FLOW CHARACTERISTICS OF GAS-BLAST FUEL INJECTORS FOR DIRECT-INJECTION COMPRESSION-IGNITION ENGINES

by

NICHOLAS JOSEPH BIRGER

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Abstract

Natural gas has a high auto-ignition temperature, therefore natural gas engines use an ignition source to promote combustion. The high-pressure direction-injection (HPDI) systems available use small diesel injections prior to the main gas injection. A new series of HPDI injectors have been developed that inject diesel and gas simultaneously through the same holes. In order to understand and control injection and combustion behavior in an engine, it is essential to understand how injection mass is related to the diesel/gas ratio and injection command parameters.

Three prototype injectors are examined. “Prototype B” most closely resembles a standard J36 HPDI injector, but has a modified diesel needle that injects diesel internally into a common diesel/gas reservoir. Prototypes “CS & CSX” have the diesel needle eliminated and replaced with a flow restrictor. The pressure difference between the diesel and the gas controls the quantity of diesel injected. A single pulse width (GPW) for the gas needle controls the fuel quantities.

An injection visualization chamber (IVC) was developed for flow measurements and optical characterization of injections into a chamber at pressures up to 80 bar. Diesel and natural gas are replaced by VISCOR® and nitrogen to study non-reacting flows. A novel feature of the IVC is a retracting shroud that allows the injector to reach steady-state prior to imaging.

For low commanded injection duration (GPW less than 0.60 ms), the relation between GPW and injected mass is non-linear, for all injectors tested. For gas pulse widths greater than 0.65 ms the Co-injectors exhibit approximately linear behavior with higher diesel fuelling quantities lowering gas flow quantities. All Co-injectors are compared to baseline gas flow quantities of a standard J36 to show design difference effects on flow quantities. The sensitivity of gas flow to diesel in injection quantities, as well as the discharge coefficient are computed and theoretically modeled for each prototype. Results suggest differing diesel/gas distributions, dependent on method of diesel introduction and actuator response.

Imaging indicates the mechanical delay of the injectors is independent of chamber backpressure but dependent on fuel supply pressure. However, gas injection quantities are increased by higher chamber backpressure. Changes in the gas/liquid ratio are reflected in different jet image characteristics. These results are compared to theory using an AMESim model developed for an existing production injector.
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Nomenclature

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<thead>
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<th>Abbreviation</th>
<th>Definition</th>
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<tr>
<td>A</td>
<td>Area</td>
</tr>
<tr>
<td>B</td>
<td>Co-injector B</td>
</tr>
<tr>
<td>Bias</td>
<td>Diesel-gas Pressure Bias</td>
</tr>
<tr>
<td>BP</td>
<td>Back Pressure</td>
</tr>
<tr>
<td>ECU</td>
<td>Engine Control Unit p</td>
</tr>
<tr>
<td>c</td>
<td>Speed of Sound</td>
</tr>
<tr>
<td>CD</td>
<td>Discharge Coefficient</td>
</tr>
<tr>
<td>CCD</td>
<td>Charge-coupled Device</td>
</tr>
<tr>
<td>CERC</td>
<td>Clean Energy Research Centre</td>
</tr>
<tr>
<td>CH₄</td>
<td>Methane</td>
</tr>
<tr>
<td>CI</td>
<td>Compression Ignition</td>
</tr>
<tr>
<td>CNG</td>
<td>Compressed Natural Gas</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon Monoxide</td>
</tr>
<tr>
<td>CO₂</td>
<td>Carbon Dioxide</td>
</tr>
<tr>
<td>cᵥ</td>
<td>Specific Heat at Constant Volume</td>
</tr>
<tr>
<td>cₚ</td>
<td>Specific Heat at Constant Pressure</td>
</tr>
<tr>
<td>CR</td>
<td>Compression Ratio</td>
</tr>
<tr>
<td>CS</td>
<td>Co-injector CS</td>
</tr>
<tr>
<td>CSX</td>
<td>Co-injector CSX</td>
</tr>
<tr>
<td>DAQ</td>
<td>Data Acquisition</td>
</tr>
<tr>
<td>DI</td>
<td>Diesel</td>
</tr>
<tr>
<td>DLPR</td>
<td>Dome Loaded Pressure Regulator</td>
</tr>
<tr>
<td>DPF</td>
<td>Diesel Particulate Filter</td>
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<tr>
<td>DPW</td>
<td>Diesel Pulse Width</td>
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EGR: Exhaust Gas Recirculation
f: Frequency
GHG: Greenhouse Gas
GPW: Gas Pulse Width
H: Hydrogen
H₂: Hydrogen Gas
HC: Hydrocarbon
HPDI: High-Pressure Direct-Injection
I2I: Co-Injector A
INJ: Injection
IVC: Injector Visualization Chamber
J36: Westport HPDI™ Injector
LNG: Liquefied Natural Gas
m: Mass
•  m: Mass Flow Rate
N: Nitrogen
N₂: Nitrogen Gas
NG: Natural Gas
NO: Nitrogen Monoxide
NO₂: Nitrogen Dioxide
NOₓ: Oxides of Nitrogen
O: Oxygen
O₂: Oxygen
OH: Hydroxide
P: Pressure
PM: Particulate Matter
Q: Flow Rate
R: Gas Constant
Rs: Sensitivity Value
Re: Reynolds Number
RH: Rusty Hut
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tr>
<td>S</td>
<td>Chamber Pressure Value</td>
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<tr>
<td>SCR</td>
<td>Selective Catalytic Reduction</td>
</tr>
<tr>
<td>SCRE</td>
<td>Single Cylinder Research Engine</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
</tr>
<tr>
<td>t</td>
<td>Time</td>
</tr>
<tr>
<td>TDC</td>
<td>Top dead centre</td>
</tr>
<tr>
<td>tHC</td>
<td>Total Unburned Hydrocarbons</td>
</tr>
<tr>
<td>UBC</td>
<td>University of British Columbia</td>
</tr>
<tr>
<td>uHC</td>
<td>Unburned Hydrocarbons</td>
</tr>
<tr>
<td>V</td>
<td>Volume</td>
</tr>
<tr>
<td>γ</td>
<td>Specific Heat Ratio</td>
</tr>
<tr>
<td>η</td>
<td>Liquid Mass Fraction</td>
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<tr>
<td>ρ</td>
<td>Density</td>
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<tr>
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Most importantly, I would like to express the deepest admiration to my family for their unconditional love and support. My heart goes out to Mom, Roman, and Ivan.
Chapter 1 - Introduction

The abundance, low cost and low-carbon content of natural gas make it a desirable fuel. It is currently used mostly for space heating, power generation, and industrial processes (US EIA 2009), functions which can make use of a broad range of alternative fuels. In recent years natural gas has sparked interest in being a viable fuel source for heavy-duty on-road vehicles. Diesel-fueled compression-ignition engines presently power the overwhelming majority of these vehicles and are subject to ever more stringent emission and fuel economy requirements. Future requirements will undoubtedly stretch the diesel-fueled compression-ignition engine to its limitations on emissions and fuel economy thus novel methods of cleaner, more sustainable propulsion will need to be understood and developed.

1.1 Limitations of Diesel Engines

The diesel engine is amongst the most reliable and efficient prime movers available today. Its high thermal efficiency has helped if become the global standard in the on and off-road transportation sector, the marine industry, and stationary power generation. In fact, the diesel engine’s widespread use in the world accounts for approximately 22% of global crude oil use (US EIA 2009). This energy use produces a substantial amount of emissions into the environment.
Since the invention of the diesel engine it has been plagued with inherently high NOx and PM emissions. In recent years these emissions have been directly linked to serious health concerns, environmental damage, and global warming. Fortunately, due to ever advancing technology, methods exist to greatly reduce these harmful emissions. Modern diesel engines produce extremely low emissions and burn less fuel than their older counterparts. Recent government legislation in North America and Europe both introduced extremely tight regulations for new diesel engines (DieselNet 2009). This has forced both engine manufacturers and operators to focus much of their resources on meeting emissions targets.

The modern methods of emission reductions are combinations of both older known technologies and radical new ideas. Everything from selective catalytic reduction techniques to advanced exhaust gas recirculation coupled with state-of-the-art fuel injection systems and particulate filters are now widely available. Modern compression ignition engines have even strayed from using conventional diesel to mixtures of biofuels, ultra-low sulphur diesel, and even dual-fueled diesel-natural gas engines are becoming increasingly popular. NOx and PM emissions from current engines are reduced enormously from levels found even a decade ago. However, many of these techniques have a trade-off between lower emissions and increased fuel consumption. Increased fuel consumption undoubtedly produces a chain reaction of greater fossil fuel demand and higher CO₂ emissions. Therefore, current and future research should focus on studying methods of simultaneously reducing emissions, fuel consumption and, in turn, CO₂.
1.2 Diesel Engine Emissions

Diesel engines are inherently more efficient than their gasoline counterparts primarily due to combustion occurring with highly lean of stoichiometric combustion at low loads. This leads to lower CO emissions than gasoline engines but also presents some negative issues. Two major problems associated with diesel engines are the high production of NOx and diesel particulate matter (soot), from locally fuel-rich regions, that is readily visible as black smoke in the exhaust stream.

NOx from combustion processes are classified as being a composition of both NO and NO₂ - typically in proportions of 90% NO and 10% NO₂. The main sources of nitrogen in NOx formation is from the combustion air, though trace amounts of nitrogen are also present in fuel.

The most important mechanism in NOx production is that of thermal NOx. Diesel engines operate in highly lean mixtures overall and the amount of power produced each cycle is directly controlled by the amount of fuel injected. Combustion of this fuel locally occurs near stoichiometric. For combustion close to stoichiometric a high combustion temperature can be expected. Formation of thermal NOx and combustion temperature are linked through the Extended Zeldovich Mechanism. The Zeldovich Mechanism predicts that at high temperatures both molecular oxygen (O₂) and molecular nitrogen (N₂), present in the combustion air, disassociate to their atomic states (O and N) and react to
produce NO and atomic O or N (Heywood 1988). The Zeldovich Mechanism was later extended by Lavoie, et. al. to include the combustion radical OH reacting with atomic nitrogen to produce NO and H. Once the combustion process is complete, rapid cooling of the product gases takes place and this quickly restricts the chemical kinetics of the NOx to relatively high values.

NOx emissions can present some serious environmental consequences. NO₂ and OH in the atmosphere react to produce nitric acid (HNO₃). This nitric acid can then accumulate and eventually precipitate as acid rain causing harm to the natural surroundings. Smog formation and respiratory ailments are also directly linked to high concentration of NOx emissions (US EPA 2009).

Particulate Matter (PM) is primarily composed of carbonous soot although aerosols in the form of tiny oil droplets may also be present. The small solid composition of soot makes the emission unique both in production and classifications. The production of soot is complex but can be understood as a phenomenon of fuel-rich regions in the combustion process. The process begins by fuel-rich regions breaking down in the early combustion stage to a pre-cursor of the form C₂H₂. In early combustion there are both H and H₂ rich regions readily available. C₂H₂ and other combustion radicals form many different two-dimensional chain compounds as combustion continues. The next step involves the abundant hydrogen bonding to form spherical surface growth around the chains. As time continues a lack of oxygen in the fuel-rich region prevent hydrogen in the chain to oxidize and agglomeration of similar spherical particles takes place. Since the
combustion is incomplete in fuel-rich regions these carbon-rich particles escape combustion and react further with any other hydrocarbons or volatile organic compounds, grow larger, and then proceed into the exhaust stream (Heywood 1988).

The size of the final soot particles is dependent on the overall quality of the fuel injection and completeness of the combustion process. PM can range in size from individual particles on the order of nanometers up to well in the micrometer range. Modern regulations are based on size and define several classifications of PM. In particular, PM$_1$, PM$_{2.5}$, and PM$_{10}$ which correspond to particles of less than 1 μm, less than 2.5 μm, and less than 10 μm in diameter.

In recent years PM size has been linked to many serious health risks such as lung disease, asthma, cardiovascular ailments, and even premature death (World Health Organization 2009). The shift from earlier PM regulation based on specific mass emissions to size-based restrictions is founded on much research in recent years on the effects of PM size and penetration in the respiratory tract. In particular, particles larger than PM$_{10}$ are often filtered by the nasal and throat passages and do not pose a substantial health risk. However, in smaller particles of size PM$_{10}$ can penetrate into the lungs and pose significant lung problems and eventual lung cancer (World Health Organization 2009). PM$_{2.5}$ sized particles are known to travel far enough into the lungs to cause problems with gas exchange and further respiratory disease. PM$_1$ particles are particularly serious as they travel far enough in the respiratory process to travel through the lungs and into the blood stream and eventually to the heart causing severe cardiovascular problems, heart
attacks, and ultimately premature death. Taking all of the above into consideration it
should come as no surprise that a sustainable society requires reductions in PM emissions
especially to those exposed to them on a daily basis.

The task of lowering diesel engine emissions are increasingly important in today’s world
as more and more people become aware of the negative health effects of diesel
combustion products. Many methods have been well known and readily available for
decades. However, in the past there was little incentive for operators to use emission
reduction technologies due to the added cost and, in many cases, fuel consumption
increase associated with such techniques. In recent years there has been a drastic increase
in demand for diesel engine emission reduction methods due to increasingly stringent
government regulation, public awareness campaigns, and added operator incentives.

1.3 Natural Gas Compression-Ignition

In recent years natural gas has become more attractive due to its relative abundance and
lower costs compared to petroleum. Natural gas can mix uniformly with combustion air,
greatly reducing PM-forming fuel-rich regions with lower overall combustion
temperatures and subsequently lower NOx emissions (Papagiannakis 2004). More
importantly, natural gas offers the highest energy to carbon ratio of any fossil fuel. This is
especially useful since current diesel emission technologies such as Exhaust Gas
Recirculation (EGR), Selective Catalytic Reduction (SCR), and Diesel Particulate Filters
(DPF) fail to address CO₂ emissions, and often increase fuel consumption, in turn, raising
overall CO₂ production. A shift to natural gas may also take some strain off depleting oil reserves and bridge the gap to renewable energy.

The biggest concern in using natural gas in a compression ignition engines (CI) is generating high enough temperatures in the combustion chamber for natural gas autoignition—occurring at 1100-1200 K (Naber et al. 1994). This problem has been overcome through a system developed by Westport Innovations (Westport 2009). The system works by utilizing existing heavy duty truck engine blocks from Cummins and using specialized HPDi™ dual-fuel injectors. The HPDi™ uses a dual-needle, dual actuator fuel injector as shown in Figure 1.1. With this system, small quantities of diesel auto-ignite while the main injection of natural gas is underway. The present generation of HPDi™ injectors has been developed for more than 10 years, and the emission and combustion characteristics of “conventional” HPDi™ are reasonably well understood (McTaggart-Cowan et al. 2003).

Figure 1.1: Westport J36 dual-fuel injector (From Brown 2008).
The Westport HPDITM system replaces over 90% of diesel (by energy content) with natural gas. Due to the cleaner combustion of natural gas, NOx reductions of 40-50%, PM reductions of approximately 80%, GHG reductions of 20-25%, with the same or better thermal efficiency, and similar engine power and torque have been verified (Dumitrescu 2000). The technology is growing rapidly in regions such as Australia where it has been successfully adapted to long haul truck routes.

The present Westport system has certain limitations that will be addressed in this work. Injection timing and complete combustion are especially important in such a system as any CO₂ savings can easily be mitigated by high CH₄ emissions – a more potent greenhouse gas. The current production injector only works on heavy duty truck cycles and simplification and reduction of engine size is of top priority for successful transition to further market adaptation.

As a result, the “Co-injector” has been developed by Rogak et al. (2008) as a simpler way of implementing the HPDITM strategy, using only a single needle and a single actuator. It is expected, if successful, that the Co-injector will result in a simplified, lower cost injector. The research prototypes in this work were built by modifying existing HPDITM injectors. However, these injectors should closely model the behavior of the simpler injector that is ultimately desired.

In co-injection, a small quantity of diesel fuel enters the injector above the needle seat and is carried into the combustion chamber when the gas injection starts. The resulting 2-
phase, transient flow is very complex, coupled to the complex mechanical behavior of the injector itself, which in this case uses electro-hydraulic actuation.

In order to interpret engine experiments properly, it is necessary to understand how actual injection mass and duration depend on fuel pressure, cylinder pressure and commanded injection duration. Finally, it is desirable to know whether some conditions lead to poor atomization or otherwise abnormal sprays. For these reasons, an Injector Visualization Chamber (IVC) was developed.

1.4 Objective and Scope

Previous Co-injector development focused on engine experiments conducted on UBC’s Single Cylinder Research Engine (SCRE) by Brown (2008) and continued by Laforet (2009). The work herein is the first thesis attempting to understand co-injection operation in a controlled, non-combustion setting, complete with theoretical simulations.

The IVC was developed for optical and flow characterization of injections into a chamber at pressures of up to 80 bar. The fuel supply system was constructed for precise control of injector fueling and injection timing. By varying injection parameters flow measurements as well as qualitative flow visualization allow each co-injector to be uniquely examined. By using the systems developed, the work herein sets out to accomplish the following objectives:
• Flow characterize flow rates of each prototype Co-injector under parameters that closely resemble parallel engine work.

• Understand which parameters influence Co-injector operation and in which ways.

• Compare experimental results to theoretical simulations using fluid mechanics and an existing Westport model to analyze any differences.

• Use qualitative flow visualization methods to study injector operation and spray patterns as well as their possible influence on engine operation.

1.5 Thesis Structure

This thesis consists of five chapters and additional appendices where required. Chapter 1 introduces the issues surrounding diesel and natural gas engines, as well as stating the objective and scope of the work. Chapter 2 gives a background of previous work done in similar fields as well as the motivation for the current study and how it differs. Chapter 3 describes the apparatus, their requirements, and their subsequent operation. Chapter 4 outlines the experimental procedures and all of the results accumulated from the various modes of testing and theoretical findings, whilst providing a discussion and interpretation of this data. Lastly, Chapter 5 presents the conclusions and gives suggestions for future work.
Chapter 2 - Background and Motivation

Diesel fuel injection has evolved greatly over time, with every new technique providing subsequently lower emissions, better fuel economy, and greater efficiency. Today’s advanced diesel fuel injection methods mainly rely on high-pressure common rail systems. These common rail systems rely on effective high-pressure fuel pumps to pressurize the fuel to upwards of 2,000 bar and store it in an accumulator-type common rail. Electronically controlled solenoid valves give precise control over the injection parameters that are ultimately controlled by the ECU. The high pressure present in the common rail provides fine atomization of the fuel and better combustion (Mahr 2002), improvements in cold starting, as well as the ability to successfully ignite alternative fuels used in conjunction.

2.1 Dual-Fuel Compression Ignition Engines

The concept of using dual-fuels in compression ignition engines is not new (Wood 1983; Taylor 1985; Song et al. 1987; Alla et al. 2000). The primary benefit of dual-fueling is replacement of much of the diesel fuel with cleaner and more abundant fuels, such as natural gas, while maintaining diesel-like efficiency with low emissions (Douville et al. 1998; Dumitrescu et al. 2000; Harrington et al. 2002; Duggal et al. 2004).

Many dual-fuel engines rely on fumigation, whereby a low-cetane number gaseous fuel (commonly methane or propane) mixes with charge air prior to induction into the engine
cylinder. This gaseous fuel/air mixture then requires an ignition source, which often entails a pilot direct-injection of diesel or other higher-cetane number fuel; similar to that of a standard direct injection diesel engines (Beck et al. 2003). Dual-fueling allows most of the diesel (80% or more by energy content) in a conventional engine to be replaced with a smaller-chain hydrocarbon fuel that results in less CO₂ combustion products to form. NOₓ and PM emissions can also be reduced through appropriate fueling methods.

Since a gaseous fuel displaces some of the charge air, this method of dual-fueling reduces the volumetric efficiency of the engine. High hydrocarbon emissions are often a problem as rapid scavenging from 2-stroke engines or valve overlap of 4-stroke engines may lead to portions of the unburned fuel/air mixture being expelled directly to the exhaust stream. Moreover, lower loads may suffer from lower thermal efficiency and higher emissions (Alla et al. 2000).

For these reasons, it is advantageous to inject the gaseous fuel at high pressures directly into the engine cylinder, late in the compression stroke - analogous to the benefits provided by a conventional high-pressure direct injection engine.

2.2 High-Pressure Direct Injection of Natural Gas

High-pressure direct injection operates by injecting natural gas into the combustion cylinder of a CI engine near the end of the compression stroke. The charge temperature in the combustion chamber at this point is still lower than the required autoignition
temperature of natural gas (Naber et al. 1994). To compensate, a small pilot injection of
diesel is injected before the natural gas. This diesel rapidly ignites and sufficiently raises
the in-cylinder temperature to natural gas autoignition levels. High-pressure direct
injection of natural gas has some key advantages. Direct injection allows the fuel rate to
control load thus eliminating throttling losses. Correspondingly, the engine can run at
higher compression ratios resulting in high thermal efficiency. Since natural gas burns
cleaner and replaces a majority of diesel, lower emissions are possible.

Diesel engine fueling with high-pressure direct injection of natural gas was first
examined by Miyake et al. (1983). Two methods of high-pressure direct injection were
presented that are both based on diesel as the preliminary ignition source. The first
method relied on a diesel pilot injection ignition source followed by a high-pressure
natural gas injection (similar to the principle behind current Westport HPDI™
technology) in a large 4-stroke diesel engine (420 mm bore). Stable engine operation was
maintained at 85% and 100% of full load using a 5% diesel-to-natural gas ratio while
maintaining equivalent thermal efficiency. The second method relied on a mixture of
diesel and natural gas to be discussed in Section 2.2.

