# EXHAUST GAS ANALYSIS AND PARAMETRIC STUDY OF ETHANOL BLENDED GASOLINE FUEL IN SPARK IGNITION ENGINE

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**IN**

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Under the guidance of

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# ABSTRACT

Today, the reserves of petroleum based fuels are being rapidly depleted. It is well known that the future availability of energy resources, as well as the need for reducing $CO_2$ emissions from the fuels used has increased the need for the utilization of regenerative fuels .Various substitutes are available to use Gasoline in SI engines with the possibility of reducing harmful exhaust gas emissions. Ethanol is a renewable fuel. It can be produced from agricultural feed stocks, such as sugarcane and also from forestry wood wastes and agricultural residues. This research is done taking commercial gasoline as reference which is originally blended with 5% ethanol. Hence 5%, 10%, 15%, 20% ethanol blends with Gasoline including the blended ethanol initially was tested in SI engines. Physical properties relevant to the fuel were determined for the four blends of gasoline. A four cylinder, four stroke, varying rpm, Petrol (MPFI) engine connected to eddy current type dynamometer was run on blends containing 5%,10%,15%,20% blends and performance characteristics, maximum pressure induced and exhaust emissions were evaluated. In terms of octane number and density, ethanol has higher value while the calorific value of gasoline is higher than ethanol. Even though higher blends can replace gasoline in a SI engine, results showed that there is a reduction in exhaust gases and increase in Mechanical efficiency, Specific Fuel Consumption and Indicated Thermal Efficiency on blending. Maximum pressure reached also increases with blending. Hence we can conclude from the result that using 10% ethanol blend is most effective and we can utilize it for further use in SI engines with little constraint on material used to sustain little increase in pressure.

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# CERTIFICATE

**DELHI TECHNOLOGICAL UNIVERSITY**

(Formerly DELHI COLLEGE OF ENGINEERING)

Date:-__________

This is to certify that report entitled **"Exhaust Gas Analysis and Parametric Study of Ethanol Blended Gasoline Fuel in Spark Ignition Engine"** by **Mr. Prakash Mahara** is the requirement of the partial fulfillment for the award of Degree of **Master of Technology (M. Tech.)** in **Thermal Engineering** at **Delhi Technological University**. This work was completed under my supervision and guidance. He has completed his work with utmost sincerity and diligence. The work embodied in this project has not been submitted for the award of any other degree to the best of my knowledge.

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ABBREVATIONS

| | |
|---|---|
| AC | Alternating Current |
| $(A/F)_s$ | Stoichiometric air/fuel Ratio |
| Ar | Argon |
| BDC | Bottom Dead Centre |
| BMEP | Brake Mean Effective Pressure |
| cc | Cubic Centimeter |
| CC | Cubic Cylinder |
| CI | Compression Ignition |
| CO | Carbon mono oxide |
| $CO_2$ | Carbon di oxide |
| CR | Compression Ratio |
| Den | Density |
| DDGS | Distillers Dried Grains |
| DC | Direct Current |
| DNS | Denatured Spirit |
| E0 | Ethanol 0% Gasoline 100% |
| E10 | Ethanol 10% Gasoline 90% |
| E25 | Ethanol 25% Gasoline 75% |
| E50 | Ethanol 50% Gasoline 50% |
| E75 | Ethanol75% Gasolione 25% |
| E85 | Ethanol 85% Gasoline 15% |

| | |
|---|---|
| E100 | Ethanol100% Gasoline 0% |
| EFI | Electronic Fuel Injection |
| FFVs | Flex Fuel Vehicles |
| fps | Frame per second |
| GC | Gas chromatography |
| GDI | Gasoline Direct Injection |
| h | Differential head across orifice (m of water) |
| $H_2O$ | Water |
| IEA | International Energy Agency |
| ISAF | International Symposium on Alcohol Fuels |
| K | Kelvin |
| kPa | Kilo Pascal |
| L/s | Litre per second |
| LPG | Liquefied Petroleum Gas |
| mA | milli Ampere |
| MEP | Mean effective Pressure |
| ml | Millilitres |
| mm | Millimetres |
| MON | Motor Octane Number |
| MPa | Mega Pascal |
| M.P.F.I. | Multi Point Fuel Injection system |
| Ms | Millisecond |
| MTBE | Methyl Tertiary Butyl Ether |
| mV | milli Volt |

| | |
|---|---|
| $NO_x$ | Oxides of Nitrogen |
| NREL | National renewable Energy Lab |
| $O_2$ | Oxygen |
| pcm | Power Train Control Module |
| Pm | Mean Effective pressure |
| Psi | Pressure per square inch |
| RON | Research Octane Number |
| RVP | Reid vapour Pressure |
| s | Second |
| SFC | Specific fuel consumption |
| SI | Spark-Ignition |
| TDC | Top Dead Centre |
| UHC | Unburned Hydro Carbon |
| ULEVs | Ultra Low Emission Vehicle |
| Vol | Volume |
| Wt | Weight |

