The Influence of Elliptical Nozzle Holes on Mixing and Combustion in Direct Injection Natural Gas Engines

by

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A thesis submitted in conformity with the requirements for the degree of Master of Applied Science
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Abstract
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Experiments were conducted to compare mixing and combustion of natural gas jets from round and elliptical nozzle holes in an optically accessible combustion bomb. A flame ionization detector was used to measure the concentration fields of the two jet types. Pressure data, combustion imaging, and hydrocarbon measurements of exhaust gas were used to compare the ignition delay, heat release, and combustion efficiency of the two nozzles.

Concentration measurements indicated that the elliptical nozzle produced jets with smaller rich core regions and lower peak concentrations at all conditions. Firing tests indicated that the two nozzles produced equivalent ignition delays. Peak heat release rates were higher for the round nozzle, while the elliptical nozzle produced smoother transitions from premixed to diffusion burning. Combustion efficiency was slightly higher for the round nozzle. Results indicate that elliptical nozzles could potentially lower NO\textsubscript{x} and particulate emissions, but further experiments are required to test this hypothesis.
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Chapter 1

Introduction

1.1 The Demand for Alternative Fuels

Significant demand for development of alternative vehicle fuels first began as a response to the oil shocks of the 1970s, as the OPEC embargo of 1973 and the subsequent Iranian embargo of 1979 revealed the dangers of excessive dependance on foreign supplies of crude oil. After world oil prices stabilized in the 1980s, the initial push for alternative technologies lost momentum and received little attention for the next decade. However, in the late 1980s and early 1990s interest in alternative fuels revived as it was recognized that many had the potential to significantly reduce emissions, which were becoming an increasing concern. It was still generally held, however, that AFVs could not expect any significant market penetration while global oil prices remained low [1]. With recent sustained increases in crude oil prices (see Fig. 1.1), the continued degradation of urban air quality, and growing concerns about global warming, factors have combined to add urgency to the demand for viable long term alternatives to traditional gasoline and diesel powered vehicles.

The primary advantage of most alternative fuels is the potential for reduction of regulated emissions. Currently the emissions of primary concern are carbon monoxide (CO), particulate matter (PM), oxides of nitrogen (NOₓ), and reactive non-methane hydrocarbons (NMHC). Carbon monoxide has well known adverse health effects, while NOₓ and NMHCs both contribute significantly to the formation of ground-level ozone, the main component of photochemical smog. High concentrations of particulate matter also generate smog, and have been linked to increased incidence of respiratory illness. Finally, carbon dioxide (CO₂) and gaseous hydrocarbons (including methane) are greenhouse
gases and therefore contribute to climate change. There is tremendous interest in CO\textsubscript{2}, and although it is not currently regulated it may be targeted for reductions in the future in an attempt to meet agreements such as the Kyoto Protocol. As mentioned above, most of the substances identified as potential alternative fuels can significantly reduce some or all of these emissions when compared to current gasoline and diesel engines.

Due to distribution and infrastructure issues, alternative fuels have typically been seen as suited to heavy duty and fleet vehicle markets, since these applications often utilize more centralized refuelling facilities. The heavy duty vehicle market is currently dominated by diesel engines due to their favourable power and efficiency characteristics. However, the overall number of diesel engines in service in North America has remained a small fraction of the total market. The low number of diesel engines in service has led to less stringent controls on allowable diesel emission levels. Recently, however, as a result of rising fuel prices the overall number of diesel engines entering the vehicle market has
Chapter 1. Introduction

increased, making diesel engines a target for more aggressive future emission controls. Figure 1.2 summarizes the proposed limits on PM and NO\textsubscript{x} for diesel engines into the near future. Note that CO is not included, since it is not a significant issue for diesel engines. The difficulty of achieving some of these targets with conventional diesel engines adds to the desirability of a feasible alternative fuel diesel technology.

1.2 Potential Alternative Fuels

Numerous alternative vehicle fuels have been identified and evaluated, each with its own benefits and drawbacks. Methanol (CH\textsubscript{3}OH), ethanol (C\textsubscript{2}H\textsubscript{5}OH), vegetable oils & vegetable oil alkyl esters (biodiesel), hydrogen, liquified petroleum gas (LPG) or propane, compressed and liquified natural gas(CNG/LNG) and electricity (derived either from hydrogen fuel cells or grid power) have all been tested with varying degrees of success. The main benefit, as discussed above, is the reduction in harmful emissions these fuels afford. The drawbacks, however, are more specific to the various fuels. For example, methanol and ethanol both suffer from cold start problems and require engine modifications to prevent corrosion [4, 5]. Vegetable oils and biodiesels can potentially ‘gel’ or polymerize [6], resulting in fuel system blockages. Grid powered electric vehicles have been hampered by lack of suitable battery technology in terms of cost, energy density and driving range. Fuel cells, one of the most highly touted alternatives, have been notoriously slow to progress towards commercial viability \footnote{For example, a prediction was made by Daimler-Benz & Ballard in 1998 that price competitive fuel cell cars and buses would be available within 6 years [7]. In 2005, 7 years later, Ballard announced plans to have price competitive vehicles available by 2010 [8].}. Finally, virtually any fuel that exists as a gas at standard conditions (LPG, CNG, H\textsubscript{2}) will suffer considerable range penalties due to lower volumetric energy densities, and require the dedication of large volumes and weights to high pressure storage tanks.

It is generally accepted that engine technologies must be optimized for a given fuel in order to fully exploit its potential [4, 5]. This is important for two reasons. First, the unique qualities of a fuel will dictate that engine operating parameters, particularly compression ratio, be set to obtain the maximum efficiency and power output available. Second, so called flexible fuel vehicles (FFVs), capable of running on two different fuels (most frequently gasoline and ethanol), require additional space and weight to accommodate multiple fuel tanks, and operators may simply fuel with gasoline alone thereby
negating the benefits of the alternative system \[9\].

Given that each potential alternative fuel has unique properties and requires uniquely optimized technology, the need for detailed study of any one particular option is clear. As natural gas is the fuel of interest in this study, its own unique properties will now be discussed in greater detail.

1.3 Natural Gas as a Vehicle Fuel

Natural gas is a plentiful and versatile resource found in many areas worldwide. The primary component of natural gas is methane, usually comprising 90 to 97% by volume, with the remainder made up of ethane, propane, butane, traces of higher alkanes, carbon dioxide and nitrogen. Natural gas has numerous possible advantages as a vehicle fuel. Its low carbon to hydrogen ratio provides for reductions in CO\(_2\) of up to 26\% \[10\], and reduces soot formation. Low levels of impurities reduce the formation of aromatics and other toxins, and sulfur dioxide(SO\(_2\)) (a source of particulate smog and a precursor to acid rain). NG also has a lower adiabatic flame temperature and wider flammability limits than gasoline or diesel, enabling reductions in NO\(_x\) by lowering peak combustion temperatures and enabling lean operation. Lean combustion can also lower fuel consumption and CO production.

Natural gas is already widely used in residential heating and power generation so there is an extensive existing distribution infrastructure around which refueling stations could be built. In addition to its uses in generation, natural gas serves as a feedstock for a multitude of other chemical compounds. Specifically, many other alternative fuels including methanol, ethanol, and hydrogen can be derived either directly or indirectly from natural gas. A recent analysis by Hekkert et al. \[11\] compared total “well-to-wheel” fuel chain efficiencies of several of these natural gas derived fuels, employed in various end-use vehicle technologies, to the efficiencies of current crude oil derived fuel chains. They found that of the fuels employed in internal combustion engines only compressed natural gas outperformed the reference crude oil chains. This led them to identify CNG IC engines as the ideal transition technology for development of a refueling infrastructure to serve future hybrid electric and fuel cell powered vehicles using gas as their feedstock. They also noted that engine efficiency was of primary importance in determining the overall efficiency of a fuel chain. Thus, CNG remains a viable option for further development as a niche alternative fuel from an economic and energy efficiency standpoint.
1.3.1 Natural Gas Engine Technologies

Of the natural gas engines currently in use, the majority operate on spark ignition cycles. Methane has a very high octane rating of 130, giving it excellent knock resistance. Natural gas blends consisting primarily of methane therefore inherit enhanced knock resistance as well. High knock resistance in turn allows natural gas SI engines to operate with compression ratios of 11 to 11.5:1, compared to typical SI ratios of 8 to 9:1 [12]. To take advantage of the potential for high compression ratios, a typical strategy has been to convert diesel engines to spark-ignition and then fuel them with natural gas. The diesel to spark ignition conversion process has some negative effects, however, in terms of performance and emissions. Wide variations in emission levels have been observed for different conversion technologies [13], and many of the converted engines suffer from extremely high total hydrocarbon emission levels. On the other hand, when existing SI engine technology is modified to accommodate NG the results can be much more positive [14]. The dependence of performance on conversion technology reinforces the need for dedicated, optimized NG engine technologies to maximize the benefits inherent to the fuel.

Spark-ignition natural gas engines can be divided into two main classes: Stoicheometric Otto-Gas Engines and Lean-Burn Otto-Gas Engines. As the name indicates, Stoicheometric Otto engines operate near or at an equivalence ratio of $\phi = 1$. Due to the large volumes of gaseous fuel required for stoicheometric combustion, a significant amount of air is displaced within the intake manifold. The fact that the fuel is already a gas when it is injected also means that there is no evaporative cooling of cylinder gases, resulting in lower charge density. Both of these effects decrease volumetric efficiency, causing power losses on the order of 5-15%, and energy specific fuel penalties of 15-20% relative to gasoline fuelled SI engines [10]. For high transient cycles this penalty can increase to up to 50% due to additional throttling losses.

The Lean-Burn Otto gas engine has several advantages that have made it the SI technology of choice for natural gas. First, lean operation provides significantly higher power and torque output by displacing less intake air, although lean-burn engines still suffer 15% fuel penalties compared to current diesel engines. Lower NO$_x$ can also be achieved due to lower peak combustion temperatures. The ability to run at very lean equivalence ratios is limited at lower loads as combustion stability degrades, reducing performance benefits at these conditions. Also, as equivalence ratios are lowered to further drive down
NO\textsubscript{x} and fuel consumption, emissions of unburned hydrocarbons will increase, resulting in a trade-off between performance and emissions [15]. Simultaneous reduction of both NO\textsubscript{x} and HC emissions cannot be achieved without unduly impacting engine efficiency, leading Corbo, Gambino and Iannaccone to conclude that “at the present development status, the NG lean burn engine does not succeed in meeting the future requirements of both low NO\textsubscript{x} emissions and low GWI [global warming impact] [15].”

In attempting to overcome the power and efficiency deficits that spark ignition technology imposes on natural gas engines while still maintaining acceptable emission performance, the use of direct injection compression ignition has been an attractive alternative. Direct injection engines can achieve much higher compression ratios than spark ignition, up to 18:1, and without intake air throttling or premixing of fuel and air they achieve higher volumetric and thermal efficiencies. By injecting the gas directly into the chamber near the end of the compression stroke, the potential for premature autoignition is reduced or eliminated. This also makes direct injection engines less sensitive to variation in the composition of the natural gas being used. Overall, this technology is theoretically capable of achieving high power densities, equivalent to current liquid diesel technologies, while still retaining the favourable emission characteristics of natural gas.

NO\textsubscript{x} and particulate matter are typically the emissions of greatest concern in diesel engines. Natural gas has a lower adiabatic flame temperature and slower burning velocity than diesel fuel, both of which will tend to lower peak cylinder temperatures and reduce NO\textsubscript{x}. The small carbon to hydrogen ratio and low impurity levels of NG also reduce the potential for particulate formation. This effectively shifts the typical NO\textsubscript{x}-particulate tradeoff (see Fig.1.3) curve towards the origin, achieving lower levels of both emissions without seriously affecting engine performance.
1.3.2 Natural Gas Vehicle Emissions

Existing natural gas engines have demonstrated significant emission benefits and are well documented in the literature. Carbon monoxide reductions of up to 95% have been reported [13, 16, 17, 18, 19, 14], along with CO\textsubscript{2} reductions of 6.5 to approximately 20% [13, 16, 20]. NO\textsubscript{x} emissions have typically been reduced, with levels up to 99% lower than comparable diesel engines [13, 19, 14]. Particulate matter has also been very successfully controlled, with reductions of up to 99% [13, 16, 19].

Hydrocarbon emissions from natural gas engines are a slightly more complex matter. Total hydrocarbon (THC) levels from NG engines are often found to be significantly higher than comparable gasoline or diesel engines, often by one or two orders of magnitude [16, 13]. In other cases, however, total hydrocarbon levels have been comparable to, or even lower than, diesel or gasoline baselines [14] (this is largely a function of engine type and combustion chamber design, as mentioned briefly above). In any case, the vast majority of HC emissions from an NG engine consist of methane [21], which is extremely unreactive, with an atmospheric life of 10 to 20 years [22, 23]. Thus, of greater concern in terms of local air quality and smog formation are NMHC levels, as these compounds account for 94 to 98% of the HC reactivity in NG engine emissions [21]. NMHC levels from NG engines are typically of the same order [16] or lower [13] than current engines, resulting in lower overall ozone formation potential [18].

Emissions of methane have been of increasing concern recently, as it is a greenhouse gas several times more powerful than CO\textsubscript{2}\textsuperscript{2}. Atmospheric methane concentrations have also shown some of the most drastic increases of all greenhouse gases [24]. However, the potential contribution of natural gas engines to atmospheric methane has been called into question, with a prediction that even significant market penetration would lead to at most a 0.025% increase in global concentrations [25]. Obviously, global warming related issues are vastly complex and can’t be dealt with in any way that would do them justice here. At the very least the emission of unburned CH\textsubscript{4} from a natural gas engine represents wasted fuel energy and lost efficiency, and should be minimized on this principle alone.

\textsuperscript{2}The exact relative global warming potential (GWP) of methane is not clear, and depends heavily on several factors such as atmospheric life of CO\textsubscript{2} and consideration of relative versus instantaneous levels of radiative forcing [23]. Values as high as 62 [15] have been used, though values of 3.7 or less (on molar basis) can be inferred from other analyses. See [23] and [22] for detailed treatments of this issue.
Chapter 1. Introduction

1.4 Objective

Over the past decade a number of projects have been carried out at the University of Toronto’s Engine Research and Development Lab with the goal of better understanding the injection and combustion processes within directly injected natural gas engines. Flow visualization techniques were used to analyse the penetration and spreading of gas jets within a simulated combustion chamber, and a model of the jet was developed [26]. A study of ignition delay for a single gas jet ignited by a glow plug was then carried out and led to the optimization of glow plug placement and the design of a glow plug shield to provide short ignition times at various operating conditions [27]. Finally, the early stages of combustion and propagation of the gas flame between multiple jets were investigated. It was found that chamber geometry, gas jet location, and glow plug shielding were the dominant factors influencing the effectiveness of combustion and flame propagation [28].

One of the remaining challenges for natural gas diesel technologies is to minimize the production of smoke during combustion, hopefully avoiding the need for particulate traps in the exhaust system. The best way to reduce smoke is to improve the mixing between the fuel jets and air within the chamber, allowing enough air entrainment into the fuel to ensure complete combustion at all conditions. Traditionally, improved mixing has been achieved by decreasing injection orifice size or increasing injection pressure. However, limits on material strength and machining capabilities lead to a point beyond which further benefits from these traditional methods become too difficult or expensive to warrant serious attention. Thus, alternative methods of enhancing fuel-air mixing may prove to be worthwhile for this engine type. Fundamental work on the structure of jets issuing from elliptical nozzle holes has suggested that the potential exists for greatly enhanced air entrainment and modified spreading characteristics from this nozzle shape that could benefit the operation of a diesel engine [29], [30].

The goal of this thesis is to expand on the previous investigations by considering a new method to enhance the mixing of natural gas and air within the combustion chamber of a direct injection NG engine. The use of elliptical, rather than round, injector nozzle holes will be examined to determine their effect on jet structure and mixing prior to combustion, as well as subsequent effects on ignition delay and combustion efficiency. It is hoped that this may prove the viability of elliptical nozzle holes as a method for reducing smoke and hydrocarbon emissions, and increasing the efficiency of combustion in engines fueled by natural gas and other gaseous fuels.
Chapter 2

Theoretical Considerations

2.1 Quasi-Static Jet Development

Proper design of an effective combustion system for a natural gas engine requires a clear understanding of the various factors influencing the injection and jet development processes. The fuel injection process in a D.I. engine is by necessity transient, so the structure and size of a jet in this environment cannot be described with standard fully-developed jet models that require invariance of major properties with respect to time. A reliable model of jet travel time, spreading rate, and mixing is further complicated by the presence of high-velocity swirl, confinement effects from nearby walls, and interactions between multiple adjacent jets. Insight into the injection process is provided by recourse to transient jet models, and design-specific consideration of factors related to the combustion chamber geometry.

2.1.1 Transient Jet Structure

Transient development of a round jet is typically modeled via the application of a two-zone model such as that described initially by Turner [31]. Turner proposed that the transient jet could be modeled as a traveling spherical head vortex ahead of a self-similar (quasi-steady state) jet. The spherical head region is fed with mass and momentum from the jet region while being slowed by viscous forces; the head vortex also entrains very little fluid from the surroundings. The quasi-steady jet region exhibits self-similarity, allowing its structure at any cross-section to be described by a single correlation. Images of the two-zone structure were captured by Rizk [32], confirming the general assumptions from
Chapter 2. Theoretical Considerations

Figure 2.1: The approach to self-similarity of an impulsively started round jet. Inset images demonstrate the general validity of the two-zone model [33].

Turner about development and mixing. Images of the early penetration of an impulsively started round jet, along with data confirming the approach to self-similarity are shown in Figure 2.1.

The two-zone model was successfully applied to describe impulsively started laminar jets [34], and later extended to turbulent jets [35]. Birch et al. [36] measured the concentration field for a turbulent methane jet, providing a general outline of the axial and radial decay within the jet. They also confirmed the near self-similarity of the normalized profiles of radial concentration within the quasi-steady region.

The early analyses and experiments described above were applied only to incompressible (subsonic) jets. When the pressure difference across a nozzle reaches the critical pressure ratio for the fuel gas, the jet issuing from a nozzle will become choked and effects of compressibility must be accounted for. The critical pressure ratio for any gas is given by:

\[
\frac{P^*}{P_0} = \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{(\gamma - 1)\gamma}}
\]

where \(\gamma\) is the specific heat ratio of the gas \(^1\). If the reservoir pressure (\(P_0\)) is increased beyond this point the jet becomes underexpanded, such that some portion of the expansion to ambient pressure must take place outside of the nozzle. Underexpansion in a

\(^1\)For natural gas, with a typical \(\gamma\) of 1.31, the critical pressure ratio is 0.544.
round jet leads to the development of a series of expansion waves that are reflected at the flow boundary and then coalesce, resulting in the formation of a “barrel shock.” The barrel shock is typically terminated by a circular Mach disc, after which the flow rapidly equilibrates into a nearly sonic jet. The structure of this near nozzle region, as described and photographed by Ewan and Moodie [37] is shown in Figures 2.2 and 2.3.

In addition to capturing images of the shock structure in the near field region, Ewan and Moodie also tested several correlations for both velocity and concentration decay to determine the proper description of the self-similar underexpanded jet [37]. The shock region immediately downstream of the nozzle is generally quite short, on the order of 1 to 3 nozzle diameters (depending on pressure ratio). The shocks cause a very rapid expansion of the jet followed by an expansion and decay typical of a fully expanded (sonic) jet. Ewan and Moodie found that the jet in these cases can be treated like
a sonic jet of equal mass flow rate issuing from a “virtual” orifice of larger diameter located at some distance from the actual nozzle. The equivalent diameter of the virtual nozzle is a function of the jet discharge coefficient and pressure ratio. Using this method, Ewan and Moodie found excellent collapse of the concentration and velocity profiles of underexpanded gas jets. This confirmed the validity of extending the model of Turner to include fully- and under-expanded jets.

Another model of transient jet structure similar to that of Turner, and especially useful in combustion applications, is described by Hyun et al. [38]. They divide the jet into four distinct regions, as seen in Figure 2.4. The potential core is the region nearest the nozzle exit. The flow is quite stable in this region, up to the point where small scale turbulence begins to form. The main jet region follows the potential core, and falls within the central portion of the jet, along the centreline and between the shear layers. The main jet region feeds momentum into the dilution region, which is largely the same as the spherical tip described by Turner. Hyun et al note that a very lean region develops between the main body of the jet and the spherical tip/dilution region. Of greatest interest in terms of mixing, ignition and combustion is the mixing flow region, encompassing the shear layers that develop at the edges of the main jet region. The shear layers develop and grow as turbulence is generated by the jet, leading to mixing of the jet gases with the surroundings. This region is where the majority of the air entrainment into the jet and jet spreading in the radial direction occurs. It is also the region where a local ignition source must be placed for best performance, as discussed in Section 2.2.

2.1.2 Unconfined Jet Penetration and Mixing

Applying the structural model of Turner, Hill and Ouelette [33] developed a model for jet penetration that would be applicable to a gas jet within an engine. They deduced
that the penetration of the jet could be represented by

\[ Z_t = \Gamma \left( \frac{\dot{M}}{\rho} \right)^{\frac{1}{4}} t^{\frac{1}{2}} \]  \hspace{1cm} (2.2)

where \( \dot{M} \) is the jet momentum flowrate, \( \rho \) is the chamber gas density and \( \Gamma \) is a constant. Taking the value \( \Gamma = 3.0 \pm 0.1 \), this relationship was found to accurately fit numerous existing data sets measuring the penetration rate of gas jets. Hill and Ouelette also found that precisely the same constant applies in the case of highly underexpanded jets, such as would be found in a DI engine.

Jennings and Jeske [39] used simulations of natural gas injection into a constant volume chamber to determine the effects of nozzle design on jet penetration. They found initially that a greater number of smaller nozzle holes provided better mixing with ambient air by reducing the size of the jet’s fuel rich core region. However, they also noted that since the mixing is largely controlled by the turbulence generated by the jet larger nozzle holes, which generate more turbulence energy, tended to offset some of the disadvantage of a larger fuel-rich core region. The extent of mixing achieved with the large holes late in the cycle was therefore only slightly lower than that of the smaller nozzle holes. The advantage of small holes early in the injection process could still prove to be important in an engine application, as it will improve the ignition delay properties and reduce the tendency to produce soot relative to a jet with a larger rich core.

In their simulations of constant volume injection, Jennings and Jeske also found that the level of turbulence energy in the surroundings (i.e. swirl intensity) had a negligible effect on the mixing rate of the jet, as the turbulence generated by the jet itself was much greater than that of the surroundings at all conditions [39]. A similar insensitivity to swirl was noted, for similar reasons, by Abraham et al. [40]. In particular, Abraham’s simulations revealed that a low inertia gas jet will tend to be convected by the chamber swirl, reducing the velocity gradient between the two flows, and limiting the generation of additional turbulence. This is an important point to note: while the presence of swirl does not appear to significantly affect the rate of mixing of the jet it can still have an important effect by bending the axis of the jet, changing its position within the chamber, especially relative to any point sources of ignition.

Abraham et al. [40] compared the mixing of gas jets with liquid diesel sprays, and the results have interesting implications for gas engine design. It may intuitively seem as though a gas jet should mix faster than an equivalent liquid spray due to the time needed
for evaporation of droplets. However, Abraham’s simulations revealed that under normal circumstances a liquid spray will in fact mix faster and spread further than an equivalent gas jet, though the gas jet will penetrate through the ambient air more quickly. The different mixing behaviour results from the effectiveness of the liquid spray at transferring momentum to the surroundings. The small droplets evaporate very quickly, transferring their momentum to the surroundings. This momentum transfer increases the turbulence levels in the mixing region, leading to quick growth of the jet shear layers and mixing of the fuel to within flammability limits. Larger droplets, on the other hand, penetrate further into the surroundings before evaporating and transferring their momentum. This means they tend to end up in regions where a large amount of ambient air is available, again allowing them to quickly mix to within flammability limits. It should be noted that further simulations revealed that when combustion is accounted for, the differences between liquid and gaseous jets were reduced [41]. However the initial results still have potentially important implications during the pre-ignition period.

2.1.3 Effects of Jet Interaction

Not surprisingly, the behaviour of a gaseous fuel jet within an engine departs significantly from the ideal situation of the free jet described above. While the jet’s underlying structure remains unchanged, the influence of adjacent jets and confining surfaces encountered within a typical combustion chamber can significantly change the actual penetration and mixing observed. In fact, the injection, ignition, and combustion properties within an engine are often most significantly influenced by the specific injector configuration and chamber geometry.

In attempting to apply the theory of under-expanded free jets described in Section 2.1.1 to injection of gaseous fuel within engines, Ouelette and Hill adapted the analysis of Ewan and Moodie to develop a model for a conical jet sheet injected through a fast acting poppet valve [42]. They found a dramatic influence of the injector configuration and chamber geometry on the structure and propagation of the injected gas plume. In experiments with small poppet angles (approximately 10°, or 160° included angle) with no nearby walls, the jet penetrates much as predicted by the model. However, when the poppet angle is increased to 20 or 30° the axis of the jet tends to bend inwards significantly, greatly reducing the radial penetration; complete collapse of the jet sheet was observed in some cases. Simulations by Ouelette and Hill also found that a conical
jet sheet achieves approximately 30% less radial penetration than an equivalent set of 7 equally spaced round nozzle holes. Similar results were reported by Jennings and Jeske [39, 43], where they first observed higher mixing rates for a 12 hole injector relative to an 8 hole injector in a simulation of injection into unconfined surroundings, and later found that the 8 hole injector outperformed the 12 hole injector once jet interaction and boundary effects were accounted for.

As a jet penetrates into a surrounding environment it entrains air along its periphery. This entrainment leads to a local drop in pressure of the surroundings. In a large chamber this pressure drop causes more surrounding air to quickly be drawn into the jet periphery, but in the case where the amount of ambient gas is limited either by the presence of a bounding wall or an adjacent jet the pressure drop cannot be counteracted by additional entrainment and tends to become amplified. Eventually the pressure deficit becomes so large that it causes the jet to deflect to fill the void, resulting either in attachment to an adjacent surface or coalescence with a neighbouring jet, as in the collapsed conical jet sheets of Ouelette and Hill. When multiple jets coalesce or are attached to nearby walls the total surface area available for entrainment is reduced, lowering the rate of mixing. Local shear stresses are also increased under these conditions, reducing penetration rates. The conical jet sheet represents the extreme case of closely spaced individual jets merging as they propagate through the combustion chamber, which is why the simulations of Ouelette and Hill showed a decreased penetration rate. The potential for jet merging, and the negative impact this has on mixing and penetration rates led Jennings and Jeske to conclude that “the optimal number of holes is that which produces the largest number of separate plumes.” [43]. This guideline balances the need for small nozzle holes to improve each individual jet’s mixing properties with the need to avoid plume merging.

2.1.4 Effects of Jet Confinement

As noted above, the deflection of jets due to lack of ambient air for entrainment can result from confining walls as well as neighbouring jets. Ouelette and Hill noted this in their poppet valve experiments, where the jet sheet from the 10 and 20° poppets attached itself to the top wall of the chamber and propagated radially along the wall, rather than penetrating significantly in the axial direction [42]. This wall attachment reduced the radial penetration of the jet by about 10% [33]. Jet attachment to the cylinder head was also found in the simulations of Jennings and Jeske [43]. As in the case of plume
merging, attachment to adjacent walls reduces the available surface area for entrainment and increases shear forces along the remaining free jet surfaces. The combined effects of plume merging and attachment to confining walls can be seen in Figure 2.5, where the individual jets penetrate well into the chamber with some deflection towards the cylinder head, and Figure 2.6, where the plumes have merged and attached fully to the head. Note the much higher fuel concentrations and reduced penetration in Fig. 2.6.

To avoid attachment the volume between the jet and wall head must be increased. The available options for increasing the distance (and therefore volume) between the jet and the head are to increase the angle at which the jet is injected relative to the head, or increase the vertical distance, or “tip height,” between the head and the nozzle hole. Jennings and Jeske investigated these two parameters in their design of an optimized injector configuration [43]. They noted that increasing the tip height of the nozzle was a much more effective method of enhancing mixing than increasing injection angle, as increased angles reduce the radial penetration of the jet into the chamber. Since the volume of the chamber increases exponentially with radius, decreasing the penetration of the jet significantly reduces the volume of air available for entrainment. The net result is that the benefits obtained by avoiding attachment to the cylinder head are more than offset by the reduced penetration. The simulations of Jennings and Jeske in fact revealed that the jets that remained fully detached from the cylinder head due to increased injection angle showed the worst mixing. Increasing tip height, on the other hand, allows the jet
to remain free from attachment without requiring a reduction in radial penetration. An increased tip height with minimal injection angle provided the best overall performance in the simulations of Jennings and Jeske. This configuration is reproduced in the ERDL combustion bomb, with the jet tip (nozzle hole) positioned halfway between the two confining walls and injected parallel to them both. Laser shadowgraph measurements of the penetration rates of various gas jets in a bomb with identical geometry to that used in this study were reported by Brombacher [26] and fit to a correlation of the form presented in Equation 2.2, with $\Gamma = 2.7$. As shown in Figure 2.7, Brombacher found good agreement with the correlation constant for the early portion of the injection, though the observed penetration for the later stages was significantly less than that predicted by the correlation. This suggests that despite the positioning and orientation of the nozzle, jet attachment to adjacent walls still plays a role in inhibiting penetration within the ERDL combustion bomb.

2.2 Natural Gas Ignition Methods and Effects

2.2.1 Autoignition of Natural Gas

In a typical Diesel engine ignition is achieved by simply compressing the cylinder gas to a high temperature where the injected fuel autoignites due to heat transfer from
the surroundings. The simplicity of this method of ignition makes it preferable to any method requiring some form of ignition assist. The ignition performance of a Diesel fuel is normally measured using the Cetane number indicating the blend of two reference fuels (n-cetane and heptamethylnonane/HMN) that produces the same ignition delay under identical engine test conditions. However, as demonstrated by Siebers [44] the cetane number has limited usefulness in the case of many alternative fuels, particularly when the relationship between ignition delay and temperature differs from that of the reference fuel blends. As a result there is no convenient index by which to rate the autoignition characteristics of a fuel like natural gas; the different factors that influence autoignition in an NG engine must therefore be captured individually.

