Investigation of Flame-Front Equivalence Ratio during Stratified Engine Combustion

by

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Abstract

Stratified engine combustion was investigated using simultaneous imaging of the fuel distribution and flame front in an optically accessible direct-injection spark-ignition engine. Planar laser-induced fluorescence of 3-pentanone doped into iso-octane and the OH radicals naturally occurring in the combustion products were imaged with two intensified CCD cameras. The 3-pentanone images provide a quantitative measure of fuel concentration and the OH images allowed for the position of the flame front to be accurately determined. These results represent the first data taken during stratified combustion using a two-camera technique. Using the image data a novel method was developed to determine the flame-front equivalence ratio during stratified combustion. The results of the method provide insights into the stratified combustion process. Additionally, engine-out NO\textsubscript{x} and CO measurements are presented and an effort to determine a correlation between the flame-front equivalence ratio and measured emissions is made where the flame-front equivalence ratio is thought to be a major factor in pollutant emission formation during stratified combustion. The effects of engine speed, engine load, spark timing, and ignition timing were investigated.

The data indicate that a wide range of equivalence ratios are present along the flame front. The limited field of view was found to significantly influence the data. The flame-front equivalence ratio data taken for conditions with varying injection and varying spark timing at equivalence ratios of $\Phi = 0.32$ and $\Phi = 0.42$ at 600 rpm showed little correlation with the measured emissions. However, the NO\textsubscript{x} data did clearly reflect the trends of peak pressure. The available field of view may have been one cause for the lack of correlation,
but the pressure trends and emissions data also indicate that combustion phasing has a
strong influence on NOx emissions with changes in spark timing of 10 crank angle degrees
causing almost a factor of two change in measured engine-out NOx.
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Appendix A
1.0 Introduction

1.1 Motivation

The need for reduction of CO$_2$ emissions and an overall reduction in energy consumption has driven the development of more efficient engines. The concept of directly injecting fuel in cylinder and igniting the air-fuel charge with a spark has been around for some time but only recently has become realizable as a possible production alternative to the port fuel injection (PFI) engines which currently dominate the automotive market in the United States.

Direct-injection spark-ignited (DISI) engines offer the benefit of significant increases in thermodynamic efficiency when compared with PFI engines, while maintaining the power, torque, and response characteristics of a PFI engine which consumers are familiar with. On the downside, DISI engines generate unburned hydrocarbons (HC) emissions, nitric oxide (NO$_x$) emissions, and particulate matter in excess of PFI engines. To compound this problem, DISI engines offer the greatest benefits when operating under stratified, overall lean operation. Under these conditions the three-way catalyst technology used on PFI engines is not applicable. Currently significant work is being done to develop lean-burn catalyst technologies, however, efficiencies of these catalysts are still quite low relative to three-way catalysts, and they suffer from problems such as sulfur poisoning that limit the effective life of the catalyst.

Pollutant emissions have so far been the major stumbling block preventing widespread implementation of DISI engines in production automobiles. The possible
efficiency increases due to DISI operation merit significant effort to better understand the processes affecting emissions formation. The general understanding of emissions formation under stratified combustion is slowly being established with much speculation of possible causes and little hard data to back up some of the theories. Recently the application of techniques such as planar laser-induced fluorescence have given greater insight into the mixture formation, preparation, and pollutant processes.

1.2 Objective

The main objective of this research was to gain an increased understanding of the factors influencing NO\textsubscript{x} formation during stratified combustion. A fundamental theory regarding the formation of NO\textsubscript{x} during stratified combustion in a DISI engine proposes that the burning zone equivalence ratio dominates the NO\textsubscript{x} formation. Inherent in this theory is the assumption that burning zone equivalence ratios are approximately stoichiometric for the bulk of the fuel being burned, thus generating the greatest amount of NO\textsubscript{x} possible. This is exacerbated by the overall lean environment. The question posed by this research was, what is the burning zone equivalence ratio and does a correlation exist between that equivalence ratio and the measured engine-out NO\textsubscript{x} emissions? Therefore, the main goal of the research was to determine if the burning zone equivalence ratio is the dominating factor for NO\textsubscript{x} formation during stratified combustion.

1.3 Approach

In order to answer the questions posed for this research the correct diagnostic techniques needed to be applied. Planar laser-induced fluorescence (PLIF) was chosen as
the principle measurement technique for this research since it allows both the ability to track the flame-front location, and the accurate determination of the equivalence ratio along the flame front. The fuel concentration measurements were performed by imaging 3-pentanone doped into the fuel (isooctane) and the flame was tracked by imaging the OH radical that naturally occurs in the combustion products. This selection of tracer molecules allows for imaging of the two species simultaneously through excitation at a single wavelength. The simultaneous imaging of fuel concentration and the OH radical to track the flame during stratified combustion has never been presented before to the author’s knowledge.

1.4 Outline

This thesis is organized as follows: Chapter 2 is an overview of the pertinent literature. Some background information on DISI engines is given including information on homogeneous operation, stratified operation, and emissions formation. In particular relevant mechanisms of NOx formation are discussed. Chapter 3 discusses the experimental setup used for the work in this thesis. An overview of the engine setup as well as the emissions test equipment, and equipment used for the planar laser-induced fluorescence imaging is given. In Chapter 4 the method used for determining flame-front equivalence ratio is detailed. The chapter includes information on the actual procedure used as well as information detailing the accuracy of the method used by application to homogeneous conditions. In Chapter 5 the results are discussed in the following order: 1. Effect of engine speed, 2. Effect of engine load (overall equivalence ratio), 3. Effect of injection and spark timing, 4. Emissions and pressure data, Chapter 6 presents conclusions and recommendations for possible future work.
2.0 Literature Review

2.1 DISI Engines

2.1.1 A Brief History of DISI Engines

The concept of directly injecting fuel into the cylinder and igniting the air-fuel charge with a spark has been in existence since the 1930’s [1]. Since then significant effort has been aimed at developing the concept so that it could be implemented in a production engine. Early efforts such as the Texaco Controlled Combustion System (TCCS) and Ford’s programmed combustion (PROCO) system utilized diesel injection systems, which were available at that time, but were not ideally suited for the task due to the system limitations [2]. In the late 1980’s and early 1990’s the development of common rail fuel systems with solenoid-actuated injectors greatly increased the flexibility of the injection process and made the possibility of implementation feasible in production.

Today several companies are producing engines for the Japanese and European markets, these include Toyota, Mitsubishi, Volkswagon, Peugeot, and Nissan. However, due to differences in emissions regulations between the United States and these markets no DISI engines are currently offered in the domestic market.

2.1.2 Advantages and Disadvantages of DISI Over PFI Operation

The DISI engine concept offers the possibility of fuel economy approaching that of a diesel engine while maintaining the power density and driving characteristics of a port fuel injection (PFI) spark-ignited engine. Several researchers have estimated that increases in
fuel economy of up to 20-25% over the entire FTP cycle are possible and that increases of up to 40% under light load are possible [2]. In general, these types of fuel economy increases have not been reached, in part, due to emissions constraints.

To achieve significant increases in efficiency with a DISI engine it must be operated with an overall lean, stratified charge over a significant portion of the operating map. Operating with a stratified charge allows the engine to remain almost completely unthrottled thus greatly reducing the pumping losses associated with light load operation in a PFI engine [2]. However, due to the overall lean operation high efficiency three-way catalyst technology cannot be applied.

In addition to the efficiency increase obtained by running under stratified-charge operation, DISI engines have other benefits as well. DISI operation allows for increased compression ratios, which increase the overall thermal efficiency of the engine [2, 3]. Under homogeneous operation the cooling of the incoming air by the direct-injected fuel spray can significantly increase volumetric efficiency [3]. Under all operating conditions the direct injection of the fuel improves transient response and fuel metering due to the elimination of a fuel film in the intake port. Direct injection also allows for the possibility of stopping the injection of fuel when the vehicle is decelerating [2].

As noted earlier not all of the characteristics of DISI operation are necessarily positive. For example, the engine-out emissions of nitric oxides (NOx), hydrocarbons (HC), and soot from a DISI engine can be significantly greater than those of a PFI engine [4,5,6]. Past research indicates that NOx emissions tend to be high, even though the overall equivalence ratios are quite lean during stratified operation. The NOx emissions represent the most significant challenge for engineers due to the oxygen present in the exhaust.
Hydrocarbon formation mechanisms in a DISI engine include piston and cylinder wall wetting, as well as quenching of the flame in the lean periphery of the stratified charge. Several other mechanisms exist, but those mentioned above are considered to be the most significant contributors [7]. Hydrocarbons are a major problem even though oxygen exists in the combustion products for oxidation. One factor contributing to this problem is the lower exhaust temperatures under stratified operation when compared with a PFI engine at a similar load. The lower temperatures mean that very little post-flame oxidation takes place in the cylinder or the exhaust port. The lower exhaust temperatures also have ramifications with respect to catalyst light-off time and overall catalyst efficiency [2, 7].

Strand [8] showed the presence of lean quenching in an optical DISI engine using PLIF of 3-pentanone. The research indicated the presence of fuel with equivalence ratios of $\Phi < 0.3$ as late as 70 crank angle degrees (CAD) after top dead center (ATDC) [8]. The modeling work done by Anderson et al. [7] indicates that the flame-front can propagate through mixtures of equivalence ratio down to $\Phi = 0.3$ and that mixtures with equivalence ratios of $\Phi = 0.2$ can oxidize in ~1.5 ms at a temperatures of 1150 K or greater, but that mixtures with $\Phi < 0.1$ at 1150 K will not oxidize completely. Both of these studies show that lean quenching of the flame in the fuel cloud periphery is of particular concern.

Soot formation, although not necessarily a major problem for PFI engines, can be problematic under stratified DISI operation. The late injection timings associated with stratified charge result in portions of the fuel having very little time to mix resulting in rich pockets of fuel in cylinder. Burning of these rich pockets results in significantly increased amounts of particulate matter of increased size and number density when compared with
PFI engines [2]. Maricq et al. [6] performed particulate matter (PM) measurements on a 1.83L production DISI engine. Their data showed a significant increase in the number and size of particles as injection timing was retarded, and as engine load and speed were increased. The maximum load for stratified-charge operation is mainly limited by the formation of soot/smoke.

DISI engines do have some advantages in pollutant emissions over PFI engines. These advantages are mainly limited to startup due to the increased accuracy in fuel metering gained with a DISI engine and include reduced carbon monoxide (CO) and HC emissions at startup. There are also lower carbon dioxide (CO₂) emissions due to the increased fuel economy [2].

2.1.3 Homogeneous Operation

For operation at loads greater than approximately 40-60% of full load, a DISI engine must be operated homogeneously similar to a PFI engine due to the large amounts of soot formed under late-injection stratified operation [7].

Homogeneous operation of a DISI engine has several intricacies not present with a PFI engine due to the in-cylinder fuel injection. One of the advantages of the DISI engine versus the PFI engine under homogeneous operation is the fact that direct injection of fuel during the intake stroke provides evaporative cooling. Anderson et al. performed tests and calculations to determine the effects of evaporative cooling on engine performance. They were able to attain an increase in the volumetric efficiency of approximately 2-3%, a decrease in charge temperature of up to 50-60° C, and an increase in the max torque output by 5-10% due to the effect of evaporative cooling of the incoming charge [3].
The injection timing during homogeneous operation is nontrivial because both the advantage of the evaporative cooling and avoiding spray impingement on the piston must be taken into account. It is also important to note that studies have shown that even while injecting during the intake stroke, where significant time is available for mixing, inhomogeneities can exist in the mixture at spark time, which can lead to combustion instability and emission problems. In general, the pollutant emissions under homogeneous operation are similar to those of a PFI engine if the in-cylinder charge is approximately homogeneous [2].

2.1.4 Stratified Operation

Ideally when operating with a stratified charge, an engine would be designed to place an air-fuel mixture at the spark plug at the time of ignition with the bulk of the mixture being slightly rich, and without any of the fuel mixing beyond the lean flammability limit. The locally rich equivalence ratio would give low NO\textsubscript{x} generation while the overall lean operation would provide a large amount of available oxygen to oxidize any products of incomplete combustion such as CO, and unburned hydrocarbons. In practice there is no engine that is able to achieve this idealized operation.

There are several components one must consider when designing an engine for stratified operation. The injector is one of the most important components of any DISI system. Several injector characteristics must be considered in the design process including atomization, spray penetration, and entrainment. The atomization characteristics of the spray will greatly affect the lifetime of the liquid spray droplets in the engine, which can greatly impact mixing, and the penetration of the spray. In general good atomization is
considered desirable, with Sauter mean diameters (SMD) on the order of 20-30 μm being typical for pressure swirl injectors commonly used in DISI applications [2].

The penetration of the fuel spray is also important because it determines, along with the injection timing, when the spray will impact the piston. Because the piston geometry is usually used to direct the fuel charge to the spark plug the spray penetration will affect the optimal ignition time, and the range of usable ignition times.

The amount of air entrainment defines the mixing process, and, therefore, the amount of fuel in regions below the lean flammability limit. Air entrainment increases the air utilization, but too much mixing at low overall equivalence ratios can decrease the local equivalence ratios and reduce combustion stability.

In addition to the injector both the piston geometry and the spark plug are important components. As stated earlier the piston geometry is generally used to the direct the fuel towards the spark plug and aid in forming an ignitable mixture at the spark gap. An important issue is the amount of spray impingement and the distribution of the fuel film on the piston, which can be a significant source of unburned hydrocarbons and soot [2].

The relative placement of the spark plug and injector are very important in DISI engines using stratified operation. There are two general injector-spark plug spacing concepts used, wide spacing and narrow spacing [2]. As the names suggest in one case the injector and spark plug are located relatively far apart in the combustion chamber whereas in the other case they are located relatively close to one another. In general it is desirable to locate the injector on the intake side of the engine due to the high temperatures on the exhaust side that can lead to injector durability issues. The spark plug should be located centrally in the chamber to minimize the distance the flame must travel under homogeneous
operation and reduce the possibility of knock [2]. The spark plug must also have high discharge energy to assure repeatable combustion so individual high-energy coils are often employed.

To achieve stable stratified operation the piston position, injection timing, and ignition timing must be properly coordinated. Timing of injection and ignition is nontrivial. In order for an engine to operate properly stratified, the piston must be at a proper distance from the injector at the start of injection to direct the stratified charge to the spark plug at the proper ignition timing. The system is constrained in several ways: first the piston position is constrained by the slider crank mechanism, secondly, the spray penetration and injection rate are constrained because injectors generally are operated at the same pressure over all operating conditions [2]. The above-mentioned constraints cause the system to operate stably over a small injection and ignition timing range. Kaiser et al. [5] performed engine-out emissions measurements on a 1.83L DISI 4-cylinder engine. Data were taken for cases with high stratification and significant changes in engine-out emissions were seen for changes in end of injection (EOI) of 10-20 CAD.

There are some additional effects to operating an engine lean and stratified. The specific heat ratio of the mixture in-cylinder is increased [2]. The combustion duration under stratified charge is short due to the concentration of fuel in a small volume. The exhaust gas temperatures are significantly reduced due to the large amount of dilution from the additional air. Lower exhaust gas temperatures are troublesome because at idle, unless other measurers are taken, the amount of time for the catalyst to light off is greatly increased, and when operating under stratified operation for long periods of time the exhaust temperature can drop low enough to significantly reduce catalyst efficiency [2].
2.1.5 NO$_x$ Formation During Stratified Charge Operation

There are several mechanisms by which NO$_x$ is formed during combustion. These include the Zeldovich mechanism, the Fenimore (prompt) mechanism, the N$_2$O intermediate mechanism, as well as several pathways for nitrogen bound in the fuel. In the environment of the IC engine the Zeldovich mechanism generally predominates. The Zeldovich mechanism is highly temperature dependent due to the high activation energy of the rate-limiting step. As a rule of thumb the mechanism does not become a significant pathway for NO$_x$ production until temperatures greater than 1800 K are reached [9].

Several researchers have shown that engine-out NO$_x$ emissions during stratified operation are greater than those of a PFI engine at the same load [4, 5]. Casarella and Ghandhi listed burning zone equivalence ratio as one possible emissions formation mechanism, stating that the highest NO$_x$ emissions will occur at equivalence ratios slightly lean of stoichiometric [10]. This theory relies on the Zeldovich mechanism being the main cause of NO$_x$ production. The higher compression ratios, and a larger compressed mass may also contribute to the NO$_x$ emissions [7].

Although the global burning zone equivalence ratio may be close to stoichiometric, throughout the cylinder significant mass of fuel exists that can be significantly richer than stoichiometric or significantly leaner than stoichoimetric. In the case of locally rich areas, the Fenimore or prompt mechanism may be another production mechanism of NO$_x$. In areas of lean mixture the N$_2$O intermediate mechanism may provide a pathway for NO$_x$ production [9].
2.2 Planar Laser-Induced Fluorescence (PLIF) Imaging

2.2.1 Description of PLIF Technique

Light formed into a sheet from a tunable laser source is sent through the flow of interest. A fraction of the incident light can be absorbed if the incident photons have energy resonant with an allowable transition in the absorbing molecules. The energy is then emitted as photons of a different wavelength than the incident laser light due to energy transfer to nearby rotational and vibrational energy levels. A portion of this light can be imaged by a detection device and quantitative and qualitative information can be gained.

If we assume a simple two energy level model in the linear fluorescence regime, the number of photons collected by the detection device, \( N_p \), is given by

\[
N_p = \eta \frac{\Omega}{4\pi} f_1(T) n_a V_c B_{12} E_\nu S
\]

where \( \eta \) is the collection efficiency of the optics, \( \Omega \) is the collection solid angle [sr], \( f_1(T) \) is the fractional population of the lower laser-coupled state, \( n_a \) is the absorbing species number density [cm\(^{-3}\)], \( V_c \) is the collection volume [cm\(^3\)], \( B_{12} \) is the Einstein B coefficient of the transition, \( E_\nu \) is the spectral fluence of the laser [J cm\(^{-2}\) Hz\(^{-1}\)]. Although (1) was derived for a two level system, the equation holds for several conditions with real molecules [11].