## SYMBOLS

| | |
|---|---|
| ° | Degree |
| % | Percentage |
| r | Compression ratio |
| K | Poly tropic Index |
| $\eta_t$ | Thermal Efficiency |
| $\eta_v$ | Volumetric Efficiency |
| n | No of strokes |
| p | Pressure |

**CHAPTER 1**

# INTRODUCTION

## 1.1 Introduction

Rising fuel prices and increased oil consumption along with the lack of sustainability of oil-based fuels have generated an interest in alternative, renewable sources of fuel for internal combustion engines, namely alcohol-based fuels. Currently ethanol is the most widely used renewable fuel with up to 10% by volume blended in to gasoline for regular engines or up to 85% for use in Flex-Fuel vehicles designed to run with higher concentrations of ethanol. Ethanol can also be used as a neat fuel in spark-ignition (SI) engines or blended up to 40% with Diesel fuel for use in compression-ignition (CI) engines [24-25]. Ethanol was introduced as a replacement for methyl tertiary butyl ether (MTBE) when it was realized that MTBE leaked onto the ground at filling stations resulting in the contamination of large quantities of groundwater. Ethanol is biodegradable, less detrimental to ground water, and has an octane number much higher than gasoline as well as having a positive effect on vehicle emissions [26]

There are lots of gases in the environment which are causing pollution and greenhouse effect and the major contributor is the transport sector due to the heavy, and increasing, traffic levels. In spite of ongoing activity to promote efficiency, the sector is still generating significant increases in $CO_2$ emissions. As transport levels are expected to rise, especially in developing countries, fairly drastic political decisions may have to be taken to eradicate this problem in the future. Furthermore, the dwindling supply of petroleum.

Today, the transport sector is a major contributor to net emissions of greenhouse gases, of which carbon dioxide is particularly important. The carbon dioxide emissions originate mainly from the use of fossil fuels,mostly gasoline and diesel oil in road transportation systems, although some originates from other types of fossil fuels such as natural gas and Liquefied Petroleum Gas (LPG). If international and national goals (such as those set out in the Kyoto protocol) for reducing net emissions of carbon dioxide are to be met, the use of fossil fuels in the transport sector has to be substantially reduced. This can be done, to some extent, by increasing the energy efficiency of engines and vehicles and thus reducing fuel consumption on a volume per unit distance travelled basis. However, since the total transportation work load is steadily increasing such measures will not be sufficient if we really want to reduce the emissions of carbon dioxide.

### 1.2 Alcohol Blended Fuels

The idea of adding low contents of ethanol or methanol to gasoline is not new, extending back atleast to the 1970s, when oil supplies were reduced and a search for alternative energy carriersbegan in order to replace gasoline and diesel fuel. Initially, methanol was considered the mostattractive alcohol to be added to gasoline. Since methanol can be produced from natural gas at nogreat cost, and is quite easy to blend with gasoline, this alcohol was seen as an attractiveadditive. However, when using methanol in practice it became clear that precautions had to betaken when handling it and that methanol is aggressive to some materials, such as plasticcomponents and even metals in the fuel system. A lesson learned was that new, more resistantmaterials had to be used in the fuel system of the vehicles as well as in the distribution system.These experiences were also of great value when ethanol came to be more commonly used as analternative to the commercial fuels, since even ethanol can be characterized as an aggressivefluid, albeit somewhat less so than methanol. The interest in producing an alternative fuel basedon biomass has also been a major factor in the early choice between methanol and ethanol.The use of E85, a mixture of 85 % ethanol and 15 % gasoline, for Flexible Fuel Vehicles has become common.

