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THE IMPACT OF FUEL DISTRIBUTION ON CYCLIC COMBUSTION VARIATIONS IN A NATURAL GAS FUELED SPARK IGNITION ENGINE

A DISSERTATION

Presented in Partial Fulfillment of the Requirements for the Degree Doctor of Philosophy in the Graduate School of The Ohio State University

by

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ABSTRACT

Combustion variations that occur between individual cycles during spark ignition engine operation are a major cause of elevated emissions, depressed fuel economy, and power fluctuations in internal combustion engines. Reducing the level of cyclic variations in natural gas engines is of particular importance as these engines typically operate at significantly higher compression ratios and with wider equivalence ratio limits than their gasoline counterparts. The role of mixture inhomogeneity in generating cyclic combustion variations is little understood due to the difficulty in determining the fuel distribution within the cylinder during operation.

The purpose of this work was to measure the level of mixture inhomogeneity in the cylinder of a natural gas engine and correlate these results with the level of cyclic combustion variations as determined from cylinder pressure data. The distribution of fuel within the cylinder was measured in three different planes, as a function of fuel injection timing and crank angle. Processed Planar Laser Induced Fluorescence images provide a two dimensional measure of the equivalence ratio in each of the given planes as a function of crank angle and fuel injection timing. The mixture formation process and the state of the mixture just prior to ignition were analyzed statistically to provide
information pertaining to the bulk distribution of the fuel and the magnitude of small scale inhomogeneities.

The state of the mixture in terms of small scale inhomogeneities, bulk maldistribution of fuel, and cyclic variations in total fuel content and distribution were found to correlate well with the level of cyclic combustion variations in the engine. More specifically, increases in the inhomogeneity of the cylinder charge were found to correlate closely with the variation of ignition delay. Variations in the overall burn duration and maximum cylinder pressure were found to be most closely related to fluctuations in the total amount of fuel present in the cylinder in a given cycle. This study has also led to the determination of an injection timing which produces the most uniform fuel distribution for this specific engine.
Dedicated to John Galt
ACKNOWLEDGMENTS

I wish to thank my current advisor, Dr. Mo Samimi for his support and intellectual input, my previous advisor Dr. Lawrence Kennedy for his contribution to this work, Mr. Keith Rogers, without whom the construction and operation of the Optical Access Engine at Ohio State would not have been nearly impossible, and Mr. Mike Flory for his assistance and support throughout my research. I would also like to recognize the financial contributions of Honda R&D of Japan, the National Science Foundation, The Ohio State University through a Presidential Fellowship, and the Ralph Kurtz Chair in Mechanical Engineering.
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CHAPTER 1

INTRODUCTION

The variations in combustion that occur between individual cycles in spark
ignition engines are a major cause of elevated pollutant levels, depressed fuel economy,
and poor driveability in IC engine powered vehicles. Discrepancies in cylinder pressure
evolution and flame propagation impede the development of more efficient, cleaner
burning power plants as engines must be designed to operate in the full range of pressures
and burn durations which may occur. These considerations are becoming increasingly
important as interest in lean-burn and alternatively fueled engines grows. In order to
minimize the impact of cycle-to-cycle variations (CTCV), the physical processes which
cause them must be understood. The purpose of this research has been to study the
degree to which one of these causes, mixture inhomogeneity, influences the amount of
CTCV which occurs in natural gas fueled engines.

The main indication of CTCV in spark ignition engines is variability in the
evolution of cylinder pressure from one cycle to the next. In particularly fast burning
cycles, the cylinder pressure rises rapidly, with the maximum pressure being higher and
occurring early in the expansion stroke. In slow burning cycles, the pressure peak occurs
later, and is considerably lower because of the larger volume into which the fuel's energy
is being release. The general result of CTCV is to create slow burn cycles in which the
spark timing is retarded from maximum brake torque timing (MBT) and fast burn cycles
in which the spark is advanced from MBT. Fast burn and slow burn cycles both create
difficulty in optimizing engine design. Fast burning cycles lead to higher cylinder
pressures which can result in auto ignition of the end gas region, producing engine knock. Knock is detrimental both in its potential to cause engine damage and its contribution to elevated emissions. Slow burning cycles result in reduced engine efficiency as the heat release occurs later in the expansion cycle. In addition, if the heat release becomes sufficiently slow, partial or complete misfire can occur. This results in both increased emissions and poor fuel economy. In order to compensate for these effects, compression ratios are lowered to prevent knock in the fastest burning cycles, and air/fuel ratios and charge dilution are limited in order to prevent partial or complete misfire in the slow burning cycles.

The degree of CTCV in spark ignition engines is known to depend on several parameters that influence the combustion process. Foremost among these causes are the in-cylinder flow field, including both bulk flow and turbulence intensity, the state of the mixture within the cylinder in terms of concentrations and spatial distribution, and the characteristics of the ignition process. The bulk in-cylinder flow field impacts CTCV both through its effect on mixture formation, and it's impact on flame kernel development. The turbulence intensity of the flow also impacts mixture formation, but has additional impact due to its influence on flame propagation velocity. Concentrations of fuel, air, and residual gases within the cylinder contribute to CTCV mainly through their effect on laminar flame speed during kernel development. The ignition characteristics of the engine have substantial control over the duration and timing of the flame initiation phase of the combustion process. Both of these items make a substantial contribution to the degree of CTCV.

The level of mixing of injected fuel, incoming air, and residual gases during the intake and compression strokes plays an important role in the quality of combustion that occurs in a given cycle. Stoichiometric and fuel rich regions in the fresh charge are easily ignited due to the higher flame speed in rich mixtures and the greater energy which is released in a given volume. Lean mixtures on the other hand are difficult to ignite consistently as the rate of heat transfer out of the flame kernel approaches the rate of heat release. The results of attempted ignition in these leaner regions range from slow flame
initiation which leads to delayed combustion, to the complete extinguishing of the flame resulting in misfire. For this reason, the degree of homogeneity of the cylinder charge has a great influence on the CTCV of a given engine.

These effects are particularly poignant in natural gas fueled engines. Due to the relatively low laminar flame speed of natural gas, the duration of flame kernel initiation is extended, and the overall duration of combustion is longer than in similar gasoline powered engines. This fact is especially disturbing when taken in the context of the desire to utilize lean burn technology in natural gas engines. The high octane number and greater flammability limits of natural gas make it a fine candidate for application in lean burn engines. On the other hand, the longer combustion duration in natural gas is extended even further as the air/fuel ratio is increased above stoichiometric. In order to reap the benefits of lean burn operation, the air/fuel ratio must be increased beyond the point of maximum nitric oxide emission. The lean limit of operation for natural gas engines is determined by the onset of unacceptable levels of CTCV. In homogeneous charge engines, the lean limit is extended by producing a consistently uniform mixture. In stratified charge engines, the lean limit can be further extended by ensuring that the volume of charge in the region of the spark plug during ignition is richer than the overall mixture. In either case, the impact of mixture strength throughout the combustion chamber, and particularly near the spark plug, on combustion quality must be better understood.

The quantification of the impact of mixture inhomogeneity on combustion variations requires that both the degree of inhomogeneity and the combustion quality for any given cycle be measured. While the quantification of combustion quality through cylinder pressure measurement is a well-established technique, the determination of mixture homogeneity is not a straightforward matter. Many techniques have been used in the past to measure mixture quality. These range from analysis of the exhaust gases, to in-cylinder sampling valves, but until recently there was no means by which the spatial distribution of fuel in the engine cylinder could be measured accurately during a single cycle. Optical techniques developed in the last twenty years make this measurement of
mixture distribution feasible in optically accessible engines during engine operation. Raman scattering provides the ability to measure the concentration of several species simultaneously, but the signal levels are too low to make two-dimensional measurements possible. Planar Rayleigh scattering has been demonstrated in an engine cylinder, but it is difficult to separate the scattering by molecules from light scattered from cylinder walls, and other components. In order to overcome these difficulties, molecules with large scattering cross sections must be used as a fuel tracer, making combustion in these engines impossible. This makes the application of Rayleigh scattering for the study of CTCV impractical.

Planar Laser Induced Fluorescence (PLIF) is an optical technique in which molecular tracers are excited with a high intensity laser sheet to an electronic state from which they emit Stokes shifted photons. PLIF is an effective technique for measuring fuel distributions during the intake and compression strokes of an internal combustion engine for several reasons. Due to high signal levels, small concentrations of molecular tracers can be utilized, thus reducing the impact on combustion events. The longer wavelength photons which are emitted from the excited species are shifted sufficiently to be easily separated from the scattered laser light, thereby significantly reducing background noise. The intensity of the PLIF signal is not heavily dependent on the pressure and temperature of the flow, making quantification of obtained images a relatively straightforward process. PLIF has been used in this study to determine the distribution of fuel within the engine cylinder as a function of crank angle and fuel injection timing.

In order to study the impact of mixture inhomogeneity on combustion quality and CTCV, other factors which play a role in this phenomenon must be understood and controlled. For this reason, Chapter 2 discusses not only the causes of CTCV but also operating parameters which influence these causes. Each of the factors discussed earlier is reviewed with respect both to its direct impact on combustion quality and to its influence on the mixture formation process. In this work, the role of each factor excluding mixture inhomogeneity has been either minimized or understood to the point
that its influence can be removed from the experimental results. The operating parameters which control the direct causes of CTCV have been studied in order to determine conditions which are of particular importance. For example, items such as the overall air/fuel ratio and percent EGR have been carefully controlled in order to make measurements of the impact of mixture inhomogeneity meaningful. Also covered in Chapter II is the importance of, and difficulty in, generating a uniform charge in natural gas engines.

Chapter 3 is dedicated to the measurement of mixture homogeneity in internal combustion engines. The various optical techniques which are available are discussed, in addition to older methods such as sampling valves, exhaust gas measurement, and flame ionization detectors. The theory of planar laser induced fluorescence is covered in the detail required for its application to concentration measurements in the engine environment. This includes an overview of tracer selection and a review of previous tracers utilized in this capacity. Measurements of fuel distributions in gasoline engines which have been carried out recently are reviewed so as to understand their relation to the current work.

Chapter 4 covers experimental methods which are used to determine combustion quality on a cycle resolved basis. This includes an overview of techniques such as exhaust gas measurement and flame front imaging, but focuses mainly on cylinder pressure based techniques. The engine cylinder pressure is of great importance in quantifying combustion events. Indicated mean effective pressure, maximum cylinder pressure, and the angle at which the maximum cylinder pressure occur have all been utilized as measures of combustion quality in the study of CTCV. With appropriate simplifying assumptions, combustion parameters such as the heat release rate and burn duration can also be determined from accurate pressure measurements. Several of these parameters have been used in this work, and thus a thorough understanding of the acquisition of accurate pressure measurements and the processing of this data is required.

The specific experimental methods which have been utilized are discussed in Chapter 5. This discussion includes an overview of the Optical Access Engine
Laboratory at The Ohio State University where all of the research described here was conducted. The experimental procedures which were used are described in detail. This covers the engine operating conditions of interest, the procedure for collecting and processing fuel distribution images and pressure traces, and the interpretation of the experimental results. The equipment which has been utilized includes the optical access engine, the PLIF excitation source, the image collection system, and various other items pertaining to the application of PLIF to engine flows. The design of both the PLIF and cylinder pressure experiments is discussed with respect to the acquisition of accurately and statistically significant results.

Chapter 6 describes the PLIF experimental results. While it is not practical to display all of the over 5000 images which have been collected, selected images are presented which give a thorough description of the evolution of the mixture field. Mean images provide an idea of how the state of the mixture in the cylinder develops with advancing crank angle for any given injection timing. These images are used to explain how the mixture reaches its final state just prior to ignition. In addition to the mean images, instantaneous images are presented. These images serve to describe the level of small scale inhomogeneity in the flow, as well as the variation in the bulk distribution from cycle to cycle. The last section of Chapter 6 is a statistical analysis of the PLIF images. This analysis provides the quantitative measure of mixture homogeneity which is required in order to determine the impact of mixture quality on combustion parameters.

Chapter 7 includes a presentation and discussion of the cylinder pressure traces which have been acquired. Cylinder pressure data have been collected for each of the injection timings tested, at 4 different equivalence ratios. A sampling of the pressure traces collected is shown so as to provide a basic understanding of the level of CTCV which is occurring in each test case. This presentation is followed a statistical analysis of the pressure data. Several parameters were chosen to represent the overall combustion quality. The criteria for the selection of these parameters, as well as their variation with injection timing is discussed.
A direct comparison of the PLIF image statistics and cylinder pressure combustion statistics is carried out in Chapter 8. As both sets of data were collected with respect to fuel injection timing, the timing itself can be removed from the analysis, resulting in a direct comparison of the distribution of fuel within the cylinder and the resulting level of CTCV. A series of graphs detail the direct relationship which has been found between mixture inhomogeneity and CTCV in the natural gas engine used in this study. Because the specific fuel injection timing has been removed, the results obtained are applicable in other engine configurations which display similar levels of mixture homogeneity to that found in this study.

The variation of combustion events from one engine cycle to another in an internal combustion engine has a detrimental effect on overall engine performance. In order to minimize the impact of these variations, their causes must be understood and quantified. The inhomogeneity of the cylinder charge is known to be a major contributor to this problem. The relatively new technique of planar laser induced fluorescence provides the opportunity to measure fuel distributions directly, and thus to study the impact of mixture inhomogeneity on combustion quality. The purpose of this research project has been to correlate mixture distribution images in a natural gas fueled engine by PLIF with measures of combustion quality determined from cylinder pressure measurements. An understanding of this correlation will be useful in the future design and implementation of natural gas fueled internal combustion engines.
CHAPTER 2

CYCLIC VARIATIONS IN SPARK IGNITION ENGINES

2.1 Introduction

The variation of cylinder pressure on a cycle resolved basis during internal combustion engine operation is a well known phenomenon. These cycle-to-cycle variations (CTCV) result in reduced engine efficiency and increased emission of unburned hydrocarbons, oxides of nitrogen, and carbon monoxide. The reduction of CTCV is receiving increased attention due to the need to further reduce engine emissions and fuel consumption in modern engine applications. The desire to develop stratified charge, lean burn engines further amplifies the necessity to understand the causes of CTCV in order to minimize its effects.

There are many operating parameters which are known to influence the degree of CTCV in spark ignition engines. Altering parameters such as overall air/fuel ratio, residual gas fraction, and engine intake geometry can reduce the level of cyclic variations that occur. In order to make substantial progress towards the minimization of CTCV though, the phenomenological causes must be understood. The root causes of CTCV must be related to the conditions of the cylinder charge during ignition and combustion. More specifically this can include the temperature, pressure, composition, distribution, and velocity of the mixture, as well as any variation in the ignition source. From this basic list emerges the physical causes of CTCV: the in cylinder flow field, including bulk flow and turbulence intensity, charge composition, including the concentration of air, fuel, and residual gases, charge distribution, and ignition characteristics.
Engine geometry and operating conditions influence the CTCV of any given engine by affecting the physical causes listed above. The intake geometry determines, to a large extent, the bulk cylinder flow, as well as affecting charge mixing. The cylinder and combustion chamber geometry affect both the turbulence intensity and the turbulence length scale. The fuel preparation system, including geometry, delivery timing, and method of delivery affect the overall air/fuel ratio in a given cycle as well as the distribution of fuel within the cylinder. The ignition system geometry, especially the spark gap and electrode shape have influence on the electrical discharge characteristics. The overall operating condition, including load, speed, spark advance, percent EGR, and overall air/fuel ratio have an impact on several of the root physical causes of CTCV. Finally, the characteristics of the fuel influence CTCV through their burning properties, mainly due to difference in energy density and flame speed. The purpose of this chapter is to give an overview of both the root causes of CTCV in internal combustion engines and the engine parameters that affect these causes. In addition, the significance of mixture inhomogeneity in producing CTCV in natural gas fueled engines is described in detail.

2.2 Phenomenological Causes of Cyclic Combustion Variations

As described above, the variation in the combustion event from one engine cycle to the next can be related to the state of the mixture contents during ignition and combustion, as well as any energy transfers with the outside boundaries. While the temperature and pressure of the mixture prior to combustion can vary from cycle to cycle, the variation is inconsequential and can be neglected as a possible cause. This leaves the fluid motion within the cylinder, the composition of the combustible charge, and the distribution of the various constituents of the charge as defining the important state of the mixture. Possible interactions of the charge with the system boundaries include heat transfer from the cylinder walls, valves, and cylinder head, as well as the ignition event which constitutes an energy input. The impact of heat transfer on the overall heat release is significant, but on a cycle to cycle basis does not vary significantly. The only
exception to this is heat transfer between the spark plug electrodes and the developing
flame kernel, which will be discussed later. Leakage from the cylinder and adsorption
and desorption in engine oil influence emission formation, but are not likely to contribute
to CTCV. Several excellent references are available which give a comprehensive
overview of the current state of knowledge concerning the causes of CTCV (Young 1981,
Ozdor 1994).

2.2.1 In-Cylinder Flow and CTCV

The flow of mixture within an engine cylinder during the intake and compression
strokes of engine operation is a highly complex and turbulent process. While large scale
flow structures may be reproducible from one cycle to the next, engine flows are very
sensitive to minor variations in geometry and engine operating point. For this reason,
even the overall bulk flow within the cylinder can vary substantially leading to CTCV. In
addition, turbulent flow is of great importance in engines. Turbulence is generated in the
intake process, but it also results from the dissemination of energy from the breakdown of
large scale structures during the compression stroke. The impact of engine flows on
mixture distribution is discussed later, while the direct influence of bulk and turbulent
flows on combustion processes is the subject of this section.

Bulk engine flows are considered to be the large scale structures which convect
the mixture around the combustion chamber during the intake and compression strokes.
Bulk flows in engines are typically discussed in terms of swirl, tumble and squish. Swirl,
a vortex whose axis is parallel to the cylinder axis is prevalent in single intake valve
configurations, and in dual intake valve systems in which the valve profiles are
significantly different. Swirling flows begin early in the intake process, and continue into
the combustion phase. Squish is a phenomena in which the mixture is forced toward the
cyliner centerline during compression. This type of flow is present mainly in bowl type
combustion chambers, which are more common in compression ignition engines than in
spark ignition engines. Tumble can be described as a vortex whose axis of rotation is
perpendicular to the cylinder axis. Tumble flow is typical in 4-valve engine geometries,
and tends to dominate other bulk flows in these engines. Four-valve engines which are the norm in modern automobiles typically have a combination of tumble and swirl during the intake and compression strokes.

The tumble vortex which is generated during the intake stroke tends to continue through the beginning of the compression stroke at which time it is broken up into small scale turbulence as the piston constricts its motion. Thus there is very little tumble motion in the cylinder at the time of ignition and during the combustion process which follows. For this reason, the major bulk flow which influences the combustion process directly is swirling engine flow.

Large scale flow structures in engines have their greatest impact during the spark ignition phase of combustion. Tagalian and Heywood (1986) studied the impact of this type of flow on flame kernel growth. Their main conclusion is that the flame kernel is convected away from the spark plug gap by bulk flows. Using high-speed schlieren images, the evolution of a large number of flame kernels was traced over the duration of their evolution. With the variations in speed and direction of the bulk flow through the spark plug gap, the convection of the flame kernel varied substantially. The importance of this finding is the effect that flame kernel convection has on CTCV. As the kernel is stripped from the plug gap it can travel towards combustion chamber surfaces, in which case the flame quenching process is affected. In addition the geometry of the spark plug can lead to variations in flame kernel development due to convection. If the kernel travels toward the spark plug ground electrode, heat transfer to the electrode increases, which slows the kernel development, and lengthens the initial combustion phase. As discussed later, the length of the flame kernel development phase of combustion is closely correlated to CTCV.

The impact of bulk engine flows on CTCV can be studied in some detail by varying engine geometry and simultaneously measuring velocities, normally by LDV, and the CTCV of the engine. LeCoz (1992) followed this approach in changing the swirl intensity by varying the valve lift of one of the valves of a four valve engine head. This study investigated the effects of both turbulence and bulk flow on engine combustion by
measuring CTCV as a function of the ensemble averaged velocity at the spark plug gap for various engine flow geometries. The major result of this work is that the large scale structures appear to have an impact on combustion stability only in mixtures which are significantly lean of stoichiometric. Large CTCV in the lean operating regime correlated well with bulk velocity at the spark plug. This is probably due to the convection of the flame towards surfaces which lead to early quenching of the flame, which is particularly important in the slow burning lean mixtures. Overall, though, LeCoz found that neither large scale nor small scale structures explained a majority of the CTCV found in the test engine. In all but the leanest operating regimes, the combined effect of both small and large scale velocity fluctuations was less than 40% of the total CTCV. Thus while in-cylinder flows are important to understanding CTCV, they are clearly not the only root cause.

High frequency velocity fluctuations, or small scale turbulence, is known to be an important parameter in the combustion process. This is true for several reasons. Small scale turbulence enhances the mixing process, leading to a more homogeneous charge. Apart from this relation, though, small scale turbulence also has a large impact on the initiation and propagation of the flame front in internal combustion engines. Early work, in which the turbulence intensity in a combustion system was correlated to burn rates (Rashidi, 1981), postulated that high levels of turbulence in the flame initiation region gives rise to high CTCV. This work though, was not conducted in an actual engine, and the results are difficult to extend into that regime. The conclusion of the author is that a reduction in small scale turbulence in the area of the spark plug would be beneficial. The error in this logic is that in an engine flow, some level of turbulence is going to be present, and that while flame initiation in laminar flow is more consistent, once in the turbulent regime, less turbulence is not necessarily better. This assertion, that additional turbulence aides combustion stability, is supported by the bulk of engine literature.

The impact of small scale turbulence on combustion stability has been studied quite extensively. A very complete study was conducted by Hill (1988) and Hill and Kapil (1989) in which experimental measures of turbulence and CTCV were acquired in
conjunction with an analysis of the turbulence scales and intensity. The author utilizes
evidence from schlieren flame kernel images to postulate that turbulence in engines tends
to be homogeneous, isotropic, and the characteristic turbulence structures have a size
significantly smaller than the spark plug gap. The Tennekes model of small scale
turbulence is used to predict the theoretical standard deviation of the ignition delay which
should result from the measured turbulence intensity and integral length scale. While the
predicted mean ignition delay follows the general trend of the data, it does not provide a
quantitative match, indicating other factors as major causes of CTCV.

Utilizing schlieren photography and LDV, Hall (1989) determined that the main
influence of small scale turbulence on combustion, especially during the initiation phase
was to cause an increase in flame front area by wrinkling. This wrinkling accelerates the
flame kernel growth process. The very early stages of kernel development have been
shown to proceed at a rate related to the laminar flame speed of the local mixture
(Tagalian and Heywood, 1986), but this wrinkling effect shortens the kernel development
phase, which is very well correlated with a reduction in CTCV.

Matthews et al (1991) studied the effect of turbulence intensity on flame
propagation in a rapid bum engine. Applying a fractal geometry treatment of turbulence,
the authors conclude that the main effect of turbulence in spark ignition engine
combustion is to stretch the flame and increase the surface area of the flame. This is the
flame wrinkling discussed previously. A statistical analysis of experimental data led to
the conclusion that equivalence ratio has a dominant affect on peak cylinder pressure, but
turbulence intensity dominates the determination of the crank angle location of maximum
pressure. Both of these parameters contribute to the indicated mean effective pressure in
any given cycle.

In summary, there are two major effects of the in-cylinder flow field on CTCV
besides its contribution to mixing. Bulk flows lead to the convection of the developing
flame kernel away from the spark plug gap which can lead to cyclic variations due to heat
transfer to spark plug electrodes or flame quenching at cylinder walls. Small scale
turbulence tends to shorten the burn duration, especially in the early flame development
region, which is known to reduce the degree of CTCV in spark ignition engines. Any experiment which attempts to measure the cyclic variability caused by factors other than the engine flow field must consider the CTCV inherent in engine operation due to small scale turbulence.

2.2.2 Charge Composition and CTCV

The constitution of the mixture trapped in the cylinder during a particular cycle clearly has a very strong influence on the heat release and pressure production which occur during the combustion process. The cylinder charge is basically made up of three constituents, air, fuel and residual gases from the previous cycle. The amount of both fuel and air is controlled by the intake geometry and pressure, as well as the residual gas remaining from the previous cycle. This leads to a complicated process in which it is difficult to maintain constant cylinder charging even in gaseous fuel systems. The addition of liquid fueling dynamics further complicates the situation and leads to variations in not only overall cylinder mass but relative air/fuel ratio as well.

The amount of fuel and residual gases are known to vary significantly, and thus have an impact on the state of the mixture during combustion, as well as the total amount of heat released in a given cycle. The first investigations into cyclic variations in engines focused on this cause (Yu, 1963). Until the late 1970’s, fuel metering was performed largely by carburetors in automotive internal combustion engine applications. While carburetors are relatively simple devices, they have several drawbacks for fuel metering. These are that they have difficulty accurately metering fuel all along the speed/torque curve of a given engine, and they do a poor job of distributing fuel between individual engine cylinders. While single point injection systems improved the fuel metering problem, at least during steady operation, they did little to correct the distribution problem. In Yu’s study, the air/fuel ratio in each individual cylinder was measured on a cycle resolved basis. As one would expect with a carbureted system, the air/fuel ratio varied not only from cylinder to cylinder but from cycle to cycle. The prescribed solution for this at the time was to operate in an overall rich of stoichiometric regime, thus
preventing lean cyclic variations in any cylinder. Clearly this is not a feasible solution due to the increased emissions and decreased fuel economy associated with rich operation. The maldistribution of fuel between cylinders and cycles was determined to result in power and fuel economy losses on the order of 5% at stoichiometric operation. Even utilizing a gaseous fuel which was essentially completely mixed, the variations between individual cycles did not decrease significantly.

The impact of both total fuel ingested and residual gas concentration was investigated by Sztenderowics and Heywood (1990). The total mass of fuel burned was calculated from cylinder pressure data, and was found to vary substantially at low load and speed, where CTCV is typically most severe. On a cycle resolved basis, the mass fraction burned correlated quite well with the IMEP for the given cycle. The total mass fraction burned varied most substantially (up to about ±5%) at light load and under continuous firing conditions. By exhaust gas analysis, the variation in total fuel burned was determined not to be due to incomplete combustion but to varying amounts of fuel trapped per cycle. While the authors provided several suggestions of possible causes of this fluctuation, none were experimentally tested. The fuel in this case was well mixed propane, suggesting that either the total mass of charge trapped in the cylinder varied from cycle to cycle, or unburned fuel remained from previous cycles to be burned in the next. The fact that cyclic variations decreased but were not eliminated when running in skip fired mode lends credence to the idea that it is some combination of these two phenomena which lead to the variation in total fuel burned per cycle.

Utilizing in-cylinder measurement of charge constituent concentrations by Raman spectroscopy, Gruenfeld et al (1994) concluded that a major cause of CTCV is the variation of fuel and residual gas trapped per cycle. The Raman technique allows the measurement of fuel concentration as well as oxygen and nitrogen, thus the relative air/fuel ratio and the residual gas fraction can be determined. By varying the fuel injection timing and measuring air/fuel ratio and residual gas content, along with combustion pressure, a strong correlation was found between the standard deviations of the given parameters. The main shortcoming of this study is the lack of spatial resolution.
Because the Raman measurements are the result of integration over the width of the combustion chamber, in the diameter of the laser beam, the local, not the overall, air/fuel ratio and residual gas fraction are found. For this reason, the results of this study are just as likely to be attributable to mixture inhomogeneity as to cyclic variations in overall air/fuel ratio.

