HIGH PRESSURE DIRECT INJECTION OF NATURAL GAS AND HYDROGEN FUEL IN A DIESEL ENGINE

by

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Date 21 DEC 2000
Emissions and performance of an HPDI single-cylinder diesel engine fueled by natural gas and by a mixture of hydrogen-natural were investigated. A DDC 1-71 engine with electronic controls was used. Natural gas or hydrogen-natural gas mixture was injected late in the compression stroke after a pilot quantity of a diesel was injected.

Engine performance and emissions have been measured over a wide range of parameters: gas injection pressure, engine load, injection delay and mass flow percentages for the gaseous fuels. These results were compared with conventional diesel fueling.

The purpose of this research was to investigate the potential for reducing the diesel exhaust emissions while maintaining high thermal efficiency. With HPDI of natural gas and hydrogen-natural gas, NOx emissions can be reduced to almost half of those with diesel fueling, by appropriately adjusting the injection timing. NOx concentration is observed to increase with gas injection pressure. For hydrogen-natural gas fueling, an increase of the hydrogen percentage produces an increase in NOx. This is attributed to the higher combustion temperature as the percentage of hydrogen increases.

Thermal efficiency of HPDI of natural gas was greater than conventional diesel fueling at high loads but almost identical for medium and low loads. For HPDI of hydrogen-natural gas mixture the thermal efficiency is less than with natural gas fueling. The drop in thermal efficiency has not yet been explained.

With hydrogen addition, the unburned hydrocarbons are greatly reduced, for almost all the engine loads. Carbon monoxide emission with HPDI was reduced for high and low loads.
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LIST of ABBREVIATIONS

1-71 DD  single cylinder Detroit Diesel engine
6-71 DD  6-cylinder Detroit Diesel engine
6F3A     HPDI injector from series III.
ABOI     absolute beginning of injection
ABDC     after bottom dead center
ATDC     after top dead center
AVL      Anstalt für Verbrennungskraftmaschinen
         (Institute for Internal Combustion Engines)
BDC      bottom dead center
BBDC     before bottom dead center
BHP      brake horse power
BOI      beginning of injection
BMEP     brake mean effective pressure
BSFC     brake specific fuel consumption
BTDC     before top dead center
c.a.     crank angle
CARB     California Air Resources Board
CH₄      Methane
CI       compressions ignition
CNG      compressed natural gas
CO       carbon monoxide
CO₂      carbon dioxide
CN  Cetane number
CR  compression ratio
DAQ  data acquisition system
DDC  Detroit Diesel Corporation
DDEC  Detroit Diesel Electronic Control
ECU  electronic control unit
EMA  engine Manufacturers Association
EPA  environmental Protection Agency
FID  flame ionization detection
GIP  gas injection pressure
HHV  higher heating value
HPDI  high-pressure direct-ignition
ID  interior diameter
ISAAC  fast data acquisition
It  injection time
LHV  lower heating value
MEP  mean effective pressure
MW  molecular weight
NDIR  non-dispersive infrared radiation
NO  nitrogen oxide
NO2  nitrogen dioxide
NOx  oxides of nitrogen
NPT  National pipe thread
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Definition</th>
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<tbody>
<tr>
<td>$n_R$</td>
<td>number of crank revolutions per power stroke</td>
</tr>
<tr>
<td>PCB</td>
<td>Piezotronics Inc.</td>
</tr>
<tr>
<td>PM</td>
<td>particulate matter</td>
</tr>
<tr>
<td>ppm</td>
<td>parts per million</td>
</tr>
<tr>
<td>PSI</td>
<td>pound per square inch</td>
</tr>
<tr>
<td>PW</td>
<td>injection pulse width</td>
</tr>
<tr>
<td>RBOI</td>
<td>relative beginning of injection</td>
</tr>
<tr>
<td>rpm</td>
<td>revolutions per minute</td>
</tr>
<tr>
<td>SAE</td>
<td>Society of Automotive Engineers</td>
</tr>
<tr>
<td>SI</td>
<td>spark ignition</td>
</tr>
<tr>
<td>S/N</td>
<td>serial number</td>
</tr>
<tr>
<td>TDC</td>
<td>top dead center</td>
</tr>
<tr>
<td>THC</td>
<td>total (unburned) hydrocarbons</td>
</tr>
<tr>
<td>UBC</td>
<td>University of British Columbia</td>
</tr>
<tr>
<td>ZEB</td>
<td>zero emission buses</td>
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## List of Symbols

- **a**: Crank radius [mm] or [in]
- **A**: Area [m$^2$]
- **B**: Cylinder bore [mm] or [in]
- **d**: Diameter [m] or [mm]
- **LHV**: Lower heating value [MJ/kg]
- **l**: Connecting rod length [mm] or [in]
- **L**: Piston stroke [mm] or [in]
- **$\dot{m}$**: Mass flow [kg/h] or [kg/s]
- **M**: Mach number
- **MW**: Molecular weight [g/mol] or [kg/kmol]
- **$\dot{M}$**: Momentum flux [N]
- **n**: Engine speed [rpm]
- **$\dot{Q}$**: Volumetric flow [m$^3$/h] or [l/s]
- **r_c**: Compression ratio
- **$\bar{R}$**: Universal gas constant, 8314 [J/molK]
- **p**: Pressure [kPa] or [bar] or [psi]
- **$p_0$**: Stagnation pressure [kPa]
- **P**: Power [W] or [kW]

\[
\Delta h = \bar{h} - \bar{h}_{f,298} = \text{enthalpy difference [kJ/kmol] between any given state and 298K}
\]

- **$h_f^0$**: Enthalpy of formation [kJ/kmol] at 298K
- **$h_{fg}$**: Evaporation enthalpy [kJ/kmol]
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tr>
<td>$H_P$</td>
<td>enthalpy of products [kJ]</td>
</tr>
<tr>
<td>$H_R$</td>
<td>enthalpy of reactants [kJ]</td>
</tr>
<tr>
<td>$s^0$</td>
<td>entropy [kJ/kmol K]</td>
</tr>
<tr>
<td>$T$</td>
<td>engine torque [Nm] or [lbf ft]</td>
</tr>
<tr>
<td>$T$</td>
<td>absolute temperature [K]</td>
</tr>
<tr>
<td>$T_0$</td>
<td>stagnation temperature [K]</td>
</tr>
<tr>
<td>$u_{fg}$</td>
<td>evaporation energy [kJ/kmol]</td>
</tr>
<tr>
<td>$v$</td>
<td>velocity [m/s]</td>
</tr>
<tr>
<td>$V_c$</td>
<td>clearance volume [dm$^3$], [m$^3$] or [in$^3$].</td>
</tr>
<tr>
<td>$V_d$</td>
<td>displacement volume [dm$^3$], [m$^3$] or [in$^3$].</td>
</tr>
<tr>
<td>$\gamma'$</td>
<td>polytropic coefficient, specific heat capacity ratio</td>
</tr>
<tr>
<td>$\eta_{th}$</td>
<td>thermal efficiency [%]</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>air-fuel ratio</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density [kg/m$^3$]</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>equivalence ratio</td>
</tr>
<tr>
<td>$\chi$</td>
<td>mole fraction</td>
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The author would like to acknowledge the continual scientific support offered by Dr. Philip Graham Hill, my thesis supervisor, throughout the research. His invaluable experience has helped the author to surpass difficult moments, which are inevitable in this line of work. Dr. Daniel Fraser, my co-supervisor, came forward with innovative suggestions, especially for hydrogen injection in a diesel engine. I am deeply indebted to both of them for their thoughtful help.

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I also wish to express many thanks to Shu Oshika, mechanical technician, John Richards, electronics technician and Jeffrey Kohne, injector technician for their willingness to put their knowledgeable skills into this project. As some of the machining and welding were done at specialized machine-shops, I would like to extend my gratitude to Igor Gavanac, welding operator and Frank van Hall, lathe operator for their high quality work they have performed.

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The friendship and productive brainstorm with colleagues Petros Lappas, Connor Reynolds, Mike Georgalis, Pete Ostafichiuk, Adrian Field etc. were very much appreciated during my UBC period.

Many thanks to Jon Mikkelsen, who has given me the opportunity of being a Teaching Assistant in the undergraduate labs, and to Mrs. Jan Mardsen, the graduate student secretary.

Finally, I owe thanks to my parents and family for their moral support despite my being abroad.

Bogdan

December 2000
To my little Alex,
As the title states, the objective of this thesis is to compare the performance and emissions of a diesel engine fueled with either natural gas or a mixture of hydrogen and natural gas with those of diesel fueling. The motivation for switching from diesel fuel to alternative fuels is to meet U.S. EPA (Environmental Protection Agency) diesel engine emission regulations. Such regulations include reduction of nitrogen oxides (NOx), particulate matter (PM) and carbon dioxide (CO2).

This chapter provides brief explanation of how a diesel engine works with injection of natural gas. The problem of strict future diesel engine regulations is discussed along with possible methods of satisfying these regulations. An investigation has been defined for a diesel engine fueled with a hydrogen and natural gas.

Continuing a series of UBC research, the natural gas (or a hydrogen and natural gas mixture) was diesel-pilot ignited and directly injected in a single cylinder diesel engine. The high-pressure direct injection (HPDI) of the gaseous fuel occurs late in the diesel cycle. It has been previously determined that the HPDI of compressed natural gas (CNG) preserves the thermal efficiency of the diesel engine over the whole load range and reduces emissions.

This thesis addresses the effects of CNG (compressed natural gas) or a mixture of CNG and hydrogen in a diesel engine. Data obtained were in-cylinder pressure measurements, exhaust emissions and thermal efficiency.
1.1. Diesel engine operations

In order to understand how a diesel engine works it is useful to preview the air-standard cycle, which is the ideal diesel cycle.

The Ideal Diesel cycle

The ideal air-standard (ideal) diesel cycle is a closed system (i.e. does not exchange any working fluid with the surroundings). An ideal cycle for the diesel engine (compression-ignition engine) is shown in fig. 1.1.1. Heat is transferred to the working fluid at constant pressure. This constant pressure process is an approximate description of the effect of the injection and burning of the fuel in presence of air.

![Diagram of the ideal diesel cycle](image)

*Figure 1.1.1. The air-standard Diesel cycle*
Using Fig. 1.1.1 to describe the process, one can observe that during process 1-2 the compression is isentropic. From state 2 to state 3 heat is added at constant pressure. At state 3, heat ceases to be added. The gas undergoes an isentropic expansion, through process 3 to 4. The ideal air-standard Diesel cycle replaces the real intake-exhaust process by a constant-volume rejection of heat (process 4→1).

The theoretical efficiency of a Diesel cycle could be expressed as follows:

\[
\eta_d = 1 - \frac{Q_{\text{rejected}}}{Q_{\text{added}}} = 1 - \frac{C_p(T_3 - T_2)}{C_v(T_4 - T_1)} = 1 - \frac{T_1 \left( \frac{T_4}{T_1} - 1 \right)}{kT_2 \left( \frac{T_3}{T_2} - 1 \right)},
\]

where \(Q_{\text{added}}\) is the area encompassed in "a-2-3-b", and \(Q_{\text{rejected}}\) is the area encompassed in "a-1-4-b".

For an ideal diesel cycle, the isentropic compression ratio is greater than the isentropic expansion ratio. For a given compression ratio, if the maximum temperature (T₃) increases, the efficiency decreases, because the addition of heat increases a lot faster than the useful work. As shown previously in Fig. 1.1.1, because the constant pressure line (process 2-3) intersects the constant volume line (process 2'-3-3'), the increase of area 3-3'-4'-4-3 (which corresponds to an increase of useful work) is smaller than the relative increase of area 3-3'-c-b-3 (which corresponds to a large heat addition). As a result the cycle efficiency decreases.

From the efficiency standpoint, if one compares the Diesel cycle (cycle 1-2-3-4-1) with the Otto cycle\(^1\) (cycle 1-2-3'-4-1), it appears that the latter has a higher efficiency. In practice, however, because in a SI engine, the mixture air-fuel is compressed together during the compression stroke, an undesirable detonation phenomenon (knock) appears when compression...

\(^1\) Otto cycle is representing the spark ignition engine
ratio is too high (maximum of 14:1, depending on the fuel). In contrast, for a CI (compression-ignition) engine, only fresh air is compressed, and therefore there is no danger of knocking at higher compression ratio (around as high as 23:1, or more).

The advantage of using in practice a higher compression ratio for the Diesel engine in respect to the Otto engine, is that it has higher thermal efficiency. This conclusion is drawn based on the compression ratio that could be practically achieved for each of these 2 types of engines of identical displacement.

The mean effective pressure (MEP) within the cylinder during a complete cycle is obtained by approximating the area included by the pressure trace, with a rectangular area with one side equal to the displacement volume (i.e. the volume obtained by multiplying the piston area with the piston stroke). The MEP is calculated such the two areas to be equal. Using this, a theoretical thermal efficiency can be calculated, which is the ideal maximum possible efficiency to aim for.

Ultimately, along with this research, a pressure versus crank angle graph would be used for comparing different conditions for the combustion of natural gas, and/or hydrogen in combination with a very small quantity of diesel, which is meant to initiate the ignition. A Diesel engine operates by compressing air in the cylinder prior to injecting fuel into the hot compressed air to initiate combustion.

**The Real Diesel cycle**

The actual compression-ignition cycle differs from the ideal air-standard Diesel mainly because it exchanges mass with the surroundings (there are many other reasons such as irreversibility due to heat losses or friction). It is an open cycle, where the working fluid is refreshed every cycle. A diesel engine is a reciprocating - piston engine with internal (heterogeneous) mixture formation and auto-ignition. The intake of the fresh air load and the
exhaust of the burnt gases occurs during the process 4 to 1. During the compression stroke the air introduced and trapped in the cylinder is heated and compressed from BTC (bottom dead center) to TDC (top dead center). In this process the air can reach 500-800 kPa and temperatures of 750-900 K. Just before TDC, the diesel fuel is injected into the cylinder at a high enough pressure to penetrate far enough into the combustion chamber. Such temperatures are sufficient to induce auto-ignition of the diesel fuel (which is injected into the cylinders before the end of the compression stroke). This method is called “high pressure late direct injection”. The injection timing and fuel quantity is controlled by an electronic injection unit (ECU).

The fuel is atomized in small droplets near the injector nozzle and has to evaporate before it can satisfactorily mix (this is still a heterogeneous mixture) with the air. In such a mixture, the air-fuel ratio (called λ and defined as the air-fuel ratio relative to the stochiometric air fuel ratio) extends from pure air (λ is infinite for the region outside of the spray) to pure fuel (λ is zero in the spray core).

![Curve of the air-fuel ratio in an individual stationary droplet](image)

**Figure 1.1.2 Air-fuel ratio for a single stationary droplet.**

---

2 Late-cycle refers to the injection of the gas near the end of the compression stroke (close to TDC)
As can be seen from the Fig. 1.1.2. (lambda versus flame range zone) the combustion takes place for a very narrow lambda range, starting from 0.3 up to 1.5. The average value for $\lambda$ is one, which represent the stoichiometric air-fuel ratio.

A typical diesel-spray is shown in Fig. 1.1.3. There are mainly two zones for the plume: a core and an upstream jet, or spray.

![Diagram of a diesel-spray injection in a diesel engine.]

**Figure 1.1.3. Fuel spray injection in a diesel engine.**

The overall challenge is to achieve a thorough mixing process of fuel (diesel) and air in the engine, in a short time (of the order of one millisecond). This also must occur at a prescribed crank angle near Top Dead Center (TDC). Direct injection of the fuel and rapid evaporation makes this possible. Because a diesel engine is operated at medium and high loads with lean fuel/air ratios, an improved conversion fuel efficiency can be obtained, mainly due to higher compression ratio. On the other hand, slower burning rates lead to lower efficiencies. The combustion process begins shortly after the start of injection and can be divided into two phases. The first phase is “the premixed flame” phase, which takes place from the time when the fuel is injected until the start of ignition. During this phase, the fuel mixes with air and ignites spontaneously. Auto-ignition is highly dependent on temperature and less dependent on cylinder pressure at the end of the
compression stroke. The second phase is the "diffusion flame" phase. This phase may include part of the injection period, but follows the premixed air and fuel. The diffusion flame burns more slowly than the premixed flame, is responsible for most of the pressure rise (generating work), and is the primary cause of NOx formation. If the injection is very advanced, the "premixed flame" phase is responsible for the pressure increase.

The ignition delay plays a significant role in NOx formation and cylinder pressure. The ignition delay in a diesel engine can be defined as the time (or crank angle) interval between the start of injection and the start of combustion. The start of injection is usually taken as the time when the injector needle opens the nozzle orifice. The start of combustion is more difficult to determine but can be done by either measuring cylinder pressure or use optical instruments to see when the first flame occurs. There is no strict correlation between the two. However, it has been observed that the pressure rises before the first flame can be seen.

The ignition quality of a diesel fuel is characterized by its Cetane Number [CN]. For diesel fuel the CN ranges from 45 to 50, on a scale out of 100 for very high ignition quality. Since injection timing is used to control combustion timing, the time delay between the injection and spontaneous combustion has to be preserved and kept short. This ensures that the maximum cylinder pressure is maintained below what the engine can tolerate (i.e. not to engine parts damages). Fig. 1.2.1. shows a typical cylinder pressure trace for a diesel engine.
A diesel engine can be operated un-throttled by controlling the amount of fuel being injected per cycle and by maintaining a quasi-constant airflow. In contrast, a SI (spark ignition) engine controls the load by throttling the intake airflow. The result of throttling is a substantial torque loss (due to the work needed for pumping loop). Because the diesel engine is throttle-free, its thermal efficiency is higher than that of a spark ignition engine.

1.2. Diesel engine performance and emissions

Advantages of diesel engine

The compression ignition engine has a higher thermal efficiency than a spark ignition engine of the same size. The higher thermal efficiency is due to higher compression ratio ($r_c$) and throttle-free operation. For diesel engines a higher CR can be reached because there is no danger of
knock. Knock, a phenomena characteristic of spark ignition engines, is a name given to a loud noise, which is transmitted to the engine structure when abnormal combustion occurs\(^4\) (i.e. destructive auto-ignition). Since diesel is directly injected into the cylinder after compression, where heterogeneously mix with the heated air there is no premixed fuel-air mixture. That is the cause of knock-free operations (i.e. there is no detonation of uniformly mixed fuel and air).

Also for a diesel engine there is no need for throttling the fresh air as the output power is controlled by the amount of diesel that is directly injected into the cylinder. The throttle-free operation is possible because the fuel burns as a diffusion flame (where burning speed is as fast as the fuel is introduced in the cylinder) and not as a propagating flame (where burning speed is strongly dependent on mixture composition). The absence of throttle and high compression ratio give a diesel engine high thermal efficiency.

Also the throttle-free operation makes it possible for a diesel engine to reach maximum torque at low speeds. Nevertheless, the diesel engine is very reliable; truck engines may operate as much as 500,000 or even 1,000,000 miles between overhauls.

Due to its performance advantages over the spark ignition engine, the diesel engine has been for a century the dominant prime mover in all industrial and heavy-duty sectors. Almost all North American heavy or light trucks and transit buses use diesel engines. As an example, approximately 75,000 transit buses are diesel powered in North America alone.

However, as diesel emission regulations are getting tighter it becomes clear that one of the biggest disadvantages of the diesel engine is that it is heavily polluting the atmosphere.

\(^4\) The spontaneous ignition phenomenon [the occurrence of the flame before the spark (preignition) or after the spark (postignition)] causes the knock and that is governed by the temperature and pressure.
Disadvantages of a diesel engine

Atmosphere pollution can be explained as follows: at a given speed and increasing engine load, the amount of fuel per cycle increases whereas the airflow per cycle is kept almost constant (i.e. rich mixture). Because the air required to be consumed by the additional injected fuel is less than it should be (stoichiometric), soot is formed. This happens at the so-called "smoke limit" load, for which insufficient oxygen finds its way to the fuel. To prevent excessive soot formation the ratio fuel/air has to be about 20% leaner than the stoichiometric ratio. The slower-burning diffusion flame is the main cause for PM (particulate matter) and unburned HC (hydrocarbons). However, even at high λ (i.e. lean mixture), some soot forms within the diffusion flame and may not be fully oxidized when it leaves the flame.

Providing more air per stroke, a turbocharger is commonly used in a diesel engine, to increase the "smoke limited" power. A turbocharger can boost the airflow up to a relative ratio of 4:1 or even higher. This means that at high load much additional air can be forced into the cylinder to compensate for the extra fuel. However, even at low loads soot formation may still be unacceptable, because soot particles formed in the diffusion flame are not fully oxidized before exhaust. The concern is to reduce the particulate matter from the engine exhaust with diameters less than 2.5 μm (according with the 1998 EPA standard for diesel engines), which are considered carcinogenic. Appendixes A1 and A2 provide emission regulations for different vehicles.

The injection timing (beginning of injection) and duration of injection (pulse width) have a great impact on NOx production, which is very dependent on maximum flame temperatures.

NOx production is also one of the major drawbacks for the diesel engine, and its future depends very much on emission limits that have to be met.
1.3. Diesel engine emissions regulations

New technologies need to be developed and applied in order to improve diesel engine emissions. As emission regulations are periodically revised one can see that the general trend is to continually reduce the concentration of nitrogen oxides (NOx), particulate matter (PM), carbon monoxide (CO), carbon dioxide (CO₂), and unburned hydrocarbons (HC), all of which contribute to global air pollution. The top two concerns for the diesel engine emissions are soot (PM or black smoke) and NOx (oxides of nitrogen, i.e. NO and NO₂).

For most of North America, the regulations of the U.S. Environmental Protection Agency (EPA) provide the standard which all diesel engine must meet. These regulations are shown for various years in Fig. 1.3.1.

![Graph showing US EPA emissions regulations over years]

Figure 1.3.1. Emissions regulations for diesel engine implemented by U.S. EPA

\(^5\) With the stoichiometric fuel - air ratio, the amount of oxygen provided is just sufficient to completely burn the fuel.
The graph represents how abruptly the emissions limits have declined and the proposed decrease rate for coming years. It should be mentioned that while some of these emission limits (for the year 2004) have been already implemented as mandatory, others (for the year 2007) are still pending as proposed values. The target emission levels for 2004 are 0.5 g/bhp-hr for NOx (oxides of nitrogen) and 0.01 g/bhp-hr for PM (particulate matter).

These plans were presented by the EPA and by the California Air Resources Board (CARB) during a SAE Clean Diesel TOPTEC symposium held October 1999. See Appendix A1 and A2.

These emissions limits are strictly regulated by CARB for diesel powered vehicles. CARB is even stricter when it comes to buses. It is anticipated that in the following 10 - 15 years California will have 15% of their transit fleet as “Zero emission buses” (ZEB). The regulation allows transit agencies the flexibility of choosing between using either diesel or alternative fuels for powering the bus fleet. Alternative fuels that can be used include CNG (compressed natural gas), LNG (liquefied natural gas), propane, methane, or methanol.

Furthermore, CARB is imposing a global limit for the whole fleet of buses. It is referred to as the “fleet average” and it’s value will be 4.8 [g/bhp-hr] for NOx. Transit companies will be forced to retire some of the old buses, which are heavily polluting the atmosphere. A time frame was established for transit companies to show that they need to have more and more ZEB’s, over a period of 15 years.

Having seen that, diesel engine manufacturers have had to move to meet all these stringent emission regulations. The Engine Manufacturers Association (EMA) is trying to meet 0.01 PM standard by October 2002, ahead of the CARB deadline, which is January 2004.
1.4. Natural gas fueling

In the light of the EPA regulations, moving from diesel fueling to natural gas fueling appears to be a wise step. The advantages of fueling a diesel engine with compressed natural gas (CNG) are multiple. The CNG can produce less pollutant emission, while preserving diesel engine efficiency. Besides, it is widely available in North America at relatively low cost.

As NOx is formed in the post flame region, which means at very high temperatures, the diesel fueling produces more NOx emissions than the NG fueling, because diesel burns at higher adiabatic temperatures. Moreover, to maximize the thermal efficiency and to prevent knock at all loads, CNG must be injected into the cylinder directly (straight into the cylinder), as in conventional diesel engine, rather than indirectly (in a premixing chamber). However, using the indirect injection of natural gas along with a small pilot diesel, Nakagawa et al. [6] has measured a NOx concentration of 0.9 [g/bhp-hr] for a thermal efficiency of 39.1%.

Because of its advantages, the direct injection method of natural gas has been preferred and many studies are in this direction. The method of injecting the natural gas directly into the cylinder is not new but requires overcoming the high pressure built into the cylinder at the end of compression stroke, where pressure is the highest. That is why the method is called HPDI (i.e. High Pressure Direct Injection). Retaining the characteristics of the diesel engine, HPDI fueling makes possible maintaining the higher compression ratio with no danger of knocking.

Chronologically, many studies have been done for testing different methods of directly injecting CNG along with diesel pilot. In the beginning, diesel pilot was injected from one side of the cylinder while CNG was injected from the center of it. That method had the advantage of maintaining the diesel pilot quantity constant through all the load ranges, but it had the
disadvantage of an asymmetrical diesel pilot, which limited the minimum amount of diesel pilot for stable ignition. In subsequent research, both CNG and diesel pilot were injected from the center of the combustion chamber, but at different horizontal angles. A different method was to simultaneously inject CNG and diesel pilot through the same injector, which has proven to produce fewer emissions. In recent years sequential injection proved that is more advantageous, and therefore accepted for the HPDI technology. A more detailed presentation of previous work is done in chapter 2.

1.5. **HPDI (high-pressure direct injection) of CNG (compressed natural gas)**

For HPDI technology, both diesel and natural gas are required to be injected directly and sequentially. The diesel pilot is injected first, followed by the natural gas as both fuels are injected late in the cycle (just about TDC). The pilot is getting self-ignited, as it would do in a conventional diesel engine.

In the following, a parallel comparison is been done between diesel and HPDI of CNG (compressed natural gas) fueling characteristics.

A) For a conventional diesel injector, the fuel is injected a few crank angle degrees before the end of the compression stroke. The temperature of the air in the cylinder is 750-900 K. As the diesel fuel passes through the nozzle orifices it atomizes. Due to high pressure and temperature within the cylinder, the fuel vaporizes, mixes with air and auto-ignites (in less than 2 ms after injection has began).

B) For a HPDI injector the processes are as follows:

The natural gas is directly injected into the diesel engine late into the cycle, but would not self-ignite at only 900-1000 K (natural gas self-ignites at about 1300 K), which is the air
temperature at the end of the compression stroke. Though in theory one could increase the air temperature to auto-ignite the natural gas by increasing engine compression ratios, it is impractical to get such a high temperature because of mechanical stresses and strength limitations of the engine metal parts. Therefore, some sort of “spark” is required to be produced, from which the natural gas could ignite. Rather than using a glow plug\(^7\) (whose temperature has to be about 1350 K) to ignite the injected NG, a pilot diesel injection (preceding the NG injection) can provide stable ignition.

Thus, into the cylinder is first injected a small quantity (3-5%)\(^8\) of diesel fuel pilot, followed by the natural gas. The HPDI of CNG and diesel fuel injection is shown in figure 1.5.1, from which one can see the pattern of the jet plumes and the injection angles (in vertical and horizontal planes).

A significant role in understanding the combustion mechanism is played by the ignition delay, which is the time interval between the beginning of injection (BOI) and the time at which combustion raises the chamber pressure by 14 kPa (2.5% mass-burned fraction). The NG combustion time was found to be almost the same as for conventional diesel fueling, despite the fact that the rate of heat release is higher for NG and diesel pilot than that of a conventional diesel.

As previous UBC studies have indicated, the HPDI pilot-CNG injection and combustion are complicated both spatially and temporally. It was also measured that the CNG fueling has a lower NOx emissions than the conventional diesel fueling. Since NOx is temperature dependent, and the CNG maximum flame temperature is lower than that of the diesel fuel, a NOx reduction will occur for CNG fueling.

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\(^{7}\) glow plug = heating device which enables ignition on cold start engine by getting up to 1100K.

\(^{8}\) This percentage is based on the energy release by that fuel quantity.
Figure 1.5.1. HPDI of CNG and diesel injection plumes and angles configuration.
1.6. HPDI of a mixture of CNG and hydrogen

The use of hydrogen could be beneficial for reducing exhaust emissions, as the products from burning hydrogen would be mainly water vapor. However, as air is used for burning the hydrogen, nitrogen oxides are present in the reaction products. The potential for reducing the unburned hydrocarbons, soot and particulate matter is an incentive for fueling the engine with hydrogen. For the majority of North American heavy duty trucks and buses powered by diesel engines the option of direct injecting a mixture of natural gas and hydrogen into the compression ignition engines might offer a reliable solution for complying with the EPA (Environmental Protection Agency) emission regulations. One aim of the work described in this thesis was to see what percentage of hydrogen has to be injected along with natural gas, so that the emissions to be reduced and the performances to be improved.

1.7. Objectives of this research

This research is focussed on improving the HPDI of CNG as well as using the HPDI for a mixture of hydrogen and natural gas for the first time in a diesel engine.

The objectives of this research are as follows:

- To measure and compare the engine performance for three different fueling methods: conventional diesel injection, HPDI of CNG ignited by diesel pilot and HPDI of CNG mixed with hydrogen ignited by diesel pilot.
- To analyze the exhaust emissions for NOx, CO, THC obtained for all three fueling methods and correlate those with the cylinder pressure for a wide range of operating conditions.
• To see how these emissions can be reduced by appropriately adjusting the injection characteristics, such as injection timing, gas injection pressure or fuel mass flow.

• To investigate the feasibility for the HPDI technology, by analyzing the NOx production-thermal efficiency trade-off. It seems that the advantage of using the natural gas-hydrogen mixture is not in NOx reduction, but in lower THC, which indicated a more complete combustion.

Summary:

• HPDI of CNG retains or even increases the high power output of the conventional diesel engine with throttle free operation while the high thermal efficiency and low speed torque are preserved. HPDI of CNG is knock-free as the diesel fueling engine compression ratio is maintained.

• Among other advantages of HPDI of NG injection it can be mentioned that it is very reliable in operation, is not vulnerable to low quality of NG (i.e. although the NG has high methane concentration and low propane concentration, the percentage can vary) and could reduce the NOx emissions (because the NG flame temperature is lower than that of a diesel and even more it can be lowered by delaying the injection timing).

• For the same pressure conditions, the diesel auto-ignition temperature is 750-900 K, while for the natural gas that is 1200-1300 K. Such a high temperature cannot be reached in a diesel engine with our present technology of engine parts manufacturing (i.e. stress and strength of material limits would be exceeded).

• Because the natural gas does not auto-ignite for the same conditions as diesel fuel, a flame has to be initiated, either from a spark plug, glow plug or from burning a diesel pilot.
• The potential for reducing the unburned hydrocarbons (THC), soot and particulate matter with CNG - hydrogen fueling and maintaining the CNG fueling thermal efficiency and NOx emissions has been an incentive for this work.

• Direct injection of a mixture of CNG and hydrogen could produce further reduction of soot, THC, and particulate matter. A NOx reduction is unlikely, because the hydrogen maximum flame temperature is grater than that of the CNG, hence NOx concentration is more prone to increase.
Chapter 2

PREVIOUS RESEARCH

One of the methods to meet the oncoming stringent US EPA (Environmental Protection Agency) emissions regulations is the HPDI of CNG late-cycle\(^1\) for heavy duty diesel engines. The advantages of CNG fueling (and HPDI in particular) over conventional diesel fueling are high load capability, higher thermal efficiency for higher loads (or identical thermal efficiency for low loads), and reduction of NOx and particulate matter. Moreover, natural gas is readily available, has a relatively low cost and reduces our dependency on imported oil. All of these facts have motivated various researchers to investigate CNG injection in a diesel engine. The following is a brief history of these trials as well as mention of previous work on hydrogen fueling.

2.1. Direct injection of natural gas in a diesel engine

In 1982, Sarsten et al [1] started direct injection of natural gas injection in a diesel engine. He used liquefied natural gas compressed to 200 bars, vaporized and directly injected into the cylinder, where it was ignited by a small pilot diesel. By self-igniting at usual diesel pressures and temperatures the pilot diesel ignited the natural gas. Sarsten et al wanted to preserve the original diesel fuel system, so that it would be easy to switch back to plain diesel.

\(^{(1)}\) Late cycle injection means that injection occurs late in compression stroke.
The diesel pilot was injected through an unmodified centrally located diesel injector whereas the gaseous natural gas was injected through a separate valve injector eccentrically placed. There were few engine modifications, and the results were promising for using alternative fuels in a diesel engine. The experiment successfully proved that a diesel engine can be fueled with natural gas. No emissions data were taken.