Wakenell et al. (1987) used a two-cylinder locomotive research engine redesigned for
natural gas fueling. They showed that by using late-cycle high-pressure direct injection
with diesel pilot the research engine performance resulted in matching rated speed and
power with slightly lower thermal efficiency than with pure diesel operation. The diesel
fueling rates were lowered to 1% of fuel energy while maintaining full power. At very
high natural gas to diesel fuel ratios audible knock became apparent and subsequently diesel amounts were increased. At diesel fueling amounts of approximately 10% stable, knock-free operation was maintained though with lower thermal efficiency. Of particular importance was the noted unstable operation at part load motivating a closer examination of injection timing parameters and their importance on combustion in high-pressure direct-injection.

Hodgins et al. (1996) and Douville et al. (1998) compared an early natural gas HPDI system with diesel-fueling of a two-stroke engine. With natural gas fueling, NOx emissions were reduced by 20-40%, CO emissions may be lowered by 15-20%, and PM emissions were almost unchanged. The thermal efficiencies were almost identical at low to medium load but were higher for natural gas fuelling at high loads. CO and PM emissions were lower at high loads than conventional fueling. Natural gas fueling had higher methane emissions at low loads but can be lowered with system refinement.

The overall viability of high-pressure direct injection of diesel pilot and natural gas as a stable and effective fueling method has been confirmed repeatedly. (Dumitrescu et al. 2000; Harrington et al. 2002). It was shown that when small quantities of diesel were injected before a natural gas main injection diesel-like thermal efficiencies were achievable over a wide range of engine load while substantially reducing NOx, PM, and CO₂ emissions in both two and four-stroke engines.
Present day, commercialized HPDI technology is available for a variety of trucking applications using the Westport GX engine system (Westport 2009). This system combines the Westport J36 HPDI injectors, with a 15 L Cummins ISX engine block utilizing EGR, and a specially designed fuel delivery system based on conditioning LNG. Duggal et al. (2004) summarized the complete systems results of the HPDI system installed on a Cummins ISX engine with EGR. The same torque, power, and efficiency were maintained while lowering NOx emissions to 0.6 g/bhp-hr and PM to 0.03 g/bhp-hr with an overall reduction of GHG.

2.3 Co-Injection of Diesel/Natural Gas

2.3.1 Prior Work

Using gas-assisted atomization of liquid fuel in compression-ignition engines can be dated back as far back as 1893 (Stone 1992). Rudolf Diesel used compressed air to assist atomization of diesel for spontaneous ignition in a pressurized combustion chamber. More recently Sovani et al. (2001) studied effervescent atomization whereby gas is mixed internally with a liquid to improve the quality of atomization – similar to co-injection. The main benefits of effervescent atomization are a decrease in the required injection pressures, larger injector orifices, and lower injection rates that still result in finely atomized liquid droplets (Sovani et al. 2001). These are all of direct benefit to engine and injector development. The fundamental process in effervescent and gas-assisted atomization depends on appropriate liquid and gas mixing that results in two-
phase flow with a lower speed of sound than the corresponding pure gas flow (Sherstyuk 2000). This lower speed of sound results in a lower velocity for choked flow. As such, a two-phase mixture with fixed-pressure fuel supplies will have a larger pressure gradient across the flow area than pure injections of liquid or gas at similar injection pressures as shown in Figure 2.1. Larger pressure gradients are fundamental to better atomization of the fluid. Moreover, the gas, in some instances, can act to reduce the effective orifice size by forcing the liquid into a tighter annular flow. Upon exiting the orifice, the gas will rapidly expand causing the liquid flow to break apart and form smaller droplets (Sovani et al. 2001).

Figure 2.1: A two-phase flow chokes at lower velocities and will experience a higher pressure gradient across the nozzle exit increasing atomization quality (From Sovani et al. 2001).
In practice, using natural gas to assist atomization of diesel for compression ignition engines was investigated many years ago with no apparent market adaptation to date (Miyake et al. 1983). In that work, stable operation of a large diesel engine (420 mm bore) with slightly longer ignition delay than separate diesel/gas injections was observed using a relatively simplified injection system of premixed diesel and natural gas. Hill et al. (1991) and Chepakovich (1993) used a natural gas blast to assist diesel atomization in a pre-chamber. Yang (2002) used a separate mixing chamber, where diesel is first injected followed by a gas injection to atomize the diesel and the subsequent mixture is then injected into the combustion chamber. The system uses relatively low pressures of 15 - 45 bar that limit the points of injection in the diesel cycle.

2.3.2 Current Work

Most recently (2005 to present), there was a renewed interest in diesel/natural gas co-injection by Rogak et al. (2008) as a means of creating a simplified injector for the HPDI strategy. A prototype Co-injector was created (Prototype A) by modifying an existing Westport J36 injector such that diesel pilot is injected internally into the common gas reservoir (detailed apparatus description is given in Section 3.1). The resulting diesel-gas two-phase mixture is then injected into the combustion cylinder. Figure 2.2 shows the distinction in this method compared to the other dual-fuelling strategies presented. Understanding the combustion process of such an injection becomes inherently more complex than a pure liquid diesel injection.
The first work performed to understand Prototype A combustion behavior was by Jones in 2005 using the UBC Single Cylinder Research Engine (SCRE) (Jones 2005a; Jones 2005b). Initial Co-injector engine results indicated that Prototype A required higher diesel fuelling rates than similar J36 operation. Moreover, engine knock was observed at some of the higher diesel injection rates. Early emissions results showed similar levels as the J36 except low engine loads where CH₄ and CO were higher. PM emissions were lower in most test cases suggesting the diesel was may have been more finely atomized or other combustion effects were present.

In 2006 it was proposed to run Prototype A using double pulse tests. In these tests a shorter "pilot" co-injection was first sent followed by a main co-injection. The short first duration was desired to be mostly diesel followed by a mainly gas second injection, though exact distributions between pilot and main injections was unclear. The double
pulse tests showed the Co-injector could produce similar NOx emission as the J36 injector under combinations of load and speed whilst maintaining similar thermal efficiency if appropriate quantities of diesel are used (Jones 2006). PM emissions were significantly lower than the J36 at high engine loads and could be reduced further with lower diesel fuelling rates up to a limit of 12 mg/injection when misfires began to occur.

McTaggart-Cowan (2006) continued Prototype A work at low engine loads using single injection tests to determine the combustion stability and ways of lowering high uHC emissions at low loads. It was found that these emissions could be reduced at low load operating conditions by increasing manifold pressures, increasing the diesel fuelling amounts, or reducing the gas supply pressure. More importantly, the amount of gas injected was significant in effecting both emissions and combustion stability. If higher gas fuelling rates are used, emissions increase and combustion stability decreases. McTaggart-Cowan suggests that diesel consumption may be lowered by reducing the gas injected. Therefore, it becomes important to study injector control parameters, as well as the precise interaction between diesel and gas and their result on injection quantities, as both have an effect on emissions and combustion stability.

Through all the engine experiments it became apparent that understanding diesel-gas interaction during injection is critical. For a constant diesel fuelling rate, Brown (2008) showed double-pulse operation of Prototype A increased ignition delay with a corresponding decrease in heat release. This is may be due to less diesel being injected in the first pulse with remaining diesel carrying over to the second injection. Less diesel in
the first injection provides less of an ignition source and the overall combustion dynamics rely on significant amounts of diesel igniting in the second injection.

In an attempt to increase the amount of diesel injected in the first pulse, Prototype A was modified in 2007 by adding an internal sleeve to the existing prototype thus creating Prototype B. The sleeve acts to reduce the diesel/gas reservoir volume whilst increasing the gas flow velocities through it. It was thought that higher fluid velocities through the reservoir would allow diesel to accumulate nearer to the tip, with correspondingly higher gas velocities scavenging the diesel more rapidly into the first pulse. With this added sleeve, Prototype B showed significant improvements over Prototype A as measured by Brown (2008). The ignition delay was reduced at moderate and low engine speeds. The amount of diesel required for stable operation also reduced by up to 20%. CO emissions were reduced moderately but NOx increases slightly due to the shorter ignition delay and slightly higher combustion temperatures. uHC and combustion stability were mostly similar to that of Prototype A tests.

From September 2007 to December 2009 Laforet expanded on the engine testing performed by Brown on Prototype B – in parallel with much of the work herein. Furthermore, an additional single-needle prototype (Prototype CS) was introduced that more closely resembled the desired injector (detailed description in Section 3.1.2). This injector allowed diesel to continuously flow into the diesel/gas reservoir. Laforet (2009) showed that double injection testing of Prototype CS was able to match or outperform Prototype B, using even lower diesel fuelling quantities. Under certain low pulse width
conditions this prototype was even able to outperform the J36. Heat release data of Prototype CS showed that more diesel was being injected and igniting in the pilot injection, suggesting diesel may be accumulating nearer to the tip, as desired.

2.3.3 Early Flow Characterization

Flow characterization of production injectors is of particular importance to Westport as it allows them to monitor variability and quality of injectors. As such, two Westport flow benches, the EFS1 and BTR2, are used to monitor flow parameters (Inokoshi 2007; Brown 2008). The EFS1 flow characterization rig allows up to 6 injectors to be connected to a common rail fuel supply and simultaneously flow tested with diesel and natural gas into a constant pressure environment. This allows the flow rate variability to be measured and compared for multiple injectors connected to the same common rail. The Westport BTR2 is used for quality control of all production injectors. It allows VISCOR® and nitrogen to be independently fired into a static backpressure of 80 bar. Each production injector is tested in this rig to determine whether simulated diesel and gas flow rates are within specified production limits. Flow rates in both rigs are determined by coriolis mass flow meter on the gas side and gravimetrically for diesel.

The Westport flow benches were used by Inokoshi of Wesport Innovations with the prototype Co-injectors in 2007 and 2008 (Inokoshi 2007). The EFS1 was used to attempt to understand the diesel/gas interactions and their effect on flow rates. Co-injector B was inserted with five blanks and VISCOR® replaced diesel to reduce flammability hazards.
The GPW region of interest from McTaggart-Cowan (2006) single-injection tests was used in these tests. Table 2.1 summarizes the test parameters for EFS1 testing. Test points were accumulated by using a full factorial of DPW and GPW (23 in total, with DPW = 0 ms and GPW = 0.45 ms omitted). Diesel flow rates were taken by averaging the change in mass of the diesel fuel supply quantity over 10-20 minutes, whilst gas flow was sampled at 1 Hz over 100 s.

<table>
<thead>
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<th>Test Parameter</th>
<th>Value</th>
</tr>
</thead>
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<td>Speed</td>
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<tr>
<td>Chamber Pressure</td>
<td>60 bar</td>
</tr>
<tr>
<td>Pulse Separation</td>
<td>1.0 ms</td>
</tr>
<tr>
<td>Gas Rail Pressure</td>
<td>23.0 MPa</td>
</tr>
<tr>
<td>Diesel Rail Pressure</td>
<td>24.2 MPa</td>
</tr>
<tr>
<td>Bias</td>
<td>1.2 MPa</td>
</tr>
<tr>
<td>GPW</td>
<td>0.45, 0.50, 0.55, 0.60, 0.65, 0.70 ms</td>
</tr>
<tr>
<td>DPW</td>
<td>0, 1.0, 1.5, 2.0 ms</td>
</tr>
</tbody>
</table>

Table 2.1: Test matrix for EFS1 flow bench testing of Co-injector B. The fluids used were VISCOR® and natural gas.

Figure 2.3 shows the flow rates from these experiments. There is a clear region of nonlinearity present in the region GPW = 0.50 – 0.65 ms. In this region the flow rates for both diesel and natural gas are independent of GPW. This region is of particular importance in understanding Co-injector operation since it is used in much of the low-load single-injection testing and suggest ambiguous fuelling rates. In the region GPW = 0.65 – 0.70 ms the injector behaves as anticipated with diesel fueling amounts directly affecting the gas delivery. At GPW = 0.45 ms the fueling rate drops considerably to very low fueling amounts. The test points were only run once and some of the odd trends (such as flow curves that cross) were quite uncertain and a partial motivation for the present thesis.
Figure 2.3: Gas injection mass as a function of Gas Pulse Width for Co-injector B on the Westport EFS1 flow bench (From Brown 2008).

The BTR2 rig was used to compare gas-only flow rates of Co-injector A and B to give early indications of the flow differences from the added sleeve. Table 2.2 summarizes the test parameters for this series of tests. 11 GPW points were selected in the range 0.50 – 3.00 ms. It should be noted that unlike a standard J36 injector where diesel and gas are separately injected, diesel flow rates could not be independently tested in the BTR2 because the Co-injector injects diesel internally. In turn, only gas flow rate comparisons were made.
<table>
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</tr>
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<tr>
<td>Pulse Separation</td>
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<td>Bias</td>
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</tr>
<tr>
<td>GPW</td>
<td>0.50 – 3.00 ms</td>
</tr>
<tr>
<td>DPW</td>
<td>0 ms</td>
</tr>
</tbody>
</table>

Table 2.2: Test matrix for EFS1 flow bench testing of Co-injector B. The fluids used were VISCOR® and nitrogen.

Figure 2.4 shows a comparison of gas flow rates for Co-injector A and B. The sleeve added to Co-injector B lowers the flow rates by 8 - 26%. Brown suggests these flow rate reductions may be from friction losses by the sleeve restricting the flow areas in the diesel/gas reservoir. The resolution at lower GPW is insufficient, particularly that of the region of interest in the EFS1 tests, to distinguish precise injector response. Moreover, the diesel interaction does not affect the gas fuelling through this flow characterization method. The plotted grey lines indicate the variability tolerances for production Westport J36 injectors. The gas flow trends for both Co-injector A and B are within the limits suggesting the gas needle and actuator response is within Westport production limits.
The EFS1 and BTR2 tests show the importance of flow testing injectors, as there are regions with unclear operational characteristics. The EFS1 results of Figure 2.2 show that diesel interaction has a profound effect on overall fuelling quantities injected and must be understood in greater detail for successful Co-injector development. The Single Cylinder Research Engine can measure flow rates, but is too cumbersome and complex for flow tests. Therefore, it became necessary to develop a system for flow testing - most preferably with engine-like conditions - so Co-injector flow characteristics can be studied. These flow-testing results motivate a critical element of the work performed herein.
2.4 Spray Visualization

The quality of diesel fuel atomization is of great importance in the overall of combustion process and emissions performance of any diesel engine. In particular, finer atomization allows for better fuel/air mixing and promotes more complete combustion, resulting in better engine performance and lower emissions (Lefebvre 1989, p. 107-110). Therefore, it is of importance to understand injection spray characteristics and which conditions promote complete combustion and low emissions, especially in relatively new technologies such as dual-fueling. Unfortunately, due to the high pressure and dynamic nature of engine processes understanding fuel flow is not a straightforward task. Many mechanisms of fuel spray formation such as jet break-up, particle size distributions, penetration lengths, and time evolution in such environments are still not fully understood (Shao et al. 2003). Fortunately, much progress is being made in spray analysis and characterization and a variety of methods exist.

Optical spray characterization has long been a staple of diesel engine development due to its non-intrusive nature. Hiroyasu and Arai (1990) set the foundation for much of today’s work by using photographic imaging to quantify many jet characteristics of diesel sprays in relatively high-pressures. In their work they used a constant volume chamber with optical access capable of 30 bar ambient pressure. Diesel was injected at pressures up to 100 MPa into varying chamber pressures and parameters such as break-up length, spray angle, spray tip penetration length, and droplet size were measured and empirical equations derived.
Naber and Siebers (1996) studied the effects of high-pressure diesel injection into ambient gas of varying density and vaporization. Diesel spray characteristics in a combustion-prone environment were of interest. Their optical vessel allowed for high ambient pressure (up to 35 MPa) and allowed the walls to be heated to engine-like temperatures of 525 K. Their results showed that ambient gas density and vaporization have profoundly larger effects on spray penetration than previously thought, setting the foundation for further work.

Shao et al. (2003) and Delacourt et al. (2004) show that photographic techniques using modern CCD technology and image processing demonstrate a powerful method for quantitative spray analysis and characterization of the diesel spray process. Their technique relies on high-speed imaging of high-pressure diesel sprays in optically accessible constant volume pressurized chambers. Images are processed for background and noise adjustments and subsequently placed through computational algorithms to quantify tip penetration and near and far field angles. Injection parameters can be varied to characterize a wide array of different diesel sprays and compare them to theory.

Schlieren and shadowgraph techniques allow visualization of gas jets, and have been used for some of the earlier work on HDPI injection (Hill et al. 1992; Hill et al. 1999). Much of this work focused on the penetration rates of gaseous jets in a constant volume chamber with similar conditions to those found in direct injection diesel engines. Chepakovich (1993) used similar techniques, however also studied two-phase gas
diesel/gas injections. Much comparison was drawn to early models predicting gaseous jet time evolution and, in the cases of two-phase injections theory was expanded through his imaging and measurement techniques.

Sprays with significant vaporization (e.g., gasoline sprays) lead to a more complex characterization problem because ideally two phases must be tracked. Combinations of scattering (for liquid) and Schlieren or LIF imaging (for the vapor) have been reported in (Lahbabi et al. 1993; De Zilwa and Steeper 2005).

For gas-assist atomization, much earlier work focuses on steady sprays at atmospheric pressure (Mansour and Chigier 1995), where windowless open test systems can be used. In fact, optical studies of atmospheric injections of the J36, Prototype A, and Prototype B were conducted at UBC by Mikawoz (2005) and Marr (2007). Mikawoz (2005) developed new techniques of fuel delivery and imaging for flow characterization. Early optical imaging showed that the VISCOR® was finely atomized during Co-injector operation but some conditions showed inconsistent sprays that required further investigation (see Figure 2.5). Marr (2007) showed that the J36 required approximately 50-100 injections before it reached steady-state operation, possibly accounting for some of the inconsistencies. Before reaching steady-state, the injector exhibited variations in the quality of VISCOR® injections. Through Marr’s work it became apparent that studying the injector should be done after reaching steady-state, as this is the regime where the injector will operate the vast majority of its time.
Figure 2.5: Early optical Prototype A co-injection images by Mikawoz (2005). Left - consistent, finely atomized jets. Right - large, inconsistent droplets are circled. Note: These co-injection images were taken well before the 50-100 injections required to reach steady-state.

Though these early Co-injector imaging techniques are useful, it is desirable to expand on them. The actual co-injection process we wish to study herein is transient, involves two phases, and is done into a high-pressure environment. An added complication is that the injector does not reach steady-state before 50-100 injections (Marr 2007). Therefore, new techniques had to be developed to overcome this. In this work, the focus was on tracking liquid droplets because initial interest is in the influence of fuel and chamber pressure on injection timing.
Chapter 3 - Apparatus and Procedures

The experimental work was performed in Rusty Hut 122 at The University of British Columbia. The experimental apparatus consisted of six key components: the injectors, the Injector Visualization Chamber (IVC), the spray extraction shroud, the high-speed camera, the fuel delivery system, and the control systems. The key components and their operation will be described in this chapter and an instrumentation list is provided in Appendix A.

3.1 Injectors

3.1.1 J36 and Co-injector B

The standard HPDI system uses a Westport production model J36 injector and consists of two concentric injectors with diesel introduced in the inner injection system and natural gas in the outer injection system, as shown in Figure 3.1. The injector operates as follows: a pulse of +5 V is initially generated via a LabView controlling program (see Section 3.6) for the duration specified by the GPW (see Section 3.1.3). Typically this value is in the range of 0.55 – 2.0 ms. This signal is then sent to a custom-built Westport injector driver box connected to a 12 V power supply. The injector driver box boosts the pulse voltage to approximately 50 V and the current to 20 A - which is then sent to the injector. The driver box has an inherent signal processing delay of approximately 0.1 ms. The final, conditioned signal triggers the diesel/gas needle actuators.
Figure 3.1: Westport J36 HPDI injector.

Figure 3.2: Schematic of Co-injector B Nozzle.
The actuators use electro-mechanical solenoid valves which open/close for a period based on the signal duration received and complicated needle dynamics. Diesel is used as the hydraulic fluid to ensure the needle is seated (See Figure 1.1 for an HPDI Injector schematic). Once an actuator is triggered an open solenoid valve allows hydraulic diesel to drain via a return line and a spring lifts the needle. The overall needle dynamics are complex and are based on parameters such as the spring force, chamber backpressure, and the internal gas pressure – all acting to overcome the diesel hydraulic pressure. As will be shown, these parameters all play a different role in overall injector flow characteristics.

The injector geometry of the first prototype Co-injector was based on the same J36 injector geometry with the following modifications. The diesel needle was shortened and a “diesel plug” was inserted below it. Diesel is injected through 7 new holes into the gas plenum, which is above the gas needle seat. This first prototype was dubbed “Prototype A” and much of the early co-injection work was based on results from this injector. It is important to note that due to actuator and fuel supply constraints, the diesel must be injected into the plenum over a short duration (2-4 ms) with a driving pressure (“diesel gas bias”) of 0.7 to 2.5 MPa. Although this is far lower than conventional diesel injection pressures, it does result in significant jet velocity, probably enough to partially atomize the diesel and distribute it inside the injector. Therefore, the diesel is not injected before the gas, but rather throughout the pilot gas injection and during part of the second (“main”) injection.
During SCRE relocation in late 2006, Prototype A was modified and the injector was renamed “Co-injector B”. A sleeve was installed on the outside of the gas needle as an attempt to reduce the distribution of diesel within the injector and increase the efficiency with which the gas flow would entrain it. This sleeve acted to change the internal geometry of the Co-injector, reducing its inner reservoir volume by 33% to 35 mm$^3$ (or enough for 30 mg of diesel) (Brown 2008). The minimum annular area of this injector decreased from 30 mm$^2$ to 10 mm$^2$. The annular area then re-expands near the tip back to an area of 30 mm$^2$ in an effort to keep the diesel/gas mixture near the tip. A simplified representation of the Co-injector B nozzle is shown in Figure 3.2 (Page 31). It should be noted that when the gas needle is fully open the proceeding flow is choked at the gas holes of the injector nozzle. However, when the needle initially lifts from its seat, or is near closing, the small lift present will be the limiting area for choked flow as long as the area is less than that of the gas holes.