2.2.3 Ignition Characteristics and CTCV

Variability in the ignition event has the potential to cause fluctuations in the overall combustion event. The phasing of the ignition event with respect to crank angle is of the utmost importance in maintaining a consistent level of IMEP. The variation in this phasing, known as spark jitter, may have in the past been a major cause of CTCV. With modern electronic ignition systems, though, the timing of the ignition event is essentially constant. Other possible sources of variation in the ignition process are the total amount of energy released to the gas, and the variation in release rate during the spark event. While increasing total spark energy and spark duration have been shown to reduce CTCV, little information exists as to the extent to which these parameters vary during operation with a given ignition system at a set operating point.

2.2.4 Mixture Inhomogeneity and CTCV

The variation of combustion related parameters due to the incomplete mixing of air and fuel prior to combustion has been the main focus of this work. The impact of fuel distribution on CTCV is of particular importance for several reasons. In gasoline engines, mixture distribution has been shown to have a major impact on CTCV, especially in multi-point fuel injection systems. Due to the inherent advantages which these systems offer in terms of fuel efficiency and emission reduction, especially during engine transients, their use will necessarily increase. In the case of natural gas engines, the use of multi-point injection systems is in its infancy, and in a similar way to gasoline, has the ability to enhance engine efficiency. While multi-point injection systems provide much more accurate fuel metering, they lead to especially poor mixture formation. This is not
necessarily the case in gasoline intake systems where multi-phase flow dominates, but in natural gas systems they present serious problems in terms of mixture distribution.

Natural gas jets have lower momentum in comparison with liquid fuel jets. Thus, despite the increased mixing area which is inherent in underexpanded, high pressure jets, the overall rate of mixing is significantly lower than in liquid jets. It has been demonstrated both analytically and experimentally, that the rate of mixing in gas jets is greatly reduced from that of liquid jets (Abraham et al, 1994). As discussed later, the importance of understanding CTCV, and especially it’s relation to mixture distribution, in the case of natural gas engines is particularly important to their eventual success. To date, there have been no investigations of CTCV in multi-point injected, natural gas fueled engines.

The main reason that mixture non-uniformity affects combustion variability is the variation of flame speed with relative air/fuel ratio. For most fuels, the rate of propagation of a flame, either laminar or turbulent, is maximum just rich of stoichiometric, and decreases rapidly as the mixture becomes lean. Natural gas has a lower flame speed and greater flammability limits than gasoline. Due to it’s greater flammability limits, it is advantageous to operate natural gas engines at lean of stoichiometric conditions resulting in further depression of the flame propagation speed. The result of this extension is a greater propensity for high levels of CTCV.

This relation between flame speed and combustion stability is demonstrated well in a study by Heywood and Vilchis (1984) comparing the operation of an internal combustion engine on propane with one operated on hydrogen. Hydrogen has a laminar flame speed approximately eight times that of propane, with a corresponding difference in turbulent flame speed. Schlieren photographs were utilized to study the flame propagation process with these two fuels in a transparent square piston engine. The hydrogen flame was found to be both more spherical and less irregular in outline than the propane flame. More importantly, the shape and size of the hydrogen flame were much more consistent from cycle to cycle than the propane flame. The early expansion phase of combustion, that is the flame kernel development, was found to proceed at essentially the
laminar flame speed. The conclusion drawn from this work was that slower flame speeds which extend the flame kernel formation phase of combustion produce significantly higher levels of CTCV.

In a complementary study by Keck et al (1987), the early flame development period was studied in detail using schlieren photography in an internal combustion engine. Flame front imaging was used both to determine the nature of the flame and the initial flame speed. The nature of the flames in this study were determined to be thin, highly wrinkled flames in which the degree of stretching was very high. While flame propagation in the main phase of combustion was found to be dependent mainly on the intensity of turbulence and the spherical burning area, development of the flame kernel was determined mainly by the laminar flame speed and the size of the first eddy burned. The result of this is the correlation between low laminar flame speeds and high degrees of CTCV. In addition, the majority of cyclic variation in combustion was found to occur during the flame initiation period. Once the flame kernel growth period is concluded, flame propagation is a fairly regular event.

The main conclusion which can be drawn from these investigations is that any condition which decreases the laminar flame speed in the region of the spark plug during ignition will cause increased levels of CTCV. As the flame initiation phase is extended, parameters such as in-cylinder flow and ignition characteristics have more time to influence the delicate balance of heat release, heat transfer, and flame convection which are occurring simultaneously. Laminar flame speed is decreased by several factors, including the concentration of residual gases, the type of fuel being used, and perhaps most importantly, the distribution of fuel within the cylinder. Poor fuel distribution results in spatially distinct rich and lean regions. If the spark plug volume contains a rich region at the time of ignition, the flame speed is high, resulting in rapid flame development with earlier and higher peak pressure. On the other hand, if a lean mixture occupies the spark plug volume, the flame kernel development is slow, leading to slower pressure rise to a lower absolute maximum pressure. For this reason, engines which
operate with poorly distributed air/fuel mixtures are more likely to suffer from extreme CTCV than those operating on truly homogeneous mixtures.

Several experimental studies have investigated this phenomenon with varying degrees of accuracy. An early study (Pundir et al, 1981), measured the CTCV of a gasoline engine operating with different intake configurations. In this study, the degree of mixture inhomogeneity was not measured, but was assumed to vary with the geometry of the injection system. Using the method described by Eltinge (1968), the authors determined the standard deviation of the mixture homogeneity from the exhaust gas composition. This method is discussed in Chapter 3. The coefficient of variation of the maximum cylinder pressure was used as the determinant of CTCV. Using the Eltinge method, the degree of maldistribution was determined as a function of intake geometry. The result of this study was that the covariance of the maximum cylinder pressure increased linearly with standard deviation of spatial air/fuel ratio. While predicting the general trend, the fact that the mixture maldistribution was not physically measured should be taken into account.

A more advanced study was carried out by Hamai et al (1986), which utilized a fast sampling valve to measure the mixture strength near the spark plug at ignition and simultaneously measure the pressure evolution for the given cycle. In this case, the heat release delay and indicated mean effective pressure were taken as the measures of CTCV. An extremely high correlation was found amongst the three parameters, variation of air/fuel ratio, variation in heat release delay, and variation in indicated mean effective pressure. All the correlation coefficients were between 0.85 and 0.90, indicating that the local air fuel ratio determined the heat release delay which in turn determined the indicated mean effective pressure of the given cycle. Overall, the local air/fuel ratio was found to fluctuate by ±1.7 air/fuel ratios in this port injected engine. Clearly, fluctuations of this size have a significant impact on the laminar flame speed, and thus the duration of flame kernel development.

In a study similar to that of Pundir et al, Sztenderowicz and Heywood (1990) attempted to produce different levels of mixture uniformity by injecting liquid fuel in the
inlet port, and upstream of the inlet port. The overall mixture nonuniformity was estimated by the Eltinge method, once again, and was found to vary with injection location as before. In this study though, no correlation was found between the degree of non-uniformity and the variation in IMEP. The only apparent differences between the studies are the engine utilized and the operating condition. In the later study, the engine was operated at the stoichiometric air/fuel ratio, and at high load. The earlier study was conducted at a lower load. The main impact of load in this case is to change the residual gas fraction. Lower loads tend to have higher levels of residual gas, due to lower intake manifold pressure. As with lean operation, high levels of residual gas tend to decrease laminar flame speed. These observations tend to suggest that there may be some limit below which the impact of laminar flame speed becomes particularly important.

A work by Cho and Santavicca (1993) investigated the impact of mixture non-uniformity on combustion. Using both laser induced fluorescence and schlieren, the impact of local mixture strength at the spark plug on flame development was studied. In this case cyclically resolved data was not available, but statistical measures of the variation of flame kernel radius at a given time with the standard deviation of spatial mixture distribution was used. Differing levels of mixture homogeneity were produced with injection of gasoline at given locations upstream of the inlet port. Laser induced fluorescence measurements of the local mixture strength confirmed the increase in homogeneity with increasing distance between the injector and the inlet port. As expected the average flame kernel radius decreases as the mixture becomes less uniform. In addition, the standard deviation of the flame kernel radius also increases continuously with increasing levels of inhomogeneity. The influence of the degree of incomplete mixing on misfire rate was also studied. The misfire rate was found to be nearly constant up to a certain level of mixture maldistribution, but after this point the number of misfires increased dramatically. The increase was from 2% misfire at a mixture non-uniformity of 24% RMS to a misfire rate of 23% at a mixture non-uniformity of 33% RMS. This once again lends credence to the concept of a laminar flame speed threshold for consistent combustion.
While there is a significant body of evidence relating mixture maldistribution to CTCV, there remains a number of pertinent points yet to be addressed. Foremost among these in this study is the influence of fuel type, that is the use of natural gas, on the correlation between local mixture strength and CTCV. Clearly, the impact of fuel maldistribution on combustion quality will vary with the overall air/fuel ratio, and the load and speed of the engine. In addition, little research is available which includes direct measurement of mixture strength and combustion parameters. The purpose of this work has been to quantify the impact of mixture inhomogeneity on cyclic combustion variations through direct measurement of each, in a natural gas fueled engine.

2.3 Parameters Influencing the Degree of CTCV

The causes of CTCV discussed above are influenced to varying degrees by a number of engine operating parameters. In order to determine the specific impact of mixture nonuniformity on CTCV, the effects of these parameters must be understood and controlled. In each case, the parameter of interest has some affect on one of the root causes. Factors which influence the cylinder constituents, or in-cylinder flow, or ignition characteristics must be limited to extract information on mixture distribution alone.

The intake geometry of the engine has an influence on in-cylinder flow and mixture distribution. In gasoline fueled engines the intake geometry also governs the two phase flow of fuel into the cylinder. In gas powered engines this impact is reduced as there is no wall film or fuel droplets. Varying intake configurations, including valve profiles, valve timing, runner length, etc. have a profound affect on the bulk flow in the engine at any given operating condition. As discussed earlier, the bulk flow, especially tumble, determine the turbulence intensity as they are broken down late in the compression stroke. Naturally, the intake and exhaust geometry also affect the total volumetric efficiency of the engine, and the residual gas fraction. Each of these factors has been controlled to eliminate their effects on CTCV in the given research. The intake and exhaust geometry are fixed for the given engine and do not play an important role.
Of course the flow field in the engine varies from one cycle to the next due to the nature of turbulence.

The method of fuel preparation in an engine has a large impact on both mixture homogeneity and the overall air/fuel ratio for a given cycle. Multi-point injection systems, such as the system utilized in this research provide for very consistent overall air/fuel ratio, but also result in poor mixture homogeneity. The degree of inhomogeneity is determined largely by the timing and duration of the injection. In gasoline engines, the fuel is injected prior to the opening of the intake valve, leading to increased vaporization and improved fuel distribution. Because this vaporization process is not necessary in gaseous fueled engines, this injection timing is not optimum. The ideal injection event would probably be a continuous injection during the intake stroke, so that as air passes the injector, it could entrain the injected fuel. Unfortunately, the injection duration is determined by the maximum required flow rate at full load and speed. The result at idle is a 5 ms injection period during a 45 ms intake stroke. One application of this research is the determination of the injection timing which produces the best mixture distribution.

The ignition system used in a given engine has a definite impact on the CTCV of the engine. Several ignition system parameters have been shown to influence cyclic variability of the early combustion phase. Foremost among these are the spark energy and spark plug geometry. The most important factors in spark plug geometry are the plug gap and electrode shape. Increasing the spark plug gap has been shown to decrease CTCV in light load, lean burn applications. The lean limit in several test engines has been extended by increasing the spark gap significantly. It is probable that this is a result of inclusion of a greater volume, some of which may contain a richer mixture. In addition, increasing the gap increases the kernel volume to electrode surface area ratio, which decreases the heat transfer from the kernel. Unfortunately, large gaps are not practical in production engines as they do not function well over the entire operating range of the engine. In addition, larger gaps require larger breakdown voltages which are not available in current automobile ignition systems. The geometry of the spark electrodes has an impact on cyclic variability of the ignition process due to heat transfer.
Smaller spark electrode have been shown to reduce CTCV by increasing the amount of heat retained by the flame kernel. This retention of heat produces a more stable kernel which speeds flame initiation even in leaner mixtures. Once again, though, the smaller geometry spark plugs are not suitable for production use due to durability problems.

The final ignition characteristic which influences CTCV is the location of the ground electrode with respect to the combustion chamber. Depending on the bulk flow velocities, certain electrode geometries produce lower levels of CTCV than others. This is due to flame kernel convection with the bulk flow. If the kernel is convected towards the ground electrode, heat transfer is increased and flame stability suffers. For this reason, the electrode position must be controlled in order to isolate the impact of mixture distribution on CTCV.

2.4 Significance of Fuel Distribution in Natural Gas Engines

There are several factors which afford particular importance to gaining an understanding of CTCV due to mixture maldistribution in natural gas engines. The application of multi-point fuel injection systems to natural gas engines is a requirement if these engines are going to compete with traditional gasoline powered engines. The performance of natural gas engines must be improved by means of increased compression ratio due to the deflated volumetric efficiency caused by the displacement of air by natural gas in the intake system. Natural gas engines operate most cleanly and efficiently well lean of a stoichiometric air/fuel ratio. Lastly, natural gas is a prime candidate for stratified charge lean burn engines.

Multi-point fuel injection systems have vastly improved the efficiency and emission levels of gasoline powered vehicles. This can be attributed to accuracy of fuel metering, superior performance during transient operation, and better mixture distribution. While natural gas engines can definitely benefit from the improved fuel metering and transient advantages of multi-point injection systems, the mixture distribution situation differs significantly for gaseous fuels. As discussed earlier, gas jets do not provide the degree of mixing afforded liquid jets. For this reason, mixture quality
will decrease with a change from single-point to multi-point injection systems. In order to overcome this potential pitfall, the impact that this increased mixture maldistribution will have on combustion stability must be understood.

The volumetric efficiency of natural gas engines is traditionally lower than that of similar gasoline engines due to the gaseous nature of the fuel. Injecting a gas into the inlet port of a throttled engine reduces the partial pressure of air in the inlet. This reduction in partial pressure lowers the total mass flow rate of air at any given operating point. The result of this reduction is lower fuel efficiency and peak torque. Fortunately, natural gas has the advantage of having a very high octane rating. This allows natural gas engines to utilize higher compression ratios compared to gasoline engines, without the risk of knock. Higher compression ratios, though, lend themselves to higher cyclic fluctuations for two reasons. As the compression ratio rises, the pressure in the cylinder at the point of ignition rises as well. The laminar flame speed of most hydrocarbons decreases with increasing pressure. Methane, the main constituent of natural gas, is particularly prone to this problem. As discussed earlier, decreases in laminar flame speed typically increase the flame kernel formation duration, and thus negatively impact CTCV. Regardless of fuel, CTCV becomes more important in high compression ratio engines, because the fastest burning cycles tend to create auto-ignition in the end gas region as discussed earlier. The greater the extent of CTCV, the more likely is the existence of cycles which burn quickly enough to result in knock in a given engine. In order to prevent these phenomena and allow the elevated compression ratios which provide higher efficiencies for natural gas engines, the causes of CTCV, especially mixture inhomogeneity, must be better understood.

Natural gas is nearly an ideal candidate for application to the concept of lean burn, stratified charge engine technology. The extended flammability limits, and high octane number of natural gas allow lean burn engines to operate lean enough as to bypass the peak of the NO production curve, and reduce NO emission without a catalyst. This ability to operate in the lean regime is important in natural gas engines as the very low levels of CO in the engine exhaust make the application of a three-way catalyst more
difficult. An engine which could operate with a wide enough range of air/fuel ratios could eliminate the need for a throttle and govern engine output with air/fuel ratio only. In order to accomplish this, the lean limit of the engine must be extended well past that which is currently available in engines. The lean limit is highly dependent on the local air/fuel ratio near the spark plug at the time of ignition. While the lean limit of homogenous charge natural gas engines is impressive, it must be extended further to allow throttleless operation. The stratified charge engine is one means by which this could be accomplished. In a stratified charge engine, fuel/air mixing is inhibited in some manner during the intake and compression strokes. This can be accomplished either by direct injection of the fuel into the cylinder, or the stratification of the intake charge by control of in-cylinder flow. This last method has been accomplished recently by injecting fuel into a single port of a dual port cylinder head. By designing the intake geometry such that strong tumble is created without a large swirl component, the fuel rich and fuel lean areas remain quite stratified during compression. In this way, a mixture strength which is richer than the overall air/fuel ratio can be obtained at the spark plug location. Clearly, for this scenario to prove feasible over the entire operating range, a greater understanding must be obtained of the impact of local mixture strength on flame initiation and CTCV.
CHAPTER 3

EXPERIMENTAL METHODS FOR FUEL DISTRIBUTION MEASUREMENT

3.1 Introduction

In order to determine a quantitative relationship between mixture homogeneity and CTCV, the direct measurement of both fuel distribution and combustion quality is required. The topic of this chapter is the determination of the first of these, the in-cylinder fuel distribution. The methods by which mixture homogeneity can be measured include both optical and non-optical techniques. While the research described in this work utilized planar laser-induced fluorescence, a brief background of other possible measurement methods is given. The purpose of these summaries is to provide insight into both the techniques that have been used in the past, and the results of the research conducted using these methods. In addition to summaries of methods which have been utilized to this point, a more detailed description is given of planar laser induced fluorescence (PLIF). This description includes sufficient theoretical detail to demonstrate the applicability of PLIF to in-cylinder mixture measurement and to emphasize the technical challenges that were overcome to determine quantitatively the relative air/fuel ratio on a spatially resolved basis from PLIF images. Finally, an overview is given of the studies which have been conducted using PLIF to determine mixture homogeneity in gasoline fueled internal combustion engines.

3.2 Mixture Distribution Measurement by Gas Sampling

The oldest, and perhaps most straightforward, method of determining mixture distribution quality is the direct sampling of in-cylinder gases. Several methods have
been employed which provide insight into the spatial distribution of fuel within the combustion chamber at various times during the intake, compression, and combustion strokes. While these methods have limited utility for cycle resolved, quantitative measurements, they can provide an overall qualitative measurement of the degree of mixing which is produced by a particular engine as a function of operating conditions.

3.2.1 In-Cylinder Sampling Techniques

One method that has been utilized by engine researchers for the determination of in-cylinder mixture distribution is the use of a gas sampling valve for the direct measurement of gas concentrations. This technique utilizes a small probe which contains a sampling valve that operates sufficiently fast as to collect a small volume of gas at a given time during the engine cycle. This captured gas is then analyzed by one of several methods to determine the concentrations of its various constituents. The main advantage of this type of system is its relative simplicity. Gas sampling valves can be adapted to most any engine geometry without major engine modification. While the opening and closing of the valve must be very fast, modern electronics make the timing and control of this event quite simple.

Unfortunately, gas sampling also has several inherent disadvantages. Among the disadvantages are the intrusion of the probe, the accuracy of the concentration measurements, and the lack of spatial resolution. The major area of interest for concentration measurements is typically near the spark plug. Inserting a probe in this area not only disturbs the fluid flow in the vicinity of the spark plug, but also provides another surface for heat transfer from the developing flame kernel. As discussed previously, both of these factors impact the early flame development period of combustion, and thus have a large influence on the degree of CTCV. In addition to being an intrusive technique, the measurements achieved through gas sampling can be erroneous for several reasons. The most obvious of these is the reaction of the gases after sampling. The cylinder charge is a highly volatile mixture, and reactions which occur between the time the mixture is sampled and the time the sample is analyzed can lead to inaccurate results. Finally, gas
sampling valves only have the ability to measure the gas concentration in a small volume, and typically provide average gas concentrations in this volume. This lack of spatial resolution makes determination of the overall degree of mixture homogeneity very difficult. For these reasons, gas sampling valves were not considered an appropriate means of determining fuel distribution for this research. Descriptions of several of the valve sampling techniques and their results are provided below in order to provide an understanding of the application of gas sampling valves to mixture distribution measurement.

The first study of the impact of fuel distribution on CTCV (Yu, 1963), utilized a fast response gas sampling valve developed specifically for that research. In this study, an electronically triggered valve opened at a specified crank angle for 2.4 ms during each engine cycle. In order to separate the gas samples taken in individual cycles, a small amount of helium was introduced into the line between each collection. The gas samples were analyzed by means of a gas chromatograph to determine the concentration of CO, CO$_2$, N$_2$, and O$_2$. From these concentrations, the air/fuel ratio, as well as the residual gas fraction can be determined. Despite the fact that isooctane supplied through a carburetor was the fuel, the mixture was assumed to be uniform in the cylinder. Thus measurements were interpreted as determining the variation in overall air/fuel ratio in a given cylinder from cycle to cycle, that is, the maldistribution of fuel between cylinders. Knowing that the cylinder charge in carburetated engines is stratified leads to the conclusion that the gas sampling valve, while measuring overall air/fuel ratio to some degree, is also measuring the charge stratification. The main result of interest in this study is that air/fuel ratios at the spark plug probe were found to vary randomly from 16.8 to 19.5. This is a very large range of air/fuel ratios which can clearly lead to high levels of CTCV as described previously.

In a more recent work (Hamai et al, 1986) a similar sampling valve was used with a thermal conductivity sensor to determine the concentrations of CO, CO$_2$, HC and NO. The sampling valve in this study was open from 13 degree BTDC to 19 degrees ATDC, thus collecting gases during a large portion of the combustion phase. The air/fuel ratio of
the sample was determined from the gas concentration measurements, although the authors did not specify the technique. The weaknesses inherent to sampling valves are clear in this study for several reasons. The long sampling time required to acquire appropriate sample sizes gives very poor temporal resolution. That is, the concentrations of the sampled gases changes tremendously during the time of sampling, making interpretation of the concentrations difficult. In addition, the single measurement location does not allow for an overall quantification of the mixture homogeneity. As in the previous study, though, the variation in air/fuel ratio was quite large, 16.2 to 19.6, confirming that significant levels of inhomogeneity exist up to and during the combustion process.

A more accurate means of determining local mixture strength is the flame ionization hydrocarbon detector. This system has been used in several recent studies (Sleightholme, 1990 and Galliot et al, 1990) to determine the extent of maldistribution of fuel in the cylinder charge. In a flame ionization detector (FID), a small stream of gases is drawn from the cylinder on a continuous basis. The gases are burned, and a detector produces a signal proportional to the number of carbon atoms in the sample. From the continuous FID traces, the number density of the fuel can be determined. This number density is then converted to relative air/fuel ratio by correcting for cylinder pressure. The main advantage of the flame ionization detector is its speed of response, on the order of milliseconds. This allows the collection of a continuous trace of the concentration throughout the engine cycle. The trace is of course delayed by the response time from the actual concentration. Flame ionization detectors have been used to measure both air/fuel ratio and residual gas fraction. Galliot et al (1990) measured the residual gas fraction that results in motored cycles following a single combusting cycle. The number of cycles required to exhaust the majority of residual gas is of interest in the this research and is discussed in chapter 5. Sleighthome (1990) used an FID to correlate charge inhomogeneity with CTCV. The author found that at lean operating conditions, the standard deviation of the FID signal was proportional to the standard deviation of the engine IMEP. Unfortunately, as discussed previously, the FID is unable to distinguish
between variations in overall air/fuel ratio, and mixture homogeneity, leading to difficulty in the interpretation of the results.

Another iteration of the sampling probe concept is the catalytic hot wire nozzle probe. In this method, a small amount of the cylinder gases are drawn over a heated wire which causes reaction in the gases. In a similar way to a hot wire anemometer, the heat released from the reaction of the gases is determined from the current required to maintain the catalytic wire at constant temperature. This instrument has been demonstrated (Li et al, 1994) as a means to measure in-cylinder mixture strength. Due mainly to the variation of probe signal with both temperature and concentrations, though, the margin of error from these measurements seems to be quite high. While the probe can measure the increase in homogeneity with crank angle, it is unable to make quantitative mixture strength measurements. The probe, of course, also suffers from the weaknesses of other sampling probes. That is their intrusion and lack of spatial and temporal resolution make them relatively useless for the quantitative determination of the homogeneity of cylinder charge on a cycle resolved basis.

3.2.2 Exhaust Gas Sampling

In addition to the in-cylinder sampling processes described above, there is a method of determining qualitatively the in cylinder mixture distribution from exhaust gas sampling. This method is used extensively mainly due to its convenience, which stems from the fact that no engine modifications are required. The process, developed by Eltinge (1968) utilizes the established influence of air/fuel ratio on exhaust emissions to estimate the degree of inhomogeneity which exists in the cylinder during combustion. The first step in the process is to assume that the cylinder volume is broken down into a large number of small control volumes, each having a distinct air/fuel ratio. For each control volume with known air/fuel ratio, the expected emissions of CO, CO₂ and O₂ can be determined as a function of only the air/fuel ratio. These emissions are calculated by assuming complete combustion of fuel to oxygen and water in the case of excess air, and a balance between H₂, CO, H₂O, and CO₂ by means of the water gas reaction model in
the case of fuel rich operation. Given these assumptions, the total measured emissions from the engine would be a summation over all of the small control volumes. Making the further assumption that the distribution of air/fuel ratios within the control volumes is Gaussian in nature, the standard deviation of the distribution can be found. This standard deviation of the air/fuel ratio distribution is taken as a measure of the mixture homogeneity.

There are several weaknesses to the above described method. While the assumption of equilibrium concentrations of emission constituents is accurate to the necessary degree, there is no reason to believe that the distribution of air/fuel ratio in the individual elements should be Gaussian in nature. For example, in liquid fuel systems, if droplets exist, the distribution will have wildly varying fuel concentrations in some of the control volumes leading to a very misleading standard deviation. Despite the drawbacks of this method, several experimental tests have demonstrated its use as a qualitative measure of mixture homogeneity. For instance, the lean limit in a gasoline powered engine was found to be greatest when the standard deviation of the distribution was minimized. Thus, while the Eltinge method can provide insight into the effectiveness of a given intake configuration in creating a uniform mixture, it is of little use in determining the relationship between CTCV and mixture homogeneity on a cycle resolved basis.

3.3 Optical Techniques for Mixture Distribution Measurement

The evolution of high powered lasers and sensitive photodetector arrays in the last two decades have led to the development of many new optical techniques for the measurement of mixture distribution in internal combustion engines. These techniques rely upon the scattering, emission, absorption or absorption and reemission of light by molecules either present in the flow or added specifically for the technique. The main advantage of optical techniques over the sampling probes described earlier is their non-intrusive nature. Optical techniques offer the possibility of determining the mixture concentration at a point or in a plane of the flow without disturbance of the flow itself.
The main obstacle which must be overcome in making optical concentration diagnostics possible is the need for optical access to the measurement volume. This need for optical access requires a compromise between accuracy in replicating actual engine operation and geometry conducive to optical measurements. The sections that follow cover the optical techniques which have to this point been utilized to make measurements of the mixture homogeneity in motored or firing internal combustion engines.