In 1983, Einang et al [2] injected natural gas at high pressure in a two-stroke marine application engine and used the pilot diesel for ignition. Their project involved a loop scavenged 300 mm bore engine running at 375 rpm. The electronic injector that they used was more precise for injection timing than the previously used mechanical ones. For natural gas injection pressure of 200 bars, the thermal efficiency was slightly better than for conventional diesel fueling. Also, NOx emissions were 25% lower for CNG fueling than those of diesel fueling.

In the same year, 1983, Miyake et al [3] proved the potential for CNG fueling in a four-stroke single cylinder diesel engine (with a large bore, 420 mm). The results have indicated higher thermal efficiency than with diesel fueling. Initially, Miyake et al developed a pilot fuel injection method, which allowed the pilot diesel to be injected from one side of the cylinder, producing a plume in the radial direction. A diesel valve was mounted on one side of the cylinder. The diesel pilot providing about 5% of the total fuel energy was required to initiate the gas ignition. Natural gas was injected through a central injector at 250 bars and the plumes were distributed radially and symmetrical.

Diesel pilot quantity was approximately constant through the whole load range. Early work showed that the asymmetrically mounted gas valve couldn't produce a uniform flame propagation for the entire CNG volume and therefore, the pilot couldn't be
kept at minimum levels (i.e. 5% of the total fuel energy). Having discovered this, Miyake et al developed a method, in which the diesel pilot was mixed with natural gas before CNG was introduced into the cylinder. A single injector was used, aligned with the cylinder axis. It was found that because of pre-mixing of the two fuels, ignition improved, the engine was more stable in operation, but the pilot diesel still required being 5 percent. At about 85% of the maximum engine load, the CNG fueling thermal-efficiency was matched with the conventional diesel fueling for the same conditions.

In 1987, another team led by Miyake [4], designed an improved fuel injection system, so that fuels were injected sequentially. The CNG injection pressure was 250 bars and the diesel pilot was injected in multiple plumes, which alternated with the gas plumes. Not only was the pilot diesel reduced (2-3%) but also the cycle-to-cycle engine variations were significantly reduced. However, the required diesel pilot wasn't small for all regimes; it rose to 10 % for some operating conditions.

The same team tested high-pressure direct injection for natural gas on a large two-stroke engine. The CNG pressure was maintained at 250 bars, and high-pressurized oil at 300 bars was used as a seal between the hydraulic actuator fluid and the natural gas. Because the gas and diesel were controlled separately through two different injectors, the engine could be operated on straight diesel, just by closing the CNG injector. It was for the first time that the diesel pilot was kept both constant and small (i.e. having only 5% of the total fuel energy). Better performance was obtained with the sequential diesel-gas injection than with the mixed diesel-gas injection. The comparison was done for high loads only, and emissions measured were only CO and smoke, both smaller than those of conventional diesel fueling.
In 1987, Wakenell et al [5] injected pressurized natural gas late in the engine cycle (i.e. just before piston reaches top dead center, TDC). They used a two-stroke locomotive engine equipped with a blower for scavenging. Tests were conducted to see what improvements the late cycle natural gas injection could have had on engine thermal efficiency. The natural gas injection pressure was 340 bars, and the injector had a hydraulically actuated valve to control the gas injection. Modifying the hydraulic pressure could have changed the gas injection timing. Separate injectors were used for natural gas and diesel fuel. As in the Miyake experiment, the natural gas was injected radially from the center of the combustion chamber, but at an angle (of 25 degrees) below the fire deck. The diesel fuel was injected at 12 and 48 degrees below the fire deck using two small injectors (with two holes each) placed offset from the natural gas central valve.

The performances obtained with CNG fueling were very close (thermal efficiency was 98% of that of diesel) to that for diesel fueling. However, at high loads the pilot quantity had to be increased (about 10% of the total energy fuel content) in order not to have knock. At lower loads the engine was unstable, and difficult to control. Many more other investigations have been done with late-cycle direct injection of natural gas. [6,7,11,12]

In 1987, Hodgins et al [8], used high-pressure direct injection of natural gas in a two-stroke single-cylinder engine. The experimental injector they used had an unshrouded poppet valve, which was designed to simultaneously inject CNG and diesel pilot. The small quantity diesel was premixed with CNG outside of the cylinder, and injected along with the CNG as a conical sheet. The injection angle was 20 degrees below the fire deck. The diesel - CNG ratio (on an energy basis) and the gas injection pressure
were modified to explore the potential of using dual fueling over a full range of loads. In terms of emissions it was found that the NOx and CO2 levels were lower than with conventional diesel operation over the entire operating range. For higher loads the thermal efficiencies exceeded those of conventional diesel fueling, while for low loads the cycle-to-cycle variations produced a lower thermal efficiency for CNG.

Continuing this line of work, Tao [9], in 1992, separated the conical sheet in independent jets, by using a modified shrouded poppet valve. His research was focused on determining which injection timing and diesel-natural gas ratio was optimum for getting the best efficiency and low NOx concentrations. Tao tested Hodgins' HPDI injector, with good results (30% NOx reduction, and almost no losses for thermal efficiency compared to that of diesel fueling). Even with very advanced ignition delays, unacceptable levels of total unburned hydrocarbon (THC) emissions were obtained. These levels were an indication of poor combustion (i.e. large injection delays) and the work showed that the external mixing of diesel pilot and CNG was the cause. By injecting the diesel pilot and the CNG sequentially the ignition delay could be reduced.

In 1992, Oullette and Hill [10] analyzed the injection spray configuration using flow visualization methods (i.e. Schlieren and high speed photography). They concluded that the injection angle (i.e. the angle in the vertical plane between the fire deck and the injection plumes axis) as much as the interlace angle (i.e. the angle in horizontal plane described by the diesel and CNG plumes axis) play essential roles in improving the combustion process (especially for flame quenching). They studied the influence of the penetration distance of the gaseous fuel, which is mainly a function of the gas injection pressure.
In 1993, Douville [13] studied and compared the performance and emissions of a single-cylinder two-stroke engine fueled with HPDI (high-pressure direct injection) natural gas for late cycle injection. He found that, over a broad range of injection timings and low to medium loads, thermal efficiency for both diesel and natural gas fueling was almost identical. However, at high loads, the HPDI fueling thermal efficiency was found to be higher than conventional diesel fueling. Most pollutants have been found to be much lower than the ones obtained with classical diesel fuel. At the same time, the cylinder peak pressure was higher for HPDI of CNG than for conventional diesels.

In 1995, Tao and Hodgins[14] reported the reduction of NOx emissions from a two-stroke V6 cylinder diesel engine fueled by an improved HPDI dual-fuel injector. The diesel pilot was injected a few crank angle degrees before the gas, which was introduced very late in the compression stroke at 160 bars. CNG produces less NOx than diesel fuel when it burns because of the lower adiabatic temperature. Measurements indicated that NOx emissions were much reduced (an average of 45-55%) for a large range of injection timings\(^2\) and loads. However, injection timing (or beginning of injection, BOI) adjustment can be used for controlling the NOx production. These findings coupled with the fact that the thermal efficiency was almost as high as for diesel fueling made the HPDI technology an alternative and competitive fueling method for diesel engines. These authors also determined that the CNG injection pressure (GIP) influenced both the engine performance and the emissions. Experiments have shown that at the maximum achievable load, thermal efficiency seemed to increase as the gas injection pressure (GIP) is increased to 180 bars.

\(^2\) Injection timing is also called "beginning of injection", or BOI.
Mtui and Hill [15,16] have done further developments of HPDI technology in 1996. They have shown the influence of the CNG presence for the diesel ignition delay and the potential for NOx emission reduction. They learned that the sequential injection of diesel and CNG would provide a better flame propagation from diesel pilot to the CNG plumes.

In 1997, Meyers et al [19], implemented the sequential diesel-natural gas HPDI on a large V6 locomotive engine. One year later, in 1998, the same team (Meyers et al. [22]) successfully tested a 16-V cylinder diesel engine fueled with natural gas and ignited by diesel pilot. The novelty was that the diesel injector tip wasn't modified for natural gas injection, so that in case of a CNG fuel system failure the engine could be fueled with diesel. The development of high swirl combustion produced NOx emissions of 2.9 g/kW-hr and a total HC emissions of 5.4 g/kW-hr. These represent an important decrease from conventional diesel fueling.

2.2. Direct injection of Hydrogen in a Diesel engine

A different method for reducing the diesel engine emissions is replacing conventional diesel fueling by direct injection of hydrogen. Hydrogen-fueled diesel engines are of interest because they offer a controlled combustion and a good potential for reducing emissions. With pure hydrogen fueling the emissions will contain nitric oxides (such as NO, NO₂, NOₓ) because of the nitrogen contained in the air, but none of
the particulate matter (PM), THC (total hydrocarbons) CO, CO₂ or CH₄, except for the PM originating from lubricating oil.

The advantage of throttle-free operation for a direct injection diesel engine can be preserved on hydrogen injection. The output power is controlled by the amount of hydrogen which enters the cylinder. Although a direct-injection type of diesel engine (used primarily for heavy duty diesel engines) has a thermal efficiency greater than the indirect injection diesel, only one previous study appears to have been done on hydrogen fueling with this engine type.

In 1990, Welch and Wallace[31] directly injected pure hydrogen into a single-cylinder diesel engine, using a continuously operating glow plug as an ignition source. Welch and Wallace were looking to see how the engine performance would improve and how the emissions would decrease relative to those for diesel fueling.

Hydrogen at 10 MPa was injected into the cylinder through a hydraulically actuated injector which controlled the timing and duration of injection, while the engine was run at 1200 rpm and 1800 rpm. Hydrogen was injected through a hydraulically actuated injector, which allowed a certain amount of gas to be introduced into the engine at a pressure well above the cylinder pressure. The optimum value of hydrogen pressure depends on factors such as injector nozzle numbers and their orientation, the size and shape of the combustion chamber and pressure ratio. The results indicated that a hydrogen-fuelled diesel engine produced higher power than diesel fueling. It was also found that injection delays were about the same as diesel fuel, and that they decreased substantially as the load increased. The peak cylinder pressure was as high as for the diesel fueling when there was advanced injection timing. For retarded injection timing
the peak cylinder pressure dropped considerably. The indicated efficiency of the hydrogen-fueled diesel engine was about 90% of that of the ordinary diesel engine at moderate loads. In order to have maximum efficiency, the hydrogen injection timing must vary with the load. At full load hydrogen fueling thermal efficiency is one fifth smaller than diesel fueling. For low loads, the hydrogen-fueled efficiency is smaller than the corresponding one for moderate loads. At 1200 rpm, the indicated thermal efficiency for hydrogen injection was about one-fifth to one-third smaller than that of conventional diesel fueling, especially for low loads. No results were shown for the measured thermal efficiency (based on the torque measurements). Delaying the injection timing produced an increase in indicated thermal efficiency.

Welch and Wallace suggested that improvement in efficiency would most likely be obtained by refining the combustion chamber and injector nozzle designs. Hydrogen fueling exhaust temperatures were different for 1200 rpm and 1800 rpm. For lower rpm (i.e. 1200 rpm) it was found that the hydrogen fueling exhaust temperature was comparable to that produced by diesel fueling for a broad BOI range, especially at low loads. At 1800 rpm the hydrogen exhaust temperature was much higher than that produced with diesel fueling, particularly high (more than double) for very late BOI (i.e. 6 c.a. degrees BTDC) and low loads. That was explained by an increase of the heat transfer due to late burning. Very advanced injection timing (19 c.a. degrees BTDC) produced higher levels of NOx (1500-1700 ppm), especially for higher loads. However, by properly adjusting the hydrogen injection timing (i.e. more retarded BOI, such 6 c.a. degrees BTDC), a function of the load, the NOx emissions can be lower than those for the standard diesel.
It was found that the smoke emissions were almost eliminated, as well as CO, CO2 and THC (the only unburned hydrocarbon came from engine oil). Consideration was paid to unburned hydrogen, which was found to vary with the engine load. The maximum unburned hydrogen percentage (0.5-0.6 %) was reached for very low (at 1200 and 1800 rpm) or very high loads (only for 1200 rpm). The optimum regime from this prospective is for medium loads. The unburned hydrogen could be a cause for a low thermal efficiency.

2.3. Summary

- The CNG or hydrogen fueling can help reduce harmful emissions from diesel engine.
- With CNG fueling it has been proven that the thermal efficiency can be almost identical to that for diesel fueling, while NOx emissions can be reduced by about half, (depending on load, injection delay and pilot quantity). Particulate matter can also be appreciably reduced (because the natural gas does not contain as much bonded carbon as fossil diesel oil).
- With pure hydrogen fueling, CO, CO2, THC emissions can be eliminated, except for products of lubrication oil emissions.
- The different fuels required some diesel engine modifications, as well as some auxiliary compressors (to compress the gaseous fuel to the required direct injection pressure) and storage tanks. Because neither CNG nor hydrogen self-ignites at the existing pressure and temperature at the end of the compression stroke, a small pilot diesel is needed.
- The maximum combustion temperature strongly influences NOx formation. Since the adiabatic flame temperature for hydrogen is greater than that of natural gas, different methods to cool down the combustion temperature may need to be employed (i.e. exhaust gas re-circulation).
- The benefits of hydrogen in reducing emissions of carbon compounds suggest possible benefits of using mixtures of hydrogen and CNG as fuel for direct injection diesel engines.
Chapter 3

ENGINE TESTING APPARATUS

3.1. Introduction

This chapter describes the testing apparatus and the instrumentation that was used for measuring the engine performance and emissions. The emissions of NOx, CH4, CO, CO2, O2, and THC (total unburned hydrocarbons) are of special interest. Two sets of data acquisition are described, as they were used to acquire and store all needed information. This chapter also estimates the experimental uncertainty for the principal measurements, and the main indicators of performance.

Description of a single cylinder diesel engine

The 1-71 Detroit Diesel engine that has been used at UBC is a naturally aspirated, two-stroke engine with forced uniflow roots blower for intake air. There are no intake valves since fresh air enters the cylinder via intake ports, which are holes slotted at the bottom of the cylinder liner. The burnt gases exit via two push-rod activated exhaust valves.

Shown in Fig. 3.1.1. is a cross-sectional drawing of the engine, from which one can see most of the engine parts. The figure shows the valve train, consisting of the camshaft, two cam followers, two spring loaded push rods, two rocker arms, and two exhaust valves. Later in this chapter an explanation is given of how a modified cylinder head was fitted to the existing engine block. The admission of fresh air and exhaust of
Figure 3.1.1. Cross section of 1-71 DD engine
burnt gases is called the scavenging process. During part of this process the intake and exhaust ports must both be open, as shown by Fig. 3.1.2.

Figure 3.1.2. Timing diagram for 1-71 DD engine (figure reproduced from GM 1-71 DDC engine, Operating manual).

In the figure one can see that as the crankshaft rotates clockwise, the intake process starts at 48° before bottom dead center (BBDC), and finishes at 48° after the bottom dead center (ABDC). From the same figure it can be seen that, as the exhaust valves open at 85.5° BBDC and close at 55° ABDC, there is an overlapping process throughout the whole intake process. Since the engine is a two-stroke engine it is
important to note that the piston is opening and closing the intake ports, as they are at the
bottom of the cylinder liner. The roots-blower communicates with the intake ports, so
that more air can be introduced if necessary.

There were two special features of the use of this engine: a new monovalve
cylinder head and a HPDI injector.

**Engine specifications**

The research engine specifications are:

*Table 3.1. Detroit Diesel 1-71 single cylinder Engine specifications*

<table>
<thead>
<tr>
<th>Type</th>
<th>Two-stroke diesel engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>Stroke</td>
<td>4 ¾ in (108 mm)</td>
</tr>
<tr>
<td>Bore</td>
<td>5 in (127 mm)</td>
</tr>
<tr>
<td>Total displacement</td>
<td>20 in³ (1.162 liters)</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>16 : 1</td>
</tr>
<tr>
<td>Maximum speed</td>
<td>1400 rpm</td>
</tr>
<tr>
<td>Gross Rated Power Output</td>
<td>15BHP (11.2kW) @ 1200 rpm.</td>
</tr>
<tr>
<td>Gross Rated BMEP</td>
<td>4.8 bar @ 1200 rpm.</td>
</tr>
<tr>
<td>Rated peak torque</td>
<td>56 lb ft (76 Nm) @ 1200 rpm.</td>
</tr>
<tr>
<td>Piston speed @ 1200 rpm</td>
<td>1000 ft/min (5 m/s)</td>
</tr>
<tr>
<td>Diesel fuel consumption</td>
<td>0.227 kg diesel /bhp-hr.</td>
</tr>
<tr>
<td>Cooling water (when a heat exchanger is used)</td>
<td>7 quarts(^{(1)}) (6.6 liters)</td>
</tr>
</tbody>
</table>

\[^{(1)}\] 1 liquid quart = 0.946 l
This engine was fueled originally with diesel liquid fuel, for which the factory rated engine characteristics (horsepower and torque) are presented in Fig. 3.1.3. From this figure one can see that the maximum horse power increases linearly (from 10 HP to 25 HP) over a broad range of engine speeds (from 600 rpm to 1600 rpm). Maximum rated torque, however, has a peak of 33 [lbs ft] at about 1200 rpm. This fact along with the stability in engine operation, were two reasons to choose 1200 rpm for test the speed of the engine.
Figure 3.1.3. Horse power and torque curves for 1-71 DDC engine fueled with diesel.
Fuel system supply

Diesel fuel, compressed natural gas or gaseous hydrogen were delivered through three independent fuel supply systems. Diesel fuel was fed into the diesel injector manifold. The diesel was required by all of the fueling methods (i.e. CNG or hydrogen-CNG mixture) as a small diesel pilot is used to initiate the ignition. Natural gas is compressed from 15 kPa supply to 20 MPa using two commercially available natural gas compressors, and then stored in both low pressure and intermediate pressure bottles. See Fig. 3.1.4.

Figure 3.1.4. Diesel and CNG injection fueling set-up for 1-71 DDC engine.
Natural gas drawn from these bottles is further compressed to 35MPa using a hydraulic activating intensifier (or high-pressure compressor). After being compressed by the intensifier the natural gas is stored in high-pressure bottles and from there it is fed into the HPDI injector manifold via a pressure regulator. The CNG injection pressures used were 130, 150 and 170 bars.

Hydrogen was taken straight from a high-pressure (2600 psi) hydrogen bottle. A regulator adjusted the pressure and a metering-valve set the hydrogen mass flow. The mixing between the CNG and hydrogen was done in a tee connector mounted upstream from the HPDI injector. The schematic of the CNG and hydrogen mixing (which is done ahead of the HPDI injector) installation is shown in Fig. 3.1.5.

The main task was to maintain equal pressure for both CNG and hydrogen flows (so there would be no back-pressure). Mass flows were measured by one flow meter for each gaseous fuel. Hydrogen was injected at smaller mass fractions than natural gas, starting with 0.25 kg/h (which represents 15% of the CNG mass flow) and then increasing it to 0.5kg/h (which represents 25% of the CNG mass flow). The adjustment of the mass flow was done with the help of a metering valve. Two electronic flow meters ("Micromotion") were used for measuring the mass flow of each gas. To inject both gases through the same HPDI injector they have to have the same pressure (i.e. 100 bars, or 130 bars GIP). To ensure an equal pressure for hydrogen and natural gas two regulators were used, one for each gas.
Figure 3.1.5. Hydrogen and CNG injection fueling set-up for the 1-71 DDC engine.
The Diesel and HPDI injectors

There were two injectors used for all engine operations: one was a factory made diesel injector (used for conventional diesel fueling) the second was a prototype "Westport Innovation Inc.", FIII series (6F3A) HPDI injector (used for both CNG or CNG and hydrogen mixture injection). The injectors were almost identical in exterior shape, but fundamentally different internally. The standard diesel injector is presented in Fig. 3.1.6. and the HPDI injector is shown in Fig. 3.1.7. The fundamental differences between the two are as follows.

In the standard diesel injector, the downward motion of the plunger pressurizes the diesel fuel when the poppet control valve is down (as indicated in Fig.3.1.6.). The poppet valve is activated by the electronically controlled solenoid only for the pulse width duration. The injection itself is done mechanically by the plunger, which is pushed downward by a rocker arm (not shown in Fig. 3.1.6.). The plunger creates pressures large enough to open the spring-loaded needle valve at the tip of the injector.

On the other hand, the HPDI injector, shown in figure 3.1.7., is more complicated than its diesel counterpart. Diesel fuel is trapped in the same manner as for diesel injector. The special feature is that both diesel liquid and natural gas are injected sequentially from the same injector. Two concentric spring-loaded needles (one for diesel and one for CNG) can freely move in and out. By adjusting the length or the spring constant of the diesel spring, one can alter the diesel pilot quantity.

Since diesel fuel is almost incompressible, it is used as hydraulic fluid to activate the CNG needle to open. In other words the CNG needle stays open as far as the hydraulic pressure within the HPDI injector remains high enough (7000-8000 psi).
Figure 3.1.6. Detroit Diesel electronically controlled injector.
Figure 3.1.7. "Westport Innovations Inc." HPDI injector.
It should be pointed out that the diesel injection pressure, about 3500 psi,
is different from the hydraulic diesel internal pressure (used for actuating the
CNG needle). CNG is fed through a designated gas port, which is different from
the diesel intake manifold. The gas that might leak is evacuated through a vent
line.
3.2. Engine instrumentation

The 1-71 DDC engine has been instrumented for:

- Torque measurements.
- Cylinder pressure measurements.
- Coolant, engine oil, intake air and exhaust gas temperature measurements.
- Intake air, diesel fuel, CNG and hydrogen mass flow measurements.
- Intake air, CNG and hydrogen pressure measurements.
- Crank angle position.
- Exhaust gas (NOx, CO, CO2, O2, CH4, THC) emission measurements.

Electronic fuel injection system

The 1-71 DDC engine injection timing is controlled by a Detroit Diesel Electronic Control (DDEC) system. The electronic fuel injection system consists of electronic unit injectors, sensors and electronic control module (ECM). The ECM controls opening (i.e. it controls the fuel quantity, which is injected) and injection timing (i.e. it correlates the fuel injection with the piston position at the end of the compression stroke). The signal sent from the electronic control unit (ECU) activates an injector solenoid valve, that operates the poppet valve from the injector. There is a time delay between the moment at which the ECU sends an injection command pulse and the moment at which the solenoid valve starts to move. That time delay is called "rising time" and is shown in Fig. 3.2.1. The rising time might influence the fuel injection process, especially at high rpm (in this case 1200 rpm).
Figure 3.2.1. The time difference between the injection command and the solenoid response
The actual rising time signal is shown in figure 3.2.2.

Figure 3.2.2. The real rising time for the injector solenoid.
As shown in Fig. 3.2.2, the actual rising time is not a square increase from zero to maximum, but a ramped up signal. The rising time differs for the diesel injector or HPDI injector. The average rising time for a HPDI injector is around one millisecond. That translates to a smaller actual injection pulse width. Also from Fig. 3.2.2, one can see that the stay-open signal is pulsating and the closing time is very short, as the closing signal is very abrupt.

**Torque measurements**

To measure the 1-71 DDC engine brake horse power\(^1\) a water-brake dynamometer was used. The dynamometer dissipates the mechanical energy developed by the engine and the corresponding engine torque can be measured by using a strain gage load cell. The engine power is transformed into heat and is dissipated by a water flow. The factory error calibration for the dynamometer is ± 0.1% maximum relative torque [Nm] reading. The water-brake is rated at 550 hp, which is more than the engine could produce. The dynamometer is controlled by an electro-pneumatical valve which regulate the water flow (i.e. the water flow regulates the torque). A 1/4 of a turn potentiometer was used to manually control the electronic dynamometer controller.

\(^1\) Brake horse power [HP], which is measured, is the net engine output.
Cylinder Pressure measurements

Cylinder pressure measurement was obtained via a piezoelectric pressure transducer mounted in the cylinder head. A X112A04 PCB piezoelectric transducer model (Serial # 7215) was used. A piezoelectric transducer is a small quartz element, which generates a current when pressure acts on it. The transducer has high output impedance (order of GΩ) which has to be matched with that of a charge amplifier input. Failing to match them will generate a load on the pressure transducer, which will then be amplified by the charge amplifier, thus giving an erroneous pressure signal. Because the resistance of the connecting cable from the quartz transducer to the charge amplifier can influence the signal (i.e. if the resistance is lowered, the cylinder pressure signal drifts) special care had to be taken not to have any dirty connections or bend, stretched and stripped cable (for which the resistance decrease). The calibration sheet for piezoelectric sensor is given in Appendix B1.

Shown in Fig. 3.2.3-3.2.5 are the cylinder pressure signals (measured by the piezoelectric sensor) for 20 consecutive engine cycles which were previously superimposed and averaged. The accuracy of those measurements depends on the type of pressure sensor. Studies of cylinder pressure measurements ([44], [46], [47]) will be presented in Appendix B2. Shown in Fig. 3.2.6. is a typical cylinder pressure for 20 consecutive cycle plotted against the crank angle degrees. All these pressure traces are averaged and the result is a single pressure trace for that regime.
Figure 3.2.3. Cylinder pressure versus volume displacement for 3 bars BMEP with diesel fueling for different BOIs at 1200 rpm. The graphs were plotted in log-log scale.
Figure 3.2.4. Cylinder pressure versus volume displacement for 130 bars GIP, 3 bars BMEP, with CNG for different BOIs at 1200 rpm. The graphs were plotted in log-log scale.
Figure 3.2.5. Cylinder pressure versus volume displacement for 130 bars GIP, 3 bars BMEP, 0.25 kg/h hydrogen and 2 kg/h CNG, for different BOIs at 1200 rpm. The graphs were plotted in log-log scale.
Figure 3.2.6. Cylinder pressure for 20 consecutive cycles for 130 bars GIP, 5 bars BMEP, with CNG fueling, at -15 c.a. degrees BOI and 1200 rpm.
Temperatures and pressure measurements

Thermocouples were used for measuring temperatures of ambient air, airbox intake air, exhaust gases, supply diesel fuel, engine coolant, engine oil, and dynamometer exiting water. Some of these temperatures were used to monitor the engine. Others were used for efficiency calculations (ambient air and airbox intake air temperatures). A calibration graph for the airbox thermocouples is presented in Appendix B3. The temperature uncertainty is ± 2 degrees, as they will be specified in table 3.6. Pressures were measured for ambient air and for the airbox.

Fuel mass flow measurements

Diesel fuel flow is measured by an "AVL\(^{(1)}\) gravimetrical flow meter (i.e. the volumetric diesel quantity fills a weighted re-circulating reservoir, which is emptied periodically). According to the "AVL" manufacturer, the maximum diesel flow measurements error is ± 0.2 % of the full scale. The natural gas flows through a "Micromotion" flow-meter device before reaching the HPDI injector. The meter compares utilizes the Coriolis effect and provides a voltage signal which is linear with mass flow rate and independent of the fluid being measured. A second "Micromotion" flow-metering device was needed to measure the hydrogen mass flow. Both "Micromotion" flow-meters are identical, but differently configured for the two gaseous fuels.

\(^{(1)}\) AVL - "Anstalt für Verbrennungskraftmaschinen", or Institute for Internal Combustion Engines, in Kleiststraße, Austria.
The natural gas one was configured from 0 to 50 kg/h, whereas the hydrogen one was configured from 0 to 4 kg/h. Shown in Appendix B4 are the relative error estimates [in percentages] for the range from zero to maximum flow. Figures 3.1.4 and 3.1.5 show schematically the diesel - CNG and CNG- hydrogen injection fueling set-up. The difference between the diesel, CNG and hydrogen mass flow uncertainties is shown in table 3.6, to be discussed in section 3.6.

Engine rpm

Engine speed was measured by a magnetic probe shaft encoder (called "SoftPot"), which was rigidly mounted on the engine crank shaft. The "SoftPot" magnetic induction probe (shown in Appendix B5) was placed over a 60-tooth gear rotating with the crank shaft. The sensor provided a signal frequency proportional to engine rpm (i.e. 1 rpm = 1 Hz). The manufacturer specifications for the "SoftPot" indicate a ± 0.1 % maximum relative error, which translates to a ± 1.5 rpm maximum uncertainty, at the maximum engine speed of 1500 rpm (see Table 3.6)

Exhaust Gas Analyzers

Exhaust gas emissions in the engine tail-pipe were measured on a volumetric basic. From all measured constituents, only the NOx, CO and THC concentrations were used in the present research. Because the THC (total hydrocarbons) or NOx (i.e. NOx represents both NO, nitric oxides and NO2, nitric dioxides) could be condensing (at low temperatures) on their way from the exhaust engine muffler to the emissions bench
analyzers, a heated sampling line was used. The sample lines were heated at a constant temperature of 190°C.

All exhaust sampling (a portion of the total exhaust gases) gases flowed through heated lines and passed through two different filters. Once they enter the emission bench analyzers, they are purged of particulate matter (PM). Since they are cleaned of PM, the gases then flow through a heated pump, to direct the gases toward each analyzer.

Shown in Fig. 3.2.7 is a schematic of the emission analyzers. The path that is followed by the exhaust gas splits in several directions. This is because the gases should be conditioned before they enter certain analyzers. For instance, the NOx and CO concentrations were measured by two different analyzers (Siemens Ultramat 22P and Siemens 21 P) using a non-dispersive infrared (NDIR) absorption detection method. Before it gets into Ultramat 22P analyzer (that is for NOx and CH₄ ppm measurements) the flow is cooled in a chiller and then splits in three paths, each of them going to the CO, CO₂ and O₂ analyzer.

A measuring apparatus (RATFISCH RS-55) using the flame ionization detection (FID) method determined the THC concentration. The 1-71 engine's THC was compared with those of a known THCs burning products in a small built-in oven.

Since a preliminary calibration for the analyzers has to be done before every engine test, a start up procedure for the emission bench analyzers is presented in Appendix B6. The calibration consists of determining zero and maximum scales for the apparatus. To determine zero, an inert gas (such as nitrogen) was passed through the analyzer to purge any other gas components. To determine the maximum concentration which could be measured (also called span), a calibration gas with known concentration
Figure 3.2.7. Schematic of the emission bench analyzers.
was admitted. Each analyzer had a different calibration, function of what gas is analyzed (CO, CO₂, CH₄, O₂, NO, NO₂). For both zero and maximum readable concentration, the analyzer output analog signal (electrical) is proportional to the gas concentration. The electrical signal voltage for all the analyzers is presented in Appendix B7, from which one can see the voltage representing minimum and maximum gas concentrations (measured in parts per million, ppm).

The uncertainty of the emission measurements is affected by two major error sources. One is due to the analyzer scale, which in most of the cases is too large for the magnitude of the gas concentration that are measured on 1-71DD engine. In other words, if the actual measurements are at the bottom of the scale, the relative uncertainty would be much larger than for full scale readings. The analyzer minimum and maximum gas concentrations, with their corresponding voltages are shown Appendix B7. Also in Appendix B7 is shown the maximum relative percentage error and the maximum reading uncertainty. As an example, a CO reading of 50-100 ppm (for CNG fueling) on a total scale of 10,000 [ppm] Siemens 21P analyzer represents only 1% of the full range. The relative maximum reading uncertainty in this case is ± 15 ppm, which given a net uncertainty of 30 ppm, which is already 30 to 50% of the actual measured reading. Similar uncertainties were encountered for NOx and THC, at different figures. Table B7b from Appendix B7 shows the averaged ppm emissions measured for NOx, CO, and THC, with the percentage of the apparatus scale utilization.

The second source of persistent (yet not constant) errors was the day-to-day calibration of the emission bench. Although the calibration procedure depends on the gas flows used for both zero or span, it is more important to maintain steady flow during the
calibration. However, it was impossible to set the flow identically for different days. For example, the flow for CO should be kept between 2 and 3 l/min, as recommended by the manufacturer, but a 2.3 [l/min] flow, say, would make a different calibration than a 2.5 l/min flow. Day-to-day reproducibility inconsistency was observed for the NOx and CH₄ analyzer, which displayed some residual value when no exhaust flow passed through them. This "hysteresis" value for the NOx analyzer was 190-230 ppm. The average measured NOx concentration ranged from 1000 to 2500 ppm, depending on the fueling method, BOI, or load. The error therefore spreads from 5% to 10%.

Having described these two major error sources, we were interested to see if data were repeatable or not. The reproducibility tests are presented in Appendix B8. All the emissions graphs were plotted for wet-basis concentration. In Appendix B10 an explanation for wet-basis of exhaust emissions is presented.