### 3.1.2 Co-injector CSX and CS

Two further prototype Co-injectors, CSX and CS, were built using J36 bodies but more closely resemble the desired single-needle injector. In these Co-injectors the diesel needle was removed entirely and replaced by Lee Restrictors as shown in Figure 3.3. This allows diesel to continuously leak into the gas plenum, the exact flow rate being controlled by the diesel gas bias. These injectors have roughly half as many moving parts, match fits and actuators as the baseline HPDI J36 injector. Co-injector CS has a nominally identical tip and gas plenum geometry as Co-injector B. Co-injector CSX closely resembles CS but varies in its sleeve geometry, with CSX having a larger sleeve (outside diameter of the
sleeve is increased from 8.2 to 8.4 mm) that reduces the plenum volume and flow area even further. The main differences between Co-injector CS, CSX and B are the internal velocities, duration, and accumulation of diesel flow into the plenum. The flow is again choked at the gas holes of the nozzle when the needle is fully open.

![Schematic of Co-injector CS and CSX Nozzle.](image)

**Figure 3.3: Schematic of Co-injector CS and CSX Nozzle.**

3.1.3 Injector Command Parameters

Under engine requirements all injectors studied use 2 fuel injections per cycle, with the three series of injectors having varying injection parameters. The J36 has two distinctly different injection processes. Pure diesel is injected through its independent diesel needle followed by a main gas injection through the gas needle. In Co-injector B three injections are needed: the diesel pre-injection, pilot gas injection and the main gas injection (Brown 2008; Brown et al. 2009). During the pre-injection, the diesel needle lifts and diesel is injected into the common gas/diesel reservoir. During the pilot and main gas injections a
mixture of diesel and natural gas is injected into the combustion chamber of varying duration. Co-injector CS and CSX operate in a similar manner to B, but the requirement for diesel pre-injection is eliminated. Since diesel is constantly allowed to leak into the gas plenum, these injectors only require two actuations of the gas needle. The co-injectors may also operate off single injection operation whereby the pilot co-injection is eliminated and only a main injection pulse is used (McTaggart-Cowan 2006). Figure 3.4 shows the sequence command injections for the three series of injectors. Figure 3.5 shows the sequence of command injections and the corresponding injector actions.

Figure 3.4: Schematic of Commanded Injections. a) Shows the pulses for a J36 type injector with separate diesel and gas injections. b) Shows the command pulses for Co-injector B. c) Shows the command pulses for Co-injector CS and CSX.
1. Pre-Injection
   Diesel needle lifts and diesel is injected into the common gas/diesel reservoir
2. Gas needle re-seats and there is still diesel in the injector.
3. Pilot Injection
   Gas needle lifts and gas/diesel mixture is injected into the combustion chamber
4. Gas needle re-seats and there is still diesel in the injector.
5. Main Injection
   Gas needle lifts and gas/diesel mixture is injected into combustion chamber.
6. Gas needle reseats and there is very little diesel in the injector.

Figure 3.5: Injection sequence for Co-injector B operation (From Brown 2008).
3.2 Injector Visualization Chamber (IVC)

The IVC was machined from a 27cm x 25cm x 25cm block of steel. Exact machining specifications are shown in Appendix B. Ports on 5 faces can accept windows, an injector holder, or a spray collection shroud (discussed in 3.3). In the configuration shown in Figure 3.6, windows are installed to give a horizontal view of the injector spray. In the summer of 2009 a new shroud mechanism was designed to allow future users a head-on view of injector spray. The IVC windows are designed for a working pressure of 80 bar, which is comparable to diesel engine cylinder pressures during the fuel injection.

Figure 3.6: Injector Visualization Chamber.
3.3 Spray Extraction Shroud

After about 10 injections, enough liquid droplets contaminate the windows (or stay suspended in the chamber) to obscure the view. From previous work it became evident that the Co-injector requires approximately 100 injections to reach stable operation after installation in the IVC (Marr 2007). After this many injections it is impossible to see subsequent injections. To avoid this problem, a retractable shroud mechanism that covers the injector tip was built. The shroud covers and drains the liquid while the injector stabilizes. It can then be retracted rapidly allowing for a clear view of the next 10 injections.

The operation of the shroud is shown in Figure 3.7. When the shroud is closed around the injector tip, the central tube is connected to vent while the annular space is connected to the high-pressure (i.e., near the chamber pressure) supply. The outer tube is made intentionally leaky so that air flows into the main chamber and maintains it at high pressure, and also into the shroud chamber below the injector, maintaining this volume at high pressure as well. The vent and pressure lines are connected to a 4-way valve. When the lines are reversed, the shroud (the outer tube) retracts. A switch is attached to the shroud that indicates when the shroud has retracted sufficiently to image the spray. The camera starts to collect images at the first injection after the shroud-open signal is sent to the computer. Depending on pressure settings the shroud has a retraction time of 1-3ms, less than 1 injection cycle.
The first shroud mechanism, used for the majority of the work herein, allowed horizontal view of the injector spray. As experiments progressed it became apparent that with the horizontal view jet penetration characteristics were difficult to quantify. Therefore, in December 2009 a new shroud mechanism, based on the same movement dynamics was built. By rearranging the old IVC configuration and manufacturing a new angled-shroud that moves from a horizontal port a head-on spray view is available. This will be particularly useful for future jet penetration measurements.
3.4 High-Speed Camera

Images are obtained with a Vision Research model Phantom v7.1 CMOS high-speed camera using a Nikon 50mm 1:1.2 lens. This camera can collect 320 x 240 pixel grey scale (14-bits per pixel) images at a frame rate of 24096 frames per second with an exposure time of less than 31 μs. Back illumination is provided by a 500 W halogen lamp directed at a white paper diffuser. Schlieren or shadowgraph optics were not used and in turn pure gas jets are not visible in the images; only jets with liquid drops are visible. However, it is believed that the drops are small (hence, tracking the gas flow quite well) for the following reasons. Firstly, the gas/liquid ratio is high enough that we always expect dispersed rather than slug flow in the injector nozzle. Secondly, optical particle sizing with a LaVision Sizemaster system found droplets were always less than 15 microns (Marr 2007). The Sizemaster measurements were done on injections into atmospheric pressure, and therefore provide only qualitative indications of behavior at higher pressure. At high pressure, many Marr (2007) experiments with the Phantom camera system show that visible (i.e, 200+ microns) drops only form when the injector is not in steady state.

Raw images have no background subtraction, and as a result there is significant texture to image background. This background (and drops on the windows) may partly obscure the sprays in the still images, but is easily distinguished from the sprays when observing the full movie. The qualitative observations discussed are based on studying the movies as well as the still images included in this work.
3.5 Gas and Liquid Supply and Measurement

The fluid control system is shown in Figure 3.8. Bottled $\text{N}_2$\textsuperscript{1} has been used to model the natural gas fuel and pressurize the IVC. Gas pressure is held at a pressure slightly below the liquid pressure using a GO Inc. Model DL57-111396-E dome-loaded regulator. An Endress + Hauser Promass 80A coriolis mass-flow meter is used on the gas supply. Two gas accumulators (total volume of 1 L) are used upstream of the injector to reduce potential cyclic pressure fluctuations from injector operation.

Figure 3.8: The IVC system schematic.

\textsuperscript{1} Tests can also be performed with lower molecular weight gases to match natural gas more closely.
VISCOR ® 1487 calibration fluid\(^2\) is used to match the density and viscosity of diesel without matching its flammability. VISCOR is supplied by a Bosch Model CP3.3 radial 3-plunger, 3-lobe diesel fuel injection pump; pressure is set with a Tescom Model 26-1762-26 back-pressure regulator. This pump was belt driven by a Marathon Electric Model 56C17F5326 1-phase AC motor. The motor is geared to operate the high-pressure pump at 300 rpm. A JABSCO Model MDXT self-priming pump was used to slightly pressurize the VISCOR prior to the high-pressure pump. A simple heat exchanger was built to keep the VISCOR from heating during prolonged pump operation. Similar to the gas supply, slightly upstream of the injector a Flowguard Model PD-0100 pulsation dampener is used to help reduce pressure fluctuations. Swagelok Model SS-4TF-15 15 micron filters are used to ensure clean VISCOR is supplied to the injector.

The VISCOR is stored in a small tank with a capacity of 750 mg. Most of the liquid flow is used for injector actuation and returned at low pressure to the supply tank. This tank sits on an Adams Lab PGW 753 electronic balance, from which the net mass of liquid injected into the IVC can be inferred. This provides accurate measurements of the liquid mass injected in each cycle, but cannot be used to determine the distribution of liquid mass between two injections in the same cycle.

Liquid, gas and chamber pressures are measured with GP50 Model 211 pressure transducers. All important apparatus calibrations can be found in Appendix C.

\(^2\) VISCOR ® 1487 has a composition of 40-70\% petroleum hydrocarbon distillate, 10-30\% Light naphthenic hydrotreated distillates, 1-5\% Calcium salt of sulfonic acids.
3.6 Control System, Data Acquisition, and Triggering

All fuel and chamber supply parameters are manually controlled by the use of hand valves and pressure regulators. Injector pulse parameters are controlled by a computer running National Instruments LabView Software Version 8.0. A LabView VI was written that allows the user to input desired pulse width durations for each of the pilot and main pulse, pulse separation, and firing frequency. The VI then outputs this onto a National Instruments Model BNC-2121 counter, which in turn generates pulses as specified. These counter pulses are then sent through a specially designed Westport injector driver box to boost the current and voltage of the signal finally sent to the injector.

The IVC pressure transducers and coriolis mass flow meter analog output signals are connected to a National Instruments Model BNC-2120 DAQ board. A VI was written that takes these signals as well as the output of the digital scale (via serial port) and records them as tab separated text files for analysis in Microsoft Excel. Data is taken at a frequency of 5 Hz, which is limited by the output frequency of the scale.

A separate camera computer was used for image recording with the camera. This computer has 550 PC Interface Board installed, which has a BNC camera trigger input and an Ethernet connection to communicate with the camera. Phantom Version 630 is installed to control the camera and adjust all settings. The high-speed image data from the camera is stored on the camera buffer and later saved in raw video format (.cin files).
These videos could then be converted to 16-bit grayscale TIFF images for post-processing with MATLAB.

Camera triggering required the installation of a normally open electromechanical contact on the spray extraction shroud. The contact was positioned such that there was no signal when the shroud is over top of the injector and the switch is closed (allowing the injector to reach steady state without fouling the windows). As soon as the shroud retracts, the switch opens and a signal is sent. The camera is triggered when it receives this signal (indicating the injector is visible) AND a pulse from the injector (indicating it is firing).
Chapter 4 - Results and Discussion

This chapter is divided into 4 main sections: Experimental Design, Results from Flow Measurements, Results from Co-injector Modeling, and Results from High-Speed Imaging. The experimental design for all the modes of testing are discussed in Section 4.1. Flow measurement results are presented in Section 4.2 by showing flow characterization for each Co-injector independently, as well a J36 baseline flow curve for comparison. In Section 4.2.5 all of the Co-injectors are compared to this baseline and each other to gain further insight into resulting flow rate differences from each design. Experimental results are compared to theory and simulations in Section 4.3. A theoretical model to further understand Co-injector fluid mechanics is introduced in Section 4.3.1. In conjunction with Westport, a comprehensive AMESim J36 model is extended to gain insight into injector dynamics and effects on flow rates. The results of the simulations run using this model are presented and explained in Section 4.3.2. Finally, high-speed imaging was used to gain qualitative understanding of the two-phase flows as well as measure some important physical parameters such as mechanical delay. Results obtained from these imaging methods are presented in Section 4.4.
4.1 Experimental Design

Flow characterization experiments of Co-injectors in the IVC were designed to match the pressures and injector durations typical of engine operation. Other experiments were chosen to further understand Co-injector dynamics - such as fuel supply and chamber pressure variations on flow rate, high gas flows to determine discharge coefficient, injector mechanical delay, and minimum pulse width for injection. These experiments include conditions that cannot be reached in actual engine operation.

4.1.1 Flow Characterization

Under normal J36 and Co-injector B operation the diesel pulse width (DPW) and gas pulse width (GPW) provide the most direct control over the amounts of diesel/gas injected. Co-injector CS and CSX use the diesel gas bias and the GPW to control these amounts. For these reasons, it is desirable to understand the flow characteristics of the two-phase injection by varying these parameters through a wide range of values and studying the differences produced by each injector.

The experimental parameters for flow characterization were chosen to resemble those used during single injection engine operation (McTaggart-Cown 2006; Brown 2008). In these experiments, there is only one injection per cycle so that the gas and diesel masses in the injection can be determined unambiguously. If there are two injections per cycle
(double-pulse testing), the flow measurements cannot be used to determine how the diesel and gas are distributed amongst the two injections.

In determining the values for pulse width variation, the region of interest studied by Brown was used as a baseline for all flow characterization (0.45 - 0.70 ms in increments of 0.05 ms). However, SCRE in-cylinder pressure varies with stroke and this region had to be slightly altered to compensate for constant IVC chamber pressure. Therefore, the region 0.55 - 0.80 ms was chosen. This region shows a similarly distinct saddle to SCRE and Westport experiments and will be the basis of much discussion. The injection frequency was 400 per minute, equivalent to 800 RPM in a 4-strok engine. This speed is close to the SCRE idle speed and produced flow rates on the order of those seen in the engine. Higher experiment speeds produced similar flow trends but with substantially higher flow rates that required more bottled nitrogen and VISCOR that were unfeasible for prolonged testing. A chamber pressure of 70 bar was used. This pressure is similar to the in-cylinder pressure at the start of main injection in SCRE testing. Diesel and gas supply pressures used for flow characterization were also set to the same values as SCRE testing for the majority of flow testing. However, flow characterization was also performed with higher supply pressure during similar work in the engine.
Using the abovementioned considerations, a full factorial experiment was completed for each injector with 3 repetitions of each test point\(^3\) in Table 4.1. Table 4.2 shows the parameters that were kept constant so the effects of only DPW (or diesel gas bias) and GPW variation could be examined. From this test matrix, fundamental flow rate curves can be constructed and compared for each injector to understand differences.

<table>
<thead>
<tr>
<th>Test Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel Pulse Width (ms)</td>
<td>0, 1.0, 1.5</td>
</tr>
<tr>
<td>Diesel Gas Bias (MPa)</td>
<td>1.0, 2.0, 4.0</td>
</tr>
<tr>
<td>Gas Pulse Width (ms)</td>
<td>0.55, 0.60, 0.65, 0.70, 0.75, 0.80</td>
</tr>
</tbody>
</table>

Table 4.1: Test matrix for Co-injector testing. Co-injector B requires variation in DPW whilst CS and CSX require variation in bias.

<table>
<thead>
<tr>
<th>Test Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>800 RPM</td>
</tr>
<tr>
<td>Chamber Pressure</td>
<td>70 bar</td>
</tr>
<tr>
<td>Pulse Separation</td>
<td>1.0 ms</td>
</tr>
<tr>
<td>Gas Rail Pressure</td>
<td>21.0 MPa</td>
</tr>
<tr>
<td>Diesel Rail Pressure</td>
<td>23.5 MPa</td>
</tr>
<tr>
<td>Bias (J36 and B)</td>
<td>2.5 MPa</td>
</tr>
</tbody>
</table>

Table 4.2: The constant parameters during flow characterization.

\(^3\) Each test point consisted of a scale measurement, diesel supply pressure, gas supply pressure, chamber pressure, and coriolis flow meter reading accumulated at 5 Hz for 4 minutes. This produced 1200 values of each parameter for final test point averaging.
4.1.2 Chamber Pressure Effects on Co-injector Flow Rates

The force balance of the injector gas needle is directly dependant on chamber pressure. As such, chamber pressure plays an important role in gas needle response and subsequently injection flow rates. To study the effect of chamber pressure on injection flow rates, a test series was constructed by using diesel and gas flow rates that were similar to engine pulse widths used by Brown and Laforet. Chamber pressure is varied under constant DPW (or diesel gas bias) and GPW. Table 4.3 shows the test matrix for this data set with 3 repetitions of each point. Aside from chamber pressure, all other parameters were kept constant as in Table 4.2.

<table>
<thead>
<tr>
<th>Test Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel Pulse Width (ms)</td>
<td>1.5</td>
</tr>
<tr>
<td>Diesel Gas Bias (MPa)</td>
<td>1.0 (CS), 2.0 (CSX)</td>
</tr>
<tr>
<td>Gas Pulse Width (ms)</td>
<td>0.65, 0.80, 1.4</td>
</tr>
<tr>
<td>Chamber Pressure (bar)</td>
<td>0, 10, 20, 30, 40, 50, 60, 70</td>
</tr>
</tbody>
</table>

Table 4.3: Test matrix for chamber pressure effects.

4.1.3 High GPW Tests

High GPW tests are used to determine the discharge coefficient for each injector. Using a sufficiently high GPW ensures the gas needle is open long enough for all diesel to effectively be scavenged from the plenum. Once all diesel is injected at the beginning of this long injection, a period of uninterrupted gas flow will ensue from which the
discharge coefficient can be computed. To determine the discharge coefficient for each Co-injector a GPW of 1.6 ms and 2.0 ms was used with the lowest possible diesel flow (DPW = 0 ms or bias = 1.0 MPa) to determine the change in injection mass per unit change in GPW. The gas supply pressure used is 21.0 MPa to a chamber pressure of 70 bar – a pressure ratio of 3. Under such pressure ratio choked flow assumptions can be made and the discharge coefficient can be computed from comparison of the experimental change in mass flow rate to the ideal choked mass flow rate for nitrogen:

\[ C_D = \frac{\dot{m}_{\text{Exp}}}{\dot{m}_{\text{ideal}}} \]  

(4.1)

with the ideal choked mass flow rate given by,

\[ \dot{m}_{\text{ideal}} = A_t \sqrt{\gamma p_o \rho_o \left( \frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}}} \]  

(4.2)

Where \( A_t \) is the total area of the injector gas holes and \( P_o \) is the gas supply pressure and density is determined from the ideal gas law using supply temperature,

\[ \rho_o = \frac{P_o}{RT_o} \]  

(4.3)

The experimental mass flow rate is measured by taking the difference in gas flow rate at 2.0 ms and 1.6 ms and dividing by difference in pulse width such that,

\[ \dot{m}_{\text{Exp}} = \frac{m_{\text{i}}^{GPW=2.0\text{ms}} - m_{\text{i}}^{GPW=1.6\text{ms}}}{\Delta GPW} \]  

(4.4)

and as shown in Figure 4.1.
Figure 4.1: A graphical illustration showing how the experimental mass flow rate is computed using the slope of the two high GPW values.

4.1.4 Co-injector Electro-Mechanical Delay

The Co-injector electro-mechanical delay is defined as the difference between the time the controller pulse is generated and the time an injection is visually confirmed. There are two main contributors to this delay: the current driver and the injector solenoid valves. The shroud sensor coupled with the first pulse sent to the injector triggers the camera imaging. Since the camera begins imaging upon a pulse sent to the injector, the mechanical delay can be determined as the time difference between the start of imaging and the first sign of injection. The influence of key Co-injector parameters on mechanical delay will be investigated. In particular, diesel is used for actuation of the gas needle and supply pressure variations will be examined on mechanical delay. Chamber pressure is
important to the force balance of the gas needle and will also be varied to determine the
effect on mechanical delay. Table 4.4 shows the test points used to do so for the Co-injectors and baseline J36.

<table>
<thead>
<tr>
<th>Test Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel Pulse Width (ms)</td>
<td>1.0</td>
</tr>
<tr>
<td>Diesel Gas Bias (MPa)</td>
<td>1.0</td>
</tr>
<tr>
<td>Gas Pulse Width (ms)</td>
<td>0.80</td>
</tr>
<tr>
<td>Supply pressure (MPa)</td>
<td>16.0, 18.0, 21.0, 25.0, 28.0</td>
</tr>
</tbody>
</table>

Table 4.4: Test matrix for determining mechanical delay as a function of gas supply pressure.

4.1.5 Minimum GPW for Injection

Optical access makes it possible to determine the minimum GPW for gas needle opening and subsequently fuel being injected. The GPW is set to a low value where no injections are initially seen. The GPW is then increased from this value until fuel injections are visible. Chamber and supply pressures can be varied to inspect their effect on needle opening as well as the DPW or bias.
4.1.6 Low Molecular Weight Gas Blend Testing

Since methane and nitrogen have different molecular properties their influence on Co-injector flow rates will also be different. In order to understand some of these differences an inert blended gas that matched the density of methane was created using a helium/nitrogen mixture (49.8% to 50.2% by volume). 4000 psi tanks were made by matching the required partial pressures of the two gases to give a fluid that matched the density of methane. Due to the limited resources of creating such blended tanks gas flow characterization was only performed for a single level of diesel fueling. The flow characterizations for the Co-injectors using this blend was performed and compared to nitrogen in Section 4.2.

4.1.7 Non-dimensional Groups

The experimental design focused on ensuring many of the parameters are varied such that they can be used to interpret Co-injector behavior in the SCRE. As a result, some key non-dimensional groups are varied from these experimental procedures.

- The pressure ratio is defined to be the ratio of gas supply pressure to chamber pressure. For flow characterization at 21 MPa it is kept at 3 and for 28 MPa it is 4. However, this ratio is varied from 3-210 when measuring chamber pressure effects.
The Reynolds number values, \( Re \), were varied through two methods. First, the gas supply pressures could be varied subsequently increasing \( Re \) with a corresponding increase in supply pressure. Secondly, \( Re \) was decreased in the gas blend testing because the density of the injected gas was lowered to match methane and lower flow rates were measured. The Reynolds Number was examined in two distinct regions of the fuel delivery system: the point of pressure readings (in Figure 3.8), and inside the Co-injector gas needle nozzles (see Figure 3.2 and 3.3).