2.2.2 Experimental Considerations

A PLIF imaging system typically consists of a laser light source, a detection/camera system, and sometimes a dopant added to the flow to track a particular quantity. All of these components have several considerations that must be taken into account when selecting a PLIF imaging system.
2.2.3 Laser Selection

Hanson et al. reported six factors to consider when selecting a laser source for a PLIF experiment: center frequency, bandwidth, tunability, intensity, temporal resolution, and beam quality [10]. The laser selected must be capable of producing a center frequency that overlaps the transition of interest. No laser beam is purely monochromatic so the bandwidth of the laser output must be taken into consideration. Too wide of a bandwidth will provide inefficient excitation, in general the bandwidth should be approximately the same width as the absorption transition for efficient excitation. In order to generate a measurable signal, a laser source must provide a high intensity beam. The temporal behavior of the laser will affect the temporal resolution of the system. Pulsed lasers with short pulse widths on the order of several nanoseconds are able to effectively freeze the flow. However, due to the relatively low pulse rate of most of these systems (~10-100 Hz) the data acquisition rates can be relatively slow compared with the events in an engine. Finally, the beam quality of the laser is important because a laser sheet of relatively uniform intensity is desirable. The intensity variations in the sheet can be corrected, but this adds noise to the data.

2.2.4 Fluorescence Detection System

The detection system used in the collection of the fluorescence data can greatly affect the quality of the data. The low light levels generated from PLIF experiments and the harsh environment of the IC engine make the selection of a detection system nontrivial for a given application. Hanson et al. listed several parameters to consider when choosing a detection system: the system quantum efficiency, the linearity of the system response, the
dynamic range of the system, the ability to shutter rapidly, the data acquisition rate, the noise associated with the system, and the spatial resolution [11].

The signal detected by an imaging system is proportional to the system quantum efficiency. Low quantum efficiency detection systems suffer from low signal-to-noise ratios in low light environments. The main benefit of a high quantum efficiency device is that it provides noise free gain.

The linearity of the detection system response must also be considered. Media such as film offer several advantages over electronics, but the nonlinear response of film makes quantitative interpretation of data difficult [11]. In conjunction with linearity, the dynamic range of the system can also limit the interpretation of data. In an engine during stratified combustion equivalence ratios as high as 3 or 4 can be reached and equivalence ratios as low as 0.05 or slightly lower may exist in the unburned region in-cylinder. A system with high dynamic range will be capable of detecting the two orders of magnitude difference in signal level where a system with a small dynamic range will only be able to accurately detect a fraction of the range of equivalence ratios present.

The ability to reject combustion luminosity when taking PLIF images in an engine will determine whether the data can be quantitatively interpreted. Two factors affect a detection systems ability to reject combustion luminosity: the ability to rapidly shutter or gate the detector, and the rejection ratio. Short gate times (~100 ns) decrease the integration of incident luminosity. During the portion of the engine cycle when the camera is not being operated a high rejection ratio (~$10^5:1$) is necessary. In the absence of a high rejection ratio the generally long integration time associated with image readout can cause significant luminosity signal to be detected by an imaging device.
Although not always of significant importance the ability of a detection system to rapidly acquire images can be useful. The data acquisition rate is relatively important when imaging inside a combusting environment that generates soot. If the imaging system used with an engine can take a picture every engine cycle the maximum amount of data can be acquired before having to clean the windows of the engine. This can result in a factor of two increase in the data acquired when can compared with a system that can image every other cycle.

The imaging system noise response will determine the overall achievable signal-to-noise ratio (SNR). Rothamer and Ghandhi highlighted the need for a high SNR when trying to resolve small-scale fuel distribution fluctuations [12]. In general the noise associated with the detection system is the dominating noise source for signal to noise ratios less than 30:1. Therefore, for most imaging experiments performed in engines, the detection system SNR will determine the PLIF system noise [12].

The spatial resolution of the detection system determines the scale of the smallest object that can be determined. Both the lens selection and the camera itself will affect spatial resolution. If the camera is intensified the spatial resolution can be limited by the channel spacing in the microchannel plate. Even if the camera used is not intensified the response of the camera to a step input will affect the spatial resolution.

### 2.2.5 Selection of Dopant for Fuel Concentration Measurements

When tracking fuel concentrations in an engine a dopant with known fluorescence characteristics is usually added to the fuel to make measurements. For maximum accuracy and fidelity of fuel concentration measurements several conditions should be met: the
concentration of the dopant selected should faithfully track the fuel concentration; the absorption cross-section and fluorescence efficiency of the dopant should have little dependence on the local thermodynamic conditions; the dopant should not greatly alter engine performance and should have little effect on the liquid phase properties of the fuel [13].

Boiling point has, in the past, been used as the main factor in determining if a dopant will faithfully track the fuel concentration. Therefore, 3-pentanone has been used with iso-octane to simulate gasoline since 3-pentanone has a boiling point of 102°C as compared to 99°C for iso-octane. However, LeCoz et al. [14] and Han and Steeper [13] have shown that non-ideal behavior during the evaporation of iso-octane/3-pentanone mixtures can occur and other fuel-dopant combinations have shown this behavior [13, 14, 15]. The main issue with the studies that have been performed is that they looked at the evaporation of the mixture at a constant temperature over long time scales (~10 min). In an engine the time scales are ~5 orders of magnitude less, therefore, many of the mass diffusion effects which have significant time to occur in a 10 min could be insignificant in the time spans occurring in a IC engine. No one has yet proven that a 3-pentanone iso-octane mixture provides significant error in tracking the fuel, and studies, such as those done by Johansson et al. [16], using the 3-pentanone –iso-octane combination have shown a strong correlation between fluorescence measurements and engine performance, indicating that 3-pentanone served well as a fuel tracer.

Kim et al. observed temperature suppressions of ~ 200 K were observed with diesel injection [17]. Therefore, the dependence of a dopant’s fluorescence on temperature is important to consider. Ghandhi et al. performed experiments with acetone and 3-pentanone
that showed the dependence of the fluorescence signal on temperature for ketones [18]. In general the temperature dependence of a dopant’s fluorescence is a function of the dopant and the excitation wavelength as shown by Thurber et al. for acetone. Figure 2.1 shows the temperature dependence of acetone fluorescence versus temperature at various excitation wavelengths [19]. Ketones exhibit similar fluorescence characteristics so the acetone data can be used to infer a relationship for 3-pentanone as well. From Fig. 2.1 it is apparent that selection of an excitation wavelength between 276 nm and 300 nm greatly reduces the temperature dependence of the fluorescence for ketones.

In addition to temperature, the fluorescence signal can also depend on pressure.

Figure 2.1. Temperature dependence of acetone fluorescence signal [19]
Pressure is homogeneous throughout the cylinder in an engine so pressure differences within an image are not an issue [12]. However, pressure differences may exist between the calibration image and the data image, therefore, the pressure must be simultaneously recorded. Thurber et al. performed experiments on the pressure dependence of acetone fluorescence using several excitation wavelengths. Their results indicated that the pressure dependence of the fluorescence can be alleviated through the use of wavelengths close to 308 nm [20]. Again, we can assume similar behavior for 3-pentanone due to the similarity of fluorescence characteristics for all ketones. Figure 2.2 shows the effect of pressure on the fluorescence of acetone in a nitrogen bath gas for three different excitation wavelengths, 248 nm, 266 nm, and 308 nm [20].

2.2.6 CCD Camera Noise

The noise characteristics of the detection system used to image the fluorescence are especially important due to the impact they have on the quantitative interpretation of the data. Charge coupled device (CCD) based detection systems are currently the standard detection device for PLIF imaging experiments. They offer the benefits of excellent linearity, good quantum efficiency, low noise, and low cost [12].

Janesick et al. stated several sources for CCD noise: dark noise, read noise, shot noise, and fixed pattern noise [21]. The dark noise and read noise of the detection system are related to the chip itself; whereas fixed pattern noise can be related to both the chip
Figure 2.2. Pressure dependence of acetone fluorescence signal [20].

to both the chip and whatever is being imaged. Shot noise is independent of the detection device and is related to the probabilistic nature of photon detection.

The generation of dark noise is due to photoelectrons thermally generated in the silicon of the detector. Dark noise is highly dependent upon the detector temperature and the readout time. Dark noise is reduced by approximately a factor of 2 for every 6°C reduction in temperature [22]. Scientific grade sensors are often cooled due to the dependence of dark noise on detector temperature. Dark noise is a constant for a constant detector temperature, exposure duration, and readout time.
Read noise is generated during the process of reading out the electrons from the detection array and is dependent upon the readout rate of the array and the readout electronics of system [21]. More expensive camera systems with cooled scientific grade CCDs feature ultra-low noise readout circuits to reduce read noise. For a given camera system at a set readout rate the read noise is a constant.

The probabilistic nature of photon detection is described well by Poisson statistics. The noise arising from detecting photons for a fixed time process is termed shot noise. Shot noise gives \( SNR = \sqrt{N_{pp}} \) where \( N_{pp} \) is the number of photons incident per detection element (pixel) [21]. In the high signal limit, the noise response of any CCD based detection system is limited by the shot noise.

A repeatable intensity pattern in an image is the source of fixed pattern noise. Possible sources of fixed pattern noise include pixel-to-pixel sensitivity variation in the CCD array, and any other repeatable pattern [21]. A good example of fixed pattern noise is the chicken wire pattern visible on images taken with intensified CCDs at low gain levels. This pattern is caused by the interfaces between the fiber-optic bundles used to couple the intensifier to the CCD. Unlike the other sources of noise, it is possible to correct for fixed pattern noise by normalizing the data image with an image of an evenly illuminated field.

Rothamer and Ghandhi measured the camera response of several camera systems, the camera noise response of a cooled, thinned, back-illuminated scientific grade CCD camera is given in Fig. 2.3 [12]. As can be seen from Fig. 2.3, at low illumination levels the noise response is approximately constant due to the noise floor of the camera system, which is composed of the sum of the read noise and dark noise. As the illumination is increased the
Figure 2.3. Experimentally determined response of a CCD based detection system [12].

Shot noise becomes a significant portion of the total noise. Once the illumination level is significantly high, the noise response begins to follow the shot-noise limited slope of 0.50 when plotted in log-log coordinates. Fixed pattern noise is not evident in the curve displayed in Fig. 2.3 due to the relatively low range of number of photons per pixel ($N_{pp}$) tested. A line with slope of $\frac{1}{2}$ is included as a reference.

2.2.7 Image Intensifiers

Due to the fact that the gate times of CCDs are generally not fast enough to provide significant rejection of combustion luminosity, it is often necessary to incorporate the use of an image intensifier in a detection system used in IC engine applications. Intensifiers provide short gate times (~1-100 ns) and high rejection ratio (~10^5:1), and therefore, are effective at rejecting light from luminous combustion environments [12]. A schematic of an
image intensifier is shown in Fig. 2.4. When a photon of light of energy $h\nu$ strikes the photocathode of the intensifier an electron is ejected from the surface with some associated quantum efficiency. The photoelectron is then accelerated toward the entrance of the microchannel plate (MCP) by a voltage potential. This potential can be biased on and off very quickly (~1 ns). With the potential biased off the gap is large enough that virtually no photoelectrons can reach the MCP. Upon entering the MCP the electrons undergo a cascade gain process similar to what occurs in a photomultiplier tube. The electrons exiting the MCP are accelerated toward a phosphor-coated surface by a potential. The electrons strike the phosphor and are converted back to photons at the phosphor wavelength with an associated quantum efficiency. This light is coupled to the detector by either a tapered fiber optic bundle or a lens [21].

**Figure 2.4.** Schematic of a microchannel plate image intensifier [12].
Figure 2.5. Experimentally determined noise response for an intensified camera system [12].

The addition of an image intensifier to a CCD detection system has some disadvantages. The overall system efficiency is quite low due to the additional quantum efficiencies associated with the photocathode and phosphor surfaces, and the gain process in the MCP causes an amplification of the incident shot noise [12]. Paul [23] reported the overall detector SNR response with an image intensifier

\[
SNR = \frac{\eta G N_{pp}}{\sqrt{\eta G^2 \kappa N_{pp} + N_x^2}}
\]  

(2)

where \(\eta\) is the photocathode quantum efficiency, \(G\) is the photonic gain, \(\kappa\) is a gain-dependent noise factor that is always greater than 1 and usually between 2 and 4, and \(N_x\) is the photon equivalent of the CCD detector noise and includes the read noise and dark noise.
In the shot noise limit where $\eta G^2 \kappa N_{pp} > N_e^2$ the effect of the intensifier, as noted by Paul, is to reduce the quantum efficiency of the system by the noise factor $\kappa$.

Rothamer and Ghandhi measured the system response of an intensified CCD based detection system. Their measured response is shown in Fig. 2.5 for various levels of gain [12]. As can be seen from the graph at medium to high gain levels the noise response is shot-noise limited through the entire range, this is due to the shot-noise amplification in the intensifier and this noise dominates over the CCD noise response even at very low illumination levels. Rothamer and Ghandhi concluded that when operating a detector with an image intensifier at moderate to high gain levels the CCD behind the detector has little effect on the overall noise response of the system [12].

2.2.8 Influence of PLIF System Noise on the Quantitative Interpretation of Data

As mentioned earlier the noise due to the detection system can greatly affect the interpretation of quantitative data. Rothamer and Ghandhi stated that the imaging system is the main contributor to noise in PLIF experiments with SNR $\approx 15$ [12] and listed other noise sources as being shot-to-shot laser sheet power and profile variations. They illustrated the effect of camera noise on the quantitative interpretation of data by looking at results taken at two significantly different SNRs. Figures 2.6 (a) and (b) show images taken at SNRs of 31 and 14 respectively along with cross-sections of these images [12]. The cross-sections of Figure 2.6 are depicted with error bars shown for the 95% confidence interval at each pixel.

The cross-section through an image of SNR = 31 shows great detail with respect to the homogeneity of the picture being imaged. In this image almost all of the relative
Figure 2.6. Images and cross-sections of image taken in a DISI engine with EOI = 180° BTDC and a picture timing of 20° BTDC (a) SNR ~31:1 (b) SNR ~14:1. [12].
changes in intensity can be confidently correlated with an actual change in fuel concentration allowing the conclusion that significant inhomogeneities exist in the image to be confidently made. In Figure 2.6 (b) the image appears to have inhomogeneities present in the area where the cross-section is being taken, but after referencing the cross-section and accounting for the error bars one cannot make a definite conclusion as to the state of the mixture. Therefore, when trying to garner information from relatively low SNR data (SNR ≳ 10) one must rely more heavily upon averaging and other statistical sampling to develop conclusive evidence.

### 2.2.9 Application of PLIF in Engines

PLIF has been applied for both qualitative and quantitative analysis in engines. In particular PLIF has been used for 2-d visualization of the flamefront, visualization and measurements of the fuel distribution, visualization and measurement of emission species such as NO, and measurement of temperature distributions in engines [24 - 29].

The hydroxyl (OH) radical has been used in many instances to track the burned region in-cylinder. OH is a naturally occurring radical in the combustion products and its spectroscopy has been well studied with significant data being available. Suntz et al., and Arnold et al. have successfully imaged the OH radical and have found it to be a valuable tool for tracking flame-front location [25, 28].

Imaging of the fuel distribution is one of the most commonly performed experiments using PLIF in engines due to the influence of the fuel distribution on combustion. Several studies have performed both qualitative and quantitative measurements of fuel distributions in both PFI and DISI engines [26-29]. Acetone and 3-pentanone are the most commonly
used dopants due to their boiling points and the readily available spectral data. Excitation is generally performed with the fourth harmonic of an Nd:YAG laser (266nm), the frequency doubled output of a tunable dye laser pump, or an excimer laser (248 nm, 308 nm).

Several studies have simultaneously imaged the fuel distribution and the flame-front using fluorescence from a dopant such as 3-pentanone and fluorescence from the OH radical [28, 29]. However very little work has been done while imaging stratified fuel distributions during combustion, and almost no work exists on simultaneous imaging of both the fuel distribution and the flame front during stratified combustion.
3.0 Experimental Setup

The experimental setup for this study consisted of three main components: the GMR Triptane (base 4) optically accessible engine outfitted for DISI operation, the laser and optics system used as an excitation source for the PLIF measurements, and the imaging system consisting of two intensified cameras used to record the fluorescence from the fuel dopant and the OH radical.

3.1 Optical Engine

3.1.1 Engine Dimensions

The optical engine used in these experiments was a single-cylinder, four-stroke engine modified for direct-injection spark-ignition operation. The basic dimensions and specification of the engine are given in Table 3.1.

<table>
<thead>
<tr>
<th>Table 3.1 – Basic engine specifications</th>
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<tr>
<td>Bore [mm]</td>
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<tr>
<td>Stroke [mm]</td>
</tr>
<tr>
<td>Clearance Volume [mm³]</td>
</tr>
<tr>
<td>Compression Ratio</td>
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<td>Connecting Rod Length [mm]</td>
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<td>Intake Valve Opening [BTDC]</td>
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<td>Exhaust Valve Closure [ATDC]</td>
</tr>
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</table>
3.1.2 Combustion Chamber Design

The cylinder head and piston of the engine were designed for wall guided swirl directed DISI operation. The piston features included a flat top with a 10 mm deep reentrant bowl located off center of the cylinder axis, as shown in Fig. 3.1. The head was flat with 1 intake and 1 exhaust valve and a centrally located spark plug. The injector was oriented tangent to the edge the piston bowl with a 70° vertical angle relative to the horizontal plane of piston crown. The spark plug was centrally located in the cylinder and the injector was located directly across the piston bowl from the spark plug as illustrated in Fig. 3.1.

