PILOT IGNITED HIGH PRESSURE DIRECT INJECTION OF NATURAL GAS FUELING OF DIESEL ENGINES

by

SILVIU DUMITRESCU

B.Sc., Polytechnic Institute Bucharest, 1992

A THESIS SUBMITTED IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE DEGREE OF

MASTER OF APPLIED SCIENCE

in

THE FACULTY OF GRADUATE STUDIES

(Department of Mechanical Engineering)

We accept this thesis as conforming to the required standard

THE UNIVERSITY OF BRITISH COLUMBIA

August 1999

© Silviu Dumitrescu, 1999
In presenting this thesis in partial fulfilment of the requirements for an advanced degree at the University of British Columbia, I agree that the Library shall make it freely available for reference and study. I further agree that permission for extensive copying of this thesis for scholarly purposes may be granted by the head of my department or by his or her representatives. It is understood that copying or publication of this thesis for financial gain shall not be allowed without my written permission.

Department of Mechanical Engineering

The University of British Columbia
Vancouver, Canada

Date Sep 24, 1999
Measurements of performance and emissions of a Diesel engine fueled with natural gas have been made with high-pressure direct-injection (HPDI). Natural gas is injected late in the compression cycle preceded by an injection of diesel pilot. Tests were performed on a DDC 1-71 Diesel engine with electronic controls. Influence of several physical parameters has been investigated in light of the results obtained by other researchers through 3-dimensional numerical simulation of the combustion process using a modified KIVA code. These were: natural gas injection pressure, absolute beginning of injection (BOI) and relative timing between injection of pilot diesel and natural gas (RBOI).

It was found that, on the range of 100 to 160 bar, combustion rate and NO\textsubscript{x} emissions increase with gas injection pressure. Best thermal efficiency results were obtained for a gas pressure of 130 bar. With HPDI fueling of a diesel engine, injection timing delay has proven to be an effective way of reducing NO\textsubscript{x} emissions. By appropriately adjusting beginning of injection, NO\textsubscript{x} reduction of up to 60% over the diesel baseline could be obtained, while preserving conventional diesel efficiency. Over the BOI and load range of study, a longer relative injection delay appears to give lower NO\textsubscript{x} emissions and improved thermal efficiency.

KIVA 2 simulations indicate that the interlace angle (defined as the angle between the diesel and gas plumes as viewed in the direction of the cylinder axis), which changes during engine operation, has a significant effect on combustion rate and NO\textsubscript{x} emissions. Experimentally it was observed that with an equal number (6) of nozzle holes for both natural gas and diesel pilot there was instability in engine operation at low load and wide scatter in emission measurements. Guided by the simulation results it was found experimentally that data reproducibility and engine operating stability could both be much improved by choosing a different number of jets for injection of natural gas. An arrangement of 7 natural gas and 6 diesel holes has proven to give better results of engine stability, thermal efficiency and exhaust emissions. KIVA 2 simulations indicate attachment of the gas jets to the fire deck for inclination angles less than about 20 degrees. The reduced contact area between the flame and oxygen lead to somewhat lower combustion rate and NO\textsubscript{x} emissions.
# TABLE OF CONTENTS

## ABSTRACT

## LIST OF TABLES

## LIST OF FIGURES

## ACKNOWLEDGEMENTS

### 1. INTRODUCTION

1.1. Diesel Engine Emissions  
1.2. Natural Gas Fueling Methods for Diesel Engines  
1.3. High Pressure Direct Injection of Natural Gas  
1.4. Objectives of the Research  
1.5. Methodology  

### 2. PREVIOUS RESEARCH

2.1. Fueling of Diesel Engines with High Pressure Direct Injection of Natural Gas  
2.2. Thermal Efficiency  
2.3. Emissions  
2.4. Gas Injection Pressure and Timing  
2.5. Summary  

### 3. EXPERIMENTAL METHOD

3.1. Apparatus  
3.2. Research Engine  
3.3. Instrumentation  
3.4. Data Acquisition  
3.5. Performance Parameters  
3.6. Test Matrices  

---

---
4. ENGINE OPERATIONAL INSTABILITY WITH PILOT INJECTION
   4.1. Evidences of Variability
   4.2. Investigation
   4.3. Reduction of Variability
   4.4. Summary

5. GAS INJECTION PRESSURE
   5.1. Introduction
   5.2. Thermal Efficiency
   5.3. Emissions
   5.4. Summary

6. ABSOLUTE INJECTION TIMING
   6.1. Introduction
   6.2. Thermal Efficiency and NOx Emissions
   6.3. CO and Unburned Hydrocarbon Emissions
   6.4. Summary

7. RELATIVE INJECTION TIMING
   7.1. Introduction
   7.2. Short Relative Injection Delay
   7.3. Load Dependence
   7.4. BOI Dependence
   7.5. Summary

8. CONCLUSIONS
   8.1. Conclusions
   8.2. Recommendations for Future Work

REFERENCES
APPENDIX 1 - Emission Standards for Urban Bus and Heavy Duty Compression Ignition Highway Engines
APPENDIX 2 - Evaluation of the Cylinder Pressure Signal
APPENDIX 3 - Properties of the Fuels
APPENDIX 4 - Calibration Curves
APPENDIX 5 - Repeatability of Typical Results
APPENDIX 6 - RBOI Study; BOI Dependence
APPENDIX 7 - Computed Interlace Angle Effects on Combustion Rate and NO\textsubscript{x} Emissions
APPENDIX 8 - Inclination Angle
APPENDIX 9 - Computed Inclination Angle Effects on Combustion Rate and NO\textsubscript{x} Emissions
LIST OF TABLES

Table 3.2.2: Detroit Diesel 1-71 engine specifications

Table 3.4.1: Measured engine parameters

Table 3.6.1: Experimental test matrix

Table 4.2.3: Numerical modeling engine test conditions

Table 4.3.3: Injector tip characteristics

Table 6.1.1: Experimental test matrix; 130 bar gas pressure, 3 tests for each load

Table 7.1.2: RBOI spring combination characteristics

Table 7.1.3: RBOI study test matrix; 1200 rpm, 130 bar gas pressure, 3 data sets for each test condition

Table A1.1: Environmental Protection Agency Emission Standard for Urban Bus and Heavy Duty Compression Ignition Highway Engines

Table A3.1: Natural Gas Composition

Table A3.2: Fuel Properties
LIST OF FIGURES

Figure 1.1.1: NOx and particulate matter emissions from production diesel engines, relative to existing and upcoming emission standards 2

Figure 1.3.1: Schematic of a two-stroke diesel engine using high pressure direct injection of natural gas with pilot ignition 8

Figure 1.3.2: Principle of operation of the high pressure direct injection technology (schematic) 9

Figure 1.3.3: Schematic of the interlace angle between a diesel and natural gas jet 11

Figure 1.3.4: Schematic of the jet inclination angle 12

Figure 3.1.1: Schematic of the experimental apparatus 29

Figure 3.2.1: Schematic of the two-stroke diesel engine using high pressure direct injection of natural gas with pilot ignition 31

Figure 3.3.1: Schematic of emissions heated instruments 34

Figure 3.3.2: Schematic of emissions cooled instruments 34

Figure 4.1.1: Variations in engine speed of the HPDI injector with 6 diesel and 6 natural gas jets 49

Figure 4.1.2: Variations in engine speed associated with the use of an HPDI injector with 6 diesel and 6 natural gas jets; target conditions: 3 bar BMEP and 1200 rpm 50

Figure 4.1.3: NOx and CH4 variability associated to unstable engine operation, when using an HPDI injector with 6 diesel and 6 natural gas jets 51

Figure 4.1.4: Variability in NOx emissions from HPDI using an injector with 6 diesel and 6 natural gas jets and conventional diesel fueling 52

Figure 4.2.1: Schematic of the interlace angle between a diesel and natural gas jet 54
**Figure 4.2.2:** Schematic of three interlace angle arrangements for a "6-6" injector

**Figure 4.2.4:** Computed cylinder pressure variation with crank angle, for interlace angles of 0, 15 and 30; 2.17 bar BMEP and 1200 rpm

**Figure 4.2.5:** Computed NO$_x$ emission variation with crank angle, for interlace angles of 0, 15 and 30; 2.17 bar BMEP and 1200 rpm

**Figure 4.2.6:** Natural gas ignition mechanism with interlace angles of 15 and 30 deg.

**Figure 4.2.7:** Combustion chamber temperature distribution (deg. K) for 0, 15 and 30 deg. interlace angles on a cut plane 4 mm below the fire deck

**Figure 4.3.1:** Variation of the average interlace angle between jets with gas needle angle of rotation. Number of diesel holes is equal to 6.

**Figure 4.3.3:** Schematic diagram of the "6-7" and "6-8" tip geometry configurations

**Figure 4.3.4:** NO$_x$ emissions from HPDI fueling using injectors with 6, 7 and 8 gas holes; 6 pilot diesel jets

**Figure 4.3.5:** THC emissions from HPDI fueling using injectors with 6, 7 and 8 gas holes; 6 pilot diesel jets

**Figure 4.3.6:** CO emissions from HPDI fueling using injectors with 6, 7 and 8 gas holes; 6 pilot diesel jets

**Figure 5.2.1:** Combustion chamber pressure development for HPDI with 3 gas injection pressures; 4 bar BMEP

**Figure 5.2.2:** Thermal efficiency variation with load for HPDI with 3 injection pressures and diesel baseline

**Figure 5.2.3:** Thermal efficiency variation with load for HPDI with 3 injection pressures and diesel baseline; upper load range

**Figure 5.3.1:** HPDI's CO emission variation with load for 3 injection pressures relative to the diesel baseline
Figure 5.3.2: HPDI's NO\textsubscript{x} emission variation with load for 3 injection pressures relative to the diesel baseline

Figure 5.3.3: HPDI's THC emission variation with load for 3 injection pressures relative to the diesel baseline

Figure 5.3.4: HPDI's CH\textsubscript{4} emission variation with load for 3 injection pressures relative to the diesel baseline

Figure 5.3.5: HPDI's diesel ratio variation with load for 100, 130 and 160 bar gas injection pressure

Figure 6.2.1: HPDI and diesel baseline's thermal efficiency variation with crank angle BTDC for 2 loads conditions; 130 bar gas injection pressure

Figure 6.2.2: HPDI and diesel baseline's NO\textsubscript{x} emission variation with crank angle BTDC for 2 load conditions; 130 bar gas injection pressure

Figure 6.2.3: NO\textsubscript{x} vs BSFC trade-off for HPDI and conventional diesel fueling; 130 bar gas injection pressure

Figure 6.3.1: HPDI and diesel baseline's THC emission variation with crank angle BTDC for 2 load conditions; 130 bar gas injection pressure

Figure 6.3.2: HPDI and diesel baseline's CH\textsubscript{4} emission variation with crank angle BTDC for 2 load conditions; 130 bar gas injection pressure

Figure 6.3.3: HPDI combustion pressure trace with no injection advance; 1 bar BMEP, 1200 rpm, gas pressure 130 bar

Figure 6.3.4: HPDI and diesel baseline's CO emission variation with crank angle BTDC for 2 load conditions; 130 bar gas injection pressure

Figure 7.1.1: Variation of hydraulic pressure inside HPDI injector passages before gas needle opening

Figure 7.3.1: Combustion pressure development for an HPDI fueled engine running with short and long RBOI
Figure 7.3.2: HPDI's thermal efficiency variation with load for long and short RBOI, relative to baseline diesel

Figure 7.3.3: HPDI's NOx emission variation with load for long and short RBOI, relative to baseline diesel

Figure 7.3.4: HPDI's THC emission variation with load for long and short RBOI, relative to baseline diesel

Figure 7.3.5: HPDI's CH₄ emission variation with load for long and short RBOI, relative to baseline diesel

Figure 7.4.1: HPDI's thermal efficiency variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP

Figure 7.4.2: HPDI's NOx emission variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP

Figure 7.4.3: HPDI's THC emission variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP

Figure 7.4.4: HPDI's CH₄ emission variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP

Figure 7.4.5: HPDI's combustion pressure development for long and short RBOI, and zero injection delay; 1 bar BMEP

Figure 7.4.6: HPDI's CO emission variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP

Figure 7.4.7: HPDI's BSFC vs NOx trade-off with long and short RBOI, relative to diesel baseline; 1 bar BMEP

Figure A2.1: Distortion of pressure signal with amplification factor on the P-V diagram

Figure A2.2: Effect of amplification factor on pressure signal during inlet ports opening
Figure A2.3: Pressure curves during exhaust valve opening obtained with different time constants and amplification factors with a Kistler charge amplifier model 5004

Figure A2.4: P-V diagram curves with different time constants and amplification factors obtained with the Kistler charge amplifier model 5004 (serial # 200295)

Figure A3.3: Lower heating value variation during the period January 1 to December 31, 1998 (BC Gas)

Figure A4.1: Cylinder pressure calibration curve - full scale

Figure A4.1bis: Cylinder pressure calibration curve - low load scale

Figure A4.2: Diesel mass flow calibration curve

Figure A4.3: CNG mass flow calibration curve

Figure A4.4: Air flow calibration curve

Figure A4.5: Airbox pressure calibration curve

Figure A4.6: Airbox temperature calibration curve

Figure A4.7: CNG pressure calibration curve

Figure A4.8: Ambient temperature calibration curve

Figure A4.9: Torque calibration curve

Figure A5.1: Typical reproducibility of thermal efficiency measurements

Figure A5.2: Typical reproducibility of NOx emission measurements

Figure A5.3: Typical reproducibility of CO emission measurements

Figure A5.4: Typical reproducibility of CH4 emission measurements

Figure A5.5: Typical reproducibility of NOx emission measurements
Figure A6.1: HPDI's thermal efficiency variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP

Figure A6.2: HPDI's NO\textsubscript{x} emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP

Figure A6.3: HPDI's THC emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP

Figure A6.4: HPDI's CH\textsubscript{4} emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP

Figure A6.5: HPDI's CO emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP

Figure A6.6: HPDI's BSFC vs NO\textsubscript{x} curves with long and short RBOI, relative to diesel baseline; 3 bar BMEP

Figure A8.1.1: Schematic of the jet inclination angle

Figure A8.2.1: Schematic of the CFD tip geometry arrangements

Figure A8.2.2: CH\textsubscript{4} mass fraction distribution in the combustion chamber - 15 degrees inclination angle

Figure A8.2.3: CH\textsubscript{4} mass fraction distribution in the combustion chamber - 20 degrees inclination angle

Figure A8.2.4: Computed cylinder pressure variation with crank angle for jet inclination angles of 10, 15 and 20

Figure A8.2.5: Computed NO\textsubscript{x} emission variation with crank angle for jet inclination angles of 10, 15 and 20

Figure A8.3.1: Thermal efficiency of two HPDI injectors having the gas jets inclined at 10 and 15 deg. from fire-deck
Figure A8.3.2: THC emissions from two HPDI injectors having the gas jets inclined at 10 and 15 deg. from fire-deck

Figure A8.3.3: CH$_4$ emissions from two HPDI injectors having the gas jets inclined at 10 and 15 deg. from fire-deck
ACKNOWLEDGEMENTS

The development of this thesis was possible with the generous support of a large collective of researchers.

First, I would like to give my deepest gratitude to Dr. Philip G. Hill, my thesis supervisor, for the invaluable lessons he taught me during the three years of our collaboration. I feel very honored and privileged to have worked with him.

I wish to thank Dr. Guowei Li, who performed all computational fluid dynamic simulations used for guidance of the optimization process. His numerical modeling results complemented the experimental program by helping give theoretical understanding of observed phenomena.

Special thanks are due to Dr. Patric Ouellette for the guidance and support given throughout the experimental research.

I would also like to thank Radu Oprea, research engineer at UBC, for his help with the apparatus and instrumentation.

The experimental program of this thesis was facilitated by the support of Westport Research Inc. Thanks are due to Robin Poworoznik and Jeff Kohne, who assembled and tested the injectors tested, and to the design team for performing the required geometry changes.

And last but not least I would like to thank all graduate students and engineers whom I never met but contributed with their effort to the development of the HPDI technology.
1. INTRODUCTION

The focus of this thesis was to determine experimentally the importance of selected design parameters on the efficiency and emissions of a diesel engine fueled with pilot ignited high pressure direct injection of natural gas. The work is part of a larger effort which included numerical modeling, some of which is reported in this thesis. This is a first step in the process of optimization of a new fueling technology.

1.1. Diesel Engine Emissions

Diesel engine emissions are now seen as a serious health problem, with their small particulate emissions linked to cancer and serious respiratory problems. In diesels, a stream of liquid fuel is injected at high pressure into the cylinder, where it disintegrates rapidly into fine droplets. In conditions of high air temperature at the end of compression, the droplets vaporize, mix with air and auto-ignite. Following a premixed combustion phase, the combustion is controlled by mixing with air. Although conventional diesel engines run with a very lean overall fuel-air mixture, there can be zones in the cylinder where the mixture remains very rich during combustion. Due to oxygen scarcity, these regions will have incomplete combustion, which leads to production of carbon monoxide and particulate matter (soot). Another characteristic of diesel combustion is the production of nitrogen oxides (NO\textsubscript{x}). Formation of NO\textsubscript{x} occurs in conditions of high temperature in the post flame region. It is a very temperature sensitive process, in the sense that a small difference in combustion temperature can produce a larger variation in NO\textsubscript{x}. Since diesel fuel has high adiabatic flame temperature, it is a significant producer of NO\textsubscript{x}.

In November 1996 the US Environmental Protection Agency announced its intention to begin regulating diesel engine emissions of very small (2.5 micron) particulate matter (PM\textsubscript{2.5}) in addition to the current regulations of 10 micron particulate matter (PM\textsubscript{10}). Diesel engines are also major producers of nitrogen oxides (NO\textsubscript{x}), gases which
participate in the formation of smog and ground level ozone. In the summer of 1998, California declared diesel exhaust to be a toxic substance.

Although engine manufacturers have managed to reduce emissions substantially in recent years, the conventional diesel engine may be unable to meet upcoming standards without compromising efficiency. Generally, with conventional diesels, when one takes measures to reduce NO$_x$, the result is an increase in fuel consumption and particulate matter and vice-versa, following a typical curve like the one in Fig 1.1.1. On the x axis, NO$_x$ emissions are plotted in grams per brake horse power hour (g/bhp-hr$^1$), and the y axis displays particulate matter in the same units. Figure 1.1.1 shows the level of emissions from today’s heavy duty diesel engines with respect to the upcoming standards, which are presented in detail in Appendix 1.

\[
\begin{array}{c|c|c|c}
\text{NO}_x \text{ (g/bhp-hr)} & 2002 & 1998 & 1996 \\
\hline
\text{Particulates (g/bhp-hr)} & 0.14 & 0.12 & 0.1 \\
0 & 0.08 & 0.06 & 0.04 & 0.02 \\
\end{array}
\]

Figure 1.1.1: NO$_x$ and particulate matter emissions from production diesel engines, relative to existing and upcoming emission standards

\textsuperscript{1} g/bhp-hr are the units chosen by the EPA for emission regulations and are maintained here instead of converting to SI units
In the past decade, significant research efforts have been dedicated to finding alternative ways of fueling that would enable heavy-duty engines to run efficiently with lower levels of exhaust emissions. One of the fuels most suitable for immediate conversion is natural gas, because of its lower cost than diesel, availability and high energy content per unit mass. But most of all, natural gas has great potential for particulate and NOX emissions reduction, due to its physical and chemical properties.

Because NOX formation is a very temperature sensitive process, methane's lower adiabatic flame temperature can lead to reduced NOX generation, for the same combustion rate. Chemistry balance shows that methane combustion also produces less CO2 emissions than conventional diesel fuel. Conversion of diesel engines to natural gas can thus help reduce greenhouse gas emissions.

1.2. Natural Gas Fueling Methods for Diesel Engines

Beck et al. [1] have reviewed the different concepts of natural gas fueling and emphasized advantages and disadvantages associated with each method. They have classified the different technologies in five major groups:
1. Single fuel, spark ignition systems
2. Dual fuel, diesel pilot, mechanical pump and governor
3. Dual fuel, diesel pilot, mechanical pump, electronic governor
4. Dual fuel, electro-hydraulic diesel pilot and fully electronic control with variable injection timing
5. Dual fuel, diesel pilot, high-pressure direct injected gas, full electronic control with variable injection timing

Each one of these systems has several variations according to the nature of the mixture, the way it is formed, fuel admission in the cylinder and mode of ignition.
While Beck's classification emphasizes somewhat the methods of ignition and engine control, these systems can also be described according to fundamental differences in the combustion process:

- Single fuel (natural gas), homogeneous mixture, spark ignition
- Dual fuel, homogeneous mixture of natural gas ignited by injection of a diesel pilot
- Dual fuel, high pressure direct injection of natural gas ignited by injection of a diesel pilot

Spark ignition (SI) of natural gas has been implemented in recent years on transit bus engines. It is a simple well known fueling system, and has most of the advantages and disadvantages of a gasoline Otto cycle spark-ignition engine. Natural gas SI engines produce low soot emissions throughout the load range, because of the combustion characteristics of their premixed charge. However, like the gasoline engine, they are still subject to knock (un-controlled combustion of the end gases due to high temperature and pressure). To avoid this undesired phenomenon, the engines converted to spark ignition require a reduction in compression ratio, which leads to lower thermal efficiency. In spark-ignition engines, load control is done by throttling the air intake. This method results in losses associated with the pumping process, and has a reduced torque capability at low speed. A transit bus engine is subject to frequent accelerations; therefore high torque at low speed offers better driveability. In addition, conversion of a diesel engine to spark ignition requires modifications of the cylinder head, and addition of a new system for accurate control of mixture composition and ignition timing.