Table 4.5 shows a summary of the maximum Reynolds values in these two regions. Typically \( Re \) varied between 0 - 13,700 for the gas supply and 0-10 for the diesel supply depending on region examined. There is a large range of intermediate values \( Re \) can assume based on injection parameters and the properties of the fluid injected. If a region of pure diesel injection is examined, \( Re \) will be much lower than a region of pure gas injection because of density and viscosity differences. Two-phase diesel/gas injections will assume values intermediate to the region stated as the density and viscosity will be intermediate to gas and diesel and will vary with gas-to-liquid ratio (Chepakovich 1993). The diesel flow will be highly laminar for both regions. The gas flow will be laminar in the supply line since piping diameters are sizably larger than in the nozzle. The subsequent turbulent flow found in the nozzle region may be desirable for diesel/gas mixing.
Table 4.5: Maximum values of Re experimentally attained with high flow rates. Since flow testing was done with a wide range of parameters most testing points assume intermediate Re values between 0 (no flow) and Re_max.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Re_max Gas Needle Nozzles</th>
<th>Re_max Supply Line</th>
</tr>
</thead>
<tbody>
<tr>
<td>VISCOR</td>
<td>9.5</td>
<td>1.4</td>
</tr>
<tr>
<td>Nitrogen - 21 Mpa</td>
<td>10,300</td>
<td>1,490</td>
</tr>
<tr>
<td>Nitrogen - 28 Mpa</td>
<td>13,700</td>
<td>1,980</td>
</tr>
<tr>
<td>Blended Gas</td>
<td>8,600</td>
<td>1,240</td>
</tr>
</tbody>
</table>

- The Mach number typically assumed very low values in the gas and diesel supply line region (maximum Mach number measured was approximately 0.01). However, the flow is assumed to be choked upon exiting the injector, this implies flow testing was conducted with a sonic Mach number of 1 after the gas/diesel exit the injector nozzle.

- The gas-to-liquid ratio is defined as the ratio of gas mass per injection to liquid mass per injection. This ratio was experimentally measured from flow testing and assumed values between 0 (pure diesel injection) and 40 (high GPW, high supply pressure, and very low diesel flows). As will be seen in Section 4.2, this ratio assumes values between 1-5 in the distinct saddle regions of flow characterization.
4.2 Results from Flow Measurements

The results in this section are shown for each Injector in order of testing. The theory associated with computed characterization parameters are also developed in this section and discussed. For a comprehensive summary of all the flow characterization results, as well as those found through imaging in Section 4.4, the reader is directed to Table 5.1 of Chapter 5.

4.2.1 Flow Characterization of J36

The J36 was used as a baseline single-injection gas flow characterization to show gas injection mass dependence on gas pulse width. Since the diesel injection mass is independently controlled via a separate injection needle the gas flow dynamics could, in turn, be studied with little to no diesel interaction in the gas plenum. The only source of leakage into the gas plenum is from the match fit associated with the diesel needle. The diesel leakage into the gas plenum was constant for the 2.5 MPa bias used and was measured to be 13.8 ± 3.3 mg/s or 2.1 ± 0.5 mg/inj at 800 RPM.

Flow characterization of the J36 injector is shown in Figure 4.2. A distinct “saddle” region of nonlinear gas flow is shown in the GPW = 0.50 – 0.65 ms range. This saddle region is subsequently shown to be consistent with all injectors tested and will be
discussed in the context of Co-injector operation in Section 4.2.2. In the GPW = 0.65 – 0.80 ms range gas flow assumes a linear trend suggesting consistent actuator response and needle opening durations. If these higher GPW regions are extended further, the discharge coefficient can be determined (see discussion in Section 4.1.2). The J36 discharge coefficient was determined to be 0.938 ± 0.023 and is the baseline for subsequent Co-injector comparison.

Figure 4.2: J36 gas flow vs. GPW for DPW = 0 ms at 21.0 MPa gas rail pressure and 70 bar chamber pressure.
4.2.2 Flow Characterization of Co-injector B

Understanding diesel and gas flow quantities as functions of various parameters are of fundamental importance to understanding Co-injector operation. Figure 4.3 shows that the amount of diesel present in the common reservoir has an effect on the gas flow injected. For larger DPW, more liquid is allowed to enter the common reservoir and occupy volume that would otherwise be filled with gas. Higher liquid injection mass would also reduce the 2-phase speed of sound in the nozzle. The gas flow curves exhibit a non-linear trend with gas flows increasing sharply past GPW = 0.65 ms and exhibits a similar saddle region as previous Westport flow bench testing and as described by Brown 2008. Higher variability at lower pulse widths may be a result of differing needle opening and closing times due to actuator response. The diesel present in the plenum may be ineffectively scavenged at these points, impeding any ensuing gas flow. Furthermore, needle bounce, whereby the needle rebounds from the position of maximum lift due to excessive force of the needle opening, may also contribute to inconsistency. The gas flow curves do not intersect which is desirable since a unique gas flow rate for a given DPW and GPW can be produced.

The diesel flow rates are independent of gas flow. This is to be expected since the main parameter for diesel flow rate control is the DPW which remained constant for a given data set. The bias would also affect the diesel mass, but for this study the bias was kept constant. Table 4.6 shows the diesel flow rates for the set bias of 2.5 MPa under the 3 different DPWs used.
Figure 4.3: Co-injector B gas flow vs. GPW for 3 different DPW at 21.0 MPa gas rail pressure and 70 bar chamber pressure.

Table 4.6: Co-injector B diesel flow rates for a 2.5 MPa bias and varying DPW.

<table>
<thead>
<tr>
<th>Diesel Pulse Width (ms)</th>
<th>Diesel Flow (mg/inj)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>2.6 ± 0.6</td>
</tr>
<tr>
<td>1.0</td>
<td>8.1 ± 0.3</td>
</tr>
<tr>
<td>1.5</td>
<td>10.8 ± 0.4</td>
</tr>
</tbody>
</table>

Additional tests were run to quantify flow rates at higher supply pressures in parallel with similar supply pressures in the engine (Laforet 2009). Figures 4.4 shows the flow rates for a gas rail pressure of 28.0 MPa. The bias was kept the same as the lower pressure runs and diesel flow rates were consistent with Table 4.6.
Comparing Figures 4.3 with 4.4, it is apparent that higher supply pressure results in higher gas flow. Also, the gas flow is less affected by the diesel flow, which might be explained as follows. With the higher gas density and possibly faster needle opening, the diesel present in the plenum is injected faster, allowing unimpeded gas flow for a larger part of the injection duration. The saddle region as seen in Figure 4.3 is shifted to lower GPW implying the higher diesel supply pressure produce more consistent needle opening times in the low GPW regions.

To further extend flow testing of the injector a blended helium-nitrogen blend was created that matched the density of methane used in engine testing. Based on ideal mass flow rate calculations under choked flow conditions, it was expected that the nitrogen flow rates be 26% higher than the mixture (See Appendix D for mixture details and
calculations). Figure 4.5 shows the flow rates of the blended gas and nitrogen gas under the same conditions for comparison. The blended gas flow rates were indeed lower than nitrogen but by factors of 13-29%. The diesel flow rates matched those of the initial nitrogen flow characterization in Table 4.5.

![Figure 4.5: Co-injector B blended gas flow rates at DPW = 0 ms compared to nitrogen for the DPW = 0 ms series with gas supply pressure of 21 MPa and chamber pressure of 70 bar.](image)

Since the injector performs different in regions of extreme GPW, very high GPW and low GPW were examined. At higher GPWs of Figure 4.3 the slopes of the flow curves begin approach the same value. This is of particular importance as the discharge coefficient for the injector can be experimentally determined. Measurements of the gas flow rates were taken with DPW = 0 and 1.5 ms and GPW = 1.0, 1.6, 2.0 ms. The discharge coefficient was then calculated by comparing the slope of the flow curve to the
ideal mass flow rate under choked flow conditions described in Section 4.1.3 (see Appendix E. for the large GPW data used). The discharge coefficient for Co-injector B was found to be 0.711 ± 0.015.

At very low GPW it becomes unclear whether the gas needle opens at all as no gas flow is registered. To examine needle opening, the minimum GPW required for fuel to be injected can be determined by visual inspection. This is done by setting the DPW to a desired value then varying the GPW from a low value (where no injections are visually seen) and incrementally increasing the GPW until fuel injections can be visually confirmed. It should be noted that at this minimum GPW the injection is purely diesel as zero gas flow is measured by the data acquisition system. Table 4.7 summarizes the results under chamber pressure conditions of 70 bar using 21 MPa gas supply pressure and a 2.5 MPa bias.

<table>
<thead>
<tr>
<th>DPW (ms)</th>
<th>Min. GPW for Needle Opening (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.45</td>
</tr>
<tr>
<td>1.0</td>
<td>0.43</td>
</tr>
<tr>
<td>1.5</td>
<td>0.43</td>
</tr>
</tbody>
</table>

Table 4.7: Co-injector B minimum GPW required for visual confirmation of fuel being injected into the IVC.

The chamber pressure affects the gas needle dynamics and subsequently the quantity of fuel injected. Figure 4.6 shows the effect of chamber pressure on the amount of gas injected. Notice that for GPW = 0.80 ms the gas injection quantity is reduced by an order of magnitude when the chamber pressure is reduced to atmospheric pressure. For high
GPW = 1.4 ms the decrease in gas flow is linear suggesting an increase in chamber pressure has a linear effect on gas flow; useful for interpolating gas flows to an intermediate chamber pressure such that may be found in engine testing. At GPW = 0.65 ms the co-injector injects no gas at 30-40 bar. Recall from Sections 1.3 and 3.1 and that the chamber pressure acts to assist needle opening. In the closed position, the hydraulic pressure above the needle (i.e. VISCOR® in this case) is high enough to seal the needle. When the opening command is sent to the injector, a solenoid valve drains the fluid above the needle through a control orifice. Lastly, it should be noted that for chamber pressure effects the standard deviation is smaller than the symbol size of the graphs. This implies the injector performs more consistently when the parameter of variation is only the chamber pressure.

![Figure 4.6: Co-injector B gas flow vs. chamber pressure for 3 GPWs.](image-url)
4.2.3 Co-injector CSX Flow Characterization

Co-injector CSX was the first of the series of prototype co-injectors that removes the diesel needle altogether and allows diesel to continuously collect in the gas plenum. Figure 4.7 shows that the gas flow quantity is again dependent on the diesel quantity. The saddle regions at GPW less than 0.70 ms are maintained for low biases of 1 MPa and 2 MPa further suggesting actuator response is important in this region. With high diesel flow at a bias of 4 MPa, gas flow is not registered until GPW = 0.80 ms. This is due to significant amounts of diesel accumulating in the reservoir and taking a GPW of 0.80 ms or greater to effectively scavenge out all of the diesel. For completeness the 4 MPa bias test matrix was extended to GPW = 0.95 ms to include a trend of measurable gas flow rates. Furthermore, the absence of the saddle for the 4 MPa data series further suggests it is a phenomenon of low GPW actuator response and needle response. Table 4.8 shows the considerably higher diesel fueling quantities present with this co-injector prototype. The slope of the flow curves begin to match at regions of GPW greater than 0.70 ms. This implies the diesel has sufficient time to scavenge from the gas plenum and unimpeded gas flow for a finite duration can proceed. This is desirable since precise diesel and gas flow quantities for these higher GPW regions and beyond can be extrapolated and anticipated in subsequent engine use, unlike the lower GPW regions where considerable nonlinearity and variability are present.
Figure 4.7: Co-injector CSX gas flow vs. GPW for 3 different biases at 21.0 MPa gas rail pressure at 70 bar chamber pressure.

<table>
<thead>
<tr>
<th>Bias (MPa)</th>
<th>Diesel Flow (mg/inj)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>12.6 ± 0.6</td>
</tr>
<tr>
<td>2.0</td>
<td>18.5 ± 0.7</td>
</tr>
<tr>
<td>4.0</td>
<td>26.6 ± 0.2</td>
</tr>
</tbody>
</table>

Table 4.8: Co-injector CSX diesel flow rates with varying bias.

Similar to Co-injector B, Co-injector CSX was tested at a high gas supply pressure of 28 MPa. Figure 4.8 shows the flow characterization at this supply pressure. The flow rates increase as expected with the curves exhibiting a noticeable shift towards lower GPW. The saddle region exhibits a noticeable distinction – the gas flow rate decreases from GPW = 0.55 ms to GPW = 0.60 ms for both 1 MPa and 2 MPa bias. This implies inconsistent actuator response and needle bounce are of significance in this region.
Furthermore, internal pressure waves in the diesel and gas supply lines may have some effect on fuelling quantities though measures were taken to dampen them. The higher supply pressure appears to scavenge the diesel quantities faster and gas flow is registered at a lower GPW for the 4 MPa bias than the lower supply pressure. Again this series has a notable absence of the saddle since gas flow begins at GPW of 0.70 ms and greater; outside the region of inconsistent needle dynamics. Diesel flow rates remained consistent with Table 4.8.

![Graph showing gas flow vs. GPW](image)

**Figure 4.8**: Co-injector CSX gas flow vs. GPW at 28.0 MPa gas rail pressure and 70 bar chamber pressure.

Blend testing results are shown in Figure 4.9 for Co-injector CSX. Both curves maintain similar trends with the blended gas being shifted lower due to the lower density gas injections. The blended gas flows are 26 – 51% lower than the nitrogen characterization. There was considerable variability in blended gas flow tests at low pulse width, however
at higher GPW of 0.75 and 0.80 ms the gas flows remain relatively consistent with the expected 26% lower flow rates. Diesel flow rates were in line with the values of Table 4.7.

![Graph showing gas injection mass vs gas pulse width](image)

**Figure 4.9:** Co-injector CSX blended gas flow rates compared to nitrogen to bias of 1.0 MPa series with gas supply pressure of 21 MPa and chamber pressure of 70 bar.

Figure 4.10 shows the effects of chamber pressure on gas injection quantities for Co-injector CSX at 3 different pulse widths while maintaining at bias of 2.0 MPa. Linearity is maintained at GPW = 0.80 and 1.4 ms. At GPW = 0.65 ms gas is injected consistently at 80 bar chamber pressure but quickly reduces to zero with decreased needle opening at 60 bar chamber pressure. The data is representative of the linear effect that chamber pressure appears to have on gas injection quantities.
High GPW tests of 1.6 and 2.0 ms were performed with a bias of 1 MPa and 2 MPa to determine the discharge coefficient for Co-injector CSX with the slope of the two curves being averaged. The discharge coefficient for Co-injector CSX is $0.666 \pm 0.017$.

In contrast, visual examination of low GPW test was performed to determine the minimum pulse width needed for injection to occur. Table 4.9 shows the values. The large amounts of diesel present in the reservoir appear to have a small effect on the needle opening slightly sooner at the lowest diesel flow tested. However, the values remain constant and seem to reach the limitation of needle opening pulse width for any higher diesel fuelling rates. This is likely due to the limitations of actuator response being reached.

Figure 4.10: Co-injector CSX gas flow vs. chamber pressure for 3 GPWs.
<table>
<thead>
<tr>
<th>Bias (MPa)</th>
<th>Min. GPW for Needle Opening (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>0.43</td>
</tr>
<tr>
<td>2.0</td>
<td>0.43</td>
</tr>
<tr>
<td>4.0</td>
<td>0.43</td>
</tr>
</tbody>
</table>

Table 4.9: Co-injector CSX minimum GPW required for visual confirmation of fuel being injected into the IVC.

4.2.4 Co-injector CS Flow Characterization

Co-injector CS was next in the series of prototype injectors tested. This Co-injector also removes the diesel needle and allows diesel to continuously collect in the gas plenum. The injector has a slightly smaller sleeve (as described in Section 3.1.2), in turn, having a larger diesel/gas reservoir with larger flow area. Figure 4.11 shows that the flow characterization of this injector. As expected, the diesel quantity again has a profound effect on gas injection amounts. The saddle regions at GPW less than 0.70 ms are maintained further suggesting needle and actuator response are non-linear in the regions of GPW less than 0.70 ms. Table 4.10 shows even higher diesel fueling quantities present with this co-injector build. At high bias of 4.0 MPa the injector begins to register gas flow at GPW = 0.70 ms and greater, suggesting that the diesel present per injection takes considerable time to clear. Consistent with the other Co-injectors tested, the slopes of the gas flow rate curves align in the higher GPW regions after most diesel is scavenged.
Table 4.10: Co-injector CS diesel flow rates with varying bias.

<table>
<thead>
<tr>
<th>Bias (MPa)</th>
<th>Diesel Flow (mg/inj)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>14.1 ± 0.5</td>
</tr>
<tr>
<td>2.0</td>
<td>19.9 ± 0.3</td>
</tr>
<tr>
<td>4.0</td>
<td>28.2 ± 0.4</td>
</tr>
</tbody>
</table>

Co-injector CS maintained a similar trend for changes in chamber pressure to gas injection quantities. A nearly linear trend is seen in Figure 4.12 suggesting that similar gas needle dynamic effects are present in this Co-injector.
In July 2009 Co-injector CS began to perform erratically in engine testing. The injection rates were checked in the IVC confirming considerably lower diesel injection quantities while gas flow increased. This suggested the flow restrictor had become partially obstructed. The injector may have been used with contaminated fuel causing subsequent blockage of the restrictor or the restrictor may have been dislodged from its desired position. In any case, the injector was sent to Westport for repairs and was unavailable for 28 MPa and blended gas testing.

High GPW tests were performed with Co-injector CS to determine the discharge coefficient. Flow rates at GPW = 1.6 and 2.0 ms were used with a low bias of 1 MPa. The calculation is Section 3.7.3 was performed using the collected data to determine that Co-injector CS had a discharge coefficient of 0.738 ± 0.014.
Since the injector still maintains many of the same features as Co-injector B and the HPDI standard J36 the gas needle appears to operate much in the same way. Table 4.11 shows the minimum GPW for gas needle opening and its similarity to Co-injector B and CSX further proves the gas needle dynamics are comparable.

<table>
<thead>
<tr>
<th>Bias (MPa)</th>
<th>Min. GPW for Needle Opening (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>0.43</td>
</tr>
<tr>
<td>2.0</td>
<td>0.43</td>
</tr>
<tr>
<td>4.0</td>
<td>0.43</td>
</tr>
</tbody>
</table>

Table 4.11: Co-injector CS minimum GPW required for visual confirmation of fuel being injected into the IVC.

4.2.5 Injector Comparisons

Each of the prototype Co-injectors is based off a modification of the standard J36 injector. In order to understand the effects of the modifications each of the Co-injectors is compared to the baseline gas flow of the J36. Co-injector B (Figure 4.13) shows clearly less gas flow for every operating condition. At DPW = 0 ms the conditions most closely resemble the conditions of the J36 baseline. The difference in gas flows can be attributed to the modified gas plenum geometry through sleeve addition and the slightly higher diesel amounts present from leakage past the diesel needle match fit. Furthermore, the sleeve addition may cause a pressure drop resulting in a lower tip supply pressure, with subsequently lower flow rates. The saddle regions of both injector operations are consistent in the GPW 0.55 – 0.65 ms range. Beyond this region the slopes of the J36 and Co-injector B curves are different (the J36 being steeper), indicative of the different discharge coefficients measured.
Figure 4.13: Co-injector B comparison to the baseline J36 gas flow.

Co-injector CSX (Figure 4.14) shows the same non-linear saddle present as the J36 in the low GPW region. The slope of the curves is noticeably less steep than the J36. Further, a large difference in gas flow rates is seen in comparison to the J36. Co-injector CSX has two main differences – the large internal sleeve greatly reduces the plenum volume, as well as having high relative diesel flow rates, both acting to impede gas flow.
Figure 4.14: Co-injector CSX comparison to the baseline J36 gas flow.

Co-injector CS (Figure 4.15) has the highest flow rates of the three prototypes and this is evident in the comparison to the J36. The slope in the high GPW region is again shallower than the J36. The saddle region maintains its consistency at low GPW. At high diesel fueling rates (Bias = 4 MPa) the saddle region is absent as the injector begins registering gas flow GPW = 0.70 ms. This further suggests the saddle region is only a phenomena found at low GPW.

Figure 4.16 shows the comparison of flow rates between Co-injector CSX and CS. The larger sleeve of Co-injector CSX acts to translate the flow curves lower by a constant amount - approximately 20 mg/inj at higher GPW. This may be of importance to future developments of Co-injector sleeve sizing as increasing the sleeve diameter by 0.2 mm, in this instance, acts to reduce gas flow by 20 mg/inj for similar diesel fuelling rates.
Figure 4.15: Co-injector CS comparison to the baseline J36 gas flow.

Figure 4.16: A comparison of the larger sleeve on gas flow rate differences between Co-injector CS and Co-injector CSX.
Recall that each Co-injector flow characterization has gas flow rate curves for 3 different levels of diesel fuelling – as in Figure 4.3. A critical aspect of these flow characterization curves is understanding the internal diesel interaction and its ensuing effect on gas flow rates. Thus, the parameter, $R_s$, is introduced in an attempt to quantify such interactions. Let,

$$R_s = \left. \frac{\partial m_g}{\partial m_d} \right|_{GPW=0.80\,\text{ms}}$$

where $\partial m_g$ is the difference in gas flow rates at $GPW = 0.80$ ms, and $\partial m_d$ is the difference in diesel flow rate between the curves being examined. $R_s$ is computed at $GPW = 0.80$ ms due to the consistent gas flows produced in this region for all Co-injectors. This ensures the diesel/gas interaction is examined outside of the saddle region which is prone to higher variability gas flows, large errors, and inconsistent actuator response. Moreover, at $GPW = 0.80$ ms each Co-injector has non-zero flow rates for all levels of diesel fuelling allowing the diesel/gas interactions to be quantified between the three separate levels of fuelling and averaged.

