PERFORMANCE AND EMISSION CHARACTERISTICS OF A MULTI-CYLINDER SI ENGINE ON GASOLINE-LPG DUAL FUEL MODE OF OPERATION

Thesis

Submitted in partial fulfillment of the requirements for the Degree of

DOCTOR OF PHILOSOPHY

by

VIGHNESHA NAYAK

DEPARTMENT OF MECHANICAL ENGINEERING
NATIONAL INSTITUTE OF TECHNOLOGY KARNATAKA
SURATHKAL, MANGALORE – 575025

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DECLARATION

I hereby declare that the Research Thesis entitled **“PERFORMANCE AND EMISSION CHARACTERISTICS OF A MULTI-CYLINDER SI ENGINE ON GASOLINE-LPG DUAL FUEL MODE OF OPERATION”** which is being submitted to the **National Institute of Technology Karnataka, Surathkal** in partial fulfillment of the requirements for the award of the Degree of **Doctor of Philosophy** in **Mechanical Engineering** is a *bonafide report of the research work carried out by me*. The material contained in this Research Thesis has not been submitted to any other Universities or Institutes for the award of any degree.

Register Number: **ME11F06**

Name of the Research Scholar: **VIGHNESHA NAYAK**

Signature of the Research Scholar:

Department of Mechanical Engineering

Place: NITK-Surathkal

Date:

CERTIFICATE

This is to certify that the Research Thesis entitled **"PERFORMANCE AND EMISSION CHARACTERISTICS OF A MULTI-CYLINDER SI ENGINE ON GASOLINE-LPG DUAL FUEL MODE OF OPERATION"** submitted by **Mr. VIGHNESHA NAYAK (Register Number: 110664ME11F06)** as the record of the research work carried out by him, *is accepted as the Research Thesis submission* in partial fulfillment of the requirements for the award of the Degree of **Doctor of Philosophy.**

Prof. P. Mohanan
Research Guide
Date:

Chairman-DRPC

Date:

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Place: NITK

Date: **VIGHNESHA NAYAK**

ABSTRACT

Population growth over the last decades has led to tremendous growth in fossil energy demand with increased industrialization and use of vehicles. The most common fuel for internal combustion engines is still made out of oil, but continuous increases in oil prices has increased interest in alternative fuels. Strict international regulations on emissions and improving the combustion efficiency, gaseous fuels found to be better alternative fuel for conventional fuel. Gaseous fuels are promising alternative fuels due to their economic costs, high octane numbers, higher heating values and lower polluting exhaust emissions. LPG, as a relatively clean fuel, is considered one of the most promising alternative automotive fuels because of its emission reduction potential and lower price than gasoline. Turbocharger plays vital role in enhancing the boost pressure of IC engines. Turbocharging the engine will improve the combustion characteristics and reduces the NO_X emission. Dilution of intake charge is the one of the method to reduce NO_X emission. Vaporised water-methanol induction is used to reduce the emissions from the engine.

The present study deals with experimental investigations of LPG-gasoline dual fuel mode of operation on engine performance, combustion and emission characteristics with turbocharging and vaporized water-methanol induction. A stationary four stroke, four cylinders, MPFI engine capable of developing 44 kW at 6000 rpm has been modified to operate on LPG fuel. A separate gas ECU has been developed with software to operate dual fuel mode of operation. The engine operating parameters of speed, load conditions and static ignition timings are varied. A turbocharger is selected based on the exhaust mass flow energy of the engine and installed in the experimental test rig with necessary modification in the intake and exhaust manifold. The waste heat from the exhaust gas has been used to generate vapor from water-methanol mixture and induced into the intake manifold to reduce the emissions from the engine.

Initially experiments are conducted to study the performance, combustion, cycle by cycle variations and emission characteristics of the test engine fueled with different percentage of LPG by mass viz: 0%, 25%, 50%, 75% and 100%. In the next part of investigation, static ignition timings are advanced from 5 deg. bTDC to 8 deg.

bTDC and 11 deg. bTDC to analyze performance and emission characteristics. During this stage percentage of LPG and static ignition timing are optimized based on performance and emission characteristics. Experiments are conducted at full load and part loads in the engine speed range of 2000 rpm to 4500 rpm. In third stage of research, a turbocharger is installed and conducted the experiment for optimized conditions. In the last part of the investigations, the engine tests are conducted with vaporized water-methanol induction. The waste heat from the exhaust gas has been used to generate vapor from deionized water-methanol mixture. Vapor to LPG flow rates of 10, 20 and 30% (on volume basis) are used. The vapor is mixed with the intake air in the intake manifold of the engine.

From experimental investigation for dual fuel mode of operation at 5 deg. bTDC it is found that with the 50% usage of LPG, increases the brake thermal efficiency and volumetric efficiency when compared to gasoline for speed range of 2000 rpm to 4000 rpm. 100% LPG will have much lower CO and HC emissions when compared to gasoline. This is a positive effect on environment. But for other LPG-gasoline ratio these emissions going to increases when compared to 100% LPG but it is well below when compared to gasoline for all speeds. NO_X emission is more for 100% LPG almost 4 times that of gasoline for all speed conditions. For other LPG-gasoline ratio NO_X emission is lower. Combustion results revealed that as the LPG percentage increases the peak pressure also increases and it is maximum for 100% LPG for all the speed. This increase in peak pressure will indicate the LPG will give better combustion properties compared to that of gasoline. Compared to peak pressure, the variation in cycle to cycle for IMEP is less for 50% LPG at higher speed conditions. 50% LPG showed better cycle by cycle fluctuations when compared to other fuel conditions. Net heart release rate shows that gasoline will give the more heat release compare to all other fuels, but 100% LPG will release the heat little earlier than gasoline. Since peak pressure is near to TDC for 100% LPG which results in NHRR to occur earlier than gasoline. Final outcome of the research is 100% LPG will have better combustion properties compared to gasoline but cyclic fluctuations are more for 100% LPG.

Results have shown that advancing the static ignition timing will increase the BP by 12 % at 11 deg. bTDC and 7% at 8 deg. bTDC for gasoline. Whereas for 100% LPG increased in BP is 5 % at 11 deg. bTDC and 2% at 8 deg. bTDC. BTE also increased for both gasoline and LPG when advancing static ignition timing because of reduction in the fuel consumption. Also advancing the ignition timing will engine will work leaner side hence reduction in the fuel consumption. From the results it is revealed that as the static ignition timing is advanced volumetric efficiency is increases for gasoline and 100% LPG. For other fuel conditions there is not much effect of static ignition timing on volumetric efficiency. CO emission will drastically reduce when static ignition timing advanced to 8 deg. bTDC after that not significant reduction in CO emission. 100% LPG shown major reduction in CO emission is obtained while advancing the static ignition timing. But advancing the Static ignition timing resulted in increased HC emission for all fuel blends. NO_X emission also increases with advancing the static ignition timing for all fuel blends because of increase in the in-cylinder temperature. Finally after varying the static ignition timing it is found that 8 deg. bTDC with 100% LPG will resulted in better performance and emission characteristics hence these conditions are optimized for the further research work.

Using turbocharger performance characteristics are improved. For 100% LPG and gasoline with turbocharger BP and BTE is increased when compared to without turbocharger. BTE obtained is maximum at 8 deg. bTDC with turbocharger for 100% LPG when compared to all other condition. Turbocharged engine fuelled with LPG has higher volumetric efficiency as compared to engine without turbocharger for all speed and load conditions. Volumetric efficiency increases for turbocharged engine because of higher intake air pressure will increase the density of air which leads to increase in the efficiency. When compared to base fuel gasoline at 5 deg. bTDC average increase in volumetric efficiency for 100% LPG with turbocharger is 13% at same condition. Emissions are greatly reduced with turbocharger with 100% LPG when compared to gasoline with turbocharger. When compared to base fuel gasoline at 5 deg. bTDC average decrease in CO emission for LPG with turbocharger is 72% at same condition. There is no much variations in HC emission when compared LPG with and without turbocharger at full load conditions. The turbocharged engine fuelled with LPG, there will be a good decrease in NO_X for all load conditions. This is because turbocharger

will increase the charge density hence mixture becomes to lean in the combustion zone hence formation of NO_X will reduces for all load conditions. In-cylinder pressure and net heat release rate (NHRR) also greatly improved with usage of turbocharger. Maximum of 17 bar increase in the in-cylinder pressure is obtained with usage of turbocharger. Turbocharged engine gave great improvement in cycle by cycle fluctuations when compared to naturally aspirated engine. Maximum of 84% reduction in COV of IMEP is obtained for turbocharged LPG fuel. Turbocharger will give the better combustion, performance and emission characteristics for LPG fuel.

From the experimental results for deionized water-methanol induction system it is observed that as the percentage of water-methanol increases, the engine brake thermal efficiency increased for part and full load conditions. Further increase in the flow rate of water-methanol beyond 30% will reduce the brake thermal efficiency drastically. Also results show that water-methanol induction will results in reduction of brake specific energy consumption (BSEC). Water-methanol induction has good effects in decreasing NO_X emission significantly. At full load condition around 30% and 40% average reduction in NO_X emission are obtained for 20% and 30% water-methanol flow rate. HC and CO emissions are going to reduce slightly with water-methanol induction due to presence of more oxygen in the charge to the engine.

It can be seen that use of 50% LPG is superior alternative for unmodified multi-cylinder SI engine for better engine performance and emission characteristics. The use of 100% LPG is best suited for SI engines at 8 deg. bTDC advance static ignition timing with turbocharging and 20%vaporized water-methanol induction rate to get enhanced engine performance and emission characteristics.

Keywords: Gasoline, LPG, Turbocharger, Static ignition timing, Vaporized water-methanol induction, Multi-cylinder engine, SI engine, Performance, Combustion, Emission, COV.

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NOMENCLATURE

A/F	Air Fuel Ratio
BSEC	Brake specific fuel consumption (MJ/kW-hr)
bTDC	Before Top Dead Centre
BTE	Brake thermal efficiency (%)
CA	Crank angle
COV	Coefficient of variation (%)
DP	Differential pressure
deg.	Degree crank angle
ECM	Electronic control Module
EGR	Exhaust gas recirculation
I.C	Internal combustion
IMEP	Indicated mean effective pressure (bar)
LPG	Liquefied petroleum gas
Pmax	Maximum Pressure
μ	Mean
MPFI	Multi point port fuel injection
NHRR	Net heat release rate
ppm	Parts per million
rpm	Revolution per minute
SI	Spark ignition
σ	Standard Deviation
η_{vol}	Volumetric Efficiency
ϕ	Equivalence ratio
WOT	Wide open throttle

CHAPTER 1

INTRODUCTION

1.1 OVERVIEW

Hydrocarbon fuels are currently meeting more than 90% of the total energy demand. The consumption is expected to increase at much faster rate with the need to enhance human comfort and to meet day-to-day developments. An important issue with energy usage is the associated undesirable emissions. A fossil fuel source mainly emits CO_2, which is a primary greenhouse gas. The concentration of CO_2 has increased by about one-third since industrialization began and the average surface temperature of the earth is increasing because of the global warming phenomenon. Other emissions like sulphur, carbon monoxide and oxides of nitrogen are also a great threat to mankind. Use of renewable alternatives comes first while thinking about sustainable, long lasting energy sources, which also will have little or no environmental impact on usage. Recently, much research has been conducted on alternative fuels due to increasing demand for lower fuel consumption and exhaust emissions. And also they focused on improving the combustion efficiency of the engine with conventional fuels. (Chauhan et al. 2010, Huang et al.2007)

During the last decade, gaseous fuels such as liquefied natural gas (LNG) and liquefied petroleum gas (LPG) have been widely used in commercial vehicles and promising results have been obtained from the fuel economy. Among the alternatives suggested were hydrogen, methane and LPG (mainly propane).

1.2 SPARK IGNITION ENGINE

The spark ignition (SI) engine, invented by Nicolas Otto, was the sole basis for automobile transport until the 1970s, and is still the major power source for current light-duty vehicles. The possible fuels for spark-ignition engines include Gasoline, Natural gas (mostly methane), Propane/butane (LPG), Hydrogen, Methanol and derivatives, possibly blended with gasoline, Ethanol and derivatives, and possibly blended with gasoline and Synthetic gasoline fuel. Gasoline (petrol) fuel is generally used for SI engine. Gasoline began to be produced inexpensively with the advent of petroleum refining technologies such as thermal cracking and eventually catalytic cracking. As a result, gasoline became the fuel of choice for internal combustion engines. Spark-ignited engines are therefore often referred to as gasoline engines. In a SI engine, the liquid fuel and air are generally mixed prior to their arrival in the combustion chamber i.e., outside the engine cylinder. The process of preparing this mixture is called carburetion. The basic fuel system in a petrol engine consists of a fuel tank, fuel lines, fuel pump, fuel filters, air cleaner, carburetor or fuel injector and intake manifold (Mingzhang et al. 2014).

1.2.1 Drawbacks of Carburetor system:

1. In multi cylinder engines the mixture supplied to various cylinders varies in quality and quantity since the induction passages are unequal lengths and offer different resistances to mixture flow. The mixture proportion is also affected due to fuel condensation in induction manifold.
2. Carburetors with their choke tubes, jets, throttle valves, inlet pipe, bends, etc., do not give a free flow passage for the mixture. Thus there is a loss of volumetric efficiency on this account.
3. The carburetor has many wearing part. After wear and tear it operates with less efficiency.
4. Freezing may take place at low temperatures, unless special means are provided to obviate this.
5. Backfiring may take place and there is a risk of fuel igniting outside the carburetor unless flame traps are provided.

1.3 MULTIPOINT PORT FUEL INJECTION (MPFI) SYSTEM

Multipoint injection creates the ideal preconditions for satisfying the demands placed on a mixture formation system. In multipoint injection system, each cylinder is assigned a fuel injector, which injects the fuel directly ahead of that cylinder's intake valve.

The smooth running of the engine necessitates high demands being made on the A/F mixture composition of each working cycle. Precisely timed injection is significant as well as precise metering of the injected fuel mass in accordance with air drawn by the engine. Therefore in modern multipoint injection system, not only is each engine cylinder assigned an electromagnetic fuel injector, but also this fuel injector is activated individually for each cylinder. In this way, both the fuel mass appropriate to each cylinder and the correct start of injection are calculated by the electronic control unit (ECU). Injecting the precisely metered fuel mass directly ahead of the cylinder intake valves at the correct moment in time improves mixture formation. This in turn helps in preventing wetting of the intake manifold walls with fuel, thus the advantages of the multipoint injection can be fully exploited. The engine intake manifolds thus carry only the combustion air and can therefore be optimally adapted to the gas dynamic requirements of the engine.

Table 1.1 Important component of MPFI system with its function and location.

Sl. No.	Name of the component	Location	Function
1.	Throttle Position Sensor	Throttle body	To detect the degree of throttle opening and sends the signal to ECM
2.	IAC valve	Throttle body	To supply by-pass air depending on engine condition, controlled by ECM
3.	Electric fuel pump	Basically located in the fuel tank but in our lab it is fitted to the frame of the engine	To supply the fuel at a pressure to the fuel injectors through delivery pipe
4.	Fuel pressure regulator	On delivery pipe	To maintain the fuel pressure in the line. It is maintaining the pressure about 2.55 bar, and it is also called intake manifold pressure
5.	Fuel injector	Between the delivery line and intake manifold	To inject the fuel into intake port of the cylinder head according to the signal from electronic control module (ECM)
6.	Electronic control module (ECM)	In the vehicle it is placed under passenger side instrumental panel but in lab it is placed in panel board	It controls various devices in the engine. It takes various inputs from sensors according to the requirement it is giving output.
7.	Map sensor	On intake manifold	It converts the pressure change in intake manifold to voltage change and gives signal to the ECM so pressure can be measured by ECM
8.	Engine coolant temperature sensor	On thermostat case	Coolant temperature measures and converts the changes in temperature into resistance changes in thermostats

9.	Indicated air temperature (IAT) sensor	In air cleaner	It is measuring the air temperature and sends the signal to the ECM
10.	Vehicle speed sensor	On the output shaft of gearbox	Generates the signal which is in proportion to the vehicle speed and sends signal to ECM
11.	Camshaft position sensor	On left side camshaft housing	Sends the electrical pulse signals on rotation of the camshaft
12.	Crankshaft position sensor	Mounted on oil panel	Generates A/C voltage pulse on rotation of the camshaft
13.	Engine start signal	On starter	Send signal from starter circuit to ECM
14.	PSP switch	On power steering pump	To switch on the power steering pump when oil pressure is > 35 to 45 bar
15.	Diagnostic switch terminal	Located in diagnostic connector in relay/ fuse box	To send the diagnostic signal to ECM when the terminal is grounded
16.	Test switch terminal	Located in diagnostic connector in relay/ fuse box	To reset the ignition timing to initial ignition timing when the terminal is grounded
17.	Heated oxygen sensor	On exhaust manifold	It detects the concentration of oxygen in exhaust gas and sends the signal to ECM so mixture ratio can be changed

1.4 LPG AS A FUEL IN SI ENGINES

Liquefied petroleum gas (LPG) is a mixture of hydrocarbons, mostly propane (C_3H_8) and butane (C_4H_{10}) isomers. The composition of LPG depends on its end use and varies greatly according to season, country and properties of the crude oil/gas supply used and refining process (Massimo 2012). LPG, as a relatively clean fuel, is considered one of the most promising alternative automotive fuels worldwide because of its emission reduction potential and lower fuel price compared to gasoline. (Changming et al. 2010) A major disadvantage of the LPG is the NO_X emission which is greater than that for liquid fuels because of its higher peak temperature and high flame velocity compare to gasoline. LPG is well known as a clean alternative fuel for vehicles because it contains less carbon molecules than gasoline or diesel. Its higher ratio of carbon (C) to H reduces the amount of carbon dioxide (CO_2) and other non-regulated emissions, such as formaldehyde and acetaldehydes. LPG also has other many advantages such as high octane number, high combustion value, little carbon accumulation, easy storage, and low cost. Searches for alternative energy sources in automotive industry have brought forward the use of LPG in vehicles as fuel. Nowadays, LPG is widely used as fuel in cars in developed countries (Italy, Netherlands, France, Belgium, Japan and Australia).

LPG fuel is preferred as a clean alternative fuel for internal combustion engines due to easy availability and storage, low cost, high octane number, high combustion efficiency and low exhaust emissions with respect to other fuels (Gumus 2011). The flame propagation speed of LPG is higher in slight lean mixture region (equivalence ratio around 0.9 to 1.0) but that of gasoline is promoted in the rich mixtures. Hence LPG has better combustion characteristics at lean burn engines. The flame propagation speed of LPG and gasoline is nearly same when mixture is too lean (Ceviz and Yuksel 2005). LPG has higher octane number of about 112, which enables higher compression ratio to be employed and gives more thermal efficiency. Due to gaseous nature of LPG fuel distribution between cylinders is improved and smoother acceleration and idling performance is achieved. Fuel consumption is also better (Han et al. 2008). Engine life is increased for LPG engine as cylinder bore wear is reduced & combustion chamber and spark plug deposits are reduced. As LPG is stored under pressure, LPG tank is heavier and requires more space than gasoline tank. There is reduction in power output for LPG operation than gasoline operation. Starting load on the battery for an LPG engine is higher than gasoline engine due to higher ignition system energy required (Murillo et al. 2005). LPG system requires more safety. In case of leakage LPG has tendency to accumulate near ground as it is heavier than air. This is hazardous as it may catch fire. Volume of LPG required is more by 15 to 20% as compared to gasoline. LPG operation increases durability of engine and increase in the life of exhaust system (Yamin and Badran 2001).

1.5 PHYSICAL PROPERTIES AND CHARACTERISTICS

1.5.1 Density

LPG at atmospheric pressure and temperature is a gas which is 1.5 to 2.0 times heavier than air. It is readily liquefied under moderate pressures. The density of the liquid is approximately half that of water and ranges from 0.525 to 0.580 @ 15 ºC. Since LPG vapor is heavier than air, it would normally settle down at ground level/ low lying places, and accumulate in depressions.

1.5.2 Vapor Pressure

The pressure inside a LPG storage vessel/ cylinder will be equal to the vapor pressure corresponding to the temperature of LPG in the storage vessel. The vapor pressure is dependent on temperature as well as on the ratio of mixture of hydrocarbons. At liquid full condition any further expansion of the liquid, the cylinder pressure will rise by approx. 14 to 15 kg./sq.cm. for each degree centigrade. This clearly explains the hazardous situation that could arise due to overfilling of cylinders.

1.5.3 Flammability

LPG has an explosive range of 1.8% to 9.5% volume of gas in air. This is considerably narrower than other common gaseous fuels. This gives an indication of hazard of LPG vapor accumulated in low lying area in the eventuality of the leakage or spillage. The auto-ignition temperature of LPG is around 410-580 ^{o}C and hence it will not ignite on its own at normal temperature. Entrapped air in the vapor is hazardous in an unpurged vessel/ cylinder during pumping/ filling-in operation. In view of this it is not advisable to use air pressure to unload LPG Cargoes or tankers.

1.5.4 Combustion

The combustion reaction of LPG increases the volume of products in addition to the generation of heat. LPG requires upto 50 times its own volume of air for complete combustion. Thus it is essential that adequate ventilation is provided when LPG is burnt in enclosed spaces otherwise asphyxiation due to depletion of oxygen apart from the formation of carbon-dioxide can occur.

1.5.5 Odor

LPG has only a very faint smell, and consequently, it is necessary to add some odorant, so that any escaping gas can easily be detected. Ethyl Mercaptan is normally used as stanching agent for this purpose. The amount to be added should be sufficient to allow detection in atmosphere 1/5 of lower limit of flammability or odor level 2 as per IS: 4576.

1.5.6 Color

LPG is colorless both in liquid and vapor phase. During leakage the vaporization of liquid cools the atmosphere and condenses the water vapor contained in them to form a whitish fog which may make it possible to see an escape of LPG.

1.5.7 Toxicity

LPG even though slightly toxic, is not poisonous in vapor phase, but can, however, suffocate when in large concentrations due to the fact that it displaces oxygen. In view of this the vapor possess mild anesthetic properties.

Table 1.2 Physical properties of Gasoline and LPG (Masum et al. 2013, Tian et al. 2010, Zhuang and Hong 2013)

Sl. No.	Particulars	Gasoline	LPG
1.	Chemical Formulae	C_2-C_{12}	60% C_4H_{10}+40% C_3H_8
2.	Max. Vapor Pressure Saturated in bar at 65 °C	0.2882	16.87
3.	Gross calorific value in kJ/kg.	42900	49728
4.	Specific gravity (liquid) at 15 °C Water =1	0.75-0.765	0.543
5.	Stoichiometric air fuel ratio (kg/kg)	14.7	15.5
6.	Flammability limits (Upper)	7.6%	9.1%
7.	Flammability limits (Lower)	1.4%	1.90%
8.	Ignition Temperature (°C)	257	488-502
9.	Research Octane Number (RON)	95	112
10.	Max. flame temperature (°C)	2002	1985
11.	Latent heat of vaporization (kJ/kg)	180-350	426
12.	Boiling Point (°C)	25-215	-22

1.6 ADVANTAGES OF GASEOUS FUEL (LPG) OVER LIQUID FUEL (Gasoline):

1. Fuel atomization: Atomization of fuel needs external energy to atomize the fuel in the combustion chamber. Heavier fuel needs higher atomization cost. In case of gaseous fuel the same is absent, giving gaseous fuels an advantage of 2-3%.
2. Burning Speed: The ability of gaseous fuels to burn faster than liquid and solid fuels ensure no unburnt fuel going in the exhaust. This further improves the efficiency of gaseous fuels especially in case of high consumption applications.

3. Latent heat of Vaporization: During combustion process liquid fuel gets converted to vapor and the vaporization process takes away the latent heat of vaporization from the combustion heat. The latent heat of various liquid fuels varies from 378-462 kJ/ kg. Gaseous fuels does not require any vaporization hence no such heat loss from the combustion process.
4. Excess Air: All fuels except gases are burnt 100% only at slightly positive pressure. This characteristic of liquids and solids require combustion air being fed at more pressure than in case of gaseous fuels. Gaseous fuels are the only fuel which can be burnt 100% at atmospheric pressure, requires less combustion air pressure hence less stack losses. This characteristic of gaseous fuels gives an advantage of more than 10% over liquids in terms of less exhaust losses.

1.7 CYCLE BY CYCLE VARIATION

Since the birth of the engine there has been a fundamental problem, the cycle to cycle variation. This limits the engine performance and gives rise to increased emissions. The combustion process in a spark ignition engine is not repetitive from engine cycle to engine cycle. This can easily be noted if the pressure trace in the cylinder is measured. The peak pressure obtained can change in 30% from cycle to cycle in a well-functioning engine (Johanson 2003).

Historically it is the cylinder pressure that has been used to measure the fluctuations. This has led to the use of pressure related parameters to quantify the fluctuation intensity. The maximum pressure and its crank angle location are frequently used parameters (Heywood 1998). The variation in indicated mean effective pressure, IMEP, produced per engine cycle is also a well-used parameter. The standard deviation is usually normalized with the average value to give a coefficient of variation, COV of IMEP. These parameters have the benefit of requiring no modeling and the COV of IMEP shows how much torque fluctuation that the transmission etc. must tolerate. COV of IMEP is thus a good parameter to be used for transmission design and a general indicator of engine behavior. The major drawback with the parameters derived from the pressure directly is the lack of knowledge on the ongoing process. The only reasonable way the pressure can change from cycle to cycle is variation in the combustion process. Hence, there is every reason to use the pressure in a heat release model and analyze the heat release function instead of the pressure trace, especially if the origins of fluctuations are of interest. Even if more detailed information on the combustion process is required, the heat release calculation should be replaced by some form of flame location detection.

In general, the combustion in a spark ignition can be expected to fluctuate during the entire flame propagation process. The fuel and residual gases are generally not well mixed with air and hence the laminar flame speed will differ depending on location and time. The same argument can be used for the flow situation. The level of turbulence cannot be expected to be homogeneous and the mean flow situation will also change from cycle to cycle and from location to location. But even though fluctuations are expected for large flames, these flames will have the benefit of integrating out in homogeneity in the fuel, residual and flow fields. The very small flame in the early part of the combustion does not have this possibility to average out the flame speed setting parameters. Thus, this part of the combustion is expected to have the greatest problem with cycle to cycle variation (Hao, 2014). Strong cycle-to-cycle variations of engine flows often prohibit SIDI engines from reaching their full potential of efficient and clean combustion. It is because the variations of air flow, fuel and temperature distributions in the vicinity of spark plug prior to ignition all affect the early flame formation, propagation and the subsequent combustion processes.

Bizon et al. (2009) studied the reducing cyclic variability of parameters, seeking thus to modify the appropriate design and operating parameters. A qualitative analysis of cycle-to-cycle variation may aid better understanding. In diesel engines, cyclic variability is due to unsteady in-cylinder flow and injection variations (Long, 1995). Zhong et al. (2003) showed that the amount of fuel injected may vary by 23%, resulting in a cyclic variation of indicated mean effective pressure (IMEP).

1.8 EFFECT OF IGNITION TIMING

LPG is having higher self-ignition temperature than gasoline which increases the combustion duration. Also LPG is having higher calorific value and latent heat of vaporization than gasoline. When fuels are having different combustion properties then there is a need of changing the spark ignition timing. Spark ignition timing has a significant impact on the engine performance and emission characteristics when fuels are having different combustion properties. If spark timing is too advanced there is substantial increase in the cylinder pressure before piston reaches to the TDC point. Hence there will be more compression work against the exhaust gas and hence net work output will be decreased. Therefore optimum spark timing will produce the satisfactorily high cylinder pressure with peak occurring just after the TDC. Hence optimum spark timing will give the minimum compression work and maximum expansion work (Erkus et al. 2015, Alsfour 1998, Lawankar et al. 2012, Turkoz et al. 2014).

1.9 TURBOCHARGING

Turbo charging plays vital role in increasing the boost pressure of an Internal Combustion Engine and also reduce exhaust emissions. A turbocharger is a turbine driven compressor. The thermodynamic matching of the turbocharger is implemented by the means of mass flow and energy balances. The turbine and the compressor power output are identical in a steady state condition. The matching calculation is iterative, based on compressor and turbine maps, as well as the most important engine data. The turbocharger has three principal components: a compressor which gives boost pressure, a turbine which drives compressor and linked to it by a shaft, and bearing assemblies to support the shaft. When the pressure of the engine's intake air is increased, its temperature also increases. In addition, heat soak from the hot exhaust gases spinning the turbine may also heat the intake air. The warmer the intake air, the less dense, and the less oxygen available for the combustion event, which reduces volumetric efficiency. Not only does excessive intake-air temperature reduce efficiency, it also leads to engine knock, or detonation which is destructive to engines. The proper selection of turbocharger components to make up a complete engine turbocharger system is a complicated balance of many design considerations. Depending on the nature of the application, the performance may be specified at a single operating point or a wide range of different operating points. A large turbo may give more peak power, but can take more time to spool up. Turbochargers are costly to add to naturally aspirated engines, and add complexity. Adding a turbo can often cause a cascade of other engine modifications to cope with the increased power, such as exhaust manifold, intercooler, gauges, plumbing, lubrication, and possibly even the block and pistons. With the use of turbocharger NO_X emission can be decreased (Zhen and Yang 2013, Nicholas 2005, Baskharone 2006).

1.10 NO_X REDUCTION TECHNIQUE

The peak cycle temperature shoots up whenever the load is increased, which tends to accelerate NO_X formation. Several techniques have been tried to inhibit NO_X formation. Some of them are: use of EGR, Turbo-charging with intercooling, addition of diluents or water injection along with the intake charge etc. Injection of water into the intake manifold has been found to be an effective way to reduce NO_X emission in SI, CI and LPG engines.

Increasing the intake charge humidity was also reported as an efficient technique to control NO_X emission. The concept of water addition as a supplement to the internal combustion engine has been around for over 50 years. It is a well-known fact that water does not burn but it is excellent at absorbing heat due to water having a high specific heat capacity and latent heat of evaporation. The latent heat of evaporation of water is 2256 kJ/kg, which is approximately 6 times greater than that for gasoline under standard atmospheric pressure and temperature. Since it is a good absorber of heat, peak temperature in the cylinder will reduce so that the NO_X emission will greatly reduce.

Water addition, as a separate liquid or emulsion with fuel for automobile engines, has been investigated and reported in published papers extensively. These investigations are generally related to water effects on engine performance, knock, and emissions.

Production of NO_X depends on the fuel/air equivalence ratio, maximum cycle temperature, and burning rate. The NO_X emission is a function of the fuel/air equivalence ratio for different water to fuel mass ratios. The peak NO_X emissions occur at slightly lean conditions, where the combustion temperature is high and there is excessive oxygen to react with the nitrogen as a result of the tendency of dissociation. The water injection reduces the NO_X emission in the lean region having a local maximum between equivalence ratios of 0.9 and 1.0. Because the combustion process is closer to a stoichiometric ratio and produces a higher flame temperature, the NO_X emissions increased, particularly by the increase of thermal nitrogen oxide. The drop in temperature and reduction of the combustion rate with water addition are the main reasons for the NO_X reduction. The effect of 0.5 g of water addition/g of fuel yields a lower adiabatic flame temperature at a level of approximately 150 K for a stoichiometric fuel/air mixture. Therefore water is added into the fuel-air charge at the intake manifold with the induction method, which is presumably the simplest and most effective method.

1.11 PRESENT WORK

The present study deals with experimental investigations on the combustion, performance and emission characteristics of multi-cylinder SI with LPG-Gasoline dual fuel mode of operation with turbocharging and water methanol induction. For experimentation, 44.5 kW capacity Zen MPFI engine has been made in to test rig with all necessary instrumentation for measuring combustion, performance and emission parameters. Engine test rig is modified to work in LPG fuel injection after incorporating an aftermarket LPG injection kit with LPG open ECM. The engine is coupled with an eddy current dynamometer for measuring the load on the engine. Sequence of experiments are carried out with engine operating parameters of speed, load and various percentage of LPG on mass basis with different static ignition timings. To compare the results of experiments with different percentages of LPG-gasoline ratio and static ignition timings is optimized as a baseline fuel on engine performance, combustion and emission characteristics. Experiments are also carried out on the engine test rig with turbocharging for optimized fuel blend and static ignition timing. With experiment data cycle by cycle fluctuation study also carried out. To reduce the emission from the optimized fuel blend, the method vaporized water-methanol induction is employed. The waste heat from the engine exhaust gas is used to heat water-methanol mixture in a heat exchanger and it is converted into vapor state. The vaporized water-methanol at various proportions of 10, 20 and 30% of LPG fuel consumption are inducted along with intake air. Engine combustion, performance, emissions and cyclic variations are studied. Finally comparative study with baseline fuel are studied.

1.12 ORGANIZATION OF THESIS

The thesis has been organized in to 6 chapters starting with the introduction. This chapter gives the background of the problem definition and basic details. The second chapter deals with the in depth literature review covering mainly the various aspects related to combustion, cycle by cycle variations, performance and emissions of alternative fuels and the methods improve the same. Pervious works on the use of gaseous alternative fuels, turbocharging and emission reduction technology are detailed in this chapter. Based on the literature review, the objectives of the present work are also described in this chapter. The third chapter presents the introduction to engine setup, instrumentation and measurement system. Engine modification done for the LPG conversion, Turbocharging and vaporized water-methanol induction are also given. The experimental methodology and experimental procedures along with error and uncertainty analysis are described in chapter four. The results for experimental work is analyzed and given in chapter 5. The sixth chapter is devoted to bring out the important conclusions based on this work with recommendation and the scope for future work.

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CHAPTER 2

LITERATURE REVIEW

The scarcity of petroleum resources in the oil market along with the acutely growing demand of the oil threatens the security of energy production. Energy policy and planning with the related orientation have become very important in developed and developing countries nowadays. And due to the environmental problems exhaust gas emission regulations become more stringent. In this context a review of literature on the use of various alternative fuels like LPG for spark ignition engine has been done in this chapter. And also various emission reduction techniques and performance improvement methods used also been reported.

2.1 USE OF LPG IN SI ENGINE

Gaseous fuels in general are promising alternative fuels due to their economical costs, high octane numbers, high calorific values and lower polluting exhaust emissions. The benefit of these fuels is that they emit less air pollutant compare to gasoline and most of them are more economically beneficial compared to oil and they are renewable. The most common fuels that are used as alternative fuels are natural gas, LPG, ethanol, methanol and hydrogen (Schoenung 2001, Durgun 1988). Lots of works have been done on engine operating with these fuels. Following section will deals with use of LPG fuel in SI engine.

Philip Price et al. (2003) studied the thermodynamic performance of the evaporator used in the ford focus liquefied petroleum gas was calculated over the engine power range. The authors studied three parts in this paper: Evaporator study, Evaporator performance calculations and vehicle testing.

Lee and Ryu (2005), done the experiment to investigate flame propagation and combustion characteristics of LPG fuel. Laser deflection method and the high speed Schlieren photography method were employed to measure the flame propagation speed of LPG fuel. The flame propagation speed in the constant volume combustion chamber (CVCC) reached a maximum at the stoichiometric equivalence ratio. The effect of equivalence ratio on the combustion duration was rapidly increased in the lean mixture region. This result indicates that the combustion worsened in the lean equivalence ratio region.

Bayraktar and Durgun (2005) investigated the effect of combustion parameter on LPG fuelled single cylinder SI engine. Results showed that significant improvements in exhaust emissions can be achieved. However, variations in various engine performance parameters and the effects on the engine structural elements are not promising. They recommended since LPG had a high octane number, it may lead to operating with higher compression ratios, and consequently, the engine efficiency and fuel economy would be better.

Loganathan and Ramesh (2007) are investigated effect of gasoline and LPG injection into the manifold of a 145cc two stroke engine using a specially developed electronic circuit to have close control over the air fuel ratio. The maximum brake thermal efficiency with LPG was 25% and that with gasoline was 23%. The injection pulse width that resulted in the best brake thermal efficiency also resulted in the lowest HC emissions with LPG and gasoline. HC levels and exhaust gas temperature were slightly higher with LPG while NO levels were comparable in carbureted LPG. The COV of IMEP and peak pressure are lower with LPG with gasoline. The higher HC and NO_X levels with LPG may have to be tackled through after treatment device and retard spark timing respectively.

Ozcan and Yamin (2008) were investigated performance and emission characteristics of LPG powered vehicle with variable compression ratio and stroke length. The variable stroke technique can be used to improve the performance and emission characteristics of LPG fueled spark ignition engines. Shorter stroke lengths, on the other hand, caused the cylinder pressure and temperature to increase and, hence, increase the thermal and mechanical stresses on the engine.

Jothi et al. (2008) investigated the studies of exhaust gas recirculation (EGR) on homogeneous charge ignition engine. DEE was added in liquid state as drops and just before the intake manifold it mixes in the form of vapor with LPG-air mixture. At full load NO concentration considerably reduced to about 68% as compared to LPG operation without EGR.

Saraf, R.R. et al. (2009) studied the emission of newly introduced gasoline/LPG bifuel automotive engine in Indian market. Emissions were tested as per LPG Bharat stage III driving cycle. CO emissions were in range of 38.9 to 111.3ppm.

HC emissions were in the range of 18.2 to 62.6ppm. NO_X emissions were in the range of 0.8 to 3.9ppm and CO_2 emissions were from 6719.2 to 8051ppm.

Lee et al. (2009) experimentally investigated dimethyl ether (DME) blended LPG fuel in SI engine. Results showed that stable engine operation was possible for a wide range of engine loads up to 20% by mass DME fuel. Exhaust emissions measurements showed that hydrocarbon and NOx emissions were slightly increased when using the blended fuel at low engine speeds. However, engine power output was decreased and break specific fuel consumption (BSFC) severely deteriorated with the blended fuel since the energy content of DME is much lower than that of LPG. Furthermore, due to the high cetane number of DME fuel, knocking was significantly increased with DME.

Shankar and Mohanan (2010) studied on a four cylinder multipoint port fuel injection gasoline engine retrofitted to run with LPG injection with respect to combustion, performance and emission characteristics were done. The findings of the experiments suggest that higher thermal efficiency and therefore improved fuel economy can be obtained from SI engines running on LPG as opposed to gasoline. However, author reported advanced static ignition timing has an adverse effect on NO_X emissions, which increases further.

Pourkhesalian et al. (2010) were comparatively studied different alternative fuels including LPG on performance and combustion characteristics. They concluded that volumetric efficiency of the engine working on hydrogen is the lowest (28% less that gasoline fueled engine), gasoline produce more power than the all being tested alternative fuels. Liquid fuels tested will produce more power rather than gaseous fuels and they produce less NO_X.

Massimo (2012) studied the performance and emission characteristics of LPG fuelled SI engine for a passenger car with conversion kit for dual fuel mode of operation. Research also tells about LPG as a better alternative fuel for present condition with their advantages. The experimental result showed that gaseous LPG port injection system leads to noticeable performance deterioration and it is due to deterioration of volumetric efficiency and insufficient fuel delivery.

Gumus (2011) studied the effects of variation in volumetric efficiency on the engine emissions characteristics with different LPG usage levels (25%, 50%, 75% and

100%), on an engine operated with multipoint and sequential gas injection system were investigated. The results showed that volumetric efficiency decreased considerably at the use of 25% LPG level. As for the 50%, 75% and 100% LPG usage, volumetric efficiency decreased in proportion to LPG usage level. Air fuel ratio decreases with increase in LPG usage level and the minimum air fuel ratio value obtained at 100% LPG usage. At the mixture containing 25% LPG, brake specific fuel consumption and energy consumption decreased while the brake thermal efficiency was maintained. Positive results are obtained at all LPG usage levels in terms of exhaust gas emissions. With use of fuel blends containing 25%, 50%, 75% and 100% LPG, CO emissions decreased by 26.8%, 26.2%, 40.7% and 53.3% respectively. With use of fuel blends containing 25%, 50%, 75% and 100% LPG, HC emissions decreased by 27.7%, 41.4%, 53.1% and 72.6% respectively.

Sulaiman et al. (2013) analysed the performance characteristics of LPG fuelled single cylinder SI engine. Authors revealed that SI engine fuelled by LPG has slightly decreased on power output up to 4 % compared to gasoline. However, engine fuelled by LPG reduce on specific fuel consumption (SFC) to 28.38 % compared to gasoline.

Morganti et al. (2013) experimentally found out the research octane number (RON) and motor octane number (MON) for LPG. In the experiment they revealed that RON and MON are better for LPG than gasoline. Also suggested LPG can be used with higher compression ratio.

Elnajjar et al. (2013) found the effect of LPG fuel with different compositions on single cylinder SI engine experimentally. The result data indicates that different LPG fuel composition has minimal effect on the engine efficiency and has strong impact on the levels of generated combustion noise.

Erkus et al. (2013) comparatively studied carbureted and injection fuel supply methods for LPG fuelled single cylinder SI engine. The test results showed that the LPG gas injection system developed in this study can help to achieve higher engine power outputs, lower specific fuel consumption and lower exhaust emissions.

Morganti et al. (2015) investigated the auto ignition of LPG fuel in CFR engine. They found nitric oxide (NO) was to be a strong promoter of both fuel oxidation and auto-ignition. And also revealed physically reasonable concentrations of both carbon monoxide (CO) and unburned hydrocarbons (UHCs) had an insignificant effect on auto-ignition in comparison with NO

Ceviz et al. (2015) done the research on effect of LPG injection temperature on the engine performance and emission characteristics of a single cylinder SI engine. According to the test results, engine performance and NO emission characteristics can be improved by controlling the LPG temperature before injecting to the engine intake manifold. Results of the study showed that the engine brake power loss can be increased by about 1.85% and NO emissions can be decreased by about 2% as compared to the operation with the original LPG injection system.

Cinar et al. (2016) studied the performance and emission characteristics of single cylinder SI engine with LPG as a fuel for different valve lift positions. It has found that with the usage of LPG, efficiency, torque and power are reduced with increase in specific fuel consumption. However, HC and CO emissions decreased, NOx emissions increased with the usage of LPG fuel.

Kacem et al. (2016) experimentally studied the effect of LPG-hydrogen mixture in unleaded gasoline engine. They revealed that with the usage of LPG emissions are greatly reduced.

2.2 WORK RELATED TO IGNITION TIMING

Alasfour (1998) studied experimentally effect of ignition timing on NO_X emission, knock characteristics, thermal efficiency and exhaust temperature in a spark ignition engine. From the experimental results it has been found that retarding the ignition timing will causes the rise in the exhaust gas temperature and decrease in the thermal efficiency. Authors also revealed from the experiment that advancing the ignition timing causes combustion process to occur near to TDC which will increases the peak pressure and temperature within the combustion chamber leaving high level of NO_X emission. Research also discovered that advancing the ignition timing, the peak NO_X to be shifted towards lean fuel-air equivalence ratio.

Topgul et al. (2006) are experimentally investigated effect of ignition timing on performance and exhaust emission characteristics of ethanol-gasoline blends on SI engine. Authors are varied ignition timing, compression ratio and fuel blends at 2000 rpm and wide open throttle conditions. Results shown that there is no much change in the MBT when ignition timing is advanced. CO and HC emissions are slightly reduced when the ignition timing is retarded.

Kwak et al. (2007) experimentally investigated time resolved thermal HC emission characteristics of liquid phase LPG injection in SI engine. They suggested that varying ignition timing will reduce the thermal HC emissions.

Akansu and Bayrak (2011) are investigated experimentally the effect of LPG fuel on single carbureted SI engine for different spark timings. From the research work it has been found the advancing the spark timing will results in higher thermal efficiency, NO emissions and reduction in CO and HC emissions.

Erkus et al. (2015) experimentally conducted varying ignition timings on SI engine with LPG as a fuel. Researchers are conducted at wide open throttle (WOT) conditions and 4300 rpm to analyze performance and emission characteristics. The study reveals that higher octane number of LPG allowed for use of lean mixtures with advanced ignition timings without knock even under heavy load operation. Advancing the ignition timing with LPG results in better performance of the engine. Experiments showed brake thermal efficiency and BSFC values over 33% and 25% respectively. Advancing the ignition timing has not much effect on CO emission but HC and NO_X emissions are going to be increased.

Arroyo et al. (2015) studied the effect of ignition timing on supercharged gaseous fuel in SI engine. Tests are conducted at 2500 rpm and full load conditions. Ranges of ignition timings are taken 11 – 59 degree bTDC depending upon fuel used. Results revealed that advancing the ignition timing will give higher thermal efficiency. The most advanced ignition timings will have peak pressure to occur near to TDC. Tests also revealed that the flame development and rapid burning angles are more influenced by the ignition timing in the case of fuels which require more advanced spark angles, that is, fuels with less content of hydrogen. Also research discovered that cyclic irregularities is decreased by advancing the ignition timings.

2.3 WORK RELATED TO TURBOCHARGER

Eriksson et al. (2002) studied the effect of turbocharging in SI engine through modelling. Study showed that increase in the power output with decrease in the emissions were obtained.

Rao and Mohan (2003) investigated the effects of supercharging on the performance of a direct injection diesel engine with the use of untreated cotton seed oil. It was seen a reduction in brake specific fuel consumption of about 15% when the engine is run at the recommended injection and supercharging pressure compared to naturally aspirated engine.

Wu et al. (2003) studied performance analysis and optimization of a supercharged miller cycle Otto engine. Authors described the supercharged Otto engine adopted for Miller cycle version, it has no efficiency advantage but does provide increased net work output with reduced propensity to engine knock problem.

Kesgin (2005) investigated the effect of the turbo charging on the performance of a natural gas engine. He showed the effects of various parameters such as diameter of the exhaust manifold, diameter of the pipe at the turbine exit, location of the turbocharger, back pressure at the turbine exit on the efficiency of the turbocharger.

Sarvi et al. (2008) investigated the effect of engine operation mode and turbocharger on the emissions from large-scale medium-speed diesel engines. In the results, it was found that the exhaust emissions were also considerably dependent on the engine turbocharger system.

Moulin and Chauvin (2011) investigated the modeling and control of air intake system of a turbocharged gasoline engine. Authors describes the control strategy is based on feedback linearization and constrained motion planning, and makes use of a dynamic inversion of a physical representation of the system and an anti-windup scheme.

Gonca et al. (2013) studied the effect of steam injection in a turbocharged engine. Authors illustrated the variation of ratio of water mass to air mass passing through the intercooler with respect to pressures is demonstrated.

Fu &Yang (2013) proposed a new concept of steam turbo charging to boost IC engine intake pressure. The results show that IC engine power can be theoretically improved by 7.2% at most, & thermal efficiencies can be improved by 2 % points or more except at 1000 r/min by using steam turbo charging. All these can prove this boosting pressure concept is a novel technology with great energy saving potentials.

Gharehghani and Koochak (2013) experimentally investigates the thermal balance and performance of a turbocharged gas spark ignition engine. Results indicate that by increasing engine load and coolant temperature, the percentage of transferred energy to the exhaust gases increased while the percentage of coolant energy decreased. Also, experimental data reveals that using gaseous fuel and a turbocharger (TC) in the engine leads to 4.5% and 4% more thermal efficiency than gasoline and natural aspirated (NA), respectively.

2.4 CYCLE-BY-CYCLE COMBUSTION VARIATION

Lean burn is one of several effective methods for improving fuel consumption and reducing NO_X emissions in an automotive SI engine. However, lean burn operation increases the cyclic combustion variation and in the worst case deteriorates vehicle drivability. Cyclic variability is recognized as a limit for operating conditions with lean and highly diluted mixtures. The cyclic variations are caused by both chemical and physical phenomena. Of these phenomena, the variations in the residual gas fraction, the fuel–air ratio, the fuel composition and the motion of unburned gas in the combustion chamber can be taken into consideration. Cyclic variability is recognized as a limit for operating conditions with lean and highly diluted mixtures. Previous studies showed that if cyclic variability could have been eliminated, there would be a 10% increase in the power output for the same fuel consumption and power pollution of emissions from the engine.

An extensive literature survey has been done by Young (1982) to assess the state of the art relative to cyclic dispersion in combustion and its effect on the subsequent pressure development in the cylinder of a homogeneous charge SI engine. He has conducted survey ranging from 1950 to 1980. The study includes the effect of chemical factors such as equivalence ratio, charge dilution and fuel type. The physical factors include ignition system, combustion chamber geometry, engine speed, compression ratio, swirl and turbulence.

The influence of non-homogeneity on cycle-by-cycle variations was studied by Pundir et al. (1981) and shown that cyclic variations increases with increase in charge non homogeneity at a given mixture strength. Gatowski et al. (1984) have developed a heat release analysis procedure of SI engine which includes the effects of heat transfer, crevice flow and fuel injection. The model developed has been validated with experimental results. Cycle-by-cycle application of the model tend to predict small negative heat release rates at the end of combustion for fast burning cycles. The possible sources of error include the heat transfer correlation, the method used to represent the thermodynamic properties, or thermal effects in the pressure transducer.

Kalghatgi (1987) has shown that the advanced ignition systems are of practical interest for automotive applications as they influence the cyclic variations. The cyclic variations in SI engines can be reduced at source by reducing the variations in combustion. Lean burn is one of several methods for improving efficiency however increases the cyclic variations.

Yamamoto and Misumi (1987) has analyzed the cyclic combustion variation in a lean operation SI engine with the IMEP variation being subjected to multiple regression analysis to identify the causes of the cyclic variations, and found that the main cause of IMEP variations in the lean operating SI engine was the released heat quantity variations. A more detailed review of literature on cyclic variability in SI engines has been done by Ozdor et al. (1994), which review the effect of various parameters and their contributions. They also give an insight regarding the various indicators used for measurements of cycle-by-cycle variations. Accordingly the pressure related parameters such as Pmax, θPmax and IMEP are still most valuable in estimating in quantitative terms, the effect of various variables on the cycle-by-cycle variations. They have also distinguished between the factors causing and influencing the cycle-by-cycle variations.

Cheng et al. (1993) gave an over view of UBHC emission and shown that reduction in HC emission will produce lesser cyclic variations and improved engine performance, efficiency. Whitelaw and Xu (1995) has investigated the cyclic variations in a lean burn SI engine without and with swirl. Measurements of cylinder pressure and flame travel velocity in the lean limit have been done. The extent to which residual burned gas retarded the combustion rate and increased cyclic variability are quantified.

Ishli et al. (1997) have investigated the cyclic variations of IMEP under lean burn operations. They have identified three major reasons for cyclic variations of IMEP namely the burning speed during initial stage of combustion, maximum fuel mass fraction burned and variation in the late burning during late expansion stroke.

Einewall & Johansson (2000) have studied the influence of spark gap and fuel injection strategies to improve lean burn limit & shown that combustion chamber geometry is a problem in lean burn engines. Slow burn combustion chamber, with low turbulence results in high CBC variations. Kowalewicz (2001) has proposed several modes of fuelling and testing on single cylinder engine to extend the lean operating limit fuelling with preheated gasoline, LPG & with liquid butane. Liquid butane showed better at lean burning limit. Lee & Kim (2001) have concluded that it is necessary to understand the combustion process & CBC variation in combustion to improve the stability & consequently to improve the fuel economy & emissions.

Villarroel (2004) has investigated the effects of cycle-to-cycle variations (ccv) on nitric oxide (NO) emissions with an engine simulation model. The result indicates that cyclic variations must be considered when calculating the overall NO emissions.

Zervas (2004) has determined the coefficient of variation (COV) of the in-cylinder pressure on each crank angle of a number of cycles, in the case of a natural gas feed SI engine operating under lean conditions.

Kaminski et al. (2004) have analyzed the experimental time series of internal pressure in a four cylinder spark ignition engine. They performed for different spark advance angles; apart from usual cyclic changes of engine pressure they observed oscillations. Results show that for a smaller spark advance angle the system is more deterministic. Blazek (2004) has described the problems of the combustion process in SI engine which is attribution cyclic variability of the combustion process, manifested by variations combustion pressure near the peaks of the pressure. This work describe to interpretation of the process burning & its variability in-cylinder pressure measurement.

Ceviz and Yuksel (2005) have investigated on a FIAT, 1.801dm^3, carbureted four cylinder spark ignition engine and showed that using ethanol–unleaded gasoline blends as a fuel decreased the coefficient of variation in indicated mean effective pressure, and CO and HC emission concentrations, while increased CO_2 concentration up to 10 % vol., ethanol in fuel blend.

Ceviz and Yuksel (2006) have investigated the use of liquefied petroleum gas (LPG) as a fuel for spark ignition engine in terms of lean operation, and focuses on the cyclic variations and exhaust emissions and showed that use of LPG decreased the coefficient of variation in the indicated mean effective pressure, and emission. They concluded that the higher laminar flame speed of LPG and good mixing of gaseous fuels with air causes a decrease in cyclic variations, and higher H/C ratio of LPG decreases the engine emissions.

Ma et al. (2008) has carried out the experiment to analyse effect of cycle by cycle variations on engine with addition of hydrogen and CNG fuel in SI engine. The results showed that coefficient of variations in both maximum pressure and indicated mean effective pressure could be reduced by hydrogen addition and that positive effect would be more obvious as the engine was further leaned out.

Litak et al. (2009) have analyzed the cycle-to-cycle variations of peak pressure Pmax and peak pressure angle αPmax in a four-cylinder spark ignition engine by examining the experimental time series of Pmax and αPmax for three different spark advance angles. Using standard statistical techniques such as return maps and histograms it has been shown that depending on the spark advance angle, there were significant differences in the fluctuations of Pmax and αPmax.

2.5 NO_X REDUCTION TECHNOLOGIES

Franz and Roth (2000) are carried out the experiment to find out the effect of hydrogen peroxide (H_2O_2) dissolved in water was injected into the combustion chamber of a direct injection diesel engine at different crank angles. The H_2O_2/water solution is injected by a second injection system into the engine combustion chamber. The HC concentration is increases significantly and the NO_X concentration is lowered. These effects stronger for pure water spray than for H_2O_2/water solutions.

Ozcan & Soylemez, (2005) are studied the effect of water addition on combustion in a conventional SI engine. The manifold induction method is used for water addition. The water induction is accomplished over a wide range of water to fuel mass ratios of 0.2 – 0.5. Result showed that water induction reduces the NO_X emission in the lean region having a local maximum between equivalence ratios of 0.9 and 1.0. Maximum of 35% reduction in peak NOx emissions was observed. The CO and HC emissions were slightly affected by different water addition rates.

Ozcan and Soylemez, (2005) are investigated experimentally the effect of water injection on thermal balance and performance in spark ignition engine. The results showed that as the water injection level to the engine increased work output, while the losses other than unaccounted losses decreased. Additionally, the specific fuel consumption decreases, while the engine thermal efficiency increases. The average increase in the brake thermal efficiency for a 0.5 water to fuel mass ratio is approximately 2.7% over the use of LPG alone for the engine speed range studied.

Subramanian et al. (2007), are observed one of the main problem with gaseous fuelled internal combustion engines is high NO level due to rapid combustion. They introduced manifold water injection technique to reduce the NO emission at high equivalence ratios. From the experiments carried out by the authors, the water injection leads to a significant reduction in NO levels. NO dropped from 7670 to 2490ppm with a water flow rate of 5.9 kg/hr. There is no such adverse effect of water injection on the brake thermal efficiency and small reduction in HC emissions is observed. Even though water injection is very effective in reducing NO emissions, it can certainly leads to many adverse effects like corrosion, contamination etc.

Benini et al. (2009) are investigated the numerical and experimental work to find benefits and drawbacks of both water (mist) and steam direct injection within the combustion chamber of a 200N static trust turbojet. The aim of the investigations is to evaluate the impact of increasing water and steam flows onto the emission levels (NO and CO) of the engine. Steam injection reduces NO emissions upto 16% (in terms of mass fraction) when a steam flow which doubles the fuel flow is introduced, whereas a reduction of about 8% is found using water injection in the same proportion of fuel flow. Steam injection is preferred to water injection when a reduction in the NO emissions is to be pursued while maintain relatively low CO emissions.

Larbi and Bessrour (2010) are carried out analytical model based on detailed chemical kinetics employed to calculate the pollutant emissions of a marine diesel engine with water injection. They noted down water injection decreases the temperature of the combustion zone by absorbing the vaporization of latent heat and thereby increasing his heat storage capacity. This reduction in temperature implies a lowering of the NO concentration. The CO_2 emission remains practically constant.

Tauzia, et al. (2010), describes an experimental study conducted on a modern high speed common-rail automotive Diesel engine in order to evaluate the effects on combustion and pollutant emissions of water injected as a fine mist in the inlet manifold. A comparison is made with exhaust gas recirculation to evaluate the potential of inlet water injection as an in-cylinder emissions reduction device for automotive application. They found that a large reduction of NO_X emissions can be achieved with high water injection rates, at low load as well as high load conditions. A water mass of about 60-65% of the fuel is needed to obtain a 50% NO_X reduction.

Chen et al. (2010) studied NO_X reduction strategy through aqueous alcohol injection in SI engine. Using alcohol in an engine upto 20% by volume will reduce NO emission by 30%, while the torque decreased about 10%.

Munsin et al. (2013) experimentally investigated performance and emission characteristics of small SI engine with hydrous ethanol containing high water content upto 40%. Authors found that increasing water content at constant load decreased overall efficiency and NOx emission, while BSFC, HC, CO, formaldehyde and acetaldehyde emissions were increased.

Cesur et al. (2013) experimentally conducted to find the effect of steam injection in the electronically controlled SI engine. Researchers suggested that optimum steam ratio has been determined as 20% of fuel mass (S20) in terms of performance and emission parameters. The experimental results showed that torque and the effective power increase up to 4.65% at 3200 rpm, specific fuel consumption reduces up to 6.44% at 2000 rpm. There is 40% average reduction in NO emissions at 2800 rpm and it is 31.5% in HC emissions at 2000 rpm.

Balki et al. (2014) performance, emission and combustion characteristics of different alcohol. The results show that the use of alcohol fuels increased the engine torque, brake specific fuel consumption (BSFC), thermal efficiency and combustion efficiency. In addition, the cylinder gas pressure and heat release rate occurred earlier;CO_2 emission increased while HC, CO and NOx emissions decreased.

Gonca (2014) investigated the effect of steam injection in diesel engine on performance & emission characteristics. Author suggested the 20% introduction of steam will has greater reduction in NO emission with slightly increased in torque.

2.6 SUMMARY OF LITERATURE REVIEW

After going through a thorough literature review it has been summarized that several works have been done with LPG as a short term alternative fuel for the gasoline engine. Review says that LPG as a fuel will give the positive effect on the exhaust gas emission except for NO_X emission. And also it will give good agreement with the performance and combustion characteristics. It has been reported that LPG has higher flame speed and higher octane number than gasoline. The emission using

LPG like CO and HC are very less compared to gasoline. Authors also told that by advancing the static ignition timing will give the better performance than gasoline at higher speed but NO_X emission will be three times that of gasoline. For the reduction of NO_X emission several techniques are used by the authors like EGR, SCR, Water injection etc. By using steam induction technology maximum of 55% reduction in the NO_X emission is obtained with decrease in the brake thermal efficiency.

2.7 RESEARCH GAP

After summarizing the literature review it has been found that limited research was carried out with the dual fuel (Gasoline with LPG) mode of operation in multi-cylinder engine. With dual fuel mode of operation studies related to the cylinder to cylinder variation are found scanty. Most of the previous studies are done in single cylinder engines. Work related to turbocharging in engine are limited with LPG as a fuel in SI engine. And also research for NO_X emission reduction using vaporized water-ethanol induction technology is limited. Work related to turbocharging with dual fuel mode of operation is limited. So this will lead to the defining the objectives of the research work for filling the research gap.

2.8 OBJECTIVES OF THE RESEARCH

A thorough review of the available literature on the use of LPG injection in SI engine and various NO_X reduction techniques is performed. However based on the literature it is found that studies related to the dual fuel mode of operation i.e., gasoline with LPG on multi-cylinder engine are scanty. Hence the present study deals with investigating the effect of dual mode of operation on engine performance, combustion and emissions in a modified multi-cylinder SI engine with turbocharging and vaporized water methanol induction method. The engine operating parameters of speed, load and static ignition timing are varied. The main purpose of this investigation is to optimize the gasoline LPG percentage for different load conditions with speed.

2.8.1 Specific Objectives of the Research Work:

1. To modify and study the performance, emission and combustion characteristics of the existing four cylinder MPFI SI engine with LPG and dual fuel (LPG + Gasoline) mode of operation at various load and speed, and to

optimize the LPG gasoline percentage (Percentage varied from 0, 25, 50, 75 and 100%) for different speed and load conditions.

2. To study the performance, emission and combustion characteristics of the four cylinder MPFI SI engine with LPG gasoline percentage for dual fuel (LPG + Gasoline) mode of operation at various load, speed and static ignition timing, and to optimize the static ignition timing for the dual fuel mode of operation. The emissions are to be sampled without any after treatment device.
3. To study the performance, emission and combustion characteristics of the four cylinder MPFI SI engine with turbocharging for optimized LPG gasoline percentage and static ignition timing at various load and speed conditions.
4. To study the cycle by cycle fluctuations in a turbocharged engine with various speed for the optimized condition at full load.
5. To develop a vaporized water methanol induction system in the intake manifold using waste exhaust heat from the engine. To study the effect of various proportions of vaporized water methanol to optimized condition for turbocharged engine on the performance, emission and combustion characteristics. Vaporized water methanol ratio is varied from 10 to 30% by volume.
6. To make a comparative study of the dual fuel mode of operation with turbocharging and vaporized water methanol induction system with baseline fuel (Gasoline) on engine performance, emission and combustion characteristics.

2.9. SCOPE

The experimental investigation is done on a 4-cylinder spark ignition MPFI engine with LPG and gasoline dual fuel mode of operation. Experiment carried out with different percentage of LPG through electronically controlled LPG ECU. Mass flow rate of LPG, engine speed and load conditions are varied and compared with baseline gasoline fuel. Static ignition timings are advanced to get optimized percentage of LPG on performance, combustion and emission characteristics basis. A separate turbocharger has been selected based on the exhaust gas energy and fitted into the engine. Necessary modification has been done in the exhaust and intake manifold. Finally vaporized water-methanol system has been developed for a turbocharged MPFI engine. The exhaust emissions are measured in real time with a AVL 444 analyzer, with the samples being taken as raw sample i.e., without any exhaust after treatment devices in between the sampling point and the engine exhaust manifold.

CHAPTER 3

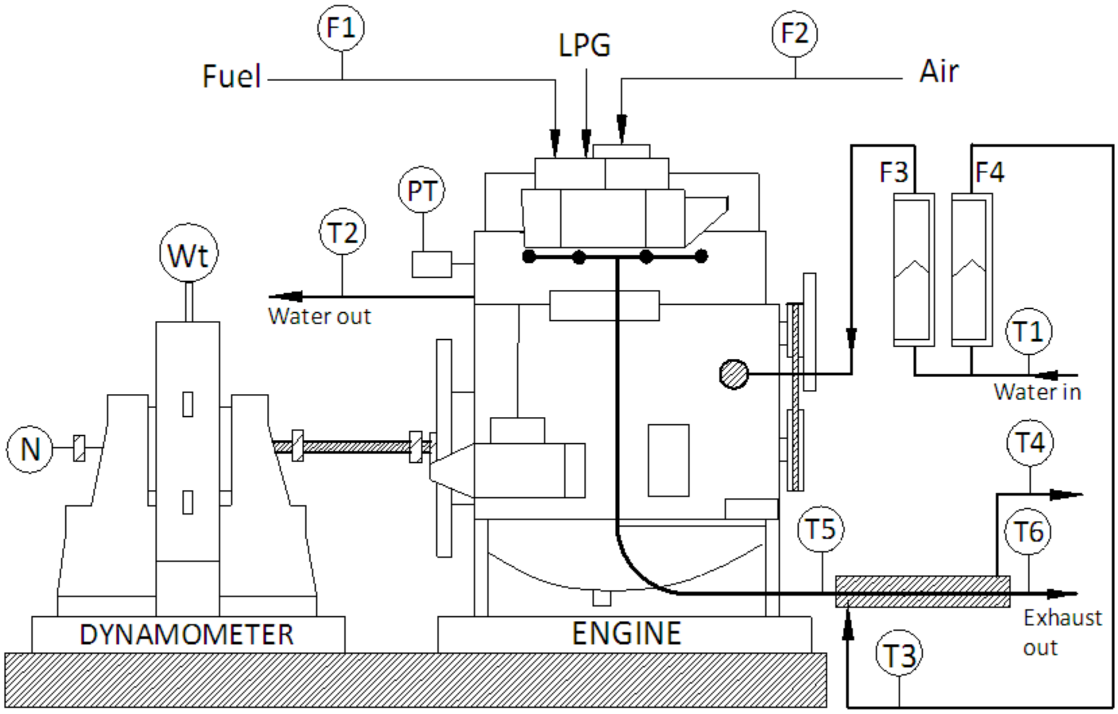
EXPERIMENTAL TEST RIG AND INSTRUMENTATION

The principles, instrumentation and modifications in the engine test rig followed during the course of research work are presented in this chapter. The modification in the experimental setup and scheme of experiments is meticulously planned in a manner to fulfill the objectives framed under present research. This chapter describes in detail engine test rig used for the research work with all instrumentation part. Necessary modification carried out in the engine setup also explained in this section with all particulars. Also measuring techniques used for different parameters are presented.

3.1 ENGINE TEST RIG

A 10L inline four cylinder engine of Maruti Suzuki Zen (max. power of 44.5 kW at 6000 rpm) with multi point port fuel injection (MPFI) system was used to acquire experimental data for this project. The engine has a single overhead camshaft layout with 4 valves per cylinder (2 intake and 2 exhaust). Fuel is injected into the intake port by a single fuel injector located in each intake runner. The engine is connected to an eddy current type dynamometer which is used to absorb power and regulate engine speed. The test rig is provided with necessary instruments for combustion pressure and crank-angle measurements, airflow, fuel flow, temperatures and load measurements. These signals are interfaced to a digital computer through an 8 channel engine interface. The set up has a stand-alone panel box consisting of air box, fuel tank, manometer, fuel measuring unit, differential pressure transmitters for air and fuel flow measurements, process indicator and engine indicator. Rotameters are provided for cooling water and calorimeter (for heat balance sheet) water flow measurement. Figure 3.1 shows the Schematic of the experimental setup, while plate 3.1 gives a view of the engine test rig.

The setup enables study of engine performance for brake power, indicated power, frictional power, BMEP, IMEP, brake thermal efficiency, indicated thermal efficiency, mechanical efficiency, volumetric efficiency, specific fuel consumption and air-fuel ratio (A/F). NI based USB-6210 data acquisition system engine performance and combustion analysis software package 'IC Engine Soft' is provided by the supplier of the test rig - M/s Apex Innovation Pvt. Ltd. Sangli, India, for on-line performance evaluation. The software also evaluate the combustion parameter for Heat Release Rate, Mass fraction Burned, In-cylinder pressure with cylinder volume for each crank angle deg. rate of pressure rise and mean gas temperature. The detailed specifications of the gasoline engine and other instrumentation mounted on the test-rig are given as Appendix I.



F1- Fuel Flow Differential Pressure (DP) unit
F2- Air Intake DP unit
F3- Rotameter (Engine)
F4- Rotameter (Calorimeter)
T1- Cooling water inlet temperature
T2- Cooling water outlet temperature
T3- Calorimeter water inlet temperature
T4- Calorimeter water outlet temperature
T5- Exhaust gas inlet temperature
T6- Exhaust gas outlet temperature
N – rpm decoder
PT- Pressure transducer
Wt – Load on Dynamometer

Fig.3.1. Schematic of the experimental setup

Plate 3.1 Engine setup with control panel

3.2 MODIFICATION OF THE ENGINE SETUP FOR OPERATION WITH LPG

The engine is modified to operate with liquefied petroleum gas (LPG) fuel. Separate four gas injectors are attached to the inlet manifold near the inlet port of each cylinder for injecting LPG. The gas injectors are operated by solenoid valves driven by 12V DC power supply. The nozzle diameter of each gas injector is determined based on the power output per cylinder and for the given engine nozzle diameter of 1.75 mm is used. A separate gas ECU has been used for driving the solenoid valves and the signals from the gas ECU controls the activation period of the gas injectors. The after-market LPG injection system manufactured by M/s Europe gas (Auto gas v 3.1 LPG) is used. The block diagram of the LPG injection system is shown in the Fig. 3.2. Domestic LPG stored in cylinders at a pressure of about 4-8 bar (Max. Vapor Pressure at 40° C is 1050 kPa gauge) which weighs 14.2 kg is used in the experiments. An unreduced pressure regulator is fitted to LPG cylinder which allows the gas to pass through it at high pressure. A copper pipe is used to supply the LPG from the cylinder to the vaporiser. An electromagnetic strainer is provided in the supply line to absorb the iron particles from the LPG cylinder which may travel along LPG. Power supply from the battery is given to activate the electromagnet. A LPG vaporiser which is provided in the supply line serves the purpose of vaporizing the fuel and supplies it at the required pressure. The function of the vaporiser is to transfer thermal energy into the LPG and to reduce the LPG (tank) pressure to the much lower system pressure such that the LPG evaporates to the superheated gas phase (Price et al. 2004). The thermal energy required is supplied by the engine cooling water which is made to pass through the vaporiser after the engine jacket circulation. To supply the required amount of gas at the required pressure for meeting the various load conditions a reference pressure from the engine inlet manifold is also connected to the evaporator.

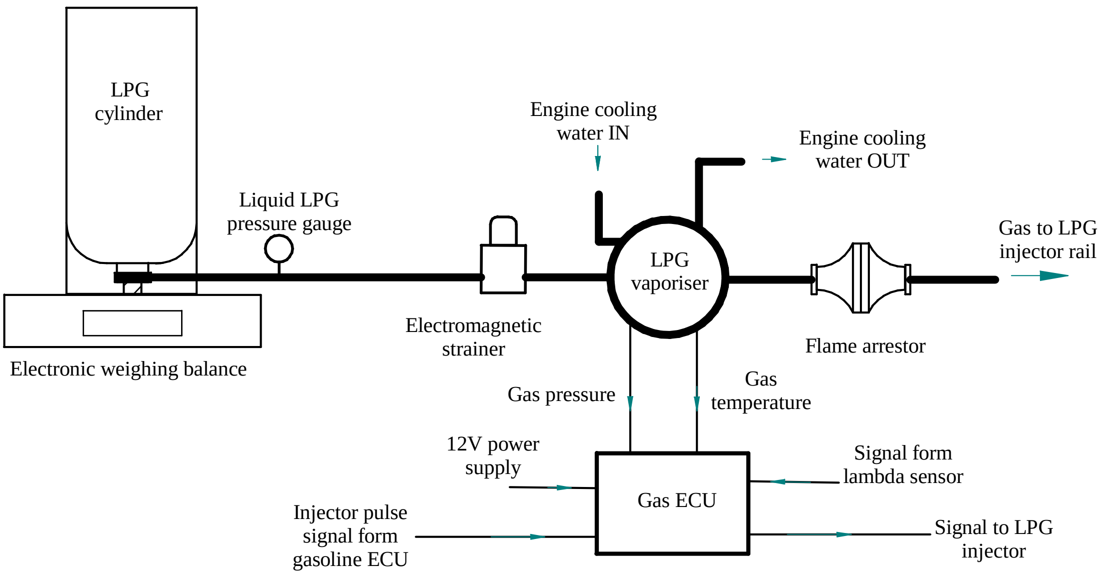


Fig.3.2. Block diagram of LPG injection system.