The dual fuel technology with homogeneous mixture uses low pressure natural gas, which is mixed with air and ignited by injection of a diesel pilot, at the end of the compression stroke. Song and Hill [2] explained the fundamental differences between dual fueling and conventional diesel. In dual fuel operation the gaseous fuel is mixed with air at lean gas-air ratio, and the mixture is then compressed. Near the end of the compression stroke, pilot diesel fuel is injected at high pressure and auto-ignites, initiating combustion of the gas-air mixture. Once ignited, combustion in dual fuel
operation is controlled by flame propagation, as opposed to the diesel combustion, which is controlled by mixing. This characteristic makes low pressure dual fueling act more like premixed combustion rather than diesel. The pilot combustion replaces the spark, to produce a reliable source of ignition for the mixtures of natural gas and air. Since this fueling method also uses homogeneous charges, it is still subject to knock, like its spark ignited counterpart. Thus, compression ratios are higher then for spark ignited engines but lower than for diesel. Generally, to obtain significant emission reduction there is a penalty in efficiency.

Several variations of low pressure dual fueling exist, and one of them is the so-called "stratified charge". In this method, zones of premixed fuel-air gas with different equivalence ratios are formed within the combustion chamber. The rich mixtures are placed around the ignition source (i.e. the pilot combustion or spark plug), and the lean ones at the periphery. Using stratified charge, the lean mixture pockets (sometimes made of just air) have less tendency to knock, and this enables the use of a higher compression ratio. Although combustion is still controlled by flame propagation (because of the mixture homogeneous character), efficiency is improved compared to that of engines using the Otto cycle.

The dual fuel, high pressure direct injection (HPDI) technology consists of high pressure injection of natural gas directly into the cylinder, with pre-injection of a small diesel pilot. Gas injection occurs late in the compression stroke, just as it would in a conventional diesel engine. Pre-injection of the pilot diesel is needed for gas ignition assist, because the temperature conditions of the end of compression (900 to 1000 K) are below the 1200 K auto-ignition temperature for high pressure natural gas. Another ignition method has also been used to provide ignition. It consists of continuous use of glow plugs that operate at 1200 K, which are mounted in the cylinder head and protrude into the combustion chamber. This method has the advantage that only one fuel is required, which simplifies injector design.

2 In this thesis dual fuel refers to simultaneous use of two fuels during combustion
The HPDI technique is unique among alternative fuel conversion strategies in that it retains the basic diesel cycle, which ensures that high efficiency and performance are preserved, while reducing emissions of NO\textsubscript{x} and particulate matter. Beck et al. [1] affirm that "from an overall performance point of view, used in large marine diesel engines, HPDI is an elegant solution". Because of its heterogeneous fuel mixture and late injection, HPDI's diesel-type combustion does not have detonation, and thus high compression ratios (and hence efficiencies) can be preserved. Mechanical losses associated with pumping of a premixed intake do not exist, because the engine operates un-throttled. As in conventional diesels, the intake charge is clean air, and overall equivalence ratios are well under unity (lean burn). HPDI's diffusive type burning does not require mixture ratio control, and is insensitive to composition of the gaseous fuel. The high pressure direct injection technology is applicable to any diesel engine, including two-stroke and four-stroke applications.

In his revision summary of the methods for natural gas fueling of diesel engines, Beck et al. [1] outlined advantages of the pilot ignited, high pressure direct injection:

- Diesel cycle efficiency (because of high compression ratio and unthrottled operation)
- No detonation limit if gas injection is simultaneous with liquid fuel pilot injection
- Lean burn, requires no mixture ratio control
- Negligible unburned fuel in the exhaust

Although it requires the presence of two fuels, the pilot injected high-pressure direct injected gas engine was found to be very suitable for large engine applications (marine, power generation, etc.).

With the aim of developing a practical natural gas fueled locomotive engine, Meyers et al. [3] designed and tested six natural gas-fueled combustion systems for an EMD 710-type engine. During the experimental program they evaluated the different fueling systems in terms of NO\textsubscript{x} and CO emissions, thermal efficiency, knock tolerance and other practical considerations. The six combustion systems can be classified as either spark-ignited with homogeneous gas-air mixtures, dual-fuel with homogeneous gas-air mixtures, or dual-fuel with heterogeneous gas-air mixtures. The fueling system with
heterogeneous mixture, called LaCHIP, used a diesel-like diffusive combustion process. High pressure natural gas was directly injected in the combustion chamber late in the compression cycle, and ignited by pre-injection of a small amount of diesel pilot.

Based on tests results and other analyses, Meyers et al. consider the LaCHIP combustion system, using a diesel pilot ignited late-cycle injection of natural gas, as the one to provide the most practical combustion system for a natural gas fueled EMD 710-powered locomotive. LaCHIP technology was able to produce a 60 % reduction in NO\textsubscript{x} emissions, while reducing CO to acceptable levels and obtaining efficiencies similar to conventional diesel.

1.3. High Pressure Direct Injection of Natural Gas with Diesel Pilot Ignition

This thesis is concerned with high pressure direct injection of natural gas fueling of a two-stroke heavy duty engine, using pre-injection of pilot diesel.

A prototype injector was developed\(^3\) which allows sequential injection of diesel and natural gas, using the same injector unit and different passages for the two fuels. Both natural gas and diesel are injected in multiple jets, using a symmetrical pattern as shown in Fig 1.3.1. The tip geometry consists of 6 gas jets injected a small distance above the 6 diesel pilot jets. Presence of several diesel plumes creates multiple ignition sites for the natural gas jets, which can auto-ignite more readily. Rapid ignition would yield a shorter premixed burning period, and hence is expected to produce lower NO\textsubscript{x} and noise. In addition to igniting the gas, diesel fuel also serves for actuation, cooling and lubrication of the injector.

\(^3\) 1990-1999 UBC/Westport Research joint project
This design has the advantage that it can be implemented with minor modifications made to the engine. An injector was designed which enables the injection of both natural gas and diesel pilot within the geometric constraints of the standard diesel injector. Operation of one version of the HPDI injector can be understood with the aid of the schematic representation in Fig. 1.3.2, which reveals both passages for diesel and natural gas.

![Schematic of a two-stroke diesel engine using high pressure direct injection of natural gas with pilot ignition](image)

**Figure 1.3.1:** Schematic of a two-stroke diesel engine using high pressure direct injection of natural gas with pilot ignition
Actuation of the injector is ensured by a hydraulic system with electronic control of the timing and duration of injection, similar to that used in standard Detroit Diesel injectors. The hydraulic pressure is generated by the downward travel of the cam-driven plunger.

**Figure 1.3.2:** Principle of operation of the high pressure direct injection (US application patent # 09/075,060) technology (schematic)
During the initial period of plunger movement, the solenoid valve remains open, to enable free flow without pressurization of diesel. At the electronic command of beginning of injection (BOI), the solenoid valve energizes to close the spool valve which blocks the fuel outflow. During the pressure build-up in the injector passages, the pilot plunger travels downward to close a relief hole. Further movement of the plunger leads to a rapid pressure build-up in the pilot needle area. When the force generated by the pressure acting on the differential area of the diesel needle overcomes the pilot spring pre-load, the needle starts moving upwards. The diesel injection continues until the pilot plunger reaches the bottom surface.

This technique limits the amount of pilot that gets injected to a small constant quantity, regardless of the total duration of injection. After the end of the diesel injection, the pressure inside the hydraulic passages, common for diesel and gas needle actuation, continues to rise until it reaches the gas needle opening pressure. When the force generated by the pressure acting on the differential area of the gas needle overcomes the gas needle spring pre-load, the needle moves upwards to cause flow through the CNG holes. Injection of natural gas takes place until the solenoid valve re-opens, relieving the hydraulic pressure.

A design characteristic of the HPDI injector is that both natural gas and diesel pilot needles are free to rotate during operation. This feature will prove to have a great importance in the combustion development, as will be shown in chapters 4 and 8.

The HPDI technology design variables include:

- **Gas injection pressure** - represents the pressure at which natural gas is injected in the cylinder. This pressure is constant during injection and cannot be changed during the engine operation.
- **Absolute injection timing** - is the beginning of injection (BOI), as commanded by the engine controller. It is given by an electronic signal, and varies according to the load and speed of the engine.
- **Relative injection timing** - represents the time delay between the diesel pilot injection and that of the natural gas. The value of this parameter is determined by the
strength of the two springs pre-loading the diesel pilot and gas needles. It cannot be changed during operation and varies with engine speed.

- **Jet interlace angle** - is defined as the angle between a diesel and the closest adjacent gas jet, as viewed in the direction of the cylinder axis (Fig. 1.3.3). For an equal number of pilot diesel and gas jets, this parameter has the same value for all pairs of jets. With rotation of the needles, the interlace angle changes during engine operation.

![Figure 1.3.3: Schematic of the interlace angle between a diesel and natural gas jet](image)

- **Jet inclination angle** - is the angle formed in a vertical plane (going through the cylinder axis) between the natural gas or diesel jets and the combustion chamber upper wall (or fire-deck), as shown in Fig. 1.3.4.

- **Pilot diesel quantity** - represents the amount of diesel fuel injected per cycle. In the current design the pilot amount is constant, and independent of engine load. Its percentage (in energy content) from the total fuel injected varies from 50 % at no load to approximately 15 % at the maximum experimental load.
1.4. Objectives of the Research

To determine the emission reduction potential of natural gas fueling, the HPDI combustion must be optimized. That is, determine a combination of geometry-design and operating parameters, that gives low NO\textsubscript{x}, CO and hydrocarbon emissions while maintaining the thermal efficiency characteristic for conventional diesel engines.

The main objective of this thesis was to determine, through numerical modeling and experiments, the importance of selected design parameters on the efficiency and emissions of a diesel engine fueled with pilot ignited high pressure direct injection of natural gas. This is a first step in optimization of the HPDI fueling technology.

The objectives of the research are:

- Establish the effect of gas needle rotation on engine performance.
- Determine if an optimum natural gas injection pressure exist, and whether or not this changes with load.
- Explore the potential for NO\textsubscript{x} reduction through (absolute) injection timing delay, while preserving efficient engine operation.
• Study the influence on efficiency and emissions of the relative injection delay between diesel pilot and natural gas.
• Determine if interlace angle should be controlled.
• Evaluate implications on efficiency and emissions of natural gas jet attachment or detachment to/from fire-deck, and if one optimum inclination angle exists.

Another objective of this study was to test the results of an inexact but detailed numerical simulation of injection, ignition and combustion, and its ability to guide the optimization process.

1.5. Methodology

The methodology employed in this research involves concurrent use of experiments and results from corresponding numerical modeling. These techniques were adopted to complement each other, in the attempt to get an accurate understanding of phenomena governing the production of engine pollutants.

KIVA 2, the computer program used for numerical calculation, solves the three-dimensional, turbulent compressible Reynolds-Averaged Navier-Stokes equations, allowing for chemically reactive flows with sprays. Originally written at the Los Alamos National Laboratories, KIVA was modified at UBC to give an improved inflow boundary condition for the natural gas injection, and two-fuel combustion. Using experimentally determined initial conditions, KIVA is able to reproduce the combustion pressure trace and predict trends for NO\textsubscript{x} formation with crank angle degrees. It can also give a detailed information of the combustion development, including 3-D temperature, pressure field and velocity distributions, as well as species concentrations in the cylinder chamber.
All numerical simulations presented in this thesis have been performed by Dr. Guowei Li\textsuperscript{4}, in response to the problems addressed.

Experimental results have been obtained using a single cylinder, two-stroke diesel engine which is representative for urban bus applications. It has accurate electronic controls for the beginning and duration of injection, and is instrumented to gather all information necessary for combustion analysis (i.e. intake and exhaust temperature and pressure, air and fuel flows, etc.). Exhaust concentrations of NO\textsubscript{x}, CH\textsubscript{4}, THC and CO can also be measured. All HPDI experiments have been compared to similar tests performed with conventional diesel fueling.

The relationship between numerical modeling and experiments is of major importance. Each one of them can be used to give information or ideas which can be used to aid the other method. Computational fluid dynamics (CFD) calculations can provide in-cylinder combustion information at small level, which experimentally require optical access and sophisticated instrumentation and is very time consuming. For example, without engine optical access, details of the geometric parameters influence on the combustion process can only be given by numerical modeling. The real impact of the same variables on engine operation (efficiency and emissions) can only come from experiments.

However, although it provides very detailed information, KIVA cannot exactly describe combustion because of a certain degree of inaccuracy associated to the models (i.e. turbulence, combustion, pollutant chemistry, etc.) it uses. Hence, it was not expected to give quantitative results, but rather qualitative ideas. Often CFD simulations can provide valuable information on the research avenues to follow or avoid, before expensive experimental research is undertaken. During this research, the extensive numerical modeling performed has produced important answers to questions that appeared during the experimental study, which might otherwise have been un-solved.

\textsuperscript{4} CFD research scientist at Westport Research
Due to the large number of variables that needed investigation, the approach chosen was to study the effect of one parameter at the time, maintaining constant all other variables. This is only the first step of a full optimization process.

A challenge of this research results from the fact that the best value (i.e. to maximize efficiency) of one parameter may possibly depend on another variable. For example, a longer relative injection delay could be used with the scope of reducing the NO$_x$ emissions. But if the natural gas injection rate is not high enough to sustain fast burning, thermal efficiency will be decreased. The relative beginning of injection (RBOI) effects are also dependent on the absolute beginning of injection (BOI), as given by the electronic control signal. With a very delayed BOI, long RBOI may still be beneficial for NO$_x$ reduction, but detrimental to the engine efficiency. Instead, a diesel-like injection advance associated to the same RBOI could produce positive results for both emissions and efficiency. This kind of interdependence between engine parameters required parallel study of some variables, using a test matrix as the one given in Section 3.6.
2. PREVIOUS RESEARCH

Section 1.2 has provided a brief introduction of the different methods used for natural gas fueling of diesel engines. This chapter reviews previous experience on the different methods for high pressure direct injection. Section 2.1 concentrates on the design solutions, whereas 2.3 and 2.4 will provide a comparative analysis of what was learned about the efficiency and exhaust emissions of these designs.

2.1. Fueling of Diesel Engines with High Pressure Direct Injection of Natural Gas

Several studies have reported successful use of the high pressure direct injection of natural gas technology. This section illustrates the various design solutions adopted.

Inspired by the necessity of a more energy efficient system for powering LNG (liquefied natural gas) carriers, Sarsten et al [4] studied the use of boil-off gas as a main fuel for the prime engine. Natural gas evaporating from the tanks would be compressed at pressures around 200 bar, then directly injected into the cylinder and ignited by combustion of a small amount of pilot diesel fuel. Their principle was to retain the diesel fuel system unaltered, such than it can operate independently in case of a gas system failure. A pilot diesel was injected through the unmodified diesel injector, and the gaseous fuel through a separate valve, placed in a newly drilled hole, offset from the diesel one. With minimal changes required to the engine, this method would become a viable alternative for diesel engines already in operation.

Encouraged by results obtained in a small engine and combustion bomb tests (1982), Einang et al. [5] have implemented the high pressure gas injection technology on a large bore, medium speed, two-stroke single cylinder diesel engine. Recent (at that time) advancements in engine controls have allowed them to use electronic commands for the
gas injector valve, which enabled a flexibility in the control of start and duration of injection un-attainable with a mechanically controlled unit.

In 1983, Miyake et al. [6] have demonstrated the feasibility of a "high output, highly efficient natural gas burning diesel engine" on a four stroke mono-cylinder diesel engine with a large bore. Several ignition techniques and injection nozzles were tested.

A first method consists of sequential injection of a diesel pilot and natural gas. In "Pilot Fuel Injection" (PFI) method, the pilot fuel injection valve is mounted on the side of the cylinder, producing only one plume in a radial direction. The gaseous fuel is injected radially from the center of combustion chamber through the pilot fuel flame developed centrally. When using the PFI method, the amount of pilot injected can be kept to a small percentage of the total energy content required for full load operation. A disadvantage comes from the asymmetry of the pilot injection pattern, which limited the minimum amount of pilot necessary for stable ignition. In the second method, called "Mixed Fuel Injection" (MFI), a pilot fuel is injected in advance into the gas fuel, and consequently the two fuels are injected in mixed form through a centrally located injection valve. With the MFI method, pre-mixing of the fuels provides smaller fluctuations in combustion and stable engine operation. However, the minimum amount of pilot required for stable ignition represents a higher percent of the total fuel energy than in the PFI case.

By using an improved pilot fuel injection system, Miyake et al. significantly reduced the cycle to cycle variation, as well as the minimum amount of pilot necessary for stable ignition. This system consists of sequential injection of diesel pilot and high pressure natural gas (250 bar) through the same injector. The stability improvement is due to a symmetrical combustion pattern - the pilot jets are equally distributed around the circumference and interlaced with the gas ones.

Miyake et al. [7] pursued the development of the improved natural gas injector, with the scope of implementation of the high pressure direct injection of natural gas technology, on a large bore, low speed two-stroke engine. Their new injector design used high pressure oil at 300 bar as a seal between the hydraulic actuator fluid and the natural gas, which is injected at 250 bar. The two valves for pilot oil and natural gas can be
controlled independently, which enables the engine to operate solely on oil fuel, in case of a gas system failure. With gas fuel operation, the amount of oil injected per stroke is constant, and represents 5% of the total energy content supplied at full load. The need for a cleaner burning engine to drive a heat and power plant prompted Biwa et al. [8] to adapt Miyake et al. high pressure gas injection principle also to a medium speed engine. Their conversion has proven successful.

In 1987, Wakenell et al. [9] studied the technical feasibility of the high pressure late cycle injection of natural gas fueling, on a medium speed, blower scavenged two-stroke locomotive diesel engine. Their secondary objectives were to obtain adequate engine performance levels for rail application, develop a system oriented toward retrofit of in-service locomotives, and realize any potential improvements in thermal efficiency due to the use of the high pressure late cycle approach. A two-cylinder version of an EMD 567B engine was converted to operate in a dual fuel mode, with natural gas as the primary fuel and with pilot injection of a small amount of diesel, as ignition source for the gas. Due to space limitations, the natural gas was stored in liquefied form in a cryogenic tank, then compressed at 340 bar and subsequently vaporized for injection in gaseous phase. Using this method, the relatively low pressure LNG could be pressurized in liquid form to the high pressures needed for injection, with a minimum amount of work expended.

In order to minimize cost and changes required to the cylinder head, their newly designed gas injector was dimensioned to fit in the outer casing of the original diesel injector, and uses a hydraulically actuated solenoid valve for the injection of natural gas. The hydraulic solenoid gas injector valve is actuated by a diesel fuel injector pump, driven off the camshaft of the engine, through an adjustable differential mechanism which allows variations in injection timing. Natural gas was injected radially from the center of the combustion chamber at an angle of 25 degrees off the plane of the cylinder head. For pilot diesel, commercially available two-hole pencil injectors were installed offset from the gas valve, in grooves machined in the cylinder head at 30 degrees from horizontal. The pilot was injected 12 and 48 degrees off the cylinder head plane. Without optimized
injectors and injection system, Wakenell et al. achieved rated speed and load with as much as 98.2% (on an energy basis) natural gas fueling. At higher load regimes their engine incurred audible knock, which could only be eliminated by increasing the amount of pilot to 10%. At low load instead, dual fuel operation was not possible because of unstable engine operation. The authors attribute this result to an un-optimized injector design, and not to the use of natural gas.

Using a mixed fuel injection technique similar to that of Miyake et al [7], Hodgins et al. [10] experimented with high pressure direct injection of natural gas fueling. A two-stroke Detroit Diesel single-cylinder engine was fitted with a poppet valve injector, designed for simultaneous injection of natural gas and diesel pilot fuel. Before injection in the cylinder, the high pressure natural gas is mixed with small amounts of diesel pilot, in a mixing reservoir inside the injector. The hydraulically actuated and electronically controlled injector generates a conical sheet with some swirl, due to the tangential entry of the gas port into the mixing reservoir. Tests were performed with different diesel/gas ratios (on an energy basis), several injection pressures, pilot fuel with different autoignition properties and full range of load. Their results have proven that with HPDI of NG, full load capability of the diesel engine has been met and exceeded.