Blends with other percentages of ethanol in gasoline are commonly used in various countriesaround the world, especially Australia (officially 10 %), Brazil (up to 25 %), Canada (10 %),Sweden (5 %) and the USA (up to 10 %). There is still debate about whether, how and to whatextent ethanol in gasoline may affect the materials in the vehicle and cause excessive wear ofparts in the fuel system and the engine. However, in the USA, car manufacturers have agreedthat use of gasoline with up to 10 % ethanol will not affect the warranties of their vehicles [1].Since both methanol and ethanol have considerably lower energy contents (15.7 MJ/l and 22.4MJ/l, respectivel) compared with gasoline (approximately 44 MJ/l) use of an alcoholcontainingblend may affect the power output of the engine to varying degrees, depending on itsdesign. According to the calculations, adding ethanol to a finalvolume of 10 % to a gasoline with an energy content of 42.2 MJ/litre will decrease energy content value by4.1 %.

### 1.3 Ethanol as a Blend

In the medium term ethanol produced from grain will probably be the most important alternative fuel for replacing gasoline, and in the long term ethanol producedfrom cellulose might take over from grain ethanol.Today, ethanol accounts for a substantial part of the alternative fuel market. The advantages of ethanol are that it can:

- Provide a viable alternative to reduce the greenhouse effect.
- Be produced domestically, thereby reducing dependence on imported petroleum.
- Be easily mixed with gasoline.
- Be used (and already is on a wide scale) as oxygenate in gasoline.
- Create new jobs in the country related to its production.

From an international perspective, most research up to 1990 was focused on blends of methanoland gasoline, but some studies were carried out on ethanol-gasoline blends. Since these studieswere carried out in the USA, it can be assumed that they mainly included vehicles with efficientemission control systems, but at the same time technical features of cars in the USA have historically differed, at least in part, from those in Sweden. It should also be noted that for a longtime 10% ethanol has been added to commercial gasoline in many parts of the world. In the USAthere is considerable experience of adding higher proportions of ethanol to gasoline than thoseallowed by gasoline regulations in Sweden (Europe). The primary advantage of adding a bio based alcohol to gasoline is that it reduces net $CO_2$ emissions but it also has other positiveeffects, such as increasing the octane value of the fuel and reducing the benzene content of theexhaust gases. The use of alcohol blended gasoline and neat fuel alcohols as substitutes for neat gasoline have become matters of interest in many countries. The International Energy Agency (IEA), established in 1974, follows the development, and data and other experience from various trials have been presented and discussed at symposia organized by the International Symposium on Alcohol Fuels (ISAF).

### 1.4 Advantage of blending ethanol with Gasoline

- There is intense interest world-wide in using ethanol as an automotive fuel, especially inblending ethanol with gasoline. Blending ethanol in a commonly used fossil fuel isgenerally seen as an easy way to introduce an alternative such as bio-ethanol without costlychanges of the fleet of vehicles on the road.
- Ethanol can easily be blended in gasoline by well-known methods. Ethanol has a lowerheating value than gasoline, which will reduce the energy content of the fuel. However thiscan be partly offset by the higher octane value of ethanol.
- The main conclusion from using ethanol-gasoline blends in practice is that blends with upto 15 percent ethanol will not have any significant negative effects on the wear of theengine or vehicle performance.
- No significant difference can be seen in regulated emissions when comparing the use ofblended fuel (with up to 10-15% ethanol) to the use of neat gasoline. Concerningunregulated emissions views differ.
- There will be a slight increase (~2-3%) in fuel consumption when shifting from neatgasoline to a 10 percent ethanol-gasoline blend, depending on the design of the vehicle.Cold starts, in particular, will affect fuel consumption more when using blended gasolinethan when using neat gasoline.

### 1.5 Co-products of ethanol [23]

The co-products that results when making ethanol are dependent on the medium used to produce the ethanol. Table 1 shows a summary of the co-products and what they are used for.

| BY-PRODUCTS/CO-PRODUCTS | USED FOR |
|---|---|
| Flour, Corn Oil, Corn Meal, Corn Grits | Used in producing food for human consumption |
| Fibrotein TM | Used as a high fibre and protein food additive |
| Corn Gluten Meal and Corn Gluten Feed | Used as high protein animal feed additives |
| Amino Acids | Used as animal feed additives |
| Dry Distiller's Grains | Used as high protein and energy animal feed |
| Carbon Dioxide | Used as a refrigerant, in carbonated beverages,to help vegetable crops grow more rapidly ingreenhouses, and to flush oil wells |

**Table 1.1: Summary of by-products/co-products made through ethanol production.**

In practice, about two-thirds of each tonne of grain (i.e., the starch) is converted to ethanol. The remaining by-product is a high protein livestock feed which is particularly well suited for ruminant animals such as cattle and sheep. This by product is also known as Distillers' Dried Grains, DDGS. The protein in this material is utilized more efficiently in ruminant nutrition than are other high-protein feed ingredients such as soybean meal. This by-product of ethanol production is particularly good for Canadian dairy, beef and sheep production. It improves the competitive position globally of producers of these farm commodities. The manure from livestock can be used as a major source of fertilizer in grain crop production [21].