In the absence of a simple metric such as Cetane number, the ignition quality of methane and natural gas mixtures is determined by directly measuring or computing the ignition delay (the length of time between the start of injection and the onset of combustion). An “acceptable” ignition delay for a direct injection engine is typically taken to be 2 ms. The 2 ms standard is frequently cited when evaluating the performance of a fuel or ignition system [12, 45, 46, 47, 48]. As shown in Figure 2.8, combustion efficiency degrades rapidly as ignition delay is pushed beyond this threshold, as significant quantities of fuel begin to mix beyond flammability limits prior to ignition. Naturally, this will negatively impact overall engine efficiency as well.

Figure 2.7: Non-dimensional jet penetration in combustion bomb with $\Gamma = 2.7$ [26].

![Jet Penetration Distance Graph](image-url)
It should be noted that ignition delay can be defined in several different ways. The most frequently used definition is the pressure delay, where ignition is defined as the point where cylinder pressure first rises above some threshold value. The threshold typically corresponds to an increase of 3 standard deviations above the standard motored pressure trace [45], or to an increase indicating combustion of some small percentage of the total fuel mass [46, 47]. Luminous delay where ignition is measured by the first appearance of light from the flame has been used in several experiments, and agrees very well with pressure delay [45]. In the case of ignition modelling, the mass fraction of fuel burned can be computed directly and used to define the point of ignition independent of the chamber pressure; it has been suggested that this method may be preferable as it ignores variations in mixture heat capacity or pressure changes induced by endothermic reactions early in the ignition process [49]. As computation of this sort is unavailable in experimental work, one or more of the other methods must be employed. Unless otherwise noted the results discussed below use some form of pressure delay, often supported by calculation of luminous delay as well.

Experiments by Fraser et al. where methane and natural gas were injected into a high-temperature combustion bomb and allowed to autoignite found that ignition delay times of less than 2 ms required bulk gas temperatures of between 1200 and 1250K [45]. As shown in Figure 2.9, the necessary ignition temperatures and the relationship between temperature and ignition delay of the gaseous fuels is significantly different than methanol or liquid cetane. With density and mixture composition held constant, increasing bulk gas temperature caused ignition delay to decrease steadily, while varying the pressure from 5 to 55atm did not significantly change the observed delay times. The range of necessary temperatures determined by Fraser would correspond to compression ratios of 26:1 to 32:1, beyond those of current direct injection engines. Extending the work of Fraser to higher pressures, Naber et al. found a minor effect on ignition delay of increasing pressure [46, 47]; the effect was slightly less than first order for pure methane and became
less significant as higher hydrocarbons were added to the fuel mixture [47] (see below for further discussion of fuel composition). With the pressure effect accounted for, Naber et al. estimate that 2 ms ignition delays can be achieved at a temperature of approximately 1150K, slightly less than that reported by Fraser.

The reluctance of methane to autoignite is largely due to the chemical kinetics of the reaction; the CH\textsubscript{3} methyl radical that is the primary product of methane oxidation is quite slow to react, so there is a lack of highly active radicals available in the early stages of ignition to push the reaction forward. Specifically, other alkyl radicals tend to produce significant quantities of HO\textsubscript{2} which contributes to rapid radical generation, while CH\textsubscript{3} does not. Generating enough radicals to initiate the rapid phase of combustion thus requires higher temperatures. As mentioned above, the necessary temperatures of 1150 to 1200K are beyond the normal range of temperatures achievable in a high compression ratio Diesel engine. This suggests that for NG to be used as a fuel in a direct-injection engine some method must be found to either reduce the autoignition temperature of the fuel, or to generate a sufficiently high temperature through some method other than compression alone.

**2.2.2 Effect of Gas Composition and Additives**

Since it is largely a lack of radicals that inhibits the ignition of natural gas under normal engine operating conditions, one method of aiding the reaction is to include components
within the fuel that generate greater quantities of radicals when oxidized. Higher alkanes such as ethane, propane and butane, which occur naturally in most commercial NG mixtures, are obvious candidates to fill this role. Fraser et al. found a trend towards lower ignition delay times associated with increasing concentrations of ethane [45]. The effect was small and was only consistent at low pressures, but increasing ethane content from 0 to 10% still reduced the temperature needed for a 2 ms ignition delay from 1250K to 1200K. Similarly, in tests using hot surface ignition, Æsøy and Valland [50] found that pure methane took 2-3 times longer to ignite than a natural gas blend, and required temperature increases of approximately 100K in order to reach comparable delay times. Naber et al. tested a wider range of fuel blends and found significant reductions in ignition delay with increasing volumes of higher alkanes, as shown in Figure 2.10. Based on the results shown in the figure, Naber et al. estimated that a typical natural gas blend should be able to autoignite at temperatures of approximately 1100K, a further reduction from the original predictions by Fraser. However, since the composition of natural gas can vary significantly depending on the source, temperatures of at least 1250K would still be required to ensure reliable performance independently of the fuel blend used.

Fuel additives other than higher alkanes can also be used to increase radical production during the ignition period. For example, Agarwal and Assanis [49] found that even very small quantities of hydrogen peroxide significantly speed up ignition. The breakup
of a single peroxide molecule produces two highly active hydroxyl radicals by the reaction \( \text{H}_2\text{O}_2 + \text{M} \rightarrow \text{OH}^- + \text{OH}^- + \text{M} \). Thus, relatively small quantities of peroxide can generate very large quantities of hydroxyl radicals that will push the reaction forward. The effect is strong enough that no additional benefit was found when \( \text{H}_2\text{O}_2 \) content was increased from 5% to 10%. Æsøy and Valland tested additional additives such as ethylene and hydrogen in a combustion bomb with hot-surface ignition, and again found significant reductions in ignition delay time with additive concentrations of as little as 4% [48].

These results suggest that the use of fuel additives is a good potential solution for improving ignition delay in NG engines, but it is not without its difficulties. The composition of NG may vary with time, location, and distributor, so a design that was optimized for one blend might perform poorly, or not at all, if the mixture composition changed. Careful control of the mixture would ensure reliable operation, but would significantly increase the complexity and expense of gas distribution, limiting the advantage of the pre-existing infrastructure. The addition of higher alkanes is also limited by the storage pressure and fuel line temperature of the gas, as high pressures will lead to condensation of propane and butane within the fuel system if they are present in high concentrations.

### 2.2.3 Ignition Assist Methods

With practical difficulties limiting the use of fuel additives for reducing autoignition temperature, the other option is to provide some additional ignition source for a DI NG engine. Three methods of ignition assist have been previously implemented for this engine type. Spark ignition is a well established method of producing ignition within Otto cycle engines where a homogeneous charge of fuel and air is burned. In a direct injection engine, however, this method of ignition assist is much less reliable, as it requires that an ignitable mixture be present in the relatively small volume around the spark gap at the precise moment the spark is fired. The stratification of the fuel within the air combined with high turbulence levels and rapid mixing in a direct injection engine makes it very difficult to consistently control the composition of the fuel/air mixture at such a precise location. Spark plug longevity is also a significant concern in heavy duty engines. Thus, spark ignition is a very difficult method to implement for a direct injection engine.

A second, more popular method for producing ignition in a DI natural gas engine is known as (Diesel) pilot injection. In this method a small quantity of liquid diesel fuel is injected just prior to, or simultaneously with, the main natural gas injection. The
diesel fuel easily autoignites under most conditions, and the subsequent energy release is sufficient to ignite the natural gas. Pilot diesel injection has been explored by several researchers, and is capable of providing reliable and repeatable ignition with good engine performance in both emissions and power [51, 52]. The main drawback to pilot ignition is the requirement for two separate fuel tanks, both requiring monitoring and refuelling. The use of regular diesel in the form of a pilot spray also tends to slightly increase the emissions from a natural gas engine relative to pure natural gas for the reasons discussed in Chapter 1.

The third method of ignition assist employed in DI NG engines, and that used in this study, is hot surface ignition. In this method, a glow plug or other hot surface is placed in the combustion chamber and heats the surrounding volume enough to provide local ignition of the fuel, with the flame then propagating throughout the rest of the chamber. This method will be discussed further in the following section.

2.2.4 Hot Surface Ignition Assist

The most common hot surface used for ignition assistance in direct injection engines is the glow plug. Glow plugs have traditionally been employed in Diesel engines to provide ignition under cold start conditions. In the case of a fuel with poor autoignition characteristics such as natural gas the glow plug is used continually to provide a local source of ignition from which the combustion process can then propagate. The glow plug works simply by creating a localized area of elevated temperature. Introducing the fuel jet into this high temperature region allows the standard ignition mechanisms to take place, first within the immediate vicinity of the plug, and then propagating outwards to the rest of the jet/chamber.

As would be expected, increasing the glow plug surface temperature reduces the observed ignition delay, as shown in Figure 2.11. This is perfectly analogous to the observations above of decreasing ignition delay with increased bulk gas temperature for cases of unassisted compression ignition. The figure further reveals that the benefit diminishes as surface temperature is continually increased. This points to the fact that the ignition delay in this configuration consists of two separable phases: physical delay and chemical delay. Physical delay consists of the time required for the jet to penetrate to the glow plug and mix to within flammability limits (although these two processes are generally achieved simultaneously), plus any delay in the response of the injector.
Chemical delay consists of the time required after an ignitable mixture is present at the glow plug for a sufficient radical pool to form to carry the combustion reaction forward spontaneously. The physical delay thus imposes an absolute minimum on the overall ignition delay, while chemical delay can vary across a very wide range under the influence of the factors discussed above such as mixture temperature and pressure, fuel composition, etc. In Figure 2.11, as glow plug surface temperature is increased the chemical delay decreases and the overall ignition delay asymptotically approaches the physical delay time.

Also notable in Fig. 2.11 is the significantly lower ignition delays for the natural gas mixture relative to pure methane. This is a manifestation of the chemical delay time, and is again analogous to the influence of fuel additives discussed earlier. As noted by Æsøy and Valland [50, 48], fuel composition has a significant effect on ignition delay when using a hot surface. They note [48] that “ignition delay can vary by a factor of 2-3 for natural gas when compared to pure methane.” Much like the case of unassisted ignition, designs should attempt to accommodate fuel composition fluctuations as much as possible.

Thus, while the basics of glow plug assisted ignition are governed by the same factors as ignition via any other method there are additional complicating factors. The placement of the glow plug, the chamber geometry, and details of the flow field within the chamber influence both the temperature at the glow plug surface and the flammability of the gas.
Influence of Injection Angle

In addition to the expected effects of bulk gas and hot surface temperature noted above, work by Abate uncovered a strong influence of injector angle and glow plug shielding within the ERDL bomb. In the case of an unshielded glow plug there was a clear optimal injection angle that provided the fastest and most repeatable ignition [27], as shown in Figure 2.12. The sensitivity of ignition delay to injection angle, especially in the case of the unshielded glow plug, is due to jet proximity on one hand, and impingement of the jet on the plug on the other. For positive ("downstream") angles above the threshold an ignitable mixture is not presented at the glow plug surface, and the bulk of the fuel may actually be swept away from the plug by the swirling flow before ignition can take place. For negative injection angles the glow plug will likely be presented with an excessively rich portion of the jet, and may also be significantly cooled by impingement of the cold core region; either circumstance will appreciably delay ignition.

Influence of Glow Plug Shielding

In an attempt to overcome the difficulties associated with the bare glow plug Abate tested two glow plug shield designs, and found similar benefits resulting from both. An L-shaped shield was first used that largely eliminated swirl within the bomb chamber.
Reducing the swirl velocity significantly reduced the dependence on injection angle by both increasing the residence time of the jet near the glow plug and reducing convective cooling of the plug by the chamber gases (see Fig. 9.18 in [27]). Ignition delay was still higher in cases where the jet impinged directly on the glow plug or where the injection angle was so large that an only extremely lean mixture was presented at the plug’s location.

While the initial tests proved the potential for benefits from glow plug shielding, eliminating swirl within a real engine is not a practical solution, as it would negatively influence the subsequent mixing and combustion processes. As a result, a round shield was designed that protected only the area surrounding the glow plug. The shield incorporated a hole on the downstream side to allow fuel from the jet to enter, while preventing convective cooling due to swirling chamber gases. The round shield was also successful at increasing the range of angles over which acceptable ignition delay times could be achieved. There was, however, still an optimum injector angle, implying that the round shield design reintroduces the potential negative effects of high swirl on ignition delay, while retaining the benefit of decreased convective cooling and increased fuel residence time next to the glow plug (see Fig. 9.20 and related discussion in [27]).

The analysis of Abate was extended to a 2-hole injector by Fabbroni, who discovered that the round glow plug shield described above was not suitable for use with multiple jets. The size and placement of the shield was not a problem when only a single jet was injected, but when a second nozzle hole was added one of the jets tended to be deflected by the shield significantly away from its initial path, which seriously impacted the resulting combustion efficiency [28]. To overcome this a third shield was designed, which was significantly smaller and less intrusive than the previous two designs. The new shield is effectively a sheath that fits very close to the glow plug surface, with only a small gap between them. The small shield is also inclined in the same direction as the plug, minimizing the disruption of the surrounding flow. The small shield design was effective at improving the efficiency of combustion with multiple jets by allowing a clear path for flame propagation from one jet to the next. The shield retains the benefits of the previous design by minimizing convective heat transfer losses and increasing fuel mixture residence time near the plug surface. For practical purposes then, this shield design is considered optimal under the current combustion bomb chamber setup and will be used throughout the present study.

The above discussion indicates that while factors such as bulk gas temperature and
pressure, and glow plug surface temperature significantly influence ignition delay, an equal or greater effect can be attributed to specific elements of chamber design and geometry such as injection angle and glow plug shielding. While optimal chamber conditions and hot surface parameters can be deduced from the work cited here, a specific optimization process similar to that outlined above for the ERDL combustion bomb will be required for any effective combustion chamber design.

2.3 Elliptical Jets

2.3.1 Basic Jet Structure

It has long been understood that spreading and mixing characteristics differ for jets issuing from nozzles of “non-standard” shape relative to the more well-studied cases of planar and circular jets. The irregularities along the perimeter of these non-standard jets can lead to strong self-induction and instability of vortical structures, which in turn alter the development of the jet as it expands into its surroundings. In the case of elliptical jets of limited aspect ratio, the self-induction process has several interesting implications. ‘Limited aspect ratio’ in this context refers to jets where the ratio of major axis length to minor axis length is no more than approximately 3:1.

The clearest physical manifestation of the self-induction process in an elliptical jet is that of axis switching. This relatively complex process is caused by the fact that the transport velocity of a vortex structure is locally proportional to the structure’s curvature [53]. This means that a structure with variable curvature will have different downstream velocities at different points along its perimeter, and will thus deform out of its original plane. The initial process of deformation will further alter the curvature of the structure, feeding back into the local velocities. In the case of an isolated elliptical vortex ring, the velocity is thus initially higher at the major axis ends of the ring. The subsequent process of deformation and feedback ultimately results in a planar structure in which the major and minor axes have switched, as shown in Figure 2.13. A more detailed discussion of the self-induction and axis switching process can be found in Hussain & Husain [30]. Depending on the stability of the flow, this process of switching can repeat numerous times in the z-axis direction before the structure is ultimately broken down by mixing and shear.

The case of a continually fed jet with an elliptical cross-section is considerably more
complex than that of an isolated vortex ring, but studies have found that the same large scale processes discussed above can still have a significant influence. As identified and discussed by various authors [29, 30], as a jet of initially elliptical cross section develops, the switching of major and minor axes is readily observable, and can recur several times as the jet progresses downstream. Of greater interest, however, is that the process of axis switching in a jet has the additional effect of significantly enhancing ambient mass entrainment relative to comparable round jets. As the process of axis switching takes place the long axis of the jet is drawn inwards towards the centerline drawing in with it some quantity of ambient fluid. Similarly, as the minor axis is pushed outward it will tend to eject some quantity of the jet fluid into the surroundings. As Hussain & Husain describe it, the jet’s structures “act as pumping devices to mix ambient and core fluids [30].” The fact that this enhancement of mixing is achieved entirely passively, with no active control over the jet required, makes it especially appealing for practical applications such as fuel injection.

Ho & Gutmark studied the variation in mass entrainment rates for small aspect ratio elliptical jets [29], and found a “phenomenal” increase in the quantity of ambient fluid drawn into the jet. They report from 3 to 8 times more mass entrainment for the elliptical jets compared to standard round jets, occurring mainly within the initial portion of the jet. The majority of the additional mass is entrained into the jet near the minor axis, as ambient fluid is drawn in to this region by the axis switching process as described above. Figure 2.14 shows the greatly increased ratio of mass entrainment for an elliptical jet, particularly in the minor axis segments.

While the studies discussed above provide a good description of an incompressible
elliptical jet, any gaseous jet used in an engine application will be underexpanded, subject to significant compressibility effects. Schadow et al. [54] broadened earlier analyses to look at fully- and under-expanded elliptic jets to determine the extent of compressibility effects. In a study of a 3:1 aspect ratio elliptic jet, they noted the same axis switching phenomenon as in incompressible jets, with the additional observation that multiple shifts took place quite rapidly within the shock cells very near the nozzle exit. They further noted that the mixing rate of the elliptic jet was “significantly higher” than the comparable round jet, though this increase was not quantitatively stated.

### 2.3.2 Elliptical Nozzles for Fuel Injection

Based on the observed mixing enhancement and increased spreading rate described above, employing elliptical nozzle profiles would appear to be an ideal strategy for direct injection engines. In particular, an improved mixing rate will tend to decrease the relative proportion of (fuel rich) diffusion burning, which should in turn reduce the formation and persistence of soot. To this end, a few previous workers have studied the influence elliptical nozzle holes on engine performance and pollutant formation. A series of tests utilizing elliptical nozzles with both vertical and horizontal orientations in standard
Diesel engines of varying size measured the effect on emissions performance relative to a round nozzle reference case [55, 56, 57]. In general the results were not encouraging, as overall performance varied only slightly between the different nozzle configurations, with slight benefits in some areas (lower NO\textsubscript{x} was observed for some load conditions with the elliptical nozzles [56, 57]) which were offset by a loss of performance for other areas. Particularly discouraging was the fact that the elliptical nozzles tended to produce more smoke than the reference nozzle at most conditions [56], precisely the opposite of the desired effect.

However, while these results suggest that elliptical nozzles are not beneficial to engine emission performance, all of the above tests were carried out with liquid Diesel sprays, as opposed to gaseous jets. The fundamental studies discussed in Section 2.3.1 were all carried out using gaseous jets of one sort or another, so no firm inferences can be drawn about the likelihood of the axis switching process transferring to a liquid spray entering a gaseous surrounding. Even if one assumes that the same processes take place in the elliptical liquid spray the work of Ho & Gutmark [29] and Schadow et al. [54] suggests that much of the additional mass entrained into a gaseous elliptical jet is gained in the region quite close to the nozzle exit. In a liquid spray the near field region of the jet will tend to be considerably less broken up, presenting a significant barrier to any additional mixing that could otherwise be achieved.

Thus, while the initial studies of fuel injection via elliptical nozzle holes reflect unfavourably on the prospects for liquid spray injection, the fundamental mechanisms described in the literature should still be applicable to gaseous jets, making the use of elliptical holes in gas fuelled engines a viable area of investigation.
Chapter 3

Experimental Apparatus

The apparatus used for this study is based on the combustion bomb designed by Cheung [58], and has previously been used for studies by Abate [27] and Fabbroni [28]. Details of the design and setup of the apparatus can be found in those references, and will be only briefly summarized as necessary here.

3.1 CFR Research Engine and Combustion Bomb

The heart of the apparatus is a single-cylinder Co-operative Fuels Research (CFR) engine. The CFR is the ASTM certified standard engine for octane and cetane rating of automotive fuels. To facilitate octane testing the CFR cylinder’s position relative to the piston is adjustable. This controls the clearance volume, which in turn controls compression ratio. For use in the combustion bomb experiments the clearance volume of the engine has been minimized, and the compression ratio is boosted further by a plate bolted to the top of the piston. The engine is driven by an adjustable 3-phase DC motor that is capable of achieving speeds of up to 400 RPM.

An optically accessible combustion bomb was designed for this research to allow the use of visualization techniques to directly observe combustion within the chamber. The combustion bomb, shown in Figure 3.1 consists of a steel block that is bolted to the CFR engine, with gas exchange achieved by coupling the bomb block to the spark plug port. The front wall of the combustion bomb consists of a 0.875 inch thick window of A-1 optical grade quartz held in place with a retaining bracket. The rear wall consists of a removable steel block that contains various holes for placement of glow plugs, temperature probes, and the natural gas injector. Figure 3.1 shows all the main components of the
Chapter 3. Experimental Apparatus

Figure 3.1: Exploded view of main combustion bomb components.

The combustion bomb is designed to simulate the combustion bowl of a medium-sized diesel engine with significant swirl. The combustion chamber has a diameter of 50.8 mm (2 inches) and a depth of 12.7 mm (0.5 inch). In order to calculate the compression ratio the volume of the combustion bomb charging passage, clearance above the CFR piston, and any other passages must be accounted for. From Table 3.1 in Abate [27], a total clearance volume of 39.44 cm$^3$ was calculated along with a displaced volume of 611.73 cm$^3$, for a total compression ratio of 16.51:1.

Bulk gas swirl within the combustion bomb is generated by a tangentially oriented intake port (see Fig. 3.2). Previous attempts at determining the magnitude of the swirl and bomb velocity profiles have met with limited success, but through analysis of combustion photographs showing the flame front being convected by chamber swirl Abate [27] estimated an angular velocity of approximately 3622 RPM for operation of the CFR engine at 230 RPM.
3.1.1 Combustion Bomb Heating

In order to adequately exhaust burned gases and ensure an identical clean charge of fresh air each for each fired cycle the CFR engine is skip-fired, with combustion occurring only once every 5 cycles. As a result, the combustion bomb block does not maintain sufficiently high temperatures during testing from the heat release of combustion alone. Instead, a rope heater and an Omega CN9000A temperature controller are used to externally maintain the temperature of the block. The temperature controller uses a k-type thermocouple probe inserted into the rear combustion bomb block to monitor the temperature. The CN9000A also incorporates a PID controller that is capable of self-tuning to provide optimal response. The rope heater is held in place by metal plates fastened to three sides of the main block. The combustion bomb is then wrapped with fiberglass rope insulation to minimize heat loss to the surroundings.

3.1.2 Oil Heating and Circulation

An external pump is used to circulate oil through the CFR engine. A bypass valve attached to the pump is used to control the oil pressure, which is maintained between 50 & 70 psi. The oil circulation system incorporates a pressure-sensitive switch that prevents the DC motor from driving the engine if oil pressure is too low. An oil heater built into the crankcase is used to maintain the oil temperature at approximately 110°F.
3.1.3 Coolant Heating and Circulation

Coolant circulates through a jacket surrounding the cylinder of the CFR engine. Under normal operation water is used as a coolant to keep cylinder temperatures low. However, as in the case of the combustion bomb block, skip-firing the CFR engine keeps cylinder wall temperatures quite low during combustion bomb testing. This necessitates that the coolant be externally heated to maintain the desired cylinder wall temperature. A Haake N3 water bath is used to heat and circulate the coolant. The bath is filled with 100% ethylene glycol and the temperature is typically maintained at 150°C. Coolant temperature is monitored by a k-type thermocouple located in the return line, after circulating through the coolant jacket. Due to the placement of the thermocouple the indicated temperature is typically up to 5 degrees lower than the temperature set on the Haake N3 controller.

3.1.4 Intake Air Heating

Naturally, the temperature of air inducted into the combustion bomb will significantly affect peak cycle temperatures and pressures. A 1000W Chromalox heater is attached to the CFR engine’s intake pipe in order to provide preheating for intake air. The heater is controlled by a CAL 9900 temperature controller and a k-type thermocouple located just downstream of the heater, and can provide intake air temperatures of between 50 and 350°C. Similar to the combustion bomb block heater, the CAL 9900 contains a self-tuning PID controller. Due to the large range of temperatures and pressures encountered in the intake manifold, it is difficult to tune the controller to provide the best response across all conditions, but tuning at an intermediate setpoint generally provides a good rise time and avoids significant overshoot.

3.1.5 Intake Manifold Pressure Control

The intake of the CFR engine is connected to a compressed air line in order to duplicate the effects of supercharging, a very common feature of diesel engines. Intake pressure is set manually with a regulator, and is monitored with a GM MAP (Manifold Absolute Pressure) sensor and the LabView instrument SetP.vi (see Appendix B). Intake pressures of 0 to 30 psig are available from the regulator. A two-way valve is incorporated into the system to allow the MAP sensor to monitor either the intake or exhaust manifold.
pressure. During regular operation and data acquisition intake pressure is monitored, with only occasional checks of exhaust port pressure required.

### 3.2 Hot Surface Ignition Assist

#### 3.2.1 Glow Plug Position and Orientation

As discussed in Chapter 2, natural gas will not autoignite under standard diesel engine conditions, so some form of ignition assist is necessary to ensure reliable combustion. In the ERDL combustion bomb a glow plug is used to provide the necessary temperatures for ignition.

The design of the combustion bomb provides three ports at different radial distances from the nozzle where the glow plug can be installed. Early experiments and simulations found that the best performance in terms of reliable ignition times was obtained with the glow plug in the port nearest the injector, and this positioning of the plug was used exclusively in Abate [27] and Fabbroni [28]. This positioning provides a distance of 6.17 mm between the exit of the jet nozzle and the centre axis of the glow plug, as measured along the bomb’s rear wall. Due to space constraints, the glow plug is inclined at 20° towards the centre axis of the bomb, so the distance from the nozzle to the plug varies slightly across the depth of the chamber. A cross-section of the rear bomb block with the glow plug installed is shown in Figure 3.3.

#### 3.2.2 Glow Plug Temperature Control

The surface temperature of the glow plug was originally controlled using a DC pulse width modulation system designed by Cheung [59] based on work by Kong & Thring [60]. However, it was later found that this system was incapable of reliably maintaining the surface temperature at the desired setpoint. Due to the rapidly changing conditions in the combustion bomb and the comparatively slow response of the plug to changes...
in power, it was deemed unfeasible to use this type of control system in the current apparatus [28].

As a result of these difficulties with active surface temperature control, the experiments by Fabbroni were carried out by maintaining a constant power setting for the glow plug. The power level must be sufficient to keep the surface temperature high enough at the correct point in the cycle to provide reliable ignition. A high power transformer is used to provide constant voltage & current to the glow plug. Power is supplied to the transformer by an adjustable autotransformer that provides from 0 to 120 VAC. Voltage is measured directly by the data acquisition system, and current is monitored with a Fluke 80J-10 current shunt. These measurements allow real time calculation of resistance and power. Steady state resistance is correlated with surface temperature, and these relationships are determined by an off-line calibration (see Section 4.1.2).

3.2.3 Glow Plug Shielding

Conditions in the combustion bomb are such that convective heat transfer losses from the glow plug are highest at the point in the cycle that ignition is taking place. The high density, high velocity intake gases tend to “sweep away” the envelope of heated air around the glow plug, and impingement of the cold gas jet can further drastically reduce the plug’s surface temperature. As such, reliable ignition in under 2ms cannot be achieved without some form of glow plug shielding [27, 28]. As described in Section 2.2.4, the shield currently used in the ERDL bomb consists simply of a close fitting metal sheath that surrounds the plug and is attached to the back wall of the bomb using a small socket head cap screw. A hole is cut in the side of the shield to allow fuel and air to reach the glow plug surface, while preventing the jet from impinging directly on the plug.

3.3 Natural Gas Injector

The gas injector used in this study is the same as that used by Fabbroni [28], and is the most recent redesign of the concept first described in Green and Wallace [61]. A solenoid is used to provide rapid response times and a strong return spring ensures that minimum injection durations can be kept low, providing a wide range of fuel delivery options. The solenoid is controlled via a driver circuit connected to the primary data acquisition computer. Gas line pressure and temperature are monitored with a Schaevitz
3.3.1 Gas Injector Nozzle Design

For the present study, two new nozzles were designed. First, an elliptical nozzle with an aspect ratio of approximately 2:1 (major axis to minor axis) was required. The long axis of the hole was aligned axially within the combustion bomb chamber. The ‘vertical’ alignment of the hole was chosen due to the fact that elliptical jets have typically been observed to spread more quickly in the minor axis plane than the major axis plane. The major axis plane in this alignment is restricted by the combustion bomb walls, and jet attachment to those walls should be avoided as much as possible, so minimizing the spread in this direction will potentially reduce the negative effects of wall attachment detailed in Chapter 2.

It was desired that the elliptical hole have an area comparable to that of the round holes previously studied in the combustion bomb. However, the difficulty of machining an elliptical hole imposed a practical limit on just how small an area could be achieved. It was also found to be impractical to EDM drill a hole with a truly elliptical contour. Instead, an elongated hole with semicircular ends was used, as this could be more easily produced by EDM drilling. The elliptical nozzle that was produced was inspected with a high power optical microscope to determine its dimensions and to look for any major flaws in the contour. An image from the microscope is shown in Figure 3.4.

The figure reveals that there are no serious flaws in the contour of the hole. One end of the hole is slightly wider than the other but the difference is slight, and the difficulty of the EDM drilling process made it unlikely that a significantly better contour could be
achieved. The scale shown in Figure 3.4 was used to calculate an approximate nozzle hole area, revealing that the hole was significantly larger in total area than the small round holes employed previously. As a result a new round hole nozzle with a comparable area (#78 drill) had to be produced. The nozzle body design and hole location were unchanged from those used by Fabbroni [28]. After both nozzles were complete they were tested to characterize their mass flow properties, as described in Section 4.2.

3.4 Data Acquisition Systems

A total of three PCs are used to collect data and control the experimental equipment used with the combustion bomb system. Each CPU controls a distinct portion of the experiment, though all three are also linked in various ways, as described below.

3.4.1 Data Acquisition System 1

The bulk of the experimental data is collected by a 100MHz Pentium PC running Windows 95 and Labview 4.1. The computer is equipped with a National Instruments AT-MIO-16E-2 data acquisition board running in differential mode, which allows for up to 8 simultaneous double-ended inputs. The board also allows for various triggering, output, and counter functions. The board uses a 12-bit analog to digital converter, for a quantization error of 1 in 4096 (or approximately 0.02% of the voltage range). Board gain is set automatically by the software based on the upper and lower voltage limits selected for a given measurement. The available board ranges and gains are summarized in Table 3.1. Full specifications for the data acquisition board can be found in Appendix A of the User Manual [62].

The AT-MIO board is connected to an SC-2070 termination board, which is in turn connected to a series of BNC connectors that are used to provide easy access for sensor and control inputs and outputs. The termination board provides a jumper-selectable temperature sensor that serves as a cold junction reference for thermocouple measurements. Specifications for the SC-2070 are available in [63].

