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Department of Mechanical Engineering

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Date April 11, 2003

DE-6 (2/88)
The method of fueling diesel engines with high-pressure direct injection (HPDI) of natural gas reduces regulated emissions while maintaining diesel-cycle efficiency. The relative injection timing between the pilot and natural gas, the absolute injection timing, and the injection pressure were varied in operation of a modified heavy-duty single-cylinder diesel engine. The emissions studied included nitrogen oxides (NOx), carbon monoxide (CO), particulate matter (PM) and total hydrocarbons (THC). The engine was operated at three speeds and three equivalence ratios. The effects of changing exhaust pressure and boost pressure were also examined.

Exhaust back pressure significantly affects all emissions, but not gross efficiency. For consistent effect on emissions, an absolute back pressure of 150 kPa was selected for the standard test procedure.

The relative injection timing (RIT) significantly affects most emissions, but does not appear to affect PM or specific fuel consumption. With respect to RIT, burn duration correlates well with emissions. All pollutant emissions are at a minimum when the RIT is set to 1.8 ms for the conditions tested.

The timing of the 50% cumulative heat release (HR50) is a good variable for relating NOx and efficiency to injection timing. The reduction of specific NOx emissions as a function of HR50 with retarded injection is approximately independent of equivalence ratio. Specific fuel consumption as a function HR50 is nearly independent of load and speed; the best fuel consumption occurs when HR50 is approximately 5° after top dead center. The effects of retarding absolute timing on CO depend upon the equivalence ratio. Total hydrocarbon emissions increase with excessively retarded absolute timing. The effects of timing on PM depend on speed and equivalence ratio.

The relationship between NOx and efficiency and HR50 does not change with injection pressure. Increasing injection pressure appears to increase mixing, which generally improves CO emissions at high equivalence ratio, but worsens THC and CO emissions at a low equivalence ratio, late timing. Moderately changing engine air flow-rates with a constant equivalence ratio does not significantly affect emissions.
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LIST OF SYMBOLS AND ABBREVIATIONS

SYMBOLS

ϕ Equivalence Ratio
A/F Air to Fuel Ratio (mass)

ABBREVIATIONS

BD Burn Duration
°c.a. Degrees Crank Angle
CI Compression Ignition
CO Carbon Monoxide
CO2 Carbon Dioxide
COV Coefficient of Variation
DAQ Data Acquisition System
EOC End of Combustion
EOI End of Injection
EVC, EVO Exhaust Valve Closing, Opening,
GPW Gas Pulse Width
GSOI Gas Start of Injection
HPDI High Pressure Direct Injection
HRR Apparent Heat Release Rate
HR50 50% Heat Release (c.a.)
IMEP	Indicated Mean Effective Pressure
ISFC	Indicated Specific Fuel Consumption
IVC, IVO	Intake Valve Closing, Opening
MBT	Timing For Maximum Brake Torque
NOx	Nitrogen Oxides
OEM	Original Equipment Manufacturer
P_{inj}	Injection Pressure
P_{cyl}	Cylinder Pressure
PM	Particulate Matter
PPW	Diesel Pilot Pulse Width
PSOI	Pilot Start of Injection
RIT	Relative Injection Timing
SCRE	Single Cylinder Research Engine
SOC	Start of Combustion
SOI	Start of Injection
TDC, ATDC, BTDC - top dead center crank position, After TDC, Before TDC
TEOM	Tapered Element Oscillating Microbalance
THC	Total Hydrocarbons
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1. INTRODUCTION

1.1 Preliminary Remarks

The diesel engine is known for its high efficiency, high torque capabilities, and durability. For these qualities, diesel engines have become the most prevalent engine type for medium and heavy-duty applications including most commercial ground transportation such as tractor trucks and locomotives. A recent EPA[1] air pollution study in the United States however, found that diesel engines contribute 33% of national nitrogen oxides (NO\textsubscript{x}) emissions, and 70% of mobile sub-2.5\textmu m particulate matter (PM) emissions. Studies have shown that diesel particulate emissions are toxic and NOx emissions have adverse health effects in addition to creating photochemical smog and acid rain[1,2]. These emissions are of considerable concern in urban areas with high concentration of traffic.

Globally, governments are recognizing the health risks associated with diesel emissions and legislating severe reductions from current diesel emission levels[3]. Methods are under development for reducing these emissions include exhaust gas recirculation, exhaust after-treatment, fuel additives, and fuel substitution. Westport Innovations has developed a fueling system that employs pilot-ignited, late cycle, high-pressure direct injection (HPDI\textsuperscript{TM}) of natural gas. It has been shown that Westport's natural gas HPDI\textsuperscript{TM} system reduces NO\textsubscript{x}, PM, and CO\textsubscript{2} emissions while preserving the thermal efficiency of the diesel engine [4,5,6,7]. The emission reductions in a US FTP cycle test were 45%, 85%, and 71% for NOx, nmHC, and PM respectively as compared to 1998 EPA emission requirements[7]. These emissions have been shown to be affected by altering injection parameters such as injection pressure and timing[6,7,8].

1.2 Motivation for Research

As diesel engine emissions are known to be detrimental to health and the environment, it is imperative that engine injection parameters are tuned to minimize emissions. Engine injection parameters that are adjustable without modification are pilot injection timing, gas injection timing, and injection pressure. It is well-known that with injection timing, the trade-off for NOx reduction is decreased efficiency; however, the effects of the timing between the pilot and natural gas are not well understood. Changing the injection pressure will affect the rate of fuel injection and turbulent mixing in the cylinder and, as the combustion is mostly mixing-limited, this is
expected to influence the engine performance and emissions. Many of the previous HPDI™ studies were conducted with 2-stroke diesel engines and proof-of-concept prototype injectors. The capabilities of new HPDI™ injectors have overcome the limitations of the earlier prototype injectors, where the injector used for this study uses less pilot and provides more control of injection. Supercharging has been shown to improve emissions in diesel engines. However, supercharging CI engines fuelled with directly injected natural gas has been insufficiently explored to date. The general focus of this study is to experimentally improve the fundamental understanding of changing injection parameters on engine performance.

1.3 Methodology

The measurements of performance and emissions were conducted with UBC’s single cylinder research engine (SCRE) fueled with pilot ignited HPDI™ of natural gas. The engine is a 4-stroke, modified Cummins ISX heavy-duty diesel engine, configured to run with Westport HPDI™ technology, and commissioned to serve as a flexible platform to perform fundamental studies[9]. Using a single-cylinder engine provides a better opportunity than a multi-cylinder engine to study the effects of changing injection parameters. A single injector simplifies control, and avoids the extra maintenance costs and experimental noise of multiple injectors. The facility is equipped with auxiliary systems for independent control of air supply, exhaust back-pressure, and fuel pressure. The engine system includes instrumentation of all flow rates, temperatures, emissions, and high-speed in-cylinder pressures. This experimental system provides more control over operating parameters than a standard engine with the advantage that controlled experiments can be used to fundamentally investigate the effects of injection parameters on emissions.

1.4 Thesis Overview

The combustion mechanism for pilot-ignited HPDI engines is explained in chapter 2. Chapter 2 also outlines pollutant formation and summarizes previous research, which leads into the specific objectives of this study. The experimental apparatus and experimental methodology are explained in chapter 3. The effects of exhaust back-pressure are established in chapter 4 to support a reliable testing procedure. The experimental results of changing injection parameters are presented in the order of absolute injection timing, relative injection timing, and, finally, injection pressure in consecutive chapters. Conclusions and recommendations are offered in chapter 8. The emissions measured include NOₓ, PM, carbon monoxide and total hydrocarbon
emissions. The performance measures are in terms of efficiency and qualitative observations of heat release data.
2. BACKGROUND AND OBJECTIVES

This chapter provides background knowledge regarding high-pressure direct injection (HPDI) of natural gas in compression ignition engines. The differences between natural gas and diesel fuels are highlighted and the use of pilot-ignition for natural gas is introduced. The combustion mechanics of the HPDI process are subsequently considered. The formation of major pollutant emissions from compression ignition (CI) engines are characterized. A brief literature review is discussed with respect to parametric studies that have been conducted with HPDI. Finally, the research objectives of this study are explained.

2.1 Methane and Diesel

The natural gas used in these experiments is composed of 96% methane, 2% ethane, 1% nitrogen, 0.5% propane, and smaller amounts of higher hydrocarbons\[10\]; therefore the properties of methane dominate combustion. Methane has a lower adiabatic flame temperature than diesel; the lower combustion temperatures reduce NOx formation\[5\]. The stoichiometric air-to-fuel (A/F) ratio of this natural gas is 16.8\[10\] and the stoichiometric ratio for the \#2 diesel fuel is approximately 14.4\[12\]. However, the natural gas has a lower heating value of approximately 49.1 MJ/kg\[10\], which is higher than the heating value for diesel at approximately 42.8 MJ/kg\[12\]. This means that when using the equivalent of 1 g of diesel energy content, natural gas requires only 14.6 g of air, where diesel requires 14.4 g of air. As a result, the breathing requirements of a diesel engine fueled with natural gas are very similar. However, methane will auto-ignite within an engine timescale of 2ms only when it is at temperatures of at least 1035 K\[13\], as compared to 650K for diesel\[14\]. The maximum in-cylinder temperature for a 19:1 compression ratio engine reaches approximately 900 K, which is why diesel pilot ignition can be used for natural gas HPDI\textsuperscript{TM}. The diesel pilot will hereafter be referred to as ‘pilot’.

2.2 Combustion of Pilot-Ignited Direct Injection of Natural Gas

Dumitrescu \[8\] describes the HPDI combustion mechanism as being the same as the three phases of diesel combustion: premixed, mixing-controlled and late combustion. As shown in Figure 2-1, this is an insufficient description for HPDI combustion as the pilot ignition and subsequent delay before gas injection are a notable events in the combustion process. It will be shown that the relative timing of these events is a significant factor in HPDI combustion. These
extra processes essentially form 5 major components of the pilot-ignited HPDI combustion process. The first (ab) and second stages (bc) of the diesel pilot combustion are identical to normal diesel combustion of ignition delay and premixed combustion phase as described by Heywood [14]. The small quantity of diesel burns rapidly and likely proceeds directly into a late combustion phase (c) and mixing of burned products. An ignition delay of the natural gas occurs as it mixes sufficiently with the air and burned pilot gases, after which the mixture ignites (d). The ignition delay of the natural gas is much shorter than the diesel auto-ignition delay and is not noted in the illustration. Ignition of natural gas transpires via local interaction of high temperatures and radicals resulting from the diesel combustion. From this second ignition, the HPDI combustion process mimics diesel process, progressing from pre-mixed combustion (de) to mixing-controlled combustion (ef) to late burning combustion (fg) phases. The late burning phases occur when partially burned combustion products are relatively slowly oxidized as they mix with oxygen.

![Figure 2-1 Typical HDPI heat release rate diagram. 'P' and 'G' denote the pilot and natural gas fuels. 'SOI' and 'EOI' refer to 'start' and 'end' of injection.](image)

2.3 Pollution Formation

The formation mechanisms of individual emission species in compression ignition engines are essential to describing the effects of changing various injection parameters and operating conditions.
2.3.1 Nitrogen Oxides

Nitrogen oxides (NOx) are one of the contributors, along with SO₂, to acid rain[2]. Locally, they are toxic compounds and can also photochemically react to form highly toxic smog. As detailed in Warnatz et al. [15], the four accepted modes of NO formation in CI engines, some of which oxidizes to NO₂. The first and most dominant production mode is a thermal mechanism where an equilibrium system relationships favors NO at high temperatures above 1700K. These reactions are kinetically limited in engines by the short time spent at high temperature conditions. The radicals within the flame provide another mechanism for NO formation, known as the 'prompt' or Fenimore mechanism. The prompt mechanism is less temperature sensitive than the thermal mechanism and involves CH radicals, which occur only in the flame itself. The N₂O mechanism for NO formation is also an equilibrium relationship, similar to the thermal mechanism. However, the limiting chemistry is a third body reaction and as such is highly pressure sensitive, but as the activation energy requirements are lower than thermal mechanism, this progresses NOx production at lower temperatures than via a non-thermal mechanism. The least dominant mechanism for NO production in CI engines is fuel-borne nitrogen, which is chemically bound in the fuel and is released in combustion reactions. This mode is the least significant in HPDI fueling as the small amount of nitrogen in the diesel is a negligible component of the total fuel and the nitrogen content in natural gas is not chemically bound, but a molecular constituent[9]. The amount to which each mechanism contributes to total NOx production is dependent upon in-cylinder conditions such as temperature and engine speed. Not all NO created is emitted in the exhaust as some NO decomposes after peak cylinder temperature via a reverse of the thermal mechanism [16].

2.3.2 Total Hydrocarbons (THC)

There are many different hydrocarbons that form in CI engine combustion, some of which are nearly inert, others which react readily to form smog, and others that are carcinogens[14]. Most of the hydrocarbons are from fuel that does not completely. Essentially two major mechanisms are responsible for fuel escaping combustion: over-lean or over-rich at time of flame propagation[14]. In either case, the fuel residuals will oxidize throughout the expansion stroke. Over rich and over-lean conditions can both occur in the combustion zone, however over-lean is generally the most prevalent in CI engines as the combustion is mostly non-premixed and in overall lean conditions[14]. Other possible sources of unburned hydrocarbons include wall
quenching of the flame, and also fuel escaping the injector. Wall temperature can significantly affect THC concentration, with increased temperatures promoting more oxidation. Unburned hydrocarbons are most prevalent in CI engines at idle, over fueling (acceleration), and at retarded timings with high cyclic variability.[14]

2.3.3 Carbon Monoxide

The health risks associated with carbon monoxide (CO) inhalation arise from the preferred attraction of CO to hemoglobin over oxygen, which can cause asphyxiation. The dominant factor of CO production is equivalence ratio (\(\phi\), defined as the ratio between actual air mass and stoichiometric air mass) regardless of which fuel is burned. As equivalence ratio is increased, CO production is nominal until \(\phi\) approaches unity and then dramatically increases beyond this point[14]. As such, CO production in compression ignition engines is primarily produced in the rich core of the jet as fuel is injected[14]. It can also be formed under ultra-lean conditions when the CO oxidation is limited by low temperatures[15].

2.3.4 Particulate Matter

Soot, or particulate matter (PM), is considered a respiratory health risk, particularly a lung irritant, and, recently, small particles have also been associated with increased mortality rates[2,17]. Soot is different from unburned hydrocarbons in that it precipitates once exhausted and has a very high carbon-to-hydrogen ratio. It is generally accepted that soot formation occurs in two basic steps of particle formation and particle growth, as summarized by Heywood[14]. Particle formation results from incomplete combustion as some fuel is pyrolyzed in the fuel-rich core of a fuel jet, forming soot precursors. These precursors are the nuclei of particle growth. Natural gas is less likely to form soot than diesel in a CI engine[4]. Most of the soot formed during combustion is later oxidized when mixed with adequate air during the expansion stroke, before the exhaust valve opens. Once the burnt gas is exhausted, the soot precipitates as the exhaust stream is mixed with and cooled by ambient air. Some fraction of the particulate matter is attributable to unburned lubricating oil, which is adsorbed by the soot.

2.3.5 Engine Variables that Affect Performance and Emissions

There are several parameters that can be adjusted to affect pollution formation from engines: speed, air flow-rate, overall equivalence ratio, and load of the engine, all of which influence combustion and thereby emissions. The fuel injection rate and timing also affect the efficiency and emissions from CI engines[14]. The injection timing affects the conditions in
which combustion occurs. The rate of fuel injection influences the mixing rates in the engine and thus affects performance and emissions. An unusual characteristic of this engine system is adjustable back-pressure, which will be shown to considerably affect emissions.

2.4 Previous Studies

Much research has been conducted on proof of concept for diesel pilot-ignited, late-cycle direct injection of natural gas in diesel engines[18,19,20]. This technique is the only dual fuel system which retains the overall performance and thermal efficiencies of a diesel cycle engine[19,21]. Two basic methods have been used to introduce both fuels including mixing the fuels [20, 22], and sequential injection of the pilot[6,20]. Early work using two injectors [18,20] found problems associated with this configuration at low loads, although at high loads the ratio of pilot could be reduced to 2%. Further developments of the injection system to a sequential injection through a single injector improved stability and reduced the amount of diesel[18]. There has been a drive to minimize the pilot in all cases and different configurations influence the amounts of minimum pilot for stable combustion.

Ouellette [23] investigated the fundamentals of natural gas injection and combustion interaction between diesel and natural gas with experiments and modelling. Although most of the jet behavior study was conducted with conical injection, the combustion modelling is qualitatively applicable. The simulations showed that optimal combustion occurred when the natural gas jet was delivered through burning diesel (i.e. after diesel auto-ignition), indicating that both timing and injection configuration is important. Detailed simulations by Li et al. [24] and experiments by Dumitrescu [8] confirmed that the configuration of the diesel pilot and gas jet is important for performance. This led to the conclusion that a difference in number of holes between pilot and diesel is required for consistent combustion.

A new injector, J-31, was designed by Westport to incorporate electronic control of both pilot and gas injection. The J-31 injector was studied in a six-cylinder, turbocharged Cummins ISX engine for soot investigations by Baribeau [17] and injection schemes by Harrington et al.[7]. McTaggart-Cowan [9] commissioned a single-cylinder research engine (SCRE) at UBC in the interest of studying HPDI of natural gas under more control and with exhaust gas recirculation. The UBC research engine, a modified 6-cylinder Cummins ISX, also employed the J-31 injector.
Diesel Injection Studies

Extensive timing and pressure sweeps conducted by Stumpp et al. [25] on a turbocharged, 6-cylinder diesel engine showed that NOx increased with increased diesel injection pressure for all timings. There was a NOx versus PM and efficiency trade-off with respect to changing the injection timing. The study presented data showing a lower limit to NOx reduction by retarding timing. The minimum NOx level was increased by increasing injection pressure. However, increasing pressure decreased the PM for all injection timings. As the engine was turbocharged, air-to-fuel ratios were changing with timing. Studies have shown that supercharging a diesel engine enhances mixing and reduces ignition delay, and the combination of these effects increase the relative amount of diffusive (non-premixed) burning which decreases the NOx production[26,27].

HPDI Absolute Injection Timing

Absolute timing studies of an HPDI system vary the pilot and natural gas injections together such that the relative time between them is fixed. Douville [28] conducted timing studies with an injector that mixed diesel and natural gas in a naturally aspirated, 2-stroke, Detroit Diesel 1-71 single cylinder engine. The study found that with retarding injection at low load increased THC and CO emissions and decreased NOx emissions. At high load, retarded injection caused decreases in THC, and NOx with no effect on CO emissions. Dumitrescu [8] investigated timing using a single injector with separate holes for diesel and natural gas and the same engine as Douville [28]. Again, NOx decreased with retarded injection. The study indicated that THC and CO were almost unaffected by retarding injection. In both studies, the timing was based upon the diesel pilot, which constituted 25-50%, and 15-50%, for Douville [28] and Dumitrescu [8] respectively. Baribeau [17] studied three injection timings for effect on CO and NOx emissions on a turbocharged six-cylinder Cummins ISX engine with a sequential injector. The NOx emissions were consistently reduced by retarding injection, but effects on CO emissions depended on the operating conditions.

HPDI Relative Injection Timing

The relative timing between the pilot and natural gas injections is important to combustion in an HPDI engine. Relative injection timing of diesel and natural gas in a conical injection pattern was simulated by Ouellette [23] with KIVA2 code using simplified combustion chemistry. The results showed that some delay of the natural gas after the pilot injection produced
the most favorable combustion for THC and CO emissions and the highest burn rate. Very limited experimental studies by Wakenell et. al [18] were conducted with a multi-injector system on the effects of relative injection timing. As the injector configuration was found to be important for combustion however, the results are not directly relevant to a single injector design. Relative injection timing using a single injector with fuel-specific holes has only been investigated by Dumitrescu [8] at 1200 rpm. Two relative injection timings were attained by changing injector pre-load springs and the change in relative timing was inferred from controller information. The shorter RIT was the minimum possible relative timing for stable operation of the engine. The estimated relative times between the gas and pilot start of injections were 3.7 and 5.9 °c.a.. The shorter RIT was found to adversely affect efficiency and all emissions under all engine conditions tested. Limitations of the injector were acknowledged and further study with more relative injection flexibility was suggested.

**HPDI Injection Pressure**

The effects of injection pressure on engine emissions and performance were also studied by Dumitrescu [8] and Douville [28]. Both studies were conducted with the same single-cylinder engine, operating at one speed. Both studies found increasing injection pressure increased NOx emissions. No consideration however, was given for combustion occurring earlier due to increased injection rate. The study by Douville [28] showed little change in efficiency with varying injection pressure between 100 and 140 bar except at high load, where an intermediate pressure (120 bar) gave best thermal efficiency. Dumitrescu [8] found similar results with a sequential fuel HPDI injector when pressure was varied from 100 to 160 bar, where the optimal pressure was found to be 130 bar. Baribeau [17] studied injection pressure with a variable pressure scheme set according to operating conditions, which varied between 14 and 25 MPa. The results showed that the effect of injection pressure on CO emissions was dependent upon engine conditions. Harrington et al. [7] remarked that a variable injection pressure/timing scheme may result in the best possible emissions across different operating conditions. The study also noted that high injection pressure may be required at high speed operation to ensure combustion does not extend too late in the cycle.

**Particulate Matter**

An injection pressure and timing study of PM was conducted by Baribeau [17]. The test points were taken from the AVL 8-mode test cycle. The results of timing were inconclusive as
Timing had different effects on PM at different test points. Increasing injection pressure reduced PM at low loads, but did not affect PM at high loads. Preliminary PM studies were conducted by Brakel [30] on the SCRE engine that verified the validity of TEOM measurements against gravimetric samples. The investigation also included CO emissions and a brief study of back-pressure influence where increasing back-pressure was found to affect both PM and CO emissions. The CO and PM emissions were found to correlate only at some operating conditions. No other literature was found on PM from direct injection of natural gas in CI engine.

A preliminary study of back-pressure on the SCRE by McTaggart-Cowan [9] found no measurable effect of back-pressure on indicated power or NOx emissions. There was no other literature regarding details on effects of back-pressure on emissions or engine operation.

Summary

The effects of back-pressure warrant further investigation as it is crucial to establishing a reliable test procedure and valid comparison with turbocharged engines. This review of literature indicates that a comprehensive timing/pressure study is required to determine the actual effects of increasing injection pressure, rather than indirect effects arising from changes in injection rate. It is also apparent that PM emissions are poorly understood in natural gas HPDI engines, and as such PM emissions should be included in the timing/pressure study.

2.5 Objectives

The objectives of this investigation are summarized as follows:

- Establish the effects of back-pressure as a component of a reliable testing procedure for examining injection parameters.
- Study the effects of changing the absolute injection timing.
- Study the effects of changing the relative injection timing to determine if there is an optimum relative timing.
- Study the effects of changing injection pressure, while accounting for corresponding changes in timing of the combustion event.
- Study the effects of changing the supercharging rates.

The performance measures that will be used are the NOx, CO, THC, and PM emissions, as well as indicated specific fuel consumption. These measures will be correlated with heat release and in-cylinder pressure data for explanation of effects.
3. EXPERIMENTAL METHOD

The experiments in this study were conducted in the Department of Mechanical Engineering at the University of British Columbia using a single-cylinder modified Cummins ISX 400 diesel engine, fueled by Westport Innovations, Inc. HPDI technology[9]. Test operating parameters of the engine were computer-controlled through custom-built Westport control system. The test engine is instrumented for high-speed in-cylinder data acquisition in addition to emissions and performance measurements. Measurements were recorded using a computer controlled data acquisition system and the data was analyzed with spreadsheets and Matlab programs developed by Westport and UBC specifically for this engine.

3.1 Experimental Apparatus

The experiments were conducted on the apparatus described in the Master’s theses of McTaggart-Cowan [9] and Brakel [30]. A new component in this system is a reciprocating compressor plumbed into the air intake system. The engine is a modified Cummins ISX engine and the original specifications for the engine are listed in Table 3.1. The engine is factory-altered such that only one cylinder fires. The other five cylinders have blocked valves and holes in the pistons. The flywheel on the research engine is the largest available to maximize inertia. The air intake manifold, fuel rails and functioning piston are all from a production 6-cylinder engine.

Alterations were also performed by Westport to enable the engine to run on HPDI. This requires a custom Westport control system that replaces the Cummins electronic controls. A separate fuel rail in the ISX engine is used for the gas supply. The diesel circulates continually, while the gas rail terminates at the injectors. The dual fuel system requires custom dummy injectors for the non-firing cylinders to prevent diesel from leaking into either the cylinders or the gas rail. The operating diesel injector was replaced with a Westport HPDI J-31 injector. A conceptual injector schematic shown in Figure 3-1. The injector is orientated vertically in the cylinder and the relative position of diesel and natural gas jets. In the ISX engine, the injector is centered in the cylinder and the piston bowl is toroidal shaped. The geometric characteristics of the injector are detailed in Table 3.2.
Table 3.1: Engine Specifications

<table>
<thead>
<tr>
<th></th>
<th>Single-cylinder</th>
<th>6-cylinder</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement Volume</td>
<td>2.5 L</td>
<td>Rated Power (1800 rpm)</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>19:1</td>
<td>300 kW</td>
</tr>
<tr>
<td>Bore</td>
<td>137 mm</td>
<td>6-cylinder Rated Torque</td>
</tr>
<tr>
<td>Stroke</td>
<td>169 mm</td>
<td>(1200 rpm)</td>
</tr>
<tr>
<td>Piston Bowl (combustion</td>
<td>Toroidal Bowl</td>
<td>1966 N-m</td>
</tr>
<tr>
<td>chamber)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>IVO</td>
<td>-3° ATDC</td>
<td></td>
</tr>
<tr>
<td>IVC</td>
<td>188° ATDC</td>
<td></td>
</tr>
<tr>
<td>Connecting Rod</td>
<td>262 mm</td>
<td></td>
</tr>
<tr>
<td>EVO</td>
<td>502° ATDC</td>
<td></td>
</tr>
<tr>
<td>EVC</td>
<td>722° ATDC</td>
<td></td>
</tr>
</tbody>
</table>

Figure 3-1  Westport HPDI™ Injector Schematic
Table 3.2: J-31 Injector Geometric Dimensions

<table>
<thead>
<tr>
<th></th>
<th>Angle from Firedeck</th>
<th>Hole Diameter (mm) +/- 0.01</th>
<th>Number of Holes</th>
<th>Total Flow Area (mm$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>18°</td>
<td>0.12</td>
<td>7</td>
<td>0.079</td>
</tr>
<tr>
<td>Natural Gas</td>
<td>18°</td>
<td>0.71</td>
<td>8</td>
<td>3.2</td>
</tr>
</tbody>
</table>

3.1.1 Auxiliary Systems

To start the engine and to overcome the friction from the other 5 cylinders during low-load conditions, a Baldor ZDM 4110T-5 35kW electric motor provides supplemental torque. A map of the approximate engine operating capabilities is shown in Figure 3-2. The lower load limit that the SCRE can operate at increases with speed, and high speed, low load operation is not possible. At higher loads excess power is absorbed by an inductor-type, 150 kW General Electric TG dynamometer. The electric motor drives the engine through a toothed belt attached to the rear shaft of the dynamometer. Torque is transferred between the engine and the dynamometer through a spider coupling.