3.2.1 LPG Engine Control Unit (Gas ECU)

The function of the gas ECU is to open and close the gas injector at appropriate time to control the duration of injection. Concept of working of gas ECU for bi-fuel application is based on master slave theory (Khatri et al. 2009). A sequential gas injection controller of IV generation OSCAR-N OBD CAN is used. The gasoline injector opening signal pulse from the pre-installed Gasoline ECU is fed to gas ECU as an input. The gas ECU modifies the gasoline pulse width using a correction factor and sends it to gas injectors. This correction factor is calculated based on the density of liquid gasoline and gaseous LPG. It also takes into account the signals from the other sensors such as exhaust lambda sensor (indicating oxygen content in the exhaust gases) and inlet manifold absolute pressure indicating the engine load. When the engine is running with LPG as fuel, the emulator system in the gas ECU cuts off gasoline injection signals and gives the emulated signal to the gasoline ECU so that it doesn't give a fault signal. A switch provided in the control panel is use to switch between LPG and gasoline fuel operation. The specifications of the gas ECU is given as Appendix II.

3.2.2 Safety Measures and Flame Arrestor

Gaseous fuels are difficult to handle compared to liquid fuels and thus they are considered to be more dangerous than liquid fuels. Hence at most care should be taken while handling the gaseous fuels. LPG is mainly a mixture gaseous fuel of propane and butane. Both these fuels have very low ignition temperature normally in the range of 40^0C. So the contingencies of auto ignition and thus explosion are rather more here. Also the flames may propagate back into the pipeline which may trigger the fuel in the gas cylinder to explode. To avoid flash back, a flame arrestor is connected in series in the fuel supply line. From the flame arrestor LPG is passed to the gas injector rail. Flame arrestors are the equipment's which quench flames that are propagating back to the cylinder. They prevent the propagation of flame from the exposed side of the unit to the protected side by the use of wound crimped metal ribbon type flame cell element called as Honeycomb. This construction produces a matrix of uniform openings that are carefully constructed to quench the flame by absorbing the head of the flame. This provides an extinguishing barrier to the ignited vapor mixture. Under normal operating conditions the flame arrestor permits a relatively free flow of gas or vapor through the piping system. If the mixture is ignited and flame begins to travel back through the piping, the arrestor will prohibit the flame from moving back to the gas source. Plates 3.2 and 3.3 respectively show the gas injectors and the flame arrestor used in the LPG system.

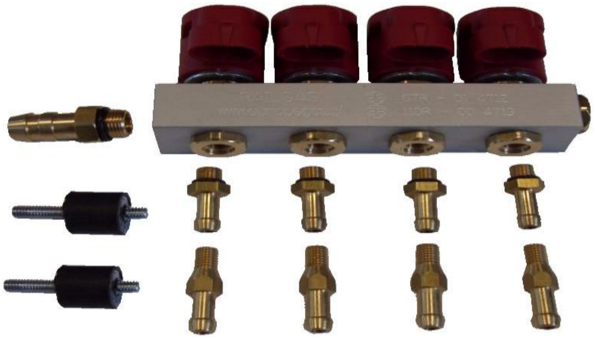


Plate 3.2 Gas injectors

Plate 3.3 Flame Arrestor

The leakage in LPG pipeline can happen in two ways. One such chance is through the injector to the cylinder and other one is through any leaks in the pipeline. The safety measure to avoid this leakage is by conducting the periodic leak checking of both the injector and the pipelines. The new injector may not have any leakage problems but as the time progress due wear and tear of the parts, the injectors may be subjected to some leakage problems. These leakages may be of very small quantity but are sufficient enough to auto ignite. Since the LPG gas molecules are denser than the air, it will settle down inside the cylinder and during cranking it may auto ignite and thus causes trouble. Use of ordinary pipes increases the chances of leakage of gases to the atmosphere which may enhance the chances for LPG to mix with the air. This may even led to hazardous explosions. Hence seam-less copper pipes are used to avoid the chances of leakage of gas through the pipes to the atmosphere.

3.3 TURBOCHARGER SETUP AND INSTALLATION

The schematic diagram of turbocharged MPFI engine is shown in figure 3.3. Turbocharger consists of turbine and compressor. Exhaust gas energy is used to run the turbine. A wastegate regulates the exhaust gas flow that enters the exhaust-side driving turbine and therefore the air intake into the manifold and the degree of boosting. Depending upon the required output pressure the compressor, exhaust gas energy is used through waste gate. Excess exhaust gas will goes in the exhaust line. Intake air pressure is measured interms of pressure gauge. The boost pressure is maintained around 0.25 bar. The compressed air temperature is more than room temperature, so intercooler (heat exchanger) is used to bring down the temperature of the compressed air. The warmer the intake air, the less dense, and the less oxygen available for the combustion event, which reduces volumetric efficiency. Not only does excessive intake-air temperature reduce efficiency, it also leads to engine knock, or detonation which is destructive to engines. The pressure ratio is maintained 1.25. The pressure difference between the intake manifold and atmosphere is measured in terms of mercury U-tube manometer. Through waste gate, the intake pressure is controlled. Plate 3.4 and 3.5 shows the turbocharger setup in an engine.

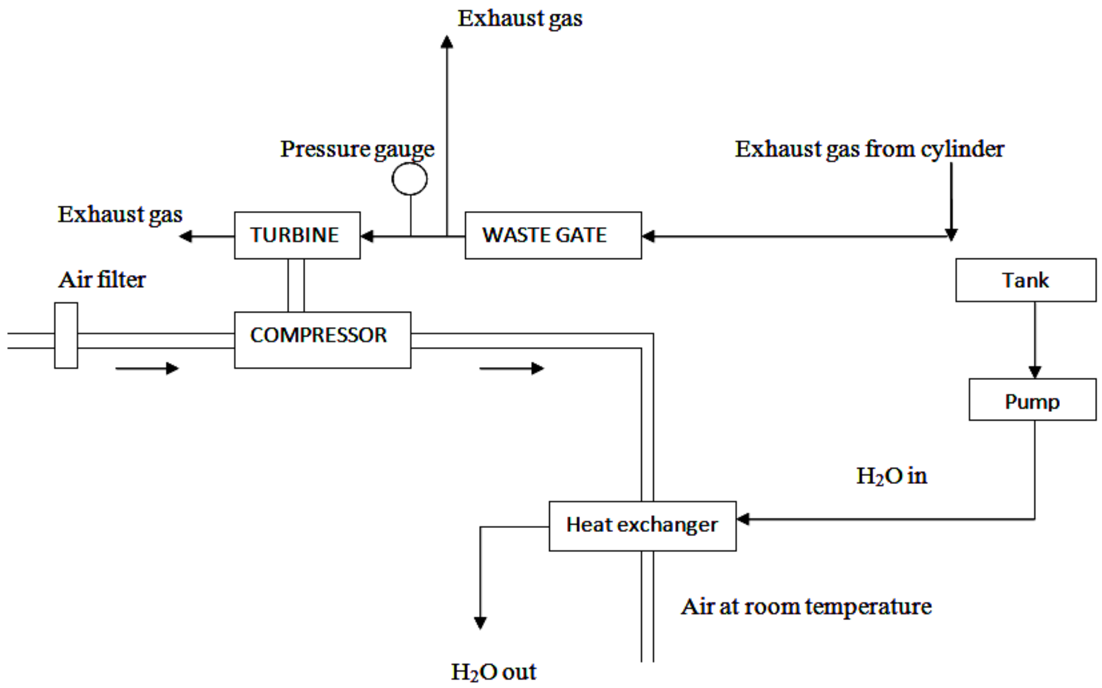


Fig. 3.3. Schematic diagram of turbocharger setup for engine.

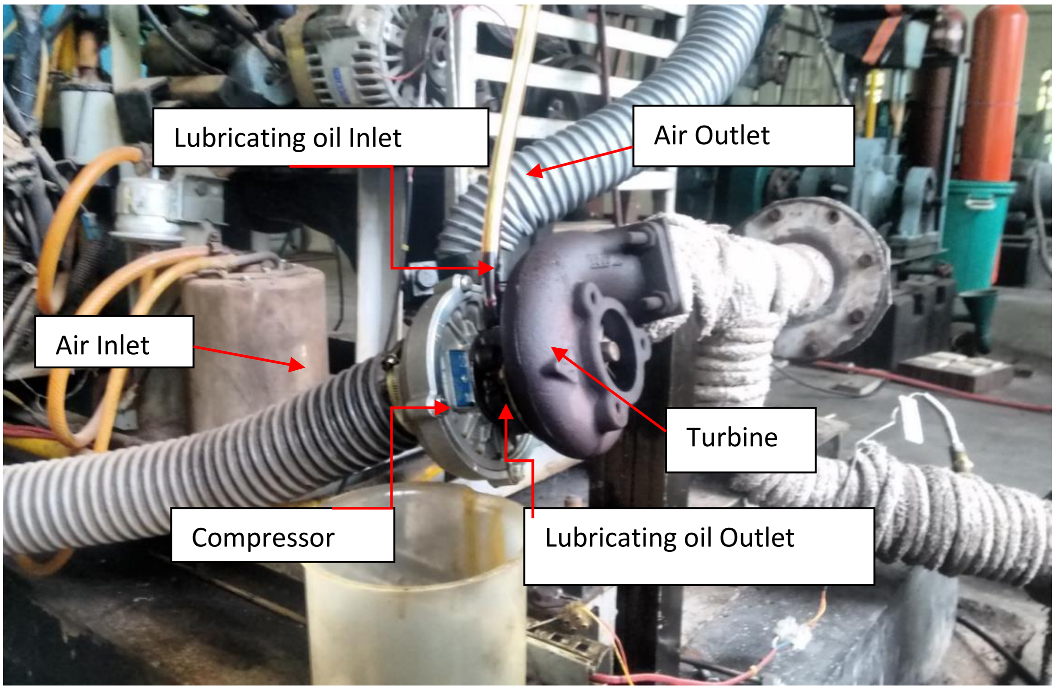


Plate 3.4 Turbocharger setup

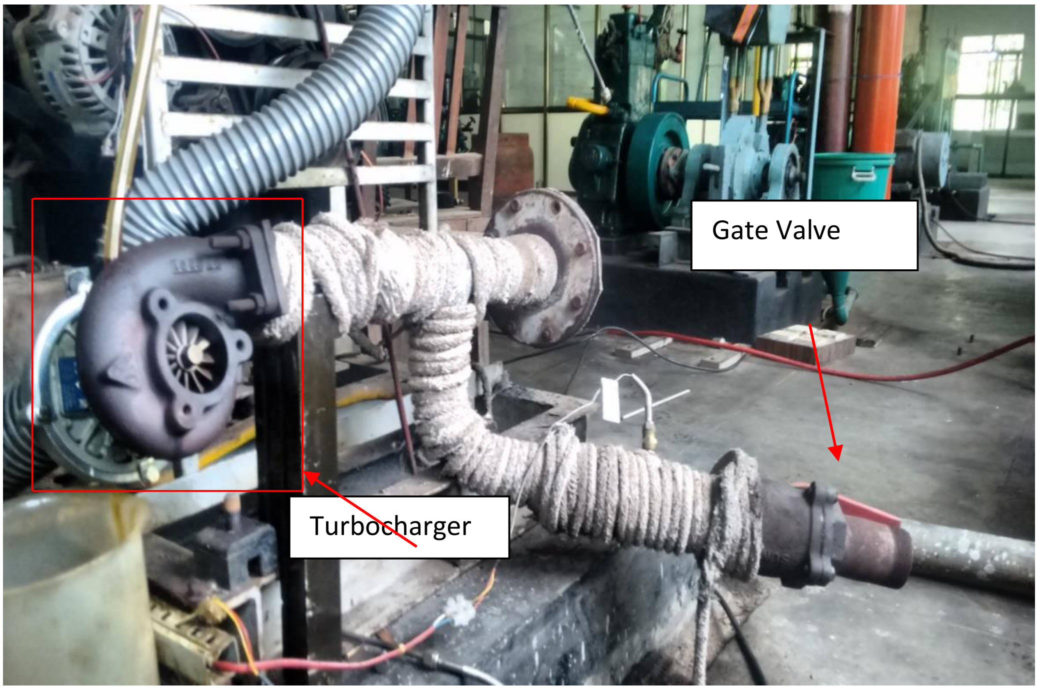


Plate 3.5 Turbocharger setup with gate valve

3.4 VAPORIZED WATER-METHANOL INDUCTION SYSTEM DEVELOPMENT

LPG combustion in SI engine results in higher emissions of harmful NO_X even though the emissions of CO and HC are reduced substantially. Use of after treatment devices may reduce the NO_X emissions. In this work a method of reduction of NO_X in the engine itself is described. Among several in cylinder NO_X reduction techniques, the method of vaporized water-methanol induction with intake air is used by developing a device to supply vaporized water-methanol at various proportions. The vaporized water-methanol is produced with the help of heat of engine exhaust gases, which is otherwise is lost to the surroundings. De-ionized water along with methanol in the proportion of 60:40 is stored in a container and a low power pump is used to pass the water through a copper tube of $1/4^{th}$ inch size. The copper tube is coiled around the engine exhaust pipe such that the water-methanol mixture and exhaust gases pass in a counter flow way so as to maximize the heat transfer between the two fluids. Since the flow rate of the available pump was higher than the required water flow rate (max. 3 liters per hour), a bypass system is provided after pump so that the excess water returns back to the sump. Sufficient length of copper coiling is provided so that the water-methanol will be completely vaporized and vapor is inducted in to the engine manifold. The amount of vaporized water-methanol to be inducted is decided based on the LPG fuel flow rate at each operating condition. Before admitting vaporized water-methanol in to the engine manifold at each operating condition with a specific vaporized water-methanol flow rate, it is ensured that liquid form is completely converted into vapor form. Fig.3.4 represents the schematic block diagram of the vaporized water-methanol induction setup.

Vaporized water-methanol to LPG fuel mass ratios of 0.1, 0.20 and 0.3 are used at each operating condition. A provision is made in the inlet manifold just before the throttle valve to induct vapor continuously to ensure good mixing of vaporized water-methanol with the intake air. The water-methanol flow rate is measured and controlled manually by a rotameter of range 0.1 lph to 10 lph with a least count of 0.1 lph. Plates 3.6, 3.7 and 3.8 detail the various systems. Plate 3.6, 3.7 and 3.8 shows entire vaporized water methanol induction system with copper coil and intake manifold.

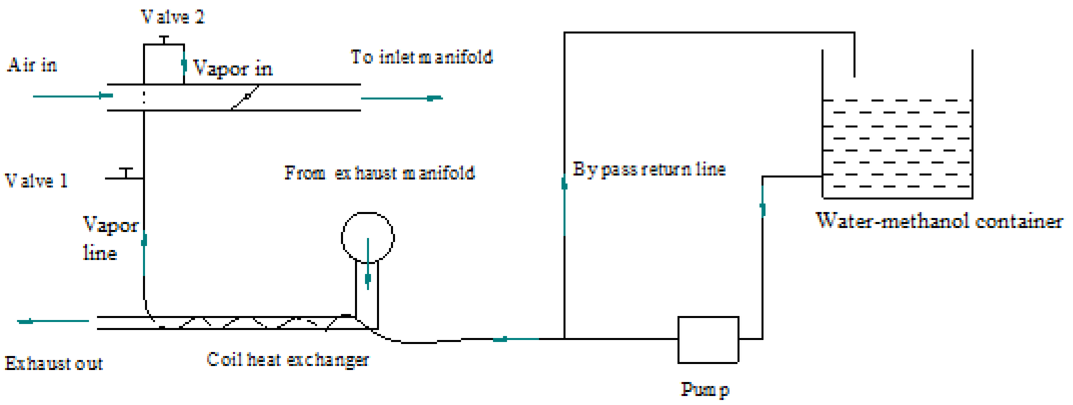


Fig. 3.4 Block diagram of vaporized water-methanol induction system

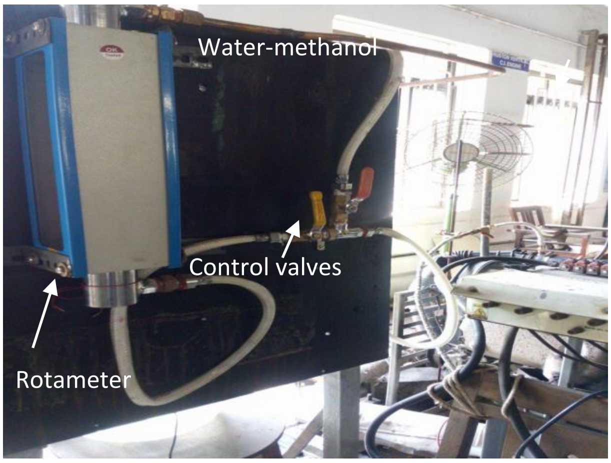


Plate 3.6. Vaporised water-methanol induction system

Plate 3.7 Heat exchanger for production of vapor of water-methanol.

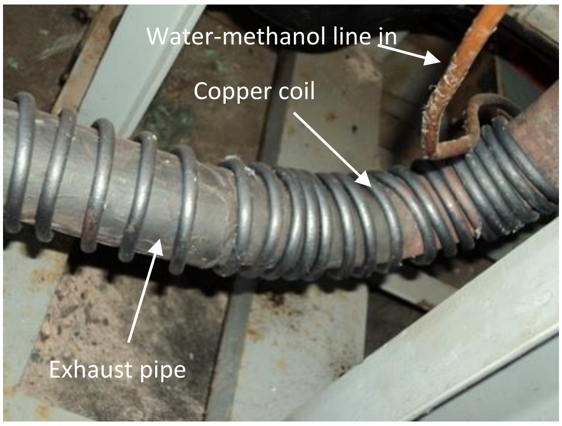


Plate 3.8 Copper coils to the exhaust pipe for the production of vaporised water-methanol

3.5 MEASUREMENT SYSTEM

The test bed is fully instrumented to measure the different parameters during the experiments on the engine. A detailed description of the different measurement systems used for evaluating the engine performance and emission is given in this section.

3.5.1 Cylinder Pressure measurement

A piezo-electric pressure transducer is used for recording the cylinder pressure for number of consecutive cycles for combustion variability studies. A PCB Piziotronics Inc, built piezoelectric pressure transducer is installed in the engine cylinder head of 1st cylinder. The sensor is flush mounted and it measures the pressure trace in the cylinder with 1degree crank-angle resolution. The pressure crank-angle data is acquired on a digital computer operating on windows 8.1 system through National Instrument (NI) based data logger NI USB 3210. An 'IC Engine Soft' software is installed to get required data from different sensors through NI USB 3210 data logger. The software provided is capable of data logging a maximum of 100 consecutive combustion cycles. The sensor body is continuously circulated with cooling water so as to maintain the sensor at a constant temperature. A rotary encoder is fitted on the engine output shaft for crank angle signal. Both signals are simultaneously scanned by an engine indicator (electronic unit) and communicated to computer. The software in the computer draws pressure crank-angle and pressure volume plots and computes indicated power of the engine and other combustion properties.

3.5.2 Air and Fuel Flow Measurements

As the air flow during engine suction is pulsating, for satisfactory measurement of air consumption an air box of suitable volume fitted with orifice is used for damping out the pulsations. The differential pressure across the orifice is measured by water manometer and pressure transmitter. The flow across the orifice is connected via a parallel section to the U- tube manometer and the air intake differential pressure (DP) unit. The DP unit senses the pressure difference across the orifice, which is sent to the transducer. The transducer gives a proportional output as DC voltage (analog signal), which is converted into digital signal which will be in turn processed by the computer software program to get the air flow rate in mm of water column and kg/h.

The fuel consumed by the engine is measured by determining the volume flow of the fuel in a given time interval and multiplying it by the density of the fuel. A glass burette having graduations in ml is used for volume flow measurement. Time taken by the engine to consume this volume is measured by stopwatch manually. Alternatively a differential pressure transmitter working on hydrostatic head principles is used for fuel consumption measurement. The fuel tank is connected to a burette for manual fuel flow measurement and to a fuel flow DP transmitter unit. The fuel line is connected to a two-way fuel cock which can be kept either in tank position or measuring position. When kept in measuring position, the fuel to the engine goes from the burette. The pressure head difference is sensed by the fuel DP transmitter. It gives proportional analog signal, which through the NI USB 3210 hardware goes to the 'IC Engine soft' which calculates the fuel flow rate in kg/h. It is essential to enter the values of density and the lower calorific value of each test fuel to the software 'IC Engine soft' before operating with that test fuel.

LPG flow rate is measured on the mass basis, to minimize the error in measurement while operating the system under varying pressures. An electronic weighing balance of 100 kg capacity with a least count of 10 gram and a stopwatch is used to measure the flow rate of LPG. For the experiment, the LPG cylinder is placed in upside down direction, so that LPG will be flowing in liquid form to the vaporizer which can provide the required amount of fuel.

3.5.3 Engine Speed Measurement

Engine speed is sensed and indicated by an inductive pickup sensor in conjunction with a digital rpm indicator, which is a part of the eddy-current dynamometer controlling unit. The dynamometer shaft rotating close to inductive pickup rotary encoder sends voltage pulse whose frequency is converted to rpm and displayed by digital indicator in the control panel, which is calibrated to indicate the speed directly in number of revolution per minute.

3.5.4 Load Measurements

The brake load is measured by an eddy current dynamometer. It consists of a stator on which a number of electromagnets are fitted and a rotor disc coupled to the output shaft of the engine. When rotor rotates eddy currents are produced in the stator due to magnetic flux set up by the passage of field current in the electromagnets. These eddy currents oppose the rotor motion, thus loading the engine. These eddy currents are dissipated in producing heat so that this type of dynamometer needs cooling arrangement. Regulating the current in electromagnets controls the load. A moment arm measures the torque with the help of a strain gauge type load cell mounted beneath the dynamometer arm. The analog load cell signal through the ADC card is fed to the computer to give load in kg. The dynamometer is loaded by the dynamometer loading unit situated in the control panel.

3.5.5 Temperature Measurement

Chromel-Alumel thermocouples connected to digital panel meter are positioned at different locations to measure the following temperatures: jacket water inlet temperature (T_1), jacket water outlet temperature (T_2), inlet water temperature at calorimeter (T_3), outlet water temperature at calorimeter (T_4), exhaust gas temperature before calorimeter (T_5) and exhaust gas temperature after calorimeter (T_6). All sensors, which sense the temperatures of respective locations, are connected to the control panel, which gives the digital reading of the respective temperatures. These are also interfaced to the computer through NI hardware.

3.5.6 Static Ignition Timing Measurements

The device used to measure the static ignition timing is ignition timing gun. This ignition timing illuminates the light when engine is running and keep the light on the flywheel which is having the scale for the measurement of ignition timing. The flywheel has the scale of 10 divisions with least count 2 deg. bTDC. To change the static ignition timing, the ignition distributor assembly is loosened and is rotated slightly in the direction of rotation of flywheel to retard the timing and in the direction opposite of rotation of flywheel to advance the timing.

3.5.7 Exhaust Emission Measurements

An AVL Digas 444 five gas Exhaust gas analyzer is used to measure the various exhaust emissions of carbon monoxide (CO, %volume), carbon dioxide (CO_2, %volume), unburnt hydrocarbons (HC, ppm) oxygen (O_2, % volume) and oxides of nitrogen (NO_X ,ppm), the five gas analyzer is calibrated by the supplier AVL prior to the use and necessary precautions are taken to see the proper working of it by regular check up of all types of filters and cleanliness of probe was maintained. Leakage test and zero adjustments are done regularly. The engine test rig has no catalytic converter, and thus the emission readings taken are raw emissions, without being treated. The specifications of the analyzer are given as Appendix III. All instruments are calibrated prior to their use in the tests.

CHAPTER 4

METHODOLOGY AND EXPERIMENTAL PROCEDURE

In this chapter scheme of the experimental work is presented in order to fulfill the research objectives. Scheme and methodology of the research work framed carefully and divided into four phases. Initially engine tests are conducted with dual fuel mode of operation with different percentages of LPG for different speed and load conditions at factory set static ignition timing. In the second phase of investigation, experimental research carried out with different advanced static ignition timings to study performance and emission characteristics. During this stage LPG percentage and static ignition timings are optimized. In the third phase of the experiment, engine setup is modified to operate with turbocharging for the optimized condition. In the final phase of the research work, a vaporized water-methanol induction system for the engine is developed to reduce the emissions from the engine.

4.1 SCHEME OF ENGINE EXPERIMENTAL STUDIES

The engine experimental study involved four distinct stages. The first stage of the experiment involves the steady state engine performance, combustion and emission characteristics evaluation with LPG-gasoline dual fuel mode of operation. During this stage LPG-gasoline ratio is varied from 0%, 25%, 50%, 75% and 100%. During this first stage, experiment will be carried out with varying speed from 2000 rpm to 4500 rpm in steps of 500 rpm and varying load 25%, 50%, 75% and 100% of full load with set static ignition timing of 5 deg. bTDC. For the conversion of dual fuel mode operation gasoline injector should be switch off by giving the faulty signal and switch on the LPG injector, so that cylinder will work in LPG mode of operation and remaining will work in gasoline mode. In this stage cycle by cycle fluctuations studies are also carried out. At this stage LPG-gasoline ratio is optimized for factory set static ignition timing.

In the second stage of experiments, the static ignition timing is varied from 5 deg. bTDC to 8 and 11 deg. bTDC to analyze performance and emission characteristics of different LPG-gasoline ratio. During this stage speed and load conditions are varied similar to the first stage of experiment. In this stage static ignition timing is advanced because LPG is gaseous fuel, having higher auto-ignition temperature compared to gasoline. In this LPG-gasoline ratio and static ignition timing are optimized with respect to speed and load conditions. The scheme of the experiment for first and second stage i.e., dual fuel mode of operation with varying static ignition timing are shown in the figure 4.1.

To check the static ignition timing, an "ignition timing gun" (timing light) is used. It is connected to the battery positive and negative terminals. Another probe of the timing gun is hooked to the cable connected to the spark plug of the first cylinder. The engine is started and kept at static condition for 2-3 minutes. Now the timing gun is used to illuminate the pulley connected to the engine flywheel and the static ignition timing can be read on a scale with the help of timing gun light.

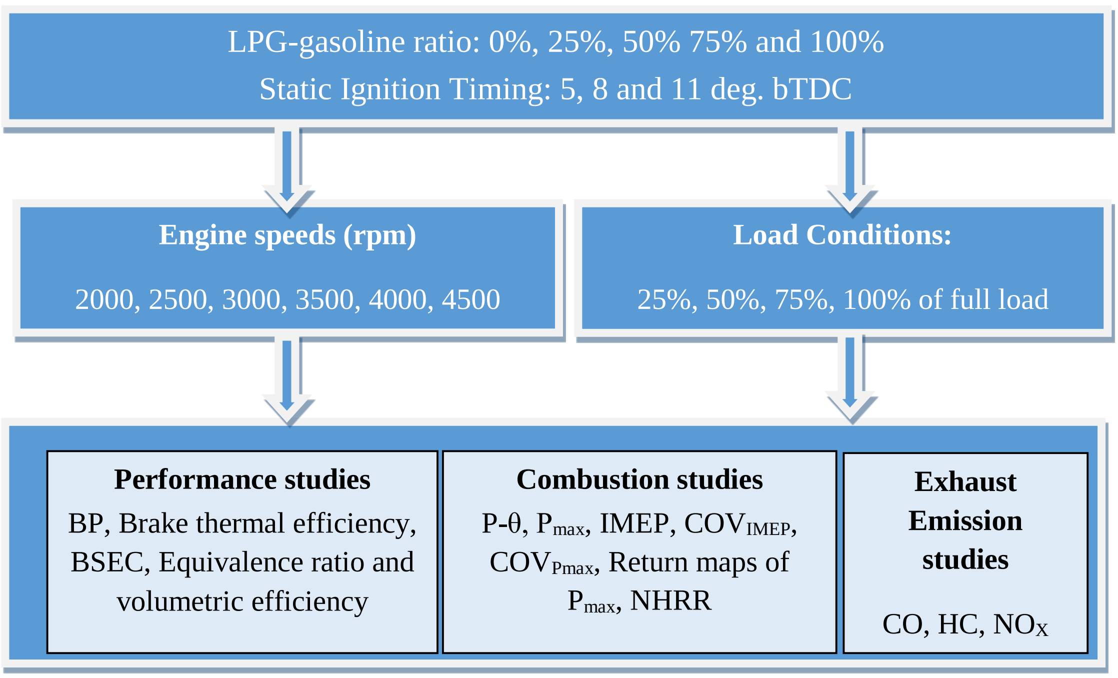


Fig. 4.1 Scheme of experiments with varying LPG-gasoline ratio and static ignition timing.

The recorded pressure-crank angle data for 100 consecutive cycles are used for calculating the indicated mean effective pressures (IMEP), COV_{IMEP}, COV_{Pmax}, and net heat release rate (NHRR). Analysis of the obtained data is performed and results are plotted.

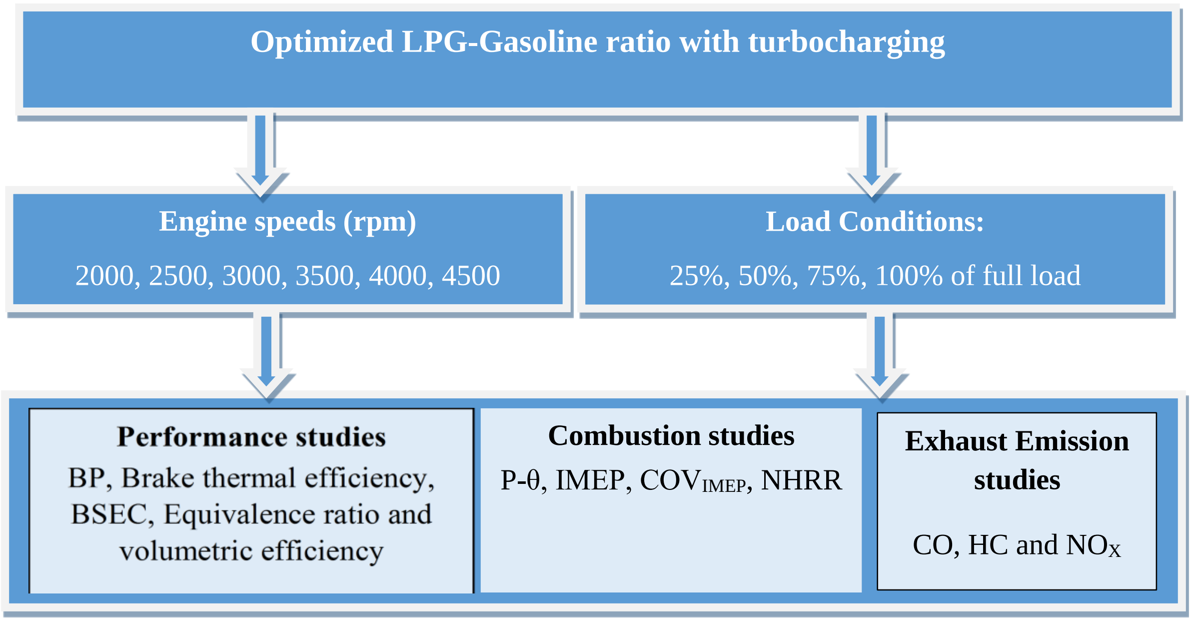


Fig 4.2 Scheme of experiments for turbocharging

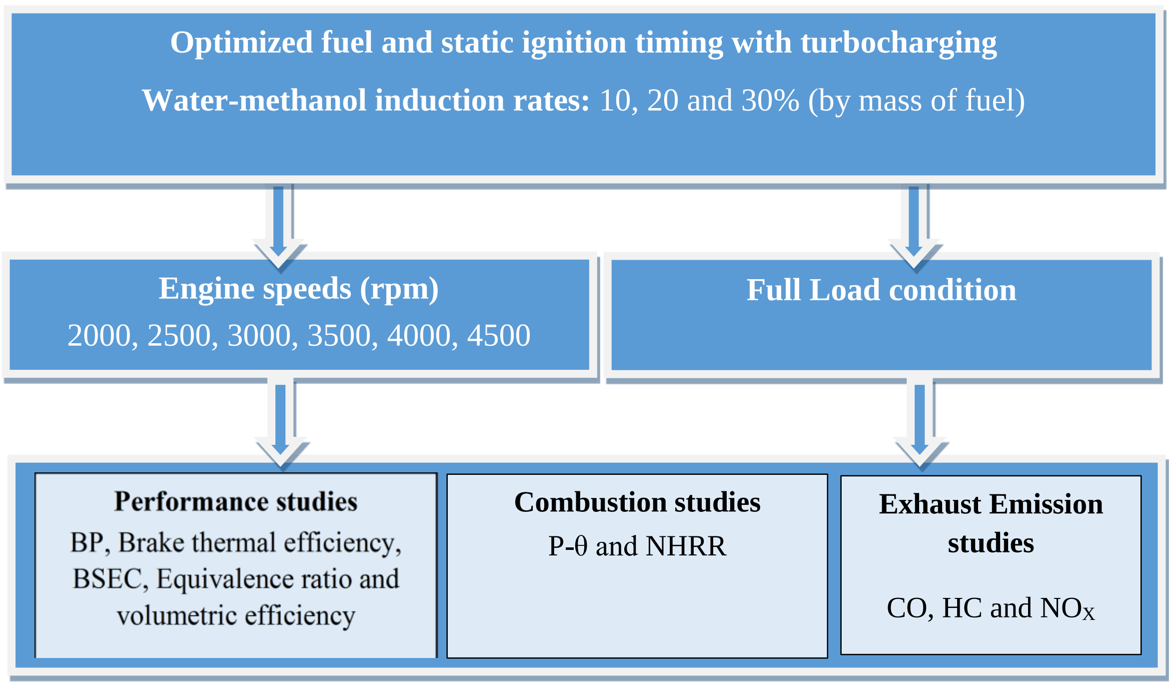


Fig 4.3 Scheme of experiments vaporized water-methanol induction system.

In the third stage the engine testing is performed with turbocharging for optimized static ignition timing and LPG-gasoline ratio with various operating parameters of speed, and load conditions. In this stage of experiment, a turbocharger is selected based on the exhaust mass energy, and fitted into the engine with necessary modification in the exhaust and intake manifold. For this optimized LPG-gasoline ratio experiments are carried out with turbocharger to investigate performance, combustion and emission characteristics. Also cycle by cycle fluctuation studies are carried out for 100 cycles. In this stage speed is varied from 2000 rpm to 4500 rpm in steps of 500 rpm and load varying load 25%, 50%, 75% and 100% of full load for optimized LPG-gasoline ratio and static ignition timing. The scheme of the experiment for turbocharger operation is shown in the figure 4.2.

In the last stage of the engine testing is performed with gaseous LPG injection with vaporized water-methanol induction. Vapor of water-methanol mixture is produced from using waste heat from exhaust gases. Precisely measured water at rates of 10, 20 and 30% by mass of LPG is converted in to vapor form and is inducted in to the intake air stream. In this stage speed is varied from 2000 rpm to 4500 rpm in steps of 500 rpm at full load condition for turbocharged, optimized LPG-gasoline ratio and static ignition timing. The scheme of the experiment for turbocharger operation is shown in the figure 4.3.

4.2 DETERMINATION OF IMEP AND COV$_{IMEP}$

The area enclosed by the p-v diagram of an engine gives the indicated work done by the gas on the piston. The IMEP is a measure of the indicated work output per unit swept volume, a parameter independent of the size and number of cylinders in the engine and engine speed.

IMEP is defined as:

$$IMEP = \frac{W_i}{V_s} \tag{4.1}$$

where W_i is the indicated work in Newton metres

V_s is the swept volume per cylinder in cubic metres

The IMEP can be computed by experimental pressure & volume data for a 0-720° crank angle by the following equation (Brown 2001):

$$IMEP = \frac{\Delta\theta}{V_s} \sum_{i=n_1}^{i=n_2} P(i) \frac{dV(i)}{d\theta} \tag{4.2}$$

P(i) is cylinder pressure at crank angle I in Pascals

V(i) is cylinder volume at crank angle I in cubic metres

V_s is cylinder swept volume in cubic metres

n_1 is BDC induction crank angle

n_2 is BDC exhaust crank angle

The software uses the pressure crank-angle history for the determination of the indicated mean effective pressure (IMEP). When logging consecutive combustion cycles the software calculates the average IMEP. The coefficient of Variation (COV) is the standard deviation in IMEP divided by the mean IMEP (Ceviz and Yuksel 2006), and is usually expressed in percent. It is defined as

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\mu_{IMEP}} \tag{4.3}$$

Where σ is the standard deviation and μ is the mean value.

The standard deviation is given by

$$\sigma = \sqrt{\frac{\sum_1^n (IMEP_i - \mu_{IMEP})^2}{(n-1)}} \tag{4.4}$$

Where n= Number of combustion cycles.

4.3 HEAT RELEASE RATE

In-cylinder combustion pressure data is very useful information, which could be used to quantify the combustion behavior of the fuels inside the engine. Traditionally, engineers, during the engine design and optimization process, perform in-cylinder pressure measurements to determine peak pressure, rate of change in pressure, estimated rate of heat release, mass-burned fraction, and the charge temperature. Rate of heat release analysis shows the estimated rate of heat release during the combustion process. The results provide a quantified assessment of combustion rate and the means to diagnose combustion process (Catania et al. 2001). Heat release analysis is generally applied to compression ignition engines, although there is no reason why it cannot be used in spark ignition applications. Heat release analysis computes how much heat would need to have been added to the cylinder contents, in order to produce the observed pressure variations. In the present work, an effort is made to determine a single zone heat release rate and combustion temperature in a SI engine, using experimentally obtained average pressure-crank angle data. Heat release rate is computed for 100 consecutive combustion cycles at every test point.

4.3.1 Thermodynamics of heat release

The heat-release analysis is carried out within the framework of the first law of thermodynamics when the intake and exhaust valves are closed, i.e. during the closed part of the engine cycle. The simplest approach is to regard the cylinder contents as a single zone, whose thermodynamic state and properties are modeled as being uniform throughout the cylinder and represented by average values. The basis for the majority of the heat-release models is the first law of thermodynamics, i.e. the energy conservation equation. For an IC engine, the cylinder contents are a single open system. The First Law of Thermodynamics as applied to this case is given by:

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \sum_i \dot{m}_i h_i = \frac{dU}{dt} \tag{4.5}$$

Where Q is the heat transferred in Joules, p is the pressure in pascals, V is the volume m^3, m_i is the mass of fuel injected, h_i is the enthalpy in J/kg and U is the internal energy in J. Since the only mass crossing the system boundary is the fuel injected, the mass-enthalpy term reduces to a "mass of fuel enthalpy" term. Assuming that the enthalpy and internal energy are sensible terms (using a baseline of 298 K) and that the net heat released defined as the difference between the energy released through combustion and the energy lost to heat transfer from the system walls, the equation 4.5 can be rewritten as:

$$\frac{dQ}{dt} = p\frac{dV}{dt} + \frac{dU}{dt} \tag{4.6}$$

The heat transfer dissipated through the system boundary presents a problem only at the end of combustion where temperatures have risen. If we further assume that the contents of the cylinder can be modeled as an ideal gas, then the equation 4.6 can be rewritten as:

$$\frac{dQ}{dt} = p\frac{dV}{dt} + mC_v\frac{dT}{dt} \tag{4.7}$$

Where C_v is the specific heat at constant volume. Differentiation of the perfect gas law with R assumed constant provides a means of eliminating the temperature term which is generally unavailable in pressure analysis to give

$$\frac{dQ_{Net}}{dt} = \left(1 + \frac{c_v}{R}\right) p \frac{dV}{dt} + \left(\frac{c_v}{R}\right) V \frac{dp}{dt} \tag{4.8}$$

Substituting the specific heat ratio , provides the final equation used in the analysis with the result being equally valid when substituting the independent variable , or crank angle, for time, t, the net heat release combustion model of Krieger and Borman is obtained (Eyidogan et al, 2010).

$$\frac{dQ_{Net}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \tag{4.9}$$

where γ is the ratio of specific heats, Q_{net} is the net heat release rate in Joules per degree, p is the in-cylinder pressure in Pascals, V is the in-cylinder volume in cubic metres.

The in-cylinder heat transfer occurs by both convection and radiation, where convection constitutes the major part. Heat is transferred by both convection and radiation occurring between in-cylinder gases and cylinder head, valves, cylinder walls, and piston during the engine cycle. By taking into account the effects of heat transfer to the cylinder walls, the gross heat release can be calculated as follows:

$$\frac{dQ_{Gross}}{d\theta} = \frac{dQ_{Net}}{d\theta} + \frac{dQ_{ht}}{d\theta} \tag{4.10}$$

The mean charge temperature T for the single-zone model is found from the state equation $pV = mRT$, assuming the total mass of charge m_c and the mass specific gas constant R to be constant. These assumptions are reasonable since the molecular weights of the reactants and the products are essentially the same. If all thermodynamic states (p_{ref}, T_{ref}, V_{ref}) are known or evaluated at a given reference condition such as Inlet Valve Close (IVC), the mean charge temperature T is computed as

$$T = p.V.\frac{T_{ref}}{p_{ref}.V_{ref}} \tag{4.11}$$

The cylinder volume at IVC is computed using the cylinder volume given in the above equation for θ_{IVC} and is therefore considered to be known. The two other states at IVC (P_{IVC}, T_{IVC}) are considered unknown and have to be estimated.

The rate of pressure rise is calculated using a simple numerical differentiation:

$$\frac{dp}{d\theta} = \frac{p_{i+1} - p_{i-1}}{\theta_{i+1} - \theta_{i-1}} \tag{4.12}$$

Once the rate of heat release rate is computed for 25 consecutive combustion cycles for a given test point, the peak heat release rate HRR_{max} is computed for every combustion cycle. The HRR_{max} of 25 cycles are used to compute coefficient of variation of HRR_{max} (COV_{HRRmax}).

$$COV_{\text{HRRmax}} = \frac{\sigma_{\text{HRRmax}}}{\mu_{\text{HRRmax}}} \tag{4.13}$$

Where σ is the standard deviation and μ is the mean value.

The standard deviation is given by

$$\sigma = \sqrt{\frac{\sum_1^n(\text{HRRmax}_i - \mu_{\text{HRRmax}})^2}{(n-1)}} \tag{4.14}$$

Where n= Number of combustion cycles.

4.4 ERROR AND UNCERTAINTY ANALYSIS

Error is associated with various primary experimental measurements and the calculations of performance parameters. Errors and uncertainties in the experiments can arise from instrument selection, condition, calibration, environment, observation, reading and test planning. Uncertainty analysis is needed to prove the accuracy of the experiments.

The uncertainty in any measured parameter is estimated based on Gaussian distribution method with confidence limit of ±2σ (95.45% of measure data lie within the limits of ±2σ of mean). Thus uncertainty of any measured parameter is given by:

$$w_i = \frac{2\sigma_i}{\bar{x}} \times 100 \tag{4.15}$$

Experiments are conducted to obtain the mean ($\bar{x}$) and standard deviation (σ_i) of any measured parameter (x_i) for a number of readings. This is done for speed, load, time for a specified amount of air and fuel flow etc. For the analysis, 20 sets of readings are taken at the same operating condition. The uncertainty values for speed, load, air flow rate, fuel flow rate, exhaust gas temperature and emissions of NO_X, HC, CO are calculated using equation (4.15).

A method of estimating uncertainty in experimental results has been presented by Kline and McClintock (1953). The method is based on careful specifications of the uncertainties in the various primary experimental measurements. Suppose a set of measurements is made and the uncertainty in each measurement may be expressed with the same odds. These measurements are then used to calculate some desired results of the experiments. The uncertainty in the calculated result can be estimated on the basis of the uncertainties in the primary measurements.

If an estimated quantity R depends on 'n' independent measured parameters $x_1, x_2, x_3, \ldots, x_n$. Then R is given by

$$R=R(x_1, x_2, x_3, \ldots, x_n) \quad (4.16)$$

Let w_R be the uncertainty in the result and $w_1, w_2, \ldots, w_n$ be the uncertainties in the independent measured parameters. R is the computed result function of the independent measured parameters $x_1, x_2, x_3 \ldots x_n$ as per the relation $x_1 \pm w_1, x_2 \pm w_2, \ldots x_n \pm w_n$). If the uncertainties in the independent variables are all given with the same odds, then the uncertainty in the result having these odds is given as (Adnan et al. 2012):

$$w_R = \left(\left[\frac{\partial R}{\partial x_1} w_1\right]^2 + \left[\frac{\partial R}{\partial x_2} w_2\right]^2 + \ldots\ldots + \left[\frac{\partial R}{\partial x_n} w_n\right]^2\right)^{1/2} \quad (4.17)$$

Using the equation (3.29) for a given operating condition, the uncertainties in the computed quantities such as mass flow rates of air and fuel, brake power, brake thermal efficiency are estimated. The estimated uncertainty values at a typical operating condition are given table 4.1.

Table 4.1 Uncertainty values of various parameters.

Sl. No.	Parameter	Uncertainty (%)
1	Speed	±0.25
2	Torque	±0.32
3	Air flow rate	±1.05
4	Fuel flow rate	±0.81
5	Exhaust gas temperature	±0.50
6	NO_X emission	±5.91
7	HC emission	±5.50
8	CO emission	±3.77
9	Brake power	±0.3
10	Brake thermal efficiency	±0.1
11	Volumetric efficiency	±0.4

4.5 DETERMINATION OF STOICHIOMETRIC A/F FOR GASOLINE, BUTANE AND PROPANE

Air contains both oxygen and nitrogen, if sufficient oxygen is available, a hydrocarbon fuel can be completely oxidized. The carbon in the fuel is then converted to carbon dioxide (CO_2) and the hydrogen to water (H_2O). When the products are at low temperatures the nitrogen is not significantly affected by the reaction. Consider the complete combustion of a general hydrocarbon fuel of average molecular composition C_aH_b with air. The overall complete combustion equation is

$$C_aH_b + (a + b/4)\,[O_2 + 3.76N_2] \rightarrow aCO_2 + (b/2)H_2O + 3.76\,(a + b/4)N_2 \qquad (4.18)$$

Equation (1) defines the stoichiometric proportions of fuel and air; i.e., there is just enough oxygen for conversion of all the fuel into completely oxidized products. The stoichiometric air/fuel depends on the fuel composition. Because the composition of the combustion products is significantly different for fuel-lean and fuel-rich mixtures, and because the stoichiometric air/fuel ratio depends on fuel composition, the ratio of the stoichiometric air/fuel ratio to the actual ratio (or its inverse) is more informative parameter for defining mixture composition, which is called as equivalence ratio ϕ.

$$Equivalence\ ratio\ \phi = \frac{(A/F)_{stoichiometric}}{(A/F)_{actual}} \qquad (4.19)$$

Calculation of stoichiometric A/F:

For gasoline:

$C_8H_{18} + 12.5\,[O_2+3.76N_2] \rightarrow 8CO_2 + 9H_2O + 3.76\times12.5N_2$

Stoichiometry A/F ratio = (mass of air)/(mass of fuel)

$(A/F)_{stoichiometric}$ = 12.5× [2×16+3.76×2×14] / (12×8+18) **= 15.05**

For propane:

$C_3H_8 + 5\,[O_2+3.76N_2] \rightarrow 3CO_2 + 4H_2O + 3.76\times5N_2$, $(A/F)_{stoichiometric}$= **15.6**

For butane:

$C_4H_{10} + 6.5\,[O_2+3.76N_2] \rightarrow 4CO_2 + 5H_2O + 3.76\times6.5N_2$,

$(A/F)_{stoichiometric}$ = **15.39**

For LPG (60% butane and 40% Propane):

$4\,C_3H_8 + 6\,C_4H_{10} + 59\,[O_2+3.76N_2] \rightarrow 36\,CO_2 + 46\,H_2O + 3.76\times59\,N_2$

$(A/F)_{stoichiometric}$= **15.46**

CHAPTER 5
RESULTS AND DISCUSSION

A four cylinder, multipoint fuel injection engine has been modified to work on LPG injection system with turbocharger and vaporized water-methanol induction system. A set of experiments has been done on the engine operated at six engine speed conditions for four load conditions with varying static ignition timing to analyze the performance, combustion and emission characteristics of engine. The results and discussion chapter is divided into four segments. The first segment deals with the study of LPG-gasoline dual fuel mode of operation with different percentage of LPG at factory set static ignition timing. In this section performance, combustion cycle by cycle variations and emission characteristics are analyzed. In the second segment static ignition timing are advanced and compared with factory set static ignition timing. After this stage LPG percentage and static ignition timing are optimized. In the third segment turbocharger is fitted in the engine with necessary modification to study performance, combustion cycle by cycle variations and emission characteristics for optimized condition. In the last segment effect of vaporized water-methanol induction to performance, combustion and emission characteristics on turbocharged engine is analyzed.

5.1 EFFECT OF LPG-GASOLINE DUAL FUEL MODE OF OPERATION

In this section effect of different percentage of LPG in MPFI 4-cylinder SI engine on performance, emission, combustion and cycle by cycle variations are analyzed for different speed and load conditions.

5.1.1 Performance Characteristics

Performance characteristics involves the study of parameters like brake power (BP), brake thermal efficiency (BTE), equivalence ratio, volumetric efficiency and brake specific energy consumption (BSEC) on engine speed and load conditions.

5.1.1.1 Brake Power

The following figures 5.1 to 5.4 show the variations of brake power with the engine speeds at different load conditions. At quarter load condition, torque and mean effective pressure decreases more rapidly with increasing speeds. This may be attributed to the reduced air flow in to the cylinder as the throttle area is reduced. The pumping component of total friction also increases as the engine is throttled thus decreasing the mechanical efficiency (Heywood 1998). This has resulted in reduced engine power output at higher speeds at quarter load for all the fuels. However 100% LPG produces comparatively higher BP at speeds above 2500 rpm compared to gasoline. Since at higher engine speeds, flames generated by LPG propagate faster than that of gasoline. Hence LPG generates more power output than that of gasoline. BP will be maximum for 25% LPG at 4500 rpm. This might be due to higher consumption of fuel as seen from figure 5.17 & 5.18 at higher speed which leads to higher BP.

At all loads, the engine generates constant torque for a particular throttle valve opening position. Hence brake power increases linearly with engine speeds. The influx of fuel is more in wide open throttle compared to part throttles. This results in the higher power generation in the higher load condition for all the fuels. But for 50% and 75 % of LPG and gasoline, as the speed increases beyond 3500 rpm the power output going to be reduced for all the load conditions. At full load 4500 rpm, for 50%LPG there is decrease in the power output of 1.1kW compared to gasoline. This indicates the load carrying capacity of the engine decreases with mixture of LPG with gasoline. Average decreased in power output for 50%LPG when compared to gasoline is 3.5% at full load, 6.1% at 75% of full load and 7% at 50% of full load at 4500 rpm. This decrease in the power output might be due to severe cycle by cycle variation characteristics in the cylinder for the mixture of fuel. Figure 5.5 shows the effect of torque on engine speed.

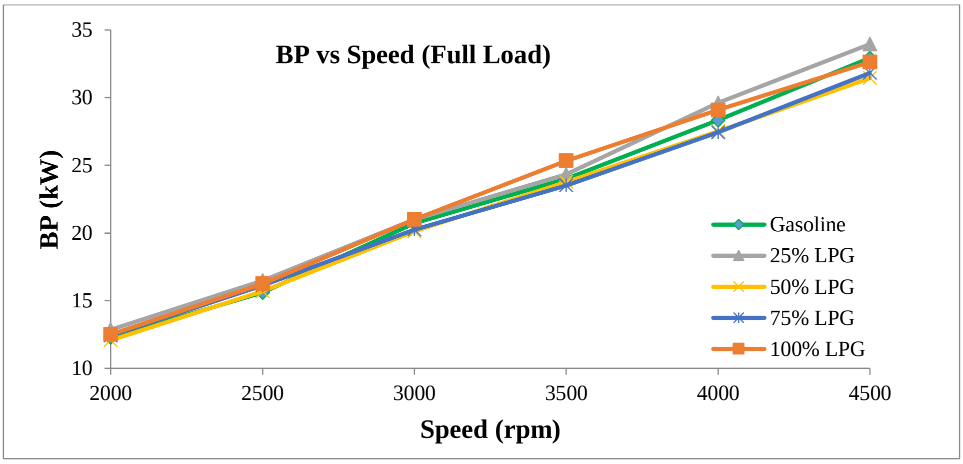


Fig. 5.1 BP vs Speed at full load

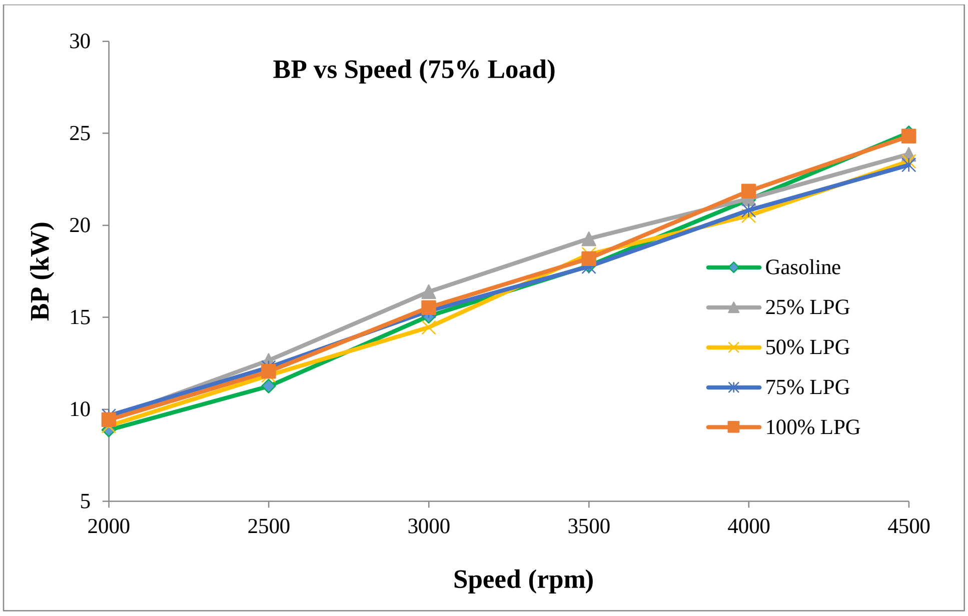


Fig. 5.2 BP vs Speed at 75% load

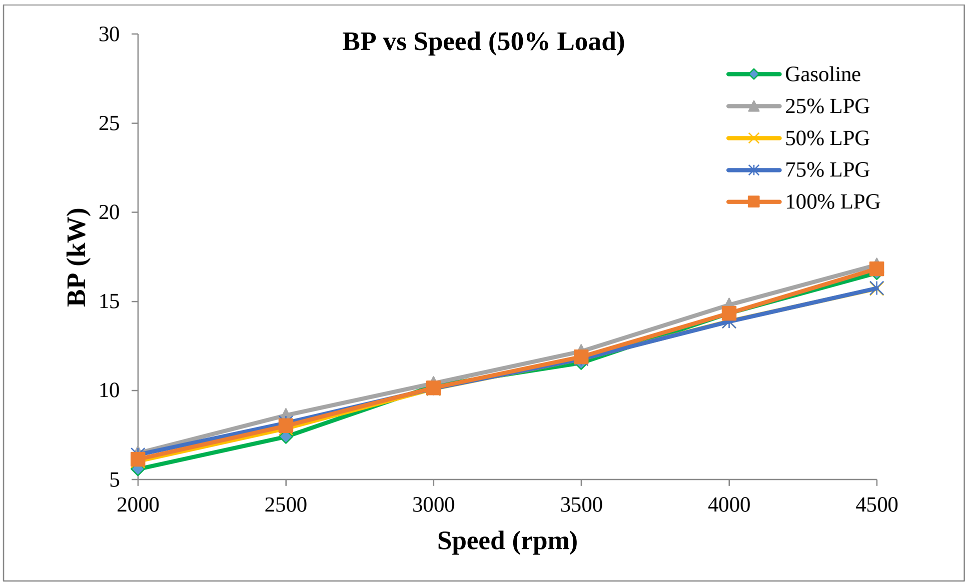


Fig. 5.3 BP vs Speed at 50% load.

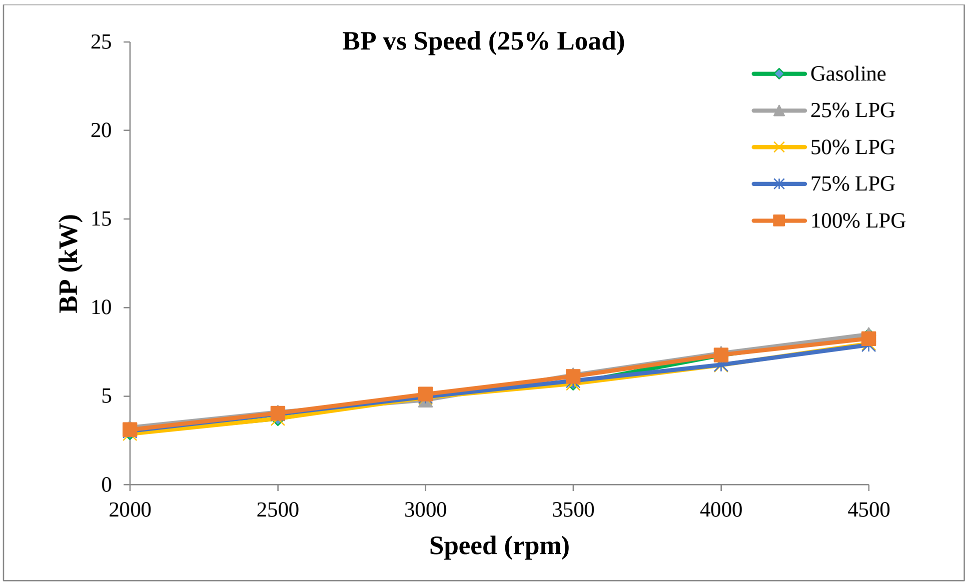


Fig. 5.4 BP vs Speed at 25% load.

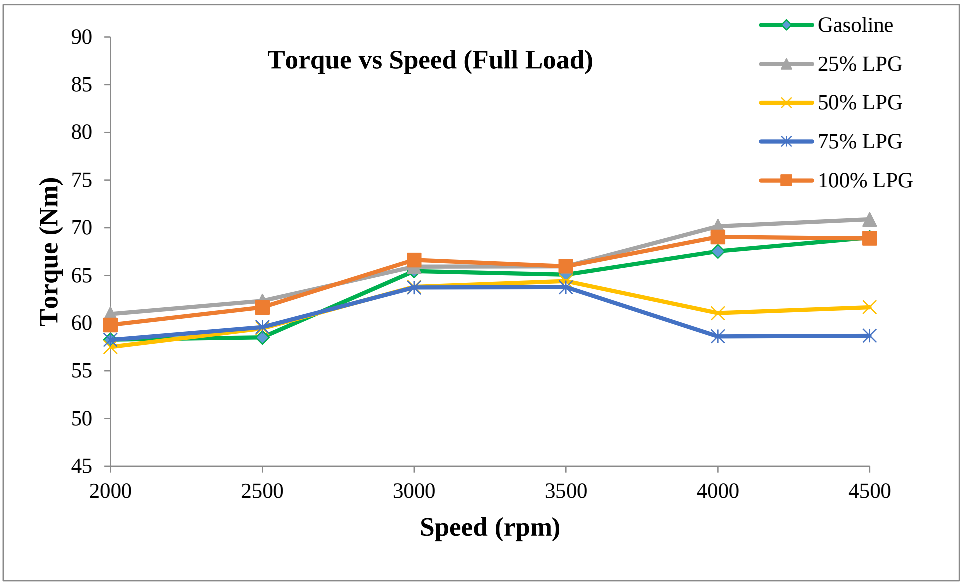


Fig. 5.5 Torque vs Speed at full load.

5.1.1.2 Brake Thermal Efficiency

The variations of Brake thermal efficiency for LPG, gasoline and dual fuel mode at all the operating conditions are shown from Figure 5.6 to 5.9.

The characteristics of these curves shows that the brake thermal efficiency increases with increase in speed but after 4000 rpm it decreases for all the fuel at all load conditions and it is maximum for 75% of full load condition. This is because at

75% load condition engine will work in the economy zone where fuel consumption is taken utmost care. At full load condition power output is more important than fuel economy. At higher engine speeds, the higher flame propagation speed of LPG negates the effect of ignition temperature. Here the time duration for each cycle is very low which demands more rate of combustion to get the complete combustion of the fuel. The lower propagation speeds of gasoline flames cannot afford the requisite combustion rate; instead the engine takes more fuel to generate the required torque. The collective outcome of these factors lowers brake thermal efficiency of the engine for gasoline at higher engine speeds.

The lean operation decreases the flame speed and the burning rate, and the reduction in burning rate results in an increase in the overall combustion duration. Since the ignition temperature of LPG is higher than the gasoline, ignition delay and thus combustion duration is more for LPG (Ceviz et al. 2005). Since there is a decreases in the average burning rate, to accommodate this effect engine consumes more fuel which in turn decreases its efficiency. Hence 100% LPG has lower efficiency than gasoline. Also volumetric efficiency of LPG is lesser than gasoline at higher load condition which will lead to reduction in the efficiency. During dual fuel mode of operation LPG having higher flame speed and gasoline having lesser flame speed, due to this combustion duration will decreases when compared to 100% LPG which will lead to reduction in fuel consumption at higher load condition. Hence 50% LPG is showing higher Brake thermal efficiency compared to gasoline at 100%, 75% and 50% load conditions. Beyond 50% increase in the LPG percentage leads to decrease in the efficiency due to increase in combustion duration. Brake thermal efficiency is increased by percentage average of -4.3% for 100% LPG, 2.1% for 50% LPG, 0.16% for 25% LPG, and 1.2% for 75% LPG at 75% of full load condition. But for the part load condition 50%LPG is giving the comparable results with petrol and better than LPG fuel because if increment in the volumetric efficiency and at this condition engine is working nearly in stoichiometric region. At 75% load there is not much difference in brake thermal efficiency between 50%LPG and gasoline but at 50% load there is 2% increase in average value of brake thermal efficiency for 50%LPG from 2000 to 4000 rpm when compared to neat gasoline.

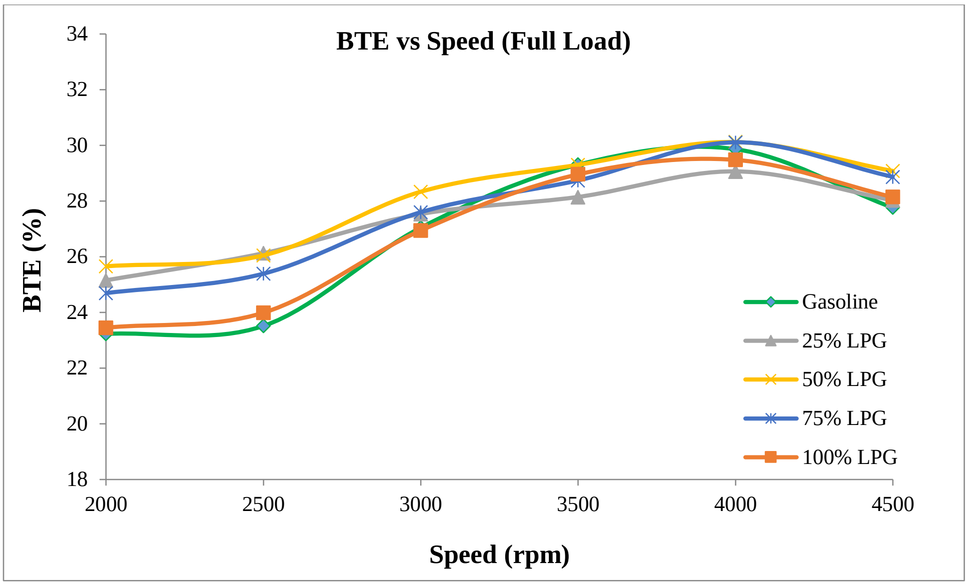


Fig. 5.6 BTE vs Speed at full load

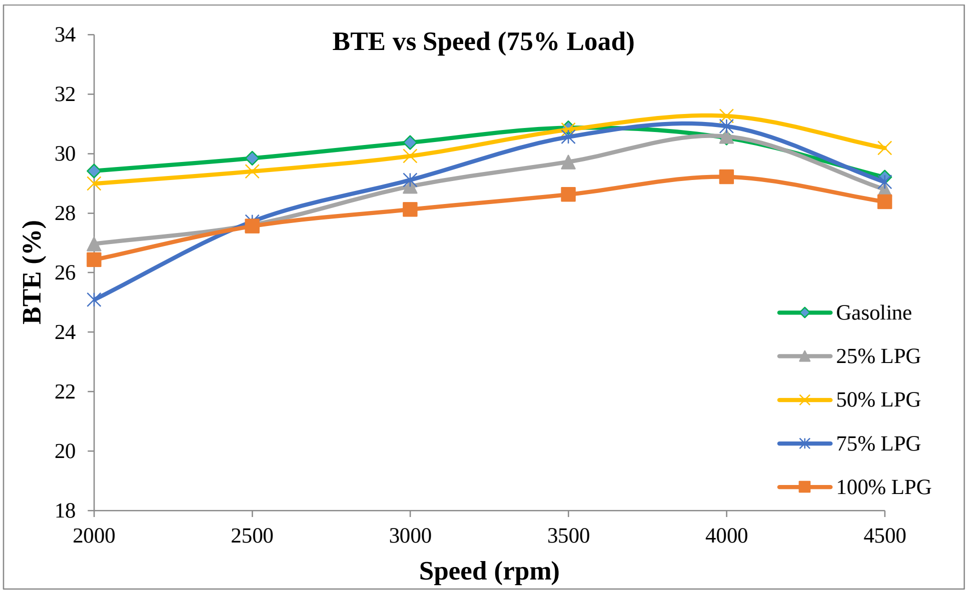


Fig. 5.7 BTE vs Speed at 75% load

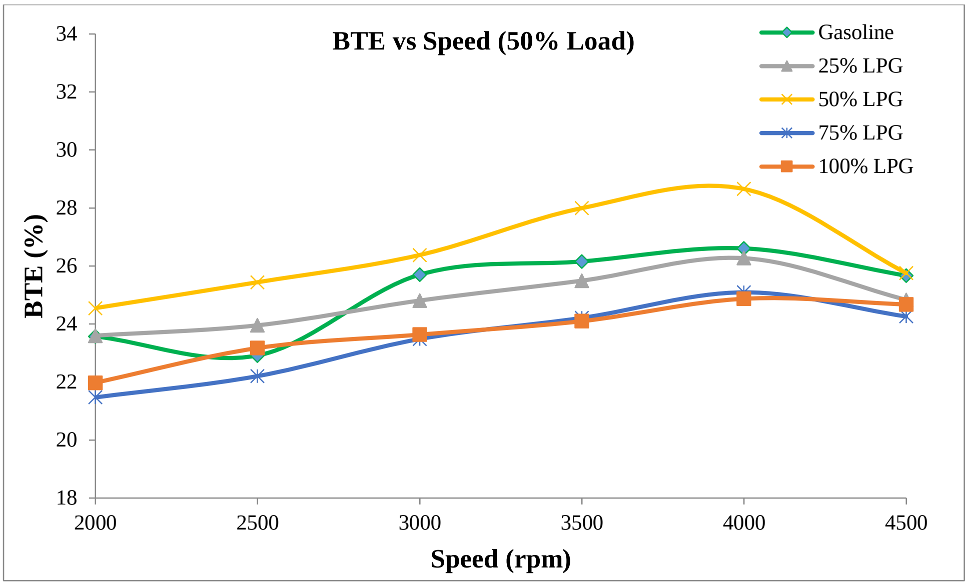


Fig. 5.8 BTE vs Speed at 50% load

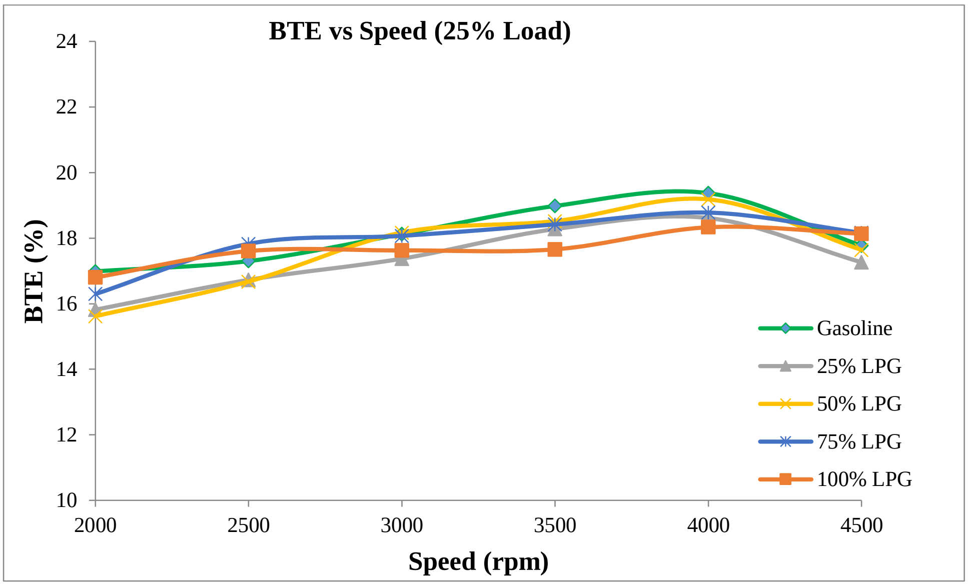


Fig. 5.9 BTE vs Speed at 25% load

5.1.1.3 Equivalence Ratio

The figures 5.10 to 5.13 indicate the variations of equivalence ratio of LPG, gasoline and dual fuel operation at various operating conditions. This equivalence ratio will determine the weather engine is working in leaner, stoichiometric or richer mixture strength. As the engine speed increases the equivalence ratio will try to remains in same value. In LPG fuelled engine combustion will take place in leaner region so as the speed increases quantity of fuel will increase so it will reach near stoichiometric region. But at full load since quantity of fuel going to be more LPG will work in slight richer region. Whereas for gasoline it will also work in richer region for all load condition since it is in liquid form and higher density quantity of fuel going inside is always higher. At quarter load both gasoline and LPG will work in richer mixture because of the quantity of air going inside is less compared to fuel due to restriction in passage from throttle valve.