Under the late injection timings typical of stratified operation the injector orientation caused the spray to impinge upon the piston surface, rebound, and convect toward the centrally located spark plug. The intake port was designed to generate swirl in the same direction as the fuel spray. The swirl was intended to interact with the fuel and

![Figure 3.1. Optical DISI engine combustion chamber layout.](image)
aid in the convection of the spray towards the spark plug while maintaining a tightly stratified mixture. When timed properly the injection and subsequent convection generated a stratified mixture located in the region of the spark plug.

### 3.1.3 Optical Access

Optical access to the combustion chamber was achieved through the use of a steel spacer ring, a Bowditch piston, and an access port in the side of the cylinder block. Figure 3.2 illustrates the components used to gain optical access to the combustion chamber. The steel spacer ring had two diametrically opposed quartz windows allowing the laser sheet to enter and exit the engine cylinder. Light from the fluorescence was transmitted through a sapphire window in the bottom of the piston bowl. The interchangeable piston cap was attached to the Bowditch piston. The piston body was approximately 25 cm long and had three legs attaching the top portion connected to the piston cap to the bottom portion used to attach to the connecting rod. The piston design allowed a $45^\circ$ mirror to be placed below the

![Figure 3.2. Optical access in the DISI optical engine.](image-url)
piston cap in the side of the cylinder block, as shown in Fig. 3.2. The mirror reflected the fluorescence light out of the engine through a hole in the cylinder block and onto two CCD cameras used to make measurements. The light path through the engine is depicted in Fig. 3.2.

3.1.4 Fuel Injectors, and Fuel System

Two different injection systems were utilized for this experiment, a Chrysler-designed high-pressure swirl-type injector, and an Orbital air-assisted injector. The Chrysler injector was operated by sending a TTL signal with the desired injection pulse width to a 70 V Chrysler-designed driver. Two 4-amp peak 1-amp hold Air Products driver circuits were used to operate the Orbital injection system. The Orbital injection system was used as a means of preparing a completely homogeneous mixture for calibrating the data images and for operating the engine homogeneously.

The Orbital injector was operated at an air pressure of 550 kPa with a 69 kPa differential between the fuel and air. The injector operates by injecting the fuel into a chamber in the back of the air-fuel injector 4 ms before the air-fuel injection takes place. The air-fuel injector then opens and air flows over and through the fuel for 3 ms creating a finely atomized spray. This injector was located in the intake system 1 m before the intake port. When operating the engine with this injector the fuel air charge entering the cylinder was considered completely homogeneous.

Both injection systems were calibrated such that the mass of fuel for a given injection duration was known. Changing the injection duration controlled the mass of fuel injected.
A single piston-accumulator-based fuel system was used to supply high pressure fuel to both injectors at the same time. The system provided gauges for monitoring the high pressure fuel (5.2 MPa) used to supply the Chrysler injector, the air pressure supplied to the air-assist injector, and a differential pressure gauge for monitoring the pressure difference between the air and fuel for the air-assist injector.

The accumulator, uses a piston to separate the nitrogen, supplied from bottles of compressed nitrogen, from the fuel eliminating the possibility of large quantities of nitrogen dissolving into the fuel. A pressure regulator was adjusted to step the pressure down to 619 kPa necessary to for the air-assist injector rail. This setup allowed both injectors to be operated concurrently, which was an important feature for acquiring homogeneous mixture images immediately before and after the data image sets.

### 3.1.5 Engine Timing

A programmable engine controller was used to send TTL pulses to the injector drivers, spark coil driver, lasers, and cameras. The controller used was built in house. The engine controller was synchronized with the engine via a BEI shaft encoder coupled to the crankshaft. The encoder provided resolution of 0.25 crank angle degrees (CAD) for the precise timing of engine events, and image acquisition.

### 3.1.6 Pressure Measurement

Pressure measurements were acquired with a Kistler model 7603 piezo-electric pressure transducer connected to a Kistler model 5004 charge amplifier. The charge amplifier and pressure transducer combination was calibrated with a dead weight pressure tester and gave correlation coefficients ($R^2$) greater than 99.99%. The signal from the charge
amp was recorded with a Hi-Techniques Win600 data acquisition system running
REValation engine acquisition software.

3.1.7 Experimental Running Conditions

Running conditions were chosen for this experiment in order to investigate the
changes in flame-front equivalence ratio under a wide variety of operating conditions. Data
were acquired at engine speeds of 600 rpm and 1200 rpm. Engine coolant temperature was
controlled to 65°C for the 600 rpm data, and 60°C for the 1200 rpm data. All of the data sets
were taken under atmospheric intake conditions with an intake pressure of 98.5 ±1kPa.

The baseline running conditions are shown in Table 3.2. These conditions were
determined by performing injection and ignition timing sweeps and optimizing the indicated
mean effective pressure (IMEP) and coefficient of variation (COV) of IMEP. The one
exception to this is the $\Phi = 0.69$ case, where the condition was chosen because it
represented a slightly stratified mixture, which could be used to test the sensitivity of the
flame-front detection method described in Chapter 4.

The conditions tested range from completely homogeneous, to highly stratified. The
homogeneous data were taken because the equivalence ratio at the flame front was known
exactly allowing the data to be used as a test case for the image correction procedure and to
check the validity of the method used to determine the flame-front equivalence ratios.

The stratified cases tested were chosen to determine the effect of overall equivalence
ratio on the flame-front equivalence ratio. The time for mixing in the $\Phi = 0.27 - \Phi = 0.62$
cases increases by only 5 CAD, therefore, one would expect significant differences in the
fuel distributions due to the significant change in overall equivalence ratio.
In addition to the baseline cases shown in Table 3.2 several other conditions were tested for the equivalence ratios of $\Phi = 0.32$ and $\Phi = 0.42$. These conditions included both a sweep of spark timing while holding the injection timing constant, and a sweep of injection timing while holding the spark timing constant. In the case of varying spark timing data were taken by varying the spark $\pm 5$ CAD from the baseline case in 2.5 degree increments.

<table>
<thead>
<tr>
<th>Equivalence Ratio</th>
<th>Injection Type</th>
<th>Engine Speed (rpm)</th>
<th>EOI (BTDC)</th>
<th>Spark (BTDC)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.27</td>
<td>Stratified</td>
<td>600</td>
<td>47.5</td>
<td>15</td>
</tr>
<tr>
<td>0.32</td>
<td>Stratified</td>
<td>600</td>
<td>50</td>
<td>17.5</td>
</tr>
<tr>
<td></td>
<td>Stratified</td>
<td>1200</td>
<td>55</td>
<td>17.5</td>
</tr>
<tr>
<td>0.42</td>
<td>Stratified</td>
<td>600</td>
<td>52.5</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>Stratified</td>
<td>1200</td>
<td>55</td>
<td>20</td>
</tr>
<tr>
<td>0.52</td>
<td>Stratified</td>
<td>600</td>
<td>55</td>
<td>15</td>
</tr>
<tr>
<td></td>
<td>Stratified</td>
<td>1200</td>
<td>57.5</td>
<td>20</td>
</tr>
<tr>
<td>0.61</td>
<td>Stratified</td>
<td>600</td>
<td>57.5</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>Stratified</td>
<td>1200</td>
<td>60</td>
<td>20</td>
</tr>
<tr>
<td>0.69</td>
<td>Stratified</td>
<td>600</td>
<td>180</td>
<td>20</td>
</tr>
<tr>
<td></td>
<td>Stratified</td>
<td>1200</td>
<td>180</td>
<td>30</td>
</tr>
<tr>
<td>0.81</td>
<td>Homogeneous</td>
<td>1200</td>
<td>N/A</td>
<td>30</td>
</tr>
<tr>
<td>0.89</td>
<td>Homogeneous</td>
<td>600</td>
<td>N/A</td>
<td>20</td>
</tr>
</tbody>
</table>
Injection timing was varied ± 10 CAD from the baseline case in 5 degree increments. The varying of injection and spark timing were performed at both engine speeds tested.

3.1.8 Emissions Measurement (NOx and CO)

Measurements of NOx and CO were made using the Wisconsin Small Engine Consortium emissions bench. The exhaust gas was sampled from a mixing tank located approximately 2 m from the exhaust valve of the engine. To completely remove the water vapor from the sample it was passed through two filters and an ice bath prior to being sent to the emissions bench. The ice bath was maintained at 0°C through the use of a slurry of water and ice. The output of the ice bath was connected to the emissions bench using the shortest possible length of Teflon tubing in order to make the transit time from the ice bath to the analyzers as short as possible.

Only the NOx and CO analyzers on the emissions bench were used. These analyzers were connected in parallel to the emissions sampling line. The NOx analyzer used was a ThermoElectron 10A chemiluminescent analyzer. A Horiba PIR-2000 non-dispersive infrared (NDIR) analyzer was used for the CO measurements. The voltage output of the analyzers used was acquired with a Hi-Techniques Win600 data acquisition system running REValation engine acquisition software. The signals were recorded for 100 engine cycles and were averaged later.

The main reason for trying to reduce sample transit time was the fact that the optical engine was not able to fire continuously for an extended period, so the engine may never have had time to reach a steady state. When firing the heat transfer to the piston and cylinder surfaces caused the top-sealing ring on the piston and the rider rings to expand
creating higher friction along the cylinder, in turn, causing the rings to heat up even more. This process built on itself to the point that when you stopped the engine it was hard to rotate the crankshaft of the engine while this would normally require only a small amount of force. This process became even more acute at 1200 rpm due to the increased sliding friction; therefore, emissions data were only taken for the 600 rpm conditions.

The piston ring friction affected the available time for sampling for almost all of the operating conditions. Due to this fact a test plan was devised that would give repeatable results while not damaging the engine and, hopefully, still show measurable differences in the different running conditions being tested. Engine load or equivalence ratio had a significant effect on the rate at which the piston and rings were heated, and the testing procedure had to be varied for each load condition. In general the procedure used consisted of 4 steps: 1. motor the engine, 2. fire the engine, 3. take emissions data, and 4. stop the engine and let it cool. This procedure was repeated for all of the data sets taken at each load condition. The specific durations for the different load conditions for each step of the process are shown in Table 3.3.

Table 3.3.  Emissions testing procedure test durations

<table>
<thead>
<tr>
<th>Equivalence Ratio</th>
<th>coolant temperature (°C)</th>
<th>Motored Duration (s)</th>
<th>Fired Duration (s)</th>
<th>Rest Duration (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.32</td>
<td>66</td>
<td>120</td>
<td>150</td>
<td>180</td>
</tr>
<tr>
<td>0.42</td>
<td>66</td>
<td>120</td>
<td>90</td>
<td>240</td>
</tr>
<tr>
<td>0.52</td>
<td>60</td>
<td>120</td>
<td>75</td>
<td>240</td>
</tr>
<tr>
<td>0.61</td>
<td>60</td>
<td>120</td>
<td>60</td>
<td>240</td>
</tr>
<tr>
<td>0.81</td>
<td>60</td>
<td>120</td>
<td>60</td>
<td>N/A</td>
</tr>
</tbody>
</table>
3.2 **PLIF Imaging**

The PLIF imaging setup consisted of three main components: the tunable laser light source, optics for forming the laser light sheet, and cameras to image and digitize the fluorescence images.

The dopant used to take all of the data was 3-pentanone (diethyl-ketone), which was doped into pure iso-octane (2,2,4-trimethylpentane) fuel at a concentration of 10% by volume. The boiling points of 3-pentanone and iso-octane are 102°C and 99°C, respectively. Due to the fact that the boiling points match so closely they can be considered coevaporative and one can assume that the 3-pentanone faithfully tracks the iso-octane concentration (see § 2.2.5).

3.2.1 **Lasers and Optics**

The laser system used for this experiment consisted of two main components, a Neodymium:Ytrium Aluminum Garnet laser (Nd:YAG) (Spectra Physics model GCR-170), and a tunable dye laser (Lambda Physik, Scanmate 2E) using a Rhodamin 6G dye dissolved in methanol as the dye solution. Figure 3.3 shows the optics and relative laser positioning used.

The 1064 nm output of the Nd:YAG was frequency doubled to 532 nm using a harmonic generator installed on the output of the laser. A dichroic mirror that reflected the 532 nm light and transmitted the 1064 nm light was used at the exit to reject any of the residual 1064 nm light. The laser beam was then reflected through two right angle reflecting prisms to the dye laser.
The 532 nm pump beam entering the dye laser was used to pump the dye in the pre-amp/oscillator and in the main amplifier. The dye concentrations in the pre-amp/oscillator cuvet and main amplifier cuvet were 0.12g/L and 0.4g/L, respectively. A grating in the dye laser provided the ability to tune the laser over a limited wavelength range, specific to the dye, within 0.05 nm. For this study the grating was set to an output wavelength of 567.85 nm. Figure 3.4 shows the optics setup used on the output side of the laser to form the beam into a sheet. The output from the dye laser was collimated and then frequency doubled to 283.93 nm. Upon exiting the doubling crystal the beam was directed through two Pellin-Broca prisms to disperse the ultraviolet (283.93 nm) light from the 567.85 nm radiation. After exiting the second Pellin-Broca prism the beam passed through a right-angle reflecting prism, which oriented the beam normal to the optics table and normal to a 1 m focal length spherical lens that focused the light at the center of the cylinder. The
Figure 3.4. Optics Setup.

spherical lens caused the beam to converge in all directions. The beam was oriented parallel to the table again by sending it through another right-angle prism. A 100 mm focal length cylindrical lens followed the right-angle prism and was used to ultimately diverge the beam into a sheet. A dielectric mirror situated ~20 cm from the engine was used to reflect the laser sheet into the engine through the quartz windows located in the spacer ring. The end
result was a sheet approximately 200 μm thick and 20 mm wide. Due to the long focal length of the spherical lens the sheet thickness remained approximately constant throughout the area imaged, and the sheet width varied relatively little over the area imaged.

### 3.2.2 Camera Setup

For this experiment two cameras were used to simultaneously image the OH and 3-pentanone fluorescence. Both cameras were intensified. The camera used to image the OH was assembled from parts purchased individually. It consisted of a Pulnix model TM-9701 camera with a video rate front illuminated 768 × 484 pixel interline transfer CCD, coupled to a DEP model XX1450DH intensifier, which had an S-10 photocathode and a P43 phosphor. The camera featured a digital output (EIA-422) that was captured with an EPIX inc. PIXCI D frame grabber.

A Roper-Scientific PI-Max camera was used to capture the fluorescence from the 3-pentanone. The camera featured a full frame 512 × 512 pixel cooled CCD fiber optically coupled to the intensifier. The CCD temperature was maintained at -20°C for all tests even though the dark noise was small when compared with the intensifier noise. This camera came with its own controller and a PCI image acquisition card.

The intensifiers provided the ability to gate the exposure times of the cameras to provide significant rejection of luminosity during combustion. The gate times used to take the data were 150 ns for the 3-pentanone images and 400 ns for the OH images. The duration of the laser pulse was also very short (~8 ns), effectively freezing the motion and giving good temporal resolution. However, due to these short durations the timing of events
relative to one another needed to be very precise and even small amounts of electronic jitter in the timing signals caused problems.

In addition to the issues of jitter in the timing signals, the issue of simultaneously acquiring images from two separate cameras and frame grabbers was challenging. The Pulnix camera required three TTL signals to be triggered, an RS-422 interface was used on the frame grabber thus two signals were required to trigger the CCD of the camera, one of negative polarity and one of positive polarity. The frame grabber looks for the voltage differential between the two signals to change sign to signify a trigger. Another signal was necessary to turn the intensifier on and off since it operated independent of the CCD. In addition to the previously stated difficulties the two cameras had different readout rates and array sizes, and the data acquisition boards they used had different data transfer rates. The final problem was related to the fact that when you acquire images with both cameras on a single computer the instantaneous data transfer rate overwhelms the PCI bus of the computer causing the acquisition of both images to be slowed down. Therefore, a unique timing scheme utilizing two Berkeley Nucleonics model 555 pulse/delay generators was developed to provide the necessary triggers for both cameras and the lasers. Figure 3.5 shows a schematic of the timing connections used. The image acquisition process was initiated by the engine controller sending a pulse to one of the pulse generators 181 \( \mu s \) before the desired picture timing. The pulse generator then output one pulse to the laser to trigger the flash lamps of the laser and one pulse to the PI-Max camera to trigger the programmable timing generator in the camera controller, as soon as the PI-Max camera received the trigger it would output a pulse to the other pulse generator if it were going to
42

Figure 3.5. Schematic of connections for camera timing used.

take a picture that particular cycle. Then the second pulse generator would trigger the intensifier for the Pulnix camera with the correct exposure duration. In addition to sending a trigger to the first pulse generator the engine controller was also used to trigger the CCD of the Pulnix camera. This was possible because the trigger to turn the CCD on and off did not require precise timing since the exposure duration was set by the intensifier exposure duration. The Pulnix CCD exposure was set to ~8 ms.

3.2.3 Camera Field of View, Spatial Resolution

Spatial resolution of the camera systems was an important consideration for this experiment due to the small thickness of the flame-front under the conditions tested. Flame-front thicknesses were expected to be ~100 μm range. Intensified cameras have limited
spatial resolution, and exhibit a gradient response to a step input as illustrated in Fig. 3.6. In order to fully resolve the flame front you need several pixels within the front. However, a tradeoff exists between resolution and field of view - as you increase the resolution by increasing magnification your field of view decreases proportionally.

Another important factor to consider when determining what magnification/resolution to use for an experiment is that intensifiers exhibit a maximum possible resolution due to the channel spacing in the Mirco Channel Plate (MCP), which for the PI-Max camera was 45 lp/mm.

After considering the tradeoffs in regards to this experiment, a 200 mm focal length $f/4$ NIKKOR lens was chosen for imaging the 3-pentanone fluorescence with the PI-Max camera. The lens was set to the maximum aperture to achieve the maximum possible signal. For the OH fluorescence the choice of lens was simple because the only lens capable of transmitting UV light was a 105 mm focal length $f/4.5$ NIKKOR lens.