During typical experiments, the DAQ board samples five separate inputs: injection pressure, from the Schaevitz transducer; intake manifold pressure, from the MAP sensor; combustion bomb pressure, from the Kistler transducer; injector solenoid current pulses; and either the output from the fast flame ionization detector (FFID, see Section 3.6) or
Table 3.1: Data acquisition board ranges & gains.

<table>
<thead>
<tr>
<th>Board Gain</th>
<th>Input Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>±10V</td>
</tr>
<tr>
<td>1</td>
<td>±5V</td>
</tr>
<tr>
<td>2</td>
<td>±2.5V</td>
</tr>
<tr>
<td>5</td>
<td>±1V</td>
</tr>
<tr>
<td>10</td>
<td>±500mV</td>
</tr>
<tr>
<td>20</td>
<td>±250mV</td>
</tr>
<tr>
<td>50</td>
<td>±100mV</td>
</tr>
<tr>
<td>100</td>
<td>±50mV</td>
</tr>
</tbody>
</table>

combustion bomb temperature from a fine wire thermocouple, depending on the experiment being conducted. Each channel is scanned once per scan clock pulse (1800 times per revolution, see below). The channel scan clock is set to its maximum value of 500 000 Hz, which corresponds to an inter-channel delay of $2\mu$s, the settling time of the board. This effectively isolates the various inputs, preventing signal bleed between channels. In practice, it is still recommended that very low voltage inputs be scanned first, as some cross-talk between channels was observed by Fabbroni [28].

Timing of experiments is controlled by an AVL shaft encoder connected to the CFR engine. The encoder generates 1800 symmetric pulses per revolution (5 pulses per degree of crank angle), as well as a single trigger pulse each time the cylinder reaches TDC. All pulses are converted to TTL levels and then passed to the remaining circuitry. The Crank Degree Marker (CDM) pulses are sent directly to both the DAQ board and the camera timing subcircuit (see Section 3.5). At the DAQ board, the CDM pulses are routed to a programmable input pin identified in the software as the clock source; thus, the data acquisition VI advances one time step for every 0.2 degrees of crank revolution.

TDC trigger pulses are inverted from LOW to HIGH, and sent to an AND gate, which receives its second input from another encoder wheel that generates a pulse indicating the window for arrival at a non-firing TDC (ie. TDC at the end of the exhaust stroke). Thus, the AND gate produces a pulse precisely at TDC of the exhaust stroke every cycle. This pulse is sent to the DAQ system as well as the imaging circuitry. The input of this pulse to the DAQ board is specified within the software as the trigger to start sampling data. Thus, after the VI is started it waits for the first non-firing TDC pulse,
then samples the five input channels five times per crank angle degree until the specified length of acquisition has been reached, at which point the data in the buffer is written to a file.

### 3.4.2 Data Acquisition System 2

The second data acquisition system is used to monitor and control glow plug temperature during experiments. This system was initially much more complicated, when active control of glow plug temperature via pulse width modulation was being attempted. The glow plug monitoring process is much simpler now, and the separate system has been retained mainly out of convenience. A 90MHz 486DX PC running Labview 4.1 is equipped with a National Instruments AT-MIO-16E-10 DAQ board, connected to an SC-2070 termination board and BNC connectors, much like Data Acquisition System 1. The specifications of the two boards are quite similar, with the exception of a longer settling time for System 2 (10µs as opposed to 2µs). Specifications for the acquisition and termination boards are available in the relevant manuals [62, 63].

Glow plug current is measured using a Fluke 80J-10 current shunt, which provides outputs of 0-100 mV for currents of 0-10 A. Voltage is measured between the glow plug terminal and the nearest rear combustion bomb block bolt (ground). A voltage divider at the BNC connector reduces the voltage by half to avoid overloading the board, so voltages must be multiplied by 2 in the software. Resistance and power are calculated based on the voltage and current measurements obtained. For calibration of glow plugs a thermocouple is occasionally connected as an additional input, but this is not used during the course of regular data collection.

### 3.5 Image Acquisition System

A Princeton Instruments Intensified Charge Coupled Device (ICCD) is used to capture images of ignition and combustion events within the combustion bomb. A CCD consists of a photosensitive array which develops a charge proportional to the incident light intensity, forming a digital image. An Intensified CCD, as the name implies, intensifies the incoming light, making the ICCD suitable for low light applications. The greater intensity of the incoming light also reduces the minimum exposure times, so ICCD cameras can operate much faster than a device relying on ambient light intensity alone. However, care must
be taken when using an ICCD to avoid overexposure, as excessive light intensities can damage the array.

The ICCD system used in the present experiments is operated in “kinetics mode,” which allows for very high frame rates to be achieved. This mode is enabled by masking a portion of the CCD array, so that only a small portion is exposed to incident light. After an image is captured the charge can be very rapidly shifted into the masked portion of the array, much faster than it could be transferred by the controller into the software. With the camera used in these experiments the masked portion of the array is large enough to store 11 frames before the charge must be transferred to the software. Thus, by controlling the timing between frames a series of images spanning the combustion process can be generated for each unique injection event, with the slower data transfer process occurring between subsequent injections.

3.5.1 Light Filtering

Due to the intensity of light produced by the combustion in the chamber it is necessary to filter some of the incident light out in order to make image analysis possible. In particular, since the early stages of combustion emit primarily blue light, a filter that blocks incident light in all but a narrow band in the blue portion of the visible spectrum is used. Figure 3.5: Transmission data for blue ICCD camera Figure 3.5 shows the transmission data for the filter material that is used on the ICCD camera. It should be noted that white light produced in the later stages of combustion will radiate across the full visible spectrum, so some portion will fall within the blue filter band. This can make it difficult in practice to separate the different phases of the combustion process. The implications of this are discussed in greater detail later, when the combustion imaging results are presented.
3.5.2 Imaging Control Subcircuit

The ICCD camera system is controlled by a 90MHz Pentium EISA PC. Although quite old, this system is required to maintain hardware compatibility and allow the maximum possible data transfer rate when capturing images. The ICCD system parameters are set using Princeton Instruments’ WinView 1.4 software (see [64]).

The CCD array, intensifier and lens are mounted together on a tripod facing the combustion bomb’s window. The system uses two controllers: a PG-200 Programmable Pulse Generator that generates the high voltage pulses to activate the image intensifier, and an ST-138 Detector Controller that synchronizes between the camera and the computer, controlling the transfer of captured images, and also contains the controls for the camera’s cooling system. Details about the operation and capabilities of the camera, pulse generator, and controller can be found in [65], [66], and [67] respectively. Further details regarding the specific setup of the CCD system for this experiment are available in [59] and [58].

The precise timing of image acquisition during experiments is controlled by external circuitry linked to both the CCD system and the LabView VIs that control data collection. A schematic of the components and connections is shown in Figure 3.6. At the start of data acquisition, the DAQ board sets the voltage to 0V at the DAC0OUT pin, which connects to the Inhibit input on the PG-200. Thus, image acquisition can only take place while the VI is actively running and data is being collected, preventing possible damage to the ICCD array from overexposure.

The portion of the cycle to be captured is set using the Variable Window Data Acquisition (VWDA) box. This box has two inputs, and two adjustable counters that are set manually by the user. Input pulses from the shaft encoder are counted, and output pulses are generated for all inputs that fall between the lower and upper limits specified by the counters. The output pulses from the Variable Window are sent to the Adjustable Burst Fire (ABF). The ABF also has two adjustable counters, labeled FIRE and TOTAL, which are set by the user. The ABF divides the input pulses from the Variable Window, generating \( \frac{x}{y} \) output pulses for every \( y \) input pulses, where \( x \) and \( y \) are the values of FIRE and TOTAL, respectively. The VWDA and ABF are both reset by the TDC pulse each cycle.

Output pulses from the ABF are sent as one of two inputs to the Camera Trigger Controller (CTC). The second input to the CTC comes from an additional gating circuit,
Figure 3.6: Layout of image acquisition and control circuitry.

the ‘L3221 One-Shot’. The L3221 uses two inputs to generate a single high-level TTL pulse, lasting from the start of injection to the next non-firing TDC of each cycle. The CTC therefore generates output pulses for all inputs received from the ABF that fall within the window specified by the L3221. The CTC output pulses are sent to the External Trigger In connection on the PG-200, activating the high voltage pulses to the image intensifier on the CCD and capturing a single frame for each successive pulse, to a total of 11 frames. The ST-138 then downloads the contents of the CCD array and transfers it to the controlling PC.

The net result of all of the above circuitry is that images are captured during a user specified portion of a fired cycle, at a user-determined rate, and only during active data collection. This allows the captured images to be precisely correlated to the experimental data for the cycles that are captured.

3.6 Fast Flame Ionization Detector

The current study employs a Cambustion HFR400 fast flame ionization detector (FFID) to sample for the presence of hydrocarbons both within the combustion bomb and in the engine’s exhaust system. Specific details of the FFID’s setup for each experiment will be discussed later. The basic principles of the FFID’s operation are discussed below.
Flame ionization detectors work on the principle that when a hydrocarbon molecule is ‘burned,’ it releases a number of ions that is (almost) directly proportional to the number of carbon atoms it contains. By sampling a hydrocarbon-laden gas, combusting it in a carbonless hydrogen flame, and then measuring the quantity of ions released, the FID generates an output voltage that is directly proportional to the concentration of carbon atoms (from hydrocarbon sources) in the sample. If the average carbon number of the hydrocarbon being sampled is known, then the output voltage of the FID can be used to calculate the overall molecular concentration (or equivalence ratio) of the hydrocarbon.

A Fast FID (FFID) operates on this same basic principle, but uses modifications to the gas handling system to provide better frequency response. In a standard FID, the sampled gas is mixed with the hydrogen fuel well before combustion takes place. This allows the sample to mix and disperse within the sampling line before being burned and measured by the FID sensor. As a result, the FID cannot accurately reflect rapid concentration changes in the sample. The modifications made to create the Fast FID involve mixing the hydrogen fuel with the sample gas immediately prior to the nozzle where the fuel/sample is burned, and keeping the FID sensor relatively close to the sampling region. Both of these changes help to prevent dispersion of the sample prior to combustion. The only remaining source of sample dispersion is that which occurs within the transfer tube as the sample flows to the sensor head. By minimizing axial dispersion in the sample line, a very rapid response to changes in concentration can be achieved.

A schematic diagram of the FFID sampling head is shown in Figure 3.7. Sample flow within the FFID begins at the tip of the sampling probe. The gas mixture first flows up the Transfer Sample Line (1), which connects to the T-top assembly (2) in the constant pressure (CP) chamber (3). A portion of the sample then flows through the FID tube (4). At the end of the FID tube, very near the nozzle hole, the sample mixes with the hydrogen-air mixture flowing through the outer nozzle tube and is burned. The ions generated by the combustion are sensed by the collector electrode (5) in the FID chamber (6), generating the output signal.

For any sample gas to flow, the chambers in the assembly must be maintained at successively lower pressures (i.e., $P_{\text{sample}} > P_{\text{CP}} > P_{\text{FID}}$). A vacuum pump draws air from both the CP and FID chambers (7), while bleed air (8 & 9) supplied from the FID control unit allows the pressure to be set independently for the two chambers.

The CP chamber provides a buffer that allows the sample pressure to vary without impacting the flowrate of gas into the FID chamber. As sample pressure varies, the mass
flow through the transfer tube and into the T-top will change. With the end of the T-top tube open, excess mass will flow into the CP chamber keeping the pressure at the inlet of the FID tube constant (aside from any small change in dynamic pressure). A constant pressure in the T-top in turn keeps the mass flow into the FID chamber constant. The ability to maintain constant sample flow with large sample pressure variations is a necessity for engine applications as fluctuations will always exist in the region being sampled (whether in the combustion chamber or intake/exhaust manifold). An accurate FID output over time requires that sample flow rates do not vary significantly, as this would change the flux of ions at the collector even though the hydrocarbon concentration in the sample remains constant. Maintaining this “pressure independence” is a key factor in the operation of the FFID, and is discussed in Chapter 5.

This chapter described the layout and functionality of the equipment used to carry out the experiments that make up the body of this study. Before conducting these experiments however, it was first necessary to ensure that the operational characteristics of all the critical components were well understood, and that the relevant properties could be reliably predicted. The next chapter will summarize the preliminary work carried out with the aim of characterizing this equipment, establishing the base upon which the main results of this study will rely.
Chapter 4

Validation of Experimental Apparatus

As discussed in Chapter 3 the ERDL combustion bomb apparatus consists of several semi-independent systems working together. A successful analysis of experimental results requires that the characteristics of each of these systems be well understood. For the present study, a number of changes were made to the experimental setup that could potentially alter results from those obtained in the previous combustion bomb experiments. The following sections summarize the experiments carried out to validate the different elements of the apparatus, with special attention paid to those sections where modifications have been made.

4.1 Glow Plug Monitoring and Calibration

4.1.1 Measurement Strategy

In earlier combustion bomb experiments ([59, 26, 27]) a DC pulse-width modulation system was used to provide feedback control of glow plug core temperature. This system was developed by Cheung [59], modeled on a similar control setup described by Kong and Thring [60]. The pulse-width modulation system was ultimately abandoned as it was found to provide inadequate control of glow plug temperature at steady-state, owing to the relatively slow thermal response of the glow plug (see [28]). In lieu of the feedback system, Fabbroni used a constant power setting to maintain high glow plug surface temperatures. The temperature-resistance relationship obeyed by most glow plugs can be
used effectively to estimate surface temperature during combustion bomb experiments. Thus, calibration curves of temperature vs. resistance are required for all glow plugs to be used in testing. Calibration curves are obtained by powering the glow plug at a range of setpoints while recording current and voltage values as described in Section 3.2.2. Resistance and power values are both calculated from the simultaneous voltage and current measurements. Surface temperature is measured with a fine wire (0.003 in. diameter) exposed junction S-type thermocouple. A small loop of wire containing the junction is left protruding from the end of the ceramic insulator and is draped over the plug and allowed to hang under slight tension. This provides excellent thermal contact between the junction and the glow plug surface, and also evenly heats the leads of the thermocouple in the vicinity of the junction, minimizing conduction losses.

The response of the glow plug depends heavily on its surrounding environment: ambient air temperature, air movement, the contact area between plug and fixture, and the heat capacity and conductivity of the fixture will all affect the steady state temperature and resistance of the glow plug for a given applied voltage. Additionally, the length and placement of lead wires within the circuit will change the voltage at different positions, while the location of voltage probes will alter the indicated voltage. Changing the measured voltage will in turn change the calculated resistance and power and alter the steady state correlations. For the tests carried out here it was desired that the test conditions should match as closely as possible the conditions that would be present in the combustion bomb during testing. As a result, all present tests were carried out with the glow plug installed in the rear combustion bomb block, as shown previously in Figure 3.3. In this configuration there is significant contact area between the glow plug body and block, and a portion of the heating element itself also contacts the block. This has a significant impact on the temperatures achieved for a given set of conditions, as much of the energy generated by the plug is quickly conducted away and into the block. The impact of heat transfer to the bomb block on the subsequent tests is discussed further below.

### 4.1.2 Calibration Experiments

For this study a new set of glow plugs different from those used by Abate and Fabbroni had to be obtained. The original GM/Isuzu plug had been discontinued for some time and could no longer be purchased so a suitable aftermarket replacement had to be found.
Chapter 4. Validation of Experimental Apparatus

Figure 4.1: Temperature vs. resistance data for Denso DG-143 glow plugs. Solid lines indicate correlations from Fabbroni [28].

The new plug needed to be dimensionally identical with similar voltage/resistance/power characteristics (50W nominal power at 5V). Three different models of glow plug were tested as potential replacements: Wellman model W541, a 5V/50W plug; NGK model Y104, a 10.5V/50W plug; and Denso model DG-143, also a 5V/50W plug. Full details of the testing for the different plugs can be found in Appendix E. As stated in the appendix, the Denso DG-143 was ultimately selected for use in the combustion bomb for the present study. Figure 4.1 shows the temperature-resistance data gathered for a selection for the Denso plugs. The figure also includes sample correlations from the earlier calibrations by Fabbroni with the old GM plugs, showing that the Denso plugs achieve equal or greater surface temperatures at comparable resistances.

When used in the combustion bomb, the glow plug is operated at a constant power. Thus it is useful to have temperature-power correlations available in addition to the temperature-resistance curves just discussed. Figure 4.2 shows the curves of temperature vs. power obtained for both the Denso and NGK glow plugs. The temperature-power curves for both types of plug are well fit by logarithmic correlations. The correlations indicate that the Denso plugs provide significantly higher temperatures than the NGK plugs at the limiting operating conditions near and above 100W. In the previous study
with the combustion bomb all glow plugs were operated at a constant power of 100W. However, the results of the current tests indicate that constant power operation will not provide consistent performance as surface temperature variations of greater than 100K are possible for different plugs from the same manufacturer operating at equal power consumption. As a result, the glow plug operation strategy for the current tests will be to adjust the power of an individual plug in order to achieve an equivalent average ignition delay at a preselected operating condition. This should eliminate any variability in performance stemming from variability between glow plugs, helping to isolate the desired factors within the combustion bomb.

### 4.2 Gas Injector and Nozzle Testing

The performance of the natural gas injector used in the combustion bomb experiments has been thoroughly examined and reported on by previous workers (see [61, 59, 26, 27, 28]); these previous tests made it unnecessary to undertake a full study of the injector’s characteristics for the present work. However, two new nozzles were machined for the present study, both of which had to be fully characterized before being put into service. The results of the nozzle testing, as well as some additional tests to verify the injector’s
performance are reported here.

### 4.2.1 Injector Nozzle Mass Flow Testing

As described in Chapter 2, for all injector pressures above the critical pressure (approximately 1.84 times the maximum chamber pressure for methane), the flow from the injector nozzle will be choked. The choking will occur at the point between the reservoir and the chamber with the minimum cross-sectional area. The cross-sectional area combined with the reservoir pressure and speed of sound for the injected gas will determine the mass flow rate achieved from the nozzle. For reliable and predictable injector performance this minimum area needs to occur at the exit plane of the nozzle. This condition on the nozzle area is easily achieved with the gas injector, especially in cases where only a single nozzle hole is utilized. However, for at least some portion of each injection the flow will actually be choked upstream of the nozzle hole, between the injector’s pintle and seat. As the pintle lifts up off the seat or returns to it there will always be a brief period during which the area between the pintle and seat is less than that of the nozzle. During this time the flow rate from the nozzle will be less than its steady state value. Once the pintle has lifted sufficiently from the seat the choke point will move downstream to the nozzle exit. For sufficiently long injections the period of choking at the pintle will be very small compared to the overall injection length, and the injector will behave as though the full steady state flow rate was achieved for the entire injection. However, as injection durations become shorter, the period of reduced flow rate will account for more and more of the total injection and will cause the measured mass flow to deviate from the steady value. This becomes important if very short injection durations are going to be used during subsequent testing, as you cannot simply extrapolate the steady mass flow rate to arbitrarily short injections. Thus, for injection durations below the threshold where the steady flow assumption loses its applicability the flow rate must be determined from a specific test at that condition, rather than being inferred from correlations based on other data.

The mass flow rates of the different nozzles were determined with the same test procedures used by previous workers [26, 27, 28]. The basic test procedure is as follows.

1. The test cylinder was evacuated using a vacuum pump.

2. The evacuated cylinder was weighed using a digital scale with a 0.01 g resolution (in practice, the scale’s precision is ± 0.02 g).
3. The cylinder was connected to the injector.

4. The MassTest2.vi program was initiated, performing between 125 and 400 injections (as set by the user, depending on the injection pressure and duration).

5. The cylinder was re-sealed and disconnected from the injector.

6. The cylinder was weighed again, and the final mass recorded.

The average mass per injection for the given set of injector conditions is then determined by dividing the change in cylinder mass by the total number of injection events.

Mass flow tests were carried out for each nozzle to determine the injector’s response across various possible conditions. Injection durations between 1 and 10ms were tested, as well as injector pressures of between 7 and 11MPa. The results of these tests for both nozzles are shown in Figures 4.3 and 4.4.

With reference to the two figures it can be seen that for injection durations of 2 ms and up, the average mass per injection increases linearly with injection duration at all pressures. This indicates that above the 2 ms threshold both nozzles behave as though they are choked at the nozzle exit plane for the full duration of the injection. Thus, the slopes of the various linear trendlines indicate the choked mass flow rates for the nozzles at the various test pressures. For injection durations of less than 2ms the average mass per injection begins to deviate from the linear relation, as the portion of the injection where the flow is choked between the pintle and seat becomes more significant. The figures also reveal that the mass flow rate from the elliptical nozzle hole is slightly higher than that of the round hole nozzle at equal pressures.

For the injections of duration \( \geq 2 \) ms, a 2-variable linear regression was performed to quantify the combined influence of injection pressure and duration. Equations 4.1 and 4.2 show the resulting correlations for the round and elliptical nozzles.

\[
m_{\text{avg}}(\text{mg}) = 1.63t + 1.41P - 10.6 \quad (4.1)
\]

\[
m_{\text{avg}}(\text{mg}) = 1.83t + 1.50P - 10.4 \quad (4.2)
\]

In both equations \( t \) is the injection duration in ms and \( P \) is injector gas pressure in MPa.

4.2.2 Gas Injector Response

In addition to knowing the expected mass flow from the gas injector, it was also necessary to have an estimate of the delay time between the arrival of the injector pulse and
Figure 4.3: Round nozzle mass flow test results.

Figure 4.4: Elliptical nozzle mass flow test results.
when gas first starts to exit the nozzle. To determine this opening delay the injector was fired into a very small volume chamber where the pressure was monitored by the Kistler pressure transducer. The opening delay time of the injector is then defined as the time between the firing of the injector pulse, and the first point where the pressure transducer’s signal rises to $3\sigma$ above its steady value. Tests were carried out at a variety of gas pressures to check for any influence this might have on the injector’s response time. The results are shown in Figure 4.5. The figure shows that the opening delays are quite consistent at approximately 0.6 ms, and there is no apparent trend with injection pressure. The scanning rate of the DAQ board during these experiments imposed a quantization limit of 0.1 ms on the time measurements, so it is likely that the actual opening durations have even less variability than what is shown in the figure. In subsequent data analysis the opening delay of 0.6 ms will be subtracted from all times that are referenced to the start of injection in order to compensate for this offset.

### 4.3 Range of Motored Combustion Bomb Conditions

In order to interpret the data from the combustion bomb experiments it is critical that a complete and accurate understanding of the conditions within the chamber prior to and during combustion be available. As in previous studies, combustion bomb conditions were determined by measuring temperature and pressure during motored cycles while varying intake conditions throughout their full available ranges.
4.3.1 Combustion Bomb Pressure Measurement

Sample combustion bomb pressure traces for low, intermediate, and high intake pressure cases are shown in Figures 4.6, 4.7 and 4.8. Simulations by Cheng [69] have shown that pressure equalizes very rapidly between the combustion bomb and the cylinder. Thus, it is reasonable to treat the pressure histories obtained within the bomb as applicable to the combined combustion bomb/cylinder system.

The motored pressure traces obtained from the ERDL combustion bomb are generally quite straightforward. As seen in Figure 4.6, with no boost in intake pressure the pressure in the combustion bomb is slightly affected by both the intake and exhaust valve activations. At the start of the intake stroke the bomb pressure is higher than the intake pressure; when the intake valve opens the pressure drops slightly and then remains steady. Likewise, at the end of the expansion stroke the pressure in the combustion bomb has fallen slightly below that in the exhaust manifold, and pressure increases when the exhaust valve opens. In the cases with elevated intake pressures (Figures 4.7 and 4.8), the opposite behaviour is observed. When the intake valve is opened the measured pressure rapidly increases to the intake pressure, then remains steady during the rest of the intake stroke. At the end of the expansion stroke the system pressure has generally fallen to
nearly the pressure of the exhaust manifold, and only a slight drop is observed at the opening of the exhaust valve.

Peak pressure is typically obtained 3 to 6 degrees before TDC on the compression stroke, with the offset tending towards higher values as intake pressure is increased. The peak value offsets were very consistent, with a cycle-to-cycle standard deviation typically less than 0.5 CAD at a given condition. Likewise, the values of peak pressure were extremely consistent between cycles, with standard deviations of less than 1% of the peak value obtained at all conditions. There was no apparent effect of intake conditions on the cycle-to-cycle variability of peak pressure.

A multiple linear regression was performed of peak combustion bomb pressure on intake temperature, intake pressure and combustion bomb wall temperature. It was found that wall temperature did not increase the statistical significance of the regression, so peak pressure depends only on intake pressure and temperature as specified by Equation 4.3:

\[ P_{\text{peak}}[\text{kPa}] = 2763 - 1.63T_{\text{intake}} + 142P_{\text{intake}} \]  

(4.3)

with \( T_{\text{intake}} \) in °C and \( P_{\text{intake}} \) in psig. This correlation is used later to approximate the necessary intake conditions to produce desired peak conditions for firing tests.
Figure 4.8: Averaged motored combustion bomb pressure trace, high intake pressure case.

### 4.3.2 Combustion Bomb Temperature Measurement

The combustion bomb temperature traces obtained during motoring tests are considerably more complex than the pressure histories just discussed. To examine the various features of interest, a series of motored temperature histories are shown in Figure 4.9. The figure includes cases at low (a & b), intermediate (c & d), and high (e & f) intake pressures, with a low temperature (a, c & e) and high temperature (b, d & f) intake case at each pressure. As with the pressure histories, intake and exhaust valve openings and closings are denoted by vertical lines.

Cycle-to-cycle variations in peak temperature were minimal at all conditions, as in the case of peak pressure. The standard deviation of peak temperature averaged 0.5% of the peak value, and was less than 1% in all the tested cases. Peak temperatures occurred slightly earlier than peak pressures, generally between 5 and 9 CAD before TDC on the compression stroke. The location of the temperature peak was more variable, with a cycle-to-cycle standard deviation of between approximately 1 and 3 CAD at a given condition.

A key point to note about the combustion bomb temperature histories is that of
Figure 4.9: Motored bomb temperature traces: low and high intake temperature, and low, medium and high pressure cases.
the ‘step’ or ‘hiccup’ in the temperature trace that occurs at or near 300 CAD at all
conditions. This was discussed previously by Fabbroni [28], who suggested that it was
due to the onset of significant blowby losses at this point in the cycle. However, upon
examining the data of Fig. 4.9, this explanation does not appear to be adequate. There
is undoubtedly significant mass lost due to blowby during the compression stroke (see
Section 4.3.5), but for it to cause a drop in temperature such as that observed in Figure
4.9 the cylinder gases would actually have to be expanding. In fact, with the other
properties held constant, a loss of mass would actually cause an increase in temperature
simply by the ideal gas relation. Finally, the complete absence of any aberration in
the cylinder pressure traces indicates that blowby losses alone cannot account for the
observed temperature effect.

It is much more likely that the observed effect is due to the presence of tempera-
ture gradients within the combustion bomb/cylinder system. While pressure equilibrates
throughout the system very rapidly, without sustained vigorous mixing temperature gra-
dients could easily develop, particularly between the gases in the cylinder and gases in
the combustion bomb. After the intake valve opens and the pressure stabilizes early in
the intake stroke, there will be very little gas exchange between the cylinder and the
bomb. Since the combustion bomb walls are maintained at temperatures between 20
and 90°C higher than the cylinder walls, the cylinder gas will transfer significantly more
heat to the adjacent walls during the intake stroke. As the compression stroke begins,
there will at first be only a small quantity of gas being forced from the cylinder into
the combustion bomb, and the temperature will rise as expected. Then, as the piston
accelerates through the middle portion of the compression stroke, larger quantities of
gas will enter the combustion bomb, mix with the higher temperature gases there, and
briefly cause the indicated temperature to drop, or significantly slow its rise. The con-
tinued compression, combined with the more thorough mixing of the combustion bomb
and cylinder gases will then allow the temperature to resume its expected rise towards
TDC. This hypothesis is supported by noting that, in Figure 4.9, the magnitude of the
temperature aberration tends to increase with combustion bomb wall temperature. As
combustion bomb temperature increases, so does the difference between bomb temper-
ature and cylinder wall temperature. This will in turn magnify the difference in heat
transfer between the two zones, causing the cylinder gas temperature to be increasingly
lower than the combustion bomb gases early in the compression stroke.

Complete details of the gas exchange processes between the intake manifold, cylinder
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and combustion bomb cannot be deduced from temperature measurements taken within the combustion bomb alone. To obtain a more complete understanding, including the gas movement during the intake and exhaust strokes, simultaneous measurement of the temperatures in all 3 zones would be necessary for the full cycle. With the extremely small clearance volume currently maintained in the cylinder, temperature measurement in that zone would be extremely difficult, and would likely require some modification of the apparatus. However, even without such comprehensive data the combustion bomb temperature histories themselves are considered very reliable, especially towards TDC, which is precisely where the greatest accuracy is needed for the present work.

As was the case with the pressure measurements discussed above, a range of peak combustion bomb temperatures was obtained for varying intake conditions. However, peak temperature was not correlated to intake conditions alone. Rather, a multiple linear regression on intake temperature, combustion bomb wall temperature, and peak combustion bomb pressure was found to provide a good prediction of peak temperature. Peak pressure was selected rather than intake pressure for this correlation because it is measured directly during each cycle and will thus provide a good prediction of peak temperature even if, for example, a small leak from the system prevents the peak temperature from reaching the level predicted by intake conditions alone. The correlation determined for peak temperature is shown in Equation 4.4:

\[
T_{\text{peak}}[\text{K}] = 599 + 0.252 T_{\text{intake}} + 0.649 T_{\text{bomb}} - 0.0014 P_{\text{peak}}
\]

(4.4)

where intake and combustion bomb temperatures are in °C, and \( P_{\text{peak}} \) is in kPa.

4.3.3 Range of Peak Motored Combustion Bomb Conditions

With the two correlations of Equations 4.3 and 4.4, the overall range of available peak combustion bomb conditions can be determined, as shown in Figure 4.10. The regions outlined in the figure denote the range of peak conditions available at different combustion bomb wall temperatures. In practice there is a continuous relation between combustion bomb wall temperature and the peak conditions, and the outlined regions in the figure are simply illustrative examples. It should be noted that it may not be possible to obtain all of these conditions in practice, especially very near the extremes of the areas outlined in Figure 4.10. This is a consequence of the fact that as intake pressure and mass flow in the intake manifold increase, the intake air heater has a more difficult time reaching
high set points. Thus, at the highest available intake pressures there is a limit on the intake temperature that can be achieved.

