OPTIMIZING THE COMPRESSION RATIO OF A CNG CONVERTED DIESEL ENGINE

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CERTIFICATE OF APPROVAL

The thesis titled "Optimizing the Compression Ratio of a CNG Converted Diesel Engine**."** Submitted by Golam Saklayen, Roll No: 100710041F, Session: October-2007 has been accepted as satisfactory in partial fulfillment of the requirement for the degree of the Master of Science in Mechanical Engineering.

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This is to certify that the work presented in this thesis is an outcome of the investigation carried out by the author under the supervision of Dr. Md. Ehsan, Professor, Department of Mechanical Engineering, Bangladesh University of Engineering and Technology (BUET), Dhaka and it has not been submitted elsewhere for the award of any degree or diploma.

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Dedicated

To

My parents

Who decay to let us grow

ABSTRACT

An experimental investigation on optimizing the compression ratio (CR) of diesel to retrofitted complete CNG converted engine was conducted. The investigation was done by experimental setup buildup.

In the present experimental study a CNG converted diesel engine (HINO WO4D, 4.009 liter) was used for investigation. The modified diesel engine was fitted with a 3-stage diaphragm type pressure regulator CNG fuel system, a mixing chamber with throttle and a spark ignition system. The converted engine was run only CNG.

The existing diesel engine compression ratio of 19.2 would be reduced to a range between 11 to 13. The clearance volume in the cylinder was increased by removing the material from the piston top surface, while keeping the stroke lengths the same. During change of each piston set the gaskets, piston rings and lubricating oil was changed to keep the physical parameters to be identical. Engine performance measurements were taken for each set of compression ratios using a hydraulic dynamometer setup with necessary instrumentation. The wide open throttle conditions were used for evaluation of engine performance. A theoretical study of the comparative analysis of the stresses on the piston top was done for different piston top thicknesses and all was found within safe range.

The investigation was carried out in wide open throttle (WOT) and 1450 to 2300 rpm as better running condition and data obtained was compared within three CRs and the better or optimum performance CR setup was selected. Basing on the peak pressure, volumetric efficiency , brake specific fuel consumption (Bsfc) and temperature distribution for CR=12 was considered as an optimized performance setup for a diesel bus engine converted to CNG.

ACKNOWLEDGEMENTS

The author wishes to express his sincere appreciation to Professor Dr. Md. Ehsan of Mechanical Engineering Department, Bangladesh University of Engineering and Technology (BUET), Dhaka, Bangladesh for his guidance, invaluable suggestions and constructive criticism throughout this investigation and for painstakingly reading the manuscript and suggesting its improvement. His encouragement, patience and careful supervisions are gratefully acknowledged.

The author is also indebted to Mr Aynul Haque Assistant manager Navana CNG Ltd, for his invaluable suggestions and cooperation in all phases of CNG conversion and CNG diagnostic checkup.

The author gratefully acknowledges the financial assistance given by Bangladesh University of Engineering and Technology (BUET), Dhaka.

Sincere thanks are offered to the staff of the machine shop, welding shop of mechanical engineering department of BUET, Dhaka for their kind co-operation in constructing, fabricating and assembling different parts and components of the experimental set-up and special thanks to Md. Rokon Uddin, instructor of Heat engine laboratory mechanical engineering department of BUET, Dhaka for assisting in all phases of the investigation including his full time presence.

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NOMENCLATURE

BHP	Brake horse power
Bmep	Brake mean effective pressure
Bsfc	Brake specific fuel consumption
C	Half the distance of the height at failure surface
C_d	Co-efficient of discharge
CR	Compression ratio of the engine
$D_{orifice}$	Orifice diameter
Esfc	Equivalent diesel specific fuel consumption
H	Manometer deflection
I	Moment of inertia
M	Bending moment of piston element
$\dot{m}_{ng}$	Mass flow rate of natural gas
N	Engine rpm
$\dot{Q}_{f}$	Volume flow rate of natural gas
LHV_{g}	Lower heating value of natural gas
LHV_{diesel}	Lower heating value of diesel
V_d	Piston displacement volume
W	Uniform Load
W_L	Applied Load
ρ_{water}	Density of Water
ρ_{air}	Density of air
η_{v}	Volumetric efficiency
η_{t}	Thermal efficiency

CHAPTER-1
INTRODUCTION

1.1 General

The present crisis of conventional fossil fuels are widening the application of high potential alternative fuels like natural gas. Natural gas is an excellent engine fuel. Its combustion and emissions characteristics are superior to any other realistic competing fuel; it provides the engine designer with considerable flexibility in the selection of design parameters. To make the most of its advantages it is necessary to design an engine to match its specific properties. Moreover, the economic and environmental benefits have directed many existing internal combustion engine users to modify there hardware for using natural gas. The better alternatives for a diesel engine is either to use natural gas with some diesel in dual fuel mode or physically convert the diesel engine almost into a petrol engine which could be run on natural gas only. The second option involves major hardware change, but has the advantage of using a single fuel ie. natural gas, which is still much cheaper. The less availability and high price of petrol/gas engines that could be used for powering buses, has also made the second option more popular in many parts of the developing world, specially in Bangladesh.

All diesel engines can be converted to natural gas engine. Where the power level of the engine after conversion depends on numerous issues, such as natural gas quality, compression ratio (CR), power level of the original diesel engine, emission levels required etc. The compression ratio of the typical diesel engine, when converted directly to spark ignition operation, is too high to operate on natural gas without knock. So the compression ratio needs to be decreased. A properly converted engine can make almost as much power using natural gas with diesel. Diesel engines converted to natural gas generally require added components as well as some mechanical changes to the engine.

There is very limited systematic scientific study of setting the conversion parameters, particularly the compression ratio for optimized operation of such CNG converted diesel engines in Bangladesh. The items related to the present study will be discussed briefly in subsequent paragraphs and chapters.

1.2 CNG as an Alternative Fuel

Natural gas is a clear, odorless, gaseous mixture of hydrocarbons primarily composed of methane. Natural gas is commonly used in power generation and industry as well as in homes for heating and cooking. In vehicles, it is used as compressed natural gas (CNG), which is stored in special cylinder tanks at pressure of 3,000 psi. Natural gas is drawn from wells or extracted in conjunction with crude oil production.

Natural gas has been used in light and heavy duty natural gas vehicles (NGVs) since the mid 1960s. As with all fuels, natural gas is flammable. Natural gas is lighter than air and quickly dissipates when released. Natural gas is only flammable when the fuel/air mixture is in between 5% and 15% by volume. Below 5%, there is not enough fuel for ignition, and above 15% there is not enough oxygen for ignition [1].

The composition of NG also varies between countries. The principle ingredient of NG is methane. Methane makes up to 96% of NG. Aside of methane, the composition contains small portions of other gases such as ethane, propane, butane, pentane and hexane. It can also contain nitrogen, helium, carbon dioxide and hydrogen sulphide. In Bangladesh natural gas contains 95% methane with a very low amount of sulphur. The composition of NG obtained from different gas field of Bangladesh was shown in Table E-1 Appendix E.

CNG has few safety features or properties (Table-E-2, Appendix E) that make it an inherently safer fuel than petrol or diesel. Those are:

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a) Higher Octane number in the range of 120 to 130, which is considerably higher than 93 to 99 Octane for good quality petrol. A high Octane number ensures that CNG fuel can run at higher compression ratio without any knocking phenomena to the piston that will cause damage to the engine

b) NG has a specific gravity of 0.65. This means it is lighter than air so if leakage occurs from cylinders, it just rises up and dissipates into the atmosphere.

c) It has a self ignition temperature of 700^0C as opposed to 455^0C of petrol which helps for prevention of knocking.

d) For combustion to take place, CNG has to mix with air within a narrower range within 5-15% by volume.

e) CNG cylinder designed and manufactured with special material to withstand the high safety specification, which makes fuel storage more safer than petrol or diesel tank.

1.3 CNG Conversion of Diesel Engine

There are two methods of converting diesel engine into CNG engine to utilize NG as the main fuel. Those are :

a) *Dual-fuel*, where the vehicle runs both on diesel and natural gas with the combustion of diesel used to ignite the natural gas.
b) *Dedicated*, which runs entirely on natural gas instead of diesel.

A third method (bi-fuel) applicable for SI engine. All the three types can be manufactured from the start to use natural gas by original equipment manufacturers (OEM) as dedicated conversion or converted as retrofitted from engines/vehicles that were originally manufactured to run on diesel only .

1.3.1 Retrofitted Conversion

This option is a dedicated gas system which is to replace diesel fuel system with a similar gas fuel air ignition system that has been designed and setup to run on gas only shown in Figure-1.1. The system is comparatively cheaper however it only allows us to drive on CNG.

Figure-1.1: A Truck Van with Retrofitted Complete CNG Conversion

To convert to retrofitted CNG engine the conversion activities /modifications as follows:

i) Removal of the diesel injection system and replacing it with an spark ignition system.

ii) Installation of a gas carburetor or gas injection system as required to deliver the CNG.

iii) Modifying pistons to reduce the compression ratio or replace with new pistons of a selected design and material. The new compression ratio will normally be kept between 11 and 15 depending on the gas composition.

iv) Modifying the cylinder head to accommodate a spark plug or design and manufacture a new gas head as required. This new head may be available from the factory or may need to be specially engineered and manufactured for the conversion.

v) Inlet and exhaust valves and seats from the diesel engine may be used providing the materials are compatible for the use of gas otherwise they must be replaced with valves and seats of selected material. This is especially important for a turbocharged engine.

vi) Mechanical and electronic installation of the kit, installing components such as gas regulator, valves, cylinder, fuel switch, hoses etc. Remove the diesel injection system and replace with an spark ignition system.

A typical standard retrofit CNG conversion system essentially consists of the following components, the locations are indicated in Figure- 1.2 :

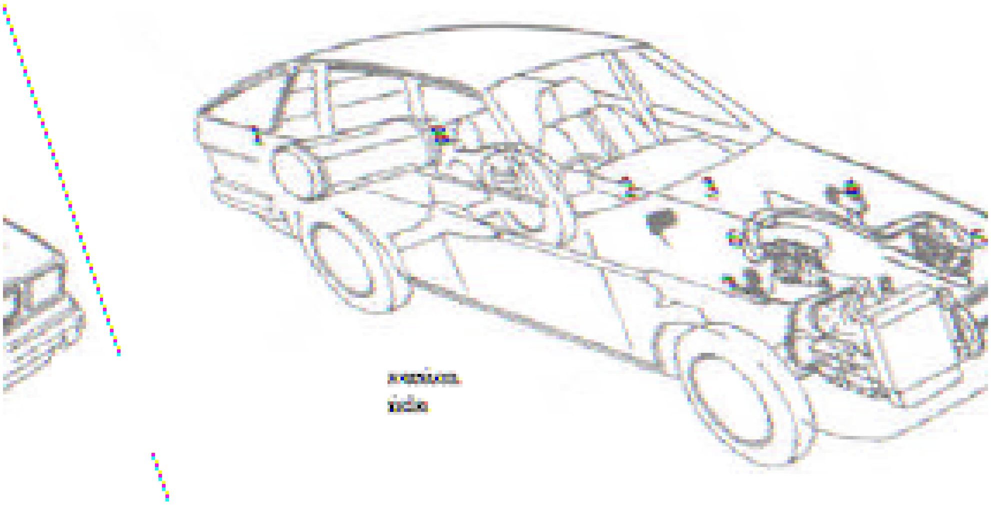


Figure-1.2: Retrofitted CNG Conversion components (1. CNG Storage Cylinder, 2. Fuel Selector Switch, 3. High Pressure Piping,4. Refueling Valve, 5. Pressure

Regulator, 6. Gas-Air mixing Chamber,7. Petrol Solenoid Valve , 8. Gas Vapor Enclosure)

i) CNG Storage Cylinder

The Cylinder is used to store CNG at a working pressure of 200 bar. It is fitted with a shut-off valve and a safety burst disc. Cylinders may be of seamless steel structure or reinforced carbon fiber composite. 50, 60, 90 water liter size cylinders are commonly used. At a pressure of 200 bar and ambient temperature CNG is compressed to about 3.7 times of equivalent petrol volume. Due to the cylindrical shape and wall thickness the packing efficiency is poor and the real car space is occupied by the cylinders is much more. The shut-off valve with the cylinder may have overflow limiter facilities built-in.

ii) Fuel Selector Switch

The selector switch is fitted at the dash board, enabling the driver to choose either the CNG mode or the petrol /diesel mode of operation. The electronics built in this unit also ensures safety by switching off the gas solenoid whenever the engine is switched off. It also indicates the quantity of CNG available in the cylinder.

iii) High Pressure Piping

The high pressure pipe connects the refueling valve to the CNG cylinder and pressure regulator.

iv) Refueling Valve

The refueling valve is used to refuel the CNG cylinder.

v) Pressure regulator

The pressure regulator has a solenoid valve to shut-off gas supply to the engine. The CNG stored at a high pressure in the cylinder, is reduced to just below the atmospheric pressure by this unit. This negative pressure is also a safety feature that will not allow gas to pass through when the engine is not running. Generally this is provided with a gas filter element.

vi) Gas-Air mixing Chamber

The gas-air mixer is a unique component, specially designed to suit each engine model. It precisely meters gas fed into the engine.

vii) Petrol Solenoid valve

Petrol solenoid valve is used to cut off petrol supply to the engine when it is run on CNG.

vii) Gas vapor Enclosure

Fitted near the cylinder, the vapor bag/enclosure is used to enclose the cylinder valve and the pipes connecting it and is vented out of the car.

Engine which is equipped with electronic fuel injection (EFI) system and electronic control unit (ECU) include the following components:

i) Spark Timing Advance Processor (STAP/TAP)

The slower flame propagation speed in methane-air mixture makes it necessary to increase the spark advance for CNG operation of a petrol engine. This unit allows optional spark advances (eg. additional 6^0, 9^0, 12^0, 15^0 BTDC), generally during accelerating conditions. The processor modifies the spark advance signal to the electronic ignition system accordingly. Use of this improves the peak power and acceleration performance of the vehicle running on CNG, while retaining the performance with petrol.

ii) Emulator/Simulator

The EFI injectors are generally kept non-functional at CNG operation. On the other hand, in some engines feedback signals from the fuel injectors operation are sent back to the electronic control units(ECU). Under CNG operation the emulator simulates dummy feedback signals to the ECU, maintaining its normal operation as with petrol. This component may not be necessary for all models of EFI engines.

iii) Oxygen Sensors and Speed sensors

Often these are required in close-loop gas flow-control CNG conversion systems.

iii) Active Mixing Valve

Active mixing valve controls the flow of gas into the mixing chamber according to engine requirements sensed.

iii) Gas Injectors

Gas injectors are used for conversion systems, where just like multipoint fuel injection system, individual gas injectors control gas flow into each cylinder controlled by a processor unit.

1.3.2 Dual fuel Diesel-CNG conversion

The second method, which is to mix a small amount of diesel fuel with the natural gas in the combustion chamber. The diesel ignites from the piston compression, subsequently igniting the natural gas. This is commonly referred to as dual-fuel operation. This dual-fuel approach has definite disadvantages as it provides poor low-speed performance, produces higher exhaust emissions, and has a tendency to knock [2]. The compression ratio is kept the safer for both diesel and dual operation.

Figure-1.3: Duel Fuel Diesel-CNG engine

In dual fuel operation, the gaseous fuel-air mixture mixing with air intake – fumigation. This mixture is compressed during the compression stroke. Near the end of the compression stroke, diesel fuel is injected in order to initiate the combustion of gaseous-air mixture. Due to its function to start the combustion, the diesel in dual fuel operation is referred to as pilot diesel. The changeover of the mode from duel-fuel to straight diesel or vice versa can take place while the engine runs. Engine may get 20% diesel and 80% gas. The conversion cost is less and saving would be more than 50% [3]. Since compression ratio is not changed, the knocking may limit the order of replaced of diesel by gas.

1.3.3 Omnitek Retrofit Conversion Technology

Omnitek is the provider of a real world solution for diesel engines. The Omnitek Diesel-to-Natural Gas Conversion Technology as in Figure-1.4 is used around the world. The system converts diesel engines into clean burning natural gas engines at a fraction of the cost of a new engine.

Omnitek is currently providing this solution to customers in 12 countries and has been used to converted over 4000 diesel engines around the world.

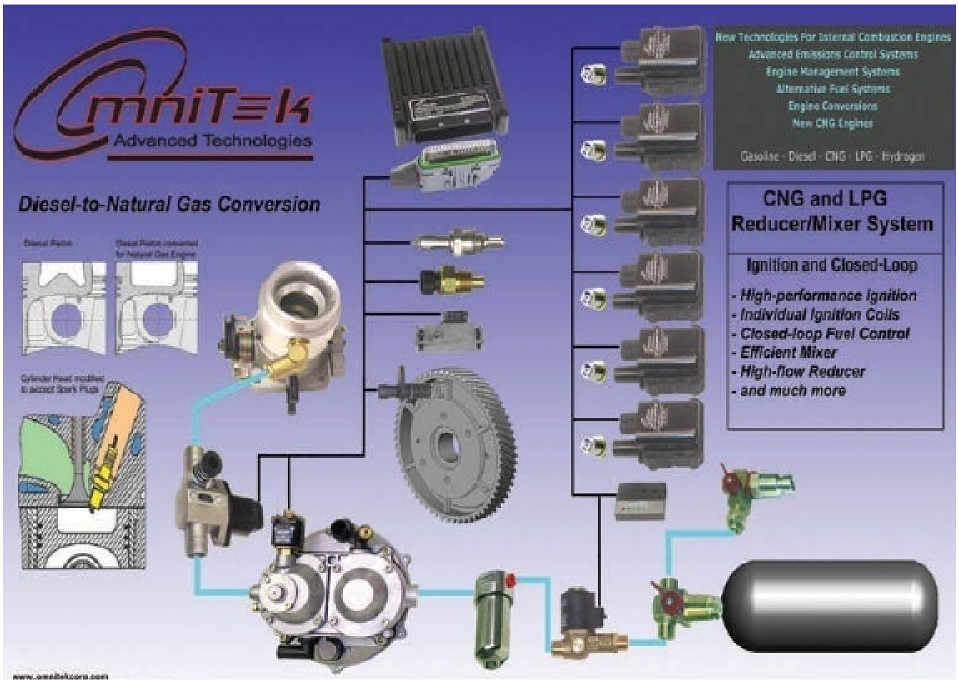


Figure-1.4: Omnitek CNG Conversion kits

Conversion kits for diesel engines come in many sizes and can be divided into two groups:

i) Engines without a turbocharger - can use a simple reducer/mixer system (CIMS).

ii) Engines with a turbocharger - must use electronic fuel injection (ECM).

In Omnitek conversion basically the diesel engine undergoes a major rebuilt and is in the process of transformed from a diesel engine to a gas engine. The steps of conversion include the followings [4]:

i) Disassemble engine.

ii) Checking of components and replacement as necessary.

iii) Modification of pistons for gas use (lower compression ratio).

iv) Modification of cylinder head for spark plugs.

v) Installation of camshaft sensor and timing wheel.

vi) Reassemble engine.

vii) Installation of throttle body, ignition system, gas mixer or fuel injectors.

viii) Tuning of the engine (fuel and ignition).

Details parameters of the Omnitek components will be discussed in Chapter 3 under section 3.2.

1.4 Variation of Compression Ratio

The compression ratio is an important parameter that can be used to predict the performance of any internal-combustion engine. It is a ratio between the gas volumes in the combustion chamber and cylinder, when the piston is at the bottom of its stroke and the volume when the piston is at the top of its stroke.

The ratio is calculated by the following formula:

$$\mathrm{CR} = \frac{(\pi b^2 s)/4 + V_c}{V_c} \tag{1.1}$$

where ,

b = cylinder bore (diameter)

s = piston stroke length

V_c = volume of the combustion chamber (including head gasket interface gap). This is the minimum volume of the space into which the fuel and air is compressed, prior to ignition. Because of the complex shape of this space, it usually is measured directly rather than calculated.

Generally Engine power, peak pressure, torque and fuel conversion efficiency increases with increase of CR. Since the original diesel engine operates under a high compression ratio, the conversion to CNG engine system might promote knocking in the combustion chamber due to self-ignition characteristic of CNG. Although CNG has a high octane rating of 130, it will still self ignite when operates in a high pressure combustion chamber due to compression effect of the compression stroke. Therefore, the compression ratio of the engine needs to be modified and reduced to obtain the optimum performance for the CNG converted diesel engine.

Diesel engine can work better in higher CR as 17-22 and produce higher power and torque. To convert it to complete CNG operated the compression ratio to be reduced in a range between 11 to 15 [5], [6] and [7]. CR can be adjusted by three different methods [8]:

a) Modifying of piston groove or bowl

In this method piston is modified by milling the piston head to create a recessed bowl shape. The size of the bowl depends on the size of the piston. It is usually suitable for large piston because small piston with recessed bowl can cause thermal stress to build up in the piston head and piston skirt. This may cause failure to the piston.

b) Modifying the length of the connecting rod

Here the connecting rod length is adjusted to adjust the displacement volume of each cylinder thus CR varies. This method is very costly and complicated to be constructed. Improper design will cause vibration and thermal stress to build up in the piston.

c) Insertion of plate onto the piston head

In this method a plate with a known thickness is added between the piston head and the cylinder block and act as a spaces and seal between the engine block and the piston head . The shape of the plate will follow the shape of the top of the piston head. It is less costly, easier to construct and no complicated calculation require like other methods. However cam operation needs to be modified to adjust with the extra thickness of the plate inserted.

d) Other methods

Shorter cam duration can improve the effectiveness of increasing the CR. Besides cam duration, the cam's Lobe Centerline Angle (LCA) and the cam advance are important to increase the CR. A wider LCA (numbers getting larger) promotes greater increase in CR than a tight LCA (numbers getting smaller) .

In Chapter 3 details on the design procedure for the modification of piston head to reduce the compression ratio was explained further.

1.5 Necessity of the Study

Over last three decades research and development on the engine has been addressing about the better combustion technology within the cylinder volume. With advanced technology the use of alternative fuel in diesel engine is possible with some modifications. The CR of diesel engine is to be decreased in order to convert it to CNG for reducing knock. Thus the power output reduces and less power may be obtained. In our country the diesel buses converted to CNG at a lower CR may not be suitable for the best overall performance. No such research has been carried out for the converted CNG engines in our country and no data available for this. Most of the companies are doing the compression reduction based on theoretical trial and error method through field experiences. So, to investigate the best CR of CNG bus engine by NG composition of Bangladesh, this is a relevant topic to be studied. The data from these research works could be useful to the engineers, CNG conversion companies and the researchers of Bangladesh to design and set the correct/ optimum conversion parameters.

1.6 Objective with specific aim and possible outcome

In the study of optimizing the compression ratio of CNG converted diesel engine the specific objectives are as follows:

a) To operate the converted diesel engines with different compression ratios, using pistons of different clearance volumes.
b) To study the performance of a CNG converted diesel engine with different compression ratios.
c) To study the comparative stress levels in the modified piston.
d) To optimize the compression ratio based on the best overall performance using locally available compressed natural gas as fuel.
e) To identify the most suitable compression ratio of operating CNG converted diesel engines used for buses in Bangladesh.

1.7 Scope of the Study

This section contains the brief description of the different themes which has been presented in the various chapters.

The brief description of CNG conversion of Diesel engine with its types and compression ratio variations have been incorporated in **chapter 1**. The importance of the study and the aim of the study have also been included in this chapter. In **chapter 2** the brief survey of the various related literatures have been provided. Usually the research works which are directly related to the present study have been included in this chapter. Some works which are inline with the present study have also been included.

The description of the experimental set-up, parameter measurement setup and data acquisition system have been included in **chapter 3** in a nutshell.

The most important part of the thesis is the results and discussion, which has been provided in **chapter 4**. The performance parameters are compared for different CR have been discussed elaborately in this chapter. Finally in **chapter 5** the conclusions and the recommendations for future researchers have been given.

CHAPTER-2
REVIEW OF LITERATURE

2.1 General

With the decreasing of world oil reserve engine users and technologists become partially biased to the use of alternative fuel. There are number of research groups, automobile companies, environmental workers and the inventors worked very hard to get the best possible type of fuel replacement of the conventional one. By then millions of vehicles have been converted to alternative fuel as NGVs. Research on different parameter change and performance studies have been carried out on those types of NGVs [8]. It was found that NG composition has a significant role in combustion. So the research has been carried out on different composition of NGs at different types of engines. Later some of them worked with variation of effects on engine performance at variable compression ratio. Some simulation carried out by computational fluid dynamics (CFD) to simulate the activities at different compression ratio for CNG engine [9]. A study on a retrofitted complete converted bus engine data and results are not probably available in detail, although this is a problem of practical significance. In this chapter a brief description of some of the papers related to the present state of the problem will be mentioned.

2.2 Existing Research Work

The parameters of variable compression ratio have been studied by a number of research groups as in [5], [6], [8], [9], [10] and [11]. Olsson et al [5] showed that compression ratio influence on maximum load of a natural gas turbocharged homogeneous charged compression injection (HCCI) engine. In his experiment piston with interchangeable flat steel crowns was used to allow 4 different compression ratios between 15:1 and 21:1. In the study it was found that compression ratio demonstrated no significant difference in the later ignition timing, rate of heat release is little or not at all affected by compression ratio. Peak cylinder pressure is higher for a higher compression ratio but NOx emissions are lower

Rao et al [9] carried out experiments on a single cylinder four stroke variable compression ratio diesel engine. Test were carried out for 7 compression ratios between 13.2 and 20.2. Results showed an improved performance and emission characteristics at compression ratio 14.8, where the compression ratio lesser or greater than 14.8 showed a drop in brake thermal efficiency and emission Characteristics.

LIM Pei Li [8] studied the effects of different compression ratio on the performance of a four stroke, Ford direct injection diesel engine operating on dual fuel system using CNG as the main fuel. Based on CFD method using FLUENT simulation software the compression ratio chosen as optimized value to operate the CNG-diesel engine without knock was found to be 16.6:1 .

Nirendra N. Mustafi et al [2] carried out research study on influence of particulate matter (PM) during vehicle emission. PM emission for dual fueling is found to be reduced by about 70%. Results are compared between the diesel and dual-fuel operations and also using between natural gas or biogas fueling for a particular engine operating condition.

S Dasappa [6] worked on power output prediction of diesel engine converted to gaseous engine by empirical simulation, where compression ratio being one of the primary factors. The simulation data and its comparison with other references predicted a change of 1.5% in efficiency for unit change of compression ratio for gas engines under consideration.

Lim Pei Li [8] investigated that for an unit increase in CR, the peak cylinder pressure increases with approximately 100-110 psi. Consequently, component parts of the engine such as the head gasket, connecting rods, crank and blocks are subjected to thermal stress. More CR is more susceptible to knocking . Preventive methods are to find a fuel that has a high octane rating about 115 or higher, a better design on the piston bore or the combustion chamber to improve the swirling action and the injection timing.

Korean Car makers Kia (sportage 83 liters,1520 kg) , Deawoo (Cielo 75 liters, 1103.5 kg) and Hyundai (Accent 95.1 Liters, 970 kg) performed the field test of natural gas buses for the first time in Korea during 1998. From the field test it was ascertained that performance of NGV found similar to that of gasoline and diesel except the NOx, HC reduction by 30-40% and PM emission is zero [12].

Number of research based studies have been carried out regarding variable compression ratio (VCR) engines [10,11]. Variable compression ratio (VCR) technology has long been recognized as a method for improving the fuel economy . Various designs include modification of the compression ratio by moving the cylinder head, variation of combustion chamber volume using a secondary piston or valve, variation of piston deck height (top thickness) , modification of connecting rod geometry, moving the crank pin within the crankshaft, and moving the crankshaft axis [10].

To control the VCR of the combustion chamber a mechanism invented by a group of inventors . The mechanism with modification can also be employed to control the maximum compression pressure or the maximum combustion pressure of the engine [11].

D & M Performance Machine Shop [7] suggested that CR in a gasoline/petrol powered engine with detonation should usually not be much higher than 10:1. Engines running exclusively on LPG or CNG, the CR may be higher, due to the higher octane rating of these fuels. Engine with a 'ping' or 'knock' sensor and an electronic control unit (ECU), the CR can be as high as 13:1 (2005 BMW K1200S) .In a turbo charged or super charged engine, the CR will be around 8.5:1. In a diesel engine, the CR will be 14:1 and higher.

T. M. French [13] presented in his technical note that the results on fuel efficiency (miles per equivalent gallon) for CNG vehicles are not conclusive. Different tests and studies have varied results ranging from losses to gains in fuel efficiency relative to gasoline or diesel for vehicles converted to CNG. Basically, a vehicle designed for and dedicated to CNG use only should be expected to have a greater efficiency than a gasoline or diesel fueled engine, with an efficiency improvement in the range of 5 to 15% relative to a gasoline engine.

According to National Renewable Energy Laboratory (NREL) [14] from 2004 transit buses are a key role market for natural gas vehicles. Increasingly, transit agencies have been choosing natural gas buses as a way to cut air pollution and boost energy security. American Public Transit Association (APTA) reported ,approximately 12% of transit buses were powered by natural gas and 25% buses were waiting for NG conversion.

California Air Resources Board [15] reported from their statistics that the variation in CNG composition seen throughout the South Central Coast and southern San Joaquin Valley that can adversely affect engine performance. The effects can include misfire, stumble and underrated operation as well as engine knock and overheating. It was confirmed that those effects are dependent on the engine’s ability to tolerate or compensate for the variation in fuel composition.

According to project report for Bangladesh Road Transport Corporation [3].NG of Bangladesh typically contains mostly methane 95-97% which does not contain toxic elements such as mercury or arsenic. CNG vehicles emit more NOx in comparison with gasoline, however de-NOx converters would be able to reduce the NOx emissions to acceptable levels. CNG results in 10-20 .percent less power output with existing engines.

Position paper of International Association for Natural Gas Vehicles (IANGV) [16] reported that thermal efficiency of an engine increases with an increase of CR. In practice the higher friction losses that go with this increase result in an efficiency maximum being reached at a CR in the range 15 to 18. Since natural gas has a high octane rating (knock resistance), it can be used in naturally aspirated engines with CR values not far from the optimum. Petrol is knock limited and cannot be used at CR values needed for optimum efficiency; it is generally used in engines with CR values in the range 8 to 10. The high compression ratio of the diesel allows high thermal efficiencies to be attained. These efficiencies can generally be matched at full load with natural gas using the Otto cycle since the knock limited compression ratio is quite close to the optimum.

Fire apparatus manufacturers' association (FAMA) [17] reported with the experience that during comparison between a Cummins C8.3 diesel engine and Cummins C-Gas Plus engine showed an average fuel economy that was 5.17 mpeg (miles per energy equivalent gallon) for the natural gas trucks and 6.73 mpeg for the diesel trucks. This represented a 23.2% fuel economy penalty for the natural gas trucks. It is reasonable to assume, therefore, that fire apparatus powered by natural gas engines will also have a lower fuel efficiency. They further reported that fleet testing of CNG pick-up and delivery vehicles by UPS also showed a diesel-equivalent fuel economy penalty of 27 to 29%.

Bhuiyan [18] investigated that a petrol generator run on closely regulated natural gas flow showed improvement in several aspects. The A/f ratio was found to be between 16-21.The air flow rate was found to be about 55-65% of the petrol. The overall efficiency was found to be improved reaching about 16% at rated capacity. At full load condition the bsfc value was significantly less than the petrol data.

Tanjia Zakir et al [19] investigated the performance test on a diesel engine with partial replacement of diesel 60-100% with NG at 2250 rpm. Where it was found that power decreases to 75.50 % when diesel replacement was 60% . The maximum brake thermal efficiency was noticed 33.12% at 60% and more in 100% diesel replacement. Diesel mass flow was reflected equivalent to 20% of NG flow. A conclusion was made that total bsfc in dual fuel operation usually found lower value and relatively more economical in case of fuel consumption consideration.

S.M Nuruzzaman [20] investigated the effects on a diesel engine with partial replacement of diesel 30-100% with NG at 2200 rpm with an engine model F1L21OD with CR=17. Where it was found that volumetric efficiency and thermal efficiency was 73% and 33.55% respectively for 100% diesel. For some load the thermal efficiency reduces to 17% for 30% diesel and 70% NG condition. Same conclusion as [18] was made that total bsfc in dual fuel operation usually found lower value and relatively more economical in case of fuel consumption consideration.

Beside this number of inventors and conversion companies working with the latest technologies to improve the power of the converted engine in terms of fuel feeding, variable compression ratio improvement, environmental benefit and economy set up of engine [4],[8],[16],[21],[22],[23],[24] etc.

CHAPTER-3
EXPERIMENTAL SET-UP

3.1 General

In experimental investigation to find the optimal compression ratio of a CNG engine, a diesel engine (HINO WO4D, 4 liter) was converted to CNG . The test was conducted with three set of pistons of different clearance volumes. The displacement volume was kept the same, hence the CR varied. The applied load was calculated by coupling a hydraulic dynamometer and operating the engine at wide open throttle (WOT). In this chapter a brief description regarding the construction of the experimental setup, performance parameters setup and data acquisition method have been provided systematically.

3.2 Experimental Set up

The preparation of buildup for experimental setup was completed in two phases (i) Diesel engine conversion and preparing the engine for test and (ii) Performance parameter setup

3.2.1 Diesel Engine Specification (Model: HINO WO4D) before Conversion:

Table 3.1: Engine Specification before Conversion

Type		4 stroke, Diesel, Vertical 4 cylinder in line, Overhead valve, Water cooled.
Combustion system		Direct injection
Bore and stroke		104 × 118
Piston displacement		4.009 L
Compression ratio		19.2 : 1
Max. output		85 KW at 3200 rpm/min
Max. torque		284 N-m at 1900 rpm/min
Max. Engine speed		3200 rpm/min
Compression pressure		3.53 – 3.82 MPa at 350 rpm
Idling revolution		600 – 650 rpm
Firing order		1-3-4-2 (A number of a cylinder is counted from the timing gear side)
Aspiration		Naturally aspirated.
Direction of rotation		Counter clock wise viewed from flywheel.
Dry weight		Approximately 3285 N
Valve seat angle	Intake	30^0 BTDC
	Exhaust	45^0 BTDC
Valve face angle	Intake	30^0 BTDC
	Exhaust	45^0 BTDC
Valve Timing (flywheel travel)	Intake opens	8^0 Before top dead center
	Intake closes	48^0 After bottom dead center
	Exhaust opens	60^0 Before bottom dead center
	Exhaust closes	8^0 After top dead center
Valve clearance (When cold)	Intake	0.30 mm
	Exhaust	0.45 mm
Engine oil pump	Type	Full forced pressure feed by gear pump

	Drive	By gear from the cam shaft
Engine oil cooler		Multi plates type, water cooled
Injection nozzle	Type	Multi hole nozzle type
	Nozzle opening pressure.	21.57 MPa
Coolant pump	Type	Forced circulation by volute pump
	Drive	By V-belt from crank shaft
Thermostat	Type	Wax type, bottom bypass system.
	Valve opening temperature	82^{0} C

3.2.2 Necessary apparatus required for conversion:

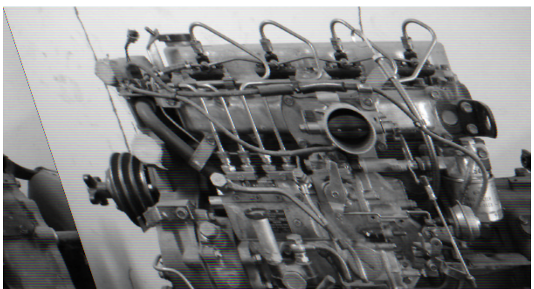
Table 3.2: Apparatus Required for Conversion

Serial	Apparatus	Number/ Quantity
01	ECU with harness	01 set
02	Spark plug with wire	01 set
03	CNG Regulator	01
04	CNG Switch	01
05	Heat meter, oil meter	01
06	Engine mounting	03
07	Battery (12 Volt)	02
08	Ignition coil	04
09	Radiator with cap	01
10	Radiator fan	01
11	Alternator	01
12	Self starter motor	01
13	O_2 Sensor	01
14	MAP sensor	01
15	Timing sensor	01
16	Coolant sensor	01
17	FB2W Engine complete	01
18	Fly wheel	01
19	Silencer box with pipe	01
20	Ignition key	01

3.2.3 Sequential steps in CNG conversion of Diesel Engine

The following steps are followed during conversion of diesel engine to CNG.

i) **Dismantle the Diesel Engine**



First of all the diesel engine was dismantled for necessary modification as shown in Figure 3.1 to 3.7.

