EFFECT OF COMPRESSED NATURAL GAS ON PERFORMANCE AND EMISSION OF A 4-STROKE SPARK IGNITION ENGINE

AWANG BIN IDRIS

MASTER OF ENGINEERING (AUTOMOTIVE)

UNIVERSITI MALAYSIA PAHANG

UNIVERSITI MALAYSIA PAHANG

CENTER FOR GRADUATE STUDIES

We certify that the thesis entitled "Effect of Compressed Natural Gas on Performance and Emission of a 4-Stroke Spark Ignition Engine " is written by Awang bin Idris . We have examined the final copy of this thesis and in our opinion; it is fully adequate in terms of scope and quality for the award of the degree of Master of Engineering in Automotive. We herewith recommend that it be accepted in fulfillment of the requirements for the degree of Master of Engineering specializing in Automotive.

Name of External Examiner Signature

Profesor Dr. Ahmad Kamal Ariffin bin Mohd Ehsan

Jabatan Kejuruteraan Mekanik & Bahan,

Fakulti Kejuruteraan,

Universiti Kebangsaan Malaysia,

43600 Bangi, Selangor.

Name of Internal Examiner Signature

Profesor Madya Dr. Mustafizur Rahman

Fakulti Kejuruteraan Mekanikal,

Universiti Malaysia Pahang,

Karung Berkunci 12, 25000 Kuantan,

Pahang.

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AWANG BIN IDRIS

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DECLARATION OF THESIS

Author's full name : Awang bin Idris

Date of birth : 05/10/1967

Title : EFFECT OF COMPRESSED NATURAL GAS ON PERFORMANCE AND EMISSION OF A 4-STROKE SPARK IGNITION ENGINE

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PROESSOR DR. ROSLI ABU BAKAR
NAME OF SUPERVISOR

Date: 14 NOVEMBER 2009 Date: 14 NOVEMBER 2009

SUPERVISOR'S DECLARATION

We hereby declare that we have checked this thesis and in our opinion, this thesis is adequate in terms of scope and quality for the award of the degree of Master of Engineering (Automotive).

Signature

Name of Supervisor : DR. ROSLI ABU BAKAR

Position : PROFESSOR

Date : 14 NOVEMBER 2009

Signature

Name of Co-supervisor: DR. AHMAD SYAHRIZAN SULAIMAN

Position : LECTURER

Date : 14 NOVEMBER 2009

STUDENT'S DECLARATION

I hereby declare that the work in this thesis is my own except for quotations and summaries which have been duly acknowledged. The thesis has not been accepted for any degree and is not concurrently submitted for award of other degrees.

Signature

Name : AWANG BIN IDRIS

ID Number : MMA05001

Date: : 14 NOVEMBER 2009

Dedicated to my beloved family and friends

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ABSTRACT

This thesis deals with experimental study of a four-stroke spark ignition engine. The objective of this thesis is to evaluate the performance and emission characteristics of the engine while using conventional fuel, gasoline and alternative fuel, compressed natural gas (CNG). The engine operated under a steady state condition at wide-open throttle condition. The performance and emissions test was performed with various constant loads at different speed within the range of 1500 rpm to 4000 rpm with 500 rpm interval. The first experiment is executed by using gasoline and followed by CNG. The engine performance and emissions such as air-fuel ratio, torque, brake power, brake specific fuel consumption, efficiency, the concentration of CO, CO_2, HC, and NO_x of gasoline and CNG were measured. The results demonstrated that the potential of reducing emissions while applying CNG as fuel is obvious. However the performance of CNG is reduced as the brake power of engine decrease around 25% compare to gasoline engine. During operate with CNG the engine emissions of CO, CO_2 and HC shows a significant reduction but the NO_x emission is highly increased compare to gasoline. The results and analysis will be useful for the development of dedicated gas engine in the near future.

ABSTRAK

Tesis ini membentangkan keputusan ujikaji enjin nyalaan bunga api bagi menilai prestasi dan ciri-ciri emisi apabila menggunakan bahan api gasolin dan gas asli termampat (CNG). Bagi menilai prestasi engine, operasi dilakukan dengan kelajuan enjin antara 1500 ppm hingga 4000 ppm, di bawah keadaan mantap dengan pendikit terbuka luas (wide-open throttle). Ujian emisi dijalankan dengan menggunakan beban tetap yang berbeza pada setiap kelajuan. Prestasi dan keluaran engin seperti nisbah udara-bahan api, daya kilas, kuasa brek, penggunaan bahan api tentu brek CO, CO_2, HC dan NO_x dari CNG diukur dan dibandingkan dengan bahan api gasolin. Keputusan kajian menunjukkan CNG sangat berupaya mengurangkan keluaran berbanding dengan gasolin. Walau bagaimana pun, prestasi CNG berkurangan, di mana kuasa brek menurun sehinga 25% pada berbanding dengan gasolin. Semasa beroperasi dengan CNG keluaran engine seperti CO, CO_2 dan HC sangat berkurangan tetapi keluaran NOx meningkat dengan begitu tinggi berbanding dengan gasolin. Keputusan dan analisa yang dibuat sangat berguna untuk kajian lanjut bagi tujuan pembangunan enjin gas pada masa depan.

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LIST OF SYMBOLS

B	Bore
C_F	Correction factor
C_d	Coefficient of drag
L	Stroke
m_a	Air flow rate
$\dot{m}_f$	Fuel flow rate
$\dot{m}_a$	Air flow rate
ΔP	Pressure diffrent
$P_{i,m}$	Measured indicated power
$P_{s,\,d}$	Standard dry-air absolute pressure
P_m	Measured ambient-air absolute pressure
$P_{i,\,s}$	Standard indicated power
P_b	Brake power
P_f	Measured friction power
$P_{i,\,g}$	Indicated gross power
$P_{b,\,m}$	Measured Brake power
P_{atm}	Atmosphere pressure
P_i	Measured indicated power
Q_{LHV}	Low heating value

$\bar{S}_p$	Mean piston speed
T_s	Standard ambient temperature
T_{amb}	Temperature during experiment
V_d	Displaced volume
ρ_{air}	Air density
t_f	Time of fuel consumption in second
η_{bth}	brake thermal efficiency
η_{bth}	Mechanical efficiency
η_v	Volumetric efficiency

LIST OF ABBREVIATIONS

AFR	Air-fuel ratio
BMEP	brake mean effective pressure
BSFC	Brake specific fuel consumption
BDC	Botton dead centre
CNG	Compressed natural gas
CO	Carbon monoxide
CO_2	Carbon dioxide
DOHC	Double over-head cam
EGR	Exhaust gas recirculation
FMEP	Friction mean effective pressure
HC	Hydrocarbom
LPG	Liquified Petroleum Gas
PMEP	Pump mean effective pressure
SULEV	Super ultra low emissions vehicle
sfc	Specific fuel consumption
SI	Spark ignition
TDC	Top dead centre
TWC	Three-way catalytic converter
ULEV	Ultra low emission vehicle
WOT	Wide-open throttle

CHAPTER 1

INTRODUCTION

1.1 Introduction

It is a well known fact that the fossil fuel reserves all over the world are diminishing at an alarming rate and a shortage of crude oil is expected in early decades of this century (Aslam et al., 2006). The needs of alternative fuel is becoming the subject of interest to replace conventional fuel. The best candidate to replace conventional fuel is compressed natural gas (CNG). The used of CNG for vehicles date back to the 1930's in Italy. The late 70's however witnessed the launching of and commitment to a growing and developing industry. There are more than 3 million natural gas vehicles worldwide with major concentrations in Argentina, Brazil, Italy, Pakistan, USA and Egypt (Bardan, 2005). The main factors which stimulate interest in using CNG as fuel are energy security, emission reduction which can reduce the greenhouse gas effect and the cost is very much lower compared to gasoline and diesel and other alternatives fuel. However the main problem of using CNG as fuel is the performance degradation due to slow burning of the gas because of low energy content per unit volume (Heywood, 1988).

At the present time, there are many vehicle using natural gas as fuels such as lorries, buses, motorcars, ferries and etc. Most of the vehicles that are using natural gas is using the retrofitted method which needs an aftermarket conversion kit to be installed. The majority of these vehicles are converted gas vehicles adapted to use CNG in bi-fuel systems. The renaissance of interest in natural gas is springing up anew in both research and development as a preparation for the gas – age (co-existence of NGV, biogas and hydrogen). Having recognized the potential for a gas-age period, the present task remains to develop the required technology to a standard whereby transition to hydrogen or bio-gas fuel may be carried out when required, as a result of economic, social, technological or political changes.

1.2 Malaysian Scenario

According to Sahari et al., (2005) the development of natural gas vehicle in Malaysia was carried out by Petronas and started in 1986 as in 1984 they have carried out a study on the usage of natural gas in Malaysia. The history of the development of natural gas application in Malaysia is shown in the chronological order in the Table 1.1.

Table 1.1: The history of natural gas application for vehicle in Malaysia

Year	Activities
1984	Petronas' journey in promoting NGV began with a study on the usage of Natural gas in Malaysia as an alternative fuel for transportation.
1986	Petronas' embarked on an NGV Pilot Programme to improve its understanding of the technology involved and to lay the framework for a commercial program.
1991 to 1994	Petronas' commitment continued with the development with the Malaysian Standard (MS1204) jointly with SIRIM for the construction of NGV stations.
1995	Petronas NGV Sdn. Bhd was incorporated to spearhead the promotion and development of NGV in Malaysia.
1996 to 1999	Developing a system for safety inspection, involving the Road Transport Department and PUSPAKOM's inspection of NGV vehicles and the approval of NGV conversion workshops. Standard for importation and re-testing of NGV cylinder were also established with the Department of Occupational Safety and Health, Malaysia (DOSH).
1998	Malaysia's first mono-fuel vehicle, the Enviro 2000 NGV taxi which was used during XVI Commonwealth Games.

2004	A special committee on NGV, in October 2004 to study and recommended suitable strategies to accelerate the implementation of Malaysia’s NGV program

Resource: Petronas NGV Sdn Bhd.

The growth rate of vehicles in Malaysia is quite high. In the year 2005, 1.02 million new vehicles were registered and 55% of new registrations were cars (The Star, 19th February, 2006). In a certain state the number of vehicles is more than its population. For example in Penang, there are 1.47 million vehicles compares to 1.4 million populations (Utusan Malaysia, 16th March, 2006). This contributes to the increased demand on fuel usage and air pollution is worsened. This kind of situation is happened all over the world and the needs of alternative fuel, like CNG is inevitable.

Concerning to the environmental pollution produced by vehicle fuel the Malaysia government is encouraging the use of CNG engines as an alternative. It will reduce the CO, CO_2 and NO_x emissions. As the price of CNG is very much lower compare to conventional fuel - gasoline and diesel, it will reduce the cost also. Furthermore the CNG engines are comparable to the existing engines. Performance wise, the CNG engine is expected to be same with existing engine by using advanced technology and further research in the future. The Malaysian government has implemented a few components to encourage the use of CNG as alternative fuel included:

i. Ensure the continuous availability and fair pricing of the CNG. The current price of CNG is RM0.68 per litre. Compare to the petrol and diesel the price of CNG are lower 65% and 57% (The Star, 19th February, 2006).

ii. Additional reduction of the road tax from existing level (Mardani et al., 2001):
   a. Monogas or dedicated vehicles 50% off
   b. Bi-fuel vehicles 25% off
   c. Dual fuel vehicles 25% off

iii. Tax incentives and other financial incentives for encouraging and facilitating purchase of new buses and other vehicles and construction of CNG outlets.

The commercialization of CNG vehicles in Malaysia is far behind compare to other countries such as Argentina, Brazil and Pakistan. Currently, there are 40 service stations in Malaysia, providing CNG refuelling facilities to a population of around 15,600 natural gas vehicles. However the development of CNG refuelling stations and the construction of CNG pipelines are still in progress. The government of Malaysia plans to develop 94 stations to serve 54,000 vehicles by the year 2010 (Daud, 2005). In Malaysia the natural gas is supplied to NGV stations through natural gas pipe line (conventional system) and trailers (mother-daughter systems)

The use of CNG is predicted to increase in the near future because of the increasing price of petrol and diesel fuel. On February 28, 2006 the price of the fuel increased by 15.6% and recently it was increased again around 40% for petrol. According to this scenario, the government and private sectors were encouraged by economists and politicians to use alternative fuel, ultimately CNG. As a result of those suggestion one of the major players in transportation of Malaysia, NADICORP Holdings Bhd has taken an initiative to use the CNG by early 2007, for 800 buses stage by stage. (Berita Harian, 5th April, 2006).

1.3 Problem Statement

Energy is the key for the development of a nation but its stock is limited. To ensure the continuous supply of energy mainly depending on gasoline and diesel, many countries are looking for alternative fuel. It is well known that CNG is one of the best candidates of alternative fuel. However the previous research of Catania et al. (2000) indicated the degradation of performance of retrofitted CNG engine**.** The main reason of the lower performance of retrofitted CNG engine is the energy content per unit volume in CNG is lower than gasoline (Heywood, 1988). As the main disadvantage of retrofitted CNG engine is the performance degradation, the data of the research will be useful to identify the problems and improves the performance of the engine. The effects of performance parameters are thoroughly investigated.

1.4 Objectives of the Research

The objectives of this study are as follows.

i. To analyze the behaviour of retrofitted engine performance in term of power, torque and efficiency using CNG as a fuel.

ii. To evaluate the engine emission and fuel consumption while operating with CNG.

1.5 Scope of the project

This project is to investigate a retrofitted four stroke 1.5L spark ignition engine performance and emission during operating between 1500 – 4000 rpm. The entire experiment is executed at wide open throttle (WOT) conditions. The engine specification is remaining unchanged and the engine was not installed with the Three Way Catalytic Converter (TWC) of emission control.

1.6 Thesis Organization

This thesis consists of five chapters including introduction. Chapter 2 comprises a literature survey on the subject of world fuel supplies, CNG as alternative fuel, the characteristics of CNG, previous research on CNG vehicles, engine performance and emission. Chapter 3 concentrates on the methodology of the research: the flowchart, experimental set-up, instrumentation, measurement the important parameter involved in these studies. Chapter 4 provides detailed information on result and discussion on this research which are experimental results of engine performance and emission for both type of fuel, gasoline and CNG. The summary of all the present work and recommendations for further work are also presented in this chapter.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

The purpose of this chapter is to present the previous studies related to CNG as alternative fuel application on four stroke spark ignition engine while using gasoline and CNG. The literatures survey consist of four distinguished subjects, which are (i) world fuel supply, (ii) compressed natural gas as alternative fuels, (iii) Characteristics of CNG, (iv) development of vehicle powered by CNG and (iv) previous research on CNG engine performance and emission.

2.2 World Fuel Supplies

Energy is a vital resource for development but its availability is bounded. In accordance to the information illuminated in energy conservation handbook, Japan, Germany, France and Italy import more than 90% of their oils and America imports more than half of the oils consumed. In all countries studied, transportation consumes approximately one-fourth to one-third of the total energy consumption (Conservation Energy Handbook, Tokyo 1991).

During last few decades supply disruptions and cost increases of petroleum had awaken the world to realize the nearly close to depletion of the petroleum supply. This had initiated a dramatically accelerated interest and mission in searching for alternative fuels for automobiles and other power primers. The need for these fuels has become a high priority for many countries, particularly those with little or shortage of indigenous hydrocarbon resources, and investigations were initiated into possible use of natural gas, coal, oil, shale, uranium, vegetable oils etc. The world fuel supply continues to change. With engines operating at peak efficiencies, both thermodynamically and with absolute clean combustion, there still needs to be a continuous source of fuel to be burnt. The analysis of determining the time that petrol fuels supply will be adequate for the market demand depends on the rate at which the current fuel resources and the state of political relations with oil producing countries (Ferrone, 1991).

2.3 CNG as Alternative Fuel

A review on some of the recent research and development activities on the use of CNG as fuel substitute is present, and results obtained from using CNG as fuel in internal combustion engines. The opportunities for further technology and development are investigated even though CNG will not entirely displace petroleum as source of fuel.

Energy consumption in developing countries has increased rapidly in past decades and will probably continue to do so. This increase is essential for economic growth and improved standards of living. For many developing countries the main sources of this energy come from imported petroleum. Increasing use of petroleum will intensify local air pollution and magnify the global warming problems caused by CO_2 gas emission. The present supply outlook and the environmental consideration suggest that an urgent need exist for the development of economically and environmentally sound alternatives to petroleum fuels for modern internal combustion engines deserve the attention of both developed and developing countries (Energy Information Administration, 1998)

CNG should be considered as the main candidate for alternative fuel to prolong the existence of fuel. The seven criteria of alternative fuel outlines are listed in Table 2.1 (Semelsberger et al., 2005). Furthermore Dennis Dart, United Kingdom bus manufacturer, has already evaluated eight promising alternatives fuels which are listed in Table 2.2.

Table 2.1: Criteria affecting the suitability of alternative fuel (Semelsberger et. al., 2005)

Item	Criteria
Availability	Production facilities, pre-existing infrastructure and the natural resources as raw material in the form of fossil fuels
Economics	The fuel production and distribution cost, cost of constructing new facilities, cost of raw material used, and the cost of retrofitting old equipment to process the new fuel or to replace them with new technology
Acceptability	Inherently safe in handling and refuelling and also inherent health risks to humans or animal life.
Environmental & Emissions	Reduce fuel effect to global warming and if in the event of large scales release, it must be low affect the environment
National security	The material must be readily available and process without reliance on foreign materials
Technology	Must be commercially available and emerging technology can process the fuel
Versatility	It must be versatile in application & can be manufactured using various feedstock

Table 2.2: Dennis Dart fuel requirements for various alternative fuels (Stratton, 1996)

Fuel Types	Storage Pressure (bar)	Fuel Storage Volume (litres)	Fuel container weight (kg)	Fuel Storage Temperature (^{0}C)	Calorific Value NET (MJ/kg)
Diesel	1	135	30	15	42.9
Petrol	1	160	35	15	43
CNG	200	540	460	15	47.2
LPG	8	230	70	15	46.1
LNG	6	260	80	-161	47.2
Methanol	1	300	70	15	19.7
Electric	1	-	5000+	15	-
Hydrogen	300	270	950	15	119.8

2.4 Characteristics of CNG

Methane is a dominant component but ethane and heavier hydrocarbons such as propane, butanes, etc are in natural gas up to a maximum of equilibrium vapour pressure. Only saturated hydrocarbons are found in gas. The general formula in this series is C_nH_{2n+2}.

The natural gas composition has impact on emission, performance and safety. In terms of this issue, the effect of natural gas composition can be outline as follows:

a. knock tendency
b. fuel stratification in storage cylinder
c. fuel metering which has a primary effect on engine performance and emission
d. Corrosion occurrence due to impurities in fuels.

Natural gas has many positive characteristics (Table 2.3) that help to reduce engine emissions and in transportation it helps to reduce the fuel cost. With a gaseous fuel mixture formation is very good, even at cold start conditions. It does not wet the intake manifold or cylinder walls as gasoline does. Thus an exact stoichiometric air-fuel ratio (AFR) can be maintained during transient engine operation

The high octane number in the range of 120 to 130 (Heywood, 1988; Poulton, 1994) of natural gas allows high compression ratio in optimize the natural gas, which means an increase efficiency, thus lowering the raw emission (Heck, 1998) without knocking phenomena to piston that will cause damage to the engine. This leads to a reduction of CO_2 emission up to 23% at equal energy conversion (Heitzer et al., 2000). However antiknock rating higher than that required for knock-free operation does not improve engine performance.

Even considering the fact that methane is a much powerful greenhouse gas than CO_2; the overall green house effect of a NGV is still lower than conventional gasoline vehicles due to the low levels of unburnt methane in the exhaust. However not only engine emissions are reduced with natural gas, emission during transport, storage and refuelling (fuel cycle emissions) are also below standard liquid fuels. The heating value in other words, heat of combustion or calorific value of fuel is the amount of heat produced when the fuel is burnt completely. There are two values of heating value. They are higher heating value and lower heating value. The difference between them is that higher heating value exceeds the lower the lower heating value by the energy supplied by water vapour in condensing. A fuel with low heating value provides less heat on combustion which means less power than the same amount of fuel with high heating value fuel, more consumption of it would be necessary. If natural gas is used in optimized engine, better fuel consumption will be obtained.

Energy content based on volume, determines the vehicle range. To increase the range, a high density of fuel is preferred because heating value per unit volume if fuel is greater. This explains why the liquid state is of primary interest for storage problem. According to this data vehicle running on CNG travel only 26% as far as it could on gasoline (Wozniak et al., 2000; Weide, 2000).

Vapour pressure of a fuel is a prime importance of drivability of vehicles under all conditions. One of the most common methods to measure fuel volatility is the Reid Method. There is no need to vaporize CNG, contrarily liquid fuels have to be vaporized before they are introduced into engine. This characteristics of CNG makes the cold start problems and low temperature emissions due to cold enrichment, minimum. In other words the lower AFR means that chemical energy released per kilogram of stoichiometric mixture burnt during combustion is greater. It is calculated by dividing the lower heating value of the fuel by the AFR. This explains why gasoline has a greater heat release although it has a lower heat of combustion. The heat vaporization of a fuel affects the volumetric efficiency positively by decreasing the temperature of fuel-air mixture which means makes the mixture dense. Although natural gas has higher heat vaporization, it is already in gaseous state when it is inducted into engine and it does not provide this cooling effect. On the other hand cold starting problems would occur with higher heat of vaporization on internal combustion engines.

Thermal efficiency increases with lower flame temperature due to the reduced heat losses from an engine. The lower visibility (luminosity) also decreases heat loss by radiation. Flame temperature is also a parameter in NOx. Lower flame temperature reduces NOx emission. Flame speed defines the relative motion of the flame front towards the unburnt mixture. Stoichiometric AFR is strongly determined by flame speed. The fast combustion rate provides more efficient torque development. On the other hand, the increasing thermal and mechanical burdens along with increased combustion temperatures cause higher thermal losses, combustion noise and NOx emissions.

When released to the atmosphere, CNG will mix with the air and become flammable only when the mixture is within 5 to 15 percent natural gas. If the mixture is less than 5 percent natural gas, it doesn't burn and if the mixture is more than 15 percent natural gas, there is not enough oxygen to allow it to burn. Because natural gas is lighter than air, it quickly dissipates when released from a tank. This kind of property is shown that CNG does not explode easily and is safe to use.

Table 2.3: Combustion related properties of Gasoline and CNG (Heywood, 1988)

Properties	**Gasoline**	**CNG**
Motor octane number	80-90	120
Research octane number	92-98	120
Molar mass (kg/mol)	110	16.04
Stoichiometric air-fuel ratio	14.6	16.79
Stoichiometric mixture density (kg/m3)	1.38	1.24
Lower heating value (MJ/kg)	43.6	47.377
Lower heating value of stoichiometric mixture(MJ/kg)	2.83	2.72
Flammability limits (volume % in air)	1.3-7.1	5-15
Spontaneous ignition temperature (^{0}C)	480-550	645

2.5 Research on CNG Vehicle

Most of the research performed by automotive manufacturers, universities or research organizations is focuses on how to increase the performance of the vehicle by increasing the engine power to equal its gasoline vehicle counterpart, increase the driving range, reducing the weight of CNG storage, exhaust emissions meet or better than standard, and meet all motor vehicle standard and trunk storage space comparable to gasoline vehicles (Wozniak et al., 2000). However, every effort will be made to use as many as possible common parts with gasoline engines. Emission control and engine efficiency are two of the most important design considerations in current engine development. The impending introduction of emissions standards such as Euro IV and V, Japan and New Japan Long Term and the Californian Super Ultra Low Emission Vehicle (SULEV) is forcing the automotive engineer to develop new technologies to combat engine emissions. The end result could be more efficient engines with simultaneous reduction of all major pollutants relative to gasoline engines of equivalent capacity.

There are three possible alternative systems of having an equal or higher efficiency than diesel engines (Shioji, 2000). The first at a homogeneous-charge lean-burn spark-ignition operation, a deep-bowl combustion chamber can significantly suppress the cycle-to-cycle variations under a fuel-lean condition at a higher swirl ratio, thus achieving low NO_x emissions at various engine loads by regulating mixture concentrations. The second is dual-fuel natural gas operation with gas-oil pilot injection gives a higher thermal efficiency and much less smoke emission compared to the diesel operation over a wide range of loads, as long as adequate control of pilot-fuel amount, injection timing and throttle opening area are selected. And the third is high-pressure natural-gas direct injection operation is successfully achieved with a glow-plug ignition-assist, exhibiting slightly higher thermal efficiency than the gas-oil operation only at higher loads. However, at middle and light loads incomplete combustion occurs and thermal efficiency significantly lowers.