$R_s$ is a measure of the gas flow rate sensitivity on the amount of diesel being injected. Higher $R_s$-values represent the gas flows being more susceptible to flow rate reductions for a given amount of diesel being present. Table 4.12 shows the computed $R_s$-values for each Co-injector flow characterization (both at 21 and 28 MPa where measured).

<table>
<thead>
<tr>
<th></th>
<th>B - 21 Mpa</th>
<th>B - 28 Mpa</th>
<th>CSX-21 MPa</th>
<th>CSX-28 MPa</th>
<th>CS - 21 Mpa</th>
</tr>
</thead>
<tbody>
<tr>
<td>$R_s$</td>
<td>1.15 ± 0.03</td>
<td>0.60 ± 0.04</td>
<td>1.58 ± 0.19</td>
<td>1.32 ± 0.04</td>
<td>1.79 ± 0.05</td>
</tr>
</tbody>
</table>

Table 4.12: $R_s$-values for the flow characterizations performed on each Co-injector.
The $R_S$-values confirm the qualitative trends of the curves. Co-injector B at 21 MPa gas supply pressure has less variation in gas flow than both Co-injector CSX and CS. It should be noted that the method of diesel introduction in Co-injector B is fundamentally different than Co-injector CSX and CS, leading to a different sensitivity to subsequent gas flows. In comparison, Co-injector CS has the highest gas flow sensitivity per quantity of diesel. This may be a factor of internal geometry, with Co-injector CS having a smaller sleeve that is less impeding on gas flow than the larger sleeve in Co-injector CSX.

As mentioned in Sections 4.2.2 and 4.2.3 it appeared that the high supply pressure of 28 MPa had less variability in gas flow than the 21 MPa supply pressure. This is confirmed in both cases – Co-injector B and CSX both have lower $R_S$-values, thus less gas flow variability, for the 28 MPa data than their corresponding $R$-values at 21 MPa.

Recall from Figures 4.6, 4.10, and 4.12 that the Co-injector gas flow rates appear to linearly increase with chamber pressure while holding all other parameters constant. To further understand this, the precise flow rate increase can be computed at each chamber pressure by linearly interpolating the points. Therefore,

$$S = \frac{\Delta m_g}{\Delta P_c}$$

(4.6)

where $S$ is denoted as the average rate of change of the gas flow rate, $m_g$, under varying chamber pressures, $P_c$ – equivalent to the average slope of the curves. Table 4.13 shows the average $S$-value (in units of mg/inj-bar) at each constant GPW curve and for each of the Co-injectors. Notice that at GPW = 0.65 ms both Co-injector B and CSX have a
considerably lower $S$-value than at the higher GPWs. This may be due to GPW = 0.65 ms being a pulse width associated with the saddle region found in flow characterization. This implies that the inconsistent actuator response in the saddle regions of flow characterization is independent of chamber pressure. At GPW = 0.80 ms all of the Co-injectors have statistically similar flow rate changes under varying chamber pressure. This also holds true at the high GPW = 1.40 ms, suggesting consistent Co-injector operation. It appears that the $S$-value slightly increases as GPW increases from 0.80 to 1.40 ms – though the change is not statistically significant.

<table>
<thead>
<tr>
<th>GPW (ms)</th>
<th>Co-injector</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>B</td>
<td>CSX</td>
<td>CS</td>
</tr>
<tr>
<td>0.65</td>
<td>0.31 ± 0.09</td>
<td>0.35 ± 0.03</td>
<td>-</td>
</tr>
<tr>
<td>0.80</td>
<td>0.55 ± 0.06</td>
<td>0.57 ± 0.09</td>
<td>0.52 ± 0.04</td>
</tr>
<tr>
<td>1.40</td>
<td>0.66 ± 0.06</td>
<td>0.64 ± 0.05</td>
<td>0.57 ± 0.08</td>
</tr>
</tbody>
</table>

Table 4.13: Average slope, $S$ (in units of mg/inj-bar), of the linear interpolations performed on the curves in Figures 4.6, 4.10, and 4.12.

A comparison of discharge coefficients is shown in Table 4.14. As can be seen, the Co-injectors all have a significantly lower discharge coefficient than the J36. The major influence on this parameter appears to be the internal geometry differences from the added sleeves. The J36 has no added sleeve and very minimal amounts of diesel crossover leakage into the gas plenum, subsequently having a relatively high discharge coefficient. The added sleeves acts to reduce the discharge coefficient for Co-injectors B and CS in comparison to the J36. This is to be expected as the gas flow is further restricted by the sleeve addition. The difference between discharge coefficients in Co-injector B and CS is not, however, statistically significant. This is expected as both
Co-injector B and CS have identical internal geometry. Co-injector CSX has an even more profound sleeve, which acts to further reduce the discharge coefficient.

<table>
<thead>
<tr>
<th>Injector</th>
<th>Discharge Coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>J36</td>
<td>0.938 ± 0.023</td>
</tr>
<tr>
<td>B</td>
<td>0.711 ± 0.015</td>
</tr>
<tr>
<td>CSX</td>
<td>0.666 ± 0.017</td>
</tr>
<tr>
<td>CS</td>
<td>0.738 ± 0.014</td>
</tr>
</tbody>
</table>

Table 4.14: A summary of injector discharge coefficients.

Lastly, an important aspect of gas flow characterization is the ability to deduce the regions in which primarily diesel is injected, with gas flows less than the resolution of the experimental setup. Each Co-injector had the minimum GPW measured for injection to be visually observed, while the minimum GPW from the flow curves for gas to be registered is also known. In turn, a range of GPWs can be constructed that shows what is assumed to be a pure injection of diesel that pools near the injector tip. After gas flow is registered it is known that the diesel is sufficiently scavenged near the tip because lower pressure gas is allowed a flow passage. Table 4.15 shows the regions of diesel-only injections. From the table it is clear that the amount of diesel fuelling has an effect on the GPW required to scavenge the diesel. Higher diesel fuelling rates require higher GPWs before gas flows are registered. The pulse width ranges of Table 4.14 are consistent with the noted high diesel distributions in the GPW ranges of the first pulse in double pulse engine testing presented in the work of Brown et al. (2009) and Laforet (2009).
<table>
<thead>
<tr>
<th>Co-injector</th>
<th>DPW/Bias</th>
<th>GPW Range for Pure Diesel Flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>0 ms</td>
<td>0.43 – 0.50 ms</td>
</tr>
<tr>
<td>B</td>
<td>1.0 ms</td>
<td>0.43 – 0.55 ms</td>
</tr>
<tr>
<td>B</td>
<td>1.5 ms</td>
<td>0.45 – 0.60 ms</td>
</tr>
<tr>
<td>CSX</td>
<td>1.0 MPa</td>
<td>0.43 – 0.60 ms</td>
</tr>
<tr>
<td>CSX</td>
<td>2.0 MPa</td>
<td>0.43 – 0.60 ms</td>
</tr>
<tr>
<td>CSX</td>
<td>4.0 MPa</td>
<td>0.43 – 0.80 ms</td>
</tr>
<tr>
<td>CS</td>
<td>1.0 MPa</td>
<td>0.43 – 0.50 ms</td>
</tr>
<tr>
<td>CS</td>
<td>2.0 MPa</td>
<td>0.43 – 0.60 ms</td>
</tr>
<tr>
<td>CS</td>
<td>4.0 MPa</td>
<td>0.43 – 0.70 ms</td>
</tr>
</tbody>
</table>

Table 4.15: GPW regions where Co-injector flow is primarily diesel, as gas flow is lower than the resolution of the experimental system.

4.3 Results from Co-injector Modeling

4.3.1 Fluid Mechanics Modeling

An important comparison can be drawn between Co-injector B and CS from their measured diesel and gas flow rates. Since the internal geometry is equivalent between these two injectors, the only difference between the two is the method of diesel introduction. From Brown (2008) and Laforet (2009) it is thought that the high internal diesel injection velocities of Co-injector B promote uniform mixtures of diesel/gas prior to injection. Co-injector CS, however, is thought to have the diesel pool near the tip in a stratified layer, since the diesel continuously leaks and most of the pressure drop occurs through the flow restrictor rather than in the plenum. If diesel is pooled near the tip, an injection will consist of mainly diesel, followed by mainly gas after the diesel has been
scavenged\textsuperscript{4}. The two hypothesized methods of diesel/gas mixing are shown in Figure 4.17. Under choked flow assumptions, the speed of sound for liquid and gas are distinctly higher than for a uniform mixture of the two (Sherstyuk 2000). Thus, the mass flow rate is expected to be higher for an injection with distinct liquid and gas portions — as is the case in the higher measured flow rates of Co-injector CS than B.

Figure 4.17: A simplified representation of two methods of diesel/gas mixing within a Co-injector gas plenum. Left — uniform mixing takes place, where the diesel is suspended as small droplets throughout the gas. Right — the diesel is allow to pool near the injector tip, forming a stratified mixture of diesel and gas.

If similar diesel fuelling rates are considered between the injectors (Coinjector B: DPW = 1.5 ms or 10.8 mg/inj, Co-injector CS: Bias = 1.0 MPa or 14.1 mg/inj), Co-injector CS has higher gas flow rates (with slightly higher diesel fuelling) than Co-injector B at every GPW as shown in Figure 4.18. Assuming that the ideal case where gas needle response is

\textsuperscript{4} In the engine work of Laforet (2009) and Brown et al. (2009) heat release results also suggested that the second generation of prototypes (Co-injector CSX and CS) have higher amounts of diesel being injected in the early portion of injections.
identical\(^5\) (which may be valid as the Co-injectors are built from the same body and the only notable difference so far is the method of internally injecting diesel), calculations can be applied to show that different fuel injection methods might be present. In turn, to further understand such mixing differences, a theoretical model will be established that shows the difference such mixtures would have on Co-injector flow rates and the implications it would have on the experimental results of Section 4.2.

![Graph of gas flow rates for Co-injectors](image)

**Figure 4.18**: A comparison of the gas flow rates for each Co-injector at similar diesel fuelling rates.

\(^5\) If \(GPW = 0.55\) ms is examined for Co-injector CS a gas flow of 20 mg/inj is registered. This implies that the gas needle opens faster for this Co-injector due to a more responsive actuator - subsequently triggering gas flow at lower GPW. Non-idealized actuator response implications will be discussed later in this section.
If the Co-injectors are assumed to have uniform mixing of diesel and gas, such that the liquid is distributed as fine droplets within the gas, the resulting two-phase mixture can be assumed to be a quasi-gas (Chepakovich 1993). If the volume occupied by the liquid is also assumed to be negligible to the overall volume of the gas the equivalent specific heats and gas constant of the two-phase mixture can be introduced as,

\[ c_{v,\text{quasi}} = (1 - \eta)c_{v,g} + \eta c_l \]  
(4.7)

\[ c_{p,\text{quasi}} = (1 - \eta)c_{p,g} + \eta c_l \]  
(4.8)

\[ R_{\text{quasi}} = c_{p,\text{quasi}} - c_{v,\text{quasi}} = (1 - \eta)(c_{p,g} - c_{v,g}) \]  
(4.9)

\[ \gamma = \frac{c_{p,\text{quasi}}}{c_{v,\text{quasi}}} = \frac{(1 - \eta)c_{p,g} + \eta c_l}{(1 - \eta)c_{v,g} + \eta c_l} \]  
(4.10)

\[ \eta = \frac{m_l}{m_l + m_g} \]  
(4.11)

where \( m_g \) is the mass of gas, \( m_l \) is the mass of liquid, \( \eta \) is the liquid mass fraction, \( c_{v,g} \) and \( c_{p,g} \) are the specific heats at constant volume and pressure of the gas, \( c_l \) is the specific heat of the liquid, \( c_{v,\text{quasi}} \) and \( c_{p,\text{quasi}} \) are the specific heats of the resulting quasi-gas mixture, and \( R_{\text{quasi}} \) is the gas constant.

By substituting the measured diesel and gas values at each GPW for Co-injector B into Equations 4.7 – 4.11 a theoretical flow rate, \( m_{\text{quasi}} \), can be computed using the choked flow equations of 4.1 and 4.2.
\[ m_{\text{quasi}} = C_D A_i \sqrt{\gamma \rho_o P_o \left( \frac{2}{\gamma+1} \right)^{\gamma+1}} \]  \hspace{1cm} (4.12)

where

\[ \rho_o = \frac{P_o}{R_{\text{quasi}} T_o} \]  \hspace{1cm} (4.13)

and Equations 4.12 and 4.13 now vary due to the difference of the liquid addition to the quasi-gas.

The performance of the model will be assessed by comparing a theoretical injection time, \( t_{\text{inj}} \), with the commanded gas pulse width (GPW). The theoretical time, \( t_{\text{inj}} \), would be the time needed to inject a certain amount of diesel and gas assuming that the needle opens and closes instantaneously and completely. Implicitly, this assumes that the experimentally measure discharge coefficient represents a fully-open injector. In actuality, the commanded GPW differs from the injection duration due to electromechanical delays and varying actuator response between injectors, but if it is assumed these are the same for all injectors, then comparing \( t_{\text{inj}} \) with GPW can provide information on the fluid mechanics.

From the calculated flow rate, \( m_{\text{quasi}} \), the time needed to reach the measured flow quantity can now be computed,

\[ t_{\text{inj}} = \frac{m_{\text{Exp}}}{m_{\text{quasi}}} \]  \hspace{1cm} (4.14)

where \( m_{\text{Exp}} \) is the experimentally measured mass of diesel and gas injected at any GPW such that,
If this same quasi-gas model is applied to Co-injector CS, \( m_{\text{Exp}} \) will be significantly higher as seen by the higher experimental flow rates, whilst calculation shows that \( m_{\text{quasi}} \) remains very similar to that of Co-injector B, thus \( t_{\text{inj}} \) will increase. However, the Co-injectors are compared at the same GPWs so the time frame for injections is assumed to not change between injectors. Therefore, Co-injector CS must have a different physical method of injecting higher quantities diesel and gas within a similar timeframe to Co-injector B.

Fortunately, if the diesel and gas are considered to be stratified and injected separately (i.e. a pure diesel injection, followed by a pure gas injection), these higher injection quantities can be further accounted for. First, the flow rate of a pure diesel injection will undeniably be much higher than a gas or quasi-gas injection based on density considerations alone. The flow rate of diesel through an injector can be modeled from Stone (1992) as,

\[
\dot{m}_{\text{diesel}} = C_D A_s \sqrt{2 \rho_{\text{diesel}} \Delta P}
\]  

(4.16)

where \( \Delta P \) is the pressure difference between the diesel supply and chamber. Thus, the time required for pure diesel injection, \( t_{\text{diesel}} \), is given by,

\[
t_{\text{diesel}} = \frac{\dot{m}_{\text{diesel}}}{\dot{m}_{\text{diesel}}}
\]  

(4.17)

Since the diesel flow rate is significantly higher than the quasi-gas flow rate the time the quantity of diesel in Co-injector CS will be injected in a very short time compared to the
overall injection time, \( t_{inj} \). In turn,

\[
t_{gas} = t_{inj} - t_{diesel}
\]  

(4.18)

is the amount of time remaining for the pure gas injection to take place. The flow rate for this pure gas flow will be slightly lower than the quasi-gas flow rate, however there will be less mass to inject in the allotted timeframe since the diesel mass has been entirely injected. From here, the addition of \( t_{gas} \) and \( t_{diesel} \) will give \( t_{inj} \) for this injection method and can be compared to the quasi-gas calculations of Co-injector B.

The abovementioned technique is applied in example calculations for Co-injector CSX and CS in Appendix F and is compared to Co-injector B. Figure 4.19 compares \( t_{inj} \) for the two models applied to each Co-injector\(^6\) and Table 4.16 shows the residual sum of the squares (RSS) values. By assuming that Co-injector B injects the diesel/gas mixture as a homogenous quasi-gas and Co-injector CS and CSX inject the fuel as a stratified pure diesel then pure gas injection, the flow rates in the given time (as specified by GPW) are in closest agreement as measured by the residual sum of squares. However, this method does not account for the entire discrepancy in flow rates.

\(^6\) \( t_{inj} \) is less than GPW for each point, as expected, since it represents the sustained time of maximum needle opening. A given GPW will have a definite period where the needle is in partially open/closed stages and inherent mechanical delays.
Figure 4.19: A comparison of the theoretical time for injection of both diesel and gas for each injector. ‘quasi’ denotes the homogenous mixture model and ‘stratified’ denotes pure diesel injection proceeded by pure gas injection.

<table>
<thead>
<tr>
<th></th>
<th>$B_{\text{quasi}}$</th>
<th>$B_{\text{stratified}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>CSX$_{\text{quasi}}$</td>
<td>2.74E-02</td>
<td>4.06E-02</td>
</tr>
<tr>
<td>CSX$_{\text{stratified}}$</td>
<td><strong>1.32E-02</strong></td>
<td>1.58E-02</td>
</tr>
<tr>
<td>CS$_{\text{quasi}}$</td>
<td>2.87E-01</td>
<td>3.42E-01</td>
</tr>
<tr>
<td>CS$_{\text{stratified}}$</td>
<td><strong>1.84E-01</strong></td>
<td>2.27E-01</td>
</tr>
</tbody>
</table>

Table 4.16: A comparison of the RSS-values for each Co-injector model. The lowest RSS-value for each comparison is denoted in bold and indicates closest agreement between models.
From Figure 4.19 it should be noted that if the flow rate differences were entirely attributed to fluid mechanic differences the curves in agreement should directly align. However, this is clearly not the case. Actuator response, and its effect on needle opening/closing times, seems to be a very important factor in the differences presented in these curves - with injector-to-injector actuator differences being apparent. Furthermore, the assumption that time of injection is constant for each is clearly too idealized. While fluid mechanics can account for some of the differences, the rest must be from actuator variations.

To further understand the two models, a comparison can be made of the predicted gas mass flow rates under identical diesel fueling rates. If the quasi-gas model is rearranged, the gas mass injected can be solved as a function of injection time by rearranging Eq. 4.13 such that,

\[ m_{\text{quasi}} = m_{\text{quasi}} \tau \]  

(4.19)

Where \( \tau \) is the theoretical injection time that assumes instantaneous and complete needle opening and closing. \( m_{\text{quasi}} \) is the total mass injected through this model and constitutes both diesel and gas,

\[ m_{\text{quasi}} = m_{\text{diesel}} + m_{\text{gas}} \]  

(4.20)
Furthermore, Recall from Equations 4.7 – 4.11 that if all injection parameters are kept constant, $m_{\text{ quasi}}$ is a function\(^7\) of the diesel and gas mass injected through the liquid mass fraction, $\eta$. Therefore,

$$m_{\text{ quasi}} = f(m_{\text{diesel}}, m_{\text{gas}}) \quad (4.21)$$

Now, to compute the theoretical gas flows it will be assumed that diesel the fueling rate is kept constant. This simplifies Equation 4.21 such that,

$$m_{\text{ quasi}} = f(m_{\text{gas}}) \quad (4.22)$$

Lastly, by combining Equations 4.19, 4.20, and 4.22, the injected mass quantity is now an implicit function of $m_{\text{gas}}$ and $\tau$ through,

$$m_{\text{gas}} = f(m_{\text{gas}})\tau - m_{\text{diesel}} \quad (4.23)$$

where $m_{\text{diesel}}$ is held constant. From Equation 4.23 the theoretical gas injection mass can now be computed as a function of any desired injection time given a constant diesel fueling rate.

A similar derivation will now be shown for the stratified model. For the stratified model the injection time is a function of both the time for diesel and gas to be injected. In the case of modeling the theoretical flow rate, $\tau$ becomes,

$$\tau = t_{\text{diesel}} + t_{\text{gas}} \quad (4.24)$$

\(^7\) The exact expression for $f(m_{\text{diesel}}, m_{\text{gas}})$ is inherently complex when $m_{\text{diesel}}$ and $m_{\text{gas}}$ are substituted into Eq. 4.11 and is shown in Appendix F for completeness.
where $t_{diesel}$ is given by Equation 4.17. $t_{gas}$ is a straightforward term to compute and takes the usual form,

$$
t_{gas} = \frac{m_{gas}}{m_{gas}}
$$

(4.25)

Since $m_{gas}$ is the mass flow rate of a pure gas injection, Equation 4.11 is largely simplified by not having to consider the liquid mass fraction. In fact, each term becomes a constant based on injection parameters and the fluid being injected (assumed to be nitrogen), thus reducing to an idealized choked flow and a constant value. If Equations 4.11, 4.24, and 4.25 are now combined and rearranged the theoretical mass flow rate as a function of $\tau$ can be calculated as

$$
m_{gas} = m_{gas} \left( \tau - \frac{m_{diesel}}{m_{diesel}} \right)
$$

(4.26)

Again, the injected gas mass will be compared under a constant diesel fueling rate, therefore both $m_{diesel}$ and $m_{gas}$ are constant. In turn, Eq. 4.26 becomes a linear function of the injection time, $\tau$.

Equations 4.23 and 4.26 are solved and plotted for values of $0 \leq \tau \leq 0.80$ ms to show the theoretical flow rates under similar conditions to the experimental GPWs used. Diesel flow rates are kept constant at 10 mg/inj and 20 mg/inj – comparable to experimental data. The discharge coefficient was set to 1 to attribute any flow rate differences to the
fluid mechanics involved. The results for both models are shown in Figure 4.20 and are compared to that of a pure gas injection under idealized choked flow conditions. As can be seen, the stratified model predicts gas flow rates that are higher, in both cases, than the corresponding quasi-gas model. The gas flow rates for the stratified model are 3.8 – 5.5 mg/inj higher than the corresponding quasi-gas model in this region. Again, this may account for some of the flow rate differences experimentally measured between Co-injectors. Overall, the models show considerably higher flow rates at all injection times than any of the experimental data. This is to be expected as the needle is assumed to instantaneously open to maximum lift and, again, instantaneously close. Moreover, when the constant diesel flow rates are added to the theoretical gas mass values, both models suggest slightly higher overall flow rates than the pure gas model.