**Data acquisition system**

A computer-controlled data acquisition (called "slow DAQ") was used for monitoring the emissions, flow, pressure and temperature signals as well as logging the crank angle position, the engine RPM and the dynamometer torque. Once signals were converted from analog to digital, amplified and filtered they were fed to a data file within an IBM-PC computer. A UBC developed software was used to coordinate and store all raw data (see Appendix B9, for a sample). The slow data acquisition had to be configured for different fueling methods such as CNG or diesel, since it can display only 16 channels at once. A sample of DDC1.cfg (for diesel fueling) and DDC2.cfg (for CNG fueling) is presented in Appendix B10.
A second data acquisition (called "fast DAQ" or ISAAC) was used for acquiring the cylinder pressure signal and the BDC (bottom dead center) position of the cylinder per each engine cycle. The BDC signal was used to trigger the start of the data acquisition while the crank angle signal was used as an external clock. A charge amplifier unit amplifies (from -10V to +10 V) the instantaneous pressure signal captured by the non-cooled piezoelectric (quartz) pressure transducer thread mounted in the combustion chamber. Through the use of data acquisition, cylinder pressure values were grabbed over 20 engine cycles, and then averaged. To visualize the cylinder pressure trace in real time, an oscilloscope was used. A sample of the cylinder pressure output file is given in Appendix B11.
3.3. Experimental procedure

The engine was tested at constant speed, 1200 rpm, for which the engine torque was maximum. Three fueling methods were used: conventional diesel, CNG with diesel pilot, and a mixture of CNG - hydrogen with diesel pilot. The BOI (beginning of injection) was varied from 25 to 7 degrees crank angle (c.a.) before top dead center (BTDC) for CNG fueling and from 21 to 9 c.a. degrees BTDC for the CNG and hydrogen fueling. The BOI increments were taken in 2 c.a. steps. For both modes of gaseous fueling the influence of the very early or very late injection delay (BOI) was measured.

Engine loads were measured for brake mean effective pressure (BMEP), which is calculated as 
\[ \text{bmep} = \frac{2\pi T}{V_d} \]
where \( T \) is the measured engine torque, and \( V_d \) is the volume displacement. The engine loads were chosen to be low (1 bar BMEP), medium (3 bar BMEP) and high (5 bar BMEP). However, 5 bar BMEP could only be reached with CNG or hydrogen and CNG mixture fueling, because the diesel engine could not provide torque equivalent to 5 bar BMEP. For the diesel fueling, the engine was tested at 1 and 3 bar BMEP respectively. The load of 1 bar BMEP turned out to be very low for the CNG-hydrogen mixture. Hence, normal pulse width was too short to allow enough hydrogen mass low to enter the cylinder. Therefore, for the CNG-hydrogen mixture fueling the engine was tested only for 3 and 5 bar BMEP.

Tests were conducted with two injectors: a conventional diesel injector and a "Westport Innovation Inc." HPDI (high pressure direct injection) dual fuel injector. The factory (Detroit Diesel) diesel injector was used for conventional diesel fueling. Data obtained with diesel fueling is referred to as diesel baseline data. The HPDI dual fuel
injector has been used to sequentially inject both diesel pilot and CNG or diesel pilot and a mixture of CNG and hydrogen.

Two different hydrogen mass flows were used in combination with CNG: 0.25 kg/h and 0.5 kg/h, which roughly represented 12.5% and 25% respectively of the CNG mass flow. Since the rpm was kept constant, the diesel pilot varied only with load. The CNG mass flow rate varied with engine load and hydrogen mass flow. For higher loads, both pilot and CNG mass flow increase.

For CNG fueling three different gas injection pressures (GIP) were used: 130, 150 and 170 bars. For hydrogen and CNG mixture fueling, 100 and 130 bars were used as GIPs. Cylinder pressure was recorded and averaged over 20 consecutive cycles for any fueling method. The cylinder pressure was plotted for different fueling methods, BOIs, loads or hydrogen percentages. Also local pressure variation, when the injection occurs, was analyzed for different fueling methods, loads, and injection timings.

Thermal efficiency was compared for all of the above cases. The relative thermal efficiency error was calculated based on the cumulated manufacturer's apparatus errors and the estimated uncertainty assigned for measurements. Losses within the HPDI injector were also analyzed and estimated.

NOx, CO and THC emissions were plotted, analyzed and compared for different conditions (i.e. different loads, BOIs, GIPs) or fueling methods (i.e. diesel fuel, CNG, CNG and hydrogen, for two different hydrogen percentages). These points were selected to cover an entire engine load range, using different gas injection pressures and injection delays.
3.4. Cylinder head modifications

This section describes all the engine modifications that have been done to accommodate a different cylinder head on our existing single cylinder engine. Originally, the adaptation of a different cylinder head to the 1-71 DD engine was intended for optical visualization. Later, the optical visualization objective was omitted from this project. However new tests were done with the new cylinder head.

In order to insert an optical probe perpendicularly into the cylinder head, one of the two exhaust valves was replaced with a plug through which the optical probe (endoscope) was to pass. Plugging one of the exhaust valves would have had little geometrical influence on the combustion chamber configuration, but the inbox air temperature and pressure would have increased around 30% compared with the 2-valves cylinder head configuration.

However, not willing to sacrifice the 1-71 DD cylinder head, a different cylinder head\(^1\) (from a 3-71 DD engine) was obtained and modified. Most of the details of the modified cylinder head are shown in Fig. 3.4.1 and 3.4.2.

Freshly cut water-jackets were covered by welding cast-iron walls on one side of the cylinder head. Because the cast-iron becomes brittle on welding, the cylinder head was initially heated up to 700-800\(^0\) C in an oven and then welded in a continuous move. To avoid cracks formatted due to non-uniform or accelerate cooling, the welded cylinder head was placed in a protective medium for a two-day cooling process. The final cooling consisted of burying it in sand for a week.

\(^1\) Although different in sizes, both cylinder heads were from the same two-stroke Detroit Diesel engine family
From Fig. 3.4.1. one can see that after being cut off, the cylinder head had its water jackets (coolant passages) exposed. These needed to be covered with a wall. The diesel fuel rails were rerouted. The exhaust manifold was modified and one of the exhaust valve was removed and replaced with a plug that was welded in place.
The following steps briefly and chronologically describe the cylinder head modification process.

1) Before being cut, the cylinder head was stripped of the right exhaust valve. Also the valve spring, its sealing O-ring, the valve seat and the valve guide were removed.

2) The one-cylinder head portion was separated from the three-cylinder head by using a bench-saw.

3) A specialized welding shop did the cast-iron welding, grinding and surfacing operations for the sawed cylinder head. A cast-iron plug was machined and inserted instead on one of the exhaust valve. Some coolant holes and stud bolts were relocated, whereas other coolant passages were modified to match the 1-71 DDC engine block configuration.

4) Several cylinder head machining operations were carried out:
   a) All the flat surfaces of the cylinder head were resurfaced on the milling machine.
   b) Push-rods holes from the cylinder head were bored and reamed eccentrically, to allow the original push-rods to freely pass through.
   c) Oil passages were bored to a larger diameter. Two oil holes have been threaded with coarse NPT (national pipe thread) at the bottom of them and with SAE facing seal threads for the top facing surface.
   d) Engine oil passages were partially plugged since the oil channel configuration had to be re-routed to match with that from the 1-71 DDC engine block.
e) Two SAE type plugs were modified. One was used to connect the oil supply line to the motor oil filter and the second one to completely plug the extremity of an oil passage. A special SAE reamer tool needed was used.

f) Coolant holes from the modified cylinder head were plugged and eccentrically milled to match the engine block pattern.

g) Diesel rail configuration were modified. A steel spacer was mounted.

h) The pressure transducer holder diameter was reduced and mounted so that the sensor will be flush with the fire deck.

i) The sealing method for the pressure transducer holder was changed. The banjo-fitting sealing solution was preserved, but the underneath O-rings were removed as such, the holding bolts did not poke into the water jacket as for the original cylinder head.

j) A three-step hole was drilled perpendicular to the cylinder head fire-deck. An NPT threat has been machined to mount the pressure transducer holder. A pilot hole was drilled to make a concentrically thread for the pressure transducer holder.

k) An aluminum flange-adapter has been manufactured for the thermostat.

l) A customized 3 holes bracket has been manufactured to hold the injector push rod spring in place.

m) Two mounting holes for the spring bracket were threaded at an enlarged size into the cylinder head.

n) One side of the cylinder head has been covered with an aluminum plate.

___

1 "Igor Gavanac and sons Co." have done the cast-iron welding
o) A bronze space bushing for the deactivated rocker arm shaft was made.

p) Lathe machining of the oil adapters.

5) Manufacturing cylinder head gaskets using a template. The template was identical for the old or new cylinder head.

6) Several other cylinder head modifications were made to ensure proper sealing against the exhaust gases (plugs and gaskets surface grinding), coolant (installing a double O-ring on a spacer plate) and engine oil (pressurized oil hoses and connectors were needed) (see drawings).

After doing all these modifications, the engine was tested with both diesel and CNG fuels at different loads, BOIs or engine speeds. Minor corrections were needed to be done before we were able to run the engine at full capacity. Successful tests were conducted thereafter for diesel, CNG and hydrogen. No further engine problems were encountered. Appendixes C1-C9 show photos or scaled drawings which were needed for modifying the cylinder head.

\(^3\) The whole process lasted 16 months.
Figure 3.4.2. The 3-71 cylinder head after all the modifications were done.

From Fig. 3.4.2, the wall needed to convert part of a 3-71 (3 cylinder) head to a 1-71 (one cylinder) head is shown.
3.5. Engine characteristics for the modified cylinder head

As the 3-71 DD cylinder head was substantially modified to fit the 1-71 DD engine, and one of the exhaust valves was removed a comparative analysis of engine characteristics and emissions was needed. Therefore measurements with both cylinder heads were conducted for HPDI of CNG at 130 bars GIP (gas injection pressure), 3 bars BMEP (brake mean effective pressure) at 1200 rpm.

For these conditions, as it can be seen from Fig. 3.5.1. the airbox intake pressure has increased for the single-valve cylinder head compared with that of two-valve cylinder head (which it will referred in the following as the original cylinder head). The variation of pressure air in the airbox was constant over a broad BOI range (i.e. from 5 to 15 c.a. degrees BTDC) for both cylinder heads. The pressure increase for the modified cylinder head is $1/7^{th}$ of the original cylinder head. It was believed that the engine will be given a slightly higher emissions, due to the decrease in breathing capacity. Figure 3.5.2. shows a decrease of $1/6^{th}$ for the volumetric air flow of the modified cylinder head compare to the original one. That was anticipated, because the breathing capacity was reduced.

Knowing the exhaust valve diameter ($39.725 \text{ mm}^2$) and the piston speed at 1200rpm (5m/s) one can simply calculate the volumetric flow which would have passed through the exhaust valve:

$$\dot{Q} = Av = \frac{\pi d_{EV}^2}{4} \cdot v = \frac{\pi (39.725)^2}{4} \cdot 5 = 6.2 \cdot 10^{-3} \text{ m}^3 / \text{s} = 6.2l / \text{s}.$$
Figure 3.5.1. Intake (airbox) pressure [kPa] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
Figure 3.5.2. Airflow (dv/dt) [l/s] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
Figure 3.5.3. Air massflow (dm/dt) [kg/hr] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
Figure 3.5.4. Intake temperature [K] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOLs, with one and two exhaust valves.
Figure 3.5.5. NOx (wet) [ppm] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
Figure 3.5.6. Thermal Efficiency [%] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
Figure 3.5.7. CNG massflow \([\text{dm/}dt]\) [kg/hr] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
The air mass flow variations with BOI shown in Fig. 3.5.3. are small, both for the two-valve cylinder head and the one-valve cylinder head. For a late BOI (i.e. from 5 to 11 c.a. degrees BTDC) the air mass flow for one-valve is smaller than that of the two-valve by about 10%. For early BOI, the difference is as low as 5%.

The intake air temperatures plotted in Fig. 3.5.4, are larger for the modified cylinder head than for the original one. This is because with greater flow restriction the blower pressure ratio is larger. With higher cylinder inlet temperature the NOx production can be higher. As shown in Fig. 3.5.5, the NOx production ranged from 10% (for 5 c.a. degrees BTDC) up to 25% (for 15 c.a. degrees BTDC), higher for the single-valve head. The NOx variation is similar for both modified and original cylinder heads for the whole BOI spectrum.

The thermal efficiency for both cases, as is indicated in Fig. 3.5.6, is slightly larger (2-3% increase) for the one-valve compared to two-valve, especially for late BOI (ranging from 5 to 11 c.a. degrees BTDC). Once the injection delay is further delayed (11 to 15 c.a. degrees BTDC) the difference between the two efficiency drops to zero.

As is shown in Fig. 3.5.7, the CNG mass flow to be injected in the engine is generally stable at 1.8 - 2 kg/hr for one-valve and 1.7-1.8 kg/hr for two-valve. The maximum CNG mass flow appears to be larger for very late BOI than for more advanced injection delay. It is mentioning that for very early BOI (i.e. 15 c.a. degrees BTDC) both CNG average mass flows were identical for one or two-valve. Natural gas mass flow fluctuates because of dynamometer fluctuation to maintain a constant load. If the load cannot be kept constant (within reasonable error margins), the engine rpm would not be
Figure 3.5.8. BSFC [g/kW-hr] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
Figure 3.5.9. BSFC of NOx [g/kW-hr] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
Figure 3.5.10. Trapping efficiency [%] for 3 bar BMEP, 130 bar GIP @ 1200 rpm for different BOIs, with one and two exhaust valves.
constant, and the engine accelerations will definitely influence the fuel mass flow and consumption.

As one can see from Fig. 3.5.8, which indicates the brake specific fuel consumption \([\text{g/kW-h}]\), the consumption is slightly increased for one-valve than for two-valve. A similar trend as for thermal efficiency graph could be seen here. Once the BOI is later than 13 c.a. degrees BTDC, the brake specific fuel consumption of two-valve exceeds that of one-valve.

The NOx brake specific consumption \([\text{g/kW-h}]\), shown in Fig. 3.5.9, has a similar variation as the NOx \([\text{ppm}]\) versus BOI, shown in figure, namely the NOx emissions are higher for one-valve than those for two-valve. A large difference could be seen for early BOIs.

From Fig. 3.5.10, one can see that the trapping efficiency curves (i.e. the air quantity which is trapped inside the cylinder after the intake ports were closed) resemble with the air mass flow curves (Fig. 3.5.3.), namely the almost constant trapping efficiency for two-valve opposes a less linear variation for one-valve. It could be seen that starting from 9 c.a. to 15 c.a. degrees BTDC, the trapping efficiency for one-valve continually decreases to reach the correspondent trapping efficiency for two-valve (about 65%).
Summary of performance and emissions of the one-valve head

- By removing one exhaust valve and plugging the port, engine breathing capacity was reduced. This affected efficiency and emissions. However, the engine can be operated at full torque and rpm.

- Corresponding intake air pressure from the airbox for one valve is 15% larger than for two valves. The volumetric airflow for one-valve is 15% smaller than the corresponding one for two-valve.

- From the emissions study, it can be concluded that corresponding NOx for the one-valve head is increasing as much as 20-25% compared to that of the two-valve, especially for early BOIs. For very late BOIs there is no difference in NOx production between the two.

- The air mass flow of the one-valve head is an average about 10% lower than that of two-valve, especially at very late BOI (i.e. 5 c.a. degrees BTDC).

- Taking into consideration all the modifications it was concluded that the modified cylinder head would serve as a useful tool for comparing the effects of fuel composition variations, as well as for future use in optical study of ignition and combustion.
3.6. Preliminary calculations

Before CNG, hydrogen and diesel testing was done, some preliminary calculations were done to ensure that safety or satisfactory flow conditions were attained.

Fuel mass flow calculations (shown in Appendix D1) for diesel fuel and CNG were done to see what is the ratio of the two. Momentum fluxes were then calculated, to estimate the influence of GIP (gas injection pressure) upon jet penetration. No jet travel distance was calculated. In Appendix D2 the choked mass flow calculation for hydrogen and CNG is compared against the measured mass flows (using a flow meter).

Mole fractions for hydrogen and CNG were determined, and then mixture polytropic coefficient. As computed choked mass flow was larger than the measured one, a flow meter flow verification was done. That is also presented in Appendix D3. The flow calibration was done to determine how accurate the flow meter readings were.

To ensure that no supersonic hydrogen velocities flow through the tubes, a velocity calculation was done in Appendix D4. It was found that while the velocities vary with tube diameter, they are far less (in a range of 1-2 m/s) than supersonic ones.

Appendix D5 presents the lower heating value fluctuations for a span of three months.

As NOx production is highly dependent on temperature, a simple adiabatic flame temperature calculation was done for both methane (which is approximating CNG) and hydrogen. These calculations are shown in Appendix D6. The purpose was to compare them and to explain why, with higher adiabatic flame temperature for hydrogen one could
expect to have greater NOx emissions. The iterative tabulated results are shown in Table D6a and Table D6b.

An important task was to calculate the thermal efficiency for the hydrogen-CNG fueling, which it is presented in the following.

3.6.1. Thermal efficiency calculations

The thermal efficiency for an engine can be expressed as:

$$\eta_{th} = \frac{W}{Q} = \frac{P}{m_f LHV}$$

where, $P = 2\pi \frac{n}{60} T$, is the brake power [kW]

$W$ is work output, $Q$ is heat input, $m_f$ is fuel mass flow [kg/hr], $T$ is the output torque [Nm], $n$ is the engine speed [rpm], and $LHV$ is the lower heating value [MJ/kg].

Therefore the thermal efficiency is a calculated value based on some constants (LHV of the specific fuel), some parameters which are kept constant (rpm,) and some parameters which are varied (engine load, variation of the fuel mass flow is a consequence of varying the BOI, pulse width PW, torque T). For almost identical BOI, engine speed, the dual fuel injector the actual mass flow for diesel pilot varies from 0.25 kg/hr up to 0.35 kg/hr, and the CNG mass flow varies from 1.8 kg/hr to 2.3 kg/hr. The hydrogen mass flows were either 0.25kg/h or 0.5 kg/h.

For an HPDI dual-fuel injector the injected fuel is comprised from 5% diesel and 95% CNG. For multiple fuels the thermal efficiency is calculated as:

$$\eta_{th} = \frac{W_{\text{output}}}{Q_{\text{input}}} = \frac{P}{\sum_{i=1}^{n} m_f LHV_i}$$
where, the summation of the products of the \( m_f \) and \( LHV_i \) extends over all fuels used for the combustion (\( i = 1, \ldots, n \) fuels).

The results of these calculations are shown in Appendix B6 as they were used to plot thermal efficiency traces in section 5.4.

Along with the thermal efficiency, the combustion efficiency can be defined as the fraction of the fuel energy supplied which is released in the combustion process.

\[
\eta_c = \frac{H_{\text{Reactants}}(T_{\text{Reactants}}) - H_{\text{Products}}(T_{\text{Products}})}{\sum_{i=1}^{n} m_{f_i} LHV_i}
\]

where \( m_f \) are the average mass flows for fuels and oxygen (from the air), \( H \) represent the enthalpy at ambient pressure of both reactants (fuels and air) and products (exhaust gases) and \( LHV_i \) represents the lower heating value for all the fuels.

The combustion efficiency gives an indication of the combustion completeness, and has an impact on CO emissions.

**The thermal efficiency uncertainty calculation**

Determining the thermal efficiency is very important for comparing two different fueling methods. The accuracy of these calculations is equally important, because if the difference between the two compared efficiencies is in the error range field it could not be concluded which thermal efficiency is bigger or smaller.

To estimate the error for the thermal efficiency one has to take into account two types of systematic errors: one is due to any of the measuring apparatus (which is provided by the apparatus' manufacturer) and second is assigned to the reading measurements (that is estimated based on experiment uncertainty).
A. Errors due to apparatus accuracy

Shown in table 3.6. are uncertainty estimates provided by instrument manufacturers.

*Table 3.6.1. Uncertainty estimates provided by instrument manufacturers.*

<table>
<thead>
<tr>
<th>Number</th>
<th>Measured parameter</th>
<th>Rated (manufacturer) uncertainty</th>
<th>Absolute value</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>%</td>
<td></td>
</tr>
<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>BOI</td>
<td>± 0.2</td>
<td>±0.04 c.a. degrees</td>
</tr>
<tr>
<td>2</td>
<td>Ambient Temperature</td>
<td>± 2</td>
<td>±1°C</td>
</tr>
<tr>
<td>3</td>
<td>PW</td>
<td>± 0.2</td>
<td>±0.04 c.a. degrees</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 temperature</td>
<td>± 2</td>
<td>±2°C</td>
</tr>
<tr>
<td>7</td>
<td>Intake air flow</td>
<td>± 0.5</td>
<td>±0.6 kg/h</td>
</tr>
<tr>
<td>8</td>
<td>CNG pressure</td>
<td>± 1</td>
<td>±0.2 bar</td>
</tr>
</tbody>
</table>

|        | Averaged flow           | Averaged error                   |
|        |                         |                                  |
| 9      | Diesel mass flow        | ± 2                              | 0.45 kg/h                       | ±0.009 kg/h |
| 10     | CNG mass flow           | ± 2                              | 2 kg/h                          | ±0.04 kg/h |
| 11     | H₂ mass flow            | ± 1                              | 0.25 kg/h                       | ±0.002     |
|        |                         |                                  | 0.5 kg/h                        | ±0.005     |
B. Assigned errors for hydrogen mass flow readings

For determining the "Micromotion" flow meter output signal of the hydrogen flow an ampere-meter was used. The current fluctuation was $\pm 0.5$ mA for both flows: 5 mA for 0.25 kg/h and 6 mA for 0.5 kg/h. A $\pm 0.1$ mA reading error has been assigned for all of these indications. This reading error translates to 0.002 kg/h or 0.005 kg/h maximum reading mass flow uncertainty, which correspond to 0.25 kg/h or 0.5 kg/h hydrogen mass flow.

C. Calculation of the thermal efficiency errors

The thermal efficiency is calculated using formula:

$$
\eta_{th} = \frac{W_{output}}{Q_{input}} = \frac{P}{\sum_{i=1}^{n} \dot{m}_i LHV_i \omega} = \frac{\omega T}{\sum_{i=1}^{n} \dot{m}_i LHV_i} = \frac{2\pi \frac{n T}{60}}{\sum_{i=1}^{n} \dot{m}_i LHV_i}
$$

(Heywood [34], page 164)

The thermal efficiency uncertainty is calculated using the following formula:

$$
\frac{d\eta}{\eta} = \sqrt{\left(\frac{dT}{T}\right)^2 + \left(\frac{d\omega}{\omega}\right)^2 + \left(\frac{d}{\sum_{i=1}^{n} \dot{m}_i LHV_i} \sum_{i=1}^{n} \dot{m}_i LHV_i \right)^2}
$$

(5)

where $d\left(\sum_{i=1}^{n} \dot{m}_i LHV_i \right) = \sum_{i=1}^{n} \dot{m}_i \left( LHV_i \right) + \sum_{i=1}^{n} d\dot{m}_i \left( LHV_i \right)$

(6)
A sample calculation for 130 bars GIP, 3 and 5 bars BMEP, 19°c.a. degrees BTDC, 0.25 kg/h or 0.5 kg/h hydrogen in mixture with CNG, and pure CNG, at 1200 rpm is shown in Appendix D7.

At 3 bars BMEP, for either 0.25 kg/h or 0.5 kg/h hydrogen mass flow, the thermal efficiency uncertainty is 3-3.5%. At 5 bars BMEP, for either 0.25 kg/h or 0.5 kg/h hydrogen mass flow, the thermal efficiency uncertainty is 2-2.5%. (see Appendix D7).

3.6.2. The engine output work

To clarify the previous discussion about the output work, a brief description of it is offered in the following paragraph.

A two-stroke engine cycle starts by getting the fresh air load through the intake windows, which is then compressed during the compression stroke. Just before TDC (top dead center) the ignition occurs (due to higher temperature and pressure), followed by the expansion stroke (or power stroke) which produces useful work. Just before BDC (bottom dead center) the exhaust valves open, allowing the burned gases to exit from the cylinder. The operation of cleaning the cylinder of the burned gases and filling it with fresh air (which is a combined intake and exhaust process) is called scavenging process.

A cylinder pressure versus displaced volume [i.e. the volume described by the sweep of the piston from the BDC (bottom dead center) up to the TDC (top dead center)] is showed in figure 3.6.2.
2-stroke

Figure 3.6.2. The p-V diagram for a two stroke diesel cycle engine.

The DDC 1-71 single cylinder engine used in this research has a uniform-scavenged flow configuration, which means that the intake is done through the intake ports (placed at the bottom of the cylinder) and the exhaust is done through two exhaust valves, placed at the top of the cylinder.

The indicated work per cycle (per cylinder) can be obtained by integrating around the curve to obtain the area enclosed on the diagram:

\[ W_{\text{indicated}} = \int p \, dV, \]

where \( dV \) is taken infinitesimal for which the pressure gradient can be considered zero (pressure is constant for that very small "dv" element).

The power per cylinder can be expressed as:

\[ P_{\text{indicated}} = \frac{W_{\text{indicated}} \cdot N}{n_R}, \]
where $n_R = 1$ for two-stroke cycles engine, is the number of revolutions per each power stroke per cylinder.

### 3.6.3. Brake mean effective pressure

Brake mean effective pressure (BMEP) is useful for comparing performance of different engine designs. It is not dependent on the size of the engine. As was mentioned before, BMEP is obtained by dividing the output work per cycle by the cylinder-displaced volume per cycle.

It has units of force per unit area, therefore could be named a pressure.

Thus $\text{mep} = \text{work per cycle/ displacement}$ or,

\[ \text{mep} = \frac{Pn_R}{V_d} = \frac{Pn_R}{V_d n} \quad \text{or} \quad \text{mep} = \frac{6.28n_R T}{V_d} \tag{7} \]

where $n_R =$ number of crank revolutions for each power stroke per cylinder.

For a two strokes cycle engine $n_R=2$.

$n =$ revolutions per seconds

\[ \text{mep}[\text{kPa}] = \frac{P[kW]n_R \times 1000}{V_d[\text{dm}^3]n[\text{rev/s}]} \quad \text{or} \quad \text{mep} = \frac{6.28n_R T[Nm]}{V_d[\text{dm}^3]} \]

The area obtained by multiplying the MEP with the volume swept is equal to the area of the pv diagram. For this 2 stroke research engine there is no pumping loop, so basically the area encompassed by the close loop is described by the rectangle of sides mep, and $V_d$. For the DD 1-71 engine, the displacement volume is 70.9 cubic inches ($V_d = 70.9 \text{ in}^3 = 1.1618 \text{ dm}^3 = 1.1618 \text{ l}$).
3.6.4. Lower and Higher heating value

The differences in chemical composition and heating values for hydrogen and natural gas (considered to be mostly methane) can produce an adiabatic flame temperature difference. In table 3.6.4, the heating values for diesel fuel, natural gas and hydrogen are presented.

Table 3.6.4. The LHV and HHV for diesel, CNG and hydrogen.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>phase</th>
<th>LHV</th>
<th>HHV</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural gas(C\textsubscript{n}H\textsubscript{3.8n}N\textsubscript{0.1n})</td>
<td>Gaseous</td>
<td>49.6</td>
<td>56</td>
</tr>
<tr>
<td>Diesel (C\textsubscript{n}H\textsubscript{1.7n})</td>
<td>Liquid</td>
<td>42.8</td>
<td>45.5</td>
</tr>
<tr>
<td>Hydrogen (H\textsubscript{2})</td>
<td>Gaseous</td>
<td>120.9</td>
<td>142</td>
</tr>
</tbody>
</table>

Where LHV represents the lower heating value, and

HHV represents the higher heating value.

The heating value or the calorific value of the fuel is the magnitude of the heat of reaction at constant pressure or at constant volume at a standard temperature [usually at 25\degree C, or 298 K] for the complete combustion of unit mass of fuel.

The more commonly used is the heating value at constant pressure.

The LHV is used when the H\textsubscript{2}O formed as a reaction product is all in the vapor phase, and the HHV is used when the H\textsubscript{2}O formed is all condensed to the liquid phase.

\textsuperscript{1} These values were taken from John B Heywood - Internal combustion engine fundamentals, page 915 - McGraw-Hill 1988.
The relation between them is:

\[ HHV_p = LHV_p + \left( \frac{m_{H2O}}{m_f} \right) h_{fg H2O} \]  \hspace{1cm} (8)

where, the indices "p" stands for constant pressure, \( \left( \frac{m_{H2O}}{m_f} \right) \) is the ratio of \( H2O \) produced to mass of fuel burned, and \( h_{fg H2O} \) is the evaporation enthalpy for water which is formed in the reaction. The evaporation enthalpy of formation is:

\[ h_{fg} = h_g - h_f \]

where \( h_g \) is the enthalpy of saturated liquid and \( h_f \) is the enthalpy of the saturated vapor.

Similar formula can be expressed for constant volume heating value:

\[ HHV_v = LHV_v + \left( \frac{m_{H2O}}{m_f} \right) u_{fg H2O} \]  \hspace{1cm} (9)

where, \( u_{fg H2O} \) is the evaporation internal energy: \( u_{fg} = u_g - u_f \)

The relation between the internal energy and the enthalpy is:

\( u = h - Pv \), where \( P \) is pressure and \( v \) is specific volume.

Calculating the heat input (HI) per unit time:

Heat/s [W] = Gas consumption [m³/s] * LHV [J/m³]

For CNG, the HI = gas consumption * 45.

\( H_2 \), the HI = gas consumption * 120.
Chapter 4

MEASUREMENTS WITH DIRECT INJECTION OF CNG FUELING

4.1. Introduction

The performance and emissions of the 1-71 DDC single cylinder engine were investigated for CNG fueling using a HPDI (high pressure direct injection) injector.

The late-cycle HPDI gas injection in a diesel engine produces important changes in engine performance (cylinder pressure, engine load) and emissions (such as NOx, CO, THC, particulate matter) compared with diesel fueling. These changes are analyzed in this chapter, and compared to the diesel baseline. The goal of injecting natural gas in a diesel engine is to reduce the emissions (particularly NOx, and soot) while preserving the high efficiency and sturdiness of the diesel engine.

In this phase of research, the single cylinder engine was run in the following regimes:

*Table 4.1. Test matrix of engine regimes*

<table>
<thead>
<tr>
<th>BOI</th>
<th>BMEP</th>
<th>GIP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Crank angle degrees BTDC</td>
<td>bars</td>
<td>bars</td>
</tr>
<tr>
<td>5 to 25</td>
<td>1,3 and 5</td>
<td>130</td>
</tr>
<tr>
<td>5 to 25</td>
<td>1,3 and 5</td>
<td>150</td>
</tr>
<tr>
<td>5 to 25</td>
<td>1,3 and 5</td>
<td>170</td>
</tr>
<tr>
<td>5 to 25</td>
<td>1 and 3</td>
<td>Diesel baseline</td>
</tr>
</tbody>
</table>
Table 4.1. (which is a test matrix) represents the variation of BOI (beginning of injection), BMEP (brake mean effective pressure or engine load) and GIP (gas injection pressure) in four series of tests. The engine speed (rpm) was maintained at 1200 rpm and the gas injection pulse width (PW) was varied to produce the required BMEP at each GIP and BOI. The pulse width for diesel pilot injection was approximately 4 crank angle degrees (c.a.) and the relative injection delay (RBOI) was approximately 6 degrees. With diesel fueling the engine could not be operated at 5 bars BMEP.

The gas injection pressure influences the mass flow, which is choked at the exit of the injector nozzle holes. With HPDI fueling the CNG accounts for 95% of the total energy released. The decrease of NOx concentration (as will be shown in this chapter) is mainly due to the gas injection, affected by the GIP.

By varying the engine load for the same BOI and GIP, we can study and compare the CNG fueling engine performances (such as cylinder pressure and thermal efficiency) and emission levels on one side, versus the diesel fueling performances and emissions, on the other side. It is a trade-off between thermal efficiency and NOx production (which is very dependent to high temperatures).
4.2. Cylinder Pressure

The variation of the cylinder pressure with crank angle depends on parameters such as BOI (beginning of injection), GIP (natural gas injection pressure) and engine load. A typical cylinder pressure versus crank angle degrees for the 1-71 Detroit Diesel engine fitted with a CNG injector is shown in Fig. 4.2.1.

**Figure 4.2.1.** A typical cylinder pressure trace for single cynder 1-71 DD engine. The graph was plotted for -11 c.a. BTDC BOI, 3 bar BMEP load and 150 bars GIP at 1200 rpm. The dieslpilot is only 5%.