Lean burn occurs when the air/fuel ratio is greater than stoichiometric. The relative air/fuel ratio (λ) is obtained by dividing the actual air/fuel ratio by the stoichiometric air/fuel ratio. It has been found that lean operations ($\lambda >> 1.0$) can significantly decreased NOx and CO emissions. Other advantages include increased overall efficiency due to changed specific heat ratios and reduced pumping losses. However, limits exist on how lean an engine may be run before the occurrence of misfiring and high cyclic variability resulting from irregular ignition and fluctuating flame propagation. The bi-fuel Lotus Elise implements multi-point port injection technology along with advanced, integrated gas storage and fuel control systems to the 1.8 Liter engine (Durrel et al., 2000). Following on from the success of this technology demonstrator, Lotus has investigated the use of direct injection – a technology becoming very important in gasoline vehicle – with CNG. The test results on CNG at stoichiometric conditions are shown that the emissions reductions of 60% HC, 26% NO_x, and 21% of CO_2 and the torque reduction of 9% compared to gasoline. The torque as well as power reduction are believed due to lower density of CNG and the lower calorific value of the CNG/ air mixture. The use of exhaust gas recirculation (EGR) to decrease NO_x emissions can be applied just as effectively to a CNG engine as a gasoline engine showing the adaptability of CNG as a fuel that is responsive to emissions reduction techniques already implemented with gasoline. The successful story of Honda Civic GX, an advanced natural gas vehicle could be meet the Super-Ultra-Low Emission Vehicle (SULEV) emission standard in California and also meet the future European and Japanese emission standards (Suga et al., 2000). University of Melbourne develop of a CNG engine with ultra lean burn low emissions potentials named Hydrogen assisted jet ignition (HAJI) to achieve reliable combustion and low NO_x emission, whilst direct injection is used to improve thermal efficiency and decrease HC emissions. It replaces the spark plug in a conventional spark ignition engine and virtually eliminates the lean limit. It consists of small prechamber (approximately 0.7% of TDC main chamber volume), a hydrogen injector and a spark plug in the engine (Wang et al., 2000)

The dedicated CNG engine- vehicle technology involves with engineering design and development of three areas – engine, gas storage and vehicle packaging completely and they are trying to obtain the certification and to produce by line in plant. Nissan CNG S-class van is the first CNG small van inline production that certified as a ULEV by Ministry of Transport in Japan (Yoshikawa et al., 2000) The major points of developing dedicated CNG engine-vehicles are reducing the exhaust emissions to comply with the emission standard such as ULEV, SULEV, Euro IV or Euro V, to achieve the greater cruising range by a single charge, to secure roomy luggage compartment by using space effectively and acceleration comparable to the gasoline model. Another successful development of a prototype of a dedicated CNG engine-vehicle is a mid-size sedan, done by The John Hopkins University Applied Physics Laboratory (JHU/APL), in conjunction with DaimlerChrysler and Lincoln Composites. It has a city/highway driving range of 480km, ample trunk capacity (354 liters), and acceleration comparable to the gasoline model and achieving ultra-low exhaust emission. The 2.4-liter double overhead cam (DOHC) engine was modified for natural gas operation with high-compression pistons, hardened exhaust valve, a methane-specific catalytic converter, and a multipoint gaseous fuel injection system. The chassis was repackage to increase space for fuel storage with a custom-designed, cast aluminum, semi-trailing arm rear suspension system, a revised flat trunk sheet metal floorpan, and by equipping the car with run-flat tires (Wozniak et al., 2000).

On-board CNG fuel storage presents unique challenges for the commercialization of natural gas vehicles. Vehicle range, storage system cost, durability, weight and compatibility of component materials are all key issues. Until now the limiting factor in all efforts to increase the fuel efficiency and driving range of passenger car and light duty vehicles fuelled with natural gas was the extra weight added to vehicle by the gas storage tanks. Recent developments in composite materials are promising to reduce drastically the total fuel/tank mass by combining thinner tank walls, non-cylindrical irregular tank shapes and higher storage pressure, albeit at considerable cost. At the same time new gas storage tanks should be thoroughly tested in order to prove that they are capable of meeting the very high safety standards existing worldwide. The test is include pressure cycling, environmental exposure, hydrostatic burst, bonfire test, penetration test, permeation test, and accelerated stress rupture. Thiokol Propulsion has successfully designed and demonstrated a two cell conformable tank capable of storing CNG at 248 MPa, provides up to 50% more storage within available vehicle space compared to conventional cylindrical tanks and safe operation over the tank's 15-20 year. Additional challenges for this effort include incorporation of lower cost manufacturing component and techniques (Haarland et al., 2000).

The University of Oklahoma introduced composite fuel, trademarked as "Super-Gas^{TM}", a combination of propane (LPG) and CNG (50/50 mass mixture), is stored in liquid form at relatively low pressures (1460 psig) as compared to traditional CNG systems. Its gives greater range is reached when driving at the same conditions (Super-Gas™ vs. CNG) and using the same tank size with half pressure of CNG (approx. 1500 psig), but retains the high energy density of LPG (Sanchez et al., 2000). In Japan, the first NGV conversion was carried out from 1991. To apply lighter-weight and lower cost cylinders for NGV service in Japan, the characteristics of the following 2 types of cylinder design are being researched and tested at Faber in Italy and Powertech in Canada (Kikihara et al., 2000)

- Type 1 steel cylinder: maximum tensile strength not greater than 1095 N/mm^2 according to ANSI/IAS NGV2-1998

- Type 2 steel FRP wrapped: Maximum tensile strength is same as Type 1 steel cylinder.

Gas Research Institute (GRI) introduced Type 3 cylinder as a comprehensive strategy to help promote the natural gas vehicle (NGV) industry through the optimization of NGV storage technologies (Gambone et al., 2000). Dynatek Industries, sought to further optimize their existing NGV cylinder design by reducing manufacturing cost, improving life cycle performance and increasing impact durability. Another type of natural gas storage tank is Type 4 with plastic line, fully over-wrapped with composite. For weight critical vehicle applications the Type 3 and Type 4 tanks are more commonly use today. The Type 4 tanks is a very lightweight design where the liner material is used for preventing gas permeation and composite over-wrap is design to carry the entire pressure loading. There is also a development of integrated storage system by JHU/APL. These design elements enhance the integration of the ISS container with the vehicle to provide a driving range and trunk capacity comparable to gasoline-powered automobile. It was developed using all-composite, small diameter cylinders encapsulated within the high strength fiberglass shell with impact-absorbing foam. When CNG, LPG, and other gas fuels were used for combustion in vehicles' engines, a larger degree of valve seat wear was observed and it was difficult to provide the same wear resistance as for gasoline engines. Therefore, Honda R&D and Nippon Piston Ring, developed the mechanism of valve seat wear in gas fuel engines was analyzed and an alloy valve seat was developed (Sato et al., 2000). Concerning larger gas engines, in the medium- and heavy-duty size range, there are a development in the component such as what has been done by Advanced Engine Component Ltd. has researched, developed and commercializing its state of the art dedicated spark ignited multi-point fuel injected technology for heavy duty bus and commercial engines from 1984 (Middleton, 2000)

At the present, there are conversions of diesel engines; the converted diesel engines are based on either lean-burn homogeneous concept or stoichiometric mixtures (Bakar et al., 2007). Although lean homogeneous engines generate less NO_x emissions than stratified lean engines, they are in principle unable to reach the very low fuel consumption of their stratified counterparts. Lean burn engine technology includes turbocharging in order to maintain a high specific power output expected from large diesel engines, which is reduced at high air-fuel ratios. From the control point of view and flexibility in using natural gas of varying composition, lean burn engine are in a weak position since they require different type of oxygen sensor in order to operate as a closed-loop system; such sensors have only recently become available and are less tested compared to their stoichiometric counterparts. Southwest Research Institute do research on the development of an ultra low emission, high efficiency, natural gas fueled heavy duty engine. The engine was inline 6-cylinder model with a displacement of 8.1 liters. A spark-ignited, lean burn, open chamber combustion system was used, and the engine was turbocharged and aftercooled. The engine had full electronic control fueling, spark timing, boost pressure, and throttle position (Kubesh et al., 2000). Overall, the NO_x versus efficiency tradeoff for the engine was improved significantly. Also, in comparison to the Californian Air Resources Board Ultra Low Emissions Vehicle (CARB ULEV) standards, the composite test result were roughly 60% of ULEV standard for NO_x + NHMC, and the result for the lowest NO_x cycle were below one half of the NO_x + NHMC standard. Carbon monoxide and particulate matter were also well below the ULEV limits, again with no oxidation catalyst being used. Stoichiometric CNG engines are now under development by Cummins Westport and John Deere (Kubesh et al., 2004; Duggal et al., 2004) and have been commercialized by IVECO engines for numbers of years. IVECO have a range of heavy duty stoichiometric CNG engines in production, which range up to 228 kW and over 1000NM of torque. This is adequate for heavy duty applications. It will be also possible to improve the efficiency of such engines by use of supercharging, increased compression ratio, and exhaust recirculation. This could bring the efficiency of engines equal to lean burn CNG engine.

The development of CNG engine-vehicle technology only will not satisfy the market need except the refueling technology developed concurrently. To meet this requirement the design of refueling station must target the area of improvement included the safety, minimizing life-cycle cost, raising the comfort level and improving the general performance. Chiyodai Kikai Works Co., Ltd. and Yamaha Motor Co., Ltd. have jointly developed a gas engine driven vehicle refueling appliance, in collaboration with Tanabe Pneumatic Machinery Co., Ltd and YEC Co., Ltd. Electronic self-diagnostics always monitor the operation of this refueling appliance with safety devices such as an automatic shutoff valve, etc (Saruta, 2000). With the mentioned development target, the NGV refueling appliances that are comparatively compact, light weight, less expensive, and quite would provide the market with wider choice of refueling measures, and also of meeting anticipated demand of diversified natural gas vehicle types from heavy duty to light duties, over-the-road, in plant, and other future demands.

2.5.1 Engine Performance

Engine performance is an important factor. It means how well an engine is doing its job in relation to the input energy or how effectively it provides useful energy in relation to some other comparable engine. Engine performance is a related parameter like speed, inlet pressure, temperature output, air fuel ratio etc. the useful range of all these parameters is limited by various factors, like mechanical stresses, knocking, overheating etc. Due to this factor, there is a practical limit of maximum power and efficiency obtainable from an engine. The performance of the engine is judge from the point of view of the two main factors such as engine power and engine efficiency. The overall engine efficiency related to other efficiencies is encountered when dealing with theory, design and operation of engines. The energy flow through the engine is expressed in three distinct terms. They are indicated power (P_i), friction power (P_f) and brake power (P_b). Indicated power can be computed from the measurement of forces in the cylinder and brake power is computed from the measurement of forces at the crankshaft of the engine. The friction power can be estimated by perform motoring test of the engine. Figure 2.4 shows fuel energy distribution in IC engine system. It can be realized that reducing fuel energy losses will increase fuel conversion efficiency as well as brake power.

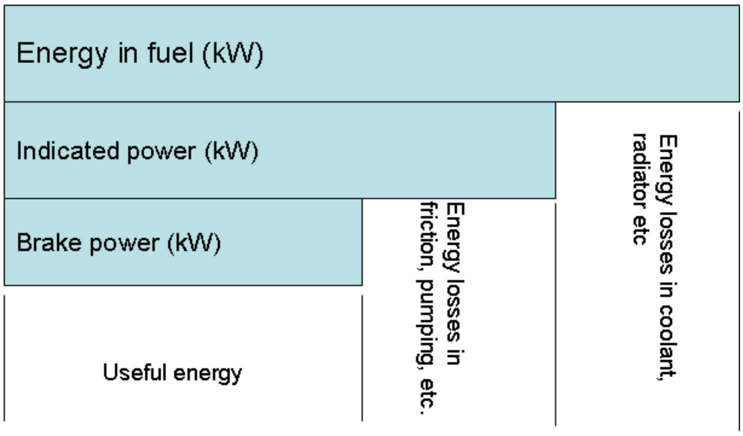


Figure 2.4: Energy Distributions (Ganesan, 2004)

The practical engine performance parameters of interest are power, torque, and specific fuel consumption. Power and torque depend on an engine's displaced volume. The major operating variables that affect spark-ignition performance, efficiency and emissions at any given load and speed are: spark timing, fuel-air (or air-fuel) ratio relative to the stoichiometric ratio and fraction of the exhaust gases that are recycle for NO_x emission control. Load is, of course, controlled by varying the inlet manifold pressure.

Bangale et al. (1995) found the maximum torque of 52.97 Nm at 2000 rpm for gasoline, and for CNG the maximum torque is 41.4 Nm at 4000 rpm. This research conducted experiment at Automotive Research Association of India (ARAI) for 800 cc engine. The maximum torque of CNG compare to gasoline is drop by 21.3%. One of the influence factors that affected to the torque is fuel flow rate. Kalam et al., 2005 carried out an experimental study and found the flow rate of CNG is lower 11% compare to gasoline. The maximum flow rate of gasoline and CNG was at 5.0 kg/hr and 4.35 kg/hr respectively.

For the brake specific fuel consumption (BSFC), Bangale et al., (1995) and Aslam et al. (2005) was found a range difference. Bangale et al found that the BSFC of CNG was reduced by 15.8% and Aslam found 18% reduction of CNG compare to gasoline. The result shows that the drops of BSFC occur as the speed in low range, nearly level at medium speed and increase at high speed.

The maximum brake power for gasoline and CNG are 47 kW and 39 kW respectively. It can be seen from the results that the brake power for CNG reduces 15% compare to gasoline (Aslam et al., 2005). In another research by Kalam et al. (2005), it is found that the CNG maximum brake power is reduce by 15 - 20% compared to gasoline. Bangale et al., (1995) at ARAI found that the CNG maximum brake power reduced by 13.2% compare to gasoline. According to Aslam et al. (2005) the result of brake mean effective pressure (BMEP) of CNG is reduce 16% compare to gasoline. The brake thermal efficiency is based on the brake power of the engine. This efficiency gives an idea of the output generated by the engine with respect to heat supplied in the form of fuel. In modern engines brake thermal efficiency of almost 28% is obtainable with gas and gasoline spark-ignition engine having a moderate compression ratio and as high as 36% or even more with high compression ratio (Ganesan, 2003). A research conducted at ARAI (Bangale et al., 1995) found that the maximum BTE is 25.2 % for gasoline and 35% for CNG.

Volumetric efficiency is used as an overall measurement of the effectiveness of a four stroke cycle engine and its intake and exhaust system as an air pumping device. The typical value of volumetric efficiency for natural aspirated engines is in the range between 80 to 90 % (Heywood, 1998). Beside heat transfer, volumetric efficiency is one of the factors that control the engine performance. Mechanical efficiency is the mechanical losses in an engine. The losses occur at mechanical movement such as piston, bearing, power absorbed by engine auxiliaries, ventilating action by the flywheel and work charging cylinder with fresh charge and discharging the exhaust gas during the exhaust stroke. In general mechanical efficiencies of engine vary from 65 to 85% (Ganesan, 2003). According to Heywood the typical values for modern automotive engine at wide-open throttle is 90% at speed within the range of 1800 – 2400 rpm, decreasing 75% at maximum rated speed and as engine is throttled, mechanical efficiencies decrease, eventually to zero at idle operation.

2.5.2 Emissions

(a) Emission Requirements

As long as there are combustion engines, pollutants will be produced. Therefore, it is important to reduce their volume by as much as possible. All hydrocarbon based fuels are mostly composed of hydrocarbons. If a complete and uniform combustion can be achieved, there would be only be water (H_2O) vapour and carbon dioxide (CO_2) produced which are both harmless. The physical characteristics of engines and the constituents of fuels do not allow these ideal conditions to be realised.

Vehicle exhaust emissions are identified as the major source of air pollution in urban areas. The causes of air pollution are many and complex. Urban expansion and rapid population growth have resulted in more vehicles and contributing to worsening environment. Realising this, most government and local authorities are planning to tighten or introduce stricter emission regulations for vehicles. The level of emissions from vehicles can be reduced by establishing strict emissions standards and making sure that the vehicles continue to meet these standards throughout their useful life. In order to comply with these emissions regulations, either more advance aftertreatment emissions control devices have to be incorporated into the vehicle or to use some alternative fuels which are capable of providing cleaner combustion.

The automotive air-pollution became apparent in 1940s in the Los Angeles basin. In due course it became clear that the automotive was a major contributor to hydrocarbon and oxide of nitrogen emissions, as well as the prime cause of high carbon oxides levels in urban areas (Heywood, 1988).

The principal harmful components are CO, HC and NO_x. Not everything comes out from the exhaust is toxic. The percentage of CO_2 indicates the degree of combustion. For Lambda = 1, the CO_2 percentage should be 14% by volume. The nearer we approach it, the better. Alternative fuels are found to have good potential in solving the emission problem, and helping to reduce urban CO and ozone levels (Walsh, 1989).

(b) Trophospheric Ozone

The formation of photochemical smog as a result of chemical reactions involved with both hydrocarbons and nitrogen dioxides in the presence of sunlight. HC facilitate smog formation, they are simply the by-product of incomplete combustion or lack of oxygen. It is powerful oxidant and irritant. The major strategy for reducing smog is to restrict the HC emissions and NO_x control is also necessary. There is critical need to consider controls on both NO_x and reactive HC if overall oxidant levels are to be lowered (McRae et al., 1987).

It is a powerful oxidant and irritant. It causes numerous short and long term effects on heart and lung healths. The immediate effects of excessive amount of photo smog causes individuals suffer of irritation, cough and chest discomfort, headaches, upper respiratory illness, increased asthma attacks and reduce pulmonary function as a result of this problem (Walsh, 1989; Lee, 1992).

The testimony presented by Thomas (1987) also reflected the current air quality standard tends to understate the health effects that elevated ozone concentrations occurring on some days during the hot summers in many of our urban areas reduce lung function, not only for people with pre-existing respiratory problems, but even for people in good health. This reduction in lung function may be accompanied by symptomatic effects such as chest pain and shortness of breath.

Some new studies also indicate that photochemical pollutants inflict damage on forest ecosystems and seriously impact the growth of certain crops (Mackenzie et al., 1988). Numerous studies also demonstrated that healthy young children suffer adverse effects from exposure to ozone at levels below the current air quality standards (Walsh, 1989).

(c) Carbon Monoxide (CO)

Carbon monoxide is a toxic gas formed by incomplete combustion of fuels. It exacerbates heart disease and compromise brain function. High level of carbon monoxide in urban air has been resulted entirely from motor vehicle emissions. The CO problem is critical because of the abundance evidence relating CO exposure to adverse health effects. A recent assessment revealed that 'these findings demonstrate that low levels of CO produce significant effects on cardiac function during exercise in subjects with coronary artery disease' (Walsh, 1989).

CO emissions are dangerous as even a very small amount can change the composition of the blood and it combines easily with the haemoglobin, allowing less oxygen to be absorbed into the bloodstream. Another finding concluded that 'Given the magnitude of the effect that we have observed for a very prevalent cause of death, exposure to vehicular exhaust, in combination with underlying heart disease or other cardiovascular risk factors could be responsible for a very large number of preventable death' (Walsh, 1989). Moreover, some recent evidence also indicates that CO may contribute to elevated levels of tropospheric ozone (McDonald et al., 1987).

(d) Oxides of Nitrogen (NO_x)

Emissions of NO_x are formed during combustion and causes regional acid deposition problems. Oxides of nitrogen emitted to the atmosphere is eventually deposited in downwind areas, in the form of nitrate particles or dissolved in rain. Next to sulphur dioxide, NO_x emissions are the most prominent pollutant contributing to acidic deposition. The deposition is responsible for the damage to vulnerable ecosystems due to acidification. NO_x emissions have also been shown to effect vegetation adversely. Some scientists speculate that NO_x is a significant contributor to the dying forests throughout central Europe (Whetstone et al., 1983). NO_x emissions are responsible for approximately one-third of the acidity of rainfall.

Further NO_x emissions have been known to cause damage and deteriorate effects to a wide range exposed materials, including textiles dyes, plastics, rubber and is responsible for a portion of the brownish coloration in polluted air or smog. NO_x emissions produce a variety of adverse environmental effects as mentioned above, it also produce adverse health effects. NO_x emissions also react chemically with other pollutants to form ozone and other toxic pollutants (Walsh, 1989). NO_x emissions are generated by the amount of high energy coming from the nitrogen and oxygen and they are very toxic. At high levels, they impair breathing and are the cause of intoxications (Bosch-Test Technology Catalogue). As reported by Lindvall, exposure to NO_x emissions is linked with increased susceptibility to respiratory infection, increased airway resistance in asthmatics, and decreased pulmonary function (Lindvall, 1982).

It is demonstrated that most areas of US under short term exposures to NO_x have resulted in a group of school children suffering from a wide range of respiratory problems, as well as increased sensitivity to bronchoconstrictors by asthmatics (Orehek et al., 1976; Mostardi et al., 1981). These effects are even more adverse when nitrogen dioxide and sulphur dioxide occur simultaneously.

(e) Carbon Dioxide

Concern about the increasing release of greenhouse gases such as CO_2 has grown out of research that documents the buildup of gases in the atmosphere and estimates the implications of continued accumulations. Carbon dioxide is largely transparent to incoming solar radiation, but can absorb infrared radiation reemitted by the Earth. Because of this energy trapping property, such a gas is referred to as a greenhouse gas (Department of Energy, 1991).

(f) Hydrocarbon

The pollutants commonly classified as hydrocarbons are composed of a wide variety of organic compounds. They are discharged into the atmosphere when some of the fuel remains unburned or is only partially burned during the combustion process. Most unburned hydrocarbon emissions result from fuel droplets that were transported or injected into the quench layer during combustion (Heywood, 1998). This is the region immediately adjacent to the combustion chamber surfaces, where heat transfer outward through the cylinder walls causes the mixture temperatures to be too low to support combustion.

Partially burned hydrocarbons can occur for a number of reasons:

i. Poor air and fuel homogeneity due to incomplete mixing, before or during combustion
ii. Incorrect air-fuel ratios in the cylinder during combustion due to maladjustment of the engine fuel system.
iii. Excessively large fuel droplets (diesel engines)
iv. Low cylinder temperature due to excessive cooling (quenching) through the walls or early cooling of the gases by expansion of the combustion volume caused by piston motion before combustion is completed.

All of these conditions can be caused by either poor maintenance or faulty design. Therefore, the lowest emissions will be achieved only by proper maintenance of engines designed specifically for low emissions.

2.5.3 Emissions of Engine

There are four main parameters which to the emission of the engine including chemical composition of the fuel, homogeneity of the air-fuel mixture, air-fuel ratio and ignition timing. Air- fuel ratio has the greatest influence on emission. The main components of emission are CO, CO_2, NO_x and HC. The normal operating range for a conventional SI engine using gasoline fuel is $12 \leq A/F \leq 18$. Lean burn occurs when the air/fuel ratio is greater than stoichiometric. The relative air/fuel ratio (λ) is obtained by dividing the actual air/fuel ratio by the stoichiometric air/fuel ratio. Lean operations ($\lambda >> 1.0$) can significantly decreased NO_x and CO emissions.

A research at Lotus found the test result on CNG at stoichiometric conditions showed emissions reductions of 60% HC, 26% NO_x, and 21% CO_2 and the torque reduction of 9% compared to gasoline (Durrel et al., 2000). The exhaust, CO emission depends on availability of oxygen. In other word fuel - air ratio (F/A) is important to determine CO level. If F/A is less than 1 (lean mixture, it means there is more oxygen than required to burn fuel. If F/A is bigger than 1 (rich mixture), there is less oxygen than necessary to burn all the fuel. When F/A is equal to 1, there is just enough air is allowed to burn all the fuel.