After first generating the peak condition correlations seen above, it was noted that there were some significant differences from the ranges reported by both Abate [27] and Fabbroni [28]. The observed values of both peak pressure and peak temperature were lower than those reported in either of the previous combustion bomb studies. Peak temperatures, in particular, were significantly lower. In addition, the functional relationship between peak pressure and peak temperature had changed, with increases in peak pressure having minimal effects on peak temperature when other set points were held constant. Abate [27] observed that there was a significant dependance of peak combustion bomb temperature on peak pressure, with peak temperatures increasing significantly with pressure while intake and wall temperatures were held constant. Fabbroni [28] observed the same relationship, but found that the correlation was not as strong as that reported by Abate. This change was attributed to the presence of cross-talk between data acquisition channels, in which a portion of the high voltage pressure transducer signal ‘bled’ into the much lower voltage thermocouple signal, causing temperature values to be
overestimated. [28] This explained why temperatures would be overreported, and why the effect would increase with increasing pressure, as higher transducer voltages caused greater amounts of signal bleed. In the present work, however, the relationship between peak pressure and temperature has almost completely disappeared, with increasing peak pressures having almost no effect, or even a slightly negative effect, on peak temperature.

After thoroughly checking the data acquisition system for faults, and ensuring that all of the thermocouples and heaters attached to the engine were functioning properly, it was concluded that the observed temperature discrepancies did not result from an equipment failure. This is further confirmed by the individual temperature traces such as those in Figure 4.9, which show no unusual features. Short time-scale fluctuations in temperature are still well resolved, indicating that lag between the indicated and actual temperatures was not rounding off peak values and causing them to be underestimated. It was next suspected that mass loss due to leaks and blowby was causing the observed pressure/temperature deficit. After checking the combustion bomb and finding it relatively leak-free, it was decided that a more significant rebuild of the CFR engine was necessary to ensure that blowby was minimized as much as possible. To that end, a new set of piston rings was installed, and the CFR cylinder itself was replaced with a newly honed cylinder that met nominal tolerances. After breaking in the new piston rings and rebuilding the CFR engine the peak combustion bomb conditions were again measured and found to be largely unchanged (this latter set of data is that which is reported in Fig. 4.10). This strongly suggested that the observed values were in fact accurate, and that the discrepancy was not due to any faults within the apparatus or measurement system. This was further verified by a series of experiments carried out with a flow meter, as described in Section 4.3.4 below.

Considering the observed trends in the peak conditions plot, it is not immediately apparent that any relationship between pressure and temperature such as that suggested by Abate and Fabbroni should be expected. Assuming that the cylinder gases undergo an approximately polytropic compression, their properties should be governed by the formula of Equation 4.5.

$$\left( \frac{P_2}{P_1} \right)^{\frac{n-1}{n}} = \frac{T_2}{T_1} = \left( \frac{V_1}{V_2} \right)^{n-1}$$  \hspace{1cm} (4.5)

In the CFR engine, this relationship is fixed by the compression ratio, and thus varying \( P_1 \) while keeping \( T_1 \) constant should have no effect on \( T_2 \), at least in the absence of other factors. Naturally, heat transfer and mass loss will cause the actual compression process
to deviate from the ideal of Equation 4.5. However, as the cylinder wall temperature used in this study is the same as that used by both Abate and Fabbroni, and is verified by two independent measurements, there is no reason to suspect that the magnitude of heat transfer is different than in those cases. Also, as will be shown in Section 4.3.5, testing indicates that mass lost via blowby is not significantly higher in the present study than it was previously. Ultimately, the difference between the observed temperatures in the present work and those reported by Fabbroni would be best explained by some residual signal bleed that was not fully eliminated during Fabbroni’s experiments. Although no major changes were made to the data acquisition system for the present work that would readily explain the observed changes, all other potential explanations were eliminated by the verification testing.

4.3.4 Intake Flow Rate Testing

As discussed above, the peak condition testing in the combustion bomb raised some questions about the reliability of the pressure and temperature measurements. To provide an additional check on these values, a laminar flow meter was installed in the intake line for the CFR engine in order to quantify the volume and mass flow rates into the engine at various intake conditions. Due to the setup of the engine and the intake system, the flow meter had to be installed in such a way that measurements were only available for atmospheric intake conditions. However, this still provides a good set of additional data for verification, as discussed below.

The primary reason for conducting the flow meter measurements was to verify the accuracy of the pressure and temperature measurements within the combustion bomb. From the motored data sets of pressure and temperature, it is possible to use the ideal gas law to calculate a mass flow rate for the system, as well as the extent of blowby losses in the CFR engine. Mass flow is calculated simply by using the pressure and temperature data, along with the known volume of the cylinder/combustion bomb system as a function of crank angle to determine the total mass trapped in the cylinder at two points in the cycle: at an early point in the compression stroke, when temperature and pressure have equalized between the cylinder and bomb but significant blowby has not yet begun, and at the end of the exhaust stroke. The two calculated masses therefore represent the total charge mass trapped per cycle, and the mass of residual gas that remains in the system at the end of the cycle. Subtracting the residual from the total mass determines the
mass inducted into the system per cycle, which can then be combined with engine speed to determine a mass or volumetric flow rate at standard conditions. The laminar flow meter returns a value for the volumetric flow rate directly. By comparing the flow rates determined by the two methods the accuracy of both systems can be verified.

A TSI Model 2013 flowmeter, with a range of 0-25 SCFM, was used to measure the volumetric flow rate of air in the intake manifold. Since the flow rates encountered in the CFR engine were in practice quite low, it was found to be quite difficult to accurately tune the sensor’s linearized output so the non-linear voltage was measured instead, and converted to a flow rate in post-processing. Calibration points provided by the manufacturer were fit with an exponential correlation that was then used to interpolate flow rates from the measured voltages. The calibration curve and correlation used are included as Figure D.4 in Appendix D. The intake manifold flow rate was calculated from the flowmeter data using the mean output voltage of the averaged motored cycle trace. Measurements were taken at four intake temperatures and four engine speeds. The difference between the two methods of flow rate measurement was then calculated for each of the conditions. The results are shown in Figure 4.11. The figure clearly indicates that the difference between the two calculation methods starts out quite large, then decreases

Figure 4.11: Percent difference between volumetric flow rates calculated via alternative methods.
significantly as engine speed increases. It should be noted that in every case the flow rate calculated from the pressure and temperature data was the higher of the two values.

The tendency for the two values to converge at higher engine speeds can be attributed largely to the error induced in the flow meter by the highly transient nature of the intake flow. The flow meter used in these experiments monitors heat transfer and temperature with separate elements in order to determine flow rate. As discussed by Welch [70], this type of flow meter is prone to errors from several sources when measuring unsteady flows. Errors can result from velocity profile distortions and time lag of the sensing element, both of which will be exacerbated by fluctuations. For a single cylinder engine running at very low speeds, there will be large pulsations within the intake manifold, and this will in turn lead to significant error, as shown in Figure 4.11. It would thus be expected that, as the flow rate within the intake manifold increases and becomes more uniform, the error in the flowmeter’s output would decrease. Figures 4.12 and 4.13 show two representative flowmeter curves, at low and high speed engine operation, respectively. The two figures demonstrate that although significant pulsations are present at both low and high speeds, the flow in the high speed case does not drop as low in the early and late portions of the cycle. Thus, the flow through the meter is somewhat steadier in the high speed cases, and should help to reduce the magnitude of the errors present.

Clearly, even at the highest speeds achievable in the CFR engine, there is significant unsteadiness in the intake manifold flow. Thus, it is not surprising that the calculated flow rate values do not completely converge. However, the trend in the data suggests that
Figure 4.13: Intake flow rate trace from laminar flowmeter, high speed case.

if the engine speed was further increased, or if the intake flow could be stabilized (with a plenum, for example), the discrepancy between the observed values would continue to decrease. This solidifies the reliability of the pressure and temperature data collected within the combustion bomb, confirming that they provide a reasonable base upon which other calculations can be based.

4.3.5 Blowby Losses in the CFR Engine

As discussed above, the ideal compression process within the CFR engine would be a polytropic process following an index of approximately 1.3. However, due to a combination of heat transfer and mass lost due to blowby past the piston rings in the CFR cylinder, the actual compression process diverges significantly from this ideal. Figure 4.14 shows a typical pressure-volume trace for a motored cycle on logarithmic axes. The divergence between the ideal polytropic line and the actual pressure trace demonstrates the significance of the heat and mass losses. The figure also reveals that in the early portion of the compression, the polytropic index is quite closely followed, with significant divergence occurring more towards the end of the stroke.

As discussed in Section 4.3.4, flowmeter tests indicated the validity of calculating cylinder masses using the measured motored pressures and temperatures. Thus, by calculating the total (initial) trapped mass based on pressure and temperature early in the compression stroke, and the final mass based on values at the end of the compression stroke, the amount of mass lost due to blowby during each cycle can be estimated.
Calculating the blowby losses for the range of motored bomb data sets yielded values of between 20 and 35%. These values are quite high, but are in general agreement with those reported by Fabbroni [28].

There are a number of possible explanations for the magnitude of the blowby losses in the CFR cylinder. First, the fact that the engine is run at very low speeds in these experiments means that there will be a significantly longer time for blowby can occur. Assuming that the flow past the ring pack is choked this speed effect will be very significant since the instantaneous blowby rates will otherwise be comparable to those in a high speed engine. Tests with the CFR engine at speeds from 200 to 300 RPM revealed a slight tendency for blowby to decrease with speed, as shown in Figure 4.15. The observed effect was relatively small, with reductions in the total blowby of 1 or 2%.
across the range of tested speeds. However, if this effect was consistent up to speeds to 1500 RPM or more, the blowby losses would be reduced to levels much more consistent with typical engine operation.

An additional factor contributing to the significance of blowby losses in the CFR cylinder is lubricant starvation. Effective sealing between piston rings and cylinder walls relies on a thin layer of lubricating oil. The CFR engine uses a splash lubricating system, which may be less effective at continuously supplying oil to the cylinder walls at low speeds. Additionally, two oil control rings are used instead of one to prevent oil from reaching the combustion bomb and obscuring the quartz window. The extra oil ring not only cuts down on the oil film available for proper sealing, but also leaves one less regular ring past which cylinder gases need to pass before having a clear path into the crankcase. It would be expected then, that the CFR engine in its current configuration would suffer greater than usual blowby losses regardless of speed, and the low speeds serve simply to amplify the losses further.

This chapter summarized the tests carried out in characterizing and calibrating the equipment that will be used throughout this experimental program. Having presented and discussed the results of these validation tests, focus will now shift to the experiments that make up the main body of this research. The next chapter will examine the detailed setup and calibration procedure for the FFID, plus the data analysis and presentation methods that underlie the results. Following that, a chapter will be dedicated to the summary, presentation and discussion of the results of the FFID testing carried out in the ERDL combustion bomb.
Chapter 5

Fast FID Calibration and Data Analysis

The first major set of experiments in the present study used a fast flame ionization detector to measure local hydrocarbon concentrations within the gaseous fuel jet. The basic principles of the FFID’s operation were discussed in Section 3.6. Before beginning testing within the combustion bomb, however, a series of preliminary experiments were required to characterize the behaviour of the FFID under the specific test conditions within the ERDL combustion bomb. The main topics of interest were the sample transit time between the sampling probe and the FFID chamber, the pressure independence of the FFID, the response time of the FFID to step changes in concentration, and the functional form of the FFID’s response to a wide range of sample concentrations. Each of these issues are addressed in the sections that follow.

5.1 Sample Transit Time Analysis

The time required for sample gas to travel from the tip of the sampling probe to the FFID chamber will affect some aspects of the FFID’s response and the subsequent data analysis. First, a knowledge of the transit time is required in order to reference the FFID output signal back to ‘real time’ events within the combustion bomb (as with the opening delay of the gas injector). Second, the transit time of the sample can have an effect on the frequency response of the FFID when dealing with rapidly changing sample concentrations. As discussed in Crawford and Wallace [71], for a gas sample flowing through a pipe some mixing within the sample is unavoidable, particularly at the leading

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and trailing edges of the sample where concentration gradients are the greatest. If a significant amount of mixing is occurring between the leading edge of the sample gas and the ambient air in the tube, or within the sample itself, sharp gradients in concentration will become spatially dispersed within the tube, causing the resulting FID response to be dispersed temporally.

Two factors will significantly influence the sample transit time. The dimensions of the sample transfer tube is the first of these factors. Naturally, shorter sample transfer lines will produce shorter transit times. The limiting factor in this case is simply how close the FID head can be be placed to the sample location. In the present case, a tube of sufficient length to allow various probe positions within the combustion bomb was also necessary, but neither of these factors was a significant issue, since the ERDL combustion bomb is relatively freely accessible. The diameter of the transfer tube will also slightly influence transit time, as smaller diameter tubes will cause greater frictional losses and retard the flow. Larger sample transfer lines will reduce transit times, but will increase the mass flow rate into the T-top, which may cause the T-top exit to choke, endangering the pressure independence of the FID (see below).

The other factor affecting transit time is the pressure distribution throughout the FID sampling system. In most cases, especially when working with a high pressure sample, the flow within the sample transfer tube will be choked. This imposes a limiting value on the time required to reach the T-top. Changing temperatures within the combustion bomb will effect the speed of sound of the gas, but these changes will have a minimal effect on the overall transit time. In these high pressure sampling cases, the vacuum in the CP chamber will not significantly influence transit time. However, there is still a finite and potentially significant time required for the sample to flow through the T-top and FID tubes. Increasing the FID chamber vacuum (ΔP-FID) will significantly reduce this time. The diameter of the FID tube will have similar effects here as in the transfer line. Smaller diameters will produce increased friction, but also allow higher ΔP values with less chance of flame extinction.

The relevant dimensions of the FID setup chosen for this study are summarized in Table 5.1. For this configuration, estimates of the transit time for various sampling pressures and chamber vacuums can be provided by the Cambustion SATFLAP3 software. The software does not simulate highly transient flows well, but the values provide a rough estimate of what sort of response to expect. For the range of sample pressures and vacuums desired, the software indicates that transit times of between 1.6 and 2.0 ms should
Table 5.1: Fast FID component dimensions.

<table>
<thead>
<tr>
<th>Component</th>
<th>Diameter (in×1000)</th>
<th>Length (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sample Transfer Line</td>
<td>15</td>
<td>6.0</td>
</tr>
<tr>
<td>T-Top</td>
<td>44</td>
<td>0.970</td>
</tr>
<tr>
<td>FID Tube</td>
<td>9</td>
<td>0.750</td>
</tr>
</tbody>
</table>

Figure 5.1: FFID sample transit times for various test pressures.

In order to check the travel time of the FFID with the chosen setup, tests were carried out in which the gas injector was fired into a small volume, directly at the probe tip of the FFID. Accounting for the opening delay of the injector, the measured delay between the firing of the injector and the first response of the FFID signal provides a good estimate of the travel time. The results of the tests are shown in Figure 5.1. A data acquisition scan rate of 5000 Hz was used during these tests, providing a maximum temporal resolution of 0.2 ms.

The figure reveals that, for low test pressures, transit time can be quite high and variable. However, beyond a threshold pressure (approximately 1 MPa in these tests), the transit time reaches a steady value, and is unaffected by further pressure increases. This is to be expected, since transit time will be relatively fixed once the flow within the
sample tube becomes choked. The tests indicate that the selected FFID setup results in a transit time of approximately 1.4 ms, slightly lower than that predicted by the SATFLAP3 software.

### 5.2 FID Response Time Testing

As described above, there is inevitably some dispersion of the sample as it flows from the inlet of the sample line to the FID chamber. The extent of this dispersion determines the frequency response, or time constant of the FFID system (if a true step change in concentration could be presented at the exit of the FID tube, there is no mechanism that would prevent this from being reflected in the output signal). A fast frequency response is crucial in cases where large or rapid changes in sample concentration need to be resolved accurately. The concentration field of a transient gas jet is such a case. In order to provide a reasonably accurate representation of the concentration field the dispersion of the sample must be kept to a minimum.

In addition to estimating transit time as described above, Cambustion’s SATFLAP3 software also provides estimates of response time for different FFID configurations. For the setup of Table 5.1 the estimated response time is 0.2 ms. However, as outlined in Crawford and Wallace [71], transient effects can change the extent of sample dispersion when performing in-cylinder sampling near TDC. A rapidly accelerating flow may remain laminar well beyond the usual critical Reynolds’s number, and the larger concentration gradients present at the leading edge of a laminar pipe flow will increase the rate of dispersion, degrading the response time. As a result, some further confirmation of the response of the FFID was desired.

To determine the response time of the FFID, tests were carried out using a modified solenoid valve to provide a high pressure step change in gas concentration to the sample head. A fast acting solenoid valve was modified by drilling a small hole immediately downstream of the valve seat. Fitting an FFID sample transfer tube to this passage and sealing the usual valve outlet provides as short and direct a path as possible between the upstream gas and the downstream sampling head. Minimizing the distance between the gas source and sampling head minimizes the sample dispersion, providing as close an estimate to a true step change in concentration as possible. The use of this apparatus is discussed in detail in Liu and Wallace [72], where it was used for a similar purpose. In the present tests, however, it was found that the dispersion in the sample line between the
solenoid valve and FFID head could not be reduced sufficiently to provide a good estimate of what would be encountered in the combustion bomb. The sample line fitted to the solenoid valve was longer and had a larger diameter than the probe used in the combustion bomb, and a true step change could not be preserved in the sample. Initial tests measuring jet concentrations within the combustion bomb also showed much faster responses than tests with the solenoid valve. It was also noted that the jets would not be presenting step changes, but would rather be changing on approximately the same time scale as the estimated response of the FFID. Taken together, these considerations make the demands on FFID response time somewhat less stringent than in previous applications, where step changes due to induction of premixed fuel-air charges were being measured at very high engine speeds, and signal peak degradation was therefore of primary concern. In the tests described in Chapter 6, a “steady” period was generally encountered during the main portion of fuel injection where concentration did not change significantly over several sampling steps. This indicated that the response of the FFID was fast enough that peak degradation was not a concern.

5.3 FID Output Calibration Curves

The response of the FFID is slightly unusual in that it passes through two distinct linear response regions as the hydrocarbon concentration is increased. A typical response curve, covering the full range of concentrations which the FFID can resolve, is shown in Figure 5.2.

At low concentrations ($y \leq 13000$ ppm C), there is a purely linear response from the FFID with an intercept of zero. Within the range $13000 \leq y \leq 30000$, the response transitions from the initial linear region with zero intercept, to a second linear region with a non-zero intercept and smaller slope. These two initial regions are shown in greater detail in Figure 5.3. For $y \geq 30000$, the response transitions fully into the second region and continues out to concentrations of approximately $200000$ ppm C. Beyond this point the response becomes exponential and highly erratic, so it represents the practical upper limit that can be resolved by the FFID.

It should be noted that the specific calibration curves of Figures 5.2 and 5.3 apply only to one configuration of the FFID. Changing tube sizes and pressure differentials will change the flow rate of hydrocarbons to the FID chamber, thereby changing the voltage output. However, the form of the response, with the two distinct linear regions, and the
Figure 5.2: Full-range calibration curve for FFID.

Figure 5.3: FFID response for low concentrations, with output voltage normalized to unity gain.
concentrations at which the transitions between regions occur, are not changed by these factors. So, when sampling from regions with relatively low hydrocarbon concentrations (ie. \( y \leq 13000 \text{ ppm C} \)), a single calibration is sufficient to describe the FFID’s output. For samples where concentrations will reach or exceed 30000 ppm C, two separate calibration curves will be required. During combustion bomb experiments the FFID was calibrated before each round of testing by providing a calibrated gas mixture of high purity methane and air, prepared with two Matheson rotameters. The calibration curves used for the rotameters are included in Appendix D.

5.4 FFID Data Reduction and Presentation

The jet concentration measurements taken with the FFID in the combustion bomb consist of numerous individual data sets, each representing the time history of concentration at a particular location in the jet field. The FFID probe was inserted through the thermocouple port in the combustion bomb block, and its axial position was easily adjustable. Different angular positions were obtained by rotating the natural gas injector relative to the probe. For each test condition, the FFID probe was used to sample at four axial positions, and up to 10 angular positions. Thus, between 30 and 40 data sets had to be subsequently combined to obtain a time history of the concentration field. The steps carried out in summarizing and presenting the results presented in the following chapter are summarized below.

First, the FFID output was averaged over the 10 fired cycles sampled in each data set. This averaged voltage trace was then converted to concentration based on the appropriate calibration data. Concentrations were then converted to values of equivalence ratio. The relevant portion of each cycle was then extracted, and the data from different locations were aligned by their time indices. This produced a spatial matrix of equivalence ratio values for each time step. Figure 5.4 shows a surface plot of a typical set of data. \(^1\) Each point on the surface represents the average FFID output over 10 cycles, at a particular time step, and at a single distance-angle position.

The data shown in the figure are somewhat rough, so the values were next interpolated over a finer grid in order to produce a smoother final plot. Care was taken to ensure

\(^1\)Note that the \( x \)-axis values in the figure are angular positions, even though they are displayed on a linear axis. Thus, the plot does not accurately display the shape of the jet, but merely illustrates the general form of the data.
that the interpolation process did not significantly alter the shape or distribution of the subsequent contour plots (see below). Figure 5.5 shows the same data set as above after interpolation. Note that the $X$ values have been converted to radians, although they are still displayed on a linear axis.

From the surface plot of Figure 5.5 contour plots can be easily extracted. Plots will typically be presented with a limited number of contours at important equivalence ratios displayed. Since the flammability limits of natural gas fall roughly within the 0.5 to 2.0 range of equivalence ratios, these values will frequently be used as inner and outer boundaries. Figure 5.6 shows a basic contour plot with four levels displayed, drawn from the data of Figs. 5.4 and 5.5.

5.4.1 FFID Data Uncertainty and Error Analysis

As mentioned above, the individual data points such as those shown in the surface plot of Figure 5.4 are generated by averaging the FFID output over 10 injection events. To estimate the uncertainty associated with these data points, standard deviations were calculated along with the mean for some representative data sets. Figure 5.7 summarizes the results for the same data set depicted in Fig. 5.4 by taking lateral slices through the jet field at the four sampling positions, and plotting concentration and standard deviation.
Figure 5.5: Interpolated equivalence ratio surface plot.

Figure 5.6: Contour plot with equivalence ratios of 0.5, 1.0, 1.5 and 2.0.
Figure 5.7: Standard deviations of representative FFID data. Figures are for axial position relative to nozzle exit: (a) 1.0 in., (b) 0.75 in., (c) 0.5 in., (d) 0.25 in. Note changing x-axis scaling for different axial positions.

The figure reveals several important facts. First, there is very little cycle-to-cycle variability in the data in the central portion of the jet. This will be extremely important in analyzing results later since this is the region containing the highest concentrations, and is also where the most important comparisons between the two nozzles will be drawn. Secondly, the variance of the data increases quite suddenly and dramatically as you move towards the periphery of the jet and the measured concentrations drop off. This is not
wholly unexpected, as this is the mixing flow region of the jet where the shear layer between injected fuel and ambient air develops. Significant turbulence and mixing in this region will lead to a highly variable mixture being presented at the sampling location. The physical basis of the variation is further confirmed by the fact that in Fig. 5.7(b), (c), and (d) there is a peak in the standard deviation at the point where the concentration drops nearly to zero. Beyond this point there is little or no fuel being presented to the probe, and the variability of the signal again starts to drop off.

The overall implication of the pattern of variability in the data presented above is that the FFID itself is quite reliable, and does not introduce a significant amount of variation into the data. Rather, the majority of the uncertainty in the concentration measurements will have a physical basis in the variability of the fuel-air mixing process within the combustion bomb. In the locations where more consistent mixtures are presented to the sampling probe the spread of the data is small enough to give a high level of confidence to the comparisons between different data sets.
Chapter 6

FFID Jet Testing Results and Discussion

6.1 Test Conditions and Setup

Jet concentration fields were measured for both the round and elliptical nozzles at a total of 4 different test conditions. Short injection durations typical of those that would be used in a practical engine application were tested at low and high peak bulk gas densities. A longer injection duration was then tested at the high peak density condition. Finally, tests were conducted at the high peak density condition using a calibrated mixture of 20 000 ppm CH$_4$ in N$_2$, in an attempt to resolve a larger range of jet concentrations. The test conditions are summarized in Table 6.1.

As discussed briefly in Section 5.4, measurements of concentration were taken by inserting the FFID probe tip into the combustion bomb chamber through the thermocouple port, along the dashed line shown in Figure 6.1. Injection angles for these tests

<table>
<thead>
<tr>
<th>Test</th>
<th>Nozzle</th>
<th>Intake Temp. (°C)</th>
<th>Intake Pressure (psig)</th>
<th>Bomb Wall Temp. (°C)</th>
<th>Peak Gas Density (kg/m$^3$)</th>
<th>Injection Duration (CAD)</th>
<th>Injection Pressure (MPa)</th>
<th>Injection Angles</th>
<th>Fuel Type</th>
</tr>
</thead>
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<tr>
<td>A1</td>
<td>Round</td>
<td>300</td>
<td>0</td>
<td>180</td>
<td>10.05</td>
<td>1.4</td>
<td>11.1</td>
<td>-25 to 20</td>
<td>Natural Gas</td>
</tr>
<tr>
<td></td>
<td>Elliptical</td>
<td>300</td>
<td>0</td>
<td>180</td>
<td>10.05</td>
<td>1.4</td>
<td>10.2</td>
<td>-25 to 20</td>
<td>Natural Gas</td>
</tr>
<tr>
<td>A2</td>
<td>Round</td>
<td>220</td>
<td>15</td>
<td>200</td>
<td>20.24</td>
<td>1.4</td>
<td>11.2</td>
<td>-25 to 15</td>
<td>Natural Gas</td>
</tr>
<tr>
<td></td>
<td>Elliptical</td>
<td>220</td>
<td>15</td>
<td>200</td>
<td>20.24</td>
<td>1.4</td>
<td>10.3</td>
<td>-25 to 15</td>
<td>Natural Gas</td>
</tr>
<tr>
<td>A3</td>
<td>Round</td>
<td>220</td>
<td>15</td>
<td>200</td>
<td>20.24</td>
<td>3</td>
<td>11.2</td>
<td>-25 to 15</td>
<td>Natural Gas</td>
</tr>
<tr>
<td></td>
<td>Elliptical</td>
<td>220</td>
<td>15</td>
<td>200</td>
<td>20.24</td>
<td>3</td>
<td>10.3</td>
<td>-35 to 15</td>
<td>Natural Gas</td>
</tr>
<tr>
<td>A4</td>
<td>Round</td>
<td>220</td>
<td>15</td>
<td>200</td>
<td>20.24</td>
<td>2</td>
<td>11.2</td>
<td>-25 to 15</td>
<td>Natural Gas</td>
</tr>
<tr>
<td></td>
<td>Elliptical</td>
<td>220</td>
<td>15</td>
<td>200</td>
<td>20.24</td>
<td>2</td>
<td>10.3</td>
<td>-25 to 15</td>
<td>20000ppm CH$_4$ in N$_2$</td>
</tr>
</tbody>
</table>

Table 6.1: Test conditions for FFID jet concentration measurements.
are measured relative to the FFID probe. Aligning the probe with the centreline of the jet is taken to be 0°. Rotation of the injector in the direction of bulk gas swirl is taken as positive, and rotation opposite the swirl direction is taken as negative. Measurements were taken at intervals of 5° of injector rotation. Positive rotation was cut off at the point where the jet periphery was no longer being presented to the probe, and negative rotation was limited at a point where the majority of the concentration field had been resolved. For each angular position, measurements were obtained at 4 axial locations (0.25, 0.5, 0.75 and 1.0 in. from the injector nozzle).

In order to ensure comparable results, the mass and momentum flow rates from the two nozzles were to be kept as close as possible. Equal mass flow rates would be achieved by balancing Equation 6.1,

$$\rho_r A_r V_r = \rho_e A_e V_e$$

where \(\rho\) is fuel density, \(A\) is nozzle orifice area, and \(V\) is the gas velocity at the nozzle exit, with the subscripts \(r\) and \(e\) denoting the round and elliptical nozzles, respectively. For equal fuel temperatures, the values of \(V_r\) and \(V_e\) will be fixed and equal at the speed of sound. Thus, since \(A_e\) is slightly larger than \(A_r\), it was necessary to make \(\rho_r\) larger than \(\rho_e\) by increasing the injection pressure in order to obtain equal mass flows. Since momentum flow rate is given by \(\dot{M} = \dot{m} V\), and \(V\) is fixed as just described, maintaining equal mass flow rates ensures equal momentum flow rates automatically.
6.2 Natural Gas - Low Density Case

The first set of tests described here is for injection of natural gas into the combustion bomb with a relatively low peak gas density. The chosen injection duration was 1.4 CAD, or approximately 1 ms.

Using the methods described in Chapter 5, surface maps of equivalence ratio were generated for each time step of interest during the main portion of injection, and contour plots were then extracted from this data. Figure 6.2 shows the concentration fields measured for each nozzle. The plots show contours for equivalence ratios between $\phi = 0.25$ and $\phi = 2.0$ (concentrations lower than 0.25 are not shown, while all concentrations higher than 2.0 are depicted by a single colour region).

First, a few general points are worth noting about the data in the figure. The approximate timing for each frame is determined by taking the crank angle where the FFID signal first becomes appreciable, then accounting for the estimated sample travel time ($\sim 1.4$ ms), as well as the injector opening delay and jet travel time ($\sim 0.6$ ms, combined). This yields an estimate of 2 ms, or 2.6 to 2.8 CAD, for the total expected lag. In this case, the first signal is observed at approximately 358.6 CAD, so the ‘actual’ crank angle at which the contours from the first frame would have developed is approximately 356.0. Subsequent frames advance at the sampling interval of 0.2 CAD. Note that after 1.4 CAD (a time equivalent to the injection duration), starting at the 8th frame of Figure 6.2, the jet contours begin to recede, signalling the end of the injection. This indicates that sample dispersion within the FFID is not significantly altering the signal duration.