From Fig. 4.2.1, one can observe that the cylinder pressure is nearly atmospheric for the scavenging process. At about 100 crank angle degrees before Top Dead Center (TDC) the pressure starts increasing. That is due to the compression process and to the roots blower pressure ratio (for this research engine that is 1.2 to 1). The roots blower feeds air to the cylinder at about 1.25 bars. At 11 degrees crank angle (in this example) BTDC, the injection of the diesel pilot begins. The following increase of the cylinder pressure is
small and is due to the diesel combustion. As it can be seen from Fig. 4.2.2. (which is a magnified figure of the hatched zone from Fig. 4.2.1.) just before 0 degree (which corresponds to the Top Dead Center, TDC) the natural gas is injected. The delay between the diesel and gas injection is known as relative beginning of injection (RBOI). The pilot diesel fuel injection hasn’t yet finished and gas is flowing into the cylinder. The natural gas is injected and the following increase of the pressure is large, mainly due to the gas combustion. The engine cycle continues with the expansion stroke, which produces useful work. The cycle ends with the scavenging process, in which the burnt gases are purged from the cylinder while fresh air is introduced. The scavenging process is characteristic for two-stroke engines and there is a portion of the cycle in which fresh air and exhaust gas are mixed (see Fig. 4.2.1.)

![Graph of the cylinder pressure due to diesel and natural gas combustion](image)

**Figure 4.2.2.** Representation of the beginning and end of the injection for diesel and natural gas. The graph was plotted for 11 c.a. BTDC BOI, 3 bar BMEP load and 150 bars GIP at 1200 rpm.
Figure 4.2.2. shows the beginning and the end of the pilot and gas injections. The engine was run at constant speed (1200 rpm), for 130, 150, 170 bars GIP (injection pressure), for a load of 1, 3 and 5 bars BMEP (brake mean effective pressure), and a sweep of injection delay from 25 to 5 c.a. degrees BTDC (before Top Dead Center).

For most of the tests defined in section 4.1. cylinder pressure [kPa] was plotted against crank angle degrees for different BOIs (beginning of injection). (See Fig. 4.2.3 - 4.2.9).

In each case the cylinder pressure suffers a small increase due to diesel pilot injection and combustion. That appears as a small protuberance on the left side of the pressure trace. The amount of diesel that is injected in the cylinder is equivalent to only 5 percent of the total energy released based on multi-fuel burning. The time delay between the diesel injection and the formation of the first flame fronts is very short and is a function of the engine speed, engine load, and the beginning and duration of injection.

Throughout this research the single-cylinder engine was operated at 1200 rpm because, for a broad range of engine loads (from minimum load, zero bars BMEP, up to maximum load of 5 bars BMEP, for a CNG injector), the engine operated smoothly at that speed. Also, the maximum torque and power were attainable at 1200 rpm.

For this engine speed the delay between diesel injection and the moment where the first ignition sites are formed is 3 or 4 crank angles degrees [15]. This corresponds to 0.4 to 0.5 milliseconds.
Fig 4.2.3. Cylinder Pressure [kPa] for 3 bars BMEP, Diesel baseline @ 1200 rpm for different BO's [c.a. degrees BTDC]
Fig 4.2.4. Cylinder Pressure [kPa] for 3 bars BMEP, 130 bars GIP @ 1200 rpm for different BOIs [c.a. degrees BTDC]
Fig 4.2.5. Cylinder Pressure [kPa] for 5 bars BMEP, 130 bars GIP @ 1200 rpm for different BOIs [c.a. degrees BTDC]
Fig 4.2.6. Cylinder Pressure [kPa] for 1 bar BMEP, 150 bars GIP @ 1200 rpm for different BOIs [c.a. degrees BTDC]
Fig 4.2.8. Cylinder Pressure [kPa] for 5 bars BMEP, 150 bars GIP @ 1200 rpm for different BOIs [c.a. degrees BTDC]
Fig 4.2.9. Cylinder Pressure [kPa] for 5 bars BMEP, 170 bars GIP @ 1200 rpm for different BOIs [c.a. degrees BTDC]
The maximum cylinder pressure can vary dramatically function of the load, BOI, GIP and PW. For early BOI's (i.e. 19, 21, 23, 25 c.a. BTDC) and medium loads (i.e. 2, 3 bar BMEP) this 1.16 liter engine can produce an instantaneous (for several c.a.) cylinder pressure as high as 7000 or 8000 kPa. Moreover, if the GIP (natural gas injection pressure) is high enough (170 bar) the cylinder pressure can reach 8000 kPa.

![Cylinder Pressure for 3 bar BMEP, BOI =-5 c.a. BTDC, 150 GIP @ 1200rpm](image)

**Figure 4.2.10.** Cylinder pressure trace for single cylinder 1-71 DD engine for 5 c.a. BTDC BOI, 3 bar BMEP load and 150 bars GIP at 1200 rpm.

As shown in Fig. 4.2.10., for late BOIs (i.e. 5 c.a. BTDC) the situation is changing so that the cylinder pressure after the gas combustion is smaller than the cylinder pressure after the diesel injection and combustion. Similar characteristics are seen for pressure traces at 5, 7, 9 c.a. degrees BTDC. This is due to two factors: the diesel pilot itself does
not burn properly in either premixed (diesel mixed with air to the combustible limits) or diffusive (diesel burns as far as is injected in the engine) phases; and the natural gas combustion is delayed during the expansion stroke. The conclusion is that the thermal efficiency is low and emissions are high. For late combustion the burning could be incomplete, the flame temperature decreases, therefore the NOx production decreases.

The late cycle fuel injection might explain why, for very late BOIs (i.e. the beginning of injection is very close to TDC, top dead center), large PWs (i.e. duration of injection) are required. This is practically proved by the fact that the engine is easier to control for early BOIs rather than for late BOIs. Misfiring or random firing could be another explanation for rough engine running with late BOI. Since NOx decreases with late BOI, whereas CH4, CO and THC decrease with early BOI, there is a BOI range (from 13 to 19 c.a.\(^3\)) which is best, taking all emissions into account.

The cylinder pressure increase due to fuel injection can be seen only for BOI greater than 11 c.a. degrees BTDC. The cylinder peak pressure traces for higher BOI are substantially increasing with engine load.

\(^3\) That is shown in paragraph 4.5. (see fig.4.5.1-4.5.3., 4.6.1, 4.7.1. for different BOI's, loads and GIP's)
4.3. Injection pulse width

The duration of the injection, called Pulse Width (PW), is the nominal but not actual opening time for the injector needles\(^1\). The pulse width always increases with load and depends somewhat on beginning of injection (BOI) and gas injection pressure (GIP). One can classify the important variable as independent (engine load, injection pressure and BOI) and dependent (pulse width (PW), thermal efficiency, emissions). Here we take PW as the interacting dependent variable.

As it is shown in figure 4.3.1, the PW variation with engine load has a minimum (or a plateau).

\[\text{Fig 4.3.1. Dependency of pulse width with BOI, for 130 bars GIP for different engine loads and 1200 rpm.}\]

\(^1\) The research injector that was used had two concentrically needles, one for gas and one for diesel.
Figure 4.3.1 shows the measured values for the pulse width. Since at a given load minimum pulse width means maximum efficiency one can see from Fig. 4.3.1. that there is a range of BOI in which the efficiency is nearly constant. For injection BOI earlier than about 25 degrees BTDC, and later than 5 degrees BTDC the engine efficiency suffers considerably.

\[ \text{PW [c.a.]} \text{ vs GIP [bar]} \text{ for 3 bars BMEP and 1200 rpm} \]

![Graph showing relationship between PW (pulse width) and GIP (gas injection pressure)]

\[ \text{PW [c.a.]} / \text{GIP [bars]} \]

\[ \text{GIP [bar]} \]

**Figure 4.3.2.** Dependency of PW (pulse width) on GIP (gas injection pressure) for 3 bars BMEP load, 25 c.a. to 5 c.a. degrees BTDC and 1200 rpm.

Shown in Fig. 4.3.2.is the variation of the pulse with the BOI and the GIP. It can be seen that the gradient of PW is relatively steep for small changes in GIP. The PW decreases sharply from 12 to 7 c.a. degrees. There is a plateau for 160 - 170 bars GIP,
which suggests that for a high enough injection pressure, the penetration through the combustion chamber is producing sufficiently fast, so that there is no need for a large PW. This observation could be used for future design for natural gas intensifier requirements.

Figure 4.3.3. Dependency of PW * GIP versus GIP (gas injection pressure), for 3 bars and 1200 rpm.

Shown in Fig. 4.3.3. is the variation of PW*GIP, which is approximately proportional to the fuel mass flow per engine cycle. The pulse width depends mainly on BOI and engine load. If thermal efficiency, engine load and engine speed were constant then the mass flow should be proportional with the constant and proportional with product of the choked flow injector pressure and pulse width. As it can be seen from figure 4.3.3. the product PW * GIP is almost constant for a GIP variation from 130 to 170
bars. Additional information to understand these phenomena is given by the thermal efficiency curves, shown in figure 4.3.4.

**Fig 4.3.4.** Thermal efficiency variation function of the engine load BMEP (bars) at 1200 for 130 bars GIP (gas injection pressure)

The graph was plotted for different engine loads (2,3,4,5 bars BMEP), maintaining the injection gas pressure constant (130 bars GIP). From Fig. 4.3.4 one can notice that there is almost no variation for thermal efficiency at a given engine load for a broad range of BOI (i.e. from 25 c.a. BTDC to 5 c.a. BTDC). Also, the highest thermal efficiency is attained at the highest load (5 bar BMEP), with an average of 31%, and the lowest thermal efficiency is almost 14% for 2 bars BMEP. The variation of the thermal efficiency for 5 bars is almost linear, for a broad range of BOIs (from 3 c.a. BTDC to 25
c.a. BTDC), which once again indicates a very good stability in engine operation at high loads.

Similarly, for smaller loads, an almost constant variation of the thermal efficiency is observed. The difference is that the efficiency is decreasing with load. Thus for 4 bars BMEP, the average thermal efficiency is 25%, for 3 bars BMEP that is 20%, and for 2 bars BMEP that is only 13%.

A peculiar thermal efficiency variation with BOI was obtained for 4 bars BMEP, where the quasi constant trend was replaced by a slight parabolic trend, with a maximum at 19 c.a. degrees BOI, and a minimum at 3 c.a. degrees BOI. However in all curves thermal efficiency was almost independent of BOI.

After making these preliminary tests (in terms of thermal efficiency) for the gas injection pressure it was decided to do all the following performance and emissions tests for 130, 150 and 170 bars GIP (gas injection pressure) for CNG (compressed natural gas). The gas injection pressure can be varied from a regulator upstream of the injector, and does not require any injector modification.

For determining which GIP (gas injection pressure) is optimum from the combustion process, thermal efficiency and emissions standpoint multiple tests were done with different GIP.
As Fig. 4.3.5. shows for 3 bar BMEP, the thermal efficiency is greater for 170 bars GIP than it is for 130 GIP.

The reason for choosing high CNG injection pressure is to have injection pressure high enough to overcome the pressure of the compressed air during the compression stroke, but at the same time not to over-penetrate the combustion chamber.

However, if the GIP is too low, the gas plume travel would be restricted to a short distance from the injector tip further into the combustion chamber.

---

\[\text{Thermal efficiency [%] vs. BOI [c.a.] for 3 bars BMEP at 130,150,170 bars GIP and 1200 rpm}\]

Figure 4.3.5. Thermal efficiency vs. GIP for 130, 150, 170 bars GIP, at 3 bars BMEP and 1200 rpm.\(^2\)

\(^2\) The graphs for 1 and 3 bars BMEP are shown in section 4.4.- Thermal Efficiency sub-chapter.
If the GIP is too high, that means the gas plume is traveling too fast and the penetration distance is long. The natural gas plumes that should be enflamed because of the diesel pilot plumes could either extinguish the pilot from behind or get enflamed very close to the piston crown or cylinder walls. This is known as wall impingement, and it is detrimental because of a heat transfer to the walls is higher that it should be. If the fuel is injected faster into the cylinder, the mixing-controlled (diffusive phase) combustion is getting faster, producing an increase of the flame temperature and thus higher NOx emissions and unburned mass fractions.

\[ 3 \text{ bar BMEP, 170 bar GIP} \]

\[ 3 \text{ bar BMEP, 130 bar GIP} \]

\[ 3 \text{ bars BMEP, 150 bars GIP} \]

**Figure 4.3.6.** The variation of NOx production with GIP's at different BOI's for 3 bars BMEP.

However the effect of increasing the natural gas injection pressure (GIP) is not strong. From Fig. 4.3.6. it can be seen that for 170 bar GIP and 25 c.a. degrees BTDC
BOI, the production of NOx is higher than for lower GIP's (i.e. 150 or 130 bars GIP) for identical BOI. This is due to higher injection pressure, which might produce higher turbulence, and thus more burning before TDC, hence higher temperature. The production of NOx (from 700 to 1900 ppm) is steadily increasing with the increase of the GIP (from 130 to 170 bars). The NOx dependency on BOI is strong. This is because the higher flame temperature is associated with high cylinder pressure and temperature strongly affects NOx.
4.4. Thermal Efficiency

From Fig. 4.4.1., 4.4.2., and 4.4.3 the thermal efficiency traces can be described as followed:

For 3 bars BMEP one can see that, regardless of the GIP (gas injection pressure) variation (i.e. 130 bars, 150 bars, 170 bars), the thermal efficiency for natural gas almost overlaps the thermal efficiency for diesel fueling. The variation range is 24 to 27%. For the diesel baseline, at 3 bars BMEP, the thermal efficiency is large (27%) for early BOIs and decrease slightly (up to 22%) for late BOIs. Among possible causes for the efficiency decrease with late BOI is incomplete combustion, or even misfiring.

For an engine load of 1 bar BMEP, the difference between the natural gas injection and diesel injection efficiencies is large. For all GIPs the thermal efficiency for CNG ($\eta_{th}$ for CNG is 16%) was about 40% greater than the efficiency for the diesel baseline ($\eta_{th}$ for diesel is 12%). The variation for both of them is almost constant.

For low loads, it can be concluded that the efficiency of the CNG fueling with the HPDI is superior to the conventional diesel injection. These results, correlated with the NOx emissions (which are lower for HPDI fueling) indicate the advantages of the HPDI fueling over diesel fueling.

For 5 bars BMEP engine load, one can observe that there is available data only for HPDI of CNG, because with conventional diesel fueling the engine couldn't provide 5 bars BMEP (i.e. cannot provide enough torque). The curvature of the thermal efficiency traces for CNG is almost linear with BOI. Making a comparison between 130, 150 and 170 bars GIP for thermal efficiencies (shown in Fig. 4.4.4.), it appears that maximum efficiency can be obtained for a BOI ranging from 11 c.a. to 15 c.a. degrees BTDC, which
Figure 4.4.1. Thermal Efficiency [%] vs. BOI for 1, 3, 5 bar BMEP, 130 bar GIP, CNG and 1, 3 bar BMEP, diesel @1200 rpm.
Figure 4.4.2: Thermal Efficiency [%] vs. TDI for 1,3,5 bar BMEP. 150 bar GIP, CNG and 1,3 bar BMEP. Diesel @ 1200 rpm.
Figure 4.4.3. Thermal Efficiency [%] vs. BOI for 1,3,5 bar BMEP, 170 bar GIP, CNG and 1,3 bar BMEP, diesel @1200 rpm.
Figure 4.4.4. Thermal efficiency variation with 130, 150 and 170 bars GIP, function of the engine load 1, 3, 5 bars BMEP, at 1200 rpm.
is in good correlation with the conclusions drawn based on Fig. 4.3.1 (dependency of PW with BOI). At 5 bars BMEP the values for thermal efficiencies were about 30%.

An interesting case is for 150 GIP, where the thermal efficiency for HPDI of CNG was closer to the efficiency for diesel fueling, than for 130 or 170 bars GIP. This fact suggests that at 150 bars GIP and 3 bars BMEP the best trade-off thermal efficiency, load and pulse width can be reached.
4.5. **NOx formation**

NOx represents the group of NO (nitric oxide) and NO\textsubscript{2} (nitrogen dioxide) which are formed within the engine, but the preponderant compound is NO.

The nitric oxide (NO\textsubscript{x}) is formed in high-temperature burned gas. The higher the burned gas temperature is the higher the rate of formation of NO\textsubscript{x} is. It is mainly formed in the last part of the combustion stroke.

The Zeldovich mechanism of NO formation shows how the molecule of nitrogen contained in the air (21 % oxygen, 79 % nitrogen) partially converts to NO:

\[
\begin{align*}
O + N_2 & \rightarrow NO + N \\
N + O_2 & \rightarrow NO + O \\
N + OH & \rightarrow NO + H
\end{align*}
\]

The rate of formation of NO\textsubscript{x} depends on species concentrations and on temperature. NO\textsubscript{x} forms in both the flame front and the postflame gases, but because the flame reaction zone is very thin (0.1 mm) and the residence time within the zone is short, the NO\textsubscript{x} formation in the postflame gases is a lot bigger than that produced in the flame front.

For straight diesel, according to Heywood ([34], page 586), in the premixed (or uncontrolled) combustion, the fuel-air mixture burns due to spontaneous ignition and flame propagation, whereas during the mixing controlled combustion (which follows) the mixture burns in a diffusion flame. It is the diffusion flame and the temperature changes due to compression of the already burned gases, which change the composition of any gas element. Mixture which burns early in the combustion process is very important since
it is compressed to a higher temperature, increasing the NO formation rate, as combustion proceeds and cylinder pressure increases.

Figure 4.5.1. presents the NOx formation for both HPDI and conventional diesel fueling, at 1, 3, 5 bars BMEP, and 130, respectively 1 and 3 bars BMEP. From Fig. 4.5.1. is seen that the NOx emission reduces with load and with injection delay.

For 1 bar BMEP, the HPDI fueling has a very slight variation of NOx formation, being almost constant at 500-600 ppm. The effect of BOI is very light, and for injection delays smaller than 15 c.a. degree (i.e. 15, 13, 11, 9, 7, 5 c.a. BTDC) there is no further decline of NOx. It is justifiable then to compare the NOx plots with the efficiency plots, to see which of them dictates the best operational regime. As is shown in Fig. 4.4.1 the thermal efficiency is almost constant for the whole range of BOIs at 1 bar BMEP. But for larger loads (i.e. 3 or 5 bars BMEP) the curves decline for late injection delay, starting with 11 c.a. degrees BTDC. From this perspective (maximizing thermal efficiency) it is beneficial to keep the injection delay at 11 c.a. BTDC or above.

As shown in Fig. 4.5.1 for 3 and 5 bars and HPDI fueling it can be seen a sharp and continue decrease of NOx formation from 25 c.a. BOI up to 5 c.a. BOI, before top dead center. Also it can be seen that is a very small difference between the NOx variation for 3 and 5 bars for the HPDI fueling. This indicates that past 3 bars BMEP, the load doesn't affect the production of NOx, which is beneficial for the HPDI technology.

The NOx concentration in the exhaust for the diesel fueling is almost double that for HPDI fueling for both comparable loads (i.e. 1 and 3 bars BMEP). The rate of NOx formation decreases with BOI in the same manner as for HPFI fueling. The impossibility of reaching higher loads with diesel fueling, prevents us from making a comparison with
Figure 4.5.1. NOx (wet) emissions vs. BOI for 1, 3, 5 bar BMEP, 130 bar GIP, CNG, and 1, 3 bar BMEP, diesel @1200 rpm.
corresponding NOx plots for HPDI of CNG. The potential of HPDI fueling of reaching higher loads (during the tests it was briefly reached even 6 bars BMEP) with practically maintaining the NOx emissions is very promising.

As is shown in Fig. 4.5.2, and 4.5.3., if the natural gas injection pressure (GIP) is increased (i.e. 150 bars, or 170 bars), the slope of the NOx emissions plots increases. In other words the injection delay (i.e. BOI is very late) has a bigger effect in reducing the NOx concentration. One particular change for higher GIP's is that the NOx concentrations for CNG and Diesel are very close, which indicates that a lower gas injection pressure is beneficial for reducing NOx formation. As indicated in Fig. 4.5.4, NOx emission reduces with increasing the engine load. In the same time for early injection delays the NOx production increases. (see Fig. 4.5.4.). From Fig. 4.5.5. it can be seen that for 130 bars GIP, the NOx production is the smallest. From the same figure one can see that the NOx is increasing from an average 30-40 to 60-70 [ g/kWh ] function of the load increase. (see Fig. 4.5.5.)

Summary:

To summarize this section, it can be said that NOx is very dependent on the temperature of the diffusion flame; it can be reduced by delaying the injection. For this engine the thermal efficiency does not strongly decrease as the BOI is delayed from 21 to 11 c.a. degrees. NOx emissions increase with increasing engine load. The trade-off between the NOx formation and thermal efficiency gives us a compromise regime for HPDI fueling operation. It can be concluded that for low loads the NOx emissions is not influenced by the BOI regardless of the GIP's, whereas for higher loads in order to reduce the NOx production a reduce of GIP's is required, as much as using late injection.
Figure 4.5.2: NOx (wt) emissions vs. BOI for 1.35 bar BMEP, CNG and 1.3 bar BMEP, diesel @ 1200 rpm
Figure 4.5.3. NOx (wet) emissions vs. BOI for 1, 3, 5 bar BMEP, 170 bar GIP, CNG and 1,3 bar BMEP, diesel @1200 rpm
NOx [ppm] variation with BMEP [bars] function of different BOI's for 130 bars GIP CNG and 1200 rpm.

Figure 4.5.4. NOx(wet) emissions versus engine load [bars BMEP] for the CNG HPDI fueling at 130 bars GIP and 1200 rpm.
Figure 4.5.5. Brake specific NOx [g/kWh] emissions vs. BOI for 130, 150, 170 bars GIP and 1, 3, 5 bars BMEP at 1200 rpm.
4.6. Carbon monoxide (CO)

Carbon monoxide emissions were monitored, analyzed and compared among HPDI and diesel fueling. The study was done for the lowest GIP (gas injection pressure), 130 bar, because it was observed previously a very little variation (less than 5%, according to figures 4.5.1, 4.5.2, 4.5.3 for NOx emissions and Fig. 4.4.1, 4.4.2 and 4.4.3 for thermal efficiency) of NOx and thermal efficiency for different GIP's (150 bars or 170 bars). It was concluded that one GIP analysis would reveal enough variation for CO at different injection delays (i.e. different BOI's).

**Figure 4.6.1. HPDI and diesel CO [ppm] variation for 130 GIP and different loads at 1200 rpm**
The engine was run for 1, 3 and 5 bars BMEP for the HPDI fueling and 1, and 3 bars for the diesel fueling (See Fig. 4.6.1). As is shown in Fig. 4.6.1, for very late BOIs (i.e. 5 and 7 c.a. BTDC) the concentration of the CO is high for both diesel and CNG fueling. In these regimes, the CO emissions are increasing from 150-170 ppm at high loads (5 bars BMEP) to 270-300 ppm for medium loads (3 bars BMEP). The maximum concentration of carbon monoxide with HPDI fueling is reached at low loads (1 bar BMEP) and very early injection timing.

For the same BOIs with diesel fueling, the values are three times larger, or more than those for HPDI fueling. The CO concentration is drastically increasing for 1 bar BMEP for conventional diesel injection, where figures are in the range of 1300-1400 ppm.

Once the BOI is further advanced (i.e. 9, 11, 13 c.a. degrees BTDC) the CO emissions reduces drastically. For these injection timings the CO concentration varies very little from 150 ppm to 220 ppm, depending on the engine load: at low loads CO concentration is higher than for high loads.

It is clearly seen that if the injection delay is too late, the natural gas cannot find and mix with the available air in a relative short time remaining for the combustion (i.e. before the scavenging process starts) producing CO.

Figure 4.6.2 shows the CO brake specific production [g/kWh] versus injection delay [BOI in c.a.] for 130, 150, 170 bars GIP [gas injection pressure] at 3 bars BMEP. It can be seen that the CO concentration increases very little with the GIP as the overall CO production is small.
Figure 4.6.2. Brake specific CO [g/kWh] vs. BOI for different GIPs [bars] for 3 bars BMEP and 1200 rpm.
In the present research, the level of CO was found to be almost constant for advanced BOI (i.e. from 25 to 15 c.a.) with an average value of 150-160 [ppm] regardless of the engine load or the fueling system. The only significant CO increases were for late BOIs and low loads for HPDI fueling, which however are far lower than the CO concentration for diesel fueling.
4.7. **Total (unburned) hydrocarbon (THC)**

The unburned hydrocarbons are the consequence of incomplete combustion of the hydrocarbon fuel. THC is expressed in parts per million carbon atoms [ppm] and is a measure of the combustion inefficiency.

The unburned gases are produced by the flame quench. When the flame quenches it leaves a layer of unburned gases ahead of the flame. Study of the causes for flame-quenching mechanisms could explain the THC formation. The flame could quench because of walls and piston crown configuration (small passages, or crevices).

For the DDC 1-71 research engine the flame should propagate at an angle to the cylinder walls, because the injector nozzle holes' are inclined 10 degree below the fire deck. The cold jet and the hot flame impinges the cylinder walls at a 80 degree angle and shouldn't be quenched by the walls (it is known that a flame may quench when entrance a thin volume, such as the region between the piston and the cylinder wall) but it could partially quenched because the jets are very close to the fire deck (10 degrees).

By relating the heat release within the flame to the heat loss to the walls (where quenching might occur) the formation of the unburned hydrocarbon can be studied. Figure 4.7.1 displays the unburned hydrocarbon (THC) for 1,3 5 bar BMEP for HPDI fueling, respectively for 1,3 bar BMEP diesel fueling, for 130 bar GIP, at 1200 rpm.
Figure 4.7.1. HPDI and diesel THC [ppm] variation for 130 GIP and different loads at 1200 rpm.

From Fig. 4.7.1. it can be seen that the variation of the THC [ppm] for HPDI fueling for a broad range of injection timings (BOI varies from 5 to 25 c.a.) and 3 loads BMEP, is relatively smooth. For HPDI fueling the THC concentration is almost constant (170 -180 ppm) for all three loads for a BOI varying from 25 to 15 c.a.

The difference start at 13 c.a. were the THC for 5 bars BMEP is less (190 ppm) than that for 3 or 1 bar BMEP (210 ppm). For CNG injection, the maximum value for THC emissions is reached at 390 [ppm] for low load (1 bar) and late injection (BOI = 5 c.a BTDC), whereas for similar conditions the conventional diesel injection produces
1250 [ppm]. Based on these graphs it could be concluded that the most unfavorable regimes are for low loads and very late injection, because they produce the most unburned hydrocarbon.

One other correlation can be done between the cylinder pressure for very late BOIs and THC formation. As was shown in Fig. 4.7.2 the effect of the late injection is that the pressure peak due to gas combustion is lower than the (see Fig. 4.7.2) ones for earlier BOIs.

Figure 4.7.2. Cylinder pressure for HPDI fueling, late injection (BOI = -5 c.a BMEP) at 130 bars GIP, 3 bar BMEP and 1200 rpm.
4.8. Summary

- The shape of the cylinder pressure-crank angle curve is different for the HPDI fueling than for the diesel fueling. The HPDI one is sharper than the diesel one right after the end of the compression stroke and beginning of injection. Because two different injectors (diesel and HPDI) with two different injection rising times\(^1\) have been used it appears that diesel cylinder pressure builds faster (is shifted toward the left side of the pressure vs. c.a. degrees BTDC graph) than that of the CNG fueling. 
  
In fact, the cylinder peak pressure varies with the BOI and the engine loads. For very late BOI the burning occurs very late, influencing negatively the cylinder pressure. 
The rate of pressure increase depends not only on the ignition delay but on injection delay (different injectors).

- Thermal efficiency is preserved for HPDI compared to that of the diesel fueling. The advantage of the natural gas fueling upon diesel fueling is that the engine is capable of more torque. For HPDI fueling the thermal efficiency varies from 22% to 32% for engine load varying from 1 to 5 bars BMEP for the same GIP. Also thermal efficiency varies with GIP from 22% (for 130 bars GIP) up to 30% (for 170 bars GIP) for different loads. The optimum thermal efficiency can be reached for a BOI varying from 9 c.a. to 21 c.a. degrees BTDC.

- The HPDI fueling reduces the NOx emissions by 50-55%, compared with diesel fueling. For 130 bars GIP NOx production is the lowest. The increase in GIP produce

\(^1\) The difference between the injection rising times is explained in subchapter 3.2.
an increase of NOx. Engine load influences NOx emissions (i.e. for low loads the emissions are 30-35% lower than that for higher loads).

The NOx concentration can be reduced by delaying the injection timing of CNG (i.e. BOI is retarded). It is practically lowest for the latest BOI that was used (5 c.a. degrees BTDC). But the injection timing cannot be indefinitely retarded because the combustion would be incomplete being shifted into the expansion stroke. A very late injection produces a decrease of the combustion products temperature, thus a decrease in NOx formation, but also an undesirable decrease of thermal efficiency.

- Sensible CO reduction for CNG fueling in respect with diesel fueling. For early BOI (25 c.a. to 15 c.a.) this phenomena is not of great importance. For late injection timing (15 to 9 c.a.), an almost 60-70% reduction occurs, going up to 90-100% for very late BOI (-5 c.a. BTDC). GIP has little influence (less than 5%) on CO emissions. The optimum BOI for getting least CO varies from 19 to 13 c.a. BTDC.

- Total unburned hydrocarbons (THC) are reduced by HPDI fueling, especially for low loads. Within the natural gas injection, the influence of the engine load on THC reduction is important just for late BOI (ranging from 9 c.a. to 5 c.a. BTDC).

- There is a trade off balance for GIP-BMEP-BOI and low NOx-THC-CO and high thermal efficiency. The best conditions are 130 bars GIP, 3 bars BMEP and (9 to 19 c.a.) BOI.

For low loads, high GIP and/or late injection delay the NOx, THC, and CO increase and thermal efficiency is less than optimum.
Chapter 5

MEASUREMENTS WITH DIRECT INJECTION CNG - HYDROGEN FUELING

5.1. Introduction

The focus of this part of the research is to externally mix the natural gas and the hydrogen at different percentage mole fractions and then directly inject this mixture into the diesel engine, using an HPDI injector.

Such high-pressure direct injection of hydrogen with diesel pilot hasn't been investigated before. A relative low gas injection pressure (100 bars GIP) was chosen for preliminary tests with hydrogen. This pressure is not optimum for HPDI technology, but it is a good starting point for noticing the changes induced by hydrogen injection on the engine performances and emissions. Encouraged by these results, the injection pressure was later increased to 130 bars GIP to provide a direct comparison with CNG data that were taken at the same GIP. It wasn't the pure hydrogen injection that we were interested in, but rather a mixture of natural gas and hydrogen. For this, we mixed different percentages of hydrogen with CNG. The hydrogen was injected at smaller mass fractions than natural gas (around 2 kg/h), starting with 0.25 kg/h and then increasing to 0.5 kg/h.

At 100 GIP, for an engine load of 1 bar BMEP, the PW required was approximately 9 crank angle degrees. This PW provided enough injection time for a 0.25 kg/h hydrogen mass flow but was not sufficiently long for a 0.5 kg/h hydrogen mass flow. At 0.5 kg/h hydrogen mass flow the engine couldn't be operated stable at 1200 rpm and 1 bar BMEP. Once the engine load was increased to 3 bars, the average pulse width needed was 15 crank angles, which was long enough for both 0.25 kg/h and 0.5 kg/h hydrogen mass flows.
For higher loads, such as 5 bars BMEP, the corresponding PW was found to be greater than 25 crank angle degrees, which is the maximum on the controller scale. In addition, for a pulse width of 25 c.a. degrees, the engine rpm couldn't be held constant at 1200 rpm as desired.

All the above reasons led to increasing the gas injection pressure to 130 bars. A test matrix of the points where data was taken is shown in table 5.1.

Table 5.1. The test matrix and the mass flows for 130 bars GIP, 3 and 5 bars BMEP.

