Study of Ignition and Emissions in a Direct Injected, Compression Ignition Natural Gas Engine

by

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A thesis submitted in conformity with the requirements for the degree of Master of Applied Science
Graduate Department of Mechanical and Industrial Engineering
University of Toronto

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Abstract

Natural gas direct injection and glow plug ignition assist technologies were implemented in a single-cylinder, optically-accessible engine. Initial experiments studied the effects of injector and glow plug shield geometry on ignition quality. Injector and shield geometric effects were found to be significant, with only two of 20 tested geometric combinations resulting in reproducible combustion. Further experiments explored the effects of equivalence ratio and intake pressure on ignition delay, engine performance, and exhaust emissions. Combustion was found to proceed in a stratified-premixed mode at lower equivalence ratios, and a free-mixing mode at the higher equivalence ratios. Both combustion modes resulted in high NOx emissions. Stratified-premixed combustion produced higher hydrocarbon emissions, and lower levels of particulate matter and carbon monoxide, when compared to free-mixing combustion. Higher intake pressure was found to reduce all emissions levels. This effect was largely attributed to better charge mixing achieved from pressure-driven increase in engine swirl momentum.
Acknowledgments

I would like to sincerely thank Professor James Wallace for taking me on as student, guiding me through this project, and accommodating my research interests. Dr. Wallace has made my Master’s experience very positive and valuable by going beyond the regular duties of a supervisor. Dr. Wallace has been an influential mentor in my technical, academic, professional, and personal development. He has shown incredible patience, which has been tested by my multiple work-related degree start delays, deviations in academic plans, as well as ongoing project issues.

I would also like to thank my spouse Maria Kandaurova. Maria has provided endless support and encouragement over the duration of this thesis project. Beyond support at home, she has helped me with the organization of the lab, reassembly of the data acquisition system, as well as many other tasks. If permitted, I would gladly grant her a part of my degree in recognition of her unwavering support.

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# Nomenclature

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<tr>
<th>Abbreviation</th>
<th>Description</th>
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<tbody>
<tr>
<td>AI</td>
<td>analog input</td>
</tr>
<tr>
<td>AO</td>
<td>analog output</td>
</tr>
<tr>
<td>ATDC</td>
<td>after top dead center</td>
</tr>
<tr>
<td>BTDC</td>
<td>before top dead center</td>
</tr>
<tr>
<td>CAD</td>
<td>crank angle degree</td>
</tr>
<tr>
<td>CFD</td>
<td>computational fluid dynamics</td>
</tr>
<tr>
<td>CFR</td>
<td>Cooperative Fuels Research</td>
</tr>
<tr>
<td>CI</td>
<td>compression ignition</td>
</tr>
<tr>
<td>CNC</td>
<td>computer numeric control</td>
</tr>
<tr>
<td>CR</td>
<td>compression ratio</td>
</tr>
<tr>
<td>DAQ</td>
<td>data acquisition</td>
</tr>
<tr>
<td>DI</td>
<td>digital input</td>
</tr>
<tr>
<td>DI</td>
<td>direct injection</td>
</tr>
<tr>
<td>DING</td>
<td>direct injection natural gas</td>
</tr>
<tr>
<td>DO</td>
<td>digital output</td>
</tr>
<tr>
<td>DP</td>
<td>differential pressure</td>
</tr>
<tr>
<td>EC</td>
<td>elemental carbon</td>
</tr>
<tr>
<td>EDM</td>
<td>electric discharge machining</td>
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<tr>
<td>EEPS</td>
<td>engine exhaust particle sizer</td>
</tr>
<tr>
<td>EFV</td>
<td>excess flow valve</td>
</tr>
<tr>
<td>EGR</td>
<td>exhaust gas recirculation</td>
</tr>
<tr>
<td>ERDL</td>
<td>Engine Research and Development Laboratory</td>
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<tr>
<td>ESM</td>
<td>exhaust sampling manifold</td>
</tr>
<tr>
<td>EVC</td>
<td>exhaust valve close</td>
</tr>
<tr>
<td>EVO</td>
<td>exhaust valve open</td>
</tr>
<tr>
<td>FFID</td>
<td>fast flame ionization detector</td>
</tr>
<tr>
<td>FTIR</td>
<td>Fourier transform Infrared Spectrometer</td>
</tr>
<tr>
<td>GDI</td>
<td>gasoline direct injection</td>
</tr>
<tr>
<td>HV</td>
<td>hand valve</td>
</tr>
<tr>
<td>IMEP</td>
<td>indicated mean effective pressure</td>
</tr>
<tr>
<td>ISFC</td>
<td>indicated specific fuel consumption</td>
</tr>
<tr>
<td>IVC</td>
<td>intake valve close</td>
</tr>
<tr>
<td>IVO</td>
<td>intake valve open</td>
</tr>
<tr>
<td>LFE</td>
<td>laminar flow element</td>
</tr>
<tr>
<td>LNG</td>
<td>liquefied natural gas</td>
</tr>
<tr>
<td>MAP</td>
<td>manifold air pressure</td>
</tr>
<tr>
<td>NG</td>
<td>natural gas</td>
</tr>
<tr>
<td>NLD</td>
<td>needle lift delay</td>
</tr>
<tr>
<td>OC</td>
<td>organic carbon</td>
</tr>
<tr>
<td>OD</td>
<td>outer diameter</td>
</tr>
<tr>
<td>PAH</td>
<td>polycyclic aromatic hydrocarbons</td>
</tr>
<tr>
<td>PI</td>
<td>pressure indicator</td>
</tr>
<tr>
<td>PM</td>
<td>particulate matter</td>
</tr>
<tr>
<td>PRV</td>
<td>pressure regulating valve</td>
</tr>
<tr>
<td>ROHR</td>
<td>rate of heat release</td>
</tr>
<tr>
<td>RPM</td>
<td>revolutions per minute</td>
</tr>
<tr>
<td>SAE</td>
<td>society of automotive engineers</td>
</tr>
<tr>
<td>SDV</td>
<td>shutdown valve</td>
</tr>
<tr>
<td>SI</td>
<td>spark ignition</td>
</tr>
<tr>
<td>SOI</td>
<td>start of injection</td>
</tr>
<tr>
<td>SR</td>
<td>safety relay</td>
</tr>
<tr>
<td>TDC</td>
<td>top dead center</td>
</tr>
<tr>
<td>TDP</td>
<td>threshold differential pressure</td>
</tr>
<tr>
<td>TI</td>
<td>temperature indicator</td>
</tr>
<tr>
<td>TOI</td>
<td>time of ignition</td>
</tr>
<tr>
<td>TSOI</td>
<td>true start of injection</td>
</tr>
<tr>
<td>ULSD</td>
<td>ultra low sulphur diesel</td>
</tr>
<tr>
<td>US EPA</td>
<td>United States Environmental Protection Agency</td>
</tr>
<tr>
<td>VDC</td>
<td>voltage, direct current</td>
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Chapter 1

1 Introduction

1.1 Natural Gas Advantages

There are several motivations for the study and development of natural gas engines. In the recent years the market price of natural gas in North America has seen a decline, while the reserves have grown due development of hydraulic fracturing technology. Even at today’s low oil prices, natural gas remains cost competitive. Natural gas is also well suited to substitute oil and coal because the North American natural gas supply and distribution infrastructure are already well established.

Environmental considerations play a large role when considering natural gas as an alternative fuel. Natural gas produces approximately 30% less CO2 when compared to oil on same fuel energy content basis [1]. When compared to coal for power generation, the decrease in CO2 emissions can be as high as 50% [2]. Many climate researchers argue that using natural gas as an interim fuel, during the transition to renewable energy, is the only realistic way to cut current CO2 emission levels. A study by Cathles et al. showed that substituting the current coal and some oil consumption with natural gas over the next 50 years can achieve approximately 40% of the benefit that zero-carbon energy sources would if substituted in the same fashion [1].

1.2 Natural Gas in Engine Applications

In engine applications natural gas can be used in spark ignition (SI) and direct injection (DI) internal combustion engines. In fact natural gas has been used as a fuel for spark ignition (SI) engines for many decades. SI engines operating on natural gas have been developed for large fleet vehicles, such as city busses. Passenger car gasoline engines have also been modified to run on natural gas. There are several motivations to study the use of natural gas in direct injection engines, as opposed to SI engines. DI engines offer potential advantages in engine performance, thermal efficiency, and improved exhaust emissions. DI engines can achieve higher efficiency for two main reasons. Unlike SI engines, which control load by throttling intake air flow, DI engines control load by varying the amount of injected fuel. Throttling is not required and
pumping losses are avoided, resulting in efficiency gains. DI engines are also not limited by knock. SI engines are. The absence of a knock limit allows for higher compression ratios and results in higher thermal efficiency. Overall, SI engines have an efficiency of up to 36%, while DI engines can reach efficiencies as high at 45%.

The major challenges in direct injection natural gas (DING) engine development have been primarily associated with the poor compression ignition properties of natural gas, and more recently with emissions associated with natural gas combustion. These issues are further discussed in Chapter 2.

1.3 Thesis Objectives

Over the last several years, the Engine Research and Development Laboratory (ERDL) at the University of Toronto has been developing an ignition assist and fuel injection technology which helps overcome the kinetic limitations of natural gas in an engine environment. Both modelling and experimental work has been conducted. The experimental work has been performed in a fixed-volume combustion bomb coupled to a Cooperative Fuels Research (CFR) engine to simulate cyclical compression typically seen in an engine. The combustion bomb provides optical access, which allows visual study of fuel injection and ignition. While this original setup was effective for studying the fundamentals of hot surface ignition, it did not allow for simulation of complex flow fields typically seen in a direct injection engine. In addition, the nature of the apparatus required skip-fired operation in order to purge the exhaust products out of the combustion bomb between fired cycles. This feature made it impractical to measure the combustion emissions and did not allow for continuously fired operation. The objectives of this thesis were:

1. To implement the ignition assist technology in an engine environment and prove continuously fired operation capability
2. To explore the effect of pressure and equivalence ratio on natural gas ignition performance in an engine environment
3. To collect and study the first emissions measurements associated with the hot surface ignition assist technology
Chapter 2

2  Literature Review

2.1  Natural Gas Compression Ignition & Ignition Delay

Natural gas has slow ignition kinetics, resulting in a low cetane number. In simpler terms, natural gas is a poor diesel fuel. It is very difficult to ignite through compression within a required time frame available in a typical diesel engine. This time frame is termed the ignition delay and is defined as the time from the start of injection (SOI) to the start of combustion of the air/fuel mixture. In order to effectively use natural gas in a modern diesel engine, an ignition delay of less than 2 ms is required for desirable performance, efficiency, and emissions characteristics.

![Graph showing compression ignition temperature of natural gas and diesel fuel](image)

*Figure 2.1 – Compression Ignition Temperature of Natural Gas and Diesel Fuel [1]*

Fraser et al. [2] showed that unassisted compression ignition of natural gas in a diesel engine requires the bulk air/fuel mixture to reach temperatures of 1200 to 1300 K for an ignition delay of 2 ms. For intake air conditions of 1 atm and 325 K, unassisted compression ignition would require compression ratios from about 26:1 to 32:1 [3], [2]. Such high compression ratios are not practical as they would require a typical engine to be redesigned to withstand much higher combustion pressures. This would result in a much heavier engine. Higher pressures would also negatively impact engine efficiency and performance due to a large increase in frictional losses.
[3]. Figure 2.1 shows the temperature and compression ratios that are required for autoignition of natural gas and diesel fuels with ignition delays of 2 ms [1].

2.1.1 Effects of Ignition Delay on Engine Operation

If the ignition delay is longer than 2 ms, combustion will start well after top dead center (TDC), resulting in lower power output and lower efficiency. Aesoy and Valland [4] showed that engine efficiency starts to be variable at ignition delays greater than 2 ms, and rapidly deteriorates at ignition delay values greater than 3 ms. Figure 2.2 shows a plot of engine efficiency vs. ignition delay.

![Figure 2.2 – Engine Efficiency as a Function of Ignition Delay [4]](image)

Longer ignition delay will also result in more flammable mixture available at the start of combustion. Once ignition occurs, the pressure rise can be excessively quick, resulting in shock loads on the engine. A higher proportion of premixed combustion can also lead to higher NOx emissions. Hydrocarbon emissions are also expected to increase, since the longer delay will result in higher amounts of over-lean mixture that will not participate in the combustion process [3], [2]. Ignition delay of less than 2 ms can be readily achieved using diesel fuel, but multiple experiments have shown that achieving this ignition delay with natural gas requires some form of ignition assist [5], [3], [6], [2].
2.1.2 Measuring Ignition Delay

Ignition delay is typically determined by measuring in-cylinder pressure rise or identifying the first appearance of luminosity produced by the burning fuel/air mixture. Siebers and Edwards [7] showed that the two approaches yielded substantial differences for liquid fuels such as methanol, ethanol, isooctane, and cetane. Luminous ignition delay was consistently lower than the in-cylinder pressure ignition delay. For gaseous fuel, such as methane and natural gas, however, Fraser et al. [2] demonstrated that the two methods produce comparable ignition delay measurements. Most studies of natural gas and methane combustion use the pressure measurement technique as it is simpler to implement.

2.1.3 Physical and Chemical Components of Ignition Delay

In their experiments, Naber and Siebers [7] and Aesoy and Valland [1], [4] recognized that the overall ignition delay can be broken down into the physical and chemical delay components. Although the two research groups have slightly varying definitions for these components, generally, the physical delay represents the time it takes for the jet to penetrate the combustion chamber and mix with the air to the appropriate ignitable level. The chemical delay represents the time required for the ignition reaction to start. Naber and Siebers saw the physical and chemical delay components as not additive. Their view was that chemical delay could start before the physical delay period is over. On the other hand, Aesoy and Valland interpreted the overall ignition delay to be a summation of the physical and the chemical delay terms. These differences indicate that the two ignition delay components do not have universal strict definitions. However, the two-component ignition delay theory offers a valuable concept for understanding the overall ignition delay phenomenon.

2.1.4 Effect of Natural Gas Composition on Ignition Delay

Natural gas is a mixture of hydrocarbon gasses typically consisting of methane (up to 100%), ethane (2–10%), propane (1–4%), and other gasses in trace amounts [2]. The exact composition of natural gas varies widely depending on the source of supply. For example, Liss et al. reported that commercially available natural gas in the United States can vary from 75% to 98% in methane content [7]. Several studies have shown that the minor constituents of natural gas can have a significant effect on its compression ignition kinetics [2], [7], [1], [4], [8], [9].
Fraser et al. [2] investigated the ignition characteristics of methane blended with 5, 10, and 15% ethane. This study demonstrated that an increase in ethane concentration of the natural gas results in a slight decrease of the ignition delay. Naber and Siebers [7] conducted a similar study but used a range of natural gasses with compositions typical of the US market. These included extremes of high-methane LNG-sourced natural gas, and propane-rich natural gas supplied under propane/air peak shaving practices. Results of this study showed that ignition delay decreases with increasing ethane, propane, and butane content. In engine applications, this variability could result in performance or emissions issues. The study suggested the possibility of using species such as ethane, propane, and butane to limit the ignition delay times of natural gas to desired values of less than 2 ms [7].

Aesoy and Valland [1], [4] conducted a similar investigation of the effects of the constituents of natural gas on ignition delay. They reported that the ignition delay of natural gas can be as much as two to three times lower than for pure methane. Aesoy and Valland’s results were consistent with the results of Naber and Siebers and showed higher alkanes to decrease ignition delay of natural gas. It was also shown that higher carbon alkanes were more effective ignition improvers. Aesoy and Valland classified natural gas constituents into two classes [1], [4]:

**Ignition Quality Improvers** – Constituents that lower the ignition delay when compared to pure methane (ethane, propane, butane, ethylene, hydrogen)

**Ignition Quality Deteriorators** – Constituents that are inert/incombustible and have no chemical effect, but affect combustion by absorbing some of the combustion heat (nitrogen, carbon dioxide)

### 2.1.5 Effect of TDC Temperature on Ignition Delay

In their study of natural gas ignition, Naber and Siebers [7] found temperature to be the single most significant parameter influencing ignition delay. Experimental data showed that TDC temperature has a strong Arrhenius type effect on ignition delay (higher temperature leading to lower ignition delay) at temperatures below 1200 K. Above 1400 K the ignition delay approached a limiting value of the physical delay component.
2.1.6 Effect of Ambient Pressure or Ambient Density on Ignition Delay

Naber and Siebers [7] observed that ambient density, which is essentially ambient pressure, has an effect on ignition delay. Ignition delay of each of the tested natural gas blends tested in the study reduced with increasing density for temperatures below 1300 K. This effect was found to be less than first order. The strongest effect was observed for pure methane and weakest for a high-ethane natural gas blend.

2.1.7 Effect of Injection Pressure on Ignition Delay

Aesoy and Valland [1] conducted an extensive investigation of natural gas injection, ignition and combustion, and explored the effects of various factors on the ignition delay. Their work showed that the injection pressure of the natural gas had a varying effect on the ignition delay. Initially, as the injection pressure increased, ignition delay decreased. At a certain point, however, an ignition delay minimum was reached. Beyond this point increasing the injection pressure actually raised the ignition delay.

2.2 Ignition Assist Technologies

All studies exploring compression ignition of natural gas in a direct injected diesel engine concluded that the temperatures of 1200 – 1300K or compression ratios of up to 1:32 are required to achieve ignition within 2 ms after SOI [3], [2], [7], [1], [4]. As these conditions are not achieved in modern diesel engines, ignition assist technologies are required to make natural gas ignitable with 2 ms in a typical diesel environment.

Several ignition assist technologies have been studied for direct injected natural gas (DING) engines. One method to assist combustion of natural gas is to precede natural gas injection with a pilot injection of a high-cetane fuel such as diesel. This technology was developed and commercialized by Westport Research Inc. [10]. Another approach uses a glow plug to locally increase the temperature of the fuel/air mixture in order to initiate combustion. The glow plug assist method is discussed in further sections as it may offer an elegant solution for assisting natural gas ignition without the added complexity of a second pilot fuel system.
2.2.1 Glow Plug Ignition Assist

A glow plug can be used to assist natural gas ignition by acting as an ignition site. In order to ignite the natural gas/air mixture, glow plug temperatures of 1200–1400 K are required [3], [1]. The effectiveness of the glow plug depends on its temperature, position, and the velocity of the gas jet.

Unshielded Glow Plug

Fabbroni et al. [3] explored the viability of using an unshielded glow plug as a means of ignition assist. Experiments were performed in a combustion bomb coupled to a rapid compression machine to simulate in-engine conditions and flow patterns. Two gas jets positioned at 60° and 20° from the glow plug were used to simulate jet–jet interaction. Experiments were conducted at three different temperatures and pressures, and two different equivalence ratios. It was demonstrated that an ignition delay of under 2 ms could not be achieved with an unshielded glow plug due to cooling of the glow plug from contact with the expanding gas jets.

Shielded Glow Plug

As previously shown by Aesoy and Valland [1], shielding of the glow plug can be used to improve glow plug performance. Shielding offers the following benefits:

- Shielding limits the heat losses from the glow plug to the swirling air flow
- Shielding limits the cooling of the glow plug by the gas jets
- Shielding can increase the residence time and, therefore, the heat transfer to the gas that is trapped between the shield and the glow plug

Fabbroni et al. [3], [6] tested the performance of several shield configurations. The study found that, generally, shielding is effective at igniting the contacting fuel jet. Larger size shields, however, deflected adjacent fuel jets and impeded flame propagation from one fuel jet to another. Further development by Fabbroni [11], and later by Habbaky [12] and Chown [12], refined the shield design. The latest design consisted of a thin Inconel tube installed concentrically around the glow plug tip. The gap between the glow plug and the shield was set to
0.5 mm. A circular opening in the glow plug wall was provided to expose a portion of the glow plug surface to contact with the fuel jets.

2.3 Jet Structure and Behavior

Fuel jet structure and behavior are critical for direct injection engines. Jet dynamics influence the ignition delay and the overall combustion process by directly affecting the fuel/air mixing characteristics, and the distribution of the fuel/air mixture in the combustion chamber.

In a diesel engine environment, fuel penetration of the combustion chamber needs to occur within the 2 ms ignition delay criteria, as discussed in previous sections. Depending on the fuel, liquid or gas, the jet characteristics vary [11].

2.3.1 Jet Penetration

When jets are injected into a quiescent environment, their shape develops as the fluid travels outward from the injection orifice. For incompressible transient jets, Rizk [12] showed that the ratio of the jet head diameter to the penetration distance of the jet (D/Z) initially changes, but approaches a constant value once the jet has penetrated approximately 10–15 nozzle diameters from the injection orifice [12]. This constant ratio results in a jet shape that is referred to as self-similar, or fully developed. This condition implies that the jet shape is constant, and remains congruent as the jet penetrates further out from the injection source.

In order to estimate jet penetration with respect to time, Hill and Ouellette [12] used a jet model first proposed by Turner. The Turner model considers the jet to consist of two parts: a quasi-steady-state jet region, and a travelling spherical vortex head. This model is depicted in Figure 2.3.
Using conservation of momentum, the Turner jet model, and the assumption of self-similarity, an equation relating jet penetration to the momentum injection rate ($M$), ambient density ($\rho$) and time ($t$) was derived for incompressible jets [12]:

$$Z_t = \Gamma^*(M/\rho)^{1/4} t^{1/2} \quad (Eq. 2.1)$$

The constant $\Gamma$ in this equation was calculated to be equal to 3. Hill and Ouellette [12] confirmed this value with experimental data of Witze [12]. Through analysis of experimental results of multiple jet injection studies, Hill and Ouellette further expanded these concepts and validated the above relationship for compressible and incompressible jets (including under-expanded), and sonic and subsonic jets. Their analysis also showed that the jet penetration constant value of 3 is also valid in these cases [12].

**Engine Conditions Effect on Jet Penetration**

Hill and Ouellette [12] validated Turner’s jet model and verified that the self-similarity assumption holds for all types of injections into a quiescent environment. They also investigated the validity of these assumptions in an engine environment [13]. Jet interaction with the combustion chamber wall, the combustion chamber flow field (or swirl), and the impact of injection duration on jet penetration were studied. To test jet-wall interaction, methane was injected at an angle of $10^\circ$ to a closely positioned wall [12]. This was done to simulate the effect that a cylinder head surface may have on a gas jet. Different injection times and stagnation to chamber pressure ratios were tested. Experimental observations showed a slight departure from self-similarity at the lower pressure ratios. The observed deviation, however, was small. The study concluded that jet self-similarity is a reasonable simplifying assumption for gas jet injection close to a wall surface [12].

*Figure 2.3 – Turner’s Model of a Transient Turbulent Jet [12]*
Further investigation was carried out thorough computer modelling of jet penetration [13]. The k–ε turbulence model was used to model the jet. The model was tested against previous experimental studies and was optimized to match experimental data. The optimized model was then used to study the effects of combustion chamber turbulence on jet penetration. Results showed that increased turbulence decreases jet penetration, but increases lateral spreading of the jet. The effect of injection duration was also explored. Results showed that the penetration rate of a short burst jet is the same as that of a continuous jet for up to twice the injection duration of the short burst jet [13].

2.3.2 Concentration Decay in Gaseous Jets

In order to understand how a gas jet behaves once injected into a combustion space, it is important to understand how the concentration of the gas decays with axial distance from the injection orifice. The concentration in subcritical gas jets decays along the jet path in a hyperbolic manner, according to the following equation [14]:

\[ \eta_{\text{mean}} = k d^* \left( \frac{\rho_a}{\rho_g} \right)^{1/2} / (z+a) \]  

(Eq. 2.2)

Where:

\( d \) – orifice diameter

\( \eta_{\text{mean}} \) – mean axial concentration

\( a \) – virtual origin displacement

\( k \) – axial decay constant

\( z \) – downstream distance

\( \rho_a \) – ambient gas density

\( \rho_g \) – injected gas density

In choked, supercritical flow, if the upstream pressure of the jet fluid is increased, the expansion of the jet to ambient conditions takes place downstream of the orifice. The jet behaves as if it
were expelling from an orifice of a larger diameter, called the pseudo-diameter [14]. Birch et al. [14] conducted a study which defined the pseudo-diameter and experimentally validated that the axial concentration decay equation for subcritical flow was valid for supercritical conditions if the pseudo-diameter is used in place of the nozzle diameter. The study also confirmed that the dependency of the decay constants for different gasses is dependent only on specific heats and densities of the gasses. The pseudo-diameter was defined as [14]:

\[ d_{ps}/d = (0.582 \cdot C_d \cdot P/P_a)^{1/2} \]  

(Eq. 2.3)

Where:

- \( d_{ps} \) – pseudo-diameter
- \( d \) – orifice diameter
- \( P \) – upstream pressure
- \( P_a \) – ambient pressure
- \( C_d \) – discharge coefficient

2.3.3 Gas Jet Mixing

Jet development and mixing with air is critical, as this process controls the physical delay component of ignition delay, and controls the amount of flammable fuel/air mixture available for combustion. Abraham et al. [15] conducted a study in which jet development and mixing of a gas jet was compared to a liquid spray. Spray and jet mixing was compared by quantifying the rich (\( \phi > 2 \)), flammable (0.5 \( \leq \phi \leq 2 \)), and lean mixtures (\( \phi < 0.5 \)) of the gas jets and liquid sprays. The study was based on computational model results of fuel injection in a diesel engine.