The overall development process in both jets is largely the same. The axis of both jets is bent slightly in the direction of the bulk gas swirl. The deflection is not very great in either case, which is to be expected as the momentum of the jets is much higher than that of the bulk gas, especially at this test condition. The momentum difference is especially important during the main injection period while the jet is still being actively fed with momentum from the nozzle. Also visible in the figure is the clear spreading of the jet as it impinges on the combustion bomb wall. A high degree of interaction between the jet and the wall is generally undesirable, as the wall is an extinction zone for combustion. Thus, any fuel near the wall will not be burned and will contribute to exhaust HC emissions. As a result, if a jet contains large areas of high fuel concentration, the impingement of that fuel onto the wall will tend to increase hydrocarbon emissions, in addition to the increased tendency for soot production associated with rich fuel mixtures.
Figure 6.2: Equivalence ratio contours for round and elliptical nozzles, low peak gas density case.
Figure 6.3: $\phi = 2.0$ contours for round and elliptical jets, low peak density case.

The importance of this becomes clear when comparing the extent of the fuel rich jet cores, as discussed below.

The most significant difference noted between the two jets in Figure 6.2 is in the extent of the fuel rich ($\phi \geq 2.0$) core region. Figure 6.3 shows the extent of the $\phi = 2.0$ contour at selected times steps for the early, middle and late portions of the injection. As shown in the figure, there is a significant difference in the extent of the rich regions of the two jets at this condition. In the early stages of injection, the core of the elliptical jet is smaller both in width and in penetration through the chamber. After the jets reach steady state in the mid-portion of the injection, the width of the elliptical jet core is nearly the same as that of the round jet, as shown in the 357 CAD frame of Figure 6.3, which depicts the maximum extent of both jets. However, the penetration of the rich core region is still less for the elliptical jet, as evidenced by the fact that the $\phi = 2.0$ contour never reaches the combustion bomb wall. Compare this to the round jet, where the rich core region reaches the wall and then begins to spread along it as the jet reaches its maximum extent. Also evident in Figure 6.3 is the more rapid destruction of the rich core of the elliptical jet after the gas flow from the injector is cut off. This suggests that the fuel within the $\phi \geq 2$ contour of the round jet is reaching higher concentrations than in the elliptical jet during the main injection, allowing the elliptical core to mix down to within flammability limits more quickly after fuel stops being injected. Unfortunately the response of the FFID makes precise measurement of these very high concentrations impossible with pure natural gas. This will be addressed later in the tests conducted.
with a more dilute calibrated gas mixture.

These initial results suggest that even though the overall extent of the two jets is similar, the fuel rich core of the elliptical jet has been diminished somewhat relative to the round jet. As noted in Chapter 2, the additional air entrained into an elliptical jet is largely gained within the core region, where the axis switching process is taking place; this is consistent with the diminished size of the core region seen in Figure 6.2. A smaller rich core is ideal for combustion, as it will reduce both the amount of fuel in the wall region, and the amount of rich diffusion burning that tends to produce soot.

6.3 Natural Gas - High Density Case

The main results for the second test case, with a higher peak gas density in the combustion bomb, are shown in Figure 6.4. As with the low density case, contours are included for equivalence ratios in the range of $0.25 \leq \phi \leq 2.0$ in each case.

The figure reveals that the development of the jets differs somewhat from the low density case. The progress of the developing plume towards the wall takes slightly longer in the high density case; this is not unexpected, as the higher density chamber gas will exert a greater retarding influence on the fuel. There is also a notably greater deflection and deformation of the fuel jet in the direction of swirl in this case. As the density of the chamber gas increases, the momentum of the swirling flow will also increase and this will in turn add to the influence of the swirling cross-flow on the jets.

As an example of the influence of the swirling flow, the spreading of the jet in the upstream direction after it impinges on the combustion bomb chamber wall is notably less for this high density case. While in the low density case of Figure 6.2 the jet spread nearly the same distance at the wall in both directions, the downstream spreading is amplified and the upstream spreading is nearly eliminated in Fig. 6.4. The effects of this jet deflection at higher chamber densities was later noted during firing tests, as the angle between the jet and glow plug had to be reduced as intake pressure was increased in order to prevent the fuel from being swept away from the hot surface and degrading ignition properties.

The distribution of the fuel within the jet for this case is also significantly different, with much smaller rich cores for both jets relative to the low density case. As noted in Chapter 2 the mixing rate for gas jets is largely determined by the turbulence generated by the jet itself, rather than ambient density or intensity of swirl, since the momentum
Figure 6.4: Equivalence ratio contours for round and elliptical nozzles, high peak gas density case.
Chapter 6. FFID Jet Testing Results and Discussion

Figure 6.5: $\phi = 2.0$ contours for round and elliptical jets, high peak density case.

The ratio of the gas jet to ambient flow remains very large in either case, so similar mixing rates would be expected in both the low and high density chamber conditions. Increasing the density of entrained fuel while maintaining comparable turbulence levels will thus increase the total mass of air entrained into the jet which contains the same fuel mass in both cases, thereby reducing local concentrations and equivalence ratios throughout.

As in the low density gas case, there is a notable difference in the sizes of the fuel rich cores for the two different jets. Figure 6.5 shows the boundary of the $\phi \geq 2.0$ region for both jets at three time steps. The figure reveals that the core region of the elliptical jet is again diminished relative to the round jet, especially in the early and late stages of injection. A rich core was found to develop earlier in the round jet, and at the first time step displayed in Fig. 6.5 the core of the elliptical jet is again both narrower and shorter than that of the round jet. During the steady portion of the injection, the core of the elliptical jet remains slightly shorter, although the difference is not as notable as in the low density case, and the width of the two jet cores is essentially the same. The rightmost frame in Fig. 6.5 shows the rapid destruction of the elliptical jet core after the injection ends. The rich core regions of both jets are eliminated entirely in the subsequent time steps, but it remains clear from Figures 6.4 and 6.5 that higher concentrations persist longer, and over larger regions in the round jet.

Since an equivalence ratio of approximately $\phi = 0.5$ represents the lower limit for flammability of methane in air, it is also useful to look at comparative contours at that
Figure 6.6: $\phi = 0.5$ contours for round and elliptical jets, high peak density case.

Figure 6.6 shows contours of $\phi = 0.5$ for the high density gas case at the same time steps as in Fig. 6.5. The figure shows that, unlike the rich core regions, the more dilute areas of the jet are virtually identical in extent throughout the injection process. The similarity of the low concentration contours was initially something of a surprise, as previous studies of elliptical jets (see Chapter 2) indicated that they had a tendency to spread more rapidly in the minor axis plane than a comparable round jet. However, these studies were all carried out with low density, low velocity jets in unconfined surroundings. It is believed that the geometry of the combustion bomb chamber prevents a similar effect from appearing in this case. Significant additional jet spreading in the radial direction within the combustion bomb (the minor axis plane of the jet) would require that additional air be entrained from the transverse direction. However, this direction is highly constrained in the combustion bomb chamber due to the close proximity of the rear block wall and quartz window. If there is limited space, and thus a limited quantity of ambient fluid between the jet and these walls, the necessary mass of air will not be available for entrainment from this region and the spreading of the jet will be retarded. In fact, as the penetration results of Brombacher [26] showed, it is likely that the jet is attached to one or both of these walls, which would certainly prevent any additional entrainment.

The fact that the $\phi = 0.5$ contours are so similar for both jets, while the $\phi = 2.0$ region is larger in the round jet suggests that it is only the distribution of the fuel within the jet that is different in the elliptical case, rather than the overall size or shape of the
jet itself. This means that more fuel will be mixed to within the flammability limits in the elliptical jet, which should provide favourable conditions for both ignition and the early stages of combustion.

6.4 Natural Gas - Extended Injection

Since the power output of a Diesel engine is controlled by the quantity of injected fuel (i.e. the injection duration), a range of injection lengths will be encountered in any practical engine application. Due to the fact that a very short injection was used for the first two test conditions with the FFID it was unclear whether the results presented above would hold true for significantly longer injections. Thus, an additional test condition with a longer injection duration was added to ensure that the comparisons drawn already would remain valid at different operating conditions. The results of this test are shown in Figure 6.7. Note that a larger time step is used between frames in this figure, although all of the major stages of the jet’s development are still captured. For this set of tests, a larger number of downstream positions were measured from the elliptical gas jet. This additional downstream data can be seen in Fig. 6.7, although it does not reveal any important new information.

The first item to note from the figure is that it appears to confirm that the short injections were in fact producing jets that reached steady state, since the extent of the jets in Fig. 6.7 is not significantly different from those in Fig. 6.4. Figure 6.8 shows overlapping contours of $\phi = 2.0$ for the two jets from both the long and short injection cases. From the figure, it is clear that the length of injection has at most a very small impact on the extent of the rich fuel core, which is slightly larger for the extended injection condition in both cases. This suggests that for similar chamber conditions the size, shape, and distribution of fuel within the gas jet produced by the injector will be essentially unchanged by injection duration. Of course, the subsequent combustion process may be impacted, but the initial mixing and development stages should be relatively similar.

Since lengthening the injection duration did not produce any significant changes in the development of the gas jets, it is not surprising that the earlier comparisons drawn between the elliptical and round nozzle holes can also be seen in Fig. 6.7. The diminished size of the rich core region in the elliptical nozzle jet is again evident, with a maximum penetration distance into the chamber slightly less than that of the round jet. The much more rapid destruction of the elliptical jet’s rich core in the late stages of the injection
Figure 6.7: Equivalence ratio contours for round and elliptical nozzles, extended injection test.
Figure 6.8: \( \phi = 2.0 \) contours for long (solid) and short (dashed) injections, for both elliptical and round jets.

is also evident. The outer portions of the jets are again quite similar in both shape and extent. The mechanisms responsible for the differences in jet development will be the same in the case of long and short injections.

### 6.5 Calibrated Gas Mixture Tests

Within the jets of natural gas tested so far, fuel concentrations occasionally exceeded the upper limit that could be well resolved by the FFID. Also, since the FFID was being operated at its maximum span to accommodate a wide range of concentrations, there was some concern that very low equivalence ratios would not be accurately captured. As a result, a set of tests was carried out using a calibrated mixture of 2% \( \text{CH}_4 \) in \( \text{N}_2 \), rather than natural gas, as the fuel. By utilizing a gas jet that would contain at most 20000 ppm of \( \text{CH}_4 \) the span of the FFID could be set such that the full range of fuel concentrations could be resolved in the first linear response region, as described in Chapter 5. Tests with the calibrated gas mixture were performed at the high density chamber condition described above as this more accurately matched the conditions that would be found in a diesel engine, and also because it would provide a further check on the data from the long and short injection tests. An injection duration of 2.0 CAD was chosen, part way between the durations used for the results already presented above. The contour maps obtained for the two nozzles are shown in Figure 6.9.
Figure 6.9: Equivalence ratio contour plots for round and elliptical nozzle holes. Fuel is 2% CH₄ in N₂.
As hoped for, the results of Fig. 6.9 resemble those of both Figs. 6.4 and 6.7. A very similar process of core formation, steady state portion, and core destruction is evident in all three cases. That is, the core region of the elliptical jet forms more slowly, reaches a slightly smaller ultimate extent, and is more quickly dispersed after injection than in the round jet. Figure 6.10 illustrates these three stages for the calibrated gas jets, and shows clearly how the comparison between the two nozzles for this case matches that from the earlier test cases.

Comparing the data of Fig. 6.9 with that of Fig. 6.4, the overall size of the core regions in the calibrated gas jets is somewhat smaller than in the tests with pure natural gas. There are a few possible explanations for this discrepancy. The most likely explanation is that the contour plots for the earlier tests were scaled based on the assumption that the jets being measured consisted of pure methane. Since the actual composition of the natural gas being injected wasn’t known, and since the primary purpose was to draw comparisons between cases rather than draw a precise outline of the individual jets, this was not a significant source of concern. However, since the natural gas mixture will contain some quantity of higher alkanes, the pure CH$_4$ assumption will tend to overestimate the extent of a given contour to some degree. For an average natural gas mixture (93-95% CH$_4$) with some inert components the difference will be minimal, but if the inert component is smaller, or there is a greater than average quantity of ethane or propane, it could become more significant. Additionally, since the hydrocarbon concentrations being in the present tests are so small, the results will be more sensitive to
small calibration errors, either in the FFID, or in the composition of the gas itself. The gas mixture used for these tests was certified at 2.04% (or 20400 ppm), with a quoted tolerance of ±2%, or approximately ±400 ppm. A variation of this size could attribute for some portion of the observed discrepancy.

Finally, comparison plots of the $\phi = 0.5$ contours for the two jets were drawn for the early, middle, and late stages of injection. The plots are summarized in Figure 6.11. As mentioned above, since $\phi = 0.5$ represents the approximate lower flammability limit for methane/natural gas in air, these plots give an idea of the size and shape of the “useful” portion of the jet.

The contour plots of Figure 6.11 demonstrate once again that despite having a diminished core size the overall envelope of the elliptical jet is not significantly different from that of the round jet. For all of the time steps depicted, the upstream boundaries of both jets overlap almost completely. The downstream boundary of the round jet is slightly wider than the elliptical jet, although the difference is notable only during the early and late stages of the injection. As far as presenting an ignitable mixture to the glow plug, then, the two jets are essentially identical. The placement/angle of either nozzle relative to a shielded glow plug located in the upstream direction would not need to be altered to obtain the same ignition performance. The wider spread in the downstream direction could potentially help flame propagation between jets in the round nozzle case, although the difference between the two cases is small and would likely be quickly overwhelmed.

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**Figure 6.11:** $\phi = 0.5$ equivalence ratio contours for round and elliptical jets, calibrated gas mixture.
by convection due to swirl.

It was hypothesized earlier that the more rapid destruction of the rich fuel core in the elliptical gas jets was the result of lower peak concentrations due to enhanced mixing. It was also noted that for straight natural gas injection, the fuel concentration in the rich core of the jet occasionally exceeded the limit that could be resolved by the FFID, and this hypothesis could therefore not be accurately assessed. In the present tests, however, the full range of concentrations within the jet can be resolved, and peak values can therefore be compared with greater confidence. Examining the data for both the round and elliptical nozzles, it was found that the peak concentration occurred in both cases at the same time step. Figure 6.12 shows the concentration distribution across the jet field at this time step for an axial position 0.25 in. from the nozzle, where the peak concentration was measured. The points in this figure are the 10-cycle averages taken directly from the source data, with no interpolation or smoothing applied. The error bars depict the $3\sigma$ (99.7%) confidence limits for each point.

The figure confirms the previous suspicion that higher maximum concentrations were being achieved in the round jet. The difference between the two nozzles is significant at $3\sigma$ in the central portion of the jet, where the comparison is being drawn. Unsurprisingly,
Figure 6.13: Equivalence ratios across round and elliptical jets. 1.0 in. from nozzle exit, 357.8 CAD. Note change of scale from Fig. 6.12.

the confidence limits expand significantly as you move to the edges of the jet, since the mixture in this region will be considerably less uniform, and the passages of larger pockets of air or fuel will increase the cycle-to-cycle variability. The difference observed in the more consistent central region would explain why, after the injection ends, the $\phi = 2.0$ contour of the elliptical jet disappears more rapidly, since a smaller amount of additional mixing is necessary to bring those regions of the jet to within flammability limits. The lower peak concentration observed in the elliptical jet is a further potential advantage during combustion, since it will likely reduce the length of the diffusion burning process, thereby decreasing the potential for soot.

At the same time step depicted in Fig. 6.12, slices were also taken through the jet field at the furthest axial position, 1 inch from the injector nozzle. The results are shown in Figure 6.13. This figure reveals that although the elliptical jet reaches lower peak values in the core region, it reaches equivalent relative concentrations in the downstream locations. As discussed in Chapter 5, the variability of the mixture at this far axial position is much greater than that of Fig. 6.12, especially away from the centre of the jet, so none of the differences shown in Fig. 6.13 are statistically significant (hence the error bars are omitted). It should also be noted that, although the elliptical jet reaches
higher concentrations as shown in Fig. 6.13, none of the concentrations at this location exceed the upper flammability limit of $\phi = 2.0$. 
Chapter 7

Combustion and Heat Release Data
Analysis Methods

7.1 Calculation of Ignition Delay

The procedure followed in this study to determine ignition delay is largely the same as that used by Fabbroni [28]. First, an average motored pressure trace is generated from all of the non-firing cycles in a given data set. Then, for each firing cycle, a difference vector is generated by subtracting the average pressure trace from the firing trace. Figure 7.1 shows the average, fired, and differential pressure traces for a sample case.

The start of injection is found by averaging the injector current measurements over the initial portion of the compression stroke, then looking for the point where the current rises above 3 standard deviations from the mean. There is generally very little if any noise in the injector current trace, so determining this point is unproblematic. The point of ignition is found, similarly, by looking for the point where the pressure difference trace crosses the $3\sigma$ line permanently (see Figure 7.2). If ignition is defined instead as the point where the $3\sigma$ line is first crossed, spurious pressure spikes in the early stages after injection due either to early flame kernel development or to the injection event itself can cause ignition delay to be significantly underestimated. Thus, to find the point where the permanent crossing occurs, the pressure differential trace is examined by starting at its maximum value and working backwards.

Determination of the $3\sigma$ point for the pressure trace also presents some challenges. It is necessary to calculate the mean and standard deviation using a base signal that does not systematically vary over its length, or else the mean value of the portion of the signal...
Figure 7.1: Motored (average), fired, and differential pressure traces for a typical fired cycle.

Figure 7.2: Close-up view of ignition event. Pressure differential trace, with mean and $3\sigma$ lines, plus injector current are shown.
Figure 7.3: Influence of base signal length on calculated ignition delay. Mean and standard deviation of 10 consecutive fired cycles is shown for various base signal lengths.

being examined for ignition may differ from the overall base signal mean. This suggests that a shorter base signal is preferable, as large variations in the mean value would be less likely. However, it is also necessary to use a signal that has sufficient point-to-point variation to provide a realistic value of standard deviation. Suppose, for example, a very short portion of the base signal (i.e. 5 to 10 CAD before injection) is used. If the pressure differential trace over this interval happens to be significantly ‘smoother’ or ‘rougher’ than it is generally, the position of the $3\sigma$ line will be under- or over-estimated, impacting the calculated point of ignition. Ultimately, some balance must be struck that will provide a base signal of sufficient length to accurately reflect the variability of the signal as a whole, without being so long that it allows for the possibility of a significant shift in the mean value over its length. A series of calculations was run on several datasets, using base signals of different lengths, to determine the effect on the calculated ignition delays. The results are shown in Figure 7.3. The figure reveals that as the base signal is shortened, the resulting ignition delay tends to be both shorter and more variable. This is not surprising, since the shorter base signal will tend to have a smaller $3\sigma$ threshold, making it more likely that early signal peaks will be identified as ignition events. However, for base signals of approximately 90 CAD and higher, the ignition delay and standard deviation values level out, indicating that the length of the
signal is no longer significantly impacting the calculation. This same trend was observed for each of the data sets studied. An excessively long base signal was still deemed less desirable, as it did have a slight tendency to overestimate the reported delay times in some cases, so a base signal length of 100 CAD was selected as appropriate, and used in all further calculations.

7.2 Heat Release Calculation

A first law energy balance is used to determine the instantaneous net rate of heat release (ROHR) within the combustion bomb system. The use of the net ROHR effectively eliminates the need to account for heat transfer losses during combustion. This greatly simplifies the calculation procedure, and avoids the need for uniquely determined coefficients for the system. The same form of the heat release is used in this study as was used by Fabbroni [28]. The basic differential form of the first law is generally stated as

$$\frac{dQ_{net}}{d\theta} = \frac{dU}{d\theta} + \frac{dV}{d\theta} + h_b \frac{dm_b}{d\theta}$$  \hspace{1cm} (7.1)

where the subscript $b$ denotes blowby gases. The third term on the right therefore represents the energy carried out of the system by blowby gases. Treating the working fluid as an ideal gas, $dU$ is typically replaced by $mc_v dT$. However, this expansion neglects variation in both $m$ and $c_v$. This is usually a valid assumption but in the present case, with larger than usual blowby losses, it becomes necessary to account for variation in mass. By substituting $mc_v T$ for the $U$ and taking the full differential, plus replacing $dm_b/d\theta$ with $-dm/d\theta$, Equation 7.1 becomes

$$\frac{dQ_{net}}{d\theta} = mc_v \frac{dT}{d\theta} + mT \frac{dc_v}{dT} \frac{dT}{d\theta} + c_v T \frac{dm}{d\theta} + p \frac{dV}{d\theta} - h_b \frac{dm}{d\theta}$$  \hspace{1cm} (7.2)

In this formulation, $dc_v/d\theta$ has been expanded via the chain rule, since the term $dc_v/dT$ can be directly calculated, as $c_v$ is a function of temperature only. The $dT/d\theta$ term is eliminated in the final formulation of the equation by substituting $PV/mR$ for $T$ and expanding the differential.

The blowby gases that escape during compression will pass through the very narrow region between the piston and the wall, where a thermal boundary layer will have formed. Thus, the enthalpy of the blowby gases will be different from the enthalpy of the bulk gas in the cylinder. To account for this, the specific enthalpy of the blowby gases is
evaluated at a temperature halfway between that of the bulk gas and the cylinder walls. This, along with some additional substitutions from ideal gas relations and collecting like terms, yields the final form of the net ROHR equation shown in Equation 7.3.

\[
\frac{dQ_{\text{net}}}{d\theta} = \left( c_v + T \frac{dc_v}{dT} \right) \frac{V}{R} \frac{dP}{d\theta} + \left( c_p + T \frac{dc_v}{dT} \right) \frac{P}{R} \frac{dV}{d\theta} - \frac{dm}{d\theta} \left[ \frac{c_p (T + T_{\text{wall}})}{2} + T^2 \frac{dc_v}{dT} \right]
\]

Many of the terms in Eq. 7.3 cannot be directly drawn from the raw experimental data. The following sections will therefore explain how these quantities are determined, from both experimental and theoretical considerations.

### 7.3 Modeling Cylinder Mass

During motored testing, measurements of both pressure and temperature from the combustion bomb were available to calculate the instantaneous system mass. During firing tests the temperature cannot be directly measured, as the thermocouple would be quickly destroyed. However, knowledge of both temperature and mass is still required in order to carry out heat release calculations, so it is necessary to have some functional description of the system mass in the absence of direct temperature data. The development of an appropriate system mass model is described in this section.

#### 7.3.1 Temperature and Mass at Key Cycle Points

The development of correlations for the mass fraction throughout the full cycle is described below in Section 7.3.2. However, before developing a mass model for the full cycle, it was necessary to determine correlations for temperature and mass at some specific points in the cycle. These correlations will later be used to reconstruct traces of total system mass from mass fraction data, and the total mass traces will then be used to reconstruct temperatures for the complete cycle.

The first necessary correlation is for the temperature at BDC of the compression stroke. As discussed earlier, the measured temperature within the combustion bomb is considerably higher than the temperature in the cylinder during the early parts of the compression stroke. Thus, simply using the measured combustion bomb temperature at BDC will significantly overestimate the average system temperature, and thus underestimate the total trapped mass. Instead, following the procedure used by Fabbroni [28], the temperature is found by extrapolating along the polytropic compression line from a
point in the cycle where the system temperature is relatively homogeneous but before the onset of significant blowby, back to BDC. The extrapolated BDC temperature can then be correlated to intake conditions. The correlation found from the motored data sets is shown in Equation 7.4.

\[ T_{BDC}[K] = 365 + 0.0823T_i[^\circ C] + 0.890T_{wall}[^\circ C] \]  

(7.4)

With Eq. 7.4, the true average system temperature at BDC can be determined from intake conditions alone, and can then be used to calculate the total trapped mass.

It was also necessary to have an estimate of the temperature and mass at TDC on the compression stroke. As discussed earlier, a correlation for peak temperature was developed based on intake conditions during motored testing. A number of correlations were tested to determine which predictors resulted in the best results for TDC temperature based on motored results. It was found that peak temperature was the most significant predictor, with some additional influence of combustion bomb wall temperature. The resulting correlation is shown in Equation 7.5.

\[ T_{TDC}[K] = 1.03T_{peak}[K] - 0.169T_{wall}[^\circ C] \]

(7.5)

As with the case of BDC conditions, the temperatures calculated with Eq. 7.5 can be combined with measured pressure and known volume to determine system mass at TDC. As discussed below, these masses help to determine the specific form of the reconstructed mass traces for firing cycles.

### 7.3.2 Mass Fraction Correlations

The process of developing a mass model is greatly simplified by the fact that mass and temperature data are only required for calculations in the portion of the cycle near top centre where injection and combustion occur. It is further simplified by the fact that motored testing revealed that the indicated system mass, when normalized to a mass fraction, follows the same general functional form in this region, independent of intake conditions. Figure 7.4 shows several typical system mass traces reconstructed from motored pressure and temperature data. Note that the various curves in Fig. 7.4 all share similar shapes in the region of interest near TDC, and that the changing intake conditions tend only to shift the curves relative to one another.

The mass fraction traces such as those in Figure 7.4 are calculated relative to the plateau value achieved during the intake stroke, which reveals the total mass trapped
before the onset of significant blowby (see Figure 4.14 and related discussion). The calculated mass is averaged over the region from 290 to 310 CAD, and the full cycle is then divided by this value to determine the mass fraction.

The next step in the analysis was to generate correlations that would allow the mass fraction trace for any cycle to be reconstructed without access to direct temperature measurements. This was accomplished by breaking the cycle into different zones. From 300 CAD to 370 CAD (60° BTDC to 10° ATDC), the mass fraction data was fit in a least-squares sense to an equation of the form:

\[
M_{frac} = A_1 + (E_1 - A_1) \exp \left[-\left(\frac{\theta - B_1}{C_1}\right)^D_1\right]
\]  

(7.6)

This correlation was performed for all of the motored data sets, and the resulting values of \(B_1, C_1, D_1,\) and \(E_1\) were then averaged to produce an overall correlation independent of intake conditions. The correlation constants did not vary a great deal for the different motored data sets, so the average equation is in fact applicable for the full range of combustion bomb conditions, as verified below. The value of \(A_1\) was determined on a cycle-by-cycle basis during subsequent analysis, where it was calculated to make the correlation match the mass fraction at TDC calculated from Equation 7.5.

The same procedure was carried out from 370 CAD to 390 CAD, with the functional
Table 7.1: Correlation constants for mass fraction reconstructions near TDC.

<table>
<thead>
<tr>
<th>Constant</th>
<th>Zone 1 (300-370 CAD)</th>
<th>Zone 2 (370-390 CAD)</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>288.10</td>
<td>359.98</td>
</tr>
<tr>
<td>C</td>
<td>58.34</td>
<td>22.98</td>
</tr>
<tr>
<td>D</td>
<td>5.657</td>
<td>7.011</td>
</tr>
<tr>
<td>E</td>
<td>1.016</td>
<td>0.965</td>
</tr>
</tbody>
</table>

form as shown in Equation 7.7.

\[
M_{frac} = A_2 - (E_2 - A_2) \exp \left[ -\left( \frac{\theta - B_2}{C_2} \right)^{D_2} \right]
\] (7.7)

In this case, \( A_2 \) was calculated for each cycle such that the values of \( M_{frac} \) from Equations 7.6 and 7.7 would overlap at 390 CAD. The average correlation constants eventually determined by this method are summarized in Table 7.1.

For all crank angles during the compression and expansion strokes that fell outside of these two ranges, mass fraction was set to a constant value. From 180 to 300 CAD, \( M_{frac} = 1.0 \), and from 390 to 450 CAD \( M_{frac} = M_{frac}(390) \), as calculated by Eq. 7.7.

The primary function of the mass model is to allow reconstruction of the combustion bomb temperature history. Thus, the multi-zone correlation described above needs to be able to produce an accurate temperature trace, especially in the region near TDC where combustion properties will be evaluated. Figure 7.5 shows a typical motored temperature trace along with the reconstructed temperatures based on the model. The figure shows very good agreement between the two traces. The agreement between measured and calculated temperatures varies somewhat across different intake conditions, but is consistently very good in the region immediately surrounding TDC, where it is most critical to the subsequent analysis.

### 7.3.3 Specific Heat of the Working Fluid

The working fluid in the combustion bomb is assumed to be air for all calculations. Small amounts of natural gas will obviously be present after injection, but this will have minimal effects on the properties of the gas, especially after combustion has commenced. The specific heat capacity, and its derivative with respect to temperature, is used in several places in the ROHR equation. The heat capacity of the gas is calculated at each
Figure 7.5: Comparison of measured and reconstructed temperatures for compression and expansion strokes, representative cycle.

This specific heat correlation, taken from Moran & Shapiro [73], used $T$ in K, and is valid from 300 to 1000 K. Calculating $c_v$ and $dc_v/dT$ from Eq. 7.8 is trivial.

\[
\frac{c_v}{R} = 3.653 - 1.337 \times 10^{-3} T + 3.294 \times 10^{-6} T^2 \\
- 1.913 \times 10^{-9} T^3 + 0.2763 \times 10^{-12} T^4
\]  
(7.8)

7.3.4 Signal Derivatives

While calculating the derivative of the specific heat capacity may be trivial, for signals acquired from the DAQ system the process is considerably more complex. The unavoidable presence of noise in the signal makes the instantaneous calculation of a derivative impractical, so some form of signal filtering becomes necessary. As discussed in Fabbroni [28], the choice of filter applied to the data can significantly impact the resulting derivative calculation. Following the method of Fabbroni, a Savitsky-Golay filter is used for combustion bomb data, as it is generally accepted that this method is best able to preserve short time scale events in the signal, unlike simpler averaging techniques.

The Savitsky-Golay filter works by fitting a polynomial of degree $n$ (specified by the user), to a series of points, or frame, of length $F$ (also user-specified). The relative sizes of $n$ and $F$ determine the trade-off between amount of smoothing and the goodness of the
polynomial fit. Matlab’s Savitsky-Golay filter design routine can also provide a matrix of differentiation filters (for calculating first through fifth derivatives) that allow estimation of the signal derivatives at the midpoint of the frame. By incrementally applying the filtering routine, advancing the frame along the input signal, estimates of the derivative can be calculated for almost the full data set (excluding some points at the start and end).