In dual fuel mode of operation 50%LPG will always work near stoichiometric region whereas other two combinations is working in richer region for all the load and speed conditions. This is because for this fuel mixture, quantity of air going inside that is volumetric efficiency is more when compared to that of fuel hence it will work in a stoichiometric region. But at higher load it is working in slightly richer region.

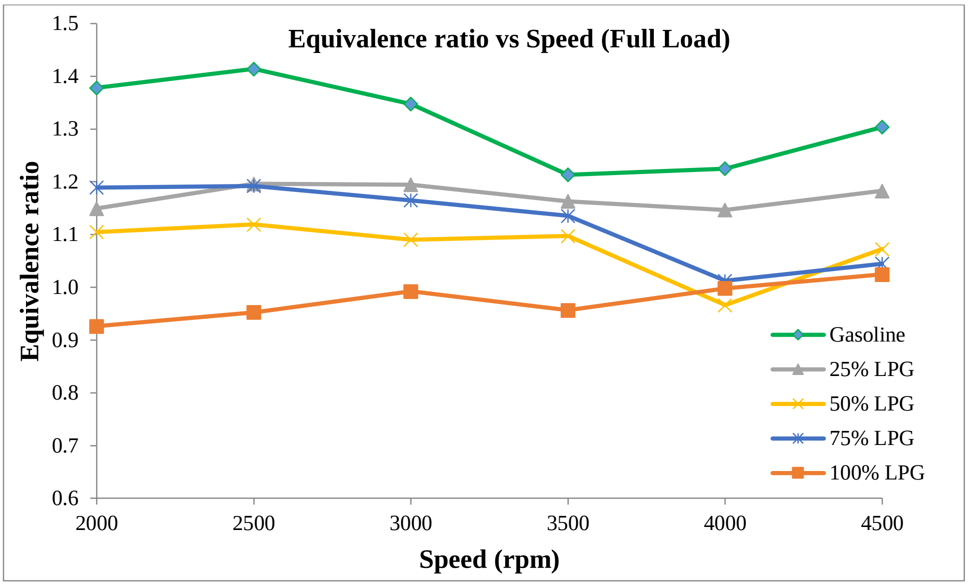


Fig. 5.10 Equivalence ratio vs Speed at full load

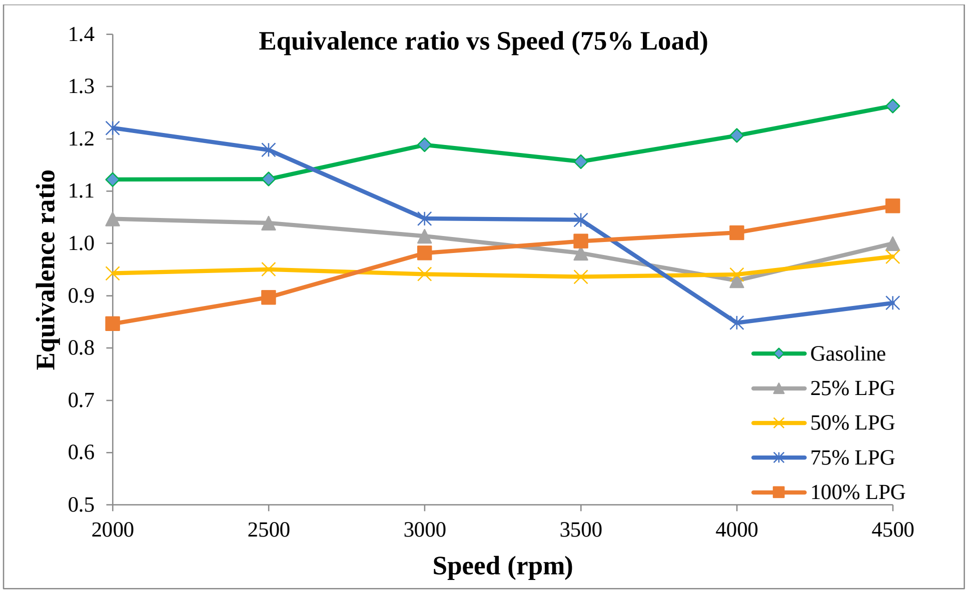


Fig. 5.11 Equivalence ratio vs Speed at 75% load

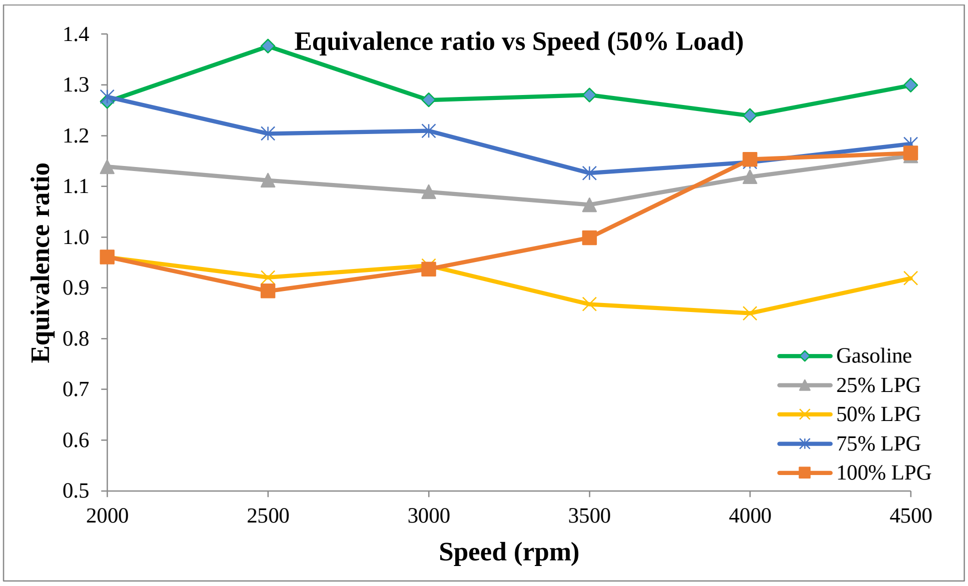


Fig. 5.12 Equivalence ratio vs Speed at 50% load

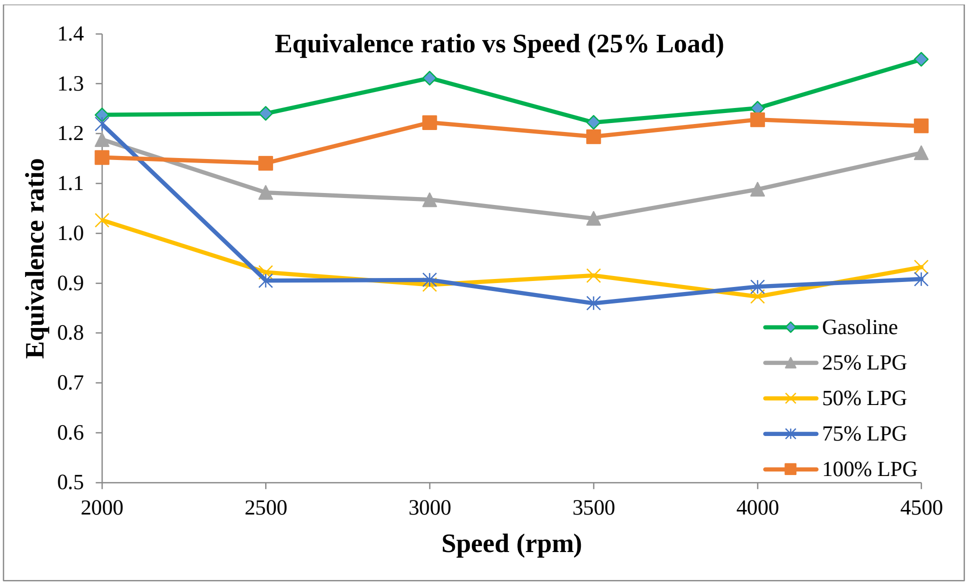


Fig. 5.13 Equivalence ratio vs Speed at 25% load

5.1.1.4 Volumetric Efficiency

Volumetric efficiency is a measure of effectiveness of an engine's induction process. The induction system includes the air-filter, injector and throttle plate in spark ignition engine. They all restrict the amount of air which an engine of given displacement can induct. Volumetric efficiency of a normally aspirated engine is the ratio of the actual volume flow rate of air into the cylinder at atmospheric pressure and temperature conditions surrounding to the engine to the rate at which the volume displaced by the piston. The figures 5.14 to 5.17 indicate the variations of volumetric efficiency of LPG, gasoline and dual fuel operation at various operating conditions. As the speed increases the volumetric efficiency also increases till the maximum power after that speed it is decreases with increase in the speed for all fuel and load conditions. As the load increases the volumetric efficiency also keep on increases for all the fuel and speed conditions. At higher load volumetric efficiency of LPG is lesser compared to gasoline because gaseous fuel will displaces more amount of air compare to liquid fuel. But at part load condition the LPG is working in leaner region the quantity of fuel injected is less compared to gasoline therefore volumetric efficiency will be more for LPG. But in dual fuel mode of operation especially 50%LPG mixture will have higher volumetric efficiency. At full load 20% and 23% increase in the volumetric efficiency is obtained for 50%LPG for 4500 rpm when compared to LPG and gasoline respectively. In dual fuel mode of operation the volumetric efficiency is increased when compared to LPG and petrol.

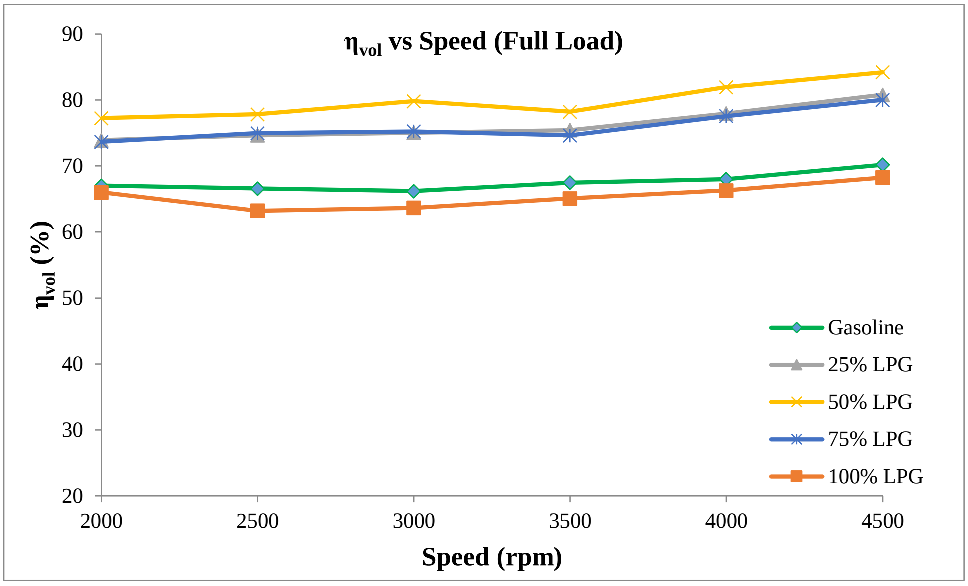


Fig. 5.14 η_{vol} vs Speed at Full load

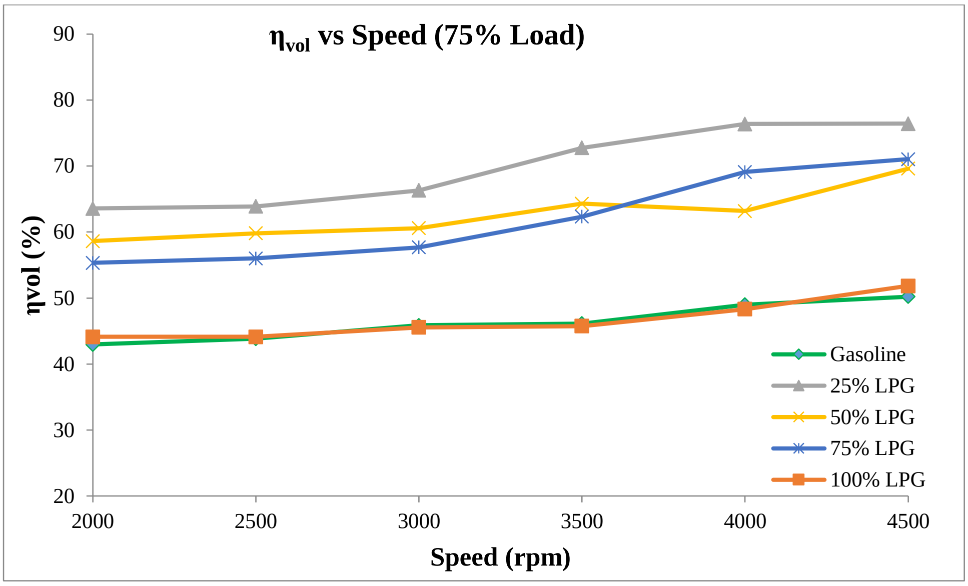


Fig. 5.15 η_{vol} vs Speed at 75% load

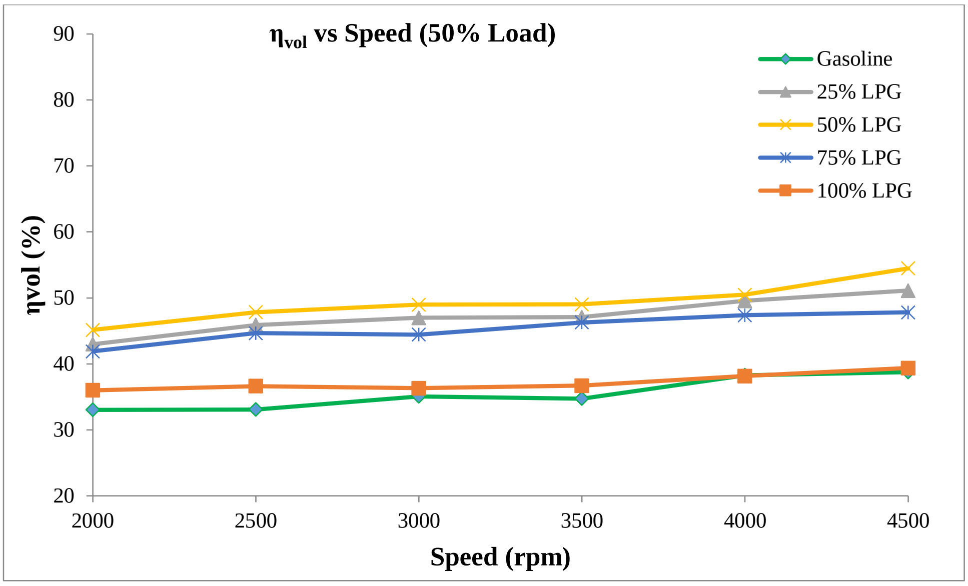


Fig. 5.16 η_{vol} vs Speed at 50% load

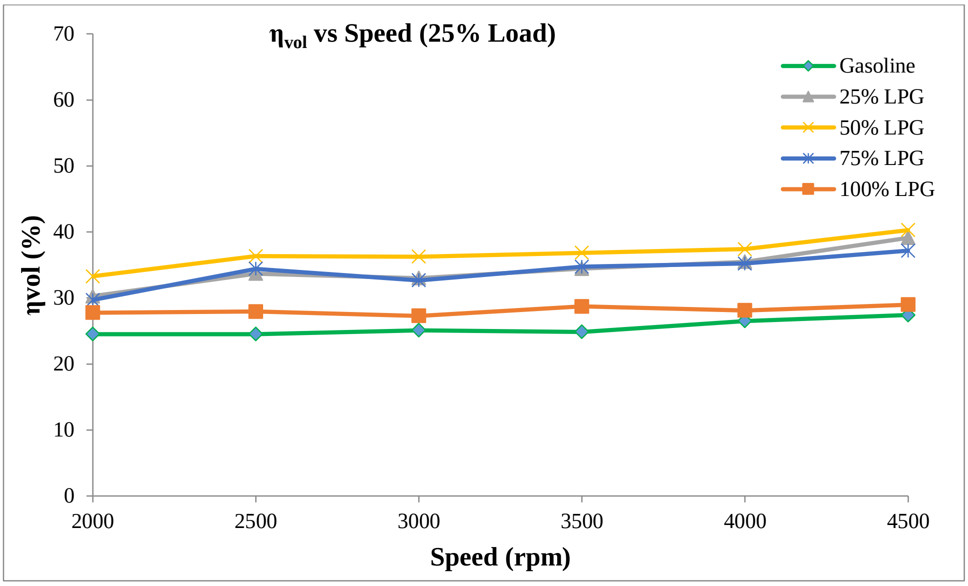


Fig. 5.17 η_{vol} vs Speed at 25% load

5.1.1.5 Brake Specific Energy Consumption

Brake specific energy consumption will indicate to produce one kilowatt of power in one hour how much energy is consumed. This comparison will give the fuel economy for an engine where different calorific value of fuels are being used. The figures 5.18 to 5.21 indicate the variations of brake specific energy consumption of LPG, gasoline and dual fuel operation at various operating speed and load conditions.

As the speed increases BSEC decreases attaining a speed of 4000rpm then starts increasing for all load and fuel blend conditions. This is the point where fuel consumption going to minimum. At quarter load, the brake power generated by the engine for all fuels is low even at higher engine speeds. This results in higher specific energy consumption.

At higher load condition, the BSEC of 100% LPG is higher than that of gasoline for all engine speeds. This can be the result of the higher flame propagation speed and self-ignition temperature of LPG lead into higher combustion duration. Due to this more consumption of fuel to the engine takes place. Hence increase in BSEC is found for 100% LPG. As the speed increases quantity of fuel going to increase as a result of which BSEC also increases compared to gasoline. In dual fuel mode of operation 50%LPG mixture will gives the lower BSEC at 100%, 75% and 50% of full load when compared to 100% LPG and gasoline. The equivalence ratio is lean for 50%LPG which decreases its burning rate. The higher combustion duration demands more quantity of fuel for LPG. Hence 50%LPG have lower BSEC at lower engine speeds. The optimal energy consumption for 50%LPG can be achieved by running the engine at 4000 rpm at higher throttle opening positions.

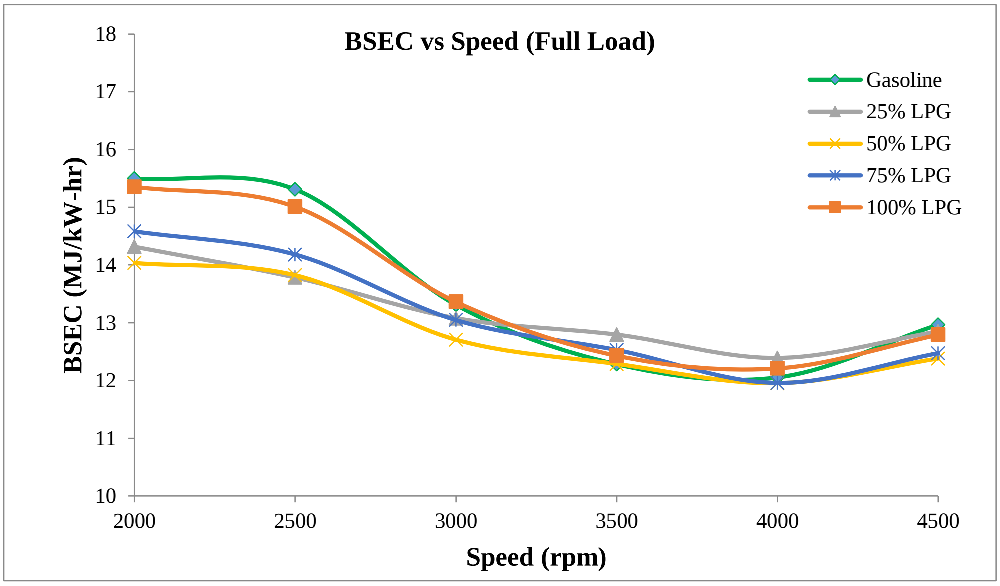


Fig. 5.18 BSEC vs Speed at full load

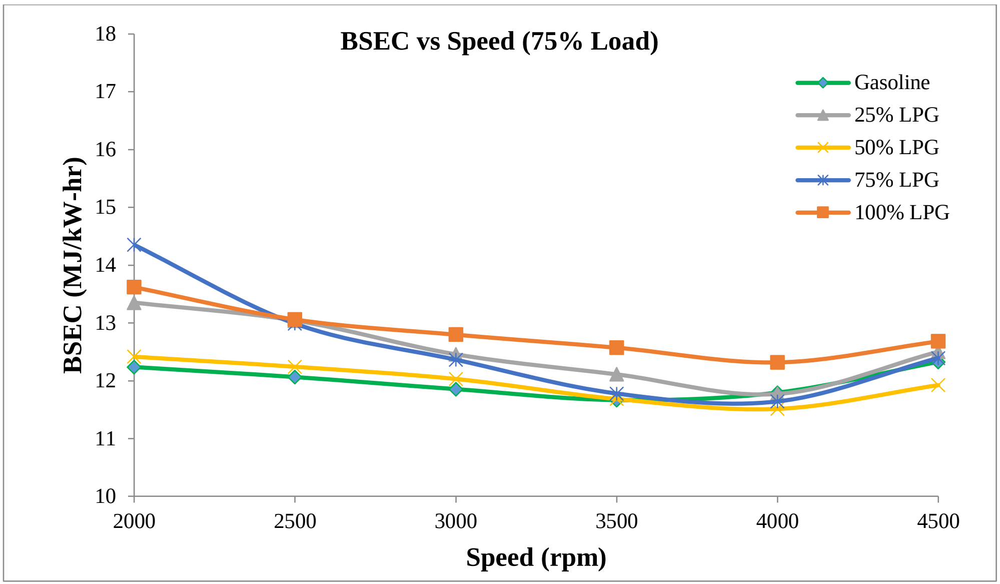


Fig. 5.19 BSEC vs Speed at 75% load

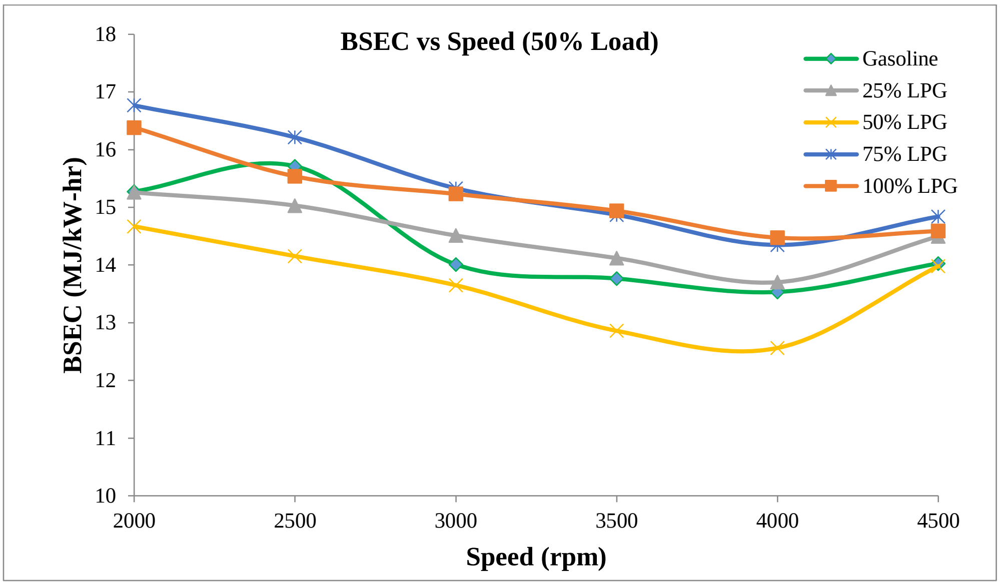


Fig. 5.20 BSEC vs Speed at 50% load

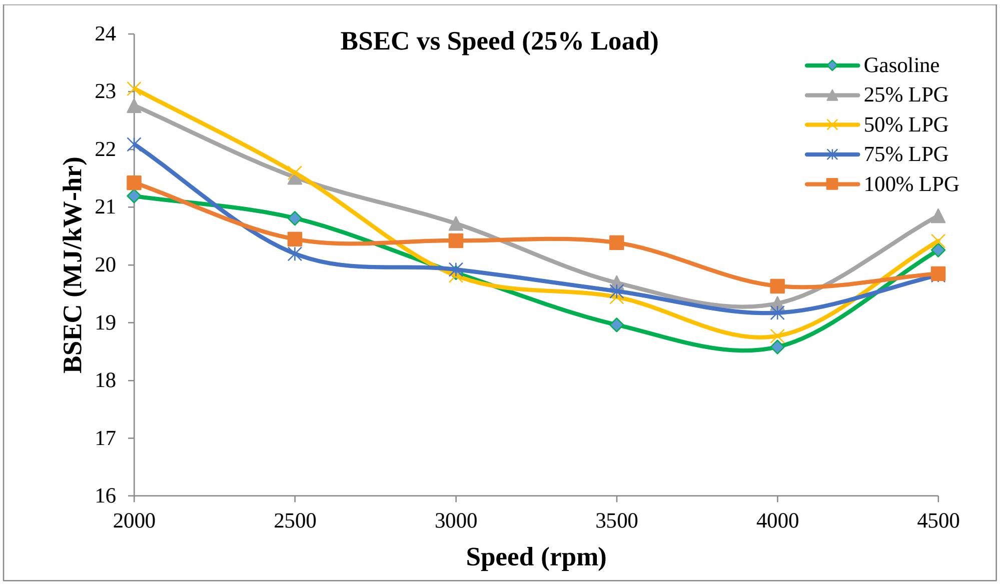


Fig. 5.21 BSEC vs Speed at 25% load

5.1.2 Emission Characteristics

Emission characteristics involves the study of parameters like carbon monoxide (CO), hydrocarbon (HC) and oxides of nitrogen (NO_X) emissions on engine speed and load conditions.

5.1.2.1 CO emissions

The comparison of CO emissions of LPG, gasoline and at dual fuel operation at various operating speed and load conditions are shown in the figure 5.22 to 5.25. From the figures it can be inferred that CO emissions of LPG is far less than that of gasoline. The CO emissions are reduced from an average value of 5.08 % to around 1.69% when we use LPG instead of gasoline. For LPG, at all load condition the CO emissions are found to be less than 2% which is well within the limits of EURO 5 pollution norms. The higher flame propagation speed and proper mixing of gaseous LPG with the air enhances the combustion and thus reduces the CO emissions. During dual fuel mode of operation as the percentage of LPG increases the CO emission is going to be reduced and it is minimum for LPG for full load and 75% of full load. This is because of carbon content in LPG is less as well as proper mixing of LPG with air enhance the reduction in CO emission.

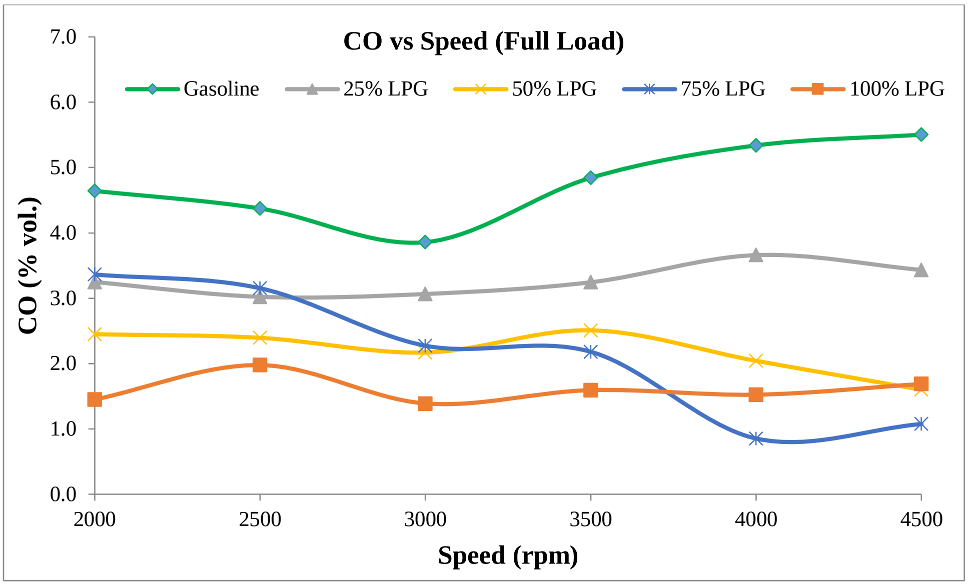


Fig. 5.22 CO vs Speed at full load

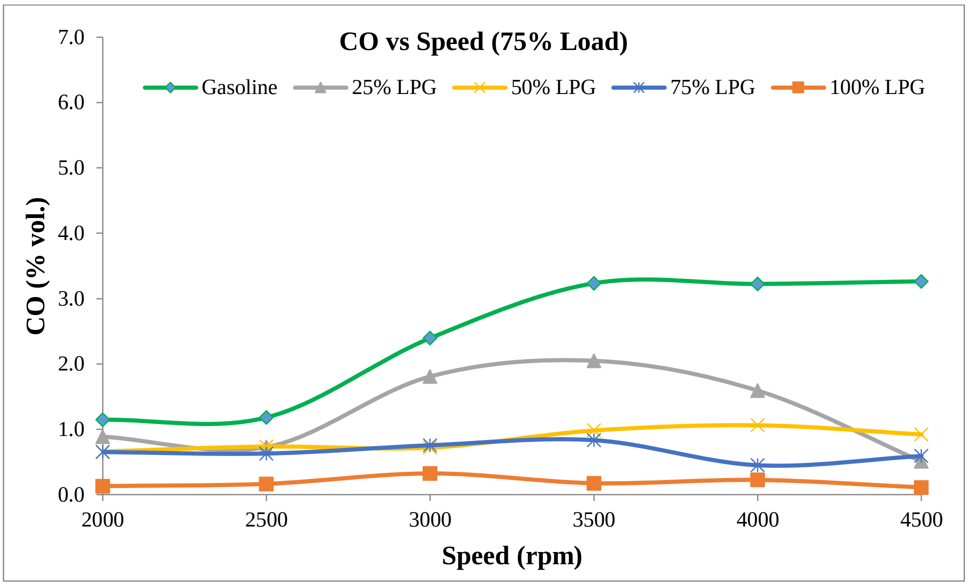


Fig. 5.23 CO vs Speed at 75% load

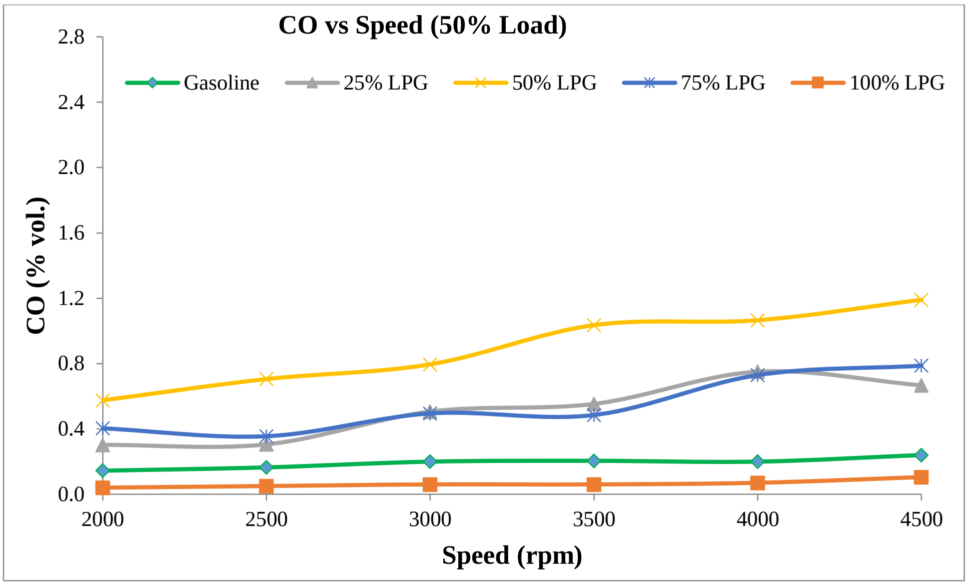


Fig. 5.24 CO vs Speed at 50% load

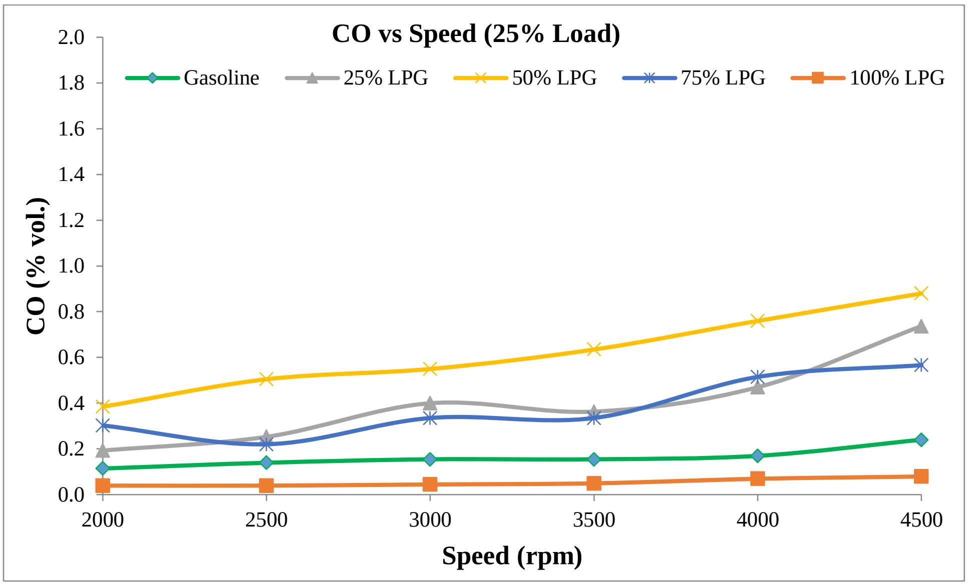


Fig. 5.25 CO vs Speed at 25% load

At 75% of full load and 4000 rpm, the reduction in CO is 92% for LPG, 50% for 25%LPG, 67% for 50%LPG, and 85% for 75%LPG obtained when compared to gasoline. At lower load conditions, as the speed increases CO emission will increases for all fuels which will indicate incomplete combustion is occurring. For fuel-rich mixtures CO concentrations in the exhaust increases steadily with increasing in equivalence ratio, as the amount of excess fuel increases. The equivalence ratio increases along with the speed. At higher load conditions gasoline working very richer region so CO emission will be more compared to LPG, whereas introduction of LPG will work in leaner region. And also at higher speeds combustion duration will be less so complete combustion of fuel will not take place so CO emission will more for gasoline.

5.1.2.2 HC emission

Figures 5.26 to 5.29 show the variation in hydrocarbon emissions on LPG and gasoline at different throttle valve openings. As the speed increases fuel will consumed more and combustion will takes place completely for all load and fuel blends. At full load and 25% of full load, the richer mixture will results in higher HC emissions for all fuels and engine speeds. LPG will have minimum HC emission when compared to gasoline this is because gasoline will work in richer region therefore unburned hydrocarbon emission is more. At full load, for gasoline HC emission is 4 times and 3 times that of LPG at 2000 rpm 4500 rpm respectively. While percentage of LPG in a mixture the HC emission is going to be reduced. At full load 4000 rpm, 42% for 25%LPG, 75% for 50%LPG, 85% for 75%LPG reduction in HC emission is obtained. This reduction in the HC emission is due the fact that LPG combustion temperature is more than that of gasoline and also higher flame propagation speed, because of this complete combustion of fuel take place which will be resulted in substantial reduction in HC emission at all throttle open conditions (Heywood 1998).

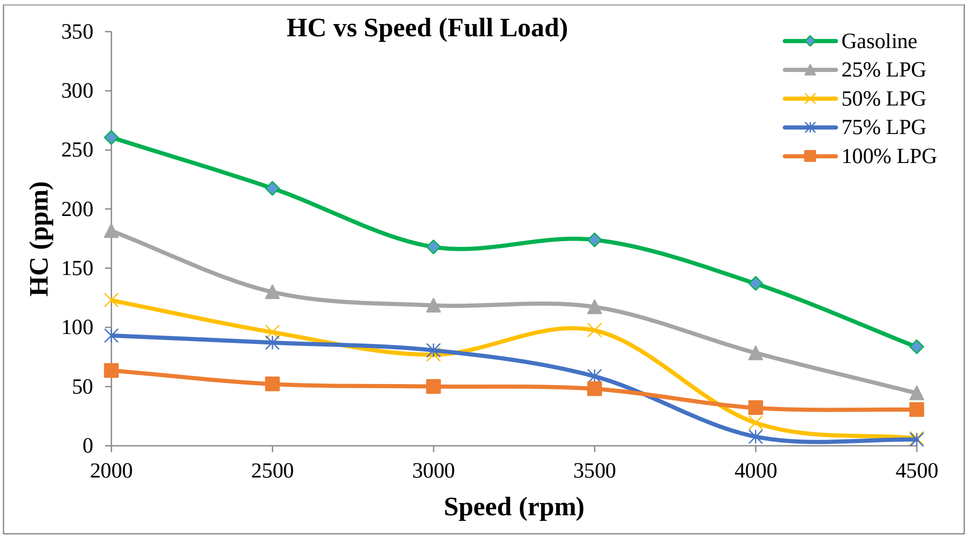


Fig. 5.26 HC vs Speed at Full load

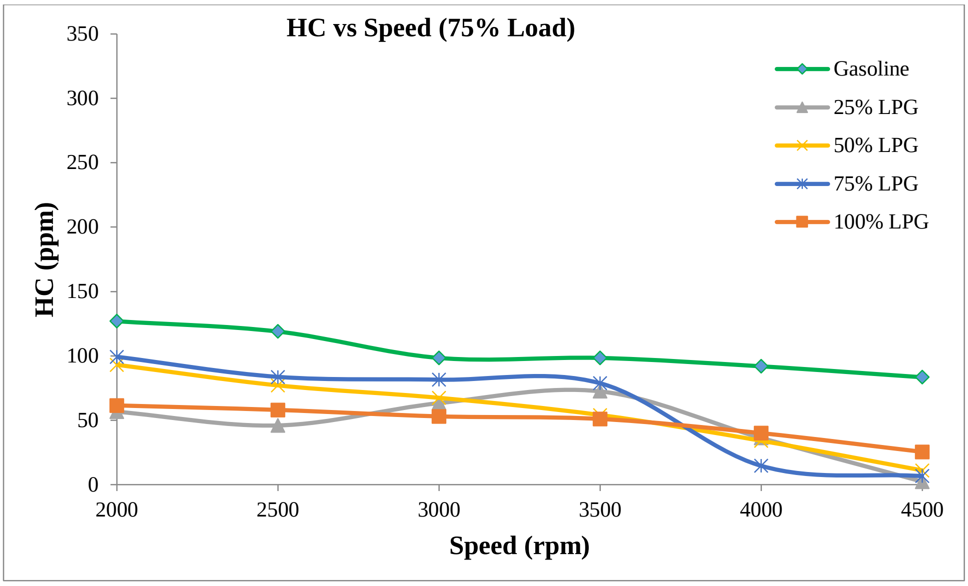


Fig. 5.27 HC vs Speed at 75% load

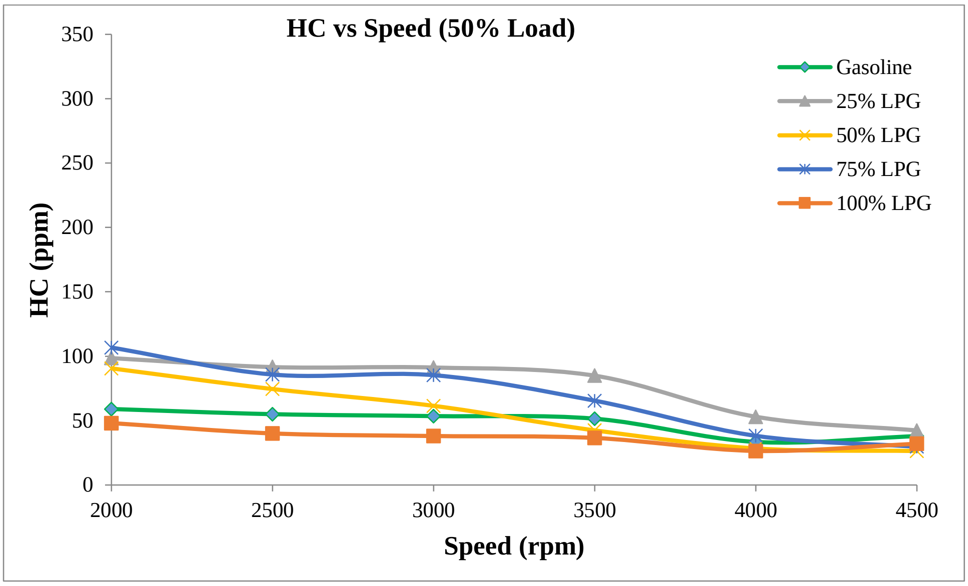


Fig. 5.28 HC vs Speed at 50% load

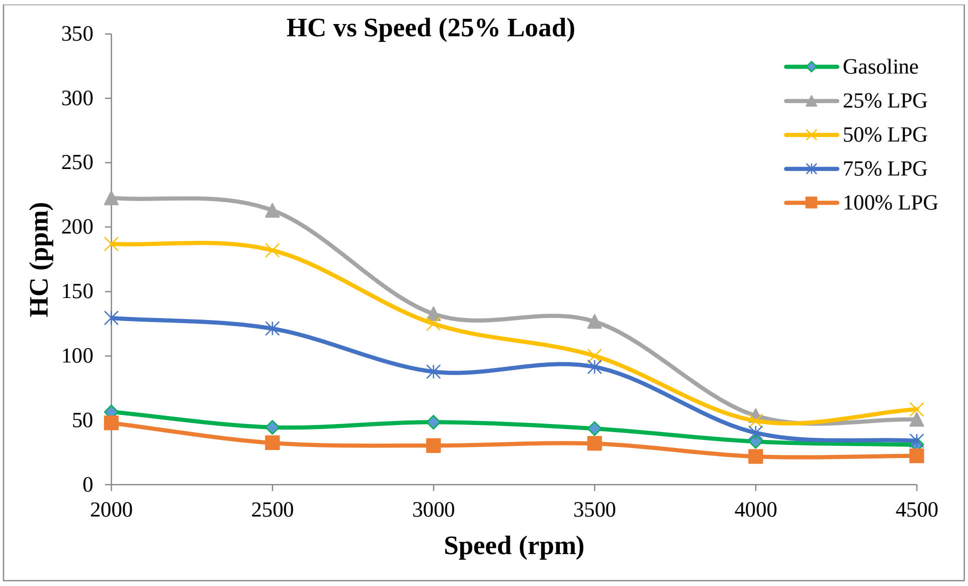


Fig. 5.29 HC vs Speed at 25% load

5.1.2.3 NO_X emission

Figures 5.30 to 5.33 show the variation of NO_X at different engine speeds and throttle position openings. One of the most important engine variables that effect NO_X emission is equivalence ratio. It can been seen that as the speed increases NO_X emission will almost remain constant for the gasoline but for LPG the NO_X will increases as speed increases for all load conditions.

Due to higher flame propagation speed of LPG at stoichiometric equivalence ratio and proper mixing of gaseous fuels with air causes an increase in the burning rate of the fuel and thus results in the complete combustion of the fuel. Hence the cylinder pressures and combustion temperatures of LPG are higher than those obtained for gasoline and other fuel mixtures. As a final outcome of this, more NO_X emissions occur in LPG combustion at higher speeds. The value of NO_X emission of LPG is almost three times the emission of gasoline at all load conditions. At full load 4000 rpm the value of NO_X reaches nearly 954ppm for LPG, whereas for gasoline value comes around 244ppm, for 25%LPG is around 408ppm, 50%LPG is around 338ppm, for 75%LPG is around 434ppm are obtained.

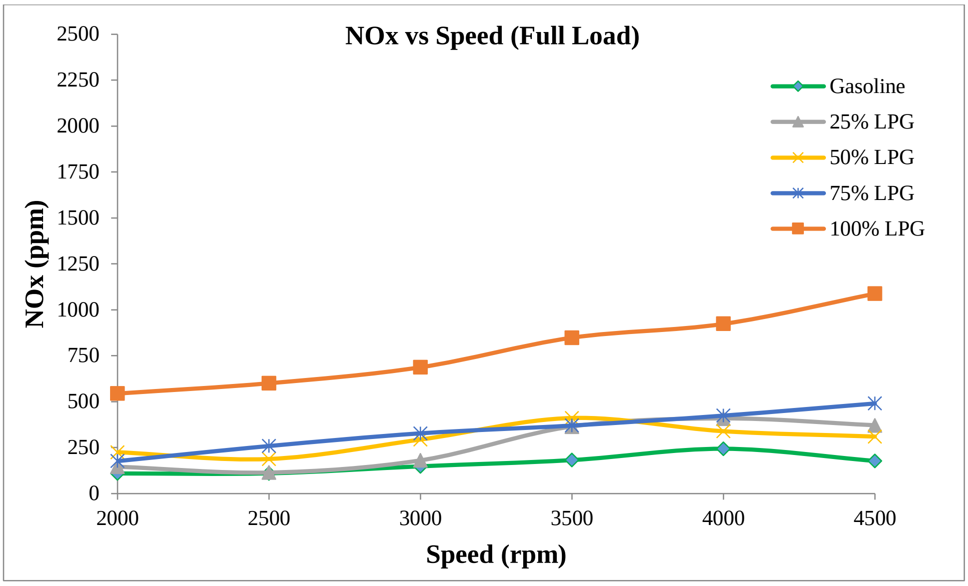


Fig. 5.30 NOx vs Speed at full load

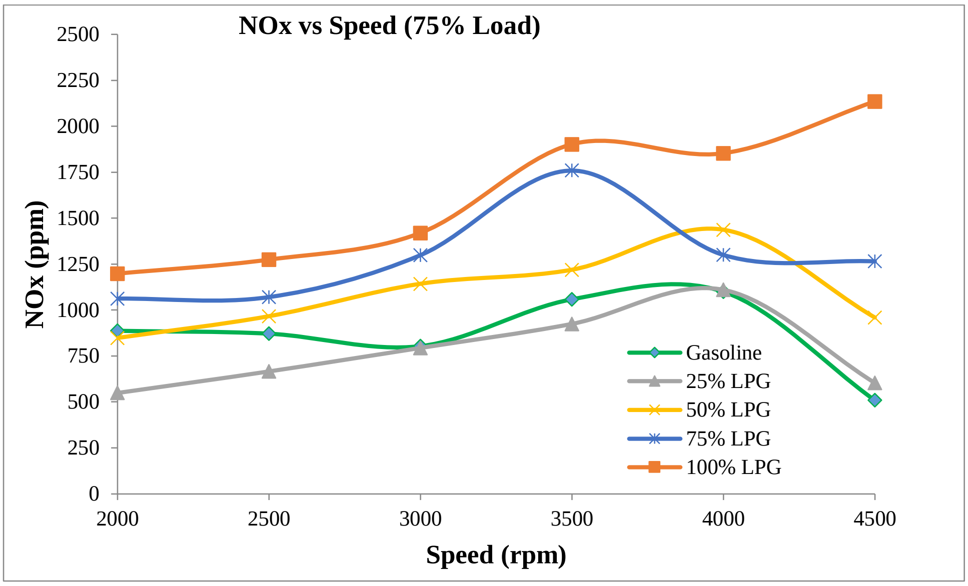


Fig. 5.31 NOx vs Speed at 75% load

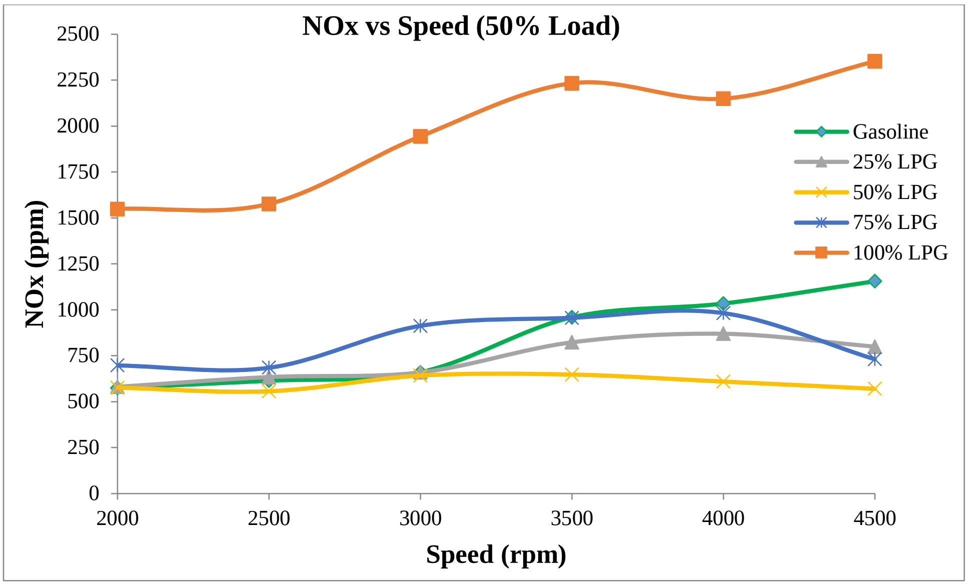


Fig. 5.32 NOx vs Speed at 50% load

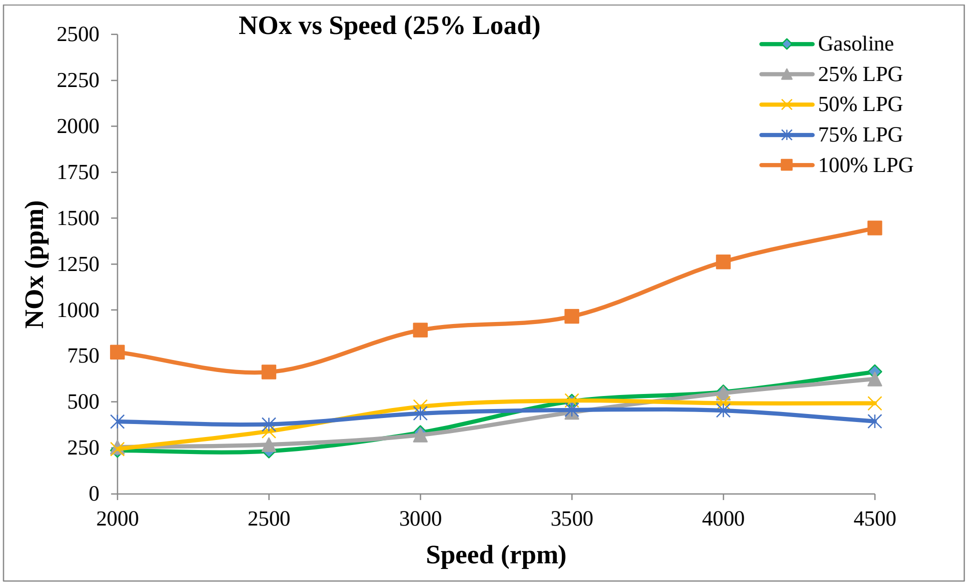


Fig. 5.33 NOx vs Speed at 25% load

5.1.3 Combustion Characteristics and Cycle by Cycle variations

Combustion characteristics and cycle by cycle variations involves the study of parameters like cylinder pressure vs crank angle (P-θ) diagram net heat release rate (NHRR), return time map and coefficient of variation (COV) on engine speed at full load conditions.

5.1.3.1 P-θ Diagram

Figures 5.34 to 5.39 show the variation of the cylinder pressure for 100 consecutive combustion cycles with all the fuel at full load and all speed conditions. It is observed that there is considerable variation in the pressure for the same operating conditions from one cycle to another. As the LPG percentage increases the peak pressure also increases and it is maximum for 100% LPG for all the speed. Since IMEP getting inside the combustion chamber is more as well as flame speed is high for LPG therefore maximum peak pressure will occur. At 4500 rpm the percentage increase in peak pressure is 20% for LPG, 9% for 25%LPG, 3% for 50%LPG, 1% for 75%LPG when compared to gasoline at full load. This increase in peak pressure will indicate the LPG will give better combustion properties compared to that of gasoline. But at lower speed the percentage increase in the peak pressure is very less. Figures 5.46 and 5.47 respectively show the variation of the maximum cylinder pressure of each cycle (P_{max}) and IMEP of each cycle for 100 consecutive combustion cycles with gasoline fuel. It is clear from the figures that there is considerable variation in the pressure related parameters for the same operating conditions from one cycle to another. Compared to peak pressure, the variation in cycle to cycle for IMEP is less. In the figure 360 deg. represents the TDC point.

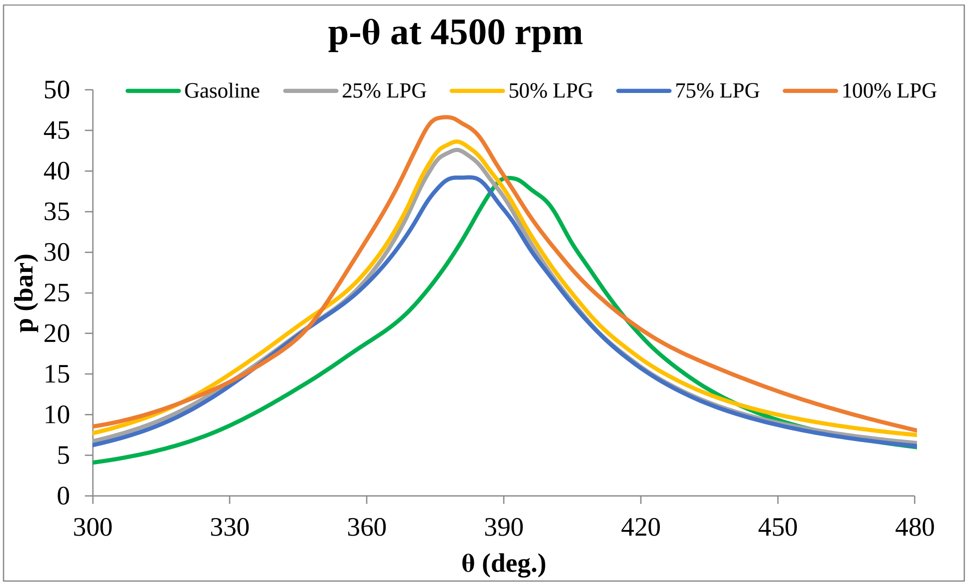


Fig. 5.34 Pressure vs Crank Angle at Full load for 4500 rpm

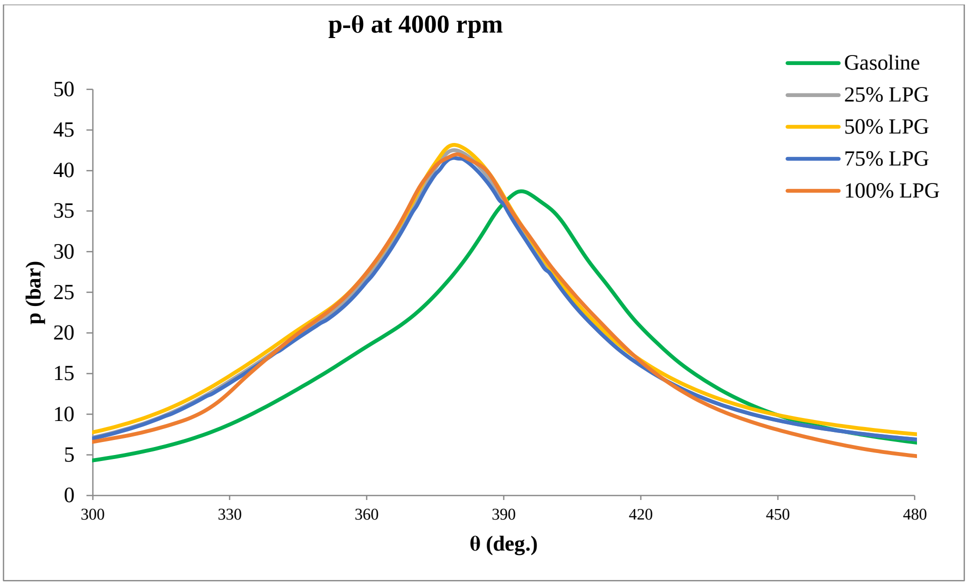


Fig. 5.35 Pressure vs Crank Angle at Full load for 4000 rpm

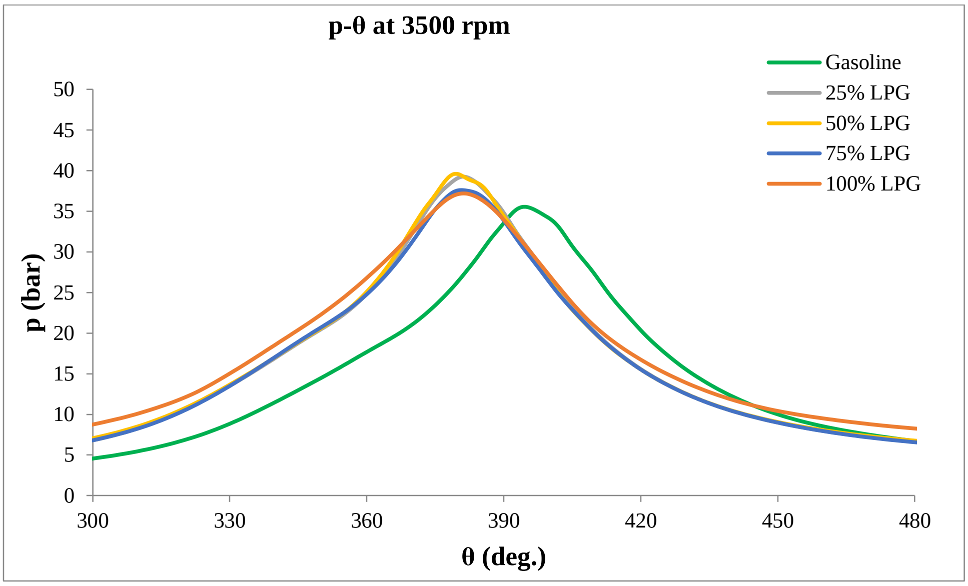


Fig. 5.36 Pressure vs Crank Angle at Full load for 3500 rpm

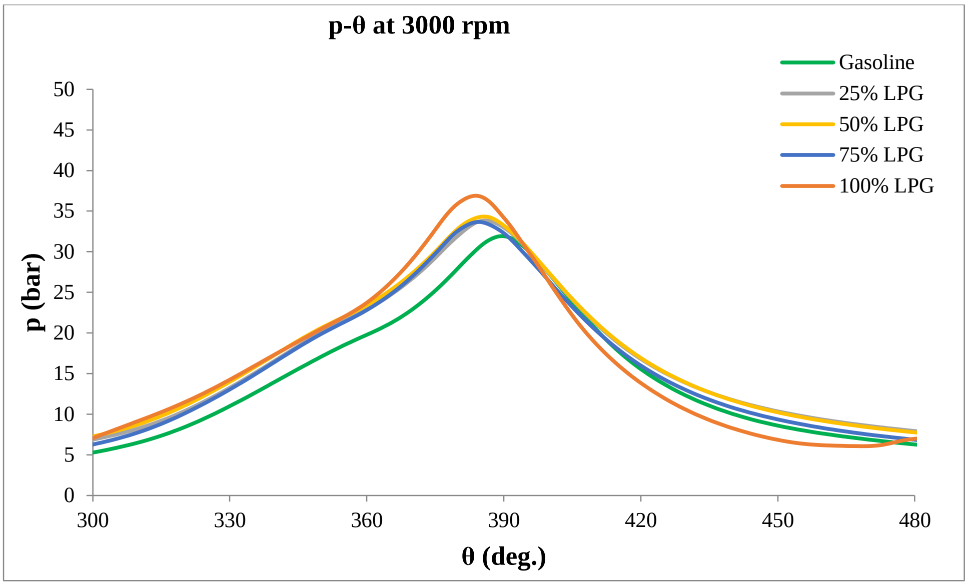


Fig. 5.37 Pressure vs Crank Angle at Full load for 3000 rpm

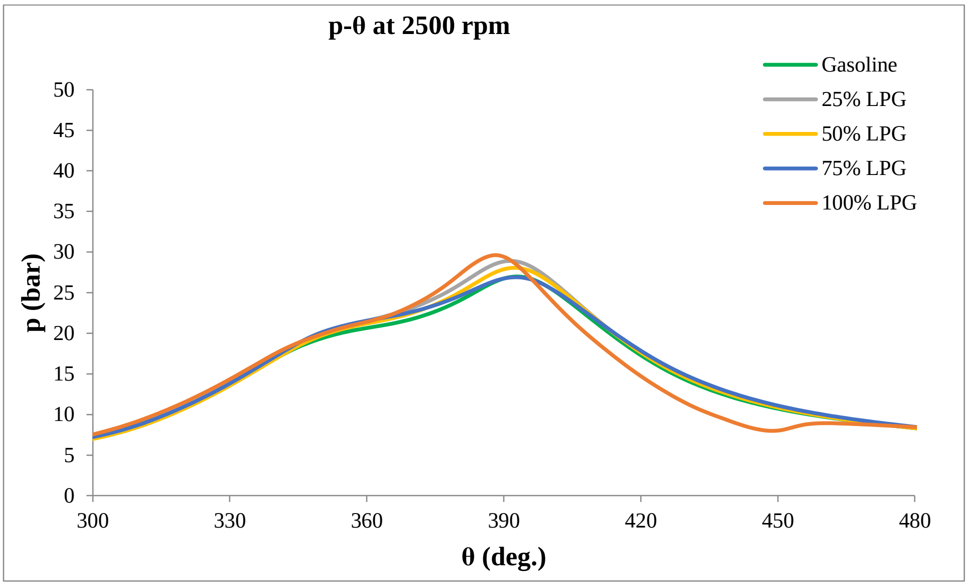


Fig. 5.38 Pressure vs Crank Angle at Full load for 2500 rpm

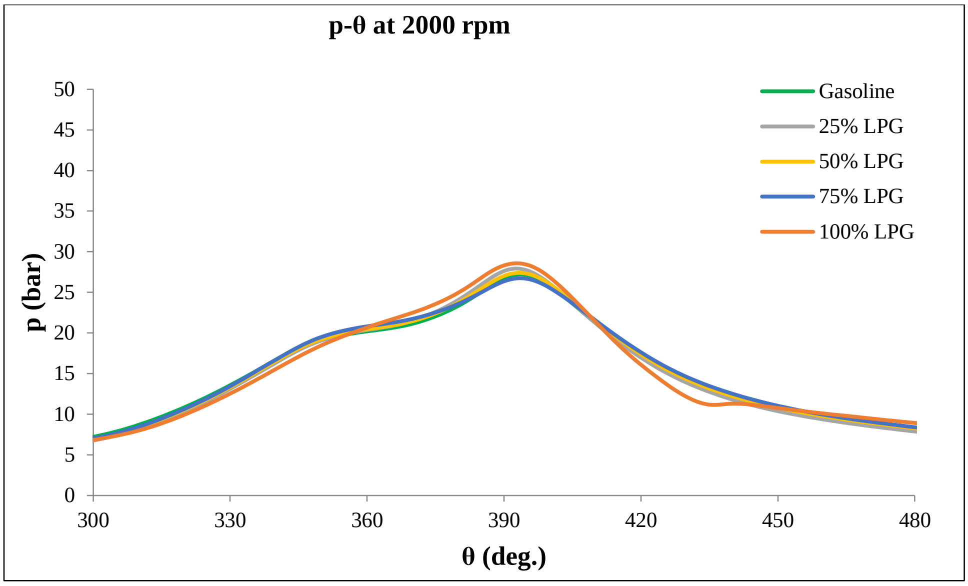


Fig. 5.39 Pressure vs Crank Angle at Full load for 2000 rpm

5.1.3.2. Pmax and IMEP

Figure 5.40 to 5.45 shows the variation in maximum peak pressure with 100 consecutive cycles at full load and different speed. It will give the how the variation in the peak pressure will occur from cycle to cycle. Combustion variations are presented with the generalized plots of peak pressure (P_{max}) and indicated mean effective pressure (IMEP) for 100 consecutive. Considerable variations in the peak pressure and IMEP trends are observed for various fuels. From the peak pressure trends it can be observed that the fluctuations are high for gasoline and 75% of LPG usage level and it is lower for LPG and 50% of LPG usage level at lower speed. But higher speed the fluctuations are high compared to lower speed for all fuels except for 50% usage of LPG. The 50%LPG show relatively low fluctuations in both P_{max} and IMEP for all speed.