Abraham et al. [15] determined that jet development occurs in three stages as the jet moves farther out from the source of injection. From the start of injection to about 1/3 of the full jet development distance, interaction of the gas jet with the ambient air is low and is limited to a narrow region at the periphery of the jet. This limited amount of mixing results in a predominantly rich jet. As the jet moves further from 1/3 to 2/3 of the full jet development distance, mixing at the jet periphery becomes more dominant. The mixing layer grows inward
from the periphery towards the center axis of the jet. This creates the flammable fraction of the jet. Past this point, from about 2/3 to full jet development length, air continues to mix with the jet, but results in a lean mixture. This jet development sequence is depicted in Figure 2.4 [15].

![Figure 2.4 – A Schematic of Gas Jet Development](image)

2.3.4 Spray Mixing

Spray development and mixing proceeds in much the same sequence as for gas jets, but with the added effect of evaporation of the fuel. In a spray, the mixing of the fuel and air is greatly affected by the droplet size. Larger droplets result in a slower rate of evaporation than for smaller ones [15].

In their modeling work, Abraham et al. [15] showed that gas jets mix slower than liquid sprays. This is contrary to intuition, as it is commonly assumed that the gas jet should be a better mixer since it is not limited by vaporization rates of droplets. Simulations, however, revealed that a spray exhibits a much faster boundary shear layer growth than the gas jet. Modelling several different spray drop sizes demonstrated that both large and small droplets enable a higher percentage of a flammable mixture. For large droplets the evaporation rate is low. The core of the spray develops a higher percentage of the flammable phase by limiting the rich phase formation. For small droplets, the evaporation is much quicker, increasing the transfer of their momentum to the surrounding air. The transferred momentum increases turbulence at the spray
boundary and increases mixing of the vaporized fuel with the ambient gas. This leads to an increased flammable fraction [15].

Abraham’s analysis showed that as the droplet size decreases, spray mixing approaches the mixing rates of a gas jet [15]. In fact, a study by Hill and Ouellette [13] demonstrated that diesel sprays exhibit identical penetration and mixing characteristics as natural gas jets when the diesel spray droplet size is smaller than approximately 5 microns.

2.3.5 Enhancing Gas Jet Mixing

Many of the studies looking at natural gas jets in an engine environment look for ways to improve the gas jet mixing characteristics, since improved mixing would yield faster ignition and more complete combustion. The most common methods evaluated in the reviewed literature are briefly discussed below.

Swirl – A common approach of improving mixing is to use a flow field, such as swirl created in an engine environment. The swirl velocity, however, is low close to the center of the combustion chamber and the injector (center mounted), and grows in the radial direction. A study by Abraham et al. [15] concluded that swirl has very little effect on jet mixing.

Injector Hole Size – The scaling analysis of Abraham et al. [15] showed that developing the maximum fraction of flammable mixture quickly can be achieved by decreasing the injector hole diameter. This approach was tested and confirmed in a study by Picket and Siebers [16].

Injector Hole Shape – A study by Gutmark et al. [17] showed that elliptical jets with a 2:1 ratio of the major axis nozzle width to the minor axis nozzle width could entrain several times more fluid than a circular jet. In this study, the elliptical nozzle gas jet was found to exhibit an axis switching phenomenon with the minor and major axes of the jet switching as the jet travelled out from the injection source. The jet’s minor axis was found to entrain fluid much more effectively than the jet’s major axis. It was also noted that the entrainment of the jet’s major axis was similar to that of a circular jet. The effectiveness of the elliptical jet was attributed to the outer jet vortex ring being stretched in the minor axis, resulting in higher entrainment of quiescent fluid [17]. Exploring the use of elliptical jets in an engine environment presents a possible opportunity to obtain better fuel air mixing.
2.4 Emissions

In addition to ignition and combustion properties, fuel jet development in a direct injection engine has a significant effect on engine emissions. Traditional diesel engines have been known to produce high NOx and particulate matter (PM), also called soot, emissions. Tree and Svensson [18] reviewed a multitude of diesel emissions studies and compiled an approximate depiction of the operating domain for typical diesel engines. This domain is shown in Figure 2.5 [18].

![Figure 2.5 – Diesel Engine Soot and NOx Operating Domain [18]](image)

The diffusion flame in a typical diesel engine produces a structure where the outer flame region of the fuel jet is combusting at near stoichiometric conditions, but is at temperatures that exceed 2600 K. These conditions correspond to the bottom right portion of the figure and result in high NOx formation levels. The area of the jet inside the flame envelope is at lower temperatures than the flame, but is at a much higher equivalence ratio. These conditions correspond to the high soot region of the figure and result in high soot/PM formation [16], [18].

NOx emissions contribute to smog formation, while soot has been linked with adverse health effects such as respiratory illness and cancer. The first emissions standards for CO, NOx, and hydrocarbons for heavy-duty diesel engines were instituted in 1974. However, soot/PM emissions did not become regulated until the 1980s. Since that time, diesel exhaust emissions standards have spread to cover non-road equipment, locomotives, and marine engines. They have
also become more stringent. The historic evolution of the heavy-duty diesel engine on-road regulatory limits for NOx and PM emissions are shown in Figure 2.6 [19].

In response to the increasingly stringent regulations, diesel technology has continuously progressed. Modern diesel engines employ technologies such as exhaust after treatment, including wall-flow diesel particulate filters (DPFs), diesel oxidation catalysts (DOCs), and selective catalytic reduction (SCR) systems. The following sections cover the fundamentals of NOx and soot/PM formation in compression ignition engines.

2.4.1 NOx Emissions

NOx emissions consist of NO and NO₂. In combustion engines NO accounts for most of the total NOx level. The dominant mechanism for NOx formation is termed the thermal NOx mechanism or the Zeldovich mechanism. In this mechanism NO is formed by oxidation of atmospheric (molecular) N₂. Initial NO formation rate is defined with the following rate expression [20, p. 171]:

$$\frac{d[NO]}{dt} = 2k[/eq][O]_eq[N2]_eq$$

(Eq. 2.4)
Where:

\[ \frac{d[NO]}{dt} \text{ is the rate of NO formation in kmol/m}^3 \]

\( k \) is the rate constant of \( 1.8 \times 10^{11} \exp\left[-38,730/T(K)\right] = \text{m}^3/\text{kmol s} \)

\([O]_{eq} \text{ is the equilibrium oxygen radical concentration in kmol/m}^3 \text{ from the following equilibrium reaction: } O_2 \leftrightarrow 2O\)

\([N_2]_{eq} \text{ is the equilibrium nitrogen concentration in kmol/m}^3 \text{ from the following equilibrium reaction: } O_2 + N_2 \leftrightarrow 2NO\)

Temperature has by far the single most significant effect on NO formation. Higher temperature leads to higher rates of NO formation. Temperature affects the overall rate constant of Eq. 2.4, as well as the equilibrium constants for the O and N\(_2\) equilibrium. Higher O\(_2\) concentration also increases NO formation rate [21, p. Eq. 11.11]. In DI engines, NO forms both in the flame front and in the post-flame gasses. Gasses spend a relatively short time in the flame front. Post-flame gasses dominate NO production since they have a longer residence time at higher temperature [21, p. 574].

### 2.4.2 Particulate Matter (PM) Emissions

PM, or soot, is formed from unburned fuel. Particles nucleate from a vapor phase to a solid phase in fuel-rich, high-temperature areas. Soot formation consists of the following processes [18]. These are depicted in Figure 2.7.

1. Pyrolysis – a high-temperature, limited-oxidation chemical conversion of fuels to soot precursors such as unsaturated hydrocarbons, polyacetylenes and polycyclic aromatic hydrocarbons (PAH)
2. Nucleation – formation of soot particles (nuclei) from gaseous precursors
3. Surface growth – soot nuclei absorb gaseous hydrocarbons and grow in mass
4. Coalescence and agglomeration – processes by which soot particles combine into larger particles or form composite particle clusters
5. Oxidation – a process that converts carbon and hydrocarbons to CO, CO$_2$ and H$_2$O, thus competing with the soot formation processes.

![Figure 2.7 – Particulate Matter/Soot Formation Process][18]

To explain how soot forms in a diffusion flame, Dec proposed a model shown in Figure 2.8. As fuel leaves the injector orifice, the fuel droplets begin to evaporate. Evaporation is controlled by the rate of energy transferred into the jet. The liquid jet is self-similar and penetrates to a maximum length, called the liquid length. The fuel is most concentrated in the center of the jet, with the outer layer at a lower fuel concentration. The jet expands with axial distance. Air entrainment occurs at the outermost layer. The diffusion flame occurs at a distance from the nozzle, termed the lift-off length. The flame is fuel rich at the lift-off length, but gets progressively leaner in the increasing axial and radial directions. The length of the stoichiometric flame layer is termed the flame length [18].

![Figure 2.8 – Dec’s model of soot in a diffusion flame][18]

Soot particles first appear just downstream of the liquid length. These particles are small and uniformly distributed. They form due to the initial premixed burn phase of the flame.
development. As the diffusion flame begins to form after the premixed phase, larger soot particles form at the leading edge of the jet. The diffusion flame enhances the soot growth rate. In the later stage of the diffusion flame, soot becomes contained by a layer of OH$^-$ and gets oxidized before leaving the jet. Soot, however, can survive combustion if the time to complete burnout is insufficient or parts of the flame become extinguished [18].

Lift-off length is a critical parameter in soot formation as it determines the amount of air entrained into the fuel jet. Longer lift-off length results in more air entrainment and decreases soot formation. Many factors affect the lift-off length and, therefore, the sooting characteristics of the diffusion flame. Higher ambient temperature results in a shorter lift-off length, which leads to lower air entrainment and higher sooting. Increased injection pressure increases jet velocity and increases the lift-off length, resulting in lower sooting. Higher ambient density decreases the lift-off length, but increases air entrainment. This results in an insignificant overall effect. Decreasing injection orifice size tends to improve air entrainment and thus decreases sooting [18].

**Effect of Fuel Structure on Soot Formation**

Fuel structure, or the way all the atoms are organized in a molecule, has an effect on sooting characteristics. While most researchers agree that composition is important for both premixed and diffusion flames, the opinions diverge on the influence of structure. Most agree that structure plays an important role in diffusion flames, but may be less important for premixed flames [18].

Ring structures in fuel molecules are the largest contributors to soot formation. Multiple studies have indicated that fuels having higher aromatic content increase the fuel’s propensity to soot. However, some studies indicated that the increased sooting was attributed to a higher C/H ratio, and not necessarily the aromatic structure. For non-aromatic fuels, higher chain length, ring circumference, higher degree of branching, and presence of carbon double bonds also increase sooting. Further, higher molecular weight and higher degree of isomerization increase sooting tendency [18].

**Effect of Fuel Composition/Fuel Oxygen on Soot Formation**

Oxygen content, whether mixed in with the fuel or as part of the fuel structure, generally decreases soot formation. Oxygen that is mixed in with the fuel initially increases soot formation
until a critical equivalence ratio is reached. Beyond this point soot formation quickly falls off to zero. It is important to note that an equivalence ratio of one is not necessary to eliminate soot. As long as the conditions are adequate to oxidize fuel to CO, soot will not form. Several studies have indicated that soot elimination can be achieved with oxygen mass fractions of 27–35% [18].

Other important species present in fuels are carbon, hydrogen and sulphur. Higher carbon content increases the sooting likelihood of a fuel, as it is the main constituent of soot. Higher hydrogen content, on the other hand, decreases sooting potential. The decrease is linear with hydrogen wt%. Sulphur worsens sooting by oxidizing and attaching itself to soot particles, thus increasing their mass [18].

**Effect of Engine Design on Soot Formation**

Many engine parameters affect soot formation. Some of these are captured below [18]:

**Swirl** – Swirl usually does not have enough momentum to significantly impact jet air entrainment, but it does increase turbulence in the late injection stages. The increased turbulence increases soot burnout. If swirl is too high, however, it can cause negative jet-jet interactions and delay mixing, which in turn, results in more soot surviving combustion [18].

**Injection Timing** – Advanced injection timing results in lower particulate emissions, but higher NOx. Advanced timing actually increases soot formation, but since the end of injection comes earlier, higher temperature is reached. The higher temperature facilitates more complete soot burnout and results in lower overall exhaust soot content [18].

**Intake Temperature and Pressure** – Higher intake temperature increases the rate of soot formation and the rate of oxidation, leading to overall lower soot exhaust content. Increased intake pressure increases air entrainment in the fuel jet, and leads to lower in-cylinder and exhaust soot levels [18]. Refer to Section 5.2.1 for further discussion on jet entrainment at higher intake pressure.

**Engine Transients** – Rapid change from low load to high load conditions results in increased sooting since the increase in air lags behind the increased fuel injection. This lag results in oxygen deficient combustion and higher sooting [18].
**Multiple Injections** – In this approach a small amount of fuel is injected at the end of the main fuel injection event. This post-injection causes higher temperatures and better mixing which aids in the burnout of soot formed in main injection/diffusion flame event [18].

**Water Emulsions** – The use of fuel-water emulsions has shown a reduction in both the soot and NOx emissions levels. Jets of fuel-water emulsions increase the lift-off length and result in more air entrainment, leading to lower soot formation [18].

### 2.4.3 Soot Sources in DING Engines

Jones et al. [11] conducted a study that looked at the PM emissions in a direct injection dual-fuel engine running on natural gas (as the main fuel) and diesel fuel (pilot injection for combustion initiation). The study was performed using a Detroit Diesel six-cylinder engine that has been modified to run on one piston, in a dual-fuel arrangement.

Study results showed that the pilot-generated PM concentration varied from 4 to 40 % of total soot, and the lube oil and natural-gas-generated PM contribution varied from 60 to 96 %. The highest pilot PM contribution occurred at low load with no exhaust gas recirculation (EGR). This indicates that the pilot contribution to PM emissions can be expected to be high with the engine in idle. At high load, pilot injection accounted for approx. 5% of the PM emissions, indicating that natural gas has a significant contribution to the total PM emissions [11].

The lubricating oil’s contribution to PM emissions was also estimated by measuring the trace metal content (as a tracer for oil PM). Only the upper bounds of oil PM contributions could be determined. The contribution values were found to range from high possible percent PM contribution at low load conditions, to 20% possible PM contribution at high load conditions. By comparing the high load maximum 20 % contribution of lubricant to the 96% contribution of both natural gas and oil, Jones et al. [11] concluded that natural gas is the primary contributor of PM emissions at high loads.
Chapter 3

3 Experimental Setup

The experimental setup for the work presented in this thesis was based on the modification of the Ricardo Hydra single-cylinder engine and dynamometer platform that has been previously used for several engine studies at the ERDL. The Hydra setup was previously modified by Cheung to allow optical access to the combustion chamber using a Bowditch piston [22]. Chown later built on Cheung’s original design and established a high-level plan for integrating the glow plug ignition assist technology into the Hydra engine [13]. The experimental setup scope of this thesis was extensive and included: the redesign, modification, fabrication and implementation of the original proposed mechanical design by Chown [23]; and design and implementation of multiple auxiliary systems.

To make the body of this thesis more concise, the following sections describe the completed direct injected natural gas (DING) engine system as used for thesis experiments. A more detailed description of the design development undertaken in the scope of this thesis can be found in Appendix A1.

3.1 Optically Accessible, Direct Injection Natural Gas (DING) Engine

The DING engine is a modified Ricardo Hydra single-cylinder engine that has been extended to accommodate the optically accessible Bowditch piston design. A simplified illustration of the setup is shown in Figure 3.1 for the purpose of clarifying the optical access concept. The extension consists of a spacer block, a second Lister STW water-cooled cylinder block, and a Lister ST air-cooled electrically-heated cylinder head. The spacer block holds a stationary mirror mounted at 45° to the horizontal plane. The Lister STW cylinder block is mounted on top of the spacer block, housing the Bowditch piston. The Lister ST cylinder head is mounted on top of the Lister STW cylinder block. The Bowditch piston has an open vertical side slot that reveals the stationary mirror at all times during engine operation. The piston deck is equipped with a quartz window, which is fastened to the piston with a bolted piston crown. A horizontal line of sight is established through the open side slot of the Bowditch piston to the stationary mirror. The mirror reveals the view of the combustion chamber through the quartz window.
In this extended DING engine design, the experimental combustion chamber is bound by the Bowditch piston crown, the quartz window, the Lister STW cylinder block, and the Lister ST cylinder head. The original Hydra piston and cylinder block see no combustion and are solely used to couple the Bowditch piston to the crankshaft via the Hydra connecting rod. The active cylinder bore is set by the Lister STW block bore of 3.75 in. (93.53 cm). The stroke of 3.5 in. (8.89 cm) is set by the Hydra crankshaft. The displacement of the active cylinder is 38.66 in$^3$ (633.46 cm$^3$).

It is important to note that the optical access capability was not utilized in the thesis experiments. The quartz window was replaced with a solid aluminum puck. This was done to limit the potential risks associated with the first-time implementation of the shielded glow plug ignition assist technology in an engine environment. Optical studies were deferred to a time when the DING engine operation is well tested and understood.
3.1.1 Combustion Chamber Geometry & Compression Ratio

Combustion chamber design of the DING engine was based on the combustion chamber geometry of the CFR combustion bomb apparatus. A simplified diagram of the DING engine combustion chamber is shown in Figure 3.2.

![Diagram of DING Engine Combustion Chamber](image)

Figure 3.2 – DING Engine Combustion Chamber

The combustion chamber space consists of the small clearance volume between the top of the piston crown and the cylinder head, and the cylindrical bowl volume profiled into the piston crown. The diameter of the bowl is 2.0 in. (50.80 mm), as replicated from the diameter of the CFR combustion bomb apparatus. The depth of the bowl is 0.535 in (13.59 mm), measured from the cylinder head plane to the surface of the quartz window.

The depth of the bowl is based on two critical criteria. The first criterion is adequate piston clearance from the cylinder head. Due to its elongated design, the Bowditch piston can expand significantly in length due to combustion heat. Insufficient clearance may result in collision of the elongated piston with the cylinder head. In order to avoid mechanical interference, the minimum required cold clearance was estimated to be 0.062 in. (1.58 mm). This value was calculated based on the thermal expansion of the Bowditch piston at a maximum average temperature rise of 200° C. To provide an added safety factor, the cold piston clearance was set to 0.085 in (2.08 mm). Calculation of thermal expansion is included in Appendix A2.

The second criterion was the compression ratio (CR). Compression ratio directly affects engine cycle efficiency, as well as injection mixing, ignition and flame propagation. As much as practicable, an attempt was made to match the CR of the DING engine to the 14.8:1 CR of the CFR combustion bomb apparatus. Due to the clearance requirement and the aim to preserve the
piston bowl geometry close to that of the combustion bomb apparatus, the final compression ratio of the DING engine was set to 13.17:1. A detailed calculation of the compression ratio can be found in Appendix A3.

### 3.1.2 Fuel Injector Nozzle Tip and Glow Plug Geometry

Work conducted on the CFR combustion bomb apparatus showed that the geometry of the fuel injector nozzle tip, the glow plug and the glow plug shield directly affect combustion characteristics. As much as practicable, the CFR combustion bomb apparatus arrangement was replicated in the DING engine. The injector tip was positioned normal to the cylinder head plane, in the middle of the piston bowl. The depth of the injector tip was chosen to be 0.245 in. (6.22 mm), measured from the cylinder head plane to the centerline of the injector orifices (see Figure 3.2). This depth places the plane of injection in the middle of the cylindrical combustion bowl volume. The glow plug and shield were positioned at an axis angle of 21.3° from the injector tip axis with the tip pointed in the direction of the injector. The depth of the glow plug was set such that the center of the shield opening is in line with the opposing injector orifice. The horizontal distance from the injector nozzle orifice to the centerline of the glow plug was set to 0.229 in. (5.81 mm) to be consistent with the equivalent distance of the CFR combustion bomb apparatus (see Figure 3.2).

The implemented cylinder head design enables the adjustment of the injector and glow plug shield angles. The injector is equipped with a degree indicator system, and the shield is inscribed with marks at every 10° of shield rotation (allows to set the shield angle). Figure 3.3 shows a simplified illustration of the injector nozzle tip and glow plug.

![Figure 3.3 – Injector and Shield Angle Illustration](image-url)
It should be noted that the injector and glow plug angles share the same datum line and are both defined to be positive in the direction of the engine swirl field.

### 3.1.3 Bowditch Piston Rings

The nature of the optical design requires the underside of Bowditch piston to be open, in order to allow optical access. This makes conventional splash lubrication impractical. For this reason, the piston is outfitted with piston rings made from Vespel® SP-21 graphite-filled polyamide material (manufactured by Cook Compression). These rings allow for short-term oil-less operation of the DING engine. The ring design incorporates an overlapped gap which enables thermal expansion of the rings. Extensive experimentation with the piston ring materials and gap geometry was conducted. This work is documented in Appendix A4.

### 3.2 Cussons Dynamometer and Control Unit

As mentioned previously, the DING engine is based on a modification of single-cylinder Ricardo Hydra engine and testbed system. The testbed system was retained in the DING setup. The Hydra testbed includes a Cussons dynamometer and a Cussons control unit. The dynamometer is coupled to the DING engine. It is electrically driven and allows motoring of the engine in unpowered operation, and absorption of engine output power in fired operation. The Cussons control unit includes the following features which were utilized in the DING experiments:

- a potentiometer for setting the engine speed
- readout of engine torque, engine speed, and intake air flow rate
- engine oil pump and heater control
- engine coolant pump and heater control
- safety trips for low oil and coolant flow, and for high oil and coolant temperature

The engine oil and coolant reservoir, pumps, and heaters were provided on the testbed system. Both oil and coolant systems use a water cooled heat exchanger which allows rough temperature control by throttling of the cooling water flow. Cooling water was provided from the building water supply system.
3.3 Fuel Injector

The fuel injector used in the study was originally developed by Green and Wallace [24]. It was later modified over the course of the work conducted on the CFR combustion bomb apparatus. A simplified diagram of the fuel injector configuration used in this thesis is shown in Figure 3.4.

![Simplified Section Diagram of DING Injector](image)

Figure 3.4 – Simplified Section Diagram of DING Injector

The injector is actuated by an electromagnetic solenoid. When no power is applied, the spring loaded injector needle seals the fuel flow by pressing the needle tip into a seat installed in the injector nozzle’s tip. When the solenoid is powered, the armature attached to the injector needle is pulled back. In response, the needle retracts and the seal at the needle tip is opened. Fuel is allowed to flow from the injector inlet through the injector’s internals, and finally past the seat and out of the injector orifices. In the initial design the injector needle tip was made of O1 tool steel and the nozzle seat was made from Vespel® (a high temperature polyamide). With this configuration, the injector sealed poorly, allowing fuel to leak out in the shut-off position. Since leakage would directly affect emissions measurements, the sealing mechanism underwent extensive redevelopment over the course of this thesis. Key modifications included fabrication of a new injector needle, changing the seat material to copper, and changing the needle tip material to Vespel®. The resulting configuration provided good sealing, improved wear characteristics and easy serviceability of the seal components. Details on the redevelopment of the injector sealing system are provided in Appendix A5.

Fuel Injector Nozzle

Three nozzles were available for the fuel injector: 0.2 mm diameter circular 9-hole, 0.3 mm diameter circular 9-hole, and 0.3 x 0.4 mm elliptical 9-hole. Nozzle selection was based on each
nozzle’s ability to deliver fuel at a rate that would enable injection durations in excess of the ignition delay period over a wide range of equivalence ratios. This requirement is critical for achieving non-premixed mode combustion. Preliminary tests showed that only the 0.2 mm diameter circular 9-hole nozzle was sized appropriately for the DING engine, with the latter two nozzles resulting in excessive fuel flow rate. All thesis experiments were conducted using the 0.2 mm diameter circular 9-hole nozzle.