Carbon dioxide is another by-product produced when making ethanol. Carbon dioxide, given off in great quantities during fermentation will be collected and cleaned of any residual alcohol, compressed and sold as an industrial commodity [22].

**1.6 Objectives and Structure of the Thesis**

This present study focuses on the examination of ethanol blended fuels and the exhaust emissions by them for use in internal combustion engines by focusing on the following areas that have yet to be explored:

A comparison of the UHC, CO, and $CO_2$ emissions of ethanol-gasoline blends (E5, E10, E15, E20), neat and to determine the effect that blended gasoline has on Mechanical efficiency, specific fuel consumption,Indicated Thermal Efficiency.Variation of pressure with crank angle is also determined.

Previous studies using a similar setup and an engine fueled with ethanol showed a modest decrease in unburned hydrocarbon emissions and a corresponding increase in nitrogen oxides emissions.

Chapter 4 describes in detail the test engine setup and specifications, the emissions analyzers and other measurement equipment, the test fuels, and the experimental procedures and operating conditions used in this study. Chapter 5 focuses on the emissions of the blended fuels discussed previously.

The emissions of ethanol-gasoline blends were compared on the basis of the engine load and rpm. Chapter 5 presents the results from the engine and the impact on UHC, CO, and $CO_2$ emissions and variation of specific fuel consumption and Mechanical efficiency. Chapter 6 provides concluding remarks as well as recommendations for future research and improving the performance of the engine used in this study.

### 1.7 Expected Outcome

An important step in efforts to solve the problem of crude oil is to replace fossil source energy with bioenergy. In the transport sector this means either introducing bio fuels and using adapted vehicles, or blending bio fuels with petroleum-based fuels for use with present vehicle fleets. The two alternatives are not, of course, mutually exclusive. However, blending bio fuels with petroleum-based fuels for use by the present conventional vehicle fleets has the advantages that even using quite low blending concentrations will result in substantial total volumes of gasoline being substituted by bio fuels, and that the present infrastructure for distributing fuels can be used.

In order to reduce absolute amounts of these emissions we have to go further and an additional measure that will be required is to replace fossil vehicle fuels with renewable ones. Primarily, especially in the short term, this means bio-based fuels. Probably the best candidate bio fuels to replace gasoline in the short term are alcohols. Alcohols can be blended with gasoline or used as neat fuel in both optimized spark ignition engines and compression ignition engines.

## CHAPTER 2

# LITERATURE REVIEW

### 2.1 Background

The issue of whether higher ethanol blends can successfully be used in conventional vehicles is a key to their expanded deployment. However, there are divergent expert opinions on whether E10–E25 could detrimentally affect fuel control system operation and materials in a conventional vehicle. This section presents the current views of experts and their findings. Attaining a definitive answer will require further study.

Little specific testing has been completed on higher ethanol blends. Rather, the testing has focused either on E85 (intermediary points were not assessed in flex-fueled vehicle [FFV] testing) Rhoad [4] or on 10% ethanol blends. Under contract to National renewable Energy Lab (NREL), the Nexum Research Corporation of Ontario, Canada, conducted a fuel efficiency and emissions analysis on E20, E40, and E85 blends. Blends of 15% were also considered in the late 1970s. Although Brazil has had an extensive blended fuel program, vehicle systems in Brazil were initially optimized for neat fuels. Due to this prior optimization, impacts on existing (e.g., conventional) fuel systems were not extensively observed when blends ranging from E22 -E24 became widely used.

Therefore, without additional testing, it is difficult to say with absolute certainty what the effects of higher ethanol blends will be on emissions, drivability, and material compatibility. Some air pollution and automotive specialists choose to err on the side of conservatism and assume that effects occurring with E85 fuels could also occur to some degree with other ethanol blends. However, other experts postulate that higher ethanol blend effects could be limited [25].