Engine powered by gaseous fuel like natural gas have lower CO concentrations in the exhaust for the same air-fuel ratio, because of higher hydrogen to carbon ratio and more homogeneous mixtures and can be operated much leaner. According to Aslam et al. (2005), CO produces by CNG is less 80% compared to gasoline and it is believe it came from the characteristic of CNG that they mix better during the operation. Bangale et al. (1995) carried out another comparative study of exhaust emission. The authors found that CNG reduces CO_2 emission 81% compared to gasoline. At moderate level of concentration, CO_2 is not considered an air pollutant. However, it is considered the major greenhouse gas. At higher concentrations, it is major contributor to global warming. CO_2 is the major component of the exhaust in the combustion of any HC fuel. Because of the growing number of motor vehicles, along with more factories and other sources, the amount of carbon dioxide in the atmosphere continues to grow. At upper elevations in the atmosphere, this higher concentration of CO_2 along with other greenhouse gases creates a thermal radiation shield. This shield reduces the amount of thermal radiation energy allowed to escape from the earth, raising slightly the average earth temperature. The most efficient way of reducing the amount of CO_2 is to burn less fuel (i.e., use engines with higher thermal efficiency).

Aslam et al. (2005) and Bangale et al. (1995) were found the same percentage of CO_2 concentration reduction occurs while operating with CNG compare to gasoline. The CNG produces less concentration by 50%. On the other hand, the mechanisms which cause HC emission can be defined as follows,

- The part of the air-fuel mixture not reached by the flame front, such as the charge coming from crevice volume between piston, rings and cylinders.

- Flame quenching at the combustion chamber walls, leaving a layer of unburned air-fuel mixture adjacent to wall.

- Absorption of fuel vapour into oil layers on the cylinder during intake and compression strokes and desorption of fuel vapour into the cylinder during the expansion and exhausts strokes.
- Incomplete combustion because of slow flame front.

Bangale et al. (1995) and Aslam et al. (2005) were found that the same amount of reduction of HC while operating by CNG compare to gasoline. Both research shows that the reduction of HC emission is 50%. According to Stone, 1998, NO_x are reactive with hydrocarbons resulting smog formation. It is also toxic itself. Although, emission of CO and HC is related to incomplete combustion, NO_x emission mainly depends on combustion temperature and combustion duration. While NO and NO_2 occur together, NO is predominant one. NO nominally accounts for approximately 90% of total NO_x. The rate of formation of NO is primarily dependent on the temperature of burned gases. It is obvious that the temperature control is the main parameter in the control of NO_x. The compression ratio, air-fuel ratio, fuel composition, ignition timing and combustion mixture diluents such as exhaust gases define the temperature of combustion. Exhaust gas acts as a diluter of the inlet charge and reduces the peak flame temperature. Higher heat capacity of CO_2 and H_2O provides exhaust gas more effective than same amount of air (Stone, 1999).

Time is the second factor on the occurrence of NOx. The contribution of NO_x formation is more significant when the mixture is burned early. Because the early burned mixture has more time to reach equilibrium. Another appearance of time factor is at different engine speeds. At higher engine speeds contribution of NO_x is lower due to the less time required to achieve the engine cycle. It is found that higher NOx concentration produce by the engine while operating with CNG compare to gasoline (Kalam et al., 2005 and Bangale et al., 1995). The range is between 22-33%. According to Aslam et al. (2005) more reduction of NOx concentration can be achieved by using TWC.

2.6 Summary

This chapter is dedicated the literature findings suitable to this study. Primarily reviews have been made on world fuel supplies, the fact why CNG is suitable as alternative fuel and CNG characteristics. The other part of review is focus on previous research performed on CNG vehicles, included research on its performance and emissions. The descriptions have stressed out some findings from other experimental research. The next chapter of thesis will explain clearly about the research methodology and experimental procedure.

CHAPTER 3

METHODOLOGY

3.1 Introduction

The performance and emissions of an engine while running on CNG are major concerns of the research. The dedication of this chapter is to present the experimental methods for the performance and emissions of the engine. Experimental setup describes engine test rig, important parameters and measurement technique including cylinder pressure, rate of air and fuel consumptions, engine speed, engine torque and engine emission. The experiments, gasoline and CNG were performed without any modification.

3.2 Overall Research Methodology

The flowchart of research methodology in this study is shown in Figure 3.1. Chronologically, the first activities of the research is designing and developing the test rig of the engine followed by installation of experimental apparatus included apparatus for engine performance and apparatus for exhaust emissions, calibration, equipment functionality test and experiment. The first experiment is performed using gasoline as fuel, followed by CNG as fuel with the same condition as performed with gasoline. After all the data being collected, the differences of engine performance and emissions of gasoline and CNG were analyzed.

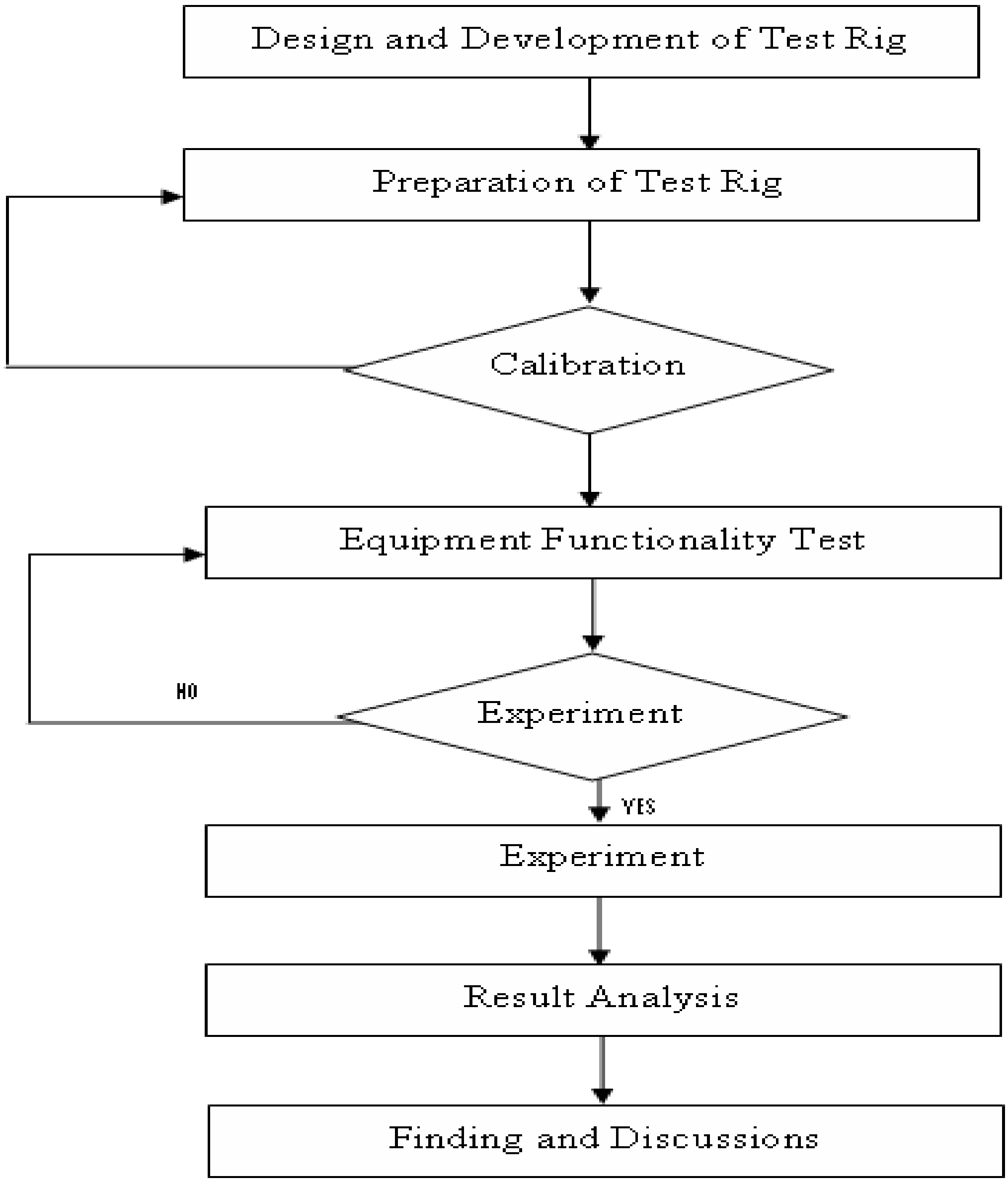
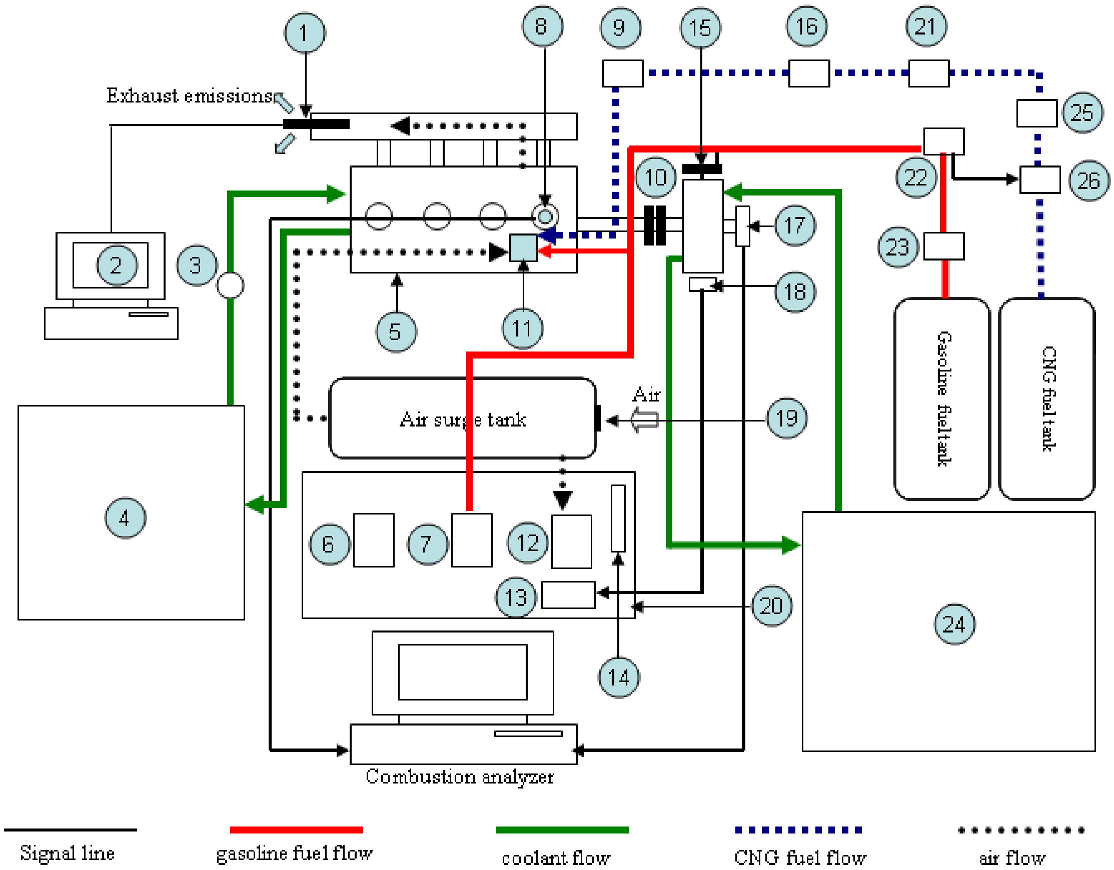


Figure 3.1: Flowchart of the Studies

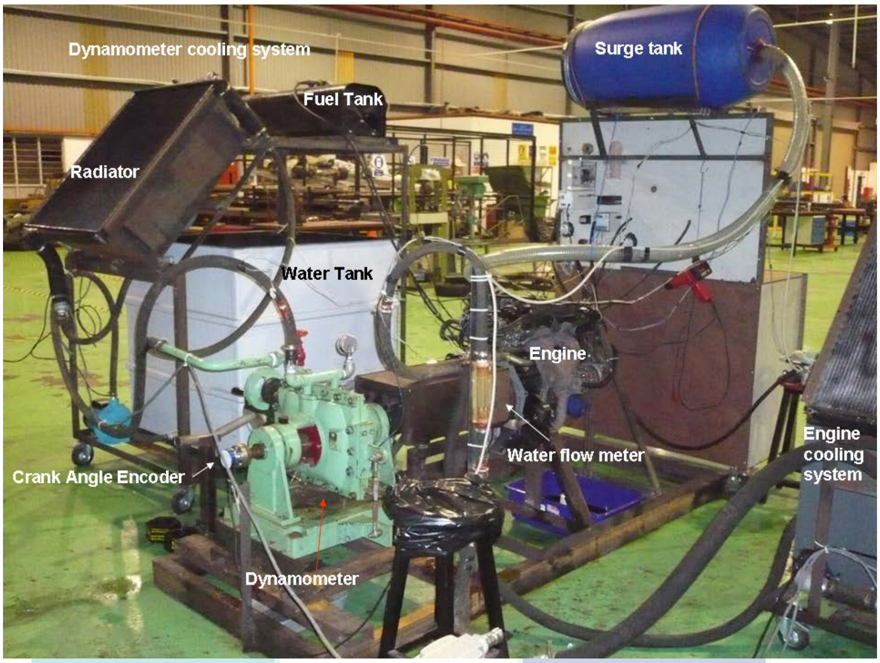
3.3 Experimental Apparatus

The experimental apparatus employed for all experimental investigation covered in this project include (i) apparatus for engine performance and (ii) apparatus for engine combustion analysis and (iii) engine exhaust emission study. The schematic diagram in Figure 3.2 shows the general layout of test engine, instrumentation and sensor.



1. Emission probe
2. Exhaust gas analyzer
3. Water flow meter
4. Engine cooling system
5. Engine
6. Temperature sensor
7. Fuel flow meter
8. Pressure transducer
9. Mixer
10. Engine-dynamometer coupling
11. Carburetor
12. Manometer
13. Load indicator
14. Throttle controller
15. Dynamometer controller
16. CNG flow meter
17. Crank angle encoder
18. Load sensor
19. Orifice
20. Instrumentation panel
21. Power valve
22. Change-over switch
23. Cut-off solenoid valve
24. Dynamometer cooling system
25. Pressure regulator
26. High pressure gauge

Figure 3.2: Schematic diagram of the engine test-rig



(a) Rear view of original test rig

(b) Front view original test rig

Figure 3.3: Figure of original test rig

3.4 Description of Apparatus

The engine performance characteristics studies were performed at the Automotive Laboratory, Faculty of Mechanical Engineering Universiti Malaysia Pahang. Proton Magma 4G15 engine is considered in this study. This is four-stroke, four cylinder type engine. The engine specification gives in Table 3.1. The engine specification remained unchanged except the cooling system. It was modified to incorporate portable systems.

Table 3.1: Proton Magma 4G15 engine specification

Parameter	**Specification**
Model	4G15P naturally-aspirated
Engine Type	Spark-ignition
Cubic Capacity (cm^3)	1,488
Cylinder bore (mm)	75.5 x 82
Piston stroke (mm)	82
Compression ratio	9.2:1
No. of cylinders and valves per cylinders	4 cylinders with 3 valves per cylinder
Maximum output (DIN) PS/rpm	86/6000
Net (kW/rpm)	(64/6000)
Maximum torque (DIN) kg.m/rpm	12.5/3500
Net (Nm/rpm)	122/3500
Carburetor	Down-draft 2-barrel

A Froude type hydraulic dynamometer ATE – 60 LC with a maximum capacity of 170 brake horsepower, operating range of 5501 - 7500 rpm was used to maintain the variation of loads at different speeds. Engine power absorption was through an impeller keyed to the main shaft. The main shaft is mounted on rolling contact bearings housed in stator casing. Brake power output was measured with a pneumatic load cell with pressure indicator which was connected to the dynamometer. The dynamometer was controlled manually which enabled the engine to be run in loading mode only: the dynamometer was used purely as a dynamometer and absorbed the power generated by the engine, for power measurement. As the dynamometer is not incorporated with any automatic regenerative system it cannot perform motoring function such as an electric motor.

Measurement of air flow rate going into internal combustion engine always presents some difficulties since the flow of air is intermittent or pulsating in the inlet manifold to the cylinder (Ganesan, 2004). In this project a big cylindrical surge tank or quieting tank with capacity 0.1885 m^3 is installed to draw the intake air from ambient through a calibrated hose with an internal diameter of 50mm to inlet manifold. At the other side of the cylinder is an orifice a diameter of 100mm.

The air flow rate into the inlet manifold is measured by a manometer, comprising of a hose branched out from a 50 cm main pipe immediately before entering the air filter. The pressure drop across this calibrated is used in conjunction with the calibration with calibration chart or calibrated equation to calculate the actual air flow into the engine. The calibration chart was procured through the manual calibration of pressure drop against an anemometer which gives a direct reading of air flow velocity in m/sec covering the full range of flow rate interested. To measure the flow rate of gasoline flow rate, a gravimetric method is introduced. It is used a bulb glass. This is a special made of 50 ml bulb glass tube (gravimetric method) and the measurement is performed with the aid of stop watch.

The flow rate of the CNG is measured by Yokogawa flow meter; model RAMC01-T4SS-62M1-T90NNNPT/SD/IE1 a metal short-stroke rotameter. It is installed between the power valve and mixer. The reading of the gas flow rate is taken from the electronic display panel. However manual reading is also available.

Water is used as a cooling medium. Cool water at estimated initial temperature of 22^0 C is supplied from the main water source to the engine through the inlet pipe, and it is discharged to the outlet pipe which is linked directly with the water tank. The hot water from the outlet will mix with the cool water having intermediate temperature of around 85^0C ($80\text{-}90^0C$), and the water keeps circulating between the engine and water tank. The water tank is placed at the same level with the engine, however the power of the pump will provide sufficient pressure head to meet flow requirement of the cooling water. A thermometer is mounted at the inlet and another at the outlet pipe, and a valve is used to control the inlet flow which in turn controls the temperature. This system is fabricated at the university workshop.

Water is used as cooling medium. Cool water at estimated temperature of 22^0 C is supplied from the main water source to the engine through the inlet pipe, and it is discharged to the outlet pipe which is linked directly with the water tank. The hot water from the outlet will mix with the cool water having intermediate temperature of around 65^0C, and the water keeps circulating between the dynamometer and water tank. The water tank is placed at the same level with the engine, however the power of the pump will provide sufficient pressure head to meet flow requirement of the cooling water. A thermometer is mounted at the inlet and another at the outlet pipe. The volume of the water filled inside this tank is $1m^3$. This system is also fabricated at the university workshop.

A vertical manometer is used for measurement of pressure drop across air surge tank. The total height of manometer is 1000 mm. A special type of liquid with density, ρ, 0.692 kg/m^3 is introduced to replace water. The pressure drop is directly taken from the manometer the unit of Pascal / Nm. Strobotester is used to measure the engine speed (rpm) and ignition timing manually. The brand of SINCRO model DG83-D is introduced.

3.4.1 Combustion Analysis

A unit of combustion analyzer, Dewetron CA was employed to study the phenomenon of engine combustion. Figure 3.4 shows Dewetron CA combustion analyzer. Crank position obtained from high resolution encoder through Kistler 2613B1. Measurement of combustion process is accessible using the combustion pressure sensor. The latest design of the sensor allowed the sensor to be inserted in the spark plug like adapter which design for dual-functional operation; to supply spark and to measure the combustion pressure. The installation of the spark plug type pressure sensor is presented in Figure 3.5**.** It is installed in the first engine cylinder of the engine**.** The usage of combustion pressure sensor is made simpler with the integration of Dewetron CA 5000, a computerized based combustion analyzer. The voltage signal from the sensor is directly send to the Dewetron CA software and interpreted as the pressure data.

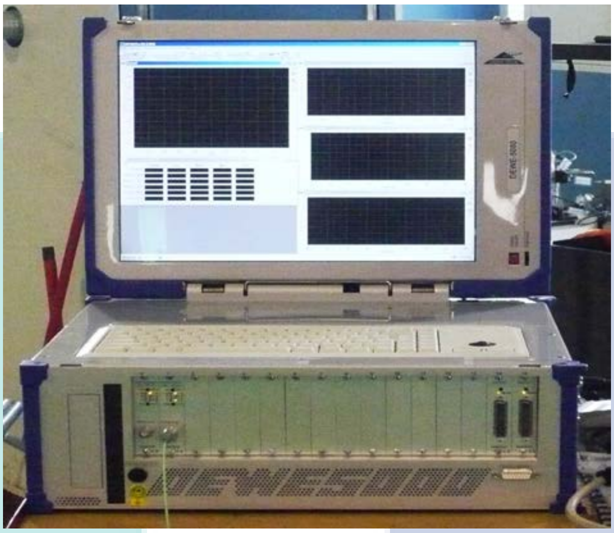


Figure 3.4: Dewetron CA combustion analyzer

Figure 3.5: Installed pressure sensor spark plug

Measurement of speed is accomplished using the Kistler CAM system of the encoder of Kistler 2613B, the signal conditioner 2613B1, and the line terminator which is connected to a measuring instrument. The encoder is installed on a bolted adaptor located at the end of the dynamometer shaft (Figure 3.6). The flexible coupling enables the encoder to function well under the vibratious condition.

Figure 3.6: Bolted crank angle encoder

Figure 3.7: Crank angle encoder and data acquisition kit

A measuring spark plug of Kistler 6117BCD15 is installed to the first engine cylinder to measure the pressure rise and peak pressure in the cylinder. This type of spark plug can be installed as a normal spark by using the mounting key to avoid from the damage because of over screw. Coupled usage of the crank angle encoder and the spark plug type of combustion pressure sensor during cranking phase will allow determination of the zero crank angle position based on the maximum compression pressure during non-firing condition.

Figure 3.8: Pressure sensor spark plug

3.4.2 Exhaust Gas Analysis

A KOEN exhaust gas analyzer of model KEG500 was used to measure the concentration of gas species, such as NO_x, HC, CO, CO_2 and O_2. There two other measurements can be made by this equipment, air surplus rate (lambda – λ) and the air fuel ratio (AFR).

The technology of this analyzer consists of automatic measurement with microprocessor control and self test, auto-calibration before every analysis, high degree of accuracy of analysis.

The probe of the exhaust gas analyzer was inserted inside the exhaust for a few second to get the data of the emission. All the data will be printed automatically by a built in printer. After self calibration and at rest the exhaust gas analyzer show the value of zero of percent of CO, CO_2, HC and AFR but the value of O_2 is shown as 20.75%. It is mean that the equipment is in the right condition and is ready to use as shown in the Figure 3.9.

Figure 3.9: KEG-500 exhaust gas analyzer

3.5 Experimental Procedure

The experimental procedure of the study is divided into three sections (i) procedure for engine dynamometer evaluation, (ii) procedure for exhaust gas analysis and (iii) procedure for engine combustion analysis.

All investigations with respect to engine performance, fuel consumption and exhaust emission are carried out both with gasoline and CNG. However, instead of covering the whole engine operating range, this experiment only cover a range of speed from 1500rpm to 4000 rpm at wide open throttle (WOT) setting. This is mainly to prevent unnecessary breakdown of the engine halfway through the running test as this can be perceived from the age of this particular test engine.

Adjustment were made as per manufacture's instructions before running the test and the engine ran-in before running according to the manufacturer's recommendation, or otherwise the engine ran-in until corrected brake power is maintained within a variation of 1% over a period of eight (8) hours running. Prior to engine test, the following readings are required to be recorded:

- Engine make
- Engine model
- Engine serial number
- Type of fuel system
- Compression ratio
- Test location
- Test date
- Test number

Prior to commencement of each engine test and before any data being taken or recorded, the engine is maintained at part load until the coolant temperature is brought up to normal operating temperature. In this project the coolant temperature is maintained at 80 +/- 5^0C.

The steady state tests comprises of running the engines until steady condition attained. Data were collected either automatically by computer acquisition system, DEWE-CA 500 or manually then stored the acquire data in the computer memory for further analysis. The readings taken are:

- Engine speed
- Engine Torque
- Cylinder pressure
- Air and fuel consumptions
- Temperature of the engine cooling water
- Ignition timing
- Ambient temperature
- Temperature of engine cooling Water

The parameters such as engine speed, engine torque, cylinder pressure, air and fuel consumptions and air flow rate is considered as an important parameter to be measured. The other parameter such as temperature of engine cooling water and temperature of dynamometer cooling water is to avoid overheating while performing the experiments. Ambient temperature is a parameter to be controlled to ensure the experiment ran under the right condition.

