STATUS OF THESIS

Title of thesis	Optimization of Injection Parameters in Compressed Natural Gas Direct Injection (CNG-DI) Spark Ignition Engine

I FIRMANSYAH

hereby allow my thesis to be placed at the Information Resource Center (IRC) of Universiti Teknologi PETRONAS (UTP) with the following conditions:

1. The thesis becomes the properties of UTP.
2. The IRC of UTP may make copies of the thesis for academic purposes only.
3. This thesis is classified as

	Confidential
√	Non-confidential

If this thesis is confidential, please state the reason:

-

The contents of the thesis will remain confidential for - years.

Remarks on disclosure:

-

Endorsed by

Signature of Author	Signature of Supervisor
Lubang Buaya No 69, Jakarta, Indonesia	Universiti Teknologi PETRONAS
Date : 30 JULY 2007	Date :

APPROVAL PAGE

UNIVERSITI TEKNOLOGI PETRONAS

Approval by Supervisor (s)

The undersigned certify that they have read, and recommend to The Postgraduate Studies Programme for acceptance, a thesis entitled "**Optimization of Injection Parameters in Compressed Natural Gas Direct Injection (CNG-DI) Spark Ignition Engine**" submitted by (Firmansyah) for the fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

____

Date

Signature : ______________________

Main supervisor : Assoc. Prof. Dr. Abdul Rashid A. Aziz.

Date : ______________________

Signature : ______________________

Co-Supervisor : ______________________

Date : ______________________

TITLE PAGE

UNIVERSITI TEKNOLOGI PETRONAS

Optimization of Injection Parameters in Compressed Natural Gas Direct Injection (CNG-DI) Spark Ignition Engine

By

Firmansyah

A THESIS

SUBMITTED TO THE POSTGRADUATE STUDIES PROGRAMME AS A REQUIREMENT FOR THE

DEGREE OF MASTER OF SCIENCE

MECHANICAL ENGINEERING

BANDAR SERI ISKANDAR,

PERAK

2007

DECLARATION

I hereby declare that the thesis is based on my original work except for quotations and citations which have been duly acknowledged. I also declare that it has not been previously or concurrently submitted for any other degree at UTP or other institutions.

Signature : ____________________

Name : FIRMANSYAH____________________

Date : 30 JULY 2007____________________

ACKNOWLEDGEMENT

First and foremost, I would like to give my sincere thanks to ALLAH SWT, the almighty God, the source of my life and hope for giving me the strength and wisdom to complete the research.

I am most grateful to my supervisor Assoc. Prof.. Dr. Abdul Rashid. A. Aziz for giving me an opportunity and guidance to pursue a master degree. Many times, his patience and constant encouragement has steered me to the right direction. I would also like to thank my co-supervisor Mr. M. Arif. M. Noor for his suggestions, criticism and helps. Further, I would like to thank Assoc. Prof. Dr. Chalilullah Rangkuti for the support and guidance.

I would like also express my gratitude to all Research officers from Center for Automotive Research, UTP, En. Shahrul, En Gezmir, En. Saiful, En. Razali, En Fahmi, Ms Hasniza for their effort in helping during the experimental works. Special thanks also to my postgraduate friends for their encouragement and friendship. My sincere thanks also to postgraduate office staffs, Kak Norma, Kak Haslina, Pn Kamaliah and Bang Fadhil for their assistance during my study.

At last and most importantly, I would like to thank my family for their open-mindedness and endless support. They are always close to my heart.

ABSTRACT

Energy security and environmental problems boosts the development on automotive industry. CNG as an alternative fuel has been applied in the vehicle to reduce the engine emissions with drawback in engine performance. It has lower performance compared to gasoline due to lower volumetric efficiency, lower energy content, and longer combustion durations. However, high octane number of CNG is beneficial in increasing engine efficiency. Direct injection application on CNG engine is expected to increase the volumetric efficiency of the engine to get further increase in engine performance while still maintaining the low engine emissions. In order to improve the engine performance, the objectives of this research are investigating the combustion characteristics of CNG direct injection engine and optimization of injection parameters to obtain high performance engine.

This research was carried out on a dedicated natural gas-fuelled spark ignition engine with a compression ratio of 14:1. This research was carried out in two stages; simulation process and experimental works. The simulation stage used GT-POWER software to predict the experimental results while the experimental works was carried out on a central direct injection system, where the injector was placed on the centre of the combustion chamber with spark plug offset by 6 mm from the center. A wide open throttle condition was set to examine the best performance of CNG-DI engine within operating range from 2000 RPM to 5000 RPM. The test was conducted according to SAE standard for Engine performance and Testing (SAE J1995, Engine Power Test Code-Spark ignition and Compression ignition-Gross power rating). Injection parameters such as injection timing, injection pressure and injector spray angle were investigated to find out the effect of the injection parameters to the CNG-DI engine performance. While, ignition timing is adjusted to achieved the maximum brake torque (MBT).

This research has come out with the injection strategy to get an optimum condition for direct injection operation with CNG fuelled engine. Firstly, port simulated injection timing is good to be applied in high engine speed condition with lower injection pressure due to better mixing capabilities. While secondly, partial and full direct

injection systems with high injection pressure is good to be applied in lower engine speed due to better volumetric efficiency. Furthermore, partial direct injection systems have shown better performance for wide engine speed. As for the injector spray angle, each injector has its optimum operating condition where the WAI injector performs better at high engine speed. In contrary, the NAI injector has higher performance at lower engine speed.

ABSTRAK

Gas asli sebagai bahan api alternatif telah digunakan pada enjin untuk mengurangkan kotoran asap walaupun terjadi kejatuhan prestasi enjin. Kejatuhan prestasi enjin gas asli adalah disebabkan oleh kecekapan isi padu yang rendah, ketumpatan tenaga yang rendah dan tempoh pembakaran gas asli yang lama. Walau bagaimanapun, kelebihan menggunakan gas asli bagi aplikasi enjin adalah kemampuan enjin yang dapat mempunyai nisbah mampatan lebih tinggi disebabkan nombor oktana yang tinggi. Sistem suntikan terus untuk enjin gas asli dijangka untuk meningkatkan isi padu bagi memperolehi prestasi enjin gas asli yang tinggi dengan kotoran asap yang rendah. Kajian ini bertujuan untuk pengoptimuman parameter-parameter suntikan diperlukan bagi memperolehi hasil terbaik daripada enjin gas asli dan penyelidikan karakteristik pembakaran gas asli.

Kajian ini dijalankan terhadap enjin gas asli khusus-nyalaan bunga api dengan 14 nisbah mampatan. Sistem suntikan terus berpusat digunakan. Penyuntik bahan api ditempatkan di pusat kebuk pembakaran di samping palam pencucuh yang ofset sebanyak 6mm daripada pusat kebuk. Bukaan pendikit penuh ditetapkan untuk memeriksa prestasi terbaik enjin CNG-DI dalam lingkungan operasi daripada 2000 ppm ke 5000 ppm. (ppm= pusingan per minit). Ujian dijalankan berdasarkan piawaian SAE untuk Engine performance and Testing (SAE J1995, Engine Power Test Code-Spark ignition and Compression ignition-Gross power rating). Penyelidikan kesan parameter-parameter suntikan seperti pemasaan suntikan, tekanan suntikan dan sudut semburan penyuntik terhadap prestasi enjin CNG-DI telah dijalankan. Di samping itu, pemasaan pencucuhan dilaraskan untuk memperolehi daya kilas brek maksima (MBT). Hasil eksperimen dikelaskan berdasarkan setiap parameter-parameter suntikan seperti berikut.

Kajian ini telah menemukan strategi suntikan untuk mendapatkan keadaan yang optimum bagi operasi suntikan langsung dengan enjin yang menggunakan bahan api gas asli. Pertamanya, masa suntikan liang yang diselakukan adalah baik untuk diaplikasi dalam keadaan halaju tinggi enjin dengan tekanan suntikan yang lebih rendah disebabkan oleh keupayaan pembauran yang lebih baik. Keduanya, system

suntikan langsung separa dan penuh dengan tekanan suntikan tinggi adalah baik untuk diaplikasi dalam halaju rendah enjin disebabkan oleh kecekapan isi padu yang lebih baik. Tambahan pula, system suntikan langsung separa telah menunjukkan prestasi yang lebih baik untuk halaju luas enjin. Manakala untuk sudut semburan pemancit mempunyai keadaan operasinya yang optimum di mana pemancit WAI berfungsi dengan lebih baik pada halaju tinggi enjin. Sebaliknya pemancit NAI mempunyai prestasi lebih tinggi pada halaju rendah enjin.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

ABBREVIATIONS

ABDC	After Bottom Dead Center
AFR	Air Fuel Ratio
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTDC	Before Top Dead Center
CNG	Compressed Natural Gas
CO	Carbon Monoxide
COV	Coefficient of variation
DI	Direct Injection
EGR	Exhaust Gas Recirculation
FMEP	Friction mean effective pressure
HC	Hydrocarbon
IANGV	International Association for Natural Gas Vehicle
IMEP	Indicated Mean Effective Pressure
MBT	Maximum Brake Torque
NAI	Narrow Angle Injector
NO_x	Nitric Oxide
RCM	Rapid Compression Machine
UHC	Unburned Hydrocarbon
VVT	Variable Valve Timing
WAI	Wide Angle Injector
WOT	Wide Open Throttle

CHAPTER I

INTRODUCTION

1.1 Background

The rapid development of automotives industry is mostly driven by increasing concern of energy security and environmental problems. Hydrocarbon fuels have become the main energy source for over centuries and creating high dependencies on using hydrocarbon as fuels in every aspect in life. Hydrocarbons as a non-renewable energy source have a limited supply that eventually will be depleted especially for gasoline and diesel that have been used on the transportation sector for decades. It has forced the automotive industry to find an alternative fuel to reduce the dependencies on gasoline and diesel as fuel.

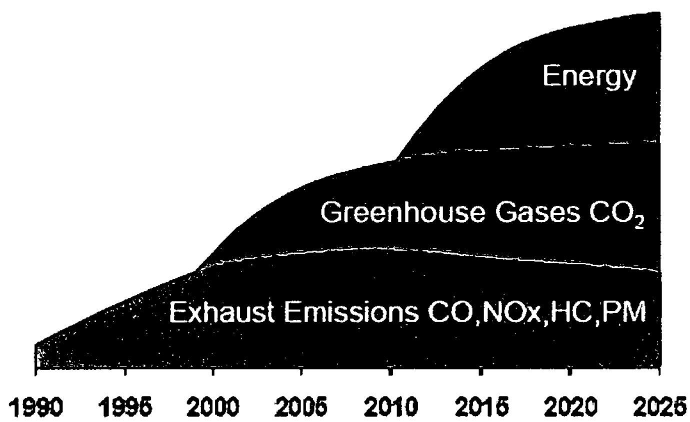


Figure 1-1 The evolution of environmental thinking[1]

Furthermore, environmental problems caused by the rapid development of automotive industry have become the concern for the future generation as seen in Figure 1-1 other than energy security. Automotive sector has contributed a large amount of air pollution and it has deteriorated the environment. In particular, air pollution such as CO and NO_x can bring hazard to humans. Therefore, the environmental regulator has created strict regulations that limit the exhaust emission allowed from vehicle to ensure the environment condition and reduce harmful gasses that are dangerous to humans. Natural gas that has methane as the main composition is the cleanest fuel after hydrogen that makes natural gas as an option to solved environmental problem (Figure 1-2).

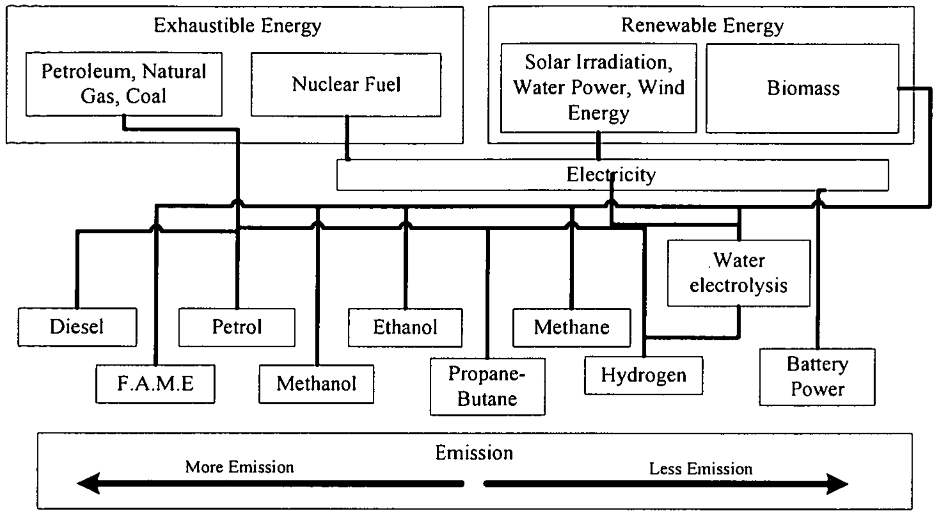


Figure 1-2 Energy option for vehicle[1]

The Europe Commission has discussed the direction of alternative fuel for the next decade mainly to overcome the problems stated earlier. The Commission proposed by 2020 the use of 10% natural gas, 8% biofuels, and 5% hydrogen for transportation, while the rest of the percentages is using the conventional fuel (gasoline and diesel). This perspective gives the picture that the natural gas development as fuel source for the next decade will rapidly increase especially in transportation sectors[1] because of natural gas potential that can gives significant contribution to both energy security and emissions problems.

The conversion to natural gas as vehicle fuel had been steadily increasing in many countries. Statistical data provided by IANGV (International Association for Natural Gas Vehicle) shows that Malaysia is in the 20th position among all the countries that has been using natural gas vehicle in the transportation sector after Argentina, Brazil, and Pakistan as the biggest three of natural gas vehicle fleet. (See Table 1-1)

In order to solve the energy security and environmental problems, automotive industry also has come out with systems that reduce emission and also fuel consumption. Technological developments such as direct injection (DI), variable valve timing (VVT), exhaust gas recirculation (EGR) and on-board diagnosis system (OBD) are proven to exhibit the ability to reduce fuel consumption and the emission levels to a very low limit.

Table 1-1 Natural gas vehicle statistics[2]

Position	Country	Vehicles*	Refuelling Stations	VRA**	Last Updated
1	Argentina	1,459,236	1,400	32	5-Dec-06
2	Brazil	1,357,239	1410		7-Mar-06
3	Pakistan	1,300,000	1230		7-Apr-06
4	Italy	410,000	558		6-Dec-06
5	India	334,658	321		Apr 06
20	Malaysia	19,000	46	1	6-Dec-06
	TOTALS	6,080,582	10,068	9,176	

* Includes both OEM and converted NGVs

** VRA = Number of Vehicle Re-fuelling Appliances

The natural gas products consist of CNG (compressed natural gas), LNG (liquefied natural gas) and LPG (liquid petroleum gas). Research has been done to compare the application of natural gas product in the internal combustion engine. LPG has higher thermal efficiency and lower fuel consumption although it has an issue with safety and high demand for household sector. CNG shows better starting and smooth acceleration but higher safety issues due to its specific gravity which is lower than air [3].

In today's application, CNG fuel was mainly applied in combination with other fuels in the bi-fuel or tri-fuel vehicle system. The fact that all fuel has different properties gives compensation to the engine setting which has to be adjusted to facilitate both fuel and create un-optimized setting for each fuel. Dedicated CNG engine is expected to utilize the advantages of using CNG as fuel which has high octane number to be applied with higher compression ratio and increase the engine efficiency. Moreover, Natural gas as an alternative fuel has potential to emit low exhaust emissions because of simplicity of the chemical bonding and has low H/C ratio.

Currently most CNG fuelled vehicles employed conventional engine technology for fueling system that use mixer to create air and fuel mixture and the engine performance for these systems is lower compared to gasoline engine. In order to improve the engine performance, the automotive company has employed fueling systems with multi port injection[4]. The new systems are claimed to perform better compared to mixer systems. Further development of CNG engine technology has focused on increasing the engine power that can be achieved by applying current SI engine technology to the CNG fuelled vehicle[4-6].

With the recent engine technological development, CNG engine has reach the direct injection stage in its application. In the implementation of EGR on CNG engine resulting as the NO_x and HC emission were very low especially with stoichiometric condition. However, EGR decreased the brake efficiency of the engine[7]. While CNG engine with turbocharger is beneficial at increasing power due to higher intake charge, the power increased is still lower at low engine speed[8]. Direct injection (DI) systems in CNG engine is another way to increase the volumetric efficiency. DI system could also enhance the mixing process forming turbulence inside the combustion chamber, due to high speed delivery

1.2 Problem Statement

The implementation of mixer and port injection system to the natural gas engine gives lower torque and power output compare to gasoline direct injection because of low volumetric efficiency of the gaseous fuel. CNG as an engine fuel has slow flame speed so that advancing ignition timing is needed to get a complete combustion. However, advance ignition timing would result in longer time fuel heat to be transferred to the wall and thus decreasing the engine efficiency. In addition, slow flame propagation of CNG is equals to longer combustion duration. Longer combustion duration could cause some portion of fuel left unburned during the exhaust stroke, and high hydrocarbon emission will be detected in the emission. Moreover, unburned fuel is indicating combustion efficiency of an engine, so that the combustion efficiency of CNG engine is lower compare to gasoline fuelled engine. As an outcome, the employment of CNG as an alternative fuels has been resulted 13-30% lower engine performance due to lower volumetric efficiency, combustion efficiency and longer combustion duration.

1.3 Objectives

The objectives of this research are:

1. To investigate the combustion characteristics of CNG using a direct injection system on a spark ignition engine
2. To obtained a higher power output of the CNG direct-injection engine by optimizing Injection Parameters.

1.4 Scope of Works

This research is basically focused on injection parameters on CNG direct injection spark ignition engine. The injection parameters are injection timing, injection pressure and injector spray angle on the range of engine speed from 2000 RPM to 5000 RPM. Moreover, this research was carried out in the full load wide open throttle condition on which ignition timing was set to get the maximum brake torque for each injection parameters.

1.5 Theses Organization

This thesis consists of seven chapters which are introduction, literature review, theoretical background, simulation, methodology, results and discussion, and conclusion. A whole range of systematic experimental works had been performed to meet the objective describe in chapter 1 of this thesis.

Chapter 1 describes the background of this research that also covered the current development of engine technology mainly for spark ignition engine. Further, chapter 1 consist of explanation of both advantages and disadvantages in this engine development technology. In addition, the objective and scope of works for this research is also oulined

Chapter 2 describe the past work that had been done on this research area and the position of this research in the current area. Mainly, it explains the fueling system technology of spark ignition engine using CNG as fuel and its development along with the characteristics of each technology.

Chapter 3 presents the theoretical background for this research which will be used as the basic for simulation and analysis stage. Chapter 3 elaborates the theory on engine performance along with its efficiency and combustion characteristics that occurs inside the combustion chamber. Further, this chapter briefly explains about emission resulting from the combustion process.

Chapter 4 discussed on the simulation process including the assumption being used, parameters of input and the model building process. It also explains the simulation results that will be used as the tools to set up the experimental investigation and help to explain the experimental results analysis.

Chapter 5 specifies the equipment that had been used in this research together with its measurement capabilities and characteristics. The data gathering process is also included in this chapter together with the constraint for each parameters being researched, as well as the equation used to analyze the data that had been gathered.

Chapter 6 illustrates the experimental results on engine performance such as torque, power, fuel consumption and engine efficiency. Furthermore, detailed analysis on the combustion process that occur in the combustion chamber and emission resulted during this process is carried out for every parameters in the certain range of engine speed.

Chapter 7 contains the concluding remarks of this research and proposed work for the future development.

CHAPTER II

LITERATURE REVIEW

This chapter will elaborate the development in internal combustion engine as well as the fueling systems technology. The information on CNG application in engine with the advantageous and the disadvantageous of this fuel will also be explained with the combustion strategy that has been employed to get better results derived from the engine.

2.1 Internal Combustion Engine development

Engines have been used as tools to generate power for centuries. In the early 1769, steam engine was used and became the platform for the next chapter of engine development. Throughout the 18^{th} and 19^{th} centuries, people started to search for alternative systems due to the expensive and complex systems of the steam engines. The internal combustion engine was a promising alternative system. Over the years, there were numerous proposals on patents for internal combustion engines. Otto and Langen introduced the first concept of internal combustion engine in 1866. It was assembled as a long vertical cylinder with piston and piston rod and it had a very low power-weight ratio. This first generation of internal combustion engine was called the Otto engine and it used gasoline as fuel source. Rudolph Diesel founded an alternative system to Otto engine in 1892, which is known as the diesel engine as shown in Figure 2-1. Diesel engine required higher compression ratio to achieve the auto ignition. Typically the compression ratio is above 19:1, while for Otto engine typical compression ratio is about 10:1 and it is limited by the knock limit. As the result, diesel engine has higher efficiency and lower fuel consumption compared to Otto engine.

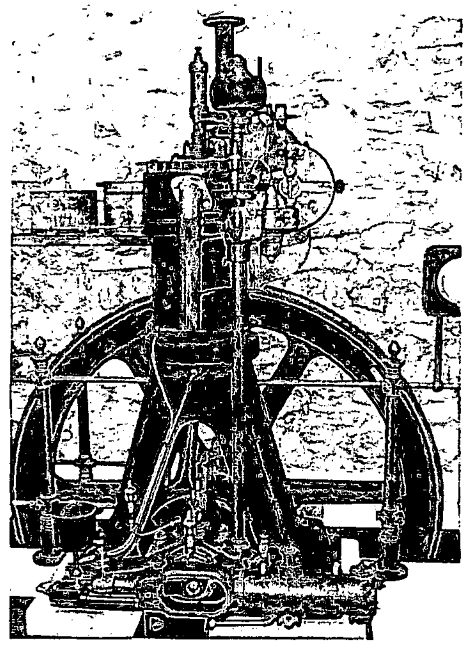


Figure 2-1 Diesel engine built by *MAN AG*
Source: www.wikipedia.com

The widespread utilizations of Otto and Diesel engine along the years had brought up energy and environment issues such as fuel depletion and environmental pollution. These issues have resulted in increased technological development of internal combustion engines. Continues improvement was done to increase the engine efficiency and to reduce emissions. The use of alternative fuels were considered to overcome some of these issues. Technology improvement includes variable valve timing (VVT), direct injection (DI), variable compression ratio, and on board diagnosis. The use of alternative fuel such as CNG, LNG, alcohol bond hydrocarbon, and hydrogen was considered to solve not only the emission issues, but also the fuel depletion issues[1].

Technology improvement and alternative fuel were affecting emission level of an engine in many ways. Through these developments, spark ignition engine performance and emission results are getting better each day with NOx is 80% lesser than compression ignition engine. However, the green house gas (CO2) emission of SI engine is higher compared to compression ignition (CI) engine[9].

2.2 CNG as an alternative fuel

The automotive industry is seen in the natural gas as a promising fuel to reduce the dependency on crude oil supply, as natural gas is expected to be available for a longer term compared to crude oil. The number of vehicle which is converted from diesel or gasoline to natural gas is increasing, and dedicated CNG-fueled vehicle shows significant increase in some countries[1]. The use of natural gas as engine fuel actually started way back in 1974 and 1979 where gas engine was used due to the shortage of gasoline and diesel fuels. Another factor that contributed to the development of the CNG engines is the tightening emission regulations in many developed countries to the allowable emission level of the conventional engine.

Increasing environmental concern boosts the CNG engines development because of the fact that the use of CNG resulted in a lower emission. Exhaust emission gases from engines such as NO_x, CO, HC, CO_2, and particulate matter have polluted the air. Release of the exhaust gas emissions into the atmosphere has contributed to smog pollution and the increase in the levels of dangerous ground level ozone.

Natural gas is primarily composed of methane and has a high H/C ratio. This makes CNG favorable fuel to reduce CO_2 and CO emission[9-11]. Although hydrocarbon emissions of CNG are lower than conventional SI engine using port injection, the HC emission in form of methane shows contradictory results as it is higher compared to gasoline-fueled engine. Some experimental works have found that NO_x emission of CNG engine was higher than conventional SI engine[12-14]. Table 2-1 shows the emission standard that is applied for CNG engine.

Table 2-1 Emission standard for CNG engine[15]

	NMHC	THC	CO	NO_X	PM
	[g/kWh]				
EURO I	-	1.1	4.5	8	0.36
EURO II	-	1.1	4	7	0.15

The use of CNG from the point of view of engine efficiency has some advantages and the disadvantages. Advantages of CNG include, firstly, CNG possesses high octane number (RON130) that could be applied to a higher compression ratio and increase the engine thermal efficiency. And secondly, the flammability limits of CNG fuel are wider than gasoline, so it can operate on a very lean limit[16, 17]. As for the results of wider flammability limits, CNG can be operated at a lower fuel consumption than that of gasoline.

Disadvantages of CNG as fuel are that, firstly, power produced from CNG engine shows approximate 10% lower output compared to gasoline fuel[18-23]. This is due to gaseous state of CNG that decrease the engine volumetric efficiency[24, 25]. Secondly, energy content of CNG is lower, and has longer combustion duration[26]. And thirdly, natural gas engine conversion from gasoline fueled engine requires some modification on the fueling systems, valve train and ignition system[19]; However, other systems in CNG fueled engine basically operate on the same fundamental concept as that of gasoline fueled engine. Table 2-2 summarizes the conversion effect of spark ignition engine from gasoline -fueled to CNG-fueled vehicle.

Table 2-2 Conversion effect of Gasoline to CNG in Spark ignition engine[3]

Changes - Conversion from Gasoline to CNG	Effect	Scale
Change in fuel	Loss of power	8% to 15%
Lower energy density of fuel	Loss of range	40% or more
Increase in fuel storage volume	Loss of available space	signifcant
Increase in storage weight	Loss of acceleration	noticeable
Improvements due to methane fuelling	Improved emissions	signifcant
Fuel costs	Saving on fuel costs	30% or more
Savings in engine maintenance	Cleaner oil and longer life	noticeable

2.3 The Development of Natural Gas engine

CNG was first used as fuel for vehicles since 1920 to provide power for heavy commercial vehicle with spark ignition engine. And its rapid development happened during oil shocks in 1974 and 1979[27]. CNG seems to be the best alternatives those days because of its availability and affordable price. However the use of CNG engine was slowly terminated in 1953 as sufficient amount of gasoline became available and heavy truck with spark ignition engine fell out of favor. In the year after, the use of

CNG as fuel was slowly phased out by the use of gasoline and diesel in vehicle as these fuels were readily available. Recently the demands on CNG engine are steadily increasing as the world is facing the depletion of fuel source and increased concern on environmental condition. Further more, CNG development was pushed up by the strengthening emission regulation and with the full support from the government institutions.

By the end of the 20th century, many automotive industries and government institutions were giving their full attention concerning CNG engine development. These changes are mostly driven by stricter emission regulations that emphasize these institutions to find the best alternatives fuel. Current problem faced by conventional diesel and gasoline engine is the emissions emitted consist of CO_2, NO_x, CO and hydrocarbon. These emissions give detrimental effect to the environment as it can bring green house effect and some are harmful to humans.

As an alternative, CNG fuelled engine are developed to meet the power and efficiency of the available engines which are gasoline and diesel engine[25, 27, 28], with significantly lower harmful emission.

Among the innovations that the engineers and researchers have come up with regards to CNG engine are:

i. Engine format:

   Dedicated / Monofuel with higher compression ratio with supercharger or turbocharger, bi-fuel / dual fuel, tri-fuel engine

ii. Fuel delivery systems:

   Carburetor, Single port injection, Multi port injection, and Direct injection

iii. Combustion strategy:

   Stoichiometric and Lean burned combustion

2.3.1 Engine Format:

i. *Dedicated/Monofuel* engine is a dedicated engine that uses only natural gas as its fuel source. A dedicated engine has the advantage of being optimized to operate on natural gas. Thus it ensures maximum efficiency and minimum emission results. As the vehicle has been optimized for natural gas engine, the distance it can travel depends on the CNG volume carried by the vehicle
ii. *Bi-fuel/dual fuel* is the engine which operates on either natural gas or gasoline. Bi-fuel engine generally relies on gasoline to start the engine, thus a small portion of gasoline is still required to get a good cycle. These engines are usually optimized to run on one fuel. The deficiency in running CNG fuel are compensated by the use of a supercharger or turbocharger.
iii. A relatively recent technology, a *tri-fuel engine* combines a flex-fuel (flexible fuel) vehicle and natural gas engine vehicle. A flex fuel vehicle uses gasoline and ethanol, either exclusively or blended together. Tri-fuel vehicles first entered the market in 2005 in Brazil, where ethanol and NG were widely used as a transport fuel[29].

2.3.2 Fuel delivery system:

Fuel delivery system has been investigated along with other technology development and it has been developed over the years, from conventional Carburetor system to a sophisticated Direct-injection system. The purpose of the development is to ensure the precise amount of fuel supply that is required for combustion process. Along with precise control of fuel, mixing control of fuel and air is important parameters that are affected by fuel delivery methods. Controllable mixing can ensure the good and stable combustion takes place in the combustion chamber.

i. Carburetor

Carburetor is among the first fueling system that comes along with engine technology. It delivers fuel with a less complicated system and it has less control to the amount of fuel that enters the system as shown in Figure 2-2. More complicated carburetor has been introduced to control fuel delivery in such way. However, due to its nature, carburetor system has its limitation to control fuel to a very specific amount.

The basic principal of carburetor is differential pressure that occurs during intake process inside the intake manifold and throttle. High velocity of air entering intake manifold will pass the throttle and reduce the pressure in the throttle, while the reduction in diameter from intake to throttle will create vacuum condition. Pressure difference between fuel rail and throttle will pull out some amount of fuel and instantly mix with air. A homogeneous mixture will be created and enter the combustion chamber.

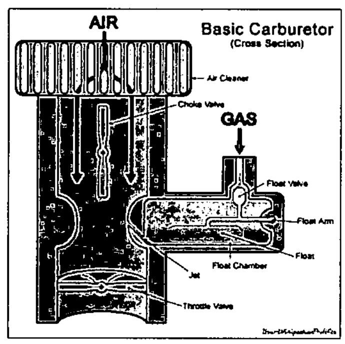


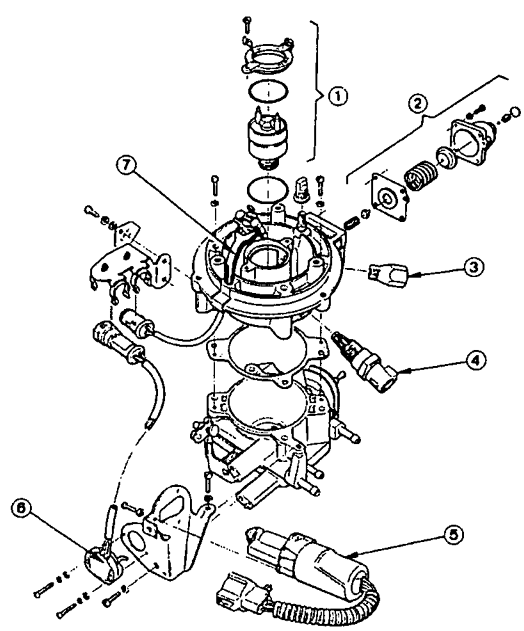
Figure 2-2 Carburetor

CNG fueled engine with carburetor system had 13-30% drop in engine performance compared to gasoline engine, and 13-17% performance drop if it used more complex system with additional AFR controller. As expected from CNG, the exhaust emission emitted from CNG engine was 90% and 50% lower for CO and HC emissions respectively compared to gasoline engine[20, 30].

ii. Single point injection / Throttle body fuel injection.

The improvement had been done to improve the control on fuel delivery with Single point injection (SPI), which was basically a replacement for carburetor systems. SPI was an incorporated electronically. It controlled fuel injector valves into the throttle body and was designed as a bolt-in replacement for carburetor, so the automakers did not have to make any drastic changes to their engine design.

Figure 2-3 shows the main component of SPI systems. The SPI system consists of fuel pump, fuel filter and carburetor unit. Power to the system is supplied by the power relay, which has built-in timer and allows throttle valves to continue to operate after the engine is switched off. SPI basically is a carburetor derivative. However, instead of having the induction vacuum sucking the fuel into venture, an injector forced fuel into the air flow. This system allows more precise control on fuel delivery by injecting the amount of fuel required by the combustion process. Furthermore, it can decrease the fuel consumption of an engine. CNG application on SPI system can increase the engine performance compared to carburetor system due to its precise control in regulating air-fuel ratio[30].



1 Fuel injector assembly
2 Fuel pressure regulator components
3 Fuel inlet connector
4 Air charge temperature (ACT) sensor
5 Throttle valve control motor
6 Throttle position sensor (TPS)
7 Fuel injector wiring

Figure 2-3 Single point injections
Source: Ford 1.4 CFI engine maintenance manual

SPI system of CNG had 14.4% lower IMEP output at 4500 RPM and 13.8% lower at rated speed in comparison with gasoline fuel because CNG fuel has small effect on cooling the intake charge as gasoline does. As the results, inlet temperature for CNG will be higher and reduces the volumetric efficiency of the engine[31].

iii. Multi Port Injection (MPI)

As the engine design advanced, single point injection (SPI) was phased out and replaced by multi-port fuel injection (MPI). This design placed single injector for each cylinder and the injector is normally attached either on the intake manifold (employed by 80% of MPI system) as seen on Figure 2-4 or on the cylinder head upstream (20% of MPI system employed), and both aimed right at the intake valve. Mostly, MPI system in gasoline engine inject the fuel when the intake valve is closed so that there will be an associated time lag between the injection event and fuel/air mixture induction into the combustion chamber. During this time lag, fuel and air will be mixed and create homogeneous mixture before entering the combustion chamber[32]. While, MPI systems of CNG engines inject the fuel when the intake valve is open, which eliminates the time lag factor[4, 33].

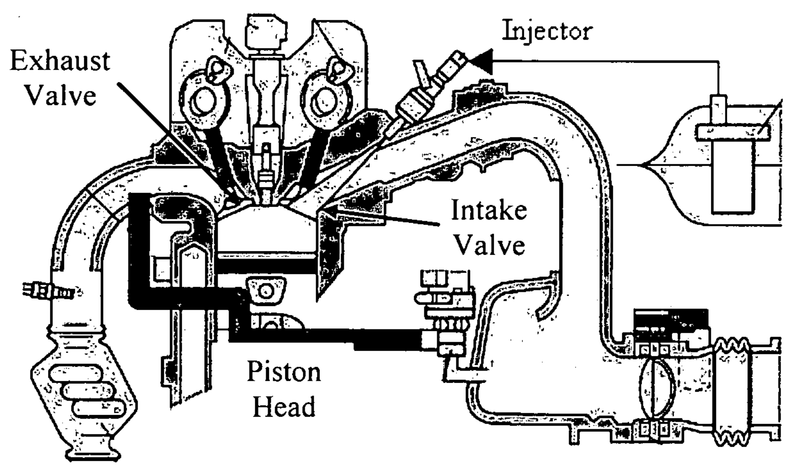


Figure 2-4 Multi port injection (MPI) [34]

The MPI system consists of three major parts, which are injectors, fuel pump and ECU. ECU is mainly used to control the injection event parameters such as injection timing and fuel required by each cylinder, such that by employing ECU, MPI systems use more sophisticated system compared to SPI system[2, 35]. However, MPI system utilization can accurately deliver the fuel requirement for each cylinder as such that the fuel metering will be enhanced and fuel consumption will be decreased. Moreover, MPI usage somehow increases the control on combustion process which improve the engine performance[33]. Along with increasing engine performance, the emissions emitted were also decreasing especially for CO and HC emission compared to carburetor system as shown in Table 2-3.

On the perspective of CNG fueled engine, MPI systems had 9.95% and 10.4% lower IMEP values for 4500 RPM and rated RPM respectively compared to MPI-gasoline. Further, it also has 3% to 5% better performance if compared to SPI-CNG system[31].

Table 2-3 Tailpipe emission (un-catalyzed) and fuel composition[36]

Vehicle		Tailpipe emissions-measured				Fuel consumption-measured			
Model	Fuel system	HC (g/km)	CO (g/km)	Nox (g/km)	HC+NOx (g/km)	(g/km) Absolute	(g/km) Relative	(l/100 km)	(km/l)
Large 2S' (150-200 cm3 2-stroke)	Carburetor (baseline)	8.5	10.9	0.04	8.5	33.4	-	4.43	22.6
	aSDI' (Euro I system)	1.06	1.47	0.09	1.16	19.6	-41%	2.58	38.7
		1.76	1.82	0.06	1.82	20.4	-39%	2.7	37
Large 4S' (150-200 cm3 4-stroke)	Carburetor (baseline)	0.85	11.1	0.17	1.02	246		3.33	30
	SePI' (Devel system)	0.5	1.34	0.25	0.72	20.5	-17%	2.79	35.9

iv. Direct Injection (DI) System

The most recent technology development on fuel delivery system is direct injection. Over the past three decades, a research goal has been to develop an internal combustion engine that combines the best feature from spark ignition and compression ignition engines. Such an engine would exhibit a brake specific fuel consumption approaching that of a diesel engine, while maintaining the operating characteristics and specific power output of gasoline engine. Direct injection-four stroke-spark ignition engine believes to be the answers to this objective because it does not throttle the inlet mixture to control the load. In direct injection system (Figure 2-5), the fuel is directly injected to the combustion chamber instead of separate chamber or port and the load is controlled by varying the amount of fuel that is injected to the cylinder.

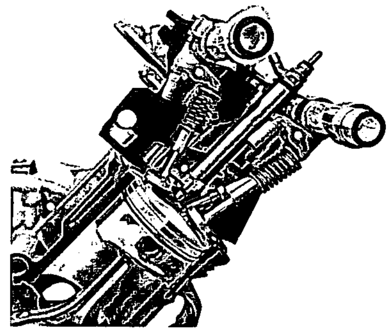


Figure 2-5 Direct injection systems
Source: http://www.emercedesbenz.com/

Typical spark ignition direct injection (SI-DI) engine consists of high pressure fuel pump, injector and ECU, where the ECU function is to control the amount of fuel injected and injection timing.