<table>
<thead>
<tr>
<th>BOI</th>
<th>BMEP</th>
<th>Hydrogen</th>
<th>Average CNG mass flow</th>
<th>H2/CNG mass fraction</th>
<th>H2/CNG molar fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>[c.a. BTDC]</td>
<td>bars</td>
<td>[kg/h]</td>
<td>[kg/h]</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>9, 11, 13, 15, 17, 19</td>
<td>3</td>
<td>0.25</td>
<td>2.04</td>
<td>12.3</td>
<td>1</td>
</tr>
<tr>
<td>9, 11, 13, 15, 17, 19</td>
<td>5</td>
<td>0.25</td>
<td>2.11</td>
<td>11.8</td>
<td>0.9</td>
</tr>
<tr>
<td>9, 11, 13, 15, 17, 19</td>
<td>3</td>
<td>0.5</td>
<td>1.90</td>
<td>26.3</td>
<td>2.1</td>
</tr>
<tr>
<td>9, 11, 13, 15, 17, 19</td>
<td>5</td>
<td>0.5</td>
<td>2.08</td>
<td>24</td>
<td>1.9</td>
</tr>
<tr>
<td>9, 11, 13, 15, 17, 19</td>
<td>3</td>
<td>0</td>
<td>2.22</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>9, 11, 13, 15, 17, 19</td>
<td>5</td>
<td>0</td>
<td>2.29</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

As can be seen from table 5.1, two loads (3 and 5 bars BMEP), and two hydrogen mass flows (0.5 kg/h and 0.25 kg/h) were used for testing. The CNG mass flows decreased with decreasing engine load or with increasing hydrogen mass flows. The CNG values shown in table 5.1 were averages values obtained in each test. Mass fractions and molar fractions were calculated as was shown earlier in chapter 3.6 (preliminary calculations).
5.2. Cylinder Pressures

Cylinder pressure variation was compared for diesel, CNG and a mixture of hydrogen and CNG fueling. Initially, 100 bars were used as gas injection pressure (GIP) for both CNG and hydrogen. In Figs. 5.2.1 and 5.2.2, cylinder pressure traces are shown, for a mixture of 0.25 kg/h or 0.5 kg/h hydrogen and natural gas. From these figures one can see that the pressure peak is quite low, even for very early BOIs, and that is due to the low GIP (100 bars). The intent to start off with 100 bars GIP was to see if hydrogen injection will produce any changes in cylinder pressure. At 3 bars engine load the cylinder pressure was plotted for BOI ranging from 19 c.a. to 9 c.a. degrees BTDC for a 0.5 kg/h mass flow hydrogen and CNG mixture and compared with the corresponding ones for pure CNG.

The results are shown in Figs. 5.2.3 and 5.2.4, in which both cylinder pressure traces are plotted on the same graph. It can be seen that the effect of hydrogen was quite substantial, with an increase of the pressure peak of 20-25% in respect with that produced by CNG combustion. It can be also seen that while for early BOI the peak cylinder pressure increases for both CNG and CNG and hydrogen mixture, for late BOI, only the mixture of hydrogen with natural gas produces a sharp peak pressure increase.

These qualitatively good results were the incentive to inject the hydrogen and CNG at 130 bars GIP which, for CNG is associated with low emissions, good thermal efficiency and high torque. The cylinder pressure for different BOI for 3 and 5 bars BMEP, 0.25 kg/h or 5 kg/h H₂ and 2 kg/h CNG at 130 bars GIP and 1200 rpm is shown in Figs. 5.2.5 - 5.2.8.
Fig 5.2.1. Cylinder Pressure [kPa] for 3 bars BMEP, 100 bars GIP with 0.25kg/h $H_2$ and 2 kg/h CNG for different BOIs [c.a. degrees BTDC] @ 1200 rpm.
Fig 5.2.2. Cylinder Pressure [kPa] for 3 bars BMEP, 100 bars GIP with 0.5 kg/h H₂ and 1.88 kg/h CNG for different BOIs [c.a. degrees BTDC] @ 1200 rpm.
Comparison of cylinder pressure for 3 bars BMEP, 100 bars GIP, with 0.5 kg/h Hydrogen and CNG, and only with CNG @ 1200rpm for BOI = -9 c.a BTDC

Comparison of cylinder pressure for 3 bars BMEP, 100 bars GIP, with 0.5 kg/h Hydrogen and CNG, and only with CNG @ 1200rpm for BOI = -11 c.a BTDC

Comparison of cylinder pressure for 3 bars BMEP, 100 bars GIP, with 0.5 kg/h Hydrogen and CNG, and only with CNG @ 1200rpm for BOI = -13 c.a BTDC

Fig 5.2.3. Cylinder Pressure [kPa] for 3 bars BMEP, 100 bars GIP with 25% (0.5kg/h) H₂ and 2 kg/h CNG and CNG for 9, 11, 13 BOIs [c.a. degrees BTDC]@ 1200 rpm.
Comparison of cylinder pressure for 3 bars BMEP, 100 bars GIP, with 0.5 kg/h Hydrogen and CNG, and only with CNG @ 1200rpm for BOI = -15 c.a BTDC

Comparison of cylinder pressure for 3 bars BMEP, 100 bars GIP, with 0.5 kg/h Hydrogen and CNG, and only with CNG @ 1200rpm for BOI = -17 c.a BTDC

Comparison of cylinder pressure for 3 bars BMEP, 100 bars GIP, with 0.5 kg/h Hydrogen and CNG, and only with CNG @ 1200rpm for BOI = -19 c.a BTDC

**Fig 5.2.4. Cylinder Pressure [kPa] for 3 bars BMEP, 100 bars GIP with 25% (0.5 kg/h) H₂ and 2 kg/h CNG and CNG only, for 15, 17, 19 BOIs [c.a. degrees BTDC] @ 1200 rpm.**
From Figs. 5.2.5. and 5.2.7. (3 bars BMEP, for 0.25 kg/h H\(_2\) and 0.5 kg/h H\(_2\) mass flow) it can be seen that the cylinder peak pressure slightly decreases with the increase of the hydrogen mass flow, especially for very advanced BOI. However, from Figs. 5.2.6. and 5.2.8. (5 bars BMEP, for 0.25 kg/h H\(_2\) and 0.5 kg/h H\(_2\) mass flow) it can be seen that the quantity of hydrogen injected does not affect the pressure trace.

To illustrate the differences between the diesel, CNG and the mixture of hydrogen and CNG cylinder pressure traces were plotted for 3 bars BMEP, 130 G\(\text{ip}\), and 1200 rpm for BOI varying from 19 to 9 c.a. degrees BTDC, as shown in Figs. 5.2.9. - 5.2.10. From these figures one can see that the hydrogen injection increases the cylinder peak pressure compared to the CNG fueling. The amount of hydrogen introduced in the cylinder also makes a difference in cylinder peak pressure increase. For late BOI (Fig. 5.2.9.) the CNG injection fails to produce a cylinder pressure trace close to that of the hydrogen and CNG mixture injection. In contrast with the CNG fueling, the mixture of hydrogen and CNG produces a noticeable higher peak cylinder pressure. For early BOI (Fig. 5.2.10.) both hydrogen percentages produce an almost identical peak pressure, whereas the CNG peak pressures are smaller but very close to those produced by the mixture. Thus it appears that the addition of hydrogen to CNG considerably improves the burning rate.
Fig 5.2.5. Cylinder Pressure [kPa] for 3 bars BMEP, 130 bars GIP with 0.25 kg/h H₂ and 2.05 kg/h CNG for different BOI's (c.a. degrees BTDC) @ 1200 rpm.
Fig. 5.2.7: Cylinder Pressure (kPa) for 3 bars BMEP, 1.30 bars GIP with 0.5kg/h H₂ and 1.87 kg/h CNG for different BOIs (c.a. degrees BTDC)@1200 rpm.
Fig 5.2.8. Cylinder Pressure [kPa] for 5 bars BMEP, 130 bars GIP with 0.5 kg/h H₂ and 2.1 kg/h CNG for different BOIs [c.a. degrees BTDC]@ 1200 rpm.
Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and with 0.25 kg/h and 0.5 kg/h hydrogen and CNG at 1200 rpm for BOI = 9 c.a. BTDC

Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and with 0.25 kg/h and 0.5 kg/h hydrogen and CNG at 1200 rpm for BOI = 11 c.a. BTDC

Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and with 0.25 kg/h and 0.5 kg/h hydrogen and CNG at 1200 rpm for BOI = 13 c.a. BTDC

Figure 5.2.9. Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and 0.25 kg/h or 0.5 kg/h hydrogen and CNG mixture @ 1200 rpm for BOI = 9, 11, 13 c.a. BTDC.
Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and with 0.25 kg/h and 0.5 kg/h hydrogen and CNG at 1200 rpm for BOI = 15 c.a. BTDC

Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and with 0.25 kg/h and 0.5 kg/h hydrogen and CNG at 1200 rpm for BOI = 17 c.a. BTDC

Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and with 0.25 kg/h and 0.5 kg/h hydrogen and CNG at 1200 rpm for BOI = 19 c.a. BTDC

Figure 5.2.10.
Comparison of cylinder pressures for 3 bars BMEP, 130 bars GIP, CNG and 0.25 kg/h or 0.5 kg/h hydrogen and CNG mixture @ 1200 rpm for BOI = 15, 17, 19 c.a. BTDC
5.3. Injection Pulse Width

While the engine was run at 130 bars GIP on CNG, the direct injection of hydrogen was done intermittently (only for taking data). The mass flow percentage of hydrogen within the mixture can be either 0.25 kg/h or 0.5 kg/h in respect with CNG mass flow, which was adjusted according to the engine load or hydrogen mass flow. (see table 5.1.). Following the hydrogen injection, the engine speeds up for 10-15 seconds (for the same PW as was used for CNG) and then the rpm drops abruptly, so that the PW needs to be increased.

As was mentioned in section 4.3., the pulse varies with load, so one can expect to have large PW for high loads and small PW for low loads. Shown in Fig. 5.3.1. is the pulse width variation function of the engine load and the hydrogen percentage within the mixture.

From Fig. 5.3.1. one can see that for the same hydrogen mass flow, say 0.5 kg/h, the correspondent PW for high loads (5 bars BMEP) is bigger (with about 1 c.a.) than that for medium loads (3 bars BMEP) for a whole BOI range (from 19 c.a. to 9 c.a. degrees BTDC). Similar PW offset (about 1 c.a.) can be seen for lower hydrogen mass flow (0.25 kg/h) for the same BOI variation. When the pulse width (i.e. opening time for the injector needles) is large, more fuel (in this case hydrogen and natural gas mixture) could be injected in the engine. Thus, it makes sense that for higher loads the PW is increased.

From Fig. 5.3.1. it could be concluded that the pulse width for hydrogen and CNG mixture is larger than the CNG pulse width, for both hydrogen mass flow percentages, with about 18-20%. The fuel mass flow increases with pulse width, but is
Figure 5.3.1. Comparison of PW vs. BOI for 3, 5 bars BMEP, 130 bars GIP, for CNG, and 0.25 kg/h or 0.5 kg/h H2 and CNG mixture @1200 rpm.
not proportional to PW. As is shown in Fig. 5.3.2, the total pulse width is comprised of two independent injections, one for diesel the other for CNG. Here pulse width is understood to be the total PW. Thus the fuel mass flow depends on fuel molecular weight (MW) and polytropic coefficient ($\gamma$).

![Diagram of Pulse Width Injection for Diesel and CNG](image)

**Figure 5.3.2. Pulse Width Injection for Diesel and CNG.**

The hydrogen and CNG pulse width for high loads is 10% bigger than the pulse width for medium loads. Also from Fig. 5.3.1 it can be seen that the CNG pulse width trace decreases with late BOI, whereas the hydrogen and CNG mixture pulse width appears to increase with late BOI, especially for low hydrogen mass flows (0.25kg/h). The slope of the PW trace is lean, as the PW variation is about 2 c.a. for a whole BOI range (i.e. from 19 c.a. to 9 c.a. degrees BTDC). These trends can be observed for both high (5bars BMEP) and medium-low (3 bars BEP) loads.
5.4. Thermal Efficiency

As shown in Fig. 5.4.1 the calculated thermal efficiencies for 3 bars BMEP, 130 bars GIP and 0.25 kg/h or 0.5 kg/h hydrogen mass flow in a mixture with CNG are quite low compared with those of the CNG fueling (averaging 24-25%). It is seen that with increasing hydrogen mass flow within the CNG-hydrogen mixture (i.e. from to 0.5 kg/h) the thermal efficiency decreases from 15-16% corresponding to 0.25 kg/h, respectively to 12-13% corresponding to 0.5 kg/h.

To check whether the apparent drop in efficiency with hydrogen fueling could have been due to error in hydrogen measurement, a re-calibration of the hydrogen flow meter was performed, using compressed air through the micro-motion flow sensor (which registers mass flow rate independently of the molecular weight of the flowing gas). The results of this test which is documented in Appendix D3, indicates that, within experimental error the previously recorded hydrogen flow rates are correct.

To check whether the internal losses of the injector could have been radically different with CNG and CNG-plus-hydrogen fueling, an analysis was made of the internal stagnation pressure losses for both types of fueling and losses due to both sudden expansion from the needle valve port to the injector sac and due to friction choking of the injector nozzle passages. The analysis, whose results are provided in Appendix D2, shows that for a representative injector nozzle tip geometry the internal stagnation pressure losses are essentially the same for both CNG and CNG-plus-hydrogen fueling. The choked flow rates are significantly different, requiring the pulse width to be significantly longer for the latter case.
These two analyses, however, fail to explain why the efficiency apparently drops seriously with hydrogen addition to CNG; further work will be required to provide an explanation.

In terms of thermal efficiency variation with injection delay, it also can be seen from Fig. 5.4.1 that the hydrogen mass flow does not influence the thermal efficiency much, for most of the BOI range. For BOIs ranging from 17 to 13 c.a. degrees BTDC, the two corresponding thermal efficiencies for the two hydrogen mass flows have almost a linear variation with BOI ranging from 19 to 9 c.a. degrees. A slight drop in thermal efficiency can be seen, for 0.25 kg/h hydrogen mass flow, at 9 c.a. degrees BTDC. For the lower hydrogen mass flow (i.e. 0.25 kg/h) the thermal efficiency decreases a couple of percentages points for very late BOI (i.e. 9 c.a. degrees BTDC) or very early BOI (i.e. 19 c.a. degrees BTDC).

Figure 5.4.2. shows the thermal efficiency for 130 bars GIP, 5 bars BMEP, at 1200 rpm, and two hydrogen mass flows: 0.25 kg/h or 0.5 kg/h. As one can see from Fig. 5.4.2. for high loads (i.e. 5 bars BMEP), the hydrogen-CNG mixture still produces low thermal efficiency compared to that of the CNG (average of 29%).

Thermal efficiency has an average value of 23-24% for 0.25 kg/h hydrogen mass flow, and an average value of 17-18% for 0.5 kg/h hydrogen mass flow. Although at 5 bars BMEP, the thermal efficiency for both hydrogen mass flows is greater than the corresponding ones for 3 bars BMEP, they are still low compared with the CNG fueling. At 5 bars BMEP, a similar thermal efficiency decrease could be seen with increasing the hydrogen mass flow, as for 3 bars BMEP. Judging from Fig. 5.4.1. and 5.4.2., it can be
Figure 5.4.1. Thermal efficiency [%] vs. BOI for 3 bars BMEP, 130 bars GIP with CNG, 0.25kg/h H2 and 0.5kg/h H2 @ 1200 rpm.
Figure 5.4.2. Thermal efficiency (%) vs. BOI for 5 bars BMEP, 130 bars GIP, with CNG, 0.25 kg/h H2 and 0.5 kg/h H2 @ 1200 rpm.
estimated a decrease of thermal efficiency of 10-12 percentage points with the increase of the hydrogen flow.

For 130 bars GIP most of the thermal efficiency traces have a maximum for the BOI ranging from 17 c.a. to 13 c.a. degrees BTDC. As presented in Fig. 5.4.1 and 5.4.2., the thermal efficiency improves with a couple of percentage points for BOI ranging from 17 c.a. to 13 c.a. degrees BTDC. From Fig. 5.4.2 it can be concluded that for high loads (5 bars BMEP) the hydrogen injection improves the thermal efficiency substantially if is mixed in a sufficient percentage (25% in this case), whereas for medium-low loads (3 bars BMEP) the hydrogen fails to produce even same thermal efficiency as for CNG. Moreover, for 3 bars BMEP load, the changes in hydrogen percentage cannot improve the thermal efficiency so that to near the CNG thermal efficiency.

In other words, the mixture of hydrogen and CNG seems to decrease the thermal efficiency for both loads (i.e. 3 and 5 bars). The decrease is much larger than the estimated uncertainty of the efficiency calculations. This conclusion could be put in correlation with the fact that the pulse width has to be increased (8-10%) for hydrogen and CNG mixture, regardless of the hydrogen percentage.

For such a low thermal efficiency figures, the calculated uncertainties of thermal efficiency of 2-3 % (as it has been shown in chapter 3.6., and in Appendix 3.6.) become important.

**p-V diagram and indicated work.**

To calculate indicated work one needs to plot the p-V diagram.

\[ W_{\text{indicated}} = \int p\,dV \]
Measured cylinder pressures were recorded for each crank angle degree. To plot the p-V diagram one need to calculate volume.

The cylinder volume \( V \) at any crank position \( \alpha \) is (Heywood, page 43, eq. 2.4.):

\[
V = V_c + \frac{\pi B^2}{4} \left( l + a - s \right)
\]

where: \( V_c = \frac{V_d}{r_c - 1} \), is the clearance volume, \( V_c = 7.7467 \times 10^{-3} [m^3] \) for 1-71 DD engine.

\( V_d = 70.9 [in^3] = 0.001162 [m^3] \) is the total displacement volume for 1-71 DD engine.

\( r_c = 16 \), is the compression ratio for 1-71 DD engine.

\( B = 0.108 [m] \) is the cylinder bore for 1-71 DD engine.

\( l = 0.254 [m] \) is the connecting rod length for 1-71 DD engine.

\( a = \frac{L}{2} = \frac{0.127}{2} = 0.0635 [m] \), is the crank radius for 1-71 DD engine.

\( L = 0.127 [m] \), is the piston stroke for 1-71 DD engine.

\[
s = a \cos \alpha + \left( l^2 - a^2 \sin^2 \alpha \right)^{1/2} [m],
\]

is the distance between the crank axis and the piston pin axis.

\( \alpha \) [degrees] crank angles crank shaft.

\( \alpha \) varies from 0° to 360° (TDC = 180° crank angle, and BDC = 0° = 360°).

With these notations, (eq. 2.4, Heywood,) can be rewritten as:

\[
\frac{V}{V_c} = 1 + \frac{1}{2} \left( r_c - 1 \right) \left[ R + 1 - \cos \alpha - \left( R^2 - \sin^2 \alpha \right)^{1/2} \right]
\]

and \( R = \frac{l}{a} \) (3)

The volume \( V \) was calculated for \( \alpha \) varying from 0 to 360, in increment of one degree.

As is shown in table 5.4.1, for each volume \( V \), the pressure was considered constant.
The integral of indicated work \( W_{\text{indicated}} = \int p \, dV \) became a finite summation:

\[
W_{\text{indicated}} = \sum_{0}^{\Delta V} p\Delta V .
\] (4)

The p-V diagram for diesel, CNG and H\(_2\)-CNG mixture fueling is shown in Fig. 5.4.3.

The results for 13 c.a. degrees BTDC of beginning of injection, 3 bars BMEP, and 1200 rpm, are as follows:

**Table 5.4.1. Cylinder peak pressure and indicated work for three fueling methods for BOI= 13 c.a. degrees BTDC, 3 bar BMEP at 1200 rpm.**

<table>
<thead>
<tr>
<th>Fueling method</th>
<th>Cylinder peak pressure [kPa]</th>
<th>Indicated work [kJ]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>5815 kPa at 30 c.a. ATDC</td>
<td>99.8</td>
</tr>
<tr>
<td>CNG</td>
<td>5153 kPa at 34 c.a. ATDC</td>
<td>104.5</td>
</tr>
<tr>
<td>0.5kg/h H(_2)-plus-CNG</td>
<td>5495 kPa at 28 c.a. ATDC</td>
<td>95.7</td>
</tr>
</tbody>
</table>

From 5.4.1 one can see that even though the cylinder peak pressure for hydrogen-CNG mixture fueling is higher than that of CNG fueling, the indicated work is smaller than for the CNG. In fact the hydrogen-CNG mixture indicated work is slower than the diesel one. Nevertheless one can say that the cylinder peak pressure is reached faster with hydrogen-CNG mixture than for CNG, or diesel. That's due to high hydrogen flame speed, compared to CNG one. (See Fig. 5.4.3.)
Figure 5.4.3. $p$-$V$ diagram for Diesel, 0.5 kg/h H2-plus-CNG, and CNG at 3bars BMEP, BOI=13°c.a. BTDC and 1200 rpm.
5.5. Total hydrocarbon analysis. Comparison with the CNG fueling.

Hydrogen burning does not produce unburned hydrocarbon, because hydrogen doesn't contain any hydrocarbons. By injecting some hydrogen along with CNG into the engine, one should expect to have a decrease of THC.

Figure 5.5.1. displays the total hydrocarbon emissions for 100 bars GIP for both CNG and CNG and two hydrogen mass flows (0.25 kg/h and 0.5 kg/h) at 3 bars BMEP. For almost all the BOI range, the levels of THC emissions were very low, with an average of 50-60 ppm for 0.5kg/h hydrogen mass flow, and 80-90 ppm for 0.25kg/h. These values were 3-4 times lower than for CNG fueling for the same load (i.e.3 bars). For retarded BOI (i.e. 11 c.a. to 9 c.a. degrees BTDC) the THC produced by the hydrogen and CNG mixture is 14-15 times smaller than that for the diesel fueling. This ratio decreases abruptly to 3-4 times for early BOI (i.e. 19 c.a.). From Fig. 5.5.1. one could see a big reduction of the unburned hydrocarbon even for a small hydrogen percentage compared with the CNG or diesel fueling. Theoretically, the hydrogen injection has the potential of reducing completely the unburned hydrocarbons.

Shown in Fig. 5.5.2, for large gas injection pressure (130 bars GIP) and the same hydrogen percentage (0.25 kg/h), the THC traces have similar shape as for 100 bars GIP (shown in Fig. 5.5.1). The increase in the GIP facilitated us to run the engine at even higher loads (5 bars BMEP), for which the THC emission levels were lower than those for the 3 bars BMEP. At 3 bars BMEP, the THC emissions for CNG fueling are 2-3 times larger (ranging from 190 ppm to 250 ppm) than for the hydrogen and CNG mixture (ranging from 95 to 45 ppm). A THC concentration increase can be observed for CNG fueling for retarded BOI (i.e. 11 c.a. to 9 c.a. degrees BTDC), while for the same
Figure 5.5.1. THC (wet) vs. BOI for 3 bar BMEP, 100 bars GIP, for CNG, CNG and H2 and diesel @1200 rpm
Figure 5.5.2. THC (wet) [ppm] vs. BOI for 3 bars BMEP, 130 GIP, for CNG, 0.25 kg/h or 0.5 kg/h H2 and CNG mixture @1200 rpm
conditions the mixture of hydrogen and CNG produces a decrease of the THC emissions. Consequently, the THC emissions were the lowest for very late injection delay as the hydrogen was injected very late into the cycle.

From Fig. 5.5.3, one can see that the overall THC have increased with load increase. By comparing Fig.5.5.2. and 5.5.3. it can be seen that the THC emissions increased with 30-40 % from 3 bars BMEP to 5 bars BMEP. However, for 5 bars BMEP an increase in hydrogen percentage (0.5 kg/h from 0.25 kg/h, in this case) produces a further decrease in THC concentration (from a range of 140 - 90 ppm down to 130-80). This conclusion is in good agreement with the conclusions drawn from 100 bars GIP case. For any similar conditions (i.e. same loads, and same GIPs), the hydrogen and CNG mixture fueling produces considerably less THC emissions than CNG fueling. This is to probably due to the cylinder pressure traces which is quite different for CNG fueling than for the H2-CNG mixture fueling. The increase of the peak cylinder pressure for the hydrogen was due to the rapid pressure building, which can be correlated with the fact that the hydrogen burns faster than the CNG and requires a shorter combustion time.

The decreased unburned hydrocarbons with hydrogen addition seems inconsistent with the apparent decrease in efficiency and raises the question as to whether the decrease in efficiency may have been due to unburned hydrogen. This further emphasizes the need for future work to explain the effect of hydrogen addition to CNG on thermal efficiency.
Figure 5.5.3. THC (wet) [ppm] vs. BOI for 5 bars BMEP, 130 GIP, for CNG, 0.25 kg/h and 0.5 kg/h H2 and CNG mixture@1200 rpm
5.6. **Nitrogen oxide emissions analysis. Comparison with the CNG fueling.**

Reducing the nitrogen oxide (NOx) is one of the main concerns of the HPDI technology. By injecting a fraction of pure hydrogen along with CNG into the diesel engine, the NOx emissions tend to increase, because the hydrogen burns faster and at higher temperature than the natural gas.

As is shown in Fig. 5.6.1., for 100 bars GIP, 3 bars BMEP, the NOx concentration has increased for both hydrogen percentages (0.25 kg/h or 0.5 kg/h hydrogen mass flow), but not very much, compared to those of the CNG fueling.

The injection timing is influencing the decrease of NOx in the same manner for all fueling methods (diesel, CNG and hydrogen and CNG mixture) such that the NOx concentration can be drastically reduced by delaying the BOI (i.e. the NOx production is minimum for very late BOI, such 9 c.a. degrees BTDC). However, judging from Fig. 5.6.1, the NOx emissions are lower (by about 20% for advanced BOI and 5% for retarded BOI) for hydrogen and CNG mixtures than for diesel fueling.

By increasing the gas injection pressure to 130 bars GIP, one can see the NOx emissions variation in Fig. 5.6.2. It is known that higher injection rates (which depend on the gas injector nozzle hole area) and injection pressure (GIP) result in higher fuel-air mixing rates, hence the higher heat release, and probably higher NOx emissions. Because of the low density of the hydrogen, it is necessary to deliver a large volume (0.5 kg/h hydrogen mass flow is equivalent to 67% hydrogen molar fraction) of hydrogen to consume all the available air comprised in the cylinder. Since the injector used hasn't had its gas holes enlarged to accommodate more hydrogen fuel, the PW required was
Figure 5.6.1. NOx (wet) vs. BOI for 3 bar BMEP, 100 GIP, with CNG, H2 and CNG, and diesel @1200 rpm
Figure 5.6.2. NOx (wet) [ppm] vs. BOI for 3 bars BMEP, 130 bars GIP, with CNG, 0.25 kg/h and 0.5 kg/h H2 and CNG mixture @1200 rpm.
increased. As one can see from Fig. 5.6.2., for larger hydrogen mass flow (0.5 kg/h) the 
NOx emissions increases with 20% compared to those of lower hydrogen mass flow 
(0.25kg/h mass flow). Also, NOx is drastically influenced by the injection timing, and 
can be decreased almost 40% by delaying the injection timing. Probably for an injection 
very close to TDC, the NOx emissions would be very low, but the thermal efficiency 
would not be maximum. For 3 bars, and a BOI ranging from 19 c.a. to 15 c.a. degrees 
BTDC, the hydrogen and CNG mixture produces less NOx emissions than the 
corresponding CNG fueling.

As it can be seen from Fig. 5.6.3., for 5 bars BMEP, and 0.25 kg/h hydrogen mass 
flow, the NOx concentration is lower for the CNG-hydrogen mixture than for CNG 
fueling. To further investigate the hydrogen influence, its percentage was increased to 0.5 
kg/h mass flow.

As shown in Fig. 5.6.3., the effect of increasing hydrogen percentage is an 
increase in NOx production, by roughly 30%. It is more accentuated for late BOI (i.e. 9 
c.a. degrees BTDC) where the difference is 35-40 % in favor of the smaller hydrogen 
percentage. The NOx variation with injection timing follows the same trend as for CNG 
fueling (i.e. NOx emissions decrease with more retarded BOI). From Fig. 5.6.3. it can be 
concluded that, by increasing the percentage of hydrogen one might expect to have an 
increase in NOx production.

However, in all cases, the NOx production can be controlled (i.e. decreased) by 
properly adjusting the injection timing (i.e. late BOI).
Figure 5.6.3. NOx (wet) [ppm] vs. BOI for 5 bars BMEP, 130 bars GIP, with CNG, 0.25 kg/h and 0.5 kg/h H2 and CNG mixture @1200 rpm.
5.7. Carbon monoxide analysis. Comparison with the CNG fueling.

In general, the carbon monoxide (CO) emissions for the hydrogen and CNG fueling are not reduced beyond what the CNG fueling can produce. As shown in Fig. 5.7.1., the preliminary tests with 100 bars GIP and 0.25 kg/h or 0.5 kg/h hydrogen and an average 1.9 kg/h CNG, indicate that CO levels for 0.5 kg/h hydrogen are 40-60% greater than those for CNG fueling, whereas for 0.25kg/h hydrogen flow the CO emissions are more than double compared with the same CNG-fueling. The CO dependency with BOI could be explained by the available time for the existing air to combine with the injected fuels, such that a complete combustion could take place. Should the injection delay be retarded, the gaseous fuels may not have time to combine with the oxygen available in the engine cylinder so that more carbon monoxide would be produced. One can see from Fig. 5.7.2. that for 130 bars GIP, 3 bars BMEP and for low hydrogen mass flow (i.e. 0.25kg/h), the CO production is somewhat greater for CNG fueling than for the hydrogen and CNG mixture. If the hydrogen percentage is increased, for the same load, the CO production is almost the same as for CNG fueling, especially for late BOIs (i.e. from 13 to 9 c.a. degrees BTDC). For more advanced BOI (i.e. 19 c.a. up to 13 c.a. degrees BTDC) and 3 bars BMEP, the CO production is almost within the same range for both CNG and hydrogen-CNG mixture.

As shown in Fig. 5.7.3, for greater loads (i.e.5 bars BMEP) the departure of the hydrogen and CNG mixture CO trace from that of the CNG fueling is even larger. For 0.25kg/h H2 , the CO emissions vary from 220 to 320 ppm for a BOI ranging from 19 to 9 c.a. degrees BTDC. For the same rpm, load and BOI range, the CO emissions produced by the CNG fueling vary from 210 to 360 ppm. For any of the fueling methods, it can be
Figure 5.7.1. CO (wet) vs. BOI for 3 bar BMEP, 100 bars GIP, with CNG, H2 and CNG mixture, and diesel @1200 rpm
Figure 5.7.2. CO (wet) [ppm] vs. BOI for 3 bars BMEP, 130 bars GIP, for CNG, 0.25 kg/h and 0.5 kg/h H2 and CNG mixture @1200 rpm.
seen that for late BOIs, the CO is increasing, as the combustion is not complete. Figure 5.7.3 shows the influence of the hydrogen percentage (0.25 kg/h or 0.5 kg/h) for high loads (5 bars BMEP). By increasing the hydrogen percentage, the CO concentration increased by almost 10-15% compared to the CNG fueling.

It can be concluded that although it is not increasing greatly, the CO production for hydrogen and CNG is higher than with CNG fueling. It appears that by further increasing the hydrogen percentage could produce more CO.
Figure 5.7.3. CO (wet) [ppm] vs. BOI for 5 bars BMEP, 130 bars GIP, for CNG, 0.25 kg/h and 0.5 kg/h H2 and CNG mixture @1200 rpm
5.8. Summary

- For 130 bars GIP, 3 and 5 bars BMEP, at 1200 rpm, the hydrogen injection increases the cylinder pressure peak by 20-30% as a function of the hydrogen percentage (i.e. 0.25 kg/h or 0.5 kg/h). Also, the shape of the cylinder pressure is sharper than that obtained with CNG fueling. For similar engine loads, rpm and BOIs, the pulse width of hydrogen-CNG mixture fueling is larger than that for CNG fueling with roughly 2 - 3 c.a. degrees. The different hydrogen percentages (i.e. 0.25 kg/h or 0.5 kg/h) influence the peak cylinder pressure (i.e. for greater percentage the cylinder peak pressure increases). The injection timing influences the maximum peak pressure, such that by increasing the gas injection pressure (GIP) for both hydrogen and CNG mixture and for CNG, the peak pressure increases. It appears that with hydrogen the ignition delay is smaller than for CNG fueling.

- With hydrogen-CNG mixture fueling the thermal efficiency decreased drastically. The cause is not yet explained, and cannot readily be explained as due to increased internal losses through the injector, which could produce an increased stagnation pressure drop. The plotted data shows a thermal efficiency of 16% for 0.25 kg/h hydrogen flow, respectively 13% for 0.5 kg/h hydrogen flow, at 3 bars BMEP, which are very low compared with the CNG thermal efficiency of 24% for the same conditions. At high loads (5 bars BMEP) the 0.25 kg/h hydrogen mass flow thermal efficiency is 22-23%, respectively 18% for 0.5 kg/h, which are low compared with that of CNG fueling (28-29%). At low hydrogen mass flow (0.25 kg/h) the thermal efficiency is higher than for that of a high hydrogen mass flow (0.5 kg/h).
influence of the hydrogen percentage is considerable, especially for low loads. The thermal efficiency for hydrogen-CNG fueling is maximum for a BOI ranging from 19 c.a. to 13 c.a. degrees BTDC.