![Figure 3-2 Engine map of the single cylinder engine](image)

The intake air-flow rate is supplied independent of engine operation with a choice of two compressors: a Lysholm Technologies 1600AX supercharger screw compressor or an Ingersoll-Rand reciprocating compressor. The supercharger is driven by a speed-controlled 35 kW Baldor electric motor. The Ingersoll-Rand has the capacity to supply more air at higher pressures than
the screw compressor and an approximation of these capabilities are shown in Figure 3-2. A schematic of the air-exchange system is detailed in Figure 3-3. When the intake air is routed through the supercharger, the air passes through a filter before compression, after which it is then cooled by a water-cooled heat exchanger. When the air is compressed by the Ingersoll-Rand, the air is condensed and sent to a storage drum before reaching the test cell. Once the air reaches the test cell, the air is filtered and then the flow rate is controlled with a manual regulator. The condenser after the Ingersoll-Rand reduces the relative humidity of the intake air from ambient conditions and, as such, emissions cannot be directly compared with screw-compressor engine operation. A three-way valve isolates the compressor air systems from each other.

To shield the compressors from pulsations from the engine, a 132L surge tank is located between the three-way valve and the engine intake. The air system is also piped to optionally allow exhaust gas to recirculate into the engine intake flow, but this option was not used for this study. Exhaust gas from the engine is routed through a manifold and then passes through another 132L surge tank which isolates the back-pressure valve and EGR system from the pulsating effects of the engine. To simulate the back-pressure effects of a turbocharger, a manually controlled Bernard Electric type OA8 electric actuator is coupled to a Fisher Posi Seal type A41, 2” butterfly valve situated downstream of the surge tank.
Figure 3-3  Engine Air-Flow Schematic

The diesel supply is stored at ambient pressure before it is pressurized by a Dynex PF1001 displacement pump. A mixing valve controls the diesel temperature by mixing hot return diesel with diesel flowed through a heat exchanger and mixing valve. This ensures constant diesel fuel temperatures throughout testing. A custom dome-loaded, self venting Go regulator is used to maintain the gas rail pressure and a +50 psi differential diesel rail pressure.

The engine coolant is plumbed for both heating and cooling. Cooling is employed during engine operation and flow is channeled through an external water/coolant heat exchanger for heat dissipation. This system is regulated by an engine thermostat to maintain coolant temperature of 80°C. Before start-up, coolant is pumped through a heating loop including a 1.5 kW process heater for starting purposes and to reduce warm-up times. Two other heaters are attached to the engine; a 1.5 kW block heater and a 1.5 kW immersion oil heater. At lower speeds and loads the engine temperature is slightly lower than 80°C as not enough heat is generated by the single operating cylinder at these conditions.
3.1.2 System Controls

The experimental apparatus is controlled through a combination of manual, automatic, and computer controls. The most important controller is the Westport HPDI injection controller. This controller provides independent control of both pilot and gas injection. The parameters are set manually on a computer interface, timed by milliseconds rather than crank angle. A schematic of the different timings relative to top dead center (TDC) of the compression stroke is depicted in Figure 3-4. The pilot pulse width (PPW) is the commanded time in ms for a cavity to fill with diesel inside the injector. The pilot is then injected into the engine at the commanded pilot start of injection (PSOI), with respect to TDC in ms. The commanded relative injection timing (RIT) setting determines the time between the start of pilot injection and the start of gas injection. After the RIT, the gas pulse width (GPW) is the commanded times for the pulse of natural gas. The start of gas injection (GSOI) is calculated by adding the RIT to the PSOI. As there are hydraulic effects, the exact start of injection is slightly delayed after the command. All timings in this study are reported relative to top dead center (TDC) of the compression stroke, in terms of absolute time (ms) or crank angle (c.a.).

![Diagram of Injection Control Scheme](image)

Figure 3-4 Injection Control Scheme

The electric motor driving the engine is torque-controlled with a potentiometer connected to a Baldor series 18H vector drive. The dynamometer is speed-controlled with a Digalog 1022A PID controller. Supercharger throughput is controlled by a potentiometer connected to a Baldor variable speed drive series 15H. The exhaust back-pressure is set using a potentiometer wired directly to the butterfly-valve actuator. The inlet air temperature is regulated by setting an Omega 77000 series PID controller to the desired temperature. This controller mitigates the water flow through the intake air heat exchanger.
3.2 Instrumentation

The new components in this system include a more sensitive diesel-flow rate measurement, an intake-manifold pressure transducer, and additional heating elements for the particulate measurement system. As the critical focus of this study was emissions, much attention was given to the exhaust sampling and the emission bench. The exhaust gas was sampled just after the exhaust manifold and routed through a 7m stainless steel heated line to the emission bench containing the gas analyzers. The sampling line is heated to prevent water from condensing and particulate matter from precipitating and adsorbing on the sample line walls. The exhaust sample passes through 2 sizes of heated filters and a heated sample pump before being split. One hot stream is sent directly to a flame ionization detector, the other stream is routed through a condenser and then into the other analyzers. As such, measurements for the second stream require a conversion from dry-based emissions based on calculated water content. The gases measured and their respective analyzers are summarized in Table 3.3. The high range \(\text{CO}_2\) analyzer is used for exhaust measurements and the low range \(\text{CO}_2\) analyzer is used for PM dilution ratio measurements. The \(\text{CO}, \text{CO}_2,\) and \(\text{NO}_x\) emission measurements were measured and reported in accordance with SAE recommended practice [32].

<table>
<thead>
<tr>
<th>Gas Analyzed</th>
<th>Principle</th>
<th>Make</th>
<th>Model</th>
<th>Range</th>
<th>Uncertainty (+/-)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\text{CO} (\text{Low range})_2)</td>
<td>Non-Dispersive InfraRed</td>
<td>California</td>
<td>Model</td>
<td>0 - 2 %</td>
<td>0.04 %</td>
</tr>
<tr>
<td>(\text{CO}_2 (\text{High range}))</td>
<td>NDIR</td>
<td>Beckman</td>
<td>880</td>
<td>0 - 20 %</td>
<td>0.2 %</td>
</tr>
<tr>
<td>(\text{O}_2)</td>
<td>Paramagnetic</td>
<td>Siemens</td>
<td>Ultramat 22P</td>
<td>0 - 21 %</td>
<td>0.1 %</td>
</tr>
<tr>
<td>(\text{CO})</td>
<td>NDIR</td>
<td>Siemens</td>
<td>Ultramat 21P</td>
<td>0 - 10 000 ppm</td>
<td>20 (0 - 2000) ppm</td>
</tr>
<tr>
<td>(\text{THC})</td>
<td>Flame ionization detector</td>
<td>Ratfisch RS 55</td>
<td>0 - 1000 ppm</td>
<td>10 (0 - 2000 - FS)</td>
<td></td>
</tr>
<tr>
<td>(\text{NO}_x)</td>
<td>Chemiluminescent</td>
<td>API</td>
<td>200 AH</td>
<td>0 - 3000 ppm</td>
<td>0.5 % Reading</td>
</tr>
</tbody>
</table>

The physical operating parameters of the engine are captured by a collection of thermocouples for temperature, pressure transducers for process pressure measurements, and an eddy-current probe for speed. The intake air-flow rate is determined with either a turbine meter...
installed before the supercharger or a venturi from the Ingersoll-Rand. High-speed instrumentation include an intake manifold piezo-resistive pressure transducer and an in-cylinder, flush-mounted, piezo-electric pressure transducer, which are referenced against rotary position via an optical shaft encoder. The torque absorbed by the dynamometer is measured through a load cell. The gas-flow rate is determined with a coriolis meter and the diesel consumption is determined through a regression fit of fuel reservoir mass readings. A summary of the non-analyzer instrumentation employed is shown in Table 3.4.

<table>
<thead>
<tr>
<th>Instrument</th>
<th>Make</th>
<th>Model</th>
<th>Sensitivity</th>
<th>Range</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermocouples</td>
<td>Omega</td>
<td>type-K</td>
<td>0.7 C</td>
<td>0 - 1370 C</td>
<td>1.1 C</td>
</tr>
<tr>
<td>Process Pressure transducers</td>
<td>Energy-Kinetics</td>
<td>209</td>
<td>n/a</td>
<td>0 - 345 kPa</td>
<td>0.85 kPa</td>
</tr>
<tr>
<td>In-cylinder Pressure transducer</td>
<td>AVL</td>
<td>QC33C</td>
<td>&lt; 1.5 kPa</td>
<td>20 MPa</td>
<td>1 % IMEP</td>
</tr>
<tr>
<td>Speed (RPM)</td>
<td>Digilog</td>
<td>H25D</td>
<td>2 rpm</td>
<td>8000 rpm</td>
<td>0.125%</td>
</tr>
<tr>
<td>Incremental Optical Shaft Encoder</td>
<td>BEI</td>
<td>H25D</td>
<td>0.5 °</td>
<td>8000 rpm</td>
<td>0.25 degree</td>
</tr>
<tr>
<td>Intake manifold pressure</td>
<td>PCB</td>
<td>1501</td>
<td>0.06 kPa</td>
<td>0 - 600 kPa</td>
<td>&lt;0.9 kPa</td>
</tr>
<tr>
<td>Dynamometer Load Cell</td>
<td>Artech Industries</td>
<td>20210</td>
<td>0.1 N</td>
<td>0 - 1.1 kN</td>
<td>N/A</td>
</tr>
<tr>
<td>Diesel Mass Scale</td>
<td>Tara Systems</td>
<td>SE-Count</td>
<td>0.9 mN</td>
<td>44.5 N</td>
<td>*see section 4.2.2</td>
</tr>
<tr>
<td>Turbine Air Flowmeter</td>
<td>Superflow</td>
<td>6”</td>
<td>n/a</td>
<td>n/a</td>
<td>1%</td>
</tr>
<tr>
<td>Coriolis Gas Flow-meter</td>
<td>Micromotion</td>
<td>custom</td>
<td>n/a</td>
<td>0-15 kg/hr</td>
<td>2.46% reading</td>
</tr>
<tr>
<td>Venturi pressure transducer</td>
<td>Autotran</td>
<td></td>
<td></td>
<td></td>
<td>~10%</td>
</tr>
</tbody>
</table>

### 3.3 Data Acquisition and Analysis

A computerized data acquisition system (DAQ) captured and recorded all the instrumentation signals using a National Instruments Labview 6i and NiDaq software platforms. The Pentium III 533MHz computer employs a National Instruments 64 channel, 1.25 MS/s, 12-bit
PCI-MIO-16E-1 DAQ card that collects the analogue signals routed through a SCXI 1001 chassis. Two collection processes were employed using the DAQ, one being 45, 720° cycles (4 strokes) of high-speed acquisition of the in-cylinder and intake manifold pressures, which were referenced with respect to crank angle. The low-speed acquisition process logged all other analogue instruments at 1 Hz.

To achieve sufficient accuracy for all measurements, the low-speed data sampling period was 5 minutes for each test point. The associated high-speed data was taken at the beginning and end of the slow-speed sampling period. Due to oil temperature variations and torque fluctuations caused by the electric drive system, indicated mean effective pressure (IMEP) instead of brake torque must be used to determine engine load. The IMEP values were obtained from the pre and post-test high-speed data. If these values were within 1% of each other, only the first sample was retained. Otherwise, the IMEP values were averaged. The DAQ computer also had the capability to calculate real-time heat release based on 20-cycle averages of in-cylinder pressure data. When this option was enabled, no data could be logged. Diesel flow rate and PM emissions rate were regression-fit calculated from the ‘slow’ data. All other signals are instantaneous measurements which were checked against Chauvenet’s Criterion and then averaged. The criteria involves an assumption that the distribution is gaussian and a reading is rejected if the probability of obtaining a particular deviation from the mean is less than 0.5*n, where n is the number of samples. Details are included in Appendix E.

**Particulate Matter**

To measure particulate matter (PM), a portion of the exhaust stream is routed through a mini-dilution tunnel. The diluted exhaust is sampled at a constant flow rate with a Rupprecht & Patashnick series 1105 Tapered Element Oscillating Microbalance (TEOM). The commissioning of this instrument is detailed by Brakel [30]. The sample air is drawn through a continuously-weighed filter which measures real-time mass accumulation from which instantaneous mass rates and concentrations may be derived. The filter is a hydrophobic PFA (teflon)-coated borosilicate glass which, in conjunction with sample filter temperature of 50°C (well above ambient conditions), reduces the impact of humidity changes. The 1.4 factor difference between filter samples and the TEOM reading found by Brakel [30] appears to have been improved to 1.1 by fully heating the sample line to the TEOM instrument [31].
**Diesel Flow Rate**

The diesel fuel system is a recirculating loop which is characterized by slight fluctuations with time and as such direct measurement of the small net diesel consumption is not possible. Diesel reservoir mass measurements are taken for several minutes and then the data is fit by linear regression to determine diesel consumption. The diesel error component of the total fuel measurement is relatively insignificant when at least 100 samples is taken[29]. By using longer data sets, or at higher diesel fueling rate, the relative uncertainty is reduced. This system uses a scale with finer resolution than used by McTaggart-Cowan [9] and Brakel [30].

3.4 **Test Matrix**

The tests were divided into four subsets: exhaust back-pressure; absolute injection timing; relative injection timing between the pilot and natural gas; and injection pressure. The parameters that were varied for each subset of test is listed in Table 3.5, including the different engine conditions, called the test modes. The different engine conditions are detailed in Table 3.6. For each test mode, the air and fuel flow rates were held constant. The speeds were varied between 800 and 1600 rpm, and equivalence ratio was varied at 1200 rpm between 0.3 and 0.5. The equivalence ratios are represent most of the range of typical turbo-charged 6-cylinder operation at 1200 rpm. The maximum speed tested, 1600 rpm, is the fastest that the SCRE engine can be ran at medium load. The 23 MPa maximum injection pressure is the maximum attainable pressure for a continuous supply. All tests were conducted with a PPW of 0.65 ms which (in theory) allows the same volume of diesel to be injected every cycle regardless of speed/load. The RIT setting of 1.8 ms is a recommended Westport value used for general engine operation. The inlet air temperature at the aftercooler was set to 30°C.

Increasing the equivalence ratio for a constant air-flow rate corresponds to an increase in engine load. Holding the charge-air mass per cycle constant at different speeds provides a good comparison of performance and emissions as the in-cylinder conditions are identical between different speeds. An air-flow rate of 4 g/cycle was selected from the approximate mid-load point of 6-cylinder operation at 1200 rpm. This was extended to other speeds to negate effects due to volumetric efficiency.
Table 3.5: Test Matrix

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Exhaust Back-pressure</th>
<th>Absolute Injection Timing</th>
<th>Relative Injection Timing</th>
<th>Injection Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Modes</td>
<td>1-5</td>
<td>1-5</td>
<td>2,3</td>
<td>1-7</td>
</tr>
<tr>
<td>PPW (ms)</td>
<td>0.65</td>
<td>0.65</td>
<td>0.65</td>
<td>0.65</td>
</tr>
<tr>
<td>RIT (ms)</td>
<td>1.8</td>
<td>1.8</td>
<td>0-5.7</td>
<td>1.8</td>
</tr>
<tr>
<td>GSOI (°ATDC)</td>
<td>-5, +10</td>
<td>-10 to +10</td>
<td>-5, +5</td>
<td>-10 to +10</td>
</tr>
<tr>
<td>Injection Pressure</td>
<td>19</td>
<td>19, 23</td>
<td>19</td>
<td>19</td>
</tr>
<tr>
<td>(MPa)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Back Pressure</td>
<td>10-190</td>
<td>50</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>(kPa)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 3.6: Test Modes: Operating Conditions

<table>
<thead>
<tr>
<th>Mode Number</th>
<th>Air Flow (g/cycle)</th>
<th>Speed</th>
<th>Equivalence Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4</td>
<td>1200</td>
<td>0.3</td>
</tr>
<tr>
<td>2</td>
<td>4</td>
<td>800</td>
<td>0.4</td>
</tr>
<tr>
<td>3</td>
<td>4</td>
<td>1200</td>
<td>0.4</td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>1600</td>
<td>0.4</td>
</tr>
<tr>
<td>5</td>
<td>4</td>
<td>1200</td>
<td>0.5</td>
</tr>
<tr>
<td>6</td>
<td>3</td>
<td>1200</td>
<td>0.4</td>
</tr>
<tr>
<td>7</td>
<td>6.5</td>
<td>1200</td>
<td>0.4</td>
</tr>
</tbody>
</table>

3.5 Data Processing Calculations

Equivalence Ratio

\[ \phi = \frac{(A/F)_{stoichiometric}}{(A/F)_{actual}} \]  \hspace{1cm} (Eq. 3.1)

Where A/F is the mass-based air-to-fuel ratio

IMEP

Due to variation in oil temperature which affects brake torque, and brake loads that are sometimes negative, IMEP is used to determine engine load. The gross IMEP is used for analysis,
which excludes the work in the pumping loop (only considers compression and power strokes). IMEP is determined from:

\[ IMEP = \int p \, dV \]  

(Eq. 3.2)

Where \( p \) and \( V \) are the in-cylinder pressure and volume. In-cylinder pressure was pegged at \( 180^\circ \) using intake manifold pressure as suggested by Randolph [33]. The IMEP is calculated at discrete \( 0.5^\circ \) intervals of \( P \) and \( dV \). A second order discretization was used as follows:

\[ IMEP = \sum p_n \frac{(V_{n+1} - V_{n-1})}{2} \]  

(Eq. 3.3)

The coefficient of variation (COV) is used to determine the relative variation of cycle to cycle combustion. For example, the COV of IMEP is calculated as follows:

\[ COV_{IMEP}(\%) = \left( \frac{\sigma_{IMEP}}{IMEP} \right) \times 100 \]  

(Eq. 3.4)

Where \( \sigma_{IMEP} \) is the standard deviation of IMEP over the number of cycles of data.

Heat Release Rate

The apparent net heat release rate (HRR) of the in-cylinder gas is the difference between chemical energy released and heat transfer from the cylinder. The net heat release rate is less than the gross heat release rate due to crevice region effects and losses due to heat transfer. The HRR is inferred from the in-cylinder pressure and volume, with ideal gas assumptions, as follows [14]:

\[ HRR = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{(\gamma - 1)} \gamma \frac{dp}{d\theta} \]  

(Eq. 3.5)

Where \( \gamma \) is the specific heat ratio of the in-cylinder gas and \( d\theta \) is the change in crank angle.

A seven point, third order smoothing algorithm was applied for one iteration to the in-cylinder pressure trace as follows[39]:

\[ p_n^{i+1} = \frac{1}{21} (-2p_{n-3} + 3p_{n-2} + 6p_{n-1} + 7p_n + 6p_{n+1} + 3p_{n+2} - 2p_{n+3}) \]  

(Eq. 3.6)

Where \( i \) denotes the iteration. This technique has been shown to give consistent heat release rate[40], and a smoothed and unsmoothed heat release curve are included in Appendix F. The HRR was calculated with a second order discretization using the in-cylinder pressure data and a constant \( \gamma \) of 1.30 that approximates the value of the hot combustion gases. By integrating
the HRR over a period from -30° to +70° ATDC, the percentage of heat release, start of combustion (10% heat release), and burn duration (10-90% heat release) can also be examined.

**Fuel Equivalence**

For consistency, the mass flow rate of the natural gas is converted to an equivalent mass flow of diesel on an energy basis and added to the diesel pilot flow as follows:

\[ m_{\text{fuel}} = m_{\text{pilot}} + m_{\text{NG}} \left( \frac{LHV_{\text{CNG}}}{LHV_{\text{diesel}}} \right) \]  

(Eq. 3.7)

Where \( m \) is the mass flow and LHV is the lower heating value for the corresponding fuels.

**Efficiency**

The measure for efficiency used in this discussion is the Indicated Specific Fuel Consumption (ISFC) which essentially only considers in-cylinder combustion/heat transfer effects and neglects mechanical friction. As such, this is a good measure of combustion efficiency for comparison of different speeds. The ISFC is calculated as follows:

\[ isfc = \frac{m_{\text{fuel}}}{P_{\text{ind}}} \]  

(Eq. 3.8)

Where \( P_{\text{ind}} \) is the indicated power calculated as:

\[ P_{\text{ind}} = \frac{\text{IMEP} \cdot V_d \cdot N}{n_R} \]  

(Eq. 3.9)

Where \( V_d \) is the displacement volume of the cylinder, \( N \) is the engine speed and \( n_R \) is the number of crank revolutions per power stroke.

**Emissions**

Most emissions reported in this study are standardized against indicated power in indicated specific emissions. The term *indicated specific* refers to the mass of pollutant produced per a fixed amount of indicated energy (kW-hr), which is calculated from the in-cylinder pressure. This standardizes the results so that comparisons can be made at different speeds, loads, and also against different engines. For example, the indicated specific carbon monoxide emissions are calculated as:
As the CO and NO\textsubscript{x} exhaust sampling systems pass through a drier before the analyzers, a wet/dry correction factor must be employed as follows:

$$[x_{i,\text{corr}}] = [x_{i,\text{dry}}](1 - x_{H_2O})$$  \hspace{1cm} (Eq. 3.11)

To correct for humidity changes, the measured NO\textsubscript{x} value was divided by a correction factor, where the correction factor K is defined as\[32\]:

$$K = 1 + 7 \cdot \left(0.04 \cdot \frac{F}{A} - 0.004 \right) \cdot (\omega - 10.7) + 1.8 \cdot \left(-0.116 \cdot \frac{F}{A} + 0.005 \right) \cdot (T_{\text{intake}} - 29.4)$$  \hspace{1cm} (Eq. 3.12)

where F/A is the overall fuel/air ratio and \(\omega\) is the specific humidity of the intake air.

**Particulate Matter**

The rate of particulate matter emitted from the engine is calculated as following:

$$\dot{m}_{\text{pm, ex}} = \frac{\dot{m}_{\text{pm, teom}}}{\dot{m}_{\text{dil}}} \cdot \dot{m}_{\text{exhaust}}$$  \hspace{1cm} (Eq. 3.13)

Where \(\dot{m}_{\text{pm, teom}}\) is the measured PM accumulation rate in the TEOM and the mass flow of raw exhaust through the TEOM, \(\dot{m}_{\text{dil}}\) is given by:

$$\dot{m}_{\text{dil}} = \rho \cdot Q_{\text{teom}} \cdot \frac{1}{\text{DR}_{\text{wet}}}$$  \hspace{1cm} (Eq. 3.14)

Where \(\rho\) and \(Q_{\text{teom}}\) are the density and sample flow rate through the TEOM, and the wet dilution ratio is given by\[30\]:

$$\text{DR}_{\text{wet}} = \frac{[CO_2]_{\text{ex, dry}}(1 - [H_2O]_{\text{ex}}) - [CO_2]_{\text{dil}}}{[CO_2]_{\text{tot, dry}} \left(1 - \frac{H_2O_{\text{ex}}}{\text{DR}_{\text{dry}}}ight) - [CO_2]_{\text{dil}}}$$  \hspace{1cm} (Eq. 3.15)

Where \(ex,\text{ dil},\) and \(tot\) denote the concentrations in exhaust, dilution air, and diluted exhaust respectively\[30\]. The dry dilution ratio is given by:

$$\text{DR}_{\text{dry}} = \frac{[CO_2]_{\text{ex, dry}} - [CO_2]_{\text{dil}}}{[CO_2]_{\text{tot, dry}} - [CO_2]_{\text{dil}}}$$  \hspace{1cm} (Eq. 3.16)
3.6 Error Analysis

The absolute measurement error based on instrumentation uncertainty was calculated for reported values. For a function where the value \( R = R(x_1, x_2, ..., x_n) \) and \( w_1, w_2, ..., w_n \) are the respective uncertainties for \( x_1, x_2, ..., x_n \), then the global uncertainty for \( R \) is determined as follows:

\[
\omega_R = \left\{ \sum_{i=1}^{n} \left[ \frac{\partial R}{\partial x_i} \omega_{x_i}^2 \right] \right\}^{0.5}
\]  
(Eq. 3.17)

For example, based on equations the uncertainty in measurement of \( P_{ind} \) is derived from (Eq. 3.9) and (Eq. 3.17) as:

\[
\omega_{P_{ind}} = \frac{V_d}{n_r} \left\{ \omega_N^2 \cdot IMEP^2 + \omega_{IMEP}^2 \cdot N^2 \right\}^{0.5}
\]  
(Eq. 3.18)

The maximum measurement error for \( P_{ind} \) is calculated to +/- 1.1 kW.