**Fuel Injector Settings**

Injection dynamics and flow characteristics are highly sensitive to two critical fuel injector parameters: the needle spring preload and the needle lift distance. The injector needle spring preload controls the needle’s sealing force and has an effect on injector closing time. The preload can be adjusted by the thread engagement of the spring retaining locknut. A needle preload of 40 lb was set in the injector preparation and calibration phase of each test. This value was determined optimal in previous work of Chown [23].

The needle lift distance is determined by the gap between the solenoid and the armature. The lift distance controls how quickly the injector opens, since the solenoid pull force decreases with the square of solenoid-armature gap distance. The lift distance also controls the size of the gap between the needle tip and the nozzle seat when the injector is actuated. This gap throttles the fuel flow before it reaches the nozzle offices. The needle lift can be set by adjusting thread engagement between the solenoid casing and the injector body. The solenoid casing is first engaged until it lightly contacts the armature. It is then backed off by 75° in the counterclockwise direction. This sets a needle lift distance of 0.010 in. (0.265 mm). This lift distance was determined optimum during preliminary experiments.

### 3.4 Glow Plug and Shield

The glow plug used in this thesis was a ceramic-tipped Bosch GLP4 Duraterm® glow plug (part number 0250523002). This glow plug type was previously used by Chown on the CFR combustion bomb apparatus since it can reach higher temperatures than can be achieved by conventional metal-tipped glow plugs. The ceramic design can reach temperatures of 1400 K at 11 VDC rated voltage.
The glow plug shield designed by Chown for the CFR combustion bomb apparatus was used in the DING setup [13]. The shield consists of a thin Inconel tube that is mounted concentrically around the glow plug tip. The gap size between the glow plug and the shield is 0.5 mm. A circular opening is provided to allow fuel jet contact. A drawing of the shield is provided in Appendix D of Reference [13].

### 3.5 Fuel System

A system diagram of the DING fuel system is show in Figure 3.5.

![Figure 3.5 – DING Engine Fuel System](image)

Natural gas is stored in two high pressure carbon steel tanks (TK-1 and TK-2) that are mounted on casters for mobility. The tanks are filled to a pressure of 3000 psig using a filling station that compresses low pressure utility gas from an Enbridge supply line. Average composition of the natural gas used in the thesis experiments is provided in Appendix A6. Each of the natural gas tanks has an independent isolation valve (HV-1&2). Outlets of each tank’s isolation valves are joined to common line with a common secondary isolation valve (HV-3). Tanks are stored outside of the test cell. The gas is then routed inside the test cell via a high pressure hose. The inlet of the hose is equipped with an excess flow valve (EFV-1) which shuts off the natural gas flow in case of hose rupture. The hose is then routed to a solenoid operated shutdown valve (SDV-1), which shuts the flow of gas in case of a power outage or an emergency shutdown.
Stainless steel tubing is used downstream of the shutdown valve. The gas pressure is reduced to the desired injection pressure by a hand operated pressure regulator (PRV-1). An additional excess flow valve (EFV-2) is provided for added safety. The gas is then filtered through a 15 micron filter (FIL-1). An isolation valve (HV-4) is provided before the gas is routed to the fuel injector (INJ-1) via a manifold (MF-1) and a flexible hose. A vent line with an isolation valve (HV-5) routed to a vacuum trench is provided to enable depressurization of the system when it is not in use. Natural gas injection pressure is measured by an AST4700 absolute pressure transducer (PI-1), which is recorded in the data acquisition system (DAQ). Temperature is measured with a K-type thermocouple (TI-1) and is displayed on the readout of the auxiliary control unit. The natural gas injector is solenoid-driven (closed when not energized). To prevent injector opening in an upset scenario, the injector power supply circuit is equipped with two safety relays. Safety relay SR-1 cuts the power to the injector whenever there is a trip event on the dynamometer control unit. Safety relay SR-2 cuts the power to the injector if the engine speed is below the low alarm (600 rpm) or above the high alarm (2000 rpm).

3.6 Air Intake System

The air intake system was designed to enable naturally aspirated and boosted operation of the DING engine. A system diagram of the DING air intake system is show in Figure 3.6.

Figure 3.6 – DING Engine Air Intake System

Under boosted operation, air is supplied from a 100 psig compressed air utility connection. Isolation valve (HV-6) is opened and isolation valve (HV-7) is closed. The utility air is reduced to boost pressures with a hand operated pressure regulator (PRV-2). Boost pressure is adjustable
from 0 to 20 psig. The air is passed through a 100 micron filter (FIL-2) and then through a laminar flow element (LFE-1). The laminar flow element is used to measure the air flow rate by correlating the flow rate to the pressure drop across the laminar flow element. Differential pressure transmitter (DP-1) measures the pressure drop, while a pressure transmitter (PI-2) measures the absolute pressure of the flowing air. These signals are recorded by the data acquisition system (DAQ). A K-Type thermocouple (TI-2) indicates the temperature of the flowing air on the auxiliary control unit and is also input by the user into the data acquisition system. The pressure drop, pressure, and temperature are used to calculate the mass air flow rate in the data processing routine. After the flow element, the air is passed to an intake plenum (PL-1), then through a Chromalox 1000W heater (HT-1), and finally to the intake port of the DING engine. The plenum is a cylindrical tank (12 in. diameter, 13 in. long) that acts as a flow buffer to smooth out the flow spikes that are typical for single-cylinder reciprocating engines. The heater is used to preheat the air to simulate compression-heated turbocharged air. The heater is controlled by the auxiliary control unit, with temperature feedback provided by a k-type thermocouple (TI-3) installed just downstream of the heater. Intake manifold pressure is recorded by an absolute pressure transducer (PI-3) installed just upstream of the intake valve. Naturally aspirated air intake is enabled by closing isolation valve (HV-6) and opening isolation valve (HV-7). In this case air is drawn from the test cell.

3.7 Exhaust and Emissions Sampling System

The DING engine exhaust system was designed to safely route exhaust to a vacuum trench while allowing sampling and analysis of the exhaust gasses. A system diagram is shown in Figure 3.7. The custom-built stainless steel exhaust system consists of 2.0 in. diameter stainless steel tubing, two flexible couplings (FC-1&2), the exhaust sampling manifold (ESM-1), and a muffler (MF-1). The flexible couplings isolate excessive engine vibration from the downstream exhaust components. An exhaust plenum (not shown) can be installed upstream of the sampling manifold in order to provide smoothing of the exhaust flow. The muffler is used to absorb excessive noise created in the exhaust system. The exhaust system is heat traced and insulated from the exhaust port up to the muffler with 1 in. thick fiberglass insulation. The heat tracing allows preheating of the exhaust system prior to testing, and the insulation keeps the system hot during tests. Keeping the system hot prevents condensation of exhaust gasses. Exhaust temperature is measured at the
exhaust port of the DING engine with a K-type thermocouple (TI-4). The temperature is displayed on the auxiliary control unit.

A Cambustion HFR-400 Fast Flame Ionization Detector (FFID) was installed to measure unburned hydrocarbon concentration in the exhaust. The FFID exhaust sample is taken through the FFID probe installed on exhaust port of the Lister ST head, just downstream of the exhaust valve. The close proximity of the probe to the exhaust valve and the quick response time of the FFID apparatus allow for measurement of unburned hydrocarbon concentrations within each engine cycle. FFID measurements are recorded in the data acquisition system (DAQ) with reference to the crank angle position. It should be noted that FFID measurements were only taken in the engine commissioning phase of the project. Thesis experiments used the FTIR analyzer for measurement of hydrocarbon species.

Unlike the FFID, all remaining emissions analyzers shown in Figure 3.7 provide time-averaged emissions measurements. The sampling connections for these instruments are provided on the exhaust sampling manifold (ESM-1), which consists of a 3.0 in. (76.2 mm) diameter, 36 in. (914.4 mm) long stainless steel tube. The following subsections describe the different emissions analyzers, listed in order of their connections on the exhaust sampling manifold.
**Connection S-1 – Dekati Diluter & Filter Cart (for future work)**

A KF-40 vacuum flange connection is provided at location S-1 for connection of a Dekati diluter. The diluter can be used with an exhaust sample filter to collect gravimetric exhaust particle samples. Gravimetric analysis was not performed in this study. The Dekati diluter connection was provided for future experiments.

**Connection S-2 – Engine Exhaust Particle Sizer (EEPS)**

Exhaust gas for particulate measurement is sampled with a TSI 379020A thermodiluter head connected at point S-2. The diluter head uses a 10 cavity rotating disk to withdraw a metered amount of exhaust gas and dilute it with preheated, HEPA-filtered dry air supplied from the TSI 379020A diluter control unit. Dilution of the raw exhaust sample is required for several reasons. Firstly, raw exhaust can reach temperatures higher than 400 °C, which could damage the EEPS analyzer. Dilution reduces the temperature of the sample to manageable levels. Secondly, particle concentration of raw exhaust can be at the top or higher than the maximum of the EEPS measurement range. Dilution of the raw exhaust brings the particulate concentration down into the measurement range. The dilution ratio is controlled by setting the dilution air flow rate and the rotational speed of the rotating thermodiluter disk. The diluted sample is routed to the diluter control unit and then to a TSI 379030 thermal conditioner, which heats the diluted sample to 300 °C in order to evaporate volatile components. This is done to prevent formation of nano-droplets which could erroneously register as solid particles by the EEPS analyzer. From the thermal conditioner, the sample is routed to a TSI 3090 Engine Exhaust Particle Sizer. The EEPS records one-second averages of the particle concentration as well as the particle size distribution for particles in the 5.6–560 nm range. The EEPS exhaust is passed through the LICOR CO₂ analyzer before being expelled into the vacuum trench. The LICOR analyzer is used to log the diluted exhaust CO₂ concentration. This measurement, along with the FTIR measurement of CO₂ in the raw exhaust, is used to determine the actual dilution ratio on the basis of a CO₂ mass balance. The dilution ratio calculation is covered in Chapter 4.

**Connection S-3 – Emissions Bench**

An Emissions Bench connection is provided at connection point S-3. A three-way switching valve (HV-7) is used to select between sampling the exhaust gasses from the manifold or
sampling the span gasses from the calibration line. When sampling the exhaust, the sample gas is passed through a heated filter (FIL-3) and then through a heated flexible sample line routed to the Emissions Bench. Both the heated filter and the sampling line are set to maintain a temperature of 191°C (376°F). This prevents condensation of the sample prior to reaching the analytical equipment in the Emissions Bench. When calibrating the Emissions Bench, span gasses of a known concentration are passed through a distribution manifold on the Emissions Bench to a calibration line. The calibration line routes the gasses to the three-way switching valve (HV-7), which diverts them to the heated filter (FIL-3) and the heated sample line. The calibration gasses replicate the exhaust sample path in order to be conditioned the same as an exhaust sample. The Emissions Bench consists of four analyzers. These are listed in Table 4.1 along with the species measured by each analyzer.

<table>
<thead>
<tr>
<th>Analyzer</th>
<th>Type</th>
<th>Measured Species</th>
</tr>
</thead>
<tbody>
<tr>
<td>CAI 600 HFID</td>
<td>Flame Ionization Detector</td>
<td>total hydrocarbons in ppm (propane basis)</td>
</tr>
<tr>
<td>CAI 600 HCLD</td>
<td>Chemiluminescence Gas Analyzer</td>
<td>NO, NO₂, NOx in ppm</td>
</tr>
<tr>
<td>CAI 601 P/NDIR</td>
<td>Paramagnetic O₂/Non-Dispersive IR CO₂</td>
<td>O₂ in %, CO₂ in %</td>
</tr>
<tr>
<td>CAI 602 NDIR</td>
<td>Non-Dispersive IR</td>
<td>CO in ppm</td>
</tr>
</tbody>
</table>

The Emissions Bench was used in the thesis experiments but was later found to be out of specification. For this reason, the Emissions Bench data was omitted from this thesis.

**Connection S-4 – Fourier Transform Infrared Spectrometer (FTIR)**

FTIR samples are taken and connection point S-4. Similarly to the Emissions Bench, the exhaust sample is passed through a heated filter (FIL-4) and then to the FTIR analyzer through a heated line. Both the filter and heated line are kept at a temperature of 191°C (376°F) to prevent sample condensation. The FTIR collects the raw sample absorption spectrum and interprets concentrations of various gasses by comparison with standard spectra. The gasses to be detected are defined in the *FTIR recipe*. The recipe used in the thesis experiments is covered in Chapter 4.

**Sample Probes**

The sampling probes used for the FTIR, Emissions Bench, and the EEPS system were designed to collect the exhaust sample across the entire cross section of the exhaust sampling manifold. The probes were mounted perpendicularly to the exhaust flow. A simplified diagram of an emissions probe is shown in Figure 3.8.
Each probe is made from 3/8 in. stainless steel tubing that has been perforated with 3/16 in. diameter holes. Three holes are drilled 1 in. apart in four sets around the circumference of the probe. The probe is capped at the end to force sample collection through the collection holes. The FTIR and Emissions Bench probes are installed in the horizontal configuration. The EEPS probe is installed vertically to capture the entire cross-section of the particulate vertical size gradient. This gradient is expected due to different settling velocities of different particle sizes.

### 3.8 Data Acquisition and Control System

The data acquisition and control system (DAQ) from the CFR combustion bomb apparatus was reused for the DING engine. The system’s hardware was overhauled to replace damaged components and was installed in a rack mounted cabinet. The DAQ system has two primary purposes: control of the fuel injector, and data acquisition of DING engine instruments. Table 3.2 shows the main input and output signals of the DAQ system.
Table 3.2 – DAQ System Inputs and Outputs

<table>
<thead>
<tr>
<th>Channel</th>
<th>Type</th>
<th>Instrument</th>
<th>AI/DI Measurement or AO/DO Function</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>AI</td>
<td>Kistler 6121 pressure transducer</td>
<td>in-cylinder pressure measurement</td>
</tr>
<tr>
<td>3</td>
<td>AI</td>
<td>Injector power supply current monitor (CI-1)</td>
<td>injector current measurement</td>
</tr>
<tr>
<td>4</td>
<td>AI</td>
<td>MAP 1 (PI-3)</td>
<td>intake air pressure measurement</td>
</tr>
<tr>
<td>5</td>
<td>AI</td>
<td>MAP 2 (PI-2)</td>
<td>laminar flow element pressure measurement</td>
</tr>
<tr>
<td>6</td>
<td>AI</td>
<td>AST4700 pressure transducer (PI-1)</td>
<td>natural gas pressure measurement</td>
</tr>
<tr>
<td>7</td>
<td>AI</td>
<td>Cussons differential pressure transducer (DP-1)</td>
<td>laminar flow element flow measurement</td>
</tr>
<tr>
<td>8</td>
<td>DI</td>
<td>Camshaft hall effect sensor</td>
<td>top dead center pulse counter</td>
</tr>
<tr>
<td>9</td>
<td>AO</td>
<td>Injector power supply</td>
<td>injector actuation pulse</td>
</tr>
<tr>
<td>10</td>
<td>DI</td>
<td>AVL Optical Encoder</td>
<td>crank angle degree counter</td>
</tr>
</tbody>
</table>

Legend: AI – analog input, DI – digital input, AO – analog output, DO – digital output

The control and data acquisition functionality are timed with crank angle of the DING engine’s crankshaft rotation. The crank angle degree position (CAD) is provided by an AVL optical encoder that was coupled to the crankshaft of the DING engine. The encoder provides a pulse at each 0.2 CAD. The control and acquisition logic are defined in a LabVIEW interface. The original control program for the CFR combustion bomb apparatus was developed by Fabbroni [25]. This program was modified to create new codes to allow skip-fired and continuously fired operation of the DING engine. Several new LabVIEW programs were also created for various experimental purposes. Table 3.3 outlines the LabVIEW programs used in the thesis experiments.

Table 3.3 – DAQ LabVIEW Programs

<table>
<thead>
<tr>
<th>LabVIEW Program</th>
<th>Program Purpose</th>
<th>Program Origin</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydra DING 2015.vi</td>
<td>Injection control and full test data acquisition – used in skip-fired tests only</td>
<td>Modified version of BombDAQ_C2.vi</td>
</tr>
<tr>
<td>Hydra DING 2015 Emissions.vi</td>
<td>Injector control and in-cylinder pressure acquisition only – used for fired emissions tests only</td>
<td>Modified version of BombDAQ_C2.vi</td>
</tr>
<tr>
<td>MassTest_C.vi</td>
<td>Injector control only – used for injector calibration</td>
<td>Original code by Fabbroni [25].</td>
</tr>
<tr>
<td>FFIDCal_C.vi</td>
<td>FFID signal readout – used for FFID calibration</td>
<td>Original code by Fabbroni [25].</td>
</tr>
<tr>
<td>Intake Pressure Readout.vi</td>
<td>Average intake pressure readout – used to set DING intake pressure</td>
<td>New Code</td>
</tr>
<tr>
<td>Pressure Transducer Calibration.ci</td>
<td>Simultaneous Readout of the MAP 1, MAP 2, and Omega PX309 pressure transducers – used for pressure transducer calibration</td>
<td>New Code</td>
</tr>
</tbody>
</table>

Details on the DAQ hardware and software design are available in Reference [25].
3.9 Apparatus Calibration

All instrumentation used in the DING engine experiments was calibrated prior to conducting tests. Calibration of the in-cylinder pressure (Kistler 6121), intake pressure (MAP1/PI-3), laminar flow meter pressure (MAP2/PI-2), laminar flow element differential pressure (DP-1), and the natural gas (AST4700/PI-1) pressure transducers was performed in the DING engine commissioning phase. Calibration procedures and calibration curves are presented in Appendix A7. Fuel injector calibration was performed prior to every test day. Fuel injector preparation and calibration procedures are documented in Appendix A8.
Chapter 4

4 Experimental Design

The experimental program in this thesis consisted of three major test programs:

1. Shield and injector angle screening tests
2. Skip-fired ignition delay tests
3. Fired emissions tests

The design of each test program is explained in the subsequent sections. Key calculation methodologies are also presented to explain generation of results from the measured data.

4.1 Glow Plug Shield and Injector Angle Screening Tests

Initial commissioning of the DING engine showed poor combustion performance. Ignition often had to be achieved by supplying the glow plug with over 15.5 VDC power, well in excess of its design rating of 11 VDC. This practice was unsustainable since at this voltage level the glow plug survived several minutes of operation and then burnt out, shattering in the reciprocating engine. Further commissioning showed that a 14 VDC glow plug voltage level struck a balance between glow plug life and the ability to initiate combustion. Commissioning tests also indicated that different glow plug shield and injector angles resulted in vastly different performance. Only a few combinations were tested during commissioning, warranting further investigation.

The effects of the glow plug shield and injector angles on combustion initiation were investigated by audible detection. Occurrence of combustion events was easily detected audibly, by ear. This methodology was confirmed by comparing early audible detection test results with combustion pressure traces. The observed performance was classified into the following categories:

- no combustion
- poor combustion, > 75% misses
- poor combustion, 50–75% misses
- repeatable combustion, < 25% misses
- repeatable combustion, no misses
Leveraging the findings from a similar test sequence done on the CFR combustion bomb apparatus by Chown [23], the test was limited to shield angles that place the shield opening downstream of the engine swirl. Upstream orientation would result in swirl cooling of the glow plug surface. Glow plug shield angle was tested in 20° increments, ranging from 0° to 80° in engine swirl direction. Injector angle was tested in 10° increments, ranging from 0° to 30° in engine swirl direction. Figure 4.1 shows an illustration of the tested shield and injector angle combinations. It should be noted that Figure 4.1 is provided for the visualization only and does not represent actual fuel jet contours.

![Figure 4.1 – Shield and Injector Angle Screening Test Matrix](image)
For each shield-injector angle combination, five different injection durations were tested. Injection durations were evenly spaced and covered equivalence ratios ranging from 0.1 to 0.43. Successful shield-injector angle combinations were tested at additional injection durations corresponding to an equivalence ratio of 0.5. In total, 20 different shield-injector arrangements were tested over five different injection durations. Each test consisted of 20 fired cycles, with 3 skipped cycles in between fired cycles. Tests were conducted at an intake pressure of 11 psig, ambient air temperature of 25°C, and an engine speed of 1000 rpm.

4.2 Skip-Fired Ignition Delay Tests

The second test program explored the effects of intake pressure and equivalence ratio on ignition performance. The key investigated parameter was the ignition delay. Since the ignition delay calculation requires motored (unfired) cycles as a reference, skip-fired testing methodology was required. The following sections explain the setup and results calculation methodology for the skip-fired test series.

4.2.1 Engine Operating Conditions

The following test conditions were determined for the skip-fired ignition delay tests:

**Engine Speed:** Engine speed of 1000 rpm was selected for the skip-fired test series. This was deemed as the top speed setting at which the engine can operate continuously with an acceptable level of risk with respect to the mechanical integrity of the Bowditch piston assembly and the oil-less polyamide piston rings. Engine speed in excess of 1000 rpm posed a risk of engine damage.

**Intake Pressure Range:** Three intake pressures of 5, 10, and 15 psig were selected for the test. These values are typical of the boost levels found in diesel engines.

**Intake Air Temperature:** Ambient air was used for the experiments. Although the capability to heat intake air was installed, it was not used due to the already extensive experimental scope.

**Start of Injection:** Optimal injection timing was determined by testing different injection timing starting with 350 CAD BTDC. Injection timing was incrementally advanced until the combustion pressure trace deviated from the preceding motored trace just prior to TDC. This
approach was applied across the proposed test intake pressures and equivalence ratios. It was found that start of injection (SOI) at 335 CAD BTDC was a good fit for all test conditions.

**Equivalence Ratio Range:** Equivalence ratio is controlled by the fuel injection duration. The maximum injection duration was limited to avoid fuel injection past 30 CAD ATDC. Beyond this value the combustion chamber volume expands rapidly with increasing crank angle and combustion energy is captured less efficiently. Keeping to this criterion, four nominal equivalence ratios of 0.2, 0.3, 0.4, and 0.5 were selected. These values were determined through extensive fuel injector calibration and are based on the widest range of equivalence ratios that was achievable with the injector nozzle at all the test pressures. This equivalence ratio range is also representative of diesel equivalence ratios from idle to approximately mid-load conditions [21, p. 53].

**Fuel Injection Pressure:** Fuel injection pressure of 1800 psig was selected to enable choked flow of the natural gas fuel against the peak motored in-cylinder pressure at each of the three intake pressures. This is desired so the injection flowrate is constant for all test conditions. The highest mean motored pressure of 5738 kPag (832 psig) was achieved for the 15 psig intake pressure. The ratio of the peak motored pressure to the injection pressure $P_{\text{motored}}/P_{\text{injection}}=0.462$ is well under the critical pressure ratio of 0.546 [21, p. 909] for natural gas, thus assuring choked flow of the natural gas fuel.

**Glow Plug Voltage:** Glow plug voltage of 14 VDC was used as it was determined to provide a good balance between glow plug life and glow plug power required to achieve repeatable combustion.

**Glow Plug Shield and Injector Angle:** A glow plug shield angle of 60° and an injector angle of 0° in engine swirl direction were used for all tests. These were determined as optimal settings from the earlier screening test sequence.

### 4.2.2 Skip-Fired Test Data Acquisition

Table 4.1 lists the parameters tracked by the LabVIEW data acquisition system (DAQ) for the skip-fired ignition delay tests. Each parameter was logged with respect to the crank angle at a 0.2 CAD resolution for the entire duration of each skip-fired test.
Table 4.1 – DAQ Parameters Tracked in Ignition Delay Tests

<table>
<thead>
<tr>
<th>DAQ Channel</th>
<th>Tracked Parameter</th>
<th>Used In Calculating</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>In-cylinder pressure</td>
<td>ignition delay, ROHR, IMEP</td>
</tr>
<tr>
<td>3</td>
<td>Injector solenoid current</td>
<td>ignition delay</td>
</tr>
<tr>
<td>4</td>
<td>Intake pressure</td>
<td>in-cylinder pressure trace</td>
</tr>
<tr>
<td>5</td>
<td>Intake laminar flow meter pressure</td>
<td>actual equivalence ratio</td>
</tr>
<tr>
<td>7</td>
<td>Intake laminar flow meter flow rate</td>
<td>actual equivalence ratio</td>
</tr>
</tbody>
</table>

Other parameters remained relatively constant during a test. These were manually entered into the DAQ system at the beginning of each test.

4.2.3 Skip-Fired Test Key Calculations

The majority of the calculations required for the skip-fired ignition delay tests were carried out in a MATLAB routine (*HydraDAQDataProcessing2015R2.m*). Key calculations are explained in the following sections.