Direct injection system has several advantages compared to MPI system. These advantages are higher fuel economy, drive-ability, AFR controllability, combustion stability, better emission and increasing engine efficiency by 10-20%. However, DI has some limitation with a very complex control system, low injector performance and durability and higher amount of local NO_x[37].

In the CNG fueled engine, direct injection system offers a better control on AFR [12] which affects the exhaust emission of CNG engine. In addition, it can provide very low allowable lean limit down to 0.1 equivalence ratio, which indicates a very low fuel consumption[38]. Further, DI employment on CNG engine could reduce the cycle-by-cycle variation from 23.7% to 4% and 1.6% respectively for late and early injection mode with AFR equals to 23[39]. DI also offers faster combustion rate because of its capability to generate turbulence[40]. However, a drawback of DI-CNG engine is emitting high level of NO_x emission due to higher combustion temperature resulting from increased combustion process.

Other research that compared the direct injection method with mixer system and the results state that the maximum brake power for DI systems is 7.7% higher than that mixer engine, while the thermal efficiency of DI is increased up to 98.8% which resulted in decrease of NOx emissions[41].

2.3.3 Combustion strategy

Currently there are two types of combustion strategy that has been utilized on CNG-DI engine, stoichiometric and lean-burned combustion. Both systems are based on the specific amount of fuel injected to the system.

i. Stoichiometric Combustion

Stoichiometric or theoretical combustion is the ideal combustion process during which a fuel is burned completely and it depends on the fuel composition. CNG which primarily composed of methane had a stoichiometric air-fuel ratio around 16.9. Stoichiometric systems offer exceptionally clean combustion especially with the employment of EGR and three-way catalyst, where it can reduce more than 90% NO_x and methane emission[12]. On the other hand, stoichiometric system on CNG engine performance was down to 20% for BSFC and 15% for BMEP compared to gasoline fueled engine[14, 42]. Furthermore, CNG stoichiometric has slower burning rate and longer ignition delay compared to gasoline engine.[43]

Stoichiometric combustion has been used in light duty applications because it can be equipped by three-way catalyst and after treatment technology to meet light duty emission standard. The stoichiometric natural gas engine performs remarkably low emission results thanks to high catalyst efficiency compared to lean-burn combustion. However, the emission results are highly dependant on the stability of air-fuel ratio of the engine. As changes in AFR values could affect the emission results[44].

In addition, the engine performance could be improved and exhibit high tolerant on gas property variations with, the use of feedback control which is using an oxygen sensor to control air-fuel ratio. As an effect, high thermal efficiency of stoichiometric engine come at a price of higher combustion temperature and reduced fuel economy[2].

ii. Lean-burned Combustion

Lean-burned combustion is the combustion process during which a fuel is burned in the ratio that provides excess air. Lean burn combustion offers low fuel consumption with capability to operate under a very lean limit, where the lowest lean limit of CNG is approaching 0.02 equivalence ratio way below the limit of gasoline direct injection engine. With 0.3[45] equivalence ratio, the engine performance of lean burn combustion shows at least twice the power produced by stoichiometric combustion but with the consequences of much higher NO_x and HC emission emitted from lean burn combustion method[46]. Compared to stoichiometric combustion, lean burn combustion shows higher values for coefficient of variation due to variation of mixture properties in each engine cycle. Controlling mixture formation is the main concern for researchers to get the stable engine operation using lean burn combustion system[47].

The application of lean burn engine is in the range of medium to heavy duty application, in which low fuel consumption and reducing emission without after treatment technology are the main concern of its development[2].

2.4 Compressed Natural Gas Direct Injection (CNG-DI) Engine

Emission reduction is one of a key task for automotive industries. Therefore, energy resources offering advantage with respects to emission savings are of high interest. CNG offers this benefit by emitting lower emission compared to liquid fuel due to its chemical properties. Additionally, CNG has potential to increase thermal efficiency of spark ignition engine due to its combustion properties such as high knock resistance and extreme stratification capabilities for lean air-fuel ratio.

For current passenger car standard applications, a power drop of approximate 10% is noticed by the use of CNG, which occurs from reduced volumetric efficiency. But this drawback can be compensated by direct injection of CNG straight into the combustion chamber. It was noticed that direct injection system has 9-35% higher IMEP compared to MPI system from the effect of improved volumetric efficiency[39, 48]. However, emission level of CNG-DI engine is higher compared to MPI system especially for NO_x and HC emission near stoichiometric condition.[49] Controlled combustion with controlling mixture preparation is hoped to solve this problem. Furthermore, regulating injection parameters such as injection timing, injection pressure and spray angle of injector to control the mixture preparation is hoped to solve a drawback in emission and it could increase the power resulted by the engine.

There are two types of mixture preparation method, homogeneous charge and stratified charge[32, 37, 50].

2.4.1 Stratified charge operation

Stratified charge is the mixture condition which some amount of fuel stratification exists before the combustion takes place. This condition is achieved by injecting the fuel relatively late during the cycle. Stratified charge operation was applied on idle-medium engine load and engine speed in gasoline direct injection engine in order to achieve maximum fuel economy.

An experimental study to improve direct injection parameters on natural gas engine performance has been done. As the outcome that the combustion and emission parameters are not significantly influenced by the modes of fuel injection such as the number of injectors, position of the injectors and the arrangements of injector regards to spark position. Furthermore, the pressure rise due to combustion for direct injection is higher compared to homogeneous/port injection[38].

As mentioned before, time availability for mixing is important on natural gas engine combustions. Degree of stratification of fuel inside the combustion chamber is an important parameter. Controllable charge stratification can make the combustion process better. Study on the possibility of controlling the charge stratification was done. CNG-DI stratified combustions systems can realize overall shorter combustion compared to homogeneous system. And the combustion efficiency can be maintained more than 0.92 in the range of equivalence ratio from 0.1 to 0.9[17].

The basic characteristics of CNG direct injection have been investigated. Stratified combustion strategy was investigated at rapid compression machine (RCM) that has 10 compression ratios. The result shows that compared to homogeneous system, stratified combustion has higher heat release rate for wide range of ϕ. Furthermore, emission result shows that for CO and CO_2 emission, stratified charge has better performance. In contrary, hydrocarbon emission of natural gas is higher in respect with gasoline. Higher heat release rate and peak pressure lead to high NO_x emission. However, the NO_x emission result is the same as gasoline. The combustion duration of CNG is faster for initial combustion (0-10%) in the combustion process, but slower at main stager of combustion (10-90%). Equivalence ratio has less effect on combustion duration of CNG[51]. Injection timing has significant effect on combustion of CNG, which can be seen on the variation of heat release rate pattern and emission obtained from CNG combustion. Moreover, the possibility of CNG-DI on spark ignition engine has been researched[17]. Wide operational range of equivalence ratio and high-energy conversion efficiency are the basic thought for CNG-DI implementation on spark ignition engine. The combustion duration of CNG with stratified system is shorter than CNG-homogeneous by significantly decreasing time interval between injection timing and ignition timing. Pressure rise due to combustion is insensitive to the injection timing parameter[52].

Combustion stability of CNG-DI engine is high with less cycle-by-cycle variation at the maximum pressure rise. The of CNG-DI is approximately 2% for wide range equivalence ratio. Furthermore, the combustion stability of CNG-DI is less sensitive to equivalence ratio and the variation of NO_x shows interdependence with variation of rapid combustion duration. For CO and hydrocarbon, variation of late combustion duration is related to the variation CO and HC emission[53].

2.4.2 Homogeneous charge operation

Homogeneous charge is the control strategy in which the mixture of fuel and air had achieved homogeneous mixture before the ignition event can be realize with the use of early injection timing on direct injection system. The reason of this is with the use of early injection timing, significantly longer time for mixture preparation is available.

Homogeneous systems on gasoline direct injection offers good exhaust emission results thanks to stoichiometric after treatment systems. Combines with direct injection, homogeneous systems offers the possibility to reduce fuel consumption that has been addressed as the disadvantage of the system compared to stratified combustion[6].

CNG direct injection homogeneous systems was investigated by Zuohua Huang .et al. (2001). The experiment was done on rapid compression machine (RCM). It was stated that combustion efficiency of CNG direct injection operated at homogeneous charge is higher than that of stratified systems at stoichiometric condition. While at lean operation, stratified charge stratification has higher combustion efficiency[17].

CNG direct injection engine operated at homogeneous charge had lower methane emission, but it had higher nitric oxide emission than that of stratified charge stratification. These emission trends may be due to a higher combustion efficiency of homogeneous charge compare to stratified systems[38, 45].

There are few experiments on homogeneous charge direct injection CNG engine because of the higher fuel consumption limits compared to stratified charge[17, 45]. But the emission of homogeneous charge systems which is lower compared to stratified systems especially using catalytic converter made homogeneous systems favorable. Longer combustion duration as the disadvantage of homogeneous combustion systems is the obstacle that needs to be solved. Furthermore, the problems with methane and NOx emission on CNG engine need to be reduced. Implementation of direct injection systems at CNG engine is beneficial for combustion enhancement process. Momentum generated by CNG injection is 3 times higher than gasoline. Higher turbulence intensity is achievable. The pressure difference between injection pressure and ambient pressure are determining parameters for momentum generation inside the combustion chamber. The higher the pressure difference the greater is the momentum generated by injection is [26]. It was observed that the concentration of NOx drops to a very low level at stoichiometric air/fuel ratio. It leads to NOx emissions on CNG-DI engine can be reduced by increase the equivalence ratio (ϕ).

Homogeneous charge operating method is designed to meet the requirement of medium-high engine loads. In order to facilitate these needs, the power output of a CNG direct injection engine has to be increased by increasing the turbulence level inside the combustion chamber since turbulence is needed to enhance the combustion speed of CNG homogeneous mixture especially to improve the mixing process mainly at low engine speed[54]

From the literature review, it can be conclude that the possibility of direct injection systems to be applied on CNG engine could increase the performance of the engine. In order to get more power on the engine, one has to optimize the parameter that controls the injection events in direct injection systems. Injection parameters such as injection timing[41, 49, 52, 53, 55-58], injection pressure [32, 37, 41, 50, 59]and type of injectors[32, 59, 60] had proven to be important parameters that affect the combustion and performance of direct injection engine. Therefore, the research will be focused on optimizing the injection parameters in order to improve the power output of CNG direct injection engines.

CHAPTER III

THEORETICAL BACKGROUND

3.1 Thermodynamics principles

This chapter provides criteria by which to estimate the performance of internal combustion engines. Most important are thermodynamics cycles based on ideal gas undergoing ideal process. However, internal combustion engine follows mechanical cycle. The start and end points are mechanically similar in the cycle for internal combustion engine, whether it is two-stroke or four-stroke mechanical cycle.

It is very convenient to compare internal combustion engine with ideal air standard cycle as a simple basis for comparison. Air standard cycles have limitations as air and fuel mixtures do not behave as an ideal gas. Despite this, the simple air standard cycles are very useful, as they indicate trend.

Commonly, there are 3 types of air standard cycles which are used to represent internal combustion system. These are the ideal air standard Otto cycle for spark ignition engine, the ideal air standard Diesel cycle for compression ignition engine, and air standard dual cycles that combine Otto and Diesel cycle. In this chapter we only discussed the ideal air standard Otto cycle as representative of the experimental engine.

3.1.1 The ideal air standard Otto cycle.

The Otto cycle is usually used as basis for spark ignition engine and high-speed compression ignition engines. The cycle consists of four non-flow processes. The compression and expansion process are assumed to be reversible adiabatic, and thus isentropic.

The processes are as follows (Figure 3-1):

1-2 Isentropic compression of air through a volume ratio V_1/V_2, the compression ratio r_v

2-3 Addition of heat Q_{23} at constant volume

3-4 Isentropic expansion of air to the original volume

4-1 Rejection of heat Q_{41} at constant volume to complete the cycle

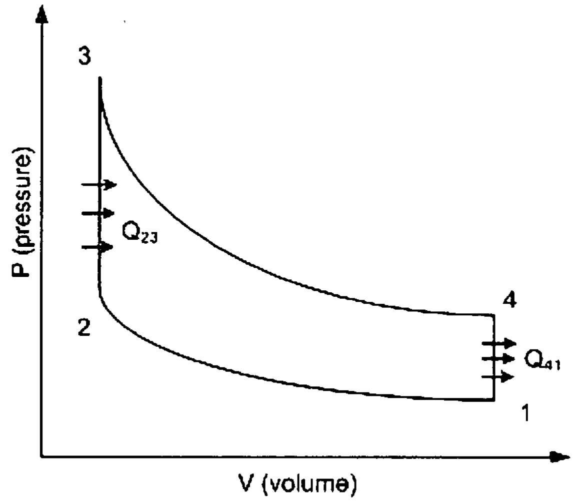


Figure 3-1 Pressure-Volume diagram for air standard Otto cycle.

3.1.2 Mechanical cycles

Internal combustion engine operates in mechanical cycle, not a thermodynamic cycle. Nevertheless, it is convenient to analyze engine behavior using hypothetic cycles from thermodynamic cycle. The mechanical cycle can be represented by pressure-volume diagram in Figure 3-2.

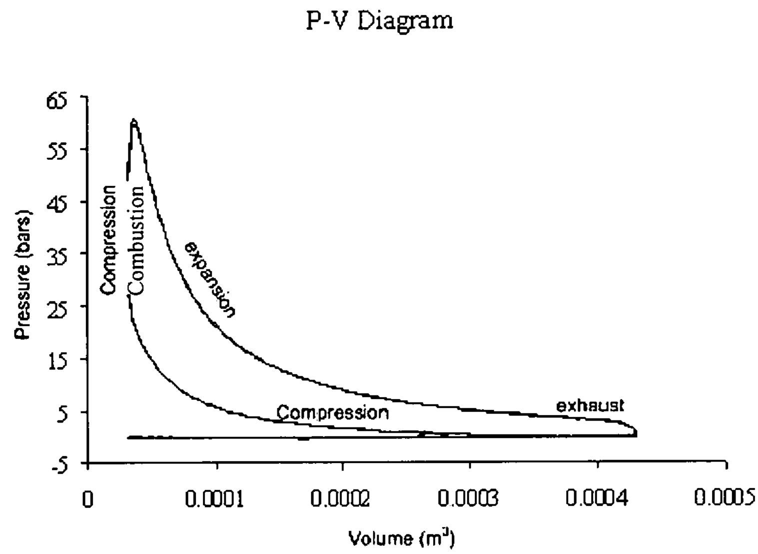


Figure 3-2 Mechanical cycles of internal combustion engine

Figure 3-2 was the pressure profile of researched engine. It shows that the combustion process occurs neither at constant volume nor constant pressure. The dependency to the piston movement and the mechanical cycle such as valve opening/closing, ignition, and injection (direct injection) events are determining parameters for the performance of the engine.

3.1.3 Comparison between thermodynamic and mechanical cycle

For simplicity, the mechanical cycle can be idealized as constant volume combustion. These idealized conditions are widely used as a basis for simplest computer model. In comparison, Otto cycle assumes that:

1. air behaves as a perfect gas with constant specific heat capacity and the process are reversible
2. there is no induction or exhaust process, but a fixed quantity of air and no leakage
3. heat addition is from external sources, which is different with internal combustion engine
4. heat rejection to the environment to complete the cycle, as opposed to the exhaust stroke and blow down.

With these assumption that has been made in the Otto cycle, there are some difference expected between theoretical and actual results as represented in Figure 3-3 includes (1) the maximum pressure is considerably less than the theoretical ones, (2) there is entropy degradation during expansion process primarily due to heat transfer process to coolant. And (3) "lost work" which is the area different between theoretical and actual cycle as shown in Figure 3-3. The "lost work' could mainly be attributed to the following, (1) heat loss, (2) mass loss, (3) finite burn rate and (4) finite blow-down rate. These losses result in the actual efficiency being less than that of the equivalent ratio of fuel-air cycle by a factor ranging from 0.8 – 0.9.

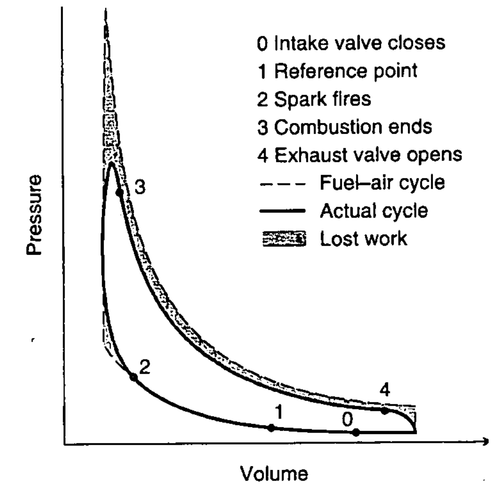


Figure 3-3 A comparison of an actual cycle with its equivalent fuel-air cycle
Source: Ferguson. C. R., Kirkpatrick. A. T., Internal combustion engine applied thermosciences pg.96

3.2 Basic calculation of performance parameters for internal combustion engines.

The indicators of internal combustion engine performance are important information for user to determine the engine performance level. Torque, power, brake mean effective pressure (BMEP), and brake specific fuel consumption (BSFC) are the parameters that have been used to represent the performance of engines. Further discussion on each parameter will be presents next.

(a) Torque (Nm)

The ability of the engine to do work is presented as torque. Dynamometer has been used to measure torque output of the engine tested. The torque reading from dynamometer already included the overall efficiency of the engine. The forces applied to the crank-shaft by the combustion without any mechanical loss is recognized as indicated torque. For the simulation process, indicated torque can be calculated from the cylinder pressure input.

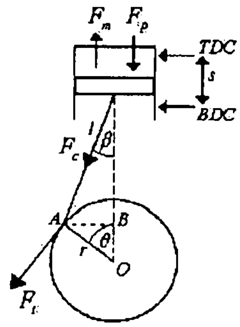


Figure 3-4 Forces acting in the engine cycles

Based on Figure 3-4, Torque is the forces acting to the crank shaft (F_t) times radius of Crank shaft. In equation form:

$$T_i = r \bullet F_t \qquad 3\text{-}1$$

where F_t is the component of the pressure force which is orthogonal to the crank shaft. By trigonometric computation, F_t can be derived from pressure by:

$$F_t = p(\theta)A_p \frac{\sin(\theta+\beta)}{\cos\beta} \qquad 3\text{-}2$$

The mechanical efficiency of the systems used in this experiment can be calculated by the difference of the measured and indicated torque by:

$$\eta_m = \frac{T_i - T_m}{T_i} \qquad 3\text{-}3$$

where : T_i = Indicated torque, T_m = measured torque

(b) Power (Kilo-Watt)

Power is defined as work rate done by the engine that can be derived from torque by using equation:

$$P = \frac{2\pi NT}{60} \qquad 3\text{-}4$$

where : P = power output (W), N = Engine speed (RPM) and T = Engine Torque (Nm)

(c) Brake Mean Effective Pressure (BMEP)

BMEP is a measure work output from an engine, and not of the pressure of an engine even-though Pascal arises as BMEP unit. BMEP is also a derivation from engine torque by equation :

$$p_b = \frac{2\pi.2T}{V_s} \qquad 3\text{-}5$$

where: T = engine torque (Nm), V_s = cylinder Swept Volume (m^3)

(d) Brake Specific Fuel Consumption (BSFC)

Fuel consumption has become an important parameter with the current situation of decreasing energy source. The main target of new engine technology is mainly based on fuel consumption and emission. Fuel consumption can be evaluated as BSFC, where BSFC is the fuel consumption of an engine that is used to produce one power output value. Derive from BSFC definition, the relationship between fuel consumption and power are:

$$BSFC = \frac{FC \cdot 3600}{P} \qquad 3\text{-}6$$

where: FC = fuel consumption (g/s), P = Engine power output (kWh)

(e) Air-Fuel Ratio (AFR)

AFR is a comparison of fuel and air entering the combustion chamber. The value of air fuel ratio is an important indicator especially to control the combustion in the engine. AFR value are dependant on types of fuels that is used. Each fuel has its own specific air-fuel ratio for complete combustion. Chemically balanced air fuel ratio is called stoichiometric AFR. Stoichiometric AFR can be calculated using chemical equation, for example, the balance equation for CH_4 with air is:

$$CH_4 + 2(O_2 + 3.76N_2) \Leftrightarrow CO_2 + 2H_2O + 2(3.76N_2) \qquad 3\text{-}7$$

Hence,

$$AFR_{stoich} = {2 * (1 + 3.76) * 28.9} \Big/ {(1 * 12) + (4 * 1)} = 17.19 \qquad 3\text{-}8$$

(f) Volumetric Efficiency

The amount of air that enters the combustion chamber is depends on the efficiency of the whole intake systems. In order to get inside combustion chamber, air has to go through some parts. Air filters, Intake runner, intake port and intake valve are the main parts that build complete intake system. Total loss comes from intake systems can be observed by calculate volumetric efficiency of the engine. Volumetric efficiency defined as the comparison of actual air mass that enters the chamber to the total displacement volume of specific engine.

$$\eta_v = \frac{2(\dot{m}_a + \dot{m}_f)}{\rho_o V_d N} \qquad 3\text{-}9$$

where: η_V = Volumetric Efficiency, $\dot{m}_a$ = Actual air mass that enters the combustion chamber (g), $\dot{m}_f$ = Actual fuel mass (g), V_d = Displacement volume (m^3), ρ_o = Air intake density (kg/m^3), N = Engine speed (RPM)

In the equation, $\dot{m}_f$ is the mass of fuel inducted. For a direct injection system with injection after intake valve closed (IVC), $\dot{m}_f = 0$. Volumetric efficiency is one of the important parameters to the engine performance in which high volumetric efficiency value of engine is expected. There are alternative that has been proposed to increase the volumetric efficiency of the engine. Turbo/Super charger and Variable valve timing (VVT) are the most popular CNG engine technologies that have been implemented to new vehicle systems.

(g) Injector flow rate

Injector flow rate is estimated by the estimation of fuel velocity at the exit of the injector orifice based on pressure difference between the fuel supplied pressure and the ambient pressure in the combustion chamber. The equation to estimate the velocities as follows[56]:

$$V_{CNG} = \sqrt{2\frac{k-1}{k}\frac{P_{CNG}}{\rho_{CNG}}\left(1-\frac{P_{ambient}}{P_{CNG}}\right)^{(k-1)/k}} \qquad 3\text{-}10$$

where: V_{CNG} = Injection velocity of CNG (m/s), P_{CNG} = Injection pressure of CNG fuel (Pa), ρ_{CNG} = CNG density (kg/m^3), $P_{ambient}$ = Atmospheric pressure (Pa), k = Polytropic index (k = 1.33)

3.3 Combustion Analysis for Internal Combustion Engine

Combustion as the heart and soul of internal combustion engine is another important process to be examined for the engine performance. A good combustion process by definition is controlled and complete combustion process in the IC engine. Controlled combustion process in Spark-Ignition engine is when the combustion is initiated by a spark, while uncontrolled combustion occurs when combustion is initiated by a hot spot. Initiation of combustion by spark is the beginning of a normal combustion process. There are several factors that can affect the combustion in the spark ignition engine. Examples are fuel composition, certain engine design, operating parameters and combustion chamber deposits.

Figure 3-5 represents the combustion process in spark ignition engine which can be divided into 4 distinct phases; (1) spark ignition ; (2) early flame development ; (3) flame propagation ; (4) flame termination.

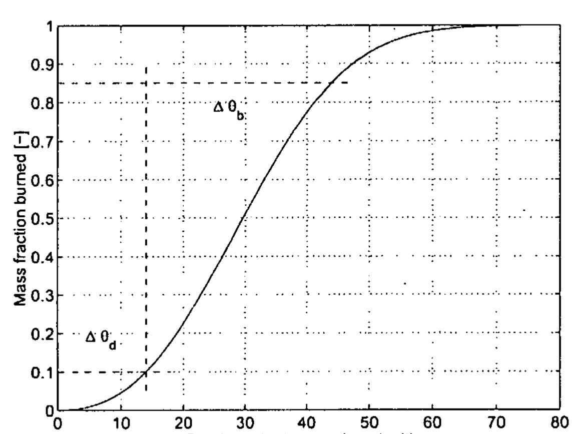


Figure 3-5 Combustion stages in Spark Ignition Engine

The first stage of combustion is corresponding to the initiations of combustion by spark, θ_{ignd}. This stage is dependant on the properties of the mixture. The combustion process needs an amount of energy to be initiated which is called activation energy. The energy requirement can be delivered by the combustion chamber condition such as pressure, volume and temperature. In the spark ignition engine this required energy also has to be delivered from spark plug. The spark-plug initiate combustion by applying large amount of energy to the mixture in a short time. After the spark initiation, the combustion is not always instantly started. The favorable condition that fulfill the activation energy has not yet been reached. The time needed from the start of ignition to the start of combustion is called the ignition delay.

The second stage of combustion is corresponding to the early flame development. Early flame development is mainly due to a chemical process which depends on the nature of the fuel on the temperature and pressure of air/fuel mixture. This stage also depends on the vicinity of the spark plug. Afterwards, the flame starts to propagate. Flame propagation mostly depends on the fuel availability at instance. Higher fuel availability around the spark can increase the combustion rate at initial stage. At this rate, fuel delivery is important. The exact control in order to get rich condition around the spark before ignition is needed. Injection parameters, combustion chamber shapes, and in-cylinder flow characteristics are the parameters that have been controlled to get expected behavior of initial combustion. The initial combustion duration is defined as the beginning 0-10% of total combustion duration.

The third stage after initial combustion is rapid combustion. Rapid combustion is defined as the following 0-90% of total combustion duration. This stage is dependant on the degree of turbulence in in-cylinder flow, prevailing temperatures, pressures and also fuel availability. By improving the combustion rate at this stage, higher pressure and better combustion will be achieved, with the side effect of nitric-oxide emission in the combustion product increasing with the increase of pressure and temperature. Hence, optimum value of this performance and emission parameters have to be discovered.

Final stage or flame termination basically is the process that will consume the fuel that is still unburned. At this stage, fuel availability is important. This stage will be limited by the time of exhaust valve open. The fuel that is still left unburned on the final stage will go through with the combustion product to the exhaust. End of combustion occur at the final 90-100% of total combustion duration.

Combustion rate is limited by the time available during the process. Total time available in crank angle degree depends on ignition timing, exhaust valve opening, and engine speed. In a well designed engine, the end of stage 2 combustion process is close to the point of maximum cylinder pressure. In practice, for optimum performance and efficiency the maximum pressure should occur between 5-20° ATDC.

Definition of each step of combustion process can be analyzed using pressure reading from the engine data. Detail explanation on determination of combustion process can be seen in the next discussion.

3.3.1 Cylinder Pressure

Good understanding of the thermodynamic process in internal combustion engine depends on the accuracies of experimental work. Pressure and temperature measurement are the common tools to analyze combustion process that can be obtained by experimental process. Other observation such as flame propagation, in-cylinder flow behavior, and mixture distribution needs complicated measurement systems and devices. The most common equipment to measure pressure is the pressure transducer since this enables us to study the pressure changes throughout the cycle.

Observation and detail analysis of pressure reading from pressure transducer can be used to determine heat release rate, indicated energy by the combustion process (IMEP), burn rate and combustion efficiency. In-cylinder pressure (kPa or bars) usually presented against crank angle (degree) or cylinder volume (m^3). Pressure analysis has been briefly explained in the thermodynamics principal section. Pressure is build during the compression stroke, and will increase drastically as a result of combustion process. It then drops as soon as the expansion stroke and exhaust valve open takes place.

The area inside the cycle is considered as IMEP, and the pressure difference between motoring cycle and working cycle is defined as the pressure build by the combustion process to calculate heat release. Further discussion on analysis of cylinder pressure will be presented next.

3.3.2 Mean effective pressure (MEP)

The work transferred from the gas to the piston can be calculated using cylinder pressure data. Corresponding to cylinder pressure versus instantaneous volume (p-V) diagram, indicated work can be calculated (Figure 3-6). This indicated work is obtained by integrating the curve area enclosed on p-V diagram

$$W_c = \oint p dV \qquad 3\text{-}11$$

in four-stroke engine, the indicated work have two definitions,

- gross indicated work per cycle, W_c, work delivered to the piston by compression and expansion stroke at the power cycle
- net indicated work per cycle, $W_{c,n}$, is the work delivered to the piston over entire cycle.

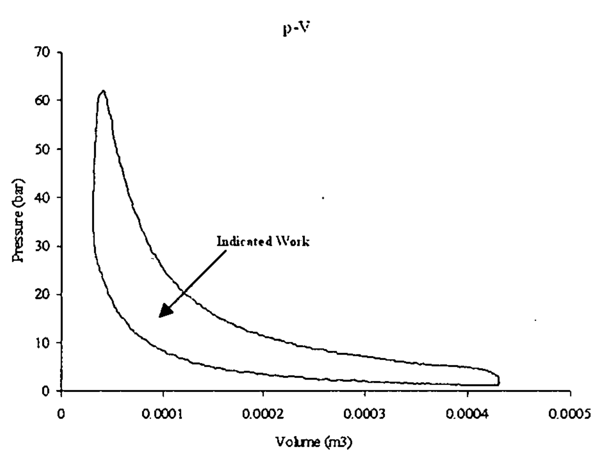


Figure 3-6 Indicated Mean Effective Pressure

The terms of indicated work often refer to the gross indicated work because in the engine analysis, total work at the power cycle is the indicator for engine performance. This indicated work per cycle over displacement volume is also called indicated mean effective pressure. The variation of IMEP over speed range is related to the engine volumetric efficiency. At high speed it will gradually decrease, especially with part load condition that have higher slope compared to full load. Another important measurement from cylinder pressure data is COV (coefficient of variation) of IMEP. This parameters indicate the cyclic variability of the combustion. It is defined as the standard deviation of IMEP divided by mean IMEP,

$$IMEP_{avg} = \frac{1}{n}\sum_{i}^{n} IMEP_i \qquad \text{3-12}$$

Standard deviation of IMEP defined as:

$$\sigma_{IMEP} = \sqrt{\frac{\sum_{i=1}^{n}\left(IMEP_{avg} - IMEP_i\right)^2}{n-1}} \qquad \text{3-13}$$

So, coefficient of variation of IMEP is:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP_{avg}} \qquad \text{3-14}$$

where, n is the number of samples.

COV also indicate the stability of the engine. In order to get a good drivability, COV value should not exceed 10%[Heywood]. There are three factors that influence COV:

- The variation in gas motion in the cylinder during the combustion, cycle-by-cycle
- The variation in the amounts of fuel, air, and residual gas to a given cylinder each cycle
- Variation in mixture composition within the cylinder at each cycle-especially near the spark plug-due to variations of mixing.

Direct injection systems have higher IMEP compared to manifold systems due to its higher engine volumetric efficiency. And also, IMEP value in this system also affected by injection timing parameters. Early injection timing has lower IMEP and lower COV (coefficient of variation) of IMEP because of the slower burn rate and higher stability of the mixture. Contrary, late injection timing have higher IMEP and higher COV. The fast burn rate and in-stability of the mixture are the main reason.

3.3.3 Heat release analysis

In Otto cycle the combustion is assumed to occur at the constant volume condition, respectively. Actual cycles of spark ignition engine do not match with these simple models, and more realistic model is required. The prevailing combustion model is the vibe function [vibe,1970], which is often spelled as Weibe function. Weibe function represents typical heat release and or "burn fraction" curve for spark ignition engine Figure 3-4. The burn fraction figure plots the cumulative heat release fraction versus crank angle. The characteristics features of the heat release curve are basically same as the combustion stage, the first slope is the spark ignition to the start of combustion (in the graph it will be represent as a zero value line). The second part is the initial flame propagation, indicated by a small slope region from the start of combustion to 10% of the combustion duration. It will be followed by rapid combustion, which have a higher slope gradient and the last stage being the flame termination region. These stages will be seen as an S-shape curve which can be represented by equation

$$x_b(\theta) = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^{n+1}\right] \qquad \text{3-15}$$

where

θ	= Crank angle at instance (degree)
θ_s	= Start of the combustion (degree)
θ_d	= Combustion duration (degree)
$n+1$	= Weibe form factor
a	= Weibe efficiency factor

further, the burn rate is given by its differential form

$$\frac{dx_b(\theta)}{d\theta} = \frac{a(n+1)}{\theta_d}\left(\frac{\theta - \theta_s}{\theta_d}\right)^n \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^{n+1}\right] \qquad \text{3-16}$$

The parameters a and n are adjustable according to the best fit to experimental data. The start of combustion is at $\theta = \theta_s$, with $x_b(\theta_s) = 1 - \exp(0) = 0$. The completeness of combustion in Weibe function is consider at 90%-99% of mass fraction burned, because Weibe function shows asymptotically approach 1. The value of 'a' and 'n' as the parametric value with $a = 5$ and $n = 2$ is well fitted with the experimental data. However, these values also depends on engine parameters such engine speed, engine load and particular type of engine[61].

Ignition delay parameter is not the same as start of combustion angle. On Rassweiler and Withrow, start of combustion defined as the pressure rise due to combustion $p_c \geq 0$. The ignition delay depends on the strength of mixture around the spark at ignition timing. So, ignition delay is the crank angle duration from the ignition point to the start of combustion. Applying the duration of ignition delay to Weibe function as:

$$x_b(\theta) = 1 - \exp\left[-a\left(\frac{\theta - \theta_s - \theta_{ignd}}{\theta_d}\right)^{n+1}\right] \quad \text{3-17}$$

where, θ_{ignd} = ignition delay duration (degree)

In the spark ignition engine, ignition timing is a known parameters. Applying this equation to find the best fit to the experimental data, stages of the combustion process could be predicted more accurately.

The differentiated Weibe function is used to produce mass fraction burned or a normalized heat release rate. The absolute value of the heat release rate, dQ_{ch}, is given by the chemical energy of fuel with the fuel mass m_f, specific heating value of fuel q_{HV}, and combustion efficiency η_c as:

$$\frac{dQ_{ch}}{d\theta} = m_f q_{HV} \eta_c \frac{dx_b}{d\theta} \quad \text{3-18}$$

where $m_f q_{HV} \eta_c$ is the total energy released by the combustion process.

The importance of accurate determination of each combustion stage can be seen in the weibe function equation. False prediction of burn duration, start of combustion and ignition delay can affect the result of heat release and mass fraction calculation.

There are seven methods commonly used for ignition delay and combustion parameters determination. From the result, it is stated that every method has its own limitation. For this experiment the author has select the "dP" and "d ln P" method to determine the combustion parameters[62].

The "dP" method determine the combustion parameters by calculating the difference between motoring cycle and firing cycle. In spark ignition engine, spark timing is a known variable. The pressure difference will start either immediately ignition point or after some delay. The start of combustion (SOC) will be recognized as the difference between motoring pressure and combustion pressure more than 1 bars.

The "d ln P" method uses the algorithm derivative of the in-cylinder pressure. Combustion limits correspond to the two local minimum points of the d ln P versus crank angle curve.

3.3.4 Combustion Efficiency

Combustion efficiency is the difference between available energy (fuel) and the final value of heat released by combustion. Defined as

$$\eta_c = \frac{Q_{out}}{Q_{in}} = \frac{\sum Q_{ch}}{m_f q_{HV}} \qquad \text{3-19}$$

Where $\sum Q_{ch}$ is the total heat released by the combustion process, m_f is fuel mass, and q_{HV} is the specific heating value of fuel.

3.4 Emission

3.4.1 Hydrocarbon emission

Heywood [1988] stated that there are four possible HC emission formation mechanism in spark-ignition engine; (1) flame quenching at the combustion chamber walls, leaving a layer of unburned fuel-air mixture adjacent to the wall; (2) the filling of crevice volumes with unburned mixture. When the flame is quenched at the crevice volume, mixture will escape along with the product from primary combustion process; (3) absorbtion of fuel vapor to oil layers on the cylinder wall during intake and compression and followed by desorption of fuel vapor in to the cylinder during expansion and exhaust stroke; and (4) incomplete combustion in a fraction of the engine operating cycles occurs when the combustion quality is poor due to engine transients operation, non-optimum control of EGR, AFR, and ignition timing.

Table 3-1 have shown that the main contributor of hydrocarbon emissions is crevice volume. The level of unburned hydrocarbon in the exhaust gases is generally specified by the total concentration of atom carbon expressed as part per million (ppm). In general, hydrocarbon emission is categorized into two: first is VOC (volatile organic compound) which consist of reactive hydrocarbon that have unstable bonding and tend to react with atmosphere; second is methane emissions that have very stable bonding and will not react in air. CNG, which have more than 80% methane composition, mostly faces the problems of incomplete combustion resulting from unburned methane in high percentage[63].

Table 3-1 HC sources

Source	%HC
Combustion chamber crevice	38
Single-wall flame quenching	5
Oil film layers	16
Combustion chamber deposits	16
Exhaust valve leakage	5
Liquid fuel	20

Sources: Alkidas.A.C., Combustion Chamber Crevices:the major source of engine-out hydrocarbon emission under fully warmed conditions. Progress in Energy and Combustion Science. 25. 1999. 253-273

Higher methane composition in the exhaust gas could cause a problem on three-way catalyst usage[64].

3.4.2 Nitric Oxide (NOx)

Nitric Oxide is one of the major engine emission problems. Nitric oxide mainly consist of NO (Nitrogen Monoxide) and NO2 (Nitrogen dioxide), with NO being the biggest percentage in NOx emission. The source of NOx emission is the oxidation of atmospheric nitrogen during the combustion process. However if the fuel contains significant amount of nitrogen, the oxidation of the fuel is an additional source of NO. The governing mechanism of nitric oxide formation in the combustion of near stoichiometric condition have three major reactions, which are:

$$O + N_2 \Leftrightarrow NO + N \tag{3-20}$$

$$N + O_2 \Leftrightarrow NO + O \tag{3-21}$$

$$N + OH \Leftrightarrow NO + H \tag{3-22}$$

The above reaction also called *Zeldovich* mechanism is also designated as thermal NO. These reactions significantly contribute to the formation of nitric oxide during combustion process.