The dynamometer water supply is pressurized in the range of 1.0 – 1.75 bar for operation ranging from 1000 to 6000 rpm. In order to supply the water as well as to release the heat generated by the dynamometer operation, a customized cooling system that consist a heat exchanger, cooling fan and special water reservoir are connected with the outlet water pipeline. As for the engine sufficient flow rate of water is required to cool down especially when running at full power. Mass flow rate of water in the range of 4 to 15 litres per minute is supplied accordingly appropriate with engine speed and load.

The mass flow rate of gasoline fuel is determined by the time taken over fuel consumption from a 50 ml bulb glass tube, a gravimetric method with the aid of stop watch. Yokogawa flow meter is used (Model RAMC01-T4SS-62M1-T90NNNPT/SD/IE1) to measure the CNG flow rate. The gases that are analyzed are CO, CO_2, HC and excess oxygen using a unit of exhaust gas analyzer, model KEG 500. The exhaust gas calibration procedure consists of three phases, (i) zero calibration, (ii) span calibration and (iii) checker trimmer adjustment.

The analyzer was warm up for a few minutes in PURGE mode while the probe was exposed to clean air, before calibration. For zero calibration, with analyzer still in PURGE mode, meter reading for HC, CO, CO_2 and O_2 were set to zero using AUTO ZERO key. Subsequently the span calibration was performed. The analyzer was set in the standby mode and a span gas container nozzle was pressed into the span gas inlet on the analyzer for 10 seconds. Next the SPAN POT for HC, CO, CO_2 and O_2 were adjusted to give meter readings indicated by the span gas composition and the analyzer in the PURGE mode and the meter readings of HC, CO, CO_2 and O_2 set to zero, the checker trimmer was adjusted to give readings of 5000+/- 150 ppm, 5+/-0.14% vol. and 10+/-0.45% vol. respectively while the SPAN CHECK key was pressed.

3.6 Performance Parameter

There are five important performance parameter of the engine is presented in this section. They are (i) brake power (P_b), (ii) brake mean effective pressure (BMEP), (iii) air fuel ratio, (iv) brake specific fuel consumption (BSFC) and (v) engine efficiency. Engine efficiencies are included brake thermal efficiency, mechanical efficiency and volumetric efficiency.

3.6.1 Brake Power

Engine torque is measured with a dynamometer. The engine is clamped on a test bed and the shaft is connected to the dynamometer rotor. The rotor is coupled hydraulically to a stator which is supported in low friction bearings. The stator is balanced with the rotor stationary. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights. The torque exerted by the engine is *T:*

$$T = Fs \tag{3.1}$$

The power *P* delivered by the engine and absorbed by the dynamometer is the product of torque and angular speed. It known as brake power**,** $P_,$

$$P = \frac{2\pi NT}{60 \times 10^4} \tag{3.2}$$

Refer to Heywood, 1988 the correction factor $C_{F,}$

$$C_F = \frac{P_{s,d}}{P_m - P_{v.m}} \left(\frac{T_m}{T_s} \right)^{1/2} \tag{3.3}$$

Where $P_{s,d}$ is standard dry-air absolute pressure , P_m is measured ambient-air absolute pressure, $P_{v,m}$ is measured ambient-water vapour partial pressure, T_m is measured ambient temperature and T_s is standard ambient temperature, K

Standard condition according to European and American practice: atmosphere pressure 1 bar = 750 mmHg at temperature 25^{o}C (298 K), relative humidity, 30%.

The rated brake power is corrected by using Equation (3.3) to correct the indicated power and making the assumption that friction power is unchanged. Thus

$$P_b = C_F P_i - P_f \tag{3.4}$$

Where P_b is brake power (standard), P_i is indicated power measured by DEWE-CA500 and P_f is friction power measured by DEWE-CA 500

3.6.2 Mean Effective Pressure

Mean effective pressure is the average pressure inside the cylinders of an internal combustion engine based on calculated or measured power output. It increases as manifold pressure increases. For any particular engine, operating at given speed and power output, there will be a specific indicated mean effective pressure *(IMEP)* and a corresponding brake mean effective pressure *(BMEP).* They are derived from the indicated and brake power respectively.

$$BMEP = \frac{P_b n_r \times 10^3}{V_d N} \tag{3.5}$$

3.6.3 Air-Fuel Ratio

The relative proportions of the fuel and air in the engine are very important from the standpoint of combustion and the efficiency of the engine. This is expressed either as ratio of the mass of the fuel to that of the air or vice versa. In the SI engine the fuel-air ratio practically remains a constant over a wide range of operation.

Air flow rate

$$m_a = \frac{C_d \pi d^2}{4}\left[\sqrt{\frac{2(\Delta P)}{\rho_{air}}}\right] \tag{3.6}$$

where, $C_d = 0.64$, $d^2 = (0.1)^2$, $\Delta P = pressure\ (Pa)$

Air density

$$\rho_{air} = \frac{P_{atm}}{R \times T_{ambient}}, \tag{3.7}$$

where, $P_{atm} = 101325Pa$, $R = 287\,J/kgK$, $T_{ambient}$ is temperature during experiment

Air flow rate in kg/s

$$\dot{m}_a = \rho_a \times m_a \tag{3.8}$$

Fuel flow rate in kg/s

$$\dot{m}_f = \frac{0.0384}{t_f}, \tag{3.9}$$

Where t_f is time for fuel consumption

Thus, air-fuel ratio is the ratio of air flow rate and fuel flow rate.

3.6.4 Specific Fuel Consumption

The fuel consumption characteristics of an engine are generally expressed in terms of specific fuel consumption in kilograms of fuel per kilowatt-hour (kg/kWh). It is an important parameter that reflects how good the engine performance is. It is inversely proportional to the thermal efficiency of the engine.

$$sfc = \frac{\dot{m}_f}{P} \tag{3.10}$$

$$\text{Thus, } BSFC = \frac{\dot{m}_f}{P_b} \tag{3.11}$$

3.6.5 Efficiency

It is common practice in engineering to establish a figure of merit for a device by comparing the actual performance of the device with the performance it would have had under some arbitrary set of ideal conditions. The ratio of actual performance to the ideal performance is called the efficiency of device. Since the ideal performance is usually unattainable, the efficiency is usually less than 1. In internal combustion engines, one of the most important efficiencies is the thermal efficiency, which is define as

$$\text{Thermal efficiency, } \eta_t = \frac{\text{Work deliver by the engine}}{\text{Chemical energy in the fuel}}$$

$$= \frac{\text{Work output}}{\text{Fuel energy input}}$$

Brake Thermal Efficiency,

Brake thermal efficiency is the ratio of brake power *Pb*, to the input fuel energy in appropriate units.

$$\eta_t = \frac{P_b}{\dot{m}_f \times Q_{LHV}} \times 100\% \tag{3.12}$$

Mechanical Efficiency

Mechanical efficiency is defined as the ratio of brake power, P_b (delivered power) to the indicated power (power provided by the piston). It can also defined as the ratio of brake thermal efficiency to the indicated thermal efficiency,

$$\eta_m = \left(\frac{P_{b,m}}{P_{i,g}}\right) \times 100\% = \left(1 - \frac{P_f}{P_{i,g}}\right) \times 100\% \qquad (3.13)$$

Volumetric Efficiency

This is one of the very important parameters which decide the performance of four-stroke engines. Four stroke engines have distinct suction strokes and therefore the volumetric efficiency indicates the breathing ability of the engine. It is to be noted that the utilization of the air is what going to determine the power output of the engine. Hence, an engine must be able to take in as much air as possible.

Volumetric efficiency is defined as the volume flow rate of air into the intake system divided by the rate at which the volume is displaced by the system.

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} \times 4V_d \times N} \times 100\% \qquad (3.13)$$

Where, $\dot{m}_a$ unit in kg/s; $\rho_{a,i}$ unit in kg/m^3; $4V_d$ unit in m^3; N unit in rev/s

3.7 Summary

This chapter is described the procedure of experiments. In the first part of the content, (i) the engine test-rig is presented and (ii) the subscription of all important equipment and experimental methods has been described. In the next part of the chapter, the important parameter formulation of the study is presented such as brake power, brake mean effective pressure, brake specific fuel consumption and engine efficiency. The next chapter of the thesis is presented the discussions of the experimental results.

CHAPTER 4

RESULT AND DISCUSSIONS

4.1 Introduction

This chapter presents the experimental results of gasoline and CNG operated at wide-open throttle condition. In order to compare gasoline and CNG over a range of operating conditions it is necessary to adopt a common abscissa unit for graphing. Graphing results are included cylinder pressure against crank angle (CA) degree and engine performance. The performance of the engine are included torque, power, mean effective pressure, fuel consumption, efficiency, exhaust emission and air-fuel ratio against engine speed.

4.2 Cylinder Pressure

The evaluation of spark ignition engine is mainly based on cylinder pressure which is very much influenced by engine torque, speed, air-fuel ratio and ignition timing at compression ratio and throttle opening constant for wide-open throttle. In the study instantaneous cylinder pressure is measured at each engine crank angle for four-stroke cycle from -360^o to 360^o variation of cylinder pressure in bar with crank angle degree. The distribution of cylinder pressure against crank angle for gasoline and CNG are shown in Figure 4.1 and 4.2 respectively. It is clearly shown that, the peak cylinder pressure of CNG is lower than gasoline.

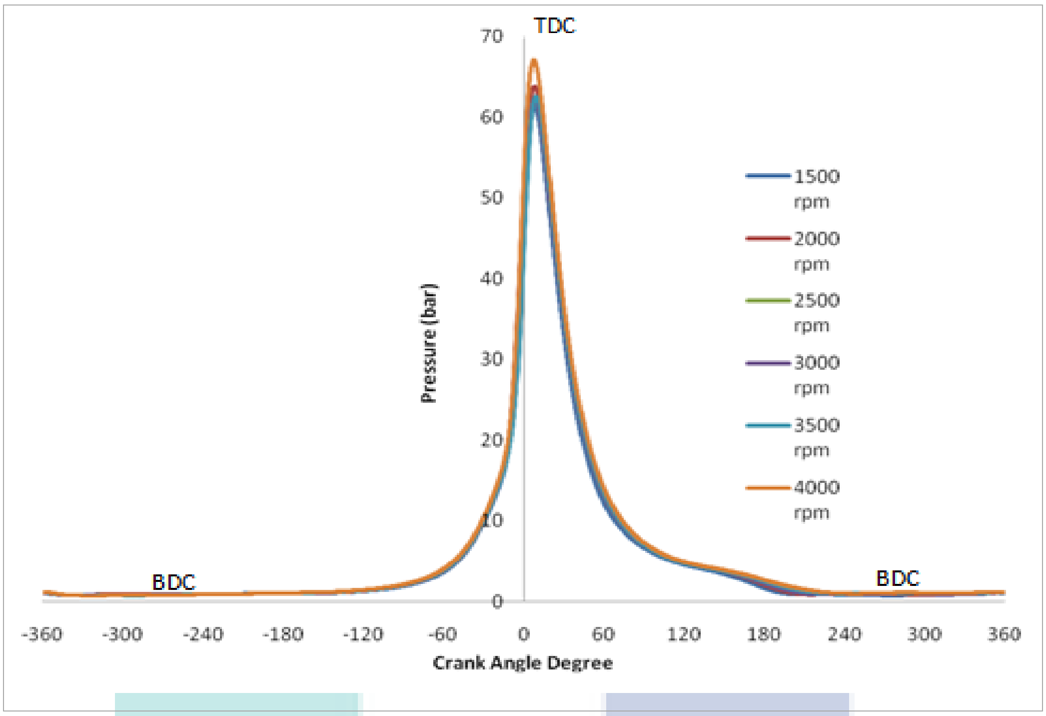


Figure 4.1 : Cylinder pressure distribution against crank angle degree of gasoline at wide-open throttle condition

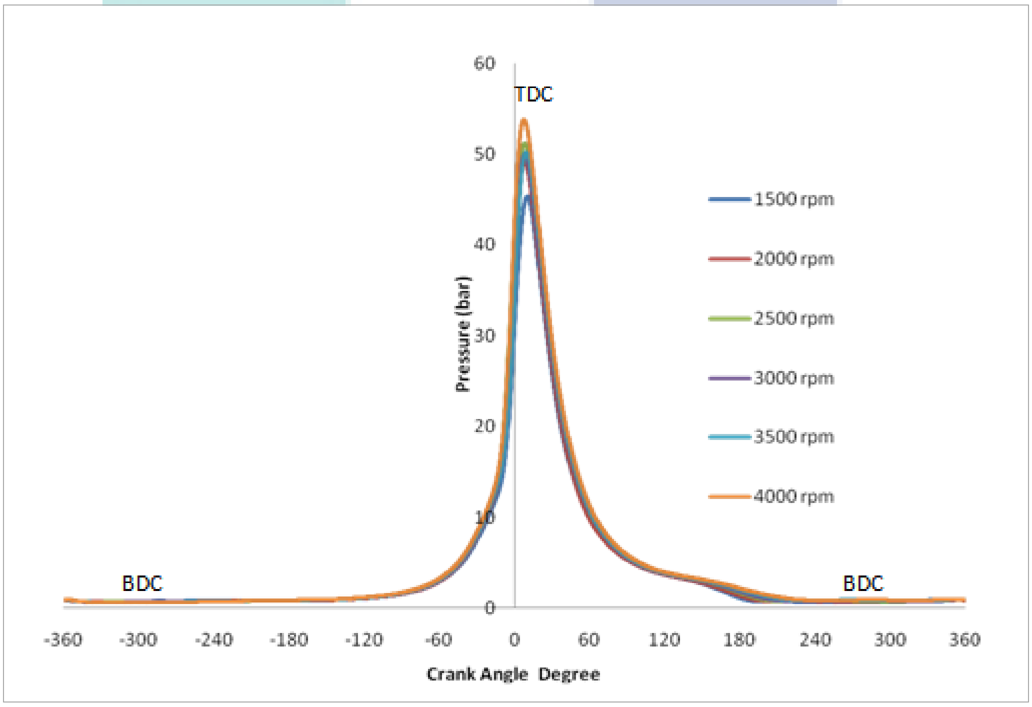


Figure 4.2 : Cylinder pressure distribution against crank angle degree for CNG at wide-open throttle condition

Comparing the peak pressure of gasoline and CNG, the reduction of the peak pressure of CNG is around 20% at different engine speed. This phenomenon gives an indication that the performance of the engine will be reduced as expected in the literature study. Principally, to obtain high peak pressure for gasoline, the timing of peak pressure must occurs as close as possible to top dead centre (TDC) point approximately 5-10^{o} CA after TDC and for 8 -12^{o} for CNG.

Table 4.1: Measured maximum peak pressure of gasoline and CNG at TDC

Engine speed (rpm)	Peak pressure of gasoline (bar)	Peak pressure of CNG (bar)
1500	55	44
2000	61	49
2500	64	51
3000	62	50
3500	62	50
4000	67	54

The experimental results show that the lowest peak pressure is produced at 1500 rpm of engine speed for both type of fuel. At 1500 rpm, gasoline fuel produces 55 bar cylinder pressure; air-fuel ratio, 13.4 and the engine torque, 101.6 Nm. At the same engine speed, CNG produced 44 bar cylinder pressure; air-fuel ratio, 14.2 and the engine torque, 75 Nm. The air-fuel ratio for both type of fuel is close to rich burn condition.

The highest peak pressure is produced at 4000 rpm of engine speed both gasoline and CNG. It is found that for the gasoline, air-fuel ratio, 13.0; and the engine torque, 107.8 Nm. The same trend also occur while the experiment used CNG. Its produce the highest peak pressure at 4000 rpm of engine speed with air-fuel ratio, 13.8 and the engine torque is 80 Nm. However, the torque is slightly low at high speed because of high friction. The air-fuel ratio also slighter richer at high speed because of more quantity of unburnt hydrocarbon occur at that stage.

The rate of pressure rise in an engine combustion chamber exerts a considerable influence on peak pressure developed, the power produced and the smoothnesswith which the forces are transmitted to the piston. The rate of pressure rise mainly depends upon the mass rate of combustion of mixture in cylinder. It is clear that the lower rate of combustion longer time is required for completion of combustion which necessitates the initiation of burning at an early point on the compression stroke. Also, the higher rate of combustion results the higher rates of pressure rise producing higher peak pressures at a point closer to top dead center.

4.3 Engine Performance

All of the results presented are based on the data collected from the experiment. There are a few results directly collected from the display panel or reading through the gauge such as load, fuel flow rate, air flow rate, emission, engine cooling temperature, dynamometer cooling temperature and ambient temperature. The first engine cylinder is used for the experiment. Combustion analyzer (Dewetron CA) gives the results of a cylinder that needed to multiply by four to obtained overall results. The correction factor is applied to obtain the precise result. It is included the calculation of the brake power, brake specific fuel consumption, mean effective pressure, engine efficiency etc., as mentioned in Subsection 3.7. The gas exhaust emission analyzer (KOEG 500) shows the air-fuel ratio on its panel and recorded manually into the log book while performed the experiment.

4.3.1 Air-Fuel Ratio

Air-fuel ratio is an important substance to the engine performance. The ratio of the air and fuel influences the resultant peak pressure and the timing for peak pressure consequently engine performance (Heywood, 1988). The air-fuel ratio is dependent of load and engine speed. For wide-open throttle condition, the air and fuel flow are increase as speed increase.

Figure 4.3 presents a variation of air-fuel ratio against engine speed at wide-open throttle for gasoline and CNG fuel. It can be seen that the air-fuel ratios of CNG are higher than gasoline throughout the engine speed. The maximum air fuel ratio is obtained at 3000 rpm for both fuels. The stoichiometric mixture obtained at maximum air fuel ratio. The maximum air fuel ratio for CNG and Gasoline are 15.1 and 12.3 respectively. According to Heywood (1988), the air-fuel ratios of gasoline and CNG at stoichiometric combustion are 14.6 and 16.79 respectively. The obtained experimental results agrees with Heywood and Kalam et al. (2005) results.

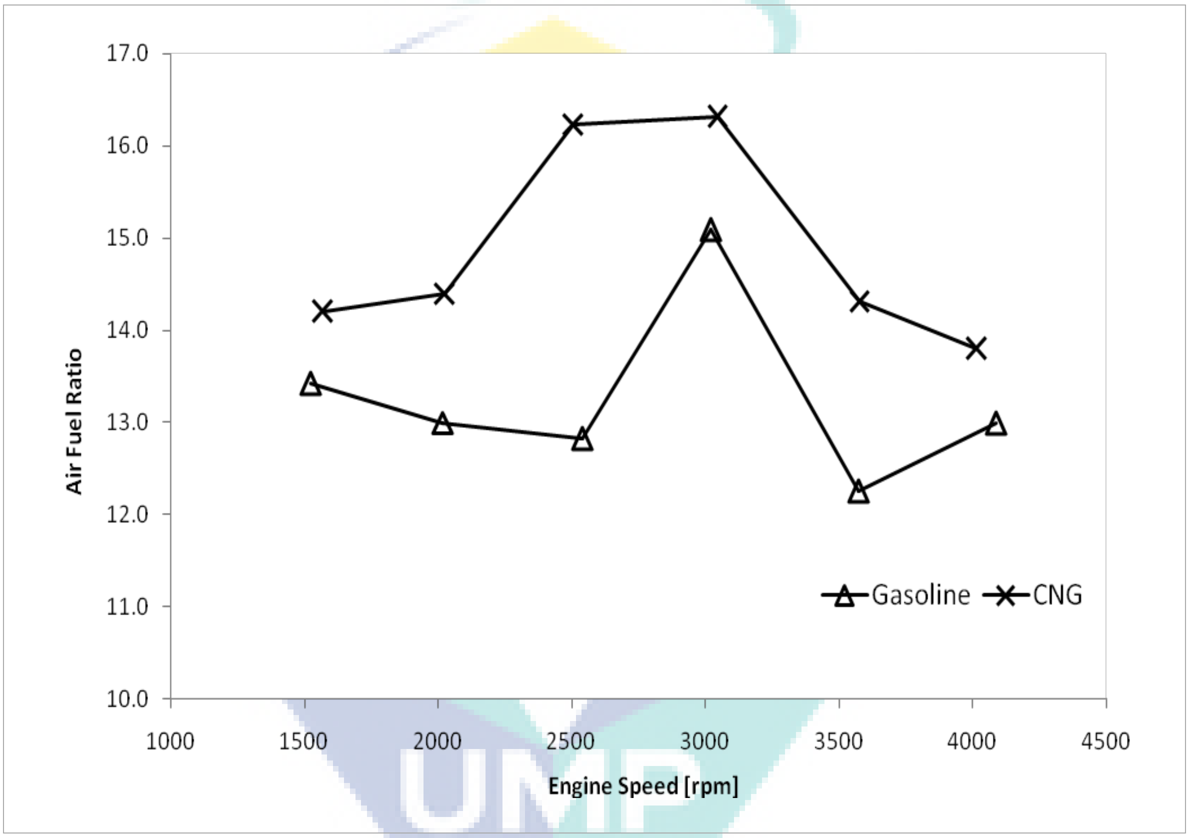


Figure 4.3: Air-fuel ratio at different engine speed for wide-open throttle condition

4.3.2 Torque

Maximum engine torque is imposed to ensure that the engine is operated at full power. Figure 4.4 presents the variation of engine torque with speed for gasoline and engine. The obtained results show that the range of torque for gasoline and CNG are 100-110 Nm and 75-82 Nm respectively. The maximum engine torque is achieved 110 Nm and 82 Nm at 2500 rpm for gasoline and CNG respectively. However, the torques produced by CNG, 26% lower than gasoline but the same trend is occurred, while the highest torque is achieved at the engine speed of 2500 rpm with the value of 82 Nm and the range of the torque of engine speed is between 75-80 Nm. The value of CNG maximum torque reduction 26 % is comparable with the finding of 21.3% reduction Bangale et al. (1995) (Table 4.2)

Table 4.2: CNG torque reduction compare to gasoline

Referred results	Maximum CNG torque reduction (%)
Bangale et al. (1995)	21.3
Experimental result	26

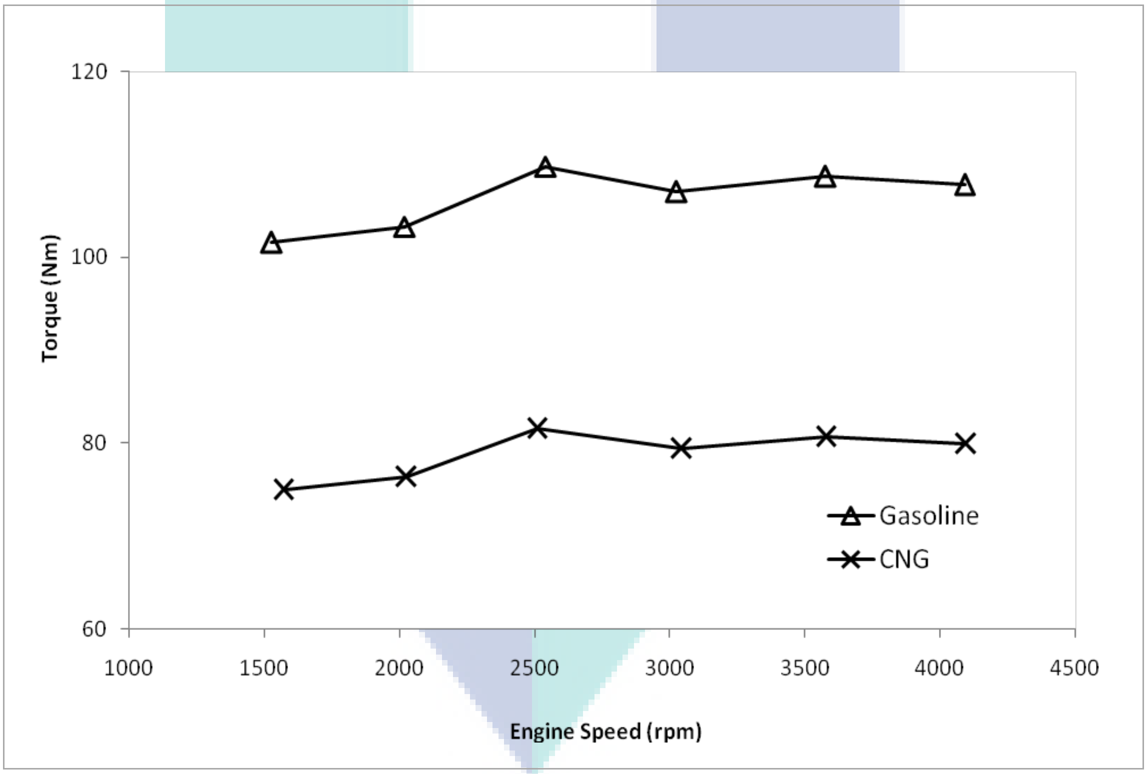


Figure 4.4: Torque at difference engine speed for wide-open throttle condition

The engine cannot withstand a higher load at high engine speed thus reduced the engine torque. This is due to the increase friction in engine components and it is dominant at high speed (Stone, 1999). More work produced in the engine cylinder is used to overcome the increased friction. However, the maximum torque is lower than the rated torque as given by the manufacturer specification which has the value approximately about 122 Nm at 3500 rpm (Table 3.1) and achieve earlier. This deviation is due the measured engine torque values have subtracted with inertia load imposed by the hydraulic dynamometer prior to the loading execution.