4.2.3.1 Ignition Delay Determination

Ignition delay ($\tau_{ig}$) was the key parameter of this test program. The MATLAB code was used to process the acquired test data and to calculate all test results. Ignition delay for every fired cycle was calculated as the time between the true start of injection (TSOI) to the time of ignition (TOI).

$$\tau_{ig} = time_{@TOI} - time_{@TSOI}$$  \hspace{1cm} (Eq. 4.1)

Determination of the True Start of Injection (TSOI)

The commanded the start of injection (SOI) is defined by the user in the *Hydra DING 2015.vi* control and data acquisition file. Actual fuel injection, however, occurs after the injector solenoid receives power and the injector needle lifts. To detect the true start of injection (TSOI) for each fired cycle, the MATLAB routine scans through the acquired injector current trace until it detects a current rise above a threshold value. From this time point, 0.55 ms of needle lift delay (NLD) time is added to account for the time required for the injector needle to lift and allow the fuel to flow. The NLD time was experimentally determined during injector calibration. It was shown to be constant across different injection durations.
\[ TSOI = time_{(injector\ current\ rise)} + NLD \]  
(Eq. 4.2)

**Time of Ignition (TOI)**

Time of ignition (TOI) is determined by comparing the fired cycle in-cylinder pressure trace with the average pressure trace of the preceding motored cycles. Since the in-cylinder pressure signal can be quite noisy, all pressure traces are first smoothed using a Savitzky-Golay filter function (part of the MATLAB Signal Processing Toolbox). The optimum filter settings were determined through iterative testing of several early result files. Filter polynomial order 4 and a frame size of 51 were determined as settings that provide the best balance between noise reduction and good pressure signal definition. Once filtered, the MATLAB code averages the pressure traces of the motored cycles preceding each fired cycle. A difference between the fired pressure trace and the average motored pressure trace is determined for each motored-fired cycle set. The code then calculates a threshold differential pressure (TDP) value. This value is equal to three standard deviations (3\(\sigma\)) of the differential pressure trace over 100 CAD prior to the commanded start of injection. The MATLAB routine scans the fired pressure trace and finds the first point where 5 consecutive fired cycle pressure points exceed the average motored pressure by more than the TDP value. This point is defined as the time of ignition. Figure 4.2 illustrates the procedure of determining TSOI and TOI values.
The above methodology requires at least one motored cycle to precede each fired cycle, necessitating skip-fired operation of the engine. The number of the skipped cycles was determined in the commissioning stage by testing the dynamometer’s ability to control engine speed. One and two skipped cycle operation resulted in poor speed control due to the inability of the dynamometer to vary engine load in response to the fired cycles. Three skipped cycles were sufficient for lower equivalence ratio tests, and four skipped cycles were required for higher equivalence ratio tests.

4.2.3.2 Actual Equivalence Ratio ($\phi_a$)

For the skip-fired tests the actual test equivalence ratio was calculated by the MATLAB code using the following equation:

$$\phi_a = \frac{(F/A)}{(F/A)_{stoic}}$$

(Eq. 4.3)
Where:

\( \varphi_a \) is the actual equivalence ratio

\((F/A)\) is the actual fuel to air mass ratio

\((F/A)_{stoic} \) is the stoichiometric fuel to air mass ratio

The calculation was performed using the intake air flow measured by the laminar flow meter and the fuel mass injection rates established during injector calibration.

### 4.2.3.3 Rate of Heat Release (ROHR)

To enable combustion mode analysis, the MATLAB code was designed to calculate differential net rate of heat release (\( \text{ROHR}_{\text{net}} \)) curves for each skip-fired test. First, the net rate of heat release (\( \text{ROHR}_{\text{net}} \)) is calculated for the motored and the fired pressure traces using the following equation:

\[
\text{ROHR}_{\text{net}} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{1 - \gamma} V \frac{dP}{d\theta}
\]

\((\text{Eq. 4.4})\)

Where:

\( \text{ROHR}_{\text{net}} \) is the calculated net rate of heat release in kJ/CAD

\( \frac{dV}{d\theta} \) is the change in cylinder volume in m\(^3\)/CAD, calculated from the engine geometry

\( \frac{dP}{d\theta} \) is the change in pressure in kPa/CAD, calculated from the in-cylinder pressure trace

\( P \) is the in-cylinder pressure in kPa, given by the in-cylinder pressure trace

\( V \) is the cylinder volume in m\(^3\), calculated from the engine geometry

\( \gamma \) is the ratio of specific heats for air \((c_p/c_v)\)

\( \theta \) is the crank angle in crank angle degrees (CAD)
The calculated ROHR\textsubscript{net} is a composite term which accounts for several energy inputs and sinks. The major ones are the combustion heat input rate and the heat loss rate (heat loss to the cylinder walls). The minor contributors are the enthalpy addition from the injected fuel, and the loss of enthalpy from piston blowby. Since these contributions are difficult to calculate on individual basis, they are all accounted for in the ROHR\textsubscript{net} term. The above equation was derived from the first law of thermodynamics using assumptions of quasi-equilibrium conditions and ideal gas behavior.

In the MATLAB code, Eq. 4.4 is implemented in a discrete form with the $\frac{dV}{d\theta}$ and $\frac{dP}{d\theta}$ terms replaced with discrete versions, $\frac{\Delta V}{\Delta \theta}$ and $\frac{\Delta P}{\Delta \theta}$. A constant value of the specific heat ratio $\gamma = 1.325$ is used (an average of two typical values suggested in [21, p. 510]). Once ROHR\textsubscript{net} is known for the fired and the preceding average motored pressure traces, the differential net rate of heat release ROHR\textsubscript{dnet} is calculated:

$$ \text{ROHR}_{dnet} = \text{ROHR}_{\text{net fired}} - \text{ROHR}_{\text{net motored}} \quad (\text{Eq. 4.5}) $$

The resulting ROHR\textsubscript{dnet} term represents the net rate of heat release added by the combustion event. Once plotted, ROHR\textsubscript{dnet} curves show high levels of noise. This noise is a result of the discrete stepwise calculation of the heat release, which amplifies any unfiltered residual noise from the pressure trace filtering. To enable examination of the general shape of the heat release, the ROHR\textsubscript{dnet} curves are filtered with a Savitzky-Golay algorithm using a polynomial order of 4 and a frame size of 201. This influences the depicted magnitude of the heat release, but makes the curve’s shape easy to interpret. To capture the true magnitude of the heat release, the maximum heat release, ROHR\textsubscript{max}, and the total heat release, ROHR\textsubscript{total}, are computed from the unfiltered ROHR\textsubscript{dnet} curves.

It should be noted that ROHR\textsubscript{dnet} curves are simply referred to as ROHR curves in subsequent sections of this thesis.
4.2.3.4 Mean In-Cylinder Temperature

Mean in-cylinder temperatures were computed in MATLAB from intake valve closure (IVC) to exhaust valve opening (EVO). Temperature was calculated using the ideal gas law along with the following assumptions:

- Air mass inducted into the cylinder (measured by the DAQ system) remains constant from IVC to EVO. Blowby losses are assumed to be negligible, and are excluded from the calculation for simplicity.

- Fuel is assumed to be injected at a constant rate from the start of injection (SOI) and lasts for the injection duration (defined in the LabVIEW control software). Fuel mass injection rate is determined from injector calibration.

- Total in-cylinder mass at any crank angle position between IVC and EVO consists of the inducted air and the fuel injected up to that point in the cycle.

- As the fuel combusts, the total moles of in-cylinder gasses remains constant (assumption validated in Section 4.5.1.5)

With the in-cylinder pressure and engine volume known (pressure measured in the DAQ system, volume calculated from engine geometry), and the in-cylinder mass and molar content defined (defined above), the calculation of temperature with the ideal gas law is straightforward. The MATLAB code calculates the mean in-cylinder temperature at every 0.2 CAD step between IVC and EVO. It should be noted that the calculated temperatures represent the mean in-cylinder temperatures. In reality, the temperature in the cylinder varies from the cold air charge temperature to the high temperatures of the flame front. The calculated mean in-cylinder temperatures, however, are useful in determining mean temperature trends with respect to equivalence ratio and intake pressure.

4.2.3.5 Indicated Mean Effective Pressure (IMEP)

Indicated mean effective pressure (IMEP) is a parameter calculated by dividing the work per engine cycle by the cylinder volume displaced per cycle. This parameter is indicative of engine performance, with higher values being favorable. There are two types of IMEP, gross and net.
The gross IMEP, or $\text{IMEP}_{\text{gross}}$, only accounts for work done over the compression and expansion strokes. The net IMEP, or $\text{IMEP}_{\text{net}}$, accounts for the entire engine cycle, including intake and exhaust strokes. $\text{IMEP}_{\text{gross}}$ is especially useful in combustion analysis as its values are directly tied to the work of combustion and exclude the mechanical inefficiencies, which may vary engine to engine, of the intake and exhaust strokes. $\text{IMEP}_{\text{gross}}$ and $\text{IMEP}_{\text{net}}$ were calculated using the following formula:

$$\text{IMEP} = \frac{(\sum_{\theta=a}^{b} P_{\theta} \Delta V_{\theta})}{V_{d}} \quad (\text{Eq. 4.6})$$

*Where:*

- $\text{IMEP}$ is the calculated indicated mean effective pressure in kPa
- $P_{\theta}$ is the in-cylinder pressure in kPa at crank angle $\theta$
- $\Delta V_{\theta}$ is the change in cylinder volume in m$^3$, from $\theta$ to $\theta+0.2$ CAD
- $V_{d}$ is the engine displacement in m$^3$
- $a$ & $b$ are the summation limits in CAD

For $\text{IMEP}_{\text{gross}}$, the summation is carried out only over the compression and expansion strokes ($a=180$, $b=540$ CAD). For $\text{IMEP}_{\text{net}}$, the summation is carried out over the entire four-stroke cycle ($a=0$, $b=720$ CAD). It should be noted that since data is acquired every 0.2 CAD, the change in volume is defined as $\Delta V_{\theta} = V_{\theta+0.2} - V_{\theta}$.

### 4.3 Combined Skip-Fired and Fired Emissions Test Sequence

Figure 4.3 shows a combined test sequence that includes test preparation, the skip-fired test series, and the fired emissions test series. It should be noted that the fired emissions tests were conducted right after completion of the skip-fired tests for each of the tested intake pressures. This was done to share the injector calibration and engine preparation efforts, and to keep test conditions for the two test types as close to one another as possible. The latter requirement is critical as it allows for comparison of the different test results between the two test types. Figure 4.3 summarizes the 12 tested conditions: 3 different intake pressures (5, 10 and 15 psig) and four
nominal equivalence ratios ($\phi_n$ of 0.2, 0.3, 0.4 and 0.5). These test conditions were used for both the skip-fired ignition delay, and the fired emissions tests. Several key components of this test sequence are discussed below.

![Figure 4.3 – Combined Skip-Fired and Fired Emissions Test Sequence](image)

**Test Preparation**

The major components of test preparation are shown in the first column of Figure 4.3. Due to injector needle tip wear from previous tests, the fuel injector had to be completely overhauled prior to each test. Key steps of the overhaul included: thorough cleaning of the injector components, resurfacing of the injector needle tip, resetting of the injector spring preload, and resetting of the injector lift distance. Once overhauled, the injector was calibrated and reinstalled in the DING engine cylinder head. Details on injector overhaul and calibration procedures are documented in Appendix A8.

The DING engine itself also had to be torn down and overhauled prior to every test. Key steps included: engine block disassembly, thorough cleaning and lubrication of the piston rings and cylinder surface; cleaning and reinstallation of the engine cylinder head and valve train; and installation of a new glow plug. An engine test preparation check sheet is presented in Appendix B1.
After reassembling the injector and engine block, the engine was warmed up by motoring, heated recirculation of the engine oil and coolant, and resistance heating of the cylinder head. During this time the emissions equipment, used in fired emissions test, was calibrated. After checking all engine and auxiliary systems, the intake pressure was set at the air supply regulator while motoring the engine. A LabVIEW program, *Intake Pressure Readout.vi*, was created to display the average intake pressure. Using this program allows for precise setting of the intake pressure. Once the intake pressure was set, the testing sequence was initiated.

**Motored Tests**

Motored tests were an important feature of the both the skip-fired ignition delay and the fired emissions test sequences. Motored tests allowed monitoring of engine compression by indicating the peak motored pressure. If the pressure deviated significantly from the previous tests at the same intake pressure setting, tests were halted until the underlying issue was resolved. One example of such an issue was severe wear or breakage of the polyamide piston rings. For the fired emissions tests, the motored tests were also used to establish the value of the frictional torque. This is further explained in Section 4.5.1.2.

### 4.4 Fired Emissions Tests

The aim of the fired emissions tests was twofold:

1. Prove and study continuous operation of the engine
2. Collect the first emissions measurements for the glow plug ignition assist technology

Considerations and the design of the continuously fired tests are discussed in the following subsections.

#### 4.4.1 Mechanical Considerations

The optical access capability of the DING engine limits the speed of continuous fired tests. The additional Bowditch piston creates an unnatural load for the engine testbed system, which is designed to operate without this additional mass. In addition, the polyamide oil-less rings have not been previously used under continuously fired engine operation. Potential for piston ring damage was unknown.
The engine’s capability of continuously fired operation was investigated through a series of careful commissioning tests. Engine speed, engine load (controlled by injection duration), engine intake pressure, and test duration were incrementally increased. Engine sound was carefully monitored to detect operational issues. Tests showed that operation at 1000 rpm at an intake pressure of 15 psig and an equivalence ratio of approximately 0.5 was sustainable. Test duration was limited to 6 minutes in length. This figure was selected to minimize the risk of piston ring damage, while allowing engine exhaust emissions levels to start to stabilize by the end of the test. Running longer test durations would increase the risk of piston ring or engine failure. In fact, one set of the polyamide piston rings failed during later commissioning tests. Implementation of new Teflon-based piston rings was attempted as a contingency, but proved unsuccessful. The last remaining set of polyamide piston rings was reserved for the thesis experiments. This set, however, had to be modified to avoid engine seizure from excessive thermal expansion. To mitigate the risk of failure to this last ring set, the initial test procedure was revised to include re-oiling the underside of the piston rings between different intake pressure test series. Further discussion of the Teflon piston ring trials and modification of the remaining polyamide piston ring set are documented in Appendix A4.

### 4.4.2 Fired Test Data Acquisition

Data acquisition for the fired tests varied from the skip-fired tests. Due to computing power limitations the control and data acquisition system was unable to continuously record the parameters listed in Table 4.1. A simplified version of the skip-fired LabVIEW program, *Hydra DING 2015 Emissions.vi*, was created to act solely as a control system for natural gas injector. All test parameters were recorded manually. Table 4.2 lists the tracked parameters, their source, as well as the results calculations that require these parameters. All of the listed parameters were recorded at motored conditions prior to the test, and then at 30 sec, 1 min, 2 min, 3 min, 4 min, 5 min, 6 min into the test.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Units</th>
<th>Source</th>
<th>Used In Calculating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake Air Temperature</td>
<td>°C</td>
<td>Auxiliary Control Unit</td>
<td>ISFC</td>
</tr>
<tr>
<td>Intake Air Flow</td>
<td>in H₂O</td>
<td>Cussons Control Cabinet</td>
<td>ISFC</td>
</tr>
<tr>
<td>Torque</td>
<td>Nm</td>
<td>Cussons Control Cabinet</td>
<td>IMEPₙ𝑒𝑡</td>
</tr>
<tr>
<td>Exhaust Temperature</td>
<td>°C</td>
<td>Auxiliary Control Unit</td>
<td>Test monitoring</td>
</tr>
<tr>
<td>Engine Speed</td>
<td>rev/s</td>
<td>Cussons Control Cabinet</td>
<td>ISFC, IMEPₙₑᵗ</td>
</tr>
</tbody>
</table>

*Table 4.2 – Parameters Tracked in Fired Emissions Tests*
4.4.3 Fired Test Emissions Measurement

Emissions were measured from approximately two minutes prior, to full completion of every fired test. Pre-test measurements allowed for establishing background emissions levels and for verification of the emissions equipment. The FTIR analyzer was used to measure all gaseous emissions. The “DING-simple” recipe, a simplified subset of the MKS “NG Engines 1Hz R3” recipe, was used for processing of the FTIR spectra. The EEPS analyzer was used to measure particulate emissions. The LICOR instrument was used to measure the CO₂ levels of the EEPS diluted exhaust. Table 4.3 summarizes the measured emissions.

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Sampling Rate</th>
<th>Emissions Measured</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>FTIR</td>
<td>0.98 Hz</td>
<td>CO₂, CO, H₂O, CH₄, C₂H₆, C₂H₄, NO, NO₂ (“DING-simple” recipe)</td>
<td>ppm or % (wet basis)</td>
</tr>
<tr>
<td>EEPS</td>
<td>1 Hz</td>
<td>Particle count by size in the 6.04 to 523.3 nm size range</td>
<td>#/cm⁶</td>
</tr>
<tr>
<td>LICOR</td>
<td>1 Hz</td>
<td>CO₂ in diluted exhaust stream (used for EEPS dilution ratio calculation)</td>
<td>ppm (wet basis)</td>
</tr>
</tbody>
</table>

4.4.4 Fired Test Key Calculations

Calculations of key performance parameters for fired emissions test are covered below.

4.4.4.1 Actual Equivalence Ratio (φₐ)

Injector performance can change throughout the fired test due to changes in engine cylinder head temperature, injector seal wear, as well as other factors. Calculation of equivalence ratio on the basis of constant known fuel injection amount (determined from calibration) may yield inaccurate estimates of the actual equivalence ratio. A better approach is suggested in Reference [21, p. 151]. This method uses the measured emissions concentration to back-calculate an air/fuel ratio on a basis of a carbon balance. Equivalence ratio is then determined from the air/fuel ratio. The equations for this method originated from Reference [21, p. 151], but were modified. Methane was assumed to be the fuel and the major contributor to unburnt hydrocarbons. This assumption is deemed valid since the natural gas fuel used in the experiments is >94% methane and FTIR measurements showed non-methane unburnt hydrocarbons to be lower than 5% of the unburnt methane concentration. The modified equations are presented below [24]:
\[
\frac{A}{F} = \frac{MW_{\text{air}}}{MW_{\text{fuel}}} \left[ \frac{100 + (CH_4) - 0.5(CO) + 0.5(H_2O)}{(CH_4) + (CO) + (CO_2)} \right] - 2 \]  
(Eq. 4.7)

\[
H_2O = 2 \left( \frac{(CO_2)+(CO)}{(CO)/[3.8(CO_2)]+1} \right) \]  
(Eq. 4.8)

\[
\varphi_a = \frac{(A/F)_{\text{stoic}}}{(A/F)} \]  
(Eq. 4.9)

Where:

- \(\varphi_a\) is the calculated actual test equivalence ratio
- \((A/F)\) is the calculated mass air to fuel ratio
- \((A/F)_{\text{stoic}}\) is the stoichiometric mass air to fuel ratio (determined from stoichiometry)
- \(MW_{\text{air}}\) is the molecular mass of air (28.96 g/mol)
- \(MW_{\text{fuel}}\) is the molecular weight of methane fuel (16.04 g/mol)
- \((CH_4), (CO), (CO_2)\) are the molar concentrations of combustion products in % (wet basis)
- \((H_2O)\) is the molar concentration of water formed by combustion in % (calculated from Eq. 4.8)

### 4.4.4.2 Indicated Net Torque \((T_i)\)

Indicated net torque \((T_i)\) was calculated to support several key calculations. The Cussons dynamometer cabinet includes a display of the engine brake torque \(T_b\) (engine torque at the shaft). In order to estimate the indicated torque \(T_i\), knowledge of the frictional torque \(T_f\) (torque required to overcome engine friction) is required. To estimate the frictional torque, brake torque was measured at motored conditions before and after each of the fired emissions tests. The motored brake torque after each test was found to be lower than prior to the test. This is likely a consequence of the depleting cylinder wall oil layer and piston ring wear over the duration of each test. Frictional torque at each minute of the continuous test was estimated by linear interpolation between motored brake torque before and after each test. The indicated torque was then calculated as follows:
\[ T_i = T_{b(test)} - T_{f(estimated)} \]  
(Eq. 4.10)

Where:

\( T_i \) is the calculated indicated net torque in Nm

\( T_{b(test)} \) is the brake torque in Nm measured during the fired test

\( T_{f(estimated)} \) is the frictional torque in Nm estimated by interpolation between motored brake torque measured before and after each fired test

### 4.4.4.3 Net Indicated Mean Effective Pressure (IMEP\textsubscript{net})

Since an in-cylinder pressure trace was not available for fired emissions tests, only the net mean effective pressure (IMEP\textsubscript{net}) was calculated for each test. This calculation used the following formula [24]:

\[ IMEP_{net} = \frac{2\pi T_i}{V_d} \]  
(Eq. 4.11)

Where:

IMEP\textsubscript{net} is the net indicated mean effective pressure in kPa

\( T_i \) is the indicated net torque in Nm

\( V_d \) is the engine displacement in dm\textsuperscript{3}
4.4.4.4 Indicated Specific Fuel Consumption (ISFC)

Indicated specific fuel consumption (ISFC) was calculated to determine the fuel consumption of the DING engine. ISFC was calculated using the following formula [24]:

\[ ISFC = \frac{\dot{m}_f}{2\pi NT_i} \]  
(Eq. 4.12)

Where:

- \( ISFC \) is the calculated indicated specific fuel consumption in g/kW-h
- \( \dot{m}_f \) is the fuel mass injection rate in g/hr (back calculated from the actual equivalence ratio and the measured air mass flow)
- \( N \) is the engine speed in rev/s
- \( T_i \) is the indicated net torque in Nm

4.4.4.5 Combustion Chemistry Calculations

A simplified approach was used to determine combustion chemistry of natural gas. Since the natural gas used in all tests was > 94% methane, combustion chemistry is modelled as methane combustion:

\[ CH_4 + \frac{2}{\phi} (O_2 + 3.773N_2) \rightarrow aCO_2 + bCO + cCH_4 + dH_2O + eH_2 + fO_2 + gN_2 \]

The chemistry is subject to conservation of C, H, O, and N atoms. An atom balance is performed to define the stoichiometric coefficients:

- \( C \) atoms: \( 1 = a + b + c \)
- \( H \) atoms: \( 4 = 4c + 2d + 2e \Rightarrow e = 2 - 2c - d \)
- \( O \) atoms: \( 2/\phi(2) = 2a + b + d + 2f \Rightarrow f = 2/\phi - a - 0.5b - 0.5d \)
- \( N \) atoms: \( 2(3.773)(2)/\phi = 2g \Rightarrow g = 7.546/\phi \)
Using the coefficients determined above, the stoichiometric coefficients of the reactants and products can be defined as follows:

<table>
<thead>
<tr>
<th>Species</th>
<th>Reactant moles</th>
<th>Product moles</th>
</tr>
</thead>
<tbody>
<tr>
<td>CH₄</td>
<td>1</td>
<td>c</td>
</tr>
<tr>
<td>CO₂</td>
<td>0</td>
<td>a</td>
</tr>
<tr>
<td>CO</td>
<td>0</td>
<td>b</td>
</tr>
<tr>
<td>H₂O</td>
<td>0</td>
<td>d</td>
</tr>
<tr>
<td>H₂</td>
<td>0</td>
<td>2−2c−d</td>
</tr>
<tr>
<td>O₂</td>
<td>2/φ</td>
<td>2/φ−a−0.5b−0.5d</td>
</tr>
<tr>
<td>N₂</td>
<td>7.546/φ</td>
<td>7.546/φ</td>
</tr>
<tr>
<td>Total</td>
<td>1+9.546/φ</td>
<td>2+9.546/φ−c+0.5b−0.5d</td>
</tr>
</tbody>
</table>

The total moles of products can be expressed as $N_{product} = 2+9.546/\phi−c+0.5b−0.5d$. Mole fractions of the exhaust components are defined as follows:

$$\chi_{CH₄} = \frac{c}{N_{prod}} = \frac{c}{2 + 9.546/\phi − c + 0.5b − 0.5d}$$

$$\chi_{CO₂} = \frac{a}{N_{prod}} = \frac{a}{2 + 9.546/\phi − c + 0.5b − 0.5d}$$

$$\chi_{CO} = \frac{b}{N_{prod}} = \frac{b}{2 + 9.546/\phi − c + 0.5b − 0.5d}$$

$$\chi_{H₂O} = \frac{d}{N_{prod}} = \frac{d}{2 + 9.546/\phi − c + 0.5b − 0.5d}$$

Since the mole fractions of CH₄, CO₂, CO, and H₂O are all measured by the FTIR, above equations can be solved (4 equations, 4 unknowns). With b, c, and d known, $N_{product}$ can be determined and the remaining moles fractions of H₂, O₂, and N₂ (not measured by the FTIR) can be calculated from the equations below:

$$\chi_{H₂} = \frac{2 − d − 2c}{N_{prod}}$$

$$\chi_{O₂} = \frac{2/\phi − a − 0.5b − 0.5d}{N_{prod}}$$

$$\chi_{N₂} = \frac{7.546/\phi}{N_{prod}}$$
For ideal combustion the above calculation can be greatly simplified by assuming conservation of moles from the reactant to the product side. This is valid in ideal combustion of methane due to methane’s unique stoichiometry. The number of reactant moles equals the product moles as shown in idealized combustion equation:

\[
CH_4 + 2(O_2 + 3.773N_2) \rightarrow CO_2 + 2H_2O + 7.546N_2
\]

In real combustion, the presence of CO and H\textsubscript{2}, however, can offset the conservation of moles assumption. Reactions involved in forming these species do not conserve moles from the reactant to the product side. To test the impact of these species, sensitivity analysis was performed. Results for 5 psig intake pressure test series were used for the analysis since they exhibited the highest levels of CO in the exhaust. Coefficients a, b, c, and d were iterated until the calculated mole fractions of CO and CH\textsubscript{4} matched the test results. The calculated mole fractions are presented in Table 4.4.