Parallel experiments and 3-D numerical simulations by Ouellette and Hill [11] have revealed a bi-stable character of the conical sheet generated by the unshrouded poppet used by Hodgins et al. For injection angles below 20 degrees, the jet would tend to attach to the upper chamber wall whereas for angles above this limit the jet would collapse downwards and attach to the piston crown. This behavior was due to pressure differences between the upper and lower sides of the jet. By using the same injector with a castellated end sleeve to break the conical sheet into separate jets, the pressure on both sides of the sheet would balance yielding a stable jet. Hence, subsequent experiments carried out by Tao et al. [12] were done exclusively using the shrouded poppet injector. They investigated the injection timing and fuel ratios required for best efficiency over the full load range, at a speed of 1200 rpm.
Fundamental studies have brought more insight into the natural gas ignition and combustion in diesel engines. Mtui and Hill [13] have shown that when mixed with natural gas, the ignition delay of diesel was significantly affected by the presence of methane. Injecting natural gas late in the compression cycle, the temperature of the intake charge is unchanged, but if the diesel is mixed with the gas prior to injection, its vapor will first come in contact with the cold natural gas jet. This suggests that the gas should contact the products of combustion rather than the cold vapor of diesel to have successful ignition. In light of these results, subsequent tests with HPDI of NG performed at UBC have used prior injection of the pilot diesel. This method also offers NOx reduction potential because of the split injection.

Using sequential injection of diesel and natural gas, Hodgins et al. [14] have successfully converted a commercial two stroke diesel engine, to high pressure direct injection of natural gas fueling with pilot ignition. Their electronically controlled and hydraulically actuated injector unit was used for injection of both fuels, and provided fast, flexible, accurate and repeatable operation with proven, reliable, existing technology.

Based on successful results obtained during the evaluation of different natural gas fueling systems [3], Meyers et al. [15] developed and optimized the LaCHIP combustion technology for a passenger locomotive application, with the aid of detailed CFD simulations and experiments carried out on a single cylinder engine. Following the work of Meyers et al., Bourn et al. [16] have implemented and tested the LaCHIP technology on an EMD 16-cylinder 710 engine. The prototype gaseous injection system has proven excellent reliability and combustion stability under all operating conditions. Testing has also shown that LaCHIP can use the original diesel injector without tip modification, such that the engine can still operate at full power using conventional diesel fueling, in case of failure of the high pressure natural gas system.
2.2. Thermal Efficiency

Generally, the motivation for conversion to alternative methods of fueling is the reduction in regulated exhaust emissions. But, for the conversion to be successful, penalties in thermal efficiencies must be avoided or limited.

In the case of high pressure direct injection of natural gas in diesel engines, theoretical considerations are in favor of the method. To achieve good thermal efficiency, an engine must first be able to operate at high compression ratios (air standard efficiency increases with compression ratio). Since HPDI uses a fresh air intake charge and heterogeneous fuel mixture, it is not subject to knock, and so can operate at compression ratios of conventional diesels. Another requirement for having good efficiency is a rapid ignition, fast combustion, and concentrated around the top dead center.

With HPDI, ignition delay should not be affected, because the start of combustion is given by the diesel pilot combustion. In consequence, the duration of combustion could be adjusted for high thermal efficiency by appropriately choosing the gas injection pressure (with diesel-like fueling, combustion rate depends on injection rate, which is a function of the gas pressure). Therefore, high pressure direct injection of natural gas fueling should be able to produce diesel-like thermal efficiency.

The degree of success in obtaining high thermal efficiency reported by previous researchers, is believed to be dependent on the level of technology optimization achieved.

With 73 % natural gas fueling, Einang et al. [5] achieved thermal efficiency slightly better than with 100 % diesel operation. They believe this is possibly due to a more rapid energy conversion, seen in the steeper pressure rise and higher maximum pressure. Since both dual fuel and full diesel operation were required, only one of the system could be optimized. For the LNG carriers, diesel was chosen to be the primary fuel. Hence, the diesel injector remained placed in the center of the combustion chamber, and due to
symmetry, was expected to give more efficient use of the available air than the gas valve. Higher thermal efficiency is expected with the natural gas injector located centrally.

Using "pilot fuel injection", at 85 % of maximum engine load and 5 % diesel pilot fuel energy input, Miyake et al. [7] matched thermal efficiency of the same engine running on conventional diesel fueling. In conclusion to their study, they state that "almost equal" performance has been obtained with natural gas fueling as with 100 % diesel operation. The power, thermal efficiency and reliability of the natural gas fueled medium speed engine developed by Biwa et al. [8], have proven to be the same as with conventional diesel fueling, even when the methane was diluted with nitrogen.

Without optimized injectors and injection system, Wakenell et al. [9] achieved rated speed and load with as much as 98.2 % (on an energy basis) natural gas fueling, but with a penalty of 10 % in thermal efficiency. It appears however that the injection timings they used were excessively advanced, and in consequence running at high load was affected by knock. The high ignition advance led to an increase in the amount of gas mixed to flammability limits, which produced a large premixed combustion phase. This was possibly the source of their noise, since diffusive burning cannot produce knock.

In 1992, Hodgins et al. [10] have proven that using high pressure direct injection of natural gas mixed with the diesel pilot, full load capability of the diesel engine can be met and exceeded. Thermal efficiencies were similar at low and medium load and increased at higher loads. However, these results were obtained at the expense of the CO and unburned hydrocarbons emissions, which were above the levels of conventional diesel. Hodgins et al.'s [14] improved design, with sequential injection of diesel and natural gas, has reached only about 75 % of the diesel engine full load capability, at 160 bar injection pressure. Throughout the load range achievable, thermal efficiencies were some 5 % lower than for the diesel baseline.

The LaCHIP technology using a gas injector improved by Meyers et al. [15], has demonstrated the ability to produce full power with 75 % reduction in NOx, while
maintaining low carbon monoxide and total hydrocarbon emissions. During steady state
testing of the LaCHIP fueled multi-cylinder engine, Bourn et al. [16] have shown that the
fuel efficiency penalty, for the 75-percent NO\textsubscript{x} reduction target, is approximately 10-
percent. A diesel equivalent efficiency could also be achieved with a lower, but
significant reduction in NO\textsubscript{x} (60 %).

2.3. Emissions

Most researchers whose work was described in this chapter undertook conversion of
existing diesel engines with the main aim of reducing exhaust emissions. Because of the
character of their combustion, diesels are responsible for production of most particulate
matter and NO\textsubscript{x} emissions from engine exhaust.

The diffusive type burning of the diesel liquid jets generates large amounts of soot in the
fuel rich zones of the flame. As it finds available air the soot continues to oxidate, and
leaves the engine exhaust in quantities much smaller then those formed during
combustion. The process of soot oxidation "freezes" as the cylinder temperature lowers
during the piston expansion. Efficiency of the soot depletion process depends in great
part of the quality of the fuel mixing process with air. Although conventional diesels run
with lean overall mixtures, locally there can be regions with high equivalence ratios
(rich), which suggests that some of the fuel will not find the air for complete oxidation.
This is the case for high load operation, which is most responsible for soot emission.

Diesels are also major producers of nitrogen oxides. NO\textsubscript{x} formation is a process very
sensitive to temperature. Because the diesel fuel has an elevated adiabatic flame
temperature and because of high compression ratios, diesel combustion generates large
amounts of NO\textsubscript{x}. Emissions of carbon monoxide and unburned hydrocarbons from diesel
engines are generally very low, because of large excess of air.
When used in fueling of diesel engines, natural gas is expected to produce much reduced emissions of particulate matter. Its gaseous form should help improve the mixing process with air and give more complete oxidation of the soot particles formed in the rich zones of the flame. The lower carbon-to-hydrogen ratio of the methane (main constituent in natural gas) is expected to produce lower emissions of particulate matter.

The use of natural gas is expected to give reduction in NO\textsubscript{x}, because of the lower adiabatic flame temperature of the methane. With sequential injection of diesel and natural gas the burning process will be split, which should lower combustion temperature. Hence, due to this character of combustion, additional NO\textsubscript{x} reduction could be obtained. Carbon monoxide and unburned hydrocarbon emissions from natural gas fueled engines are expected to be of the same order of magnitude as those produced by conventional diesels.

Einang et al.'s [5] gas fueling system with an un-optimized injector nozzle, gave slightly higher smoke readings. At full load, NO\textsubscript{x} emissions were reduced by 24 % relative to diesel operation. Hydrocarbon emissions were low and constant throughout their tests.

The method of high pressure direct injection of mixed natural gas and diesel, experimented by Hodgins et al. [10], has proven potential for nitrogen oxides and carbon dioxide reduction. However, emissions of hydrocarbon and carbon monoxide were increased and smoke levels comparable to those of conventional diesels. With the same injector, Tao et al. [12] obtained a 30 percent reduction in NO\textsubscript{x} emission rate without serious loss of efficiency, by delaying injection timing. Also, re-circulation of as little as 15 percent of the exhaust gas was shown to strongly reduce NO formation in the cylinder, with some loss in thermal efficiency and increase in unburned hydrocarbons. However, mixing the pilot diesel and natural gas, there is potential for very long ignition delays and poor combustion, followed by unacceptable levels of hydrocarbon emissions.

During recent tests performed at UBC and the Colorado Institute for Engine Research [17], utilization of the HPDI technology on a multi-cylinder two-stroke diesel engine led
to a reduction of 45% in NO\textsubscript{x} and of 75% in soot. Unburned hydrocarbons were slightly high and attributed to poor idling. All other emissions were within regulated levels. These results however, were obtained using the stock controller settings of a Detroit Diesel 6V-92 engine, which are optimized for conventional diesel operation. The same engine running on dual fuel mode is expected to have different optima, because of different combustion characteristics.

LaCHIP technology using a gas injector developed by Meyers et al. [15] and implemented on a single cylinder engine, has demonstrated the ability to produce full power with 75 % reduction in NO\textsubscript{x}, while maintaining low carbon monoxide and total hydrocarbon emissions. Bourn et al.'s [16] steady state testing by has shown that their program target of 75 % reduction in NO\textsubscript{x} emissions can also be achieved when fueling a multi-cylinder locomotive engine. Additionally, CO\textsubscript{2} emissions have been reduced by nearly 25 percent compared to diesel engine operation.

2.4. Gas Injection Pressure and Timing

As described in section 1.3, the injection pressure and timing are expected to have significant effects on efficiency and emissions. However, little information is available about the impact of these parameters. In the early development stage of the high pressure direct injection of natural gas, most researchers have used analytical methods to determine the appropriate gas injection pressure, and focused their study on optimization of other parameters, generally geometry.

Hodgins et al. [10] performed tests with the high pressure injection of mixed diesel and gas using several injection pressures, but their results did not give a clear trend. After adopting the sequential injection for diesel and natural gas method, Hodgins et al. [14] achieved only about 75 % of the conventional diesel load capability. Experiments on that
range have indicated that at the maximum achievable load, thermal efficiency seemed to increase with gas injection pressure.

In their development of the LaCHIP fueled multi-cylinder engine, Bourn et al. [16] have shown that gas injection timing and pressure are the only parameters they could control which significantly affected NO\textsubscript{x} and efficiency. Retarding gas BOI timing and reducing gas injection pressure caused a simultaneous reduction in NO\textsubscript{x} and thermal efficiency at all load regimes. Emissions of CO have shown significant sensitivity to those changes which affected in-cylinder mixing.

2.5. Summary

- High pressure direct injection (HPDI) of natural gas has proven to be a viable concept. The technology has been implemented in large, medium and small size engines, generally lower speed. Some designs have proven reliability.
- Comparison between different natural gas fueling methods (including spark ignition, dual fuel, etc) seems to indicate that the high pressure direct injection technology is more suitable for heavy duty applications.
- High pressure natural gas can be injected separately or mixed with pilot diesel. Separate injection of the pilot has produced a reliable and repeatable ignition source for natural gas. Mixed injection has given long ignition delay and reduced thermal efficiency.
- Injection of the natural gas and diesel pilot has been done using one or separate injectors. Both methods have proven successful. The symmetry jet arrangement of the one injector design recommends it as a better one from gas ignition and thermal stress view point.
- Using high pressure direct injection of natural gas, thermal efficiency can be equal or greater than that of the same engine operating on conventional diesel fueling.
- Research on heavy duty/transit bus mono and multi cylinder engines has shown the potential for emission reduction while preserving a high diesel-like efficiency. With
improved durability of the HPDI technology, there is great motivation for creating an optimized design.

- In any high pressure direct injection of natural gas design solution, the pilot diesel and gas jet geometry arrangement must be optimized to maximize efficiency.
- There is very little information on the effects of gas injection pressure and timing. So far it is known that retarding gas BOI timing and reducing gas injection pressure cause a simultaneous reduction in NO\textsubscript{x} and thermal efficiency.
- Nothing was found in literature about the effects of relative beginning of injection between diesel and natural gas on efficiency and emissions.
- With HPDI of natural gas technology, the interlace and inclination angles (as defined in Section 1.3) have un-known effects on combustion. So far, no information could be found about these two physical parameters and their importance for emissions and efficiency.
3. EXPERIMENTAL METHOD

The purpose of this chapter is to describe the experimental apparatus used in the research, including test engine, instrumentation, data acquisition and exhaust gas emission measurement systems. The chapter also presents the engine performance parameters calculated by the data processing software, and the experimental uncertainties associated to experimental measurements.

3.1. Apparatus

The experimental apparatus used in this research was described in the Master's theses of Gunawan (1992) and Douville (1994). It consists of a diesel engine with torque and speed control, and an emissions console capable of measuring several components present in the exhaust gas. The new component in the experimental apparatus is the HPDI injector. It has evolved from the design presented in [10] and [12], where the pilot and gas were mixed together, to have separate passages for diesel and natural gas, which now are injected sequentially. Details of the new injector have been presented in Section 1.2 of Chapter 1.

Measured parameters are: engine speed, torque, air and fuel mass flows, natural gas pressure, beginning of injection (BOI) and injection duration, intake air pressure and temperature, exhaust gas temperature, cylinder pressure, crank angle position and exhaust concentrations of methane, total hydrocarbons, carbon monoxide, nitrogen oxides and carbon dioxide.

Figure 3.1.1 shows schematically the experimental apparatus set-up, including the engine test cell and the control room. Interaction between the user and engine is done through a control panel, which gathers information from sensors and displays it in engineering units. Load and speed control, plus the commands for beginning and duration of injection, are sent from the control panel.
Mounted on one end of the crank shaft is a water brake dynamometer used for load measurement and engine control. The force produced by the engine is given by a load cell installed on its casing. On the free end of the dynamometer is placed the sensor which measures engine speed.

The engine cylinder pressure is sampled in the cylinder head. Signals from the transducer and electrical actuation of the HPDI injector are recorded and displayed on an oscilloscope. On the free end of the crank shaft is mounted an optical encoder which generates the triggering signal for data collection. Pilot diesel consumption is measured with a mass flow meter connected in a closed loop with the HPDI injector. The natural gas mass flow is given by a high sensitivity sensor located outside the engine test cell. To measure the engine air consumption, a laminar flow element was placed at the air intake, just after the filter. Ambient air temperature is measured at the same location.
Temperatures of the intake and exhaust charge are measured in the airbox and immediately downstream of the exhaust valve, respectively. The boost intake pressure is given by a gage placed in the airbox. Exhaust gas is sampled a few feet away from the engine and sent to the analyzers through heated lines. These are located in a separate room adjacent to the test cell. Information from all sensors is collected in the data acquisition, which is controlled by a computer.

### 3.2. Research Engine

The research engine is a naturally aspirated, two-stroke Detroit Diesel 1-71, retrofitted with DDEC 1 (Detroit Diesel) electronic controls of the injector unit. A schematic of the engine cylinder is given in Fig. 3.2.1. Engine specifications and performance capabilities are outlined in Table 3.2.2:

#### Table 3.2.2: Detroit Diesel 1-71 engine specifications

<table>
<thead>
<tr>
<th>Engine type</th>
<th>Two-stroke, naturally aspirated, blower-forced uni-flow scavenging</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>Bore and stroke (mm)</td>
<td>108 x 127</td>
</tr>
<tr>
<td>Connecting rod (mm)</td>
<td>254</td>
</tr>
<tr>
<td>Displacement (liters)</td>
<td>1.162</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>16:1</td>
</tr>
<tr>
<td>Rated power</td>
<td>11.2 kW @ 1200 RPM</td>
</tr>
<tr>
<td>Rated BMEP</td>
<td>4.8 bar @ 1200 RPM</td>
</tr>
<tr>
<td>Rated torque</td>
<td>76 Nm @ 1200 RPM</td>
</tr>
<tr>
<td>Inlet port closure</td>
<td>55° ABDC</td>
</tr>
<tr>
<td>Exhaust valve open</td>
<td>95° ATDC</td>
</tr>
</tbody>
</table>
Originally a mechanically controlled engine, the Detroit Diesel 1-71 has the flexibility of using two control circuits:

- the DDEC electronic control, a commercially available computerized system which determines duration and beginning of injection according to gathered information from sensors,
- a more flexible controller, designed in-house at UBC, which enables the user to freely choose beginning of injection (BOI) and pulse width (PW) duration, regardless of operating parameters. Such system permits evaluation of the engine performance characteristics and emissions, with different scenarios.
However, there are also some limitations to working on a single cylinder engine. The maximum load capability of the engine running on conventional diesel is 4.5 bar BMEP, which is only half the load output of the multi-cylinder 6V-92 engine. Also, the maximum speed that can be achieved is 1400 rpm, although the engine could run stable only around 1200 rpm. This speed is significantly lower then the 2100 rpm speed capability of the multi-cylinder engine.

Engine operation is controlled though adjustment of load, injection timing and duration. The dynamometer is an electro-pneumatically controlled water brake, which measures the output torque and dissipates the energy produced by the engine. The load absorbed is proportional to the amount of water that has to be displaced by straight vanes, which rotate at the engine speed inside the dynamometer casing. Water level in the housing is varied through the electro-pneumatically controlled inlet and exhaust orifices. The dynamometer's rotor is connected to the engine shaft and the casing moves freely around its axis. The force exerted on the stator during engine operation is measured by a load cell (strain gage) connected to the casing. Force measurement, converted into torque through multiplication with the stator moment arm, is used to determine the brake mean effective pressure (BMEP), which is a measure of engine load.

3.3. Instrumentation

This section will describe the instruments used for combustion analysis, with special emphasis on exhaust emissions measurement devices and their accuracy limitations.

Gases monitored in the research include CH$_4$, CO, THC, and NO$_x$. Nitrogen dioxides and hydrocarbons emissions present in engine exhaust can condense at low temperatures. Thus, in order to get measurements reflecting the actual levels from engine, sampling lines have to be heated to 190° C. Since the same temperature requirement is also needed for two of the analyzers, instruments have been grouped in separate cabinets.
The heated enclosure, shown schematically in Fig 3.3.1, contains the Ratfish FID RS55
total hydrocarbon analyzer, a sampling pump, two particulate matter filters and the
Anarad NO₂ to NO converter. The Siemens NO/CH₄ and CO analyzers, and the
Beckman CO₂ and O₂ instruments are placed in the cool instruments cabinet. Their
arrangement is illustrated in Fig 3.3.2. For measurement of CO, CH₄ and NO
concentrations, the non-dispersive infrared absorption (NDIR) method is used; total
hydrocarbon concentration is determined with an analyzer using the flame ionization
detection (FID) principle. The NDIR instruments need a cool and dry sample.

From the sampling location, exhaust gases flow through the heated lines to the heated
instruments. By passing through two filters of different size selection, the sample will
first be cleaned of particulate matter and then separated in two paths. One path goes to
the FID instrument to measure total hydrocarbon emissions, and the other leads to the
heated pump. From the pump, the flow is separated again into two paths. One path leads
to the NO₂ to NO converter and then through the chiller to the CH₄/NO analyzer, in the
cool instruments enclosure. The other path goes through the chiller and into the cool
instruments enclosure, where it is subsequently separated into three paths that lead to the
CO, CO₂ and O₂ analyzers.
Figure 3.4.1: Schematic of heated emission instruments enclosure

Figure 3.4.2: Schematic of cool emission instruments enclosure

1 Courtesy of Yinchu Tao [18]
Before each test and every two hours after that, each emission analyzer is calibrated using two gases of known composition. To determine the zero, the apparatus uses an inert gas like nitrogen. For the span$^2$, a calibration gas with a composition close to the maximum range of measurement is needed. The instruments output an electrical signal which is proportional to the concentration of emission measured. This method of measurement is believed to have good accuracy. The detailed checking of the emissions apparatus has proven that as long as steady flow is established, the value measured remains unchanged (i.e. at the passage of calibration gases through instruments no variation in the value measured occurs). However, errors are still significant. They can be due to calibration off-set which occurs during testing. This is believed to be one major source of error. The relative and net uncertainty in emission measurements is given in Table 3.4.1.

Appendix 5 shows the degree of reproducibility of typical emission data sets. Generally, emission measurements taken in different days display similar trends, but can have slight offsets in the values measured. These shifts (more obvious in the case of CH$_4$, THC and CO emissions) could be attributed to small inaccuracy of the analyzer zeroing process. The Siemens infrared analyzers determine slightly different points at each zeroing. These small changes could result in absolute shifts of the emission measured of up to 10 ppm, which represents as much as 20% of the values measured.