4.3.3 Power

In the previous chapter (Figure 2.4) is discussed about the energy distribution of chemical energy from the fuel. They are indicated power and brake power. Indicated power is based on indicated net work, thus a measure of the force available within the cylinder. The experiment indicated power is taken from the combustion analyzer of the data acquisition systems. The useful power that drives the vehicle is the brake power off-course lower than indicated power because of energy losses in friction, pumping, etc.

i. Indicated Power

Figure 4.5 is shown variation of indicated power in kW with engine speed at wide-open throttle condition. The indicated power increases with engine speed increases. It lies within the range of 16-52 kW and 13-41 kW for gasoline and CNG respectively. The reduction 19.5% of indicated power while operating with CNG. The indicated power is strongly affected to the brake power of the engine. The brake power is derived by minusing the friction power occur in the engine cylinder.

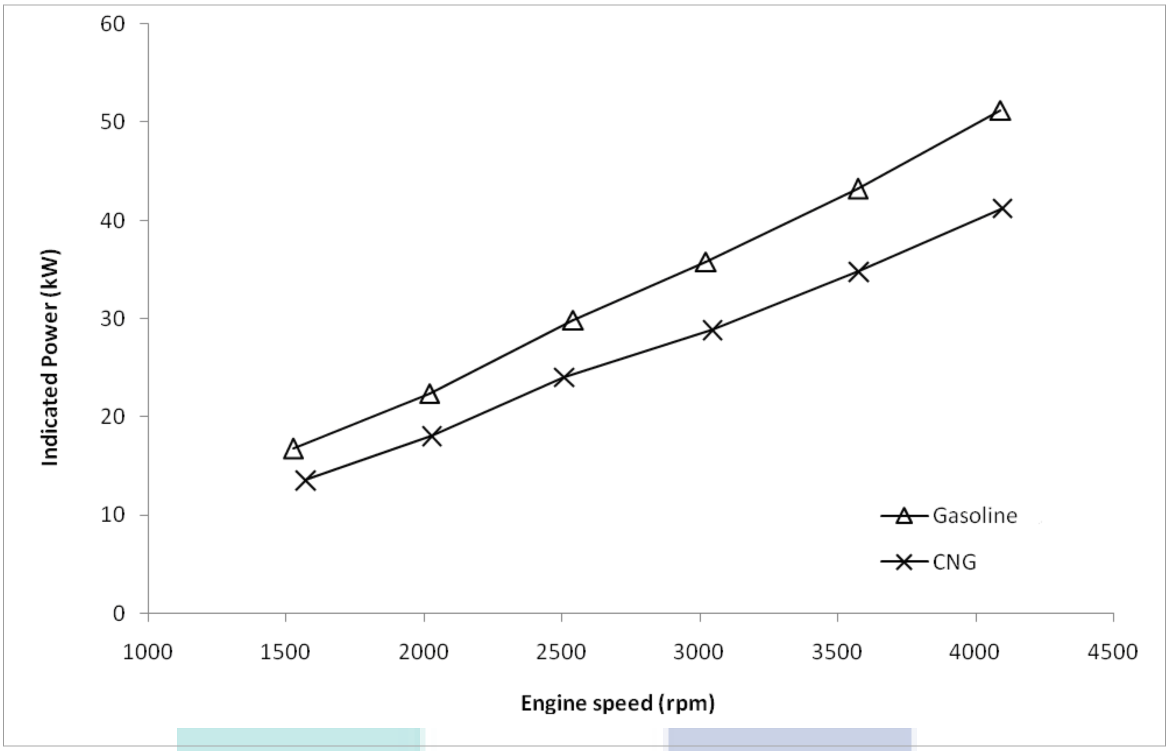


Figure 4.5: Indicated power at different engine speed for wide-open throttle condition

ii. Brake Power

Brake power is the output power of an engine measured by developing the power into a brake dynamometer on the output shaft. Figure 4.6 is shown variation of brake power in kW with engine speed at wide-open throttle condition. At wide-open throttle condition, the brake power for gasoline lies between 16 kW at 1500 rpm to 47 kW at 4000rpm and for the CNG, the brake power lies between 12kW at 1500 rpm to 35kW at 4000rpm. The graph shows that the brake power is proportional with the engine speed for both type of fuel.

It is found that the maximum brake power at wide-open throttle condition of CNG reduce by 26 %. Table 4.3 is comparing the experimental result with other findings. It shows the reduction of power while operating with CNG is slightly lower compare to gasoline. However, the current result of maximum brake power is comparable to other findings as Aslam et al. (2005) found the reduction of CNG brake power is 15%, Kalam et al. (2005) found within the range of 15 – 20% reduction and the earlier finding by Bangale et al.(1995) is 13.2%.

Table 4.3: CNG brake power reduction compare to gasoline

Referred results	Range of CNG brake power reduction
Bangale et al. (1995)	13.2
Aslam et al. (2005)	15
Kalam et al. (2005)	15-20
Experimental result	26

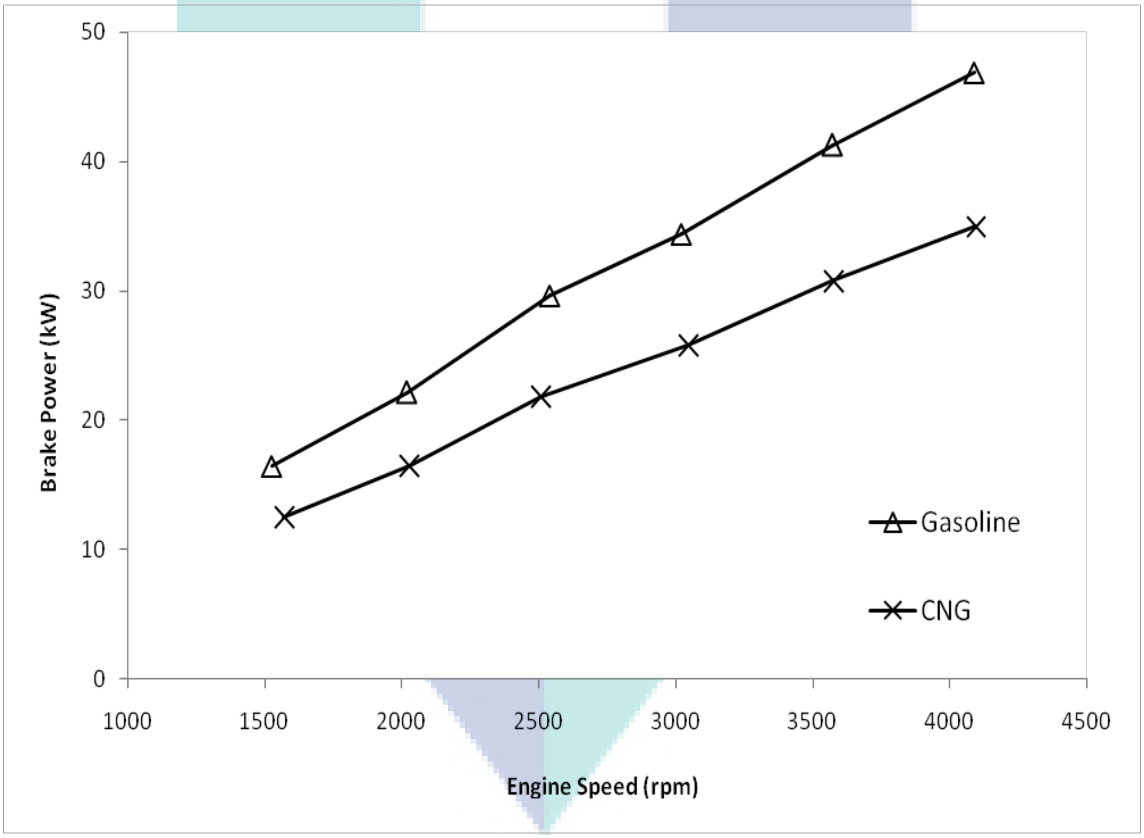


Figure 4.6: Brake power at different engine speed for wide-open throttle condition

4.3.4 Fuel Consumption

There are two ways of expressing fuel consumption viz. by volume or by weight during a specific time. For automobiles it is expressed in terms of kilometres per litre. In this experiment the fuel flow rate is measure by weight during a specific time.

i. Fuel Flow Rate

Figure 4.7 is shown the variation of torque against engine speed at the wide-open throttle operation of gasoline and CNG. The maximum and minimum fuel flow rate is of course related to the speed of the engine. During the engine operating with the gasoline fuel, the range of fuel flow rate is within 5.5 kg/hr at 1500 rpm and 15 kg/hr at 4000. For the CNG, the fuel flow rate is lower than gasoline. It lies between the range of 4 kg/hr at 1500 rpm to 7.4 kg/hr at 4000 rpm and it is between 25-51% of reduction of fuel flow rate. The maximum fuel flow rate reductions occur at 3500 rpm at with 51% reduction compare to gasoline. However, the overall reduction of CNG fuel flow rate of the experiment is higher than the finding of Kalam (Kalam et al., 2005)

Table 4.4: CNG fuel flow rate reduction compare to gasoline

Referred results	Maximum CNG fuel flow rate reduction (%)
Kalam et al. (2005)	11
Experimental result	51

It's observed that the increasing speed increases fuel consumption due to produce high brake power and the fuel consumption for gasoline fuel are higher at every running speed as compared to CNG fuel which is the reason to produce lower brake power by CNG fuel.

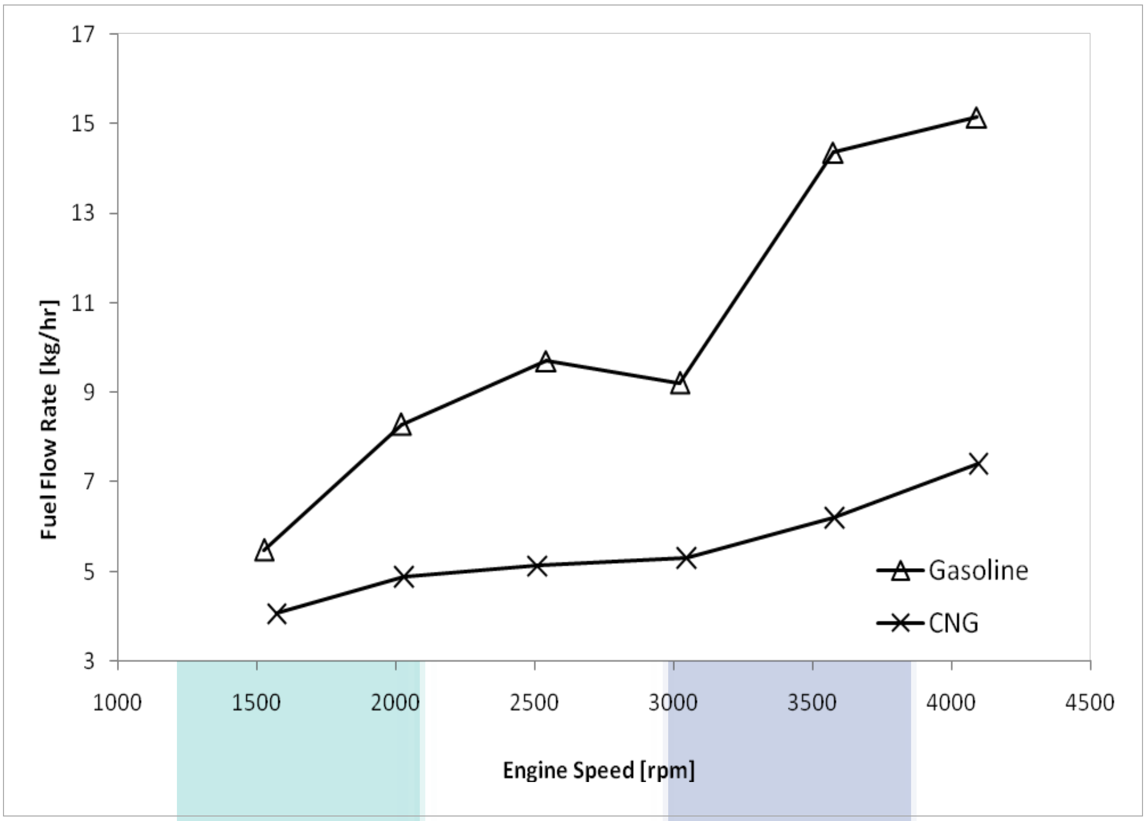


Figure 4.7: Fuel flow rate at different engine speed for wide-open throttle condition

ii. Brake Specific Fuel Consumption

The brake specific fuel consumption (BSFC) curve of Figure 4.9 is for wide open throttle speed operation for both type of fuel, gasoline and CNG. While operating with the gasoline, the graph shows that the BSFC is varying in the range of 0.33 kg/kWh to 0.37 kg/kWh. For the CNG operation it is varied between 0.21 kg/kWh to 0.32 kg/kWh. The graph shows a minimal trend which agreed with typical plot of conventional engine. Overall, the most optimum fuel utilization is achieved at 3000 rpm of engine speed with the value of BSFC about 0.27kg/kWh for gasoline and 0.21 kg/kWh for CNG. The lowest BSFC point is coincide with the leanest equivalence ratio.

Table 4.5: BSFC reduction of CNG compare to gasoline

Referred results	Range of CNG BSFC reduction (%)
Bangale et al. (1995)	15.8
Kalam et al. (2005)	15 -18
Experimental result	34

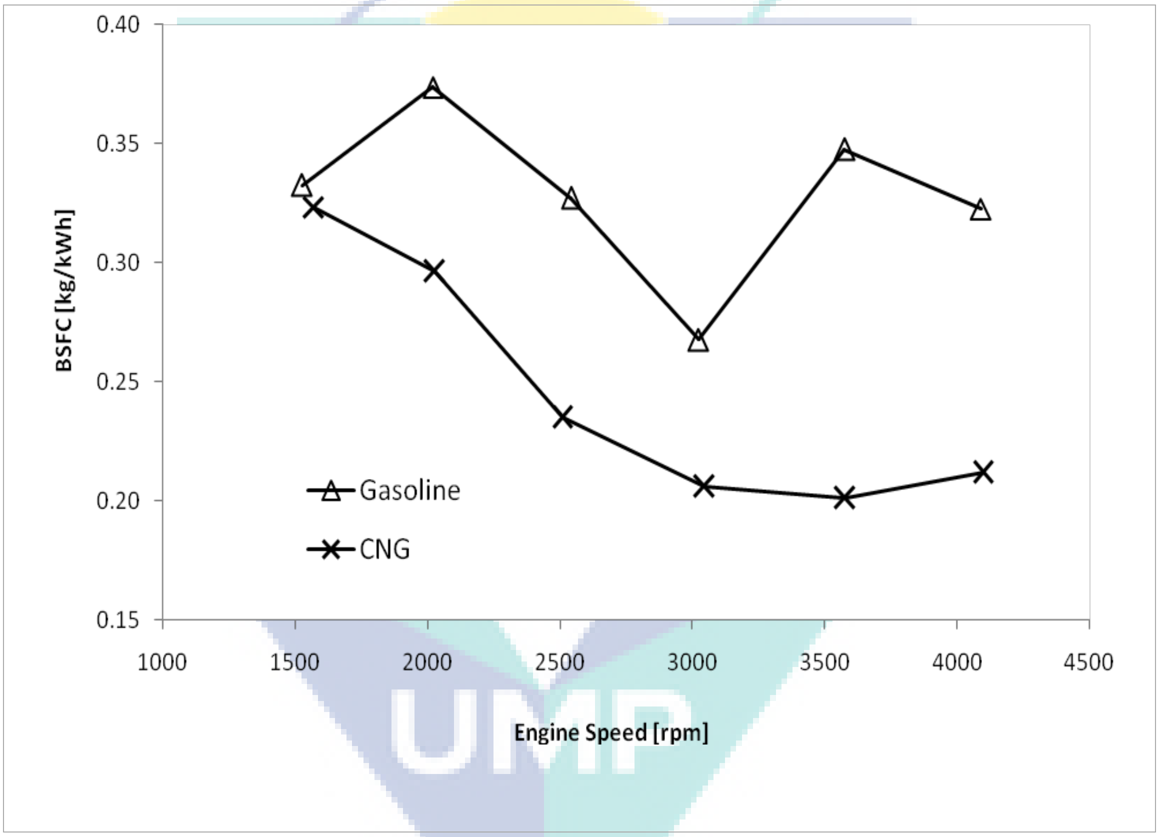


Figure 4.8: BSFC at different engine speed for wide-open throttle condition

The value of CNG maximum BSFC reduction compare to gasoline is 34% occur at 3500 rpm at wide-open throttle. It is a very low reduction compare to the finding of Aslam et al. 2005 with 18% drop at wide-open throttle condition. Bangale et al. (1995) and Kalam et al. (2005) also got a lower reduction of BSFC as 15 – 18% and 15.8% (Table 4.5). However the curves of the graph are shown the same trend with them whereby the BSFC increase as the speed in low speed range and drop at the high speed. This is because, at low speeds, the heat loss to the combustion chamber walls is proportionally greater, resulting in higher fuel consumption for the power produced. At high speeds, the friction power is increasing at a rapid rate, resulting in a slower increase in BSFC. It is observed that BSFC of gasoline was always higher than CNG throughout the speed range.

4.3.5 Brake Mean Effective Pressure

Mean effective pressure is a good parameter for comparing engine with regards to design or output because it is independent of both size and speed. If the torque is used for engine comparison, a larger engine will always look better. If power is used as the comparison, speed becomes very important (Pulkrabek, 2004). Figure 4.9 illustrates the brake mean effective pressure (BMEP) at different engine speed for wide open throttle of gasoline and CNG. The calculated brake mean effective pressure gives the value in the range of 3.5 MPa at 1500rpm and 3.7 MPa for at 4000 rpm for gasoline. For CNG, it is lies within and 2.5 MPa at 1500 rpm to 2.7 MPa at 4000 rpm.

The maximum BMEP of gasoline is 3.77 MPa, at 2500 rpm of engine speed and for CNG is 2.81 MPa at 2500 rpm of engine speed. However, for CNG operation, the 26% maximum reduction of the brake mean effective pressure of experimental results is acceptable compare to the finding of Aslam et al., (2005) with 16% reduction. It is shown in the Table 4.6. The reduction in BMEP with CNG operation is seen throughout the speed range. Part of this brake mean effective pressure loss is due to longer ignition delay and lower flame propagation of CNG. As combustion starts earlier with respect to TDC, there is a greater amount of negative work done on the piston before TDC compare to gasoline. The remainder of BMEP loss is due to displacement of air by CNG.

Table 4.6: CNG BMEP reduction of CNG compare to gasoline

Referred results	Maximum CNG BMEP reduction (%)
Aslam et al. (2005)	16
Experimental result	26

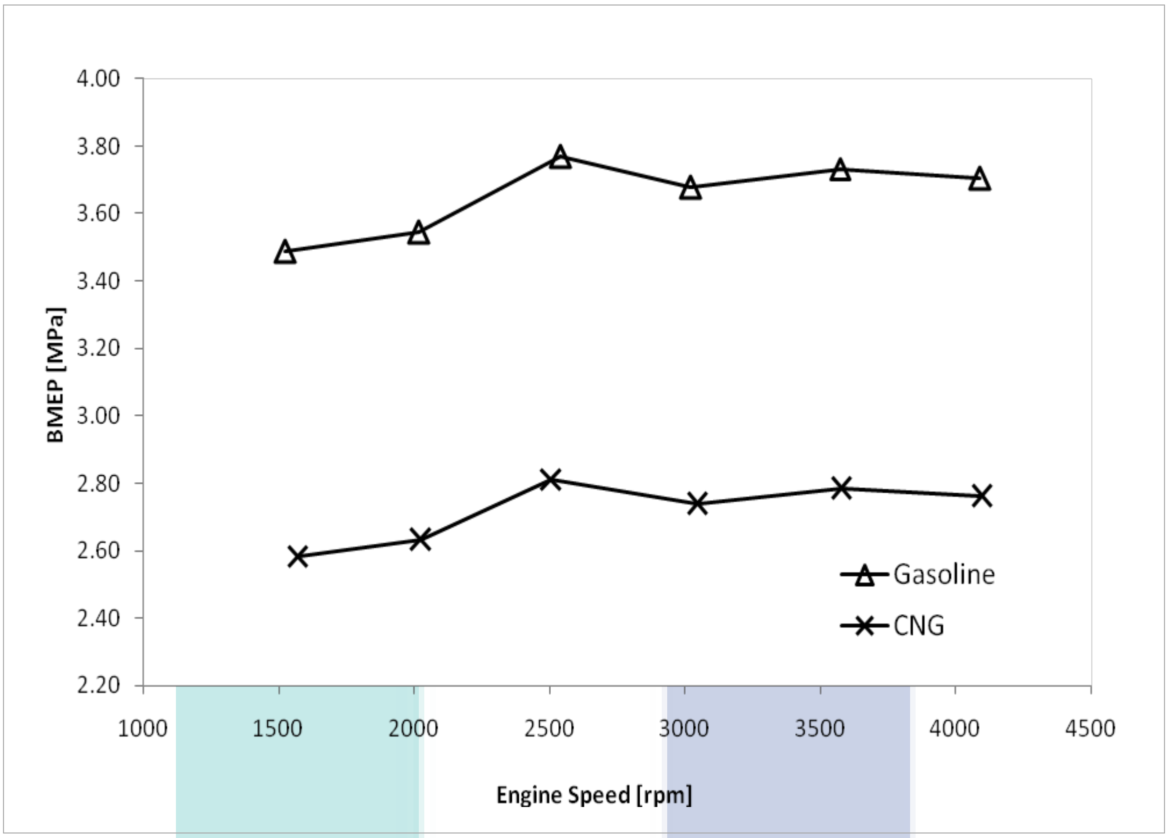


Figure 4.9: BMEP at different engine speed for wide-open throttle condition

4.3.6 Efficiency

i. Brake Thermal Efficiency

The brake thermal efficiency, is telling us how much of the fuel power is converted into brake power. Figure 4.10 illustrates the brake thermal efficiency for wide-open throttle with variable speed range of 1500 – 4000 rpm. It is observed that, at the beginning at low speed the brake thermal efficiency is low, increase until reach at a state of stoichiometric condition of air-fuel ratio, and then it's lower again when speed increased.

For the gasoline fuel, the measured brake thermal efficiency values fall within the range of 23% to 30%. The value is still within the typical range for natural aspirated SI engine where the value is around 30% (Heywood, 1988). When operating with CNG at wide-open throttle condition the range of brake thermal efficiency is between 24% - 39%. This trend also was found by Bangale et al. (1995), brake thermal efficiency of CNG higher than gasoline. Comparing the result of experiment, the brake thermal efficiency of the experimental result is higher than the finding of Bangale. However, the obtained experimental results are good agreement with Bangale et al. (1995) results. It is shows that brake thermal efficiency of CNG are higher than gasoline.