The requirement of simple and dependable model to predict the nitric oxide formation using CNG fuel and spark ignition engine has been extensively researched. Nitric oxide formation model are mostly based on Zeldovich mechanism. NO formation becomes significant at temperature above 1200 °C in methane/air combustion. Nitric oxide emissions significantly increase with increasing temperature. Furthermore at high temperature combustion exceeding 1600 °C, the thermal NO formation became important because it controls the formation rate of nitric oxide. For temperature less than 1500 °C, another NO formation route has been suggested known as N_2O/NO mechanism.

$$N_2 + O + M \leftrightarrow N_2O + M \tag{3-23}$$

$$N_2O + O \leftrightarrow NO + NO \tag{3-24}$$

$$N_2O + O \leftrightarrow N_2 + O_2 \tag{3-25}$$

Also, Bozelli and Dean proposed a new route that is dominant at low temperature NO formation.

$$N_2 + H \leftrightarrow NNH \tag{3-26}$$

$$NNH + O \leftrightarrow NO + NH \tag{3-27}$$

Thermal NO mechanism, N_2O/NO and NNH mechanism are the common NO formation mechanism that is used to models and predict the nitric oxide emission. On the effect of air/fuel ratio, simplified model shows a good trends, though the value is lower than experimental result. The trend shows that as the air/fuel ratio increase the thermal NO also increase[65].

Other researcher had compared the use of 4-, 5-, and 9- step reduced mechanism and compare it to detailed 276-step mechanism models. This model is used to predict CO and NOx emission. It was found that 9-step mechanism gives a reliable instantaneous reaction rate and composition for broad region of operation. 5-step mechanism shows less accurate prediction for equivalence ratio 1, but it shows good agreement with 9-

step and detailed step models for equivalence ratio less than 0.65. It also use N_2O/NO formation mechanism to model the reduced mechanism of NOx formation[66].

Furthermore, thermal NO and N_2O/NO mechanism is used to predict nitric oxide emission in spark ignition engine. Combine with multi-zone combustion models, Raine et al. found that multi-zones prediction shows good agreement for NOx concentration at 4000 ppm. At the higher level of NOx, the model shows over estimation. At the lower level of NOx, it predict slightly lower than the experimental result. Single-zone prediction shows higher level of NOx compare to multi-zone model[67].

From all experimental result and models, it was found that temperature and equivalence ratio is the most influential parameter in the formation of nitric oxide. Other variables that also determine the NOx emission is burned gas fraction of the in-cylinder unburned mixture, and spark timing.[61]

In order to control nitric oxide emission, there are two common way. First is Exhaust gas recirculation (EGR), this method circulates the exhaust gas back into the combustion chamber. EGR will reduce flame temperature and flame speed by diluting exhaust gas into the chamber. Second is the catalytic converter, it will capture nitric oxide molecule in the exhaust gas by its reactive metal that react with nitric oxide.

3.4.3 Carbon Monoxide (CO)

Carbon monoxide is formed during the combustion process. CO emission is mainly caused by less amount of oxygen entering the systems to perform complete combustion. In addition, in high temperature products, even with lean mixtures, dissociation ensures significant CO levels are still produced. Later, the dissociation process will freeze as the burned gas temperature falls during expansion process. Rich mixture condition boosts incomplete combustion which leads resulting CO and hydrocarbon emission. Therefore, air fuel ratio is an important variable that affect emission in spark ignition engine.

CO exhaust level in the chamber is kinetically controlled. In premixed hydrocarbon-air flames, CO concentration increases rapidly in the flame zone to a maximum value, which is larger than equilibrium value for adiabatic combustion. CO formation principal reaction steps in hydrocarbon combustion mechanism, can be summarized by

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RCHO \rightarrow RCO \rightarrow CO \qquad 3\text{-}28$$

Where, R stands for radical hydrocarbon such as CH_3, CH_2, C_2H_2 etc that has higher tendency to react according to the reaction process described in reaction chain 3-20. CO can also be formed because of not enough time available during the formation of CO2, which is described in reaction 3-21.

The principal of CO oxidation in hydrocarbon-air flames is

$$CO + OH \Rightarrow CO_2 + H \qquad 3\text{-}29$$

Note that CO formed in the combustion process via this path when oxidized to CO2 at slower rate.

CHAPTER IV

ENGINE PERFORMANCE SIMULATION

In this section, numerical prediction of engine performance and emission will be discussed. GT-Power is one of the many 'state of the art' commercially available engine-modeling tools. Such codes shares common assumption for modeling a compressible flow ie. a one dimensional simplification of a flow governing equations of mass, momentum and energy conservation.

A GT-POWER model was build based on the researched engine specification. The engine was 400 cc Spark ignition, with CNG fuel. There are three major steps to simulate with GT-POWER; building the model; validation; and simulation. Each step will be discussed further in the next discussion.

Simulation stage has been used to investigate and predict the behavior of the CNG-DI engine. In this chapter, prediction of the characteristics of CNG direct injection spark-ignition engine is presented.

4.1 Building the model

Primarily, engine model have to be similar to the actual engine. Figure 4-1 represents actual engine and Figure 4-2 the GT-POWER model.

The following gives some modeling detail and assumptions included in the model:

- A simple weibe function for spark ignition engine was used to model the combustion heat release rate
- Wall temperatures for inlet manifold elements were estimated and assumed time-invariant.
- The diffuser ducting to the inlet plenum was modeled as a succession of step change sudden expansions.
- Stoichiometric AFR for methane was applied.

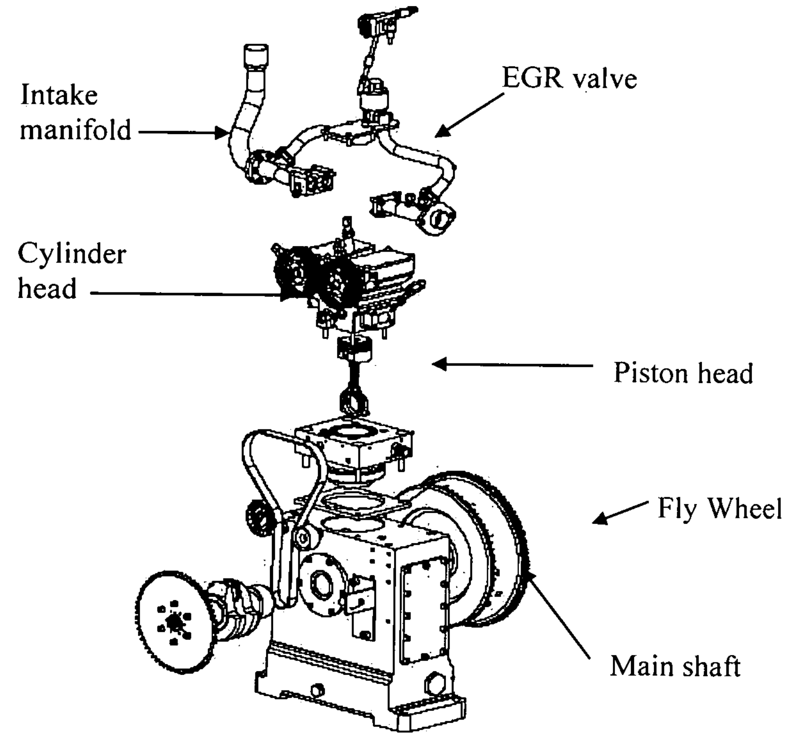


Figure 4-1. CNG direct injection spark ignition experimental engine

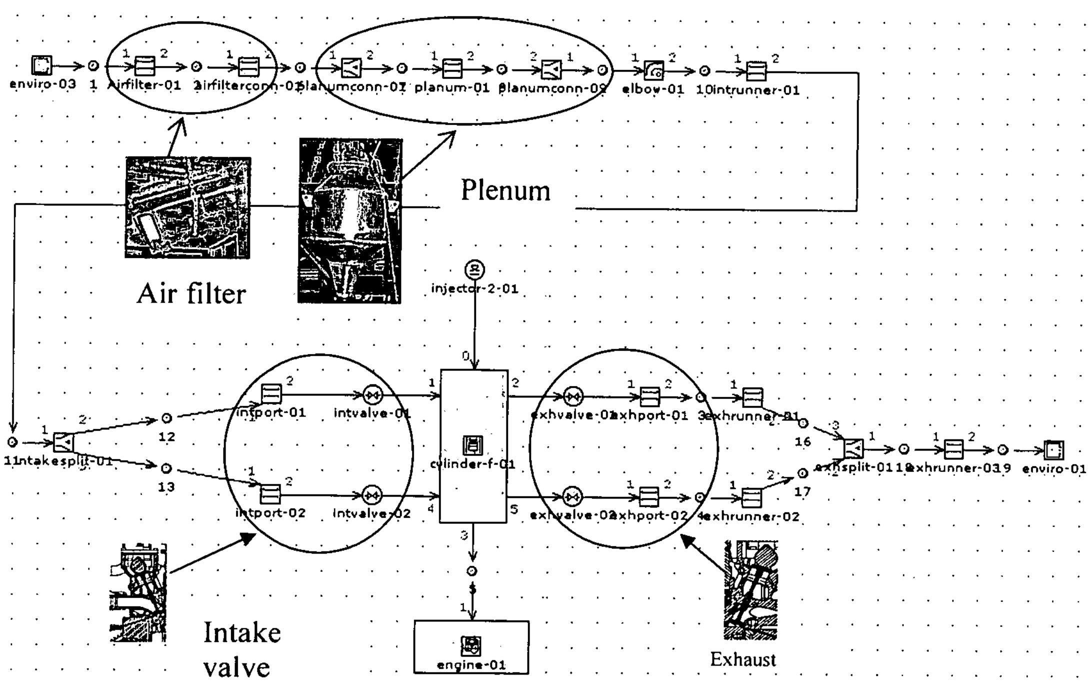


Figure 4-2 CNG-DI engine model

Continued:

- FMEP model constants were initially adjusted to give best agreement with experiment and were then fixed for the study with multiply factor was needed from the recommended values to get the very best fit to the experimental results.

- Wall temperatures for cylinder, piston and head were assumed.

Other assumption that is used in this model is the use of methane as fuel to represent CNG. The use of methane in this simulation is basically due to major composition of methane in CNG as shown in appendix A.

The engine model was built using the dimensions of real engine from air intake to exhaust. Every dimension of runner, junction and elbow followed the dimension of the experimental engine. Direct injection was modeled by direct fuel supply to engine cylinder. Injector delivery rate was calculated based on pressure difference between injector pressure and in-cylinder pressure using equation:

$$V_{CNG} = \sqrt{2\frac{k-1}{k}\frac{P_{CNG}}{\rho_{CNG}}\left(1-\frac{P_{ambient}}{P_{CNG}}\right)^{(k-1)/k}} \tag{4-1}$$

where: V_{CNG} = Injection velocity of CNG (m/s)

P_{CNG} = Injection pressure of CNG fuel (Pa)

ρ_{CNG} = CNG density (kg/m^3)

$P_{ambient}$ = Atmospheric pressure (Pa)

k = Polytropic index (k = 1.33)

Injection timing was defined in the GT-Power input, with the injection start angle as reference input. One-dimensional GT-Power calculation method has limitation on simulating injector spray angle. Hence, in this simulation, injector spray angle shall not be simulated by GT-Power.

4.2 Model Validation

To validate the GT-Power simulation, there are some processes that have to be carried out. These validation process have to be done orderly to get the same characteristics between the engine model and experimental engine. These sequences are used to validate the model engine:

- Model intake manifold pressure have to be the same with the experimental engine. This calibrates flow losses between the inlet and measurement location.
    - This step is applied at the highest speed. The losses will be in the highest value at rated speed and it will be easiest to see slight changes in the manifold pressure
    - Comparing the whole range of speeds
- Calibrating the volumetric efficiency
- The cylinder pressure profile of the model is the common way to validate the model. There are some important pressure profiles that the model must have in order to have the same values as actual engine.
    - Pressure profile during compression stroke
    - Pressure profile during combustion using heat release calculation provided by GT-Power.
- Last step of the validation process is to find the correct FMEP value from measured motoring data. This step would match the reading of the brake torque and power of the model to the actual engine.

4.3 Simulation Result

4.3.1 Injection timing

Injection timing simulation was done at 3000 RPM and full load condition. Injection timing range was from 300 °BTDC to 100°BTDC. The injection pressure for this simulation is 18 bars.

Figure 4-3 represents the pressure profile of varying injection timing parameters. It shows that early injection timing have the lowest maximum pressure (300 - 250°BTDC) which about 64 bar. While, late injection timing has higher maximum pressure value. Figure 4-3 also shown that for injection timing less than 160°BTDC the pressure profile is almost similar one another. These results are also shown by performance graph in Figure 4-4.

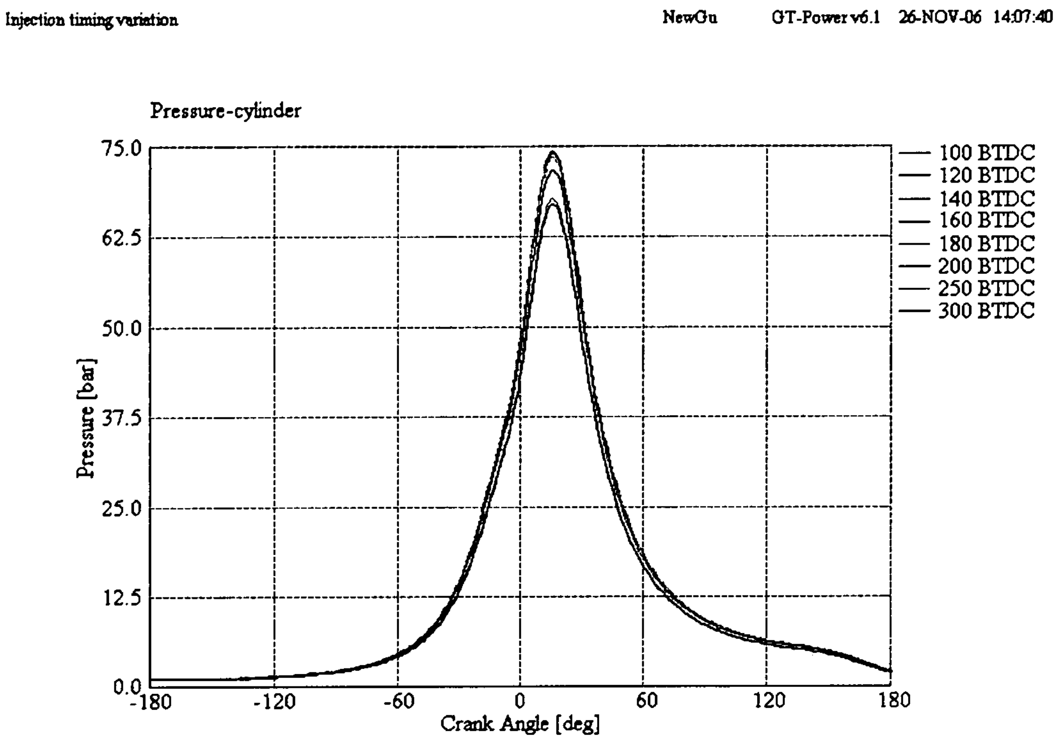
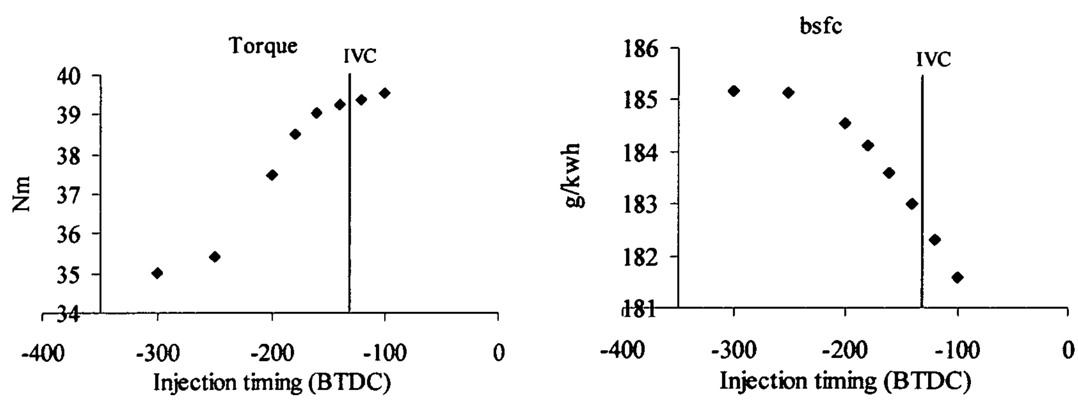
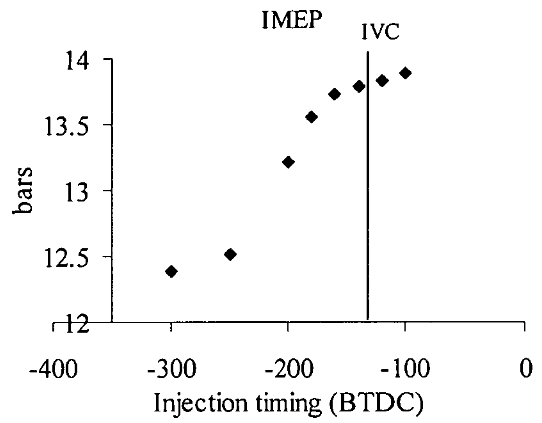


Figure 4-3 Injection timing variation (GT-Power)



(a) (b)



(c)

Figure 4-4 (a,b,c) Engine performance for various injection timing

The torque characteristic depicted in Figure 4-4 reveals performance for late injection timing is almost similar, only a small gradient of torque can be seen in the injection timing range of 100-160 °BTDC. Similar behavior is also shown by IMEP graph BSFC graph shows that 300°BTDC injection timing have the highest value, 185.6 g/kwh, and its descending as injection timing is retarded. 100°BTDC injection timing gave the lowest value of bsfc of 181.5 g/kwh. These characteristics of injection timing effect to the engine performance may be due to the injection timing affecting the volumetric efficiency of the engine. The volumetric efficiency will increase as injection timing retarded as shown in Figure 4-5.

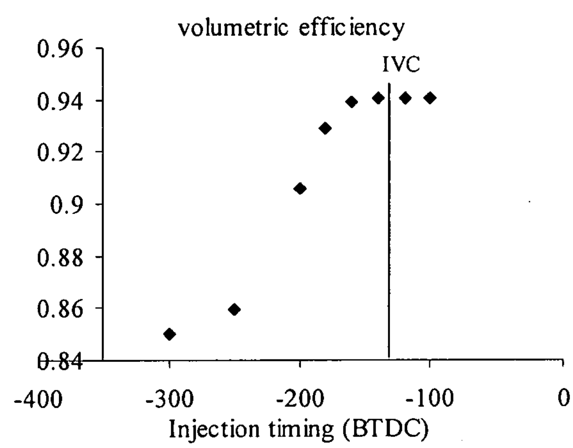


Figure 4-5 Volumetric efficiency for various injection timing.

This simulation result shows that volumetric efficiency is the main reason for the performance difference of various injection timing parameters in spark ignition direct injection engine. The relation between volumetric efficiency of the engine to the performance is almost linear according to Figure 4-6.

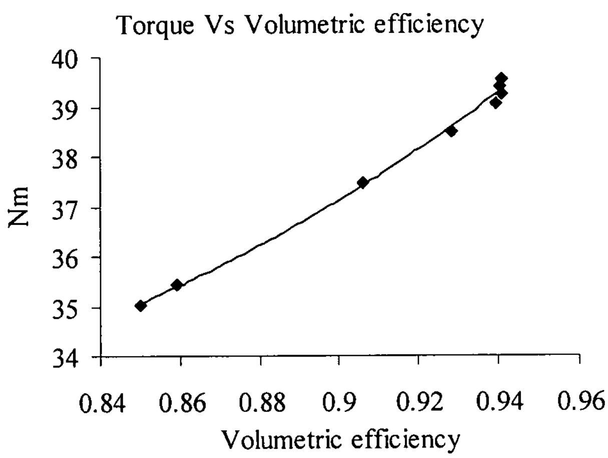


Figure 4-6 Volumetric efficiency effect to the Torque of SI-DI engine

Summary of the performance of the GT-Power model is presented in Table 4-1.

Table 4-1 Performance prediction of CNG-DI engine

CNG-DI Engine

Injection Timing parameters

Engine Performance Predictions (SI)-Performance

	100BTDC	120BTDC	140BTDC	160BTDC	180BTDC	200BTDC	250BTDC	300BTDC
Brake Power [kW]	12.4	12.4	12.3	12.3	12.1	11.8	11.1	11.0
Brake Power [HP]	16.7	16.6	16.5	16.5	16.2	15.8	14.9	14.8
Brake Torque [N-m]	39.5	39.4	39.2	39.1	38.5	37.5	35.4	35.0
IMEP [bar]	13.55	13.50	13.45	13.39	13.22	12.88	12.22	12.09
FMEP [bar]	1.10	1.10	1.10	1.10	1.10	1.09	1.07	1.06
PMEP [bar]	-0.34	-0.34	-0.34	-0.34	-0.33	-0.32	-0.30	-0.29
Air Flow Rate [kg/hr]	39.2	39.2	39.2	39.2	38.7	37.8	35.8	35.5
BSAC [g/kW-h]	3159	3172	3184	3194	3203	3210	3221	3222
Fuel Flow Rate [kg/hr]	2.3	2.3	2.3	2.3	2.2	2.2	2.1	2.0
BSFC [g/kW-h]	181.6	182.3	183.0	183.6	184.1	184.6	185.2	185.2
Volumetric Efficiency [%]	94.1	94.1	94.1	93.9	92.9	90.6	85.9	85.0
Volumetric Efficiency (M) [%]	94.1	94.1	94.1	93.9	92.9	90.6	85.9	85.0
Trapping Ratio	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
A/F Ratio	17.39	17.40	17.40	17.40	17.40	17.40	17.40	17.40
Brake Efficiency [%]	39.6	39.5	39.3	39.2	39.1	39.0	38.9	38.9

The NOx emission result for various injection timing can be seen in Figure 4-7. It shows that 160°BTDC injection timing gave the highest result of NOx emission of

17.6 g/kwh. Injection timing at 100°BTDC have the lowest NOx value of 16.6 g/kwh. The maximum NOx for 160° BTDC is due to the peak pressure at 160°BTDC, the highest among all injection timing, 74.4 bars. This peak combustion resulted in high development of NOx concentration.

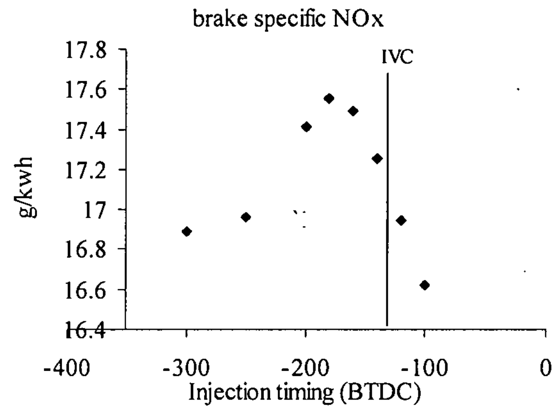


Figure 4-7 brake specific NOx for various injection timing.

As shown in Figure 4-8 Carbon monoxide emission shows the lowest maximum value for late injection timing (100°BTDC) at 0.00995 g., while early injection timing shows high CO concentration at 0.013 g. This figure also shows that for injection timing range of 300-200°BTDC, 180-160°BTDC the CO emission is almost similar. While, for late injection timing 140-100°BTDC the difference is significant.

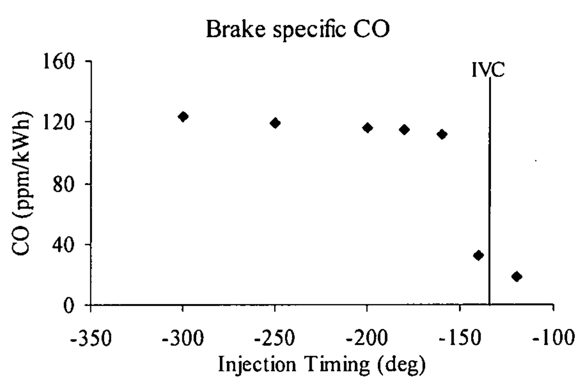


Figure 4-8 Brake specific CO emission of CNG-DI engine for various injection timing

4.3.2 Injection Pressure

Injection pressure parameters mainly affect the injection window of the injector. Higher injection pressure means there is less time needed to deliver the required fuel. Figure 4-9 illustrates the injection window for each injection pressure changes.

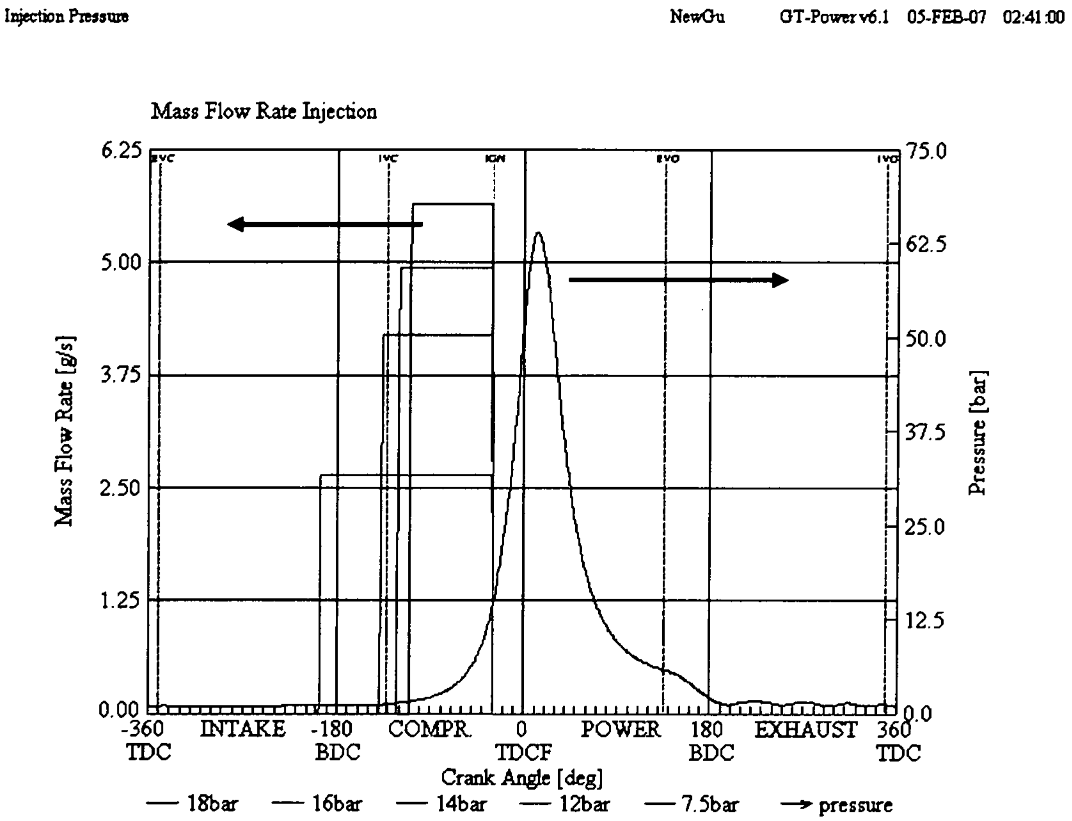


Figure 4-9 Injection window for various injection pressures.

Figure 4-9 also showed that for every injection pressure there are retarding limits for injection timing since all injection event must be completed before ignition is started to prevent backfire. The lower the injection pressure the lesser injection timing can be retarded to get homogeneous mixture. If at operational lower injection pressure operation, the start of injection is close to TDC, fuel will be injected during the ignition and it will gives unwanted effect to the engine.

Figure 4-10 elaborates the volumetric efficiency of the model for various injection pressures at 300°BTDC injection timing on 3000 RPM as a baseline where all injection events occur while intake valve is open. It shows that 16 bars injection timing gave the lowest volumetric efficiency, 83.65%, while 20 bar and 12 bar injection pressure have high volumetric efficiency values of 84%. For the 12 bars injection pressure, this behavior may due to the speed of injected fuel that create a local pressure drop near intake valve which enhance the intake air flow. While 20 bar injection pressure have high volumetric efficiency may due to the small injection window that reduce the interruption of the intake air flow enters the combustion chamber.

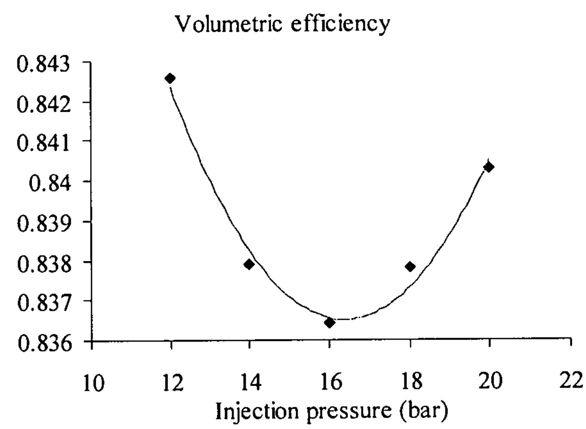


Figure 4-10 Volumetric efficiency for various injection pressures

For late injection timing when all the valve are closed, only injection pressure of 16 bar and above are possible due to their smaller injection duration. At this condition, the volumetric efficiency is only affected by the induction of air and manifold pressure more and not due to fuel induction process.

Figure 4-11 represents the pressure profile for various injection pressures with fix end of injection timing so that for higher pressure that has shorter injection duration the injection events is done after intake valve closed. It can be shown from the figure that the combustion pressure will increase as the injection pressure increases. This is due to higher volumetric efficiency and to the higher injection velocity of higher injection pressure, which adds to the kinetic energy of the mixture. While for lower injection pressure, some part of the injection event occurs during intake valve open. Hence, the results lower injection pressure will result to a lower volumetric efficiency compared to high injection pressure.

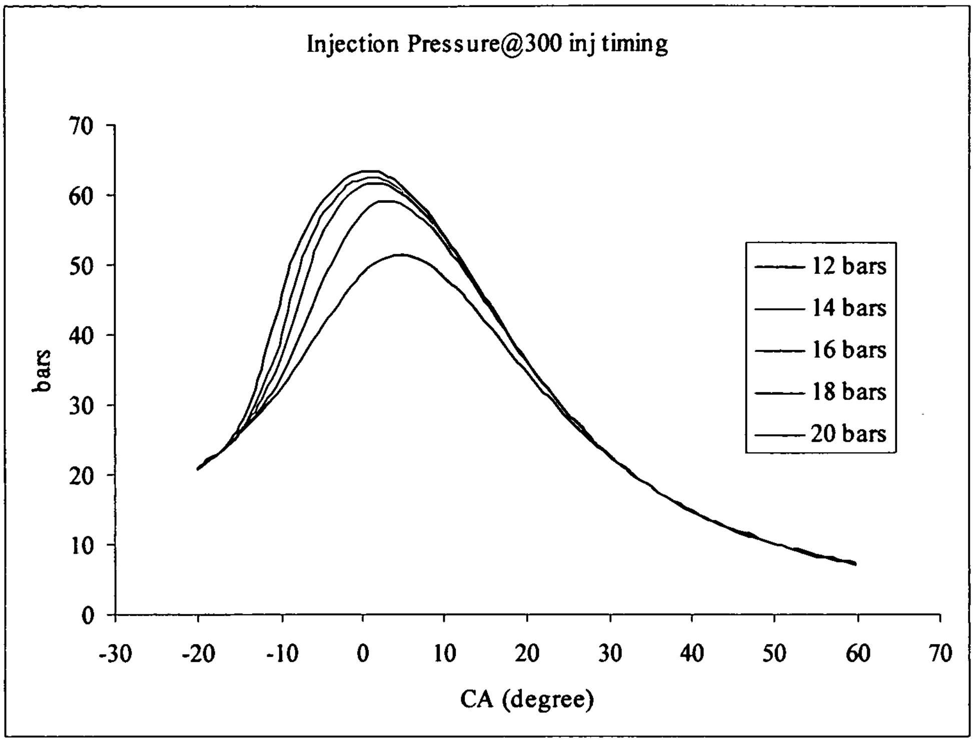


Figure 4-11 Pressure profile for injection pressure variation

Simulation stage has been done to predict the experimental results. It shows that the injection timing and injection pressure has significant effect to the engine performance of CNG-DI engine. These simulation results will be carried out as the prediction and preliminary assumptions of the experimental data. Moreover, these data will be compared with the experimental data at the results and analysis sections. In the next chapter, experimental method and equipment used in this research will be discussed.

CHAPTER V

EXPERIMENTAL WORKS

This section describes the specified equipment that has been used in this research together with its characteristics and measurement capabilities. Data gathering process along with the constraint of the parameters being researched was also included in this section as well as the equation being used to analyze the data gathered.

5.1 Engine Test-bed, Instrumentations and Engine Testing

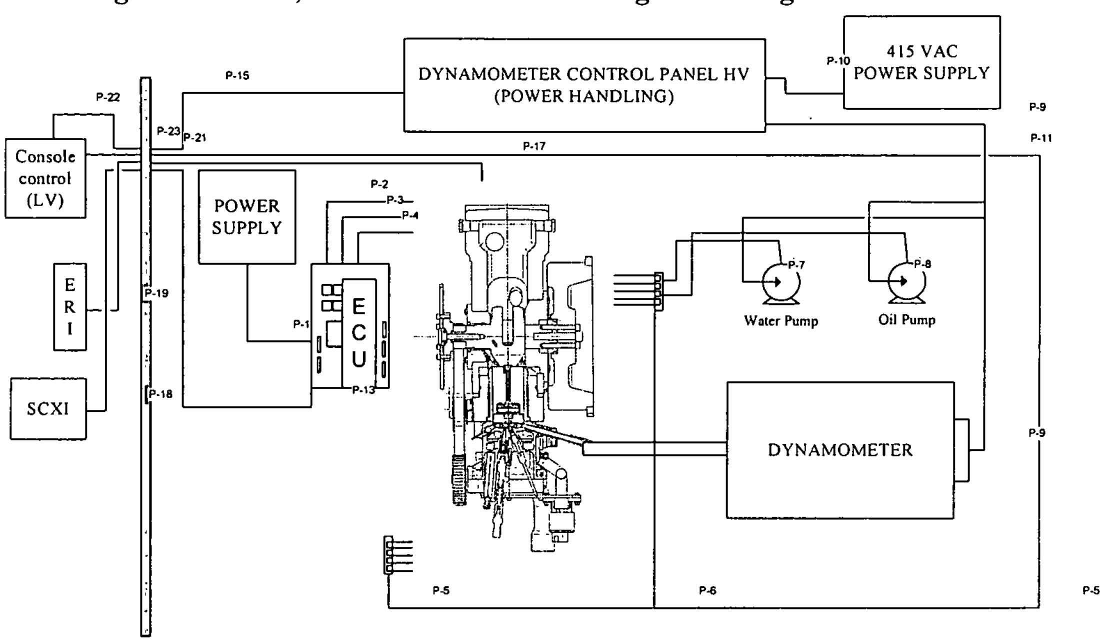


Figure 5-1 Schematic of engine test-bed

Engine schematic setup is shown in Figure 5-1. A four-stroke single cylinder research engine was used to investigate CNG direct injection engine performance. With 400 cc displacement volume, this engine has a compression ratio of 14. Engine parameters such as ignition timing, injection timing and air fuel ratio are controlled by ECU that is connected with ECU Remote Interface (ERI) installed in the personal computer (Table 5-1).

Table 5-1 Engine Specifications and Test bed

Engine Properties	
Displacement volume	399.25 cm^3
Cylinder Bore	76 mm
Cylinder Stroke	88 mm
Compression Ratio	14
Exhaust Valve Closed	ATDC 10°
Exhaust Valve Open	BBDC 45°
Inlet Valve Open	BTDC 12°
Inlet Valve Closed	ABDC 48°
Dynamometer	Eddy Current with maximum reading is 50Nm
ECU	Orbital Inc

Figure 5-2 shows the geometry of the combustion chamber with injector and spark plug positioning. In the combustion chamber, the injector is placed in the centre of the combustion chamber and spark plug is placed next to the injector. Modified spark plug is used to improve CNG combustion and it has longer tip compared to standard spark plug.

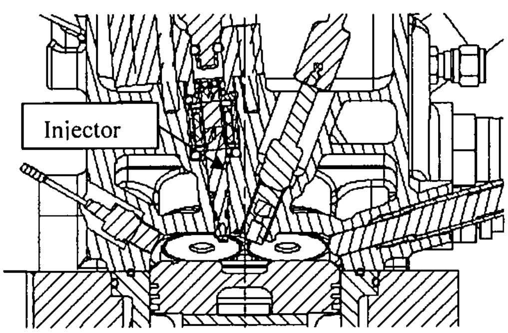


Figure 5-2 Injector and Spark position

Experimental work was done in the homogeneous condition. Homogeneous piston was used as illustrated in Figure 5-3 to achieve homogeneous condition of the mixture. This piston has a small cup placed in the centre of the piston head.

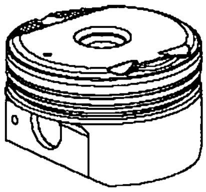


Figure 5-3 Homogeneous piston head shape

In order get the power reading generated by the engine, a dynamometer is coupled to the engine. Eddy current dynamometer is used to measure the brake torque as the engine output.

The scheme of the fueling system can be seen in Figure 5-4. The pressure regulator is used to reduce the input from CNG tanks, which are at 200 bars, down to 30 bars. Micro-motion fuel flow meter is used. It is placed after pressure regulator and it has a sensitivity of 0.001 g/s. Next, after the fuel flow meter, the inlet fuel pressure control system is placed. This system is coupled with a compressor to maintain the pressure along the fuel rail to injector to get small variability of fuel injection rate. Detail specification of fuel flow meter is shown in Table 5-2.