7.4 Combustion Efficiency Calculation

Measurements taken with the FFID in the exhaust manifold are used to calculate combustion efficiencies during firing combustion bomb tests. Combustion efficiency is calculated for each cycle using equation 7.9:

$$\eta_{\text{comb}} = 1 - \frac{m_{\text{fuel,unburned}}}{m_{\text{fuel, injected}}}$$  \hspace{1cm} (7.9)

The terms are essentially straightforward. Total injected fuel mass is determined by the correlation developed from the fuel injector tests described in §4.2. The unburned fuel mass is determined by the FFID measurements of hydrocarbon concentration in the exhaust. Figure 7.6 shows typical FFID output traces for several cycles from one firing test. The signal typically begins to rise very shortly after EVO at 510 CAD, and quickly reaches a relatively steady value which is maintained throughout the exhaust stroke.
This steady value is averaged and converted to concentration using the appropriate FFID calibration curve. This provides the average molar concentration of CH$_4$ in air. The molar concentration is easily converted to a mass fraction, which is then multiplied by the total system mass at the start of the exhaust stroke to yield the total mass of unburned fuel that is then substituted into Eq. 7.9.
Chapter 8

Firing Tests - Results and Discussion

8.1 Test Conditions

Firing tests were carried out at several conditions for both the round and elliptical nozzles. Three peak motored bulk gas temperatures were tested at each of three peak pressures for both a long and short injection duration. This provided a total of 18 test conditions for each nozzle, or 36 conditions altogether. The test conditions used are summarized in Table 8.1.

As discussed in Chapter 4, the glow plug power setting used for these tests was determined by performing a number of firing test runs with different power settings. The tests were carried out at a combustion bomb condition that was considered relatively unfavourable for ignition. The power necessary to produce a delay of 2ms or less at that condition was then used in all subsequent tests. For the test condition and glow plug chosen, this resulted in a required power setting of 110 W. Fortunately the first glow plug to be calibrated in this manner lasted throughout the remaining experiments, so this calibration step did not need to be repeated.

8.2 Ignition Delay Results

The first parameter of interest in the results of the firing tests is the ignition delay. As discussed in Chapter 7, pressure delay is calculated from the combustion bomb data using the method described in Section 7.1. The delay is calculated independently for each fired cycle, and is then typically averaged over the 10 cycles recorded for a given test condition.
Table 8.1: Intake and peak combustion bomb conditions for firing tests.

8.2.1 Ignition Delay - Temperature Effects

At each peak pressure condition, three peak temperatures were tested. Unfortunately, a large range of peak temperatures was not available with the present test engine set up (see Ch. 4 for discussion of available combustion bomb conditions), but enough variation could be introduced to uncover any significant effects. Figure 8.1 summarizes results for the short injection case, showing temperature effects for two pressure conditions, along with standard deviations (as a percent of the mean).

Comparing the data from the two different nozzles, the figure reveals that there is no significant, consistent difference in either ignition delay or variability. The similarity in performance of the two nozzles was expected, based on the similarity of the outer jet envelopes observed in the FFID testing. The results from the two jets also reveal that there is a significant amount of variability in the measured ignition delays, with standard deviations of 30 to 50% not uncommon. Again, based on the error analysis presented in Chapter 6, this is not surprising, since it was clear that the mixture in this region varied significantly between cycles. This sort of variability is an unavoidable consequence of the highly stratified fuel distribution in a direct injection engine. It should be noted, however, that despite the variability the measured ignition delay was consistently below the 2 ms threshold at all conditions.
Figure 8.1: Temperature effects on ignition delay for short injection. Mean and standard deviation shown for: (a) low, and (b) high pressure conditions, with injection duration of 1.4 CAD.

The charts in Figure 8.1 reveal that increasing peak combustion bomb temperatures have no significant impact on the resulting ignition delay. This initially seems surprising, since temperature is generally a crucial factor in ignition delay length, as discussed in Chapter 2. Although temperature is important in the ERDL combustion bomb it is the hot surface temperature, rather than the bulk gas temperature, that has the greatest impact. At all of the available operating conditions the hot surface of the glow plug in the combustion bomb is several hundred degrees warmer than the bulk gas. Thus, when an ignitable mixture reaches the immediate vicinity of the glow plug the heating at the plug’s surface quickly overwhelms any influence of the bulk gas temperature. Since the bulk gas temperature is in general 400 K or more below the unassisted autoignition temperature of natural gas, it is unlikely that any significant quantity of pre-ignition reactions are taking place prior to the mixture reaching the glow plug and the gas temperature therefore becomes an insignificant factor.
Figure 8.2: Ignition delay and standard deviation, variation with peak combustion bomb pressure. Results shown for: (a) low, and (b) high temperature cases, with injection duration of 1.4 CAD.

### 8.2.2 Ignition Delay - Pressure Effects

Ignition delay results were grouped by temperature and plotted against peak combustion bomb pressures in order to examine what effect this might have on ignition delay times. Representative results are shown in Figure 8.2, along with plots of standard deviation, again expressed as a percentage of mean value. As was the case with the temperature results, there is not a significant or consistent difference between the round and elliptical nozzles evident in the figure. This is again as was anticipated, for the same reasons discussed above.

Evident in the charts in Fig. 8.2 is a trend towards decreasing ignition delay as peak pressure is increased. This requires some additional explanation, as the influence of pressure on ignition delay was generally found to be small except at very high pressures, as discussed in §2.2. The effect in the present case, however, stems from more direct physical
considerations. As the results of the FFID jet tests discussed earlier showed, as the peak bulk gas density (and hence pressure) increases there is a significant enhancement of mixing. A doubling of peak gas density from 10 to 20 kg/m$^3$, for example, produced a much smaller rich core in both the elliptical and round nozzle jets (see Figs. 6.2 and 6.4). By allowing the jet to mix more quickly to within flammability limits, there will be a significant reduction in ignition delay time as peak pressure and density are increased. Additionally, the calibration of the glow plug set point discussed above, and the determination of the ideal injector angle, were both performed at an elevated peak pressure condition. This was done to ensure adequate performance at all test conditions, but it may well have had the added effect of influencing the relative positions of the jet and glow plug at the different pressure conditions. It was noted in Chapter 6 that the axis of the jet is bent more significantly by the bulk gas swirl as peak density is increased, and the upstream side of the jet is significantly more “flattened” as well. With the injector angle optimized for an intermediate to high peak pressure, the fuel jet at the low pressure could end up presenting more of the core region to the glow plug. This will add to the effect of the slower mixing and tend to further increase ignition delay.

While there is a trend towards lower ignition delay as peak combustion bomb pressure increases, there is no significant effect on the variability of the data. The standard deviation as a percent of mean ignition delay remains fairly steady with increasing pressure, and actually increases in some cases. This would indicate that the absolute variance of the data is remaining nearly the same as ignition delay decreases. This would be expected as a result of the physical variability of the highly stratified fuel mixture being presented to the glow plug, as observed in the increasing variability of the FFID data at the periphery of the jet reported in Chapter 6.

8.3 Heat Release and Combustion Imaging Results

Using pressure data from firing cycles and the various correlations and methods developed from the tests of Chapter 4 and Chapter 7, combustion events were analyzed for both nozzles. Instantaneous and cumulative heat release calculations, combined with analysis of combustion images taken with the ICCD camera system provide additional information about the two nozzle designs under test.
Figure 8.3: Burn duration results for short injection case with low peak temperature. Error bars indicate one standard deviation.

8.3.1 Burn Duration Analysis

Numerically integrating the ROHR curves for firing cycles and calculating the time from ignition to a point where the net heat release curve stabilizes provides a good estimate of overall burn duration for each cycle. The results of burn duration calculations for the short injection cases at low temperature are shown in Figure 8.3. The figure indicates that there is no significant difference between the overall burn duration for the two nozzles and, with one exception, there is little difference in the variability of the data. The large standard deviation of the elliptical data at the low pressure condition is due to the presence of two cycles in which the burn duration was significantly higher.

There appears to be a slight trend in the results of Fig. 8.3 towards shorter average burn durations as the peak motored pressure in combustion bomb is increased. Figure 8.4 shows the results for the high peak temperature case, and confirms this general trend. The additional figure also confirms that there is no significant difference in the overall burn durations achieved by the two different nozzle designs. Comparing between the figures also reveals that peak bulk gas temperature has no significant impact on the burn duration.

The influence of peak pressure on the overall burn duration is explained by the mixing behaviour observed with the FFID concentration measurements, and is the same as the
effect noted above on ignition delay. Increasing pressure and density of the chamber gases enhances the extent of fuel-air mixing within the gas jets, which will increase the quantity of fast premixed burning, and will also increase the rate of subsequent diffusion burning to some extent. Increasing peak temperature has little effect since the pre-ignition bulk gas temperature is quickly overwhelmed by the heat release of combustion.

The burn durations calculated for the long injection cases show essentially the same trends as those for the short injection. Naturally, the overall length of combustion is somewhat longer in these test cases. Increasing the injection length from 1 ms (1.4 CAD) to 2.2 ms (3.0 CAD) increases the average burn duration across all conditions by approximately 40%, from 4.47 ms to 6.24 ms. However, the longer burn duration doesn’t increase the variability of the data, with the standard deviation at each condition averaging 8% of the mean value. Figure 8.5 shows the average burn duration at the intermediate peak temperature condition for the long injection case.

8.3.2 Heat Release Rates and Image Analysis: Short Injection Cases

While the results of the overall burn duration calculations revealed no difference between the two nozzles, consideration of the combustion images reveals that there is a clear difference in the progress of the combustion process, even if the overall time taken is largely the same. Figures 8.6 and 8.7 show the combustion images captured for the two nozzles.

Figure 8.4: Burn duration results for short injection case with high peak temperature. Error bars indicate one standard deviation.
Figure 8.5: Burn duration results for long injection case with intermediate peak temperature. Error bars indicate one standard deviation.

at the low peak temperature, intermediate peak pressure case and the intermediate peak temperature, high peak pressure case as representative examples, respectively.

Comparing the series of images in the two figures, the elliptical nozzle appears to be producing a higher intensity flame, covering a greater area of the chamber during the main phase of combustion. This would likely indicate that a greater amount of premixed combustion is taking place in the elliptical case compared to the round. To further investigate, image histograms were generated for each individual frame in the images of Figure 8.6, and the average number of pixels above a given intensity level was calculated. Figure 8.8 shows the average quantity of pixels above an intensity of 16 (from a 256 level grayscale image), or 6.25% of maximum intensity. This is reveals the approximate amount of the image that is enflamed at each time step. Figure 8.9 shows the quantity of pixels above 50% intensity, indicating the amount of the image where the most intense combustion is taking place. The histogram counts for each time step are adjusted relative to a value of zero for the first frame in order to account for variation in background illumination and the glow plug.

The histogram data in the two figures confirms the first impression from Figs. 8.6 and 8.7 that the elliptical nozzle reaches a greater degree of illumination, both in overall extent and in intensity. The two nozzles are generally quite close in the very early stages of combustion as the first flame kernel emerges from around the glow plug, but the elliptical nozzle peaks at significantly higher values than the round nozzle. The two cases
Figure 8.6: Combustion images for short injection, low peak temperature, intermediate peak pressure case.
Figure 8.7: Combustion images for short injection, intermediate peak temperature, high peak pressure case.
Figure 8.8: Average quantity of illuminated pixels for both elliptical and round nozzles. Short injection, low temperature, intermediate pressure case. Counts indicate all pixels with intensity greater than 16 out of 256 grayscale levels, or 6.25% of maximum intensity. Then begin to converge again in the late stages of the burn process.

In order to correctly interpret the data obtained from the images and histograms, it is necessary to simultaneously consider the heat release data. The magnitude and location of the peak heat release rate was determined for each measured cycle. The average location of the peak heat release rate was consistently very similar for the two nozzles both in absolute terms, and relative to the measured start of ignition. However, as shown in Figure 8.10 there are differences in the peak rate of heat release. The figure reveals that the peak rate of heat release from the round nozzle (as a percentage of the total heat release), is consistently higher than that of the elliptical nozzle. The individual differences shown in the figure are not generally significant at high levels of confidence (i.e. much more than one standard deviation), but the trend of higher peaks from the round nozzle is consistent across virtually all of the tested conditions. Based upon the observations of the concentration fields for the two jets presented in Chapter 6 it seems surprising that premixed combustion would be higher for the round nozzle. Looking at the development of the gas jets measured by the FFID, both nozzles appear to be relatively well developed before ignition occurs. It is possible that in the time that elapses before
Figure 8.9: Average quantity of high intensity pixels for both elliptical and round nozzles. Short injection, low temperature, intermediate pressure case. Counts indicate all pixels with intensity greater than 128 out of 256 grayscale levels, or 50% of maximum intensity.

Figure 8.10: Peak rates of heat release for round and elliptical nozzles at low peak temperature, short injection case.
Figure 8.11: Average heat release curve and high-intensity image histogram counts for elliptical nozzle. Short injection, intermediate peak temperature, high peak pressure.

ignition more fuel is mixing to beyond flammability limits in the elliptical nozzle jet, which could potentially reduce the amount of premixed fuel available for combustion. It is also possible that the initial propagation of the flame from the glow plug to the rest of the jet is impeded in the elliptical nozzle jet due to the presence of overly lean mixture, or to some other unforeseen difference in the dynamic behaviour of the jet. Unfortunately the available data do not provide a conclusive explanation for this observation, although some of the differences are further examined below, as well as their implications.

The peak rate of heat release from both nozzles, for the low temperature, intermediate pressure test condition of the images and histograms displayed above, occurs at approximately 357.7 CAD. Referring back to the histograms of Fig. 8.8 or 8.9, it is clear that the peak in energy release is occurring before the peak in flame luminosity for both nozzles. This is confirmed by Figures 8.11 and 8.12, showing the simultaneous progress of the averaged heat release rate and image illumination for both nozzles for the intermediate temperature, high pressure condition of Fig. 8.7.

The overlayed heat release curves & histogram counts confirm that the illumination within the combustion chamber peaks slightly after the peak rate of heat release for both the round and elliptical nozzles. The increased illumination in the chamber after the
Figure 8.12: Average heat release curve and high-intensity image histogram counts for round nozzle. Short injection, intermediate peak temperature, high peak pressure.

peak rate of energy release is likely due to the production and combustion of soot during the diffusion burning phase. Although the filter being used admits light primarily in the blue part of the visible spectrum, high intensity white light radiates across the full spectrum, and will emit some quantity of radiation in the blue region. If the white light is intense enough, then, it could produce the sort of peaks observed in the histograms shown above. To examine this, consider the case of a black body radiating at the adiabatic flame temperature of methane, approximately 2250 K. Figure 8.13 shows the spectral emissive power for such a body, along with a “filtered” spectrum obtained by applying the transmission data for the blue filter used with the ICCD camera. The figure quite clearly shows that the long wavelength radiation has significantly more emissive power than the shorter blue light, even after filtering. For the early, premixed portion of combustion this is not a concern, since primarily blue light is incident on the CCD. However, once a white spectrum develops from the oxidation of soot there is a strong likelihood that it will dominate the incident light regardless of filtering. The modeling of this flame as a black body is only a rough estimation, so the data of Fig. 8.13 should not be taken as indicative of the true incident radiation on the CCD array, but merely as an indicator of the potential for white light to overwhelm the images. The possible implications of this
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Figure 8.13: Emissive power for black body radiation at 2250 K: base spectral power and filtered intensity.

are discussed further below.

Thus far, the results presented appears to lead to different conclusions about the combustion properties of the two nozzle designs under test. The higher peak heat release rates from the round nozzle seem to indicate a greater quantity of premixed combustion, while the higher degree of illumination in the combustion images suggests a greater premixed phase for the elliptical nozzle. However, since the illumination peaks slightly after the heat release curves as shown in Fig. 8.11 and 8.12, it remains uncertain whether the peak illumination is occurring during the premixed or diffusion burning phase. If the illumination being measured for the elliptical nozzle is the result of diffusion burning (i.e. oxidation of soot), it would suggest that the elliptical nozzle may produce more particulate emissions, rather than less, as initially hoped and predicted by the FFID measurements. To further examine this the average heat release curves for each nozzle were plotted simultaneously in order to observe the relative heat release profiles. Figure 8.14 shows the heat release curves at the intermediate peak temperature and high peak pressure case. The curves are normalized relative to their maximum values since the shape of the curves, rather than the absolute magnitudes, is of interest in this case (the significance of peak heat release rates was discussed above).

The results shown in Fig. 8.14 are representative of what is seen at most test condi-
Figure 8.14: Heat release rate profiles for elliptical and round nozzles. Short injection, intermediate peak temperature, high peak pressure case.

It shows that the energy release from the round nozzle climbs slightly more quickly, but then also drops rapidly before entering the diffusion burning phase. The heat release profile from the elliptical nozzle, on the other hand, transitions more smoothly between the two burn phases, and also has a wider plateau and slower descent during the pre-mixed burn. This can be observed in individual ROHR curves, and is thus not simply the result of averaging. The peak values from the image histograms described above generally occurred in the region where the transition between premixed and diffusion burning takes place. With more fuel being consumed in the elliptical jet at this stage, this helps to explain the apparent contradiction mentioned above. It indicates that the peak image intensities are not occurring exclusively within the diffusion burning phase, and are therefore not necessarily associated purely with production of soot. In fact, looking back at the combined heat release/histogram curves above, the peak illumination in the round nozzle occurs well into the diffusion burning phase, while the peak from the elliptical nozzle occurs much closer to the transition between premixed and diffusion burning. As mentioned, these results are consistent across most test conditions. The differences diminish somewhat as peak pressure is reduced and combustion becomes more variable, but the elliptical nozzle consistently maintains the wider, smoother heat release profile.
8.3.3 Heat Release Rates and Image Analysis: Long Injection Cases

As in the short injection cases presented above, full sets of combustion images for representative conditions with the long (3.0 CAD) injection are presented below. The light intensity from combustion in these cases was considerably higher than in the short injection cases, so the images below are scaled to a different range of pixel intensities. Direct comparison between the two nozzles remains valid. Increasing the length of injection naturally increased the overall burn duration meaning that in some cases, especially at low pressure, the full combustion process was not captured by the camera. Thus, images for the mid-temperature, mid-pressure case and the high-temperature, high-pressure case are shown below in Figures 8.15 and 8.16, respectively.

Simple inspection of the figures reveals a similar pattern to that observed in the short injection cases above, with similar overall burn durations achieved with both nozzles, but significantly higher light intensities produced by the elliptical nozzle jets. Histograms were again produced for each frame of the images, and averaged over the 10 cycles captured at each condition. Figure 8.17 shows the results for both low and high intensity cutoff values at the intermediate temperature & pressure condition, while Figure 8.18 shows the same for the high temperature, high pressure condition.

The image histograms for these two conditions show much the same pattern as those presented earlier. In both of the cases shown the elliptical nozzle peaks with much more illuminated area than the round nozzle at both low and high intensity cutoffs. It should also be noted that for both of the cases shown, the area illuminated in the elliptical nozzle images is decreasing by the end of the captured window, while the illuminated areas in the round nozzle images is holding steady, and increasing in several cases. Since the overall burn durations are quite similar for both nozzles, this suggests that the highly luminous burning taking place in the late stages with the round nozzle is the result of pockets of rich fuel mixture producing soot. The fact that the peak intensity occurs much earlier in the elliptical nozzle images makes it more difficult to judge whether it is the result of molecular emission or soot production & consumption (or some combination of both). However, considering the fact that the illumination level is generally quite low at the point where heat release peaks, any subsequent increases in illumination are more likely from diffusion burning (i.e. soot production). If this is true, there is the possibility that the soot produced very late in the combustion process will remain, and then be
Figure 8.15: Combustion images for long injection, intermediate peak temperature, intermediate peak pressure case.
Figure 8.16: Combustion images for long injection, high peak temperature, high peak pressure case.
exhausted as particulate emissions, while the relatively lengthy remaining combustion after the peak illumination in the elliptical nozzle could consume much of the particulate matter formed.

The averaged heat release curves for the long injection conditions presented above are again quite similar to those shown earlier for the short injection cases. Figure 8.19 shows the normalized heat release rates for the intermediate temperature & pressure case, and confirms that the heat release curve produced by the elliptical nozzle drops off less sharply after peaking, resulting in a smoother transition to the diffusion burning process. The peak in illumination for the elliptical nozzle at this condition occurs very near 361 CAD, while there is no clear peak in the illumination of the round nozzle images. As noted above, this implies that illumination peaks after heat release, and it is therefore diffusion burning that drives the light output at this point.

The possibility that the premixed burning process is somewhat slower or more distributed in the elliptical case is implied by both the average ROHR curves shown above and the lower peak heat release rates discussed in the previous section. A lower peak rate of heat release would have beneficial effects on the production of NOx by reducing peak temperatures within the combustion chamber. To examine this possibility, along with the relative rates of soot formation in the two nozzle configurations, would be an important aspect of future testing.
Figure 8.18: Image histograms for long injection, high peak temperature, high peak pressure case. Left chart shows percentage of pixels above 6.25% illumination, right chart shows percentage of pixels above 50% illumination.

Figure 8.19: Normalized average heat release rate curves for round and elliptical nozzles. Long injection, medium peak temperature, medium peak pressure case.
Figure 8.20: Combustion efficiency variation with temperature for: (a) low, (b) medium, and (c) high peak pressure cases. Injection duration of 1.4 CAD. Error bars indicate one standard deviation.

8.4 Combustion Efficiency Results

8.4.1 Combustion Efficiency with Short Injection

Combustion efficiencies were calculated for each recorded cycle using data obtained from the FFID in the exhaust stream of the CFR engine, using the method described in Chapter 7. The results, showing the influence of increasing peak temperature at three pressure levels, are shown in Figure 8.20.

The figure reveals no clear or consistent trends when comparing the results for the two nozzles. In general the differences between the round and elliptical nozzle efficiencies are not significant at a high level of confidence. In those cases where the difference is significant at a single standard deviation, there is a slight tendency for the round nozzle to perform better. The figure does indicate that the performance of both nozzles is quite...
good, with mean efficiencies ranging from 80 to almost 95%. There is also a trend towards increasing combustion efficiencies as peak temperature is increased in all three cases. As there was no significant effect of increasing peak temperature detected in the tests this trend is not the result of pre-ignition mixing to below flammability limits, but rather indicates that a higher peak temperature improves the subsequent combustion process itself. As discussed above, the influence of the bulk gas temperature on the ignition process is overwhelmed by the temperature of the glow plug. However, the bulk gas temperature can still affect how quickly the initial flame kernel will propagate out from near the glow plug through the remaining fuel. As the flame in the combustion bomb propagates from the “upstream” side of the jet to the “downstream” side, in the direction of swirl, it is likely that more downstream fuel will be mixed beyond flammability limits if the flame propagation process is slow to reach it, and this will degrade the combustion efficiency.

In all three cases shown in Fig. 8.20 the variability of the data decreases substantially as the mean efficiency increases. This reflects the fact that an increasing mean combustion efficiency indicates more repeatable combustion from cycle to cycle. That is, the lower efficiency is the result of an increasing range rather than simply an overall shift towards lower values. As the peak bulk gas temperature in the bomb is increased there is little change in the maximum efficiency observed at different conditions, but the minimum efficiency increases significantly, from as little as 70% at low temperature conditions up to 85 to 90% or higher at high temperature.

Results of the combustion efficiency calculations were also plotted versus peak pressure for the various temperature levels, as shown in Figure 8.21. The figure reveals little new information not discussed above, except for the fact that increasing peak pressure within the combustion bomb has no appreciable effect on combustion efficiency. Given the apparent decoupling of ignition delay and efficiency just discussed, this is not unexpected. The decreasing variability with increasing temperature is again visible between the three figures.

The apparent lack of influence of the nozzle design on combustion efficiency was initially somewhat surprising, given the observed differences between the two jets discussed in Chapter 6. However, on further reflection it is not clear that any significant influence should have been expected. Since the overall envelopes of the two jets were in all cases so similar there is likely little difference in the amount of premixed fuel that falls below flammability limits; the elliptical nozzle might be expected to push slightly more fuel into
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Figure 8.21: Combustion efficiency variation with peak pressure for: (a) low, (b) medium, and (c) high peak temperatures, short injection duration case. Error bars indicate one standard deviation.

this region, which could account for the small differences noted above. After ignition the low speed of the engine means that there is a great deal of time for combustion to proceed to completion, which would negate any potential advantage of the smaller core in the elliptical jet. Of course, the combustion duration can still have an important impact on performance and soot production, as discussed earlier. However, in terms of the ability of the flame to propagate throughout the jet field and for combustion to proceed, there is little to suggest that the two jets’ structures should play a significant role.

As discussed in Chapter 2, efficiency can be heavily influenced by ignition delay, with large delays seriously degrading performance as shown in Fig. 2.8. That figure also indicates, however, that efficiency is not seriously impacted until the ignition delay crosses a certain length threshold (approximately 2 ms in that case). Before that point, ignition delay and efficiency are independent of one another. In the tests being presented
here the ignition delay was in almost all cases at or below the 2 ms threshold, with a maximum of just under 2.3 ms observed for a single cycle. Figure 8.22 plots combustion efficiency directly against ignition delay for the measured cycles across all conditions, showing the decoupling of ignition delay and combustion efficiency described above.

### 8.4.2 Combustion Efficiency with Long Injection

Results for combustion efficiency calculations with the longer injection duration are presented next. Figure 8.23 shows the mean combustion efficiency versus peak bulk gas temperature for three different pressure levels. The results for the long injection are quite similar to those for the short injection. In general the differences between the two nozzles are not significant at very high confidence levels, but the round nozzle does perform slightly better under most conditions. The trend towards increasing efficiency and decreasing variability with increasing peak temperature is also apparent.

The most notable difference between the long and short injections is that the long injection case results in higher efficiencies at nearly all conditions. The mean efficiencies shown in Fig. 8.23 are all above 90% and are in the range of 96 to 98% at the highest temperature condition. Clearly the combustion process is consuming nearly all of the
fuel in these cases, with only a very small amount being exhausted. This suggests that
the fuel which is escaping the combustion process is that small portion which mixes
to beyond flammability prior to ignition or in the early part of combustion, or falls
in the extinction region very near the combustion bomb walls. This quantity of fuel
should be the same in the long and short injection cases, since the ignition delay is not
significantly different, and since the long injection provides more fuel, the over-mixed
portion will comprise a smaller percentage of the total and result in a higher efficiency.
The additional fuel and extended burn duration will also release significantly more energy
and drive up the temperature in the combustion bomb chamber, which could in turn aid
in driving the combustion reaction to completion more effectively. As was the case with
the short injection, increasing the peak combustion bomb pressure has no significant
effect on either the mean combustion efficiency or the variability of the data.

Figure 8.23: Combustion efficiency variation with peak temperature for: (a) low, (b) in-
termediate, and (c) high peak pressures, long injection duration case. Error bars indicate
one standard deviation.
Chapter 9

Conclusions and Recommendations

9.1 Conclusions

The different areas investigated during this study have produced results which suggest that elliptical nozzle holes for gaseous fuel injection are a potentially positive but ultimately unproven method enhancing direct injection engine operation.

The results of the FFID jet concentration measurements presented in Chapter 6 showed that there are small but significant differences between the two nozzles under test. The elliptical nozzle consistently produced jets with smaller rich core regions, but which achieved nearly identical overall extent as the jets from the round nozzle. It was also observed that after injection was cut off, the core region of the elliptical jet tended to be destroyed more rapidly and completely than that of the round jet. Measurements of peak concentration obtained by using a calibrated gas mixture confirmed that higher peak concentrations were being produced in the round nozzle jets, with differences that were significant at very high levels of confidence. The very tight confidence limits on the data from the central portions of the jets also confirmed that the FFID produced reliable and repeatable measurements, with the observed variation in data at further downstream and off-centre locations attributed to physical variability of the mixture in those regions.

The results of firing tests within the combustion bomb generally indicated smaller differences between the two nozzle designs, with one exception. Ignition delay performance from both nozzles was excellent, with delays of less than 2 ms achieved at all of the test conditions and no significant differences between the two designs. This was attributed to the fact that the overall jet envelopes were very similar for the two nozzles, and they would therefore present similar mixtures to the glow plug at the same times.
The same trends were also observed in both nozzles, with increasing peak pressures leading to shorter ignition delays, and little to no effect of peak temperature. The pressure effect is attributed to the increased mixing rate causing ignitable mixtures to develop sooner, while the bulk gas temperature is assumed to be overwhelmed in its influence by the temperature of the hot surface. There was no significant difference observed in ignition delay when longer injection durations were used, as would be expected.

The results of the heat release analysis from the two nozzles allowed for a number of comparisons. The overall burn duration was found to be similar for the two nozzle designs, with differences rarely reaching statistical significance at more than a single standard deviation. Burn durations were found to decrease with increasing peak temperature, and there was again little effect of peak temperature. The reasoning for this is similar to that for the ignition delay results, with increasing pressure leading to enhanced mixing and therefore faster combustion, while bulk gas temperature was quickly overwhelmed by the heat release of combustion and had no significant influence.

Looking from overall burn durations to instantaneous heat release rates, it was found that the peak rate of heat release from the round nozzle jet was typically higher than that of the elliptical nozzle. It was hypothesized that this could possibly be attributed to over-leaning in the elliptical jet causing less premixed fuel to be available, but the possibility of another mechanism being responsible cannot be discounted. This is one area that requires additional investigation, as discussed in the following section.

While peak heat release rates were somewhat higher for the round nozzle jets, it was noted that when comparing the normalized heat release rate curves of the two nozzles, the elliptical nozzle produced a wider plateau in the premixed combustion phase, and transitioned more smoothly to the diffusion burning phase. This could indicate that less pure diffusion burning was taking place in the elliptical nozzle jets, which would be beneficial from a soot production standpoint. Additionally, lower peak heat release in the elliptical nozzle jets would likely lead to lower NO\textsubscript{x} production. As discussed below, these are two areas that need particular attention in any future studies of elliptical fuel injector nozzle technology.

Combustion images were analyzed as a supplement to the heat release data captured in the combustion bomb, and revealed some of the most significant differences between the two nozzle designs. Image histograms indicated that at virtually all conditions the elliptical nozzle jets produced more intensely luminous flames, and illuminated a greater area of the chamber. The peaks in luminosity generally occur near the transition from
premixed to diffusion burning in the elliptical nozzle jets, while the flame from the round nozzle jet generally peaked later, well into the diffusion burning phase. It was difficult to determine from the images how much of the luminosity of the elliptical jet flame was the product of soot production. It was noted, however, that an earlier peak in soot production in the elliptical jet would allow for more complete oxidation, potentially reducing the quantity of particulate emissions. Conversely, the later peak in the round jet flame would not allow as much time for subsequent oxidation. In the absence of reliable measurements of the absolute rate or quantity of particulates produced by the flames, no firm conclusion can be drawn about which of the two nozzle designs would result in higher emission levels. This should be addressed in any future work, as discussed below.