Table 4.7: Brake thermal efficiency of CNG compare to gasoline

Referred results	Maximum CNG brake thermal efficiency (%)
Bangale et al. (1995)	28
Experimental result	39

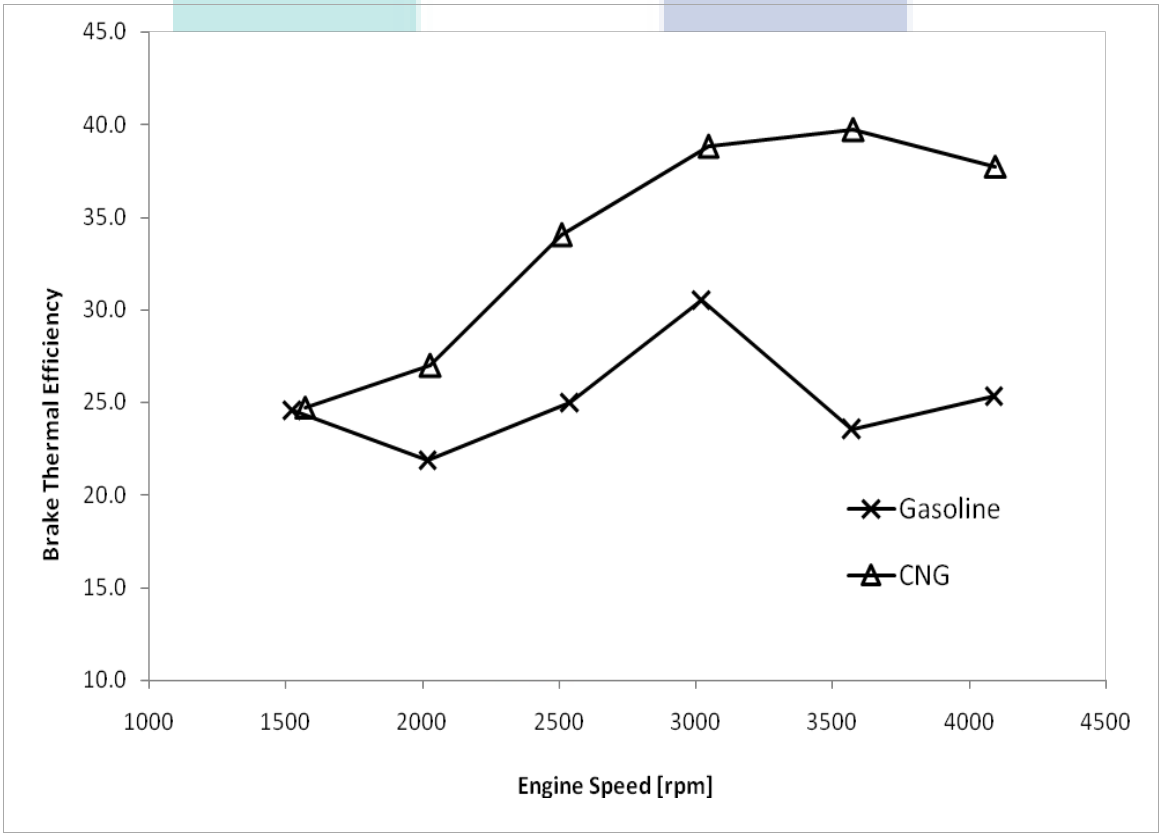


Figure 4.10: Brake thermal efficiency at different engine speed for wide-open throttle condition

ii. Mechanical Efficiency

Figure 4.11 presents the measured of mechanical efficiency at different engine speed for wide-open throttle condition. While operating with the gasoline fuel, the measured mechanical efficiency values fall in the range of 91% to 99%. This value is still in typical range for naturally aspirated spark ignition (SI) engine where the value is slightly higher than the typical value 65 - 90% at low speed and decrease to 75% at maximum rated speed (Heywood, 1988; Ganesan, 2004). The value of the peak efficiency is about 99% at 2500 rpm of engine speed. This point coincides with the highest torque point. It expected that the mechanical efficiency will be further decrease at higher engine speed approximating the value of 70%. However the current measurement is limited to the speed of 4000 rpm only thus gives moderate reduction efficiency. The experimental results show that the overall efficiency of CNG is lower than gasoline. It is found that the values of mechanical efficiency of CNG are lower than the gasoline as shown in Table 4.8.

Table 4.8: Typical value of mechanical efficiency of naturally aspirated SI engine

Referred results	Range of typical value of mechanical efficiency (%)
Heywood (1988)	90
Ganesan (2004)	65-85
Experimental results	84-92 for CNG 91-99 for gasoline

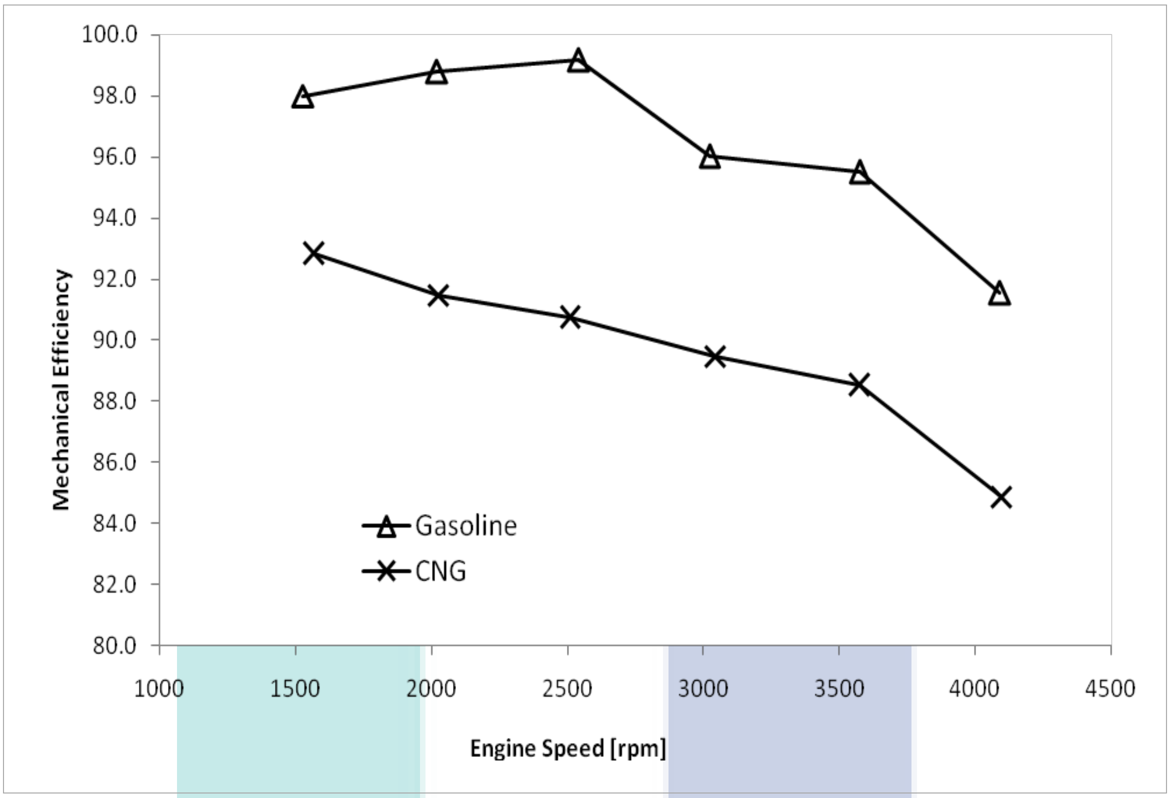


Figure 4.11: Mechanical efficiency at different engine speed for wide-open throttle condition

The range of mechanical efficiency reduction while operating with CNG lies between 84% -92%. It is believe due to lower flame propagation, that gives the lower torque of the engine. However, the same trend with the gasoline operation shows by the curve of CNG; the mechanical efficiency reduce by increasing the engine speed.

iii. Volumetric Efficiency

Figure 4.12 illustrate the variation of volumetric efficiency as function of engine speed for wide-open throttle condition. While operating with gasoline fuel, the volumetric efficiency of the engine falls within the range of 83% to 89%. And while operating with CNG its range is within 73% to 87%. From the review, the typical volumetric efficiency is about 80% to 90% and gas engine has a very much lower volumetric efficiency since gaseous fuel displaces air and therefore the breathing capacity of the engine is reduce (Heywood, 1988; Ganesan, 2004; Gupta, 2006) as hown in Table 4.9. However, the CNG engine will operating leaner than gasoline engine. The maximum reduction of volumetric efficiency of CNG is about 12% compare to gasoline.

Table 4.9: Typical value of volumetric efficiency of naturally aspirated SI engine

Referred results	Typical volumetric efficiency (%)
Heywood (1988)	80-90
Ganesan (2004)	80-85
Gupta (2006)	80-90
Experimental results	71-87 for CNG 80-90 for gasoline

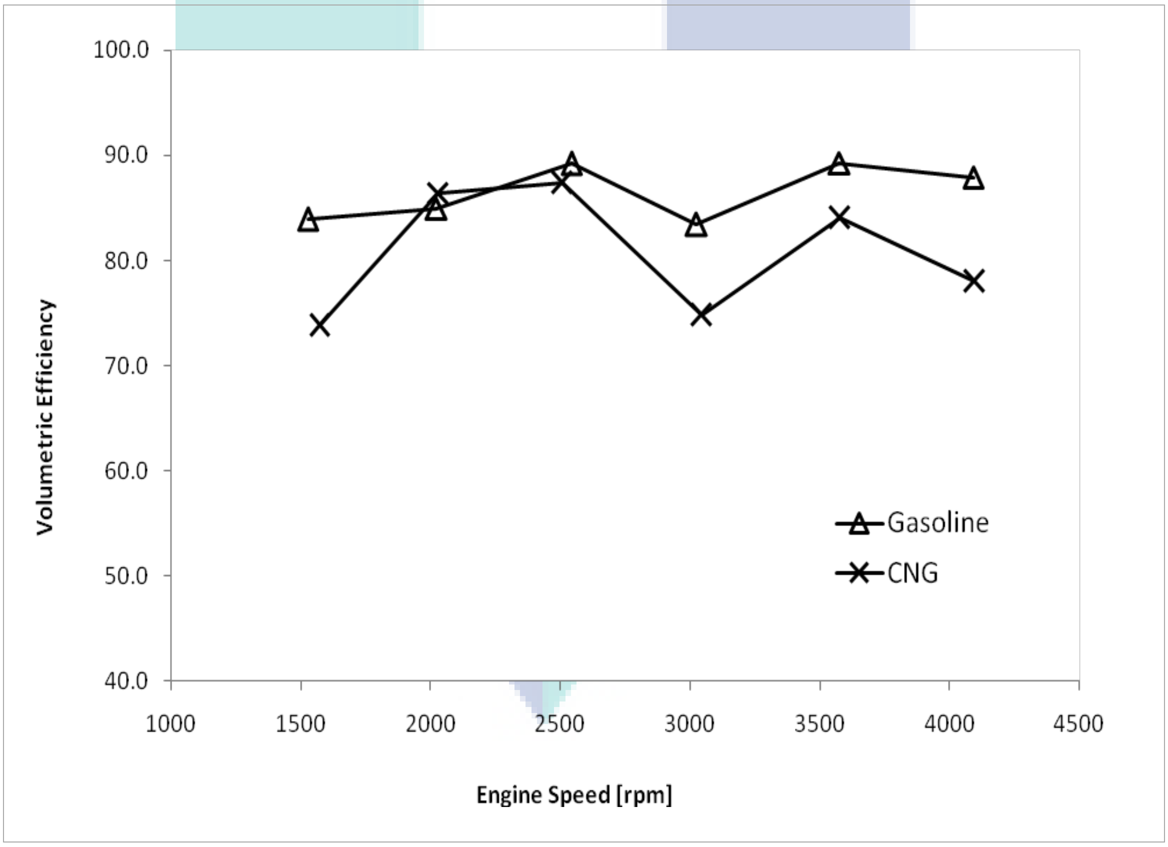


Figure 4.12: Volumetric efficiency at different engine speed for wide-open throttle condition

4.3.7 Emissions of the Engine

In this experiment, the emissions measured are included (i) carbon monoxide (C), (ii) carbon dioxide (CO_2), (iii) hydrocarbon (HC) and (iv) oxide of nitrogen (NO_x). It is found that the overall result of emission shown the same trend compare to the finding by other researcher. Even though the result of is higher than the other finding; it is acceptable as the exhaust gas emission analyser is calibrated by the supplier before performing the experiments.

i. Carbon Monoxide

Figure 4.13 illustrates the variation of CO concentration as function of engine speed for both fuels at wide open throttle. It is observed that the range of CO concentration for wide-open throttle operation of gasoline is between 1.8% and 1.0%. For CNG at wide-open throttle condition the range is between 1.3% and 5.2%. However from the graph curve it is showed that the concentration of CO is not proportion with the engine speed. The reduction of CO concentration of CNG operation lies between 48% and 74% within the engine speed between 1500-4000 rpm. It is observed that the CO concentration at the low speed and high speed are higher than middle speed. The lowest CO concentration occurred at engine speed between 2500rpm and 3000rpm, when is stoichiometric combustion occurred.

From the literature review, the reduction of CO concentration is around 80% at wide-open throttle operating condition (Bangale et. al., 1995 and Aslam et. al., 2005) as shown in Table 4.10. It is because CNG burns leaner than gasoline.

Maximum CO is generated when an engine runs rich, such as when starting or when accelerating. Even when the intake air-fuel mixture is stoichiometric or lean, some CO will be generated in the engine. Poor mixing, local rich regions and incomplete combustion will create some CO (Stone, 1999). In this study, the lowest concentration of CO is occurred at stoichiometric mixture at the engine speed of 3000 rpm with 15.1

air-fuel ratio. For the gasoline fuel and for CNG it's occurring at 2500 rpm and 3000 rpm with air-fuel ratio 16.2 and 16.3 respectively.

Table 4.10: CO concentration reduction of CNG compare to gasoline

Referred results	Range of reduction CO concentration of CNG (%)
Bangale et al. (1995)	81
Aslam et al. (2005)	80
Experimental results	48-74

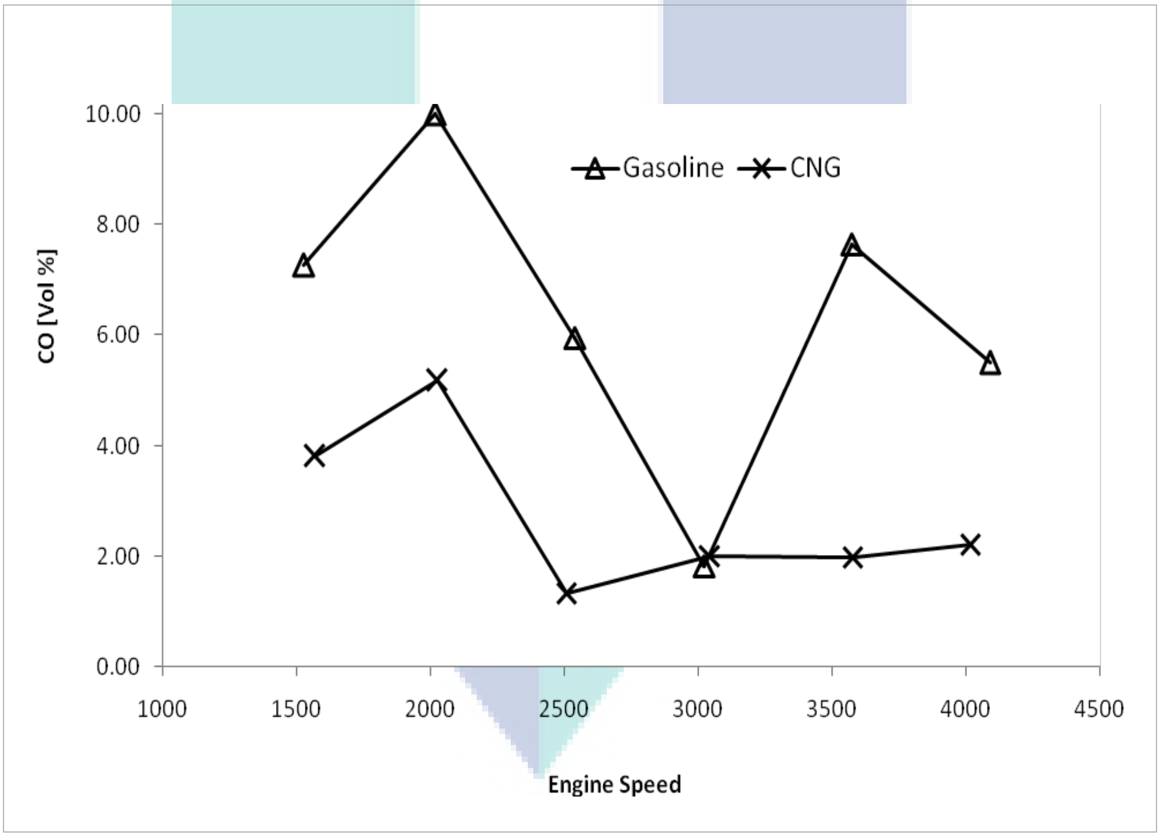


Figure 4.13: CO concentration at different engine speed for wide-open throttle condition

ii. Carbon Dioxide

Figure 4.14 shows the variation of CO_2 concentration with engine speed of gasoline and CNG at wide-open throttle condition. The range of CO_2 concentration for gasoline operation is lies between 7.9% and 13.8%, for CNG it is lies in the range of 8.3% and 11.3%. It is observed that the CO_2 concentration is fluctuated and doest show the proportional trend with the engine speed. Sometimes the CO_2 concentration of CNG is higher than CO_2 concentration of gasoline and vice-versa. However, the lowest drop of CO_2 concentration of CNG and gasoline is occurring at the engine speed of 3000 rpm with 11.2% for CNG and gasoline 13.8%. Therefore the maximum reduction of CNG of CO_2 concentration is 19% compare to gasoline.

This result is comparable with the finding of Bangale et al. (1995) and Aslam et al. (2005). They were found the reduction of CO_2 concentration is between 12 - 20% at wide open throttle operating condition as shown in Table 4.11. The lowest CO_2 concentration occurred at engine speed between 2500 rpm and 3000 rpm, when stoichiometric mixture is occurred.

According to Stone (1999), at moderate level of concentration, carbon dioxide is not considered an air pollutant. However, it is considered a major greenhouse gas and, at higher concentrations, is a major contributor to global warming. CO_2 is a major component of the exhaust in the combustion of any hydrocarbon fuel. Because of the growing number of motor vehicles, along with more factories and other sources, the amount of carbon dioxide in the atmosphere continues to grow. At upper elevations in the atmosphere, this higher concentration of carbon dioxide, along with other greenhouse gases, creates a thermal radiation shield. This shield reduces the amount of thermal radiation energy allowed to escape from the earth, raising slightly the average earth temperature. The most efficient way of reducing the amount of CO_2 is to burn less fuel (i.e., use engines with higher thermal efficiency).

Table 4.11: CO_2 concentration reduction of CNG compare to gasoline

Referred results	Maximum reduction of CO_2 concentration of CNG (%)
Bangale et al. (1995)	12
Aslam et al. (2005)	20
Experimental results	19

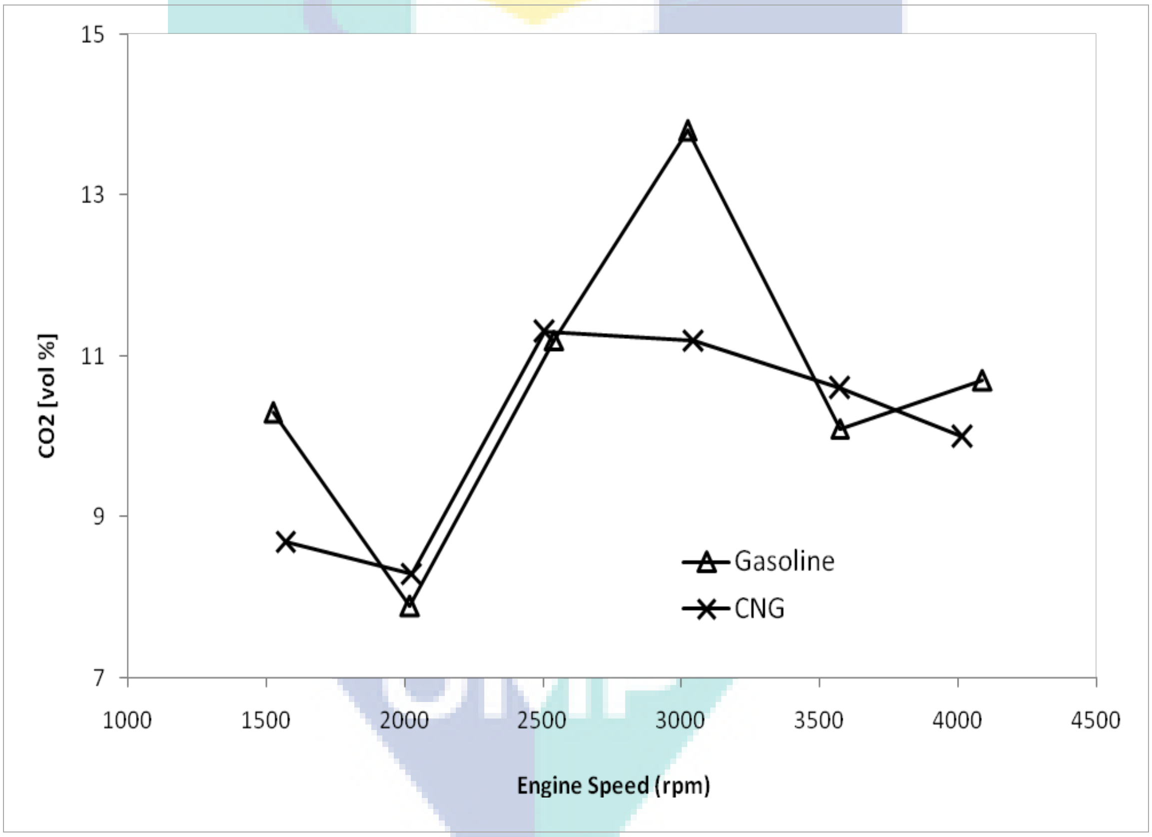


Figure 4.14: CO_2 concentration at different engine speed for wide-open throttle condition

iii. Hydrocarbon

Figure 4.18 shows the variation of the HC concentration with speed of gasoline and CNG at wide open throttle. During the operation with gasoline the hydrocarbon concentration is fluctuate within the range of 49 ppm and 154 ppm. The same trend goes to CNG while operating at wide-open throttle. The range of hydrocarbon concentration is also fluctuating within the range of 13 ppm and 50 ppm. The highest HC concentration maximum reduction of CNG is 87% compare to gasoline occurred at 2000 rpm of engine speed.

From the literature review, the reduction of HC concentration is about 50% at wide open throttle operating condition (Bangale et. al., 1995 and Aslam et. al., 2005) as shown in Table 4.12. The value of maximum HC concentration reduction for the experimental results are occurred at 87% is achieved while operating with CNG. It is observed that the HC concentration at the low speed and high speed are higher than middle speed. The lowest HC concentration occurred during the stoichiometric combustion at the engines speed of 3000 rpm for both fuels with 10 ppm for CNG and 49 ppm for gasoline. However, the average reduction of hydrocarbon concentration is 62.5%.

Table 4.12: HC concentration reduction of CNG compare to gasoline

Referred results	Maximum reduction CO concentration of CNG (%)
Bangale et al. (1995)	50
Aslam et al. (2005)	50
Experimental results	60

According to Stone (1999), exhaust gases leaving the combustion chamber of an SI engine contain up to 6000 ppm of hydrocarbon components, the equivalent of 1 – 1.5% of the fuel. That statement shows that the concentration of hydrocarbon from the current result is very low.

Causes of HC emission includes nonstoichiometric air-fuel ratio, incomplete combustion, crevice volumes, leak past the exhaust valve, valve overlap, deposits on combustion chamber walls and oil on combustion chamber walls.