Figure 4.21 shows the difference in gas flow of each model in relation to the pure gas flow – referred to as the ‘gas mass deficit’. It can be seen that the quasi-gas model clearly predicts less gas mass than the corresponding stratified model. However, a very important conclusion can be drawn from this. Whilst the stratified gas model predicts higher flow rates, this is not enough in itself to account for all the experimental gas mass flow differences registered between Co-injectors B and CS. Once more this suggests another factor is involved.
Figure 4.20: The theoretical flow rate of both the quasi-gas and stratified models, for two different fueling rates (10 mg/inj and 20 mg/inj), are shown. The flow rate of a pure gas injection is also shown for comparison.

Figure 4.21: The gas mass deficit of each model compared to the theoretical pure gas mass flow.
This calculation method reiterates that the underlying assumption of identical actuator response between Co-injectors is highly idealized. Actuator response between injectors is known to be quite variable as evident by the manufacturer tolerance curves of Section 2.3.3 and discussed in Inokoshi (2007). To account for the large differences in gas flow between Co-injector B and CS (see Figure 4.18, specifically GPW = 0.55 ms) there must be a difference in actuator response – with Co-injector CS having a more responsive actuator. Simply put, the difference in gas flow rates for similar injector bodies cannot be entirely attributed to fluid mechanic differences. However, from Figure 4.18 Co-injector B and CSX have flow variations well within the possibility of being attributed to fluid mechanic differences.

Lastly, it should be noted that in all likelihood there will be a combination of a two-phase diesel/gas and pure diesel being injected during actual Co-injector operation. The calculations herein are solely meant to attempt to build on this theory and theoretically prove the feasibility of differing fuelling strategies being present, since IVC and SCRE experiments show differing flow between Co-injectors, which are built from the same injector bodies. A more comprehensive fluid mechanics model, perhaps based on two-phase, two-fluid compressible flow (such as Haseli 2009), may lead to a more in-depth understanding of the differences in Co-injector mixing and fluid dynamics.

---

8 Recall Co-injector CSX has a larger sleeve than Co-injector B, thus it should have inherently lower gas flow. However, gas flow rates are similar as seen in Figure 4.18 (with CSX having higher diesel flow) suggesting the flow rate might be brought higher through stratified mixing.
4.3.2 AMESim Modeling

The simulation software AMESim (AMESim 2009) was used for simulation of an existing Westport HPDI Injector. AMESim is a software package used to simulate 1D hydraulic, electrical, and mechanical components that may be linked to produce multidisciplinary system models. Each component has the associated physics programmed and can be readily linked through relevant boundary conditions to form complex systems.

An existing AMESim HPDI injector model was provided for this study by Westport (Kostka 2008) and extended by Gu (2009). The model allows for parameters such as needle lift, needle lift duration, and gas/diesel flow rates to be simulated under a wide variety of conditions. The HPDI model provides useful information about the Co-injectors as the injectors use the same gas needles as the J36.

In the model used, the gas needle lift is a function of time and hydraulic parameters. Gas mass flow is computed from the needle lift using isentropic mass flow. The mass flow equation across the needle seat is solved based on the states of the sac and plenum (the sac and plenum pressures, temperatures and enthalpies must be known before the flow equation can be calculated). The sac and plenum states are found by calculating the mass and enthalpy flows to/from each volume and integrating for the remaining mass and enthalpy at each time step. The AMESim model uses a Peng-Robinson equation of state to connect the density and enthalpy to temperature and pressure.
The model was used to simulate gas flow and maximum needle lift as a function of chamber pressure. Parameters were chosen to most closely resemble that of the Co-injector. The working fluid was set to be nitrogen and the gas supply pressure was set to 21 MPa with a 2.5 MPa bias. The DPW was set to 0 ms so only leakage past the diesel needle into the plenum would be the only source of diesel flow (modeled as 2.5 mg/s of leakage). The GPW was set to 0.80 ms. The simulation was run using time steps of 0.01 ms for a duration of 3 ms. This provides 300 iterations for each simulation and was consistent with the procedures used by Westport and Kostka (2008).

Figure 4.22 shows the gas flow as a function of chamber pressure for the simulation run. The simulation results show some similarity to the measurements of Figure 4.3. Simulation flow rates are higher than the experimental data but this can partly be attributed to injector differences. Figure 4.23 shows the maximum needle lift for the same simulation. The similar increasing trend suggests maximum needle lift plays an important role in the gas flow. The needle lift duration did not change significantly for constant GPW with varying chamber pressures, further suggesting the needle lift height is the controlling parameter for gas flow. The delay between the commanded start of injection and first registered gas flow of the injector (equivalent to mechanical delay) did not vary under chamber pressure variation in the simulation.
Figure 4.22: Simulation gas flow as a function of chamber pressure for GPW = 0.80 ms.

Figure 4.23: Simulation maximum needle lift as a function of chamber pressure for GPW = 0.80 ms.
The dynamics of needle lift are an important mechanism in determining gas injection quantities. In particular, two important needle lift profiles are shown in Figure 4.24 and 4.25. Figure 4.24 shows a region with a high GPW such that maximum needle lift is attained and maintained for a definite period while Figure 4.25 shows a low GPW that allows only partial needle opening. An important aspect of these differing needle lift profiles is the fact that the maximum needle lift varies. Since Figure 4.25 does not reach maximum lift, the gas flow is restricted by a smaller cross-sectional area for the entirety of the injection duration. In contrast, when the needle reaches maximum lift and sustains its position, a period of consistent, unobstructed gas flow with maximum flow area can ensue.

Figure 4.24: A typical needle profile for a high GPW, whereby the needle reaches its maximum position and maintains it for a finite period of time.
The gas flow rates were simulated for varying GPWs using gas supply pressures of 21 and 28 MPa. The DPW was again set to 0 ms as this most closely resembles the co-injector. Varying the DPW has little effect as the two fuels are independently controlled in a production HPDI injector, aside from the leakage. Figure 4.26 shows increasing flow with GPW as expected for the simulations. The experimentally measured values are also plotted for comparison. Of importance in the simulations is the absence of the saddle region found in the experimental data, suggesting that this region is a non-idealized phenomenon of actuator and needle response. Both simulation curves are well within an order of magnitude agreement of the experimental data. The 21 MPa data overestimates the flow rate in the saddle region (0.55–0.70 ms) by as much as 5.8 mg/inj and then proceeds to underestimate the actual data by up to 10.6 mg/inj in the higher GPW region (0.70-0.80 ms). The 28 MPa simulated data has higher flow rates as expected, but underestimates the actual data by 3.4-18.2 mg/inj depending on GPW. The absence of the saddle region in the simulation appeared to cause this deviation from the experiments.

Figure 4.25: A typical needle profile for a low GPW where the needle reaches an intermediate needle opening (less than the maximum needle opening possible) and immediately begins to close.
As mentioned, the simulated gas flows have a noticeable omission of the saddle region found in the low gas pulse width region. To further extend the model and understand possible reasons for the saddle region the interaction between the gas needle and actuator were examined. The model assumes a harmonic oscillation between the two. As such, the damping coefficient was examined, and reduced by 60% and 90% and flow rates computed. The reduction in damping coefficient had no effect on the gas flow rates. This would suggest that gas needle oscillations (also referred to as needle bounce) likely has little to no effect on the gas flow rates in the saddle region of experimental flow testing, further implying that at the low GPW regions the gas needle does not have sufficient time to reach and maintain a maximum opening position.
4.4 Results from High-speed Imaging

4.4.1 Injector Electro-Mechanical Delay

The shroud sensor coupled with the first pulse sent to the injector trigger the camera imaging. Since the camera begins imaging upon a pulse sent to the injector, the mechanical delay can be determined as the time difference between the start of imaging and the first sign of injection. Mechanical delay measurements were averaged for a cycle of 5 injections after the injector reached steady state. The errors are one-sided and equal the frame rate. This is because the major uncertainty is that the injection begins in the time interval between the last frame of no injection and the first frame of injection. Figure 4.27 shows the mechanical delay of the Co-injector gas needle. For comparison the mechanical delay of the smaller J36 pilot needle is shown. Both are independent of DPW, GPW and chamber pressure as expected by force balance and simulation. However, gas and diesel supply pressures do have an effect on mechanical delay as confirmed by Figure 4.28. The reduction in mechanical delay due to higher supply pressures is thought to be due to the subsequent higher hydraulic pressure being able to drain faster through the control orifice, thus allowing the spring to lift the needle sooner. Moreover, the plenum gas pressure acts to assist the gas needle in opening. In turn, at higher supply pressures the needle opening time is expected to be faster.

---

9 Recall the electro-mechanical delay includes two components: the delay in the injector driver (0.1 ms) and the delay in the actuators (remaining delay).

10 The area of gas needle exposed to the chamber pressure is much less than that of the total area of the gas needle and plays a smaller role on overall force balance than supply pressure.
Figure 4.27: Injector mechanical delay vs. chamber pressure for DPW = 1.0 ms and GPW = 0.80 ms.

Figure 4.28: Mechanical delay as a function of gas supply pressure of the tested injectors.
4.4.2 Spray and Jet Characteristics

The images selected in this Section are of the highest quality accumulated. However, it should be stated that excluded images consistently showed the same spray and jet characteristics under identical operating conditions. The ultimate difference between the images selected in Figures 4.29 - 4.34 and those excluded is the overall clarity. Any window contamination from repeated testing or improper shroud operation acted to distort the overall quality of the image. It was crucial to ensure the IVC windows were free from contamination before an image set was taken. The images of the highest quality and those worth discussion are of Co-injector B and are presented herein. Co-injector CSX was sparsely imaged due injector malfunctions in February and August 2009 preventing a full test matrix of images to be produced with the same quality of Co-injector B – hence only mechanical delay was accumulated for Co-injector CSX.

Figures 4.29-4.34 show injections from the production J36 injector and Co-injector B\textsuperscript{11}. The images shown were taken at 70 bar chamber pressure and use the constant parameters found in Table 3.2. The co-injector images correspond to points found on the flow curves of Figure 4.3. Images are compared using a constant DPW and low/high

\textsuperscript{11} The printed hardcopy of this work may not provide adequate resolution to distinguish the features described in this section. As such, a supplemental electronic copy of the images entitled “Co-injector B Images – N. Birger Feb. 2010.pdf” is provided to the reader. The file locations of the full electronic video clips of the injectors can be found in Appendix G.
GPW. A time interval of 41 μs between frames was used. In each spray figure, images are taken at the following times:

a) 4th frame after start of injection (0.164 ms),

b) 25th frame (1.025 ms)

c) 100th frame (4.1 ms).

Figure 4.29 a) shows a shroud dimension for reference. Figures 4.29 and 4.30 show the standard J36 injecting diesel and gas separately. Both figures show very similar liquid jets in a) and b). This is to be expected because the DPW are the same and gas has not yet been injected. In the 100th frame (c), the injection from the gas holes of the injector is visible, indicating that some liquid is actually carried with the gas injection. In fact, for this injector, about 2.5 mg/inj of liquid leaks (unintentionally) into the gas plenum, as determined by IVC measurements with DPW = 0 ms.

Figures 4.31 and 4.32 show the co-injector diesel spray for a low diesel flow (DPW = 0 ms, only the leak past the needle match fit is allowed to enter the plenum). Figures 4.31 a) and 4.32 a) show a clearly wider, more dispersed jet for the co-injection process than the J36 pilot injection. Furthermore, the jet in 4.31 a) is much darker than in 4.32 a) indicating that a higher diesel/gas ratio is present in this injection. This is to be expected since Figure 4.31 is for a much lower GPW, which reduces the gas flow. As the injection evolves Figure 4.31 b) and 4.32 b) show distinctly different spray qualities. 4.31 b) remains much darker than 4.32 b) and has a less pronounced jet shape. 4.31 c) and 4.32 c) contrast the effects higher gas flow has on constant diesel flow. 4.32 c) has a more
dispersed less dense amount of diesel visible. This is to be expected as the gas acts to scavenge the diesel and disperse it faster.

Figures 4.33 and 4.34 show the Co-injector spray for a higher amount of diesel flow (DPW = 1.0 ms). The sprays in 4.33 a) and 4.34 a) are similar in shape to that of 4.31 a) and 4.32 a). It is difficult to tell if the jet is denser based on a 4.32 a) – 4.33 a) comparison, but 4.32 a) – 4.34 a) shows a visibly darker jet. As the jets evolve, 4.33 b) and 4.34 b) clearly show different sprays. 4.34 b) with the higher quantity of gas is clearly lighter, more dispersed, and occupies a larger volume than 4.33 b). This can only be attributed to more gas injected. 4.33 c) and 4.34 c) show very different time evolutions of the jets. In 4.33 c) there are dark regions present. However, in 4.34 c) the spray appears to rather dispersed and again occupies a larger volume. Any remnant of the initial jet shape appears to be gone and a diffused cloud of diesel is all that is visible.

It is evident that the Co-injector and J36 have different qualitative spray patterns. The J36 liquid injection has narrower, more well defined sprays that maintain their shape far longer than the two-phase coinjector jet. The volume occupied by diesel spray is larger for the two-phase jet than the J36 jet. This implies that for a given quantity of diesel the Co-injector will have diesel more evenly dispersed which may be especially useful in engine applications.
Figure 4.29: Images of J36. DPW = 1.0 ms, GPW = 0.55 ms.

Figure 4.30: Images of J36, DPW = 1.0 ms, GPW = 0.80 ms.
Figure 4.31: Co-injector B images.  
DPW = 0 ms, GPW = 0.55 ms.

Figure 4.32: Co-injector B images.  
DPW = 0 ms, GPW = 0.80 ms.
Figure 4.33: Co-injector B images. DPW = 1.0 ms, GPW = 0.55 ms.

Figure 4.34: Co-injector B images. DPW = 1.0 ms, GPW = 0.80 ms.
Chapter 5 - Conclusions

5.1 IVC and Apparatus Conclusions

An initial objective of this work was to develop a consistent flow characterization technique. The Injector Visualization Chamber (IVC) allows injector flow characterization and imaging of gas and liquid injectors at engine-relevant pressures. This is valuable because chamber backpressure has been shown to be critical in determining diesel/gas injection quantities. An innovative retractable shroud allows the injector to reach steady-state without fouling the windows; this is an essential for studies of new gas-assist injectors ("Co-injectors") but may be useful for studying other injectors. The IVC provides resolutions that are approximately ± 0.50 mg/inj for diesel flow and ± 0.10 mg/inj for high GPW (0.70 ms or greater) up to ± 1.00 mg/inj for the low GPW region where injector behavior is variable and flow rates approach the lower limit of detection capabilities. This is sufficient to draw important conclusions from prototype Co-injector flow characterization. Furthermore, the IVC is capable of quickly and easily interchanging prototype injectors if any abnormalities are detected in SCRE or flow testing. This saves time and valuable resources if injector operational problems are detected.
5.2 Flow Characterization Conclusions

A comprehensive summary of the experimentally measured quantities and computed values are shown for each Co-injector in Table 5.1. The discharge coefficient and mechanical delay show the fundamental parameters that are limited by injector design. Diesel flow rates are shown for all DPW/Biases tested. The gas flow rate ranges are shown for 21 MPa and 28 MPa supply pressures for both nitrogen and blended gas (where applicable). The minimum needle opening times are shown as well as the corresponding range of pure diesel flow. The computer characterization parameters $S$ and $R_S$ are also presented.
<table>
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<tr>
<th>Parameter</th>
<th>B</th>
<th>CSX</th>
<th>CS</th>
</tr>
</thead>
<tbody>
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<td>( C_D )</td>
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<td>0.922 ± 0.019</td>
<td>-</td>
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</tr>
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<td>8.1 ± 0.3</td>
<td>10.8 ± 0.4</td>
</tr>
<tr>
<td>1.0</td>
<td>12.6 ± 0.6</td>
<td>18.5 ± 0.7</td>
<td>26.6 ± 0.2</td>
</tr>
<tr>
<td>1.5</td>
<td>14.1 ± 0.5</td>
<td>19.9 ± 0.3</td>
<td>28.2 ± 0.4</td>
</tr>
<tr>
<td>2.0</td>
<td>20.3 ± 7.5</td>
<td>0 - 65.0</td>
<td>0 - 43.1</td>
</tr>
<tr>
<td>4.0</td>
<td>29.1 - 74.2</td>
<td>17.8 - 66.4</td>
<td>0.7 - 63.3</td>
</tr>
<tr>
<td>1.0</td>
<td>0 - 25.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas Flow Range* - 21 MPa (mg/inj)</td>
<td>8.7 - 47.0</td>
<td>3.0 - 42.1</td>
<td>0.2 - 37.6</td>
</tr>
<tr>
<td>Gas Flow Range* - 28 MPa (mg/inj)</td>
<td>33.3 - 81.8</td>
<td>19.4 - 76.3</td>
<td>20.0 - 76.3</td>
</tr>
<tr>
<td>Blended Gas Flow Range (mg/inj)</td>
<td>0.1 - 39.6</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Min. Needle Opening t - 21MPa (ms)</td>
<td>0.45</td>
<td>0.43</td>
<td>0.43</td>
</tr>
<tr>
<td>GPW Range for Pure Diesel Flow (ms)</td>
<td>0.43 - 0.50</td>
<td>0.43 - 0.55</td>
<td>0.45 - 0.60</td>
</tr>
<tr>
<td>GPW (ms)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.65</td>
<td>0.31 ± 0.09</td>
<td>0.55 ± 0.06</td>
<td>0.66 ± 0.06</td>
</tr>
<tr>
<td>0.80</td>
<td>0.35 ± 0.03</td>
<td>0.57 ± 0.09</td>
<td>0.64 ± 0.05</td>
</tr>
<tr>
<td>1.40</td>
<td>0.52 ± 0.04</td>
<td>0.57 ± 0.08</td>
<td></td>
</tr>
<tr>
<td>GPW (ms)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.65</td>
<td>1.15 ± 0.28</td>
<td>1.58 ± 0.19</td>
<td>1.79 ± 0.05</td>
</tr>
<tr>
<td>0.80</td>
<td>0.60 ± 0.04</td>
<td>1.32 ± 0.04</td>
<td></td>
</tr>
<tr>
<td>1.40</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 5.1: A characterization summary of the experimentally measured parameters and computed values of each Co-injector tested.

*Note the gas flow ranges listed are from GPW = 0.55 ms to 0.80 ms.
The IVC has clarified some important relationships between fluid pressures, flow rates and injector dynamics for each of the Co-injectors tested.

- The Co-injector gas flow quantities are directly affected by the amount of liquid present. If more liquid is present in the injector plenum less gas will be injected. Fortunately, a unique gas flow rate is set by a given DPW (for Co-injector B) or bias (for Co-injector CSX and CS) and GPW (fluid pressures being equal). The liquid flow rate, as expected, is independent of GPW.

- Every injector tested (including the J36) has a region of nonlinearity at low GPWs (0.55 – 0.65 ms). The reproducibility in the GPW of this “saddle” region with each injector tested indicates it is a phenomenon of Co-injector actuator response at these low GPWs. For GPWs greater than 0.70 ms the injectors work in a predictable manner, with gas flow rates increasing linearly.

- Injector supply pressure variations have profound effects on the quantities of diesel and gas injected. Co-injector gas flows increased twofold based on gas supply pressure increases from 21 MPa to 28 MPa, with no change in diesel fueling rates (since the bias was kept consistent during gas pressure variation). Higher injector supply pressures showed less gas flow sensitivity per quantity of diesel injected at a specific GPW, as evident by lower $R_s$-values at 28 MPa than 21 MPa.
• By varying chamber pressure it was shown that the gas flow could be lowered by an order of magnitude through a 70 bar decrease in chamber pressure. Co-injector mechanical delay, however, is independent of chamber backpressure.

• By varying the composition of the gas supply (through matching the density to methane) the Co-injectors exhibited flow rates that were lower than the corresponding nitrogen tests. Through the calculations of Appendix D it was expected that the gas flow rates be lowered by 26% at each condition. Experimental results for Co-injector B showed that gas flow rates were reduced by 13-29% during blended gas testing. Co-injector CSX showed flow rates were lower by 26-51%. However, both injector flow rates mainly strayed from the expected gas flow in the low GPW region, where flow non-linearity is present. At higher GPW (greater than 0.70 ms) both Co-injector B and CSX were consistently near the expected 26% lower gas flow rate.

• The gas flow rates of Co-injector B are less sensitive to the presence of diesel than both Co-injector CSX and CS due to the method of internal diesel mixing as computed by the $R_S$-value. Co-injector CS, and its larger plenum volume in comparison to Co-injector CSX, displays the highest sensitivity. Both Co-injector B and CSX have less sensitive gas flow rates to the presence of diesel at 28 MPa gas flow rates.
• The $S$-value suggests the rate of gas flow change with varying chamber pressure is lower when examined at GPWs in the non-linear saddle region (GPW = 0.65 ms) than in regions of consistent gas flow (GPW = 0.80 ms and 1.40 ms). This further implies non-linear Co-injector operation in the saddle region due to differing actuator responses.

• The discharge coefficient was measured for each injector tested. Results showed that the production J36 (without sleeve, and with very little liquid present during gas injection) had the highest discharge coefficient. This is to be expected as Co-injector internal geometry includes volume-reducing sleeves that restrict the amount of gas being injected. Co-injector B and CS have identical inner geometries and, in turn, statistically similar discharge coefficients. Co-injector CSX has a larger and more restricting inner sleeve and subsequently the lowest discharge coefficient.