3.7 Experimental Uncertainty

To determine the experimental uncertainty for this study, a repeatability test was conducted by repeating two set-points over several test days. The repeatability test was conducted at two operating conditions, 15 times over 5 test days to determine the uncertainty of the measurement due to calibration, experimental procedure, and engine variability. The engine operated for both test points at 1200 rpm, 144 kg/hr of intake air, with 50 kPag exhaust back-pressure. One repeatability point was at \( \phi \) of 0.5 with GSOI of +5°c.a., and the other repeatability point was at \( \phi \) of 0.3 and GSOI of -5°c.a. The maximum standard deviation between the two repeatability points is used for the experimental uncertainty. The error is assumed to be normally distributed and a 95% confidence interval (1.96 standard deviations) is used. The maximum measurement and experimental uncertainties of the performance measures are reported in Table 3.7. The repeatability analysis estimates the experimental uncertainty, includes random errors introduced by experimental method, but excludes systematic errors. The experimental uncertainty is more useful when comparing measurements from the same system and, as such, is used for errors in plots. The repeatability uncertainty is generally lower than the instrument uncertainty for these experiments.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Measurement Uncertainty</th>
<th>Repeatability Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>NOx (g/kW-hr)</td>
<td>1.1</td>
<td>0.54</td>
</tr>
<tr>
<td>tHC (g/kW-hr)</td>
<td>0.12</td>
<td>0.063</td>
</tr>
<tr>
<td>tHC (g/hr)</td>
<td>0.85</td>
<td>0.93</td>
</tr>
<tr>
<td>CO (g/kW-hr)</td>
<td>0.33</td>
<td>0.22</td>
</tr>
<tr>
<td>CO (g/hr)</td>
<td>2.7</td>
<td>6.1</td>
</tr>
<tr>
<td>PM (mg/kW-hr)</td>
<td>N/A</td>
<td>2.9</td>
</tr>
<tr>
<td>ISFC (g/kW-hr)</td>
<td>15.7</td>
<td>3.1</td>
</tr>
</tbody>
</table>
4. EXHAUST BACK-PRESSURE

The first set of experimental tests presented are the effects of back pressure on performance and emissions. The effects of exhaust back-pressure on the SCRE intake manifold pressure, heat release, and exhaust emissions were examined to establish a reliable testing procedure for later tests. Back-pressure testing was carried out at each speed and equivalence ratio in the test matrix. The effect of back pressure at different timing was also investigated. The sampling requirements include a minimum back pressure to provide sufficient flow to the emissions instrumentation. The objective of this portion of the study was to find a single exhaust pressure that results in a reliable, repeatable testing state.

As exhaust back-pressure is raised, the pressure of the gases in the cylinder at exhaust valve closure (EVC) is higher and therefore a greater residual exhaust mass is retained by the engine. This is similar to applying exhaust gas recirculation to the engine. Increasing the exhaust residuals increases the specific heat of the charge air, dilutes the oxygen concentration, and alters the chemical kinetics by changing the concentrations of \( H_2O \) and \( CO_2 \) species[14,34]. These changes lessen the amount of nitrogen oxides (NOx) emitted, increase particulate matter (PM) formation, as well as limit the amount of oxidation of fuel and carbon monoxide (CO)[34]. A study by Ladommatos[34] isolated chemical, thermal, and dilution effects to determine that the most significant mechanism in PM formation and NOx reduction is the dilution of oxygen concentration. The preliminary study of back pressure by McTaggart-Cowan [9] indicated little effect on NOx or indicated power output within experimental error. It will be shown that adding exhaust back-pressure can induce appreciable changes in emissions.

4.1 Non-Combustion Engine Effects

The mechanical effects of raising back pressure are examined for an operating condition of 1200 rpm, 4g/cycle air, \( \phi \) of 0.4, with a GSOI of +15\(^\circ\). How back pressure affects in-cylinder pressures, exhaust temperatures, efficiencies, and brake power is discussed. A six-cylinder version of this engine usually runs with a turbo-charger, with induced back pressures between 60 and 90% of intake manifold pressure[35]. However, with exhaust gas recirculation (EGR), exhaust back-pressures may be much higher than in a non-EGR engine. For these experiments, back pressure was varied from 10 kPag to approximately 180 kPag in small increments below 60
kPag and larger increments above 60 kPag. Here and throughout this chapter, the pressures described are gauge pressures unless otherwise indicated.

The in-cylinder pressure-volume trace for high and low back pressure is shown in Figure 4-1. While not visible, the peak cylinder pressure was 3% higher with 180 kPag back pressure than with 24 kPag back pressure. As combustion occurs well after top dead center, this likely indicates more mass in the cylinder and, as the fresh air-flow rate is the same, the additional mass must be residual exhaust gas. The indicated power output value is the same within error and, as such, gross efficiency is not affected, which concurs with results of McTaggart-Cowan [9].

Increasing back pressure however, does increase the amount of work done to pump gas out of the engine. As a result, more work is done by the engine and engine power output is negatively affected as shown in Figure 4-2. The amount of work done by the engine to pump exhaust out is dictated by the characteristics of the both intake and exhaust manifold at different speeds. It appears that the effect of back pressure on power consumption is quite similar between different speeds, except at high back pressures when more power is lost at high speed. It is expected that more power is lost at high speed, based on the same back pressure. The engine lost power because increased work is done to expel the exhaust gas. The work is absorbed by the exhaust gas, resulting in higher exhaust temperatures. As shown in Figure 4-3, the temperature increase is linear with back pressure increasing, with $R^2$ value of 0.99 for $\phi$ of 0.4. Changing back pressure also affects the volumetric efficiency of the engine, which decreases slightly with increasing back pressure as shown in Figure 4-4. Qualitatively, the engine was found to run more quietly with the initial application of back pressure.

![Pressure-volume diagram for two back-pressures 1200 rpm, 4g/cycle of air, GSOI +15°ATDC.](image)

Figure 4-1
Figure 4-2  Effect of back pressure on engine brake power for different speeds. The power difference is referenced to brake power at minimum back pressure for each speed.

Figure 4-3  Exhaust manifold temperature versus back pressure at 1200 rpm, air=4g/cycle, GSOI=-5°.
4.2 Intake Manifold Pressure

Examining the instantaneous intake manifold pressure gives some indication of how back pressure is affecting the engine. Intake airflow was kept constant at 144 kg/hr, injection settings were held constant and only back pressure valve settings were varied. The intake manifold pressure was measured approximately 45 cm from the intake valve at high speed with respect to crank angle as shown in Figure 4-5. The pressure traces indicate that there are significant sinusoidal oscillations in the intake manifold pressure occurring at the same frequency, regardless of engine back-pressure. Pulsations are expected in the intake manifold due to single-cylinder operation; however, they are exacerbated by a flexible connector near the intake which allows the manifold volume to change. The intake manifold is designed for a 6-cylinder engine and not for single-cylinder operation, and this probably adds a minor effect to the pressure oscillation. The intake-manifold pressure rose with increasing back pressure and the supercharger speed was increased to maintain a consistent airflow. This implies increased mass in the cylinder with the same net flow through the cylinder. There is a non-linear increase of manifold pressure with change in back pressure, with an increasing effect at higher back pressure.

As the back pressure is measured downstream from the exhaust manifold, past the surge tank and after several meters of piping, the actual pressure at the exhaust manifold is unknown. To compound this, there are likely fluctuations in the exhaust manifold, which are not measured.
due to high exhaust temperature: piezo-transducers are not able to withstand the constant high temperatures found in engine exhaust.

There is apparently a perturbation in the intake manifold pressure shortly after top dead center (TDC) between the intake and exhaust strokes, and another at approximately 140° c.a. The earlier perturbation is larger and is magnified in Figure 4-6 and it is presumed that the second perturbation is a reflection of the first. The perturbation is apparent in all cases except 55 kPag back pressure at approximately 20° c.a. after TDC. At back pressures above 55 kPag, the pulsation increases the intake manifold pressure at +20° c.a.. A possible explanation for the positive pulsation in the intake manifold is back flow from the cylinder to the intake manifold transducer. With 15 kPag of back pressure, the perturbation is a reduction of intake manifold pressure at the same point, which may correspond to a surge of air into the cylinder from the intake manifold. The perturbations suggest a rapid air exchange between the cylinder and intake manifold at intake valve opening (IVO). As shown in Figure 4-1, there is relative pressure difference between IVO and the intake stroke (approximately intake manifold pressure) for high and low back pressures. This pressure differential likely causes a pulse of mass transfer of air, or pressure wave, as the intake valve is opened. After some time (approximately 20° at 1200 rpm) this pulsation reaches the intake manifold transducer. As there is a 5° overlap between IVO and
exhaust valve closure (EVC), there may also be a direct exchange of mass between intake and exhaust manifolds.

Figure 4-6  Early stroke intake manifold pressure for 4 back pressures at 1200rpm, 4 g/cycle of air.

Intake manifold pressure is compared at different speeds in Figure 4-7. The pressure at IVO (near 0°) for 1600 rpm is considerably different from the average pressure in the intake manifold. At 800 rpm, the difference between average manifold pressure and pressure at IVO is smaller. Based on the manifold pressures at IVO for all speeds, approximating a turbocharger would induce back pressure between 40 and 70 kPag. Because the average manifold pressure is dependent on speed and the instantaneous exhaust pressure is not known, a meaningful differential measurement could not be obtained. The difference between the average manifold pressure and the pressure at IVO would induce a bias if attempting to set the back pressure based on a difference of average measurements. Therefore, all effects of back pressure are compared at gauge values instead of differential values between intake and exhaust.
4.3 Heat Release

An examination of heat release at the same baseline case of 1200 rpm, φ 0.4, GSOI -5° ATDC was conducted to determine effects on combustion rate. The heat release rates are shown for maximum and minimum back pressure in Figure 4-8 and no difference is apparent. No drastic differences were expected as the IMEP is the same for both back pressures. The implication is that differences in emissions are not due to changes in combustion rate.

Figure 4-8  Comparison of apparent heat release rate for high and low back pressure at 1200 rpm, φ=0.4, GSOI -5°, 4 g/cycle air.
4.4 Emissions

The pollutant presented are the indicated specific emissions of NOx, THC, CO and PM. There are more exhaust residuals retained in the cylinder as back pressure is increased. Three sets of data were compared to determine the effects of back pressure on emissions. The back pressure was varied at operating conditions of 1200 rpm, GSOI -5° ATDC, 4g/cycle of air, and equivalence ratios of 0.3, 0.4 and 0.5. The effect of back pressure at different timings was also investigated at 1200 rpm, with GSOI of -5 and +15°c.a., 4g/cycle of air and φ of 0.4. The relationship of back pressure to speed was investigated with tests at 800, 1200 and 1600 rpm with φ of 0.4 and GSOI of -5° ATDC. The effects of speed and timing on emissions are discussed in the next chapter.

Equivalence Ratio

As injection timing was varied at φ of 0.3, 0.4, and 0.5, it is important to first understand the effects of changing equivalence ratio. A heat release rate (HRR) diagram shown in Figure 4-9 shows the effects of increasing equivalence ratio on combustion. As φ was increased, the first 'peak' of combustion, which corresponds to premixed combustion, was not drastically affected. However, as φ was increased, the second 'peak' of combustion, which corresponds to mixing-limited combustion, increased in magnitude. This was due to the longer injection of natural gas, which increases the total injection momentum, and thereby total mixing rates. Higher mixing rates and more fuel to burn results in more and stronger mixing-limited combustion, and the premixed combustion does not change. A comparison of the corresponding in-cylinder pressure for φ of 0.3 and 0.5 is shown in Figure 4-10. The pressure was identical until later in the combustion process, where the pressure for a φ of 0.5 continues to rise. This is due to greater energy released to the cylinder contents, which contain the energy at higher pressures and temperatures.
The reported NOx emissions are adjusted by an empirical correction factor as noted in section 3.5. This correction factor is based on diesel fuel and incorporate relative humidity and the air-to-fuel ratio. As natural gas has different stoichiometry than diesel, the correction factor may cause inaccuracies. Increasing back pressure can effect a reduction in NOx production for equivalence ratios of 0.3 and 0.4 as shown in Figure 4-11. There is some scatter in the data, but a definite downward trend in NOx production is evident as back pressure is increased. However, it appears that there is no effect at an equivalence ratio of 0.5 as the NOx emissions are not affected significantly. The effect of back pressure at different timings is shown in Figure 4-12.
pressure reduces NOx emissions more for the earlier injection timing. This trend is also confirmed at 800 rpm, found in appendix G. The effects of back pressure on NOx are much smaller than the effects of timings tested.

The NOx emissions as a function of back pressure for different speeds are shown in Figure 4-13. NOx is virtually unaffected by increases in back pressure at 1600 rpm; however, a modest effect is seen at 1200 rpm and the effect is more prominent when speed is reduced to 800 rpm. The production of NOx appears to be more affected by back pressure at lower speeds. A possible cause of the reduced effect at higher speeds is due to the reduced time of the valve overlap, which would result in less exhaust residuals retained by the engine. However, noting that back pressure has little effect on low specific emissions of NOx (at $\phi=0.5$), the reduced effect with speed may simply be consistent for lower levels of specific NOx. Further investigation may truly isolate the effect of back pressure on NOx emissions with respect to speed and timing.

Figure 4-11 NOx vs. back pressure for 3 equivalence ratios at 1200 rpm, GSOI=-5°, 4 g/cycle air.
Figure 4-12  NOx emissions vs. back pressure for 2 timings, 1200 rpm, $\phi=0.4$, 4 g/cycle air.

Figure 4-13  NOx emissions vs. back pressure for 3 speeds, GSOI=−5°, $\phi=0.4$, 4 g/cycle air.
Total Hydrocarbons

Total hydrocarbon emissions are affected very differently by back pressure as shown in Figure 4-14. Until back pressure exceeds a threshold for each equivalence ratio, the unburned hydrocarbons appear relatively unaffected by back pressure. Once back pressure exceeds this critical point, the amount of unburned hydrocarbons emitted increases dramatically. The effect of different equivalence ratio is similar to NOx, where there is a more pronounced effect at lower equivalence ratios. If a similar amount of excess hydrocarbons escape combustion at each equivalence ratio, the higher exhaust temperature for higher equivalence ratios is expected to oxidize more of the hydrocarbons. As such, the greater effects of back pressure on THC emissions at lower equivalence ratios is not surprising. There also appears to be a leveling off of THC emissions at φ of 0.3, above 110 kPag back pressure. This levelling event is unexplained.

The effect of back pressure on THC emissions at different timings is shown in Figure 4-15. There is an offset between the emissions at different timings, but the slopes of THC versus back pressure appear somewhat parallel. Therefore, it is assumed the effect of back pressure on THC emissions is similar at different injection timings.

Figure 4-14 Total hydrocarbons vs. back pressure for 3 equivalence ratios at 1200 rpm, GSOI=−5°, 4 g/cycle air.
Figure 4-15  Total hydrocarbon emissions vs. back pressure for 2 timings, 1200 rpm, \( \phi=0.4 \), 4 g/cycle air.

The effects of back pressure on THC emissions at different speeds is shown in Figure 4-16. The response of THC emissions to back pressure is very similar between 800 and 1200 rpm. However, the back-pressure threshold at which THC emissions begin to increase appears to be slightly higher for 1600 rpm than for 800. The higher threshold may be due to higher surface temperatures in the cylinder, or perhaps less exhaust residuals in the engine at higher speeds. The THC emissions rise in similar fashion in response to increasing back pressure for all speeds. For all conditions tested, back pressure values below 50 kPag do not appear to significantly affect THC emissions. The increased temperatures associated with higher back pressures should increase oxidation rates, but the exhaust manifold temperatures may be too low to affect THC emissions at that point. The increases in THC may be caused by the additional exhaust residuals which affect oxidation as noted earlier. It is not obvious what is causing the increases in THC emissions.
Carbon Monoxide

The CO emissions shown in Figure 4-17 show a radically different response to back pressure as compared to NOx or THC emissions. At \( \phi \) of 0.3 and 0.4, the CO emissions are identical and show no significant change with increasing back pressure. However at \( \phi \) of 0.5, increased back pressure resulted in a reduction of CO to a level statistically the same as at lower \( \phi \). As this result was unusual, it was repeated and two sets of data are shown in Figure 4-17. Further testing that confirms this effect at low speed and high equivalence ratios is found in appendix G. This effect was also evident at some conditions tested by Brakel [30]. The higher exhaust temperatures corresponding to increased back pressure likely increase CO oxidation. The specific CO emissions appear to approach a similar value at higher back pressures, regardless of equivalence ratio.

As shown in Figure 4-18, there is no discernible effect on CO emissions of changing back pressure at the different timings tested. The effect of back pressure at different speeds on CO emissions is shown in Figure 4-19. There appears to be no effect of back pressure at 800 and 1200 rpm. At 1600 rpm, CO emissions are higher at low back pressure compared to other speeds, but decline as back pressure is increased. The specific CO emissions appear to approach a similar
value at higher back pressures, regardless of speed. It appears that back pressure will decrease CO emissions if they are above approximately 1g/kW-hr for both cases of CO reduction.

Figure 4-17 Carbon monoxide emissions vs. back pressure for 3 equivalence ratios at 1200 rpm, GSOI=-5°, 4 g/cycle air.

Figure 4-18 Carbon monoxide emissions vs. back pressure for 2 timings, 1200 rpm, φ=0.4, 4 g/cycle air.
Particulate Matter

There is no consistent effect of back pressure on PM emissions at any equivalence ratio at 1200 rpm as shown in Figure 4-20. There is also no discernible effect at equivalence ratios of 0.3 and 0.4. For φ of 0.5, it appears that back pressure decreases PM emissions with the exception of the highest back pressure tested. There is no significant effect of back pressure on PM emissions by changing the timing at 1200 rpm, φ of 0.4 as shown in Figure 4-21. The effect of changing back pressure at different speeds is shown in Figure 4-22. There does not appear to be any effect of changing back pressure on PM emissions at 800 rpm or 1200 rpm. However, PM emissions at 1600 RPM show differences at 4 back pressures. The PM emissions at 13 and 75 kPag of back pressure are higher than PM emissions at back pressures above 145 kPag. It seems that high amounts of back pressure at high speed can reduce PM emissions, although the trend is not consistent for this testing procedure.
Figure 4-20  Particulate matter emissions vs. back pressure for 3 equivalence ratios at 1200 rpm, GSOI=-5°, 4 g/cycle air.

Figure 4-21  Particulate emissions vs. back pressure for 2 timings, 1200 rpm, φ=0.4, 4 g/cycle air.
Figure 4-22 Particulate matter emissions vs. back pressure for 3 speeds, GSOI=-5°, $\phi=0.4$, 4 g/cycle air.
4.5 Summary

Increasing back pressure:

a) increases exhaust residuals retained by the engine;
b) does not affect gross power output, which concurs with McTaggart-Cowan [9];
c) decreases brake power output;
d) increases exhaust temperature;
e) decreases volumetric efficiency;
f) can decrease NOx emissions, most strongly at earlier timings, lower equivalence ratios and lower speeds, with no significant effect at 1600 rpm, $\phi=0.4$; or 1200 rpm, $\phi=0.5$;
g) increases THC's above 50 kPag, and the effect appears to be consistent for different speeds and timings; the effect is reduced at higher equivalence ratios;
h) appears to decrease CO if CO emissions are greater than 1 g/kW-hr (e.g. 1200 rpm, $\phi=0.5$ and 1600 rpm, $\phi=0.4$);
i) causes no significant effect on CO if CO emissions are less than 1 g/kW-hr for all speeds, loads, and timings tested;
j) generally does not affect PM, but appears to decrease PM at some conditions, which coincide with back pressure induced CO reductions.

The NOx results verify that back pressure can also induce no significant effect as found by McTaggart-Cowan [9].

4.5.1 Test Procedure

An objective of this chapter was to establish a back pressure for the test procedure. For emission sampling system requirements and simulation of a turbocharger, exhaust back-pressure is necessary to the system. As turbocharged engines typically run with a back pressure that can approach 90% of the intake manifold pressure, this is a logical maximum. A turbocharger would in a six-cylinder engine would cause back pressures between 40 and 70 kPag for the conditions tested. It is impossible to choose a back pressure such that all emissions will remain unaffected at all conditions. The effects of back pressure on emissions is the smallest for most conditions when back pressure is less than 50 kPag. With that rationale, the test back pressure was chosen to be 50 kPag, which is between 15 and 40 kPag lower than intake manifold pressure at IVO depending on the speed.
5. ABSOLUTE INJECTION TIMING

This chapter examines the effects on performance and emissions of changing the absolute timing of the injection. Altering the absolute injection timing was accomplished by shifting the natural gas injection while maintaining a constant relative timing between the pilot and natural gas. The injection timing was primarily retarded past timing of best efficiency to examine nitrogen oxides (NO$_x$) reduction with corresponding performance and emission effects. The effects of injection timing were studied at a constant speed with 3 equivalence ratios ($\phi$), and a constant $\phi$ across 3 speeds. Results are compared with experimental findings of Dumitrescu [8].

The experiments were conducted in two sets of tests while gas injection was changed in 5° increments. One set of tests varied timing for $\phi$ of 0.3, 0.4, and 0.5 at 1200 rpm. The other set of tests varied the timing at $\phi$ of 0.4 for 800, 1200, and 1600 rpm. All tests presented were conducted with 4 g/cycle of air, an injection pressure 19 MPa, and a setting of 1.8 ms relative timing (RIT) between pilot and gas injection. As the RIT was fixed in time, it varied in terms of crank angle with speed. As the natural gas constitutes between 93-97% of the total energy content of the fuel, the timing of the start of gas injection (GSOI) is more critical than the timing of the diesel, and the timings reported in this chapter are referenced to GSOI. The absolute timing is hereafter simply referred to as the injection timing. The criterion used for efficiency comparison is the Indicated Specific Fuel Consumption (ISFC). The emissions reported are nitrogen oxides (NO$_x$), total hydrocarbons (THCs), carbon monoxide (CO), and particulate matter (PM). Comparisons with earlier HPDI timing studies and general diesel timing studies are included for comparison of effects on emissions.

As injection is retarded past optimum efficiency a combination of effects occur, including reduced thermodynamic efficiency and lower maximum in-cylinder pressures. A reduction in maximum in-cylinder pressure corresponds to a reduction in maximum in-cylinder temperatures. The lower temperatures are due to combustion later in the expansion stroke, which allows the burned gases to expand in a greater volume. The more retarded the timing, the further the decrease in efficiency. The late combustion reduces the time between combustion and exhaust valve opening (EVO), decreasing time available for kinetics of oxidation to occur. Late combustion in diesel engines generally decreases NO$_x$, but increases unburned fuel, intermediate species such as CO and PM, and exhaust temperatures[14].
5.1 Performance

As timing of the gas injection is changed, there are changes in the combustion and power output. As shown in Figure 5-1, as timing is retarded for a constant fuelling rate, there is a decrease in engine output. This is expected as burning the fuel later in the cycle reduces the peak pressure and the expanding gas only does work to the piston through part of the stroke. The variation in output at a given injection timing is due to differences in the fuelling rate, as the instrument error for IMEP is less than 1%.

Through study of the apparent heat release rate (HRR) shown in Figure 5-2, it is observed that heat release rates increase and burn durations decrease with retarded timings. The corresponding in-cylinder pressure curves are depicted in Figure 5-3. A comparison of burn duration at different equivalence ratios and injection timings shown in Figure 5-4 confirms the general trend of shorter burn durations at later timings for all equivalence ratios. The burn duration is the timing between 10% of total heat release and 90% of total heat release. The greater pressure difference between the injector and in-cylinder at late timings may result in higher mixing rates. However, the gas injection duration is actually longer at late timings, which implies slower injection and lower mixing rates. The explanation for the increased heat release rate with shorter burn duration at retarded timings may be due to different injection rate shapes, where for retarded timings more gas is injected later and at higher injection rates.

![Figure 5-1](image)

**Figure 5-1** Comparison of IMEP for various injection timings and 3 equivalence ratios at 1200 rpm, 4 g/cycle air.
Figure 5-2  Apparent heat release rate for various injection timings at 1200 rpm, 4 g/cycle air, $\phi=0.4$.

Figure 5-3  In-cylinder pressure for various injection timings at 1200 rpm, 4 g/cycle air, $\phi=0.4$. 
Figure 5-4 Burn duration for various injection timings and 3 equivalence ratios at 1200 rpm, 4 g/cycle air.

The variation of the start of combustion is shown in Figure 5-5, where the start of combustion (SOC) is defined as 10% heat release. There is some scatter in the data, and no consistent trend is present for $\phi$ of 0.3. The prominent scatter for $\phi$ of 0.3 is probably due to the pilot comprising almost 10% of the total energy. For each equivalence ratios of 0.4 and 0.5, the variability appears to generally increase with retarded timings. The increased variation with retarded timing is likely due to combustion occurring later in the stroke and therefore at lower in-cylinder temperatures. The variation in engine output by coefficient of variation (COV) of IMEP is shown in Figure 5-6. There is no discernible effect on variation in engine output for $\phi$ of 0.4 and $\phi$ of 0.5. The large scatter and slightly higher variability for $\phi$ of 0.3 is likely due to the short gas injection duration for low fuelling rates. The COV decreased with retarded injection, which may be explained by the increasing gas injection duration with retarded injection, which allows more stable injector operation.
Figure 5-5  Standard deviation of 50% heat release for various injection timings at 1200 rpm, 144 kg/hr air and 3 equivalence ratios.

Figure 5-6  Coefficient of variation of IMEP for various injection timings at 1200 rpm, 4 g/cycle air and 3 equivalence ratios.
An examination of the HRR shown in Figure 5-7 for different speeds gives information about combustion intensity and burn duration. For each case, the engine had the same amount of air and fuel injected per cycle. With respect to intensity, the maximum rate of heat release appears to be similar for all speeds. The 800 rpm case has a burn duration of 19° (3.96 ms) which is significantly lower than the 1200 and 1600 rpm cases which have BD’s of 25° (3.47ms) and 29° (3.02 ms) respectively. The 1600 rpm case appears to burn the fastest on an absolute time basis. As the fuel is injected at a constant rate regardless of speed, the faster burn implies that the mixing rates are higher at increased engine speeds. As the burn duration is slower in terms of crank angle, the mixing rate is not directly proportional to speed. This is not surprising as the injection momentum of the natural gas injection contributes significantly to in-cylinder mixing.

