STUDENTS' DECLERATION

I, hereby declare that the dissertation entitled **"PERFORMANCE AND EMISSIONS ANALYSIS OF CNG–DIESEL DUAL FUEL ON A VCR ENGINE"** being presented here in the partial fulfillment for the award of the Degree of Master of Engineering (Thermal Engineering), is an authentic record of own work carried out by me under the guidance and supervision of Prof. Amit Pal, Sr Lecturer, Department of Mechanical Engineering and Prof. Dr S. Maji, Head, Department of Mechanical Engineering, Delhi College of Engineering, Delhi.

I, further declare that the dissertation has not been submitted to any other Institute/University for the award of any degree or diploma or any other purpose whatsoever.

May, 2009

Pankaj Kumar
University Roll no. 12300
Master of Engineering
(Thermal Engineering)

CERTIFICATE

It is to certify that the dissertation entitled **"PERFORMANCE AND EMISSIONS ANALYSIS OF CNG–DIESEL DUAL FUEL ON A VCR ENGINE"** submitted by **Mr. Pankaj Kumar**, 11/THR/07, in partial fulfillment for the award of the Degree of Master of Engineering in Thermal Engineering, is an authentic record of student's own work carried out by him under our guidance and supervision.

It is also certified that this dissertation has not been submitted to any other Institute/University for the award of any degree or diploma.

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ABSTRACT

The rapidly depletion of petroleum reserves or resources have promoted research for alternative fuels for internal combustion engines. The C.I. Engine is a very important prime mover being used in the buses, trucks, locomotives, tractors, pumping sets and many other applications, small and medium electric power generation and marine propulsion etc. The running costs of C.I. Engines are much less than S.I. Engines and hence make them attractive for industrial, transport and other applications.

A dual fuel diesel engine is a diesel engine fitted with a fuel conversion kit to enable use of clean burning alternative fuel like compressed natural gas. Dual fuel engines have number of potential advantages like fuel flexibility, lower emissions, higher compression ratio, better efficiency and easy conversion of existing diesel engines without major hardware modifications. In view of energy depletion and environmental pollution, dual fuel technology has caught attention of researchers. It is ecological and efficient combustion technology.

The objective of the present major project work is to investigate the possibility of dual fuelling of Compression Ignition (C.I.) engine with Diesel and Biodiesel with Compressed Natural Gas (CNG) in order to reduce engine emissions and enhance its thermal efficiency. A Direct Ignition, single cylinder 4 stroke 3.5kW Variable Compression Ratio Diesel Engine was operated in dual fuel mode. The engine was initially started with Diesel injection and subsequently CNG was supplied with the incoming air. After self-ignition of diesel and blend of biodiesel fuel, CNG-Air mixture ignited

The investigation on three different compressions ratio show the use of CNG resulted in significant reduction of smoke opacity and Nitrogen oxide (NO_X) emissions with a slight penalty on CO and HC exhaust Emissions.

Our study also throws light on present limitations and drawbacks of dual fuel-engines and proposed methods to overcome these drawbacks. Analysis of recent research activities carried out to study effect of different parameters affecting performance of diesel - CNG and Biodiesel - CNG dual fuel engines is also summarized here. Future scope of research for these engines is also discussed.

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LIST OF SYMBOLS

η	Thermal Efficiency
η_f	Fuel Conversion Efficiency
η_v	Volumetric efficiency
V_d	Displacement Volume
Q_h	Lower Heating Value of Fuel
ρ_a	Air Density
λ	Fuel to Air Ratio
N	Engine Speed in rpm
r_c	Compression Ratio
γ	Ratio of Specific Heats
MJ	Mega Joules
K	Optional absorption coefficient of the obscuring matter per unit length
n	Number of soot particles per unit volume
B X	X% of biodiesel by volume in a mixture of biodiesel and diesel
P	Pressure
Θ	Crank angle

LIST OF ABBREVIATIONS

ACSF	American Clean Skies Foundation
BIS	Bureau of Indian Standards
BMEP	Brake Mean Effective Power
BTHE	Brake Thermal Efficiency
CNG	Compressed Natural Gas
CR	Compression Ratio
DFNG	Dual Fuel Natural Gas Vehicles
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
EIA	Energy Information Administrations
GDP	Gross Domestic Products
IMEP	Indicated Mean Effective Power
IMEP	Indicated Mean Effective Pressure
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
MeEf	Mechanical Efficiency
NCI	Navigant Consulting, Inc.
NDIR	Non-Dispersive Infra-Red Analyser
NGV	Natural Gas For Vehicles
PPM	Parts Per Million
SEC	Specific Energy Consumption
SFC	Specific Fuel Consumption
VCR	Variable Compression Ratio
VOC	Volatile Organic Compound

CHAPTER 1

INTRODUCTION

Energy has always played an important role in development of a country. It is considered as an index of economic growth and social development. Per capita energy consumption is considered as measure of prosperity of a country besides gross domestic products (GDP) and per capita income. The world has witnessed industrial revolution in the past century and it has also faced serious problems of indiscriminate utilization of the energy resources. The ideology was related to more energy consumption for higher industrial development and never considered better and efficient use of energy.

1.1 ENERGY CRISIS AND NEED FOR ALTERNATE FUELS

There are limited numbers of crude oil wells and reserves across the world. Since these natural resources are being used from several last centuries though every resource or reserve has its own span of life and it will come to its end some day in near future. So considering these facts, the research work and experimental analysis are in progress to check the suitability of the alternative fuel on internal combustion engines. The main concern is to reduce pollution level to minimum limit and that's why we are trying to use different alternative fuels such as biofuels, compressed natural gas (CNG), liquefied natural gas (LNG), liquefied petroleum gas (LPG), hydrogen, electricity, solar energy etc.

The Life of the automobile no longer seems to be under the control of its designers and manufactures. The benefits of the machine and the society's perception of the benefits of the machine have ensured its survival and popularity thus far in the history of mankind. The times are changing and man is coming to new crossroads that must be critically evaluated in light of new concerns and problems that were not pressing at the birth of the automobile more than a century ago.

Industrial development has brought the western world to a new frontier that is no longer about living by the sweat of our brow and by working the land. The industrial revolution and the modernization of the world has enabled us to remove ourselves from nature and to allow us to look the other way as our inventions and creations dirty our nest and foul our water. However, the time has come where the infinite has to be realized to actually be finite. Nature can not take all of our garbage any more we are approaching the pollution saturation point in which all the actions that we perform should be analyzed to take into account that we are not alone and that not only are we harming the creatures around us, we are harming ourselves. [1]

1.2 ALTERNATIVE FUEL IMPUTES

There are some very important parameters which should be considered before adaptation of an alternative fuel in an existing engine. These include:

- ❖ No or minimum modification required in design of engine,
- ❖ Use of same storage and transportation infrastructure,
- ❖ Biodegradable and non-toxic assuring safe handling and transportation,
- ❖ Capability of being produced locally and low investment cost. [2, 3].

The economic benefits of the fuels like vegetable oils, compressed natural gas, ethanol, and methanol etc. compared to the traditional petroleum resources are marginal but the environmental benefits are enormous, thus public policies need to be revised to encourage the development of these resources for which:

- land for production need to be explored
- an efficient extraction of oil from oil seeds and transesterification plant would be required
- distribution and storage facilities constructed
- monitoring of major users for detection of problems
- large scale use are needed before the technology can be recommended for general use
- the magnitude of our energy needs provides an inexhaustible market of our total agriculture production capacity at the highest possible level
- farm back to work providing for our food needs and also growing crops and livestock for energy. Energy is the only crop that we could never grow in surplus [4].

1.3 DIESEL–CNG AND BIODIESEL–CNG AS A DUAL FUEL

The dual fuel engine is a diesel engine that operates on gaseous fuels while maintaining some liquid fuel injection to provide a deliberate source for ignition. Such a system attempts usually to minimize the use of the diesel fuel by its replacement with various gaseous fuels and their mixtures while maintaining satisfactory engine performance. There are some problems associated with the conversion of a conventional diesel engine to dual fuel operation. At light load, the dual fuel engine tends to exhibit inferior fuel utilization and power production efficiencies with higher unburned gaseous fuel and carbon monoxide emissions relative to the corresponding diesel performance. Operation at light load is also associated with a greater degree of cyclic variations in performance parameters such as peak cylinder pressure, torque and ignition delay which have narrowed the effective working range for dual fuel applications in the past[5]. These trends arise mainly as a result of the poor flame propagation characteristics within the very lean gaseous fuel–air mixtures and originating from the various ignition centers of the pilot.

- In Diesel-CNG and Biodiesel-CNG dual fuel engine mixture of natural gas and air is induced in engine cylinder during the suction stroke and compressed during compression stroke.
- This air fuel mixture is ignited by injecting small quantity of diesel or biodiesel called as pilot injection in cylinder at the end of compression stroke.
- This pilot fuel ignites due to heat of compression just like diesel engine.
- Burning of diesel pilot fuel further ignites and burns CNG in the cylinder and power is produced.
- CNG has benefits like lower exhaust emissions, high octane number and wide flammability range, capability to form homogeneous air fuel mixture, low photochemical reactivity and lower toxicity of exhaust gases as compared to pure diesel.

In dedicated CNG engines existing diesel engines should be converted to spark ignition engine and CNG is ignited by spark ignition. This requires considerable changes in diesel engine like change in compression ratio, replacing injectors by spark plugs. If dual fuel technology is adopted these major changes can be avoided and existing diesel engines can be easily converted to dual fuel operation by using simple conversion system.

1.3.1 Current Dual Fuel Technology

A measured quantity of natural gas is mixed with the air just before it enters the cylinder and compressed to the same levels as the diesel engine to maintain efficiency. The natural gas mixture does not ignite spontaneously under compression, so the dual-fuel engine uses a small injection of diesel fuel, around 10% of the total energy of the fuel, to ignite the main charge of gas and air. This small 'pilot' injection acts like a multitude of microscopic spark-plugs, setting off clean and efficient combustion of the lean gas-air mixture. Natural gas burns cleaner than diesel due to its inherently low carbon content.

1.3.2 Dual-Fuel Engine Technology Facts:

- Engine runs on natural gas with diesel pilot ignition
- At full power diesel fuel is 10% of total fuel and at normal operation it is up to 85% natural gas substitution
- Runs on 100% diesel fuel until engine coolant is at operating temperature
- Under normal operation, can run solely on diesel at light loads
- Under normal operation, can run solely on diesel at full power
- Uses a standard electronically-controlled diesel engine
- Maintains electronic control of both gas and diesel injection
- Base diesel ECU is retained
- Dual-Fuel ECU controls gas operation and modifies diesel demand for the diesel engine ECU
- Automatically reverts to diesel only when gas supply is out of acceptable parameters
- Recent introduction of a combustion knock sensor protects the engine from variable gas quality

- The operating system in the technology automatically switches the engine from diesel to natural gas when the engine reaches optimal load, making it particularly suited to long haul heavy trucks. The dual-fuel can be turned off at any time, enabling the engine to operate totally on diesel. This switch from diesel or to dual fuel is almost imperceptible, even while the vehicle is moving.

1.3.3 Interfaced Dual-Fuel Technology

In addition to providing a Retro-Fit Dual-Fuel technology, Clean Air Power has been developing a manufacturer branded and supported Interfaced product. A contemporary integrated system will deliver at least 70% overall gas substitution, giving fuel savings of 9 paise/km per vehicle per year, with CO_2 savings of 26 tones yearly. Clean Air Power's integrated C_{12} product is already certified to Euro IV standards. .[6]

The dual-fuel electronic control unit (ECU) can be interfaced with original equipment manufacturers (OEMs) ECU. This enables the dual-fuel ECU to control the engine ECU, ensuring optimum conditions for dual-fuel combustion. This interface has been successfully achieved on DAF and Mercedes engines and will form the core of any Dual-Fuel application to an engine with OEM cooperation.

The 'Genesis' retro-fit product uses a standard dual-fuel ECU that interfaces with the inputs and outputs of the engine ECU, rather than with its software, enabling it to be installed independently of OEM participation.[6]

1.4 MECHANISM OF EMISSIONS FORMATION IN IC ENGINES

Introduction

Internal combustion engines have been subject to emission control techniques since the passage of the Clean Air Act in1966. Successive amendments have tightened the allowable levels of emissions emanating from new vehicles and were later extended to cover particulate emissions from diesel engines. The trend towards lower and lower allowable emissions levels appears to be continuing with particular emphasis on diesels.

This document aims to enlighten the reader as to the primary formation processes occurring within a typical compression ignition engine (also known as a diesel engine after its inventor, Rudolf Diesel). The main pollutants emitted from the exhaust of a typical diesel engine include hydrocarbons (HC), oxides of nitrogen (NO_x) and particulate matter (PM). Carbon monoxide (CO) is touched on lightly for reasons explained later.

Hydrocarbons

Hydrocarbons describe the large family of emissions composed of hydrogen and carbon in a variety of chemical bonds. These range from simple non reactive methane molecules (CH_4) to more complex and active chemical chains like benzene (C_6H_6) and butene (C_4H_8). Hydrocarbons (HC) are formed when fuel is not adequately oxidized, or burned. In diesels, incomplete combustion of the fuel results in soot formation, visible as large clouds of black smoke, containing up to 0.5% of the fuel mass. During startup, and subsequent misfire, unburnt fuel may condense and produce clouds of white smoke [23]. Overall, the level of HC emitted as a pollutant is strongly dependent upon the fuel distribution and resulting combustion inside the cylinder.

Hydrocarbon emissions can be split into two major groups: non-reactive and reactive. This grouping stems from the chemical reactivity of the molecules with respect to the formation of smog. Hydrocarbons play a secondary role in ozone formation by accelerating the formation of NO_2, which reacts with O_2 to produce ozone, the basic component of smog. The reactive components include all hydrocarbon chains except methane, which is highly stable and also gives rise to the term "non-methane organic gases" which include all non-methane hydrocarbons and oxygenates. In addition to participating in smog formation, many oxygenates are also irritants to the eyes and lungs. Further many of these molecular chains are not found in the fuel prior to combustion, demonstrating the complex chemical kinetics that occurs inside a combustion chamber.

One of the factors in the production of hydrocarbon emissions is the quenching of the flame front as it approaches the relatively colder surfaces of the cylinder walls and piston. These surfaces absorb heat energy to such an extent that combustion cannot sustain itself within the fuel-air mixture. Crevices and gaps such as those seen between the cylinder walls and piston dominate this mechanism as hydrocarbons quenched at the walls are readily oxidized later in the cycle[7]. Cold starting of an engine demonstrates this problem drastically as the relatively cold surfaces of the combustion chamber cause excessive amounts of black smoke. One source unique to direct injection diesels comes from the fuel injector tips. Fuel leftover in the nozzle tips after injection has ceased slowly evaporates and seeps slowly into the combustion chamber where it may or may not be oxidized. The major source however, contributing to HC emissions are the localized rich or lean conditions found within the combustion zones. As the spray is injected, the air mixes with the outer edges of the fuel producing very lean zones that oxidize in a non self-sustaining manner and seldom to completion. As the spray continues to mix with the air, these lean zones expand outward leaving more combustible mixtures behind in the center of the chamber. The amount of HC left unburned is then a function of the mixing rate (or turbulent swirl) of the engine, the cylinder conditions and because of its association of the prior two, the ignition delay. According to Heywood, there is a non-linear relationship between the ignition delay and the amount of HC produced. Leanness, however, is not the sole condition aiding hydrocarbon emissions. Overly rich mixtures will also result in incomplete combustion, a condition that can be caused by insufficient mixing of the oxygen in the air with the fuel spray. This is especially the case just after the injector nozzles have ceased spraying as the pressure forcing the fuel out has dropped and the remaining fuel enters the combustion chamber at low speed. The low velocity of the fuel causes undermixing of the fuel-air to occur, which of course generates an overly rich region. Desorption of HC from the layer of oil that coats the cylinder walls adds to the overall level found in exhaust gas and is controlled by the characteristics of the fuel being used and its ability to be absorbed by the oil layer.

Engine operating conditions play a role in HC emissions mainly as a function of the load on the engine. Idle and light load conditions generate overall fuel to air ratios of around 100:1 and this causes an excess of over lean regions in the injected fuel spray. Consequently, light load and idle produce substantially more HC emissions than full load [23]. On the other end of the spectrum overfueling of the engine at high loads will produce excessive HC through insufficient oxygen supplies.

The timing of the injection produces an effect on HC as well. If the timing is advanced away from top dead center and away from the optimum timing, the ignition delay lengthens, allowing a higher percentage of the total fuel injected to mix with the air and impinge on the cylinder walls. This also produces more areas of lean mixtures, hindering efficient combustion and raising the amount of unburned HC[42]. On the other hand, retarding the advance produces overly rich regions with insufficient time to combust with the end result being visible smoke. In a similar vein, lengthening the physical time that the injectors are open and spraying fuel into the cylinders reduces HC at low load, but at high load leads to an increase in smoke and particulates [8].

Particulate Matter

The distinction between particulate matter and hydrocarbon emissions is a matter of condensation temperature. Generally, heated probes in a dilution tunnel are maintained at 190°C and any hydrocarbon chain that condenses is filtered out and lumped with the soot and ash accumulations as particulate matter, which is gathered by filtering the diluted exhaust stream at a constant 52°C. Particulate formation is a major concern in diesel engine combustion and consists mainly of carbonaceous conglomerations. These clumps are formed mostly through incomplete combustion of fuel with small contributions from the lubricating oil [7]. As the fuel in the advancing flame plume combusts, pyrolytic reactions crack the hydrocarbons that have yet to pass through the flame. As these reactions occur, particulate masses form and are passed through the flame. A side effect of this process is the radiation heat transfer that is given off by the heated particulates which increases the pyrolytic reactions in the unburned fuel. If the fuel mixing is poor within the cylinder, large quantities of particulates can form [2]. Typically, above temperatures of 500 °C, the particles are composed solely of clusters of carbon, while at temperatures below this; higher molecular weight hydrocarbons condense onto the clumps. As the particulates travel through the flame front and into the more heavily oxygen populated areas, the clumps tend to oxidize and for this reason concentrations are reduced in the leaner regions of combustion.

Oxides of Nitrogen

The main source of nitrogen in the chemical formation of NO_X is atmospheric, and a very small portion is caused by nitrogen compounds found in some fuels. The fuel source is more pronounced in diesel combustion, however. The basic kinetic equations for the transformation of atmospheric nitrogen are known as the Zeldovich mechanism. These two equations have been rigorously tested and a third equation has been generally accepted to contribute significantly and as such the three are sometimes referred to as the 'extended' Zeldovich mechanism.

$$O + N_2 \Leftrightarrow NO + N - 75\frac{kcal}{mol}$$

$$N + O_2 \Leftrightarrow NO + O + 31.8\frac{kcal}{mol}$$

$$N + OH \Leftrightarrow NO + H + 39.4\frac{kcal}{mol}$$

The third equation is usually found in rich mixtures where OH is readily available. As the burned gas region behind the flame front absorbs energy from the combusting mixture, the pressure and temperature both rise significantly. It is this region's high temperature which spurs the formation of nitric oxide (NO) and in most cases, the flame front production is simply ignored. The flame front does, however, play two significant roles by providing the thermal energy required to dissociate the N_2 into N radicals and by providing the reactions which lead to the NO producing chains. The main controlling factors are the amounts of oxygen and nitrogen radicals available and the temperature of the mixture. The temperature of the mixture is especially important as there is a non-linear relation between it and the rate of formation of NO. Due to this, the formation kinetics of NO 'freeze' below a given temperature inside the cylinder as the piston continues downward on the expansion stroke. It is also this kinetic freeze which causes diesels to produce a significant amount of nitrogen dioxide (NO_2). At light load, there is a significantly large portion of the cylinder charge containing unused and relatively cool amounts of air mixing with the burning fuel. NO_2 is primarily formed in the flame front and can only be conserved by quenching, a process made easy by the generous amounts of cooler air at light load. For this reason, concentrations of NO_2 can approach 10-30% of the overall oxides of nitrogen in a diesel at light load [7]. Speed also plays a small role in NO_2 formation as lower speeds increase the residence time of NO with O_2 [8].

Fuel-air ratio also plays a significant role in the production of NO_X, with the peak formation rate occurring at a point just lean of stoichiometric. This peak can be explained by the still fairly high combustion temperatures coinciding with the high availability of nitrogen and oxygen, which is why the peak does not occur at a point slightly rich of stoichiometric where combustion temperatures are highest. As an engine strays farther and farther into the lean region, the combustion temperatures plummet and this effect dominates the kinetics of NO_X formation. However, diesels operate primarily in the lean region (when overall fuel to air ratios are considered) where high gas availability dominates.

Carbon Monoxide

Since diesel engines operate at such lean overall air to fuel ratios, and since carbon monoxide formation is generally a fuel rich combustion phenomenon, this pollutant is not significant in diesel engine exhaust. Although there are regions of very rich combustion that do produce detectable quantities of carbon monoxide, the gas is oxidized later in the cycle and reduced to negligible amounts in the exhaust stream.

1.5 GROWTH OF TRANSPORTATION ACTIVITIES

The population of the world is increasing at an alarming rate of nearly 3 percent every year. India is the world's seventh largest country (as per area) and in terms of population it stands second to China. As the seventh most populous metropolis in the world, Delhi's population is 13.8 million, with an annual growth rate of 3.8% per annum, [7]. By the end of 2050 the population of India's second largest city (Delhi) is expected to touch a level of 1.9 times of what it is today, thus almost doubling this figure of the year 2006, [8]. In recent years per-capita income in Delhi has grown at roughly 5% per annum and at 3000$ (USD), it is twice the national average.

As the economies grow, transportation activities also tend to expand. Such as populations of megalopolis cities, motor vehicles, motor fuel consumption and air pollution all have increased. The global vehicle stock is expected to be approximately double from about 640 million at present to about 1 billion vehicles by the year 2025. It is envisioned that the transportation sector will show as large as about 7% global increase during next decade. The air quality crisis in most of the cities is often attributed in large measure (40–80%) to vehicular emissions.[9]

1.6 GROWTH TREND OF MOTOR VEHICLES

The vehicle population in India is growing at an exponential rate and is fast approaching the 50 million mark [10]. Delhi faces the same transportation, economic and environmental challenges of other megacities. Population, motor vehicles, pollution, and traffic congestion are all increasing. In the past 30 years, its population more than tripled and vehicles increased almost fifteen fold. By 2000, Delhi had about 2.6 million motor vehicles - 200 for every 1,000 inhabitants, a rate far higher than most cities with similar incomes. Most of these vehicles are small, inexpensive motorcycles and scooters, rather than automobiles. This proliferation of vehicles in a relatively poor city is indicative of the strong desire for personal transport–a phenomenon observed virtually everywhere. Delhi is an emerging example of how this desire can now be met with relatively low incomes. Delhi is expected to continue growing at a rapid rate into the foreseeable future. Its population is expected to surpass 22 million by 2020 and motor vehicles, including cars, trucks, and motorized two- and three-wheelers, are expected to grow at an even faster rate. The domestic auto industry is predicting car sale increase of ten percent per year. With an extensive network of roads and increasing income, there is every reason to expect vehicle sales and use to continue on a sharp upward trajectory. That is why emission standards & fuel quality specifications are being tightened progressively and significant improvements in vehicle emissions have been achieved.

The auto-fuel policy in India has prepared the road map for emission norms in the coming years. The entire vehicle fleet, motorized and non-motorized is growing rapidly. From 1975 to 1998, the car population increased from about 68,000 to almost 800,000, and the motorized two wheelers from about 100,000 to almost 2 million. With continued income growth, the motor vehicle population is expected to continue expanding at a high rate (see Table 1.1). The number of bicycles and cycle rickshaws is also very large and increasing, though the number is unknown since many owners do not comply with the requirement for annual registration. It is estimated by the authors that as many as 300,000 cycle rickshaws currently travel on Delhi roads.

Table 1.1: Motor vehicles in use in Delhi, 1990-2020 (thousands)

Year	Scooters and motorcycles	Cars/ jeeps	Auto rickshaws	Taxis	Buses	Freight	All motor vehicles
1971	93	57	10	4	3	14	180
1980	334	117	20	6	8	36	521
1990	1077	327	45	5	11	82	1547
2000	1568	852	45	8	18	94	2584
2010	2958	1472	103	14	39	223	4809
2020	6849	2760	209	28	73	420	10336

Data Source: Transport department, Government of National Capital Territory of Delhi

Buses form the backbone of the transport system in Delhi. As a generalization, buses are the most economically and environmentally efficient means of providing transport services to most people. In Delhi, buses constitute less than one percent of the vehicle fleet, but serve about half of all travel demand. Since 1992, Delhi has turned increasingly to the private sector to help expand and improve bus service. This decision was a response to the widely acknowledged shortcomings of public bus service, including escalating costs, poor maintenance, high labor costs, an aging bus fleet, and erratic service. Bus service was expanded in 1996 by adding more buses, with buses per route increasing from 0.8 to 1.7. The regular fixed-route bus system now comprises about 4,000 privately operated buses and 3,760 publicly operated buses. It is complemented by 5,000 private charter buses that provide point-to-point service during peak hours to subscribers who pay a monthly fee for a guaranteed seat.

Despite these expanded transit services, at both the lower and upper end of the market, overall transit use continues to lose market share. Buses accounted for 57 percent of total passenger kilometers in 1990, dropping to about 49 percent in 2000 (see Table1.2). This drop is largely due to increased use of motorized personal vehicles in upper income households, mostly two-wheelers but also cars and the expanding population of very poor immigrants who cannot afford to ride the bus.

Table 1.2: Historical and forecasted travel demand in Delhi, 1990-2020, billion passenger kilometers (Motorized Travel Only)

Year	Two Wheelers	Cars & jeeps	Auto rickshaws	Taxis	Buses	Rail Transit	Total
1990	8.0 (17)	8.6 (18)	3.4 (7)	0.3 (<1)	27.2 (57)	0.0 (-)	47.5 (100)
2000	14.8 (16)	29.0 (31)	3.5 (4)	0.4 (<1)	46.8 (49)	0.0 (-)	94.4 (100)
2010	33.8 (15)	61.6 (28)	7.6 (3)	0.6 (<1)	105.0 (48)	10.4 (5)	219.1 (100)
2020	102.6 (20)	153.3 (30)	15.8 (3)	1.3 (<1)	220.0 (44)	10.4 (2)	503.4 (100)

Note: Figures in parentheses are percentages.,Source: [Ref. 10]

Rapid Transit

- To reduce traffic congestion and air pollution, the Delhi and national governments are building an integrated multi-modal rapid transit system known as MRTS. This system includes 198.5 kilometers of electrified rail lines and is estimated to cost 150 billion Indian Rupees (US $4 billion) at 1996 prices. The first phase calls for 55 kilometers of rail and bus ways, one-fifth of it have been completed till 2005, with projected passenger loads of 3 million passengers per day. The plan includes 115 new feeder bus routes on existing roads. The second phase is scheduled for completion in 2021 and will carry 22 million passengers. In 2004 Mass Rapid Transit System known as METRO was also introduced in account of to reduce the pollution.
- Since long time City buses are providing transport to majority of peoples (nearly 57%) But they haven’t been given any preference like in terms of separate lane for fast movement. Now in Delhi, Bus Rapid Transit is introduced as a pilot project on some specific routes. This system reduces the time taken in travelling and reduces unnecessary traffic jam problem and hence may encourage the people to use the public transport system.

CHAPTER 2

LITERATURE REVIEW

In literature review I have gone through a number of research papers studies on dual fueling of C.I. engines and experimental analysis on suitability of CNG in these engines. The results and analysis on the dual fueling of C.I. engines is analysed.

2.1 STUDIES ON CNG

2.1.1 Introduction

Natural gas was first used as fuel in China during the Shu Han dynasty in AD 221-263. The gas was obtained from shallow wells near seepages and was distributed locally through piping made of hollowed-out bamboos. Since then, there are no records on the usage of natural gas until the early 17th century in Northern Italy, where it was used as a fuel to provide lighting and heating. As time moved on, the usage of natural gas spread to North America, Canada, New Zealand and Europe.

The usage was limited to domestic and industry heating. When the world turned into the 20th century, the usage of natural gas expanded to most part of Western Europe and USA. Exploration for the natural gas source was more active after the post-war years. It became a commercial item in the form of liquefied natural gas [11] for exports and imports. The gas fields or the natural gas resources are mainly found in Asia and Middle East countries. These include Malaysia, Brunei, Algeria, Libya, Saudi Arabia, Kuwait and Iran. By 1980s, these countries became the main exporters of natural gas.

Lapin, et al. (2002) conducted a study to examine the mutagenic effects of exhaust emissions from a CNG fueled refuse hauler without any emission control device. In this study, diluted and cooled PM samples were collected isokinetically on a 20 in by 20 in polytetrafluoroethylene glass fiber filters. The PM samples collected on the filter were solvent extracted, and subjected to ames bioassay. Results from the dose response assay were positive indicating mutagenic activity. Thus, the need for retrofitting the existing

CNG-fueled vehicles with a reliable and durable exhaust after treatment device, and improving the engine technology to address the increase in nanoparticles and genotoxic exhaust emission was substantiated. [12]

A year long study by Lev-On, et al. (2001) that focused on chemically characterizing the exhaust emissions from trucks and buses running on different test fuels, with and without exhaust after treatment devices showed that the introduction of ultra low sulfur diesel (ULSD) along with diesel particulate filter (DPF) significantly reduced diesel exhaust emissions both gaseous and PM emissions from diesel engines. The CNG-fueled vehicles exhibited emissions of non-regulated compounds and nanoparticles that were greater by a factor of 15-20 than corresponding emissions from a diesel engine equipped with an exhaust after treatment device.[13]

Mathis et al. (2004) studied the influence of volatile organic compounds (VOCs) on nucleation of nanoparticles in the exhaust of a modern light-duty diesel vehicle. In this study, different organic compounds, with a diverse molecular structure, were added to the dilution air. The size distribution and the particle concentrations were measured using a Scanning Mobility Particle Sizer (SMPS) while varying the sample temperature and relative humidity. The results showed a large variation in the number concentration of nucleation mode particles in response to the varying sampling conditions and different organic compounds. Any increase in the number concentration of nanoparticles in the exhaust of CNG engines despite the absence of carbonaceous soot nucleating sites has been linked not only with the ash and heavier hydrocarbon content in the lubrication oil but also to the non-regulated sulfur content in the lube oil and volatile organic compounds in the natural gas exhaust [14,15]. A novel exhaust after treatment device was developed by West Virginia University (WUV) and Lubrizol to reduce the soluble organic fractions in the natural gas exhaust, promoting further reduction of PM emissions both by mass and number concentration. The exhaust after treatment device comprised of a catalyzed particulate filter and an oxidation catalyst, which trapped the ash, produced from lube oil additives and oxidized the heavier hydrocarbons from incomplete combustion of lube oil; thereby, minimizing nanoparticle formation.[16]

Increasing availability of gaseous fuel and the demand to use them for power generation has led to manufacturing of the gas engines. Most of the engines are modified from diesel engines to run on gas by introducing the ignition, gas governing and carburetion systems along with change in compression ratio and in some cases change in the combustion chamber. Each of the system components plays an important role in the overall performance of the engine. It is assumed, that the effect of ignition time, ignition quality and the mixture ratio control for a given combustion chamber design, are chosen in such a way that they are the best. Over the last three decades research and development on the engine has been addressing the use of technology for better combustion within a cylinder volume, by improving amongst other aspects, the spray characteristics of fuel and mixing with air. Recent development in the injectors and combustion chamber designs have led to very compact engines with the primary aim to reduce the weight, while improving the overall conversion efficiency. In the bargain, the excess air factors that were in the range of 25–30 % have been restricted to about 15-20% in most of the engines. [17]

The present designs of gas engines adopt most of the hardware related to diesel engines. Thus, a diesel engine is considered as a benchmark for the comparison of the power output of the gas engine. In the analysis, only 4 stroke engine designs are considered.

2.1.2 Choice of CNG as an Alternative Fuel

The conscious endeavor towards searching for alternatives for the polluting conventional fuels had started a long time ago. But it was also important to establish the feasibility both technical and commercial of the alternatives. Given its characteristics and unmatched advantages, CNG was the obvious choice for an alternative automobile fuel.

CNG has emerged as an attractive alternative automobile fuel due to its clean burning characteristics and very low amount of exhaust emissions. As a fuel, it is clean, economical and has been in use worldwide to power vehicles. Petrol driven vehicles can use CNG by installing a Bi-Fuel Conversion kit and the converted vehicle has the flexibility of operating either on CNG or petrol. Diesel Engines can also be converted to run on CNG by installing a dual fuel kit or converting the existing diesel engine into a Spark Ignition one.

Table 2.1: Comparison of Fuel Properties [17]

PROPERTIES	NATURAL GAS	DIESEL
Boiling Point (K @1 Atm)	147	433-655
Density (kg/cum)	128	785-881
Auto Ignition Temperature (K)	900	477-533
Flash Point (K)	124	325
Octane / Cetane Number	130	46-51
Flammability Limits Range	5.0-15	0.7-5
Net Energy Content (MJ/Kg)	49.5	43.9
Combustion Energy (KJ/cum)	24.6	36
Vaporization Energy (MJ/cum)	215-276	192

2.1.3 Composition Of Natural Gas

Generally, natural gas is one of the hydrocarbon families, made up of carbon and hydrogen atom. There are different compounds in natural gas such as methane, ethane, propane and iso-butane as well as other non-hydrocarbon compounds such as carbon dioxide and nitrogen. The natural gas is assumed to consist of mainly methane, ethane and propane. Their respective composition percentage of the typical natural gas is shown in Table 2.2.

It is very important to know the composition of the natural gas used for the analysis because different composition has different effect on the combustion process in the diesel engine. Unfortunately, there is no standard reference for the design of a standard CNG diesel engine because the natural gas composition varies in different countries. This posed a problem to the engineer in designing the fuel feeding system and the injection system for the CNG-diesel engine.

Table 2.2: Composition percentage of Natural Gas [18]

Constituent	Formula	Volume %
Methane	CH_4	70-90
Ethane	C_2H_6	0-5
Propane	C_3H_8	0-5
Butane	C_4H_{10}	0-5
Pentane	C_5H_{12}	0-5
Hexane (& higher)	C_6H_{14}	Trace
Benzene (& higher)	C_6H_6	Trace
Carbon dioxide	CO_2	0-8
Oxygen	O_2	0-0.2
Nitrogen	N_2	0-5
Hydrogen Sulphide	H_2S	0-5
Rare Gases	He,Ne,Ar,Kr,Xe	Trace
Water	H_2O	Trace

Moreover, the variation in the natural gas composition brought difficulties in the improvement of engine performance and minimization of the exhaust gas pollution. Since the proportion of methane in natural gas is the largest compared to other gases like propane and ethane, the main characteristic of natural gas can be directly related to the characteristic of methane. To configure this problem with variation of natural gas composition, the Natural Gas Vehicles (NGV) Coalition in USA has recommended a general guideline of natural gas composition used for the emission test certificate. This test is carried out to help the certification of the engine's performance and its exhaust gas pollution characteristics that are affected by the gas composition [19]

2.1.4 Properties of Natural Gas

Physically, natural gas is colorless, tasteless, relatively non-toxic [20] and not a volatile organic compound (VOC)[8]. It exists in our environment at normal temperature and pressure, which gave it its name. To use natural gas as fuel in vehicles, it has to be compressed at a high pressure of about 18- 20 MPa at normal temperature in vessels before it can be supplied to the engine's combustion chamber. Generally, natural gas is lighter than air with a vapour density of 0.68 relative to air [21]. Therefore, if leaking happens, it will not cause explosion but instead it will disperse to the atmosphere. Natural gas has a high auto-ignition temperature compared to gasoline or diesel, which is the lowest temperature for it to ignite through heat alone and without any spark or flame [22]. Higher ignition temperature means that natural gas is more difficult to ignite. This can significant reduce the fire hazard, and constitute anti-knocking ability especially when it is compressed in a very high pressure in the combustion chamber. This property is certainly useful for the design of a dual-fuel engine. The ignition temperature for natural gas is about 900 K.

Other physical properties such as the flammability limits range, octane rating, Wobbe Index and flash point also play an important role in the analysis of compression ratio and combustion efficiency of the engine. The Wobbe Index is a measure of the fuel interchangeability with respect to its energy content and the air-fuel ratio [22].

Flash point is the minimum temperature for an ignition. The flash point for natural gas is approximately 180° at normal pressure. The flammability limit range is the concentration of natural gas in air to cause an explosion. This is between the lower explosive limit (LEL) of 5% to the upper explosive limit (UEL) of 15%.

If the concentration of natural gas is more or less than this range, an explosion would not occur. This will certainly reduce the risk of explosion of CNG in air due to leaking because natural gas can only burn in air when the concentration of CNG is high. With this wide range of a lean mixture of CNG and air can be used for the CNG-diesel engine to promote better exhaust emission properties. The octane rating is an important property in determining the compression ratio of the engine. For natural gas, the octane number is approximately 130. This is much higher than gasoline with an octane number of 96[22]. This property is important as it determines the time needed for the natural gas and air to mix homogeneously in the combustion chamber to minimize knocking or detonation.

2.1.5 Advantages of Using CNG

CNG has four big safety features that make it an inherently safer fuel than petrol, diesel, or LPG.

1. CNG has a specific gravity of 0.587. This means that it is lighter than air so if it leaks, it just rises up and dissipates into the atmosphere. On the ground other fuels will form a hazardous puddle on should a leak occur.
2. It has a self-ignition temperature of 700 degree centigrade as opposed to 455 degree centigrade of petrol.
3. CNG has to mix air within small range of 4 to 14 percent by volume for combustion to occur. This is a far narrower range than for petrol.
4. CNG cylinders are designed and built with special materials to the highest safety specifications, which make storage far safer than petrol tanks.

The life of an engine increases by using CNG. Lubricating oil life is extended considerably because CNG does not contaminate and dilute the crankcase oil. Due to the absence of any lead content lead fouling of plugs is completely eliminated and plugs life is greatly extended. Another aspect, which increases engine life, is that CNG enters the engine in the form of a gas whereas petrol or diesel enters in the form of spray or mist.

A big advantage of CNG is that it is virtually pollution free. It is a natural gas mainly composed of methane; therefore, its exhaust emissions consist of water vapors and a small fraction of carbon mono-oxide. As there are no carbon and other particles in the exhaust, the exhaust fumes are negligible. CNG has a good mixture distribution quality. When the correct proportions are brought together they mix thoroughly and rapidly, which improves the combustion efficiency of the engine.

The Research Octane Number of CNG is 130 as compared to 87 of premium motor gasoline. The equipment is not complex and gives many trouble free years of service. However, as for all fuels, to maintain maximum efficiency it is advisable to have a routine check after every 10,000 km. The CNG market is more stable than the gasoline market. CNG generally costs 15 to 40 percent less than gasoline or diesel. CNG requires more frequent refueling, however, because it contains only about a quarter of the energy by volume of gasoline.

In addition, CNG vehicles cost between $3,500 to $6,000, more than their gasoline-powered counterparts. This is primarily due to the higher cost of the fuel cylinders. As the popularity and production of CNG vehicles increases, vehicle costs are expected to decrease.

• Air Quality – Most studies indicate a reduction in NOX of approximately 50% and PM of more than75%.

• Potential Fuel Cost Savings – Viking Freight Study showed average fuel costs per mile of $0.11 for CNG and $0.16 for diesel when natural gas and diesel fueling were on site (31% fuel cost savings)

• Political Benefits – Most fleets switch to natural gas because of political benefits

- Meet government requirements

- Promote energy security

- Enhance public image

CNG can be used in Otto-cycle (gasoline) and modified Diesel cycle engines. Lean-burn Otto-cycle engines can achieve higher thermal efficiencies when compared with stoichiometric Otto-cycle engines at the expense of higher NOx and hydrocarbon emissions. Electronically-controlled stoichiometric engines offer the lowest emissions across the board and the highest possible power output, especially when combined with EGR, turbocharging and intercooling, and three way catalytic converters, but suffer in terms of heat rejection and fuel consumption. A suitably designed natural gas engine may have a higher output compared with a petrol engine because the octane number of natural gas is higher than that of petrol.

The cost of running a car on CNG is 73 per cent cheaper than petrol and 45 per cent cheaper than diesel at current rates. One kg of CNG in Delhi is priced at Rs 18.90 and can run 20.85 km, whereas petrol costs Rs 50.56 per litre and offers a mileage of 15 km, and diesel costs Rs 32.35 per litre and runs 20 km.

2.1.6 Natural Gas under the Earth

CNG is usually found underneath the surface of the earth. As natural gas has a low density, once formed it will rise towards the surface of the earth through loose, shale type rock and other material. Most of this methane will simply rise to the surface and dissipate into the air. However, a great deal of this methane will rise up into geological formations that 'trap' the gas under the ground. These formations are made up of layers of porous, sedimentary rock (kind of like a sponge, that soaks up and contains the gas), with a denser, impermeable layer of rock on top. This impermeable rock traps the natural gas under the ground. If these formations are large enough, they can trap a great deal of natural gas underground, in what is known as a reservoir.

There are a number of different types of these formations, but the most common is created when the impermeable sedimentary rock forms a 'dome' shape, like an umbrella that catches all of the natural gas that is floating to the surface. There are a number of ways that this sort of ‘dome’ may be formed. For instance, faults are a common location for oil and natural gas deposits to exist. A fault occurs when the normal sedimentary layers sort of split, vertically, so that impermeable rock shifts down to trap natural gas in the more permeable limestone or sandstone layers. Essentially, the geological formation which layers impermeable rock over more porous, oil and gas rich sediment has the potential to form a reservoir.

The picture below shows how natural gas and oil can be trapped under impermeable sedimentary rock, in what is known as an anticlinal formation. To successfully bring these fossil fuels to the surface, a hole must be drilled through the impermeable rock to release the fossil fuels under pressure. Note that in reservoirs that contain oil and gas, the gas, being the least dense, is found closest to the surface, with the oil beneath it, typically followed by a certain amount of water.

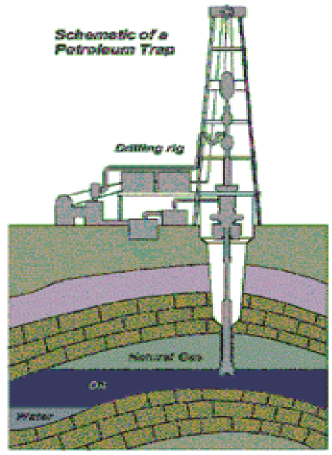

Figure 2.1: Schematic of petroleum trap

With natural gas trapped under the earth in this fashion, it can be recovered by drilling a hole through the impermeable rock. Gas in these reservoirs is typically under pressure, allowing it to escape from the reservoir on its own. In addition to being found in a traditional reservoir such as the one shown above, natural gas may also be found in other 'unconventional' formations.[18]

2.1.7 World Natural Gas Reserves

The EIA, in conjunction with the Oil and Gas Journal and World Oil publications, estimates world proved natural gas reserves to be around 5,210.8 Tcf. As can be seen from the graph, most of these reserves are located in the Middle East with 1,836.2 Tcf, or 34 percent of the world total, and Europe and the Former U.S.S.R. with 2158.7, or 42 percent of total world reserves. The United States, by this calculation, possesses 3 percent of the world total natural gas reserves.

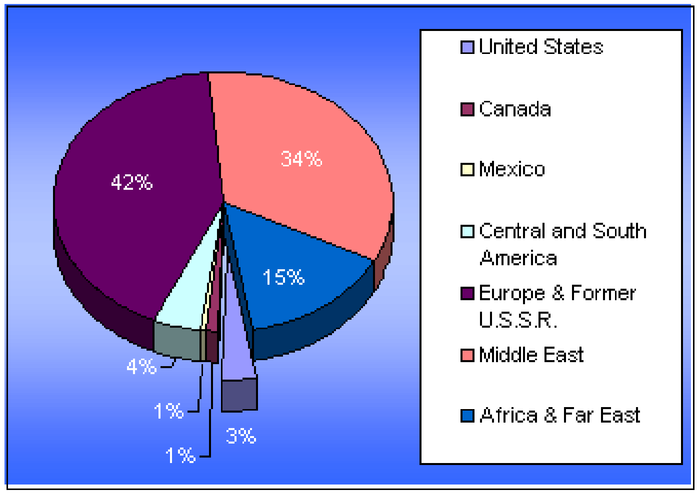

Figure 2.2: World natural gas reserves

A study released by the American Clean Skies Foundation (ACSF) and conducted by Navigant Consulting, Inc. (NCI) concludes that the United States has 2,247 trillion cubic feet (Tcf) of natural gas proved reserves and unproved technically recoverable resources, including major contributions from unconventional resources from three sources: tight sands, coalbed methane and especially from shale. Reserves at that level would supply natural gas for 118 years at current production levels, according to the report.

Total US natural gas production reached 19.3 Tcf/year (52.9 Bcf/day) by the end of 2007, a 4.3% increase over the 18.5 Tcf/year (50.7 Bcf/day) level at the end of 2006. Over the last decade, production from unconventional sources has increased almost 65%, from 5.4 Tcf/year (14.8 Bcf/day) in 1998 to 8.9 Tcf/year (24.4 Bcf/day) in 2007. Unconventional production has increased from 28% of total production in 1998 to 46% in 2007.[23]

2.1.8 CNG Worldwide Experiences:

The first natural gas engine was invented in the mid-1800. Since the 1930's, Italy has been using natural gas as a vehicle fuel and developing its associated technologies. During World War II some cars in Australia were converted to run on "town gas". The adoption of CNG as a vehicle fuel is now accelerating worldwide. In the last decade, the number of Compressed Natural Gas Vehicles (CNGV's) around the world has jumped by 500% from around 1 million vehicles in 1996 to in excess of 5 million vehicles in 2006.[23]

The online Newspaper DNA (Daily News and Analysis) states that according to statistics available with the Bharat Petroleum Corporation Limited (BPCL), the consumption of natural gas in Mumbai, India has risen by a whopping 374% over the past 3 years. To improve security of energy supply and reduce greenhouse gas emissions, the European Union has set itself an objective to have substituted 20% of all traditional fuels used in the road transport sector (petrol and diesel) with alternative fuels before the year 2020. Three alternative solutions are seen as promising: biofuels, natural gas and hydrogen. By 2020, at least 50% of alternative fuel substitution is expected to be natural gas.

Canada

Canada is a large producer of natural gas, so it follows that CNG is used in Canada as an economical motor fuel. Canadian industry has developed CNG-fueled truck and bus engines, CNG-fueled transit buses, and light trucks and taxis. Both CNG and propane refueling stations are not difficult to find in major centers.

New Zealand

The New Zealand CNG program commenced in 1979. By 1985, over 100,000 vehicles were on CNG and the NZ program was leading the world. All these vehicles were converted from gasoline or diesel and amounted to over 10% of the national fleet that had access to natural gas. There were 450 refueling stations. In 1985, the Government announced the withdrawal of incentives and this resulted in the end of the vehicle conversion program. Since then, the NGVs of that era have reached the end of their useful lives and have not been replaced. There are now about 1500 vehicles on CNG including 40 buses in Hamilton and Auckland. There is a strong NZ export industry mainly in refueling equipment but also in vehicle conversion technology and services.

Europe

In Germany, CNG-generated vehicles are expected to increase to two million units of motor-transport by the year 2020. The cost for CNG fuel is between 1/3 and 1/2 compared to other fossil fuels in Europe**.**

South America

Argentina and Brazil are the two countries with the largest fleets of CNG vehicles. Conversion has been facilitated by a substantial price differential with liquid fuels, locally-produced conversion equipment and a growing CNG-delivery infrastructure. A ‘Blue-network’ of CNG stations is being developed on the major highways of the Southern Cone (including Chile and Bolivia) to allow for long-haul transportation fuelled by CNG.

Asia

In India, CNG costs are at Rupees 18.90 per kg compared with Rs.50.56 per liter of petrol. The cost saving is immense along with reduced emissions and environmentally friendlier cars. CNG has been made mandatory for all public transport in the Indian capital city of New Delhi. CNG has grown into one of the major fuel sources used in car engines in Iran, Pakistan, Bangladesh and India. The use of CNG is mandated for the public transport system of India’s capital New Delhi as well as for the city of Ahmedabad in the state of Gujarat. The Delhi Transport Corporation operates the world’s largest fleet of CNG buses.

In Singapore CNG is increasingly being used by public transport vehicles like buses and taxis, as well as goods vehicles. However, according to Channel News Asia on April 18, 2008, more owners of private cars in this country are converting their petrol-driven vehicles to also run on CNG motivated no doubt by fiercely-escalating petrol prices these days.

Oceania

During the 1970s and 1980s, CNG was commonly used in New Zealand in the wake of the oil crises, but fell into decline after petrol prices receded. Brisbane and Transperth in Australia have both adopted a policy of only purchasing CNG buses in future. Transperth is purchasing 451 Mercedes-Benz OC500LE buses, including 58 articulated buses, while Brisbane Transport has purchased 216 Scania L94UB and 180 MAN 18.310 models as well as ordering up to 30 articulated CNG buses on MAN chassis. In the 1990s Benders Buslines of Geelong, Victoria trialled CNG buses for the Energy Research and Development Corporation.

2.1.9 Advances in the Exploration and Production Sector

Technological innovation in the exploration and production sector has equipped the industry with the equipment and practices necessary to continually increase the production of natural gas to meet rising demand. These technologies serve to make the exploration and production of natural gas more efficient, safe, and environmentally friendly. Despite the fact that natural gas deposits are continually being found deeper in the ground, in remote, inhospitable areas that provide a challenging environment in which to produce natural gas, the exploration and production industry has not only kept up its production pace, but in fact has improved the general nature of its operations. Some highlights of technological development in the exploration and production sector include:

- 22,000 fewer wells are needed on an annual basis to develop the same amount of oil and gas reserves as were developed in 1985.
- Had technology remained constant since 1985, it would take two wells to produce the same amount of oil and natural gas as one 1985 well. However, advances in technology mean that one well today can produce two times as much as a single 1985 well.
- Drilling wastes have decreased by as much as 148 million barrels due to increased well productivity and fewer wells.

- The drilling footprint of well pads has decreased by as much as 70 percent due to advanced drilling technology, which is extremely useful for drilling in sensitive areas.
- By using modular drilling rigs and slim hole drilling, the size and weight of drilling rigs can be reduced by up to 75 percent over traditional drilling rigs, reducing their surface impact.
- Had technology, and thus drilling footprints, remained at 1985 levels, today's drilling footprints would take up an additional 17,000 acres of land.
- New exploration techniques and vibrational sources mean less reliance on explosives, reducing the impact of exploration on the environment.

2.1.11 CNG Conversion

In conventional Spark Ignition/Petrol/Gasoline Engines, conversion to CNG as the fuel requires a CNG Conversion Kit whereas in Compression Ignition /Diesel engines major modifications are required in the engine cylinder head.

Figure 2.3 Components of a CNG kit

Major components of CNG kit for carburetor fitted petrol vehicle are illustrated below:

1. Pressure Regulator.
2. Petrol Solenoid Valve with manual override switches. (Stops petrol flow when operating on CNG).
3. On-Off valve and refueling connector. (Opens or stops gas flow to the regulator and includes a refueling device).
4. Control Module/Change-over Switch (Electronic control component with fuel selection switch).
5. CNG level Indicator (LED Indicator).
6. Gas Air Mixer.
7. CNG cylinder with valve, vapour bag & bracket.
8. Petrol hose.
9. Low-pressure gas hose.
10. Ignition advance processor.
11. High pressure gas tube.
12. Wire harness.
13. NRV in petrol return line.
14. Pressure gauge.

A CNG Conversion Kit costs approximately Rs.29000 with a 10 Kg Cylinder and Rs.32000 with a 12 Kg cylinder. These conversion kits are tested and certified by IARI for every type/model of vehicle. CNG cylinders are manufactured from a special steel alloy and are seamless in construction. Their compact size allows them to easily fit even in a small car.

CNG cylinders are designed and built in such a way so as to withstand high pressure. The maximum pressure in a CNG cylinder is up to 200 kg/cm^2. CNG cylinders are safe as they are manufactured as per specific requirements and tested before use, in accordance with international specifications and standards and they are duly approved by Chief Controller of Explosives.

2.1.12 Difficulties For Developing CNG Infrastructure

The following difficulties were faced in developing CNG infrastructure:

- Limited natural gas allocation leading to delay in management decisions on expenditure commitment.
- Uncertainty about conversion of vehicles & CNG demand.
- Lack of indigenous technology.
- Capital intensive project - a mother station cost would be 5-6 times the cost of a petrol pump & pipeline need to be in place.
- Infrastructural constraints (Electricity, land etc.).
- Delay in getting permissions from statutory authorities.
- Objection from local people, encroachment.
- Low storage capacity of on board cylinders, thus requiring frequent refills.

2.2. STUDIES ON BIODIESEL

2.2.1 Introduction

Various products derived from vegetable oils have been proposed as an alternative fuel for diesel engines. Today ‘biodiesel’ is the term applied to the esters of simple alkyl fatty acids used as an alternative to petroleum based diesel fuels. Importance of biodiesel in the recent context increases due to increasing petroleum prices; limited fossil fuel reserves and environmental benefits of biodiesel viz. decrease in acid rain, decrease in emission of CO_2, SO_x and un-burnt hydrocarbons during the combustion process. Beside these factors, its easy biodegradability, production of biodiesel is considered as an advantage over that of fossil fuels.

The main commodity sources of biodiesel in India are non edible oils obtained from plant species such as Jatropha, Pongamia pinnata, Thumba oil etc. Biodiesel contains no petroleum, but it can be blended at any level with petroleum diesel to create a biodiesel blend or can be used in its pure form. Just like petroleum diesel, biodiesel operates in compression ignition engine, which essentially require very little or no engine modifications because biodiesel has properties similar to petroleum diesel fuels. It can be stored just like petroleum diesel fuel and hence does not require separate infrastructure.

The use of biodiesel in conventional diesel engines results in substantial reduction of un-burnt hydrocarbons, carbon monoxide and particulate matters. Biodiesel is considered as a clean fuel since it has almost no sulphur, no aromatics and has about 10% built in oxygen, which helps it to burn fully. Its higher cetane number improves the ignition quality even when blended in the petroleum diesel

2.2.2 <u>Chemistry of Biodiesel</u>

Biodiesel is made using the process of transesterification. In the transesterification of different type of oils, triglycerides react with an alcohol, generally methanol or ethanol, to produce esters and glycerin. To make it possible, a catalyst is added to the reaction.

$$
\begin{array}{lllllll}
CH_2\text{-OOC-}R_1 & & & & R_1\text{-COO-}R' & & CH_2\text{-OH} \\
| & & & \text{Catalyst} & & & | \\
CH\text{-OOC-}R_2 & + & 3R'OH & \longleftrightarrow & R_2\text{-COO-}R' & + & CH\text{-OH} \text{ ------- (2.1)} \\
| & & & & & & | \\
CH_2\text{-OOC-}R_3 & & & & R_3\text{-COO-}R' & & CH_2\text{-OH} \\
 & & & & & & \\
\text{Glycerides} & & \text{Alcohol} & & \text{Esters} & & \text{Glycerin}
\end{array}
$$

where, the term R represents to different alkyl groups.