Figure 3.1: Dismantled Engine Hino WO4D

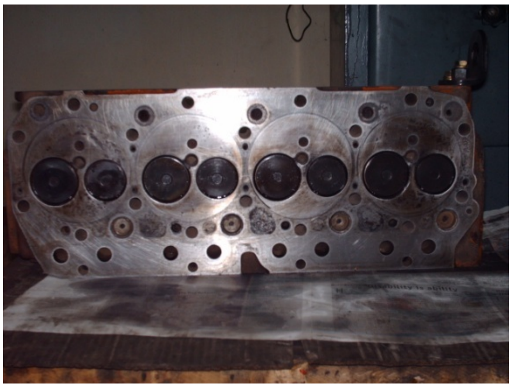


Figure 3.2: Engine head dismantling

Figure 3.3: Lub oil Pump Assembly

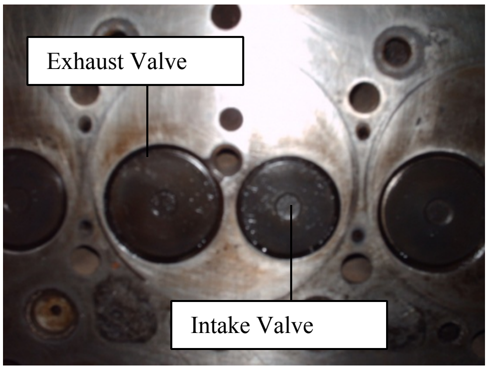


Figure 3.4: Intake and Exhaust valve

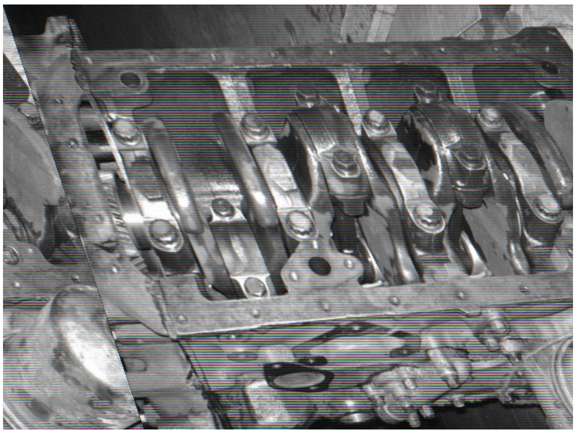


Figure 3.5: Engine Crank shaft assembly

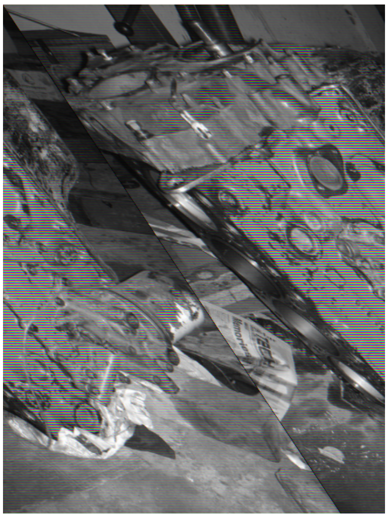


Figure 3.6: Engine body dismantling   Figure 3.7: Lub oil sump dismantling

ii) Removal of Fuel Injection System

Diesel fuel injection system with fuel injector , high pressure piping and high pressure pump has been removed from its position The pump drive is separated from the pump.

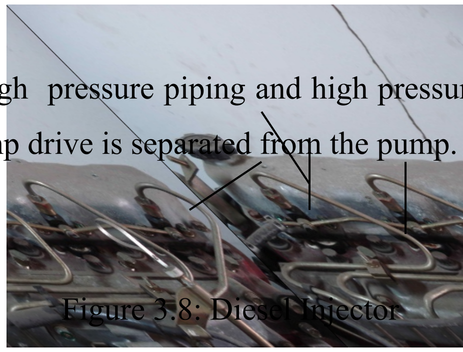
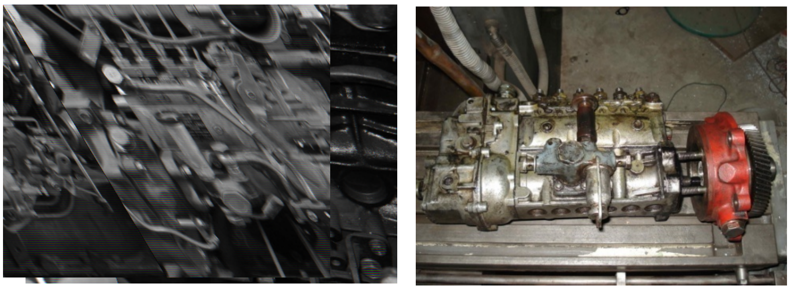


Figure 3.8: Diesel Injector



(a) (b) (c)

Figure 3.9 (a),(b),(c): Fuel pump and fuel line removal steps

iii) Modification in Piston head (Changing to lower Compression Ratio)

The engine has high compression ratio 19.2 is decreased to SI engine range 9-12. Initially the piston had a recess at the top face which constitutes a part of the combustion chamber. To change the compression ratio the piston was modified by removing material from piston head.

Data for piston head modification is given below:

Table: 3.3 Parameters before and after CNG Conversion

Data	**Before Modification**	**After Modification**
Engine Capacity	4009 cc (Cubic Centimeter)	4009 cc
Engine Displacement per cylinder	1002.3 cc	1002.3 cc
No. of Cylinder	4	4
Compression Ratio	19.2	11
Volume of combustion chamber at TDC	42.33 cc	87.3 cc
Clearance Volume	55 cc	100.04 cc
Piston Diameter	104 mm	104 mm
Piston head Cutting volume for modification	-	38.6 cc
Piston Head cut diameter	54.7 mm	74.46 mm
Piston Head cut depth	18 mm	18 mm
Piston height	144 mm	144 mm

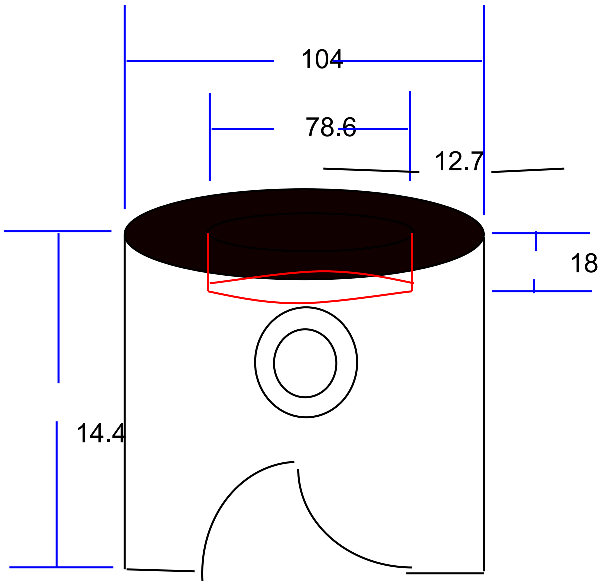
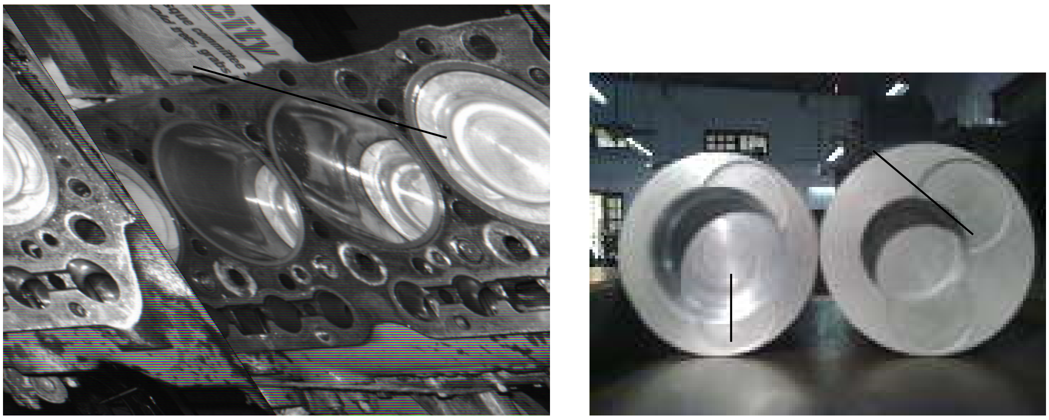


Figure 3.10: Dimension for Piston Modification(in mm)

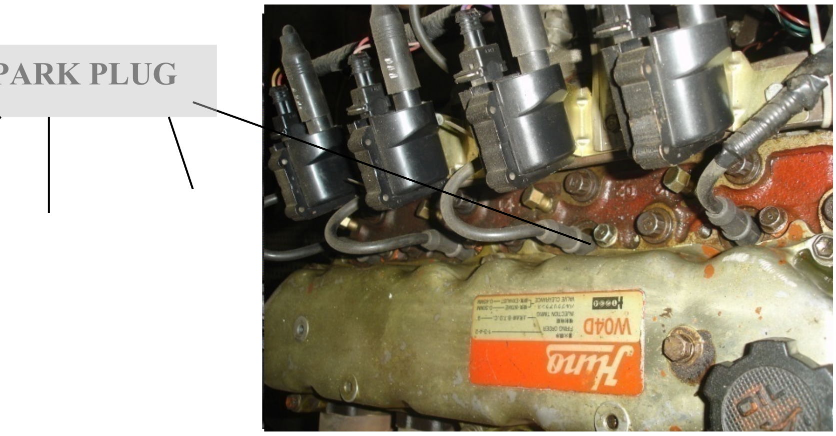


(a) (b)

Figure 3.11: (a) Piston after modification, (b) Piston before modification

iv) Engine Head Modification and spark plug installation

Engine head was modified and spark plug was installed in place of injectors.



(a) (b)

Figure 3.12: (a) Engine head modified , (b) Spark plug installed

v) Reassemble of the parts after being modified

The engine components were reassembled for modification (engine head, oil sump, modified piston set, radiator etc.)

vi) Installation of Carburation and Ignition Management System (CIMS)

The Omnitek Carburetion and Ignition Management System (CIMS) was specifically designed for natural gas (CNG) or propane (LPG) applications and features an integrated ignition and carburetion controller to manage the gas metering and ignition timing for lowest fuel consumption and emissions.

All components have been optimized to work together and provide a reliable, robust system that is easily integrated into retrofit and OEM applications. All in compact and waterproof enclosure to withstand even the harshest tropical environment.

The CIMS features enhanced closed-loop fuel control for lowest emissions and fuel consumption, a high-speed stepper motor driver for fast and accurate mixture control and a powerful ignition system for easy starting and a no-backfire guaranty, as well as a dashboard-switch selectable 2nd ignition map for low grade fuel applications. Latest inductive ignition technology provides long-duration multi-spark capability for multi coil and coil-per-plug applications. This is designed to control normally aspirated or turbo-charged engines. Perfect for use on diesel-to-CNG engine conversions. Different Components or kits which are fixed with converted engines are as follows:

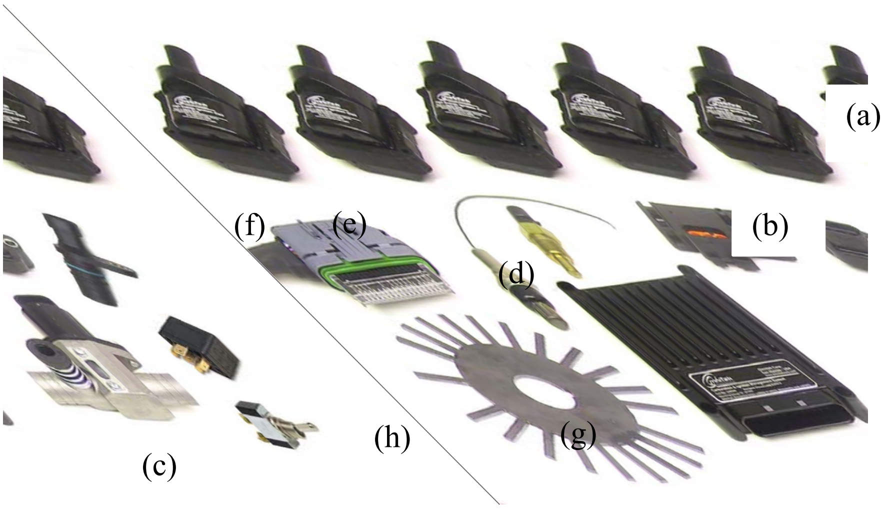


Figure 3.13: Different Components of CIMS (a) Ignition coil, (b) Engine speed sensor, (c) Timing disk, (d) MAP sensor, (e) ECTS, (f) Oxygen sensor, (g) Gas valve, (h) ECM

a) **Ignition Coils**

The CIMS uses the inductive ignition approach to store electrical energy and increase voltage. The system has been designed for use only with the coils supplied with the system. One of each coil is dedicated for a particular spark plug and connected to that.

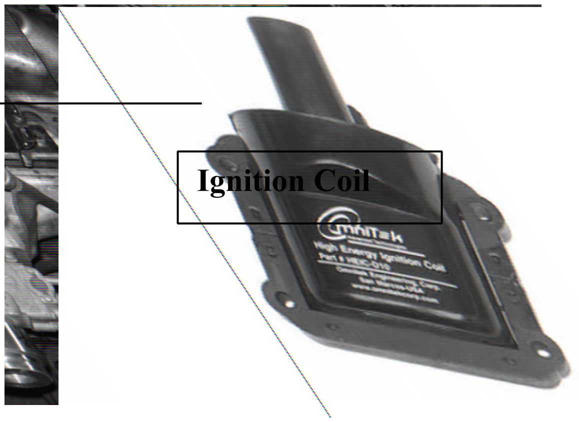


Figure 3.14: Ignition Coil

Specifications :

Primary resistance: .06 Ω , spark energy: 60 MJ , secondary voltage: 45 kV max , pre-spark suppression diode (for higher vehicle voltage levels) of high side: ratched-type post , temperature range: -40°C ~ +121°C .

b) **Engine Speed Sensor**

The inductive magnetic pickup sensor is used to
angular position of the engine. This is a rugged
of metal, operates without contact and provides

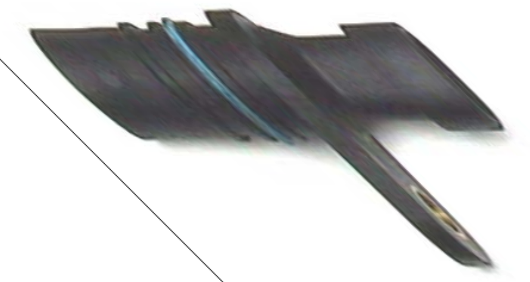


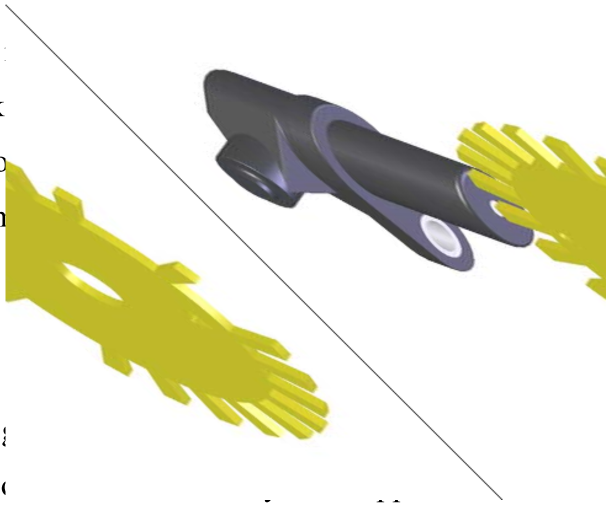
Figure 3.15: Variable Reluctance Engine Speed Sensor

Specifications

Type : Variable Reluctance, nominal working gap: 1.0 mm (0.5~1.5 mm), resistance:0.91kΩ, electrical connection: sealed waterproof connector, temperature range: -40°C~+121°C, robust and durable 25% glass filled polymer housing. The cable is aluminum enforced polyester foil for superior signal shielding.

c) Timing Disk

The timing disk is a multi-tooth design
accuracy and quicker starts. Timing disk
It is ensured that ensure that the directio
timing disk rotation (the arrow on the tin



d) Wiring harness

The supplied wiring harness uses 20
wiring is colour coded for easy installati

e) Manifold Absolute Pressure Sensor (MAP Sensor)

The MAP sensor is used to measure the absolute pressure in the engine's intake manifold. This is an excellent indicator of engine load (since it changes with throttle position) and can be used to obtain optimum ignition timing throughout all operational ranges. Use of a MAP sensor allows benefits in fuel consumption, exhaust emissions, response, power, torque and durability.

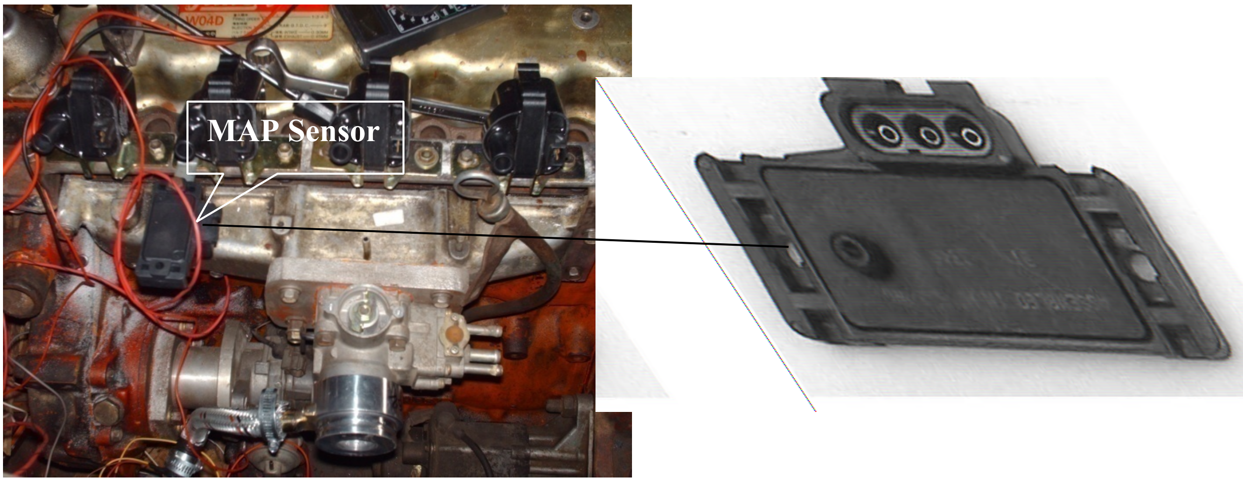


Figure 3.17: MAP Sensor

Specifications

Intake manifold pressure: 20 to 115 kPa absolute , analog output signal , operates from a regulated 5 V supply voltage , mountable directly to intake manifold or with a vacuum hose, electrical connection: 3 pin sealed waterproof connector, temperature range: -40°C ~ +121°C.

f) Engine Coolant Temperature Sensor (ECTS)

The engine coolant temperature sensor lets the CIMS module know whether the engine is cold or warmed up. This is important since combustion characteristics are closely related to engine temperature. Use of an ECTS can provide benefits in cold starting, cold engine power and torque, resistance to spark knock in an overheat situation and idle stability.

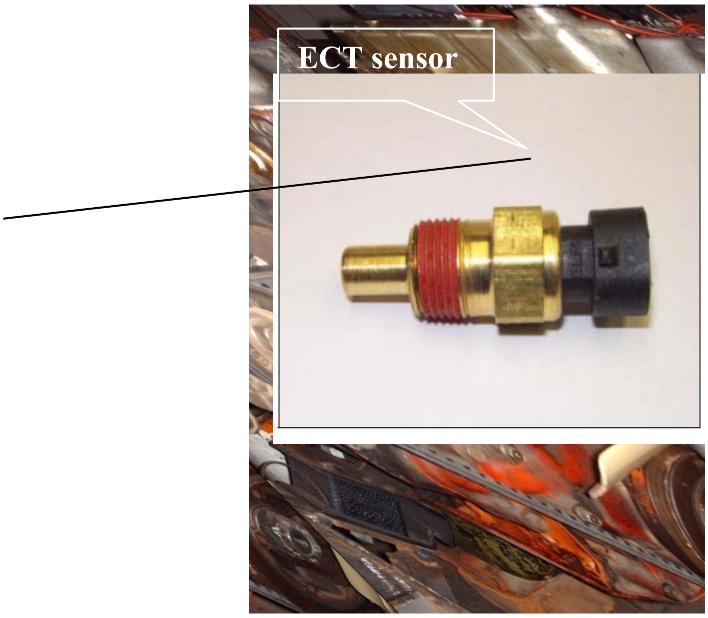


Figure 3.18: ECT Sensor and its placement on the engine

Specifications:

Negative Temperature Coefficient (NTC) thermistor design, brass body, 3/8" NPT thread supplied with sealing compound, operates from a regulated 5 V supply voltage, low resistance across the unit provides a high signal voltage (high coolant temperature), electrical connection: 2 pin sealed waterproof connector, high resistance across the unit provides a low signal voltage (low coolant temperature).

g) Fuel Shut-Off Solenoid

The CIMS controller is capable to limit the engine RPM and the vehicle speed (if a vehicle speed sensor is used) by closing the natural gas solenoid. The appropriate settings can be changed using the Omni Watch software.

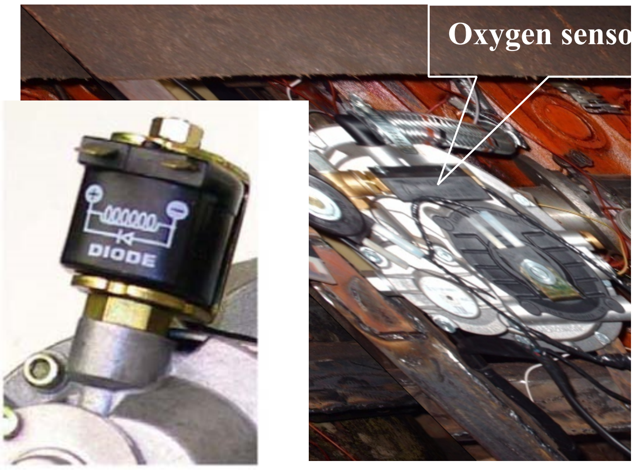


Figure 3.19: Fuel Shut-off Solenoid

h) Oxygen Sensor

Oxygen sensor is located between engine exhaust manifold and catalytic converter. The oxygen sensor normally known as the lambda sensor contains zirconium dioxide solid electrolyte, which compares the oxygen content in the exhaust and the normal atmosphere and the potential difference created due to variation of oxygen levels will provide a milli volt output. This output feedback given to the microprocessor (ECU) for the adjustment of the fuel delivery to the engine. The oxygen sensor output will be varying be operates in the stoichiometric condition input variation. Figure 3.20: O

i) Stepper Motor (Gas valve)

The voltage from the oxygen sensor is fed to the ECU. V high voltage, indicating a rich mixture, it responds by duration slightly to weaken the mixture. As the mixture w voltage goes low, and so the ECU enriches the mixture.

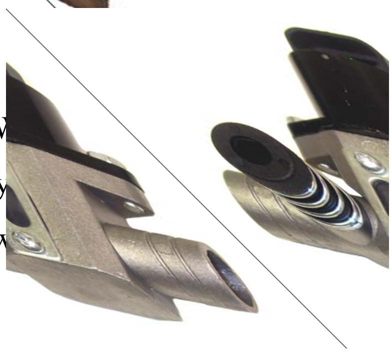


Figure 3.21: Gas valve

j) Throttle valve and Gas Mixing Chamber

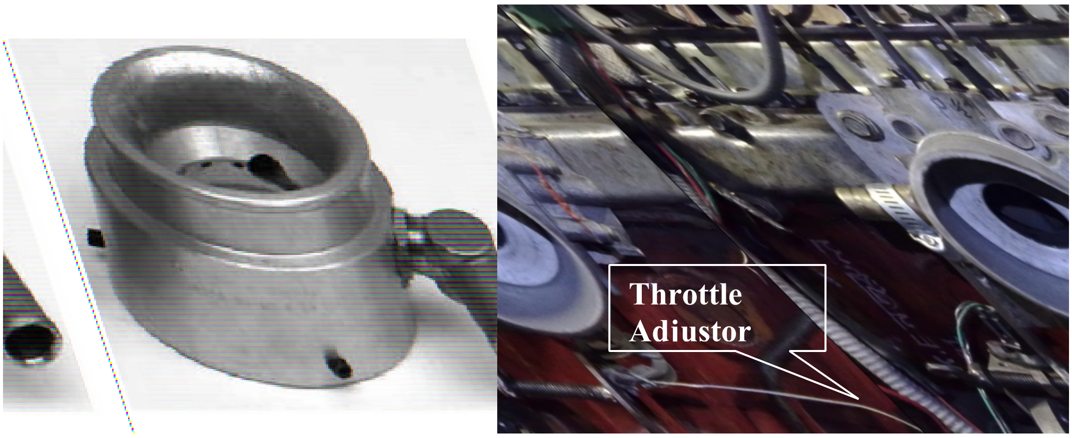
The mixing chamber is was used to mixing up air and NG in a chamber space.

Currently two types of mixers are available:

1) In cases where the air filter is just above the carburetor.

2) In cases where the air filter is separate from the carburetor, placing the mixer on the hose that joins them and not over the carburetor.

In this case later setup was followed for this experiment. The pressure regulator and the mixer was connected by means of a fire resistant hose, in which it was also installed the "maximum gas flow adjuster" (Power Screw). The gas air mixer was fitted as Figure 3.22.Throttle valve was operated by incorporating manual throttling by adjusting accelerating linkage as shown in Figure 3.22.



(a) (b)

Figure 3.22: (a) Mixing chamber and (b) Throttle valve adjusting mechanism

vii) Installing the Gas system

It includes CNG regulator, its accessories, line to CNG cylinder, gas pressure measuring gauge fixing up. The regulator used here is a control device to reduce the natural gas pressure allowing a regular flow of gas every time the engine requites it. It is equipped with diaphragms three natural gas pressure reduction stages that allow stability of flow. The regulator is heated with the liquid of the engine cooling circuit, prevents the natural gas freezing during the fall in pressure phase.

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.

The Components of the regulator as shown in Figure 3.23 are: A: Gas outlet connector, B: Idle speed setting screw, C: Plus contact for idle speed solenoid valve, D: High pressure solenoid valve, E: Heating connector, F: Gas inlet connector and G: Idling solenoid valve.

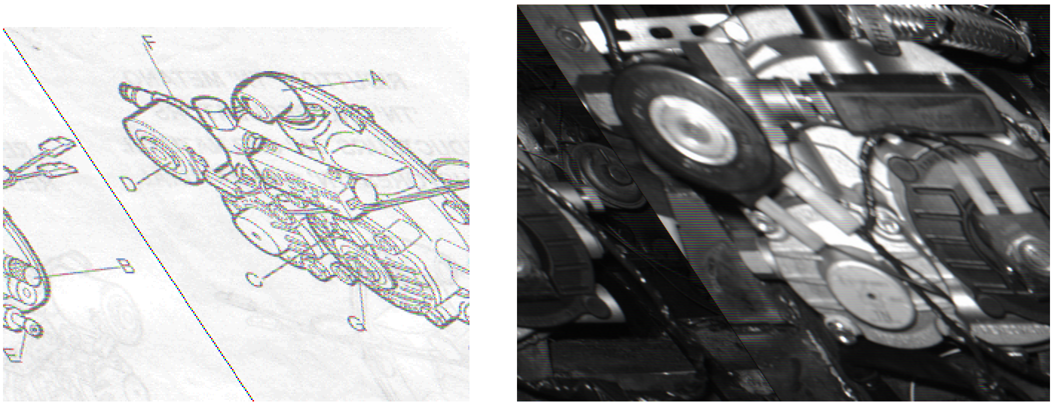


Figure 3.23: CNG Regulator

Specifications:

Regulator type: 3 stages with electronic starting device and vacuum controlled idling, heating: from engine cooling circuit liquid, inlet pressure: 220 bar, 1st stage adjustment pressure: 4 bar and 2nd stage adjustment pressure: 1.5 bar, power supply: 12V DC

CNG cylinder used to supply the NG as shown in Figure 3 standard Type-1 CNG cylinder capable of withstand 200 period without failure.

Figure 3.24: CNG Cylinder

viii) **Tuning of the engine (fuel and ignition)**

Test was being carried out whether the fuel system and spark ignition system was workable or not. This time the full conversion components or setup as Figure-3.25 were tested for reliable fitting. OmniWatch software made on running diagnostic checkup and log data possible as in Figure 3.25 to 3.30.

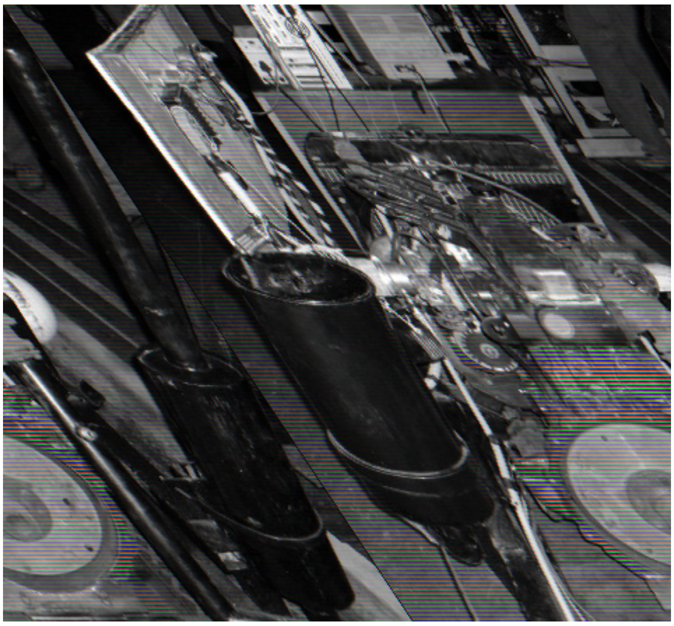


Figure 3.25: CNG Converted Diesel engine setup

Omnitek has designed a user-friendly PC software interface for use with the CIMS controller. OmniWatch allows the user to load calibration and firmware files, monitor system variables, troubleshoot problems, log data and view operational data.

From the ECM data list we can get data from sensors, ignition timing, feedback fuelling, processor performance and these can be obtained and summarized, analyzed and modified for the proper operating condition.

Using the Omniwatch software after connecting the ECM with a computer following data can be obtained:

- ECM Data List
  - Summary
  - Sensors
  - Ignition Timing
  - Feedback Fuelling
  - Processor Performance
  - Statistics

RPM	--------	Units
Intake Manifold Pressure	--------	Units
Engine Coolant Temperature	--------	Units
Engine Operating Mode	--------	Units
Sync Status	--------	Units
Malfunction Indicator Light Status	--------	Units
Oil Pressure Switch Status	--------	Units
Battery Voltage	--------	Units
Throttle Position Sensor	--------	Units
O2 Sensor Voltage	--------	Units
Stepper Motor Actual Position	--------	Units
Vehicle Speed	--------	Units
Final Ignition Timing	--------	Units
Commanded Dwell	--------	Units

Figure 3.26: Omni Watch summary data screen

RPM	---------	Units
Timing Disk Delta Tooth Time	---------	Units
Timing Disk Delta Missing Tooth Time	---------	Units
Vehicle Speed	---------	Units
Vehicle Speed Delta Pulse Time	---------	Units
Battery Voltage	---------	Units
Intake Manifold Pressure	---------	Units
Intake Manifold Pressure Sensor Raw	---------	Units
Throttle Position Sensor	---------	Units
Throttle Position Sensor Raw	---------	Units
Engine Coolant Temperature	---------	Units
Engine Coolant Temp Sensor Raw	---------	Units
O2 Sensor Voltage	---------	Units
Oil Pressure Switch Status	---------	Units

Figure 3.27: Omni Watch sensor screen

RPM	---------	Units
Intake Manifold Pressure	---------	Units
Engine Coolant Temperature	---------	Units
Transient Manifold Pressure Rate	---------	Units
Basic Ignition Timing	---------	Units
ECT Timing Advance	---------	Units
Transient Ignition Timing Adder	---------	Units
Final Ignition Timing	---------	Units
Multispark Count	---------	Units
Commanded Dwell	---------	Units
Diag Dwell Reduction Fire Order 1	---------	Units
Diag Dwell Reduction Fire Order 2	---------	Units
Diag Dwell Reduction Fire Order 3	---------	Units
Diag Dwell Reduction Fire Order 4	---------	Units
Diag Dwell Reduction Fire Order 5	---------	Units
Diag Dwell Reduction Fire Order 6	---------	Units
Diag Dwell Reduction Fire Order 7	---------	Units
Diag Dwell Reduction Fire Order 8	---------	Units

Figure 3.28: Omni Watch Ignition timing screen

RPM	--------	Units
Intake Manifold Pressure	--------	Units
Engine Coolant Temperature	--------	Units
Transient Manifold Pressure Rate	--------	Units
O2 Sensor Voltage	--------	Units
Target O2 Volts	--------	Units
O2 Error	--------	Units
O2 Sensor Crosses	--------	Units
O2 Crosses per Second	--------	Units
Open / Closed Loop Fuelling	--------	Units
O2 Warm / Cold	--------	Units
O2 Normal / Stuck	--------	Units
O2 Rich / Lean	--------	Units
Stepper Motor Base Position	--------	Units
Transient Enrichment Steps	--------	Units
Short Term Trim	--------	Units
Long Term Trim	--------	Units
Final Fuel Trim	--------	Units
Stepper Motor Desired Position	--------	Units
Stepper Motor Actual Position	--------	Units

Figure 3.29: Omni Watch feedback fuelling screen

RPM	--------	Units
ntake Manifold Pressure	--------	Units
Engine Coolant Temperature	--------	Units
Reset Arm Counter	--------	Units
Main Loop Time	--------	Units
128Hz Loop Time	--------	Units
Maximum Loop Time Recorded	--------	Units

Figure 3.30: Omni Watch processor performance screen

Maximum Vehicle Speed Recorded	--------	Units
Total Engine Run Time	--------	Units
Total Engine Idle Time	--------	Units
Total Engine Cranking Time	--------	Units
Total MIL On Time	--------	Units
Total High Voltage Time	--------	Units
Total Engine Cold Time	--------	Units
Total Engine Overheat Time	--------	Units
Total Engine Severe Overheat Time	--------	Units
Total Vehicle Moving Time	--------	Units
Total Distance Driven	--------	Units
Total Open Loop Time	--------	Units
Total Long Term Trim Disabled Time	--------	Units
Total Mixture Too Rich Time	--------	Units
Total Mixture Too Lean Time	--------	Units
Total Fuel Cutoff Time	--------	Units
Total Key Cycles	--------	Units

Figure 3.31: Omni Watch statistics screen

Technical assistance was supported from NAVANA CNG Ltd for conversion of the diesel engine and setup of Omnitek equipment.

3.2 Performance parameter Setup

To set the performance parameter the converted engine to be turned into an experimental setup shown in Figure 3.32. Experimental setup buildup was followed the following steps:

Figure 3.32: Experimental Setup (CNG converted diesel engine)

a) Mounting the engine on test bed frame

To get the performance test the converted engine was mounted first on a fabricated tough mild steel U channel frame as Figure 3.33 to turn it into an experimental setup. Consideration was made in design and fabrication so that the centre of the shaft of hydraulic dynamometer would be in line with the centre of the fly wheel as Figure 3.34. The frame was designed by keeping enough space for opening the engine head and oil sump to change the new piston set without taking the engine out of the frame.

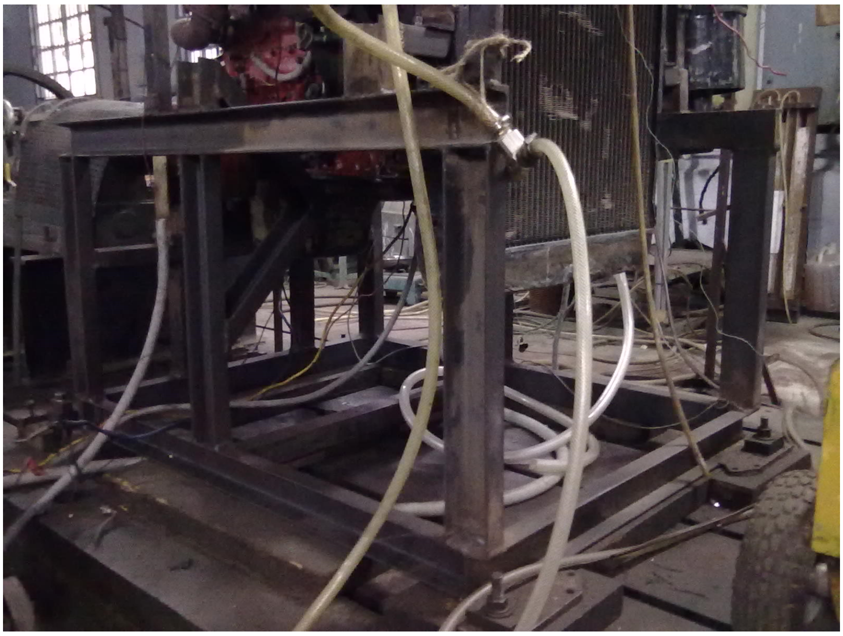


Figure 3.33: Fabricated U channel frame

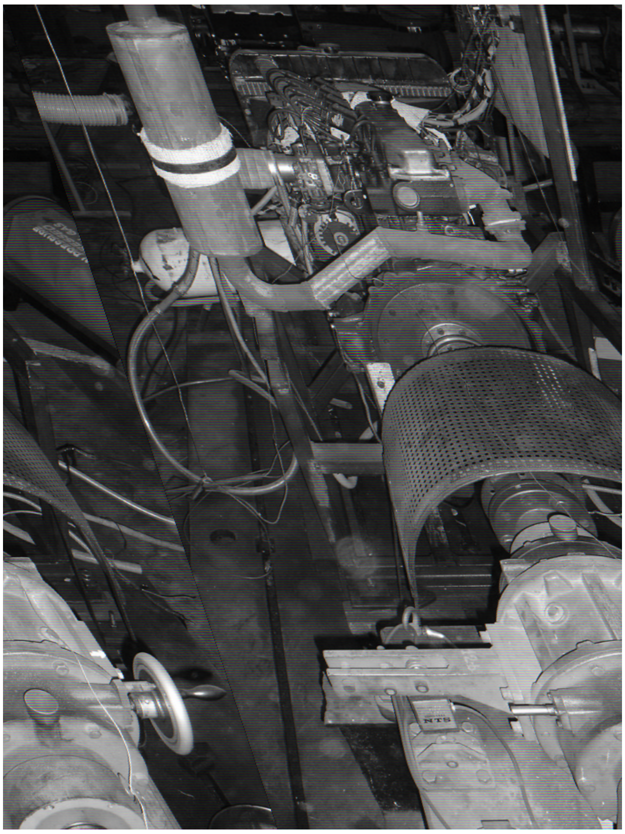


Figure 3.34: Experimental Setup with dynamometer coupling.

b) Applied load / torque and power measurement

A coupling flange as Figure 3.37 was made and fixed with the flywheel at one end and dynamometer coupling with other end tightly bolted face to face. Now the hydraulic or water brake dynamometer, model no TFJ -250L as figure 3.35 was used to apply the load on it. It consisted of load cell transducer to measure the reaction force acting on the dynamometer. A magnetic type tachometer was used to measure the engine speed. Two

LED displays were being used to show the load in kg and engine speed in rpm instantly as Figure 3.36.