![Figure 5-7](image)

**Figure 5-7** Apparent heat release rate for various injection timings and 3 speeds at φ=0.4, with 50% Heat Release= +10° ATDC. GSOI: 800 rpm = +5°; 1200 rpm = 0°; 1600 rpm = -5°.

It can also be inferred from the different injection timings depicted in Figure 5-7 that the centroids of injection and heat release will shift with respect to GSOI as speed is changed. This is because the injection rate is constant on an absolute time basis, but the injection crank duration gets longer as speed is increased. There may also be a hydraulic delay between commanded GSOI and true GSOI, which will exacerbate the shift. This means that the engine performance will change for a fixed GSOI across different speeds, as the timing of the combustion event influences engine performance. The timing of the cumulative 50% heat release (HR50) approximates the center of combustion and, as such, the HR50 was chosen as the independent
variable for efficiency comparison. The effects of injection timing on ISFC at different speeds are shown in Figure 5-8. The ISFC is plotted against both GSOI and HR50 and the trends for each speed collapse onto one another when plotted against HR50, but not for GSOI. This illustrates a relationship between the HR50 timing and efficiency. The indicated fuel consumption will be lower than the brake fuel consumption as indicated power neglects friction and auxiliary engine loads. By neglecting these parasitic losses however, indicated efficiency can provide fundamental comparisons between different engine conditions. This provides knowledge for better understanding of the pilot ignited HPDI natural gas combustion process.

Figure 5-8  Indicated specific fuel consumption for various injection timings at $\phi=0.4$, 4 g/cycle air and 3 speeds, plotted against a) GSOI and b) 50% Heat Release Crank Angle.
The most efficient performance appears to be obtained when the HR50 occurs at approximately 5° ATDC, which appears to be the local minimum for specific fuel consumption. While running at this optimum timing, 1600 rpm appears slightly more efficient than 800 rpm, but this difference is eliminated as injection is retarded. The efficiency difference at optimum timing is likely due to different heat transfer at different speeds. The peak heat transfer to the piston and cylinder walls occurs at the high peak temperatures and pressures associated with combustion near piston top dead center (TDC). The transition through this point is faster at higher speeds and there is less heat lost because less time is available for heat transfer. Increased speeds however, will introduce more mechanical friction and reduce brake efficiency. It appears that changing the speed for injection timings retarded past optimum did not affect the efficiency significantly.

As equivalence ratio is increased, the centroids of injection and combustion shift later in the cycle. For this reason, HR50 is used to compare efficiency different equivalence ratio. As shown in Figure 5-9, the indicated specific fuel consumption is statistically within error for each equivalence ratio tested at 1200 rpm. In terms of gross efficiency performance, there are no significant differences between different fuelling rates.

Figure 5-9 Comparison of ISFC for various injection timings and 3 equivalence ratios, 1200 rpm, 4 g/cycle air.
5.2 Nitrogen Oxides

This section examines the effects of injection timing on NO\textsubscript{x} emissions with particular consideration given to the timing of combustion. It is widely accepted that peak cylinder temperature is the most critical factor governing NO\textsubscript{x} production in CI engines\cite{14}. For combustion occurring after TDC, the HR50 may correspond to the timing of maximum cylinder temperature as both pressure and temperature drop after combustion as the cylinder volume expands. The timing of 50% heat release will approximate the centroid of combustion for retarded timings, whereas GSOI is merely an indication of where injection begins. Plotting NO\textsubscript{x} emissions as functions of HR50 and GSOI are compared in Figure 5-10. The specific NO\textsubscript{x} emissions appear as independent trends for each \( \phi \) plotted against GSOI, but when plotted against HR50 they collapse within error into a single trend. This indicates that HR50 is indeed a meaningful independent variable for comparing NO\textsubscript{x} emissions for an HPDI engine. For consistency, all further emissions comparisons are made using HR50 instead of GSOI as the independent variable.

For all loads tested, there was a decrease of NO\textsubscript{x} emissions with retarded timing as shown in Figure 5-10. As the thermal mechanism of NO\textsubscript{x} production dominates in most engine applications, the lower peak temperatures associated with retarded timing likely influence the reduction of NO\textsubscript{x} emissions. However, as the 50% heat release is increased beyond 15° ATDC, there are no further significant reductions in specific NO\textsubscript{x}. This agrees with the data found in Dumitrescu\cite{8}, where brake specific NO\textsubscript{x} emissions at high load showed a lower limit for retarded injection. It seems there is a limit to the extent which NO\textsubscript{x} emissions can be mitigated by retarding gas injection.

As specific NO\textsubscript{x} emissions appear to be the same regardless of equivalence ratio, this implies that changing the \( \phi \) (at least between 0.3-0.5) does not affect NO\textsubscript{x} production. The amount of NO\textsubscript{x} emitted however, is roughly proportional to the amount of fuel burned for a fixed HR50. This is true as both specific NO\textsubscript{x} emissions and specific fuel consumption do not significantly change with equivalence ratios at a constant HR50 for the conditions tested. A confirming figure is found in Appendix H. This proportionality could be due to the largely mixing-limited combustion in HPDI engines, which proportion is consistent for all timings. The relationship between NO\textsubscript{x} production and HR50 could be examined with diesel fuel to see if the
relationship holds true for diesel. The apparent collapse of NO\textsubscript{x} production at different equivalence ratios onto a single trend is unique to gross indicated specific power, which references against zero power. The trend will not collapse with brake specific power as there is a constant amount of mechanical friction and auxiliary engine loads at any given speed, although the HR50 timing beyond which no further reductions in NO\textsubscript{x} will be the same.

Figure 5-10  Nitrogen oxides emissions for various injection timings and 3 equivalence ratios at 1200 rpm, 4 g/cycle air
(a) plotted against Gas Start of Injection
(b) plotted against HR50
The NO\textsubscript{x} emissions were also observed to be dependent on speed changes. As shown in Figure 5-11, the NO\textsubscript{x} production for \( \phi \) of 0.4 is significantly higher at 800 rpm than at 1200 or 1600 rpm. This is an expected result as the kinetics of NO\textsubscript{x} formation are limited by less available time at high temperatures at faster speeds, even though in-cylinder surface temperatures are higher. Also, the mixing rates are lower at lower speeds, and as such, the products of combustion take longer to mix with cooler excess air. It also appears that NO\textsubscript{x} is reduced as HR50 is retarded to 20° ATDC at 1600 rpm. As no NO\textsubscript{x} reduction is gained at 1200 rpm beyond 15° ATDC, the HR50 limit for NO\textsubscript{x} reduction via retarded injection appears to increase with speed. The NO\textsubscript{x} emissions at 1600 rpm were biased compared to lower speeds as the intake manifold temperature rose to 31°C for the 1600 rpm case, as compared to 25°C for 800 and 1200 rpm. Increasing the intake charge air temperature will result in more NO\textsubscript{x} emitted\cite{14}. The aftercooler air outlet temperature was set to 30°C, but at lower speeds (and airflow rates), the inlet air was cooled throughout the piping. The difference between speeds may be shown to be more proportional with tests at a different aftercooler setting. As the piping is ambient temperature, it would be appropriate to reduce the aftercooler setting to approximately 25°C.

Figure 5-11  Nitrogen oxides production for various timings and 3 speeds, \( \phi=0.4, 4 \text{ g/cycle air.} \)
5.3 Total Hydrocarbons

The total hydrocarbon emissions are an indication of relatively how much fuel is escaping combustion. For comparison of THC emissions at different injection timings, both the raw THC emissions and normalized emissions are plotted in Figure 5-12. The first observation that can be made is that raw levels of THC are similar across equivalence ratios for early injection timings and decline modestly as injection is slightly delayed. The THC emissions begin to rise significantly as injection is further delayed, and this effect is more pronounced for low $\phi$ than for high $\phi$. This compares well with Dumitrescu [8] whose results indicate that retarded timings at low loads had increased CH4 and THC emissions. Hydrocarbon oxidation rates slow as temperature in the cylinder drops [14]. As the in-cylinder temperatures are higher for higher $\phi$, the correlation of oxidation with temperature may explain why unburned hydrocarbons are lower at retarded timings for higher $\phi$. The variability of combustion noted in section 5.1 may also be linked to the increased THC emissions.

The precise source of the apparent equivalent raw THC emissions at early timings is unknown, with several possibilities: a similar amount some of the fuel may escape combustion despite the length of injection, or some lubricating oil may be expelled. The possibility also exists of diesel leakage from the injector, which would be very small and likely be consistent regardless of quantity of gas injected.
Figure 5-12  Total Hydrocarbon production for various timings and 3 equivalence ratios at 1200 rpm, 4 g/cycle air
(a) non-normalized, (b) normalized

The effects of speed on THC emissions are shown in Figure 5-13. The amount of THC emitted is the same within experimental error for 1200 and 1600 rpm, however the THC emissions at 800 rpm are slightly lower. As shown in Figure 5-7, the combustion intensity is similar for between each speed. Therefore, the lower THC emissions at 800 rpm is probably due
to the increased time available for the residuals to oxidize. The effects of retarding injection on THC emissions at different speeds appears to be consistent as the increase in THC emissions with retarded injection is similar. Total hydrocarbon emissions do not collapse onto a single trend with HR50. As discussed earlier in section 5.1, plotting against HR50 actually shifts later with increasing equivalence ratio and speed. As THC emissions at lower equivalence ratios are more affected than at higher equivalence ratios, using HR50 as the independent variable instead of GSOI causes the trends between equivalence ratios to diverge. Therefore, HR50 is not a good independent variable for comparing total hydrocarbon emissions at different equivalence ratios. This also means that the THC emissions are not related to the timing of peak cylinder temperature in the way that NO\textsubscript{x} emissions relate.

![Figure 5-13](image)

**Figure 5-13** Total hydrocarbon emissions for various timings and 3 speeds at $\phi=0.4$, 4 g/cycle of air.

### 5.4 Carbon Monoxide

The CO production at different equivalence ratios is presented in the form of raw emission rates in Figure 5-14. As can be seen in Figure 5-14(a.), the absolute CO production for equivalence ratios of 0.3 and 0.4 is the same for early injection timings within experimental error. At a very late injection timing, CO production increases for $\phi$ of 0.3 and $\phi$ of 0.4. This differs slightly from the findings of Dumitrescu [8], where no increase in CO was found. This may be
due to different timings tested or a difference in engine or injector performance. The CO could be produced in local rich conditions; as a result of the variability of combustion mentioned in section 5.1; quenching of the CO oxidation process as residuals are mixed with cooler bulk air; or a combination of the three. The CO emissions at $\phi$ of 0.5 did not exhibit consistent behavior, increasing with retarding injection to approximately $12^\circ$C.A., then decreasing dramatically to $20^\circ$C.A. before rising again at extremely late injection. The injection timings corresponding to HR50 of 5 and $10^\circ$ C.A. were repeated to confirm the CO levels. The repeated points confirmed the results within experimental error. The relatively high CO emissions at $\phi$ of 0.5 around $12^\circ$C.A. are indicate poor air utilization of the fuel. Poor air utilization is likely a result of interaction of the fuel jet with the piston bowl.

The effect of injection timing on CO emissions at different speeds is shown in Figure 5-15. Carbon monoxide production is observed to be relatively consistent for different speeds and HR with the exception of 1600 rpm, HR50 of $15^\circ$C.A.. If the observed increase in CO production is simply due to limited availability of oxidation time, it would be expected to have lower CO levels at low speed, which has more time available for oxidation. The CO emissions measured at 1200 and 1600 rpm for late timing appears to be equivalent. Therefore, CO emissions appear to be more affected by equivalence ratio and timing than by speed.

As with the THC emissions, CO emissions at late timings and lower equivalence ratios are more affected by retarded timing than at higher equivalence ratios. Also similar is that HR50 as an independent variable causes CO emission trends to diverge at different equivalence ratios. Therefore, plotting against HR50 exacerbates CO and THC emissions caused by retarded timing and is not the best independent variable for comparison.
Figure 5-14  Carbon monoxide emissions for various injection timings and 3 equivalence ratios at 1200 rpm, 4 g/cycle air.
(a) non-normalized
(b) normalized
Figure 5-15  Carbon monoxide emissions for various timings and 3 speeds at $\phi=0.4$, 4 g/cycle of air.

5.4.1 Particulate Matter

The effect of changing injection timing on PM emissions at different equivalence ratios is shown in Figure 5-16. No trends are apparent within experimental error. For most timings, it appears that PM production is the same within error for all equivalence ratios. However, there is a maximum PM emitted for $\phi$ of 0.5 at HR50 of $+10^\circ$ATDC. Interestingly, this coincides with the same conditions that generate the maximum CO production. Testing at increased equivalence ratios and/or extending the sampling period may be more provide more informative regarding injection timing and particulate matter emissions.

The effect of speed on PM production is shown in Figure 5-17. It appears that more PM is generated at 1600 rpm than lower speeds. There is no statistical difference in PM production between 800 and 1200 rpm for all timings tested. The effect of timing on PM production at 1600 rpm is inconsistent, increasing with retarded timing to $15^\circ$ c.a., then decreasing at $20^\circ$ c.a. and then increasing again. When comparing the CO emissions to the PM emissions at 1600 rpm versus timing, the trends appear to somewhat correlate. This requires further investigation. The 1600 rpm, $\phi=0.4$ case is the only condition, where this engine shows a NO$_x$-PM trade-off with timing prevalent in diesel engines [14,25].
Figure 5-16  Particulate matter emissions for various timings and 3 equivalence ratios at 1200 rpm, 4g/cycle air.

Figure 5-17  Particulate matter emissions for various injection timings and 3 speeds at $\phi=0.4$, 4g/cycle air.
5.5 Summary

- By using 50% heat release (HR50) instead of GSOI injection timing, the NO\textsubscript{x} production was found to collapse to a single curve for all equivalence ratios at a constant speed and air flow rate. NO\textsubscript{x} emissions are proportional to fuel burned for a fixed HR50.
- Using 50% heat release was found to correlate efficiency well at different speeds.
- Retarding absolute injection timing:
  a) improves efficiency until HR50 reaches +5°c.a. and then decreases efficiency;
  b) decreases specific NO\textsubscript{x} emissions until 50% heat release reaches approximately +15°c.a., after which specific NO\textsubscript{x} emissions are constant. This result is consistent with the findings of Dumitrescu [8];
  c) modestly decreases THC emissions for slight retard, and then increases with further retard. Higher speeds and lower equivalence ratios more affected. This is consistent with data found in Dumitrescu [8];
  d) increases CO emissions for $\phi$ of 0.3 and 0.4 at all speeds tested; the raw CO emissions are the same at early timings;
  e) increases CO emissions for $\phi$ of 0.5 at 1200 rpm as 50% heat release reaches +12°c.a., then decreases CO to +22°c.a., then increases CO beyond +22°c.a.. This may be explained by natural gas impinging on the piston bowl;
  f) causes no significant effect on PM emissions for $\phi$ of 0.3, 1200 rpm; nor for $\phi$ of 0.4 at 800 or 1200 rpm;
  g) increases and decreases PM emissions in a manner that seems to correlate with CO emissions for $\phi$ of 0.5 at 1200 rpm;
  h) generally increases PM emissions for $\phi$ of 0.4 at 1600 rpm.
- Increasing speed:
  a) reduces absolute time of burn duration, implying increased mixing rates;
  b) improves efficiency for optimum timing, with no effect on efficiency at later timings;
  c) reduces specific NO\textsubscript{x} emissions and delays the apparent 50% heat release limit for NO\textsubscript{x} emission improvements;
d) does not significantly affect CO production or PM emissions between 800 and 1200 rpm;
e) increases THC emissions;
f) increases PM emissions at 1600 rpm as compared to 1200 rpm;
6. RELATIVE INJECTION TIMING

This chapter examines the effects on performance and emissions of changing the relative injection timing (RIT) between the diesel pilot and the natural gas. The relative timing between the injections of natural gas and diesel will dictate the amount of mixing of natural gas and air before ignition. This amount of mixing influences the proportions of pre-mixed and mixing-limited combustion, which may affect the emissions and engine efficiency. As diesel substitution with natural gas in direct injection diesel engines has proven to reduce emissions while maintaining efficiency[4,5,8], diesel quantity should be as low as possible for best emissions. Wakenel et al. [18] found there was a minimum amount of diesel pilot required to promote good ignition of the natural gas in a pilot-ignited, natural gas fueled diesel engine[18]. Optimization of the relative injection timing between the natural gas and the pilot will facilitate the minimum amount of diesel required for good ignition.

Results are compared with injection delay simulations from Ouellette [23] and experimental results from Dumitrescu [8]. For this study, an RIT of 1.8 ms is considered normal, the value at which all other studies were conducted on this engine[9,30]. Relative timings below 1.8 ms are considered ‘short’ and timings above 1.8ms are considered ‘extended’. All tests were conducted with 4g fresh air per cycle at an overall equivalence ratio (\(\phi\)) of 0.4. The effect of RIT on heat release is considered for separate cases of short and extended RIT. The effect of RIT on efficiency and emissions is compared for a set of tests where both gas and pilot injections were varied to maintain a constant 50% heat release (HR50). The influence of speed and absolute timing on the effects of RIT are also considered.

6.1 Performance

**Short Relative Injection Timing**

For this set of tests, the pilot start of injection (PSOI) was held constant at -5° ATDC and the natural gas injection timing varied. The purpose of this test is to clearly examine ignition and heat release at short RIT. By maintaining a constant pilot injection, the conditions during pilot autoignition are consistent and therefore effects of changing relative timing are independent of changes in diesel ignition. Nielsen et al. [36] investigated the effect of methane on diesel autoignition by inducting methane into the intake manifold of a diesel engine. The results indicated that methane inhibited the diesel autoignition. It was unclear however, whether the
effect was due to chemical effects or lower in-cylinder temperatures due to reduced specific heat of the charge air mass. Mtui et al.
[37], in study of an early sequential HPDI injected engine found there was no increase in autoignition delay of diesel when natural gas was injected shortly after the diesel. The diesel represented 30-50% of total energy. Much smaller amounts of pilot were used for the current study, and injection of natural gas before the diesel pilot is possible with this injector. The heat release data for pilot ignition at relative timings where natural gas was injected before and after the diesel pilot for a constant PSOI is shown in Figure 6-1. It appears that the initial change in heat release rate (-2° ATDC), which is assumed to be the diesel ignition event, is unchanged for the range of RIT tested. This agrees well with the experimental results of Mtui et al. [37], even with negative relative timings. It appears that natural gas does not affect diesel autoignition for this injector configuration.

![Figure 6-1](image)

**Figure 6-1** Comparison of apparent heat release rate for various short relative injection timings at \( \phi=0.4 \), PSOI=-5° ATDC, 1200 rpm.

As gas start of injection (GSOI) is advanced from +8° to -5°, the combustion appears to become more premixed as the maximum heat release rate increases and the burn duration decreases. As the gas is injected before the diesel ignites for GSOI of -5° and -12°, the natural gas likely mixes with more air before ignition. Of interest is the negative RIT (GSOI -12°) results in a lower apparent heat release rate than the 0 RIT case. However, as the natural gas is injected further before diesel ignition, it likely creates a mixture with leaner components, and the additional time for mixing creates a more voluminous combustible mixture. This is consistent with simulations presented by Ouellette [23], which indicated that no injection delay resulted in over-mixed methane. Also, the flame speed becomes slower with leaner mixtures[15]. Premixed flame speed is the fastest when the mixture fraction gradient is the sharpest. The gradient will be
the most sharp with some premixing and but the gradient will even out with further mixing (i.e. leaner mixture). The slower flame speed in a greater volume of combustible mixture is the most likely explanation for lower heat release rate and longer burn duration at negative relative timings. Also of interest is the shape of the heat release rate for negative RIT where the change in heat release appears rough (0-4° ATDC) before continuing into the next portion of combustion with smooth changes in heat release rate. While both fuels are present and reacting in a heterogeneous fuel mixture they are likely competing for oxygen; also, combustion kinetics may be different from a single fuel environment.

**Variable Pilot Injection Timing**

For these tests, the gas injection timing was held constant at -5° ATDC and the pilot injection timing varied in small increments between -32 and +7 °c.a. relative to gas injection. Positive crank angle indicates that the pilot was injected after the natural gas. The heat release for a range of extended RIT (PSOI -22 to -37°c.a.) is shown in Figure 6-2. The ignition of the natural gas appears to retard as RIT is extremely long. Combustion of the diesel may not be focused enough to ignite the new jet area. This is because the long RIT allows the diesel pilot to completely burn and mix with cool bulk air, which reduces ignition capability and quality. The gas jet may also travel further to reach the burning pilot, which would allow more natural gas to mix with air before igniting, resulting in more premixed combustion. The shape of the heat release rate at long relative timing also seems to change, where the ‘peak’ of combustion occurs at a later crank angle. This may be due to weak ignition of the natural gas.

![Figure 6-2 Comparison of apparent heat release rate for various extended relative injection timings at φ=0.4, 1200 rpm, GSOI=-5°.](image)
An examination of the heat release data of extended RIT shown in Figure 6-2 yields an apparent minimum threshold before the diesel pilot will ignite. When the diesel pilot is injected at -27° ATDC and at -37° ATDC, the heat release rate is negligible until approximately -20° c.a.. The minimum auto-ignition temperature of diesel is approximately 650 K [14], and according to polytropic compression relations, the in-cylinder temperatures reaches autoignition threshold at approximately -30° ATDC. After the diesel reaches 650 K and is mixed with oxygen, there is a delay while the diesel acquires enough energy to activate combustion reaction[14]. At 1200 rpm, this appears to take approximately 10 degrees.

The effects of RIT on the combustion events: start of combustion (SOC), end of combustion (EOC) and 50% heat release (HR50) are shown in Figure 6-3. The SOC is defined as the point where 10% total heat release is attained, and EOC is defined at 90% of total heat release. For these heat release comparisons only one standard deviation is used for experimental error. This error estimate is employed as engine conditions were not changed except for relative injection timing, and thus the experimental variation between relative injection tests is much smaller than the repeatability tests, where engine conditions were dramatically adjusted between samples. The SOC appears to be the most affected by RIT, where it increases as RIT is changed from 12°. The increased SOC changes more strongly for short RIT. The HR50 appears less affected by RIT than SOC, as HR50 is only shifted for RIT of 0 and 32°c.a.. The EOC does not appear to move consistently with RIT as SOC shifts. It is interesting that the EOC does not change much for very short or long relative timings even though the SOC does. This implies that the combustion rate must be higher for those timings than for a ‘normal’ relative injection timing.
Figure 6-3 Comparison of combustion events at $\phi=0.4$, GSOI=-5° ATDC, 1200 rpm at various RIT.

The burn duration, which is the difference between the SOC and EOC is another measure for examining combustion. The effect of relative injection timing on burn duration is shown in Figure 6-4. The maximum burn duration occurs with a relative injection timing between 8 and 22°c.a. This range likely corresponds to the maximum proportion of mixing-limited combustion as compared to other relative timings. As RIT is increased from this point, the burn duration declines slightly. As RIT is reduced from 8°c.a. the burn duration decreases to relative injection of -4°c.a., where pilot is injected after natural gas. This minimum burn duration is where the natural gas has mixed sufficiently to burn the fastest. When the relative injection of the pilot is delayed to -7°c.a., the burn duration appears to increase. This indicates that the combustible natural gas mixture is likely more lean for pilot injections delayed beyond approximately -4°c.a.. Simulations of 3 injection delays presented by Ouellette [23] indicated that injection delay of 0.25 ms resulted in the shortest burn duration, and 0.5 ms delay was faster than no delay. The results from the current study indicate that slightly negative injection delay results in the fastest combustion. The difference could be due to the dissimilar configurations between simulation and this injector, or possibly a difference between commanded and actual start of injection times.
Figure 6-4  Burn duration for various relative injection timings at $\phi=0.4$, GSOI=-5° ATDC, 1200 rpm.

The effect of changing relative injection timing on variability of engine output is shown in Figure 6-5. The scatter in the data is noticeable, and only negative relative injection timings show moderately higher variability. It appears that RIT does not strongly affect engine output variability, therefore the COV of indicated mean effective pressure will not be used for further comparisons.

The effect of changing relative injection timing on the variability of combustion events is shown in Figure 6-6. The two parameters chosen for the variability of combustion are the standard deviations of start of combustion and of 50% heat release. Both parameters increase with negative relative timing, and increase dramatically for relative timing above 20°. Because the correlation coefficient is 0.98, only the standard deviation 50% heat release will be used for performance comparisons at different operating conditions. The high correlation implies that the entire combustion event is varying with RIT and not simply the start of combustion. The increased variability is expected coincide with increased CO and THC emissions as the variation is an indication of combustion stability.
Figure 6-5 Coefficient of variation of IMEP for various relative injection timings at $\phi=0.4$, GSOI=-5° ATDC, 1200 rpm.

Figure 6-6 Standard deviation of combustion events for various relative injection timings at $\phi=0.4$, GSOI=-5° ATDC, 1200 rpm.
6.2 Variable Pilot and Natural Gas Injection

The tests presented in this section were accomplished by varying the injection of both fuels to maintain a constant point of 50% heat release. This was done as HR50 was previously shown in section 5.2 to be a marker for efficiency and NOx production. This provides a consistent comparison for emissions and efficiency. The influence of timing on RIT effects was examined at 1200 rpm, for of HR50 at +5° and +15° ATDC held within 0.5°. The increments for the relative timings were chosen with baseline points of -0.5, 0, 1.8 (standard), 3 ms plus two variable timings, one short and one extended. These points are illustrated in Figure 6-7 with ‘A’ and ‘B’ denoting the short and extended timings respectively. The GSOI was held constant and the HR50 was determined for RIT of 1.8 ms from real-time heat release analysis on the DAQ computer. The two relative timings where HR50 shifted +0.5° relative to that timing were arbitrarily chosen as the variable timings. These variable points were dependent upon engine conditions.

![Figure 6-7](image)

Figure 6-7 Illustration of the determination of the variable relative injection timing settings.