- The hydrogen and CNG mixture fueling reduces drastically (3 or 4 times compared to the CNG fueling) the unburned hydrocarbons (THC). This trend is even more accentuated for higher hydrogen percentage into the mixture. The THC production varies with the load, such that for higher loads the THC emissions increases. THC can be controlled with injection delay, such that for very late injection THC emissions timing for CNG+H2 be reduced. The THC reduction is substantial (15 times or more for very advanced BOI) if we compare the diesel fueling with hydrogen and CNG mixture fueling. Again this comparison should be made for the whole BOI range (i.e. for late BOI, the reduction is about 2 times).

- With hydrogen and CNG mixture fueling the increase in NOx emissions was about 20% for advanced BOI and about 5% for retarded BOI. The increase in NOx production is mainly due to higher adiabatic flame temperature for hydrogen than for CNG. As a confirmation, the NOx increased with increasing the hydrogen mass flow. However, the NOx emissions obtained with hydrogen and CNG were less than for diesel fueling. More advance hydrogen and CNG mixture timing produced lower NOx. The engine loads affect the NOx production (higher loads produce less NOx). By appropriately adjusting the BOI, one can obtain a reduced NOx concentration, for a given load or GIP. An increase in GIP has little effect on NOx production for both gaseous fueling methods
The hydrogen and CNG mixture fueling has an impact on CO reduction in respect with CNG fueling. If for early BOI (19 c.a. to 13 c.a: degrees BTDC) the CO levels are almost constant (around 200 ppm) for both CNG and hydrogen and CNG mixture fueling, for late injection timing (13 to 9 c.a.), an almost 90% reduction occurs. The GIP has little influence (less than 5%) on CO emissions. The optimum BOI for getting least CO varies from 19 to 13 c.a. BTDC.
CONCLUSIONS AND FUTURE RECOMENDATIONS

6.1. Conclusions

Based on the investigation of the performance and emissions of single-cylinder two-stroke diesel engine with CNG and hydrogen-CNG mixtures, the following conclusions can be made:

- With CNG fueling the engine load capability increased with about 20% compared to diesel fueling. The hydrogen-CNG fueling engine load capability is almost the same as the CNG fueling over a wide load range.
- For advanced injection, the peak cylinder pressure of hydrogen-CNG fueling is 10% larger than that of the CNG fueling. For retarded injection the peak cylinder pressure of hydrogen-CNG is as low as that for CNG fueling. Hydrogen mass flow increase produces a peak cylinder pressure increase for the same injection timing.
- While the thermal efficiency for CNG is about the same as for diesel fueling for low and medium loads, at higher loads it may be as much as 1.10 times (see Fig. 4.4.1). An increase in gas injection pressure (GIP) produces a slight increase of thermal efficiency. With 25% mass flow percent hydrogen in natural gas the thermal efficiency based on lower heating values is about 0.6 times than that of CNG fueling. Because the hydrogen has a relatively low density, the volumetric flow rate and the injection pulse width are larger than with CNG, at a given load and speed.
- The NOx emissions are reduced by 50-55% by using CNG instead of diesel, for all loads. A further reduction can be reached by delaying the injection timing. For the hydrogen-CNG mixture, the NOx is lower than that of diesel level, but 10-15% higher compared to CNG fueling. Injection delay helps reducing the NOx for a hydrogen-CNG mixture fueling, too. Increasing the hydrogen percentage produces an increase in NOx emissions. A GIP (gas injection pressure) increase produces a NOx emissions increase for both pure CNG and hydrogen-CNG mixture fueling.

- The total unburned hydrocarbon (THC) is much reduced with hydrogen-CNG fueling. If the reduction is 50-60% for CNG fueling compared to that of the diesel fueling, by injecting the hydrogen-CNG mixture the reduction as much as 90%, even more for late injection timing. The THC can be reduced to almost zero by injecting pure hydrogen.

- The carbon monoxide emissions produced by the hydrogen-CNG mixture fueling are similar to those produced by CNG fueling for the whole load range. A slight increased is observed for late injection. No noticeable effect was seen for the increase of the hydrogen mass flow.
6.2. **Future work recommendations**

- Further study is needed to determine the cause of reduced efficiency with addition of hydrogen to the CNG fuel supply. The role of hydrogen in reducing emissions of total hydrocarbons (and probably particulate matter) should be further studied if thermal efficiency does not need to be sacrificed.

- The investigation should be extended to pure hydrogen fueling. That means either modifying the injector nozzle holes diameter, or changing the pulse width from the ECU, or both.

- A study of the gas injection pressure (GIP) increase influence on the engine performance and emissions is desirable. A possible way of increasing the GIP is to compress both hydrogen and CNG gases with the same intensifier.

- Attention should be given to lowering the hydrogen combustion temperature, which could reduce the NOx emissions, making them comparable with those of pure CNG injection (which are low compared to those of diesel fueling).
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Practical measurements references:


Appendix A1

Emission Standards for USA and Canada for heavy-duty truck and bus engines

Model year 2004 and later
In October 1997 EPA adopted a new emission standard for model year 2004 and later heavy duty diesel engines used in trucks and buses. The standards reflect the provisions of the Statement of Principles (SOP) signed in 1995 by the EPA, California-ARB, and the leading manufacturers of heavy-duty diesel engines. The goal is to reduce NO\textsubscript{x} emissions from highway heavy-duty engines to levels approximately 2.0 [g/bhp*hr] beginning in 2004. Manufacturers will have the flexibility to certify their engines to one of the two options given in Table 1. Emissions are measured over the US Transient FTP cycle.

Table 1
EPA Emission Standards for MY 2004 and Later Heavy-Duty Diesel Engines, [g/bhp*hr]

<table>
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<th>Option</th>
<th>NMHC + NO\textsubscript{x}</th>
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All emission standards other than NMHC and NO\textsubscript{x} applying to 1998 and later model year heavy duty engines (Table 2) will continue at their 1998 levels. The standards will be subject to a review process in 1999 which may either tighten or relax the above emission limits.

EPA established a revised useful life for the heavy duty diesel engine service class of 435,000
miles, 22,000 hours, or 10 years, whichever occurs first, for all pollutants beginning in model year 2004.

The new EPA standards are expected to be adopted by California, which has the authority to set its own vehicle emission standards, so manufacturers will be able to use a single engine or machine design for both markets.

Model year 1987-2003

The MY 1987-2003 US federal (EPA) and California (ARB) emission standards for heavy duty diesel truck and bus are summarized in Table 2 and Table 3. All emissions are tested using the US Transient FTP and expressed in \( \text{[g/bhp\&[g/bhp\*hr]} \)

Table 2

EPA Emission Standards for Heavy-Duty Diesel Engines, \([\text{g/bhp}\*\text{hr}]\)

<table>
<thead>
<tr>
<th>Year</th>
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<th>NOx</th>
<th>PM</th>
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<table>
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* - in-use PM standard 0.07
Table 3
California Emission Standards for Heavy-Duty Diesel Engines, [g/bhp*hr]

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<th>NOx</th>
<th>PM</th>
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Appendix A2

U.S. EPA, California ARB plans for heavy duty diesel engine emission standard

(Date: October 7, 1999)


Environmental regulatory authorities in the U.S. revealed their planned new emission standards for new heavy-duty highway engines, which target emission levels of about 0.5 g/bhp-hr for oxides of nitrogen (NOx) and 0.01 g/bhp-hr for particulate matter (PM). These plans were presented by the federal Environmental Protection Agency (EPA) and the California Air Resources Board (ARB) during the SAE Clean Diesel TOPTEC, held last week in Florida, and during the “Exploring New Technologies For Clean Air” symposium in California earlier this week.

The new standard is expected to require exhaust gas after-treatment technologies, including diesel particulate traps and NOx trap/catalyst systems. Ultra low sulfur diesel (ULSD) fuel will play a key “technology enabler” role in making the standard feasible. ULSD fuels improve the emission reduction effectiveness and durability of diesel oxidation catalysts and are expected to enable future, advanced emission controls.

Diesel fuel sulfur levels of between 5 - 40 ppm are considered by the EPA, while the ARB pointed at the 30 ppm mark. Current U.S. standards allow for up to 500 ppm of sulfur in highway diesel fuel. The final fuel sulfur specification will depend on after-treatment technology needs for NOx and PM control, refinery production technology feasibility, costs, ability to maintain fuel quality in the distribution system, and testing tolerances.
Ultra low sulfur diesel regulation for HD engines would be also beneficial for light-duty vehicles, which need to use the same after-treatment technologies to meet the proposed Tier 2 standard. EPA sees a potential for 70-90% reduction in PM emissions and 60-90% reduction in NOx emissions, relative to today’s standards. That translates to about 0.01 g/bhp-hr PM standard, down from current 0.1 g/bhp-hr, and about 0.5 g/bhp-hr NOx standard, down from today’s 4 g/bhp-hr (about 2 g/bhp-hr in 2004). Both standards are considered by the EPA as forcing exhaust gas after-treatment in heavy-duty diesel engines.

The Clean Air Act requires a 3-year stability period between new emission standards. Since the latest adopted HD diesel emission standard comes to power in 2004, the earliest introduction of the new federal standards is possible in the year 2007. The EPA expects a phased-in beginning of the standard in 2007. In California, the ARB Chairman Alan Lloyd has directed the ARB staff to develop a program to reduce emissions from new heavy-duty diesel engines to the above levels beginning in 2007.

A Notice of Proposed Rule Making (NPRM) on the new federal HD standards is expected to be published in early 2000.

Both the EPA and CARB indicated they would also regulate substantial reductions in PM emissions from non-road diesel engines. The federal non-road engine regulation, expected in 2001 (ANPRM in early 2000), would also consider a non-road fuel quality standard similar to the current highway requirement of 500 ppm sulfur cap.
California adopts new emission standards for urban buses

Source: California ARB, February 25, 2000

The California Air Resources Board (ARB) adopted a new regulation to reduce emissions from the state’s transit buses. The bus rule, which will be phased-in beginning in 2002, affects about 8,500 urban buses at approximately 75 California transit agencies.

Various components of the regulation, to be introduced over the next ten years, focus on the reduction of particulate matter (PM) and nitrogen oxides (NOx) emissions. Diesel exhaust PM contributes to mortality and contains substances known to cause cancer, while NOx contributes to ozone, the main harmful component of urban smog. The rule calls for cleaner engines, cleaner diesel fuel, retrofit with diesel particulate filters to reduce exhaust PM emissions from older diesel buses, use of zero emission buses (ZEB) and reduced exhaust PM and NOx from new diesel engines.

The regulation allows transit agencies the flexibility of choosing between either a diesel or alternative fuel “path” to lower air emissions. Agencies may choose to use low-emission alternative fuels such as compressed or liquefied natural gas, propane, methanol, electricity, fuel cells or other advanced technology.

Continued use of diesel brings with it a requirement to use low-sulfur (15 ppm) diesel fuel beginning July 1, 2002, and cut PM emissions from new diesel buses down to 0.01 g/bhp-hr beginning in 2004. A new lower NOx standard of 0.2 g/bhp-hr applies to both diesel and alternative fuel bus engines sold to California transit agencies starting in 2007.

Today’s bus engine standards are 0.05 g/bhp-hr for PM and 4.0 g/bhp-hr for NOx.
Appendix B1:
A typical piezoelectric calibration sheet for a pressure transducer is shown.

PCB PIEZOTEKNICS INC.

CALIBRATION CERTIFICATE

Model: X112A04
Serial #: 17215
Description: Pressure Sensor Station: Bomb'e Huille
Type: Charge

Sensitivity: 0.948 pC/PSI
Linearity: 0.46% FS

Date: 7/31/98
By: Chris Talebkhan
Cert #: 11572

Notes:
1. Calibration is traceable to NIST up to 15000psi static and complies with ISO 10012-1 and former MIL-STD-45662A.

PCB PIEZOELECTRONICS, INC.

Picocoulombs vs. Pressure [psi]

3425 WALDEN AVENUE, Depew NY 14043
Tel: 716-684-0001 Fax: 716-684-0987
e-mail: sales@pcb.com Web: www.pcb.com
Appendix B2:

Pressure transducer calibration analysis

The use of different types of pressure transducers can produce different in-cylinder pressure readings. Also the location where the pressure is mounted in the combustion chamber can affect the absolute pressure readings. Randòlph [46], Hodgins and Mtui [47], and others have done detailed studies in this direction.

Hodgins and Mtui [47] tested 5 different piezoelectric pressure transducers (4 were made by PCB, and one was a water cooled AVL made) to measure the cylinder pressure for a single-cylinder Detroit Diesel engine (1-71DDC). For each of the pressure sensors the signal was amplified using the same charge amplifier (Kistler 5001), with the identical settings. An ISAAC fast data acquisition was used to acquire the pressure signals, together with the bottom dead center (BDC) signal and the crank angle signal from an optical encoder (SoftPot). They used an averaged cylinder pressure for 20 consecutive cycles. The indicated mean effective pressure was calculated as:

\[ IMEP = \frac{W_{\text{indicated}} [\text{bar}]}{V_{\text{cylinder}}} \]

where \( W_{\text{indicated}} \) is the indicated work as represented by the area bounded by the p-V diagram curve, and \( V_{\text{cylinder}} \) is the displaced volume of the cylinder.

The coefficient of variability (COV), defined as the standard deviation in IMEP for each cycle divided by the IMEP was calculated to assess the degree of cycle-to-cycle variation of pressure signals:
Their results at 3 bars BMEP, and 1200 rpm indicated a large discrepancy in peak pressure indicated by the piezo-transducers\(^{(1)}\). As shown in Fig. B2, there is a relative cylinder peak pressure difference of about 1600 kPa between AVL (water-cooled) sensor and PCB A10. The difference represents 20% of the maximum cylinder pressure.

\[
COV_{\text{IMEP}} = \frac{\sigma_{\text{IMEP}}}{\text{IMEP}_{\text{avg}}} \cdot 100\%.
\]

---

\(^{(1)}\) they used PCB 112M297a, PCB112M297b, PCB112M298, PCB112A10 and AVL8QP500c (water-cooled).

---

**Figure B2.** Cylinder pressure vs. c.a. [degree ABDC] for 5 bars BMEP, 1200 rpm and different pressure transducers.
The cause of discrepancies could be from either a calibration error (sensitivity values differ from those given in the calibration sheets) or thermal effects (induced by high temperatures during combustion). Transducer 112M297a appeared to have the lowest thermal effect. The lowest COV (1.24%) of all transducers used was found for PCB 112M297b. Hodgins and Mtui concluded that the measured cylinder pressure with depend on the pressure transducer used. They based their analysis on a comparison between the calculated cylinder pressure at the end of compression stroke with the measured cylinder pressure for the same conditions (end of compression stroke, which is TDC). The cylinder pressure at TDC can be calculated as:

\[ p_a V_d \gamma = p_c V_c \gamma, \]

where \( V_d \) is the volume displacement, \( V_c \) the combustion chamber volume, \( p_a \) the airbox pressure, and \( p_c \) is the pressure at the end of compression stroke. \( \gamma \) is the polytropic coefficient, which is found in literature (\( \gamma = 1.4 \)).

Figure B2b shows how the pressure transducer was mounted. On the original cylinder head the pressure sensor was placed 1.25 inches from the injector tip. This dimension was kept identical for the modified cylinder head, such that no effects on the cylinder pressure measurements would occur.

Studies of the cylinder pressure sensor mounting were done by Randolph [46], who pointed out that a maximum 10% difference could occur from mounting the sensor in different locations. The signal drifting could be also spread on as much as 8-10 crank angle degrees just because of the mounting.
Figure B2b. Pressure transducer mounting in cylinder head of 1-71 DDC engine.
Appendix B3

Airbox temperature calibration

Date: August 1999

For measuring the intake air temperature a thermocouple has been used. To read the temperature in degrees Celsius a voltage-temperature conversion had to be made. The voltage reading variations were linear.

Table B3 Airbox temperature calibration

<table>
<thead>
<tr>
<th>Measurements</th>
<th>AIR</th>
<th>WATER</th>
</tr>
</thead>
<tbody>
<tr>
<td>Voltage [V]</td>
<td>0</td>
<td>3.122</td>
</tr>
<tr>
<td>Thermometer [°C]</td>
<td>-29.82</td>
<td>21.5</td>
</tr>
<tr>
<td>Digital reading [°C]</td>
<td>-29.82</td>
<td>20.2</td>
</tr>
</tbody>
</table>

The slope of the line is:

\[
\text{Slope} = \frac{98.5 - 20.2}{7.975 - 3.122} = 16.13
\]

and interception with axis is at -29.82 (see figure B3)

Figure B3. Airbox voltage vs. thermocouple temperature
Appendix B4:

Micro Motion Mass Flowmeter Summary

Date: 29 September, 2000

Process Operating Conditions: Gas

Fluid Name: Natural Gas (SG = 0.667)

Maximum Rate : 15 kg/h

Maximum Pressure: 5000psi  Maximum Temperature: 75 degree F.

Viscosity: 0.01cP

<table>
<thead>
<tr>
<th>Meter Model #</th>
<th>Flow ± % Rate</th>
<th>Density ± g/cc</th>
<th>Pressure Drop psi</th>
<th>Velocity ft/sec</th>
<th>Reynolds Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>*DH012S</td>
<td>2.46</td>
<td>0.032</td>
<td>0.9</td>
<td>11.5</td>
<td>224577</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Rate [kg/hr]</th>
<th>Mass Flow [%] rate</th>
<th>Pressure Drop [psi]</th>
<th>Velocity [ft/sec]</th>
<th>Reynolds Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>15.00</td>
<td>2.5</td>
<td>0.9</td>
<td>11.5</td>
<td>224577</td>
</tr>
<tr>
<td>13.50</td>
<td>2.7</td>
<td>0.7</td>
<td>10.4</td>
<td>202160</td>
</tr>
<tr>
<td>12.01</td>
<td>2.9</td>
<td>0.6</td>
<td>9.2</td>
<td>179743</td>
</tr>
<tr>
<td>11.30</td>
<td>3.1</td>
<td>0.5</td>
<td>8.7</td>
<td>169181</td>
</tr>
<tr>
<td>10.51</td>
<td>3.2</td>
<td>0.4</td>
<td>8.1</td>
<td>157326</td>
</tr>
<tr>
<td>9.01</td>
<td>3.7</td>
<td>0.3</td>
<td>6.9</td>
<td>134909</td>
</tr>
<tr>
<td>7.51</td>
<td>4.3</td>
<td>0.2</td>
<td>5.8</td>
<td>112492</td>
</tr>
<tr>
<td>6.02</td>
<td>&gt;5</td>
<td>0.2</td>
<td>4.6</td>
<td>90075</td>
</tr>
<tr>
<td>4.52</td>
<td>&gt;5</td>
<td>0.1</td>
<td>3.5</td>
<td>67658</td>
</tr>
<tr>
<td>3.02</td>
<td>&gt;5</td>
<td>0.0</td>
<td>2.3</td>
<td>45241</td>
</tr>
<tr>
<td>1.52</td>
<td>&gt;5</td>
<td>0.0</td>
<td>1.2</td>
<td>22824</td>
</tr>
<tr>
<td>0.03</td>
<td>&gt;5</td>
<td>0.0</td>
<td>0.0</td>
<td>407</td>
</tr>
</tbody>
</table>
Figure B4. "Micromotion" relative error estimates [ %] for a flow ranging from zero to maximum.
Appendix B5

"SoftPot" optical encoder schematics and electrical interface is presented here:

![Figure B5a. US Digital optical encoder](image)

US Digital Corporation @11100 NE 34th Circle, Vancouver, WA 98682-8714, USA.

Electrical interface

![Figure B5b. Pull-up resistor on HEDS-9X40 encoder.](image)

To insure reliable encoding performance, the HEDS-9040 and 9140 three channel encoder modules require 2.7 kΩ (± 10%) pull-up resistors on output pins 2, 3 and 5 (channel I, A and B) as shown in figure B2b. These pull-up resistors should be located as close to the encoder module as possible (within 4 feet). Each of the three encoder module outputs can drive a single TTL load in this configuration.
### Appendix B6: Start up procedure for emission bench analyzers

<table>
<thead>
<tr>
<th>Step</th>
<th>Procedure</th>
<th>Additional Information</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Open gas cylinders</td>
<td>Compressed air, H₂, CH₄, CO, CO₂, NO, N₂, HC. Do not adjust regulators on cylinder tops.</td>
</tr>
<tr>
<td>2</td>
<td>To select a heated sample stream:</td>
<td>Turn the selected engine (Ricardo) dial to ON position, the other switches should be in OFF position.</td>
</tr>
<tr>
<td>3</td>
<td>Set enclosure temperature at 190 C</td>
<td>Ogden dial</td>
</tr>
<tr>
<td>4</td>
<td>To power the Emissions Bench:</td>
<td>Turn ON all the switches, (except the two engines which are not selected), inside the breaker panel.</td>
</tr>
<tr>
<td>5</td>
<td>Slide all 6 switches towards the desired position to read emissions on the data acquisition systems inside cabinet #2 (black box)</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Start the heated pump on Raffisch RS-55</td>
<td>Press PUMP button ON.</td>
</tr>
<tr>
<td>7</td>
<td>Purge fuel line</td>
<td>Wait 20-40 mins to reach 190 C</td>
</tr>
<tr>
<td>8</td>
<td>Start up heated HC analyzer</td>
<td>Replace fuel hose inside cabinet #1 on the back of Raffisch.</td>
</tr>
</tbody>
</table>
| 9    | Calibrate CO₂ analyzer (Beckman 880) | Turn the CO₂ switch to zero (top of cab #2) and select "3" on Chesseil Display.  
Range: 0 - 5V for 0 - 20%  
Adjust the flow rate to 1.5 l/min using "Flow to analyzers" knob.  
Press "Zer", "Enter". Wait for readings on Chesseil. Adjust with arrows to read 0% on the left of the display, then press "Enter" again.  
Turn switch to SPAN (top of cab #2).  
Press "Span", "Enter". Wait for readings on Chesseil. Adjust with arrows to read between 9-18.5% on the left of the display, then press "Enter".  
NOTE: Before starting to calibrate, make sure system temperature on oven is approx 190 C.  
0% - 0.016V  
4.6V - 18.5% |
| 10   | Calibrate CO analyzer (Siemens 21P) | Turn the CO switch to ZERO and select "4" on Chesseil display.  
Range: 0 - 5V for 0 - 10,000 ppm  
Adjust the flow rate to 2 - 3 l/min using "Flow to analyzers" knob.  
Press "3H" button to zero the gas and again when zeroing is complete.  
0.001 V - 0 ppm  
Turn CO switch to SPAN.  
Adjust the CO pot. to read (Chesseil display) 1.04V (207/ppm)  
1.04 V - 207/ppm |
| 11   | Calibrate O₂ analyzer (Oxymat 5E) | Turn the O₂ switch to ZERO and select "5" on Chesseil display.  
Range: 0 - 5V for 0 - 21%  
Adjust the flow rate to 0.7 l/min using "Flow to analyzers" knob.  
Enter "111" to make Code 1 light go out.  
(Note: Oxymat 5E should be left on overnight before testing so that accurate calibration can be completed)  
Set analyzer to "Calibration" mode by pressing "Meas/Cal" button.  
Press "5", then press "Enter" to zero the analyzer.  
Turn the O2 switch to SPAN. Reset Flow to 0.7 l/min.  
Press "8", then press "Enter" to span, wait until "not ready" light is off.  
CH₄ Range: 1 - 5V for 0 - 5000 ppm  
Adjust the flow rate to 2 - 3 l/min using "Flow to analyzers" knob.  
Press "3H" button to zero the gas and again when zeroing is complete.  
Select "1" on the Chesseil and adjust the pot. to read 4.043 on Chesseil display if required.  
4.043 V - 3950 ppm  
Turn CH₄ switch to OFF and turn CH₄ switch to SPAN.  
Select "2" on the Chesseil and adjust the pot to read 3.584 on Chesseil display if required.  
3.584 V - 1938 ppm |
| 12   | Calibrate NOx and CH₄ analyzer (Ultramat 22P) | Turn NO and CH₄ switches to ZERO (linked).  
CH₄ Range: 1 - 5V for 0 - 5000 ppm  
Adjust the flow rate to 2 - 3 l/min using "Flow to analyzers" knob.  
Press "3H" button to zero the gas and again when zeroing is complete.  
Select "1" on the Chesseil and adjust the pot. to read 4.043 on Chesseil display if required.  
4.043 V - 3950 ppm  
Turn CH₄ switch to OFF and turn NO switch to SPAN.  
Select "2" on the Chesseil and adjust the pot to read 3.584 on Chesseil display if required.  
3.584 V - 1938 ppm |
Appendix B7

Accuracy of the emission bench measurements

The analyzer specifications provided by the manufacturer are tabulated as follows:

Table B7a. Accuracy of the emission bench analyzers

<table>
<thead>
<tr>
<th>Analyzer</th>
<th>Minimum value</th>
<th>Maximum value</th>
<th>Analog Voltage</th>
<th>Maximum relative error (%)</th>
<th>Maximum reading net uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>NOx, Ultramat 22P</td>
<td>0</td>
<td>3,000 ppm</td>
<td>1-5 V</td>
<td>± 5</td>
<td>± 10 ppm</td>
</tr>
<tr>
<td>CH4, Ultramat 22P</td>
<td>0</td>
<td>5,000 ppm</td>
<td>1-5 V</td>
<td>± 20</td>
<td>± 10 ppm</td>
</tr>
<tr>
<td>CO2 , Backman 800</td>
<td>0</td>
<td>20%</td>
<td>0-5 V</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>CO , Siemens 21P</td>
<td>0</td>
<td>10,000 ppm</td>
<td>0-5 V</td>
<td>± 20</td>
<td>± 15 ppm</td>
</tr>
<tr>
<td>O2 , Oxymat 5E</td>
<td>0</td>
<td>21%</td>
<td>0-5 V</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>THC, Ratffish RS-55</td>
<td>0</td>
<td>10,000 ppm</td>
<td>0-10 V</td>
<td>± 20</td>
<td>± 10 ppm</td>
</tr>
</tbody>
</table>

It should be pointed out that during the study of emission measurements most of the values that were produced by the 1-71 engine were too small in comparison with the analyzer scale. Table B7b shows the actual measured emissions obtained throughout this research. The data for the emissions is given for all the fueling methods combined, from the lowest to the highest value.
Table B7b. Measured emissions data for 1-7l DD engine.

<table>
<thead>
<tr>
<th>Emission measured</th>
<th>range</th>
<th>Maximum value</th>
<th>The percentage utilization of the apparatus scale</th>
</tr>
</thead>
<tbody>
<tr>
<td>NOx</td>
<td>750-2500 ppm</td>
<td>3,000 ppm</td>
<td>25% - 85%</td>
</tr>
<tr>
<td>THC</td>
<td>100-1500</td>
<td>10,000 ppm</td>
<td>1% - 15%</td>
</tr>
<tr>
<td>CO</td>
<td>100-1500ppm</td>
<td>10,000 ppm</td>
<td>1% - 15%</td>
</tr>
</tbody>
</table>

By making a comparison, between these values and the analyzer ranges, one can see that for CO and THC readings are at the lowest limit of the scale range.

Pressure Settings for the Gas Analyzers

In the followings it is shown the pressure range for emission analyzers.

RATFISCH Total Hydrocarbon Analyzer (Flame ionization detector) Model 55

Analyzer Conditions:

Temperature: 180 Celsius

**Pressure Settings:** Sample/span-gas/zero-gas: 200 mbar

Fuel: (Hydrogen): 0-0.5 bar

Combustion air: 0-0.8 bar

Siemens Oxymat 5E Gas Analyzer

Sample gas flow: 0.3 to 1 l/min

Permissible sample gas pressure: up to 5 mbar

Permissible purging gas overpressure: up to 250 mbar.
Siemens Ultramat 21P or Gas Analyzer

Flow adjustment rate: Approx. 2 to 3 l/min by built in diaphragm pump

Fault indication at flow rates less than 1 l/min.

Beckman non-dispersive infrared (model 880)

Sample pressure: Maximum 10 psig (69kPa), standard

Sample flow rate: Approx. 1 to 2 SCFH (500 to 1000 cc/min)

Electric sample gas cooler (CGEK 5)

Operating pressure (absolute pressure):

0.5 – 3.5 bar at +100° C (sample gas inlet temperature).

0.5 – 2.5 bar at +140° C (sample gas inlet temperature).

Table B7c. Maximum sample gas flow rate in each cooling system.

| Sample gas flow rate [sample gas inlet temperature +100° C, operating pressure (absolute pressure) 1 bar] |
|--------------------------------------------------|--------------------------------------------------|
| 1 sample gas circuit | 2 sample gas circuit |
| +5°...<+45° C | 250 l/h | 250 l/h |
Appendix B8:

Repeatability tests

The consistency of the emission data measurement is shown in the following. Tests were conducted on three successive days for CNG fueling and hydrogen-CNG mixture fueling. Factors that could influenced the measurements were: fluctuations in CNG lower heating value concentration, emission bench analyzers calibration, ambient air temperature, pressure and humidity. Throughout the test the gas injection pressure (GIP), engine load, mass flows for CNG and hydrogen and engine rpm were kept consistently constant, so that wouldn't be a error factor.

For CNG fueling repeatability tests were conducted on 16th, 17th and 18th of September 2000, for 1,3 and 5 bars BMEP, 19 c.a. degrees BTDC for BOI, 130 bars GIP and 1200 rpm. The results for brake specific NOx, CO and THC in [g/kWh] are shown in Fig. B8a. From that one can see the reproducibility of the results with a minimum deviations. It was estimated that the maximum variation was 2-3%, and that was caused by all the reasons mentioned above along with instrument errors or wear of the prototype HPDI injector.

For hydrogen-CNG mixture fueling the tests were done on 7th, 8th and 9th of October 2000, for 3 and 5 bars BMEP, 13 c.a. degrees BTDC for BOI, 130 bars GIP and 1200 rpm. The results for the specific NOx, CO and THC in [g/kWh] are shown in Fig. B8b.

Based on these tests was concluded that the measurements are consistent within a certain accuracy. To illustrate the CNG daily variation of LHV, see Appendix D6.
Figure B8a. Brake specific NOx, CO and THC [g/kWh] vs. load for 130 bars GIP, 19c.a. degrees BTDC, at 1200 rpm with CNG fueling.
Figure B8b. Brake specific NOx, CO and THC [g/kWh] vs. load for 130 bars GIP, 13c.a. degrees BTDC, at 1200 rpm with 0.5kg/h hydrogen and CNG fueling.
Appendix B9:

Description of acquiring and processing engine data

All the measured parameters that are useful for the combustion analysis are collected by using two types of data acquisition: the slow and the fast data acquisition.

As it was mention in chapter 3.2., the “slow speed” data acquisition collects all the information required for monitoring the engine and for further calculations with only one exception: the cylinder pressure. The cylinder pressure is acquired by the “fast speed” data-acquisition (ISAAC). In this appendix is shown what type of output files each data acquisition produces.

The “slow speed data acquisition” creates 3 types of output files: *.raw, *.prn and *.emd.

