Charge Coupled Device camera recording and computational analysis of flame propagation in a spark-ignition engine

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Charge Coupled Device camera recording and computational analysis of flame propagation in a spark-ignition engine.

by

Simon Robinson BSc

A Doctoral Thesis
Submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy of the Loughborough University of Technology

1996

Supervisor: Professor J C Dent PhD, CEng

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ABSTRACT

Homogeneous charge combustion in a four stroke cycle spark-ignition engine was studied using through-piston-photography with a gated-intensified CCD camera. Analysis of computer stored multiple exposed flame front images was carried out for various engine conditions, in-conjunction with the test data and cylinder pressure signals. Representative turbulence scales were inferred from the flame propagation and cylinder pressure data. Fractal analysis of flame edge contours resulted in a fractal dimension $D_3$ in the range 2.12 to 2.23 corroborating data presented elsewhere. A correlation is presented here between the standard deviation of peak cylinder pressure and the fractal dimension $D_3$. 
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Finally my family and friends for supporting my endeavours and encouraging the conclusion of my work.
PUBLICATIONS

STATEMENT OF ORIGINALITY

This is to certify that the Author is responsible for the work submitted in this thesis, that the original work presented is his own except as specified in acknowledgements or with references and that neither the thesis nor the original work contained therein has been submitted to this or any other institution for a higher degree.
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NOMENCLATURE

$A_f$ Flame front surface area
$\tau$ Flame front radius
$r_b$ Flame burned equivalent radius
$r_c$ Flame centre position
$S_b$ Turbulent burning speed
$S_L$ Laminar burning speed
$u_f$ Flame propagation velocity
$u_e$ Flame front entrainment velocity
$u_g$ Gas expansion velocity
$V_f$ Flame volume
$V_{fu}$ Volume ahead of flame front
$\rho_u$ Unburned gas density
$\rho_b$ Burned gas density
$m_b$ Burned charge mass
$m$ Total charge mass
$x_b$ Mass fraction burnt
$v_p$ Instantaneous piston velocity
$V_{cyl}$ Instantaneous cylinder volume
$N_E$ Engine speed
$B$ Cylinder bore diameter
$S$ Piston stroke
$D_v$ Maximum valve lift
$L_v$ Inlet valve diameter
$\eta_{vol}$ Volumetric efficiency
$u_i$ induced velocity at valve seat
$n$ Polytropic index
$\gamma$ Ratio of specific heats

$u'$ Turbulent Intensity
$\eta$ Kolmogorov scale
$\lambda$ Taylor microscale
$L_g$ Gibson length scale
$L$ Integral length scale
$\tau$ Characteristic time scale
$l_n$ Characteristic eddy size
$v_n$ Eddy rotation velocity
$\epsilon_n$ Dissipation of turbulent kinetic energy
$D_3, D_2$ Fractal dimension
$D_T$ Turbulent fractal dimension
$D_L$ Laminar fractal dimension
$A_T$ Turbulent flame surface area
$A_L$ Laminar flame surface area
$l$ Contorted curve length
$N$ Number of segments
$\epsilon$ Segment length
1.0 INTRODUCTION

The following chapter discusses the background to current engine development with reference to this research, then presents an overview of the research objectives. The structure of this thesis is outlined at the end of the chapter.

1.1 Spark-Ignition Engine Development

Throughout the last decade, concern over the exhaust emissions from the Internal Combustion (IC) engine has become a major priority for the Automotive Industry. This is to meet with the public demands for an improved environment and the more efficient use of fossil fuels.

Dale and Oppenheim [1] and Borman [2] have reviewed recent and past developments toward improving the prime mover in automotive engineering. This includes minimising pollutants, maximising efficiency and utilising a range of fuels. They emphasise the importance of a coordinated programme of research to meet the expectations for improvements in emissions and efficiency.

Improving exhaust emissions essentially involves the lowering of carbon monoxide (CO), oxides of nitrogen (NOx), hydrocarbons (HC), particulates and carbon dioxide (CO2). Carbon dioxide, the so called greenhouse gas, is an inevitable product of complete combustion. This can only be reduced by improved fuel economy and hence greater engine efficiency.

Carbon monoxide is both poisonous and damaging to the environment and a product of incomplete combustion, improving burning efficiency reduces CO. Nitrogen oxides are linked to breathing disorders and can contribute to the formation of photo-chemical smog. These can be reduced through reduced combustion temperatures or by exhaust gas after-treatment. Hydrocarbons are both unpleasant and smog forming, and usually
arise from cycle misfire or flame quenching leaving unburnt fuel to be exhausted. These can also be reduced through improved combustion efficiency. Exhaust gas concentrations of NO$_x$, CO and HC from a gasoline engine can be reduced with a 3-way catalyst, this is through the oxidation of the carbon based compounds to CO$_2$ and reduction of the nitrogen oxides to nitrogen. The cost of the catalyst, the need to use lead free petrol and stoichiometric mixture operation represent the limitations in this after treatment system. Burning air-fuel mixtures at very close to the stoichiometric mixture results in poorer fuel economy and higher CO$_2$ levels than for Diesel or lean burn spark-ignition engines.

Emissions and economy can be improved by making SI engines run at leaner mixtures. Leaner mixtures result in lower heat release and lower burning temperatures and so a reduction of NO$_x$ emission. However a resultant impairment in CO and HC emissions can occur if the air-fuel mixture is taken beyond the lean limit of the engines combustion system. (Figure 1.1)

![Figure 1.1 - Demonstrates effect of Air-Fuel ratio on emissions](image)

2
Lean mixtures have slower burning speeds than stoichiometric mixtures, and are prone to a higher variation of flame propagation speeds. This is reflected in a variable rate of pressure rise, and delay in the start of combustion. As a consequence the cyclic variability of Indicated Mean Effective Pressure (IMEP) and hence variation in torque output of an engine is therefore greater with lean mixtures.

To reduce cyclic variability, burning of the fuel should be more predictable and faster. Predictability means that the ignition timing control will consistently produce maximum torque at Minimum advance for Best Torque (MBT) timing. The timing is not compromised to reduce the engine's sensitivity to knocking. A faster burn means fuel can be burned closer to top dead centre hence the heat release is close to the point of maximum temperature of the cycle. There are a number of important parameters in controlling the effectiveness and stability of cylinder pressure development. These include combustion chamber shape, air motion, air-fuel mixing, exhaust residual scavenging and spark-ignition source.

To achieve the goals of accurate and repeatable control of combustion and hence improve performance, economy and emissions, advanced engine research tools need to be applied. Amann\textsuperscript{3} and Dyer\textsuperscript{4} discuss developments of new techniques for assessment of fundamental controlling elements in combustion. Through techniques of combustion visualisation, the assessment of the physical factors affecting lean flame development consistency can be considered. By understanding the influence of the various parameters on the lean flame development it should be possible to optimise the following: turbulence intensity, bulk motion, residuals concentration, combustion chamber shape and intensity of the spark source. An improved compromise between pollutants and efficiency being achieved.
1.2 The objectives of this research

The objective of this work is to understand the effect of a number of engine parameters on the combustion processes in a spark-ignited engine through flame visualisation. The major parameters being the effect of increasing turbulence on flame structure particularly with lean mixtures.

The research engine study uses an optically accessed four-valve, twin overhead-camshaft, pent-roof combustion chamber, common to modern production gasoline engines. This allows proven trends from this study to be related more easily to practical production engines. The research programme is a parametric study in which the influence of engine speed, air-fuel ratio and engine induction system on flame structure is assessed, within the constraints imposed by the through-piston photographic arrangement of the engine design.

1.3 The thesis structure

The thesis is in eight chapters as follows:

1. Introduction
2. Literature Survey
3. Experimental Procedure
4. Flame Picture Analysis
5. In-cylinder Charge Turbulence
6. Fractal Analysis
7. Results and Discussion
8. Conclusions and Recommendations
• **Introduction**, Covers the background of the thesis concluding with it's objectives.

• **Literature Survey**, Detailed appraisal of the published literature related to the study. It is divided into spark ignition combustion, photographic techniques and fractal analysis.

• **Experimental Procedure**, Covers a description of the equipment, analysis of the method of obtaining experimental data and the test programme.

• **Flame Picture Analysis**, Discusses the software for extracting quantitative data from the flame images.

• **In-Cylinder Charge Turbulence**, Discusses the evaluation of in-cylinder charge turbulence parameters, from the flame images and cylinder pressure data.

• **Fractal Analysis**, Discusses the fractal analysis of the combustion images to quantify flame front structure.

• **Results and Discussion**, this section presents and discusses the key results from this research programme.

• **Conclusions and Recommendations**.
2.0 LITERATURE SURVEY

The following chapter is a review of the literature supporting the methods adopted for this programme of research. Three sections present different aspects of the research. These are, the investigation into Spark-Ignition (SI) engine combustion, combustion photography and the fractal geometry of turbulent flame fronts. In conclusion the proposed methodology adopted for this research is outlined.

In order to achieve improved emissions and drivability from lean burn engines a more complete understanding of the characteristics of flame propagation in a lean environment must be gained. The following literature provides background on SI combustion, in particular cyclic variability of flame propagation and turbulent flame burning models. The visual observation and recording techniques are discussed to assess the best approach to achieve minimal engine modifications with maximum combustion information. Fractal analysis is a new area under investigation relating to turbulent flame propagation and the relevant literature is discussed.

2.1 Spark-Ignition Engine Combustion

In developing low emission, drivable engines, one option as outlined in the Introduction is to use lean burn technology. This requires the combustion system to be optimised to make use of lean air-fuel ratio mixtures in the engine across the speed and load range. Drivability and emissions will be worse with lean mixtures in conventional engines due to slower burning rates. This is because the chemical burning process is slow due to the lower available fuel energy. Drivability is improved by reducing cyclic variability, which can be achieved through increased burn rate. In-cylinder temperatures are higher during the combustion process, due to the reduced duration for the combustion event and the consequent higher rate of heat release. In addition cycle temperatures are higher due to piston position during the main part of the flame propagation. This is apparent in both magnitude and phasing of peak cylinder pressure relative to engine TDC. The
reduction of burn rate due to mixture strength increases the influence of other parameters, such as mixture velocity, air-fuel mixing, fuel atomisation, exhaust gas residuals and the ignition system. Variation in combustion duration due to any of the more dominant effects listed above can be reduced by increasing the burn rate. Increasing turbulence levels will increase the flame wrinkling and hence burning surface area leading to a faster reaction rate.

By reducing cyclic variation, smoother, more economical and cleaner engines can be produced. Reduction in burn variation allows ignition timing to start closer to the knock limit. This means engines can have lower octane requirements or higher compression ratios. Increasing in-cylinder charge turbulence by altering the gas flow into the cylinder improves entraining of the fresh air-fuel mixture into the traversing flame front with the effect of increasing burning surface area and overall burn rate. The inlet port and combustion chamber design have a significant effect on gas motion. Improving volumetric efficiency of the engine flow through the inlet ports improves engine fuel economy, the characteristic swirl or squish motions have been particularly important in improving combustion efficiency.

Swirl or tumble motion is rotational bulk flow of fluid within the cylinder, often conserved from the inlet port. Swirl being rotational in the horizontal plane about the centre of cylinder bore, whilst tumble is rotation in the vertical plane of the inlet port. Squish is inwards flow generated near piston Top Dead Centre (TDC). Fluid is forced inwards from the small clearance volume between the combustion chamber and piston, (Figure 2.1).

The following section discusses the principle factors influencing turbulent flame propagation, that have been reported in the literature. The relative importance of air-fuel mixture strength, fuel atomisation, turbulence and exhaust residuals are considered. The flame propagation mechanism that leads to effective burning is discussed. This introduces the concept that burning consists of a random initiation of a laminar type
flame kernel, which wrinkles under the effect of local turbulence levels. The ensuing combustion entrains fresh mixture into the flame front enhancing the burning process. The scale of the localised turbulence levels influences the burning rate.

Figure 2.1 - In-cylinder air motion.

Matsushita et al [5] in their development of a lean burn engine have utilised a control valve to adjust swirl levels with load. A lean mixture sensor provides feedback control for a multipoint fuel injection system, this results in a stable and consistent engine performance. The engine has improved drivability, economy and emissions when compared to traditional engines operating with lean mixtures.

A major contribution to the understanding of spark-ignition engine performance was made by Patterson[6]. Patterson investigated engine behaviour to understand the primary factors in controlling combustion variation. This was by a thorough analysis of
in-cylinder pressure measurements. Patterson quantified cyclic variation statistically in terms of percentage variation in cylinder pressure. The work investigated mixture strength, mixture turbulence, fuel atomisation, ignition system and exhaust residuals. Using high speed data acquisition of cylinder pressure from a V8 multi-cylinder engine and a single cylinder version of the engine (two valves per cylinder). Both cycle to cycle and cylinder to cylinder variations were considered.

Patterson states that swirl and turbulence levels are primary influences on burn rate. By introducing a shrouded mask onto an intake valve swirl levels were increased. Increases in mixture entrainment through swirl or turbulence increase the combustion reaction zone increasing the speed of combustion, (Figure 2.2).

![Figure 2.2 - Patterson, effects of shrouded valve on cyclic variability.](image)

A given engine geometry will produce variations in mixture velocity around the spark plug at ignition due to fluctuation in the mean flow and turbulence. These variations occur from cycle to cycle, and in a given cycle are non-uniform across the combustion
space, hence producing variability in the combustion process and cylinder pressure
development. Patterson's use of shrouded valves increases swirl and doubles
combustion rate and results in half the combustion variation.

The improvement of fuel-air mixing and distribution has a significant effect on cyclic
variation. However, fuel-air mixing is of more importance at leaner mixtures, this was
based on a comparison of liquid indolene and propane fuels. The indolene-air mixture
generated greater irregularity than did the propane-air mixture. Therefore the improved
homogeneity of gaseous supply of propane and air generates a faster burn than the less
homogeneous vaporised indolene and air. The effect of homogeneity is more apparent
when the air-fuel mixture is lean, (Figure 2.3). However, well optimised turbulence
through the use of shrouded valves resulted in a greater improvement than achieved by
optimising fuel-air homogeneity in this study.

Figure 2.3 - Patterson [6], effects of fuel mixing
on cyclic variability.

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Patterson tested the effects of fuel-air distribution by utilising an air driven atomiser to produce 1 micron fuel droplets ensuring fuel-air homogeneity throughout induction. The fuel-air distribution between cylinder to cylinder was improved significantly, however cylinder to cylinder variations in pressure rise were still in evidence. Measurements of air-fuel ratio showed a very close balance across the cylinders as compared to the standard carburettor, the atomising fuel supply was providing a leaner mixture of 18.5:1 air-fuel ratio as opposed to 16.6:1 to achieve better economy and yet a reduced pressure rate variation was observed at 1200 rpm and about the same at 1600 rpm. A significant benefit was obtained from optimising fuel-air mixing and had effectively reduced the sensitivity of cyclic variability on engine speed. Patterson concluded that a strong dependence exists between fuel distribution and cylinder to cylinder pressure variation.

According to Patterson, high energy discharge ignition systems were shown to be of little benefit in terms of reducing cylinder pressure variation, assuming the standard spark system has been optimised. However deterioration in spark ignition control systems leading to poor repeatability of spark timing was deemed to be responsible for 10-15% of cyclic variability.

Patterson had shown that a small variation in exhaust residuals had very little effect in improving the cyclic variation. The residual contribution was reduced by disabling the spark for a number of cycles to scavenge the cylinders adequately, whilst residuals were increased by increasing exhaust back pressure. From the results presented, however it suggests possibly the spread in the frequency histogram of pressure rate remains similar despite increases in average combustion rate, for a reduction in residuals. It could be presumed that on a percentage basis cyclic variability has reduced. In addition it is difficult to judge the thermal effects of allowing the engine to cool over the non firing cycles, on engine irregularity. Dilution of the intake air represents a better method of evaluating residual effects.
The conclusions that can be drawn from Patterson is that assuming the fuel-air supply is homogeneous and lean, spark and residual scavenging are optimised, then the predominant factors involved in combustion rate and cyclic variability is that of gas motion, arising from the induction process and piston-combustion chamber interaction. By optimising turbulence levels and bulk gas motion, combustion rates can be increased, and hence cyclic irregularities are reduced. More recent research has made use of research engines that are designed to be very similar to production engines, whilst providing optical access to the combustion chamber. This allows a more detailed assessment of the fundamental effects of the controlling parameters on combustion variation. The work discussed below presents studies of early flame propagation and flame cyclic variability, which improves the understanding of flame initiation and combustion variability. The concepts are then discussed in relationship to the current thick flame entrainment models associated with spark-ignition combustion. Additional background information is discussed in Appendix I, this contains a brief introduction to turbulence, particularly the terminology.

Hamamoto et al [7] studied cycle to cycle fluctuation of combustion of lean mixtures using cylinder pressure analysis and Schlieren photography, in an optically accessed side valve single cylinder engine. The very early flame kernel development recorded in the Schlieren photographs, displayed a greater fluctuation in the kernel location than in the kernel size. The authors concluded that whilst the early flame kernel fluctuation cannot be significantly improved through changes in mixture strength and air motion, the cyclic variability of the overall combustion duration can be improved by reducing the combustion period and reducing the influence of the early stage of combustion. That is, rapid turbulent burning is less affected by this initial large variation in flame position, than slower less turbulent burning.

Nakanishi et al [8] studied the effect of residual gases in the cylinder on cyclic variability of the combustion process in a side valve single cylinder engine. They considered the effects of combustion chamber shape, turbulence, spark ignition source and air-fuel
They found increasing the squish in the chamber reduced cyclic variability. However, unlike Patterson they found high energy ignition systems and longer (20ms) duration multi-spark ignition caused reductions in cyclic variability. In addition, increased dilution of the mixture with air or exhaust residuals was found to increase burn rate variability. That is, leaning the mixture or using Exhaust Gas Recirculation (EGR) reduces burn temperature and rate, with an adverse effect on burn rate stability.

The work of Hill [9], considers research into cyclic variation in SI engines, which showed a strong correlation between leaner air-fuel ratio and larger standard deviation in burning times as would be expected. Hill maintains that spark timing is very repeatable, but variability in combustion initiation occurs and controls the subsequent combustion event. There is a greater effect on overall variability due to factors early in the flame development. The later stages of flame development tend to be more repeatable. Hill reports on work which correlates peak cylinder pressure to flow velocity magnitudes near the spark plug and concludes that the small scale turbulent structure effects the variability of the early flame kernel and hence the remainder of the flame development. This is based upon estimates of the mean random ignition time delay, $\lambda/(4S_L)$, correlating to $\sigma_0$, where $\lambda$ is Taylor Microscale, $S_L$ is the laminar burning velocity and $\sigma_0$ is standard deviation in burning time. Experimentally derived burning times correlate well with the estimated ignition time delay. This relies on the assumption that the turbulence is both homogeneous and isotropic.

To achieve accurate and useful analysis of flame picture data the quantification of flame data must be carried out. The early flame development analysis by Keck et al[10] and the work of Beretta et al[11] used photography and cylinder pressure analysis to investigate flame development. Keck used the MIT square piston engine, equipped with two valves and tested low and high swirl and low and high squish. Swirl was controlled using a shrouded and unshrouded inlet valve, whilst squish was varied with a stepped and flat top piston. The square piston engine allowed the authors to undertake Schlieren photographic studies through the cylinder wall. The engine has a low compression ratio.
and less than ideal combustion chamber. Beretta[11] used a transparent piston engine, based on a single cylinder of a Ford V-8 engine, the compression ratio was 7.86, higher than the 5.75 of the square piston engine. The engine has two valves and spark source at one end of the combustion chamber. The photographs were taken through the piston into the combustion chamber. The flame photographs[10][11] of the researchers were digitised into a computer and subsequently processed.