At 1200 rpm, the variable timings were 2° and 29° c.a. at HR50 of +5°, and 6° and 29° c.a. at HR50 of +15°. By using the same strategy at 800 rpm, HR50 of +15°, the timings were 3° and 25° c.a.. The extended variable RIT was utilized as an arbitrary limit of how long the relative timing may be before substantial deterioration of natural gas ignition. The different limits at different operating conditions suggest that operating conditions influence the effects of relative injection timing. The influence of speed on RIT effects was examined by comparing 800 rpm
and 1200 rpm at a constant HR50 of +15° ATDC. As negative relative timings caused audible knock and degradation of emissions, the minimum RIT at 800 rpm was set to 0°.

### 6.2.1 Performance

To compare performance, relative injection timing was varied to determine impact on efficiency, burn duration, and variability of 50% heat release. The indicated specific fuel consumption is constant within experimental error for all relative timings for each absolute timing at 1200 rpm as shown in Figure 6-8. As noted earlier, a slightly negative relative timing resulted in the shortest burn duration. As burn duration becomes shorter, the engine performance becomes closer to the ideal otto engine cycle. As such, it was expected that efficiency would improve with the most rapid burning. However, it appears that the change in burn duration is not significant enough to alter engine efficiency for this fuelling rate. This may be due to incomplete combustion or the change in burn duration is not large enough to affect efficiency. Despite the lack of change in efficiency for this fuelling rate, shorter burn duration has the potential to improve efficiency and should be tested at other loads and higher speeds. The results of this study differ from the findings of Dumitrescu [8], which found decreased thermal efficiency for short relative timing. It was noted by Dumitrescu [8] that the decreased thermal efficiency may have been due to injector limitations at short RIT, rather than due to the combustion effects of a short RIT.

![Figure 6-8](image_url) Efficiency for various relative injection timings at φ=0.4, 1200 rpm, with HR50 timings of +5 and +15° ATDC.
As shown in Figure 6-9, the longest burn duration at 1200 rpm occurs at a relative timing of 13°c.a. (1.8 ms) for both absolute timings. The minimum burn duration occurs at a negative relative timing of -4° for both absolute timings. The minimum burn duration is lower at HR50 of +5° than +15, presumably due to the higher cylinder pressures and temperatures which promote combustion. The results indicate that there is little influence of changing absolute injection timing on burn duration for zero RIT. It is apparent that RIT does not affect the short burn duration for retarded timings that was noted in chapter 5.

The effect of RIT on burn duration is shown for two speeds in Figure 6-10. The results are shown in absolute time format and also in crank angle format. It is apparent that burn duration does not collapse onto a single trend versus RIT for either crank angle or absolute time. The relative timing effects at short RIT are dependent on the diesel pilot autoignition delay and the distance the natural gas jet travels to reach the burning pilot. Diesel autoignition delay is strongly dependent on pressure and temperature, which are weakly affected by speed. The gas jet speed is the same, regardless of engine speed. Both of these factors are mostly time dependent phenomenon and this is apparent in Figure 6-10-a for short RIT, where the proportional decrease in burn duration appears to match on an absolute time basis. The effects at long RIT probably have more to do with mixing and cooling of the diesel pilot after combustion, where mixing is roughly proportional to speed. This is apparent in Figure 6-10b, where after the maximum burn duration for each speed, the slope of the burn duration vs. RIT seem to match. Therefore, it appears the characteristics of burn duration at short RIT are time dependent, while the burn duration characteristics at long RIT are speed dependent.
Figure 6-9  Comparison of burn duration for various relative injection timings at $\phi=0.4$, 1200 rpm, with HR50 timings of $+5^\circ$ and $+15^\circ$ ATDC.
Figure 6-10 Burn duration for various relative injection timings at two speeds, $\phi=0.4$, HR50=+15° plotted against:
(a) absolute timing, (b) crank angle

The effect of RIT on variability appears similar for both absolute timings tested at 1200 rpm as shown in Figure 6-11. Only at a negative RIT does the variation appear to be lower an earlier absolute timing. This could be due to higher in-cylinder temperatures which should
promote more consistent ignition of the natural gas. The influence of speed on combustion variability due to RIT is shown in Figure 6-12. It appears that with short RIT, the variability at 800 rpm is slightly lower than that of 1600 rpm. There is not much difference in variation at extended RIT.

![Figure 6-11](image)

**Figure 6-11** Standard deviation of 50% heat release for various relative injection timings at two absolute timings, $\phi=0.4$, 1200 rpm.

![Figure 6-12](image)

**Figure 6-12** Standard deviation of 50% heat release for various relative injection timings at two speeds, $\phi=0.4$, HR50=+15°ATDC.
6.2.2 Emissions

Nitrogen Oxides

The effects of relative injection timing on NO\textsubscript{x} emissions at 1200 rpm is shown in Figure 6-13. The minimum NO\textsubscript{x} emissions for both absolute timings occur at relative injection timing of 13° (1.8 ms). As RIT is extended from 13°, there is a slight increase in NO\textsubscript{x} emissions. As relative injection timing is shortened from 13°, there is a dramatic increase NO\textsubscript{x} emissions. The effects of RIT on NO\textsubscript{x} emissions are stronger for the HR50 of +5 as compared to the HR50 of +15. This likely has to do with the lower in-cylinder temperature at later absolute timing. It is interesting that at negative RIT the apparent heat release rate is lower for than for 0 RIT, but the NO\textsubscript{x} emissions are higher. An explanation for this is that the natural gas mixture is more voluminous before ignition at very negative relative timings. Once burned, the products of the mixture may take longer to mix with the cooler bulk air, which allows more time for NO\textsubscript{x} chemical kinetics to proceed.

The effects of RIT on NO\textsubscript{x} emissions at different speeds is shown in Figure 6-14. The minimum NO\textsubscript{x} production at 800 rpm is level between 9° and 14° relative timings. At 800 rpm, the NO\textsubscript{x} increases from a minimum of 6 g/kW-hr to 12 g/kW-hr at 0 RIT. At 1200 rpm, the NO\textsubscript{x} increases from a minimum of 4.4g/kW-hr to 8.6 g/kW-hr. It appears that the effects of relative timing on NO\textsubscript{x} emissions are greater at lower speeds. This is probably due to increased mixing rates at higher speeds.

Dumitrescu [8] found that short RIT increased NO\textsubscript{x} emissions while holding PSOI constant. Chapter 5 illustrated that shifting combustion earlier in the cycle increases NO\textsubscript{x} emissions. It is not surprising that the 3° shorter RIT in [8], which shifted the gas combustion earlier in the cycle, resulted in increased NO\textsubscript{x} emissions. The results from the current study indicate that the increased NO\textsubscript{x} found by [8] would be partly due to premixing, and partly due to shifting the combustion event.
Figure 6-13  Comparison of NO\textsubscript{x} production for various relative injection timings at two absolute timings, $\phi=0.4$, 1200 rpm.

Figure 6-14  NO\textsubscript{x} emissions for various relative injection timings at two speeds, $\phi=0.4$, HR50=$+15^\circ$ATDC.
Total Hydrocarbons and Carbon Monoxide

The results of the effects of relative injection timing on both total hydrocarbons and carbon monoxide are shown in Figure 6-15 for two absolute timings. As with NO\textsubscript{x} emissions, the minimum CO and THC emissions occur at an RIT of 13\textdegree. At HR50 of +5\textdegree, changing RIT increases both THC and CO emissions for both extended and negative relative timings. Relative timing appears to affect CO and THC emissions more strongly at HR50+15. The in-cylinder pressures and temperatures are lower for the later absolute timing, making it more difficult to ignite the natural gas mixture. In addition, there is less time for THCs and CO to oxidize at late absolute timings. The presence of more incompletely burned products at short RIT for late absolute timing indicates that combustion is less efficient and as a result, the corresponding shorter burn durations are less likely to improve engine efficiency (at late absolute timings). The more moderate levels of THC and CO emissions for short RIT at early timings provide more opportunity for the short burn duration to improve efficiency (at early absolute timings).

The simulations of Ouellette [23] found high levels of CO formation with no ignition delay. This was mostly due to impingement of the gas jet on a wall as the flame could not reach the end of the jet before the jet reached the wall. The 1200 rpm experimental results of Dumitrescu [8] indicated that higher levels of CO and THC emissions occur with a 3\textdegree RIT, as compared to a 6\textdegree RIT. The trends of CO and THC emissions in this study agree with trends at short RIT found by Dumitrescu [8] and CO results of Ouellette.

The effects of RIT on THC and CO emissions at different speeds is shown in Figure 6-16. The minimum THC and CO emissions at 800 rpm occur at the same relative timing as with NO\textsubscript{x} emissions between 9\textdegree and 14\textdegree. The effects of changing RIT on THC and CO emissions appears similar for both speeds.
Figure 6-15  Carbon monoxide and total hydrocarbon emissions for various relative injection timings at two absolute timings, $\phi=0.4, 1200$ rpm.

Figure 6-16  Carbon monoxide and total hydrocarbon emissions for various relative injection timings at two speeds, $\phi=0.4, HR50=+15^\circ$ATDC
**Particulate Matter**

There was no statistical change in PM emissions for the relative timing sweeps at each operating condition. The experimental error masked any effect of relative timing, partly because the PM emissions at this equivalence ratio are very low. To determine the effect of RIT on PM, this study should be repeated for an engine operating condition where the engine is known to produce higher levels of PM.

**Overall Emission Discussion**

To examine a perceived correlation between emissions and burn duration, the correlation coefficient between burn duration and emission species was calculated using:

\[
\rho_{x,y} = \frac{Cov(X, Y)}{\sigma_x \cdot \sigma_y}
\]

(Eq. 6.1)

Where \(\sigma\) is the standard deviation for each variable and the \(Cov\) is given by:

\[
Cov(X, Y) = \frac{1}{n} \sum_{i=1}^{n} (x_i - \mu_x)(y_i - \mu_y)
\]

(Eq. 6.2)

The correlation coefficient for each emission species, averaged over each mode tested is listed in Table 6.1. It is apparent that the maximum burn duration corresponds to minimum emissions when changing relative timing at all speeds and absolute timings tested. This likely has to do with increasing the proportion of pre-mixed combustion, which burns faster, but causing more emissions. Burn duration could be used for determining optimal RIT, without requiring lengthy emission data samples and analysis, although this should be verified. The flat optimal emission/ burn duration response to changing relative timing at 800 rpm suggest that more pilot is injected than required for good ignition. This can be inferred as an optimal minimum amount of pilot will likely have an distinct maximum burn duration versus RIT. The accuracy of the burn duration measurements can be improved by averaging more cycles of in-cylinder data. This correlation will not hold for extreme negative relative timings where the burn duration increases or for extremely long RIT where the ignition of the natural gas is very weak. The exact limits for good correlation are unknown. This correlation is only valid for a fixed operating condition.
Hypothetically, smaller quantities of pilot in the current configuration would result in a smaller window of good ignition. This is because a smaller quantity of diesel pilot will cool faster and thereby reduce the ignition capability. Higher speed or high swirl may require more amounts of pilot, as the pilot will disperse and cool faster than at low speeds. Conversely, the amount of diesel may be reduced at low rpm as it does not disperse as quickly. Also, RIT may become significant at high EGR, as EGR will increase the ignition delay[14], which will affect ignition strength due to mixing.

The minimum amount of pilot is a complicated problem involving engine conditions plus injection duration, injection rate and shape, injector configuration, and nozzle shape. The sheer number of variables involved suggest that numerical simulation studies be conducted to optimize the diesel charge before experiments. Noting that the diesel accounts for as little as 3% of the total fuel (at high load), further reduction of pilot would result in only nominal emission reductions due to fuel substitution. However, optimized pilot charge may improve natural gas ignition, where minimal pre-mixing of the natural gas should result in the lowest possible NO\textsubscript{x} formation.

The emission results indicate that optimal timing between gas and pilot injection occurs at approximately 13° and 8-14° for 1200 and 800 rpm respectively. The optimum timing for both speeds included the ‘normal’ timing of 1.8 ms. The emission response was more flat in terms of absolute time at 800 rpm than at 1200 rpm. This is likely influenced by enhanced mixing at higher speeds. The fact that optimal emissions resulted at delayed sequential injection concurs with diesel/CH\textsubscript{4} simulations by [23], which determined that an injection delay of gas after diesel reduces chance of fuel over-mixing and impingement. In those simulations, increasing the injection delay from zero decreased CO formation and reduced unburned CH\textsubscript{4}. The results of these experiments indicate that a minimum of premixed combustion, with good ignition of the natural gas, brings about the best possible emissions.

<table>
<thead>
<tr>
<th>Emission Species</th>
<th>Correlation Coefficient with Burn Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon monoxide</td>
<td>-0.97</td>
</tr>
<tr>
<td>Nitrogen oxides</td>
<td>-0.98</td>
</tr>
<tr>
<td>Total hydrocarbon</td>
<td>-0.96</td>
</tr>
</tbody>
</table>
6.3 Summary

- There is an optimum relative injection timing (or range of timings) for a minimum of all emissions at fixed operating condition, the optimum relative timing appears to be independent of speed or absolute injection timing.

- The optimum relative timing included the ‘normal’ 1.8 ms (9° and 13° for 800 and 1200 rpm respectively). The emission response was flatter at 800 rpm than at 1200 rpm on an absolute time basis.

- The efficiency is not significantly affected by relative injection timing. However, the ability to reduce burn duration provides potential for improved efficiency at early absolute timings.

- When 50% heat release is held constant, burn duration correlates well with emissions. The longest burn duration is produced by combustion most favorable for emissions.

- For short relative injection timings, an optimal delay avoids premixed combustion-i.e. it is desirable for the diesel pilot to be ignited before the NG is injected. It seems that combustion appearing more pre-mixed generates worse emissions. This event is more dependent on absolute time than by crank angle.

- Agreement with the emission trends at short relative timings with Dumitrescu [8], however the efficiency findings between studies differ. The difference in thermal efficiency found in that study was likely due to injector design.

- At extended relative injection timings, apparent heat-release data indicate that natural gas ignition by pilot is weakened by mixing of the pilot, therefore higher speeds and high swirl engines may require more diesel for good ignition of the natural gas.

- Future study should include simulation of the diesel pilot with a focus of improving ignition of the natural gas.

- The effect of relative injection timing on particulate matter emissions should be studied at conditions generating more particulate matter (higher equivalence ratios).
An examination of the heat release data of extended RIT shown in Figure 6-2 yields an apparent minimum threshold before the diesel pilot will ignite. When the diesel pilot is injected at -27° ATDC and at -37° ATDC, the heat release rate is negligible until approximately -20° c.a.. The minimum auto-ignition temperature of diesel is approximately 650 K [14], and according to polytropic compression relations, the in-cylinder temperatures reaches autoignition threshold at approximately -30° ATDC. After the diesel reaches 650 K and is mixed with oxygen, there is a delay while the diesel acquires enough energy to activate combustion reaction[14]. At 1200 rpm, this appears to take approximately 10 degrees.

The effects of RIT on the combustion events: start of combustion (SOC), end of combustion (EOC) and 50% heat release (HR50) are shown in Figure 6-3. The SOC is defined as the point where 10% total heat release is attained, and EOC is defined at 90% of total heat release. For these heat release comparisons only one standard deviation is used for experimental error. This error estimate is employed as engine conditions were not changed except for relative injection timing, and thus the experimental variation between relative injection tests is much smaller than the repeatability tests, where engine conditions were dramatically adjusted between samples. The SOC appears to be the most affected by RIT, where it increases as RIT is changed from 12°. The increased SOC changes more strongly for short RIT. The HR50 appears less affected by RIT than SOC, as HR50 is only shifted for RIT of 0 and 32° c.a.. The EOC does not appear to move consistently with RIT as SOC shifts. It is interesting that the EOC does not change much for very short or long relative timings even though the SOC does. This implies that the combustion rate must be higher for those timings than for a ‘normal’ relative injection timing.
The burn duration, which is the difference between the SOC and EOC is another measure for examining combustion. The effect of relative injection timing on burn duration is shown in Figure 6-4. The maximum burn duration occurs with a relative injection timing between 8 and 22°c.a.. This range likely corresponds to the maximum proportion of mixing-limited combustion as compared to other relative timings. As RIT is increased from this point, the burn duration declines slightly. As RIT is reduced from 8°c.a. the burn duration decreases to relative injection of -4°c.a., where pilot is injected after natural gas. This minimum burn duration is where the natural gas has mixed sufficiently to burn the fastest. When the relative injection of the pilot is delayed to -7°c.a., the burn duration appears to increase. This indicates that the combustible natural gas mixture is likely more lean for pilot injections delayed beyond approximately -4°c.a.. Simulations of 3 injection delays presented by Ouellette [23] indicated that injection delay of 0.25 ms resulted in the shortest burn duration, and 0.5 ms delay was faster than no delay. The results from the current study indicate that slightly negative injection delay results in the fastest combustion. The difference could be due to the dissimilar configurations between simulation and this injector, or possibly a difference between commanded and actual start of injection times.

Figure 6-3  Comparison of combustion events at $\phi=0.4$, GSOI=-5° ATDC, 1200 rpm at various RIT.
Figure 6-4  Burn duration for various relative injection timings at φ=0.4, GSOI=-5° ATDC, 1200 rpm.

The effect of changing relative injection timing on variability of engine output is shown in Figure 6-5. The scatter in the data is noticeable, and only negative relative injection timings show moderately higher variability. It appears that RIT does not strongly affect engine output variability, therefore the COV of indicated mean effective pressure will not be used for further comparisons.

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Figure 6-5  Coefficient of variation of IMEP for various relative injection timings at $\phi=0.4$, GSOI=$-5^\circ$ ATDC, 1200 rpm.

Figure 6-6  Standard deviation of combustion events for various relative injection timings at $\phi=0.4$, GSOI=$-5^\circ$ ATDC, 1200 rpm.
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At 1200 rpm, the variable timings were 2\textdegree and 29\textdegree c.a. at HR50 of +5\textdegree, and 6\textdegree and 29\textdegree c.a. at HR50 of +15\textdegree. By using the same strategy at 800 rpm, HR50 of +15\textdegree, the timings were 3\textdegree and 25 \textdegree c.a.. The extended variable RIT was utilized as an arbitrary limit of how long the relative timing may be before substantial deterioration of natural gas ignition. The different limits at different operating conditions suggest that operating conditions influence the effects of relative injection timing. The influence of speed on RIT effects was examined by comparing 800 rpm
and 1200 rpm at a constant HR50 of +15° ATDC. As negative relative timings caused audible knock and degradation of emissions, the minimum RIT at 800 rpm was set to 0°.

6.2.1 Performance

To compare performance, relative injection timing was varied to determine impact on efficiency, burn duration, and variability of 50% heat release. The indicated specific fuel consumption is constant within experimental error for all relative timings for each absolute timing at 1200 rpm as shown in Figure 6-8. As noted earlier, a slightly negative relative timing resulted in the shortest burn duration. As burn duration becomes shorter, the engine performance becomes closer to the ideal otto engine cycle. As such, it was expected that efficiency would improve with the most rapid burning. However, it appears that the change in burn duration is not significant enough to alter engine efficiency for this fuelling rate. This may be due to incomplete combustion or the change in burn duration is not large enough to affect efficiency. Despite the lack of change in efficiency for this fuelling rate, shorter burn duration has the potential to improve efficiency and should be tested at other loads and higher speeds. The results of this study differ from the findings of Dumitrescu [8], which found decreased thermal efficiency for short relative timing. It was noted by Dumitrescu [8] that the decreased thermal efficiency may have been due to injector limitations at short RIT, rather than due to the combustion effects of a short RIT.

![Figure 6-8 Efficiency for various relative injection timings at \( \phi = 0.4 \), 1200 rpm, with HR50 timings of +5 and +15° ATDC.](image)
As shown in Figure 6-9, the longest burn duration at 1200 rpm occurs at a relative timing of 13°c.a. (1.8 ms) for both absolute timings. The minimum burn duration occurs at a negative relative timing of -4° for both absolute timings. The minimum burn duration is lower at HR50 of +5° than +15, presumably due to the higher cylinder pressures and temperatures which promote combustion. The results indicate that there is little influence of changing absolute injection timing on burn duration for zero RIT. It is apparent that RIT does not affect the short burn duration for retarded timings that was noted in chapter 5.

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Figure 6-9  Comparison of burn duration for various relative injection timings at $\phi=0.4$, 1200 rpm, with HR50 timings of +5 and +15° ATDC.
Figure 6-10  Burn duration for various relative injection timings at two speeds, 
$\phi=0.4$, HR50=$+15^\circ$ plotted against: 
(a) absolute timing, (b) crank angle

The effect of RIT on variability appears similar for both absolute timings tested at 1200 rpm as shown in Figure 6-11. Only at a negative RIT does the variation appear to be lower an earlier absolute timing. This could be due to higher in-cylinder temperatures which should
promote more consistent ignition of the natural gas. The influence of speed on combustion variability due to RIT is shown in Figure 6-12. It appears that with short RIT, the variability at 800 rpm is slightly lower than that of 1600 rpm. There is not much difference in variation at extended RIT.

Figure 6-11 Standard deviation of 50% heat release for various relative injection timings at two absolute timings, $\phi=0.4$, 1200 rpm.

Figure 6-12 Standard deviation of 50% heat release for various relative injection timings at two speeds, $\phi=0.4$, HR50=$+15^\circ$ATDC.
6.2.2 Emissions

Nitrogen Oxides

The effects of relative injection timing on NO\textsubscript{x} emissions at 1200 rpm is shown in Figure 6-13. The minimum NO\textsubscript{x} emissions for both absolute timings occur at relative injection timing of 13° (1.8 ms). As RIT is extended from 13°, there is a slight increase in NO\textsubscript{x} emissions. As relative injection timing is shortened from 13°, there is a dramatic increase NO\textsubscript{x} emissions. The effects of RIT on NO\textsubscript{x} emissions are stronger for the HR50 of +5 as compared to the HR50 of +15. This likely has to do with the lower in-cylinder temperature at later absolute timing. It is interesting that at negative RIT the apparent heat release rate is lower for than for 0 RIT, but the NO\textsubscript{x} emissions are higher. An explanation for this is that the natural gas mixture is more voluminous before ignition at very negative relative timings. Once burned, the products of the mixture may take longer to mix with the cooler bulk air, which allows more time for NO\textsubscript{x} chemical kinetics to proceed.

The effects of RIT on NO\textsubscript{x} emissions at different speeds is shown in Figure 6-14. The minimum NO\textsubscript{x} production at 800 rpm is level between 9° and 14° relative timings. At 800 rpm, the NO\textsubscript{x} increases from a minimum of 6 g/kW-hr to 12 g/kW-hr at 0 RIT. At 1200 rpm, the NO\textsubscript{x} increases from a minimum of 4.4g/kW-hr to 8.6 g/kW-hr. It appears that the effects of relative timing on NO\textsubscript{x} emissions are greater at lower speeds. This is probably due to increased mixing rates at higher speeds.

Dumitrescu [8] found that short RIT increased NO\textsubscript{x} emissions while holding PSOI constant. Chapter 5 illustrated that shifting combustion earlier in the cycle increases NO\textsubscript{x} emissions. It is not surprising that the 3° shorter RIT in [8], which shifted the gas combustion earlier in the cycle, resulted in increased NO\textsubscript{x} emissions. The results from the current study indicate that the increased NO\textsubscript{x} found by [8] would be partly due to premixing, and partly due to shifting the combustion event.
Figure 6-13  Comparison of NO\textsubscript{x} production for various relative injection timings at two absolute timings, $\phi=0.4$, 1200 rpm.

Figure 6-14  NO\textsubscript{x} emissions for various relative injection timings at two speeds, $\phi=0.4$, HR50=+15\textdegree ATDC.
Total Hydrocarbons and Carbon Monoxide

The results of the effects of relative injection timing on both total hydrocarbons and carbon monoxide are shown in Figure 6-15 for two absolute timings. As with NO\textsubscript{x} emissions, the minimum CO and THC emissions occur at an RIT of 13\textdegree. At HR50 of +5\textdegree, changing RIT increases both THC and CO emissions for both extended and negative relative timings. Relative timing appears to affect CO and THC emissions more strongly at HR50+15. The in-cylinder pressures and temperatures are lower for the later absolute timing, making it more difficult to ignite the natural gas mixture. In addition, there is less time for THC and CO to oxidize at late absolute timings. The presence of more incompletely burned products at short RIT for later absolute timing indicates that combustion is less efficient and as a result, the corresponding shorter burn durations are less likely to improve engine efficiency (at late absolute timings). The more moderate levels of THC and CO emissions for short RIT at early timings provide more opportunity for the short burn duration to improve efficiency (at early absolute timings).

The simulations of Ouellette [23] found high levels of CO formation with no ignition delay. This was mostly due to impingement of the gas jet on a wall as the flame could not reach the end of the jet before the jet reached the wall. The 1200 rpm experimental results of Dumitrescu [8] indicated that higher levels of CO and THC emissions occur with a 3\textdegree RIT, as compared to a 6\textdegree RIT. The trends of CO and THC emissions in this study agree with trends at short RIT found by Dumitrescu [8] and CO results of Ouellette.

The effects of RIT on THC and CO emissions at different speeds is shown in Figure 6-16. The minimum THC and CO emissions at 800 rpm occur at the same relative timing as with NO\textsubscript{x} emissions between 9\textdegree and 14\textdegree. The effects of changing RIT on THC and CO emissions appears similar for both speeds.
Figure 6-15 Carbon monoxide and total hydrocarbon emissions for various relative injection timings at two absolute timings, $\phi=0.4$, 1200 rpm.

Figure 6-16 Carbon monoxide and total hydrocarbon emissions for various relative injection timings at two speeds, $\phi=0.4$, HR50=+15°ATDC
Particulate Matter

There was no statistical change in PM emissions for the relative timing sweeps at each operating condition. The experimental error masked any effect of relative timing, partly because the PM emissions at this equivalence ratio are very low. To determine the effect of RIT on PM, this study should be repeated for an engine operating condition where the engine is known to produce higher levels of PM.