A problem encountered during the study was the accuracy of measuring small concentrations of CH$_4$ and NO$_x$. The methane emissions sampled are around 100 ppm (parts per million), whereas the maximum value measured by the analyzer is 3804 ppm. This means that the apparatus operates only in the first 4% of its range, where the relative error is greater. Similarly, measurements of NO$_x$ were performed mostly in the first 15% of the analyzer scale.

One peculiarity of this instrument (yet not understood) is the display of some residual emissions value when only air flowed through the system. For CH$_4$ these values were as large as 40 ppm, when most measurements indicated values around 100 ppm. NO$_x$ measurements had the same problem, but the residual value recorded is usually around 25 ppm, with respect to natural gas operation values around 250 ppm.

$^2$ The maximum concentration which can be measured
Engine speed measurements are done with a magnetic induction probe encoder. The sensor is placed over a 60-tooth gear rotating with the crank shaft, which is located on the free end of the dynamometer. After passing through the data acquisition board, engine speed is displayed on both control panel and the computer screen.

Instruments required for collecting cylinder pressure data include a piezoelectric pressure transducer coupled with a charge amplifier. The pressure transducer generates an electrical charge signal proportional to the pressures measured, which is then converted into a voltage signal by the charge amplifier. Engine cylinder pressure was sampled at each crank angle degree with a PCB 112B11 piezoelectric transducer, mounted in the cylinder head 11 mm recessed from the fire-deck. After the charge amplifier, the pressure signal is recorded by the ISAAC high speed data acquisition, and then sent to the computer. Because cylinder pressure signal accuracy is indispensable, separate attention has been devoted to it. A detailed explanation of the procedure required for obtaining reliable pressure data and a routine for checking results are given in Appendix 2.

To determine the electronic beginning and end of injection, the electrical actuation signal for the HPDI injector is also recorded and displayed on the oscilloscope. Triggering signal for data collection is given by the optical encoder mounted on one end of the crank shaft, which acts as a bottom dead sensor indicator as well as external clock.

Instantaneous and average values of the pilot diesel consumption are measured with an AVL gravimetric mass flow meter, which is connected in a closed loop with the HPDI injector. The system consists of continuous weight measurement of a re-circulating tank filled with diesel, which has a fuel line leaving to the engine and another one returning from it. Real-time weight measurement of the tank indicates the net fuel consumption during operation.

Natural gas\textsuperscript{3} from the building network is supplied to the injector from a series of bottles, which are pressurized with two commercial compressors. After passing through a

\textsuperscript{3} Natural gas composition and its thermodynamic properties (supplied by BC Gas) are given in Appendix 3.
pressure regulator, the compressed natural gas (CNG) mass flow is measured with a Micro Motion high capacity transducer. This operates on the Coriolis acceleration principle, and is placed outside the engine test cell. The Micro Motion sensor has a maximum error of ±0.4% at full range, but its level of inaccuracy is believed to be higher, because the instrument operated on the first 5% of its range, where accuracy is reduced (all 1-71 engine measurements were in the range of 0.5-2.5 kg/hr, whereas the transducer span is 50 kg/hr).

A laminar flow element measuring the pressure differential across a restriction placed in the air intake, is used to determine the engine absolute air flow consumption. For conversion into standard conditions, temperature measurements are taken at the air filter intake. Other important parameters for combustion analysis are the temperatures of the intake and exhaust charge, as well as the boost pressure. Temperatures are measured with thermo-couples in the airbox and immediately downstream of the exhaust valve, respectively. The intake pressure is measured with a gage manometer mounted in the airbox.

The calibration curves for all instruments are given in Appendix 4 and their maximum (relative and net) error is shown in Table 3.4.1.
3.4. Data Acquisition

A software package written in the Alternative Fuels Laboratory at UBC coordinates data acquisition and the electronic control module (ECM) sending commands to the injector unit. Data collection is done separately for the "low speed" and "high speed" type of signals which are received from the engine.

Information from all sensors except cylinder pressure, bottom dead center and crank angle interval passes through an interconnect board and subsequently to the A/D card installed in the computer. Following digitization of the signals acquired, the computer displays engineering parameters relevant to combustion performance and emissions analysis. Each value of the engine parameter measured is sampled 100 times and then averaged by the software. The interconnect board is capable of receiving up to 16 channels. Measured engine parameters, their channels and instrument errors are shown in table 3.4.1.

Table 3.4.1: Measured engine parameters

<table>
<thead>
<tr>
<th>Channel</th>
<th>Measured parameter</th>
<th>Maximum relative error(^1) (%)</th>
<th>Maximum reading net uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>CNG mass flow</td>
<td>±0.4</td>
<td>±0.01 kg/hr</td>
</tr>
<tr>
<td>1</td>
<td>Beginning of injection</td>
<td>±0.2</td>
<td>±0.04 CA deg</td>
</tr>
<tr>
<td>2</td>
<td>Ambient temperature</td>
<td>±2</td>
<td>±1 deg Celsius</td>
</tr>
<tr>
<td>3</td>
<td>Pulse width</td>
<td>±0.2</td>
<td>±0.04 CA deg</td>
</tr>
<tr>
<td>4</td>
<td>Torque</td>
<td>±0.1</td>
<td>±0.1 Nm</td>
</tr>
<tr>
<td>5</td>
<td>Engine speed</td>
<td>±0.1</td>
<td>±1.5 rpm</td>
</tr>
<tr>
<td>6</td>
<td>Intake air pressure</td>
<td>±2</td>
<td>±2 deg Celsius</td>
</tr>
</tbody>
</table>

\(^1\) All instrument error, except emissions, were given by manufacturer. For emission instruments the error was deduced from the data on repeatability of measurements. Since all results were affected by data variability, trendlines were used to estimate the errors. The largest variation between trendlines of data taken in different dates is considered the maximum net experimental uncertainty. Typical repeatability of the emission measurements is given in Appendix 5. This method does not take into account the systematic error, which is unknown.
Crank angle variation of the cylinder pressure signal is recorded with a separate "high speed" data acquisition (ISAAC), independent of the "low speed" system used for the rest of the sensors. Bottom dead centre (BDC) and crank angle interval signals, generated by an optical encoder mounted on the crank shaft, are used to trigger and respectively determine the time intervals at which ISAAC needs to sample the pressure transducer. As much as 100 engine cycles can be recorded and stored by ISAAC. Data is then downloaded to the PC and processed for combustion analysis by other two programs written at UBC. Final output is a file containing the cylinder pressure variation with crank angle degree, together with information about the engine operation regime.

### 3.5. Performance Parameters

#### Thermal Efficiency

Thermal efficiency of an engine $\eta_{th}$ is defined as the ratio between the rate of shaft or brake work output and the rate of heat input:
where the rate of work output is the brake power $P$ produced by the engine and heat input is equal to:

$$\dot{Q} = m_f \cdot LHV$$

(3.2)

where $m_f$ is a measured parameter that represents the fuel mass flow and LHV is the lower heating value of the fuel.

Therefore, thermal efficiency of the engine can be expressed as:

$$\eta_{th} = \frac{P}{m_f \cdot LHV}$$

(3.3)

The brake power $P$ is a calculated parameter which can be obtained from measurements of the engine speed $N$ and torque output $T$:

$$P = 2\pi \frac{N}{60} T$$

(3.4)

in which $N$ is in revolutions per minute.

If more fuels are involved, then the thermal efficiency becomes:

$$\eta_{th} = \frac{P}{\sum_i (m_f LHV_i)}$$

(3.5)
where \( m_i \) is the mass flow of the fuel \( i \) and \( LHV_i \) is its lower heating value.

In order to have a common basis for comparison under different atmospheric conditions, the brake power is corrected to standard air conditions: 100 kPa inlet air pressure and 25\(^\circ\)C inlet air temperature. Correction is made according to the SAE standard J1349 JUN85 [19]. The fuel lower heating values used in the efficiency calculations were 43200 kJ/kg and 49417 kJ/kg\(^2\) for the diesel pilot and compressed natural gas, respectively. These values vary from day to day, as shown in Fig. A3.3, Appendix 3. During the period of tests, the natural gas lower heating value did not change by more then 1.2 %.

All engine parameters have a cumulated error because of the error in measured parameters. Its magnitude was determined using the theory of propagating errors, which assumes that experimental data has a Gaussian distribution about the mean. The uncertainty of measured parameters, necessary in the calculation of the composite error, was determined experimentally or was given by the instrument manufacturers. The relative error in brake power can be calculated as follows [20]:

\[
\frac{\delta P}{P} = \sqrt{\left(\frac{\delta N}{N}\right)^2 + \left(\frac{\delta T}{T}\right)^2}
\]

Both measurements of engine speed and torque have a relative error of ±0.1%. Hence, the relative error in brake power will be ±0.14 %. The relative error in thermal efficiency can be found in a similar manner. After arranging terms, the cumulated error becomes:
Typical relative errors in diesel, natural gas mass flow measurements and LHV are ±2 %, ±8 % and ±1.2 %, respectively. Using a mass ratio of diesel to natural gas of 30 %, the relative errors in torque, speed and the fuel lower heating values defined above, the maximum relative error in thermal efficiency is ±6.4 %. At maximum load, this produces an absolute uncertainty in efficiency of ±1.8 %. To increase the level of confidence in the experimental results, a large number of points were taken at each operational regime. The maximum absolute scatter in data obtained with diesel operation is 1 %. Appendix 5 illustrates the reproducibility of typical efficiency data sets.

**Brake Mean Effective Pressure**

Mean effective pressure is an engine parameter which characterizes the work produced by an engine regardless of its size. It is defined as the ratio between the work produced W per cycle and the cylinder displacement volume V_d:

\[
MEP = \frac{W}{V_d}
\]  

(3.6)

If the work W is the torque measured at the dynamometer, then MEP is called brake mean effective pressure (BMEP).

With the aid of Eq. 3.4, the brake mean effective pressure can be written as:

---

\[ \delta \eta = \left[ \left( \frac{\delta N}{N} \right)^2 + \left( \frac{\delta T}{T} \right)^2 + \left( \frac{LHV_{\text{dil}} \cdot \delta \left( \frac{\delta m_{\text{dil}}}{m_{\text{dil}}} \right)}{m_{\text{dil}} \cdot \frac{LHV_{\text{gas}}}{m_{\text{gas}}} + LHV_{\text{gas}}} \right)^2 + \left( \frac{\delta LHV_{\text{gas}}}{LHV_{\text{gas}}} \right)^2 \right]^{1/2} + \left( \frac{m_{\text{dil}} \cdot LHV_{\text{dil}} + LHV_{\text{gas}}}{m_{\text{gas}} \cdot LHV_{\text{gas}}} + 1 \right) \]

\[ \left( \frac{\delta m_{\text{gas}}}{m_{\text{gas}}} \right)^2 \]

\[ \left( \frac{LHV_{\text{gas}}}{m_{\text{gas}}} \right)^2 \]

\[ \left( \frac{\delta LHV_{\text{gas}}}{LHV_{\text{gas}}} \right)^2 \]

\[ \left( \frac{\delta m_{\text{gas}}}{m_{\text{gas}}} \right)^2 \]

\[ \left( \frac{\delta LHV_{\text{gas}}}{LHV_{\text{gas}}} \right)^2 \]

\[ \left( \frac{\delta m_{\text{gas}}}{m_{\text{gas}}} \right)^2 \]
Following the procedure described above, it can be shown that the relative error in BMEP is equal to the relative error in torque, which is ±0.1%.

The combustion process efficiency can also be estimated with the aid of the indicated mean effective pressure (IMEP) which represents the area inside the P-V diagram, determined through cylinder pressure sampling. IMEP is defined as the ratio between the indicated work and the cylinder displacement volume $V_d$.

\[
IMEP = \frac{W_{\text{ind}}}{V_d}
\]  

$W_{\text{ind}}$ can be determined through numerical integration of the cylinder pressure, when the kinematic characteristics of the engine are known.

\[
W_{\text{ind}} = \int PdV
\]

**Wet-Basis and Brake Specific Emissions**

The experimental apparatus used in the research measures volumetric concentrations of carbon monoxide (CO), nitrogen oxides (NO\(_x\)), methane (CH\(_4\)), total hydrocarbons (THC) and carbon dioxides (CO\(_2\)). With the exception of THC, all emissions need to be free of exhaust water, therefore the analyzers output dry percentages. Since the amount of water in the exhaust is significant (a complete chemical reaction should only give water, carbon dioxide and nitrogen), the values measured are considerably higher than their actual concentration. To obtain an accurate analysis of the combustion efficiency and level of emissions, the data collected must be converted to wet-basis. This consists of correcting the measured values, to account for the water vapor present in the exhaust.
In performing this conversion, the data acquisition software uses the algorithm derived by Heywood [21]. The dry $x_i^{\ast}$ and wet $x_i$ mole concentrations of a species $i$ obtained by combustion of a generic fuel of composition $C_nH_m$ are related through:

$$x_i = (1 - x_{H_2O}^{\ast}) x_i^{\ast}$$  \hspace{1cm} (3.10)

Where

$$x_{H_2O} = \frac{m}{2n} \left[ \frac{x_{CO}^{\ast} + x_{CO_2}^{\ast}}{1 + \frac{x_{CO_2}^{\ast} - x_{CO}^{\ast}}{K_1 x_{CO_2}^{\ast} + \frac{m}{2n} (x_{CO}^{\ast} + x_{CO_2}^{\ast})}} \right]$$  \hspace{1cm} (3.11)

The presence of two fuels requires adoption of an average composition for the $C_mH_n$, based on the ratio between the fuel flows of diesel and natural gas. The constant $K_1$ relates the wet basis concentrations of $CO_2$, $H_2O$, $CO$ and $H_2$ in the following way:

$$K_1 = \frac{x_{CO}^{\ast} x_{H_2O}^{\ast}}{x_{CO_2}^{\ast} x_{H_2}^{\ast}}$$  \hspace{1cm} (3.12)

Since the concentration of $H_2$ is not measured directly, values for the empirical constant $K_1$ need to be taken from published exhaust gas composition data (Heywood [21]). With the aid of formulae (3.10), (3.11) and (3.12) the mole fractions of $CO$, $CO_2$, $NO_x$ and $CH_4$ can be converted from dry to wet basis. The flame ionization detector measures THC concentrations already in wet basis, so they can be used directly. The same sets of formulae enable wet-basis mole fractions calculations of $N_2$, $H_2$ and $H_2O$, which are not measured directly.
Brake specific emissions are a way of expressing the exhaust characteristics of an engine regardless of its size. They are defined as the mass flow of one species $m_i$ per unit brake power $P_b$.

$$b_{si} = \frac{m_i}{P_b}$$  \hspace{1cm} (3.13)

The mass flow rate of a component $m_i$ can be determined as follows:

$$m_i = x_i \cdot \frac{\dot{M}_{exh}}{M_{exh}} \cdot M_i$$  \hspace{1cm} (3.14)

Where $M_i$ and $M_{exh}$ are molecular weights of the component $i$ and respectively exhaust. The exhaust molecular mass can be found adding the molecular weights of the components, reduced by their mass fraction in the mixture.

$$M_{exh} = \sum_i (x_i \cdot M_i)$$  \hspace{1cm} (3.15)

in which the species included in the summation are CO, CO$_2$, NO, THC, O$_2$, H$_2$O and H$_2$. Only NO is considered in the calculation since it is the main component of NO$_x$.

The mass flow of exhaust is the summation between the measured mass flows of air, diesel and natural gas.

$$\dot{M}_{exh} = \dot{M}_{air} + \dot{M}_{diesel} + \dot{M}_{gas}$$  \hspace{1cm} (3.16)

Brake specific emissions are customarily expressed in grams per brake horse power hour, g/bhp-hr.
3.6. Test Matrices

This section illustrates the experimental test matrix, including values of the physical parameters corresponding to all conditions.

Table 3.6.1: Experimental Test Matrix

<table>
<thead>
<tr>
<th>Variable to be analyzed</th>
<th>Gas injection pressure</th>
<th>Absolute beginning of injection</th>
<th>Relative beginning of injection</th>
<th>Interlace angle</th>
<th>Inclination angle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load (bar BMEP)</td>
<td>0-4</td>
<td>0-4</td>
<td>0-4</td>
<td>0-4</td>
<td>0-4</td>
</tr>
<tr>
<td>BOI (degrees BTDC(^3))</td>
<td>Adjusted for start of combustion at TDC(^4)</td>
<td>0-15</td>
<td>0-15</td>
<td>Adjusted for start of combustion at TDC</td>
<td>Adjusted for start of combustion at TDC</td>
</tr>
<tr>
<td>RBOI</td>
<td>Long(^5)</td>
<td>Long</td>
<td>Short(^6), long</td>
<td>Long</td>
<td>Long</td>
</tr>
<tr>
<td>GIP(^6) (bar)</td>
<td>100, 130, 160</td>
<td>130</td>
<td>130</td>
<td>130</td>
<td>130</td>
</tr>
<tr>
<td>Number of gas jets</td>
<td>7</td>
<td>7</td>
<td>7</td>
<td>6, 7, 8</td>
<td>6</td>
</tr>
<tr>
<td>Gas jets inclination (degrees)</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>10, 15</td>
</tr>
</tbody>
</table>

All tests were performed at 1200 rpm\(^7\) because at this speed the engine had smooth operation. At 4 bar BMEP, the engine running on conventional diesel produced high smoke emissions, and therefore this was the maximum experimental load. The injection

\(^3\) Before top dead center  
\(^4\) Top dead center  
\(^5\) To be defined in Chapter 7  
\(^6\) Gas injection pressure  
\(^7\) Revolutions per minute
pressure range was dictated by two conditions: beyond the highest pressure, 160 bar, the engine had unstable operation, and below the lowest limit, 100 bar, the injector would not operate choked. The number of diesel jets used for all geometries was 6 and their inclination from fire-deck 10 degrees. In certain tests, BOI was adjusted to obtain start of combustion at TDC, such that meaningful comparisons could be made between similar burning patterns.
4. ENGINE OPERATIONAL INSTABILITY WITH PILOT IGNITION

This chapter is dedicated to a problem that appeared during testing of the HPDI technology, which led to failure of the initial studies. The problem was an unstable engine operation at low load and large variability in emission measurements during steady state testing. Unstable engine operation means wide speed variation which can lead to un-controllable over-speeding or halting. The following sections will give evidences of emission variability, explain the origin of engine instability and describe the investigation that identified a solution to the problem.

4.1. Evidences of Variability

Initial experiments, with a prototype natural gas injector having 6 nozzle holes for both natural gas and diesel pilot, revealed engine operational instability at low load and gradual time-wise variability in exhaust emission measurements. Low frequency fluctuations of the combustion pressure trace slope were also observed with a time scale of some 2 minutes.

All these phenomena occurred during steady state testing, where load and speed need to be maintained constant. Instead, the engine operated with speed oscillations of up to 15 % (of the average) and variability in emissions, even when the amount of fuel injected per cycle and load setting were kept unchanged. The same engine running on conventional diesel fueling did not display any of these symptoms, and so the instability was attributed to the pilot ignited high pressure direct injection of natural gas.

In order to perform any experimental study, accuracy and repeatability of the data is indispensable. So, to continue the research, the origin of these problems had to be identified, their mechanism understood and measures had to be taken to eliminate it. Figure 4.1.1 reveals the magnitude of engine speed changes during steady state experiments where load was varied from 4 to 0 bar brake mean effective pressure
(BMEP). Targeted engine speed was 1200 rpm. The x axis displays the number of data points, which were collected manually at intervals of approximately 8 seconds.

![Graph showing engine speed and BMEP variations](image)

**Figure 4.1.1:** Variations in engine speed of the HPDI injector with 6 diesel and 6 natural gas jets

At 3 bar BMEP, the detailed graph in Fig. 4.1.2 reveals an almost sinusoidal speed behavior. Also, one can see how the amplitude of these variations increases with load reduction (Fig. 4.1.1).
Figure 4.1.2: Variations in engine speed associated with the use of an HPDI injector with 6 diesel and 6 natural gas jets; target conditions: 3 bar BMEP and 1200 rpm

Variability in the data, especially NO$_x$, was also obtained at high load, when the engine ran in alternating metastable regimes, with changes in speed of smaller magnitude. In Fig. 4.1.3 one can see oscillations in NO$_x$ and CH$_4$ emissions corresponding to the variations in engine speed of Fig. 4.1.2.
Figure 4.1.3: NO\textsubscript{x} and CH\textsubscript{4} variability associated to unstable engine operation, when using an HPDI injector with 6 diesel and 6 natural gas jets.

The variability of emissions and efficiency data taken with this prototype injector created difficulty in determining clear trends. Figure 4.1.4 is an example of these results, showing NO\textsubscript{x} emission variation with load, for HPDI and diesel fueling. One comment must be made about this result: the variation in NO\textsubscript{x} emission obtained with this 6-6\textsuperscript{1} injector is as large as 130 ppm (40% of the average value measured), which is an order of magnitude greater than the maximum net uncertainty due to instrument error (10 ppm). Therefore the variability in NO\textsubscript{x} emissions cannot be attributed to measurement error.