The analysis of Keck et al[10] assumed that the flame front approximates to a spherical expanding cloud, the spark source being centred away from the combustion chamber walls. However the wrinkled nature of the very early kernel does lead to inaccuracy in using a spherical approximation. In addition errors occur due to the depth of the flame across the combustion chamber space. The measured flame shadow area allows the radius, $r_f$, to be determined from:

\[ r_f = \sqrt{\frac{A_f}{\pi}} \quad (2.1) \]

\[ V_b = \frac{4}{3} \pi r_f^3 \quad (2.2) \]

The flame volume can then be estimated from 2.2 assuming only a limited amount of surface wrinkling has occurred. From the volume estimate, $V_b$ the proportion of mass fraction burnt, $x_b$ was estimated using the unburned and burned gas density, $\rho_u$, $\rho_b$, respectively.

\[ x_b = \frac{m_b}{m} = \frac{(\rho_u/\rho_b)V_b/V}{1-(1-p/\rho_u)V/V} \quad (2.3) \]

It was estimated by Keck that $x_b$ was within ±30% of the correct mass fraction. This however is an estimate of burned gas from the flame picture. The mass fraction burnt
\( x_b \) can be estimated from cylinder pressure using equation 2.4.

\[
x_b = \frac{(p/p_m)^{\gamma n} - 1}{(\rho/\rho_s - 1)} \tag{2.4}
\]

where \( p_m \) is the isentropic pressure curve and the polytropic exponent, \( n=1.35 \). The equation 2.5 and 2.6 below show how the burned gas radius \( r_b \) is obtained.

\[
V_b = \frac{m_s}{\rho_s} = \frac{x_m}{\rho_s} \tag{2.5}
\]

\[
V(r_s - r_c, \theta) = V_s \tag{2.6}
\]

where \( r_s \) is the radius of the spherical surface centred at \( r_c \), with the piston position at \( \theta \) crank degrees. The shadow radius \( r_l \) when compared with the estimated burn radius \( r_b \) shows a considerable entrainment of unburned mixture after the flame exceeds 10 mm. Prior to this the estimated radii are similar. However it is clear that the photographs detect the passage of the flame more accurately than can be estimated from pressure records. The burned gas behind the flame front is of considerably lower density than the unburned mixture therefore small burned mass detected through pressure deviations can equate to a much larger flame volume. The results of flame position and standard deviation are presented in Table 2.1 and summarised in schematic form Figure 2.4.

<table>
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<tr>
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<th>( \bar{x}_c \pm \delta x_c )</th>
<th>( \bar{y}_c \pm \delta y_c )</th>
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<td>Base</td>
<td>0.8 \pm 2.1</td>
<td>-2.5 \pm 2.1</td>
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<tr>
<td>Shroud</td>
<td>2.8 \pm 2.0</td>
<td>0.5 \pm 2.4</td>
</tr>
<tr>
<td>Squish 1</td>
<td>2.8 \pm 1.7</td>
<td>0.2 \pm 4.9</td>
</tr>
<tr>
<td>Squish 2</td>
<td>4.9 \pm 1.7</td>
<td>7.8 \pm 1.8</td>
</tr>
</tbody>
</table>

Table 2.1 - Mean Flame centre displacement in mm from spark plug (note: positive values are in direction away from walls)
Figure 2.4 - The effects of swirl and squish on flame centre.

Keck [10] corroborates the work discussed above that the early flame centres move, due to random walk in turbulent flow and convection in the mean flow. This is particularly evident in the squish 2 case presented in the Table 2.1 and Figure 2.4. The squish forces the kernel downwards and inwards away from the spark location. It is probable that the swirl case has a strong deflection in the direction perpendicular to the viewing direction. Whilst all the cases presented indicate movement away from the cylinder wall, (in the case squish 2 the nearest wall is that of the piston step). The variation in displacement of the flame from the spark plug results in significant variation in spherical burning area when wall contact occurs and hence disturbing the overall period of combustion. The flame velocity accelerates from the laminar flame speed to a fast turbulent burning.

Beretta et al [11] used high speed film to record the emitted light from the propagating flame front. Only 10-13 engine cycles were fired and recorded, the first of which had
no exhaust residuals from the previous motored cycle. The first recorded in-cylinder pressure signal represents engine motoring data, which is used for reference and calibration purposes. The optical configuration is nearer to the practical engine than the later work of Keck[10], but still, intermittent engine firing will result in some inconsistency due to poor wall temperature stability. The authors reported that subsequent to the first cycle the combustion was typical in terms of exhaust residuals. Testing was conducted between 833 to 1233 rev/min, representing slower than typical engine operating speed. The analysis of the flame pictures used by Beretta et al[11] was different to that of Keck[10] due to the through piston photography arrangement. The flame was assumed to consist of a spherical surface bounded by the combustion chamber surface and the quartz piston, Figure 2.5. The centre is assumed to originate at the same height as the spark.

\[ \text{Spark plug} \quad \text{Combustion chamber} \quad \text{Flame front} \]
\[ \text{Cylinder} \quad \text{Quartz Piston} \]

Figure 2.5 - Beretta[11] et al combustion chamber and flame front.

The outer profile of the cloud viewed through the piston was traced and digitised as the flame moved across the chamber. To each flame front profile a best fit circle was determined using the method of least squares, Figure 2.6.
The analysis of the cylinder pressure data was used to derive mass fraction of fuel-air mixture burned and hence determine the burned radius, $r_b$, as with the Keck et al\cite{10}. In comparison with the flame radius, $r_f$, the burned radius, $r_b$, is similar, however the difference $r_f - r_b$ grows with $r_f$ from 2mm to up to 8mm, Figure 2.7.

The data shows an increasing difference between burned gas radii estimate and the flame radii for $r_f$ larger than 10mm. The difference between radii tends to become constant above 30mm at approximately 6mm. This further supports the model of entrainment of fresh mixture behind the flame front before it is completely burnt. During the combustion process the flame centre displacement changes during the initial burning stage of the flame. The displacement increases until the flame radius approaches 30mm, Figure 2.8. Beretta et al\cite{11} suggest most of the displacement occurs when the flame is below 10mm radius. They determine a characteristic flame radius, $l_c$, required for stabilisation based on the initial slope of the graph and the stable flame displacement $\delta_c$. This is not considered in the later work of Keck\cite{10} where the flame centre displacement are much smaller. This term is considered, by Beretta\cite{10}, to be important in controlling
the cyclic variability of the process. However this term was not correlated against test conditions with higher cyclic variability to validate this statement.

Figure 2.7 - Beretta\textsuperscript{[11]} et al Flame radius and burned gas radius.

Figure 2.8 - Beretta\textsuperscript{[11]} et al Flame centre displacement against flame radius.
The characteristic speeds, the flame speed $u_f$ obtained from the rate of change of flame radii, $r_f$ and the burning speed $S_b$ from the rate of change of burned radii, $r_b$ are calculated and plotted. The difference between $u_f$ and $S_b$ is due to the entrainment of fresh unburnt fuel-air mixture and is the gas expansion speed ahead of the flame front $u_e$. It was shown that the flame speed accelerates from the laminar flame speed, $S_L$ up to 10 times that speed, Figure 2.9.

![Figure 2.9 - Beretta\cite{11}, Flame speed and turbulent burning speed.](image)

The turbulent burn-up model discussed by Beretta\cite{11} has been based on the earlier work by Blizard and Keck\cite{12}. The model of the flame burning process assumes that turbulent fuel-air eddies are entrained into the burning flame front, which are subsequently burned at the laminar flame speed. The eddies persist from induction into the combustion phase and are assumed to be spatially homogeneous. The flame propagates through the mixture at a speed determined by the rate at which eddies are entrained $u_e$. These eddies are ignited once behind the flame front to burn at the laminar flame speed $S_L$. They can then burn in the characteristic time $\tau = l_e/S_L$ where $l_e$ is the characteristic eddy radius. The engine speed, geometry, size and spark advance influence $l_e$, whilst stoichiometry, fuel type, residual, inlet air density and spark advance determine $S_L$. 

20
This model was further developed by Tabaczynski[13][14]. This combines the turbulence flow structure and the burning of entrained unburned eddies in the mixture behind the flame front. The individual eddy is no longer considered a single entity that burns at laminar flame speed. The eddy is of integral size $L$, Figure 2.3, the internal small scale structure consists of vortex tubes of diameter $\eta$, the Kolmogorov micro-scale which are spaced at intervals of $\lambda$, the Taylor micro-scale. The molecular processes control the chemical reactions and can be considered instantaneous at the Kolmogorov scale, whilst at the microscale level, burning occurs at laminar speed, (Appendix 1).

![Figure 2.10 - Schematic of turbulent eddy [13].](image)

Daneshyar and Hill[15] review literature on the processes involved in mixture preparation and combustion for SI engines. The effects of engine variables are outlined. Turbulence Intensity, $u'$ increases linearly with engine speed, whilst the Integral length scale is approximately independent of engine speed. The Taylor microscale was found to reduce with increased compression ratio. Daneshyar and Hill[15] discuss typical values of scales in the study of turbulent combustion in spark-ignition engines. The integral scale, $L$ is
typically of the order of the combustion chamber height approximately 10mm. The Taylor microscale, $\lambda$ of order 1 mm and the Kolmogorov scale 0.01 mm. Typical turbulence intensities, $u'$ are in the range of 1-6 m/s, laminar flame speed, $S_L$ approximately 0.5 m/s and laminar flame thickness 0.01 mm.

The combustion model of Blizard and Keck\cite{12} is extended by Daneshyar and Hill\cite{15} to include the effects of the reaction zone thickness. It is composed of a random ignition delay, a period for the combustion front to propagate across the cylinder and a period to burn the entrained charge behind the flame front. Results from the model compare well with experimental data presented elsewhere in the literature.

High speed Schlieren photography of opposing double kernel, (twin spark source), flame propagation in a constant volume fan stirred bomb was conducted by Groff\cite{16}. The turbulent burning velocity is evaluated by measuring the approaching flame profile against time. The two converging flame fronts cause the gas expansion velocity, $u_g$ ahead of the front, to cancel out. Effectively the flame propagation velocity $u_r$ is equivalent to the entrainment velocity, $u_e$ (refer to equation 2.7 below). The propagation velocity, $u_r$ is obtained using a single spark and recording flame propagation from a single kernel in the combustion bomb.

$$u_r = u_f - u_e \quad (2.7)$$

To improve high compression lean mixture burning Checkel and Thomas\cite{17} worked with a turbulent charge in a constant volume combustion bomb. They utilised perforated plates that were drawn through the chamber and induced turbulence within the air-fuel mixture. They concluded that small scale turbulence was better than large scale turbulence at improving combustion, however evidence suggests that the small scale turbulence decays more rapidly than larger scale. In an engine large scale motions that
have been well preserved from the intake can be further broken down to small scale turbulence just prior to ignition.

Milane and Hill [18] undertook a study in a combustion bomb, in which turbulence was created in a mixture of Propane and air prior to ignition. They use a swirling charge motion in a combustion bomb to improve their understanding of swirl enhanced combustion. The objective of the study was to obtain the entrainment velocity \(u_e\), defined by the expression above in equation 2.7. Where \(u_f\) and \(u_g\) are the flame propagation and gas expansion speed respectively. The combustion bomb was instrumented for photography and pressure measurement. Measurements of flame area were made from photographs for which a hemi-spherical combustion cloud was assumed for calculating the flame radius. The entrainment of unburned gas into the flame front is controlled by the burning velocity of the flame and the velocity of the gas ahead of the flame front, due to compression of unburned gas and the expansion of burned gas.

The propagation speed is estimated from the radius of flame clouds and spacing of images. The mean gas velocity is estimated by dividing the rate of change of unburned gas volume by the flame front area. Using the conservation of mass and isentropic compression for the unburned gas, \(u_g\) can be expressed as in equation 2.8.

\[
\frac{dP}{dt} = \frac{V_u}{A_f} \gamma P \quad (2.8)
\]

Where \(V_u\), \(A_f\) and \(P\) are unburned volume ahead of the flame front, flame surface area and cylinder pressure respectively. From 2.7 and 2.8 Milane and Hill calculate the entrainment velocity of fresh mixture into the flame front. Their estimates of entrainment velocity compare well with results presented elsewhere in the literature.
In summary the literature shows that fundamental to the improvement of spark-ignition combustion, particularly where lean mixtures are concerned, is the judicious use of controlled levels of turbulence. The combustion process in the spark-ignition engine takes place in a turbulent flow field. This flow field is dependant on piston motion, (engine speed), chamber geometry and the induction process. The fully optimised combustion system requires refinement of the combustion chamber and induction process to improve combustion. The optimised charge combustion begins with an ignited kernel away from the combustion chamber walls after a random initiation period, which will progress from a laminar burning kernel to a rapid moving wrinkled flame front engulfing fresh mixture. The mixture continues to burn behind the flame front until the fresh mixture is exhausted. The reaction zone is thick due to the small scale localised turbulence, which increases the burning flame area resulting in fast burns. The model of turbulent combustion is continually improved to further understand combustion fundamentals and a means of improving the optimum combustion system.

Cyclic variability is important in influencing spark-ignition engine drivability and emissions, particularly where lean air-fuel ratio mixtures are utilised. Assuming the mixture strength and residual contributions are optimised then the turbulent flow field will be most significant in controlling cyclic repeatability. The early kernel development within the turbulent flow field has an important role in the future stability and repeatability of combustion duration.

Keck et al[10] and Beretta et al[11] experimentally demonstrate the convection of the kernel in the bulk flow of the charge and random walk in the turbulence. This random motion of the initial laminar kernel positions the centre of the propagating flame randomly in relationship to the combustion chamber walls. This determines the contact time in the combustion process between the walls and the flame, which will influence the overall combustion duration and effectiveness. The engines used in this work are not very close to current production engines, therefore the spark plug position, combustion chamber design and valve orientation are not comparable with a production engine.
2.2 Photographic Techniques

The following section considers work based on utilising photographic records of the combustion event. Firstly it outlines the methods of access for combustion photography and their development to more advanced techniques. The compromises involved in providing useful images are discussed, this includes access to the combustion chamber, speed of combustion, the provision of adequate illumination, the nature of the combustion recording and the concessions to production engines. The duration and speed of the combustion event complicate the process of recording. Light emissions may not be adequate at the exposure times required to freeze the propagating flame.

2.2.1 Origins of current technique

The concept of visualising combustion with the aid of high speed photography has been used since it was first tried by Withrow in the 1930's and is discussed in considerable detail in Rassweiler and Withrow [9], who were the first to correlate engine cylinder pressure records from a pressure balanced diaphragm transducer with the film record of the propagating flame. The engine was a side valve engine operating at 4.6:1 compression ratio with visual access through a window in the cylinder head. The engine was operated at 900 rev/min, wide open throttle and 13:1 Air-fuel ratio. To reduce the thermal stress on the glass window in the cylinder head the engine was regularly skip fired. To provide combustion pictures the engine triggered a camera capable of up to 5000 frames per second (fps). The results provided were dealt with in some detail to establish links between pressure and inflamed volume.

Side valve engines have been replaced by more powerful overhead valve designs. Access through the cylinder head of overhead valve engines is much more difficult.
The present generation of commercial spark-ignition engines rely on overhead valves and higher compression ratios (10:1), therefore the introduction of through piston photography by Bowditch [20] in the 1960's provided a means of viewing the combustion process in this type of engine configuration. (Figure 2.11). This paper outlines the detail design and potential mechanical problems facing a user of this technique. In addition various qualitative studies are discussed that can be carried out using this technique.

2.2.2 Progress in engine combustion photography

The use of the photographic technique in engine combustion studies can be for fundamental characterisation of engine performance parameters, and as a diagnostic
technique for modifying or developing engine design. The following literature considers
the range of techniques available with their limitations and benefits. The main
experimental factors are access to the combustion chamber, the available light and the
speed and repetition of the combustion events. The main techniques discussed use either
high speed film cameras, still cameras or electronic video cameras. Access to the
combustion space and combustion process is necessary to record the light emission from
combustion. Usually quartz windows are positioned in the cylinder head, piston or liner
walls to accommodate optical access. To improve both the light available or to freeze
flame progression, flames can be seeded and a flash or pulsed lasers can be used to
illuminate the event. Mixture density distribution can be recorded using shadowgraph
or Schlieren techniques.

A high speed photographic study of the effects of EGR on combustion was carried out
by Nakanishi et al [8]. The engine studied utilised a quartz head side valve single
cylinder. To record photographically pictures of lean mixture combustion, with low
light emission, a Photosonic intermittent high speed camera capable of 800 fps was used.
To freeze the flame adequately the camera exposed each frame for only $250\mu s$, much
less than the $1250\mu s$ between frames. This provided more light on the film than did an
Hitachi high speed rotating prism camera capable of 800-4000 fps.

The high speed film camera (HSFC) has been used successfully for many years in the
study of the internal engine combustion process. The images in Figure 2.12 are taken
from Rashidi [21]. They show clearly the flame development and propagation during a
single combustion cycle in an engine, it is relatively easy to evaluate propagation rates
and flame position information. The flame profiles can be measured in relation to a
reference grid scaled to the cylinder head. The timing of each frame is known from the
camera speed and can be used with the flame profile displacement to determine flame
propagation speeds. The flame edge and flame shape can be compared qualitatively to
assess factors effecting combustion effectiveness.
The Hycam range of high speed film cameras are frequently used in engine combustion studies, Rashidi used a 2000fps rotating prism camera with 70μsec exposures. Rashidi used the Bowditch method of through-piston-photography and seeded the fuel mixture to improve flame luminosity. It was possible to obtain photographic records of an adequate quality of combustion, without additional illumination.

![Image](image)

Figure 2.12- Rashidi[21].

The HSFC has a limited film length for a given test, which restricts the number of consecutive cycles that can be recorded. Film exposure is a synchronous event resulting in continuous and contiguous frames of images taken throughout the whole cycle. In a conventional 4 stroke engine, of the 720 degrees of the cycle, perhaps 40 degrees are of interest, less than 6% of a film is utilised per cycle. So for 20 consecutive cycles at 6 degree intervals 2400 frames are required of which around 150 will carry information. A typical test may require 400 feet of film of which only 130 feet contains useful information.
There are 40 frames per foot on 16mm film which means 5400 frames are available (40 cycles). The 300 or so usable frames have to be extracted and time stamped to the engine cycle and crank angle position from the 5400 frames. Rates of at least 2000 fps with HSFC are required to produce results with relatively small steps in flame development, for example 4.5 degree steps at 1500 rev/min.

Exposures of approximately 100$\mu$S duration are necessary for freezing the flame, whilst allowing enough light to adequately expose an image of the flame onto the film. The image registration of geometry and flames can be difficult with HSFC. The cameras cannot be easily mounted on an engine, due to the precision components. If the camera is not engine mounted it will be isolated from engine vibrations, this may result in flame position variation, which is not combustion cycle to cycle variability.

Generally HSFC analysis has relied on manual recording of the flame profiles, which is then available to quantify flame propagation rates and flame position. This tends to limit the number of cycles that can be readily analysed. Film can be digitised automatically through film to video instrumentation but further reductions in image quality will be incurred. This introduces an additional delay into the analysis process, firstly chemical processing the film followed by electronic digitising, which requires time and careful control of experimental records.

To improve the amount of information resolved, additional light can be utilised. The shadowgraph or Schlieren techniques can be adopted. DeSoete \[22\] studied the early stages of combustion viewed in a bomb with the laser shadowgraph technique and an image convertor camera. This allowed the flame distortion to be closely examined.

The shadowgraph passes a collimated light beam through the combustion event casting a shadow onto a screen, Figure 2.13. The density fluctuations alter the refractive index of the mixture causing light gradients on the screen. The Schlieren technique is similar except that a cutoff aperture acts as a filter to remove some refracted light and reduces
the light from the high density gradient regions, Figure 2.14. Both techniques usually require two access windows in line to permit the light to enter and exit the combustion chamber. This usually limits the practical use of this technique on modern engines. The shadowgraph and Schlieren techniques provide pictures that are representative of mixture density distribution throughout the chamber. The shadowgraph is sensitive to the second derivative of fluid density, whilst Schlieren responds to the first derivative.