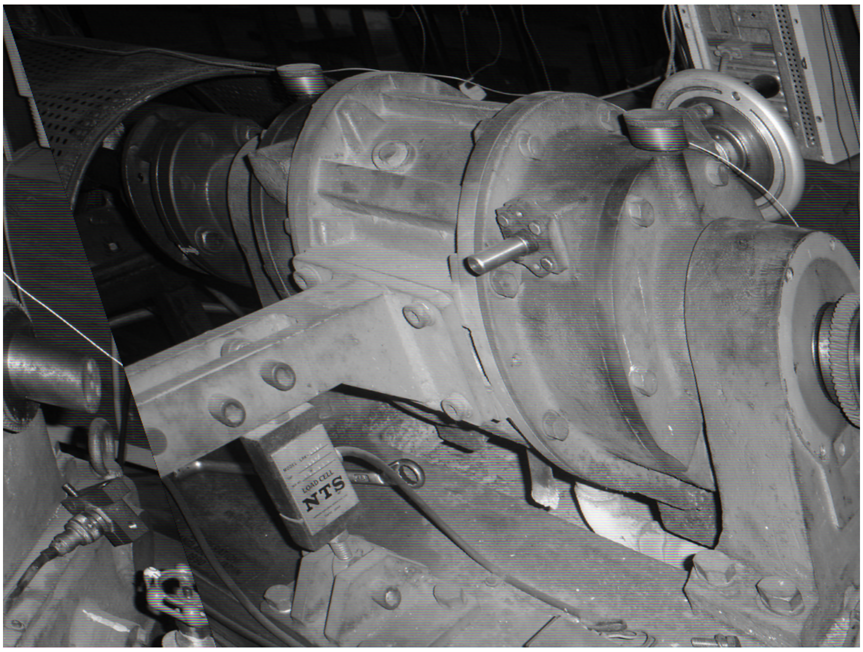


Figure 3.35: Hydraulic Dynamometer

Figure 3.36: Digital meter for load and rpm

Figure3.37:Coupling flange

Other parameter like, Brake horse power obtained as:

Power (P) = (W_L x N)/ Dynamometer constant (C) (3.1)

Torque was measured as:

Torque (T) = P/2 π N (N/m) (3.2)

where C=2500, N = particular rpm

c) **Temperature measurement**

All temperatures were measured using K-type thermocouple setup as shown in Figure 3.38 where a selecting nob was used to select the temperature record for lub oil, coolant and exhaust gas. _

Fig 3.38 : Temperature measuring unit.

d) Air flow measurement

The air-flow meter provides a means of measuring the amount of air being drawn into the engine under various operating conditions. Air flow is measured by having the engine draw its air through a set of precision flow-nozzles (each 30 mm in diameter) into a pulse-damping drum, then through a flexible hose with the air filter housing as Figure 3.39,3.41 and 3.42.Since all air entering the carburetor is drawn through the flow-nozzles which restrict the air flow. The air-flow rate can be determined by measuring the pressure across the flow-nozzle. The pressure difference across the nozzle is measured in millimeter of water by a U tube manometer as Figure 3.40.

From the pressure difference we get : $V_1 = [h \times 2g \times (\rho_{water} - \rho_{air}) / \rho_{air}]^{1/2}$ (3.3)

Mass flow rate of air can be calculated as :

$$m(air) = C_d \times (\pi/4 \times d^2) \times V_1 \times 3600 m^3/hr \times \rho_{air} \quad (3.4)$$

Figure 3.39: Air filter Housing

Figure 3.40:U tube manometer

Figure 3.41:Air flow measurement setup

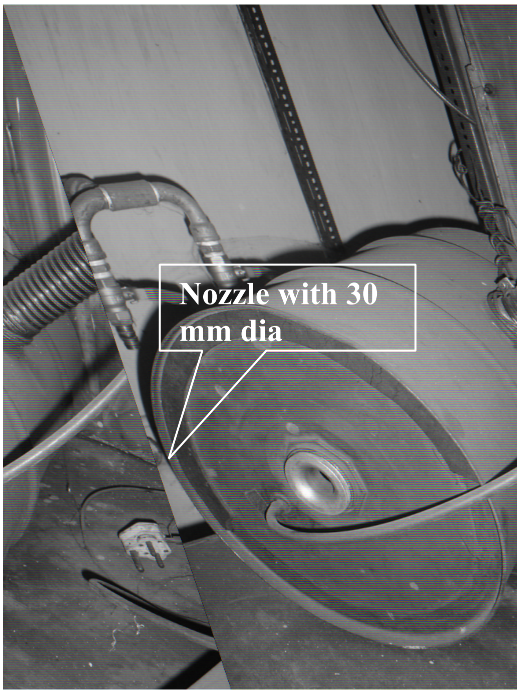


Figure 3.42: Pulse damping drum

e) Gas flow measurement

Flow of gas was measured by rotameter flow percentage (Figure 3.44), which already being calibrated in m^3/h and provided us a calibrated data graph with an equation y = 0.1895x + 0.5853 as Figure 3.46. Here 'x' being the rotameter % and y being the gas flow rate in m^3/h. Mass flow rate of CNG can be found out this way for each data set.

Figure 3.43: Gas storage

Figure 3.44: Rotameter

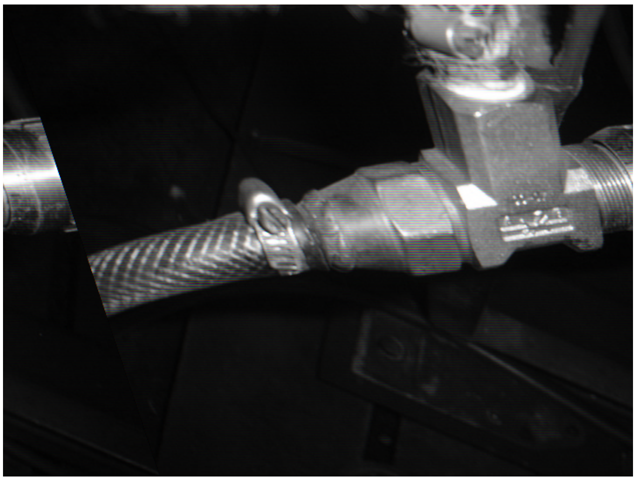


Figure 3.45: Manual Gas Flow control nob

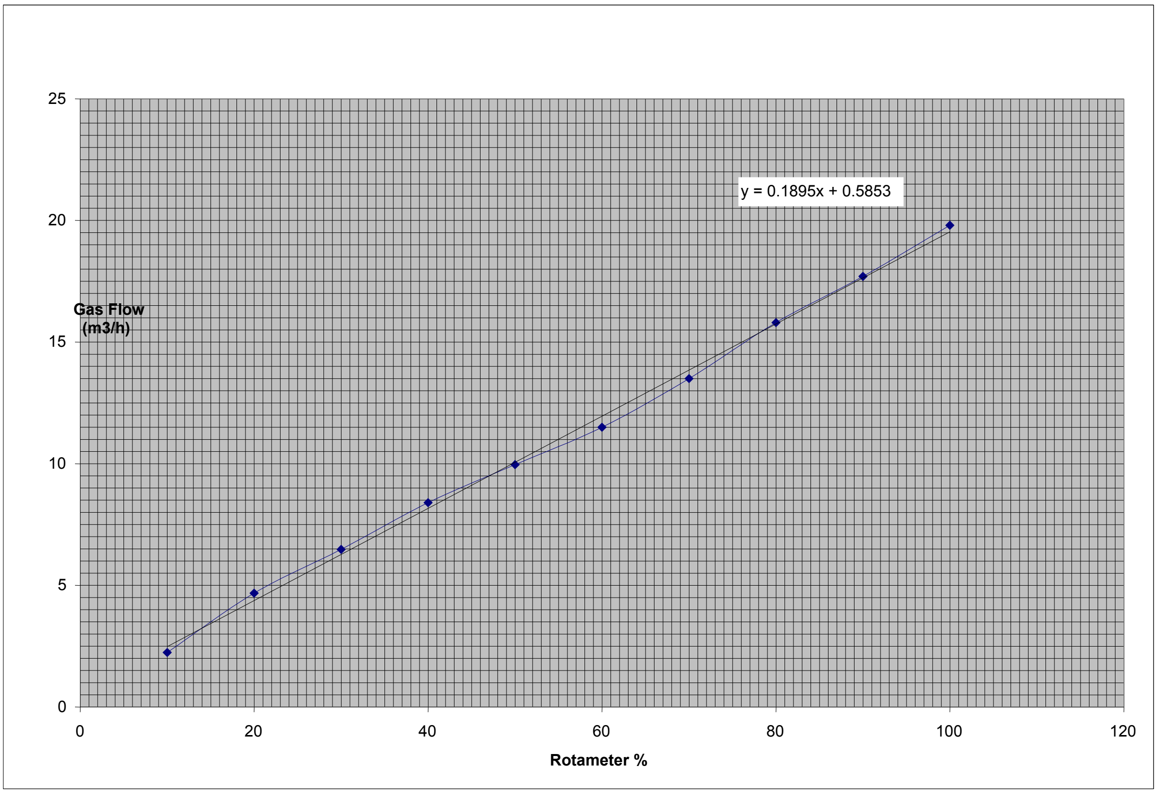


Figure 3.46: Rotameter calibration

f) **Compression ratio setup**

For new or adjusted compression ratio the first set of piston was replaced each time. Material from the piston head were removed to create a recessed bowl as per the need of new clearance volume as shown in Figure 3.47 to 3.49, which was possible by varying bowl volume of the piston head.

The piston size was suitable enough for any further machining. Three set of pistons were used for CR=11, CR=12 and CR=13 as figure 3.52, 3.54, 3.56. The cutting volume was being measured each time by cutting the piston in lathe and filling up groove with water, which reflect the original bowl volume in practical. Then the pistons are placed in the cylinder as Figure 3.50 to work with a new setup.

Figure 3.47: New piston (for Diesel) before modification to CNG operation

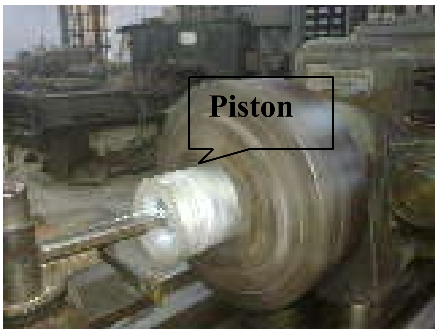


Figure 3.48: Piston head machining

Figure 3.49: Piston before and after modification to CNG operation

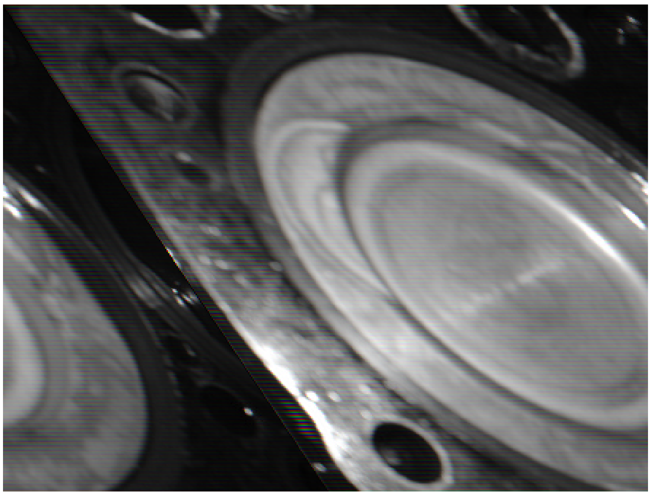


Figure 3.50: Modified piston placement in cylinder

Design of modified pistons theoretically will take the dimension as of Figure,3.50, 3.52, 3.52, 3.53, 3.54, 3.55, 3.56 ,3.57and 3.58. The theoretical calculation for cutting diameter of piston bowl for a particular CR were shown in details in APPENDIX D of this paper.

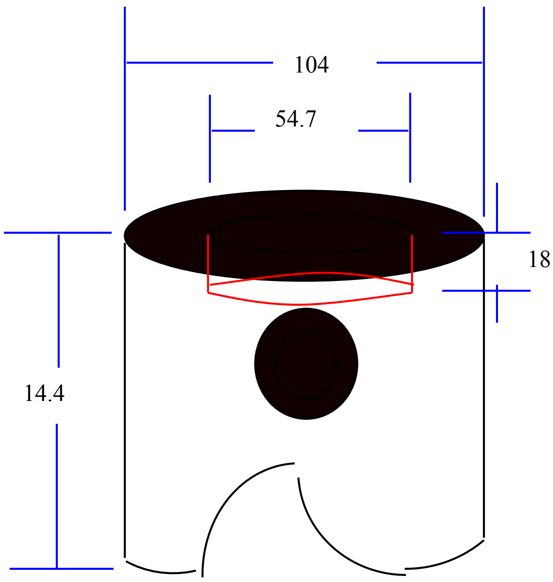


Figure 3.51: Dimension of Piston Before Modification (in mm)

(a)

(b)

Figure 3.52: Post machining piston of CR=11 (a) Piston top and (b) Side view

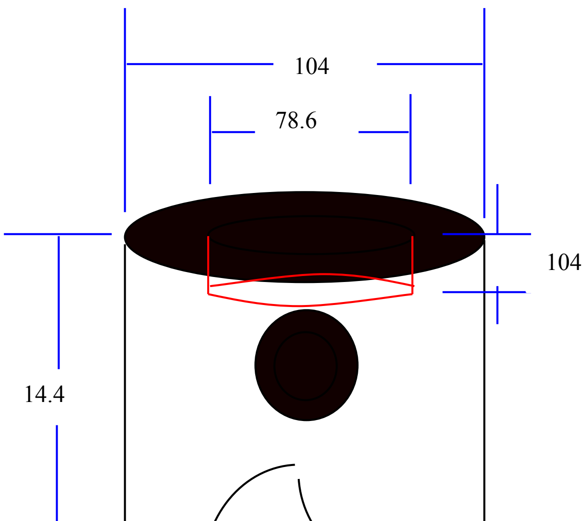


Figure 3.53: Dimension of CNG converted Piston (CR=11)

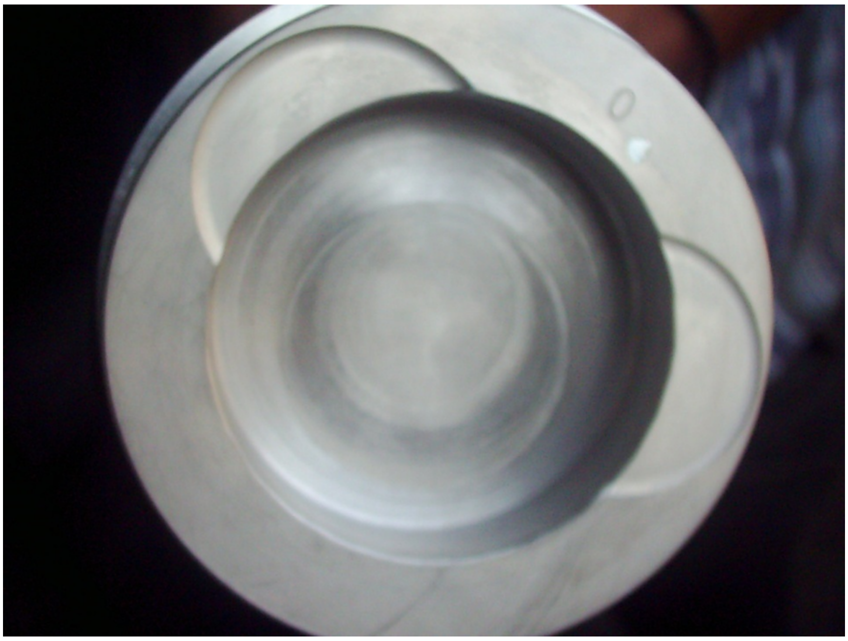


Figure 3.54: Post machining piston top view (CR=12)

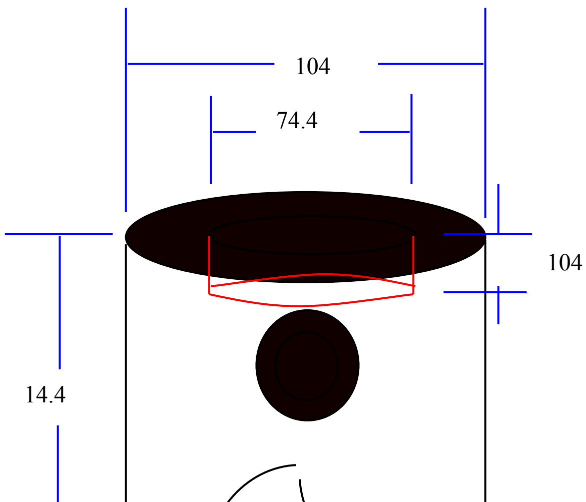
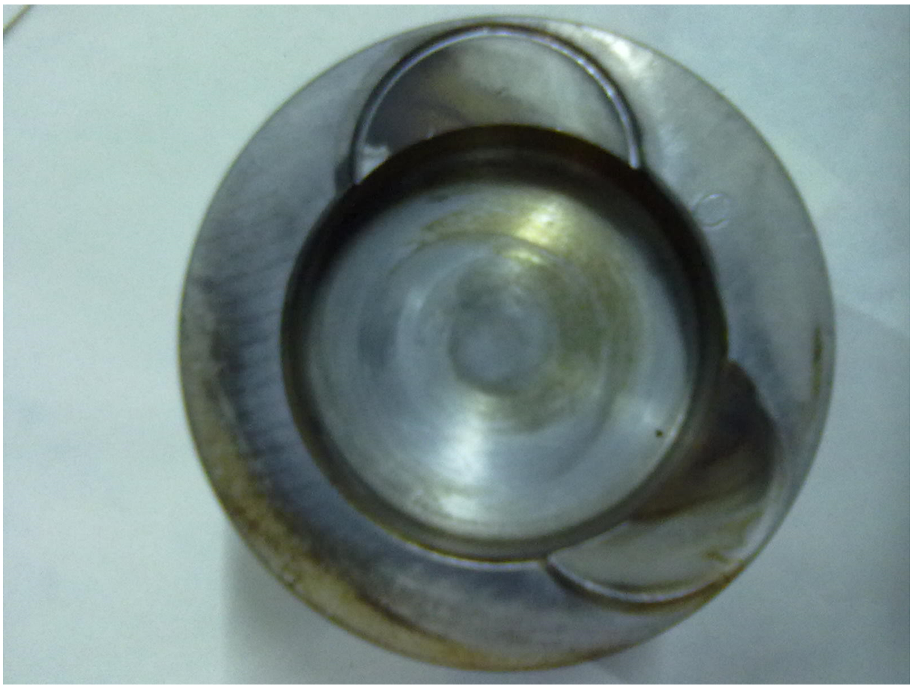
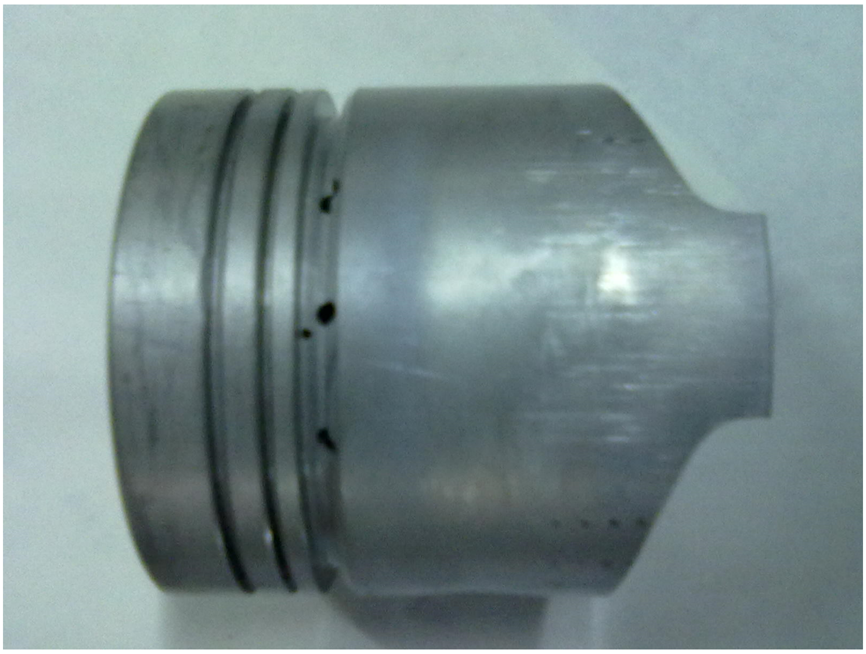


Figure 3.55: Dimension of CNG converted Piston (CR=12)



(a)



(b)

Figure 3.56: Post machining piston of CR=13 (a) Piston top and (b) Side view

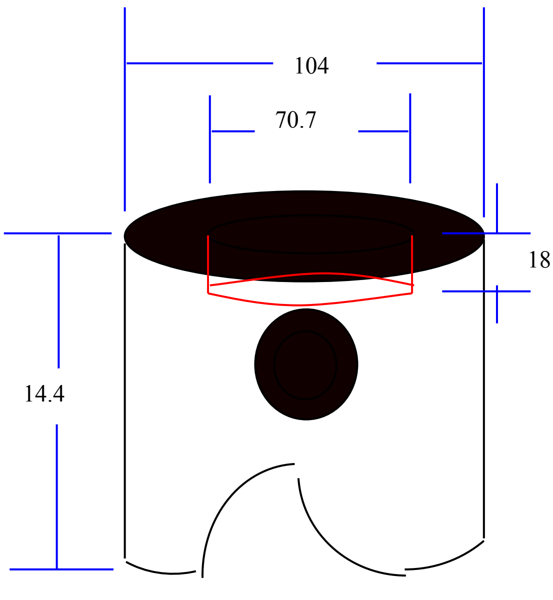


Figure 3.57: Dimension of CNG converted piston (CR=13)

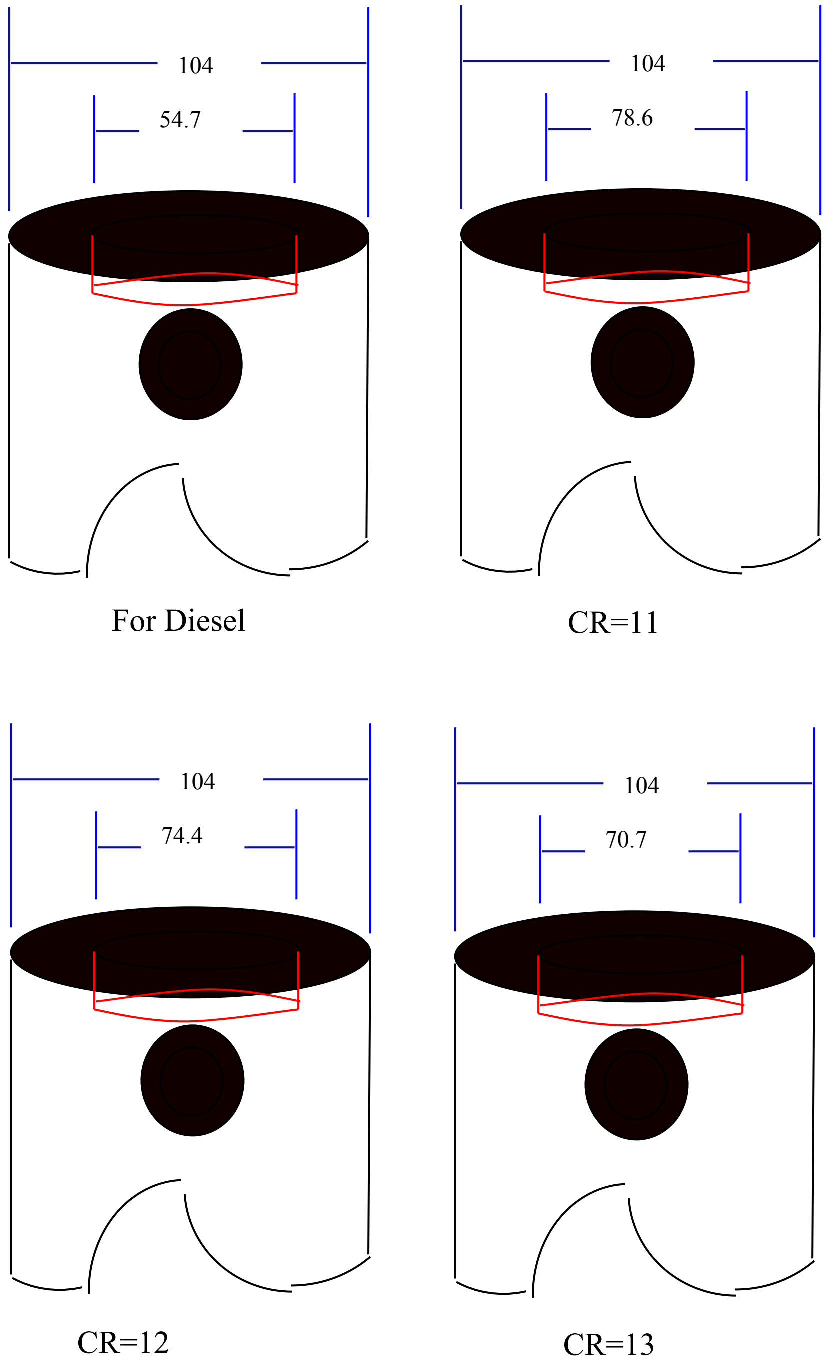


Figure 3.58: Dimension of cutting diameter of different setup pistons

Theoretical Stress Analysis For Piston

A theoretical stress analysis was carried out for the both piston types used for diesel engine and modified piston for CNG engine with different compression ratios. For analysis a Representative Volume Element (RVE) of piston was considered a simple beam with a reaction force of connecting rod at the centre to depict the full piston. Maximum peak pressure of diesel engine 1000 psi (68 bar) was taken to find out the maximum stress developed on the piston element in diesel operation. Which was further compared with CNG operated piston at different CRs with their obtained maximum stress from the peak pressure of the individual CR. The Basic Equations were used as :

For ($0 < X < L/2$)

Shear Force, $V = -WX$ (3.5)

Moment , $M = -WX.X/2 = -WX^2/2$ (3.6)

For ($0 < X < L$)

Shear Force, $V = -WX + WL$ (3.7)

Moment , $M = -WX^2/2 + WLX - WL^2/2$ (3.8)

Maximum Stress = MC/I (3.9)

Where , W = Uniform force in N/m

X = Distance zero to full diameter of piston

L = Full Piston diameter=104 mm

C = Distance from the centre to the top of the thinnest cross section

I = Moment of inertia = $1/12\ bh^3$

Effective Moment of Inertia , Ie was found out as :

$I_e = 1/12\ bh_e^3 = 1/12b\ [h+h'f'(d)]^3$ (3.10)

If $d \rightarrow 0$ then h' =18 and if d=d then h'=0

From Figure 3.59 for cutting diameter 0 to d the h_e (effective height) was found out from the interpolated value between 10 to 28 mm.

d= piston cutting dia

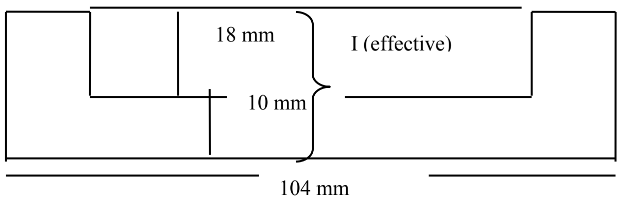


Figure 3.59: Stress analysis with effective moment of inertia

From the stress analysis of ANNEXURE B the Table 3.4 was formed as below:

Table 3.4: Stress analysis comparison

	Maximum Stress Developed on piston (MPa)	CNG piston Stress compared to maximum Diesel Design piston Stress	Relative Stress factor
Design for Diesel Engine	649.27	100 %	1
Design for CNG with CR=11	97.03	14.94 %	6.69
Design for CNG with CR=12	131.34	20.23%	4.94
Design for CNG with CR=13	163.69	25.21 %	3.97

g) Other parameter setup

Other performance parameters related to applied load, torque, power, air flow, gas flow rate are shown as sample calculation in APPENDIX C of this paper.

3.3 Data Acquisition

Data from the performance test of CNG engine at varying CRs were found out both from Omnitek software interface by PC data logging as Figure 3.58 and from direct display meter at ambient condition each time with different CRs. Data were then processed in a spread sheet to put into necessary calculation to get the other performance values which are not directly found out from the investigation or display meter. The performance data found out from the experiment is displayed in APPENDIX D of this paper.

Figure 3.60: PC Interfacing of data logging

CHAPTER-4
RESULTS AND DISCUSSION

4.1 General

The results of the experimental investigation regarding the measured performance parameters have been discussed for three compression ratios 11, 12 and 13 in this chapter. The diesel engine 110 hp (considering present rated power 80 hp) and maximum 3000 rpm after CNG conversion was allowed to run upto its highest rpm till the smooth running condition. It was practically found that over 2300 rpm unusual vibration of engine and knocking was observed .So, the highest rpm 2300 was considered, from where gradually load was increased and thus rpm change occurred. Below 1450 rpm for further increase of load engine was found to completely shutdown. The rpm range between 1450 to 2300 for all CR were found in smooth running condition. So, for performance comparison and data logging this range was considered. All the data were taken in WOT condition to get maximum power output in 1450 to 2300 rpm range. Data logging was done every after 10 minutes initial warmup and in same atmospheric condition. Thus performance data for three CRs were taken and compared in graphically. The results are discussed mainly basing on the related graph plotted with the obtained data. The graphs were discussed in this chapter in subsequent paragraphs. Details of data are shown in APPENDIX D..

4.1.1 Applied Load, Power and Brake Mean Effective Pressure (Bmep) for CR=11

In Figures 4.1, 4.2 and 4.3 maximum applied load 40 kg, maximum power of 29.55 hp (36.93% of rated power) and maximum Brake mean effective pressure (Bmep) 3.59 bar was observed in the region of 1700 to 1800 rpm. From the peak load, power and torque for further increased or decreased of rpm within the range of 1450 to 2300 rpm all data were found as a decreased value.

4.1.2 Air Flow rate , Volumetric Efficiency , Fuel flow rate , Air / Fuel Ratio and Engine Manifold pressure for CR=11

In Figure 4.4 it was found that air flow rate increased gradually with increase of rpm. The flow variation within 138.62 kg/h to 170.38 kg/h was observed within 1450 to 2300 rpm. It was observed that the volumetric efficiency of the CNG converted engine has been decreased almost linearly with the increase of engine speed within the range 1450 to 2300 rpm as in Figure 4.5. The maximum efficiency was found to be 66.68% at lower engine speed (1450 rpm) and minimum efficiency 51.80% was obtained at a higher experimental engine speed (2300 rpm).

In Figure 4.6 it was observed that the mass flow rate of NG found 19.6% lower than the equivalent diesel flow rate at same engine speed range. Both the flow rate represented very little variation from 1450 to 2300 rpm. Maximum and minimum flow rate for NG was found to be 10.93 kg/h and 10.42 kg/h respectively

A little gradual variation was observed in A/F ratio distribution within a rpm range. As in Figure 4.7 the variation was found in A/F ratio as an increasing trends between 13.30 to 15.81 within the operating speed range. All the values were less than the stoichiometric A/F ratio (17.2) of natural gas.

It was observed that for CR=11 the engine manifold pressure at WOT changed significantly with the increase of engine speed. The variation found within the range 73.9 kPa (at lower speed 1450 rpm) to 93.9 kPa (at higher speed 2300 rpm) as in Figure 4.8.

4.1.3 Brake Specific Fuel Consumption (Bsfc) and Thermal Efficiency for CR=11

It was found from Figure 4.9 that the minimum bsfc would be 0.49 kg/kW-h at 1810 rpm. Maximum bsfc value 0.88 kg/kW-h was attained at maximum experimental engine speed 2300 rpm. Bsfc varied from 0.49 to 0.88 kg/kW-h within 1450 to 2300 rpm range.

Thermal efficiency was increased linearly upto its peak value 14.54% at 1810 rpm then linerly decreased for further increase in engine speed as shown in Figure-4.10. Minimum efficiency 7.98% was obtained at a rpm 2300 (maximum experimental rpm).

So, maximum fuel utilization from the CNG fuel 14.54% can be possible if the engine run at nearly 1800 rpm.

4.1.4 Temperature Measurement for Lubricant Oil, Coolant and Exhaust Gas for CR=11

In Figure 4.11 very less variation of lubricant oil temperature was observed with the change of engine speed. Temperature changes was found within the range of 89^0 to 103^0C when engine speed was varied within 450 to 2300 rpm.

In Figure 4.12 very less variation of coolant (water) temperature was observed with the change of engine speed. At varying engine speed 1450 to 2300 rpm the temperature was changed within the range of 75^0 to 81^0C .

In Figure 4.13 it was observed that there found very little or almost no variation in exhaust gas temperature change upto engine running speed 1900 rpm. Further increases in engine speed upto 2300 rpm has a significant stepwise temperature rise from 433^0C to 564^0C .