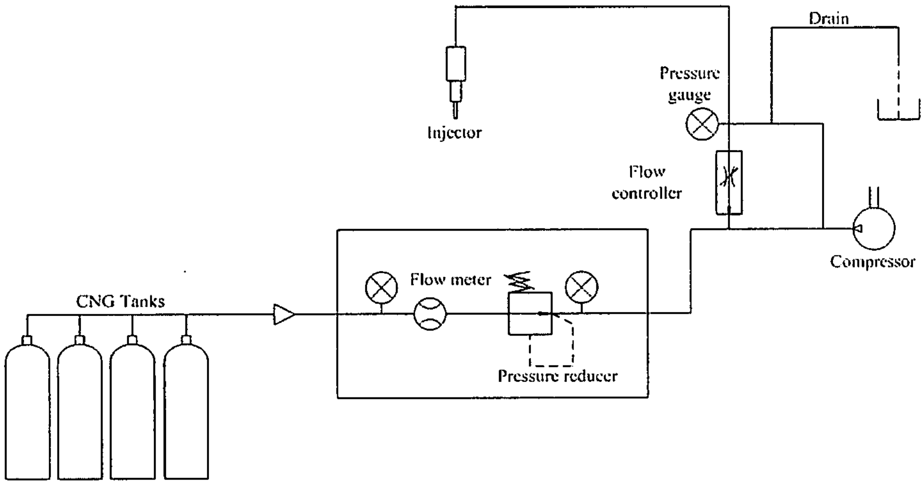


Figure 5-4 CNG fueling system

Table 5-2 Micro-motion fuel flow meter specifications

Flow accuracy:	+/-0.05% of flow rate
Gas accuracy:	+/-0.35% of flow rate
Density accuracy:	+/-0.0002 g/cc
Wetted materials:	304L, 316L Stainless Steel or Nickel Alloy
Temperature rating:	-400 to 800°F (-240 to 427°C)
Pressure rating:	1450 - 6000** psi (100 to 413** bar)

5.1.1 Pressure sensor

A number of methods have been used to measure pressure as a function of cylinder volume. We will restrict our attention to piezoelectric transducers (Figure 5-5), since it is the method that is used in this experiment.

The piezoelectric effect is the generation of an electric charge to a solid by a change in pressure. There are two primary piezoelectric effects: (1) the transversal effect in which charges on the x-planes of the crystal result from the forces acting upon the y-plane, and (2) the longitudinal effect in which charges on the x-planes of the crystal result from forces acting upon the x-plane.

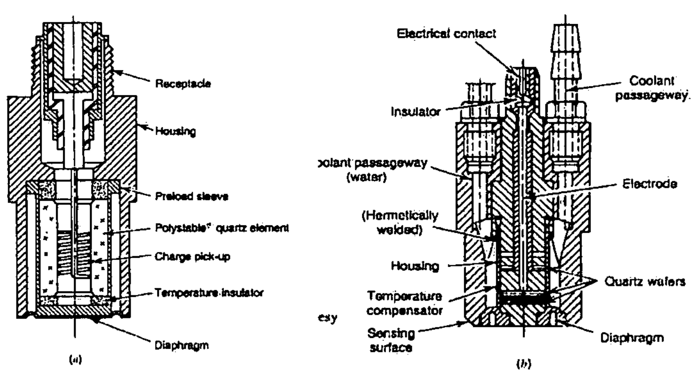


Figure 5-5 Quartz piezoelectric pressure transducers. (a) Courtesy of Kistler Instrument Corp. (b) Courtesy of AVL Corp.
Source: Ferguson. C. R., Kirkpatrick. A. T., Internal combustion engine applied thermosciences pg.120

Piezoelectric transducer can be obtained with internal cooling passage and with a temperature compensator. Note that increasing temperature will make the casing expand and uncompressed crystals from load. This cooling is an important operational procedure to get correct pressure reading.

A typical result of measuring cylinder pressure as a function of cylinder volume is shown in Figure 5-6. A crank angle encoder is used to establish the top dead center position and the phasing of cylinder pressure to crank angle.

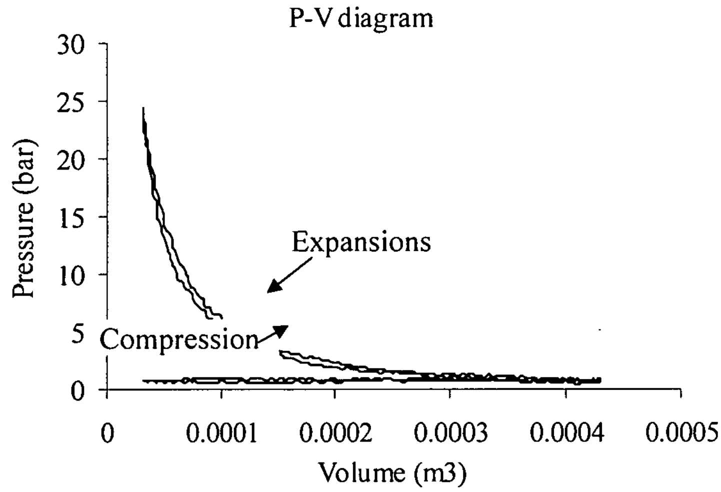


Figure 5-6 Pressure as a function of volume in the engine cylinder

5.1.2 Exhaust Gas Analyzer

A multi component analysis of exhaust gas emission, shown in Figure 5-7 (manufactured by GASMET^{TM}) was used to measure the exhaust gas. General specification of this analyzer is provided in Table 5-3.

The stationary GASMET^{TM} FTIR analyzer consists of four systems:

1. **Gasmet**^{TM} Cr-2000 gas analyzer with LN_2 detector and fast response sample cell
2. **Gasmet**^{TM} sampling system
3. Heated Filter Unit
4. **Gasmet**^{TM} Industrial Computer

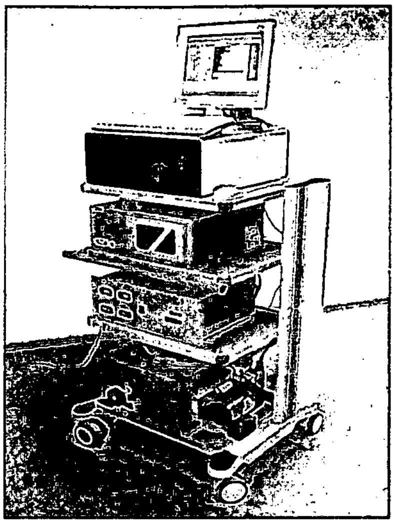


Figure 5-7 GASMET^{TM} stationary FTIR analyzer

Table 5-3 General specification of GASMET^{TM} FTIR analyzer

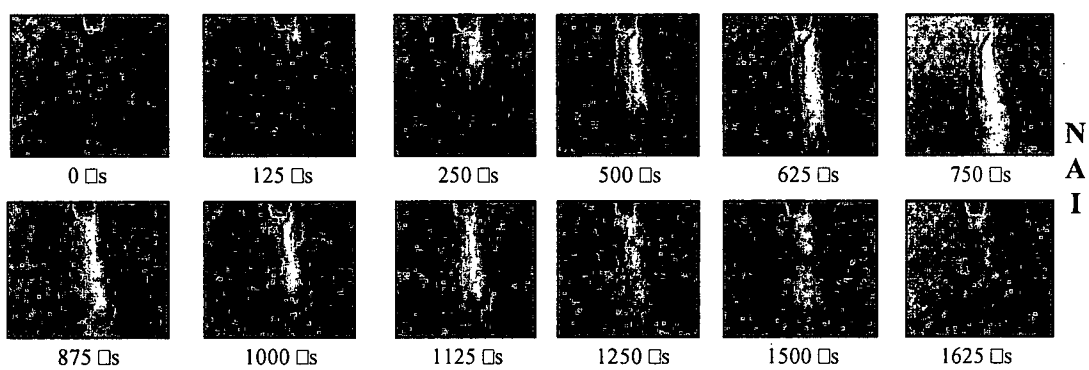
General parameters	
Measuring Principle:	Fourier Transform Infrared, FTIR
Performance:	simultaneous analysis of up to 50 gas compounds
Response time:	typically <<25s, depending on the gas flow and measurement
Operating temperature:	15-25°C non condensing
Storage temperature:	-20 - 60°C non condensing
Power supply	100-115 or 230V / 50-60 Hz
Power consumption:	300 W
Measuring parameters	
Zero point calibration:	24 hours, calibration with nitrogen
Zero point drift:	<2 % of measuring range per zero point calibration interval
Sensitivity drift:	none
Linearity deviation:	<2 % of measuring range
Temperature drifts:	<2 % of measuring range per 10 K temperature change

Pressure Influence:	1 % change of measuring value for 1 % sample pressure change. Ambient pressure changes measured and compensated

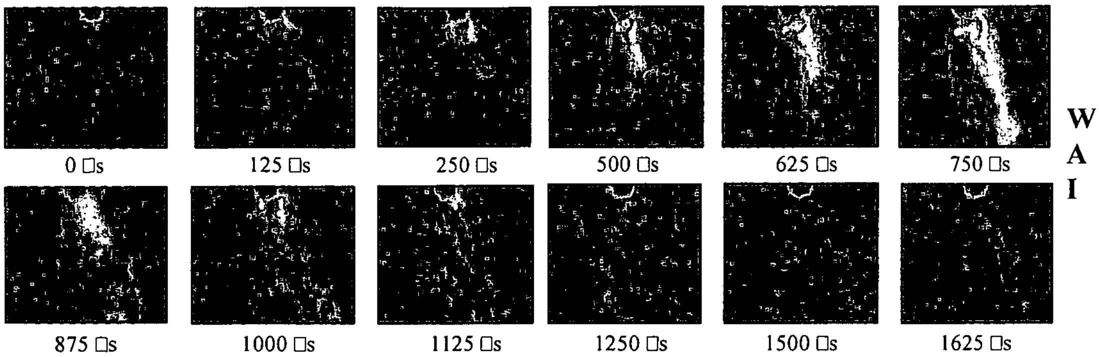
5.1.3 Injectors

Injector is a very important component in the combustion system on spark-ignition direct-injection engine, in fact, this is the most influential component. Injector characteristics such as spray angle, penetration length, fuel delivery rate is known to be critical to the mixing, charge stratification and combustion stability of an engine. Finding the optimum performance of injector to match the chamber geometry, in-cylinder flow field and spark location, is the essence of an injector design.

In this experiment, two kinds of injector spray angle were used. One is a NAI that has a spray angle of 30 degree, and other is a WAI with a 70 degree spray angle. Figure 5-8 shows the injector spray image at atmospheric condition for both injector, WAI and NAI.



(a)



(b)

Figure 5-8 (a, b) Injection sequence at atmospheric condition for both injector spray angle

5.2 Injection parameters and Data collection

Data collection for these experiments follows SAE standard for Engine performance and Testing. SAE J1995 i.e Engine Power Test Code-Spark ignition and Compression ignition-Gross power rating.

The effect of different injection parameters to the performance of CNG direct injection engine was investigated. The injection parameters such as injection timing, injection pressure, and type of injectors are proven to be significantly affecting direct injection engine performance. While, the other injection parameters such as injector location, number of injector does not have significant effect to the performance and combustion of CNG direct injection engine.

5.2.1 Injection timing

Injection timing was set from early to late injection timing. As baseline, very early injection timing was used. At very early injection timing, the fuel was injected when intake valve starts to open, 300 °BTDC. Very early injection timing at direct injection systems can be considered as port injection. Even tough, there are slight difference between port injection and early injection at direct injection systems, very early injection timing can be considered as port injection because of its mixing behavior are almost the same

Further effect of injection timing was investigated. Injection timing variables was starts from 250 °BTDC to 100 °BTDC with increments 20. The experiment was done on full-load condition and homogeneous systems.

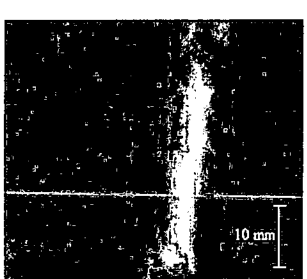
5.2.2 Injection pressure

Injection pressure was varied in range of 7.5 bars to 18 bars. Delivery rate of injectors was the most significant parameter affected by injection pressure changes. Higher injection pressure resulted smaller injection window needed by injector to delivery the fuel required by the systems. As the required window is smaller, the injection timing can be retarded closed to TDC. Injection timing closed to TDC is very useful to be implemented in stratified combustion system.

5.2.3 Injector spray angle

Injector spray angle variable is expected to affect the engine performance. There are two different spray angle used in this experiment, Narrow angle injector (NAI) and wide angle injector (WAI). Injector spray angle was affecting the mixing rate of the mixture. Wider injector angle gives higher percentage of air entrapment during the injection process, so it is expected to wider angle injector have higher mixing rate compare to narrow angle injector.

Figure 5-8 represents the injection sequence for both injector spray angle. This image is using pressure 18 bars, and was done in atmospheric condition. It can be seen that for WAI the fuel disappearance rate is faster compare to NAI. It is assumed that the fuel disappears because it was already mixed with air, so in conclusion that WAI have better mixing rate compare to NAI.



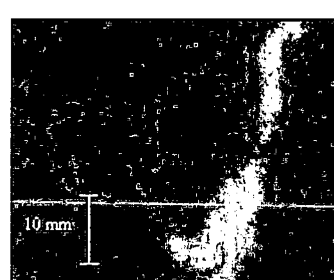


Figure 5-9 Injector image

hile, Figure 5-9 shows the injection events penetration. It can be seen that the spray angle trapped the air in the center of fuel injected. WAI shows larger trap region compared to NAI which is tends to mix better.

5.3 Engine performance parameters

The method that commonly applied to measure torque is the dynamometer that is supported by trunnions bearings and restrained from rotation only by a strut connected to a load cell. Whether the dyno absorb or provide power, a reaction torque is applied to the dyno. Hence, if the force applied to the strut is F, then torque applied to the engine is:

$$\tau_i = r \bullet F \qquad 5\text{-}1$$

Where, r is defined as the length of lever arms that is connected to the load.

The power P delivered by the engine and absorbed by the dynamometer is the product of torque and angular speed:

$$P = 2\pi N\tau \qquad 5\text{-}2$$

Where, N is the crank shaft rotational speed (RPM)

Since the work has been done by the engine, the work done through one revolution , or 2π radians, is $2\pi\tau$, it follows another parameters named mean effective pressure (MEP). MEP value can be generated from torque measurement, with:

$$mep = \frac{2\pi\tau}{V_d} \qquad 5\text{-}3$$

Where V_d is total volume displacement of the engine.

If the engine absorbing energy, then brake mean effective pressure (BMEP) is determined. So, BMEP is the external shaft work per unit volume by the engine.

Brake specific fuel consumption (*BSFC*) is the fuel flow rate $\dot{m}_f$ divided by the brake power P

$$BSFC = \frac{\dot{m}_f}{P} = \frac{\dot{m}_f}{2\pi N\tau} \qquad 5\text{-}4$$

The BSFC is a measure of engine efficiency. It inversely connected to the engine efficiency. The lower BSFC value the better the engine. By using BSFC as comparison tools, two different engine can be compared as long as they operate on the same fuel.

Another performance parameter of importance is the volumetric efficiency, η_v. It is defined as the mass of fuel and air inducted into the cylinder divided by the mass that would occupy the displaced volume at the density ρ_o in the intake manifold. Note that volumetric efficiency is a mass ratio not a volume ratio. For a four-stroke engine the volumetric efficiency is

$$\eta_v = \frac{2(\dot{m}_a + \dot{m}_f)}{\rho_o V_d N} \qquad 5\text{-}5$$

In the equation, $\dot{m}_f$ is the mass of fuel inducted. For a direct injection system, $\dot{m}_f = 0$. further, the density value can not be applied to turbocharger or supercharger systems. The density value can only be used for standard condition.

5.3.1 Combustion analysis

Combustion analysis can be generated from the pressure reading resulted by the pressure sensors. PC-based combustion analysis hardware and software is used to acquire and analyze the pressure data. The scheme of the systems can be seen in Figure 5-10 below.

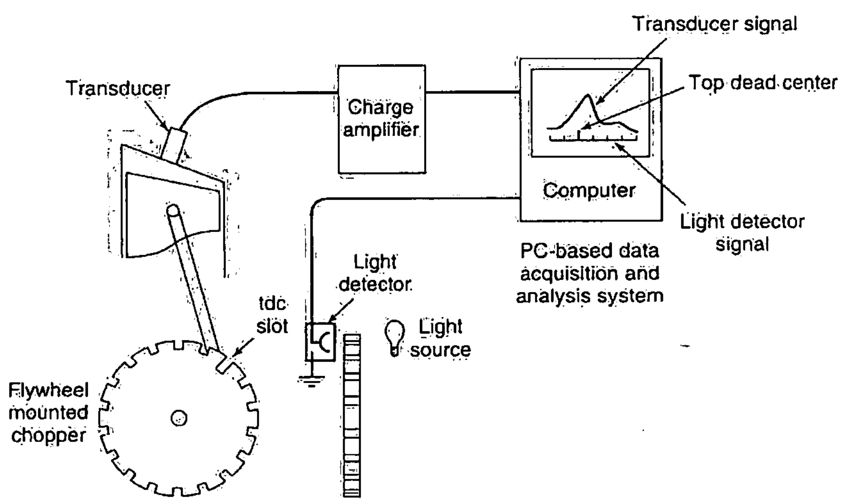


Figure 5-10 Typical pressure measurement system
Source: Ferguson. C. R., Kirkpatrick. A. T., Internal combustion engine applied thermosciences pg.121

The hardware consist of high-speed data acquisition and dedicated signal processor. The software performs statistical and thermodynamics analysis of the pressure data at real-time. Measurements of cylinder pressure can be used to determine not only the location of peak pressure, but also the instantaneous heat release, and burn fraction.

Lab-View high speed data acquisition is used to obtain the reading of cylinder pressure, with minimum 100 power cycles is recorded as the result of the experiment.

5.3.2 Indicated Mean Effective Pressure (IMEP)

This part of IMEP analysis is focusing on IMEP calculation procedure from cylinder pressure data. Burn and Emtage [1] there are many alternatives equation to get the result of IMEP integration from cylinder pressure data. The best equation which has the least error is:

$$IMEP = \frac{\Delta\theta}{V_d} \cdot \sum_{i=n_1}^{n_2} p(i)\frac{dV(i)}{d\theta} \qquad \text{5-6}$$

Where $p(i)$ is cylinder pressure at crank angle i, $V(i)$ is cylinder volume at crank angle i, V_d is cylinder displacement volume, and n_1 and n_2 refers to the BDC crank angle position.

It also stated that the IMEP errors is significantly affected by the crank angle resolution, pressure transducer accuracy and engine geometry. These errors can be significantly reduced if the crank angle resolution is less than 10 crank angle degree.

5.3.3 Heat release rate analysis.

Diagnosing internal combustion engine had been done in order to improve engine performance since years. Combustion analysis is important diagnostic in automotives research and development due to its relation to work producing process. Accurate diagnostic on combustion always been related to in-cylinder pressure data so that the combustion process can be distinguished.

The reduction of the effects of volume changed, heat transfer, and mass loss on cylinder pressure is called heat release analysis and is done within frame of first law of thermodynamics. The simplest approach is to regards the cylinder contents as single zone, whose thermodynamics state is considered as uniformed throughout the cylinder and represented by average value. Another approach is considering the cylinder contents as divided zones. Considering the properties and composition of each zones. Two-zones methods is more accurate compare to single-zone analysis. But, single-zone methods has been widely used in on-line processing to calculate heat release rate from pressure reading. It needs less computational time compare to two-zones methods.

The basic principal to analyze heat release from cylinder pressure data is the first law of thermodynamics; the energy conservation equation.

$$dQ - dW = dU - \sum_i h_i dm_i \qquad 5\text{-}7$$

where dU is the internal energy changes of the mass in the system, dQ is the heat transferred to the systems, and dW is work done by the systems, $\sum_i h_i dm_i$ is the enthalpy flux across the system boundary. Possible flow, are: flows that through the valves; fuel that directly injected into the cylinder; flows from crevice region; and piston ring blow-by.

In single-zone analysis there are assumption had to be made[61], which is:

- the in-cylinder contents through the chamber is uniform
- the modeled combustion represents as heat release
- heat released as product of combustion occurs uniformly in the chamber
- the gas mixture is ideal gas

Crevice effect are usually small, a sufficiently accurate model for their overall effect is to consider a single aggregate crevice volume where the gas is at the same pressure as the combustion chamber, but at the different temperature. Since this crevice region are narrow, an appropriate assumption that the crevice is at the wall temperature, by inserting the crevice model into equation (8), gives the chemical energy or gross heat release rate as:

$$\frac{dQ_{ch}}{d\theta} = \left(\frac{1}{\gamma - 1}\right) V \frac{dp}{d\theta} + \left(\frac{\gamma}{\gamma - 1}\right) p \frac{dV}{d\theta} + V_{cr}\left[\frac{T'}{T_w} + \frac{T'}{T_w(\gamma - 1)} + \frac{1}{bT_w} \ln\left(\frac{\gamma - 1}{\gamma' - 1}\right)\right] \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} \qquad 5\text{-}8$$

An example of the use of equation (5-8) to analyze an experimental pressure versus crank angle for a conventional spark ignition engine is shown in Figure 5-11.

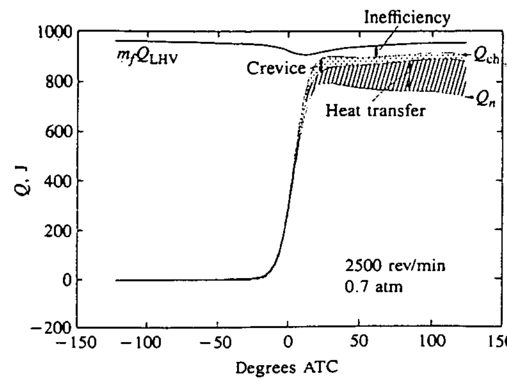


Figure 5-11 Results of heat release analysis showing the effect of heat transfer, crevices, and combustion inefficiency.
Source: Heywood. J. B, Internal Combustion Engine Fundamentals. Pg.388

The lowest curve shown is the net heat release. By adding the heat transfer and crevice effect gives chemical energy release by combustion process. On the top of the curve is the lower heating value of the fuel multiply by the amount of fuel enters the chamber as the total energy input. The gap between the gross heat release and total chemical energy input is the inefficiency of the combustion process. Inaccuracies of the cylinder pressure reading can also contribute to this different, by this analysis we can also determine the degree of error of the cylinder pressure reading.

(a) Rassweiler-withrow model

The Rassweiler-Withrow model was originally presented in 1938 and its still used for detemining mass fraction burned because of its simplicity and computational efficient. The cornerstone for this methods is the fact that pressure volume relation can be modeled by polytropic relation

$$pV^n = \text{constant} \tag{5-9}$$

where the constant exponent n values is 1.25 to 1.35 in range. This n values gives a good fit to experimental data for both compression and expansion process. This exponent value is termed as polytropic index.

Based on eq. 6 Rassweiler-Withrow models have some assumption in which there is no crevice effect taking into account $(h'-u+c_vT)\frac{dm_{cr}}{d\theta}=0$, the heat transfer value is zero $dQ_{ch}=dQ$, the specific heat value can be assumes as constant value and represented as polytropic index, $\gamma = n$, and assuming that there is no release of chemical energy during the compression phase prior to the combustion or during the expansion phase after the combustion, therefore $dQ = 0$. This assumption yields to:

$$dp = -\frac{np}{V}dV \qquad 5\text{-}10$$

from which polytropic index is found by integration and noting that n have constant value.

When considering combustion, equation (5-9) can be rewritten as

$$dp = \frac{n-1}{V}dQ - \frac{np}{V}dV = dp_c + dp_v \qquad 5\text{-}11$$

where dp_c is the pressure change due to combustion and dp_v is the pressure change due to volume changes.

In Rassweiler-Withrow models, the actual pressure change $\Delta p = p_{j+1} - p_j$ during the interval $\Delta\theta = \theta_{j+1} - \theta_j$, is assumed to be made up of a pressure due to combustion Δp_c, and pressure rise due to volume change Δp_V,

$$\Delta p = \Delta p_c + \Delta p_v \qquad 5\text{-}12$$

justified by equation (5-11). The pressure change due to volume change during interval $\Delta\theta$ is given by polytropic relation, which gives

$$\Delta p_v(j) = p_{j+1,v} - p_j = p_j\left(\left(\frac{V_j}{V_{j+1}}\right)^n - 1\right) \qquad 5\text{-}13$$

applying $\Delta\theta = \theta_{j+1} - \theta_j$, (5-12) and (5-13) yields to the pressure change due to combustion as

$$\Delta p_c(j) = p_{j+1} - p_j\left(\frac{V_j}{V_{j+1}}\right)^n \qquad 5\text{-}14$$

by assuming that the pressure rise due to combustion occurs in interval $\Delta\theta$ is proportional to the mass of mixture that burns, the mass fraction burned at the end of the process becomes

$$x_b(j)=\frac{m_b(j)}{m_b(total)}=\frac{\sum_{i=0}^{j}\Delta p_c(i)}{\sum_{i=0}^{N}\Delta p_c(i)} \qquad 5\text{-}15$$

where Δp_c is the result of equation (14), and N is the total number of crank angle intervals. The result of the mass fraction burned is S-curve. From this mass fraction burned, one can generate each step of the combustion process. Start of combustion, combustion duration and ignition delay is the properties that can be recognize by analyze the mass fraction burned curve.

The total heat released by the combustion can be approximate by

$$\Delta Q=\frac{V_{j+1/2}}{n-1}\Delta p_c(j) \qquad 5\text{-}16$$

where the volume during the interval j is approximate with $V_{j+1/2}$ (the volume at the middle of the intervals).

Rassweiler-Withrow methods have high dependency on the accuracy of the polytropic index value[68]. The polytropic index value for spark ignition engine commonly in range of 1.25 to 1.35. So the determination of correct polytopic index value is very important before doing the cylinder pressure analysis. Polytropic index can be found using the cylinder pressure reading at the motoring cycle. By implement the first law of thermodynamic analysis and find the best polytropic index that fits the cylinder pressure at motoring condition.

(b) Apparent heat release models

The heat release model that is derived from the first law of thermodynamics was proposed by Krieger and Borman [1967]. It was called the computation of net heat release. Apparent heat release model assume there is no heat transfer and crevice effect. Hence, the apparent heat release dQ can be expressed as:

$$dQ=\frac{1}{\gamma-1}Vdp+\frac{\gamma}{\gamma-1}pdV \qquad 5\text{-}17$$

this apparent heat release analysis have the same expression as Rassweiler and Withrow, based on the assumption of polytropic index value.

There are different result shows by these methods. Rassweiler and Withrow shows shortest combustion duration compare to apparent method.

Both methods is preferable for combustion analysis because of the least computational time, and less parameters required to analyze the combustion in the engine

5.4 Device Calibration

Calibration process is required to ensure the data accuracy. In this experiment, routine calibration mainly was done to dynamometer, pressure data acquisition system, and exhaust gas analyzer.

5.4.1 Dynamometer calibration process

Dynamometer was calibrated using calibrated weight as reference. The weight was put on one side of dynamometer and see the reading in the control panel, and after the first reading, the weight was gradually increase according to sequence in Table 5-4.

After dynamometer achieve its maximum reading (48 Nm), the weight was unload and control panel reading supposed to show zero reading. For other arm of dynamometer, same calibration procedure was applied.

Table 5-4 Load Sequence for Dynamometer Calibration Process

No	Weigth (kg)	Torque (Nm)
1	1	4
2	2	8
3	5	20
4	10	40
5	12	48

5.4.2 Pressure data acquisition systems calibration

Pressure reading was calibrated by using pressure testing device for engine. The procedure for pressure sensor calibration was:

- Turn-on all systems
- Unplug the spark plug from its position

- Install the manual pressure testing device on the spark position
- Motoring the engine in a low RPM condition (<300 RPM)
- Measure the reading by data acquisition systems
- Compare both result from manual pressure device and pressure sensor reading

Maximum pressure was compared using this procedure. The manual pressure reading device was showing pressure gauge reading. If the pressure sensors shows the same reading with pressure gauge, than in order to get absolute pressure reading, the whole pressure sensor reading need to add by 1.

5.4.3 Exhaust gas analyzer calibration

Exhaust gas analyzer calibration was done before starting any experimental works. GASMET exhaust gas analyzer already equipped with self calibration procedure. This calibration procedure is called *zero calibration.* Zero calibration is necessary to be done before starting the experiment. It will measure the background spectrum for subsequent sample spectrum measurement. Zero calibration can only be valid if the instrument is in steady state condition with certain cell temperature and interferometer temperature.

For zero calibration, the sample must filled with pure substance such as N_2 to make sure that there is no unwanted sample in the test cell. FTIR system will create background data base of the pure substance analysis as the base line for the measurement process. A typical background spectrum is presented in Figure 5-12. The background spectrum is represents the actual absolute intensity of infrared radiation that is transmitted through zero gas filled sample.

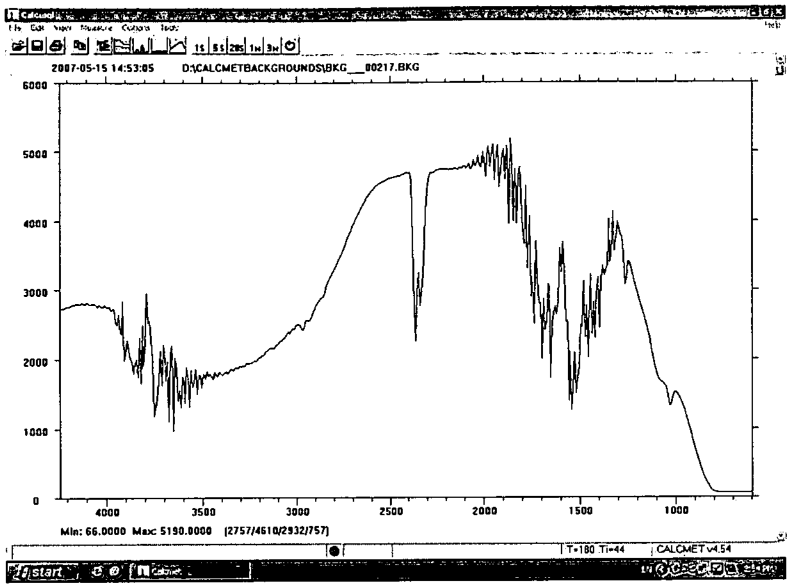


Figure 5-12 Calibration spectrum on FTIR system for emission analysis

The engine specification and experimental methods described in this section. Its also describes the calibration and data analysis process that will be used to obtained the experimental results. In the next sections, data presentation and analysis is discussed to get an optimum condition of the engine along with the analysis of the combustion process occurs during the process to get better understanding on CNG-DI engine.

CHAPTER VI

RESULT and ANALYSIS

6.1 Effect of Injection Timing to The Engine Performance of CNG-DI Engine

Analysis on the effect of injection timing to the performance of CNG-DI engine will be discussed mostly on this particular section. The engine test bed is a single cylinder research engine (SCRE) with a compression ratio of 14. The test was conducted at full load condition with ignition point set at MBT (Maximum Brake Torque). A high pressure injector (18bar) was used in a central injection system. Air-fuel ratio was set at stoichiometric condition that is 17.4 for CNG gas. Furthermore, 300 °BTDC injection timing was used as a baseline to simulate the port injection systems as comparison for other direct injection systems. By applying 300 °BTDC injection timing, fuel was injected during intake stroke which is assumed to have the same mixture properties as port injection systems (see Figure 6-1).

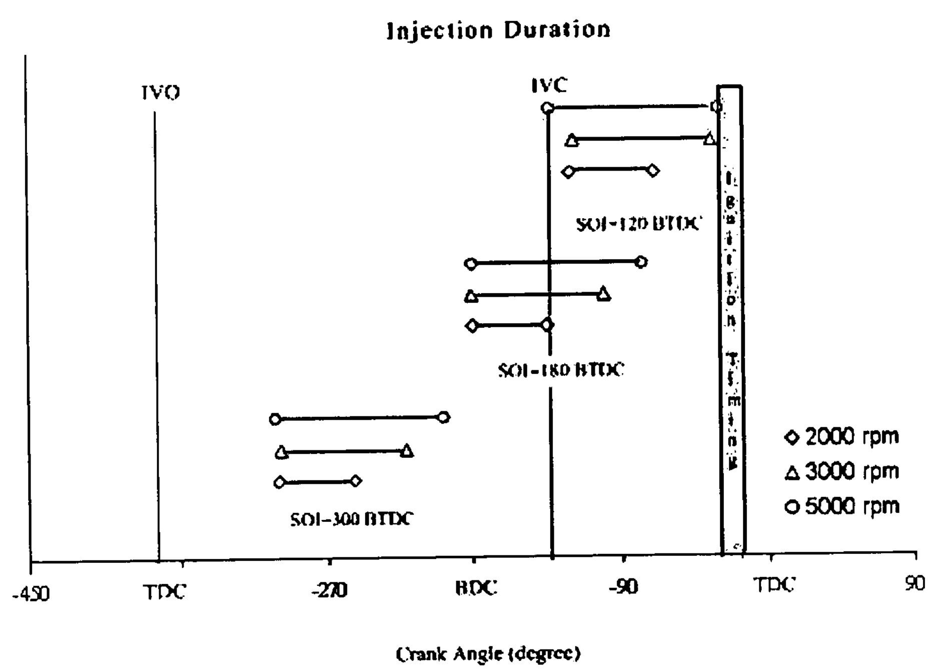


Figure 6-1 Injection duration for different injection timing.

The injection duration is increased along with the increasing of speed. However, retarding injection timing is limited by ignition timing parameters. So that the minimum limit of retarding injection timing for every speed is different. At higher speed, injection timing has to be advanced.

6.1.1 Engine Performance

Engine performance characteristics are shown in Figure 6-2. The torque curve shows tendency to increase up to 3500 RPM for all injection timing. Especially for injection timing 300°BTDC, torque keeps increasing until 4000 RPM, and then it starts to decrease (Figure 6-2.a).

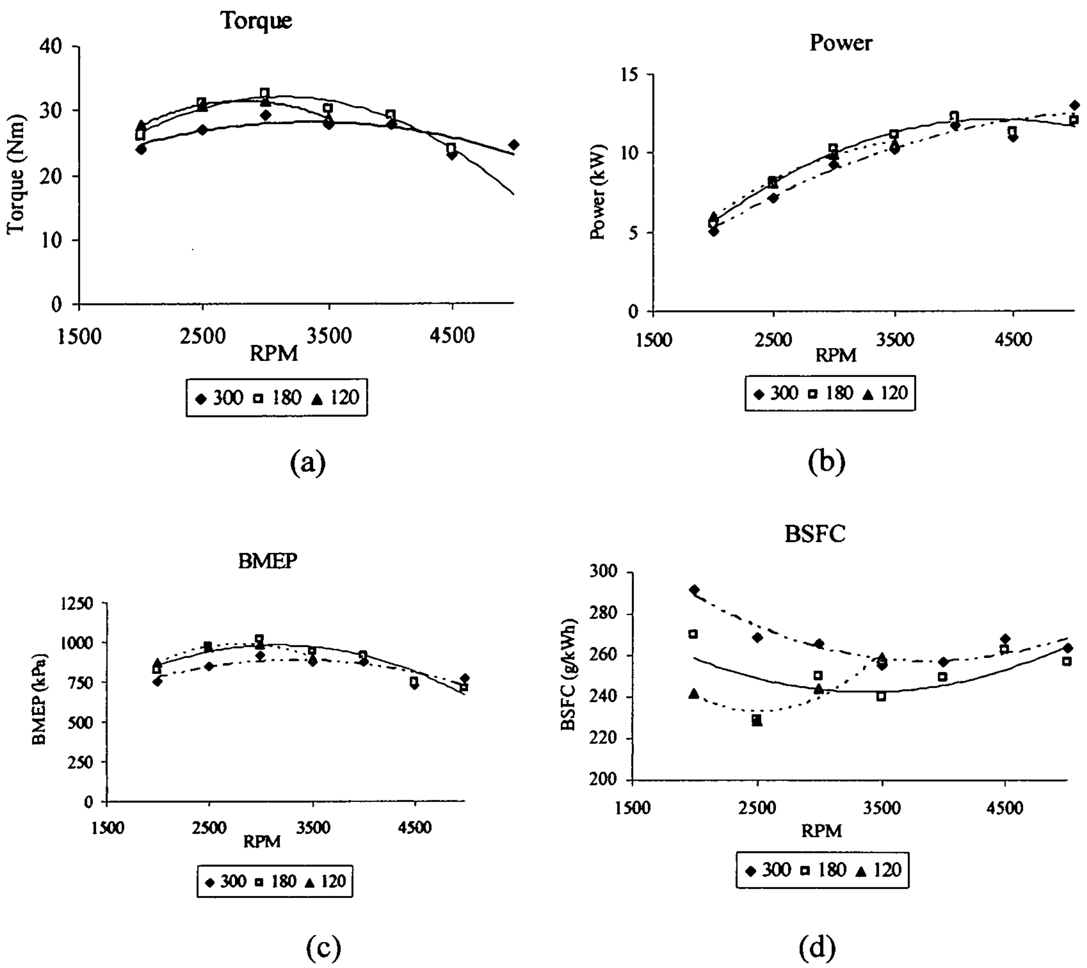


Figure 6-2 Engine Performance characteristics for different injection timing

At lower speed, late injection timing seems to have 10-20 % higher torque compared to very early injection timing. But at high speed, very early injection timing has a better torque.