Figure 2.13 - Schematic of Shadowgraph technique.

Figure 2.14 - Schematic of Schlieren technique.
These are used primarily to provide a sharp contrast of the steep gradients in temperature of the mixture in the combustion chamber. This is representative of the density changes between the unburned and burned mixture.

Gatowski et al [23] used Schlieren photography in the Massachusetts Institute of Technology (MIT) square piston engine with the Bowditch configuration. This allowed visual access through the sides of the square cylinder without distortion. A collimated beam of light from a spark source provides the means of producing the image and a 2000 fps camera records the Schlieren images of the density fluctuation in the ignited early flame kernel. The images are clear and show the early flame development and the effects of turbulence distorting the flame front.

Witze[24], used a laser strobing shadowgraph technique to investigate the effects of swirl and the air-fuel mixture on the flame propagation. A Hycam high speed camera operating at 5000 fps and a stills camera were used to record the shadowgraph, (Figure 2.15). Here the laser produces a beam of light that is reflected off the mirrored piston surface and projected onto a Mylar screen. The strobing of the laser can be as short as 28μs effectively freezing the combustion event. Sufficient contrast is obtained on the screen due to the laser light source, which can be recorded onto conventional photographic film. HSFC or a still Polaroid camera record the image on the screen. The film speed available for the still camera allows a short 28μs exposure to be used. The film image was 48mm providing a good quality sharp image with rapid turn around due to the instant Polaroid pictures. The HSFC requires 125μs for a compromise between the light and speed of exposure. The 16mm image size meant that a longer focal distance is necessary reducing the available light. This does however allow more than one frame to be recorded during combustion of a single engine cycle. The still camera was the preferred method of recording images for the bulk of the engine testing, despite only taking a single frame for each engine cycle.
2.2.3 Laser sheets

The photographic techniques as discussed above record a 2-dimensional image of light intensity emanating from a volume of burning fuel/air mixture. This can either be with flame light emissions or through the utilisation of an additional light source as with shadowgraph or Schlieren. By using suitable seeding of the gas mixture it is possible to improve visualisation of the flame. If the light source is in the form of a laser sheet the combusting flame can be recorded as a 2-D cross-section. This allows very short
exposures to be taken when using high powered pulsed lasers (10ns). The quality of the frozen image and the thin cross section of flame resolves greater detail of flame front distortions due to turbulent flow fields. However these techniques require horizontal and vertical access to the combustion chamber resulting in major engine modifications.

Baritaud [25] used a 2-D flame front visualisation technique, based on Mie-scattering from a particle seeded flow. The lower seeding density in the hot burned gas produces a lower intensity of scattered light than the unburned region and so the flame zone can be identified. Essentially this was a qualitative study considering the nature of the flame front. The degree of flame contour wrinkling was discussed as engine speed is increased. The turbulence levels increase with engine speed as does the amount of flame wrinkling.

Zur Loye and Bracco[26] investigated flame structure with a 2-D laser sheet imaging method. They used a Reticon Charge Coupled Device (CCD) video camera to record the image, whilst Baritaud used a vidicon video camera. Image slices of flames were taken by introducing a sheet of laser light, produced with a cylindrical lens, into the seeded combustion charge, (Figure 2.16). A CCD camera focuses perpendicularly onto the plane of laser light to take an image of combustion, the event is shuttered by strobing the laser for a very short duration, (10ns) in synchronism with the cameras exposure.

The analysis of the data was performed by computer. It was possible to correct images for background lighting and produce intensity histograms to separate out the foreground from the background. The image is then reduced to a binary image representing the foreground and background, which can be adjusted if the boundary is not clearly delineated. This allows an edge position contour to be determined from the boundary between the two regions. From the edge contour the authors could determine flame area, thickness of the flame front and radius of curvature of the flame front. The resolution of the CCD camera used by Zur Loye and Bracco[26] was only 100x100 pixels which restricts the level of accuracy when determining the flame characterisation parameters.
More recently Ziegler et al \cite{27}, examined flame structure using a 2-D laser sheet in a square piston engine. The engine is optically accessed through the four cylinder walls. The authors used a vertical laser sheet to pass through the combustion chamber in the region of the spark plug. A rotating prism camera was focused onto scattered light from the combustion in the plane of the laser sheet. They used a copper vapour laser capable of operating at pulse durations as short as 20ns. The camera is capable of up to 80,000 fps. Recorded photographic images are transferred to a MicroVAX computer system for processing. Similar to the previous workers, image processing generates a flame contour, which is available for quantitative analysis. The experimental setup has allowed the authors to process 5000 frames of data, giving a sufficient data set to investigate possible correlations.

The technique of using laser sheets allows a thorough analysis of flame fronts to be performed. However, the engines require considerable modification to provide sufficient optical access, which means that a larger number of assumptions have to be made in drawing conclusions from the work, and relating these to more practical engine designs. The laser images are of high quality in comparison with conventional combustion imaging using HSFC and shadowgraphs. In addition to a good contrast and level of
detail, they provide a true two dimensional slice through the expanding flame cloud. A topographical representation of the flame can be visualised by moving the position of the sheet through the flame cloud in subsequent combustion cycles, and a quantitative picture of the flame geometry developed.

2.2.4 Video Cameras

Video cameras have been used in research carried out by workers discussed previously. However, their development in recent years has been significant. It is intended that a brief description of the technology is included followed by a discussion of how the work presented here was able to utilise video techniques. The use of video cameras have become more predominant due to the benefits they offer over alternative photographic techniques. The speed in displaying recorded images is superior to the film camera, the high light sensitivity, robustness and easy electronic control all prove to be beneficial, however the framing rate is too slow to generate multiple images during the progress of a single combustion event. The computer processing of images is most useful in extracting relevant data.

There are a number of variations in video camera type, but the two main types are vidicon tube cameras and the Charge Coupled Device (CCD) cameras. The former are the basis of the older camera technology used until recently in television broadcasting. The latter are gradually replacing vidicon cameras for television purposes, but dominate totally home use and most industrial applications. More recently image intensifiers have become available to modify the operation of video cameras. They effectively attach to the front of a tube camera or CCD and allow the light signal to be amplified onto the imaging surface. High voltage plates generate greater photon activity and higher light levels at the image sensor. An additional benefit of intensifiers allows them to be shuttered for very short intervals. This permits rapid exposures to be recorded of less than 1µs, much faster than conventional CCD shuttering techniques, which shutter over periods of about 1ms.
A simplified schematic of a frame transfer CCD is shown in Figure 2.17. A lens focuses the image onto the optically exposed surface of the CCD, which generates charge in the array of charge wells across the CCD. The quantity of charge varies with time and light intensity. An image is stored on the surface of the CCD which can then be transferred, after a period of exposure of the light sensitive portion of the CCD to the covered area. This occurs by exciting adjacent charge wells to shift charge potential across the CCD. Each vertical line of the CCD acts as a serial shift register allowing the total portion of the exposed CCD to be shifted to the covered portion.

![Figure 2.17 - Schematic of CCD frame transfer array.](image)

Cameras without frame interlacing operate at a framing rate of 25Hz, that corresponds to 15.625kHz the line scan rate of a 625 line system. With frame interlacing only 388 lines are scanned in sequence at frame rate of 50Hz to reduce visible flicker in the resultant image. A frame will be transferred from the CCD as a serial array of pixels
at 15.625 kHz or 64μs for a line. Each pixel is converted from charge potential into a voltage level that can either be digitised into a framestore or converted into an analogue CCIR signal. CCIR is the broadcast standard for black and white television signals.

The framestore permits a television frame to be stored as a digital representation of the image. Each line of the image, pixel by pixel is converted into a series of numbers. A framestore could consist of 512 x 512 pixels using 8 bits to represent each pixel. Like the CCD a framestore provides a list of storage locations to record light level for each pixel. The image being converted into 256 discrete light intensity levels for an 8 bit representation of 512 line by 512 rows. A framestore usually consists of fast Random Access Memory (RAM), similar to computer program memory. The input is designed to read a standard composite Television signal. Whilst the output can be displayed directly onto a Television display monitor or accessed digitally through a computer interface.

One of the earliest uses of video camera was in 1979 by Steinberger et al [28]. A Pulsed illuminated TV system provided real time viewing of the combustion event from consecutive cycles. This permitted either shadowgraph or Schlieren type investigations to be carried out. Combustion was photographed through a transparent piston with a pulsed laser source providing illumination through a transparent cylinder head. A Charge Injection Device (CID) camera was used with a short duration 10-100μs flash recorded onto a Video Tape Recorder (VTR). This allowed a qualitative study of cyclic variation to be conducted. The CID camera is very similar to a CCD camera with the exception that each pixel is read in a similar fashion to computer RAM memory.

The benefits of a TV camera system were further illustrated by Kozuka et al [29] who used a shuttered TV camera. The experimental engine employed Bowditch, through-piston photography, whilst the camera was a high speed shuttered vidicon TV camera system. This provided 30-800μs exposure with 33ms repeat rate. A development of this system with an image intensifier was employed by Iwashita and
Saito \[30\] to study knock. Camera intensifiers allow much lower light levels to be recorded without resorting to an additional light source. It can magnify the light intensity by about 20,000 and shutter over periods from 200ns-50\(\mu\)s. This provides a superior light sensitivity and shuttering performance than would be possible with film techniques. The authors analysed the images with a computer system determining flame area by evaluating a binary image and counting the pixels of the burnt flame. This was then related to the onset of knock recorded in the pressure signals.

Checkel and Thomas \[17\] photographed combustion in a turbulent mixture in a bomb using a range of measurement methods which included, pressure, ionisation probe, direct photography and a photomultiplier tube with a suitable filter to record emission from CH molecules in the propagating flame front. The authors use an Imacon 790 camera, which is a vidicon camera that has a complex control system to deflect the image from an intensifier to a screen phosphor. This phosphor screen can hold up to 16 multiple images that can be recorded by a Polaroid camera or CCD. This system is extremely expensive and has very low resolution, in a single framing mode it will use a standard TV frame of 276x388 pixels. To achieve 16 pictures the horizontal and vertical resolution will each be divided by 4, resulting in 69x97 pixels for each image.

The use of CCD cameras is generally restricted by exposure period and light sensitivity, Bates\[31\], very recently used a Gated-Intensified CCD. This permits rapid low light exposures of the combustion process to be recorded. No fuel additives or additional light sources are necessary and the image system can record multiple events in a single cycle. The engine investigated used overhead valves, through-piston access and an optically transparent cylinder liner made from sapphire.

Bates arranged two synchronised cameras to record flame propagation from the side through the sapphire cylinder liner and from below through the quartz piston. A combustion image recorded through the piston is shown in Figure 2.18, whilst that recorded through the sapphire liner is in Figure 2.19.
Figure 2.18 - Bottom view of flame, Bates [31].

Figure 2.19 - Side view of flame, Bates [31].
The gated-intensifier improves light sensitivity with magnification from 50-20,000 and exposures as short as several microseconds. Repeat shuttering can be achieved to generate multiple exposures of the combustion event onto the CCD. This can be spaced such that two separate flame clouds can be identified.

The equipment allowed a detailed quantitative analysis of engine results from an overhead 3 valve engine operating at up to 4000 rev/min at 9:1 compression ratio. The sapphire liner and quartz piston effect heat transfer from the combustion chamber and could result in some effects on the combustion system as compared to the production engine.

High speed video has also become possible more recently, but exposure periods are not very short. Strickland [32] used a Kodak high speed video of combustion in a constant volume bomb to produce Laser shadowgraphs which were processed to generate a 2-D optical map of flame propagation. The camera can record up to 2000fps at 238 x 192 lines of an 8 bit image.

Computer processing of the combustion image produces a black and white map of the detail of the shadowgraph. This is then thinned down until each of the black regions consist of a line one pixel wide. The image size is thus reduced and the image is divided into small blocks. These blocks are correlated between successive frames to produce a 2-D array of displacement vectors.

Photographic, video and laser based techniques each have merits depending on their application. It is considered here that the major benefits of video techniques are as follows. The CCD camera lends itself to computer storage and analysis, thus speeding up and increasing analysis capacity. It provides extremely high speed shuttering and at very low light levels. CCD cameras are more compact and robust than any alternative photographic devices, however they are more intrusive than optical sensors.
2.2.5 Optical sensors

Optical sensors are linked closely with photographic studies as a means of both combustion analysis or engine control systems. They can provide the ability of visual analysis without significant modification to an engine. The relationship between information detected by sensors and that by cameras means that practical applications for diagnostics and control can be derived from optical combustion analysis, where suitable sensors can replace the camera.

Spicher and Kollmeier [33] utilise a modified cylinder head fitted with a grid of optical fibres connected to photomultiplier tubes to observe knocking in a Rotax single cylinder engine. This approach directly senses a quantitative light signal that can be monitored with transient recorders. Analysis of the signals allows comprehensive investigation of knock characteristics.

The simplest use of optical probes has been in a spark plug, Witze et al [34], have positioned eight optical fibres around the edge of a conventional spark plug. These act as flame arrival sensors by triggering voltage comparators on the outputs of photomultipliers connected to each fibre. This very early detection gives eight displacement velocity vectors from the spark plug. Bulk convective flow can be inferred from the velocity vectors determined from the time taken from the spark to the arrival at the sensor. Two subsequent papers [35][36] have been written about the potential for incylinder flame propagation investigations in standard production engines. However the devices are sensitive to each application, whilst they are simple to fit and use, the setting up of the triggering levels is complicated and the devices are prone to erroneous triggering from noise and reflections.

Measurement of infrared emissions with an optical sensor, (Remboski et al [37]), has been employed as a potential sensor for engine management systems. A photodiode measures light level that correlates with locations and magnitudes of peak pressure.
2.2.6 Summary of photographic techniques

A number of techniques have been presented as a means to study turbulent combustion in spark-ignition engines. This includes the engine optical access, the means of illuminating the combustion event and the method of recording the event. Generally high speed film cameras have been used with either the Schlieren technique or using seeded fuel-air mixtures to increase combustion light emission. The most common means adopted for visual access is either through the cylinder head or through the piston with appropriate transparent windows. The latter is more cumbersome, however, an engine nearer to that of a production engine can be studied.

More recent techniques utilising laser based illumination and video cameras offer the potential for automated recording and processing of combustion images. In addition these approaches improve the clarity of observations. However specific techniques are not universal and apply well to certain investigations. Video cameras offer convenience, light sensitivity, timed synchronisation and indefinite test durations as compared to high speed film cameras. The film cameras do allow the total combustion event of a single cycle to be recorded, which alternatively is only possible using an Imacon 790 video camera, however this has a low resolution and is expensive. The laser and Schlieren techniques provide flame structure detail not available by line-of-site combustion studies, however they do require additional complexity and cost when setting up the experimental system. In addition the increased optical access can require engines further removed from production engines.

2.3 Fractal dimension of flames

The investigations with flame contours discussed above have been concerned with correlations between spherical flame size and the irregularity of flame kernel position. This relies on the assumption that the flame front is spherical. In addition flame wrinkling has only been considered in a small minority of investigations. Recent work
has been carried out into Fractal geometry in relation to turbulent boundaries. This allows quantitative analysis of flame wrinkling to be determined. The following section considers firstly a brief introduction to fractal geometry, followed by a review of the literature considering fractals applied to turbulent flame propagation in SI engines. The concepts described allow a more detailed characterisation of flame shape, which can then be correlated with other combustion parameters.

The fractal dimension is introduced as a numerical description of the order of irregularity in a surface or edge to a shape. This is discussed in relation to turbulent flame propagation. Finally the utilisation of fractal dimension in flame propagation modelling is discussed.

The interest in fractals stems largely from Benoît Mandelbrot's own fascination with this subject area. Mandelbrot gives a detailed account in his book on the subject [38] and in early papers details the relation to turbulence [39][40]. Euclidian geometry has dimensions of 1, 2 and 3 for a line, square or cube, fractal dimensions are non-integer and can be considered as an object having a degree of convolutedness taking it beyond the normal Euclidian dimension. An area of a rough surface has a dimension greater than 2.

Fractals in relation to premixed turbulent flames was investigated by Gouldin [41]. The classical illustrative example of fractals is introduced: How long is the coast of Great Britain?, the answer depends on the scale of measurement used. If a 1 metre rule was used it would appear longer than if a 1 mile rule was used. This is illustrated in Figure 2.20 demonstrating that highly distorted curves can only be measured absolutely if the measurement scale is equal to the smallest irregularity. The measurement length varies with the measurement scale. The Euclidean geometry would measure equally the length of the contour assuming the measurement scale is suitably small. For very rough contours, this is not the case. The fractal dimension is a parameter used to characterise the relation between the scale of measurement and the length of the contour. This is an evaluated parameter for roughness of the contour that can be used to describe it.

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Sreenivasan and Meneveau [42] discuss the application of fractals to turbulent flow. Turbulent flows are considered to be flows consisting of hierarchical scales of eddies, (Appendix 1). As large eddies become unstable they break down into smaller and smaller eddies until stability is achieved at a molecular level. They conclude that turbulence cannot be proven to be fractal in nature, but turbulent behaviour can be considered fractal over a short timescale. Further work is required to establish if fractals have a contribution to the understanding of turbulence.

To make use of fractals in investigating turbulent boundaries the fractal dimension can be evaluated. This is representative of the irregularity of the boundary shape. The length of an edge can be measured by using a polygon of N elements each of $\varepsilon$ length, the total length $l$ is then $N\varepsilon$. As $\varepsilon$ is varied the length $l$ varies according to the relationship in equation 2.9.

\[
l \propto \varepsilon^{1-D_f} \quad (2.9)
\]
The fractal dimension is represented as $D_2$ and can be evaluated by taking logarithms. Various approaches in the literature have been adopted to calculate the fractal dimension. A commonly used method measures a contour length by determining the number of image pixels within a radius $\varepsilon$ from the contour. This is repeated for a range of $\varepsilon$ values and plotted. Using equation 2.10 the fractal dimension $D_2$ can be determined from the slope of the log-log plot.

$$\log(N(\varepsilon)) = (2 - D_2)\log(\varepsilon) \quad (2.10)$$

From the evaluation of fractal dimension it becomes clear that the relationship only holds for a range of $\varepsilon$ values and that 'cutoff' values occur at both extremes of the measurement scales. As $\varepsilon$ is reduced the length $N(\varepsilon)$ may become Euclidean, in approaching the inner and outer cutoff the slope ceases to be linear and unpredictable, (Figure 2.21). Therefore the fractal dimension is representative of the fractal character of the contour bounded between the scale defined by the inner and outer cutoff.