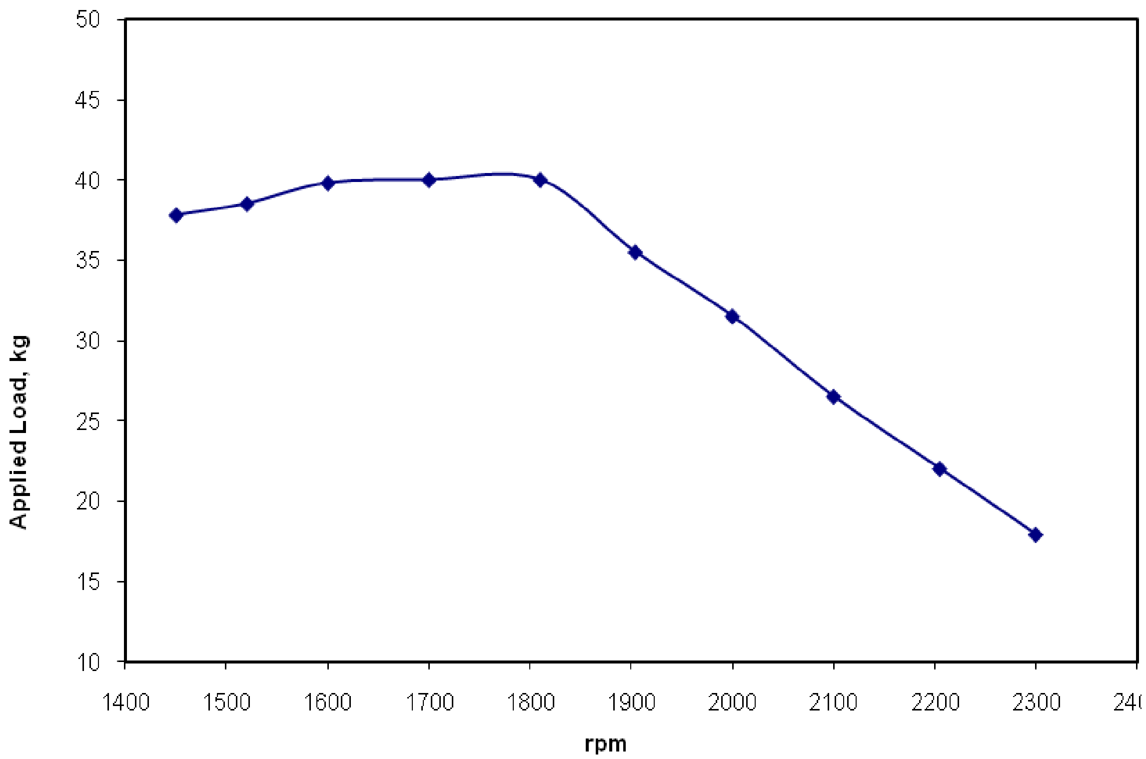


Figure 4.1: Applied load for CR=11

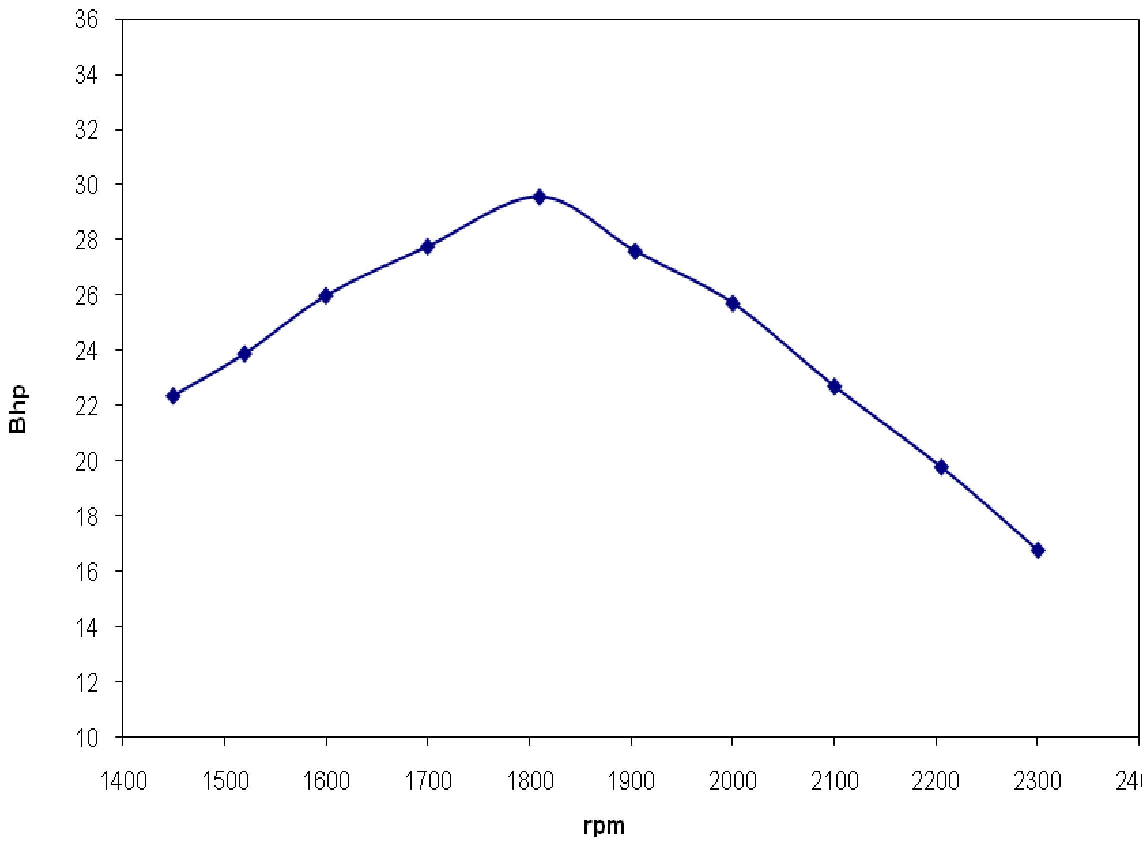


Figure 4.2: Brake Horse Power for CR=11

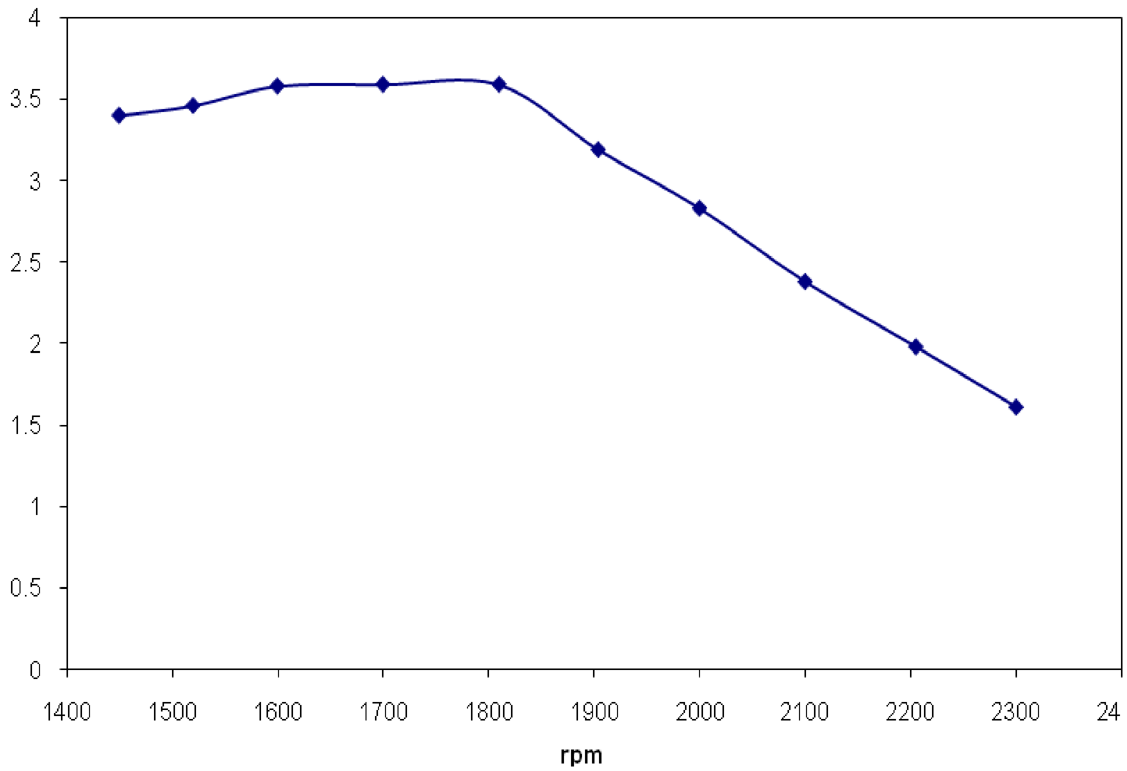


Figure 4.3: Brake Mean effective Power (Bmep) for CR=11

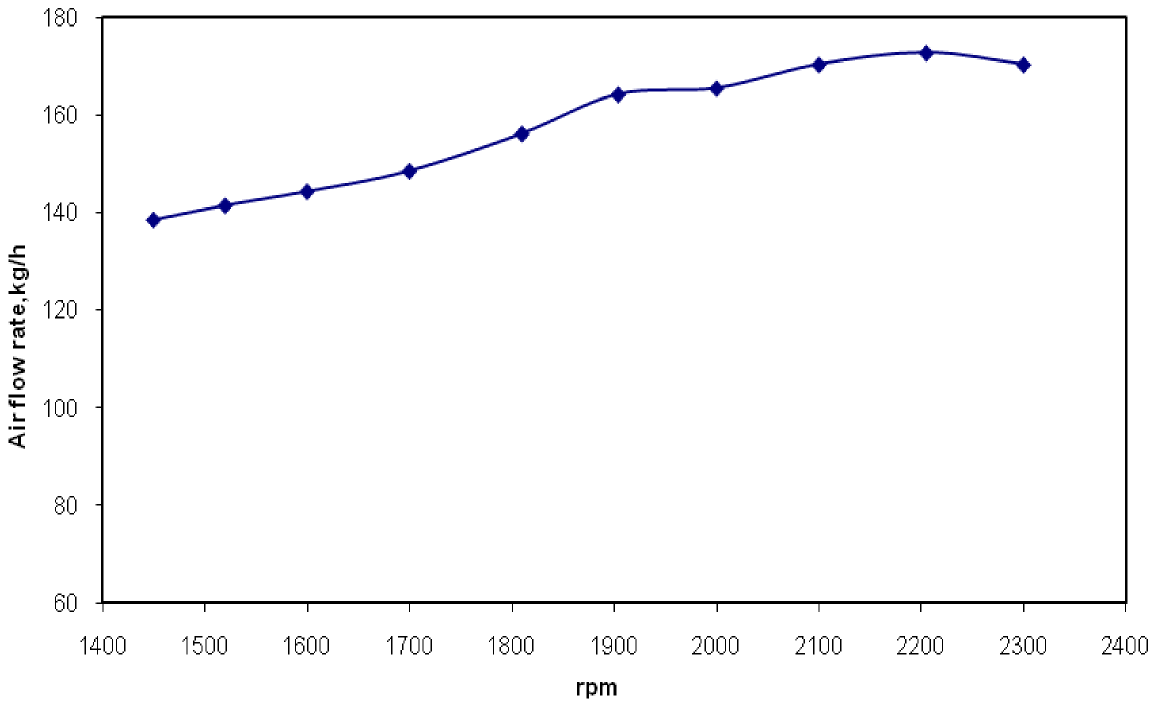


Figure 4.4: Air Flow Rate for CR=11

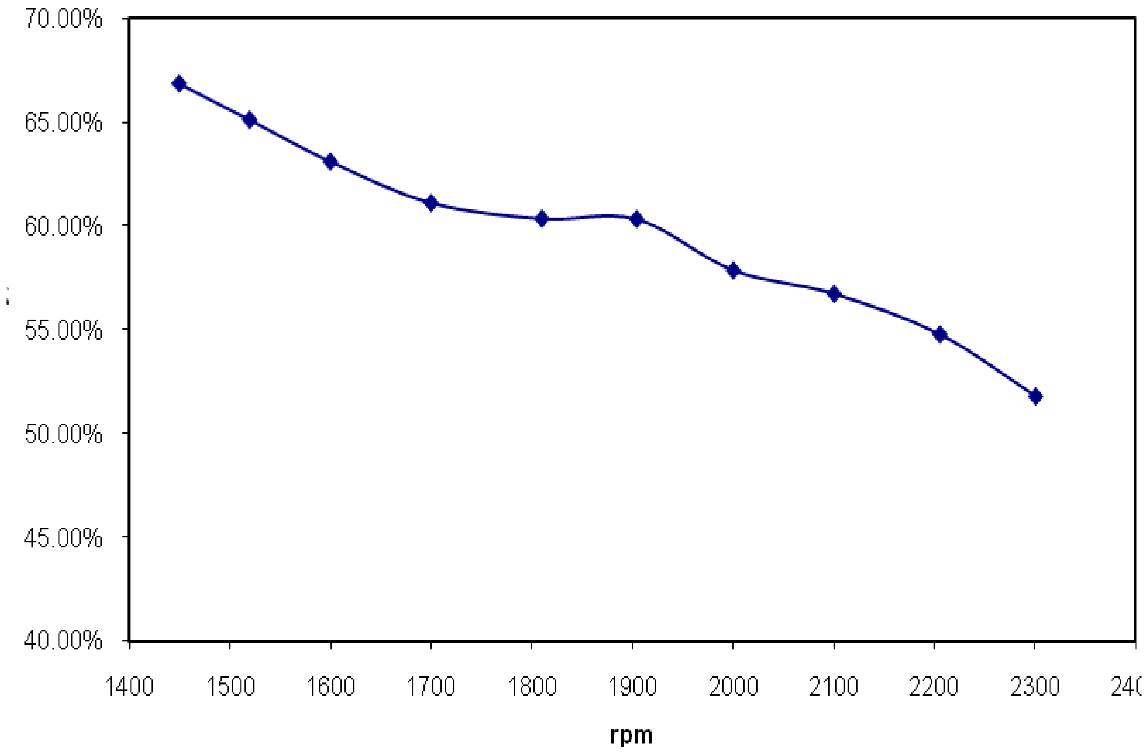


Figure 4.5: Volumetric Efficiency for CR=11

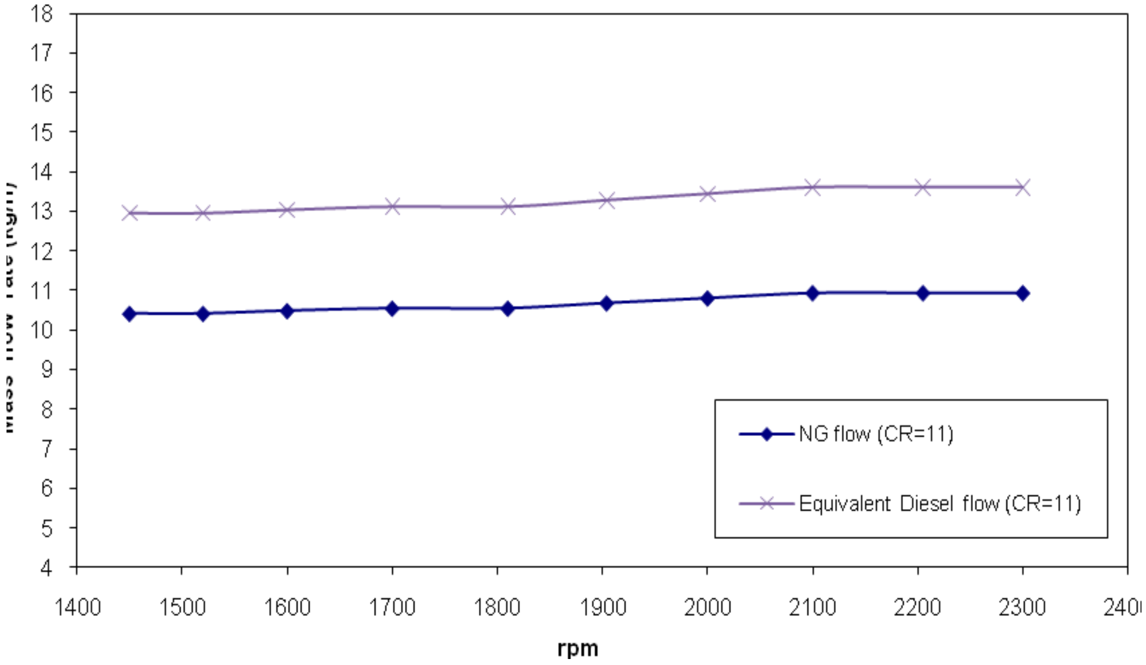


Figure 4.6: Mass flow rate NG and Equivalent Diesel flow for CR=11

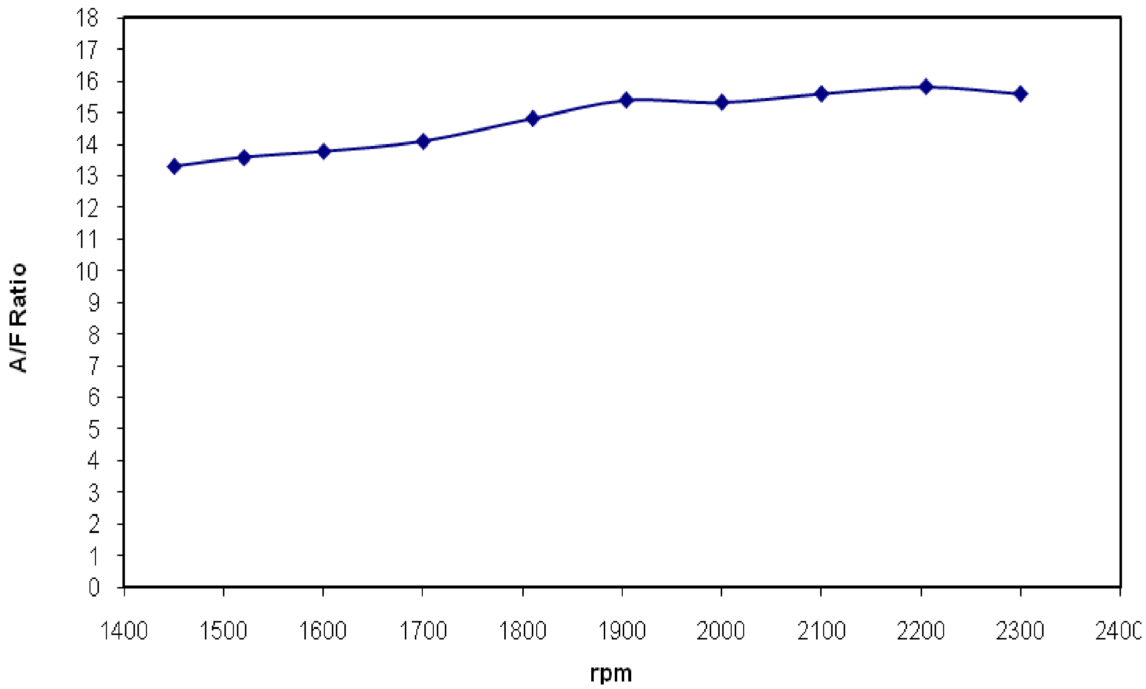


Figure 4.7: A/F ratio distribution for CR=11

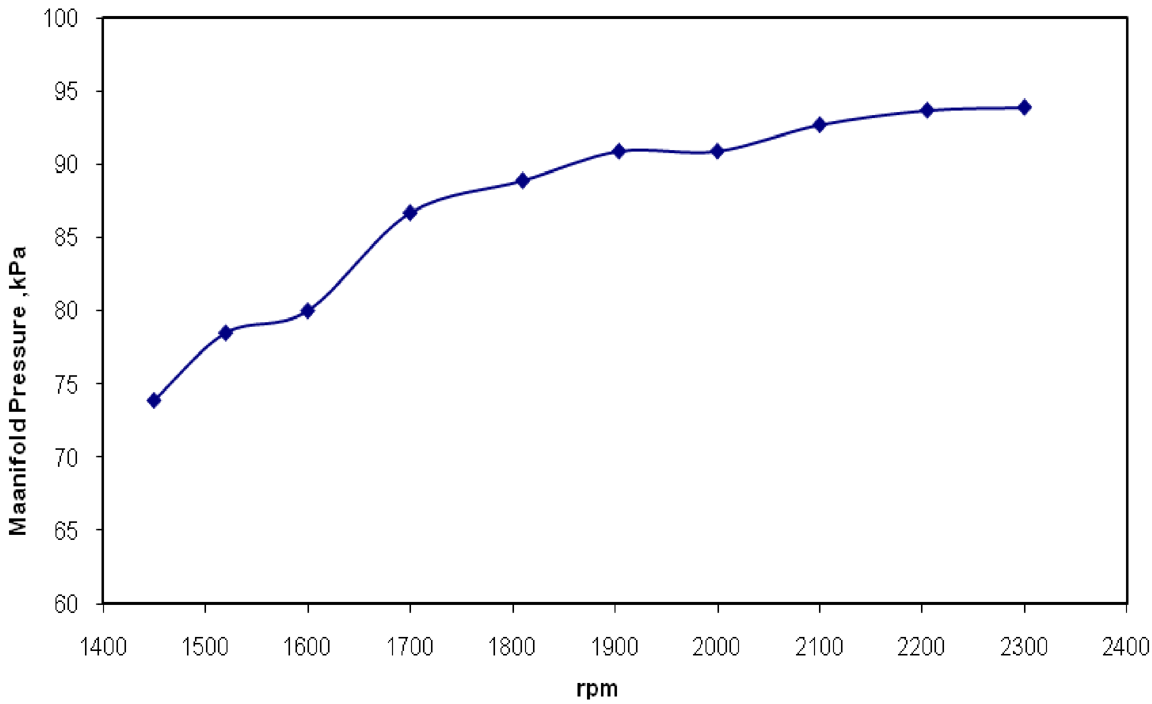


Figure 4.8 Manifold Vacuum Pressure for CR=11

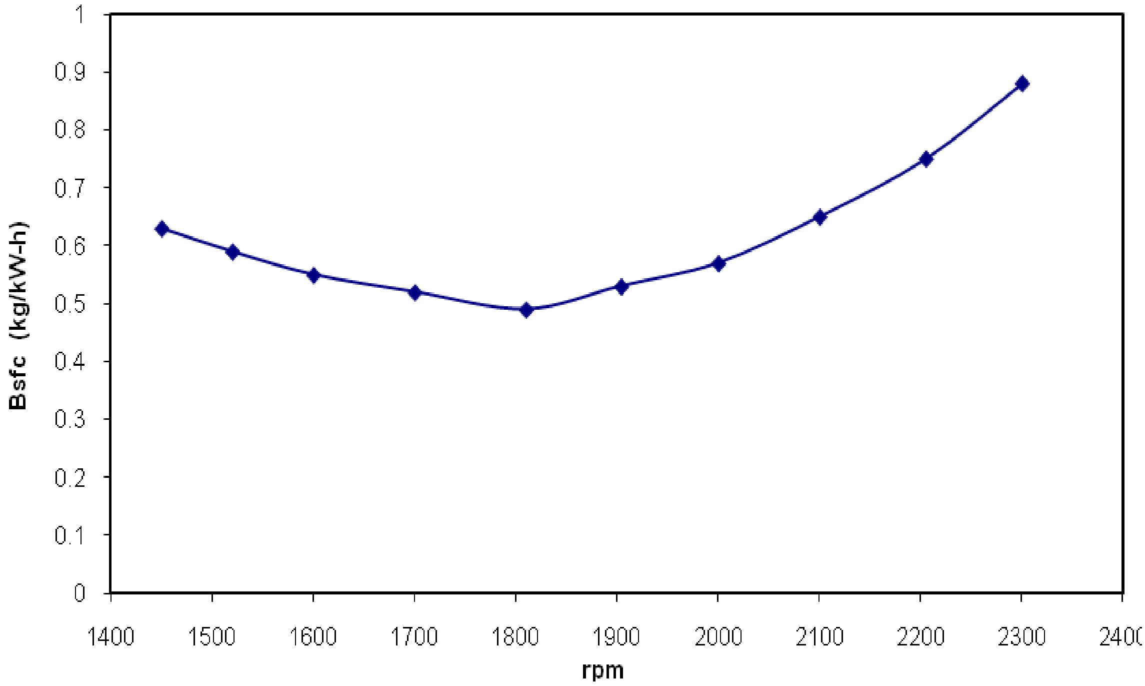


Figure 4.9: Brake Specific Fuel Consumption (Bsfc) for CR=11

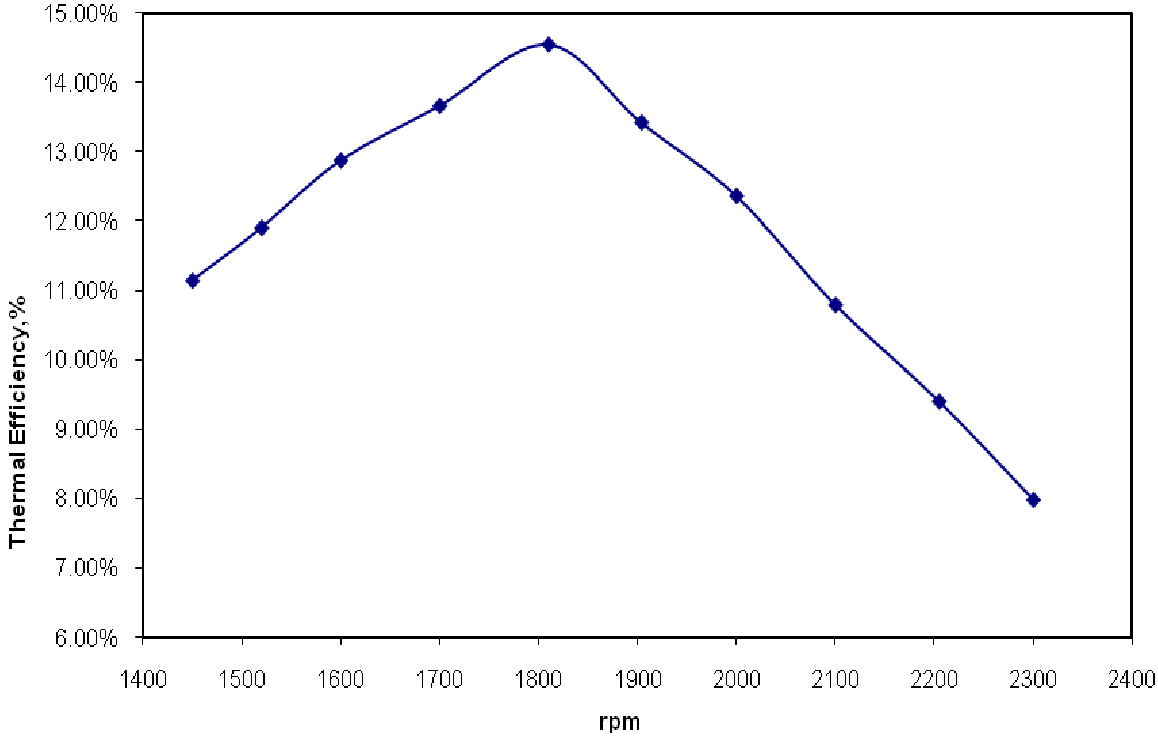


Figure 4.10: Thermal Efficiency for CR=11

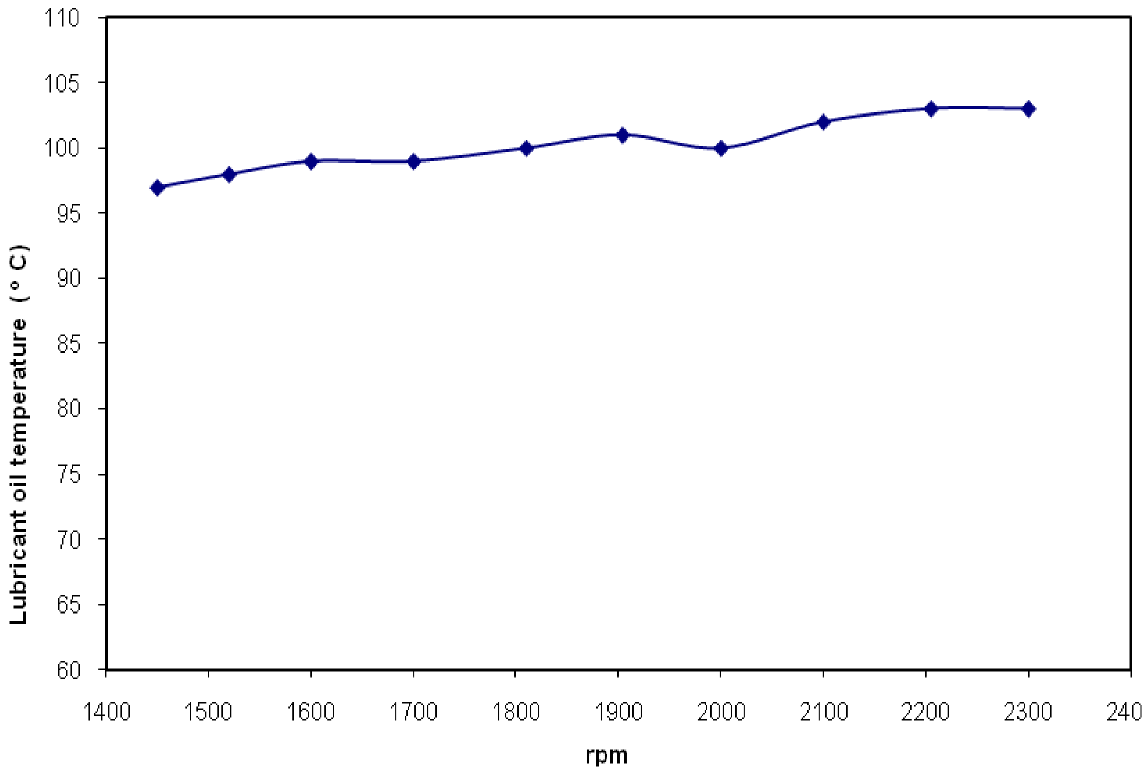


Figure 4.11: Lubricant oil temperature for CR=11

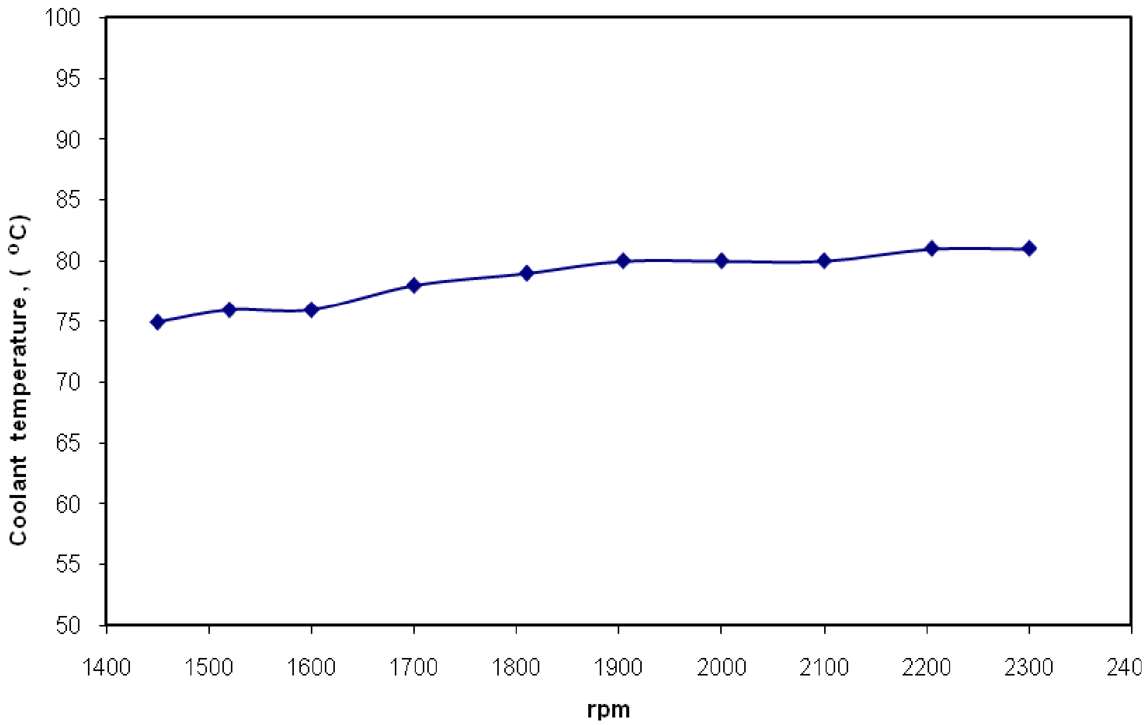


Figure 4.12 Coolant (water) temperature for CR=11

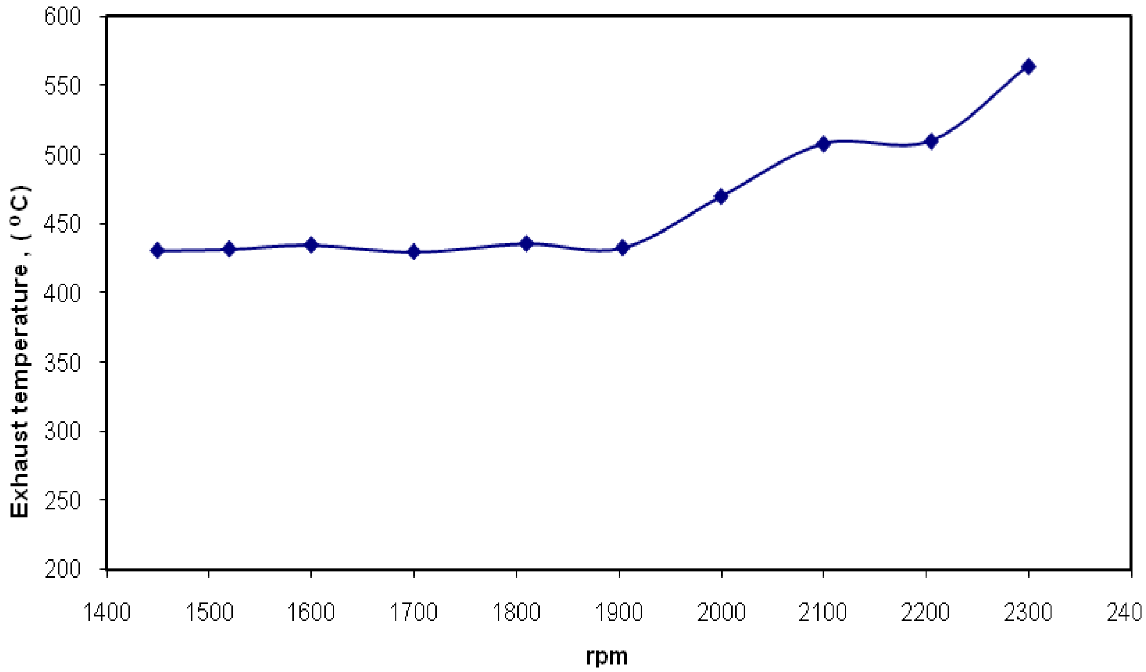


Figure 4.13 Exhaust gas temperature for CR=11

4.1.5 Applied Load, Power and Brake Mean Effective Pressure (Bmep) for CR=12

In Figures 4.14, 4.15 and 4.16 maximum applied load 47 kg, maximum power of 34.72 hp (43.4% of rated power) and maximum Brake mean effective pressure (Bmep) 4.22 bar was observed in the region of 1700 to 1800 rpm. From the peak load, power and torque for further increased or decreased of rpm within 1450 to 2300 range all data were found as a decreased value.

4.1.6 Air Flow rate , Volumetric Efficiency , Fuel flow rate , Air / Fuel Ratio and Engine Manifold pressure for CR=12

From figure 4.17 it was found that air flow rate increased gradually with increase of rpm. The flow variation within 157.93 kg/h to 183.10 kg/h was observed within 1450 to 2300 rpm. It was observed that the volumetric efficiency of the CNG converted engine has been decreased almost linearly with the increase of engine speed within the range 1450 to 2300 rpm as in Figure 4.18. The maximum efficiency was found to be 76.15% at lower engine speed (1450 rpm) and minimum efficiency 55.66% was obtained at a higher experimental engine speed (2300 rpm).

It was observed in Figure 4.19 that the mass flow rate of NG found 19.6% lower than the equivalent diesel flow rate at same engine speed range. Both the flow rate represented very little variation from 1450 to 2300 rpm. Maximum and minimum flow rate for NG was found to be 10.68 kg/h and 10.3 kg/h respectively.

It was observed little variation in A/F ratio distribution within a rpm range. As in Figure 4.20 the variation was found in A/F ratio between 15.09 to 17.79 within the operating speed range. Except few all the values are less than the stoichiometric A/F ratio (17.2) of natural gas.

The engine manifold pressure at WOT was changed significantly with the increase of engine speed. The variation found within the range 82.3 kPa (at lower speed 1450 rpm) to 94 kPa (at higher speed 2300 rpm) as in Figure 4.21.

4.1.7 Brake Specific Fuel Consumption (Bsfc) and Thermal Efficiency for CR=12

It was found in Figure 4.22 that the minimum bsfc would be 0.42 kg/kW-h at 1810 rpm. Maximum bsfc value 0.69 kg/kW-h was attained at maximum experimental engine speed 2300 rpm. Bsfc varied from 0.42 to 0.69 kg/kW-h within 1450 to 2300 rpm range.

Thermal efficiency increases linearly upto its peak value 17.09 % at 1810 rpm then linerly decreases for further increase in engine speed as in Figure-4.23. Minimum efficiency 10.23 % was obtained at a rpm 2300 (maximum experimental rpm). So, maximum fuel utilization from the CNG fuel 17 % can be possible if the engine run at nearly 1800 rpm.

4.1.8 Temperature Measurement for Lubricant Oil, Coolant and Exhaust Gas for CR=12

In Figure 4.24 a variation in lubricant oil temperature was observed with the change of engine speed. Temperature variation was found within the range of 87^0 to 104^0C when engine speed was varied within a speed from 1450 to 2300 rpm, the coolant temperature was changed within the range of 68^0 to 80^0C as in Figure 4.25. It was observed in Figure 4.26 that the gradual variation of exhaust temperatures between 451^0C to 621^0C was found for 1450 to 2300 rpm range. The maximum exhaust temperature was attained 621^0C at 2300 rpm..