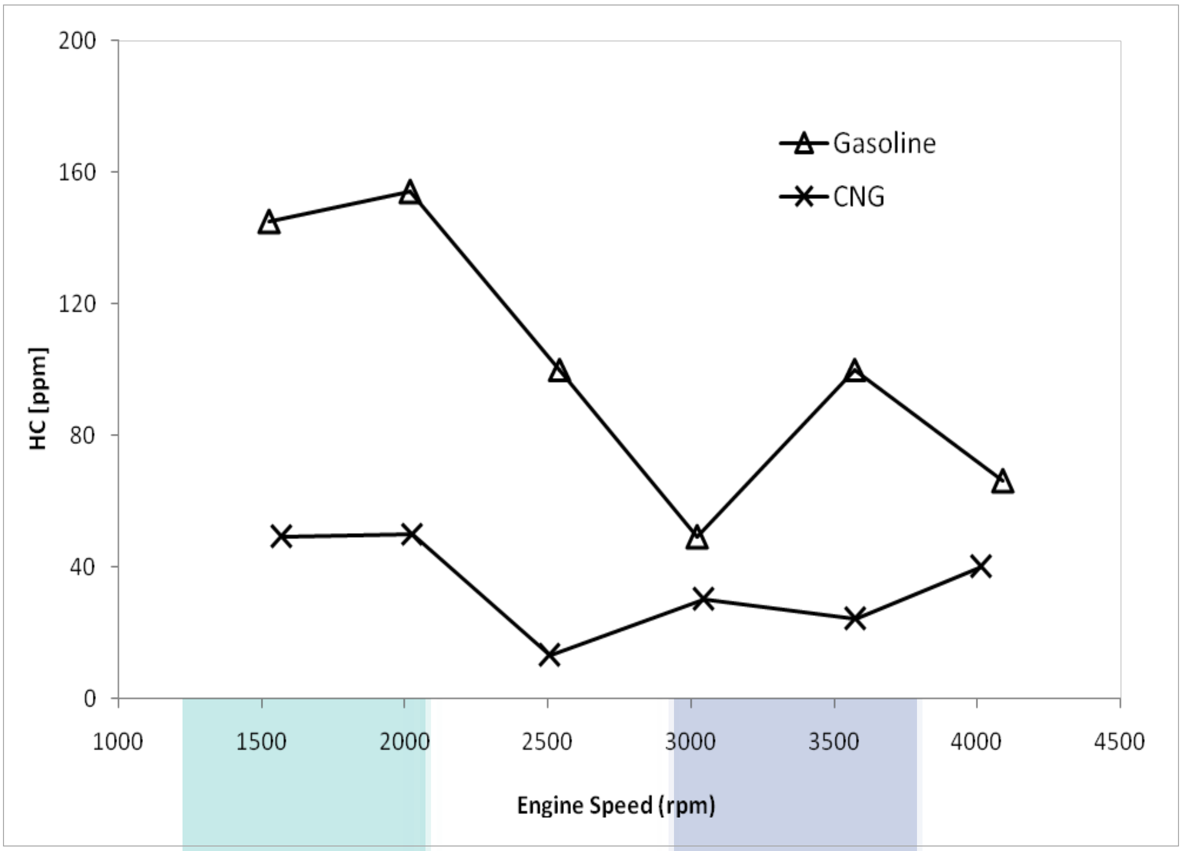


Figure 4.15: HC concentration at different engine speed for wide-open throttle condition

iv. Oxide of Nitrogen

Figure 4.16 shows the variation of NO_x concentration with speed. The engine operation is performed at wide open throttle for each type of fuel, gasoline and CNG. The figure shows that the trend of NO_x concentration is low at engine speed of 1500 rpm to 2000 rpm; high concentration of NO_x at 2500 rpm and 3000 rpm and becoming at the rest of the engine speed. The range of NO_x concentration is lie within 135 ppm and 872 ppm for the gasoline operation and for the CNG operation the range of NO_x concentration is within 119 ppm and 1525 ppm. The amount of the NO_x concentration is acceptable and according to Stone (1999), exhaust gases of an engine can have up to 2000 ppm of nitrogen.

The maximum NO_x concentration at wide open throttle during operating with gasoline is 872 ppm occurred at 3000rpm of engine speed and the maximum CO concentration at the same condition of operation for CNG is 1525 ppm, occurred at 4000 rpm of engine speed. The overall results of NO_x concentration of CNG is much higher comparing to gasoline. According to Heywood (1988), CNG produces higher temperature compare to gasoline that produces lower temperature during the exhaust stroke. The higher temperature during the exhaust stroke is due to slow flame propagation and the highest temperature of CNG occurred slightly after TDC. It's make the exhaust gas temperature of CNG is higher than gasoline.

Bangale et al. (1995) and Aslam et al. (2005) found the NO_x concentrations increase in the range of 22% to 33% during wide-open throttle operation. The average value of maximum CNG of NO_x concentration increased 43% at wide-open throttle. It is observed that the NO_x concentration at the low speed and high speed are lower than middle speed. The highest NO_x concentration occurred at engine speed between 2500 rpm and 3000 rpm, when is stoichiometric combustion occurred. In the current study, the NO_x emission produces by the CNG is very high compare to gasoline but it is still within the limit.

Table 4.13: Increment of NO_x concentration of CNG compare to gasoline

Referred results	Maximum increase of NO_x concentration of CNG compare to gasoline (%)
Bangale et al. (1995)	22
Aslam et al. (2005)	33
Experimental results	43

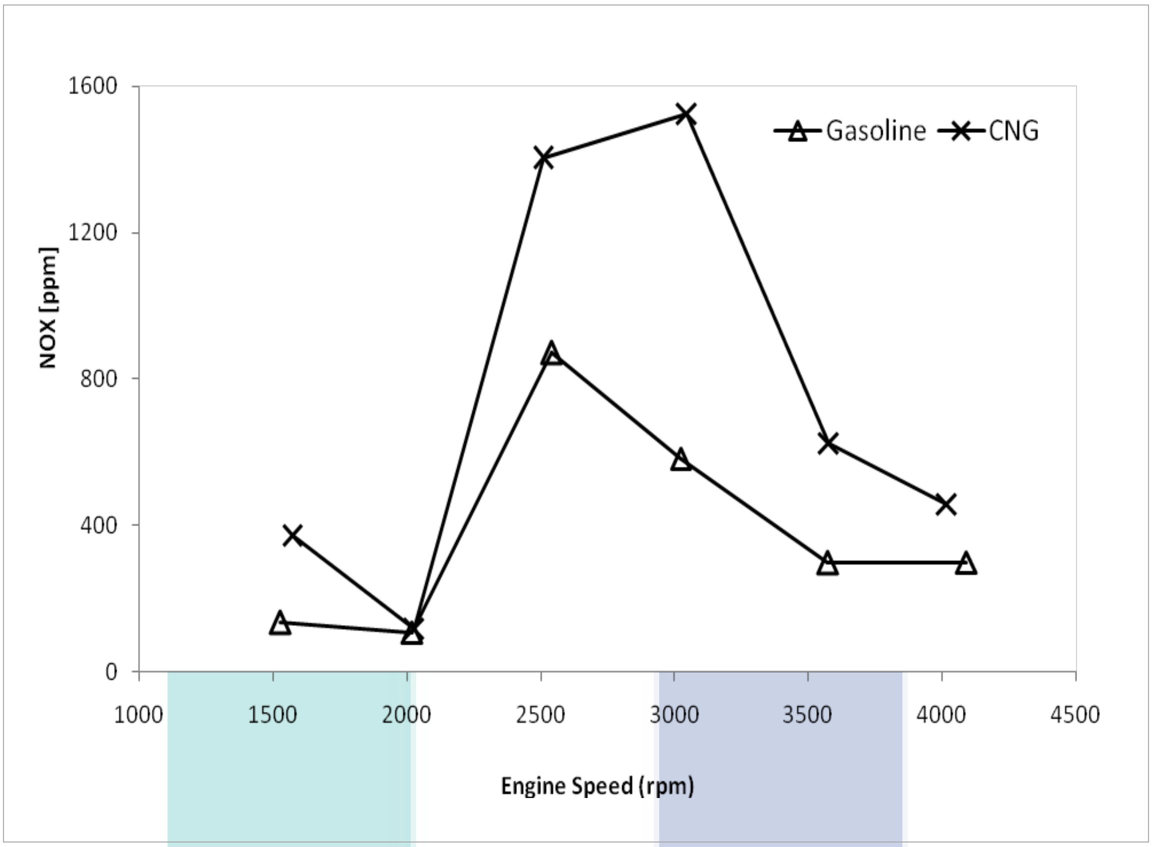


Figure 4.16: NO_x concentration at different engine speed for wide-open throttle condition

4.4 Summary

This chapter has presented the experimental results of engine test-rig. It is consist of the engine torque, air-fuel ratio, cylinder pressure, engine power, fuel consumption, efficiency and engine exhaust emission. The experimental results have been discussed and compare with previously published literature for validity purposes. Eventhough the engine speed is limited the range of 1500 rpm to 4000 rpm; the findings are very valuable for further research. The next chapter of the thesis will be summarizing the results, conclusion and recommendation for future work.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 Introduction

The aim of this thesis was to study the behaviour retrofitted engine performance of CNG on a 4-stroke spark ignition engine in term of power, torque, fuel consumption engine efficiency and engine emission while operating with CNG. The investigation was performed at engine speed in the range of 1500 to 4000 rpm. All the experimental exercise is performed at wide-open throttle. The data is collected and the results show a significant finding for future research on CNG as alternative fuel. Previous and recent of numerous researchers still explore and study of this area using variety of techniques to protect the poor fuel consumption and emission level.

5.2 Conclusion

The present study, has demonstrated that retrofitted CNG fuelled engines have a potential for maximum brake thermal efficiency and significant reduction of emissions. The following concluding remarks can be drawn from the present study

- While operating with the gasoline fuel the maximum brake power is produces at maximum engine speed for both type fuels, there are 46.9 kW produces by gasoline and 35 kW produces by CNG. CNG produces less by 25% brake power compared to gasoline.

- Retrofitted CNG produces 25% less brake mean effective pressure and consumes 13% less brake specific fuel consumption at wide-open throttle condition with CNG compare to gasoline

- The engine shows a maximum of 39% of brake thermal efficiency produces by CNG compare to gasoline at wide-open throttle condition. The average increment of brake thermal efficiency of CNG compare to gasoline is 23%.

- The average retrofitted engine reduced CO by around 52%, CO_2 by 8.6% and HC by 62% and the NO_x emissions are increased by 43% while operating at wide-open throttle condition.

Retrofitted CNG fuelled engines can be used for the moment for economic, environment and energy security reason and providing the future gas-age period.

5.3 Recommendations for the Future Research

There is still scope for the further study to improve the CNG engine performance. This section describes some possibilities for extending work for the investigation of CNG engine of the performance and emission of the CNG dedicated and dual-fuel engine. It is suggested to study the effectiveness of Three-Way catalytic Converter of reducing exhaust emission of the engine it is also suggested to study the engine using electronic fuel injection system. It is believe through application both of the technology the engine performance will increase and the exhaust engine emission will reduce significantly for the same type of engine.

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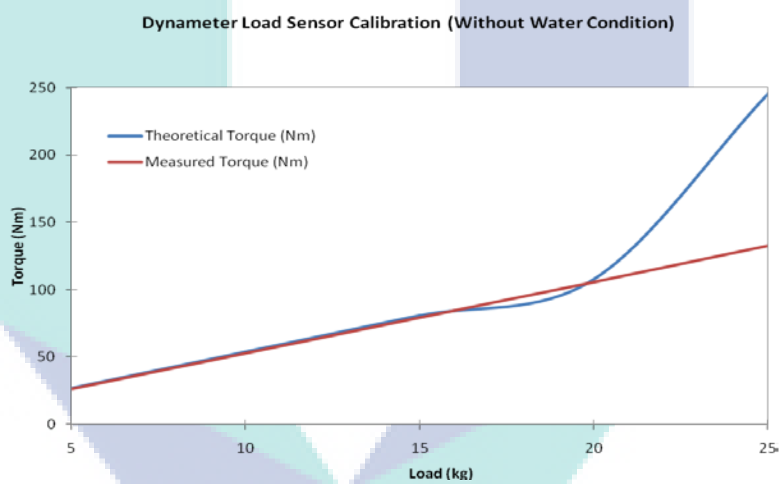
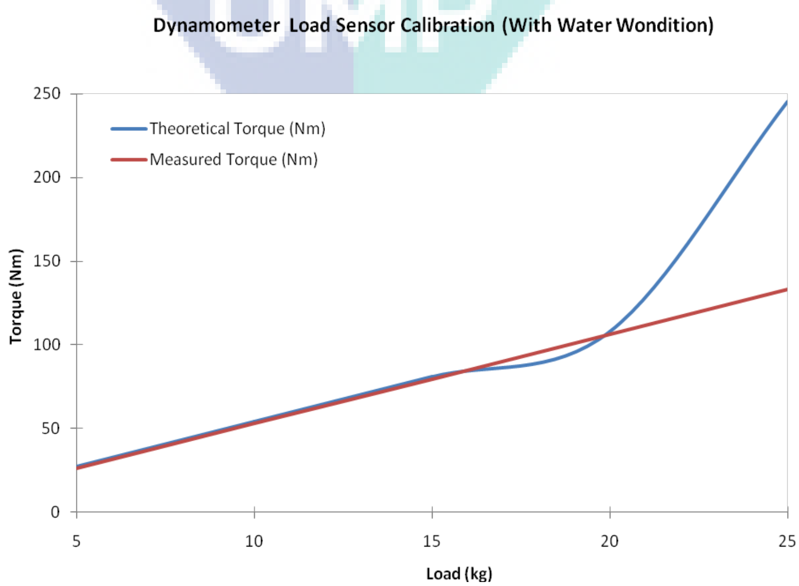
APPENDICES

APPENDIX A

Dynamometer Load Cell Calibration

Calibration Graph





APPENDIX B

The Estimated of Maximum Gas Flow Rate of Gas Flow Meter

According to Kalam et al., (2005) for the CNG maximum specific fuel consumption, SFC is and 380g/kWh maximum brake power, BP is 39kW. These data is used to estimate the gas flow meter operating range.

i. Mass flow rate

Therefore gas flow rate = SFC x BP

= 380 x 39 = 14 280 g/h

= 14 280 x 1 x 10^{-3} / 3600

= 4.117 x 10^{-3} kg/s (maximum gas flow rate)

ii. Volumetric flow rate

Density of CNG, $\rho = 0.717 kg/m^3$

ρ = mass flow rate / volumetric flow rate

$\therefore$ Volumetric flow rate = mass flow rate/ρ

= (4.117 x 10^{-3} kg/s)/ (0.717kg/m^3)

= 5.74 x 10^{-3} m^3/s or 20.663 m^3/h

If we need maximum flow rate of the flow meter of 10 x 10^{-3} kg/s

The volumetric flow rate will be, 10 x 10^{-3}/0.717

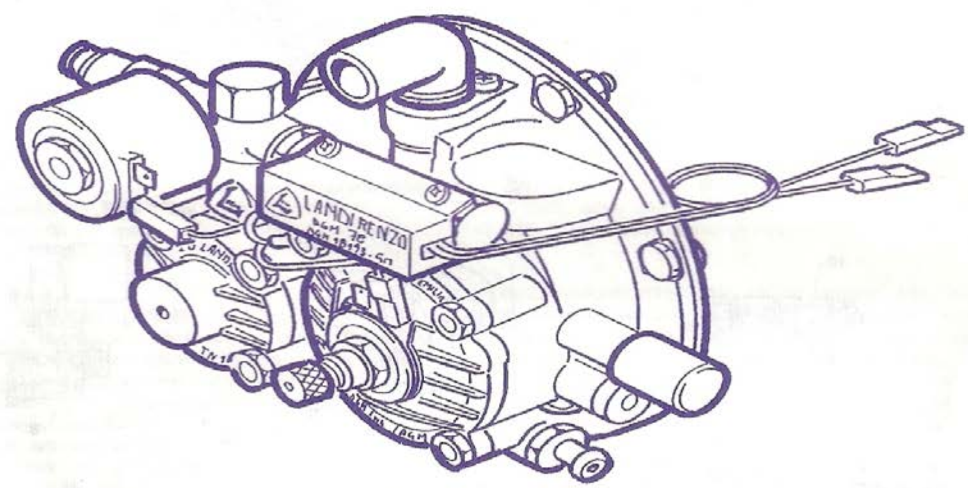
= 0.0139 m^3/s or 50.2 m^3/h

For the application in this investigation, the best model used is RAMC01-T4SS-62M1-T90NNNPT/SD/IE1 a Metal Short-stroke Rotameter a product of Yokogawa.

APPENDIX C

MANUAL OF 'TN SIC' CNG REGULATORS

RIDUTTORI 'TN SIC' METANO
'TN SIC' CNG REGULATORS
REDUCTEURS 'TN SIC' GAZ NATUREL
REDUCTORES 'TN SIC' GAS NATURAL



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Thank you for purchasing a **LANDI RENZO** pressure regulator type **'TN SIC'**, the reliable and technologically advanced device to install a CNG conversion system on vehicles with catalytic converter, injection system, carburettor and turbo-charger.
Correctly installed, your pressure regulator will give many years of excellent performance.
To ensure that you achieve peak performance from the conversion system, please read this installation and setting guide thoroughly.

LEGEND (Fig. 1)

A) Gas outlet connector
B) Idle speed pipe (the pipe is not installed on the regulator, but is supplied separately. We suggest its use only in case of stalling during idle / out-of-idle condition. If used, to position always at the same direction of gas outlet connector)
C) Sensitivity screw
D) Plus contact for idle speed solenoid valve
E) Idle speed setting screw
F) High pressure solenoid valve **(not present on the TN1 SIC and TN1 SIC Turbo-Charged versions)**
G) Gas inlet connector
H) Heating connector

1. TECHNICAL SPECIFICATION

Electronic control device to reduce the natural gas pressure allowing a regular flow of gas every time the engine requires it.
It is equipped with three natural gas reduction stages that allow stability at both high and low pressures and a high-pressure solenoid valve upstream from the first stage. The absorption of heat, taken from parts of the reduction unit, heated with the liquid of the engine cooling circuit, prevents the natural gas freezing during the fall in pressure phase.
The flow of gas necessary for engine idling has a positive pressure from the second stage and is activated by means of a gas pipe separated from the main flow. It includes an electronic starting device with a built-in safety system that trips and shuts off the gas solenoid valves if the engine is switched off or even stalls.

SPECIFICATION:
Regulator type: 3 stages with electronic starting device and idling at positive pressure
Use: motor transport (suitable for vehicles with catalytic converter, injection, carburettor and turbo-charger)
Type of fluid: CNG (Compressed Natural Gas)
Body: GDALSI 13 Fe UNI 5079
Heating: engine cooling circuit liquid
Test pressure: 300 bar
Inlet pressure: 220 bar
First stage adjustment pressure: 3.5-4 bar
Second stage adjustment pressure: 1,5 bar
Power supply: 12V DC
H. P. solenoid valve coil power capacity: 20W
Idling solenoid valve coil power capacity: 18W

VERSIONS:
TN1 SIC (standard): up to 130 HP
TN1 B SIC (standard): up to 130 HP
TN1 B SIC (oversize): from 130 HP to 190 HP
TN1 C SIC (oversize): from 130 HP to 190 HP
TN2 C SIC (super overs.): from 190 HP to 220 HP
TN2 C/S: from 220 HP to 250 HP
TN3 SIC: over 250 HP
TN1 SIC Turbo-Charged: for turbo-charged engines up to 200 HP

2. REGULATOR OPERATION (Fig. 2)

When solenoid valve (1) is open, natural gas enters the first-stage chamber (4). The pressure exerted by the gas on the chamber walls dilates diaphragm (8) thus over-coming the resistance of spring (9). Being connected to lever (3), diaphragm (8) also acts on the first-stage valve (2) generating a pressure balance.
The gas then passes from the first-stage chamber (4) to the second-stage chamber (11). Gas flow is metered by the pressure exerted by diaphragm (10) in controlling the opening and closing of valve (13).
The vacuum generated by the engine causes the third-stage diaphragm (6) to move axially. Being connected to lever (5), this diaphragm causes valve (7) to open so that the gas reaches the engine through outlet (19). Valve (7) is sealed when spring (14) is adequately set.
The starting and idle-speed device consists of solenoid valve (17) which is controlled via an electronic device.

Plunger (15) moves thus leaving hole (18) open. The gas coming from the second-stage chamber (11) flows out of this hole thus allowing the engine to run at idle speed. If the engine stops, the coil bècomes de-energised and plunger (15) closes outlet hole (18).
Idle-speed setting is achieved via adjuster (16). At starting, the electronic device energises coil (17) so that plunger (15) leaves (18) open thus letting the required flow of gas through. The regulator has a built-in electronic transducer which shows how much gas is available in the cylinder.

3. GENERAL NOTICES

To install the regulator, the following instructions must be observed:

- install the regulator in the engine compartment as close as possible to the point where the mixer is to be installed, securing it integrally with the bodywork using the screws provided;
- position the regulator away from air intake components for the ventilation and heating of the passenger compartment;
- position the regulator at a distance not inferior to 150 mm from the exhaust pipes or silencer. In case the distance is inferior to the minimum value, but not greater than 75 mm, it is necessary to insert between the elements a metal sheet (or equivalent material) with a minimum thickness of 1 mm.
- position the regulator in parallel with the direction of travel and in an upright position so that it is easily accessible to allow adjustment and maintenance work;
- check that the regulator is placed in a lower position than the highest point of the radiator in order to prevent air bubbles forming in the water circuit;
- take care not to position the regulator so that the bleed plug is above the distributor or above the ignition coil;
- carefully clean the natural gas supply pipes before they are finally connected to the regulator to prevent any impurities getting inside the regulator;
- check that with the engine running there is no leakage from the water pipes (generally connected to the passenger compartment heating circuit);
- never change for any reason the position of the regulator gas inlet assembly (G) because that operation could alter the setting of the first stage valve: in this way you may risk to prevent the gas flow and an eventual pressure increase in the first stage could cause the opening of the security valve and the consequent gas leak.

LANDI RENZO S.p.A., beside declining warranty, will not assume responsibility for damage derived from tampering said component and the regulator in general.

- check that the regulator heats up quickly by means of the engine cooling circuit connection.

Every time the engine cooling circuit is emptied it is necessary to restore the level of liquid, taking care any air bubbles are eliminated as they could prevent the regulator from heating.
The regulator gas outlet should be connected to the mixer preventing the connecting pipe (which must be as short as possible) from having any bends or pockets.
The regulator is supplied with securing brackets to position the regulator in the engine compartment. These brackets will need to be adapted in relation to the point of the engine compartment chosen for securing.

4. SETTING PROCEDURE FOR REGULATORS TYPE 'TN SIC'

***with exhaust gas analyser* (Fig. 1)**

4.1 CATALYSED CAR

The first step is to adjust peak speed:

- take the engine at about 3,500 r.p.m. until reading on the Tester Programmer Mod. V05 that the default value is recorded.

The second step is to adjust idling speed:

- with the engine running, turn the idle speed setting screw (E) (clockwise it decreases, anticlockwise it increases) until, on the Tester Programmer Mod. V05, the number of steps of the linear electromechanical actuator indicated in menu 'Display' at the word MOT is equal (or as close as possible) to the value indicated at the word DEF.
- always check that the Lambda scale LED's indicating carburation are flashing properly.

• Check by the gas analyser that the Lambda value is about 1.000, CO and HC values are tending to zero and CO_2 value is about 11-13%.
For deeply details see 'Installation and adjustment manual of the Lambda Control System A1 V05' or 'Tester Programmer Mod. V05 instruction manual' for the procedure of recording the carburation.
Having regulated idling and peak speeds, carry out a test on the road.

4.2 INJECTION CAR
The first step is to adjust peak speed:
• take the engine at about 3,500 r.p.m. and turn the peak speed regulator located on the start petrol solenoid valve, between the regulator and the mixer, to take the values of CO, HC and CO_2 as shown in the table.

The second step is to adjust idling speed:
• with the engine running, turn the Idle speed setting screw (E) (clockwise it decreases, anticlockwise it increases) to take the values of CO, HC and CO_2 as shown in the table.
Having regulated idling and peak speeds, carry out a test on the road.

4.3 CARBURETTED CAR
The first step is to adjust peak speed:
• take the engine to approximately 3,500 r.p.m. and turn the peak speed regulator located between the regulator and the mixer to take the values of CO, HC and CO_2 as shown in the table.

The second step is to adjust idling speed:
• with the engine running, turn the Idle speed setting screw (E) (clockwise it decreases, anticlockwise it increases) to take the values of CO, HC and CO_2 as shown in the table.
Having regulated idling and peak speeds, carry out a test on the road.