The overall process is normally a sequence of three consecutive steps, which are reversible reactions. In the first step from triglycerides, Diglycerides is obtained. From Diglycerides, Monoglycerides is produced and in the last step, from Monoglycerides, glycerin is obtained. In all these reactions esters are produced. The stoichometric relation between alcohol and the oil is 3:1. However, an excess of alcohol is usually more appropriate to improve the reaction towards the desired product [24, 25]

$$\textbf{Triglycerides (TG) + R'OH} \underset{k_2}{\overset{k_1}{\longleftrightarrow}} \textbf{Diglycerides (DG) + R'COOR}_1 \text{ ------ - } \quad (2.2)$$

$$\textbf{Diglycerides (DG) + R'OH} \underset{k_4}{\overset{k_3}{\longleftrightarrow}} \textbf{Monoglycerides (MG) + R'COOR}_2 \text{ --- } \quad (2.3)$$

$$\textbf{Monoglycerides (MG) + R'OH} \underset{k_6}{\overset{k_5}{\longleftrightarrow}} \textbf{Glycerin (GL) + R'COOR}_3 \text{ ------- } \quad (2.4)$$

where k_1, k_2, k_3, k_4, k_5, k_6 are the catalyst used which may be acidic catalyst, alkali catalyst or lipase as a catalyst. These are described below.

<u>Alkali Catalyst</u>

These catalysts can be used with methanol or ethanol as well as any kind of oils, refine, crude or frying. The main alkali catalysts are.

a. Sodium hydroxide (NaOH).

b. Potassium hydroxide (KOH).

<u>Acidic Catalyst</u>

Acid transesterification is a great way to make biodiesel if the sample has relatively high free fatty acid content. The main acidic catalysts are.

a. Sulfuric acid (H_2SO_4).

b. Sulfonic acid.

2.2.3 Resources of Biodiesel

Many developed countries have active biodiesel programs. Currently biodiesel is produced mainly from field crop oil like rapeseed, sunflower etc. in Europe and soybean in US. Malaysia utilizes palm oil for biodiesel production while in Nicaragua it is Jatropha oil [26].The productions of vegetable oil globally and in India are given in Table 2.3 and Table 2.4.

Table 2.3: Global Productions of the Major Vegetable Oils

Oil	Production (million tonnes)
Soybean	27.8
Rapeseed	13.7
Cottonseed	4.0
Sunflower	8.2
Peanut	5.1
Coconut	3.5
Linseeds	0.6
Palm	23.4
Palm kernel	2.9
Olive	2.7
Corn	2.0
Castor	0.5
Seasame	0.8
Total	**95.2**

Table 2.4: Vegetable oil production in India

Oil	**Production (million tonnes)**
Groundnut	1.40
Soya	0.82
Rape / Mustard	1.55
Sunflower	0.30
Sesame	0.26
Castor	0.25
Niger	0.03
Safflower	0.09
Linseeds	0.10
Cottonseed	0.44
Coconut	0.55
Rice Bran	0.55
Oil from expelled cakes	0.28
Minor oilseeds	0.05
Total	**6.67**

Source: Table 2.3 & 2.4 [Ref.:26]

2.2.4 Properties of Biodiesel

A general understanding of the various properties of biodiesel is essential to study their implications in engine use, storage, handling and safety.

Density/ Specific gravity

Biodiesel is slightly heavier than conventional diesel fuel (specific gravity 0.88 compared to 0.84 for diesel fuel). This allows use of splash blending by adding biodiesel on top of diesel fuel for making biodiesel blends.

Cetane Number

Biodiesels has higher cetane number than conventional diesel fuel. This results in higher combustion efficiency and smoother combustion.

Viscosity

In addition to lubrication of fuel injection system components, fuel viscosity controls the characteristics of the injection from the diesel injector (droplet size, spray characteristics etc.). The viscosity of methyl esters can go to very high levels and hence, it is important to control it within an acceptable level to avoid negative impact on fuel injection system performance. Therefore, the viscosity specifications proposed are same as that of the diesel fuel.

Flash point

Flash point of a fuel is defined as the temperature at which it will ignite when exposed to a flame or spark. The flash point of biodiesel is higher than the petroleum based diesel fuel. Flash point of biodiesel blends is dependent on the flash point of the base diesel fuel used, and increase with percentage of biodiesel in the blend. Thus in storage, biodiesel and its blends are safer than conventional diesel. The flash point of biodiesel is around 160 ^{0}C.

Cold filter plugging point (CFPP)

At low operating temperature fuel may thicken and not flow properly affecting the performance of fuel lines, fuel pump and injectors. Cold filter plugging point of biodiesel reflects its cold whether performance. It defines the fuels limit of filterability. Biodiesel thicken at low temperatures so need cold flow improver additives to have acceptable CFPP.

Cloud point

Cloud point is the temperature at which a cloud or haze of crystals appear in the fuel under test conditions and thus becomes important for low temperature operations. Biodiesel generally has higher cloud point than diesel fuels.

Aromatics

Biodiesel does not contain any aromatics so aromatic limit not specified.

Stability

Biodiesel age more quickly than fossil diesel fuel due to the chemical structure of fatty acids and methyl esters present in biodiesel. Typically there are up to 14 types of fatty acid methyl esters in the biodiesel. The individual proportion of presence of these esters in the fuel affects the final properties of biodiesel. Saturated fatty acid methyl esters (C14:0, C16:0, C16:0) increase cloud point, cetane number and improve stability whereas more polyunsaturates (C18:2, C18:3) reduce cloud point, cetane no. and stability.

There are three types of stability criteria:

1. Oxidation stability
2. Storage stability
3. Thermal stability

Iodine number

Iodine number refers to the amount of iodine required to convert unsaturated oil into saturated oil. It refers to the amount of unsaturated fatty acid in the fuel. One value of iodine number can be obtained by using several grades of unsaturated acids. So an additional parameter, linolenic acid (C18: 3) content is specified and limited to 15% in Austrian Standard ON C 1191.

Acid number/ Neutralization number

Acid number reflects the presence of free fatty acids or acid used in manufacture of biodiesel. It also reflects the degradation of biodiesel due to thermal effect. The resultant high acid number can cause damage to injector and also result in deposit in fuel system and affect life of pumps and filters.

Some of the important properties of biodiesel proposed by BIS (Bureau of Indian standards) are given in Table 2.9. Properties of the biodiesel from different oil are given in Table 2.9.

Table 2.5: Summary of proposed BIS (Bureau of Indian Standards) for biodiesel [27]

Standard / specification		**Proposed BIS**
Density @ 15°C	g/cm^3	0.87 – 0.90
Viscosity @ 40°C	mm^2/s	3.5 - 5.0
Flash point	°C	>=100
Sulphur, max.	%mass	0.035
CCR,100%distilation residual max.	%mass	0.05
Sulphated ash, max.	%mass	0.02
Water. max.	mg/kg	500
Total contamination, max.	mg/kg	20
Cetane no		>=51
Acid no	mg KOH/g	<=0.8
Methanol	%mass	<=0.02
Ester content	%mass	>=96.5
Diglyceride	%mass	<=0.2
Triglyceride	%mass	<=0.2
Free glycerol	%mass	<=0.02
Total glycerol	%mass	<=0.25
Iodine no		<=115
Phosphorus	ppm	<=10
Alkaline matter(Na,K)		<=10
Distillation, T 95%	°C	<=360

Table 2.6: Properties of biodiesel from different oils [27]

Vegetable oil methyl esters (biodiesel)	Kinematic viscosity (mm^2/s)	Cetane No.	Lower heating value (MJ/kg)	Cloud point (^oC)	Pour point (^oC)	Flash point (^OC)	Density (kg/l)
Peanut	4.9	54	33.6	5	–	176	0.883
Soya bean	4.5	45	33.5	1	-7	178	0.885
Babassu	3.6	63	31.8	4	–	127	0.875
Sunflower	4.6	49	33.5	1	–	183	0.860
Tallow	–	–	–	12	9	96	–
Diesel	3.06	50	43.8	–	-16	76	0.855
20% biodiesel	3.2	51	43.2	–	-16		0.859

2.3 STUDIES ON DUAL FUEL

2.3.1 Introduction

Razavi and Karim (2005) conducted the experimental work and evaluate that an increase in the cetane number of the pilot fuel when operating the engine on low heating value gas fuel mixtures, represented by 35% CH_4 and 65% N_2 by volume, decreases the ignition delay significantly, while maintaining its characteristic variations with the increased admission of the gaseous fuel. The decrease is relatively less marked when operating with this large pilot liquid fuel quantity when a high cetane number fuel is employed. It can also be seen that the employment of relatively lower pilot quantity can increase significantly the ignition delay. The limits of nitrogen and carbon dioxide admissions were obtained by keeping the pilot and gas (methane) quantity employed constant during the test. By adding the amount of nitrogen or carbon dioxide in the methane and changing the load to adjust the speed of the motor at 1000 rpm. The result of the limits tests for the nitrogen and carbon dioxide admissions for two different cetane number, pilot quantities, and equivalence ratios. The ignition delay increases gradually by adding the inert gasses (i.e. nitrogen or carbon dioxide) to certain points, but later on it increases sharply to the limit point when the satisfactory engine operation will be impossible. In these tests the constant equivalence ratio was assumed, although there were small variations of this quantity mainly due to the changes of the volumetric efficiency of engine when the percentage of nitrogen or carbon dioxide admissions was varied. [28]

The project conducted by California Energy Commission Sacramento, California shows that The C-10 DFNG engines in dual-fuel mode had lower emissions of the following pollutants as compared with the C-10 D engine (ranges represent lowest/highest value for all three drive cycles tested): NOx (27%–60%), PM (54%–64%), and CO_2 (14%–19%), and increases of CO (634%–860%) and non-methane hydrocarbons (NMHC) (697%–1,718%). In the two variable load tests (EPA UDDC and the WVU CBD), the C-10 DFNG may have had higher combined NOx and NMHC emissions. However, the smog-forming potential of these combined emissions compared to diesel combined NOx and

HC emissions cannot be quantified because specific hydrocarbons vary in their ozone reactivity. Because the HC emissions were not speculated in this study, the reactivity of the DFNG and diesel exhaust emissions cannot be compared. Even so, the simple addition of NOx + NMHC emissions indicates a 17% increase compared with the C-10 D engine for the variable load cycles. In contrast, on the 55-MPH Steady State test, the C-10 DFNG engine had 42% lower NOx + NMHC emissions compared with the C-10 D.[21]

2.3.2 Advantages and Disadvantages of Using CNG-Diesel Engine

The advantage of converting conventional diesel engine to dual fuel CNG-diesel engine is to convert the engine back to 100% diesel operation easily. It does not require a spark ignition or an electrical system to start the combustion process. Instead like spark ignition engine operating on premixed combustion process, the dual fuel engine works in a diffusion combustion process with a high pressure. Besides that, conversion of diesel engine is more economical than conversion of spark ignition engine due to less modification on the original diesel engine [24].

It has fuel flexibility .It can be operated on full diesel mode if natural gas is not available. Duel fuel engine have potential benefits like higher power density, better efficiency and lean burn combustion capacity which results in reduced misfire NO_X emission. Diesel pilot fuel provides lubrication to valves and rings which results in longer maintenance intervals and reduces maintenance cost. The major advantage of dual fuel natural gas engine compared to spark ignition engine is its ability to burn very lean air fuel mixtures with high efficiency [29].

Since dual fuel engine operates at low compression ratio such as 16:1, the original diesel engine only needs minimum modification to suit the CNG diesel operation. Therefore, the cost of conversion for diesel engine is lower compared to spark ignition engine. Another advantage of using CNG-Diesel engine is the increase in power output compared to the original diesel engine, which must be within the range of up to 3000rpm. If more than 3000rpm, the power output of CNG diesel engine would decrease slowly due to knocking in the rapid combustion process.

Since CNG has high octane rating and high ignition temperature, the CNG-diesel engine has a higher resistant to knock than the conventional diesel engine. However, mild knock still will occur; hence a careful consideration for the compression ratio is necessary to minimize the knock. The CNG-Diesel engine can be used at a low substitution level to lower the emission gases. It is more environmental-friendly and has a good energy security because it produces less hazardous emissions such as carbon monoxide, nitrogen oxides, sulphur oxides and particulate matters.

Moreover, the benefits and the good properties of methane as the main component in natural gas reduce the risk of explosion when leakage occurs. However, there still exist some limitations to the CNG-Diesel engine. For instance, the fuel control system is more complicated compared to the spark ignition system. Adjustment to the mixing ratio of CNG-Diesel and the control of the injector pump are difficult to set correctly for a homogeneous mixture. The variation in the composition of natural gas around the world creates difficulties to certificate the engine performance and exhaust pollution characteristic in the fuel control system. Another important disadvantage of CNG-diesel engine that caused significant withdrawal from vehicle buyers is the supply system.

Normally, dual fuel vehicles need to have separate tanks to store the natural gas in compressed tanks, which are in liquid form. This increases the weight of the vehicle causing reduced power output, limited storage space and posing other drivability problems to the vehicle owner. The refueling time is longer approximately twice as much as the refueling time for normal gasoline vehicles. There are lesser refueling station for natural gas compared to other conventional fuel such as diesel and gasoline [25]. Moreover, it has shorter driving range due to lower energy density of natural gas. Lastly, it has a low efficiency when operating in part load or no load condition but this disadvantage is not significant because the vehicle would be in idling position.

2.4. STUDIES ON AIR POLLUTION:

2.4.1 Introduction

Well we are living in the global community and have to realize that we are affecting others in everything that we do. Historically, the automobile has been a noxious polluter all around. The resources needed to produce and operate a car were a factor and the byproducts from the production and operation of the car were a factor. The different type of pollutants can be broken down into categories based on the different areas that they affect. The automobile produces air, water, and noise pollution.

Air pollution levels greatly exceed national and World Health Organization health-based standards, and transportation is by far the largest source of the pollution. Countries world over, especially in the developing world, are experiencing rapid urbanization. The share of the world's population living in cities is reported to have grown from about 35 percent in 1970 to almost 50 percent in 2001 and this number is expected to increase to more than 60 percent by 2030 [28]. One of the many consequences of the increased economic activity that accompanies urbanization is increased vehicle use. Concentrations of conventional air pollutants, including sulfur dioxide (SO_2), particulates (PM_{10} and $PM_{2.5}$), ozone (O_3), nitrogen dioxide (NO_2), carbon monoxide (CO), and air toxics, are rising in many cities and are in many cases already well above the World Health Organization's guidelines for ambient air-quality standards.

Yet Delhi, India, once ranked among the world's most polluted mega-cities, has proven to be an exception. Delhi was once considered to be the world's fourth most polluted city. Delhi has, in the past few years, made surprising strides toward improving its air quality. Data collected by the Central Pollution Control Board (CPCB) suggest that levels of suspended particles, possibly one of the main indicators for respiratory health, have at least stabilized and possibly even fallen. Despite popular resistance and widespread protests in 2001, when all public transport–related commercial vehicles were ordered to convert to compressed natural gas (CNG), residents of Delhi today believe there has been significant change, and they approve of it. Of the various measures taken to control air pollution levels in Delhi especially from vehicular emissions, the best known of these measures is the court-mandated conversion of all commercial passenger vehicles—buses, three-wheelers, and taxis—to compressed natural gas (CNG). The conversion of buses from diesel to CNG has helped to reduce PM10, CO, and SO_2 concentrations in the city.

Based on Air Quality Monitoring Data generated from Air Quality Monitoring Stations under NAMP, the trends in annual average concentration of Sulphur Dioxide (SO_2), Nitrogen Dioxide (NO_2) and Respirable Suspended Particulate Matter (RSPM) in Delhi have been derived and presented below. The Air Quality Trend information in metro cities generated following information.

Most of the pollutants from I.C. engines are emitted through the exhaust. However, some total organic compounds (TOC) escape from the crankcase as a result of blowby (gases that are vented from the oil pan after they have escaped from the cylinder past the piston rings) and from the fuel tank and carburetor because of evaporation. Nearly all of the TOCs from diesel C.I. engines enter the atmosphere from the exhaust. Crankcase blowby is minor because TOCs are not present during compression of the charge. Evaporative losses are insignificant in diesel engines due to the low volatility of diesel fuels. In general, evaporative losses are also negligible in engines using gaseous fuels because these engines receive their fuel continuously from a pipe rather than via a fuel storage tank and fuel pump.

The primary pollutants from internal combustion engines are oxides of nitrogen (NOx), hydrocarbons and other organic compounds, carbon monoxide (CO), and particulates, which include both visible (smoke) and nonvisible emissions. Nitrogen oxide formation is directly related to high pressures and temperatures during the combustion process and to the nitrogen content, if any, of the fuel.

The other pollutants, HC, CO, and smoke, are primarily the result of incomplete combustion. Ash and metallic additives in the fuel also contribute to the particulate content of the exhaust. Sulfur oxides also appear in the exhaust from IC engines. The sulfur compounds, mainly sulfur dioxide (SO_2), are directly related to the sulfur content of the fuel.

2.4.2 Nitrogen Oxides - Nitrogen oxide formation occurs by two fundamentally different mechanisms. The predominant mechanism with internal combustion engines is thermal NOx which arises from the thermal dissociation and subsequent reaction of nitrogen (N_2) and oxygen (O_2) molecules in the combustion air. Most thermal NOx is formed in the high-temperature region of the flame from dissociated molecular nitrogen in the combustion air. Some NOx, called prompt NOx, is formed in the early part of the flame from reaction of nitrogen intermediary species, and HC radicals in the flame. The second mechanism, fuel NOx, stems from the evolution and reaction of fuel-bound nitrogen compounds with oxygen. Gasoline, and most distillate oils, have no chemically-bound fuel N_2 and essentially all NOx formed is thermal NOx.

2.4.3 Total Organic Compounds

The pollutants commonly classified as hydrocarbons are composed of a wide variety of organic compounds and are discharged into the atmosphere when some of the fuel remains unburned or is only partially burned during the combustion process. Most unburned hydrocarbon emissions result from fuel droplets that were transported or injected into the quench layer during combustion. This is the region immediately adjacent to the combustion chamber surfaces, where heat transfer outward through the cylinder walls causes the mixture temperatures to be too low to support combustion. Partially burned hydrocarbons can occur because of poor air and fuel homogeneity due to incomplete mixing, before or during combustion; incorrect air/fuel ratios in the cylinder during combustion due to maladjustment of the engine fuel system; excessively large fuel droplets (diesel engines); and low cylinder temperature due to excessive cooling (quenching) through the walls or early cooling of the gases by expansion of the combustion volume caused by piston motion before combustion is completed.

2.4.4 Carbon Monoxide

Carbon monoxide is a colorless, odorless, relatively inert gas formed as an intermediate combustion product that appears in the exhaust when the reaction of CO to CO_2 cannot proceed to completion. This situation occurs if there is a lack of available oxygen near the hydrocarbon (fuel) molecule during combustion, if the gas temperature is too low, or if the residence time in the cylinder is too short. The oxidation rate of CO is limited by reaction kinetics and, as a consequence, can be accelerated only to a certain extent by improvements in air and fuel mixing during the combustion process.

2.4.5 Smoke, Particulate Matter, and PM-10

White, blue, and black smoke may be emitted from IC engines. Liquid particulates appear as white smoke in the exhaust during an engine cold start, idling, or low load operation. These are formed in the quench layer adjacent to the cylinder walls, where the temperature is not high enough to ignite the fuel. Blue smoke is emitted when lubricating oil leaks, often past worn piston rings, into the combustion chamber and is partially burned. Proper maintenance is the most effective method of preventing blue smoke emissions from all types of IC engines. The primary constituent of black smoke is agglomerated carbon particles (soot).

2.4.6 Sulfur Oxides

Sulfur oxide emissions are a function of only the sulfur content in the fuel rather than any combustion variables. In fact, during the combustion process, essentially all the sulfur in the fuel is oxidized to SO_2. The oxidation of SO_2 gives sulfur trioxide (SO_3), which reacts with water to give sulfuric acid (H_2SO_4), a contributor to acid precipitation. Sulfuric acid reacts with basic substances to give sulfates, which are fine particulates that contribute to PM-10 and visibility reduction. Sulfur oxide emissions also contribute to corrosion of the engine parts.

2.5 Emissions Control

Steps Taken to Control Vehicular Pollution

In recent years, several important initiatives have been taken to control pollution from vehicular emissions. Here an analysis of those steps is presented in a chronological order:

- CPCB started the vehicular pollution control activity as early as 1989 when The Ministry of Environment and Forest (MoEF) constituted an expert committee on vehicular emission, which for the first time notified the emission standards for both manufacturing stage and in-use vehicles under the Environment (Protection) Act in February 1990.

- Fuel Quality Specifications were laid down for the first time. Moreover the emission norms for vehicles were made more stringent as compared to those laid down in 1990. Gasoline lead and benzene levels were specified to be 0.15 gm/lt and 5% respectively. Sulphur content in diesel was brought down to 0.5%.
- In 1998, Government took important measures to curb air pollution due to vehicular emissions. These include- phasing out lead from gasoline, reduction of sulphur content in diesel to 0.25% and benzene content to 3%. Emission norms for catalytic converter were made more stringent and hot start was replaced by cold start. Banning of more than 15 years old vehicles and introduction of pre-mix 2T oil in retail outlets are other significant steps initiated in 1998.
- The campaign for vehicular pollution control gained momentum in 2000 with the introduction of EURO-II emission norms for passenger cars and EURO-I norms for other vehicles. Introduction of low smoke 2-T oil, reduction of sulphur content in diesel to 0.05% and gasoline benzene reduction to 1% were some of the major steps taken. Moreover phasing out more than 8 years old buses from Delhi, replacement of pre-1999 autos/taxis with vehicles on clean fuels and conversion of post 1999 autos to CNG mode were some of the initiatives taken in 2000.
- 2001 was marked by The Supreme Court's order of replacing passenger commercial vehicles in Delhi with CNG driven ones. Introduction of 0.05 % sulphur diesel was an important initiative taken in 2001.
- On 1 Jan 2002, came the much-awaited "Interim Report on Automobile Fuel Policy" by Mashelkar Committee. According to R A Mashelkar, CNG is an environmentally acceptable fuel which is suitable for Delhi; provide the appropriate engines are used. Then on 6 April, in a landmark judgment advancing the fundamental right to life, The Supreme Court imposed an exemplary fine of Rs.500 per day from 01 April on 8000 diesel run buses in Delhi for failing to convert to CNG. It also directed the Delhi Government to phase out 800 diesel buses per month from May 01.

Engine efficiency has been steadily improved with improved engine design, more precise ignition timing and electronic ignition, more precise fuel metering, and computerized engine management.

Advances in engine and vehicle technology continually reduce the toxicity of exhaust leaving the engine, but these alone have generally been proved insufficient to meet emissions goals. Therefore, technologies to detoxify the exhaust are an essential part of emissions control.

2.6 EMISSION NORMS

Emissions standards are requirements that set specific limits to the amount of pollutants that can be released into the environment. Many emissions standards focus on regulating pollutants released by automobiles (motor cars) and other powered vehicles but they can also regulate emissions from industry, power plants, small equipment such as lawn mowers and diesel generators. Frequent policy alternatives to emissions standards are technology standards (which mandate the use of a specific technology) and emission trading.

Standards generally regulate the emissions of NO_x, sulfur oxides, particulate matter (PM) or soot, carbon monoxide (CO), or volatile hydrocarbons (see carbon dioxide equivalent)

<u>CO_2 emission</u>

Within the European Union, road transport is responsible for about 20% of all CO_2 emissions, with passenger cars contributing about 12%.

The target fixed at Kyoto Protocol was an 8% reduction of emissions in all sectors of the economy compared to 1990 levels by 2008-2012.

Relative Carbon dioxide emissions from transport have risen rapidly in recent years, from 21% of the total in 1990 to 28% in 2004, but currently there are no standards for CO_2 emission limits for pollution from vehicles.

EU transport emissions of CO_2 currently account for about 3.5% of global CO_2 emissions.

Obligatory labeling

The purpose of Directive 1999/94/EC of the European Parliament and the Council of 13 December 1999 relating to the availability of consumer information on fuel economy and CO_2 emissions in respect of the marketing of new passenger cars is to ensure that information relating to the fuel economy and CO_2 emissions of new passenger cars offered for sale or lease in the Community is made available to consumers in order to enable consumers to make an informed choice.

In the United Kingdom, the initial approach was deemed ineffective. The way the information was presented was too complicated for consumers to understand. As a result, car manufacturers in the United Kingdom voluntarily agreed to put a more "consumer-friendly," colour-coded label displaying CO_2 emissions on all new cars beginning in September 2005, with a letter from A (<100 CO_2 g/km) to F (186+ CO_2 g/km) . The goal of the new "green label" is to give consumers clear information about the environmental performance of different vehicles. Other EU member countries are also in the process of introducing consumer-friendly labels.

Non-existent obligatory vehicle emission limits

The CO_2 emission limits generated by vehicles are nowadays subject to a voluntary agreement (in this differ from the obligatory limits in the U.S. CAFE legislation) between the EU and the auto manufacturers. The ultimate EU target with voluntary agreements are to contribute, is to reach an average CO_2 emission (as measured according to Commission Directive 93/116/EC) of 120 g/km for all new passenger cars by 2012.

However, as it becomes increasingly clear that the agreement will not deliver (having achieved only 160 g/km in 2005, from 186 g/km in 1995) lawmakers have started considering regulation.

In late 2005, the European Parliament passed a resolution in support for mandatory CO_2 emission standards to replace current voluntary commitments by the auto manufacturers and labeling.

In late 2006, in response to a new report, by the European Federation for Transport and Environment documenting lack of progress on the voluntary targets, the European

Commission announced that it was working on a proposal for legally-binding limit CO_2 emissions from cars. According to the mentioned European Federation for Transport and Environment study, Fiat is the best performer in Europe.

On 7 February 2007 the European Commission published its key draft proposal (COM 2007 0019) EC legislation to limit average CO_2 emissions from the European fleet of cars to 120 g CO_2/km. Some people interpreted this as meaning that all manufacturers would have to average 120 g for their fleet, but this is not the case. Some volume manufacturers of smaller cars such as Fiat, Renault and Peugeot-Citroen are already quite close to the target whilst smaller volume manufacturers of higher emissions cars such as BMW, Mercedes, Audi, Saab and Porsche are a long way from reaching this target. Not surprisingly the French and Italian manufacturers want a blanket target whereas the German manufacturers feel a blanket target would destroy their industries.

The environmental group T&E insists on the need for a longer-term target that doubles fuel efficiency of new cars over the next decade, 80 g/km by 2020. It says new-car emissions from European producers slipped to 160 grams per kilometer (g/km) on average last year (reduced only 0.2 percent in 2006), still way off a voluntary goal of 140 g/km by 2008.

<u>Non-CO_2 fuel emission: stages and legal framework</u>

The stages are typically referred to as Euro 1, Euro 2, Euro 3, Euro 4 and Euro 5 fuels for Light Duty Vehicle standards. The corresponding series of standards for Heavy Duty Vehicles uses Roman, rather than Arabic numerals (Euro I, Euro II, etc.)

The legal framework consists in a series of directives, each amendment to the 1970 Directive 70/220/EEC. Here is a summary list of the standards, when they come into force, what they apply to, and which EU directives provide the definition of the standard.

- Euro 1 (1993):
    - For passenger cars - 91/441/EEC.
    - Also for passenger cars and light trucks - 93/59/EEC.
- Euro 2 (1996) for passenger cars - 94/12/EC (& 96/69/EC)

- Euro 3 (2000) for any vehicle - 98/69/EC
- Euro 4 (2005) for any vehicle - 98/69/EC (& 2002/80/EC)
- Euro 5 (2008/9) and Euro 6 (2014) for light passenger and commercial vehicles – 2 007/715/EC

These limits supersede the original directive on emission limits 70/220/EEC.

The classifications for vehicle types are defined by:

- Commission Directive 2001/116/EC of 20 December 2001, adapting to technical progress Council Directive 70/156/EEC on the approximation of the laws of the Member States relating to the type-approval of motor vehicles and their trailers
- Directive 2002/24/EC of the European Parliament and of the Council of 18 March 2002 relating to the type-approval of two or three-wheeled motor vehicles and repealing Council Directive 92/61/EEC

In the area of fuels, the 2001 Biofuels Directive requires that 5.75% of all transport fossil fuels (petrol and diesel) should be replaced by biofuels by 31 December 2010, with an intermediate target of 2% by the end of 2005. However, MEPS have since voted to lower this target in the wake of new scientific evidence about the sustainability of biofuels and the impact on food prices. In a vote in Strasbourg, the European parliament’s environment committee supported a plan to curb the EU target for renewable sources in transport to 4% by 2015. They also said that a thorough review would be required in 2015 before the EU could progress to an 8-10% mark by 2020.

2.7 EUROPEAN EMISSION STANDARDS

European emission standards for passenger cars (Category M_1*), g/km

Table 2.7: Emissions Standards for Passenger Cars

Tier	Date	CO	HC	NO_x	HC+NO_x	PM
Diesel						
Euro 1†	July 1992	2.72 (3.16)	-	-	0.97 (1.13)	0.14 (0.18)
Euro 2	January 1996	1.0	-	-	0.7	0.08
Euro 3	January 2000	0.64	-	0.5	0.56	0.05
Euro 4	January 2005	0.5	-	0.25	0.3	0.025
Euro 5 (future)	September 2009	0.5	-	0.18	0.23	0.005
Euro 6 (future)	September 2014	0.5	-	0.08	0.17	0.005
Petrol (Gasoline)						
Euro 1†	July 1992	2.72 (3.16)	-	-	0.97 (1.13)	-
Euro 2	January 1996	2.2	-	-	0.5	-
Euro 3	January 2000	2.3	0.2	0.15	-	-
Euro 4	January 2005	1.0	0.1	0.08	-	-
Euro 5 (future)	September 2009	1.0	0.1	0.06	-	0.005**
Euro 6 (future)	September 2014	1.0	0.1	0.06	-	0.005**

* Before Euro 5, passenger vehicles > 2500 kg were type approved as light commercial vehicle N1 – I

** Applies only to vehicles with direct injection engines

† Values in brackets are conformity of production (COP) limits

Emission standards for light commercial vehicles

European emission standards for light commercial vehicles ≤1305 kg (Category N1 - I), g/km

Table 2.8: Emissions Standards for N1-I

Tier	Date	CO	HC	NO_x	$HC+NO_x$	PM
Diesel						
Euro 1	October 1994	2.72	-	-	0.97	0.14
Euro 2	January 1998	1.0	-	-	0.7	0.08
Euro 3	January 2000	0.64	-	0.5	0.56	0.05
Euro 4	January 2005	0.5	-	0.25	0.3	0.025
Euro 5 (future)	September 2009	0.5	-	0.18	0.23	0.005
Euro 6 (future)	September 2014	0.5	-	0.08	0.17	0.005
Petrol (Gasoline)						
Euro 1	October 1994	2.72	-	-	0.97	-
Euro 2	January 1998	2.2	-	-	0.5	-
Euro 3	January 2000	2.3	0.2	0.15	-	-
Euro 4	January 2005	1.0	0.1	0.08	-	-
Euro 5 (future)	September 2009	1.0	0.1	0.06	-	0.005*
Euro 6 (future)	September 2014	1.0	0.1	0.06	-	0.005*
* Applies only to vehicles with direct injection engines						

European emission standards for light commercial vehicles 1305 kg – 1760 kg (Category N1 - II), g/km

Table 2.9: Emissions standard for N1-II

Tier	Date	CO	HC	NO_x	$HC+NO_x$	PM
Diesel						
Euro 1	October 1994	5.17	-	-	1.4	0.19
Euro 2	January 1998	1.25	-	-	1.0	0.12
Euro 3	January 2001	0.8	-	0.65	0.72	0.07
Euro 4	January 2006	0.63	-	0.33	0.39	0.04
Euro 5 (future)	September 2010	0.63	-	0.235	0.295	0.005
Euro 6 (future)	September 2015	0.63	-	0.105	0.195	0.005
Petrol (Gasoline)						
Euro 1	October 1994	5.17	-	-	1.4	-
Euro 2	January 1998	4.0	-	-	0.65	-
Euro 3	January 2001	4.17	0.25	0.180	-	-
Euro 4	January 2006	1.81	0.13	0.1	-	-
Euro 5 (future)	September 2010	1.81	0.13	0.075	-	0.005*
Euro 6 (future)	September 2015	1.81	0.13	0.075	-	0.005*
* Applies only to vehicles with direct injection engines						

European emission standards for light commercial vehicles >1760 kg max 3500 kg. (Category N1 - III), g/km

Table 2.10: Emissions standard for N1-III

Tier	**Date**	**CO**	**HC**	**NO_x**	**$HC+NO_x$**	**PM**
Diesel						
Euro 1	October 1994	6.9	-	-	4.9	0.25
Euro 2	January 1998	1.5	-	-	0.96	0.17
Euro 3	January 2001	0.95	-	0.780	0.86	0.1
Euro 4	January 2006	0.95	-	0.39	0.46	0.06
Euro 5 (future)	September 2010	0.74	-	0.28	0.35	0.005
Euro 6 (future)	September 2015	0.74	-	0.125	0.215	0.005
Petrol (Gasoline)						
Euro 1	October 1994	6.9	-	-	1.7	-
Euro 2	January 1998	5.0	-	-	0.8	-
Euro 3	January 2001	5.22	0.29	0.210	-	-
Euro 4	January 2006	2.27	0.16	0.110	-	-
Euro 5 (future)	September 2010	2.27	0.16	0.082	-	0.005*
Euro 6 (future)	September 2015	2.27	0.16	0.082	-	0.005*

Emission standards for Lorries and Buses

Whereas for passenger cars, the standards are defined in g/km, for lorries (trucks) they are defined by engine power, g/kWh, and are therefore in no way comparable. The following table contains a summary of the emission standards and their implementation dates. Dates in the tables refer to new type approvals; the dates for all type approvals are in most cases one year later (EU type approvals are valid longer than one year).

The official category name is heavy-duty diesel engines, which generally includes lorries and buses.

EU Emission Standards for HD Diesel Engines, g/kWh (smoke in m^{-1})

Table 2.11: Emissions standard for HD Diesel

Tier	**Date**	**Test cycle**	**CO**	**HC**	**NO_x**	**PM**	**Smoke**
Euro I	1992, < 85 kW	ECE R-49	4.5	1.1	8.0	0.612	
	1992, > 85 kW		4.5	1.1	8.0	0.36	
Euro II	October 1996		4.0	1.1	7.0	0.25	
	October 1998		4.0	1.1	7.0	0.15	
Euro III	October 1999 EEVs only	ESC & ELR	1.0	0.25	2.0	0.02	0.15
	October 2000	ESC & ELR	2.1	0.66	5.0	0.100.13*	0.8
Euro IV	October 2005		1.5	0.46	3.5	0.02	0.5
Euro V	October 2008		1.5	0.46	2.0	0.02	0.5

* For engines of less than 0.75 dm³ swept volume per cylinder and a rated power speed of more than 3,000 per minute. EEV is "Enhanced environmentally friendly vehicle".

Emission standards for Large Goods Vehicles

Table 2.12: Euro Norm Emissions for category N2, EDC (2000 and up)

Standard	**Date**	**CO (g/kWh)**	**NO_x (g/kWh)**	**HC (g/kWh)**	**PM (g/kWh)**
Euro 0	1988-1992	12.3	15.8	2.6	none
Euro I	1992-1995	4.9	9.0	1.23	0.40
Euro II	1995-1999	4.0	7.0	1.1	0.15
Euro III	1999-2005	2.1	5.0	0.66	0.1
Euro IV	2005-2008	1.5	3.5	0.46	0.02
Euro V	2008-2012	1.5	2.0	0.46	0.02
Euro VI					

Emission Norms in India

An Overview

1991 - Idle CO Limits for Gasoline Vehicles and Free Acceleration Smoke for Diesel Vehicles, Mass Emission Norms for Gasoline Vehicles.

1992 - Mass Emission Norms for Diesel Vehicles.

1996 - Revision of Mass Emission Norms for Gasoline and Diesel Vehicles, mandatory fitment of Catalytic Converter for Cars in Metros on Unleaded Gasoline.

1998 - Cold Start Norms Introduced.

2000 - India 2000 (Eq. to Euro I) Norms, Modified IDC (Indian Driving Cycle), Bharat Stage II Norms for Delhi.

2001 - Bharat Stage II (Eq. to Euro II) Norms for All Metros, Emission Norms for CNG & LPG Vehicles.

- 2003 - Bharat Stage II (Eq. to Euro II) Norms for 11 major cities.
- 2005 - From 1 April Bharat Stage III (Eq. to Euro III) Norms for 11 major cities.

2010 - Bharat Stage III Emission Norms for 4-wheelers for entire country whereas Bharat Stage - IV (Eq. to Euro IV) for 11 major cities. Bharat Stage IV also has norms on OBD (similar to Euro III but diluted).

Table 2.13: Indian Emissions Standards (4 Wheelers)

Standard	Reference	Date	Region
India 2000	Euro 1	2000	Nationwide
Bharat Stage II	Euro 2	2001	NCR*, Mumbai, Kolkata, Chennai
		2003.04	NCR*, 10 Cities†
		2005.04	Nationwide
Bharat Stage III	Euro 3	2005.04	NCR*, 10 Cities†
		2010.04	Nationwide
Bharat Stage IV	Euro 4	2010.04	NCR*, 10 Cities†

* National Capital Region (Delhi)

† Mumbai, Kolkata, Chennai, Bengaluru, Hyderabad, Ahmedabad, Pune, Surat, Kanpur and Agra

The above standards apply to all new 4-wheel vehicles sold and registered in the respective regions. In addition, the National Auto Fuel Policy introduces certain emission requirements for interstate buses with routes originating or terminating in Delhi or the other 10 cities.

For 2-and 3-wheelers, Bharat Stage II (Euro 2) is applicable from April 1, 2005 and Stage III (Euro 3) standards would come in force preferably from April 1, 2008, but not later than April 1, 2010 .[29]

Enhanced environmentally friendly vehicle

Enhanced environmentally friendly vehicle or EEV is a term used in the European emission standards for the definition of a ‘clean vehicle’ > 3.5 tonne in the category M2 and M3. The standard lies between the levels of Euro V and Euro VI.

Cycle beating

For the emission standards to deliver real emission reductions it is crucial that the test cycles under which the emissions have to comply as much as possible reflect normal driving situations. It was discovered that engine manufacturers would engage in what was called 'cycle beating' to optimize emission performance to the test cycle, while emissions from typical driving conditions would be much higher than expected, undermining the standards and public health. In one particular instance, research from two German technology institutes found that for diesel cars no 'real' NO_x reductions have been achieved after 13 years of stricter standards.

CHAPTER 3

VCR ENGINE

3.1 INTRODUCTION

The concept of variable compression ratio (VCR) promises improved engine performance, efficiency, and reduced emissions. The higher cylinder pressures and temperatures during the early part of combustion and small residual gas fraction owing to higher compression ratio give faster laminar flame speed. Therefore, the ignition delay period is shorter. As a result, at low loads, the greater the compression ratio, the shorter is the combustion time. The time loss is subsequently reduced. Therefore, it seems reasonable that fuel consumption rate is lower with high compression ratios at part load.

The VCR or, more correctly, variable expansion ratio can make a significant contribution to thermo dynamic efficiency. The main feature of the VCR engine is to operate at different compression ratios, depending on the vehicle performance needs. A VCR engine can continuously vary the compression ratio by changing the combustion chamber volume. In a VCR engine, thermodynamic benefits appear throughout the engine map. At low power levels, the VCR engine operates at a higher compression ratio to capture high fuel efficiency benefits, while at high power levels the engine operates at low compression ratio to prevent knock. The optimum compression ratio is determined as a function of one or more vehicle operating parameters such as inlet air temperature, engine coolant temperature, exhaust gas temperature, engine knock, fuel type, octane rating of fuel, etc. In a VCR engine, the operating temperature is more or less maintained at optimum, where combustion efficiency is high. It has been proven that a VCR engine develops much more power for the same engine dimensions, i.e. it is very compact and has a high power-to-weight ratio without any penalty on specific fuel consumption. In other words, reducing the engine capacity at the same power leads to reduction in fuel consumption owing to reduced pumping, friction, and heat losses.

3.2 NEED FOR VCR ENGINE

The present challenge in automotive engine technology is the improvement of thermal efficiency and hence the fuel economy and lower emission levels. One of the key features affecting thermal efficiency is the compression ratio, which is always a compromise in fixed compression ratio spark ignition (SI) engines the formula for air standard cycle efficiency is

$$\eta = 1 - [1/r_c^{(\gamma - 1)}]$$

Where η is the efficiency of the cycle; r_c is the compression ratio; and γ is the ratio of specific heat of air at constant volume to specific heat at constant pressure, or approximately 1.40. Higher compression ratio results in higher thermal efficiency and improved fuel economy in the internal combustion engine. Generally, the operating conditions of SI engines vary widely, such as stop and go city traffic, highway motoring at constant speed, or high-speed freeway driving. In a conventional SI engine, the maximum compression ratio is set by the conditions in the cylinder at high load, when the fuel and air consumption are at maximum levels. If the compression ratio is higher than the designed limit, the fuel will pre-ignite causing knocking, which could damage the engine. Unfortunately, most of the time SI engines in city driving conditions operate at relatively low power levels under slow accelerations, low speeds, or light loads, which lead to low thermal efficiency and hence higher fuel consumption. As the engine load decreases, the temperature in the end gas drops, so that high compression ratio could be employed without the risk of knocking in naturally aspirated or boosted engines. Raising the compression ratio from 8 to 14 produces an efficiency gain from 50 to 65 per cent (a 15 per cent gain), whereas going from 16 to 20 produces a gain from 67 to 70 per cent (a 3 per cent gain). Figure 3.1 shows the effects of compression ratio with respect to thermal efficiency.

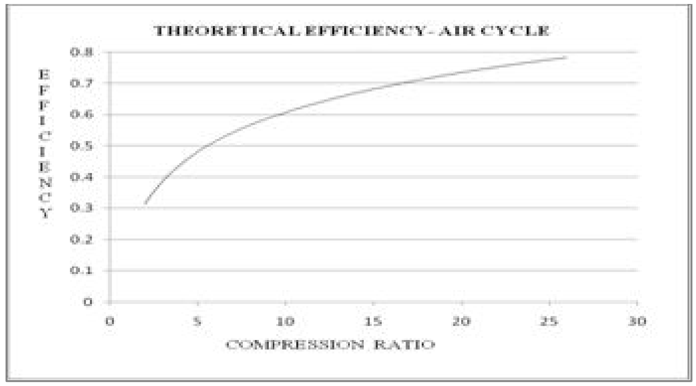

Figure 3.1: Effect of Compression Ratio on Thermal Efficiency

The characteristics that have the dominant effects on thermal efficiency are compression ratio and air–fuel mixture strength. The fuel–air cycle efficiency increases with the compression ratio in the same manner as the air standard cycle efficiency, principally for the same reason, i.e. more scope for expansion work. Figure 3.2 shows the influence of 'per cent theoretical fuel' (fuel–air mixture strength) on thermal efficiency at different compression ratios [31]. The 'per cent theoretical fuel' is the ratio of actual fuel–air ratio to chemically correct fuel–air ratio, in percentage terms. The maximum output is obtained when the air–fuel capacity of the engine is utilized, i.e. when the maximum amount of fuel can be burnt efficiently. The maximum brake mean effective pressure (b.m.e.p.) in SI engines is 12 bar whereas in diesel engines it is 18 bar. A higher compression ratio increases the pressure and temperature of the working air–fuel mixture, which increases the tendency of the engine to knock. For knock, key variables are: end-gas temperature, pressure, and composition; time/speed; and fuel octane rating [7].

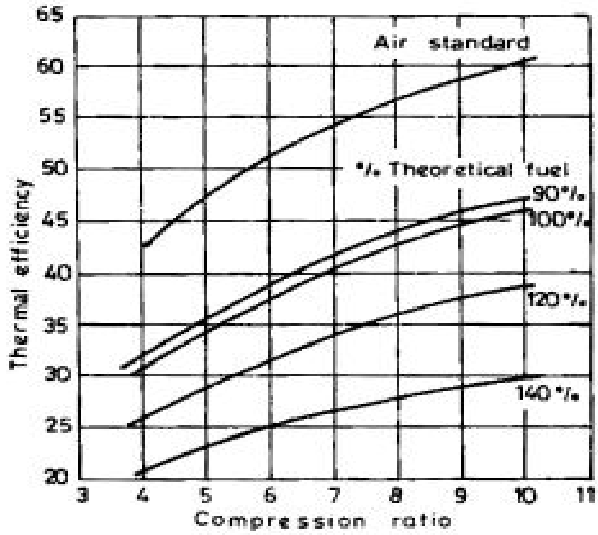

Figure 3.2: Effect of Compression Ratio and Mixture Strength on Thermal Efficiency

3.3 VARIOUS VCR APPROACHES

Designing and successfully developing a practical VCR engine has long been a challenge to the automobile industry. Many innovative patents have been filed and different designs developed to modify the compression ratio. A few approaches are discussed below.

3.3.1 Moving the crankshaft axis

FEV, Germany has chosen to alter the position of the crankshaft. In their engine, crankshaft bearings are carried in an eccentrically mounted carrier that can rotate to raise or lower the top dead centre (TDC) positions of the pistons in the cylinders (Fig. 3.3). The compression ratio is adjustable by varying the rotation of the eccentric carrier. Mounting the crankshaft on eccentric bearings is simple in that the reciprocating assembly itself is unchanged. In fact, the engine requires an offset fixed-position output shaft; a coupling is required between the movable crankshaft end and the fixed output shaft. The compression ratio is adaptable from 8 to 14 approximately by varying the rotation of the eccentric carriers through 55° [7].

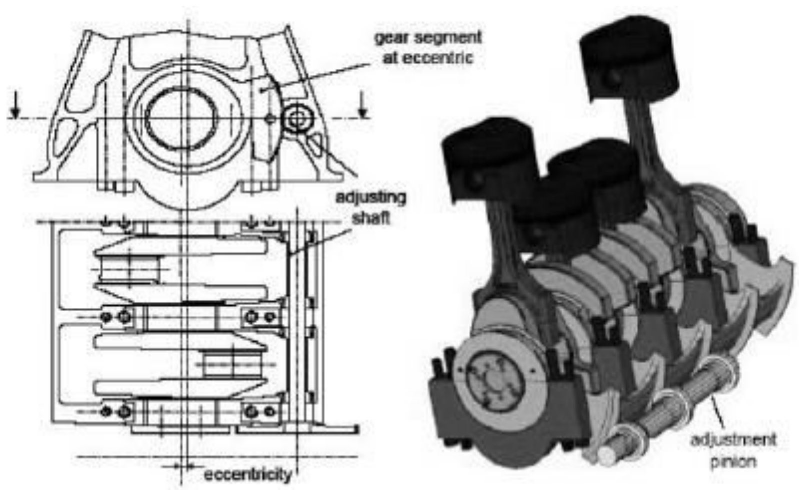

Figure. 3.3: Eccentric Crankshaft Mounting

3.3.2 Modification of the connecting rod geometry

The Nissan project uses a multi-link system to achieve VCR by inserting a control linkage system between the connecting rod and the crankshaft, and connecting this to an actuator shaft, so that the compression ratio can be varied. This project was incorporated in a four-cylinder engine without major modification of the engine block. The shorter crank throw allowed room for the link system, which was anchored by an eccentric rotary actuator. Compression was varied from 10 to 15 approximately by a 70° rotation of the actuator, while at TDC; the piston position was changed by 3.1 mm (Fig. 3.4). Examining the details of multi-link system operation reveals some advantages. The most striking advantage is that of maximum piston accelerations. Tension forces acting through the connecting rod and piston at TDC represent one of the factors limiting piston speed, so a geometry that reduces the peak piston acceleration would allow either an increase in sustainable engine speed or an increasing stroke, either of which is useful in terms of increasing power output [32].

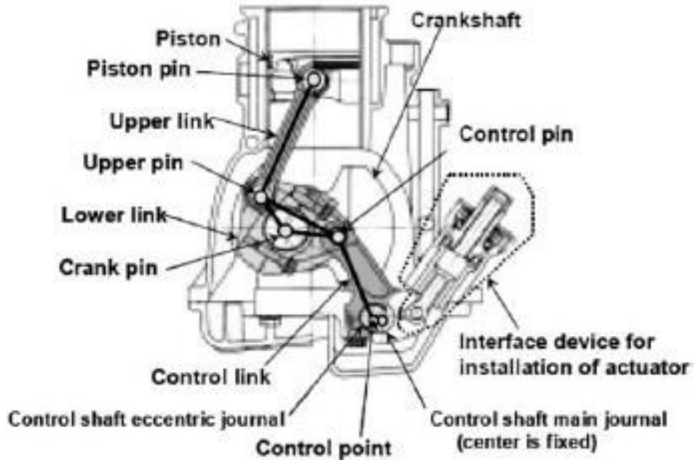

Figure. 3.4: Multi-link VCR Configuration

3.3.3 Moving the cylinder head

The moving head concept (Saab Automobile AB) combines a cylinder head with cylinder liners into a monohead construction, which pivots with respect to the remainder of the engine. The lower half of the block includes the crankcase and engine mounts, and carries the crankshaft, gear box, oil cooler, and auxiliaries. The upper half includes the cylinders, their liners, camshafts, and an integrally cast cylinder head. This part is referred to as the monohead (Fig. 3.5). Saab has enabled a tilting motion to adjust the effective height of the piston crown at TDC. The linkage serves to tilt the monohead relative to the crankcase in order to vary the TDC position of the piston. By means of actuator and linkage mechanism the compression ratio can be varied from 8 to 14. A screw type supercharger provides a 2:1 boost pressure when wide open throttle conditions occur[33]. This system gives wide fuel flexibility, with reduced CO_2 emissions proportional to fuel consumption. Saab recognized that the fuel efficiency of the VCR engine would be low without high-pressure supercharging.

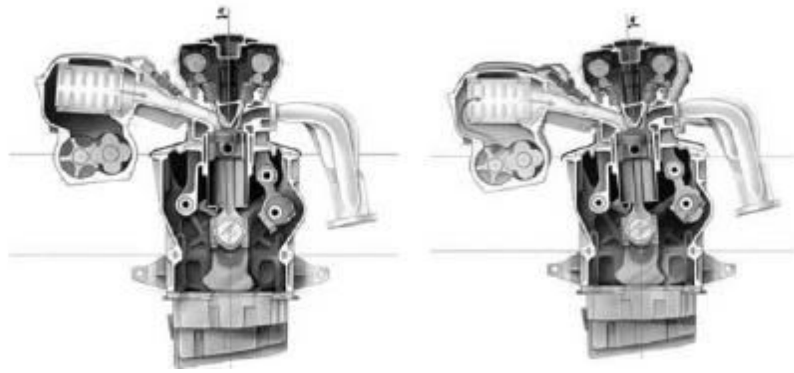

Figure. 3.5: The Saab VCR engine

3.3.4 Variation of combustion chamber volume using a secondary piston or valve

Ford has patented a means to vary combustion chamber volume by using a secondary piston or valve (Fig. 3.6). The piston could be maintained at an intermediate position, corresponding to the optimum compression ratio for a particular condition. The volume of the combustion chamber is increased to reduce the compression ratio by moving a small secondary piston which communicates with the chamber [37]. However, this would require a finite length bore in which the piston could travel, which raises questions of sealing, packaging, and durability varying combustion chamber geometry compromises the area available for intake and exhaust valves, while moving the cylinder head and barrel is feasible in a research engine but harder to accomplish in a production vehicle. The cylinder head cooling needs to be improved by an efficient cooling system and the auxiliary piston needs proper lubrication for efficient functioning of the VCR engine.

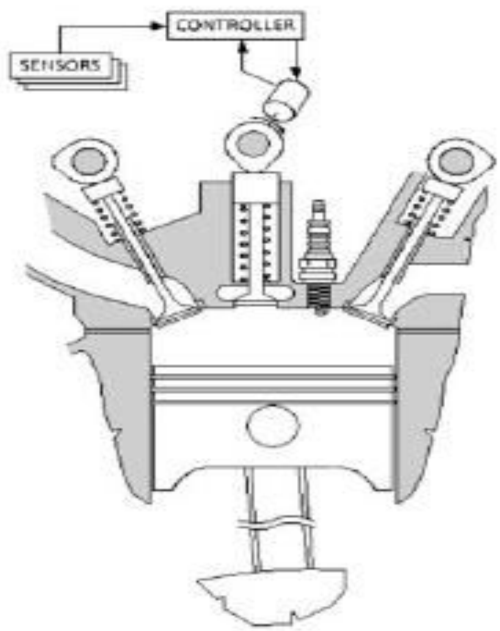

Figure. 3.6: Ford VCR Engine

3.3.5 Variation of piston deck height

The Daimler-Benz VCR piston design shows variation in compression height of the piston and offers potentially the most attractive route to a production VCR engine, since it requires relatively minor changes to the base engine architecture when compared to other options (Fig. 3.7). Unfortunately, it requires a significant increase in reciprocating mass and, more importantly, a means to activate the height variation within a high-speed reciprocating assembly[38]. This is typically proposed by means of hydraulics using the engine lubricating oil; however, reliable control of the necessary oil flow represents a major challenge. This is claimed to reduce the peak firing loads so that the compression ratio variation becomes self acting rather than externally controlled. A side-effect would be the momentary variation in clearance volume during the combustion event, which would, in turn, increase, then reduce the volume available to the expanding gases.

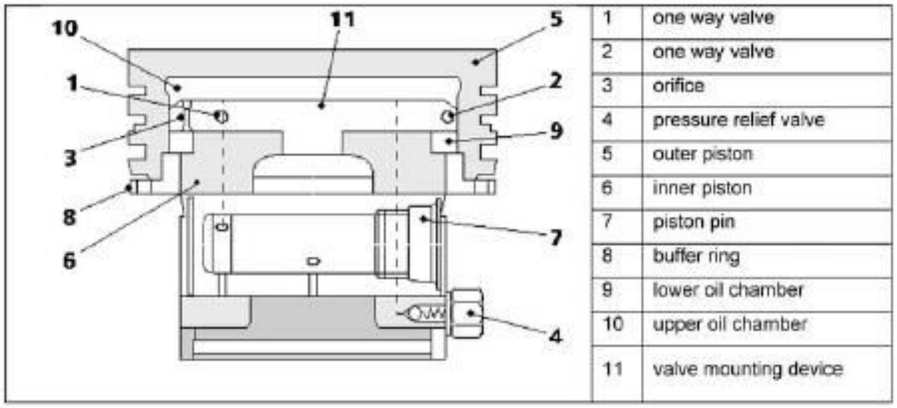

Figure. 3.7: Daimler-Benz VCR Piston

3.3.6 <u>Pressure reactive piston</u>

The University of Michigan developed a pressure reactive piston for SI engines. The pressure-reactive piston assembly consists of a piston crown and a separate piston skirt, with a set of springs contained between them (Fig. 3.8). This piston configuration allows the piston crown to deflect in response to the cylinder pressure. As a piston crown deflects, the cylinder clearance volume increases, lowering the effective compression ratio and reducing peak cylinder pressure. This mechanism effectively limits the peak pressures at high loads without an additional control device, while allowing the engine to operate at high compression ratio during low load conditions [39]. It can be easily adapted to the conventional engine with only changes to piston and connecting rod design. Brake specific fuel consumption improvements of the pressure-reactive piston engine over baseline engine at light loads ranges from 8 to 18 per cent. The pressure-reactive piston shows higher heat transfer losses because of higher surface-to-volume ratio and produces higher hydrocarbon emission at part load owing to higher compression ratio and more crevice volume (piston crown design).