### 2.2 Technical Issues [27]

#### 2.2.1 Fuel Control System

The fuel control system is critical to the normal operation and emission control of the vehicle. It is also sensitive to changes in fuel composition. For higher blends of ethanol to work well in conventional vehicles, the fuel control system must be able to compensate for differences between ethanol blends and gasoline (due predominantly to ethanol's higher oxygen content).

#### 2.2.2 Air/Fuel Ratio

Due to greater availability of range authority and to advanced emission control technology it is possible that the fuel control systems on Technology Class 5 vehicles could compensate for higher ethanol blends by re-calibrating the air/fuel ratio settings accordingly. These vehicles are equipped with superior "block learning capabilities" with the ability to adapt the base fuel control settings to accommodate the higher oxygen levels. A key point is that the adaptation process must occur in the closed loop phase of the vehicle operation, i.e., hot operation. Adaptive learning does not take place in the open loop, or transient, start-up phase. It is also important to note that systems may vary among vehicle manufacturers. Although each is striving toward a common goal, algorithms in fuel control systems are considered to be highly proprietary, and consequently, may produce different results [25].

### 2.2.3 Oxygen Sensor

An integral component of the fuel control system is the oxygen sensor, which is located in the exhaust stream. The higher the oxygen content in the exhaust, the greater the voltage transmitted from the oxygen sensor to the computer.

Despite the differences in oxygen levels, current practice in the automotive industry is to use the same oxygen sensor in both dedicated gasoline vehicles and E85 flex fuel vehicles (FFVs); with negligible impacts on system operation [3,25]. Further, the results of tests conducted for NREL (by Nexum Research Corporation) on E20 blends utilizing a standard oxygen sensor, indicate that standard oxygen sensors operate effectively in an E17-E24 environment. Stoichiometric conditions were achieved despite the additional oxygen present.

However, an opposing view suggests that the oxygen sensor and its interaction with the power train control module (PCM) has a limited authority range of about 4% oxygen, and can therefore not properly calibrate for a 17% ethanol blend which approaches 6% oxygen. In certain vehicles, the oxygen sensor could have a limited ability to transmit voltage, and could be unable to transmit voltage levels commensurate with the level of oxygen present in the fuel.

$NO_x$ emissions may be elevated due to the PCM's inability to compensate for higher oxygen levels. The argument further maintains that Ultra Low Emission Vehicle ULEVs and other future technology vehicles will require an air/fuel trim within a very tight range to achieve emissions compliance. Experts in the field maintain that automakers prefer oxygen levels no higher than 2 wt. % so that a tighter range can be maintained [2]. However, $NO_x$ levels may also be reduced, independently of the fuel control system equipment, due to the reduced combustion temperatures of ethanol blends. In addition, if increases in $NO_x$ are noted, the oxygen sensor can be used to compensate for the increased oxygen level [3].

2.2.4 Fuel Quality

The effect of wider oxygen parameters on a tightly controlled air/fuel ratio relates to the broader issue of fuel quality. Automakers are focusing considerable attention on narrowing fuel specifications wherever possible. Proposed petitions would confirm the T-50 specification at 170° (50% of the fuel evaporates at 170°), and eliminate the one psi Reid vapour Pressure RVP variance for ethanol blends. Because RVP measures how easily a liquid evaporates, a higher RVP means that more of the fuel can evaporate, contributing to the formation of ground-level ozone. To limit the possibility of such emissions, EPA has set progressively tighter limits on RVP in fuels.

Because it is difficult for ethanol blends to meet the 170° T-50 specification, ethanol blenders have traditionally relied on the RVP variance to compete in oxygenated fuel markets. However, the variance may not be necessary for higher ethanol blends since RVP decreases are seen as the volume of ethanol increases. Areas of study should include assessing base fuel qualities that would optimize higher ethanol blend use.

### 2.2.5 Hot Operation

Automakers have voiced concerns about higher ethanol blends impacting hot driveability. The volatility of 5.7%-10% ethanol blends is approximately 1 psi higher than unleaded gasoline unless a low-RVP base fuel is used. Vapor lock, and difficult hot start can potentially occur more often in a fuel with a higher volatility, particularly in summertime conditions. However, in current practice, manufacturers have already engineered solutions to this problem since E10 is so widely used in today's gasoline market [3].

### 2.2.6 Materials Compatibility

Alcohol fuels have different physical and chemical properties than gasoline, which affects their compatibility with fuel system components. However, using corrosion-inhibiting additives, as well as the chemical composition and physical properties of the base fuel, affect the degree of materials incompatibility.