\textsuperscript{1} 6-6 means respectively 6 diesel pilot and 6 natural gas jets.
Figure 4.1.4: Variability in NO\textsubscript{x} emissions from HPDI using an injector with 6 diesel and 6 natural gas jets and conventional diesel fueling.

To help identify the sources of engine instability and variability in emissions, two questions had to be answered first:

- Does the engine instability come from the combustion process or are there other factors producing it?
- Is there a relationship between the engine instability, combustion rate change and data variability?
4.2. Investigation

Detailed examination of the engine behavior indicated that the increase in combustion rate (seen on the oscilloscope sampling cylinder pressure) coincides with an acceleration of the engine and higher NO\textsubscript{x} generation. This result made sense, because an increase in combustion rate would produce higher flame temperature and therefore increased NO\textsubscript{x}. The relatively small changes in pressure development produced more obvious differences in the NO\textsubscript{x} production, because of its high sensitivity to temperature (Figs. 4.1.2 and 4.1.3). Also, higher burning rate led to faster combustion and more work produced; in conditions of constant load, this would result in acceleration of the engine.

Such observations pointed out that these phenomena are inter-related, and they all come from the combustion process. So to eliminate the emissions variability, we had to understand what causes the change in combustion rate and then take measures to remove its oscillations.

As described in Chapter 1, with the current injector design, the gas needle is free to rotate during engine operation. Since the diesel pilot jets emerge from the rotary gas needle, and the gas jets from the fixed injector body, the diesel jets will change position relative to the gaseous ones during operation. Implications of the changes in interlace angle (as defined in chapter 1 and shown in Fig. 4.2.1) could be as follows: with pilot ignited HPDI fueling, diesel combustion is used as ignition assist for the natural gas jets. The shifting of relative position between diesel and gas jets will produce changes in the contact area between natural gas and the diesel combustion sites. This will possibly influence the gas ignition area and thus combustion rate and NO\textsubscript{x} emissions. An investigation was made to see if the interlace angle could be the cause for the burning rate variations. In the absence of engine optical access, special emphasis was given to numerical modeling.
CFD simulations have been performed for an HPDI injector with a design configuration consisting of 6 diesel and 6 natural gas jets, equally spaced around the circumference, and three interlace angles: 0, 30 and 15 degrees (Fig. 4.2.2). These angles were chosen to represent two extreme cases of angular variation and an intermediate one. In the two-stroke engine used for research, the intake charge has a clockwise swirl motion. Hence, for the case of 15 degrees interlace angle, the diesel jets were placed "upstream" from those of natural gas, such that with a clockwise motion of the air, the gas plumes become surrounded by the pilot flame. Numerical modeling was performed using the geometry of a 6V-92 Detroit Diesel engine, which is similar to that of the 1-71 Detroit Diesel used in experiments. Due to symmetry, the KIVA computations were performed on a 60 degree sector of the combustion chamber. Hence, the boundary conditions on the two radii were periodic. Natural gas was injected at 185 bar, and the diesel and gas jets were both inclined at 10 degrees from fire-deck. Four test conditions were chosen from the AVL 13-mode testing standard, and are defined in Table 4.2.3. They are a combination of two extreme cases of load and speed.
Figure 4.2.2: Schematic of three interlace angle arrangements for a "6-6" injector

Table 4.2.3: Numerical modeling engine test conditions

<table>
<thead>
<tr>
<th>Test</th>
<th>Speed (rpm)</th>
<th>Load (%)</th>
<th>BMEP (bar)</th>
<th>BOI (deg ATDC)</th>
<th>Diesel/Gas (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mode 3</td>
<td>1200</td>
<td>25</td>
<td>2.17</td>
<td>-6</td>
<td>24.10</td>
</tr>
<tr>
<td>Mode 6</td>
<td>1200</td>
<td>100</td>
<td>8.95</td>
<td>-6</td>
<td>8.65</td>
</tr>
<tr>
<td>Mode 8</td>
<td>2100</td>
<td>100</td>
<td>6.33</td>
<td>-10</td>
<td>8.54</td>
</tr>
<tr>
<td>Mode 11</td>
<td>2100</td>
<td>25</td>
<td>1.75</td>
<td>-12</td>
<td>23.92</td>
</tr>
</tbody>
</table>
Because the load and speed capabilities of the 6V-92 engine are superior to those of 1-71, the only numerical modeling case that pertains to experiments is Mode 3, 2.17 bar BMEP and 1200 rpm. These results will be compared with experimental emissions data.

CFD simulations on Mode 3 operating conditions revealed two important results:

- Combustion rate changes with interlace angle. Fifteen and zero degree interlace angles produced the fastest combustion rate, the 15 deg. one being slightly higher. Thirty degree interlace angle had the slowest burning rate. Figure 4.2.4 shows the computed pressure development with crank angle ATDC (after top dead center), for HPDI injector with interlace angles of 0, 15 and 30 degrees.

- Variation of the interlace angle has influence on NO\textsubscript{x} emissions. The interlace angle cases which produced greater NO\textsubscript{x} emissions correspond to those having higher combustion rate. This result makes sense, because faster combustion yields higher flame temperature and thus more NO\textsubscript{x}. Hence, the cases of 15 and 30 deg. interlace

![Figure 4.2.4: Computed cylinder pressure variation with crank angle, for interlace angles of 0, 15 and 30; 2.17 bar BMEP and 1200 rpm](image)
angle produced the highest and lowest NO$_x$, respectively. Figure 4.2.5 shows the computed NO$_x$ generation with crank angle ATDC, for the same simulation case as in Fig. 4.2.4.

![Graph of NO$_x$ emission variation with crank angle](image)

**Figure 4.2.5:** Computed NO$_x$ emission variation with crank angle, for interlace angles of 0, 15 and 30; 2.17 bar BMEP and 1200 rpm

Numerical modeling confirmed qualitatively experimental results. If the gas needle rotates, combustion rate and NO$_x$ formation change according to the interlace angle formed between the diesel and gas jets. However, quantitatively the results are not in perfect agreement. The maximum experimental NO$_x$ (Fig. 4.1.4) is 50 % greater then the minimum emissions measured. CFD results instead show only a 14 % increase of maximum versus the minimum emissions computed. This discrepancy is attributed to poor performance of KIVA's NO$_x$ or combustion model.
An explanation for the combustion rate changes with interlace angle was suggested by analysis of the KIVA combustion development animation. At 15 degree interlace angle, due to charge swirling motion, the gaseous jet injected becomes entirely surrounded by the hot flame of pilot diesel combustion, and thus the high ignition area gives fast flame development. For the 30 degrees angle instead, the gas jet ignites on a much smaller area only on its sides, because it's at the furthest distance from the diesel combustion sites. A schematic of the natural gas ignition mechanism for interlace angles of 15 and 30 deg. is given in Figure 4.2.6.

![Interlace angle = 15](image)

![Interlace angle = 30](image)

**Figure 4.2.6**: Natural gas ignition mechanism with interlace angles of 15 and 30 deg.

Figure 4.2.7 shows the computed combustion chamber temperature distribution on a cut plane 4 mm below the fire-deck, at the same instant (2.5 crank angle ATDC) after injection (BOI 5 deg. BTDC), for interlace angles of 0, 15 and 30 deg. respectively.
The areas with high combustion temperature are represented with dark shades of grey. At zero deg. interlace angle, the charge swirling motion produces a small offset of the pilot flame during injection time delay, such that the natural gas jet comes in contact with high temperature gases from diesel combustion on approximately 75% of its area (2-D).

Figure 4.2.7: Combustion chamber temperature distribution (deg. K) for 0, 15 and 30 deg. interlace angles on a cut plane 4 mm below the fire deck

In the 15 deg. case, the natural gas jet becomes entirely surrounded by the high temperature region belonging to the pilot combustion. Having maximized gas jet ignition area, the combustion rate will be highest. With an interlace angle of 30 deg. instead, due
to a greater distance between jets, the natural gas comes in contact with high temperature gases on a much smaller area, only on its sides. The small jet ignition area will lead to the lowest burning rate of all cases. Computational results for the remaining operating conditions have proven small influence of the interlace angle on combustion rate but more significant impact on NO\textsubscript{x} emissions. These results are presented in Appendix 7.

In conclusion, variation of the interlace angle during engine operation was found to be responsible for the changes in combustion rate, engine instability and also the variability in exhaust emissions. The interlace angle is thus an important parameter, and should be controlled to optimize combustion. However, since eliminating needle rotation was impossible during the timeline of this thesis, another solution has been chosen to deal with the operational instability and variability in emissions.

4.3. Reduction of Variability

The combination of 6 diesel and 6 natural gas jets is perhaps the worst case, because with needle rotation, the interlace angle (equal for all pairs of diesel-gas jets) varies widely from 0 to 30 degrees. However, when choosing a different number of holes for gas and diesel pilot, the interlace angle is not the same anymore for all pairs of jets. To evaluate the effects of gas needle rotation when using different number of holes for pilot and gas, the average interlace angle (defined as the linear average of the interlace angles for all pairs of diesel-gas jets) can be used to compare different tip geometries. It was decided to experiment injectors with unequal number of jets for the diesel and gas based on the observation that the average interlace angle will go through smaller variations during gas needle rotation.

What combination of diesel vs. gas natural number of holes is best, required a different study. Practical considerations suggested that the number of diesel jets be left unchanged and the number of gaseous jets increased to 7, 8 or 9. Since the interlace angle was found to be responsible for the combustion rate changes, the objective was to make its average
go through minimum variations during needle rotation. Figure 4.3.1 shows computational results of the average interlace angle variation with gas needle rotation angle, obtained with 7, 8 and 9 gas jets. At each degree of relative rotation between the diesel and gas needles, the interlace angle between any pair of diesel-gas jets changes and so is its average. Since the average angle has repetitive variation, it is enough to study its change over one period (equal to the angle between two adjacent diesel or gas jets). The x axis displays the relative angle of rotation between one diesel and gas jet. Figure 4.3.1 indicates that a combination 6 diesel - 8 gas jets ("6-8") would be best, because it maintains a constant average interlace angle of 15 degrees at any needle rotation angle.

![Figure 4.3.1: Variation of the average interlace angle between jets with gas needle angle of rotation. Number of diesel holes is equal to 6.](image)

Since the "6-7" combination also has small interlace angle variations, "6-7" and "6-8" injector tips have been designed and machined, to determine experimentally which one
will give the best efficiency-stability-emissions results. Figure 4.3.2 shows a schematic diagram of the jet arrangement for the 6-7 and 6-8 injectors.

![Schematic Diagram](image)

**Figure 4.3.3**: Schematic diagram of the "6-7" and "6-8" tip geometry configurations

The characteristics of the three injector tips compared are given in Table 4.3.3:

**Table 4.3.3: Injector tip characteristics**

<table>
<thead>
<tr>
<th></th>
<th>&quot;6-6&quot;</th>
<th>&quot;6-7&quot;</th>
<th>&quot;6-8&quot;</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of diesel holes</td>
<td>6</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>No. of gas holes</td>
<td>6</td>
<td>7</td>
<td>8</td>
</tr>
<tr>
<td>Diesel jet inclination angle (degrees)</td>
<td>10</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Gas jet inclination angle (degree)</td>
<td>10</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>Gas hole diameter (inch)</td>
<td>0.021</td>
<td>0.021</td>
<td>0.0195</td>
</tr>
<tr>
<td>Gas injection pressure (bar)</td>
<td>130</td>
<td>109</td>
<td>130</td>
</tr>
</tbody>
</table>
The 7 and 8 gas hole injectors used in this stage of research had been designed to have a 15 deg angle from the fire deck because CFD simulations have revealed that with 10 deg inclination the gas jets tend to cling to the top wall (Coanda effect\(^2\)). The attachment will reduce the contact area between natural gas and air, yielding possibly a lower combustion rate and reduced NO\(_x\). Detailed information about jet inclination angle effects will be given in Appendix 8.

To maintain constant gas injection rate in conditions of increased hole area, the injection pressure (GIP) during the 7 and 8 gas hole injector testing has been reduced accordingly. For gas pressures above 130 bar the nozzle is choked throughout the entire injection process. Hence, with constant exit velocity, the gas density (which is proportional to pressure) must be reduced to compensate for the area increase. All experiments have been performed at 1200 rpm, 130 bar GIP (or equivalent) and load varying from 0 to 4 bar brake mean effective pressure (BMEP). Beginning of injection (BOI) has been adjusted at each load to obtain start of combustion at top dead center (TDC).

Figures 4.3.4, 4.3.5 and 4.3.6 show a comparison between the emissions of NO\(_x\), total hydrocarbons (THC) and CO, respectively, obtained with HPDI injectors having 6, 7 and 8 natural gas holes. Also plotted on the same graphs is the conventional diesel fueling data, called "baseline".

\(^2\) Jet attachment to the combustion chamber upper wall because of pressure differential between the upper and lower side
The results obtained with "6-7" and "6-8" injector tips can be summarized as follows:

- By using 7 and 8 gas holes, the engine stability was dramatically improved. The 7 hole tip gave better stability results than the 8 hole one. This is a surprising result in light of the theory presented above. Mathematics recommended the 8 hole tip as the one with best stability potential, because of its constant average interlace angle during gas needle rotation. Experimental results though, suggest a different mechanism: what matters is how many pairs of jets are aligned at one time, and not just the average interlace angle. These act as a combustion accelerator, which explains why the 8 hole injector, with 2 pairs of jets aligned simultaneously, behaves more unstably.

- Thermal efficiency obtained with the 6-7 and 6-8 tips is similar to that of the 6-6 injector.
Figure 4.3.5: THC emissions from HPDI fueling using injectors with 6, 7 and 8 gas holes; 6 pilot diesel jets

- With both new injector tips, the large variability in NO\textsubscript{x} emissions recorded with the prototype injector was eliminated. Moreover, the 6-7 and 6-8's levels are low, similar in magnitude and below the average trendline going through the emission points of the 6-6 injector. This is a very important result, because it shows that by choosing a different number of gas holes, not only was the variability eliminated, but also the level of emissions significantly reduced. The decrease in NO\textsubscript{x} obtained with the 6-7 and 6-8 injectors is on the order of 25%, much greater then the 5 % uncertainty in emission measurement. No explanation could yet be found for this significant NO\textsubscript{x} reduction.

- Emissions of unburned hydrocarbons are low and similar to those of the 6-6 injector. They consist mostly of methane.

- Carbon monoxide emissions from the 6-6 and 6-7 hole tips are similar in magnitude and significantly lower then for the diesel baseline. At high load (4 bar BMEP) the 6-
8 hole tip produces greater CO emissions than the 6-6 and 6-7 injectors. An explanation for this may be that having 8 jets rather than 6 or 7, the natural gas mixing with air will be affected, due to a closer interaction between fuel plumes.

**Figure 4.3.6**: CO emissions from HPDI fueling using injectors with 6, 7 and 8 gas holes; 6 pilot diesel jets

The experimental results prompted a tip geometry change from 6 to 7 natural gas holes. This modification eliminated the engine instability and reduced the variability in emissions. All following optimization experiments have been performed with this design.
4.4. Summary

- Large experimental variations in NO\textsubscript{x} emissions (especially at low load operation), engine operational instability and combustion rate changes, obtained with an HPDI injector having 6 diesel and 6 natural gas jets, have been attributed to changes in interlace angle. These results suggest the necessity to control and optimize this physical parameter.

- Numerical modeling simulations have proven that, at low load and speed, the interlace angle has significant impact on combustion rate and NO\textsubscript{x} emissions. The smaller the interlace angle (i.e. the closer the jets), the higher the combustion rate and the greater the NO\textsubscript{x} production.

- Experimental studies with different interlace angles should be performed to confirm the numerical modeling results.

- "6-7" and "6-8" injector tip designs have been tested. The "6-7" produced best stability-emissions results and was chosen to replace the prototype 6-6 injector tip for the rest of the analysis.

- Changing the number of gas holes from 6 to 7 and maintaining 6 diesel pilot jets, the variability in NO\textsubscript{x} emissions has been eliminated and the engine stability substantially improved. Moreover, the emissions of NO\textsubscript{x} have been reduced by up to 30 % with respect to the 6-6 HPDI injector. This is considered to be the principal finding of this research.
5. GAS INJECTION PRESSURE

With HPDI fueling of diesel engines, gas injection pressure is a factor of great importance to the combustion process because, together with injector hole area, it determines the rate at which natural gas is injected in the cylinder. The purpose of this chapter is to show the effect that variations in gas injection pressure can have on the emissions and efficiency of a high pressure directly injected natural gas fueled engine. Special attention is dedicated to NO\textsubscript{x}.

5.1. Introduction

The diesel combustion process is composed of three phases: premixed combustion, mixing-controlled combustion and late combustion. In the premixed phase, the fuel injected during ignition delay which has mixed with air to combustible limits, burns rapidly as in a spark ignited engine. The next phase is diffusive burning, which proceeds only as fast as the fuel is mixed with the air in the cylinder. Finally, in the late combustion (after the end of injection), a small fraction of fuel which escaped the first two burning phases, together with soot and fuel-rich combustion products, will now be oxidated as more air becomes available.

Diesel engine efficiency can be increased by having faster combustion, which will produce a thermodynamic cycle closer to the ideal air standard (where most combustion proceeds at constant volume and the remaining at constant pressure). Since most of the combustion takes place during the mixing-controlled phase, one way to obtain high efficiency is by having a fast injection. The faster the injection, the faster the burning, as will be shown.

The combustion mechanism of HPDI fueling is the same as for conventional diesel engines. Hence, high thermal efficiency can be achieved through fast combustion, which requires rapid injection of natural gas. The prototype HPDI injector uses constant gas
injection pressure (GIP), and was designed to operate choked for all engine operating conditions. It thus has mass flow rate constant during injection and proportional to GIP and nozzle area ([11] and [22]). The higher the gas pressure, the faster the combustion. A high injection pressure is also expected to produce more turbulence, hence better mixing. Gas injection pressure has significant influence on nitrogen oxide production as well. Fast diffusive burning due to high natural gas pressure will produce increased flame temperature and therefore increased NO\textsubscript{x}.

But a high injection pressure can also be detrimental because the gaseous jets can over-penetrate the combustion chamber. This can result in jet impingement on the wall, flame quenching, and increased heat transfer losses because combustion proceeds around the periphery of the cylinder. With jet over-penetration more fuel can get trapped in the piston ring crevices, resulting in high unburned hydrocarbon emissions.

From the view point of the work needed for natural gas compression, there is incentive to have a low gas pressure. A higher GIP increases compression work, which reduces the engine mechanical efficiency. However, insufficient injection pressure can result in poor mixing of the gas with air, slow combustion and low thermal efficiency. Gas injection pressure effects may also depend on the value of other parameters. Jet penetration is a function of GIP, but also of the hole size. The mixing with air depends on injection pressure, and on the number of jets. In order to separate the effects of gas pressure from the influence of other parameters, the remaining variables have been kept constant during experiments.

All tests were performed with an injector tip having 6 diesel and 7 natural gas jets, both equally spaced around the circumference. This arrangement was chosen because it provided best operational stability and data repeatability, as presented in chapter 4. Tests were done at steady state, for loads varying from 0 to 4 bar BMEP (maximum limit for the diesel fueled engine) and constant speed (1200 rpm). At each one of the test points, BOI was adjusted such that the start of combustion occurs at TDC. The reason for doing this was to obtain a similar combustion pattern, such that meaningful comparisons
can be made between the efficiency and emissions of different gas pressures tests. It was found that, for the same load and speed conditions, BOI did not vary with pressure by more than 0.3 deg CA.

5.2. Thermal Efficiency

Figure 5.2.1 displays the experimental pressure development in the combustion chamber as a function of crank angle after top dead center (ATDC), for HPDI fueling with three gas injection pressures. An increase in slope during the burning phase indicates that combustion rate increases with natural gas pressure. One would therefore expect that thermal efficiency should also increase with injection pressure throughout the range chosen for experiments.

![Figure 5.2.1: Combustion chamber pressure development for HPDI with 3 gas injection pressures; 4 bar BMEP](image)
Figure 5.2.2 illustrates experimental results of thermal efficiency variation with load, obtained using HPDI with three injection pressures. On this graph and throughout the rest of the thesis, HPDI is compared with the "baseline", which represent results obtained from the same engine running on conventional diesel fueling. The x axis displays brake mean effective pressure (BMEP), which is a measure of the engine load.

Before explaining the experimental results, a few comments about the error in thermal efficiency need to be made. At maximum load, the net experimental uncertainty in thermal efficiency due to instrument error (derived in chapter 3) is ±1.8 % absolute. As can be seen in Fig. 5.2.2, the variability in efficiency measurements is as large as 1.5 % absolute, of similar magnitude with the cumulated instrument error. The variability can be attributed either to changes in combustion rate caused by gas needle rotation, or to measurement error. To deal with this scatter, best curve fits through the data points have been introduced, and evaluation will be made analyzing their trendlines. There is in the
HPDI data evidence of some periodic variation even with uneven hole number. Also, when comparing diesel with HPDI data one must consider the greater error in natural gas measurements. Appendix 5 includes typical results of data repeatability for thermal efficiency and exhaust emissions.