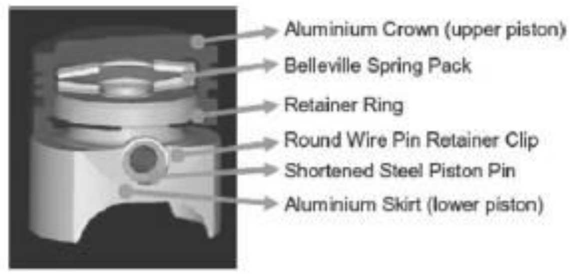

Figure. 3.8: Pressure-Reactive Piston Cross-Section

3.3.7 Moving the crankpins

Gomecsys has proposed to move the crankpins eccentrically to effect a stroke change at TDC. Figure 3.9 shows the Gomecsys VCR engine in which moveable crankpins form an eccentric sleeve around the conventional crankpins and are driven by a large gear [37]. Differences in the TDC position may vary up to 10 mm with a rotation of the ring-gear of only 40°. By rotating the ring-gear slightly to the right or to the left, while the crankshaft is at the TDC position at the end of the compression stroke, the position of the eccentric can be lifted or lowered. Note that lifting the eccentric at one TDC automatically causes the other TDC to be lowered accordingly. In order to effectively downsize the engine, a two-cylinder inline engine is a perfect solution for small cars; the two- cylinder Go Engine concept is small and lightweight, and total power train costs are comparable with a small four-cylinder engine. Applications involving staggered crankpin geometry would be less elegant, requiring multiple gear drives [38].

Figure. 3.9: Gomecsys VCR Engine

Table 3.1: Summary of Various VCR Approaches [39]

Serial no.	Geometric approach	Unique feature	Commercial barrier
1	Moving the crankshaft axis	The engine crankshaft bearings are carried in an eccentrically mounted carrier and the compression ratio is adjustable by varying the rotation of the eccentric carrier	The engine requires an offset fixed-position output shaft; a coupling is required between the movable crankshaft and the fixed output shaft
2	Modification of connecting rod geometry	Effective length of the connecting rod is varied by a multi-link system to change the compression ratio of the engine	Leads to increase in vibration owing to intermediate members in the connecting rod
3	Moving the cylinder head	The compression ratio of the engine is varied by adjusting the slope of the upper half of the engine in relation to the lower half	Reworking of the entire engine structure is necessary
4	Variation of combustion chamber volume using a secondary piston or valve	The volume of the combustion chamber is varied by moving a small secondary piston or valve, which communicates with the chamber	Introduction of additional elements within the crowded combustion chamber environment threatens to compromise ideal geometry and layout of the valves and ports. Engine-out emission performance is likely to be undetermined by additional crevice volumes
5	Variation of piston deck height	Variation in total height of the piston by means of hydraulics or springs to change the cylinder clearance volume	Significant increase in reciprocating mass and higher heat transfer losses owing to higher surface-to-volume ratio
6	Moving the crankpin	Vary the TDC positions by moveable crankpins, driven by large ring-gears	Staggered crankpin geometry would be less elegant, requiring multiple gear drives

3.4 EFFECT OF COMPRESSION RATIO ON POWER OUTPUT AND EFFICIENCY

The power delivered from an engine is directly proportional to the mean effective pressure (mep) developed in the engine cylinder. The fuel conversion efficiency, i.e. sum of thermal and mechanical efficiency, increases with increase in compression ratio, implying the power output increases with compression ratio under a given set of operating conditions.

From the relationships [1] on engine performance we have:

Power = $\eta_f \eta_v N V_d Q_{h\vartheta} \rho_a \lambda / 2$ (1)

Mean effective pressure

= $\eta_f \eta_v N V_d Q_h \rho_a \lambda$ (2)

Otto Cycle Efficiency = $1 - 1/ r_c^{\gamma-1}$ (3)

where, η_f, η_v, V_d Q_h, ρ_a and λ are the fuel conversion efficiency, volumetric efficiency, displacement volume, lower heating value of the fuel and air density, and fuel to air ratio respectively. N is the engine speed (rpm). r_c, is the compression ratio and γ the ratio of specific heat.

From the above expressions (1) – (3), power output is related to the compression ratio through the mean effective pressure and the cycle efficiency. Heywood [2] indicates that only few studies have focused on the effect of compression ratio on engine performance and efficiency over a wide range of compression ratios in the case of a spark ignited engines. The basic limitation in using higher compression ratio for spark-ignited engines arises out of the properties of the fuel used. The range of operating compression ratio found in the literature is between 8 and 14. For spark-ignited engines with the compression ratio less than 12, for a unit change of compression ratio, the output changes by about 3 % [23].

In order to establish the effect of compression ratio on the power, data on Cummins gas engines is used from published data [24]. Using the data, the engine power output is estimated for different derating factors; 1 – 3 % in power output per unit change in compression ratio. Figure 3.10 shows the plot of percentage error for various models at 1, 2 and 3 % power change for unit change in compression ratio.

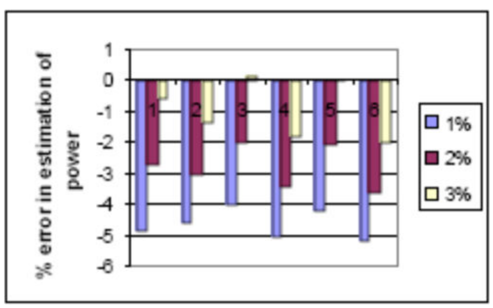

Figure 3.10: Plot of % error in estimation of power for different engine model at for 1, 2 and 3 % loss of power per unit change in compression ratio

Error is calculated comparing the estimated power with the rated power indicated by the manufacturer for various compression ratios. For the various engine models, the calculated error is minimum at 3 % change in power for a unit change in compression ratio. The negative values indicate the over estimation of power. Using this information, conversion factors for the power output from commercial gas engines can be established. Based on these estimates, a 3 % change in power occurs for a unit change in compression ratio is chosen for further analysis in the paper.

From the simple analysis a fairly accurate estimation of the power output from engines is possible, if the compression ratios of all the engines and the output of one of the engines are known. Thus the scaling law for correcting the compression ratio appears reasonably well established.

Engine efficiency also gets affected due to the change in compression to similar extent [23]. Depending upon the cylinder sizes and the operating conditions, Heywood indicates that, for a unit change in the compression ratio in the range of 9 to 11, the relative change in efficiency is between 1 and 3 percent. Further in a detailed work carried out the by Kerley and Thruston [25], the effective change in efficiency is found to be in the range of 1 – 1.4 % per unit change in compression ratio. It is also found from the work of Sridhar et al [18], that the factor is about 1.3 per unit change in compression ratio with different operating conditions.

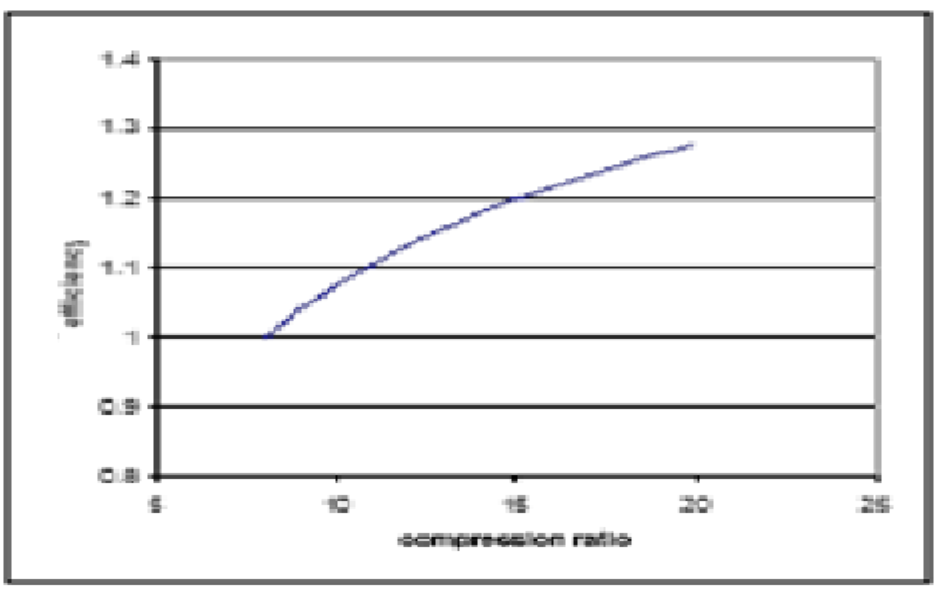

Figure3.11 Variation of Efficiency with Compression Ratio [Ref.21]

Fig.3.11 shows the variation of normalized efficiency with compression ratio using the ideal efficiency relation (eq. 3) for $\gamma = 1.4$. Efficiency obtained with compression ratio of 8 is used to normalize the efficiency calculated using other compression ratios. From the experimental analysis Heywood presents the details of an 8-cylinder engine with wide open throttle having similar features as in Figure 2.4. The change in efficiency is about 3 % per unit change in compression ratio in the range of $r_c < 12$ and is about 1.8 to 2.4 in the range of compression ratio between 12 and 17. Based on the results available from other sources presented here and the non availability of any commercial data in this regard, it is assumed a change of a 1.5 % in efficiency for unit change of compression ratio for the gas engines under consideration.

3.5 EFFECT OF FUEL CALORIFIC VALUE ON POWER

The energy content in the air-fuel mixture has an effect on the output of the engine. Depending upon the calorific value and the stoichiometric air requirement, the energy content in the engine cylinder, which is denoted by the energy density (MJ/m^3), determines the power developed in that cylinder. In order to compare the performance of the engine using different fuel, the energy densities for various fuel is obtained as indicated below.

Diesel

Based on the stoichiometric requirements, the A/F for diesel ($C_{14.4}H_{24.4}$) is 15 and with an excess air factor of 15 %, the airflow per unit weight of the diesel used is 18. This is about the range of designed air availability for most of the high - speed engines operating at nominal ratings. In the cylinder, the energy density at the nominal rating of the engine would be about 2.83 MJ/m^3 of the mixture.

Natural gas

At stoichiometry, the A/F requirement for natural gas is about 17. Excess air factor in the range of 5% for gaseous fuel combustion results in an air requirement of say 18 kg per kg of natural gas. With lower calorific value of 45 MJ/kg the energy density inside the cylinder volume would be about 3.0 MJ/m^3.

Producer gas

Using the similar logic as that of natural gas, evaluating producer gas fuel in engines, we have the following; the calorific value of the gas taken at 5.2 MJ/kg would result in an energy density of 2.4 MJ/m^3. This value is lower than that of diesel and about 75 % the value of natural gas. [25]

Biogas

The calorific value of biogas with 75 % methane and 25 % carbon dioxide is about 23.6 MJ/kg. At stoichiometry, A/F is about 10.5 and with 5 % excess air, the energy density is 2.3 MJ/m^3. This value is lower by about 5 % in comparison to producer gas. Table 2.3 shows the energy densities of various fuels. It is clear that the energy density of natural gas is higher than other gaseous fuels.

Table 3.2: Properties of Various Fuels

Fuel	**Lower heating value MJ/kg**	**A/F**	**Energy density MJ/ m^3**
Diesel	42.5	18	2.83
Natural Gas	45.0	18	3.00
Producer Gas	5.2	1.2	2.40
Biogas	23.6	11	2.30

The usage of natural gas as a vehicle fuel was discovered back in the early 1920s in Italy. The usage was not popular then due to the fact that natural gas was more commonly used in domestic and industry heating as well as to generate electricity. However, after the World War 2, there is growing interest on the usage of natural gas as vehicle fuel. This interest had led to establishment of approximately 1200 refueling stations and 1500 sub stations for natural gas in Italy by the early 1950s [40].

In 1991, Italy became the leading country in the research of natural gas vehicles. Italy had about 235000 gasoline vehicles and 20 diesel vehicles that were converted to natural gas. Natural gas is compressed in a high pressure tank of 18-20MPa to form compressed natural gas (CNG). The country with the second highest natural gas vehicles is Argentina, which has about 100000 gasoline vehicles and 10 converted diesel vehicles [41].

By 2003, Argentina overtook Italy and became the leading country in the world to convert vehicles to use CNG (Refining Margins Dip, then Recover, April 2004). The trend towards converting gasoline and diesel vehicles to use CNG is still quite unpopular in Malaysia, compared to more developed western countries.

However, due to the limitation of the crude petroleum oil reserves, which should last for another 15 years, Malaysia has since resolved to do more researches and experiments to use alternative fuels like natural gas. This is because the country’s natural gas reserves would last for about 80 – 90 years. Review showed that up until December 1994, there were about 900 vehicles converted to use CNG as fuel in Malaysia.

Hence, from the history of natural gas, the expansion of the usage of natural gas in automotive industry becomes clearer.

CHAPTER 4

DEVELOPMENT OF EXPERIMENTAL SETUP

4.1 DIESEL - CNG AND BIODIESEL - CNG PROJECT

In this project work a number of experiments have been performed on a single cylinder VCR engine by varying the compression ratio from 13 to 17.5and also varying the flow rate of CNG.from 0 LPM to 20 LPM .

4.1.1 Methodology of Experimental Setup

We and Kirloskar DI Diesel Engine develop a VCR (Variable Compression Ratio) Engine Test Setup to study the effect of change of compression ratio on the engine performance. The compression ratio is changed without stopping the engine and without altering the combustion chamber geometry by specially designed tilting cylinder block arrangement. The setup consists of single cylinder, four stroke, VCR Diesel engine connected to eddy current type dynamometer for loading is used with necessary instruments for combustion pressure and crank-angle measurements. These signals are interfaced to computer through engine indicator for P-θ & P-V diagrams. Sensors are used interfacing airflow, fuel flow, temperatures and load measurement. The set up has stand-alone panel box consisting of air box, two fuel tanks for duel fuel test, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator and engine indicator. Rotameters are provided for cooling water and calorimeter water flow measurement. The setup enables varying the compression ratio for measurement of engine performance parameters like brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, Mechanical efficiency, volumetric efficiency, specific fuel consumption, A/F ratio and heat balance. Lab view based Engine Performance Analysis software package “Enginesoft LV” is used for on line performance evaluation. A computerized Diesel injection pressure sensor is used for the measurement of combustion chamber pressure.

Normally, the diesel engine produce excessive knocking that can damage the engine, besides that it produces gases like carbon monoxide, nitrogen oxides, unburnt hydrocarbon, smoke, soot and other forms of black carbon as well as particulate matter, such as lead. All the gases are harmful to the environment and human kind; they can cause greenhouse effect, acid rain and air pollution and thus harm the environment. Due to these effects, human beings may suffer from various diseases, such as lung cancer, breathing difficulties, poisoning and skin cancer etc.

Figure 4.1: VCR C.I. Engine

The main features of the setup are as follows:

• CR changing without stopping the engine

• No alteration in Combustion chamber geometry

• Arrangement for duel fuel test

• PΘ-PV plots, performance plots and tabulated results

• Online measurements and performance analysis

• Data logging, editing, printing and export, Configurable graphs

• IP, IMEP, FP indication

• Combustion analysis

The whole conversion of diesel engine into CNG is very costlier because a large number of modifications are required. For that reasons it is economical to use diesel engine as a dual fuel engine i.e. CNG-Diesel with minor modification. In my project we are trying to analyze how such minor modification effect in reducing the emissions, cost of conversion diesel to CNG and other harmful effects on human and atmosphere using the diesel engine in dual fuel mode.

In diesel –CNG dual fuel engine mixture of natural gas and air is induced in engine cylinder and is compressed during compression stroke. This air fuel mixture is ignited by injecting small quantity of diesel called as pilot injection in cylinder at the end of compression stroke. This diesel pilot fuel ignites due to heat of compression just like diesel engine. Burning of diesel pilot fuel further ignites and burns compressed natural gas in the cylinder and power is produced. In dedicated CNG engines existing diesel engines should be converted to spark ignition engine and CNG is ignited by spark ignition. This requires considerable changes in diesel engine like change in compression ratio, replacing injectors by spark plugs. If dual fuel technology is adopted these major changes can be avoided and existing diesel can be easily converted to dual fuel operation by using simple conversion system.

4.1.2 THE EXPERIMENTAL SETUP

The test engine used was a single cylinder, water cooled, direct injection, normally aspirated, four stroke diesel engine 87.5 mm bore, 110 mm stroke and a variable compression ratio (12-17.5). The gaseous fuels employed were fumigated into the intake air at a point in the engine manifold just outside the cylinder. The engine was coupled directly to an electric dynamometer, which permitted the engine to operate under partial motoring conditions representing negative brake output. Four different commercial and three other prepared diesel fuels were used, having different cetane numbers. Various pilot quantities were employed with commercials methane Natural Gas and a low heating value gas made up of a mixture of methane and nitrogen or carbon dioxide. For any set of operating conditions, the pilot fuel was kept constant while the amount of the main gaseous fuel was gradually increased. The ignition delay period was established from pressure-time records obtained using a water-cooled piezoelectric transducer. The injection timing was established using an electric inductance transducer. The average values obtained forms several consecutive cycles were used. Throughout these tests, the injection timing was kept constant and the engine was operated at 1000 rev/min, under normally aspirated conditions. For the conversion of diesel C.I. engine into a dual fuel engine, some fundamental modifications were required for proper mixing of air and CNG, for which the set up shown below was prepared:

Product :VCR Engine test setup 1 cylinder, 4 stroke, Diesel (Com.)

<u>Engine</u> Make Kirloskar, Type 1 cyl. 4 stroke Diesel, water cooled, power 3.5kW at 1500rpm, stroke 110mm, bore 87.5mm. 661cc, CR17.5, Modified to VCR engine CR 12 to 18

<u>Dynamometer</u> Type eddy current, water cooled, with loading unit

<u>Air box</u> M S fabricated with orifice meter and manometer

<u>Piezo sensor</u> Range 5000 PSI, with low noise cable

<u>Crank angle sensor</u> Resolution 1 Deg, Speed 5500 RPM with TDC pulse.

<u>Data acquisition device</u> NI USB-6210, 16-bit, 250kS/s.

<u>Piezo powering unit</u> Make-Cuadra, Model AX-409.

<u>Digital milivoltmeter</u> Range 0-200mV, panel mounted

<u>Temperature sensor</u> Type RTD, PT100 and Thermocouple, Type K

<u>Temperature transmitter</u> Type two wire, Input RTD PT100, Range 0–100 Deg C,

Output 4–20 mA and Type two wire, Input Thermocouple,

Range 0–1200 Deg C, Output 4–20 mA

<u>Load indicator</u> Digital, Range 0-50 Kg, Supply 230VAC

Load sensor Load cell, type strain gauge, range 0-50 Kg

Fuel flow transmitter DP transmitter, Range 0-500 mm WC

Air flow transmitter Pressure transmitter, Range (-) 250 mm WC

Software "EnginesoftLV" Engine performance analysis software

Rotameter Engine cooling 40-400 LPH; Calorimeter 25-250 LPH

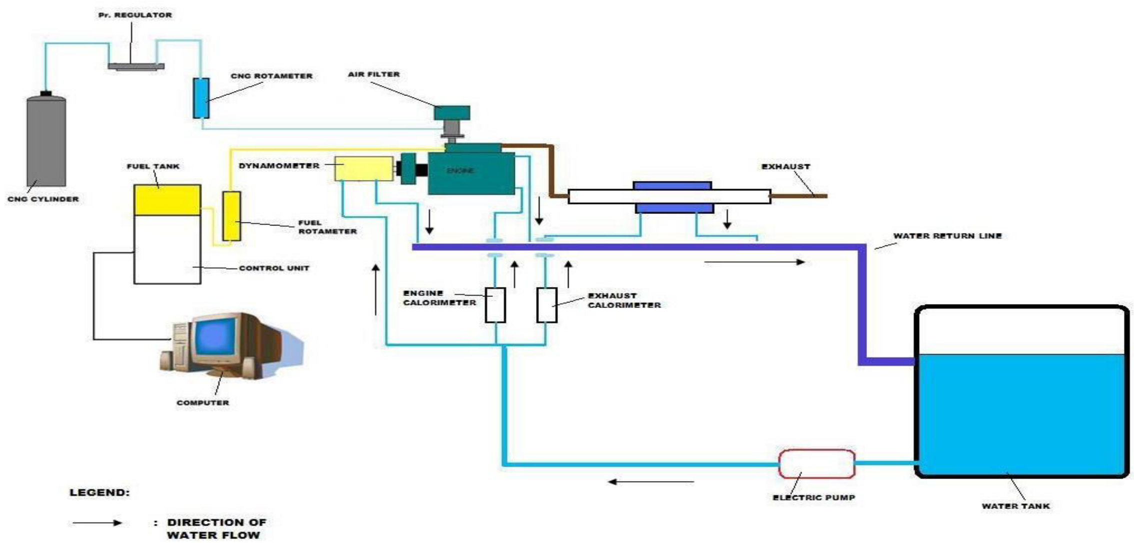

Figure 4.2: Schematic Layout of VCR Engine Test Set Up

Figure 4.3 a Actual Diesel-CNG VCR Engine Test Setup

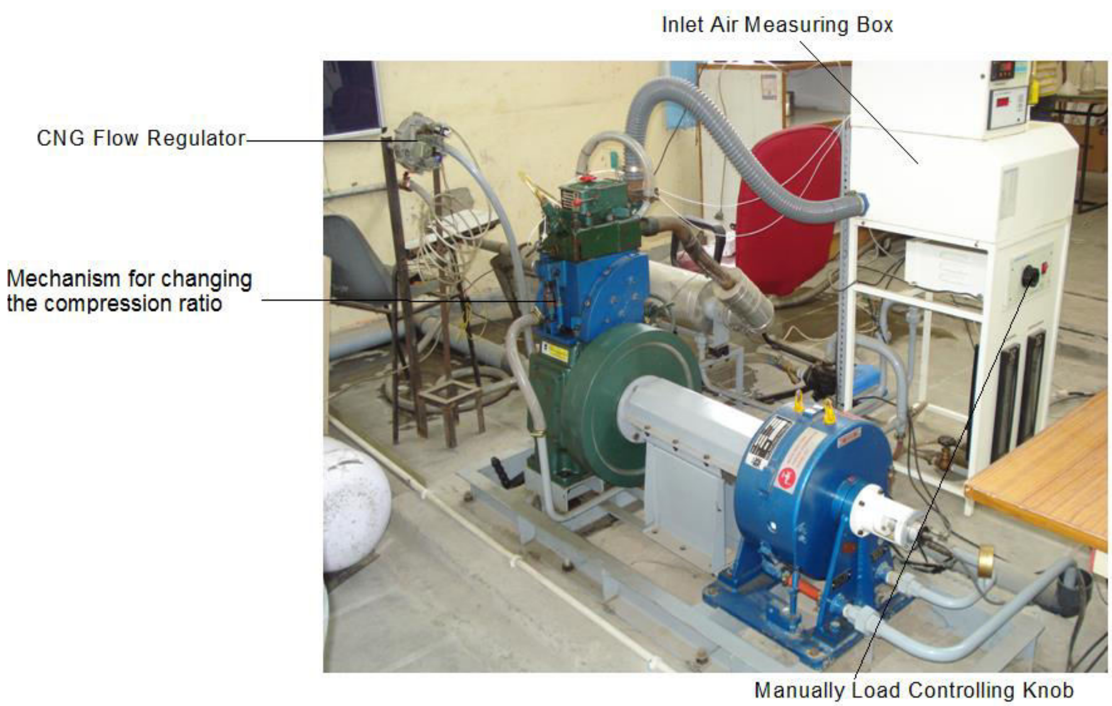

Figure 4.3 b: Actual Diesel-CNG VCR Engine Test Setup

4.2 GAS ANALYSERS

Non-dispersive infra-red (NDIR) analyzers

Working principle

In the NDIR analyzer the exhaust gas species being measured is used to detect itself. This is done by selective absorption. The infrared energy of a particular wavelength or frequency is peculiar to a certain gas in that the gas will absorb the infrared energy of this wavelength and transmit the infrared energy of other wavelengths.

Measurement of CO using NDIR

The absorption band for carbon monoxide is between 4.5 and 5 microns. So the energy absorbed at this wavelength is an indication of concentration of CO in the exhaust gas. The NDIR analyzer consists of two infrared sources, interrupted simultaneously by an optical chopper. Radiations from these sources pass in parallel paths through a reference cell and a sample cell. Exhaust gas from the tailpipe of the engine/vehicle is passed through the sample cell during the measurement of exhaust emissions. The reference cell is filled with an inert gas, usually nitrogen, which does not absorb the infrared energy for the wavelength corresponding to the compound being measured.

A closed container filled with only the compound to be measured works as a detector. The detector is divided into two equal volumes by a thin metallic diaphragm. When the chopper blocks the radiation, the pressure in both parts of the detector is same and the diaphragm remains in the neutral position. As the chopper blocks and unblocks the radiations, the radiant energy from one source passes through the reference cell unchanged whereas the sample cell absorbs the infrared energy at the wavelength of the compound in the cell [31].

The absorption is proportional to the concentration of the compound to be measured in the sample cell. Thus unequal amounts of energy are transmitted to the two volumes of the detector and the pressure differential so generated causes movement of the diaphragm of the detector. This changes the capacitance between the diaphragm and a fixed probe, thereby generating an a.c. signal which is amplified and, after rectification to d.c. displayed on a meter.

Non-dispersive infra-red (NDIR) analyzers

The signal is a function of the concentration of the compound to be measured. The NDIR can accurately measure CO, CO2 and those hydrocarbons which have clear infrared absorption peaks. However, usually the exhaust sample to be analyzed contains other species which also absorb infrared energy at the same frequency.

Gas chromatography

In gas chromatography first the individual constituents are separated from the mixture of gas. After separation, each compound can be separately analyzed for concentration. This is the only method by which each component existing in an exhaust sample can be identified and analyzed. However, it is very time consuming and the samples can be taken only in batches. Gas chromatograph is primarily a laboratory tool. In addition to the above methods such as mass spectroscopy, chemiluminescent analyzers, FID, smokemeters and electrochemical analyzers are also used for measuring exhaust emissions.

Flame ionization detector (FID) (Working principle)

A hydrogen–air flame contains a negligible amount of ions. However even if a trace amount of organic compound such as HC is introduced into the flame, a large number of ions are produced. If a polarized voltage is applied across the burner jet and an adjacent collector, an ion migration will produce a current proportional to number of ions and thus HC concentration present [30] in the flame.

The output of FID depends on the number of carbon atoms passing through the flame in unit time. Doubling the flow velocity would double the output. Hexane (C6H14) would double the output of propane (C3H8).

Chemiluminescent Analyzer

Chemiluminescent is the emission of light energy resulting from a chemical reaction. If NO and O3 are brought together a chemical reaction takes place which produces NO2 in activated state. Activated NO2 emits light

$$NO + O3 \rightarrow NO2 + O2$$

$$NO2 \rightarrow NO2 + \text{light emissions}$$

To measure NO concentration the gas sample to be analyzed is send to a reaction chamber, where it combines with ozone. The resulting chemiluminescent is monitored through an optical filter by a high sensitivity photomultiplier positioned at one end of the reactor. The filter photo multiplier combination responds to light in a narrow wavelength band unique to the above reaction. The output from the photo multiplier is linearly proportional to the NO concentration. To measure NOx concentration (i.e. NO plus NO2) the sample gas flow is diverted through an NO2-to-NO converter. The chemiluminescent response in the flow reactor to the converter effluent is linearly proportional to the NOx concentration entering the converter. The photo multiplier output is amplified and given to an indicator for measuring NOx concentration directly.

Figure 4.5: AVL Gas Anlyser (AVL DIGAS 444)

4.4 SMOKEMETERS

There are two basic types of smoke meters which are used to measure smoke density.

(i) Filter darkening types

(ii) Light extinction type.

The light extinction type of meters can measure both white and black smoke whereas the filter paper darkening type meters can give only black smoke. The light extinction meter can be used for continuous measurements while the filter type can be used only under steady state conditions.

<u>Bosch smoke meter</u>

Bosch smoke meter is filter darkening type. A measured volume of exhaust gas is drawn though a filter paper which is blackened to various degrees depending upon the amount of carbon present in the exhaust. The density of soot is measured by determining the amount of light reflected from the sooted paper. The diameter of the filter paper, the sample volume etc., all are well defined.

<u>Hartridge smoker meter</u>

This smoke meter works on the light extinction principle. A continuously taken exhaust sample is passed through a tube of about 46 cm length which has a light source at one end and photocell or solar cell at the other end. The amount of light passed through this smoke column is used as an indication of smoke level.

This smoke density is defined as the ratio of electric output from the photocell or solar cell when sample is passed through this smoke column to the electric output when clean air is passed through it. The fraction of the light transmitted through the smoke (T) and the length of the light path (L) are related y the Beer-Lambert law.

$$T = e^{-KL}$$

$$K = n\,A\,\theta$$

Where K is the optional absorption coefficient of the obscuring matter per unit length, n the number of soot particles per unit volume; A the average projected are of each particles; and θ the specific absorbance per particle.

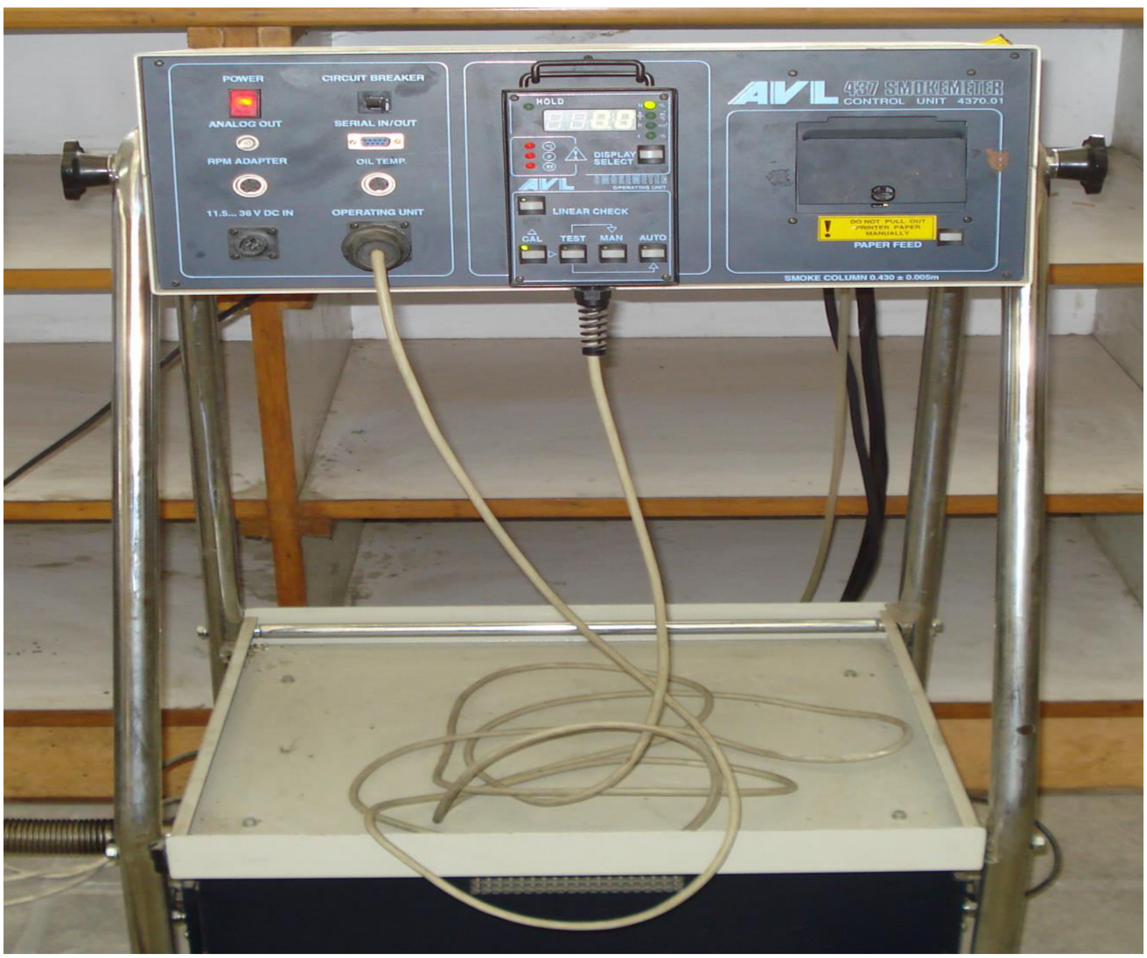

Figure 4.5: AVL Smokemeter

CHAPTER 5

OBSERVATIONS AND ANALYSIS

5.1 METHODOLOGY OF THE EXPERIMENTAL SETUP

A single cylinder, Variable Compression Ratio Engine (Kirloskar make) with range of CR from 12-17.5 was used and was operated on pure fuel mode as well as dual fuel mode. The fuels used were Diesel, blends of Biodiesel (B 20, B 40, and B 60) and CNG. The blend prepared in the laboratory itself by mixing appropriate volume of diesel with Thumba oil (biodiesel).

A set of reading was obtained first by running the engine with diesel at CR of 17.5 and varying the load from idle to rated load of 3.5 KW in steps of 1 up to 3 KW and then to 3.5 KW. The engine performance characteristics were recorded by using the software Engine Soft and instrumentation provided by the National Instruments. The emissions were recorded for each load by using Gas Analyzer AVL Di Gas 444 and the opacity was recorded by Smokemeter (AVL 437).

A set of reading was then recorded for the operation of the engine in dual fuel mode. For this, the CNG conversion kit was switched on and the flow of the CNG was set at 10 LPM. The rotameter was calibrated for LPG and hence the flow rate was converted to that of CNG taking into account the difference in densities of LPG and CNG.

Then, the engine was run on blends of Biodiesel (B20, B40 and B60 in order) and the parameters were recorded as above. The calorific value and density values were entered accordingly in the Engine Soft. Also, the blends were used to run the engine in the dual fuel mode. Care was taken that the engine was started and switched off with pure diesel only.

Similar sets of readings were recorded for the compression ratio of 15 and 13. For this purpose, the engine was started at compression ratio of 17.5 and then the compression ratio was changed by using the tilting head arrangement. The compression ratio was indicated by the rings made on the lever of the arrangement. The observations were made for pure fuel operation of the engine with diesel, B20, B40 and B60 and then dual fuel operation of the same fuels as listed.

Another set of reading was obtained for compression ratio of 17.5 by changing the flow rate of CNG to 20 LPM.

The parameters reordered are listed below:

i) **Load**: It is the brake power output of the engine and is recorded in KW. The engine is loaded by an Eddy Current Dynamometer whose load can be varied by a control knob on the main control panel.

ii) **Brake Thermal Efficiency (BTHE)**: It is the ratio of the brake power of the engine to the thermal energy supplied by the fuel.

iii) **Specific Fuel Consumption (SFC)**: It is the consumption of the fuel in kg/hr per KW of the brake output of the engine.

iv) Using the Gas Analyzer the emissions were recorded. The **emissions** recorded were Carbon Monoxide in percent, carbon dioxide in percent, unburnt Hydrocarbons in PPM and Nitrogen oxides in PPM. The smokemeter was used to measure the **opacity** of the smoke.

When the engine is used in Dual fuel mode, the engine is supplied with CNG at a constant flow rate. But the calculations made by the Engine Soft did not take this into account. Hence, the Brake thermal Efficiency was recalculated. Also, the SFC is the consumption of diesel only and thus another parameter termed as specific energy consumption is defined which is the total energy supplied by the fuel (diesel + CNG) per KW of brake power.

The various observations made are listed below in tabular form. Also the Graphs comparing these parameters are listed.

5.2 OBSERVATIONS

5.2.1 COMPRESSION RATIO: 17.5

a) DIESEL

Table 5.1: Performance characteristics with diesel

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.67	4.86	5.13	215.56
2.00	1.00	11.41	33.16	0.75	31.55
3.00	2.00	21.60	63.33	0.40	16.66
4.00	3.00	29.29	83.30	0.29	12.29
5.00	3.50	32.04	91.27	0.27	11.23

Table 5.2: Emissions with Diesel

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.04	25	2.2	274	0.6
2	1	0.04	22	2.4	286	0.7
3	2	0.04	21	3.2	443	0.8
4	3	0.04	23	3.7	574	0.8
5	3.5	0.03	23	3.6	572	0.8

b) B 20

Table 5.3: Performance characteristics with B 20

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.70	4.38	5.18	211.76
2.00	1.00	12.93	35.14	0.68	27.84
3.00	2.00	22.99	62.32	0.38	15.65
4.00	3.00	30.98	85.14	0.28	11.62
5.00	3.50	32.48	89.29	0.27	11.08

Table 5.4: Emissions with B 20

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.06	18	2.6	299	1
2	1	0.06	16	3.2	394	1.2
3	2	0.07	19	3.8	535	1.6
4	3	0.06	23	4.4	702	1.6
5	3.5	0.07	19	4.6	755	1.5

c) B 40

Table 5.5: Performance characteristics with B 40

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.98	5.42	4.54	181.81
2.00	1.00	14.66	41.09	0.61	24.55
3.00	2.00	26.93	67.70	0.33	13.36
4.00	3.00	31.47	87.68	0.29	11.43
5.00	3.50	34.86	98.03	0.26	10.32

Table 5.6: Emissions with B 40

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.05	28	2.1	226	0.6
2	1	0.05	22	2.8	324	0.9
3	2	0.04	24	3.3	458	1
4	3	0.04	25	3.8	586	1
5	3.5	0.04	14	4.1	654	0.9

d) B 60

Table 5.7: Performance characteristics with B 60

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	2.05	5.47	4.51	175.60
2.00	1.00	13.62	37.41	0.68	26.43
3.00	2.00	23.97	66.67	0.39	15.01
4.00	3.00	30.98	85.95	0.30	11.62
5.00	3.50	34.50	92.35	0.27	10.43

Table 5.8: Emissions with B 60

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.05	16	2.2	215	0.7
2	1	0.05	20	2.9	318	0.7
3	2	0.05	18	3.3	391	0.8
4	3	0.04	15	4	546	0.7
5	3.5	0.04	14	4.1	611	0.8

e) DIESEL + CNG, with CNG flowrate=10 lpm

Table 5.9: Performance characteristics with diesel + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.97	5.08	0.45	182.74
2.00	1.00	13.30	35.08	0.18	27.06
3.00	2.00	23.18	61.64	0.14	15.53
4.00	3.00	32.02	86.53	0.11	11.24
5.00	3.50	35.86	92.58	0.11	10.03

Table 5.10: Emissions with Diesel + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.12	426	2	141	0.6
2	1	0.14	358	2.4	205	0.6
3	2	0.14	283	2.7	259	0.7
4	3	0.14	222	3.1	357	0.8
5	3.5	0.15	223	3.6	433	0.8

f) B 20 + CNG

Table 5.11: Performance characteristics with B 20 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.84	5.03	0.78	195.65
2.00	1.00	13.52	36.41	0.17	26.627
3.00	2.00	22.93	62.21	0.14	15.69
4.00	3.00	31.66	83.65	0.12	11.37
5.00	3.50	34.42	89.81	0.12	10.45

Table 5.12: Emissions with B 20 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.15	359	2.4	169	0.6
2	1	0.18	303	3	271	0.8
3	2	0.19	248	3.5	370	1.3
4	3	0.2	197	4.1	502	1.3
5	3.5	0.17	170	4.1	568	1.1

g) B 40 + CNG

Table 5.13: Performance characteristics with B 40 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	2.07	5.76	0.26	173.91
2.00	1.00	14.80	40.32	0.12	24.32
3.00	2.00	24.80	66.88	0.12	14.51
4.00	3.00	33.62	86.77	0.10	10.70
5.00	3.50	38.00	95.12	0.10	9.47

Table 5.14: Emissions with B 40 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.13	432	2.1	166	0.6
2	1	0.14	342	2.5	228	0.8
3	2	0.16	283	3	297	0.8
4	3	0.16	227	3.4	409	1
5	3.5	0.16	207	3.7	482	0.8

h) B 60 + CNG

Table 5.15: Performance characteristics with B 60 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	2.14	5.55	0.20	168.22
2.00	1.00	15.41	39.53	0.10	23.36
3.00	2.00	25.77	65.61	0.11	13.96
4.00	3.00	34.25	85.30	0.10	10.51
5.00	3.50	37.51	93.72	0.10	9.59

Table 5.16: Emissions with B 60 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.13	396	1.9	115	0.7
2	1	0.16	342	2.6	195	0.9
3	2	0.17	278	3.1	271	0.8
4	3	0.17	214	3.6	389	1
5	3.5	0.16	192	3.9	454	0.8

5.2.2 COMPRESSION RATIO: 15

a) DIESEL

Table 5.17: Performance characteristics with diesel

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.42	5.33	6.05	253.52
2.00	1.00	10.99	34.31	0.78	32.75
3.00	2.00	21.50	63.27	0.40	16.74
4.00	3.00	29.06	83.56	0.29	12.38
5.00	3.50	32.22	88.44	0.27	11.17

Table 5.18: Emissions with Diesel

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.35	175	2.4	20	0.8
2	1	0.3	110	2.9	48	8.2
3	2	0.2	79	3.7	136	21.8
4	3	0.15	65	4.3	266	26.8
5	3.5	0.16	52	4.7	373	61.1

b) B 20

Table 5.19: Performance characteristics with B 20

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.70	4.95	5.15	211.76
2.00	1.00	12.62	36.52	0.70	28.59
3.00	2.00	22.17	64.19	0.40	16.23
4.00	3.00	29.62	84.19	0.30	12.15
5.00	3.50	32.26	91.49	0.27	11.15

Table 5.20: Emissions with B 20

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.21	75	2.4	45	1.4
2	1	0.15	95	3.9	176	7.4
3	2	0.13	97	4.3	277	16.5
4	3	0.1	67	5.9	543	55.6
5	3.5	0.09	70	5.7	557	56.6

c) B 40

Table 5.21: Performance characteristics with B 40

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.80	5.16	5.00	200
2.00	1.00	13.23	37.84	0.68	27.21
3.00	2.00	22.01	63.41	0.41	16.35
4.00	3.00	29.22	82.90	0.31	12.32
5.00	3.50	32.26	91.26	0.28	11.15

Table 5.22: Emissions with B 40

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.18	45	2.6	60	10.9
2	1	0.15	51	3.2	104	2.2
3	2	0.12	54	3.8	190	6.5
4	3	0.09	104	4.4	356	23.6
5	3.5	0.08	126	5.1	497	48.6

d) B 60

Table 5.23: Performance characteristics with B 60

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.76	4.77	5.24	204.54
2.00	1.00	12.78	35.35	0.72	28.16
3.00	2.00	22.23	59.57	0.42	16.18
4.00	3.00	28.94	79.77	0.32	12.43
5.00	3.50	31.23	87.07	0.30	11.52

Table 5.24: Emissions with B 60

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.2	25	2.6	50	1.7
2	1	0.16	30	3.3	109	4.2
3	2	0.12	34	4	224	17.2
4	3	0.1	44	4.7	376	37.4
5	3.5	0.09	58	5.3	490	49.9

e) DIESEL + CNG with CNG flowrate=10 lpm

Table 5.25: Performance characteristics with diesel + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.27	5.23	2.84	283.46
2.00	1.00	10.69	35.63	0.33	33.67
3.00	2.00	22.67	64.71	0.14	15.88
4.00	3.00	31.85	82.08	0.11	11.30
5.00	3.50	35.60	88.27	0.11	10.11

Table 5.26: Emissions with Diesel + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.37	635	2.5	20	38
2	1	0.36	443	2.9	42	10.4
3	2	0.34	366	3.5	82	11
4	3	0.27	274	4.1	172	48.9
5	3.5	0.25	252	4.4	236	55.7

f) B 20 + CNG

Table 5.27: Performance characteristics with B 20 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.79	5.10	0.91	201.11
2.00	1.00	13.40	37.94	0.18	26.86
3.00	2.00	22.49	61.29	0.15	16.00
4.00	3.00	31.58	82.21	0.12	11.39
5.00	3.50	35.03	91.50	0.11	10.27

Table 5.28: Emissions with B 20 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.3	509	2.3	25	1.5
2	1	0.31	411	2.9	51	6.8
3	2	0.24	309	4.4	206	25.6
4	3	0.19	232	5.2	349	41.9
5	3.5	0.16	206	5.8	461	60.9

g) B 40 + CNG

Table 5.29: Performance characteristics with B 40 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.13	5.07	3.91	318.58
2.00	1.00	13.98	34.50	0.15	25.77
3.00	2.00	26.12	62.57	0.10	13.78
4.00	3.00	33.91	84.42	0.10	10.61
5.00	3.50	37.32	88.30	0.10	9.64

Table 5.30: Emissions with B 40 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.29	533	2.5	18	1.4
2	1	0.31	420	3.2	54	3.8
3	2	0.26	313	3.9	147	21
4	3	0.23	312	4.6	239	42
5	3.5	0.22	307	4.3	272	46.5

h) B 60 + CNG

Table 5.31: Performance characteristics with B 60 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.13	5.24	4.01	318.58
2.00	1.00	12.55	35.18	0.23	28.68
3.00	2.00	25.77	59.67	0.11	13.96
4.00	3.00	33.41	79.40	0.11	10.77
5.00	3.50	37.41	87.44	0.10	9.65

Table 5.32: Emissions with B 60 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.28	508	2.3	29	6.7
2	1	0.31	400	3	76	11.5
3	2	0.3	317	3.6	133	15.6
4	3	0.25	236	4.3	241	28.3
5	3.5	0.22	221	4.7	328	34.7

5.2.3 COMPRESSION RATIO: 13

a) DIESEL

Table 5.33: Performance characteristics with diesel

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.09	5.37	7.88	330.88
2.00	1.00	10.99	35.95	0.78	32.75
3.00	2.00	21.50	65.23	0.40	16.74
4.00	3.00	29.06	83.81	0.29	12.38
5.00	3.50	32.22	89.83	0.27	11.17

Table 5.34: Emissions with Diesel

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.46	340	2.45	34	42.5
2	1	0.39	78	2.9	50	6.7
3	2	0.3	41	3.7	105	17.1
4	3	0.23	40	4.4	231	44.1
5	3.5	0.21	39	4.7	240	60.1

b) B 20

Table 5.35: Performance characteristics with B 20

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.78	5.47	4.94	202.70
2.00	1.00	13.14	40.78	0.67	27.40
3.00	2.00	22.46	66.26	0.39	16.03
4.00	3.00	28.74	84.91	0.31	12.52
5.00	3.50	32.34	93.96	0.27	11.13

Table 5.36: Emissions with B 20

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.29	64	2.3	24	3.1
2	1	0.27	59	3.1	73	21.1
3	2	0.2	69	3.9	187	32.2
4	3	0.14	105	4.5	369	50.8
5	3.5	0.11	111	5	474	69.4

c) B 40

Table 5.37: Performance characteristics with B 40

Sl NO	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.53	4.71	5.89	235.60
2.00	1.00	11.86	38.37	0.76	30.34
3.00	2.00	21.84	64.63	0.41	16.48
4.00	3.00	27.81	82.30	0.32	12.94
5.00	3.50	30.23	89.54	0.30	11.90

Table 5.38: Emissions with B 40

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.38	68	2.3	21	3.9
2	1	0.33	43	3.2	53	7
3	2	0.26	44	4	152	20.4
4	3	0.16	100	6.5	465	34.2
5	3.5	0.17	77	5.5	435	38

d) B 60

Table 5.39: Performance characteristics with B 60

Sl NO	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.78	6.28	5.20	202.70
2.00	1.00	13.30	38.68	0.69	27.07
3.00	2.00	23.23	65.94	0.40	15.49
4.00	3.00	31.38	88.62	0.29	11.47
5.00	3.50	34.98	97.98	0.26	10.29

Table 5.40: Emissions with B 60

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.24	47	2.3	41	2
2	1	0.21	32	3.1	103	3.3
3	2	0.18	36	3.7	209	8.7
4	3	0.15	46	4.5	352	13.2
5	3.5	0.14	56	4.7	442	30.9

e) DIESEL + CNG

Table 5.41: Performance characteristics with diesel + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.02	4.42	4.55	352.94
2.00	1.00	9.58	42.55	0.43	37.57
3.00	2.00	20.50	70.53	0.18	17.56
4.00	3.00	30.05	84.22	0.13	11.98
5.00	3.50	34.07	91.30	0.12	10.56

Table 5.42: Emissions with Diesel + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.35	659	2.8	31	90.9
2	1	0.42	552	3.4	50	64.5
3	2	0.44	430	3.9	75	88.2
4	3	0.38	311	4.4	123	92.2
5	3.5	0.34	267	4.7	154	63.7

f) B 20 + CNG

Table 5.43: Performance characteristics with B 20 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.31	5.98	2.71	274.81
2.00	1.00	10.12	38.74	0.39	35.57
3.00	2.00	24.56	66.29	0.12	14.65
4.00	3.00	35.56	85.78	0.09	10.12
5.00	3.50	38.35	92.27	0.09	9.38

Table 5.44: Emissions with B 20 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.3	612	2.1	28	7.5
2	1	0.34	501	2.8	69	15.3
3	2	0.33	408	3.5	130	33.6
4	3	0.29	342	4.2	219	51
5	3.5	0.25	334	4.6	293	66.7

g) B 40 + CNG

Table 5.45: Performance characteristics with B 40 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.36	3.14	2.53	264.70
2.00	1.00	11.66	23.63	0.28	30.87
3.00	2.00	22.68	70.03	0.15	15.87
4.00	3.00	33.65	84.24	0.10	10.69
5.00	3.50	37.49	91.25	0.10	9.60

Table 5.46: Emissions with B 40 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.29	744	2.5	34	45.2
2	1	0.33	623	3.1	56	56.1
3	2	0.41	546	3.7	83	65.7
4	3	0.38	412	4.3	118	76.5
5	3.5	0.31	302	4.8	200	66.6

h) B 60 + CNG

Table 5.47: Performance characteristics with B 60 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.60	6.28	1.58	225
2.00	1.00	13.75	30.29	0.17	26.18
3.00	2.00	23.67	64.76	0.14	15.20
4.00	3.00	32.40	92.67	0.12	11.11
5.00	3.50	37.08	92.08	0.11	9.70

Table 5.48: Emissions with B 60 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.37	594	2.5	14	22.3
2	1	0.37	460	3	31	35.5
3	2	0.36	350	3.2	53	39.6
4	3	0.3	293	3.7	108	46.6
5	3.5	0.28	268	4.6	206	57.6

5.2.4 FLOW RATE OF CNG: 20 LPM, CR = 17.5

a) DIESEL + CNG

Table 5.49: Performance characteristics with diesel + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.05	4.91	0.38	342.857
2.00	1.00	7.23	36.83	0.20	49.792
3.00	2.00	14.57	56.71	0.12	24.708.
4.00	3.00	20.02	79.61	0.12	17.98
5.00	3.50	29.23	83.32	0.03	12.31

Table 5.50: Emissions with Diesel + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.08	967	1.2	27	1
2	1	0.1	845	1.6	41	1.3
3	2	0.1	665	1.7	51	2.4
4	3	0.11	528	1.8	73	3.8
5	3.5	0.11	497	1.9	86	10.3

b) B 20 + CNG

Table 5.51: Performance characteristics with B 20 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	1.01	4.86	0.69	355.73
2.00	1.00	8.66	35.32	0.06	41.60
3.00	2.00	16.97	59.47	0.04	21.21
4.00	3.00	25.33	81.14	0.03	14.21
5.00	3.50	29.25	84.42	0.03	12.30

Table 5.52: Emissions with B 20 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.1	904	1.3	52	0.9
2	1	0.14	827	1.8	82	0.7
3	2	0.24	1020	3.2	150	1.8
4	3	0.26	802	3.8	228	3.8
5	3.5	0.23	708	3.8	251	8.1

c) B 40 + CNG

Table 5.53: Performance characteristics with B 40 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	0.99	5.57	0.88	362.31
2.00	1.00	8.59	42.34	0.07	41.93
3.00	2.00	16.61	61.53	0.05	21.66
4.00	3.00	24.32	81.56	0.04	14.80
5.00	3.50	29.80	91.32	0.02	12.08

Table 5.54: Emissions with B 40 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.08	876	1.1	19	0.7
2	1	0.09	745	1.2	28	1.6
3	2	0.1	506	1.3	30	3
4	3	0.1	433	1.6	52	12.4
5	3.5	0.1	384	1.7	65	16.4

d) B 60 + CNG

Table 5.55: Performance characteristics with B 60 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	0.98	5.37	1.03	367.34
2.00	1.00	8.76	37.55	0.05	41.09
3.00	2.00	17.03	59.05	0.04	21.13
4.00	3.00	24.96	76.77	0.03	14.42
5.00	3.50	29.41	84.35	0.03	12.24

Table 5.56: Emissions with B 60 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.14	1407	2.1	41	1.4
2	1	0.18	1243	2.6	62	2.2
3	2	0.2	1067	3.1	89	2.8
4	3	0.23	893	3.6	127	9.6
5	3.5	0.24	796	3.9	156	10.6

5.2.5 FLOW RATE OF CNG: 20 LPM, CR = 15

a) DIESEL + CNG

Table 5.49: Performance characteristics with diesel + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	0.85	5.29	2.25	423.52
2.00	1.00	7.93	34.56	0.15	45.39
3.00	2.00	16.09	64.08	0.14	22.37
4.00	3.00	25.04	83.24	0.10	14.37
5.00	3.50	29.03	89.45	0.06	12.40

Table 5.50: Emissions with Diesel + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.08	967	1.2	27	1
2	1	0.1	845	1.6	41	1.3
3	2	0.1	665	1.7	51	2.4
4	3	0.11	528	1.8	73	3.8
5	3.5	0.11	497	1.9	86	10.3

b) B 20 + CNG

Table 5.51: Performance characteristics with B 20 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	0.75	5.09	3.67	480.00
2.00	1.00	8.95	37.05	0.02	40.22
3.00	2.00	17.71	63.02	0.02	20.32
4.00	3.00	26.54	82.35	0.01	13.56
5.00	3.50	30.97	92.21	0.01	11.62

Table 5.52: Emissions with B 20 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.1	904	1.3	52	0.9
2	1	0.14	827	1.8	82	0.7
3	2	0.24	1020	3.2	150	1.8
4	3	0.26	802	3.8	228	3.8
5	3.5	0.23	708	3.8	251	8.1

c) B 40 + CNG

Table 5.53: Performance characteristics with B 40 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	0.99	5.34	0.88	363.63
2.00	1.00	7.89	37.87	0.07	45.62
3.00	2.00	17.56	64.45	0.05	20.50
4.00	3.00	24.95	85.95	0.04	14.42
5.00	3.50	30.89	93.32	0.02	11.65

Table 5.54: Emissions with B 40 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.08	876	1.1	19	0.7
2	1	0.09	745	1.2	28	1.6
3	2	0.1	506	1.3	30	3
4	3	0.1	433	1.6	52	12.4
5	3.5	0.1	384	1.7	65	16.4

d) B 60 + CNG

Table 5.55: Performance characteristics with B 60 + CNG

Sl No	LOAD (kW)	BTHE (%)	MeEf (%)	SFC (kg/kW-hr)	SEC (MJ/kW-hr)
1.00	Idle	0.98	5.37	1.03	367.34
2.00	1.00	8.76	37.55	0.05	41.09
3.00	2.00	17.03	59.05	0.04	21.13
4.00	3.00	24.96	76.77	0.03	14.42
5.00	3.50	29.41	84.35	0.03	12.24

Table 5.56: Emissions with B 60 + CNG

Sl No	Load (kW)	CO (%)	HC (ppm)	CO_2 (%)	NOX (ppm)	OPACITY (%)
1	Idle	0.14	1407	2.1	41	1.4
2	1	0.18	1243	2.6	62	2.2
3	2	0.2	1067	3.1	89	2.8
4	3	0.23	893	3.6	127	9.6
5	3.5	0.24	796	3.9	156	10.6

5.3 GRAPHS

5.3.1 PRESSURE – CRANK ANGLE CURVES(P-θ) Curves

The P-θ curve shows the instantaneous pressure at a given crank angle. It is useful in determining the peak cylinder pressure, the rate of pressure rise, ignition delay and also to determine the IMEP. The curves are drawn for a given load and that of different fuels are represented together for the purpose of the comparison. The curves for the dual fuel operation are also plotted. Finally, the effect of the flow rate of CNG is studied by comparing the plots at different flow rate.