![Graph](image.png)

Figure 2.21 - Inner and Outer 'cutoff' for fractal dimension.
The specific application to flames assumes that surface wrinkling of flamelets in a turbulent propagating flame is fractal. Mantzaras et al. [43] utilised the topographic laser pictures of flame fronts from the earlier work of Zur Loye and Bracco [26] to derive the fractal dimension. Applying the technique described in Sreenivasan and Menevau [42] to the flame fronts, representing the boundary between dark and light pixels, dimensions are determined for various experimental operating conditions. It was estimated that the fractal dimensions are of the order of 2.36. It was not possible to determine the inner cutoff because this was beyond the resolution of the camera used. They derive a model for turbulent flame speed illustrated in equation 2.11, and conclude that the flames are fractal in nature. The fractal dimension measurements are in good agreement with other workers.

\[ \frac{S_f}{S_L} = (u' / S_f)^{3D_f-1} \] (2.11)

Gouldin et al. [44] experimentally evaluated fractal dimension for premixed turbulent flames. As an alternative to the method proposed in Gouldin [41] and Mantzaras et al. [43] they utilise a more direct interpretation of path length \( l \) against \( \varepsilon \) to derive \( D \). By tracking around the perimeter with various values of \( \varepsilon \), this enabled \( D_2 \) to be derived from \( \log(l) = (1-D_2) \log(\varepsilon) \). They produced preliminary results for fractal dimensions of 2.1 well below the expected value of 2.37. An outer cutoff scale several times larger than the integral scale of the flow was determined, whilst the inner cutoff was orders of magnitudes larger than the Kolmogorov scale. This is possibly due to the low absolute turbulence intensity that is evident in their experiments. The authors expect further work is required for any conclusions to be drawn, particularly experiments using a larger variation in turbulence levels and mixture parameters.
Peters\cite{Peters} reviews investigations into laminar flamelets in turbulent combustion. The concept of a turbulence length scale, the Gibson scale, $L_g$ is introduced. This is a larger scale than the smallest possible scale in turbulent flows that is the Kolmogorov scale. (Appendix 1). The Gibson scale is defined as, $L_g=S_L^3/\varepsilon_n$, where $S_L$ is the laminar flame velocity and $\varepsilon_n$ the dissipation of turbulent kinetic energy. An eddy size $I_n$ rotates at velocity $v_n$ according to the relationship $I_n=v_n^3/\varepsilon_n$, as $S_L=v_n$ then $L_g=S_L^3/\varepsilon_n$. Figure 2.22. The eddy interacts with the oncoming flame front, large eddies cause substantial convolution, small eddies with low velocities will not wrinkle the flame front. Eddies at the Gibson scale are the smallest possible, whilst still wrinkling the flame front and therefore should be equivalent to the lower cutoff value for evaluating fractal dimension.

Figure 2.22 - Eddy entrainment into flame front.
Turbulent flame growth using a fractal model has been recently developed by Santivicca[46] who describes the growth process as the competition between turbulent convection and laminar burning. Turbulent convection acts to wrinkle the flame front, whilst laminar burning acts to smooth the flame surface. The turbulence length scales larger than the laminar flame thickness (≈0.01 mm) convectively distort the flame front, whilst those smaller than the flame front thickness effectively increase the transport rate into the flame front resulting in increased local burning rate. The turbulent flame propagation initiates as a small laminar flame kernel, this then accelerates into full turbulent combustion. The transition between laminar and turbulent burning is dependant on the turbulence intensity and scale. The turbulent level of the combustion can be related to the fractal dimension of the flame because the flame wrinkling being representative of turbulent distortion of the flame front.

The rates of wrinkling and smoothing are proportional to $u'$ and $S_L$, respectively. A heuristic model of the fractal dimension of flames was derived, equation 2.12. The fractal dimension of the flame $D_F$ is as follows:

$$D_F = \frac{S_L}{u' + S_L} D_L + \frac{u'}{u' + S_L} D_T \quad (2.12)$$

$D_L$ is the laminar flame fractal dimension given as 2.0, whilst $D_T$ is the high Reynolds number or turbulent limit to the fractal dimension, taken as between 2.37 and 2.41. $u'$ and $S_L$ are the turbulent intensity and laminar flame speed respectively. This predicted fractal dimension for a range of $u'/S_L$ compares well with measured fractal dimension of other workers. At higher turbulence levels the spread of experimental results is quite large, Figure 2.23. The model of fractal dimension is then used to predict turbulent flame speed from equation 2.13.
\[ \frac{S_T}{S_L} = \frac{A_T}{A_L} = \left( \frac{\eta}{L} \right)^{0.26} \]  

(2.13)

where \( \eta \), \( L \) are the Kolmogorov scale, (inner cutoff) and the integral length scale, (outer cutoff) respectively and \( A_T \) and \( A_L \) are the turbulent and laminar flame surface area respectively. The relationship between flame areas and the inner and outer cutoff for a fractal object is then used for the remainder of the expression. This is assuming that the Kolmogorov scale is larger than the flame thickness and the turbulent flames are wrinkled laminar flames without flame stretch occurring. This is further modified to account for small kernel size where the radius, \( R \), is less than the integral length scale, equation 2.14.

Figure 2.23 - Plot of fractal dimension against \( u'/S_L \)
This model has been compared to experimental results for a range of turbulent flows and mixtures of various laminar burning velocities. It is sensitive to the values of Kolmogorov and integral scales, but is encouragingly accurate. The major limitations are in the assumption that the inner cutoff or smallest structure scale is that of the Kolmogorov scale and that the model does not account for ignition behaviour and there is variation due to flame stretch.

Fractals can be successfully used to improve the modelling and characterisation of turbulent flame propagation in spark-ignition combustion. The impact on the fundamental understanding of turbulence and combustion requires further work to be carried out before conclusions can be made.

2.4 Proposed Methodology

The overall conclusion that can be drawn from the literature survey is that investigations into spark-ignition combustion should concentrate on a particular region of the combustion system, with a suitable experimental technique. If it is assumed that the distribution of air and fuel is homogeneous, the spark energy is sufficient and the contribution of exhaust residuals to the charge is minimal, then the combustion system optimisation is controlled by the mixture strength and the turbulent flow field. This is for a given compression ratio, engine geometry and inlet pressure. Reducing mixture strength for improved economy and emissions will introduce a greater cyclic variability in combustion duration. A means of improving the combustion rate and hence cyclic
variability is by altering the turbulent flow field. The turbulent flow field is determined by the induction process, the piston speed and the combustion chamber geometry. Modern production spark-ignition engines utilise twin overhead camshafts, central spark plug locations and compact combustion chamber designs.

To investigate the effects of the turbulence on the combustion process, the flame propagation and in-cylinder pressure must be recorded. For studies of overhead camshaft engines the only practical means of optical access is by using Bowditch through-piston-photography. However this compromises the maximum compression ratio and engine speed and in addition the optimum piston crown shape. To fully investigate the turbulent flow field the induction flow areas can be varied and the engine operating speed. The air-fuel mixture strength can be varied from stoichiometric towards the lean limit to investigate the relative effects of increasing turbulence levels on the lean combustion.

Lean mixtures have low levels of blue light emitting from the flame, to adequately record the flames intensified video cameras provide a suitable cost effective approach. The sensitivity of the camera is such that additional light is not required to record flame images. Recording images using a video camera is easily adapted to automatic storage and subsequent processing. Adopting an approach for flame analysis as used by Keck et al.[10] and Beretta et al.[11] allows inference of parameters directly from observed measurements that characterise the effects of turbulence on the flame propagation. By utilising the more recent study of fractal geometry characterisation of the flame front profile is possible. This can lead to improved models where the assumption of a spherical flame fronts does not apply.
3.0 EXPERIMENTAL PROCEDURE

The following chapter outlines the experimental technique and equipment used in acquiring engine test results for this research programme. The chapter is in two sub-sections, first an overview of the equipment and then secondly the measurement technique and test programme.

3.1 The equipment

The equipment includes the Ricardo Hydra engine test cell, video camera system, engine control and data acquisition systems.

3.1.1 Ricardo Hydra engine

The engine is technically specified in Table 3.1 and illustrated in Figure 3.1. It is a single cylinder four stroke internal combustion engine, set-up for through piston photography as discussed in 2.2.1.

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Ricardo Hydra, single cylinder</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>80 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>89 mm</td>
</tr>
<tr>
<td>Capacity</td>
<td>447 cc</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>9:1</td>
</tr>
<tr>
<td>Valve gear</td>
<td>Twin camshaft, four valve</td>
</tr>
<tr>
<td>Cylinder head</td>
<td>Pent roof, combustion chamber</td>
</tr>
<tr>
<td>Ignition</td>
<td>Microcontroller/Lumenition</td>
</tr>
</tbody>
</table>

Table 3.1 - Engine Details
Optical access is provided through an elongated piston into the combustion chamber through a quartz window insert in the flat top piston crown, Figure 2.11 & 3.2. This aluminium and quartz piston is mounted at the top of a slotted piston extension, which allows visual access to the combustion chamber through a mirror at 45 degrees to the engine axis. The extra weight and length of the piston contributes to a number of difficulties resulting in increased piston ring loading, poorer engine balance, engine speed stability and a lower maximum engine speed.

Oil surface lubrication of the upper piston section interferes with the optical clarity of the window, hence piston ring lubrication requires an alternative source. The upper piston rings are made of self lubricating material, graphite or phosphor bronze. This reduces the contamination of the windows, however the piston rings require regular inspection and more frequent replacement then would normally be required. The engine is fired on a propane-air mixture to reduce contamination of the piston rings.
The propane-air fuel mixture is produced by mixing the inducted air and propane gas, the latter supplied from a compressed gas cylinder supply. The fuel is introduced into the air supply using a demand regulator connected to a carburettor, downstream a plenum chamber reduces the pulsations through the carburettor improving the repeatability of the air-fuel proportions in the mixture. The lean limit of combustion is more dilute than would be obtained with liquid fuel, due to the improved mixing.

Quartz window failure is a hazard that can arise through mechanical or combustion effects on the window, this necessitates a complete engine rebuild. To minimise this event a strict regime to control the procedures for an engine build, engine test operations and component design was followed, (Appendix 2).

3.1.2 Engine dynamometer

Engine load control is with a Brush Electrical Machines dynamometer, this controls the load electronically to maintain a constant selected speed. The drive from the crankshaft connects to a generator/motor that starts the engine in a motoring mode and can then switch automatically to provide a load for the engine during firing. The engine is loaded by switching the brake to the regeneration mode of operation, which provides a controlled electrical load for the power generated.

The engine operator can select an engine speed and throttle position and the controller automatically adjusts the engine load to maintain a constant engine speed. The voltage to the stator windings is adjusted to effect the electrical power generated by the brake and hence a closed loop control of engine speed is enabled. The equipment displays torque and speed. Rotary dials are adjusted to demand engine speed and throttle position. The torque displays up to a 10% fluctuation as the load adjusts to keep the speed constant. Because a single cylinder engine with an extended piston was used the engine rotational speed is subject to larger fluctuations than would be normal for such an engine.
3.1.3 Engine control

The engine is controlled by a number of systems: the dynamometer, controlling load and throttle position, a gas flow meter to monitor propane gas quantity and a microprocessor to control spark timing and engine mis-fire. The acquisition of engine signals was controlled with an IBM PC linked to the spark controller. The throttle is adjusted by a rotary dial on the dynamometer control panel, which actuates a servo-motor pulley/cable arrangement to change throttle position. Spark timing is selected through a dial on the microprocessor controller card. This controller provides a camera trigger and data acquisition start signal, for the IBM PC. The microprocessor schematic and software flowchart are shown below, Figures 3.2 & 3.3.

The spark ignition signal can be defined at any point from 99 to 0 degrees before TDC. The camera signal can be set to a point in the cycle at an interval of 2 degrees. The microprocessor controls engine spark and camera timing by monitoring a crank, TDC and camshaft signal. The dial inputs on the front of the controller box determine at which point the output control signals are activated. The system software flow diagram is simplified to illustrate the principles of operation, an acquisition requires a signal from the IBM PC which then effects how the camera, and engine signals are set. The engine and camera control timing is based on three input signals, an engine camshaft sensor (CAM), two crankshaft sensors, Top Dead Centre (TDC) and two degree pulse (CRANK) signals.

The CRANK signal is a square wave TTL pulse generated by an optical sensor providing a rising edge every two degrees of engine crank. The TDC signal is an additional square pulse in synchronism with the CRANK pulse once per crankshaft revolution, timed to engine top dead centre. The CAM square pulse is not critically timed to the engine like the previous pulses, it is merely present in the compression stroke. This identifies the upward firing stroke from the upward exhaust stroke. This ensures timing cannot be 360 degrees out of phase with the engine.
Figure 3.2 - Schematic of Engine/Camera System

Microprocessor Ignition & Camera Control Box

IBM PC Data Acquisition Control and Storage

Start & DMA Signal

Crank Angle

Spark Control

Cylinder Pressure

Camera Trigger

EEV Gated Intensified Camera

Image Pixel Transfer

Ricardo Hydra Engine
Figure 3.3 - Flowchart of Engine control
The microprocessor stores a running total of the crank position 0 (Induction TDC), 360 (firing TDC) to 720 in two degree steps. The CAM signal means phase resynchronisation to the engine occurs every two crank revolutions. The microprocessor can initiate a spark control signal, camera trigger pulse and cylinder pressure acquisition start pulse, all synchronised to engine crank position. The desired position for the spark and camera being selected by the dial inputs. The camera is triggered with a TTL signal connected into the camera controller. An induction TDC signal starts cylinder pressure acquisition, ensuring cylinder pressure capture is always in phase with the camera picture, Figure 3.4.

A test switch controls an internal dummy CRANK signal to simulate a nominal 1500 rpm engine speed and can be used for setting up equipment without running the engine. A trigger mode switch changes between a remote mode and continuous operation mode.
Continuous operation provides signals to repeatedly trigger the cylinder pressure acquisition and camera. The remote mode delays the trigger until an input signal is received from the an external TTL source, generated by in this case the IBM PC. Continual operation permits setting up of camera focus, picture timing and camera gain, whilst remote mode allows test samples to be acquired.

The timing diagram of Figure 3.4 can be explained as follows. On receipt of the initiate pulse from the PC, the inhibited camera and induction signals are ready for activation on the next available cycle. The PC digitises the cylinder pressure signal, (DMA acquire), through the next two consecutive cycles after the induction TDC start signal is received. During the first of these cycles the spark is initiated as normal and a camera picture is taken. The subsequent second cycle is mis-fired, by disabling the spark and camera trigger, however cylinder pressure is still recorded. This provides each test sample of firing cylinder pressures with a 'hot motored' cycle, that is a non-fired cycle with similar temperature and gas composition to a fired cycle.

The engine then runs as normal without a camera signal, but with a spark signal. The PC reads to the disk the video image, (Image Acquire), while the engine carries on for a number of cycles. On completion of the image read a new acquisition cycle can be initiated by the PC. This can be repeated until the required number of samples have been recorded, each sample consisting of an image, a fired and mis-fired engine cylinder pressure cycle. Each image consists of 200x200 values, storage to disk takes approximately 4.5 seconds at 1500 rpm and over one hundred cycles will have passed.

The engine control relies on the manual control of some parameters, such as water coolant flow rates, fuel and air flow rate (open loop control). To maintain the desired values the operator adjusts the controls to agree with sensor outputs. There are various temperature sensors positioned on the engine, inlet, sleeve, exhaust. The fuel, propane gas, is metered through the carburettor and a flow restrictor. The fuel volume flow is measured over a fixed time measured through a gas flow meter. The air flow being
measured by the pressure drop across a calibrated orifice plate. The air and fuel passes through a butterfly throttle as in a conventional carburettor.

3.1.4 Video Camera system

The camera system is a gated-intensified Charge Coupled Device (CCD) unit provided by English Electric Valve company (EEV). The camera head is shown in Figure 3.5, it is small and robust making it possible to mount directly on to the engine. It comprises a CCD array bonded to a microchannel plate (fibre optic bundle) and a photon intensifier. The CCD responds to light falling on the frame, by utilising a pulsed high voltage, the intensifier acts like a shutter briefly amplifying light onto the CCD array. This allows very short exposure times and at relatively low light levels.

Figure 3.5 - Video camera system

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The unit supplied by EEV incorporates the electronic driving circuits for the camera, an image framestore and an IBM PC interface card. The driving circuits are contained partly in the camera head and partly in the control unit and provide the synchronising and conditioning necessary to produce a standard monochrome video signal. The framestore allows this signal to be converted back to an image and stored in local image store.

The framestore image can be viewed on a local screen and downloaded through the interface card to a PC. Because of the asynchronous sampling of pictures a framestore is necessary. The pictures are synchronous to the engine instead of to a normal video time base. The pictures displayed on a monitor direct from the camera are intermittent and flash for a brief instant. With the framestore, which acts as a buffer, the monitor continually displays the most recent capture until the next occurs.

The camera was under development by EEV at whilst in use at Loughborough University and a number of modifications were implemented to enable the research to be more useful. This was principally the adaptation of circuit control to allow the camera to be retriggered to generate multiple exposures. The TTL camera trigger signal was conditioned to provide a series of pulses to the camera controller.

The unit was modified to allow these pulses to re-activate the intensifier/shutter permitting re-exposure of the CCD. The CCD shifts the exposed frame into the covered area of the CCD after a period of 7ms from the last trigger pulse. If a further pulse is detected before the 7ms period has elapsed the uncovered area of the CCD is re-exposed by the intensifier. This can occur a number of times, until 7ms have elapsed between successive pulses. At this point the CCD transfers the composite image to the covered area of the CCD to be transmitted as the next available frame. The framestore is synchronised to capture the exposed frame from the continuous stream of blank frames being transmitted at 50hz.
3.1.5 Computer equipment

Three computer systems have been utilised in this research, two for acquisition/control and one for processing the results. Overall control of acquisition is through an IBM PC AT, with specific engine operation control by an 8085 controller card. Fortran and 'C' software perform the processing of performance data into processed results on an IBM 6150 Unix workstation.

The acquisition controller is a standard IBM AT compatible with an 80286 microprocessor, Enhanced Graphic Adapter (EGA) display, 640k RAM and an 80287 maths coprocessor. Because of the need to store large amounts of image data the storage capacity was increased to an 110 Megabyte hard disk. Real time data acquisition was provided by a Data Translation I/O card, this enables analogue and digital signals to be acquired directly to memory using Direct memory access (DMA). The Data Translation and EEV camera interface cards fitted directly into the standard IBM PC I/O slots. 'C' or Basic software can access and control directly functions provided by these cards.

The ignition control box used a dedicated 8085 microprocessor card developed to provide camera and spark control. The card provides 24 digital I/O lines that are ideal for simple control tasks. The software is written in 8085 assembler on a specialised development system, which had been installed into an EPROM.

Finally an IBM 6150 unix workstation has been used to processes the data. The system includes Fortran, 'C' compilers and image processing developer library IMPART, from the IBM Scientific Research Centre. The power of this system is necessary to improve the level of processing in developing analysis routines and for each image before quantifying data from an image.
3.2 Acquisition of engine conditions

The following section concerns the acquisition of data and the procedures governing the test programme.

3.2.1 Manually Logged Data

The general engine performance observations were recorded manually. This includes temperature and air box pressure differential, engine speed, torque, ignition timing and camera parameters, air and fuel flow measurements and emissions readings. The parameters are recorded after the engine has been allowed to settle at a given condition. The form in Figure 3.6 is filled in by the operator and entered in the computer after testing is complete. The calculated data is added by the computer, this is the information below "test by" line in the test sheet.

Figure 3.6 - Test sheet

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<th>date</th>
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<td>Barometric pressure (mmHg)</td>
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<td>Engine speed (rpm)</td>
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</tr>
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<td>Torque (NM)</td>
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<td>Ignition timing (CA BTDC)</td>
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<td>Inlet manifold pressure (cmH2O)</td>
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<td>Fuel time for 50 counts (sec)</td>
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<td>Manometer reading for air box (in)</td>
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<td>Time when flame image is taken (ca)</td>
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<td>Exposure separation time (ms)</td>
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<td>flow rate of fuel (g/s)</td>
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<tr>
<td>Air/fuel ratio by exhaust analysis</td>
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<td>Brake mean effective pressure (bar)</td>
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<td>1.695</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake specific NO emission (g/kW)</td>
<td>0.000</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
3.2.2 Automatically Logged data

The camera image output and cylinder pressure transducer data is digitised into the computer as discussed in section 3.1.3. The signal digitisation is synchronised for a complete set of results at each test point. This ensures registration of the cylinder pressure and image date in terms of cycle alignment.

An automated test point consists of a set of files for each test condition and each picture crank position, refer to section 3.2.3. For consistency 20 images are taken with associated firing and motoring cylinder pressure data. The storage of all the data of a test sample takes less than 2 minutes for 20 non-consecutive cycles.

3.2.3 Test programme

The test programme was designed to maximise the information from a minimal test matrix. Therefore a range of fuel mixture strength from stoichiometric towards the lean limit were tested. Variation in turbulence levels in the cylinder were tested by different geometrical configurations and different engine speeds. The turbulence levels were varied by using three cylinder head arrangements and three engine speeds, shown in Tables 3.2 and 3.3. The range of mixture strengths were covered with three air-fuel ratio's shown in the Table 3.3.