Effect of injection timing on power is illustrated in Figure 6-2.b, late injection timing (180 and 120°BTDC) has 15 % better power compared to very early injection timing for speed less than 3500 RPM. As the speed continues to increase, 120°BTDC reaches its operating limit as described in Figure 6-1 where the injection events have to be done before ignition point. While, 180°BTDC injection timing shows decreasing performance until 5000RPM the performance drops to 5% below early injection timing (300°BTDC).

BMEP (Figure 6-2.c) shows a good relation with torque, and has similar curves as torque. Maximum value is ~1040 kPa at 3000 RPM, and it drops down to 750 kPa at 4500 RPM for all injection timing. BSFC (Figure 6-2.d) represents specific fuel consumption for this engine. 180BTDC gives 14% lower BSFC value compared to 300°BTDC at 2000RPM, while 300°BTDC has a tendency to have small changes on BSFC with 280 g/kWh, BSFC for 180°BTDC is increasing up to 2% lower compare to 300°BTDC at 5000RPM. 120°BTDC demonstrates same trends as 180°BTDC where it has 20% lower than 300°BTDC and increases to 1% higher than 300°BTDC at 3500RPM for BSFC.

High performance of full direct injection systems at lower engine speed may due to higher volumetric efficiency achieved. While at higher engine speed, volumetric efficiency effect is compensated with better mixing with port injection simulated.

Engine volumetric efficiency was illustrated in Figure 6-3. Late injection timing (120 and 180°BTDC) has overall higher volumetric efficiency (0.83 – 0.94) than early injection timing (300°BTDC) which is about 0.73-0.83. Higher performance of late injection timing at lower RPM may be due to higher volumetric efficiency of late injection. As the speed is increasing, volumetric efficiency is decreasing. Air-fuel mixing does not have enough time to mix properly, yielding performance drops at high RPM by using late injection timing. Moreover, at higher speed, time availability for mixing and combustion is an important parameter, good performance for both parameters can be achieved by early injection, even though the volumetric efficiency for every injection-timing at this rate is almost the same.

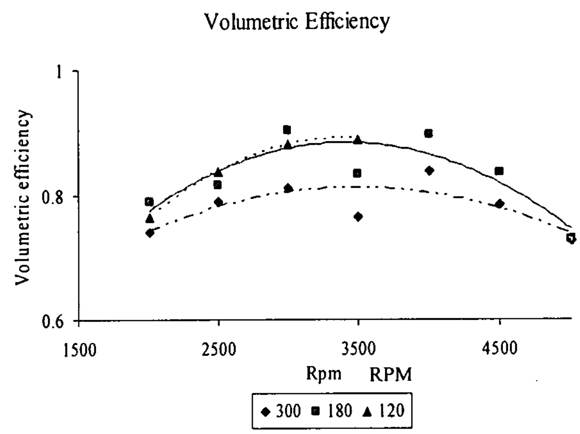
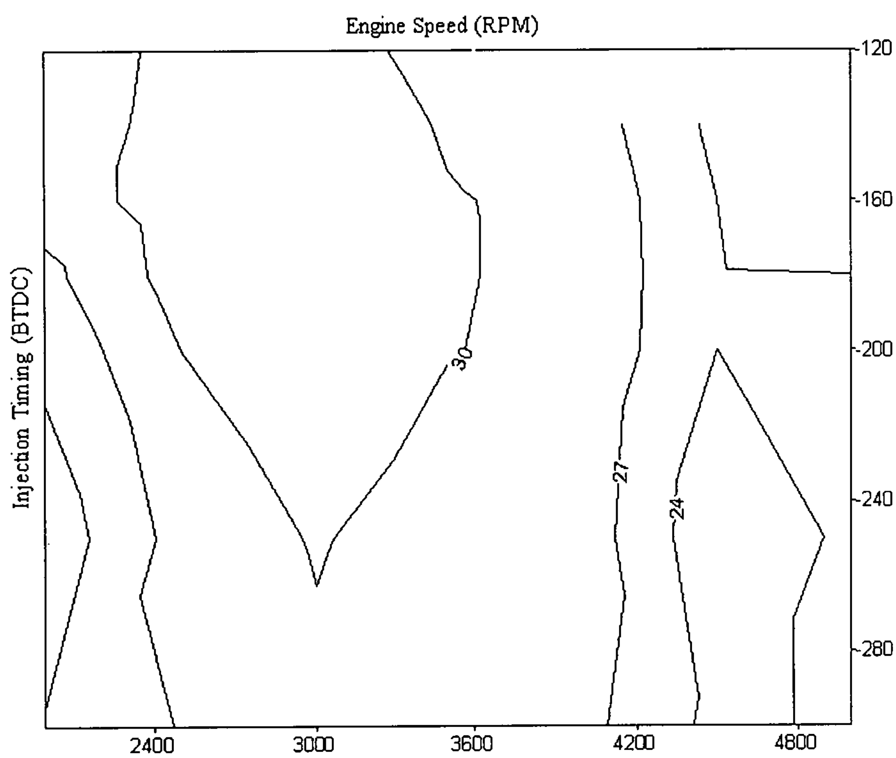
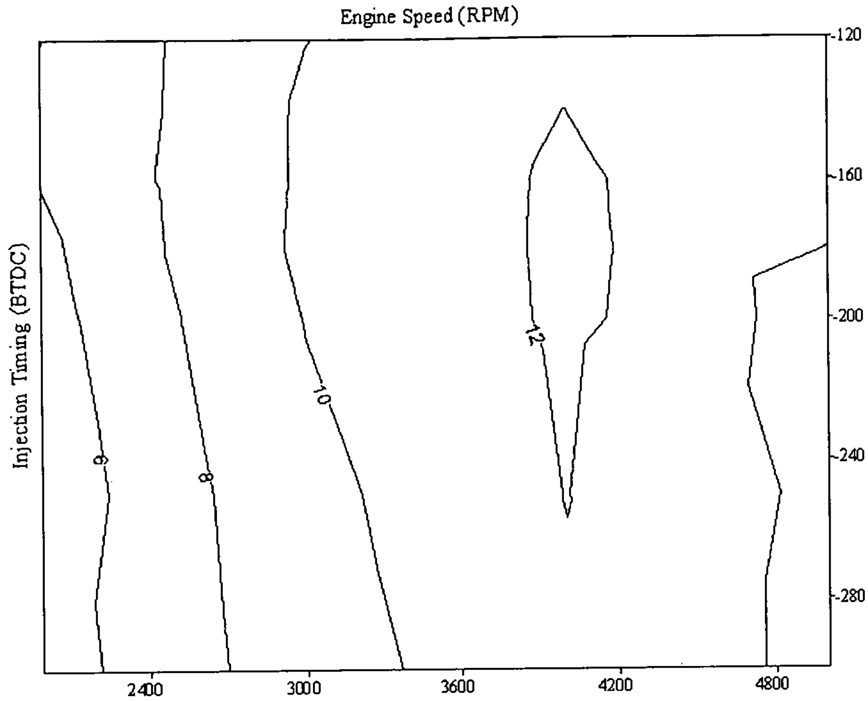


Figure 6-3 Volumetric efficiency for different injection timing.

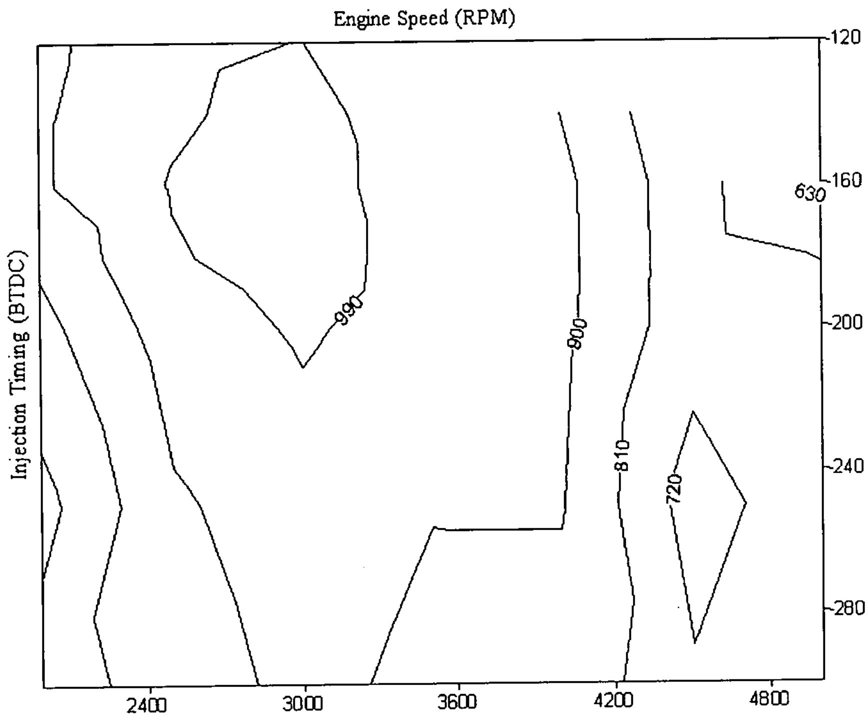
The performance mapping due to variations of injection timing can be seen in Figure 6-4.



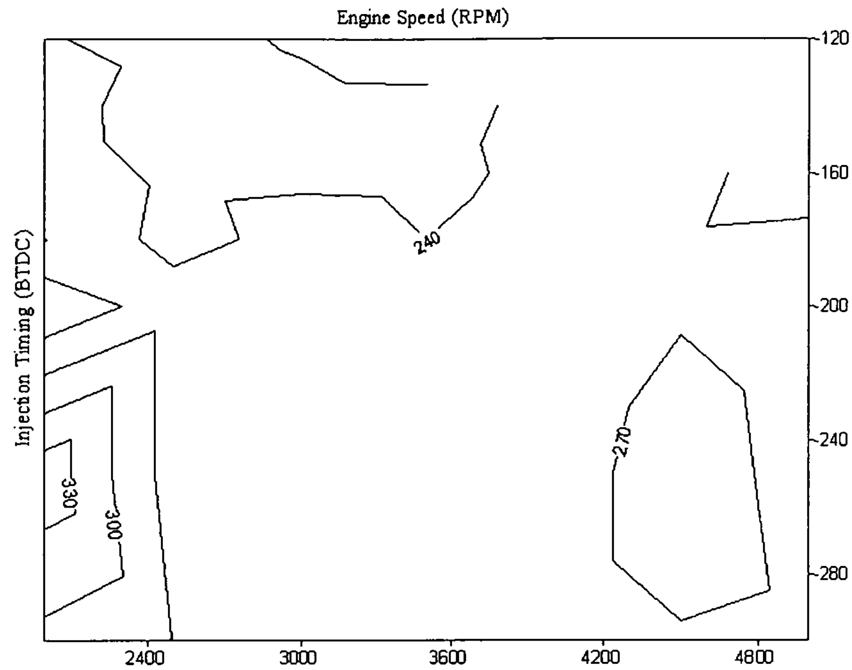
(a) Torque (Nm)



(b) Power (kW)



(c) BMEP (bar)



(d) BSFC (g/kWh)

Figure 6-4 Engine performance mapping sensitivity to the injection timing parameters

Optimum torque can be achieved as in the range of injection timing 140-180 °BTDC on 2500-3500 RPM with 32 Nm as seen in Figure 6-4.a. Moreover, the optimum power is achieved at the higher speed (more than 4500 RPM) and when early injection timing is applied (Figure 6-4.b). While, BSFC value shows the lowest at the range of 120-160 °BTDC injection timing on 2500-4000 RPM which is indicate that direct injection naturally reduced engine fuel consumption.

Injection timing effects on engine parameters are clearly shown in Figure 6-2. At low speed, is is observed that as the injection timing advanced to 300°BTDC, torque, BMEP, and BSFC drop 16% and 12% respectively compared to full direct injection method (120°BTDC). While at 3000 RPM, torque increase starts from 120°BTDC and achieves its maximum value at 160°BTDC, 32.4 Nm. After 160°BTDC, torque starts to decrease if injection timing is getting earlier. For 300°BTDC injection timing, torque reading is 28.2 Nm.

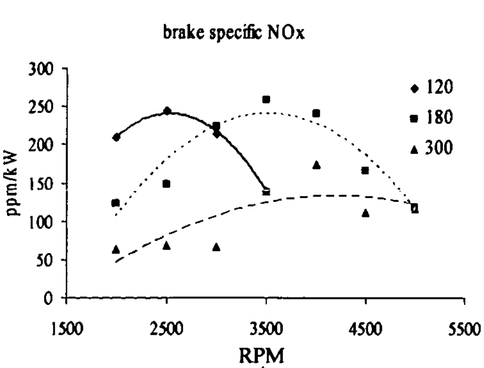
Same performance trend is also shown by power, and BMEP. For BSFC, the best value is also at 160°BTDC injection timing, around 235.3 g/kWh. At high speed, it shows different trends compared to lower engine speed. At high speed, maximum BMEP is at 300°BTDC, which has 15% better compared to late injection timing.

It can be concluded that, the late injection timing performance drops at high engine speed is due to lack of time availability for mixing and combustion. High volumetric efficiency at lower speed gives late injection timing better performances compared to early injection timing.

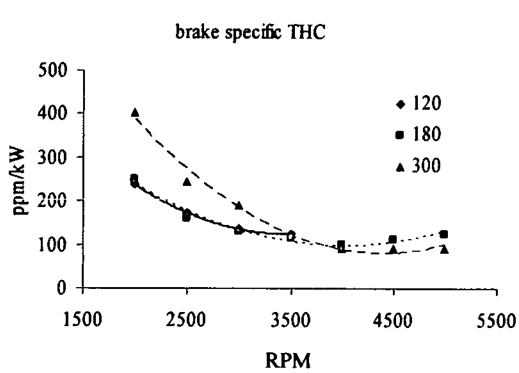
At high engine speed the volumetric efficiency is not affected so much by the injection parameter. Hence, early injection timing gives longer time for mixing as well as better performance at high speed condition.

6.1.2 Engine Emission

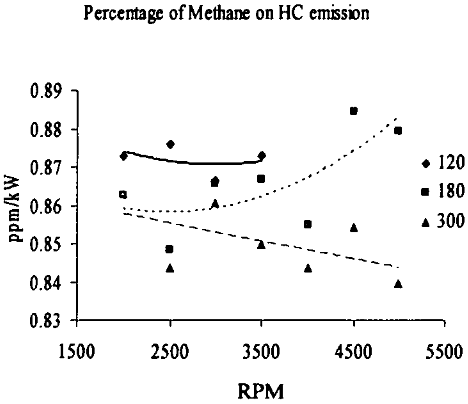
Emission of CNG-DI engines as the effect of injection timing is shown in Figure 6-5. Nitrogen oxide (NOx) for late injection timing (120°BTDC) gave high result at lower speed (<3000 RPM), as the increase in speed the NOx emission was decreasing for 120°BTDC injection timing. While for injection timing of 180°BTDC, the NOx emission shows high average result compared to other timing which is about 184.2 ppm/kW. Because of the behavior of NOx, which was formed due to high combustion temperature as well as indicates the combustion levels, 180°BTDC injection timing is considered to have the highest average of combustion efficiency.



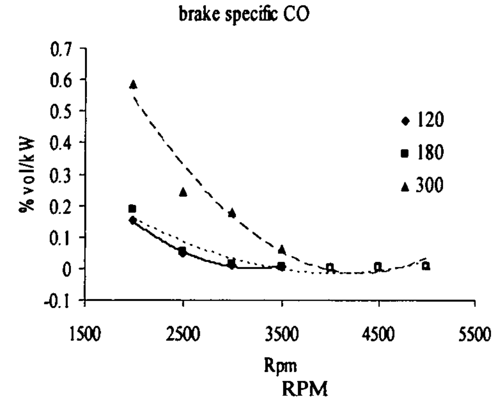
(a)



(b)



(c)



(d)

Figure 6-5 Emission characteristics for different injection timing

Equivalence to 300°BTDC injection timing, 120°BTDC injection timing has more than 88% and 180°BTDC has 71% of NOx emission.

An opposite behavior was demonstrated by hydrocarbon emission shown by Figure 6-5.b. 300°BTDC injection timing has the highest HC emission average compared to other injection timing. 178 ppm/kW HC emission from 300°BTDC was 19.5% higher compared to 180°BTDC and 9% higher compared to 120°BTDC. It is also shown at RPM more than 3500, the HC emission output was slightly different for all injection timing, even though 180°BTDC shows a bit higher HC emission on higher RPM. It can be concluded that at high RPM injection timing parameters have small effect on HC emission.

The percentage of methane emission in unburned hydrocarbon was illustrated in Figure 6-5.c. It shows that 120°BTDC injection timing had the highest percentage of methane in the HC emission at the speed less than 3500RPM. With an average of 87.6% the composition of methane emission was above methane emission for 300°BTDC injection timing that had 86%. Furthermore 120°BTDC methane percentage decreases as the speed increases and it has consistent difference with 300°BTDC methane percentage. 180°BTDC has similar value of methane percentage compared to 300BTDC which is 86%. At higher speed, there are contrast differences between 180°BTDC and 300BTDC injection timing. 180°BTDC had constant increase as the increase in RPM while 300°BTDC had decreasing trends as the increase in RPM where the maximum difference for methane percentage is 6% at 5000RPM while with 180°BTDC is higher.

CO emission was represented in Figure 6-5.d. it has the same behavior as HC and methane emission. The maximum value of CO emission for 300°BTDC was 0.6%vol/kWh, and for 180°BTDC its maximum value was 0.2 %vol/kWh. And the minimum values for all injection timing was 0.007%vol/kWh. CO also indicates completeness of combustion process. It shows that for 300BTDC the combustion efficiency is increasing with higher gradient than the trends shown by 180°BTDC. Furthermore, 120°BTDC had the same trends as 180°BTDC injection timing. But due to limitation of the fueling, 120°BTDC injection timing maximum RPM range was 3500RPM.

6.1.3 Engine Combustion

Pressure reading for various injection timing can be seen in Figure 6-6. 180°BTDC injection timing shows the highest pressure among all injection timing. It also means that the best combustion occur on 180°BTDC injection timing. Better combustion result for 180°BTDC injection timing may be due to combination of homogeneous combustion with small fuel stratification because of the injection process occurs before and after intake valve closed [Figure 6-1]. A small fraction of fuel stratification in the mixture can increase the combustion process[32]. Turbulence level during the combustion process may also be the reason of high pressure of 180°BTDC injection timing. It is known that direct injection systems can generate turbulence inside the combustion chamber, while existence of turbulence in the cylinder flow can improve the combustion process[69].

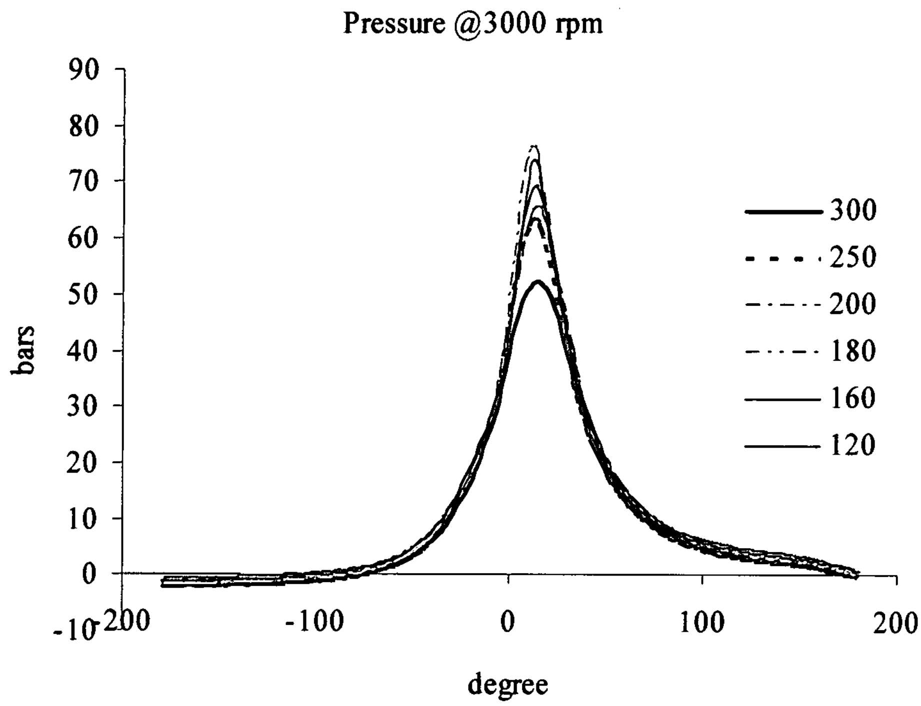


Figure 6-6 Pressure reading characteristics for different injection timing at 3000 RPM.

In the meanwhile, the lowest pressure output is 50 bars resulting from 300°BTDC injection timing that happens due to lower volumetric efficiency and highly homogeneous mixture of this injection timing. 300°BTDC injection timing is considered to have performance characteristics with port injection because the injection event happens when the intake valve opens, as seen in Figure 6-1.

Homogeneous mixture of methane at stoichiometric condition produced low flame speed[70] that needs enough time to be fully combustible. Improving combustion by adding turbulence can increase the process. But on 300°BTDC injection timing, the fuel was injected during intake process, so that the turbulence level will be lower compared to fuel injected after intake valve closed.

Combustion pressure of 68bar at 120°BTDC injection timing may be due to higher charge stratification and turbulence intensity of the mixture compared to 180 and 300°BTDC injection timing. Even though 120°BTDC injection timing pressure reading lies between 300°BTDC and 180°BTDC injection timing pressure reading.

Modest pressure reading of 120°BTDC injection timing may be due to low combustion efficiency of 120°BTDC at 3000 RPM as seen in Figure 6-9.

Figure 6-7 shows the sensitivity of IMEP to the injection timing parameters. IMEP is related to the combustion process, IMEP mainly is the work transferred from the combustion to the piston. Higher IMEP values indicate that good combustion occurs in the process. Over a wide RPM range, 180°BTDC shows high value of IMEP with maximum value occurs at 3500 RPM, 11.4 bar. During 2000 RPM condition, 180°BTDC is 9.6% and 120°BTDC injection timing is 10.6% higher compared to 300BTDC. The difference between those injection timing is decreasing as the RPM increased, especially for 120°BTDC was drop after 3000 RPM. At 5000 RPM, IMEP values between 180°BTDC and 300°BTDC injection timing had a 4% difference.

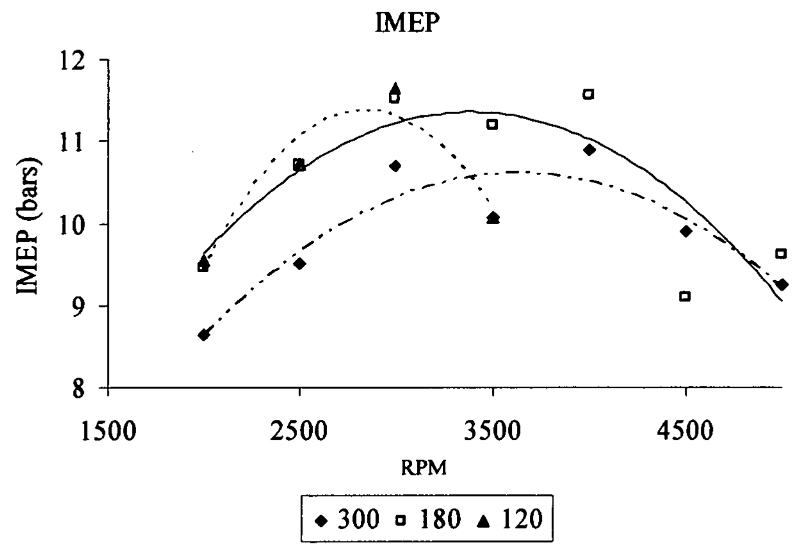


Figure 6-7 IMEP characteristics for different injection timing

Variation of IMEP was illustrated in Figure 6-8. It has been found that vehicle drive-ability problems usually occur when COV of IMEP exceeds about 10%. CNG-DI engine is proven to have good COV value which is less than 10%. Figure 6-8 shows that 300°BTDC injection timing has higher COV value compared to other injection timing 3-4%, which may be due to slow flame propagation of homogeneous mixture of CNG. 180°BTDC seems to have the lowest average value of COV. Small fraction of stratification, flow enhancement by injected fuel, and time availability to mix may be the reason of this low COV value.

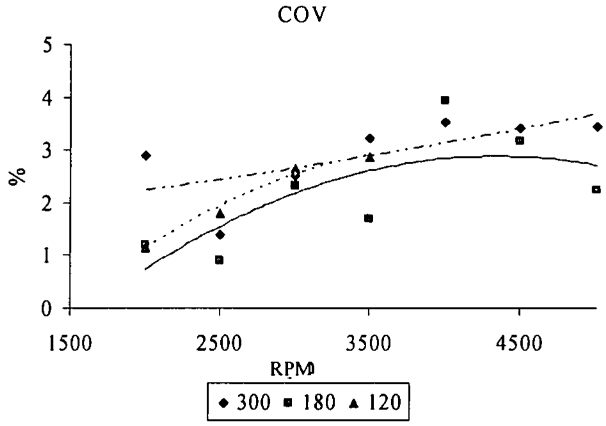


Figure 6-8 Cycle by cycle variation for different injection timing

Combustion efficiency characteristics for different injection timing are shown by Figure 6-9. At 300°BTDC injection timing, the combustion efficiency shows an increasing trend which is contrary to HC emission. This explains the source of HC emission that comes from combustion inefficiency. Maximum combustion efficiency for 300°BTDC injection timing was 0.7 at 4000 RPM and 0.6 at the minimum value at 2000 RPM. Moreover, high combustion efficiency at high RPM may also be the reason of increasing performance of 300°BTDC injection timing at high RPM condition.

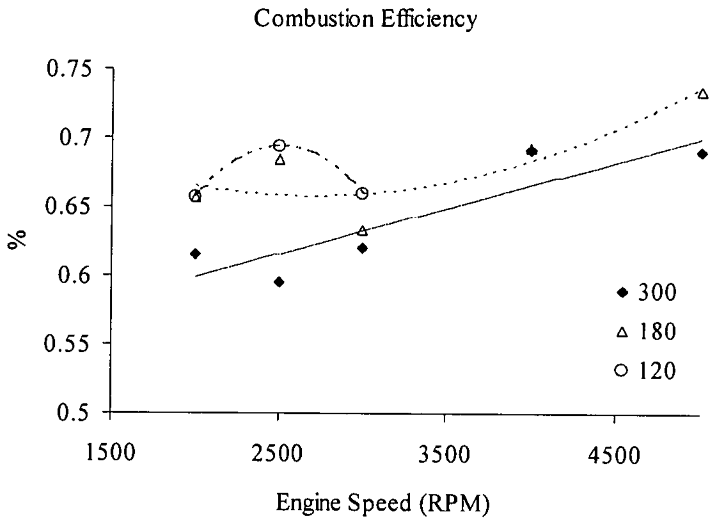


Figure 6-9 Combustion efficiency for different injection timing

Furthermore, a combustion efficiency characteristic for 180°BTDC is shown by Figure 6-9. It shows that at 2000 RPM the combustion efficiency is at the lowest state 0.65 and it is increasing with small slope until its maximum point is at 0.78 at 5000 RPM. The combustion efficiency of 180°BTDC is the highest compared to other injection timing. These results agreed with NOx emission of 180°BTDC that has high average for wide RPM range.

High combustion efficiency of 180°BTDC may due to the combination of high volumetric efficiency and better air-fuel mixing. Where, the volumetric efficiency for 180°BTDC is almost the same with 120°BTDC with larger mixing window available during the compression process.

Figure 6-9 represents the combustion efficiency for 120°BTDC injection timing. It shows that at the speed less than 3000 RPM the combustion efficiency is good. It can reach 0.76 at 2500 RPM as the maximum value for combustion efficiency. It also agrees with NOx results that indicate good combustion resulted higher NOx emissions.

Heat release characteristics for different injection timing were shown in Figure 6-10. It shows that 160°BTDC injection timing has the highest maximum heat release which is 0.08kJ/CA. This might be caused by the effect of charge stratification since

160BTDC has wider injection window after intake valve closed. 180°BTDC and 120°BTDC injection timing was placed after 160°BTDC with 0.073kJ/CA and 0.064kJ/CA. 120°BTDC low level of heat release may be due to high concentration of charge stratification that can lower the combustion temperature[37, 52] by decreasing the mixture temperature because of fuels that enter the combustion chamber.

300°BTDC injection timing has the lowest heat release, which is about 0.04kJ/CA due to its homogeneous charge stratification that prohibits longer combustion duration and leads to lower heat release on CNG-DI engine. Heat release pattern can explain the combustion process that occurs in the systems. Higher heat release shows that better combustion efficiency and higher NOx emission take place on the process.

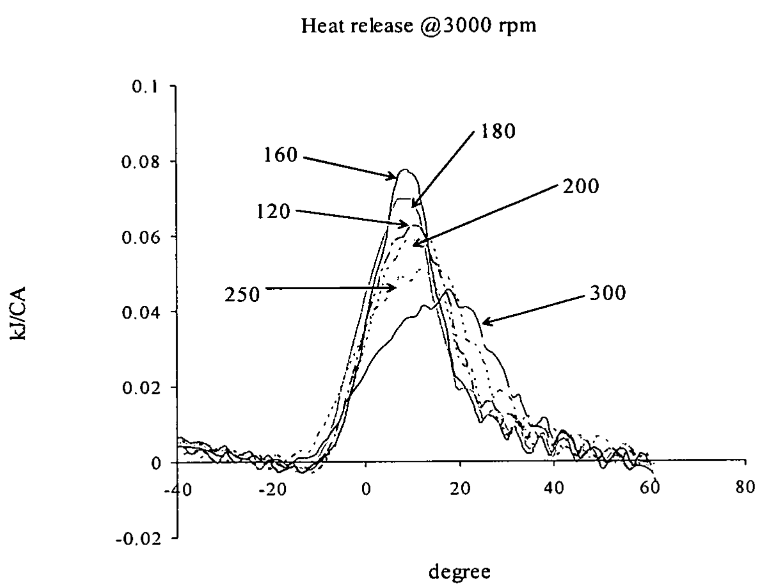


Figure 6-10 Heat release characteristics for different injection timing at 3000 RPM

Combustion duration of CNG mixture in DI engine is clearly shown in Figure 6-11. 180°BTDC and 120°BTDC seem to have same slope of main combustion stage (10-90%). The difference between those two is the initial combustion stage. It shows that 180°BTDC has a smaller angle compare to 120°BTDC. While 300°BTDC injection timing has the longest duration among all injection timing that leads to lower percentage of mass fraction burn.

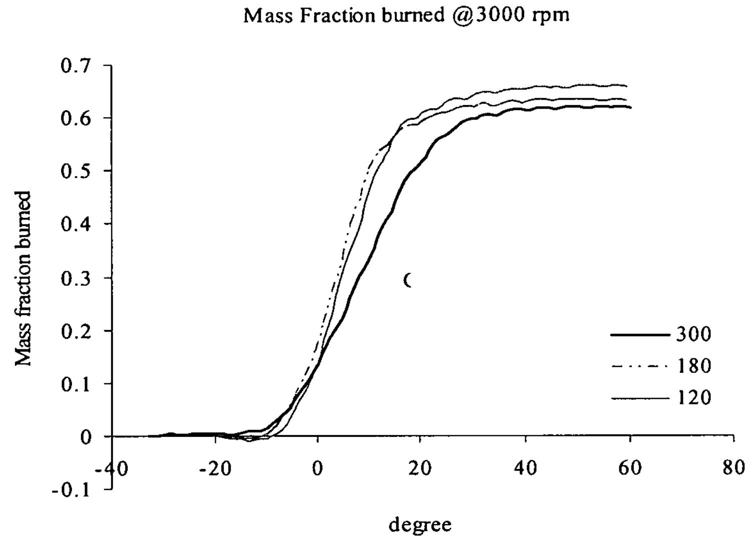


Figure 6-11 Heat release rate for various injection timing.

High RPM and full load condition requires good combustion process in order to hold up the CNG-DI engine performance. Direct injection systems offer another way to improve CNG engine performance in order to get closer performance to gasoline engine. At high RPM condition, CNG engine undergoes a drop in power due to gas form of CNG that leads to lower energy content. Larger injection window is needed to get the same energy as gasoline. So, there will be limitation to retard injection timing.

The effect of injection timing at 5000 RPM was shown in Figure 6-12. 160°BTDC has the highest value by 58 bar and 130°BTDC has the lowest by 38 bar. Low pressure result of 130°BTDC injection timing may be due to lack of mixing time in high speed condition. These behaviors can also be seen in heat release pattern in Figure 6-13.

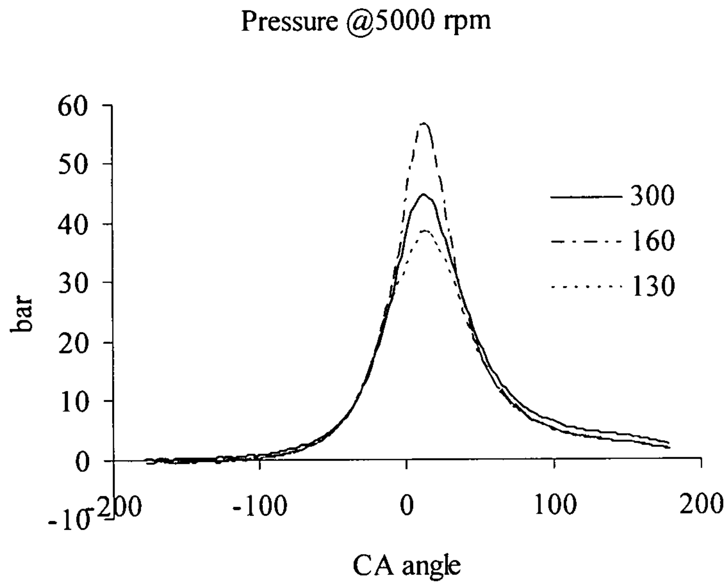


Figure 6-12 Pressure characteristics for different injection timing at 5000 RPM

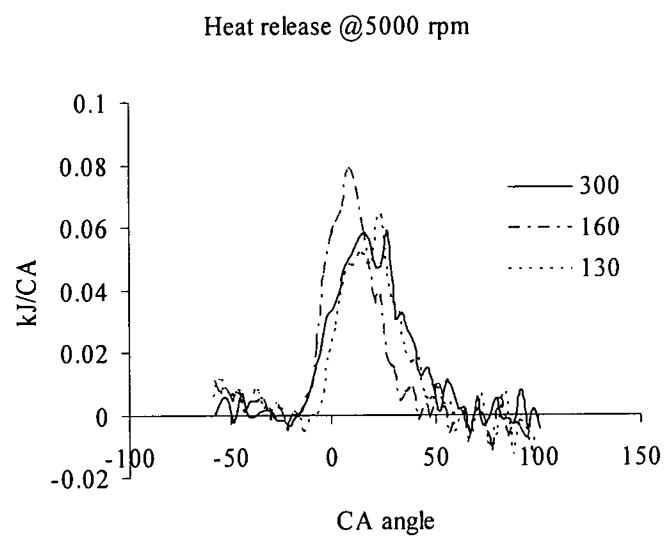


Figure 6-13 Heat release rate for different injection timing at 5000 RPM.

Rate of mass fraction burned at 5000 RPM was shown in Figure 6-14. 160°BTDC injection timing seems to be the best injection at 5000 RPM condition, in which combustion duration is shorter and combustion efficiency is higher. While 300°BTDC has slower main mass fraction burned as it occurs in homogeneous mixture. It also shows that for initial combustion, 160°BTDC and 300°BTDC have same duration. It shows that the mixture characteristics at the time of ignition for both injection timing are almost the same at high speed condition.

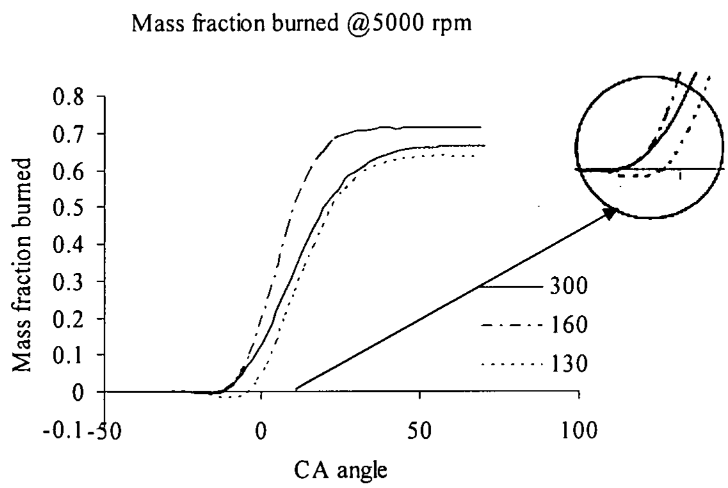
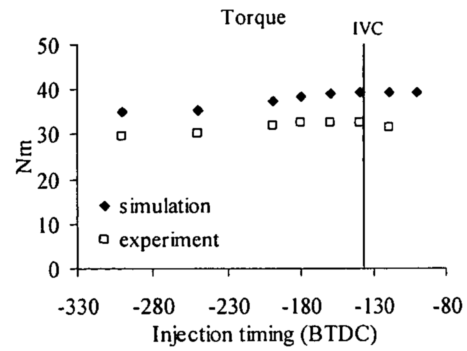


Figure 6-14 Mass fraction burned for different injection timing at 5000 RPM

Figure 6-14 have shown that the ignition delay period for late injection timing is longer compared to partial and early injection timing. Which is indicate inconsistence of the mixture due to uncontrolled high charge stratification in the combustion chamber. This phenomenon either due too rich condition or over-lean condition around the spark plug at the time of the ignition takes place.

6.1.4 Experimental results comparison with simulation results

In comparison with simulation results, the simulations give almost the same trends as experimental results. As seen in Figure 6-15, the torque reading for both simulation and experiment increased as the injection timing is retarded which mainly affect by engine volumetric efficiency. Same results shown by the BSFC graph, which is both results, show the same trends. However, there is noticeable offset between both data. This offset may due to main assumption used in the simulation where the CNG was considered as methane which have higher heating value. Another reason is the assumption that considered complete combustion occur during the combustion process.