Upon opening *.raw files generates: "Acquired Data - Average Of 100 Samples" and contains:

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Torque</td>
<td>Engine</td>
<td>Ambient</td>
<td>Airflow</td>
<td>Liq.fue</td>
<td>CNG</td>
<td>CNG</td>
<td>Intake</td>
</tr>
<tr>
<td>(N-m)</td>
<td>(RPM)</td>
<td>(°C)</td>
<td>(l/sec)</td>
<td>(kg/hr)</td>
<td>(kg/hr)</td>
<td>(bar)</td>
<td>(kPaG)</td>
</tr>
</tbody>
</table>

And

<table>
<thead>
<tr>
<th>Intake</th>
<th>Inject.</th>
<th>CO</th>
<th>CO2</th>
<th>NOx</th>
<th>O2</th>
<th>CH4</th>
<th>THC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temp.</td>
<td>BOI</td>
<td>[dry]</td>
<td>[dry]</td>
<td>[dry]</td>
<td>[dry]</td>
<td>[dry]</td>
<td>[wet]</td>
</tr>
<tr>
<td>(°C)</td>
<td>(°BTDC)</td>
<td>(ppm)</td>
<td>(%vol)</td>
<td>(ppm)</td>
<td>(%vol)</td>
<td>(ppm)</td>
<td>(ppmc)</td>
</tr>
</tbody>
</table>

The *.prn files generates: "Calculated Performance Data - Average Of 100 Samples" and contains:

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Brake</td>
<td>BMEP</td>
<td>Thermal</td>
<td>Air</td>
<td>Correct.</td>
<td>Correct.</td>
<td>Correct.</td>
<td>Diesel</td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>Power</td>
<td>MEP</td>
<td>Effic.</td>
<td>Massflow</td>
<td>Power</td>
<td>Torque</td>
<td>MEP</td>
<td>Ratio</td>
</tr>
<tr>
<td>(kW)</td>
<td>(bar)</td>
<td>(%)</td>
<td>(kg/hr)</td>
<td>(kW)</td>
<td>(N-m)</td>
<td>(bar)</td>
<td>(nrg%)</td>
</tr>
</tbody>
</table>
And

<table>
<thead>
<tr>
<th>Rel.</th>
<th>Excess</th>
<th>Scavenge</th>
<th>Chargin</th>
<th>Traping</th>
<th>Trapped</th>
<th>Trapped</th>
</tr>
</thead>
<tbody>
<tr>
<td>A/F</td>
<td>Air</td>
<td>Ratio</td>
<td>Effic.</td>
<td>Effic.</td>
<td>Airflow</td>
<td>RAFR</td>
</tr>
<tr>
<td>(dual)</td>
<td>(%)</td>
<td>(%)</td>
<td>(%)</td>
<td>(%)</td>
<td>(kg/hr)</td>
<td></td>
</tr>
</tbody>
</table>

The *.emd files generates: "Calculated Emissions Data - Average Of 100 Samples" and contains

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>##</td>
<td>CO</td>
<td>CO2</td>
<td>NOx</td>
<td>O2</td>
<td>CH4</td>
<td>THC</td>
</tr>
<tr>
<td>(Exh.Ana)</td>
<td>(g/kWhr)</td>
<td>(g/kWhr)</td>
<td>(g/kWhr)</td>
<td>(g/kWhr)</td>
<td>(g/kWhr)</td>
<td>(g/kWhr)</td>
</tr>
</tbody>
</table>

And

<table>
<thead>
<tr>
<th>CO</th>
<th>CO2</th>
<th>NOx</th>
<th>O2</th>
<th>CH4</th>
<th>THC</th>
<th>NMHC</th>
<th>H2O</th>
<th>H2</th>
</tr>
</thead>
<tbody>
<tr>
<td>[wet]</td>
<td>[wet]</td>
<td>[wet]</td>
<td>[wet]</td>
<td>[wet]</td>
<td>[wet]</td>
<td>[wet]</td>
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<tr>
<td>(ppm)</td>
<td>(%vol)</td>
<td>(ppm)</td>
<td>(%vol)</td>
<td>(ppm)</td>
<td>(ppmc)</td>
<td>(ppmc)</td>
<td>(%vol)</td>
<td>(ppm)</td>
</tr>
</tbody>
</table>

And

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>(g/mol)</td>
<td>(mass%)</td>
<td>(mass%)</td>
<td>(g/wet)</td>
<td>(mass%)</td>
<td>(mass%)</td>
</tr>
</tbody>
</table>

The "fast speed data acquisition" (ISAAC) acquires cylinder pressure data and produces binary files which are named in the following manner: *a.001, *a.002, ..., *a.020, etc.

The ISAAC samples pressure data for every crank angle for each engine cycle. Each engine cycle of pressure data is saved in a separate file, as shown above. Next set of pressure (data from different 20 consecutive engine cycles) will be automatically named *b.001, *b.002, ..., *b.020, and so on. It is necessary to know the engine operating conditions (BOI, PW, engine speed, engine load, injection pressure) in which the data was taken. To convert these binary files in ASCII text files the pdc.exe program is used.
It produces an output *.dat file data, which contains all the 20 instantaneous pressure measurements, for all the conditions (or regimes) that the engine was ran at. This file contains the un-averaged cylinder pressure data. The units for cylinder pressure in *.dat files are kPa, and that represents the absolute cylinder pressure.

To obtain the average of these 20 pressure values (which were taken in 20 consecutive cycles) for a given conditions, xpdata.exe program is used. This program uses *.dat file as an input file, and produces *.xpj as an output file. The *.xpj file contains only one set of averaged cylinder pressure, which is going to be used to plot the pressure trace for a given conditions (BOI, PW, engine speed, engine load, injection pressure). The*.xpj is a final file which contains a 2 columns table : one with c.a. (crank angle degree) and another one with cylinder pressure. With these cylinder pressure values the pressure traces were plotted against c.a. degrees.

The explanations for pdc.exe are as follows:

"v2.32"   ; PROG.VERSION$
"2"   ; # of data channels per sample
"360"   ; # of data samples per cycle
"360"   ; # of C.A. degrees per cycle
"d:\thesis\"   ; Path for binary files
"d:\thesis\"   ; Path for ASCII file(s)
"feb28c"   ; Base name for binary files
"feb28c"   ; Base name for ASCII file(s)
"001"   ; First filename extension
"020"   ; Last filename extension
"1"   ; Extension increment
"Y"   ; Autoincrement extensions
"Y"   ; Prepare for mass burn analysis
"Bottom"   ; Channel 0 title, row 1
"Dead"   ; Channel 0 title, row 2
"Center"   ; Channel 0 units
"Cylinder"   ; Channel 1 title, row 1
"Pressure"   ; Channel 1 title, row 2
"(psi)"   ; Channel 1 units
"Fuel"   ; Channel 2 title, row 1
<table>
<thead>
<tr>
<th>Title/Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Injection</td>
<td>Channel 2 title, row 2</td>
</tr>
<tr>
<td>Command</td>
<td>Channel 2 units</td>
</tr>
<tr>
<td>Hydraulic</td>
<td>Channel 3 title, row 1</td>
</tr>
<tr>
<td>Pressure</td>
<td>Channel 3 title, row 2</td>
</tr>
<tr>
<td>(psi)</td>
<td>Channel 3 units</td>
</tr>
<tr>
<td>Y</td>
<td>Keep titles in output files</td>
</tr>
<tr>
<td>Engine speed (rpm)</td>
<td>1200</td>
</tr>
<tr>
<td>Exhaust temperature (K)</td>
<td>710</td>
</tr>
<tr>
<td>Air box temperature (K)</td>
<td>350</td>
</tr>
<tr>
<td>Air box pressure (kPa)</td>
<td>122.2</td>
</tr>
<tr>
<td>Intake air mass flow (kg/hr)</td>
<td>112</td>
</tr>
<tr>
<td>Diesel mass flow (kg/hr)</td>
<td>3.0</td>
</tr>
<tr>
<td>CNG mass flow (kg/hr)</td>
<td>0</td>
</tr>
<tr>
<td>Charge amp gain (psi/V)</td>
<td>200</td>
</tr>
<tr>
<td>A/D gain (counts/V)</td>
<td>204.8</td>
</tr>
<tr>
<td>Ch. # of pressure signal (1..4)</td>
<td>360</td>
</tr>
<tr>
<td># of pressure records per cycle</td>
<td>360</td>
</tr>
<tr>
<td># of records to analyze per cycle</td>
<td>360</td>
</tr>
<tr>
<td>φC.A. increment per sample</td>
<td>1</td>
</tr>
<tr>
<td>φC.A. of 1st record (0°=BDC)</td>
<td>0</td>
</tr>
<tr>
<td>φC.A. of BOI (180°=TDC)</td>
<td>159</td>
</tr>
<tr>
<td>φC.A. of 1st sample to keep</td>
<td>0</td>
</tr>
<tr>
<td>φC.A. of last sample to keep</td>
<td>359</td>
</tr>
<tr>
<td>Pulse width duration in φC.A.</td>
<td>19.25</td>
</tr>
</tbody>
</table>
Appendix B10:

Configuration files for engine data acquisition.

This is the data acquisition configuration file for diesel fueling (called DDC1.cfg)
Last modified 10-08-1998
Engine: DDC CI 2-Stroke NA 1-Cylinder
Emissions Data: y
Displacement (cc): 1162.354
Fuel Lower Heating Values
Mode Main Pilot

<table>
<thead>
<tr>
<th>#</th>
<th>Variable Name</th>
<th>Channel</th>
<th>Constant Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Torque</td>
<td>4</td>
<td>none</td>
</tr>
<tr>
<td>2</td>
<td>Engine Speed</td>
<td>5</td>
<td>none</td>
</tr>
<tr>
<td>3</td>
<td>Ambient Air Temp</td>
<td>2</td>
<td>none</td>
</tr>
<tr>
<td>4</td>
<td>Barom. Pressure</td>
<td>none</td>
<td>none</td>
</tr>
<tr>
<td>5</td>
<td>Airflow</td>
<td>8</td>
<td>none</td>
</tr>
<tr>
<td>6</td>
<td>Liq.fuel dm/dt</td>
<td>7</td>
<td>none</td>
</tr>
<tr>
<td>7</td>
<td>CNG dm/dt</td>
<td>0</td>
<td>none</td>
</tr>
<tr>
<td>8</td>
<td>Exhaust Pressure</td>
<td>none</td>
<td>none</td>
</tr>
<tr>
<td>9</td>
<td>CNG Pressure</td>
<td>9</td>
<td>none</td>
</tr>
<tr>
<td>10</td>
<td>Intake Pressure</td>
<td>6</td>
<td>none</td>
</tr>
<tr>
<td>11</td>
<td>Intake Temp.</td>
<td>3</td>
<td>none</td>
</tr>
<tr>
<td>12</td>
<td>Exhaust Temp.</td>
<td>none</td>
<td>none</td>
</tr>
<tr>
<td>13</td>
<td>Inject. BOI</td>
<td>1</td>
<td>none</td>
</tr>
<tr>
<td>14</td>
<td>CO [dry]</td>
<td>13</td>
<td>none</td>
</tr>
<tr>
<td>15</td>
<td>CO2 [dry]</td>
<td>12</td>
<td>none</td>
</tr>
<tr>
<td>16</td>
<td>NOx [dry]</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>O2 [dry]</td>
<td>11</td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>CH4 [dry]</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>19</td>
<td>THC [wet]</td>
<td>14</td>
<td></td>
</tr>
</tbody>
</table>

Number of display columns: 16

<table>
<thead>
<tr>
<th>#</th>
<th>Variable</th>
<th>Title 1</th>
<th>Title 2</th>
<th>Units</th>
</tr>
</thead>
</table>
| 1  | Engine   | Speed   | (RPM)   | #######
| 2  | BMEP     |         | (bar)   | ######.##
| 3  | Thermal  | Effic.  | (%)     | #######.##
| 4  | Liq.fuel | dm/dt   | (kg/hr) | #######.##
| 5  | Air Box  | Temp.   | øC      | #######.#
| 6  | BOI      | øBTDC   |         | #######.##
| 7  | Air      | dm/dt   | (kg/hr) | #######.##
| 8  | TORQUE   | N-m     |         | #######.##
| 9  | CO       | (wet)   | (ppm)   | #######.##
| 10 | CO2      | (dry)   | (%vol)  | #######.##
| 11 | 45 NOx  | (wet)   | (ppm)  | ############# |
| 12 | 47 CH4  | (wet)   | (ppm)  | ############# |
| 13 | 48 THC  | (wet)   | (ppmc) | ############# |
| 14 | 16 NOx  | (dry)   | (ppm)  | ############# |
| 15 | 50 H2O  | (wet)   | (%vol) | ############# |
| 16 | 18 CH4  | (dry)   | (ppmC) | ############# |

ISAAC Configuration:

Number of Channels: 2
Channel Label Units Slope Offset
0 BDC 1.000 0.000
1 Cylinder Pressure (PSI) 1.000 0.000
Crank angle pulses/rev = 360
Data points required/rev = 360
External clock divide = 1
Number of consecutive cycles = 20
The following is a data acquisition configuration file for CNG (called DDC2.cfg)
Last modified 03-10-1998
Engine: DDC CI 2-Stroke NA 1-Cylinder
Emissions Data: y
Displacement (cc): 217869.4
Fuel Lower Heating Values
Mode Main Pilot
3 43200 0

<table>
<thead>
<tr>
<th>#</th>
<th>Variable</th>
<th>Name</th>
<th>Channel</th>
<th>Constant Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Torque</td>
<td>(N-m)</td>
<td>4</td>
<td>none</td>
</tr>
<tr>
<td>2</td>
<td>Engine Speed</td>
<td>(RPM)</td>
<td>5</td>
<td>none</td>
</tr>
<tr>
<td>3</td>
<td>Ambient Air Temp</td>
<td>(°C)</td>
<td>2</td>
<td>none</td>
</tr>
<tr>
<td>4</td>
<td>Barom. Pressure</td>
<td>(kPa)</td>
<td>none</td>
<td>none</td>
</tr>
<tr>
<td>5</td>
<td>Airflow</td>
<td>(l/sec)</td>
<td>8</td>
<td>none</td>
</tr>
<tr>
<td>6</td>
<td>Liq.fuel dm/dt</td>
<td>(kg/hr)</td>
<td>7</td>
<td>none</td>
</tr>
<tr>
<td>7</td>
<td>CNG dm/dt</td>
<td>(kg/hr)</td>
<td>0</td>
<td>none</td>
</tr>
<tr>
<td>8</td>
<td>Exhaust Pressure</td>
<td>(kPaG)</td>
<td>none</td>
<td>none</td>
</tr>
<tr>
<td>9</td>
<td>CNG Pressure</td>
<td>(bar)</td>
<td>9</td>
<td>none</td>
</tr>
<tr>
<td>10</td>
<td>Intake Pressure</td>
<td>(kPaG)</td>
<td>6</td>
<td>none</td>
</tr>
<tr>
<td>11</td>
<td>Intake Temp.</td>
<td>(°C)</td>
<td>3</td>
<td>none</td>
</tr>
<tr>
<td>12</td>
<td>Exhaust Temp.</td>
<td>(°C)</td>
<td>none</td>
<td>none</td>
</tr>
<tr>
<td>13</td>
<td>Inject. BOI</td>
<td>(°BTDC)</td>
<td>1</td>
<td>none</td>
</tr>
<tr>
<td>14</td>
<td>CO (dry)</td>
<td>(ppm)</td>
<td>13</td>
<td>none</td>
</tr>
<tr>
<td>15</td>
<td>CO2 (dry)</td>
<td>(%vol)</td>
<td>12</td>
<td>none</td>
</tr>
<tr>
<td>16</td>
<td>NOx (dry)</td>
<td>(ppm)</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>17</td>
<td>O2 (dry)</td>
<td>(%vol)</td>
<td>11</td>
<td></td>
</tr>
<tr>
<td>18</td>
<td>CH4 (dry)</td>
<td>(ppm)</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>19</td>
<td>THC (wet)</td>
<td>(ppmc)</td>
<td>14</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td></td>
<td></td>
<td>14</td>
<td></td>
</tr>
</tbody>
</table>

Number of display columns: 16
<table>
<thead>
<tr>
<th>#</th>
<th>Variable</th>
<th>Title 1</th>
<th>Title 2</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2 Engine</td>
<td>Speed (RPM)</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>21 BMEP</td>
<td>21</td>
<td>21</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>27 Diesel</td>
<td>Ratio (nrg%)</td>
<td>27</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>6 Liq.fuel</td>
<td>dm/dt (kg/hr)</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>7 CNG</td>
<td>dm/dt (kg/hr)</td>
<td>7</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>9 CNG</td>
<td>Pressure (bar)</td>
<td>9</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>22 Thermal</td>
<td>Effic. (%)</td>
<td>22</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>48 Unburn.</td>
<td>CNG (mass%)</td>
<td>48</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>43 CO</td>
<td>(wet) (ppm)</td>
<td>43</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>44 CO2</td>
<td>(wet) (%vol)</td>
<td>44</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>45 NOx</td>
<td>(wet) (ppm)</td>
<td>45</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>47 CH4</td>
<td>(wet) (ppm)</td>
<td>47</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>19 THC</td>
<td>(wet) (ppmc)</td>
<td>19</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>47 NMHC</td>
<td>(wet) (ppmc)</td>
<td>47</td>
<td></td>
</tr>
<tr>
<td>15</td>
<td>46 O2</td>
<td>(wet) [%vol]</td>
<td>46</td>
<td></td>
</tr>
<tr>
<td>16</td>
<td>16 NOx</td>
<td>(dry) (ppm)</td>
<td>16</td>
<td></td>
</tr>
</tbody>
</table>

**ISAAC Configuration:**

- **Number of Channels:** 4
- **Channel Label Units Slope Offset**
  | 0  | BDC | 1.000 | 0.000 |
  | 1  | Cylinder Pressure (PSI) | 1.000 | 0.000 |
  | 2  | Hydraulic Pressure (PSI) | 1.000 | 0.000 |
  | 3  | Command Signal | 1.000 | 0.000 |
- **Crank angle pulses/rev = 360**
- **Data points required/rev = 360**
- **External clock divide = 1**
- **Number of consecutive cycles = 20**
Appendix B11.

Cylinder pressure output file sample

In the following it is shown an example of an output *.xpj cylinder pressure format header. This was generated by pdc.exe (which converts bin data in ASCII data) and xpdata (i.e. averages the cylinder pressure data over 20 engine).

The explanations of all the numbers is given below.

1250,120.3,352.75,608.2,2258.
.001479,8.145E-06, 2.29E-05 ^; 0.173,1,14.6
360,90,20
1,.012445, .002249

Revolution per minute = 1250 rpm
P air-box (IVC) = 120.3 [ bar ]
T air-box (IVC) = 352.75 [ K ]
T exhaust = 608.2 [ K ]
NOx measured = 2258 [ ppm ]

Air flow rate = 0.001479 [ \frac{g}{cycle} ]

Diesel flow rate = 0.000008145 [ \frac{g}{cycle} ]

Natural Gas flow rate = 0.0000229 [ \frac{g}{cycle} ]

BOI = 173° from BDC (or -7° to TDC)

1 = crank angle increment

PW = 14.6° (pulse width or injection duration)

360 = total number of crank angles per engine cycle

90 = every 90 c.a. degrees the cylinder pressure placed in a different column

Unburned Diesel mass fraction = 0.012445 [ \frac{g}{cycle} ]
This header is followed by a 2-column table, in which a cylinder pressure is been tabulated by the software (xpdata.exe). A sample raw data for the cylinder pressure is shown below.

<table>
<thead>
<tr>
<th>Time (s)</th>
<th>Cylinder Pressure (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.120 34</td>
<td>37,115.62</td>
</tr>
<tr>
<td>1,118.65</td>
<td>38,117.31</td>
</tr>
<tr>
<td>2,118.99</td>
<td>39,117.98</td>
</tr>
<tr>
<td>3,118.32</td>
<td>40,116.97</td>
</tr>
<tr>
<td>4,117.98</td>
<td>41,117.31</td>
</tr>
<tr>
<td>5,117.64</td>
<td>42,118.32</td>
</tr>
<tr>
<td>6,118.65</td>
<td>43,116.30</td>
</tr>
<tr>
<td>7,117.64</td>
<td>44,117.98</td>
</tr>
<tr>
<td>8,118.99</td>
<td>45,117.98</td>
</tr>
<tr>
<td>9,117.98</td>
<td>46,117.98</td>
</tr>
<tr>
<td>10,117.64</td>
<td>47,117.98</td>
</tr>
<tr>
<td>11,117.98</td>
<td>48,120.34</td>
</tr>
<tr>
<td>12,116.97</td>
<td>49,117.64</td>
</tr>
<tr>
<td>13,116.30</td>
<td>50,120.67</td>
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<td>52,121.01</td>
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<td>16,117.31</td>
<td>53,121.01</td>
</tr>
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<td>17,116.63</td>
<td>54,121.68</td>
</tr>
<tr>
<td>18,115.96</td>
<td>55,123.36</td>
</tr>
<tr>
<td>19,115.96</td>
<td>56,123.70</td>
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<tr>
<td>21,116.97</td>
<td>58,125.05</td>
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<td>59,125.38</td>
</tr>
<tr>
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<td>60,126.39</td>
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<td>24,116.97</td>
<td>61,128.08</td>
</tr>
<tr>
<td>25,116.30</td>
<td>62,129.09</td>
</tr>
<tr>
<td>26,116.30</td>
<td>63,128.42</td>
</tr>
<tr>
<td>27,116.97</td>
<td>64,131.11</td>
</tr>
<tr>
<td>28,115.29</td>
<td>65,131.78</td>
</tr>
<tr>
<td>29,116.63</td>
<td>66,131.44</td>
</tr>
<tr>
<td>30,116.63</td>
<td>67,134.14</td>
</tr>
<tr>
<td>31,115.29</td>
<td>68,135.15</td>
</tr>
<tr>
<td>32,115.96</td>
<td>69,137.17</td>
</tr>
<tr>
<td>33,117.31</td>
<td>70,138.52</td>
</tr>
<tr>
<td>34,116.63</td>
<td>71,138.85</td>
</tr>
<tr>
<td>35,115.96</td>
<td>72,140.20</td>
</tr>
<tr>
<td>36,117.31</td>
<td>73,141.54</td>
</tr>
<tr>
<td>185,3298.74</td>
<td>220,1955.47</td>
</tr>
<tr>
<td>186,3246.20</td>
<td>221,1884.10</td>
</tr>
<tr>
<td>187,3194.04</td>
<td>222,1812.39</td>
</tr>
<tr>
<td>188,3146.24</td>
<td>223,1748.09</td>
</tr>
<tr>
<td>189,3128.39</td>
<td>224,1684.12</td>
</tr>
<tr>
<td>190,3212.22</td>
<td>225,1622.18</td>
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<tr>
<td>191,3337.46</td>
<td>226,1565.62</td>
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<tr>
<td>192,3409.84</td>
<td>227,1508.72</td>
</tr>
<tr>
<td>193,3487.27</td>
<td>228,1459.23</td>
</tr>
<tr>
<td>194,3587.26</td>
<td>229,1407.72</td>
</tr>
<tr>
<td>195,3721.59</td>
<td>230,1360.59</td>
</tr>
<tr>
<td>196,3853.56</td>
<td>231,1314.81</td>
</tr>
<tr>
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Appendix B12:

**Wet-Basis and Brake Specific Emissions.**

The emission analyzers measure the volumetric concentrations of carbon monoxide (CO), carbon dioxide (CO2), nitrogen oxides (NOx), methane (CH4), and oxygen (O2) with no water vapor percentage, thus on a dry-basis. Since water vapor formed in the engine exhaust dilutes the emissions concentration the analyzers' measurements have to be converted from dry-basis to wet-basis. Heywood [34] derived an algorithm for this conversion, described at page 152. The wet $\bar{x}$ and dry $\bar{x}^*$ mole fraction of species i are combustion products of a generic fuel $C_mH_nO_r$.

Wet and dry mole fractions are related by:

$$\bar{x}_i = (1 - \bar{x}_{H_2O}) \bar{x}_i^*$$

$$\bar{x}_{H_2O} = \frac{m}{2n} \left[ \frac{\bar{x}_{CO}^* + \bar{x}_{CO_2}^*}{1 + \frac{\bar{x}_{CO}^* + \bar{x}_{CO_2}^*}{K \cdot \bar{x}_{CO_2}^*} + \frac{m}{2n} \cdot (\bar{x}_{CO}^* + \bar{x}_{CO_2}^*) \right]$$

and

$$\bar{x}_{H_2} = \frac{\bar{x}_{H_2O} \cdot \bar{x}_{CO}^*}{K \cdot \bar{x}_{CO_2}^*}$$

Because diesel, natural gas and hydrogen are used as fuels, an average composition for $C_mH_nO_r$ based on the ratio between the fuel flows was taken.

The constant K relates the wet basis concentrations of CO2, H2O, CO and H2 as:

$$K = \frac{\bar{x}_{CO} \cdot \bar{x}_{H_2O}}{\bar{x}_{CO_2} \cdot \bar{x}_{H_2}}$$

Since $\bar{x}_{H_2}$ is not measures directly, empirical values for K are taken from literature (Heywood [34]).

The THC concentration is measured directly on wet-basis.
Specific Emissions

The measured concentration of the exhaust emissions is expressed in [ppm], parts per million, which corresponds to the mole fraction multiplied by $10^6$).

Normalized indicators of emissions levels are more useful, such as specific emissions.

As for power or efficiency, there are two types of specific emissions: indicated and brake specific emissions. They can be expressed as:

$$sNO_x = \frac{\dot{m}_{NO_x}}{P},$$

$$sCO = \frac{\dot{m}_{CO}}{P},$$

$$sHC = \frac{\dot{m}_{HC}}{P},$$

$$sPart = \frac{\dot{m}_{Part}}{P},$$

where specific emissions are the mass flow rate of pollutant per unit power output. The units are: [μg / J], [g / kW·h], or [g / hp·h].

Many of the repeatability tests data were plotted as brake specific emissions.
Appendix C1: Cylinder head component parts
Appendix C1: The modified cylinder head mounted onto the engine block.
Figure C2. 1-71 original cylinder head gasket
Figure C3. Original (unmodified) 1-71DDC cylinder head.
Figure C4. Modified (from 3-71 DDC) cylinder head.
Figure C5. Modified cylinder head assembly drawing (showing the exhaust valve plug)
Figure C6. Exhaust valve plug.
Figure C7. Cross section from the 1-71DDC cylinder head and piston.
Figure C8. Cylinder head with diesel jets and CNG jets.
Appendix C9: (see figure B2b)

Appendix D1: Calculation of the mass flows and momentum

To investigate what impact on injection, ignition and combustion the CNG pressure has, tests were conducted for 130, 150 and 170 bars GIP. The cylinder pressure and emissions were analyzed by varying the injection timing (BOI), the injection duration (PW), the engine load (BMEP) and by maintaining the engine speed constant.

Through the HPDI fuelling, both diesel and gas are injected at a constant mass flow through the injector nozzle holes. For this research a "6-7" (i.e. an injector with 6 diesel holes and 7 CNG holes) injector was used. The ratio of the holes diameters is 4 to 1, as diesel hole has 0.005 inches diameter and gas hole has 0.020 inches diameter. A drawing of the injector needles is shown in Fig. D1a. and Fig.D1b.

**Figure D1a.** The HPDI injector nozzle with two concentrical needles.
For a better understanding of the importance of the GIP for the combustion process and the NOx formation, a mass flow and momentum calculation was done for diesel and CNG.

**Calculation of the mass flow and the momentum flux for Diesel**

The mass flow for the diesel fuel injected through the HPDI injector is:

\[ m_d = \rho_d v_d A_d N_d, \]

where

\[ \rho_d = 850 \left( \frac{kg}{m^3} \right), \] is the density of the diesel fuel, (considered incompressible fluid)

\[ v_d = 100 \left( \frac{m}{s} \right), \] is the estimate diesel exit velocity through the diesel hole,

\[ d_d = 0.127 \times 10^{-3} \left[ m \right], \] is the HPDI injector diesel hole diameter,

\[ N_d = 6, \] is the number of diesel holes,

\[ A_d = \pi \frac{d_d^2}{4} \left[ m^2 \right] = 1.2667 \times 10^{-8} m^2, \] is the area of the HPDI injector diesel hole

Thus the diesel mass flow for all 6 holes can be calculated as:
\[ m_d = \rho_d v_d \ A_d \ N_d = 850 \times 100 \times 1.267 \times 10^{-8} \times 6 = 6.46 \times 10^{-3} \text{ kg/s}. \]

Knowing that the diesel injection occurs for about 5 crank angle degrees per engine cycle (which has 360 degrees), and that the engine was run at 1200 revolutions per minute, the injection duration can be calculated as 0.7 milliseconds.

The estimate diesel quantity per hour can be calculated as:

\[
(\text{Diesel pilot mass flow}) = m_d I_{\text{diesel}} 3600 = 0.16 \text{ kg/hr},
\]

where \( I_{\text{diesel}} \) is the diesel pilot injection time. This value was obtained for no engine load.

In reality, data taken from the experiments have showed that the maximum average pilot diesel injected was ranging from 0.35 kg/hr to 0.45 kg/hr (for 5 bar BMEP, which is the maximum load for the HPDI injector). Moreover, from the raw data it was noticed that the diesel consumption is function of the engine load.

The diesel momentum flux at exit is:

\[ M_d = \rho_d v_d^2 A_d N_d = m_d \ast v_d = 0.646 \text{ N}, \]

which represents a small impingement force for the diesel jet. This momentum is important for estimating the injection penetration distance of the diesel.

**Calculation of the mass flow and the momentum flux for CNG**

A similar calculation can be done for the natural gas mass flow, momentum and consumption.

Thus, the natural gas mass flow is:

\[ m_g = \rho_g v_g \ A_g N_g, \text{ where} \]

---

\(^{(1)}\) The diesel injection is electronically controlled by the ECU (electronic control unit)
\( \rho_g = 110 \left( \frac{kg}{m^3} \right) \) is the natural gas density at 170 bars GIP injection pressure\(^2\).

\( v_g = 400 \left( \frac{m}{s} \right) \), is the estimate \(^3\)(supersonic) natural gas exit velocity through the gas hole,

\(dg = 0.508 \times 10^{-3} \left[ m \right] \), is the HPDI injector natural gas hole diameter,

\( N_g = 7 \), is the number of natural gas holes for the best HPDI (it has been proved in previous researches\(^4\) that the optimum configuration from the turbulence mixing, combustion process, thermal efficiency and emissions standpoint is a "6-7" injector configuration, which means a HPDI injector with 6 holes for diesel and 7 holes for natural gas).

\[ A_d = \pi \frac{d_d^2}{4} \left[ m^2 \right] = 20.2682 \times 10^{-8} \left[ m^2 \right], \] is the area of the HPDI injector natural gas hole.

The natural mass flow for all 7 holes can be calculated as:

\[ m_g = \rho_g v_g A_d N_g = 110 \times 400 \times 20.2682 \times 10^{-8} \times 7 = 62.42 \times 10^{-3} \text{ kg/s}. \]

In contrast with the pilot injection time, the natural gas injection occurs on an average span of 40 crank angle degree (if the BOI is very early, i.e. -25 c.a. BTDC, the end of injection will occur at about +15 c.a. degree ATDC, whereas for a late BOI, the injection ends up later ATDC). For an engine speed of 1200 rpm, this number of crank angle degrees translates into 5.6 milliseconds. The estimate natural gas quantity per hour could be calculated as:

\[ (\text{CNG mass flow}) = m_g I_{\text{gas}} \times 3600 = 1.26 \text{ kg/hr}, \] where \( I_{\text{gas}} \) is the natural gas injection time.

\(^2\) According with the BC Gas, the ratio of natural gas and air is 0.52 at atmospheric pressure.

\(^3\) Patric Oullette - "High pressure injection of natural gas for diesel fueling" MASc. Thesis, 1992, UBC.

Again, the natural gas consumption was estimated for no engine load.

In fact, the maximum quantity of natural gas that was injected was 2.26 kg/hr (for 5 bar BMEP, which is the maximum load for the HPDI injector).

These calculations are meant to give an estimate figures for the fuel consumption, in [kg/hr]. It was observed a good agreement with the measured consumption.

The momentum due to the natural gas can be calculated as:

$$M_g = \rho_g v_g^2 A_g N_g = m_g * v_g = 24.97 \text{ N}$$

The ratio of diesel-natural gas momentum is 38, which illustrates a higher turbulence for the natural gas plume as it is injected into the cylinder, compared with the diesel pilot plume. It is believed that, because the natural gas has a large momentum it will travel farther into the cylinder.

An increase in GIP (gas injection pressure) would increase the gas momentum, facilitating a good mixing with the available air from the cylinder. However, an excessive GIP, would cause a very high gas momentum, and the gas might not get the chance to completely ignite from the diesel pilot. In this situation, part of the injected gas could remain unburned creating an increase in THC (total hydrocarbon) percentage. Other effects of excessive GIP are that the gas plume cannot completely mix with the available air or, that the gas impinges on the wall as a partially enflamed plume. Therefore the natural gas temperature flame, which is the main factor for NOx formation, can be easily influenced by the amount and the pressure of gas which is introduced in the cylinder.

The trade off for choosing the appropriate GIP is dictated by ensuring a better air-fuel mixing, an efficient combustion, a higher thermal efficiency and low NOx production.
Appendix D2:

Calculation of hydrogen and natural gas choked mass flows.

The percentage at which hydrogen is mixed and injected with the natural gas has to be known, such that its effects on engine performances and emissions could be analyzed.

The calculation has been done for choked flow for the gaseous fuels (CNG and hydrogen) passing through the HPDI injector gas holes. These calculated flows are to be compared against the measured mass flow for both CNG and hydrogen.