A file naming convention was taken using a letter followed by three numbers, e.g. B111. This uses the convention of the letter for the cylinder head followed by 1 to 3 for engine speed and then 1 to 3 for air-fuel ratio. The final digit is to indicate test number at a given engine condition. The test range covers a matrix of 27 test points. Each test point has a number of tests relating to the duration of the combustion event. This is because for each engine condition a combustion image and it's associated engine parameters are recorded for the image from just after spark to the point where the flame fills the available window area. This can consist of any number of images from three to nine.
To consider flame variability 20 images are taken at each of the recorded crank positions, these are indicated by one of the letters from A-T on the end of the file name.

<table>
<thead>
<tr>
<th>Head name</th>
<th>Head file label</th>
<th>Head type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Head 1</td>
<td>A</td>
<td>Four valve, 50% &amp; 50% inlet valve split</td>
</tr>
<tr>
<td>Head 2</td>
<td>B</td>
<td>Four valve, 60% &amp; 40% inlet valve split</td>
</tr>
<tr>
<td>Head 3</td>
<td>C</td>
<td>As above, 60% inlet valve disabled</td>
</tr>
</tbody>
</table>

Table 3.2 - Cylinder head configuration

<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>1000</th>
<th>1500</th>
<th>2000</th>
</tr>
</thead>
<tbody>
<tr>
<td>15:1</td>
<td>11</td>
<td>21</td>
<td>31</td>
</tr>
<tr>
<td>18:1</td>
<td>12</td>
<td>22</td>
<td>32</td>
</tr>
<tr>
<td>21:1</td>
<td>13</td>
<td>23</td>
<td>33</td>
</tr>
</tbody>
</table>

Table 3.3 - File naming convention for speed and mixture

The two different cylinder heads, head 1 and 2 are illustrated in Figure 3.7. The configurations of prime interest are the second and third of Table 3.2. The first is very similar to the second, whilst the third head has the large inlet valve disabled reducing inlet flow ports to one and the inlet cross-sectional area to 40% of the second head.

The engine speed covers the available range given the restriction associated with this particular research engine. The air-fuel ratio's cover a range from stoichiometric toward the lean limit for the stable running of this particular engine.
Figure 3.7 - Photograph of Pent roof heads
4.0 FLAME PICTURE ANALYSIS

The following chapter discusses the processing of image data taken with the video camera system. The first section provides an introduction to image processing techniques. The second section considers a method for deriving detailed information from flame images, that is edge contour, position and circle fit.

4.1 Image Processing a brief introduction.

An image is essentially a 2-dimensional representation of the enflamed gas volume. The digital image is equivalent to placing a grid over a photograph, if that photograph is monochrome each element in the grid can be assigned a value representing the level of light between 0 for black and 255 for white, see Figure 4.1 for a coarse grey scale ramp. Digital computers have been able to manipulate data performing a variety of processing techniques. These techniques can be applied equally successfully to the three dimensional data representing an image, this includes both colour and monochrome images.

![Figure 4.1 - Simple image construction.](image)

Image processing treats an image as a mathematical structure, for example a 200 x 200 matrix representing a 2-D image in which each element can have a value between 0-255. This numerical representation of an image is useful in order to allow the image to be stored, copied, printed and manipulated by the computer. The processing of an image is usually considered as mathematical, image enhancing, or feature identification. Mathematical image processing includes simple addition, subtraction and any
mathematical operation which is applied to one or more images as vectors. Image enhancing may use filtering techniques such as convolution masks, fourier analysis or contrast stretching. A convolution mask consists of 2-D array of numerical values, this is applied to an equivalent sized window area of an image. The resultant value replaces the centre pixel of the evaluated region. This can include simple weighted averages and more complicated edge detection masks. Generally the vector is multiplied with the pixel intensities and summed to produce a single value. However a median filter sorts the pixel elements in the vector and replaces the centre pixel with the middle value from the array. This results in a smoother image, without loss of high frequency information.

Fourier analysis of an image results in a spatial representation of the frequency information in the image. Filtering or modifying the Fourier transform of an image results in changing frequency components of the image. Examples of this are to reduce blurring of an image by taking a fourier transform of an image, then filtering the outer area of the resultant image and then taking the reverse fourier transform to generate a new image. Contrast stretching results in a more even spread of image tone across an image. This is by remapping an image from one set of values to a larger set, for example an image may be very dark because the majority of pixels are between 0-128. Re-assigning these to a range of 0-256, rejecting pixel information above 128, will result in a more acceptable visual image.

Feature identification can be very application specific as discussed later. Generally techniques for image segmentation can be employed to divide an image into regions according to a set of rules, for example a given intensity threshold can be used to divide an image into a region above the threshold and one below it.

Propagating flame images are of interest in this research and are of a very specific nature. The major objective is to extract salient features of the flame structure for subsequent analysis. This provides the opportunity of automatically resolving characteristic features.
4.2 Flame Picture Procedures

The procedures required to bring about the flame characterisation discussed above are considered now without specific reference to the program listings, however Figure 4.2 contains a simplified flow diagram of the procedure. The objective is to extract flame edge position from a doubly exposed line of sight flame front image, Figure 4.3. The edge contours provide quantifiable information from the qualitative pictures. This can be used to obtain flame position and location, growth rates and equivalent spherical radii.

![Flowchart of Image processing.](image)

Figure 4.2 - Flowchart of Image processing.
A method has been developed for extracting the above characteristics from a given image. This has been adapted to allow two contours to be extracted from a composite flame picture consisting of two separately timed exposures. These are recorded onto the same CCD before being digitised. The two overlaid images have two distinctive flame contours at the two time positions. These boundaries are representative of the flame position at the end of each of two exposures during a single combustion event. This separation distance and time gap allows an average flame velocity to be determined during the exposure envelope during the combustion event. A sequence of processing steps combine to automate the analysis, whilst producing consistent results.

The software consists of over 2000 lines of 'C' programming language and underwent a number of development stages before the final algorithm was implemented. The analysis is undertaken in a number of steps, which begins with image normalisation, then selection of an intensity threshold to separate the flame from the background,
followed by identification using region growing. The border of the image region is then tracked to produce a flame edge contour. This can then be approximated to with a best circle fit. The outer edge is used to determine a new intensity threshold for the inner flame, the region is identified, the contour and circle fitted, see Figure 4.2.

The separation, exposure control and intensifier gain are important in obtaining good quality results. It was established through testing, during development of this technique, that 5 closely spaced exposures of the order of 10μs followed by a single exposure of 10μs at the desired crank angle later allowed the flame contours to be clearly distinguished. The testing conducted for the results presented in Chapter 7.0 utilise the 5:1 exposure control with a separation of 6 crank degrees between the two main exposures. The initial 5 exposures being close enough to freeze motion and increase flame intensity on the CCD.

To establish the optimum image intensity, maximising the camera resolution over the range of test conditions, the camera intensifier gain was adjusted between tests. Lean mixtures produce pale blue flames with a low light intensity, therefore the gain required was higher than for the stoichiometric air-fuel ratio condition. The gain setting was determined from three parameters, white noise in the image, (present at high gain level), image blooming, (created with CCD saturation) and the later image intensity. The short exposure image can be difficult to separate from the background if the overall image intensity is too low. The gain is set near to the maximum level at which point the inner flame will begin to saturate in the centre. This is compromised by the deterioration in quality due to white noise for faint lean mixture flames and by high cyclic variability resulting in large variations from image to image during a test sample. Some images may saturate whilst others are too faint. The dynamic range of the camera allowed a sufficient span to adequately expose the conditions down to the 21:1 air-fuel ratio tests, however image noise is significantly more noticeable at these conditions. This process of setting the image gain for each new test condition allows the software to reliably and consistently produce good quality flame edge contours.
4.2.1 Development system

The computer system provided for this task is an IBM 6150 Unix workstation with development software for image processing IMPART from IBM UK Scientific Research Centre. This is written in the 'C' programming language to provide a high level structured programming environment. A degree of structure is imposed on both programming and data objects. The environment is particularly suited to handling large data structures in a modular fashion that allows a complex sequence of tasks to be constructed.

IMPART has been based on IAX an IBM research centre image processing utility, the design philosophy behind IMPART is for a library of image processing functions which can be called either from a 'C' program or from an interactive environment. The latter provides an interpretive interface for an operator to build up a command type list of routines and data structures processed on entry. However, in this project the IMPART system was in the early stages of development. The interactive environment had not yet been implemented and only a few utilities had been written. The major benefits of using IMPART have been in developing routines that conform to a recognised standard provided by IBM. The data structure format and module design underpins the image processing system.

The image data file conforms to an IBM image file standard representation KIPS and essentially consist of a 32-byte header, consisting of an 8 bytes name tag, two image pixel size (x and y) values an image type identifier and a 19 byte padding string. This is followed by a list of the x and y bytes, Appendix 3.

4.2.2 Histogram and Normalisation

The digitised flame image can be processed in a number of ways to improve signal-to-noise ratio. An image histogram provides a means of displaying the image data
without spatial information, Figure 4.4. It is essentially a frequency distribution of pixel intensity and it illustrates the dynamic range of an image. For example, a very dark image may consist of the majority of detail shifted to one end. Ideally a histogram should extend across the full range of the intensities available, that is 0-255 for an 8 bit representation. The poorly set up image system or badly lit image can be improved by redistributing the image values across the intensity range. Image quality variations due to different setup conditions can be partially corrected for by normalisation.

![Image Histogram of Figure 4.3.](image)

An image may reside in a pixel intensity range of 0-127 and appear to be relatively dark, by normalising or contrast stretching the pixels are reassigned across the available range 0-255. Even though no data is added and some data may be lost, the image may become apparently much improved. In the case of displaying images of this kind, the display will use the full 0-255 range for it's black to white grey scale and therefore by only using 0-127 the dynamic range of the video screen has been reduced significantly. Data can be lost or corrupted when it's true value is approximated after normalising. Less frequent grey level intensities at the extremes of the grey scale may be re-assigned to the maximum or minimum intensity available. In the example discussed, pixels greater than
127 after normalising could be assigned to 255, so a pixel of 130 could not be reproduced by reversing the function applied to the image. After normalising, subsequent operations applied to the image should be more consistent than to those applied to raw images.

The normalisation approach adopted here divides the image into small 8x8 pixel blocks and evaluates averages for each, to minimise noise effects. The minimum and maximum block average is used as the normalisation limits, equation 4.1. As it is possible that the old intensity value is higher than the maximum or lower than the minimum the result must be restricted to between 0 and 255. The resultant image is shown in Figure 4.5.

![Normalised Flame Image](image)

Figure 4.5 - Normalised Flame Image from Figure 4.3.

\[
\text{Intensity}_{\text{new}} = \frac{\text{Intensity}_{\text{old}} - \text{Intensity}_{\min}}{\text{Intensity}_{\max} - \text{Intensity}_{\min}} \times 255 \quad (4.1)
\]
4.2.3 Threshold selection and segmentation

An image can be segmented into separate regions to identify characteristic features of an image. A common means of segmentation is to select an intensity level as a threshold point within an image or region.

A typical simple black and white image of an object and background often results in a bimodal distributed histogram, that is it's shape consists of twin peaks, foreground and background, with an intervening trough. This, however, is only true when objects are of a consistent shading and intensity such as when even lighting is present.

The edge threshold in the flame application is complicated by a number of factors. Firstly, the intensity gradients are not sharply defined at edges and tend to be gradual. Secondly, particles from the graphite piston rings, produce bright spots disturbing the image texture. Finally, extra complications are introduced by processing images that consist of multiple exposures. This twin exposure method results in overlaid flame images, which blend into each other. Segmentation is required to divide the image into three and not just two zones.

The initial segmentation is achieved by processing the image histogram to produce a grey threshold. This threshold is used to divide the image into two regions, a background and foreground. The foreground can then be split into the two flame regions. An additional approach is then necessary for the separation of the two foreground regions. The human visual system is very good at deciding where edges should be and so visual inspection of processed images provides a quick method of validating the method.

The technique for determining the threshold level for the outer edge was based on excluding the dark or black background pixels. A high number of background pixels are present at the dark extreme, intensity zero. Through a number of test cases a set
cut off value was determined for frequency of dark pixels. Below this frequency the pixel intensity corresponding to this frequency was utilised as the image outer threshold. This was a simple and reliable means of evaluating a threshold level to separate foreground and background.

4.2.4 Region Growing

The intensity threshold from the step 4.2.3 and from the later step described in 4.2.7 is used to segment the image with a technique known as region growing[47]. The threshold divides the image into two regions one above the threshold and the other below. The inner brighter region of the doubly exposed image of the flame will be considered as part of the desired region to be segmented and is identified as having an intensity larger than the threshold, while the outer darker background is below the threshold. This technique relies on identifying elements in the desired region by changing their status.

A central bright pixel is selected, that is above the threshold in a region near the spark plug, then each of it's neighbours can be examined and identified or ignored. The selected pixel is identified by setting it's value to zero. Each of the identified pixel's neighbours are compared with the threshold in turn, those above are similarly selected, Figure 4.6. This is known as a recursive technique, the routine is called repetitively from inside itself. This is preferred to a sequential pixel by pixel examination because it is less susceptible to noise, and a single contiguous block is identified. This labelled region of flame is represented, as in Figure 4.7.

To achieve region growing discussed here a technique from graphics application can be used, that is a recursive flood fill algorithm similar to those applied to computer graphic applications. This is a relatively fast and reliable means of separating out a contiguous block from an image, and is discussed below.
The threshold is 160 for segmentation.
The neighbouring pixels to 245 the centre pixel are examined.
55, 75, 155, 95 are ignored.
200, 185, 195, 205 are selected
The selected pixels are used as the new centre pixel.

Figure 4.6 - Region growing.

Figure 4.7 - Region grow, outer flame of Figure 4.3.
4.2.5 Edge Tracking

The newly identified flame block represents the outer boundary of the second exposure taken with the video camera system. The data of interest is the edge of this zone, the flame position reached after a predefined period of time from the first exposure. The edge is extracted by simply locating the boundary and tracking the edge of the region.

The image is a matrix of grey levels, a sub-section of this matrix contains a block at a redefined grey level, for example black Figure 4.7. The position of the boundary of this region is desired. To establish the coordinates of the edge of this region a boundary pixel is first located. From this pixel boundary pixels are examined in a clockwise manner until a transition occurs from a pixel outside the region to one inside the region. This becomes the neighbouring edge point. The neighbours of this pixel are then examined in a clockwise fashion starting from the previous point in the edge until the transition is encountered.

![Figure 4.8 - Edge tracking.](image)

Gradually the routine tracks through a matrix of values identifying bordering pixels of a region in terms of x,y position until the complete contour is outlined. The example in Figure 4.8 shows the order for examination of neighbouring pixels. The pixel 4
would then become the central pixel of the eight element scan region. A list of coordinates obtained for the edge position is recorded and can be used for the next stage of the quantification process. The resultant edge is demonstrated in Figure 4.9.

![Figure 4.9 - Outer edge contour fit to flame of Figure 4.3.](image)

4.2.6 Circle fitting

The analysis discussed in chapter 2 and 5, relies on the assumption that flame fronts are spherical in geometry. This requires an equivalent circle fit to be evaluated for the flame contour. This can then be used in the subsequent burning flame front analysis to estimate flame volume and surface area, taking into account the combustion chamber geometry. In order to derive projected flame front circle diameter, the flame front area evaluated from the region growing module is assumed to be the equivalent area of the flame circle.
To obtain the circle centre location the following must be satisfied; from a nominal centre point, the difference between the edge point to centre distances and the circle radii must be minimised. An initial centre estimate is made by averaging the edge coordinates. The radius from this centre to the edge coordinates are calculated and the radius subtracted. An iterative process converges on the centre position with the least deviation between the circle radii and that of the edge point radii. The fit obtained for the inner and outer flame contours are illustrated in Figure 4.10.

![Figure 4.10 - Outer and inner edge contour fit to flame circle fit to edge for Figure 4.3.](image)

4.2.7 Inner edge threshold

The sections 4.2.4 to 4.2.6 must then be repeated at a new threshold obtained for the inner region or initial flame exposure. To establish the threshold, radial profiles from the newly derived outer flame centre to the outer edge coordinates are analysed.
Linear arrays of image intensity are taken across the radial profile these are then analysed in sequence. Essentially a point of steepest gradient is desired, this separates the brightly exposed centre region from the lesser exposed outer flame. The exposure period for the early inner flame consists of 5 short exposures, whilst for the outer later flame position a single short exposure is used.

The edge transition occurs at a sharp gradient change and by evaluating the second and third derivative of the intensity profile a position in the intensity array is determined. Figure 4.11 shows a large positive peak in the 2nd derivative where the intensity levels out at the intensity for the outer edge. This corresponds to the negative going zero crossing in the third derivative. The intensity level associated with this profile is taken and averaged with threshold values from profiles taken at regular angular increments around the flame. The profiles are smoothed to reduce effects of noise and the resultant average value used to determine the inner edge grey threshold. This threshold level can then be used in a repeat of 4.2.4 to 4.2.6.

Figure 4.11 - Intensity profile analysis.
4.3 Summary of analysis

The analysis is summarised in Figure 4.12 of the following page illustrating the image processing a flame image undergoes. This image is for the more turbulent faster burning single inlet valve test case. The image is larger and brighter with a greater spacing between flame exposures due to the higher flame propagation speed.
Figure 4.12 - Summary of flame image analysis at 4°BTDC and 2°ATDC for the single inlet valve case at 1500 rev/min, 15:1 air-fuel ratio and ignition timing 10°BTDC.
5.0 IN-CYLINDER CHARACE TURBULENCE ANALYSIS

This chapter discusses the combined analysis of flame picture results, cylinder pressure, and engine geometry used in this research. The literature survey section 2.1 discusses in more detail the work associated with the current understanding of the thick flame entrainment model. Below is presented the implementation and development of the analysis used to derive turbulence characteristics from the recorded engine data.

5.1 Thick flame entrainment model

The work of various authors discussed in section 2.1 has been adopted as a basis for the flame picture analysis. The turbulent combustion model relies on the assumption that an ignited laminar kernel rapidly evolves into a wrinkled flame front entraining fresh mixture into the progressing flame front. The thin flame models fail to accommodate the difference between estimated proportions of burned mixture and flame volumes. With appropriate recording techniques for the visible combustion flame and the in-cylinder pressure the fuel-air mixture entrained behind the flame front can be estimated. The thick flame entrainment model can be divided into a number of stages, initiation, kernel development, turbulent flame front propagation and burning of the remaining mixture behind the flame front.

This model involves understanding the effects of turbulence on the flame front. The eddies of the turbulent mixture ahead of the flame front distort the approaching flame front and are entrained and consumed. The rate of entrainment is a function of the compression of the unburnt gas ahead of the flame front and the progression of this front. The velocity of entrainment, \( u_e \), of fresh mixture is calculated from equation 5.1.

\[
    u_e = u_f - u_t
\]  

\( u_f \) and \( u_t \) are the flame front propagation speed and gas expansion speed respectively.
$u_f$ is determined from the increase in radius of the flame images for a given exposure separation time, Figure 5.1. The radius $r_2$ and $r_1$ are components in the horizontal direction, therefore the radius from flame origin assuming it remains in the same horizontal plane as the spark plug is $r_2/\cos(18^\circ)$, ($18^\circ$ is the pent roof angle).

![Figure 5.1 - Schematic of flame front propagation.](image)

Therefore $u_f = (r_2 - r_1)/(0.95t_{2,1})$ where $t_{2,1}$ is the time between the two flame radii. The gas expansion speed, $u_g$ is derived by assuming the unburned charge ahead of the flame front is subjected to isentropic compression due to flame expansion and piston motion. Therefore:

$$u_g = \frac{1}{A_f} \frac{dV_{fu}}{dt} \quad (5.2)$$

$V_{fu}$ is the volume outside flame and $A_f$ is flame front surface area. The unburned gas is compressed by the moving piston and the advancing expanding flame, Figure 5.2. The quenching effects of the chamber wall will distort the flame front, therefore the camera records the flame front edge away from the pent roof wall, Figure 5.2.
Figure 5.2 - Isentropic compression of unburned charge.