3.3.1 Raman Spectroscopy

Raman spectroscopy has been demonstrated as a possible means for measuring the concentration of fuel, air, and residual gases along a measurement line in an internal combustion engine (Sawerysyn and Heywood, 1986 and Grunefeld et al, 1994). These techniques utilize Raman scattering off molecules present in the flow. When laser light impinges on diatomic and triatomic molecules within a given flow, a small portion of the light is scattered inelastically at a wavelength which is longer than the incident light. Because the wavelength of the scattered light is a function of the allowed rotational and vibrational transitions within the particular molecule, the concentration of a given species can be determined by measuring the intensity of light scattered from the gas as a function of wavelength. In combusting systems, measurements are typically made of $N_2$, $O_2$, $H_2O$, and fuel. The degree to which the incident light is shifted during scattering is distinct to each of these constituents and can be determined from both experiment and theory.

While Raman scattering offers the possibility of measuring the air/fuel ratio in engines, there are several issues which make this measurement difficult. The primary difficulty in utilizing Raman scattering in engines is the small magnitude of the Raman scattering cross section. Even very intense lasers do not produce signal levels of sufficient magnitude to make planar measurements. The result of these small cross sections is that in the best case, Raman measurements can only be made at a single point, or along a line. Even in this limited case, the signal to noise ratio is low, making quantitative determination of air/fuel ratio difficult.
Despite the difficulties described above, several researchers have attempted to make concentration measurements in engines utilizing Raman spectroscopy. Sawersyn and Heywood (1986) utilized a Q-switched, frequency doubled YAG laser (532 nm) to provide 500 mJ per pulse for Raman scattering off the major constituents in the combustion chamber of a commercial engine. Multichannel spectroscopy allowed for the measurement of the concentrations of several species at once along a line determined by the laser beam. As is typical with engines used for optical techniques, the geometry of the combustion chamber was altered significantly. The intake and exhaust valves were moved to the cylinder walls to allow the installation of an overhead viewing window. Windows were also inserted into the cylinder walls to allow passage of the laser beam. With these modifications, the compression ratio of the engine was lowered to 4.4:1, clearly not a realistic operating point. The main result of this research was to demonstrate that the presence of the major species could be determined during engine operation. The quantitative concentrations of the species, on the other hand, could not be determined due to the low signal to noise ratio.

A more recent study conducted by Grunefeld et al (1994) averaged the Raman signal over the length of the laser beam in order to get an overall measurement of mixture strength. This resulted in considerably larger scattering signals but sacrificed any form of spatial resolution. In this study, the laser beam, a high power pulsed UV beam, was brought in through a window in the cylinder wall and exited through a similar window at the opposite side of the cylinder. The backscattered Raman signal was collected through the entrance window and separated from the beam by means of a dialectically coated mirror. In this configuration, the scattering from the entire length of the beam is collected at a single point. By monitoring the concentration of O₂, N₂ and fuel, the average relative air/fuel ratio over the length of the beam could be determined. Collecting the scattering from both the fuel and air eliminates the influence of laser power on the concentration measurement. The authors estimated an accuracy within 4% of the actual concentration. Cylinder pressure was measured simultaneously with the collected Raman scattering and some correlation was found between IMEP and the overall air/fuel ratio in the beam. The
results of this study are difficult to extrapolate to a relation between CTCV in production engines and mixture homogeneity because of the altered geometry of the engine, and the lack of spatial resolution of the measurement.

3.3.2 Rayleigh Scattering

As opposed to the Raman scattering described above, Rayleigh scattering is the elastic scattering of light off of molecules in a gas. The major differences between Rayleigh and Raman scattering are that there is only a very slight Doppler shift in wavelength in Rayleigh scattering, and the scattering cross sections are typically three orders of magnitude larger. These differences offer both advantages and disadvantages. Because the scattering cross sections are much larger, the signal levels are higher and planar measurements are possible. The lack of wavelength dependence of the scattered light, though, presents several problems. Because the scattered and incident light are at the same wavelength, scattering from walls, windows and other optics can not be distinguished from scattered light from gas molecules. This significantly lowers the signal to noise ratio in complex geometries such as an engine combustion chamber where many reflecting surfaces are present.

The elastic nature of the scattering makes it impossible to distinguish between species. While each constituent of the gas has a different scattering cross section, the wavelength of the scattered light is constant. In two component mixtures, or mixtures which have one constituent whose scattering dominates the other constituents, the determination of absolute concentration is possible. In this case, the signal produced by the minor constituents or smaller cross section molecules can be subtracted by comparison with a second image taken without the major constituent. In engine flows this can become complicated for several reasons. Most importantly, depending on the fuel, the scattering cross section of the fuel may not dominate that of background air. The limiting feature of Rayleigh scattering measurement of species concentrations in engines, though, remains the low signal to noise ratio which results from scattering off solid
surfaces. Several efforts have been made to utilize Rayleigh scattering to study mixture formation in engines. A few of these are described below.

Arcoumanis et al (1984) made measurements of concentration via Rayleigh scattering in a motored, square piston engine. Due to the small difference in scattering cross section between most gaseous fuels and air, a large cross section gas must be used. Liquid fuels make measurement even more difficult because any droplets will result in Mie scattering which would overwhelm the Rayleigh scattering signal. The fuel replacement which is used in most applications is large refrigerant molecules such as Freon. In this case Freon-12 was used. Freon-12 has a scattering cross section almost 20 times that of air. Even with the reduced scattering in the square piston engine, and the large cross section of Freon-12, small signal levels allowed only the acquisition of pointwise data. The authors were able to acquire air/fuel ratios at several points in the combustion chamber averaged over many cycles. This data yields some information as to the flow of fuel as it enters the chamber through the intake valve. The main drawbacks to this method are its lack of spatial resolution and the fact that the combustion which results from different mixture qualities cannot be studied due to the replacement of fuel with Freon-12.

Zhao et al (1992), overcame some of the shortcomings of the above described experiment by improved optical methods. In this case, the technique was extended to two dimensions by lowering background scattering, use of a more powerful laser, and collection of the scattered signal with an intensified CCD camera. The fuel was once again replaced with a large molecular tracer, Freon 113 in this case. By moving the windows through which the laser sheet enters and leaves the cylinder and filtering the inlet air to remove dust, the background scattering was reduced to the level that planar measurement could be taken. In addition, the engine was motored with a very rich mixture ($\lambda = 0.67$) which enhances the strength of the signal. Calibration of the system was made by measuring the Rayleigh scattered signal from well mixed cases in which the relative air/fuel ratio was known. Despite all of these efforts, the results obtained are still qualitative in nature. While Rayleigh scattering offers the opportunity to measure in-
cylinder fuel distribution, it is was not considered for this research because the fuel of interest, natural gas, cannot be used as a tracer due to it's small scattering cross section.

3.3.3 Infrared Absorption

A rather new method for measuring the concentration of fuel in a combustible mixture utilizes the absorption properties of hydrocarbons. Most hydrocarbons in the gaseous phase absorb infrared radiation. Winkhofer et al (1992) used this property to make measurements of fuel concentration on a cycle resolved basis in an optically accessible engine. The relation between the absolute fuel concentration and the attenuation of an incident beam was found experimentally using mixtures of known concentrations. The laser used in these studies was a continuous He-Ne laser with a wavelength of 3.39 micrometers. A HgCdTe photo-sensor measured the degree of attenuation which occurred across the cylinder bore. The beam passed through the cylinder through small windows in the cylinder walls. Due to the continuous nature of the measurement, the variation of fuel concentration with engine crank angle could be measured for any given cycle. This provided a measurement of the cyclic variation of air/fuel ratio in the measurement beam. The measurement yields only an average air/fuel ratio in the beam diameter, across the cylinder bore. While a matrix of measurement points can be used to build up two dimensional distribution maps they are both cyclically and spatially averaged. In addition the accuracy of the measurement system allows for only qualitative results.

3.3.4 Flame Emission

Another technique which bears consideration is the determination of air/fuel ratio from the visible emission of the actual combustion process. The wavelength distribution of the light emitted during the combustion process is a strong function of the air/fuel ratio in the flame. More specifically, emission from C\textsubscript{2} and CH are monitored using fiber optic cables and photodetectors. Emissions from C\textsubscript{2} are very strong in rich flames, but loose their intensity as the air/fuel ratio moves towards stoichiometric and into the lean region.
CH emissions on the other hand are quite constant across the air/fuel ratio spectrum. Several researchers (Ohyama et al, 1990 and Chou and Patterson, 1995) have used these characteristics of the flame emission spectra to determine air/fuel ratio. In both cases, fiber optic cables are installed in the engine cylinder head to capture as much of the flame emission as possible. Two cables are used, one leads to a detector which measures CH emission. The other cable measures C\textsubscript{2} emission. By taking the ratio of these intensities the effects of varying pressure and temperature are eliminated. While the relation varies substantially between fuels, the intensity ratio can be related to air/fuel ratio for any particular fuel through experimental calibration. In each of the cited references, air/fuel ratio measurements were made at several engine locations for a large number of cycles. As would be expected, the air/fuel ratio was found to vary substantially both between locations and individual cycles. While simple, this technique does not yield the type of information which was required for this work. The measurements which have been made are an average over the entire combustion duration, and are only pointwise in space. The technique has the possibility of being expanded into planar measurements, but temporal averaging would still be necessary to achieve quantitative information concerning the relative air/fuel ratio distribution.

3.3.5 Planar Laser Induced Fluorescence

Planar Laser Induced Fluorescence (PLIF) is an optical technique in which a sheet of laser light is passed through a fluid, resulting in emission from specific molecules. This technique can be utilized to determine species concentrations, temperatures, and velocities. A detailed explanation of the mechanisms of PLIF are given in several references (e.g. van Cruyningen et. al., 1990, Seitzman and Hanson, 1993).

The property of interest in this research is the measurement of fuel concentration in a plane in the cylinder of an internal combustion engine. In order to measure species concentrations using PLIF, a match must exist between the wavelength of the laser light available, and the absorption range of the species of interest. If such a match exists, as
the sheet passes through the gas mixture, a portion of the laser light is absorbed, as given by:

\[ N_{\text{abs}} = N_{\text{in}} \times e^{-\sigma(\lambda) \int C(x, y, z) \, dx} \]  

(3.1)

Where \( N_{\text{abs}} \) is the number of photons absorbed per unit time in a unit area of the flow; \( N_{\text{in}} \) is the number of photons incident on the flow per unit time and unit area, \( e^{-\sigma(\lambda) \int C(x, y, z) \, dx} \) is a factor that accounts for the attenuation of the sheet as it passes through the medium, \( \sigma(\lambda) \) is the absorption cross section for the given species at the wavelength \( \lambda \), and \( C(x, y, z) \) is the concentration of unexcited molecules of the given species at location \( x, y, z \).

The emission from the species in question at a given \( x, y \) location is then just the number of absorbed photons multiplied by the quantum efficiency of the emission of a photon from the given excited state integrated over the thickness of the sheet:

\[ N_{\text{emitted}}(x, y) = \int N_{\text{abs}}(x, y) \times \phi(\lambda) \, dz \]  

(3.2)

Where \( N_{\text{emitted}} \) is the number of photons emitted per unit area in the \( x, y \) plane per unit time and wavelength and \( \phi(\lambda) \) is the quantum efficiency of emission from the given excited state. This analysis follows the description by Lozano et al. (1992).

In order to make an accurate measurement of the concentration \( (C(x, y, z)) \) several criteria need to be met. From Eqn. 3.1, the exponential term must be essentially constant. In order for this to be the case, the medium being traversed must be optically thin, that is, only a small amount of the incident radiation is absorbed over the length of the sheet. When this is the case, the number of photons incident at the location where the sheet first enters the medium is essentially equal to the number of photons incident at the location where the sheet leaves the medium. This assumption can be tested by measuring the power in the laser sheet before it enters, and after it exits the measurement volume. If an
optically thin mixture cannot be achieved, PLIF images can be corrected to account for the attenuation of the laser sheet as it passes through the medium, though this complicates the process.

In addition to the optically thin assumption described above, several other requirements must be met to relate emitted intensity to concentration. The absorption cross section of the mixture can be a complex function of temperature, pressure, and mixture constituents. Absorption cross sections depend on temperature mainly because as the temperature of the mixture changes, a larger number of the molecules of a given species are found outside of the ground state. That is, as the temperature of the mixture is raised significantly, some of the molecules are excited out of the ground state. In addition, the molecular tracer may dissociate, at high temperatures, forming new molecules which may or may not have excitation spectra which are resonant with the incident laser light. The absorption cross section of the mixture may change with pressure due to pressure broadening of the absorption spectrum. The shape of the absorption spectrum changes with increasing pressure, this can lead to a change in the absolute value of the absorption cross section at any given wavelength. While effects such as these can be account for, they make the quantification of concentration measurements difficult.

Assuming that the absorption cross section is independent of the state of the mixture, and the attenuation of the laser sheet is negligible, then from Eqn. 3.1, the number of photons absorbed per unit sheet volume is directly proportional to concentration of the absorbing species in that volume. These photons which are absorbed are of such a wavelength as to cause the excitation of a small portion of the dopant molecules from a lower electronic state, usually the ground state, to a higher electronic state. Once the excited molecules are in the excited electronic state, there are several paths which they can follow, only one of which produces the desired fluorescence. An excited molecule can absorb an additional photon which can lead either to elevation to a higher energy level or can return the molecule to its original ground state (stimulated emission). Neither of these scenarios plays a large role in PLIF due to the small concentration of
excited molecules. The second possible path an excited molecule can take results from inelastic collisions with other molecules. In this case the molecule exchanges energy with its collision partner leading to de-excitation of the molecule. This process, known as quenching, results in the removal of molecules from the excited states without the desired emission. If molecules are sufficiently excited, the absorbed energy may lead to the dissociation of the molecule. Once again, due to the small number density of excited molecules, this is not a major factor in the type of PLIF which has been carried out. Finally, the excited molecule can emit a photon at a wavelength longer than that of the incident laser light. This can occur over short time scales (fluorescence, typically in the nanosecond range) or long time scales (phosphorescence, typically in the millisecond range). The ratio of the emission in these two ways depends on the stability of the excited electronic state.

The purpose of the quantum efficiency in Eqn. 3.2, is to account for all of the above effects. The quantum efficiency is the proportion of excited molecules which produce the fluorescence emission on which PLIF is based. The quantum efficiency of most molecules is quite low (less than 1%). More important than the absolute magnitude of the quantum efficiency is its variation with mixture conditions. With all of the processes described above competing as paths of deexcitation, the percentage of molecules which spontaneously emit a photon can be a function of temperature, pressure, and number density. Because of the complexity of the energy transfer mechanisms in larger dopant molecules, the dependence of the quantum efficiency, and absorption cross section on temperature and pressure are typically determined experimentally. Ideally, a dopant should be found which minimizes the variation of these two parameters with mixture conditions.

Given the invariance of the quantum efficiency and absorption cross section with mixture conditions, and assuming an optically thin medium, the intensity of fluorescence which is emitted from the excited medium is dependent only on the concentration of the species of interest and the intensity of the incident radiation. If the intensity of the incident radiation is either known or can be measured, PLIF provides a direct linear
relationship between intensity and concentration. Due to the difficulty in predicting the variation of PLIF signal with temperature and pressure, a more direct approach was used in this research, as described in Chapter 5.

In order to image the mixing of natural gas and air, a tracer must be added to the natural gas stream. This tracer is necessary because there are no transitions in natural gas which can be excited with available laser technology. In selecting an appropriate tracer, many criteria must be kept in mind. The most prominent among these criteria are:

- The availability of lasers in the wavelength range for species excitation
- The magnitude of the molecular absorption cross section
- The magnitude of the quantum efficiency and its dependence on flow properties
- The magnitude of the Stokes shift between incident and emitted photons
- The lifetime of the excited state from which photons are emitted
- The similarity between physical properties of the tracer and the fuel

With these criteria in mind, the selection of known tracers can be evaluated for their applicability to the given case. The range of molecular tracers which have been utilized to this point is quite large. There are simple molecular tracers such as NO, NO₂, I₂, Na, CO, and H₂O. There are several disadvantages to using these tracers for concentration measurements. In most cases, the absorption spectrum of the simple tracers is made up of a number of discrete lines as opposed to a broad band. Reaching these transitions typically requires a tunable dye laser, which is not currently available in the OSU Optical Access Engine Laboratory.

In addition to the simple molecular tracers, there are many larger molecules which can be used as tracers. The most common among these are acetone and biacetyl. The absorption cross sections, quantum efficiencies and other fluorescence properties of the two compounds are quite similar (Lozano et. al., 1992). The tracer chosen for the current work was acetone due to its greater similarity with the fuel of interest, natural gas. Though the molecular weight of acetone is much greater than that of natural gas, it is
considerably lighter than biacetyl. In addition, the boiling point of acetone is about 30 degrees Celsius less than that of biacetyl. The vapor pressure of acetone at room temperature is approximately 27.5 kPa. This relatively high vapor pressure leads to a straightforward doping procedure, and one in which the concentration of dopant in the fuel mixture can be carefully controlled and monitored.

Understanding the fluorescence properties of acetone was of the utmost important to both designing the mixture homogeneity measurement experiment and interpreting the results. The properties of the most importance are the absorption cross section and the quantum efficiency, and their dependence on properties of the flow. There are three electronic states of interest in the laser induced fluorescence of acetone. The ground state (S₀), the first excited singlet (S₁), and the first excited triplet (T₃). Acetone has a broad excitation spectrum, allowing the use of a variety of excitation sources. The spectrum extends from 210 nm to 350 nm. The maximum absorption cross section is 4.7 * 10⁻²⁰ cm² occurring at a wavelength of 275 nm. The excitation wavelength used in this research is 266 nm, at which the absorption cross section is approximately 4.0 * 10⁻²⁰ cm². This data is taken from Lozano et al and was obtained experimentally at room temperature. It has been shown in the literature (Caldwell and Hoare, 1962) that the shape of the spectrum changes with temperature. The main effect is to shift the absorption peak to longer wavelengths. This phenomenon can lead to a variation of overall quantum yield with temperature, which must be taken into account when interpreting PLIF images.

Once the acetone molecules reach the S₁ state, there are three main paths which the molecules can take. First, a small amount of acetone dissociates forming biacetyl. While biacetyl is not excited by the 266 nm wavelength used in the proposed work, it can be excited by collisions with electronically excited acetone. Fortunately, excited biacetyl emits photon mainly by phosphorescence, which is almost entirely quenched in the presence of oxygen. Thus the dissociation of acetone does not play a major role in its fluorescence properties. The second path which is taken is intersystem crossing to the first excited triplet (T₃). The large majority of molecules follow this route. The small
number of molecules which are not transferred to the T\textsubscript{3} state, fluoresce in the wavelength region from 340 nm to 700 nm. The bulk of the emitted light has a wavelength between 400 and 500 nm. The nominal quantum efficiency of this emission is 0.2%, and the lifetime of this state is approximately 4 ns. Because the dominant deexcitation path for acetone is transfer to the T\textsubscript{3} state by intersystem crossing, collisional quenching of fluorescence is negligible. For this reason, the quantum yield of acetone fluorescence is independent of molecular collisions and thus is independent of the pressure of the mixture.

The large number of molecules which are transferred to the first excited triplet state have two possible paths to deexcitation. In oxygen free environments, the main path is phosphorescence with a lifetime of 200 microseconds. This emission would mask fluorescence from the excited singlet making PLIF with acetone impossible. Fortunately, the phosphorescence of the T\textsubscript{3} state is heavily quenched with oxygen. Thus, the main path of depopulation of the T\textsubscript{3} state is collisional quenching, which does not interfere with the fluorescence measurements. More detailed descriptions of the laser induced molecular processes in acetone are available from several sources (Halpem and Ware, 1970, Caldwell and Hoare, 1962, Lozano et. al., 1992, Seitzman and Hanson, 1993).

The utilization of PLIF for the determination of relative air/fuel ratio in an internal combustion engine has been demonstrated by several research groups in the past several years. One of the early efforts (Arnold et. al., 1990) used acetaldehyde as the fluorescence tracer in a gasoline fueled engine. In this work, the dopant was added at a concentration of 1000 ppm to gasoline. The acetaldehyde was used as a flame front marker, as it decomposes almost entirely above temperatures of 900 K. The experiments were carried out in a square piston rapid compression machine, with a XeCl laser acting as the excitation source. While the signal to noise ratio was sufficiently large to distinguish burned and unburned regions (S/N = 10/1), it could not provide any quantitative information as to the distribution of the fuel within the cylinder. Applications of this type are an effective means to track flame front propagation which is an important parameter from an engine modeling perspective.
Lawrenz et al (1992) carried out a thorough investigation of possible PLIF seeding compounds and reported the fluorescence properties of six of these. While the overall quantum yield of several tracers was found to be superior to that of acetone, there are several other properties which make acetone a more useful dopant. Biacetyl, for example, has an excellent fluorescence magnitude, but its fluorescence is quenched in the presence of oxygen such that the fluorescence signal is proportional to tracer concentration but inversely proportional to pressure. At the pressures typical of internal combustion engine operation, the signal is significantly smaller than acetone. The authors selected ethylmethylketone as their tracer due to its physical similarity to gasoline. Its vapor pressure is quite similar to typical gasoline blends. Some experimental results are presented in which gasoline was entirely replaced with ethylmethylketone during motored operation in a square piston engine. These limited results demonstrate that the mixture is highly inhomogeneous during the intake stroke, and even at the end of the compression stroke the relative air/fuel ratio varies from 0.8 to 1.2 across the combustion chamber.

PLIF images were collected using biacetyl as a tracer in a more realistic optical access engine by Baritaud and Heinze (1992). In this work, a sheet of tripled Nd:YAG laser light was brought into the engine cylinder by means of large windows in the cylinder walls. Due to the geometry changes required to accommodate these windows the compression ratio of the engine was lowered to 6.2:1. The PLIF images of the biacetyl doped fuel were acquired through a transparent Bowditch type piston. While the cylinder bore was 86 mm, the images region was 38 mm by 60 mm in the center of the chamber. The collected images were processed to remove background noise, corrected for variations of signal strength with temperature and pressure, and nonlinearities in the collection optics. The authors used qualitative images to determine the mixture quality as a function of injection timing. The mixture quality was taken to be the standard deviation in the intensity on a pixel by pixel basis. For this gasoline fueled engine, the mixture distribution was best when injection occurred prior to the opening of the intake valve. As discussed previously, this is not the case for a gaseous fuel.
Wolff et al (1993) performed calibration studies in a static test cell and a motored engine in order to demonstrate the applicability of acetone as a tracer for PLIF measurements in engines. Their test cell measurements included the variation of fluorescence intensity with both temperature (300 to 800 K) and pressure (0.05 bar to 35 bar). Their conclusions, contrary to the acetone fluorescence properties discussed earlier, show no increase in the fluorescence intensity with increased temperature. The measured fluorescence intensity varied less than 15% from 300 K to 700 K. There are several possible explanations for this. One is that the excitation source may lie in the flat region of the absorption curve such that the shifting of the curve does not change the absorption cross section. The authors do not specify their excitation source. Another possible explanation is the dissociation of acetone with increasing temperature in the static cell. As the cell temperature increases, the increase in the absorption cross section could be offset by a decrease in the number density due to dissociation. The relation between fluorescence intensity and temperature is one which requires some effort in order to make quantitative concentration measurements utilizing acetone as a tracer.

In an engine similar to the one described above, Zhao et al (1994), made measurements of fuel distribution utilizing nitrogen dioxide as a fluorescence marker. In this experiment, the fuel was replaced with a mixture of nitrogen and nitrogen dioxide. The fluorescence properties of nitrogen dioxide are not as well suited to PLIF application in engines due to the strong variation of signal intensity with temperature and pressure. Low signal to noise ratios resulted in an inability to obtain quantitative results with this method. The images acquired in this work are given only in black and white distinguishing between rich and lean of stoichiometric as a function of position. This is of little use, as the range of variation is not known. The difficulty in obtaining reasonable PLIF results in gaseous flows is demonstrated by the marginal quality of the results in this work.

A recent work by Neij et al (1994) related fuel distribution images acquired by PLIF to the duration of the early flame development period. This work, performed in a gasoline powered engine, measured the relative air/fuel ratio in a region near the spark
plug, on a cycle resolved basis, using diethyl ketone as a fluorescence tracer. The authors estimated the error in the air/fuel ratio measurement to be on the order of 10%. In a small area near the spark plug gap, approximately 5 mm in radius, the mean equivalence ratio was found to vary from 0.6 to 1.0 in a set of 50 images collected in a single test run. As expected, the degree of mixture homogeneity was found to depend strongly on the timing and location of the fuel injector event.

As described above, PLIF has been demonstrated as a practical means of measuring fuel distributions in internal combustion engines. Further development has been carried out in the present work to extend the method to natural gas engines, as described in Chapters 5 and 6. This work utilized PLIF of acetone to determine the degree of maldistribution of fuel prior to combustion, and relate this parameter to the level of CTCV measured in a fired natural gas fueled engine.
CHAPTER 4

EXPERIMENTAL QUANTIFICATION OF COMBUSTION QUALITY

4.1 Introduction

The determination of the impact of mixture inhomogeneity on cyclic variations in internal combustion engines requires a means to measure both the distribution of the mixture and the quality of combustion. Methods by which the fuel distribution can be measured quantitatively were discussed in Chapter 3. The focus of this chapter is methods of quantifying the quality of combustion for a given cycle, and the overall degree of CTCV.

While the torque output of the engine, or its bulk measured emissions are traditional means of determining engine combustion quality, they provide little insight into cycle resolved combustion processes. The output torque of the engine is a function of many parameters including frictional losses, pumping losses, rotational dynamics, and combustion generated cylinder pressure. While the possibility exists to determine misfire or knock from engine torque output, detailed information concerning individual combustion events cannot be reconstructed from a torque measurement. In a similar way, the engine emissions measured in the exhaust manifold can be used to determine overall air/fuel ratio, or to detect misfire, but this method provides only an average over several cycles, and is therefore of little use in determining cycle resolved combustion quality.

The most effective way to track the combustion process on a cycle resolved basis is the measurement of cylinder pressure. Piezo-electric pressure transducers are currently available which are specifically designed for the harsh environment in the combustion
The specifics of cylinder pressure measurement are discussed in the next chapter. All measures of combustion quality which were utilized in this research are based on the measurement of cylinder pressure. Other techniques that see some application in current research, but which are not discussed here, include optical techniques for the measurement of flame front propagation and in-cylinder temperature measurement by flame spectroscopy.