Discussion published in a report by Oak Ridge National Laboratory (1988) indicates compatibility in fuel system parts with 15% ethanol blends. Testing conducted at the Technical Research Center of Finland found that vehicles fueled on E15 ran satisfactorily on stock carburetor settings without modification.In addition, eight out of ten test cars that were fueled on E15 showed less or equal wear compared to the same vehicles operated on gasoline.

Dunn and Pfistererfound that permeability issues could be remedied by applying a thin nylon veneer to the outside of nitrile rubber compounds. (Polymer permeability is a concern of automakers, viewed as a major factor in evaporative emissions).

It is likely that many of these research results have been incorporated in modern vehicle systems. In addition, years of materials testing on alternative and blended fuels have been applied to conventional systems as well. It is estimated that the sophisticated fuel system materials of today's vehicles could likely resist any additional corrosivity associated with higher ethanol blends [25].

### 2.3 Literature Survey

- N. Seshaiah tested the variable compression ratio spark ignition engine designed to run on gasoline with pure gasoline, LPG (Isobutene), and gasoline blended with ethanol 10%, 15%, 25% and 35% by volume. Also, the gasoline mixed with kerosene at 15%, 25% and 35% by volume without any engine modifications has been tested and presented the result. Brake thermal and volumetric efficiency variation with brake load is compared. CO and $CO_2$ emissions have been also compared for all tested fuels.It is observed that the LPG is a promising fuel at all loads lesser carbon monoxide emission compared with other fuels tested. Using ethanol as a fuel additive to the mineral gasoline, (up to 30% by volume) without any engine modification and without any loses of efficiency,it has been observed that the petrol mixed with ethanol at 10% by volume is better at all loads and compression ratios.

- Rodrigo C.Costa, José R. Sodre[5] compares the performance and emissions from a production 1.0-l, eight-valve, and four-strokeengine fuelled by hydrous ethanol (6.8% water content in ethanol) or 78% gasoline-22% ethanol blend. Theengine was tested in a dynamometer bench in compliance with NBR/ISO 1585 standard. The performanceparameters investigated were torque, brake mean effective pressure (BMEP), brake power, specific fuelconsumption (SFC), and thermal efficiency. Carbon monoxide (CO), carbon dioxide ($CO_2$), hydrocarbons(HC) and oxides of nitrogen ($NO_x$) exhaust emissions levels were also presented. The results showed thattorque and BMEP were higher when the gasoline-ethanol blend was used as fuel on low engine speeds.On the other hand, for high engine speeds, higher torque and BMEP were achieved when hydrous ethanolfuel was used. The use of hydrous ethanol caused higher power at high engine speeds, whereas, for lowengine speeds, both fuels produced about the same power. Hydrous ethanol produced higher thermalefficiency and higher SFC than the gasoline-ethanol blend throughout all the engine speed range studied.With regard to exhaust emissions hydrous ethanol reduced CO and HC, but increased $CO_2$ and $NO_x$ levels.

- HakanBayraktar[6]studied the effects of ethanol addition to gasoline on an SI engine performance and exhaust emissions areinvestigated experimentally and theoretically. In the theoretical study, a quasi-dimensional SI enginecycle model, which was firstly developed for gasoline-fueled SI engines was adaptedfor SI engines running on gasoline–ethanol blends. Experimental applications were carried outwith the blends containing 1.5, 3, 4.5, 6, 7.5, 9, 10.5 and 12 vol% ethanol. Numerical applicationswere performed up to 21 vol% ethanol. Engine was operated with each blend at 1500 rpm forcompression ratios of 7.75 and 8.25 and at full throttle setting. Experimental results showed that among thevarious blends, the blend of 7.5% ethanol was the most suitable one from the engine performanceand CO emissions points of view. However, theoretical comparisons showed that the blendcontaining 16.5% ethanol was the most suited blend for SI engines. Furthermore, it wasdemonstrated that the proposed SI engine cycle model has an ability of computing SI engine cycleswhen using ethanol and ethanol–gasoline blends and it can be used for further extensive parametricstudies.