Overall Emission Discussion

To examine a perceived correlation between emissions and burn duration, the correlation coefficient between burn duration and emission species was calculated using:

$$\rho_{x,y} = \frac{Cov(X,Y)}{\sigma_x \cdot \sigma_y}$$ (Eq. 6.1)

Where $\sigma$ is the standard deviation for each variable and the $Cov$ is given by:

$$Cov(X, Y) = \frac{1}{n} \sum_{i=1}^{n} (x_i - \mu_x)(y_i - \mu_y)$$ (Eq. 6.2)

The correlation coefficient for each emission species, averaged over each mode tested is listed in Table 6.1. It is apparent that the maximum burn duration corresponds to minimum emissions when changing relative timing at all speeds and absolute timings tested. This likely has to do with increasing the proportion of pre-mixed combustion, which burns faster, but causing more emissions. Burn duration could be used for determining optimal RIT, without requiring lengthy emission data samples and analysis, although this should be verified. The flat optimal emission/burn duration response to changing relative timing at 800 rpm suggest that more pilot is injected than required for good ignition. This can be inferred as an optimal minimum amount of pilot will likely have an distinct maximum burn duration versus RIT. The accuracy of the burn duration measurements can be improved by averaging more cycles of in-cylinder data. This correlation will not hold for extreme negative relative timings where the burn duration increases or for extremely long RIT where the ignition of the natural gas is very weak. The exact limits for good correlation are unknown. This correlation is only valid for a fixed operating condition.
Hypothetically, smaller quantities of pilot in the current configuration would result in a smaller window of good ignition. This is because a smaller quantity of diesel pilot will cool faster and thereby reduce the ignition capability. Higher speed or high swirl may require more amounts of pilot, as the pilot will disperse and cool faster than at low speeds. Conversely, the amount of diesel may be reduced at low rpm as it does not disperse as quickly. Also, RIT may become significant at high EGR, as EGR will increase the ignition delay [14], which will affect ignition strength due to mixing.

The minimum amount of pilot is a complicated problem involving engine conditions plus injection duration, injection rate and shape, injector configuration, and nozzle shape. The sheer number of variables involved suggest that numerical simulation studies be conducted to optimize the diesel charge before experiments. Noting that the diesel accounts for as little as 3% of the total fuel (at high load), further reduction of pilot would result in only nominal emission reductions due to fuel substitution. However, optimized pilot charge may improve natural gas ignition, where minimal pre-mixing of the natural gas should result in the lowest possible NOx formation.

The emission results indicate that optimal timing between gas and pilot injection occurs at approximately 13° and 8-14° for 1200 and 800 rpm respectively. The optimum timing for both speeds included the ‘normal’ timing of 1.8 ms. The emission response was more flat in terms of absolute time at 800 rpm than at 1200 rpm. This is likely influenced by enhanced mixing at higher speeds. The fact that optimal emissions resulted at delayed sequential injection concurs with diesel/CH4 simulations by [23], which determined that an injection delay of gas after diesel reduces chance of fuel over-mixing and impingement. In those simulations, increasing the injection delay from zero decreased CO formation and reduced unburned CH4. The results of these experiments indicate that a minimum of premixed combustion, with good ignition of the natural gas, brings about the best possible emissions.
An examination of the heat release data of extended RIT shown in Figure 6-2 yields an apparent minimum threshold before the diesel pilot will ignite. When the diesel pilot is injected at 
-27° ATDC and at -37° ATDC, the heat release rate is negligible until approximately -20° c.a.. The minimum auto-ignition temperature of diesel is approximately 650 K [14], and according to polytropic compression relations, the in-cylinder temperatures reaches autoignition threshold at approximately -30° ATDC. After the diesel reaches 650 K and is mixed with oxygen, there is a delay while the diesel acquires enough energy to activate combustion reaction[14]. At 1200 rpm, this appears to take approximately 10 degrees.

The effects of RIT on the combustion events: start of combustion (SOC), end of combustion (EOC) and 50% heat release (HR50) are shown in Figure 6-3. The SOC is defined as the point where 10% total heat release is attained, and EOC is defined at 90% of total heat release. For these heat release comparisons only one standard deviation is used for experimental error. This error estimate is employed as engine conditions were not changed except for relative injection timing, and thus the experimental variation between relative injection tests is much smaller than the repeatability tests, where engine conditions were dramatically adjusted between samples. The SOC appears to be the most affected by RIT, where it increases as RIT is changed from 12°. The increased SOC changes more strongly for short RIT. The HR50 appears less affected by RIT than SOC, as HR50 is only shifted for RIT of 0 and 32°c.a.. The EOC does not appear to move consistently with RIT as SOC shifts. It is interesting that the EOC does not change much for very short or long relative timings even though the SOC does. This implies that the combustion rate must be higher for those timings than for a 'normal' relative injection timing.
Figure 6-3  Comparison of combustion events at $\phi=0.4$, GSOI=-5° ATDC, 1200 rpm at various RIT.

The burn duration, which is the difference between the SOC and EOC is another measure for examining combustion. The effect of relative injection timing on burn duration is shown in Figure 6-4. The maximum burn duration occurs with a relative injection timing between 8 and 22°c.a.. This range likely corresponds to the maximum proportion of mixing-limited combustion as compared to other relative timings. As RIT is increased from this point, the burn duration declines slightly. As RIT is reduced from 8°c.a. the burn duration decreases to relative injection of -4°c.a., where pilot is injected after natural gas. This minimum burn duration is where the natural gas has mixed sufficiently to burn the fastest. When the relative injection of the pilot is delayed to -7°c.a., the burn duration appears to increase. This indicates that the combustible natural gas mixture is likely more lean for pilot injections delayed beyond approximately -4°c.a.. Simulations of 3 injection delays presented by Ouellette [23] indicated that injection delay of 0.25 ms resulted in the shortest burn duration, and 0.5 ms delay was faster than no delay. The results from the current study indicate that slightly negative injection delay results in the fastest combustion. The difference could be due to the dissimilar configurations between simulation and this injector, or possibly a difference between commanded and actual start of injection times.
The effect of changing relative injection timing on variability of engine output is shown in Figure 6-5. The scatter in the data is noticeable, and only negative relative injection timings show moderately higher variability. It appears that RIT does not strongly affect engine output variability, therefore the COV of indicated mean effective pressure will not be used for further comparisons.

The effect of changing relative injection timing on the variability of combustion events is shown in Figure 6-6. The two parameters chosen for the variability of combustion are the standard deviations of start of combustion and of 50% heat release. Both parameters increase with negative relative timing, and increase dramatically for relative timing above 20°. Because the correlation coefficient is 0.98, only the standard deviation 50% heat release will be used for performance comparisons at different operating conditions. The high correlation implies that the entire combustion event is varying with RIT and not simply the start of combustion. The increased variability is expected coincide with increased CO and THC emissions as the variation is an indication of combustion stability.
Figure 6-5  Coefficient of variation of IMEP for various relative injection timings at $\phi=0.4$, GSOI=$-5^\circ$ ATDC, 1200 rpm.

Figure 6-6  Standard deviation of combustion events for various relative injection timings at $\phi=0.4$, GSOI=$-5^\circ$ ATDC, 1200 rpm.
6.2 Variable Pilot and Natural Gas Injection

The tests presented in this section were accomplished by varying the injection of both fuels to maintain a constant point of 50% heat release. This was done as HR50 was previously shown in section 5.2 to be a marker for efficiency and NO\textsubscript{x} production. This provides a consistent comparison for emissions and efficiency. The influence of timing on RIT effects was examined at 1200 rpm, for of HR50 at +5 and +15°ATDC held within 0.5°. The increments for the relative timings were chosen with baseline points of -0.5, 0, 1.8 (standard), 3 ms plus two variable timings, one short and one extended. These points are illustrated in Figure 6-7 with ‘A’ and ‘B’ denoting the short and extended timings respectively. The GSOI was held constant and the HR50 was determined for RIT of 1.8 ms from real-time heat release analysis on the DAQ computer. The two relative timings where HR50 shifted +0.5° relative to that timing were arbitrarily chosen as the variable timings. These variable points were dependent upon engine conditions.

![Figure 6-7 Illustration of the determination of the variable relative injection timing settings.](#)

At 1200 rpm, the variable timings were 2° and 29°c.a. at HR50 of +5°, and 6° and 29°c.a. at HR50 of +15°. By using the same strategy at 800 rpm, HR50 of +15°, the timings were 3° and 25°c.a.. The extended variable RIT was utilized as an arbitrary limit of how long the relative timing may be before substantial deterioration of natural gas ignition. The different limits at different operating conditions suggest that operating conditions influence the effects of relative injection timing. The influence of speed on RIT effects was examined by comparing 800 rpm
and 1200 rpm at a constant HR50 of +15° ATDC. As negative relative timings caused audible knock and degradation of emissions, the minimum RIT at 800 rpm was set to 0°.

### 6.2.1 Performance

To compare performance, relative injection timing was varied to determine impact on efficiency, burn duration, and variability of 50% heat release. The indicated specific fuel consumption is constant within experimental error for all relative timings for each absolute timing at 1200 rpm as shown in Figure 6-8. As noted earlier, a slightly negative relative timing resulted in the shortest burn duration. As burn duration becomes shorter, the engine performance becomes closer to the ideal otto engine cycle. As such, it was expected that efficiency would improve with the most rapid burning. However, it appears that the change in burn duration is not significant enough to alter engine efficiency for this fuelling rate. This may be due to incomplete combustion or the change in burn duration is not large enough to affect efficiency. Despite the lack of change in efficiency for this fuelling rate, shorter burn duration has the potential to improve efficiency and should be tested at other loads and higher speeds. The results of this study differ from the findings of Dumitrescu [8], which found decreased thermal efficiency for short relative timing. It was noted by Dumitrescu [8] that the decreased thermal efficiency may have been due to injector limitations at short RIT, rather than due to the combustion effects of a short RIT.

![Figure 6-8](image)

**Figure 6-8** Efficiency for various relative injection timings at φ=0.4, 1200 rpm, with HR50 timings of +5 and +15° ATDC.
As shown in Figure 6-9, the longest burn duration at 1200 rpm occurs at a relative timing of 13°c.a. (1.8 ms) for both absolute timings. The minimum burn duration occurs at a negative relative timing of -4° for both absolute timings. The minimum burn duration is lower at HR50 of +5° than +15, presumably due to the higher cylinder pressures and temperatures which promote combustion. The results indicate that there is little influence of changing absolute injection timing on burn duration for zero RIT. It is apparent that RIT does not affect the short burn duration for retarded timings that was noted in chapter 5.

The effect of RIT on burn duration is shown for two speeds in Figure 6-10. The results are shown in absolute time format and also in crank angle format. It is apparent that burn duration does not collapse onto a single trend versus RIT for either crank angle or absolute time. The relative timing effects at short RIT are dependent on the diesel pilot autoignition delay and the distance the natural gas jet travels to reach the burning pilot. Diesel autoignition delay is strongly dependent on pressure and temperature, which are weakly affected by speed. The gas jet speed is the same, regardless of engine speed. Both of these factors are mostly time dependent phenomenon and this is apparent in Figure 6-10-a for short RIT, where the proportional decrease in burn duration appears to match on an absolute time basis. The effects at long RIT probably have more to do with mixing and cooling of the diesel pilot after combustion, where mixing is roughly proportional to speed. This is apparent in Figure 6-10b, where after the maximum burn duration for each speed, the slope of the burn duration vs. RIT seem to match. Therefore, it appears the characteristics of burn duration at short RIT are time dependent, while the burn duration characteristics at long RIT are speed dependent.
Figure 6-9  Comparison of burn duration for various relative injection timings at $\phi=0.4$, 1200 rpm, with HR50 timings of +5 and +15° ATDC.
Figure 6-10  Burn duration for various relative injection timings at two speeds, 
$\phi=0.4$, HR50=$+15^\circ$ plotted against:
(a) absolute timing, (b) crank angle

The effect of RIT on variability appears similar for both absolute timings tested at 1200 rpm as shown in Figure 6-11. Only at a negative RIT does the variation appear to be lower an earlier absolute timing. This could be due to higher in-cylinder temperatures which should
promote more consistent ignition of the natural gas. The influence of speed on combustion variability due to RIT is shown in Figure 6-12. It appears that with short RIT, the variability at 800 rpm is slightly lower than that of 1600 rpm. There is not much difference in variation at extended RIT.

Figure 6-11 Standard deviation of 50% heat release for various relative injection timings at two absolute timings, $\phi=0.4$, 1200 rpm.

Figure 6-12 Standard deviation of 50% heat release for various relative injection timings at two speeds, $\phi=0.4$, HR50=$+15^\circ$ATDC.
6.2.2 Emissions

Nitrogen Oxides

The effects of relative injection timing on NO\textsubscript{x} emissions at 1200 rpm is shown in Figure 6-13. The minimum NO\textsubscript{x} emissions for both absolute timings occur at relative injection timing of 13° (1.8 ms). As RIT is extended from 13°, there is a slight increase in NO\textsubscript{x} emissions. As relative injection timing is shortened from 13°, there is a dramatic increase NO\textsubscript{x} emissions. The effects of RIT on NO\textsubscript{x} emissions are stronger for the HR50 of +5 as compared to the HR50 of +15. This likely has to do with the lower in-cylinder temperature at later absolute timing. It is interesting that at negative RIT the apparent heat release rate is lower for than for 0 RIT, but the NO\textsubscript{x} emissions are higher. An explanation for this is that the natural gas mixture is more voluminous before ignition at very negative relative timings. Once burned, the products of the mixture may take longer to mix with the cooler bulk air, which allows more time for NO\textsubscript{x} chemical kinetics to proceed.

The effects of RIT on NO\textsubscript{x} emissions at different speeds is shown in Figure 6-14. The minimum NO\textsubscript{x} production at 800 rpm is level between 9° and 14° relative timings. At 800 rpm, the NO\textsubscript{x} increases from a minimum of 6 g/kW-hr to 12 g/kW-hr at 0 RIT. At 1200 rpm, the NO\textsubscript{x} increases from a minimum of 4.4g/kW-hr to 8.6 g/kW-hr. It appears that the effects of relative timing on NO\textsubscript{x} emissions are greater at lower speeds. This is probably due to increased mixing rates at higher speeds.

Dumitrescu [8] found that short RIT increased NO\textsubscript{x} emissions while holding PSOI constant. Chapter 5 illustrated that shifting combustion earlier in the cycle increases NO\textsubscript{x} emissions. It is not surprising that the 3° shorter RIT in [8], which shifted the gas combustion earlier in the cycle, resulted in increased NO\textsubscript{x} emissions. The results from the current study indicate that the increased NO\textsubscript{x} found by [8] would be partly due to premixing, and partly due to shifting the combustion event.
Figure 6-13  Comparison of NO\textsubscript{x} production for various relative injection timings at two absolute timings, \(\phi=0.4\), 1200 rpm.

Figure 6-14  NO\textsubscript{x} emissions for various relative injection timings at two speeds, \(\phi=0.4\), HR\textsubscript{50}=+15°ATDC.
Total Hydrocarbons and Carbon Monoxide

The results of the effects of relative injection timing on both total hydrocarbons and carbon monoxide are shown in Figure 6-15 for two absolute timings. As with NO\textsubscript{x} emissions, the minimum CO and THC emissions occur at an RIT of 13°. At HR50 of +5°, changing RIT increases both THC and CO emissions for both extended and negative relative timings. Relative timing appears to affect CO and THC emissions more strongly at HR50+15. The in-cylinder pressures and temperatures are lower for the later absolute timing, making it more difficult to ignite the natural gas mixture. In addition, there is less time for THCs and CO to oxidize at late absolute timings. The presence of more incompletely burned products at short RIT for late absolute timing indicates that combustion is less efficient and as a result, the corresponding shorter burn durations are less likely to improve engine efficiency (at late absolute timings). The more moderate levels of THC and CO emissions for short RIT at early timings provide more opportunity for the short burn duration to improve efficiency (at early absolute timings).

The simulations of Ouellette [23] found high levels of CO formation with no ignition delay. This was mostly due to impingement of the gas jet on a wall as the flame could not reach the end of the jet before the jet reached the wall. The 1200 rpm experimental results of Dumitrescu [8] indicated that higher levels of CO and THC emissions occur with a 3° RIT, as compared to a 6° RIT. The trends of CO and THC emissions in this study agree with trends at short RIT found by Dumitrescu [8] and CO results of Ouellette.

The effects of RIT on THC and CO emissions at different speeds is shown in Figure 6-16. The minimum THC and CO emissions at 800 rpm occur at the same relative timing as with NO\textsubscript{x} emissions between 9° and 14°. The effects of changing RIT on THC and CO emissions appears similar for both speeds.
Figure 6-15  Carbon monoxide and total hydrocarbon emissions for various relative injection timings at two absolute timings, Φ=0.4, 1200 rpm.

Figure 6-16  Carbon monoxide and total hydrocarbon emissions for various relative injection timings at two speeds, Φ=0.4, HR50=+15°ATDC
Particulate Matter

There was no statistical change in PM emissions for the relative timing sweeps at each operating condition. The experimental error masked any effect of relative timing, partly because the PM emissions at this equivalence ratio are very low. To determine the effect of RIT on PM, this study should be repeated for an engine operating condition where the engine is known to produce higher levels of PM.

Overall Emission Discussion

To examine a perceived correlation between emissions and burn duration, the correlation coefficient between burn duration and emission species was calculated using:

\[ \rho_{x,y} = \frac{Cov(X, Y)}{\sigma_x \cdot \sigma_y} \]  

(Eq. 6.1)

Where \( \sigma \) is the standard deviation for each variable and the \( Cov \) is given by:

\[ Cov(X, Y) = \frac{1}{n} \sum_{i=1}^{n} (x_i - \mu_x)(y_i - \mu_y) \]  

(Eq. 6.2)

The correlation coefficient for each emission species, averaged over each mode tested is listed in Table 6.1. It is apparent that the maximum burn duration corresponds to minimum emissions when changing relative timing at all speeds and absolute timings tested. This likely has to do with increasing the proportion of pre-mixed combustion, which burns faster, but causing more emissions. Burn duration could be used for determining optimal RIT, without requiring lengthy emission data samples and analysis, although this should be verified. The flat optimal emission/ burn duration response to changing relative timing at 800 rpm suggest that more pilot is injected than required for good ignition. This can be inferred as an optimal minimum amount of pilot will likely have an distinct maximum burn duration versus RIT. The accuracy of the burn duration measurements can be improved by averaging more cycles of in-cylinder data. This correlation will not hold for extreme negative relative timings where the burn duration increases or for extremely long RIT where the ignition of the natural gas is very weak. The exact limits for good correlation are unknown. This correlation is only valid for a fixed operating condition.
Table 6.1: Burn Duration Correlations

<table>
<thead>
<tr>
<th>Emission Species</th>
<th>Correlation Coefficient with Burn Duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon monoxide</td>
<td>-0.97</td>
</tr>
<tr>
<td>Nitrogen oxides</td>
<td>-0.98</td>
</tr>
<tr>
<td>Total hydrocarbon</td>
<td>-0.96</td>
</tr>
</tbody>
</table>

Hypothetically, smaller quantities of pilot in the current configuration would result in a smaller window of good ignition. This is because a smaller quantity of diesel pilot will cool faster and thereby reduce the ignition capability. Higher speed or high swirl may require more amounts of pilot, as the pilot will disperse and cool faster than at low speeds. Conversely, the amount of diesel may be reduced at low rpm as it does not disperse as quickly. Also, RIT may become significant at high EGR, as EGR will increase the ignition delay [14], which will affect ignition strength due to mixing.

The minimum amount of pilot is a complicated problem involving engine conditions plus injection duration, injection rate and shape, injector configuration, and nozzle shape. The sheer number of variables involved suggest that numerical simulation studies be conducted to optimize the diesel charge before experiments. Noting that the diesel accounts for as little as 3% of the total fuel (at high load), further reduction of pilot would result in only nominal emission reductions due to fuel substitution. However, optimized pilot charge may improve natural gas ignition, where minimal pre-mixing of the natural gas should result in the lowest possible NO\textsubscript{x} formation.

The emission results indicate that optimal timing between gas and pilot injection occurs at approximately 13\textdegree and 8-14\textdegree for 1200 and 800 rpm respectively. The optimum timing for both speeds included the ‘normal’ timing of 1.8 ms. The emission response was more flat in terms of absolute time at 800 rpm than at 1200 rpm. This is likely influenced by enhanced mixing at higher speeds. The fact that optimal emissions resulted at delayed sequential injection concurs with diesel/CH\textsubscript{4} simulations by [23], which determined that an injection delay of gas after diesel reduces chance of fuel over-mixing and impingement. In those simulations, increasing the injection delay from zero decreased CO formation and reduced unburned CH\textsubscript{4}. The results of these experiments indicate that a minimum of premixed combustion, with good ignition of the natural gas, brings about the best possible emissions.
6.3 Summary

- There is an optimum relative injection timing (or range of timings) for a minimum of all emissions at fixed operating condition, the optimum relative timing appears to be independent of speed or absolute injection timing.

- The optimum relative timing included the ‘normal’ 1.8 ms (9° and 13° for 800 and 1200 rpm respectively). The emission response was flatter at 800 rpm than at 1200 rpm on an absolute time basis.

- The efficiency is not significantly affected by relative injection timing. However, the ability to reduce burn duration provides potential for improved efficiency at early absolute timings.

- When 50% heat release is held constant, burn duration correlates well with emissions. The longest burn duration is produced by combustion most favorable for emissions.

- For short relative injection timings, an optimal delay avoids premixed combustion i.e. it is desirable for the diesel pilot to be ignited before the NG is injected. It seems that combustion appearing more premixed generates worse emissions. This event is more dependent on absolute time than by crank angle.

- Agreement with the emission trends at short relative timings with Dumitrescu [8], however the efficiency findings between studies differ. The difference in thermal efficiency found in that study was likely due to injector design.

- At extended relative injection timings, apparent heat-release data indicate that natural gas ignition by pilot is weakened by mixing of the pilot, therefore higher speeds and high swirl engines may require more diesel for good ignition of the natural gas.

- Future study should include simulation of the diesel pilot with a focus of improving ignition of the natural gas.

- The effect of relative injection timing on particulate matter emissions should be studied at conditions generating more particulate matter (higher equivalence ratios).
7. INJECTION PRESSURE

This chapter presents the effects of changing injection pressure on engine performance and emissions. The injection pressure of the fuel influences the fuel penetration, injection rate, and mixing rate. These in turn influence the fuel combustion, performance and emissions of the engine. The experiments in this section were conducted at two injection pressures and various absolute timings. The variation of injection timings are required to understand the relationships between emissions, injection pressures, and shifts in combustion events. Three sets of tests were conducted: constant air/cycle for 3 fueling rates at one speed; constant equivalence ratio for 3 air flow rates at one speed; one equivalence ratio (\( \phi \)) and constant air/cycle for 3 speeds. The 50% heat release crank angle is used as the independent variable for comparison.

7.0.1 Gas Injection Mechanics

Natural gas is injected into the cylinder through a small nozzle at sonic or near-sonic conditions. The Reynolds number of the jet at sonic velocities is approximately \( 5 \times 10^6 \), which is fully turbulent. As the jet penetrates into the cylinder it quickly mixes with the chamber air. The goal of injection is to achieve proper air utilization while avoiding over-penetration, which is fuel impingement onto the piston bowl or the cylinder walls [14]. It is also desirable to avoid under-penetration as it results in poor air utilization. Ouellette [23] noted that over-mixing of gas jets is also a potential issue for pilot ignition, defined as too much lean mixture prior to ignition. That study showed increasing injection pressures increases injection rate, mixing, and also gas jet penetration. The critical pressure ratio for sonic injection of methane at 300K is 1.84, based on the perfect gas law, injection-to-cylinder pressure ratios above this will result in an underexpanded jet. The experiments by Ouellette [23] also found that the jets observe self-similarity with no discontinuity at sonic injection. This means that the gas jet will exhibit consistent characteristics above and below the critical pressure ratio.

In the case of operation at high load, 1200 rpm, the in-cylinder pressure approaches that of the injection pressure. The peak cylinder pressure reaches 18.5 MPa at 5° ATDC while the injection duration is from -10° to +7° ATDC. It is evident that the injection at this point is no longer choked flow, as the pressure ratio is near unity. The choked state of the gas jet is dependent on the speed and load of the engine as well as timing and pressure of the gas injection.
However, from a practical perspective, the injection pressure should be kept as low as possible to minimize energy required for compressing the natural gas.

### 7.0.2 Combustion Effects

Increasing the gas pressure boosts the rate of fuel injection, which should enhance mixing rates, and thereby increase the heat release rate. An increase of injection pressure causes an increase in momentum injection rate, which thereby mixes the gas faster. Also the fuel injection rate increases, which injects more gas in the cylinder for a given crank angle (within the injection duration). The effects of changing injection pressure on heat release rate is shown in Figure 7-1 for $\phi$ of 0.5 at a fixed start of gas injection (GSOI). It appears that the pilot combustion and the ignition of the natural gas is the same for both injection pressures. However, at the higher injection pressure, the heat release rate appears to increase faster (steeper slope) than at the low injection pressure. This is probably an indication of increased mixing rates. Also, the maximum apparent heat release rate is higher for increased injection pressure. The heat release rate for 19 MPa injection pressure shows a distinctive ‘double hump’, which indicates a progression from pre-mixed to mixing-limited combustion. However, for 23 MPa, the ‘double hump’ occurs more rapidly and at a higher rate. This is probably due to a combination of the increased injection and mixing rates. It also appears that the burn duration is shorter for higher injection pressures. This is confirmed by the results shown in Figure 7-2, which shows a general reduction in burn duration with increasing injection pressure of approximately $7^\circ$. As the GSOI was the same for both injection pressures, it is apparent that the 50% heat release shifts earlier in the cycle for a higher injection pressure.