The pressure diagnostics which will be discussed in this chapter can be divided into two general categories. The first category is quantities which are based on the pressure measurement directly. The evolution of cylinder pressure in an internal combustion engine is a complicated process involving the heat release characteristics of combustion, heat transfer by convection, conduction and radiation, as well as the constantly changing combustion chamber volume. The direct use of cylinder pressure is a means to measure the summation of these effects on engine operation. The most common parameters involving direct use of combustion pressure are the maximum combustion pressure, $P_{\text{max}}$, the crank angle of occurrence of this maximum pressure, $\theta_{P_{\text{max}}}$, and the indicated mean effective pressure, IMEP. The utility of each of these parameters will be discussed in detail in this chapter. The second category of parameters derived from the measurement of cylinder pressure involve indirect measurements of combustion parameters. These parameters are based on combining the cylinder pressure measurements with simple thermodynamic engine models. By utilizing the cylinder pressure, cylinder volume, and the derivatives of these functions, in addition to empirical measures of flame propagation, quantitative measures of the combustion quality can be derived. These include mainly the heat release rate and overall heat release for a given cycle, as well as the duration of the combustion event. These parameters, and their derivation from cylinder pressure are discussed later in the chapter.
4.2 Combustion Pressure Measurements

The most easily acquired measure of engine combustion quality on a cycle resolved basis is the engine cylinder pressure. A digital data acquisition system triggered by a crankshaft mounted incremental encoder to sample voltages produced from the charge amplified signal of a piezo-electric pressure transducer is a relatively simple way of acquiring pressure traces for individual cycles. The encoder, pressure transducer, and data acquisition systems used in this work are discussed in detail in Chapter 5. Assuming that pressure traces can be acquired with sufficient resolution both in the pressure measurement and the crank angle, pressure data can be utilized as an effective measure of engine performance. While many different pressure parameters have been utilized for the quantification of combustion quality, there are three which have been found to be the most useful when studying CTCV. These are the indicated mean effective pressure, the maximum cylinder pressure for a given cycle, and the angle of occurrence of the maximum cylinder pressure. Other measures, such as the maximum rate of change of cylinder pressure and the angle of occurrence of this quantity have been used. These measures have been demonstrated in most cases to provide minimal additional information in terms of the cyclic variability of the engine.

4.2.1 Indicated Mean Effective Pressure

The indicated mean effective pressure is the most commonly used measure of engine performance. This is due to its relative invariance with engine size, and thus its application to the comparison of different engine displacements and geometries. The mean effective pressure is an excellent measure of the overall performance of the engine, as it is basically the specific work produced per unit volume. By definition the mean effective pressure is the work for a given cycle divided by the displaced volume of the engine:

\[
\text{MEP} = \frac{W_c}{V_d}
\]  

(4.1)
Where $W_c$ is the work per cycle and $V_d$ is the displaced volume of the engine.

The indicated mean effective pressure, IMEP, is simply the work delivered to the piston by the cylinder gases during a given engine cycle divided by the engine displacement. This quantity neglects frictional and all other losses and measures only the $P\cdot dV$ work delivered to the piston. This neglects the work done during the intake and exhaust strokes, known as pumping losses. Thus the net indicated mean effective pressure for a given cycle is defined as:

\[
\text{IMEP}_n = \frac{\int_{\text{BDC}}^{\text{TDC}} P \cdot dV}{V_d}
\]

(4.2)

Where $P$ is the cylinder pressure (gage pressure) and $dV$ is the incremental change in volume.

Clearly because the cylinder pressure data will be sampled data, the integral given above will become a summation over all the data points collected during the compression and expansion strokes. Thus the numerator of Eqn. 4.2 can be replaced as follows:

\[
\int_{\text{BDC}}^{\text{TDC}} P \cdot dV = \sum_{\theta = -\infty}^{\theta = \infty} \frac{P \cdot dV}{d\theta} \cdot d\theta
\]

(4.3)

The incremental change in the angle, $d\theta$, is the resolution of the encoder used to trigger the data acquisition system. The rate of change of the cylinder volume can be found as a function of engine geometry. The cylinder volume at any given crank angle is given by:

\[
V(\theta) = V_c + \frac{V_c}{2} \cdot (r_c - 1) \cdot \left[ R + 1 - \cos \theta - (R^2 - \sin^2 \theta)^{1/2} \right]
\]

(4.4)
Where \( V \) is the cylinder volume at crank angle \( \theta \), \( V_c \) is the clearance volume of the engine, \( r_c \) is the compression ratio, and \( R \) is the ratio of connecting rod length to crank radius. The rate of change of the cylinder volume is found by differentiating this function with respect to crank angle:

\[
\frac{dV}{d\theta} = \frac{V_c \sin \theta}{2} \left( r_c - 1 \right) \left( 1 + \frac{\cos \theta}{\sqrt{R^2 - \sin^2 \theta}} \right)
\]  

With this rate of change of cylinder volume as a function of crank angle known, the expression for the net indicated mean effective pressure as a function of the sampled pressure data is given by:

\[
\text{IMEP}_{n} = \frac{\sum_{\theta=-x}^{\theta=x} P(\theta) \frac{V_c \sin \theta}{2} \left( r_c - 1 \right) \left( 1 + \frac{\cos \theta}{\sqrt{R^2 - \sin^2 \theta}} \right)}{V_d}
\]  

The net indicated mean effective pressure can be calculated for any single engine cycle, and used as a measure of the efficiency of the combustion process. IMEP takes into account not only the magnitude of the pressures which occur in a given cycle, but the accuracy of the phasing of the cycle with respect to the maximum brake torque timing. For this reason, it is an excellent measure of the quality of the combustion process.

4.2.2 Maximum Cylinder Pressure

The maximum cylinder pressure which occurs during a given stroke is used for several reasons as an indication of the quality of the combustion processes in internal combustion engines. Maximum cylinder pressure is a very simple means of quantifying the combustion process as it requires no calculation, and represents the combustion quality with a single parameter. The maximum cylinder pressure which occurs in a given
cycle is a function of both the total amount of energy which is released during combustion, and the timing of that release relative to TDC of the engine.

As the amount of heat released increases, the maximum pressure naturally goes up if all else is held constant. On the other hand, the maximum cylinder pressure also increases as the effective ignition timing is advanced. The effective ignition timing here refers to the time at which significant deviation from the motored cylinder pressure trace is first observed. This timing, as discussed later, is a function of the timing of the spark as well as the ignition delay. As the effective ignition timing is advanced, the heat release occurs earlier in the cycle, at which time the cylinder volume is smaller. This leads to a higher maximum cylinder pressure. The ignition timing which corresponds to the maximum cylinder pressure is not coincident with the maximum brake torque timing as pressure which is developed by combustion before the piston reaches TDC does negative work on the piston, resulting in lower overall torque. The cyclic variation in the ignition delay results in some cycles which are advanced from MBT and some which are retarded from it. In addition, the total heat release which occurs during a given cycle is a function of both the mass of fuel trapped in the cylinder and the combustion efficiency. The combustion efficiency accounts for fuel which leaves the combustion chamber either partially or completely unoxidized. Because the maximum cylinder pressure accounts for both the total heat release and the timing of the heat release it is an excellent measure of the quality of the combustion event.

4.2.3 Angle of Occurrence of the Maximum Cylinder Pressure

The crank angle at which the maximum cylinder pressure occurs, $\theta_{\text{pmax}}$, is an excellent indicator of the duration of the flame initiation period. The crank angle of occurrence of maximum cylinder pressure is somewhat insensitive to the total mass of fuel trapped per cycle, and the total mass of fuel burned. As the timing of the ignition spark is essentially constant, as discussed in chapter two, the main factor which influences the timing of the combustion process is the ignition delay. By recording the crank angle at which the maximum pressure occurs, the duration of ignition delay can be
approximated. As was discussed earlier, the duration of the delay between the spark plug firing and the first noticeable sign of pressure increase from motored operation is very well correlated with the degree of CTCV which occurs at a given operating point. The impact of factors such as overall air/fuel ratio, mixture distribution, and in-cylinder flow which are known to affect ignition delay can be quantified with \( \theta_{p_{\text{max}}} \). As with maximum cylinder pressure, one of the main advantages of using \( \theta_{p_{\text{max}}} \) is its simplicity. It requires no additional calculation to extract it from the pressure data, and it can be stored as a single parameter which quantifies the combustion quality through the impact of ignition delay for any given cycle.

4.3 Calculated Combustion Parameters

While the direct pressure measurements discussed above offer a simple means of quantifying the combustion quality in internal combustion engines, they tend to measure the overall effect of several competing phenomenon. For example, the indicated mean effective pressure takes into account the total heat release, the timing of the heat release, and the variation in heat transfer from cycle to cycle. In order to obtain more specific information as to which of these parameters is the major contributor to cyclic variability, several methods have been developed to extract specific information about the combustion process from cylinder pressure data. These methods are based on simple combustion models and require several simplifying assumptions in order to allow calculation from only the cylinder pressure. The two most prevalent combustion related parameters which are calculated are the heat release rate and the burn duration. The methods of calculating these quantities and their major uses are discussed in detail in this section.

4.3.1 Heat Release Rate

The rate of heat release during the combustion process is an excellent means by which to characterize the combustion process. The heat release rate is a measure of the flame propagation speed, which, as discussed earlier has a major impact on CTCV.
Unfortunately, due to the complex geometries and high speed nature of reciprocating engines, the rate of heat release can not be measured directly as in combustion bomb type experiments. The heat release rate can, on the other hand, be calculated from the cylinder pressure, though to a lower degree of accuracy. The easiest means of estimating the heat release rate from cylinder pressure data results from treating the cylinder during combustion as a closed system of varying volume. If heat transfer effects and flows into and out of crevice volumes in the combustion chamber are neglected, an energy balance on the combustion chamber control volume yields:

\[ \delta Q_{ch} = mc_dT + PdV \]  (4.7)

Where \( \delta Q_{ch} \) is the incremental amount of chemical energy released, \( m \) is the total mass of the system, \( c_v \) is the specific heat at constant volume of the gas, \( dT \) is the incremental change in the gas temperature, \( P \) is the cylinder pressure, and \( dV \) is the incremental change in the cylinder volume.

There are several important assumptions implicit in this equation. Firstly, heat transfer, which accounts for approximately 15% of the total heat release is neglected. While this leads to significant error in the absolute value of the heat release, the real item of interest is the variation in heat release from one cycle to the next. The effects of heat transfer during steady operation are essentially unchanged between cycles. Thus the heat release rate calculated here is intended more as a basis for comparison than an absolute measure of the heat release. Additional assumptions include the absence of crevice volume flows and the treatment of the entire cylinder volume as a single mass. Once again, while these lead to errors in the absolute value of the heat release, they do little to impact its variation from one cycle to the next.

Applying the ideal gas law to Eqn. 4.7 gives the expression shown in Eqn. 4.8. The treatment of the cylinder gas constituents as an ideal gas with a constant value of \( R \) will lead to small additional errors but is consistent with the purpose of the derivation.
\[ \delta Q_{ch} = \left( \frac{c_v}{R} \right) V dP + \left( \frac{c_p}{R} + 1 \right) P dV \] (4.8)

Where R is the average gas constant for the cylinder constituents, and c_p is the specific heat at constant pressure. Substituting the ratio of specific heats into Eqn. 4.8, and taking the rate of change with respect to crank angle yields a measure for the heat release rate as a function only of known or measured values, namely pressure, volume, and the rate of change of these properties:

\[ \frac{dQ_{ch}}{d\theta} = \frac{\gamma}{\gamma - 1} \frac{\gamma}{\gamma - 1} P \frac{dv}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \] (4.9)

Where \( \gamma \) is the ratio of specific heats.

The ratio of specific heats can be either taken as a constant value throughout the process or can be estimated as a function of temperature. The above estimation of the heat release rate has several applications. The rate of heat release at any given crank angle can be used to estimate the flame speed and the proportion of burned and unburned mixture in the cylinder. The heat release rate shortly after ignition is a good measure of the mixture strength, as the flame speed at this time is closely related to the laminar flame speed. The heat release rate can also be integrated and normalized to yield a measure of the burned gas fraction, this provides one method of estimating burn durations as discussed in the next section. The importance of the heat release rate in analyzing combustion is as a more direct link between the collected cylinder pressure data and the actual cycle resolved combustion events. That is, the heat release rate is a measure of the burning rate in the engine, which is related directly to the properties of the combusting mixture.
4.3.2 Burn Duration

The number of crank angle degrees required for the complete combustion of the cylinder charge is an important parameter for the operation of internal combustion engines. The burn duration is influenced by several parameters including in-cylinder flow and air/fuel ratio as described in Chapter 2. Most importantly for this research, the burn duration is believed to vary significantly with the local mixture strength in the region of the spark plug during ignition. This is due primarily to the influence of mixture stoichiometry on laminar flame speed. For this reason, the burn duration is a very important parameter quantifying CTCV.

While the burn duration can be estimated using the heat release rate, as described previously, a more accurate measure is achieved by the application of a simple two zone combustion model. In this model, the compression and expansion strokes in the engine are assumed to be essentially isentropic. The combustion chamber is divided into two regions, a burned gas region and an unburned gas region, separated by a thin flame front. From the polytropic relation applied to the burned and unburned volumes:

\[ V_u = V_{u,o} \left( \frac{P_o}{P} \right)^\frac{1}{n} \]  

(4.10)

Where \( V_u \) is the volume of the unburned mixture, \( V_{u,o} \) is the volume the unburned mixture would have occupied at point of ignition, \( P_o \) is the pressure in the combustion chamber at ignition, \( n \) is the polytropic exponent (experimentally determined \( \approx 1.3 \)), and \( P \) is the cylinder pressure.

\[ V_b = V_{b,f} \left( \frac{P_f}{P} \right)^\frac{1}{n} \]  

(4.11)

Where \( V_b \) is the volume of the burned mixture, \( V_{b,f} \) is the volume which the burned mixture would occupy at the end of combustion, and \( P_f \) is the pressure in the combustion chamber at the end of combustion.
Because the flame front is considered to occupy a negligible volume:

\[ V = V_u + V_b \]  \hspace{1cm} (4.12)

Where \( V \) is the total cylinder volume.

Clearly, the burned gas fraction can be written in terms of either the burned or unburned mixture volume ratio:

\[ x_b = 1 - \frac{V_{u,o}}{V_o} = \frac{V_{b,f}}{V_f} \]  \hspace{1cm} (4.13)

Where \( V_o \) is the cylinder volume at ignition, and \( V_f \) is the cylinder volume at the end of combustion.

Combining equations 4.10 through 4.13 yields an expression for the burned gas fraction in terms of the cylinder pressure and the conditions at the beginning and end of combustion:

\[ x_b = \frac{\frac{1}{P^n} V - \frac{1}{P^n} V_o}{\frac{1}{P^n} V_f - \frac{1}{P^n} V_o} \]  \hspace{1cm} (4.14)

This is only an approximation of the actual mass fraction burned for several reasons including the neglect of heat transfer and the variation of the polytropic index over the large temperature range which occurs during combustion. Nonetheless it has been found to correlate very well with burned gas fraction measurements taken from flame visualization experiments.

There are several burn durations which are in common use in internal combustion engine research. The total burn duration \( \theta_{0-100} \) is not typically used due to the difficulty in determining accurately the exact beginning and ending points of combustion. The two most commonly used burn durations are the number of crank angle degrees between...
ignition and 2% mass fraction burned, $\theta_{0.2}$, and the duration of the bulk of combustion, between 10% and 90%, $\theta_{10-90}$. Each of these burn durations is used for specific applications. The 0% to 2% burn duration is used as a measure of the flame development angle. As discussed previously, the duration of the flame kernel development stage of combustion has a major impact on the overall duration of combustion, and thus the CTCV at a given operating point. The 0% to 2% burn duration has been found to be closely related to air/fuel ratio both overall and local to the spark plug, due to the influence of air/fuel ratio on laminar flame speed. The second major duration of interest, $\theta_{10-90}$, is used mainly as a measure of the overall flame speed during a given cycle. This flame speed is heavily dependent on both mixture strength, and turbulence intensity. For this reason, the 10% to 90% burn duration is a good measure of the influence of various operating parameters on the in-cylinder flow field. The primary burn duration of interest in this research was the flame development angle, but the rapid burning angle, $\theta_{10-90}$, was also used. The means by which CTCV was quantified in this project are discussed in detail in Chapter 7.
CHAPTER 5

EXPERIMENTAL METHODS

5.1 Introduction

The purpose of this chapter is to describe the laboratory facilities and experimental methods which were used in this research. The description of facilities includes details of the OSU Optical Access Engine which was used in all the experiments described here. The equipment used to produce the in-cylinder fuel distribution images is also discussed. This includes the excitation source and optical components, and the image collection system. The techniques used in processing the acquired images is discussed in the next chapter. In addition to the engine itself, and the PLIF equipment, several auxiliary systems are covered, including the pressure measurement system and the computer controlling data acquisition and computer timing.

The final sections in this chapter describe the experimental techniques which have been applied. The purpose of this section is to discuss how the data was collected and the design of the experiments involved. This includes an explanation of how interfering inputs were dealt with in the PLIF experiments. Both the PLIF and pressure measurement experiments are covered. The results of these experiments are presented and discussed in Chapters 6 and 7.

5.2 Laboratory Facilities

While the general experimental techniques that were utilized in this project were discussed in chapters 3 and 4, the specific equipment which was used is described in this section. The equipment consists of: the optical access engine, the laser excitation source...
and optics, the image collection system, the system which supplies and monitors air and fuel to the engine, the pressure measurement system, and the computer control system which both acquires data and determines the timing of the experiment. The physical equipment is described as well as methods by which it was utilized in this work.

5.2.1 Optical Access Engine

The OSU Optical Access Engine (OAE) is the basic tool employed in the fuel concentration measurement and combustion pressure measurement schemes. The OAE is a single cylinder, four-stroke, spark ignition internal combustion engine which has been fabricated to allow nearly unrestricted optical access to the combustion chamber during engine operation. The basic configuration of the engine, as shown in Figure 5.1, is that of an extended piston and required structure, mounted atop a single cylinder production engine. This extended piston runs in a sapphire cylinder which is clamped between the auxiliary structure and one cylinder bank of a production automotive type cylinder head.

Optical access in the engine is provided through both the top of the piston and the entire circumference of the cylinder. The piston face contains a large quartz window designed such that over 75% of the combustion chamber can be viewed from below at any given time. In addition, the piston can be assembled in two different configurations, allowing for 90% access between the two arrangements. Access through the cylinder wall is by means of a sapphire cylinder which extends well past the bottom dead center location of the piston. This configuration allows viewing of the combustion chamber throughout all four engine strokes.

The piston window is shaped such that the entire diameter, minus the portion covered by the piston rings can be viewed in one direction. In the direction perpendicular to this, the viewing area is 1.75 inches wide. The piston is a combination of the quartz window described above, and an anodized aluminum frame. The entire piston is held together with socket head cap screws, allowing easy disassembly of the unit. A single piston ring is used for both sealing the combustion chamber and as a bearing surface. This ring is a c-ring made from a low friction, high temperature polymer. The sealing surface of the
ring is held against the cylinder wall by a stainless steel spring. The piston runs in the cylinder unlubricated, as lubrication would degrade optical access.

The sapphire cylinder which provides the running surface for the piston and replaces the cast iron cylinder liner used in typical engines, was grown from a single cylinder sapphire crystal. Unlike naturally occurring sapphire, laboratory grown sapphire is transparent, with excellent transmissive properties from the UV into the infrared. In addition, the sapphire has a tensile strength similar to that of cast iron. The main weakness of the cylinder is the possibility of brittle fracture, but no problems have been experienced in the course of this research. The cylinder is 5 inches long with an inside diameter of 3.00 inches. The running clearance of the piston in the cylinder is roughly .025 inches, which is significantly higher than in a production engine. This was done so that the piston ring would form the only bearing surface, and no scuffing would occur at the piston-cylinder interface. The sapphire cylinder is approximately 0.15 inches thick. The thin cylinder wall minimizes optical distortion, and lowers the thermal stress experienced by the cylinder. The cylinder was designed to withstand pressures of 1000 psi, and the temperature cycling associated with normal operation. As an additional safety factor, experiments are run in skip firing mode, with ignition on every tenth stroke. Additional information concerning the design of the cylinder and piston assembly are available from other sources (Hiltner, 1993 and Hiltner and Kennedy 1994).

While the engine has been fabricated to simulate as closely as possible production engine operation, it has also been designed to be adaptable to a wide range of engine applications. The extended piston assembly is composed of two sections which can be separated by thin spacer elements. The use of these elements varies the clearance height of the engine, and provides for variation of the compression ratio from 4:1 up to 10:1. The clearance height in all of the experiments included in this work was fixed at 0.125 inches. This results in a compression ratio of 7.2. The physical dimensions of the engine are provided in Table 5.1. Removal of the piston from the engine can be achieved by separation of the upper and lower piston assembly. Ease of disassembly is required as the cylinder wall surface must be cleaned after 30 to 45 minutes of engine operation. This is
necessary to reduce cylinder wall deposits which produce optical noise in the tumble and cross tumble plane images, as discussed in the experimental procedure section.

In addition to the variable compression ratio feature, the engine employs a novel valve train configuration which provides for continuously variable valve timing. The valve train consists of encoder on the crankshaft and camshaft, and a high power step motor with controller. The motor controller receives pulses from a relative crankshaft encoder at a rate of 8000 pulses/revolution. The controller then divides down this pulse rate such that the motor turns at 1/2 the crankshaft speed. This system is supplemented with a feedback control system utilizing absolute encoders on the crankshaft and camshaft. Errors which are detected in the camshaft position are corrected by altering the camshaft to crankshaft speed ratio until resynchronization has occurred. While not used in this work, the phasing of the camshaft and crankshaft can also be changed during engine operation, allowing variable valve timing schemes to be tested during a single experiment. In this work, the step motor drove the production camshaft via a timing belt. Proper phasing between the crankshaft and camshaft was accomplished by setting top dead center of the engine via physical piston location measurement, and setting camshaft position from the known valve lift profile. Valve train timing with this system is accurate within ±0.3 crank shaft degrees.

The configuration of the OAE used for this research utilizes a four valve, overhead camshaft, cylinder head. This cylinder head, produced by Honda, is used on a four cylinder, 1.5 liter production engine, and is beginning production as a natural gas engine in 1998. The cylinder head has a pentroof combustion chamber shape with a centrally located spark plug. Only minor modifications have been carried out on the cylinder head to adapt it to the OAE. The number two cylinder location in the head was modified to accept the sapphire OAE cylinder. This is the second cylinder from the timing belt sprocket end of the engine. An o-ring groove was cut around the combustion chamber which allows the sapphire cylinder to seal the combustion chamber with the addition of an o-ring. The combustion chamber geometry is altered only slightly in that the typical cylinder head gasket has been replaced by an o-ring groove. Additional changes to the
cylinder head included the removal of the rocker arms driving the valves of the dormant cylinders and the addition of an encoder at the location of the production ignition distributor. The intake and exhaust valve springs were also replaced with significantly softer springs to reduce the load on the step motor. This does not affect the valve lift profile at the low engine speeds used in this work.

The fuel injection system has been modified to utilize natural gas. This includes a custom built fuel rail and a Honda prototype natural gas fuel injector. The injector is driven by a Honda produced Engine Control Unit. The mass of fuel injected per cycle is dependent only on the injection duration at a given fuel rail pressure, which is controlled by the computer timing board described later. The fuel mass per cycle is nearly proportional to the injection pulse length. The injection system operates at a pressure of 35 psig, as specified by the manufacturer. Injection of natural gas occurs at a distance roughly 6 inches from the intake valves. The port design in this engine has the two intake valve runners separated for the final 2 inches upstream of the valves. The injector is directed at the split between the two valve ports. At low speeds, one of the intake valves is deactivated to promote swirl in the engine. The impact of this geometry is discussed in Chapter 6.

The base engine which drives the extended piston of the OAE is a single cylinder production engine. The only purpose of this engine is to provide the linear motion to the extended piston. This engine has a stroke of 2.64 inches. Mounted beneath the engine is a second crankshaft and weighted piston. The purpose of this apparatus is engine balancing. The extended piston assembly weighs slightly more than 5 pounds, leading to a large unbalanced inertial force. The lower assembly offsets this force by moving a piston of the same mass 180 degrees out of phase with the upper piston. The entire engine assembly is driven by a 50 hp D.C. motor. This motor has a feedback tachometer that is used to maintain the desired engine speed. The inertia of the motor shaft also serves to damp the speed oscillations generated by the compression and expansion processes.
5.2.2 Excitation Source

An intense sheet of laser light is required to excite acetone to a state from which it will fluoresce. The laser source for this experiment is a Spectra-Physics GCR 230, 10 Hz, Nd:YAG laser. This source is capable of producing short duration (~ 10 ns) high energy (1250 mJ/pulse) pulses in the infrared region (1064 nm). The wavelength region in which acetone absorbs a significant amount of energy is in the ultraviolet (225 nm to 325 nm). This wavelength is obtained by passing the Nd:YAG output through two temperature stabilized doubling crystals. The output from these crystals is 266 nm pulses with a maximum energy per pulse of 130 mJ. The laser power during these experiments was considerably lower, between 10 and 20 mJ/pulse. This lower laser level prevents optical damage to the cylinder and results in more consistent pulse energy from cycle to cycle. The laser is water cooled to help maintain constant output, and the doubling crystals are maintained at a constant temperature to assure efficient conversion to shorter wavelengths. The entire assembly is purged with nitrogen to prevent condensation. While careful adjustment of the laser is necessary to obtain optimum results, all acquired images have been corrected for fluctuating laser power, reducing the impact of possible inconsistent laser performance.

The output from the laser's harmonic generator, described above, is formed into a thin sheet for PLIF. The beam is first turned by 90 degrees four times by dielectrically coated mirrors. This serves both to relocate the beam and filter out undesired wavelengths. A significant portion of 1064 nm and 532 nm light exits the harmonic generator with the desired ultraviolet wavelength. The four beam turning mirrors reflect light very efficiently (>99%) at 266 nm, but pass the large majority of light at other wavelengths (<0.1% reflectance). With the four mirror setup, the higher orders of light are attenuated to $10^{-12}$ of their initial strength. This removal of extraneous wavelengths is of paramount importance in that the 532 nm wavelength falls in the emission range of acetone, and easily masks the actual fluorescence if it is not removed from the beam.
The essentially pure beam which leaves the fourth mirror is spread into a thick sheet by a convex cylindrical, UV grade quartz lens with a focal length of 50 mm. This expanding sheet is then attenuated at the edges by an adjustable iris. The beam is spread such that only the center 50% of the diameter of the beam is used. As the beam has an essentially spatially Gaussian profile, this results in a much more uniform sheet than if the entire beam were used. The sheet is then passed through a 1000 mm focal length spherical UV grade lens. This lens collimates the sheet in the spanwise direction, leading to a somewhat uniform sheet with parallel edges, and also focuses the sheet with respect to thickness. The end result of the above described optics is to deliver a thin (~ 0.6 mm), collimated (height ~ 70 mm), relatively uniform sheet of 266 nm light to the OAE measurement volume. The thickness of the sheet varies as it passes through the measurement volume with its focal point at the center of the cylinder. This variation is accounted for in two ways. The linear nature of the PLIF process assures that the fluorescence signal remains consistent with varying sheet thickness. While the intensity of radiation in the thicker sections of the sheet is lower, the total number of acetone molecules excited remains stable as the volume in the sheet is larger. Similarly in the thinner portions of the sheet, the intensity increases, leading to increased production of electronically excited acetone. In addition to the natural correction for sheet thickness, the images which have been collected are always corrected for sheet profile and thickness as described in Chapter 6.

5.2.3 Image Collection System

The photons emitted by the acetone dopant when excited by the laser sheet, must be collected normal to the laser sheet in order to produce images that can be used to determine the fuel distribution. The image collection system can be seen in the OAE photographs shown in Figure 5.2. Two different image collection schemes are possible. Images can be collected in the swirl plane by passing the laser sheet through the cylinder parallel with the face of the cylinder head. In this case images are collected through the transparent piston by means of a 45 degree mirror which mounts through the lower piston.
assembly and attaches to the engine structure. The second imaging method involves passing the laser sheet through the cylinder perpendicular to the face of the cylinder head and viewing the fuel distribution through the cylinder wall. This is the arrangement shown in the photographs. The emitted light is collected by a camera lens system which removes the scattered laser light. Wavelengths in the acetone fluorescence spectrum (350-650) are passed by the lens with minimal attenuation.