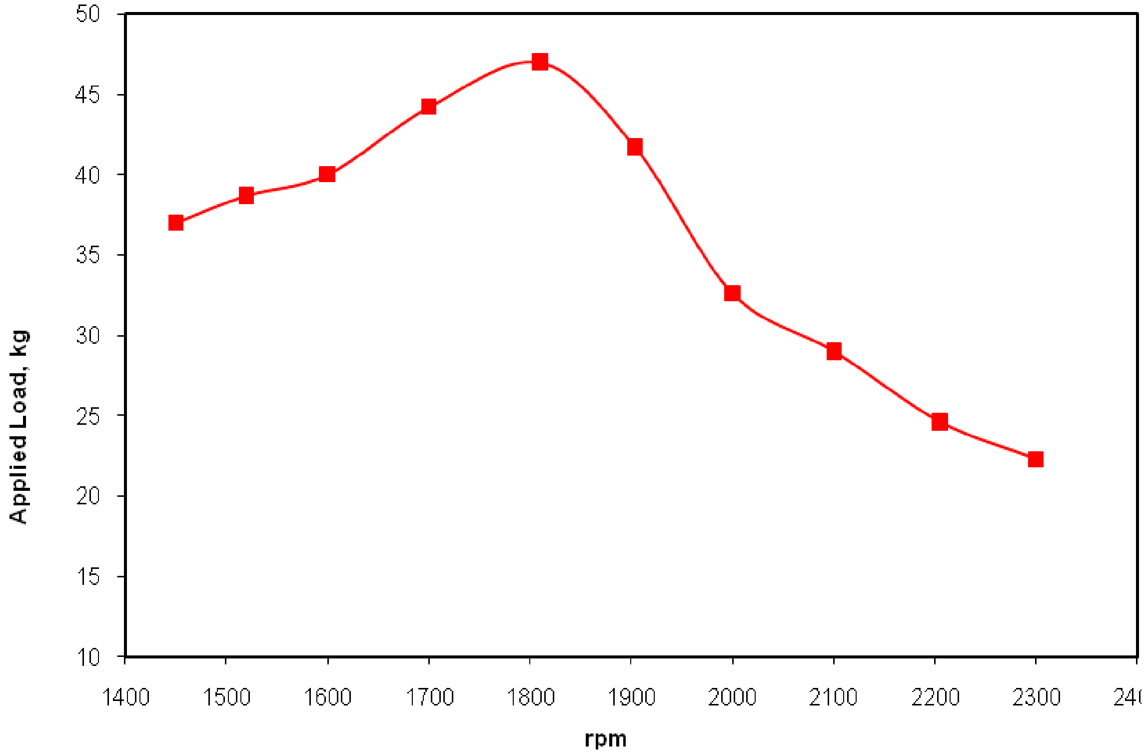


Figure 4.14: Applied load for CR=12

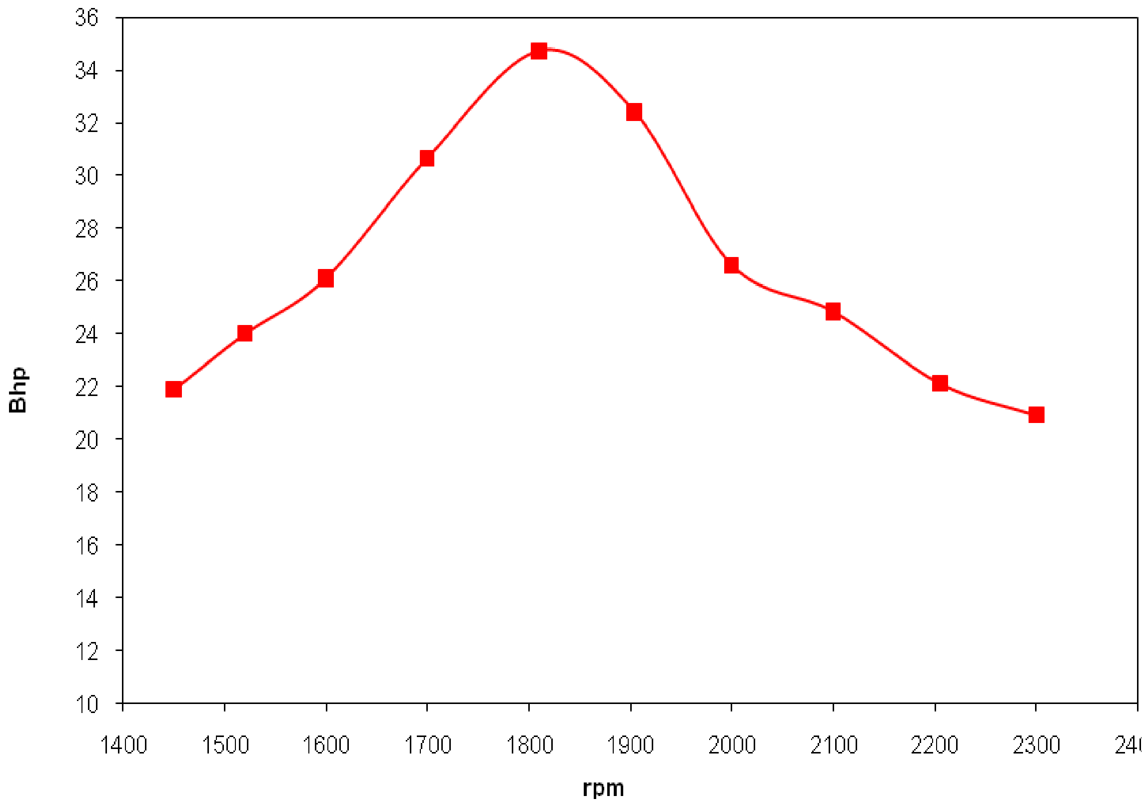


Figure 4.15: Brake horse Power for CR=12

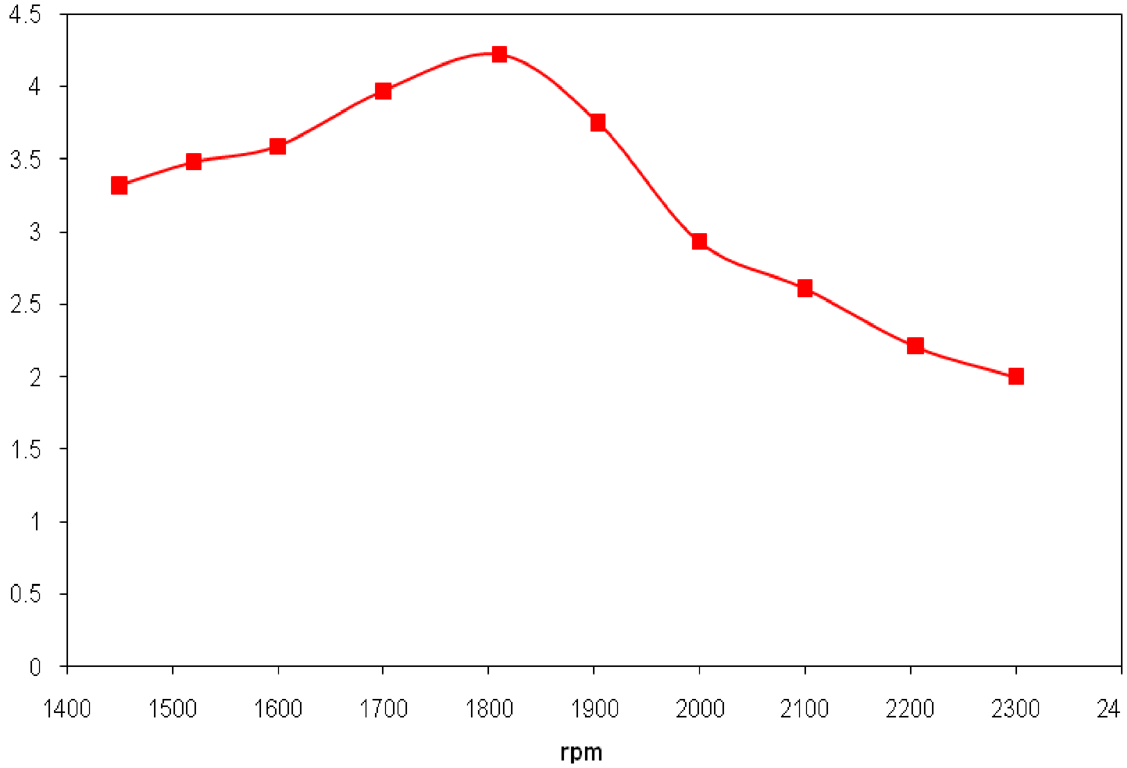


Figure 4.16: Brake Mean Effective Pressure for CR=12

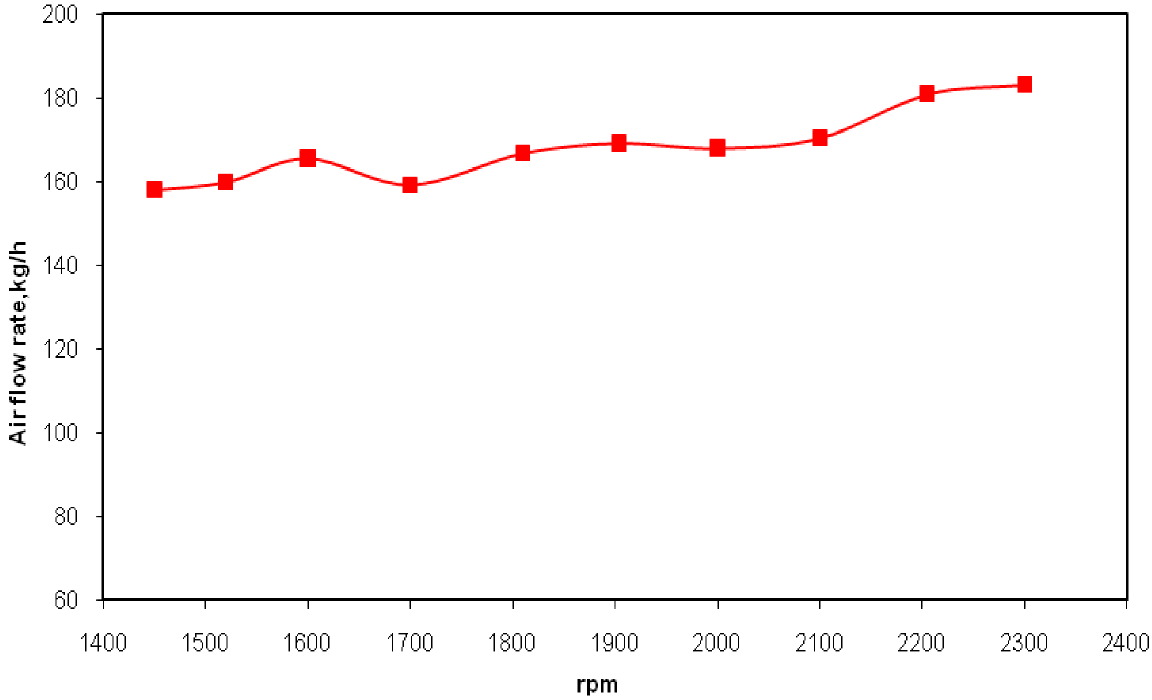


Figure 4.17: Air Flow Rate for CR=12

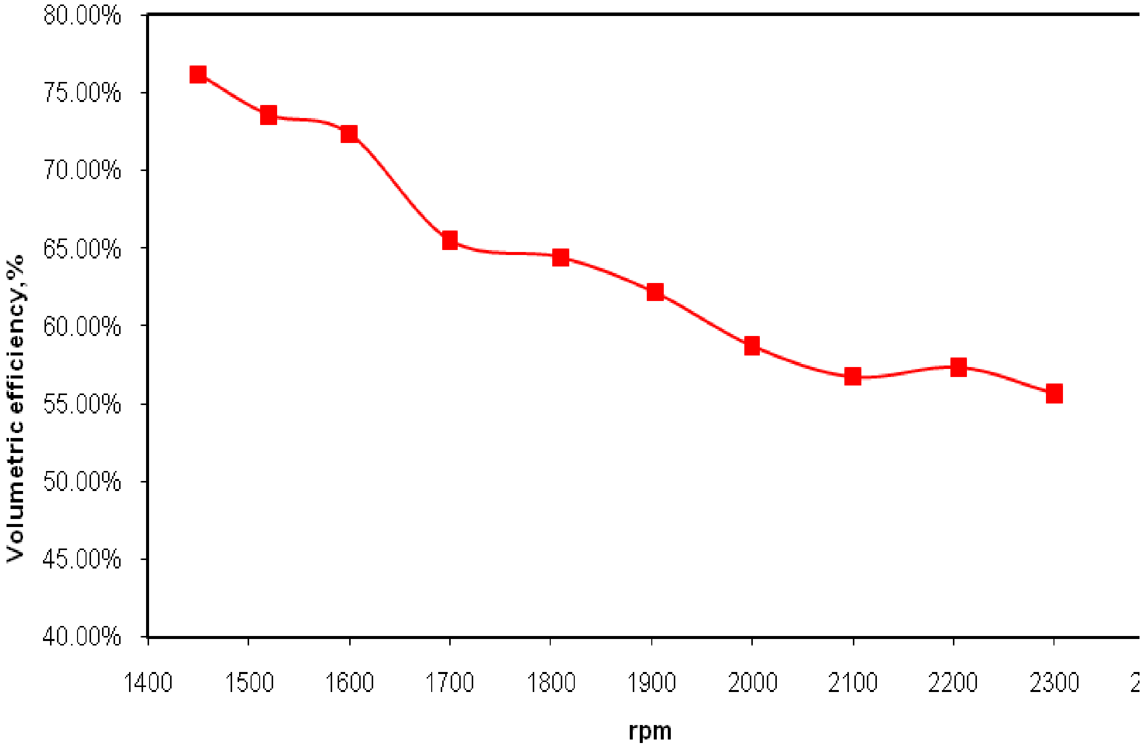


Figure 4.18: Volumetric Efficiency for CR=12

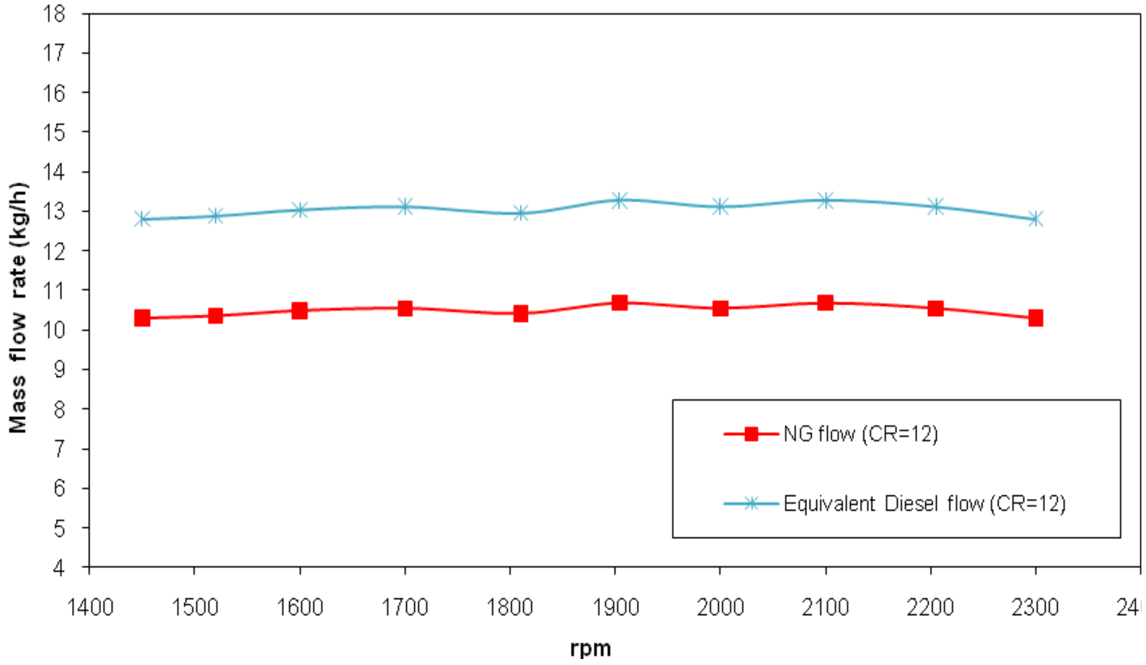


Figure 4.19: Mass flow rate NG and equivalent diesel flow for CR=12

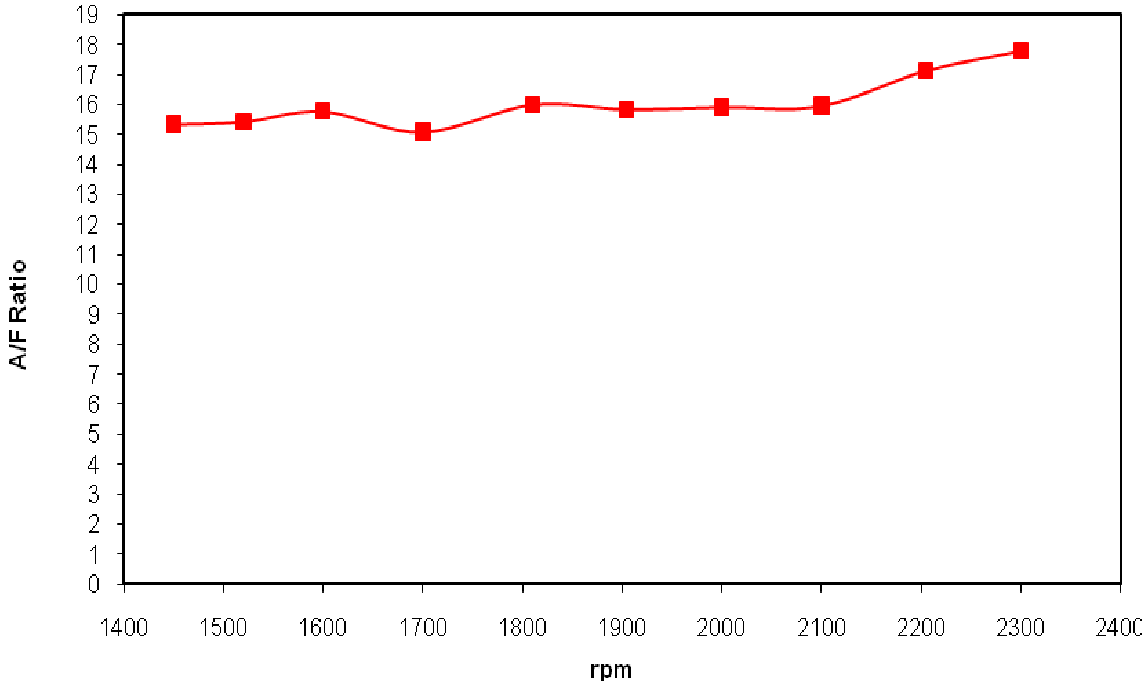


Figure 4.20: A/F ratio for CR=12

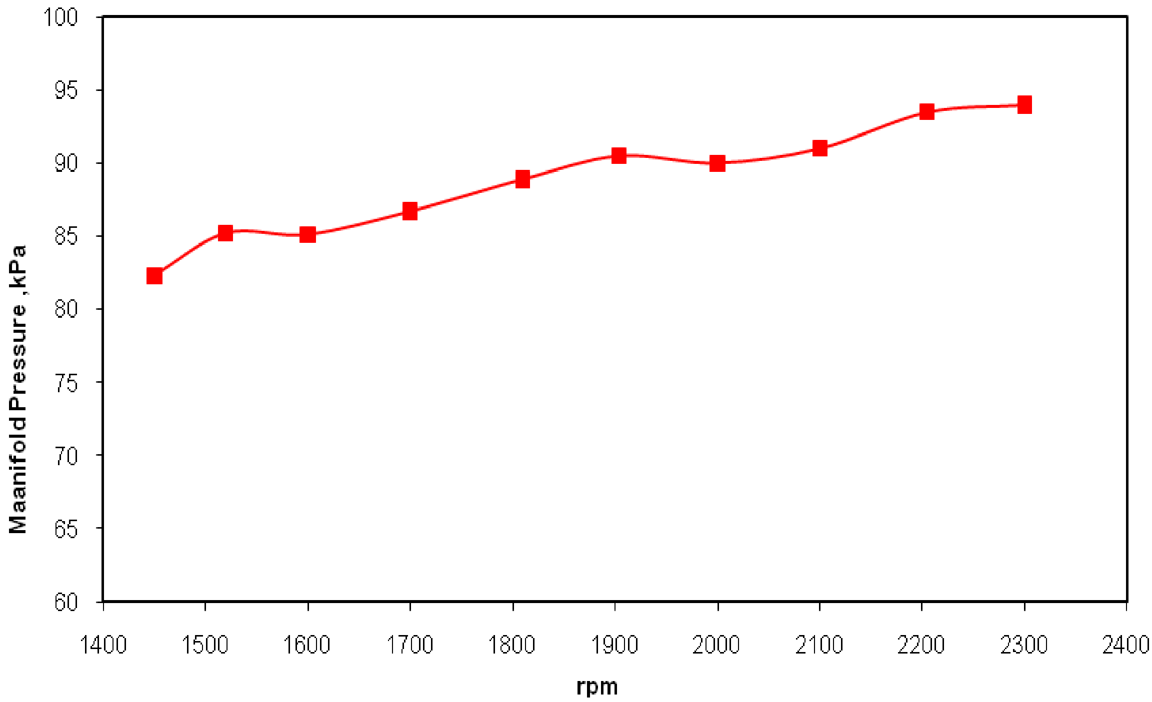


Figure 4.21 Manifold Vacuum Pressure for CR=12

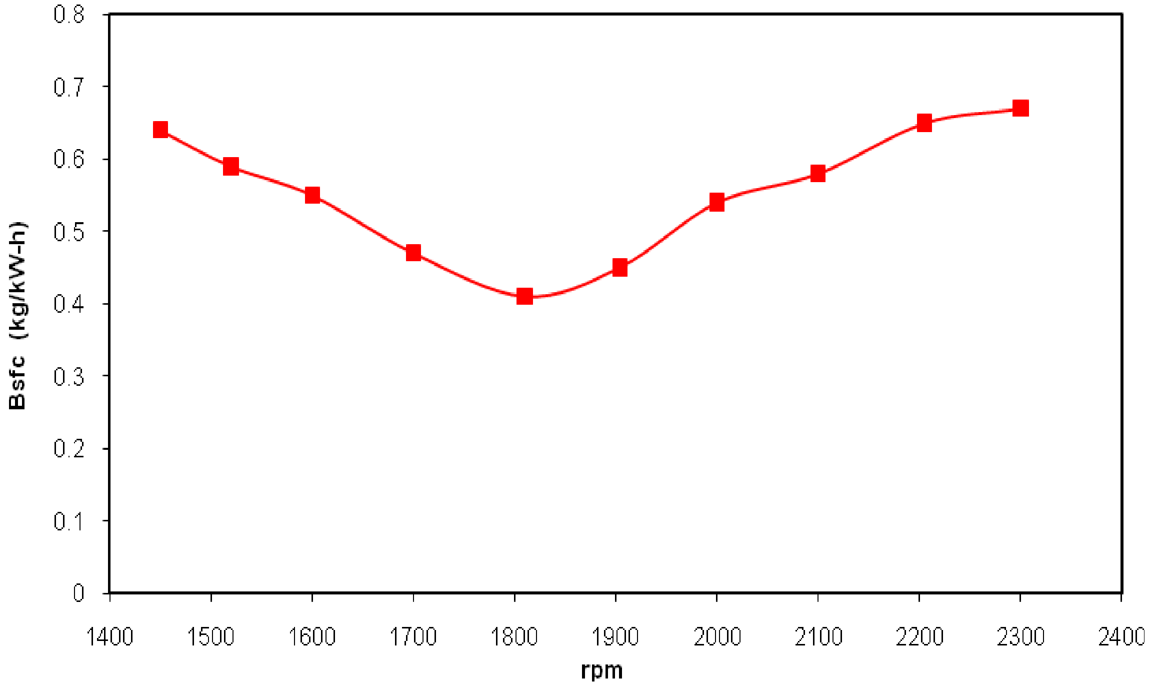


Figure 4.22: Brake Specific Fuel Consumption (Bsfc) for CR=12

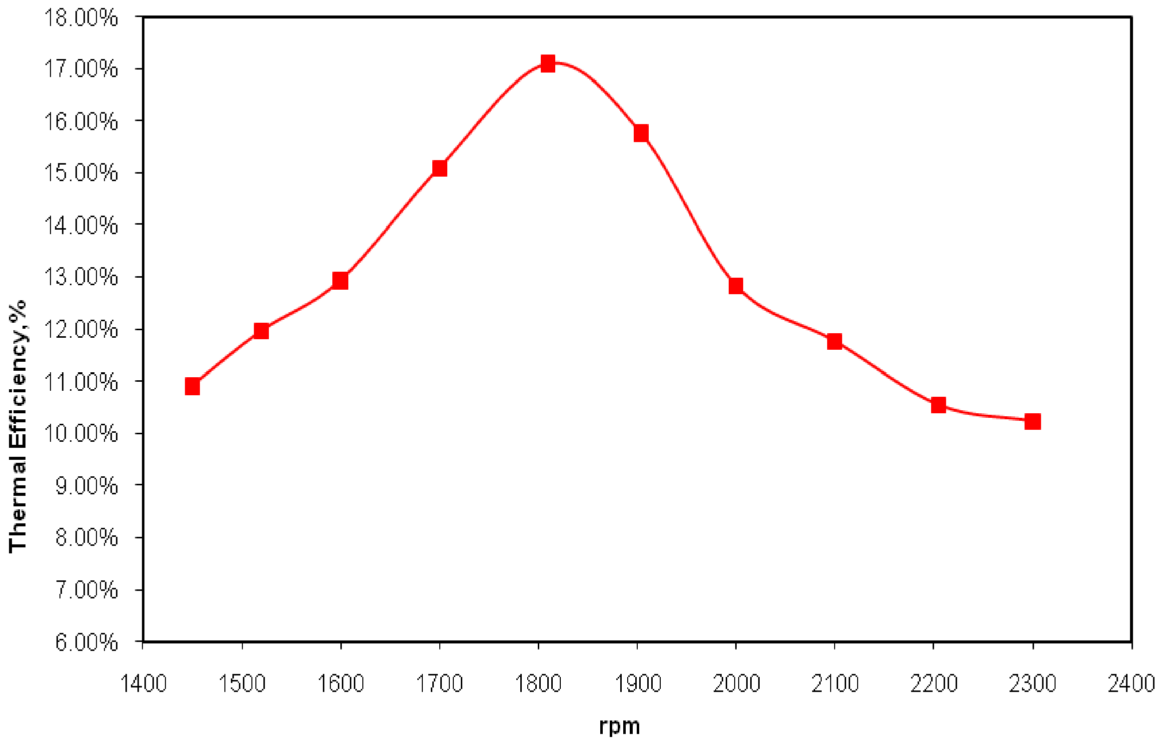


Figure 4.23: Thermal Efficiency for CR=12

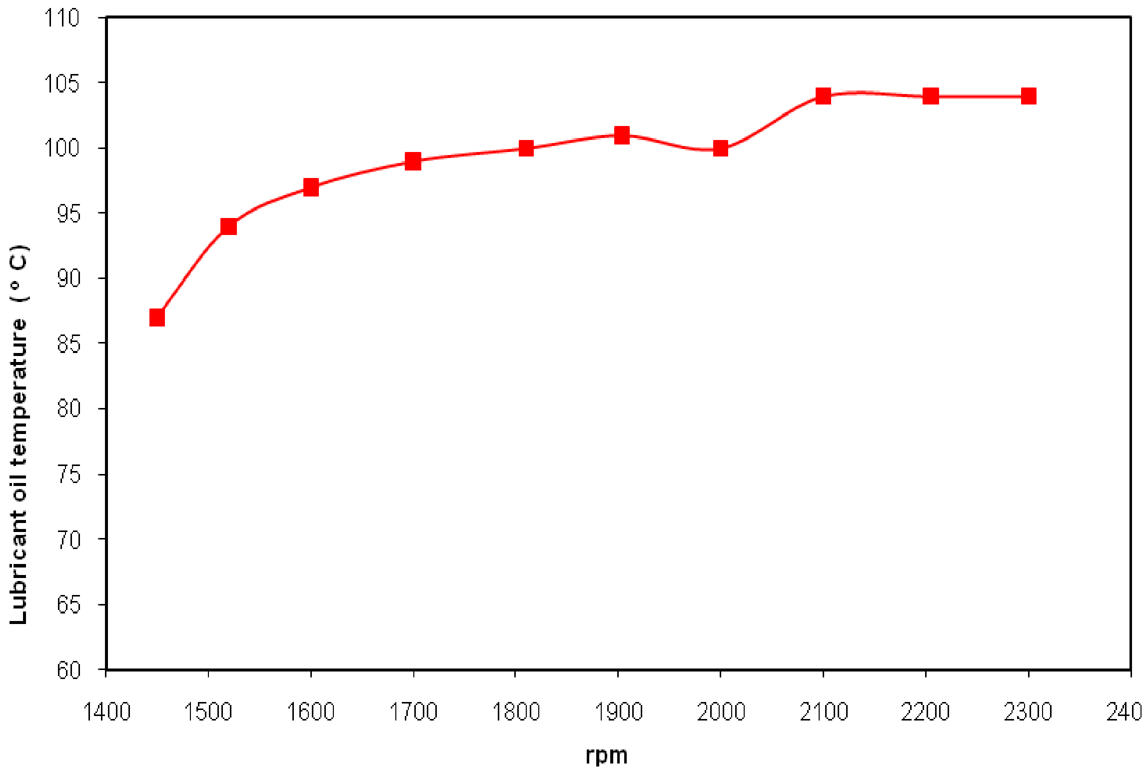


Figure 4.24: Lubricant oil temperature for CR=12

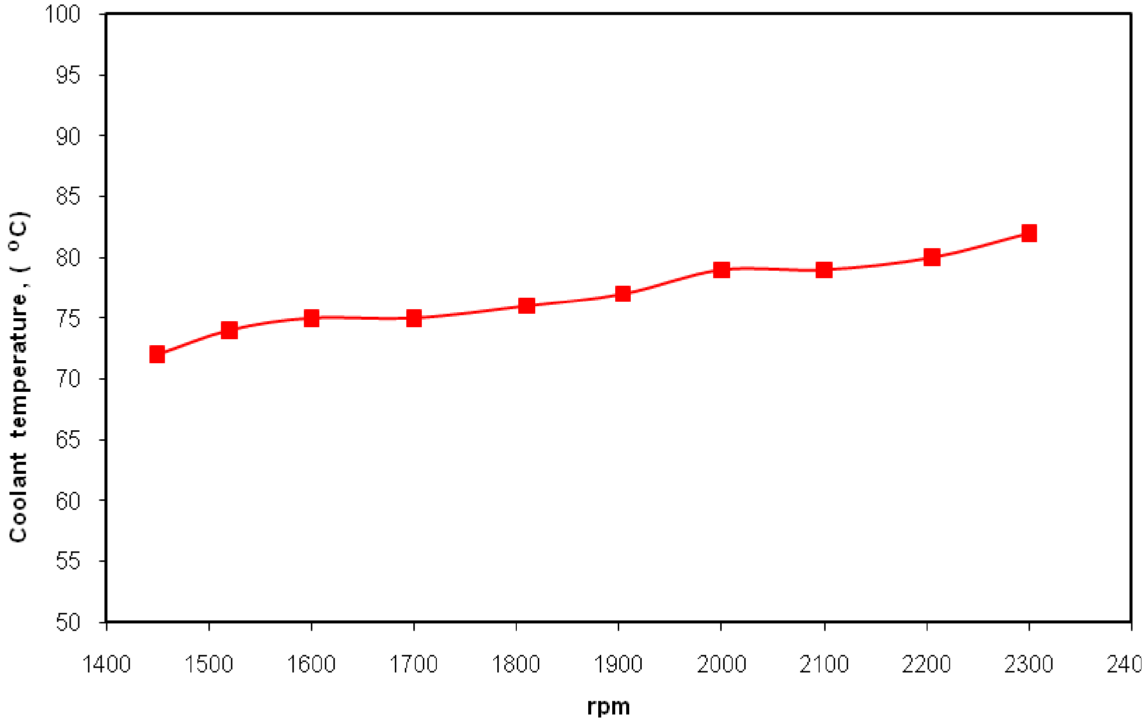


Figure 4.25 Coolant (water) temperature for CR=12

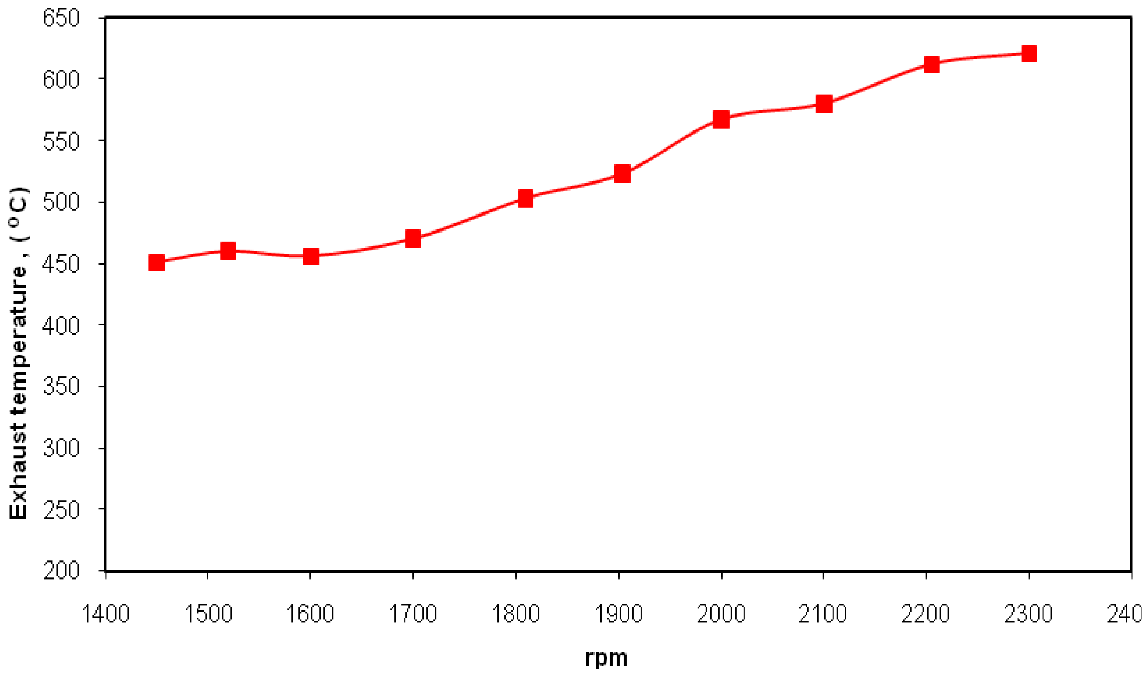


Figure 4.26 Exhaust gas temperature for CR=12

4.1.9 Applied Load, Power and Brake Mean Effective Pressure (Bmep) for CR=13

In Figures 4.27, 4.28 and 4.29 maximum applied load 45 kg, maximum power of 33.24 hp (41.55% of rated power) and maximum Brake mean effective pressure (Bmep) 4.04 bar was observed in the region of 1700 to 1800 rpm. From the peak load, power and torque for further increased or decreased of rpm within 1450 to 2300 range all data were found as a decreased value.

4.1.10 Air Flow rate , Volumetric Efficiency , Fuel flow rate , Air / Fuel Ratio and Engine Manifold pressure for CR=13

From figure 4.30 it was found that air flow rate change very gradual throughout the rpm range. The flow variation within 144.4 kg/h to 175.11 kg/h was observed within 1450 to 2300 rpm. It was observed that the volumetric efficiency of the CNG converted engine has been decreased almost linearly with the increase of engine speed within the range 1450 to 2300 rpm as in Figure 4.31. The maximum efficiency was found to be 69.63% at lower engine speed (1450 rpm) and minimum efficiency 53.24% was obtained at a higher experimental engine speed (2300 rpm).

It was observed in Figure 4.32 that the mass flow rate of NG found 19.6% lower than the equivalent diesel flow rate at same engine speed range. NG flow rate represented increased values with increase of rpm within range 1450 to 2300. Maximum and minimum flow rate for NG was found to be 12.45 kg/h and 10.93 kg/h respectively

It was observed little variation in A/F ratio distribution within a rpm range. As in Figure 4.33 the variation was found in A/F ratio between 15.09 to 17.79 within the operating speed range. All the values are less than the stoichiometric A/F ratio (17.2) of natural gas.

It was observed that for CR=13 the engine manifold pressure at WOT changed significantly with the increase of engine speed. The variation found within the range 86.7 kPa (at lower speed 1450 rpm) to 94 kPa (at higher speed 2300 rpm) as in Figure 4.34.

4.1.11 Brake Specific Fuel Consumption (Bsfc) and Thermal Efficiency for CR=13

It was found in Figure 4.35 that the minimum bsfc would be 0.49 kg/kW-h at 1810 rpm . Maximum bsfc value 0.67 kg/kW-h was attained at maximum experimental engine speed 2300 rpm. Bsfc varied from 0.49 to 0.67 kg/kW-h within 1450 to 2300 rpm range.

Thermal efficiency increases linearly upto its peak value 14.30 % at 1810 rpm then linearly decreases for further increase in engine speed as in Figure-4.36. Minimum efficiency 10.23 % was obtained at a rpm 2300 (maximum experimental rpm). So, maximum fuel utilization from the CNG fuel 14.30% can be possible if the engine run at nearly 1800 rpm.

4.1.12 Temperature Measurement for Lubricant Oil, Coolant and Exhaust Gas for CR=13

In Figure 4.37 a variation in lubricant oil temperature was observed with the change of engine speed. Temperature variation was found within the range of 99^0 to 110^0C when engine speed was varied within a speed from 1450 to 2300 rpm, the coolant temperature was changed within the range of 77^0 to 86^0C as in Figure 4.38. It was observed in Figure 4.39 that the gradual incremental variation of exhaust temperatures between 490^0C to 640^0C was observed for 1450 to 2300 rpm range. The maximum exhaust temperature was attained 640^0C at 2300 rpm.