![Figure 3.6. X-Cross section from an image of a razor blade edge.](image-url)
Figure 3.7. Field of view relative to the piston bowl for the PI-Max camera.

The combination of the 200 mm focal length lens and the PI-Max camera resulted in pixels of 48.4 μm × 48.4 μm size in the focal plane. In order to increase the SNR the pixels were binned 2 × 2 on chip resulting in a 96.8 μm × 96.8 μm size pixel in the focal plane, which is still less than the laser sheet thickness. This resulted in a field of view of 24.7 mm × 24.7 mm in the focal plane, and a magnification of 0.5. The Pulnix camera featured rectangular instead of square pixels, this fact combined with an unequal number of pixels in the horizontal and vertical directions on the array gave pixels of rectangular dimension in the focal plane. The 105 mm lens used with the Pulnix camera gave pixels of
71.0 μm (horizontal) × 83.2 μm (vertical) size. The field of view for the camera was 25.6 mm × 40.3 mm in the focal plane. The field of view for the PI-Max Camera is shown in Fig. 3.7 relative to the bowl in the piston.

The combined system resolution does not solely depend upon focal plane pixel response. There are several measures for establishing the resolution of a system including modulation transfer function (MTF) and Nyquist sampling theorem. The Nyquist theorem states that the maximum frequency that can be reconstructed from the data is ½ the sampling rate. For the PI-Max camera this gives a spatial resolution of 193.6 μm and for the Pulnix camera gives a resolution of 142.0 μm in the horizontal direction and 166.4 μm in the vertical direction.

3.2.4 Image Correction Procedure

Assuming excitation in the linear fluorescence regime where $I \ll I_{\text{sat}}$ the rate at which fluorescence photons strike the CCD, $\dot{S}$ (photons/s) is given by the light scattering equation [30]

$$\dot{S} = I X N V \left( \frac{\sigma_{\text{abs}}}{4\pi} \right) \Omega \phi \eta_{\text{opt}} \left( \frac{1}{h \nu} \right)$$

where $I$ is the laser intensity [J/m²-s], $X$ is the absorbing species mole fraction, $N$ is the number density [m⁻³], $V$ is the collection volume [m³], $\left( \frac{\sigma_{\text{abs}}}{4\pi} \right)$ is the differential absorption cross-section of the molecule [m²], $\Omega$ is collection solid angle [sr], $\phi$ is the fluorescence efficiency, $\eta_{\text{opt}}$ is the collection efficiency of the optics, $h$ is Planck’s constant [J-s], and $\nu$ is the frequency of the fluorescence [s⁻¹].
By integrating over the laser pulse duration and assuming square pixel, which is the case for the PI-Max camera, one can derive the measured fluorescence signal.

\[ S = E_p(x, y) \lambda N(x, y) \frac{\sigma_{abs}(x, y)}{4\pi} \Omega(x, y) \eta \left( \frac{1}{h \nu} \right) \]  

(2)

where \( E_p \) is the laser pulse energy [J], \( \eta \) is the collection and detection efficiency, \( l \) is the pixel dimension [cm].

In equation 2 there are three variables that exhibit thermodynamic dependence: the species number density, the differential scattering cross-section, and the fluorescence efficiency. In an internal combustion engine the pressure can be assumed uniform throughout the cylinder, but in the case of direct in-cylinder fuel injection the temperature field is not homogeneous due to the local temperature depression caused by evaporative cooling for late injection timings [17]. The thermodynamic dependences in (2) are thus illustrated through spatial dependence.

The fluorescence measurement is proportional to the fluorescing species number density. The overall number density is dependent on the local temperature and pressure through the ideal gas law equation \( N = P/kT \), where \( k \) is Boltzmann’s constant. Therefore, the fluorescence measurement can only be used to infer mole fraction by using the correct gas law. Under conditions where the gradients in temperature are small as in the case of early injection, determining mole fraction from the fluorescence measurement induces only small error, but with late injection where there can be large temperature depressions of as much as 200K the error in inferring mole fraction can become significant [12].

For most dopants used in engines one can assume that the differential scattering cross-section and fluorescence efficiency will vary with both temperature and pressure, as is
the case for ketones [12]. For this experiment the pump wavelength was chosen such that
the product of the differential scattering cross-section and the fluorescence efficiency was
relatively constant over the temperature range of interest.

In order to make the fluorescence measurements quantitative one must relate the
measured fluorescence signal to an image of known concentration. This is done by dividing
the background-corrected data images by a background-corrected calibration image, which
will be referred to as a flatfield image, resulting in the ratio of the signals being
approximately equal to the ratio of the fluorescing species number densities

\[
\frac{S_{im}}{S_{fl}} \approx \frac{N_{im}(x,y)}{N_{fl}(x,y)}
\]

(3)

assuming that laser power fluctuations are negligible and that the product of \( \sigma_{abs} \phi \) is
approximately constant, which is the case for the chosen pump wavelength. Equation
(4) relates the equivalence ratio of an air fuel mixture to the number densities of fuel and the
overall in-cylinder number density where the \( a, f, \) and \( t \) indicate air, fuel, and total
respectively.

\[
\Phi = \left( \frac{m_f}{m_a} \right)_{actual} \left( \frac{N_f}{N_t-N_f} \right)_{actual} \left( \frac{m_f}{m_a} \right)_{stoich} \left( \frac{N_f}{N_t-N_f} \right)_{stoich}
\]

(4)

Taking the ratio of the image and flatfield equivalence ratios results in

\[
\frac{\Phi_{im}}{\Phi_{fl}} = \frac{\left( \frac{N_f}{N_t-N_f} \right)_{im}}{\left( \frac{N_f}{N_t-N_f} \right)_{fl}} = \left( \frac{N_f}{N_t-N_f} \right)_{im} \left[ \frac{\left( N_t-N_f \right)_{fl}}{\left( N_t-N_f \right)_{im}} \right]
\]

(5)
where the quantity \[ \left( \frac{N_f - N_{fl}}{N_f - N_{fl}} \right)_{im} \] is approximately equal to 1 for hydrocarbon fuels because the air fuel ratios are typically on the order of 15:1 or greater and the molecular weight of the air and fuel differ by a factor of four approximately, therefore. The error in neglecting this term is at most 2.5% for a range of equivalence ratios from \( 0.1 < \Phi < 2.5 \) with a flatfield image taken at \( \Phi = 1 \) [8]. Therefore, by performing pixel-by-pixel ratios of the background-corrected raw data images with the background-corrected flatfield images the data was calibrated in terms of equivalence ratio.

\[
\frac{S_{im}}{S_{fl}} \approx \frac{N_{im}(x,y)}{N_{fl}(x,y)} \approx \frac{\Phi_{im}}{\Phi_{fl}} \tag{6}
\]

In order to make the corrections to the data several sets of pictures needed to be taken for each data set: background images taken before and after the data image set, flatfield images taken before and after the data image set, and a set of data images; all image sets consisted of 50 images.

The background images were taken at the same picture timing as the data images with the engine motored and the laser on, but with no fuel injection. The background images were later subtracted from the flatfield and data images to correct for any background signal including the dark count of the camera due to dark noise.

The flatfield images were acquired with the laser on, the engine firing, and the air-assist injector supplying a premixed homogeneous mixture of known equivalence ratio to the engine. The images were taken at the same timing as the data images. In order to prevent burned portions of the cylinder contents from showing up in the field of view, the spark timing for the flatfield images was retarded from what was used for the data images. By
firing the flatfield images the in-cylinder thermodynamic conditions were closely matched to those present in the data images, especially the effect of exhaust gas residual. This is important because it is hard to account for the residual effect since the composition and amount of the residual present is hard to determine. The individual data images with the average background subtracted were divided by the average background subtracted flatfield image.

In addition to the images, pressure data were taken for the flatfield and data image sets. The pressure traces for each picture in the data image sets were recorded and used to correct for any difference in pressure between the flatfield and data image which would result in an error in the number densities. The correction was performed by assuming polytropic compression

\[
\alpha = \frac{P_{im}}{P_{fl}} = \left( \frac{N_{fl}}{N_{im}} \right)^\gamma \quad (7)
\]

where \( P_{im} \) is the pressure in the data image at the picture timing, \( P_{fl} \) is the pressure in the flatfield image at the picture timing, and \( \gamma = 1.2 \) is the polytropic exponent determined from experimentally measured log P – log V diagram. Therefore, \( N_{im} = N_{fl} \alpha^{-\gamma} \) and the equivalence ratio determined by the method of equation (6) must be modified such that

\[
\Phi_{imcorr} = \Phi_{im}^{-\gamma} \quad (8)
\]

this correction has been applied to all of the images presented here.

In addition to correcting the 3-pentanone images as described earlier, the OH images had several corrections performed on them. Firstly, the field of view for the OH images needed to be aligned and transformed to that of the pentanone images. The alignment was
corrected by taking images of a fixed target in the focal plane with both cameras. This information was then used to correct the OH images for alignment and magnification by interpolating the pixel intensity values to the $256 \times 256$ spatial domain used for the pentanone images. After performing this pixel mapping the images were corrected for the laser sheet energy profile obtained from the average of the flatfield images taken with the pentanone camera. An average profile correction was possible because the cameras were imaging the same laser sheet.

Figure 3.8 and 3.9 illustrate the correction procedures applied to both the 3-pentanone images as well as the OH images.

### 3.2.5 Image Signal-to-Noise Ratio (SNR)

The signal-to-noise ratio (SNR) of the images directly impacts the ability to make conclusions about the data. For quantitative interpretation of data a SNR >10 is desirable with higher SNR giving more confidence in the validity of any claims made about the data. For the comparison of individual images or changes within a single image it is desirable to have a SNR of 25 or greater. When performing averages or generating statistics from a larger data set lower SNR may be tolerable.

There are several factors that impact the signal to noise ratio of a PLIF imaging system: camera noise, and pulse-to-pulse variation in both laser energy and the laser sheet profile. For the particular conditions of this experiment both the temporal and spatial noise were dominated by the shot noise amplified by the intensifiers in the two cameras used.

The attainable SNR for this experiment was limited by the use of intensified cameras that were necessary to reject the high luminosity intensity during stratified combustion, as well as the available lenses, and the attainable laser power.
Figure 3.8. Image correction process for fuel images (a) average background image (b) average flatfield image (c) raw data image (d) background and flatfield corrected image.
Relatively high amounts of gain were used with the cameras due to their low overall quantum efficiencies. The laser power for the experiment was limited by the amount of energy the dye laser could provide. For this experiment the dye laser was pumped at 200 mJ/pulse and the end result was approximately 12 mJ per pulse output at the excitation wavelength used. However, due to the proximity of the windows to the laser focal plane it is estimated that only moderate increases in laser power can be achieved before the damage threshold of the windows is exceeded.

To improve SNR it would have been possible to correct for the laser pulse-to-pulse variation. The overall SNR ratio for pentanone images was ~12:1 giving a noise level of ~8.3%, if the laser pulse-to-pulse variation in power and profile had been corrected for it would have reduce the noise only to about 7.5%, therefore, these corrections are not

![Figure 3.9](image)

**Figure 3.9.** Images illustrating correction of OH image data (a) original OH data image (b) image transformed to fuel image coordinates and corrected with the average laser sheet profile.
performed because the increase in SNR they provide would be negligible for the conditions of this experiment.
4.0 Experimental Method

This chapter details the method used to attain the results presented in chapter 5. In section 4.1 the details of the algorithms used are covered, section 4.2 shows example results under homogeneous operation, and section 4.3 discusses the fidelity and accuracy of the method.

4.1 Description of Method

The main objective of this research was to measure the flame-front equivalence ratio and see if a correlation exists between it and the measured engine out NO\textsubscript{x} emissions. To attain the flame-front equivalence ratio the flame position needed to be identified in the images, and individual pixel values representing the flame-front equivalence ratio needed to be extracted.

As discussed earlier, PLIF imaging of both 3-pentanone and OH was performed. Simultaneously imaging both of these molecules significantly increased the complexity of the experiment, however, under conditions where significant stratification exists using the 3-pentanone images alone to determine the flame-front location gives false interpretations of the images. Therefore, simultaneous imaging of the OH radical was necessary to track the flame front during stratified combustion. Figure 4.1 shows images of 3-pentanone during stratified combustion illustrating the need to determine flame-front locations through an alternative source.
4.1.1 Fuel Image Correction

The first step in determining the flame front equivalence ratio was to correct the 3-pentanone images. The pentanone images were corrected using the procedure described in
chapter 4. A threshold of 500 counts was applied to the uncorrected images prior to being divided by the flatfield images; any value below this threshold was set to zero. The threshold value was chosen because it corresponded with the approximate level of noise the background subtraction was unable to account for. An additional threshold was used to determine the extent of the image where intensities were sufficient to apply the correction. The laser sheet threshold was applied by determining where the intensity in the flatfield image fell below a value of 3000 counts. This value in general was 25-40% of the maximum values in the flatfield images taken. By thresholding to this level the edges of the corrected images did not become excessively noisy due to the decreasing SNR with decreasing signal.

4.1.2 Alignment of Images

In addition to the corrections discussed earlier, slight adjustments to the relative position of the OH images with respect to the 3-pentanone images were necessary. This was done by overlaying the corresponding OH and 3-pentanone images and then determining the horizontal and vertical offsets necessary to align the OH images to the 3-pentanone images. The original misalignment between the images was most likely due to the cameras being slightly jarred throughout a day of data taking. As the experimental system became more stable the data required little or no additional adjustment for alignment.

4.1.3 Edge Detection of Flame Front

Once the pentanone and OH images were corrected the next step was to determine the flame-front locations in the pentanone images based on the OH images. To do this an edge detection method was applied to the OH images. The algorithm used was developed by
Canny in 1986 and was designed to meet three specifications: detection error – the method should find all edges and give no false detections, location accuracy - the method should locate edges as close to the actual edge location as possible, fidelity - the method should only generate one pixel for a single edge [31]. Canny was able to determine that the first derivative of a Gaussian distribution would serve as an efficient filter to meet the criterion stated above and is given by (1) in one dimension.

\[
G'(x) = \left( -\frac{x}{\sigma^2} \right) e^{-\frac{x^2}{2\sigma^2}}
\]  

(1)

The method was implemented by first convolving the OH images with a 1-d Gaussian mask in the x and y directions. Next the images were convolved with a 1-d derivative of the Gaussian, giving the magnitude of the derivatives in the x and y directions. The non-edge values were then eliminated by what is termed nonmaximum suppression. The basic idea is to find local maximums in the gradient and set the nonmaximum points to zero. An example of the results after nonmaximum suppression is shown in Fig. 4.2. The last step of the edge detection was to threshold the remaining edge locations in order to filter out the edges of low gradient magnitude. The thresholding scheme used creates a histogram of all of the edge gradient pixel magnitudes, from the histogram it determined an upper threshold by finding the value of the 99.4 percentile pixel, this value was optimized through trial and error. Next it determined a lower threshold by taking the sum of the high pixel thresholds and the lowest pixel magnitude and dividing the sum by 2.
Once the thresholds were determined they were applied by searching through the image and setting any pixel above the high threshold to the maximum value of 65535. At each pixel location set to the maximum value the nearest neighbors were checked to see if any were above the low threshold, if they were they were also set to the maximum value. This process continued until no more pixels above the low threshold were found. The thresholding process gave well connected edges and tended to eliminate individual pixels not connected to anything. The results from a sample image after thresholding are shown in Fig. 4.2 (c).

After the Canny edge detection was applied to the OH images, the data still needed some additional filtering because not all of the edges determined from the images represented flame-front edges. Some examples of this are shown in Fig. 4.3, which shows detection of the non-flame front edge pixels originating from deposits on the piston.
Figure 4.3. Other sources of detected edge pixels (a) window stains (b) gradients in the OH concentration.
window, and from significant gradients occurring within the OH image due to changes in OH concentration throughout the image.

In order to filter out any non-flame-front pixels still present in the images several criteria were developed. Due to system noise equivalence ratios below $\phi = 0.1$ could not be considered accurate. Because the edges were located at the position of maximum gradient the equivalence ratio at the flame-front detected edge location should be above the system noise response limit of $\phi = 0.1$. Therefore, the first criterion developed, was that the equivalence ratio at the flame-front detected pixel location was greater than $\phi = 0.1$. Two criterion were developed based on the information from the OH images; the first was based on the normalized standard deviation in intensity in a small sub region surrounding the pixel of interest in the OH image, and the second was based on the average of that sub region in the OH image. A threshold was then used to filter out non-flame-front pixels. In the case of the normalized standard deviation the value chosen was 0.3 and in the case of the average the value chosen was $30 < \text{average} < 250$. The values were determined through trial and error and gave the greatest filtering of bad pixels while maintaining a significant sample size, in general $>3000$ pixels per data set. The standard deviation criterion was adopted because if a pixel was situated on the flame-front it would have a high-normalized standard deviation, therefore any pixels below a certain level did not sit on the flame-front. Both an upper and lower threshold were used for the sub region average intensity because if the pixel had a value below the lower threshold it was most likely in the middle of the fuel cloud and thus not on the flame front, and if the pixel had a value higher than the upper threshold it was most likely in the middle of the burned region and thus not on the flame-front. Figure 4.4 illustrates the effect of each filtering criterion on a sample data image,
Figure 4.4. Effect of various filtering criteria on the flame-front edges detected, shown with a 2-pixel offset (a) the unfiltered image (b) $\Phi > 0.1$ threshold applied to fuel image (c) $30 < \text{average} < 250$ threshold applied in OH image (d) standard deviation $> 0.3$ threshold applied in OH image.
where the white pixels superimposed on the fuel image represent the detected edge location after filtering.