Figure 5.2.2 shows that over the range of experimental injection pressures, the engine running on HPDI fueling produces thermal efficiencies equal to or greater than those of the baseline diesel.

To facilitate understanding of the results, a detail of the thermal efficiency plot has been given in Fig. 5.2.3. At maximum load and gas injection pressures of 130 and 160 bar, the efficiency curves have a positive slope, as opposed to the 100 bar and diesel baseline, which level off. This suggests that the HPDI fueled engine can still run efficiently at greater loads. Thermal efficiency obtained using a GIP of 100 bar is identical to that of

![Thermal efficiency variation with load for HPDI with 3 injection pressures and diesel baseline; upper load range](image)
the baseline diesel (within the accuracy of the gas measurements), but significantly lower at high load than the data taken at higher GIP. This is an indication of poor air utilization, which means that the mixing process starts to suffer because of lower penetration of the gas jets.

Data shows that with 160 bar gas injection pressure, efficiencies above 2 bar BMEP are equal to those obtained at 130 bar, whereas in the low load range efficiencies are below the 130 bar levels. This result suggests that, for this engine, there is an optimum injection pressure around 130 bar which gives maximum thermal efficiency. It is believed that for gas pressures above 130 bar over-penetration of the jets causes reduction in efficiency.

It is important to note that HPDI fueling requires compression of the natural gas. In these tests the gas was supplied from a high pressure tank, and therefore the results presented do not include the power absorbed by an on-board compressor, which would produce a penalty in efficiency. However, thermal efficiency could be reduced by at most 3 % (relative) when using a natural gas compressor\(^1\), which does not change significantly the experimental results.

### 5.3. Emissions

Carbon monoxide emissions are an indication of the degree of completion of a combustion process. Since diesels always run lean, when the engine produces CO, it is because locally the fuel did not find the air necessary for complete oxidation. That reflects poor quality of the mixing process between fuel and air.

In conditions of local oxygen scarcity, a diffusive flame, such as that made by diesel jets, produces incompletely burned hydrocarbons which will bond with each other to form the soot. This phenomenon is associated with an increase in CO emissions (of an order of magnitude or so), because of the incomplete combustion process.

---

\(^1\) The natural gas compressor absorbs approximately 3 % of the power at 120 kW.
In the absence of soot measurements, CO emissions can be used to qualitatively indicate the regime where smoke production reaches unacceptable levels. The load at which CO emissions have an exponential increase will be referred to as "the smoke limit". When diesels reach their smoke limit, even though globally they run lean, no increase in the engine load can be achieved, because of local oxygen scarcity in conditions of limited mixing. Any further addition to the amount of fuel injected will only end up as smoke in the exhaust.

The maximum load chosen for experiments was dictated by the performance and emissions of the engine running on conventional diesel fueling. 4 bar BMEP is the maximum load achievable before smoke emissions reached unacceptable levels. When running on HPDI fueling, the engine could reach higher loads without significant increase in CO emissions, suggesting greater load capability of HPDI over conventional diesel.

Figure 5.3.1: HPDI's CO emission variation with load for 3 injection pressures relative to the diesel baseline
diesel. This is possibly due to improved mixing of natural gas with air.
The carbon monoxide emissions obtained with HPDI fueling and shown in Fig 5.3.1 confirm these results. At 4 bar BMEP and injection pressures of 130 and 160 bar GIP, CO emissions are essentially unaffected by load (around 60 ppm), whereas with diesel fueling they reached 1400 ppm. When using 100 bar GIP, lower jet momentum leads to reduced mixing with air which produces a CO increase at maximum load. The HPDI emissions are still lower though than those of the baseline diesel.

Figure 5.2.1 shows that an increase in injection pressure produces visible acceleration of the burning process. The higher injection rate will eventually lead to an increased combustion rate in the mixing-limited phase, greater flame temperatures, and thus NO\textsubscript{x} emissions are also expected to rise. This trend is confirmed by the experimental results presented in Fig 5.3.2, which reveal an increase in NO\textsubscript{x} with GIP throughout the load range tested.

![Figure 5.3.2: HPDI's NO\textsubscript{x} emission variation with load for 3 injection pressures relative to the diesel baseline](image-url)
However, the data taken with HPDI fueling has a variability (25 ppm) equal in magnitude to the difference between results obtained with the 3 injection pressures. This variability is 2.5 times greater than the maximum net uncertainty (10 ppm) in NO\textsubscript{x} measurements (Table 3.4.1) and, as in the case of thermal efficiency, is believed to be caused by gas needle rotation. The trends shown in Fig. 5.3.2 are thus believed to be true, because the variability in NO\textsubscript{x} production caused by needle rotation affects in equal manner the results obtained with different injection pressures.

This results of Fig. 5.3.2 show that using HPDI, a reduction in NO\textsubscript{x} of up to 40 % over the diesel baseline can be obtained, when the start of combustion occurs at TDC. Such decrease can be due to the lower adiabatic flame temperature of the natural gas, as well as to the split combustion characteristic. This method of comparison though does not reveal HPDI's full potential for NO\textsubscript{x} reduction, which can be further improved through timing.

![Figure 5.3.3: HPDI's THC emission variation with load for 3 injection pressures relative to the diesel baseline](image-url)
control. A parallel analysis of the NO\textsubscript{x}-efficiency trade-off and timing delay NO\textsubscript{x} reduction will be discussed in the next chapter. Figure 5.3.2 also indicate a slower increase in NO\textsubscript{x} when using HPDI as opposed to diesel fueling. Since NO\textsubscript{x} is very sensitive to temperature, the lower adiabatic flame temperature of the natural gas is probably the cause of this.

The low emissions of unburned hydrocarbons (35-70 ppm) presented in Fig 5.3.3 indicate that for all injection pressures used in testing, both fuels burn almost to completion (99.85 and 99.03 % for diesel and natural gas, respectively).

The levels measured are the same magnitude as those with diesel fueling, but Fig. 5.3.4 shows they are mainly composed of methane, which is unregulated. A slight increase in unburned CH\textsubscript{4} with decreasing load is present in all cases, and is possibly due to the lower temperature regime. Methane emissions seem to increase with injection pressure, but a firm conclusion cannot be drawn because the differences obtained between the 3

![Figure 5.3.4: HPDI's CH\textsubscript{4} emission variation with load for 3 injection pressures relative to the diesel baseline](image)

77
data sets are within experimental error. Fig. 5.3.4 indicates that diesel combustion produces constant methane emissions of approx. 40 ppm.

Figures 5.3.1 through 5.3.4 displayed emission results in scattered form. The variability of all measurements is larger than the net experimental uncertainty, and is attributed to changes in combustion rate, which are produced by gas needle rotation. Since diesel is believed to be a major contributor to HPDI's soot and NOx emissions, it is important to know how much of the energy content comes from the pilot. Figure 5.3.5 shows the diesel ratio (energy percentage) variation with engine load for the three injection pressures tested.

![Figure 5.3.5: HPDI's diesel ratio variation with load for 100, 130 and 160 bar gas injection pressure](image)
5.4. Summary

- Over the gas pressure range chosen for study (100-160 bar), combustion rate increased with gas injection pressure.

- Within the level of accuracy specified in chapters 3 and 5, it appears that for the given research engine, there is an injection pressure around 130 bar for which thermal efficiency is maximized. At 130 bar GIP, the efficiency was greater (by up to 5 % of its magnitude) over the entire load range. It is expected that this optimum may change for a different nozzle area or tip geometry. Over the range of 130 to 160 bar, there seems to be little sensitivity of thermal efficiency to gas injection pressure.

- NO\textsubscript{x} increases with gas injection pressure throughout the load range. By reducing the gas pressure from 160 to 100 bar, a decrease in NO\textsubscript{x} emissions of up to 25 % could be obtained. With HPDI, 130 bar gas pressure and un-optimized beginning of injection, a 40 % reduction in NO\textsubscript{x} has been obtained over the diesel baseline.

- The lowest NO\textsubscript{x} emission has been obtained with an injection pressure of 100 bar, whereas the best efficiency has been given by a gas natural pressure of 130 bar. A compromise is thus necessary.

- Over the entire range of load and gas pressures, the research engine produced low unburned hydrocarbon emissions (35-70 ppm) and similar to those of the diesel baseline. They consisted mostly of methane (55-70 ppm), which is un-regulated.

- At gas injection pressures chosen to maximize efficiency, carbon monoxide emissions were also low (40-90 ppm) and un-affected by load (in the range of study). When using HPDI fueling, the engine could run at 25 % higher load compared to conventional diesel, without severely raising CO emissions.
6. ABSOLUTE INJECTION TIMING

The purpose of this chapter is to explore the effects of changing absolute injection timing, while relative beginning of injection is maintained constant.

6.1. Introduction

Absolute injection timing (defined in chapter 1) represents the beginning of injection, as given by the electronic command signal. It was called "absolute" to distinguish it from the relative injection timing, which in HPDI fueling represents the time (or crank angle) delay between the pilot injection and that of the natural gas.

NO\textsubscript{x} is one of the major pollutants produced by diesel engines. Therefore, one motivation for undertaking this study was to determine the HPDI's potential for NO\textsubscript{x} reduction through timing delay, while maintaining or increasing conventional diesel efficiency.

In diesel engines, NO\textsubscript{x} emissions decrease continuously with delaying injection. A later injection shifts combustion in the expansion stroke, which produces lower combustion temperatures. Conversely, thermal efficiency is maximized on a certain region of beginning of injection. For too advanced timings, combustion proceeds in the compression stroke, producing negative work. Later injection gives long combustion duration, which means departure of the cycle from the ideal air standard; thus, thermal efficiency will be affected. In consequence, timing delay NO\textsubscript{x} reduction techniques, often used by diesel engine manufacturers, are limited by thermal efficiency losses. For diesels, this method has limitations because it also comes with an increase in particulate emissions, as shown in Fig. 1.1.1. This increase is caused by the fact that with late injection, soot oxidation proceeds at lower temperatures, because combustion pressure and temperature level will be diminished by the piston expansion movement. With delayed injection there is less time available for combustion (due to reduction in the time between BOI and exhaust valve opening), and incompletely burned hydrocarbons could escape the cylinder in the form of soot.
The lower carbon-to-hydrogen ratio of the methane (main component in natural gas) molecule and better mixing properties of a gas, suggest that compressed natural gas could have potential to produce lower particulate emissions than diesel fuel. Faster mixing between gas and air will reduce ignition delay. Hence, this is expected to increase the limits of efficient burning, and enable complete soot oxidation even when using delayed injection. HPDI fueling with natural gas has already demonstrated lower particulate emissions than conventional diesel, as shown in [12] and [14]. Therefore NO\textsubscript{x} reduction by timing delay could be a method with increased potential because of its greater flexibility.

The following sections will present results of the influence of changes in injection timing on thermal efficiency and emissions. Relative beginning of injection was kept constant, and its selected value determined prior to this study. Tests were performed by varying beginning of injection (BOI) with increments of 3 degrees crank angle (CA), at loads of 1 and 3 bar BMEP, and a constant speed of 1200 rpm. In all plots, BOI is measured in crank angle degrees before top dead center (BTDC). Injection pressure was constant for all tests (130 bar). The experimental test matrix is presented in Table 6.1.1.

**Table 6.1.1**: Experimental test matrix; 130 bar gas pressure, 3 tests for each load

<table>
<thead>
<tr>
<th>BMEP (bar)</th>
<th>BOI (deg. BTDC)</th>
<th>RBOI</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0,3,6,9,12,15,18</td>
<td>Long</td>
</tr>
<tr>
<td>3</td>
<td>0,3,6,9,12,15,18</td>
<td>Long</td>
</tr>
</tbody>
</table>
6.2. Thermal Efficiency and NO\textsubscript{x} Emissions

Figures 6.2.1 and 6.2.2 display thermal efficiency and NO\textsubscript{x} emissions, respectively, for HPDI and diesel fueling, as a function of the absolute beginning of injection (BOI). At 3 bar BMEP and HPDI fueling, Fig. 6.2.2. shows a decrease in NO\textsubscript{x} emissions with delaying injection up to a BOI of 6 deg BTDC. Beyond this timing, no further reduction in NO\textsubscript{x} can be obtained. Incidentally, this is also the delay limit for maximum thermal efficiency, as displayed in Fig. 6.2.1.

The diesel engine NO\textsubscript{x} emissions continuously decrease up to a BOI of 3 deg BTDC. However, past 9 deg BTDC any further NO\textsubscript{x} reduction is made with a sacrifice in efficiency.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure6.2.1.png}
\caption{HPDI and diesel baseline's thermal efficiency variation with crank angle BTDC for 2 load conditions; 130 bar gas injection pressure}
\end{figure}

\footnote{Will be defined in section 7.1, chapter 7.}
Low load operation suggests even greater flexibility of HPDI, with constant thermal efficiency over a BOI range of approximately 12 degrees. NO\textsubscript{x} emissions decrease with injection delay for the entire BOI range. The limit is hence dictated by efficient operation. These cases suggest that one limit on delay could be set by the best trade-off between thermal efficiency and NO\textsubscript{x}. But delaying injection to reduce NO\textsubscript{x} also comes with an increase in particulate emission. Since smoke emissions could not be measured during this research, an evaluation of the delay limits could only be done considering thermal efficiency. However, an overall optimum should be sought considering all three variables, and also making sure that emissions of unburned hydrocarbons and carbon monoxide won't rise beyond regulated limits.
With HPDI fueling, both load cases display a region of constant thermal efficiency greater than for conventional diesel. The regime of high efficiency is increased towards delayed injection timings. Additional processes of atomization and evaporation necessary for a liquid fuel, make diesel combustion proceed into the expansion stroke for more advanced timings. This is possibly the reason for an "earlier" efficiency drop experienced by conventional diesel fueling. At both loads tested, HPDI has proven potential for NO\textsubscript{x} reduction through delaying fuel injection some 3 deg. with respect to the baseline diesel, without significantly affecting efficiency.

For both kinds of fueling and load cases, NO\textsubscript{x} decreases with delaying injection. Diesel however decreases with a steeper slope than HPDI, which seems to be less sensitive to BOI. The adiabatic flame temperature difference between diesel and natural gas and the split combustion pattern of HPDI can be the cause of these discrepancies. At low load, the reduction in NO\textsubscript{x} with BOI proceeds at similar rate for both fueling methods, possibly because with HPDI, the percentage of diesel in the total amount of fuel injected is much higher (around 45%).

For the higher load condition, delaying injection past a certain limit did not produce any further NO\textsubscript{x} reduction. An explanation for this could be as follows: the shifting of combustion too far in the expansion stroke, possibly led to "freezing" of the NO\textsubscript{x} depletion process due to lower temperatures. In the low load case instead, NO\textsubscript{x} drops continuously with decreasing BOI. In Fig. 6.2.2, one can see that with HPDI fueling, NO\textsubscript{x} emissions at 3 bar BMEP are lower than those of conventional diesel at 1 bar BMEP. The results obtained confirm the expected potential for NO\textsubscript{x} reduction with late injection without compromising engine efficiency.

HPDI's potential for NO\textsubscript{x} reduction through timing control is better emphasized by plotting brake specific fuel consumption (BSFC) as a function of NO\textsubscript{x}, for both HPDI and diesel fueling. The graphs displayed in Fig. 6.2.3 correspond to the data sets presented in Figs. 6.2.1 and 6.2.2.
Figure 6.2.3: NO\textsubscript{x} vs BSFC trade-off for HPDI and conventional diesel fueling; 130 bar gas injection pressure

The 3 bar BMEP load curve shows that 55 % NO\textsubscript{x} reduction is possible by using HPDI, while maintaining the diesel baseline specific fuel consumption. Data at 1 bar BMEP indicates a similar trend, with a 60 % decrease in NO\textsubscript{x} at a BOI of 3 deg BTDC, and with equivalent diesel fuel consumption. Therefore, by using natural gas in this diesel engine, there is potential for reduction in both NO\textsubscript{x} and specific fuel consumption.
6.3. CO and Unburned Hydrocarbon Emissions

While the major concern of this study is reducing NO\textsubscript{x} and preserving efficiency, one needs to make sure that delaying timing, the emissions of total hydrocarbon and carbon monoxide won't be affected. Figures 6.3.1 and 6.3.2 display total hydrocarbon (THC) and methane emissions, respectively. Within the BOI limits defined by the NO\textsubscript{x}-efficiency trade-off (3 and 6 deg. BTDC for 1 and 3 bar BMEP respectively), the HPDI unburned hydrocarbon levels remain very low (55-100 ppm) and consisting mostly of methane (55-100 ppm), which is unregulated.

![Figure 6.3.1: HPDI and diesel baseline's THC emission variation with crank angle BTDC for 2 load conditions; 130 bar gas injection pressure](image)

At 1 bar BMEP and no injection advancement, both types of fueling experience an increase in unburned hydrocarbons. This is possibly caused by a decrease in combustion temperature, in conditions of an adverse piston expansion movement. Another explanation can be as follows: the combustion pressure trace (Fig. 6.3.3) shows an
excessive ignition delay; hence, having a shorter time span available, the fuels cannot burn to completion. Part of the unburned hydrocarbon emission can also come from the fuel that has mixed beyond flammability limits during the long ignition delay period.

Figure 6.3.2: HPDI and diesel baseline's CH₄ emission variation with crank angle BTDC for 2 load conditions

However, HPDI produces only 25% of the total hydrocarbon emissions measured from the combustion of diesel, under these conditions. This reduction can be due to the difference in physical properties between fuels. At low load, the limit of BOI delay necessary for complete combustion suggested by these plots is around 3 deg BTDC, the same as the one suggested for optimum thermal efficiency described above.
Figure 6.3.4 shows that at both loads, HPDI fueling produces low emissions of CO (40-70 ppm) throughout the experimental BOI range. Moreover, carbon monoxide levels remain practically unaffected by retarding injection, as opposed to those of the baseline diesel which suffer a severe increase. This result suggests better mixing properties of natural gas. Similar reasoning as that for high THC emissions could be the explanation for greater CO emissions with diesel operation and no injection advance.
6.4. Summary

- With HPDI fueling of a mono-cylinder diesel engine, injection timing delay was found to be an effective way of reducing NO\textsubscript{x} emissions. By appropriately adjusting beginning of injection, reduction of up to 60 % (of the diesel baseline emissions) could be obtained, while preserving diesel-like efficiency.

- Injection timing delay has proven to be a more effective mean of NO\textsubscript{x} reduction when using HPDI fueling of an experimental engine, as opposed to conventional diesel.

- Within the limits of efficient engine operation, HPDI's total hydrocarbon (55-100 ppm) and carbon monoxide (40-70 ppm) emissions were very low and almost unaffected by BOI.
7. RELATIVE INJECTION TIMING

7.1. Introduction

NO\textsubscript{x} is a major pollutant produced by conventional diesel engines and its reduction through the use of HPDI fueling represents an important goal of this research. Chapter 6 described how NO\textsubscript{x} emissions can be diminished through injection timing delay. In addition to this method, with HPDI fueling, another parameter can also contribute to NO\textsubscript{x} reduction: the time delay between pilot diesel and gas injection. The purpose of this chapter is to determine the sensitivity of HPDI’s emissions and efficiency to changes in relative beginning of injection (RBOI).

The heat release curve of a conventional diesel engine (Heywood [21]) has a high peak produced by combustion of the diesel which mixed with air to flammability limits during the ignition delay period. In this premixed burning phase significant amount of NO\textsubscript{x} is generated due to the high temperature. To lower the flame temperature and thus the NO\textsubscript{x} formation during this phase, researchers are investigating a split injection pattern which can reduce the amount of premixed burning.

With HPDI fueling, a long relative injection delay can also be advantageous because the split combustion lowers the chamber temperature, which will yield low levels of NO\textsubscript{x}. If the absolute beginning of injection is also delayed, burning will proceed late in the expansion stroke. This will lead to longer combustion and lower thermal efficiency, because of departure from the ideal air standard diesel cycle. A short RBOI instead may lead to quenching by the cold gaseous jets of the just igniting diesel sites, compromising the natural gas ignition source and therefore the rest of combustion. Associated with a delayed absolute BOI, a short RBOI could also have a positive effect. Late injection produces reduction in NO\textsubscript{x}. Short RBOI (but long enough not to quench the diesel sites) means rapid ignition of the gas, small premixed burning phase and possibly additional NO\textsubscript{x} reduction.
From a design standpoint, eliminating the relative beginning of injection (RBOI) would mean that:

- the two fuels could be mixed and injected together, or
- the diesel pilot could be simultaneously injected with the gas

Mixing the fuels could simplify injector design and increase reliability. Hence the motivation to explore the possibility of eliminating RBOI.

Three-dimensionality of the problem in question suggested a layout of the relative injection timing study of the following manner:

- Different pilot/gas spring combinations were tested to encompass a sufficiently large variation in RBOI. A minimum relative delay has been sought to examine the possibility of eliminating RBOI.
- Next, efficiency and emissions were analyzed across the load range, in a similar fashion as for the gas injection pressure study. Beginning of injection (BOI) was adjusted to obtain the start of combustion at top dead center (TDC), load was varied from 0 to 4 bar BMEP, and the engine speed maintained constant at 1200 rpm.
- Subsequently, choosing two different loads, BOI has been varied with three degree increments, to encompass the engine limits of efficient operation. This method was adopted to help determine a better RBOI and its changes with load.