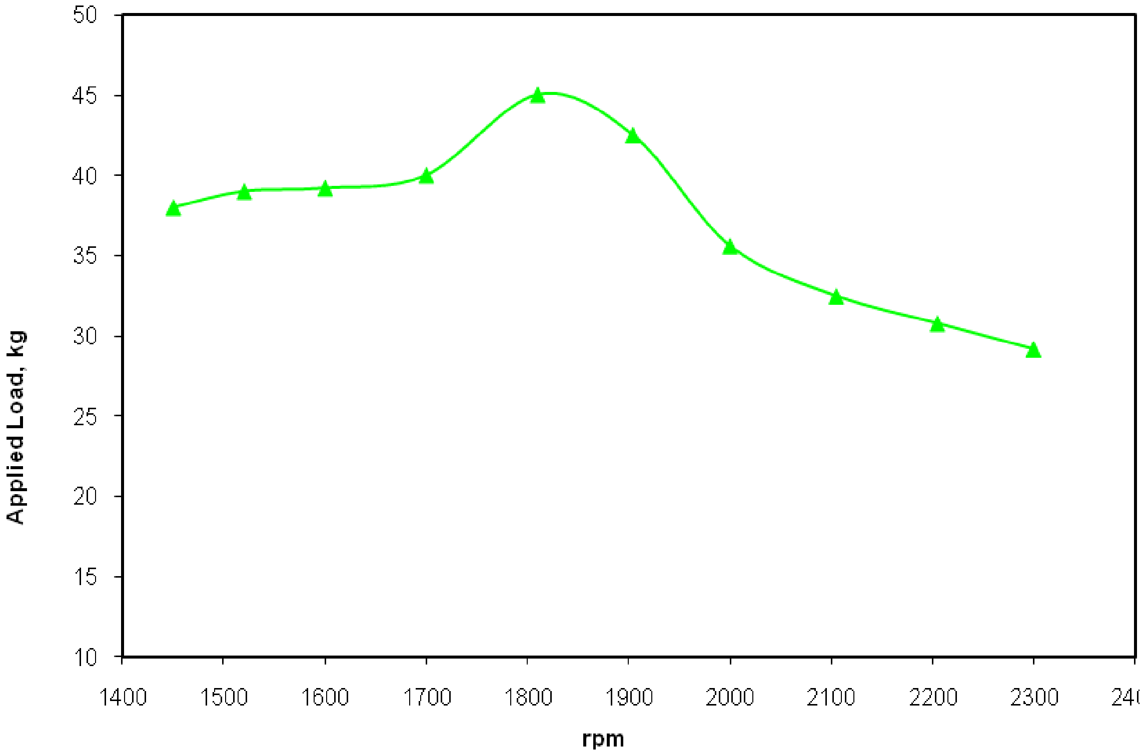


Figure 4.27: Applied load for CR=13

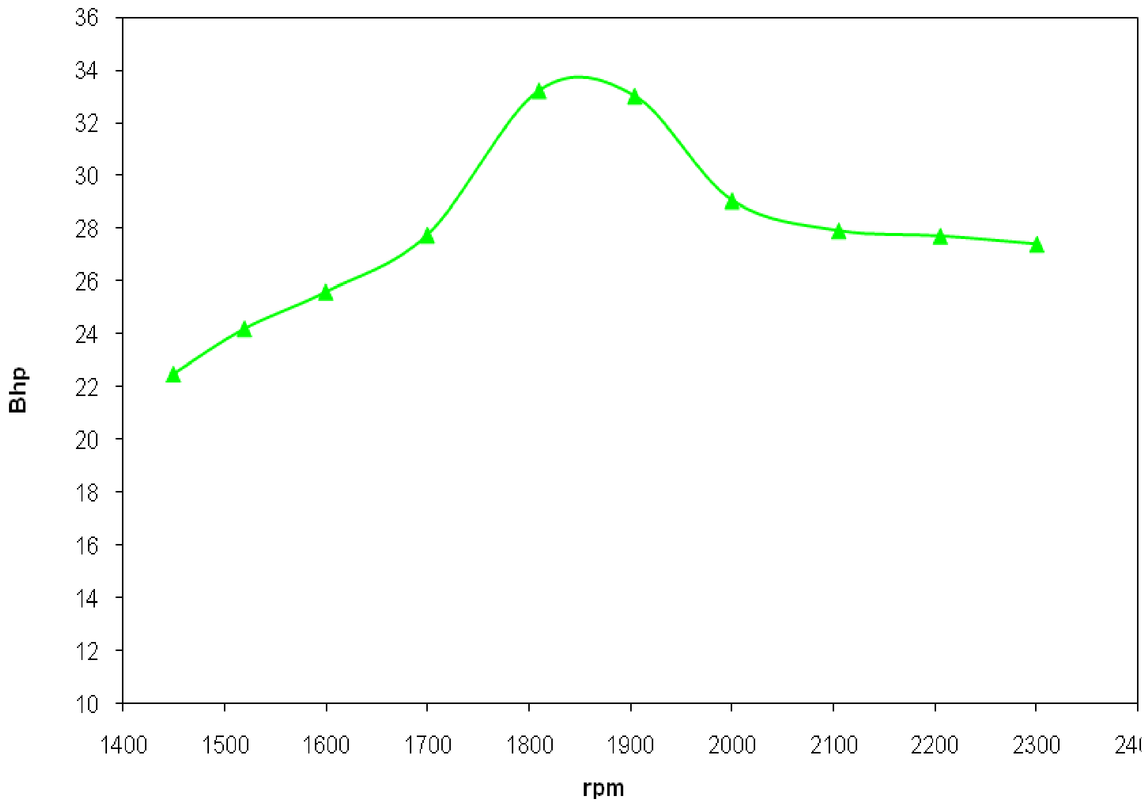


Figure 4.28: Brake horse power for CR=13

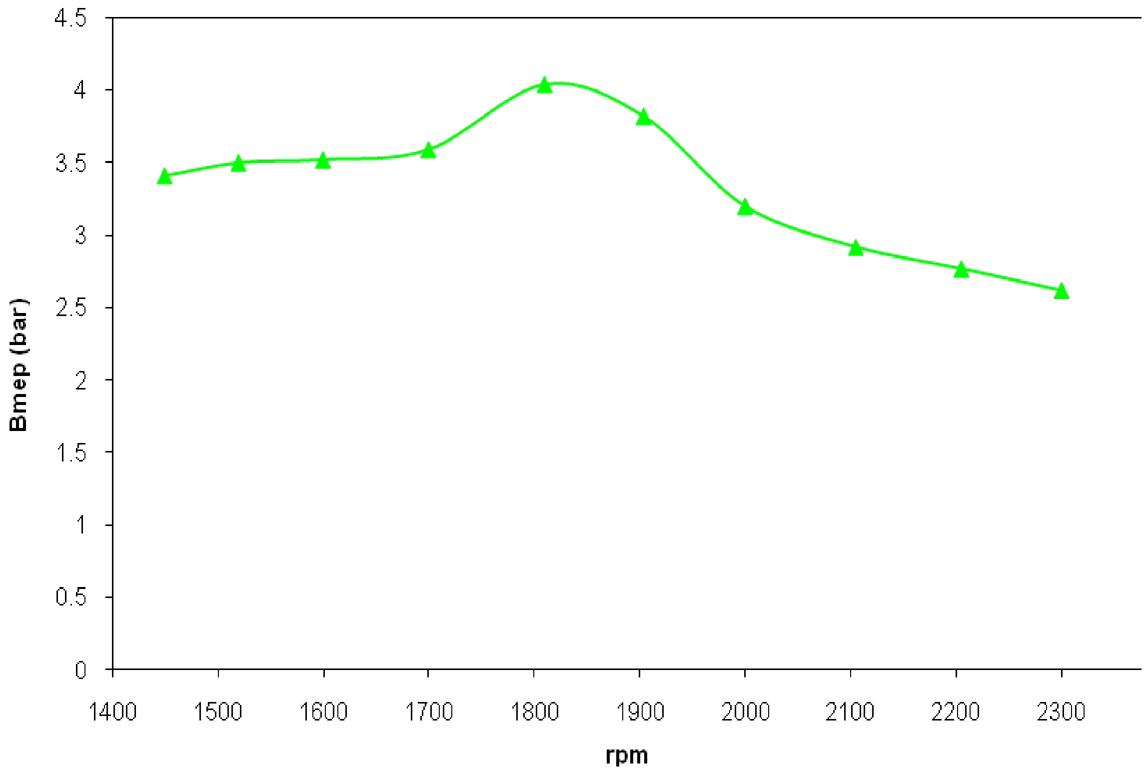


Figure 4.29: Bmep for CR=13

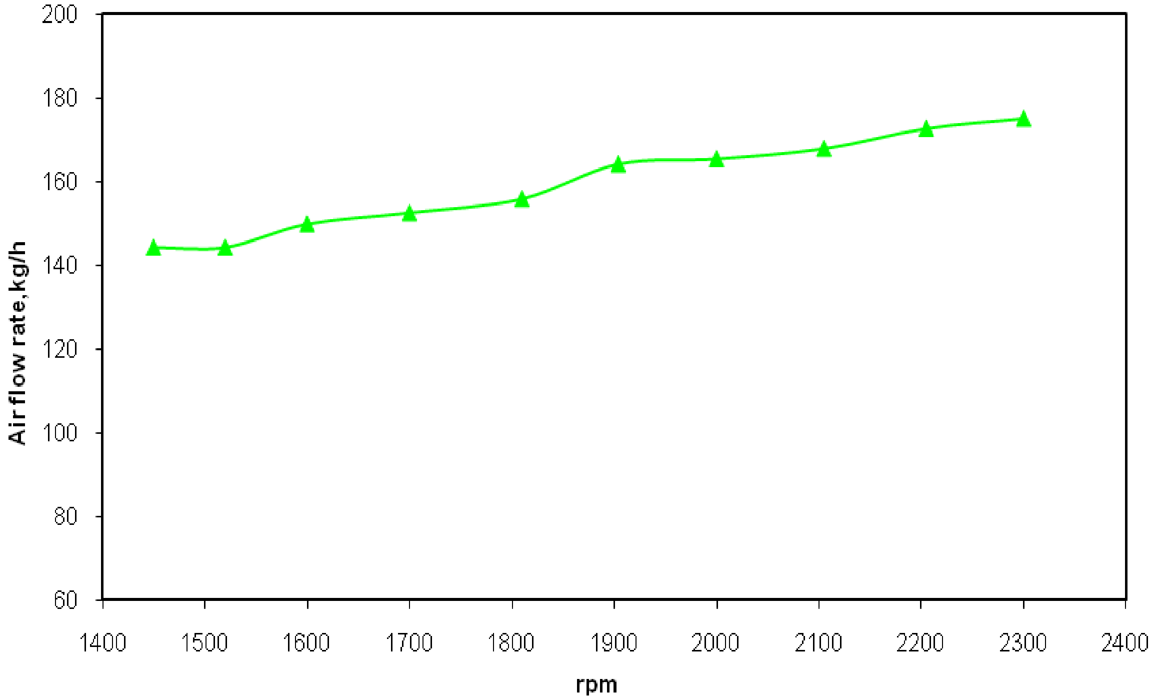


Figure 4.30: Air flow rate for CR=13

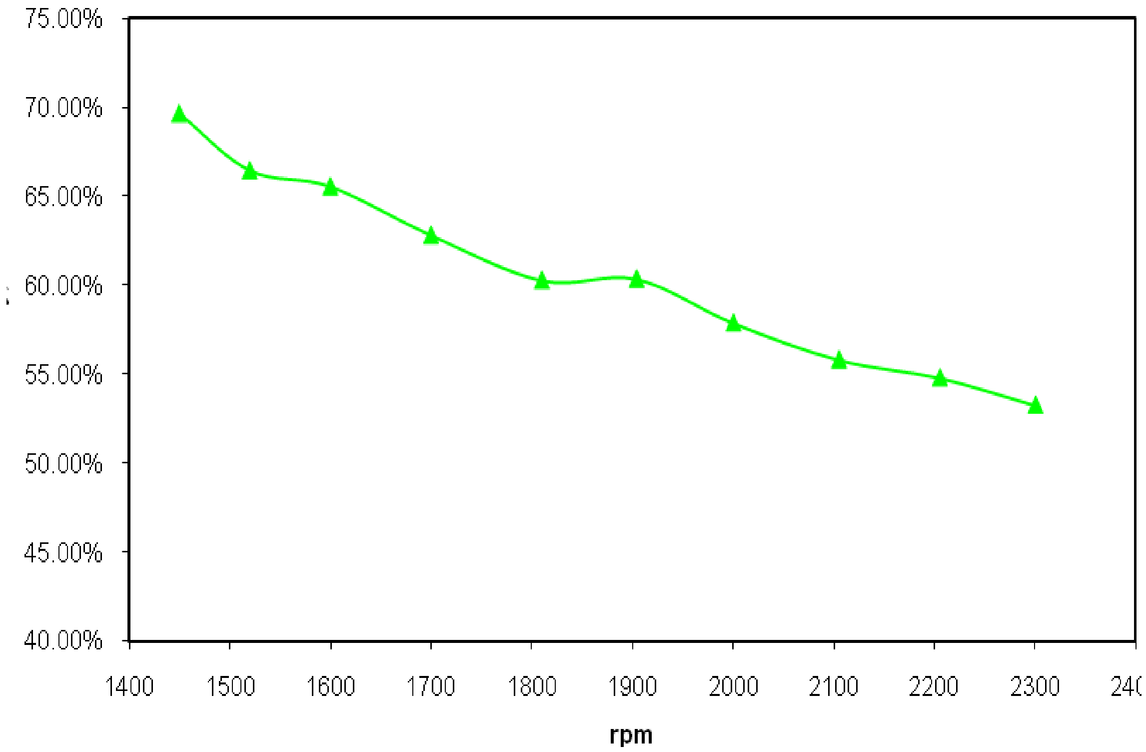


Figure 4.31: Volumetric efficiency for CR=13

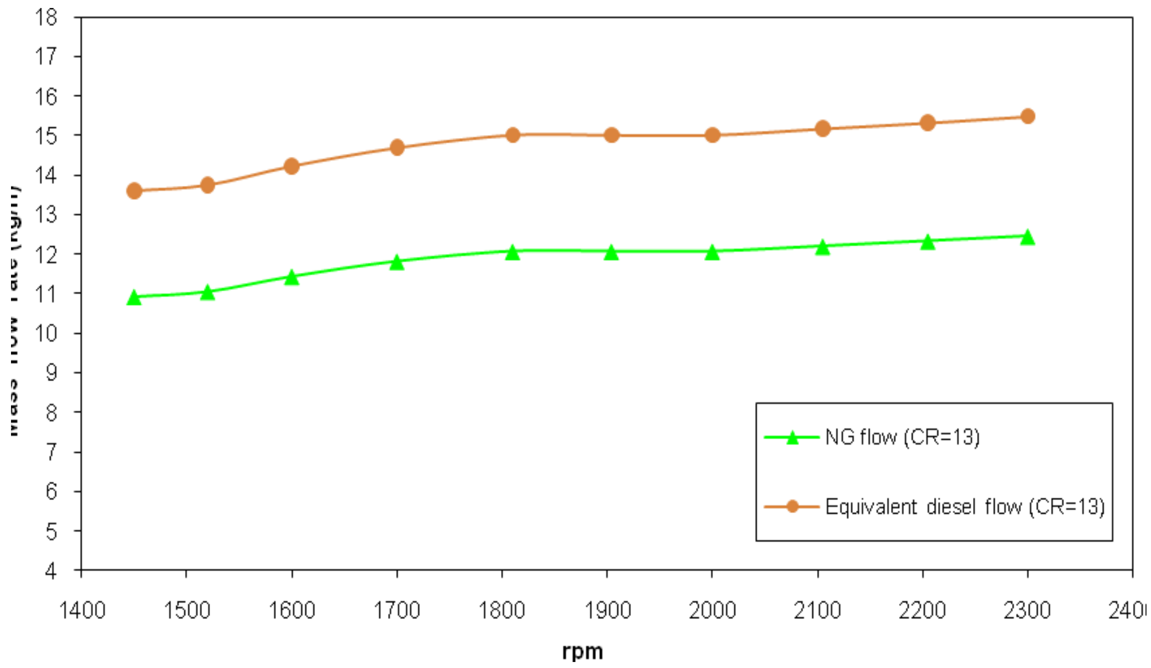


Figure 4.32: Mass flow rate NG and Equivalent Diesel flow distribution for CR=13

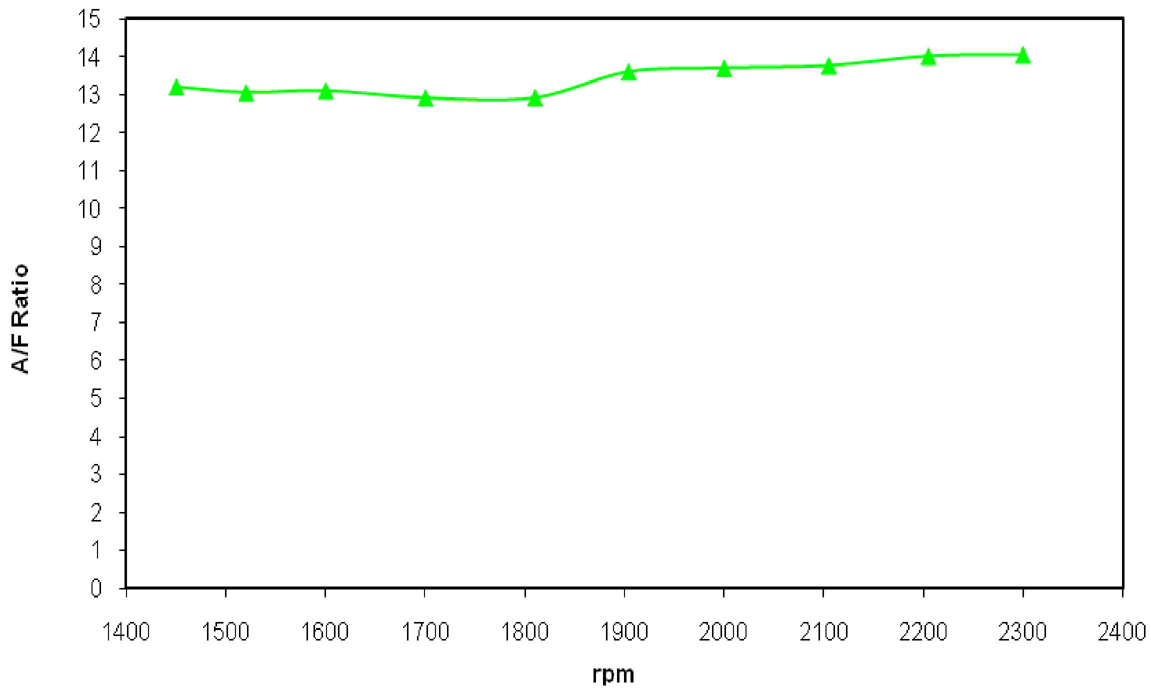


Figure 4.33: Air /Fuel Ratio for CR=13

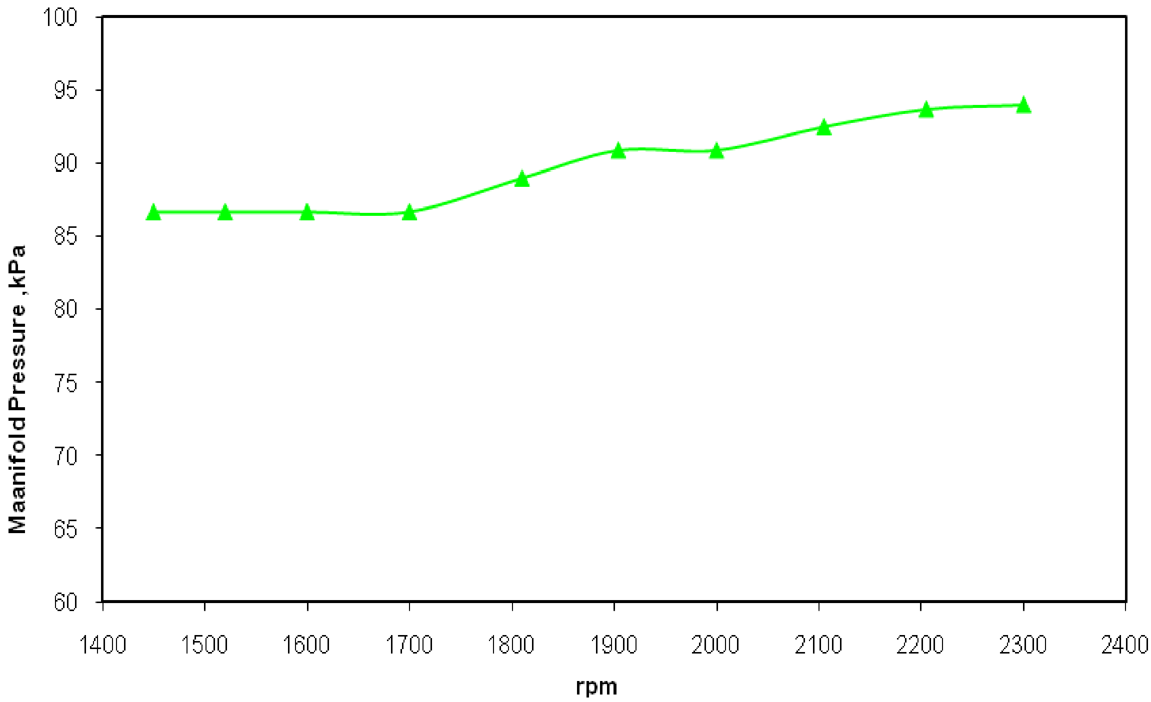


Figure 4.34: Manifold vacuum pressure for CR=13

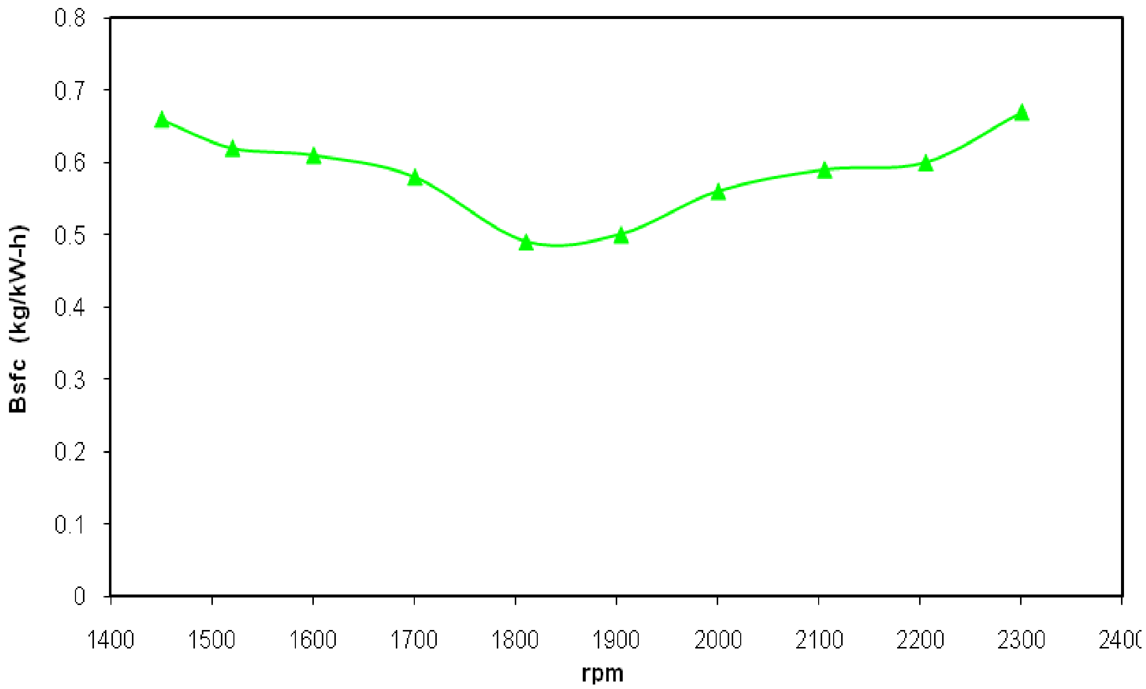


Figure 4.35: Brake specific fuel consumption (Bsfc) for CR=13

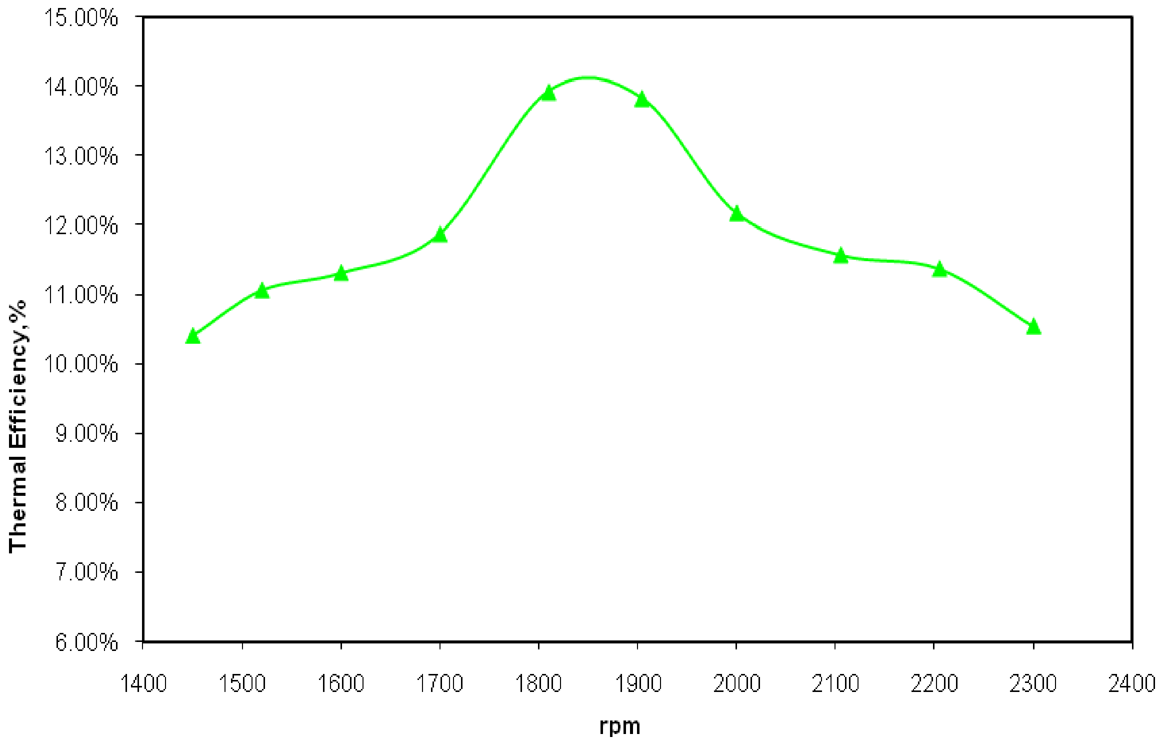


Figure 4.36: Thermal efficiency for CR=13

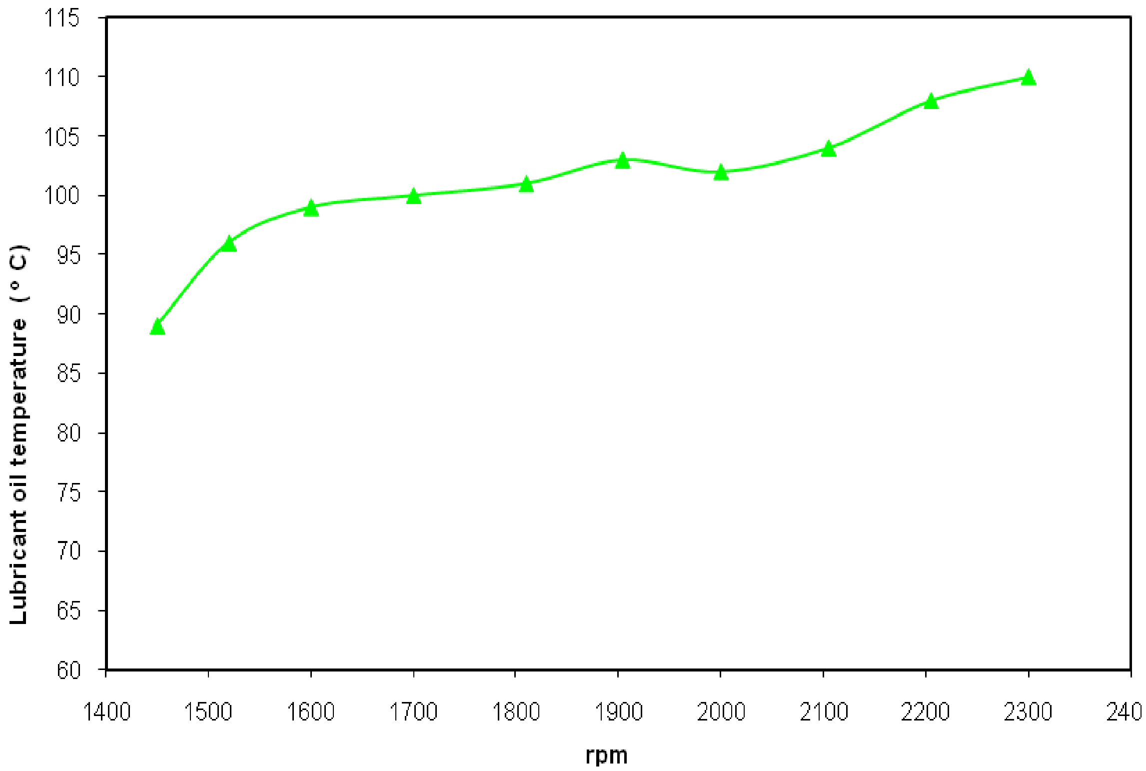


Figure 4.37: Lubricant oil temperature for CR=13

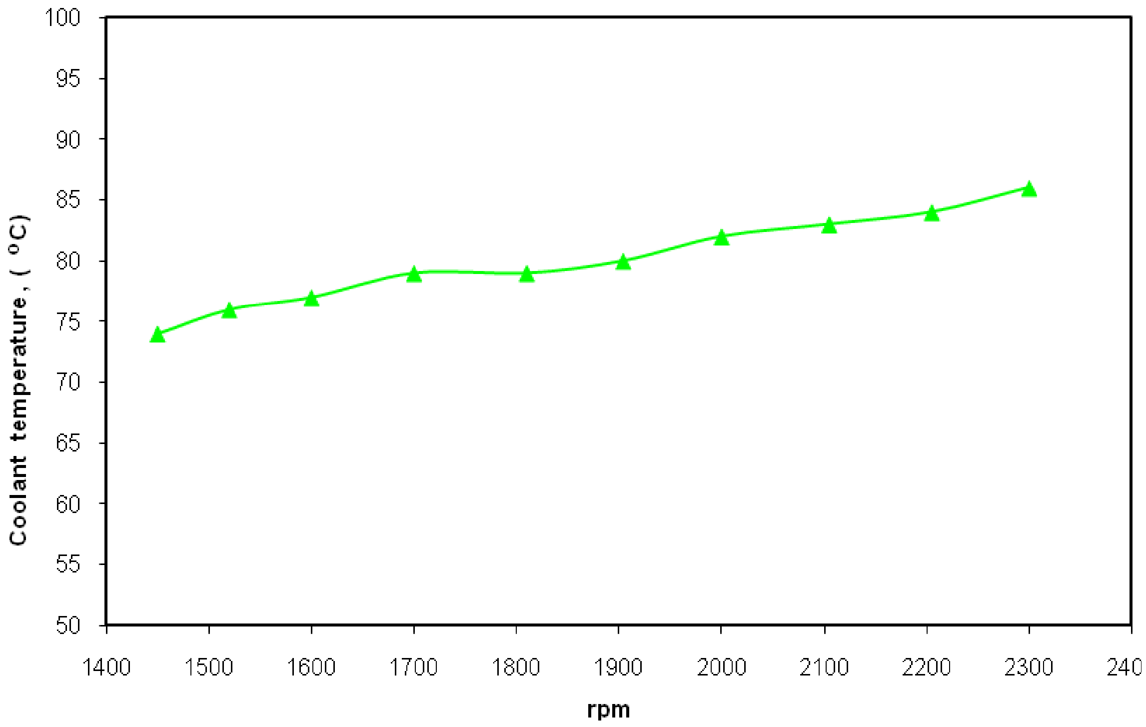


Figure 4.38 Coolant (water) temperature for CR=13

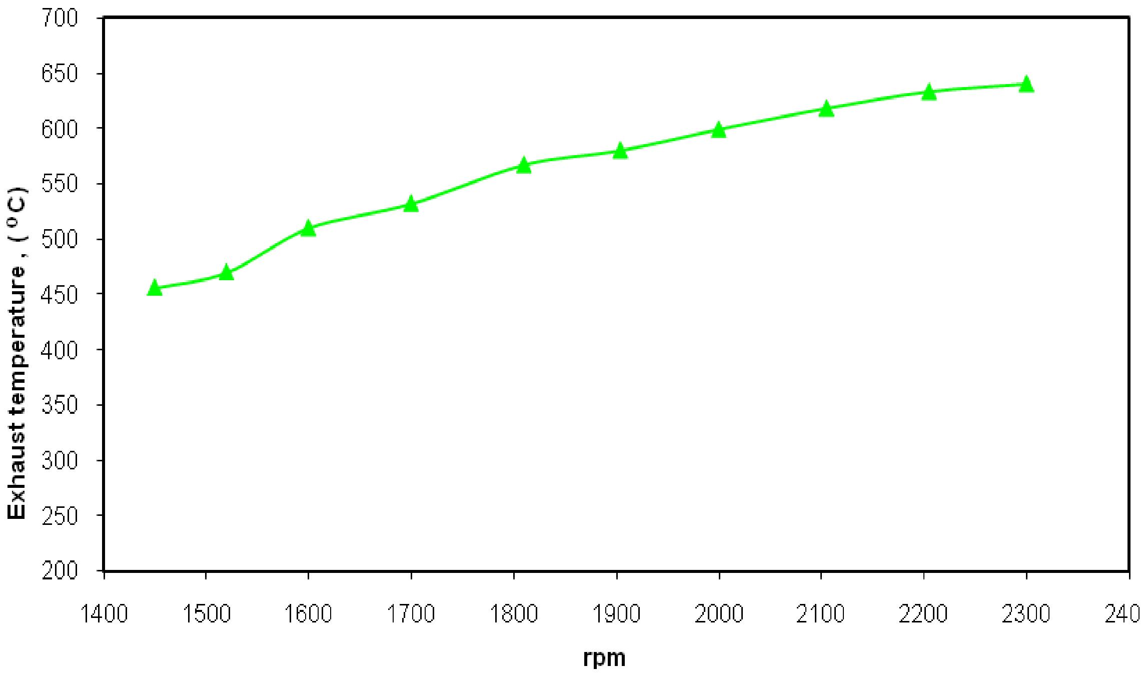


Figure 4.39 Exhaust gas temperature for CR=13

4.1.13 Comparison of Applied Load, Power and Brake Mean Effective Pressure (Bmep) for CR=11,12 and 13

From Figures 4.40, 4.41, 4.42 it was observed that peak load was found at around 1800 rpm in all CRs. The maximum peak load 47 kg was found at CR=12 thus maximum power 34.72 hp (43.4% of the rated power) was attained at this point out of three compression ratio upto its peak. At higher rpm the applied load and power was observed maximum values data in CR=13. But at this condition unusual vibration occurs and engine running condition were not smooth. So, this advantage might be discarded and thus the overall maximum power output throughout with smooth engine running condition, CR=12 was found better.

More the CR of an engine the more the peak cylinder pressure it attained. Bmep is the measure of mean pressure developed on the every power stroke. Which depends on power and engine speed. So, for higher power developed at CR=12 higher Bmep was observed with a maximum value 4.22 bar and was much less in comparing the Bmep (8-10 bar) for a SI engine.

4.1.14 Comparison of Air Flow rate , Volumetric Efficiency , Fuel flow rate , Air / Fuel Ratio and Engine Manifold pressure for CR=13

The trends of air flow rate for all CRs were found similar. For higher rpm more air was needed to the engine in all CRs. But for all the values of A/F ratios, CR=11 and CR=13 were nearly equal except the values of CR=12 was obtained little higher than other two CRs as shown in Figure 4.43. At CR=12 more air requirement was reflected to keep the mixture a lean one.

It was observed that the volumetric efficiency of the CNG converted engine decreased almost linearly by little fluctuations with the increase of engine speed within the range 1450 to 2300 rpm as in Figure 4.44 for all CRs. Throughout the engine speed range the efficiency values were found higher at CR=12 than other two CRs. The maximum volumetric efficiency found 76.15% at CR=12 (most lean mixer) , 69.635% at CR=13 and 66.85% at CR=11 for lower engine speed (1450 rpm).

From Figure 4.45 it was observed that the mass flow rate of NG for CR=11, CR=12 found almost similar upto the mid speed range. Over that speed values for CR=12 decreased a little. NG flow rate for CR=13 was found higher throughout the operating speed range. This increased values made the air fuel mixture rich at CR=13. For higher rate of air flow in CR=12 the A/F ratio was found significantly higher in CR=12 out of all three CRs and for CR=13 being the lowest as in Figure 4.46. It happened due to higher fuel attainment in CR=13 setup engine. In case of all three CRs the flow rate of NG was 19.6% lower than equivalent diesel flow.