![Figure 7-1](image_url)  
*Figure 7-1* Apparent heat release rate for two injection pressures at, 1200 rpm, $\phi=0.5$, GSOI $0^\circ$ and 4 g/cycle of air.
7.0.3 Varying Charge Air Mass

The effects of injection pressure are influenced by the ratio of the injection pressure to in-cylinder pressure. As such, the in-cylinder pressure, or charge air mass, was varied. By varying the charge air mass or variable supercharging, emissions and performance can be affected, without changing injection pressure. Supercharging influences the in-cylinder mixing rates, where the mixing rate is proportional to $P_{\text{cyl}}^{1/4}$ for a constant injection momentum[23]. To examine the effects of variable supercharging, equivalence ratio was held constant at $\phi$ of 0.4 while charge air mass was varied between 3 and 6.5 g/cycle. Emission results at 6.5 g/cycle of air are not reported due to inaccurate air flow rate measurements. Studies of supercharging with diesel found that increasing supercharging with rapid injection rates allowed for NO$_x$ reductions with few penalties from other emissions [38]. Supercharging is effective for diesel NO$_x$ reductions as the increased mixing and shorter diesel ignition delay reduces premixed combustion.
7.1 Performance

The effects of injection pressure on the performance of the engine is demonstrated by comparison of efficiency using Indicated Specific Fuel Consumption (ISFC) across a range of injection timings. The independent timing variable used for ISFC comparison is the 50% heat release crank angle (HR50). Three different parameters were varied with injection pressure; equivalence ratio, speed, and charge air mass. The equivalence ratio tests were conducted at 1200 rpm for $\phi$ of 0.3 and 0.5, with various injection timings, and injection pressures of 19 and 23 MPa. The second set of tests were conducted at $\phi$ of 0.4 for speeds between 800 and 1600 rpm. Equivalence ratio was held constant as it was previously found to affect emissions. The last set of tests were conducted at 1200 rpm and a constant $\phi$ of 0.4, while varying the charge air mass between 3 and 6.5 g/cycle. The relative injection timing (RIT) was held constant at 1.8 ms, while the absolute injection timing was varied in 5° increments.

The effects on performance of increasing injection pressure from 19 to 23 MPa at different equivalence ratios is shown in Figure 7-3. The ISFC trends appear statistically the same with changes in injection pressure for the range of fueling tested at 4 g of charge air. The results of changing injection pressures at different speeds are shown in Figure 7-4. It is reaffirmed that the gross efficiency near optimum timing is greater at 1600 rpm than 800 rpm. It is apparent that when considering ISFC in terms of HR50, injection pressure does not affect efficiency.

![Figure 7-3](image-url)  
**Figure 7-3** Efficiency for two injection pressures at two equivalence ratios, 1200 rpm, 4g/cycle of air and various timings.
Figure 7-4  Efficiency for two injection pressures at two speeds, $\phi = 0.4$, 1200 rpm, 4g/cycle of air and various timings.

The effect of changing the injection pressure at different charge air masses with $\phi$ of 0.4 is shown in Figure 7-5. The air supply for 6.5 g/cycle was a reciprocating compressor and the screw compressor was used for 3g/cycle. The efficiency at 3g/cycle was unaffected by injection pressure. There was no effect of injection pressure on efficiency trends at late timings. However, for an early injection timing at 19MPa, efficiency was lower than comparable timings at 23 MPa injection pressure. The poor efficiency at early timing, 19 MPa injection, was due to the high in-cylinder pressure, which reached 18.4 MPa at 5° ATDC, while the injection duration was from -10° to 6.3°c.a.. The low pressure ratio between the injector and cylinder that was reached likely results in poor mixing and a long burn duration, which is less efficient. As a comparison, the burn duration at 19 MPa is 32°, whereas for 23 MPa injection pressure, the burn duration is only 23°. This indicates that injection pressure has an effect at high loads and early injection timings when the in-cylinder pressure approaches injection pressure. A comparison between the efficiency curves at different charge air masses indicates that the efficiency is greater at higher charge air mass. This was likely due to the dramatic load difference between 3 and 6.5 g/cycle which was 6.5 bar and 13.5 bar IMEP respectively. At high loads there was likely a smaller percentage of energy lost to heat transfer. The ISFC was improved at high pressure despite the increased burn duration at high load, with the exception of operation at 6.5 g/cycle of air, 19 MPa, GSOI -10°.
Figure 7-5  Efficiency for two injection pressures at two air flow-rates, $\phi=0.4$, 1200 rpm, and various timings.

The earliest timing for 19 MPa injection pressure, 6.5g/cycle supercharging where efficiency appears relatively unaffected, is GSOI=0°. The in-cylinder pressure remained almost constant at 14 MPa throughout the injection duration of 10.5°c.a.. Assuming there is only a small amount of losses through injection, the pressure ratio between injector and cylinder is approximately constant at 1.37 for this engine condition. A pressure trace for this condition is included in Appendix J. The pressure ratio went from 2.55 down to 1.02 for the condition of GSOI=-10°, $\phi$ of 0.4, 6.5 g/cycle of air and 19 MPa injection. Therefore, efficiency appears unaffected when minimum ratio between the injection and in-cylinder pressure remains greater than 1.37. However, the true limit is not defined by these experiments, but lies somewhere between pressure ratios of 1.02 and 1.37. As the pressure ratio is less than the critical 1.84, apparently sonic injection is not necessary from an efficiency perspective for this engine, but the lower pressure ratio may affect emissions due to penetration and mixing effects.

The studies of injection pressure by Dumitrescu [8] found the best thermal efficiency averaged over all loads, was at intermediate pressure. The injector used in that study did not allow for independent control of the critical timing of gas injection. As such, the pilot start of combustion and RIT were held constant, which allowed the centroid of gas injection to vary with injection pressure. Douville [28] found the best thermal efficiency at high load was also an
intermediate pressure with simultaneous injection. Results from the current study suggest that such variation of the gas injection would have caused the centroid of combustion to vary with injection pressure, possibly on either side of optimum timing. This may explain the optimum intermediate injection pressure found by Dumitrescu [8] and Douville [28].

By comparing ISFC with 50% heat release crank angle, it is shown that the injection pressure does not generally affect the combustion efficiency at the conditions tested, except for high load, early timings where the in-cylinder pressure approaches injection pressure. Another consideration that was not encountered in these experiments is the desire to avoid overly late combustion due to slow injection at high speeds as identified by [7], which notes a sufficient injection pressure is required to avoid this scenario.

7.2 Emissions

The effects of injection pressure on engine emissions are presented for NO\textsubscript{x}, CO, THC, and PM emissions. For consistency, all emissions results in this chapter are compared using timing of 50\% cumulative heat release (HR50). The first set of tests that are presented were conducted at 1200 rpm for equivalence ratios of 0.3 and 0.5 with various injection timings for injection pressures of 19 and 23 MPa. The second set of tests at 1200 rpm included varying the charge air mass between 3 and 4 g/cycle at a constant \( \phi \) of 0.4. The equivalence ratio was held constant as emissions were previously found to change with equivalence ratio. The last set of tests presented were conducted at \( \phi \) of 0.4 for speeds between 800 and 1600 rpm.

7.2.1 Nitrogen Oxides

The results of changing injection pressure on specific NO\textsubscript{x} emissions at different equivalence ratios is shown in Figure 7-6. The NO\textsubscript{x} emissions for both injection pressures and both equivalence ratios follow the same trend when plotted against 50\% heat release. This implies that the increased combustion rate due to increased injection pressure does not adversely affect NO\textsubscript{x} production. This differs from the studies by Dumitrescu [8] and Douville [28], which indicated increased NO\textsubscript{x} with increasing injection pressure. Neither study however, accounted for changes in injection rate. It seems that the NO\textsubscript{x} formation does not depend on the combustion duration or intensity, but primarily depends on when the combustion occurs within the engine cycle. This also indicates that the amount of premixed combustion does not change significantly
with the increase in backpressure. This can be inferred by remembering from chapter 6 that a short RIT resulted in premixed combustion and correspondingly more NOx emissions.

Figure 7-6 Oxides of nitrogen emissions for two injection pressures at two equivalence ratios, 1200 rpm, 4g/cycle of air and various timings.

The results of increasing injection pressure on NOx emissions at different charge air masses for $\phi 0.4$ is shown in Figure 7-7. The NOx emissions follow the same trend within experimental error for both charge air masses and injection pressures. Thus, supercharging does not appear to affect HPDI natural-gas NOx emissions significantly. Likely this is because much of the combustion is already mixing-limited and the portion of premixed combustion is relatively unaffected by slight changes in the mixing rate. Further investigation is required to confirm this hypothesis for a wider range of charge air masses. As it is demonstrated that injection pressure has no significant effect on NOx production using HR50 as an independent variable, no further discussion is included for different speeds. A confirming figure is found in Appendix J. While the minimum NOx emissions at very retarded timings (HR50 > +15°c.a.) are within experimental error, the NOx levels appear slightly higher for 23 than 19 MPa injection for all cases. Perhaps a study using finer instrumentation focussed at very late injection, a small amount of NOx reduction may be obtainable by decreasing injection pressure. These changes would be small compared to retarding the injection timing from piston TDC. This is different from injection pressure results
of Stumpp et al. [25] in a diesel engine, where increasing injection pressure significantly increases the minimum possible NOx emissions.

Figure 7-7 Nitrogen Oxides emissions for two injection pressures at two air flow-rates, φ=0.4, 1200 rpm, and various timings.

7.2.2 Total Hydrocarbons

The effect of injection pressure on total hydrocarbon (THC) emissions at different timings and equivalence ratios is shown in Figure 7-8. An examination of THC emissions at φ of 0.5 reveals that there is little difference between the two injection pressures, but at φ of 0.3, the high THC emission rate at late timings is exacerbated by increased injection pressure. The increased THC emissions at retarded timings is assumed to be almost all methane. Justification for this assumption is found in Appendix K. The increased mixing rates may over-lean some of the fuel and/or quench some of the flame at lower temperatures. At higher temperatures (higher φ), the effect is less significant due to higher average temperatures and less charge air for the flame to mix with. Data in Dumitrescu [8] reveals that increasing injection pressure caused slight increases of CH4 across all loads at the timing chosen. This supports the hypothesis that increasing injection pressure may over-mix some of the fuel in HPDI of natural gas.

The effects of injection pressure on total hydrocarbon emissions at different speeds is shown in Figure 7-9. The results appear consistent with the mixing hypothesis, as at 800 rpm the
THC emissions are lower than at 1600 rpm, where more mixing occurs due to higher speed. At slow speeds, adequate time is available for oxidation of the fuel and THC emissions appear irrespective of injection pressure. At higher speeds and retarded timings however, the increased mixing effects of higher pressures appear to quench the reaction and there are more unburned hydrocarbons. The equivalence ratio presented in Figure 7-9 is 0.4, as compared to $\phi$ of 0.3 affected by injection pressure at 1200 rpm.

The results for changing the amount of supercharging at a constant equivalence ratio is shown in Figure 7-10. The amount of THC emissions is unaffected by injection pressure except for 4g/cycle between 12° and 20° ATDC. There is no statistical difference in THC emissions between 3 and 4g/cycle, and therefore THC appear unaffected by changing the amount of supercharging at a constant equivalence ratio. More mixing is expected with 4g/cycle due to a combination of higher in-cylinder pressures and more injection momentum due to more fuel mass injected. The increase in THC production due to injection pressure is slightly stronger at late injection for 4g/cycle as compared to 3g/cycle. Further study at a wider range of charge air mass is needed to determine if this is a consistent effect. The fueling rate for $\phi$ of 0.4 at 3g charge air is equivalent to $\phi$ 0.3, 4g charge air. It appears there is a greater effect on THC emissions by changing $\phi$ than by changing the charge air mass at a constant $\phi$.

![Figure 7-8](image)

**Figure 7-8** Total hydrocarbon emissions for two injection pressures at two equivalence ratios, 1200 rpm, 4g/cycle of air and various timings.
Figure 7-9  Total hydrocarbons emissions for two injection pressures at two speeds, $\phi=0.4$, 1200 rpm, 4g/cycle of air and various timings.

Figure 7-10  Total hydrocarbons at $\phi=0.4$, for two injection pressures, air flow-rates, 1200 rpm.
7.2.3 Carbon Monoxide

The effects of injection pressure on CO emissions as a function of timing for three equivalence ratios at 1200 rpm is shown in Figure 7-11. To clarify the results, the figure includes approximate trend-lines for each test condition and arrows showing the increased injection pressure. At retarded timings for a $\phi$ of 0.3, CO emissions are unaffected by injection pressure at early timings. As injection is retarded however, CO emissions are higher for injection at 23 MPa as compared to 19 MPa. This suggests that the CO oxidation process is being quenched due to either increased wall impingement or ultra-lean conditions generated by the enhanced mixing. For a $\phi$ of 0.5, CO emissions are reduced by increasing injection pressure. The higher levels of CO emission indicates poor air utilization, and lower CO emissions indicate improved mixing. It seems that the increased mixing induced by increasing injection pressure improves fuel air-utilization at higher equivalence ratios, but can also quench CO oxidation at retarded timing at lower equivalence ratios. As the CO emissions for retarded timings at $\phi$ of 0.5 are actually reduced with increasing injection pressure, it suggests wall impingement is not a factor at retarded timings. Instead, it is likely that the increases in CO emissions found at $\phi$ of 0.3 are due to quenched oxidation caused by increased mixing. It is interesting that the conditions tested appear to reach the extremes of both air under-utilization and over-mixing. Dumitrescu [8] also concluded that CO was unaffected by injection pressure, however an examination of the data yields a discernible reduction of CO emissions at high load with increasing injection pressure. Therefore, it appears that injection pressure can be used to improve air utilization for engine conditions where CO emissions require mitigation. The absolute value of the CO emissions for all of these cases is very low.
The effects of injection pressure on CO emissions at $\phi$ of 0.4 across different speeds is shown in Figure 7-12. Injection pressure does not significantly affect the relationship between CO emissions and injection timing for the speeds tested. The effect of changing injection pressure on CO emissions for different charge air masses at a constant equivalence ratio is shown in Figure 7-13. The CO production is statistically the same for both injection pressures for both charge air masses. As a result CO emissions seem unaffected by moderate changes in supercharging. As with THC emissions, CO emissions trends appear the same at a constant equivalence ratio for different supercharging rates. This suggests that equivalence ratio has more effect on CO and THC emissions than charge air mass (at constant $\phi$). Considering that the mixing rate changed by less than 10% due to a 25% change in-cylinder density, emission studies should be examined for larger differences in charge air mass. Increasing the supercharging rates is not expected to drastically affect the amount of premixed natural gas combustion, which appeared to generate more pollutant emissions in chapter 6.
7.2.4 Particulate Matter

The effects of injection pressure on PM emissions at 1200 rpm is shown in Figure 7-14. Injection pressure does not significantly affect PM emissions at \( \phi \) of 0.3. However, the PM emissions...
emissions are lower around HR50 of 10°c.a. for $\phi$ of 0.5 at 23 Mpa injection as compared to 19 MPa injection. This differs from the results of Baribeau [17], which showed improvement at low load, but no change in PM at high load. This may be due to the differences in injection pressures and loads tested between studies.

The effects of injection pressure on PM emissions across different speeds are shown in Figure 7-15. At 800 rpm, there is no apparent effect of injection pressure on PM emissions. At 1600 rpm both injection pressures exhibit low PM emissions near HR50 of 20°c.a. and then PM increases with late timing. Therefore, it appears that injection pressure has little effect on PM emissions for these conditions.

The effects of changing injection pressure and changing the charged air mass on PM emissions at $\phi$ of 0.4 is shown in Figure 7-16. There is no statistical difference in PM emissions between 3 and 4 g/cycle at 23 MPa injection. As well, there is no statistical difference between injection pressures at 4 g cycle of air. The PM data for 19 MPa at 1200 rpm, $\phi$ of 0.4, 3g/cycle was faulty and not reported. As the results indicate there is statistically no effect of injection pressure on PM emissions for $\phi$ of 0.4 at any other condition, there is likely no effect of pressure on PM for 3g/cycle at $\phi$ of 0.4.

Figure 7-14 Particulate matter emissions for two injection pressures at two equivalence ratios, 1200 rpm, 4g/cycle of air and various timings.
Figure 7-15 Particulate matter for two injection pressures at two speeds, $\phi=0.4$, 1200 rpm, 4g/cycle of air and various timings.

Figure 7-16 Particulate matter emissions for two injection pressures at two air flow-rates, $\phi=0.4$, 1200 rpm, and various timings.
7.3 Summary

- Increasing injection pressure from 19 to 23 MPa:
  a) increases combustion rate and decreased burn duration;
  b) shifts the 50% heat release earlier in the cycle for a fixed start of gas injection;
  c) causes no change in efficiency trends for 4g/cycle of charge air, or 6.5 g/cycle charge air at late injection timing, when comparing against 50% heat release;
  d) improves of engine efficiency only for early injection with 6.5 g/cycle charge air;
  e) causes no change in NOx emission trend when compared at 50% heat release, for all conditions;
  f) generally decreases CO emissions, but causes no effect on THC emissions, for $\phi$ of 0.5, at 1200 rpm;
  g) increases CO and THC emissions for late injection timings, for $\phi$ of 0.3 at 1200 rpm;
  h) no change in CO emissions, but increases THC emissions at 1600 rpm, for $\phi$ of 0.4;
  i) causes no change in THC emissions for $\phi$ of 0.4, at 800 or 1200 rpm;
  j) causes no significant change in PM emissions for $\phi$ of 0.3 or 0.4;
  k) decreases peak PM emissions for $\phi$ of 0.5, 1200 rpm, which coincides with CO reduction;

- The changes in THC and CO emissions consistent with, and are most likely due to increased mixing rates at higher injection pressure.

- Changing equivalence ratio causes more considerable change in THC and CO emissions than moderately changing the load at a constant equivalence ratio (i.e. supercharging).

- The effects on emissions of high rates of supercharging should be tested with an accurate air flow-rate measurement.
8. CONCLUSIONS

8.1 Introduction

The general focus of this study has been to provide knowledge of how changing the injection parameters affect the emissions and efficiency of a modified Cummins ISX engine fueled with pilot-ignited, high pressure direct injection (HPDI) of natural gas. This knowledge is needed to develop optimized injection strategies for similar engines. The injection parameters studied include the absolute injection timing, the relative injection timing between the diesel pilot and natural gas, and the injection pressure. The performance measures studied included the efficiency, and pollutant emissions of nitrogen oxides (NO\textsubscript{x}), carbon monoxide (CO), total hydrocarbons (THC) and particulate matter (PM).

The engine was a supercharged, single-cylinder engine and several considerations are important for operation, particularly concerning the exhaust back pressure. The effects of back pressure on emissions and efficiency were studied to determine a consistent testing procedure to simulate turbo-charged conditions. Increasing back pressure can decrease NO\textsubscript{x} emissions, though these effects are reduced as speed is increased, or injection timing is retarded. Increasing back pressure can significantly affect THC emissions, where the effects are delayed as equivalence ratio is increased. Increasing back pressure reduces CO emissions if they are greater than approximately 1 g/kW-hr, where otherwise there is no significant effect. Increasing back pressure affects PM emissions at some conditions, where PM reductions coincide with operating points where back pressure induces reductions in CO emissions. Bearing in mind the emissions, the back pressure was set to 150 kPa (absolute) which, depending on speed, was 15 to 40 kPa lower than intake manifold pressure at time of intake valve opening. The rate of supercharging was also briefly considered.

8.2 Conclusions

Based on measurements which cover a range of equivalence ratios between 0.3 and 0.5, corresponding to loads mainly between 6 and 10 bar IMEP, and speeds between 800 and 1600 rpm, the following conclusions have been drawn:
8.2.1 Relative Injection Timing

The timing between the start of injections of the pilot and natural gas was varied at 800 and 1200 rpm for an equivalence ratio of 0.4. This relative timing was found to significantly affect engine performance and emissions in the following ways:

- When the 50% cumulative heat release timing is held constant and relative timing is varied, the burn duration correlates well with NO\textsubscript{x}, THC, and CO emissions. The maximum burn duration corresponds to the lowest emissions.
- The diesel pilot should be ignited before the natural gas is injected to obtain the minimum emissions. The emission trends exhibited at short relative timings agree with data found in Dumitrescu [8].
- There is a relative injection timing that minimizes all emissions, which appears to be independent of speed or absolute injection timing. The optimum relative timing included 1.8 ms for all speeds.
- For long relative injection timings, apparent heat-release data indicates the natural gas ignition by the pilot is weakened by excessive mixing of the burned products of the pilot combustion. This being so, the minimum pilot required for good ignition of the natural gas is probably greater at higher speeds and high swirl engines.

Relative injection timing does not significantly affect efficiency or particulate matter for the conditions tested. This efficiency result differs from the reduced efficiency found by Dumitrescu [8] with short relative injection.

8.2.2 Absolute Injection Timing

Absolute injection timings sweeps were conducted with a constant relative injection between pilot and natural gas. The timing of 50% cumulative heat release (HR50) was found to be a good independent variable for comparison of NO\textsubscript{x} emissions and efficiency.

- Indicated fuel consumption as a function of HR50 is almost independent of speed. The optimum timing for efficiency occurs when the 50% heat release is approximately +5° ATDC. As timing is retarded beyond this, efficiency declines. Equivalence ratio does not affect efficiency significantly.
- Specific NO\textsubscript{x} emissions are a strong function on HR50 and are independent of equivalence ratio for constant air injection. This indicates that the timing of the combustion
event is a critical factor in NO$_x$ production for the HPDI engine. There is a limit to which NO$_x$ can be reduced by retarding timing, which occurs when the HR50 reaches a threshold of approximately 15° ATDC at 1200 rpm. A lower limit for retarded timing NO$_x$ reduction is compatible with data found in Dumitrescu [8]. The engine should be operated such that absolute injection timing is not retarded beyond the apparent limit defined by the HR50 at each speed. Beyond this point no further reductions in NO$_x$ are gained and, efficiency is reduced and other emissions deteriorate.

- The NOx emitted is proportional to the fuel burned for fixed HR50, speed, and air flow-rate for equivalence ratios between 0.3 and 0.5.
- Carbon monoxide emissions increase with retarding gas injection for late timings for $\phi$ of 0.3 and 0.4 at all speeds tested. For $\phi$ of 0.5 at 1200 rpm however, retarding injection causes CO emissions to increase, then decrease and then increase again.
- Retarding injection increases THC emissions for all conditions tested. This is expected as combustion temperatures are lower and less time is available for oxidation as timing is retarded. The effect is more pronounced at lower equivalence ratios and higher speeds.
- Particulate matter emissions are unaffected by injection timing for equivalence ratios of 0.3 and 0.4 at 800 and 1200 rpm; retarding timing generally increases PM emissions for $\phi$ of 0.4 at 1600 rpm; and retarding timing causes PM to vary in a similar manner as CO for $\phi$ of 0.5 at 1200 rpm.
- Raising speed for various timings: increases specific THC emissions; does not significantly affect CO emissions; generally increases PM emissions; decreases indicated specific NO$_x$ production; and delays the timing of the NO$_x$ reduction limit.

**8.2.3 Injection Pressure and Supercharging**

Increasing injection pressure shifts the combustion event earlier in the cycle for a constant start of gas injection, and the HR50 timing was employed for showing the effects of injection pressure on emissions and efficiency.

- As a function of HR50, efficiency is generally independent of injection pressure. Injection pressure only significantly affects efficiency trends when the in-cylinder pressure approaches injection pressure (at high load, early timing), where increasing injection
pressure improves efficiency. The findings differ from the findings of Dumitrescu [8] and Douville [28], who found the best efficiency at an intermediate pressure based on a constant start of injection and did not examine HR50.

- NO\textsubscript{x} emissions trends with HR50 are not significantly affected by injection pressure.
- Increasing injection pressure generally adversely affects tHC at late timings.
- Increasing injection pressure can either increase or decrease CO emissions depending on equivalence ratio and absolute timing. Increasing the injection pressure at an equivalence ratio of 0.4 does not significantly affect CO emissions. The complex dependence of CO emissions on injection pressure for equivalence ratios of 0.3 and 0.5 is shown in Figure 7-11.
- Increasing injection pressure can decrease PM for an equivalence ratio of 0.5 at 1200 rpm. Particulate matter emissions are not significantly affected at other engine conditions.
- Changing the load by 25% while maintaining a constant equivalence ratio of 0.4 with supercharging does not appear to affect any emissions. Changing the load by 25% with a fixed charge air mass did affect tHC and CO emissions, implying that equivalence ratio is an important factor for these emissions.

8.3 Recommendations for Future Study

This study represents a partial understanding of the pilot-ignited HPDI process. The operating conditions tested were somewhat limited. Several avenues of investigation may expand understanding and application of pilot-ignited HPDI.

- Experiments attempting minimize the diesel pilot should be conducted, using the burn duration as guidance for optimization of the relative timing. This method will be faster than taking emission measurements and requires less data processing. Heat release data may also be useful in determining the minimum amount of diesel for good ignition. Pilot quantities may likely be reduced at slow speeds as compared to high speeds.
- Injection parameters warrant further investigation for a wider range of equivalence ratios, and particular attention should be paid to the \text{NO}\textsubscript{x} versus 50% heat release trends. Emissions should be studied for higher supercharging rates with an accurate air-
flow measurement. These tests should be conducted with an injector that provides better air utilization at high equivalence ratios (i.e. new injector geometry).

- Experiments should be conducted as to whether the SAE recommended NO\(_x\) correction factor for diesel engines, which was originally determined for diesel fuel, is valid for natural gas.
- Future investigation should include variation of diesel pilot quantity and alteration of rate shape of pilot injection.
- Effects of the gas injection rate shape should be studied.
- The effect of relative injection timing on particulate matter emissions should be studied at conditions generating more particulate matter (higher equivalence ratios). To improve accuracy of PM measurements, the test period should be extended.
- The loose correlation of CO emissions trends with PM emission trends warrants further investigation.
REFERENCES


APPENDIX A. CALIBRATION

Emission Bench Procedure
Alternative Fuels Laboratory,
UBC Mechanical Engineering

NOTES

1. When calibrating - ensure the flow rate is correct for each step.
2. Do not change the pressure on the bottle regulators (exception-NOx).
3. Order gas when cylinder pressure falls below 500 psi to ensure enough lead time.
4. Do NOT turn off analyzers in cabinet #2 (breakers 2 & 6A ) unless a week without testing is anticipated.
5. Turn off Ratfisch FID (heater Oven, Pump, then Power) when not using to minimize bottled fuel and air consumption.
6. If flow rates are all too low – check heated filters in cabinet #1 and replace as necessary. Sample pump must be shut off and the filters cold. If there are still problems with flow rate, check the sample pump (not a trivial procedure).
7. Balance analyzer flow rates once sampling from hot exhaust has commenced (engine conditions may affect flow rates)
8. Wait at least 60s after turning on calibration gas before activating the calibration function.
9. Ensure that the heated enclosure has reached appropriate temperature before sampling.
10. The NOx analyser (API) is challenging to calibrate - be careful!
11. NOx component of Siemens Ultramat 22P is currently disabled (see maintenance records).