• The overall operation of single-needle Co-injectors (CSX and CS) is viable with readily controllable flow rates and low fluctuations. This is of importance as understanding the performance and flow rate response of any future prototype is valuable to subsequent data interpretation from parallel engine experiments.
5.3 Co-injector Modeling Conclusions

5.3.1 Fluid Mechanics Modeling Conclusions

Notable flow rate differences between Co-injectors were experimentally measured. The engine work of Laforet (2009) and Brown et al. (2009) suggest differences in diesel introduction methods allow for different internal diesel distributions. Two theoretical models were established that factor the diesel being uniformly distributed throughout the gas in a homogenous mixture (quasi-gas model) or distributed as a stratified diesel/gas mixture (stratified model). The models have the following implications:

- Co-injector CS has considerably higher gas flow rates than Co-injector B for equivalent diesel fuelling amounts whilst having the same internal dimensions. This appears to contradict the higher diesel sensitivity ($R_S$), but appears to be an artifact of the needle opening much earlier than the other injectors. Moreover, some of the difference may be due to the internal mixing of the diesel. Co-injector CS likely has a diesel pool near the tip of the injector with less homogeneous mixing of the two fuels. From this, the majority of diesel is injected first, followed by a mostly gaseous scavenging of the fuel. Co-injector B has the diesel internally introduced at higher velocities and likely forms a more homogenous two-phase mixture with a lower speed of sound and subsequently lower gas fuelling amounts. A similar argument is presented for Co-injector CSX in comparison to Co-injector B.
Calculations are shown that support this theory for attributing some of the gas flow differences to fluid mechanics.

- Fluid mechanics can be attributed to some of the differences in flow rates – with the stratified model predicting higher flow rates than the quasi-gas model. However, it cannot be linked as the only difference between Co-injectors as evident by Figures 4.19 - 4.21. There must be a difference in actuator response to account for the large gas flow rate differences between Co-injector B and Co-injector CS.

5.3.2 AMESim Modeling Conclusions

Some of the dynamic behavior can be explained with an AMESim model of a production Westport J36 injector. That model includes the major dynamic and hydraulic elements of the injector and was previously validated against flow measurements for the J36 injector (not the Co-injector).

- The model verified the increasing trend in gas flow rates with increasing chamber backpressure and injector supply pressure.

- The model confirmed that the injector mechanical delay is not affected by chamber pressure and is a characteristic of actuator response and diesel supply pressure – consistent with experimental results of Section 4.4.1.
• The model suggested that the maximum gas needle lift is the parameter most affected by chamber pressure and follows a similar decreasing trend as gas injection mass.

• The model was unable to replicate the saddle region consistently found in flow characterization. However, the experimental data and the theoretical gas flow rate simulation data were well within an order of magnitude agreement. The absence of the saddle region in the simulation appeared to cause much of the deviations from the experimental data.

• The model suggested that needle bounce may not be a cause for the experimental saddle region as varying the needle damping coefficient had little effect on the gas injection mass in the low GPW region.

• In the low GPW regions the needle lift may not reach maximum, as seen by the triangular needle lift profile, and the actual lift may be highly variable based on inconsistencies in actuator response. This would explain the high variability measured experimentally in the low GPW flow regions and the subsequent nonlinearity in the injection quantities of the saddle region.

These conclusions lend some credibility to the predictions of parameters not measureable in the experiments and confirm some of the experimental phenomenon measured experimentally in flow testing.
5.4 High-Speed Imaging Conclusions

The IVC has been used to uncover some interesting phenomena present in co-injection. With the camera system used in this study, individual droplets are not visible in the gas/liquid spray. Although imaging is qualitative, the IVC can detect differences in spray characteristics for different injection parameters.

- The imaging has shown two-phase co-injector jets are clearly different from pure diesel J36 jets. The volume occupied by the two-phase jet is also larger than the volume of the pure diesel jet.

- For higher GPW, the co-injector acts to disperse and dissipate the diesel faster, as indicated by a lighter (less light scattering) plume.

Imaging using the existing shroud system allowed for injector mechanical delay to be measured; a parameter that was previously unknown.

- Injector mechanical delay could be adequately measured as the time difference between commanded start of injection and first sign of injection via imaging. Injector mechanical delay was shown to be independent of chamber backpressure. However, injector fuel supply pressures had an effect: with higher supply pressures reducing the mechanical delay.
5.5 Recommendations and Future Work

An immediate source of work is to complete the flow characterization for Co-injector CS. This will in many ways also allow the researcher the opportunity to verify the repeatability of their work with the data already accumulated and extend to the points not measured. At the time of conclusion of this work, Co-injector CS is in a fully operational state and can be used for such testing.

The most significant extension of the work performed herein lies in the imaging. In the future, more controlled illumination and careful image processing may allow semi-quantitative measurements of the liquid content in the gas jets. The new shroud created in December 2009 allows for steady-state imaging, performed at a head-on profile. This will allow unimpeded views of jet penetration with no ambiguity or angle corrections from the previous side-profile. As such, this new shroud system should be used to attempt to further understand the complex two-phase flow phenomena exhibited by such Co-injectors. Furthermore, due to technical malfunctions Co-injector CSX and CS images were not of sufficient quality to include in this work. Future work should involve producing a full factorial test matrix of imaging points for the Co-injectors.

Initial MATLAB coding was performed for future semi-quantitative analysis of imaging results as shown in Appendix H. The first step in successful image analysis is background image subtraction. This is done by averaging a combination of images before and after injection occurs. The average background image is then subtracted from the time evolution of injection in each instance. This will provide a clearer view of only the jet
penetration in question. From this, distances can be computed as a function of time from an input reference dimension such as injector tip length. By understanding the two-phase jet evolution over time comparison to theory can be made for a deeper understanding of the complex processes involved.

As was done with Co-injector B, each of the test points used for flow characterization were also imaged. However, due to time restrictions, Co-injector CSX and CS were not imaged using the old shroud configuration. This imaging is of high priority when using the new shroud arrangement because it may help in understanding the diesel/gas distribution of injections with continuous leakage of diesel into the reservoir. It may be through imaging of these Co-injectors that the possible diesel “pooling” effect near the tip can be verified.

Westport Innovations has the capability to equip various injectors with needle position sensors. Such sensors would be valuable in creating needle lift profiles for any given Co-injector prototype — similar to those found theoretically in Section 4.3.2. It could also allow the investigator to study the effects of actuator response and the differences between needle opening and closing times. Furthermore, by having a needle lift sensor it would be possible to study regions of sufficiently small needle lift such that the gas flow is choked in the region between the needle and the seat rather than the nozzle.

The AMESim modeling can be extended to directly model two-phase flows of the Co-injector. This is quite a challenging task as the software needs separate coding written to
accommodate two-phase flows. However, by doing so a more thorough understanding of the overall co-injection process may ensue that could uncover other important phenomena not previously understood.

Lastly, a refined start-up procedure may be created for the IVC while using newer generations of Prototypes. Since these Co-injectors use flow restrictors, diesel is allowed to accumulate continuously and indefinitely in the gas plenum under any pressure bias. If the Co-injector is not firing for a prolonged period of time under normal supply pressures, this will cause unwanted diesel to potentially accumulate far upstream in the gas line. A check valve of proper pressure rating immediately upstream of the Co-injector will ensure that diesel will not accumulate far in the gas line. This will permit the Co-injector to reach steady-state faster.
References


Inokoshi, K. 2007. (UBC/Westport Innovations Correspondence).


Kostka, Peter. 2008. UBC/Westport Innovations Correspondence.


APPENDICES

Appendix A - Instrumentation

This appendix lists all of the instrumentation used over the course of the study. The instrumentation is listed in tables by order of fluid measurement apparatus (Table A.1), mechanical equipment (Table A.2), optical equipment (Table A.3), and miscellaneous equipment (Table A.4). The apparatus location in Rusty Hut 122 at the time of testing is shown in Figure A.1 for reference and can be found by number in each table.

Figure A.1: The apparatus location and laboratory setup of RH 122.
<table>
<thead>
<tr>
<th>Device</th>
<th>Manufacturer</th>
<th>Model</th>
<th>Range</th>
<th>Accuracy</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 Coriolis Mass Flow Meter</td>
<td>Endress + Hauser</td>
<td>Promass 80A</td>
<td>0-20 kg/h</td>
<td>±0.25% FS</td>
</tr>
<tr>
<td>2 Electronic Balance</td>
<td>Adams Lab</td>
<td>PGW 753</td>
<td>0-750 g</td>
<td>±1 mg</td>
</tr>
<tr>
<td>3 Pressure Transducer</td>
<td>GP:50</td>
<td>211</td>
<td>0-20,000 psi</td>
<td>±0.2% FS</td>
</tr>
</tbody>
</table>

Table A.1: A list of the fluid measurement apparatus used.

<table>
<thead>
<tr>
<th>Device</th>
<th>Manufacturer</th>
<th>Model</th>
<th>Range</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>4 Priming Pump</td>
<td>JABSCO</td>
<td>MDXT</td>
<td>0-360 gal/h</td>
<td></td>
</tr>
<tr>
<td>5 Main Fuel Pump</td>
<td>Bosch</td>
<td>CP3.3</td>
<td>300-1600 bar</td>
<td>Run at 300 rpm</td>
</tr>
<tr>
<td>6 Main Fuel Pump Motor</td>
<td>Marathon Electric</td>
<td>56C17F5326</td>
<td>3/4 HP</td>
<td>Run at 1725 rpm</td>
</tr>
<tr>
<td>7 Back Pressure Regulator</td>
<td>Tescom</td>
<td>26-1762-26</td>
<td>0-6000 psi</td>
<td></td>
</tr>
<tr>
<td>8 Pulsation Dampener</td>
<td>Flowguard</td>
<td>PD-0100</td>
<td>0-6000 psi</td>
<td>Charged to 3000 psi</td>
</tr>
<tr>
<td>9 Fuel Filter</td>
<td>Swagelok</td>
<td>SS-4TF-15</td>
<td>15 micron</td>
<td></td>
</tr>
<tr>
<td>10 Dome-Loaded Regulator</td>
<td>GO Inc.</td>
<td>DL57-111396-E</td>
<td>0-6000 psi</td>
<td></td>
</tr>
<tr>
<td>11 4-way Valve</td>
<td>Swagelok</td>
<td>SS-43YF2</td>
<td>0-6000 psi</td>
<td>1/8&quot; connections</td>
</tr>
<tr>
<td>12 Heat Exchanger</td>
<td>Custom Built</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table A.2: A list mechanical equipment used.
<table>
<thead>
<tr>
<th>Device</th>
<th>Manufacturer</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>13 High-Speed Camera</td>
<td>Vision Research</td>
<td>Phantom v7.1</td>
</tr>
<tr>
<td>14 Camera Lens</td>
<td>Nikon</td>
<td>50mm, 1:1.2</td>
</tr>
<tr>
<td>15 500 W Illuminating Light</td>
<td>TPI Corp.</td>
<td>F-QH-1</td>
</tr>
<tr>
<td>16 Software</td>
<td>Vision Research</td>
<td>Version 630</td>
</tr>
</tbody>
</table>

Table A.3: A list of the imaging and optical equipment used.

<table>
<thead>
<tr>
<th>Device</th>
<th>Manufacturer</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid Fuel</td>
<td>VISCOR</td>
<td>1487</td>
</tr>
<tr>
<td>Gaseous Fuel</td>
<td>Praxair</td>
<td>Nitrogen/6000 psi</td>
</tr>
<tr>
<td>Software</td>
<td>National Instruments</td>
<td>LabView v8.0</td>
</tr>
<tr>
<td>Counter</td>
<td>National Instruments</td>
<td>BNC-2121</td>
</tr>
<tr>
<td>DAQ Board</td>
<td>National Instruments</td>
<td>BNC-2120</td>
</tr>
</tbody>
</table>

Table A.4: A list of the miscellaneous equipment used and not shown in Figure A.1.
The IVC drawings are as follows:

- Completed assembly view of IVC with all components
- Exploded view of all components
- Main body casing
- Endplate
- Injector plate
- IVC leg supports
- Window
- Window adapter
MID-RANGE
SIO-BAR
OPTICAL BOMB
SUPPORT

MATERIAL: STEEL
Appendix C - Instrument Calibrations

C.1 Pressure Transducer Calibrations

The Pressure Transducers were calibrated on May 22, 2008 and re-checked on August 7, 2009. All calibrations were done by connecting the transducers in series to a high pressure nitrogen tank and Cole Parmer Model 68036 Digital Pressure Gauge. Voltage readings were performed by using a Circuit-Test DMR-3600 Digital Multimeter.

Pressure Transducer #1:
This transducer was removed, as it was a redundant measurement of the diesel pressure slightly upstream of Pressure Transducer #2.

Pressure Transducer #2:

<table>
<thead>
<tr>
<th>Pressure (psi)</th>
<th>Voltage (V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.02</td>
</tr>
<tr>
<td>200</td>
<td>0.228</td>
</tr>
<tr>
<td>408</td>
<td>0.428</td>
</tr>
<tr>
<td>604</td>
<td>0.623</td>
</tr>
<tr>
<td>804</td>
<td>0.819</td>
</tr>
<tr>
<td>1012</td>
<td>1.029</td>
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<tr>
<td>1213</td>
<td>1.228</td>
</tr>
<tr>
<td>1424</td>
<td>1.443</td>
</tr>
<tr>
<td>1615</td>
<td>1.632</td>
</tr>
<tr>
<td>1812</td>
<td>1.831</td>
</tr>
<tr>
<td>2000</td>
<td>2.022</td>
</tr>
</tbody>
</table>

Table C.1: Pressure vs. voltage measurements of Pressure Transducer #2.
Pressure Transducers #3 and #4:

<table>
<thead>
<tr>
<th>Pressure (psi)</th>
<th>PT #3 Voltage (V)</th>
<th>PT #4 Voltage (V)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>-0.022</td>
<td>0.0055</td>
</tr>
<tr>
<td>206</td>
<td>0.185</td>
<td>0.213</td>
</tr>
<tr>
<td>403</td>
<td>0.380</td>
<td>0.404</td>
</tr>
<tr>
<td>608</td>
<td>0.591</td>
<td>0.618</td>
</tr>
<tr>
<td>802</td>
<td>0.792</td>
<td>0.813</td>
</tr>
<tr>
<td>1000</td>
<td>0.986</td>
<td>1.021</td>
</tr>
<tr>
<td>1206</td>
<td>1.202</td>
<td>1.228</td>
</tr>
<tr>
<td>1420</td>
<td>1.41</td>
<td>1.429</td>
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<tr>
<td>1610</td>
<td>1.593</td>
<td>1.621</td>
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<tr>
<td>1817</td>
<td>1.803</td>
<td>1.832</td>
</tr>
<tr>
<td>2010</td>
<td>2.008</td>
<td>2.029</td>
</tr>
</tbody>
</table>

Table C.2: Pressure vs. voltage measurements of Pressure Transducer #3 and #4.

Figure C.1: A plot of the voltage vs. pressure of the three pressure transducers used and showing the linearity of their response.
C.2 Coriolis Mass Flow Meter Calibrations

The Coriolis Mass Flow Meter was cross-checked with a similar model meter at Westport Innovations on August 10, 2009. Both flow meters were connected in series to a high pressure nitrogen bottle equipped with pressure regulator and needle valve. Two tests were conducted at 1500 psi and 3000 psi – with each pressure point having three separate needle valve settings to target low, medium, and high flow rates. A comparison of the digital readouts is shown in Tables C.3 and C.4.

<table>
<thead>
<tr>
<th>Pressure=3000 psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
</tr>
<tr>
<td>----------</td>
</tr>
<tr>
<td>10</td>
</tr>
<tr>
<td>30</td>
</tr>
<tr>
<td>50</td>
</tr>
<tr>
<td>10</td>
</tr>
<tr>
<td>30</td>
</tr>
<tr>
<td>50</td>
</tr>
<tr>
<td>10</td>
</tr>
<tr>
<td>30</td>
</tr>
<tr>
<td>50</td>
</tr>
</tbody>
</table>

Table C.3: A comparison of the Wesport flow meter to the Coriolis Mass Flow Meter used in this study (UBC) conducted at 3000 psi.
Table C.4: A comparison of the Wesport flow meter to the Coriolis Mass Flow Meter used in this study (UBC) conducted at 1500 psi.

Since calibration errors could potentially propagate through both flow meters a second method of calibration was used that relied on the measurement of integrated mass flow through the Coriolis Mass Flow Meter. In this method, the flow meter was connected in series to a high pressure nitrogen bottle with pressure regulator and a 1 L accumulator at atmospheric pressure. A needle valve was connected on the inlet side of the flow meter to control the flow. Next, the nitrogen supply pressure was set to the PVC working pressure of 3000 psi. The flow rate was set to a value in the range 0.50 – 1.50 kg/h - typical flow rates used by the PVC. It was important to stay in this range as anything higher or lower could cause the flow meter to be out of range and produce inaccuracies in the total mass flow integration. The accumulator was allowed to charge until a $\Delta m$ of $\sim 0.100$ kg was registered through the flow meter. The change is pressure $\Delta P$ (measured via Cole Parmer Model 68036 Digital Pressure Gauge) was then recorded after the accumulator was allowed to reach equilibrium. From $\Delta P$, $V$, and $\Delta m$ was computed and compared to what
was registered by the flow meter through Equation C.1. This method was done five times to ensure repeatability. Measurements and calculations are shown in Table C.5 and confirm the accuracy of the flow meter.

\[ \Delta m = \frac{\Delta PV}{RT} \]  

(C.1)

<table>
<thead>
<tr>
<th>Run</th>
<th>(\Delta m) measured (kg)</th>
<th>(\Delta P) (psi)</th>
<th>(\Delta m) calculated (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.090</td>
<td>1139</td>
<td>0.091</td>
</tr>
<tr>
<td>2</td>
<td>0.104</td>
<td>1290</td>
<td>0.104</td>
</tr>
<tr>
<td>3</td>
<td>0.104</td>
<td>1288</td>
<td>0.104</td>
</tr>
<tr>
<td>4</td>
<td>0.105</td>
<td>1321</td>
<td>0.106</td>
</tr>
<tr>
<td>5</td>
<td>0.091</td>
<td>1142</td>
<td>0.092</td>
</tr>
</tbody>
</table>

Table C.5: A comparison of integrated mass flow measurements to calculated values based on the change in pressure of the accumulator.
Appendix D - Flow Rate Comparisons

Memorandum

From: Wenjun Gu
To: Dr. Rogak
Date: June 5, 2009
Subject: Comparison of Nitrogen and Blended Gas Mass Flow Rate

Known:
Blended gas contains 50% Helium and 50% Nitrogen.
Both flows are choked.

Required:
Flow rate ratio

Assumptions:
1. The flow is isentropic.
2. Both nitrogen and the blended gas are ideal.
3. They have the same stagnation pressure and temperature (\(P_0, T_0\)).
4. The tubes used are of the same dimensions.
5. Heat exchange effects on the outside part of the tubes are neglected.

For \(N_2\):

\[
R = 0.2968kJ/lkg \cdot K
\]
\[
k = 1.4
\]
For Blended Gas:

The blended gas is in 4000 psi, and is made from 2000 psi of Helium gas, and then adding N\textsubscript{2} until 4000 psi is reached.

From the partial pressure law,
\[ P_{N_2} = P_{blend} - P_{He} = 2000 \text{ psi} \]

\[ \rho_{He} = \frac{P_{He}}{R_{He} \cdot T} = \frac{2000 \text{ psi} \times 6894.76 \text{ Pa/psi}}{2077.1 \text{ J/kg K} \times 298 \text{ K}} = 22.278 \text{ kg/m}^3 \]

\[ \rho_{N_2} = \frac{P_{N_2}}{R_{N_2} \cdot T} = \frac{2000 \text{ psi} \times 6894.76 \text{ Pa/psi}}{296.8 \text{ J/kg K} \times 298 \text{ K}} = 155.91 \text{ kg/m}^3 \]

The mass of the blended gas is the sum of the mass of nitrogen and helium.

\[ m_{blended} = m_{He} + m_{N_2} \]
\[ \rho_{blended} \cdot V_{bottle} = \rho_{He} \cdot V_{bottle} + \rho_{N_2} \cdot V_{bottle} \]
\[ \rho_{blended} = \rho_{He} + \rho_{N_2} = 178.188 \text{ kg/m}^3 \]

Hence, the gas constant \( R \) for this blended gas is found.

\[ R_{blended} = \frac{P_{blended}}{\rho_{blended} \cdot T} = \frac{4000 \text{ psi} \times 6894.76 \text{ Pa/psi}}{178.188 \text{ kg/m}^3 \times 298 \text{ K}} = 519.379 \text{ J/kg K} \]
From the enthalpy and internal energy, $C_p$ and $C_v$ for this blended gas are found. Thus, their ratio $k$ is obtained.

$$h = C_p \cdot T$$

$$C_{P,\text{blended}} \cdot T = C_{P,\text{He}} \cdot T + C_{P,N_2} \cdot T$$

$$C_{P,\text{blended}} = C_{P,\text{He}} + C_{P,N_2} = 2.193 + 1.042 = 6.235 \frac{kJ}{kg \cdot K}$$

$$u = C_v \cdot T$$

$$C_{V,\text{blended}} \cdot T = C_{V,\text{He}} \cdot T + C_{V,N_2} \cdot T$$

$$C_{V,\text{blended}} = C_{V,\text{He}} + C_{V,N_2} = 3.116 + 0.745 = 3.861 \frac{kJ}{kg \cdot K}$$

$$k = \frac{C_{P,\text{blended}}}{C_{V,\text{blended}}} = 1.6149$$

Therefore, the mass flow rate of the blended gas is:

$$\dot{m}_{\text{max}} = \left( k^{\frac{1}{2}} \right) \left( \frac{2}{k+1} \right)^{\frac{1}{2(k-1)}} \cdot A^* \cdot \rho_0 \sqrt{R_{\text{blended}} \cdot T_0}$$

$$= 0.71865 \cdot A^* \cdot \rho_0 \sqrt{R_{\text{blended}} \cdot T_0}$$

$$= \frac{0.71865A^* \cdot P_0}{\sqrt{R_{\text{blended}} \cdot T_0}}$$

Comparison of the Mass Flow Rates

$$\frac{\dot{m}_{\text{N}_2}}{\dot{m}_{\text{blended}}} = \frac{0.6847A^* \cdot P_0}{\sqrt{R_{\text{N}_2} \cdot T_0}} = \frac{0.6847 \cdot \sqrt{R_{\text{N}_2}}}{0.71865 \cdot \sqrt{R_{\text{blended}}}} = 1.26$$
Appendix E - Discharge Coefficient Calculations

The discharge coefficient is computed by using the method presented in Section 4.1.2.