Prior to entering the ICCD camera the emitted light is filtered by a long pass colored glass filter. This filter was selected after an analysis of the background noise created by the PLIF technique. This experiment consisted of impinging the laser sheet on several surfaces and collecting the fluorescence that occurred with a monochromater. The wavelength spectrum of the fluorescence from quartz, sapphire, aluminum, anodized aluminum, and painted aluminum were collected along with a fluorescence spectrum of acetone. The data acquired from this experiment confirmed that the bulk of the fluorescence from the sapphire cylinder is at shorter wavelengths than the acetone fluorescence. A long pass filter was selected such that the sapphire fluorescence was reduced by an order of magnitude with only a minor reduction in acetone fluorescence signal. This analysis also led to the anodizing of most of the aluminum parts in the engine, as the anodizing was found to greatly reduce the level of fluorescent emissions from aluminum surfaces.

After this filtering, the fluorescence images are collected and focused by a Nikkor 55 mm lens set, onto an intensified CCD camera. The ICCD camera is a Princeton Instruments, 14 bit system with a 576 by 384 pixel array. The timing is controlled by a Princeton Instruments pulse generator. This combination allows for gate widths down to 5 ns. The pulse generator accepts the output pulses provided by the laser Q-switch trigger to time the camera gate. The images collected by the camera are downloaded by a Princeton Instruments ST-130 controller to a laboratory PC, where the images are processed and stored. The repetition rate of the camera is on the order of one second, which limits the collection of images to one image every several cycles. The orientation of the camera with respect to the engine is shown in Figure 5.2. The camera can be
programmed such that images are collected over only certain portions of the CCD array. This was used to limit the image size to the size of the cylinder. In addition, two pixels were binned together in each direction. This resulted in a courser image, but increased the signal level by a factor of four. The binning of pixels was done to increase the signal level without increasing the camera gain. With the pixels binned together the resolution of the system is 0.5 mm, which corresponds closely to the sheet thickness such that the pixels are spatially averaged over equal scales in all three dimensions. In addition to increasing the signal level, binning reduces the image size, expediting storage and processing. A set of 50 full images with no binning requires 18 Megabytes of storage space, while a set of binned images clipped to the cylinder size requires only 3 Megabytes. This reduces processing time by more than an order of magnitude.

5.2.4 Gas Supply System

The gas supply system for the engine provides both fuel to the fuel rail and air to the intake manifold. The fuel supply portion of the system must monitor and control not only the natural gas flow but also the flow of acetone for the PLIF experiment. The air supply system has the capability to accurately measure and control the flow rate of air through the engine.

The natural gas supply for the engine is a commercial grade of bottled gas. It contains roughly 93% methane, with the balance being propane, butane, and nitrogen. In order to perform PLIF measurements, the natural gas must be seeded with acetone. This is accomplished by passing the natural gas through a small pressure vessel filled with liquid acetone. The natural gas which passes through this tank collects an amount of acetone which is dictated by the vapor pressure of acetone. At room temperature, this is approximately 27 kPa. Experiments were run with an acetone concentration of approximately 5% by volume. A surge tank is also used in the natural gas line to limit the magnitude of oscillations which occur due to the periodic nature of the injection event. The pressure in the natural gas supply system is monitored by a pressure transducer, and is maintained at 35 psig. The flow rate of natural gas is monitored with an electronic
mass flow meter from Davis Instruments. This device uses the known specific heat of the
gas to convert a temperature change across a heating element into a mass flow rate. The
accuracy of the device is ±0.1 liters per minute. The mean mass flow rate of natural gas
through the engine is 2.6 liters per minute. This is one of the main sources of error in the
quantitative PLIF images, as discussed in Chapter 6. While the flow rate of acetone has a
direct impact on the PLIF experiments, its exact level need not be known. This is the
case due to the image processing scheme which has been developed for quantifying the
images. This process makes the final images independent of acetone flow rate, as well as
laser power and camera gain.

Several problems were encountered early in the research in terms of condensation of
acetone in the intake manifold. The problem was discovered by viewing images taken
several cycles after an injection event which still showed acetone fluorescence. This
phenomenon was caused by acetone condensation resulting from the temperature drop
across the injector. Flow through the injector is choked, leading to very high velocities at
the exit. As the temperature of the natural gas/acetone mixture dropped through the
injector, the vapor pressure of acetone was reduced and condensation occurred. This
problem was overcome by heating of the fuel rail assembly to 60 degrees Celsius. With
the incoming mixture at this temperature, the stream is essentially at ambient temperature
after passing through the injector. Experiments using Rayleigh/Mie scattering confirmed
that droplet formation out of the injector ceased with the increased fuel rail temperature.

The engine air supply system consists of a particulate filter, a water trap, a digital
mass flow meter, a throttling valve and a surge tank. Air enters the system through the
two filters to eliminate dust and water vapor. The flow rate of air is monitored by a mass
flow meter similar to the one described previously. The error in this meter is ±0.2 lpm
while the mean mass flow rate of air is 26.4 lpm. The flow rate is controlled with a
throttling valve, as the throttle body has been removed from the engine. Air flowing
through the throttling valve passes through a 5 gallon surge tank. This serves to minimize
fluctuations in intake manifold pressure. The intake manifold pressure is monitored by a
strain gauge type pressure transducer.
5.2.5 Pressure Measurement

The cylinder pressure during the compression and expansion strokes of the engine was recorded by means of a piezo-electric pressure transducer. This transducer, which is produced by Kistler instruments, is integral with the engine spark plug and thus does not require any modification to the combustion chamber geometry. The output charge from the transducer is amplified by a charge amplifier and converted into a voltage. This voltage is then sampled by the digital data acquisition system described in the next section. The pressure data was acquired with a resolution of 0.6 crank angle degrees in order to produce accurate calculations of combustion parameters. The pressure was measured only during cycles in which combustion occurred, and only during the compression and expansion strokes. This minimized the storage requirements for the pressure measurement data. The accuracy of the cylinder pressure data is discussed in Chapter 7.

5.2.6 Experiment Timing

The timing of the engine operation and image acquisition process were controlled by a timing board supplied by National Instruments. An additional data acquisition board was utilized to monitor the experiment and collect the cylinder pressure data. The trigger signal for the timing board was provided by a camshaft mounted incremental encoder. This encoder produces 1200 pulses per camshaft revolution leading a resolution of 0.6 crank angle degrees. The timing board performs 5 main functions. In order to collect images at a specific crank angle during any given cycle, a pulse must be sent to the laser to provide timing for the Q-switch pulse. The engine was skip fired, for reasons discussed later, and this presented some additional difficulties for the laser triggering. The laser produces its maximum and most consistent power output at 10 Hz. Because the engine was fired only every tenth cycle, pulses must be sent to the laser in both fired and
unfired cycles to maintain the repetition rate in the range of 10 Hz. Thus the timing board’s first task is providing a pulse to the laser during each crankshaft revolution, at a given crank angle.

The second task for the timing board concerns the gating of the ICCD camera. While the timing of the camera gate with respect to the laser pulse is controlled by the pulse generator as described earlier, it is undesirable to gate the camera with every laser pulse. Because the laser will produce pulses during both the intake stroke and the expansion stroke, the camera must be prevented from gating during the expansion and exhaust stroke laser shots. The timing board was utilized to provide an inhibit pulse to the camera during strokes in which images were not to be taken. The third and fourth tasks of the timing board are controlling the phasing of the ignition and injection events. The board provided an output pulse which triggered the initiation of the ignition event. The injection event must be controlled in terms of both timing and duration. The timing board was used to signal both the opening and closing points to the injection driver. The final task accomplished by the timing board is triggering of the data acquisition board for the collection of pressure data. As discussed previously, the pressure data was collected only during firing cycles, and only during the compression and expansion strokes. This is due to the large number of points required to accurately characterize the pressure development process.

The ignition system of the engine was developed in the laboratory specifically for this experiment. It is based on a very simple power MOSFET circuit. The ignition system is powered by a variable voltage power supply and triggered by the National Instruments timing board. The spark jitter in this system was negligible, and the spark energy was controlled by controlling the coil charging period.

5.3 Experimental Procedure

The purpose of this section is to describe in detail the methods that were used to acquire the experimental results presented in Chapters 6 and 7. The general method of acquiring PLIF images is given followed by a description of the actual process used in the
given experiments. This is followed by a description of the method used to capture cylinder pressure traces, and finally a presentation of the design of the cylinder pressure experiment.

5.3.1 Image Acquisition

Images detailing the distribution of fuel within the engine cylinder at several crank angles as a function of fuel injection timing were collected. In order to achieve high quality results careful control of the entire image acquisition technique had to be maintained.

As described previously, the excitation source for the PLIF experiments was a frequency quadrupled Nd:YAG laser. While the laser can be set to produce output pulses at a given repetition rate, in order to phase the laser with the engine, triggering signals were sent by the OAE control computer. This computer takes as its only input the encoder pulses from the engine camshaft. Each time the engine was started, it was returned to TDC of the intake stroke so that the computer would have an absolute starting point from which to count pulses from the relative encoder. The encoder generates 1200 pulses per camshaft revolution, resulting in a resolution of 0.6 crank angle degrees. The laser timing was set on the OAE computer such that a pulse was generated once per crankshaft revolution at the desired image timing. This pulse was sent to the laser, where it was used to generate a laser pulse which quickly followed the incoming pulse. Correction was made in the lag between the pulse to the laser and the laser output by observing the piston location in the acquired images and comparing it to the known location at the desired image timings. The pulse sent to the laser was adjusted so as to generate a laser pulse at the desired crank angle. As mentioned before, the laser power was set well below its maximum value. The actual output energy of the laser was not monitored but was in the 10-20 mJ/pulse range.

The laser electronics are capable of generating an output TTL signal coincident with the output of laser energy. This TTL pulse was used to trigger the ICCD camera pulse generator. This pulse generator controls the gating of the camera in terms of the timing of the gate signal and the duration of the gate signal. The timing of the gate signal was set to
be coincident with the laser pulse. The duration of the gate was set at 50 nsec. This timing was chosen by testing experimentally the signal to noise ratio as a function of gate duration. The fluorescence lifetime of acetone is very short (≈ 4ns). The relatively long gate time allows for the decay of all excited molecules as well as accounting for the deviation of the gate pulse from a perfect step function.

The pulse generator sends a high voltage pulse to the ICCD camera which activates the image intensifier. Collection of photons when no gate signal is applied is suspended due to the high blocking power of the intensifier when it is deactivated. When the intensifier is activated, the detector is exposed to the intensified output which is proportional to the intensity of the emitted acetone fluorescence. The gain of the image intensifier is adjustable on an arbitrary, non-linear scale from 0 to 10. For these experiments the gain was set to 8.0. This number was chosen to create the maximum signal without inducing the noise which is generated in the intensifier at higher gains. As described in the image processing section, the gain setting for the camera is removed from the images in the quantification process.

After the gate signal from the pulse generator has ceased, the camera controller downloads the pixel intensity values to the image collection computer, where they are stored for processing. The camera image acquisition rate is limited to roughly 1 Hz by the speed of the downloading process. This acquisition rate, in conjunction with the laser repetition rate, prevent the acquisition of more than one image per engine cycle.

All images were collected in sets of 50 for statistical purposes. For each set of images, the engine was first set to TDC of the intake stroke, and then accelerated from rest to 700 rpm over a period of 10 seconds. After the engine reached its desired operating speed, the system was allowed to reach steady state operation. This includes the laser, whose pulse energy increases steadily before leveling off after about 30 seconds. This also allowed time for the intake manifold pressure to reach its equilibrium value and for the flow through the fuel injection to reach a constant level. After 1 minute and 30 seconds, the acquisition of images was begun. An image was acquired every 10 engine cycles under motored operation. The engine was not fired during the collection of
the PLIF results, as the latest image acquisition crank angle is at 320 degrees ATDC during the compression stroke prior to ignition. Some deviation of the engine flow can be expected during motored operation as the backflow associated with high temperature exhaust gases is not present. This is an unavoidable consequence and is not thought to significantly alter the results. After a set of 50 images was collected, the images were stored by the computer, the engine was reset to TDC, and the process was begun again.

5.3.2 Design of the PLIF Experiment

The PLIF portion of this research was designed in such a way as to provide information concerning both the mixture formation process and the final state of the mixture prior to ignition. This initially involved measurements in 9 cylinder planes, for 7 different injection timings and at 4 different crank angles. The 9 original cylinder planes included 3 locations in the tumble plane, 3 locations in the cross tumble plane and 3 locations in the swirl plane. The off center locations in the tumble and cross tumble planes were dropped due to optical noise problems. Attempts to bring the laser sheet into the engine cylinder off of the cylinder centerline led to a large amount of cylinder fluorescence. Reflections at both the inside and outside cylinder surfaces led to fluorescence in the sapphire which create a very low signal to noise ratio. After processing, these image planes were dropped from the experimental data set. The final set of images consisted of data on the centerline of the tumble plane (location 2), the centerline of the cross-tumble plane (location 5) and three locations in the swirl plane (locations 7, 8 and 9). Locations 7 and 8 are not presented in this work. The cylinder measurement locations are shown in Figure 5.3.

The 7 fuel injection timings which were initially explored spanned the entire engine cycle. The final number was reduced to 5 when it was determined that any injection timing during the compression, expansion, or exhaust strokes produced virtually the same in cylinder fuel distribution. This topic is discussed in detail in Chapter 6. The final 5 injection timings which were tested were 360 degrees BTDC (injection at the end of the compression stroke), 0 degrees ATDC (injection at the beginning of the intake stroke)
and injection at 30, 60 and 90 degrees ATDC (during the intake stroke). These 5 injection timings were found to span the range of mixture formation regimes present in this engine.

Images were acquired at 4 different crank angles so as to provide a time history of the evolution of the mixture field. The crank angle timings which were selected were 90 degrees ATDC (the middle of the intake stroke), 180 degrees ATDC (the end of the intake stroke), 270 degrees ATDC (the middle of the compression stroke), and 320 degrees ATDC (near the end of the compression stroke). Complete sets of images were acquired at each of these times, for each injection timing, in each of the image planes. All images were taken during different cycles due to equipment limitations, such that there are no instantaneous images showing the mixture development in one given cycle.

Eight sets of 50 images were acquired at each crank angle location in each cylinder plane. These eight sets consists of: 1) a background image, 2) a lean flat field image, 3) a rich flat field image, 4) images with injection at 360 degrees BTDC, 5) images with injection at 0 degrees ATDC, 6) images with injection at 30 degrees ATDC, 7) images with injection at 60 degrees ATDC, 8) images with injection at 90 degrees ATDC. The background images were taken with no fuel flow, but with the laser and camera at their normal operating points. These images were used to remove extraneous noise from the other images by subtraction. The lean and rich flat field images were created by introducing the fuel/dopant mixture well upstream (= 2 meters) of the intake manifold. In this way, the fuel and air are essentially perfectly mixed by the time they reach the cylinder. These images provided a basis from which several pieces of information could be drawn. They show the flat field in terms of the laser sheet profile, camera gain, and cylinder curvature. That is, they reveal the intensity which is assigned to each pixel for a known fuel concentration. The rich and lean flat field images were taken with more and less than stoichiometric fuel flow rate respectively. The flow rate for each case was recorded from the mass flow meter. As described later, this information allows for the images to be quantified without knowledge of the temperature and pressure dependence of the acetone fluorescence.
All 8 sets of images were taken in a given plane at 90 degrees ATDC after which the piston was removed and the cylinder cleaned. After reassembly, the same experiment was repeated at a crank angle of 180 degrees in the same plane. The procedure was the same for the 270 and 320 degree ATDC crank angle timings. Collecting the images for a single plane typically took 2 full days. After a given plane was done, the laser and camera were adjusted to make measurements in another plane. The cylinder head assembly was rotated 90 degrees between the tumble and cross tumble plane images. This was necessary in order to bring the laser sheet in from the proper direction and collect images without changing the camera location. Swirl plane images were taken after the tumble and cross tumble planes were complete. The same cylinder wall cleaning procedure was maintained throughout the swirl plane measurements despite the fact that viewing through the cylinder wall was not necessary. The PLIF results acquired as described here are presented in Chapter 6.

5.3.3 Cylinder Pressure Data Collection

The cylinder pressure data was collected after the completion of all of the PLIF experiments. The cylinder pressure data was collected from the Kistler cylinder pressure transducer described previously. Acquisition of the cylinder pressure data was triggered by the camshaft encoder at a resolution of 0.6 crank angle degrees resulting in 600 points being collected during the compression and expansion strokes. Pressure traces were collected only during the firing cycles which occurred every tenth cycle. Skip firing leads to several differences between typical engine operation and that seen in the OAE. Most prominent is the absence of residual gas. Flame speeds are significantly higher in the OAE than would be expected in actual engine operation without the essentially inert residual gas. In addition, heat transfer to the walls is higher in the OAE as the wall temperature never reaches the level occurring during idle in production engines. Despite these differences, the data collected allows for comparison of combustion quality where the only difference between test cases is the fuel injection timing. The data acquisition system collected voltages which were proportional to cylinder pressure with a known gain.
on the charge amplifier. In order to find the absolute pressure, the results must be pegged to a known value during the cycle. For all of the results presented here, the pressure at the beginning of the compression stroke was set to the intake manifold pressure of 45 kPa. Motored traces were also collected to assure proper phasing between the collected cylinder pressure trace and the crankshaft angle in the engine.

5.3.4 Design of Cylinder Pressure Experiment

The data to be presented in Chapter 7 was collected in such a way as to ensure statistically significant data. One hundred and seventy five cylinder pressure traces were collected for each of 4 different fuel injection durations for each of the five fuel injection timings. While the full sets of data included 175 pressure traces, these were collected in 7 sets of 25 images each. This was done to assure that any variation in engine operating condition between test runs could be observed in the data. That is, if there is a significant difference in the pressure traces between one set of 25 images and another but not within a single set, then some operating parameter must have changed. This was found to be the case in several of the data sets. In each case, the mean pressure, maximum cylinder pressure and the standard deviation of the maximum cylinder pressure was determined. After collecting all the sets of cylinder pressure data, sets which proved to be outliers by the Dixon test were removed from the data set and the set was re-run. Of the 140 sets of traces, this was done with 6 sets.

For any given fuel injection duration, the order of testing of the five different injection timings was chosen randomly for each test run. In this way, the differences measured from different fuel injection timings could be distinguished from random error generated by varying engine parameters. This was not found to be a problem, and the order of data collection was found to be inconsequential. The four different fuel injection durations were chosen to span the range from stoichiometric to the lean limit of the engine. Due to error in the measurement of fuel flow rate, determination of the exact fuel injection duration which leads to stoichiometric operation is difficult. From observation of the pressure traces, it is certain that the stoichiometric case falls somewhere between the 4.2
msec and 4.6 msec injection durations. The 3.4 msec injection duration was the leanest case in which misfires were not a major problem. The maximum brake torque timing for each equivalence ratio was found by running tests with different ignitions timings and calculating the output torque from cylinder pressure data. This led to ignition timings of 6 degrees BTDC for 4.6 msec injection, 9 degrees BTDC for 4.2 msec injection, 12 degrees BTDC for 3.8 msec injection and 15 degrees BTDC for 3.4 msec injection. As will be demonstrated in chapter seven, the MBT timing for any equivalence ratio is also a function of the injection timing, as this influences the mean ignition delay. Despite this fact, all of the injection timing cases were run at a single ignition timing. Sample cylinder pressure traces, as well as a statistical analysis of the traces are given in Chapter 7.
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Table 5.1 Engine Design and Experimental Parameters
Figure 5.2: Optical Access Engine Laboratory Photos
Swirl Plane Locations

Tumble and Cross Tumble Plane Locations

Figure 3: PLIF Measurement Planes
CHAPTER 6

PLANAR LASER INDUCED FLUORESCENCE RESULTS

6.1 Introduction

The purpose of this chapter is to describe both the PLIF results and the means by which these results were obtained from the raw images acquired in the engine. While the raw images provided a qualitative look at the mixture formation process they required extensive processing to make them quantitatively accurate. The final processed images which are presented here can be divided into mean images and instantaneous images. The mean images are an ensemble average of the 50 images in a given set. The purpose of the mean images is to remove the small scale inhomogeneities so that the bulk maldistribution at the time of the image can be seen. The instantaneous images provide a measure of the small scale inhomogeneities as well as yielding information concerning the variation of the fuel distribution from cycle to cycle. In addition to the description of the image processing scheme and the presentation of the processed images, a statistical analysis of the images that is used in Chapter 8 in determining the correlation between fuel distribution and combustion quality, is presented.

6.2 Image Processing Method

In order to create equivalence ratio maps from the raw images that were collected, corrections had be made to offset several factors that cause deviation of the PLIF intensity from the ideal intensity given in Equation 3.2. Additional information is required to determine the absolute equivalence ratio from the arbitrary intensity units provided by the
images. The major factors which must be accounted for in correcting the PLIF images are: the pulse to pulse variation in laser energy, the spatial nonuniformity in the laser sheet, the distortion caused by cylinder curvature, the flat field gain of the individual pixels in the ICCD camera, and the fluorescence emitted from sources other than the acetone dopant. These factors and the conversion of images into equivalence ratio maps are described in the following sections.

6.2.1 Normalization

The term normalization as it is used here refers to the correction of instantaneous PLIF images for shot to shot variation in the laser pulse energy. While the mean power of the laser is stable over long periods of time (varying less than 1%), the energy contained in a single shot can vary significantly. This condition is worsened by the triggering of the laser from an external source. When triggered internally, the time between pulses is very accurately fixed, and the relation between the flashlamp pulse and the Q-switch trigger is well controlled. When triggering with a signal generated from the OAE rotation, instantaneous variations in engine speed cause triggering signals which are less accurate in time. The result is shot to shot variations in laser energy which can be as high as ±10%. As described previously, the intensity of the fluorescence signal is directly proportional to the incident radiation intensity. Thus, if left uncorrected for laser power fluctuations, the accuracy of the images would be very poor.

The traditional method of correcting for laser pulse energy fluctuations is to measure the pulse energy directly and scale each image with its known laser energy. A simplified version of this technique has been used to correct the OAE images. As discussed previously, the engine cylinder produces fluorescence as the laser sheet passes through it. While much of this fluorescence is blocked by the long pass filter, a moderate portion of what could be considered optical noise reaches the camera. Stagnant cell experiments confirmed that the level of this "optical noise" was very linear with laser pulse energy. This was done by filling the cylinder with an acetone/air mixture, and collecting PLIF images resulting from impingement with the laser sheet. As the cell
contents were constant the mean intensity should not vary from one shot to the next. The actual mean values of these images had a coefficient of variation near 8%. By measuring the intensity of emission from the cylinder wall through which the laser sheet entered the cylinder, and scaling the images according to this intensity, the coefficient of variation of means of the images was reduced to less that 1%. In this way, it was determined that the fluorescence in the cylinder wall was sufficiently linear with laser power as to be used for image normalization.

Thus the first step in the image correction process, as shown in Figure 6.1 is the normalization of images to correct for laser energy fluctuations. Each set of fifty images was scaled based on the level of fluorescence from the cylinder wall at the entrance point of the laser sheet. The mean pixel value within this region was calculated for each image. Each pixel in the image was then multiplied by a given constant divided by this value. This results in the new mean in the normalization region being the given constant. The constant used in this process was 3000, as it was close to the typical mean found in this region of the uncorrected images and thus did not radically change the scaling of the images. The same process was repeated for all image sets including the background and flat field images.

6.2.2 Background Subtraction

After the images were normalized, the next step in the correction process was subtraction of the background signal. As described earlier, for each set of images taken at a given location and crank angle, a background set of images was taken with no flow of fuel into the cylinder. Ideally with no fuel flow, the intensity measured by the camera would be zero at each pixel. This is not the case, due almost entirely to fluorescence of the optical materials and deposits on the cylinder walls. While the very short camera gate time eliminates any ambient light from the images, the UV laser light produces fluorescence in several of the materials in the engine. Prior to beginning the PLIF image acquisition, a significant amount of time was spent reducing the background signal. This included anodizing of aluminum surfaces and modifying the cylinder head to operate
without liquid lubrication. Engine oil and aluminum were both found to be major contributors to background noise. The noise produced by the sapphire cylinder was reduced as described earlier, by a long pass filter. Despite these efforts, the background noise signal remained at a level which required processing for removal.

The process of subtracting the background from the images was very straightforward. The set of background images was ensemble averaged to raise the signal to noise ratio, and then algebraically subtracted from each individual image in the injected image and flat field image sets. While this method greatly reduces the noise level in the images, some noise does remain in areas where the background signal varies significantly from one cycle to the next. Examples include the area in the swirl plane images where the laser sheet impinges on the still open intake valve. Deposits on the valve face and the valve material itself create a significant amount of noise which varies from shot to shot. This will be apparent in images presented later in this chapter. A second example of incomplete background removal is at the top of the piston ring travel. While the piston rings used produced very few deposits, the ring acts as a scraper, moving all the deposits to the area just above the top of the ring travel. This is also clear in the images shown, and becomes worse in the images taken the further from the background images. That is, because the images where injection occurred at 360 degrees BTDC were taken one set after the background, and the images taken with injection 90 degrees ATDC were taken five sets after the background, the level of ring deposits in the later images differs more from the background level than the earlier set does. While this can be seen in some of the images, the small affected area and the minor deviation from the surrounding pixels does not significantly affect the image statistics which were used to analyze the images.

6.2.3 Flat Field Correction

After normalization and background subtraction, the next image correction which was necessary was accounting for variation of pixel intensities not due to inhomogeneous charge. This is referred to as the flat field of the imaging system and accounts for several different factors. The most important contributors to the non-uniform flat field in this
experiment were the intensity profile of the laser sheet and the curvature of the engine cylinder. Each of these factors leads to the collection of a non-uniform image even in a perfectly distributed mixture. The goal of flat field correction is for the imaging system to produce a uniform intensity over all the pixels when measurements are made in a perfectly well mixed system. While this is an idealized task, the methods which have been used produce a highly uniform field and also result in the ability to measure the error that is created due to the non-uniform flat field. The techniques by which this error was analyzed and how the error was removed from the image statistics is described in section 6.5.1.

The method by which flat field correction was achieved is shown in Figure 6.1. Both a lean and rich flat field image set were taken for each image location and crank angle. The lean flat field images were used for flat field correction. The flat field images were ensemble averaged after being normalized and having the background subtracted from them as described earlier. This mean flat field image then had each pixel divided by the mean pixel value in the image. This yielded an image whose mean was one and which showed the intensity variations that the collection system would record when collecting fluorescence from a well mixed charge. Each image in each image set was then divided by the flat field image on a pixel by pixel basis. Regions which appear as less than one in the flat field image are intensified in the corrected images. Regions which appear as greater than one in the flat field image are reduced in the corrected images. In this way, the center area in the images which is brighter due to the laser sheet profile is reduced while the edges which have lower intensity are increased. Because the mean value of the flat field image is one, the overall mean of the corrected images is not greatly affected by this process. In addition to the laser sheet profile, the curvature of the cylinder is corrected for in terms of intensity, as the dimmer areas at the edges of the cylinder are raised to the level of the center. The accuracy of the flat field correction was tested by correcting the rich flat field using the lean flat field. The variations due to laser sheet distribution and cylinder curvature were reduced from ±25 % in the uncorrected rich flat field to ±2 % in the corrected rich flat field.
6.2.4 Quantification

After correcting the PLIF images by normalization, background subtraction and flat field variations, the images were still in arbitrary units. In order to make use of the image information on a quantitative level, the images had to be converted to units of equivalence ratio. This was accomplished using the corrected lean and rich flat field images.