Therefore the unburned volume follows the law below in expression 5.3.

\[ PV_{\beta}^\gamma = \text{constant} \quad (5.3) \]

\[ \therefore \log(P) \cdot \gamma \log(V_{\beta}) = \log(c) \quad (5.4) \]

Differentiating with respect to time the following is obtained, expression 5.5.

\[ \frac{1}{P} \frac{dP}{dt} + \frac{\gamma}{V_{\beta}} \frac{dV_{\beta}}{dt} = 0 \quad (5.5) \]

\[ \frac{1}{P'} \frac{dP'}{dt} + \frac{\gamma}{V_{\beta'}} \frac{dV_{\beta'}}{dt} \quad (5.6) \]
hence

\[\frac{dV_{fu}}{dt} = \frac{V_{fu} \cdot dP}{\gamma \cdot P} \quad (5.7)\]

i.e. Rate of reduction in the end gas volume \(\propto\) Rate of increase in gas pressure.

However, the rate of reduction in the end gas volume is dependant on the pressure increase from piston motion and gas expansion. Therefore, the volume change rate due to piston movement must be accounted for:

\[\frac{dV_p}{dt} = A_b \cdot \gamma_p \cdot \frac{V_{fu}}{V_{cyl}} \quad (5.8)\]

d\(V_p/dt\), gas volume change due to piston motion, \(A_b\) bore area, \(\gamma_p\) instantaneous piston speed and \(V_{cyl}\) instantaneous cylinder volume, \(V_{fu}\) unburned flame volume.

Therefore, the compression of the end gas due to flame expansion \(-dV_{fu(comp)}/dt\) is:

\[\frac{dV_{fu(comp)}}{dt} = \frac{dV_{fu}}{dt} - \frac{dV_p}{dt} \quad (5.9)\]

Therefore, the gas expansion speed can be expressed as follows:

\[u_s = \frac{1}{A_f} \left\{ \frac{V_{fu} \cdot dP}{\gamma \cdot P} - \frac{A_b \cdot \gamma_p \cdot V_{fu}}{V_{cyl}} \right\} \quad (5.10)\]
5.2 Mass loss due to piston blow-by

To obtain a reliable estimate of the gas expansion speed from the cylinder pressure data, the significant effects of low cylinder pressure must be considered. The cylinder pressure reduction is primarily due to mass loss due to gas escaping by the piston rings. This effect is particularly apparent with this experimental photographic single cylinder engine. Significant wear of the piston rings occurs during testing, this generates larger quantities of blow-by than be expected on production engines, (typically 3% according to Heywood[48]).

The isentropic pressure curve is presented in Figure 5.3 with the disabled ignition and fired cylinder pressure. This illustrates the discrepancy between the measured cylinder pressure and the cylinder pressure obtained based on isentropic compression of the inlet charge in the cylinder at bottom dead centre.

![Cylinder Pressure traces graph](image)

Figure 5.3 - Cylinder Pressure traces at 1500 rev/min, 15:1 Air/fuel ratio and ignition timing of 16° BTDC.
From the analysis leading to equation 5.7, the expression 5.11 represents the instantaneous volume change rate of the cylinder determined from the instantaneous volume and rate of change of cylinder pressure. Equation 5.12 represents the instantaneous volume change rate calculated from the instantaneous piston velocity \( v_p \) and cylinder bore area \( A_b \). These have been presented in Figure 5.4, based on isentropic compression of the initial charge.

\[
\frac{-dV_{cyl}}{dt} = \frac{V_{cyl}}{\gamma P} \frac{dP}{dt} \tag{5.11}
\]

\[
\frac{dV_{cyl}}{dt} = A_b v_p \tag{5.12}
\]

Figure 5.4 - Volume change rate at 1500 rev/min, derived from piston motion and cylinder pressure.
The analysis of the motored and fired cylinder pressure for the 1500 rev/min condition using equation 5.11 and 5.12 is presented in Figure 5.5 and demonstrates that the reduction in charge mass due to blow-by results in lower volume change rates. The open squares represent a volume change rate due to piston motion, equation 5.12, adjusted to be representative of the lower cylinder pressure is therefore \( \frac{dV_{cy}}{dt} \cdot 0.9 A_b v_p \).

![Figure 5.5 - Volume change rate at 1500 rev/min, for 15:1 air-fuel ratio ignition timing of 16° BTDC.](image)

The equation 5.10 to determine \( u_g \) evaluates the volume compression rate due to the expanding flame front and is divided by the surface area of the flame front. The calculation is applied to flame images occurring from 340 degrees to 364 degrees crank angle. Therefore an improved estimate results in equation 5.13 for the gas expansion velocity.

\[
u_g = \frac{1}{A_f} \left\{ \frac{V_{fb}}{\gamma P} \frac{dP}{dt} - 0.9 \frac{A_b v_p V_{fb}}{V_{cy}} \right\}
\]

(5.13)
5.3 Inference of Turbulent Intensity

Entrainment velocity has been used by a number of workers discussed in section 2.1 to infer turbulence intensity levels. MiIane and Hill\textsuperscript{(18)} have employed the expression in equation 5.14 to estimate turbulence intensity levels.

\[
\begin{align*}
    u_e &= S_L \cdot \sqrt{\frac{2\rho_u}{3\rho_b}} \cdot u' \\
\end{align*}
\]  

(5.14)

\(u_e\) entrainment velocity, \(\rho\) density of unburned and burned mixture, \(u'\) Turbulent Intensity, \(S_L\) Laminar Burning Velocity.

The Laminar burning speed is obtained using the empirically derived relationships from Metghalachi and Keck\textsuperscript{(49)(50)}. Laminar burning velocities are measured in a closed spherical chambers at typical engine pressures and temperatures for a given equivalence ratio. The flame is ignited and assumed to radially propagate towards the chamber walls. Ionisation probes are used to confirm the flame front shape. The burning velocity is then determined from the rate of mass burned from chamber pressure and the flame front area, the latter is determined from a laser shadowgraph signal generated at a known flame radii. The estimates rely on the assumptions that the mixture is initially of uniform temperature and composition. The thickness of the reaction zone is negligible. The pressure is spatially uniform and the reaction front is smooth and spherical.

A model has been fitted to the results by the authors which allow estimates of laminar burning velocities to be derived. Equation 5.15 shows the correlation presenting their results for laminar burning speeds.

\[
S_L - S_{L,0} \left( \frac{T_u}{T_0} \right)^a \left( \frac{P}{P_0} \right)^b
\]  

(5.15)
where $t_0 = 298$ k and $p_0 = 1$ atm are the reference temperature and pressure, and $S_{L,0}$, $\alpha$ and $\beta$ are constants for a given fuel equivalence ratio and burned gas dilutent fraction. They can be represented as follows:

$$\alpha = 2.18 - 0.8(\phi - 1) \quad (5.16)$$

$$\beta = -0.16 + 0.22(\phi - 1) \quad (5.17)$$

and

$$S_{L,0} = B_m + B_\phi(\phi - \phi_m)^2 \quad (5.18)$$

where $\phi_m$ is the equivalence ratio at which $S_{L,0}$ is a maximum with value $B_m$. Typical values for Propane are $\phi_m = 1.08$, $B_m = 34.2$ and $B_\phi = -138.7$ cm/s [49]. The laminar burning velocity, $S_L$ peaks slightly rich of stoichiometric, typically for Propane 40 cm/s at 1 atm and 300K.

### 5.4 Geometric surface and volume determination

The flame geometry was processed using known engine geometry and a spherical model for the flame progression. The flame is assumed to have propagated radially from the spark plug, (depth into the chamber) towards the cylinder walls. The hemispherical flame front radiates from the cylinder head apex, until it impinges on the piston, pent roof and cylinder wall. The volume and surface area at a particular crank angle is determined from the flame radius and position. The summation of surface area and volume elements is made by the using 3-d polar coordinates to sum over the entire closed angle of the pent roof for the given radius from the flame centre restricted only by the limits of the geometry at a particular crank angle, (Annand[51]). The angular steps of $\delta \theta_x$ and $\delta \theta_y$ are used to sum segments of the flame radius to determine the total surface area and volume bounded by the hemispherical flame surface, combustion
chamber walls, piston top and cylinder walls, Figure 5.6. Where the flame radius extends beyond a physical boundary that segment length is reduced to the limit of the boundary.

Figure 5.6 - Schematic of flame front wall interaction.

The flame surface area and volume is accurately determined assuming the flame is hemispherical and initiates at the height of the spark plug, without vertical centre motion during the propagation of the flame front. The flame centre depth information is not readily available from flame images taken with through-piston-photography. Assuming the flame radially propagates at the same rate the assumption is reasonable, however approaching the piston and wall surface could have significant effects on the flame shape and is assumed to distort the flame as shown earlier in Figure 5.2.
6.0 FRACTAL ANALYSIS

The objective of using fractal analysis on the flame image is to establish a correlation between the flame representation and the corresponding combustion performance, to aid characterisation of the combustion event. Research into the fractal representation of flame fronts is discussed more fully in the Literature survey, section 2.3. This chapter is concerned with evaluating the fractal dimension of the acquired flame fronts. These flame fronts being obtained from the image processing discussed in section 4.0.

6.1 Fractal Dimension

The discussion of the literature in section 2.3 introduced the concepts of fractal dimension for 2 dimensional and 3 dimensional objects. A highly contorted curve such as that shown in Figure 6.1 can be enscribed by a polygon of N sides of fixed length \( e \), the length of the contorted curve \( l \) is proportional to \( N \), and increases as \( e \) is decreased due to the increasing resolution of the measure \( e \). Additionally it is found that when \( \log(N) \) is plotted against \( \log(e) \) a linear relationship is exhibited. The so called fractal dimension \( D_2 \) characterising the roughness of the curve is obtained from the slope of this curve. \( D_2 \) is greater than the nominal Euclidean dimension and is expressed in this relationship, equation 6.1.

\[
l \propto N \propto e^{1-D_2} \tag{6.1}
\]

Fractal dimension, \( D_3 \) applied to a surface demonstrates a similar relationship between surface area and measurement scale, expression 6.2.

\[
A \propto e^{2-D_3} \tag{6.2}
\]
Here \( D_3 \) is the fractal dimension of a surface, \( D_3 = 2 \) for a smooth surface and satisfies \( 2 < D_3 < 3 \). Mandelbrot\(^{[38]}\) explains that due to the difficulties of estimating \( D_3 \) it is possible to determine \( D_3 \) from \( D_2 \). Evaluation of \( D_2 \) can be made from the flame contour boundary. Mandelbrot\(^{[38]}\) states, that by assuming the turbulence is isotropic, the 2 dimensional evaluation of \( D_2 \), can be used to determine \( D_3 = D_2 + 1 \), Figure 6.2.
The fractal dimension, like turbulence is dependant on scale. Engine turbulence scales exist between the Kolmogorov scale at the smallest scale to the Integral scale at the largest, (i.e. from 0.01 mm to combustion chamber height, 17.8 mm). The fractal character of an object persists between a so called inner and outer scale beyond which the fractal dimension is no longer representative of the object. Figure 6.3 illustrates the inner and outer cutoff of a surface, $\epsilon_i$ and $\epsilon_o$, respectively. The inner cutoff is usually a function of the camera resolution as the smallest scales of turbulence wrinkling the flame front are not observable. The outer cutoff represents scales which depart from fractal representation of the curve, this is representative of the largest scales in the flame front.

![Figure 6.3 - Inner and outer cutoff](image)

The flames images in this test programme are projections of three dimensional flame fronts and not two dimensional cross-sections. Therefore $D_2$, derived from the projected image, is not as shown in Figure 6.2. Sreenivasan and Meneveau discuss work carried out into fractal dimension analysis of clouds and deduce that the line of sight image is effectively superposition of many cross sections of the cloud. It is assumed that this will reduce interior fragmentation as obtained by imaging a single slice, whilst exterior roughness is increased because of interacting outer edges. It therefore may be assumed that the overall effect to fractal dimension estimate may be minor.
6.2 Fractal Dimension - determination

Sreenivasan and Meneveau\cite{42} used a technique to determine fractal dimension of turbulent boundaries for digitised images, adopted from a method proposed by Mandlebrot\cite{38}. The image is divided into reactants and products with a flame front boundary. An area within a distance \( e \) from the front is determined that is an area \( 2e_l \), where \( l \) is the length of the flame front. Equation 6.3 shows that the area \( 2e_l \) is proportional to \( e^{2-D_2} \). The slope of the logarithm of \( 2e_l \) against logarithm of \( e \) allows the co-dimension \((2-D_2)\) to be determined.

\[
2e_l = e^{2-D_2} \quad (6.3)
\]

This method is implemented as follows: A circle of radius \( e \) is drawn around each pixel of the image to determine if it intersects the flame front. The number of pixels in the image, \( N(e) \) that intersect the flame front are evaluated for a range of different \( e \). The value \( N(e) \) represents the area of the strip of width \( 2e \) about the flame front, (if the unit area is the pixel area). This is illustrated in Figure 6.4.

Therefore:

\[
N(e) = e^{2-D_2} \quad (6.4)
\]

![Figure 6.4 - Circle intersection evaluation of \( D_2 \)](image)
The outer cutoff is shown in Figure 6.5 the N(\(e\)) against \(e\) plot for a sample test results. The slope of the line is 0.81 tending towards 1.0 at the outer cutoff. The cutoff at \(e_i\) is not evident due to the camera resolution and \(e_o\) the slope changes towards Euclidean geometry. To detect \(e_i\) a resolution of significantly less than smallest physical turbulent scale is required. This scale of turbulence is smaller than the available resolution of the camera and therefore no inner cutoff can be determined.

![Figure 6.5 - Plot of N(\(e\)) against \(e\) to evaluate D_2 and outer cutoff](image)

Two inlet valve, 1000 rev/min and 21:1 air-fuel ratio, \(D_3 = 2.19\), \(L = 3.4\)mm.

The implementation of this algorithm for processing of the 6000 edges obtained, required a fast implementation of the algorithm. The method of evaluating pixels for N(\(e\)) necessitates performing a time consuming iteration on each pixel of the image, 40000 for a 200x200 image. This was improved, reducing computation time to 25% for the above technique, by utilising the edge data obtained from the analysis discussed in section 4.2.
The pixels for N(ε) will all be in close proximity to the flame edge and evaluating the possible intersection of circles for each image pixel is time consuming. The edge contours are much more compact, consisting of less than 1800 pixels. The adopted algorithm generates a filled circle, in a blank image, for all pixels within ε radius from the flame front contour. The new image then consists of a number of black pixels representing all the pixels within ε radius from the flame front contour, all the remaining pixels are black or white. N(ε) is simply determined by counting the number of black pixels in the image. This is repeated for new values of ε to determine N(ε) for a range of ε. After N(ε) is obtained for a range of ε from 1 to 20 the best fit slope and the upper cut-off limit can be determined from the resultant data, as presented in Figure 6.5.

The calibration of the algorithm has been carried out by comparison with the algorithm discussed at the beginning of the section and also by evaluation of the fractal dimension associated with the Koch curve illustrated in Figure 6.6. This can be deduced from it's definition as \( \log 4 / \log 3 = 1.262 \) \(^{[42]}\). The algorithm gives between 1.26-1.28 depending on the size and resolution of the image bitmap representing the shape.

Figure 6.6 - Image representation of Koch curve
7.0 RESULTS AND DISCUSSION

The following chapter presents the significant results produced from the study of spark-ignition flame propagation presented in this thesis. This chapter is divided into two sections spark-ignition combustion analysis, including the camera technique, flame images and cyclic variability. Then the inferred parameters of turbulence intensity and flame fractal dimension are considered.

7.1 Spark-Ignition Combustion Analysis

The results in this section are based on qualitative study of the flame images, assessment of flame contours, the evaluated flame radii and the recorded in-cylinder pressure. The results are representative of testing conducted on the single cylinder Ricardo hydra for two cylinder head configurations, at engine speeds of 1000, 1500 and 2000 rev/min, for wide open throttle with air-fuel ratios of 15:1, 18:1 and 21:1.

7.1.1 The Gated-Intensified CCD Video Camera

The Gated-Intensified CCD Camera, (GICC) technique is relatively recent and has been used in few studies to date. The GICC has proved valuable in this study providing a number of novel features not possible with alternative techniques. Combustion visualisation has been a useful diagnostic tool and is discussed in section 2.2. The camera image intensifier ensures the light sensitivity of the camera is extremely high allowing the pale blue lean mixture flame fronts to be resolved. The multiple exposure feature of the camera permits the camera to be re-exposed for durations as short as 100ns. The images presented in this section are for 5 multiple exposures at 10μs duration interspaced at 10μs intervals followed by a single exposure 6°, (666μs at 1500 rev/min) later. This allows the inner, earlier flame to be essentially multiply exposed at the same flame event, and therefore brighter and more clearly distinguished from the outer, later exposure.
The GICC technique allows image manipulation to be easily carried out. Figure 7.1, shows an image of the combustion chamber.

![Figure 7.1 - Ricardo head reference image GICC.](image)

The flame image displayed in Figure 7.2 illustrates the two exposures of a propagating flame with the overlaid extracted flame edge contour and the cylinder head geometry.

![Figure 7.2 - Flame exposures at 4°BTDC and 2°ATDC, Two inlet valve cylinder head at 1500 rev/min, 15:1 air/fuel ratio and ignition timing 16°BTDC.](image)
7.1.2 Video Pictures and Cylinder pressure analysis

The results presented below, in this section, illustrate the flame images and cylinder pressure available for qualitative study of turbulence and flame stoichiometry. Later sections will consider greater quantitative analysis of the engine test data. The Figures 7.3-7.6 present 20 flame images for each of 2 engine conditions at 2 different camera timings. Whilst Figure 7.7 and 7.8 present the associated cylinder pressure data for the two engine conditions.

The engine tests were performed with the single inlet valve activated at 1500 rev/min for 15:1, (stoichiometric) and 21:1 air-fuel ratio. Figure 7.3 and 7.4 are the stoichiometric air-fuel ratio tests and show good repeatability flame to flame over the 20 images, note the outer flame (2° BTDC) of Figure 7.3 is slightly larger than the inner flame of, (4°BTDC) of Figure 7.4 as would be expected. Flame propagation rates can be gauged by the distance between the inner and outer flame over a 6° crank angle interval, (the printed images in the Figures are 65% of full size and at 1500 rev/min the time spacing is 666μs).

The lean air-fuel ratio tests of Figure 7.5 and 7.6 are similar to those of the 15:1 air-fuel ratio case of Figure 7.3 and 7.4. This is due to high turbulence with the reduced inlet flow area. The main differences are that the 15:1 air-fuel ratio flames of Figures 7.3 and 7.4 show a larger spatial gap between the inner brighter flame and the outer flame contour. In addition the flame position and size show a greater repeatability in Figure 7.3 and 7.4. The lean case of Figure 7.5 and 7.6 show a lower concentricity between the inner and outer flames and the approximation to a circle of the flame boundary is not as good as for the Figure 7.3 and 7.4.
Figure 7.3 - Twenty Flame images, (condition C211), Single Inlet valve cylinder head, 1500 rev/min, 15:1 Air-fuel ratio Ignition 10°BTDC, Flame Exposure at 8° and 2° BTDC.
Figure 7.4 - Twenty Flame images, (condition C213),
Single Inlet valve cylinder head, 1500 rev/min, 15:1 Air-fuel ratio
Ignition 10°BTDC, Flame Exposure at 4° BTDC and 2° ATDC.
Figure 7.5 - Twenty Flame images, (condition C231), Single Inlet valve cylinder head, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 18°BTDC, Flame Exposure at 12° and 6° BTDC.
Figure 7.6 - Twenty Flame images, (condition C233),
Single Inlet valve cylinder head, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 18°BTDC, Flame Exposure at 8° and 2° BTDC.
Figure 7.7 - Cylinder pressure plot of 20 cycles for C211 test sequence. Single Inlet valve cylinder head, 1500 rev/min, 15:1 Air-fuel ratio Ignition 10°BTDC, Flame Exposure at 8° and 2° BTDC.