At Compression ratio of 17.5, the P-θ curves for the diesel and the other blends are almost similar with slight differences in peak pressure attained and thus we can conclude that the diesel can be substituted by the Biodiesel at this compression ratio. As the compression ratio is decreased from 17.5 to 15 and 13, we can observe that the suitability of diesel fuel at these low compression ratio decreases, however the trend of P-θ curve for the blends show slight variation with the change in compression ratio. Thus, at lower compression ratio the blended fuel is more suitable than diesel for CI engine. But at the rated engine power, the curves are almost similar and thus diesel can be used at low compression ratio for the rated power. The peak pressure decreases as the compression ratio is decreased, irrespective of the fuel and thus the work done per cycle is reduced which is true to the nature of the CI engines. From the P-θ curves for lower compression ratio, it is also observed that the peak pressure is greater for the blended fuel than that of pure diesel.

When the engine is run in dual fuel mode, the trend of P-θ curves is almost similar to that of pure fuels. However, there is an observable decrease in the peak pressure. This decrease is due to the low density of the CNG which reduces the volumetric efficiency of the engine and secondly as the flame velocity for CNG is low. This is further supported by the comparative study of P-θ curve with the change in flow rate of CNG. It is observed that the peak pressure decreases as the flow rate of CNG is increased. Again, we observe that the peak pressure is more for blended fuels.

The peak pressure graphs for diesel and biodiesel with flow rate of CNG (0 lpm, 10 lpm and 20 lpm) at rated load and with compression ratio 17.5 concludes that the peak pressure decreases with decrease of compression ratio. The graph also concludes that with the increase in CNG flow rate for diesel there is drastic reduction in peak pressure. For B40 blend, at 13 compression ratio and CNG flow rate of 10 lpm, shows that there is only marginal change in peak pressure. As the CNG flow rate increases the peak pressure rise decreases at constant compression ratio. The decrease in peak pressure rise is due to lower flame velocity of CNG. By varying both the compression ratio and flow rate there is drastic change in peak pressure.