- FikretYuksel, BedriYuksel [7] One of the major problems for the successful application of gasoline–alcohol mixtures as a motor fuel is the realization of a stable homogeneous liquid phase. To overcome this problem, a new carburetor was designed. With the use of this new carburetor, not only the phase problem was solved but also the alcohol ratio in the total fuel was increased.By using ethanol–gasoline blend, the availability analysis of a spark-ignition engine wasexperimentally investigated. Sixty percent ethanol and 40% gasoline blend was exploited totest the performance, the fuel consumption, and the exhaust emissions.As a result of this study, it was seen that a new dual fuel system could be serviceable bymaking simple modifications on the carburetor and these modifications would not causecomplications in the carburetor system.

- Lan-bin Wen, Chen-Ying Xin, Shyue-Cheng Yang[8]investigatedthe effect of oxygen containing additives on gasoline blended fuels on exhaust emissions for different engine speeds in a single cylinder, four-stroke, spark-ignition engine. The results indicated that CO and HC exhaust emissions were lower with the use of ethanol–gasolineand Di Methyl Carbonate (DMC)–gasoline blended fuels as compared to the use of unleaded gasoline. On the other hand, theeffect of ethanol–gasoline and DMC–gasoline blended fuels on $NO_x$ exhaust emission is insignificant.Using oxygen containing additives can increase fuel consumption as a result of the heating value ofthe blended fuels being lower than that of unleaded gasoline.

- M.A. Ceviz, F. Yuksel[9] investigated the effects of using ethanol–unleaded gasoline blends on cyclic variability and emissions in a spark-ignited engine.Results of this study 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 10vol.% ethanol in fuel blend. On the other hand, after this level of blend a reverse effect was observed on the parameters aforementioned. The 10vol.% ethanol in fuel blend gave the best results.

- Ted R. Aulich, Xinming He, Ames A. Grisanti, and Curtis L. Knudson [10] performed test to compare the evaporation rate of 10 volume percent (vol%)

ethanol-blended gasoline (E10) with the evaporation rate of its base gasoline. Weight loss, temperature, pressure, and humidity were monitored as lab-blended E10 and base gasolines were evaporated concurrently from glass cylinders placed on balances located side by side under an exhaust hood.The averaged results of four tests at about 70°F showed that the E10 lost more total weight to evaporation than the base fuel, but less gasoline. The increased weight was due to ethanol, which was present in the E10 evaporative emissions at concentrations of about 13 weight percent (wt %). In two-hour tests at temperatures near 70°F, during which 4.5 to 5.3 wt% of initial fuel samples were evaporated, E10 fuels lost an average of about 5% less gasoline than their base fuels. A similar result was obtained for a one-hour test, during which about 2.4 to 2.5 wt% of the initial fuel samples were evaporated. Gas chromatography (GC) component analysis indicated that the compositions of the ethanol-free emissions from the two fuels were similar.

- JuozasGrabys[11]investigated experimentally and compare the engine performance and pollutant emission of a SI engine using ethanol–gasoline blended fuel and pure gasoline.The results showed that when ethanol was added, the heating value of the blended fuel decreases, while the octane number of the blended fuel increases. The results of the engine test indicated that when ethanol–gasoline blended fuel was used, the engine power and specific fuel consumption of the engine slightly increase; CO emission decreases dramatically as a result of the leaning effect caused by the ethanol addition; HC emission decreases in some engine working conditions; and $CO_2$ emission increases because of the improved combustion.

- C. AnandaSrinivasan and C.G. Saravanan [12]studied Combustion Characteristics of an SI Engine Fuelled with Ethanol and Oxygenated Fuel Additives. They investigated the effects of ethanol-blended gasoline with oxygenated additives on a multi – cylinder Spark Ignition (SI) Engine. The experiments were conducted in two stages. In stage I, the test fuels were prepared using 99.9% pure ethanol and gasoline with a cycloheptanol blend, in the ratio of E69.5 + 0.5 cycloheptanol, E64.6 + 0.4 cycloheptanol, E59.7 + 0.3 cycloheptanol, E49.8 + 0.2 cycloheptanol. The remainder was gasoline. In stage II, the test fuels were prepared using 99.9% pure ethanol and gasoline with cyclooctanol blend, in the ratio of E69.5 + 0.5 cyclooctanol, E64.6 + 0.4 cyclooctanol l, E59.7 + 0.3 cyclooctanol, E49.8 + 0.2 cyclooctanol. The remainder was gasoline. Performance and emission tests were conducted on a multi – cylinder SI Engine coupled with an eddy current dynamometer. The emission tests were measured using an exhaust gas analyzer. The experimental results proved that the blend increased brake thermal efficiency more than a sole fuel, such as gasoline. The emission tests found that the CO slightly decreased, while HC and $O_2$ increased moderately and $CO_2$ and NOx appreciably decreased. In addition, combustion analyses were made with the help of combustion analyzer, in which cylinder pressure and heat release rate were analyzed.