Part 1 – Start-Up

1. Open the compressed gas cylinders fully (beside the fume hood).
2. Turn on breakers in the back of cabinet #1, except NOx converter (breaker 8A) and only select one of breakers 1A, 1B, & 3A (appropriate engine’s heated sample line).
3. Select a sample stream from the appropriate engine by turning its valve to “ON” cabinet #1, ensure all other streams are selected to “OFF”
4. Check heated enclosure temperature is set on Ogden dial to 190°C – for all engines.
5. Select the analog output path in Cabinet #2 by sliding all switches to left (Ricardo) or right (SCRE)
   a. Turn on the Ratfisch FID: Press “Power”
   b. Press “Heater-oven” button
   c. Wait 10 minute for the FID oven temperature to reach at least 150 C before attempting ignition
   d. Hold “H₂ Over” button and adjust FUEL to 0.5 bar and AIR to 0.8
e. Continue holding “H₂ Over” and press “Ignition” button - hold both buttons until the ignition light (button) goes out.
f. If ignition does not occur quickly (<20 secs), try switching to “cal gas” and repeat. If the bench has not been used for several days, it may be necessary to purge the H₂ fuel line (remove/replace the fuel hose located on the back of Ratfisch inside cabinet #1) and repeat (e).

6. Plug in NOx vacuum pump (in Fume Hood)

**Part 2 - Calibration**

1. Ensure cabinet #2 has warmed up for at least 1 hour.
2. Ensure the Ratfisch is ignited for at least 40 minutes prior to calibration (other analyzers can be calibrated in the meantime).

**ZERO CH₄ (Ultramat 22P) and NOx (API) cabinet #2**

(NOTE – NOx currently disabled on Ultramat 22P)

- Turn NO or CH₄ switches to ZERO (linked)
- Wait 10 minutes (or for reading to stabilize)

For API NOx analyzer:

- Press ‘TEST’ until sample flow rate shows at top of screen
- This should read 289-295 cc/min
- If flow-rate requires adjustment - use black regulator in back of cabinet #2. CW to increase flow (very sensitive) and CCW to decrease flow.

- **Memorize flow rate**
- press CAL
- press ZERO
- press ENTER

For CH₄ analyzer:

- Press ”>0<“ button to zero the gas and again when zeroing is complete. (Note: during zeroing, flow should read ~2.0 L/min. If it doesn't, correct it using the adjustment knob inside the cabinet).

**CALIBRATE CH₄ Analyzer (Siemens Ultramat 22P) cabinet #2**

Range: 1 - 5V for 0 - 5000 ppm

- Turn NO switch to RUN and turn CH₄ switch to SPAN.
- Check flow rate is 2.0 L/min.
- Adjust the potentiometer so that the display reads 3947ppm.

**CALIBRATE CO2 analyzer (Beckman 880) cabinet #2**

Range: 0 - 5V for 0 - 20%

- Ensure that valve on top of cab. 2 is set to "Ricardo"
- Turn the CO2 switch to zero (top of cab #2).
Check flow rate is 1.0 L/min (2 SCFH), (using "Flow to analyzers" knob).

Press "Zero" then "Enter". Adjust with arrows to read 0% on left of display, and press "Enter" again.

Turn CO2 switch to SPAN.

Check flow rate is 1.0 L/min (2 SCFH).

Press "Span", "Enter". Adjust with arrows to read ~16% on left of display, then press "Enter".

Turn CO2 switch to RUN. Adjust flow rate to 1.0 L/min (2 SCFH) again.

Calibrate (if desired) Low Range CO2 (California Analytical) cabinet #2

Range: 0-10 V for 0-2% (1) or 0-10% (2)

Note: Valves in back of cabinet should be pointed parallel right for low-range operation (intake-dilution CO2) and left for high range operation.

Select range 1 or 2 using knob on analyzer

Turn valves on LHS at back of cabinet # 2 to 'exhaust line' (both handles should point left)

Turn valve on top of cabinet 2 to "SCRE"

Turn CO2 switch to ZERO

Adjust flow rate to 1 L/min, wait 2 minutes

Adjust 'zero' potentiometer so that display reads 0. (can be done while calibrating the Beckman)

Turn CO2 switch to SPAN

Adjust flow rate to 1 L/min

Adjust 'span' potentiometer so that display reads 9.08 / 1.80 (depending on whether-range '1' or '2' is selected)

Turn CO2 switch to RUN

To measure intake/dilution CO2, switch valves at back of cabinet 2 to 'intake'. Plug in separate sample pump.

Set sample flow rate in back of cabinet #2 to 1 L/min

Set the drier air flow rate to 1.5 L/min

Calibrate CO analyzer (Siemens Ultramat 21P) cabinet #2

Range: 0 - 5V for 0 - 10,000 ppm

Turn the CO switch to ZERO.

Press ">0<" button to zero the gas and again when zeroing is complete. (Note: during zeroing, flow should read 2 L/min. If it doesn't, correct it using the adjustment knob inside the cabinet).

Turn CO2 switch to SPAN.

Check flow rate is 2 L/min, (adjust using "Flow to analyzers" knob if necessary).

Adjust the CO pot. So that display reads 2077ppm, (i.e. read 1.039 +/- 0.003V on Chessel display – for this ensure that the SCRE DAQ chassis is on). Note about 5s delay between changing pot and response

Turn CO switch to RUN. Adjust flow rate to 2 L/min again if necessary.
Calibrate O2 analyzer (Oxymat 5E) cabinet #2

Range: 0 - 5V for 0 - 21%

Note: very sensitive to flow rate-ensure proper flow rates for every step

- Turn the O2 switch to ZERO.
- Check flow rate is 0.7 L/min.
- Enter ",.111" to make Code 1 light go out.
- Set analyzer to "Calibration" mode by pressing "Meas/Cal" button.
- Press "5", then press "Enter" to zero the analyzer.
- Wait until "not ready" light is off.
- Turn the O2 switch to SPAN. (Check flow rate is 0.7 L/min).
- Press "8", then press "Enter" to span.
- Wait until "not ready" light is off.
- Press "Meas/Cal" button to set analyzer to Measure mode.
- Turn O2 switch to RUN. Adjust flow rate to 0.7 L/min again if necessary.

Calibrate THC analyzer (Ratfisch RS-55) cabinet #1

Range: 0 - 10 V for: either

- Range 4: 0 - 10 000 ppm (default)
- Range 3: (0 - 1000 ppm) (use values in brackets for calibrating this range)

- Wait 40 minutes after ignition before calibration.
- Select appropriate range.
- Select appropriate calibration gas using the valve at the bottom of cabinet #1.
- Check that ignition light is OFF. If not, follow steps e and f again (under start-up).
- Turn large black knob to "ZeroGas" position.
- Set sample backpressure at 200 mbar, maintain at every step. (sensitive)
- Adjust "Zero" on Gossen display using potentiometer.
- Turn large back knob to "CalGas"
- Set sample backpressure at 200 mbar and turn fuel knob back until Gossen display starts decreasing, then return to the maximum value. (Note: if this value is not ~3.5 on the fuel gauge, instrument needs to warm up more before calibrating).
- Adjust "Gain" pot to obtain a reading of 3947 (253.0 for range3 - note the decimal is burnt out) on the Gossen display.
- Turn large back knob to "Sample" and reset backpressure at 200 mbar.

Check chiller is cold (<3°C) and heated enclosure is 190°C.
Commence sampling from engine.
Balance (set) all flow rates – this is an iterative procedure.

Last: Calibrate NOx analyzer (API) – next page
Calibrate NOx analyzer (API)

Range: 0 - 5 V for 0 - 3000 ppm

- Check Range of instrument – should be 3000ppm for Cummins SCRE and 4500ppm for Ricardo. (Press “set”, “range”, check the range and adjust if necessary, then “enter”).
- Turn NO switch on front of cabinet #1 to SPAN
- Adjust regulator ON THE NOx CYLINDER so that the sample flow rate is exactly what was used to zero the instrument.
- Wait 2-3 minutes NO MORE/NO LESS
- Press ‘SPAN’
- Display should read 1957 ppm NOx
- Press ‘ENTER’
- Press ‘EXIT’
- Calibration (yellow) light should go out and Sample (green) light should go on
- Reset the flow rate to memorized value using black regulator in back of cabinet #2.

FINAL NOTE:

Check flow rates before recording emission data

Ensure that NOx flow-rate does not change more than +/- 1 cc/min (each 1 cc/min affects results ~0.5%)
APPENDIX B. TEST PROCEDURE

1. Start engine
2. Wait for oil temperature and diesel mass to stabilize
3. Run engine up at 12 bar load, 1200 rpm for 15 minutes
4. Record ambient conditions
5. Set desired speed with dynamometer
6. Set desired air flow rate
7. Set appropriate timing
8. Set appropriate equivalence ratio with fuel
9. Set appropriate back pressure
10. Check air flow (will be affected by back pressure)
11. Wait for exhaust temperature to stabilize
12. Take high speed data and data process immediately, check for a reasonable IMEP
13. Record time of day, commanded parameters, oil temperature, and IMEP
14. Take 300 samples (5 minutes) of data
15. Take high speed data, process immediately and check for identical IMEP
   Repeat from 4.
**APPENDIX C. LIST OF ACQUIRED PARAMETERS**

<table>
<thead>
<tr>
<th>Hand Recorded</th>
<th>Data Acquisition System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Date of the test (dd/mm/yy)</td>
<td>IMEP (kPa)</td>
</tr>
<tr>
<td>Time</td>
<td>Post Aftercooler Air Temperature (°C)</td>
</tr>
<tr>
<td>Mode Number</td>
<td>Supercharger Intake Temperature (°C)</td>
</tr>
<tr>
<td>Barometric Pressure [kPa]</td>
<td>Supercharger Exhaust Temperature (°C)</td>
</tr>
<tr>
<td>Relative humidity (%)</td>
<td>Pre-aftercooler air temperature (°C)</td>
</tr>
<tr>
<td>Ambient air temperature (°C)</td>
<td>Exhaust Manifold Temperature (°C)</td>
</tr>
<tr>
<td>PSOI [ms BTDC]</td>
<td>Intake Manifold Temperature (°C)</td>
</tr>
<tr>
<td>PW_Diesel [microseconds]</td>
<td>Exhaust BackPressure (kPag)</td>
</tr>
<tr>
<td>Gas Delay [milliseconds]</td>
<td>Pre-aftercooler Air Pressure (kPag)</td>
</tr>
<tr>
<td>PW_CNG [microseconds]</td>
<td>Post-aftercooler Air Pressure (kPag)</td>
</tr>
<tr>
<td>Engine Oil Temperature (°C)</td>
<td>Engine Speed (RPM)</td>
</tr>
<tr>
<td></td>
<td>Dynamometer Torque (N*m)</td>
</tr>
<tr>
<td></td>
<td>Vector Motor Torque (N*m)</td>
</tr>
<tr>
<td></td>
<td>Diesel fuel temperature (°C)</td>
</tr>
<tr>
<td></td>
<td>Natural Gas Fuel Temperature (°C)</td>
</tr>
<tr>
<td></td>
<td>Diesel Fuel Pressure (Mpa)</td>
</tr>
<tr>
<td></td>
<td>CNG Fuel Pressure (Mpa)</td>
</tr>
<tr>
<td></td>
<td>CNG Fuel Flow (kg/hr)</td>
</tr>
<tr>
<td></td>
<td>Intake Airflow (kg/hr)</td>
</tr>
<tr>
<td></td>
<td>CO₂ - high range (%)</td>
</tr>
<tr>
<td></td>
<td>O₂ (%)</td>
</tr>
<tr>
<td></td>
<td>CO (ppm)</td>
</tr>
<tr>
<td></td>
<td>NOₓ (ppm)</td>
</tr>
<tr>
<td></td>
<td>THC (ppm)</td>
</tr>
<tr>
<td></td>
<td>CO₂ - low range (%)</td>
</tr>
</tbody>
</table>
APPENDIX D. NATURAL GAS PROPERTIES

Table D.1 Properties of the Components of B.C. Natural Gas

<table>
<thead>
<tr>
<th>Compound</th>
<th>Molecular Fraction (%)</th>
<th>Molecular Mass (kg/kmol)</th>
<th>Lower Heating Value (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane</td>
<td>95.945</td>
<td>16.043</td>
<td>50030</td>
</tr>
<tr>
<td>Ethane</td>
<td>1.9549</td>
<td>30.070</td>
<td>47511</td>
</tr>
<tr>
<td>Propane</td>
<td>0.5547</td>
<td>44.097</td>
<td>46333</td>
</tr>
<tr>
<td>i-Butane</td>
<td>0.0689</td>
<td>58.123</td>
<td>45560</td>
</tr>
<tr>
<td>n-Butane</td>
<td>0.1116</td>
<td>58.123</td>
<td>45719</td>
</tr>
<tr>
<td>i-Pentane</td>
<td>0.0252</td>
<td>72.150</td>
<td>45249</td>
</tr>
<tr>
<td>n-Pentane</td>
<td>0.0201</td>
<td>72.150</td>
<td>45345</td>
</tr>
<tr>
<td>Hexane</td>
<td>0.0248</td>
<td>86.177</td>
<td>45103</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>0.4248</td>
<td>44.010</td>
<td>0</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>0.870</td>
<td>28.013</td>
<td>0</td>
</tr>
</tbody>
</table>

The lower heating value is for gaseous components at STP.

The molecular mass and lower heating value of the natural gas are based on the weighted average of the components.
Stoichiometric Air-to-Fuel Ratio

The stoichiometric air-to-fuel ratio is calculated on a mass-basis from the complete combustion of one mole of natural gas with dry air as follows:

\[
\frac{A}{F} = \frac{\text{moles of air} \cdot \text{molecular mass of air}}{\text{moles of fuel} \cdot \text{molecular mass of fuel}}
\]

\[
\frac{(2.038) \cdot (137.36)}{(1) \cdot (16.63)} = 16.83
\]

Natural Gas Summary:

- Molecular Weight: 16.63 kg/kmo
- Hydrogen/Carbon Ratio: 3.924
- Lower Heating Value: 49110 kJ/kg
- Air-to-Fuel Ratio: 16.83
APPENDIX E. CHAUVENET'S CRITERION

Chauvenet’s criterion is a statistical method for determining ‘outlier’ measurements, which are values determined to be anomalous and discarded. This method assumes that the sample distribution is gaussian (or ‘normal’). A reading may be rejected if the probability of obtaining a particular deviation is less than $\frac{1}{2n}$ number of samples, this is expressed by:

$$P(x_i - \bar{x}) \leq \frac{1}{2n}$$

where $n$ is the number of samples, $x_i$ is an individual sample and $\bar{x}$ is the sample mean.

For example if 20 samples are taken, samples would be rejected if:

$$P(x_i - \bar{x}) \leq 0.025$$

When comparing the $z$ value, the standard normal variable, as defined by:

$$z = \left| \frac{x_i - \bar{x}}{\sigma} \right|$$

where $\sigma$ is the standard deviation of the sample.

From normal distribution tables and a probability of 0.025, values may be rejected when corresponding $z$ values are greater than 2.24.
APPENDIX F. SMOOTHED AND UNSMOOTHED HEAT RELEASE

An unsmoothed heat release trace is found in Figure F-1 and the smoothed heat release is found in Figure F-2. The difference is apparent for the pilot combustion event occurring at approximately -16°. The pilot event becomes more distinct with smoothing, albeit more ‘squat’ shaped. The start of natural gas combustion, occurring shortly after -10° is also much more clear with smoothing. The oscillations present in the interval between the pilot and natural gas ignition are likely due to signal noise and not part of combustion. The smoothing algorithm seems to improve clarity of combustion events without affecting the net integrated heat release.

![Unsmoothed heat release trace for mode 7](image)

Figure F-1 Unsmoothed heat release trace for mode 7

![Smoothed heat release trace for mode 7](image)

Figure F-2 Smoothed heat release trace for mode 7
### APPENDIX G. BACKPRESSURE

Table G.1 Instantaneous and Average Intake Manifold Pressures

<table>
<thead>
<tr>
<th>RPM</th>
<th>Manifold IVO (kPa-a)</th>
<th>Manifold IVC (kPa-a)</th>
<th>Manifold $p_{ave}$ (kPa-a)</th>
</tr>
</thead>
<tbody>
<tr>
<td>800</td>
<td>164</td>
<td>171</td>
<td>164</td>
</tr>
<tr>
<td>1200</td>
<td>181</td>
<td>165</td>
<td>179</td>
</tr>
<tr>
<td>1600</td>
<td>184</td>
<td>175</td>
<td>200</td>
</tr>
</tbody>
</table>

Figure G-1  NOx emissions vs. backpressure at 800 rpm, $\phi$ 0.4 and 2 timings
The period of oscillation is similar for the three speeds as shown in Figure G-3, which indicates that the mechanical dynamics of the intake manifold cause the pressure oscillations and not purely gas dynamics. If the oscillation were purely gas dynamic, it should damp out after the intake stroke.
Figure H-1  Nitrogen oxides emissions normalized with fuel for various injection timings and 3 equivalence ratios at 1200 rpm, 4 g/cycle air.
I.1 Variable Pilot Injection Timing

For these preliminary tests, a broad RIT sweep was conducted with small increments by changing the injection timing of the pilot while maintaining a constant gas injection timing. The test conditions were 1200 rpm, at an equivalence ratio of 0.4 and GSOI of -5° ATDC. A rough look at how changing RIT affects emissions is shown in Figure I-1 where there is a local minimum between 1.2 and 3.0 ms. Of note is the decrease in NOx as RIT is shifted negative. The reason for this is the pilot has now been injected well after the natural gas-effecting a change in timing of the combustion.

![Figure I-1](image)

Figure I-1 Comparison of emissions production at φ=0.4, 1200 at various RIT, GSOI held constant at -5° ATDC.
Figure I-2  Standard Deviation of SOC, $\phi = 0.4$, two speeds
Figure 1-3  Particulate matter emissions at various relative timings at $\phi=0.4$, for 1200 rpm
Figure J-1  NOx emissions for two injection pressures, two speeds and various timings.
Figure J-2  In-cylinder pressure trace for GSOI=0, 6.5 g/cycle air, phi=0.4,
APPENDIX K. METHANE AND NON-METHANE EMISSIONS

A methane analyzer was available, however there was an error with the zero and data was not presented for the study. However, the sensitivity of the instrument is reliable. Methane and non-methane emissions for a timing sweep are shown in Figure K-1 and Figure K-2 respectively. Methane emissions increased for the low equivalence ratio. Non-methane hydrocarbons were negative, but showed no change with injection timing for either equivalence ratio. This indicates that the increase in total hydrocarbons found in at late timings is due to unburned methane fuel.

![Figure K-1](image.png)

Figure K-1  Methane emissions for various timings, two injection pressures, two equivalence ratios at 1200 rpm.
Figure K-2  Non-methane emissions for various timings, two injection pressures, two equivalence ratios at 1200 rpm.
APPENDIX L. CRANKANGLE OFFSET EFFECTS

As there is a known offset between the shaft encoder and the crankshaft, the effects on IMEP and heat release should be understood and quantified. To quantify these effects, apparent heat release was calculated with various offsets at an intermediate equivalence ratio, 1200 rpm, and 4 g/cycle of air. It is shown in Figure L-1 that the offset affects the apparent heat transfer from the cylinder (before combustion, near TDC), where an offset of 1.2 °c.a. (encoder signal is too early and thus the offset retards crank angle values) shows no heat transfer. There must be some heat transfer as the in-cylinder temperature is roughly 900 K and the cylinder wall temperature is approximately 380 K. As such, the offset used of 1.7 °c.a. has merit, although the exact magnitude of the heat transfer is unknown and may actually be greater than what is calculated with an 1.7 ° offset.

![Figure L-1](image)

Figure L-1 Apparent heat release near TDC for various offsets at 1200 rpm, PSOI -2°, GSOI +10°, \( \phi = 0.4 \), air = 4g/cycle, \( \gamma = 1.32 \)

How specific heat ratio affects heat release was only examined near TDC to determine if it strongly affects the apparent heat transfer from the cylinder. The sensitivity of apparent heat
release near TDC to different specific heat ratios (\(\gamma\)) is shown in Figure L-2. The shape of heat release are similar and the value of heat release at TDC is identical for each offset. According to equation 3.5, changes in the specific heat ratio will obviously affect the value of apparent heat release. A specific heat ratio of 1.30 was used for heat release calculations for combustion events and any error due to an incorrect specific heat ratio will be consistent for all plots in this study.

Figure L-2  Apparent heat release for various offsets at 1200 rpm, PSOI -2\(^\circ\), GSOI +10\(^\circ\), \(\phi = 0.4\), air = 4g/cycle, \(\gamma = 1.35\)
The effect of the offset on the apparent heat release throughout combustion is shown in figure L-3. The shape of the heat release for the main combustion event relatively unaffected, but the magnitude of the peak heat release decreases with increasing offset.

![Figure L-3](image)

Figure L-3  Apparent heat release for various offsets at 1200 rpm, PSOI -2°, GSOI +10°, $\phi = 0.4$, air = 4g/cycle, $\gamma = 1.32$

The error associated with changing the offset -0.5 and +0.5 from 1.7° is displayed in table L.1 for operation at $\phi = 0.4$, 1200 rpm, GSOI = +10°. Obviously changes in the IMEP will affect the ISFC. This is why the measurement error is so high as compared to the repeatability uncertainty as noted in Table 3.7. Any error in the offset is systematic and will not affect any comparisons between operating conditions on this engine.

| Table L.1 Measurement Offset Sensitivity for 1200 rpm, +10° GSOI. |
|--------------------|-----------------|
| **Parameter**      | **Offset**      |
| IMEP               | +5%             |
| Max HRR            | +2%             |
| Offset             | -5%             |
|                    | -2%             |
APPENDIX M. IMPINGEMENT SIMULATION

M.1 Introduction

To investigate the interaction of the natural gas jet with the piston bowl at different timings, 3 dimensional simulations were conducted using a modified KIVA 3V numerical code. It has been found that a burning jet of methane has a similar penetration to that of an unburned jet\(^1\). As such, the simulations were conducted using methane with no chemistry. The KIVA 3V code solves 3 dimensional turbulent fluid flows using a k-\(\varepsilon\) reynolds-averaged navier-stokes model and is also capable of simplified chemistry. The mesh is block-structured and the geometry, which matches the SCRE engine and injector geometry, was generated by Guowei Li of Westport Innovations. Modifications to allow dual-fuel and constant pressure injections were also made by Guowei Li. The initial conditions were validated. No sensitivity studies were conducted.

M.2 Initial Condition Validation

The progress of pressure was used to validate the initial conditions selected for this study. The simulation begins at -150° ATDC where the initial pressure was matched to the in-cylinder pressure data of the SCRE engine. The simulated air was set to the humidity of air during the experiment. The cylinder wall and cylinder head temperatures were set to the coolant temperature of 80°C. Using an initial temperature of intake manifold plus 25 K, a power regression fit of the simulated pressure-volume data from -150° to 10° ATDC is within 0.001 of experimental data (-1.371 versus -1.3705). This is considered good agreement for validation of initial conditions.

M.3 Settings

The amount of fuel injected was set to match the equivalence ratio in experiments. The experiments in question are equivalence ratio 0.4, 0.5. The conditions simulated were GSOI of 0° TDC for \(\phi\) of 0.4 and GSOI of 0° and +10° ATDC for \(\phi\) of 0.5. The injection pressure was set

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to 19 MPa to match experimental conditions. The injection mode was set to constant pressure injection. Chemistry was not enabled. The standard k-ε constants were employed.

**M.4 Discussion of Results**

The simulations results of slices extracted through the injector centerline are shown in Figure M-1 for \( \phi = 0.5 \). The slice was chosen at 15° after GSOI as that is late in the combustion event, but still during combustion where production of CO is likely to occur. The highest concentration in the legend, which indicates a rich mixture (by definition), was not obtained. This may be due to the fact that Kiva 3V simulations are known to underpredict methane jet penetration and over-mix radially.\(^1\) However, it is very apparent in Figure M-1(a) that there is much more of the richest mixture than in Figure M-1(b). The CO emissions are much higher for a GSOI of 0° than +10° for \( \phi = 0.5 \) (as found in Figure 5.13). Upon visual inspection, the area within the richest contour in Figure M-1(b) is similar to the richest contour found in Figure M-2. When comparing the emissions (in Figure 5.13) for these two cases (\( \phi = 0.5 \), GSOI = +10°, \( \phi = 0.4 \), GSOI = 0°), the CO emissions are within error. This combination of simulation data and experimental CO production concurs with the known correlation between rich fuel mixture and the amount of CO produced. There are essentially two opposing processes, CO production and oxidation. There is likely more interaction of the burning jet with the piston at early timings, however the earlier that the CO is produced, the more opportunity it has to mix and oxidize. This may be why injection at -10° ATDC exhibits lower values of CO than at GSOI of 0°, even though there is more jet interaction with the piston. Increasing injection pressure will also increase mixing, thereby reducing CO emissions, agreeing well with experimental results.

The simulations indicate that there is considerable interaction of the burning natural gas jet with the piston bowl for rich mixtures and early timings. The richness of the fuel mixture that the jet creates at the piston bowl is a likely source of carbon moxide emissions. The simulation trends coincide with the experimental results.

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Figure M-1  Contour plots of methane concentration, 15°c.a. after injection at two timings for φ=0.5.

Figure M-2  Contour plots of methane concentration, 15°c.a. after injection at GSOI of 0° for φ=0.4.