5.3.1. a COMPRESSION RATIO: 17.5

a) Load: Idle

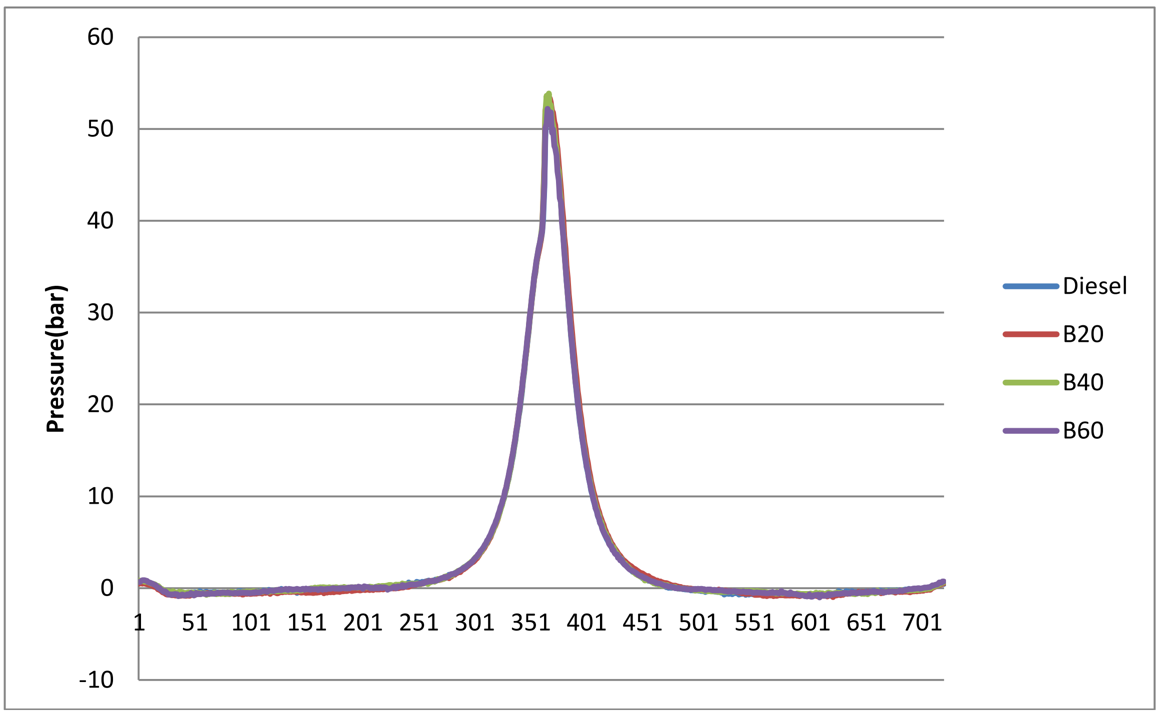

Figure 5.1a: P-θ Curve at Idle load

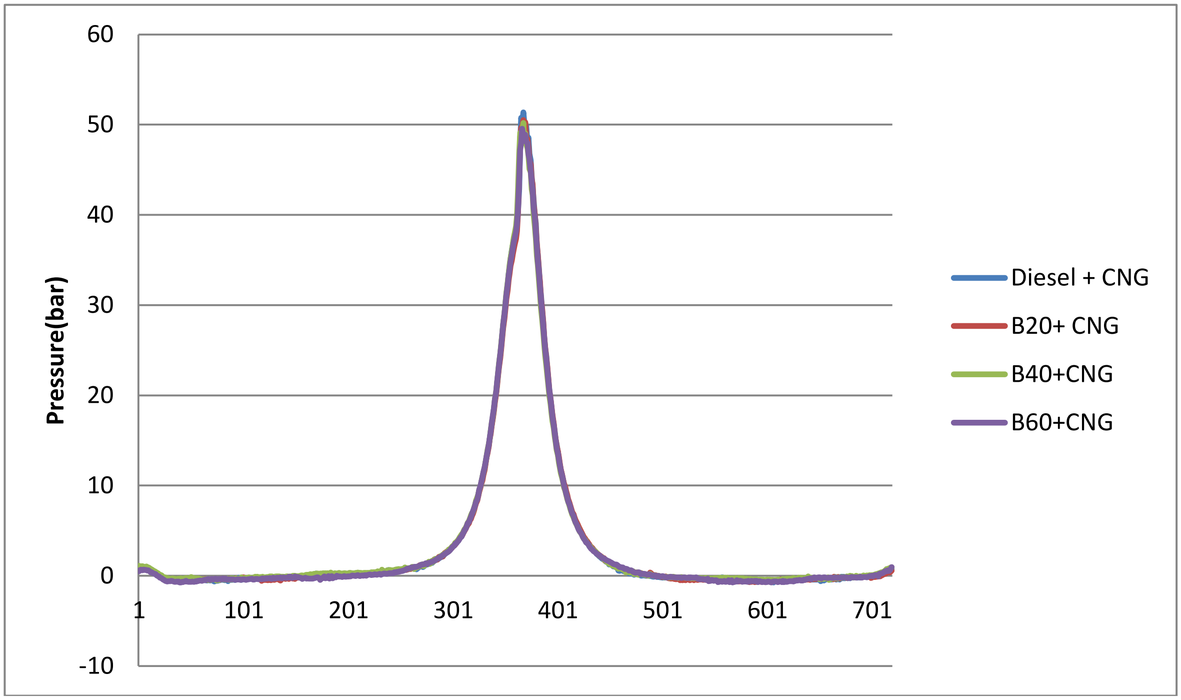

Figure 5.1b: P-θ Curve at Idle load for dual mode

b) Load: 1 kW

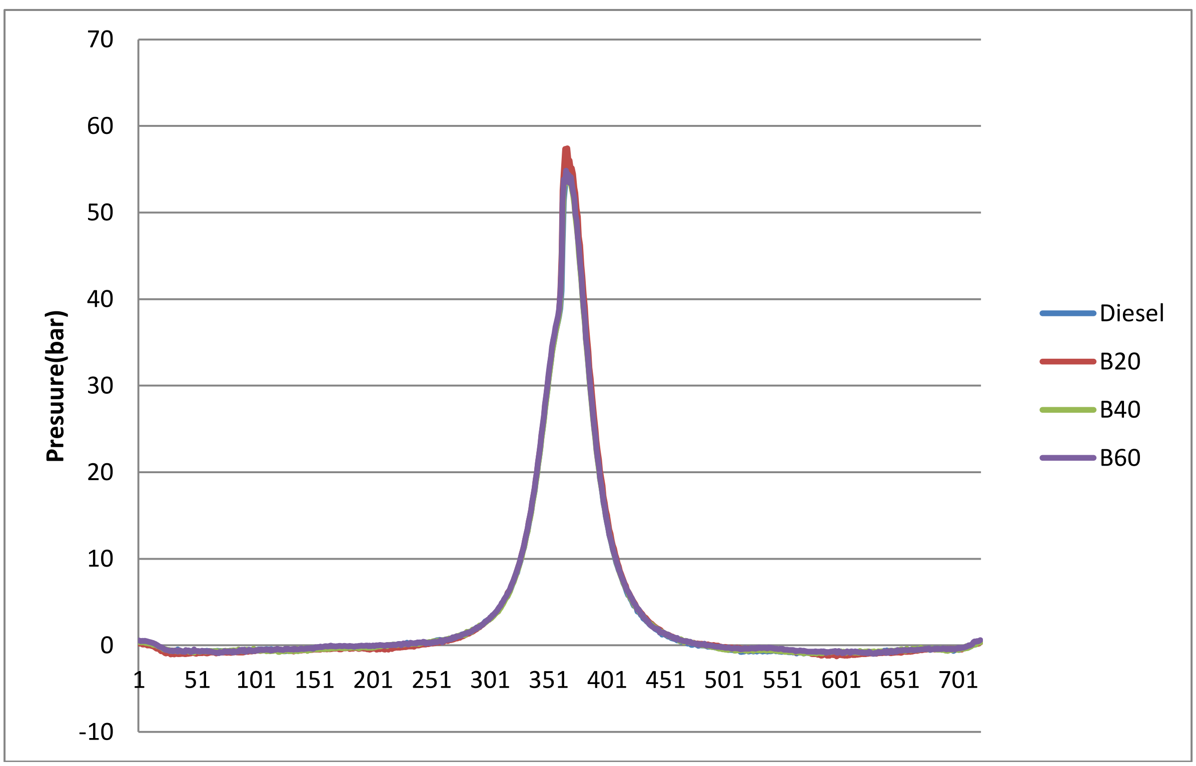

Figure 5.2a: P-θ Curve at 1 kW load

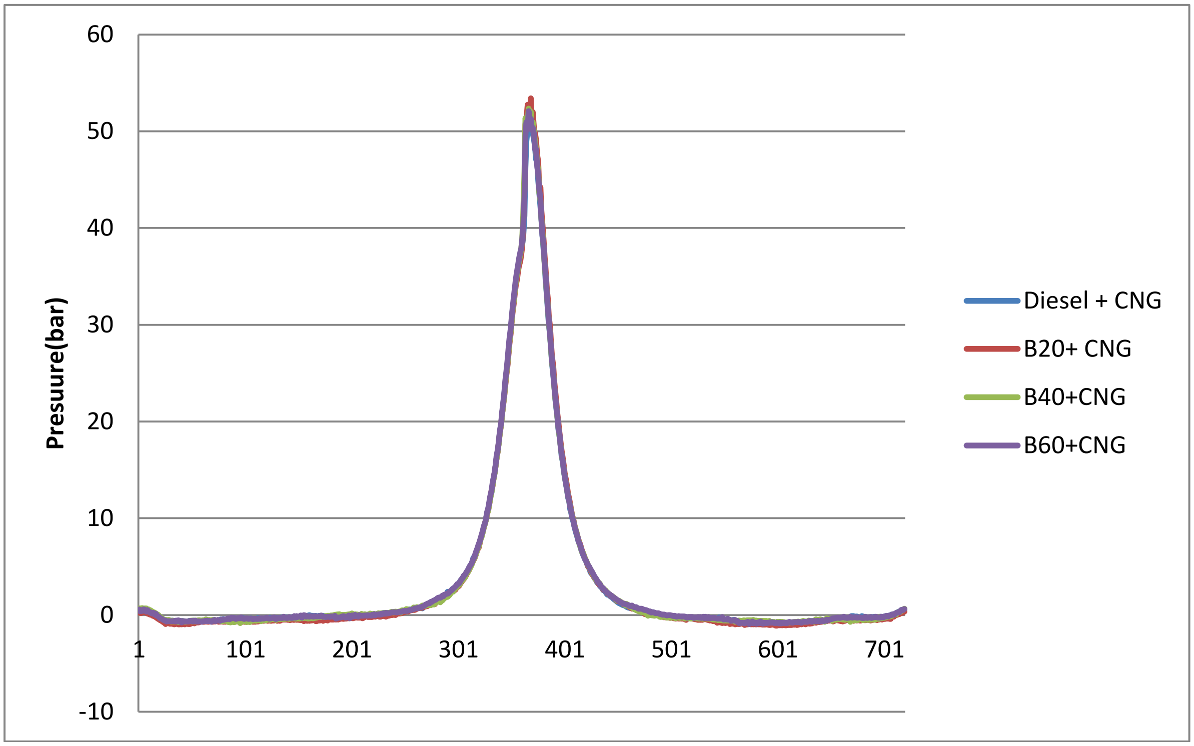

Figure 5.2b: P-θ Curve at 1 kW load for dual mode

c) Load: 2 kW

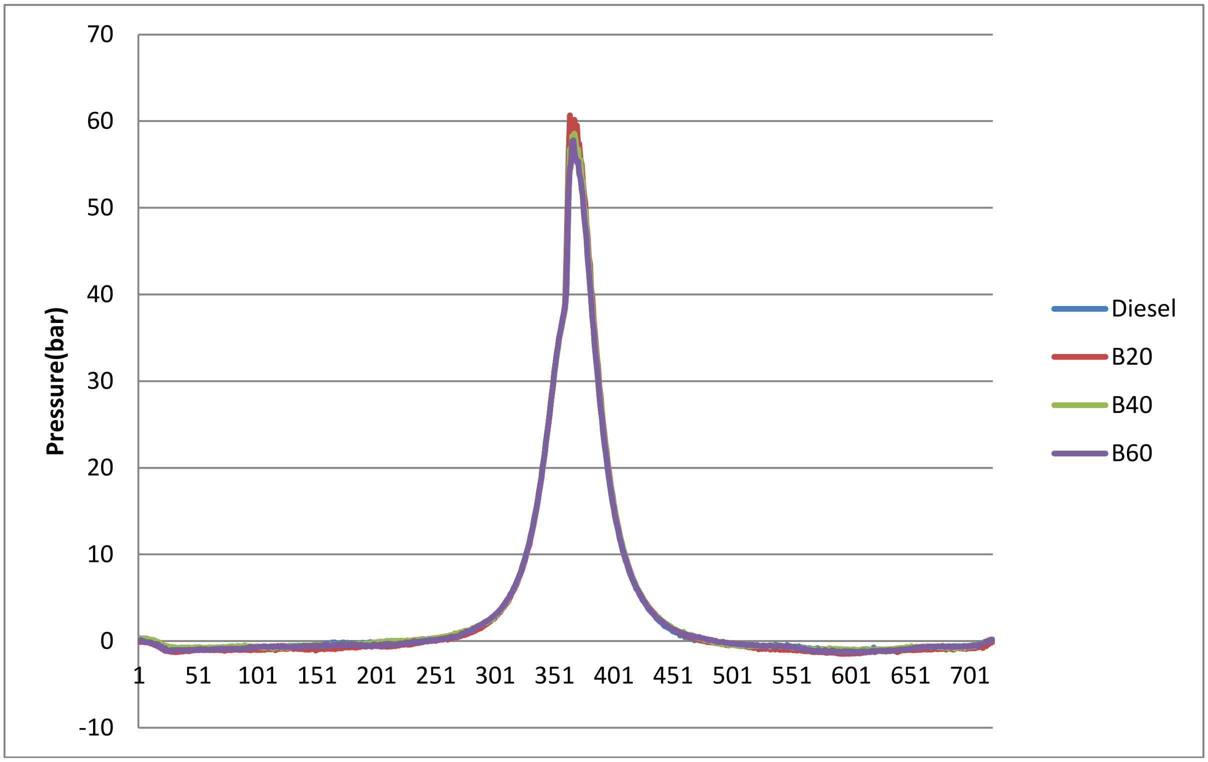

Figure 5.3a: P-θ Curve at 2 kW load

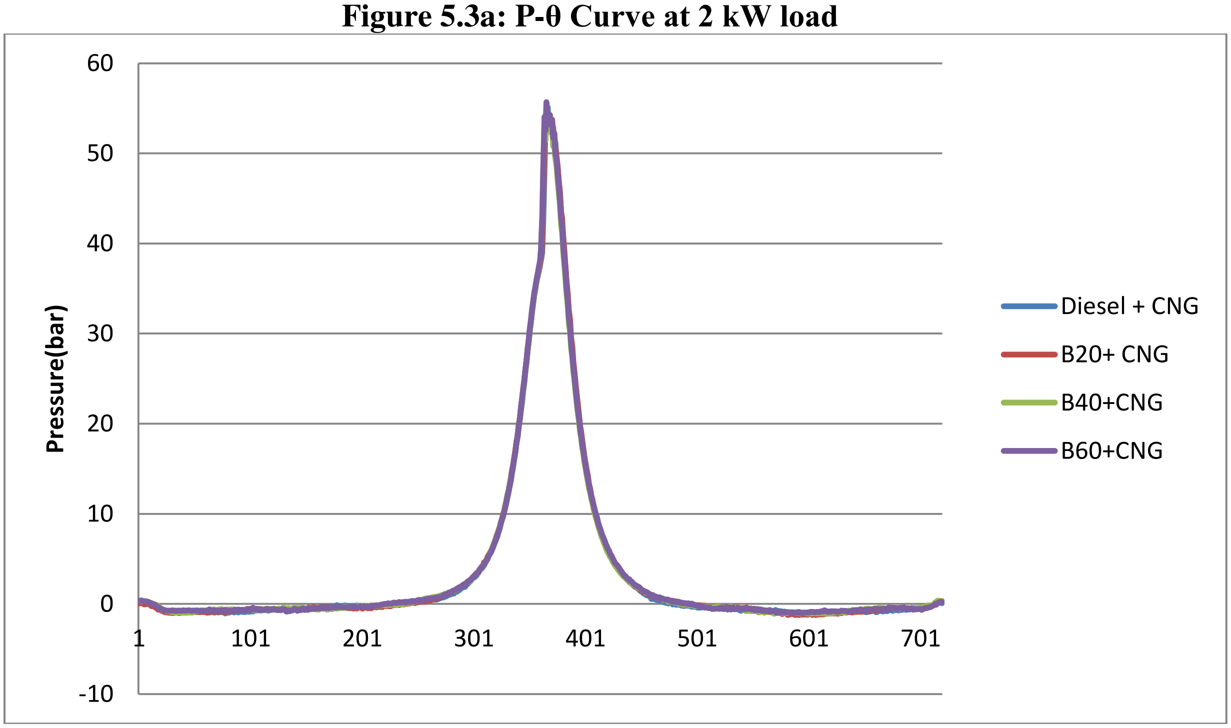

Figure 5.3b: P-θ Curve at 2 kW load for dual mode

d) Load: 3 kW

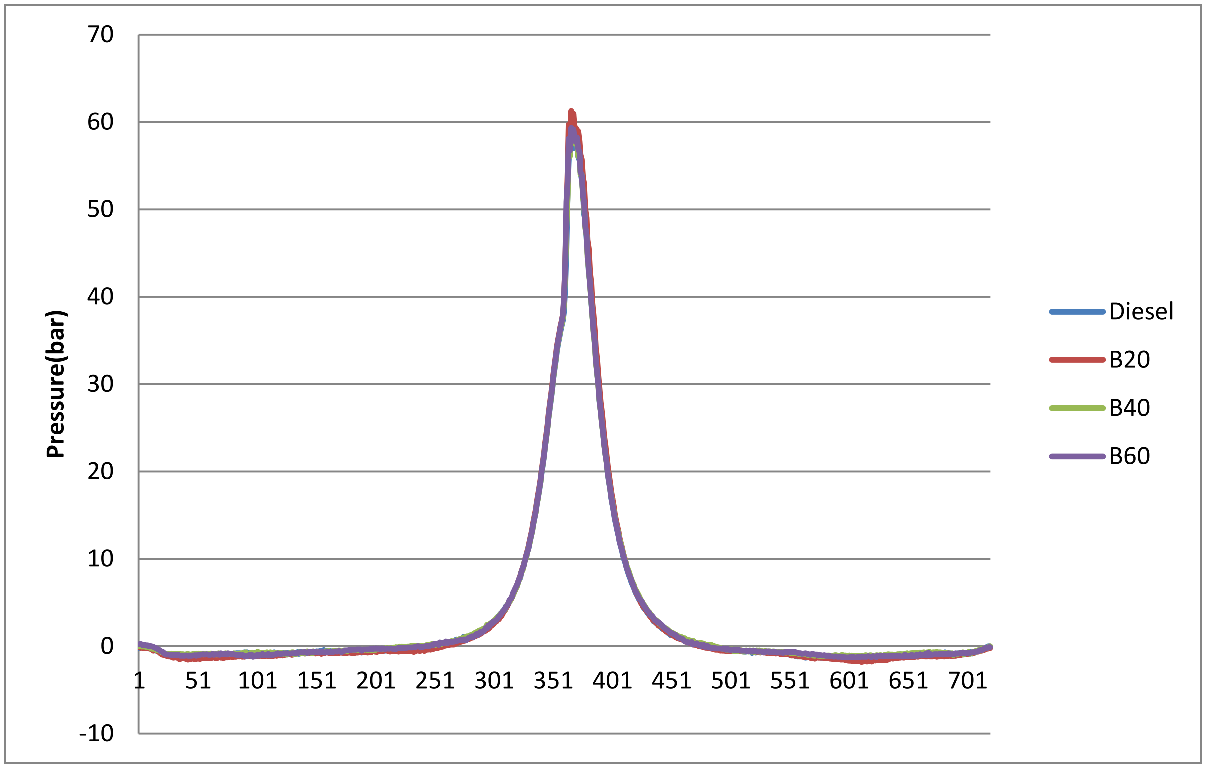

Figure 5.4a: P-θ Curve at 3 kW load

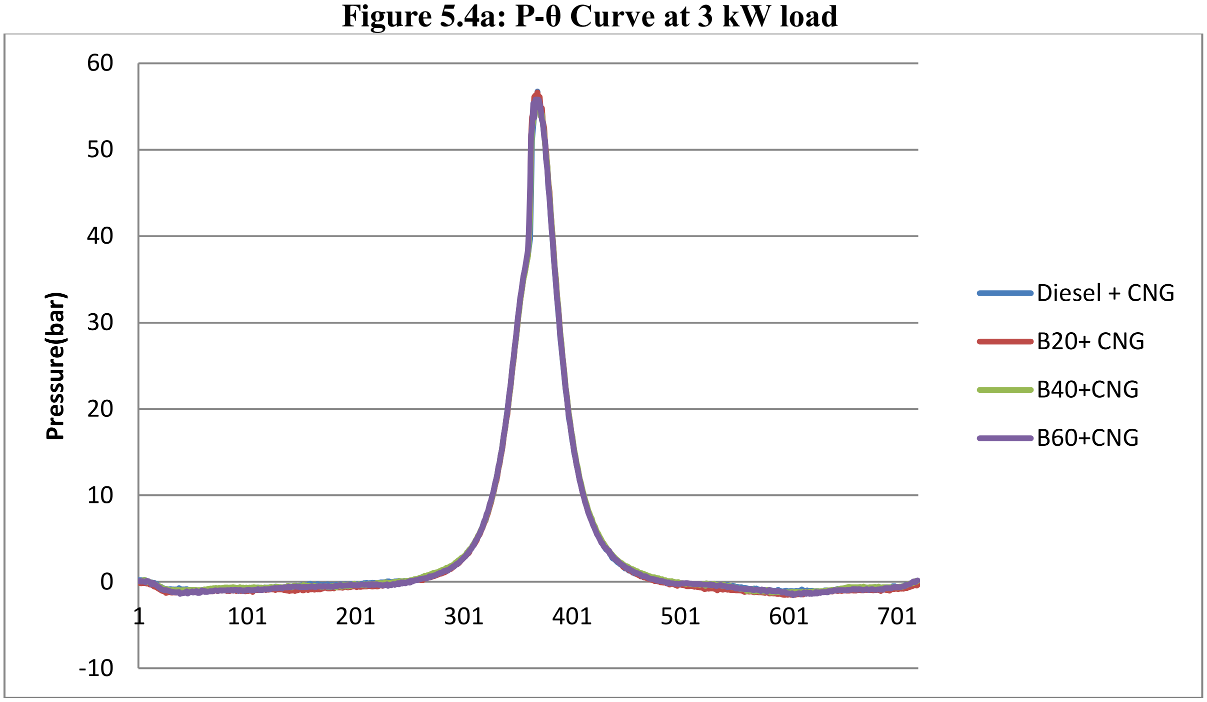

Figure 5.4b: P-θ Curve at 3 kW load for dual mode

e) **Load: 3.5 kW**

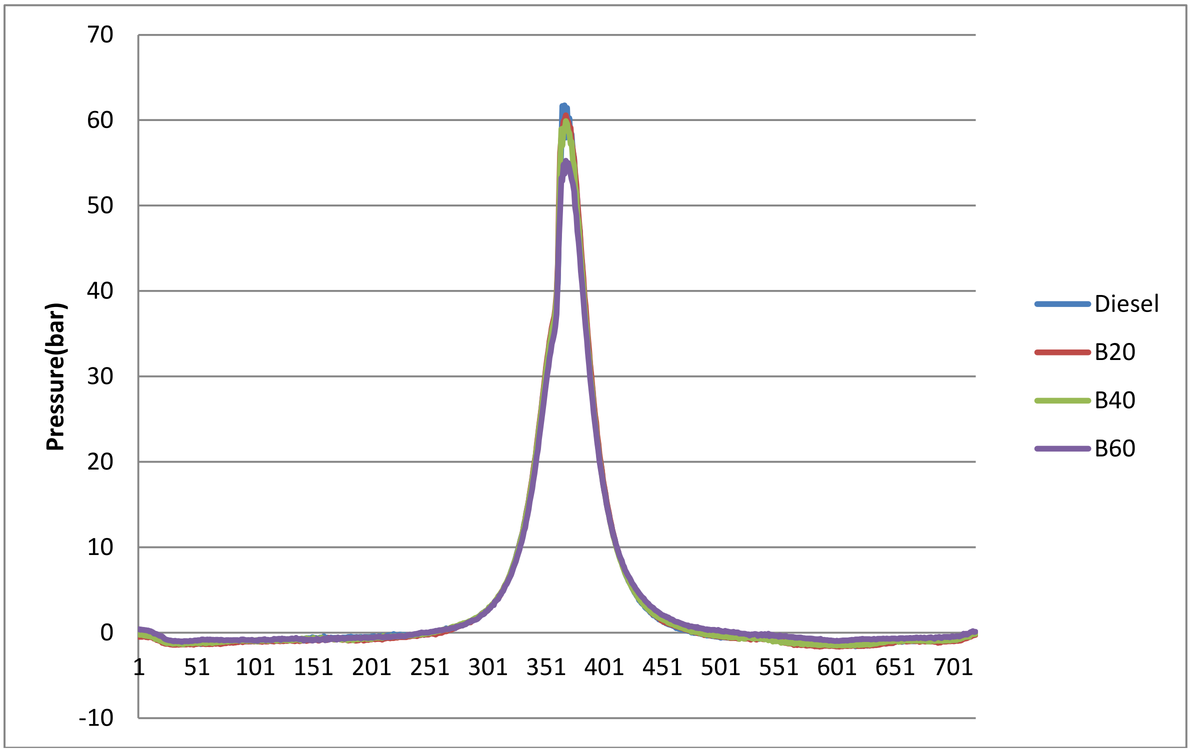

Figure 5.5a: P-θ Curve at 3.5 kW load

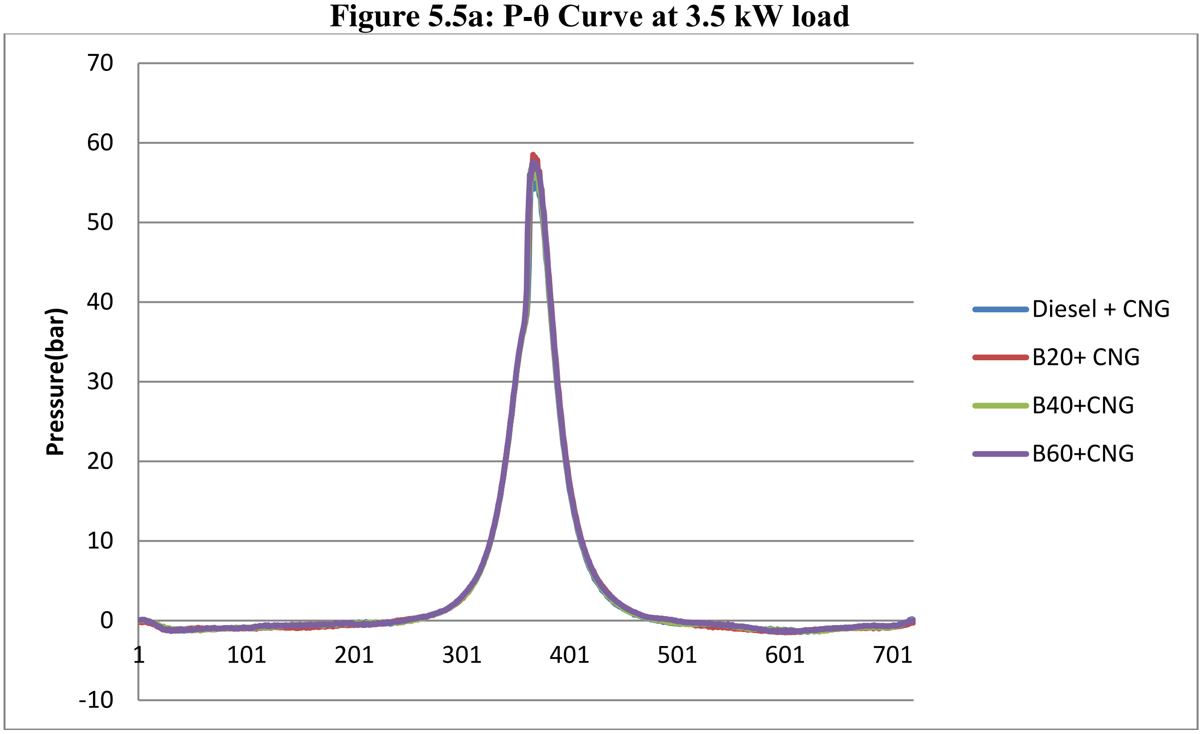

Figure 5.5b: P-θ Curve at 3.5 kW load for dual mode

5.3.1.b COMPRESSION RATIO: 15

a) **Load: Idle**

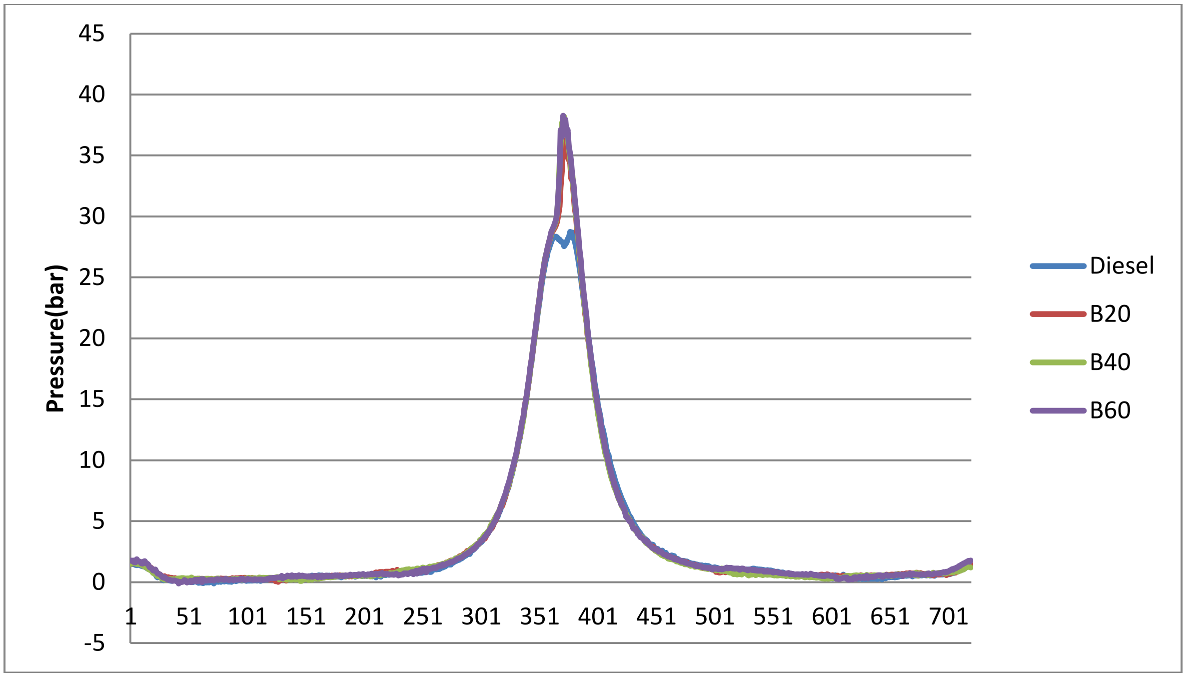

Figure 5.6a: P-θ Curve at Idle load

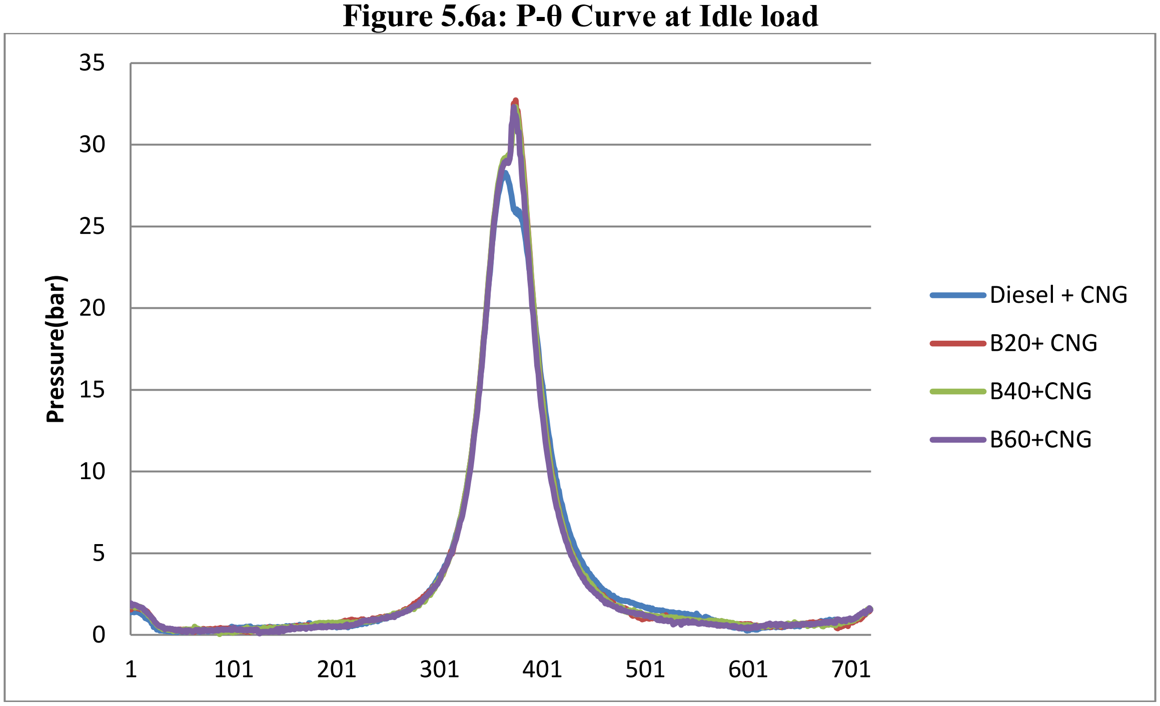

Figure 5.6b: P-θ Curve at Idle load for dual mode

b) Load: 1 kW

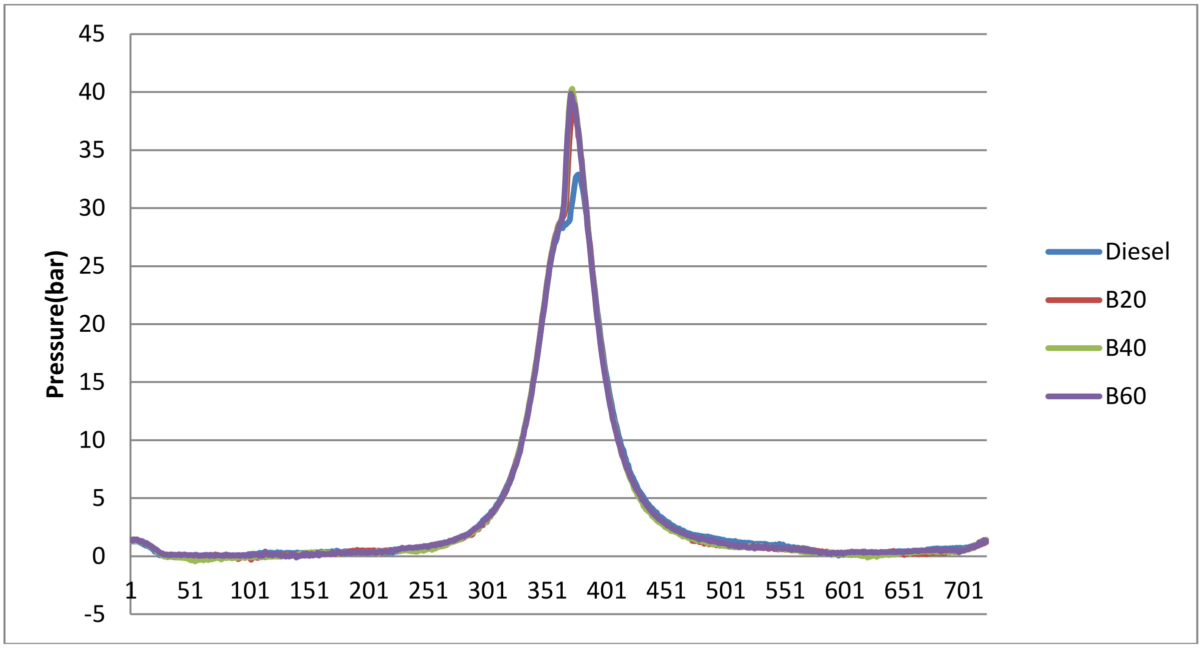

Figure 5.7a: P-θ Curve at 1 kW load

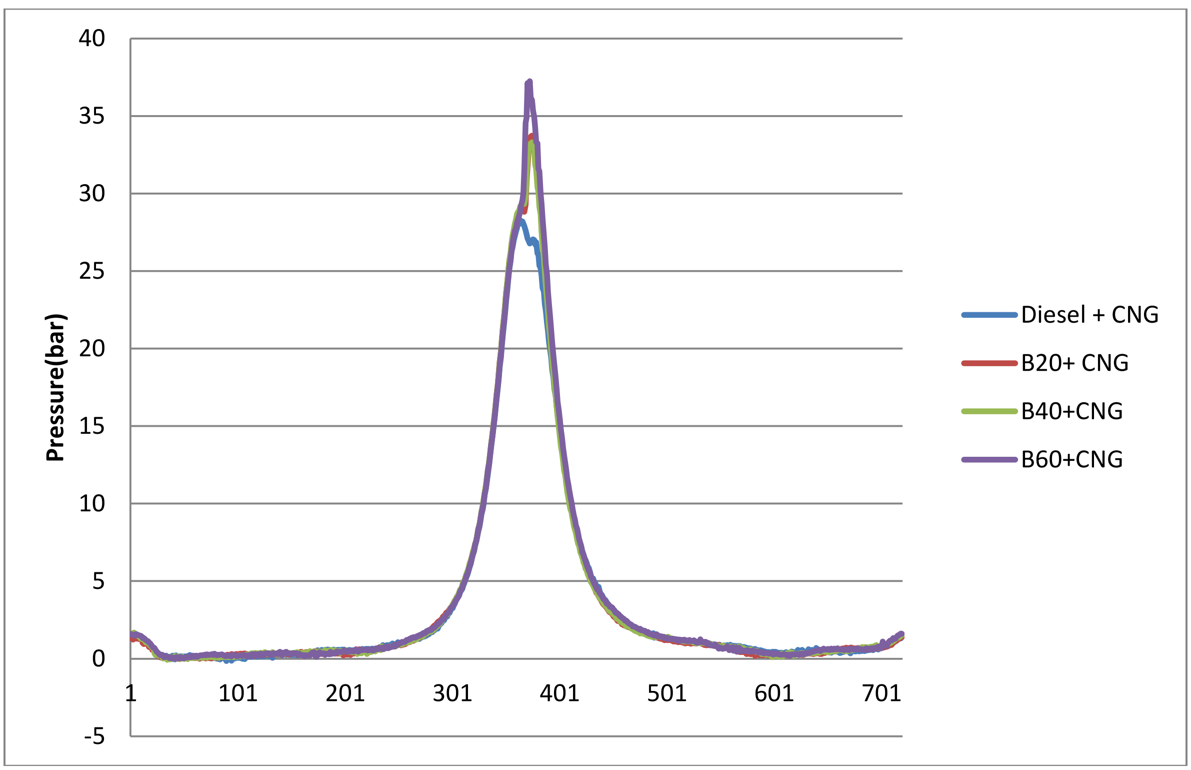

Figure 5.7b: P-θ Curve at 1 kW load for dual mode

c) **Load: 2kW**

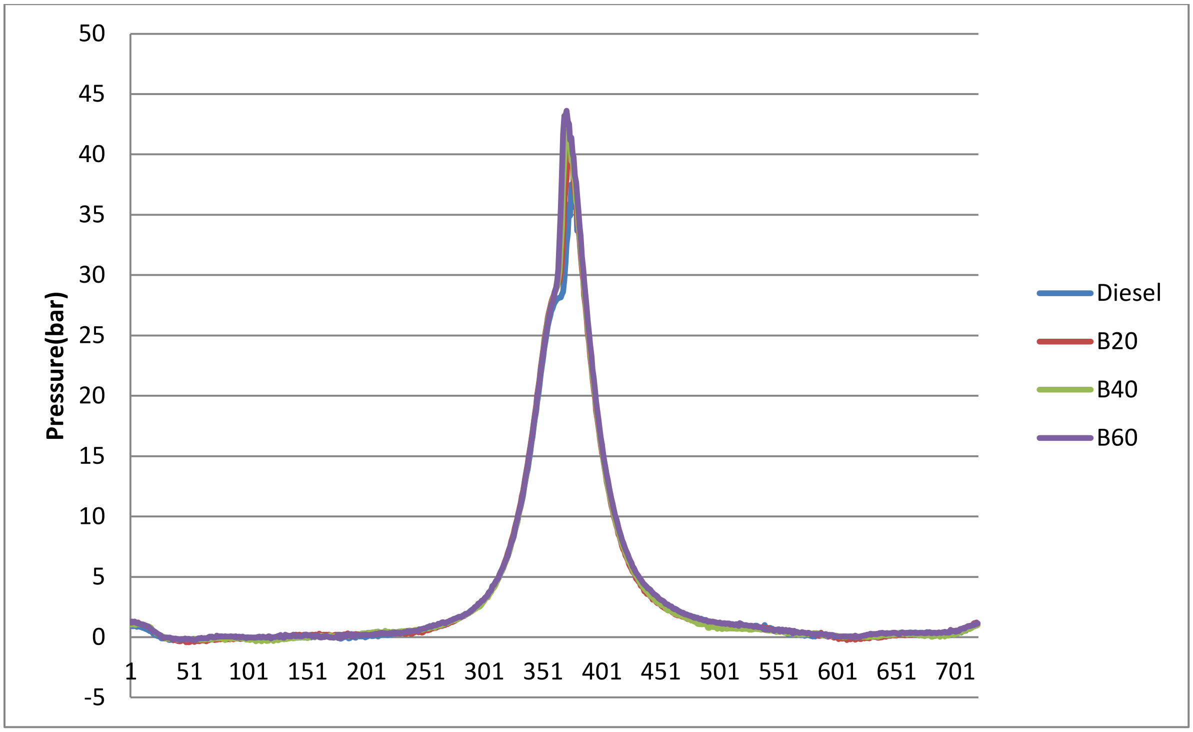

Figure 5.8a: P-θ Curve at 2 kW load

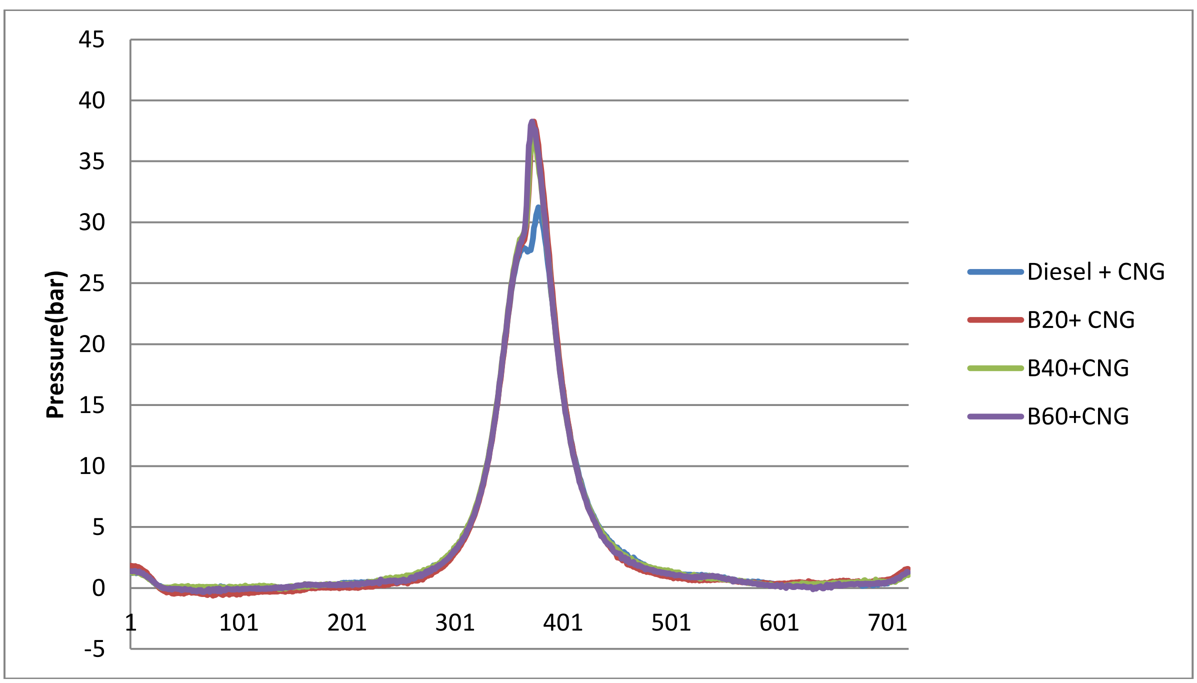

Figure 5.8b: P-θ Curve at 2 kW load for dual mode

d) **Load: 3 kW**

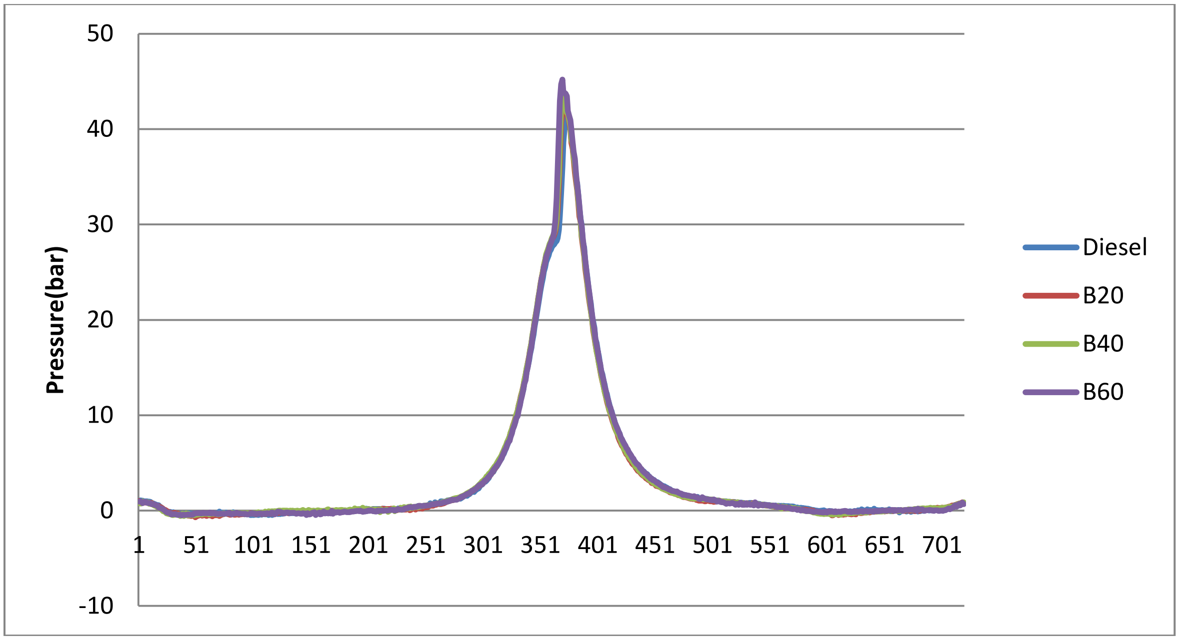

Figure 5.9a: P-θ Curve at 3 kW load

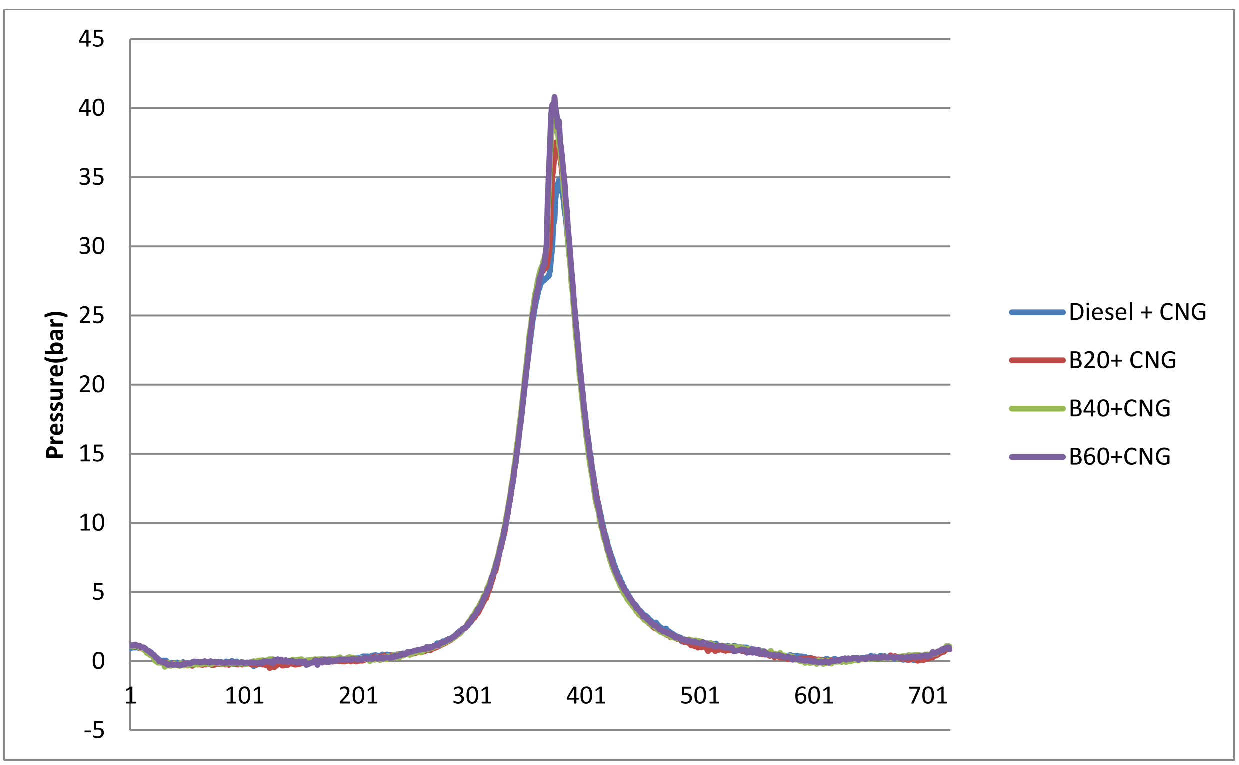

Figure 5.9b: P-θ Curve at 3 kW load for dual mode

e) Load: 3.5 kW

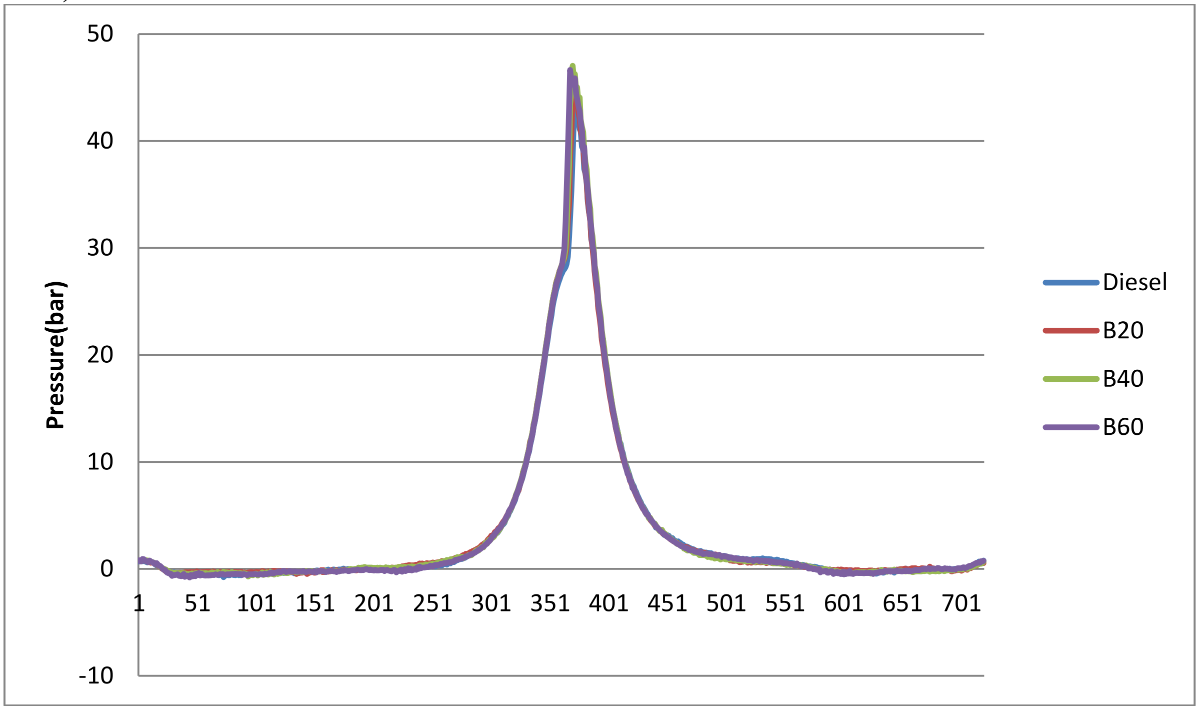

Figure 5.10a: P-θ Curve at 3.5 kW load

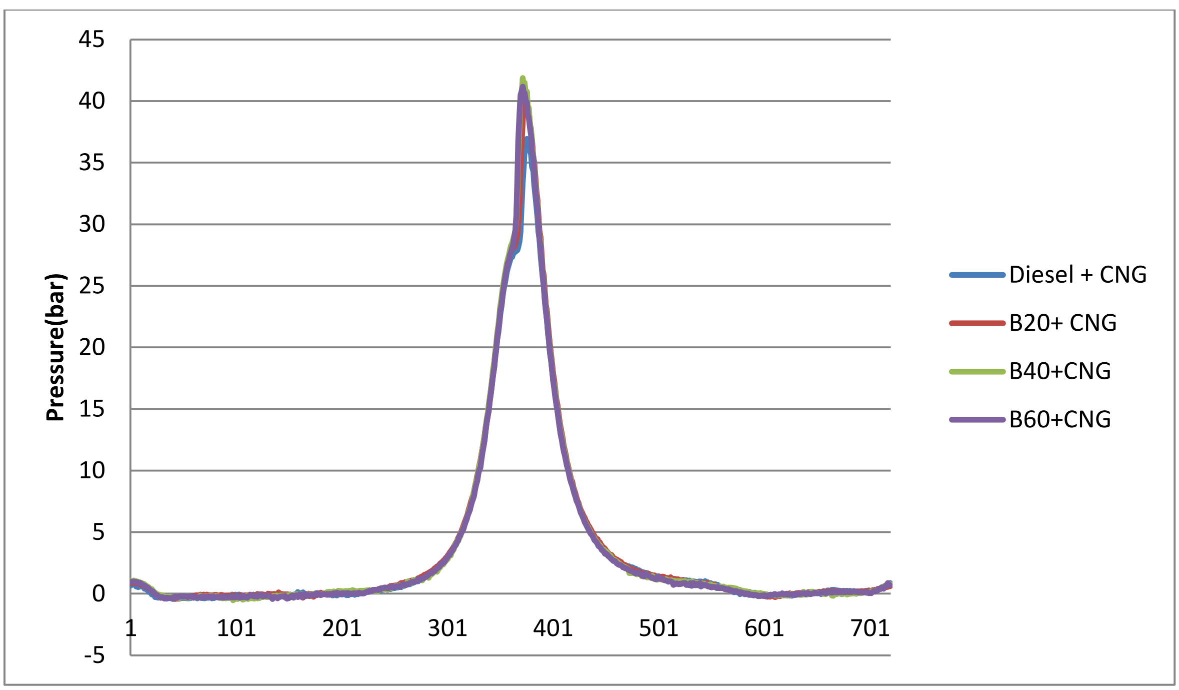

Figure 5.10b: P-θ Curve at 3.5 kW load for dual mode

5.3.1.c COMPRESSION RATIO: 13

a) Load: Idle

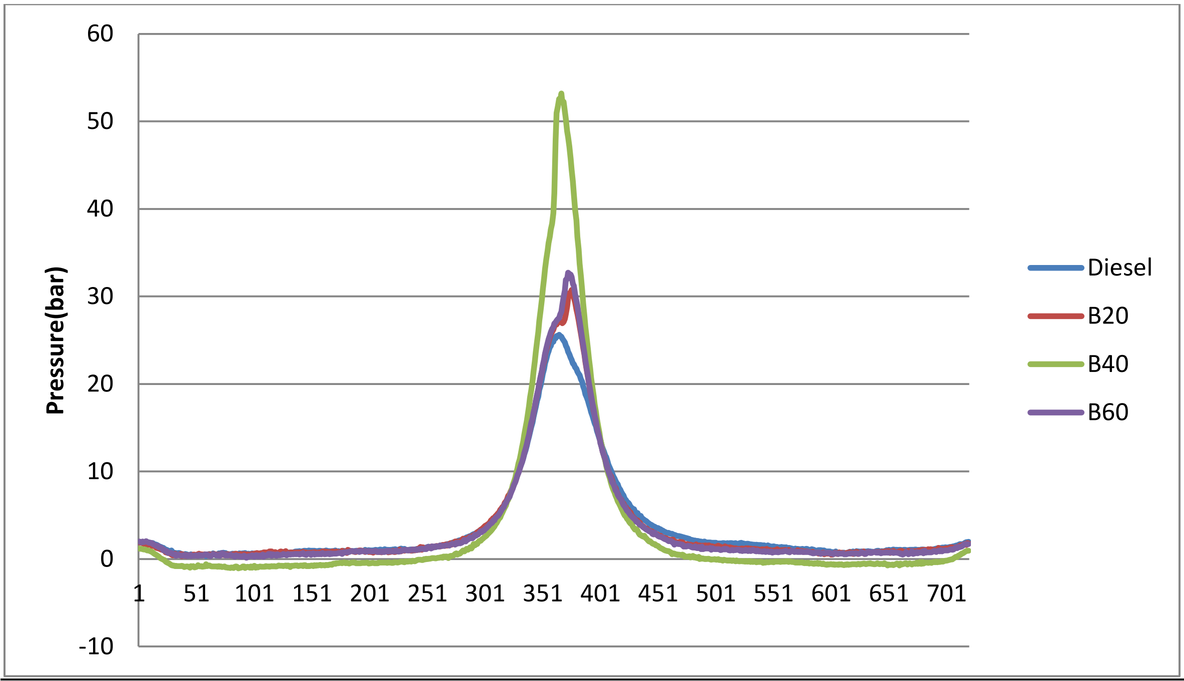

Figure 5.11a: P-θ Curve at Idle load

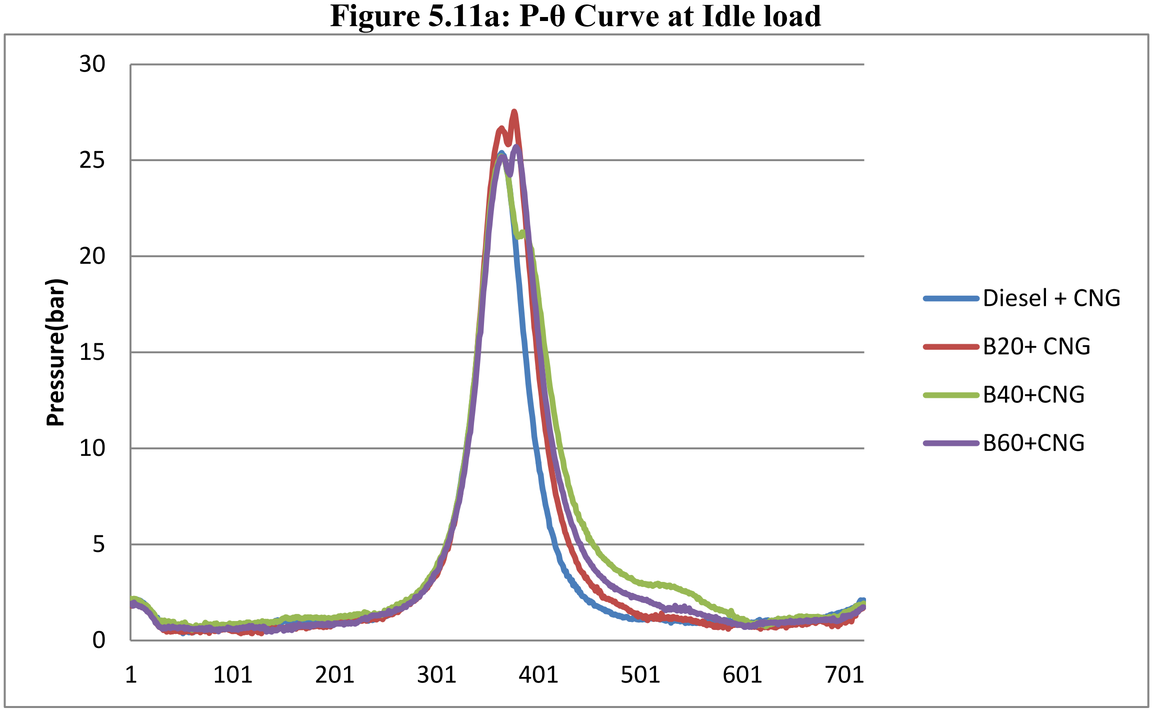

Figure 5.11b: P-θ Curve at Idle load for dual mode

b) **Load: 1 kW**

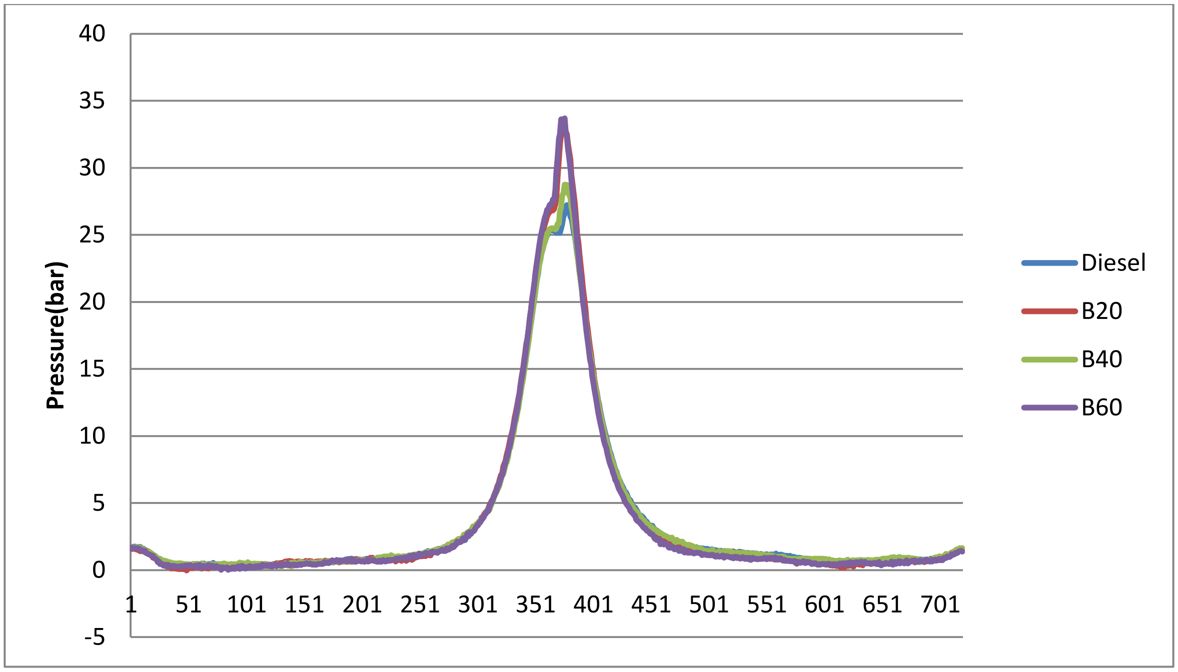

Figure 5.12a: P-θ Curve at 1 kW load

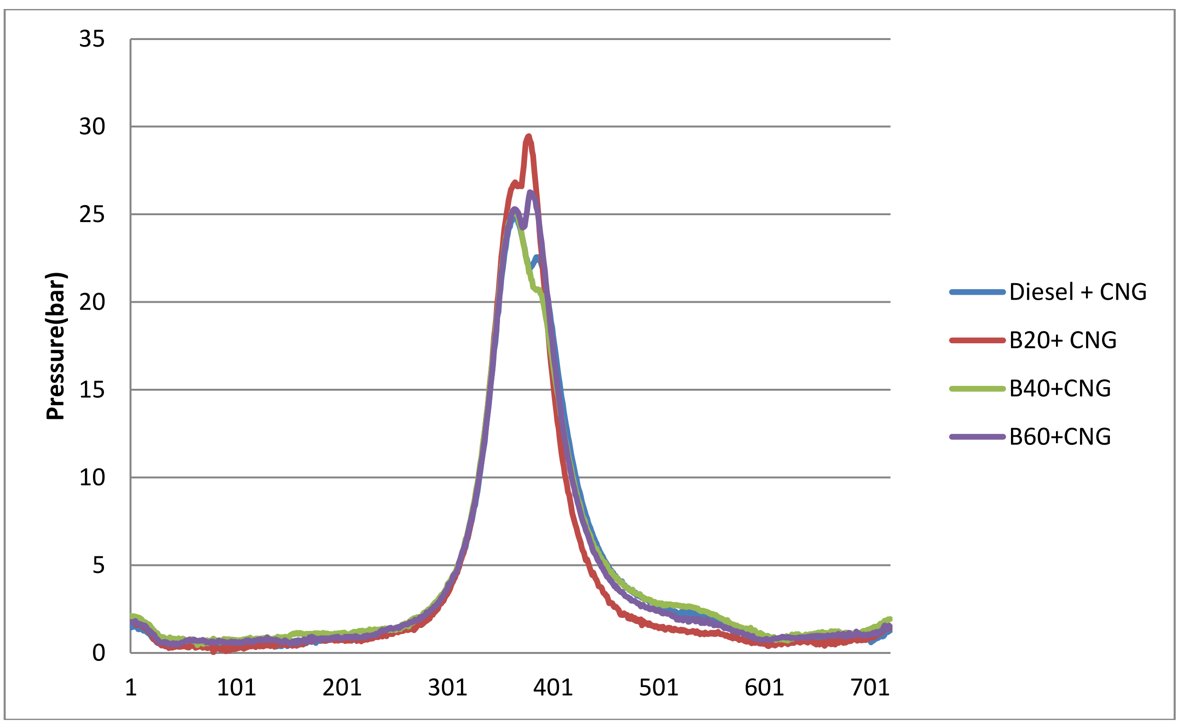

Figure 5.12b: P-θ Curve at 1 kW load for dual mode

c) **Load: 2kW**

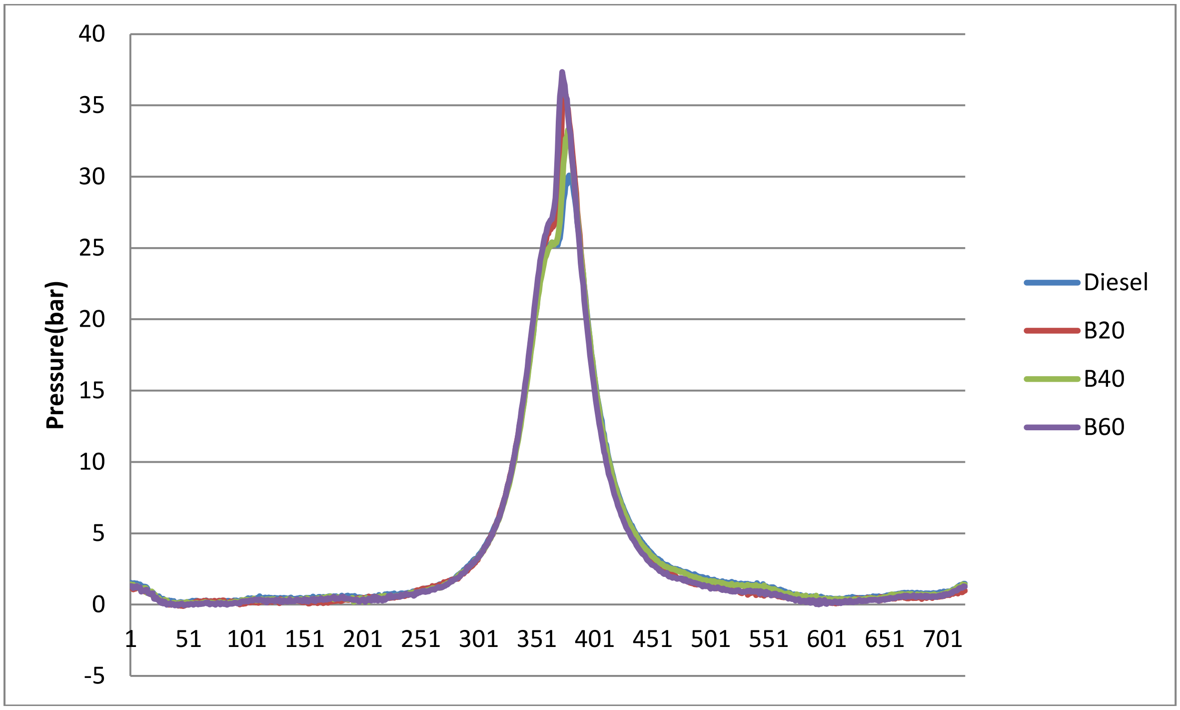

Figure 5.13a: P-θ Curve at 2 kW load

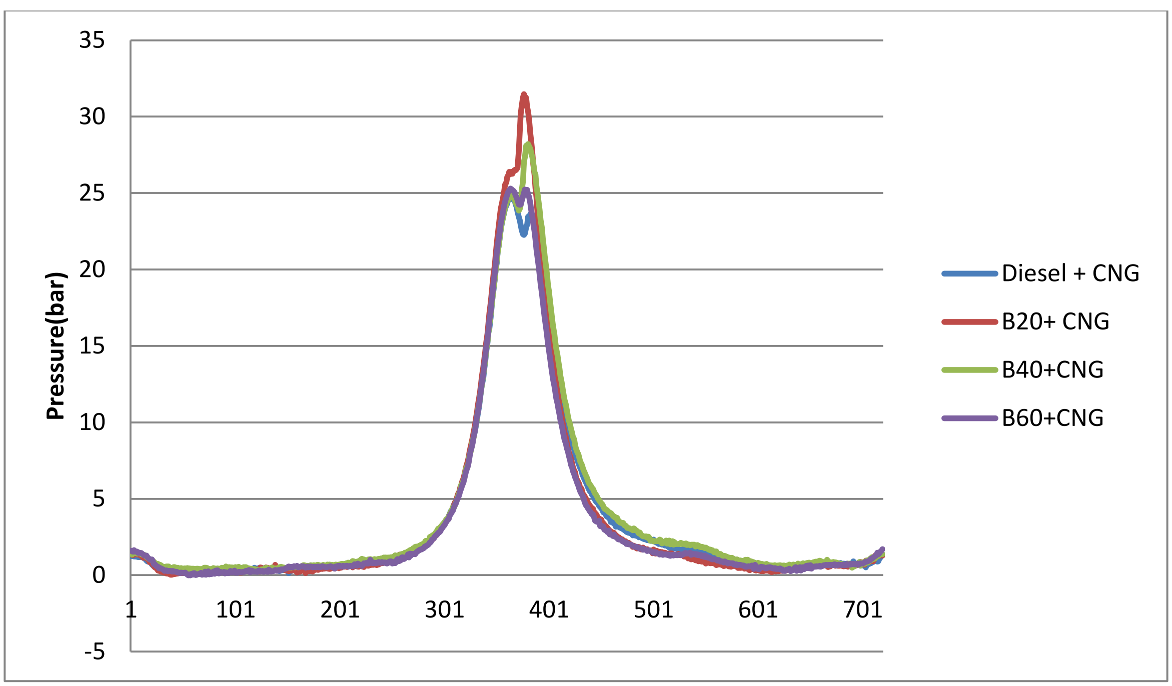

Figure 5.13b: P-θ Curve at 2 kW load for dual mode

d) Load: 3 kW

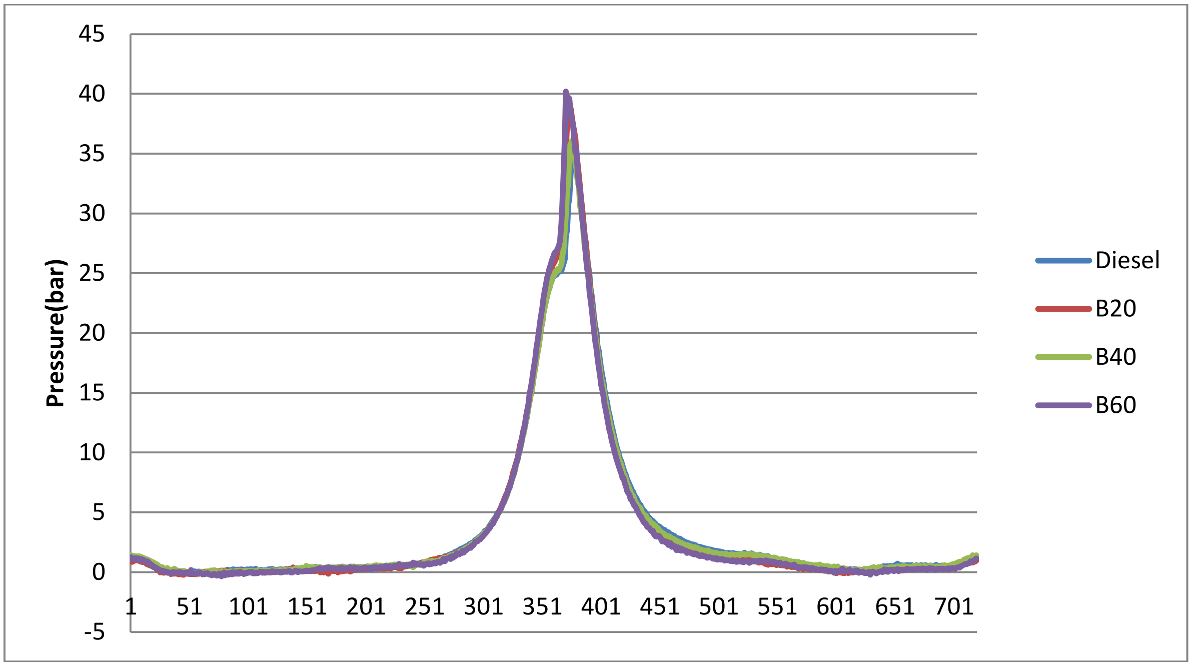

Figure 5.14a: P-θ Curve at 3 kW load

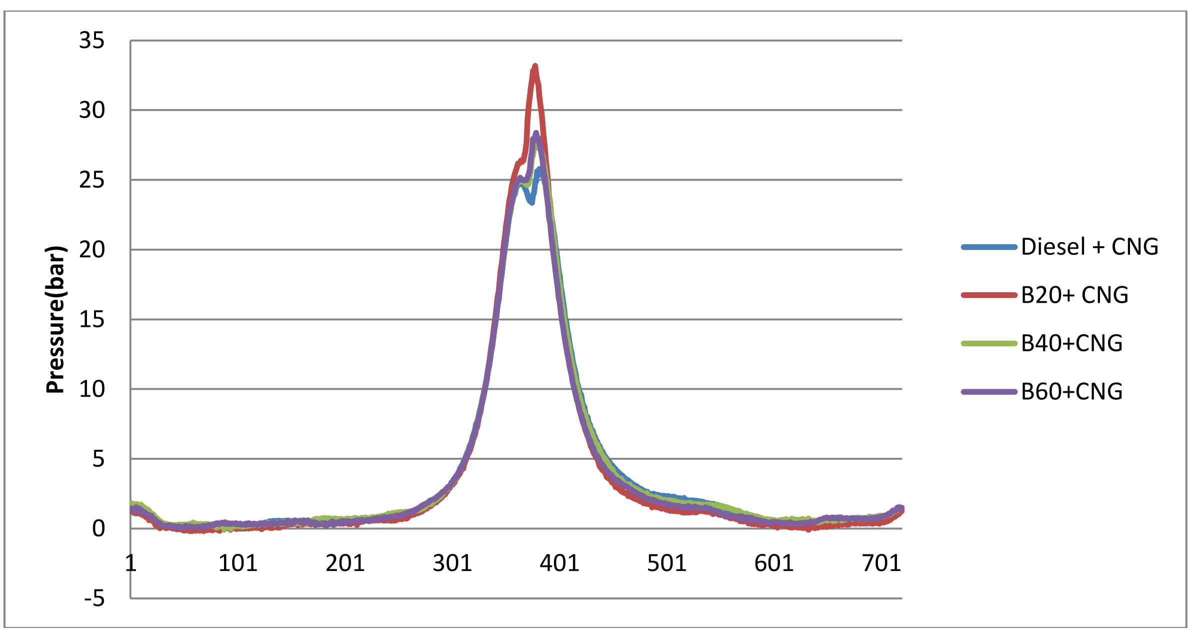

Figure 5.14b: P-θ Curve at 3 kW load for dual mode

d) **Load: 3.5 kW**

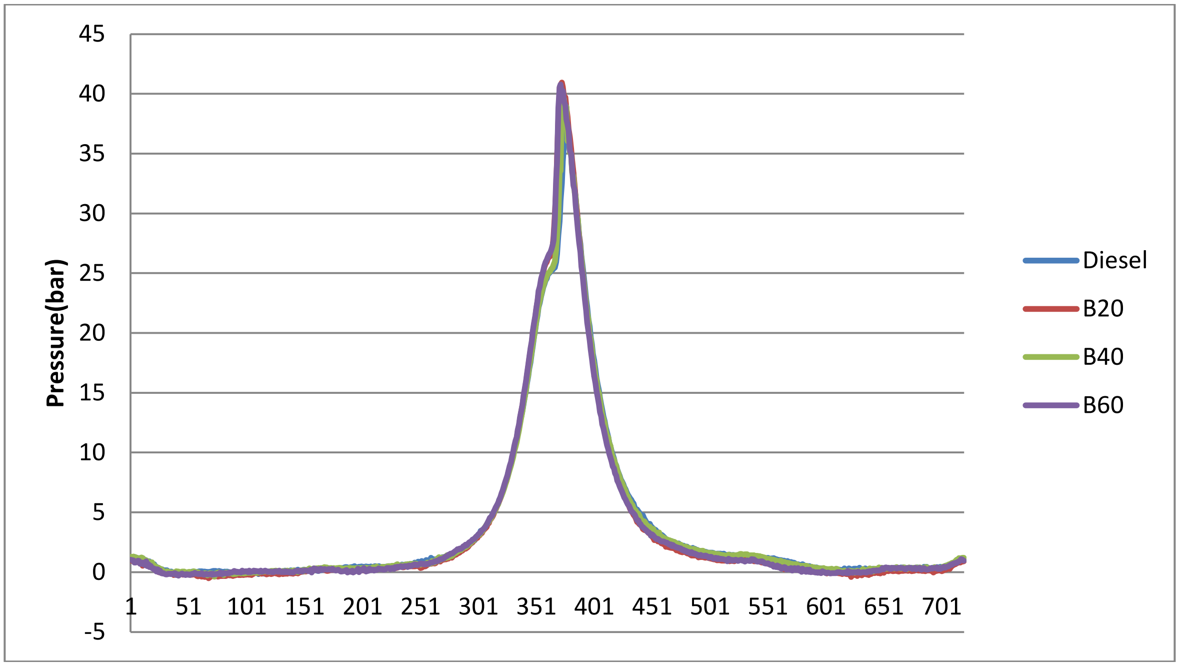

Figure 5.15a: P-θ Curve at 3.5 kW load

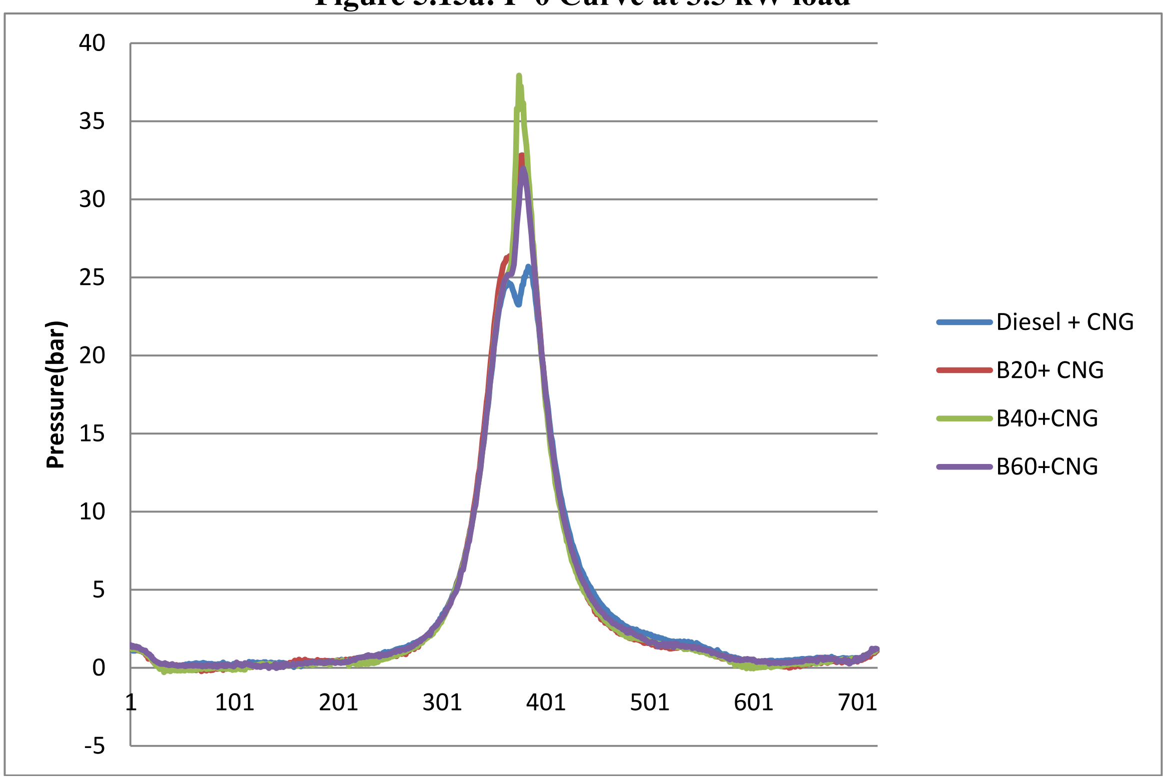

Figure 5.15b: P-θ Curve at 3.5 kW load for dual mode

5.3.1.d P-θ COMPARISON FOR VARYING CNG FLOW RATE

a) LOAD: IDLE

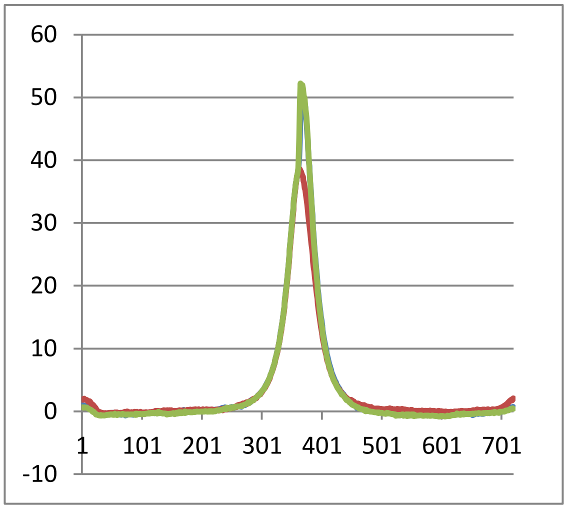

Figure 5.16a: Diesel

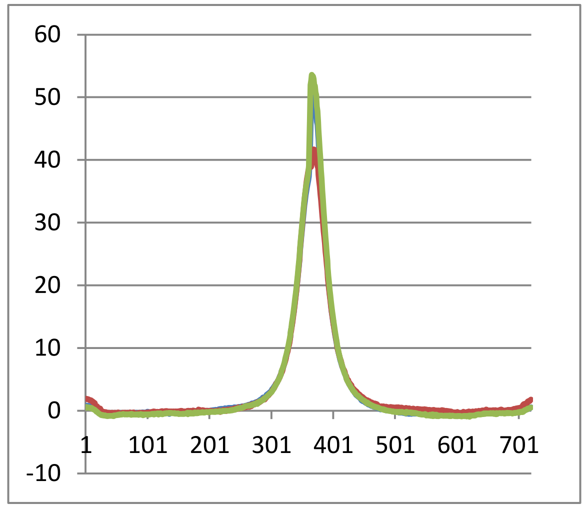

Figure 5.16b: B20

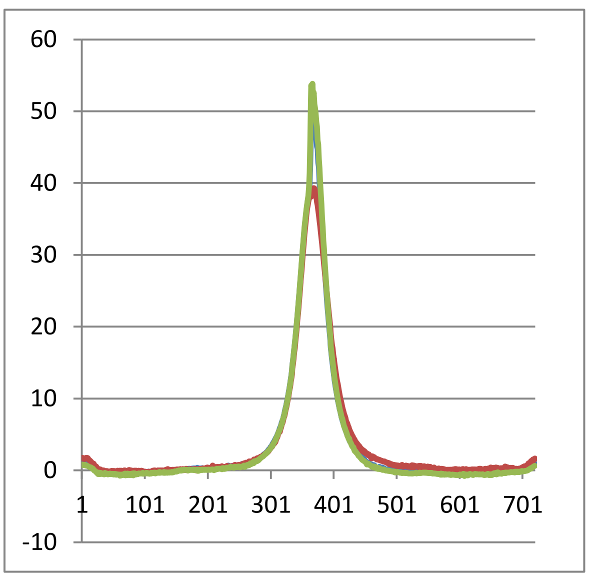

Figure 5.16c: B40

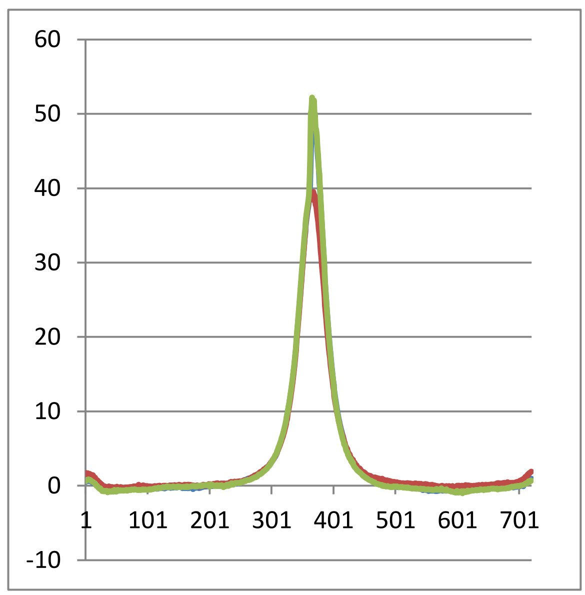

Figure 5.16d: B60

FIGURE 5.16: P-θ & FLOW RATE COMPARISON FOR IDLE LOAD

b) LOAD: 1 kW

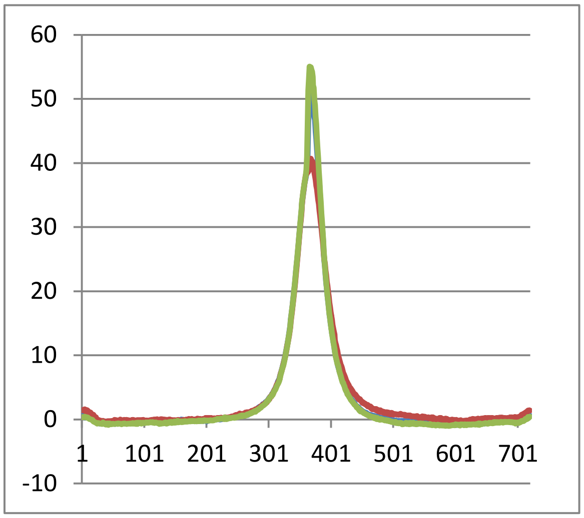

Figure 5.17a Diesel

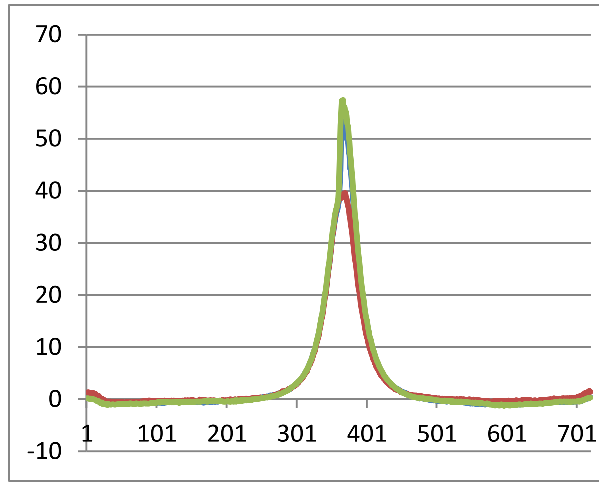

Figure 5.17b B20

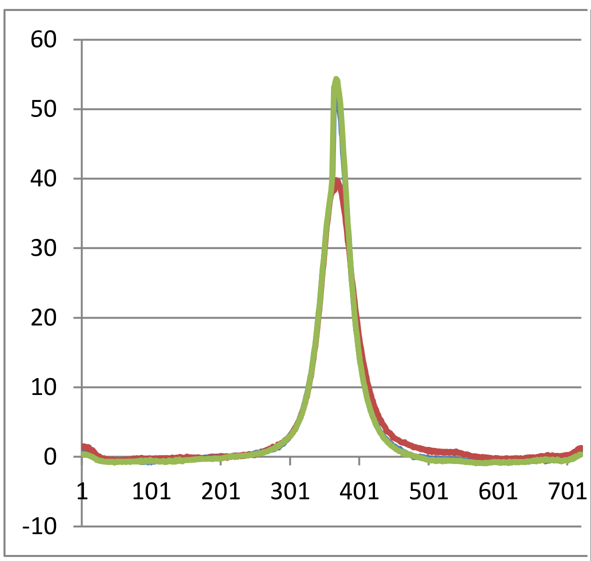

Figure 5.17c B40

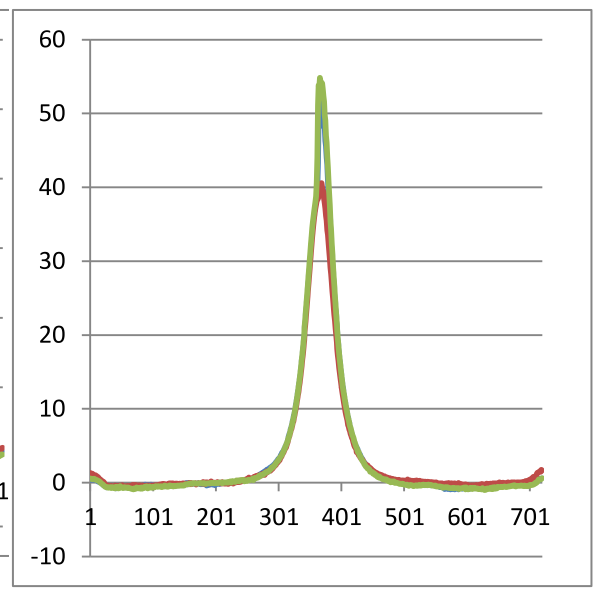

Figure 5.17d B60

FIGURE 5.17 P-θ & FLOW RATE COMPARISON FOR 1 kW LOAD

c) LOAD 2 kW

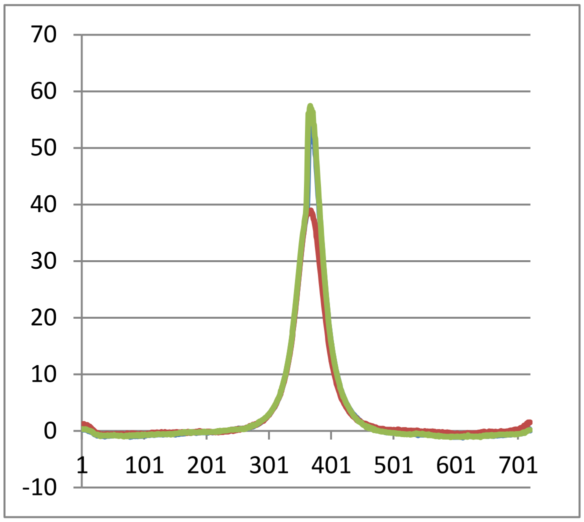

Figure 5.18a Diesel

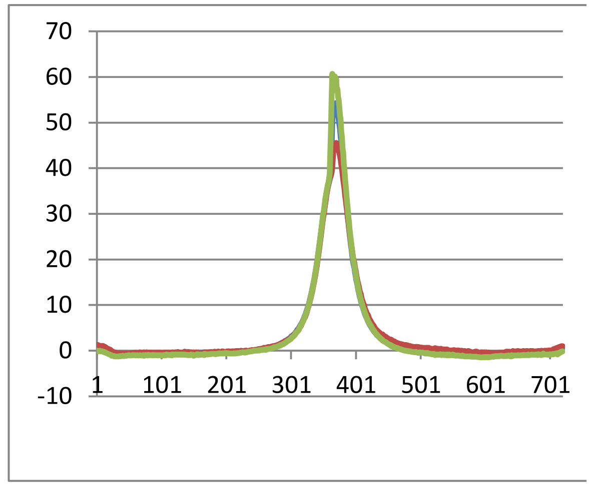

Figure 5.18b B20

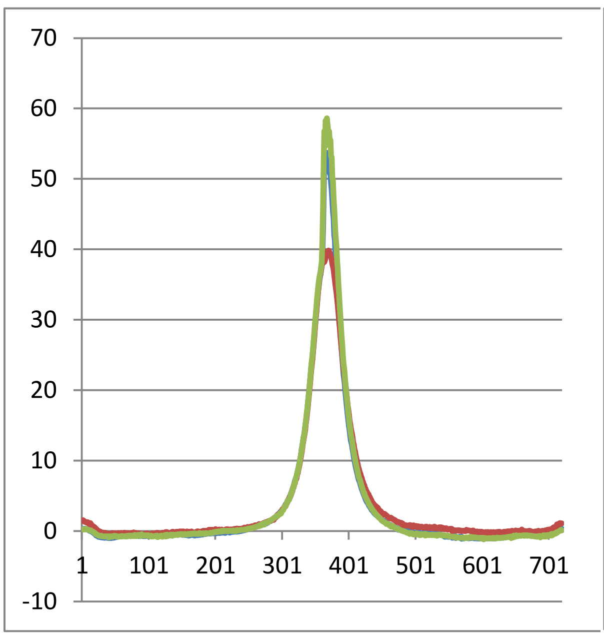

Figure 5.18c B40

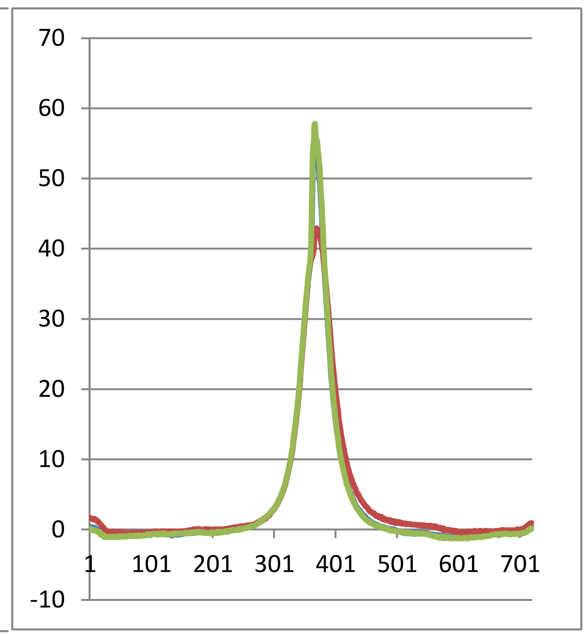

Figure 5.18d B60

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FIGURE 5.18 P-θ & FLOW RATE COMPARISON FOR 2 kW LOAD

\

d) LOAD 3 kW

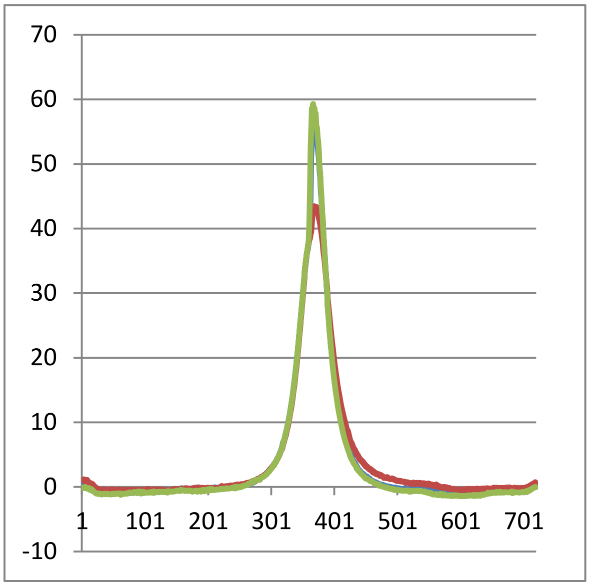

Figure 5.19a Diesel

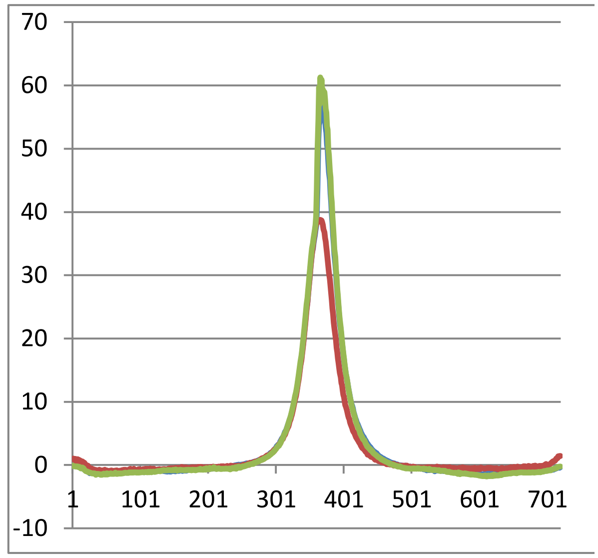

Figure 5.19b B20

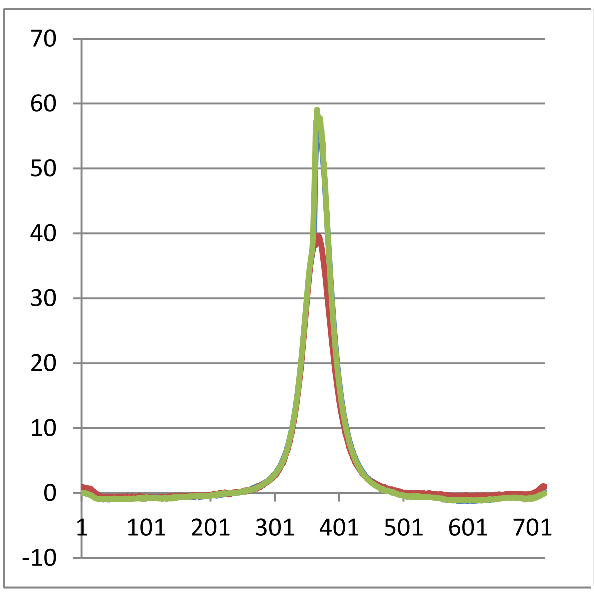

Figure 5.19c B40

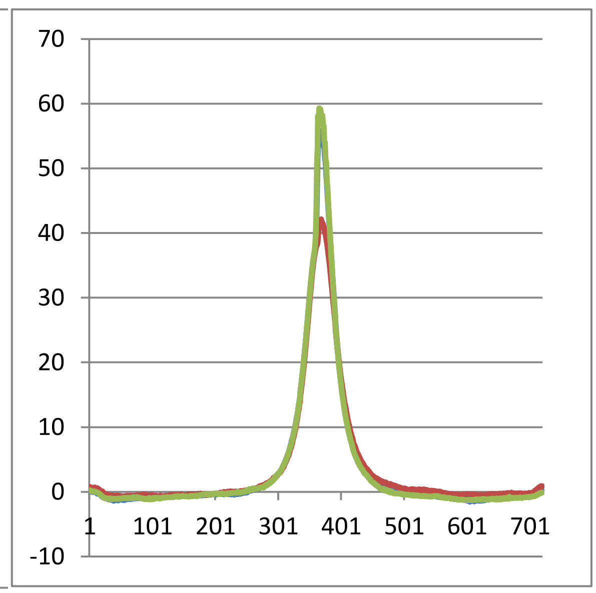

Figure 5.19d B60

FIGURE 5.19 P-θ & FLOW RATE COMPARISON FOR 3 kW LOAD

e) **LOAD 3.5 kW**

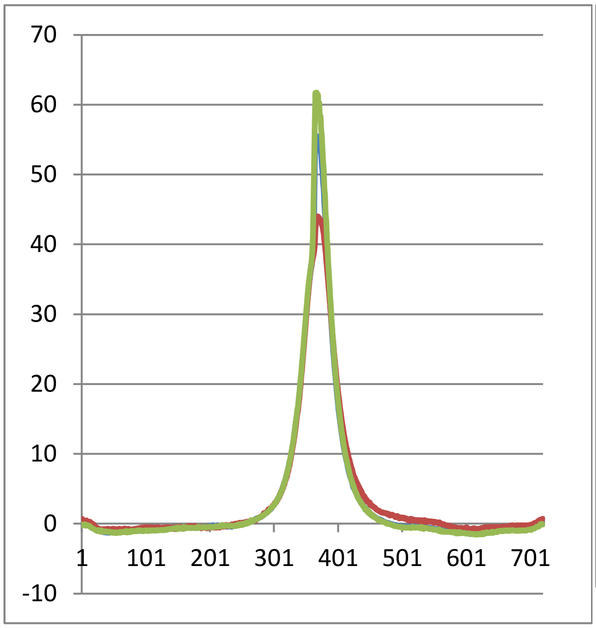

Figure 5.20a Diesel

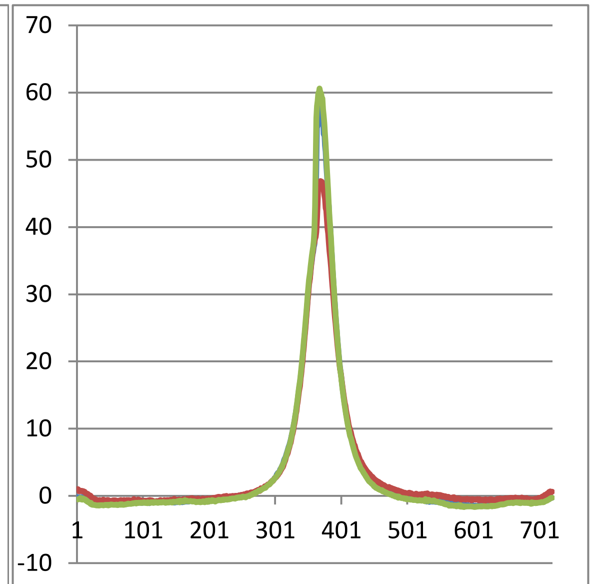

Figure 5.20b B20

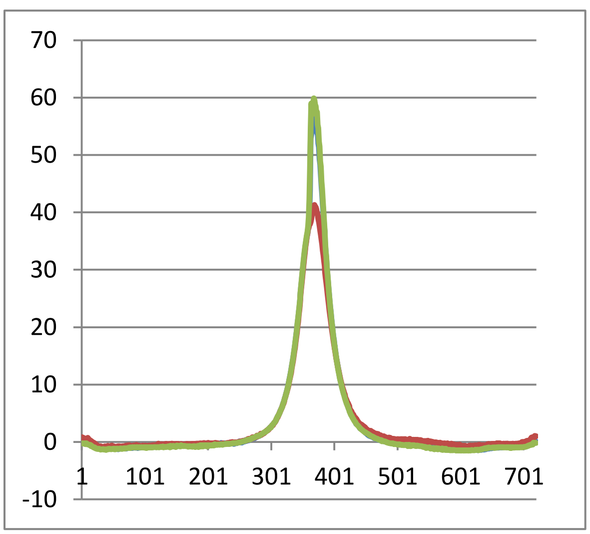

Figure 5.20c B40

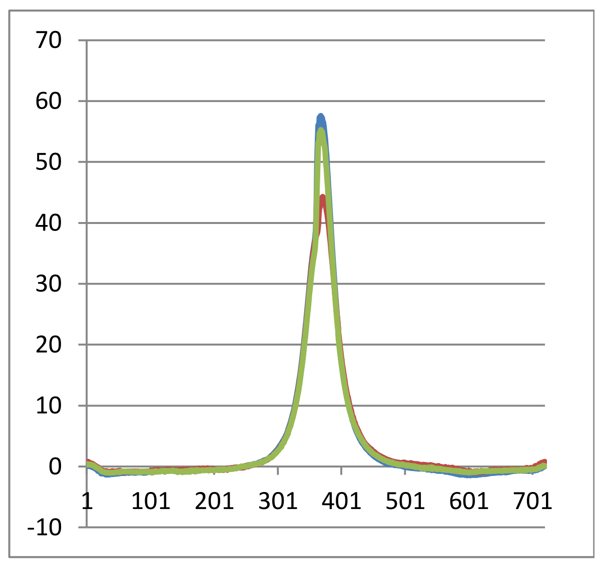

Figure 5.20d B60

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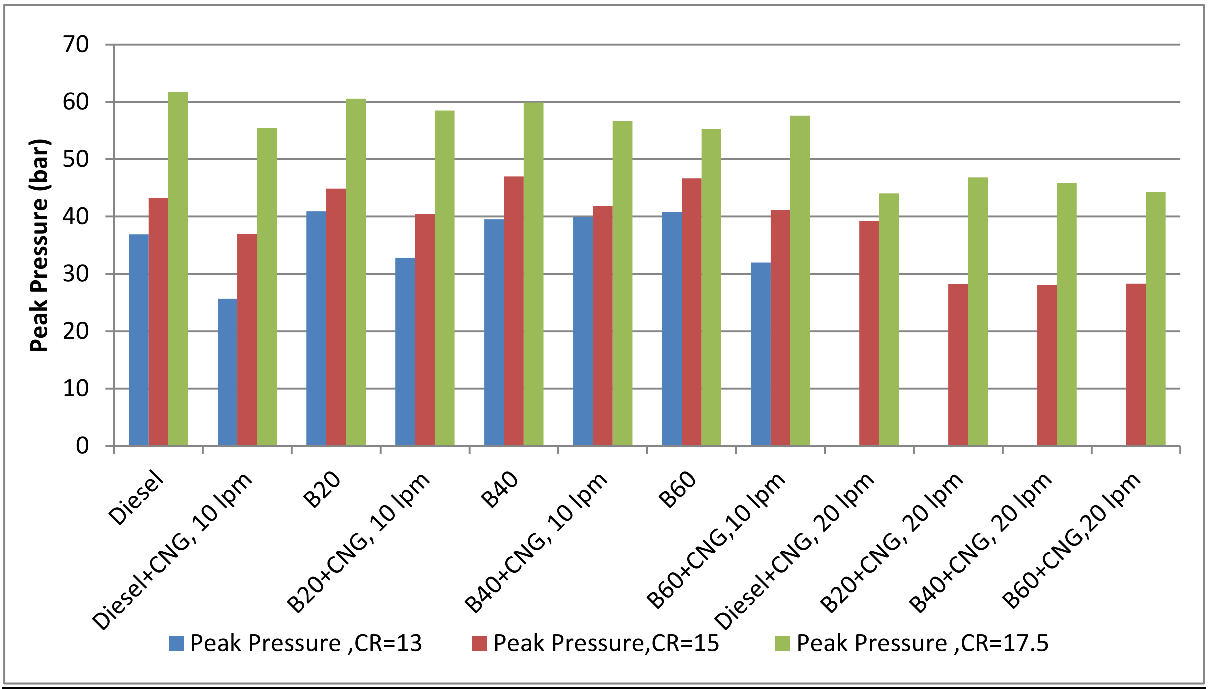
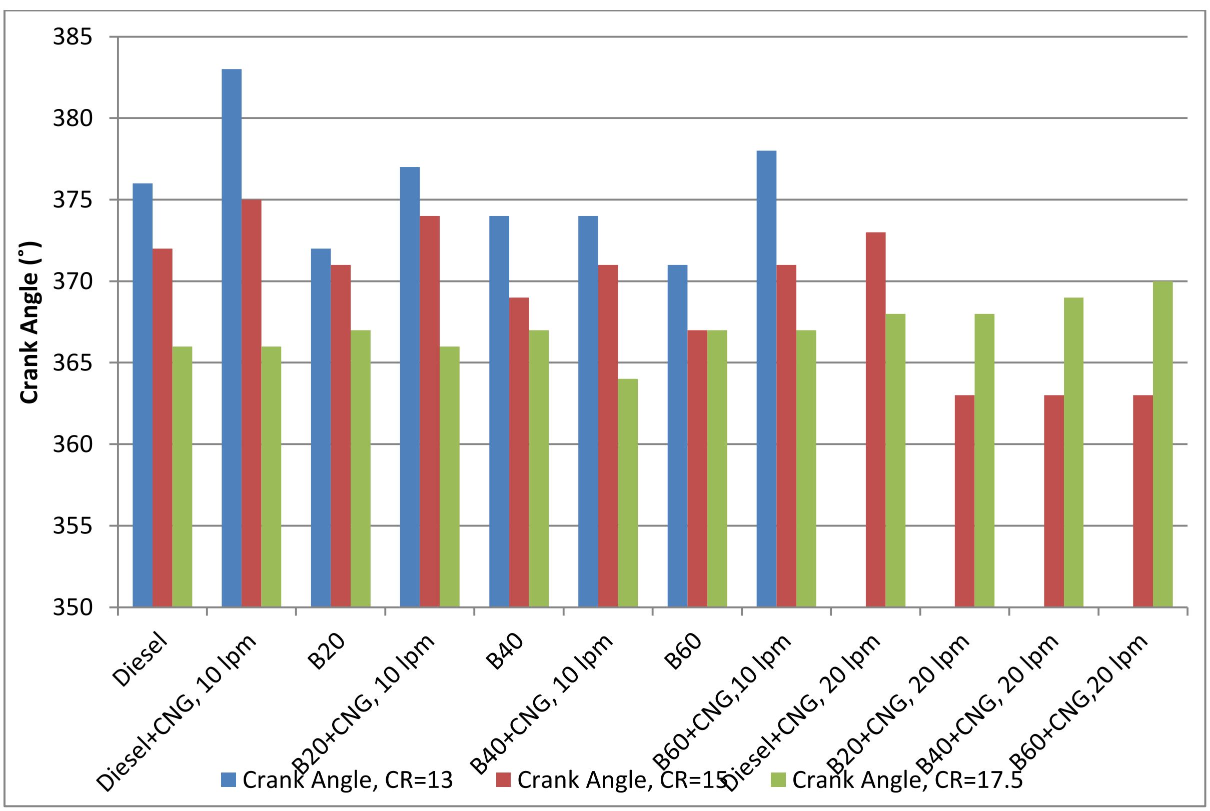
FIGURE 5.20 P-θ & FLOW RATE COMPARISON FOR 3.5 kW LOAD

5.20 a Peak Pressure Comparison for diesel, blends at different compression ratio and different flow rate at Rated Load



Crank Angle Comparison for diesel, blends at different compression ratio and different flow rate at Rated Load



5.3.2 BRAKE THERMAL EFFICIENCY

The Brake Thermal efficiency is plotted against the load and the curves for various blends are plotted together for both pure fuel and dual fuel mode.