Figure 7.8 - Cylinder pressure plot of 20 cycles for C231 test sequence. Single Inlet valve cylinder head, 1500 rev/min, 21:1 Air-fuel ratio Ignition 18°BTDC, Flame Exposure at 12° and 6° BTDC.
The outer flames in Figure 7.4 are on average 37mm in diameter, whilst those of Figure 7.6 are 31mm in diameter. The cylinder pressures of Figures 7.7 and 7.8 illustrate the cyclic repeatability of cylinder pressure development, indicating a similar trend to that observed from the flame pictures. The time from ignition to the first flame picture demonstrates the small change in cylinder pressure as compared to the variation in visible flame size, Table 7.1. The faster flame propagation speed of the 15:1 air-fuel ratio condition allows an ignition timing nearer to TDC without a detriment in peak cylinder pressure.

<table>
<thead>
<tr>
<th>Crank Angle</th>
<th>8°BTDC</th>
<th>4°BTDC</th>
<th>2°BTDC</th>
<th>2°ATDC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius (mm)</td>
<td>4.0</td>
<td>7.6</td>
<td>11.2</td>
<td>18.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Crank Angle</th>
<th>12°BTDC</th>
<th>8°BTDC</th>
<th>6°BTDC</th>
<th>2°BTDC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius (mm)</td>
<td>3.7</td>
<td>7.9</td>
<td>9.3</td>
<td>15.5</td>
</tr>
</tbody>
</table>

Table 7.1 - Flame radii for Figures 7.3-7.6.

The set of test results above are for the two extreme air-fuel ratios and for a single engine speed of 1500 rev/min and single cylinder head case. That is 80 flames out of over 2000 images, further detailed analysis of the data requires a different approach. However a sample comparison of data is presented for the 1500 rev/min condition at two air-fuel ratios for the two cylinder head configurations, (i.e. single inlet and twin inlet valve operation). Figures 7.9-7.12 show the 15:1 air-fuel ratio case with single and twin inlet valves at an early and late flame exposure crank position. Figures 7.13-7.16 show the 21:1 air-fuel ratio single and twin inlet valve case at an early and late flame exposure crank position. The Figures 7.17-7.20 present 20 cycles of cylinder pressures for the each of the engine conditions used for the images, that is single and twin inlet valve operation, 15:1 and 21:1 air-fuel ratio at 1500 rev/min.
The Figures 7.9 and 7.10 illustrate the early flame development of the 1500 rev/min 15:1 air-fuel ratio condition. The flame kernel size difference between the two exposures for the 6 crank angle degree displacement is larger for the single valve case, which is confirmed in the later images of the same engine condition, Figures 7.11 and 7.12. Figures 7.17 and 7.18 show that the faster burn improves the repeatability in cylinder pressure development and increases peak cylinder pressure, for a more retarded ignition timing of the single inlet valve condition.

The 21:1 air-fuel ratio test case presented in Figures 7.13-7.16 show the same trend as discussed for the stoichiometric case. The difference in the combustion effectiveness, for the single and twin inlet arrangement, is more evident in the lean air-fuel ratio case. Irregularity in flame uniformity, low flame intensity and small displacements between inner and outer flames are illustrated in Figure 7.16. This demonstrates a slower combustion with twin inlet valve than single inlet valve configuration. This leads to very low peak cylinder pressures and high cycle to cycle variation in cylinder pressure development, as shown in Figure 7.20, as compared to those of the single inlet valve of Figure 7.19. Although an air-fuel ratio of 21:1 is used for tests presented in Figure 7.19, a higher peak cylinder pressure and improved repeatability is achieved than those of Figure 7.18 for the twin valve, 15:1 air-fuel ratio case. This shows the strong influence of turbulence on stable combustion. These results show that the effects of air-fuel ratio can, to a certain degree, be compensated for by increased level of turbulence. However it must be noted that this engine operating speed is lower than normal engine operating speeds, where the improved breathing associated with four valve cylinder heads is of equal importance with the improved turbulence levels due to increased inlet flow velocities at these speeds.

This section is concluded with illustrations of the flame edge and geometry overlaid onto several images for a single engine condition showing four examples at three camera timing positions in Figure 7.21. The cylinder pressure shows increased peak cylinder pressure over that in Figure 7.18 for the slower engine speed.
Figure 7.9 - Single inlet valve, 1500 rev/min, 15:1 Air-fuel ratio
Ignition 10°BTDC, Flame exposure at 8° and 2° BTDC

Figure 7.10 - Twin inlet valve, 1500 rev/min, 15:1 Air-fuel ratio
Ignition 16°BTDC, Flame exposure at 10° and 4° BTDC
Figure 7.11 - Single inlet valve, 1500 rev/min, 15:1 Air-fuel ratio
Ignition 10°BTDC, Flame exposure at 4° BTDC and 2° ATDC

Figure 7.12 - Twin inlet valve, 1500 rev/min, 15:1 Air-fuel ratio
Ignition 16°BTDC, Flame exposure at 2° BTDC and 4° ATDC
Figure 7.13 - Single inlet valve, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 18°BTDC, Flame exposure at 12° and 6° BTDC

Figure 7.14 - Twin inlet valve, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 34°BTDC, Flame exposure at 20° and 14° BTDC
Figure 7.15 - Single inlet valve, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 18°BTDC, Flame exposure at 8° and 2° BTDC

Figure 7.16 - Twin inlet valve, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 34°BTDC, Flame exposure at 8° and 2° BTDC
Figure 7.17 - Single inlet valve, 1500 rev/min, 15:1 Air-fuel ratio
Ignition 10°BTDC, Cylinder Pressure - 20 cycles

Figure 7.18 - Twin inlet valve, 1500 rev/min, 15:1 Air-fuel ratio
Ignition 16°BTDC, Cylinder pressure 20 - cycles
Figure 7.19 - Single inlet valve, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 18°BTDC, Cylinder Pressure - 20 cycles

Figure 7.20 - Twin inlet valve, 1500 rev/min, 21:1 Air-fuel ratio
Ignition 34°BTDC, Cylinder pressure 20 - cycles
Figure 7.21 - Twin inlet valve, 2000 rev/min, 15:1 Air-fuel ratio, Ignition 24°BTDC, Edge data.

Figure 7.22 - Twin inlet valve, 2000 rev/min, 15:1 Air-fuel ratio, Ignition 24°BTDC, Cylinder pressure.
7.1.3 Cyclic Variability

The flame derived data for the flame velocity, flame radii and circle centre position relative to the spark plug position can be used to quantify the flame characteristic for a given turbulence or air-fuel ratio. An initial plot of the flame propagation speeds against flame radii demonstrates the acceleration of the flame as the radii becomes larger, Figure 7.23. This is comparable to Figure 2.9, Beretta et al [11] discussed earlier in the literature survey section 2.1. The flame propagation speed accelerates until a given radii is reached where the velocity becomes constant.

Further comparisons between Figure 7.23 and 7.24 clearly demonstrate the larger velocities for a given radii for the 15:1 air-fuel ratio condition. The stoichiometry is shown to be less influential than between the conditions in Figure 7.25 and 7.26 where two inlet valves are activated. The increased turbulence levels of the single valve head generate higher flame velocities in the lean mixture case of Figure 7.24 beyond that of the 15:1 case of Figure 7.25. The flame velocity appears to become constant as the flame grows in size for the twin inlet valve configuration. This shows that the 15:1 air-fuel ratio case is not affected by the increased turbulence due to the expanding flame front. The lean mixture 21:1 air-fuel ratio case with twin inlet valves, Figure 7.26, show significant fluctuation in flame propagation speed, for a given radii, particularly given the low propagation speeds. These fluctuations are reflected in the cylinder pressure development of Figure 7.20.

The fluctuation in flame images shown in the previous section indicate that flame size, position and cylinder pressure vary from cycle to cycle. The Coefficient of variation, (COV) is defined as the standard deviation of the sample expressed as a percentage of the average of that sample. By plotting the COV of peak cylinder pressure against the COV of flame radii, Figure 7.27, a good correlation is evident. This relationship is as would be expected as the flame radius is directly related to the enflamed volume and hence the proportion of the mixture contributing to increased cylinder pressure.
Figure 7.23 - Single inlet valve, 1500 rev/min, 15:1 air-fuel ratio.

Figure 7.24 - Single inlet valve, 1500 rev/min, 21:1 air-fuel ratio.
Figure 7.25 - Twin inlet valve, 1500 rev/min, 15:1 air-fuel ratio.

Figure 7.26 - Twin inlet valve, 1500 rev/min, 21:1 air-fuel ratio.
Figure 7.27 - Correlation of cyclic variability of peak cylinder pressure with flame radii for 1500 rev/min single and twin valve testing.

Figure 7.28 - Flame displacement from plug centre against flame radii for 1500 rev/min twin valve, 15:1 Air-fuel ratio.
Figure 7.29 - Correlation of flame centre displacement against Peak cylinder pressure for 1500 rev/min single and twin valve testing.

The lean 21:1 air-fuel ratio mixture for the twin inlet valve configuration shows greater than 10% fluctuation in peak cylinder pressure, this would be reflected in poor drivability. The flame radii fluctuation is greater than 20%. Improved fluctuation for the lean air-fuel ratio mixture is achieved over the stoichiometric mixture by using an increased level of turbulence with a single inlet valve activated.

The flame centre position relative to the spark plug provides an additional useful measure as compared to the flame size data. The discussion of section 2.1 Figure 2.8, Beretta et al\cite{beretta2000} concludes the flame centre displacement increases as flame radius increases to a maximum displacement. Figure 7.28 does not clearly establish the same trend, however perhaps this is due in part to the cyclic variability of both data sets. It is not possible to plot a sufficient number of points from a single engine cycle to verify this trend. In addition the very early kernel flame centre trajectory is not directly recorded only inferred from different cycles. The single valve 15:1 air-fuel ratio condition have very small displacements away from the spark plug and show little
movement with flame radii. However when plotted against peak cylinder pressure a
trend can be observed indicating a smaller displacement away from the spark plug
generates a higher peak cylinder pressures and hence lower cyclic variability. This
further corroborates the conclusions of Keck et al\textsuperscript{[10]} and Beretta et al\textsuperscript{[11]} in that the
random walk of the flame kernel in the turbulent flow field significantly effects the
consistency of the later overall combustion duration, hence combustion variability and
variations in cylinder pressure development.

7.2 Inferred Turbulence and Fractal results

Chapter 5 and 6 discuss the equations used for the analysis of flames and cylinder
pressure to infer turbulence levels and fractal characteristics of flames. The turbulence
intensity ($u'$) and length scale ($\lambda$) are derived from gas expansion speed ahead of the
flame front and flame propagation speed, obtained from cylinder pressure and flame
images respectively. These are used to evaluate an unburned mixture entrainment
velocity, which has been empirically related to turbulent intensity by Milane and Hill\textsuperscript{[18]}.

7.2.1 Mixture Entrainment velocity

The entrainment velocity, $u_e$ discussed in section 5.1 is the velocity at which fresh
mixture is consumed by the traversing flame front. This has been evaluated for each
flame image in order to determine the levels of mixing occurring in the engine. The two
extreme data sets at 1500 rev/min have been plotted in Figure 7.30 and 7.31. This
shows the relatively large entrainment velocity, Figure 7.30, of the single valve 15:1
air-fuel ratio condition as compared to the lower entrainment velocity, Figure 7.31 of
the twin valve, 21:1 air-fuel ratio case. Earlier in section 7.1.3 the cyclic variability was
shown to be considerably larger with the lean mixture, low turbulence and twin inlet
valve cylinder head. We further see here the effects of the lower entrainment velocity
in reducing the flame front burning surface area under these operating conditions.
Comparing Figures 7.23 to 7.30, it can be seen that for a 10 mm flame radius in the single valve 15:1 air-fuel ratio case the propagation velocity is 13.2 m/s and the entrainment velocity is 6.7 m/s. Whereas Figures 7.26 and 7.31 for a 10 mm flame radius in the twin inlet valve, 21:1 air-fuel ratio case the propagation velocity is 8.0 m/s and entrainment velocity is 5.2 m/s. From the expression discussed in chapter 5.0 equation 5.1 the entrainment velocity has been derived from the flame propagation speed and the gas expansion speed. It is evident that the gas expansion speed has reduced by a larger extent than the propagation speed. This is demonstrated in reduced cylinder pressures, Figure 7.20 in the lean mixture condition. The lean mixture has given rise to lower cycle temperatures and lower gas expansion speed.

The Figure 7.30 and 7.31 show an increase in entrainment velocity with flame radius indicating the progression from laminar flame to a highly wrinkled thick flame front. The entrainment velocity in Figure 7.30, for the single inlet valve test, appears to become constant or reduce as the flame radii exceeds 15 mm. This could be due to combustion chamber wall effects.

The flame propagates radially from the spark plug, the chamber height is 17.8 mm from the apex of the pent roof to the piston crown. At 9° BTDC, (flame radius of 15.9 mm), the piston is 0.7 mm from TDC position and therefore 18.5 mm below the apex of the combustion chamber. The flame propagation velocity at the 15 mm flame radii has been evaluated from exposures at 12° BTDC and 6° BTDC, which correspond to 9.8 and 22 mm flame radii average. Therefore it can be safely assumed that the flame is in contact with the piston. The propagation speed is 18.4 m/s, therefore it should take 473 μs to reach the piston assuming a constant velocity between the two exposures and piston movement is relative small the contact should occur at approximately 8° BTDC.
Figure 7.30 - Entrainment Velocity against Flame radii for Single valve, 1500 rev/min and 15:1 air-fuel ratio.

Figure 7.31 - Entrainment Velocity against Flame radii for Twin valve, 1500 rev/min and 21:1 air-fuel ratio.
7.2.2 Turbulence and length scales

The data for $u'$ and $\lambda$ has been obtained over a range of conditions and are presented as Figure 7.32 and 7.33. The single inlet valve case shows that the turbulence levels are significantly higher than those for the twin valve case. The trends show an increase in levels with respect to engine speed with similar magnitudes to those presented by Lancaster\textsuperscript{52} using hot-wire anemometry in a motored CFR engine with and without masked valves.

Similarly the microscale ($\lambda$) data shows the expected trends with increasing engine speed and higher turbulence intensity. Compared to the predictions by Tabaczynski\textsuperscript{53}, magnitudes are larger in the twin intake valve system due to the lower levels of turbulence generated. This being the case due to greater valve flow areas and lower induction velocities. Data from the single valve case correspond more closely to the results presented by Dent and Salama\textsuperscript{54} in a wedge shaped chamber.

To evaluate the trends of the data for $u'$ and $\lambda$ obtained, comparisons of the cylinder head data have been undertaken using the results of the scaling analysis from Blizard and Keck\textsuperscript{12}. The intensity $u'$ is considered to be proportional to the induced velocity at the valve seat $u_i$ at maximum valve lift $L_v$, which is obtained by equating the induced volumetric flow rate into the cylinder to that past the valve. Hence

$$u' \propto u_i \propto \eta_{volumetric} \left( \frac{B^2SN}{D_vL_v} \right)$$

where $B$ and $S$ are the cylinder bore and stroke, $N$ is the engine speed in rev/min and $D_v$ and $L_v$ the inlet valve diameter and lift respectively and $\eta_{volumetric}$ the volumetric efficiency. A comparison between heads can be considered, $B$ and $S$ are the same and inlet valve area for the single valve case is approximately 40% of the Twin valve case.
Therefore:

\[
\frac{u_s'}{u_T'} = \frac{\eta_{VOLs}}{\eta_{VOLT}^{0.4}} \quad 7.2
\]

The subscripts S and T are used for single and twin valves respectively. From measurements carried out during testing over the engine speed equation 7.2 gives the \( u_s'/u_T' = \text{constant} \approx 2.06 \) which is similar to that obtained in Figure 7.32.

For the Taylor microscale, \( \lambda \) Tabaczynski\textsuperscript{[53]} showed that for a constant compression ratio

\[
\lambda \propto \left( \frac{1}{u'} \right)^{0.5} \quad 7.3
\]

Therefore from equation 7.2 and 7.3 we would expect

\[
\frac{\lambda_S}{\lambda_T} \propto \left( \frac{u_T'}{u_s'} \right)^{0.5} = \left( \frac{1}{2.06} \right)^{0.5} = 0.7 \quad 7.4
\]

Which agrees with the data of Figure 7.33.

These results indicate that the methods presented here to evaluate characteristic values of \( u' \) and \( \lambda \) provide reasonable orders of magnitudes and trends, which are in agreement with other workers\textsuperscript{[52][53][54]}, without the direct measurement of flow velocities and turbulence levels.
Figure 7.32 - Turbulent Intensity ($u'$) against Engine Speed for Twin and single inlet valve, 15:1 and 21:1 air-fuel ratios.

Figure 7.33 - Micro length scale, ($\lambda$) against Engine Speed Twin and single inlet valve, 21:1 air-fuel ratio.
7.2.3 Fractal Analysis

The fractal dimension, \( D_z \) data characterises the fractal nature of flame front contours and can be used as a parameter in quantification of image results. \( D_z \) provides a means of correlating flame characteristics with engine parameters, as an alternative means of investigating combustion criteria. The results gathered for fractal dimension fall in the range:

\[
2.1 < D_z < 2.3
\]

Fractal dimension \( D_z \) has been plotted against peak cylinder pressure for 1000 rev/min for the twin and single inlet valve arrangements in Figure 7.34 and 7.35.

![Figure 7.34 - Peak cylinder pressure against Fractal dimension at an engine speed of 1000 rev/min the Twin inlet valves.](image-url)
Both plots show a trend of reducing cylinder pressure with reducing air-fuel ratio and increased fractal dimension. The latter being indicative of a more convoluted flame boundary as the charge is diluted. For each of the figures the laminar flame velocity is reduced, turbulence remains constant and hence the flame chemistry is less dominant and allows the charge turbulence to wrinkle the flame surface to a greater extent. The higher turbulence associated with the single inlet valve case results in increased fractal dimension for a given air-fuel ratio and an overall reduced spread in the peak cylinder pressure between the stoichiometric and 21:1 air-fuel ratio. From the 1000 rev/min data set it is apparent that the spread of fractal dimension appears to be larger in the single inlet valve case of Figure 7.35. This spread in the data lessens the clarity of the observed trend in Figure 7.35.
To further investigate the relationship between maximum cylinder pressure and fractal dimension the cyclic variability in terms of Coefficient of variation, (COV) of peak cylinder pressure was derived for each engine condition over the 20 cycle samples for a given set of flame images. This is representative of the fluctuation in cylinder pressure stability and was plotted against the average of fractal dimension for the 20 cycles, Figures 7.36-738. Figure 7.36 shows the COV in peak cylinder pressure against fractal dimension for 15:1 to 21:1 air-fuel ratio for the two and single inlet valve case at 1000 rpm. A good correlation is demonstrated between increasing air-fuel ratio, increasing fractal dimension and increasing COV of peak cylinder pressure. This is further reinforcement of the points raised with Figure 7.34 and 7.35, that the reduction in the laminar burning speed, $S_l$, due to increasing air-fuel ratio leads to both higher cyclic variability and increased fractal dimension. The reducing turbulent flame speed, as discussed earlier leads to increased cyclic variability.