Finally, hydrocarbon measurements in the CFR engine’s exhaust were used to calculate combustion efficiencies for both nozzles. It was found that the round nozzle generally achieved higher efficiencies, although the differences tended to be quite small. It is hypothesized that the observed differences are due either to over-leaning in the elliptical nozzle jets during the pre-ignition period, or due to the higher peak heat release rate in the round nozzle jet allowing more fuel to be consumed early in the combustion process before mixing beyond flammability limits.

### 9.2 Recommendations & Future Work

There are a number of areas still available to be explored in this research program.

Although a great deal of work has gone into characterizing the CFR engine and combustion bomb system, there are still some areas that could be better understood. A good understanding of the swirling flow within the combustion chamber has never been achieved, and would be a very valuable piece of additional data. Some flowrate measurements within the combustion bomb chamber were attempted during this study, but the available meter was not sufficiently sensitive to rapid changes in flow rate. Measurements of the flow in the chamber with a hot wire anemometer would be relatively simple under motored operation, although there is some question whether the sensors would survive within such a harsh environment for very long.

The intake flow rate measurements obtained in this study proved very useful in providing confirmation for the pressure and temperature measurements within the combustion bomb, but was available only at ambient intake air conditions. The functionality of this system could be greatly enhanced by installing an intake air plenum that would allow
the flow meter to be used at elevated manifold pressures. The only requirement for such a system would be that the selected plenum be able to withstand pressurization up to the 30 psig limit of the compressed air supply.

Finally, a deeper understanding of the coupled CFR cylinder/combustion bomb system would be aided by additional fast-response temperature measurements under motoring operations. Simultaneous measurement of the intake manifold, cylinder, combustion bomb, and exhaust temperatures would provide much needed insight into the gas exchange and heat transfer processes taking place in the multiple zones that make up the complete apparatus. This could shed light on issues such as the “step” in the temperature profiles that was discussed in Chapter 4, and would also add rigour to any future modeling efforts.

The jet concentration field results obtained from the FFID were generally quite satisfactory. The most significant limitation on the method was that pressure independence was lost at peak bulk gas densities much higher than those reported in this study. To compare the different nozzle designs over the full range of available combustion bomb conditions would require another method that is not limited by high pressures. A measurement probe that could be traversed axially across the chamber as well as radially would provide a 3D measurement of the concentration field, which would be an improvement over the 2D measurements reported in this work.

The greatest ambiguity remaining in the results of this study is in the correct interpretation of the combustion data gathered for the two nozzle designs. There is some question remaining about the relative quantities of premixed and diffusion burning and the implications this has on production of soot and NO\textsubscript{x}. There are several additional experiments that could help to resolve these questions. The most direct method to investigate emissions performance would be through the use of fast NO\textsubscript{x} and fast particle samplers inserted directly into the combustion bomb. However, simpler solutions could also offer answers that would make the more complex methods unnecessary. For example, an estimate of particulate emissions could be obtained by filtering the exhaust gases over an extended period of time and measuring the mass change.

Changes to the camera system could also provide further information about the combustion characteristics. The use of different filters, allowing light of different wavelengths to pass through to the detector, could be used to help distinguish between the phases of combustion. A filter that admitted only long wavelengths, for example, would be better suited to determining the onset of soot production and oxidation since it would eliminate
the ambiguity present in the results of Chapter 8 between early blue spectrum radiation and later full-spectrum radiation.

It was proposed in Chapter 8 that the observed differences in combustion efficiency between the round and elliptical nozzle jets could be the result of over-leaning in the pre-ignition period for the elliptical nozzle. However, since ignition delays were generally kept so short, it was difficult to confirm this, as no significant degradation in combustion efficiency was ever observed. By deliberately extending ignition delay, either by presenting a less ignitable gas mixture (such as pure methane) or by lowering the hot surface temperature, it would be possible to further test this hypothesis.
Bibliography


Appendix A

Equipment Operating Procedures

The following operating procedures are a slightly adapted version of those which appeared in Fabbroni [28]. Certain elements have been expanded on for clarity, and changes made to the steps themselves where appropriate.

1. Replace bomb window gaskets if necessary, and torque in place. Torque should be brought up in increments of no more than 10 in·lbs, and a proper tightening sequence should be maintained. Final torques of between 30 and 50 in·lbs, depending on operating pressure, will generally be sufficient.

2. Set injection angle by rotating gas injector and torque in place to 12 ft·lbs. Take care to keep the two sides of the tightening plate aligned. Ensure that injector leads are not connected when adjusting injection angle or gas line. Connect leads once injector is properly tightened in place.

3. Turn on oil pump.

4. Set oil heater to ‘High.’ Monitor oil temperature carefully during warm up, and reduce heater to ‘Low’ when desired temperature is reached.

5. Check coolant heater setpoint and turn heater on.

6. Set combustion bomb rope heater to desired temperature and turn rope heater on.

7. If using CCD camera system to capture combustion images:

   (a) Turn on cooling water for ICCD camera.

   (b) Turn on ST-138 controller and cooling module.
(c) Turn on PG-200 pulse generator, making sure that ‘MCP Voltage’ switch is off.

8. If FFID is being used:

(a) Clean FID sample transfer tube, T-top & FID tube, and reassemble sample head.

(b) Open air, hydrogen, and methane gas cylinders. Air and hydrogen regulators should be set to 100 psi. Methane regulator can be set to desired calibration mixture pressure.

(c) Switch on FID vacuum pump. Set desired CP chamber and ∆P-FID values.

(d) Light the FID. Monitor sample head temperature until flame stabilizes. Fine tune pressure settings if necessary.

(e) Allow at least 30 minutes for FID to stabilize (if re-lighting, less time may be required).

9. Switch on all power supplies, data acquisition PCs, and control circuitry.

10. Wait for all temperatures to reach setpoints and stabilize. This will take between 30-60 minutes.

11. Open valves on compressed natural gas tanks and switch fuel control solenoid to ‘On.’

12. Start SetP.vi and set fuel line pressure and intake manifold pressure to desired values. Release pressure in intake manifold before attempting to start the engine.

13. Open StartInjector.vi, StopInjector.vi and BombDAQ.vi on main data acquisition computer. Input appropriate values for temperatures, pressures, and injection parameters.

14. Set up the ICCD camera system, if being used:

(a) Launch WinView software on camera control PC.

(b) Capture image of background charge on ICCD array.

(c) Turn on ‘MCP Voltage’ switch on PG-200.

(d) Remove ‘Inhibit In’ connector from PG-200 and set TRIGger to INTernal.
(e) In WinView uncheck ‘Auto Store’ in the Acquisition menu.

(f) In Experiment Setup, under the Acquisition menu, set ‘# of Images’ to 1 and uncheck ‘Synchronous.’

(g) Under the Acquisition menu, select ‘Run’ to begin capturing images. Adjust the location and focus of the ICCD camera to produce optimal images. Stop image collection in WinView.

(h) Reconnect the ‘Inhibit In’ cable and set TRIGger to EXTernal on the PG-200.

(i) In WinView, check ‘Auto Store.’ Also check off ‘Synchronous’ and set ‘# of Images’ as desired (for firing tests this will usually be 10 or 11).

(j) From the Acquisition menu, select ‘Run As.’ Ensure that ‘Store Synchronously’ is checked. Set the File Increment as desired, remembering that there is an 8-character limit on WinView file names.

(k) Select ‘Run’ in WinView. The ICCD camera will now wait until external trigger pulses are received to begin image collection.

15. Calibrate the FFID, if using:

(a) Place the FFID probe tip in the calibration position, where it will be directly exposed to the calibration gas.

(b) Use the Matheson rotameters to prepare a calibration mixture, or series of mixtures, representing the expected range of concentrations to be encountered. Use the oscilloscope to set Offset and Span to ensure a good voltage response. Record calibration voltages for all rotameter mixtures.

(c) Return FFID probe tip to sampling position and tighten in place.

16. If running firing tests, run glow plug monitoring VI and switch on both glow plug power supplies. Adjust auto-transformer to achieve desired power output. Switch off second power supply until ready to begin data collection.

17. Apply power and begin motoring the CFR engine. Adjust voltage to achieve desired RPM.

18. Open compressed air line to apply intake manifold pressure as set earlier once engine is running.
19. Set desired intake air temperature and turn controller to ‘Auto.’ Allow engine to motor and wait for intake temperature to stabilize.

20. Turn glow plug power on and wait for resistance measurement to stabilize.

21. Run BombDAQ.vi to begin firing injector and collecting data & images.

22. After data collection run stop engine, release intake pressure, turn off glow plug power, and turn off intake air heater.

23. Adjust setpoints as necessary and re-run data collection process.
Appendix B

Labview Virtual Instruments

The following pages contain figures showing the front panels and block diagrams of all of the major LabView virtual instruments (Vis) used in data collection and equipment calibration and monitoring for this study.

SetP.vi is used to monitor fuel line and intake manifold pressure during equipment set-up, to ensure the desired setpoints are reached and maintained. MotoredDAQ.vi is used for recording combustion bomb pressures and temperatures, and intake flow rates during motored operation. BombDAQ.vi is an elaborated version of MotoredDAQ.vi that controls data acquisition during firing tests: monitoring input channels, controlling timing for image acquisition, and calling subroutines to start and stop the gas injector. StartInjector.vi and StopInjector.vi are called from within BombDAQ.vi to start and stop fuel injection. StartInjector.vi also contains the fuel injection parameters such as injection duration and skip fire number. MassTest2.vi was used for performing gas injector nozzle calibration experiments.
Figure B.1: SetP.vi Front Panel

Figure B.2: SetP.vi Block Diagram
Figure B.3: MotoredDAQ.vi Front Panel

Figure B.4: MotoredDAQ.vi Block Diagram, Frame 0
Figure B.5: MotoredDAQ.vi Block Diagram, Frame 1

Figure B.6: MotoredDAQ.vi Block Diagram, Frame 2
Figure B.7: BombDAQ.vi Front Panel
Figure B.8: BombDAQ.vi Block Diagram, Frame 0

Figure B.9: BombDAQ.vi Block Diagram, Frame 1
Figure B.10: BombDAQ.vi Block Diagram, Frame 2

Figure B.11: BombDAQ.vi Block Diagram, Frame 3
Figure B.12: StartInjector.vi Front Panel & Block Diagram

STOP INJECTION

To stop injector simply start this VI. Click twice. (The first click will make the window active, the second click will start the VI.)

Figure B.13: StopInjector.vi Front Panel & Block Diagram
Figure B.14: MassTest2.vi Front Panel

Figure B.15: MassTest2.vi Block Diagram
Appendix C

Matlab Scripts & Functions

C.1 MotorAvg.m

% Read data from source file (change file name & save 
% .m file before calling MotorAvg from Command Window).
prefix = ('Apr4_16');
X = dlmread(['Data file path' prefix '.txt'],',

% Read bomb setpoints from file header
Speed = X(1,1);
InTemp = X(1,2);
CoolTemp = X(1,3);
BombTemp = X(1,4);
OilTemp = X(1,5);
GPTemp = X(2,1);
GPPos = X(2,2);
GPNum = X(2,3);
GPPower = X(2,4);

% Generate crank angles and engine geometry
EngModel;

%Initialize calculated property arrays
AvgBP = zeros(3600,1); %Bomb pressure
AvgIP = zeros(3600,1); % Manifold pressure
AvgT = zeros(3600,1); % Bomb temperature
AvgFlow = zeros(3600,1); % Volumetric flowrate
AvgM = zeros(3600,1); % Trapped cylinder mass
Mfrac = zeros(3600,1);
% Sum properties over 12 motored cycles
point1 = 3;
point2 = 3602;
for i = 1:1:12;
    % Convert successive cycles of MAP voltage to pressure and add to
    % existing vector
    AvgIP = AvgIP + 65.831*X(point1:point2,3) - 3.325;
    % Convert piezo voltage to pressure. Voltages are referenced to
    % BDC by subtracting transducer reading & adding intake manifold
    % pressure.
    AvgBP = AvgBP + 1133.6*X(point1:point2,2) - ...
           1133.6*X(point1+449,2) + 65.831*X(point1+449,3) - 3.325;
    AvgT = AvgT + X(point1:point2,1);
    AvgFlow = AvgFlow + X(point1:point2,4);
    point1 = point2 + 1;
    point2 = point1 + 3599;
end

% Calculate average motored cycle profile for each property
AvgIP = AvgIP/12;
AvgBP = AvgBP/12;
AvgT = AvgT/12;
AvgFlow = AvgFlow/12;
AvgFlow = 10.^((log10(AvgFlow/2.327)/0.1582)); %Convert to SCFM

for i=1:1:length(CAD);
    % Indicated mass at each CAD from measured T & P
    AvgM(i) = AvgBP(i)*Volume(i)/(287*AvgT(i));
end

%Convert masses to mass fractions (use appropriate averaging window).
Mfrac = AvgM/mean(AvgM(1400:1600));

% Create ideal polytropic compression line
for i = 1:1:900;
    PTrope(i) = c/Volume(i+899)^1.3;
end

% Filter average temperature and bomb pressure traces (for plotting)
SmoothT = LPFil(AvgT,5);
SmoothIP = LPFil(AvgIP,5);

% Return peak motored pressure and temperature
BPmax = max(AvgBP);
\[
T_{\text{max}} = \text{max}(\text{AvgT});
\]
% Mean intake manifold pressure (psig), for correlation
\[
\text{IP} = \text{mean}(\text{AvgIP})*14.7/101.325-14.7;
\]
% Intake manifold pressure (used for scaling in plots)
\[
\text{IPmax} = \text{max}(\text{AvgIP});
\text{IPmin} = \text{min}(\text{AvgIP});
\text{Mmax} = \text{max}(\text{AvgM}); \% \text{Maximum indicated cylinder mass (for scaling)}
\]
% Call blowby subroutine to determine mass loss & flowrate indicated
% by T & P history
\text{Blowby};

% Generate plots of motored cycle conditions
% Average temperature and pressure curves
\text{figure};
\text{curve1 = line(CAD,SmoothT,'Color','k','LineStyle',':');}
\text{ax1 = gca;}
\text{set(ax1,'XLim',[0 720]);}
\text{set(ax1,'XTick',[0 90 180 270 360 450 540 630 720]);}
\text{xlabel('CAD');}
\text{ylabel('Temperature [K]');}
\text{ax2 = axes('Position',get(ax1,'Position'),'XAxisLocation','bottom',...}
\text{'YAxisLocation','right','Color','none','XColor','k','YColor','k');}
\text{curve2 = line(CAD,AvgBP,'Color','k','LineStyle','-','Parent',ax2);
set(ax2,'XLim',[0 720]);
set(ax2,'XTick',[0 90 180 270 360 450 540 630 720]);
\text{ylabel('Pressure [kPa]');}
\text{11 = line([10,10],[get(ax1,'YLim')],'LineStyle',':','Color','k');}
\text{12 = line([214,214],[get(ax1,'YLim')],'LineStyle',':','Color','k');}
\text{13 = line([15,15],[get(ax1,'YLim')],'LineStyle','--','Color','k');}
\text{14 = line([500,500],[get(ax1,'YLim')],'LineStyle','--','Color','k');}
\text{head = sprintf('Speed = %1.0f RPM, Intake Temperature = %1.1f °C,'...}
\text{'Intake Pressure = %1.1f kPa \n', Speed, InTemp, mean(AvgIP));}
\text{head2 = sprintf('Bomb Temperature = %1.0f °C,'...}
\text{'Coolant Temperature = %1.0f °C', BombTemp, CoolTemp);
\text{header = [head head2];}
\text{title(header);}
\text{legend([11 13],'IVO/IVC','EVO/EVC');}

% Intake manifold pressure & flowrate curves (add/remove comment to
% include flowrate curve)
\text{figure;}
\text{curve1 = line(CAD,AvgFlow,'Color','r');}
\text{ax1 = gca;}

set(ax1,'XLim',[0 720]);
set(ax1,'XTick',[0 90 180 270 360 450 540 630 720]);
xlabel('CAD');
ylabel('Volume Flowrate [SCFM]');
ax2 = axes('Position',get(ax1,'Position'),'XAxisLocation','bottom',...
    'YAxisLocation','right','Color','none','XColor','k','YColor','k');
curve2 = line(CAD,AvgFlow,'Color','b','Parent',ax2);
set(ax2,'XLim',[0 720]);
set(ax2,'XTick',[0 90 180 270 360 450 540 630 720]);
ylabel('Intake Mass Flow [SCFM]');
head = sprintf('Speed = %1.0f RPM, Intake Temperature = %1.1f ^oC,'...
    'Intake Pressure = %1.1f kPa \n', Speed, InTemp, mean(AvgIP));
head2 = sprintf('Bomb Temperature = %1.0f ^oC,'...
    'Coolant Temperature = %1.0f ^oC', BombTemp, CoolTemp);
header = [head head2];
title(header);

% Indicated cylinder mass based on bomb temperature and pressure
% history
figure;
curve1 = line(CAD,Mfrac,'Color','b');
ax1 = gca;
set(ax1,'XLim',[0 720]);
set(ax1,'XTick',[0 90 180 270 360 450 540 630 720]);
ylabel('System Mass Fraction');
xlabel('CAD');

% P-v diagram of cycle, with ideal polytropic compression line
figure;
curve1 = loglog(Volume,AvgBP,'Color','r');
ax1 = gca;
curve2 = line(Volume(900:1799),PTrope,'Color','b');
grid on;
xlabel('Volume (cm^-3)');
ylabel('Pressure (kPa)');

C.2 EngModel.m

%* EngModel.m
%* by Dave Wager
%* This file generate a number of engine geometry and timing
%* parameters frequently required in other routines.
% Generate array of crank angles
for i = 1:1:3600;
    CAD(i) = (i-1)*0.2;
end

% Generate cylinder geometry model
Crank = 5.715;
ConRod = 25.4;
Bore = 8.255;
Vc = 39.44;

% Calculate stroke, piston speed, volume, and volume derivative for full cycle.
for i=1:1:length(CAD);
    stroke(i) = Crank*cos(CAD(i)*pi/180) +
                sqrt(ConRod^2 - Crank^2*(sin(CAD(i)*pi/180)^2));
    dsdt(i) = (-Crank*sin(CAD(i)*pi/180) -
               Crank^2*sin(2*CAD(i)*pi/180)/(2*sqrt(ConRod^2-
                 Crank^2*(sin(CAD(i)*pi/180))^2)))*pi/180;
    Volume(i) = Vc + pi*Bore^2/4*(ConRod + Crank - stroke(i));
    dVdt(i) = -pi*Bore^2/4*dsdt(i);
end

C.3 Blowby.m

% Blowby.m
% by Dave Wager
% This program is used to calculate the extent of blowby losses for a given averaged set of motored cylinder data. Initial mass is determined from P, T & V values at a specified point, before the onset of significant losses. This is compared to the remaining mass at TDC (360 CAD, or 1800 CDM markers).

Intake = Speed/120; % Number of intake events per second

% Initial mass determined from properties partway into compression stroke, where bomb and cylinder conditions are assumed uniform.
P1 = AvgBP(1400);
T1 = AvgT(1400);
V1 = Volume(1400);
R = 287;

% Top dead centre conditions
P2 = AvgBP(1800);
T2 = AvgT(1800);
V2 = Volume(1800);

% Conditions at end of exhaust stroke, to determine residual mass
P3 = AvgBP(3600);
T3 = AvgT(3600);
V3 = V2;

m1a = P1*V1/(1000*R*T1); % Mass prior to blowby onset
m1b = mean(AvgM(1300:1500))/1000; % Alternative initial mass
mTDC = P2*V2/(1000*R*T2); % Mass at end of compression stroke
m3 = P3*V3/(1000*R*T3); % Mass at end of exhaust stroke

BBa = (m1a - mTDC)/(m1a)*100
BBb = (m1b - mTDC)/(m1b)*100

% Estimated mass flowrate, based on initial cylinder charge less
% residual mass (kg/s).
MFlow = Intake*(m1-m3);
VolFlow = (MFlow*60*3.28084^3)/0.77326; % Convert to SCFM from kg/s

C.4 LPFil.m

% LPFil.m
% Original script by Mark Fabbroni
% Current revision by Dave Wager
% This script takes an input signal 'X' and filtering window 'N'
% (where N must be odd), and produces a smoothed output 'signal'
% by applying a running average. Larger N values will produce
% greater smoothing at the expense of resolution of the output.

function [signal] = LPFil(X,N);

points = N/2 - 0.5;
for i = (points+1):1:(length(X)-points);
    sum = 0;
    % Sum and average input values over window of length N, centered
    % on index i of input signal.
    for j = -points:1:points;
        sum = sum + X(i+j);
    end
    X(i) = sum/N;
end
Appendix C. Matlab Scripts & Functions

C.5 BDCTemps.m

% * BDCTemps.m
% * by Dave Wager
% *
% * This function calculates a set of Bottom Dead Centre temperatures
% * for the combined CFR cylinder/combustion bomb system. Starting
% * from a selected point in the compression stroke (at which
% * temperature is assumed to be uniform between cylinder and bomb),
% * temperature is extrapolated back using a polytropic index to
% * determine the average system pressure at BDC. Output is a matrix
% * containing intake manifold temperature, bomb wall temperature and
% * extrapolated BDC temperature.

% BDCTemps is called with a string 'prefix' and the number of files
% to be processed.
function[temps] = BDCTemps(prefix,files)
  filepath = ('Directory where data files are stored');

  % Generate array of crank angles
  for i = 1:1:3600;
    CAD(i) = (i-1)*0.2;
  end

  % Create cylinder geometry and calculate stroke,
  % piston speed and volume.
  Crank = 5.715;
  ConRod = 25.4;
  Bore = 8.255;
  Vc = 39.44;
  for i = 1:length(CAD);
    stroke(i) = Crank*cos(CAD(i)*pi/180) + sqrt(ConRod^2 - ...
      Crank^2*(sin(CAD(i)*pi/180)^2));
    dsdt(i) = (-Crank*sin(CAD(i)*pi/180) - ...
      Crank^2*sin(2*CAD(i)*pi/180)/(2*sqrt(ConRod^2 - ...
        Crank^2*(sin(CAD(i)*pi/180))^2)))*pi/180;
    Volume(i) = Vc + pi*Bore^2/4*(ConRod + Crank - stroke(i));
  end
for k = 1:files;

% Build file name from prefix and current number being processed.
fname = sprintf('s%d',prefix,k);
X = dlmread([filepath fname '.txt'],
            ,'

% Read setpoints from file header
InTemp = X(1,2);
BombTemp = X(1,4);

% Initialize temperature array
AvgT = zeros(3600,1);

% Sum properties over 12 motored cycles
point1 = 3;
point2 = 3602;
for i = 1:1:12;
    AvgT = AvgT + X(point1:point2,1);
    point1 = point2 + 1;
    point2 = point1 + 3599;
end

AvgT = AvgT/12; % Average motored cycle temperature profile

% Extrapolate from (uniform) temperature and volume to BDC temp.
TBDC = mean(AvgT(1400:1600))*(mean(Volume(1400:1600))/...,
Volume(900))^-0.3;
temps(k,:) = [InTemp BombTemp TBDC];
end

C.6 BatchMass.m

%* BatchMass.m
%* by Dave Wager
%* This script takes as input a prefix string specifying a series of
%* motored data sets, and the number of consecutive files in the set.
%* The function returns a matrix consisting of the calculated mass
%* fraction within the bomb over a specified crank angle range for
%* each file.

function[masses] = BatchMass(prefix,files);
filepath = ('Data file directory');
% Get engine geometry and crank angles.
EngModel;

for k = 1:1:files
% Read next file in the series into X.
fname = sprintf('%s%d',prefix,k);
X = dlmread([filepath fname '.txt'],
            't');

% Read setpoints from file header
Speed = X(1,1);
InTemp = X(1,2);
CoolTemp = X(1,3);
BombTemp = X(1,4);
OilTemp = X(1,5);

% Initialize calculated property arrays
AvgBP = zeros(3600,1); % Bomb pressure
AvgIP = zeros(3600,1); % Manifold pressure
AvgT = zeros(3600,1); % Bomb temperature
AvgM = zeros(3600,1); % Trapped cylinder mass
Mfrac = zeros(3600,1); % Mass fraction

% Sum properties over 12 motored cycles
point1 = 3;
point2 = 3602;

for i = 1:1:12;
    % Convert MAP voltage to pressure and add to existing vector
    AvgIP = AvgIP + 65.831*X(point1:point2,3) - 3.325;
    % Convert piezo voltage to pressure. Voltages are referenced to % BDC by subtracting transducer reading & adding intake manifold % pressure.
    AvgBP = AvgBP + 1133.6*X(point1:point2,2) - ...
        1133.6*X(point1+449,2) + 65.831*X(point1+449,3) - 3.325;
    AvgT = AvgT + X(point1:point2,1);
    point1 = point2 + 1;
    point2 = point1 + 3599;
end

% Calculate average motored cycle profile for each property
AvgIP = AvgIP/12;
AvgBP = AvgBP/12;
AvgT = AvgT/12;
%% Use volume, with pressure and temperature profiles from above to
%% calculate indicated mass at each crank position.
for i=1:1:length(CAD);
    AvgM(i) = AvgBP(i)*Volume(i)/(287*AvgT(i));
end

%% Define trapped mass over some interval of the cycle. Calculate mass
%% fraction for full cycle by dividing by total mass.
Mtotal = mean(AvgM(1450:1550));
Mfrac = AvgM/Mtotal;

%% Return mass fraction for desired portion of the cycle.
masses(:,k) = Mfrac(1850:1950);
end

C.7 MassFit.m

%% MassFit.m
%% by Dave Wager
%%
%% This function returns the coefficients of a non-linear least squares
%% fit to a chosen functional form. Input is a text file containing a
%% matrix of mass data calculated from motored pressure and temperature
%% traces. Mass data files are generated by the BatchMass.m script.
%% This script requires the Matlab Curve Fitting Toolbox for the
%% lsqcurvefit function.
function[coeff] = MassFit(Prefix)
filepath = ('Data file directory');

%% Read mass data file as matrix X.
fname = sprintf('%s',Prefix);
X = dlmread([filepath fname '.txt'],
D = size(X);

%% Generate crank angle vector equal to length of dataset to be fit.
%% Insert the appropriate starting angle into the loop below manually.
for i = 1:1:D(1);
    CAD(i,1) = 369.8 + i*0.2;
end

%% For each column of mass data in X, either fit the coefficients of
% the desired function, or calculate the value needed to make the
% function overlap at some desired point.
for j = 1:1:D(2);
    clear x;
clear mfrac;
% clear A;
mfrac = X(:,j);
% Use the following line to generate coefficients for
% the curve fit.
x = lsqcurvefit(@(x,CAD) x(1) - (x(5)-x(1))*exp(-1*((CAD - ...
    x(2))/x(3)).^x(4)),[0.7,360,23,7,0.95],CAD,mfrac);
% Use the following line to calculate the value needed to make the
% function overlap at TDC. Manually input coefficients determined
% previously.
%A = (mfrac(D(1)) - exp(-1*((CAD(D(1)) - ...
% 294.35)/52.42)^5.23))/(1 - exp(-1*((CAD(D(1)) - ...
% 294.35)/52.42)^5.23))
coeff(j,:) = x;
end

C.8 DAQAnalysis.m

% DAQAnalysis.m
% by Dave Wager, adapted from original script by Mark Fabbroni
%
% This script generates the main results for a single set of firing
% test data: ignition delay, instantaneous and cumulative heat
% release and combustion efficiency.
path = ('File path');
fname = ('File name');
tic; % Start counter for reference during analysis.
X = dlmread([path fname '.txt'],
            [path fname '.txt']
            [path fname '.txt']
            [path fname '.txt']
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            [path fname '.txt']
            [path fname '.txt']
            [path fname '.txt']
            [path.fname]('% File path');
            % Start counter for reference during analysis.
            tic;
            X = dlmread([path fname '
            tic;
X = dlmread([path fname '.txt'],
            % User setpoints. Adjust to conditions as necessary.
            Nozzle = 1; % 1 = round, 2 = elliptical
            FIDCal = 1482; % Slope of FID calibration curve (PPM/V)
            LHV = 47.0; %MJ/kg
            InjDelay = 0.6; % Injector opening delay (ms)
            % Read equipment setpoints from file header.
            Speed = X(1,1); % RPM
            InTemp = X(1,2); % C
            CoolTemp = X(1,3); % C
BombTemp = X(1,4); % C
OilTemp = X(1,5); % F
GPTemp = X(2,1);
GPPos = X(2,2);
GPNum = X(2,3);
InjAngle = X(2,4);
WinFrom = X(2,5);
WinTo = X(3,1);
BurstTot = X(3,2);
InjNum = X(3,3);
CDMperCAD = X(3,4);
InjDur = X(3,5); % CAD
Skip = X(4,1);
SOI = X(4,2); % CAD
GPPower = X(4,3); % W
ExTemp = X(4,4);

sprintf('Data file read in %1.2f seconds',toc)

% First block of values for results file
res1 = [Speed InTemp CoolTemp BombTemp OilTemp ...
   GPTemp GPPos GPNum InjAngle WinFrom
   WinTo BurstTot InjNum CDMperCAD InjDur ...
   Skip SOI GPPower ExTemp 0];

% Determine fired cycles. The vector 'FiredCycles' will end up
% containing 10 values, indicating which of the 55 recorded cycles
% contained injection/combustion events.
c = 5;
% Find index where current rises above its base value.
while(X(c,4) < 2);
   c = c + 1;
end
% Rounding down towards zero, store the first fired cycle number.
FiredCycles(1) = fix(c/3600)+1;

% Subsequent fired cycles are determined by the skip fire number.
for i = 2:1:11;
   FiredCycles(i) = FiredCycles(1) + (Skip+1)*(i-1);
end

% Get crank angle positions and engine geometry.
EngModel;
Volume = Volume/1000; % L
\begin{verbatim}
dVdt = dVdt/1000; \% L/CAD

% Initialize vectors for storage of motored cycle data.
AvgBP = zeros(3600,1); \% Average Bomb Pressure
AvgIP = zeros(3600,1); \% Average Intake Pressure
AvgNGP = zeros(3600,1); \% Average Natural Gas Pressure

% Intialize counters to bracket cycles.
pt1 = 5;
pt2 = 3604;