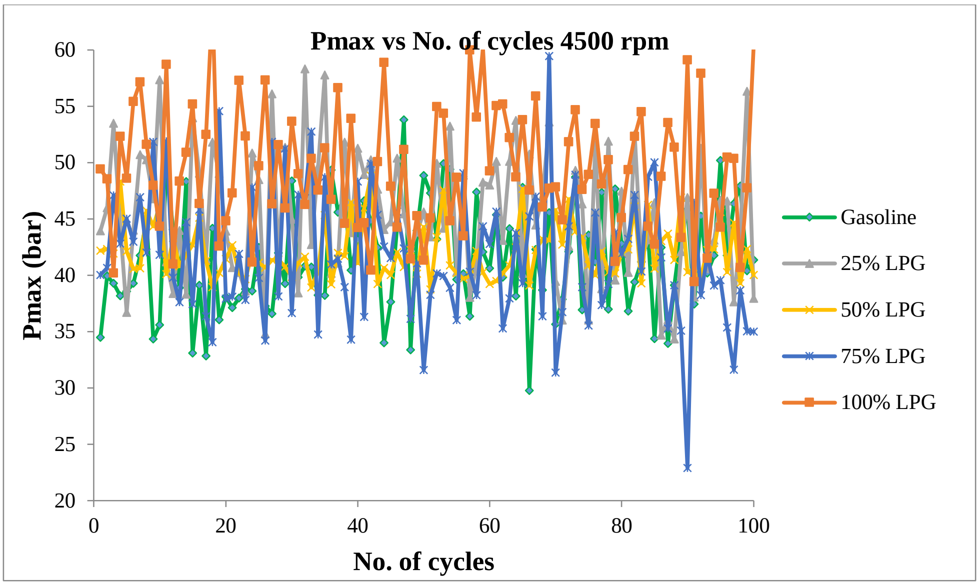


Fig. 5.40 Maximum Pressure vs No. of Cycles at Full load for 4500 rpm

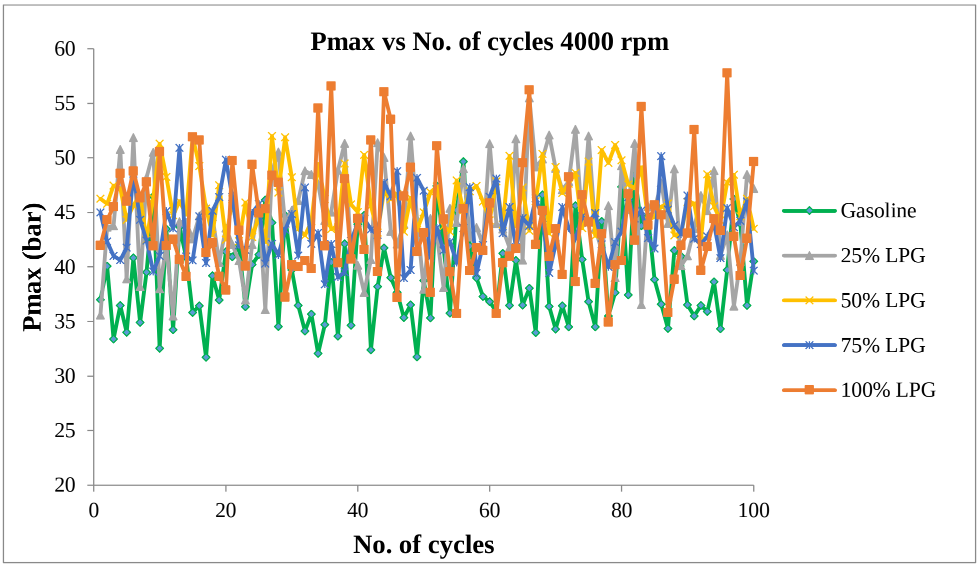


Fig. 5.41 Maximum Pressure vs No. of Cycles at Full load for 4000 rpm

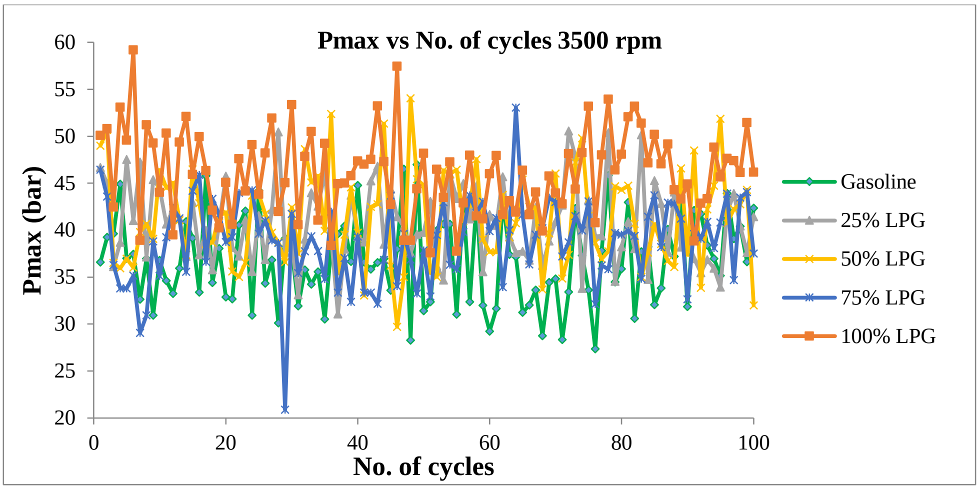


Fig. 5.42 Maximum Pressure vs No. of Cycles at Full load for 3500 rpm

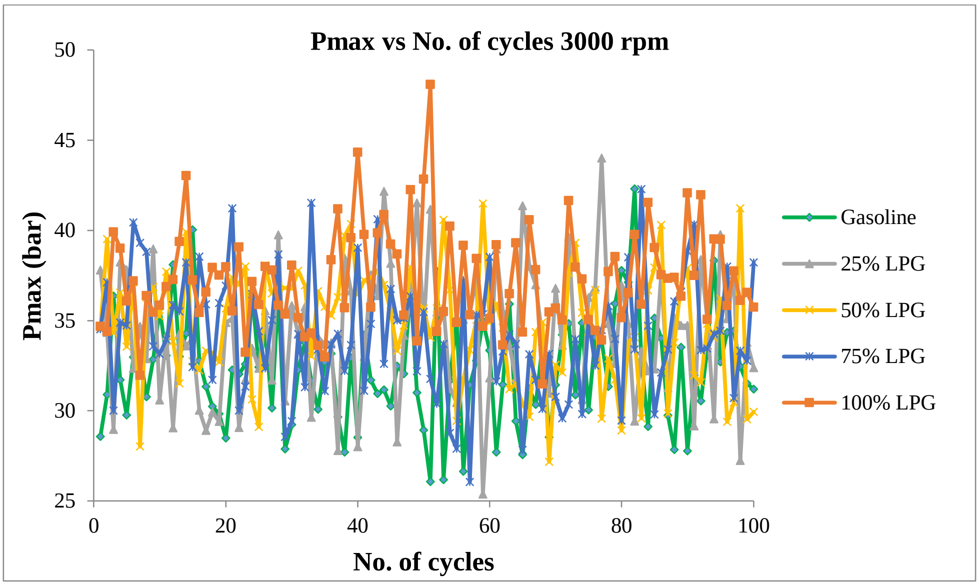


Fig. 5.43 Maximum Pressure vs No. of Cycles at Full load for 3000 rpm

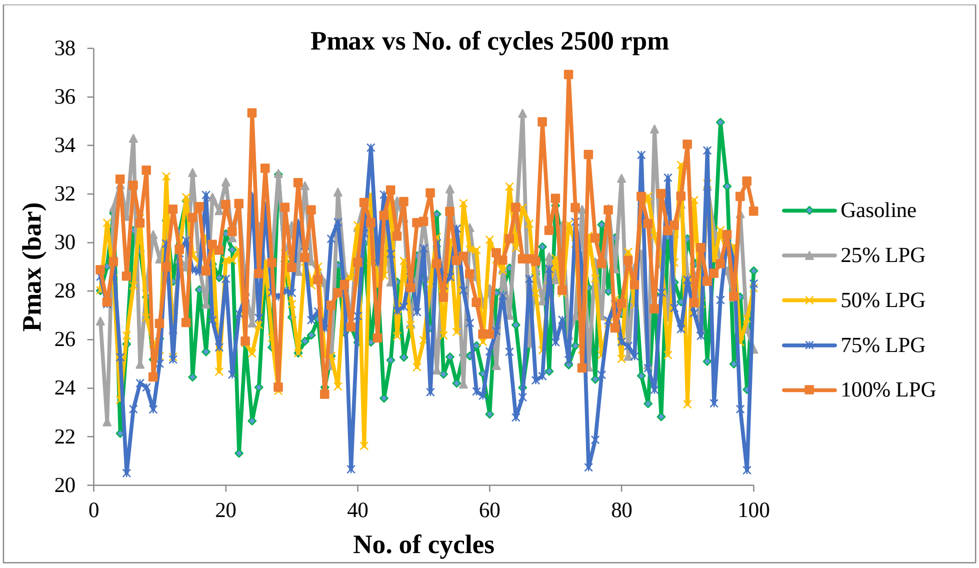


Fig. 5.44 Maximum Pressure vs No. of Cycles at Full load for 2500 rpm

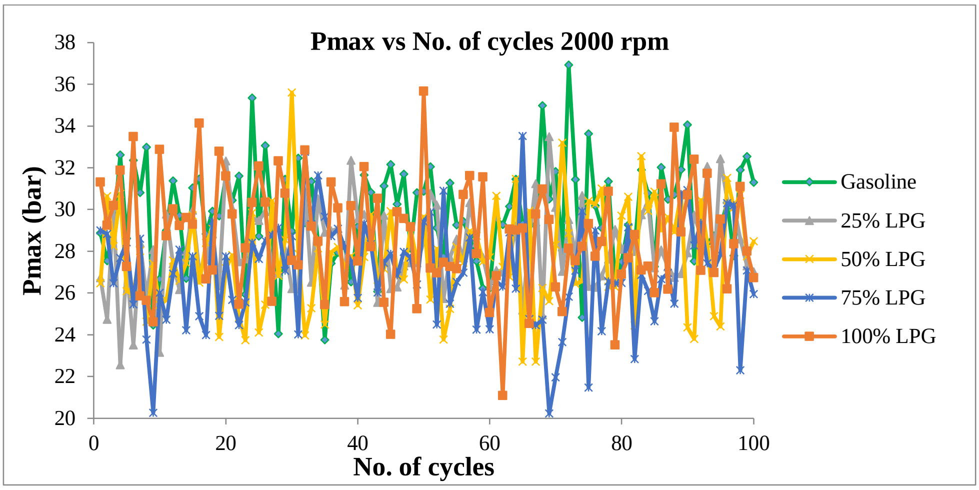


Fig. 5.45 Maximum Pressure vs No. of Cycles at Full load for 2000 rpm

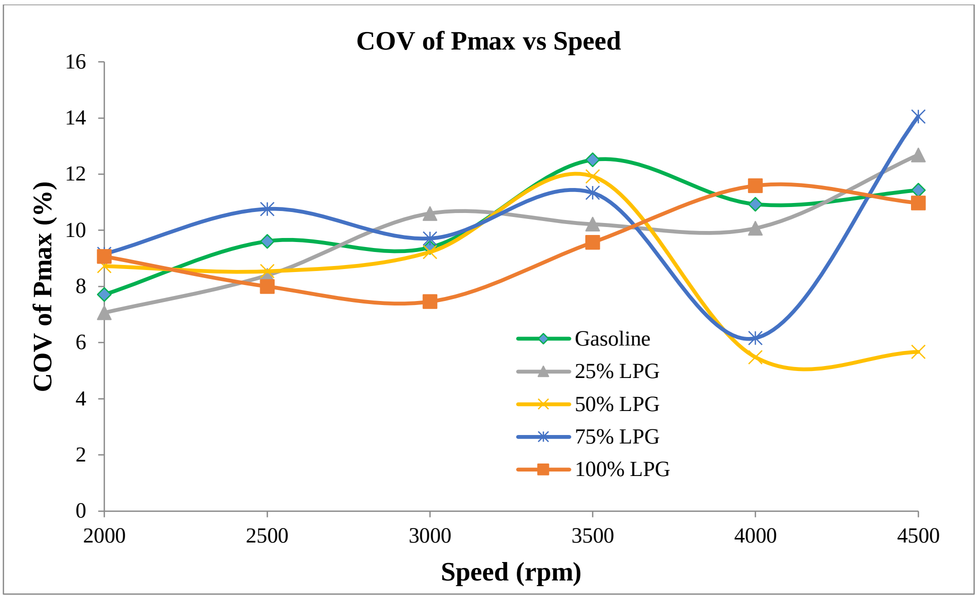


Fig. 5.46 COV of maximum pressure vs Speed at Full load

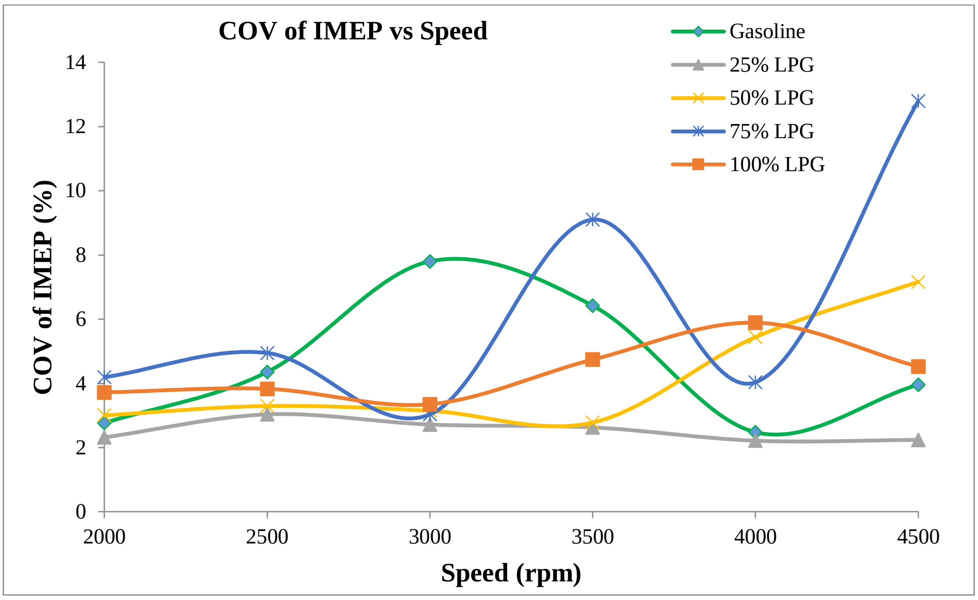


Fig. 5.47 COV of IMEP vs Speed at Full load

5.1.3.3 Return Map

Time return maps are commonly used for investigating the structure of nonlinear dynamic data and are typically constructed by plotting an observed variable against itself lagged in time. The return map presented here was constructed by plotting the P_{max} at cycle (i+L) versus the P_{max} at cycle i, where L is the return lag and typically has a value of one. Similar return map was prepared for IMEP also. For

Gaussian random data, such a map will exhibit a circular, unstructured pattern. A significantly different pattern may indicate the presence of determinism. The appearance of structure in return maps is very robust for low dimensional dynamics, even in the presence of high levels of noise. If grouping of Pmax and IMEP is symmetric then it will indicate the combustion is stabilized. Time return map of Pmax at full load and at various speed positions are shown in figure 5.48 to 5.53 and also time return map of IMEP at full load and at various speed positions are shown in figure 5.54 to 5.59. The IMEP values give an insight of combustion as it is derived from the cylinder pressure signals. At full load condition, the Pmax and IMEP of the engine increases with speed. The Pmax and IMEP developed by the engine with LPG is higher than that for gasoline and other fuel mixture. The Pmax and IMEP variation in LPG is more when compared to that of all other fuel mixture and 50%LPG usage is giving lesser variation. From figures it can observed that IMEP return map for LPG are more scattered and they are exhibiting asymmetric pattern compared to the other fuels. The return maps for the 50% usage of LPG are more symmetric. The IMEP return map indicates that till usage 50% LPG with gasoline results in more consistent power outputs.

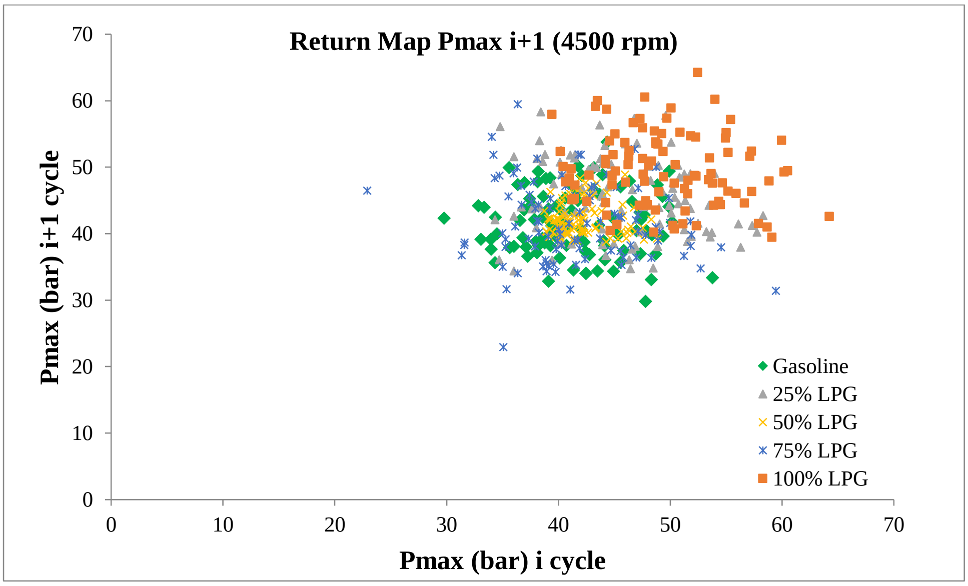


Fig. 5.48 Return map of Pmax vs Speed at Full load for 4500 rpm

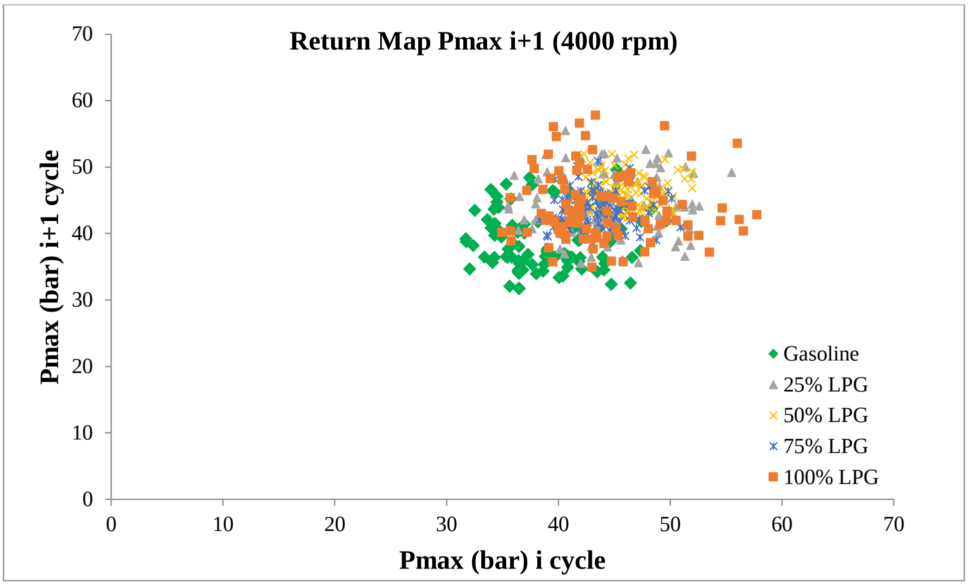


Fig. 5.49 Return map of Pmax vs Speed at Full load for 4000 rpm

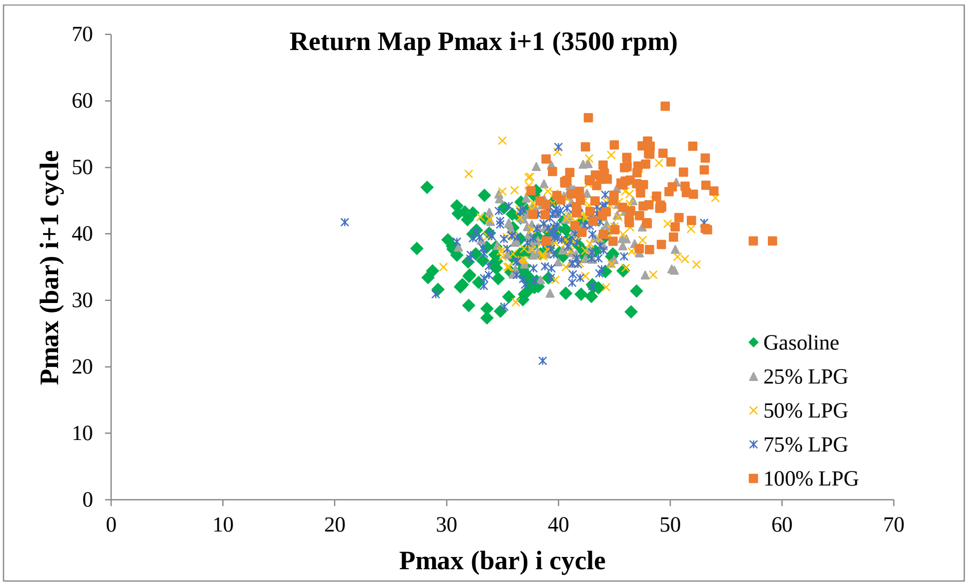


Fig. 5.50 Return map of Pmax vs Speed at Full load for 3500 rpm

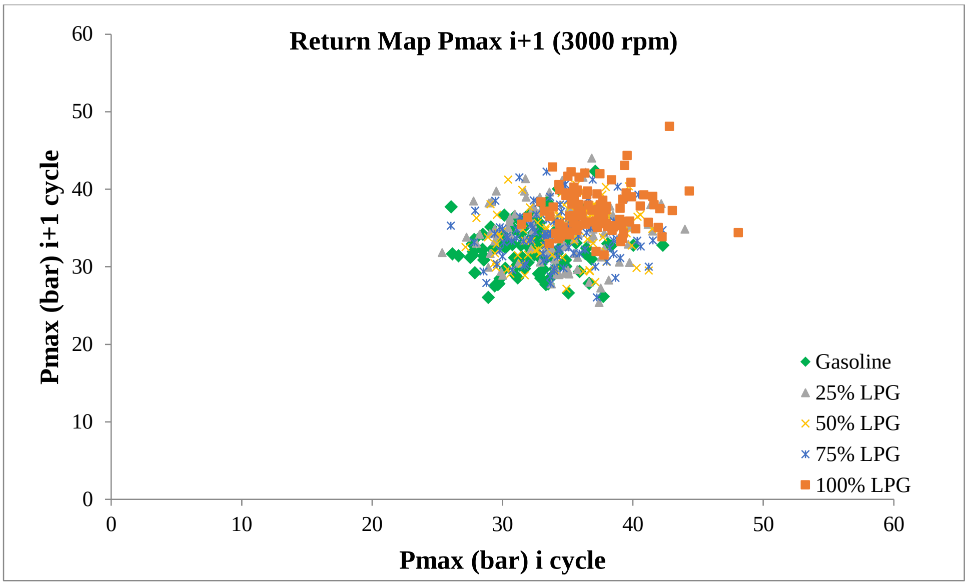


Fig. 5.51 Return map of Pmax vs Speed at Full load for 3000 rpm

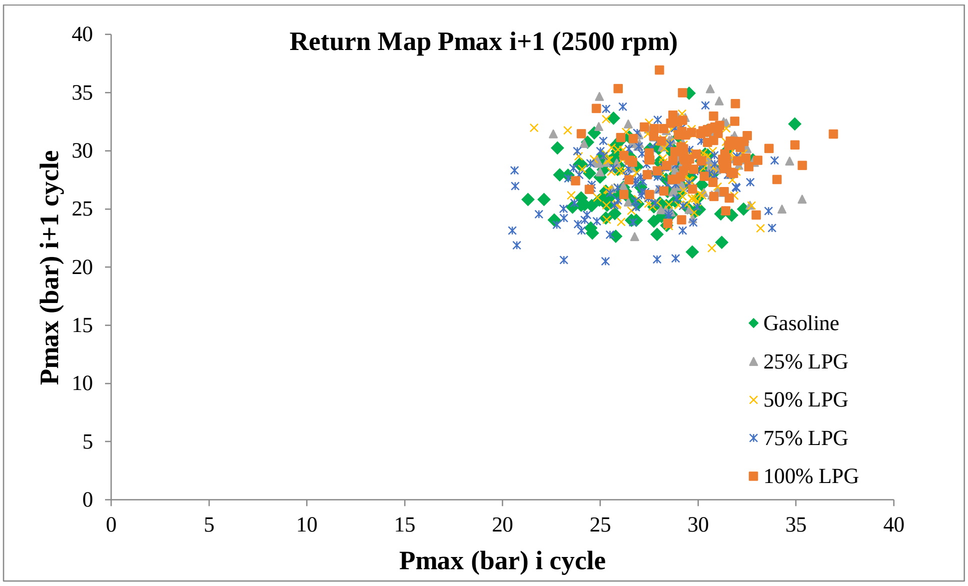


Fig. 5.52 Return map of Pmax vs Speed at Full load for 2500 rpm

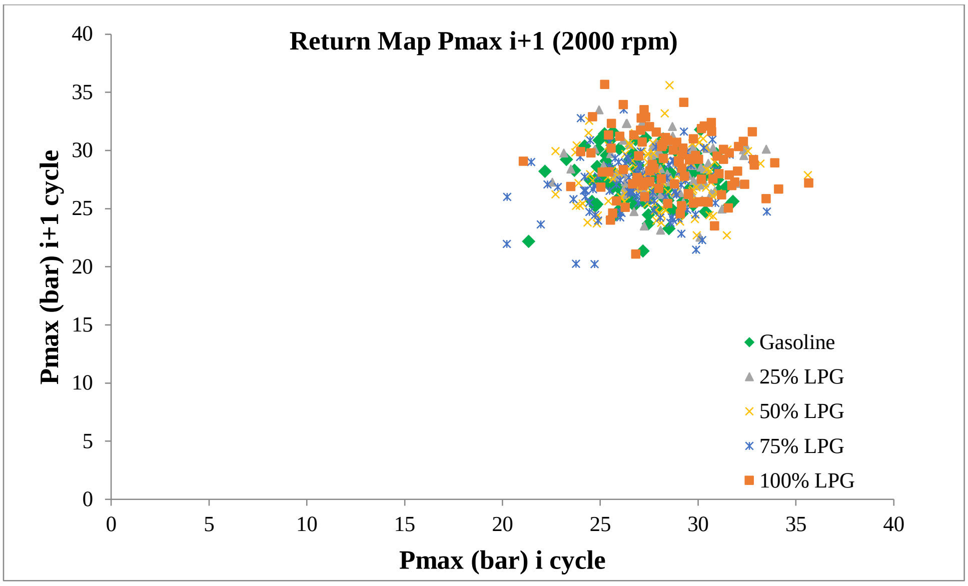


Fig. 5.53 Return map of Pmax vs Speed at Full load for 2000 rpm

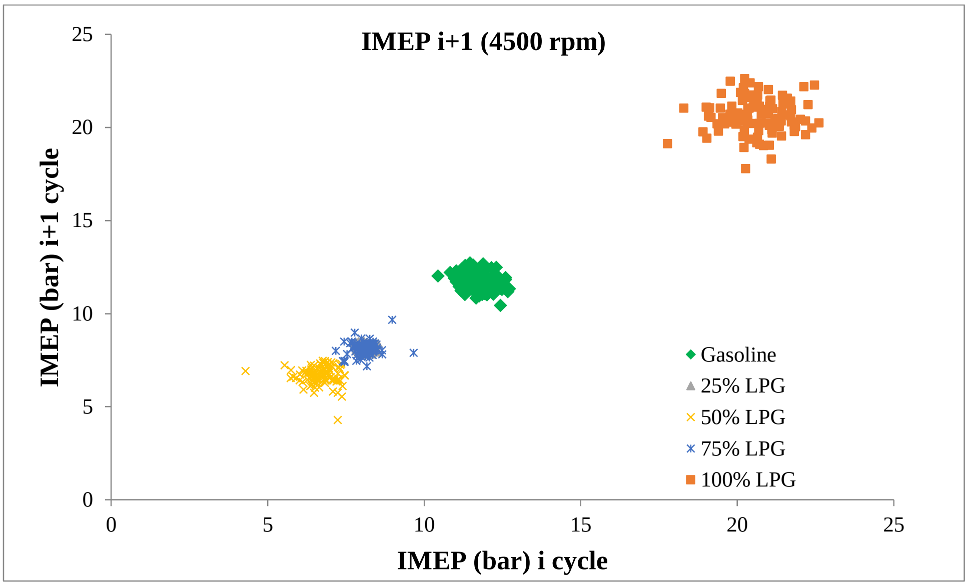


Fig. 5.54 Return map of IMEP vs Speed at Full load for 4500 rpm

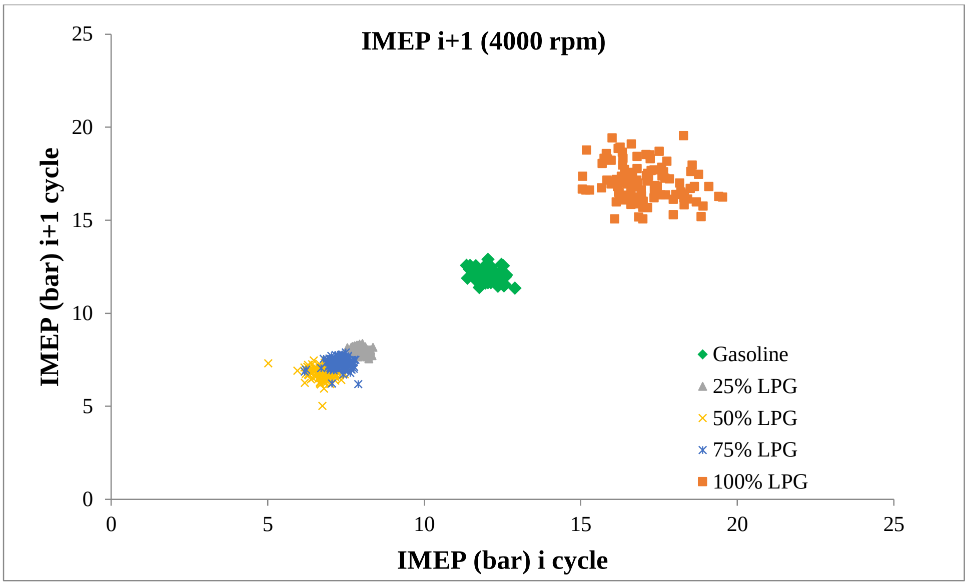


Fig. 5.55 Return map of IMEP vs Speed at Full load for 4000 rpm

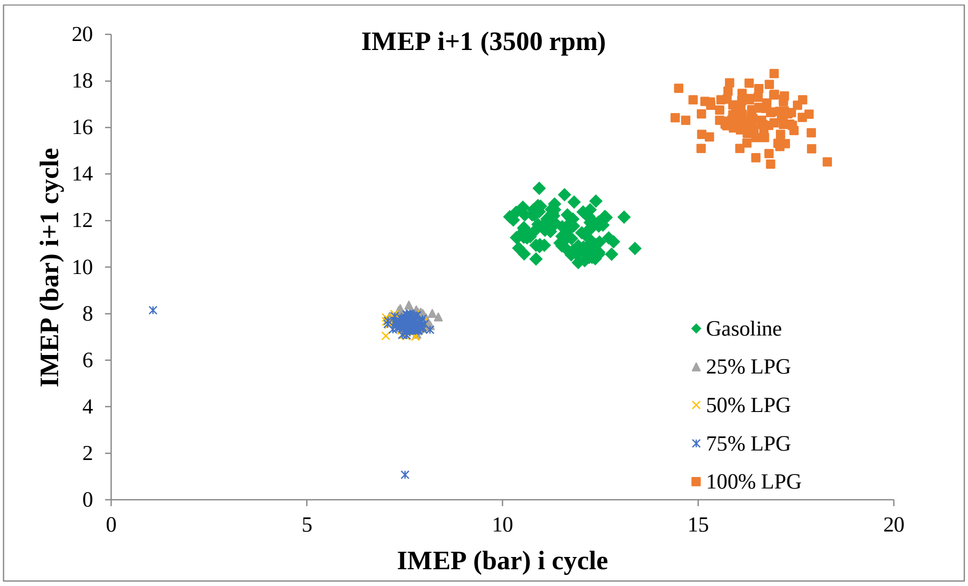


Fig. 5.56 Return map of IMEP vs Speed at Full load for 3500 rpm

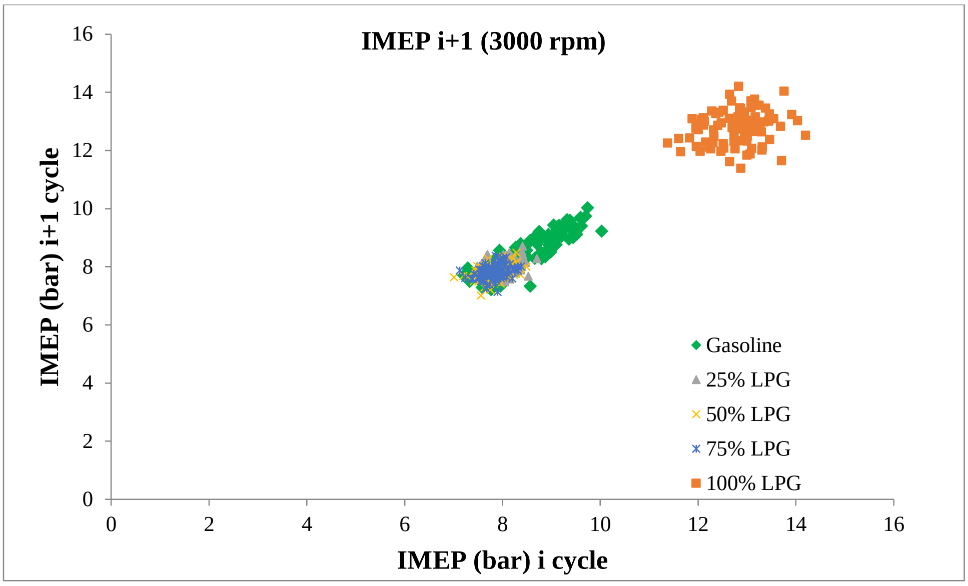


Fig. 5.57 Return map of IMEP vs Speed at Full load for 3000 rpm

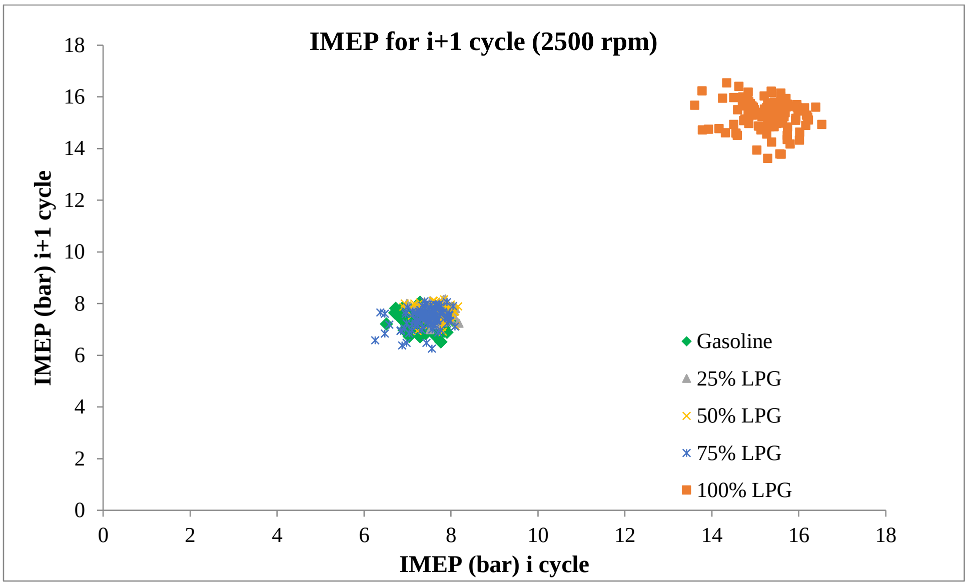


Fig. 5.58 Return map of IMEP vs Speed at Full load for 2500 rpm

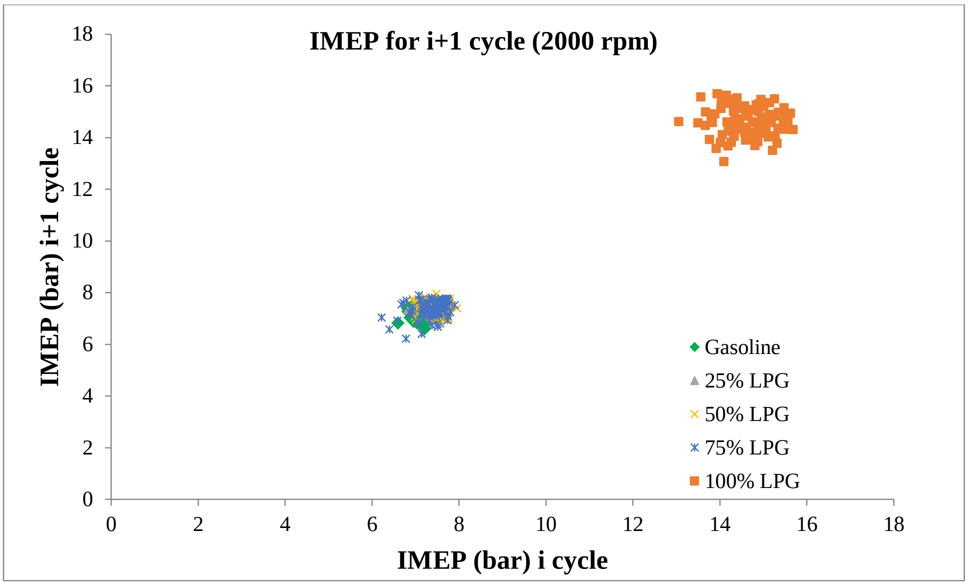


Fig. 5.59 Return map of IMEP vs Speed at Full load for 2000 rpm

5.1.3.4 Net Heat Release rate

The net heat release trends for various speeds at full load are given in figures 5.60 and 5.65 respectively. Heat release calculations are an attempt to get some information about the combustion process in an engine. Heat release rate is used in both engine performance influences in various operating conditions and same engine performances under the equal conditions. Moreover, physical and chemical properties of the fuel used in internal combustion engines are one of the main parameters which affect the heat release rate. As seen in the figures, with the increase in percentage usage of LPG in gasoline the heat release began to raise earlier than that of gasoline fuel at all the speed conditions and it is earliest for the 100% LPG. And also, the peak locations of heat release rate of increases with increase in percentage of LPG usage are wider than that of pure gasoline. This may be due to the fact that LPG has one type of hydrocarbon which will work in lean combustion. Due to this combustion chamber contains contain oxygen which improve combustion and a large amount of fuel burn takes place in the areas close to TDC. For the all test fuels, heat release rate takes place in the areas close to TDC with the increasing speed for pure LPG. (Eyidogan 2010).

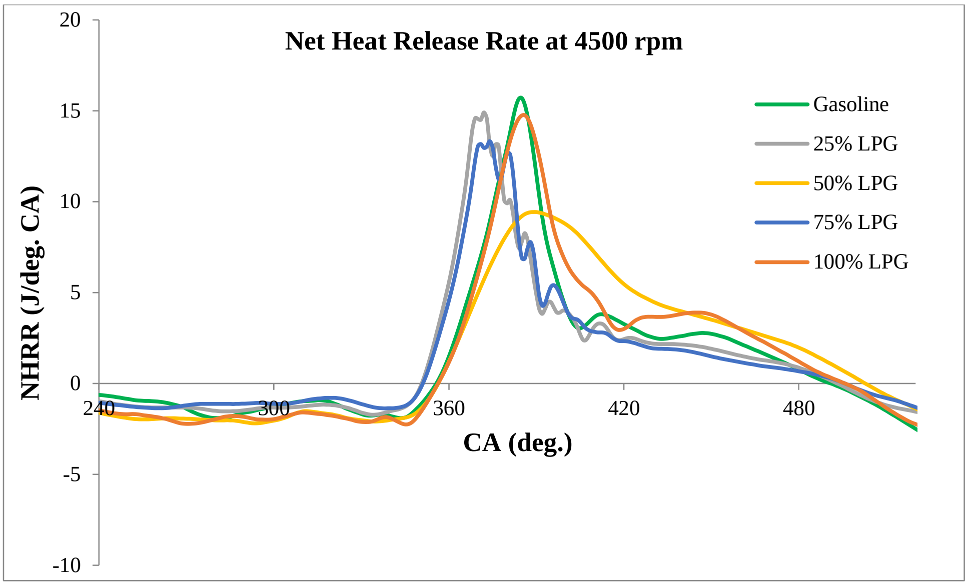


Fig. 5.60 NHRR vs Crank Angle at Full load for 4500 rpm

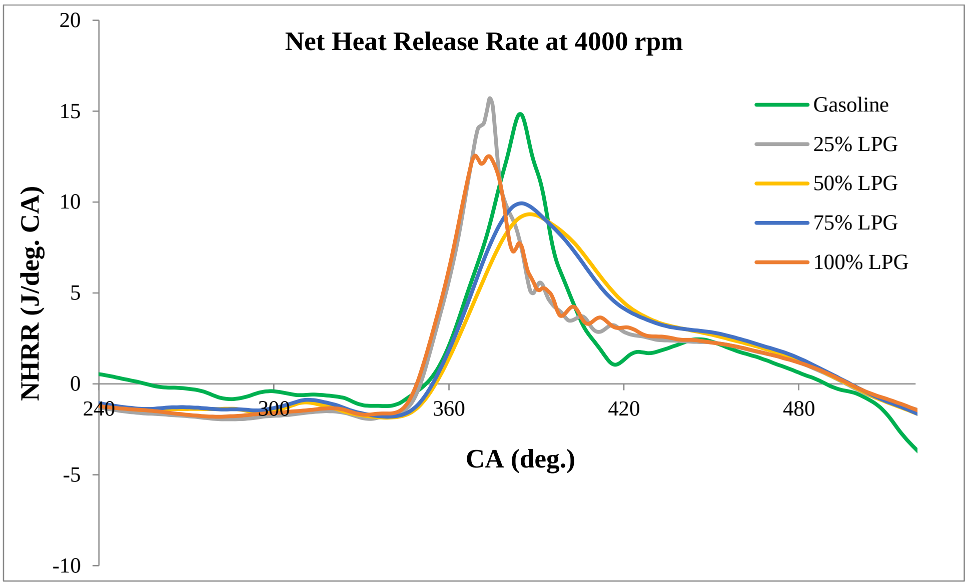


Fig. 5.61 NHRR vs Crank Angle at Full load for 4000 rpm

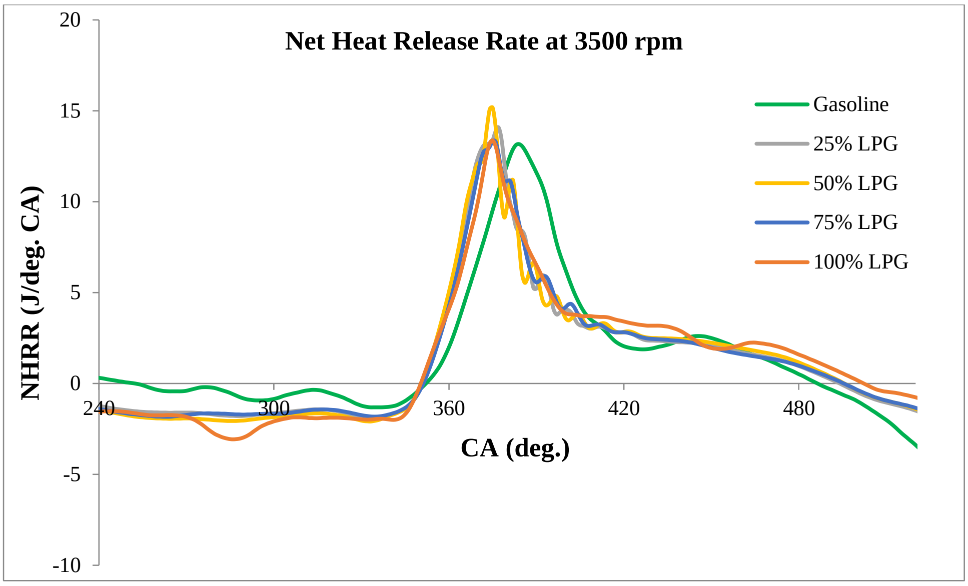


Fig. 5.62 NHRR vs Crank Angle at Full load for 3500 rpm

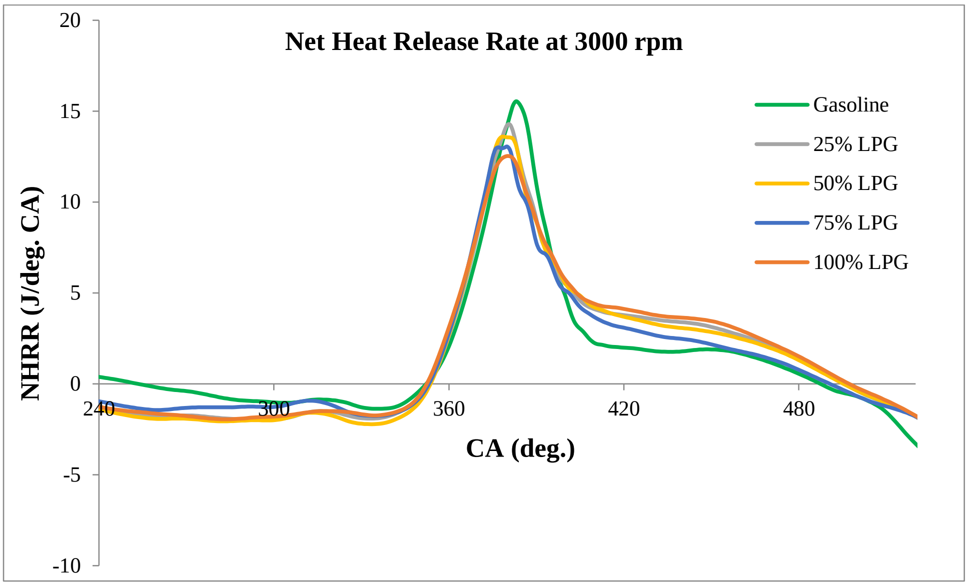


Fig. 5.63 NHRR vs Crank Angle at Full load for 3000 rpm

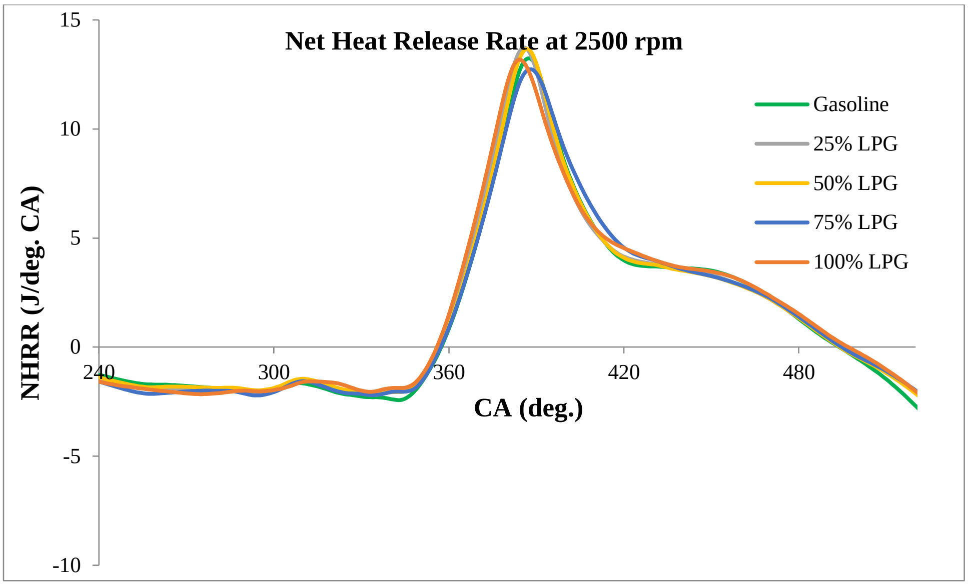


Fig. 5.64 NHRR vs Crank Angle at Full load for 2500 rpm

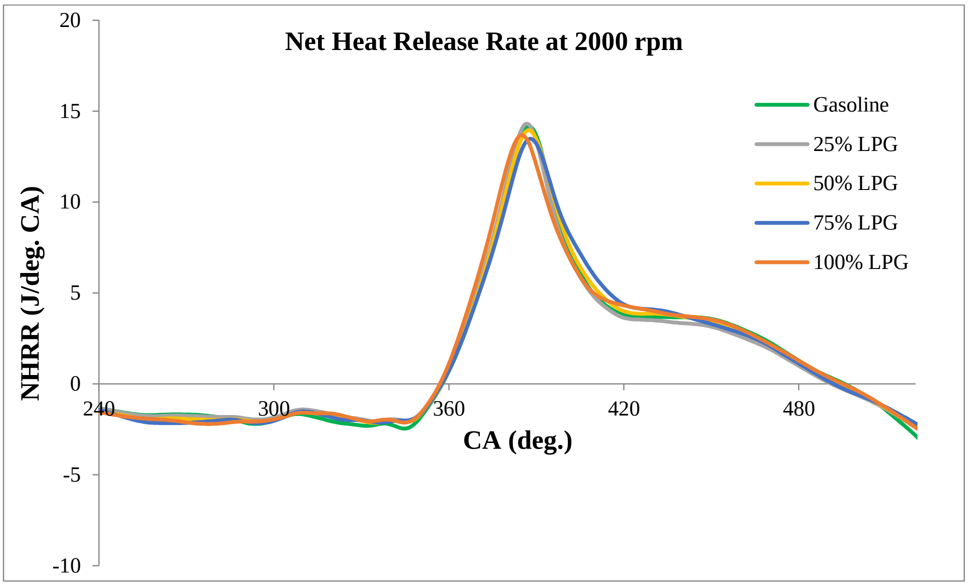


Fig. 5.65 NHRR vs Crank Angle at Full load for 2000 rpm

Summary:

- With the 50% usage of LPG the average increased in BTE is 2% for speed range of 2000 rpm to 4000 rpm at 100% and 50% of full load, and there is no much difference in BTE at 75% load when compared to gasoline.
- Volumetric efficiency is higher for 50%LPG fuel for all speed and load conditions when compared to gasoline and LPG. And also it will work in

leaner region in part load condition. Hence improvement in performance can obtain for 50%LPG usage.

- LPG will have much lower CO and HC emissions when compared to gasoline. This is a positive effect on environment. But for other LPG-gasoline ratio these emissions going to increases when compared to LPG but it is well below when compared to gasoline at 100% and 75% of full load for all speed. But at 50% load due to incomplete combustion LPG-gasoline ratio will give higher CO and HC emissions.
- NOx emission is more for LPG almost 4 times that of gasoline for all speed and load conditions. For other LPG-gasoline ratio NOx emission is lower.
- Increase in LPG ratio will give higher peak pressure and higher IMEP and combustion will take place nearer to TDC. Fluctuation in maximum pressure is more for LPG and it is minimum for 50%LPG+50%Petrol at higher operating speed. But in Fluctuation in maximum pressure is more for 50%LPG at lower speed. Time return map showed that inconsistent combustion will occur for LPG for higher speed.
- HRR is maximum for LPG and shifting towards TDC when compared to gasoline for all speed at full load. But for other LPG-gasoline ratio it not much difference when compared to gasoline.

50%LPG is given better performance results and NOx emission at 5 deg. bTDC static ignition timing for 2000 to 4000 rpm when compared to gasoline and LPG.

5.2 EFFECT OF STATIC IGNITION TIMING

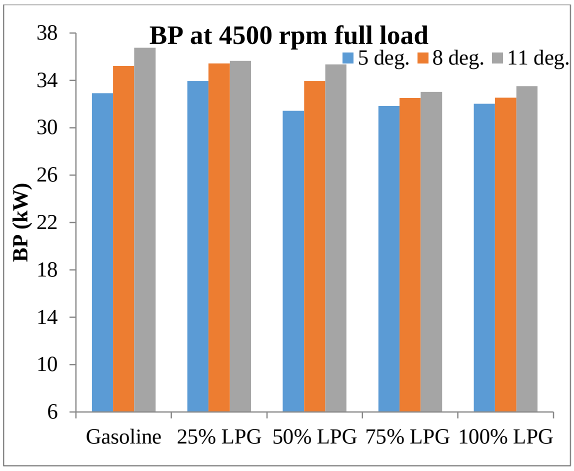
A four cylinder, multipoint fuel injection engine has been modified to work on LPG injection system. A set of experiments has been carried out with varying static ignition timing (5 deg.bTDC, 8 deg.bTDC and 11 deg.bTDC) to compare performance and emission characteristics of an engine. Parameters like engine speed is varied from 2000 to 4500 rpm in steps of 500 rpm for different engine loading and different LPG flow (0%, 25%, 50%, 75% and 100% by mass). Results are analyzed and compared with baseline fuel gasoline and factory set static ignition timing of 5 deg. bTDC. Based on analysis of results the LPG flow rate and static ignition timing are optimized.

5.2.1 Performance Characteristics

5.2.1.1 Brake Power

The following figures 5.66 to 5.69 show the effect of ignition timing on the brake power for various engine speeds at full load conditions. From the results it is revealed as the static ignition timing is advanced BP is increasing and it is maximum for 11 deg. bTDC. The general trend of increasing torque and power can attribute to the combustion process occur earlier in the cycle which will have more time availability. Therefore advancing the static ignition timing causes increased power output of an engine. Also, due to the higher flame velocity of LPG when compared to that of gasoline will reduce the flame propagation time thereby reduces the combustion duration. Hence net effect will be increased in power output of an engine. As the speed increases, the engine generates constant torque for a particular throttle valve opening position. Hence brake power increases linearly with engine speeds. The influx of fuel is more in wide open throttle compared to part throttle operations. This results in the higher power generation in the full load condition for all the fuels. At full load, 4500 rpm, average increase in power output for gasoline is 11.66% at 11 deg. bTDC, 6.92% at 8 deg. bTDC. for 100% LPG is 4.6% at 11 deg. bTDC, 1.8% at 8 deg. bTDC when compared to the 5deg. bTDC.

Figures 5.70 to 5.73, 5.74 to 5.77 and 5.78 to 5.81 show the effect of static ignition timing on BP for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively. As in full load condition similar trends are also observed in part load conditions. With advancing the static ignition timing engine power output is increasing for all fuel bend conditions.



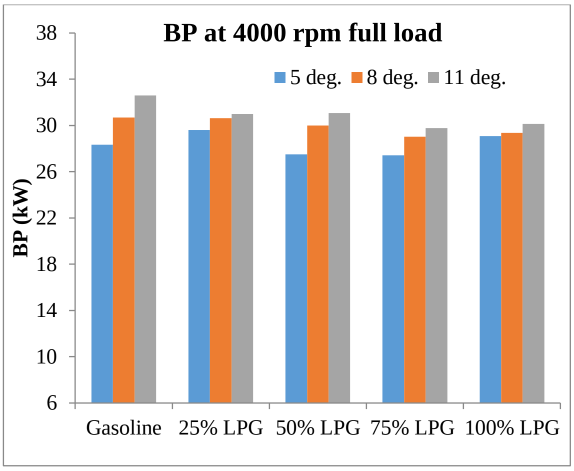
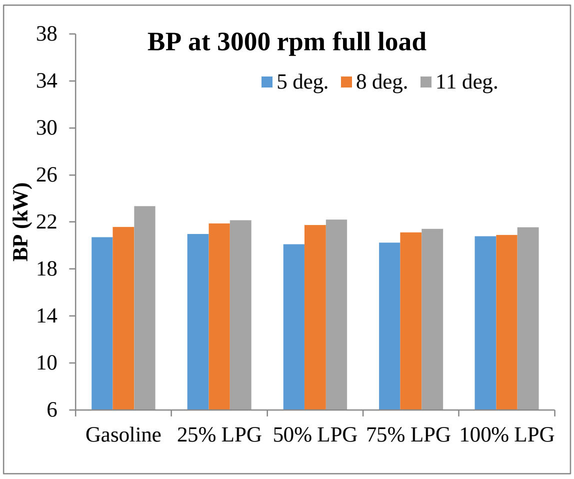


Fig 5.66 Effect of static ignition timing on BP at 4500 rpm full load condition.

Fig 5.67 Effect of static ignition timing on BP at 4000 rpm full load condition.



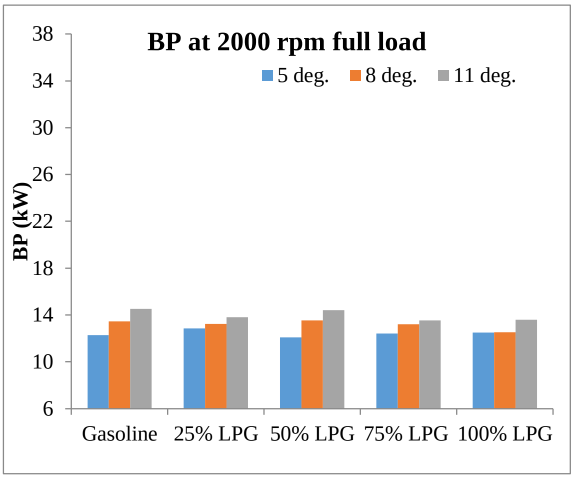
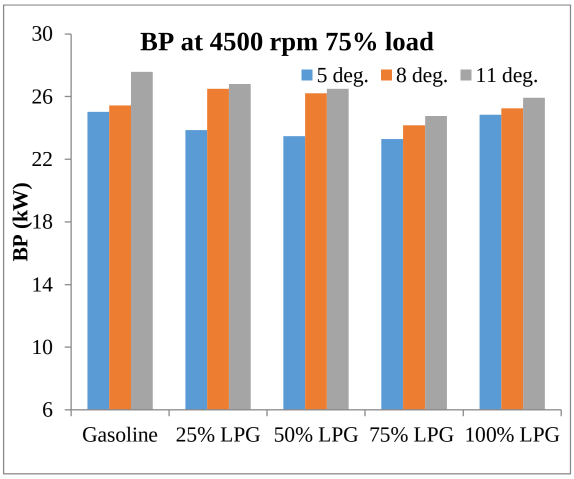


Fig 5.68 Effect of static ignition timing on BP at 3000 rpm full load condition.

Fig 5.69 Effect of static ignition timing on BP at 2000 rpm full load condition.



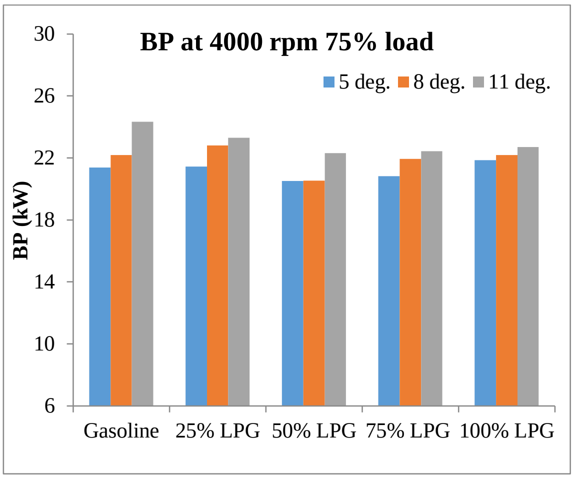


Fig 5.70 Effect of static ignition timing on BP at 4500 rpm 75% load condition.

Fig 5.71 Effect of static ignition timing on BP at 4000 rpm 75% load condition.

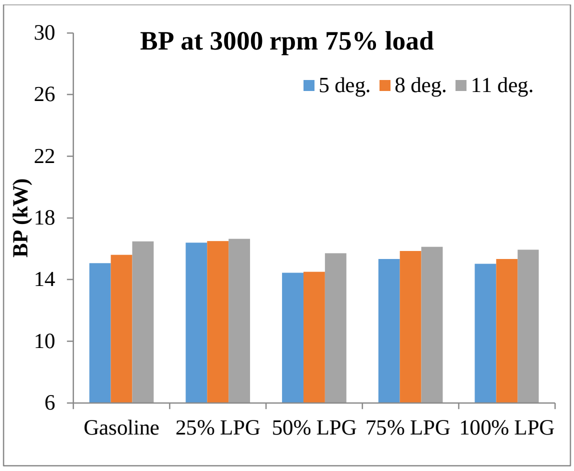


Fig 5.72 Effect of static ignition timing on BP at 3000 rpm 75% load condition.

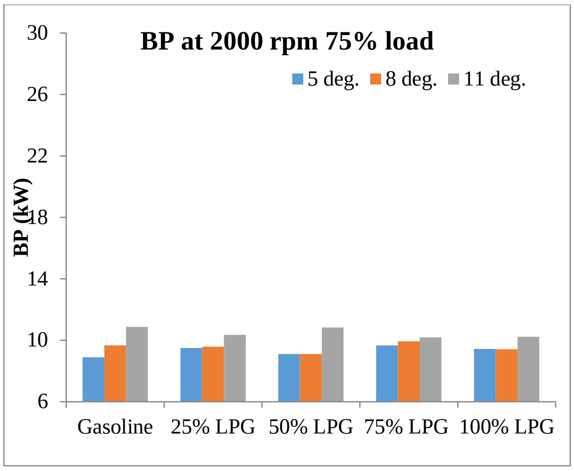


Fig 5.73 Effect of static ignition timing on BP at 2000 rpm 75% load condition.

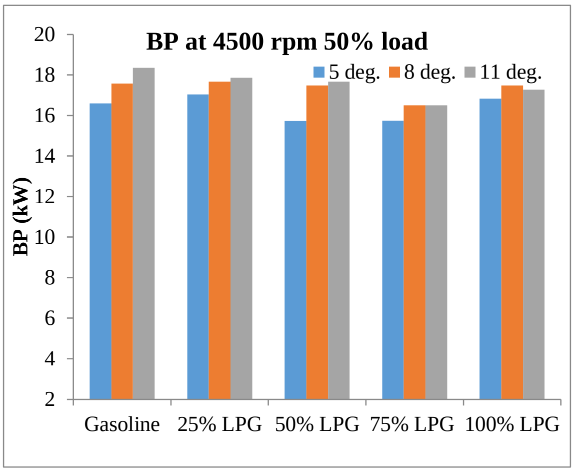


Fig 5.74 Effect of static ignition timing on BP at 4500 rpm 50% load condition.

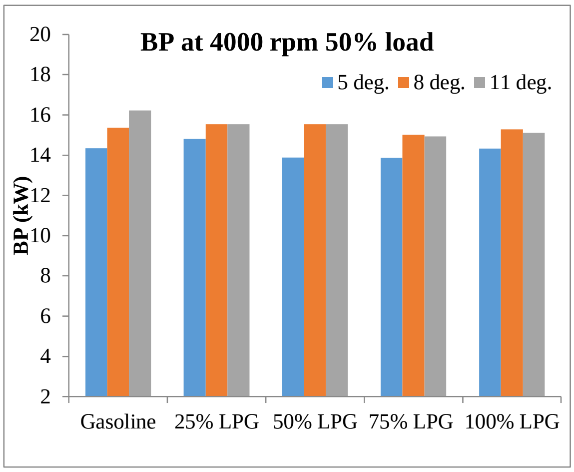


Fig 5.75 Effect of static ignition timing on BP at 4000 rpm 50% load condition.

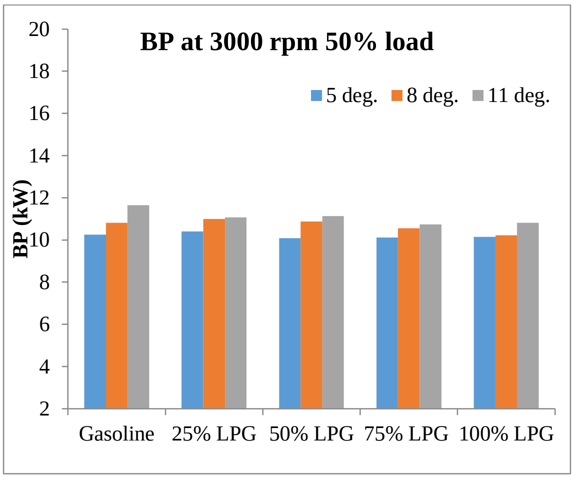


Fig 5.76 Effect of static ignition timing on BP at 3000 rpm 50% load condition.

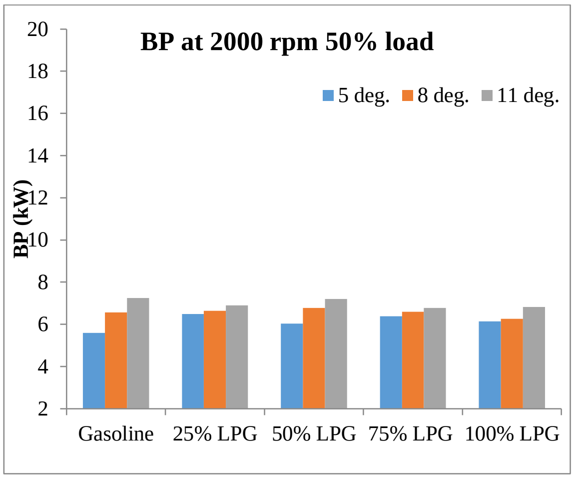


Fig 5.77 Effect of static ignition timing on BP at 2000 rpm 50% load condition.

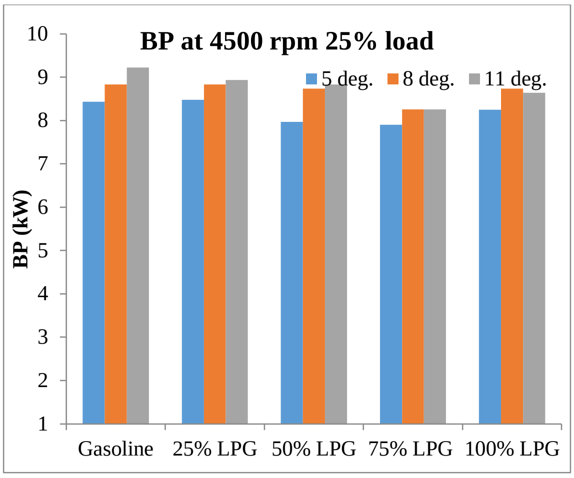


Fig 5.78 Effect of static ignition timing on BP at 4500 rpm 25% load condition.

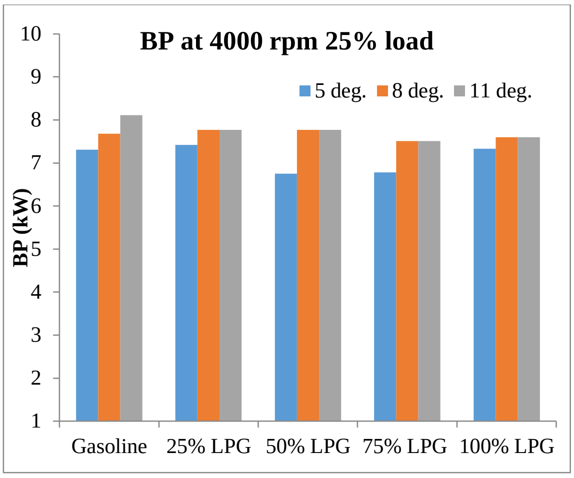


Fig 5.79 Effect of static ignition timing on BP at 4000 rpm 25% load condition.

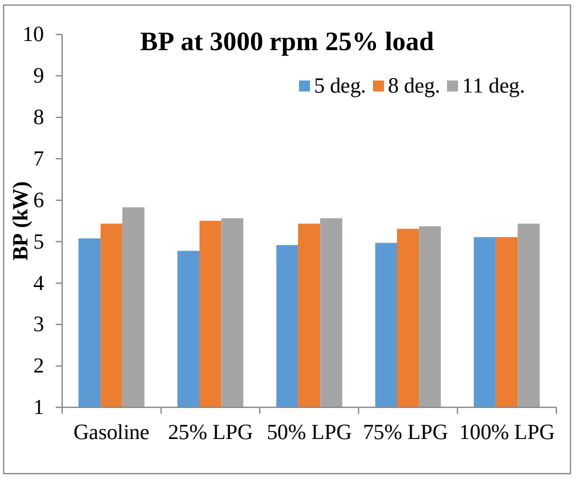


Fig 5.80 Effect of static ignition timing on BP at 3000 rpm 25% load condition.

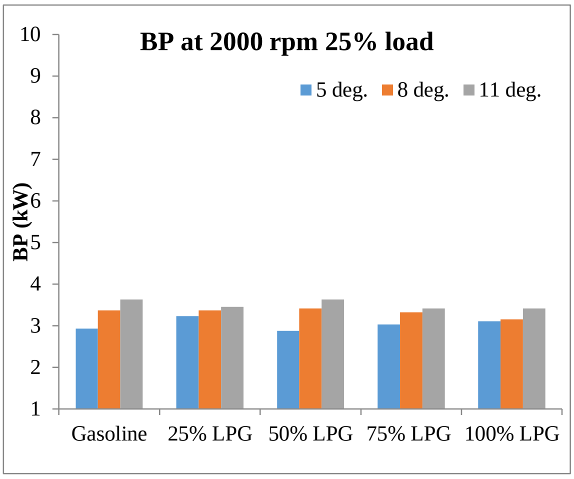


Fig 5.81 Effect of static ignition timing on BP at 2000 rpm 25% load condition.

5.2.1.2 Brake Thermal Efficiency (BTE)

The variations of Brake thermal efficiency with respect to static ignition timing for gasoline and dual fuel mode at all the engine speed and full load condition are shown in figure 5.82 to 5.85.

The comparison graph shows that the brake thermal efficiency increases with advancing the static ignition timing upto 8 deg. bTDC, beyond which decreases for gasoline. This is because too advancing the timing will results in higher compression work against the already burned gas which results in higher fuel consumption therefore decrease in thermal efficiency irrespective of increase in the power output. But as the percentage of LPG increases, BTE also increases with advancing the timing and it is maximum for 11 deg. bTDC at higher speeds. This because LPG having higher self-ignition temperature gives more time for combustion hence advancing the timing will results in higher thermal efficiency. But at lower speed since combustion duration is more for LPG, too advancing the timing will results higher fuel consumption which leads decrease in BTE. At lower engine speed 8deg. bTDC results showed better BTE. Average increase in BTE at 4500 rpm for gasoline is 8.89%, -4.79%, and for LPG is 8.56%, 14.36 at 11 deg. bTDC and 8 deg. bTDC respectively when compared to the 5deg. bTDC. Average increase in BTE at 2000 rpm for gasoline is 9.55%, 6.37%, and for LPG is 20.3%, 14.07 at 11 deg. bTDC and 8 deg. bTDC respectively when compared to the 5deg. bTDC.

Figures 5.86 to 5.89, 5.90 to 5.93 and 5.94 to 5.97 show the effect of static ignition timing on BTE for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively. As in full load condition similar trends are also observed in part load conditions. With advancing the static ignition timing BTE is increasing till 8 deg. bTDC then it starts decreasing for all fuel blend conditions. 75% load condition gives the maximum BTE because during this condition engine will work in the economy zone where fuel consumption is taken utmost care.

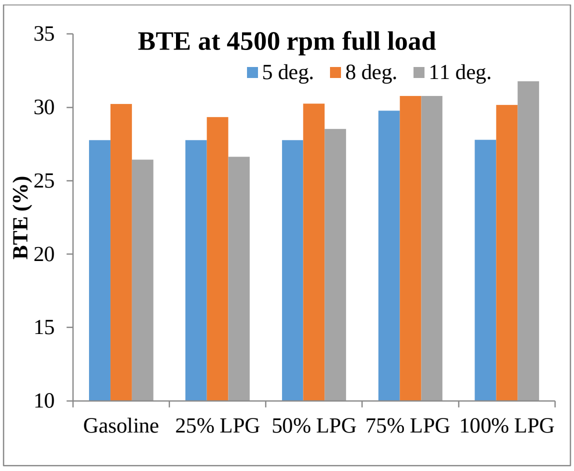


Fig 5.82 Effect of static ignition timing on BTE at 4500 rpm full load condition.

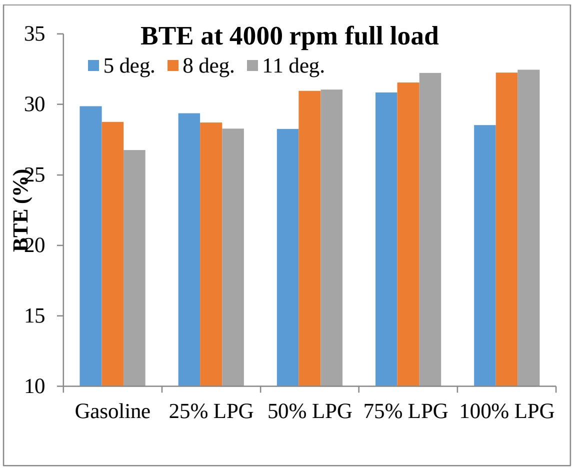


Fig 5.83 Effect of static ignition timing on BTE at 4000 rpm full load condition.

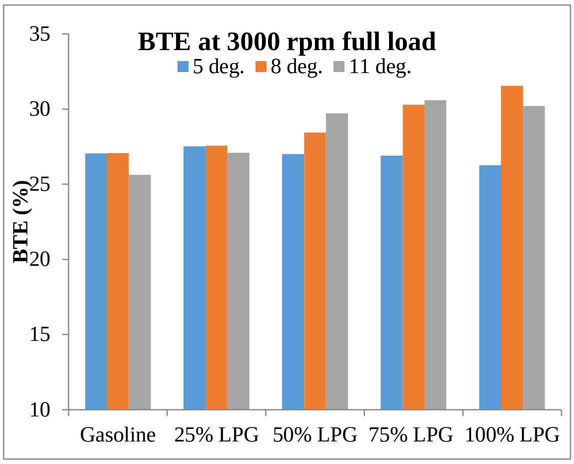


Fig 5.84 Effect of static ignition timing on BTE at 3000 rpm full load condition.

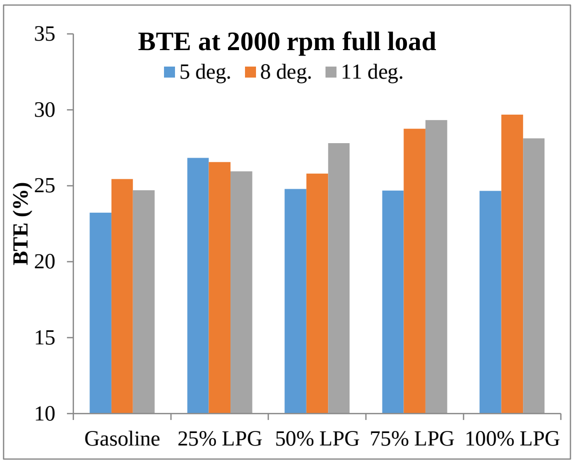


Fig 5.85 Effect of static ignition timing on BTE at 2000 rpm full load condition.

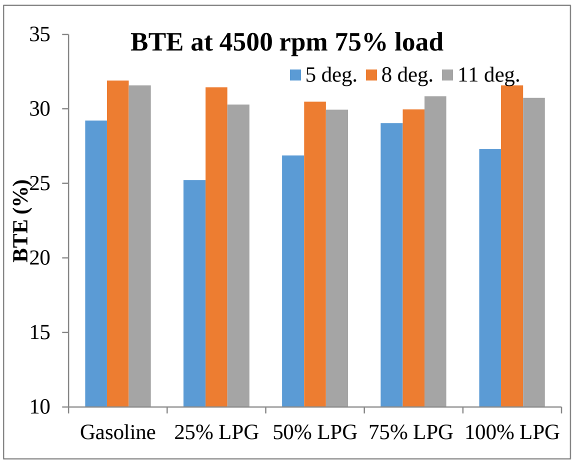


Fig 5.86 Effect of static ignition timing on BTE at 4500 rpm 75% load condition.