Recall,

\[ C_D = \frac{m_{Exp}}{m_{ideal}} \]  

(E.1)

where

\[ m_{ideal} = A_i \sqrt{\gamma \rho_o \left( \frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{\gamma-1}}} \]  

(E.2)

and

\[ m_{Exp} = \frac{m_g^{GPW=2.0ms} - m_g^{GPW=1.6ms}}{\Delta GPW} \]  

(E.3)

Nitrogen gas has the following properties:

\[ R = 296.8 \text{ J/kg} \cdot \text{K} \]
\[ \gamma = 1.4 \]

The Co-injectors were operated at a gas supply pressure of 21 MPa. The total area is a sum of each of the gas hole areas. Its value is given from the design dimensions. Therefore,

\[ P_o = 21 \times 10^6 \text{Pa} \]
\[ A_i = 3.08 \times 10^{-6} \text{m}^2 \]

Thus,

\[ \rho_o = \frac{P_o}{R \cdot T} = \frac{21 \times 10^6 \text{Pa}}{296.8 \text{J/kg} \cdot \text{K} \times 295 \text{K}} = 239.7 \text{kg/m}^3 \]
From these values the idealized mass flow rate can be readily computed as,

\[ m_{\text{ideal}} = 0.150 \text{ kg/s} \]

**E.1 Co-injector B Discharge Coefficient**

Now, Equation E.3 will be applied to the high GPW data of each co-injector and the value, \( m_{\text{Exp}} \), will be compared to \( m_{\text{ideal}} \) calculate the discharge coefficient. The high GPW data for Co-injector B is shown in Figure E.1. The average of the two slopes produced the value of,

\[ m_{\text{Exp}} = 0.1067 + 0.0023 \text{ kg/s} \]

Thus giving the discharge coefficient as,

\[ C_D = 0.711 \pm 0.015 \]

![Figure E.1: High GPW data for Co-injector B.](image-url)
E.2 Co-injector CSX Discharge Coefficient

The high GPW data for Co-injector CSX is shown in Figure E.2. The average of the two slopes produced the value of,

\[ m_{Exp} = 0.0999 + 0.0026 \text{ kg/s} \]

Thus giving the discharge coefficient as,

\[ C_D = 0.666 \pm 0.017 \]

Figure E.2: High GPW data for Co-injector CSX.
E.3 Co-injector CS Discharge Coefficient

The high GPW data for Co-injector CSX is shown in Figure E.3. The average of the two slopes produced the value of,

\[m_{\text{Exp}} = 0.1107 + 0.0035 \text{ kg/s}\]

Thus giving the discharge coefficient as,

\[C_D = 0.738 \pm 0.023\]

Figure E.3: High GPW data for Co-injector CSX.
Appendix F - Quasi-gas and Stratified Flow Modeling

F.1 Fluid Mechanics Modeling Calculations

The models assumes the following constant parameters:

- $C_D$: Dependent on Co-injector examined
- $A_i = 3.08 \times 10^{-6} m^2$ (Based on Co-injector Design)
- $c_{p,nitrogen} = 1040 J/kg \cdot K$
- $c_{v,nitrogen} = 741 J/kg \cdot K$
- $c_{viscor} = 1800 J/kg \cdot K$
- $\rho_{nitrogen} = 301 kg/m^3$
- $\rho_{viscor} = 850 kg/m^3$
- $P_o = 21 MPa$

The examples of Sections F.1.1 and F.2.2 are applied to all conditions and summarized in Tables F.1 and F.2.

F.1.1 Quasi-gas Model Example

$$m_{quasi} = C_D A_i \sqrt[\gamma+1]{\frac{\gamma P_o \rho_{\text{rel}}}{\gamma+1}}$$  \hspace{1cm} (F.1)
where the terms $\gamma(\eta)$ and $\rho(\eta)$ vary based on liquid mass fraction examined. The total 
mass injected is therefore a function of the mass of diesel and gas injected.

As an example consider the Co-injector B ($C_D = 0.711$) data point for DPW = 1.5 ms and 
GPW = 0.80 ms. The diesel and gas flow rates were 10.8 mg/inj and 37.6 mg/inj. By 
applying these values to the liquid mass fraction of Equation F.1 the following 
parameters are calculated,

\[ m_{\text{quasi}} = 0.115 \text{kg/s} \]

The total injected mass is

\[ m_{\text{quasi}} = m_{\text{diesel}} + m_{\text{gas}} = 10.8 \text{mg/inj} + 37.6 \text{mg/inj} = 48.4 \text{mg/inj} \]

Thus,

\[ t_{\text{inj, quasi}} = \frac{48.4 \text{mg/inj}}{0.115 \text{kg/s}} = 0.421 \text{ms} \]

A list of computed quasi-gas $t_{\text{inj}}$ is shown in Table F.1 and Figure F.2 for comparison. 
The example value for $t_{\text{inj, quasi}}$ is underlined.

F.1.2 Stratified Model Example

The ideal gas flow rates (assuming $C_D = 1$) for diesel and gas are given as:

\[ m_{\text{ideal, gas}} = A_1 \sqrt{\gamma p_o \rho_o \left( \frac{2}{\gamma + 1} \right)^{\gamma+1}} \]

\[ m_{\text{ideal, diesel}} = A_2 \sqrt{2 \rho_{\text{diesel}} \Delta P} \]
They assume the values of:

\[
\begin{align*}
    m_{\text{ideal, diesel}} &= 0.592 \text{kg/s} \\
    m_{\text{ideal, gas}} &= 0.150 \text{kg/s}
\end{align*}
\]

and each value will be correspondingly reduced based on Co-injector discharge coefficient. Consider example values as Section F.1.1. If the stratified model is now applied we get the following values:

\[
\begin{align*}
    m_{\text{diesel}} &= 0.421 \text{kg/s} \\
    m_{\text{ideal, gas}} &= 0.106 \text{kg/s}
\end{align*}
\]

Thus,

\[
\begin{align*}
    t_{\text{diesel}} &= \frac{m_{\text{diesel}}}{m_{\text{diesel}}} = \frac{10.8 \text{mg/inj}}{0.421 \text{kg/s}} = 0.026 \text{ms} \\
    t_{\text{gas}} &= \frac{m_{\text{gas}}}{m_{\text{gas}}} = \frac{37.6 \text{mg/inj}}{0.106 \text{kg/s}} = 0.355 \text{ms} \\
    t_{\text{inj, strat}} &= t_{\text{diesel}} + t_{\text{gas}} = 0.026 + 0.355 = 0.381 \text{ms}
\end{align*}
\]

This is example is also underlined in Table F.1
<table>
<thead>
<tr>
<th>GPW (ms)</th>
<th>$t_{inj,\text{quasi}}$ (ms)</th>
<th>$t_{inj,\text{strat}}$ (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.55</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>0.60</td>
<td>0.04</td>
<td>0.04</td>
</tr>
<tr>
<td>0.65</td>
<td>0.09</td>
<td>0.07</td>
</tr>
<tr>
<td>0.70</td>
<td>0.22</td>
<td>0.19</td>
</tr>
<tr>
<td>0.75</td>
<td>0.32</td>
<td>0.28</td>
</tr>
<tr>
<td>0.80</td>
<td><strong>0.42</strong></td>
<td><strong>0.38</strong></td>
</tr>
</tbody>
</table>

**Co-injector CSX**

<table>
<thead>
<tr>
<th>GPW (ms)</th>
<th>$t_{inj,\text{quasi}}$ (ms)</th>
<th>$t_{inj,\text{strat}}$ (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.55</td>
<td>0.00</td>
<td>0.00</td>
</tr>
<tr>
<td>0.60</td>
<td>0.17</td>
<td>0.13</td>
</tr>
<tr>
<td>0.65</td>
<td>0.18</td>
<td>0.14</td>
</tr>
<tr>
<td>0.70</td>
<td>0.22</td>
<td>0.18</td>
</tr>
<tr>
<td>0.75</td>
<td>0.36</td>
<td>0.31</td>
</tr>
<tr>
<td>0.80</td>
<td>0.44</td>
<td>0.39</td>
</tr>
</tbody>
</table>

**Co-injector CS**

<table>
<thead>
<tr>
<th>GPW (ms)</th>
<th>$t_{inj,\text{quasi}}$ (ms)</th>
<th>$t_{inj,\text{strat}}$ (ms)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.55</td>
<td>0.25</td>
<td>0.22</td>
</tr>
<tr>
<td>0.60</td>
<td>0.29</td>
<td>0.25</td>
</tr>
<tr>
<td>0.65</td>
<td>0.30</td>
<td>0.26</td>
</tr>
<tr>
<td>0.70</td>
<td>0.44</td>
<td>0.39</td>
</tr>
<tr>
<td>0.75</td>
<td>0.52</td>
<td>0.47</td>
</tr>
<tr>
<td>0.80</td>
<td>0.58</td>
<td>0.53</td>
</tr>
</tbody>
</table>

Table F.1: Computed values of $t_{inj}$ for the two models as function of GPW.
F.2 Theoretical Gas Flow Calculations

The expression for \( f(m_{\text{diesel}}, m_{\text{gas}}) \) is shown below:

\[
\chi(m_{\text{gas}}) = \frac{c_{p,\text{quasi}}}{c_{v,\text{quasi}}} = \left( 1 - \frac{m_{\text{diesel}}}{m_{\text{diesel}} + m_{\text{gas}}} \right) c_{p,g} + \left( \frac{m_{\text{diesel}}}{m_{\text{diesel}} + m_{\text{gas}}} \right) c_{i} \quad (F.1)
\]

where \( m_{\text{diesel}} \) is kept constant for varying diesel fueling levels. Thus the expression for quasi-gas choked flow becomes,

\[
m_{\text{quasi}} = f(m_{\text{gas}}) = C_D A_1 \sqrt{\chi(m_{\text{gas}}) \rho_o \left( \frac{2}{\chi(m_{\text{gas}}) + 1} \right)^{\chi(m_{\text{gas}}) + 1}} \quad (F.2)
\]

Recall from Section 4.3.1 that,

\[
m_{\text{quasi}} = m_{\text{quasi}} T \quad (F.3)
\]

and

\[
m_{\text{quasi}} = m_{\text{diesel}} + m_{\text{gas}} \quad (F.4)
\]

Therefore,

\[
m_{\text{gas}} = f(m_{\text{gas}}) T - m_{\text{diesel}} \quad (F.5)
\]

By substituting Equation F.2 the whole expression becomes,

\[
m_{\text{gas}} = T \cdot C_D A_1 \sqrt{\chi(m_{\text{gas}}) \rho_o \left( \frac{2}{\chi(m_{\text{gas}}) + 1} \right)^{\chi(m_{\text{gas}}) + 1} - m_{\text{diesel}}} \quad (F.4)
\]
This expression is implicitly solved for any desired value of $m_{diesel}$ and $m_{gas}$ using any desired equation solver or mathematical software package. The computed values of $m_{gas}$ for $m_{diesel} = 10$ mg/inj and 20 mg/inj are shown in Table F.2 and denoted as ‘Quasi-10 mg diesel’ and ‘Quasi-20 mg diesel’.

The stratified expression becomes a straightforward linear plot based on Equation 4.26 and is shown below,

$$m_{gas} = m_{gas} \left( \tau - \frac{m_{diesel}}{m_{diesel}} \right)$$ (F.5)

This expression is linear and assumes the following values for $m_{diesel} = 10$ mg/inj and 20 mg/inj.

$$m_{strat,10} = 0.150 \text{ kg/s} (\tau - 3.84 \times 10^{-5} \text{ kg})$$ (F.6)

$$m_{strat,20} = 0.150 \text{ kg/s} (\tau - 4.01 \times 10^{-5} \text{ kg})$$ (F.7)

The theoretical gas mass flows of Equations F.6 and F.7 are shown in Table and denoted as ‘Strat-10 mg diesel’ and ‘Strat-20 mg diesel’.
<table>
<thead>
<tr>
<th>Time (ms)</th>
<th>Gas Only (mg/inj)</th>
<th>Quasi - 10mg diesel</th>
<th>Quasi - 20mg diesel</th>
<th>Stratified - 10 mg diesel</th>
<th>Stratified - 20 mg diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>0.2</td>
<td>30</td>
<td>23.7</td>
<td>19.3</td>
<td>27.465</td>
<td>24.93</td>
</tr>
<tr>
<td>0.4</td>
<td>60</td>
<td>53</td>
<td>47.3</td>
<td>57.465</td>
<td>54.93</td>
</tr>
<tr>
<td>0.6</td>
<td>90</td>
<td>82.6</td>
<td>76.5</td>
<td>87.465</td>
<td>84.93</td>
</tr>
<tr>
<td>0.8</td>
<td>120</td>
<td>112</td>
<td>106</td>
<td>117.465</td>
<td>114.93</td>
</tr>
<tr>
<td>1</td>
<td>150</td>
<td>142</td>
<td>136</td>
<td>147.465</td>
<td>144.93</td>
</tr>
<tr>
<td>1.2</td>
<td>180</td>
<td>172</td>
<td>165</td>
<td>177.465</td>
<td>174.93</td>
</tr>
<tr>
<td>1.4</td>
<td>210</td>
<td>202</td>
<td>195</td>
<td>207.465</td>
<td>204.93</td>
</tr>
<tr>
<td>1.6</td>
<td>240</td>
<td>232</td>
<td>225</td>
<td>237.465</td>
<td>234.93</td>
</tr>
<tr>
<td>1.8</td>
<td>270</td>
<td>261</td>
<td>254</td>
<td>267.465</td>
<td>264.93</td>
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<tr>
<td>2</td>
<td>300</td>
<td>291</td>
<td>284</td>
<td>297.465</td>
<td>294.93</td>
</tr>
</tbody>
</table>

Table F 2: A summary of the theoretical gas mass flow computations.
Appendix G - Raw Data Location

There were 3 main computers used for data acquisition during the course of this work. The flow characterization and imaging DAQ computers are located in RH 122 as per Figure A.1 of Appendix A and the simulations/modeling data was gathered on the computer of RH 122A. All important files were initially stored on these computers. In addition, an electronic appendix to this work is provided with a copy of all raw data files, and is stored in a single folder on the shared network drive: host “origin.mech.ubc.ca”. The directory the files are stored under is “home/netapp2/rogak/nbirger/”. Access to this directory and folder can be provided by Dr. Steven Rogak. The data is stored under its corresponding uses. Major elements of the data files are organized by application with an illustration of the storage structure shown in Figure G.1 and described below.

- All Labview control and data acquisition files are stored under the subdirectory “/thesis/Labview/”. This includes “HPDI Single Pulse.vi” and “HPDI Double Pulse.vi” as the controlling interface for single and double gas needle actuations per cycle. The file “IVC Data Acquisition.vi” provides real-time pressure and flow readings as well as data recording.

- Flow Characterization data is stored under the subdirectory “/thesis/Flow data/”. The files are grouped by date/injector/supply pressure. For example the folder “/Feb. 4, 2009 – B 28 MPa/” contains the flow characterization data measured on February 4, 2009 for Co-injector B with 28 MPa supply pressure. Within this
folder data is organized by diesel supply (i.e. “/DPW = 1.0 ms - XI” or “Bias = 1.0 MPa”), where X represents which repetition of the test point was run. Within each of these folder there is a raw Excel file for each GPW tested. A typical name would be “GPW = 0.60 ms.xls.” The main folder for the period of testing (in this case “/Feb. 4, 2009 – B 28 MPa/”) contains a summary of the data – in this case named “Gas Flow Summary.xls” and “Diesel Flow Summary.xls”. The main data template to process all raw LabView data is provided in the initial subdirectory “/thesis/Flow data/” under the name “Data Template.xls”.

- Raw video clips of the imaging performed is contained under the subdirectory “/thesis/Imaging/Raw images/”. A Vision Research Cine Viewer application package is needed to view the clips in their raw form. This is downloadable from the following link:


All important MATLAB code for image processing (as describe in Appendix H) is stored in the subdirectory “/thesis/Imaging/Matlab/”. The subdirectory “/thesis/Imaging/Mech delay/” contains the mechanical delay value. If required, the raw video clips used to compute the values are located with the Raw Images.
• All important flow and injector modeling files are organized into two subdirectories. Files related to the quasi-gas and stratified modeling of Section 4.3.1 can be found under “/thesis/Modeling/Quasi-gas/”. The entirety of the quasi-gas and stratified model calculations are found in the spreadsheet “Quasi-gas Model Calculations.xls”

All files related to AMESim simulations can be found in the folder “/thesis/Modeling/Amesim/”. The simulation package AMESim Rev 8A or newer is needed to view the Co-injector models and submodels created. The associated data from the modeling is placed in the subdirectory “/thesis/Modeling/Amesim/Data/”.

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Figure G.1: Data storage structure for the work performed herein.
Appendix H - Selected MATLAB Imaging Code

The initial attempt at coding to isolate Co-injector spray features is shown using MATLAB. The code takes an input injection TIFF file as well as a background image and processes them according to the coding found in Section H.1. Future extensions should require the user to implement a systematic method of taking an entire video clip and analyze the entire file to detect parameters such as jet penetration and velocity. Section H.2 shows the code developed herein applied to a single image and the resulting spray isolation.

H.1. Image Processing Code

FUNCTION: “boundary_detect.m”

function bound = boundary_detect (background, cutoff_threshold);

    OFFSET = 0.45;

    % check bounds on threshold

    thres = cutoff_threshold;
    if ((thres<0)|(thres>1))
        min_threshold = min(min(background));
        max_threshold = max(max(background));
        thres = OFFSET*max_threshold + (1-OFFSET)*min_threshold;
    end;

    bound = (background > thres);
FUNCTION: “divide_image.m”

function divided_image = divide_image (target_image, bg_image)
    divided_image = target_image ./ bg_image;

FUNCTION: “invert_image.m”

function inverted_image = invert_image (original_image);
    inverted_image = abs(original_image - 1);

FUNCTION: “isolate_feature.m”

function feature = isolate_feature (divided_image, image_boundary)
    feature = ...
    inverted_image = invert_image(invert_image(divided_image).*image_boundary);

FUNCTION: “load_tiff.m”

function dbl_image_data = load_tiff ...
    (tiff_filename,median_filter_window,bg_flag)
    original_image = imread (tiff_filename):
    % perform median filter to take out salt-and-pepper noise
    % (if window size is 1 or less, this will be ignored)
    % Also, if window size is even, we will force it to be
    % odd to improve computational efficiency.
win = median_filter_window;
if (mod(win,2)==1)
    win = win + 1;
end;

if (win > 1)
    original_image = medfilt2 (original_image, [win win]);
end;
%
% make sure the minimum intensity is one to avoid
% division by zero error

zero_intensity_map = zeros (size(original_image));

if (bg_flag)
    zero_intensity_map = (original_image == 0);
end;
%
% check color depth and cast the zero intensity map
% accordingly
if (isa(original_image,'uint8'))
    uint_image_data = original_image + uint8(zero_intensity_map);
else
    uint_image_data = original_image + uint16(zero_intensity_map);
end;
%
% normalize image (to double precision, range 0 to 1)

dbl_image_data = im2double (uint_image_data);

FUNCTION: “process_image.m”

function [filtered_image, divided_image, isolated_image] = ...
    process_image (target_filename, background_filename);
%
% hard-coded values are known to work best

MEDIAN_FILTER_WINDOW = 5;
BOUNDARY_CUTOFF_THRESHOLD = 0.45;
%
% load and filter images

filtered_background = ...
    load_tiff (background_filename, MEDIAN_FILTER_WINDOW, 1);
filtered_image = ...
    load_tiff (target_filename, MEDIAN_FILTER_WINDOW, 0);
% perform image division

divided_image = divide_image (filtered_image, filtered_background);

% identify image boundaries

image_boundary = ...
boundary_detect (filtered_background, BOUNDARY_CUTOFF_THRESHOLD);

% isolate image feature

isolated_image = isolate_feature (divided_image, image_boundary);

H.2 Example Application

The coding in Section H.1 is now applied to a background and single image of a Co-injector B injection. The image shown uses a supply pressure of 21 MPa with DPW = 0 ms, GPW = 0.80 ms, and atmospheric chamber pressure. In this instance the Co-injector only fired a single injection, without reaching steady state, and is used as an example of potential clarity.
Figure H.1: The code first applies the function ‘load_tiff.m’ to an input background image with no injection. ‘load_tiff.m’ filters some of the inherent image noise.

Figure H.2: The function ‘load_tiff.m” is again applied to a raw TIFF spray image for processing.
Figure H.3: The function ‘divide_image.m’ divides the raw injection image in Figure H.2 by the background in Figure H.1.

Figure H.4: The function ‘boundary_detect.m’ is then applied to the divided image to identify image boundaries.
Figure H.5: Lastly, the function “isolate_feature.m” now removes the background from Figure H.4 leaving only the spray feature visible. It should be noted the whole process is automated by integrating each function into ‘process_image.m’