**Table 4.4 – Sensitivity Analysis Mole Fractions (5 psig Intake Pressure Series)**

<table>
<thead>
<tr>
<th>(\phi_a)</th>
<th>(\chi_{CH_4})</th>
<th>(\chi_{CO_2})</th>
<th>(\chi_{CO})</th>
<th>(\chi_{H_2O})</th>
<th>(\chi_{H_2})</th>
<th>(\chi_{O_2})</th>
<th>(\chi_{N_2})</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.23</td>
<td>0.001556</td>
<td>0.021550</td>
<td>0.000412</td>
<td>0.043512</td>
<td>0.000412</td>
<td>0.160987</td>
<td>0.771573</td>
</tr>
<tr>
<td>0.32</td>
<td>0.000710</td>
<td>0.031316</td>
<td>0.000396</td>
<td>0.063027</td>
<td>0.000396</td>
<td>0.139609</td>
<td>0.764546</td>
</tr>
<tr>
<td>0.40</td>
<td>0.000399</td>
<td>0.037813</td>
<td>0.001928</td>
<td>0.077553</td>
<td>0.001928</td>
<td>0.123145</td>
<td>0.757235</td>
</tr>
<tr>
<td>0.45</td>
<td>0.000292</td>
<td>0.040513</td>
<td>0.004031</td>
<td>0.085058</td>
<td>0.004031</td>
<td>0.114216</td>
<td>0.751859</td>
</tr>
</tbody>
</table>

The corresponding stoichiometric coefficients and the percent increase of total moles from the reactant side to the product side of the non-idealized combustion equation are presented in Table 4.5.

**Table 4.5 – Sensitivity Analysis of Conservation of Moles (5 psig Intake Pressure Series)**

<table>
<thead>
<tr>
<th>(\phi_a)</th>
<th>aCO\textsubscript{2}</th>
<th>bCO</th>
<th>cCH\textsubscript{4}</th>
<th>dH\textsubscript{2}O</th>
<th>(N_{React\ Total})</th>
<th>(N_{Prod\ Total})</th>
<th>Mole Increase</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.23</td>
<td>0.916350</td>
<td>0.017500</td>
<td>0.066150</td>
<td>1.850200</td>
<td>42.50</td>
<td>42.52</td>
<td>0.04%</td>
</tr>
<tr>
<td>0.32</td>
<td>0.965890</td>
<td>0.012200</td>
<td>0.021910</td>
<td>1.943980</td>
<td>30.83</td>
<td>30.84</td>
<td>0.04%</td>
</tr>
<tr>
<td>0.40</td>
<td>0.942030</td>
<td>0.048020</td>
<td>0.009950</td>
<td>1.932080</td>
<td>24.87</td>
<td>24.91</td>
<td>0.19%</td>
</tr>
<tr>
<td>0.45</td>
<td>0.903580</td>
<td>0.089900</td>
<td>0.006520</td>
<td>1.897060</td>
<td>22.21</td>
<td>22.30</td>
<td>0.40%</td>
</tr>
</tbody>
</table>

Table 4.5 shows that for the experimental results the maximum calculated error in assuming conservation of moles is 0.40%, which is not significant. Therefore, the assumption of conservation of moles can be applied with very little error. This assumption was used for the calculation of experimental results.
4.4.4.6 Combustion Efficiency

Combustion efficiency was calculated using the following equation from Reference [21, p. 154].

\[
1 - \eta_c = \frac{\sum x_i Q_{LHV_i}}{[m_{fuel}/(m_{air}+m_{fuel})]Q_{LHV_fuel}} \quad \text{(Eq. 4.13)}
\]

Where:

- \(\eta_c\) is the combustion efficiency
- \(x_i\) are mass fractions of CO, H\(_2\) and unburnt hydrocarbons
- \(Q_{LHV_i}\) are the lower heating values of CO, H\(_2\) and unburnt hydrocarbons in kJ/g
- \(Q_{LHV_fuel}\) is the fuel lower heating value in kJ/g
- \(m_{fuel}\) is the mass flow rate of fuel in g/s
- \(m_{air}\) is the mass flow rate of air in g/s

The above equation requires knowledge of the mass fractions of unburnt hydrocarbons, CO and H\(_2\). Mass fractions were calculated from mole fractions, using the following simplifying assumptions:

- Total moles of reactants is equal to the total moles of products (validated in Section 4.5.1.5)
- CH\(_4\) mole fraction was used in place of unburnt hydrocarbon mole fraction. This assumption is deemed valid as CH\(_4\) was found to be by far the most dominant unburnt hydrocarbon type (determined from FTIR analysis)
- Hydrogen mole fraction was calculated from the following equation as suggested in Reference [21, p. 150]:

\[
\chi_{H2} = \frac{\chi_{CO}\chi_{H2O}}{3.8\chi_{CO2}} \quad \text{(Eq. 4.14)}
\]
4.4.5 Emissions Data Analysis

Each of the emissions instruments used in this study logged the emissions results into a separate file. To combine all the emissions data into one file, a data compiling MATLAB routine was developed. The routine extracted and combined the emissions data from each instrument output, aligned the time axes of the different data types, and rewrote the processed results into a MS Excel format. Further processing was done in MS Excel.

4.4.5.1 FTIR Gaseous Emissions

Gaseous emissions were plotted vs. test time to examine how different specie concentrations change over the duration of the test. Generally, emissions levels showed to be at their steadiest in the last 30 seconds of each test. Emissions concentrations were averaged over this timeframe to estimate steady state values. NOx concentration was calculated as a summation of the NO and NO₂ concentrations. NOx concentration was also converted to dry basis to be consistent with other studies. All other gaseous emissions were reported on wet basis.

Gaseous Emission on Mass Basis

Emissions levels on a mass basis, in g/bhp-hr, were calculated from each emission specie’s measured exhaust mole fractions. The simplifying assumption of conservation of moles (validated in Section 4.5.1.5) was used to determine the exhaust total molar flow rate from the measured intake air and injected fuel mass flow rates. Specie mole fractions were multiplied by the total exhaust molar flow rate and by each specie’s molecular weight. The resulting emissions mass flow rate was divided by the engine’s power output (determined from the calculated indicated torque and rmp), and by a mechanical efficiency of 80% to convert the results to brake power basis.

4.4.5.2 EEPS Particulate Emissions

EEPS particulate emissions had to be processed to compensate for known soot morphology related deviations in concentration and particle sizing. The corrected data also had to be scaled up by the dilution factor to determine raw exhaust concentrations.
EEPS Soot Morphology Related Corrections

An EEPS specific correction protocol developed by Zimmerman et al [26] was applied to the data reported by the EEPS analyzer. The corrections included a concentration correction for the 8.06 to 93.1 nm bin sizes, adjustment of the bin size diameters for the original 80.6 to 523.3 nm bin sizes, and an overall concentration correction by a factor of 1/1.47 (corresponding to a correction previously used for EEPS diesel particulate emissions measurements). Further explanation of the protocol is provided in Reference [26].

Dilution Correction

The EEPS dilution ratio is set on the TSI thermodiluter and thermal conditioner modules. For all experiments the dilution ratio was set to a nominal value of 100. Previous use of the EEPS system, however, showed that the actual dilution ratio is often different from the set value and varies over the test duration. To keep track of the actual dilution ratio, CO₂ concentration of the raw exhaust, the dilution air, and the diluted exhaust were tracked. Actual dilution ratio was then calculated with a formula derived from a mass balance of CO₂:

\[ D_r = \frac{(CO_2)_e - (CO_2)_d}{(CO_2)_p - (CO_2)_d} \]

(Eq. 4.15)

Where:

\( D_r \) is the calculated dilution ratio

\((CO_2)_e\) is the CO₂ concentration of the undiluted exhaust (measured by the FTIR)

\((CO_2)_p\) is the CO₂ concentration of the diluted exhaust (measured by the LICOR analyzer)

\((CO_2)_d\) is the CO₂ concentration of dilution air (measured by the LICOR analyzer prior to the commencement of the fired test)

The soot-morphology-corrected particle concentrations were multiplied by the calculated dilution factor to determine particle concentration of the raw exhaust.
PM Emission on Mass Basis

PM emissions levels on a mass basis, in g/bhp-hr, were calculated from the soot morphology and dilution ratio corrected PM count concentrations (#/cm$^3$) for each particle bin size. Particles were assumed to be spherical and to have a mean diameter of the bin size (ie. 21.1nm bin size = 21.1 nm particle diameter). A volume for each particle size was calculated and multiplied by the particle’s count concentration to yield a particle volume fraction. Individual particle volume fractions were summed to calculate the total PM volume fraction. The simplifying assumption of conservation of moles (validated in Section 4.5.1.5) was used to determine the exhaust total molar flow rate from the measured intake air and injected fuel mass flow rates. The exhaust molar flow rate was then converted to a volumetric flow rate at the EEPS analyzer measurement conditions (35°C, 98 kPaa). The exhaust volumetric flow rate was multiplied by the PM volume fraction and by an assumed PM particle density of 1.84 g/cm$^3$ [21]. The resulting PM mass flow rate was divided by the engine’s power output (determined from the calculated indicated torque and rpm), and by a mechanical efficiency of 80% to convert the results to brake power basis.
Chapter 5

5 Experimental Results

Experimental results are organized by test program, as defined in Chapter 4. The following special nomenclature is employed in this chapter to simplify the discussion:

\[ \phi_n \text{ or nominal equivalence ratio} \] is the targeted test equivalence ratio

\[ \phi_a \text{ or actual equivalence ratio} \] or simply equivalence ratio is the actual test equivalence ratio determined through test results analysis

Figures in this chapter show results plotted at the actual equivalence ratios. Since these values vary across test conditions, figure data will often be referred to by the nominal equivalence ratio. \( \phi_n \) of 0.2 will refer to the lowest actual equivalence ratio grouping, \( \phi_n \) of 0.3 to the second lowest, \( \phi_n \) of 0.3 to the second highest, and \( \phi_n \) of 0.5 to the highest of the four. It is left up to the reader to associate the actual equivalence ratios from the figures with the nominal values in the discussion.

Different intake pressure series will often be referred to as P5, P10 and P15 for intake pressures of 5, 10, and 15 psig, respectively.
5.1 Glow Plug Shield and Injector Angle Screening Test Results

The effects of the glow plug shield and injector angles on combustion were investigated by audible detection. Results are presented in Table 5.1:

Table 5.1 – Shield and Injector Angle Screening Results

<table>
<thead>
<tr>
<th>Injection Duration (CAD)</th>
<th>$\phi_a$</th>
<th>Shield Angle 0°</th>
<th>Shield Angle 20°</th>
<th>Shield Angle 40°</th>
<th>Shield Angle 60°</th>
<th>Shield Angle 80°</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>0.09</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
</tr>
<tr>
<td>12</td>
<td>0.22</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
</tr>
<tr>
<td>18</td>
<td>0.29</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
</tr>
<tr>
<td>24</td>
<td>0.37</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
</tr>
<tr>
<td>30</td>
<td>0.43</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
<td>nc</td>
</tr>
<tr>
<td>36</td>
<td>0.50</td>
<td>n/a</td>
<td>n/a</td>
<td>n/a</td>
<td>n/a</td>
<td>n/a</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Shield-Injector Combination</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
<th>12</th>
<th>13</th>
<th>14</th>
<th>15</th>
<th>16</th>
<th>17</th>
<th>18</th>
<th>19</th>
<th>20</th>
</tr>
</thead>
</table>

LEGEND

- nc: no combustion
- a: poor combustion, > 75% misses
- b: poor combustion, 50–75% misses
- c: repeatable combustion, < 25% misses
- d: repeatable combustion, no misses
- n/a: not tested

Results show that no combustion was observed for any injector angle at shield angles of 0° and 20° (combinations 1 through 8). Combinations 9 through 11, and 14 through 20, produced combustion events of variable quality depending on the injection duration. However, these combinations failed to provide repeatable combustion for all injection durations tested. Only combinations 12 and 13 produced repeatable combustion at all the tested injection durations. The recorded data for test combinations 12 and 13 was processed to calculate average ignition delays of 2.4 and 2.1 ms, respectively. Due to a lower calculated ignition delay, combination 13, with a shield angle of 60° and an injector angle of 0°, was chosen for skip-fired ignition delay and fired emissions tests.

Results of this screening test reinforce the importance of the geometric arrangement on the ignition performance as was shown by Chown [23] on the CFR combustion bomb apparatus. In the engine environment, however, the geometry effects seem to have a much greater effect. For example, a similar test series conducted by Chown [13] on the CFR combustion bomb apparatus
showed that different shield.Injector angle combinations resulted in a difference in ignition delay. All the tested combinations, however, were capable of sustaining combustion. Major differences in engine speed, nozzle orifice size, and flow fields make a direct comparison of the CFR combustion bomb apparatus to the DING engine not practical. However, the high-level qualitative observation of vastly different performance is of note. This observation stresses the heightened importance of the glow plug shield and injection geometry on ignition performance in the DING engine environment.

Further geometric effects were not investigated in this study. Priority was given to exploring the effects of intake pressure and equivalence ratio on ignition delay as well as emissions.

5.2 Skip-Fired Ignition Delay Test Results

Effects of intake pressure and equivalence ratio on ignition delay were investigated. Three intake pressures of 5, 10, and 15 psig, and four nominal equivalence ratios of 0.2, 0.3, 0.4, and 0.5 were tested. The test for each of the 12 pressure-equivalence ratio combinations consisted of 100 fired cycles. Three skipped cycles were used between each fired cycle for $\phi_n$ of 0.2, 0.3 and 0.4. Four skipped cycles were used between each fired cycle for $\phi_n$ of 0.5. Each test condition was tested three times, each on a different day. Table 5.2 summarizes the engine test conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Parameter Set Point</th>
<th>Actual Test Value Range or Control Tolerance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake Pressure</td>
<td>5, 10, 15 psig</td>
<td>+/- 1%</td>
</tr>
<tr>
<td>Intake Temperature</td>
<td>25°C</td>
<td>25–29°C</td>
</tr>
<tr>
<td>Engine Speed</td>
<td>1000 RPM</td>
<td>+/- 1%</td>
</tr>
<tr>
<td>Head Temperature</td>
<td>100°C</td>
<td>98–109°C</td>
</tr>
<tr>
<td>Coolant Temperature</td>
<td>60°C</td>
<td>60–65°C</td>
</tr>
<tr>
<td>Oil Temperature</td>
<td>44°C</td>
<td>40–45°C</td>
</tr>
<tr>
<td>Start of Injection (SOI)</td>
<td>335 CAD</td>
<td>335 CAD</td>
</tr>
<tr>
<td>Natural Gas Pressure</td>
<td>1800 psig</td>
<td>+/- 0.5%</td>
</tr>
<tr>
<td>Glow Plug Voltage</td>
<td>14 VDC</td>
<td>+/- 0.01 VDC</td>
</tr>
</tbody>
</table>

Due to dynamometer control limitations, rpm control of the engine was poor at the beginning of the test, but steadied out about half way into each test. As the ignition delay calculation code assumes a constant rpm value, deviation of rpm has a direct effect on the calculated ignition delay. For this reason only the last 50 of 100 fired cycles were used in the results analysis. The
deviation in rpm in the second part of the test was manually recorded and is reflected in the table above.

### 5.2.1 Ignition Delay

Combined ignition delay results for all three test days are presented in Figure 5.1. Each point in Figure 5.1 represents an average of 150 measurements: three test days with 50 measurements per test. Data variability, shown as +/- two standard deviations (2σ), reflects the variation of ignition delay within each test, as well as day-to-day variability across test days. This method of showing data variability is employed for all figures in Section 5.2.

![Figure 5.1 – Ignition Delay vs. Equivalence Ratio (all tests)](image)

The results suggest several trends. The first key observation is that all test pressures and equivalence ratios result in average ignition delays below the critical value of 2 ms. Data variability, however, shows that only intake pressures of 5 and 10 psig sustain reliable ignition below 2 ms ignition delay. Although the mean ignition delay at 15 psig intake pressure is below 2 ms, the data for this pressure has the most variability. This indicates likely occurrence of occasional ignition cycles with ignition delay values above 2ms. The second observation is that equivalence ratio appears to have little to no effect on ignition delay for all tested intake pressures. This trend is expected since different equivalence ratios affect the total injection duration, but should have no effect on the short initial period of injection that happens up to the end of the physical delay period. The third observation is that the average ignition delay seems to
rise with pressure. The large variability in the data, however, warrants a closer examination of this trend.

Review of results for each test day separates the effects of the in-test and the day-to-day variability. Individual test results reinforce the trend that only intake pressures of 5 and 10 psig are likely to produce repeatable combustion at delay values below 2 ms. The observation that ignition delay may be increasing with pressure is evident in results for test day 3 at nominal equivalence ratios of 0.4 and 0.5 (refer to Figure 5.2).

![Figure 5.2 – Ignition Delay vs. Equivalence Ratio (test day 3)](image)

As mentioned in Chapter 2, ambient pressure was found to decrease ignition delay in combustion bomb experiments [7]. The trend observed in Figure 5.2 seems to violate the abovementioned effect. Reconciliation of the trend requires consideration of jet mixing dynamics and the geometric effects between the jet and the exposed glow plug surface.

Considering jet development, Equation 2.1 indicates that jet penetration is inversely proportional to the fourth root of the ambient density. Since engine charge density scales directly with the intake pressure, effects of density are essentially the same as effects of intake pressure. Taking jet penetration distance at 5 psig intake pressure as a reference value, the penetration distance for 10 psig and 15 psig intake pressures would be approximately 95% and 90% of the reference value, respectively. The lower penetration distance essentially means slower jet development speed. Slower jet development speed results in a longer time required to deliver the flammable
jet layer to the glow plug surface. This would increase the physical component of the ignition delay. Another contributing effect is engine swirl. Higher intake pressure increases intake air density and consequently the engine swirl momentum. The increased swirl momentum may be responsible for shifting the fuel jet away from optimal contact with the glow plug surface, resulting in a longer ignition delay.

The jet’s air entrainment rate was also considered in the analysis. Review of jet concentration decay correlations revealed that for supercritical jets, higher ambient pressure does not have a net influence on the jet’s concentration decay. As mentioned in Chapter 2, in supercritical jets, the expansion of the jet to ambient conditions takes place downstream of the injector orifice. In this case, Eq. 2.2 for mean concentration decay has to be corrected for the pseudo-diameter of the supercritical jet. The pseudo-diameter is defined in Eq. 2.3. The corrected concentration decay correlation ends up having no dependency on ambient pressure since the pressure ratio in the pseudo-diameter correction, once converted to a density ratio via the ideal gas law, cancels out with the density ratio term in the concentration decay equation.

5.2.2 Combustion Mode Analysis

Since the equivalence ratio is controlled solely by the injection duration, comparison of the total injection time to the ignition delay time is useful in establishing the observed combustion mode. An injection duration less than or of the same length as the ignition delay will proceed in a more premixed fashion, whereas a longer injection duration will result in a higher proportion of mixing-controlled combustion. Table 5.3 shows a comparison of the of average fuel injection duration to the average ignition delay (averaged across three test days). Percent of injection duration that occurs within the ignition delay period is calculated as a rough indicator of the premixed combustion proportion.
Table 5.3 – Ignition Delay vs. Injection Duration Comparison

<table>
<thead>
<tr>
<th>Intake Pressure</th>
<th>Nominal Equivalence Ratio</th>
<th>Average Actual Equivalence Ratio</th>
<th>Average Ignition Delay (ms)</th>
<th>Average Injection Duration (ms)</th>
<th>Percent of Injection Prior to Ignition</th>
</tr>
</thead>
<tbody>
<tr>
<td>5 psig</td>
<td>0.2</td>
<td>0.22</td>
<td>1.72</td>
<td>1.34</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
<td>0.34</td>
<td>1.72</td>
<td>1.98</td>
<td>87%</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>0.45</td>
<td>1.67</td>
<td>3.13</td>
<td>53%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.56</td>
<td>1.67</td>
<td>4.44</td>
<td>38%</td>
</tr>
<tr>
<td>10 psig</td>
<td>0.2</td>
<td>0.22</td>
<td>1.79</td>
<td>1.59</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
<td>0.33</td>
<td>1.77</td>
<td>2.82</td>
<td>63%</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>0.44</td>
<td>1.78</td>
<td>4.48</td>
<td>40%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.55</td>
<td>1.75</td>
<td>6.11</td>
<td>29%</td>
</tr>
<tr>
<td>15 psig</td>
<td>0.2</td>
<td>0.22</td>
<td>1.94</td>
<td>1.96</td>
<td>99%</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
<td>0.33</td>
<td>1.99</td>
<td>3.78</td>
<td>53%</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>0.45</td>
<td>1.96</td>
<td>5.83</td>
<td>34%</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>0.56</td>
<td>1.93</td>
<td>7.67</td>
<td>25%</td>
</tr>
</tbody>
</table>

Stratified-Premixed Combustion Mode

Table 5.3 shows that at nominal equivalence ratios 0.2 and 0.3, ignition happens after or just shortly prior to the end of the injection duration. Combustion likely occurs in a stratified-preamixed mode. This is corroborated by analysis of the ROHR curves. Figures 5.4 and 5.5 show the heat release curves for the lower equivalence ratios of 0.22 and 0.33, respectively, for an intake pressure of 10 psig, test day 3.

Both figures show a single peak typical of premixed combustion. The peak for $\phi_a=0.33$ is higher, corresponding to higher amount of injected fuel. The heat release duration is short, proceeding to approximately 40° ATDC. This combustion mode, however, cannot be called truly premixed. Mixing occurs locally, and since the mixing time is still relatively short, stratification of the...
mixed region is expected. For the purposes of later discussion, combustion at nominal equivalence ratios 0.2 and 0.3 will be termed *stratified-premixed* combustion mode.

**Free-Mixing Combustion Mode**

At higher equivalence ratios, as the injection duration grows, mixing-controlled combustion starts to develop. This is evident from ROHR curves for equivalence ratios of 0.45 and 0.56 (Fig 5.6 and 5.7). The curves show an initial heat release peak of the premixed phase which is followed by a characteristic second hump of mixing-controlled combustion. The total heat release occurs over a longer duration, lasting approximately up to 80° ATDC.

![Figure 5.6 – ROHR for \( \phi_a = 0.45 \), \( P_{\text{intake}} = 10 \) psig](image)

![Figure 5.7 – ROHR for \( \phi_a = 0.56 \), \( P_{\text{intake}} = 10 \) psig](image)

It is important to note that the mixing-controlled phase of the heat release occurs after the injector is already closed. In typical diesel combustion, the mixing-controlled heat release occurs with the fuel jet still feeding the flame. In the DING engine, the mixing-controlled phase is occurring at a time when there is no longer a jet. The mixing-controlled burn is likely occurring in the gas plume that is formed from the jet’s tail end, after the jet ceases. The mixing and air entrainment in the plume is controlled by the residual momentum of the jet and the engine swirl field. The swirl field likely has a strong effect. Since the plume is formed from the fuel jet, the jet’s momentum would push the plume outward from the center of the combustion chamber towards the cylinder wall. Further from the center, swirl velocities are higher so a large influence of swirl is expected. Complex flow and mixing patterns are likely to develop. For the purposes of later discussion, combustion at nominal equivalence ratios 0.3 and 0.4 will be termed *free-mixing* combustion mode, which consist of the initial *premixed phase* and a later *mixing-controlled*
phase. The prefix free will signify that the mixing-controlled phase occurs in an unattached free-mixing plume, rather than a jet configuration.