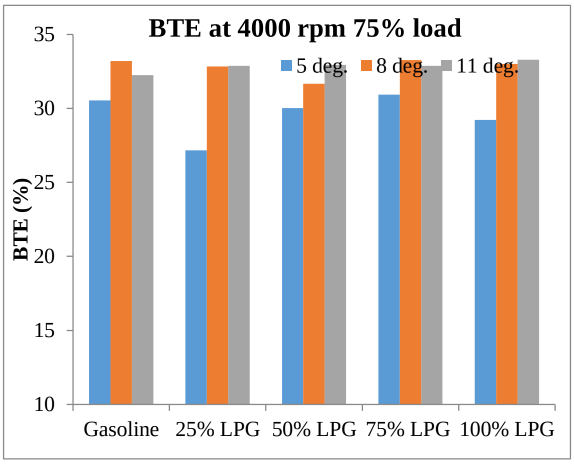


Fig 5.87 Effect of static ignition timing on BTE at 4000 rpm 75% load condition.

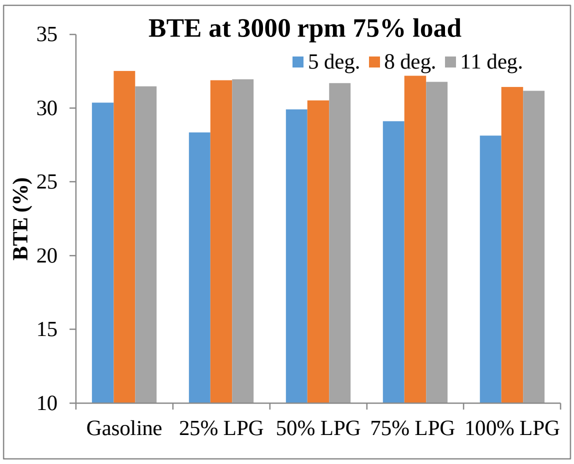


Fig 5.88 Effect of static ignition timing on BTE at 3000 rpm 75% load condition.

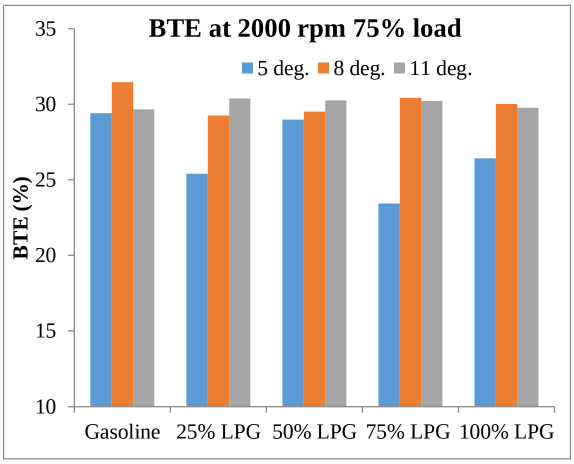


Fig 5.89 Effect of static ignition timing on BTE at 2000 rpm 75% load condition.

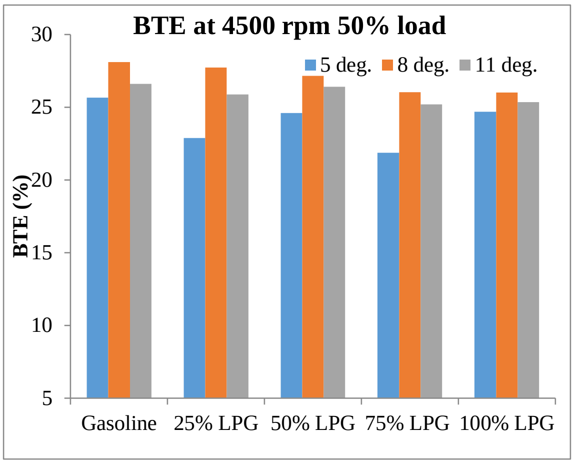


Fig 5.90 Effect of static ignition timing on BTE at 4500 rpm 50% load condition.

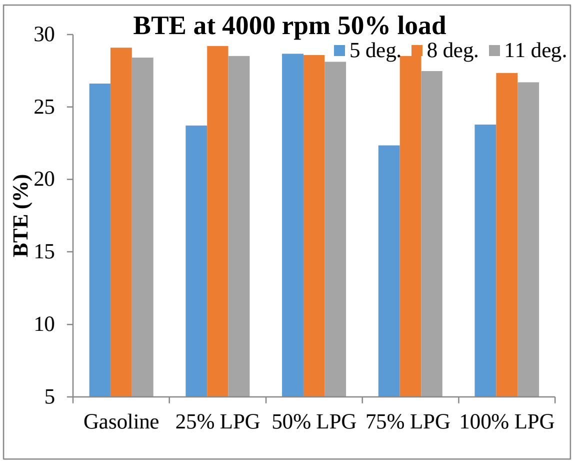


Fig 5.91 Effect of static ignition timing on BTE at 4000 rpm 50% load condition.

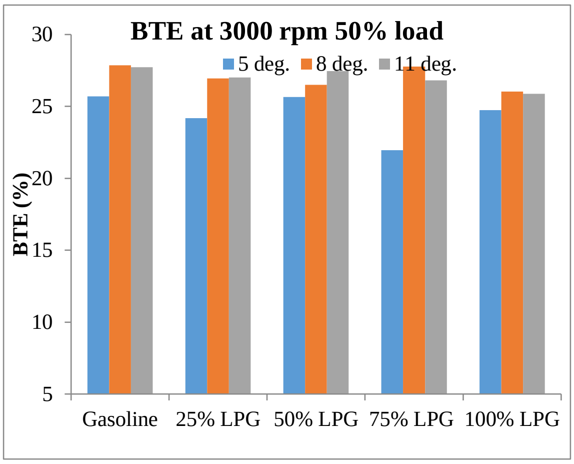


Fig 5.92 Effect of static ignition timing on BTE at 3000 rpm 50% load condition.

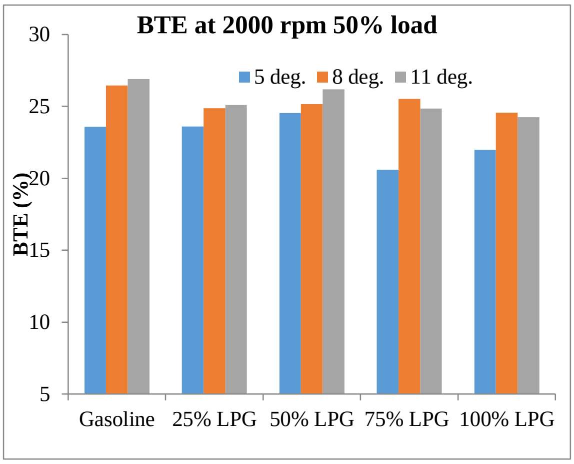


Fig 5.93 Effect of static ignition timing on BTE at 2000 rpm 50% load condition.

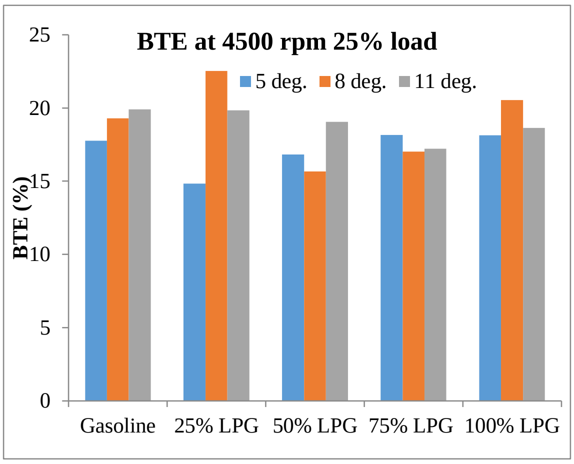


Fig 5.94 Effect of static ignition timing on BTE at 4500 rpm 25% load condition.

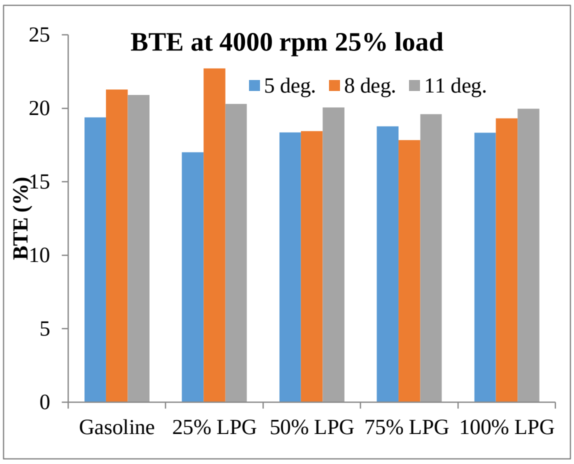


Fig 5.95 Effect of static ignition timing on BTE at 4000 rpm 25% load condition.

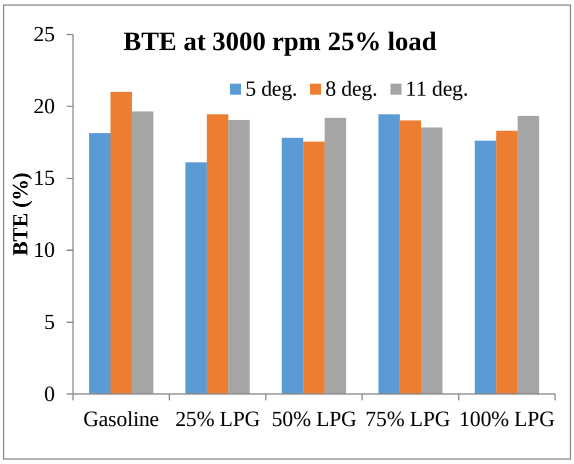


Fig 5.96 Effect of static ignition timing on BTE at 3000 rpm 25% load condition.

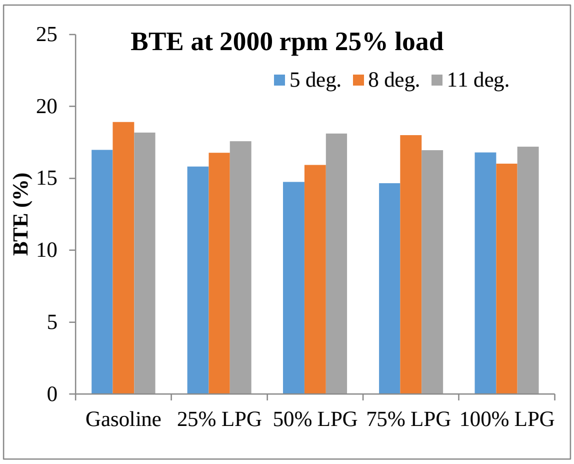


Fig 5.97 Effect of static ignition timing on BTE at 2000 rpm 25% load condition.

5.2.1.3 Equivalence Ratio (φ)

The figures 5.98 to 5.101 indicate the comparison of equivalence ratio with respect to static ignition timing for different speed and fuel blend at full load conditions. This equivalence ratio will determine the weather engine is working in leaner, stoichiometric or richer mixture. As the engine speed increases the equivalence ratio will try to remains in same value. But as static ignition timing advanced to 8 deg. bTDC engine started to work near to stoichiometric region for gasoline for all engine speed conditions. So advancing the static ignition timing resulted in the better combustion properties for gasoline. Still advancing the static ignition timing resulted in the richer mixture in the combustion chamber for gasoline.

As the percentage of LPG increases in the fuel blend, fuel-air ratio within the combustion chamber are coming closer to stoichiometric region which results in the better combustion properties. But advancing the static ignition timing there is no much variations in the equivalence ratio for the LPG fuel blend. At lower speed for 100% LPG advancing static timing to 11 deg. bTDC resulted in slightly richer region because of more fuel consumption.

Figures 5.102 to 5.105, 5.106 to 5.109 and 5.110 to 5.113 show the effect of static ignition timing on equivalence ratio for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively. At lower load, advancing the static ignition timing for gasoline resulted in decrease in equivalence ratio value for 8 deg. bTDC. Which means 8 deg. bTDC gives the better combustible properties for gasoline at lower load also and engine will work near to stoichiometric region. Still advancing the static ignition timing for gasoline resulted in slightly richer mixture when compared to 8 deg. bTDC for all lower loads. But as the percentage of LPG increased in the fuel blend there is not much variation in the equivalence ratio at 75% and 50% load conditions. At 25% load and 100% LPG, engine works in lean combustion zone where fuel economy plays important role.

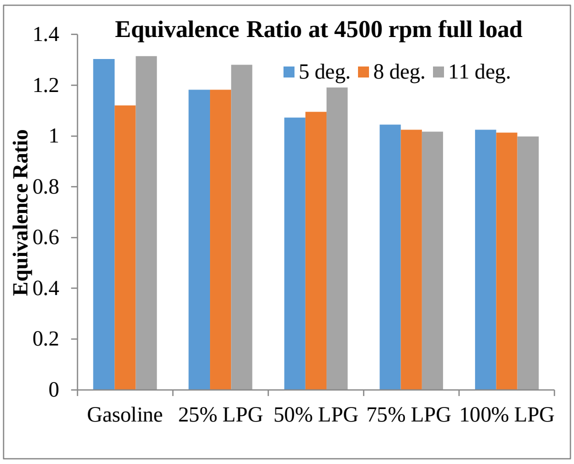


Fig 5.98 Effect of static ignition timing on φ at 4500 rpm full load condition.

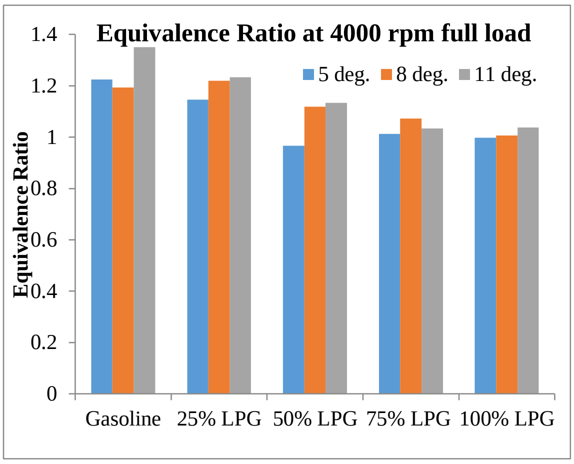


Fig 5.99 Effect of static ignition timing on φ at 4000 rpm full load condition.

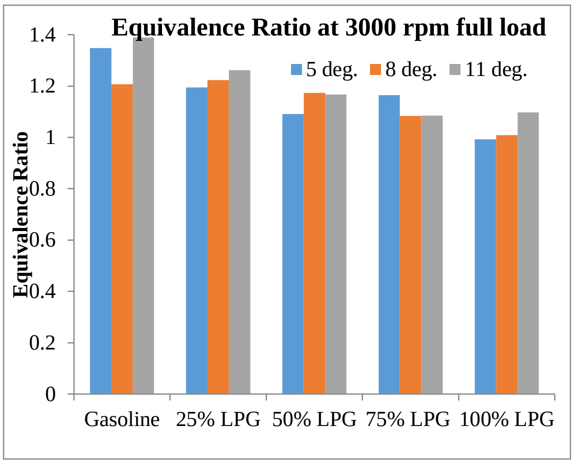


Fig 5.100 Effect of static ignition timing on φ at 3000 rpm full load condition.

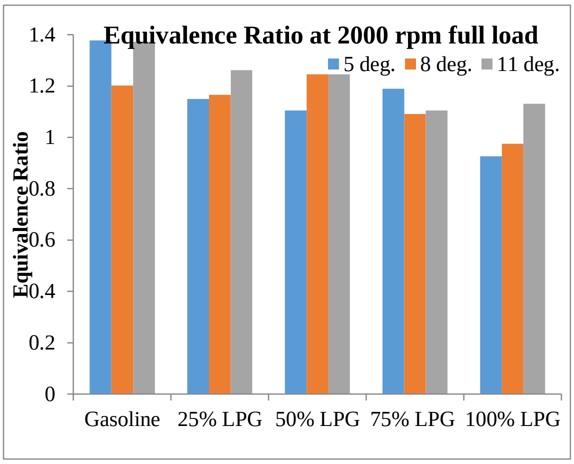
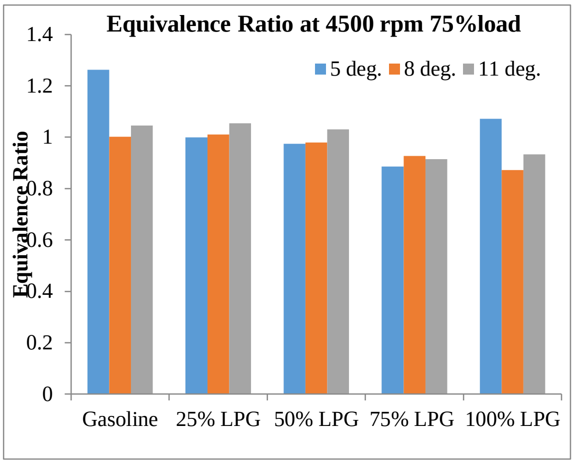


Fig 5.101 Effect of static ignition timing on φ at 2000 rpm full load condition.



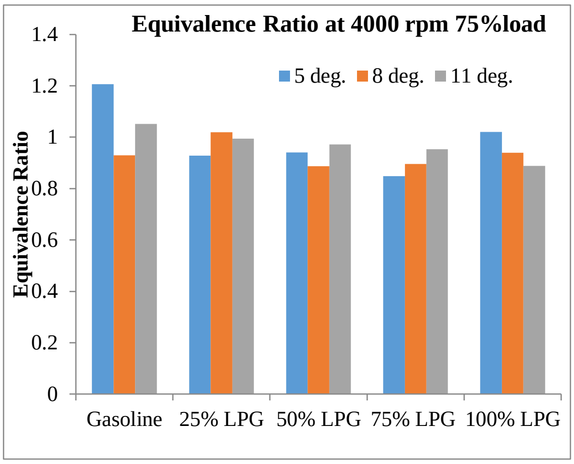
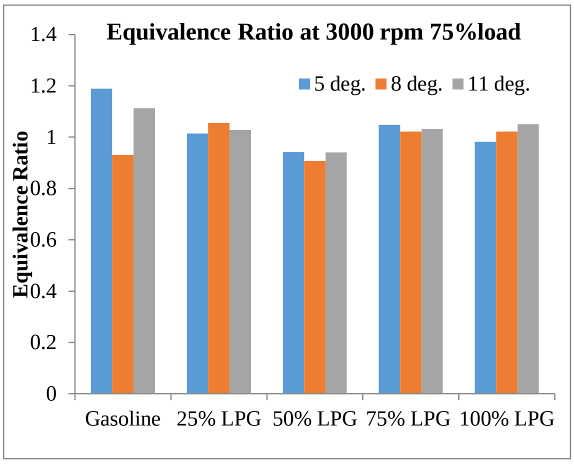


Fig 5.102 Effect of static ignition timing on φ at 4500 rpm 75% load condition.

Fig 5.103 Effect of static ignition timing on φ at 4000 rpm 75% load condition.



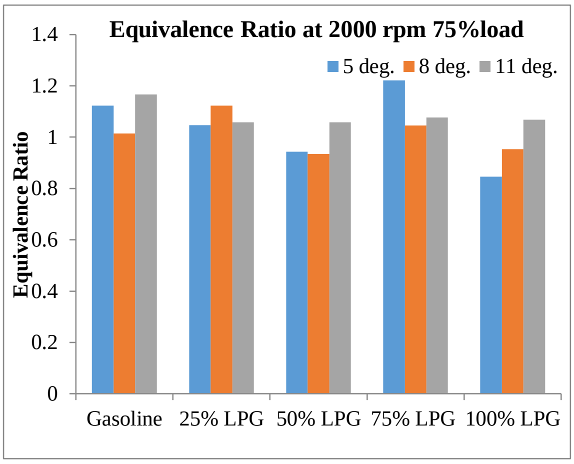
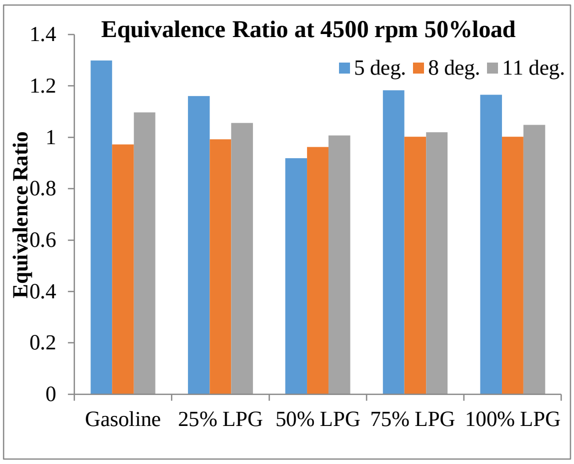


Fig 5.104 Effect of static ignition timing on φ at 3000 rpm 75% load condition.

Fig 5.105 Effect of static ignition timing on φ at 2000 rpm 75% load condition.



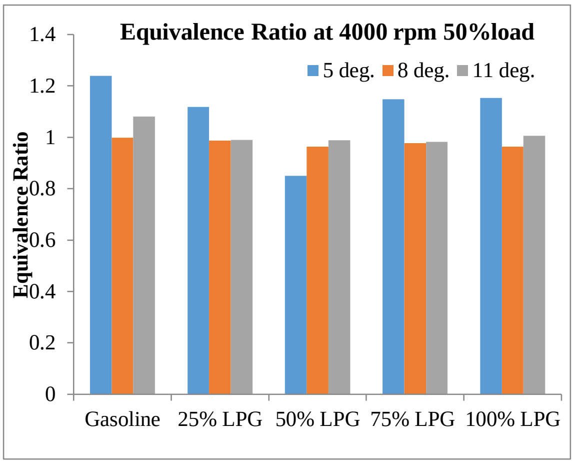


Fig 5.106 Effect of static ignition timing on φ at 4500 rpm 50% load condition.

Fig 5.107 Effect of static ignition timing on φ at 4000 rpm 50% load condition.

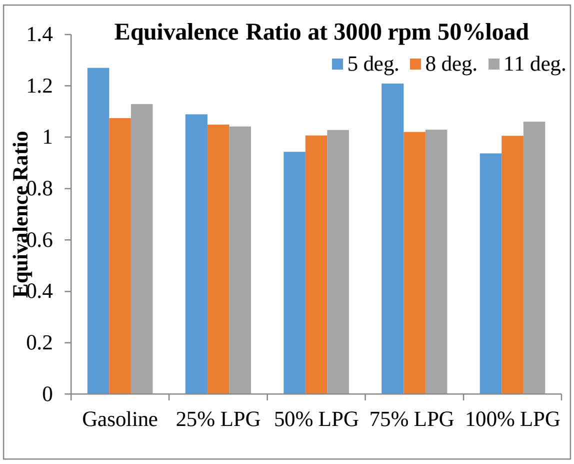


Fig 5.108 Effect of static ignition timing on φ at 3000 rpm 50% load condition.

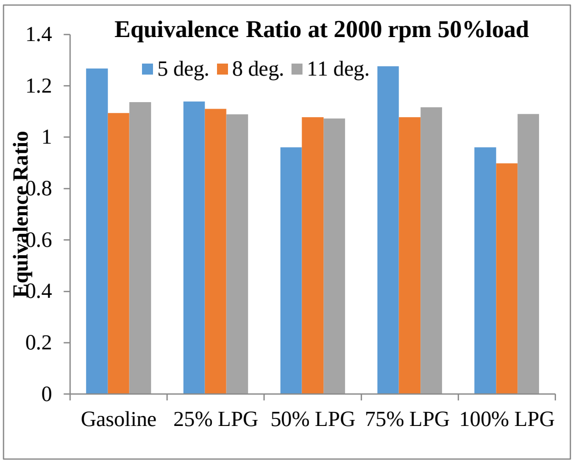


Fig 5.109 Effect of static ignition timing on φ at 2000 rpm 50% load condition.

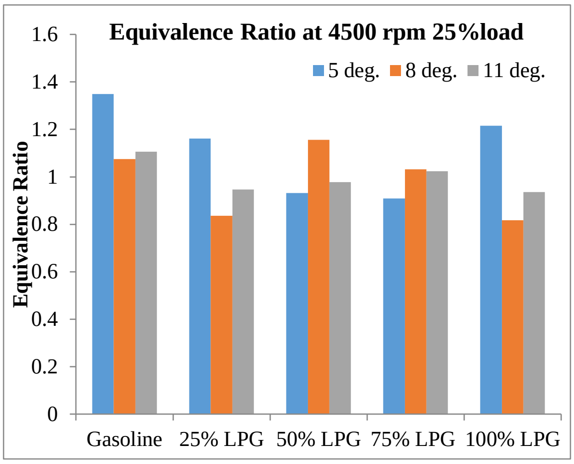


Fig 5.110 Effect of static ignition timing on φ at 4500 rpm 25% load condition.

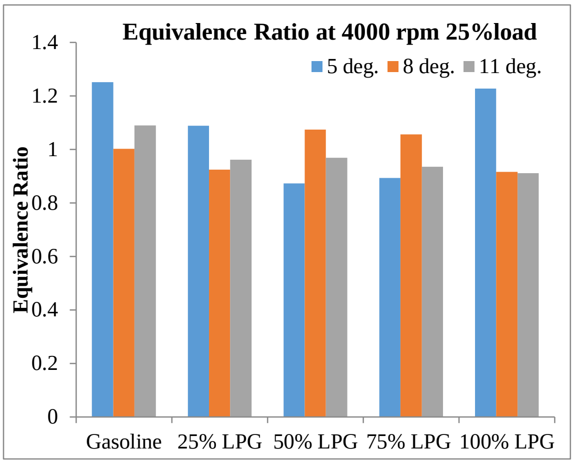


Fig 5.111 Effect of static ignition timing on φ at 4000 rpm 25% load condition.

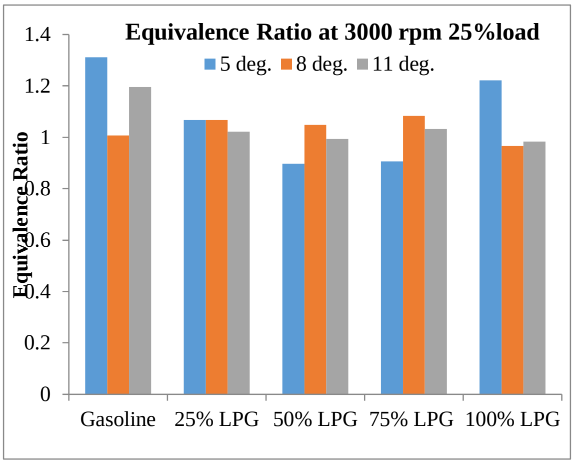


Fig 5.112 Effect of static ignition timing on φ at 3000 rpm 25% load condition.

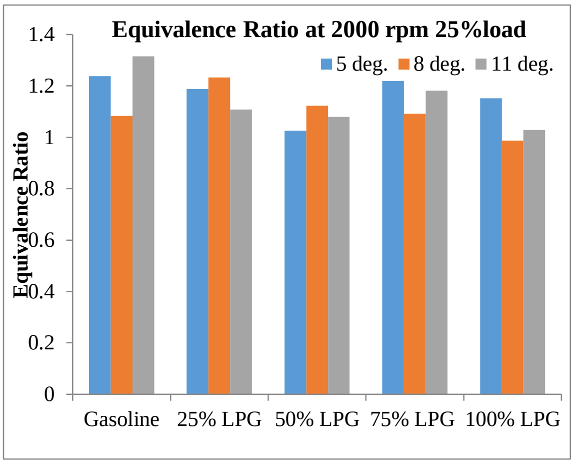
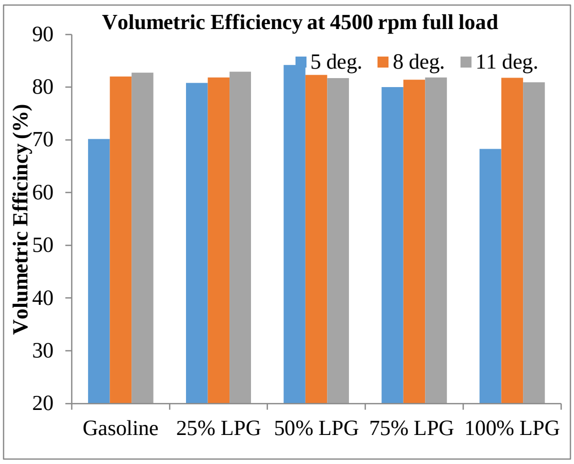


Fig 5.113 Effect of static ignition timing on φ at 2000 rpm 25% load condition.

5.2.1.4 Volumetric Efficiency

Volumetric efficiency is a measure of effectiveness of an engine’s induction process. The induction system includes the air-filter, injector and throttle plate in spark ignition engine. They all restrict the amount of air which an engine of given displacement can induct. Volumetric efficiency of a normally aspirated engine is the ratio of the actual volume flow rate of air into the cylinder at atmospheric pressure and temperature conditions surrounding to the engine to the rate at which the volume displaced by the piston. The figures 5.114 to 5.117 compares volumetric efficiency with respect to static ignition timing for different speed and fuel blend at full load condition. From the results it is revealed that for gasoline and 100% LPG as the static ignition timing is advanced volumetric efficiency is going to increase and rest fuel conditions there not much variations. Since the equivalence ratio is going to leaner side with advancing the static ignition timing for gasoline and 100% LPG fuel, volumetric efficiency will increases. For gasoline maximum volumetric efficiency found to be 82.73% at 11 deg. bTDC, whereas at 8 deg. bTDC it is 82.03% and 5 deg. bTDC 70.20%. Volumetric efficiency values for 100% LPG are 68.26%, 81.76% and 80.92% at 5,8 and 11 deg. bTDC respectively.

Figures 5.118 to 5.121, 5.122 to 5.125 and 5.126 to 5.129 show the effect of static ignition timing on volumetric efficiency for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively. At part load condition since the throttle position reduces and pumping losses will increases hence volumetric reduces as the load decreases. At part load similar trends are observed as that of full load conditions. Maximum volumetric efficiency is obtained for 100% LPG fuel in each load condition because it will work in a leaner region compared to other fuel blend.



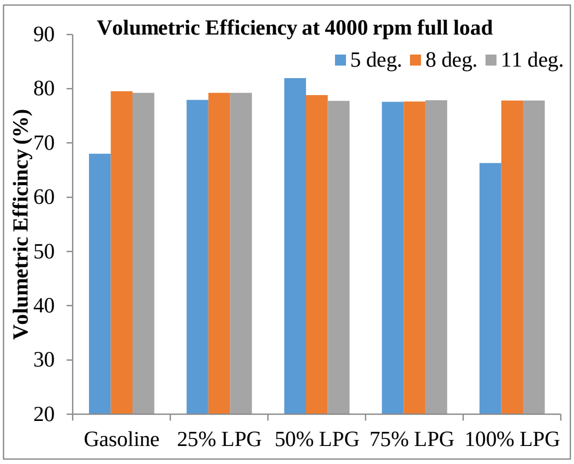
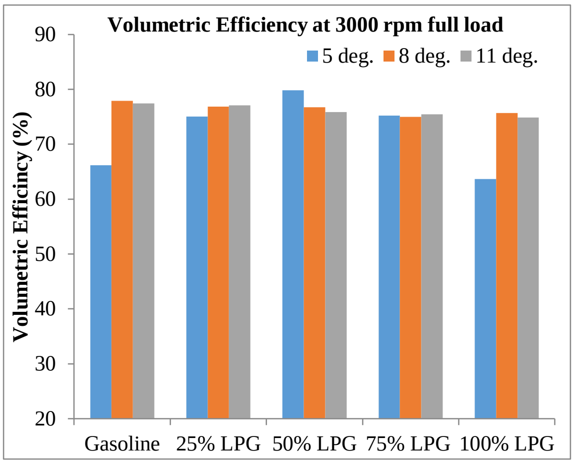


Fig 5.114 Effect of static ignition timing on Vol. Eff. at 4500 rpm full load condition.

Fig 5.115 Effect of static ignition timing on Vol. Eff. at 4000 rpm full load condition.



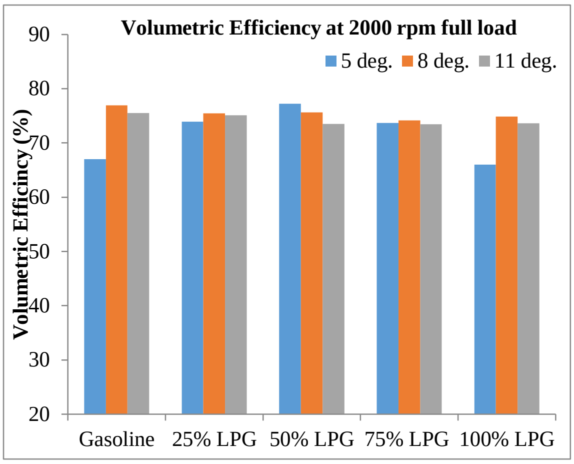
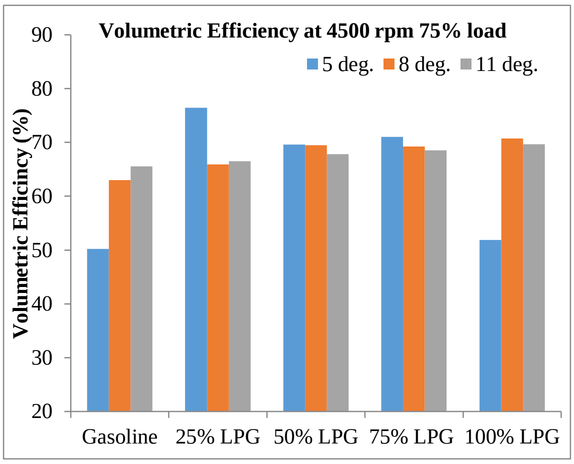


Fig 5.116 Effect of static ignition timing on Vol. Eff. at 3000 rpm full load condition.

Fig 5.117 Effect of static ignition timing on Vol. Eff.at 2000 rpm full load condition.



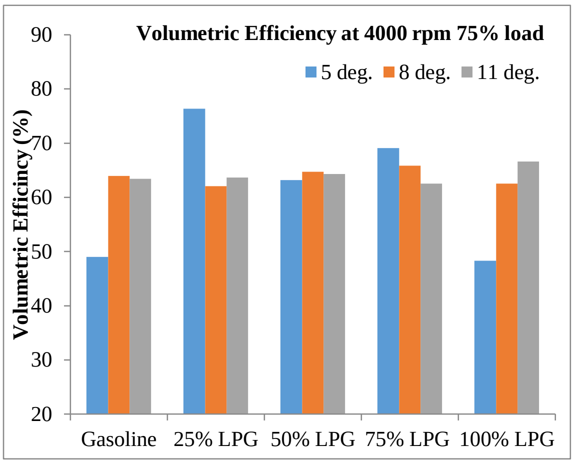


Fig 5.118 Effect of static ignition timing on Vol. Eff. at 4500 rpm 75% load condition.

Fig 5.119 Effect of static ignition timing on Vol. Eff. at 4000 rpm 75% load condition.

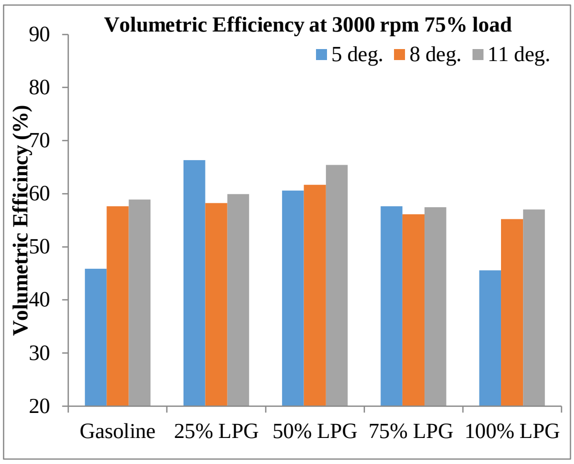


Fig 5.120 Effect of static ignition timing on Vol. Eff. at 3000 rpm 75% load condition.

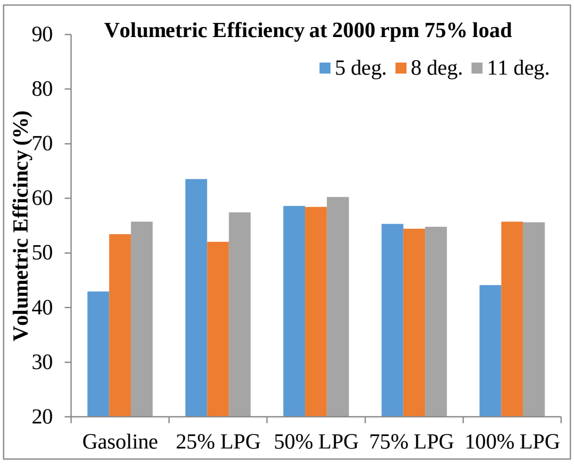


Fig 5.121 Effect of static ignition timing on Vol. Eff. at 2000 rpm 75% load condition.

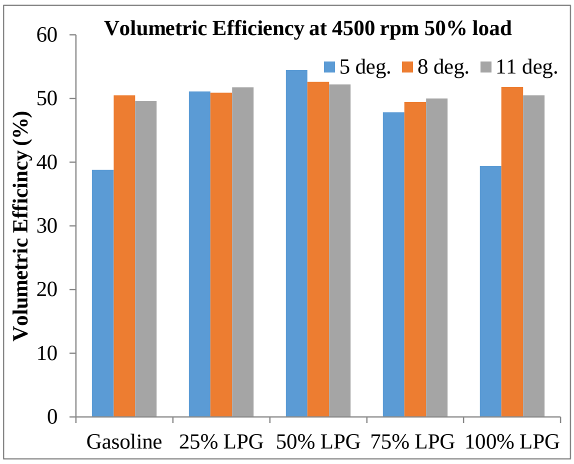


Fig 5.122 Effect of static ignition timing on Vol. Eff. at 4500 rpm 50% load condition.

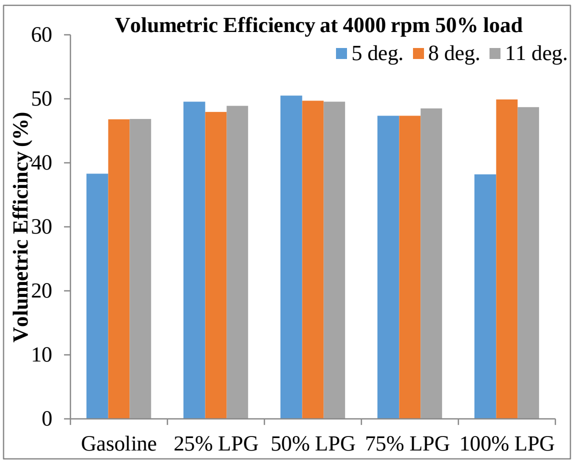


Fig 5.123 Effect of static ignition timing on Vol. Eff. at 4000 rpm 50% load condition.

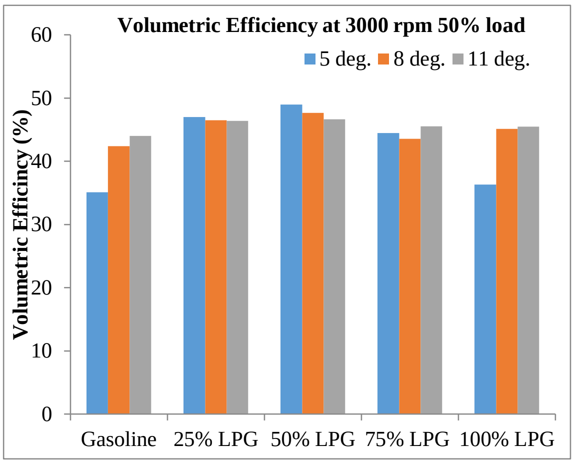


Fig 5.124 Effect of static ignition timing on Vol. Eff. at 3000 rpm 50% load condition.

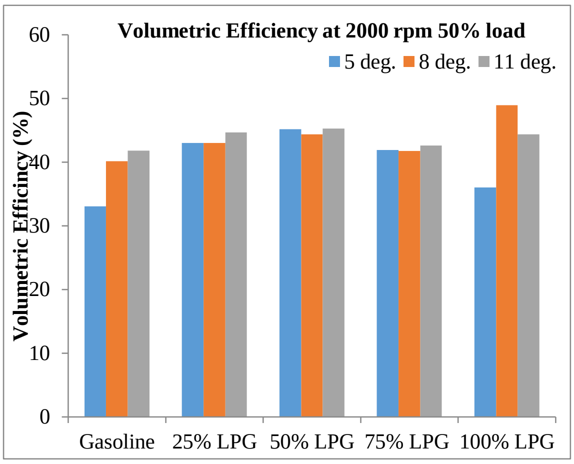


Fig 5.125 Effect of static ignition timing on Vol. Eff. at 2000 rpm 50% load condition.

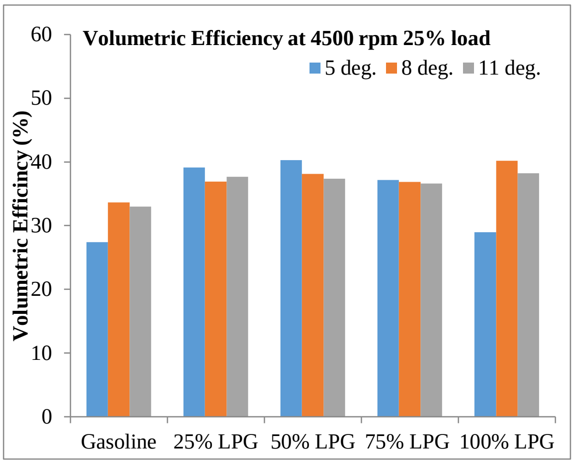


Fig 5.126 Effect of static ignition timing on Vol. Eff. at 4500 rpm 25% load condition.

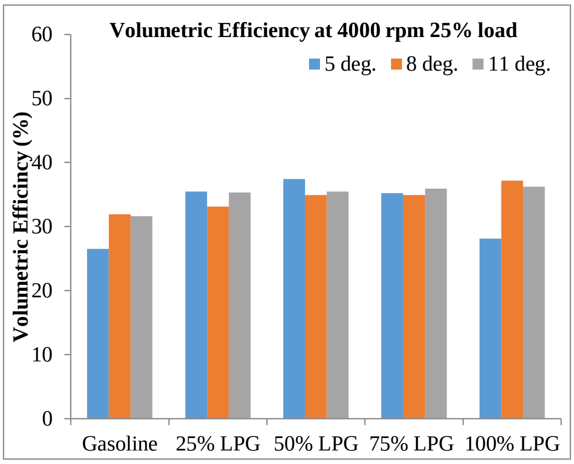


Fig 5.127 Effect of static ignition timing on Vol. Eff. at 4000 rpm 25% load condition.

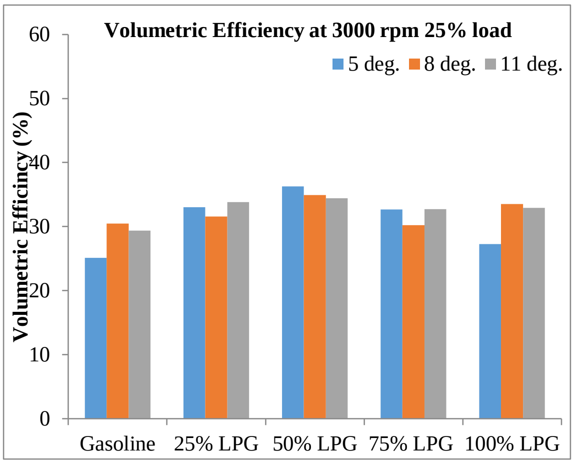


Fig 5.128 Effect of static ignition timing on Vol. Eff. at 3000 rpm 25% load condition.

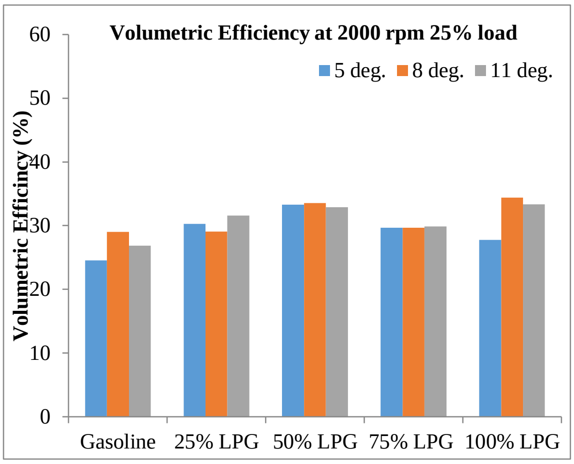


Fig 5.129 Effect of static ignition timing on Vol. Eff. at 2000 rpm 25% load condition.

5.2.1.5. Brake Specific Energy Consumption (BSEC)

Brake specific energy consumption will indicate to produce one kilowatt of power in one hour how much energy is consumed. This comparison will give the fuel economy for an engine where different calorific value of fuels are being used. The figures 5.130 to 5.133 compares the BSEC with respect to static ignition timing for different speed and fuel blend at full load condition. As the static ignition timing is advanced to 8 deg. bTDC the energy consumed to produce will be less or gasoline for all engine speed condition. Further increase in the timing will lead to increase in the energy consumption and intern fuel economy. This reduction in the energy consumption is because gasoline will work near to stoichiometric region and fuel consumption also reduced so energy consumed also reduced.

As the percentage of LPG increased, advancing the timing will results in reduction in the energy consumption and it is minimum for 11 deg. bTDC. 100% LPG gives the minimum energy consumption at 11 deg. bTDC. Since LPG is having higher calorific value than gasoline which results in the lower fuel consumption hence energy consumed for LPG will be minimum. Average decrease in the BSEC value for gasoline is 8.18% and -5.02% at 8 and 11 deg. bTDC respectively when compared to 5 deg. bTDC at 4500 rpm. Similarly for 100% LPG is 7.84% and 12.55% at 8 and 11 deg. bTDC respectively when compared to 5 deg. bTDC at 4500 rpm.

Figures 5.134 to 5.137, 5.138 to 5.141 and 5.142 to 5.145 show the effect of static ignition timing on BSEC for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively. As in full load condition similar trends are also observed in part load conditions. With advancing the static ignition timing BSEC is decreasing till 8 deg. bTDC then it starts increasing for all fuel blend conditions. 75% load condition gives the minimum BSEC because during this condition engine will work in the economy zone where fuel consumption is taken utmost care.

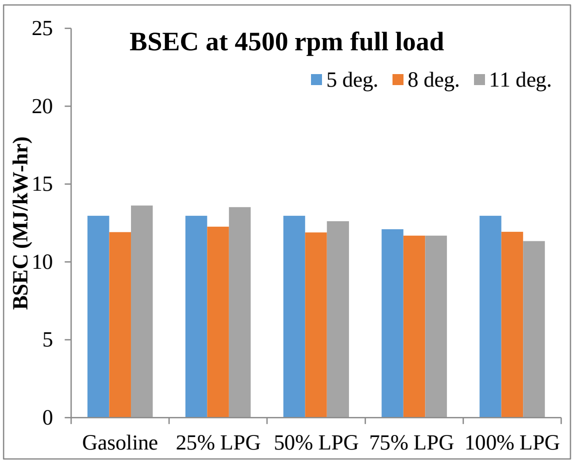


Fig 5.130 Effect of static ignition timing on BSEC at 4500 rpm full load condition.

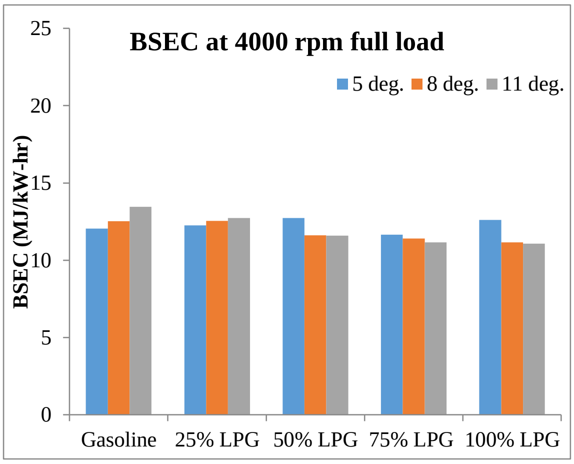


Fig 5.131 Effect of static ignition timing on BSEC at 4000 rpm full load condition.

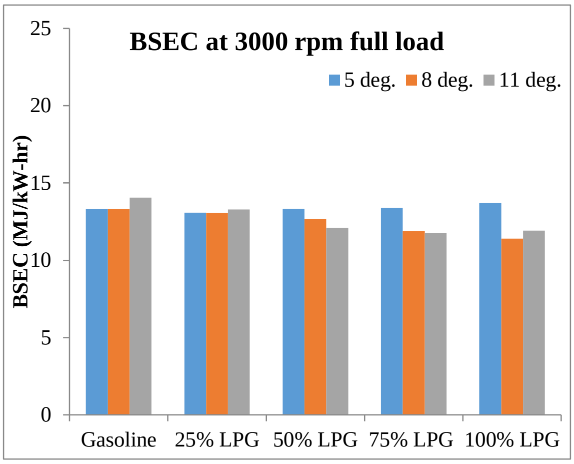


Fig 5.132 Effect of static ignition timing on BSEC at 3000 rpm full load condition.

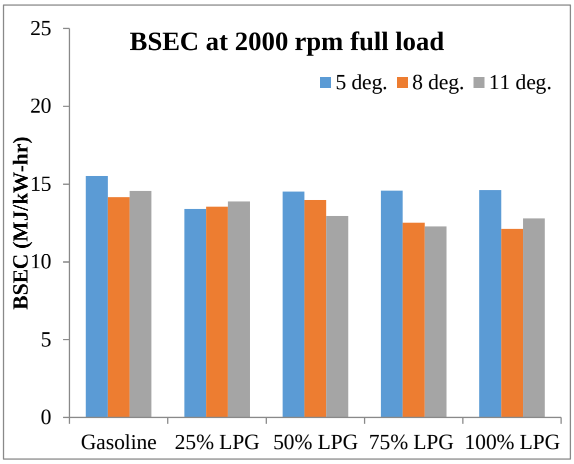


Fig 5.133 Effect of static ignition timing on BSEC at 2000 rpm full load condition.

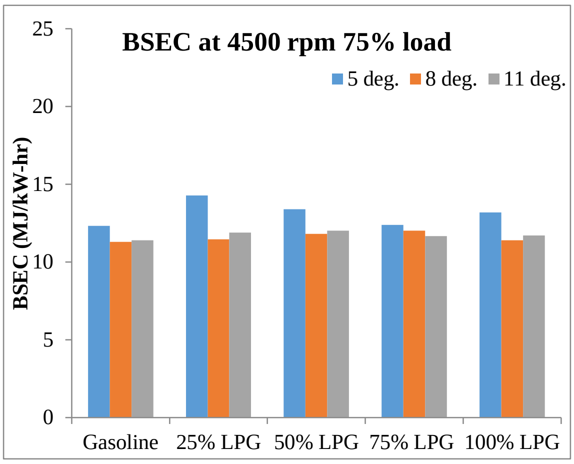


Fig 5.134 Effect of static ignition timing on BSEC at 4500 rpm 75% load condition.

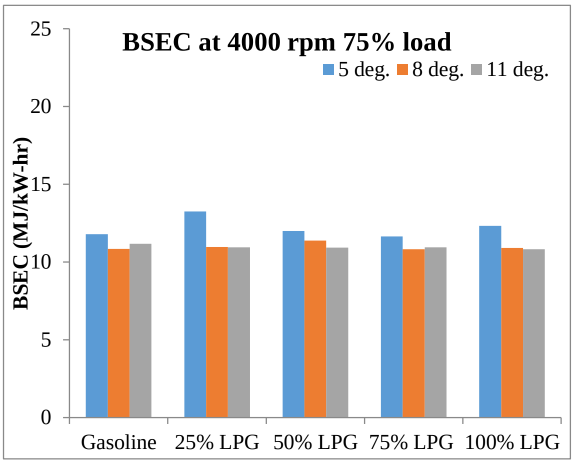


Fig 5.135 Effect of static ignition timing on BSEC at 4000 rpm 75% load condition.

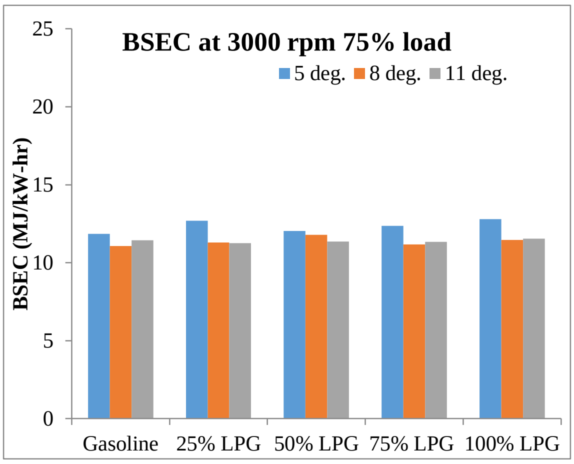


Fig 5.136 Effect of static ignition timing on BSEC at 3000 rpm 75% load condition.

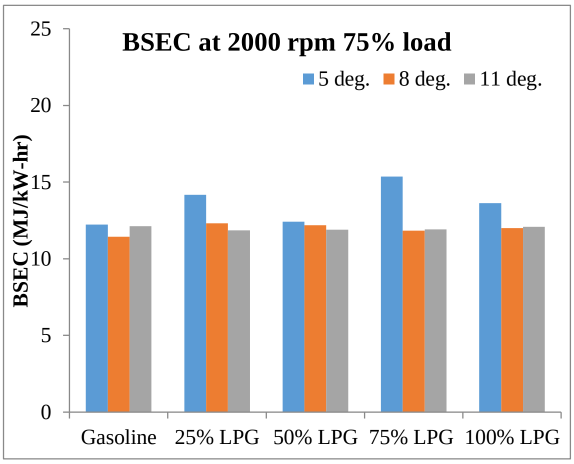


Fig 5.137 Effect of static ignition timing on BSEC at 2000 rpm 75% load condition.

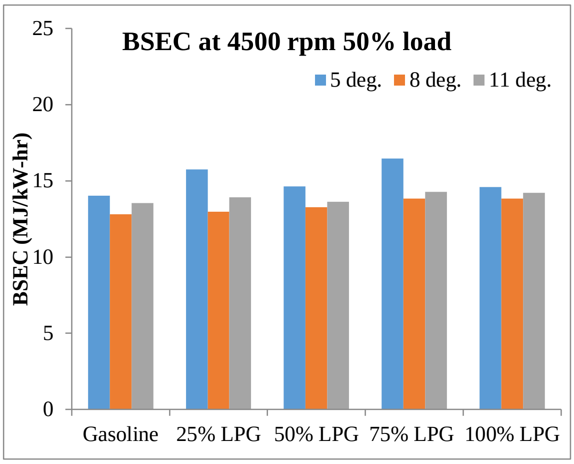


Fig 5.138 Effect of static ignition timing on BSEC at 4500 rpm 50% load condition.

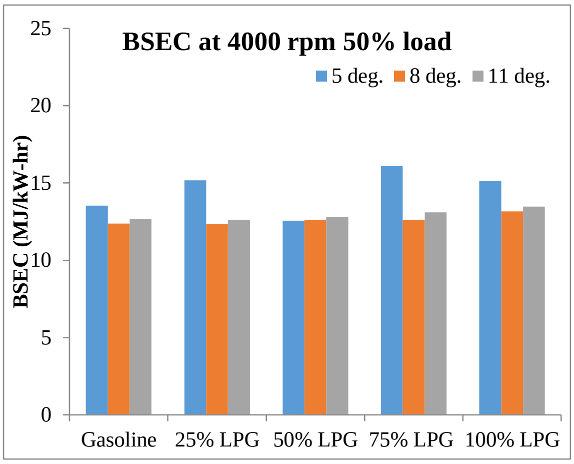


Fig 5.139 Effect of static ignition timing on BSEC at 4000 rpm 50% load condition.

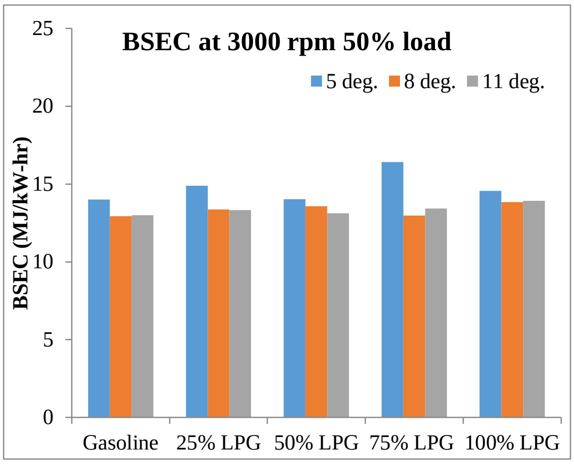


Fig 5.140 Effect of static ignition timing on BSEC at 3000 rpm 50% load condition.

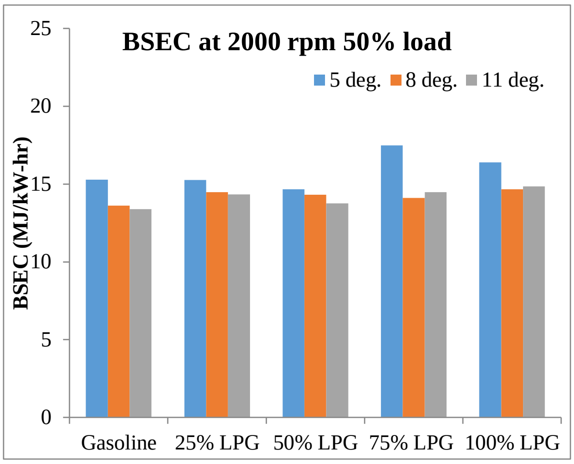


Fig 5.141 Effect of static ignition timing on BSEC at 2000 rpm 50% load condition.

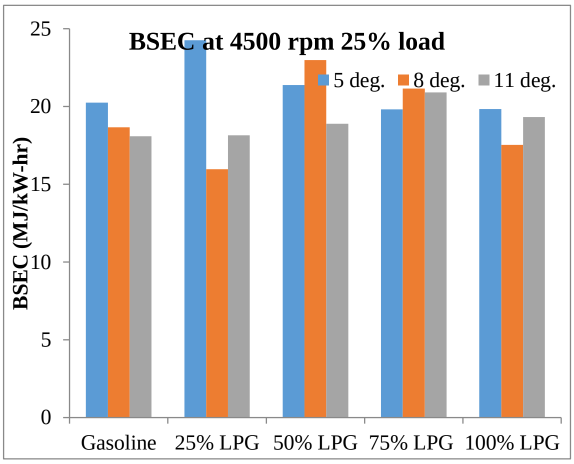


Fig 5.142 Effect of static ignition timing on BSEC at 4500 rpm 25% load condition.

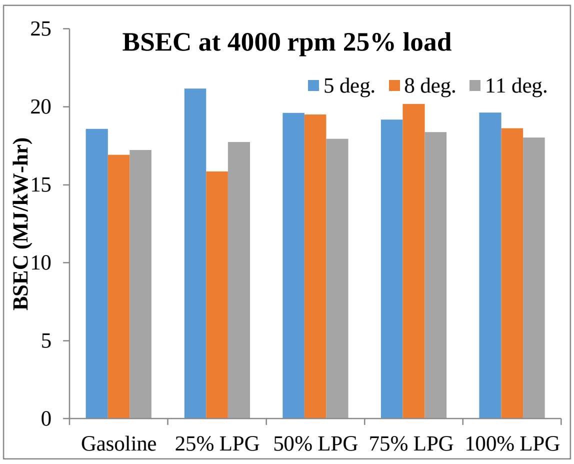


Fig 5.143 Effect of static ignition timing on BSEC at 4000 rpm 25% load condition.

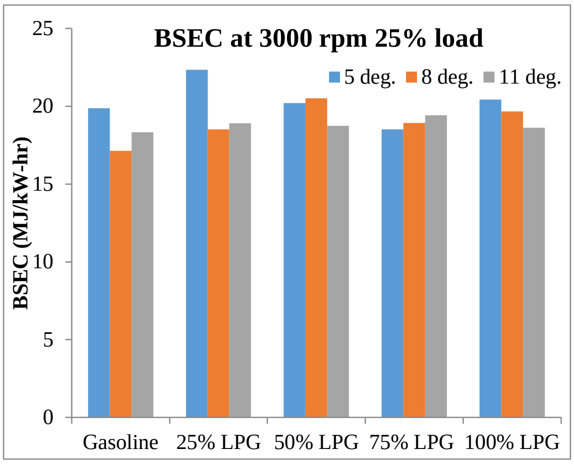


Fig 5.144 Effect of static ignition timing on BSEC at 3000 rpm 25% load condition.

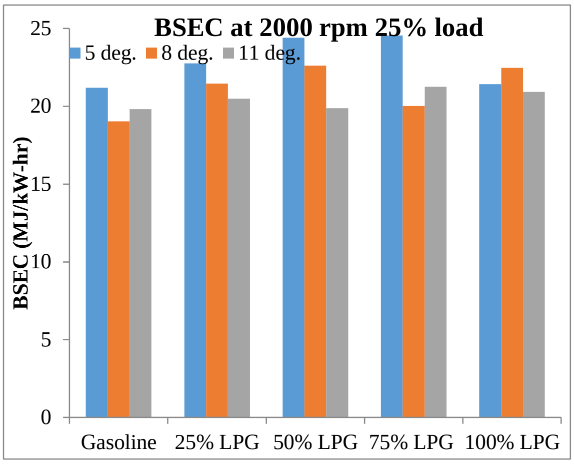


Fig 5.145 Effect of static ignition timing on BSEC at 2000 rpm 25% load condition.

5.2.2 Emission Characteristics

5.2.2.1 CO emissions

The comparison of CO emissions with respect to static ignition timing for full load condition at different speed and fuel blend operation are shown in the figure 5.146 to 5.149. CO emission is mainly influenced by the oxygen availability in the combustion zone. Hence CO emission expected to be continuing to decrease as the air-fuel mixture becomes leaner. Also CO emission requires higher exhaust temperature and longer residence time to oxidize. At full load condition as the percentage of LPG increases CO emission going to be reduced since LPG is working near to stoichiometric region and higher flame propagation speed of LPG. As advancing the static ignition timing not remarkable changes found in CO emission. At higher speed condition all the fuel blend shown decreasing characteristic for CO emission. But as the speed decreased, exhaust gas temperature and residence time reduces hence CO emission going to slightly increases for all fuel blend except for 100% LPG till 8 deg. bTDC then again reduces. Very low CO emission is obtained for 100% LPG. Average decrease in the CO emission value at 4500 rpm for gasoline is 8.34% and 14.7% at 8 and 11 deg. bTDC respectively when compared to 5 deg. bTDC. Average decrease in the CO emission value at 4500 rpm for 100% LPG is 90% and 88% at 8 and 11 deg. bTDC respectively when compared to 5 deg. bTDC.

Figures 5.150 to 5.153, 5.154 to 5.157 and 5.158 to 5.161 show the effect of static ignition timing on CO emission for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively. From the figures it can be inferred that CO emissions of LPG is far less than that of gasoline. For 100% LPG, at all load condition the CO emissions are found to be less than 2% which is well within the limits of EURO 5 pollution norms. The higher flame propagation speed and proper mixing of gaseous LPG with the air enhances the combustion and thus reduces the CO emissions. As the static ignition timing is advanced at lower load condition, CO emission is going to decrease till 8 deg. bTDC then starts increasing for 11 deg. bTDC for all fuel blend except for gasoline and 25% LPG. This is because as the ignition timing is too advanced combustion starts earlier in the cycle and net exhaust gas temperature will reduce hence increase in the CO emission obtained. As the percentage of LPG increased, CO emission is going to decrease at lower load also.

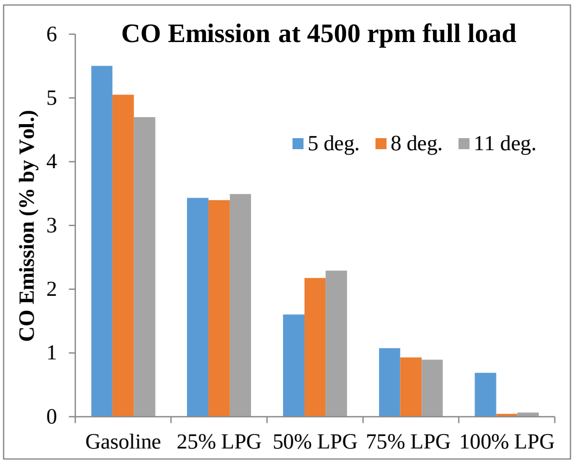


Fig 5.146 Effect of static ignition timing on CO at 4500 rpm full load condition.

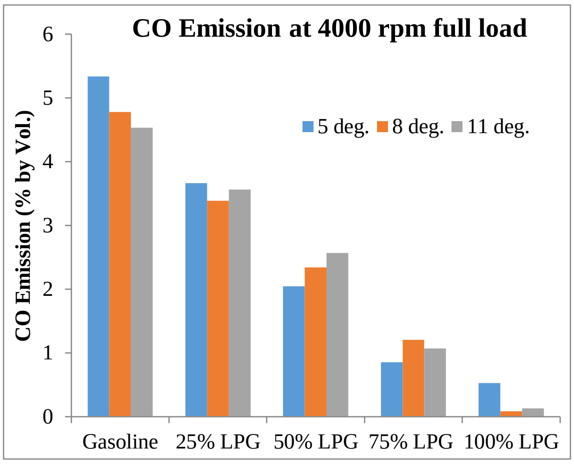
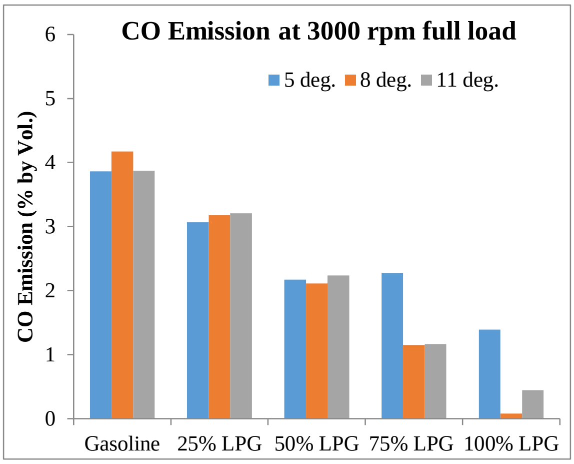


Fig 5.147 Effect of static ignition timing on CO at 4000 rpm full load condition.



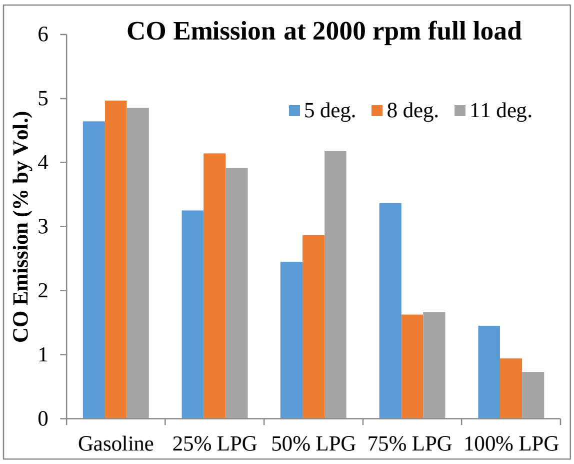
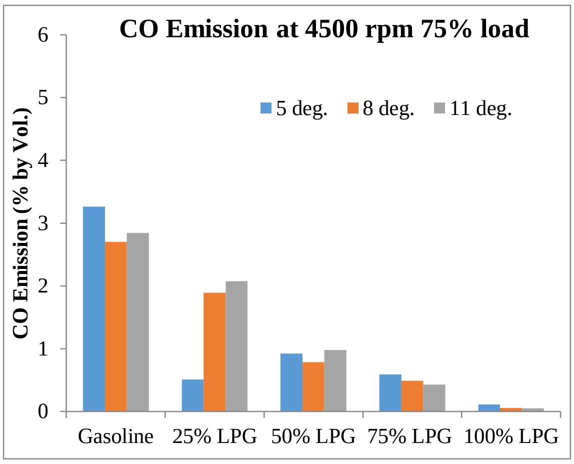


Fig 5.148 Effect of static ignition timing on CO at 3000 rpm full load condition.

Fig 5.149 Effect of static ignition timing on CO at 2000 rpm full load condition.



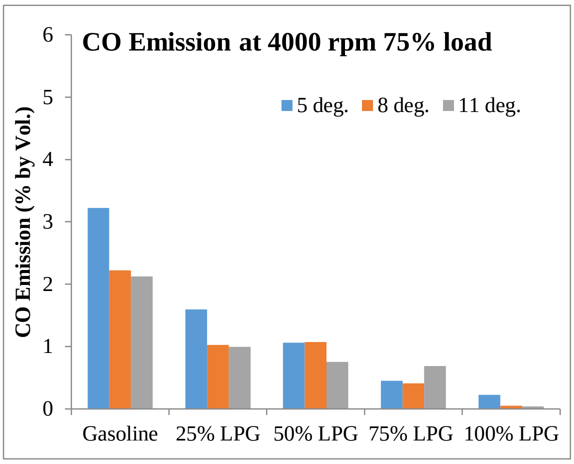
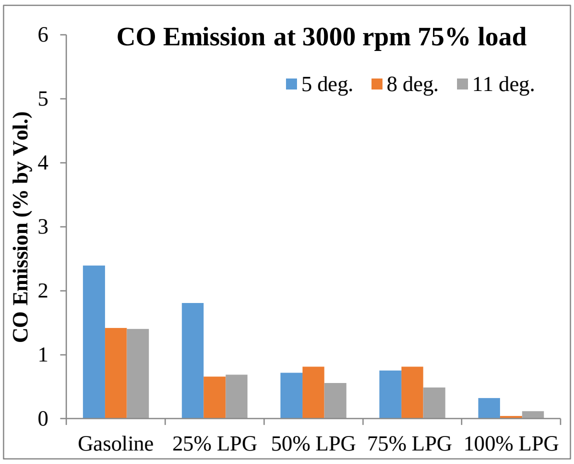


Fig 5.150 Effect of static ignition timing on CO at 4500 rpm 75% load condition.