### 4.2 Sample Results for Homogeneous Operation

Once the flame-front locations were determined from the edge detection method that information was transferred back to the fuel concentration images to sample the equivalence ratio at the flame front. As discussed above CCDs exhibit a gradient response to a step edge input, and the edge detection method locates the position of the maximum gradient. The fuel concentration value located at the position of maximum gradient will significantly underestimate the flame-front equivalence ratio, therefore, a different approach was taken. Looking at the graph of camera response for a step input, Fig. 3.7, one sees that if you move two pixels from the point of maximum gradient in the direction of increasing intensity the response curve has reached a value of ~90% of the actual value. Therefore, the approach taken to estimate the flame-front equivalence ratio was to assume that the value two pixels behind the detected edge in the direction of increasing gradient represented the flame-front equivalence ratio. There are several reasons for choosing 2 pixels as the offset. The 2 pixel offset results in a predictable underestimation of the mean equivalence ratio of ~10%. Most features in the images will have widths greater than 2 pixels for the operating conditions tested, and the imaging system resolution is approximately 2 pixels. By moving farther away from the edge there is a higher probability of inaccurately representing what occurs at the flame-front. Figure 4.5 shows two examples of the edges, after filtering, superimposed on the fuel concentration images with the 2-pixel offset applied to the edges.
4.2.1 Equivalence Ratio Histograms Under Homogeneous Operation

After the flame-front equivalence ratio was extracted from the images histograms of the equivalence ratio along the flame-front were generated. Figure 4.6 shows histograms for the homogeneous operating conditions at 600 and 1200 rpm. The 600 rpm case had a spark timing of 20° BTDC and the 1200 rpm case had a spark timing of 30° BTDC. The overall equivalence ratios were $\Phi = 0.89$ and $\Phi = 0.81$ at 600 and 1200 rpm respectively. The histograms shown are of the individual pixel equivalence ratio values with the 2-pixel offset applied. The mean value calculated from the distributions were 0.85 and 0.71, and they correspond closely with the overall equivalence ratios of $\Phi = 0.89$ and $\Phi = 0.81$. As stated
before the technique slightly under estimates the means by ~10% or less and these histograms show a slight underestimation as well.

The histogram of pixel equivalence ratio represents the number of pixel occurrences at a particular equivalence. However, the mass of fuel at the flame front is of greater interest so a mass weighting was applied. Figure 4.7 shows the results of applying a mass weighting to the pixel histograms of Figure 4.6. For the homogeneous cases this change only slightly affects the distribution by suppressing values below the mean and slightly favoring values greater than the mean.

### 4.2.2 Equivalence Ratio PDFs Under Homogeneous Operation

For each data set a different number of flame-front locations were detected. So in order to compare between distributions at different operating conditions a probability density function (PDF) $pdf(\Phi)$ where

$$P(\Phi) = \int_0^\infty pdf(\Phi) d\Phi = 1$$  \hspace{1cm} (2)

is better suited than a histogram. Figure 4.8 shows the mass weighted PDFs of equivalence ratio for the two homogeneous cases. The distributions at the two engine speeds have similar widths indicating similar relative error.

The homogeneous cases had a large number of pixels detected along the flame front (>10000) due to the ideal conditions. However, the stratified data often had a significantly lower number of pixels detected, usually 2000 – 7000. This brings up the question of whether the statistics are converged. Figure 4.9 shows the flame-front equivalence ratio PDF for the $\Phi = 0.32$ base condition at 600 rpm, along with the PDFs based on using only
Figure 4.6. Histograms of pixel equivalence ratio at (a) 600 rpm and a $\Phi = 0.89$ and (b) 1200 rpm and a $\Phi = 0.81$
Figure 4.7. Mass weighted histograms of pixel equivalence ratio at (a) 600 rpm and a $\Phi = 0.89$ and (b) 1200 rpm and a $\Phi = 0.81$. 
Figure 4.8. Effect of smoothing on PDFs of equivalence ratio along the flame front for (a) 600 rpm and a $\Phi = 0.89$ and (b) 1200 rpm and a $\Phi = 0.81$. 
Figure 4.9. Flame-front equivalence ration PDFs for $\Phi = 0.32$ and 600 rpm illustrating the convergence of the PDF.

respectively. From the PDFs it is apparent that the statistics convergence quickly. Even with only 916 the distribution is very similar to the other distributions, and it appears that the PDF is approximately converged when using only 25% (1832) of the pixels. Therefore, that all of the PDFs that are presented in chapter 5 are converged.

### 4.3 Accuracy and Fidelity of the Method

The noise sources associated with the PLIF imaging system were discussed earlier in Chapter 3. These sources affect the overall fidelity of the data presented in the PDF. In addition to the noise sources associated with the PLIF measurements, additional uncertainty arises from the method used to extract the flame-front equivalence ratio data from the fuel concentration images.
In order to reduce the variance of the distributions, Gaussian smoothing was applied to the fuel concentration images using a two-dimensional Gaussian mask with a standard deviation of 1 pixel. The results of smoothing the images are shown in Fig. 4.8 with the PDFs for homogeneous operation at 600 and 1200 rpm before and after smoothing.

A breakdown of the noise sources and magnitudes present in the PDFs of equivalence ratio for the homogeneous cases is shown in Table 4.1. Where $\sigma_{pdf}$ is defined as

$$\sigma_{pdf} = \sqrt{\int_0^{2\pi} (\phi - \mu)^2 pdf(\phi)d\phi} \quad (3)$$

where $\mu$ is the mean value of the PDF. The spatial noise was calculated from the homogeneous data images as

$$\sigma_s = \frac{1}{N_p} \sum_{p=1}^{N_p} \left[ \frac{1}{N} \sum_{x=x_1}^{N-1} \sum_{y=y_1}^{N-1} \left( S_{x,y} - \bar{S} \right)^2 \right]^{1/2} \quad (4)$$

where $S_{x,y}$ is the signal at an individual pixel location, $\bar{S}$ is the average pixel signal for the

<table>
<thead>
<tr>
<th>Noise Source</th>
<th>600 rpm (%)</th>
<th>1200 rpm (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PDF ($\sigma_{pdf}/\mu$)</td>
<td>14.0</td>
<td>19.7</td>
</tr>
<tr>
<td>Spatial ($\sigma_s/\mu$)</td>
<td>9.45</td>
<td>7.72</td>
</tr>
<tr>
<td>Phase ($\sigma_p/\mu$)</td>
<td>10.4</td>
<td>9.01</td>
</tr>
<tr>
<td>Laser Profile Correction ($\sigma_p/\mu$)</td>
<td>4.46</td>
<td>4.53</td>
</tr>
<tr>
<td>Laser Shot-to-Shot ($\sigma_{ss}/\mu$)</td>
<td>4.22</td>
<td>4.65</td>
</tr>
</tbody>
</table>
sub-region, $N$ is the number of pixels in the sub-region, and $N_p$ is the number of images acquired. The phase noise was calculated from the images as

$$
\sigma_p = \frac{1}{N} \sum_{x=x_1}^{x_2} \sum_{y=y_1}^{y_2} \left[ \frac{1}{N_p} \sum_{p=1}^{N_p} \left( S_{x,y} - \bar{S}_{x,y} \right)^2 \right]^{1/2}
$$

(5)

where $\bar{S}_{x,y}$ was the average signal at an individual pixel. Assuming that the errors add in quadrature. The error associated with correcting the images with an average flatfield image was calculated as

$$
\sigma_{lp} = \sqrt{\sigma_s^2 - \sigma_c^2}
$$

(6)

where $\sigma_c$ is the noise associated with the camera system. The error associated with not correcting for laser profile and power variations on a shot to shot basis was calculated as

$$
\sigma_{ss} = \sqrt{\sigma_p^2 - \sigma_s^2}
$$

(7)

where $\sigma_{ss}$ is the shot-to-shot laser power variation.

The means of the distributions pictured in Fig. 4.8 were $\Phi = 0.71$ and $\Phi = 0.85$ for 600 and 1200 rpm respectively. Therefore, the method consistently under-predicts the mean value. This should be considered when interpreting the results of Chapter 5.
5 Flame-Front Equivalence Ratio During Stratified Combustion

5.1 Introduction

This chapter presents results generated using the flame-front detection method described in Chapter 4. The results include PDFs of the equivalence ratio detected along the flame front as well as representative 3-pentanone and OH images. The data are presented in the following order; effect of engine speed between 600 and 1200 rpm, effect of overall equivalence ratio, effect of spark and injection timing for equivalence ratio of $\Phi = 0.32$ and $\Phi = 0.42$ at the engine speeds of 600 rpm and 1200 rpm. Additionally the results of the emissions testing will be presented. The emissions data were taken for the cases at 600 rpm for the conditions of $\Phi = 0.32$ and $\Phi = 0.42$.

5.2 Factors Affecting the Interpretation of the PDF and Image Data

Due to limitations in experimental setup and time, several factors must be considered when making conclusions about the data presented in this chapter.

5.2.1 Field of View

The fixed field of view imposes the largest limitation on the data. This results from the available lens selection and the need for relatively high magnification, which decreases the field of view. When viewing the data it is important to note that it was sampled from a
single plane in the cylinder and represented an area approximately 15 mm wide by 24 mm long.

The choice of field view was influenced by the piston position at the image timing, which was governed by the flame travel and the spark plug location. Image times were in general within $\pm 10$ CAD of TDC. The axial position of the sheet was chosen such that it passed over the piston top at TDC. Flare from the laser sheet hitting the extended-reach spark plug is difficult to subtract accurately, thus the spark plug was outside of the camera view. Also, during stratified combustion the fuel only occupies a relatively small area of the combustion chamber. When interpreting the data it is necessary to account for the effects of injection timing, picture timing, and spark timing on the position of the bulk stratified fuel cloud relative to the field of view.

5.2.2 Data Set

The time constraints related with running the engine necessitated acquiring a smaller number of picture timings and fields of view than would be considered optimal. Ideally the data set would include images at various picture timings after ignition for each operating condition, as well as repetition of the data set with several fields of view. The running conditions, for the data presented in this chapter, were optimized as discussed in Chapter 3. Chapter 3 also provides a discussion of the selection criterion and baseline engine timings.

Conclusions are drawn from image data presented throughout this chapter, and it should be noted that when such conclusions are drawn they are based upon having studied the entire data set of images and comparing entire data sets to one another; they are not based solely upon the single images shown.
The limitations of the data have been set forth so that they can be used in developing conclusions about the data. The data taken were chosen to maximize the available information while minimizing the size of the data set.

### 5.3 **Effect of Engine Speed**

Data for engine speeds of 600 and 1200 rpm were taken from equivalence ratios of $\Phi = 0.69$ down to $\Phi = 0.32$. The effect of engine speed on the fuel distribution determines the speed and load range at which the engine can operate with a stratified charge. Excessive mixing of the air and fuel at lean equivalence ratios can cause misfires and increased emissions.

#### 5.3.1 **Effect of Engine Speed, $\Phi = 0.69$**

An equivalence ratio of $\Phi = 0.69$ with an EOI of 180° BTDC resulted in a mixture approaching a homogeneous state. An equivalence ratio of $\Phi = 0.69$ was on the borderline for stable combustion under homogeneous operation. The spark timing for the 600 rpm condition was 20° BTDC and the timing for the 1200 rpm case was set to 30° BTDC. Figure 5.1 shows the mass weighted PDFs of equivalence ratio for the two speeds.

The PDF for the 600 rpm case resembles the homogeneous conditions shown in chapter 4. The main distinction between this curve and the homogenous curves is a widening of the PDF between $\Phi = 0.4$ and $\Phi = 0.6$ indicating the presence of stratification in the mixture. Figure 5.2 shows images of the 3-pentanone and OH illustrating the appearance of slight stratification for the two engine speeds. In Figure 5.2 the presence of areas both richer and leaner than the overall equivalence ratio were observed.
**Figure 5.1.** PDFs of equivalence ratio for engine speeds of 600 and 1200 rpm at $\Phi = 0.69$.

The PDF for 1200 rpm indicates the presence of significant inhomogeneity in the mixture. The PDF shows large portions of the flame-front fuel mass detected at lean equivalence ratios ($\Phi < 0.4$) and a small amount of the fuel at equivalence ratios greater than stoichiometric.

The amount of time between the EOI and picture timing for the two engine speeds is the same in crank angles ($180^\circ$), however, in real time the mixing time is reduce by a factor of two between the 600 rpm and 1200 rpm cases. From the PDFs it is apparent that the increased turbulence due to the increase in engine speed does not offset the decrease in mixing time available at the 1200 rpm condition. This results in a mixture with greater variation in equivalence ratio. The sample images of Fig. 5.2 verify this. The images also
Figure 5.2. Sample images of 3-pentanone (left) and OH (right) at (a) 600 rpm and (b) 1200 rpm for $\Phi = 0.69$
show a scaling of the flame-front wrinkling with engine speed as is expected for a relatively homogeneous condition.

The net result of the decreased mixing is the degradation of combustion stability. The 600 rpm condition ran poorly, as expected due to the lack of optimization for this condition as discussed in Chapter 3, with a coefficient of variation (COV) of indicated mean effective pressure (IMEP) of 23% and an IMEP of 4.1 bar. The 1200 rpm condition ran significantly worse with a COV of IMEP of 70%, causing a significant reduction in IMEP, down to 2.8 bar. These results verify the usefulness of the PDF information showing that the combustion quality correlates well the observations obtained from the PDFs for conditions approaching homogeneous.

5.3.2 Effect of Engine Speed, Φ = 0.61

The change from Φ = 0.69 to Φ = 0.61 resulted in a shift from a relatively homogeneous mixture to a highly stratified mixture. The EOI for the 600 rpm and 1200 rpm cases with Φ = 0.61 were 57.5° BTDC and 60° BTDC respectively. The ignition timing was 20° BTDC for both cases. The jump in injection timing from 180° BTDC to ~ 60° BTDC was necessary because an approximately homogeneous mixture would not ignite at a Φ = 0.61. Therefore, it was necessary to run the engine with a stratified air-fuel charge.

Figure 5.3 shows the flame-front equivalence ratio PDFs for both engine speeds. The PDFs observed in Fig. 5.3 span a wide range of equivalence ratios, from Φ = 0.1 to Φ = 2.2. The trace for 600 rpm appears to be composed of 3 overlapping distributions with means of Φ ~ 0.25, Φ ~ 0.8, and Φ ~ 1.3. The distribution with a mean of ~ 0.8 represents the majority of the fuel mass burning along the flame-front. Assuming an under prediction of ~ 10% in
the mean equivalence ratio this indicates the peak occurrence of flame-front fuel mass is located at a $\Phi = 0.9$. The lean distribution with a mean of $\sim 0.25$ corresponds with fuel that may have trouble oxidizing during the combustion process if local temperatures are not sufficient [7].

A significant mass of fuel with an equivalence ratio greater than $\Phi = 1.2$, and as high as $\Phi = 2.2$ was detected. The rich fuel pockets are likely to contribute to soot formation. The stratified cases with equivalence ratios of $\Phi = 0.61$ and $\Phi = 0.52$ did form significant amounts of soot and spark plug fouling became evident after $\sim 60$ s of operation.

Unlike the 600 rpm condition, at 1200 rpm the flame-front equivalence ratio PDF appears to display one continuous distribution with a mean near $\Phi \sim 0.8$. The 1200 rpm case appears to be less stratified than the 600 rpm condition, which is apparent from the images of Fig. 5.4. This is contrary to what was seen at the $\Phi = 0.69$ conditions where the

![Figure 5.3. PDFs of equivalence ratio for engine speeds of 600 and 1200 rpm at a $\Phi = 0.61$.](image-url)
Figure 5.4. Sample images of 3-pentanone (left) and OH (right) at (a) 600 rpm and (b) 1200 rpm for $\Phi = 0.61$. 
1200 rpm data appeared to be more stratified than the 600 rpm data. Both cases had approximately the same number of crankangles between EOI and the picture timing corresponding to a factor of 2 difference in real time for mixing to occur.

The piston-spray interaction at $\Phi = 0.61$ is much different than at $\Phi = 0.69$. The EOI for the $\Phi = 0.69$ case is $180^\circ$ BTDC, and the piston is located as far away from the injector as possible. In the $\Phi = 0.61$ case the piston is located relatively close to the injector and a significant amount of spray impacts the piston. Due to the lack of piston interaction in the early injection case the mixing is likely dominated by the bulk in-cylinder flow with the secondary flow present close to TDC most likely having less influence on mixing. It is possible that the increase in turbulence scaling with engine speed is insufficient to make up for the difference in available mixing time resulting in the apparent greater unmixedness observed for the case at a $\Phi = 0.69$ and 1200 rpm.

In the late injection case the bulk of the fuel is located around the reentrant bowl of the piston, and the bulk flow most likely has less influence on the mixing in the short time available. The interaction of the spray with the piston surface and the squish flow motion may be more significant contributors to the mixing process. These factors may explain the data indicating that the 1200 rpm case was more mixed than the 600 rpm case, assuming that the fuel cloud position was similar for both engine speeds.

### 5.3.3 Effect of Engine Speed, $\Phi = 0.52$

The $\Phi = 0.52$ case had similar injection and ignition timings to those used for $\Phi = 0.61$. The EOI timings used were $55^\circ$ BTDC and $57.5^\circ$ BTDC for the 600 and 1200 rpm
conditions with spark timings of 15° BTDC and 20° BTDC. Figure 5.5 shows the flame-front equivalence ratio PDFs for the 2 engine speeds.

The PDF at 600 rpm is very wide ranging from $\Phi = 0.1$ to $\Phi = 1.9$ and exhibiting a peak at $\Phi \approx 0.9$. The stratification of the fuel is evident from the PDF through the wide distribution. The peak and mean of the distribution lie around $\Phi \approx 0.9$, coinciding with the equivalence ratio of maximum NOx production. About forty percent of the mixture represented in the PDF is fuel rich indicating that significant soot production is likely under this condition.