**Effect of Pressure**

From Table 5.3 it should also be noted that for a given equivalence ratio, the percent of premixed combustion declines as the intake pressure increases. At higher pressures a longer injection duration is required to reach a given equivalence ratio. This extends the injection duration requirement and favors a higher proportion of mixing-controlled combustion. Although the average ignition delay also increases with pressure, the injection duration effect is dominant. This observation is supported by ROHR curves in Figures 5.8 and 5.9 for intake pressures of 5 and 15 psig, both at an equivalence ratio of approximately 0.56.

![Figure 5.8](image1.png)  
*Figure 5.8 – ROHR for $\phi_{av}=0.57$, $P_{intake}=5$ psig*

![Figure 5.9](image2.png)  
*Figure 5.9 – ROHR for $\phi_{av}=0.56$, $P_{intake}=15$ psig*

The shape of the curves clearly shows that the proportion of curve area under the second hump to the area under the entire curve grows with increased intake pressure.

**5.2.3 Mean In-Cylinder Temperature Trends**

Understanding the mean in-cylinder temperature is important for later discussion on emissions formation. Mean in-cylinder temperatures were calculated using the ideal gas law. The first notable temperature effect is that temperature at peak compression (at TDC) is independent from the intake pressure. This may seem counterintuitive, but is a result of the air intake system’s design. Intake air at three different intake pressures is supplied to the engine at the same ambient temperature. Peak in-cylinder pressure is of course higher for higher intake pressure, but the temperature is essentially the same. Compression work required to compress higher intake
pressure air is higher, but since more mass is inducted, the increased heat capacity of the air charge negates the expected temperature change.

Effect of equivalence ratio on in-cylinder temperature is presented in Figure 5.10 for an intake pressure of 10 psig.

![Figure 5.10 – Mean In-Cylinder Temperature for $P_{\text{intake}} = 10$ psig (test day 2)](image)

Peak motored temperature is approximately 800 K and occurs just prior to TDC. Overall, temperature levels increase with equivalence ratio due to higher heat release. Peak temperature for all series occurs around the same position, approximately at 475 CAD. Peak temperature rises from $\phi_n$ of 0.2 to 0.3. This is expected since the amount of fuel available to burn in the stratified-premixed mode is larger at $\phi_n$ of 0.3. For $\phi_n$ of 0.4 and 0.5, the peak temperature is essentially the same and is controlled by the premixed phase of the free-mixing combustion mode. For $\phi_n$ of 0.5, temperature near peak levels is maintained for a longer duration. This is a result of a longer mixing-controlled combustion phase.
Effect of intake pressure on in-cylinder temperature is presented in Figure 5.11 for the stratified-premixed combustion mode at $\phi_n$ of 0.2.

As previously discussed, for a given equivalence ratio, injection duration grows with higher intake pressure. The resulting temperature effect is that peak in-cylinder temperatures are reached later in the engine cycle. This is evident from Figure 5.11 showing the temperature curves shifting to the right with increasing pressure. It should be noted that the peak temperatures also drop slightly with increasing pressure. This is also caused by the rightward shift since a delayed peak temperature will be subjected to a drop associated with the cylinder expansion.
The temperature delay effect for the free-mixing combustion mode is shown for $\phi_n$ of 0.5 in Figure 5.12.

Figure 5.12 – Mean In-Cylinder Temperature for $\phi_n = 0.5$ (test day 3)

The peak temperature delay effect is more pronounced in the free-mixing combustion mode. The largest impact is observed for intake pressure of 15 psig, where the peak temperature is delayed until approximately 60 CAD ATDC.
5.2.4 Combustion Performance

Gross indicated mean effective pressure (IMEP<sub>gross</sub>) was calculated for each of the cycles using the in-cylinder pressure trace. Figure 5.13 shows the average IMEP<sub>gross</sub> for all the test conditions across all three test days.

![IMEP<sub>gross</sub> vs. φ<sub>a</sub> (all tests)](image)

*Figure 5.13 – IMEP<sub>gross</sub> for All Test Days*

Results show that, in general, variability in IMEP<sub>gross</sub> is within +/- 10% for nominal equivalence ratios of 0.3 through 0.5. IMEP<sub>gross</sub> values from laboratory tests of a Cummins ISX series heavy-duty six-cylinder compression-ignition natural gas engine (using pilot-ignited direct injection natural gas system by Westport Inc.) are also shown in the figure above (Jones et al. [11]). IMEP<sub>gross</sub> values for the P15 series are roughly 12% lower than values reported for the Cummins ISX engine. This indicates that the IMEP<sub>gross</sub> values in the DING experiments are reasonably close to those determined in laboratory testing of a commercial pilot-ignited natural gas powered direct injection engine.

5.3 Fired Emissions Test Results

Fired emissions tests were conducted for the same nominal equivalence ratios and intake pressures as for the skip-fired test series. Each fired emissions test was 6 minutes in duration. Gaseous emissions were measured with the FTIR analyzer and particulate emissions with the EEPS analyzer. It should be noted that in comparison to the corresponding skip-fired tests, some engine conditions changed significantly over the fired test duration. This is a result of continuous
combustion heat input of the fired tests. Key engine condition changes from 1 min to 6 min into the fired tests are shown in Table 5.4 (averaged across three test days). Test conditions that are not listed in Table 5.4 had the same control tolerance as for the skip-fired tests.

Table 5.4 – Variable Engine Conditions of Fired Emissions Tests

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Nominal Equivalence Ratio</th>
<th>$P_{\text{intake}} = 5$ psig</th>
<th>$P_{\text{intake}} = 10$ psig</th>
<th>$P_{\text{intake}} = 15$ psig</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>1m</td>
<td>6m</td>
<td>1m</td>
</tr>
<tr>
<td><strong>Air Mass Flow (g/s)</strong></td>
<td>0.2</td>
<td>7.1</td>
<td>6.9</td>
<td>9.3</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
<td>7.1</td>
<td>6.7</td>
<td>9.3</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>6.9</td>
<td>6.4</td>
<td>9.3</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>6.9</td>
<td>6.3</td>
<td>9.0</td>
</tr>
<tr>
<td><strong>Exhaust Temp (°C)</strong></td>
<td>0.2</td>
<td>169</td>
<td>198</td>
<td>167</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
<td>232</td>
<td>279</td>
<td>242</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>311</td>
<td>361</td>
<td>322</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>378</td>
<td>427</td>
<td>398</td>
</tr>
<tr>
<td><strong>Engine Speed (rps)</strong></td>
<td>0.2</td>
<td>1023</td>
<td>1035</td>
<td>1056</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
<td>1048</td>
<td>1044</td>
<td>1085</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>1055</td>
<td>1047</td>
<td>1098</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>1064</td>
<td>1052</td>
<td>1087</td>
</tr>
<tr>
<td><strong>Head Temp (°C)</strong></td>
<td>0.2</td>
<td>131</td>
<td>141</td>
<td>132</td>
</tr>
<tr>
<td></td>
<td>0.3</td>
<td>146</td>
<td>164</td>
<td>148</td>
</tr>
<tr>
<td></td>
<td>0.4</td>
<td>159</td>
<td>185</td>
<td>161</td>
</tr>
<tr>
<td></td>
<td>0.5</td>
<td>172</td>
<td>199</td>
<td>180</td>
</tr>
</tbody>
</table>

The most notable change is the cylinder head temperature. In skip-fired tests, the cylinder head was cooled with the engine air charge of the skipped cycles. In the fired emissions tests, cylinder head cooling is provided by the head cooling fan. The fan was unable to cool the head below approximately 220 °C. However, this temperature level is typical of air-cooled engines.

### 5.3.1 Indicated Specific Fuel Consumption (ISFC)

Indicated specific fuel consumption (ISFC) was calculated for all test conditions and is presented in Figure 5.14. It should be noted that all figures in Section 5.3 show each data point as an average of three values. Each of the three values corresponds to the 6th minute parameter average for each of the three test days. The variability shown reflects only the day-to-day variability between the three test days at +/- two standard deviations (2σ).
ISFC appears to be relatively constant across all tested equivalence ratios and pressures. ISFC values remain in the range of approximately 130 to 190 g/kW-h, with a mean of approx. 155 g/kW-h. An estimated typical best ISFC for diesel engines is shown in the figure at 160 g/kW-h. This value was determined by applying a mechanical efficiency of 80% to a typical best value of brake specific fuel consumption (BSFC) for a compression ignition (diesel) engine [21, p. 52]. The ISFC of the DING engine closely matches the estimated best diesel value.

Figure 5.14 – ISFC for All Test Days
5.3.2 Combustion Efficiency

Combustion efficiency was calculated for all test conditions and is presented in Figure 5.15 for all test days.

![Combustion Efficiency vs. $\phi_n$ (all tests)](image)

*Figure 5.15 – Combustion Efficiency for All Test Days*

Results show that combustion efficiency for all intake pressures is above 92% at $\phi_n$ of 0.2, above 96% at $\phi_n$ of 0.3, and above 98% for the P10 and P15 series at $\phi_n$ of 0.4 and 0.5. These values are consistent with typical values for diesel engines [21, p. 82]. Efficiency appears to be greater at higher intake pressures, indicating more complete fuel conversion. The drop in combustion efficiency for the P5 series over $\phi_n$ of 0.4 to 0.5 is attributed to increased CO emissions. As presented later in Section 5.3.5, CO emissions are significantly higher at 5 psig than for other intake pressures over $\phi_n$ of 0.4 to 0.5. High CO levels indicate incomplete fuel conversion, which results in decreased combustion efficiency. The effect of CO is demonstrated by the “P5–Low CO” curve. This curve represents the calculated combustion efficiency for the 5 psig intake pressure if CO was maintained at levels of the 10 psig intake pressure series over $\phi_n$ of 0.4 to 0.5. It should be noted that the “P5–Low CO” curve represents a hypothetical situation and is presented to demonstrate the CO emissions effect on combustion efficiency.

The following sub-sections discuss the emissions results. For each emissions type the time dependency is presented prior to discussing overall effect of pressure and equivalence ratio.
5.3.3 NOx Emissions

NOx emissions grow over the test duration, but start to flatten out at the sixth minute. An example of this behavior is shown in Figure 5.16 for test day 3 and an intake pressure of 10 psig. This trend is present for all test conditions across all test days.

![Figure 5.16 – NOx Emissions vs. Test Time (test day 3, $P_{\text{intake}} = 10$ psig)](image)

Since the test duration was limited by the durability of the polyamide piston rings, a 30 second average, from 5 min 30 s to 6 min, is reported as the best estimate of steady state NOx emissions. The combined results for all three test days are shown in Figure 5.17.

![Figure 5.17 – NOx Emissions vs Equivalence Ratio (all tests)](image)
The first notable trend is that NOx levels decrease with higher intake pressure. This trend is surprising. Absolute oxygen concentration (moles/volume) is higher at higher intake pressures. With higher oxygen concentration, NOx is expected to form at higher rates [21, p. 575]. However, the opposite is observed for all equivalence ratios. One possible explanation is the effect intake pressure has on the in-cylinder temperature. As shown in Figures 5.11 and 5.12, the peak in-cylinder temperature conditions generally occur later with increasing intake pressure. This is caused by longer injection durations required for a given equivalence ratio at higher intake pressure. NOx formation rates are highest at peak temperature, but the peak temperature delay limits the time available for NOx formation. For example, for $\phi_n=0.5$, at 5 psig intake pressure, peak temperature occurs approximately 115 CAD prior to exhaust valve opening. At 15 psig intake pressure, peak temperature occurs at approximately 70 CAD prior to exhaust valve opening. At the higher intake pressure, NOx has approximately 40% less time to form before the cylinder mixture is exhausted. Limited NOx formation time is just one possible contributor to the observed effect of intake pressure on NOx formation.

The second notable trend is that NOx emissions rise from $\phi_n$ of 0.2 to 0.3, and fall from $\phi_n$ of 0.3 to 0.5. Generally, NOx formation is expected to rise continuously with increasing equivalence ratio in lean operation. This is explained with higher temperatures resulting from higher heat release at higher equivalence ratios. In fact this is observed in the stratified-premixed combustion mode ($\phi_n$ of 0.2 to 0.3). In the free-mixing combustion mode ($\phi_n$ of 0.4 to 0.5), NOx formation declines, which is unanticipated. One possible explanation of this trend is NO freezing behavior, which has a significant effect on NOx formation in diesel engines [21, p. 587]. Fuel/air mixture which burns early in the combustion process, results in the highest rates of NOx formation since this mixture sees the highest flame and in-cylinder temperatures. Later in the combustion process, the in-cylinder temperature starts to fall due to expansion. At this point, NOx equilibrium levels fall below the early-formed NOx level. However, the high-temperature early-formed NOx cools rapidly by mixing with cool excess air and due to cylinder expansion. As a result, the early-formed NOx chemistry freezes. Early-formed NOx is unable to reach lower equilibrium levels. This situation is especially fitting to the stratified-premixed combustion mode of the DING engine. Combustion is short lived and is followed by a steep temperature fall as seen in Figure 5.10. In the free-mixing combustion mode ($\phi_n$ of 0.4 to 0.5), the temperature fall is more gradual due to the long mixing-controlled burn phase. With higher equivalence ratio this
effect is more prevalent. The gradual temperature fall may enable the decomposition of early-formed NOx to lower equilibrium levels by mitigating the NOx freeze effect.

Another contributing factor to the NOx level decrease may be reduced oxygen availability. Increase in equivalence ratio results in less oxygen available for oxidation of N\textsubscript{2}. This happens since a higher proportion of oxygen is used up for oxidizing the fuel. This is true for both DING combustion modes, but is more prevalent in the free-mixing mode since the overall equivalence ratios are higher. A second possibility is that oxygen availability is influenced by mixing. This second factor is specific to the free-mixing combustion mode only, since a large portion of the combustion process is mixing controlled. With increased equivalence ratio, the gas plume of the mixing-controlled phase gets larger and likely becomes harder to mix with the excess air. Effective mixing is important for NOx formation since it delivers excess oxygen to the region of highest temperature and high N\textsubscript{2} concentration. Both, the lower oxygen availability and impeded mixing can contribute to lower NOx levels.

**Comparison to Regulatory Levels**

In order to compare DING emissions to regulatory standards, NOx mass emissions were calculated. Results are reported on a basis of g/bhp-hr, and are presented in Figure 5.18 along with the US EPA emissions standards. It should be noted that the EPA standards are based on composite drive cycle tests for non-steady-state operation. A direct comparison to steady-state emissions of the DING is not ideal, but is useful for a high-level emissions levels assessment.
Results show that DING NOx levels exceed current standards and are more in line with 1994–1997 US EPA standards. It should be noted, however, that the DING engine was operated without exhaust gas recirculation (EGR) or exhaust aftertreatment technologies. Modern diesel engines rely on EGR and exhaust aftertreatment for compliance with the 2010+ US EPA regulatory standards [28]. Exploring use of EGR and other operating and aftertreatment strategies is left for future work.

### 5.3.4 Unburnt Hydrocarbon Emissions

The primary contributor to unburnt hydrocarbon emissions is unburnt methane. Methane emissions fall over the course of the test, levelling off at the end of the test. This trend is present across all test conditions and test days. As an example, Figure 5.19 shows methane concentration vs. time for test day 3 and intake pressure of 10 psig.
A 30 second average, from 5 min 30 s to 6 min, is shown for all tests in Figure 5.20 as a best estimate of steady state conditions.

Results show that higher variability in methane concentration, observed at $\phi_n$ of 0.2, coincides with the high variability of ignition delay values at the same nominal equivalence ratio (as shown in Figure 5.1). Since mechanisms of crevice entrapment and quenching are less prevalent in direct injection engines, this observation indicates that the natural gas jet is likely overmixing within the ignition delay timeframe and forms lean pockets of fuel/air mixture that are below the
lower flammability limit of methane. The overmixed methane survives the combustion event. This effect is especially prevalent for 5 psig intake pressure with an injection duration of approx. 1.34 ms and an average ignition delay of approx. 1.74 ms (see Table 5.3). In this case, by the end of injection, the fuel is no longer in a jet configuration and is allowed to mix freely for over 23% of the ignition delay period. The over-leaning hypothesis also explains lower combustion efficiency at the nominal equivalence ratio of 0.2. The overall trend in Figure 5.20 is that methane concentration falls with increased equivalence ratio and higher intake pressure. This dependency can be explained with the fact that higher equivalence ratios and intake pressures result in higher temperatures in the late stages of the expansion stroke, as demonstrated in Section 5.2.3. Higher late-stage temperatures result in greater rates of thermal oxidation of methane that survives combustion.

5.3.5 Carbon Monoxide Emissions

Carbon monoxide concentration was variable over the span of each test and showed either a slow decrease with time or an approximately even average concentration. The variability of the concentration was quite high for all test conditions. Figure 5.21 shows CO concentration vs. test time for test day 3 and an intake pressure of 10 psig.

![CO Emissions vs. Test Time (test day 3, P_{intake} = 10 psig)](image)

*Figure 5.21 – CO Emissions vs. Test Time (test day 3, P_{intake} = 10 psig)*
A 30 second average, from 5 min 30 s to 6 min, is shown for all tests in Figure 5.22.

![Figure 5.22 – CO Emissions vs Equivalence Ratio (all tests)](image)

With respect to equivalence ratio, all three intake pressure series show a low CO level for $\phi_n$ of 0.2 to 0.3. Over this range, the concentration levels are practically indistinguishable between the different pressure series. From $\phi_n$ 0.3 to 0.5, CO concentrations rise. The rise for the P5 series is much steeper than for the P10 and P15 series. High CO levels of the 5 psig intake pressure affect combustion efficiency, as discussed in Section 5.3.2. Higher pressure at for $\phi_n$ 0.4 and 0.5 appears to significantly lower CO emissions levels. Discussion of causes for the observed CO behavior is deferred to Section 5.3.6 where effects common to CO and PM formation are discussed.

Comparison to Regulatory Levels

Calculated CO mass emissions are presented in Figure 5.23 along with the US EPA emissions standards. As mentioned prior, the EPA standards are based on composite drive cycle tests for non-steady-state operation and are presented for high-level comparison only.
Results show that at intake pressures of 10 and 15 psig the DING engine is well within the US EPA emissions standards for CO.

5.3.6 Total Particulate Matter (PM) Emissions

Total particulate matter (PM) emissions, measured with the EEPS apparatus, do not show clear time dependence. Some tests indicate a slow decrease in PM concentration, while others show a roughly stable average with high variability. EEPS particulate concentration vs. test time is shown in Figure 5.24 for test 3, intake pressure of 10 psig.
The shape of the concentration profile over time shows excellent correlation to the CO emissions measurements. This is evident across all test conditions. Comparing Figure 5.24 to Figure 5.21 demonstrates the like shapes of the CO and PM concentration time plots. A one-minute average of total PM concentration, from 5 to 6 min, is shown for all tests in Figure 5.25.

Results for total PM concentration show similar trends to CO results. Over $\phi_n$ of 0.2 and 0.3, concentration is even and similar for all pressures. Particulate concentration starts to rise for all pressures from $\phi_n$ of 0.3 to 0.5. Higher intake pressure significantly lowers emissions levels for
\( \phi_n \) of 0.4 and 0.5. These results are consistent with the trends observed for CO emissions. Plotting CO concentration vs. particulate emissions shows excellent correlation over nominal equivalence ratios of 0.4 and 0.5. This is shown in Figure 5.26.

![Figure 5.26 – CO vs. Total PM Emissions (\( \phi_n \) 0.3 to 0.5, all pressures, all test days)](image)

The strong correlation at the higher equivalence ratios suggests that both CO and PM emissions are controlled by the same processes. Generally, the results suggest that at the two lower equivalence ratios, CO and PM concentrations are low. At the two higher equivalence ratios, CO and PM levels rise. This observation is consistent with the DING combustion modes proposed in Section 5.2.2. Stratified-premixed combustion provides ample time for the fuel to mix with oxygen, resulting in low CO and PM levels. In the free-mixing combustion mode, a large proportion of the combustion is mixing controlled. Incomplete mixing in the mixing-controlled phase results in high-temperature fuel-rich gas pockets, where CO and PM form from fuel oxidizing in a hot oxygen deficient environment. Since the proportion of the mixing-controlled phase grows with equivalence ratio, CO and PM levels increase accordingly.

The significant effect of pressure on lower CO and PM formation in free-mixing combustion (\( \phi_n \) 0.4 and 0.5) can be attributed to the following effects of higher intake pressure:

**Increased Swirl Momentum:** Higher intake pressure increases the air density, resulting in higher momentum of the engine swirl field. Increased swirl momentum increases air entrainment [27] in the free flowing fuel/air mixing-controlled plume. Higher air entrainment results in a
lower proportion of rich fuel/air pockets where CO and PM form. Higher air entrainment also increases oxidation of the CO and PM that had already formed. Lastly, increased mixing can also reduce localized combustion temperatures, which would limit the rate of CO and PM formation.

**Higher Oxygen Concentration:** Higher intake pressure increases absolute oxygen concentration (moles/volume). Higher oxygen concentration increases rates of CO and PM oxidation, leading to lower overall surviving concentration levels of CO and PM emissions.

Of the two effects mentioned, the effect of increased swirl momentum is expected to be dominant.

**Comparison to Regulatory Levels**

Calculated PM mass emissions are presented in Figure 5.27 along with the US EPA transient drive cycle emissions standards.

![Estimated PM Mass Emissions](image)

*Figure 5.27 – PM Emissions vs. US EPA Standards (all tests)*

Results show that intake pressures of 10 and 15 psig result in DING engine operation within the current US EPA transient drive cycle regulations for PM emissions.
5.3.7 PM Size Distribution

The size distribution of the particulate emissions was averaged for the last minute of each test across all test days. Figures 5.28 through 5.30 show the particulate size distribution in order of increasing intake pressure.

![PM Size Distribution (all tests, \( P_{\text{intake}} = 5 \text{ psig} \))](image)

To aid in results analysis, Table 5.5 shows the ranges of particle sizes classified by their formation processes. These values are taken from a review of soot formation in compression ignition engines by Tree and Svensson [18].

<table>
<thead>
<tr>
<th>Formation Process</th>
<th>Particle Type</th>
<th>Particle Size Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pyrolysis+Nucleation = (P+N)</td>
<td>Nuclei</td>
<td>Down to 1.5–2 nm</td>
</tr>
<tr>
<td>P+N+Coalescence = (P+N+C)</td>
<td>Primary</td>
<td>30–70 nm</td>
</tr>
<tr>
<td>P+N+C+Agglomeration = (P+N+C+A)</td>
<td>Agglomerates</td>
<td>&gt;70nm</td>
</tr>
</tbody>
</table>

DING engine test results show that the vast majority of the particles are primary particles formed by surface growth of the initial particle nuclei. Modal diameters range from 21.1 to 39.2 nm, depending of test conditions. In comparison to diesel and GDI engines, the DING engine produces smaller particles. Further discussion and comparison of DING particles to other engine types is provided in Appendix C1. A rightward shift in the size distribution occurs with increased equivalence ratio. This shift results in larger particles and in modest levels of agglomerate particle formation. The shift is most significant for an intake pressure of 5 psig, followed by the 10 and 15 psig intake pressures.
The observed equivalence-ratio-driven shift can be explained by considering the DING engine combustion modes. In the stratified-premixed combustion mode, overall particle levels are low due to the high degree of mixedness of the fuel and air. Some particles, however, form in the scarce rich fuel/air pockets. Since these rich fuel/air pockets are expected to be small and the combustion event is relatively short-lived, particles do not have sufficient time for extensive surface growth or coalescence. Lack of growth results in small particles, which are evident in the $\phi_n=0.2$ and $0.3$ data series. In the free-mixing mode combustion of $\phi_n=0.4$ and $0.5$, the long mixing-controlled phase of the combustion process presents ample opportunity for particle formation and surface growth. This results in overall larger particles.
The effect of higher intake pressure seems to counteract the equivalence-ratio-driven shift in PM size. As established in Section 5.3.6, higher intake pressure increases swirl momentum and leads to lower rates of soot formation and higher rates of soot oxidation. Higher pressure also leads to an increase in oxygen absolute concentration. These effects result not only in lower particle concentrations, but also smaller particles. Lower formation rates limit particle growth. Higher oxidation rates can oxidize the chemical species responsible for particle surface growth through absorption. Higher oxidation can also partially oxidize the formed particles to a smaller size. In general, increased pressure seems to increase mixing levels to a point where the free-mixing combustion mode starts to resemble the stratified-premixed combustion mode. This is supported by the fact that for $\phi_n$ 0.4 and 0.5, higher pressure progressively forces a PM size distribution similar to that of the stratified-premixed series for $\phi_n$ 0.2 and 0.3. This is evident in comparison of PM size distribution plots.
Chapter 6

6 Conclusions and Recommendations

6.1 Conclusions

Natural gas direct injection and glow plug ignition assist technologies were implemented in a single-cylinder optically-accessible engine. Extensive design and commissioning efforts resulted in the first successful operation of this engine type at the Engine Research and Development Laboratory at the University of Toronto. Experiments were performed to determine optimum injection and glow plug shield geometry. Ignition characteristics were studied in a skip-fired test series. Emissions characteristics were studied in a continuously fired test series. The following conclusions were reached:

Glow Plug Shield and Injector Angle Screening Tests:

- Combustion is highly sensitive to the geometric configuration of the injector and glow plug shield. Only two of 20 tested geometric configurations resulted in repeatable combustion over a range of nominal equivalence ratios from 0.1 to 0.5.