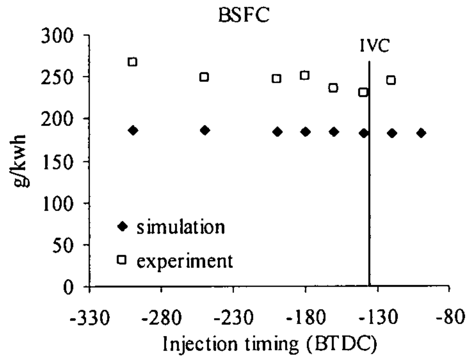


Figure 6-15 Engine performance comparison between experimental and simulation results.

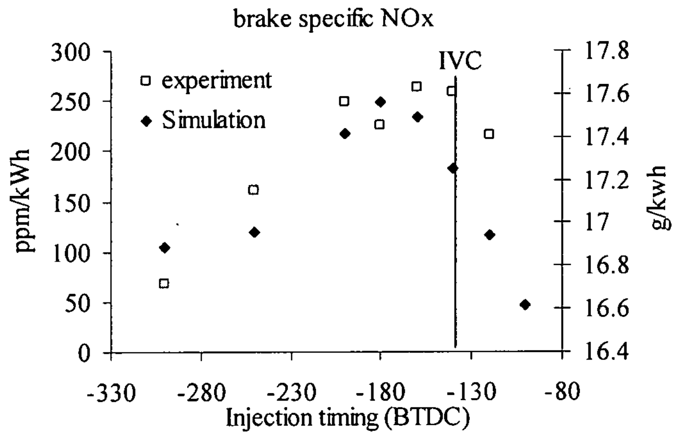


Figure 6-16 Nitric oxide emission comparison

Figure 6-16 shows simulation results have almost the same values as experimental results. NOx emissions calculation in simulation process used thermal NOx calculations as the main idea. This similar result may indicate that most NOx emissions at CNG-DI engine occur due to very high temperature at the combustion chamber.

Summary

Late injection timing (120°BTDC) shows 20% higher in torque, 15% higher in power and 20% lower in BSFC compared to 300°BTDC injection timing at the engine speed less than 3000 RPM. The reason being is that it has 10% higher volumetric efficiency compared to 300°BTDC injection timing. 120°BTDC also performs 10% higher in combustion efficiency which results 88% more NOx emissions but 19,5% lesser HC emission due to better combustion process that 120°BTDC poses.

Faster main stage of combustion process of 120°BTDC shows it has higher turbulence intensity, further it also has higher charge stratification that is indicated by faster initial combustion stage and higher percentage of unburned methane which shows inhomogeneous mixture. Increasing engine speed lowers the performance of 120°BTDC injection timing due to lesser time for fuel preparation that results lower combustion efficiency and increasing COV.

180°BTDC injection timing performs 17% higher on torque, 14,5% higher on power and 14% lower on BSFC value match up to 300°BTDC injection timing at the engine speed of 2000 RPM due to its 10% higher in volumetric efficiency. As the engine speed increases, 180°BTDC is experiencing increase in torque until it is reaching its maximum value at 3000 RPM. When the engine speed continues to increase, engine performance of 180°BTDC is decreasing down to 7% lower on torque, 5% lower on power and 2% lower on BSFC values compared to 300°BTDC injection timing.

The reason being is that 180°BTDC injection timing has smaller mixing window when the engine speed increases. Combustion efficiency of 180°BTDC is higher than 300°BTDC due to higher turbulence intensity with homogeneous mixture which is also shown by high NOx and low HC emission. Percentage of methane has shown that 180°BTDC has lower percentage than 120°TDC that shows it has better combustion process but it does not have enough time to complete the combustion. Better combustion of 180°BTDC is also shown by IMEP which has higher average compared to 300°TDC injection timing. It also demonstrates the lowest COV for wide range of engine speed.

6.2 Effect of Rail Pressure to The Engine Performance of CNG-DI Engine

Injection pressure primarily affects the fuel rate in the injection event. Injection pressure and ambient pressure difference determine the fuel injection rate. As smaller the difference it will take longer injection duration to provide the amount of fuel required in the combustion process. As seen in Figure 6-17, increase in injection pressure will decrease the injection duration needed to deliver required fuel. The effect of injection pressure to the performance, emission and combustion characteristics on CNG-DI engine will be discussed next.

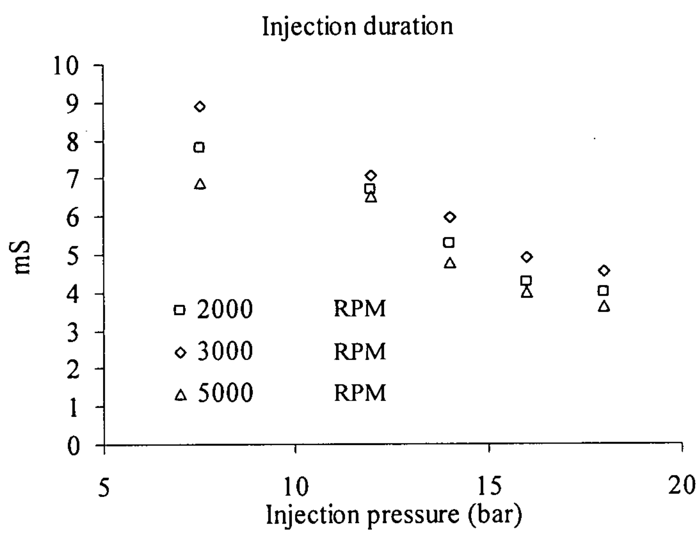
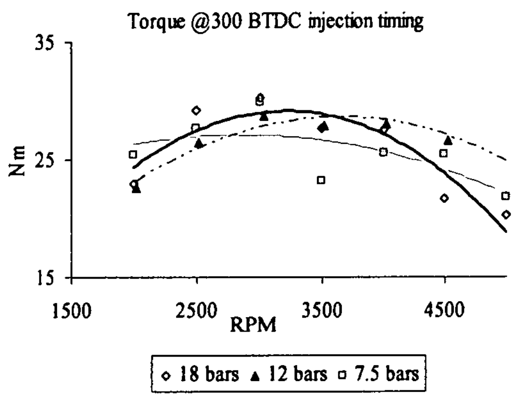


Figure 6-17 Injection duration for different injection pressure

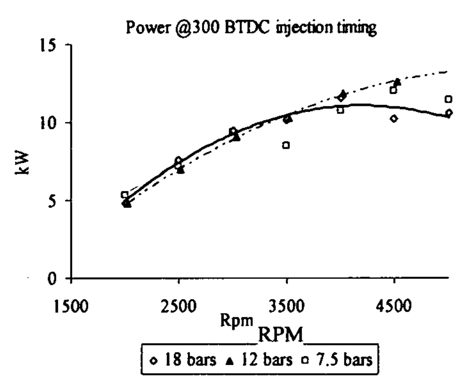
Retarding injection timing is limited if lower injection pressure was used. For example; retarded injection timing limit for 7.5 bar injection pressure is 240°BTDC, and for 12 bar injection pressure is 224°BTDC, while for 18 bar injection pressure retarding limit extends to 130°BTDC at 5000 RPM.

6.2.1 Engine Performance

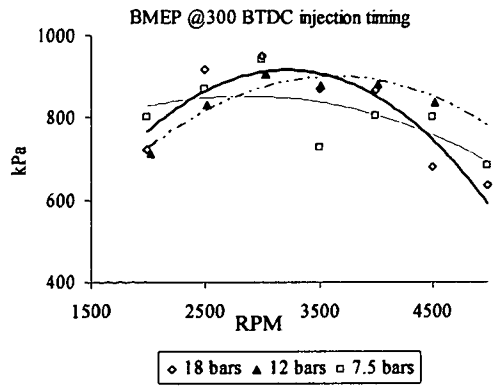
Figure 6-18 depicts the performance characteristics for different injection pressure over wide RPM range. 300°BTDC injection timing was used to determine the effect of injection pressure to the engine performance due to its capability to run over extensive RPM range. Torque, power and BMEP values show less variation for various injection pressure. Torque and BMEP has its maximum at 3000 RPM for 12 and 18 bar injection pressure while 7.5 bar injection pressure, torque and BMEP show decreasing tendency as RPM increases. Longer period of fuel injection can cause decline of volumetric efficiency that results dropping on the performance. 18 bar injection pressure shows good performance during 2000-3500 RPM. But further increase in RPM shows drop of 18bar performance due to excessive charge stratification. As for 12 bar injection pressure, in the range of 2000-3500 RPM the lower performance owes to lower injection velocity compared to 18bar that has higher effect to the mixing process. Opposing behavior showed at 3500-5000 RPM, the engine performance was steady with small drop at 5000 RPM, these matters may be due to higher volumetric efficiency because of less disturbance to air intake that enters the combustion chamber. BSFC characteristics shown in Figure 6-18.d demonstrate that 12bar and 18bar injection pressure has the same values, where it has minimum value at 3000 RPM, which is about 250g/kWh. 7.5 bar percent different characteristics, which it increases linearly with minimum values, occur at 2000 RPM with 250 g/kWh and maximum values occurring at 5000RPM with 354 g/kWh.

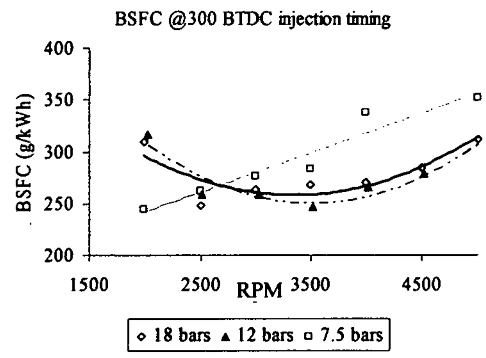


(a)



(b)



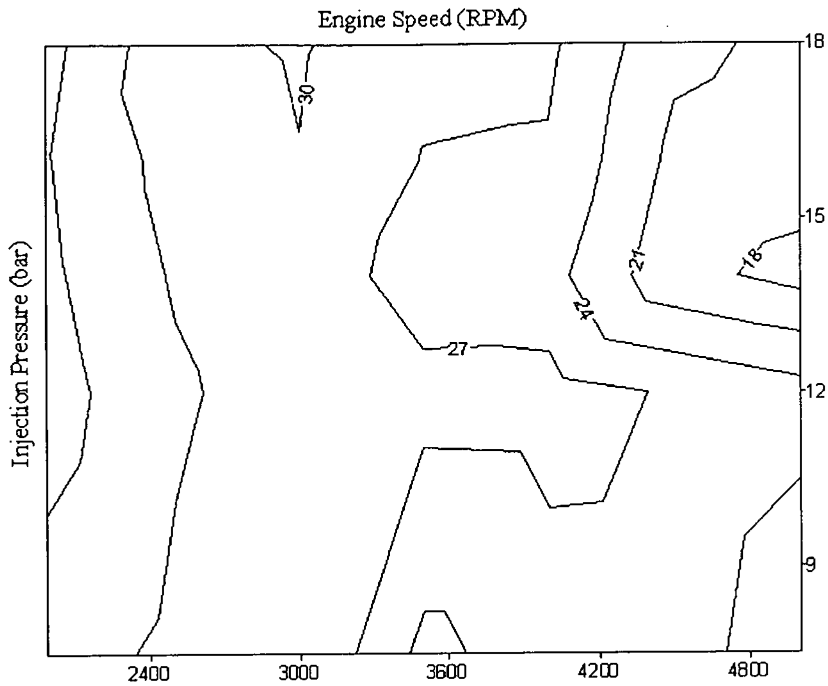


(c) (d)

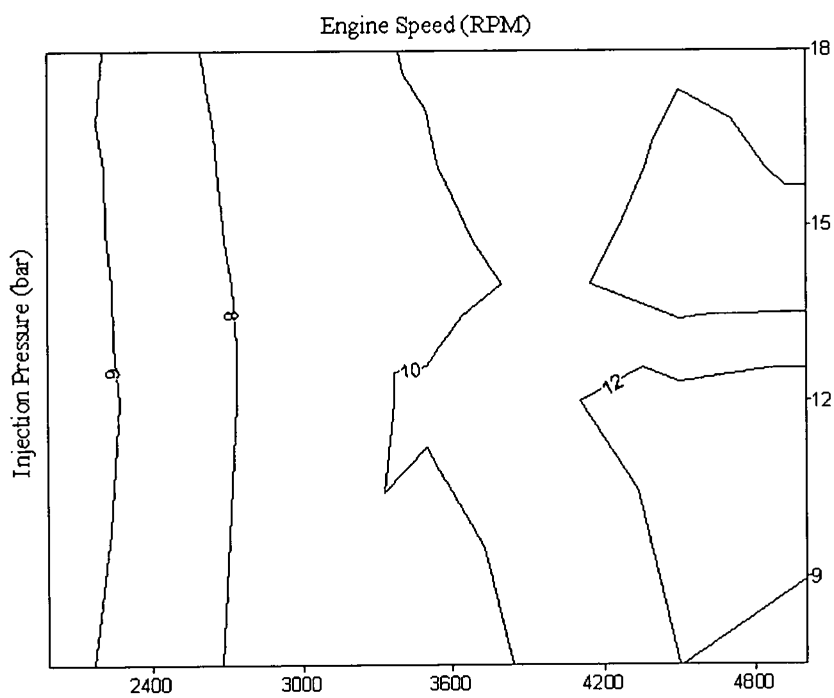
Figure 6-18 Performance characteristics for different injection pressure

Figure 6-19 characterizes the effect of injection pressure to the engine performance at different speed. 7.5 bar injection pressure shows good performance at high speed (5000 RPM). Even though the engine performance shows decreasing trends for 7.5 bar, at high rate (5000 RPM) it has higher performance match up to other injection pressure due to better mixing and increasing charge stratification for higher injection pressure. Performance characteristics for 2000 RPM show that injection pressure has small effect to the performance at this speed, where the volumetric efficiency values are slightly different for all injection pressure. While, 18 bar injection pressure displays better performance due to higher volumetric efficiency as illustrated in Figure 6-20.

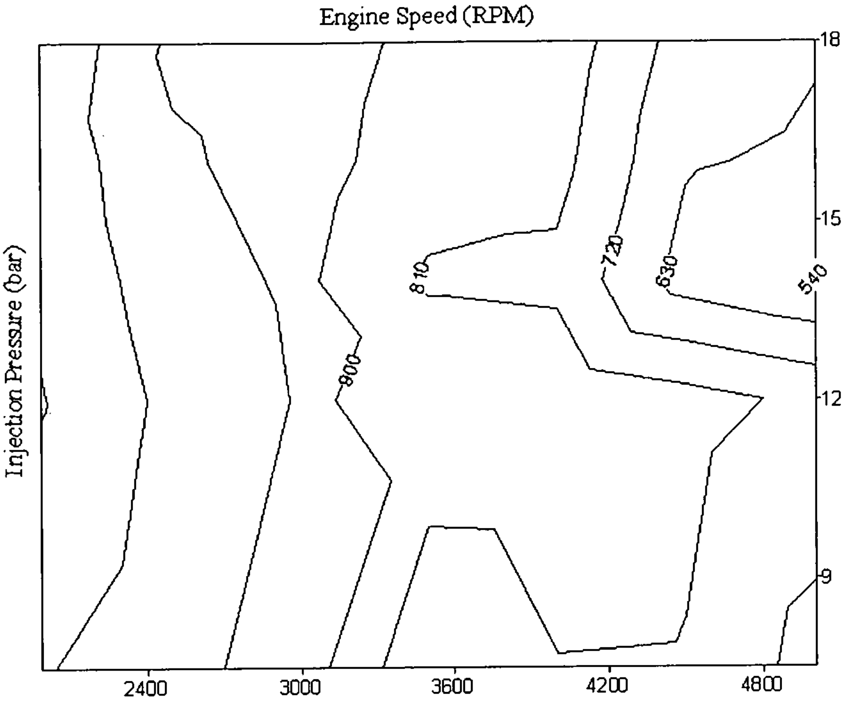
Figure 6-19 also shown the optimum power achieved with the 12 bar injection pressure at high engine speed. While at low engine speed, torque gives the highest reading on the employment of 18 bar injection pressure.



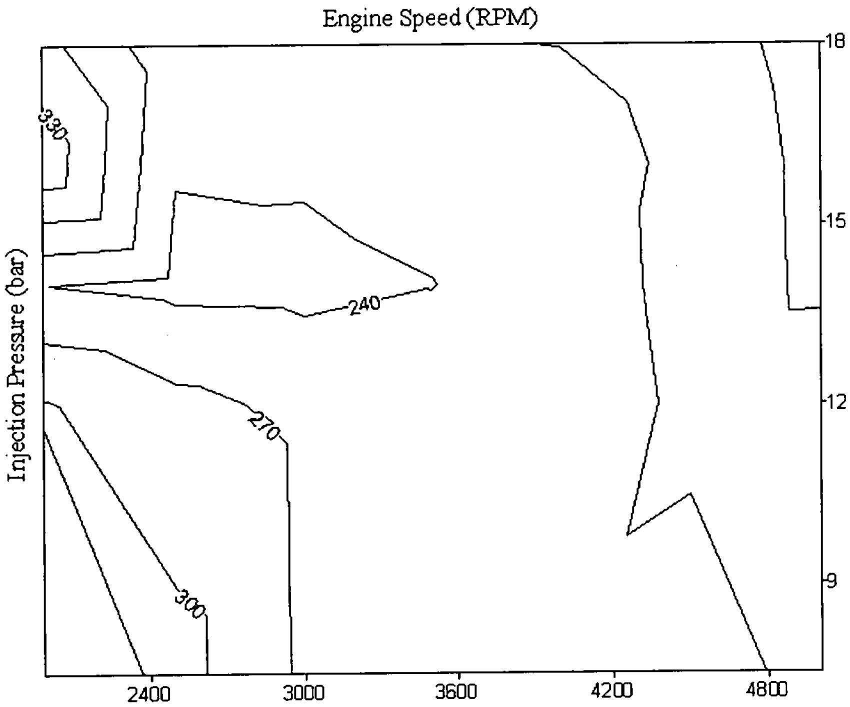
(a) Torque (Nm)



(b) Power (kW)



(c) BMEP (bar)



(d) BSFC (g/kWh)

Figure 6-19 Engine performance mapping sensitivity to the Injection pressure parameters

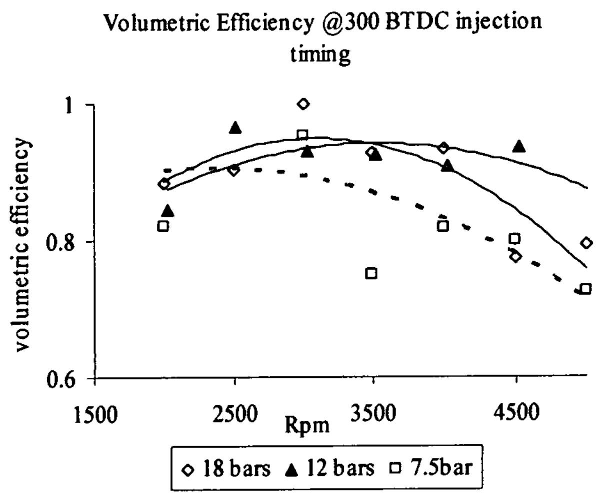


Figure 6-20 Volumetric efficiency sensitivity for different injection pressure

18 bar injection pressure performs better volumetric efficiency at lower speed (2000-3000 RPM), while at higher RPM the volumetric efficiency is declining to 0.79 but it is 8% higher than 7.5 bar injection pressure volumetric efficiency. 12 bar injection pressure shows less variation with speed change. Volumetric efficiency values range from 0.84 to 0.97. Despite the fact that volumetric efficiency is dropping as RPM increase, 12 bar injection pressure shows 0.87 volumetric efficiency value at 5000 RPM which is the highest value at that speed.

Steady volumetric efficiency value of 12 bar injection pressure could prevent performance drop at higher RPM, as illustrated in Figure 6-26. IMEP values for 12 bar injection pressure are steady with range 8.4 – 11 bar. IMEP is lower than 18 bar injection pressure at lower speed due to lower turbulence intensity but it shows almost constant value for wide RPM range so that at high RPM the IMEP is 6.2% higher compared to IMEP from 18bar injection pressure. 18 bar injection pressure IMEP demonstrates better values compared to other injection pressure due to higher turbulence created by high injection pressure. Higher turbulence intensity can promote the combustion that can be represented by high IMEP values for 18 bar injection pressure for wide speed range. Furthermore, IMEP for 18 bar injection pressure is experiencing drop at the speed more than 4000 RPM, due to dropping in volumetric efficiency which is about 21% from its maximum value.

6.2.2 Engine Emission

Next, our discussion will be on the effect of injection pressure to the exhaust emission of the engine as shown in Figure 6-21. Analysis on combustion products will be carried on to make a clear insight to the process that had been occurring during the combustion process. Emission characteristics for 7.5 bar injection pressure could reveal better combustion process that occurs at 2000 RPM. 120 ppm/kWh of NOx emission that took place between 18 bar and 12 bar injection pressure shows that the combustion of 7.5 injection pressure is better than 12 bar at this speed. While, CO emission of 7.5 bar injection pressure is almost the same as the 18bar and it has the lowest average of HC emission which is 137.6 ppm/kWh. It means that 7.5 bar injection pressure has better combustion efficiency that causes practically high NOx, low CO and HC emission at lower speed or 2000 RPM.

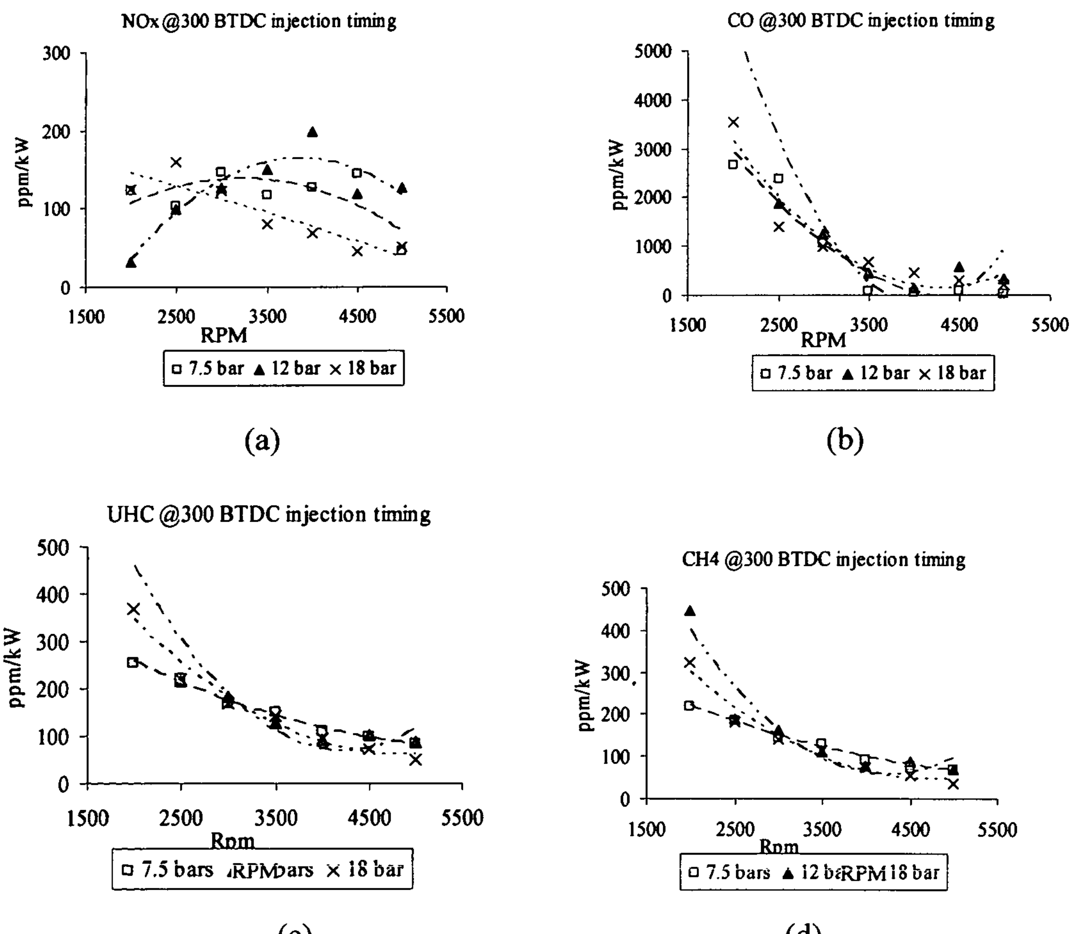


Figure 6-21 Emission characteristics for different injection pressure

In contrary, 12 bar injection pressure shows different results, where it has lower NOx emission at 2000 RPM and it increases as the speed increases. This indicates that the

combustion process employing 12 bar injection pressure is getting better as the speed increases. These behaviors are also strengthened by CO, HC and methane emission which is decreasing as the speed increase. The highest level of NOx emission for 12 bar injection pressure is 199 ppm/kWh that occur at 4000 RPM and highest CO and HC emission occuring at 2000 RPM with values of 2660 ppm/kWh for CO and 256 ppm/kWh for HC emission. Better combustion of 12 bar injection pressure may be due to combination of fair mixing and enough turbulence created during the injection process.
Emission characteristics resulting from utilizing 18 bar injection pressure have the same behavior as 7.5 bar injection pressure. NOx has its maximum value at 2500 RPM with 159 ppm/kWh and decreasing. Higher injection pressure at early injection timing could create significant disturbance to the intake flow that reduces volumetric efficiency and affects combustion process.

Combustion product characteristics relative to injection pressure can be observed in Figure 6-22. Emission output at 2000RPM demonstrated that there is a tendency of the emission to increase as the injection pressure increase. NOx levels show small changes at this RPM where unburned hydrocarbon and methane emission show significant increase that indicates low combustion efficiency due to increasing charge stratification levels together with the increasing of injection pressure. The charge stratification is decreasing when applying higher injection pressure at higher speed. Emission levels at 3000RPM for unburned hydrocarbon and methane emission show less variation with changing in injection pressure. It indicates that injection event gave a little effect to the in-cylinder flow that determines the mixing properties. At higher speed, piston stroke and intake flow behavior gave significant effect to the cylinder flow. While at rate speed (5000 RPM), injection pressure shows less effect on NOx emission. Furthermore, unburned hydrocarbon and methane emission are decreasing as the injection pressure increases. It implies that the combustion performance is increasing. CO emission has a tendency to increase as the injection pressure increases. This implies that there is better combustion of higher injection pressure. However, due to less available time or small window for combustion process, the combustion process is incomplete. This behavior may also be the one that explained the steady NOx emission results as the injection pressure increases.

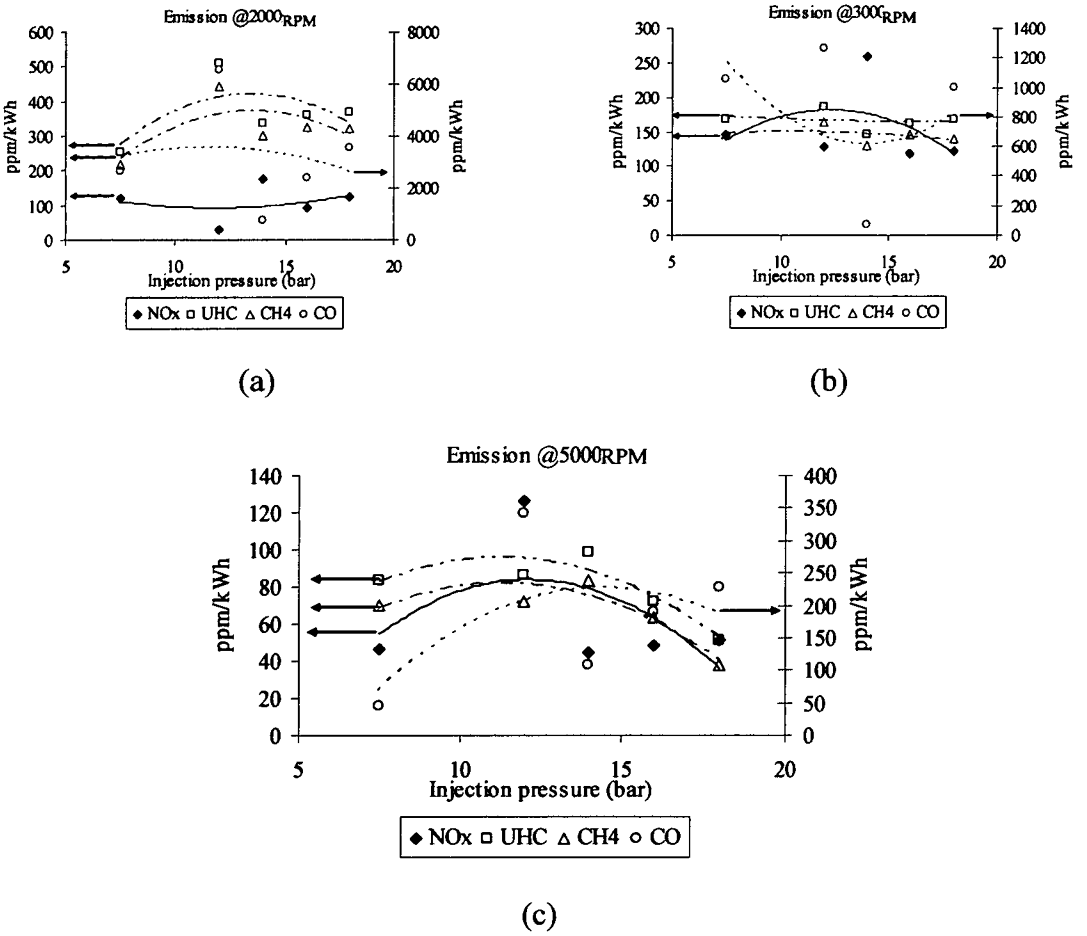


Figure 6-22 Emission characteristics for different injection pressure at various RPM

6.2.3 Engine Combustion

Combustion analysis was carried on two RPM, 3000 RPM and 5000 RPM. Maximum performance of the engine was occurring at 3000 RPM. Deeper analysis at this RPM is carried on to understand the combustion process. Pressure reading will be analyzed to get a clear picture of what happened inside the combustion chamber.

Injection pressure variation produces small effect on the performance and emission of CNG-DI engine at 3000 RPM. It is also reflected on the pressure due to combustion for various injection pressure at 3000 RPM (Figure 6-23), where the pressure result for 7.5 bar injection pressure is 62 bar. The pressure for 12 bar and 18 bar injection pressure are 70 bar and 65 bar respectively.

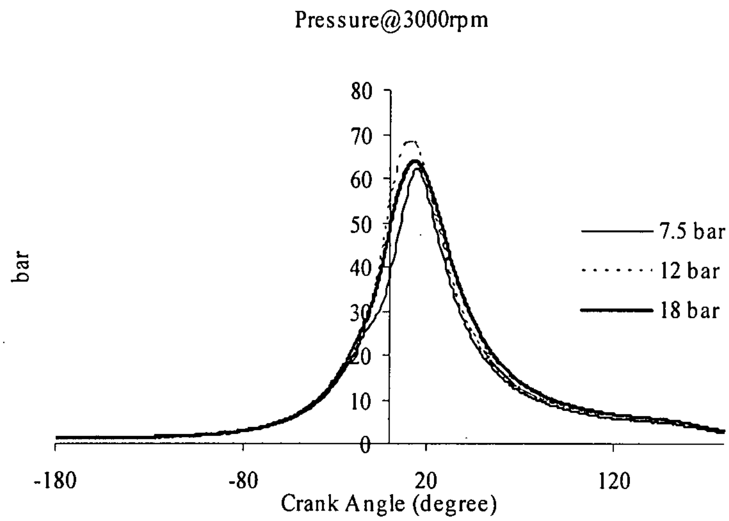


Figure 6-23 Pressure reading for different injection pressure at 3000 RPM

12 bar shows higher pressure due to better combustion process, which is also confirmed with NOx result where 12 bar injection pressure produces the highest NOx among all injection pressure. Furthermore, 12 bar good performance may be due to high volumetric efficiency and conducive stratification level in the mixture. Huang stated that charge stratification intensity in the mixing can promote the combustion process of CNG-DI engine. It is shown in Figure 6-24 that heat release rate for 12 bar injection pressure is faster than 18 bar and mostly from 7.5 bar injection pressure. Maximum peak for 12 bar injection pressure occur at 15ATDC while 18 bar and 7.5 bar injection pressure occur at 15ATDC and 18ATDC. Mass fraction burned (Figure 6-25) also describes the combustion rate of the system, in which 12 bar injection pressure shows faster initial combustion stage.

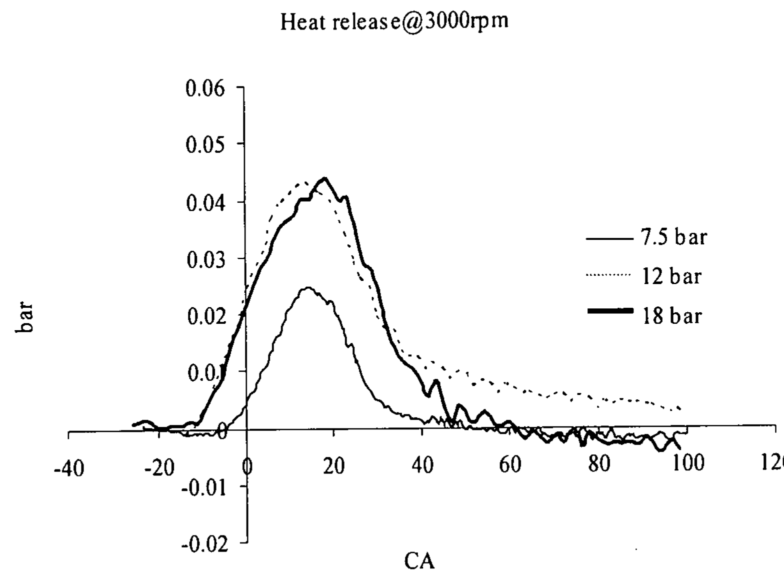


Figure 6-24 Heat release rate for different injection pressure at 3000 RPM

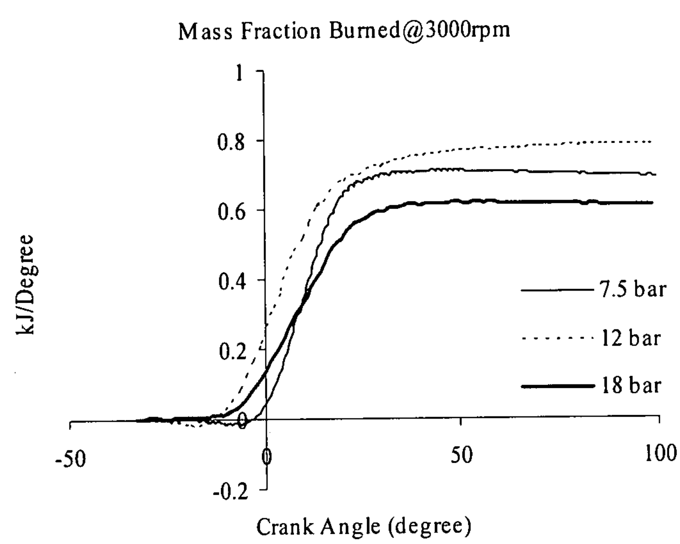


Figure 6-25 Mass fraction burned for different injection pressure at 3000 RPM

7.5 bar injection pressure exhibits curvature pressure line at the early stage of combustion (Figure 6-23), where the curve appears after the ignition point that may happen due to long ignition delay of the combustion process. Its long delayed combustion process resulted low pressure resulting from the combustion that was clearly shown in heat release rate (Figure 6-24). Heywood stated that 10-20 degree ATDC is the best crank angle, which indicates that maximum pressure should take place. But due to delayed combustion start, the pressure result for 7.5 bar injection pressure can not reach its best pressure peak. This delay may be due to homogeneous around the spark at the time of ignition point which is unfavorable to combustion process. Its homogeneous mixture is also verified by the gradient of the main combustion process (10-90%) of 7.5 bar injection pressure that is higher than other injection pressure.

IMEP values is describes at Figure 6-26, it shows that 18 bar injection pressure resulted higher IMEP on wide engine speed range from 2000 – 4500 RPM. This indicates better combustion occur. Higher turbulence intensity as the results of higher injection velocity may be the reason for this behavior.

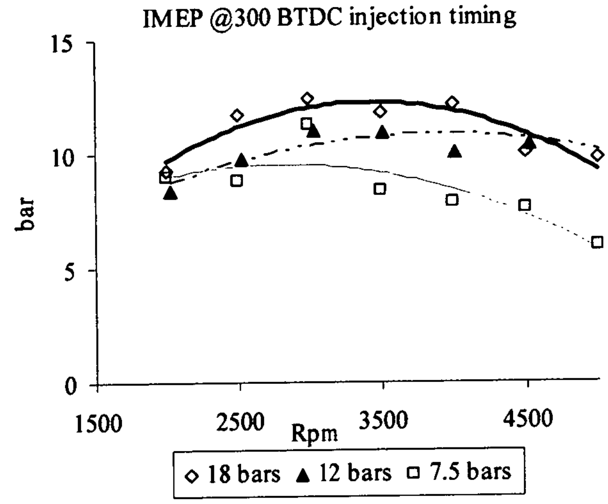


Figure 6-26 IMEP characteristics for different injection pressure.

COV characteristics for each engine speed was described on Figure 6-27 where 7.5 bar has the lowest COV compared to other injection pressure due to its homogeneous mixture while 12 bar at 2000RPM shows higher COV compared to 18 bar injection pressure. COV at 3000RPM illustrates that there is less difference between 12 -18 bar injection pressure which is 0.2. But at higher engine speed, lower injection pressure shows lower COV values than 18 bar injection pressure.