At 1200 rpm the PDF is shifted to leaner equivalence ratios with respect to the 600 rpm case. The distribution also encompasses a smaller range of equivalence ratios than at 600 rpm. This indicates that at 1200 rpm the stratified fuel cloud had mixed to a greater extent than at 600 rpm. Only 20 percent of the flame-front fuel mass is located above $\Phi = 1.0$. 

![Figure 5.5. PDFs of equivalence ratio for engine speeds of 600 and 1200 rpm at a $\Phi = 0.52$.]
Figure 5.6. Sample images of 3-pentanone (left) and OH (right) at (a) 600 rpm and (b) 1200 rpm for $\Phi = 0.52$. 
1.0 with very little fuel present at $\Phi > 1.4$ indicating a possible reduction in the soot formation due to the increased mixing. A large amount of fuel is present at lean equivalence ratios with about 15 percent located at a $\Phi < 0.3$. The fuel at lean equivalence ratios may contribute to unburned hydrocarbons.

Figure 5.6 shows images taken at the two engine speeds with $\Phi = 0.52$. The images reinforce the conclusions drawn from the PDFs. The 1200 rpm images appear to have lower equivalence ratios at the flame front and throughout the image. The changes in fuel distribution due to changing engine speed affected the operation of the engine only slightly. The COV of IMEP was 4.3 % with an IMEP of 3.3 bar for the 600 rpm case and 6.7 % with an IMEP of 2.9 bar for the 1200 rpm case.

**5.3.4 Effect of Engine Speed, $\Phi = 0.42$**

The ignition and end of injection timings used at 600 rpm and $\Phi = 0.42$ were 20 and $52.5^\circ$ BTDC respectively. At 1200 rpm, the spark was $20^\circ$ BTDC and the EOI was $55^\circ$ BTDC. Figure 5.7 shows the flame-front equivalence ratio determined PDFs at each of the engine speeds.

The 600 rpm PDF illustrated in Fig. 5.7 depicts a negatively skewed fuel distribution. The distribution is extremely broad with measurable fuel concentrations as high as $\Phi = 2.5$. However, the majority of the fuel is at equivalence ratios less than $\Phi = 0.8$, and a relatively small amount of fuel exists in the range of equivalence ratios between $\Phi = 0.8$ and $\Phi = 1.1$ generally associated with high rates of NOx production. It is interesting to note
Figure 5.7. PDFs of equivalence ratio for engine speeds of 600 and 1200 rpm at a $\Phi = 0.42$.

that the peak of the distribution corresponds almost exactly with the overall equivalence ratio, $\Phi = 0.42$.

At 1200 rpm the PDF shown in Figure 5.7 indicates a leaner burning mixture. The images of the fuel during combustion shown in Figure 5.8 support this conclusion. The 1200 rpm case also shows a significant amount of fuel below an equivalence ratio of $\Phi = 0.5$. More than half of the fuel mass detected at the flame-front in the images was at equivalence ratios less than 0.5.

The leaner burning zone equivalence ratios present at 1200 rpm may have been the cause for the increased instability in combustion. The COVs of IMEP for the 600 and 1200 rpm cases were 3.6 and 8.9 % respectively. The IMEP was significantly reduced at the 1200 rpm condition with an IMEP of 1.9 bar compared to the 2.5 bar measured at 600 rpm.
Figure 5.8. Sample images of 3-pentanone (left) and OH (right) at (a) 600 rpm and (b) 1200 rpm for $\Phi = 0.42$. 
5.3.5 Effect of Engine Speed, $\Phi = 0.32$

The ignition and injection timings for the two engine speeds at $\Phi = 0.32$ were, 17.5 and 50° BTDC at 600 rpm, and 17.5 and 55° BTDC at 1200 rpm respectively. The PDFs of the flame-front equivalence ratio measured for these cases are shown in Fig. 5.9.

The flame-front equivalence ratio distributions shown in Fig. 5.9 indicate that a relatively lean equivalence ratio exists for the majority of the fuel along the flame front observed at both engine speeds. At 600 rpm the distribution is skewed with a peak probability occurring around $\Phi \approx 0.4$. Little fuel is burning at equivalence ratios $> 0.8$ for both of the engine speeds. Based on the Zeldovich mechanism of NOx production this would suggest very low NOx production. Further discussion about this will be given in § 5.7.

The distribution at 1200 rpm is shifted to even leaner equivalence ratios relative to

![Figure 5.9. PDFs of equivalence ratio for engine speeds of 600 and 1200 rpm at a $\Phi = 0.32$.](image)
Figure 5.10. Sample images of 3-pentanone (left) and OH (right) at (a) 600 rpm and (b) 1200 rpm for $\Phi = 0.32$. 
the 600 rpm data. The extent of fuel present at lean equivalence ratios indicates that a significant amount of fuel may have mixed beyond the lean flammability limit. This is supported by the pressure measurements where several misfires were observed along with many cycles that appear to have only partially burned. The COV of IMEP at 1200 rpm was 34 % with an IMEP of 1.0 bar. At 600 rpm the engine ran considerably better with a COV of IMEP of 8.6 % and an IMEP of 1.6 bar. No misfires were present for the 600 rpm data.

Figure 5.10 shows images of the fuel and 3-pentanone for the two engine speeds. The images support the observation obtained from the PDFs of an apparently leaner equivalence ratio along the flame front. In general the flame-front propagation was approximately the same from the two engine speeds.

5.4 Effect of Engine Load (Overall Equivalence Ratio)

The following recast the data of § 5.3 to elucidate the effect of the overall equivalence ratio directly. Additionally Tables 5.1 and 5.2 are presented, which contain the summarized pressure data for all of the image data presented in this chapter.

5.4.1 Effect of Engine Load, 600 rpm

The flame-front equivalence ratio PDFs for the 5 equivalence ratios $\Phi = 0.69, 0.61, 0.52, 0.42, \text{ and } 0.32$, are displayed in Fig. 5.11. The running conditions for the various equivalence ratios were detailed in § 5.3.

The PDFs of Fig. 5.11 naturally fall into three categories. The first category includes only the $\Phi = 0.69$ condition, which represents a mixture approaching a homogeneous fuel distribution. Only slight stratification exists in this case as indicated by the images of
<table>
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<th>Running Condition</th>
<th>IMEP (bar)</th>
<th>COV of IMEP (%)</th>
<th>Peak Pressure (bar)</th>
<th>Location of Peak Pressure (ATDC)</th>
</tr>
</thead>
<tbody>
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<td><strong>Varying Equivalence Ratio</strong></td>
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<tr>
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<td>ret 5</td>
<td>1.32</td>
<td>13.6</td>
<td>18.26</td>
</tr>
<tr>
<td><strong>Varying Spark</strong></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Base Spark = 20 BTDC EOI = 52.5 BTDC</td>
<td>adv 10</td>
<td>2.31</td>
<td>42.5</td>
<td>20.42</td>
</tr>
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<td></td>
<td>adv 5</td>
<td>2.56</td>
<td>3.7</td>
<td>23.91</td>
</tr>
<tr>
<td></td>
<td>base</td>
<td>2.50</td>
<td>3.6</td>
<td>26.23</td>
</tr>
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<td></td>
<td>ret 5</td>
<td>2.36</td>
<td>3.5</td>
<td>25.59</td>
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<tr>
<td></td>
<td>ret 10</td>
<td>2.25</td>
<td>3.3</td>
<td>24.78</td>
</tr>
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<td><strong>Varying Spark</strong></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Base Spark = 20 BTDC EOI = 52.5 BTDC</td>
<td>adv 2.5</td>
<td>2.19</td>
<td>3.7</td>
<td>26.66</td>
</tr>
<tr>
<td></td>
<td>adv 2.5</td>
<td>2.29</td>
<td>3.7</td>
<td>26.05</td>
</tr>
<tr>
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<td>base</td>
<td>2.50</td>
<td>3.6</td>
<td>26.23</td>
</tr>
<tr>
<td></td>
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<td>23.86</td>
</tr>
<tr>
<td></td>
<td>ret 5</td>
<td>2.49</td>
<td>5.5</td>
<td>21.42</td>
</tr>
</tbody>
</table>

*Table 5.1. Summary of pressure data for 600 rpm image data*
Fig. 5.11. PDFs for varying equivalence ratio from $\Phi = 0.69$ to $\Phi = 0.32$ at 600 rpm.

Fig. 5.2 and the strong peak in the distribution very close to the overall in-cylinder equivalence ratio. The width of the distribution was greater than that obtained for a homogeneous condition (see Fig. 4.7.), indicating some stratification.

The second group of curves includes the $\Phi = 0.61$ and $\Phi = 0.52$ operating conditions. Under these conditions the images indicated a very wide range of equivalence ratios along the flame front and a significant portion of the fuel burning at rich conditions. These operating conditions used a late injection timing, $\sim 60$ BTDC, and produced enough soot to cause severe spark plug fouling. Therefore, curves in this category represent mixtures where significant air under-utilization occurs. The conditions in this group would benefit form augmented mixing of the air-fuel mixture prior to ignition to better utilize the
available air. This intermediate load operating range has traditionally been the most difficult for stratified-charge engine development.

The third category of PDFs consists of the $\Phi = 0.42$ and $\Phi = 0.32$ cases. These cases show a definite shift in flame-front equivalence ratio towards lower equivalence ratios from the second group of curves. The $\Phi = 0.42$ condition in particular shows a significant resemblance to the $\Phi = 0.61$ and $\Phi = 0.52$ conditions. The change in available mixing time between the group two conditions and group three conditions is small, ~ 5 CAD demonstrating the effect of decreasing equivalence ratio on the mixing of the charge. In the $\Phi = 0.32$ condition the mixture is approaching an over-mixed condition where combustion quality has begun to degrade and significant unburned fuel may exist. Pressure data taken at a condition of $\Phi = 0.27$ showed a significant increase in COV of IMEP to 24% as compared to the 8.6% obtained for the $\Phi = 0.32$ condition. The pressure data at $\Phi = 0.27$ supports the conclusion of over-mixing of the fuel as the equivalence ratio is decreased from $\Phi = 0.32$ and, therefore, at this engine speed $\Phi = 0.32$ represents the lower equivalence ratio operating limit.

5.4.2 Effect of Engine Load, 1200 rpm

The flame-front equivalence ratio PDFs for $\Phi = 0.69$, $0.61$, $0.52$, $0.42$, and $0.32$ at 1200 rpm are shown in Fig. 5.12. As was observed for the 600 rpm data the PDFs naturally segregate into three categories. The PDF for $\Phi = 0.69$ is the lone curve in the first group, while the second group consists of the curves for the $\Phi = 0.61$ and $\Phi = 0.52$ conditions, and the third group consists of the $\Phi = 0.42$ and $\Phi = 0.32$ cases.
Table 5.2. Summary of pressure data for 1200 rpm image data

<table>
<thead>
<tr>
<th>Running Condition</th>
<th>IMEP (bar)</th>
<th>COV of IMEP (%)</th>
<th>Peak Pressure (bar)</th>
<th>Location of Peak Pressure (ATDC)</th>
</tr>
</thead>
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<tr>
<td><strong>Varying Equivalence Ratio</strong></td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Equivalence Ratio = 0.32</td>
<td>0.81</td>
<td>6.02</td>
<td>3.9</td>
<td>26.51</td>
</tr>
<tr>
<td></td>
<td>0.69</td>
<td>2.80</td>
<td>70.3</td>
<td>20.26</td>
</tr>
<tr>
<td></td>
<td>0.61</td>
<td>2.23</td>
<td>79.3</td>
<td>21.37</td>
</tr>
<tr>
<td></td>
<td>0.52</td>
<td>2.94</td>
<td>6.7</td>
<td>23.54</td>
</tr>
<tr>
<td></td>
<td>0.42</td>
<td>1.94</td>
<td>8.9</td>
<td>22.35</td>
</tr>
<tr>
<td></td>
<td>0.32</td>
<td>1.03</td>
<td>34.4</td>
<td>19.49</td>
</tr>
<tr>
<td><strong>Varying injection</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Spark = 17.5 BTDC</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Base EOI = 55 BTDC</strong></td>
<td>adv 10</td>
<td>0.68</td>
<td>71.1</td>
<td>18.22</td>
</tr>
<tr>
<td></td>
<td>adv 5</td>
<td>0.92</td>
<td>27.5</td>
<td>19.48</td>
</tr>
<tr>
<td></td>
<td>base</td>
<td>1.03</td>
<td>34.4</td>
<td>19.49</td>
</tr>
<tr>
<td></td>
<td>ret 5</td>
<td>0.85</td>
<td>56.5</td>
<td>18.43</td>
</tr>
<tr>
<td></td>
<td>ret 10</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td><strong>Spark = 20 BTDC</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Base EOI = 55 BTDC</strong></td>
<td>adv 10</td>
<td>1.80</td>
<td>43.1</td>
<td>19.84</td>
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<td>adv 5</td>
<td>1.80</td>
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<td>base</td>
<td>1.94</td>
<td>8.9</td>
<td>22.35</td>
</tr>
<tr>
<td></td>
<td>ret 5</td>
<td>1.76</td>
<td>10.8</td>
<td>21.17</td>
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<tr>
<td></td>
<td>ret 10</td>
<td>1.49</td>
<td>39.8</td>
<td>19.49</td>
</tr>
<tr>
<td><strong>Varying Spark</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Base Spark = 17.5 BTDC</strong></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td><strong>EOI = 55 BTDC</strong></td>
<td>adv 5</td>
<td>0.92</td>
<td>53.9</td>
<td>20.11</td>
</tr>
<tr>
<td></td>
<td>adv 2.5</td>
<td>0.95</td>
<td>36.9</td>
<td>19.58</td>
</tr>
<tr>
<td></td>
<td>base</td>
<td>1.03</td>
<td>34.4</td>
<td>19.49</td>
</tr>
<tr>
<td></td>
<td>ret 5</td>
<td>1.86</td>
<td>10.8</td>
<td>56.5</td>
</tr>
<tr>
<td></td>
<td>ret 5</td>
<td>0.83</td>
<td>17.3</td>
<td>23.73</td>
</tr>
<tr>
<td></td>
<td>adv 10</td>
<td>1.49</td>
<td>39.8</td>
<td>19.49</td>
</tr>
<tr>
<td></td>
<td>adv 2.5</td>
<td>0.91</td>
<td>11.1</td>
<td>23.00</td>
</tr>
<tr>
<td></td>
<td>base</td>
<td>1.94</td>
<td>8.9</td>
<td>22.35</td>
</tr>
<tr>
<td></td>
<td>ret 5</td>
<td>1.76</td>
<td>12.5</td>
<td>20.23</td>
</tr>
</tbody>
</table>
The first category is representative of a mixture approaching a state of homogeneity. The flame-front equivalence ratio PDF, however, shows significant stratification relative to the homogeneous mixture PDFs of chapter 4 (see Fig.4.7).

As the injection timing retards, EOI ~ 60° BTDC, the stratification in the mixture increases despite the decrease in overall equivalence ratio for the $\Phi = 0.61$ and $\Phi = 0.52$ conditions. When compared with the 600 rpm data at the same equivalence ratios one notices a decrease in the spread of the PDFs along with an increase in the amount of fuel present at low equivalence ratios. This is indicative of increased mixing due to the increased turbulence at 1200 rpm as discussed in § 5.3.2.

![PDFs for varying equivalence ratios from $\Phi = 0.69$ to $\Phi = 0.32$ at 1200 rpm.](image)

**Figure 5.12.** PDFs for varying equivalence ratios from $\Phi = 0.69$ to $\Phi = 0.32$ at 1200 rpm.
The reduction in equivalence ratio below $\Phi = 0.52$ seems to result in overly-mixed distributions. The pressure data showed a marked increase in COV of IMEP from 8.9% at a $\Phi = 0.42$ to 34.3% at a $\Phi = 0.32$, supporting this conclusion.

The effect of increasing the engine speed from 600 to 1200 rpm appears to be increased mixing of the fuel. At low equivalence ratios this can result in poor combustion stability due to an overly lean mixture. Therefore, the lower limit of stable combustion for a stratified mixture is a function of the engine speed for this combustion system. Similar results have been observed for other stratified-charge systems [2].

### 5.5 Effect of Injection Timing

In this section the effect of varying injection timing on the flame-front equivalence ratio is discussed. Data are presented for two equivalence ratios, $\Phi = 0.32$ and $\Phi = 0.42$, for the two different engine speeds tested, 600 rpm and 1200 rpm. Five different injection timings were tested at each equivalence ratio and speed combination. The data are presented with the notation that the baseline condition corresponds approximately to the optimal operating condition with respect to COV of IMEP and IMEP. The conditions are referred to as Adv 10, Adv 5, Ret 5, and Ret 10 representing injection timings advanced 10 CAD, advanced 5 CAD, retarded 10 CAD, and retarded 5 CAD respectively. The injection, spark, picture timings for the baseline conditions are given in Table 5.3.

#### 5.5.1 Effect of Injection Timing, $\Phi = 0.32$, 600 rpm

As mentioned earlier, this running condition represents the stable limit for lean stratified combustion at a speed of 600 rpm. Therefore, small changes in injection timing
should cause significant changes in engine operation. Figure 5.13 shows the flame-front equivalence ratio PDFs for the five injection timings tested.

The interpretation of the PDFs for the cases with varying injection timing must be done with careful consideration of the stratified fuel cloud position. An important parameter when interpreting the data is the time between the end of injection and the image acquisition. Table 5.4 shows the time between EOI and image acquisition for all of the cases tested with varying injection timing.

A definite trend of increasing equivalence ratio along the flame front is seen in Fig.