Previous experiments have confirmed the linearity of the PLIF signal with concentration for constant pressure and temperature. This would allow the conversion of image intensities to equivalence ratio using stagnant cell data. This approach was not chosen for several reasons. Most importantly, the PLIF signal is dependent on both temperature and pressure. As discussed in Chapter 3, this dependence is a matter of some debate. In order to create equivalence ratio maps from test cell data, the pressure and temperature conditions from the engine would need to be replicated. This is fairly straightforward for images taken during the intake stroke as the cylinder pressure and temperature can be approximated from the intake manifold pressure. This level of simplicity does not continue into the compression stroke, as the pressure and temperature are both changing rapidly. In order to use test cell data, the pressure and temperature in the cell would need to be well above ambient to replicate these cases. In addition to these problems, stagnant cell data can be unreliable as decomposition of the dopant can occur with repeated laser pulses. If sufficient energy is absorbed by the cell gases and the gases are not replaced between shots, the level of dopant may not remain constant, leading to inaccurate measurement of dopant concentration. Finally, the use of test cell data requires the concentration of the dopant be known in an absolute sense in both the test cell and the engine. This is difficult to achieve with the simple bubbling system employed in the experiment.

The above described problems with test cell calibration were overcome by measuring the fluorescence intensity in the engine for mixtures of known equivalence ratio. As discussed previously, the rich and lean flat field images are created by
introducing a known fuel flow rate well upstream of the intake manifold, thus creating a homogeneous fuel air mixture in the cylinder. Unlike the test cell data, this mixture is necessarily at the same pressure and temperature as the injection images as the images are taken at the same crank angle. The rich and lean flat field image means provide a linear intensity map by which image intensities can be correlated with equivalence ratio. While this technique is much more accurate than test cell data, it is still the major source of error in these experiments. This is due to the inaccuracy with which the fuel flow rate is measured in the engine. The fuel flow rate is known only within ±5 % of its true value. This results in a similar error in the image intensities. The error is apparent in some of the images presented later in the chapter. The error induced by this measurement has been largely removed from the statistical analysis, as most of the statistics are based on values relative to the image mean, or comparisons within a set in which the same fuel flow rates were used in each set.

6.3 Bulk Fuel Maldistribution

The distribution of fuel in the cylinder can be viewed as a sum of contributions from bulk maldistribution and a contribution from small scale inhomogeneities. During the intake stroke and the early phases of the compression stroke, the bulk maldistribution of the fuel overshadows the contribution of small scale inhomogeneities. At some point during the compression stroke, a combination of convective and diffusive mixing has reduced the level of bulk maldistribution to the same level as the small scale maldistribution. By the time of ignition, the fuel distribution is dominated by small scale structures which are superimposed on any remaining large scale structures. The images presented in this section are meant to display the bulk maldistribution. The images have been created by ensemble averaging of the 50 images in each image set. The purpose of this section is to detail the mixture formation process for each injection timing in each of the image planes. The analysis is qualitative, with statistical quantitative analysis following in section 6.5.
6.3.1 Injection 360 Degrees BTDC of the Intake Stroke

The first test injection timing was at top dead center of the compression stroke. At this point the intake and exhaust valves are closed, and the fuel is being injected into an essentially stagnant intake manifold. It was thought prior to experimentation that this would produce a uniform in-cylinder fuel distribution due to the "premixing" time which is provided by the passing of the expansion and exhaust strokes before the intake event. As shown in this section this is not the case, and the amount of mixing which occurs in the intake manifold during this time is minimal.

Figure 6.2 shows the mean fuel distribution on the centerline of the tumble plane as a function of crank angle for this very early injection case. The intake valves are located in the upper right hand corner of each image and the bottom boundary of each image is the piston face. The active intake valve would be coming out of the page while the inactive valve would be behind the plane of the page. At 90 degrees ATDC the cylinder volume is completely filled with rich mixture. The majority of this mixture is off the scale meaning the mixture contains at least 50% excess fuel. This phenomenon is caused by the very early injection and the lack of mixing in the intake manifold. The injected fuel is directed towards the back of the intake valve by the injector, where it remains until the valve opens. Upon valve opening, this rich mixture is drawn into the cylinder with very little entrainment of intake air.

The image to the right of the 90 degree ATDC image shows the distribution when the piston has reached bottom dead center. Due to the four valve configuration and the intake manifold geometry, a strong tumble vortex is present in the cylinder at this point. The vortex is rotating in the counter-clockwise direction with respect to the images shown. The majority of the intake flow travels over the top of the intake valve, through the combustion chamber, and down the opposite wall of the cylinder. Very little mixture travels down the wall directly under the valve. The image at BDC shows the rich mixture from the previous frame being convected with the tumble vortex down along the piston face and up the opposite cylinder wall. With the rich fuel mixture cleared from the intake
port, very lean mixture is following the same path, up through the combustion chamber and down the opposite cylinder wall. At this point the charge is stratified from 50% excess fuel on the intake side of the cylinder to 50% less than stoichiometric fuel on the exhaust valve side.

Following the fuel flow to the lower left hand image, showing the distribution at 270 degrees ATDC, is somewhat more difficult as the tumble vortex is being broken down by the piston motion and the strong swirling flow generated by the deactivated intake valve configuration is beginning to dominate. In this plane, the richer mixture is located in the center of cylinder surrounded by lean mixture at the edges. The error in the mean fuel distribution scaling due to the limited accuracy of the fuel flow measurement may be causing the entire distribution to be shifter by ±5%. Despite this, the relative distribution is highly accurate, and a high level of maldistribution still exists.

The final image, in the lower right hand corner, was acquired at 320 degrees ATDC fairly close to the time of ignition. Clearly, the mixing mechanisms in the engine have greatly reduced the level of bulk maldistribution, but a measurable amount of stratification is still present. These images detail only the bulk stratification, so that while the images at 320 degrees ATDC may appear fairly uniform, this means only that whatever stratification is present is largely isotropic and homogeneous and is removed in the averaging process. Instantaneous images shown later clearly reveal this fact.

Figure 6.3 presents similar images to those just described but at the cylinder centerline of the cross tumble plane. In these images the intake valves are toward the viewer from the plane of the page and the exhaust valves are behind the plane of the page. The active intake valve is on the right hand side of the images. The first image, at 90 degrees ATDC is similar to the image in the tumble plane except for the slightly leaner area under the deactivated intake valve. This is as would be expected as the flow enters through the active intake valve forcing the residual gas remaining from the previous cycle into the area under the inactive intake valve. The image in the upper right hand corner shows the distribution at 180 degrees ATDC. As in the tumble plane, the rich area remains largely intact. Additional information provided in this image shows that the rich
area does not fill the piston face area, but is confined to the volume below the active intake valve. Combining the two images, it is clear that the rich area is strongly focused under the intake valves and more specifically under the active intake valve. Once again, at bottom dead center mixture stratification ranges from +50% to -50% fuel from stoichiometric. The progression to 270 degrees and 320 degrees is more complicated due to the interaction of swirl and tumble flows in the engine. Once again, though, a measurable level of mean mixture maldistribution is present all the way into the last frame.

The images in Figure 6.4 demonstrate the same injection timing but near the cylinder head in the swirl plane. The intake valves are at the top of each image, with the active intake valve being on the right. The images in this plane confirm information from the tumble and cross tumble planes. At 90 degrees ATDC the cylinder is filled with very rich mixture, the leanest portion of which is under the inactive intake valve. As the piston comes to bottom dead center the rich area has moved under the intake valves only, and is most concentrated under the active intake valve. Of note is the close agreement at the intersection of the three planes. For example a vertical slice through the swirl plane image at 180 degrees ATDC matches well with a horizontal slice from the top of the tumble plane centerline image. This bodes well for the accuracy of the experiment as the images were taken several months apart with significant changes being performed on the optical arrangement between the experiments. The final two images in the swirl plane show once again that while the overall mean distribution makes significant improvement from 180 degrees ATDC to 320 degrees ATDC, measurable levels of maldistribution still exist in the mean images.

6.3.2 Injection at TDC of the Intake Stroke

The images shown in Figure 6.5 detail the mixture formation process at the centerline of the tumble plane for injection at TDC of the intake stroke. The only difference between these images and those shown in Figure 6.2 is the injection timing. Injection at TDC of the intake stroke is probably the most similar to the injection timing
for a gasoline engine, and is the injection timing used in Honda's natural gas prototype. The most distinctive feature of these images is their similarity to those acquired with injection at 90 degrees ATDC. The cylinder is once again filled with very rich mixture. This is due in part to the brevity of the injection event at idle speed. The injection event lasts just under 5 msec, while the intake stroke lasts 43 msec, such that the injection event which starts at TDC is completed by 20 degrees ATDC. Due to the valve lift profile, the bulk of the intake air flow is just beginning as the injection event ends. Thus, injection at TDC of the intake stroke and injection at TDC of the compression stroke produce very similar results. There are two main differences between the two cases, and these can both be seen in the bottom dead center image in the tumble plane. The most obvious is the higher level of optical noise in the later images. As this set of images was taken further away from the background image, temporally speaking, cylinder wall deposits are increasing and are not being removed by background subtraction. The locations of these discontinuities are clear from the two images. The other difference between the images is a slight difference in the shape of the rich area at BDC. The rich area with injection at TDC is slightly behind that of injection at TDC of the compression stroke. This is as would be expected, as the fuel rich volume enters the cylinder slightly later (between 0 and 20 degrees ATDC instead of immediately upon intake valve opening). Very little difference is seen in the 270 and 320 degree ATDC images besides perhaps the shape of the rich region.

The images shown in Figure 6.6, at the centerline of the cross-tumble plane, are very similar in nature to those found in the earlier injection case. The rich area is still under the active intake valve. Note however the higher level of stratification at 270 degrees ATDC. This fact plays prominently in the final distribution, and in the combustion process with this injection timing, as will be shown in Chapter 7. The main difference which can be seen between the two injection timings in the swirl plane is the location of the rich area more under the inactive intake than under the active intake. This once again is likely due to the slightly retarded entry of the rich fuel volume into the cylinder. As will be demonstrated in the statistical analysis, the fuel distribution in the
images at 320 degrees can be deceptive due to the low resolution used to display the images. With the majority of the pixels in the stoichiometric range, a small offset resulting from inaccurate image quantification can lead to the appearance of fuel rich or lean areas. Judgment on the distribution this late in the stroke should be delayed for analysis with image statistics given in section 6.5.

6.3.3 Injection 30 Degrees ATDC of the Intake Stroke

Images showing the fuel distribution as a function of crank angle for injection at 30 degrees ATDC of the intake stroke are the first to show major deviation from the rich mixture followed by lean mixture cases shown thus far. In Figure 6.8, the fuel distribution at 90 degrees ATDC is entirely rich of stoichiometric, but much less so than in the previous cases. The flow pattern of the fuel is also evident for the first time. The richest regions show flow down the cylinder wall opposite the intake valves as well as down the cylinder wall on the side of the intake valves. Note the collision of these two regions at a kind of stagnation point on the piston face toward the right edge of the image. For this injection timing, injection occurs between 30 and 50 degrees ATDC of the intake stroke, resulting in significantly more entrainment of air as the fuel enters the cylinder than in the early injection cases. The result of this improvement is shown in the BDC image where the range of stratification has dropped from the full scale to the center three bins. The level of noise from cylinder wall deposits has again increased as this case was taken later in time than the previous cases. Once again it should be noted that the deviation of the 270 and 320 degree ATDC images should be viewed with some skepticism in terms of the mean value of the image due to both the display resolution and the accuracy of the quantification method.

Images in the cross tumble plane for the 30 degrees ATDC injection case, in Figure 6.9, show the improvement in the distribution from the earlier injection cases. At 90 degrees ATDC only a small portion of the distribution is off scale, with some portion even falling in the stoichiometric region. The rich spot under the active intake valve is not as pronounced as in earlier cases, but is still clearly present. Images at 270 and 320
degrees ATDC show improvement from similar cases with earlier injection. Figure 6.10 shows similar results in the swirl plane, with lower levels of stratification at the earliest image locations. Of note here is the appearance of poorly removed background signal from the active intake valve. At 90 degrees ATDC the intake valve is far enough open that it protrudes into the laser sheet, producing noise which cannot be removed. This was not clear in the earlier images due to the very rich mixture masking this area.

6.3.4 Injection at 60 Degrees ATDC of the Intake Stroke

Injection of fuel at 60 degrees ATDC of the intake stroke begins a new regime in the mixture formation process. This is clear from studying the mean fuel distributions shown in Figure 6.11. There is a wealth of new information contained in the image at 90 degrees ATDC. Firstly, the new fuel entering the cylinder can be seen just emerging from the combustion chamber through the strong tumble flow in the upper left hand corner of the image. This gives an idea as to the transport delay involved in the movement of fuel from the injector, through the intake port, past the valve, and through the combustion chamber. Fuel which began injection at 60 degrees ATDC is just reaching the viewable portion of the cylinder at 90 degrees ATDC.

The second, and more important point, is that the region which has not yet been reached by new fuel, is not at the lowest equivalence ratio level, but has an equivalence ratio between 0.5 and 0.9. This shows for the first time, fuel carryover from one cycle to the next. Fuel carryover is not residual gas, that is, gas trapped in the cylinder from the previous cycle, but is instead fuel which was trapped in the intake manifold from the previous cycle. As injection is moved later into the intake stroke, a portion of the fuel does not clear the intake valve before it closes near bottom dead center. This amount of fuel has been measured quantitatively, and is discussed in section 6.5.2. The existence of fuel carryover was proven by skip firing the fuel injector and measuring the fuel content of the cylinder during the cycle of injection, and during the following cycle. The measurements confirmed that injection up to 30 degrees ATDC produces very little carryover, while later injection results in an almost linear increase in fuel carryover. The
impact of this fact is seen in the tumble plane location shown in Figure 6.11. Instead of mixing nearly pure fuel with nearly pure air as in the earlier cases, the task becomes mixing newly injected fuel with a premixed charge of fuel and air.

The improvement of this mixing regime is seen in the BDC case, as the majority of the image area is in the stoichiometric region even at this early image timing. By 270 degrees ATDC, essentially the entire area is within the ±10% of stoichiometric region. This mixing improvement will be demonstrated in the statistical analysis section.

The images in Figure 6.12 demonstrate the same flow of fuel but in the cross tumble plane. At 90 degrees ATDC some fuel from the latest injection is seen entering the cylinder in the upper corners of the image, but the bulk of the volume shows the carryover fuel from the last injection. Fuel distribution at BDC is vastly improved over the early injection cases, as are the compression stroke images. The optical noise, though, has risen, appearing even in the 270 degree ATDC images. Much of the noise occurs at the top of the ring travel which is roughly 3/4 of an inch from the top of each image. The distribution at 320 degrees ATDC appears very uniform. The swirl plane images shown in Figure 6.13 present the same general trend as the other two planes. The apparently very rich area in the 90 degree ATDC image is actually noise from the intake valve face, as discussed previously.

6.3.5 Injection at 90 Degrees ATDC of the Intake Stroke

The final injection timing tested in these experiments was injection at 90 degrees ATDC of the intake stroke. The main trait of these images is their increased level of fuel carryover from the 60 degree ATDC injection case. In Figure 6.14 the fuel distribution in the tumble plane centerline is shown. At 90 degrees ATDC, the overall fuel distribution is only marginally lean of stoichiometric. In this case, the injection of new fuel has not even begun, reaffirming the fuel carryover concept. Leaner mixture is seen entering the cylinder at the upper corners, indicating that fuel carryover from the previous stroke occupies the region near the valve, but after its induction, leaner mixture follows it into the cylinder. The injection event in this case occurs between images 1 and 2. The effect
of the newly injected fuel is seen in the BDC case image. Here the new fuel is seen following the cylinder wall toward the piston face on the exhaust side of the cylinder. The distribution at BDC in the case of injection at 90 degrees ATDC has deteriorated from the injection at 60 degrees ATDC case. This is due to the very late arrival of the freshly injected fuel into the cylinder. While the initial distribution is very good, injection occurs later and provides less time for mixing with the carryover charge before reaching BDC. As is seen in the statistical analysis, injection at 60 degree ATDC is the optimum tradeoff between injection too early with minimal fuel carryover, and injection too late with insufficient time for mixing of the new charge. The compression stroke images bear this out, as the distribution is less homogenous at 270 degrees ATDC than in the earlier injection case. This trend continues in Figures 6.15 and 6.16, showing the fuel distribution in the cross tumble plane and swirl plane respectively, for injection at 90 degrees ATDC. From a qualitative observation of the mean fuel distribution images, early injection leads to the worst fuel distribution due to poor mixing in the manifold and mixture homogeneity can be improved by selecting a fuel injection timing which balances the amount of fuel carryover with the mixing time for new fuel during the intake stroke. These points are confirmed statistically in section 6.5.

6.4 Small Scale Inhomogeneities

The images presented in section 6.3 were ensemble averages of the fifty images collected at each test condition. These images provided information concerning the bulk maldistribution of fuel by removing the small scale structures through averaging. During engine operation, the distribution of fuel is some combination of the large scale distributions shown previously and small scale structures. The small scale structures can be seen only in individual images as they are largely isotropic and homogenously distributed. The purpose of this section is to provide a qualitative view of the intensity, size, and distribution of small scale structures in the cylinder as a function of both injection timing and crank angle. In addition to small scale structures, the instantaneous images presented reveal any inconsistencies in the location of the large scale structures.
from cycle to cycle. Both the displacement of large scale structures from one cycle to the next, and the small scale mixing structures contribute to the level of cyclic variation in the combustion process, as demonstrated in Chapter 8.

6.4.1 Images at 90 Degrees ATDC of the Intake Stroke

While the mean images in the last section were grouped by fuel injection timing, the images presented here are grouped by image plane and crank angle. In this way, the differences between instantaneous images for different injection timings can be studied. The images shown are simply the first images in each set of 50 and were not selected from the set for any specific reason. The instantaneous images were processed in the same way as the mean images but were not ensemble averaged. Once again, all the images were taken in separate cycles, so there are no images that show a procession in a single cycle of the instantaneous fuel distribution.

Figure 6.17 presents instantaneous images in the tumble plane at 90 degrees ATDC for all five tested injection timings. The two earliest injection timings look very similar, as they are almost entirely in the upper most bin, showing greater than 50% excess fuel. The resolution of the display makes recognition of either large or small scale structures in these images very difficult. Fuel injection at 30 degrees ATDC has significantly more air entrainment, and structures of various sizes can be seen. There is a high level of variation in the location of the large scale structures. This is to say, that the bulk maldistribution is not repeatable from one cycle to the next. The mean images presented earlier are less meaningful in this case. The later injection timings show several interesting properties. The very uniform distributions of mixture in the mean images can be seen to have resulted from isotropic small scale structures. These structures are removed by averaging and give the illusion of near perfect distribution in the mean images. Secondly, the level of cycle to cycle variation in the bulk distribution is very consistent from one cycle to the next in the later injection regimes. Figures 6.18 and 6.19 reveal similar information in the cross tumble and swirl planes respectively.
Reflections from the open intake valve can be seen in several of the images in the upper right hand corner.

6.4.2 Images at Bottom Dead Center of the Intake Stroke

Instantaneous images at the end of the intake stroke show the highest level of stratification for every injection case. Figure 6.20 shows the distributions at the centerline of the tumble plane. While the large scale structures seen in the mean images are easily distinguishable, there is significant variation in their location, particularly in the cases where fuel was injected during compression or early in the intake stroke. The level of stratification appears to be at a minimum in the 60 degrees ATDC injection case, as predicted by the mean images. While some regions show greater than 50% excess fuel, the level is vastly less than the earlier injection timings, and noticeably less than the latest injection timing tested. It is clear also that the range of equivalence ratios for each distribution is significantly higher than the range in the mean images, indicating a significant contribution from small scale structures to the overall fuel maldistribution. Figures 6.21 and 6.22 show the instantaneous images at BDC in the cross tumble and swirl planes respectively.

6.4.3 Images at 90 Degrees After Bottom Dead Center of the Compression Stroke

Figures 6.23, 6.24 and 6.25 show instantaneous fuel distributions at the midpoint of the compression stroke for the tumble, cross tumble and swirl planes locations. The early injection cases, which occur during the compression stroke and at TDC of the intake stroke show a high level of bulk maldistribution even at the late crank angle. While the mean images showed a fairly smooth fuel distribution profile, the reality, as seen in the instantaneous images is that these injection timings still have very significant stratification in terms of both large scale and small scale structures. The large scale structures do not appear in the mean images due mainly to their nearly random location in the cylinder. The rich area that was so consistently located during the intake stroke varies greatly in location during the compression stroke. This is due to the radical change in the
flow field during compression in which the dominant tumble vortex is broken down into small scale turbulence and the swirl vortex is intensified by the piston motion. This same phenomenon is seen to a lesser extent in the injection at 30 degrees ATDC case. Here the intensity of the rich areas is less than in the early case, but these areas are still clearly defined. Injection at 60 degrees ATDC departs significantly from this regime, with the almost all of the image pixels falling in the 0.7 to 1.3 equivalence ratio range. No regular large scale structures can be seen in any of the images for this injection timing case. The late injection case shows a rich area at the piston face, correlating with the flow of fuel into the cylinder late in the intake stroke. This fresh fuel has just reached the point which the early injection cases had reached at 180 degrees ATDC. This fact is somewhat offset by the premixing advantage incurred by the fuel carryover effect. The overall result is that the latest injection case is more poorly mixed than the 60 degree ATDC case but is superior to the earliest injection cases.

6.4.4 Images at 40 Degrees Before Top Dead Center

The last crank angle at which images were acquired in these experiments was 320 degrees ATDC during the compression stroke. This is roughly 25 degrees before ignition at idle. This timing was chosen because it was the last point at which low-noise measurements could be taken before the piston movement began interfering with the view of the camera or the path of the laser. The instantaneous images taken at 320 degrees ATDC are shown in Figures 6.26, 6.27, and 6.28. The images in the tumble and cross tumble planes are difficult to interpret due to the small area shown in these images. More information is available from the swirl plane images. Several conclusions can be drawn from these images. The level of maldistribution is visibly higher in the early injection cases than in the later injection cases. Both the injection at TDC of the compression stroke and injection at TDC of the intake stroke have large areas of the images still outside the stoichiometric region. The maldistribution in the last three injection timings is harder to distinguish and is treated statistically in the next section. It is clear in all the images, though, that there is some variation from one cycle to the next in the total amount
of fuel present in the image plane. Because there is an absence of large scale structures, this indicates either experimental error in the normalization process or variations in the mass of fuel trapped per cycle. The later is shown to be the case in the analysis presented in section 6.5.5. The final observation drawn from these images is that a significant level of small scale maldistribution remains even in the injection cases which result in the best fuel distribution. These small scale inhomogeneities, as discussed previously, are known to contribute to the level of CTCV in the engine.

6.5 Statistical Analysis of Fuel Maldistribution

All of the PLIF results presented up to this point have been based on qualitative observations of the acquired images. A statistical analysis has been performed to both confirm the assumptions made based on observation of the images, and to extract information from the images which is too subtle to be observed directly. All of the statistics presented are based on the assumption of a normal distribution of pixel intensities. This assumption is not required to define the mean and standard deviation as used in this chapter, but is necessary for several other reasons. Firstly, in order to remove the impact of experimental error on the standard deviation, the total variance has been assumed to be the sum of the variance due to experimental error and the variance due to true maldistribution. Secondly, the correlation coefficients presented in Chapter 8 require a normal distribution. Finally, the use of standard deviation to extrapolate information concerning the total population requires that the distributions be normal. While the distribution of pixel intensities early in the intake process deviates somewhat from a normal distribution due to the existence of large scale structures, the distribution of pixel intensities during the compression stroke is well represented by a normal distribution.

Several different properties of the PLIF images have been analyzed statistically to provide a comprehensive understanding of both small and large scale maldistribution and how they vary with crank angle and fuel injection timing. The amount of fuel carryover, the degree of maldistribution in the mean and instantaneous images and the cyclic variation in the total fuel content and distribution of fuel are all covered in this section. In
addition, the contribution of measurement error to each of the statistical quantities has been determined such that this contribution can be removed from the image statistics.

6.5.1 Analysis of Flat Field Images

The level of maldistribution which is seen in the mean and instantaneous images presented thus far is in reality some combination of true inhomogeneity in the mixture and measurement error due to optical noise. It may have been possible to determine analytically the level of noise in the images based on the quantum efficiency of emission from sapphire, the filter profile, noise in the image intensifier, etc. The approach taken in this work was to determine the measurement noise empirically for each crank angle and cylinder image location. This was done by means of analysis of the rich flat field images. These flat field images were processed as if they were an injected image case. Each image in the set was normalized, had the background subtracted and was flat field corrected using the lean flat field image. The result of this processing is a set of fifty images for which the intensity of each pixel in each frame should be identical if no measurement error is present. A statistical analysis of the variation which is present in these "perfectly mixed" images provides a wealth of information concerning the measurement technique. Once the portion of the variation found in the images due to measurement error is known, it can be subtracted from the total variation as the intensity deviations are the sum of measurement error and real inhomogeneities. This process has been carried out for each of the statistical measures presented in this chapter.

The level of inhomogeneity in the instantaneous images presented earlier in this chapter clearly varied with fuel injection timing, image plane, and image timing. The variation of interest in this study is the variation with fuel injection timing. The rich flat field images in each plane and at each crank angle were used to determine the portion of the measured maldistribution in the instantaneous images that was due to measurement error. The best measure of the maldistribution in the instantaneous images is the standard deviation of the pixel values in the given image. A perfectly distributed mixture with negligible measurement error would have a standard deviation approaching zero. The
standard deviation in the instantaneous flat field images is an indication of the measurement error in the system. This standard deviation is shown as a function of image plane and crank angle in Figure 6.29. The purpose of this graph is to show the relative values of the measurement error. Because the mean of the images is essentially one, the standard deviation is in units of equivalence ratio. There are several important pieces of information in this graph. Firstly, the measurement error in the swirl plane is consistently lower than in the other two planes. This is due to the fact that the sapphire cylinder, which causes the bulk of the optical noise is not in view in the swirl plane images. The only reason the swirl plane noise even approaches the other two planes at 90 and 180 degrees ATDC is because of the laser impingement on the valve face which is prevalent at these times in the swirl plane. A second important point is the reduction in noise with increasing crank angle. This is due to increasing cylinder pressure. As the cylinder pressure increases, the absolute number density of acetone increases, raising the signal level. This increase in signal level is not accompanied by an increase in optical noise, thus the signal to noise ratio increases, and the relative measurement error decreases. Statistically speaking, the noise level decreases to the point that at 320 degrees ATDC roughly 70% of the pixels have a value which is within 6% of their actual value. The individual values shown in the graph were used in section 6.5.4 to correct the image statistics for standard deviation in the instantaneous images.