SETTING TABLE 'TN SIC'

GAS	SPEED	LIMITS	
		bottom	top
CO (in %)	idling	0.10	0.30
	3.500 g/m	0.10	0.30
CO_2 (in %)	idling	11	13
	3.500 g/m	11	13
HC (in ppm)	idling	150	250
	3.500 g/m	30	60

5. SETTING PROCEDURE FOR REGULATORS TYPE 'TN SIC'
***without exhaust gas analyser* (Fig. 1)**

5.1 CATALYSED CAR
See point 4.1 without gas analyser check.

5.2 INJECTION AND CARBURETTED CAR
The first step is to adjust peak speed:
• take the engine to approximately 3,500 r.p.m. and turn the peak speed regulator located between the regulator and the mixer clockwise until you notice a fall in engine speed due to the mixture getting leaner; then turn this same screw very slowly anticlockwise until there is an increase in engine speed; at this stage it is not necessary to turn the screw further anticlockwise since there would only be greater consumption and no increase in efficiency.

The second step is to adjust idling speed:
• with the engine running, turn the Idle speed setting screw (E) (clockwise it decreases, anticlockwise it increases) until an optimum idling speed is obtained which is also to be checked after the road test.
Having regulated idling and peak speeds, carry out a test on the road.

6. SENSITIVITY SETTING PROCEDURE FOR REGULATORS TYPE 'TN SIC' (Fig. 1)
The regulators are already set in-house by the manufacturer. If problems arise, such as idle speed instability or acceleration gap, please check the degree of regulator sensitivity.

The setting screw (C) is not used for setting idle speed but simply to adjust the regulator sensitivity. By unscrewing it you reduce the spring load on the 3rd stage lever while by tightening it you increase the spring load on the 2nd stage lever towards closing.

In particular, since the idle speed flow is separated from the peak speed one, shifting from idle to peak speed should take place without 'carburation gaps'; such gaps may occur, above all, during too slow accelerations (too tightened screw); at the same time the regulator should remain tight without any gas leakage every time the engine is turned off (too loose screw).

In order to set sensitivity as required do as follows:

1) Remove the tube which conveys gas from the gas outlet connector to the mixer (A);
2) Tighten the sensitivity screw (C);
3) Disconnect the blue wire from the positive contact of the idle speed solenoid valve (D) and connect the same wire to the positive contact of the battery (in order to fill with gas the regulator);
4) Form a bubble with soap water on the gas outlet connector (A) and loosen the screw (C) until the gas starts coming out of the regulator and inflates the bubble;
5) From the time gas starts coming out of the regulator, tighten screw (C) again until no more gas leaks. From the moment that no more gas leaks, tighten the screw another half turn to be sure that it closes perfectly;
6) Connect again the blue wire to the positive contact of the idle speed solenoid valve (D);
7) Place the cap on the sensitivity screw (C) in order to avoid accidental or intentional changes in its setting.

Another less sensitive but more rapid system for sensitivity adjustments is as follows:

1) Fully tighten the sensitivity screw (C);
2) Turn the engine on and set idle speed by means of screw (E) until the maximum CO_2 level is attained;
3) Slowly loosen the screw (C) until a marked change (reduction) in the CO_2 level is reached;
4) From the time that this CO_2 change is observed, tighten the screw (C) until the CO_2 value is the same as in item 2.
5) Place the cap on the sensitivity screw (C) in order to avoid accidental or intentional changes in its setting.
6) Check that no acceleration gaps are observed while slowly accelerating.

After the first 500 / 1.000 Km check regulator sensitivity.

7. MAINTENANCE WORK ON THE SYSTEM

To get the best out of natural gas fuel, the engine must be tuned and regularly serviced, both as regards the mechanical and the electrical parts. In addition to the routine maintenance required by the vehicle manufacturer, it is recommended:

- every 15,000 km check/replace the air filter, change the spark plugs, check the exhaust gas with an analyser, check the efficiency of the electrical system (check there is no oxide formation in the connections);
- every 30,000 km check the valve clearance, check lambda probe efficiency (for cars with a catalytic converter); with the bleed plug, check there is no oil or other residues inside the regulator;
- every 100,000 km, if malfunctioning occurs, carry out a general overhaul of the system using our product overhaul kits, which are support of instructions showing the methods to follow.

Using spark plugs with a colder heat rating, it is wise to check that the distance of the electrodes is never greater than 1 mm.
It is advised to increase the valve clearance by 0.05 mm with respect to the specifications for petrol operation provided by the vehicle manufacturer.

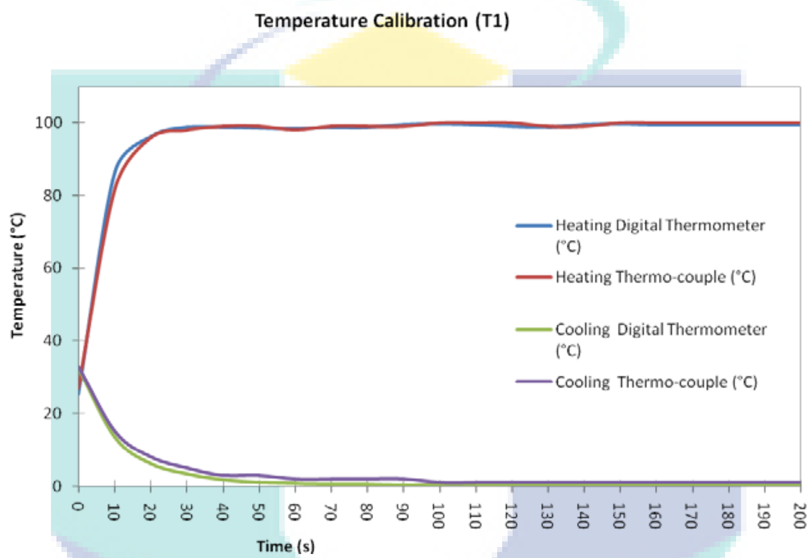
Having installed the natural gas conversion system, it is natural to travel as many kilometres as possible with this fuel: however, so as not to prejudice the correct operation of the original petrol system and fuel pump, it is advised to travel 2 - 3 km on petrol at least every 200 - 300 km (example at each CNG refuelling)

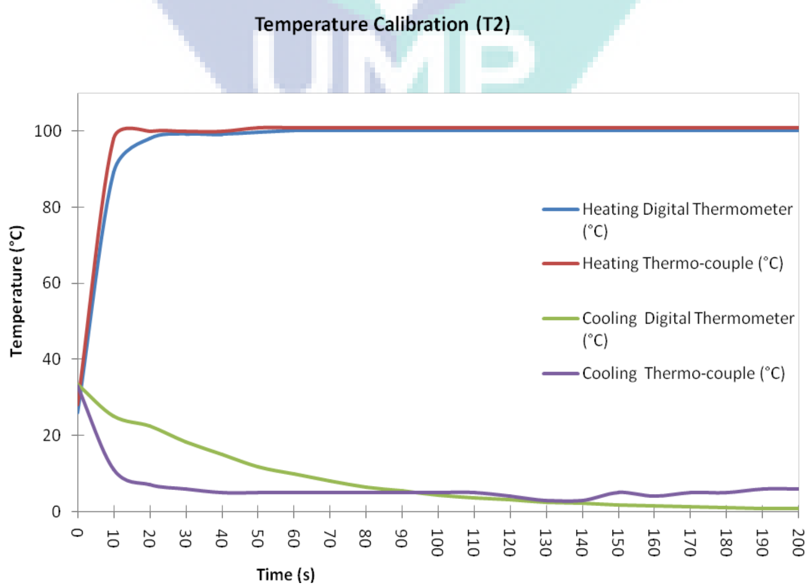
Date, descriptions and illustrations are indicatory. LANDI RENZO S.p.A. reserves the right to improve or modify them without prior notification.

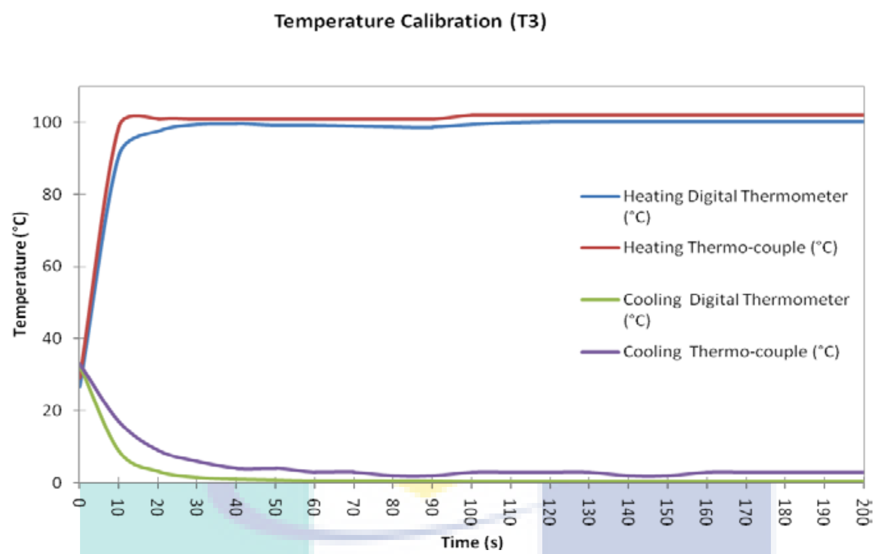
APPENDIX D

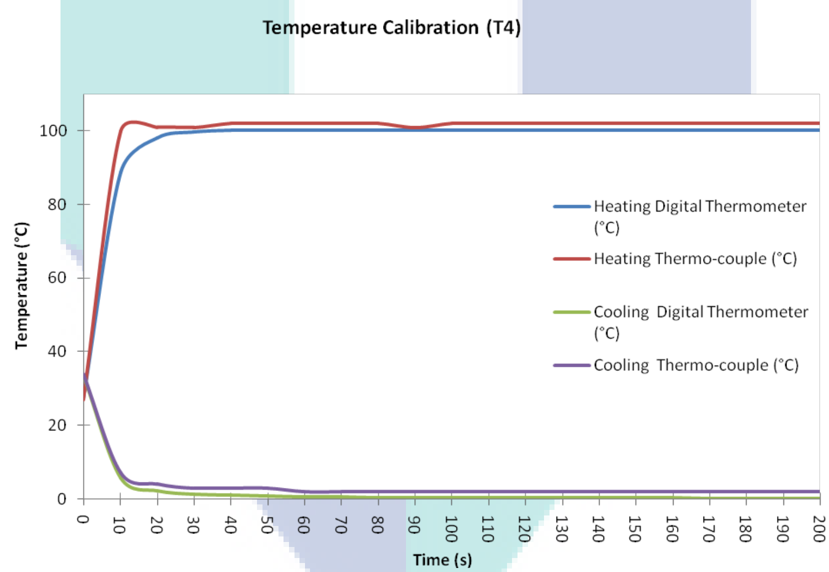
Calibration of Thermocouples

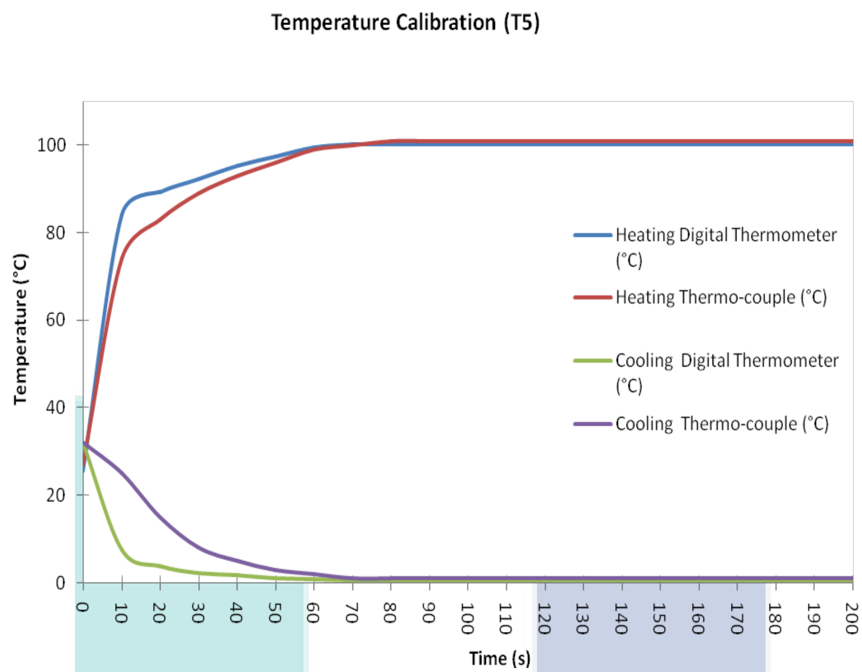
1. Calibration Graph

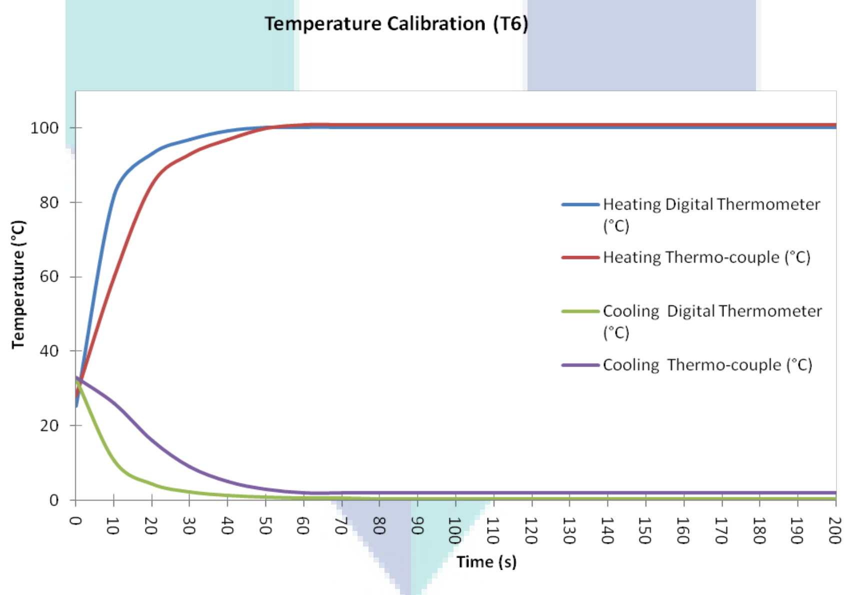





Temperature Calibration (T3)
Temperature (°C)
100
80
60
40
20
0
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
Time (s)
Heating Digital Thermometer (°C)
Heating Thermo-couple (°C)
Cooling Digital Thermometer (°C)
Cooling Thermo-couple (°C)


Temperature Calibration (T4)
Temperature (°C)
100
80
60
40
20
0
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
Time (s)
Heating Digital Thermometer (°C)
Heating Thermo-couple (°C)
Cooling Digital Thermometer (°C)
Cooling Thermo-couple (°C)


Temperature Calibration (T5)
Temperature (°C)
Time (s)
Heating Digital Thermometer (°C)
Heating Thermo-couple (°C)
Cooling Digital Thermometer (°C)
Cooling Thermo-couple (°C)


Temperature Calibration (T6)
Temperature (°C)
Time (s)
Heating Digital Thermometer (°C)
Heating Thermo-couple (°C)
Cooling Digital Thermometer (°C)
Cooling Thermo-couple (°C)

2. Calibration Data

Dynamometer Load Sensor Calibration (Without Water Condition)

Load (kg)	Theoretical Torque (Nm)	Measured Torque (Nm)	Derivation
5	26.9775	26.4	0.5775
10	53.955	52.7	1.255
15	80.9325	79.1	1.8325
20	107.91	105.7	2.21
25	245.25	132.4	2.4875

Dynamometer Load Sensor Calibration (With Water Condition)

Load (kg)	Theoretical Torque (Nm)	Measured Torque (Nm)	Derivation
5	26.9775	26.4	0.5775
10	53.955	52.8	1.155
15	80.9325	79.6	1.3325
20	107.91	106.3	1.61
25	245.25	133.0	1.8875

Temperature Calibration (T1)

Time (s)	Heating: Digital Thermometer (°C)	Heating: Thermo-couple (°C)	Cooling: Digital Thermometer (°C)	Cooling: Thermo-couple (°C)	Derivation (Hot)	Derivation (Cold)
0	25.3	27	32.6	33	1.7	0.4
10	86.7	82	13.4	15	4.7	1.6
20	96.3	96	6.1	8	0.3	1.9
30	98.7	98	3.3	5	0.7	1.7
40	98.8	99	1.8	3	0.2	1.2
50	98.6	99	1.1	3	0.4	1.9
60	98.3	98	0.7	2	0.3	1.3
70	98.7	99	0.6	2	0.3	1.4
80	98.7	99	0.5	2	0.3	1.5
90	99.5	99	0.4	2	0.5	0.6
100	99.8	100	0.4	1	0.2	0.6
110	99.6	100	0.3	1	0.4	0.7
120	99.0	100	0.3	1	1.0	0.7
130	98.8	99	0.3	1	0.2	0.7
140	99.4	99	0.3	1	0.4	0.7
150	99.8	100	0.3	1	0.2	0.7
160	99.6	100	0.3	1	0.4	0.7
170	99.6	100	0.3	1	0.4	0.7
180	99.6	100	0.3	1	0.4	0.7
190	99.6	100	0.3	1	0.4	0.7
200	99.6	100	0.3	1	0.4	0.7

Temperature Calibration (T2)

	Heating		Cooling			
Time (s)	Digital Thermometer (°C)	Thermo-couple (°C)	Digital Thermometer (°C)	Thermo-couple (°C)	Derivation (Hot)	Derivation (Cold)
0	26.1	28	33.5	33	1.9	0.5
10	89.3	98	25.1	11	8.7	14.1
20	98.0	100	22.4	7	2.0	15.4
30	99.3	100	18.4	6	0.7	15.4
40	99.2	100	15.1	5	0.8	10.1
50	99.7	101	11.8	5	1.3	6.8
60	100.2	101	9.9	5	0.8	4.9
70	100.2	101	8.1	5	0.8	3.1
80	100.2	101	6.5	5	0.8	1.5
90	100.2	101	5.4	5	0.8	0.4
100	100.2	101	4.4	5	0.8	0.6
110	100.2	101	3.7	5	0.8	1.3
120	100.2	101	3.1	4	0.8	0.9
130	100.2	101	2.5	3	0.8	0.5
140	100.2	101	2.1	3	0.8	0.9
150	100.2	101	1.7	5	0.8	3.3
160	100.2	101	1.4	4	0.8	2.6
170	100.2	101	1.2	5	0.8	3.8
180	100.2	101	1.1	5	0.8	3.9
190	100.2	101	0.9	6	0.8	5.1
200	100.2	101	0.8	6	0.8	5.2

Temperature Calibration (T3)

Time (s)	Heating Digital Thermometer (°C)	Heating Thermo-couple (°C)	Cooling Digital Thermometer (°C)	Cooling Thermo-couple (°C)	Derivation (Hot)	Derivation (Cold)
0	26.7	29	32.6	33	2.3	0.4
10	91	99	8.7	17	8	8.3
20	97.7	101	3.2	9	3.3	5.8
30	99.5	101	1.6	6	1.5	4.4
40	99.7	101	1.1	4	1.3	2.9
50	99.3	101	0.8	4	1.7	3.2
60	99.2	101	0.6	3	1.8	2.4
70	99.1	101	0.6	3	1.9	2.4
80	98.9	101	0.5	2	2.1	1.5
90	98.7	101	0.5	2	2.3	1.5
100	99.6	102	0.4	3	2.4	2.6
110	100.0	102	0.4	3	2.0	2.6
120	100.2	102	0.3	3	1.8	2.7
130	100.2	102	0.3	3	1.8	2.7
140	100.2	102	0.3	2	1.8	1.7
150	100.2	102	0.3	2	1.8	1.7
160	100.2	102	0.3	3	1.8	2.7
170	100.2	102	0.3	3	1.8	2.7
180	100.2	102	0.3	3	1.8	2.7
190	100.2	102	0.3	3	1.8	2.7
200	100.2	102	0.3	3	1.8	2.7

Temperature Calibration (T4)

Time (s)	Heating Digital Thermometer (°C)	Heating Thermo-couple (°C)	Cooling Digital Thermometer (°C)	Cooling Thermo-couple (°C)	Derivation (Hot)	Derivation (Cold)
0	27.3	27	33.6	34	0.3	0.4
10	88.7	100	5.8	7	11.3	1.2
20	98.1	101	2.3	4	2.9	1.7
30	99.7	101	1.3	3	1.3	1.7
40	100.2	102	1.0	3	1.8	2
50	100.3	102	0.7	3	1.7	2.3
60	100.3	102	0.6	2	1.7	1.4
70	100.3	102	0.5	2	1.7	1.5
80	100.3	102	0.4	2	1.7	1.6
90	100.3	101	0.4	2	0.7	1.6
100	100.3	102	0.3	2	1.7	1.7
110	100.2	102	0.3	2	1.8	1.7
120	100.3	102	0.3	2	1.7	1.7
130	100.3	102	0.3	2	1.7	1.7
140	100.3	102	0.3	2	1.7	1.7
150	100.3	102	0.3	2	1.7	1.7
160	100.3	102	0.3	2	1.7	1.7
170	100.3	102	0.2	2	1.7	1.8
180	100.3	102	0.2	2	1.7	1.8
190	100.3	102	0.2	2	1.7	1.8
200	100.3	102	0.2	2	1.7	1.8

Temperature Calibration (T5)

Time (s)	Heating Digital Thermometer (°C)	Heating Thermo-couple (°C)	Cooling Digital Thermometer (°C)	Cooling Thermo-couple (°C)	Derivation (Hot)	Derivation (Cold)
0	25.6	27	32.1	32	1.4	0.1
10	84.0	74	7.5	25	10.0	17.5
20	89.3	83	3.8	15	6.3	11.2
30	92.2	89	2.3	8	3.2	5.7
40	95.2	93	1.7	5	2.2	3.3
50	97.4	96	1.1	3	1.4	1.9
60	99.4	99	0.8	2	0.4	1.2
70	100.1	100	0.6	1	0.1	0.4
80	100.2	101	0.5	1	0.8	0.5
90	100.2	101	0.4	1	0.8	0.6
100	100.2	101	0.4	1	0.8	0.6
110	100.2	101	0.3	1	0.8	0.7
120	100.2	101	0.3	1	0.8	0.7
130	100.2	101	0.3	1	0.8	0.7
140	100.2	101	0.3	1	0.8	0.7
150	100.2	101	0.3	1	0.8	0.7
160	100.2	101	0.3	1	0.8	0.7
170	100.2	101	0.3	1	0.8	0.7
180	100.2	101	0.2	1	0.8	0.8
190	100.2	101	0.2	1	0.8	0.8
200	100.2	101	0.2	1	0.8	0.8

Temperature Calibration (T6)

Time (s)	Heating Digital Thermometer (°C)	Heating Thermo-couple (°C)	Cooling Digital Thermometer (°C)	Cooling Thermo-couple (°C)	Derivation (Hot)	Derivation (Cold)
0	25.3	28	32.6	33	2.7	0.4
10	82.1	60	10.6	26	22.1	15.4
20	93.2	85	4.4	16	8.2	11.6
30	97.0	93	2.3	9	4.0	6.7
40	99.3	97	1.3	5	2.3	3.7
50	100.1	100	0.8	3	0.1	2.2
60	100.3	101	0.6	2	0.7	1.4
70	100.3	101	0.5	2	0.7	1.5
80	100.3	101	0.4	2	0.7	1.6
90	100.3	101	0.3	2	0.7	1.7
100	100.3	101	0.3	2	0.7	1.7
110	100.3	101	0.3	2	0.7	1.7
120	100.3	101	0.3	2	0.7	1.7
130	100.3	101	0.3	2	0.7	1.7
140	100.3	101	0.3	2	0.7	1.7
150	100.3	101	0.3	2	0.7	1.7
160	100.3	101	0.3	2	0.7	1.7
170	100.3	101	0.3	2	0.7	1.7
180	100.3	101	0.3	2	0.7	1.7
190	100.3	101	0.3	2	0.7	1.7
200	100.3	101	0.3	2	0.7	1.7

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3. Awang Idris, Rosli Abu Bakar, "*The Trends of Research in Compressed Natural Gas Engine-vehicle Technology: An Overview*". Malaysian Technical Universities on Engineering and Technology 2006. KUiTTHO, Batu Pahat Johor. 19-20 December 2006.

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