Table 5.3. Timing for baseline conditions

<table>
<thead>
<tr>
<th>Condition</th>
<th>Spark</th>
<th>EOI</th>
<th>Image</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>(CAD BTDC)</td>
<td>(CAD BTDC)</td>
<td>(CAD BTDC)</td>
</tr>
<tr>
<td>Φ = 0.32, 600 rpm</td>
<td>17.5</td>
<td>50</td>
<td>0</td>
</tr>
<tr>
<td>Φ = 0.32, 1200 rpm</td>
<td>17.5</td>
<td>55</td>
<td>-5</td>
</tr>
<tr>
<td>Φ = 0.42, 600 rpm</td>
<td>20</td>
<td>52.5</td>
<td>0</td>
</tr>
<tr>
<td>Φ = 0.42, 1200 rpm</td>
<td>20</td>
<td>55</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 5.4. Time between EOI and image acquisition for the conditions tested with varying injection.

<table>
<thead>
<tr>
<th>Condition</th>
<th>Δ_{injection – image} (CAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Adv 10</td>
</tr>
<tr>
<td>Φ = 0.32, 600 rpm</td>
<td>60</td>
</tr>
<tr>
<td>Φ = 0.32, 1200 rpm</td>
<td>70</td>
</tr>
<tr>
<td>Φ = 0.42, 600 rpm</td>
<td>62.5</td>
</tr>
<tr>
<td>Φ = 0.42, 1200 rpm</td>
<td>65</td>
</tr>
</tbody>
</table>
Figure 5.13. PDFs for varying injection timing at a $\Phi = 0.32$ and an engine speed of 600 rpm.

5.13 as the injection timing becomes more advanced, in spite of the longer time available for mixing. However, the location of the fuel cloud at the set ignition timing is non-optimal because the increase in time between the EOI and ignition has allowed the fuel cloud to convect past the spark plug location. Figure 5.14 shows the postulated fuel cloud positions for the different end of injection timings. For the Adv 10 case at the time of image acquisition a significant portion of the main fuel cloud is located in the field of view resulting in relatively higher flame-front equivalence ratios. Figure 5.15 shows the pressure traces for the different injection timings detailing the main portion of heat release around TDC. From the plots of Fig. 5.15, it is evident that the rate of heat release is the slowest for the Adv 10 case supporting the postulated fuel cloud position shown in Fig. 5.14. The slow heat release indicates that at the time of ignition a lean mixture is present at the spark plug resulting in increased combustion variability with a COV of IMEP of 29%.
Figure 5.14. Estimated location of main fuel cloud for $\Phi = 0.32$ and an engine speed of 600 rpm.

As the injection timing is retarded the flame-front equivalence ratio in the field of view becomes progressively leaner as illustrated in Fig. 5.13. This is due to the effect of convection on the fuel cloud position relative to the field of view as shown in Fig. 5.14. The heat release rates increase as the injection timing is retarded until the Ret 5 condition is reached indicating an increase in equivalence ratio at the spark plug at the time of ignition. The higher flame-front equivalence ratios shown in Fig. 5.13 as the injection is advanced are strongly biased by this effect.

Although one is not able to make definitive conclusions as to how the flame-front
Figure 5.15. Pressure plots illustrating the change in combustion phasing and heat release rates for varying injection at a $\Phi = 0.32$ and 600 rpm.

equivalence ratio changes with injection timing for the majority of the fuel mass due to the influence of field of view, it is possible to investigate other features of the stratified fuel cloud. It is postulated that the majority the stratified fuel cloud is visible for the Adv 10 case so an estimation of the flame-front equivalence in the center of the fuel cloud can be made. This information indicates that in the center of the stratified fuel cloud, which is typically located in the vicinity of the spark plug outside of the field of view, the burning zone equivalence ratios are lean and with about 10% of the fuel detected at flame-front equivalence ratios of $\Phi > 1.0$ and about 50% of the fuel mass measured below an equivalence ratio of 0.6. It is likely that for the baseline and retarded conditions the equivalence ratios in the center of the stratified cloud are slightly higher due to the reduced
Figure 5.16. Images for varying injection timing at a $\Phi = 0.32$ and an engine speed of 600 rpm (a) Adv 10 (b) Adv 5 (c) Ret 5 (d) Ret 10; base conditions were shown in § 5.3.5.
mixing time.

The Adv 5 and base conditions are representative of cases where the edge of the main fuel cloud is visible and the corresponding PDFs indicate a skewed distribution with a peak at $\Phi \sim 0.4$ with very little fuel detected at equivalence ratios greater than $\Phi = 1.0$.

As depicted in Fig. 5.14 the two cases with retarded injection have little or none of the main fuel cloud in the field of view. This results in a high occurrence of low equivalence ratios in the PDFs and indicates that much of the fuel detected in this portion of the mixture may have trouble combusting and may result in lean quenching of the flame. Figure 5.16 shows sample images for 4 different injection timings, with the baseline image presented in Fig. 5.10. These images serve to verify the conclusions drawn from the PDFs.

### 5.5.2 Effect of Injection Timing, $\Phi = 0.42$, 600 rpm

The influence of increasing the overall equivalence ratio on the flame-front equivalence ratio is readily apparent from the flame-front equivalence ratio PDFs shown in Fig. 5.17. In the cases with advanced injection, in particular the Adv 5 case, significantly higher flame-front equivalence ratios are present. As was the case for the $\Phi = 0.32$ conditions the position of the stratified fuel cloud has the strongest influence on the flame-front equivalence ratios. Varying the injection timing allows different portions of the fuel cloud to be visualized.

In the Adv 10 case the bulk of the main fuel cloud has convected past the spark plug and is beginning to convect out of the field of view. Figure 5.18 shows sample images illustrating the fuel cloud positioning for the various injection timings. Figure 5.19 shows
Figure 5.17. PDFs for varying injection timing at a $\Phi = 0.42$ and an engine speed of 600 rpm.

The pressure traces for the different injection timings detailing the main portion of heat release around TDC. The slow heat release rate shown in Fig. 5.19 implies the presence of a lean mixture at the spark gap resulting in significant combustion variability and misfires with a COV of IMEP of 42%.

If the injection is retarded to the Adv 5 condition the bulk of the fuel cloud is centered in the image. With this injection timing the flame-front equivalence ratios detected in the field of view have a mean value of $\Phi \sim 0.90$ where the maximum NO$_x$ production is predicted. However, a wide distribution is evident with the bulk of the fuel mass appearing between $\Phi = 0.5$ and $\Phi = 1.50$. The periphery of the main fuel cloud was repeatably present in the area of the spark plug at ignition time so the combustion was quite stable with a COV of IMEP = 3.7%. But, due to the fact that the main fuel cloud was not in the direct vicinity of the spark plug at the time of ignition, the combustion duration for this condition was
Figure 5.18. Images for varying injection timing at a $\Phi = 0.42$ and an engine speed of 600 rpm (a) Adv 10 (b) Adv 5 (c) Ret 5 (d) Ret 10; base conditions were shown in § 5.3.4.
Figure 5.19. Pressure plots illustrating the change in combustion phasing and heat release rates for varying injection at a $\Phi = 0.42$ and 600 rpm.

significantly longer than the baseline case and cases with retarded injection. This is indicated in Fig. 5.19 where the baseline and retarded cases show a linear curve in $\log P - \log V$ coordinates at a normalized volume of 0.13, but the advanced cases are still changing slope indicating a continued heat release.

At the baseline condition the stratified fuel cloud was positioned such that periphery of the cloud was convecting into the field of view at the image time. The fuel cloud location caused a very wide distribution of equivalence ratios present at the flame-front. A sample image showing the fuel cloud portion present in the field of view was given in Fig. 5.8. Lean equivalence ratios represented a majority of the pixels detected on the flame-front, but a fair number of pixels at $\Phi > 1.50$ were also detected. With this injection timing a main portion of the stratified fuel cloud was repeatably placed in the vicinity of spark plug at the
time of ignition resulting in repeatable combustion with a low COV of IMEP = 3.6% and the highest IMEP = 2.6 bar of the injection timings tested.

With the injection retarded by 5 CAD the presence of the rich equivalence ratios was significantly reduced in the PDFs of Fig. 5.17. The distribution of equivalence ratio instead is almost exclusively lean and is centered about Φ ≃ 0.50. The fuel cloud at this injection timing would be approximately centered at the spark plug at the ignition timing giving a low COV of IMEP = 3.5% and a rapid heat release rate with a short main burn duration as indicated by Fig. 5.19.

As injection is retarded further the fuel cloud is still located in the direct vicinity of the spark plug at the time of ignition, but the extent of the flame front in the field of view is significantly reduced resulting in majority of flame-front pixels being detected at equivalence ratios less than 0.5. The reduced time available for mixing gives high flame-front equivalence ratios, but because the stratified fuel is not present in the field of view the resulting images are of an area containing very little fuel.

**5.5.3 Effect of Injection Timing, Φ = 0.32, 1200 rpm**

Unlike some of the 600 rpm data, there appears to be little difference between the flame-front equivalence ratio PDFs shown in Fig. 5.20 for varying injection timings at 1200 rpm. All of the curves show a majority of the fuel pixels at low equivalence ratios. There were very few pixels detected with Φ > 1.0 for any of the cases. The low equivalence ratios detected in all cases indicate that the fuel cloud is less stratified than at the 600 rpm running conditions.
Figure 5.20. PDFs for varying injection timing at a $\Phi = 0.32$ and an engine speed of 1200 rpm.

The images of Fig. 5.21 show that for the retarded injection cases the fuel present in the images is generally at a low equivalence ratio $\Phi \sim 0.1 - 0.4$ with a few small pocket of fuel with an equivalence ratio $\Phi \sim 1.0$ present. As the injection timing was advanced, the images showed larger pieces of the stratified fuel cloud with more fuel of $\Phi \sim 1.0$ evident. All of the injection timings tested ran relatively poorly with the base case having a COV of IMEP of 34 %, therefore the plots of pressure centered about TDC are not given for this condition.

5.5.4 Effect of Injection Timing, $\Phi = 0.42$, 1200 rpm

Figure 5.22 shows the flame-front equivalence ratio PDFs for varying injection timing at 1200 rpm, and $\Phi = 0.42$. The case with injection retarded 10 CAD is missing because one of the data files necessary to correct the images was not captured. Of the PDFs
Figure 5.21. Images for varying injection timing at a $\Phi = 0.32$ and an engine speed of 1200 rpm (a) Adv 10 (b) Adv 5 (c) Ret 5 (d) Ret 10; base conditions were shown in § 5.3.5.
Figure 5.22. PDFs for varying injection timing at a $\Phi = 0.42$ and an engine speed of 1200 rpm.

Figure 5.23. Pressure plots illustrating the change in combustion phasing and heat release rates for varying injection at a $\Phi = 0.42$ and 1200 rpm.
Figure 5.24. Images for varying injection timing at a $\Phi = 0.42$ and an engine speed of 1200 rpm (a) Adv 10 (b) Adv 5 (c) Base (d) Ret 5, the base conditions were shown in § 5.3.4.
present in Fig. 5.22 the two advanced injection cases appear to be substantially richer than
the retarded injection case. Fig. 5.23 shows the pressure traces for the different injection
timings, highlighting the main portion of heat release around TDC. The images of Fig. 5.24
indicate that as the injection timing is advanced from the baseline condition more of the
stratified fuel cloud was present in the field of view. For the base condition and the retarded
conditions little of the stratified fuel cloud was captured in the field of view.

The base condition and the condition with injection retarded 5 CAD ran the best out
of the conditions with varying injection timing with COVs of 8.9% and 10.8% respectively.
This correlates with expected position of the main fuel cloud and with the rates of heat
release apparent from Fig. 5.23.

5.6 Effect of Ignition Timing

In this section the effect of varying spark timing on the flame-front equivalence ratio
is discussed. As in § 5.5 data are presented for two equivalence ratios, \( \Phi = 0.32 \) and \( \Phi =
0.42 \), for the two engine speeds tested. Five different spark timings were tested at each
equivalence ratio/speed combination. The data are presented with the notation of the
baseline condition corresponding approximately to the optimal operating condition with
respect to COV of IMEP and IMEP. The other conditions are referred to as Adv 5,
Adv 2.5, Ret 2.5, and Ret 5 representing spark advanced 5 CAD, advanced 2.5 CAD,
retarded 2.5 CAD, and retarded 5 CAD. The injection, spark, and picture timings for the
baseline conditions were given in Table 5.3. Table 5.5 gives the time in CAD between the
EOI and image acquisition for each condition, and Table 5.6 gives the time in CAD between
the spark time and the image acquisition time. As shown in Table 5.5 the time between the
EOI and the picture time is relatively constant for most of the cases. However, due to the need to have combustion present in the field of view for all cases tested the time between spark and image timing was adjusted depending upon the operating condition.

5.6.1 Effect of Ignition Timing, $\Phi = 0.32, 600$ rpm

Figure 5.25 shows the flame-front equivalence ratio PDFs for varying spark timing at $\Phi = 0.32$ and an engine speed of 600 rpm. The PDFs indicate a clear trend of increasing equivalence ratio the more retarded the spark timing. The time between EOI and the image

<table>
<thead>
<tr>
<th>Condition</th>
<th>$\Delta_{\text{injection} \text{– image}}$ (CAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Adv 5</td>
</tr>
<tr>
<td>$\Phi = 0.32, 600$ rpm</td>
<td>50</td>
</tr>
<tr>
<td>$\Phi = 0.32, 1200$ rpm</td>
<td>60</td>
</tr>
<tr>
<td>$\Phi = 0.42, 600$ rpm</td>
<td>42.5</td>
</tr>
<tr>
<td>$\Phi = 0.42, 1200$ rpm</td>
<td>55</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Condition</th>
<th>$\Delta_{\text{spark} \text{– image}}$ (CAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Adv 5</td>
</tr>
<tr>
<td>$\Phi = 0.32, 600$ rpm</td>
<td>22.5</td>
</tr>
<tr>
<td>$\Phi = 0.32, 1200$ rpm</td>
<td>27.5</td>
</tr>
<tr>
<td>$\Phi = 0.42, 600$ rpm</td>
<td>15.0</td>
</tr>
<tr>
<td>$\Phi = 0.42, 1200$ rpm</td>
<td>25</td>
</tr>
</tbody>
</table>
acquisition was constant for all cases except for the case with spark retarded 5 CAD. Therefore, the field of view should have less of an effect on the PDFs than for the cases with varying injection timing.

The time between the spark timing and picture timing will influence the how far combustion has proceeded at the time of image acquisition. However, due to the changes in mixture present at the spark plug when varying spark timing, the time between spark and image acquisition is not the sole factor affecting the extent of combustion present in the images.

For the Adv 5 case the images of Fig. 5.26 indicate that the combustion has progressed the farthest with the flame front typically located $\frac{1}{2}$ to $\frac{3}{4}$ of the way through the field of view. For the case with spark Ret 5 CAD the images show that combustion has progressed approximately $\frac{1}{4}$ of the way through the field of view. The cases with spark
Figure 5.26. Images for varying spark timing at $\Phi = 0.32$, 600 rpm (a) Adv 5 (b) Adv 2.5 (c) Ret 2.5 (d) Ret 5, the images for the base condition were shown in § 5.3.5.
Figure 5.27. Images illustrating the appearance of combustion luminosity in the images with advanced spark timing at $\Phi = 0.32$, 600 rpm.

Figure 5.28. Pressure plots illustrating the change in combustion phasing and heat release rates for varying ignition at $\Phi = 0.32$ and 600 rpm.
timings in between these two extremes have combustion progress intermediate these cases. The extent to which this affects the PDFs is hard to estimate.

As stated earlier the PDFs indicate that as the spark timing is retarded the flame-front equivalence ratio is increasing. However, this is unintuitive because the time for mixing between the EOI and spark is increased as the spark is retarded, which should result in lower equivalence ratios at the flame-front. One indication of the richer equivalence ratios present in the advanced injection conditions is the appearance of combustion luminosity, most likely due to soot, not present in the retarded conditions. Figure 5.27 shows images from the advanced spark timings illustrating the appearance of combustion luminosity. Figure 5.28 shows the pressure traces for the spark timings detailing the area of main heat release around TDC. The traces indicate that the extent of heat release at the time of image acquisition was reduced as the spark was retarded. This may have been one of the factors influencing the flame-front equivalence ratio present in the field of view. Therefore, it appears that the field of view available is again biasing the results.

5.6.2 Effect of Ignition Timing, $\Phi = 0.42$, 600 rpm

The flame-front equivalence ratio PDFs for $\Phi = 0.42$ and varying ignition timings at 600 rpm are shown in Fig. 5.29, with sample images shown in Fig. 5.31. Figure 5.30 shows the pressure traces detailing the area of main heat release. From the sample image it is obvious that the base condition has burned throughout the field of view, and the rest of the images from the data set support this conclusion, indicating it is further along in the heat release than the other conditions; this is due to the longer time between spark and image
Figure 5.29. PDFs for varying spark timing at a $\Phi = 0.42$ and an engine speed of 600 rpm.

Figure 5.30. Pressure plots illustrating the change in combustion phasing and heat release rates for varying ignition at a $\Phi = 0.42$ and 600 rpm.
Figure 5.31. Images for varying spark timing at a $\Phi = 0.42$ and an engine speed of 600 rpm (a) Adv 5 (b) Adv 2.5 (c) Ret 2.5 (d) Ret 5 the images for the base condition were shown in § 5.3.4.
time as indicated in Table 5.6. The traces of Fig. 5.30 verify this as well. The retarded and advanced cases are at similar points in the heat release as indicated by the flame position, the location of peak pressure relative to the picture timing, and the traces of Fig. 5.30. In the retarded cases the fuel cloud has had slightly more time to convect into the field of view, but the advanced cases have a reduced time available for mixing due to the reduction in time between end of injection and ignition. These offsetting effects result in the advanced and retarded ignition cases having similar flame-front equivalence ratio PDFs. The distributions for these cases are centered about $\Phi \approx 0.7$. All of the spark timings ran well with COVs of IMEP of 5% or less.