Averaging of 50 instantaneous images yields a mean image in which random error has been reduced significantly. Figure 6.30 is a graph of the two dimensional standard deviation in the mean images as a function of location and crank angle. Note that the mean of the three locations remains essentially constant with increasing crank angle. This is the case because the random measurement error has been removed. The standard deviation values given here are less than half of those shown in the previous figure. The main contribution to the remaining error in the images is imperfect flat field correction. That is, with the random noise removed, small gradients which were not removed during the laser sheet and cylinder curvature correction process dominate the measurement error. In the mean images, in most cases roughly 70% of the pixel values are within 4% of their
actual value. Note that this does not include any error due to shifting of the entire image scale up or down due to fuel flow measurement error. As the standard deviation uses the image mean and not a value of 1.0, the error shown here does not include the deviation of the image mean from the desired image mean.

The variation in image mean is addressed in Figure 6.31. This graph was created by plotting the standard deviation of the mean values of the instantaneous images. That is, the mean value of each image was found, and the standard deviation of this set of 50 was taken. For the early image timings, this value has little meaning, as the total cylinder contents have not yet been established. At 270 and 320 degrees ATDC, though, the cylinder contents are constant, and the mean value in the images should not vary from cycle to cycle. Clearly from the graph the variation from cycle to cycle in these images is quite low, producing a standard deviation of between 0.01 and 0.03 depending on image plane. This confirms the accuracy of the image normalization method. The laser shot noise, which would be the major contributor to the changing level from shot to shot, has been reduced from ±10 % to the level seen in the chart. The remaining portion of the shot to shot variation can be removed statistically from the analysis as in section 6.5.5

Another important measure of the mixture quality is the variation of the distribution from one cycle to the next. This is measured statistically by finding the standard deviation of the 50 intensities for a single pixel location. That is, for a given cylinder location, pick the intensity value from each of the fifty images and find the standard deviation of this set. This standard deviation is a good measure of the dislocation of large scale structures from one image to the next. Once again, in the flat field images, without measurement error there should be negligible variation from one cycle to the next. As opposed to the previously discussed statistical measures, this measure of cycle to cycle variation produces a standard deviation image. This is an image of the same size as the fuel injection images which shows the level of variation at each point. These cycle to cycle variation images for the flat field are shown in Figures 6.32, 6.33 and 6.34. The standard deviations decrease with increasing crank angle, once again due to the increasing signal to noise ratio. The location of maximum variation is
clearly toward the cylinder walls in the tumble and cross tumble planes. This is due to the fluorescence from the cylinder walls. If the walls are considered as a uniform fluorescence source, the noise level will increase as slices of the cylinder are taken further from the cylinder axis. This is intuitive as the thickness of the cylinder section increases as the slice is taken at angles away from the normal. The dominant noise mechanism in the swirl plane is the active intake valve face, particularly during the valve open period. These images were used to correct similar images of the actual injection cycle to cycle variation in fuel distribution presented in section 6.5.6. The cyclic fuel distribution variation is summarized in Figure 6.35 where the mean over each of the images shown was calculated. The main trend, of course, is the decrease in noise with increased cylinder pressure.

6.5.2 Quantification of Fuel Carryover

The description of the mean fuel images given in section 6.3 introduced the concept of fuel carryover, where late injection of fuel results in a portion of the fuel not entering the cylinder on the cycle of injection. While this does not alter the mean flow rate, or the amount of fuel in the cylinder in a given cycle, it dramatically alters the mixture formation process. It is clear from the images taken with injection later in the intake stroke, that some portion of the fuel has remained in the manifold for ingestion in the next cycle. This proportion is a function of the injection timing, but is probably strongly influenced by the port geometry. In the cylinder head used in these experiments, the fuel injector directed fuel at the split between the active and inactive intake valve ports. This split is roughly 2 inches upstream of the valves. With this configuration, it is easy to imagine that the portion of the fuel which enters the inactive intake port would have difficulty finding its way into the cylinder if it was injected too late in the stroke. This is the most likely cause of the injection carryover seen in the results.

Figure 6.36 presents a quantitative analysis of this phenomenon. The data points in the figure were acquired by comparing images taken with regular firing of the fuel injector with those taken with skip firing of the injector. With regular firing, the total fuel
content at the two measurement times (270 and 320 degrees ATDC) should be constant and at a mean equivalence ratio of 1.0. This data was then compared with images taken with the injector firing every tenth stroke. Information was taken only from the centerline of the tumble plane. The agreement of the measurements at 270 and 320 degrees ATDC precludes any possibility that the perceived reduction in fuel content is caused by mixture inhomogeneity. That is, the mean value in the image plane is a good representation of the mean value of the cylinder contents.

Note that the early injection cases remain consistently just above 90% of the steady firing value. The reduction from a value of 1.0 is probably due to residual gas. In the continuous firing mode, no distinction can be made between residual gas and freshly injected charge, so that continuous firing will naturally show richer mixture by the amount of the residual gas fraction. Injection at 30 degrees ATDC begins to show a very slight decrease in the equivalence ratio of the skip fired test. Injection after this point results in a rapid reduction in the amount of fuel from a given injection which is entering the cylinder during that engine cycle. By the latest injection timing tested, 90 degrees ATDC, only 65% of the injected fuel is entering the cylinder on the first stroke after injection. This agrees well with mean images, in which the equivalence ratio at 90 degrees ATDC appears almost stoichiometric. As about half of the mixture has been drawn into the cylinder by this point, it is logical that for the first half of the charge entering the cylinder, the value would be stoichiometric if roughly half of the fuel from the previous injection has been ingested by this point.

6.5.3 Quantification of Bulk Maldistribution

The mean images which have been presented provide a clear picture of the mixture formation process in the test engine as a function of fuel injection timing. Additional information can be extracted from these images by finding the statistical level of inhomogeneity in the images. This was done for each of the mean images, at each location, for each fuel injection timing. The standard deviations from the three image planes were averaged to produce an overall standard deviation describing the level of
bulk maldistribution at a given crank angle as a function of fuel injection timing. The effect of measurement error has been removed from these figures by the method described previously.

The results of this analysis are given in Figure 6.37. At 90 degrees ATDC of the intake stroke, injection at 60 degrees ATDC has the highest standard deviation, while the two earliest and the latest injection timings have the lowest value. This is explained by several factors. The standard deviation is very high in the 60 degree ATDC injection case because of the small amount of fuel which can be seen entering the cylinder in the upper corners. This fuel, in conjunction with the overall lean distribution, creates a distribution with two separate maxima, resulting in an inflated standard deviation. The low standard deviation in the latest injection timing is due to the fact that injection has not yet occurred and the mixture entering the cylinder is fairly uniform, while being somewhat lean. The low level of standard deviation in the early injection cases is due to the use of the image mean and not the desired equivalence ratio in the calculation of the standard deviation. While the images for these cases show a very rich mixture, they show a very well mixed rich mixture. That is, while the average pixel intensity is far from stoichiometric it has a value close to that of its neighbors.

The results become much clearer by 180 degrees ATDC. Here, the nearly pure air has followed in the rich charge in the early injection cases. Therefore, the standard deviation in the mean images for early injection rises rapidly from 90 to 180 degrees ATDC. The same is true, to a lesser extent for injection at 30 degrees ATDC. The latest injection timing shows a similar increase, but for the opposite reason. As opposed to the ingestion of very lean mixture between 90 and 180 degrees ATDC, in the latest injection case, nearly pure fuel has been drawn in. Thus the well mixed cylinder contents at 90 degrees ATDC have been supplemented with a fresh charge of highly stratified mixture. After BDC of the intake stroke, the cylinder contents are essentially fixed for all of the injection timings. The result is an improvement in the fuel distribution due to convective and diffusive mixing. Note that the rate of decrease in standard deviation is proportional to the standard deviation itself. As would be expected, stronger concentration gradients
promote faster mixing. Despite this fact, neither the very early nor the very late injection timings can improve to the level of injection at 60 degrees ATDC by the time of the last image. While the other four injection timings are collected around a standard deviation of 0.035, the case of injection at 60 degrees ATDC is significantly lower at just over 0.02. The impact of this result on combustion quality will be discussed in later sections.

6.5.4 Quantification of Small Scale Inhomogeneities

While the mean images depend mainly on the bulk maldistribution of fuel in the cylinder, the instantaneous images are affected more strongly by small scale inhomogeneities. An analysis similar to the one described in the previous section was applied to the instantaneous images. In this case, the standard deviation in each of the 50 images was found, and the average of these standard deviations was calculated. Figure 6.38 shows the results of this analysis. The graph is very similar to the previous graph indicating that both large and small scale inhomogeneities are dependent on fuel injection timing in the same manner. The main difference between the two graphs is the greater spread at 320 degrees ATDC in the second graph. Here, all but the two earliest injection timings are separated by a visible amount. Of particular note, though, is that injection at TDC of the intake stroke has a slightly higher standard deviation than injection at TDC of the compression stroke, probably providing some measure of the amount of intake manifold mixing which occurs during the valve closed period.

6.5.5 Cyclic Variation in Total Fuel

Measurement error was shown in section 6.5.1 to produce a standard deviation of roughly 2% between the mean values of 50 images in a set images showing well mixed charge. Analysis of the images with port injection revealed a consistently higher level of variation, indicating that the mass of fuel trapped in the cylinder per cycle varies to a measurable degree. This information is shown graphically in Figure 6.39. Here, the mean value of each of the 50 images in a set were found, and then the standard deviation of these values was calculated. The measurement error was removed from these values as
described in earlier sections. The image timing of most consequence here is 320 degrees ATDC. Here, the consistency of fuel concentration in the cylinder from one cycle to the next is shown to be very dependent on injection timing. Injection at TDC of the intake stroke appears to be consistently higher than the other injection timings. This proves to be a key point in the analysis of combustion quality. It must be noted, though, that the total level of the standard deviation is only in the range of 2 percent of the total fuel. Thus in the large majority of engine cycles, the cylinder charge is within ±2 % of stoichiometric.

6.5.6 Cyclic Variation in Fuel Distribution

The final measure of fuel maldistribution to be considered is the cyclic variation in fuel distribution. This is measured by calculating the standard deviation of the 50 pixels from an image set which were taken from the same location in each image. This produces an image that shows not only the level of variation, but its distribution throughout the image plane. Once again the effects of measurement error have been removed from the resultant standard deviations.

Figure 6.40 shows the cyclic fuel distribution images at location 2 at 90 and 180 degrees ATDC of the intake stroke. The level of cyclic fluctuation is clearly highest in the early injection timing cases during the intake stroke. This trend continues in Figure 6.41 for the 270 and 320 degree cases. While the first two injection timings show the highest levels of cyclic variation, it is difficult to distinguish the lower level from the images. The same can be said at locations 5 and 9 as shown in Figures 6.42 through 6.45. These images are summarized in Figure 6.46 where the average level of cyclic variation has been found over each image, and the average of this value over the three planes has been calculated. While the same trend is found as in the small and large scale inhomogeneities, some important differences are seen. Injection at TDC of the intake stroke has the highest levels of cyclic variation in fuel distribution. This is followed closely by the earliest injection. Once again, the difference between these two is probably due to the small amount of mixing which occurs in the intake manifold between TDC of
the compression stroke and TDC of the intake stroke. This mixing time reduces, by a small amount, the level of cyclic variation in fuel distribution in the earliest injection case. The other three injection timings are closely packed at a standard deviation near 0.03. Injection at 60 degrees ATDC once again produces the most consistently well mixed cylinder charge.
Figure 6.1: Flow Chart of Image Processing Scheme
Figure 6.2: Mean Fuel Distribution vs. Crank Angle
(Location 2, 360 degrees BTDC Injection Timing)
Figure 6.3: Mean Fuel Distribution vs. Crank Angle
(Location 5, 360 degrees BTDC Injection Timing)
Figure 6.4: Mean Fuel Distribution vs. Crank Angle
(Location 9, 360 degrees BTDC Injection Timing)
Figure 6.5: Mean Fuel Distribution vs. Crank Angle (Location 2, 0 degrees ATDC Injection Timing)
Figure 6.6: Mean Fuel Distribution vs. Crank Angle
(Location 5, 0 degrees ATDC Injection Timing)
Figure 6.7: Mean Fuel Distribution vs. Crank Angle (Location 9, 0 degrees ATDC Injection Timing)
Figure 6.8: Mean Fuel Distribution vs. Crank Angle (Location 2, 30 degrees ATDC Injection Timing)
Figure 6.9: Mean Fuel Distribution vs. Crank Angle
(Location 5, 30 degrees ATDC Injection Timing)
Figure 6.10: Mean Fuel Distribution vs. Crank Angle
(Location 9, 30 degrees ATDC Injection Timing)
Figure 6.11: Mean Fuel Distribution vs. Crank Angle
(Location 2, 60 degrees ATDC Injection Timing)
Figure 6.12: Mean Fuel Distribution vs. Crank Angle
(Location 5, 60 degrees ATDC Injection Timing)
Figure 6.13: Mean Fuel Distribution vs. Crank Angle
(Location 9, 60 degrees ATDC Injection Timing)
<table>
<thead>
<tr>
<th>ATDC Timing (°)</th>
<th>Equivalence Ratio</th>
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</thead>
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<td>90° ATDC</td>
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</tr>
<tr>
<td>180° ATDC</td>
<td>1.5-1.3</td>
</tr>
<tr>
<td>270° ATDC</td>
<td>1.3-1.1</td>
</tr>
<tr>
<td>320° ATDC</td>
<td>1.1-0.9</td>
</tr>
<tr>
<td></td>
<td>0.9-0.7</td>
</tr>
<tr>
<td></td>
<td>0.7-0.5</td>
</tr>
<tr>
<td></td>
<td>&lt; 0.5</td>
</tr>
</tbody>
</table>

Equivalence Ratio

Figure 6.14: Mean Fuel Distribution vs. Crank Angle
(Location 2, 90 degrees ATDC Injection Timing)
Figure 6.15: Mean Fuel Distribution vs. Crank Angle
(Location 5, 90 degrees ATDC Injection Timing)
Figure 6.16: Mean Fuel Distribution vs. Crank Angle
(Location 9, 90 degrees ATDC Injection Timing)
Figure 6.17: Instantaneous Fuel Distributions
(Location 2, Images at 90 degrees ATDC)
Fuel Injected 360 degrees BTDC

Fuel Injected 0 degrees ATDC

Fuel Injected 30 degrees ATDC

Fuel Injected 60 degrees ATDC

Fuel Injected 90 degrees ATDC

>1.5  1.5-1.3  1.3-1.1  1.1-0.9  0.9-0.7  0.7-0.5  <0.5

Equivalence Ratio

Figure 6.18: Instantaneous Fuel Distributions
(Location 5, Images at 90 degrees ATDC)
Equivalence Ratio

Figure 6.19: Instantaneous Fuel Distributions
(Location 9, Images at 90 degrees ATDC)
<table>
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<th>Equivalence Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>&gt; 1.5</td>
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Figure 6.20: Instantaneous Fuel Distributions
(Location 2, Images at 180 degrees ATDC)
Fuel Injected 360 degrees BTDC

Fuel Injected 0 degrees ATDC

Fuel Injected 30 degrees ATDC

Fuel Injected 60 degrees ATDC

Fuel Injected 90 degrees ATDC

> 1.5  1.5-1.3  1.3-1.1  1.1-0.9  0.9-0.7  0.7-0.5  < 0.5

Equivalence Ratio

Figure 6.21: Instantaneous Fuel Distributions
(Location 5, Images at 180 degrees ATDC)
Fuel Injected 360 degrees BTDC

Fuel Injected 0 degrees ATDC

Fuel Injected 30 degrees ATDC

Fuel Injected 60 degrees ATDC

Fuel Injected 90 degrees ATDC

| >1.5 | 1.5-1.3 | 1.3-1.1 | 1.1-0.9 | 0.9-0.7 | 0.7-0.5 | <0.5 |

**Equivalence Ratio**

Figure 6.22: Instantaneous Fuel Distributions
(Location 9, Images at 180 degrees ATDC)

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Figure 6.23: Instantaneous Fuel Distributions (Location 2, Images at 270 degrees ATDC)

Equivalence Ratio

> 1.5  1.5-1.3  1.3-1.1  1.1-0.9  0.9-0.7  0.7-0.5  < 0.5
Fuel Injected 360 degrees BTDC

Fuel Injected 0 degrees ATDC

Fuel Injected 30 degrees ATDC

Fuel Injected 60 degrees ATDC

Fuel Injected 90 degrees ATDC

> 1.5  1.5-1.3  1.3-1.1  1.1-0.9  0.9-0.7  0.7-0.5  < 0.5

**Equivalence Ratio**

Figure 6.24: Instantaneous Fuel Distributions
(Location 5, Images at 270 degrees ATDC)

133
Figure 6.25: Instantaneous Fuel Distributions
(Location 9, Images at 270 degrees ATDC)
Fuel Injected 360 degrees BTDC

Fuel Injected 0 degrees ATDC

Fuel Injected 30 degrees ATDC

Fuel Injected 60 degrees ATDC

Fuel Injected 90 degrees ATDC

> 1.5  1.5-1.3  1.3-1.1  1.1-0.9  0.9-0.7  0.7-0.5  <0.5

Equivalence Ratio

Figure 6.26: Instantaneous Fuel Distributions
(Location 2, Images at 320 degrees ATDC)
Fuel Injected 360 degrees BTDC

Fuel Injected 0 degrees ATDC

Fuel Injected 30 degrees ATDC

Fuel Injected 60 degrees ATDC

Fuel Injected 90 degrees ATDC

> 1.5  1.5-1.3  1.3-1.1  1.1-0.9  0.9-0.7  0.7-0.5  < 0.5

Equivalence Ratio

Figure 6.27: Instantaneous Fuel Distributions
(Location 5, Images at 320 degrees ATDC)

136
Fuel Injected 360 degrees BTDC

Fuel Injected 0 degrees ATDC

Fuel Injected 30 degrees ATDC

Fuel Injected 60 degrees ATDC

Fuel Injected 90 degrees ATDC

> 1.5  1.5-1.3  1.3-1.1  1.1-0.9  0.9-0.7  0.7-0.5  < 0.5

Equivalence Ratio

Figure 6.28: Instantaneous Fuel Distributions
(Location 9, Images at 320 degrees ATDC)

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Figure 6.29: Standard Deviation Due to Measurement Error in Instantaneous Images
Figure 6.30: Standard Deviation Due to Measurement Error in Mean Images
Figure 6.31: Standard Deviation Due to Measurement Error in Total Fuel Content
Standard Deviation

Figure 6.32: Measurement Error in Cycle to Cycle Variation in Fuel Distribution at Location 2
Figure 6.33: Measurement Error in Cycle to Cycle Variation in Fuel Distribution at Location 5

Standard Deviation
Figure 6.34: Measurement Error in Cycle to Cycle Variation in Fuel Distribution at Location 9
Figure 6.35: Standard Deviation Due to Measurement Error in Cyclic Variation in Fuel Distribution
Figure 6.36: Fuel Carryover Resulting from Delayed Injection
Figure 6.37: Fuel Maldistribution in Mean Images
Figure 6.38: Fuel Maldistribution in Instantaneous Images
Figure 6.39: Cycle to Cycle Variation in Total Fuel Content
Image Timing = 90° ATDC

Injection Timing = 360° BTDC

Injection Timing = 0° ATDC

Injection Timing = 30° ATDC

Injection Timing = 60° ATDC

Injection Timing = 90° ATDC

Image Timing = 180° ATDC

Standard Deviation

Figure 6.40: Cycle to Cycle Variation in Fuel Distribution at Location 2
Figure 6.41: Cycle to Cycle Variation in Fuel Distribution at Location 2 (270 and 320 ATDC)
Figure 6.42: Cycle to Cycle Variation in Fuel Distribution at Location 5 (90 and 180 ATDC)
Figure 6.43: Cycle to Cycle Variation in Fuel Distribution at Location 5 (270 and 320 ATDC)

Standard Deviation
Figure 6.44: Cycle to Cycle Variation in Fuel Distribution at Location 9 (90 and 180 ATDC)
Image Timing = 270° ATDC

Inj. Timing = 360° BTDC

Inj. Timing = 0° ATDC

Inj. Timing = 30° ATDC

Inj. Timing = 60° ATDC

Inj. Timing = 90° ATDC

Image Timing = 320° ATDC

Standard Deviation

Figure 6.45: Cycle to Cycle Variation in Fuel Distribution at Location 9 (270 and 320 ATDC)
Figure 6.46: Cycle to Cycle Variation in Fuel Distribution
CHAPTER 7

CYLINDER PRESSURE RESULTS

7.1 Introduction

The equivalence ratio maps which have been created confirm that the fuel injection timing in the test engine has a measurable impact on the homogeneity of the cylinder charge. The fuel distribution in the combustion chamber is known to effect combustion quality due mainly to variations in the equivalence ratio of the small volume of fuel in the spark plug gap during ignition. Because it was not possible to measure directly the equivalence ratio in the spark plug gap, statistical methods are necessary to deduce the empirical relation between mixture inhomogeneity and combustion quality, particularly CTCV.

In order to determine this relationship, cylinder pressure traces have been acquired in the Optical Access Engine, as described in Chapter 5. One hundred and seventy five pressure traces were collected for each fuel injection timing, for each of four different equivalence ratios. The pressure traces were collected in sets of 25 such that any deviation in the results due to factors other than injection timing could be detected. As described previously, for each fuel injection timing, the means and standard deviations of the maximum cylinder pressure were found to be consistent from set to set, and thus any variations which are measured are due to the injection timing alone. The purpose of this chapter is to present a sample of the cylinder pressure traces, and describe the statistical methods by which the results were analyzed.
7.2 Cylinder Pressure Traces

A sample of the cylinder pressure traces which have been collected are displayed in this section, to provide a qualitative view of the results. For each test case, the first 25 pressure traces collected are shown in a pressure versus crank angle figure. These figures show the relative magnitude of the cylinder pressure, as well as the variation of the pressure traces from cycle to cycle. The cylinder pressures are all given in pressure relative to ambient, and in units of Pascals. The crank angle is in degrees ATDC of the intake stroke, thus 360 degrees is TDC of the compression stroke. Several general observations can be made of the cylinder pressure traces as a whole. The flame speed in these tests is considerably higher than in a production engine, due to skip firing of the engine. Because of the skip firing, the residual gas which would typically be present in the cylinder is replaced with a stoichiometric fuel/air mixture from the previous stroke. Thus the dilutent effects of the residual gas fraction are not present. Also clear from the traces is the noise spike introduced by the ignition system. A slight jump in the apparent cylinder pressure occurs at the point of ignition due to interference in the collection system from the breakdown in the ignition coil.

The cylinder pressure traces for an injection duration of 4.6 msec are shown in Figures 7.1 through 7.5. It is difficult to distinguish between the pressure traces except for the case of injection at TDC of the intake stroke. This set of traces shows both a lower overall average maximum pressure, and more scatter in the pressure data. Figure 7.6 shows mean traces generated by averaging all 175 traces in each set. The two highest maximum cylinder pressure values are for injection at 60 and 90 degrees ATDC. These are followed by injection at 30 degrees ATDC, injection at TDC of compression, and finally, injection at TDC of the intake stroke. This last injection timing shows a significant decrease from the other pressure traces. It is also clear that the maximum cylinder pressure varies inversely with the angle of occurrence of maximum cylinder pressure. This suggests that the maximum cylinder pressure variation is due mainly to variations in the ignition delay or the flame speed. While the total amount of energy being released is relatively constant, the release is phased differently with respect to
combustion chamber volume. This speculation is confirmed by the statistical analysis given later.

Figures 7.7 through 7.11 contain samples of the cylinder pressure traces for an injection duration of 4.2 msec. As was discussed previously, due to inaccurate measurement of fuel flow rate, it is difficult to determine the exact fuel/air ratio. From calibration measurements of the injector, and the cylinder pressure traces shown here, it appears that the stoichiometric fuel/air ratio falls somewhere between the 4.2 and 4.6 msec injection cases, such that 4.6 msec injection leads to a slightly rich equivalence ratio and 4.2 is slightly lean. For this reason, it is difficult to distinguish between the pressure traces of the two cases. Once again in this case, the average maximum cylinder pressure is noticeably lower only for injection at TDC of the intake stroke. In addition, the level of variation of the maximum cylinder pressure appears to be at a minimum for the injection at 60 degrees ATDC. The mean cylinder pressure traces shown in Figure 7.12 are similar to the 4.6 msec case. Once again, injection at 60 and 90 degrees ATDC produce nearly identical results with the highest maximum cylinder pressure. Slightly different in this case is the coincidence of the injection at TDC of compression and 30 degrees ATDC of intake. Once again, injection at TDC of the intake stroke produces a noticeably lower and delayed maximum cylinder pressure.

Pressure traces for an injection duration of 3.8 msec are shown in Figures 7.13 through 7.17. The equivalence ratio for this case is near 0.8. With this leaner condition, the maximum cylinder pressures which occur are significantly reduced, and the level of CTCV has increased as well. The dependence of these factors on fuel injection timing has not been dramatically affected. Injection at TDC of the intake stroke shows the lowest maximum cylinder pressure, the latest angle of occurrence of maximum cylinder pressure and the greatest cyclic variation in maximum cylinder pressure. The mean traces, shown in Figure 7.18, are much more closely congregated than in the previous cases. This indicates that as the mixture becomes leaner, the dependence of CTCV on injection timing is reduced.
The final injection duration which was tested was a duration of 3.4 msec. This corresponds to an equivalence ratio near 0.6. From the cylinder pressure traces, it is clear that this is approaching the lean limit of the engine. Figures 7.19 through 7.24 show the cylinder pressure traces for this case. While no complete misfires are seen in the data, the total fuel present was clearly not burned for several cycles. Injection at TDC of the intake stroke, at 30 degrees ATDC and at 90 degrees ATDC have a particularly high rates of partial burn cycles. While far from perfect, the other two injection timings are significantly better in terms of CTCV. The mean traces are closely cropped in Figure 7.24. The main contribution to differences between cycles is probably the number of partial misfires and not the phasing of the combustion process.

7.3 Cylinder Pressure Analysis

The statistical analysis presented in this section is intended to determine the relationship between combustion quality and fuel injection timing in the test engine. Specific measures of combustion quality must be selected in order to make a quantitative judgment of this relationship. Many of the possible parameters which have been used in past works were discussed in Chapter 4. A few of these properties were chosen for analysis in this section.

Several possible measures of combustion quality were not used due to difficulties in collecting cylinder pressure data in the OAE. The primary problem which was encountered was leakage from the cylinder during the compression and expansion strokes. This leakage was apparent when viewing graphs of cylinder pressure versus cylinder volume. In motored cycles, the pressure returned to a lower value at the end of the expansion stroke than it had been at the beginning of compression. The leakage was estimated from these graphs to be roughly 5% of the charge mass. There are several reasons for this leakage. The most dominant cause is probably leakage past the valves. The cylinder head used in this work was a production head which had never been run with combustion prior to theses tests. While valves and valve seats are carefully machined, a high quality seal is not typically achieved for the first 10 to 20 hours of
engine operation. Another possible source of leakage is the piston ring. As described earlier, a single piston ring was used with no wall lubrication. Some leakage probably occurred past this ring.