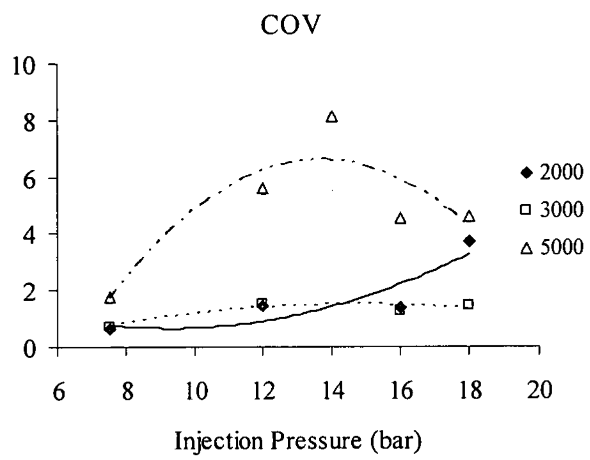


Figure 6-27 COV values for different injection pressure at various engine speed

On the other hand, 18 bar injection pressure shows opposing trends as 7.5 bar injection pressure. It has shorter ignition delay but lower slope for the main stage of the combustion process which may be due to higher stratification intensity as stated by Huang et al. Due to longer period of the combustion main stage, the heat release and mass fraction burned of 18 bar injection pressure are lower than 12 bar injection pressure.

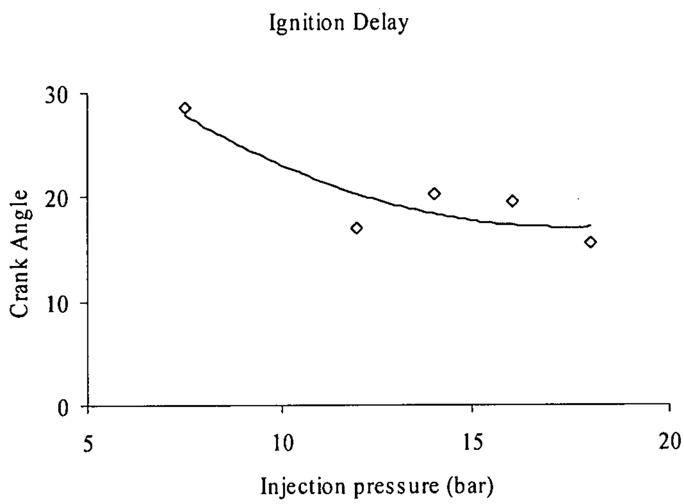


Figure 6-28 Ignition delay for different injection pressure at 3000 RPM

Specific illustration on the duration of each combustion stages for different injection pressure at 3000 RPM was shown in Figure 6-29. The trends for the duration of initial combustion process which is highly affected by charge stratification level around the spark at ignition time had demonstrated increasing trends as the injection pressure increased. It means that the charge stratification intensity is lower as the injection pressure increases due to higher turbulence intensity.

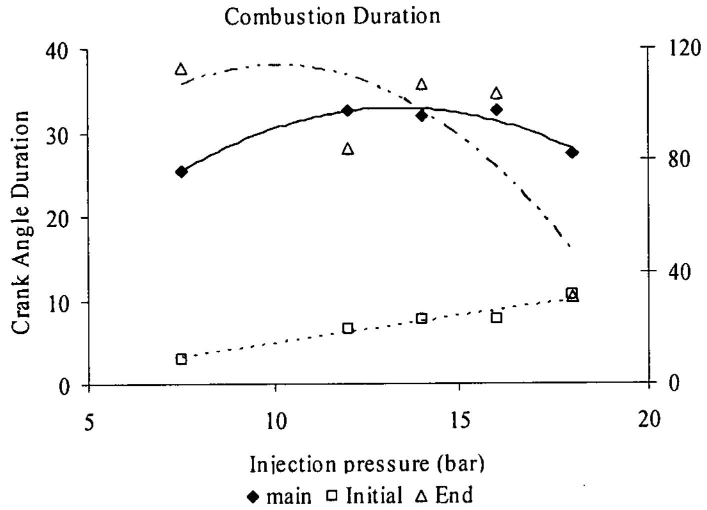


Figure 6-29 Combustion stages for different injection pressure at 3000 RPM

Combustion duration at the main combustion stage has tendency to increase even though the difference is small. Main stage is mainly affected by the turbulence intensity of the mixture. Slight difference shows that the spray-induce flow at high RPM has little effect.

For the final stage, the curve tends to decline. Final stage of the combustion process mostly is affected by the fuel availability in the mixture. Appointed to fast duration of initial and main stage of the combustion process, higher injection pressure shows faster duration at the final combustion stage.

Due to increase in charge stratification intensity, the combustion efficiency is decreasing as the injection pressure increases (Figure 6-30). 0.8 which is maximum value for combustion efficiency occurs at 12 bar injection pressure and the minimum value occurs at 18 bar injection pressure which is 0.615. This shows that homogeneous mixture is required to get better performance especially at high load and high RPM condition, but with little amount of stratification could enhance the combustion process and reduce harmful emission.

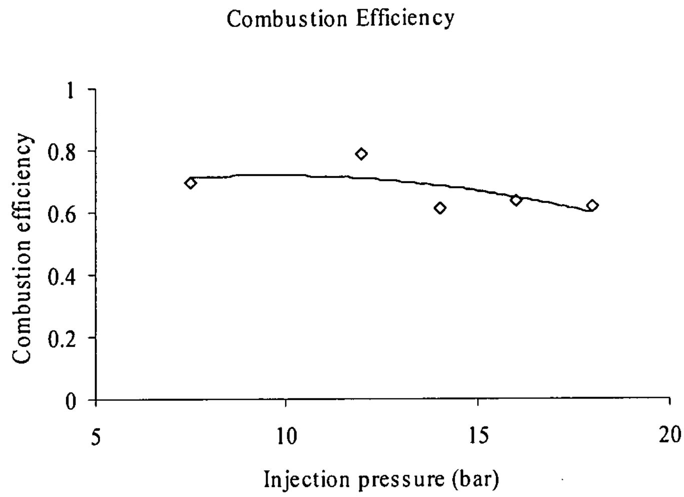


Figure 6-30 Combustion efficiency for different injection pressure at 3000 RPM

Next, Combustion analysis at 5000 RPM will be discussed. The performance drop of CNG-DI engine will be analyze to examine the parameters that affect the performance of the engine, in this case injection pressure.

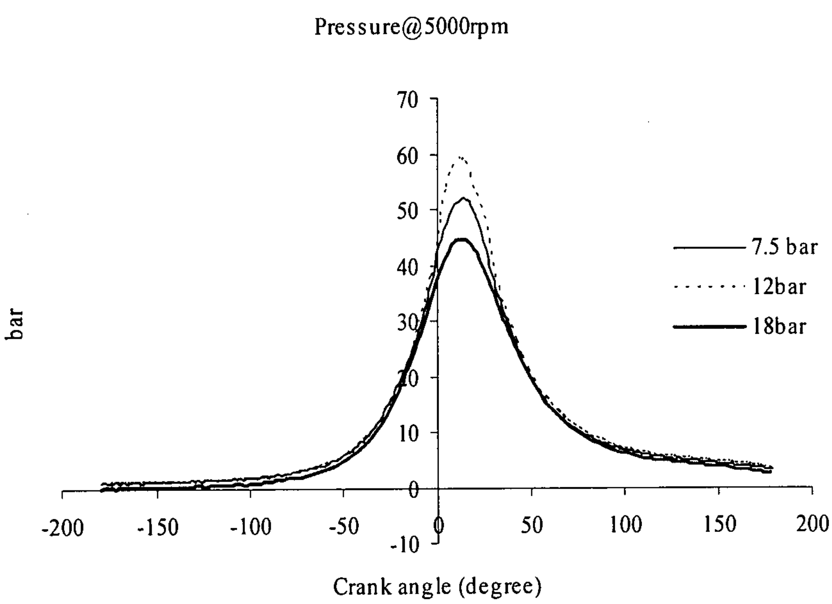


Figure 6-31 Pressure reading for different injection pressure at 5000 RPM

The experiment was done with 300°BTDC injection timing due to applicability for every injection pressure. Delivered from Figure 6-31, Significant effect can realize from pressure record at 5000 RPM, where 12 bar injection pressure exhibits the highest injection pressure among all injection timing with 60 bar. Consistent volumetric efficiency of 12 bar injection pressure is probably the reason for high pressure resulting from combustion process. Heat release rate (Figure 6-32) also explains that 12 bar injection pressure has faster main burn duration although the mass fraction burned Figure 6-33 for 12 bar injection pressure is lower than 7.5bar injection pressure.

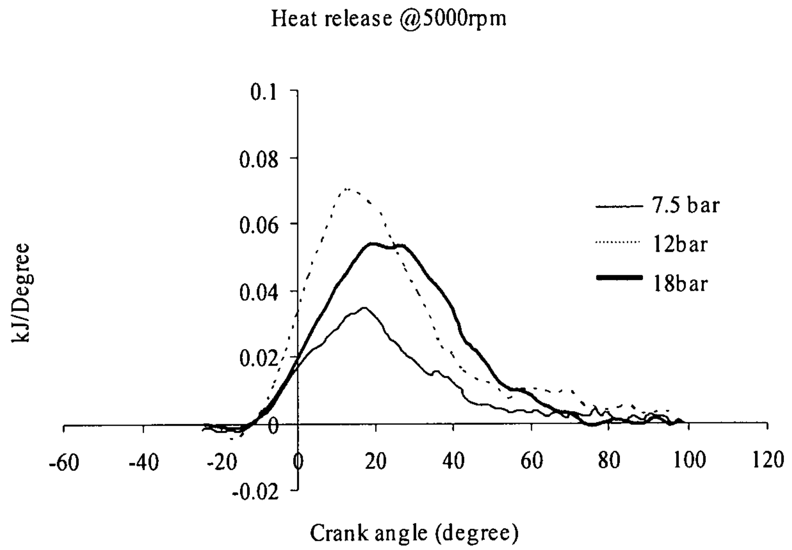


Figure 6-32 Heat release rate for different injection pressure at 5000 RPM

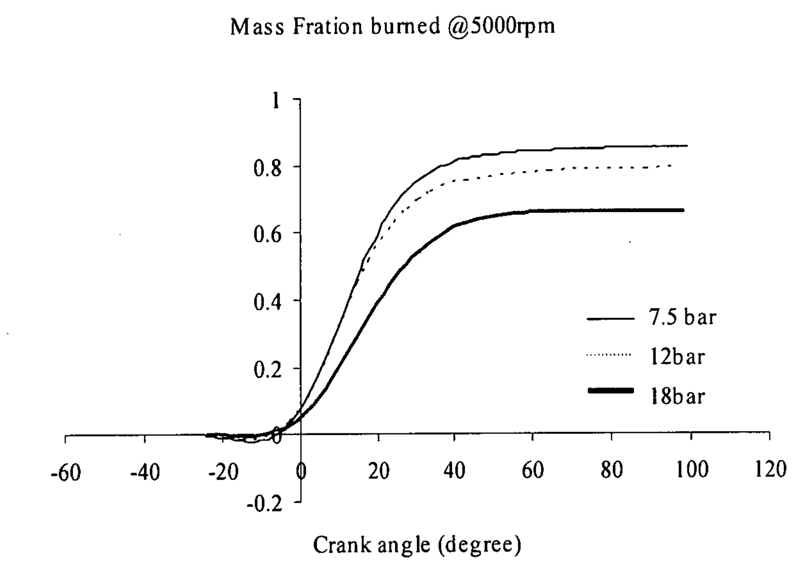


Figure 6-33 Mass fraction burned for different injection pressure at 5000 RPM

Heat release rate of 12 bar injection pressure shows that it has the highest heat release rate compared to other injection pressure due to combination of better mixing and small charge stratification that can enhance the combustion process. Mixture stratification could fasten the initial combustion, which was shown in Figure 6-29 It is depicted that 7.5 bar and 12 bar injection pressure have faster initial combustion period compared to 18bar injection pressure.

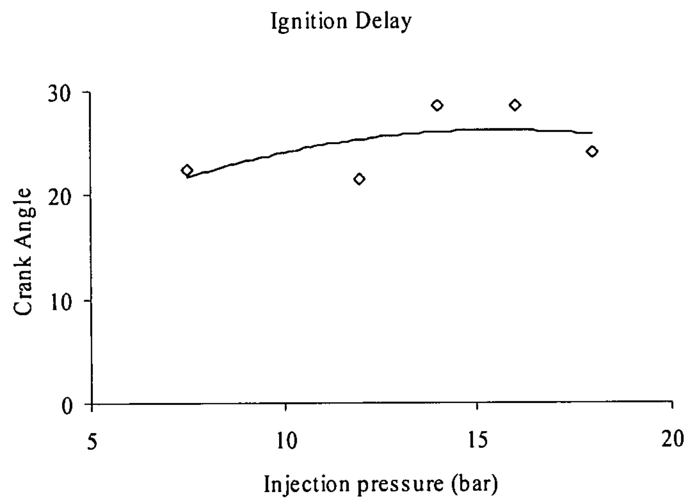


Figure 6-34 Ignition delay for different injection pressure at 5000 RPM

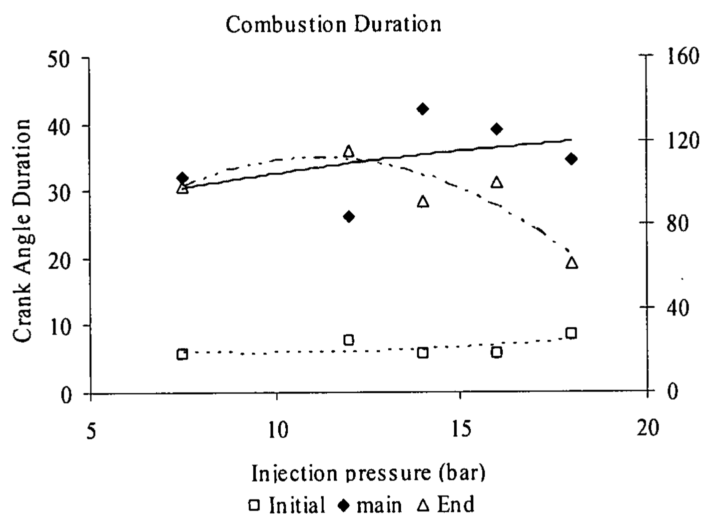


Figure 6-35 Combustion duration for different injection pressure at 5000 RPM

Moreover, the ignition delay period for various injection pressures was illustrated in Figure 6-34 where 7.5bar and 12 bar injection pressure have shorter delay period compared to 18bar injection pressure at 5000 RPM. Favorable condition for ignition which is fuel existence around the spark at the time of ignition may be created by intermediate injection velocity of lower injection pressure.

Main burned duration for 7.5 bar and 12 bar injection pressure as shown in Figure 6-35 is longer than 18bar injection pressure. This is probably caused by high turbulence intensity that 18bar injection pressure occupied due to high injection velocity that could enhance the combustion process of 18bar injection pressure but also could cause instability in the combustion process shown by high COV at 5000RPM with the employment of 18bar injection pressure.

Figure 6-36 illustrated combustion efficiency for different injection pressure at 5000RPM, where it shows that combustion efficiency is decreasing as the engine speed increases. It has 0.85 for 7.5 bar injection pressure and decreases to 0.6 if 18 bar injection pressure is applied at 5000RPM.

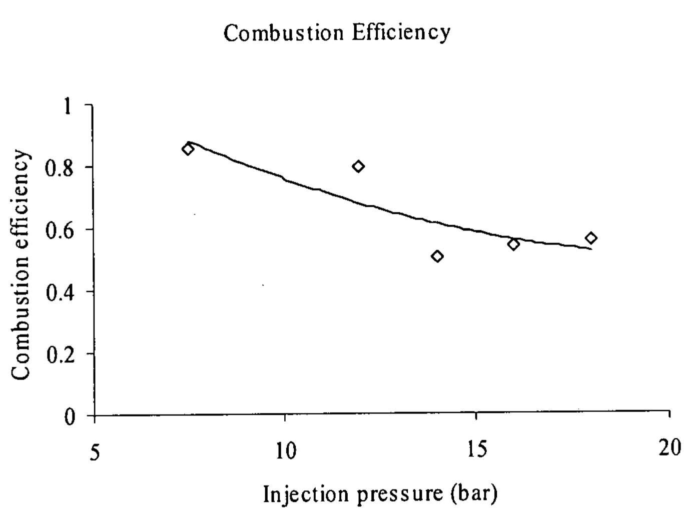
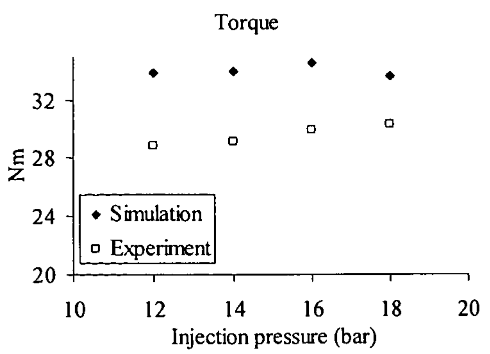
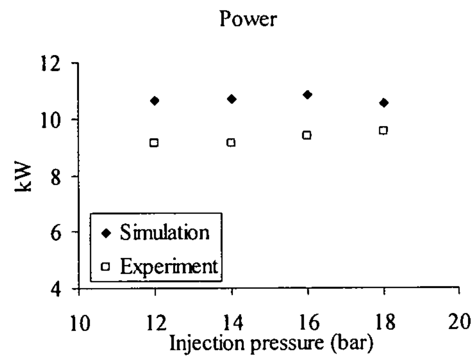


Figure 6-36 Combustion efficiency characteristics for different injection pressure at 5000 RPM

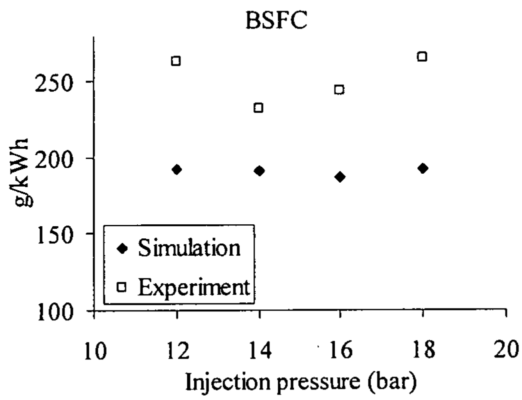
6.2.4 Experimental results comparison with simulation results



(a)



(b)



(c)

Figure 6-37 (a, b,c) Engine performance comparison

Figure 6-37 describes the comparison on the engine performance of the simulation and experimental results on injection pressure. It shows that in the simulation, the torque tends to increase as the injection pressure increase, while at the experimental results there is optimum injection pressure which is not the highest injection pressure. Limitation of the simulation which can not analyze further on flow interaction between the injection events and the intake air may be the reason of these differences. Furthermore, Figure 6-37 c on BSFC shows that the experimental results is higher compared to simulations. These gaps between both data may also due to assumption of complete combustion inside the combustion chamber.

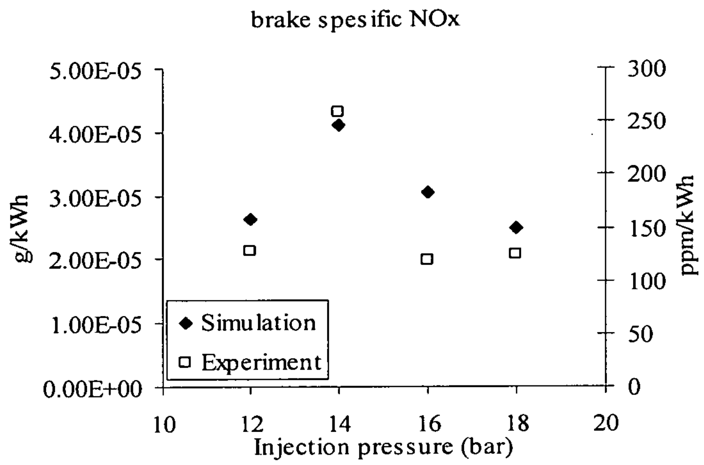


Figure 6-38 NOx emission comparison

NOx results from both experimental and simulation results shows the same trends which has the highest point at 14 bar injection pressure. Which shows better combustion occur during the combustion process as illustrated in Figure 6-38.

Summary

The effect of various injection pressures on the performance of CNG-DI engine shows that there are optimum injection pressures for every engine condition. It is shown that 18 bar injection pressure perform better performance on the speed below than 3500 RPM while 12 bar injection pressure shows good performance at the engine speed more than 3500 RPM. This behavior is because of the volumetric efficiency effect that gives same results as the engine performance.

The emission resulted for these three types injection pressure shows contradict behavior between each other as the engine speed is increasing. 18 bar injection pressure shows decreasing NOx emission as the speed increase while 12 bar injection pressure NOx emissions is increasing which is agreed with the performance results. CO emission for 18 bar injection pressure shows increasing trends while 12 bar injection pressure CO emission is decrease as the engine speed increase.

On the combustion process, IMEP values for 18 bar injection pressure is the highest for all engine speed range, although 18 bar injection pressure has slower burning rate compared to 12 bar injection pressure. Further, 18 bar injection pressure shows high variation at lower speed and decreases as the engine speed increase.

6.3 Effect of Injector Spray Angle to The Engine Performance of CNG-DI Engine

Injector Spray angle variation that consists of two types of injector angle, Narrow Angle Injector (NAI) and Wide Angle Injector (WAI), has its main purpose to improve mixing rate between air and fuel mixture. Figure 5-9 represents the image of injector angle at atmospheric condition.

In direct injection engine, injections properties are significant in generating turbulence in the combustion chamber. Cone-angle and penetration length are most common injector properties. Wide cone-angle injector and narrow cone-angle injector used in this experiment could help the understanding of the effect of different injector cone-angle to the natural gas engine performance. Naturally, increasing angle of injection spray could improve the mixing rate of the mixture with compensation lower velocity due to larger distribution area for injected gas. Injection sequence for both injectors is depicted in Figure 5-8 respectively.

Injection image was acquired by using schilieren method which is distinguishing the density difference between the injected gas and surrounding air. The image was recorded at the speed of 0.125μS per frame where injection duration used is 0.750 mS. This experiment was carried on atmospheric condition with controlled environment to prevent any interference from air flow of the surrounding to the injection process. After the injection stops at the time of 0.75 mS, we could notice that at 1.25 mS the intensity of injected gas for WAI is lesser than NAI. This behavior can be assumed that the gas is already mixed with surrounding air. At the time of 1.625 mS, the WAI injector shows very small intensity while NAI still shows residue from previous injection process. Based on these injection images, one can conclude that WAI injector mixing rate is faster than NAI injector.

6.3.1 Result of Engine Performance

Mechanical performance for early injection timing can be seen on Figure 6-39. It is shown that at Figure 6-39.a, torque output for NAI at speed below 3000 RPM is higher compared to WAI. In contrary, at higher speed WAI shows higher performance. Spray induced flow would probably be the reason to this behavior. NAI has higher influence to the in-cylinder flow. Turbulence generated by injection with NAI is higher than WAI. On the contrary, in the extreme condition, full load and high RPM, time constraint is important. Even though volumetric efficiency of WAI injectors is lower than NAI, WAI shows better performance due to its characteristics. Higher spray angle that it has gives WAI higher capability to mix fuel and air within a short time.

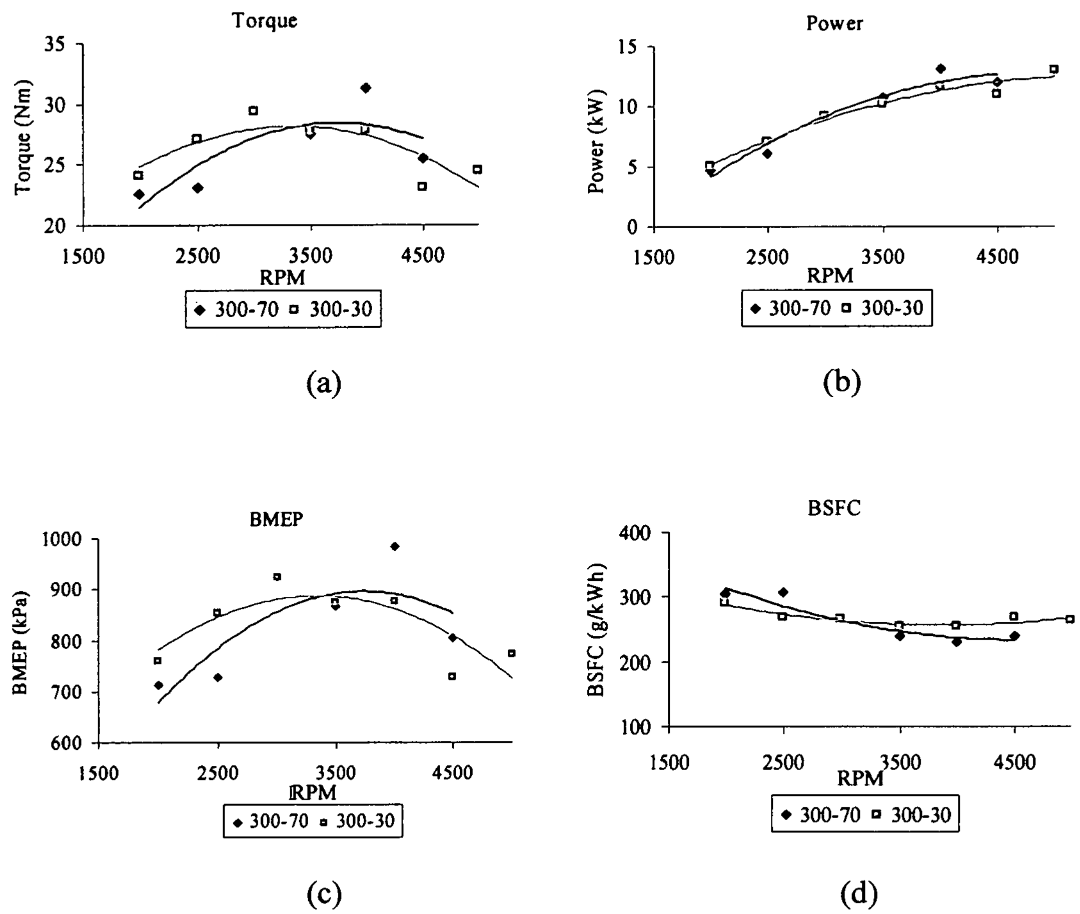


Figure 6-39 Performance at 300°BTDC Injection timing for different Spray angle injectors

It was also seen in Figure 6-39.b and Figure 6-39.c, power and BMEP have the same characteristics with torque. Maximum torque for NAI is 13 kw and WAI is 13.2 kw at 5000 RPM. BMEP shows maximum value at 3500 RPM for both injectors. BSFC (Figure 6-39.d) shows that WAI has better value at RPM less than 3000 RPM. And NAI is better at higher speed. And both injectors have the same performance at 3000 RPM for BSFC. The volumetric efficiency values for both injectors at 300°BTDC is shown in Figure 6-40, where NAI have higher volumetric efficiency for all range of engine speed.

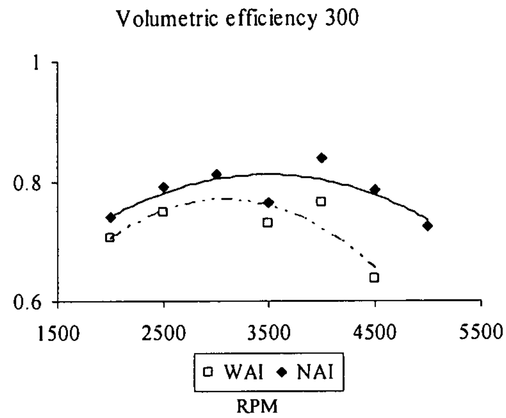
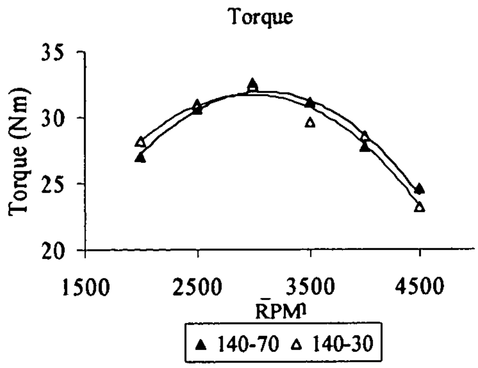


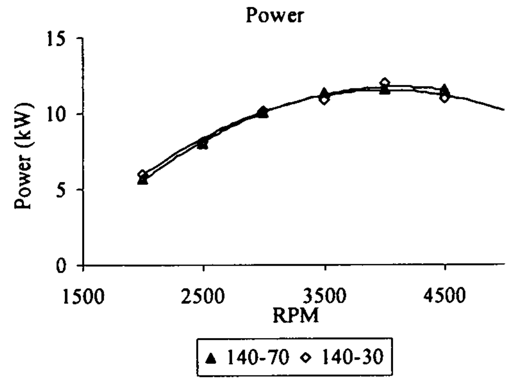
Figure 6-40 Volumetric efficiency for different injector at 300 injection timing

Figure 6-41 elaborates mechanical performance for both injectors at late injection timing (140 °BTDC). Torque in Figure 6-41.a shows that both injectors have slight difference at wide range of speed even though it can be seen that there are the same characteristics as very early injection timing is shown.

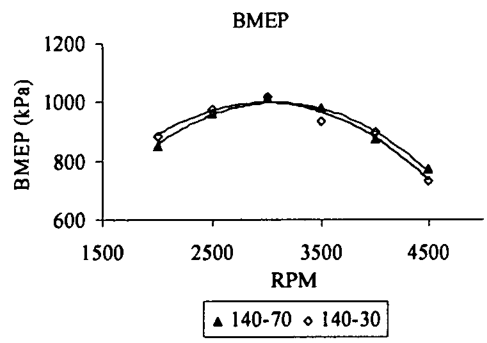
WAI has better performance at higher speed while NAI performs better at lower speed. Maximum torque for both injectors occurs on 3000 RPM, 32.7 for WAI and 32.4 for NAI injector. Power produced by both injectors is shown in Figure 6-41.b. The difference of power is not significant for wide range of speed. And it is 1-6 % range. It is also shown that maximum power for both injectors is at 4000 RPM. Higher speed shows significant drop of power. Closed interconnection of BMEP with torque makes BMEP have the same behavior with torque (Figure 6-41.c). Furthermore, BSFC values are shown in Figure 6-41.d. WAI has higher BSFC at late injection timing for every speed. The biggest difference is at 3000 RPM on 18% . It means that NAI has lower fuel consumption than WAI injector at late injection timing that can be attributed to higher volumetric efficiency of NAI at high speed.



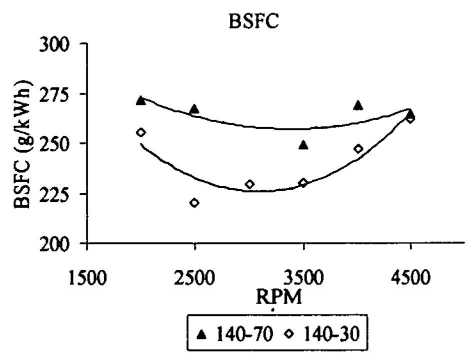
(a)



(b)



(c)



(d)

Figure 6-41 Performance at 140°BTDC injection timing for different Spray angle injectors

Figure 6-42 depicted the engine volumetric efficiency for both injectors at 140°BTDC injection timing. It is shown that lower engine speed WAI injectors has 0.85-0.97 efficiency while NAI 0.78 to 0.85. Note that the intake valve is closed at 132°BTDC, so that small portion of fuel was injected when intake valve is open which affecting volumetric efficiency of the engine. The trends shows that WAI angle gives small disturbance to intake process due to the fuel was distributed within larger area compared to NAI injection process.

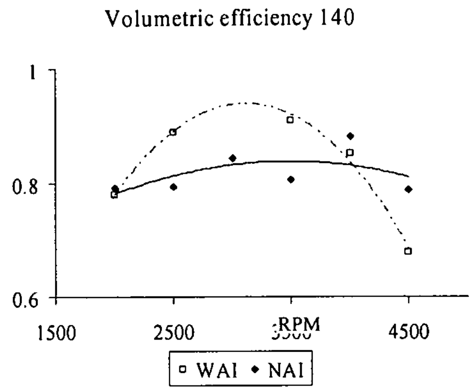


Figure 6-42 Volumetric efficiency for different injector at 140 injection timing

6.3.2 Result of Engine Emission

Emission is one of important parameter in engine performance discussion. There are three major exhaust gas emissions the author will be focusing, total unburned hydrocarbon (THC), nitric oxide (NOx), and carbon monoxide (CO). The emission result for early and late injection timing for both injectors will be presented.

Combustion products for these types of injectors are shown in Figure 6-43. Hydrocarbon emission levels decreases as the speed increases. It means that combustion with CNG fuel is getting better as speed is increasing. The lowest level of THC is at 4500 RPM which is 65 ppm/kW. Mixture formation for early injection timing is considered to be at homogeneous state when ignition is timing. As the speed increases, the mixing window is getting smaller, local stratification will occur in the chamber. When local stratification of the mixture around the spark is good, it can enhance the combustion process. In conclusion, it was probably due to local stratification of fuel around the spark. At the higher speed combustion of CNG is better. The same behavior was also shown by methane emission. Because of methane is the major composition of CNG, methane emission result can also indicate the combustion efficiency of CNG. It is shown that combustion efficiency increases as methane decreases. And the speed increases until its optimum speed at 4000 RPM. In this case, injector types have little effect on hydrocarbon emission of CNG-DI engine.

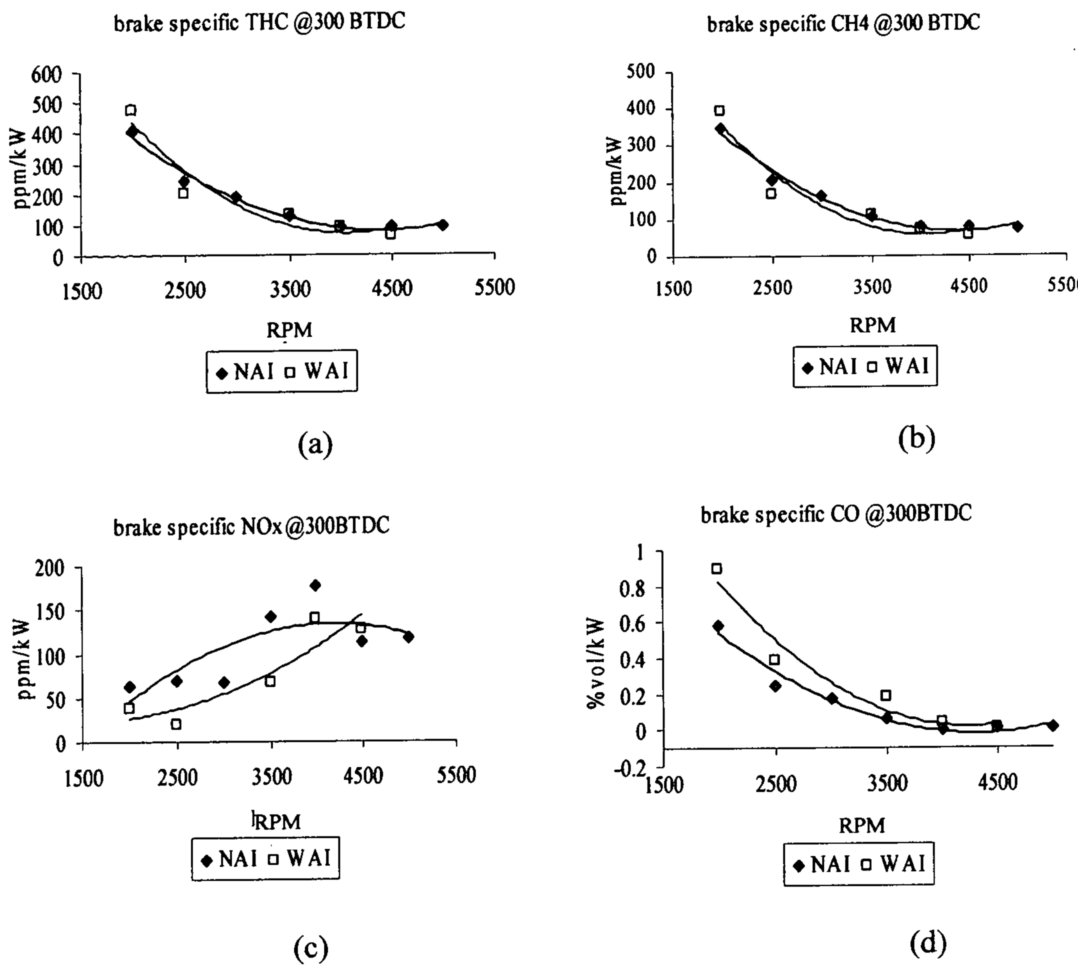


Figure 6-43 Emission at 300°BTDC injection timing for different injectors

Higher NOx emission resulted from NAI compared to WAI at speed lower than 4000 RPM. And at speed more than 4000 RPM, WAI shows same NOx trends with NAI injector. The highest NOx emission is achieved at 4000 RPM with NAI injector with concentration of 176 ppm/kW. NOx emission occurs where high temperature condition happens in the combustion chamber. Higher temperature also means better combustion happens at this rate. As a consequence, NOx emission is higher. This phenomenon also agreed with the trends of hydrocarbon emission. While CO (Figure 6-43.d) shows completeness of combustion, high CO concentration means that the combustion is incomplete. WAI shows higher CO emission compared to NAI for all speed, 0.3 %volume/kW for WAI and 0.15 %volume/kW for NAI. More homogeneous mixture achieved by WAI may be the reason of this matter. Commonly, methane has lower flame speed at near stoichiometric condition. At this condition, the turbulence level of the mixture is determined because NAI may have higher turbulence generated, so the combustion time for NAI is shorter compared to WAI.