- Glow plug power affects combustion characteristics. A voltage level of 14 VDC was found optimal. Lower voltage causes less repeatable combustion, higher voltage significantly limits glow plug life.

Skip-Fired Ignition Delay Tests:

- An ignition delay of under 2 ms is achievable in the DING engine. Higher intake pressure was found to extend ignition delay times. This effect is attributed to higher intake pressure causing slower jet penetration, and jet deflection away from optimal glow plug contact caused by a pressure-driven swirl momentum increase.

- Higher equivalence ratios and higher intake pressures result in longer fuel injection delays. As a result, combustion proceeds in two modes:
o **Stratified-Premixed Combustion** ($\phi_a \approx 0.2$ and $0.3$): Most of the fuel is injected prior to ignition, within the ignition delay period. Fuel has an opportunity to mix with air prior to the start of combustion. The fuel/air mixture is stratified.

o **Free-Mixing Combustion** ($\phi_a \approx 0.4$ and $0.5$): Longer fuel injection durations result in a two-phase combustion process: an initial premixed burn phase followed by a mixing-controlled burn phase. The fuel injection is over prior to the beginning of the mixing-controlled phase. A plume of free-mixing gas is formed after injection ceases. Mixing-controlled burning occurs in a plume vs. a jet (in diesel engines). Mixing is controlled by the engine swirl field.

- Higher intake pressure results in delay of peak in-cylinder temperature conditions in both combustion modes.

**Fired Emissions Tests:**

- NOx emissions depend on the combustion mode. In stratified-premixed combustion, NOx emissions increase with equivalence ratio due to higher temperatures associated with higher heat release. In free-mixing combustion, NOx levels fall with equivalence ratio due to lower oxygen availability and a possible mitigation of the NOx freeze effect.

- NOx emissions fall with higher intake pressure. Higher intake pressure delays peak in-cylinder temperature conditions, resulting in lower residence time available for high-temperature NOx formation.

- CO and PM emissions are highly dependent on the combustion mode. In stratified-premixed combustion, CO and PM emissions are low due to the high degree of mixing between the fuel and air. In free-mixing combustion, the developing mixing-controlled phase results in higher rates of CO and PM formation in the fuel-rich pockets of the mixing-controlled plume.

- In the free-mixing combustion mode, higher intake pressure reduces CO and PM formation rates and PM particle size by increasing the engine swirl momentum. Increased swirl momentum results in higher air entrainment, smaller amount of fuel-rich regions, higher oxygen availability, and lower flame temperatures. All effects reduce the
formation and increase the oxidation rates of CO and PM. These same effects also reduce PM particulate size by limiting surface growth time, oxidizing the surface growth species, and by increasing oxidation of the formed particles.

- The DING engine produces particles with modal diameters from 21.1 to 39.2 nm, which are smaller in comparison to diesel and GDI engines.

- The DING engine is capable of continuous operation with IMEP and combustion efficiency levels that are comparable to commercial direct injection engines. DING engine emissions performance is the range of current regulations for CO and PM. Without exhaust gas recirculation or exhaust aftertreatment, the DING engine operates above the regulated levels for NOx emissions.

6.2 Recommendations and Future Work

The following future work items are proposed in order to progress the research and development of the glow plug assisted direct injection natural gas engine technology.

**Exhaust Gas Recirculation (EGR):** EGR should be implemented on the DING engine since it is a standard NOx emissions technology implemented on most modern compression ignition engines. Operating the DING engine with EGR should reduce NOx emissions and will help determine the PM/NOx tradeoff characteristics of the DING engine.

**Computational Fluid Dynamics (CFD) Modelling:** Higher intake pressure was found to have multiple benefits in terms of emissions performance. One of the key suggested effects of higher pressure is higher swirl momentum, which leads to higher air entrainment rates. In order to better understand the jet/swirl mixing dynamics, CFD modelling of the jet and air swirl is proposed. Modelling results can yield valuable perspective on the flow and combustion characteristics.

**Glow Plug Shield and Injector Angle Optimization:** The screening tests presented in this thesis revealed that combustion is highly sensitive to the geometric arrangement of the injector, and the glow plug shield. The setting selected from the screening tests was found optimal for an intake pressure of 11 psig and natural gas injection pressure of 12 MPag. The skip-fired and fired emissions tests conditions, however, varied in intake pressure and used an injection pressure of 12.4 MPag. Performing a more detailed screening test for a range of injection and intake...
pressures is recommended as it may reveal important geometric dependencies of the ignition assist system.

**Emissions Test Time Limitation Improvement:** The time for the fired emissions tests was limited by the integrity of the polyamide piston rings. Thesis experiments were performed with only one piston ring set since another set failed in earlier tests. Piston ring design should be reviewed and new piston ring spares should be procured. This will allow for operational flexibility and hopefully enable the extension of the fired test duration beyond 6 minutes. A longer test duration can yield better steady state emissions data.

**Emissions Bench Reinstatement:** The Emissions Bench data was recorded in the thesis experiments, but was found to have multiple issues. Emissions Bench operational issues should be resolved with the vendor so it can be used in future experiments for confirmation of FTIR emissions measurements.

**Gravimetric PM Sampling:** Gravimetric sampling of PM emissions is suggested for implementation. Gravimetric sampling will enable: measurement of PM emissions on mass basis, analysis of elemental carbon to organic carbon (EC/OC) content, determination of soot morphology. EC/OC analysis is of special interest as it can help determine whether the primary particle growth mechanism is coalescence and agglomeration, or surface growth by absorption of volatile species.

**New Elliptical and Circular Orifice Nozzles:** Testing of an elliptical orifice injector nozzle vs. a circular nozzle has been a long-term interest at the ERDL. Elliptical nozzle orifices can increase fuel gas jet mixing and possibly lead to lower ignition delay and improved emissions performance. Past work on the CFR combustion bomb apparatus attempted to explore elliptical orifice nozzle performance, but faced two limitations. The first was that the original elliptical nozzle was matched in orifice area to the area of a circular orifice nozzle. However, elliptical and circular orifices of the same area have different discharge coefficients so the two nozzles exhibited different flow characteristics. The second limitation was that the CFR combustion bomb apparatus did not allow continuous fired running and emissions measurements. Design of new elliptical and circular orifice nozzles of matched flow characteristics is suggested for implementation. Some preliminary sizing calculations and additional nozzle design improvements are presented in Appendix D1.
**Optical Combustion Investigation:** Although optical access was implemented in the design of the DING engine, it was not used in the thesis experiments due to time limitations. All experiments were performed with the optical quartz window replaced with a solid aluminum puck. Since engine operation has been proven, optical tests can now be conducted. Special care should be taken on initial tests since the quartz windows have never been used on the DING engine. Mechanical integrity of the quartz window should be carefully tested by incrementally exposing the quartz window to a higher number of fired cycles. Optical investigation can reveal a great deal of information about fuel injection and combustion processes.

**DAQ System Modifications:** As mentioned in Chapter 4, the LabVIEW data acquisition program for fired emissions tests was limited to control of the fuel injector. The data collection capability was removed since it overloaded the data acquisition computer. The base data acquisition and control program dates back several years and was designed for the CFR combustion bomb apparatus. Development of a LabVIEW program, and upgrading the data acquisition computer may allow simultaneous injector control and data acquisition for fired emissions tests. Another feature of interest would be to implement the capability for multiple injections in the same engine cycle. This can enable new operating strategies that are currently used in new generation diesel engines.
References


Appendix A – Supplementary Information for Chapter 3

Appendix A1 – DING Engine Design Development, Modifications, and Commissioning

Lister ST Cylinder Head Modifications

The Lister STW cylinder block that houses the Bowditch piston can be used with a water-cooled Lister STW cylinder head, or an air-cooled Lister ST head. Modification of the water-cooled head to accept the natural gas injector would require boring through the coolant passages. This was deemed not practical. The aluminum air-cooled ST head does not have coolant passages. Instead, fins are used for heat dissipation. Since aluminum fins are simpler to modify, the air-cooled head was selected for use on the DING engine.

The original Lister ST head has an angled port (angled at 21.3° to the normal of the head plane), which is meant to hold a diesel injector. Since the intent was to mount the natural gas injector nozzle normal to the cylinder head plane, the original injector port could not be used. The original angled injector port was repurposed for mounting the glow plug. A custom adaptor was designed and installed in the original injector port (Drawing DING–DWG–1) to hold the glow plug shield and glow plug. A copper sealing ring (Drawing DING–DWG–2) was installed into the adaptor to seal the glow plug shield to the adaptor. With the glow plug position fixed, the natural gas injector had to be located in between the intake and exhaust ports. The Lister ST head had to be milled out to accommodate the natural gas injector. The distance between the glow plug axis and the injector orifice was replicated from the CFR combustion bomb apparatus and set to 0.299 in. (as shown in Figure 3.2). The injector tip had to fit through the narrow web between the intake and exhaust valve seats. The exact position of the injector tip had to selected such that the long injector body and solenoid casing would not interfere with the valve stem guides, valve springs, or rocker arms. Selection of this location was challenging and required extensive geometric study of the head assembly. Only one location without injector-valve train interference was found. The depth of the injector was set by the design requirement of placing the injection orifices at 0.245 in. from the cylinder head plane (as shown in Figure 3.2). At this depth, the larger section of the injector nozzle was found to interfere with the walls of the intake and exhaust ports. The geometry required to accept the injector nozzle would pierce through the intake and exhaust port walls. Since no other suitable injector location was found, the nozzle had
to be chamfered in order to avoid interference. Figure A1.1 shows a simplified illustration of the Lister ST head mill profile and the required nozzle chamfer modifications.

**Figure A1.1 – Simplified Lister ST Head Mill Profile and Injector Nozzle Modification**

The injector nozzle has to be able to safely withstand the 3000 psig supply pressure of the natural gas tank (in case the pressure regulator upstream of the injector fails). Since the nozzle modification reduced the local wall thickness of the injector nozzle, the modified nozzle was pressure-tested in a SolidWorks simulation. The simulation showed that the modified nozzle can withstand pressures well in excess of 3000 psig.

In order to seal the injector to the head, a ¼ in. female stainless steel Swagelok® tube fitting was turned down and press-fit in to the cylinder head. The injector tip was fitted with a ¼ in. brass ferrule. When installed, the injector tip seals to the cylinder head at the ferrule junction.

In addition to accommodating the fuel injector, the Lister ST head was modified to hold the Kistler 6121 pressure transducer. The transducer port was positioned at the edge of the cylinder head at a location where the head’s material thickness was sufficient enough to accommodate a slotted adaptor of the Kistler pressure transducer.

Once the design of cylinder head modifications was finalized, a 3D model of the head geometry modifications was developed in SolidWorks (“Lister ST Cylinder Head Mill Profile.SLDprt”). This model was used to mill the geometry required for accommodating the natural gas injector and pressure transducer into the original Lister ST head. Milling was done on a 3-axis CNC mill. Subsequent modifications were performed on a manual mill.
Lister Cylinder Head Heating and Cooling

In order to control the temperature of the cylinder head, heating and cooling capability was implemented. In order to provide heating, a high-temperature heat cable (McMaster-Carr 3641K16) was wrapped around the aluminum cooling fins and connected to a temperature controller on the auxiliary control unit. In order to provide cooling, a Dayton 1TDR6 blower was installed on the DING engine equipment support frame. The discharge of the blower was routed to the cylinder head via a flexible 3 in. hose, and a custom built shroud which mounts onto the cylinder head. The blower power supply was connected to the head temperature controller on the auxiliary control unit. In order to provide temperature feedback, a 1/16 in. diameter K-type thermocouple was installed roughly midway into the thickness of the cylinder head. The temperature controller was programmed to turn on the heater if the head temperature is below the set point, and to turn on the cooling fan if the head temperature is above the set point.

Valve Train Modifications

The Ricardo Hydra engine was originally equipped with a cam box meant for operation of the Hydra engine in Spark Ignition (SI) mode. When initially tested, the camshaft emitted a sharp cyclical mechanical sound. Disassembly of the cam box revealed that the one of the camshaft lobes was worn and that the sound was produced by the intake valve tappet hitting the wear site. It was also noticed that lubrication oil was only provided to the camshaft bearing. No direct lubrication was provided to the cam lobes. Lack of cam lobe lubrication was suspected to be the reason for the lobe wear. A spare Hydra cam box, designed for operation of the Hydra as a diesel engine, was available. This cam box was installed in place of the SI cam box. The lubrication system was modified to deliver a stream of oil to each of the intake and exhaust cam lobes through 1/8 in. diameter flexible tubing. Replacement of the cam box and the lubricant delivery modification resolved the cam lobe wear issues.

The valve timing on the DING engine was initially set such that the intake valve maximum lift occurs at 110 CAD ATDC (as recommended from prior work on the Hydra engine [25]). Motored tests, however, showed that engine compression was suboptimal. To aid in compression, advancing valve timing so that the intake valve closes earlier in the compression stroke, was desired. Since modification of the valve timing may result in valve interference with
the piston, valve lift profiles vs. crank angle were carefully measured. The position profiles of the piston crown’s top deck and of the crown’s valve recess surfaces were calculated from the known engine geometry. The profiles were combined into a positional model that shows potential valve interference with a shift in timing. The model assumes a hot piston/head clearance of 0.023 in. (0.085 in. cold clearance minus 0.062 in. thermal elongation of the Bowditch piston). The resulting positional model was used to determine the earliest valve timing without valve interference. This timing corresponded to an intake valve maximum lift occurring at 104 CAD ATDC. This timing was set on the DING engine. The position profiles for the set valve timing are shown in Figure A1.2.

![Piston Crown & Valve Position Profiles](image)

**Figure A1.2 – Piston Crown and Valve Position Profiles**

It should be noted from the figure above that although the intake valve intersects the position of the piston crown’s top deck, no clashing occurs since the surface of crown’s valve recess does not intersect the intake valve position. The resulting intake and exhaust valve closing and opening times are shown below:

- Intake Valve Opening (IVO) = 20 BTDC
- Exhaust Valve Opening (EVO) = 48 BBDC
- Intake Valve Closing (IVC) = 48 ABDC
- Exhaust Valve Closing (EVC) = 4 ATDC
**Lubricant and Coolant System Preparation**

The lubricant system on the Hydra engine was flushed. The oil filter was replaced and the engine was filled with new oil (Mobil Super 1000 5W–30). The coolant system was flushed to remove old coolant and corrosion products. All coolant hoses were replaced with new silicone high-temperature hose.

**Cylinder Leak down Test**

A cylinder leak down test was performed using a Snap-On MT 324 cylinder leak detector. The crank position was set to TDC. The leak detector was zeroed using 60 psig supply air pressure, and was then connected to the combustion chamber via the glow plug adapter (with glow plug removed). Several consecutive tests showed a consistent leakage rate of 12%. Considering that a conventional rebuilt engine will leak at or below 10%, the result of 12% was an indication of good sealing of the polyamide piston rings.
Appendix A2 – Bowditch Piston Thermal Elongation Calculation

Elongation of the Bowditch piston is critical to consider in order to avoid clashing of the piston crown with the cylinder head or the intake and exhaust valves. To estimate the potential elongation, thermal expansion was calculated for an average temperature rise of 200º C (360º F). The 200º C value was determined in commissioning tests and is roughly based on the maximum measured cylinder head temperature reached in continuous fired operation. With the overall piston and crown length of 13.2 in., expansion was calculated to be 13.2 in x 13 x 10⁻⁶ in/ºF in x 360º F = 0.062 in. This value was used as the minimum piston clearance for the DING engine.
Appendix A3 – DING Engine Compression Ratio Calculation

Calculation of compression ratio was quite challenging as it is highly sensitive to the volumetric measurement of the intricate features of the combustion chamber geometry. Volumes around existing features such as the valve seat and pressure transducer adapter crevices were carefully measured by recording the volume of ethanol (dispensed from a 5 ml pipette) used to fill these crevice features. The remaining geometry was modelled in SolidWorks CAD software. The calculation of the compression ratio is shown below.

### Compression Ratio Calculation

<table>
<thead>
<tr>
<th>Engine Geometry Parameters</th>
<th>Stroke 3.5 in (8.89 cm)</th>
<th>Bore 3.75 in (9.525 cm)</th>
<th>Connecting Rod 6.22 in (15.801 cm)</th>
<th>Cold Clearance 0.085 in (0.2159 cm)</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Feature</th>
<th>Volume (cu in)</th>
<th>Measurement Source</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stroke</td>
<td>3.5 in</td>
<td>8.89 cm</td>
<td></td>
</tr>
<tr>
<td>Bore</td>
<td>3.75 in</td>
<td>9.525 cm</td>
<td></td>
</tr>
<tr>
<td>Connecting Rod</td>
<td>6.22 in</td>
<td>15.801 cm</td>
<td></td>
</tr>
<tr>
<td>Cold Clearance</td>
<td>0.085 in</td>
<td>0.2159 cm</td>
<td></td>
</tr>
</tbody>
</table>

**Volume Features (empty spaces +ve, solid features -ve)**

<table>
<thead>
<tr>
<th>Feature</th>
<th>Volume (cu in)</th>
<th>Measurement Source</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clearance Disk Volume</td>
<td>0.9388 cu in</td>
<td>calculated</td>
<td>3.75&quot; OD (bore), 0.085&quot; deep (cold clearance)</td>
</tr>
<tr>
<td>Intake Valve Crevice</td>
<td>0.1092 cu in</td>
<td>pipette measurement</td>
<td></td>
</tr>
<tr>
<td>Exhaust Valve Crevice</td>
<td>0.0677 cu in</td>
<td>pipette measurement</td>
<td></td>
</tr>
<tr>
<td>Kistler Hole Crevice</td>
<td>0.0082 cu in</td>
<td>pipette measurement</td>
<td></td>
</tr>
<tr>
<td>Crown Height Disk</td>
<td>4.9701 cu in</td>
<td>calculated</td>
<td>3.75&quot; OD (bore), 0.45&quot; deep (crown height)</td>
</tr>
<tr>
<td>Crown Part Volume</td>
<td>-2.7498 cu in</td>
<td>Solidworks model</td>
<td></td>
</tr>
<tr>
<td>Bolt + Washer Volume</td>
<td>-0.1446 cu in</td>
<td>Solidworks model</td>
<td></td>
</tr>
<tr>
<td>Bolt Hole</td>
<td>-0.0878 cu in</td>
<td>calculated</td>
<td>0.255&quot; OD, 0.215&quot; deep</td>
</tr>
<tr>
<td>Window/Puck</td>
<td>-0.0594 cu in</td>
<td>calculated</td>
<td>2.75&quot; OD, 0.010&quot; deep</td>
</tr>
<tr>
<td>Gasket</td>
<td>-0.0839 cu in</td>
<td>calculated</td>
<td>2.75&quot; OD, 2&quot; ID, 0.030&quot; deep</td>
</tr>
<tr>
<td>Crevise Volume</td>
<td>0.2302 cu in</td>
<td>calculated</td>
<td>3.75&quot; OD (bore), 3.671&quot; ID (piston ID), 0.5&quot; deep (distance to first piston ring)</td>
</tr>
</tbody>
</table>

| Injector Tip             | -0.0137 cu in  | Solidworks model   | Installed 0.320" deep                            |
| Glow Plug & Shield       | -0.0086 cu in  | Solidworks model   | Installed 0.380" deep (vertical distance head plane to center tip of plug) |

| Clearance Volume          | 3.1765 cu in   | 52.0543 cm³        |                                                    |
| Swept Volume              | 36.6563 cu in  | 633.4649 cm³       | 0.375" OD (bore), 3.5" deep (stroke)              |
| Max Volume                | 41.8329 cu in  | 685.5192 cm³       |                                                    |

**Compression Ratio** 13.1693
Appendix A4 – Piston Ring Experiments and Modifications

Initial DING engine tests were performed with Vespel® SP-21 polyamide piston rings, supplied by Cook Compression for a previous project using the Ricardo Hydra setup. In initial DING engine tests, the polyamide piston rings performed reasonably well. Since only one spare set of polyamide piston rings was available, Cook Compression was contacted to order more spare piston ring sets. A new material, TrueTech 3210®, was proposed by Cook Compression’s engineering department. TrueTech® is a composite material consisting of a Teflon® matrix filled with bronze powder. Teflon® allows for oil-less lubrication, while the bronze filler is required for heat dissipation. Two sets of TrueTech piston rings were ordered from Cook Compression.

At a later project stage, one of the original polyamide piston rings broke in the middle of a fired test. The DING engine was refitted with the new TrueTech® piston rings. Initial tests of the TrueTech® piston ring design were unsuccessful. The engine ceased past approximately 2 minutes of continuous fired running. Excessive thermal expansion of the piston rings was suspected. In order to allow for more expansion, the original expansion gap of 0.060 in. was remachined to a width of 0.100 in. No performance improvement was noted, the engine seized after approx. 2 min of running. Three incremental modifications of the gap to a final width of 0.250 in. resulted in only minor performance improvement. From conducting motored (unfired) tests with the TrueTech® piston rings, it was noted that the cylinder block and head temperature raised drastically, when compared to a similar test performed with polyamide piston rings. This indicated that the TrueTech® material has a much higher effective friction coefficient than the original Vespel® SP-21 polyamide material.

Due to unsuccessful implementation of the new design, the DING engine was refitted with the last set of the original polyamide piston rings. Testing of this ring set showed that the polyamide rings were also exhibiting excessive thermal expansion. The engine ceased after about 3 minutes of continuous fired tests. To allow for thermal expansion, the original expansion gap of 0.060 in. was increased to 0.230 in. Inspection of the previously failed polyamide piston ring pointed out an additional design flaw of the piston ring gap design. The corner of the piston ring expansion gap was sharp and acted as a stress raiser. In fact, the failure of the original polyamide ring occurred at this corner. In order to eliminate stress concentration, widening of the expansion gap was done on a vertical mill using a 1/16 in. ball end mill. The ball end mill was used to impart a
radius on the corner of the expansion gap. Figure A4.1 shows a simplified diagram of the polyamide piston ring modification.

Figure A4.1 – Vespel® Piston Ring Modification

Testing of the modified polyamide piston rings was successful. The modified ring set was used for all thesis experiments.

**Fuel Injector Modifications**

The fuel injector used in CFR combustion bomb experiments had to be modified to fit the DING engine configuration. The original body of the injector was redesigned by Chown [13] to fit the geometry of the modified Lister ST head. The modified body was designed to be much narrower than the original, in order to avoid clashing with the valve train of the DING engine. Chown’s new injector body design, however, did not provide adequate internal passages for the natural gas to reach the injector nozzle. The gas flow was impeded by the injector needle bushing installed to guide the injector needle. Although flow channels were provided in the bushing, they were covered up by the injector needle spring. Minimizing the pressure losses of natural gas as it makes its way from the supply side to the injector orifices is critical for achieving choked flow at the nozzle orifices. In order to minimize flow restrictions, a new injector needle guide bushing with internal flow passages was designed (DING–DWG–3). In order to accommodate this new bushing design, the injector body designed and fabricated by Chown had to be modified. The original recess that holds the injector needle guide bushing was extended to a depth of 0.850 in.
from the injector o-ring recess face. Further injector modification addressed injector sealing issues. These modifications are covered in Appendix A5.
Appendix A5 – Injector Seal Redesign and Modifications

The fuel injector was assembled and tested with the modified injector body. A leakage issue was found at the injector tip seal. With the solenoid energized, the injector should not allow any flow of fuel. Significant leakage, however, was found with a bubble leak test. Review of the seal design revealed two possible leakage paths. These are presented in Figure A5.1, below.