With the current injector design, the time delay between injection of the pilot and gas is controlled by springs pre-loading two needles. Having a stronger pilot spring or a weaker gas spring will both result in a decrease in RBOI. However, this mechanical method of injection delay control has limited flexibility and only small changes could be obtained. In this study, two different RBOIs have been compared which, for the rest of the chapter, will be called "short" and "long".

In absence of an optical method for measuring RBOI, its magnitude can be approximated through analysis of three pieces of information: pilot vs. gas needle ratio of opening times, pilot vs. gas needle ratio of opening pressures, electronic pulse width (defined as the duration of injection in crank angle degrees), and combustion pressure trace. Analysis of the pressure variation inside the injector passages during needle opening time
has revealed a linear dependence with time, as shown in Fig. 7.1.1. Moreover, the pressure slopes before and after the pilot needle opening are very similar. Since measurements of these quantities could only be done at low speed, a dimensionless ratio between the opening time and pressure of the two needles could be used to estimate the RBOI magnitude. It was observed that, regardless of the injector operating speed, the ratio between the opening time for diesel and natural gas needles is always the same.

![Figure 7.1.1: Variation of hydraulic pressure inside HPDI injector passages before gas needle opening](image-url)

Therefore, calculating this ratio and using the total injector opening time measured by the engine controller, an absolute value of the RBOI can be deduced. In addition, the similarity of triangles OAC and OBD (Fig. 7.1.1) suggests that the ratio of opening pressures should be equal to the ratio of opening times of the two needles. In Fig. 7.1.1, shifting of the AC segment to the left will lead to an increase in RBOI.
Table 7.1.2 gives the pilot and gas needle opening time and pressures measured on the pop tester\(^1\), the electronic pulse width at 3 bar BMEP given by the engine controller, and the values of RBOI calculated under the assumption made above.

**Table 7.1.2: RBOI spring combination characteristics**

<table>
<thead>
<tr>
<th>RBOI</th>
<th>Pilot opening pressure (psi)</th>
<th>Gas opening pressure (psi)</th>
<th>Pilot opening delay (ms)</th>
<th>Gas opening delay (ms)</th>
<th>Electronic pulse width(^2) (CA deg.)</th>
<th>Ratio of opening times</th>
<th>Calculated RBOI (CA deg.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Short</td>
<td>4092</td>
<td>11630</td>
<td>4.19</td>
<td>9.06</td>
<td>6.80</td>
<td>0.56</td>
<td>3.7</td>
</tr>
<tr>
<td>Long</td>
<td>3900</td>
<td>12700</td>
<td>4</td>
<td>11</td>
<td>9.25</td>
<td>0.64</td>
<td>5.9</td>
</tr>
</tbody>
</table>

Taking the ratios of pilot to gas opening times, the RBOI of short and long delay represent 0.56 and 0.64 of the gas needle opening time, respectively. At 1200 rpm, this yields a short and long RBOI of 3.7 and 5.9 deg, respectively.

For ease of description and understanding of the RBOI's separate effects on load and BOI, this chapter has been subdivided in sections following the research layout presented in page 46. The experimental test matrix is given in Table 7.1.3.

**Table 7.1.3: RBOI study test matrix; 1200 rpm, 130 bar gas pressure, 3 data sets for each test condition**

<table>
<thead>
<tr>
<th>Test #</th>
<th>BMEP (bar)</th>
<th>BOI (deg. BTDC)</th>
<th>RBOI</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0 to 4</td>
<td>Combustion at TDC</td>
<td>Long</td>
</tr>
<tr>
<td>2</td>
<td>0 to 4</td>
<td>Combustion at TDC</td>
<td>Short</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>0, 3, 6, 9, 12, 15, 18</td>
<td>Long</td>
</tr>
</tbody>
</table>

\(^1\) Electro-mechanical instrument used for testing injector operation before installing on engines. The pop tester enables measurement of pressure inside the injector passages and the times of needle opening, using a pressure transducer signal recorded on the oscilloscope.

\(^2\) Recorded at a load of 3 bar BMEP and 1200 rpm.

93
7.2. Short Relative Injection Delay

The "short" relative delay selected for study represents the lowest RBOI limit of stable engine operation, and the results obtained were used to estimate the possibility of injecting simultaneously the pilot and gas.

At low load operation and "short" RBOI, significant effort was necessary to bring and maintain the engine running at steady state. Below this time delay value, the engine was impossible to run, because of speed variations as large as 500 rpm. One reason for this unstable engine behaviour could be the increased interaction between the cold gaseous jets and the diesel ones, which may have caused quenching of the pilot flame. However, a faulty injector operation could also have been the cause. The solenoid operated spool valve has a minimum recommended aperture time of 1 ms, and at no load operation the needle opening time is close to this limit. In consequence, the gas needle may not be able to close at the electronic command signal, and thus increase the amount of gas injected in cylinder. For constant load setting conditions, this would produce a speed increase and unstable engine behaviour.

This preliminary study suggests that injecting the diesel and gas simultaneously may not be a viable solution, and should be the subject of another study. For the present time, this question remains un-answered. Mixing the two fuels prior to injection is another question which requires investigation.
7.3. Load dependence

In addition to the methods described in section 7.1, a good indication of the difference in RBOI is given by comparison of the pressure curves. Figure 7.3.1 shows the pressure development with crank angle degrees, for an HPDI injector with long and short RBOI operating at 4 bar BMEP, 1200 rpm and 130 bar gas injection pressure (GIP). In order to obtain a similar combustion pattern, BOI was adjusted such that the start of combustion (separation of the pressure curve from motoring one) occurred at top dead center (TDC). At identical conditions BOI did not vary by more then 0.4 deg. In the long RBOI test case of Fig. 7.3.1, BOI was 0.1 deg. crank angle greater then for the short one.

![Combustion pressure development for an HPDI fueled engine running with short and long RBOI](image)

**Figure 7.3.1:** Combustion pressure development for an HPDI fueled engine running with short and long RBOI
Figure 7.3.1 indicates that:

- The pilot diesel ignition delay is not affected by the natural gas injection. With both RBOIs, the peak due to the pilot combustion occurs at the same instant and has the same magnitude.
- The relative ignition delay is indicated by the distance between the diesel peak combustion and the start of gas combustion. A longer plateau in the pressure trace for the "long" RBOI case is a consequence of the difference in RBOI's.
- Duration of combustion is longer in the case of long RBOI. This is indicated by a later peak pressure belonging to natural gas combustion. In addition, the maximum pressure lowers in magnitude.

The following three graphs represent the variation with load of thermal efficiency, NOx and CH4 emissions respectively.

**Figure 7.3.2:** HPDI's thermal efficiency variation with load for long and short RBOI, relative to baseline diesel
Fig. 7.3.2 shows that a longer RBOI resulted in an improved thermal efficiency. Since Fig. 7.3.1 revealed a longer combustion duration for the "long" RBOI case, one would have expected a lower thermal efficiency. Longer combustion means departing from the ideal air standard diesel cycle, and thus efficiency should decrease. The experiments do not confirm this expectation. No satisfactory explanation could yet be found for these results. A more erratic behaviour of the engine running with short RBOI suggested an increased interaction between the pilot flame and the cold gas jets. However, Fig. 7.3.1 indicates no change in the pilot's ignition delay and combustion, thus not confirming this hypothesis.

The "short" relative delay of these data sets represents the lowest limit of stable operation (with small cycle to cycle variations). With "long" RBOI, the engine had smoother running throughout the load range.

Figure 7.3.3: HPDI's NO\textsubscript{x} emission variation with load for long and short RBOI, relative to baseline diesel
The results of Fig 7.3.2 suggest that what is believed to be our "long" RBOI is probably more a medium one. Testing an injector with greater RBOI flexibility would eventually prove a decrease in thermal efficiency when using a longer relative injection delay. From the thermal efficiency stand point, an optimum RBOI seems to be somewhere in the mid-range, but its value should be the subject of another study.

As explained in the previous paragraphs, a short RBOI would yield a faster combustion and thus a higher flame temperature. This should result in an increased NO\textsubscript{x} production. Fig 7.3.3 confirms this fact: with short RBOI the combustion is faster and the levels of NO\textsubscript{x} are higher across the load range.

![Graph showing THC emission variation with load for long and short RBOI, relative to baseline diesel.](image)

**Figure 7.3.4:** HPDI's THC emission variation with load for long and short RBOI, relative to baseline diesel

When using a long RBOI, experiments show a reduction in NO\textsubscript{x}, which is more significant at high load. An explanation for this could be given by the fact that at low
load the percentage of diesel is higher. With its greater adiabatic flame temperature, diesel is probably the cause for most NO\textsubscript{x} produced at low load.

Figures 7.3.4 and 7.3.5 show very low levels of unburned hydrocarbons with both RBOIs. However, the "short" delay produced up to 30 % greater emissions. The low levels obtained suggest that having a short delay does not disturb the pilot flame or ignition of the gas to the point of having sensible unburned hydrocarbon emissions. The slightly greater emissions seen in the case of short RBOI can be due to an increase in the amount of natural gas that mixed beyond flammability limits before ignition. Since with short RBOI the gas is injected closer (temporally) to the pilot, more CNG will mix beyond flammability limit, and result in unburned hydrocarbon emissions.

![Graph showing CH\textsubscript{4} emission variation with load for long and short RBOI, relative to baseline diesel](image)

**Figure 7.3.5:** HPDI's CH\textsubscript{4} emission variation with load for long and short RBOI, relative to baseline diesel
7.4. BOI Dependence

In order to determine RBOI effects across the entire load range, data of section 7.3 was taken by adjusting BOI to obtain beginning of combustion at top dead center. Hence, using a similar burning pattern, meaningful analysis could be made. But the optimum conditions for efficiency and emissions may not be achieved when combustion occurs at TDC, and thus the sensitivity of these variables to changes in BOI was also analyzed.

"BOI dependence" of the RBOI study was performed at two different loads (a low and a medium one), constant speed and injection pressure (Table 3.6.1). Results obtained at low load will be presented in this section, and the remaining data is given for reference in Appendix 6.

![Figure 7.4.1: HPDI's thermal efficiency variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP](image)
Figure 7.4.1 shows that a "long" RBOI produces higher thermal efficiency across the BOI range. With HPDI fueling, both RBOIs display greater operational flexibility due to a larger constant thermal efficiency range than for conventional diesel fueling. However, the short relative delay case produced lower efficiency than diesel for a significant part of the BOIs tested. This is one of the reasons why the long RBOI spring combination has been chosen for performing the remaining experiments.

Figure 7.4.2 represents the variation of NO\textsubscript{x} emissions with BOI. For both load cases (Fig. A6.2), the long RBOI produced lower emissions throughout the BOI range. This result was expected because the lower combustion rate (shown in Fig. 7.3.1) possibly decreased the flame temperature and therefore NO\textsubscript{x}.

![Graph showing NO\textsubscript{x} emissions variation with BOI](image)

**Figure 7.4.2:** HPDI's NO\textsubscript{x} emission variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP

The following graphs illustrate the variation with BOI of the total hydrocarbon and methane emissions. Data (Figs. A6.3 and A6.4) indicates that at higher load (3 bar
BMEP), injection advance can be eliminated completely, without having a significant increase in unburned hydrocarbons. The levels measured in both RBOI tests are low (55-75 ppm), and similar to those produced by conventional diesel engines.

In the case of 1 bar BMEP and no injection advance instead, the short RBOI experimental data reveals a sharp increase in hydrocarbon emissions, as well as carbon monoxide (Figs. 7.4.3, 7.4.4 and 7.4.6 respectively). However, these high emissions are not of concern, since they belong to a regime outside the efficient engine operation (with 0 deg. injection delay, Figs 7.4.1 and 7.4.2 indicated a sharp drop in thermal efficiency, and no significant NOx reduction past 3 deg. BTDC, respectively). Throughout the rest of the BOI regime, both RBOIs produced very low emissions of unburned hydrocarbons. Methane seems to be predominant, representing about 80% of the THC.

*Figure 7.4.3: HPDI's THC emission variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP*
The difference between the results obtained with long and short RBOI (low load case) is very intriguing and, in the interest of gaining understanding of the combustion physics (or chemistry), an explanation will be attempted. It is however a qualitative interpretation, because important pieces of information like the relative timing between injection and ignition delay are unavailable. CFD simulations or optical measurements of the combustion process could give better interpretation of these results, but have not been performed during this study.

Information given by combustion pressure development (Fig. 7.4.5) and injection pulse width was used in explaining the high hydrocarbon emission at low load and no injection advance, given by the short RBOI. In conditions of extremely retarded injection, ignition delay has increased substantially. The combustion pressure traces taken for both RBOIs indicate the following:
• Ignition delay of the pilot diesel (and therefore of combustion) was not affected by the injection of natural gas (as previously shown by Mtui and Hill [13]).

• The start of combustion occurred around 18 deg. ATDC for both RBOIs. The total injection duration measured by the engine controller was 10 and 13.5 deg. CA for the short and long RBOI respectively. Because the start of combustion occurred after the end of injection, the natural gas had a very long time for mixing, and possibly a good part of it mixed with air beyond flammability limits. Since with long RBOI the injection of gas occurred later, smaller amounts of it had time to mix beyond flammability. The difference in unburned hydrocarbon emissions from short and long RBOI, can possibly be explained by the difference in time available for mixing, before the start of combustion.

![Figure 7.4.5: HPDI's combustion pressure development for long and short RBOI, and zero injection delay](image)

Data (Fig. A6.5) taken at higher load, show that both RBOIs produce low (30-100 ppm) and similar carbon monoxide emissions. With delayed injection timings, they are much...
lower than those produced by conventional diesel fueling (700 ppm). As explained in section 6.2, the lower time necessary for mixing of natural gas with air is possibly the reason for a more complete combustion.

![Graph showing CO emission variation with BOI](image)

Figure 7.4.6: HPDI's CO emission variation with BOI for long and short RBOI, relative to baseline diesel; 1 bar BMEP

At low load, in the BOI range of efficient operation, CO emissions (shown in Fig. 7.4.6) are also low (5-75 ppm) for both RBOIs. With no advance, the short delay produces higher CO emissions. This is possibly due to an incomplete oxidation of the unburned hydrocarbons which escaped the main combustion. Although increased, the CO production with short RBOI (400 ppm) is still lower than that of conventional diesel (900 ppm).

Section 6.2 suggested that a better way of evaluating HPDI's NO\textsubscript{x} reduction potential through timing delay is to plot experimental data in the form BSFC (brake specific fuel
consumption) vs. NO\textsubscript{x}. These curves are generated by unifying data points corresponding to different injection timings. The left side corresponds to no injection advance.

Fig 7.4.7 indicates that at low load, HPDI with short RBOI produced lower emissions of NO\textsubscript{x} than conventional diesel, but sacrificed fuel consumption. At higher load (Fig. A6.6), there is little potential for NO\textsubscript{x} reduction when using a short RBOI. HPDI fueling with "long" RBOI, gave reduction in NO\textsubscript{x} of up to 60 % over the diesel baseline, while preserving its low specific fuel consumption.

![Figure 7.4.7: HPDI's BSFC vs NO\textsubscript{x} curves with long and short RBOI, relative to diesel baseline; 1 bar BMEP](image-url)
7.5. Summary

- It appears that eliminating RBOI (simultaneously injecting the diesel pilot and natural gas) may not be a viable solution, because an overly short time delay before the gas injection may possibly quench the pilot diesel ignition sites. However, further studies need be performed to determine whether the engine instability recorded is due to combustion phenomena or faulty injector behaviour.

- HPDI fueling using a longer RBOI produced higher thermal efficiency (up to 13% of its magnitude), across the load and BOI ranges tested, than the case with shorter relative injection delay. This is an unexpected and yet unexplained result. Over the same operating range, emissions of NO\textsubscript{x} from the injector with longer RBOI were as much as 25% lower and the engine stability improved.

- In the range of efficient engine operation, both RBOIs tested produced low emissions of unburned hydrocarbons (40-120 ppm) and carbon monoxide (30-80 ppm), similar to those of conventional diesel.

- The results obtained in this study suggested continuation of the research using the spring combination that gave a longer RBOI.

- Optimization studies should be performed with an HPDI injector having more flexible RBOI control, and which enables the possibility to quantify precisely the time delay magnitude.
8. CONCLUSIONS AND RECOMMENDATIONS

8.1. Conclusions

Measurements of performance and emissions of a Diesel engine fueled with natural gas have been made using high-pressure direct-injection (HPDI). With HPDI technology, natural gas is injected late in the compression cycle preceded by an injection of diesel pilot. The influence of several physical parameters has been investigated experimentally and coupled with 3-dimensional numerical simulation of the combustion process performed by other researchers, using a modified KIVA code.

Based on the results obtained at 1200 rpm and in the range of 0 to 4 bar BMEP on a two-stroke, mono-cylinder research engine, the following conclusions can be drawn:

- High pressure direct injection of natural gas has proven to be an efficient method of fueling for diesel engines, with significant potential for emission reduction. Using HPDI, the thermal efficiency of a diesel engine was improved while decreasing NO\textsubscript{x} emissions by up to 40 percent.

- In the range of 100 to 160 bar, the increase in natural gas injection pressure produced an increase in combustion rate and NO\textsubscript{x} emissions. Within the experimental measurement accuracy, there seems to be an injection pressure around 130 bar for which thermal efficiency is maximized. At this pressure the efficiency was increased by up to 5 % (relative).

- With HPDI fueling of a diesel engine, injection timing delay has proven to be an effective way of decreasing NO\textsubscript{x} emissions. By appropriately adjusting the beginning of injection, reduction of up to 60 % (of the diesel baseline emissions) was obtained, while preserving conventional diesel efficiency.

- A shorter relative delay between the injection of pilot diesel and natural gas gave reduced thermal efficiency, engine stability and up to 25 % greater NO\textsubscript{x} emissions.

- Slow periodic variations in NO\textsubscript{x} emissions (especially at low load operation), associated with engine instability and combustion rate changes, were obtained with an HPDI injector having 6 diesel and 6 natural gas jets. These variations have been attributed to changes in interlace angle. By using different number of nozzle holes...
for pilot and gas, the engine stability was substantially improved and variability in data eliminated. An injector tip geometry arrangement of 6 diesel and 7 gaseous jets equally spaced, produced the best results of engine stability, thermal efficiency and exhaust emissions. Adopting this geometric configuration, the average NOₓ emissions from the 6-6 injector were reduced by up to 30%. Numerical modeling simulations suggest that, at low load, interlace angle has significant influence on combustion rate and NOₓ emissions. The smaller the interlace angle (i.e. the closer the jets), the higher the combustion rate and the greater the NOₓ production.

- In light of experimental results obtained in this work, KIVA 2, the 3-D program used for numerical simulations, has proven to give qualitatively useful results. It had significant importance in explaining the reasons for engine instability and variability in NOₓ emission measurements, experienced at low load with an injector having 6 diesel and 6 natural gas jets.

- HPDI of NG fueling produced low unburned hydrocarbon emissions (35-100 ppm) and similar to those of conventional diesel engines. They consisted mostly of methane (45-70 ppm), which is un-regulated.

- At operating conditions chosen to maximize efficiency, the carbon monoxide emissions (40-90 ppm) were low and unaffected by load (in the range of study).

- Numerical simulations indicate that for inclination angles of 10 and 15 deg., the gas jets cling to the fire-deck. At an inclination of 20 deg., the jets do not attach to the upper wall. Combustion rate and NOₓ emissions increase with jet inclination angle.
8.2. Recommendations for Future Work

- Experimental results on the 1-71 engine and CFD simulations demonstrate the importance of the interlace angle and suggest the necessity to control it. Optimizing its value for low load operation could produce substantial reduction in NO\textsubscript{x} emissions and eliminate engine variability. However, it is not known that an optimum interlace angle depends strongly on engine speed, RBOI, load, injection pressure and possibly other variables. Testing of injectors with different interlace angle should thus be pursued. This study must also include determination of the optimum number of nozzle holes.

- To maximize HPDI's NO\textsubscript{x} reduction potential, a more flexible technique of RBOI control should be implemented. Optimization studies should be performed with an injector which enables the possibility to quantify precisely the time delay magnitude.

- Experimental testing of injector tips with different jet inclination angles must be continued. Numerical simulation results suggest that this physical parameter could be optimized. It is suggested that the diesel and gas jet inclination angles be changed simultaneously.

- Further studies need be performed to determine whether the engine instability recorded with injectors having short RBOI is due to combustion phenomena or faulty injector behaviour.

- A study of pilot diesel minimization should be pursued. Reduction in the pilot amount could lead to further decrease in NO\textsubscript{x} and particulate emissions. Most NO\textsubscript{x} from HPDI fueling is believed to be produced by the diesel pilot.