The Brake Thermal efficiency when the engine runs in diesel and its blends is in the range of 32% - 34% at the rated load of 3.5 kW. When the engine is run in Dual Fuel Mode the BTHE is slightly higher compared to that of pure fuel engine. This trend is observed irrespective of the compression ratio of the engine. Also it is observed that B 60 tends to give higher efficiency than that of diesel. We also observe that as the flow rate of the CNG is increased in the dual fuel mode, the BTHE of the engine decreases. This is because the energy supplied by the CNG is not utilized up to the full potential.

In a dedicated diesel engine heterogeneous mixture of diesel and air is formed i.e. diesel availability varies through out the combustion chamber resulting in improper combustion, whereas, in dual fuel engine, homogenous mixture of air and CNG is compressed. Now injected diesel produces a flame front which sweeps the whole CNG-air mixture resulting in a proper combustion throughout the chamber consequently. This results in higher thermal efficiency

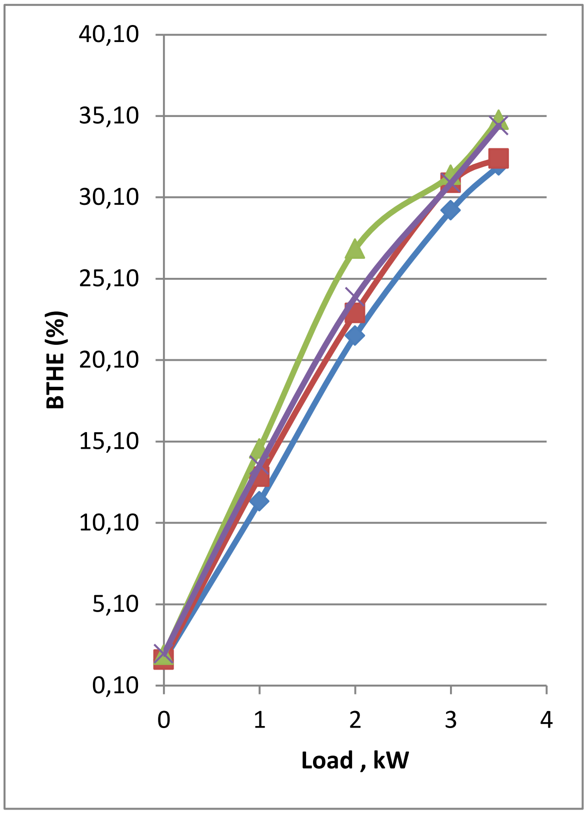


Figure 5.21a Pure Fuel, CR 17.5

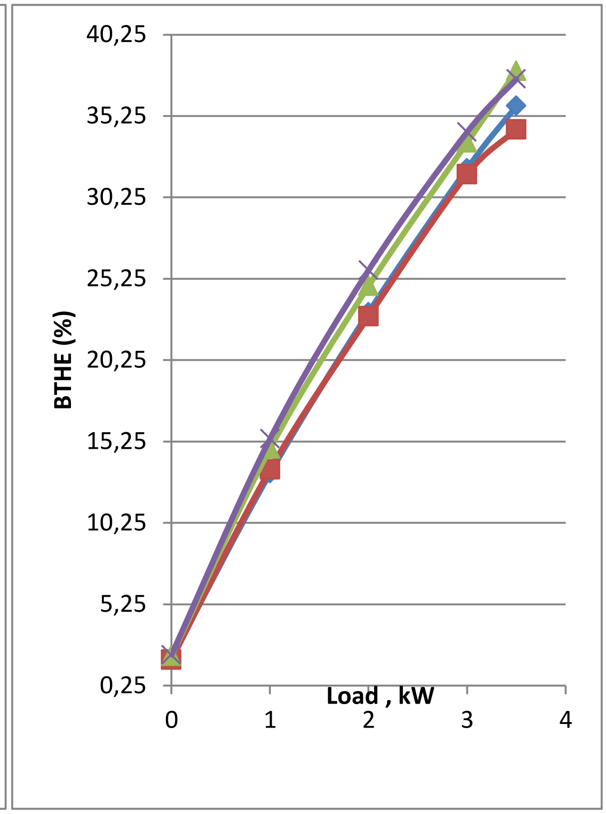


Figure 5.21b Dual Fuel CR 17.5

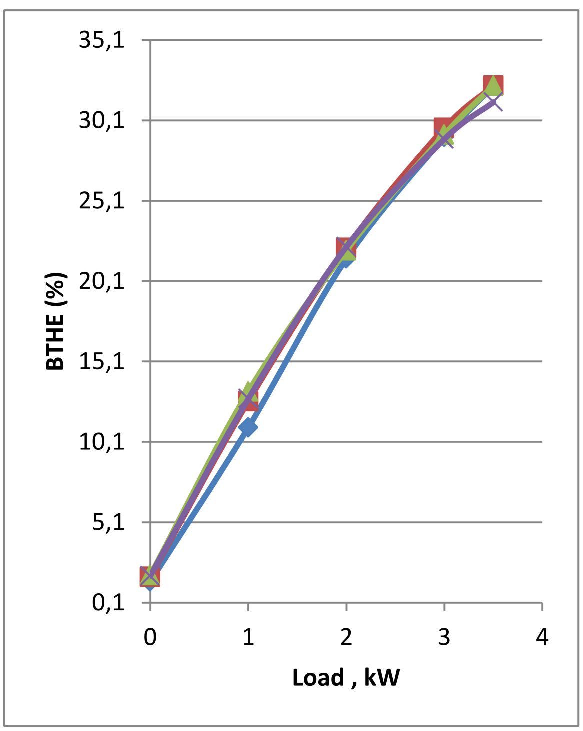


Figure 5.21c Pure Fuel, CR 15

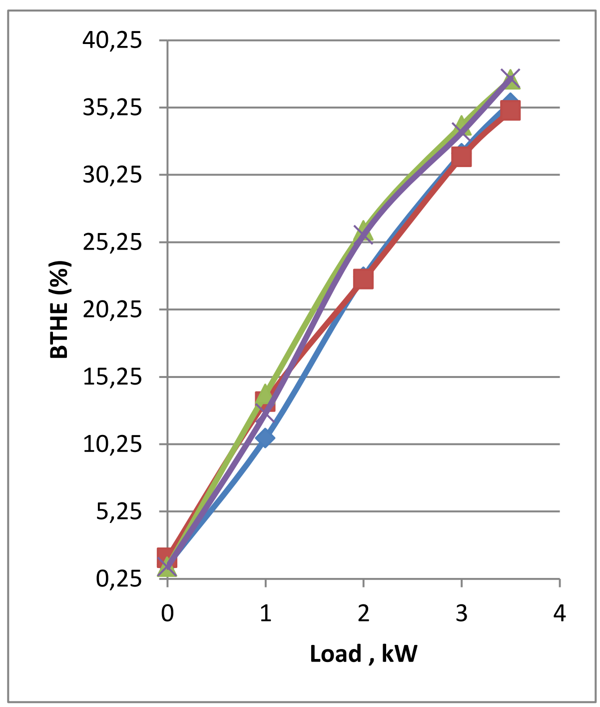


Figure 5.21d Dual Fuel, CR 15

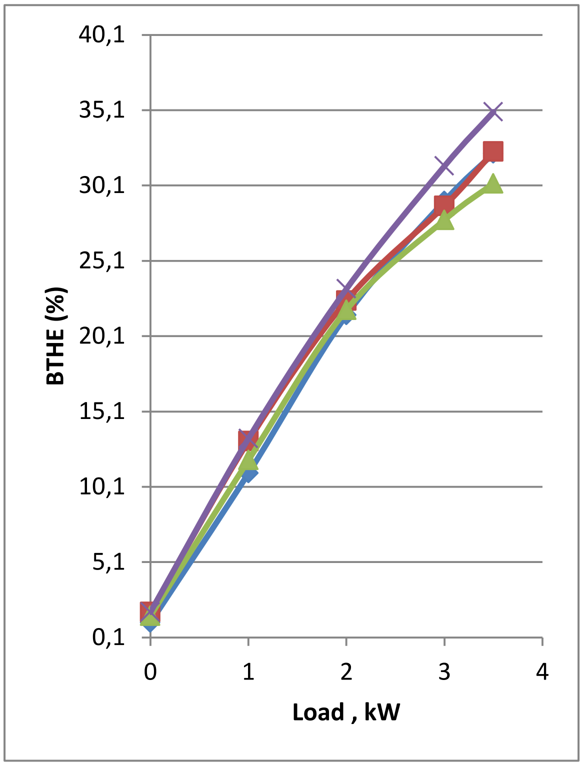


Figure 5.21e Pure Fuel, CR 13

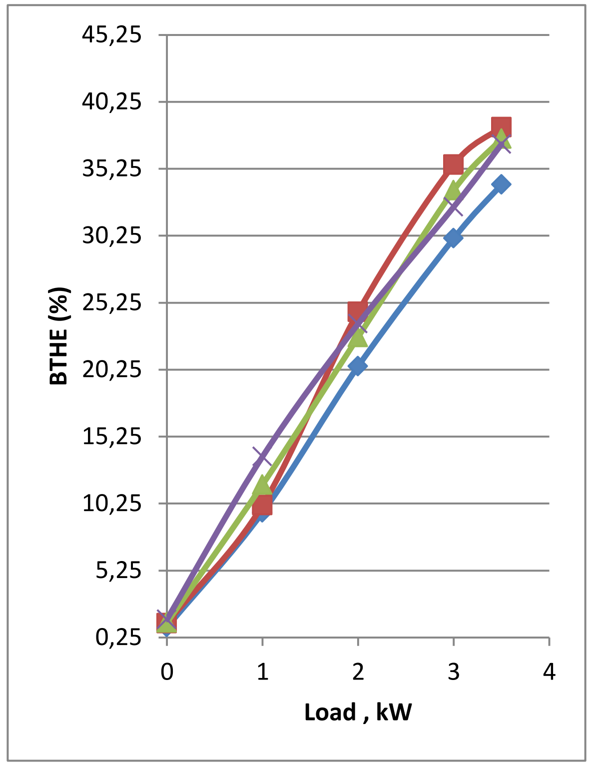


Figure 5.21f Dual Fuel, CR 13

Diesel B 20 B 40 B 60

Figure 5.21 Brake Thermal Efficiency Comparisons

5.3.3 Mechanical Efficiency

The following graphs compare the mechanical efficiency of the engine for the different fuels for both pure fuel and dual fuel operation.

We can conclude that as the load on the engine is increased the mechanical efficiency of the engine is increased. The mechanical efficiency at the rated load of 3.5 kW is about 92%, which is way higher than about 5% at idling. The mechanical efficiency trend line is almost same for the different fuels and also for a compression ratio. Also the mechanical efficiency in dual fuel mode operation shows similar trends.

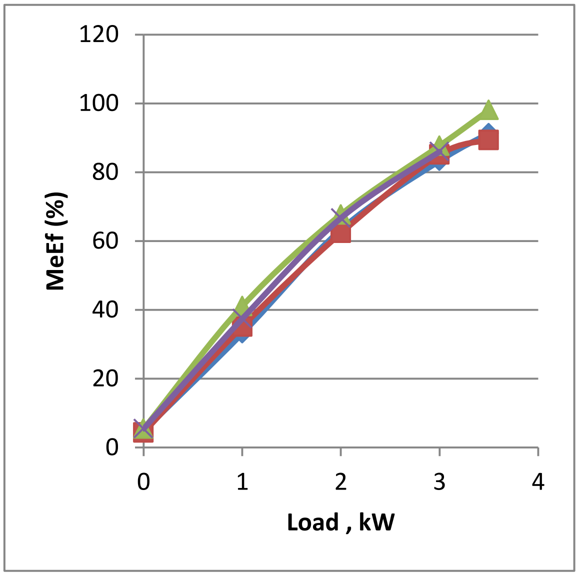


Figure 5.22a Pure Fuel, CR 17.5

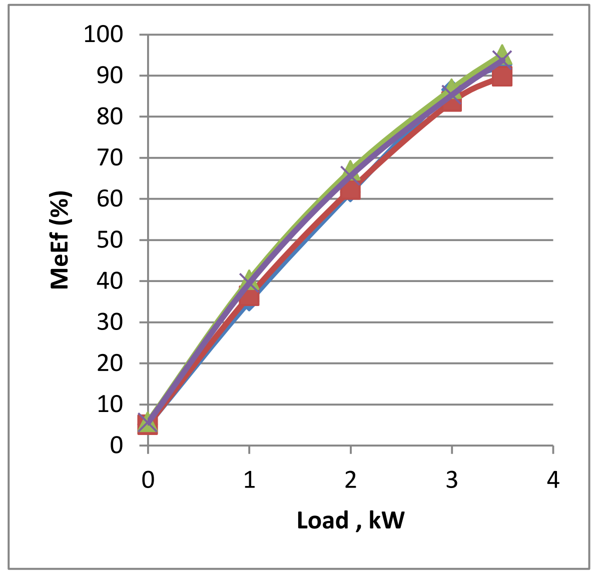


Figure 5.22b Dual Fuel, CR 17.5

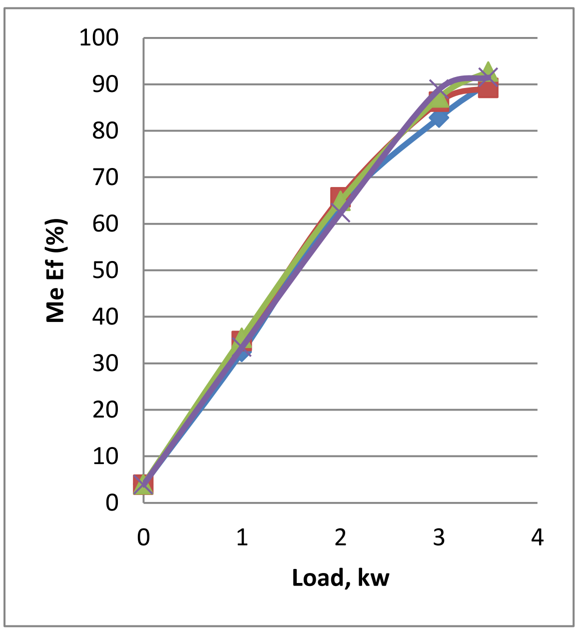


Figure 5.22c Pure Fuel, CR 15

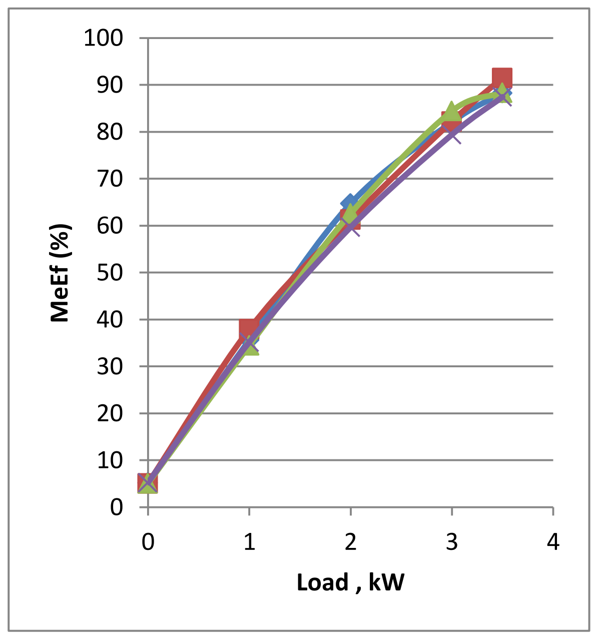


Figure 5.22d Dual Fuel, CR=15

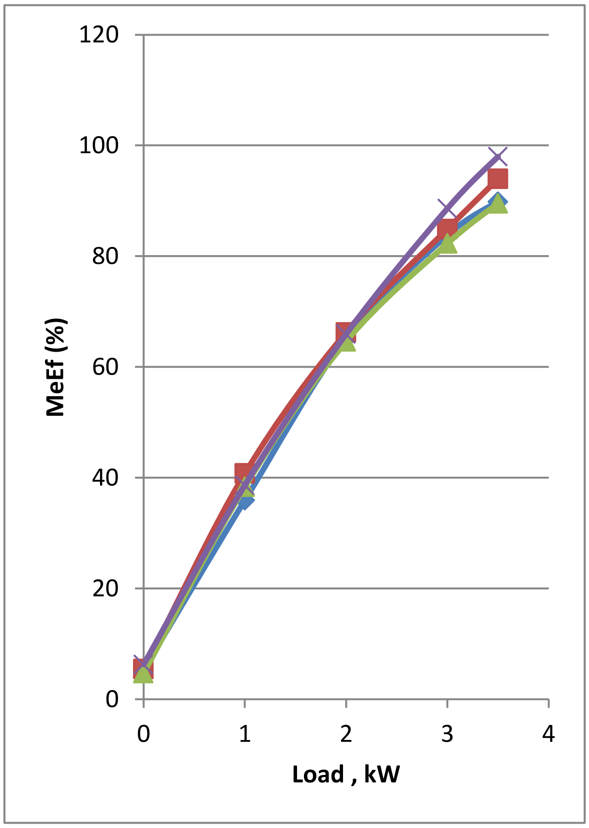


Figure 5.22e Pure Fuel, CR 13

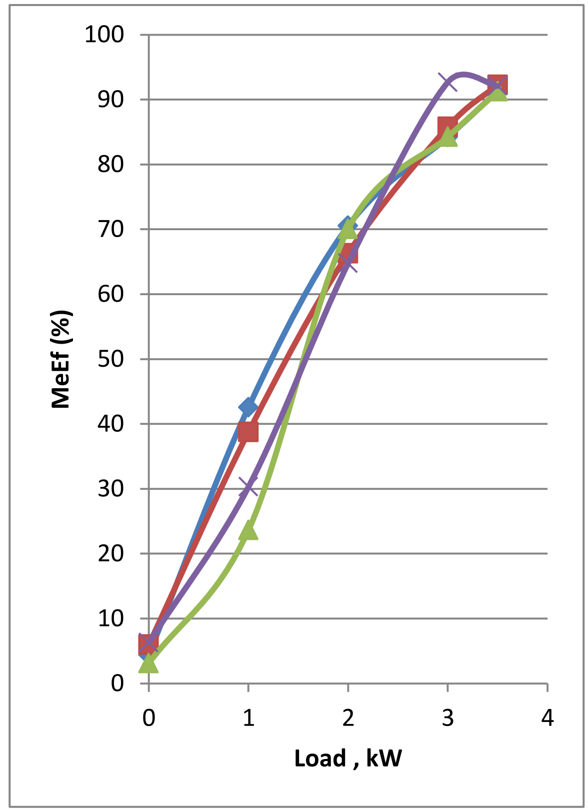


Figure 5.22f Dual Fuel, CR 13

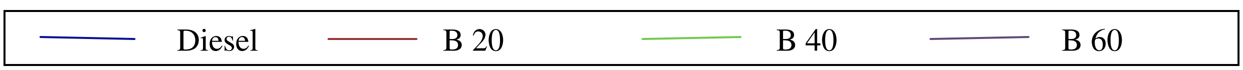


Figure 5.22 Mechanical Efficiency Comparisons

5.3.4 Specific Fuel Consumption

The specific fuel consumption of the engine with load is plotted. The graphs for different fuels are presented together.

It is observed that specific fuel consumption in a Dual Fuel Engine is lesser than a dedicated diesel engine. During the dual mode, diesel is used for pilot injection and thereafter CNG and diesel are combusting in the chamber simultaneously. Since CNG combustion is contributing to extra energy generation, to keep the engine speed constant governor attached to fuel pump reduces the diesel supply. Also the total fuel (CNG + Diesel) in terms of weight is less as compared to dedicated diesel mode since calorific value of CNG is more than that of diesel. Also as the CNG flow rate is increased the SFC decreases and as the load is increased the SFC decreases. These trends are observed irrespective of the compression ratio and the fuel used.

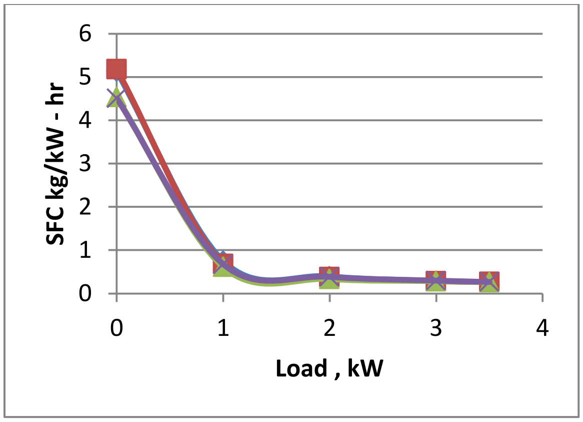


Figure 5.23a Pure Fuel, CR 17.5

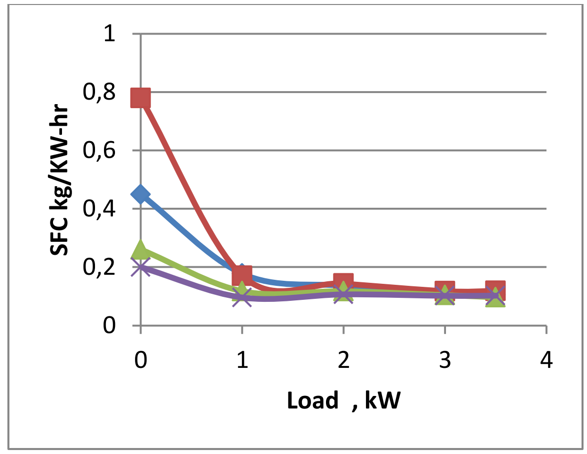


Figure 5.23b Dual Fuel, CR 17.5

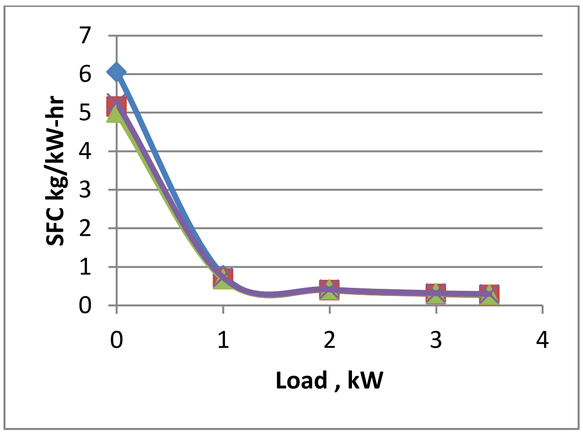


Figure 5.23c Pure Fuel, CR 15

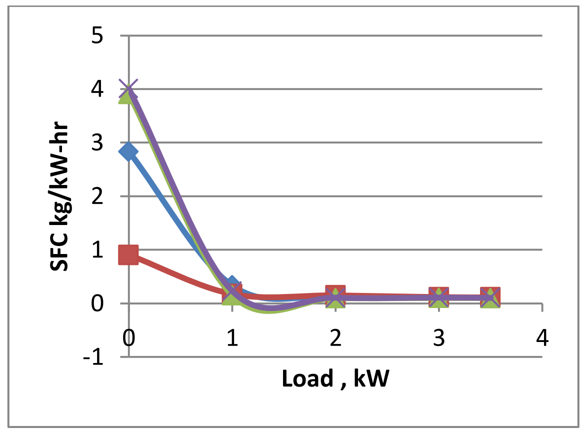


Figure 5.23d Dual Fuel, CR 15

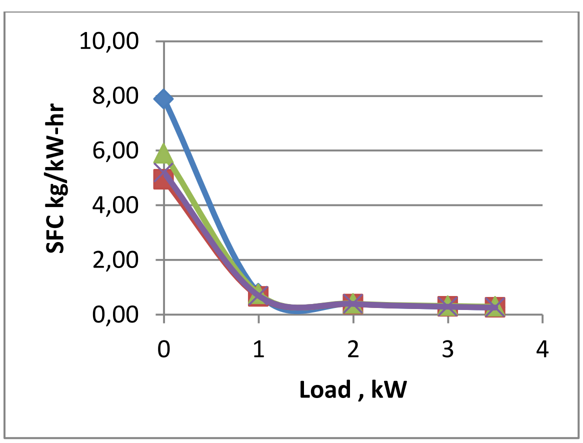


Figure 5.23e Pure Fuel, CR 13

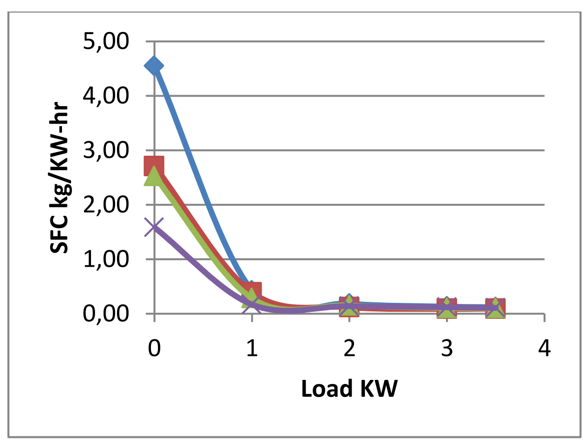


Figure 5.23f Dual Fuel, CR 13

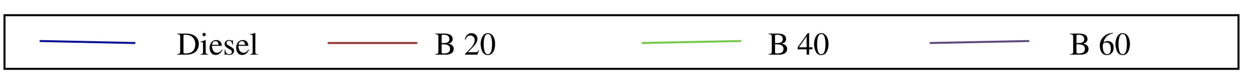


Figure 5.23 Specific Fuel Consumption Comparisons

5.3.5 Specific Energy Consumption

The graph below compares the specific energy consumption for different fuels in both pure fuel and dual fuel operation.

In case of the dual fuel operation, the SFC accounts only for the fuel supply of diesel. Thus, it is unjust to compare only the SFC at dual fuel operation with that to in pure fuel. The CNG should also be accounted for the dual fuel mode. Thus we compare the Specific energy consumption, which is defined as the total energy supplied by the fuel (Diesel + CNG) per unit Brake Power. From the SEC curves, we can observe that there is a sudden decrease in SEC as the load is increased from idle to 1 kW. And then there is a further decrease in the SEC as the load is increased. However, the SEC is higher for a higher flow rate of CNG as the thermal energy supplied to engine increases.

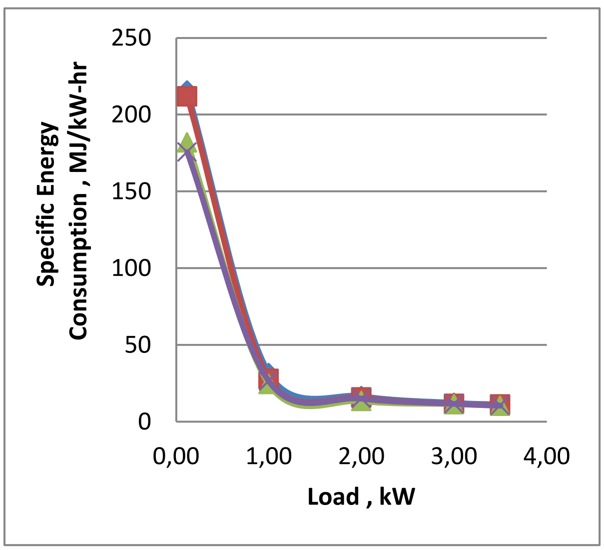


Figure 5.24a Pure Fuel, CR 17.5

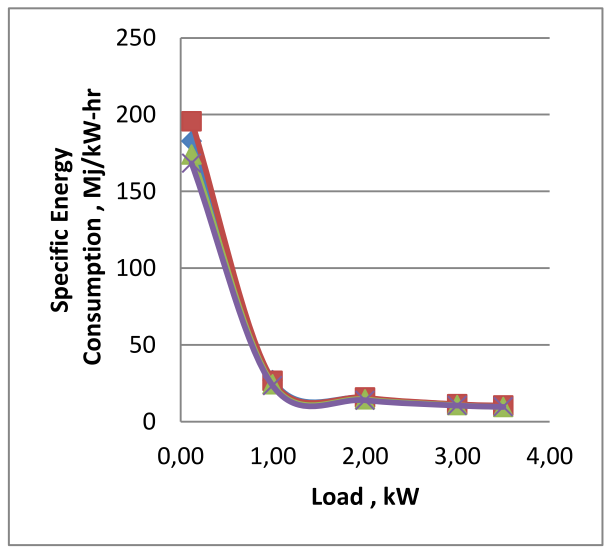


Figure 5.24b Dual Fuel, CR 17.5

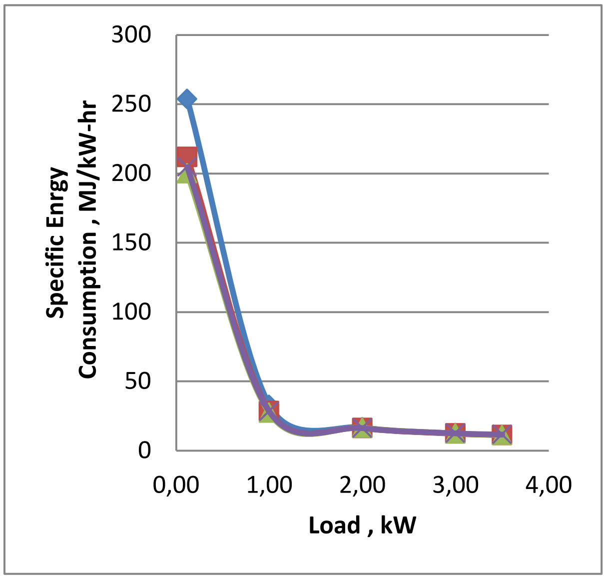


Figure 5.24c Pure Fuel, CR 15

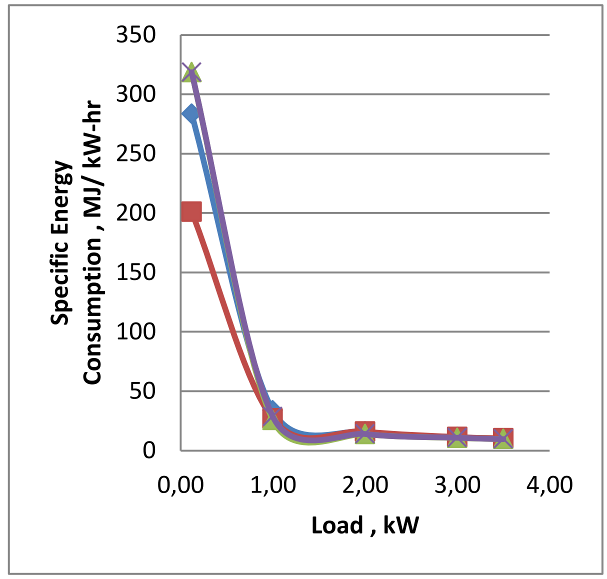


Figure 5.24d Dual Fuel, CR 15

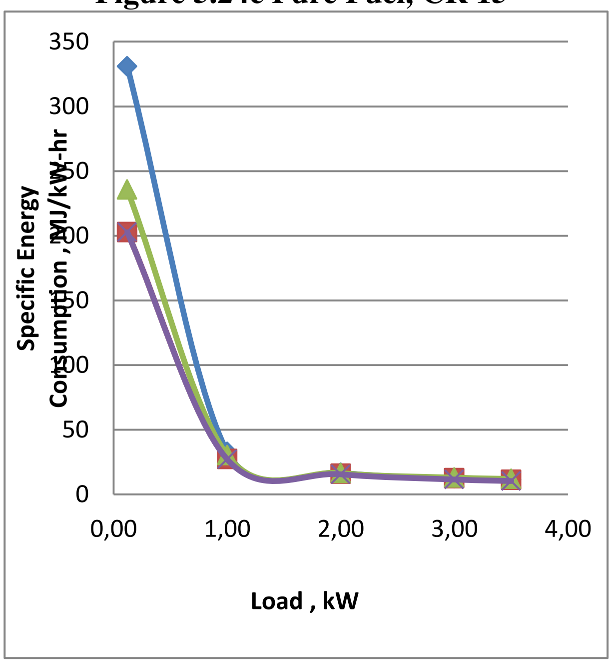


Figure 5.24e Pure Fuel, CR 13

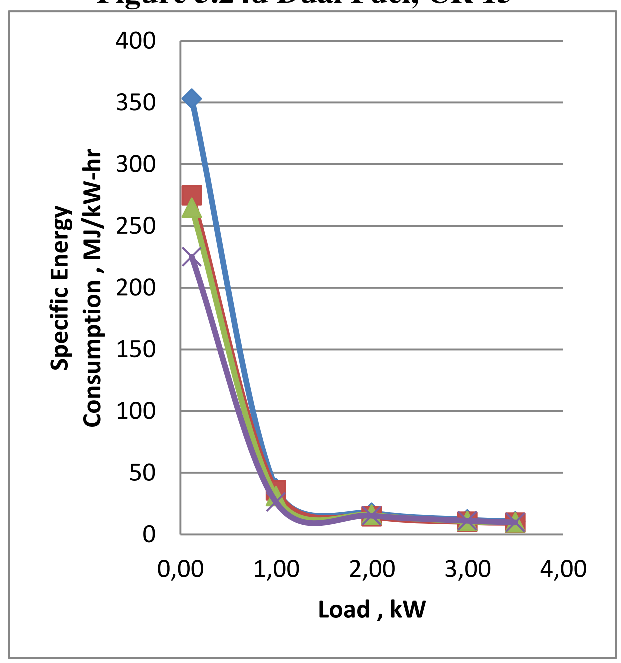


Figure 5.24f Dual Fuel, CR 13

Diesel B 20 B 40 B 60

Figure 5.24 Specific Energy Consumption Comparisons

5.3.6 ENGINE PERFORMANCE FOR VARYING CNG FLOW RATE

The engine performance characteristics vary with the flow rate of CNG in dual fuel operation. The graphs below show the variations in these for the different fuels.

a) **BTHE Comparison**

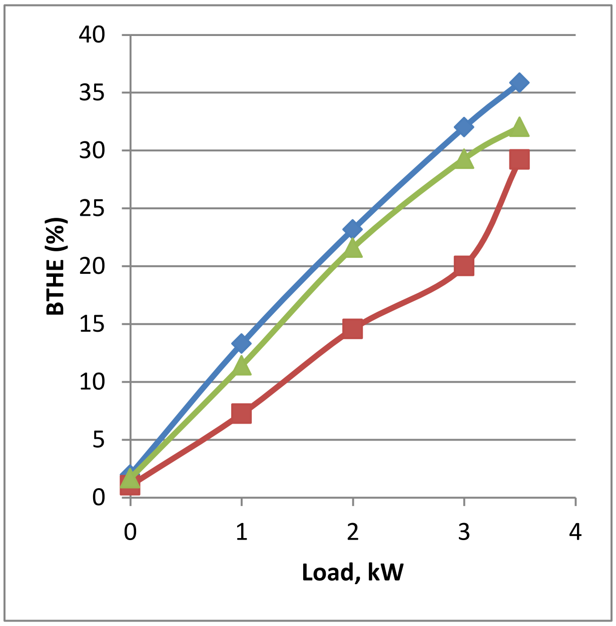


Figure 5.25a Diesel

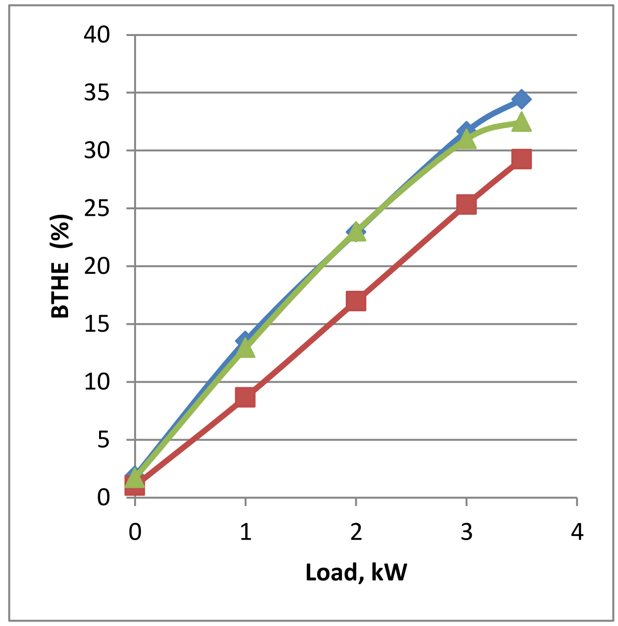


Figure 5.25b B20

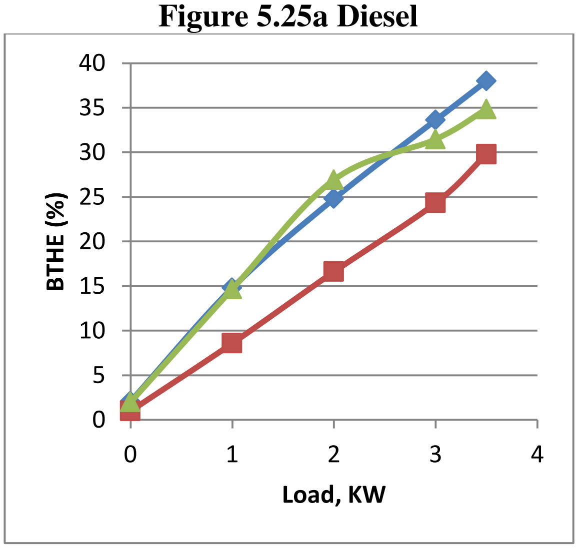


Figure 5.25c B40

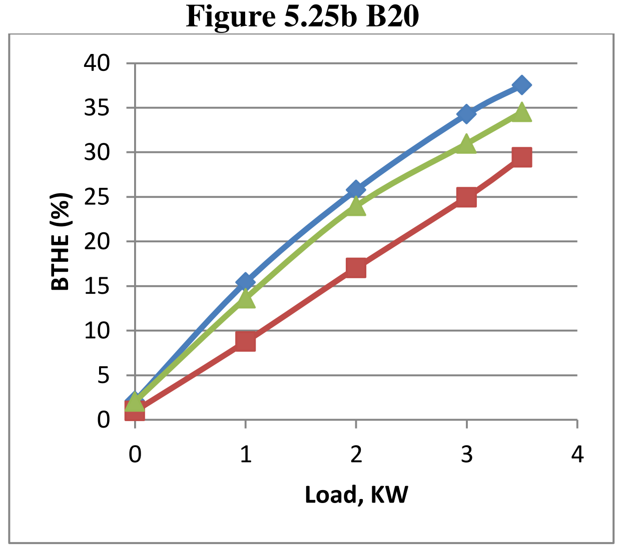


Figure 5.25d B60

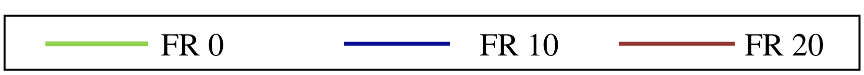


FIGURE 5.25 BTHE Comparisons(Dual Fuel)

b) **MeEf Comparison**

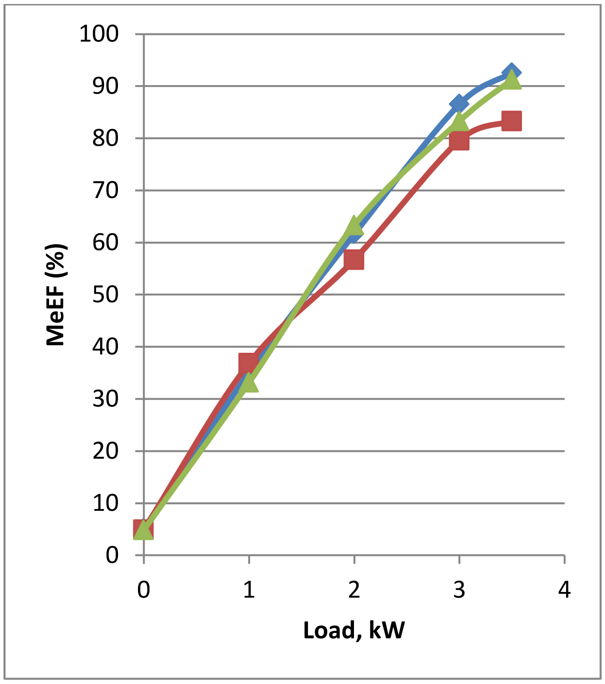


Figure 5.26a Diesel

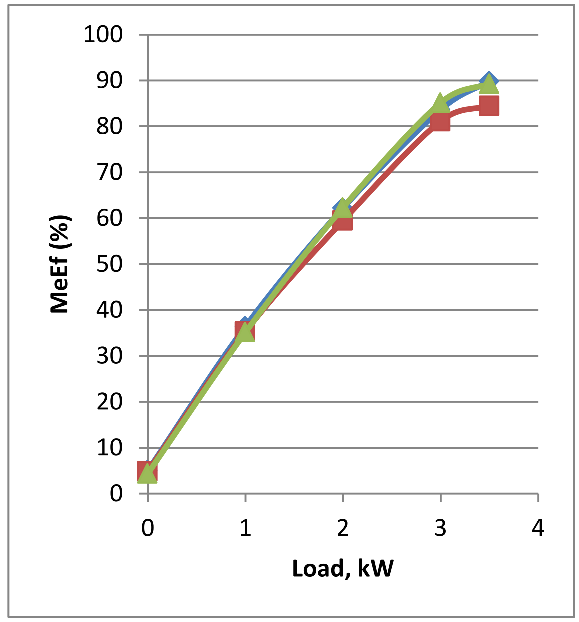


Figure 5.26b B20

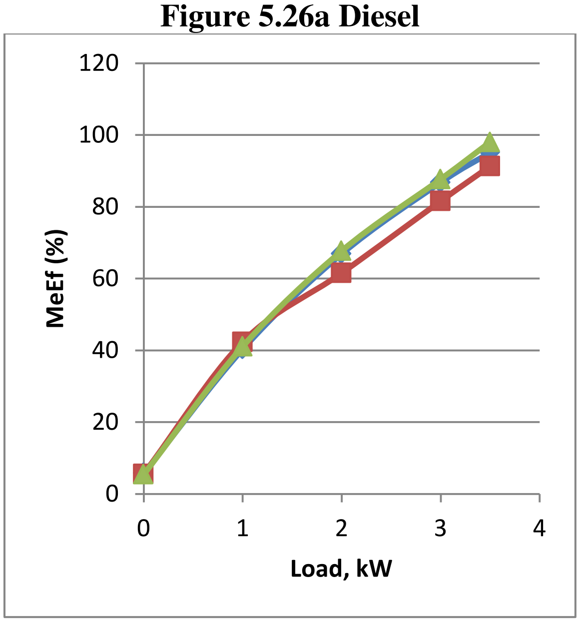


Figure 5.26c B40

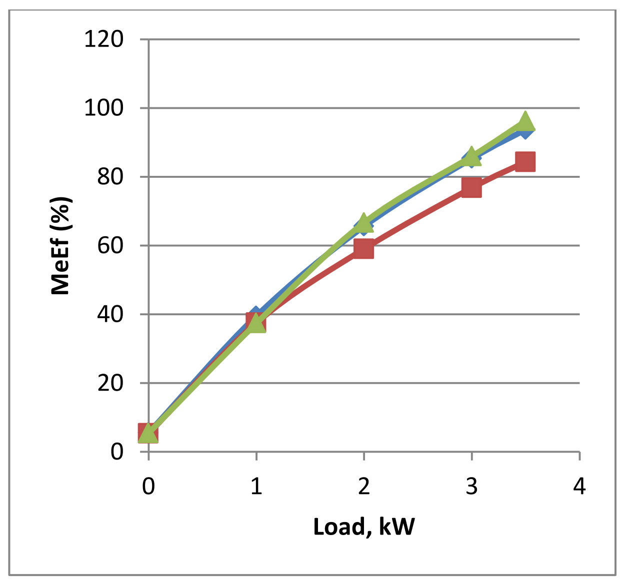


Figure 5.26d B60

FIGURE 5.26 MeEf COMPARISON (Dual Fuel)

c) **SFC :Comparison**

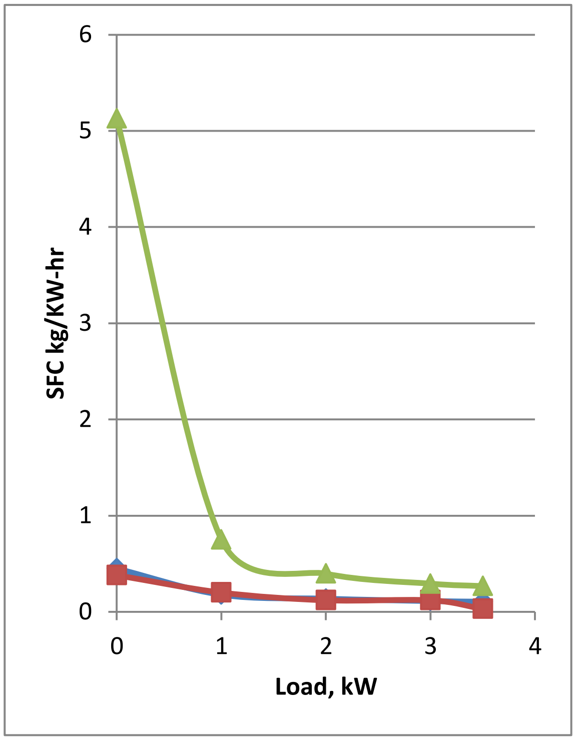


Figure 5.27a Diesel

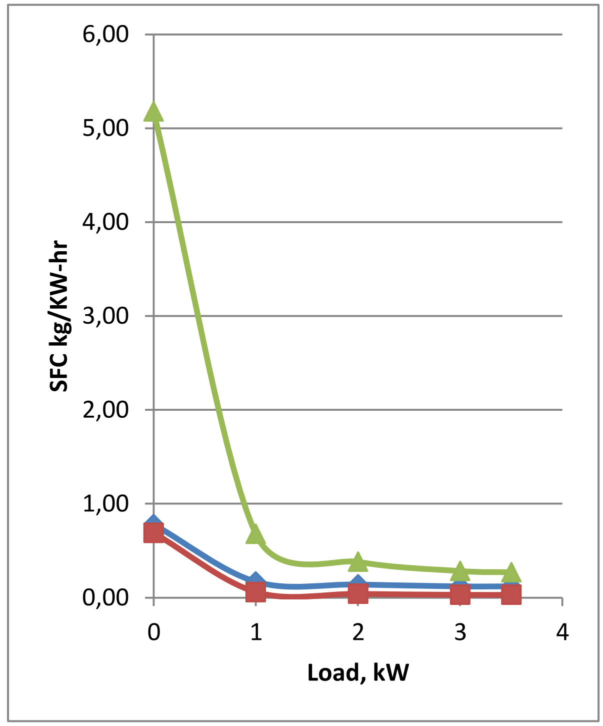


Figure 5.27b B20

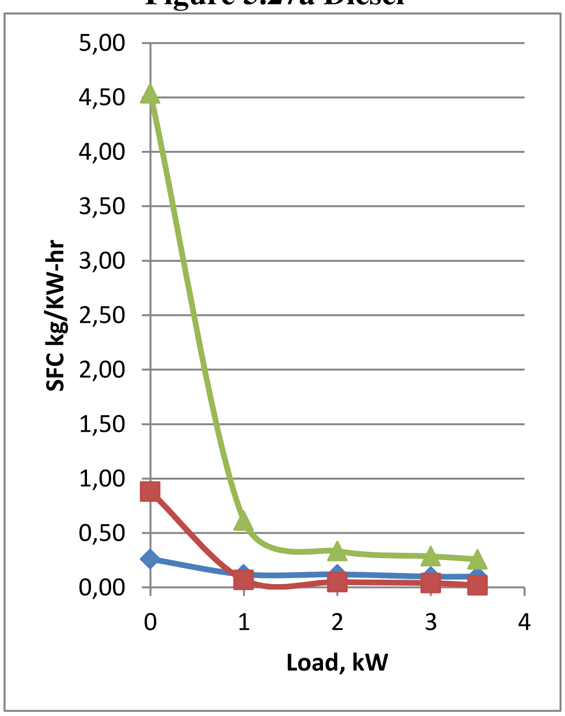


Figure 5.27c B40

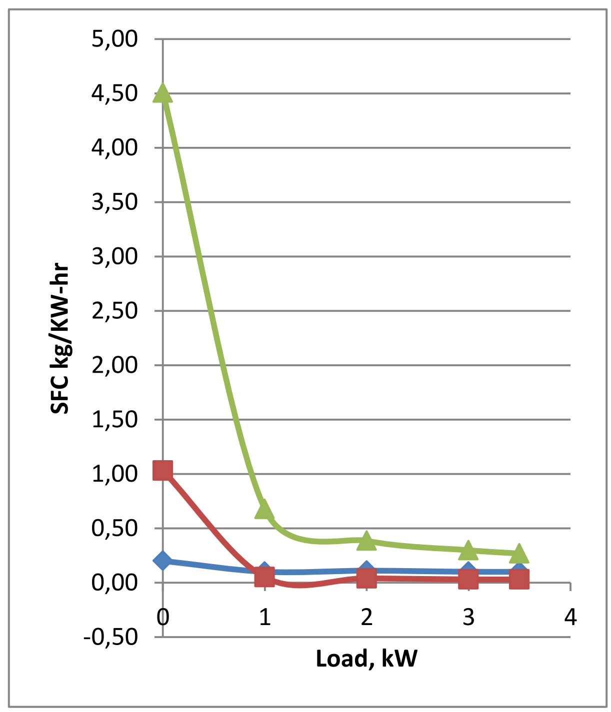


Figure 5.27d B60

FR 0 FR 10 FR 20

FIGURE 5.27 SFC Comparison (Dual Fuel)

d) **SPECIFIC ENERGY CONSUMPTION COMPARISION**

e)

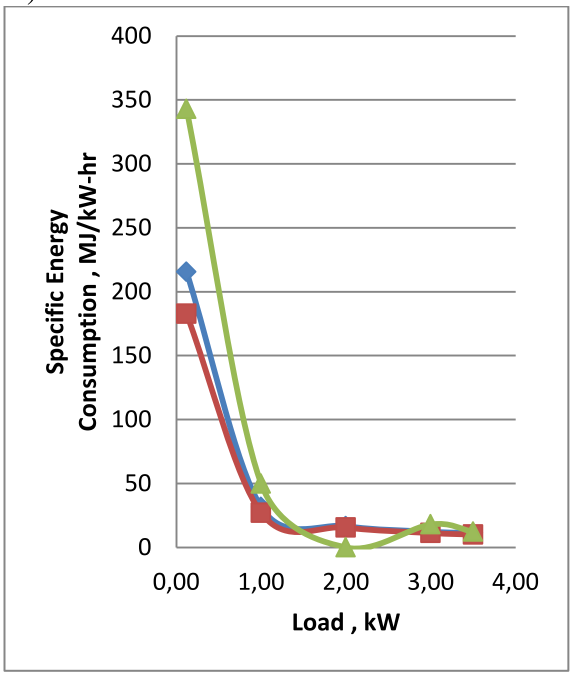


Figure 5.28a Diesel

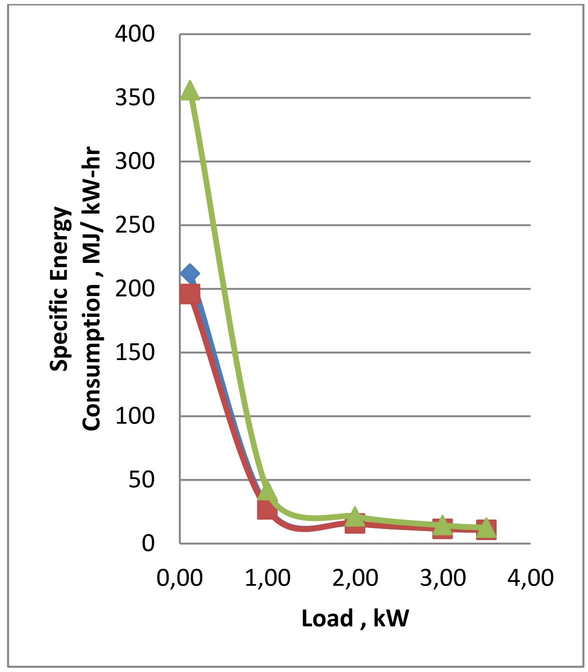


Figure 5.28b B20

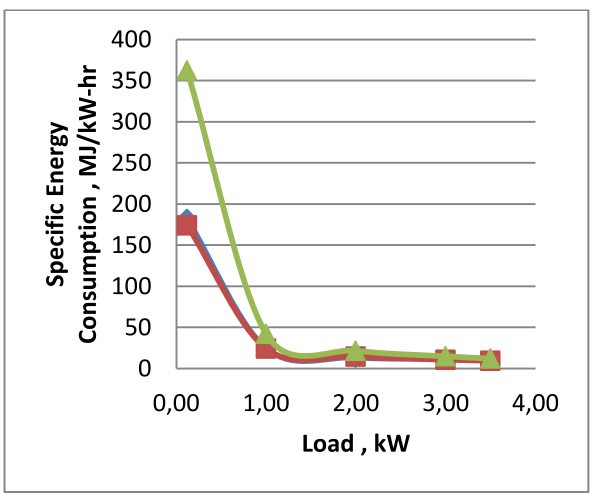


Figure 5.28c B40

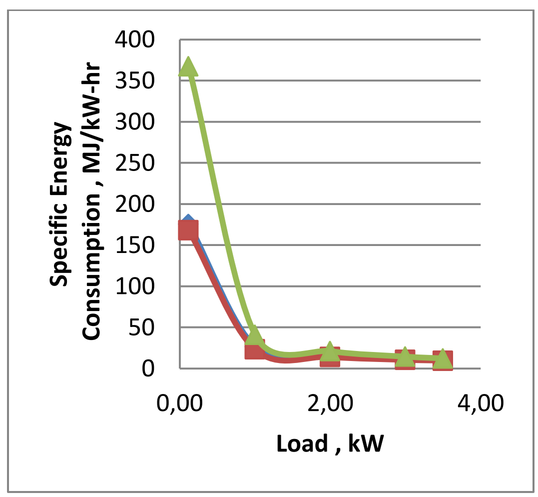


Figure 5.28d B60

FR 0 FR 10 FR 20

FIGURE 5.28 SEC Comparison

5.3.7 EMISSIONS

The engine emissions are plotted against the load in the following graphs. The emissions for various fuels in both pure fuel and dual fuel mode are represented. Also, the effect of the compression ratio can be studied from the following graphs.

a) CO EMISSIONS

It is observed that CO Emissions for the compression ratio of 17.5 is least for diesel and it further decreases as the load is increased to the rated power. At CR of 17.5 the CO emissions is maximum for B20 and when the engine is run in dual fuel mode the emissions increases, though the increase is very slight.