![Figure 7.36 - COV in Peak Cylinder Pressure against Fractal dimension for 1000 rpm](image-url)
Figure 7.37 - COV in Peak Cylinder Pressure against Fractal dimension for 1500 rpm

Figure 7.38 - COV in Peak Cylinder Pressure against Fractal dimension for 2000 rpm
Figure 7.37 shows a similar trend for the twin inlet valve arrangement with a slightly increased fractal dimension and cyclic variability at the lean air fuel ratio. However for the single valve higher turbulence case the trend it unclear indicating a very low cyclic variability in peak cylinder pressure, but a large spread in fractal dimension. Figure 7.38 shows that at 2000 rpm both the single and twin inlet valve cases show inconclusive trends, however the fractal dimension spread is high for the single valve condition and small for the twin inlet valve, whilst the for COV of peak cylinder pressure shows a large spread for the twin valve case and a small spread for the single valve case. The two inlet valve condition can be explained in part by referring to Figure 7.39 and equation 2.12.

\[ D_F = \frac{S_L}{u' + S_L} D_L + \frac{u'}{u' + S_L} D_T \]  

(2.12)
Figure 7.39 shows the variation of fractal dimension, D₂, with u'/Sₛ obtained in this study plotted with the model of Santivicca et al[46] equation 2.12. The slightly lower D₂ trend, is shown with the dashed line using a value of Dᵣ = 2.25 instead of Dᵣ = 2.35, (Dₛ = 2.0). This indicates that the fractal dimension will tend to a value near to Dᵣ. Resulting in a possible trend associated with the twin inlet valve case of Figure 7.38. The equation illustrates the chemistry effects of increasing laminar flame speed with the effect of smoothing the flame front contour, whilst increasing turbulence tends to increase the entrainment of mixture behind the flame front leading to increased turbulent burning and distorted flame front boundary.

The reduced fractal dimension observed in Figure 7.39 can be explained by considering non-engine based studies by Goix et al[55] and Gouldin et al[56], which produce values in the range 2.1-2.2 nearer to those produced here than those in engine based studies such as Santivicca et al[46]. It is likely that the results of Mantzaras et al[43] used by Santivicca[46] are strongly influenced by the higher swirl motion associated with the engine configuration. With their high swirl values and lean mixtures (φ=0.59) there is a considerable effect due to the bulk gas motion in distorting the propagating flame front, contributing to higher D₂. It is feasible that the line of sight viewing of the combusting cloud associated with the GICC technique may be responsible for creating a slight averaging effect as compared to a slice from a laser sheet. This line of sight integration will tend to give a less ragged flame shape, which will reduce interior fragmentation of convoluted flame boundaries and result in a lower D₂.

The single valve case demonstrates a spread of fractal dimension from slightly larger than 2.1 to 2.26, however the Figure 7.36 to 7.38 show that whilst cyclic variability is relatively low for all engine speed and air fuel ratios the COV of peak cylinder pressure is lowest at low engine speed with the highest fractal dimension. The overall spread in fractal dimension is large for a single set of data at a given air-fuel ratio. A closer inspection of the results indicates a trend associated with the different flame images. The single valve data set consists of a reduced number of points for each condition due to the
high flame propagation velocities and hence rapid progression of the flame front across the piston window. Therefore for a given condition a reduced number of samples are taken typically 3 or 4, whilst the twin inlet case required 4 to 9 samples at 2 crank degree intervals.

The reduced data set of the single inlet valve testing shows a much greater spread for a given condition than does the twin inlet valve tests. This appears to complicate the relationship between cyclic variability and flame front contour. A single data set should consist of flame images at different times for the same condition, therefore a similar COV of peak cylinder pressure should be measured, whilst it appears that in the single inlet valve case fractal dimension varies with flame position and size. Figure 7.23 to 7.26 show the single valve arrangement has a high flame acceleration in comparison to the twin inlet valve configuration. The rapid flame acceleration is possibly resulting in flame contour irregularity changes that vary from the flame initiation to the fully developed flame, which are more apparent than for two inlet valves. The data indicates that fractal dimension decreases at large flame radii, this may be due to the cylinder wall effects. To determine more conclusive trends the contributing factors need to be assessed by acquiring greater samples of flame contours at different sizes and positions over these operating conditions.

From the evaluated outer cutoff, $e_0$ as discussed in section 2.3 an estimate of Integral Length scale L was made. Figure 7.40 illustrates the Integral length scale plotted against engine speed for the two configurations of inlet valves. The values are of the same magnitude to those reported by Lancaster[52] and are similar to those measured by Zur Loye et al[57], where L is considered to be equal to 0.21 x clearance height. In this study the clearance height to piston face at TDC to the spark plug is 17.8mm which would make $L = 3.7 \text{ mm}$. 

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Figure 7.40 - Fractal Integral length scale against Engine Speed
8.0 CONCLUSIONS AND RECOMMENDATIONS

The following chapter presents concluding points to this research, followed by recommendations as to possible further work.

8.1 Concluding remarks

The work presented covers a range of techniques and analysis pertaining to spark-ignition combustion and in-cylinder combustion analysis. The conclusions from this body of research are presented below.

The GICC technique has been developed here, when published, it was unique and novel providing a significant contribution as an alternative technique for in-cylinder engine flame propagation studies. It uses recently developed Gated-intensified CCD camera technology in-conjunction with modifications originating from this research programme to enable high speed multiple exposure. This allows the flame propagation speed to be determined. Combustion imaging studies, only possible with High speed film cameras or additional laser illumination, are now possible using this technique.

The image processing algorithms developed for extracting flame edge contours have shown that quantitative data can be reliably and quickly extracted from the complex flame images associated with this multiple exposure technique. Generic or commercially available image processing routines can extract data from simple flame images, however it will be some time before these techniques can be applied to more complex multiple exposed images.

A comprehensive data set of engine test results including flame images, cylinder pressure and performance measures has been established for a range of in-cylinder charge stoichiometry, inlet port and piston speed induced turbulence levels.
The direct imaging of the line of sight propagating flame front has been proven to be successful as a means of determining flame front propagation speeds and indirectly inferring turbulence characteristics, this is based on the assumption that the flame front approximates to a hemispherical surface. Whilst it is only possible to normally determine turbulence parameters through direct measurement of flow velocities and fluctuations through laser based techniques or hot-wire anemometry.

The analysis of flame contours has demonstrated that, in addition to flame size and flame location, more characteristic quantities relating to the flame front geometry can be determined using Fractal theory. This fractal characterisation of the flame front roughness correlates with peak cylinder pressure fluctuation at the lower engine speeds of 1000 rpm and 1500 rpm, however this does not hold at higher engine speeds and for the 1500 rpm and higher for the single inlet valve case. The fractal dimensions determined for the flame edges are lower than presented elsewhere and likely to be due to the imaging technique, however trends are similar to those presented in the literature. The evaluation of cloud fractal dimension and composite Koch curves representative of superposition of image slices indicate that estimated $D_2$ values may be reduced from the true $D_2$ of the image slice, due to the reduction in interior fragmentation. The fractal dimension of flame contours at higher turbulence levels, that results in relatively low cyclic variability appear to have a large spread in fractal dimension.

Cyclic variability of turbulent combustion as expressed by the COV of peak cylinder pressure appears to correlate to the COV in flame radii, (flame size). The cyclic variability of cylinder pressure correlate more directly to the flame displacement away from the spark plug, that is increased displacement results in larger cyclic variability. This is consistent with other workers and further evidence supporting the effects of significant influence of the early flame kernel position.

The research results have demonstrated the importance of matching the in-cylinder turbulent flow field to the mixture stoichiometry and that through controlled levels of
turbulence, lean mixtures can produce levels of cyclic variability lower than stoichiometric mixtures. The test results based on a modern production four valve cylinder head suggest that to provide optimum engine performance across the engine operating speed range requires the control of inlet flow area. This can be reduced at low engine speeds to induce higher port generated turbulence, whilst at high engine speeds the volumetric efficiency is maintained by increasing flow area.

8.2 Recommendations for future work

The results indicate that this technique has shortcomings as compared to the laser sheet imaging technique, particularly when detailed analysis of the outer flame contour is desired. Further work could be conducted to improve the technique and establish the effect of line of sight imaging by utilizing a variable thickness laser sheet to investigate the effect of the superposition of the irregularity in the plane perpendicular to the viewing angle. The effective fractal dimension for the faster propagating flame associated with the single inlet valve case could be investigated over a greater number of flame sizes or crank positions by utilizing a smaller interval between flame images.

In addition higher resolution cameras or imaging of a small area of the flame front could be carried out to evaluate a lower fractal dimension cutoff, representative of the Gibson scale, $L_g$. The pixel resolution would have to be of the order of 0.01 mm, as compared to the current resolution 0.3 mm/pixel.

The GICC technique lends itself to in-cylinder combustion investigations for combustion abnormality. The repetitive capture of flame propagation leads to an improved understanding of the consistency and effectiveness of the combustion event from cycle to cycle. Further investigations into cyclic variability would be desirable at part throttle conditions and possibly investigating component design changes.
By using high quality emissions analysis equipment it is recommended that investigations relating flame characteristics to Hydrocarbon emissions, for a range of mixture stoichiometry be conducted. A correlation between flame data, either flame position, flame radii or fractal dimension may exist with the levels of Hydrocarbon emissions. This may be evident due to the poor combustion of very lean mixtures, giving rise to irregular flame fronts and poor combustion efficiency. Alternatively high levels of turbulence may lead to flame quenching resulting in high hydrocarbon emissions and altered flame characteristics.

Clearly further work is required to verify the use of this technique, particularly where the fractal analysis of flame fronts would be undertaken.
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APPENDIX 1: An introduction to turbulence theory

Most flows in nature are turbulent for example jet streams in the upper troposphere, water currents and cumulus clouds. Laminar flow is more usually the exception not the rule for natural flows. These tend to arise in environments of small dimensions with fluids of high viscosity for example lubricating oil flows around a bearing surface. The random nature of turbulent flows makes quantification difficult and therefore analysis is based on characterisation and qualitative studies of flows. It is therefore important to fully describe all the terminology in this field. The terms used to describe turbulent flows and characterise them are outlined below from Tennekes and Lumley [58].

- **Irregularity**, the randomness of the flow tends to lead to a non-deterministic approach to the study and analysis of such flow. Statistical techniques are usually required in the analysis of turbulent flow.

- **Diffusivity**, the nature of turbulence is to diffuse outwards into the surrounding flow generating rapid mixing and increased rates of momentum and hence higher heat and mass transfer.

- **Large Reynolds number**, as the Reynolds number is increased the laminar flow becomes unstable due to interactions of the viscous terms and the non-linear inertial terms in the equations of motion.

- **Three dimensional vorticity fluctuation**, the turbulence involves three dimensions and is rotational resulting in high levels of vorticity. The vorticity maintenance mechanism is known as vortex stretching and is absent in two dimensional flows. Cyclones in the atmosphere are essentially two dimensional and hence not turbulent.
APPENDIX 1 (Continued) : An introduction to turbulence theory

- **Dissipation**, turbulent flows are always dissipative, the viscous shear stresses increase the internal energy and reduces kinetic energy, this energy must be maintained to maintain turbulent flow.

- **Continuum**, turbulence is a continuum phenomenon governed by the equations of fluid mechanics at scales larger than the molecular length scales.

- **Turbulent flow**, turbulence is not a property of the fluid it is of a result of the flow. With large Reynolds numbers the major characteristic of a flow is independent of the molecular properties of the fluid. Non-linear terms in the equations of motion means the initial and boundary conditions are important. The Navier Stokes equation cannot be solved and so no general solution for turbulent flows can be determined and used to classify turbulent flows.

**Turbulence applied to Spark-ignition engines**

The turbulence generated inside the combustion chamber plays an important part in controlling the fuel-air burning rates during combustion phase of the internal combustion engine operating cycle. The influence of combustion chamber shape, inlet system geometry have major effects on the turbulent flow field up to and during combustion. This effects fuel economy, drivability and exhaust tail pipe emissions. The importance of the flow field has been long understood. Measurement and hence quantification of turbulence levels has always been complex and met with limited success. The turbulence is generated by high shear flows during the inlet phase of the cycle and shear flows are also generated during compression due to the break down of large scale motions created during induction and chamber squish effects.
APPENDIX 1 (Continued) : An introduction to turbulence theory

The engine is a periodic device and is therefore not readily analysed by using time averaging techniques. The technique most widely used is an ensemble averaging approach, many tests are carried out and the results averaged over time slices of the engine cycles. The characterisation of the flow is divided into a mean flow $\bar{U}$ and a fluctuating velocity component $u$. A third term is used to describe turbulent flows and this is the characteristic length scale or integral length scale $L$. The instantaneous velocity $U(\phi)$ in the cylinder at any crank angle $\phi$ is expressed as:

$$U(\phi) = \bar{U}(\phi) + u(\phi) \quad (1)$$

where

$$U(\phi) = \frac{1}{N} \sum_{i=1}^{N} U(\phi_{in}) \quad (2)$$

where $n$ is the number of strokes in an engine cycle. The so called turbulence intensity $u'$ can be determined by:

$$u'(\phi) = \sqrt{\frac{1}{N} \sum_{i=1}^{N} \left[ U^2(\phi_{in}) - \bar{U}^2(\phi) \right]} \quad (3)$$

This is representative of the instantaneous fluctuation in the flow during an engine cycle. The integral or macro length scale, $L$ of the flow is defined as the integral of the autocorrelation coefficient of the fluctuating velocity at two adjacent points in the flow with respect to a variable distance between the points and is a measure of the large scale structure of the flow field.

$$L_x = \int_0^\infty R_x \, dx \quad (4)$$
APPENDIX 1 (Continued) : An introduction to turbulence theory

where

\[ R_x = \frac{1}{N} \sum_{i=0}^{N-1} \frac{u(x_i)u(x_i+x)}{u_i'u_i+x} \]  (5)

It is however difficult to obtain this data due to the requirement of two measurement probes. Investigators have more successfully determined length scales through a correlation for the integral time scale of turbulence \( L_i \). This is defined as a correlation between velocities at a fixed point in space, but separated in time.

\[ L_i = \int_0^\infty R_i \, dt \]  (6)

where

\[ R_x = \frac{1}{N} \sum_{i=0}^{N-1} \frac{u(\tau)u(\tau-i)}{u_i'u_i} \]  (7)

This can be determined with a single probe and the non-stationary autocorrelation coefficient \( R_i \). The length scale \( L_x \) is equal to the product of the mean flow \( \bar{U} \) and \( L_i \).

\[ L_x = \bar{U}L_i \]  (8)

This is only valid if the mean flow velocity is constant, the turbulence is homogeneous and the relative turbulent intensity \( u'U \) is small compared to one. If no mean flow exists in the chamber near TDC of the compression stroke then an alternative relationship must be employed using a constant \( C \) of order 1.

\[ L_x = Cu'L_i \]  (9)
The integral length scale is representative of the large-scale structures and is at a scale of the order of the height of the combustion chamber. Superposed on this scale are a range of eddies smaller in size produced from the breakdown of these larger eddies. The Taylor microscale, $\lambda$ is a smaller scale which is a measure of the fluctuating strain rate of the turbulent field and is thought to be important in the ignition delay problem. This is obtained from the correlation curve of Figure 1 derived from expression (4) and (5). The Taylor microscale, $\lambda$ is determined by the second derivative of the correlation coefficient $R$.

The following equations are representations of the micro length scale and time scale respectively.

\begin{align}
\lambda_{x}^2 &= -2/(\delta^2 R/\delta x^2)|_{x_o} \quad (10) \\
\lambda_{t}^2 &= -2/(\delta^2 R/\delta t^2)|_{t_o} \quad (11)
\end{align}

Figure 1 Correlation coefficient $R_x$ and estimate of $\lambda_x$
APPENDIX 1 (Continued): An introduction to turbulence theory

The Kolmogorov scale, $\eta$ is representative of the smallest of eddies in turbulent motion, molecular viscosity acts to dissipate small scale kinetic energy into heat.

$$\eta = \left(\frac{v^3}{\varepsilon}\right)^{1/4} \quad (12)$$

where $v$ is kinematic viscosity and $\varepsilon$ the turbulent energy dissipation rate per unit mass.

Finally the Gibson scale is defined as the smallest scale that interacts with the flame front.

$$L_\alpha = \frac{u_f^3}{\varepsilon} \quad (13)$$

Where $u_f$ is the flame front propagation speed. This has the characteristics of a lower cutoff scale, because the eddies smaller than $L_\alpha$ and larger than $\eta$ will not wrinkle the flame front due to the low circumferential velocity. Larger eddies distort the flame front around the eddy.

The flame structure is further characterised by the dimensionless parameters of the Reynolds number and the Damkohler number. The turbulence is defined by the turbulent Reynolds number where $Re_T = u'L/v$. The characteristic turbulent eddy turnover time is defined as $\tau_T = L/u'$ and the characteristic chemical reaction time is $\tau_L = \delta_L/S_L$, where $\delta_L$ is the thickness of the laminar flame and $S_L$ is the laminar flame speed. The Damkohler number, $Da$ is then defined as follows:

$$Da = \frac{\tau_T}{\tau_L} = \left(\frac{L}{\delta_L}\right)^{\frac{S_L}{u'}} \quad (14)$$

The Damkohler number is an inverse measure of the influence of the turbulent flow on the chemical processes. The regimes of turbulent flames are illustrated in Figure 1.
The two regimes that are normally identified are those of distributed reactions and reaction sheets. In the distributed reaction regime, chemical reactions proceed in distributed reaction zones and thin sheet flames do not occur. This regime exists when the laminar flame thickness, $\delta_L$, is greater than the Integral length scale, $L$. The energy carrying eddies are significantly smaller than the flame thickness and therefore a laminar flame cannot exist. The reaction sheet regime consists of propagating laminar flame fronts, which are wrinkled and convoluted by turbulence. This occurs when the Kolmogorov scale is significantly greater than the laminar flame thickness. However, as can be seen from Figure 1, typical turbulent combustion for a spark-ignition engine is predominantly in the reaction sheet regime. The area below the reaction sheet line tends to be where the laminar flame speed is low due to high levels of residuals in the air-fuel mix. Generally, however, one would expect a thin reaction sheet, wrinkled and convoluted by turbulent flow.
APPENDIX 2 : Engine operation/running

To operate the engine a sequence of steps were followed, these are listed below:

1. Oil heater and coolant fan, depending on air temp.
2. Follow check list: Exhaust isolation valve opened,
   Exhaust drain valve closed,
   Oil and water pumps on
3. Power to dynamometer switched on.
5. Check ignition box run mode, spark timing.
6. Coil power switch.
7. Motor or Auto mode of dynomometer.
8. Motor ; Load on, motor speed adjust.
   (Gas valve open, choke, engine takes off,
   Switch to Auto, Fuel on)
9. Auto ; Fuel on, gas valve open,
   Engine start to 800rpm
   (choke, engine takes off)
10. Load on.
11. Set mains water flow to control sleeve temperature.
12. Adjust throttle, A/F ratio, spark timing to suit.

To shutdown the following steps should be taken:

1. Turn gas and mains water off.
2. Water and oil pump isolation switch.
3. Dynomometer isolation switch.
5. Switch off ancillary equipment.

The gas and mains water valves are closed whenever the engine stops, to prevent unnecessary gas build up and over cooling of the engine. The exhaust valves are to prevent condensation leaking back to the head and isolate exhaust gases from other sources engines in the pipe network.
APPENDIX 3 : File formats:

The image data file conforms to an IBM image file standard representation KIPS and essentially consist of a 32-byte header, consisting of an 8 bytes name tag, two image size x and y values, an image type identifier and a 19 byte padding string. This is followed by a raster list of the x and y bytes.

The engine test sheet, results file, flame circle fit and fractal dimension output files are all ASCII files, which list each of the test cycles in turn.

The firing cylinder pressure, motoring cylinder pressure and heat release files are binary files. Each file starts with a 2 byte integer for the number of points and is followed by a list of those points, two bytes per point.

The edge data is also a binary file starting with a two byte integer for the file size followed by a list of x,y coordinates, each point is two bytes long.