- Particulate matter measurements need be performed to determine the level of soot reduction of HPDI fueling with respect to conventional diesel fueling. Also, a NO\textsubscript{x}-particulate trade-off is needed to establish new limits for injection timing delay.
REFERENCES


20. "Mech. 372 Undergraduate Laboratory Manual", Department of Mechanical Engineering, University of British Columbia

112


APPENDIX 1

Emission Standards for Urban Bus and Heavy Duty Compression Ignition Highway Engines

Table A1.1: Environmental Protection Agency Emission Standard for Urban Bus and Heavy Duty Compression Ignition Highway Engines

<table>
<thead>
<tr>
<th>Model Year</th>
<th>CO, g/bhp-hr</th>
<th>HC, g/bhp-hr</th>
<th>NOx, g/bhp-hr</th>
<th>PM, g/bhp-hr</th>
</tr>
</thead>
<tbody>
<tr>
<td>1990</td>
<td>15.5</td>
<td>1.3</td>
<td>6.0</td>
<td>0.60</td>
</tr>
<tr>
<td>1991-1993</td>
<td>15.5</td>
<td>1.3</td>
<td>5.0</td>
<td>0.25</td>
</tr>
<tr>
<td>1994-1997</td>
<td>15.5</td>
<td>1.3</td>
<td>4.0</td>
<td>0.10</td>
</tr>
<tr>
<td>1998+</td>
<td>15.5</td>
<td>0.5</td>
<td>4.0</td>
<td>0.10</td>
</tr>
<tr>
<td>2004+</td>
<td>15.5</td>
<td>0.5</td>
<td>2.4\textsuperscript{1} NOx+NMHC</td>
<td>0.10</td>
</tr>
</tbody>
</table>

\textsuperscript{1} Cumulated value for NOx and non-methane hydrocarbons with a limit of 0.5 on NMHC
APPENDIX 2

Evaluation of the Cylinder Pressure Signal

The sensitivity of a pressure transducer and its components carrying the charge signal require special care in operation. This appendix outlines a procedure which produced the most reliable cylinder pressure results as well as some checks that can be done to ensure accuracy of the measurements.

The high impedance signal generated by the transducer requires proper insulation while is sent to the charge amplifier. Any drop in the resistance along the way can cause a signal drift and therefore a distorted one. All connections, the cable used and the charge amplifier itself need to have matching impedances, otherwise overloading of the transducer will occur. In the oily environment surrounding the mounting port, extra care must be taken to prevent contamination of the connection with the cable. Microscopic inspection performed by the manufacturer (PCB) after short usage of a transducer indicated a drop in the contact impedance of the connection on the order of $10^6$ ohms (half of the normal operating value) due to oil contamination. That resulted in overloading of the transducer (indicated by the charge amplifier) and erroneous pressure measurement.

Hodgins and Mtui [24] evaluated five piezoelectric pressure transducers in the UBC engine lab using a Detroit Diesel 6V-92 engine. They studied the thermal characteristics of different models and makes of pressure transducers using the relationship between the measured IMEP (indicated mean effective pressure), BMEP (brake mean effective pressure), TFMEP (total frictional mean effective pressure) and theoretical values of TFMEP calculated from literature. Two experimental transducers designed for high temperature applications, two commercial models plus a water-cooled one were compared. They found substantial differences in peak pressure as well as in the IMEP.
The prototype transducers have given a negative TFMEP work indicating strong thermal effects or an error of calibration.

Douville [27] has analyzed in detail the effect of charge amplifier time constant and geometry of mounting ports on the pressure transducer signal. Evaluating the indicated diagram, his results indicated similar pressure-volume variation with medium and long time constant but obtained considerably higher pressure on compression and lower pressures on expansion using a short setting. The distortion was also reflected in an exaggerated scavenging loop.

Thermal effects induced by high temperatures during combustion (with mean values around 1000 K) can distort the pressure signal of a piezoelectric transducer in the expansion stroke, therefore a reliable method of checking its accuracy had to be developed. The checking technique used in this research uses the engine parameters (geometry) and a simple thermodynamic analysis to estimate the compression pressure at TDC. The exhaust valve closure moment coupled with the bore, stroke and conrod dimensions can be used to determine the cylinder volume at the exhaust valve closure (Heywood, 1988). Calculating the combustion chamber volume at TDC using the engine geometric compression ratio, one can find the actual compression ratio, after exhaust valve closure. Having measured the airbox pressure and using published experimental values for the polytropic coefficient, the compression pressure can thus be estimated. The validity of this method relies therefore on the accuracy of the airbox pressure measurements, which becomes another important parameter for the analysis.

Preliminary testing with a PCB 112B11 (serial # 16011) pressure transducer indicated discrepancies with Douville's data (taken with a PCB 112A12 transducer) at identical testing conditions. The motoring pressures obtained in this research were 23 % higher and the airbox pressure about 7 % lower. The higher airbox pressure in Douville's data is believed to be attributed to the presence of a muffler on the exhaust pipe (initial data sets in this research were taken without a muffler).
A detailed checking procedure has thus been performed to identify the sources of error in all instruments involved in collecting pressure data. Three different transducers and three charge amplifiers were tested. The effects of the charge amplifier's time constant and amplification factor have been investigated. The models PCB 462A (serial # 2270) and Kistler 503 (serial # 1033) charge amplifiers indicated distortion of the pressure signal when using different amplification factors as seen in Figs. A2.1 and A2.2.

Figure A2.1 shows high variations in the cylinder pressure curves during the opening of the intake ports, obtained with different amplification factors. This makes it very difficult to determine the actual cylinder pressure at the inlet port closure, a value necessary for the combustion model.

![Figure A2.1: Distortion of pressure signal with amplification factor on the P-V diagram](image-url)
A better picture of the amplification factor effects is displayed in Figure A2.2, which represents the P-V diagram obtained for constants of 100, 200 and 500 units/volts. Both compression and expansion slopes change with varying constants, creating an exaggerated scavenging loop in the case of 100 units/volts. These two graphs show an unreliable operation of the two charge amplifiers which can be due to lack or wrong calibration.

A third charge amplifier has been tested, and this gave correct results. The more recent Kistler model 5004 (serial # 200295) has shown no variation of the pressure signal with amplification factor as can be seen in Figure A2.3.

Two time constants have also been tested. Again, no sensitive influence over the pressure signal has been observed. This finding is in contrast with Douville's report of
variations of the pressure signal with charge amplifier time constant, which could be due to the use of one of the charge amplifiers described above. For the following tests a short time constant was used because a medium time constant often caused overloading of the transducer (as shown by the flashing power led, feature present only on the Kistler model 5004). Figure A2.3 represents the P-CA deg diagram obtained with short and medium time constants and amplification factors of 100-1000 units/volts. One additional finding of the analysis of pressure signal was the existence of a pre-compression before exhaust valve closure of around 10 kPa. This may explain at least in part the differences between the peak compression pressures obtained in this research and those predicted with the previous model which assumed the cylinder pressure equal to the airbox pressure at exhaust valve closure.

**Figure A2.3:** Pressure curves during exhaust valve opening obtained with different time constants and amplification factors with a Kistler charge amplifier model 5004
The best results were found therefore using a Kistler charge amplifier model 5004 (serial # 200295) with short time constant and an amplification factor of 200 units/volt (for higher values the smoothing process becomes more difficult). The three transducers used have been re-calibrated by the manufacturer and gave similar results with peak motoring pressures varying no more than 5%. The model used for the rest of the research is a PCB 112B11 (serial # 17172). Figure 4 shows the degree of repeatability of the data sets.

**Figure A2.4**: P-V diagram curves with different time constants and amplification factors obtained with the Kistler charge amplifier model 5004 (serial # 200295)
## APPENDIX 3

**Fuel Properties**

### Table A3.1: Natural Gas Composition

<table>
<thead>
<tr>
<th>Constituent</th>
<th>Percent by Volume</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane, CH₄</td>
<td>89.68</td>
</tr>
<tr>
<td>Ethane, C₂H₆</td>
<td>4.68</td>
</tr>
<tr>
<td>Propane, C₃H₈</td>
<td>1.73</td>
</tr>
<tr>
<td>Butane</td>
<td>0.67</td>
</tr>
<tr>
<td>Pentane</td>
<td>0.15</td>
</tr>
<tr>
<td>Nitrogen, N₂</td>
<td>2.89</td>
</tr>
<tr>
<td>Carbon Dioxide, CO₂</td>
<td>0.20</td>
</tr>
</tbody>
</table>

### Table A3.2: Fuel Properties

<table>
<thead>
<tr>
<th></th>
<th>Natural Gas</th>
<th>Diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lower Heating Value, kJ/kg</td>
<td>49,417¹</td>
<td>43,220²</td>
</tr>
<tr>
<td>Density, kg/m³</td>
<td>0.6³</td>
<td>860</td>
</tr>
<tr>
<td>Cetane Number</td>
<td>n/a</td>
<td>45</td>
</tr>
</tbody>
</table>

¹ BC Gas
² Diesel fuel no. 2, Chevron
³ At standard conditions: P=101.325 kPa and T=298 K
Figure A3.3: Lower heating value variation during the period January 1 to December 31, 1998 (BC Gas)
APPENDIX 4

Calibration Curves

1. Cylinder Pressure Sensor

Model used: PCB 112B11
Serial number: 17172
Sensitivity: 1.066 pC/PSI for 0-3000 PSI
1.074 pC/PSI for 0-300 PSI
Calibration date: Jul 7, 1998

Figure A4.1: Cylinder pressure calibration curve - full scale
Figure A4.1bis: Cylinder pressure calibration curve - low load scale
2. Diesel Mass Flow

Figure A4.2: Diesel mass flow calibration curve
3. CNG Mass Flow

![CNG Mass Flow Calibration Curve](image)

**Figure A4.3**: CNG mass flow calibration curve
4. Air Flow

Figure A4.4: Air flow calibration curve
5. Airbox Pressure

Figure A4.5: Airbox pressure calibration curve.
6. Airbox Temperature

Figure A4.6: Airbox temperature calibration curve

![Graph showing the relationship between measured temperature and transducer signal. The x-axis represents the transducer signal (Volt), ranging from 0 to 12, and the y-axis represents the measured temperature (deg C), ranging from 0 to 120. A straight line is drawn through the data points, indicating a linear relationship between the two variables.]
7. CNG Pressure

Figure A4.7: CNG pressure calibration curve
8. Ambient Temperature

Figure A4.8: Ambient temperature calibration curve
9. Torque

Strain Gage Load Cell

Applied Force (Nm)

Transducer Signal (Volt)
APPENDIX 5

Repeatability of Typical Results

1. Thermal efficiency

![Graph showing thermal efficiency vs. BMEP](image)

**Figure A5.1:** Typical reproducibility of thermal efficiency measurements

The data sets presented in Fig. A5.1 were taken in three consecutive days\(^1\). Test conditions were: 130 bar gas injection pressure, 1200 rpm, BOI adjusted to obtain beginning of combustion at TDC. It is assumed that LHV was constant during one test, but may have varied from day to day.
The tests performed with HPDI injectors have produced large variability in thermal efficiency, as shown in Fig. A5.1. These variations, in the order of 2.5 %, are greater than the maximum net uncertainty due to instrument error (0.32 %). Therefore, they are believed to be caused by fluctuations in combustion rate due to gas needle rotation (as is described in chapters 4 and 8), and not by inaccuracies of the instruments measuring fuel flows and output torque.

The 1.1 % trendline variation in thermal efficiency results is also greater than the maximum net uncertainty due to instrument error (0.32 %). This is believed to be due to modification of the spring constants ("softening") after injector operation, which may have altered the relative injection timing. Detailed accuracy checking, with an injector which enabled steady state operation, has given a maximum error in thermal efficiency of 1 % at maximum engine load.

1 Mar 16, 17 and 18, 1999
2. NO\textsubscript{x} emissions

As in the case of thermal efficiency, the scatter in emission measurements shown in all repeatability graphs is not due to instrument error, but to the variations in combustion rate caused by needle rotation during engine operation. Since the variability in emissions is a problem that affects equally all results, the degree of repeatability will be inferred analyzing the trendlines.

The maximum net uncertainty in NO\textsubscript{x} measurements is 10 ppm, which represents approximately 5 % relative error. It is believed to be due to calibration offset which occurs from day to day.
3. Carbon monoxide emissions

**Figure A5.3**: Typical reproducibility of CO emission measurements

The variability in CO emissions due to needle rotation can be as large as 30% of the average value measured. Changes in trendline from day to day can be up to 20% (or 15 ppm) and are believed to be due to calibration offset. Although the relative error is significant, the uncertainty is not of concern because the levels measured are very low (100 ppm max).
4. Methane emissions

The maximum net uncertainty in methane measurements is 10 ppm, which represents approximately 20% relative error. It is believed to be due to calibration offset which occurs from day to day.

Figure A5.4: Typical reproducibility of CH₄ emission measurements
5. Total hydrocarbon emissions

The maximum net uncertainty in total hydrocarbon measurements is 10 ppm, which represents 20% relative error. It is believed to be due to calibration offset which occurs from day to day.

During the research there were erroneous results which could not be explained. In some tests, the percentage of methane determined with the NDIR analyzer was greater than the total hydrocarbon emissions measured with the FID detector. Although each analyzer operates on different principles, the results obtained measuring the same compound should be identical.

Figure A5.5: Typical reproducibility of NO\textsubscript{x} emission measurements
APPENDIX 6

RBOI Study - BOI Dependence

Appendix 6 displays efficiency and emission graphs pertaining to the relative injection timing study. They are referred to in Chapter 7, but were introduced in the appendix to free up the main text of the thesis.

Figure A6.1: HPDI's thermal efficiency variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP
Figure A6.2: HPDI's NO\textsubscript{x} emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP

Figure A6.3: HPDI's THC emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP
**Figure A6.4:** HPDI's CH$_4$ emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP

**Figure A6.5:** HPDI's CO emission variation with BOI for long and short RBOI, relative to baseline diesel; 3 bar BMEP
Figure A6.6: HPDI's BSFC vs NO\textsubscript{x} curves with long and short RBOI, relative to diesel baseline; 3 bar BMEP
Computed Interlace Angle Effects on Combustion Rate and NO\textsubscript{x} Emissions

This appendix shows computational results of pressure development and NO\textsubscript{x} production with interlace angles of 0, 15 and 30 degrees, corresponding to the operating conditions of Table 8.2.2. The x axis displays crank angles after top dead center.
APPENDIX 8

JET INCLINATION ANGLE

8.1. Introduction

The purpose of this appendix is to determine the potential sensitivity of HPDI's combustion and emissions to variations in inclination angle. As a reminder, the jet inclination angle is the angle formed in a vertical plane (going through the cylinder axis) between the natural gas or diesel jets and the combustion chamber upper wall (or fire-deck), as shown in Fig. A8.1.1. For numerical simulations, KIVA code uses the injection angle measured between the natural gas or diesel jets and the cylinder axis (Fig A8.1.1). This will be called included angle, and is equal to (90°-inclination angle).

![Figure A8.1.1: Schematic of the jet inclination angle](image)

The prototype injector design with 6 diesel pilot and 6 natural gas holes has both sets of jets inclined at 10 deg from the upper combustion chamber wall. Preliminary KIVA simulations have indicated that, with such a low inclination angle, the gas jets cling to the fire-deck. The implications of this attachment could be as follows:

- Reduction of the interface between natural gas and air, which could produce lower combustion rate due to diminished oxygen availability. This may result in thermal efficiency decrease but also in NO\textsubscript{x} reduction.
- Development of combustion in the squish area, rather than in the piston bowl, possibly increasing heat transfer to the cylinder walls and reducing thermal efficiency. With higher jet penetration, a greater amount of natural gas can be trapped in the crevice between piston and cylinder, which can lead to increased unburn hydrocarbon emissions.

Hence, the questions to be answered are:
- What is the injection angle at which the jet detaches from the upper wall?
- Is there an optimum inclination angle from an efficiency-emissions standpoint? In other words, is jet attachment a good or a bad thing?

For the jet inclination angle study, initial emphasis was given to numerical simulations. Experiments with injector tips having different inclination angles are to be performed.

8.2. KIVA Predictions

Detailed 3-D simulations on the 6V-92 engine geometry have been performed for gas jet inclination angles of 10, 15 and 20 deg from fire-deck. The engine test conditions used were the same as for the interlace angle study, which are described in Table 8.2.2.

Preliminary experimental results have shown that increasing only the inclination of the gas jets is detrimental to combustion. With this geometric setup (diesel jets inclined at 10 deg.), engine stability was worsened and thermal efficiency reduced at higher loads. Therefore, all CFD tests were performed for a diesel pilot jet angle equal to that of the natural gas jets.

Figure A8.2.1 shows schematically the three tip geometry arrangements used in computations and the included angle of the jets.
The KIVA combustion animations revealed the following results:

- For inclination angles of 10 and 15 deg, the natural gas jets tend to cling to the top wall - the so-called "Coanda" effect
- At 20 deg. inclination angle, complete jet separation from fire-deck was observed. In this case combustion proceeded in the piston bowl rather than in the squish area. This is expected to produce reduction in methane emissions, due to lack of gas penetration in the crevice between piston and cylinder.
Figures A8.2.2 and A8.2.3 show the methane concentration in combustion chamber, for computations with jet inclination angles of 15 and 20 degrees, respectively. Both graphs represent in-cylinder methane distribution at the same instant after beginning of injection. One can see the jet attachment in Fig. A8.2.2, and how it separated from the upper wall when the inclination angle was increased to 20 deg. (Fig. A8.2.3).

Figure A8.2.2: CH4 mass fraction distribution in the combustion chamber - 15 degrees inclination angle
Analysis of the computed combustion development and exhaust emissions data gives some insight on the effects of increasing jet inclination angle. In Fig. A8.2.4, pressure traces from computations with 10, 15 and 20 degrees are plotted together, as a function of crank angle ATDC (after top dead center). This graph suggests that combustion rate increases with jet inclination angle, possibly due to an increased contact area between the natural gas and air. This phenomenon could be observed better at low load (both high and low speed).
Figure A8.2.4: Computed cylinder pressure variation with crank angle for jet inclination angles of 10, 15 and 20

Figure A8.2.5 shows NO\textsubscript{x} generation during combustion, corresponding to the pressure traces of Fig. A8.2.4. The computed NO\textsubscript{x} emissions also increase with jet inclination angle, possibly because of the higher combustion rate.

Numerical modeling results of the combustion rate and NO\textsubscript{x} emissions for all operating conditions studied are given in Appendix 9.
8.3. Experimental Results

The first experiments compared the prototype 6-6 injector, which has both diesel and gas jets inclined at 10°, with a modified version of the 6-6 injector that had the gas jets oriented at 15° from the fire-deck. Due to design limitations the diesel jet injection angle could not be modified, and remained 10°. Since the gas jets are injected above the diesel ones, there will be increased interaction between them when the gas needle rotates. Hence, comparison with the simulation results cannot be made because these were performed having both diesel and natural gas jets injected at the same angle from the fired deck. However, the results obtained are important and will be presented in this section.
Experiments with the modified version of injector revealed a very unstable engine behavior at low load. At no load the engine was very difficult to be kept running at steady state.

Figure A8.3.1 shows that having only the gas injected at 15°, thermal efficiency is reduced at high load with respect to the prototype injector design.

![Figure A8.3.1: Thermal efficiency of two HPDI injectors having the gas jets inclined at 10 and 15 deg. from fire-deck](image)

Higher unburned hydrocarbon emissions from the 15° injector have been obtained throughout the load range and consisting mostly of methane. These results are shown in Figs. A8.3.3. and A8.3.4. Carbon monoxide and NOx levels were similar for the two injectors, and reduced with respect to the diesel baseline.
Figure A8.3.2: THC emissions from two HPDI injectors having the gas jets inclined at 10 and 15 deg. from fire-deck.
Figure A8.3.3: CH$_4$ emissions from two HPDI injectors having the gas jets inclined at 10 and 15 deg. from fire-deck

The increased interaction between the natural gas and diesel jets has proven detrimental to combustion. These results suggest that the following experiments should be performed with both diesel and natural gas jets injected at the same angle.
8.4. Summary

Numerical simulations with different inclination angles indicate that:

- For inclination angles of 10 and 15 deg., the gas jets cling to the fire-deck. At an inclination of 20 deg., the jets are separated from the upper wall.
- At low load operation, the computed combustion rate and NO\textsubscript{x} emissions increase with jet inclination angle.

Preliminary experimental studies, which compared an injector\textsuperscript{1} having the natural gas jets inclined at 15 deg. from fire-deck with the prototype injector\textsuperscript{2}, produced the following results:

- Unstable engine behaviour at low load.
- Reduced thermal efficiency at high load.
- Increased unburned hydrocarbon emissions.

\textsuperscript{1} Diesel jets inclined at 10 deg.
\textsuperscript{2} Diesel jets inclined at 10 deg.
APPENDIX 9

Computed Inclination Angle Effects on Combustion Rate and NO$_x$ Emissions

This appendix shows computational results of pressure development and NO$_x$ production with inclination angles of 10, 15 and 20 degrees, corresponding to the operating conditions of Table 8.2.2. The x axis displays crank angles after top dead center.