The two mass flows shouldn't be expected to be the same, because of the discharge coefficient through the metering valves and injectors holes.

A. Calculated choked mass fuel flows.

To calculate the mole fraction one can use:

\[
\chi_{H_2} = \frac{\dot{m}_{H_2}}{\dot{m}_{H_2} + \frac{\dot{m}_{CNG}}{MW_{CNG}}} \frac{MW_{CNG}}{MW_{H_2}}
\]

where \( \dot{m}_{H_2} \) and \( \dot{m}_{CNG} \) are mass flows for hydrogen and natural gas.

\[
\chi_{CNG} = \frac{\dot{m}_{CNG}}{\dot{m}_{H_2} + \frac{\dot{m}_{CNG}}{MW_{CNG}}} = 1 - \chi_{H_2}
\]

Since the natural gas is 95.5% methane, the molecular weight for natural gas will be taken as for methane, \( MW_{CNG} = 16 \) [kg/kmol].

The molecular weight for hydrogen is \( MW_{H_2} = 2 \) [kg/kmol].

Molecular weight of the CNG-hydrogen mixture is:
\[ MW_{\text{mix}} = MW_{H_2} x_{H_2} + MW_{\text{CNG}} x_{\text{CNG}} \]

Calculation of polytropic coefficient \( \gamma_{\text{mix}} \):

From \[ \frac{\gamma_{\text{mix}}}{\gamma_{\text{mix}} - 1} = \frac{\gamma_{H_2} - 1}{\gamma_{H_2}} x_{H_2} + \frac{\gamma_{\text{CNG}} - 1}{\gamma_{\text{CNG}} - 1} x_{\text{CNG}} \], \( \gamma_{\text{mix}} \) can be found as:

\[
\gamma_{\text{mix}} = \frac{\gamma_{H_2} - 1 x_{H_2} + \gamma_{\text{CNG}} - 1 x_{\text{CNG}}}{\gamma_{H_2} - 1} \]

Choked mass flow (through the injector's gas holes nozzle) for the mixed gases could be calculated for 7 gas holes of the HPDI injector (as the CNG - hydrogen mixture uses only the HPDI gas holes).

\[
m_{\text{choked}} = \frac{A_{\text{gas holes}} P_0 \sqrt{\gamma}}{R} \frac{2}{\gamma + 1} \left( \gamma + 1 \right)^{-\frac{\gamma + 11}{2(\gamma - 1)}} \left[ \frac{\text{kg}}{s} \right], \quad \gamma = \gamma_{\text{mix}}
\]

where \( A_{\text{gas holes}} = \frac{N \pi d^2}{4} \) is the area for 7 gas holes of the HPDI injector nozzle; \( \bar{R} = 8314.5 \left[ \frac{J}{\text{kmolK}} \right] \); is the universal gas constant, \( p_0 \) is the gases injection pressure (130 bars) considered to be the stagnation pressure; \( T_0 \) is the atmospheric temperature (293.15K), and \( MW_{\text{mix}} \) is the molecular weight of the mixture.

The theoretical choked mass flow is compared with the actual measured mass flow for the fuel mixture, that is:

\[
m_{\text{measured}} = \left( m_{\text{CNG+H}_2} \right) \left( \frac{1}{360} \right) \left[ \frac{1}{PW} \right] = \left( m_{\text{CNG+H}_2} \right) \left( \frac{1}{10PW} \right) \left[ \frac{\text{kg}}{s} \right]
\]
The ratio of the theoretical choked mass flow and the measured mass flow is:

\[ \frac{m_{\text{measured}}}{m_{\text{choked}}} \],

and that is expected to be 0.7-0.8, due to the discharge coefficients of the metering valve and injector nozzle gas holes. The calculation of these two mass flows is done for 130 bars GIP, 5 bars BMEP, with CNG and zero hydrogen, CNG and 0.25 kg/h hydrogen, and CNG and 0.5 kg/h hydrogen fueling at 1200 rpm. The graph for the ratio of the measured mass flow and the choked mass flow is shown in Fig. D2., from which one can see that for 0.5 kg/h hydrogen mass flow the mixture mass flow becomes nearly choked (i.e. the ratio tops with 0.5 which is closer to 0.7 than 0.3, which is the correspondent ratio for 0.25 kg/h hydrogen mass flow).

B. Friction choking in a constant area tube (at the tube's end)

Constant adiabatic flow with friction for constant-area can be represented by:

\[ \frac{4c_f L}{D} = \frac{1 - M^2}{\gamma M^2} + \frac{\gamma + 1}{2\gamma} \ln \left( \frac{(\gamma + 1)M^2}{2 \left( 1 + \frac{\gamma - 1}{2} M^2 \right)} \right) \]  

(1)

where \( c_f \) is friction coefficient (considered constant)

\( M \) is the Mach number

\( L \) is the length of the duct necessary to change the Mach number from \( M=0 \) to \( M=1 \).

\( D \) is the diameter of the tube

\( \gamma \) is the polytropic coefficient.

From eq. (1), chose \( c_f = 0.005 \), and solve for \( M \). Find \( M=0.85 \)

Assume that \( M_i=M_e=0.85 \)
Knowing Mach number at the exit of the tube (i.e. injector nozzle of length L), one can calculate the stagnation pressure at exit. For that we need this formula:

\[ p_{\text{oin}} = \frac{1}{M} \left[ \frac{2}{\gamma + 1} \left( 1 + \frac{\gamma - 1}{2} M^2 \right) \right]^{\frac{\gamma + 1}{2(\gamma - 1)}} \]  \hspace{1cm} (2)

Guess stagnation pressure \( p_{\text{oi}} \) (close to GIP, gas injection pressure), and solve for \( p_{\text{oe}} \).

Once \( p_{\text{oe}} \) is calculated, the mass flow required can be determined, using:

\[ m = \rho A v = \frac{p_{\text{oe}}}{RT} AM \sqrt{\gamma RT} = \frac{p_{\text{oe}} \sqrt{\gamma}}{\sqrt{RT}} A \]  \hspace{1cm} (3)

From eq. (3) once can see that \( m \propto p_{\text{oe}} \).

Tabulated calculations are presented in the following table D2d.
Ratio of the measured total fuel flow rate versus the calculated total fuel choked mass flow rate for 130 bars GIP, 5 bars BMEP, with CNG, CNG and 0.25 kg/h H2, CNG and 0.5 kg/h H2 fueling, for different BOIs @ 1200 rpm.

**Figure D2.** Ratios of choked low and measured flow for CNG and hydrogen-CNG mixture.
Table D2d. HPDI-6F3 Nitrogen flow data (BC ResearCh Reprt Jan 27, 1999 Project No 6-05-638

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<td>p=(g+1)/(g-1)</td>
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Loss Calculation Assumptions:
Assume Po for Nozzle is static pressure ps at valve exit
Assume Po for nozzle is static pressure in sac
Assume M=1 at nozzle exit
Assume constant Cf in nozzle
Figure D2d. HPDI - 6F3A Injector with 7 nozzle holes and 0.020 in. diameter Flow Tests with Nitrogen
Appendix D3:

Calibration of "Micromotion" flow-meter with compressed air.

To ensure that the "Micromotion" flow-meter used for hydrogen measurements indicates the mass flow correctly, a calibration was done, using compressed air (from a bottle). The air passed through a bottle regulator first, then through the flow-meter. That generates an output current [mA] when a mass flow is detected. According with the manufacturer specification the "Micromotion" flow-meter produces 4 mA for zero flow and 20 mA for maximum flow. Since for hydrogen the flow-meter was configured for 4 kg/h, a linear variation graph was plotted (see Fig. D3a).

![Graph of H2 mass flow (kg/h) vs. current (mA) for Micromotion flowmeter](image)

"mass flow" = 0.25"mA" - 1

Figure D3a. Hydrogen mass flow vs. current.
Table D3. Air calibration of the "Micromotion" flow-meter.

Date: 21/11/2000
Compressed air was used
The flow was adjusted to 6 mA. The overall current variation is from 5.97 to 6.05 mA
The discharge pressure was 500 psi (second stage regulator gauge)
The bottle contains 6380 liters of expanded air (according with the manufacturer label)
The readings were taken at every 5 minutes for 4 hours.
The mass flow was kept constant all the time.
The initial pressure in the bottle was 12000 kPa, and at the end 8000 kPa
Because the discharge rate was so slow (for very low pressure) the bottle temperature was unmodified.

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<th>Time</th>
<th>Pressure left in the bottle[kPa]</th>
<th>z</th>
<th>t</th>
<th>m</th>
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242
As one can see from the graph, a current of 5 mA corresponds to 0.25 kg/h hydrogen mass flow, and 6 mA corresponds to 0.5 kg/h hydrogen mass.

To verify these findings the calibration was done for air. The air was drained from the bottle as the time was recorded. Figure D3b shows the results:

![Calibration of Micromotion flow sensor with air](image)

*Figure D3b. Kilograms of air vs. time [h] gives mass flow.*

The calculations are in table D3.
Appendix D4:

Calculating the hydrogen velocity through the tubes

For determining what pipe diameter to use, a simple calculation was done, to check for the hydrogen velocity not to be supersonic.

To calculate the velocity through the pipes, we assume that the thermal efficiency produced by the CNG fueling is equal to that produced by the hydrogen fueling.

\[ \eta_{\text{H}_2} = \eta_{\text{CNG}} \]

Or \[ \frac{P}{\dot{m}_{\text{CNG}} LHV_{\text{CNG}}} = \frac{P}{\dot{m}_{\text{H}_2} LHV_{\text{H}_2}} \]

or \[ \dot{m}_{\text{H}_2} = \dot{m}_{\text{CNG}} \frac{LHV_{\text{CNG}}}{LHV_{\text{H}_2}} = 0.4167 \dot{m}_{\text{CNG}}, \]

where \( LHV_{\text{CNG}} = 50 \text{ MJ/kg} \) is the lower heating value for methane (good approximation for natural gas) and \( LHV_{\text{H}_2} = 120 \text{ MJ/kg} \) is the lower heating value for hydrogen.

From this calculation the diesel pilot lower heating value and mass flow can be omitted from both sides, canceling out.

Calculating the velocities for 2 different mass flows: 0.5 kg/h hydrogen versus 2 kg/h natural gas, considered as 25%, and 0.25 kg/h hydrogen versus 2 kg/h natural gas, which is considered 12.5% from the CNG flow. For preliminary calculations it is considered that the gaseous mixture has a percentage of 25% \( \text{H}_2 \) and 75% CNG.

Thus \[ \dot{m}_{\text{CNG}} LHV_{\text{CNG}} = 0.75 \dot{m}_{\text{CNG}} LHV_{\text{CNG}} + 0.25 \dot{m}_{\text{H}_2} LHV_{\text{H}_2} \]

\[ \dot{m}_f = \rho A v \Rightarrow v = \frac{\dot{m}_f}{A \rho} \]

For calculating the cross sectional area of the tubes, we need to know the interior diameter of the tube. For a \( \frac{1}{4} '' \) stainless steel tube, the ID(interior diameter) is:
ID = d_e - d_{walls} = 0.250 - 0.070 = 0.180 [in] = $4.572 \times 10^{-3}$ [m], and the cross sectional area is

$$A_1 = \frac{\pi d_1^2}{4} = 1.6417 \cdot 10^{-5} [m^2]$$

For a 1/2 " stainless tube, the ID(interior diameter) is :

ID = d_e - d_{walls} = 0.50 - 0.130 = 0.370 [in] = $9.398 \times 10^{-3}$ [m], and the cross sectional area is:

$$A_1 = \frac{\pi d_1^2}{4} = 6.9368 \cdot 10^{-5} [m^2]$$

The hydrogen density has to be calculated at the injection pressure. Since the mass flow meter has a maximum rated pressure above which it cannot be operated (it will be damaged), is a good assumption to calculate the hydrogen density at that allowable pressure. According to the manufacturer specifications, the maximum gas pressure for the "Micromotion" mass flow meter cannot exceed 2600 [psi] = 17.9 [MPa].

Since molecular hydrogen is considered ideal gas, for each mole of H₂ there are $22.4 \text{ dm}^3$. A mole of hydrogen has a molecular weight $MW_{H_2} = 2$ [g].

Thus the hydrogen density can be calculated at maximum flow meter pressure as:

$$\rho_{H_2} = \frac{MW_{H_2}}{N_{mole_{H_2}}} \cdot \frac{P_{max}}{P_{ambient}} \cdot \frac{T_{operate}}{T_{ambient}} = \frac{2 \cdot 10^{-3} [kg]}{22.4 \cdot 10^{-3} [m^3]} \cdot \frac{179}{1.013} = 15.7 \left[ \frac{Kg}{m^3} \right],$$

where the temperature of hydrogen injection was taken equal to the ambient temperature (i.e.293.15 K).

Therefore at maximum allowable pressure (almost 18 MPa) for the "Micromotion" mass flow meter, the hydrogen density is 15.7 kg/m$^3$. This is the highest hydrogen density in these conditions.

It is then possible to estimate the hydrogen velocity through the stainless lines, using
The calculations could be done for two different cross sectional area for the tubes.

For 1/4" tube : \( v_{1/4} = \frac{0.825 \text{ kg}}{3600 \text{ s}} \cdot \frac{1}{15.7 \frac{\text{ kg}}{\text{ m}^3} \cdot 1.64 \cdot 10^{-8} \text{ m}^2} = 0.91 \text{ m/s} \).

For 1/2" tube : \( v_{1/2} = \frac{0.825 \text{ kg}}{3600 \text{ s}} \cdot \frac{1}{15.7 \frac{\text{ kg}}{\text{ m}^3} \cdot 6.93 \cdot 10^{-8} \text{ m}^2} = 0.21 \text{ m/s} \).

**Conclusion:**

The calculated velocities for hydrogen are far smaller than supersonic velocity in the air

(\( \approx 340 \text{ m/s} \)).

**Note:**

Iterations could be done for smaller pressure, such as p= 2000 psi=13.6 MPa. That was in fact the injection pressure for both CNG and hydrogen.

Hydrogen density can be calculated as \( \rho_{\text{H}_2} = 11.98 \text{ [kg/m}^3\text{]} \).

And the correspondent velocities through the 1/2" and 1/4 " tubes, are:

\( v_{1/2} = 0.27 \text{[m/s]}, \text{ and } v_{1/4} = 1.16 \text{[m/s]} \). (still a lot smaller than 340 m/s).

Based on these calculations, all the connection between the hydrogen bottle and the flow meter, and further to the mixing tee, and into the engine, were done with 1/4" stainless steel tubing.

It was ensured that by using that tubing size, the hydrogen flow does not have velocities close to supersonics.
Appendix D5:

Nominal Natural Gas Analysis for Gas Received from Westcoast Energy Inc.

<table>
<thead>
<tr>
<th></th>
<th>% Volume</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitrogen</td>
<td>0.7</td>
</tr>
<tr>
<td>Carbon Dioxide</td>
<td>0.2</td>
</tr>
<tr>
<td>Methane</td>
<td>95.5</td>
</tr>
<tr>
<td>Ethane</td>
<td>2.9</td>
</tr>
<tr>
<td>Propane</td>
<td>0.5</td>
</tr>
<tr>
<td>Iso-Butane</td>
<td>0.05</td>
</tr>
<tr>
<td>n-Butane</td>
<td>0.1</td>
</tr>
<tr>
<td>Penthane</td>
<td>0.04</td>
</tr>
<tr>
<td>Hexanes</td>
<td>0.01</td>
</tr>
<tr>
<td><strong>Total</strong></td>
<td><strong>100</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Gross Higher Heating Value</th>
<th>[MJ/m^3]</th>
</tr>
</thead>
<tbody>
<tr>
<td>at 15 °C and 101.325 kPA</td>
<td>38.6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Relative Density</th>
<th>ro gas/ ro air</th>
</tr>
</thead>
<tbody>
<tr>
<td>at 15 °C and 101.325 kPA</td>
<td>0.58</td>
</tr>
<tr>
<td></td>
<td>0.7134 kg/m^3</td>
</tr>
</tbody>
</table>

| Average Molecular Weight  | 16.78 |

<table>
<thead>
<tr>
<th>Total sulphur</th>
<th>[mg S / m^3]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>25 (18 ppm)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Hydrogen Sulphide</th>
<th>[mg/m^3]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2 (1.5 ppm)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Water content</th>
<th>[mg/m^3]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30 (40 ppm)</td>
</tr>
</tbody>
</table>

The decrease in BC Gas Content of LHV is due to the removal of heavier components from its composition. The total sulfur includes naturally occurring sulfur compounds plus added odorant (approx. 5 mg S/ m^3 or 3.5 ppm) plus hydrogen sulfide.

The monthly and daily variation of lower heating value (LHV) is given in Fig. D5a and D5b.
Figure D5a. Monthly variation of LHV for CNG, according with BC-Gas.
Figure D5b. Daily variation of LHV for CNG, according with BC-Gas.
Appendix D6:

Adiabatic flame temperature

To illustrate the importance of the temperature on NOx formation, the adiabatic flame temperatures was calculated and compared for both methane and hydrogen oxidation.

Calculation for methane

Since the adiabatic flame temperature characterize diesel combustion and since NO forms at combustion temperature, the adiabatic flame temperature is an indicator of NO production. Later in this research a mixture of hydrogen and CNG was used as fuel for the engine, so a comparison of flame temperatures is should relate to NOx production. In these calculations the combustion product dissociation has been neglected, thus the calculated flame temperature will be overestimated. For this consideration the injection pressure doesn't influence these calculations.

First, a stoichiometric methane-air ratio (Φ =1) oxidation reaction will be considered for calculating the adiabatic flame temperature.

\[
\begin{align*}
\text{CH}_4 + 2(\text{O}_2 + 3.76\text{N}_2) & \rightarrow \text{CO}_2 + 2\text{H}_2\text{O} + 7.52 \text{ N}_2 \\
\text{Reactants} & \quad \text{Products}
\end{align*}
\]

Reactants: Products

\[
H_R = \sum_{i=1}^{n} n_i (h_{f,i} + \Delta h_i) \quad \text{and} \quad H_P = \sum_{e=1}^{m} n_e (h_{f,e} + \Delta h_e)
\]

where \(H_R\) and \(H_P\) are the enthalpy of reactants, respectively products.

\(h_{f,298}^0\) is the enthalpy of formation at standard temperature.

\(\Delta h_i = h-h_{298}^0 = (\Delta h_i)_{298,01\text{MPa},T_P}\) is difference in enthalpy between any given state and the enthalpy of ideal gas at 298.15 K, 0.1MPa.
If $H_R = H_P$, it is possible to find the adiabatic flame temperature by successive iterations. Thus,

$$H_R = \sum_{i=1}^{n} n_i (h^o_f + \Delta h_i) = h^o_{fCH_4} = -74873 \left( \frac{kJ}{kmol} \right)$$

$$H_R = \sum_{i=1}^{n} n_i (h^o_f + \Delta h_i) = 1(h^0_{fCO_2} + \Delta h_{CO_2}) + 2(h^0_{fH_2O} + \Delta h_{H_2O}) + 7.52(h^0_{fN_2} + \Delta h_{N_2})$$

(See Table D6a.)

To make a temperature comparison between reaction with the theoretical air required and the reaction with 10% excess air, a second equation is written as:

$$CH_4 + 2(1.1)(O_2+3.76N_2) \rightarrow 0.95CO_2 + 0.05 CO + 2H_2O + 0.225 O_2 + 8.27 \quad (2)$$

Similarly the reactants and products enthalpy can be calculated (See Table D6a.)

From these calculations it could be concluded that for methane oxidation with theoretical quantity of air or with 10% excess air, a reducing of adiabatic flame temperature with 200 degrees is possible, thus the potential for decreasing the NOx production.

**Calculation for hydrogen**

In the following the hydrogen adiabatic flame temperature is calculated. The purpose is to compare the two adiabatic temperatures, see which one is higher, and validate the fact that the NOx concentration is influenced by the high temperature.

The adiabatic flame temperature for hydrogen is calculated as:

$$H_2 + 1/2(O_2+3.76N_2) \rightarrow H_2O + 1.887 N_2 \quad (3)$$

Reactants Products
The maximum adiabatic flame temperature can be achieved with a stoichiometric mixture fuel-air ratio ($\Phi = 1$), thus it can be controlled by the amount of excess air that is used. As was mentioned before the production of NO, is maximum for such temperature.

The hydrogen will be taken at 298 K and the air at 800K.

$$H_R = \sum_{i=1}^{n} n_i (h_{fi} + \Delta h_{i}) \quad \text{and} \quad H_P = \sum_{e=1}^{m} n_e (h_{fe} + \Delta h_{e})$$

where $H_R$ and $H_P$ are the enthalpy of reactants, respectively products.

$h^{0}_{f, 298}$ is the enthalpy of formation at standard temperature.

For $H_R = H_P$, one can determine the adiabatic flame temperature by successive iterations.

Thus,

$$H_R = \sum_{i=1}^{n} n_i (h^{0}_{fi} + \Delta h_{i}) = h^{0}_{fi, 298}$$

$$H_R = \sum_{e=1}^{m} n_e (h^{0}_{fe} + \Delta h_{e}) = h^{0}_{fe, 298} + \Delta h_{H_2O} + 1.887 + \Delta h_{N_2}$$

For calculation and iterations see Table D6b.

From these calculations the temperature which satisfies the conditions $H_R = H_P$ is 3325 K.

This temperature is due to hydrogen burning only and is susceptible of producing a lot of NO, In fact as it will be shown the NO production doesn't increases dramatically, because the hydrogen is combined in different percentages with CNG, and therefore the overall flame temperature is lower than that for pure hydrogen.
Table D6a. The adiabatic flame temperature calculations

The calculations were done for methane (CH4) using Gordon van Wylen's Fundamentals of classical Thermodynamics textbook (pages 558-560, 575, 757-760, 770).

The first calculation and two iterations were done for 100% air. The second calculation and two iterations were done for 110% air.

ATTENTION: Use equation 4.5.1. for the 100% case, and 4.5.2. for the 110% case.

<table>
<thead>
<tr>
<th>T</th>
<th>hCO2</th>
<th>dhCO2</th>
<th>hCO</th>
<th>dhCO</th>
<th>hH2O</th>
<th>dhH2O</th>
<th>dhO2</th>
<th>dhN2</th>
<th>Hp for 100% air</th>
<th>Hp for 110% air</th>
</tr>
</thead>
<tbody>
<tr>
<td>hch4=-74873</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>p770</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2000</td>
<td>-393522</td>
<td>91439</td>
<td>-110527</td>
<td>56743</td>
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<td>72788</td>
<td>59176</td>
<td>56137</td>
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<td>-133335.4</td>
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<tr>
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<td>-393522</td>
<td>97500</td>
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<td>-241826</td>
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<td>59750</td>
<td>-174414</td>
<td>-85213.97</td>
</tr>
<tr>
<td>2200</td>
<td>-393522</td>
<td>103562</td>
<td>-110527</td>
<td>64012</td>
<td>-241826</td>
<td>83153</td>
<td>66770</td>
<td>63362</td>
<td>-130823.8</td>
<td>-37098.16</td>
</tr>
<tr>
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<td>109670</td>
<td>-110527</td>
<td>67669</td>
<td>-241826</td>
<td>88447</td>
<td>70611</td>
<td>67001</td>
<td>-86762.48</td>
<td>11525.745</td>
</tr>
<tr>
<td>2400</td>
<td>-393522</td>
<td>115779</td>
<td>-110527</td>
<td>71326</td>
<td>-241826</td>
<td>93741</td>
<td>74453</td>
<td>70640</td>
<td>-42700.2</td>
<td>60150.825</td>
</tr>
</tbody>
</table>

A) Iterations for theoretic air

<table>
<thead>
<tr>
<th>HR=-74873 MJ/kmol</th>
<th>HR=74873 MJ/kmol</th>
</tr>
</thead>
<tbody>
<tr>
<td>2300 -393522 109670</td>
<td>-241826 88477 67001</td>
</tr>
<tr>
<td>2350 -393522 112725</td>
<td>-241826 90692 68820</td>
</tr>
<tr>
<td>2400 -393522 115779</td>
<td>-241826 93741 70640</td>
</tr>
<tr>
<td>2300 -393522 109670</td>
<td>-241826 88477 67001</td>
</tr>
<tr>
<td>2325 -393522 111198</td>
<td>-241826 89720 67911</td>
</tr>
<tr>
<td>2350 -393522 112725</td>
<td>-241826 90962 68820</td>
</tr>
</tbody>
</table>

Therefore the adiabatic flame temperature for methane calculated for a stoichiometric fuel-air ratio was found 2325 K (2052 C)
Therefore the adiabatic flame temperature for methane calculated for an excess air of 110% (i.e. +10% in respect with theoretical air required) was found 2125 K (1852 °C).

It can be seen that by adding more air, the adiabatic flame temperature can be decreased with 200 degrees.

**Conclusion:**

> The adiabatical flame temperature for Methane will be compared with the adiabatical flame temperature for Hydrogen to see which is greater.

> The NOx formation is very dependent on high temperature, and since the adiabatical flame temperature is the highest, it will theoretically produce the maximum NOx.
Table D6b. The adiabatic flame temperature for H2

<table>
<thead>
<tr>
<th>T [K]</th>
<th>hH2</th>
<th>hH2O</th>
<th>dhH2O</th>
<th>dhN2</th>
<th>Hproducts</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>p761</td>
<td>p770</td>
<td>p760</td>
<td>p757</td>
<td></td>
</tr>
<tr>
<td>2000</td>
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<td>72788</td>
<td>56137</td>
<td>-63107</td>
</tr>
<tr>
<td>2200</td>
<td>59865</td>
<td>-241826</td>
<td>83153</td>
<td>63362</td>
<td>-39109</td>
</tr>
<tr>
<td>2400</td>
<td>66915</td>
<td>-241826</td>
<td>93741</td>
<td>70640</td>
<td>-14787</td>
</tr>
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<td>2600</td>
<td>74082</td>
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<td>104520</td>
<td>77693</td>
<td>9301</td>
</tr>
<tr>
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<td>83323</td>
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<td>92715</td>
<td>59675</td>
</tr>
<tr>
<td>3200</td>
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<td>137756</td>
<td>100134</td>
<td>84883</td>
</tr>
<tr>
<td>3400</td>
<td>103736</td>
<td>-241826</td>
<td>149073</td>
<td>107577</td>
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</tr>
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<td>115042</td>
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</tr>
<tr>
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<td>119077</td>
<td>-241826</td>
<td>171981</td>
<td>122526</td>
<td>161362</td>
</tr>
<tr>
<td>4000</td>
<td>126864</td>
<td>-241826</td>
<td>183552</td>
<td>130027</td>
<td>187087</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>T [K]</th>
<th>hH2</th>
<th>hH2O</th>
<th>dhH2O</th>
<th>dhN2</th>
<th>Hproducts</th>
</tr>
</thead>
<tbody>
<tr>
<td>3200</td>
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<td>-241826</td>
<td>137756</td>
<td>100134</td>
<td>84882.858</td>
</tr>
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<td>143414.5</td>
<td>103855.5</td>
<td>97563.8285</td>
</tr>
<tr>
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<td>103736</td>
<td>-241826</td>
<td>149073</td>
<td>107577</td>
<td>110244.799</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>T [K]</th>
<th>hH2</th>
<th>hH2O</th>
<th>dhH2O</th>
<th>dhN2</th>
<th>Hproducts</th>
</tr>
</thead>
<tbody>
<tr>
<td>3300</td>
<td>99961.5</td>
<td>-241826</td>
<td>143414.5</td>
<td>103855.5</td>
<td>97563.8285</td>
</tr>
<tr>
<td>3350</td>
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<td>-241826</td>
<td>146244</td>
<td>105716.5</td>
<td>103905.0355</td>
</tr>
<tr>
<td>3400</td>
<td>103736</td>
<td>-241826</td>
<td>149073</td>
<td>107577</td>
<td>110244.799</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>T [K]</th>
<th>hH2</th>
<th>hH2O</th>
<th>dhH2O</th>
<th>dhN2</th>
<th>Hproducts</th>
</tr>
</thead>
<tbody>
<tr>
<td>3300</td>
<td>99961.5</td>
<td>-241826</td>
<td>143414.5</td>
<td>103855.5</td>
<td>97563.8285</td>
</tr>
<tr>
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<td>100905.5</td>
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<td>144829.5</td>
<td>104786.5</td>
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</tr>
<tr>
<td>3350</td>
<td>101849</td>
<td>-241826</td>
<td>146244</td>
<td>105716.5</td>
<td>103905.0355</td>
</tr>
</tbody>
</table>
Table D7. Uncertainty in Thermal Efficiency E

\[ E = 2 \cdot \pi \cdot T \cdot N / (60 \cdot H) \]

<table>
<thead>
<tr>
<th></th>
<th>LHVd</th>
<th>dT</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>LHVg</td>
<td>50</td>
<td>dN</td>
<td>20</td>
</tr>
</tbody>
</table>

\[ \text{d}E / E = \sqrt{ (\text{d}T / T)^2 + (\text{d}N / N)^2 + (\text{d}LHVh / LHVh)^2 } \]

\[ H = (M_d \cdot \text{LHVd} + M_g \cdot \text{LHVg} + M_h \cdot \text{LHVh}) \cdot 1000 / 3600 \]

**Typical Values for 5 bar bmep, for 19 c.a. degrees BOI.**

<table>
<thead>
<tr>
<th>T (Nm)</th>
<th>N (rpm)</th>
<th>bmep</th>
<th>Md (kg/hr)</th>
<th>Mg (kg/hr)</th>
<th>Mh (kg/hr)</th>
<th>H (x100%)</th>
<th>E</th>
<th>dMd</th>
<th>dMg</th>
<th>dMh</th>
<th>dH</th>
<th>dT/T</th>
<th>dN/N</th>
<th>dH/H</th>
<th>dE</th>
</tr>
</thead>
<tbody>
<tr>
<td>92.9</td>
<td>1197</td>
<td>5.02</td>
<td>0.513</td>
<td>2.274</td>
<td>0.000</td>
<td>37.782</td>
<td>0.280</td>
<td>0.009</td>
<td>0.040</td>
<td>0.000</td>
<td>0.664</td>
<td>0.011</td>
<td>0.017</td>
<td>0.018</td>
<td>0.027</td>
</tr>
<tr>
<td>93.8</td>
<td>1204</td>
<td>5.07</td>
<td>0.478</td>
<td>2.140</td>
<td>0.250</td>
<td>43.894</td>
<td>0.245</td>
<td>0.009</td>
<td>0.040</td>
<td>0.002</td>
<td>0.731</td>
<td>0.011</td>
<td>0.017</td>
<td>0.017</td>
<td>0.026</td>
</tr>
<tr>
<td>94.1</td>
<td>1201</td>
<td>5.09</td>
<td>0.430</td>
<td>2.030</td>
<td>0.500</td>
<td>50.182</td>
<td>0.197</td>
<td>0.009</td>
<td>0.040</td>
<td>0.005</td>
<td>0.832</td>
<td>0.011</td>
<td>0.017</td>
<td>0.017</td>
<td>0.026</td>
</tr>
</tbody>
</table>

**Typical Values for 3 bar bmep**

<table>
<thead>
<tr>
<th>T (Nm)</th>
<th>N (rpm)</th>
<th>bmep</th>
<th>Md (kg/hr)</th>
<th>Mg (kg/hr)</th>
<th>Mh (kg/hr)</th>
<th>H (x100%)</th>
<th>E</th>
<th>dMd</th>
<th>dMg</th>
<th>dMh</th>
<th>dH</th>
<th>dT/T</th>
<th>dN/N</th>
<th>dH/H</th>
<th>dE</th>
</tr>
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<td>56.2</td>
<td>1206</td>
<td>3.04</td>
<td>0.352</td>
<td>2.210</td>
<td>0.000</td>
<td>34.948</td>
<td>0.185</td>
<td>0.009</td>
<td>0.040</td>
<td>0.000</td>
<td>0.664</td>
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<td>0.438</td>
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<td>41.786</td>
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<td>0.040</td>
<td>0.002</td>
<td>1.227</td>
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<td>0.017</td>
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<td>0.019</td>
<td>0.016</td>
<td>0.028</td>
<td>0.037</td>
</tr>
</tbody>
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Notes: Thermal Efficiencies errors were calculated and compared for very close BMEP figures (almost constant load).

Conclusion:

For this example (19 c.a. degrees BTDC) the calculated thermal efficiency is about 2.5% for 5 bars BMEP and about 3.5% for 3 bars BMEP, for any of the CNG, or CNG-hydrogen mixture injection.