% Sum properties across all non-fired cycles.
for i=1:1:55;
    if (i ~= FiredCycles(1) & i ~= FiredCycles(2) & ... 
        i ~= FiredCycles(3) & i ~= FiredCycles(4) & ... 
        i ~= FiredCycles(5) & i ~= FiredCycles(6) & ... 
        i ~= FiredCycles(7) & i ~= FiredCycles(8) & ... 
        i ~= FiredCycles(9) & i ~= FiredCycles(10) & ... 
        i ~= FiredCycles(11));
        AvgIP = AvgIP + 65.831*X(pt1:pt2,5) - 3.325;
        AvgBP = AvgBP + 1133.6*X(pt1:pt2,2) - 1133.6*X(pt1+449,2) ... 
                + 65.831*X(pt1+449,3) - 3.325;
        AvgNGP = AvgNGP + 3651.81*X(pt1:pt2,3) + 92.18;
    end
    pt1 = pt2 + 1;
    pt2 = pt1 + 3599;
end

% Divide vectors by constant to generate average pressure traces.
AvgIP = AvgIP/44; \% kPa
AvgBP = AvgBP/44; \% kPa
AvgNGP = AvgNGP/44; \% kPa
GP = mean(AvgNGP);

% Discard the first fired cycle with incorrect injection timing.
FiredCycles = FiredCycles(2:11);
res1(3,:) = FiredCycles;

% Determine injected fuel mass
if (Nozzle == 1)
    if (InjDur == 3.0)
        FuelMass = 1.63*InjDur*0.725 + 1.41*GP/1000 - 10.6; \% mg
    else
        FuelMass = 4.17 - 0.6313*(11000 - GP)/1000; \% mg
    end
end
\end{verbatim}
else if (Nozzle == 2)
    if (InjDur == 3.0)
        FuelMass = 1.83*InjDur*0.725 + 1.5*GP/1000 - 10.4; % mg
    else
        FuelMass = 1.0948*GP/1000 - 5.5379; % mg
    end
end
end

FuelNRG = LHV*FuelMass; % [MJ/kg*mg] = J

% Store measured values for fired cycles. Each cycle will be stored % as a separate column within the array for a given property.
for i = 1:1:10;
    pt1 = (FiredCycles(i) - 1)*3600 + 5;
    pt2 = pt1 + 3599;
    FiredFID(:,i) = X(pt1:pt2,1); % PPM
    FiredBP(:,i) = 1133.6*X(pt1:pt2,2); % kPa
    InjCurr(:,i) = 4*X(pt1:pt2,4);
    FiredIP(:,i) = 65.831*X(pt1:pt2,3) - 3.325; % kPa
    FiredBP(:,i) = FiredBP(:,i) - FiredBP(450,i) + FiredIP(450,i);
end

% Use Ignition routine to calculate ignition delays for fired cycles.
for i = 1:1:10;
    [a(i), b(i), IgnDelay(i), ActInjDur(i), Pdiff(:,i)] = ... 
        Ignition(CAD, FiredBP(:,i), AvgBP, InjCurr(:,i),...
            SOI, Speed, InjDelay);
end
res1(4,:) = a; % CAD
res1(5,:) = b; % CAD
res1(6,:) = IgnDelay; % ms
res1(7,:) = ActInjDur; % ms

sprintf('Ignition delays calculated after %1.2f seconds',toc)

% Use correlations from motored data to determine temperatures at key % cycle points
Tbdc = 265 + 0.0597*InTemp + 0.646*BombTemp; % K
Tpk = 599 + 0.252*InTemp + 0.649*BombTemp - 0.0014*max(AvgBP); % K
Ttdc = 1.03*Tpk - 0.169*BombTemp; % K

% With calculated temperatures and measured (non-firing) pressures, % calculate key cylinder masses, and mass fraction at TDC.
Mtotal = AvgBP(900)*Volume(900)/(Tbdc*287); % kg
Mtdc = AvgBP(1800)*Volume(1800)/(Ttdc*287); % kg
MFttdc = Mtdc/Mtotal;

% Set mass fraction to 1.0 for initial portion of compression stroke
for i = 1:1:600;
    Mfrac(i,1) = 1;
end

% Calculate TDC intercept for first mass fraction correlation.
B1 = 288.1038; C1 = 58.3361; D1 = 5.6574; E1 = 1.0161;
A1 = (MFttdc - E1*exp(-1*((360 - B1)/C1)^D1))/(1 - exp(-1*((360 - B1)/C1)^D1));

% Calculate mass fraction over range of first correlation
for i = 601:1:950
    Mfrac(i,1) = A1 + (E1 - A1)*exp(-1*(((179.8 + 0.2*i)-B1)/C1)^D1);
end

% Determine intercept point for second correlation.
B2 = 359.9848; C2 = 22.9818; D2 = 7.0111; E2 = 0.9653;
A2 = (Mfrac(950,1) + E2*exp(-1*((370 - B2)/C2)^D2))/(1 + exp(-1*((370 - B2)/C2)^D2));

% Calculate mass fraction over range of second correlation
for i = 951:1:1050
    Mfrac(i,1) = A2 - (E2 - A2)*exp(-1*(((179.8 + 0.2*i) - B2)/C2)^D2);
end

% Set mass fraction to constant for remainder of expansion stroke
for i = 1051:1:1800
    Mfrac(i,1) = Mfrac(1050,1);
end

for i = 1:1:3600
    if i < 901;
        Mass(i) = Mtotal; % kg
    else if i < 2701;
        Mass(i) = Mfrac(i-900)*Mtotal; % kg
    else
        Mass(i) = 0; % kg
    end
end

% Calculate mass rate of change for full cycle.
dmdt = derivative(CAD,Mass,1,4,17,0); % kg/CAD

sprintf('Heat release calculations starting after %1.2f seconds',toc)

% Specific heat correlation constants
alph = 3.653; beta = -1.337*10^-3; gam = 3.294*10^-6;
del = -1.913*10^-9; eps = 0.2763*10^-12;

%Calculate heat release curve for average motored cycle
for i = 1:1:3600
    if i < 901;
        AvgBT(i) = Tbdc;
    else if i < 2700;
        AvgBT(i) = AvgBP(i)*Volume(i)/(Mtotal*Mfrac(i-900)*287);
    else
        AvgBT(i) = AvgBT(2699);
    end
end

% Differentiate average motored pressure trace
dPavgdt = derivative(CAD,AvgBP',1,4,17,0);

% Calculate ROHR for average motored cycle.
for i = 1:1:3600
    cp = .287*(alph + beta*AvgBT(i) + gam*AvgBT(i)^2 + del*AvgBT(i)^3 + eps*AvgBT(i)^4); % kJ/kg/K
    cv = cp - 0.287; % kJ/kg/K
    dcvdT = beta + 2*gam*AvgBT(i) + 3*del*AvgBT(i)^2 + 4*eps*AvgBT(i)^3; % kJ/kg/K^2
    MotorHR(i) = (cv + AvgBT(i)*dcvdT)*Volume(i)*dPavgdt(i)/0.287 + ...
                 (cp + AvgBT(i)*dcvdT)*AvgBP(i)*dVdt(i)/0.287 + ...
                 1000*(0.5*cp*(AvgBT(i) + BombTemp + 273.15) + ...
                       AvgBT(i)^2*dcvdT)*dmdt(i); % J/CAD
end

NetHeat = zeros(3600,10);
if (InjDur == 1.4);
    BurnCalc = 120;
else
    BurnCalc = 150;
end

% Calculate results for fired cycles.
for j = 1:1:10
    % Generate calculated temperatures for full cycle.
    for i = 1:1:3600
        if i < 901; % Intake stroke
            Tcalc(i,j) = Tbdc;
        else if i < 2700; % Compression & expansion strokes
            Tcalc(i,j) = FiredBP(i,j)*Volume(i)/
                         (Mtotal*Mfrac(i-900)*287);
        else
            Tcalc(i,j) = Tcalc(2699); % Exhaust stroke
        end
    end
end

dPdt(:,j) = derivative(CAD,FiredBP(:,j)',1,4,17,0); % kPa/CAD

for i = 1:1:3600
    cp = .287*(alph + beta*Tcalc(i,j) + gam*Tcalc(i,j)^2 + ...
               del*Tcalc(i,j)^3 + eps*Tcalc(i,j)^4); % kJ/kg/K
    cv = cp - 0.287; % kJ/kg/K
    % Manually differentiate cv with respect to temperature
    dcvdT = beta + 2*gam*Tcalc(i,j) + 3*del*Tcalc(i,j)^2 + ...
           4*eps*Tcalc(i,j)^3; % kJ/kg/K^2
    %r1(i,j) = (cv + Tcalc(i,j)*dcvdT);
    %r2(i,j) = Volume(i);
    %r3(i,j) = dPdt(i,j);
    ROHR(i,j) = (cv + ...
                  Tcalc(i,j)*dcvdT)*Volume(i)*dPdt(i,j)/0.287 + ...
                 (cp + Tcalc(i,j)*dcvdT)*FiredBP(i,j)*dVdt(i)/0.287 + ...
                 1000*(0.5*cp*(Tcalc(i,j) + BombTemp + 273.15) + ...
                      Tcalc(i,j)^2*dcvdT)*dmdt(i); % J/CAD
    ROHR(i,j) = ROHR(i,j) - MotorHR(i);
end

% Calculate average output of FID signal
FIDAvg(j) = 0;
for i = 1:1:900;
    FIDAvg(j) = FIDAvg(j) + FiredFID(510*5+i,j);
end
FIDAvg(j) = FIDCal*FIDAvg(j)/900; % PPM (molC/mol*10^6)

% Calculate combustion efficiency
UBFuelMass(j) = 1000*16/28*FIDAvg(j)*Mass(2500)/10^6; % gCH4
CombEff(j) = 1 - 1000*UBFuelMass(j)/FuelMass; % [1000*g/mg]
% Calculate net heat release for a fixed portion of the cycle
ign = uint16(5*b(j));
for i = ign:1:(ign + BurnCalc);
    if (ROHR(i,j) > 0);
        NetHeat (i+1,j) = NetHeat(i,j) + ... 
            0.2*0.5*(ROHR(i,j)+ROHR(i+1,j)); % J
    else
        NetHeat(i+1,j) = NetHeat(i,j);
    end
end

subROHR = ROHR(ign:(ign+BurnCalc),j);
[ROHRmax(j),MaxLoc(j)] = max(subROHR);
MaxLoc(j) = MaxLoc(j) + ign;

% Look for point where net heat release reaches 5% of its final
% value to define burn start.
c1 = ign;
while(NetHeat(c1,j) < 0.05*NetHeat((ign+BurnCalc),j));
    c1 = c1 + 1;
end
startburn(j) = c1;
c2 = startburn(j) + 1;

% Look for point where net heat release stabilizes for 5
% consecutive points to define burn end
while(~(NetHeat((c2-5),j) == NetHeat(c2,j)));
    c2 = c2 + 1;
end
endburn(j) = c2;
BurnDur(j) = endburn(j) - startburn(j);

% Fill out remainder of cycle with end value of Net Heat Release.
NetHeat((c2+1):length(CAD),j) = NetHeat(c2,j);
sprintf(['Heat release calculations complete'
    'for cycle %.1f after %.1f seconds'],j,toc)
end

res1(8,:) = FIDAvg;
res1(9,:) = CombEff;
res1(10,:) = CombEff2;
res1(11,:) = startburn;
res1(12,:) = endburn;
res1(13,:) = BurnDur;
res1(14,:) = ROHRmax;
res1(15,:) = MaxLoc;

dlmwrite([path1 path2 fname '_results.txt'],res1,'\t');

% Append heat release data to results file
dlmwrite([path fname '_results.txt'],...
    ROHR,'delimiter','\t','-append');
dlmwrite([path fname '_results.txt'],...
    NetHeat,'delimiter','\t','-append');

C.9 Derivative.m

% Derivative.m
% Original script by Mark Fabbroni
% Current revision by Dave Wager
%
% This function takes an input signal ('signal') and reference
% ('CAD'), and calculates the 'P'th order derivative by incrementally
% applying a Savitsky-Golay filter using a polynomial of selected
% order ('order') to a data window (of length 'frame'). See MatLab
% documentation for details of the filter design and derivative
% estimation. Input with Figs = 1 to enable plot of results.

function[dXdt] = derivative(CAD,signal,P,order,frame,Figs)

% Initialize variables & get filter coefficients
Der = 0; B = 0; G = 0;
[B,G] = sgolay(order,frame);

% Incrementally estimate derivative of order P for central points of
% input signal. Note that initial and final (frame-1)/2 points will
% be excluded. Note also that derivatives are calculated at 0.2 CAD
% intervals. If the desired output is rate of change relative to CAD,
% the resulting Der signal must be multiplied by 5.
for i = 1:1:(length(signal)-frame);
    Der(i - 1 + (frame + 1)/2) = ...
        (factorial(P))*G(:,P + 1)'*signal(i:i+frame-1)';
end

% Fill out trailing portion of signal derivatives.
for i = (length(signal) - frame):1:length(signal);
    Der(i) = 0;
end
dXdt = 5*Der;

% If figure option is enabled in input, generate a plot of initial
% signal, filtered signal, and estimated derivatives.
if (Figs == 1);
    Fit = sgolayfilt(signal,order,frame);
    limits = [0 720];
    ticks = [0:90:720];
    figure;
    h11 = line(CAD,signal,'LineStyle','none','Marker',...
               'o','Color','r');
    h13 = line(CAD,Fit,'Color','g');
    ax1 = gca;
    set(ax1,'XColor','k','YColor','k');
    ylabel('Signal(Units)');
end

C.10 Ignition.m

% Ignition.m
% Original script by Mark Fabbroni
% Current revision by Dave Wager
% This script uses inputs: CAD (cycle crank angle positions); FiredBP
% (a pressure trace from a single fired cycle); AvgBP (averaged
% motored cycle pressure); InjCurr (injector current trace); SOI
% (crank angle at start of injection); Speed (engine speed in RPM)
% and InjDelay (physical injector opening delay); and returns
% outputs: a (crank angle position at start of injector pulse); b
% (crank angle position where pressure rise indicates start of
% combustion); IgnDelay (ignition delay in ms); and ActInjDur
% (actual injection duration adjusted for opening and closing delays).

function[a,b,IgnDelay,ActInjDur] = ...
    Ignition(CAD,FiredBP,AvgBP,InjCurr,SOI,Speed,InjDelay)

% Vector of pressure differential between average motored bomb
% pressure and one cycle’s fired bomb pressure.
Pdiff = FiredBP - AvgBP;

% Determine "steady" values (average and standard deviation) of
% pressure and current during the compression stroke, prior to the
% onset of injection/combustion.
Pbase = Pdiff((SOI-100)*5:SOI*5);
Cbase = InjCurr((SOI-100)*5:SOI*5);
PdiffAvg = mean(Pbase);
PdiffStd = std(Pbase);
CAvg = mean(Cbase);
CStd = std(Cbase);
P3Sigma = PdiffAvg + 3*PdiffStd;
C3Sigma = CAvg + 3*CStd;

% Determine the point at which injector current crosses the 3 sigma 
% line. The CAD index at this point is returned as 'a'
c1 = (SOI - 250)*5;

while ~(InjCurr(c1) > C3Sigma & InjCurr(c1+1) > C3Sigma ... 
    & InjCurr(c1+2) > C3Sigma);
    c1 = c1 + 1;
end
a = CAD(c1);

% Continuing from the index found above, find the point where injector 
% current drops back below the 3 sigma point. The difference between 
% these two indices indicates the duration of the injection (minus 
% any physical delay associated with the response of the injector.
c1 = c1 + 1;
while(InjCurr(c1) > C3Sigma);
    c1 = c1 + 1;
end
d = CAD(c1);

% Determine the point where pressure differential between motored and 
% fired traces crosses the 3 sigma line by starting at peak pressure 
% and working backwards. This point indicates the onset of combustion. 
% The CAD index at this point is returned as 'b'.
[m,c2] = max(LPFilter(Pdiff,5));
while ~(Pdiff(c2-1) < P3Sigma);
    c2 = c2 - 1;
end
b = CAD(c2);

% Calculate ignition delay as the time between the start of the 
% injector pulse and the onset of combustion determined from the 
% pressure trace above.
IgnDelay = 1000*(b-a)/(Speed*6)-InjDelay;

% Calculate actual injection duration.
ActInjDur = 1000*(d-a)/(Speed*6)-InjDelay;

% Uncomment the following section to include a plot of injector
% current and pressure differential with average and 3-sigma lines.
figure;
l1 = line(CAD(1750:1800),Pdiff(1750:1800),'Color','k');
ax1 = gca;
set(ax1,'XLim',[350 360]);
set(ax1,'XTick',[350 352 354 356 358 360]);
xlabel('CAD');
ylabel('Pressure [kPa]');
l2 = line(get(ax1,'XLim'),[PdiffAvg,PdiffAvg],'LineStyle','--');
l3 = line(get(ax1,'XLim'),[P3Sigma,P3Sigma],'LineStyle','--');
ax2 = axes('Position',get(ax1,'Position'),'XAxisLocation','bottom','...
'YAxisLocation','right','Color','none','XColor','k','YColor','k');
curve2 = line(CAD(1750:1800),InjCurr(1750:1800),...
'Color','r','Parent',ax2);
set(ax2,'X Lim',get(ax1,'X Lim'));
set(ax2,'XTick',get(ax1,'XTick'));
ylabel('Injector Current [A]');

C.11 FIDAnalysis.m

% FIDAnalysis.m
% by Dave Wager
%
% This script functions mainly to generate output vectors of average
% concentration/FID signal for the selected portion of a cycle.
% Output vectors are later compiled into contour plots using the
% FIDContour.m script. Additional data in this file are left in to
% allow additional plotting/analysis as necessary.

clear all
X = dlmread('File path & name','\t');

Speed = X(1,1);
InTemp = X(1,2);
CoolTemp = X(1,3);
BombTemp = X(1,4);
OilTemp = X(1,5);
DP_FID = X(2,1);
CPVac = X(2,2);
GPNum = X(2,3);
InjAngle = X(2,4)
WinFrom = X(2,5);
WinTo = X(3,1);
BurstTot = X(3,2);
InjNum = X(3,3);
CDMperCAD = X(3,4);
InjDur = X(3,5);
Skip = X(4,1);
SOI = X(4,2);
FIDGain = X(4,3);
ExTemp = X(4,4);

% Determine fired cycles. The vector 'FiredCycles' will end up containing 10 numbers, indicating which of the 55 cycles contained injection events.
c = 5;
% Look for index where the current pulse rises above its base value.
while(X(c,4) < 2);
    c = c + 1;
end
% Rounding down towards zero, store the first fired cycle number.
FiredCycles(1) = fix(c/3600) + 1;
for i = 2:1:11;
    FiredCycles(i) = FiredCycles(1) + (Skip+1)*(i-1);
end
% Discard the first fired cycle with incorrect injection timing.
FiredCycles = FiredCycles(2:11);

% Get crank angle positions and engine geometry
EngModel;

% Initialize vectors for storage of motored cycle data.
AvgBP = zeros(3600,1); % Average Bomb Pressure
AvgIP = zeros(3600,1); % Average Intake Pressure
AvgNGP = zeros(3600,1); % Average Natural Gas Pressure

% Intitalize counters to bracket cycles.
pt1 = 5;
pt2 = 3604;

% Sum pressures across all non-fired cycles.
for i=1:1:55;
    if (i ~= FiredCycles(1) & i ~= FiredCycles(2) & ... & i ~= FiredCycles(3) & i ~= FiredCycles(4) & ...
i ~= FiredCycles(5) & i ~= FiredCycles(6) &...
i ~= FiredCycles(7) & i ~= FiredCycles(8) &...
i ~= FiredCycles(9) & i ~= FiredCycles(10));

AvgIP = AvgIP + 65.831*X(pt1:pt2,3) - 3.325;
AvgBP = AvgBP + 1133.6*X(pt1:pt2,2) - 1133.6*X(pt1+449,2) + ...
65.831*X(pt1+449,3) - 3.325;
AvgNGP = AvgNGP + 3651.81*X(pt1:pt2,3) + 92.18;
end
pt1 = pt2 + 1;
pt2 = pt1 + 3599;
end

% Divide vectors by constant to generate average pressure traces.
AvgIP = AvgIP/44;
AvgBP = AvgBP/44;
AvgNGP = AvgNGP/44;

% Store measured values for fired cycles. Each cycle will be stored
% as a separate column within the array for a given property.
for i = 1:1:10;
    pt1 = (FiredCycles(i) - 1)*3600 + 5;
    pt2 = pt1 + 3599;
    FiredFID(:,i) = X(pt1:pt2,1);
    InjCurr(:,i) = 4*X(pt1:pt2,4);
end

% Average FID signals over 10 fired cycles.
FIDAvg = zeros(3600,1);
for i = 1:1:10;
    FIDAvg = FIDAvg + FiredFID(:,i);
end
FIDAvg = FIDAvg/10;

% Extract desired portion of average FID signal. Use SD to calculate
% variation between cycles.
c1 = 1775;
for i=1:1:35;
    c1 = c1 + 1;
    Conc(i,1) = FIDAvg(c1);
    SD(i) = std(FiredFID(c1,:));
end

% Return average and standard deviations of concentration.
Conc
SD'
C.12 CompositeImg.m

% CompositeImg.m
% by Dave Wager
% This script creates composite images from a series of individual
% image strips. Input requires a string with the file prefix of the
% series of images, and the number of files to be processed.

Prefix = ('File prefix');
Files = 7;
Path = ('File path');

for g = 1:1:Files;
    for h = 1:1:10;
        fname = sprintf('%s_%d_%d',Prefix,g,h);

        % Read next image in sequence and append to composite.
        baseimg = imread([Path fname '.png']);
        composite(139*(h-1)+1:139*h,:) = baseimg;
    end

    % Write complete composite to file and proceed to next set of
    % images.
    outfile = sprintf('%s_%d',Prefix,g);
    imwrite(composite,[Path outfile '.png']);
    clear composite,
    clear outfile;
end

C.13 ImgFrames.m

% ImgFrames.m
% by Dave Wager
% This script creates individual frames from a series of image strips.
% Input requires a string with the file prefix of the series of
% images, and the number of consecutive files to be processed.

Prefix = ('File prefix');
Files = 9;
Path = ('File path');
for g = 1:1:Files;
    for h = 1:1:10;
        fname = sprintf('%s_%d_%d',Prefix,g,h);

        % Read next image in sequence.
        baseimg = imread([Path fname '.png']);

        % Crop the base image to desired dimensions, removing unused
        % portions at the top and bottom of the strip.
        cropimg = baseimg(80:218,:);

        for i = 1:1:10;
            % Crop out the 'i'th frame of the strip.
            f = cropimg(:,(104*i-103):104*i);
            % Generate a filename for the cropped frame.
            s = sprintf('%s_%d_%d_frame%d',Prefix,g,h,i);
            % Write the cropped frame to a new file.
            imwrite(f,[Path s '.png']);
            clear f;
            clear s;
        end
        imwrite(cropimg,[Path fname '.png']);
    clear fname;
    clear baseimg;
    clear cropimg;
end
end

C.14 ImgHist.m

%* ImgHist.m
%* by Dave Wager
%* Given a file prefix and number of files in the series, this script
%* calculates histograms for individual frames, assuming 10 frames
%* per image strip.

Prefix = ('no4');
Files = 8;

for g = 1:1:Files;
    for h = 1:1:10;
        for j = 1:1:10;
C.15  HistSort.m

%* HistSort.m
%* by Dave Wager
%* Companion subroutine to ImgHist.m. Given same prefix and number of
%* files, this routine sorts the frame histograms for images taken in
%* a single engine run. Assumes 10 images per dataset. Output aligns
%* counts from different images by frame for easier post-processing.
Prefix = ('no4');
Files = 8;
for g = 1:1:Files;
    outfile = zeros(320,10); % Initialize output array
    for h = 1:1:10;
        fname = sprintf('%s_%d_%d',Prefix,g,h);
        X = dlmread(['File path' fname '.txt'],
            for j = 1:1:10;
            outfile((32*(j-1)+1):32*j,h) = X(:,j);
        end
        clear X;
        clear fname;
    end
    outfilename = sprintf('%s_%d',Prefix,g);
    dlmwrite(['File path' outfilename '.txt'],outfile,'
        clear outfile;
    end
end
Appendix D

Instrument Calibrations

D.1 Schaevitz Pressure Transducer

Calibration points for the Schaevitz strain gauge pressure transducer used for this study were taken using a dead weight tester. However, it was noted during testing that the tester had an oil leak that got larger as the test pressure was increased. Comparing the calibration points captured to the data from Abate [27], as shown in Figure D.1, it was found that the current calibration points start out quite close to the predicted values, but begin to diverge at higher pressures. This suggested that the oil leak was causing the dead weight tester to produce lower pressures than expected for the applied weight. As a result, the original calibration from Abate was used in this study.

D.2 Kistler Pressure Transducer

The Kistler piezoelectric pressure transducer does not measure absolute pressures, but only changes in pressure. It was calibrated by inserting the transducer assembly into a small chamber which was then charged to various pressures. The chamber pressure was measured using the Schaevitz strain gauge transducer. After pressurizing the chamber the Kistler transducer output was allowed to stabilize, after which the chamber was rapidly depressurized. The maximum change in voltage was measured and correlated with the pressure change. The resulting calibration curve is shown in Figure D.2
Figure D.1: Schaevitz pressure transducer calibration. Current data points and correlation from Abate.

Figure D.2: Kistler piezoelectric pressure transducer calibration curve.
Appendix D. Instrument Calibrations

D.3 GM Manifold Absolute Pressure Transducer

The GM MAP sensor used to determine intake pressure in this study was calibrated by connecting it to a pressurized chamber which was also monitored by the Schaevitz strain gauge transducer. Various absolute pressures between 1 and 3 bar were applied, and the measurements of the two transducers were then correlated. The resulting curve is shown in Figure D.3.

D.4 TSI Laminar Flow Meter

Calibration data points for non-linear output voltage supplied by the manufacturer were used to generate a calibration curve for the TSI laminar flow meter. The data was fit well by a logarithmic correlation, and the equation determined was subsequently used to calculate the indicated flow rates.

D.5 Matheson Rotameters

Two Matheson rotameters were used to prepare calibration mixtures for the Cambustion FFID. Calibration points were supplied by the manufacturer, and correlations to these
data sets were used to allow automatic calculation of flow rates and concentrations. Air flow rates were measured with a Tube 6 rotameter with a stainless steel float ball, and methane flow rate was measured with a Tube 602 rotameter with a glass ball. The calibration curves and correlations are shown in Figures D.5 and D.6 respectively. The curves shown are for 0 psig within the tubes. Calibration gases were mixed at 10 psig, so pressure correction factors had to be applied to both flow rates. For equal tube pressures the same pressure correction applies, so the overall concentration of the mixture is unaffected.
Figure D.5: Matheson rotameter calibration curve, Tube 6 with stainless steel float.

Figure D.6: Matheson rotameter calibration curve, Tube 602 with glass float.
Appendix E

Glow Plug Calibration and Testing

The following section contains extended details about the calibration and comparison of the three glow plugs evaluated for this study, as introduced in Section 4.1.2. The Wellman W451s were the first plugs to be tested. Initial measurements indicated that the Wellman plugs provided much lower temperatures than the previous model. This was initially thought to be the result of the manufacture of the plugs. However, when one of the original glow plugs was tested to confirm the validity of the calibration curves reported by Fabbroni, it was also found to be much cooler at similar voltage/power levels [28]. While the slope of the temperature-resistance relationship had not significantly changed, the entire curve was shifted towards higher resistances and lower temperatures at all conditions. After examining the data acquisition system, it was determined that the discrepancy likely resulted from differences in the experimental setup between the original and new calibrations.

First, as mentioned previously, the fixture used to hold the glow plug can have a significant impact on steady state temperature. For example, it was found that moving the glow plug from the bomb block to a smaller L-shaped aluminum bracket with greatly reduced contact area led to an increase of nearly 200K for an identical voltage input. The type of fixture used by Fabbroni in generating his calibration curves is unknown, but a different setup could clearly account for a large portion of the observed discrepancy between the new and old calibration curves.

The second change made for the current tests was to alter the location of the voltage probes. In the previous setup voltage was measured across the terminals of the power supply. In this configuration the calculated steady state resistance reflects the resistance of the glow plug, plus the combined resistance of all leads and any contact resistance
Figure E.1: Glow plug temperature vs. resistance data for Wellman W451. Solid lines indicate correlations reported by Fabbroni for previous plugs [28].

present at the connections in the circuit. Lead and contact resistances are quite small but given that changes in resistance of only 0.3 Ω are typical across the full operating range of the glow plug, they are not necessarily negligible. To determine specifically the steady state resistance of the glow plug itself the voltage probes were relocated to measure between the terminal of the plug and the nearest bolt on the bomb block, where the ground wire was also located.

Without details about the previous experimental setup the original values reported by Fabbroni could not be reproduced exactly, but taken together the changes made for the current work account for most of the observed discrepancy.

With the experiment set up as just described, steady state measurements of resistance and surface temperature were taken at various power levels for the Wellman plugs. The results are shown in Figure E.1, along with two representative curves reconstructed from Fabbroni’s correlations.

The figure shows that although the Wellman plugs all have very similar responses they still have significantly higher resistance and lower temperature than the previous plugs, even with the above changes applied. The slope of the temperature-resistance curves is also consistently lower than that of the GM plugs. Tests in the combustion
bomb confirmed that the Wellman glow plugs could not provide fast or reliable ignition at reasonable power levels, as it was found that the plugs had to driven to levels in excess of 120W to achieve 2ms ignition delays. At this level of power the plugs quickly burned out. While the Wellman glow plugs would certainly be capable of performing well within a standard diesel engine the high temperatures demanded for ignition of natural gas are simply out of their range of operation.

The NGK Y-104 plug was tested next; since it is designed to run at a different nominal voltage, comparisons of resistance with the original GM plug are uninformative. However, temperature measurements were still obtained within the range of 50 to 100W of power, the operating range of the previous plugs. The results indicated that the NGK plug could achieve higher surface temperatures than the W541, and was a potentially viable replacement. Tests within the bomb confirmed that the performance was better than the W541, but was still slightly poorer than the original plug.

The next glow plug to be tested was the Denso DG-143. The same measurement procedure was used and the steady state temperature-resistance data, along with the curves from Fabbroni, are shown in Figure E.2.

As the figure indicates, the performance of the Denso plugs relative to the original
model is much more encouraging, as they achieve comparable temperatures to the GM plugs at even lower resistances. The slope of the Denso correlations is still lower than those of the GM plugs, but equal or greater temperatures are indicated for the DG-143 at all points within the range of interest.