The cylinder leakage created difficulty in calculating heat release parameters, as they depend heavily on a polytropic relationship between pressure and volume before and after combustion. For this reason, attempts to calculate a net heat release rate resulted in large errors, and the data was not used. Additionally, the phasing of pressure to volume in calculation of IMEP and heat release rate are of the utmost importance. While great care was taken in assuring that the camshaft and crankshaft were properly phased, the resolution of the camshaft relative encoder was only 0.6 crank angle degrees. Thus even with proper alignment, the pressure versus crank angle data has errors up to 0.3 degrees. While this error is small in an absolute sense, IMEP and heat release rate are very sensitive to the pressure-volume relationship near TDC. The combustion parameters which are described in the following sections were chosen both for their pertinence to the combustion quality, and for their relative insensitivity to the above described problems.

7.3.1 Dependence of Ignition Delay on Injection Timing

The delay between the firing of the spark plug and the combustion process during a given cycle is thought to be the factor which is most strongly influenced by the homogeneity of the cylinder charge. This is due to the dependence of the kernel formation process on the equivalence ratio of the gases within the spark plug volume. A rich charge will lead to faster kernel development than a lean kernel because of the balance between the energy being released by the combustion charge and the losses through convection and conduction to the spark plug and surrounding cylinder volume. At the conditions investigated here, the average time which it takes to burn the first 10 percent of the fuel is roughly equal to the amount of time which it takes to burn the last 90 percent.

Throughout this analysis the 0 to 10 % burn duration is referred to as the ignition delay. Various measures have been used in the literature to identify the ignition delay
ranging from 2% of the fuel burned up to 10%. Ten percent was chosen here to help correct for the inaccuracy of the cylinder pressure data at the time of ignition. The interference between the coil breakdown and the cylinder pressure acquisition created noise in the pressure measurement around the time of the spark, which made the 0-2% and 0-5% burn durations less accurate. The ignition delay used here should perhaps more accurately be described as the early burn duration. In any case, the 0-10% burn duration was found using the mass fraction burned calculations described in Chapter 4.

Figure 7.25 shows the impact of injection timing on the absolute value of the ignition delay for the four different fuel injection durations tested. The data of most interest here is that of the 4.2 and 4.6 msec injection duration, as these are the cases close to stoichiometric. The trend in the graph shows an increase in ignition delay between the earliest and second earliest injection timings. This agrees well with observations made from the raw cylinder pressure data. Injection at TDC of the intake stroke showed lower maximum cylinder pressures due to delayed combustion initiation. A gradual decrease in the ignition delay is seen as injection is moved further back in the intake stroke. While the trend is clear, the absolute magnitude of the change is only 1 to 2 crank angle degrees. The trend changes significantly as the fuel/air ratio is reduced, with the latest injection timing having the longest ignition delay in the leanest test case.

More important than the mean value of the ignition delay is the variation in the ignition delay from one cycle to the next. It is here that the role of mixture inhomogeneity is most clear. The coefficient of variation of the ignition delay is used in this analysis. This is simply the standard deviation of the measurements divided by the mean value. The result is a variation in terms of percentage of the mean. The coefficient of variation of ignition delay is shown as a function of fuel injection timing in Figure 7.26. Once again the key data is that for the cases near stoichiometric. This data is clearly dependent on the injection duration, with the highest levels of COV of ignition delay occurring for the early injection cases. The COV decreases for injection at 30 and 60 degrees ATDC, but increases for the latest injection timing. This behavior is similar to that shown for the PLIF images in terms of the relation of mixture homogeneity with
injection timing. The correlation between these variables will be shown more directly in Chapter 8. The main change of COV of ignition delay with equivalence ratio is a more dramatic dependence on injection timing. For the shortest injection duration, the worst case shows variations approximately 50% higher than the best case.

7.3.2 Dependence of Burn Duration on Injection Timing

The second parameter which has been calculated from the cylinder pressure data is the rapid burn angle. Referred to here as the burn duration, this is the number of crank angle degrees required for the mixture to go from 10% burned gas fraction to 90% burned gas fraction. While the early flame development and the extinction of the flame at the cylinder walls requires a significant amount of time, the time between 10% and 90% burn fraction is quite short due to the unimpeded turbulent flame propagation through the combustion chamber. The burn duration is thought to be less dependent on mixture homogeneity than on overall equivalence ratio. This is clear because the ignition delay is determined largely by the equivalence ratio in a small region, while the overall flame propagation moves through the bulk of the mixture, thus smoothing over the small scale variations in fuel/air ratio.

The dependence of mean burn duration on injection timing is shown in Figure 7.27. The variation with equivalence ratio is clearly what is expected: Burn durations increase with leaner mixtures due to the decrease in turbulent flame speed. The variation with injection timing is similar to that of the ignition delay, though probably for different reasons. The mean burn duration will increase if the variation in equivalence ratio from cycle to cycle increases, even with a constant mean equivalence ratio. This is due to the non-linear nature of the flame speed. The maximum flame speed for methane occurs very close to an equivalence ratio of 1.0 and decreases rapidly in either rich or lean mixtures. Thus as the mean equivalence ratio in a given cycle fluctuates further from unity, the mean flame speed over many cycles will decrease even if the mean equivalence ratio over many cycles is one. This is probably the case for the data in Figure in 7.27, and this concept is supported by the data in Figure 7.28. This figure shows the COV of burn
duration as a function of equivalence ratio. The significantly higher levels of COV for injection at TDC of the intake stroke will be shown to correlate well with the variation in the mean equivalence ratio in individual cycles in Chapter 8.

### 7.3.3 Maximum Pressure and Angle of Occurrence of Maximum Pressure

While each of the previously described combustion parameters were calculated from the cylinder pressure traces, some information can be extracted directly from the traces without processing. The two parameters of greatest importance are the maximum cylinder pressure and the crank angle at which it occurs. As was discussed in the presentation of the raw cylinder pressure data, strong changes in the mean maximum cylinder pressure can be seen from one injection timing to the next, but the variation in maximum cylinder pressure is difficult to judge by observation alone. Figures 7.29 and 7.30 detail the COV of maximum cylinder pressure and the standard deviation of the angle of occurrence of maximum cylinder pressure. In Figure 7.29, the observations of the raw pressure traces are confirmed in that the largest cyclic variation in maximum pressure occurs with injection at TDC of the intake stroke. The cyclic variation in the other four injection cases fall within 1% of each other (except for the leanest case), though the variation does seem to decrease with injection occurring later in the intake stroke. As discussed previously, these variations in maximum cylinder pressure are more likely due to changes in phasing of the combustion event than to the fraction of the fuel which is burned. This is clearly not the case for the leanest test, where a large number of partial misfires had a strong influence on the mean maximum cylinder pressure. The variation in the angle of maximum cylinder pressure confirms the ignition delay information. Longer ignition delays naturally lead to a later pressure peak due to the phasing of the process. Chapter 8 explores the direct correlation of these parameters with the mixture homogeneity found in the PLIF images.
Figure 7.1: Cylinder Pressure Data
(Injection Timing= 360 deg. BTDC, Injection Duration=4.6 ms)
Figure 7.2: Cylinder Pressure Data
(Injection Timing= 0 deg. ATDC, Injection Duration=4.6 ms)
Figure 7.3: Cylinder Pressure Data
(Injection Timing= 30 deg. ATDC, Injection Duration=4.6 ms)
Figure 7.4: Cylinder Pressure Data
(Injection Timing= 60 deg. ATDC, Injection Duration=4.6 ms)
Figure 7.5: Cylinder Pressure Data
(Injection Timing= 90 deg. ATDC, Injection Duration=4.6 ms)
Figure 7.6: Mean Cylinder Pressure Traces
(Injection Duration=4.6 ms)
Figure 7.7: Cylinder Pressure Data
(Injection Timing= 360 BTDC, Injection Duration=4.2 ms)
Figure 7.8: Cylinder Pressure Data
(Injection Timing= 0 deg. ATDC, Injection Duration=4.2 ms)
Figure 7.9: Cylinder Pressure Data
(Injection Timing= 30 deg. ATDC, Injection Duration=4.2 ms)
Figure 7.10: Cylinder Pressure Data
(Injection Timing= 60 deg. ATDC, Injection Duration= 4.2 ms)
Figure 7.11: Cylinder Pressure Data
(Injection Timing = 90 deg. ATDC, Injection Duration = 4.2 ms)
Figure 7.12: Mean Cylinder Pressure Traces
(Injection Duration = 4.2 ms)
Figure 7.13: Cylinder Pressure Data

(Injection Timing = 360 deg. BTDC, Injection Duration = 3.8 ms)
Figure 7.14: Cylinder Pressure Data
(Injection Timing= 0 deg. ATDC, Injection Duration=3.8 ms)
Figure 7.15: Cylinder Pressure Data
(Injection Timing= 30 deg. ATDC, Injection Duration=3.8 ms)
Figure 7.16: Cylinder Pressure Data
(Injection Timing= 60 deg. ATDC, Injection Duration=3.8 ms)
Figure 7.17: Cylinder Pressure Data
(Injection Timing = 90 deg. ATDC, Injection Duration = 3.8 ms)
Figure 7.18: Mean Cylinder Pressure Traces
(Injection Duration = 3.8 ms)
Figure 7.19: Cylinder Pressure Data
(Injection Timing= 360 deg. BTDC, Injection Duration=3.4 ms)
Figure 7.20: Cylinder Pressure Data
(Injection Timing= 0 deg. ATDC, Injection Duration=3.4 ms)
Figure 7.21: Cylinder Pressure Data
(Injection Timing = 30 deg. ATDC, Injection Duration = 3.4 ms)
Figure 7.22: Cylinder Pressure Data
(Injection Timing= 60 deg. ATDC, Injection Duration=3.4 ms)
Figure 7.23: Cylinder Pressure Data
(Injection Timing= 90 deg. ATDC, Injection Duration =3.4 ms)
Figure 7.24: Mean Cylinder Pressure Traces
(Injection Duration = 3.4 ms)
Figure 7.25: Impact of Injection Timing on Ignition Delay
Figure 7.26: Impact of Injection Timing on Coefficient of Variation of Ignition Delay
Figure 7.27: Impact of Injection Timing on Burn Duration
Figure 7.28: Impact of Injection Timing on Coefficient of Variation of Burn Duration
Injection Duration = 4.6 ms
Injection Duration = 4.2 ms
Injection Duration = 3.8 ms
Injection Duration = 3.4 ms

Figure 7.29: Impact of Injection Timing on Coefficient of Variation of Maximum Cylinder Pressure

0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16

-360 0 30 60 90

Coefficient of Variation of Maximum Cylinder Pressure (%)

Injection Timing (Crank Angle Degrees)
Figure 7.30: Impact of Injection Timing on Standard Deviation of Angle of Occurrence of Maximum Cylinder Pressure
8.1 Introduction

The PLIF results presented in Chapter 6 confirm that in the given test engine a measurable degree of mixture inhomogeneity of the cylinder charge exists near the end of the compression stroke, and is a strong function of the fuel injection timing. This maldistribution has been analyzed quantitatively in terms of small and large scale structures, as well as the cyclic variation of the total fuel content and the mixture distribution. The cylinder pressure analysis which has been presented demonstrates the dependence of combustion quality on fuel injection timing in the given test engine. In both the PLIF and cylinder pressure results, the applicability of the results is largely limited to this specific cylinder head and test engine arrangement. While the mixture formation process in similar engines may be well represented by the acquired results, the key parameter in determining the impact of injection timing on combustion quality is the final state of the mixture prior to ignition. The purpose of this chapter is to extract information from the acquired images and combustion parameters to draw conclusions applicable to a wide range of engines, concerning the impact of mixture inhomogeneity on combustion quality.

Several parameters concerning the relation between the in-cylinder fuel distribution and combustion quality are discussed in this chapter. In each case, the information is extracted from PLIF and cylinder pressure results which have already been explained. For each parameter of interest ten data points have been used. This data
consists of the five fuel injection timings which were tested, and the combustion data from the 4.2 and 4.6 msec injection cases. The shorter injection durations have not been used because quantitative fuel distribution images were not acquired for non-stoichiometric cases. The combustion parameters which are discussed here are those related directly to the level of CTCV in the engine. The impact of mixture uniformity on the mean value of burn duration and ignition delay are thought to be due only to the higher level of variability with different injection timings. The mean value of these flame speed parameters is influenced by the level of variation in the parameters due to the non-linear nature of flame speed. As mentioned previously, the maximum flame speed for methane is just rich of stoichiometric, and the flame speed drops off rapidly as the equivalence ratio moves away from 1.0. This reduction in flame speed is probably the main factor which causes mixture inhomogeneity to effect the mean ignition duration and burn duration. For this reason, only the level of cyclic variation of the combustion parameters covered in Chapter 7 are considered here.

8.2 The Impact of Cycle to Cycle Fuel Distribution Variations on Combustion

The level of cyclic variation in the distribution of fuel in the cylinder of the engine was found to correlate very strongly with several of the combustion quality parameters. The relation between these parameters was measured by calculating the correlation coefficient based on the mean CTCV in fuel distribution and the COV combustion parameters for injection durations of 4.2 and 4.6 msec. All of the correlation coefficients for the data presented in this chapter are shown in Table 8.1.

Figure 8.1 is a plot of the available data for the relation between cyclic fuel distribution and the coefficient of variation of the ignition delay. As described previously, the level of variation in the fuel distribution from cycle to cycle was extracted from the PLIF images by determining the standard deviation for the 50 pixels in each image set which occupy the same image location. This is a measure of how much the equivalence ratio at a given point changes from cycle to cycle. No attempt was made to utilize the spatial information provided by this calculation. The data presented is based
only on the mean over the entire image of the CTCV of fuel distribution. If the spark plug area had been imaged, a more direct comparison could be made. The figure shows a strong correlation between the cyclic variation in fuel distribution and the cyclic variation in ignition delay. This is as expected. As described earlier, the ignition delay is mainly a function of the equivalence ratio in the spark plug gap. If small scale inhomogeneities dominate the mixture field, and these inhomogeneities are isotropic and homogeneously distributed, as shown in Chapter 6, than the equivalence ratio in the spark plug gap does vary from cycle to cycle. The dependence of the variation on fuel injection timing was demonstrated. The range of variation in the standard deviation of the cycle to cycle fuel distribution was almost a factor of two, and this factor caused an increase in the COV of ignition delay of about 30 percent.

The other main impact of the cyclic fluctuation of fuel distribution was found to be on the COV of maximum cylinder pressure. From the cylinder pressure traces, it was clear that for equivalence ratios near stoichiometric, the main determinant of the cylinder pressure was the phasing of the combustion process. Thus, the COV of maximum cylinder pressure is inherently tied to the ignition delay. Figure 8.2 shows the relationship between the cyclic fuel distribution and the COV of maximum cylinder pressure. In this case, doubling the level of cyclic fuel maldistribution effectively doubles the level of COV of maximum cylinder pressure. The correlation coefficient for this dependence is once again shown in Table 8.2.

8.3 The Impact of Cycle to Cycle Fuel Quantity on Combustion Quality

The total mass of fuel present for a given cycle was shown to vary to a measurable degree in Chapter 6. The variation in total fuel quantity was found to hover near one percent for most of the fuel injection timings, but was more than twice this value for injection at TDC of the intake stroke. This increased fluctuation is probably due to the variability of the flow field in the intake port at the time of injection. While the earliest injection occurs with essentially no flow in the intake port, and the last three injection timings occur at periods of high intake flow rate, fuel injected near TDC of the intake
stroke is being injected into a flow which is just beginning to be accelerated by the pressure drop across the intake valve. It is logical to assume that the flow pattern at this time is much more variable on a cyclic basis than either before valve opening or during times of large mass influx to the cylinder.

The comparison of cyclic fuel quantity and COV of burn duration is shown in Figure 8.3. Fuel quantity showed little impact on COV of ignition delay, but correlated well with burn duration. This is in agreement with theory, as the overall burn duration depends heavily on the mean equivalence ratio of the mixture by means of the turbulent flame speed. The figure demonstrates the large separation in cyclic variation of total fuel between injection at TDC of intake and the other fuel injection timings. This separation is mirrored in the COV of burn duration data. The same type of correlation is shown in Figure 8.4 for the impact of cyclic fuel quantity on cylinder pressure. The maximum cylinder pressure is clearly dependent on the total mass of fuel present, and this dependence is shown experimentally by the high level of correlation between the COV of maximum cylinder pressure and the standard deviation of the cycle to cycle fuel quantity.

8.4 Impact of Mean and Instantaneous Fuel Distribution on Combustion Quality

The standard deviation of the pixel values in the instantaneous PLIF images provides a measure of the quality of the fuel distribution for a given cycle. The standard deviation near the end of the compression stroke quantifies the complex mixture field which is created as the tumble vortex is broken down into small scale turbulence. A higher standard deviation suggests that the amplitude of the equivalence ratio fluctuations in the mixture is of greater magnitude. More intense fluctuations should lead to higher levels of CTCV, once again through the flame kernel formation mechanism. This is indeed seen to be the case in Figure 8.5 where a very strong correlation is found between the average standard deviation in the instantaneous images and the COV of ignition delay. The advantage of injection at 60 degrees ATDC can be seen here. This injection timing has been shown to lower the level of small and large scale mixture inhomogeneity in the cylinder, thereby reducing the variation in the ignition delay by more than 2%.
The mean images presented in Chapter 6 revealed that even near the end of the compression stroke, some bulk maldistribution still exists for several of the fuel injection timings tested. Some correlation is seen between the variation in ignition delay period and the level of bulk maldistribution present. This relation is shown in Figure 8.6. The correlation is much weaker than for the other parameters, and without the significantly lower maldistribution which is present for the 60 degree ATDC injection case, very little correlation would be seen at all. The conclusion to be drawn from this observation is that it is the small scale inhomogeneities which dominate the cyclical fluctuations in the engine at this level of bulk maldistribution. This may not be true if higher levels of bulk maldistribution were present. The mixing in this engine is probably superior to that found in most other engines due to the combination of tumble and swirl which have been created by the head and manifold design. A cylinder head in which both intake valves are operable at idle, for example, may show significantly higher levels of bulk maldistribution, in which case it may play a greater role in the combustion process, and distribution of the bulk rich and lean areas with respect to the spark plug would determine the combustion quality and operable limits in the engine. In the case of this engine, though, it is the small scale inhomogeneities and the cyclic variation in the fuel distribution that are the dominant cause of cycle to cycle combustion variations.
<table>
<thead>
<tr>
<th>Cycle to Cycle Fluctuation in Fuel Distribution</th>
<th>COV of Ignition Delay (%)</th>
<th>COV of Burn Duration (%)</th>
<th>COV of Maximum Pressure (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.8806</td>
<td>0.6278</td>
<td>0.8122</td>
</tr>
<tr>
<td>Cycle to Cycle Variation in Total Fuel Quantity</td>
<td>0.6291</td>
<td>0.7697</td>
<td>0.8457</td>
</tr>
<tr>
<td>Mean Value of Standard Deviation in Instantaneous Images</td>
<td>0.8785</td>
<td>0.4131</td>
<td>0.6493</td>
</tr>
<tr>
<td>Mean Value of Standard Deviation in Mean Images</td>
<td>0.7681</td>
<td>0.0816</td>
<td>0.3446</td>
</tr>
</tbody>
</table>

Table 8.1: Correlation Coefficients for Fuel Distribution and Combustion Parameters
Figure 8.1: Correlation Between Cycle-to-Cycle Mixture Distribution and Coefficient of Variation of Ignition Delay
Figure 8.2: Correlation Between Cycle-to-Cycle Mixture Distribution and Coefficient of Variation of Maximum Cylinder Pressure
Figure 8.3: Correlation Between Cycle-to-Cycle Fuel Quantity and Coefficient of Variation of Burn Duration
Figure 8.4: Correlation Between Cycle-to-Cycle Fuel Quantity and Coefficient of Variation of Maximum Cylinder Pressure
Figure 8.5: Correlation Between Maldistribution in Instantaneous Images and Coefficient of Variation Ignition Delay
Figure 8.6: Correlation Between Mean Image Maldistribution and Coefficient of Variation of Ignition Delay
CHAPTER 9

CONCLUSIONS

The degree of mixture inhomogeneity and level of cyclic combustion variation have been determined experimentally as a function of fuel injection timing in a natural gas fueled engine. The mixture inhomogeneity was evaluated by means of equivalence ratio maps acquired in three different planes in the engine cylinder by Planar Laser Induced Fluorescence. The cyclic variation in combustion was determined statistically from the analysis of cylinder pressure traces collected over many cycles in the fired engine. The relation of both fuel distribution and cylinder pressure to injection timing allowed details of the correlation between mixture inhomogeneity at the time of ignition and the resulting combustion quality to be found. A statistical analysis of this correlation revealed a strong relationship between the uniformity of fuel distribution and the repeatability of the combustion process.

The sets of 50 images in each cylinder test plane, for each injection timing were processed to provide qualitative and statistical information detailing the mixture formation process. The mean images confirmed that the mixture formation process in the test engine is heavily dependent on fuel injection timing. The two earliest injection timings, at TDC of the compression stroke and at TDC of the intake stroke, were very similar. This similarity results from the low level of mixing which occurs in the intake manifold during the valve closed period. In each of these early cases a rich volume of mixture was formed just upstream of the intake valve prior to entry into the cylinder. During the first half of the intake stroke, this rich charge is drawn into the cylinder.
During the last half of the intake stroke, a very lean mixture is ingested. The strongly stratified charge which results from this process is greatly reduced during the compression stroke, but for these two early injection cases, a measurable level of bulk maldistribution still exists near the end of the compression stroke. Injection at 30 degrees ATDC of the intake stroke results in a less stratified charge than the early injection timings because the freshly injected fuel entrains air as it enters the previously opened intake valve. A high level of stratification is still present at BDC of the intake stroke, but the mean distribution is more uniform by the crank angle of the last measurement at 320 degrees ATDC, during the compression stroke.

Injection at 60 and 90 degrees ATDC results in a phenomenon which is referred to here as fuel carryover. In each of these cases, due to the flow field and geometry in the intake manifold, a portion of the injected fuel for a given cycle is trapped in the intake manifold upon valve closing and enters the cylinder during the following cycle. The amount of carryover fuel varies almost linearly from about 8% with injection at 30 degrees ATDC to almost 45% with injection at 90 degrees ATDC. This carryover fuel is beneficial to the mixing process, as newly injected fuel is combined with a premixed air/fuel mixture as it is drawn into the cylinder. This contrasts sharply with the early injection timings which showed the fuel and air entering the cylinder at distinct times. While the impact of fuel carryover is positive to a degree, injection later in the intake stroke results in the introduction of newly injected fuel at a later crank angle than with early injection. The quality of the mixing process is a compromise between the desire for some fuel carryover and the desire to allow the maximum mixing duration for newly injected fuel. Injection at 60 degrees ATDC is the optimum injection timing in this engine. Injection prior to this point results in minimal fuel carryover, while later injection does not allow sufficient mixing time for the freshly injected fuel.

The information found in the mean images was analyzed statistically along with information obtained from the direct instantaneous images. As predicted by the qualitative analysis of the PLIF images, injection at 60 degrees ATDC showed the lowest level of mean maldistribution at the final measurement crank angle. In addition, the level
of inhomogeneity in the instantaneous images was found to be lowest in the images taken with injection at 60 degrees ATDC. The total quantity of fuel in the cylinder on a cycle to cycle basis was found to be independent of injection timing except for injection at TDC of the intake stroke. The level of cyclic variation for this case was more than twice the level found in the other cases. This variation is most likely due to the variability of the intake flow around the time of injection for this case. The cyclic variation in the distribution of fuel was also analyzed, and the lowest level of variation was once again found in the case of injection at 60 degrees ATDC.

The combustion quality was determined as a function of injection timing by an analysis of cylinder pressure traces collected in the engine under skip fired operation. Pressure traces were collected at each of the injection timings used in the PLIF experiments and with 4 different fuel injection durations. Combustion quality was analyzed based on the time taken to reach 10% mass fraction burn after ignition and the time between 10% and 90% mass fraction burned. In addition the maximum cylinder pressure and the angle of occurrence of maximum cylinder pressure were studied. The early flame development period was found to be a strong function of the injection timing. The maximum cylinder pressure and angle of occurrence of maximum cylinder pressure were also found to be dependent on the fuel injection timing. This dependence can be explained almost entirely by the variation in the injection delay.

By eliminating the injection timing from the PLIF and combustion results, the direct impact of mixture homogeneity on combustion quality was analyzed. The most direct correlation which was found was between the mixture statistics and the ignition delay. This agrees well with the theory of flame kernel development. The early flame development period is thought to depend mainly on the equivalence ratio in the spark plug volume during ignition. The variation in this fuel concentration from cycle to cycle is clearly dependent on the homogeneity of the cylinder charge. Thus higher levels of mixture inhomogeneity indicate more intense fluctuation from cycle to cycle in the spark plug volume equivalence ratio and thus in the ignition delay. The cyclic variation in fuel distribution and the level of inhomogeneity in the instantaneous images were both found
to be strongly linked to the ignition delay. Over the range of mixture inhomogeneities found, the coefficient of variation of the ignition delay was found to vary by up to 30%. In addition to the effect of mixture homogeneity, the variation in the total quantity of fuel trapped in the cylinder from cycle to cycle was found to impact both the bulk burn duration and the maximum cylinder pressure. This once again agrees closely with theory, as bulk burn duration is determined by the equivalence ratio by way of the turbulent flame speed, and the maximum cylinder pressure varies with the total amount of heat release. In summary, the variations in the state of the cylinder charge caused by varying the fuel injection timing are of sufficient magnitude to have a direct impact on the quality of combustion in the test engine. Less specific to this engine is the dependence of combustion on the mixture homogeneity. The existence of spatial equivalence ratio fluctuations in the cylinder at the time of ignition have been found to be a major contributor to the cyclic combustion fluctuations found in natural gas engines at low speed and load.

Several components of this research warrant further study, including the development of the experimental PLIF technique, its extension to direct cycle resolved measurements in the spark plug volume, and the investigation of a wider range of inhomogeneity cases. Improving the accuracy of the experimental technique would allow a more precise analysis of the mixture distribution, allowing for a direct measurement of the relation between fuel distribution and combustion quality. The main weakness of the technique used in this research is inaccurate quantification of the acquired images. The main component of the quantification error is poor accuracy in the fuel flow rate measurement. This problem would be most easily resolved by controlling the flow of fuel in the rich and lean flat field cases via a fuel injector at an upstream location. This would allow the spread between the rich and lean cases to be controlled and would provide exact correlation of the flat field flow rates with the flow rate during engine operation. The elimination of this error would allow a much more complete analysis of both the bulk fuel maldistribution and the cyclic variation in the total fuel quantity.
The ability to measure the equivalence ratio of the charge in the spark plug volume prior to ignition would facilitate comparisons between mixture homogeneity and combustion quality on a cycle resolved basis. The main hurdle to be overcome to make this measurement possible is the ability to direct the laser sheet through this volume. While this would require some modification of the production engine geometry, it would provide direct information on the effect of mixture homogeneity on the early flame development process.

The level of mixture inhomogeneity in the test engine utilized for this work is probably considerably lower than that found in other engines. The swirl induced by the inactive intake valve in conjunction with the tumble which results from the intake runner and port geometry generates a high level of mixing which effectively reduces the intensity of equivalence ratio fluctuations at the time of ignition. Additional information would be gained by testing other engines with other intake configurations. This would yield information as to the impact of intake geometry on the fuel/air mixing process and would also provide additional data points in the correlation between mixture homogeneity and combustion quality. This information is required if the overall contribution of mixture inhomogeneity to the level of cycle to cycle combustion variations in natural gas engines is to be understood.
BIBLIOGRAPHY


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