Fig 5.151 Effect of static ignition timing on CO at 4000 rpm 75% load condition.



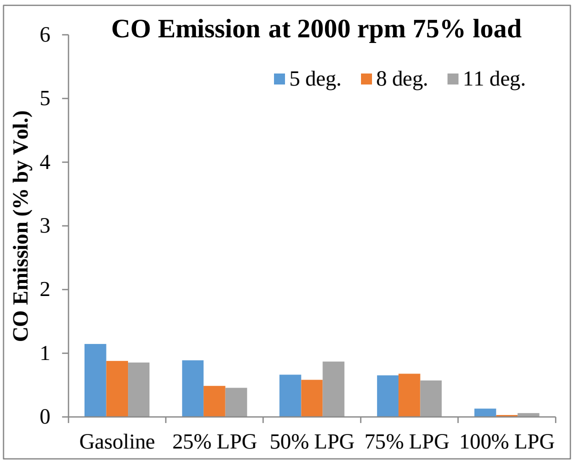


Fig 5.152 Effect of static ignition timing on CO at 3000 rpm 75% load condition.

Fig 5.153 Effect of static ignition timing on CO at 2000 rpm 75% load condition.

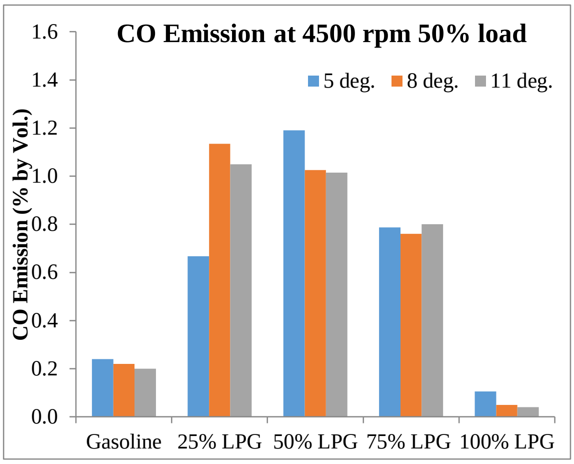


Fig 5.154 Effect of static ignition timing on CO at 4500 rpm 50% load condition.

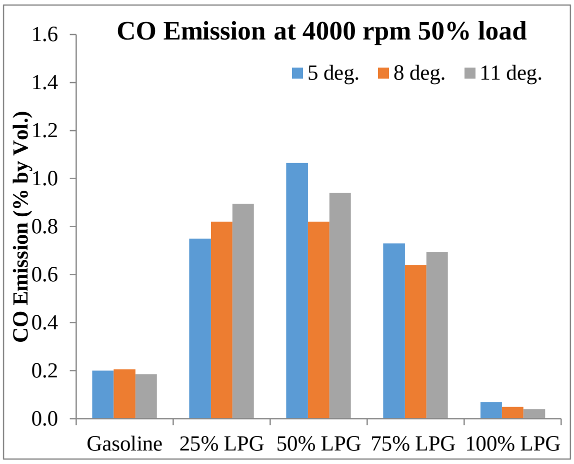


Fig 5.155 Effect of static ignition timing on CO at 4000 rpm 50% load condition.

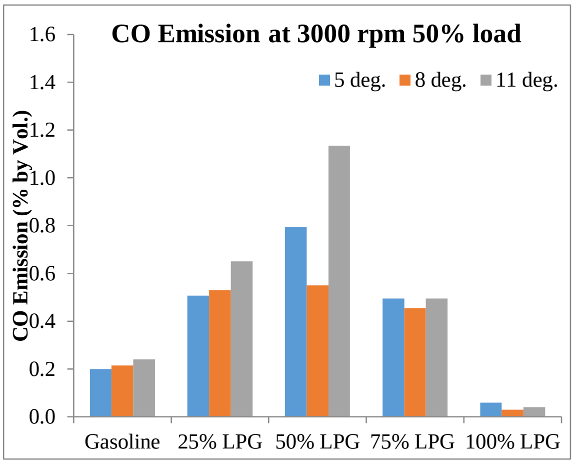


Fig 5.156 Effect of static ignition timing on CO at 3000 rpm 50% load condition.

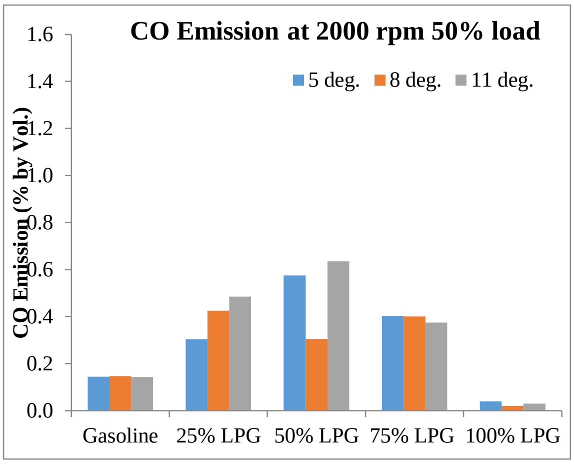


Fig 5.157 Effect of static ignition timing on CO at 2000 rpm 50% load condition.

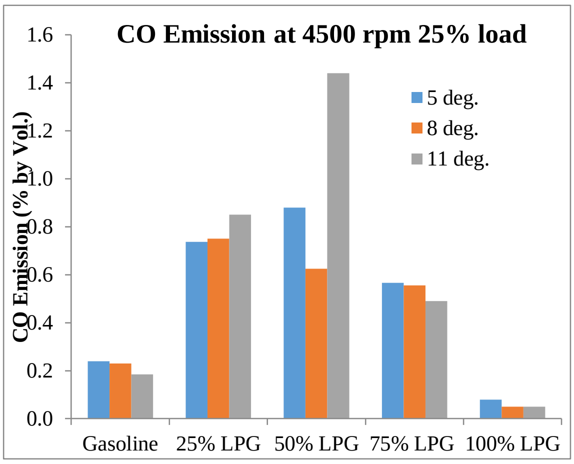


Fig 5.158 Effect of static ignition timing on CO at 4500 rpm 25% load condition.

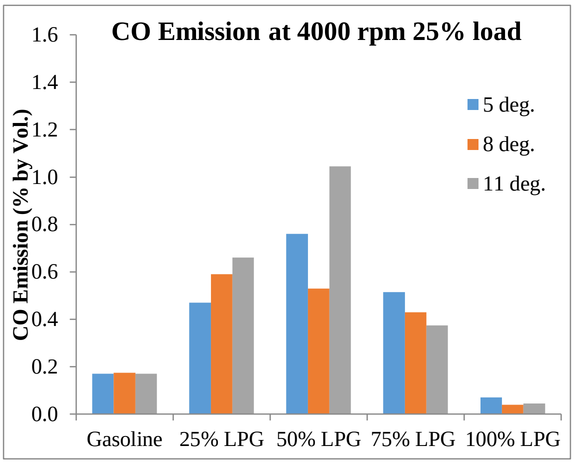


Fig 5.159 Effect of static ignition timing on CO at 4000 rpm 25% load condition.

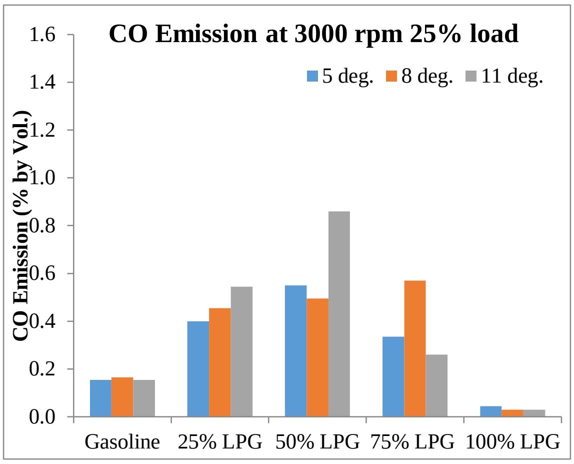


Fig 5.160 Effect of static ignition timing on CO at 3000 rpm 25% load condition.

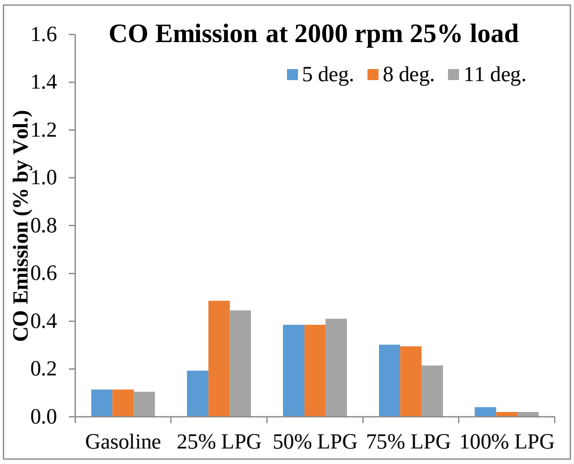
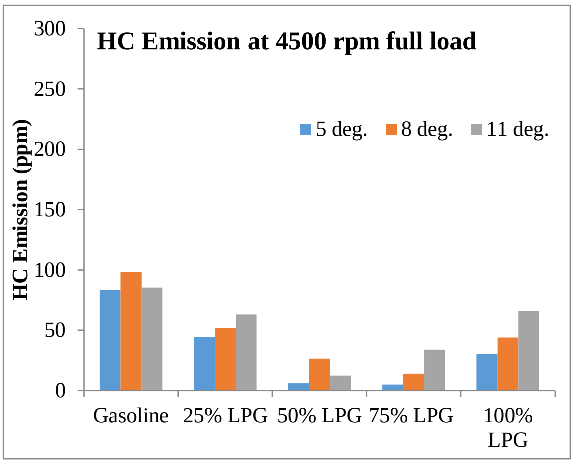


Fig 5.161 Effect of static ignition timing on CO at 2000 rpm 25% load condition.

5.2.2.2 Hydro Carbon (HC) emission

Figures 5.162 to 5.165 show the comparison of HC emissions on static ignition timing for different fuel blend and speed at full load condition. HC emissions are mainly influenced by the combustion quality and availability of oxygen. To achieve minimum HC emission best parameter is to decide is equivalence ratio. Whenever equivalence ratio is 0.9 then HC emission is going to be minimum. Figures 5.166 to 5.169, 5.170 to 5.173 and 5.174 to 5.177 show the effect of static ignition timing on HC emission for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively.

As static ignition timing advanced HC emission is increases for all loading conditions with gasoline and 100% LPG fuel. This is mainly because of two reasons. First, as static ignition timing is advanced in-cylinder pressure will rise which results in greater mass of hydrocarbons trapped in the crevice volume. Second, due to advance in the ignition timing exhaust gas temperature will reduce hence trapped hydrocarbon in the crevices volume will have less oxidation which results in higher HC emissions. In this way HC emission is increasing for advanced static ignition timing. Reduction in the HC emission at higher speed is due the fact that LPG combustion temperature is more than that of gasoline and also higher flame propagation speed, because of this enhanced combustion of fuel take place which will be resulted in substantial reduction in HC emission at all throttle open conditions (Heywood 1998).



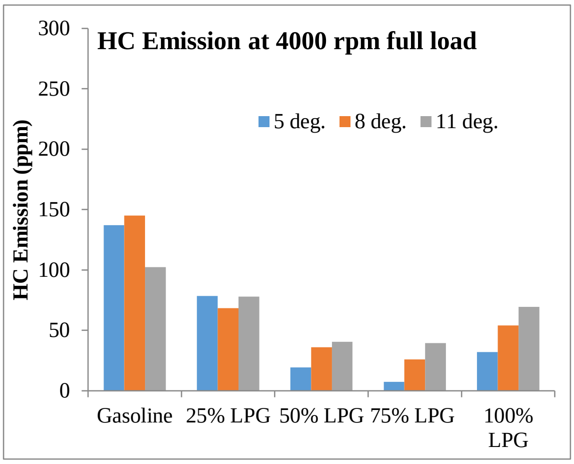
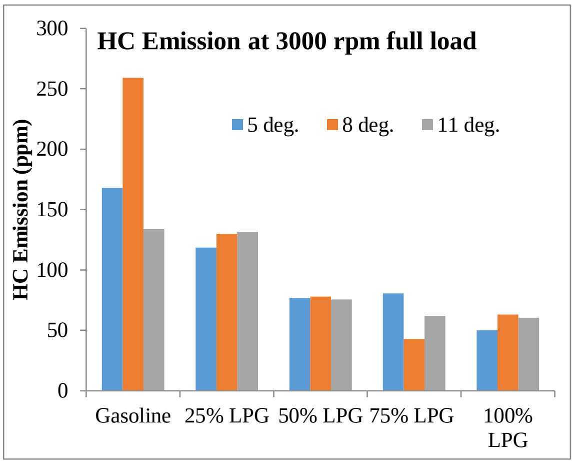


Fig 5.162 Effect of static ignition timing on HC at 4500 rpm full load condition.

Fig 5.163 Effect of static ignition timing on HC at 4000 rpm full load condition.



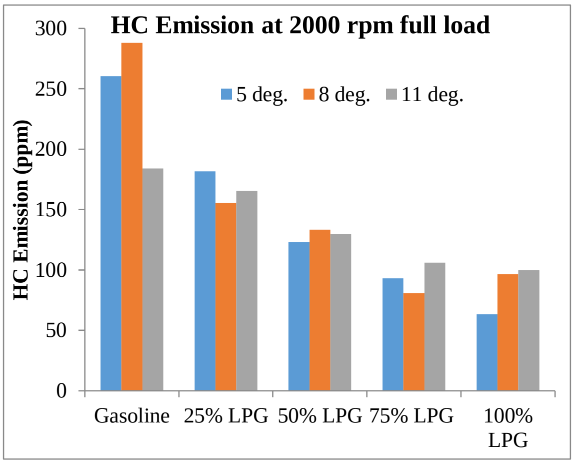
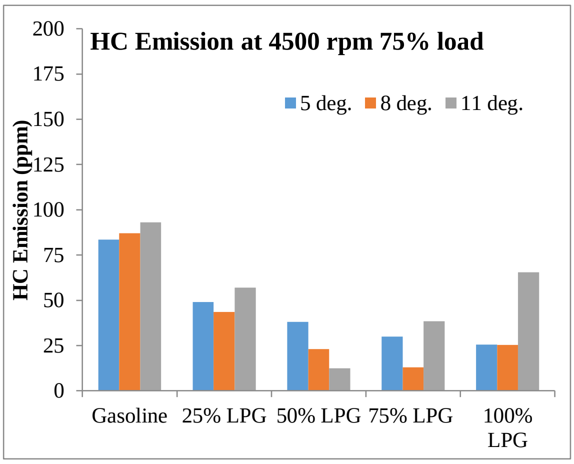


Fig 5.164 Effect of static ignition timing on HC at 3000 rpm full load condition.

Fig 5.165 Effect of static ignition timing on HC at 2000 rpm full load condition.



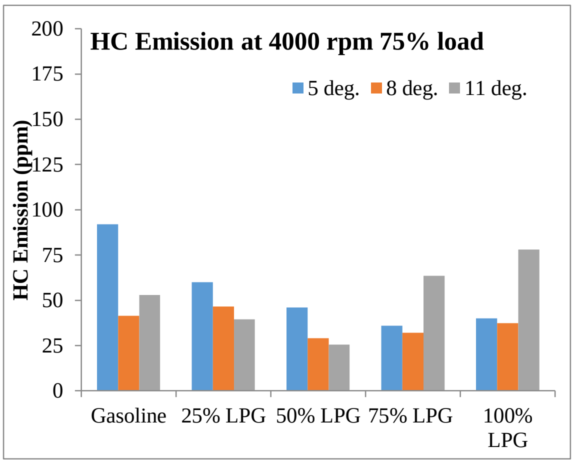


Fig 5.166 Effect of static ignition timing on HC at 4500 rpm 75% load condition.

Fig 5.167 Effect of static ignition timing on HC at 4000 rpm 75% load condition.

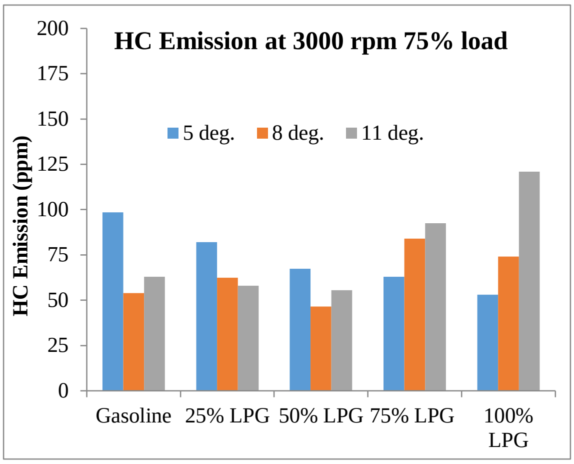


Fig 5.168 Effect of static ignition timing on HC at 3000 rpm 75% load condition.

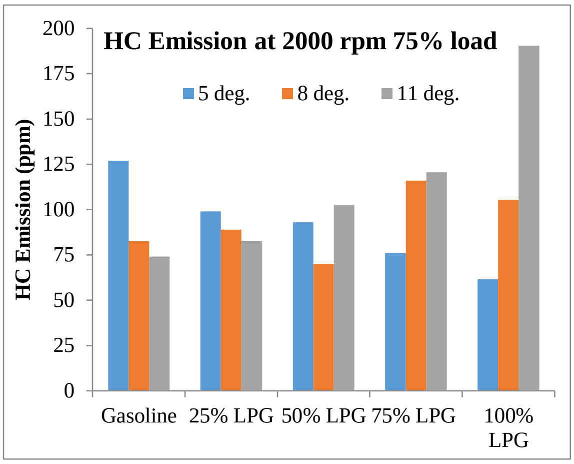


Fig 5.169 Effect of static ignition timing on HC at 2000 rpm 75% load condition.

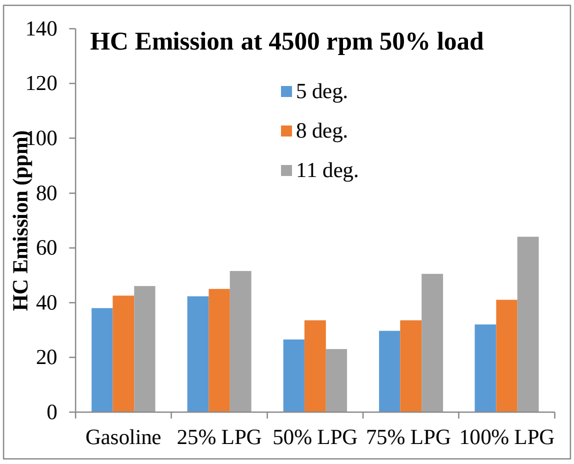


Fig 5.170 Effect of static ignition timing on HC at 4500 rpm 50% load condition.

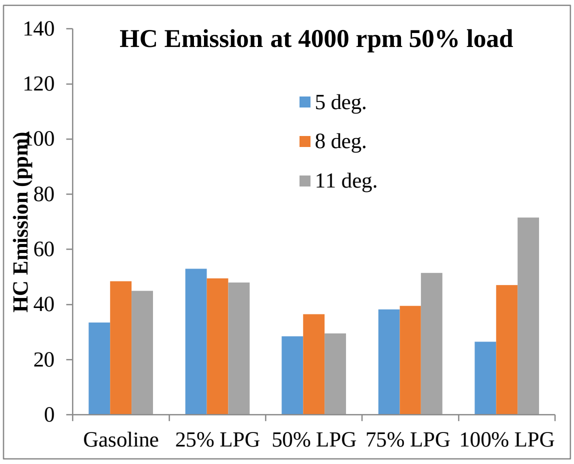


Fig 5.171 Effect of static ignition timing on HC at 4000 rpm 50% load condition.

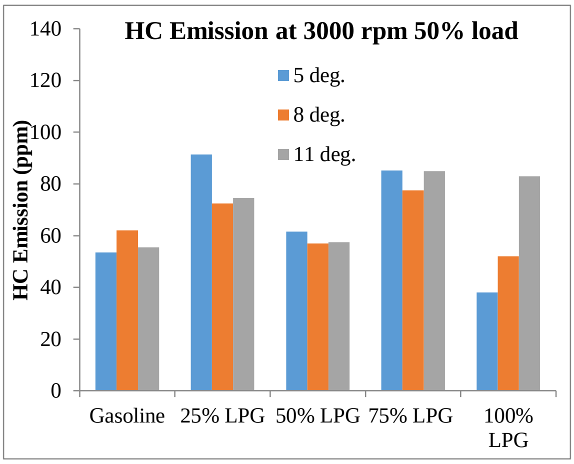


Fig 5.172 Effect of static ignition timing on HC at 3000 rpm 50% load condition.

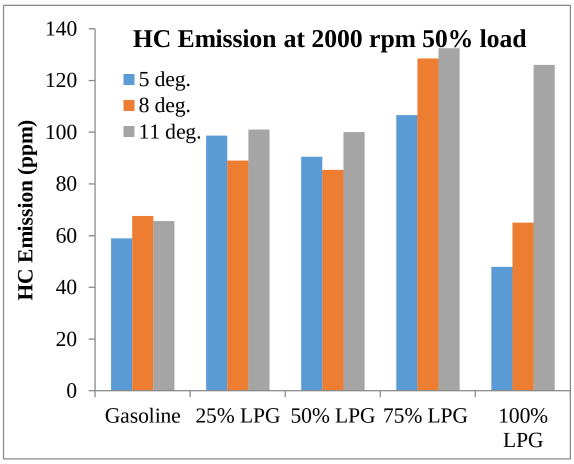
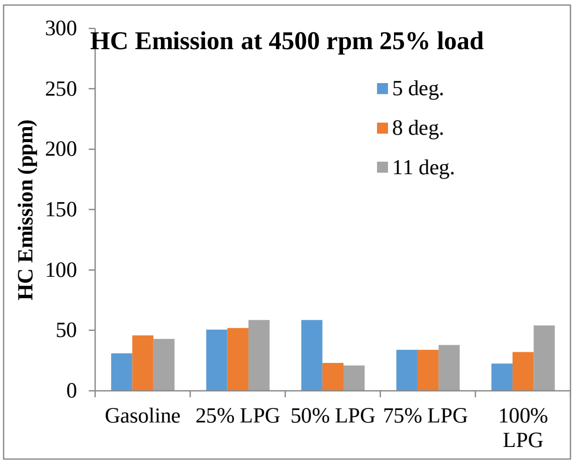


Fig 5.173 Effect of static ignition timing on HC at 2000 rpm 50% load condition.



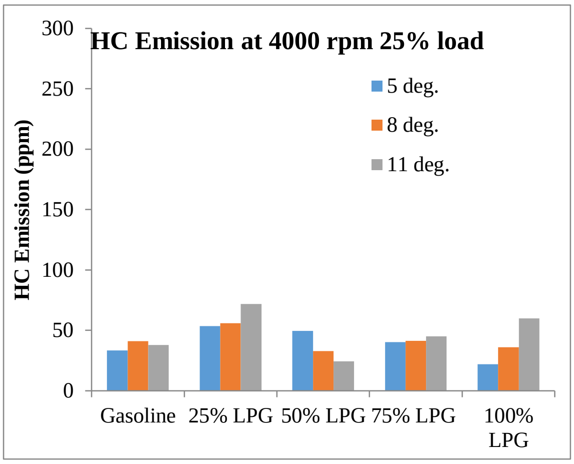
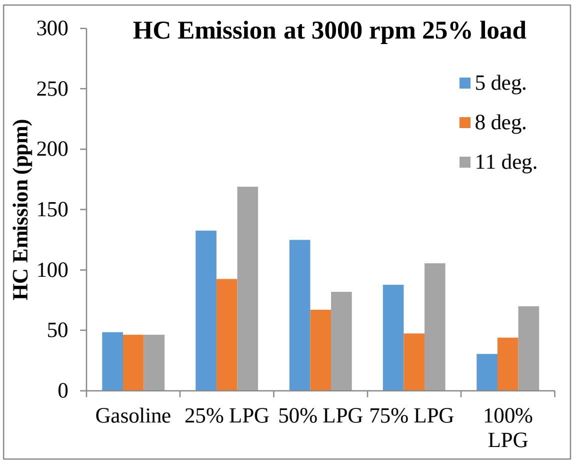


Fig 5.174 Effect of static ignition timing on HC at 4500 rpm 25% load condition.

Fig 5.175 Effect of static ignition timing on HC at 4000 rpm 25% load condition.



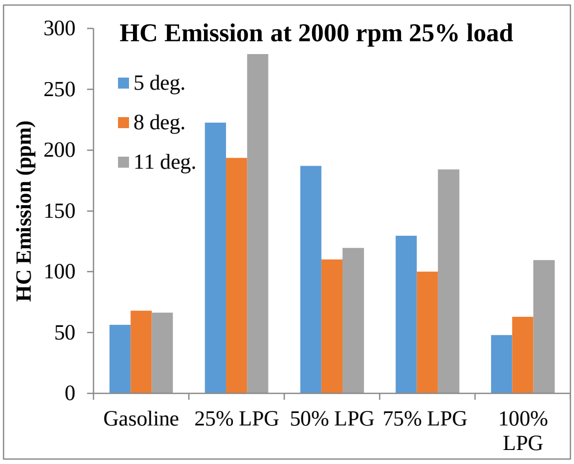


Fig 5.176 Effect of static ignition timing on HC at 3000 rpm 25% load condition.

Fig 5.177 Effect of static ignition timing on HC at 2000 rpm 25% load condition.

5.2.2.3 NO_X emission

Figures 5.178 to 5.181 show the comparison of NO_X with respect to static ignition timing for different engine speeds and fuel blend at full throttle position openings. NO_X concentration in the exhaust is mainly depending on the flame temperature and availability of oxygen. One of the most important engine variables that effect NO_X emission is equivalence ratio. The level of NO_X emission is exponentially dependent on in-cylinder temperature, as in-cylinder temperature increases the rate of NO_X formation. Figures 5.182 to 5.185, 5.186 to 5.189 and 5.190 to 5.193 show the effect of static ignition timing on NO_X emission for various engine speed and fuel blend at 75%, 50% and 25% of full load conditions respectively.

During full load condition as the static ignition timing is advanced NOx concentrations will also increases since equivalence ratio is near to stoichiometric region. NO_X emission is maximum for 100% LPG fuel at all speed and load conditions. As the timing is advanced the in-cylinder temperature rises and better combustion will take place in the cylinder hence NO_X concentration will increases. In part load condition also similar trends are obtained in the reduced concentration. Also as the percentage of LPG increases NO_X concentration will increases exponentially for all load and speed conditions. At full load, 4500 rpm increase in NO_X concentration for 100% LPG are 71% and 107% at 8 and 11 deg. bTDC respectively when compared to 5 deg. bTDC.

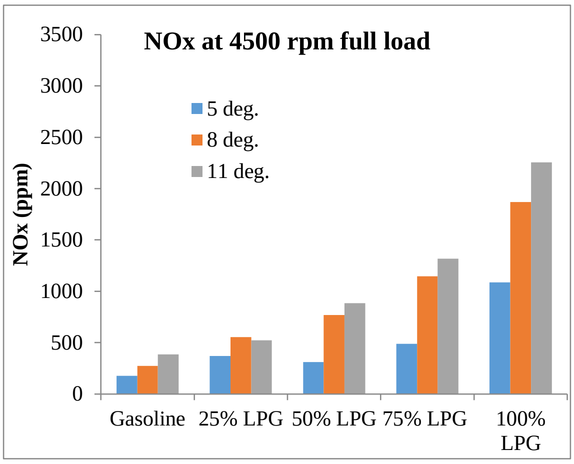


Fig 5.178 Effect of static ignition timing on NO_X at 4500 rpm full load condition.

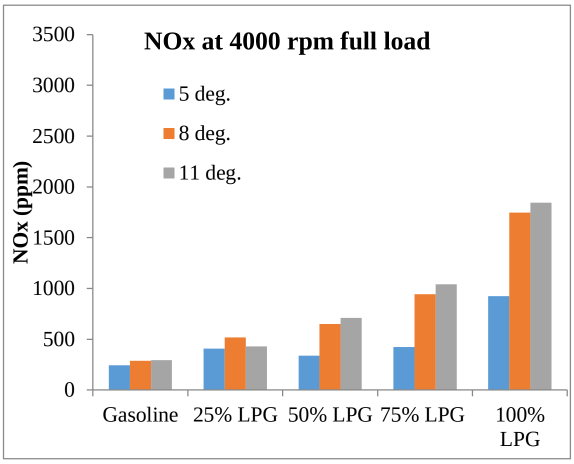


Fig 5.179 Effect of static ignition timing on NO_X at 4000 rpm full load condition.

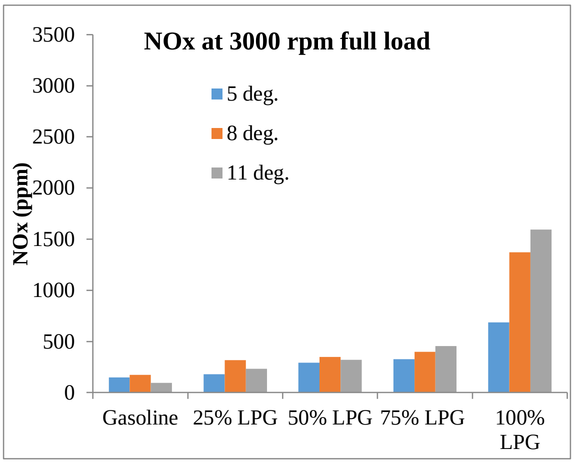


Fig 5.180 Effect of static ignition timing on NO_X at 3000 rpm full load condition.

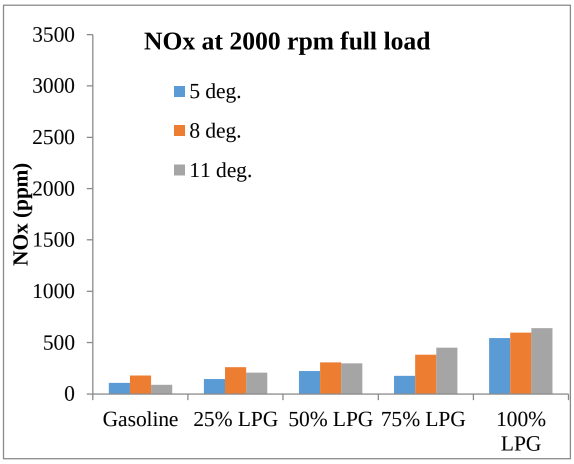


Fig 5.181 Effect of static ignition timing on NO_X at 2000 rpm full load condition.

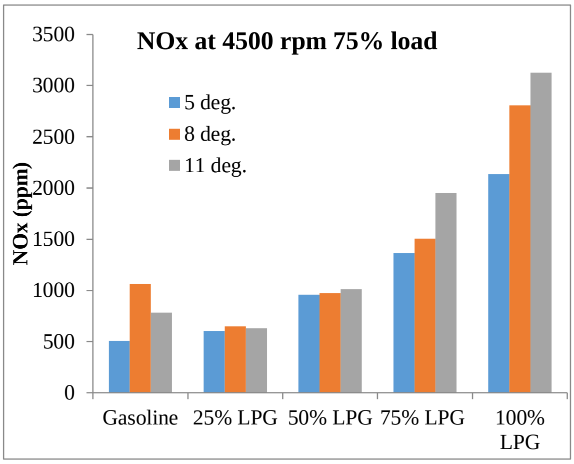


Fig 5.182 Effect of static ignition timing on NOx at 4500 rpm 75% load condition.

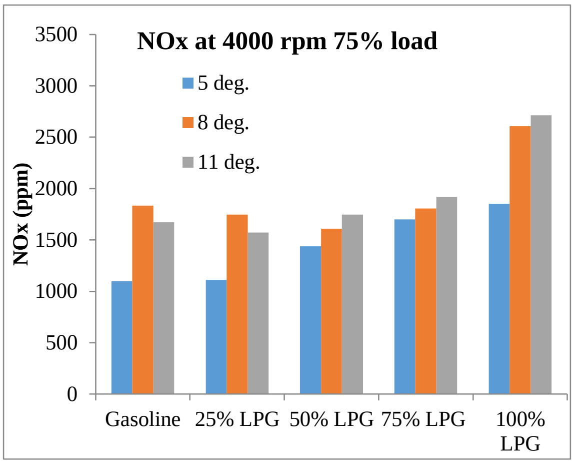


Fig 5.183 Effect of static ignition timing on NOx at 4000 rpm 75%load condition.

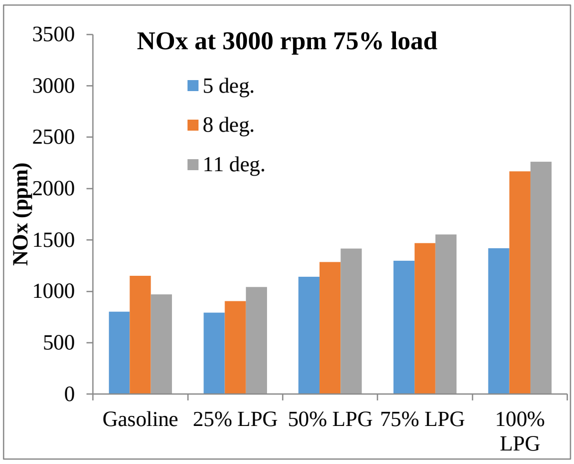


Fig 5.184 Effect of static ignition timing on NOx at 3000 rpm 75%load condition.

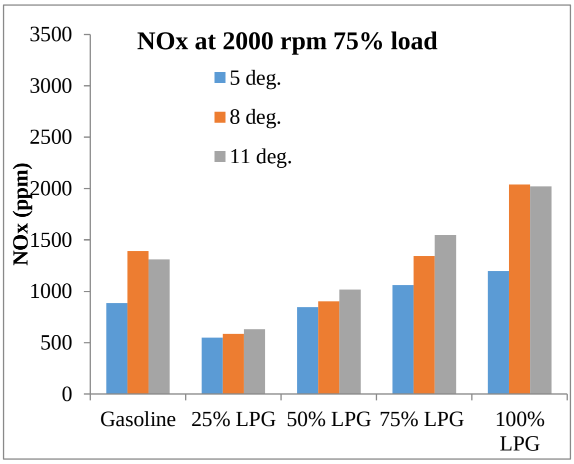


Fig 5.185 Effect of static ignition timing on NOx at 2000 rpm 75%load condition.

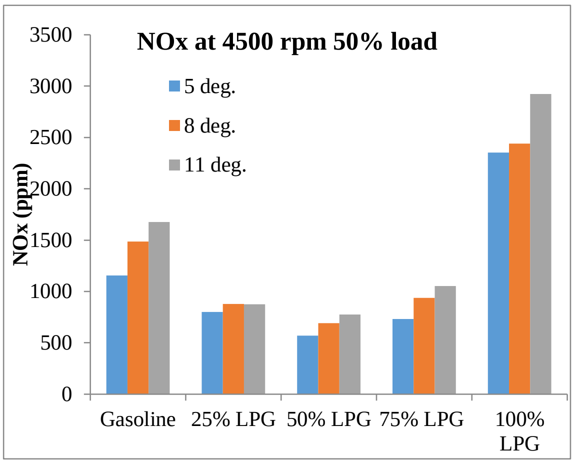


Fig 5.186 Effect of static ignition timing on NOx at 4500 rpm 50% load condition.

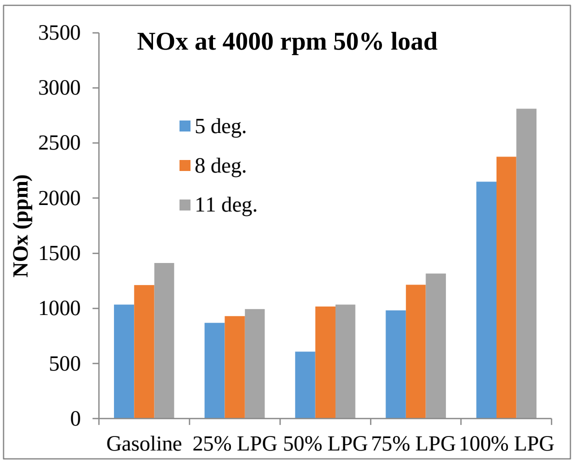


Fig 5.187 Effect of static ignition timing on NOx at 4000 rpm 50% load condition.

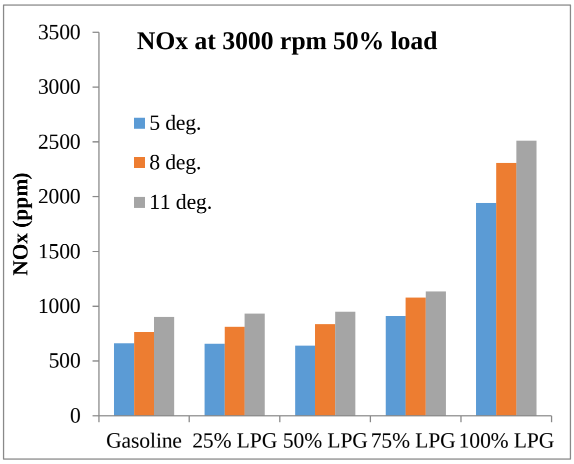


Fig 5.188 Effect of static ignition timing on NOx at 3000 rpm 50% load condition.

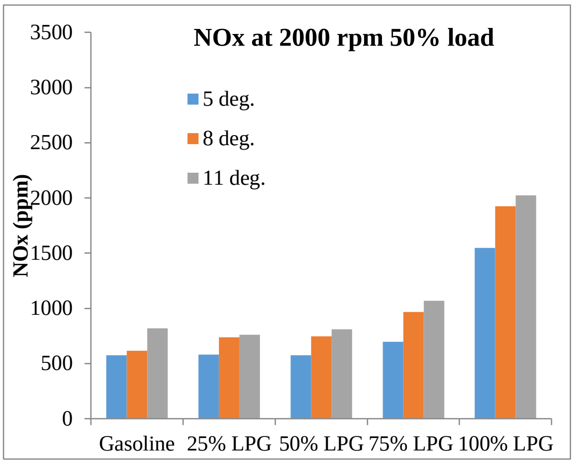


Fig 5.189 Effect of static ignition timing on NOx at 2000 rpm 50% load condition.

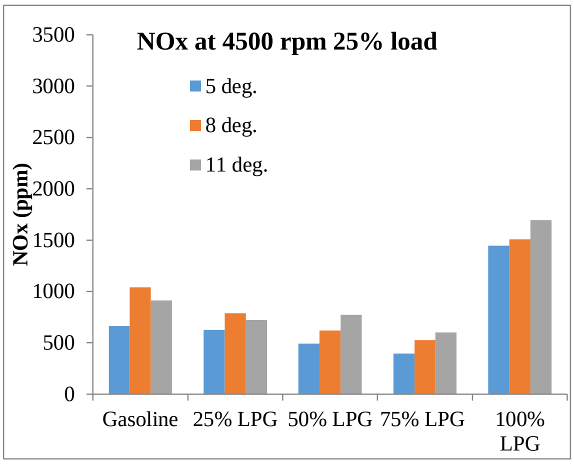


Fig 5.190 Effect of static ignition timing on NOx at 4500 rpm 25% load condition.

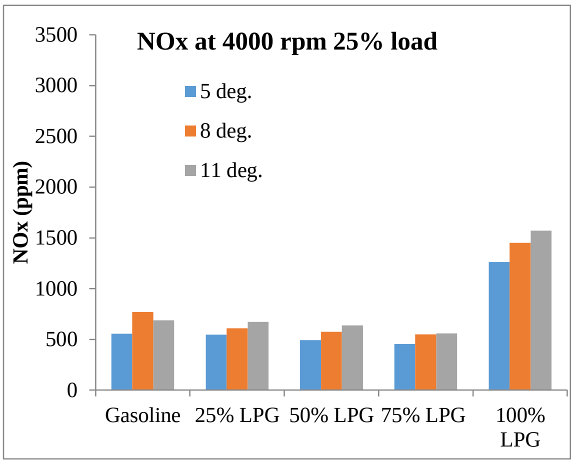


Fig 5.191 Effect of static ignition timing on NOx at 4000 rpm 25% load condition.

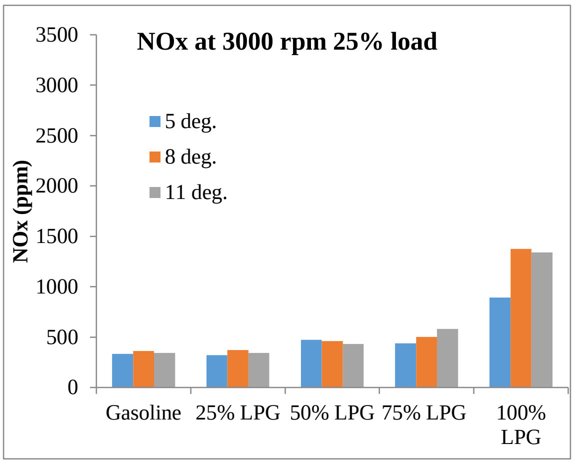


Fig 5.192 Effect of static ignition timing on NOx at 3000 rpm 25% load condition.

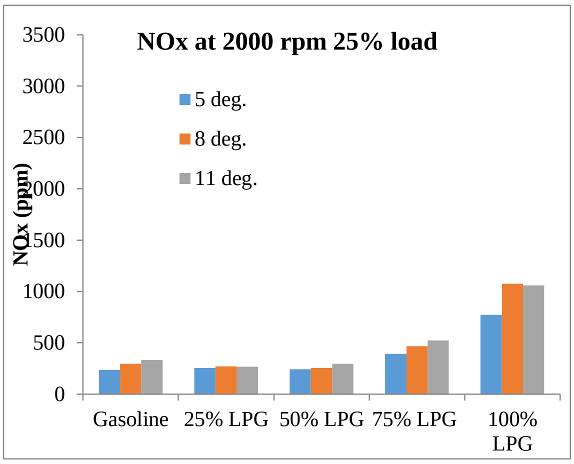


Fig 5.193 Effect of static ignition timing on NOx at 2000 rpm 25% load condition.

Summary: The results of the experiments with advancing static ignition timing on engine performance and emission characteristics can be summarized as follows:

- Results have shown that advancing the static ignition timing will increase the BP by 12 % at 11 deg. bTDC and 7% at 8 deg. bTDC for gasoline. Whereas for 100% LPG increased in BP is 5 % at 11 deg. bTDC and 2% at 8 deg. bTDC.
- BTE also increased for both gasoline and LPG when advancing static ignition timing because of reduction in the fuel consumption. Also advancing the ignition timing will engine will work leaner side hence reduction in the fuel consumption.
- As the ignition timing advanced volumetric efficiency will increases for all fuel conditions because of decrease in equivalence ratio.
- CO emission will drastically reduce when static ignition timing advanced to 8 deg. bTDC after that not significant reduction in CO emission. This is because since equivalence ratio is reduced availability of oxygen will increase and CO emission will decrease.
- 100% LPG shown major reduction in CO emission is obtained while advancing the static ignition timing.
- But advancing the Static ignition timing resulted in increased HC emission for all fuel blends. This is because greater mass of KC trapped in crevice volume.
- NO_X emission also increases with advancing the static ignition timing for all fuel blends because of increase in the in-cylinder temperature.

Finally after varying the static ignition timing it is found that 8 deg. bTDC with 100% LPG will resulted in better performance and emission characteristics hence these conditions are optimized.

5.3 EFFECT OF TURBOCHARGING

In the previous study static ignition timing has been varied and experimental results for various operating conditions have been investigated. It is analyzed and optimized with respect to performance and emission characteristics at 8 deg. bTDC and 100% LPG conditions. In this section to enhance the power, performance and combustion characteristics at optimized operating condition a turbocharger is incorporated and tested. A four cylinder, multipoint fuel injection engine has been modified to work on 100% LPG injection system with a turbocharger. A set of experiments has been carried out for 100% LPG and gasoline fuel to analyze performance, combustion and emission characteristics of an engine with turbocharger. Engine speed is varied from 2000 to 4500 rpm. The results are analyzed and compared with baseline fuel gasoline and factory set static ignition timing of 5 deg. bTDC.

5.3.1 Performance Characteristics

5.3.1.1 Brake Power

The figures 5.194 to 5.197 show the variation of brake power of gasoline, LPG with and without turbocharger at various operating speed and full load conditions and it is compared with baseline condition. Gasoline with turbocharger will produce higher BP at all speed and load condition compared to LPG. This is because gasoline will work in a richer region compared to LPG and produces comparatively higher BP. The effect of turbo charging on SI engine increase the power output for both fuels. This is because charge density as well as volumetric efficiency will increase which leads to the better combustion. Hence power output of the engine as well as load carrying capacity will increase. For turbocharged LPG also power output is increased when compared to without turbocharger and baseline fuel. But compared to turbocharger gasoline power out is slightly less because of it is working in slight leaner region for all load and speed conditions. The influx of fuel is more in wide open throttle compared to part throttles. This results in the higher power generation in the larger load condition for all the fuels. As the load increases the power also increases for both LPG and gasoline using turbocharger.

The average increase in the power output for turbocharged gasoline at 4500 rpm and full load is 2 kW compared to without turbocharger at 8 deg. bTDC and 4.2 kW when compared to gasoline at 5 deg. bTDC (baseline) condition. For LPG with turbocharger at same condition is 3.3 kW compared to without turbocharger at 8 deg. bTDC and 3.8 kW when compared to LPG at 5 deg. bTDC (baseline) condition.

When compared to base fuel gasoline at 5 deg. bTDC average increase in power output for LPG with turbocharger is 2.42 kW at same condition.

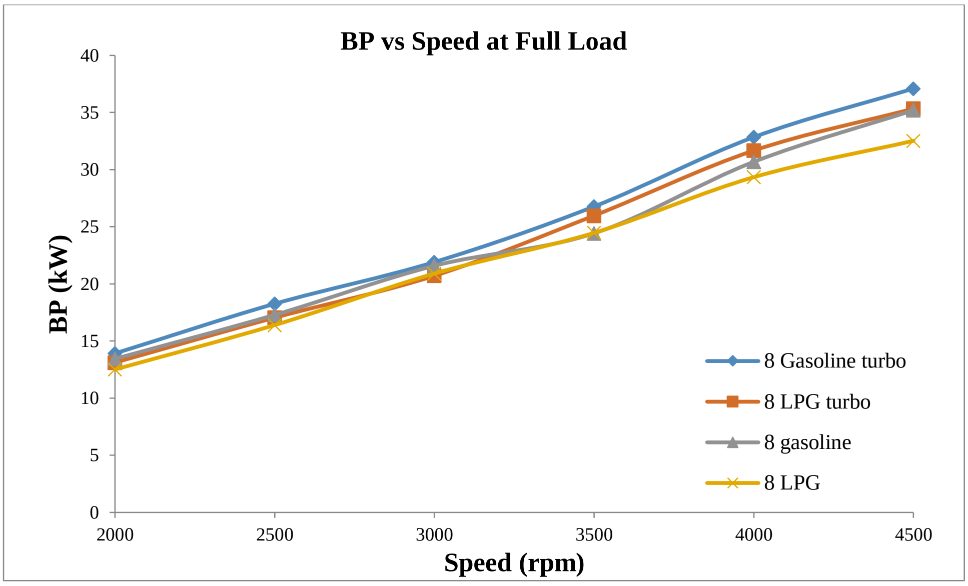


Fig. 5.194 Effect of BP vs Speed at full load condition.

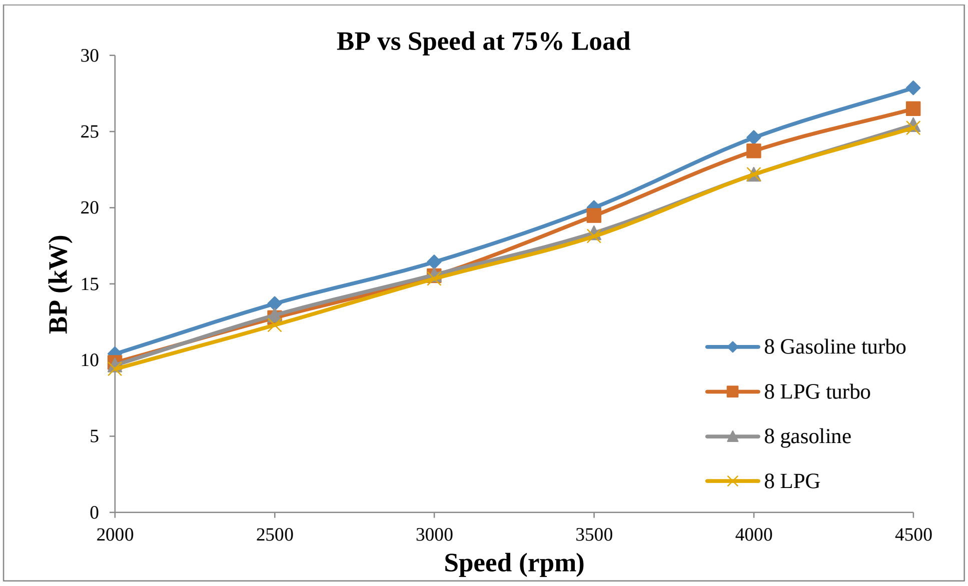


Fig. 5.195 Effect of BP vs Speed at 75% of full load condition.

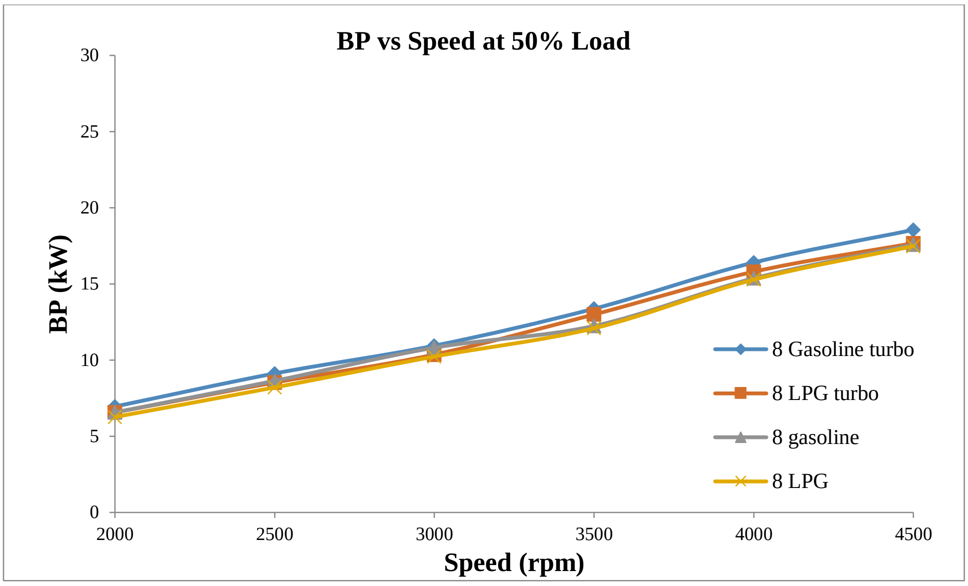


Fig. 5.196 Effect of BP vs Speed at 50% of full load condition.

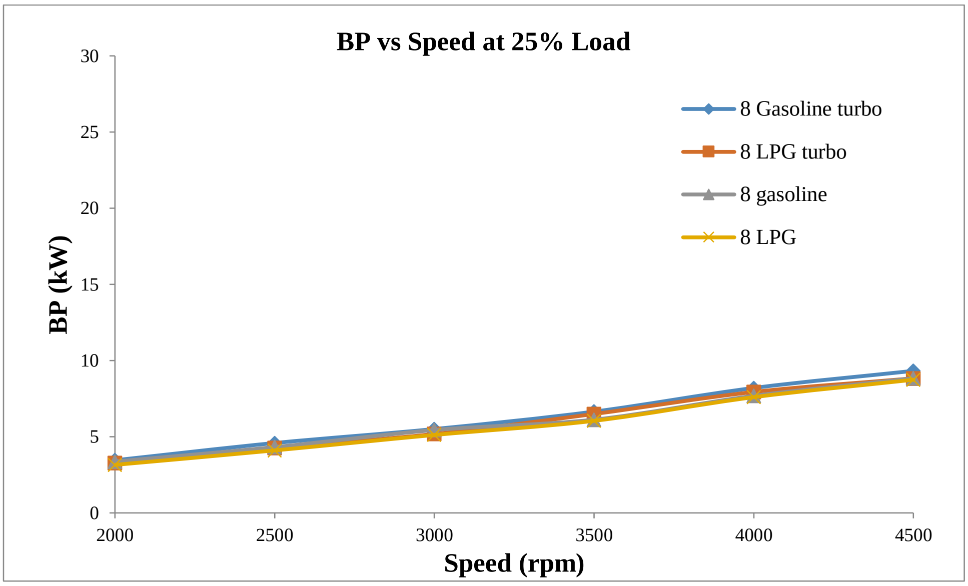


Fig. 5.197 Effect of BP vs Speed at 25% of full load condition.

5.3.1.2 Brake Thermal Efficiency:

The following figures 5.198 to 5.201 show the variations of brake thermal efficiency of gasoline, LPG with and without turbocharger at various operating speed and full load conditions and it is compared with baseline condition. The characteristics of these curves show that the brake thermal efficiency increases with increase in speed but after 4000 rpm it starts decreases. Brake thermal efficiency found to be maximum at 75 % load condition for all fuel. This is because engine will work in the economy zone where fuel consumption is taken utmost care. At full load condition power output is more important than fuel economy. At higher engine conditions, due to the higher flame propagation speed of LPG and higher charge density negates the effect of ignition temperature. Here the time duration for each cycle is increased which demands less fuel consumption for the combustion fuel. Hence LPG with turbocharger will have higher BTE when compared to gasoline. The lower propagation speeds of gasoline flames cannot afford the requisite combustion rate; instead the engine takes more fuel to generate the required torque. The collective outcome of these factors lowers BTE of the engine for gasoline at higher engine load conditions. Turbocharged LPG will have higher brake thermal efficiency than gasoline irrespective of power increment in gasoline when compared to LPG.

The lean operation decreases the flame speed and the burning rate, and the reduction in burning rate results in an increase in the overall combustion duration. Since the ignition temperature of LPG is higher than the gasoline, ignition delay and thus combustion duration is more for LPG (Ceviz et al. 2005). Since there is a decreases in the average burning rate, to accommodate this effect engine consumes more fuel which in turn decreases its efficiency. Hence LPG with turbocharger has lower efficiency than gasoline at lower load conditions. Since using turbocharger volumetric efficiency of LPG is more than without turbocharger condition which lead to higher brake power hence higher BTE is obtained when compared to the without turbocharger and baseline condition. And also turbocharger will increases the intake air pressure, which leads more oxygen available for combustion and better combustion is achieved.

The average value of BTE for turbocharged gasoline at 4000 rpm and full load is 31.74%, whereas for gasoline without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 28.75% and 29.86% respectively. For LPG with turbocharger at same condition is 34.15%, whereas for LPG without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 32.27% and 28.54% respectively. When compared to base fuel gasoline at 5 deg. bTDC average increase in BTE for LPG with turbocharger is 4.3% at same condition.

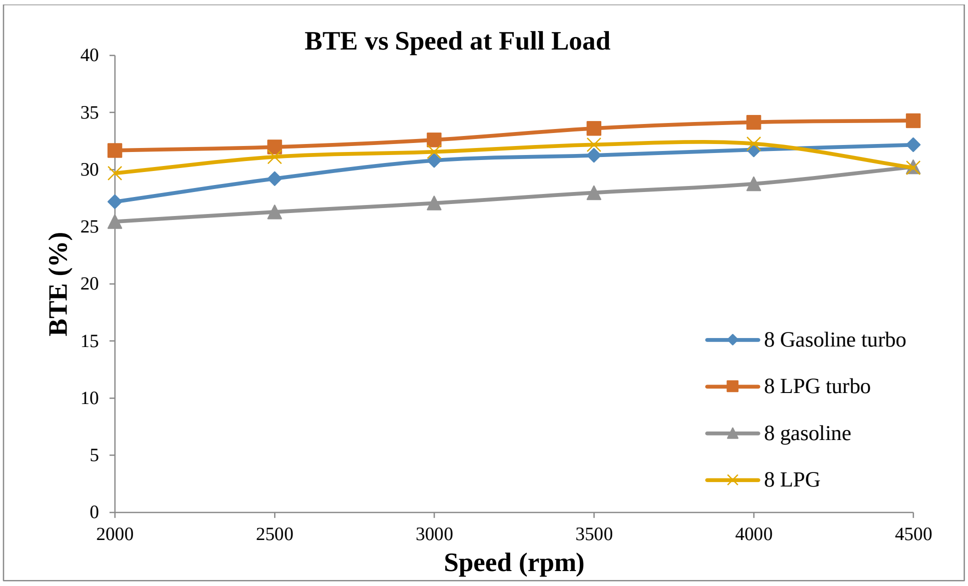


Fig. 5.198 Effect of BTE vs Speed at full load condition.

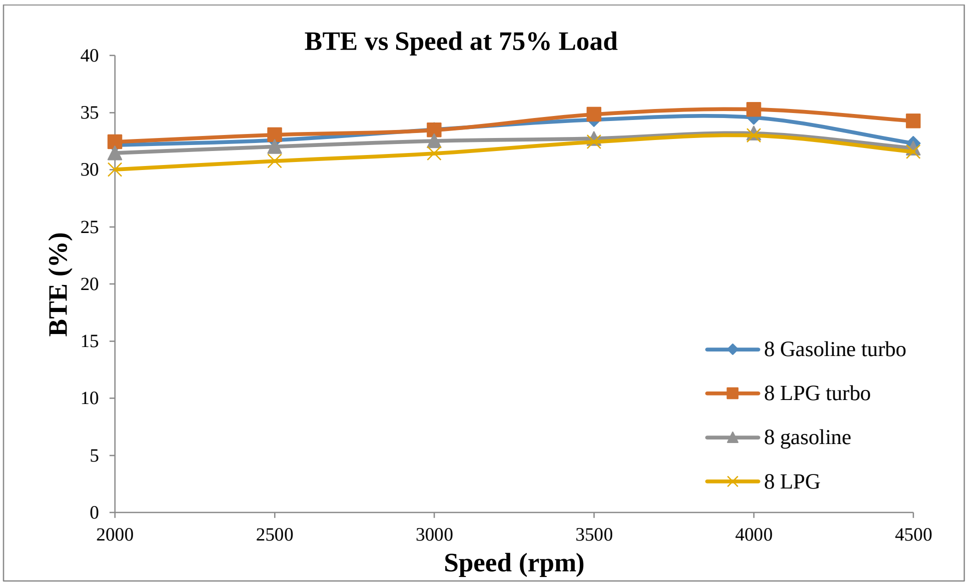


Fig. 5.199 Effect of BTE vs Speed at 75% of full load condition.

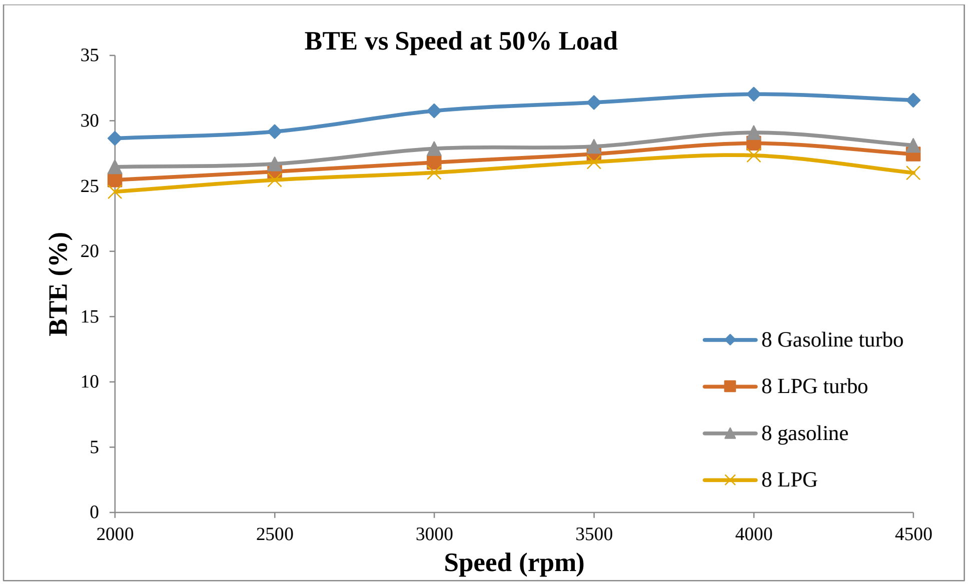


Fig. 5.200 Effect of BTE vs Speed at 50% of full load condition.

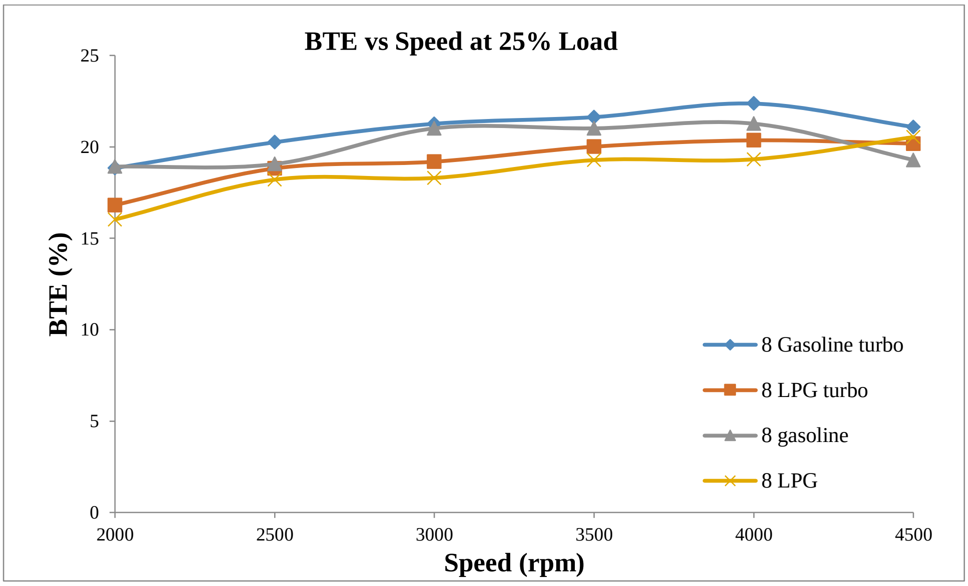


Fig. 5.201 Effect of BTE vs Speed at 25% of full load condition.

5.3.1.3 Equivalence ratio:

The figure 5.202 to 5.205 indicates the variations of equivalence ratio of gasoline, LPG with and without turbocharger at various operating speed and full load conditions and it is compared with baseline condition. This equivalence ratio will determine whether engine is working in leaner, stoichiometric or richer mixture. As the engine speed increases the equivalence ratio will try to remains in same value.

As seen from the graph, turbocharged engine fuelled with gasoline has lower equivalence ratio as compared to engine without turbocharger. But in case of higher load condition turbocharged gasoline engine will give higher equivalence ratio as compared to LPG fuelled engine. In case of turbocharged engine fuelled with LPG has lower equivalence ratio as compared to engine without turbocharger for all speed and load conditions. With the introduction of turbocharger, equivalence ratio has been reduced for both LPG and gasoline, and working near to stoichiometric region.

In LPG fuelled engine combustion will take place in leaner region so as the speed increases quantity of fuel will increase so it will reach near stoichiometric region or slightly richer region. But at part load and higher speed condition LPG with turbocharger will work in leaner region whereas LPG at 5 deg. bTDC is in richer region. This is because at part load condition there is restriction of air which leads to lesser air-fuel ratio hence equivalence ratio is high.

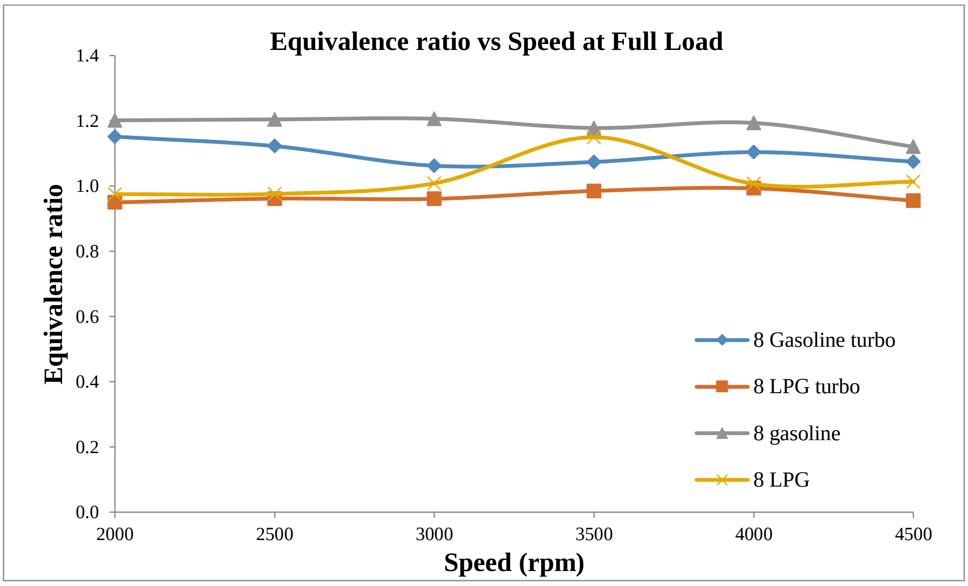


Fig. 5.202 Effect of Equivalence ratio vs Speed at full load condition.

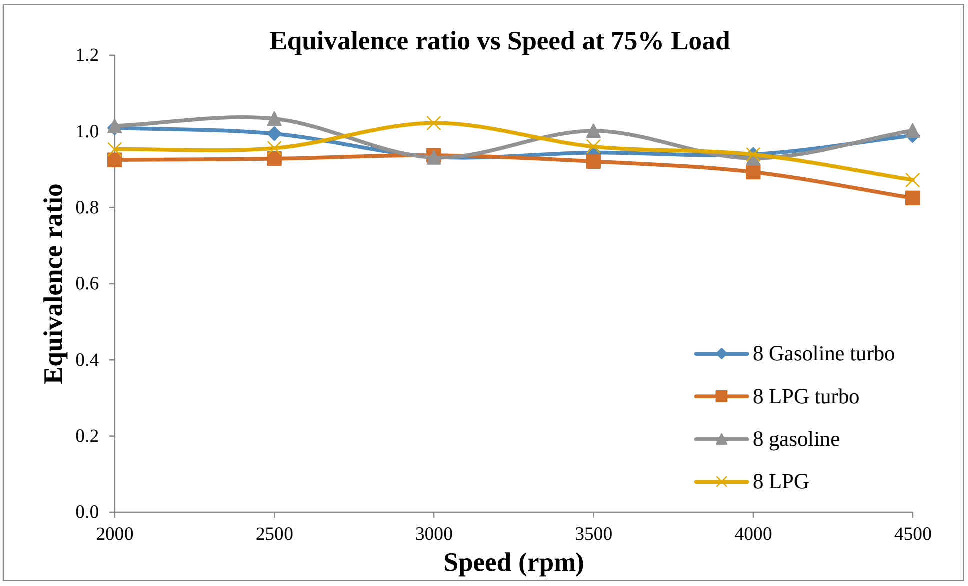


Fig. 5.203 Effect of Equivalence ratio vs Speed at 75% of full load condition.

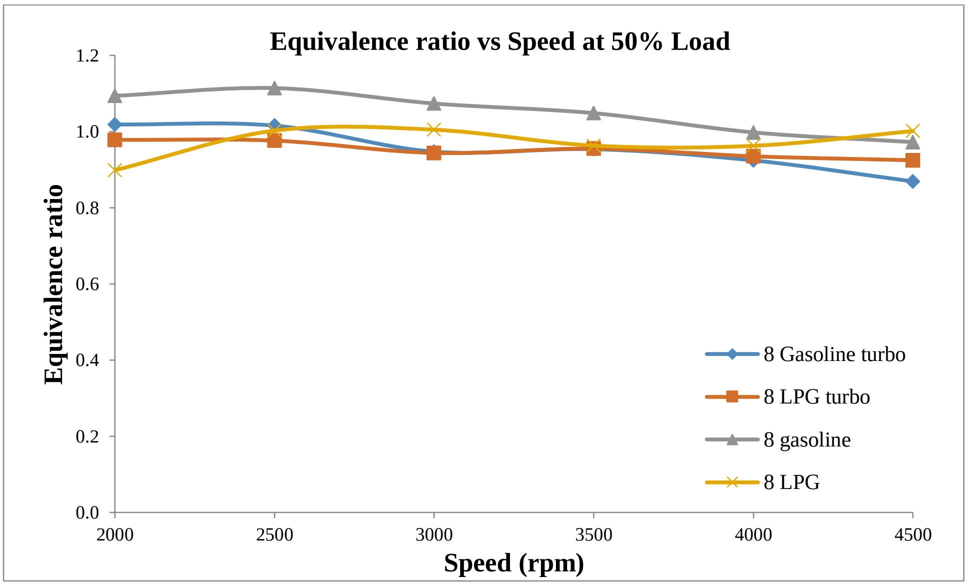


Fig. 5.204 Effect of Equivalence ratio vs Speed at 50% of full load condition.

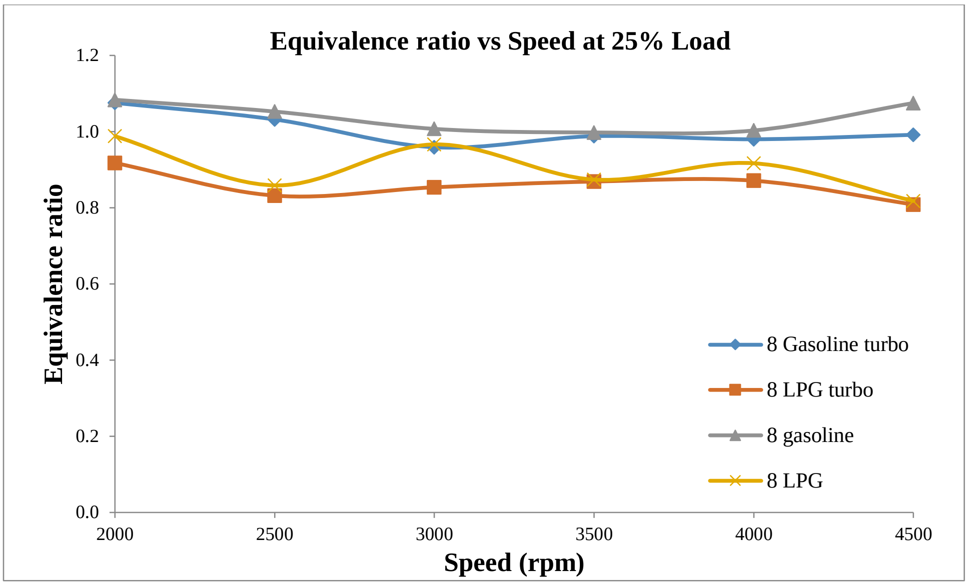


Fig. 5.205 Effect of Equivalence ratio vs Speed at 25% of full load condition.