Figure A5.1 – Injector Seal Leakage Paths

Leakage Path 1

Leakage path 1 was investigated. In the original nozzle design, the injector needle seat was made from Vespel® polyamide material. The seat was press-fit into the injector tip. The injector needle was made from tool steel. As with many seal designs, the injector seal design relied on contact of the hard surface (needle tip) and a deformable surface (Vespel® seat). The original needle was found to be out-of-round and with a worn tip surface. Both deficiencies contributed to imperfect contact of the tip and seat, and resulted in leakage and rapid Vespel® seat deterioration. To mitigate these effects, a new needle was designed and fabricated (DING–DWG–04). Special care was taken in the design and fabrication process to assure that the new injector needle is within +/- 0.001 in. straightness over its entire length. This was achieved by careful machining of a single piece of precision-ground tool steel round rod stock. The original needle design was also improved by incorporating threaded needle spring retention locknuts (DING–DWG–05), instead of a fixed spring retention flange of the original needle design. This modification allows for adjustment of the needle spring preload without the need for shimming.
Testing of the new needle showed a significant reduction in injector seal leakage. However, some leakage was still present and required further investigation.

**Leakage Path 2**

Due to high wear rates, the Vespel® needle seat needed to be frequently replaced and reinstalled. It was suspected that upon installation, the seat could be damaged by the press-fit process. Inspection of the surface finish of the seat installation site (the recess in the nozzle tip that accepts the seat) revealed a rough machined finish. It was suspected that the rough finish scored the sides of the Vespel® seat as it is being installed into the nozzle tip. It was also suspected that the frequent reinstallation would, over time, damage the surface of the accepting seat site. Both scenarios would result in leakage via path 2 (see Figure A5.1). This hypothesis was confirmed by installing a Vespel® plug (identical to the Vespel® seat, but with no center hole) and pressurizing the nozzle. Leakage path 2 was observed. Different levels of press-fit were tested by increasing the outer diameter of the Vespel® seat. This was done in an attempt to eliminate injector seal leakage via path 2. Some improvement was achieved, but some injector seal leakage was still present.

It was concluded that the key issue of the original seal design was the damage imparted on the seat and the seat site by the installation process. To eliminate this issue, a new design was proposed and implemented. The seat material was changed from Vespel® to solid copper, and the needle was modified to have a Vespel® tip (DING–DWG–06). The advantage of the metal seat is that once properly installed, it would not have to be replaced on a frequent basis, as its wear rate would be much less than that of the original Vespel® seat. The advantage of having a Vespel® needle tip is that the deformable surface would no longer be prone to damage due to frequent reinstallation, and could now also be easily refinished due to the accessibility of the needle tip. An additional advantage of this design was that the Vespel® material is likely to see much lower temperatures when installed on the needle tip vs. the original Vespel® seat design. The tip locations is farther from the combustion heat and the metal injector needle acts like a fin to dissipate any heat transferred to the Vespel® needle tip.
Appendix A6 – Natural Gas Composition

Average composition of the natural gas used for all thesis experiments, as provided by Enbridge for the Toronto area, is presented in Table A6.1.

*Table A6.1 – Toronto Area Natural Gas Composition (November 2015)*

<table>
<thead>
<tr>
<th>Constituent</th>
<th>Vol %</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane (CH₄)</td>
<td>94.46</td>
</tr>
<tr>
<td>Ethane (C₂H₆)</td>
<td>3.73</td>
</tr>
<tr>
<td>Propane (C₃H₈)</td>
<td>0.44</td>
</tr>
<tr>
<td>n–Butane (C₄H₁₀)</td>
<td>0.04</td>
</tr>
<tr>
<td>i–Butane (C₄H₁₀)</td>
<td>0.03</td>
</tr>
<tr>
<td>n–Pentane (C₅H₁₂)</td>
<td>0.00</td>
</tr>
<tr>
<td>i–Pentane (C₅H₁₂)</td>
<td>0.00</td>
</tr>
<tr>
<td>Hexanes Plus (C₆H₁₄+)</td>
<td>0.00</td>
</tr>
<tr>
<td>Nitrogen (N₂)</td>
<td>0.83</td>
</tr>
<tr>
<td>Carbon Dioxide (CO₂)</td>
<td>0.47</td>
</tr>
</tbody>
</table>
Appendix A7 – Instrument Calibration

In-Cylinder Pressure (Kistler 6121) Transducer Setup and Calibration

A Kistler 6121 pressure transducer (SN# 647456) coupled to a Kistler 5004 charge amplifier was used for in-cylinder pressure measurements. This transducer has a sensitivity of 13.8 pC/bar (0.138 pC/kPa). With maximum anticipated cylinder pressure of 20,000 kPa and voltage output range of –10 to +10V, an amplifier output ratio of 2000 kPa/V is desired. To achieve this, the amplifier settings are set as:

- Range Scale = 50 mechanical units/volt
- Sensitivity Range = 1–11
- Sensitivity Potentiometer = 5.52 pC/mechanical unit
- Time Constant = medium

**Note:** Refer to Appendix A of “Cylinder Pressure Measurement System User’s Guide” and to the “Kistler Model 5004 Dual Mode Amplifier” documents for more information.

Following the setup of the Kistler 5004 amplifier, the Kistler 6121 pressure transducer was calibrated. Since this piezoelectric transducer responds to changes in pressure, it was calibrated against the AST4700 pressure transducer, which responds to absolute pressure. The Kistler and the AST4700 pressure transducers were mounted in the natural gas fuel manifold. Pressurized natural gas was then introduced into the manifold, and the Kistler and AST4700 sensor readings were allowed to stabilize. The pressure was then rapidly released into the bleed line. The changes in voltage of both transducers were recorded using a LabVIEW virtual instrument program “KISTLER CALIBRATION.vi”. The pressure change was determined by using the AST4700 reading. The calibration for the AST4700 instrument was provided by the manufacturer. The change in the Kistler voltage was recorded and plotted against the pressure change. In total six pressures were tested. Each test was performed three times. The results were averaged. The resulting data, shown in Figure A7.1, was fit with a best fit line to obtain the Kistler transducer calibration constant of 2042.70 kPa/V.
The calibration data for the AST4700 transducer was provided by the manufacturer as a set of two calibration points: 0.000 V at 0 psig, 4.992 V at 2500 psig. This calibration is valid for the life of the transducer (as per AST). The zero point was verified at the laboratory and showed to have changed over time to about −0.05 V at 0 psig. Deviation from the original calibration was deemed negligible.

MAP 1 and MAP 2 Absolute Pressure Transducer Calibration

The experimental setup has two absolute pressure transducers measuring pressure at the intake manifold (MAP1 Transducer) and at the laminar flow meter (MAP 2 Transducer). These transducers were calibrated against a new omega PX309–300A5V pressure transducer that came with a factory NIST traceable calibration certificate. The Omega PX309, MAP1, MAP2 transducers were connected to a common manifold, and pressurized with compressed air. Pressure was controlled with a regulator. A LabVIEW virtual instrument program “Pressure Transducer Calibration.vi” was used to track the pressure indicated by the omega PX309 transducer and the voltages produced by the MAP1, MAP2 transducers. Pressures in increments of 5 psig were tested starting from atmospheric pressure up to the highest pressure that each of the tested transducers could measure in its output voltage range. Voltage vs. pressure was plotted
for each transducer. A best fit line was generated to reflect each transducer’s calibration. The calibration graphs are presented below.

**Figure A7.2 – Intake Pressure (MAP 1) Transducer Calibration**

\[
y = 63.498467x + 1.275535 \\
R^2 = 0.999593
\]

**Figure A7.3 – Laminar Flow Meter Pressure (MAP 2) Transducer Calibration**

\[
y = 43.447647x - 2.041144 \\
R^2 = 0.991081
\]

**Laminar Flow Meter Calibration**

In order to calibrate the laminar flow meter, the intake system was rerouted to a gas displacement meter. This apparatus allows volumetric flow rate measurement by timing the period required to displace a known gas volume. An Arduino® based system was used to time the displacement events. Multiple flowrates were tested. The voltage output from the laminar flow meter was
tracked using LabVIEW. Calibration results, flowrate vs. laminar flowmeter voltage, are shown in the graph below.

**Figure A7.4 – Laminar Flowmeter Calibration Curve**

For fired emissions tests, intake air flow was tracked by recording mm H₂O values (pressure drop across laminar from element) displayed on the Cussons control cabinet “Air Meter” readout. The following curve was established to relate pressure drop values to laminar flowmeter voltage. This curve can be used in conjunction with the curve above to convert mm H₂O pressure drop values to air flow measurements.

**Figure A7.5 – Laminar Flowmeter Pressure Drop vs. Output Voltage**
Appendix A8 – Fuel Injector Preparation and Calibration

Prior to every test the fuel injector was fully disassembled, cleaned, reassembled and recalibrated. The following subsections describe injector servicing and calibrations procedures, and present injector calibration settings and results.

Injector Servicing Procedure:

1) Remove injector from DING engine head
2) Undo injector nozzle flange bolts, remove injector nozzle and injector nozzle o-ring
3) Unscrew the solenoid casing cap while counter-locking the injector body with the solenoid casing, remove solenoid o-ring
4) Remove armature: hold the injector needle by counter-locking the spring retention nuts, unscrew armature holding it by the two opposing perimeter notches
5) Remove the injector needle and spring, unscrew and remove the spring retaining locknuts
6) Unscrew armature casing from the injector body, remove injector angle indicator and the injector mounting cross plate
7) Thoroughly clean all components with isopropyl alcohol
8) Resurface the Vespel® needle tip:
   • mount needle in a 1/8” diameter collet on a lathe
   • using a very sharp high-speed steel tool bit, cut the tip on a 59° angle at 3000 rpm, removing just enough to achieve a clean surface
   • polish the tip on the lathe with a rag dipped in 600 grit lapping compound, making sure to not round the tip edges
   • inspect the tip for smooth surface finish
9) Clean the injector tip copper seat with a cotton swab dipped in 600 grit lapping compound
10) Inspect the copper seat surface using a bore scope, make sure the surface is free of dirt and oxides
11) Clean the injector tip with isopropyl alcohol (best to use an ultrasonic bath)
12) Flow test the nozzle with 10 psig compressed air in a water bath to make sure all injector orifices are flowing equally
13) Reinstall the injector mounting cross plate, the injector angle indicator, and couple the injector body and solenoid casing
14) Reinstall the injector needle, spring, spring retaining locknuts and armature
15) Reinstall solenoid o-ring and install solenoid casing cap

16) Adjust the injector spring preload by setting and counter-locking the position of the spring retaining nuts, use the injector assembly apparatus to set the needle preload to 40 lb

Note 1: refer to Reference [23] Appendix C.2 for instruction on using the injector assembly apparatus

Note 2: Special care needs to be taken not the damage the needle tip when setting the preload. Make sure that the needle is allowed to slide back in the spring compression direction without the armature clashing with the solenoid (set a very large needle lift distance for this step in the servicing procedure)

17) Install new injector tip o-ring and install nozzle

18) Set the needle lift distance by hand tightening the solenoid casing until it lightly touches the armature, then back off the solenoid casing counterclockwise by 75°

19) Lock the armature in place with the armature locknut, make sure that the armature casing does not slip with respect to thread engagement with the injector body
**Injector Calibration Procedure:**

1) Install injector on the DING engine auxiliary frame (a special mounting bracket is provided to securely hold the injector)

2) Connect the flexible natural gas hose to the injector fuel port

3) Pressurize the injector with natural gas

4) Check injector tip for leakage with a bubble test

5) Connect injector power

6) Disable the safety relays (used when running) by bypassing the injector power cable around the relays (use quick connect terminals)

7) Install heated adapter onto the injector nozzle tip

8) Connect the adapter’s thermocouple and power leads to the cylinder head controller on the auxiliary control unit

9) Turn on controller and set a set point of 200°C

10) Turn on and tare weight scale

11) Unlock the valve on the test cylinder

12) Turn on the DAQ system

13) In LabVIEW, open MassTest_C.vi

14) Check that the cylinder pressure is at 400 psig, bleed pressure into vacuum trench if necessary.

**CAUTION:** *Never inject fuel into a cylinder full of air or oxygen. If required, fill and purge cylinder with nitrogen to remove air/oxygen content.*

15) Enter the number of injections, the injection frequency in Hz, and the injection duration in ms

16) With the sample cylinder at 400 psig and the isolation valve closed, place the cylinder on the weight scale and press “TARE”

17) Attach the cylinder to the injector adapter with the modified Swagelok® quick connect union

18) Open the sample cylinder isolation valve

19) Run the LabVIEW program

20) When injections are over, quickly close the isolation valve, remove the sample cylinder and weight it
21) Record results, repeat steps 14–20 three times for each injection duration tested

22) After calibration completion be sure to close the sample cylinder isolation valve. Lock the valve handle with a nut and bolt and cap the valve end with a tube fitting cap.

*Note*: When setting a new injection duration in LabVIEW, run the newly set duration for 5 injections into room air, then change the number of injections back the desired test amount. LabVIEW retains the old settings for the first injection duration, the procedure above clears the old setting.

**Injector Calibration Settings**

Table A8.1 presents the injector calibration settings used for the thesis experiments.

*Table A8.1 – Injector Calibration Settings*

<table>
<thead>
<tr>
<th>Setting</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injector Spring Preload</td>
<td>Approx. 40 lb</td>
</tr>
<tr>
<td>Injector Needle Lift</td>
<td>~75° thread engagement (0.010 in)</td>
</tr>
<tr>
<td>Number of Injections/Test</td>
<td>100</td>
</tr>
<tr>
<td>Number of Tests Recorded per Injection Duration</td>
<td>3</td>
</tr>
<tr>
<td>Injection Frequency</td>
<td>16.67 Hz</td>
</tr>
<tr>
<td>Injector Nozzle Adapter Temp</td>
<td>200° C</td>
</tr>
<tr>
<td>Sample Cylinder Pressure</td>
<td>400 psig</td>
</tr>
</tbody>
</table>

**Injector Calibration Curves**

Figure A8.1 shows the pre-test calibration curves for the three test days for the 0.2 mm diameter 9-hole circular orifice nozzle.

*Figure A8.1 – Pre-Test Injector Calibration (0.2 mm diameter 9-hole circular orifice nozzle)*

\[
y = 0.769130x + 7.217382 \\
R^2 = 0.991091
\]
Appendix B – Supplementary Information for Chapter 4

Appendix B1 – Engine Test Preparation

The following checklist was used for engine test preparation:

<table>
<thead>
<tr>
<th>Table B1.1 – Engine Test Checklist</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Injector Calibration</strong></td>
</tr>
<tr>
<td>resurface needle, assemble injector, record needle preload and lift</td>
</tr>
<tr>
<td>check injector tip for leakage at injection pressure</td>
</tr>
<tr>
<td>calibrate injector at 200°C</td>
</tr>
<tr>
<td><strong>Pre Test</strong></td>
</tr>
<tr>
<td>switch heated lines to FTIR &amp; Emissions Bench (EB)</td>
</tr>
<tr>
<td>install EEPS setup (connect: air line, exhaust line, computer cable)</td>
</tr>
<tr>
<td>turn on EB &amp; FTIR lines, heated filters and exhaust heater</td>
</tr>
<tr>
<td><strong>Engine Assembly</strong></td>
</tr>
<tr>
<td>clean piston and install piston rings</td>
</tr>
<tr>
<td>clean and install cylinder block</td>
</tr>
<tr>
<td>oil the cylinder wall and hand crank engine (clean off top excess oil)</td>
</tr>
<tr>
<td>check crown bolts for tightness</td>
</tr>
<tr>
<td>clean, lube and reinstall valves</td>
</tr>
<tr>
<td>install injector into head</td>
</tr>
<tr>
<td>clean glow plug and shield, install and set shield angle</td>
</tr>
<tr>
<td>install new glow plug (if required)</td>
</tr>
<tr>
<td>clean head and crown</td>
</tr>
<tr>
<td>install centering ring and copper gasket, install head</td>
</tr>
<tr>
<td>connect thermocouple, Kistler cable and glow plug power, bolt on shroud</td>
</tr>
<tr>
<td>install valve rockers and pushrods</td>
</tr>
<tr>
<td>hand tighten head, torque injector cross plate bolts to 6 Nm</td>
</tr>
<tr>
<td>check Bowditch piston bottom bolts (connection to Hydra)</td>
</tr>
<tr>
<td>turn on Cussons dyno unit, coolant heater, oil heater, and head heater</td>
</tr>
<tr>
<td>set valve clearance: 0.006” go, 0.008” no go</td>
</tr>
<tr>
<td>connect intake and exhaust piping</td>
</tr>
<tr>
<td><strong>Start-up</strong></td>
</tr>
<tr>
<td>check safety trips</td>
</tr>
<tr>
<td>fill natural gas line</td>
</tr>
<tr>
<td>warm up head to 100°C (120°C set point), engine coolant to 60°C</td>
</tr>
<tr>
<td><strong>Calibration</strong></td>
</tr>
<tr>
<td>calibrate EEPS, EB and FTIR analyzers</td>
</tr>
<tr>
<td>do not forget to check calibration ranges and span O2 with room air</td>
</tr>
<tr>
<td>reset the EEPS dilution ratio</td>
</tr>
<tr>
<td>turn EB sampling valve to sample exhaust</td>
</tr>
<tr>
<td><strong>Testing</strong></td>
</tr>
<tr>
<td>set intake pressure with LabVIEW VI</td>
</tr>
<tr>
<td>check torque calibration</td>
</tr>
<tr>
<td>motor engine and set rpm</td>
</tr>
<tr>
<td>adjust air flow meter cable to avoid signal noise</td>
</tr>
<tr>
<td>perform experiments</td>
</tr>
</tbody>
</table>
Appendix C – Supplementary Information for Chapter 5

Appendix C1 – DING PM Size Distribution vs. Other Engine Types

Figure C1.1 shows a particle size distribution for a 1997 Cummins 3.9 L engine operated on Ultra Low Sulphur Diesel (ULSD). Mode 2 corresponds to a high engine speed condition at an approximate equivalence ratio of 0.5. Mode 9 corresponds to a low engine speed condition at an approximate equivalence ratio of 0.3.

The distribution shown for the Cummins diesel is much wider than that of the DING engine, and has a larger proportion of agglomeration mode particles. The modal diameters are 40 and 59.4 nm, which is somewhat higher than the DING modal diameters of 21.1 to 39.2 nm (depending on test conditions).

*Figure C1.1 – Diesel PM Size Distribution [28]*
Figure C1.2 shows a PM size distribution for a 2012 Ford 2.0 L GDI engine [29].

The distribution shown for the GDI engine is much tighter than that of the DING and Cummins diesel engines. The modal diameter is 70 nm. Approximately 30–40% of the particles are in the agglomeration particle size range. Comparison of the DING engine PM size distribution shows that DING operation results in smaller particles vs. other engine types.
Appendix D – Supplementary Information for Chapter 6

Appendix D1 – New Elliptical and Circular Orifice Injector Nozzle Design

Nozzle Sizing Criteria

DING engine commissioning tests showed that the 0.3 mm diameter circular and the orifice-area-matched elliptical injector nozzles designed by Chown for the CFR bomb apparatus were not optimally sized for the DING engine setup. Both nozzles proved to be oversized, delivering too much fuel in a timeframe that was often shorter than the critical ignition delay value of 2 ms. This resulted in primarily premixed combustion, instead of the desired mixing-controlled combustion process. The following high-level criteria were identified for the nozzle flow performance. These criteria apply to both the circular and elliptical nozzles:

- Fuel flow should be choked
- Injection characteristics should allow equivalence ratios of 0.2 – 0.6 for:
  - 1000 RPM engine speed
  - Intake pressures of 34–103 kPag (5–15 psig)
  - Natural gas supply pressure of 12.4 MPag (1800 psig)
  - Minimum injection duration of 2 ms (12 CAD @ 1000 rpm)
  - Maximum duration of 55 CAD – based on the latest end of injection at 30 CAD ATDC with an SOI at 25 CAD BTDC

The small 0.2 mm diameter 9-hole circular orifice nozzle was found to fit the above criteria quite adequately. The range of actual equivalence ratios achieved for different injection durations for this nozzle are presented in Figure D1.1. These results were taken from the skip-fired test series.
### Sizing an Elliptical Orifice to Circular Orifice for Equivalent Flow

Jets issuing from elliptical orifices with a major to minor diameter ratio of 2:1 have been shown to be able to entrain more air than jets issuing from a circular orifice of equivalent flow rate characteristics [20]. Sizing of elliptical orifices to match the flow characteristics of a circular orifice cannot be done simply on orifice area basis. The area matching approach was used by Chown and proved to result in differing flow characteristics between the 0.3 mm diameter circular 9-hole nozzle and the flow-area-matched elliptical orifice nozzle. The flow characteristic difference results from the fact that elliptical orifices have been shown to have higher discharge coefficients than circular orifices. Reference [32] provides a comparison of discharge coefficients for circular, and a 2:1 axis ratio elliptical orifices of equal area. Table D1.1 summarizes the discharge coefficients for three different ratios of upstream to downstream pressure. This information was taken from the graph on page 23 of Reference [32].

#### Table D1.1 – Circular and Elliptical Orifice Discharge Coefficients [32]

<table>
<thead>
<tr>
<th>$P_{\text{upstream}}/P_{\text{downstream}}$</th>
<th>2</th>
<th>2.2</th>
<th>2.4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Discharge Coef. ($C_{dc}$) – circular orifice</td>
<td>0.754</td>
<td>0.773</td>
<td>0.786</td>
</tr>
<tr>
<td>Discharge Coef. ($C_{de}$) – 2:1 axis ratio elliptical orifice</td>
<td>0.771</td>
<td>0.791</td>
<td>0.803</td>
</tr>
<tr>
<td>$C_{dc}/C_{de}$</td>
<td>0.978</td>
<td>0.977</td>
<td>0.979</td>
</tr>
</tbody>
</table>
The average ratio of the circular orifice to elliptical orifice discharge coefficient is 0.978. Since the area of an orifice is inversely proportional to the discharge coefficient, elliptical and circular orifice areas of equivalent flow characteristics can be approximately related with the following relationship:

\[ A_{2:1\text{ ellipse}} = A_{\text{circle}} \left( \frac{C_{dc}}{C_{de}} \right) \]

Where:

- \( A_{2:1\text{ ellipse}} \) is the area of the 2:1 axis ratio elliptical orifice
- \( A_{\text{circle}} \) is the area of a circular orifice of equivalent flow characteristics
- \( C_{dc} \) is the circular orifice discharge coefficient
- \( C_{de} \) is the 2:1 axis ratio elliptical orifice discharge coefficient

Using the above equation, the total flow area of a 2:1 axis ratio elliptical 9-hole nozzle that corresponds to the flow characteristics of a 0.2 mm diameter circular 9-hole nozzle can be calculated as 0.276 mm\(^2\). This area corresponds to 9 elliptical orifices, each with a major diameter of 0.280 mm and a minor diameter of 0.140 mm.

**Proposed Nozzle Design Improvements**

**Corrosion Resistant Construction:** Injector nozzles fabricated by Chown [13] were made from 01 tool steel. Rust was found to develop deep inside the injector nozzles and is suspected to be a major contributor to nozzle performance deterioration and variability. To eliminate rusting and flow performance deterioration, new nozzles are recommended to be manufactured from stainless steel or titanium.

**Integrated Needle Seat:** Development of the injector seal design, described in Appendix A5, showed that a copper seat and a Vespel® injector needle tip result in improved sealing and serviceability of the seal assembly. For new nozzles, the injector seat is proposed to be included into the geometry of the nozzle itself. This will eliminate the possibility of fuel leaking around the seat, as was found to occur in the press-fit seat design (see leakage path 2 in Figure A5.1). It is also recommended for the final internal seat profile to be finished with electric discharge
machining (EDM). This method can be used to achieve much finer seat surface finish, which should improve sealing contact of the seat surface with the Vespel® injector needle tip surface.
Appendix E – Mechanical Part Drawings

1. DING–DWG–1 – Glow Plug Adapter
2. DING–DWG–2 – Glow Plug Adapter Sealing Ring
4. DING–DWG–4 – Threaded Injector Needle
5. DING–DWG–5 – Injector Spring Retention Locknuts
6. DING–DWG–6 – Vespel Needle Tip