This increase can be accounted to the insufficient oxygen for the complete combustion of the fuel, as CNG has replaced some of the air entering the cylinder. The flow rate of CNG is constant irrespective of the load in our experimental set up. But in actual practice it is varied according to the load, resulting in complete combustion of CNG and thus reduction in CO Emissions.

The combustion of CNG being slow (less flame velocity) full combustion is not taking place. The adiabatic flame temperature of the CNG is low so the peak temperature of the chamber is also low. So according to the equation

$$CO + OH \longrightarrow CO_2 + H \quad [7] \ldots\ldots\text{eq } 1$$

The rate constant K of this equation is directly proportional to exp (T/1102).

Hence if the peak temperature achieved is less, CO is not completely converted into CO_2.

At lower compression ratio of 15 and 13, the CO emissions for the diesel fuel are higher than that of for CR of 17.5.

This can be explained from the fact that as CR decreases the delay period increases, therefore more fuel is present when combustion starts. Also the minimum auto ignition temperature decreases due to decreased density of the compressed air, resulting in less closer contact of the molecules with air. Thus insufficient combustion takes place [42] and CO increases.

Also proper conversion of CO to CO_2 is not taking place (eq 1) as peak temperature is less as explained above.

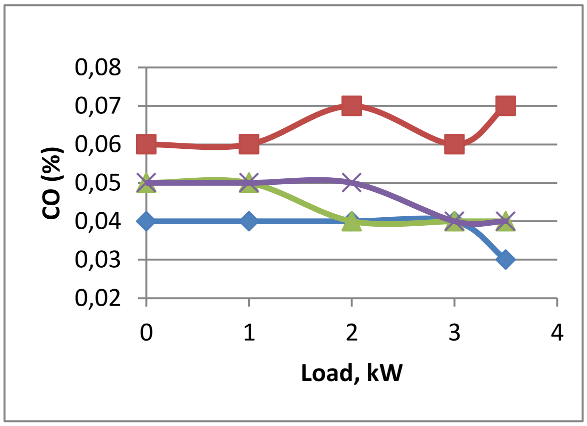


Figure 5.29a Pure Fuel, CR 17.5

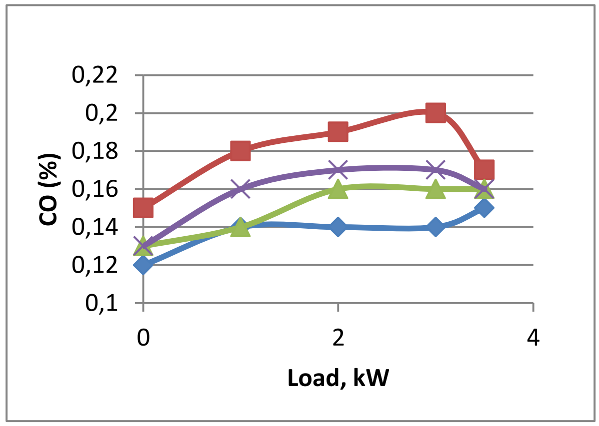


Figure 5.29b Dual Fuel, CR 17.5

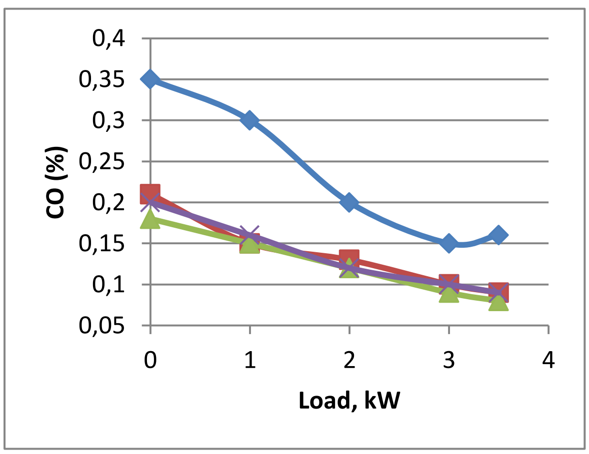


Figure 5.29c Pure Fuel, CR 15

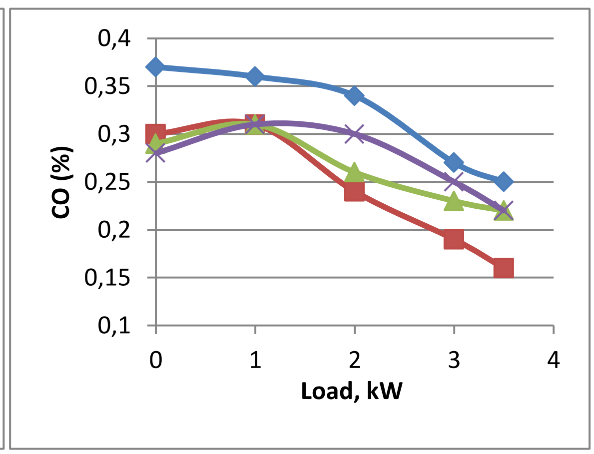


Figure 5.29d Dual Fuel, CR 15

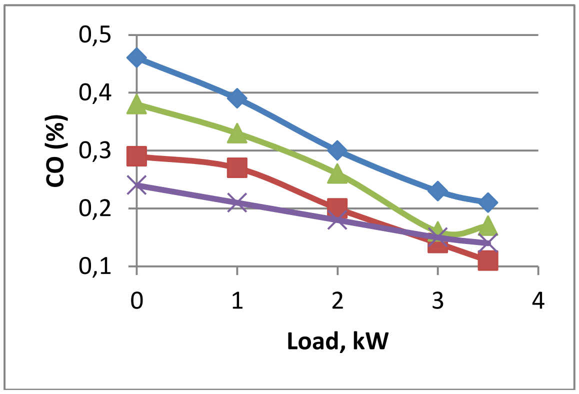


Figure 5.29e Pure Fuel, CR 13

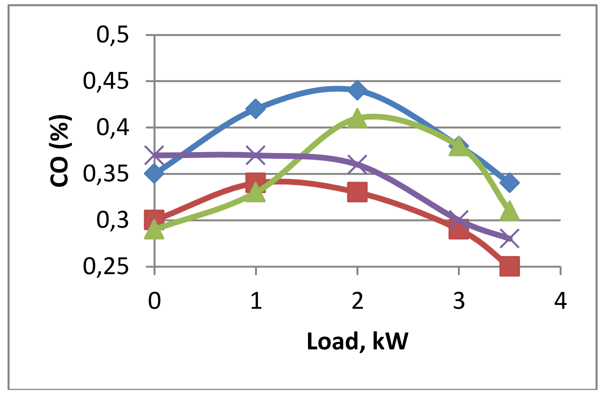


Figure 5.29f Dual Fuel, CR 13

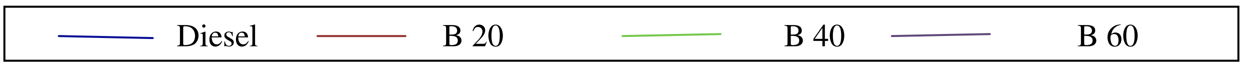


Figure 5.29 CO Emissions Comparisons

b) HC EMISSIONS

It is observed that the HC Emissions is increased when the CR of the engine is decreased and this trend is observed for both pure fuel and dual fuel mode.

This can be explained from the fact that as CR decreases the delay period increases, therefore more fuel is present when combustion starts. Also the minimum auto ignition temperature decreases due to decreased density of the compressed air, resulting in less closer contact of the molecules with air. Thus insufficient combustion takes place [Mathur & Sharma] and HC increases.

When the engine is run at dual fuel mode the HC emissions increases about 12 times. Also, when the flow rate of CNG is increased, we observe that the HC emissions increase. This is due to the fact that with CNG combustion process is slowed down. Hence part of the Fuel is left unburnt. Another factor may be the insufficient air present in the chamber.

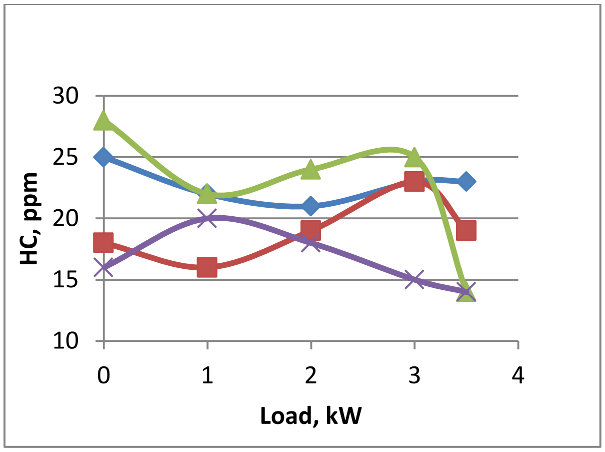


Figure 5.30a Pure Fuel, CR 17.5

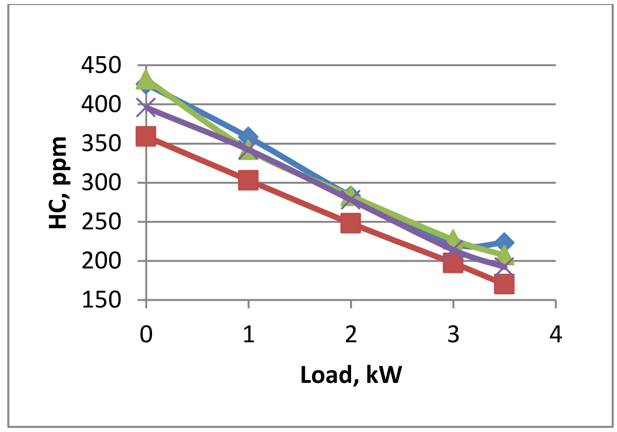


Figure 5.30b Dual Fuel, CR 17.5

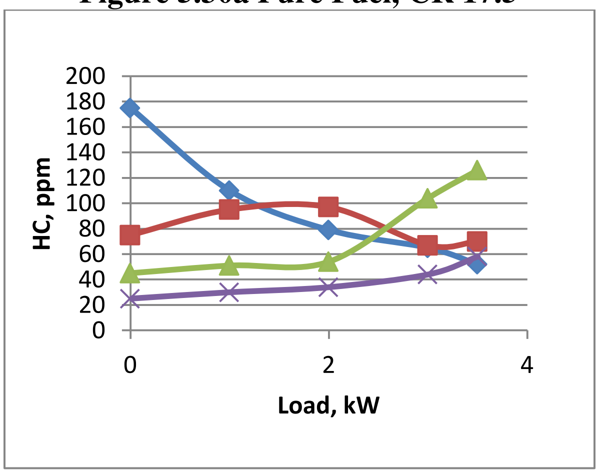


Figure 5.30c Pure Fuel, CR 15

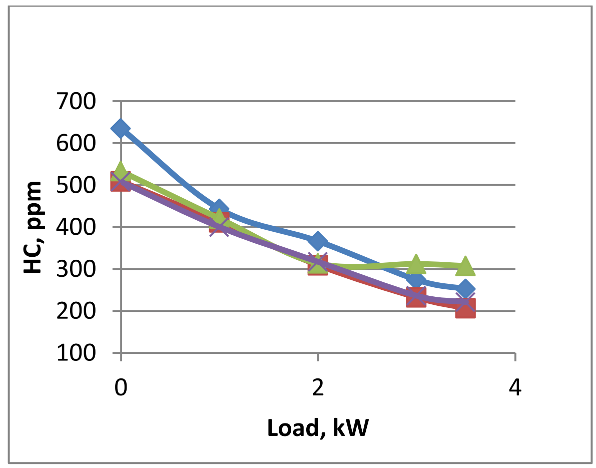


Figure 5.30d Dual Fuel, CR 15

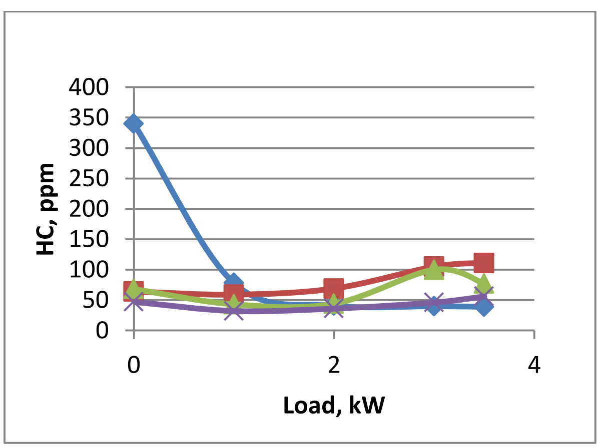


Figure 5.30e Pure Fuel, CR 13

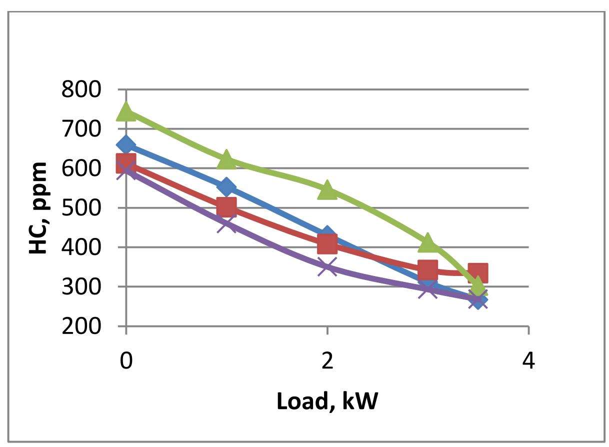


Figure 5.30f Dual Fuel, CR 13

Diesel B 20 B 40 B 60

Figure 5.30 HC Emissions Comparisons

c) CO_2 EMISSIONS

In our observation CO_2 follows an inverse pattern of CO. When CO increases, CO_2 decreases. This is due to improper conversion of CO to CO_2 due to decrease in peak temperature (less adiabatic flame temperature). Hence, the readings are in agreement with the theoretical analysis.

Also as the flow rate is increased, the CO_2 further decreases which is also due to further decrease in the peak temperature and is in agreement to the above explanation.

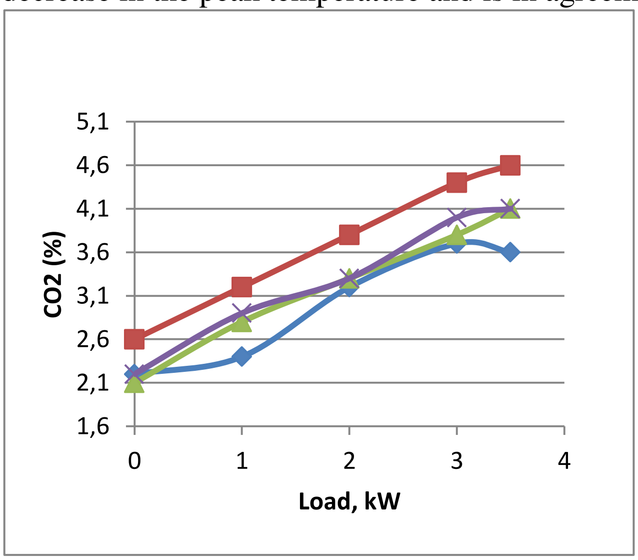


Figure 5.31a Pure Fuel, CR 17.5

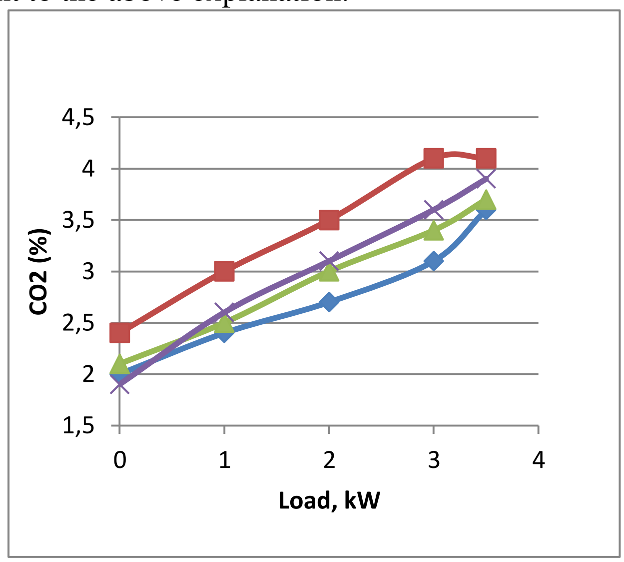


Figure 5.31b Dual Fuel, CR 17.5

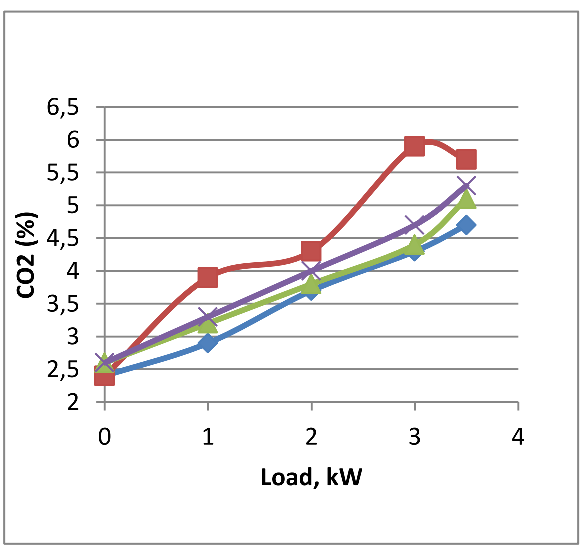


Figure 5.31c Pure Fuel, CR 15

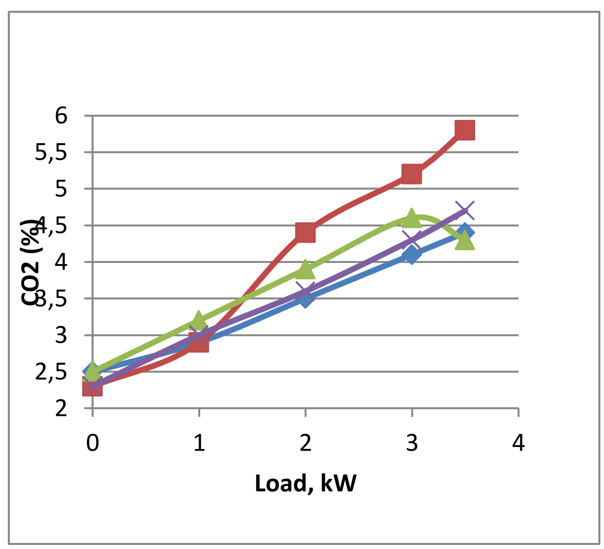


Figure 5.31d Dual Fuel, CR 15

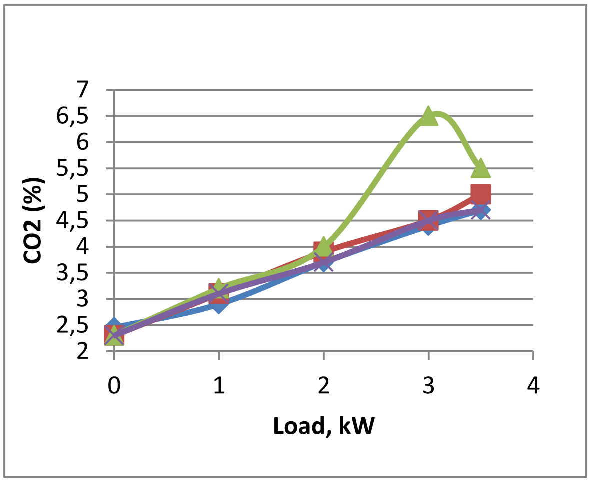


Figure 5.31e Pure Fuel, CR 13

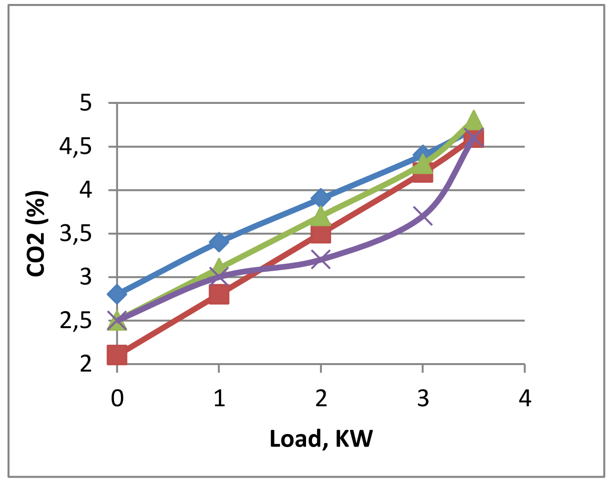


Figure 5.31f Dual Fuel, CR 13

Diesel B 20 B 40 B 60

Figure 5.31 CO_2 Emissions Comparisons

d) NOx EMISSIONS

We observe that the NOx emission is minimum for diesel and maximum for B 20. There is an observable drop in the emissions with the reduction of the compression ratio. The NOx emission is lower when the engine is run in the dual fuel mode. Also when the CNG flow rate is increased the NOx emissions reduce to almost one-fifth. With the increase in load, the NOx emissions increases and thus is maximum for rated load.

High peak temperature and availability of oxygen are two main reasons for the formation of $NO_{x.}$ NO_x is directly related to the adiabatic flame temperature. So as CNG is introduced NO_x decreases as expected. When CR is decreased, NO_x further reduces due to further drop in temperature and pressure. This can be verified from the fact that as flow rate of CNG increases NO_x further reduces.

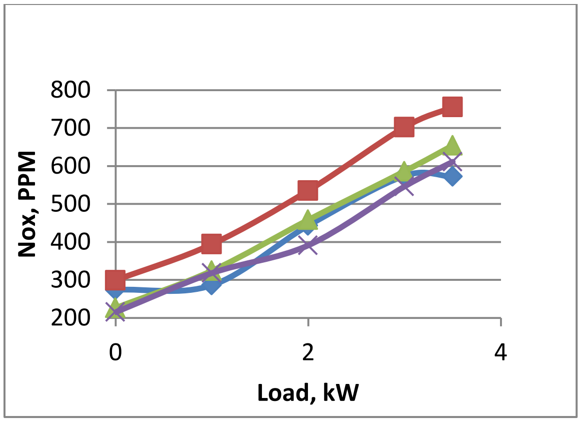


Figure 5.32a Pure Fuel, CR 17.5

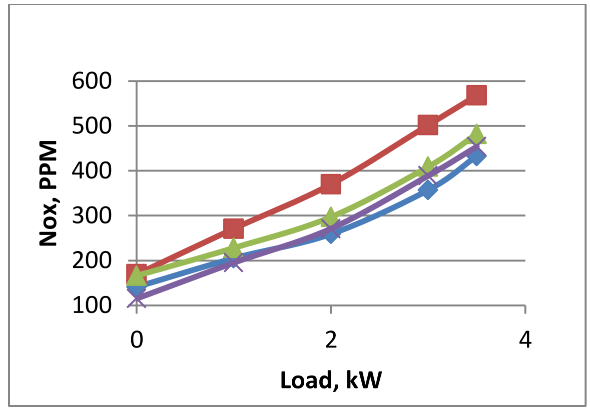


Figure 5.32b Dual Fuel, CR 17.5

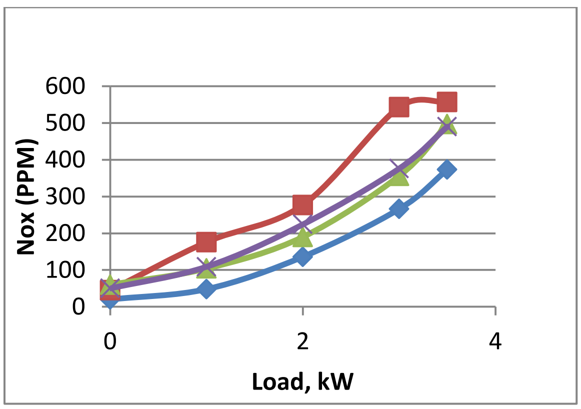


Figure 5.32c Pure Fuel, CR 15

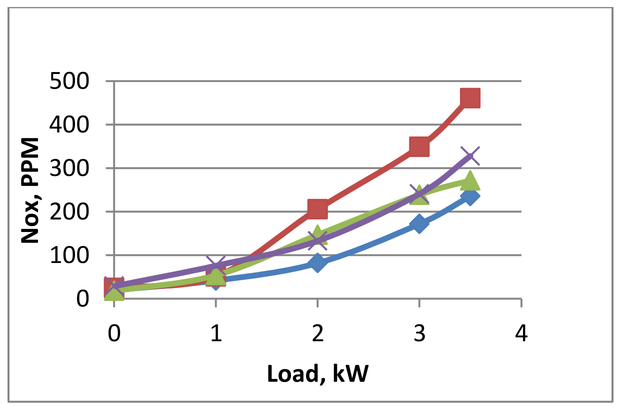


Figure 5.32d Dual Fuel, CR 15

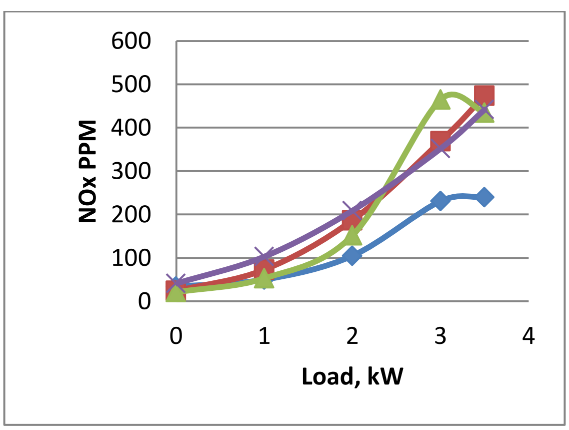


Figure 5.32e Pure Fuel, CR 13

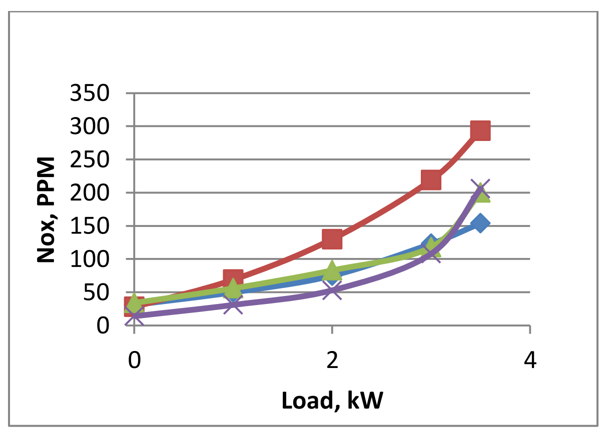


Figure 5.32f Dual Fuel, CR 13

Diesel B 20 B 40 B 60

Figure 5.32 NOx Emissions Comparisons

f) **OPACITY**

We observe that as the compression ratio is decreased the opacity of the exhaust gases increases and is as high as 50% for the compression of 15 and 65% for CR of 13. This is due to the improper combustion, more fuel is to be injected and hence amount of UBHC, PAH (polynuclear Aromatic Hydrocarbons) etc. increases therefore opacity increases.

In dual fuel engine, the amount of diesel consumed is reduced due to its replacement with combustion of CNG – air mixture. A flame front is formed by the ignition of diesel which sweeps the homogeneous mixture of CNG and air. Now at the attainment of SIT of CNG, it burns smoothly and exhaust contains less unburnt fuel. Hence smoke is reduced with the use of CNG. The soot particles, form primarily from the carbon in the diesel fuel. Since in CNG the hydrogen/carbon ratio is high because of presence of smaller hydrocarbon as compared to diesel, soot formation is less. Consequently the Particulate matter emission will also decrease with the use of CNG, and hence reduction in smoke. Also with the increase in load the smoke increases because big droplets of diesel are injected into the combustion chamber whose nucleus remains unburnt (as the fuel auto ignites in case of diesel engine) and hence increase in smoke is observed, whereas in dual engine the fuel is burnt by flame propagation hence better combustion and reduction in smoke.

Soot particles are formed primarily due to diesel fuel, as the amount of diesel injected increases at higher loads, soot particles and hence opacity increases.

Also the amount of oxygenated Hydro carbons, PAH (polynuclear Aromatic Hydrocarbons) decreases which contribute in the formation of PM

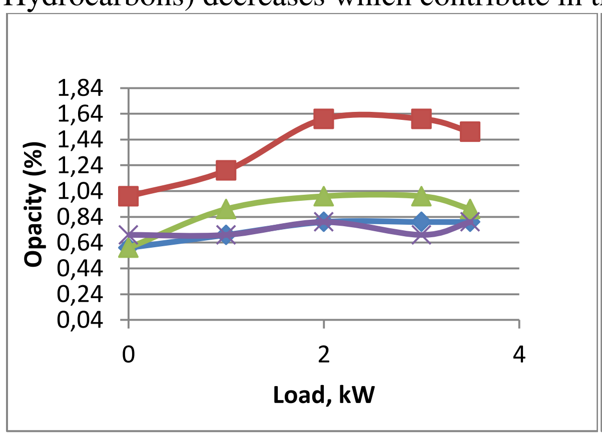


Figure 5.33a Pure Fuel, CR 17.5

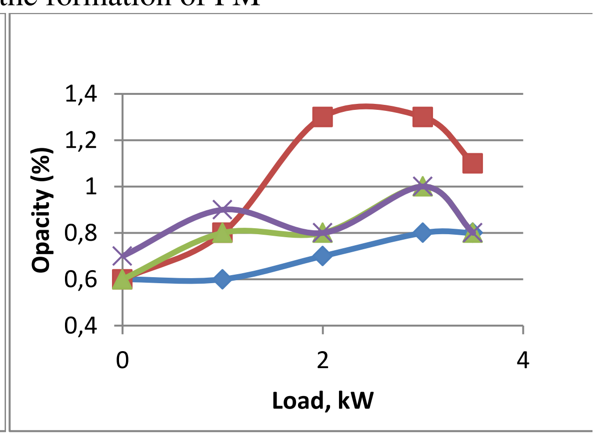


Figure 5.33b Dual Fuel, CR 17.5

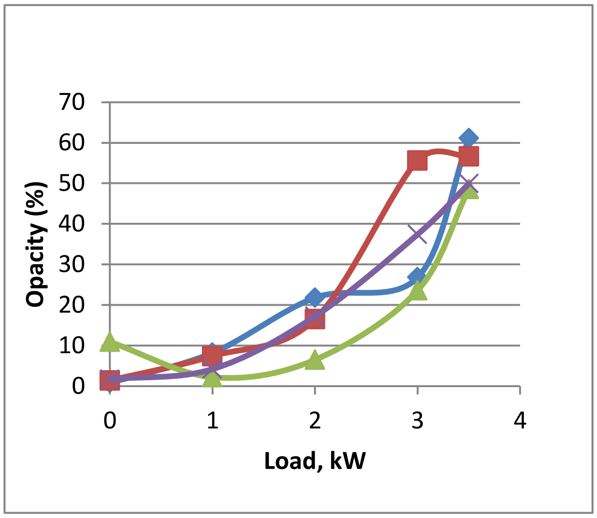


Figure 5.33c Pure Fuel, CR 15

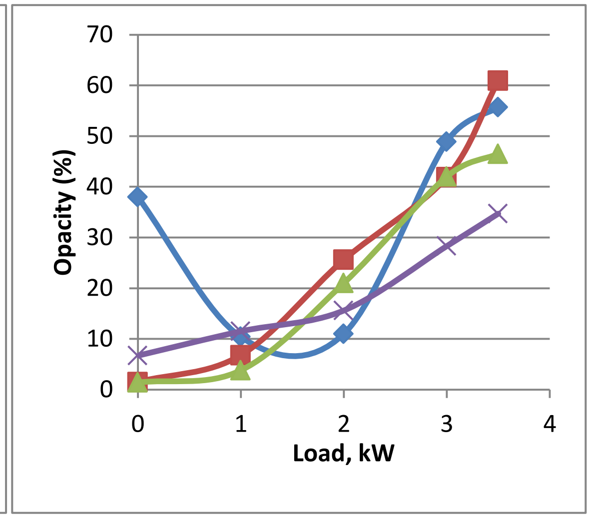


Figure 5.33d Dual Fuel, CR 15

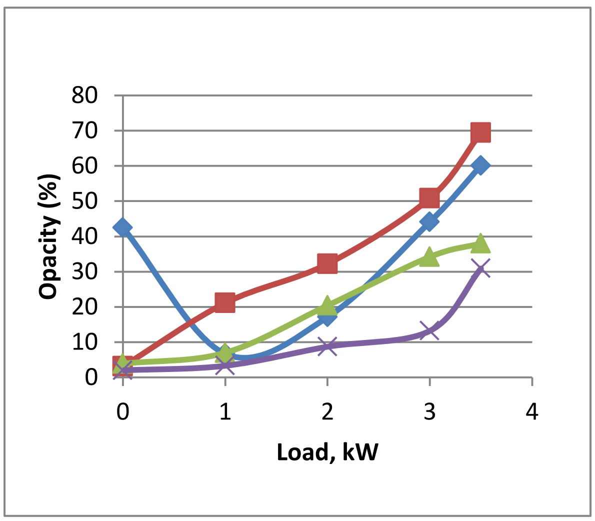


Figure 5.33e Pure Fuel, CR 13

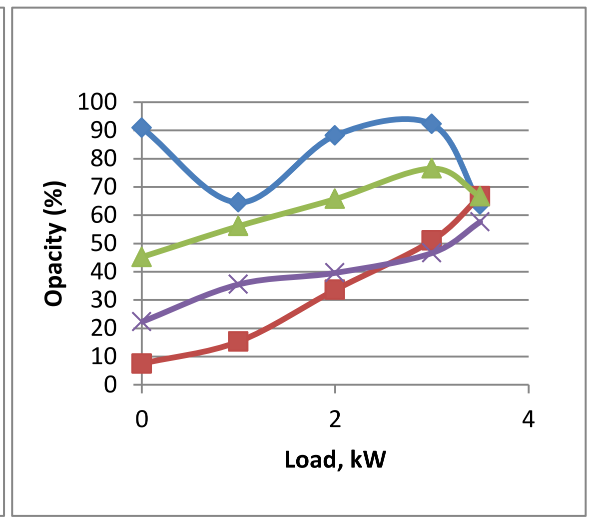


Figure 5.33f Dual Fuel, CR 13

Diesel B 20 B 40 B 60

Figure 5.33 OPACITY Comparisons

5.3.7 EMISSIONS COMPARISON BY VARYING CNG FLOW RATE

It is observed that with the change in the flow rate of the CNG in the dual fuel operation, the emissions vary. The followings graphs depict these variations.

a) **CO Comparison**

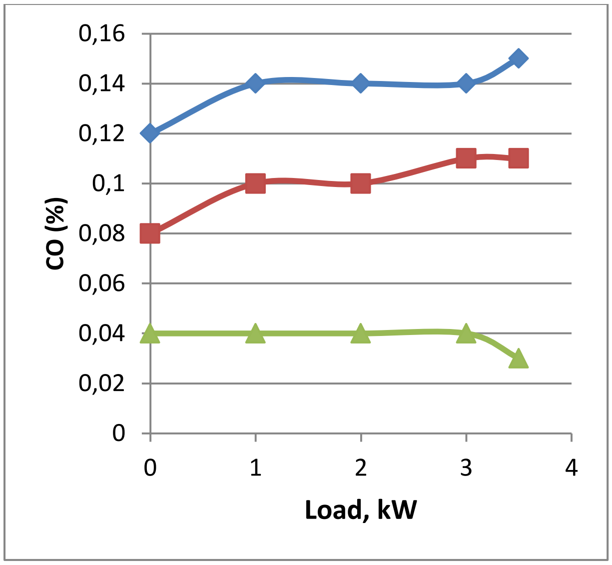


Figure 5.34a Diesel

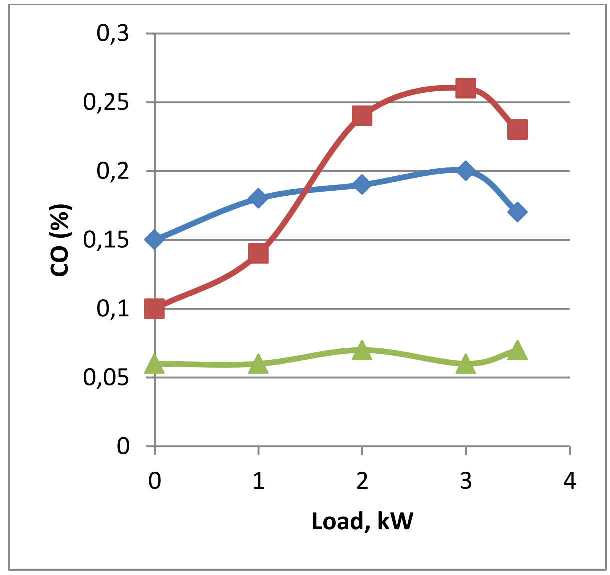


Figure 5.34b B20

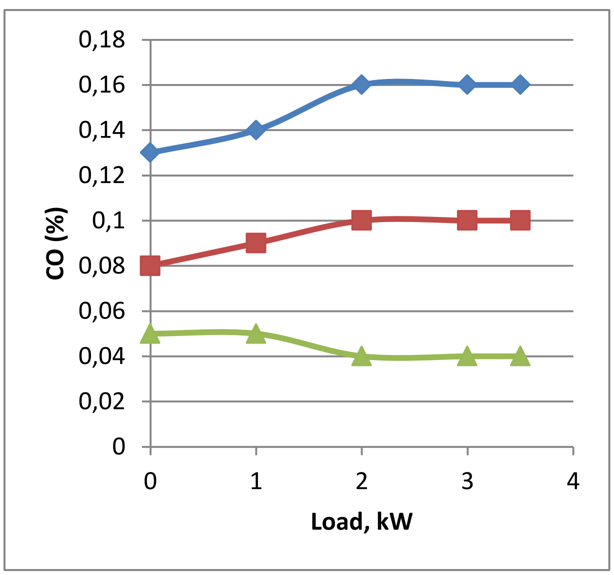


Figure 5.34c B40

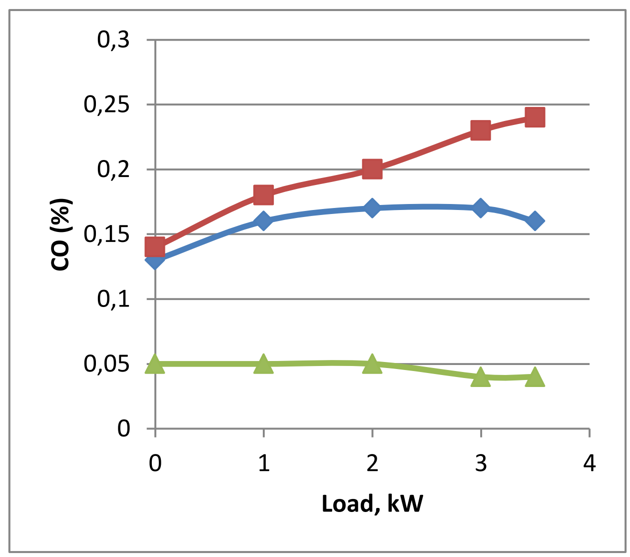


Figure 5.34d B60

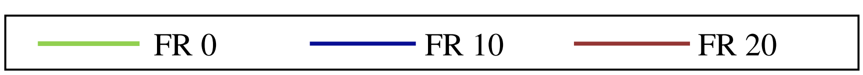


FIGURE 5.34 CO Emissions Comparison (Dual Fuel)

b) HC Comparison

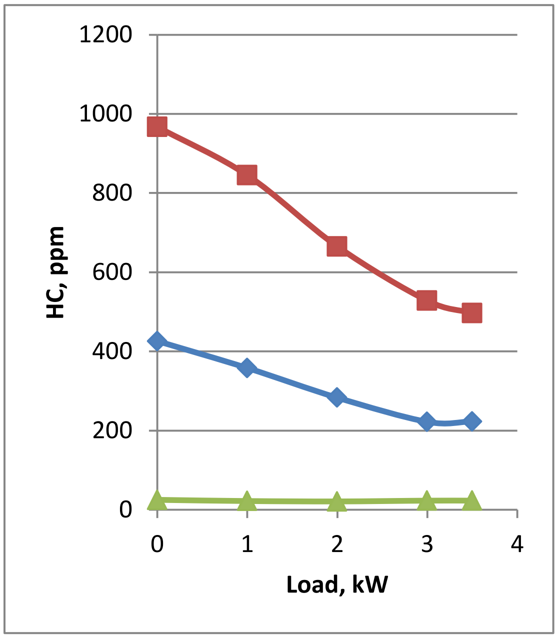


Figure 5.35a Diesel

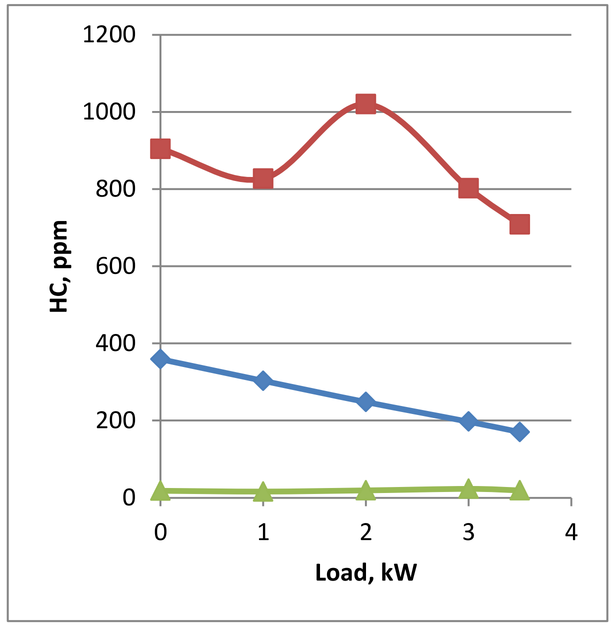


Figure 5.35b B20

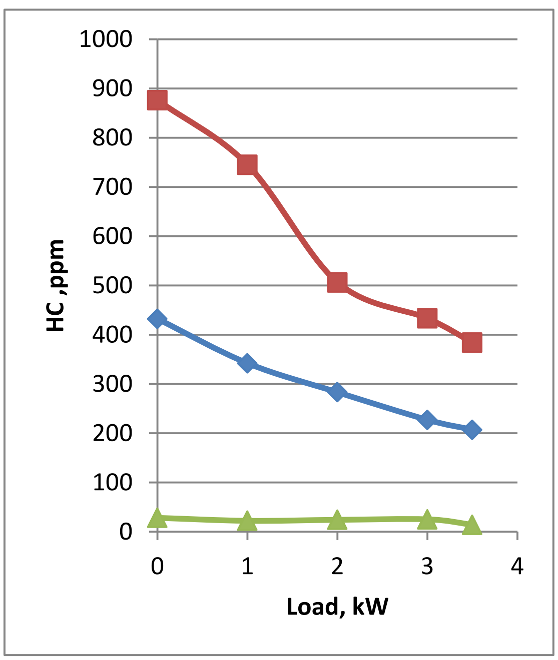


Figure 5.35c B40

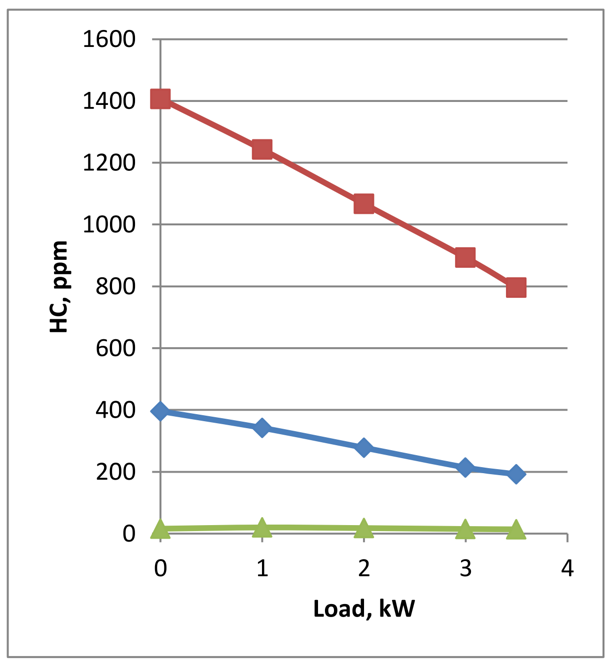


Figure 5.35d B60

FIGURE 5.35 HC Emissions Comparison (Dual Fuel)

c) CO_2 Comparison

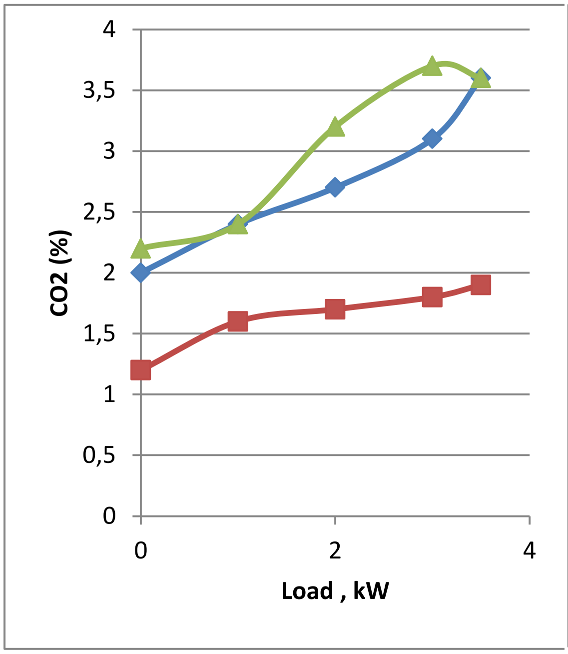


Figure 5.36a Diesel

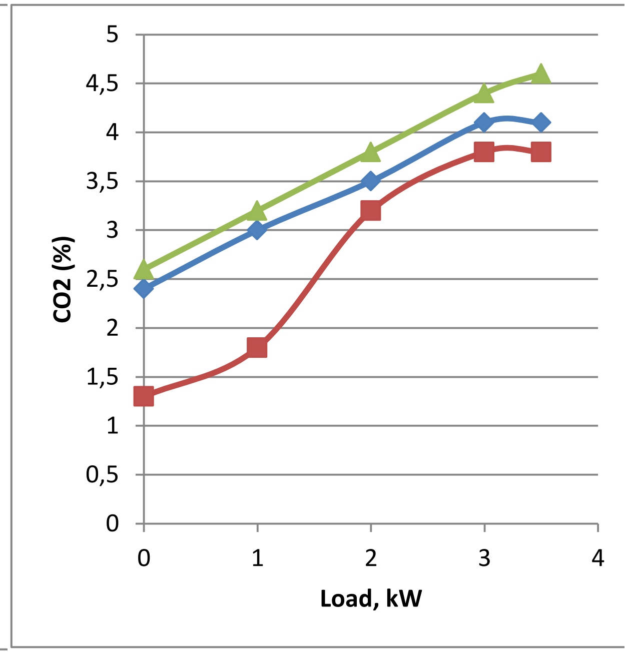


Figure 5.36b B20

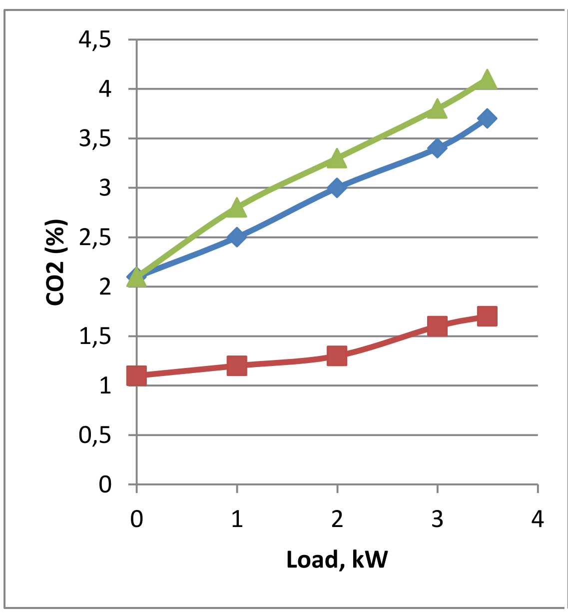


Figure 5.36c B40

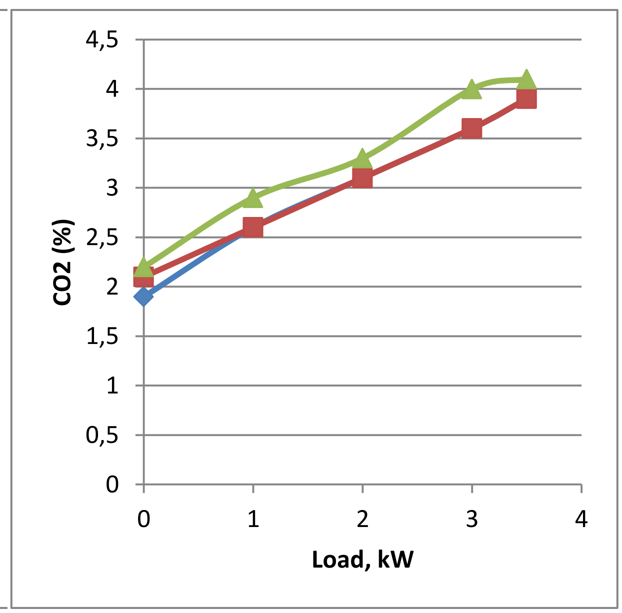


Figure 5.36d B60

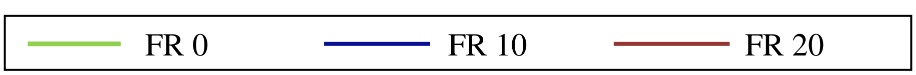


FIGURE 5.36 CO_2 Emissions Comparison (Dual Fuel)