### 5.6.3 Effect of Ignition Timing, $\Phi = 0.32$, 1200 rpm

Figure 5.32 shows the flame-front equivalence ratio PDFs for the cases with $\Phi = 0.32$ and varying spark timing at 1200 rpm. Figure 5.33 shows sample images for the
Figure 5.33. Images for varying spark timing at a $\Phi = 0.32$ and an engine speed of 1200 rpm (a) Adv 5 (b) Adv 2.5 (c) Ret 2.5 (d) Ret 5 the images for the base condition were shown in § 5.3.5.
ignition timings. All of the cases had approximately the same time between EOI and the picture timing (~60 CAD). The flame-front equivalence ratio PDFs show similar distributions in the field of view, and the flame-front propagation was similar in all cases even though the time between spark and picture timing decreased by 7.5 CAD degrees going from the most advanced case to the retarded cases. The changes in flame propagation time along with the changes in peak pressure indicate a lower rate of heat release at the advanced timings. All of the cases ran poorly with COVs of 30 % or greater. This was most likely due to the fuel mixing beyond the lean ignition limit throughout much of fuel cloud, and the ignitable mixture only being in small regions for which cycle-to-cycle variability could greatly alter the placement relative to the spark plug. In general for the cases tested the equivalence ratios were lean of stoichiometric with most pixels having $\Phi < 0.8$.

5.6.4 Effect of Spark Timing, $\Phi = 0.42$, 1200 rpm

The flame-front equivalence ratio PDFs at $\Phi = 0.42$, shown in Fig. 5.34, again show a trend of increasing equivalence ratio with retardation of the spark. All of the cases had a similar EOI of $\sim 55^\circ$ BTDC. But the time between spark and the picture decreased in 2.5° increments. Another point to note is that the time between injection and spark increases in 2.5° increments. Figure 5.35 shows the pressure traces detailing the area of main heat release around TDC. The traces indicate that the advanced timings are slightly further along in the heat release. These effects are most likely the cause for the shift in the distribution to higher equivalence ratios as the spark time is retarded. By increasing the time between injection and spark the relative position of the fuel cloud and spark plug is altered. In the cases with retarded ignition the flame is chasing the fuel cloud moving into the field of view
Figure 5.34. PDFs for varying spark timing at a $\Phi = 0.42$ and an engine speed of 1200 rpm.

Figure 5.35. Pressure plots illustrating the change in combustion phasing and heat release rates for varying ignition at a $\Phi = 0.42$ and 1200 rpm.
Figure 5.36. Images for varying spark timing at a $\Phi = 0.42$ and an engine speed of 1200 rpm (a) Adv 5 (b) Adv 2.5 (c) Ret 2.5 (d) Ret 5 the images for the base condition were shown in § 5.3.4.
whereas in the case of an advanced spark time the flame is burning into the main bulk of the fuel cloud so a higher equivalence ratio is present with the retarded conditions.

Figure 5.36 shows sample images during combustion; from these images it is evident that the retarded cases result in a mixture of higher equivalence ratio present in the field of view, resulting in higher flame front equivalence ratios. It should be noted that the distributions detected along the flame-front were in general lean for all of the spark timings tested.

5.7 Emission Results

As discussed in Chapter 3 engine-out NO\textsubscript{x} and CO emission data were only taken at 600 rpm under the conditions of varying injection timing and varying spark timing at $\Phi = 0.32$ and $\Phi = 0.42$. These conditions correspond with the PDFs and images discuss in § 5.5.1, 5.5.2, 5.6.1, and 5.6.2 respectively. The traces of pressure versus crank angle are provided in Appendix A.

5.7.1 Injection Timing Effect, $\Phi = 0.32$, 600 rpm

Figure 5.37 shows the NO\textsubscript{x} and CO emission measurements plotted versus injection timing. Figure 5.38 shows the mean peak pressure versus the mean location of peak pressure for the emissions data.

Apparent from the graph is that the CO emissions increase steadily as the injection is retarded. This indicates an increase in the burning zone equivalence ratio for the bulk of the fuel mass. This directly contradicts the PDFs shown in Fig. 5.13, but as was discussed earlier this is considered to be due to the influence of the limited field of view.
Figure 5.37. Emission measurements for varying injection timing at a $\Phi = 0.32$ and an engine speed of 600 rpm.

Figure 5.38. Peak Pressure versus crank angle of peak pressure of emissions data.
NOx emissions start at a minimum for the case with injection advanced 10 CAD and increase until the base condition timing was reached were the emissions plateau and then decrease slightly as the injection is retarded further. The trend in NOx follows the trend in peak pressure and the two correlate closely as indicated in Fig. 5.39.

The case with injection advanced 10 CAD ran poorly due to the relatively low equivalence ratio mixture near the spark plug at the time of ignition. This resulted in the mixture initially burning at low equivalence ratios and then beginning to combust higher equivalence ratio pockets of fuel as the flame progressed. The NOx emissions were low because the heat release was significantly slower than in the other cases as illustrated by

Figure 5.39. NOx emission versus peak pressure for varying injection and spark timing at $\Phi = 0.32$ and an engine speed of 600 rpm.
having the latest location of peak pressure and the lowest peak pressure of any of the injection timings, and because the condition ran poorly.

When the injection was advanced to the Adv 5 CAD case the engine ran stably with a COV of 6.2%. This was a result of the increased equivalence ratio in the area of spark plug. However, due to the increased time for mixing and the location of the main fuel cloud the location of peak pressure was relatively late. This resulted in a lower peak pressure relative to the base and retarded injection cases and resulted in the highest IMEP, 1.7 bar, due to the relatively favorable combustion phasing.

The baseline condition shows a large increase in peak pressure compared to the advanced injection cases along with the earliest location of peak pressure. The increase in peak pressure coincides with a corresponding increase in NOx shown in Fig. 5.39.

As the injection timing is retarded the stratified fuel cloud becomes richer due to the reduced time for mixing and an increase in CO is seen while the NOx stays relatively constant with a slight decrease seen as the injection timing is retarded 10 CAD. The decrease in NOx corresponds with a pressure decrease as well.

### 5.7.2 Ignition Timing Effect, $\Phi = 0.32$, 600 rpm

The emission measurements of CO and NOx for varying spark timings are shown in Fig. 5.40. The CO remained relatively constant over the entire ignition timing range tested with a slight decrease evident from the most advanced to most retarded injection timings. The decrease in CO is most likely attributable to the increase in available mixing time between the EOI and spark timing in combination with a slightly different fuel cloud position at the time of spark.
The NO\textsubscript{x} emissions appear to decrease linearly as the ignition timing is retarded along with the pressure as indicated in Fig. 5.39. Because the CO changes relatively little over this range, it can be assumed that the global burning zone equivalence ratio varies only slightly. If this assumption is valid then the change in NO\textsubscript{x} emissions is effected through a change in peak pressure, and hence peak cylinder temperature, due to the changes in combustion phasing with changing spark timing. The change in NO\textsubscript{x} is significant with a factor of two decrease from the most advanced to the most retarded conditions tested.
5.7.3 Injection Timing Effect, $\phi = 0.42$, 600 rpm

Figure 5.41 shows the NO$_x$ and CO emission measurements for varying injection timing at $\Phi = 0.42$. Figure 5.42 shows the mean peak pressure plotted versus mean location of peak pressure. The CO emission increased steadily as the injection timing was retarded. As mentioned earlier this indicates an increase in the global burning zone equivalence ratio as the injection is retarded. The case with injection Adv 10 had a COV of 63% where as the case with Adv 5 had a COV of 23%. The base condition and retarded conditions ran stably with COVs of 5% or less indicating that the bulk of the stratified fuel cloud was located in the vicinity of the spark plug at the spark timing for these conditions. The reduced time for mixing is the main influence causing increased CO emissions for the retarded injection conditions.

The NO$_x$ emissions follow a similar trend to the $\Phi = 0.32$, 600 rpm data with varying injection timing. As discussed in § 5.7.1 the condition with injection Adv 10 CAD was close to the lean ignition limit at the spark location at spark time. This resulted in unstable combustion, several misfires were present, as well as the initial flame-front equivalence ratios being lean. The end result was an increase in the combustion duration giving a lower and later peak pressure corresponding to low NO$_x$ production.

For the Adv 5 condition the bulk of the stratified cloud was closer to the spark plug at the time of ignition than for the Adv 10 case. The fuel cloud location caused the combustion to be more stable than the Adv 10 CAD condition resulting in a significant increase in peak pressure. The NO$_x$ had a significant increase corresponding to the increase in peak pressure as indicated in Fig. 5.43.
Figure 5.41. Emission measurements for varying injection timing at a $\Phi = 0.42$ and an engine speed of 600 rpm.

Figure 5.42. Peak Pressure versus crank angle of peak pressure of emissions data at a $\Phi = 0.42$ and an engine speed of 600 rpm.
The base condition had the highest peak pressure and ran well, indicating a favorable mixture located at the spark plug at the ignition timing. The base condition also corresponded with the peak NO\textsubscript{x} production. As the injection was retarded the peak pressure decreased and the location of peak pressure moved before TDC indicating a very short combustion duration. Even though the pressure decreased for the two cases with retarded injection the NO\textsubscript{x} remained relatively constant with a slight decrease noticeable. The NO\textsubscript{x} did not decrease as fast as the peak pressure because the peak pressures occurred before TDC resulting in the mixture residing at the a high pressure and temperature for a longer period of time when compared with the advanced injection conditions. The base condition
and retarded conditions had similar NO\textsubscript{x} production even though the equivalence ratio changed by a significant amount as indicated by the CO emissions.

5.7.4 Ignition Timing Effect, $\Phi = 0.42$, 600 rpm

Figure 5.44 shows the emission measurements made at $\Phi = 0.42$ and 600 rpm with varying spark timing. The data shows trends very similar to those of the $\Phi = 0.32$ data. In Fig. 5.44 the NO\textsubscript{x} emissions decrease steadily while the spark is being retarded from the Adv 5 CAD condition. The CO emissions decrease slightly as the spark is retarded but are relatively constant between the Adv 2.5 and Ret 2.5 conditions. The slight decrease in CO is most likely due to the slight increase in available mixing time between the EOI and the

![Graph showing emission measurements for varying spark timing at $\Phi = 0.42$ and 600 rpm.](image)

**Figure 5.44.** Emission measurements for varying spark timing at a $\Phi = 0.42$ and an engine speed of 600 rpm.
spark time as the spark is retarded, resulting in slightly lower global flame-front equivalence ratios in the stratified fuel cloud.

Figure 5.42 shows the peak pressure plotted versus the location of peak pressure. The relationship is linearly decreasing with increasing location of peak pressure ATDC. The NO$_x$ emissions mirror the linear decrease.

### 5.8 Discussion

The PDF and image results presented in this chapter provide insights into what occurs during stratified combustion. Although the data were only taken in a small region of the engine cylinder they can be used to draw some significant insights into the state of the flame-front equivalence ratio during combustion, and how the equivalence ratio is affected by varying certain parameters such as ignition and injection timing.

Much of the data presented in this chapter were for late injection timings, which are of significant interest for stratified combustion. The data presented in § 5.3 – 5.6 show a large array of different fuel distributions under different conditions. Many of the cases correspond to data taken in the periphery of the main fuel cloud. These cases indicate the existence of a large area of the flame front burning into very lean equivalence ratios, down to the detection limit of $\Phi \approx 0.1$. The low equivalence ratios are also apparent in the images. These areas of low equivalence ratio may contribute to the HC formation due to lean quenching of the flame as observed by Strand, and as predicted by Anderson et al [8, 7].

The data with varying injection timing allowed for the bulk of the stratified fuel cloud to be observed in the field of view. The PDFs of equivalence ratio shown in Figs. 5.13 and 5.17 for the cases with injection Adv 10 and Adv 5 indicate that during the combustion
of the bulk-stratified fuel cloud a large range of equivalence ratios exist. In the case with $\Phi = 0.32$ we notice that the majority of the distribution is lean even in the main fuel cloud.

With an equivalence ratio of $\Phi = 0.42$ the distribution is very wide and centered at a $\Phi \approx 1.0$. The PDF response in both cases is significantly broader than what is expected if the bulk of the fuel were combusting at one equivalence ratio as in the homogeneous test conditions. Therefore, the existence of a wide range of equivalence ratios at the stratified cloud flame front is real and these two distributions indicate that in general the fuel is burning at a wide range of equivalence ratios both rich and lean of stoichiometric.

Air utilization and mixing of the stratified charge was an issue apparent at many of the running conditions. In the cases with equivalence ratios approaching the upper load limit of late injection stratified combustion, $\Phi = 0.5 - 0.6$, the mixture was significantly rich due to the short available mixing time. This was noticeable in the PDFs of Fig. 5.11 where the cases with $\Phi = 0.52$ and $\Phi = 0.61$ showed the presence of fuel burning at the flame-front equivalence ratios greater than $\Phi = 1.4$. These rich equivalence ratios resulted in rapid fouling of the spark plug when the engine was run for $\sim 60$ s or more causing repeated misfires and coating a large portion of the in-cylinder surfaces with soot.

The over-mixing of the fuel resulted in much of the fuel mixing beyond the lean ignition point in the cases at 1200 rpm with a $\Phi = 0.32$. The increased mixing was most likely due to the increased turbulence near TDC caused by the squish air motion. All of the late injection cases at 1200 rpm were mixed to a greater extent than the 600 rpm conditions. At the $\Phi = 0.32$ case this resulted in poor combustion stability with the best COVs achieved being only 30%.
The emissions data taken at the 600 rpm conditions with varying injection and varying spark bring some very interesting insights when coupled with the knowledge gained from the flame-front equivalence ratio measurements. The NO\textsubscript{x} emissions for all of cases tested correlated very closely with changes in peak pressure or basically the combustion phasing. For cases with varying spark timing this is very obvious. The data for varying spark also indicate that the flame-front equivalence ratio does not play as significant a role as one might expect. This coupled with the knowledge that a broad range of equivalence ratios are present at the flame-front indicates that combustion phasing is the dominant parameter. The dependence of the NO\textsubscript{x} emissions on peak pressure indicate that the formation is highly coupled with the temperature, which points towards the Zeldovich mechanism being the key contributor to NO\textsubscript{x} formation.
6 Conclusions and Recommendations

Simultaneous planar laser-induced fluorescence imaging of 3-pentanone doped into iso-octane and OH naturally present in the combustion products was performed to determine the local equivalence ratio and the location of the flame front within the images during stratified combustion. A method was developed to automatically determine the flame-front equivalence ratio under both homogeneous and stratified engine combustion. The accuracy and fidelity of the method were verified using homogeneous test conditions where the flame-front equivalence ratio was known \textit{a priori} (Chapter 4).

The method was applied to experimental data taken for varying engine speed between 600 and 1200 rpm. The PDFs of the flame-front equivalence ratio showed an increase in the mixing for the late injection cases, whereas for the early injection condition the 1200 rpm case was less mixed than the 600 rpm condition.

The limited field of view was found to have a large influence on the data taken with varying injection timing, and varying spark timing. In some instances the main stratified fuel cloud was visible and in other cases it was not. This made it necessary to use the knowledge of the field of view relative to the spark plug and injectors when discussing the PDFs shown in sections 5.3-5.6.

The images and the PDFs both indicated that a wide variety of equivalence ratios exist during stratified combustion. This information coupled with the emissions and pressure data to show that the formation of NO$_x$ is relatively insensitive to the flame-front equivalence ratio during stratified combustion except to the extent that they affect the combustion phasing. Instead the phasing of the combustion appears to be the most
significant parameter. This was apparent in the emissions data where a factor of 2 decrease in NO\textsubscript{x} could be obtained by retarding the spark 10 CAD from the most advanced condition, while the CO measurement indicated only a slight change in the burning zone equivalence ratio over that range.

The wide range of equivalence ratios present in the images, and in cases where the edge of the main fuel cloud was observed, and the large portion of flame-front trying to propagate into lean equivalence ratios suggest that a significant mass of fuel could end up unburned in the overall lean conditions due to bulk quenching of the flame.

Based upon the results presented in this study, it is recommend that more work be done with the method to developed a further understanding of the emissions formation during stratified combustion. One of the major influences on the image data obtained for this study was the field of view. Based on knowledge developed in this study, experiments with fields of view situated on both sides of the spark plug locations are suggested. The data for varying spark and injection should be repeated. These data coupled with further emission measurements could give a definitive answer to the influence of the local flame-front equivalence ratio on the NO\textsubscript{x} emissions formation. It would also be helpful to obtain data at spark time to determine the state of the distribution at that particular instance.

In addition to the experimental work suggested above, it is recommended that further work be done to obtain heat release information from the experimental pressure traces obtained for the various running conditions. In conjunction with this an accurate estimate of the maximum bulk gas temperatures for the various conditions tested would be of significant interest to determine if the temperatures reached are great enough for the Zeldovich mechanism to contribute significant amounts of NO\textsubscript{x}.
References


Appendix A  Average Pressure Traces for Emission Measurements

Figure A.1. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying injection, Adv 5.

Figure A.2. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying injection, Adv 10.
Figure A.3. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying injection, Base.

Figure A.4. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying injection, Ret 5.
Figure A.5. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying injection, Ret 10.

Figure A.6. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying ignition, Adv 5.
Figure A.7. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying ignition, Adv 2.5.

Figure A.8. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying ignition, Base.
Figure A.9. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying ignition, Ret 2.5.

Figure A.10. Average pressure trace for $\Phi = 0.32$ at 600 rpm with varying ignition, Ret 10.
Figure A.11. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying injection, Adv 10.

Figure A.12. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying injection, Adv 5.
Figure A.13. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying injection, Base.

Figure A.14. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying injection, Ret 5.
Figure A.15. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying injection, Ret 10.

Figure A.16. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying ignition, Adv 5.
Figure A.17. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying ignition, Adv 2.5.

Figure A.18. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying ignition, Base.
Figure A.19. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying ignition, Ret 2.5.

Figure A.20. Average pressure trace for $\Phi = 0.42$ at 600 rpm with varying ignition, Ret 5.