5.3.1.4 Volumetric Efficiency:

Volumetric efficiency is a measure of effectiveness of an engine's induction process. The induction system includes the air-filter, injector and throttle plate in spark ignition engine. They all restrict the amount of air which an engine of given displacement can induct. Volumetric efficiency of a normally aspirated engine is the ratio of the actual volume flow rate of air into the cylinder at atmospheric pressure and temperature conditions surrounding to the engine to the rate at which the volume displaced by the piston. The figures 5.206 to 5.209 indicate the variations of volumetric efficiency of gasoline, LPG with and without turbocharger at various operating speed and full load conditions and it is compared with baseline condition.

As the speed increases the volumetric efficiency also increases till the maximum power, after that speed it is decreases with increase in the speed for all fuel and load conditions. As the load increases the volumetric efficiency also keep on increases for all the fuel and speed conditions. At higher load volumetric efficiency of LPG is lesser compared to gasoline because gaseous fuel will displaces more amount of air compare to liquid fuel. But at part load condition the LPG is working in leaner region the quantity of fuel injected is less compared to gasoline therefore volumetric efficiency will be more for LPG.

As seen from the graph, in case of turbocharged engine fuelled with gasoline has higher volumetric efficiency as compared to LPG and without turbocharger for all speed and higher load conditions. This is because gaseous fuel will displaces more amount of air compare to liquid fuel hence decrease in volumetric efficiency. Also, turbocharged engine fuelled with LPG has higher volumetric efficiency as compared to engine without turbocharger for all speed and load conditions. Volumetric efficiency will increases for turbocharged engine because of higher intake air pressure will increase the density of air which leads to increase in the efficiency.

But experiments showed at part load condition turbocharged engine fuelled with LPG having more volumetric efficiency when compared to gasoline. This is because at part load condition LPG works in too leaner region whereas gasoline works in richer region. And also liquid density is more and amount of air entering is restricted by throttle hence reduction in the volumetric efficiency for gasoline when compared to LPG for turbocharged engine at part load condition. When compared to baseline condition for both fuel for a turbocharged engine volumetric efficiency is increased for all speed and load conditions.

The average value of volumetric efficiency for turbocharged gasoline at 4500 rpm and full load is 84.58%, whereas for gasoline without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 78% and 70.2% respectively. For LPG with turbocharger at same condition is 83.18%, whereas for LPG without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 76.79% and 68.3% respectively. When compared to base fuel gasoline at 5 deg. bTDC average increase in volumetric efficiency for LPG with turbocharger is 13% at same condition.

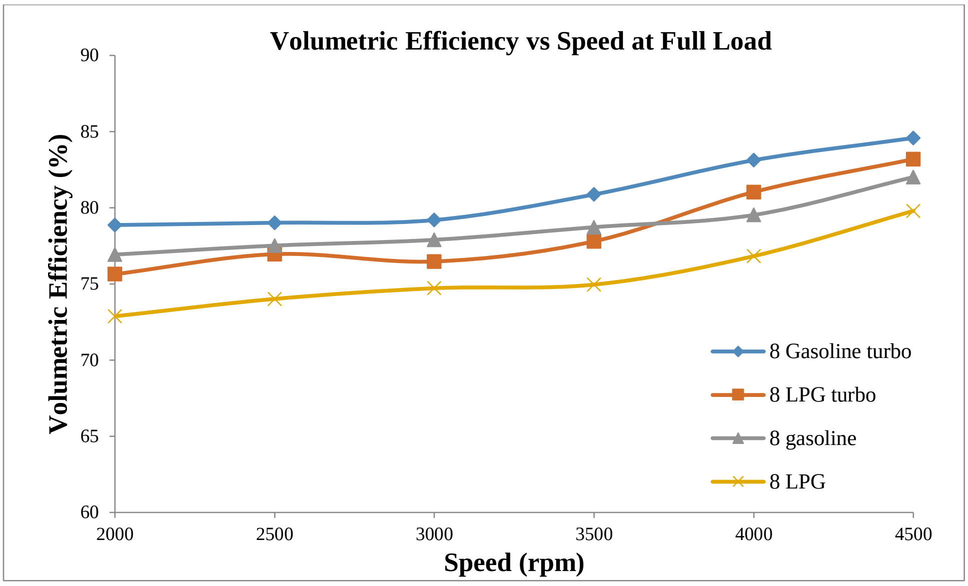


Fig. 5.206 Effect of Volumetric efficiency vs Speed at full load condition.

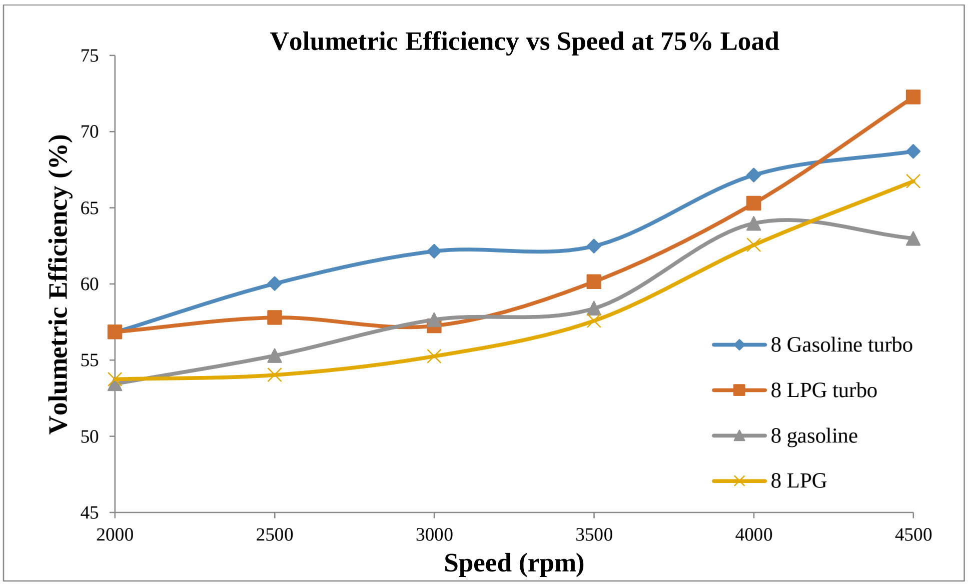


Fig. 5.207 Effect of Volumetric efficiency vs Speed at 75% of full load condition.

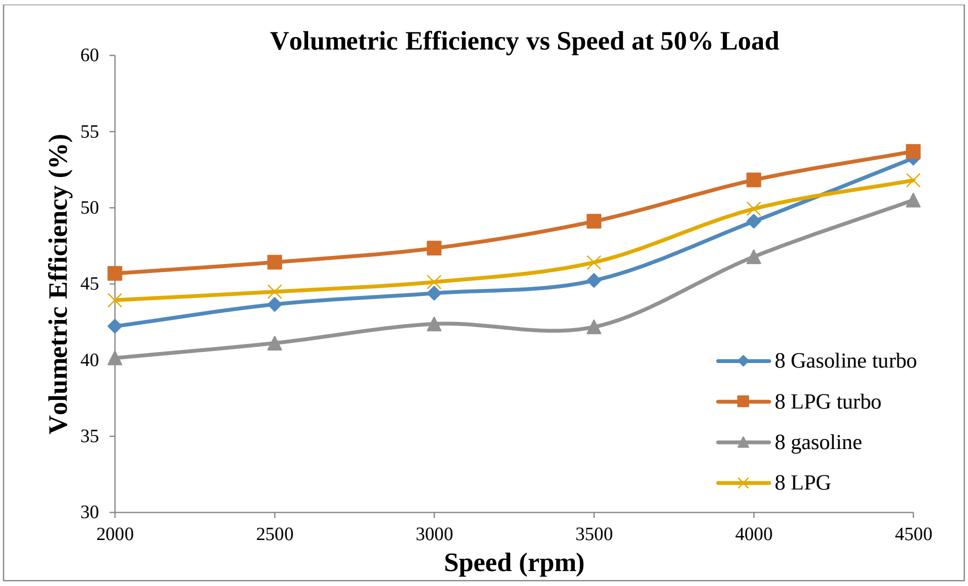


Fig. 5.208 Effect of Volumetric efficiency vs Speed at 50% of full load condition.

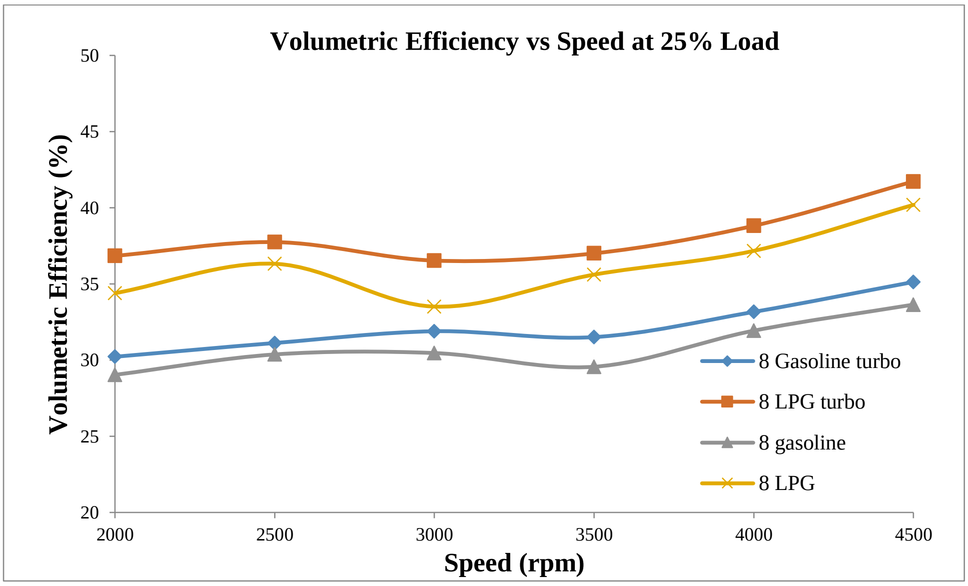


Fig. 5.209 Effect of Volumetric efficiency vs Speed at 25% of full load condition.

5.3.1.5 Brake Specific Energy Consumption:

Brake specific energy consumption will indicate to produce one kilowatt of power in one hour how much energy is consumed. This comparison will give the fuel economy for an engine where different calorific values of fuels are being used. The figures 5.210 to 5.213 indicate the variations of brake specific energy consumption of gasoline, LPG with and without turbocharger at various operating speed and load conditions and it is compared with baseline condition. As the speed increases BSEC will decreases till 4000rpm then starts increasing for all load and fuel condition. This is the point where fuel consumption going to minimum.

At full load and 75% conditions, the BSEC of LPG is lower than that of gasoline at all speeds for turbocharged engine. This can be a result of higher mass flow rate of air which work in a leaner region in turn results in lesser quantity of fuel. And also due to the higher flame propagation speed and better miscibility of gaseous LPG with the air will reduced fuel consumption. At this condition LPG with turbocharger will have better fuel economy when compared to all other cases.

At part load condition for turbocharged engine with LPG fuel BSEC increases when compared to gasoline. This is because to accommodate lower burning rate of LPG in the lean condition more fuel is consumed. Hence LPG at lower load condition will have higher BSEC compared to gasoline for turbocharged engine. But when compare to baseline condition BSEC of turbocharged engine fuelled with LPG having lesser. Hence LPG with turbocharger has lower efficiency than gasoline at lower load conditions. This is because of turbocharging will increases air density hence oxygen content in the intake manifold.

The average decrease in the BSEC for turbocharged gasoline at 4000 rpm and full load is 9.4%, when compared to without turbocharger at 8 deg. bTDC and 6% when compared to 5 deg. bTDC (baseline) conditions. For LPG with turbocharger at same condition is 5.5%, when compared to without turbocharger at 8 deg. bTDC and 16% when compared to 5 deg. bTDC (baseline) conditions. When compared to base fuel gasoline at 5 deg. bTDC average decrease in BSEC for LPG with turbocharger is 16% at same condition.

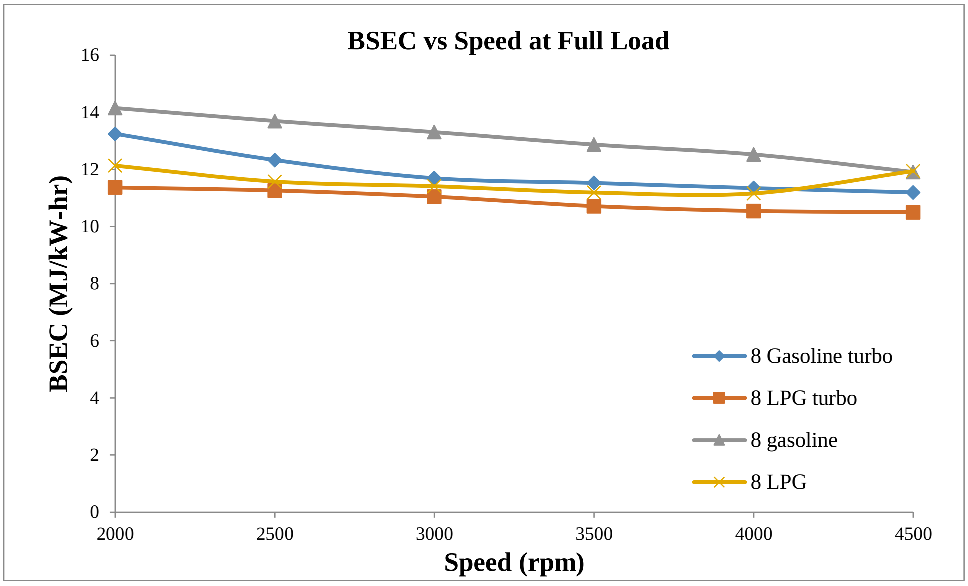


Fig. 5.210 Effect of BSEC vs Speed at full load condition.

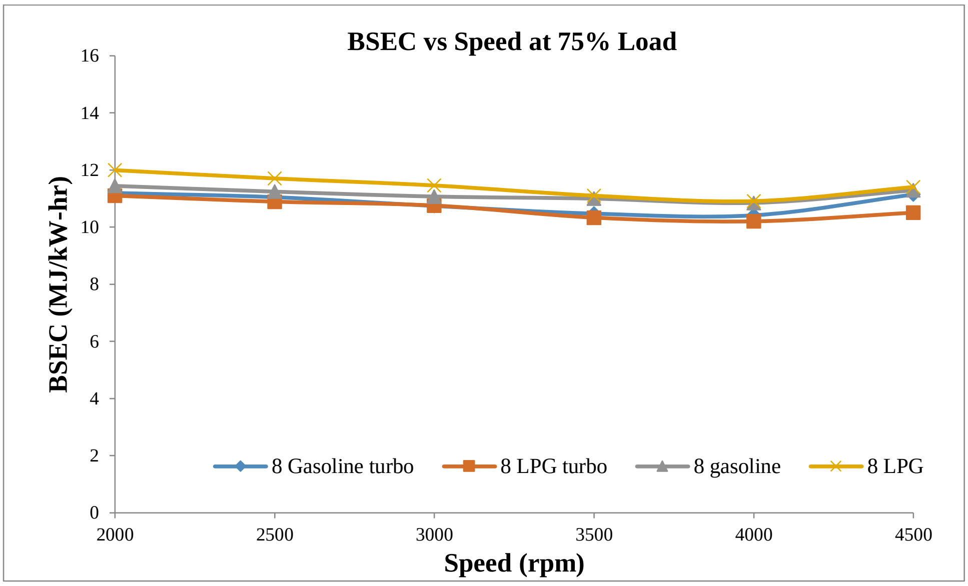


Fig. 5.211 Effect of BSEC vs Speed at 75% of full load condition.

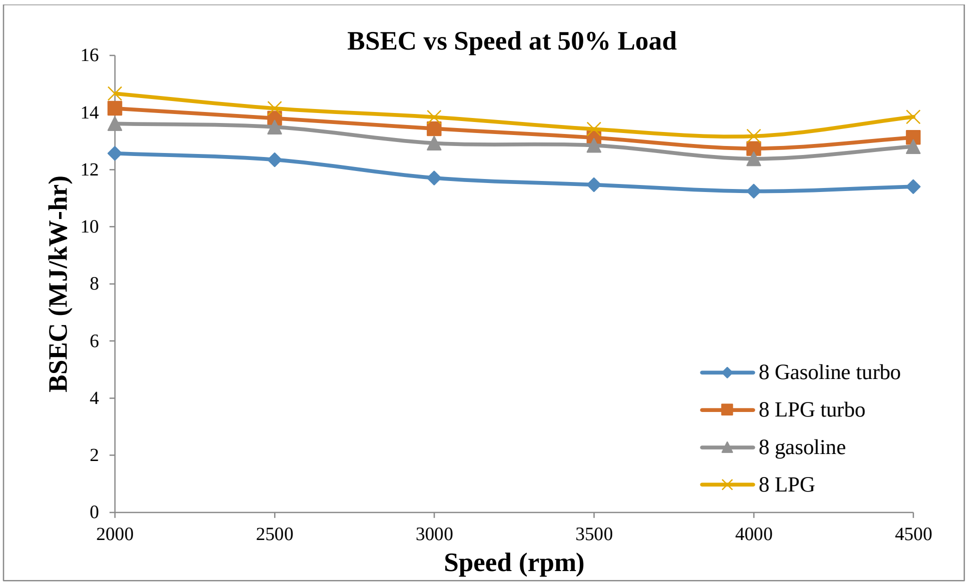


Fig. 5.212 Effect of BSEC vs Speed at 50% of full load condition.

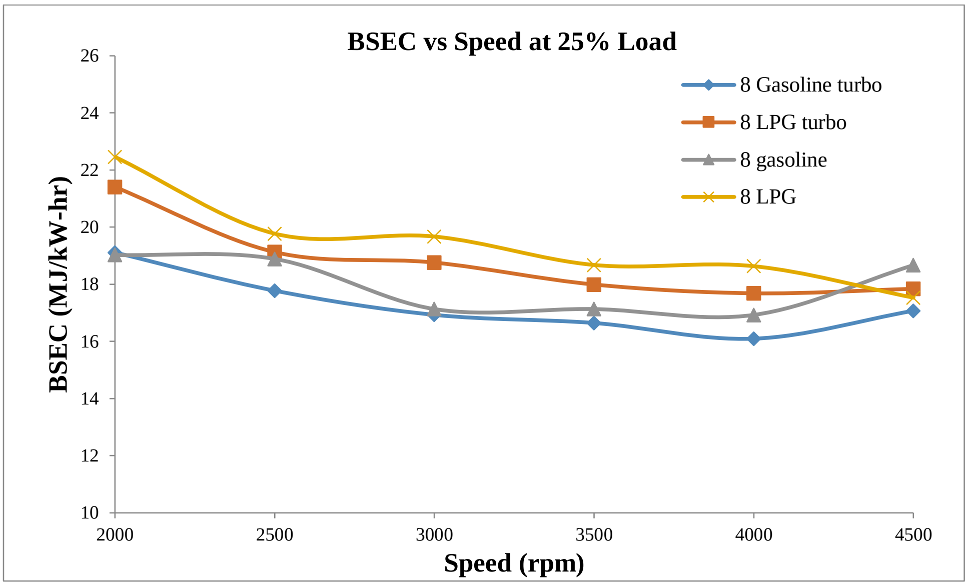


Fig. 5.213 Effect of BSEC vs Speed at 25% of full load condition.

5.3.2 Emission Characteristics

5.3.2.1 CO Emissions:

The comparison of CO emissions of gasoline, LPG with and without turbocharger at various operating speed and load conditions and it is compared with baseline condition are shown in the figure 5.214 to 5.217. From the results it can be inferred that CO emissions of LPG is far less than that of gasoline with and without turbocharger at all speed and load conditions. For LPG with turbocharger will increase CO emission when compare to without turbocharger at 8 deg. bTDC for all speed and load conditions. This slight increase in the turbocharged engine might be because of increase in the residence time due to decrease in the exhaust temperature. But there is no much variation in CO emission for turbocharged LPG engine at 8 deg. bTDC when compared to 5 deg. bTDC.

At all load condition gasoline working very richer region so CO emission will be more compared to LPG, whereas introduction of LPG will working in leaner region. And also at higher speeds combustion duration will be less so complete combustion of fuel will not take place so CO emission will more for gasoline. For gasoline with turbocharger will increase the CO emission at all speed and load conditions except for 75% load. This increase in CO emission might due to increase in the residence time. And also fuel consumption will increase which in turn increases the CO emission for gasoline.

The average value of CO emission for turbocharged gasoline at 4500 rpm and full load is 5.28%by vol., whereas for gasoline without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 5.05% by vol. and 5.51% by vol. respectively. For LPG with turbocharger at same condition is 1.55% by vol., whereas for LPG without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 0.05% by vol. and 1.69% by vol. respectively. When compared to base fuel gasoline at 5 deg. bTDC average decrease in CO emission for LPG with turbocharger is 72% at same condition.

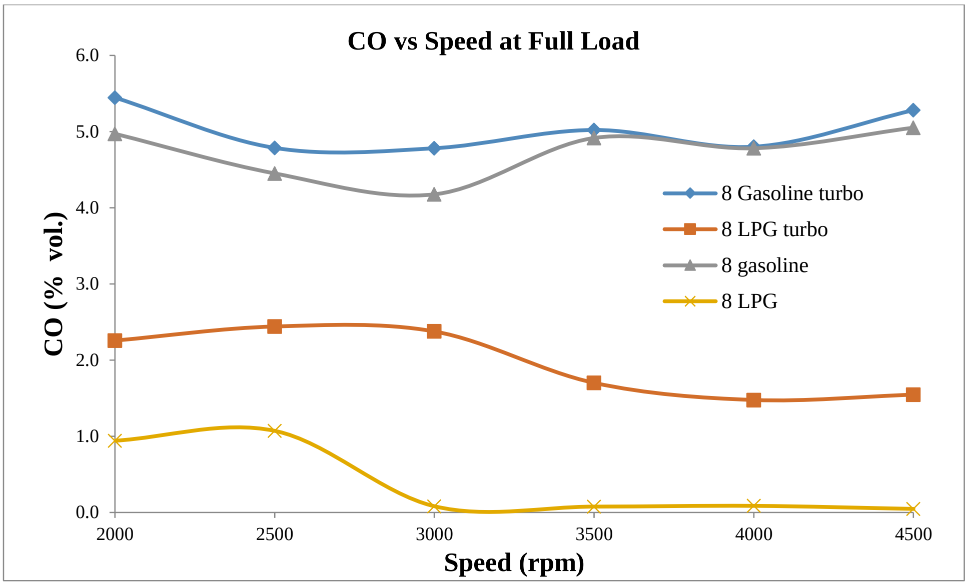


Fig. 5.214 Effect of CO emission vs Speed at full load condition.

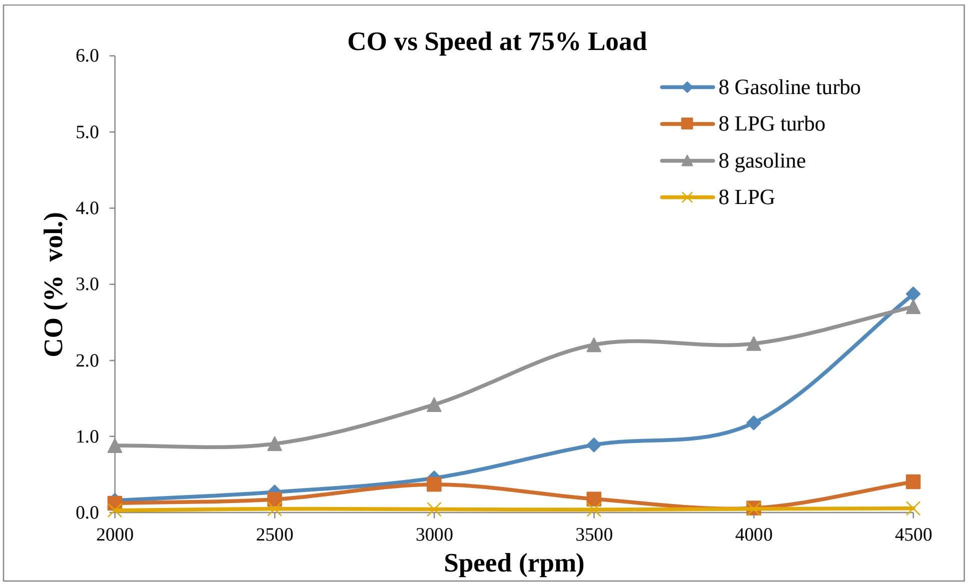


Fig. 5.215 Effect of CO emission vs Speed at 75% of full load condition.

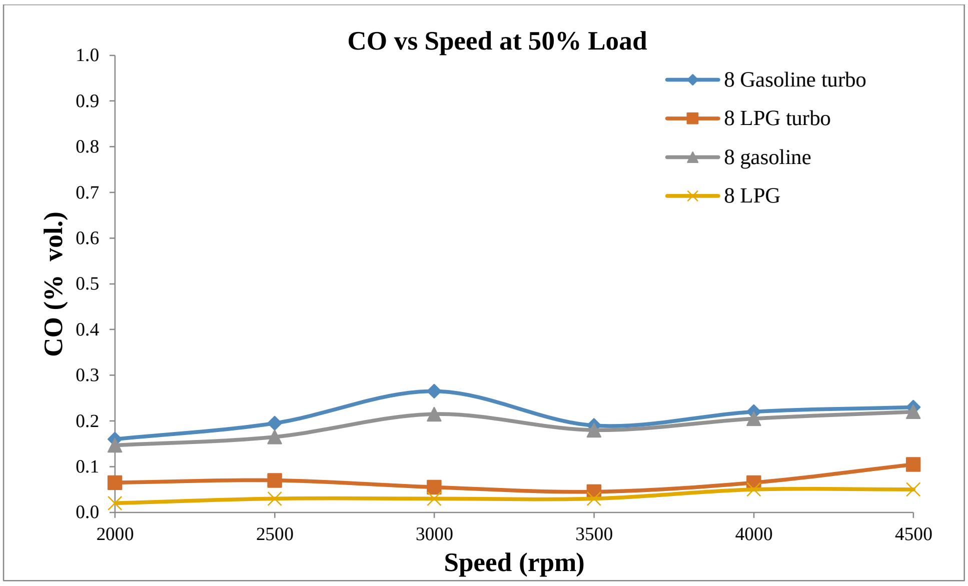


Fig. 5.216 Effect of CO emission vs Speed at 50% of full load condition.

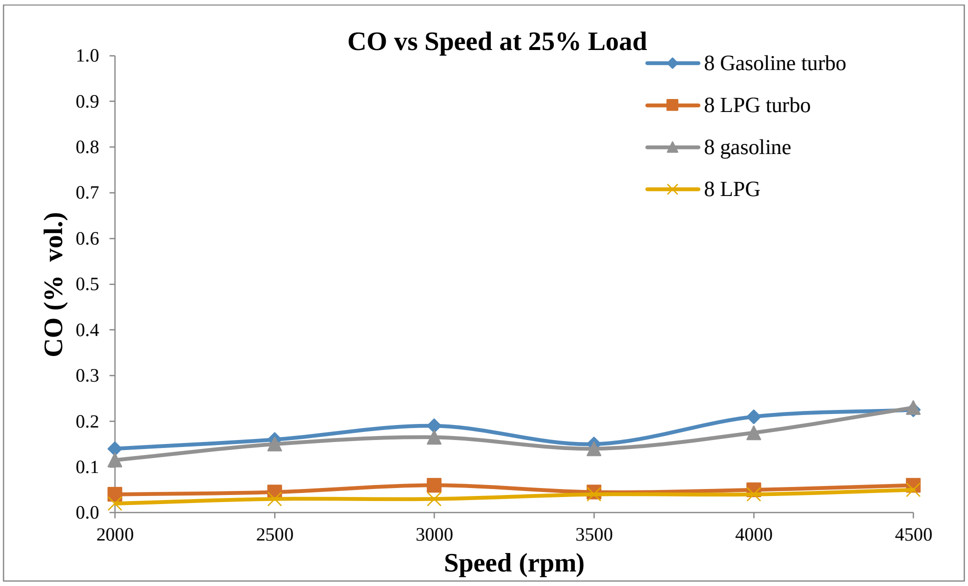


Fig. 5.217 Effect of CO emission vs Speed at 25% of full load condition.

5.3.2.2 HC Emissions:

Figure 5.218 to 5.221 shows the variation in hydrocarbon emissions of gasoline, LPG with and without turbocharger at various operating speed and load conditions and it is compared with baseline conditions. As the speed increases fuel will gets burned more and combustion will takes place completely for all load and fuels. At full load, the richer mixture will results in higher HC emissions for all fuels and engine speeds. LPG will have minimum HC emission when compared to gasoline this is because gasoline will work in richer region therefore unburned hydrocarbon emission is more. For gasoline HC emission is 4 to 3 times that of LPG at 2000 rpm 4500 rpm respectively. Turbocharged engine fuelled with LPG there will be a good decrease in the HC emission as compared to an engine without turbocharger at all load conditions. This is because advancing the ignition timing with turbocharger will lead to more time availability with increase in the oxygen content and turbulence which leads to higher oxidation of fuel hence decrease in HC emissions. But for gasoline with turbocharger also showed the same trend for HC emissions at all load conditions. This decrease in HC emission is advancing the ignition timing with turbocharger will improve the combustion of fuel due to increase in the oxygen content and more time availability during combustion.

The average decrease in the HC emission for turbocharged gasoline at 2000 rpm and full load is 30.5%, when compared to without turbocharger at 8 deg. bTDC and 23.4% when compared to 5 deg. bTDC (baseline) conditions. For LPG with turbocharger decrease in HC emission at same condition is 5%, when compared to without turbocharger at 8 deg. bTDC and -30% when compared to 5 deg. bTDC (baseline) conditions. When compared to base fuel gasoline at 5 deg. bTDC average decrease in HC emission for LPG with turbocharger is 65% at same condition.

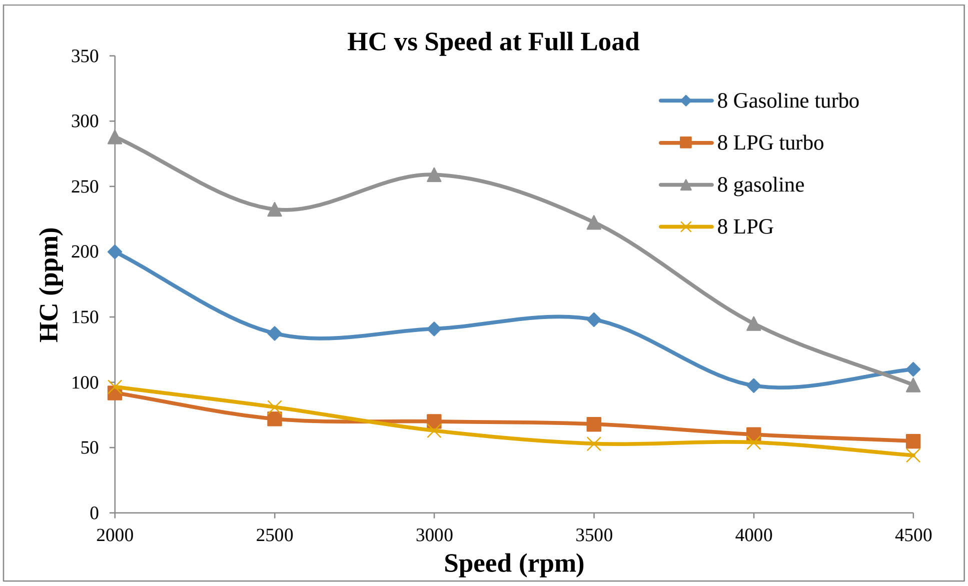


Fig. 5.218 Effect of HC emission vs Speed at full load condition.

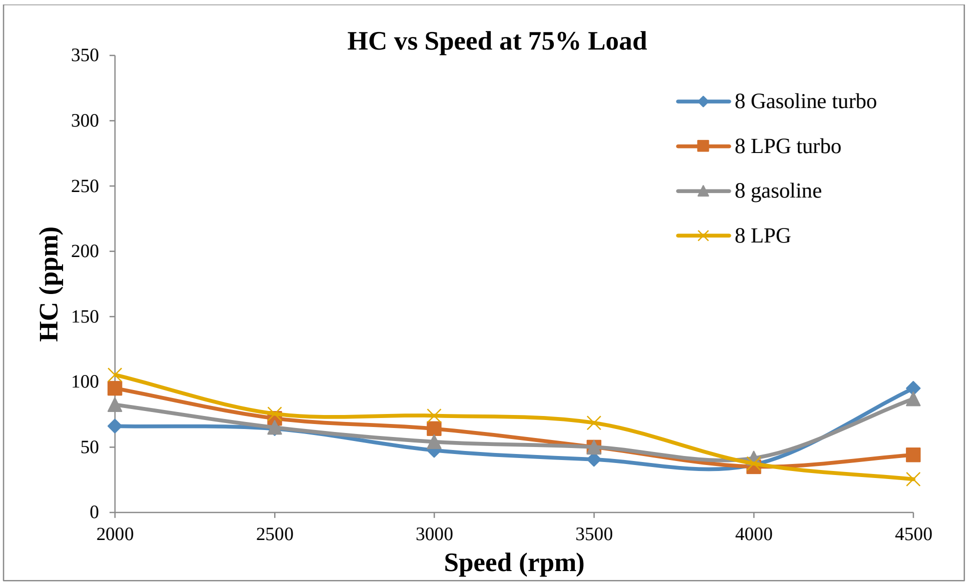


Fig. 5.219 Effect of HC emission vs Speed at 75% of full load condition.

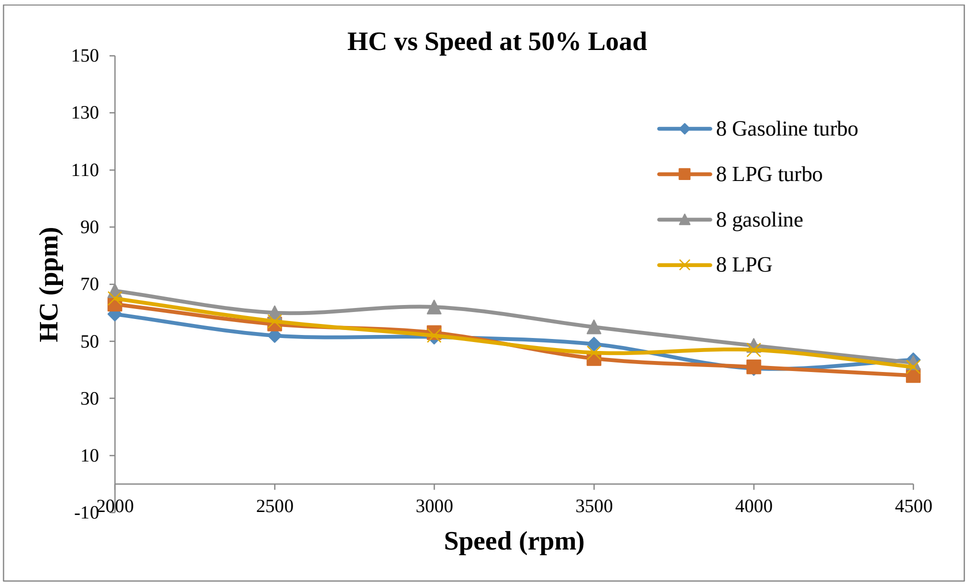


Fig. 5.220 Effect of HC emission vs Speed at 50% of full load condition.

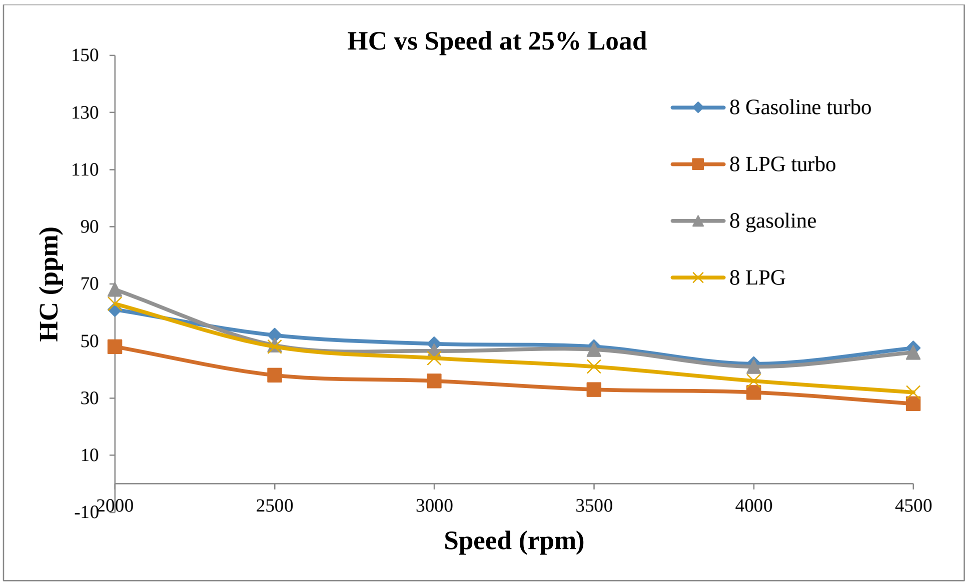


Fig. 5.221 Effect of HC emission vs Speed at 25% of full load condition.

5.3.2.3 NO_X Emissions:

Figures 5.222 to 5.225 show the variation of NO_X emission of gasoline, LPG with and without turbocharger at various operating speed and load conditions and it is compared with baseline conditions. One of the most important engine variables that effect NO_X emission is equivalence ratio. It can be seen that as the speed increases NO_X emission will almost remain constant for the gasoline but for LPG the NO_X will increases as speed increases.

Due to higher flame propagation speed of LPG at stoichiometric equivalence ratio and proper mixing of gaseous fuels with air causes an increase in the burning rate of the fuel and thus results in the complete combustion of the fuel. Hence the cylinder pressures and combustion temperatures of LPG are higher than those obtained for gasoline and other fuel mixtures. As a final outcome of this, more NO_X emissions occur in LPG combustion at higher speeds.

The turbocharged engine fuelled with LPG, there will be a good decrease in NO_X for all load conditions. This is because turbocharger will increase the charge density hence mixture becomes to lean in the combustion zone hence formation of NO_X will reduces for all load conditions. But for gasoline there is not much variations obtained in NO_X emissions. For turbocharged gasoline engine will slightly reduce the NO_X emissions.

The average decrease in the NO_X emission for turbocharged gasoline at 4500 rpm and 75% of full load is 56%, when compared to without turbocharger at 8 deg. bTDC and 12% when compared to 5 deg. bTDC (baseline) conditions. For LPG with turbocharger increase in NO_X emission at same condition is 30.7%, when compared to without turbocharger at 8 deg. bTDC and 9% when compared to 5 deg. bTDC (baseline) conditions. When compared to base fuel gasoline at 5 deg. bTDC average increase in NO_X emission for LPG with turbocharger is 282% at same condition.

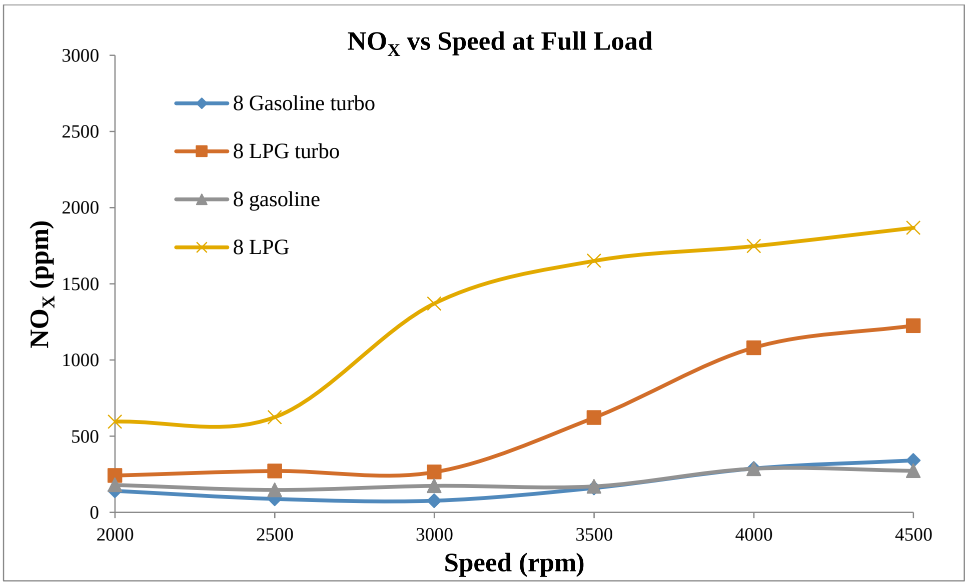


Fig. 5.222 Effect of NO_X emission vs Speed at full load condition.

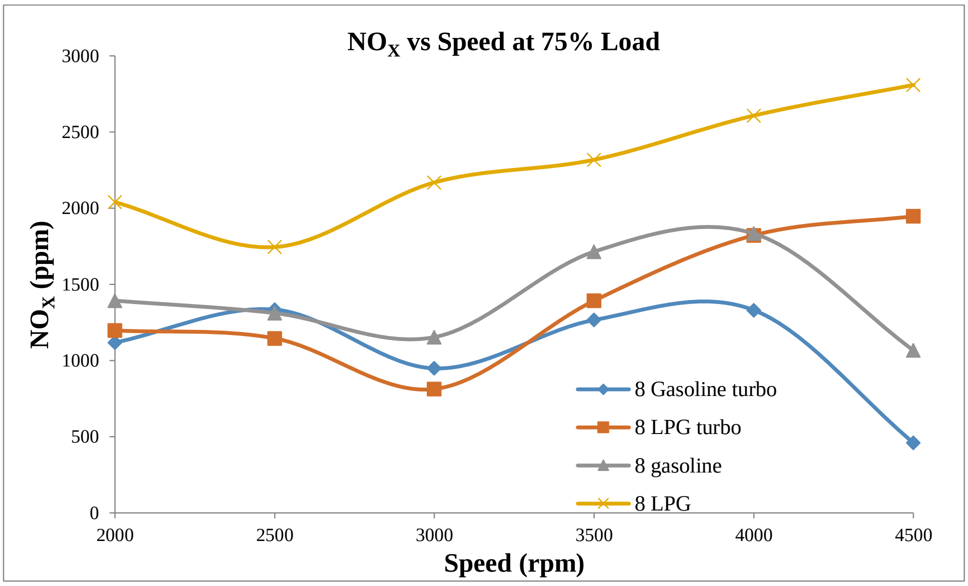


Fig. 5.223 Effect of NO_X emission vs Speed at 75% of full load condition.

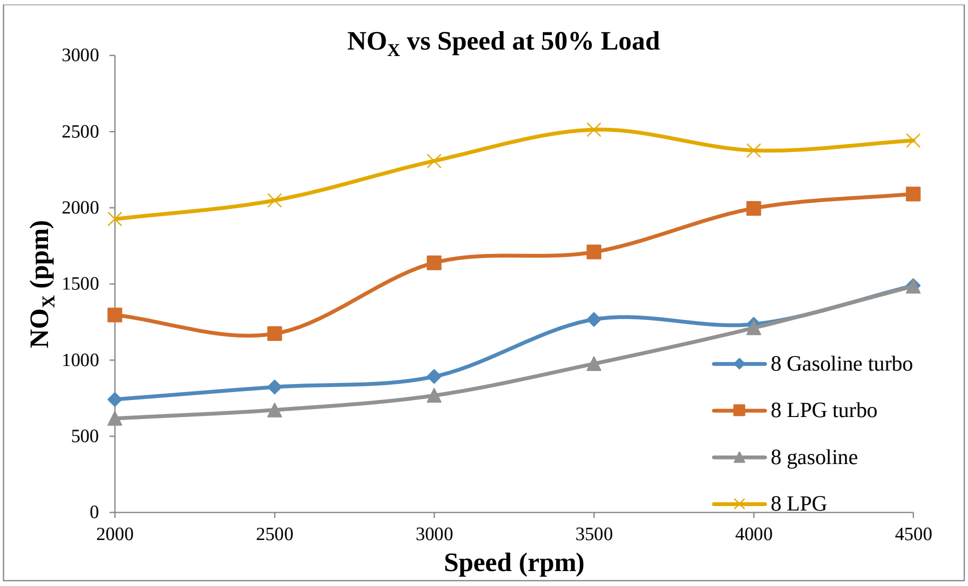


Fig. 5.224 Effect of NO_X emission vs Speed at 50% of full load condition.

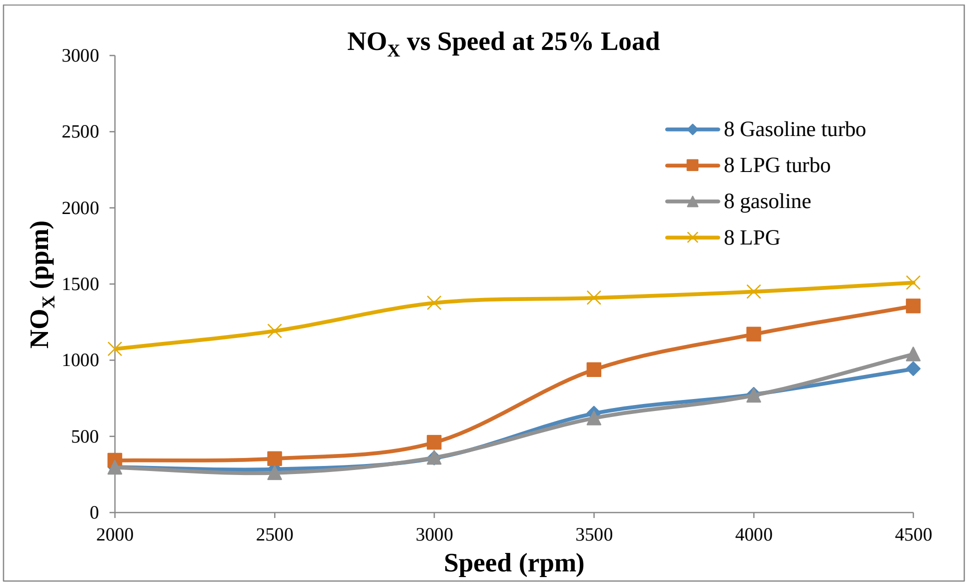


Fig. 5.225 Effect of NO_X emission vs Speed at 25% of full load condition.

5.3.3 Combustion Characteristics

5.3.3.1 P-θ Diagram

Figures 5.226 to 5.231 show the variation of the average cylinder pressure for 100 consecutive combustion cycles of gasoline, LPG with and without turbocharger at various operating speed and full load conditions, and it is compared with baseline conditions. It is observed that there is considerable variation in the pressure for the same operating conditions from one cycle to another. If we use turbocharger in an engine the peak pressure also increases. Since IMEP getting inside the combustion chamber is more because more charge density and turbulence as well as flame speed is high for LPG therefore maximum peak pressure will occur. With the use of turbocharger in gasoline fuelled engine will give very good increase in pressure as compared to LPG fuelled engine with turbocharger due good turbulence which enhances the combustion of gasoline compared to LPG. Here 360 deg. CA is the TDC point. Also from the results it is revealed the advancing the timing and using LPG will results in shifting the peak pressure location towards the TDC point. Hence with usage of LPG in SI engine enhance the combustion properties.

The average value of cylinder pressure for turbocharged gasoline at 4500 rpm and full load is 56.38 bar at 384 deg. CA, whereas for gasoline without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 40.08 bar at 380 deg. CA. and 39.15 bar at 394 deg. CA. respectively. For LPG with turbocharger at same condition is 51.52 bar at 383 deg CA, whereas for LPG without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 41.78 bar at 379 deg. CA and 40.38 bar at 378 deg. CA respectively. When compared to base fuel gasoline at 5 deg. bTDC average increase in in-cylinder pressure for LPG with turbocharger is 31.6% at same condition.

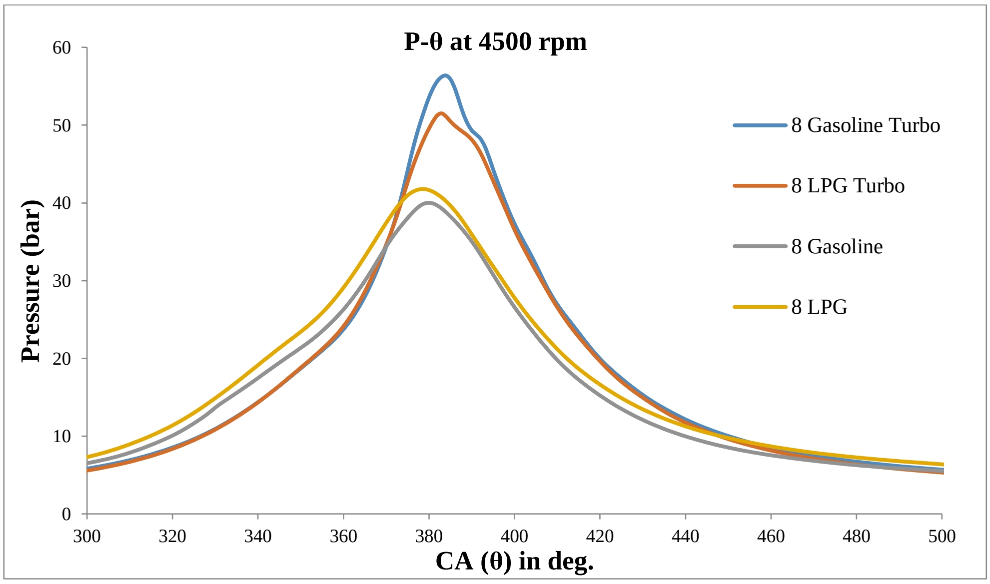


Fig. 5.226 Effect Pressure v/s Crank Angle at Full load for 4500 rpm

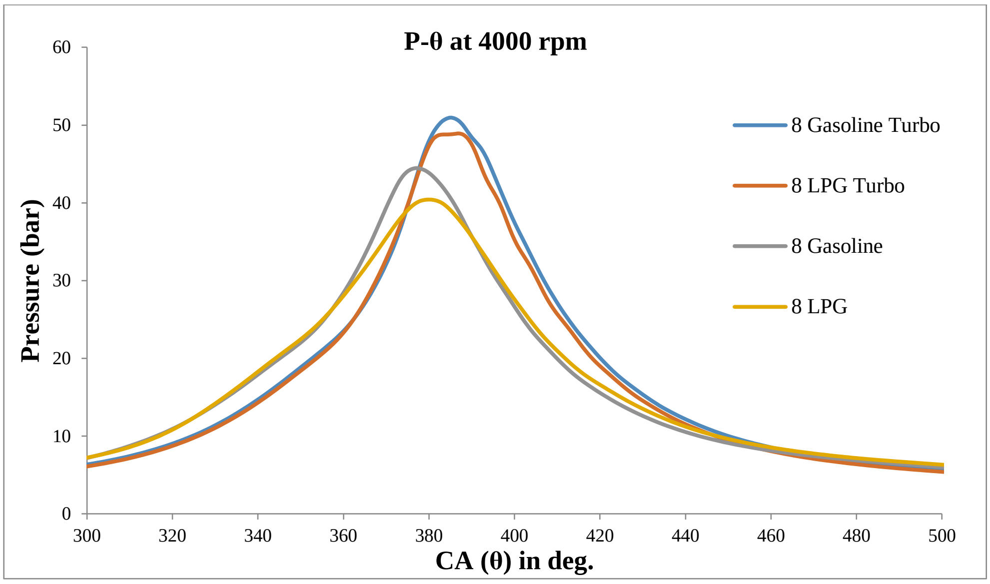


Fig. 5.227 Effect Pressure v/s Crank Angle at Full load for 4000 rpm

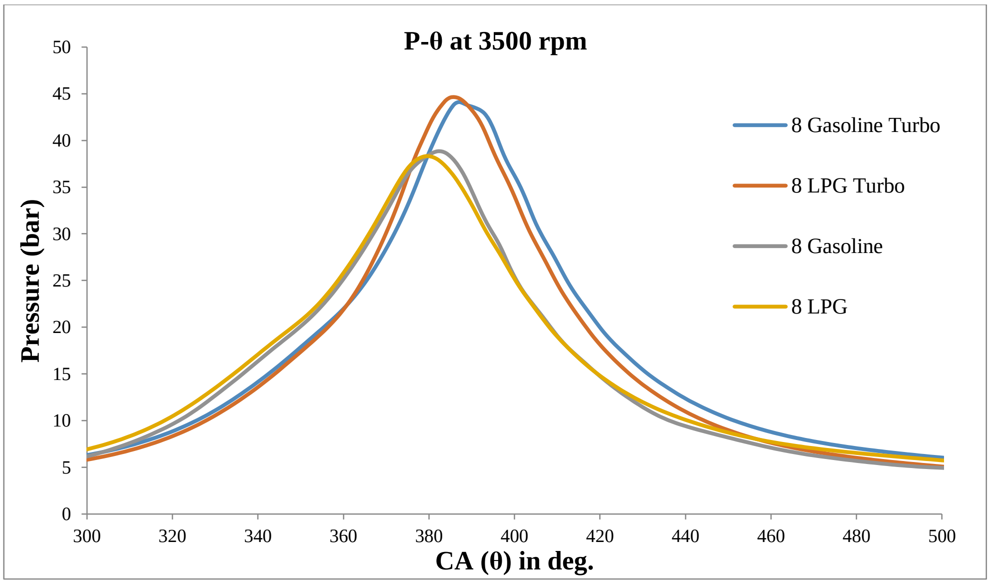


Fig. 5.228 Effect Pressure v/s Crank Angle at Full load for 3500 rpm

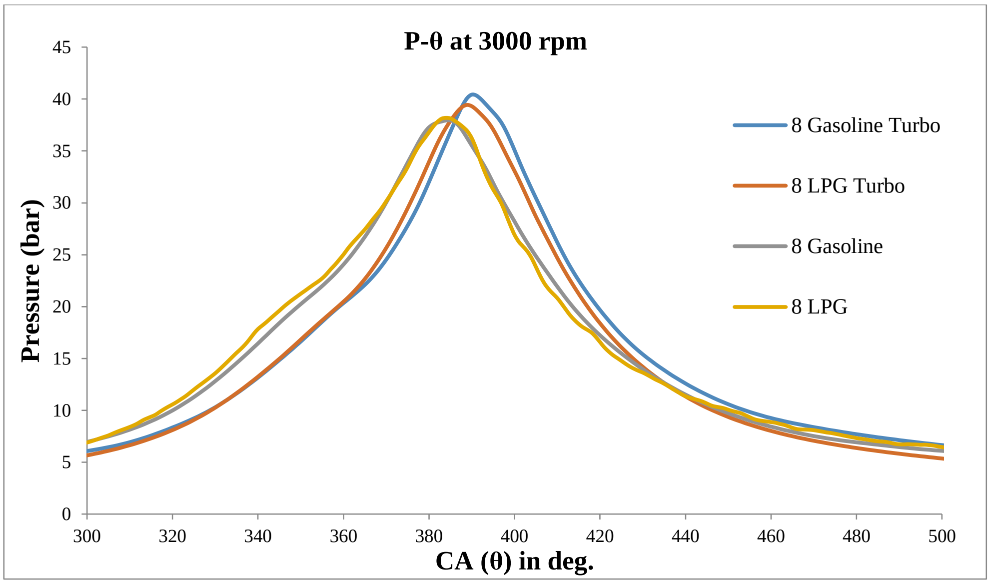


Fig. 5.229 Effect Pressure v/s Crank Angle at Full load for 3000 rpm

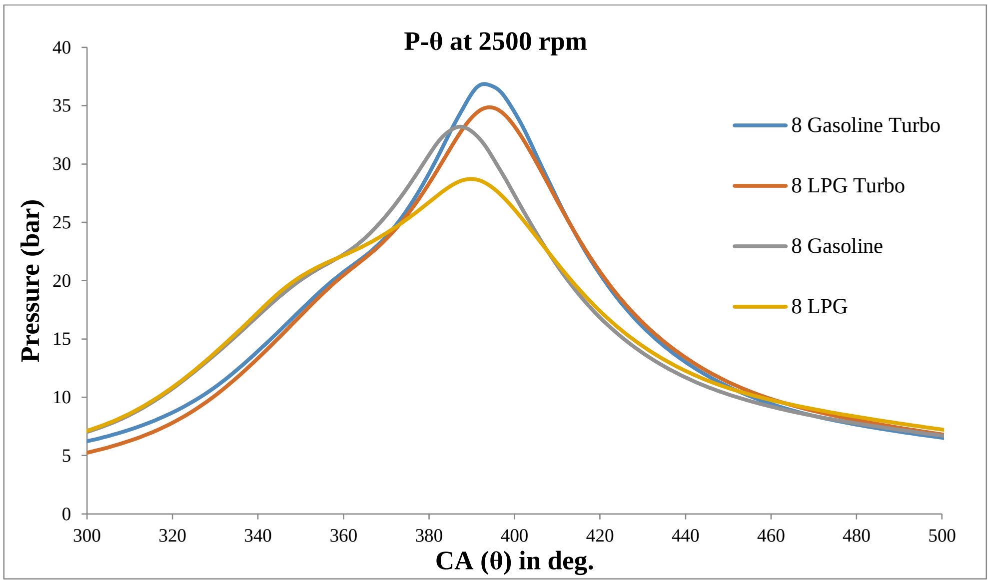


Fig. 5.230 Effect Pressure v/s Crank Angle at Full load for 2500 rpm

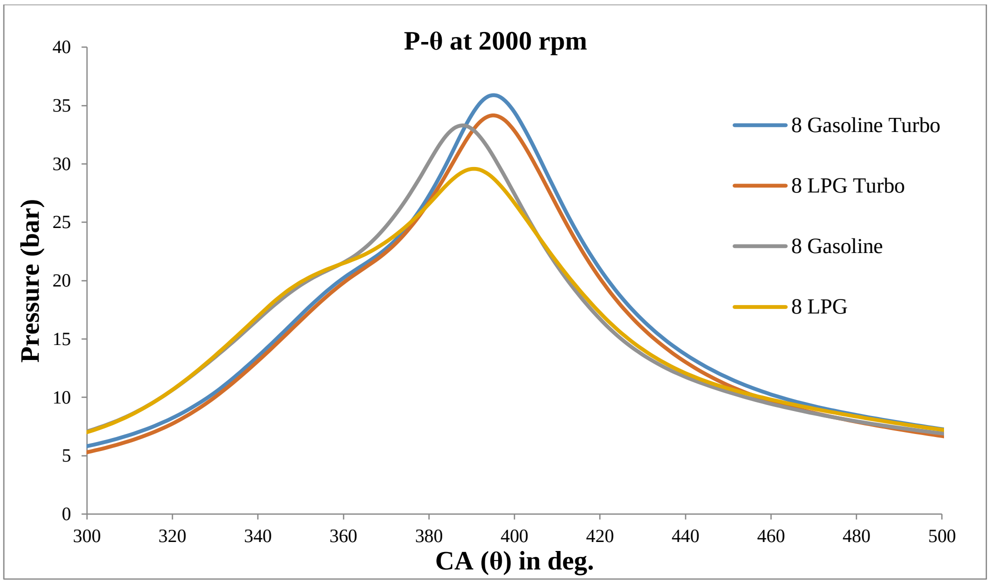


Fig. 5.231 Effect Pressure v/s Crank Angle at Full load for 2000 rpm

5.3.3.2 Net Heat release rate:

The net heat release trends for of gasoline, LPG with and without turbocharger at various operating speed and full load conditions are shown in figures 5.232 to 5.237 and it is compared with baseline conditions.. Heat release calculations are an attempt to get some information about the combustion process in an engine. Heat release rate is used in both engine performance influences in various operating conditions and same engine performances under the equal conditions. Moreover, physical and chemical properties of the fuel used in internal combustion engines are one of the main parameters which affect the heat release rate.

As seen in the figures, with the use of turbocharger in an engine heat release rate began to raise as compared to gasoline and LPG alone without turbocharger at all the speed conditions. And also, the peak locations of heat release rate of increases with increase with the use of turbocharger in an engine. This may be due to the fact that LPG has one type of hydrocarbon which will work in lean combustion. Due to this combustion chamber contains more oxygen which improves combustion and a large amount of fuel burn takes place in the areas close to TDC. For the all test fuels, heat release rate takes place in the areas close to TDC with the increasing speed for pure LPG (Eyidogan 2010). With the use of turbocharger in gasoline fuelled engine will give very good increase in heat release rate as compared to LPG fuelled engine with turbocharger. The flame propagation speed of LPG is faster than that of gasoline at the range of lean to stoichiometric equivalence ratios, but at the rich mixtures range flame speed of petrol is superior to that of LPG. Hence in turbocharger gasoline fuel will give enhanced combustion properties than LPG.

The average value of cylinder pressure for turbocharged gasoline at 4500 rpm and full load is 27.54 J/deg. CA at 379 deg. CA, whereas for gasoline without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 15.28 J/deg. CA at 382 deg. CA. and 15.54 J/deg. CA at 383 deg. CA. respectively. For LPG with turbocharger at same condition is 23.25 J/deg. CA at 378 deg CA, whereas for LPG without turbocharger at 8 deg. bTDC and 5 deg. bTDC (baseline) condition are 12.51 J/deg. CA at 372 deg. CA and 14.76 J/deg. CA at 369 deg. CA respectively. When compared to base fuel gasoline at 5 deg. bTDC average increase in in-cylinder pressure for LPG with turbocharger is 49.6% at same condition.

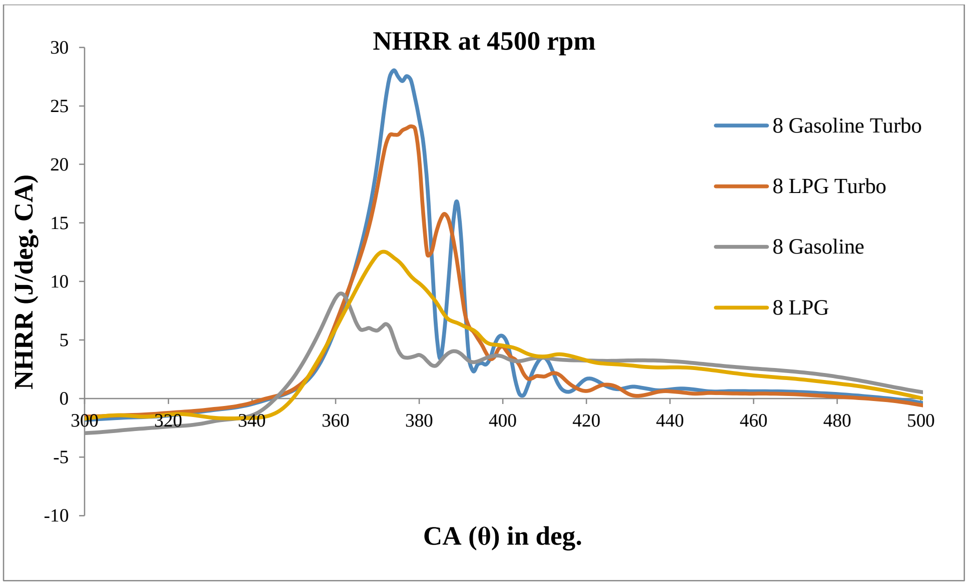


Fig. 5.232 Effect NHRR v/s Crank Angle at Full load for 4500 rpm

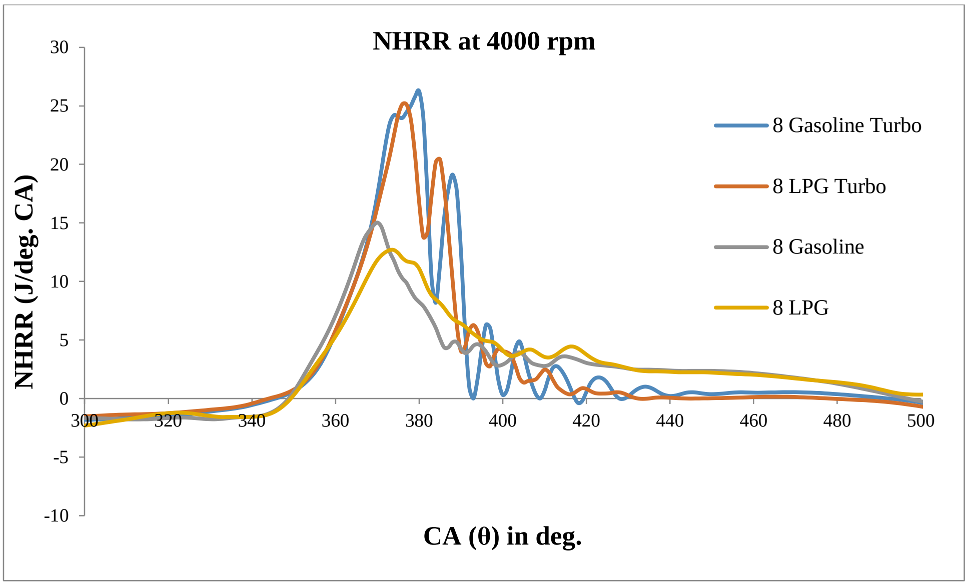


Fig. 5.233 Effect NHRR v/s Crank Angle at Full load for 4000 rpm

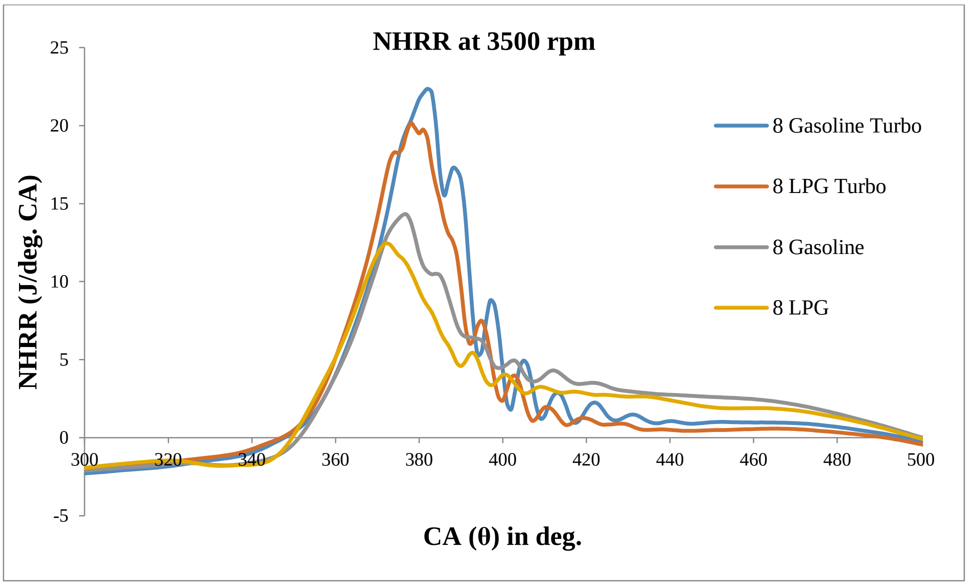


Fig. 5.234 Effect NHRR v/s Crank Angle at Full load for 3500 rpm

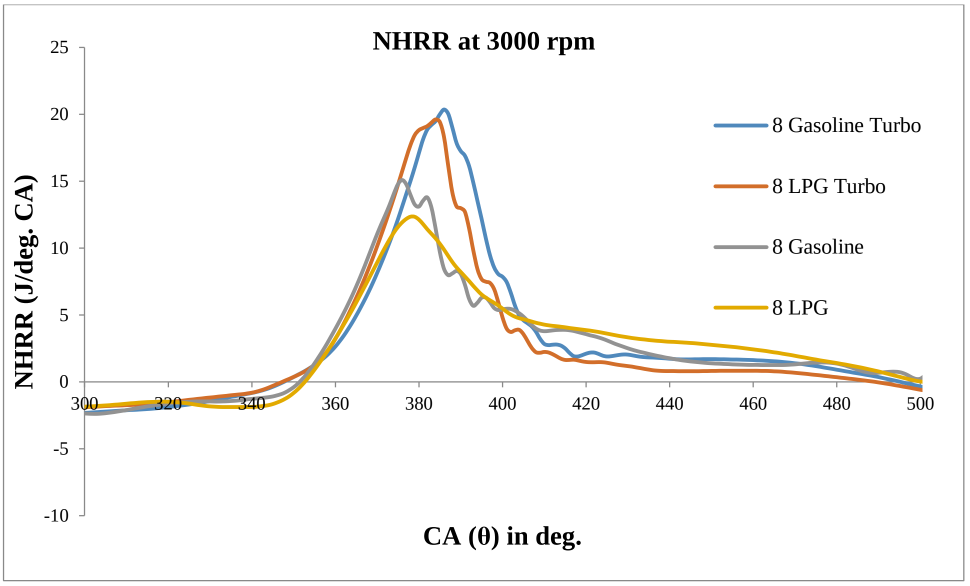


Fig. 5.235 Effect NHRR v/s Crank Angle at Full load for 3000 rpm

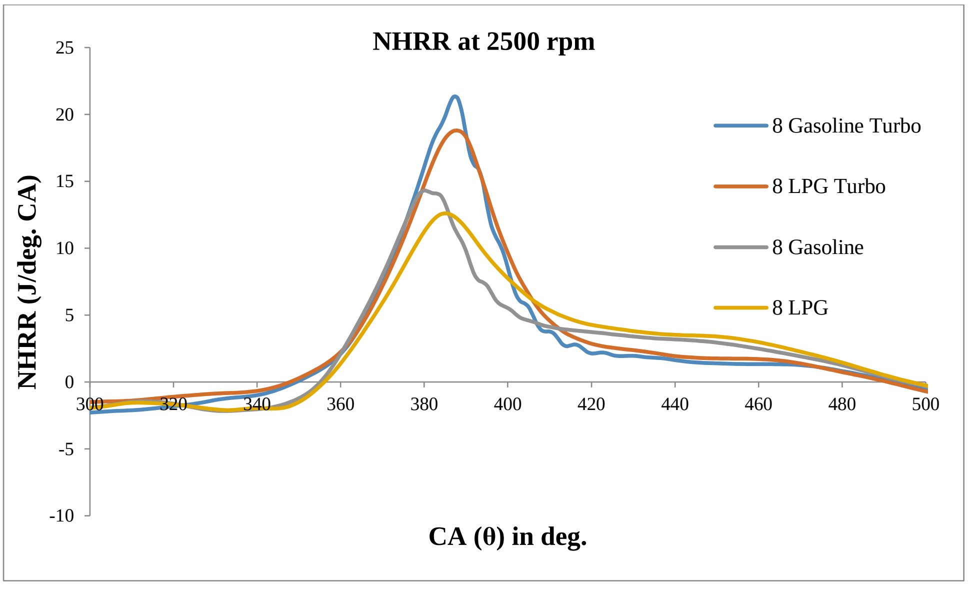


Fig. 5.236 Effect NHRR v/s Crank Angle at Full load for 2500 rpm

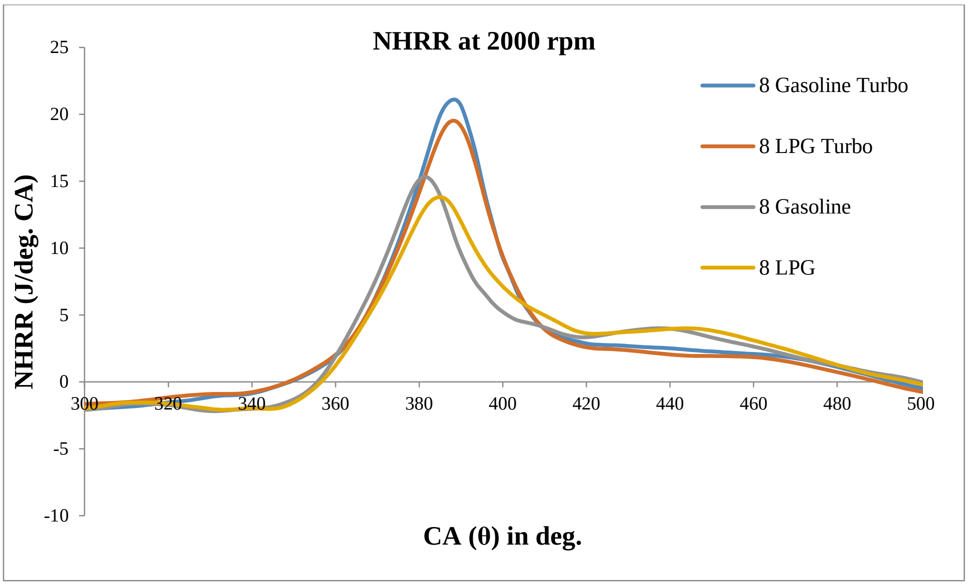


Fig. 5.237 Effect NHRR v/s Crank Angle at Full load for 2000 rpm

5.3.3.3 IMEP and COV of IMEP

Figure 5.238 to 5.243 shows the variation in IMEP with 100 consecutive cycles at full load and different speed. Graph will indicates how IMEP inside engine cylinder is varies from the cycle to cycle. From the figures it is evident that as the turbocharger engine will improves the combustion properties by reducing the variation from cycle to cycle for both gasoline and LPG when compared with without turbocharger. From the IMEP trends it can be observed that the fluctuations are high for gasoline and LPG without turbocharger and it is lower for LPG with turbocharger. Also from figure 5.244, which indicates COV of IMEP is less for LPG with turbocharger hence LPG with turbocharger will enhance the combustion properties with increase in engine life. This decrease in the COV of IMEP for turbocharged engine might be more homogeneous mixture formed due to turbulence within the cylinder. Also LPG having good lean combustion characteristics with wider flammability limit when compared to gasoline which decreases the variations inside cylinder thus combustion stability is increases for LPG. Also due to higher flame propagation speed of LPG in lean condition will decrease the combustion duration hence cycle by cycle variations are reduced. Therefor LPG with turbocharger will have a greater combustion stability when compared to the gasoline.