d) NOx Comparison

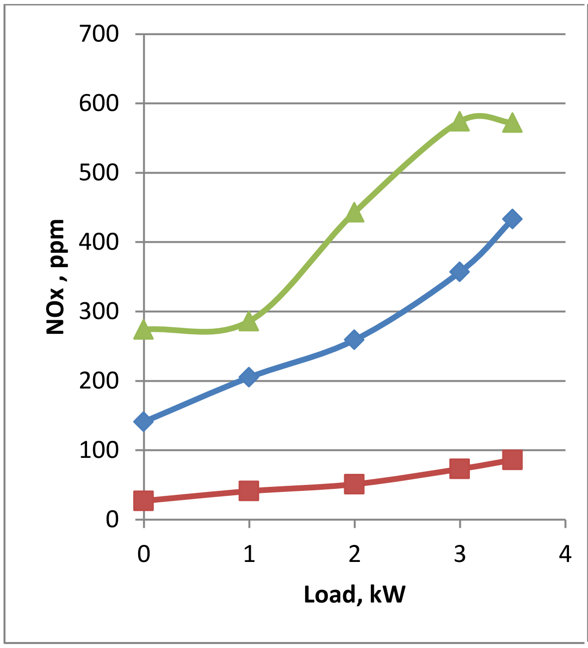


Figure 5.37a Diesel

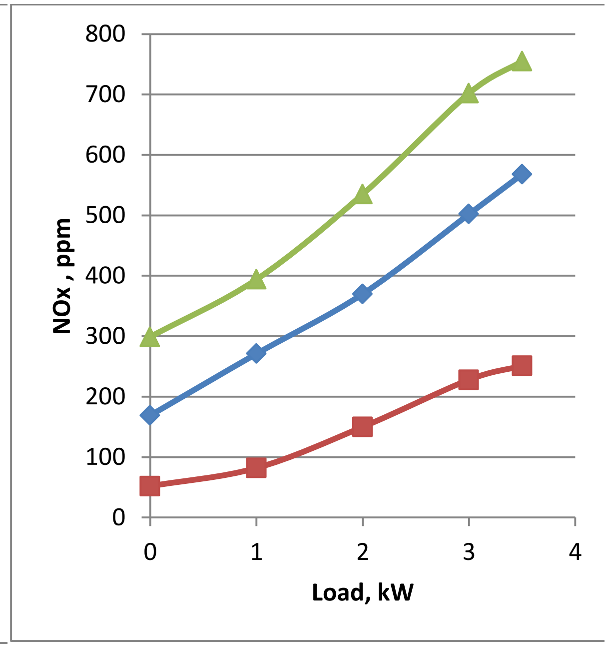


Figure 5.37b B20

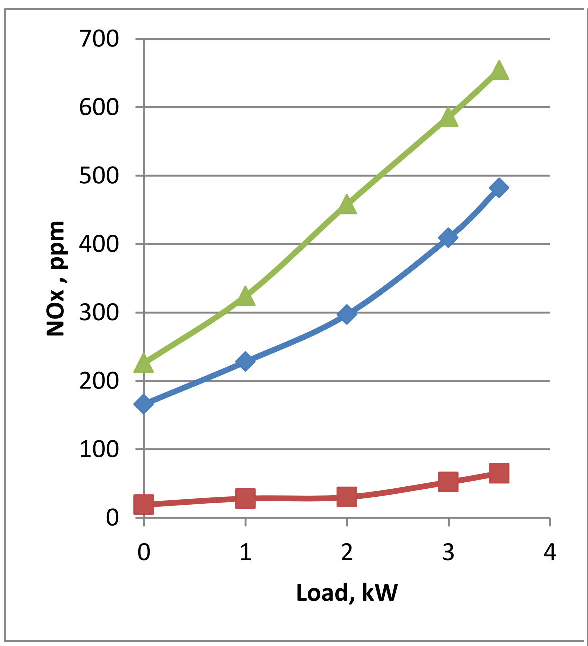


Figure 5.37c B40

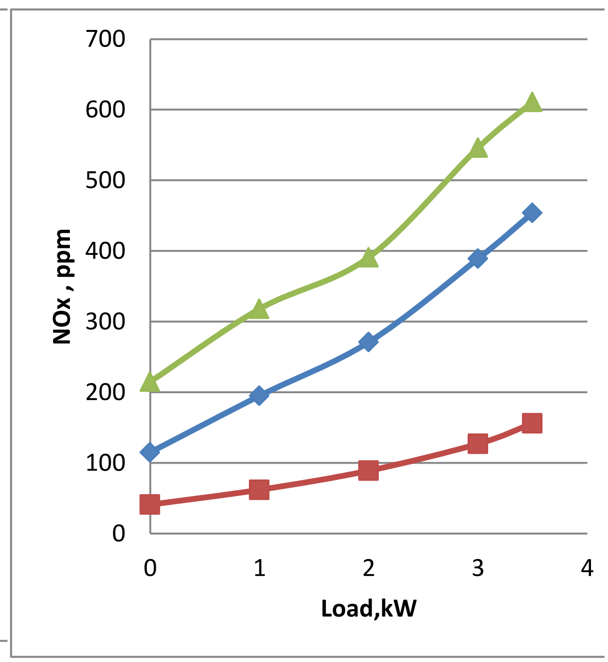


Figure 5.37d B60

FR 0 FR 10 FR 20

FIGURE 5.37 NOx Emissions Comparison (Dual fuel)

e) OPACITY

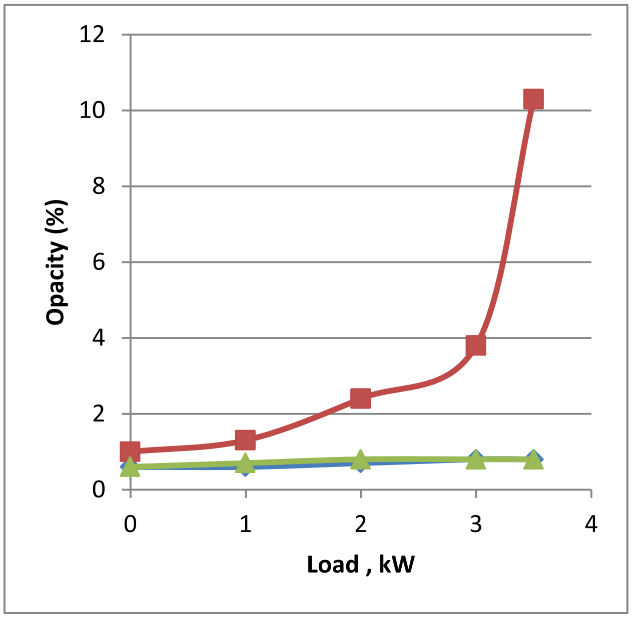


Figure 5.38a Diesel

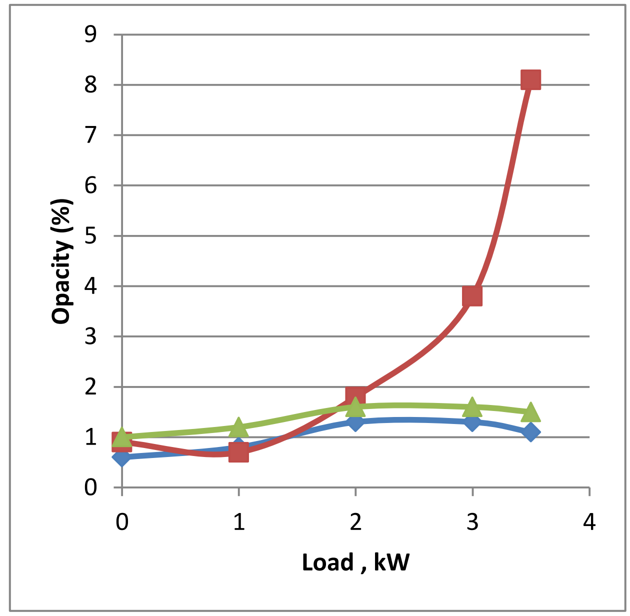


Figure 5.38b B20

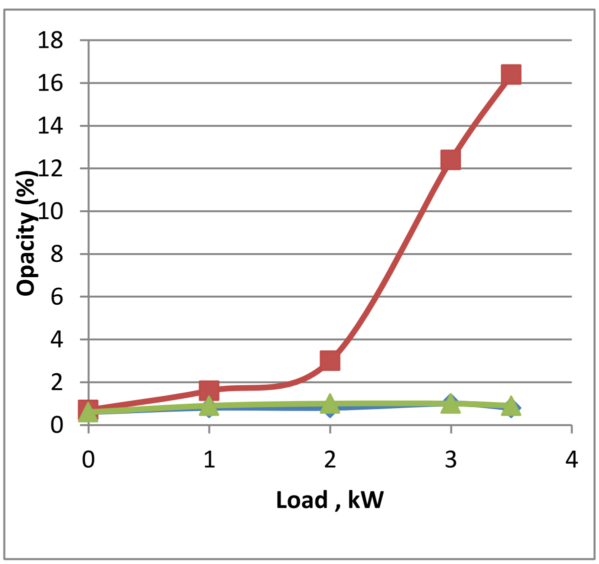


Figure 5.38c B40

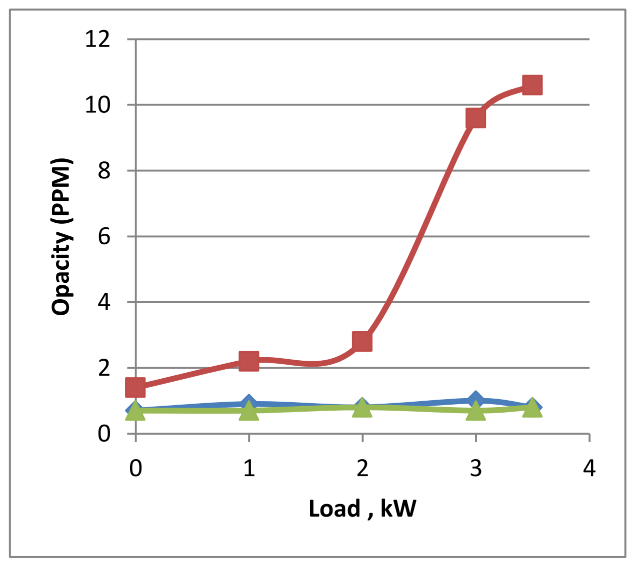


Figure 5.38d B60

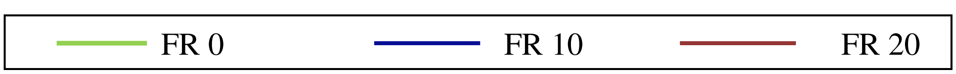


FIGURE 5.38 OPACITY Comparisons

5.3.8 Emissions Comparison for Diesel, B20, B40, B60, Diesel+CNG, B20+CNG, B40+CNG and B60+CNG

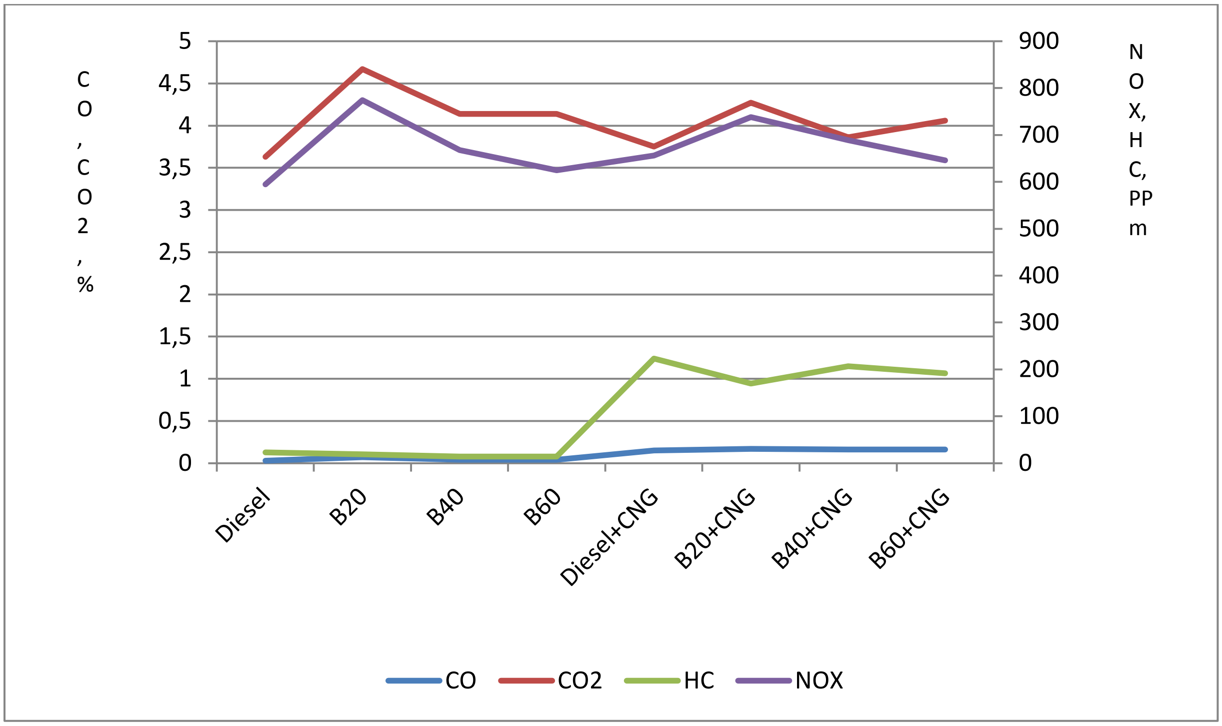


Figure 5.39a Comparison of Emissions at Compression Ratio =17.5 at Rated Load

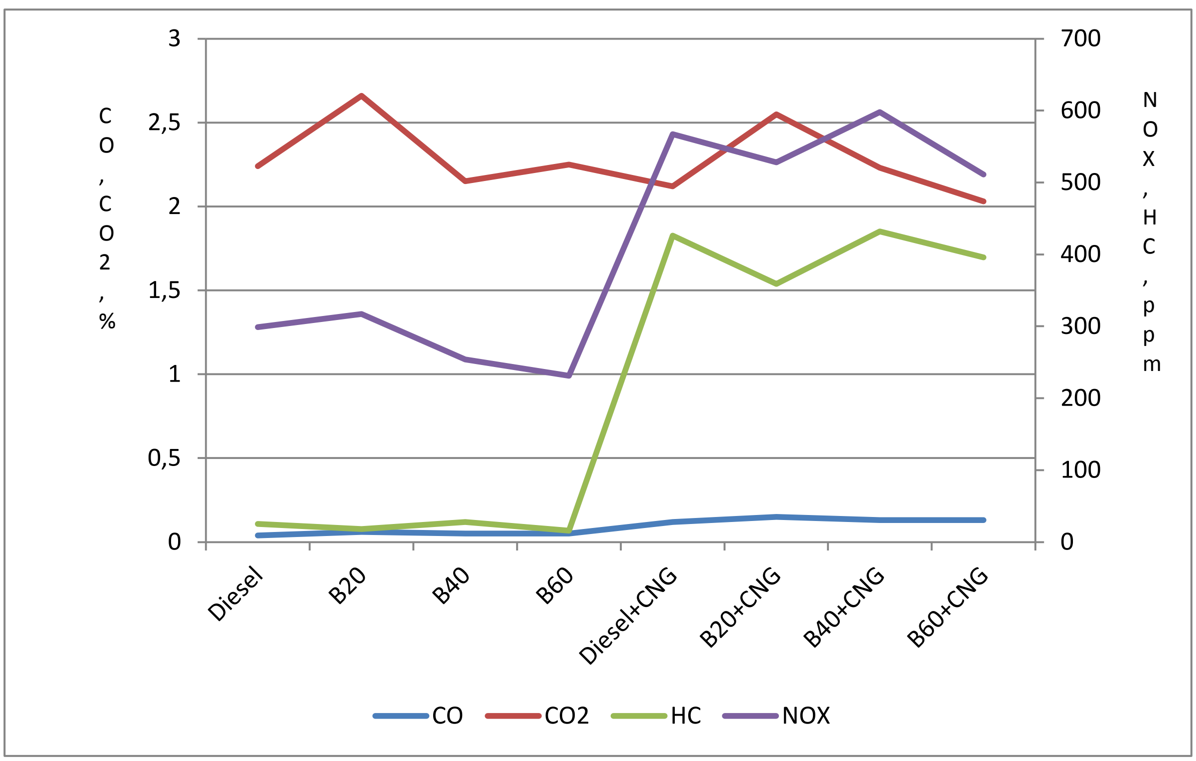


Figure 5.39b Comparison of Emissions at Compression Ratio =17.5 at Idle

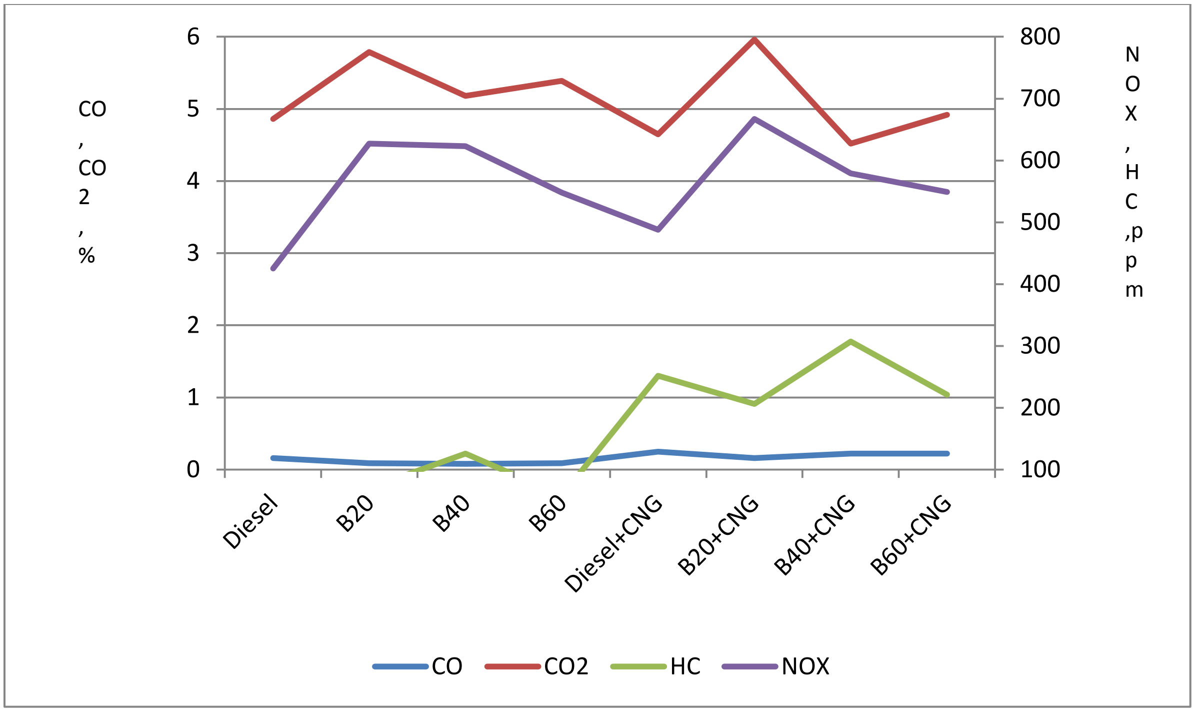


Figure 5.39c Comparison of Emissions at Compression Ratio =15 at Rated Load

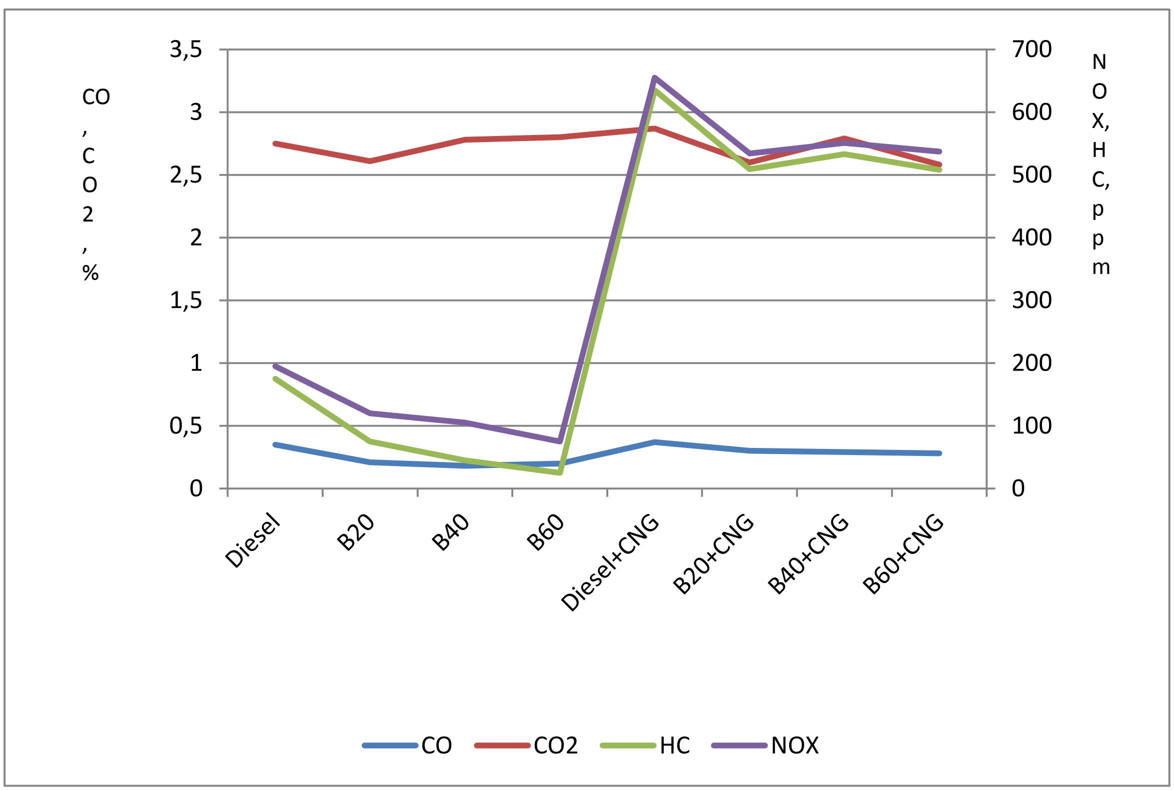


Figure 5.39d Comparison of Emissions at Compression Ratio =15 at Idle

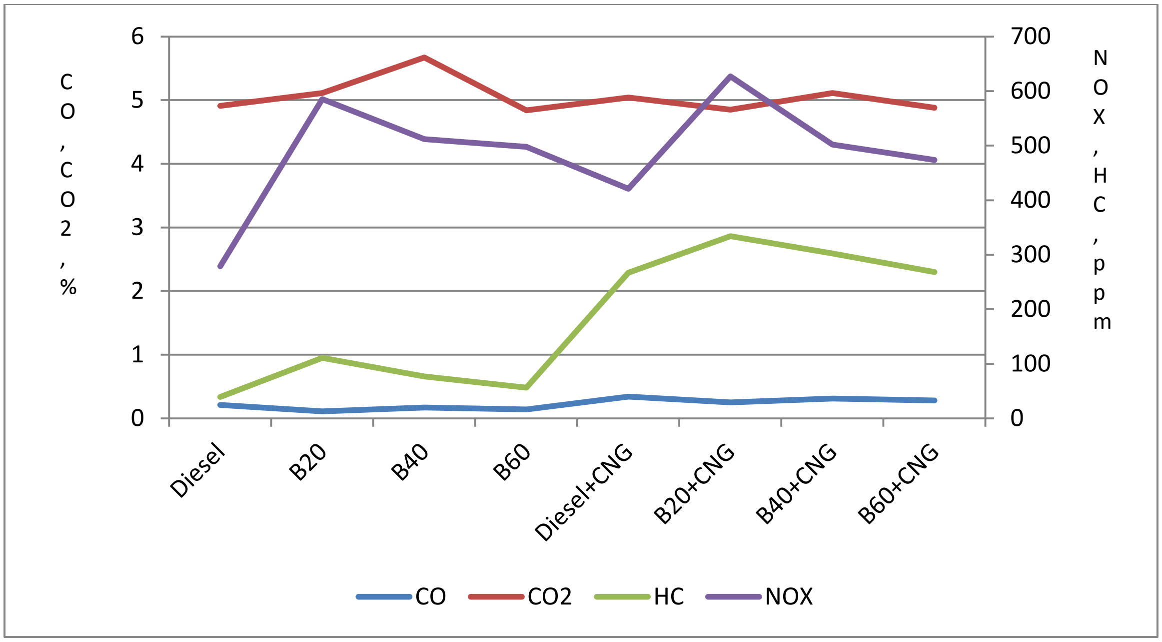


Figure 5.39e Comparison of Emissions at Compression Ratio=13 and Rated Load

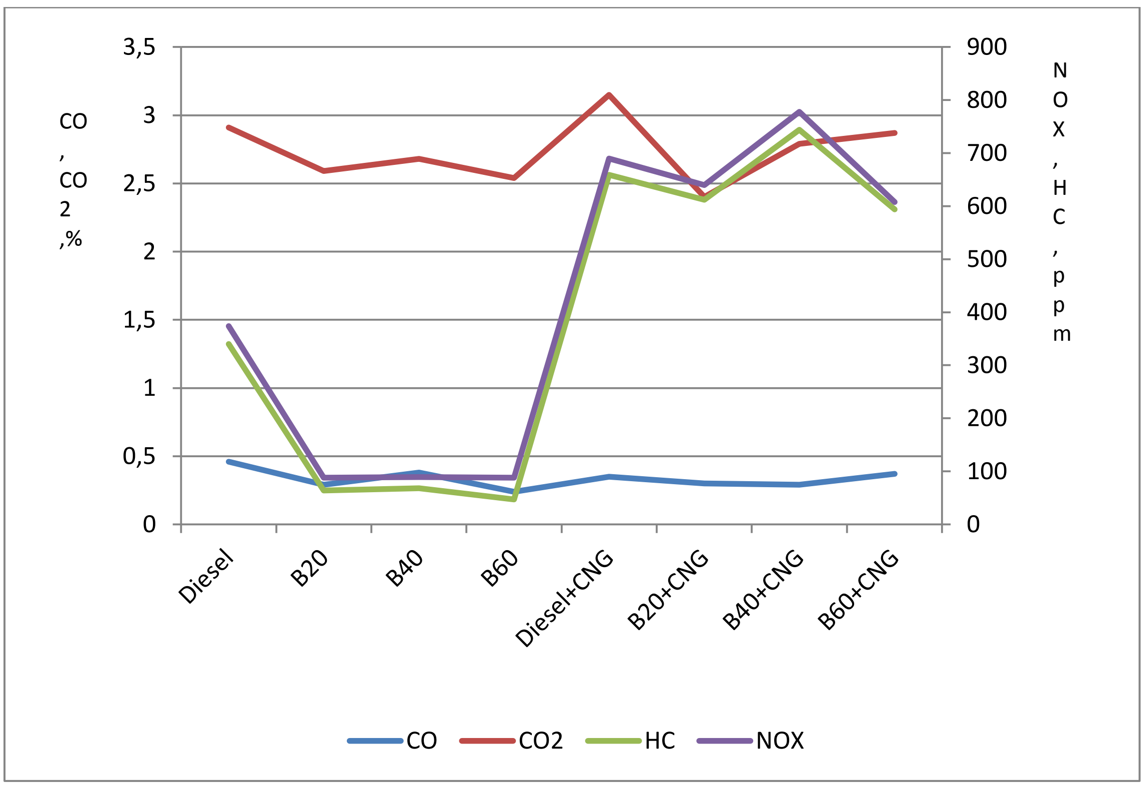


Figure 5.39f Comparison of Emissions at Compression Ratio =13 at Idle

5.3.8 EFFECT OF COMPRESSION RATIO

The following graphs show the variation of the engine emissions with the compression ratio. The graphs are for the rated load of 3.5 kW.

a) **CO Emissions**

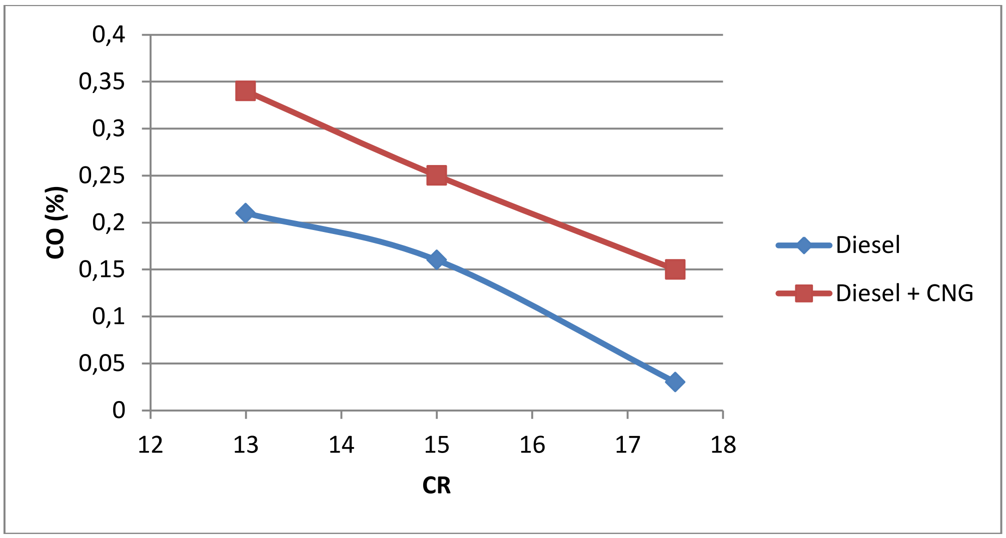


Figure 5.40 CO vs. CR

b) HC Emissions

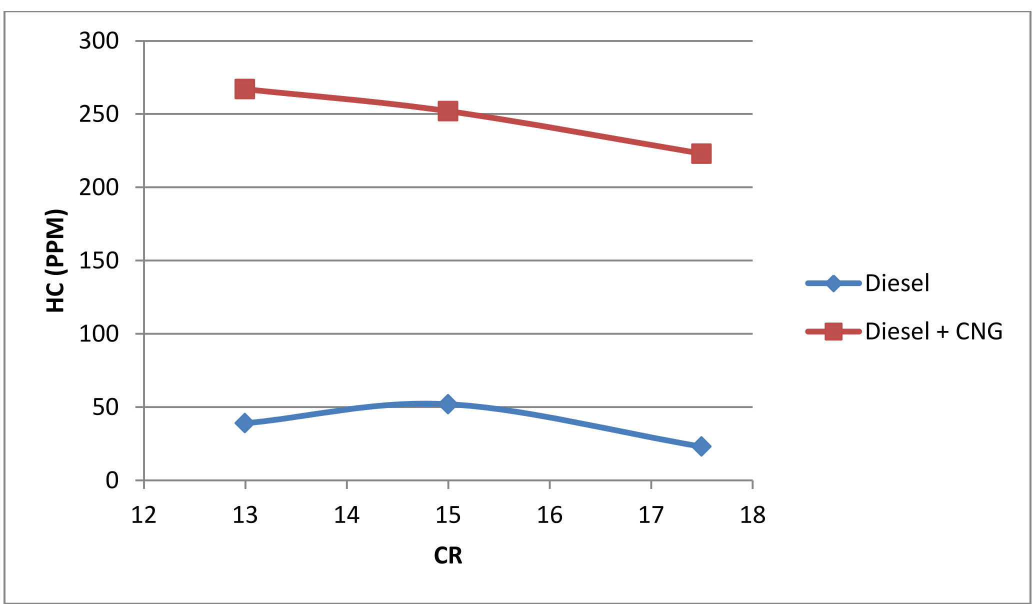


Figure 5.40 HC vs. CR

c) CO_2 Emissions

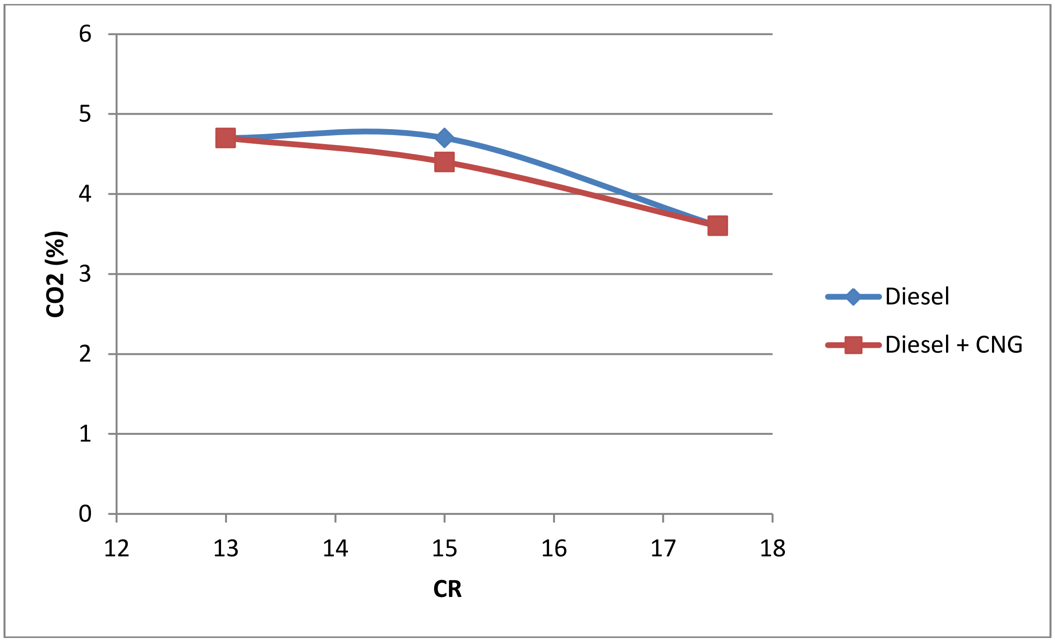


Figure 5.41 CO_2 vs. CR

d) NO_X Emissions

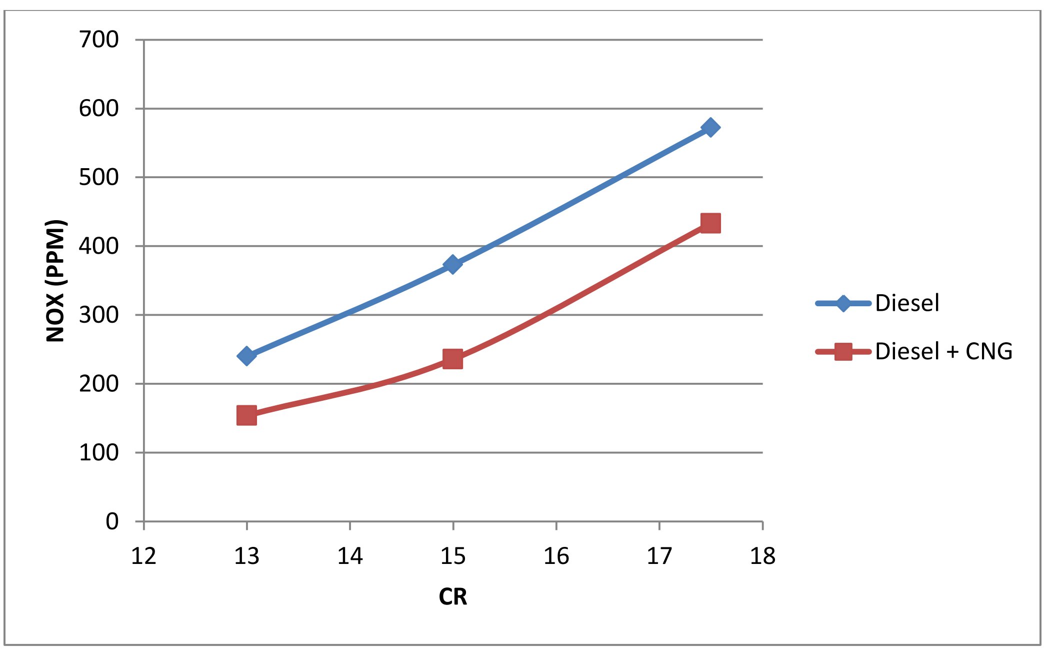


Figure 5.42 NOX vs. CR

e) Opacity

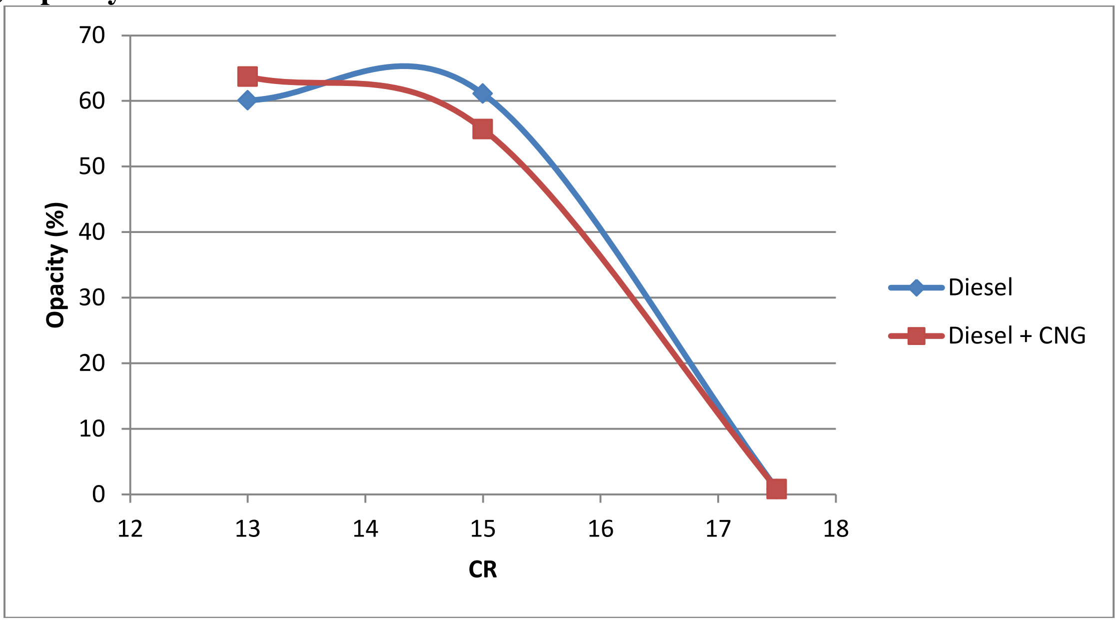


Figure 5.43 Opacity vs. CR

5.4 BREAK EVEN ANALYSIS

The Brake Even Analysis is done to determine the point (Hours of engine operation) when the operation in dual fuel break evens with that of pure fuel. For this, we have considered the case of operation of the engine for the compression ratio of 17.5 at the rated load of 3.5 KW.

a) **Case 1 (When engine runs only on Diesel)**

Assuming that engine runs most of the time at rated load (3.5 KW in this case)

Diesel cost = 32 Rs /Ltr

Specific fuel consumption = 0.2675 Kg/KW-hr

Compression ratio = 17.5

Now,

Diesel consumed per hour = 0.2675x3.5 Kg/hr

= 0.93625 Kg/hr

Therefore total cost for running the engine at diesel per hour (at rated load)

= 0.93625x32 Rs/hr

= 29.96 Rs/hr

b) **Case 2 (When engine runs in Dual Mode)**

Specific fuel consumption (diesel) = 0.105542 Kg/KW-hr

Compression ratio = 17.5

Diesel consumed per hour = 0.105542x3.5 Kg/hr

= 0.369397 Kg/hr

Now calculating the consumption of CNG for this case (10 LPM)

Pressure of CNG in the cylinder = 220bar

Cylinder volume = 60 liters

Pressure at the inlet of venturi = 0.8bar

According to Boyle's Law,

$$P_1V_1=P_2V_2$$

$$60x220=V_2x0.8$$

$$V_2 = 16500 \text{ liters}$$

Mass of CNG in the cylinder = 12kg

Therefore 12kg of CNG is equal to 16500 liters

Hence, 10 LPM of CNG is equal to 7.272×10^{-3} Kg/min

Or, 10 LPM of CNG is equal to $7.27 \times 10^{-3} \times 60 = 0.43636$ Kg/hr

Also, cost of CNG is 19.2 Rs/kg

Hence, total cost for running engine in dual mode per hour is

$$0.36397 \times 32 + 0.43636 \times 19.20 \text{ Rs/hr} = 11.82 + 8.378 = 20.198 \text{ Rs/hr}$$

Taking the initial cost for installing the CNG kit = 32000 Rs

Hence, the breakeven point for the system in hours

$$32000 + 20.198 \times t = 29.96 \times t$$

$$\Rightarrow (29.96 - 20.198)t = 32000$$

$$\Rightarrow t = 32000/9.762 = \mathbf{3278\ hours}$$

Thus, the operation in dual fuel would reach the break even point after 3278 hours of operation.

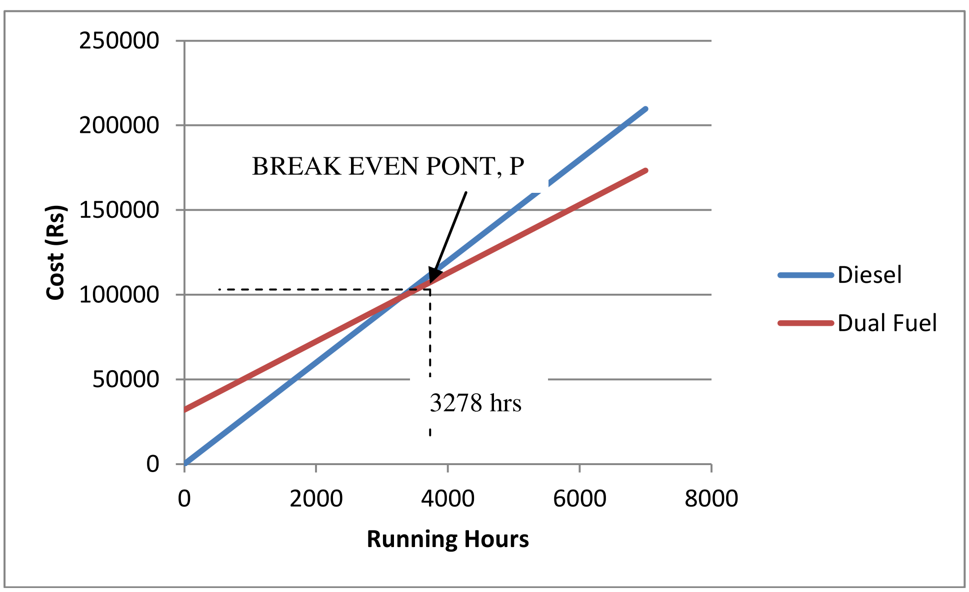


Figure 5.45: Break Even Analysis

5.5 HEAT BALANCE

PURE DIESEL

a) Heat Supplied by Fuel = Fuel Flow x Cal Val = 11235.9x3.5 = 39325.65 KJ

b) Heat Equivalent to Useful Work = BP x 3600 = 3.5x3600 = 12600 KJ

Heat Equivalent to Useful Work in% =

(Heat Equivalent to Useful Work x100)/Heat Supplied By Fuel = **32.04%**

c) Heat in Jacket Cooling Water = F3 x C_{pW} x (T 2 − T1) = 350 x 4.2 x 11.16

= 16405.55 KJ

Heat in Jacket Cooling Water in% =

(Heat in Jacket Cooling Water x100)/Heat Supplied By Fuel = 16405.55/39325.65 x 100 = **41.71 %**

d) Heat in Exhaust (Calculate C_{pex} value)

C_{pex} = (F 4 x C_{pw} x (T 4 − T 3))/((F1+F 2) x (T 5 − T 6)) KJ/Kg^{0}K = (175 x 5.49 x 4.2)/(18.45+0.936)(78.08) = 2.6658 KJ/Kg^{0}K

Where,

C_{pex}	Specific heat of exhaust gas,	2.6658 kJ/kg^{0}K (calculated above)
C_{pw}	Specific heat of water,	4.2 kJ/kg^{0}K
F1	Fuel consumption,	0.936 kg/hr
F2	Air consumption,	18.45 kg/hr
F4	Calorimeter water flow	175 kg/hr
T3	Calorimeter water inlet temperature	20.89 ^{0}K
T4	Calorimeter water outlet temperature	26.38 ^{0}K
T5	Exhaust gas to calorimeter inlet temp.	225.32 ^{0}K
T6	Exhaust gas from calorimeter outlet temp,	147.24 ^{0}K

Heat in Exhaust (KJ / h) = (F1+F 2) x C_{pex} x (T 5 − Tamb)

= 19.386 x 2.6658 x (225.32 – 30) = 9958.58 KJ

Heat in Exhaust % = (Heat in Exhaust x 100)/Heat Supplied By Fuel

= 9958.58/39325.65 x 100 = **25.32 %**

e) Heat to radiation and unaccounted (%) = Heat Supplied by Fuel (100%) − {(Heat Equivalent to Useful Work (%) □+ Heat In Jacket Cooling Water (%) + Heat to Exhaust (%)} = 100 – (32.04+41.71+25.32) = 100 – 99.07 = **0.93 %**

<u>DUAL FUEL MODE</u>

a) Heat Supplied by Fuel = Fuel Flow x Cal Val = 15515.85 + 19636.2

= 35152.05 KJ

b) Heat Equivalent to Useful Work = BP x 3600 = 3.5x3600 = 12600 KJ

Heat Equivalent to Useful Work in% =

(Heat Equivalent to Useful Work x100)/Heat Supplied By Fuel = **35.862**%

c) Heat in Jacket Cooling Water = F4 x C_{pw} x (T 2 − T1)

= 350 x 4.2 x 11.16 x 9.36

= 13759.2 KJ

Heat in Jacket Cooling Water in% =

(Heat in Jacket Cooling Water x100)/Heat Supplied By Fuel = 13759.2/35152.05 x 100 = **40.65%**

d) Heat in Exhaust (Calculate C_{pex} value)

C_{pex} = (F 4 x C_{pw} x (T 4 − T 3))/

((F1+F 2+F3) x (T 5 − T 6)) KJ/Kg^{0}K = (175 x 4.57 x 4.2)/(0.369425 + 0.43636 + 21.29)(75.5) = 2.014 KJ/Kg^{0}K

Where,

C_{pex}	Specific heat of exhaust gas	2.014 kJ/kg^{0}K (calculated above)
C_{pw}	Specific heat of water	4.2 kJ/kg^{0}K
F1	Fuel consumption	0.369425 kg/hr
F2	Air consumption	21.29 kg/hr

F3 CNG consumption 0.43636 kg/hr

F4 Calorimeter water flow 175 kg/hr

T3 Calorimeter water inlet temperature 21.25 ^{0}K

T4 Calorimeter water outlet temperature 25.92 ^{0}K

T5 Exhaust gas to calorimeter inlet temp. 218.90 ^{0}K

T6 Exhaust gas from calorimeter outlet temp. 143.40 ^{0}K

Heat in Exhaust (KJ / h) = (F1+F 2+F3) x C_{pex} x (T 5 − Tamb)

= 22.095 x 2.014 x (218.90 – 30) = 8406.22 KJ

Heat in Exhaust % = (Heat in Exhaust x 100)/Heat Supplied By Fuel

= 8406.22/35152.05 x 100 = **23.913 %**

e) Heat to radiation and unaccounted (%) = Heat Supplied by Fuel (100%) − {(Heat Equivalent to Useful Work (%) □+ Heat In Jacket Cooling Water (%) + Heat to Exhaust (%)} = 100 – (35.862+39.1+23.913) = 100 – 98.875 = **1.125 %**

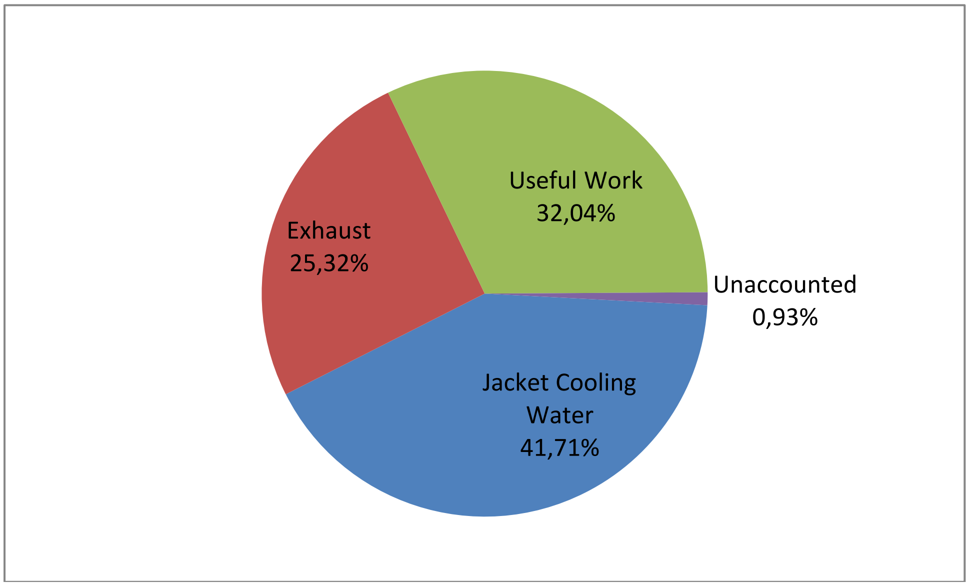


Figure 5.46 : Heat Balance for diesel

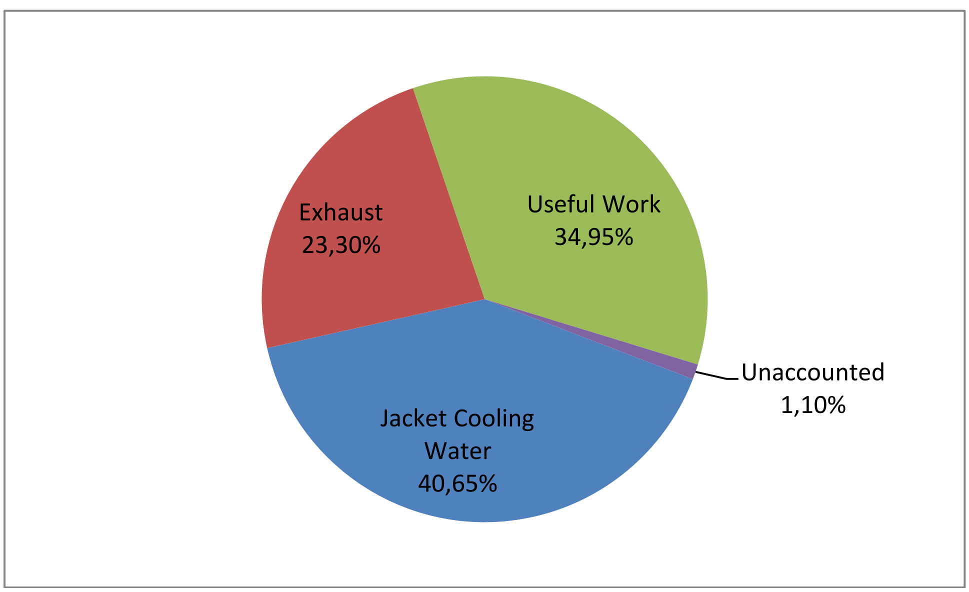


Figure 5.47: Heat Balance in Dual Mode

CONCLUSION AND RECOMMENDATIONS

CHAPTER 6

6.1 CONCLUSION

Since the C.I. engine is considered as dirtiest engine but by implementing CNG in dual fuel mode provides both advantages of less emission and minor design modification. Results of diesel and biodiesel with CNG show that the emission level reduces by 10% and the efficiency also increases by 2-3%. In our project, it is also verified.

In general, the results of data analysis of pollution levels in Delhi, before and after implementation of CNG as an automotive fuel, point to the success of a number of policies implemented but also to a number of areas of growing concern. The policies that lead to the switchover to CNG from conventional fuels has helped reduce SO_2, CO and PM10 concentrations but, contrary to popular belief, have not contributed to increase of NO_x levels. Similarly the policies that lead to the reduction of sulphur content in diesel appear to have helped to reduce air pollution by reducing SO_2 and PM10 concentrations. But it is unfortunate that the gains accrued by switchover to CNG from diesel for buses are being lost by increased pollution from three-wheelers. The recent trend in our country towards an increase in the proportion of Diesel-fuelled cars also is having a somewhat mixed impact on air quality. If the number of all vehicle types continues to increase then the gains from CNG implementation would be negated and lost. As it is not reasonable to limit the mobility of people, it is important for the government to augment the Public Transport infrastructure and advocate and promote the usage of Public Transportation means such as buses, Metro etc.

6.2 SCOPE FOR FUTURE WORK

- An advanced, multiport gas injection system and electronic diesel pump lever control can be designed to provide complete electronic control of both fuels.
- To further minimize tailpipe emissions, an oxidation catalyst can be employed, as well as an insulated electronically controlled, closed-loop exhaust gas recirculation (EGR).
- Urban trucks make good conversion targets since they are relatively large and operate much of the day, making each one a significant fuel consumer. Most buses and commercial trucks now operate on diesel fuel because of the excellent fuel economy, durability and reliability of diesel engines
- The dual-fuel system can be converted to drive-by-wire operation. The former direct connection between the accelerator pedal and the diesel pump lever can be replaced by a pedal position sensor which gives a demand for torque to the ECU. The ECU takes this demand into consideration along with engine speed and other factors to set fuel quantity and to determine what fraction of that quantity is supplied as natural gas and what fraction as diesel. By avoiding overfueling during transients, this system has the potential to reduce emissions and improve fuel economy even when running on diesel,
- At low load operating points, the combination of low overall fuel requirements and a minimum diesel pilot would make the natural gas mixture too lean to burn. In this operating region, a "cylinder selective skip-fire" system can be used to optimize emissions and natural gas utilization. The ECU skips gas injection on alternate cylinders in the firing order while injecting approximately twice the amount of gas in others The average cylinder output meets the required low load while the cylinders which receive gas have a rich enough mixture for good combustion.
- One problem which arises with natural gas injection systems is the variability of gas injection quantities as gas density varies with changes in gas pressure and temperature. The pressure regulator selected for this conversion system should be

a high-stability regulator which includes an integrated shutoff and gas filter as well as thermal control using engine coolant

- **Direct Injection of Gas and Diesel Oil**

  In this system, natural gas is injected at the end of the compression stroke. The injection contains a pilot of 3 to 15 percent diesel fuel depending on the load characteristics of the application. The diesel fuel is injected milliseconds before the natural gas fuel, which instantaneously ignites the mixture. The system uses on dual fuel injector per cylinder that contains two needles (a needle within a needle with separate injector holes for each fuel). The diesel fuel also lubricates the injector and other moving parts

- **Slow speed engines conversion (multipoint fuel fumigation):**

- gas fuel dosed into inlet channel by individual solenoid valve during the suction stroke
- original speed governor replaced by actuator to provide a constant ignition portion
- original speed governing shifted to gas fuel
- suitable for slow speed engines with s big overlap of valves

6.3 RECOMMENDATIONS

The CNG implementation process in Delhi has been a success. Delhi today has one of the world's largest public transport fleet of buses running on CNG. Although CNG implementation has helped reduce pollutant concentrations in Delhi, but the gains have been negated by- poor technology, increase in number of diesel cars and an overall increase in the number of all types of vehicles. In view of the above, the recommendations are as follows:

- Government and the regulating agencies need to aggressively advocate and promote the use of Public Transportation means. It also needs to increase the infrastructure of public transport.
- Emission norms for diesel cars need to be more stringent.
- Any city or authority thinking of implementing CNG as an automotive fuel must learn from the mistakes committed by agencies involved in CNG implementation in Delhi.
- Moreover as Safety is the most important parameter while handling CNG, therefore adequate infrastructure must be developed for inspection of CNG vehicles, especially public transport vehicles.

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