Higher value of A/F ratio means the mixer was a lean one. At too rich conditions the engine will have less oxygen to burn with the fuel, causing a rise in hydrocarbon emission or in the worst case it will stall the engine. Within three CRs setup, the A/F ratios for CR=12 was found higher than other two CRs at corresponding rpm as shown in Figure 4.46. This reflected the use of lean mixer in CR=12 than other CRs which was infact desirable. Effect of more air flow rate in CR=12 could make the mixer a lean burn. A/F ratios for CR=12 were close but less than the stoichiometric A/F ratio (17.2) of natural gas.

Variation of Manifold pressure was found within 73.9 to 93.9 kPa , 82.3 to 94 kPa, 86.7 to 94 kPa for CR=11, CR=12 and CR=13 respectively as Figure 4.47.

4.1.15 Comparison of Brake Specific Fuel Consumption (Bsfc) and Thermal Efficiency CR=11, 12 and 13

In Figure 4.48 it was found that the minimum bsfc 0.41 kg/kW-h at 1810 rpm for CR=12. All the bsfc values of CR=12 was found lower than other two CRs except for 2200 and 2300 rpm where the values of CR=13 was the lowest at unsmooth running condition. It was very clear from the Figure 4.48 that the brake specific fuel consumption for CR=12 was the lowest and it represented an economical engine running setup. The nature of the curve found was a concave because of heat loss in the lowest rpm and high speed friction dominated at the higher rpm range. As the minimum bsfc found was the optimum of this two considerations.

Thermal efficiency increased linearly upto its peak at 1810 rpm as the value 14.54% for CR=11, 17.30% for CR=12 and 14.54% for CR=13 and then gradually decreased for further increase in engine speed as in Figure-4.49. Through out the engine speed range the efficiency values for CR=12 were found higher than other two CRs. It was depicted that best fuel utilization possible when the engine was setup at CR=12. The thermal efficiency found was relatively less, the probable cause might be unburnt fuel in the cycle.

4.1.16 Comparison of Lubricant Oil, Coolant and Exhaust Temperature Measurement for CR=11, CR=12 and CR=13

Similar types of moderate variation in lubricant oil temperature was observed for all CRs. This was possible to work in same environment and duration of data logging. Lubricant oil temperature was observed within a range 97^0 to 103^0C, 87^0 to 104^0C, 99^0 to 110^0C for CR=11, CR=12 and CR=13 respectively. Highest temperature 110^0C was attained at 2300 rpm in CR=13 as shown in Figure 4.50 .

The coolant temperature change was observed as increased value with increase of engine speed. within a range 75^0 to 81^0C , 68^0 to 80^0C , 77^0 to 86^0C for CR=11, CR=12 and CR=13 respectively were shown in Figure 4.51. Highest temperature 86^0C was attained at 2300 rpm in CR=13 and the lowest temperature was 68^0 found out at 1450 rpm in CR=12.

In Figure 4.52 the engine exhaust gas temperature change was observed within a range 431^0 to 564^0C, 451^0 to 621^0C, 7490^0 to 640^0C for CR=11, CR=12 and CR=13 respectively. Highest temperature 640^0C was attained at 2300 rpm in CR=13 and the lowest temperature was 431^0 found out at 1450 rpm in CR=11. Exhaust temperature for CR=11 shown a lower value than other CRs throughout the rpm range. This was probably happened due to compression ratio variation. For higher compression ratio more pressure developed so more temperature attained in the cylinder hence more exhaust temperature rises occurred.

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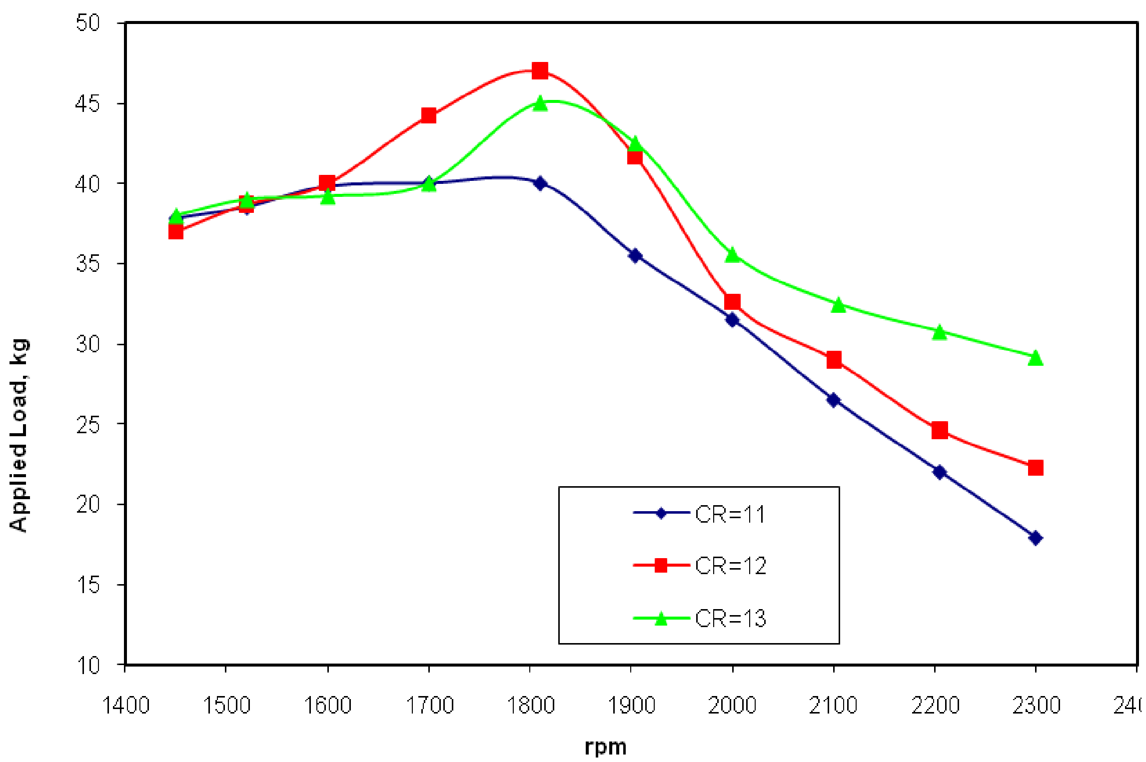

Figure 4.40: Applied load for CR=11, 12 and 13

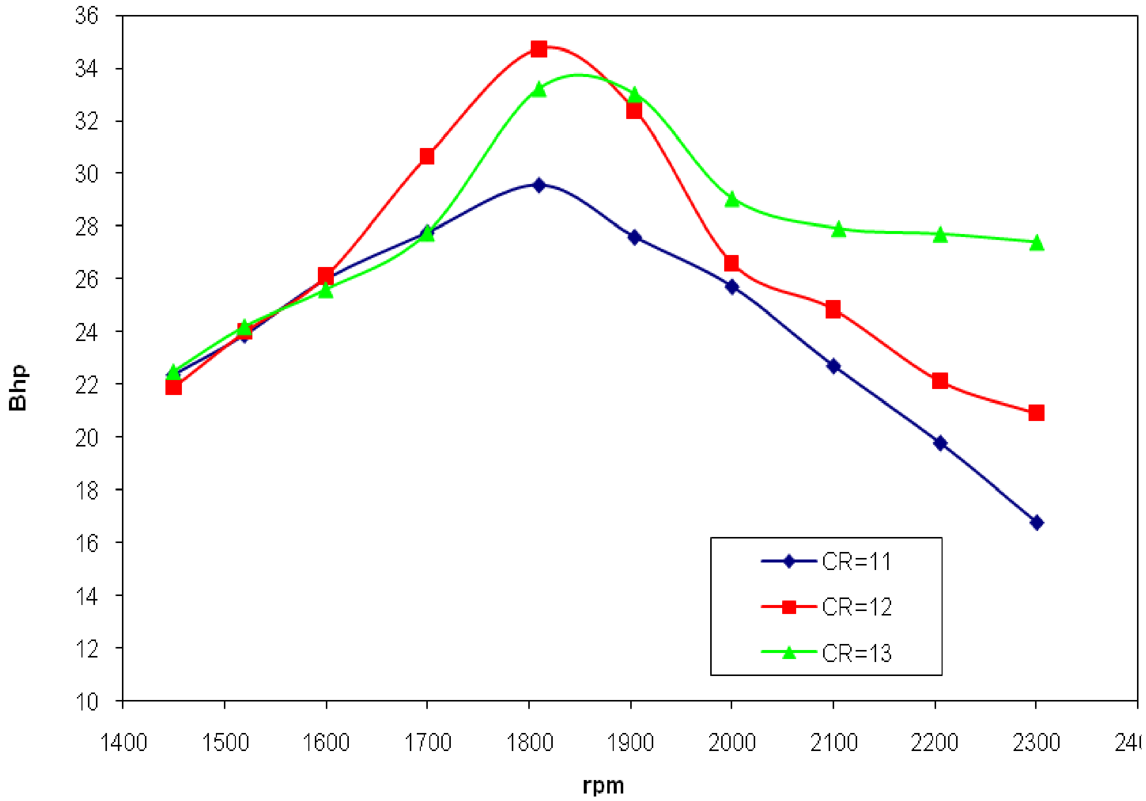

Figure 4.41: Power for CR=11,12 and 13

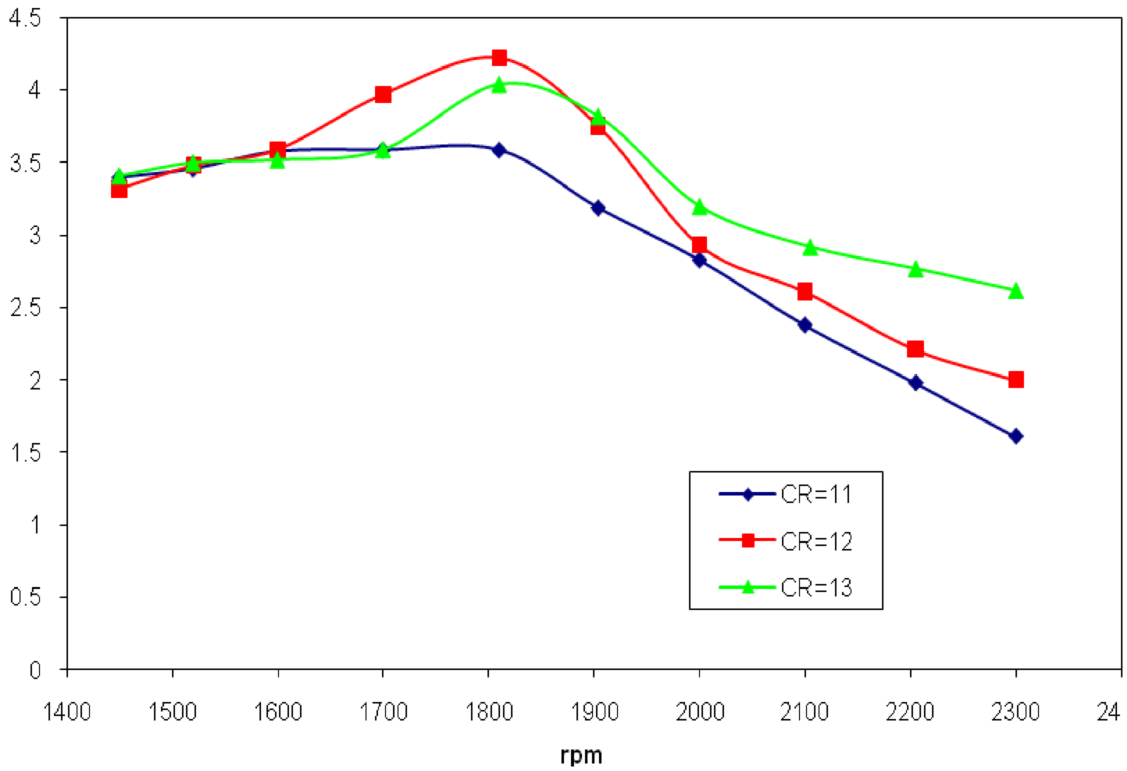

Figure 4.42: Bmep for CR=11,12 and 13

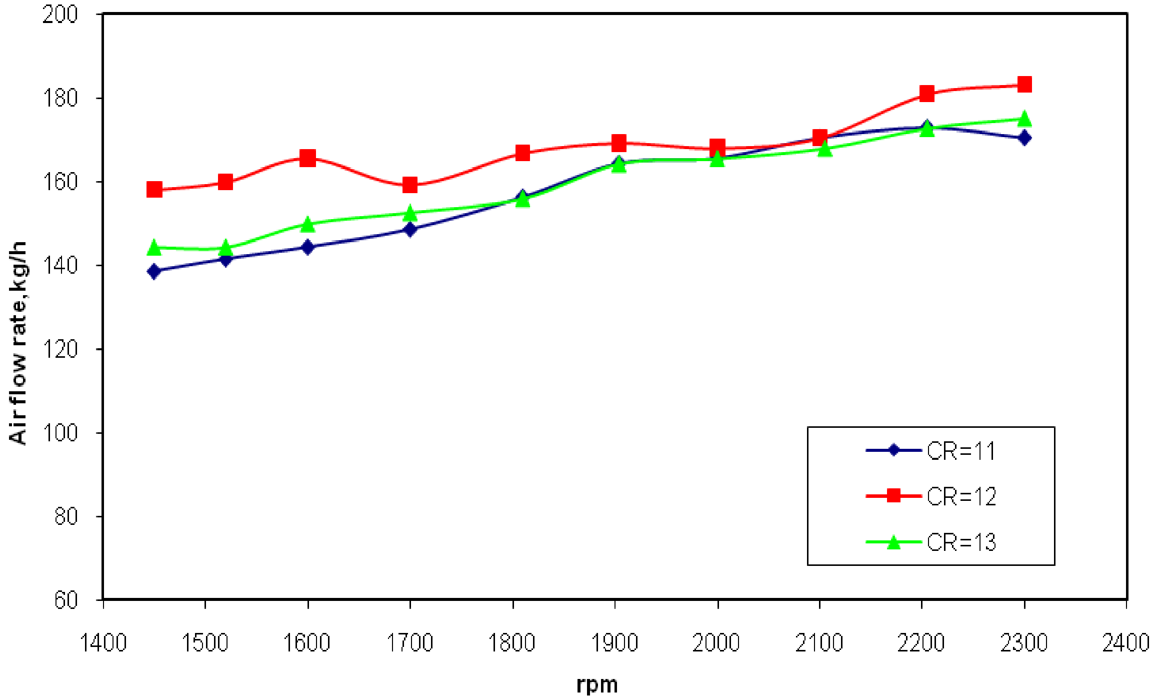

Figure 4.43: Air flow rate for CR=11,12 and 13

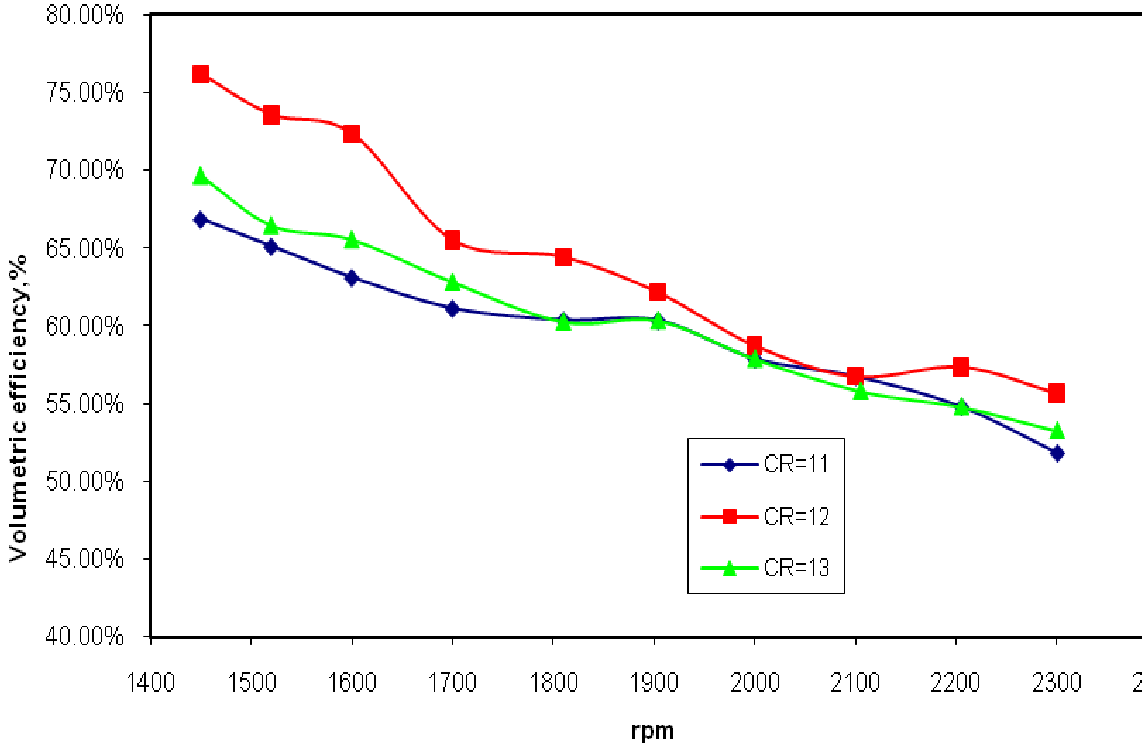

Figure 4.44: Volumetric efficiency for CR=11,12 and 13

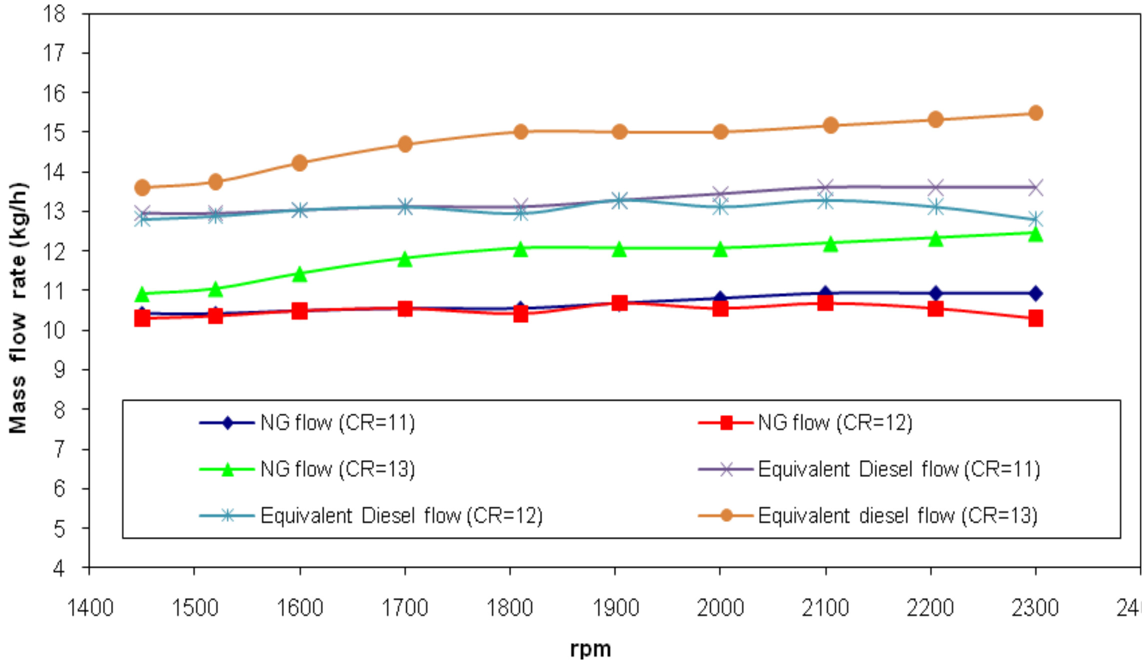

Figure 4.45: NG flow rate and equivalent diesel flow rate for CR=11, 12 and 13

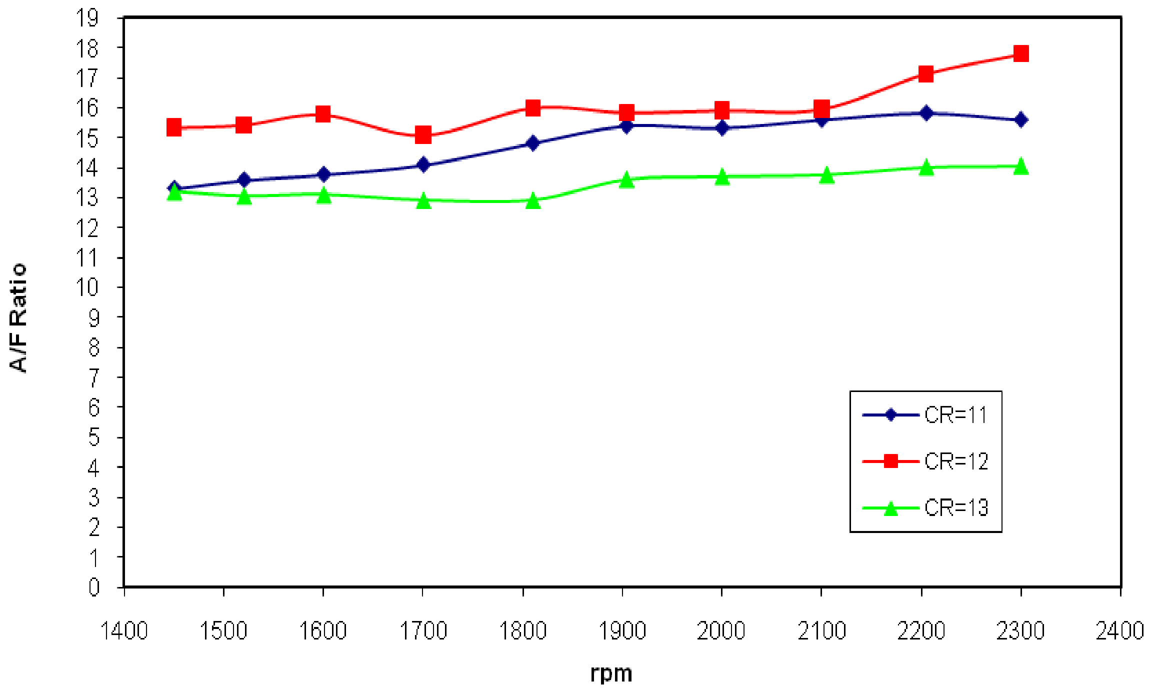

Figure 4.46: A/F ratio for CR=11, 12 and 13

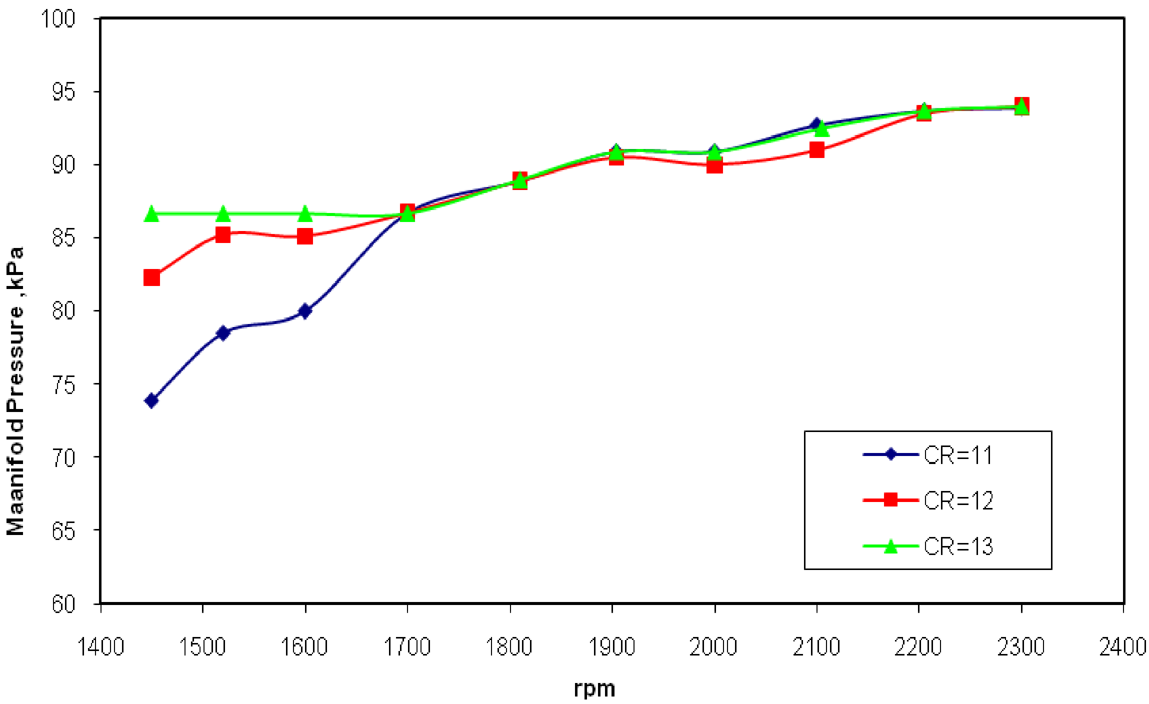

Figure 4.47: Manifold vacuum pressure for CR=11, 12 and 13

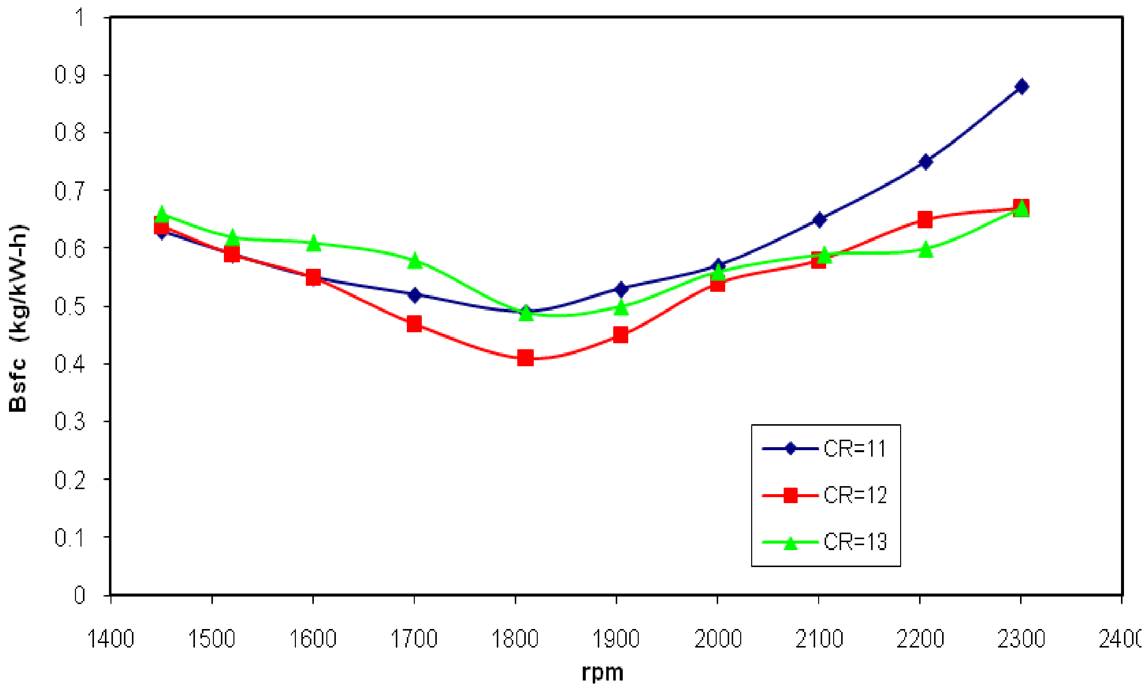

Figure 4.48: Brake specific fuel consumption (Bsfc) for CR=11, 12 and 13

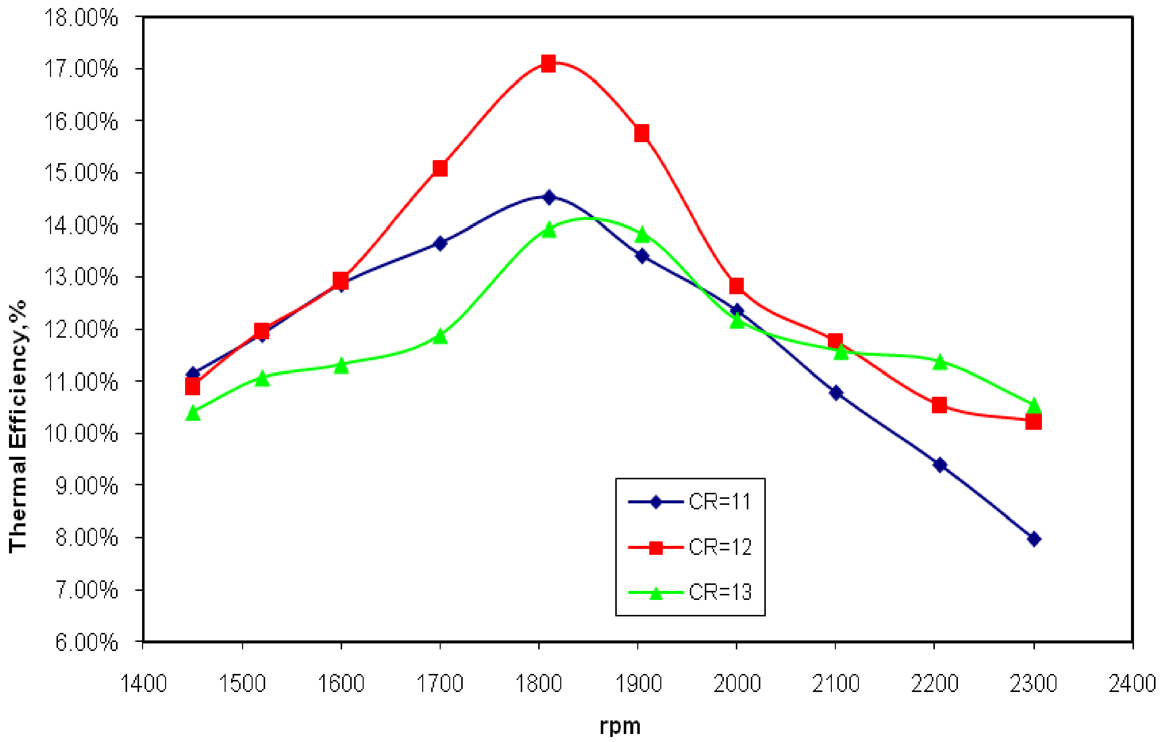

Figure 4.49: Thermal efficiency for CR=11, 12 and 13

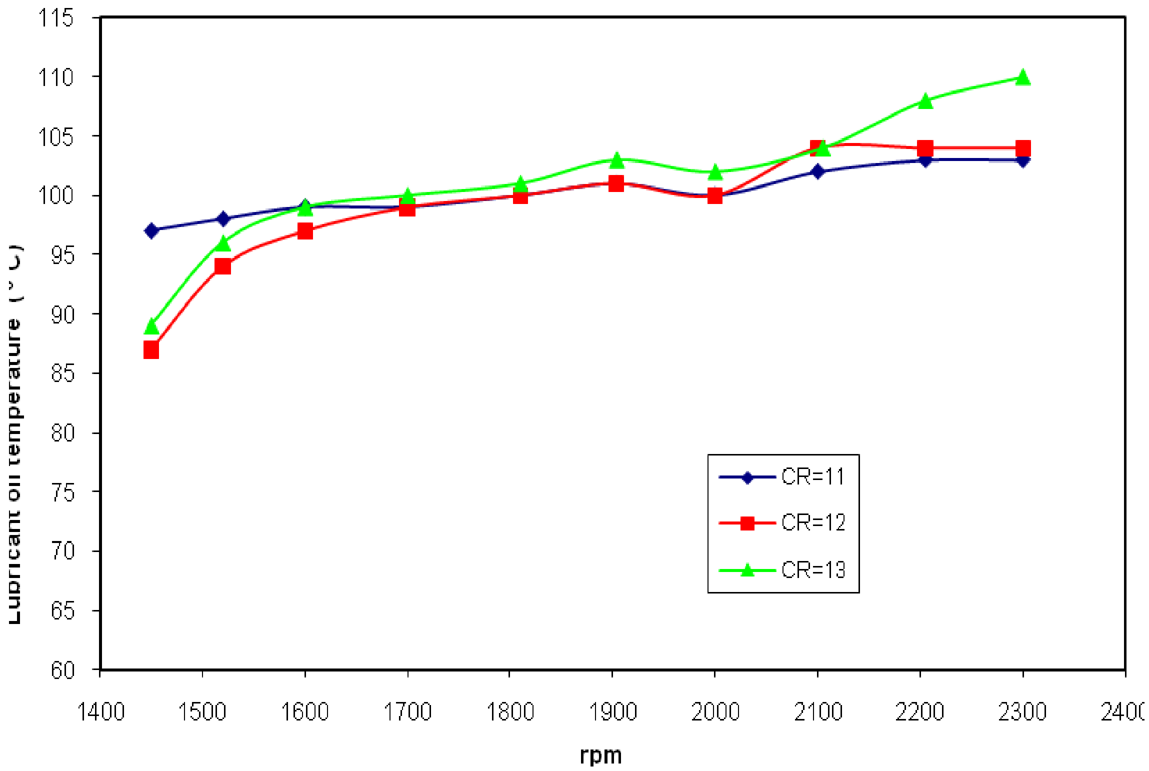

Figure 4.50: Lubricant oil temperature for CR=11,12 and 13

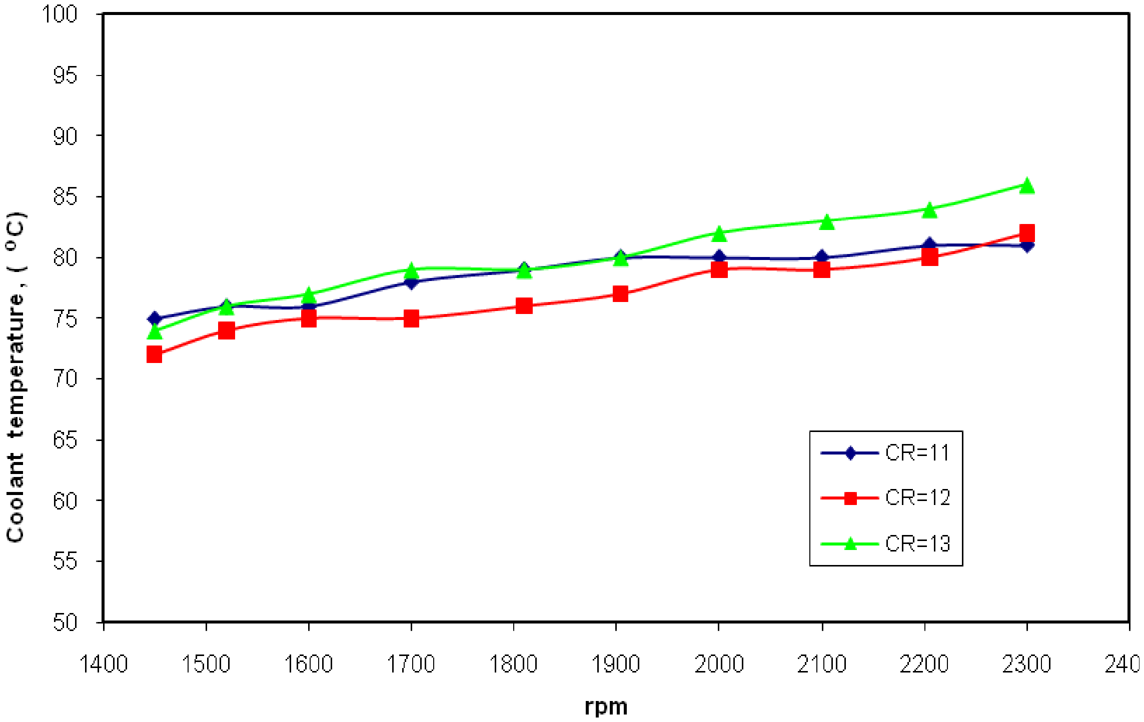

Figure 4.51: Coolant (water) temperature for CR=11,12 and 13

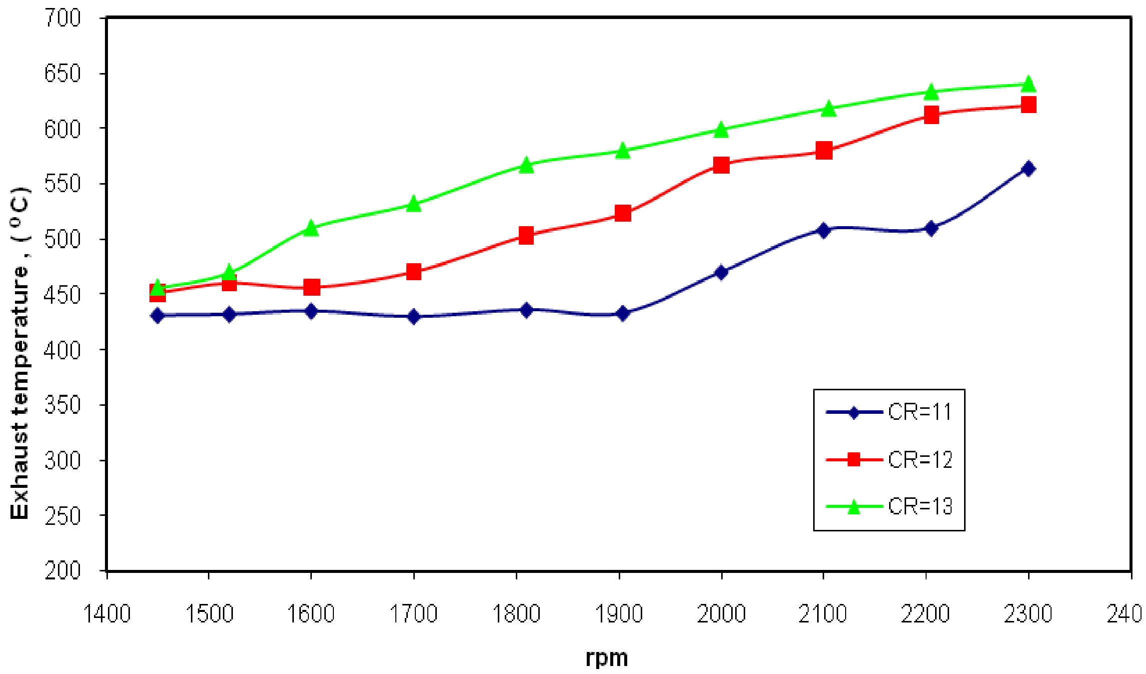

Figure 4.52: Exhaust gas temperature for CR=11, 12 and 13

4.2 Optimum Performance Setup from the Results of Performance Parameters

There was no exact performance data or yardstick available for the performance evaluation of a CNG converted diesel engine with different CRs. To find the optimum performance setup the performance parameter comparison was given due importance for this purpose. From analysis of Figure 4.1 to figure 4.52 and data table of APPENDIX D was considered to be decision yardstick .Following points were considered in deciding the optimum CR setup:

a. The rated power 80 hp of the diesel converted engine was considered 75% of maximum power (110 hp). From Figure 4.42 peak of the brake horse power was found at CR=12 out of three CRs setup. After 1900 rpm power was observed more in CR=13 at the later part of the engine operating speed. Maximum hp was found 34.72 (43.42% of the rated power) in CR=12 which was 4.26% more than CR=13 and 15% more than CR=11. Practically over 1900 rpm it was found no knock tendency in CR=12 but in CR=13 it was observed a significantly vibration and unusual running condition at higher rpm. So, to keep in a safe design CR=12 considered better power , peak load and torque.

b. Maximum Volumetric efficiency was found 76.15% in CR=12 which was 8.56% less in CR=13 and 12.2% less in CR=11 as in Figure 4.44. So, the air utilization or air flow was best suited in CR=12 than that of other two CRs.

c. In considering Bsfc, the value for CR=12 was the minimum throughout the engine operating range as shown in Figure 4.48. So, in case of fuel economy CR=12 setup was found most economical.

d. Maximum thermal efficiency was found 17.30% in CR=12 which was 16.82 % less in CR=13 and 14.9% less in CR=11 as in Figure 4.49. In CR=12 all the efficiencies shown a greater values than other CRs corresponding to a particular rpm. It was well understood from the graph of figure 4.46 that fuel utilization in CR=12 represented better result.

e. Throughout the engine operating range A/F ratio for CR=12 shown a greater values as in Figure 4.46 and closely near to stoichiometric (17.2) in some rpm. Where A/F ratios for other two CRs setup found lower values far from stoichiometric. A/ F ratios of CR=12 was found as a relatively lean mixture than other two CRs and better ignitable, whereas lowering A/ F ratio as a too rich mixture become incombustible for the engine.

f. During theoretical stress analysis of piston in APPENDIX B at different CRs, the piston design at CR=12 was found with a significantly higher amount safety factor against failure due to stress.

g. All the temperature data for lubricant oil, coolant water and exhaust gas etc. shown a moderate result for CR=12 among all three CRs as shown in

Figures 4.50, 4.51 and 4.52. The variation was not that significant for change of CR.