The decreased in COV of IMEP for LPG with turbocharger are 71% and 84% when compared to LPG and gasoline without turbocharger at 4500 rpm.

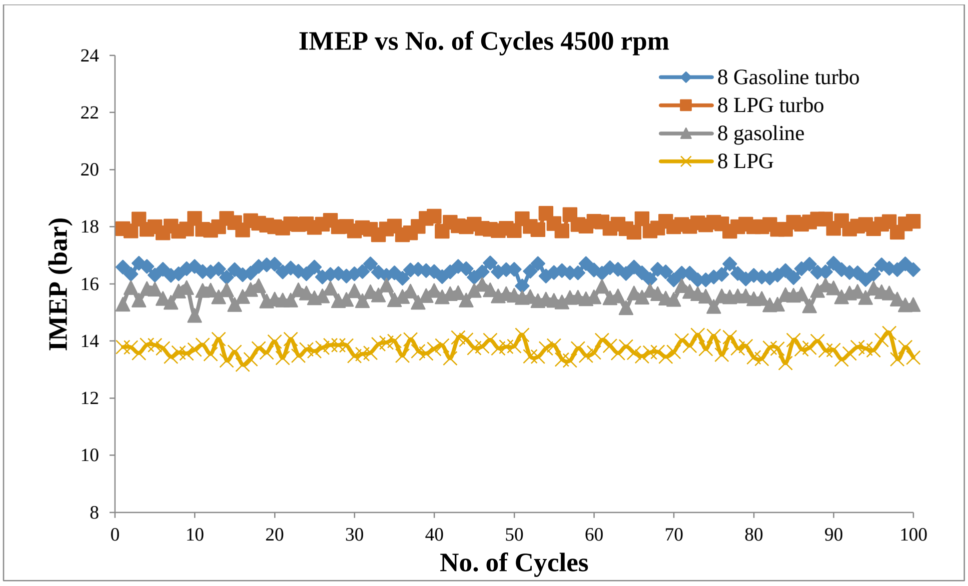


Fig. 5.238 Variation of IMEP vs No. of cycles at Full load for 4500 rpm

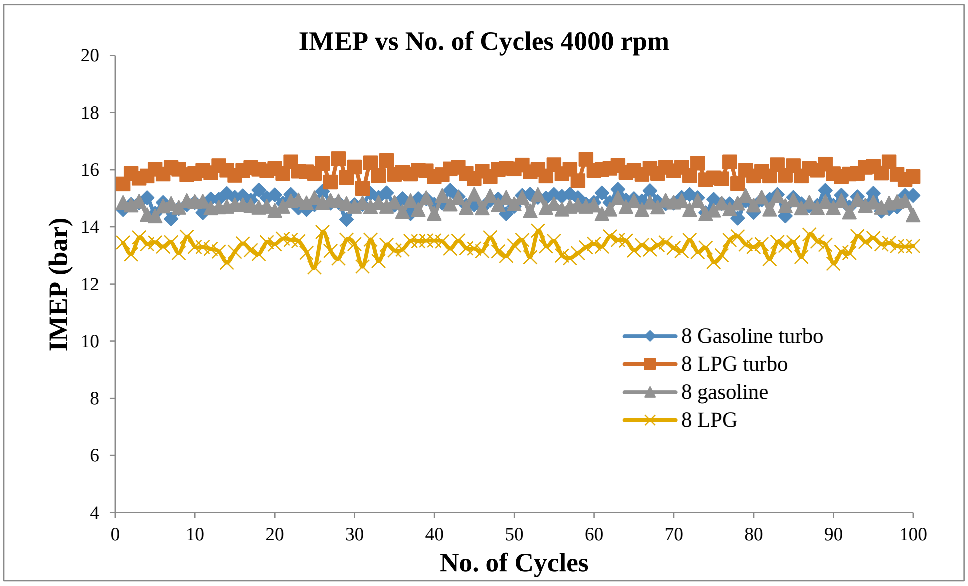


Fig. 5.239 Variation of IMEP vs No. of cycles at Full load for 4000 rpm

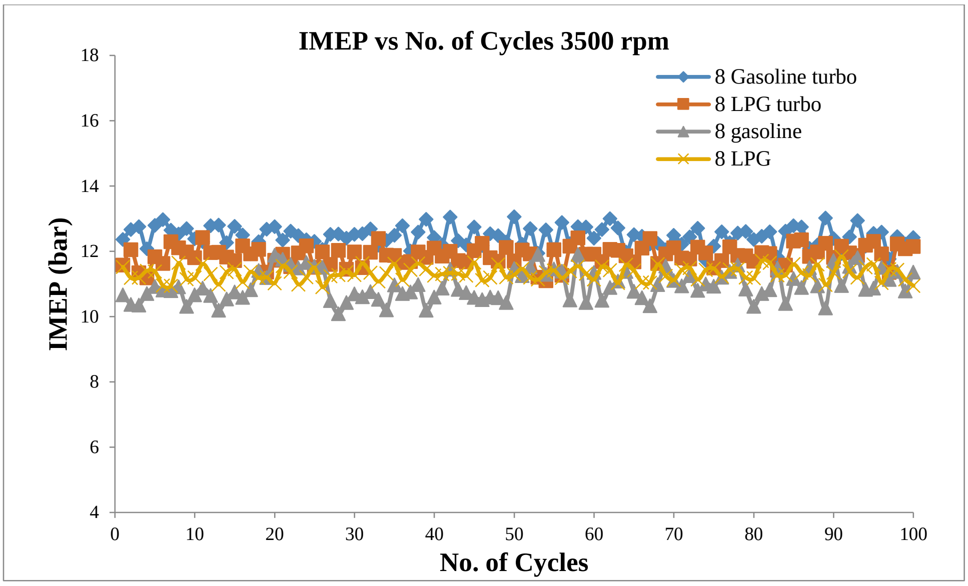


Fig. 5.240 Variation of IMEP vs No. of cycles at Full load for 3500 rpm

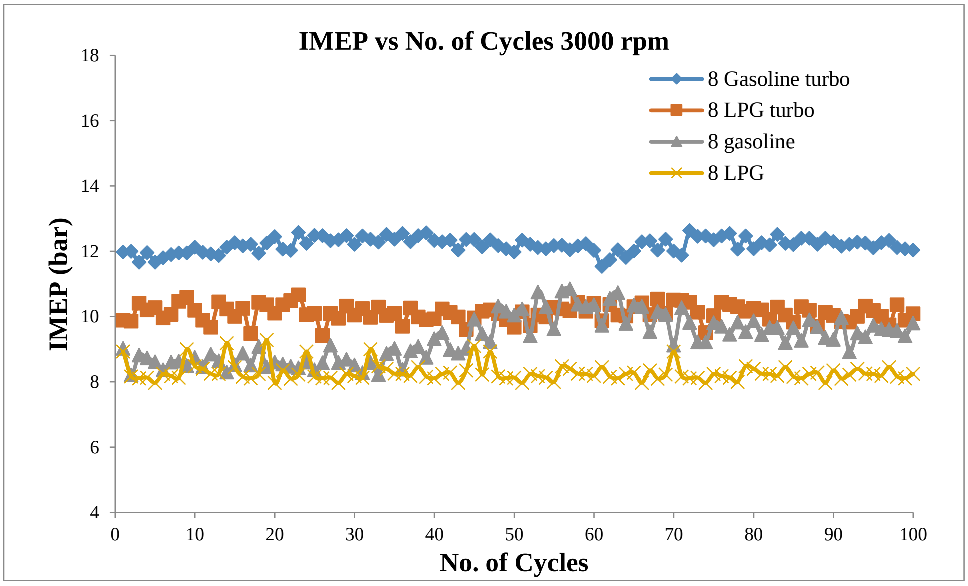


Fig. 5.241 Variation of IMEP vs No. of cycles at Full load for 3000 rpm

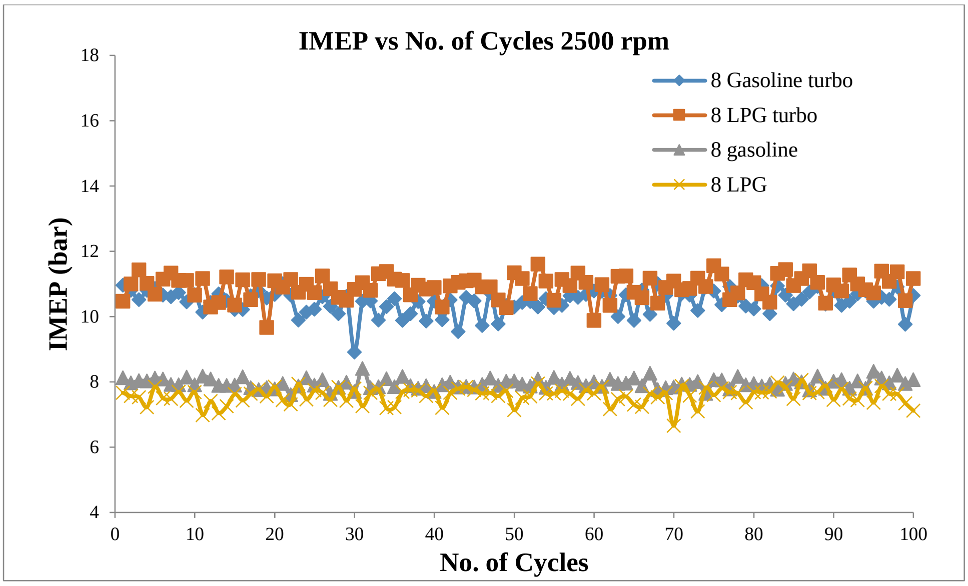


Fig. 5.242 Variation of IMEP vs No. of cycles at Full load for 2500 rpm

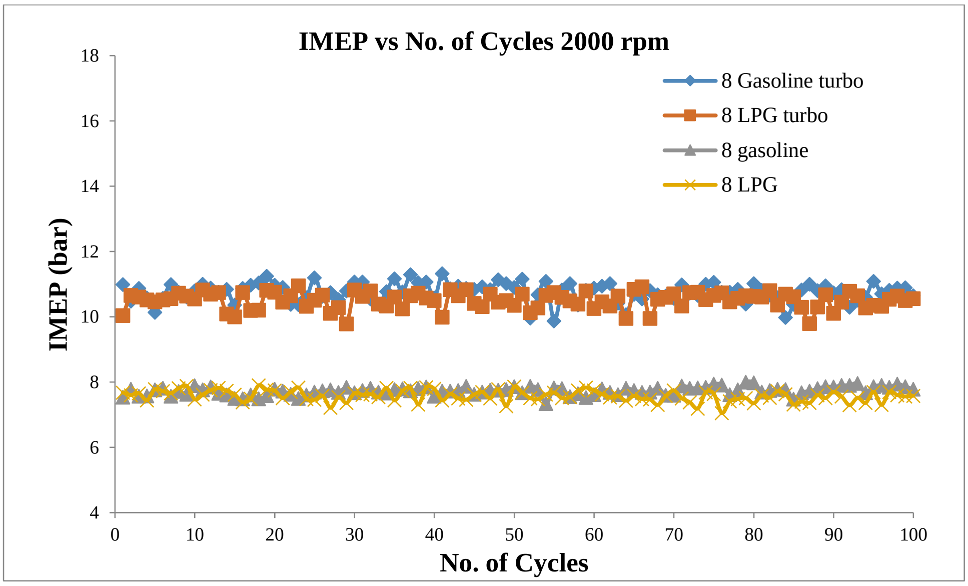


Fig. 5.243 Variation of IMEP vs No. of cycles at Full load for 2000 rpm

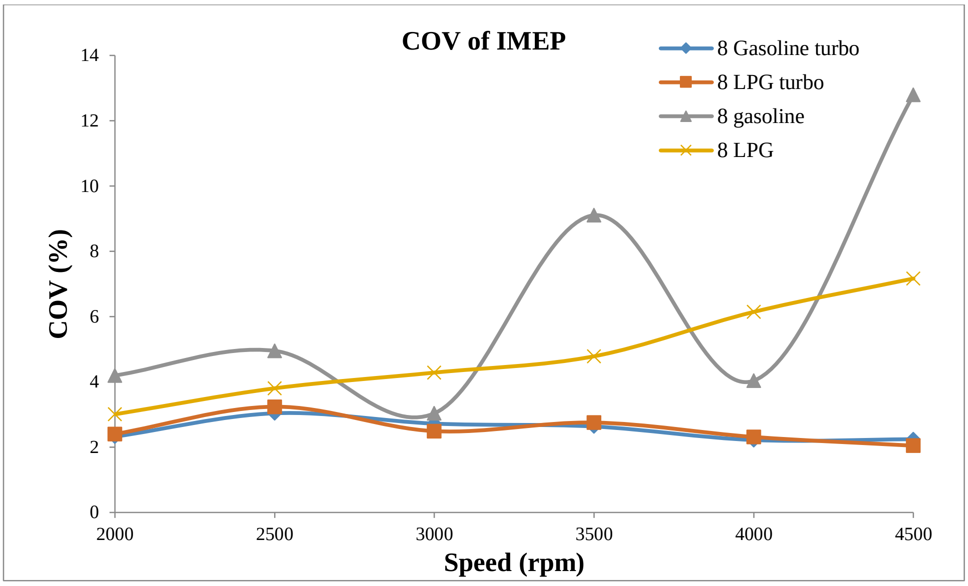


Fig. 5.244 COV of IMEP vs speed at Full load.

Summary:

- Using turbocharger performance characteristics are improved. For 100% LPG with turbocharger BTE is increased when compared to gasoline with turbocharger. BTE obtained is maximum at 8 deg. bTDC with turbocharger for 100% LPG when compared to all other condition.
- Emissions are greatly reduced with turbocharger with 100% LPG when compared to gasoline with turbocharger.
- Maximum of 56% reduction in the NO_X emission is obtained for a turbocharged LPG fueled engine at full load.
- In-cylinder pressure and NHRR also greatly improved with usage of turbocharger. Maximum of 17 bar increase in the in-cylinder pressure is obtained with usage of turbocharger.
- With turbocharger cycle by cycle variations are reduced. The decreased in COV of IMEP for LPG with turbocharger are 71% and 84% when compared to LPG and gasoline without turbocharger at 4500 rpm. Turbocharged engine will have better combustion stability than natural aspirated engine.
- Turbocharger will give the better combustion, performance and emission characteristics for LPG fuel.

5.4 EFFECT OF VAPORISED WATER-METHANOL INDUCTION

In previous section turbocharger with 100% LPG at 8 deg. bTDC static ignition timing gave better combustion and performance characteristics. Now, to improve the emission characteristics of LPG fueled turbocharged engine, vapor of water-methanol induction system has been used. In this stage, separate system has been developed to produce and control the vapor of water-methanol induction to intake manifold. A set of experiments has been conducted for 100% LPG in a turbocharged engine by varying percentage vapor of water-methanol (10%, 20% and 30% of fuel consumption) by mass. Experiments are conducted at full load condition to analyze the performance, combustion and emission characteristics. Engine speed is varied from 2000 to 4500 rpm in steps of 500 rpm at optimized 8 deg. bTDC static ignition timing for 100% LPG with turbocharger.

5.4.1 Performance characteristics

5.4.1.1 Brake Power

The figure 5.245 shows the variations of brake power for vaporized water-methanol induction at full load condition for LPG fuel with turbocharger. It can be seen that with induction of vaporized water methanol to the engine there is not much effect on BP. Although induction of vaporized water methanol reduces the volumetric efficiency of the engine as seen from the figure 5.240, the oxygen additive in the induction will nullify the power reduction of the engine. Therefore addition of methanol along with the water will maintain the same power output and load carrying capacity of the engine.

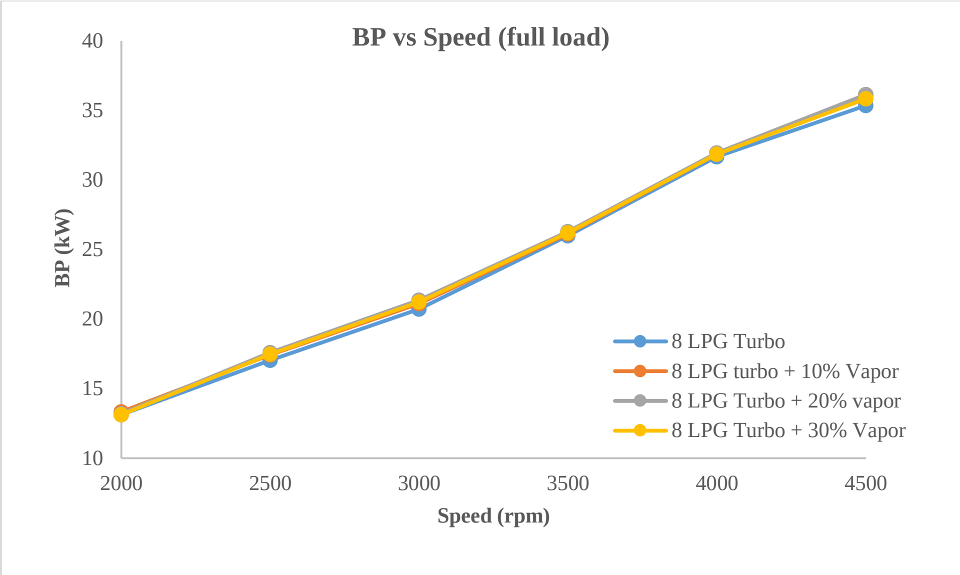


Fig. 5.245 Effect BP v/s Speed at Full load condition.

5.4.1.2 Brake Thermal Efficiency

The variations of brake thermal efficiency with engine for different percentage of vapor of water methanol flow rate at full load condition is as shown in figure 5.246. As the percentage of water methanol increases the brake thermal efficiency will increases till 20% of flow rate then start to decreases. It can be said that the reason of this improvement in the efficiency is owing to the enthalpy increase and better atomization is carried out with inducted vapor into the cylinder. Also vapor of methanol with steam increases the oxygen content in the cylinder and better combustion will takes place. Due to this there if the reduction in the fuel consumption as seen from figure 5.242, which owes towards increase in the thermal efficiency. Beyond 20% of vapor, there drastic reduction in the volumetric efficiency which in turn increases the fuel consumption and hence decrease in the thermal efficiency. Average increase in brake thermal efficiency is 1.4%, 2.2% and 2.2% for 10%, 20% and 30% vapor respectively when compared to LPG with turbocharger at 4500rpm.

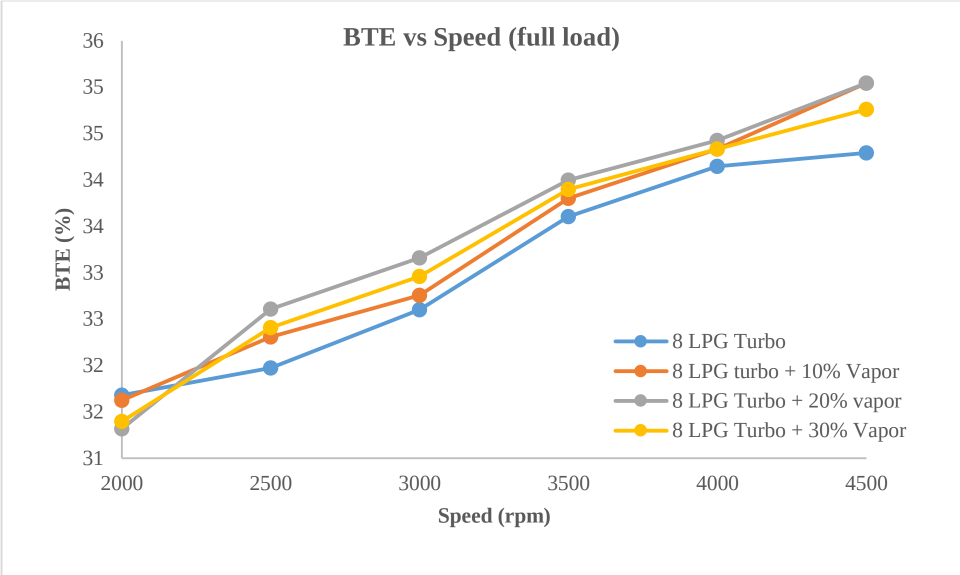


Fig. 5.246 Effect BTE v/s Speed at Full load condition.

5.4.1.3 Equivalence ratio

An introduction of vapor of water methanol in the intake manifold will reduce the mass of air consumption as it displaces air in the intake manifold. Hence equivalence ratio will be higher. Figure 5.247 shows the variation of equivalence ratio for vaporized water methanol induction for different speed at full load condition and compared with turbocharged LPG engine. From the graph it is revealed that as the percentage of vapor increases equivalence ratio also increases which indicates engine with vaporized water methanol will work in a slightly richer region. This is because introduction of vapor will reduces the volumetric efficiency hence engine will work in richer region.

5.4.1.4 Volumetric efficiency

Figure 5.248 shows the effect of vaporized water methanol induction on the volumetric efficiency for different speed condition at full load of the engine. From the graph it is clear that as the percentage of vapor increases volumetric efficiency is going to decrease for all speed condition. This is because as vapor is inducted into the intake manifold which displaces the air hence mass flow rate of intake air will decreases. Since mass flow rate of air decreases volumetric efficiency also decreases. Also vapor of methanol present along with steam will compensate the reduction in the volumetric efficiency hence there is no reduction in the power output of the engine. This is the advantage of oxygen additive along with the steam which increases the performance characteristics of the engine. The average reduction in the volumetric efficiency are 3% for 10% vapor, 4.6% for 20% vapor and 6.9% for 30% vapor obtained when compared without vapor induction.

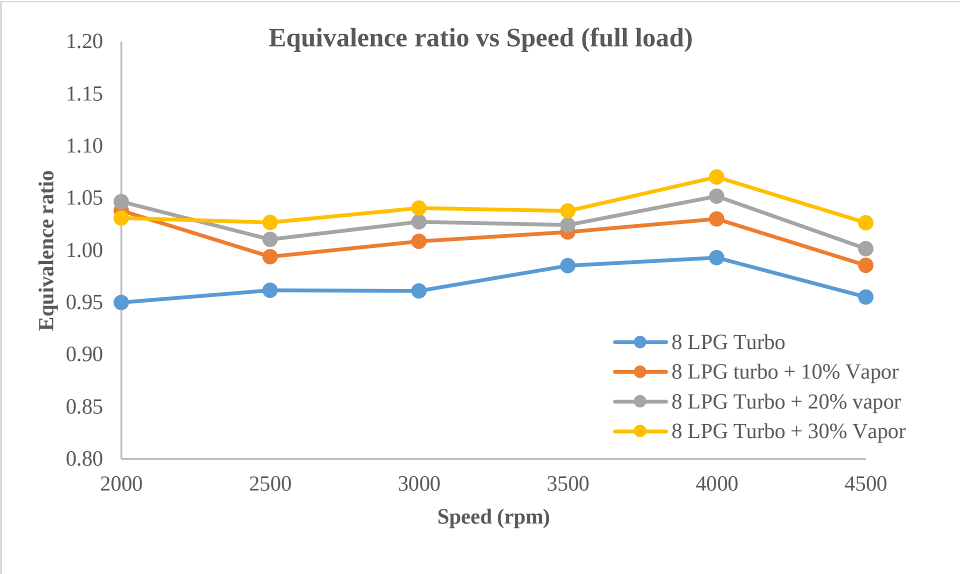


Fig. 5.247 Effect Equivalence ratio v/s Speed at Full load condition.

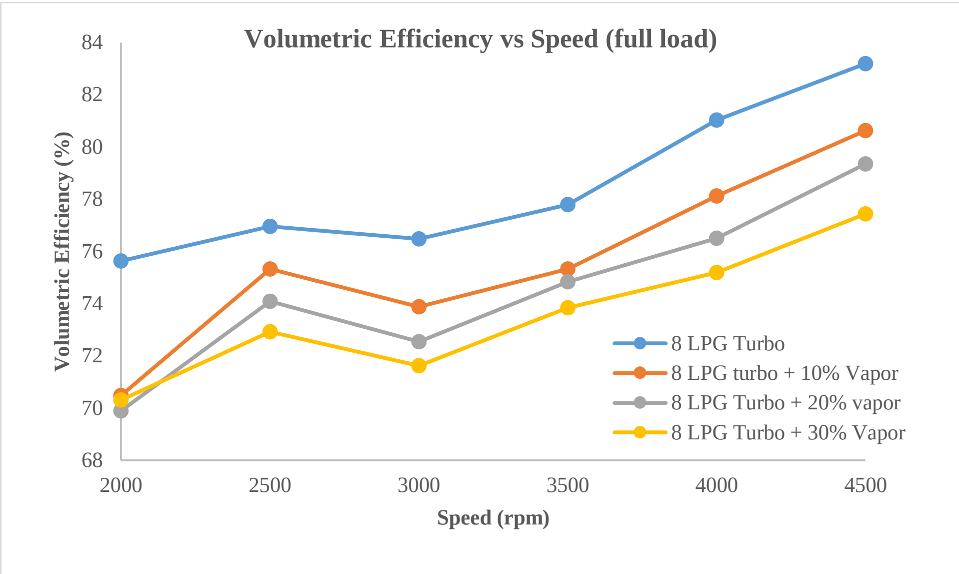


Fig. 5.248 Effect volumetric efficiency v/s Speed at Full load condition.

5.4.1.5 Brake specific energy consumption

BSEC will indicates the fuel economy of the engine. Figure 5.249 indicates the effect of vaporized water methanol induction on the BSEC for different speed condition at full load of the engine. From the experiment reading it is revealed that as the percentage of vapor increase energy consumed to produce power will decreases which indicates the fuel consumption is reduced. This is due to introduction of water methanol vapor will improvement in vaporization and mixing processes which leads to a shorter combustion reaction. Also presence of oxygen additive along with the steam will improve the combustion inside the cylinder. Hence BSEC will reduced for vaporized water methanol induction. Average decrease in the BSEC are 1.3%, 2.1% and 2.1% for 10%, 20%% and 30% of vapor when compared to without vapor condition at 4500 rpm.

5.4.2 Emission characteristics

5.4.2.1 CO Emission

Figure 5.250 depicts the variations of CO emission with engine speed for vaporized water-methanol induction at full load condition for LPG fuel with turbocharger. From the graph it indicates as the percentage of vapor introduced (i.e. 10% vapor) there is decrease in the CO emission, but further increase in the vapor there is not much change in CO emission. This is because introduction of methanol along with water increases the oxygen content in the combustion chamber hence enhancement in the combustion inside cylinder takes place. But further increase in the percentage vapor will reduces the volumetric efficiency hence there is not much variations in the CO emissions. Average decrease in the CO emissions are 13%, 14.8% and 14.2% for 10%, 20%% and 30% of vapor when compared to without vapor condition at 4500 rpm.

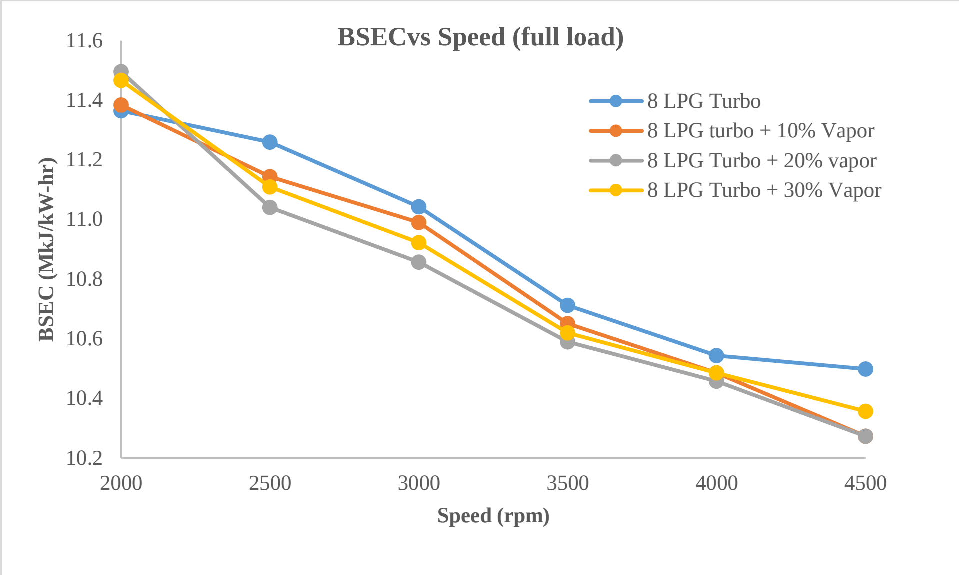


Fig. 5.249 Effect BSEC v/s Speed at Full load condition.

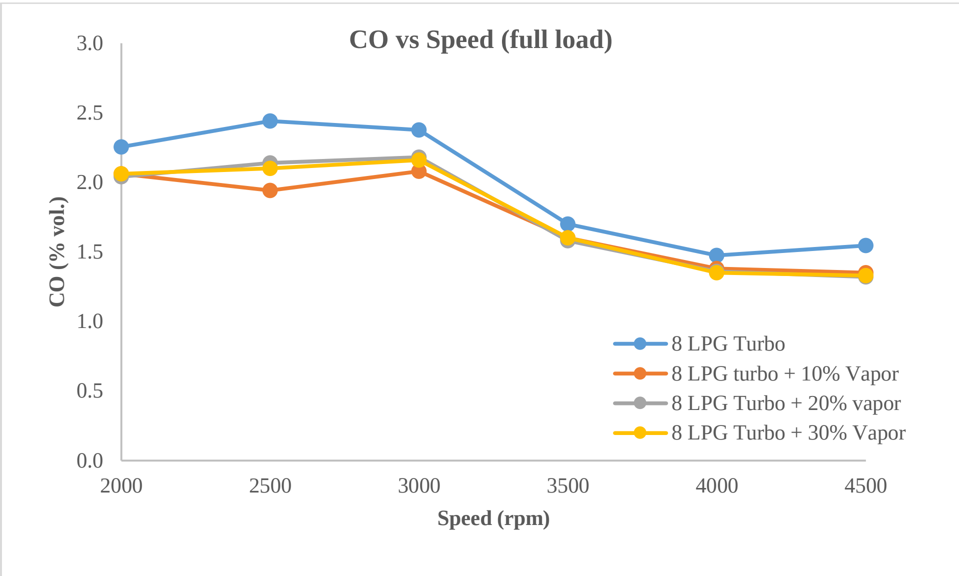


Fig. 5.250 Effect CO v/s Speed at Full load condition.

5.4.2.2 HC Emission

Figure 5.251 depicts the variations of HC emission with engine speed for vaporized water-methanol induction at full load condition for LPG fuel with turbocharger. From the experiment results it specifies as the percentage of vapor increases HC emission is decreased. The reduction in HC could be explained owing to the presence of the fuel vapor interface with very low interfacial tension which causes to a better atomization of the fuel during injection. Higher contact with the air during the burning process is resulted in a finer dispersion of the fuel droplets. Another possible reason reducing in HC may be explained with the improvement in vaporization and mixing processes which leads to a shorter combustion reaction. Average decrease in the HC emissions are 18%, 23.5% and 21.8% for 10%, 20% and 30% of vapor when compared to without vapor condition at 4500 rpm.

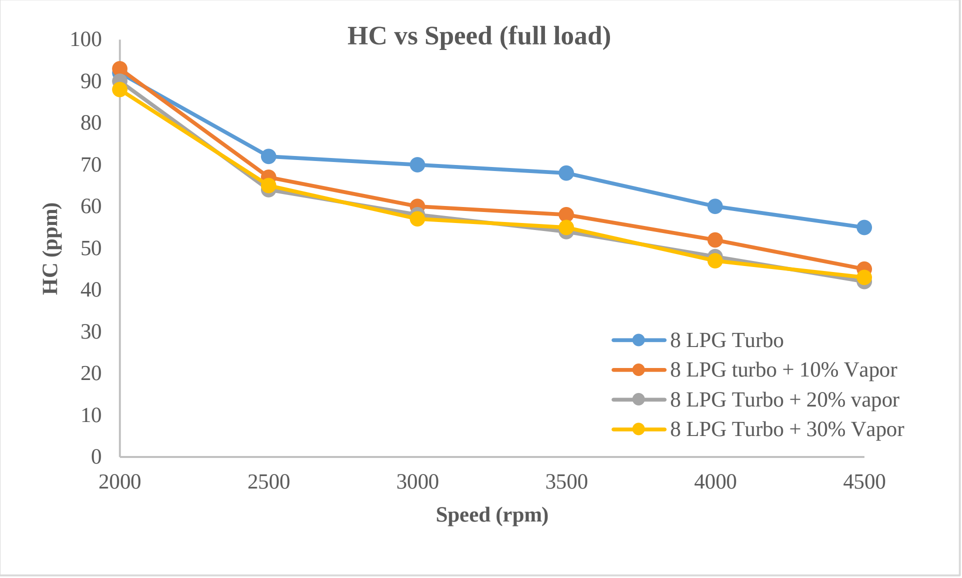


Fig. 5.251 Effect HC v/s Speed at Full load condition.

5.4.2.3 NOx Emission

Figure 5.252 depicts the variations of NOx emission with engine speed for vaporized water-methanol induction at full load condition for LPG fuel with turbocharger. From the experiment findings it implies as the percentage of vapor increases NOx emission is decreased. The interest in water injection techniques is due to the fact that water in the form of micrometer sized droplets exerts some positive effects on the combustion of the fuel and exhaust emissions, frequently NOx. Steam injection to SI engines also showed same positive effects as water injection techniques, such as reduced NOx and improved combustion efficiency. It can be concluded that the finely atomized droplets are mixed with air instantly and more homogenously after being injected into the combustion chamber. The combined effect of vaporization absorbing heat, relatively high molar heat capacity of water and increased partial pressure of oxygen puts down the peak combustion temperature and thus decreases the nitrogen oxides formation. Average decrease in the NOx emissions are 34.3%, 37.5% and 40.75% for 10%, 20% and 30% of vapor when compared to without vapor condition at 4500 rpm.

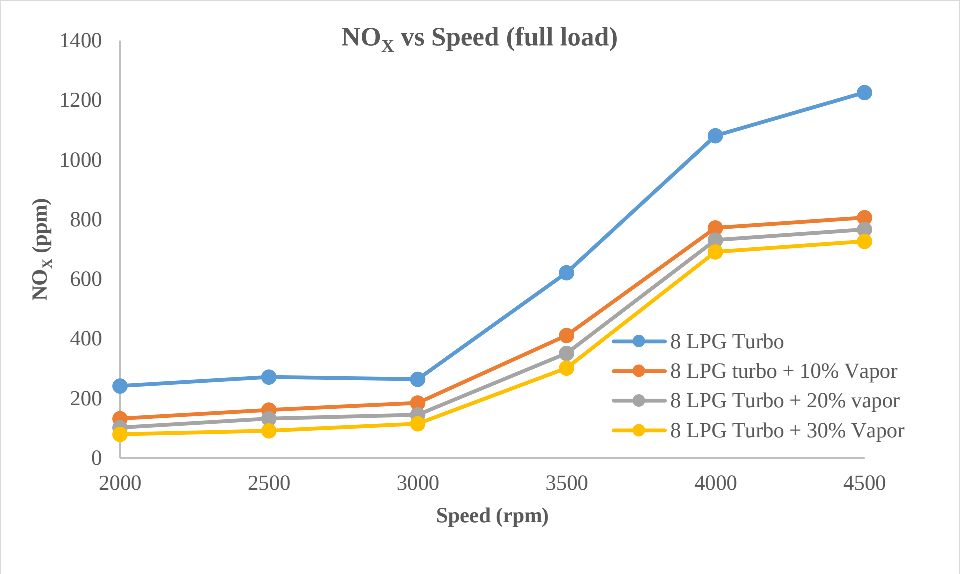


Fig. 5.252 Effect NOx v/s Speed at Full load condition.

5.4.3 Combustion Characteristics

5.4.3.1 P-θ Diagram

Figure 5.253 to 2.258 shows the effect of vaporized water-methanol induction on cylinder pressure. From the results it is revealed that as the percentages of vapor increased the cylinder pressure decreases at higher speed condition. But al lower speed as the percentage of vapor increases there is slight increased cylinder pressure is obtained. Engine working with vapor consumes lower work during compression and produces higher work during expansion period. This is the main reason why the engine running with vapor produces more efficiency and brakes torque according to standard SI engine. During compression, finer steam droplets contribute better air-fuel mixing and cause to decrease the compression temperature and pressure in the cylinder as steam absorbs more heat. Also from the graphs it is seen that as the vapor percentage increases up to 20% the peak pressure occurs near to TDC which indicates better combustion is obtained with vaporized water-methanol induction system. Also presence of methanol along with water will increases the oxygen content in the cylinder which improves the homogeneity of the charge.

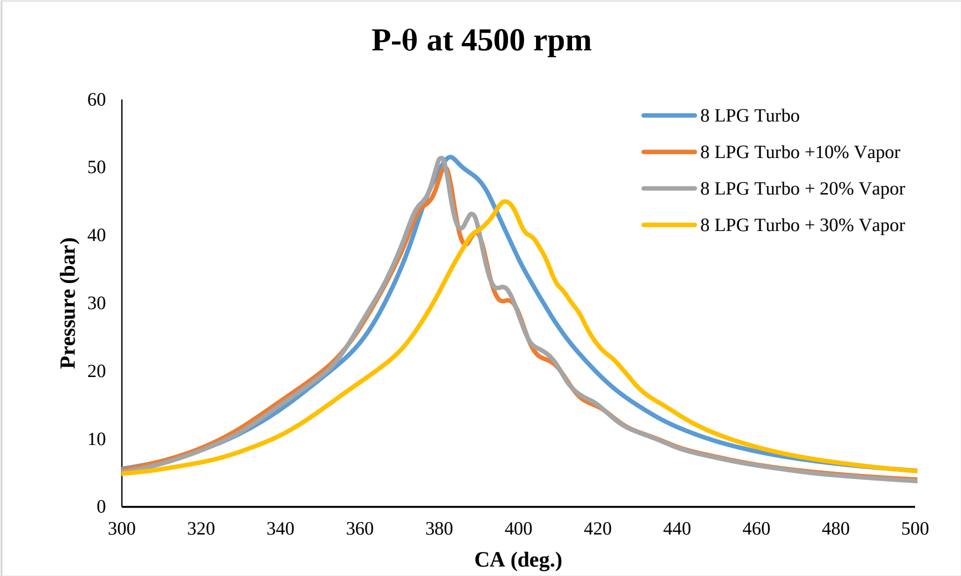


Fig. 5.253 Effect Pressure v/s Crank angle at Full load condition for 4500 rpm.

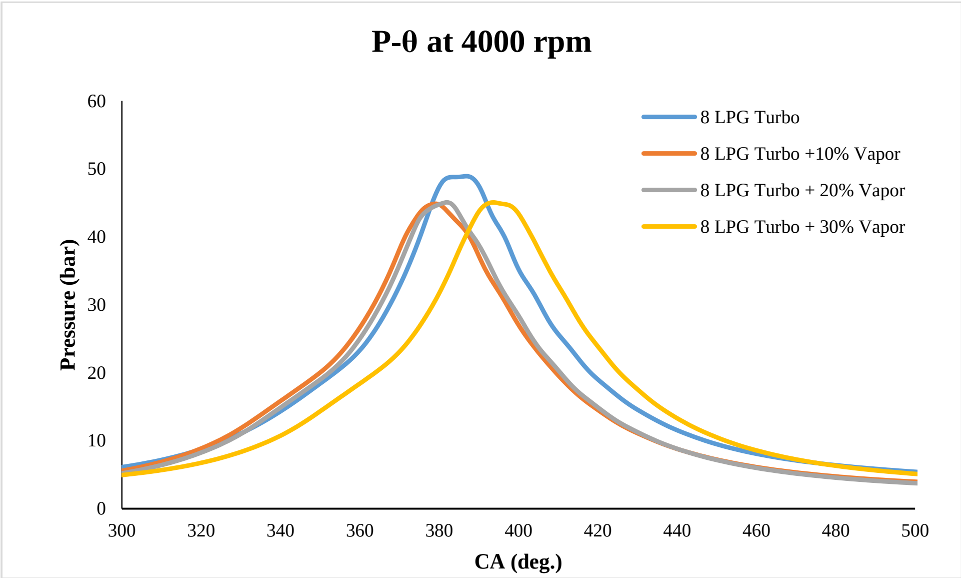


Fig. 5.254 Effect Pressure v/s Crank angle at Full load condition for 4000 rpm.

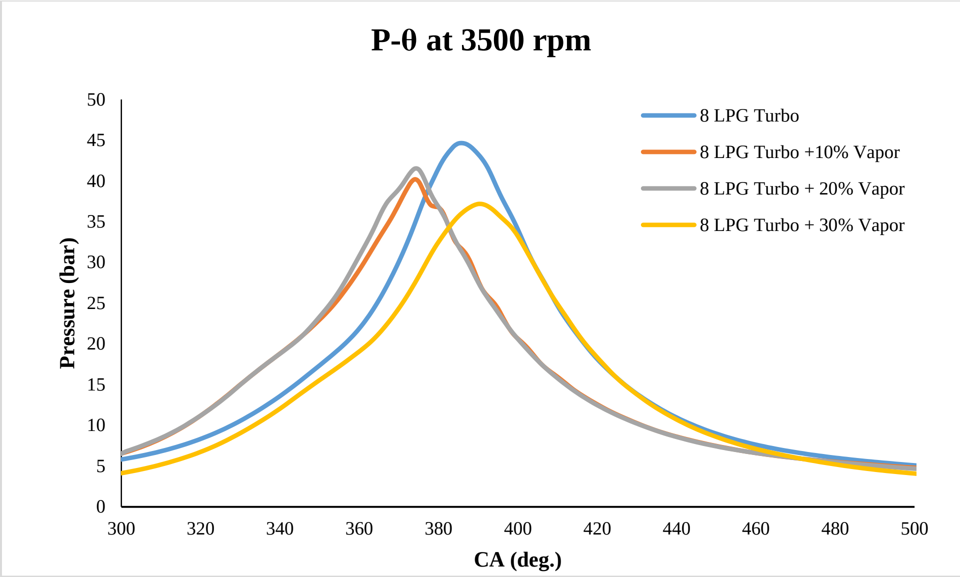


Fig. 5.255 Effect Pressure v/s Crank angle at Full load condition for 3500 rpm.

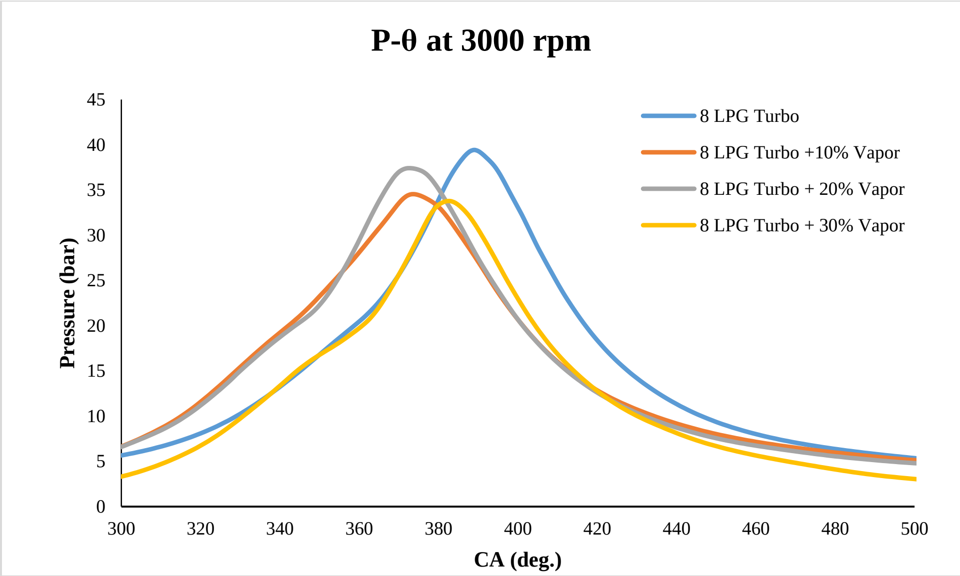


Fig. 5.256 Effect Pressure v/s Crank angle at Full load condition for 3000 rpm.

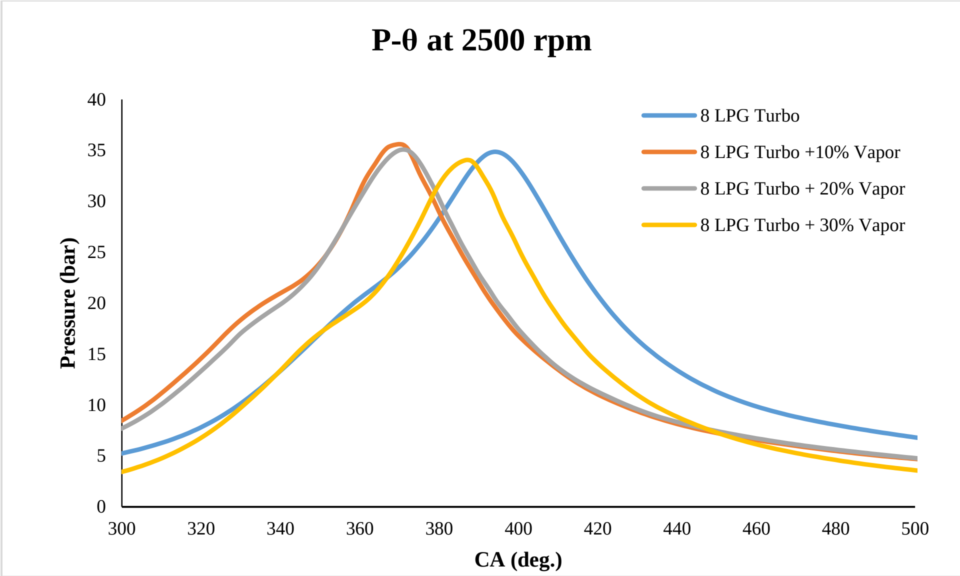


Fig. 5.257 Effect Pressure v/s Crank angle at Full load condition for 2500 rpm.

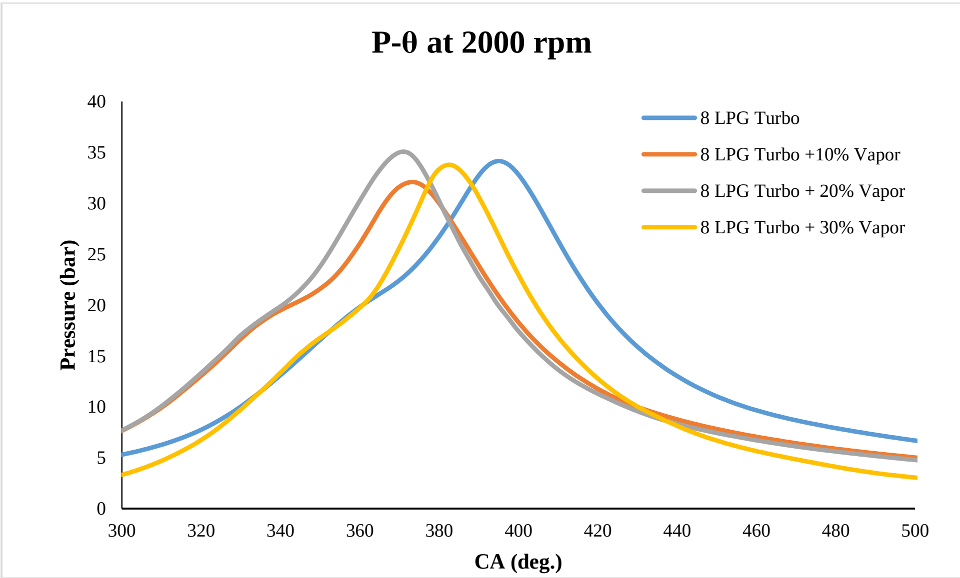


Fig. 5.258 Effect Pressure v/s Crank angle at Full load condition for 2000 rpm.

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CHAPTER 6

CONCLUSION AND SCOPE FOR FUTURE WORK

The present work was focused on the study of performance, combustion and emission characteristics of a four cylinder multipoint port fuel injection gasoline-LPG dual fuel mode of operation along with turbocharger and deionized water-methanol induction. The extensive experimental study conducted on the MPFI 4 cylinder SI engine with different percentage usage of LPG, different speed and load conditions by varying static ignition timings.

The major findings of the experimental based research work can be summarized as follows:

- With the 50% usage of LPG the average increased in BTE is 2% for speed range of 2000 rpm to 4000 rpm at 100% and 50% of full load, and there is no much difference in BTE at 75% load when compared to gasoline.
- Volumetric efficiency is higher for 50% LPG fuel for all speed and load conditions when compared to gasoline and LPG. And also it will work in leaner region in part load condition. Hence improvement in performance can obtain for 50% LPG usage.
- LPG will have much lower CO and HC emissions when compared to gasoline. This is a positive effect on environment. But for other LPG-gasoline ratio these emissions going to increases when compared to LPG but it is well below when compared to gasoline at 100% and 75% of full load for all speed. But at 50% load due to incomplete combustion LPG-gasoline ratio will give higher CO and HC emissions.
- NOx emission is more for LPG almost 4 times that of gasoline for all speed and load conditions. For other LPG-gasoline ratio NOx emission is lower.
- Increase in LPG ratio will give higher peak pressure and higher IMEP and combustion will take place nearer to TDC. Fluctuation in maximum pressure is more for 100% LPG and it is minimum for 50% LPG at higher operating speed. But in Fluctuation in maximum pressure is more for 50%LPG at lower speed. Time return map showed that inconsistent combustion will occur for LPG for higher speed.

- NHRR is maximum for LPG and shifting towards TDC when compared to gasoline for all speed at full load. But for other LPG-gasoline ratio it not much difference when compared to gasoline.
- 50%LPG is given better performance results and NOx emission at 5 deg. bTDC static ignition timing for 2000 to 4000 rpm when compared to gasoline and LPG.
- By advancing the static ignition timing to 8 deg. bTDC resulted in increase in BP obtained for all fuel blends.
- BTE also increased for both gasoline and 100% LPG when advancing static ignition timing because of reduction in the fuel consumption. Also advancing the ignition timing will engine will work leaner side hence reduction in the fuel consumption for 100% LPG.
- Emission like CO drastically reduced as the percentage of 100% LPG increased when static ignition timing is advanced to 8 deg. bTDC
- But advancing the Static ignition timing resulted in increased HC emission for all fuel blends.
- NO_X emission also increases with advancing the static ignition timing for all fuel blends because of increase in the in-cylinder temperature.
- It is found that 8 deg. bTDC with 100% LPG will resulted in better performance and emission characteristics hence these conditions are optimized.
- Using turbocharger performance characteristics are improved. For 100% LPG with turbocharger BTE is increased when compared to gasoline with turbocharger. BTE obtained is maximum at 8 deg. bTDC with turbocharger for 100% LPG when compared to all other condition.
- Maximum of 56% reduction in the NO_X emission is obtained for a turbocharged LPG fueled engine at full load.
- In-cylinder pressure and NHRR also greatly improved with usage of turbocharger. Maximum of 17 bar increase in the in-cylinder pressure is obtained.

- Turbocharged engine resulted in lower cycle by cycle fluctuations when compared to natural aspirated engine. The decreased in COV of IMEP for LPG with turbocharger are 71% and 84% when compared to LPG and gasoline without turbocharger at 4500 rpm. LPG with turbocharged engine show enhanced combustion characteristics with better combustion stability than naturally aspirated engine with speed ranges from 3500 to 4000 rpm.
- Turbocharger will give the better combustion, performance and emission characteristics for LPG fuel.
- From the experimental results it is observed that as the percentage of water-methanol increases, the engine brake thermal efficiency increased for full load conditions. Further increase in the flow rate of water-methanol beyond 30% will reduce the brake thermal efficiency drastically.
- Also results show that water-methanol induction will results in reduction of brake specific energy consumption (BSEC).
- It has been found that NO_X emissions have reduced significantly by 20 - 40% over the entire operating range with the induction of vaporized water-methanol which has worked as a cooling means for the fuel-air charge and slowing the burning rates, resulting in reduction of the peak combustion temperature. At full load condition around 30% and 40% average reduction in NO_X emission are obtained for 20% and 30% water-methanol flow rate.
- HC and CO emissions are going to reduce slightly with water-methanol induction due to presence of more oxygen in the charge to the engine.
- Beyond 20 % vapor ratio, the benefit of NOx reduction is marginal. Hence LPG+20% vaporized water-methanol at 8 deg. bTDC with turbocharger is better choice compared to LPG and gasoline from the point of view of improved engine performance and reduced exhaust emissions.

It can be concluded that use of 50% LPG is superior alternative for unmodified multi-cylinder SI engine for better engine performance and emission characteristics. The use of 100% LPG is best suited for SI engines at 8 deg. bTDC advance static ignition timing with turbocharging and 20%vaporized water-methanol induction rate to get enhanced engine performance and emission characteristics.

6.1 FUTURE WORK

1. Oxygen enrichment for LPG fuelled engine can be used and compared with turbocharged engine.
2. Direct injection of water in to combustion chamber can be devised and the results can be compared.
3. Other NO_X reduction techniques like EGR, SCR can be used and compared with the steam induction technique when the engine is running on LPG.
4. Liquid LPG (LiLPG) injection can be done instead of using vaporizer, and the results can be compared.
5. Combustion studies with more than 500 combustion cycles can be done.
6. With the installation of a programmable ECU, the engine can be run with different equivalence ratios at a given operating condition so that the lean burn limit of LPG can be studied. With the help of programmable ECU, the pulse width of LPG injection can be modified.
7. Engine studies can be conducted in the modified setup using CNG as another alternate gaseous fuel, since the existing gas injection system can be used for injecting CNG also.
8. Computer simulation studies of LPG combustion and NO_X reduction by water-methanol induction can be done.

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APPENDIX I

Specifications of the experimental setup

Engine	Make: Maruti , Model: Zen MPFI, Type: 4 Cylinder, 4S, Petrol (MPFI), water cooled, Power: 44.8 (60 BHP) kW @ 6000 rpm, Torque: 78.5 Nm (8 kgm) @ 4500rpm, stroke: 61mm,bore: 72mm, 993 cc, CR 9.4:1, 4-valves per cylinder, SOHC
Dynamometer	Make: Saj test plant Pvt. Ltd., Model: AG80, Type: eddy current, water cooled, with loading unit
Dynamometer Loading unit	Make: Cuadra, Model AX-153, Type :variable speed, Supply 230V AC.
Propeller shaft	Make: Hindustan Hardy Spicer, Model: 1260, Type: A, with universal joints
Air Box	M S fabricated with orifice meter and manometer
Fuel tank	Capacity 15 lit with glass fuel metering column
Manometer	Make: Apex, Model: MX-104, Range 100-0-100 mm,Type U tube
Fuel measuring unit	Make Apex, Glass, Model:FF0.090
Piezo sensor	Make: PCB Piezotronics, Model: HSM111A22, Range:5000 psi, Diaphragm stainless steel type & hermetic sealed
Calorimeter	Type: Pipe in pipe
Crank angle sensor	Make Kubler-Germany, Model- 8.3700.1321.0360 , Dia: 37mm , Shaft Size: Size 6mmxLength 12.5mm, Supply Voltage 5-30V DC
Engine indicator	Make-Cuadra, Model AX-104, Type Duel channel
Engine interface	Make-Cuadra, Model AX-408, No of channels 8.
Temperature sensor	Type: RTD, PT100 and Thermocouple, Type K
Load sensor	Make: Sensotronics Sanmar Ltd., Model: 60001,Type S beam, Universal, Capacity 0-50 kg, Load cell type: strain gauge,
Fuel flow transmitter	Make: Yokogawa, Model: EJA110-EMS-5A-92NN, Calibration range 0-500 mm H_2O, Output linear DP transmitter,

Rotameter	Make: Eureka, Engine cooling 100-1000 lph; Calorimeter 25-250lph
Pump	Type Monoblock
Add on card	Make: Dynalog, Model - PCI1050, Resolution 12 bit, 8/16 input, Mounting PCI slot
Software	EngineSoft - Engine performance analysis software
Overall dimensions	W 2000 x D 2750 x H 1750 mm

APPENDIX II

Specifications of gas ECU

Model: Sequential gas injection controller of IV generation OSCAR-N OBD CAN of Europe Gas

Sl No	Parameters	Specifications
1	Processor	16bit / 50MHz
2	Voltage Supply	12 volt DC
3	Input Signals	Gas Temperature
		Gas pressure
		Petrol Injection Time
		O_2 - sensor
4	Output Signals	Gas injectors

APPENDIX III

Specifications of the five gas exhaust analyzer

Make: AVL

Measured values	Measurement range	Resolution
CO	0 ... 10 % Vol.	0.01 % Vol.
HC	0 ... 20,000 ppm	10 ppm
CO_2	0 ... 20 % Vol.	0.1 % Vol.
O_2	0 ... 22 % Vol.	0.01 % Vol.
NO	0 ... 5,000 ppm	1 ppm
Lambda	0 ... 9.999 calculated	0.001

APPENDIX IV

<u>Sample Graphs for effect of BMEP on BTE</u>

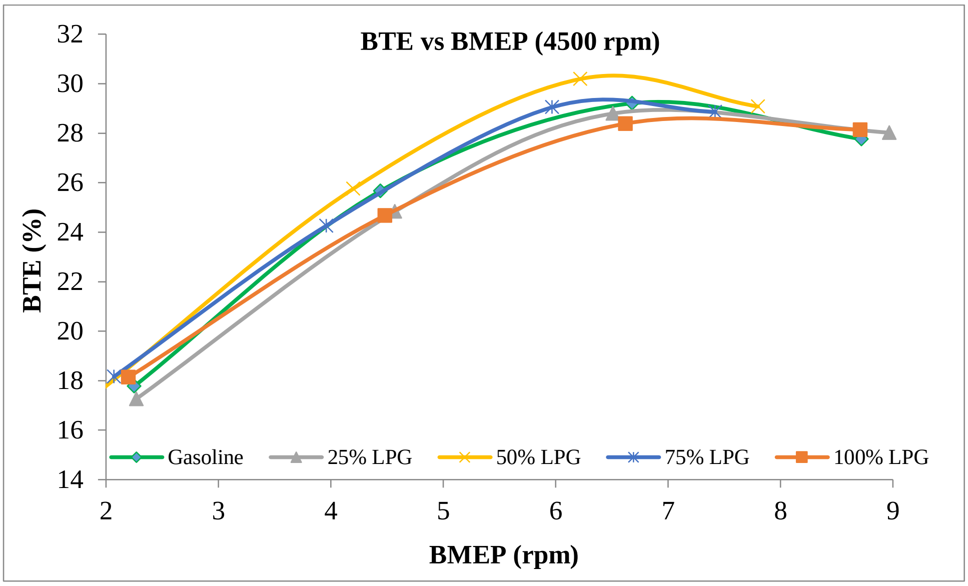


Fig. iv.i Effect of BMEP on BTE at 4500 rpm.

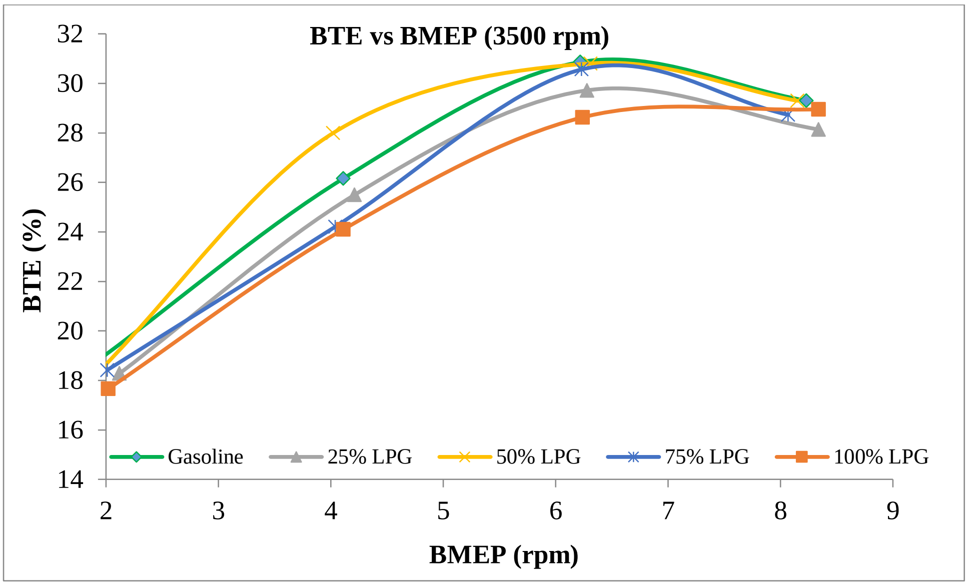
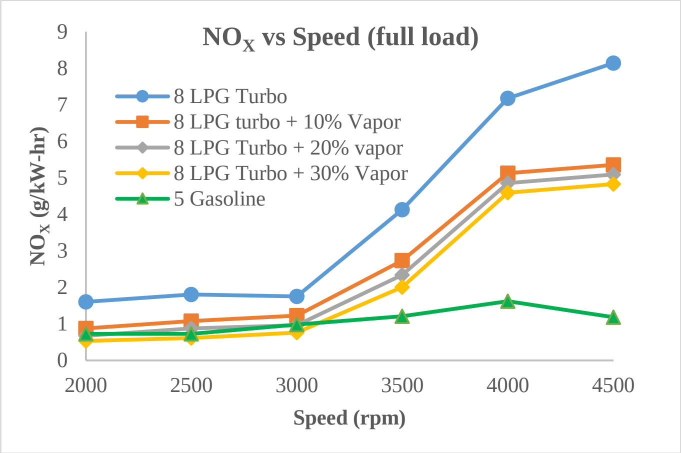


Fig. iv.ii Effect of BMEP on BTE at 3500 rpm.

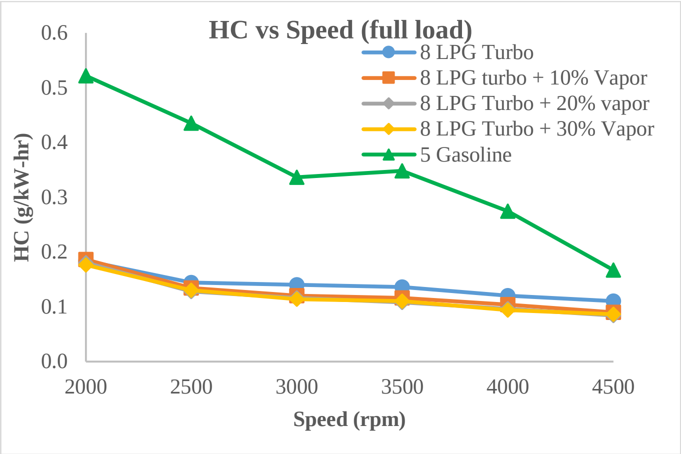
APPENDIX V

Sample Graphs for Emission in mass concentration

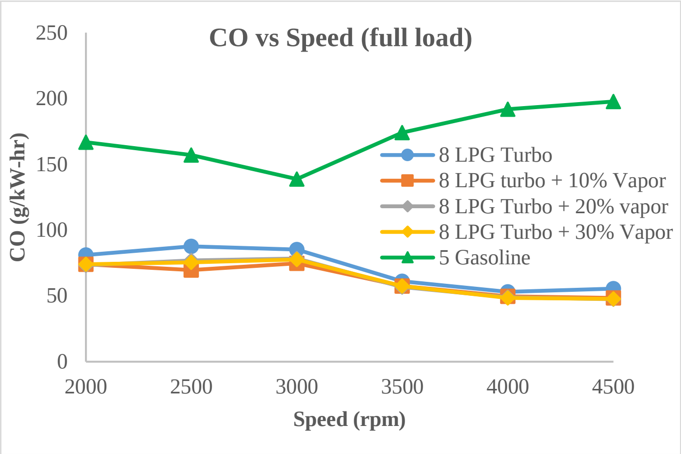
NO_X Emission for vaporized water-methanol induction



HC Emission for vaporized water-methanol induction



CO Emission for vaporized water-methanol induction



LIST OF PUBLICATION BASED ON Ph.D. RESEARCH WORK

International Journal:

1. **Vighnesha Nayak**, Shankar K S, Dinesha P and Mohanan P, " Cycle by cycle variations of LPG-gasoline dual fuel on a multi-cylinder MPFI gasoline engine" *Biofuels,* (2017): DOI: 10.1080/17597269.2017.1302669. (Accepted for publication)
2. **Vighnesha Nayak**, Shankar K S, Dinesha P and Mohanan P, "An experimental investigation on performance and emission parameters of a multi-cylinder SI engine with gasoline-LPG dual fuel mode of operation" *Biofuels,* (2016), Vol. 8, Issue 1, pp 113-123, DOI:10.1080/17597269.2016.1211396.
3. **Vighnesha Nayak**, Rashmi G. S., and Mohanan Padmanabha, "Combustion Characteristics and cyclic variations of a LPG fueled MPFI four cylinder gasoline engine", *Energy Procedia,* (2016), Vol. 90, pp 470-480.
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2. Bhimrao Patil, **Vighnesha Nayak**, Shankar K. S. and Mohanan P. " Performance and Emission characteristics of MPFI multicylinder gasoline engine using LPG with vaporized water methanol induction" *Proceedings of International Conference on Alternative Fuels for IC Engines (ICAFICE-2013),* MNIT Jaipur, 6-8th February 2013, pp 331-336.

National Conference:

1. **Vighnesha Nayak**, Shankar K. S., and Mohanan P., "Cyclic variations in MPFI engine on LPG-Gasoline dual fuel mode of operation", *Proceedings of the 24th National Conference on IC Engines and Combustion* UPES Dehradun, India: 13-16 30th Oct to 1st Nov, 2015: Paper No: UPES/NCICEC/0074, pp 321-326.

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BIO-DATA

Vighnesha Nayak

Department of Mechanical Engineering, MITE, Moodabidre

Home Town: Yekkar

- Post Graduate (2011) M.Tech. in Thermal Engineering from National Institute of Technology Karnataka (NITK), Surathkal
- Graduate in Mechanical Engineering (2009) from Canara Engineering College with First class with Distinction and Gold Medalist Mechanical department.

International Journal:

- Parashuram C., **Vighnesha N.**, Shivaprasad K V, Bedar P., Kumar G.N., "An experimental study on combustion and emission analysis of four cylinder 4-stroke gasoline engine using pure hydrogen and LPG at idle condition", *Energy Procedia,* (2016), Vol. 90, pp 525-534.
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- Dinesha P., **Vighnesha Nayak,** P. Mohanan, "Impact of injection timing on the performance and emission characteristics of a diesel engine fueled with Cardanol blends", *Proceedings of the 10th Asia-Pacific Conference (ASPACC-2015) on Combustion,* July 19-22, 2015 Beijing, China.