Figure 6-44 presents the emission outcome for both injectors at late injection timing. THC emission was depicted in Figure 6-44.a. WAI had high concentration of THC for speed below 3000 RPM, 40 % at 2000 RPM. And, it has somewhat same level at higher speed which is 1.5% at 3500 RPM. High turbulence level for NAI may be also the reason for better combustion with NAI injectors. This result can also be seen on methane emission in Figure 6-44.b, where at lower speed WAI has higher methane emission compared to NAI and almost equal at higher speed.

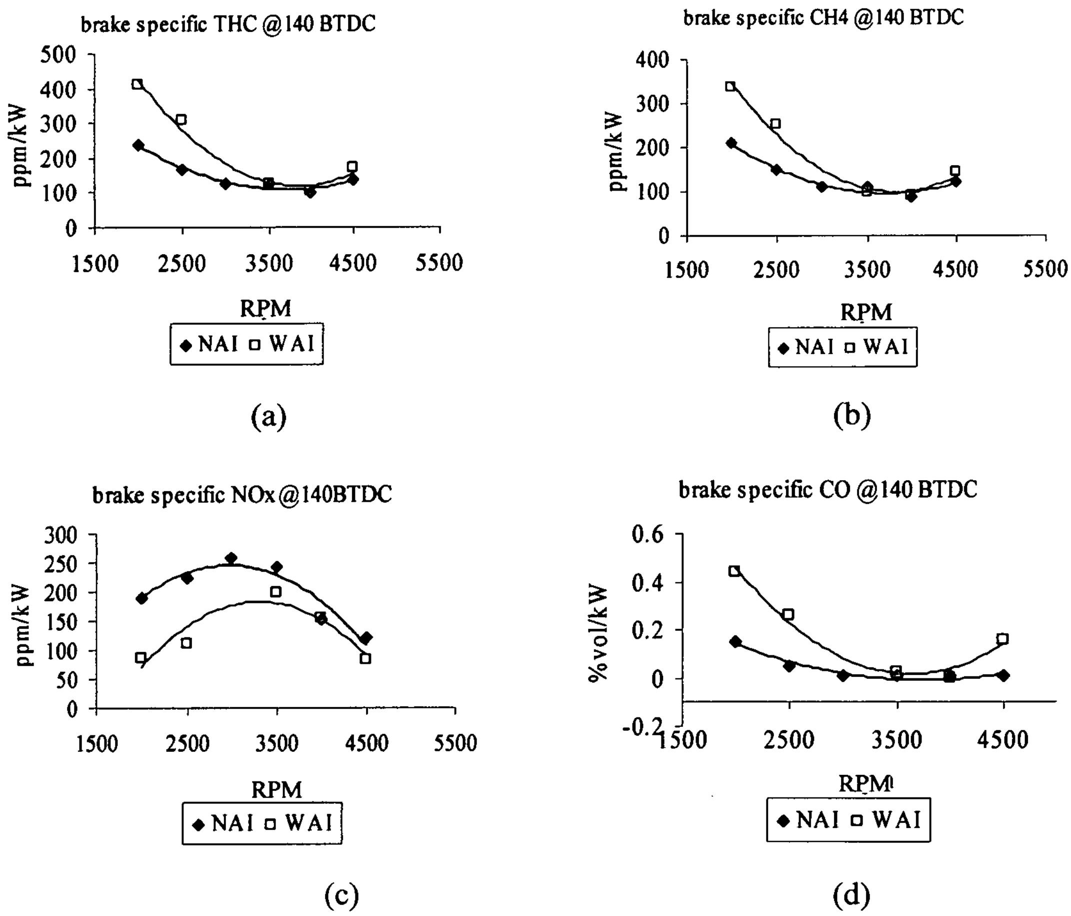


Figure 6-44 Emission at 140 °BTDC injection timing for different injectors

Nitric oxide emission result is portrayed in Figure 6-44.c. It shows that NAI injector has higher NOx emission for all speed compared to WAI. The difference of NOx emission is significant between both injectors, 35.3%. Where, the average for NAI is 198.89 ppm/kW while WAI has 128.64 ppm/kW in average.

Figure 6-44.d represents CO emission from both injectors for 140 °BTDC injection timing. CO emission result had similar trend as hydrocarbon emission. WAI has higher CO emission at lower speed and almost the same with NAI at higher speed operation. It is also shown that WAI combustion is improving as the speed increases.

6.3.3 Result of Engine Combustion

The analysis of the combustion process using pressure reading from the engine will be carried out. The heat release rate will be calculated by using Rassweiler and Withrow method. The combustion analysis will be concentrated at 140°BTDC injection timing for 3000 RPM, and 300 °BTDC injection timing for 5000 RPM speed. The limitation of injection window made at high speed the injection timing needs to be advanced. And, retarding injection timing at high speed can cause unstable combustion.

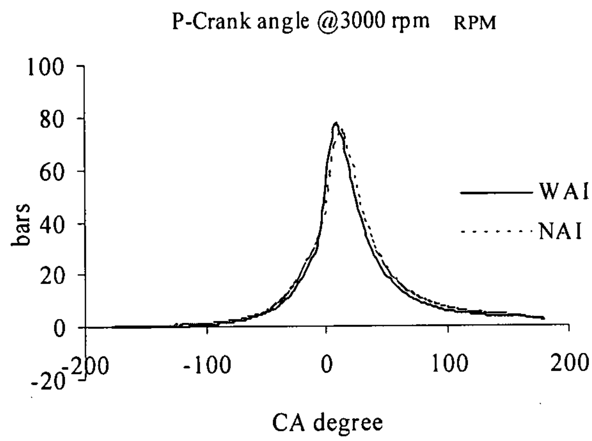


Figure 6-45 Pressure-Crank angle diagram at 3000 RPM with 140 °BTDC injection timing.

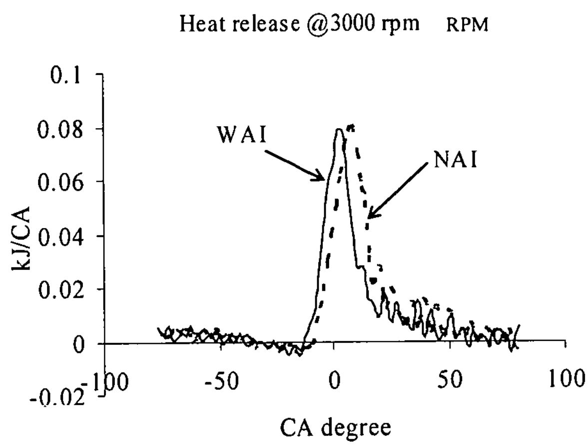


Figure 6-46 Heat release rate diagram at 3000 RPM with 140 °BTDC injection timing.

Only a small difference can be realized from pressure resulting from combustion for both injectors, which is 77.4 bars for WAI and 74.5 bars for NAI injector (Figure 6-45). Furthermore, the locations of the peak pressure are different, where for WAI the peak pressure occur at 9.5°BTDC and 12°BTDC for NAI. It shows that peak pressure for WAI occurs earlier than NAI due to faster combustion duration of WAI. Nevertheless NAI has higher maximum heat release (Figure 6-46). The rate heat release difference for both is 6° where WAI is earlier than NAI. NAI has higher values, 0.085 kJ/CA. And the difference from WAI in 1.7% . higher value of heat can affect emission output especially NOx. This behavior has explained NOx emission which is higher for NAI at 140°BTDC for 3000 RPM compared to WAI injector.

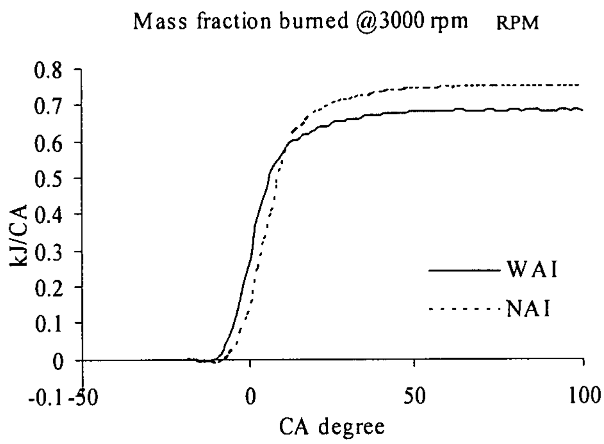


Figure 6-47 Mass fraction burned for different injector spray angle at 3000 RPM

Faster combustion duration for WAI injector (Figure 6-47) owing to degree could lead to lowering the combustion efficiency that is confirmed with CO emission. WAI has higher CO emission in respect to NAI injector. Furthermore, increasing performance of WAI at high speed could also be explained with faster combustion rate of WAI.

Pressure reading at 5000 RPM with 300°BTDC injection timing is illustrated in Figure 6-48. There are noticeable differences that can be seen in term of maximum pressure where WAI had higher pressure compared to NAI. WAI has 50.6 bars and NAI has 44.6 bars.

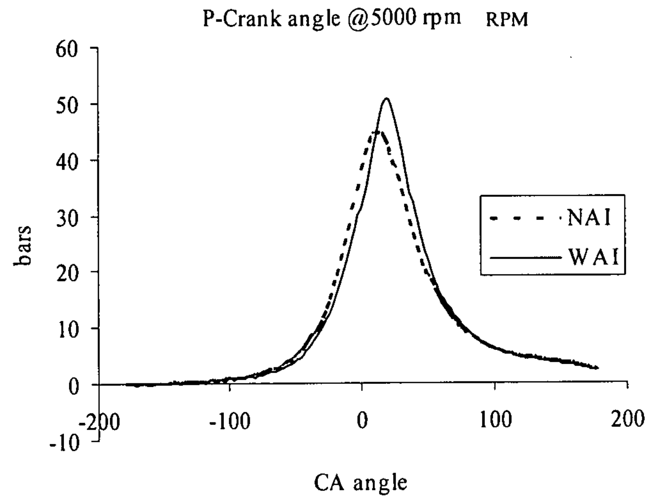


Figure 6-48 Pressure-Crank angle diagram at 5000 RPM with 300 injection timing.

In Figure 6-49. heat release rate is represented. It was clearly shown that WAI had higher heat released compared to NAI, which WAI got 0.099 kj/CA and NAI had 0.058 kJ/CA. The difference is almost 50%. This has shown that WAI is suitable for high-speed application.

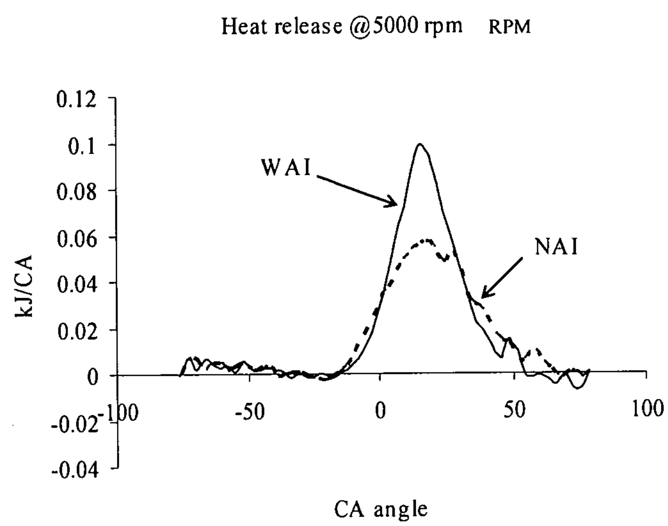


Figure 6-49 Heat release rate diagram at 5000 RPM with 300 injection timing.

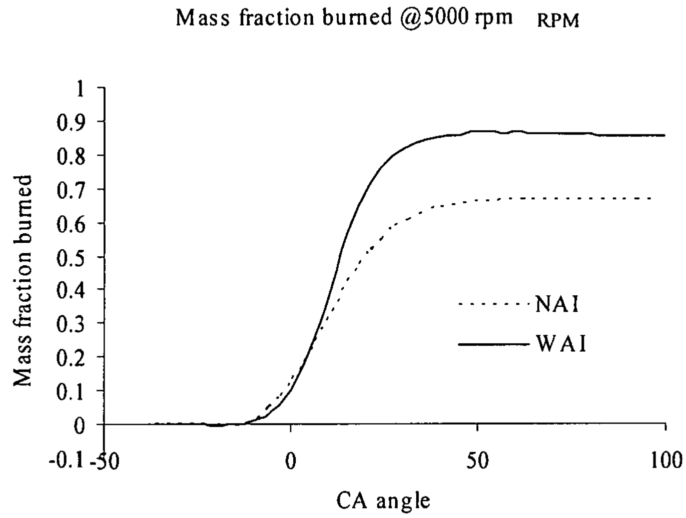


Figure 6-50 Mass fraction burned for different injector spray angle at 5000 RPM with 300 injection timing

Figure 6-50 illustrate the mass fraction burned graph where the WAI has higher mass fraction burned and longer initial combustion. These indicates the mixture of WAI is more homogeny compared to NAI at 5000 RPM

Summary

At early injection timing, the power output is significant. NAI has 10% higher at 2000 RPM compared to WAI. In contrast, WAI had higher 4.5% compared to NAI at high RPM (5000 RPM). Emission for both injector at early injection timing is slightly different. For late injection timing, the power outputs were almost the same. The difference is less than 5%. Except for BSFC, WAI had 7.95% in average of all speed. Late injection timing emission result shows that for THC, methane, and CO, WAI had higher value compared to NAI. 40% for THC, 22.5% for CO and 35.3% for NOx at 2000 RPM. In higher speed, emission difference is small, which is only 1.5%. Combustion behavior of both injectors were slightly different at 3000 RPM which is only 1.7%. Furthermore, in high RPM (5000 RPM), WAI shows higher heat release and maximum pressure.

It can be concluded that NAI injector is suitable for low speed application that needs bigger mixing window, and WAI injector is suitable for high speed operation due to its higher mixing capability.

Overall summary of chapter VI.

Table 6-1 summarizes the effect of injection parameters to the performance of CNG-DI engine.

Table 6-1 (a, b, c) Summary of effect of injection parameters to CNG-DI engine performance

300 BTDC as baseline				
	Low Engine speed (2000-2500 RPM)	Medium engine speed (2500-3500 RPM)	High engine speed (3500-5000 RPM)	
Torque	+ 8 to 14%	+ 8.6 % to 11.2%	+ 3% to - 3%	180 BTDC
NOx	2 - 3 times	+ 80 % to 30 %	+ 48 to 1.5 %	
HC	- 37 % to 34 %	- 31% to +7%	+ 22 % to 37 %	
Max. pressure	+ 5 to 11%	+ 7.6 % to 13.6%	+ 1% to - 5%	
Torque	+ 13.6 to 14.8 %	+ 7.5 to 3.4 %	-	120 BTDC
NOx	4 - 5 times	+ 10 to -1 %	-	
HC	- 41% to 30%	- 27 % to -1 %	-	
Max. pressure	+ 11.6 to 17.6 %	+ 5 to 8 %	-	

(a)

7.5 bar injection pressure as baseline				
	Low Engine speed (2000-2500 RPM)	Medium engine speed (2500-3500 RPM)	High engine speed (3500-5000 RPM)	
Torque	+ 13.6 to 14.8 %	+ 7.5 to 3.4 %	-	18 bar
NOx	2 - 3 times	+ 80 % to 30 %	+ 48 to 1.5 %	
HC	- 37 % to 34 %	- 31% to +7%	+ 22 % to 37 %	
Max. pressure	+ 8.7 to 12.8 %	+ 6.5 to 2.4 %	-	
Torque	+ 13.6 to 14.8 %	+ 7.5 to 3.4 %	-	12 bar
NOx	+ 4 to 5 times	+ 10 to -1 %	-	
HC	- 41% to 30%	- 27 % to -1 %	-	
Max. pressure	+ 14.6 to 16.8 %	+ 6.5 to 2.4 %	-	

(b)

Narrow angle injector as a baseline				
	Low Engine speed (2000-2500 RPM)	Medium engine speed (2500-3500 RPM)	High engine speed (3500-5000 RPM)	
Torque	+ 0 to 12 %	+ 14.8 % to 21.5 %	- 4 % to +1 %	WAI at 300 BTDC
NOx	- 40 % to 66 %	+ 1 % to 4 %	+97 % to 117%	
HC	- 12 % to 62 %	- 62 % to 73 %	- 82 % to 88 %	
Max. pressure	- 1 to + 10 %	+ 8.8 % to 15.5 %	- 6 % to +2 %	
Torque	+ 0 to 10 %	+ 1.4 % to 5.3 %	- 17.3 % to 31.2 %	WAI at 140 BTDC
NOx	+ 4 to 5 times	+ 5 to 6 times	+ 87 % to 137 %	
HC	- 36 % to 56 %	- 72.8 % to 78 %	- 67 % to 78 %	
Max. pressure	- 1 to 8 %	+ 1 % to 7 %	- 15.3 % to 35.2 %	

(c)

CHAPTER VII

CONCLUSIONS and RECOMMENDATION

7.1 CONCLUSIONS

The experimental works along with simulation processes had been carried out in order to obtain the optimum performance of CNG-DI engine. The effect of injection parameters such as injection timing, injection pressure and injector cone angle to the engine performance has been investigated. Conclusions on the effect of each injection parameter are as follows:

1. Early injection timing has better performance compare to late injection timing at high speed operation due to the fact that the penalty in lower volumetric efficiency is compensated with better air-fuel mixing
2. Late injection timing resulted high engine performance at low engine speed (2000-3000 RPM) due to higher volumetric efficiency.
3. Partial Direct injection (180°BTDC) shows high performance over wide range of engine speed
4. Faster combustion rate is achieved with 180°BTDC injection timing due to high volumetric efficiency and near homogeneous mixture especially at 3000 RPM
5. Reducing the injection pressure could increase the engine performance at high engine speed due to better air-fuel mixing is achieved.
6. High injection pressure (18bar) performs better performance at the engine speed less than 3000 RPM due higher kinetic energy to promote the mixing.
7. While, medium injection pressure (12bar) shows better performance at high engine speed due to better volumetric efficiency.
8. Each injector spray angle gives optimum performance at specific engine performance because of the mixture characteristics.
9. Narrow angle injector (NAI) performs better at low engine speed due to higher kinetic energy introduce to the combustion chamber.

10. While, wide angle injector (WAI) performs better at high engine speed due to faster mixing rate.
11. The simulation results show good agreement with experimental results with understandable difference due to assumption used in the simulation.

From the conclusions drawn above, the optimum engine performance for each engine speed can be delivered to get competitive performance of CNG-DI engine by adjusting the injection parameters. In summary, at higher engine RPM, early injection timing with lower injection pressure results in a higher engine power, while at lower and medium engine RPM, late injection timing with higher injection pressure results in a better performance.

7.2 RECOMMENDATIONS

Volumetric efficiency of the gaseous engines apparently gives significant effect to the engine output. Further, investigation using the turbocharger or supercharger is recommended on CNG-DI engine may further increase the power output of the engine.

Understanding the mixing process inside the combustion chamber is essential to control the combustion process of CNG-DI engine and increased the power output of the engine as well as to lower the exhaust emission emitted from the engines. Extended examinations on the interaction between injected fuel and in-cylinder flow needs to be carried out to get clear picture of controlling the mixture to control the combustion process. A flow visualization studies is recommended to understand the mixing process. While for fuel economy at idling and partial load, an investigation in the direction of stratified charge is the future recommendations to complete this current study of full-load WOT condition.

REFERENCES:

1. IANGV, *Pathways for Natural Gas Into Advance Vehicles*, G. Harris, Editor. 2002, IANGV.
2. IANGV, *IANGV reports*. 2000.
3. Mistry, C.S., *Comparative Assessment on Performance of Multi cylinder Engine Using CNG, LPG and Petrol as a Fuel.* SAE International, 2005(2005-01-1056).
4. Kato, K., et al., *Development of Engine for Natural Gas Vehicle.* SAE, 1999(1999-01-0574).
5. Pischinger, S., M. Umierski, and B. Huchtebrock, *New CNG Concepts for Passanger Cars: High Torque Engines with Superior Fuel Consumption.* SAE, 2003(2003-01-2264).
6. Achleitner, E., H. Backer, and A. Funaioli, *Direct Injection Systems for Otto Engines.* SAE, 2007(2007-01-1416).
7. Einewall, P. and O. Jönsson, *Ultra-clean natural gas engine with closed-loop lambda control, high turbulence combustion chamber, EGR and a three-way catalyst*, Lund University.
8. d'Ambrosio, S., E. Spessa, A. Vasallo, M. Ferrera, and C. Peletto, *Experimental Investigation of Fuel Consumption, Exhaust Emissions, and Heat release of a Small-Displacement Turbocharged CNG engine.* SAE International, 2006(2006-01-0049).
9. Freund, E., *The future of technologies to reduce CO2 emission of road transport*, in *Technology future presentation.* 2006: Paris.
10. Wang, M.Q. and H.-S. Huang, *A Full Fuel-Cycle Analysis of Energy and Emission Impacts of Transportation Fuels Produced From Natural Gas*, U.S.D.o. Energy, Editor. 1999, Centre for Transportation Research Argonne National Laboratory.
11. Bakar, R.A., *Design and Development of a New CNG (Compressed Natural Gas) Engine (Rekabentuk dan Pembangunan Enjin CNG Baru).* 2002, PUSAT PENGURUSAN PENYELIDIKAN UNIVERSITI TEKNOLOGI MALAYSIA.
12. Cho, H.M. and B.-Q. He, *Spark Ignition natural gas engines - A review.* Energy Conversion and Management, 2006.
13. Maji, S., R. Ranjan, and P.B. Sharma, *Comparison of Emission and Fuel Consumption from CNG and Gasoline Fueled Vehicles-Effect of Ignition Timing.* SAE International, 2000(2000-01-1432).
14. Maji, S., P.B. Sharma, and M.K.G. Babu, *A Comparative Study of Performance and Emission Characteristics and Gasoline on a SIngle Cylinder S.I. Engine.* 2004(2004-28-0038).
15. Corbo, P., M. Gambino, S. Iannaccone, and A. Unich, *Comparison Between Lean-Burn and Stoichiometric Technologies for CNG Heavy-Duty Engine.* SAE, 1995(950057).
16. Maji, S., P.B. Sharma, and M.K.G. Babu, *Experimental Investigations on Performance and Emission Characteristics of CNG in a Spark Ignition Engine.* SAE International, 2005(2005-26-344).

17. Huang, Z., et al. *Feasibility of CNG DI Stratified Combustion Using a Spark-Ignited Rapid Compression Machine.* in *The fifth International Symposium on Diagnostic and Modeling of Combustion in Internal Combustion Engines.* 2001. Nagoya: COMODIA.
18. IANGV. *Pathways for Natural Gas Into Advance Vehicles.* 2000: IANGV.
19. SIlveira, B.H.d., et al., *Development of a Concept Vehicle for Compressed Natural Gas.* SAE Brasil, 2004(2004-01-3452).
20. Majid, Z.A., Z. Yaacob, and M.P.K.I. Piau, *natural gas motorcycle-emission.* 2000, IANGV.
21. Durell, E., J. Allen, D. Law, and J. Heath, *Installation and development of direct injection system for bi-fuel gasoline and compressed natural gas engine.*
22. Kalam, M.A., H.H. Masjuki, M.A. Maleque, and M.A. Amalina, *Performance and Emission Characteristics of a retrofitted bi-Fuel Spark Ignition engine.*
23. Esfahanian, V., A.S. Zadeh, M.M.N. Azadani, and M. Mirsoheil. *Simulation and Comparison of the Performance of CNG and Gasoline Engines Using FlamePropagation Model Considering the Effects of Some of the Parameters of Combustion Chamber.* in *ANGVA International Confrence.* 2005. Malaysia.
24. www.iangv.com (1997) *Natural Gas Vehicle Industry IANGV Position paper.* **Volume,**
25. IANGV, *Natural Gas Vehicle Industry Position Paper 1997.* 1997.
26. Atkinson, C.M., M.L. Traver, C.J. Tennant, R.J. Atkinson, and N.N. Clark, *Exhaust Emissions and Combustion Stability in a Bi-Fuel Spark Ignition Engine.* SAE International, 1995(950458).
27. nylund, n., j. laurikko, and m. ikonen, *pathways for natural gas into advanced vehicles.* 2002, IANGV.
28. Suga, T., T. Muraishi, and R. Bienenfeld. *Potential of a low emission Natural gas Vehicle for the 21th Century.* in *NGV 2000.* 2000. Yokohama, Japan: IANGV.
29. Mintz, M.M., M.Q. Wang, and A.D. Vyas, *Fuel-Cycle Energy and Emissions Impacts of Propulsion SYstem/Fuel Alternatives for Tripled Fuel-Economy Vehicles.* SAE, 1999(1999-01-1118).
30. Mello, P., G. Pelliza, R. Cataluna, and R.d. Silva, *Evaluation of the maximum horsepower of vehicles converted for use with natural gas fuel.* Fuel, 2006. **85**(14-15): p. 2180-2186.
31. Loukanine, V.N., A.S. Khatchiyan, I.G. Shishlov, and R.K. Khamidullin. *Analysis of Different Ways to Develop Low-Emission Natural Gas Engines.* in *NGV 2000.* 2000. Yokohama, Japan: ANGVA.
32. Zhao, F., D.L. Harrington, and M.-C. Lai, *Automotive Gasoline Direct-Injection Engines.* 2002: SAE International.
33. Czerwinski, J., P. Comte, and Y. Zimmerli, *Investigations of the Gas Injection System on a HD-CNG-Engine.* SAE International, 2003(2003-01-0625).
34. Nagano, M., S. Watanabe, Y. Sukegawa, and K. Amou. *Port-injection Engine-control System for Environmental Protection.*
35. Stone, R., ed. *Introduction to Internal Combustion Engines.* 3rd ed. 1999, SAE International.
36. Bell, S.R., G.A. Loper, and M. Gupta. *Combustion characteristics of a natural gas fueled spark ignited engine.* 1993. Houston, TX, USA: Publ by ASME, New York, NY, USA.

37. Zhao, K., et al., *Study on combustion system of a spark ignition natural gas engine.* Tongji Daxue Xuebao/Journal of Tongji University, 1998. **26**(1): p. 1-8.
38. Shiga, S., S. Ozone, H.T.C. Machacon, T. Karasawa, and H. Nakamura, *A study of the combustion and emission characteristics of compressed –natural –gas direct-injection stratified combustion using a rapid compression machine.* Combustion and Flame, 2002. **129**: p. 1-10.
39. Arcoumanis, C., H. Flora, J.W. Kim, and H.M. Xu, *Injection natural gas engine for light duty applications.* IMEchE, 2001. **C588**(013).
40. www.iangv.org *Engine Technology.* **Volume**,
41. Hayashida, M., T. Yamato, H. Sekino, and K. Sugahara, *Investigation of Performance and Fuel Distribution of a Direct Injection Gas Engine Using LIF Measurement.* SAE, 1999(SAE 1999-01-3291,JSAE 9938046).
42. Catania, A.E., D. Misul, E. Spessa, and G. Martorana, *Conversion of a Multivalve Gasoline Engine to Run on CNG.* SAE International, 2000(2000-01-0673).
43. Catania, A.E., D. Misul, A. Mittica, and E. Spessa, *A refined two-zone heat release model for combustion analysis in SI engines.* JSME International Journal, Series B: Fluids and Thermal Engineering, 2003. **46**(1): p. 75-85.
44. Kim, J.M., S.B. Han, and R.R. Raine, *Combustion Stability of Natural Gas Engine Operating at Idle.* SAE, 2005(2005-01-3446).
45. Shiga, S., et al., *Basic Aspect of Combustion of CNG In-cylinder Direct-Injection with Spark-Ignition.* SAE International, 2005(2005-26-352).
46. Beroun, S. and J. Martins, *The Development of Gas (CNG, LPG and H2) Engines for Buses and Trucks and their Emission and Cycle Variability.* SAE, 2001(2001-01-0144).
47. Hassaneen, A.E., K.S. Varde, A.H. Bawady, and A.A. Morgan, *A Study of The Flame Development and Rapid Burn Durations In a Lean-Burn Fuel Injected Natural Gas S.I. Engine.* SAE, 1998(981384).
48. Kim, J., *Development of direct injection CNG engine.* NGV report, 2000.
49. Kim, C.U. and C.S. Bae, *Speciated hydrocarbon emissions from a gas-fuelled spark-ignition engine with various operating parameters.* Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 2000. **214**(7): p. 795-808.
50. Zuo, C. and K. Zhao. *Study on the combustion system of a spark ignition natural gas engine.* 1998. Dearborn, MI, USA: SAE, Warrendale, PA, USA.
51. Huang, Z., et al., *Basic Characteristics of direct injection combustion fuelled with compressed natural gas and gasoline using a rapid compression machine.* Proc. Instn Mech. Engrs. Part D: j. Automobile Engineering, 2003. **217**.
52. Huang, Z., et al., *Combustion characteristics of natural gas direct injection combustion under various fuel injection timings.* Proc. Instn Mech. Engrs. Part D: j. Automobile Engineering, 2003. **217**: p. 935-941.
53. Huang, Z., et al., *Effect of Fuel Injection Timing Relative to Ignition Timing on the Natural-Gas Direct-Injection Combustion.* Journal of Engineering for Gas Turbines and Power ASME, 2003. **217**: p. 53-61.
54. Abdel-Rahman, A.A., *On the emissions from internal-combustion engines: A review.* International Journal of Energy Research, 1998. **22**(6): p. 483-513.

55. Huang, Z., et al., *Correlation of ignitability with injection timing for direct injection combustion fuelled with compressed natural gas and gasoline.* Proc. Instn Mech. Engrs. Part D: j. Automobile Engineering, 2003. **217**: p. 499-506.
56. Huang, Z., et al., *Correlation of ignitability with injection timing for direct injection combustion fuelled with compressed natural gas and gasoline.* Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering, 2003. **217**(6): p. 499-506.
57. Goto, Y., *Mixture formation and ignition in a direct injection natural gas engine.* JSME International Journal, Series B, 1999. **42**(2): p. 268-274.
58. Wang, J., Z. Huang, B. Liu, K. Zeng, J. Yu, and D. Jiang, *Effect of fuel injection timings and hydrogen fraction on combustion characteristics of direct-injection engine.* Hsi-An Chiao Tung Ta Hsueh/Journal of Xi'an Jiaotong University, 2006. **40**(7): p. 767-770.
59. Kakuho, A., K. Yamaguchi, Y. Hashizume, T. Urushihara, and T. Itoh, *A Study of Air-Fuel Mixture Formation In Direct-Injection SI Engine.* SAE, 2004. **2004-01-1946**.
60. Ortmann, R., S. Arndt, J. Raimann, R. Grzeszik, and G. Wurfel, *Methods and Analysis of Fuel Injection, Mixture Preparation and Charge Stratification in Different Direct Injected SI Engines.* SAE, 2001(2001-01-0970).
61. Heywood, J.B., *Internal Combustion Engine Fundamentals.* 1988: McGraw-Hill.
62. Zervas, E., *Comparative study of some experimental methods to characterize the combustion process in a SI engine.* Energy, 2005. **30**: p. 1803–1816.
63. Alkidas, A.C., *Combustion-chamber crevices: the major source of engine-out hydrocarbon emissions under fully warmed conditions.* Progress in Energy and Combustion Science, 1999. **25** p. 253–273.
64. Mollah, M.A.H., M.S.A. Molla, S.M. Sapuan, and M.M.H.M. Ahmad, *Performance and emission characteristics of a compressed natural gas fuelled spark ignition engine.* International Energy Journal, 2001. **2**(1): p. 7-14.
65. Loffer, G., R. Sieber, M. Harasek, H. Hofbauer, R. Hauss, and J. Landauf, *NOx Formation in natural gas combustion-a new simplified reaction scheme for CFD calculations.* Fuel, 2006. **85**: p. 513-523.
66. Cannon, S.M., B.S. Brewster, and L.D. Smoot, *Stochastic Modeling of CO and NO on Premixed Methane Combustion.* Combustion and Flame, 1998. **113**: p. 135-146.
67. Raine, R.R., C.R. Stone, and J. Gould, *Modeling of Nitric Oxide Formation in Spark Ignition Engines with a Multizone Burned Gas.* Combustion and Flame, 1995. **102**: p. 241-255.
68. Grimm, B.M. and R.T. Johnson, *Review of SImple Heat Release Computation.* SAE International, 1990(900445).
69. Huang, Z., et al., *Visualization study of natural gas direct injection combustion.* IMEchE, 2003. **217**(Proc. Instn Mech. Engrs Part D: Journal Automobile Engineering): p. 667-676.
70. Huang, Z., et al., *A basic behavior of CNG DI combustion in a spark-ignited rapid compression machine.* JSME International Journal, Series B: Fluids and Thermal Engineering, 2002. **45**(4): p. 891-900.
71. Huang, Z., et al., *Study of cycle-by-cycle variations of natural gas direct injection combustion using a rapid compression machine.* Proc. Instn Mech. Engrs. Part D: j. Automobile Engineering, 2003. **217**: p. 53-61.

72. Huang, Z., et al., *NO/NO2 concentration of direct injection stratified combustion under constant volume condition fuelled by compresed natural gas and gasoline.* IMEchE, 2003. **217**(Proc. Instn Mech. Engrs. Part D: Journal Automobile Engineering): p. 935-941.
73. Huang, Z.-H., K. Zeng, and Z.-L. Yang, *Emission characteristics of direct injection natural gas engine using rapid compression machine.* Ranshao Kexue Yu Jishu/Journal of Combustion Science and Technology, 2002. **8**(3): p. 241-247.

APPENDIX A

CNG composition in Malaysia

Component	Leanest	Reachest	Unit
Methane	96.42	89.04	%
Ethane	2.29	5.85	%
Propane	0.23	1.28	%
Iso-Butane	0.03	0.14	%
N-Butane	0.02	0.10	%
Iso-Pentane	N/A	N/A	%
N-Pentane	N/A	N/A	%
N-Hexane	N/A	N/A	%
Condensate	0.00	0.02	%
Nitrogen	0.44	0.47	%
CO_2	0.57	3.09	%
Gross Heating Value	38130	38960	Kj/Kg

APPENDIX B

General categories of gasoline direct injection injectors are presented as follows:

Classification categories for Gasoline-DI Injectors

Actuation Mechanism	Single solenoid Dual solenoid piezoelectric hydraulic Cam
Fluid State	Single fluid Air-assist
Primary Atomization method	Sheet (swirl-plate) Pressure (hole-type) Pressure (slit-type) Turbulence (compound plate) Pneumatic (air-assist) Cavitations Impingement
Nozzle Configuration	Swirl Slit Multi-hole Cavity
Pintle Opening Direction	Inwardly Opening Outwardly Opening
Spray Configuration	Hollow-cone Solid-cone Fan Offset Multi-plume Shaped

Source: Zhao.F.Gasoline Direct Injection Engines

APPENDIX C

LITERATURE REVIEW on CNG-Direct Injection

Author	Masaru Hayashida et.al	Seiichi Shiga et.al	Deanna E.Wang et.al	Zuohua Huang et.al	Jongwoo Kim	Ke Zeng et.al
Engine Type	Single cylinder, OHV, Water cooled, Four stroke	RCM	Single Cylinder, DOHC, Four stroke, Hydrogen assisted injection	RCM	4 Cylinder, pent-roof cylinder head	Single cylinder, Four stroke
Ignition System	Spark Ignition	Spark Ignition	Spark Ignition	Spark Ignition	Spark Ignition	Spark Ignition
Bore (mm)	94	80	89	80	79.5	100
Stroke (mm)	90	180	66	180	95.5	115
Displacement (cc)	624	904	410	904	1900	903
Compression ratio	11.4	10	11.6	10	10.4	8
Piston Crown	Disk	Flat		Flat	Flat	Bowl-in-shape
Injector Position	Angled, Intake direction	Side Injection		Side Injection	Side injection	Side Injection
Injection Pressure	5 Mpa	9 Mpa	2 Mpa	9 Mpa	20 bar and 80 bar	80 bar
Spark Position	Exhaust valve side	Center		Center	Center	Center
Valve Numbers	2		4		4	2
Max. BMEP	n/a	n/a	n/a	n/a	n/a	0.4 Mpa at 180BTDC
Engine speed	1200				1600	1200
Throttle	WOT	n/a	WOT	n/a	part	WOT
Equivalence ratio	0.83	0.6 and 1	0.56	0.6 and 1	0.6 to 1	0.5

LITERATURE REVIEW on CNG-engine fueling method development

Literature on Fueling Method of CNG Engine

Year	Author	Research fields			
		CR	Fueling system	Operating condition	Observed parameters
2005	C.C.O. Reynolds et.al	9.26	Mixer	Strafied Condition Part load SI engine	Partially Stratified Charge composition
2005	Chetan S. Mistry	n/a	Carburetor	Variable Load Variable Speed SI engine	Comparison with Gasoline and LPG
2000 / 2005	S Maji et.al	8	Mixer	SI engine WOT	Comparison of CNG and Gasoline as engine fuel Spark timing AFR Compression ratio
1995	P. Corbo et.al	10.8	Carburetor	WOT	Comparison between stoichiometric and lean burn in CNG Heavy-Duty Engine
2003	Jan Czerwinski et.al	11	Multi Port Injection	SI engine Turbocharge engine WOT	Comparison of MPI with Port injection Injection Timing Injection Pressure
2002	Jan-Ola Olsson et.al	Varied	Port Injection	WOT HCCI engine Lean condition	Compression ratio

2000	A. E. Catania et.al	10.35	Multi Point Injection	Injection Pressure 7 bar WOT SI engine	Comparison with Gasoline
1999	Kichiro Kato et.al	11	Port Injection	WOT Injection pressure 764 kPa SI engine	Comparison with Gasoline
1998	Joseph M. Brault	12.5	Multi Port Injection	Starting Condition SI engine	Injection timing