Basing on the above discussion it was well decided and clearly found out that the optimum performance possible in CR=12 setup engine. CR below 11 was not designed as maximum performance condition already achieved in higher CR. Where maximum power, economical fuel consumption, moderate compression pressure and sustainable and well suited temperature data made the setup of CR=12 as the better performance CR setup among other CRs.

APPENDIX D

Data Table: For Performance Parameter measurement

Data For CR=11

RPM	Load (kg)	BHP (BS)	Power (BS), kW	Torque (N-m)	Mano, H (m)	Rota%	m_a (kg/h)	m_f (kg/h)	Bsfc (NG), kg/kW-h	Eqvl Dsl (kg/h)	Esfc Dsl kg/kW-h	A/F ratio	Vol. Eff	Thermal eff	Lub oil Temp, ^0C	Coolant Temp, ^0C	Exh Temp, ^0C	Man pr, kPa	mep, (bar)
2300	17.9	16.80	12.36	51.31	0.071	83	170.38	10.93	0.88	13.60	1.13	15.59	51.80%	7.98%	103	81	564	93.9	1.61
2205	22	19.80	14.56	63.07	0.073	83	172.76	10.93	0.75	13.60	0.96	15.81	54.78%	9.40%	103	81	510	93.7	1.98
2100	26.5	22.71	16.71	75.97	0.071	83	170.38	10.93	0.65	13.60	0.84	15.59	56.73%	10.79%	102	80	508	92.7	2.38
2000	31.5	25.71	18.91	90.30	0.067	82	165.51	10.80	0.57	13.44	0.73	15.32	57.86%	12.36%	100	80	470	90.9	2.83
1904	35.5	27.59	20.29	101.77	0.066	81	164.27	10.68	0.53	13.28	0.67	15.39	60.33%	13.42%	101	80	433	90.9	3.19
1810	40	29.55	21.73	114.67	0.060	80	156.24	10.55	0.49	13.12	0.62	14.81	60.35%	14.54%	100	79	436	88.9	3.59
1700	40	27.76	20.41	114.67	0.054	80	148.59	10.55	0.52	13.12	0.66	14.09	61.11%	13.66%	99	78	430	86.7	3.59
1600	39.8	25.99	19.12	114.10	0.051	79.5	144.40	10.49	0.55	13.04	0.70	13.77	63.10%	12.87%	99	76	435	80.0	3.58
1520	38.5	23.89	17.57	110.37	0.049	79	141.54	10.42	0.59	12.96	0.76	13.58	65.11%	11.90%	98	76	432	78.5	3.46
1450	37.8	22.37	16.45	108.36	0.047	79	138.62	10.42	0.63	12.96	0.81	13.30	66.85%	11.14%	97	75	431	73.9	3.40

Data For CR=12

RPM	Load (kg)	BHP (BS)	Power (BS), kW	Torque (kg-m)	Mano, H (m)	Rota%	m_a (kg/h)	m_f (kg/h)	Bsfc (NG), kg/kW-h	Eqvl Dsl (kg/h)	Esfc Dsl kg/kW-h	A/F ratio	Vol. Eff	Thermal eff	Lub oil Temp	Coolant Temp	Exh Temp	Man pr, kPa	mep, (bar)
2300	22.3	20.93	15.40	63.93	0.082	78	183.10	10.30	0.67	12.81	0.85	17.79	55.66%	10.23%	104	82	621	94	2.00
2205	24.6	22.14	16.28	70.52	0.080	80	180.86	10.55	0.65	13.12	0.83	17.14	57.35%	10.54%	104	80	612	93.5	2.21
2100	29	24.86	18.28	83.14	0.071	81	170.38	10.68	0.58	13.28	0.75	15.96	56.73%	11.76%	104	79	580	91	2.61
2000	32.6	26.61	19.57	93.46	0.069	80	167.96	10.55	0.54	13.12	0.69	15.92	58.72%	12.83%	100	79	567	90	2.93
1904	41.7	32.41	23.84	119.54	0.070	81	169.18	10.68	0.45	13.28	0.57	15.85	62.13%	15.76%	101	77	523	90.5	3.75
1810	47	34.72	25.54	134.74	0.068	79	166.74	10.42	0.41	12.96	0.52	16.00	64.41%	17.09%	100	76	503	88.9	4.22
1700	44.2	30.67	22.56	126.71	0.062	80	159.22	10.55	0.47	13.12	0.60	15.09	65.49%	15.09%	99	75	470	86.7	3.97
1600	40	26.12	19.21	114.67	0.067	79.5	165.51	10.49	0.55	13.04	0.70	15.78	72.33%	12.93%	97	75	456	85.1	3.59
1520	38.7	24.01	17.66	110.94	0.063	78.5	159.86	10.36	0.59	12.89	0.75	15.43	73.54%	11.96%	94	74	460	85.2	3.48
1450	37	21.90	16.11	106.07	0.061	78	157.93	10.30	0.64	12.81	0.82	15.34	76.15%	10.91%	87	72	451	82.3	3.32

Data For CR=13

RPM	Load (kg)	BHP (BS)	Power (BS), kW	Torque (kg-m)	Mano, H (m)	Rota%	m_a (kg/h)	m_f (kg/h)	Bsfc (NG), kg/kW-h	Eqvl Dsl (kg/h)	Esfc Dsl kg/kW-h	A/F ratio	Vol. Eff	Thermal eff	Lub oil Temp	Coolant Temp	Exh Temp	Man pr, kPa	mep, (bar)
2300	29.2	27.41	20.16	83.71	0.075	95	175.11	12.45	0.67	15.49	0.79	14.06	53.24%	10.54%	110	86	640	94	2.62
2205	30.8	27.72	20.39	88.30	0.073	94	172.76	12.33	0.60	15.33	0.77	14.02	54.78%	11.37%	108	84	633	93.7	2.77
2105	32.5	27.92	20.54	93.17	0.069	93	167.96	12.20	0.59	15.18	0.76	13.77	55.79%	11.57%	104	83	618	92.5	2.92
2000	35.6	29.06	21.37	102.06	0.067	92	165.51	12.07	0.56	15.02	0.72	13.71	57.86%	12.17%	102	82	599	90.9	3.20
1904	42.5	33.03	24.29	121.84	0.066	92	164.27	12.07	0.50	15.02	0.63	13.61	60.33%	13.83%	103	80	580	90.9	3.82
1810	45	33.24	24.45	129.00	0.060	92	155.97	12.07	0.49	15.02	0.63	12.92	60.25%	13.92%	101	79	567	89	4.04
1700	40	27.76	20.41	114.67	0.057	90	152.66	11.82	0.58	14.70	0.74	12.92	62.79%	11.87%	100	79	532	86.7	3.59
1600	39.2	25.60	18.83	112.38	0.055	87	149.96	11.44	0.61	14.23	0.78	13.11	65.53%	11.31%	99	77	510	86.7	3.52
1520	39	24.20	17.80	111.80	0.051	84	144.40	11.06	0.62	13.75	0.79	13.06	66.43%	11.06%	96	76	470	86.7	3.50
1450	38	22.49	16.54	108.94	0.051	83	144.40	10.93	0.66	13.60	0.84	13.21	69.63%	10.40%	89	74	456	86.7	3.41

CHAPTER-5
CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

The following conclusions could be drawn from the information revealed from this study.

1. The converted diesel engine could be successfully run with natural gas but only for a limited speed range, with all three compression ratios (11,12 and 13). The initial speed range of the engine rated 800-3000 rev/min was limited to 1400-2300 rev/min only.
2. The highest net power out put from the engine was found to be in the order of 34 hp which is much less (only about 43%) of the estimated rated power of the original diesel engine. This was far less than the expected performance claimed by the Retrofitting Equipment Manufacturer. Absence of a base performance data running on diesel was a short coming for an accurate comparison.
3. The Air-Fuel ratio of operation with natural gas indicated a rich mixture. This was due to the fact that gas flow rate could not be regulated beyond a limit, with smooth engine operation. The extra fluid impedance created at the air intake for the mixing chamber with throttle valve and change in mixing turbulence patters in the combustion chamber (as piston shape is changed) may be responsible for such limitations.
4. The highest thermal efficiency achieved was about 17%, running with a compression ratio of 12. Which resulted in high brake specific fuel consumption, indicating the engine was producing less power compared to the fuel intake.
5. Though the engine with piston tops creating compression ratio of 11, 12 and 13 could successfully run in the above mentioned speed range, based on - Peak power,

Brake specific fuel consumption rate, Volumetric efficiency and the smoothness of engine operation compression ratio 12 gave better result compared to compression ratios of 11 and 13 .

5.2 Recommendations

For further study in relation to the present work the following recommendations could be suggested :

1. Base data of engine operation on diesel is essential to accurately compare the performance of the converted engine. This needs to be performed before the one-way modification of the engine.
2. If the converted engine could be run reliably on for a wider speed and power range, study with compression ratios with smaller incremental and a wider range in fractional value (eg. 11, 11.5,12, 12.5 ….15 etc) could be more informative.
3. In cases where the natural gas flow in the engine could not be regulated below a range, using a turbo charger may increase the air flow and improve the combustion and engine operation.
4. If an Emission analysis is incorporated in the investigation then more precise information regarding the combustion process could be revealed.
5. Similar investigation for finding the optimum compression ratio could be done with other alternative fuels like M85, Biogas etc.

APPENDIX C

Sample Calculation for Performance Parameter Measurement

For CR=11 AT 2300 RPM

Derating:

At 1 atmospheric pressure, 30ºC (dry bulb) and 87% relative humidity by derating from lab condition to British standard condition (BS 5514) calculated α=0.98, β=1.0

Power Measurement:

- Brake Horse Power, = = 16.47 HP (Lab)
- Brake Horse Power,
- Power (kW) =
- Torque (N-m) =

Air Flow Rate Measurement:

- Mass Flow Rate of Air:

Manometer Height , H(m)=0.0710 m

Here,

2

2

Fuel Flow Rate Measurement:

- Mass Flow Rate of Natural Gas:

From Rotameter Calibration graph of Figure- 3.46 Gas flow equation Rota % +0.5853

Here, Rotameter % = 83%

Volume gas flow,m^3

Air-fuel Ratio Measurement:

- Air- Fuel Ratio, A/F ratio =170.38/10.93=15.59

Brake Specific Fuel Consumption Measurement:

kg / kW-h

Equivalent Diesel Flow Measurement:

-

Equivalent Diesel Flow Measurement:

- = 1.13 kg / kW-hr

Volumetric Efficiency Measurement:

- = 51.80 %

When , = = = 0.0040095 m^3

2

Thermal Efficiency Measurement:

- =

Brake Mean Effective Pressure Measurement:

- = = 161 kPa = 1.6 Bar

Here , When, = Number of Crank revolution=2

= = = 4.0095 dm^3

APPENDIX D

Data Table: For Performance Parameter measurement

Data For CR=11

RPM	Load (kg)	BHP (BS)	Power (BS), kW	Torque (N-m)	Mano, H (m)	Rota%	m_a (kg/h)	m_f (kg/h)	Bsfc (NG), kg/kW-h	Eqvl Dsl (kg/h)	Esfc Dsl kg/kW-h	A/F ratio	Vol. Eff	Thermal eff	Lub oil Temp, ^0C	Coolant Temp, ^0C	Exh Temp, ^0C	Man pr, kPa	mep, (bar)
2300	17.9	16.80	12.36	51.31	0.071	83	170.38	10.93	0.88	13.60	1.13	15.59	51.80%	7.98%	103	81	564	93.9	1.61
2205	22	19.80	14.56	63.07	0.073	83	172.76	10.93	0.75	13.60	0.96	15.81	54.78%	9.40%	103	81	510	93.7	1.98
2100	26.5	22.71	16.71	75.97	0.071	83	170.38	10.93	0.65	13.60	0.84	15.59	56.73%	10.79%	102	80	508	92.7	2.38
2000	31.5	25.71	18.91	90.30	0.067	82	165.51	10.80	0.57	13.44	0.73	15.32	57.86%	12.36%	100	80	470	90.9	2.83
1904	35.5	27.59	20.29	101.77	0.066	81	164.27	10.68	0.53	13.28	0.67	15.39	60.33%	13.42%	101	80	433	90.9	3.19
1810	40	29.55	21.73	114.67	0.060	80	156.24	10.55	0.49	13.12	0.62	14.81	60.35%	14.54%	100	79	436	88.9	3.59
1700	40	27.76	20.41	114.67	0.054	80	148.59	10.55	0.52	13.12	0.66	14.09	61.11%	13.66%	99	78	430	86.7	3.59
1600	39.8	25.99	19.12	114.10	0.051	79.5	144.40	10.49	0.55	13.04	0.70	13.77	63.10%	12.87%	99	76	435	80.0	3.58
1520	38.5	23.89	17.57	110.37	0.049	79	141.54	10.42	0.59	12.96	0.76	13.58	65.11%	11.90%	98	76	432	78.5	3.46
1450	37.8	22.37	16.45	108.36	0.047	79	138.62	10.42	0.63	12.96	0.81	13.30	66.85%	11.14%	97	75	431	73.9	3.40

Data For CR=12

RPM	Load (kg)	BHP (BS)	Power (BS), kW	Torque (kg-m)	Mano, H (m)	Rota%	m_a (kg/h)	m_f (kg/h)	Bsfc (NG), kg/kW-h	Eqvl Dsl (kg/h)	Esfc Dsl kg/kW-h	A/F ratio	Vol. Eff	Thermal eff	Lub oil Temp	Coolant Temp	Exh Temp	Man pr, kPa	mep, (bar)
2300	22.3	20.93	15.40	63.93	0.082	78	183.10	10.30	0.67	12.81	0.85	17.79	55.66%	10.23%	104	82	621	94	2.00
2205	24.6	22.14	16.28	70.52	0.080	80	180.86	10.55	0.65	13.12	0.83	17.14	57.35%	10.54%	104	80	612	93.5	2.21
2100	29	24.86	18.28	83.14	0.071	81	170.38	10.68	0.58	13.28	0.75	15.96	56.73%	11.76%	104	79	580	91	2.61
2000	32.6	26.61	19.57	93.46	0.069	80	167.96	10.55	0.54	13.12	0.69	15.92	58.72%	12.83%	100	79	567	90	2.93
1904	41.7	32.41	23.84	119.54	0.070	81	169.18	10.68	0.45	13.28	0.57	15.85	62.13%	15.76%	101	77	523	90.5	3.75
1810	47	34.72	25.54	134.74	0.068	79	166.74	10.42	0.41	12.96	0.52	16.00	64.41%	17.09%	100	76	503	88.9	4.22
1700	44.2	30.67	22.56	126.71	0.062	80	159.22	10.55	0.47	13.12	0.60	15.09	65.49%	15.09%	99	75	470	86.7	3.97
1600	40	26.12	19.21	114.67	0.067	79.5	165.51	10.49	0.55	13.04	0.70	15.78	72.33%	12.93%	97	75	456	85.1	3.59
1520	38.7	24.01	17.66	110.94	0.063	78.5	159.86	10.36	0.59	12.89	0.75	15.43	73.54%	11.96%	94	74	460	85.2	3.48
1450	37	21.90	16.11	106.07	0.061	78	157.93	10.30	0.64	12.81	0.82	15.34	76.15%	10.91%	87	72	451	82.3	3.32

Data For CR=13

RPM	Load (kg)	BHP (BS)	Power (BS), kW	Torque (kg-m)	Mano, H (m)	Rota%	m_a (kg/h)	m_f (kg/h)	Bsfc (NG), kg/kW-h	Eqvl Dsl (kg/h)	Esfc Dsl kg/kW-h	A/F ratio	Vol. Eff	Thermal eff	Lub oil Temp	Coolant Temp	Exh Temp	Man pr, kPa	mep, (bar)
2300	29.2	27.41	20.16	83.71	0.075	95	175.11	12.45	0.67	15.49	0.79	14.06	53.24%	10.54%	110	86	640	94	2.62
2205	30.8	27.72	20.39	88.30	0.073	94	172.76	12.33	0.60	15.33	0.77	14.02	54.78%	11.37%	108	84	633	93.7	2.77
2105	32.5	27.92	20.54	93.17	0.069	93	167.96	12.20	0.59	15.18	0.76	13.77	55.79%	11.57%	104	83	618	92.5	2.92
2000	35.6	29.06	21.37	102.06	0.067	92	165.51	12.07	0.56	15.02	0.72	13.71	57.86%	12.17%	102	82	599	90.9	3.20
1904	42.5	33.03	24.29	121.84	0.066	92	164.27	12.07	0.50	15.02	0.63	13.61	60.33%	13.83%	103	80	580	90.9	3.82
1810	45	33.24	24.45	129.00	0.060	92	155.97	12.07	0.49	15.02	0.63	12.92	60.25%	13.92%	101	79	567	89	4.04
1700	40	27.76	20.41	114.67	0.057	90	152.66	11.82	0.58	14.70	0.74	12.92	62.79%	11.87%	100	79	532	86.7	3.59
1600	39.2	25.60	18.83	112.38	0.055	87	149.96	11.44	0.61	14.23	0.78	13.11	65.53%	11.31%	99	77	510	86.7	3.52
1520	39	24.20	17.80	111.80	0.051	84	144.40	11.06	0.62	13.75	0.79	13.06	66.43%	11.06%	96	76	470	86.7	3.50
1450	38	22.49	16.54	108.94	0.051	83	144.40	10.93	0.66	13.60	0.84	13.21	69.63%	10.40%	89	74	456	86.7	3.41

APPENDIX B

1. **Stress Analysis For Piston: Type - Diesel Engine**

For a typical diesel engine peak pressure =1000psi = 68 bar = 68 x 10^5 N/m^2. So, uniform force acting on the piston element was W= 68 X 10^2 N/m

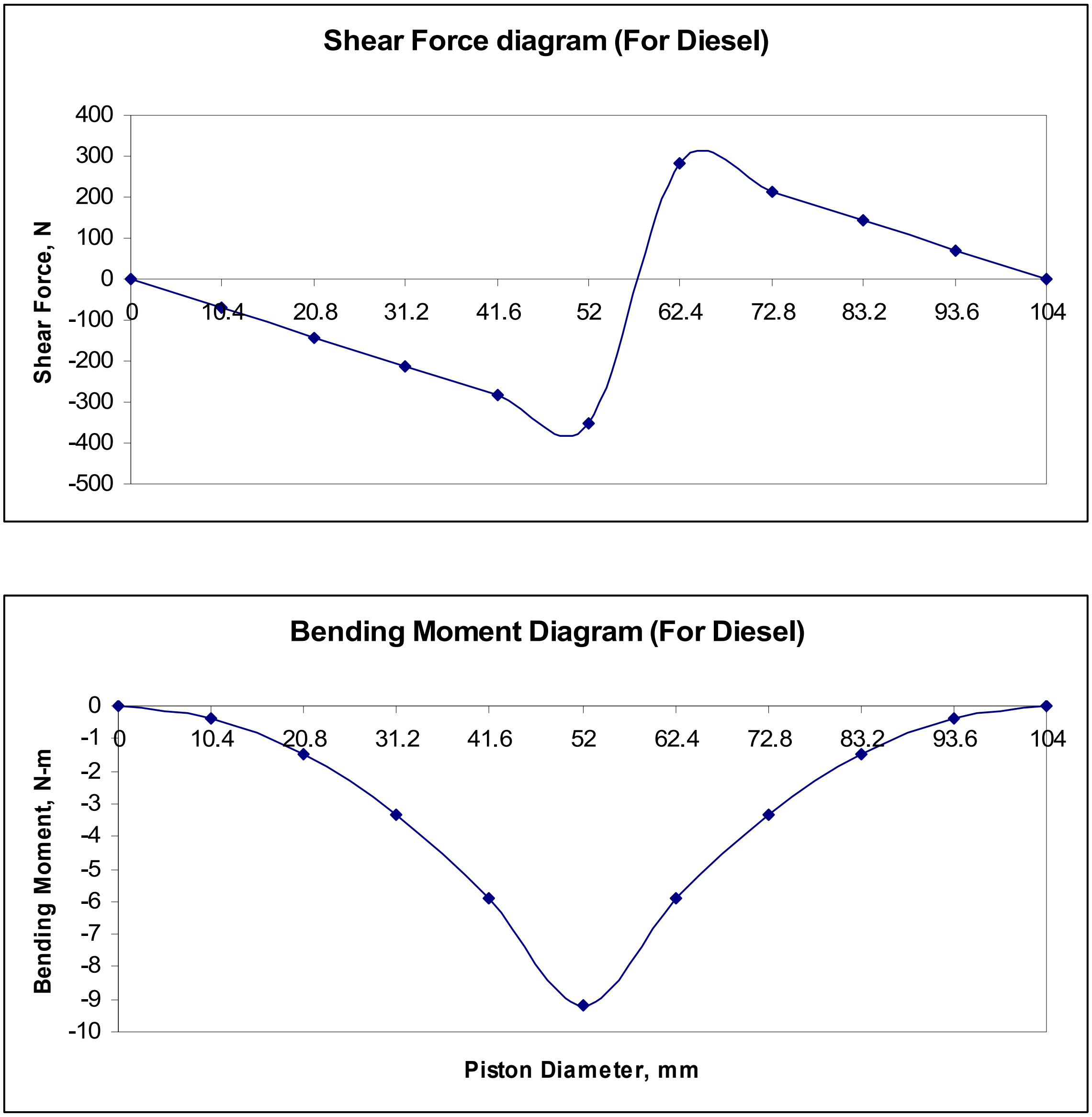

Figure C-1: Stress Analysis of Piston (Diesel engine)

X (Distance to wards x axis)	W = Uniform Force in N/m	V= Shear Force , N	R= Reaction force , N	Bending Moment (M), N-m
0	6800	0	707.2	0
10.4	6800	-70.72	707.2	-0.367744
20.8	6800	-141.44	707.2	-1.470976

31.2	6800	-212.16	707.2	-3.309696
41.6	6800	-282.88	707.2	-5.883904
52	6800	-353.6	707.2	-9.1936
62.4	6800	282.88	707.2	-5.883904
72.8	6800	212.16	707.2	-3.309696
83.2	6800	141.44	707.2	-1.470976
93.6	6800	70.72	707.2	-0.367744
104	6800	0	707.2	0

Table C-1: Stress Analysis of Piston (Data for Diesel engine)

From Figure C-1 and Table C-1 the maximum stress was found out as below:

Where , Maximum Stress = MC/I

M= 9.1936 N-m

C= 5 mm= .005 m

Effective moment of inertia,

$I = 1/12\ bh_e^3$

Interpolating for Diesel design piston , cutting diameter = 54.7 from diameter (0-104 mm) and (cutting depth 10-28 mm)

$h_e = 9.47$ mm

$I = I_{effective} = 1/12\ (.001)\ (.00947)^3 = 7.08 \times 10^{-11}\ m^4$

$= (9.1936)(0.005)/\ (7.08 \times 10^{-11}) = 649.27$ MPa

2. **Stress Analysis For Piston: Type – CNG Engine CR=11**

The peak pressure of the experimental CNG engine with CR=11 was considered as an spark ignition engine setup where the peak pressure for an SI engine nearly 30 bar=30 x $10^5\ N/m^2$. So, uniform force acting on the piston element was W= 30 X 10^2 N/m.

Figure C-2: Stress Analysis of Piston (CNG CR=11)

X (Distance to wards x axis)	W = Uniform Force in N/m	V= Shear Force , N	R= Reaction force , N	Bending Moment (M), N-m
0	3000	0	312	0
10.4	3000	-31.2	312	-0.16224
20.8	3000	-62.4	312	-0.64896
31.2	3000	-93.6	312	-1.46016
41.6	3000	-124.8	312	-2.59584

52	3000	-156	312	-4.056
62.4	3000	124.8	312	-2.59584
72.8	3000	93.6	312	-1.46016
83.2	3000	62.4	312	-0.64896
93.6	3000	31.2	312	-0.16224
104	3000	0	312	0

Table C-2: Stress Analysis of Piston (CNG CR=11)

From Figure C-2 and Table C-2 the maximum stress was found out as below:

Where , M= 4.056 N-m

C= 5 mm= .005 m

Effective moment of inertia,

$I = 1/12\ bh_e^3$

Interpolating for CR=11 piston cutting diameter = 78.6 from diameter (0-104 mm) and (cutting depth 10-28 mm)

h_e = 13.60 mm

$= 1/12\ (.001)\ (.0136)^3 = 2.09 \times 10^{-10}\ m^4$

Maximum Stress = $MC/I = (4.056)(0.005)/ (2.09 \times 10^{-10}) = 97.03$ MPa

3. **Stress Analysis For Piston: Type – CNG Engine CR=12**

From PV^k = constant equation Peak pressure for CR=12 calculated as

P_{12}=

Calculated Peak pressure of the experimental CNG engine with CR=11 was found 34 bar = $34 \times 10^5\ N/m^2$. So, uniform force acting on the piston element was W= 34 X 10^2 N/m.

Figure C-3: Stress Analysis of Piston (CNG CR=12)

X (Distance to wards x axis)	W = Uniform Force in N/m	V= Shear Force , N	R= Reaction force , N	Bending Moment (M), N-m
0	3400	0	353.6	0
10.4	3400	-35.36	353.6	-0.183872
20.8	3400	-70.72	353.6	-0.735488
31.2	3400	-106.08	353.6	-1.654848
41.6	3400	-141.44	353.6	-2.941952
52	3400	-176.8	353.6	-4.5968
62.4	3400	141.44	353.6	-2.941952
72.8	3400	106.08	353.6	-1.654848
83.2	3400	70.72	353.6	-0.735488
93.6	3400	35.36	353.6	-0.183872
104	3400	0	353.6	0

Table C-3: Stress Analysis of Piston (CNG CR=12)

From Figure C-3 and Table C-3 the maximum stress was found out as below:

Where , M= 4.5968 N-m

C= 5.25 mm= .005 m

Effective moment of inertia,

$I = 1/12\ bh_e^3$

Interpolating for CR=11 piston cutting diameter = 74.4mm from diameter (0-104 mm) and (cutting depth 10-28 mm)

h_e = 12.88 mm

$I = 1/12\ bhe^3 = 1/12\ (.001)\ (.01280)^3 = 1.75 \times 10^{-10}\ m^4$

Maximum Stress = $MC/I = (4.5968)(0.005)/\ (1.75 \times 10^{-10}) = 131.34$ MPa

4. **Stress Analysis For Piston: Type – CNG Engine CR=13**

From PV^k = constant equation Peak pressure for CR=13 calculated as

P_{13}=

Calculated Peak pressure of the experimental CNG engine with CR=13 was found 37 bar = $37 \times 10^5\ N/m^2$. So, uniform force acting on the piston element was $W= 37 \times 10^2$ N/m.

Figure C-3: Stress Analysis of Piston (CNG CR=13)

X (Distance to wards x axis)	**W = Uniform Force in N/m**	**V= Shear Force , N**	**R= Reaction force , N**	**Bending Moment (M), N-m**
0	3700	0	384.8	0
10.4	3700	-38.48	384.8	-0.200096
20.8	3700	-76.96	384.8	-0.800384
31.2	3700	-115.44	384.8	-1.800864
41.6	3700	-153.92	384.8	-3.201536
52	3700	-192.4	384.8	-5.0024
62.4	3700	153.92	384.8	-3.201536

72.8	3700	115.44	384.8	-1.800864
83.2	3700	76.96	384.8	-0.800384
93.6	3700	38.48	384.8	-0.200096
104	3700	0	384.8	0

Table C-3: Stress Analysis of Piston (CNG CR=13)

From Figure C-3 and Table C-3 the maximum stress was found out as below:

Where , M= 5.0024 N-m

C= 5 mm= 0.005 m

Effective moment of inertia,

$I = 1/12\ bh_e^3$

Interpolating for CR=11 piston cutting diameter = 70.7mm from diameter (0-104 mm) and (cutting depth 10-28 mm)

$h_e = 12.24$ mm

$I = 1/12\ bhe^3 = 1/12\ (.001)\ (.01224)^3 = 1.528 \times 10^{-10}\ m^4$

Maximum Stress = $MC/I = (5.0024)(0.005)/(1.528 \times 10^{-10}) = 163.69$ MPa

APPENDIX A

Theoretical Calculation For Modification of Piston Head

1. **For CR=19.2 (Before Modification- Diesel Engine)**

Bore, B=104 mm=0.104 m

Stroke, L = 118 mm = 0.118 m

) = 1.5 mm = 0.0015 m

= 18 mm = 0.018 m

Displaced Volume per cylinder = =

10^{-3} m^3

= 1002.39 cc

Or, 19.2-1=

Clearance Volume, = Gasket Clerance +Cutting Volume

Or, 55) +

Or, 55 - 12.74 = cutting Volume

Or, Cutting Volume= 42.33 cc

Cutting Volume = 42.33

or, 42.33

or, D= 54.7 mm

Before modification the piston for diesel engine with CR=19.2 the Diameter of the cutting volume was $D_{CR=19.2} = 54.7$ mm

2. **For CR=11**

Bore, B=104 mm=0.104 m

Stroke, L = 118 mm = 0.118 m

) = 1.5 mm = 0.0015 m

= 18 mm = 0.018 m

Displaced Volume per cylinder = =

10^{-3} m^3

= 1002.39 cc

Or, 11-1=

Clearance Volume, = Gasket Clerance +Cutting Volume

Or, 100.04) +

Or, 100.04 - 12.74 = cutting Volume

Or, Cutting Volume= 87.3 cc

Cutting Volume = 87.3

or, 87.3

or, D= 78.58 mm

To modify the piston into CR=11 the Diameter of the cutting volume will be $D_{CR=11}$=78.58 mm

1. **For CR=12**

Bore, B=104 mm=0.104 m

Stroke, L = 118 mm = 0.118 m

) = 1.5 mm = 0.0015 m

= 18 mm = 0.018 m

Displaced Volume per cylinder = =

10^{-3} m^3

= 1002.39 cc

Or, 12-1=

Clearance Volume, = Gasket Clerance +Cutting Volume

Or, 91.126) +

Or, 91.126 - 12.74 = cutting Volume

Or, Cutting Volume= 78.39 cc

Cutting Volume = 78.39

or, 78.39

or, D= 74.46 mm

To modify the piston into CR=11 the Diameter of the cutting volume will be $D_{CR=12}$=74.46 mm

1. **For CR=13**

Bore, B=104 mm=0.104 m

Stroke, L = 118 mm = 0.118 m

) = 1.5 mm = 0.0015 m

= 18 mm = 0.018 m

Displaced Volume per cylinder = =

10^{-3} m^3

= 1002.39 cc

Or, 13-1=

Clearance Volume, = Gasket Clerance +Cutting Volume

Or, 83.53) +

Or, 83.53 - 12.74 = cutting Volume

Or, Cutting Volume= 70.79 cc

Cutting Volume = 70.79

or, 70.79

or, D= 70.70 mm

To modify the piston into CR=13 the Diameter of the cutting volume will be $D_{CR=13}$=70.7 mm

APPENDIX E

Table-E-1 Typical values of some related properties of three fuels.

Property	Natural gas	Gasoline	Diesel
Formula	CH_4	C4-C12	C8-C25

H-Content %weight	25	12-15	13-16
Density kg/m3 (Ambient, 25 ^{0}C)	0.66	730	840
Vapour density, compared to air	Lighter	Heavier	Heavier
Boiling point Temp ^{0}C, atmp	-162	27-225	188-343
Latent heat of vaporization kJ/kg (Ambient, 25 ^{0}C)	-	349	233
Flash point temperature, ^{0}C	-188	-43	74
Auto Ignition (SIT)Temperature, ^{0}C	540	257	316
Octane/Cetane Number	120	90-100	40 – 55
Flammability limit, Vol%	5 – 15	1.4 - 7.6	1 – 6
Flammability limit	34.3 – 10.2	25 - 4	-
A/ F ratio, Stoichiometric (mass)	17.2	14.7	14.7
Flame propagation Speed , m/s	0.43	0.5	-
Common Compression ratio	9 – 12	9 –12	16 – 24
Lower Heating value MJ/kg MJ/liter	 50 32	 0.033 42	 44 36

Table E-2: Chemical composition of the NG obtained from different gas fields in Bangladesh

Name of gas field	**CH_4**	**Ethane**	**Propane**	**Iso-Butane**	**N-butane**	**High composition**	**N_2**	**CO_2**
Bakhrabad	94.2	3.65	0.72	0.2	0.1	0.24	0.42	0.47
Begumganj	95.46	3.19	0.64	0.17	0.04	-		0.3
Narsingdi	94.79	2.49	0.6	0.2	0.15	0.13	0.34	0.6
Feni	95.71	3.29	0.65	0.15	0.05			0.15
Habiganj	97.6	1.31	0.27	0.08	0.04	0.06	0.38	0.07
Kamta	95.36	3.57	0.47	0.09	-	-		0.51
Megna	95.15	2.83	0.6	0.16	0.09	0.07	0.37	0.53
Titas	97.33	1.72	0.35	0.08	0.05	0.06	0.3	0.11
Shahbajpur	93.68	3.94	0.71	0.2	0.07	0.04	0.46	0.9

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