- Jatin H. Vaghela[13]investigatedthe effects of ethanol–gasoline (E5, E10) & methanol–gasoline (M5, M10) fuel blends and 4-spark plugs ignition are on 2-stroke single cylinder SI engine for analyzing the performance and combustion characteristics. The tests were performed using an electric dynamometer while running the engine at constant speed of 3000 rpm and at four different engine load conditions (0.5, 1, 1.5 2 kW). The results obtained from the use of alcohol–gasoline fuel blends with 4-spark plugs were compared to those of single conventional SI engine. The results indicated that when all four spark plugs and alcohol–gasoline fuel blends were used, the brake specific fuel consumption (BSFC) and exhaust emission were decreased. Exhaust gases namely, carbon dioxides ($CO_2$), carbon monoxide (CO) and total unburned hydrocarbons (HC) were measured using multi exhaust gas analyser. Performance and exhausts emissions were compared with conventional gasoline engine with all working 4-spark plugs, using alcohol-gasoline fuel blends. This ignition system shows significant improvement for exhaust emission and also fuel consumptions at different load conditions.

- HuseyinSerdarYucesu [20] examined the effect of compression ratio on engine performance and exhausts emissions at stoichiometric air/fuel ratio, full load and minimum advanced timing for the best torque in a single cylinder, four strokes, with variable compression ratio and spark ignition engine.With increasing compression ratio up to 11:1, enginetorque increased with E0 fuel, at 2000 rpm engine speed.Compared with the 8:1 compression ratio, the increment ratio was about 8%. At the higher compression ratios0.4 the torque output did not change noticeably. At 13:1compression ratio compared with 8:1 compression ratio,the highest increment was obtained for both fuels E40and E60 as nearly 14%.

- At 11:1 compression ratio compared with 8:1, the BSFCof E0 fuel reached minimum value and decreased about10%, after this compression ratio the BSFC increased.The considerable decrease of BSFC was about 15% withE40 fuel at 2000 rpm engine speed. The highest improvementsof BSFC were obtained with E60 fuel as 14.5%and 17% at 3500 and 5000 rpm engine speeds,respectively.
- The fuels containing high ratios of ethanol; E40 and E60had important effects on the reduction exhaust emissions.The maximum decrease was obtained with E40and E60 fuels at 2000 rpm engine speed. The averagedecreases were found to be 11% and 10.8% with E40and E60, respectively. The better decrease was obtainedwith HC compared with CO. The maximum decrease inHC emission was obtained using E60 as average of16.45% at 5000 rpm engine speeds.

- C.AnandaSrinivasan and C.G.Saravanan[15]investigated the effects of ethanol and unleadedgasoline with 1, 4 Dioxan blends on multi-cylinder SI engine.The experimental results revealthe increase in brake thermal efficiency for the blends whencompared to that of sole fuel. In this investigation, the emissiontests were made with the help of AVL Di Gas analyzer, in whichCO, $CO_2$, HC, $NO_X$ were appreciably reduced and $O_2$ increasedfor all the blends when compared to sole fuel.

- ŞehmusAltun, HakanF.Oztop[18]experimentally investigated the effect of unleaded gasoline and unleaded gasolineblended with 5% and 10% of ethanol or methanol on the performanceand exhaust emissions of a spark-ignition engine. The engine tests were performed by varying the enginespeed between 1000 and 4000 rpm with 500 rpm period at threefourththrottle opening positions.The results showed that brake specificfuel consumption increased while brake thermal efficiency,emissions of carbon monoxide (CO) and hydrocarbon (HCs)decreased with methanol-unleaded gasoline and ethanol-unleadedgasoline blends. It was found that a 10% blend of ethanol or methanolwith unleaded gasoline works well in the existing design of engine andparameters at which engines are operating.

- Bang-Quan He, Jian-Xin Wang [16] investigated the effect of ethanol blended gasoline fuels on emissions and catalyst conversion efficiencies in a spark ignition engine with an electronic fuel injection (EFI) system. Result showed that ethanol can decrease engine-out regulatedemissions. The fuel containing 30% ethanol by volume can drastically reduce engine-out total hydrocarbon emissions(THC) at operating conditions and engine-out THC, CO and $NO_x$emissions at idle speed, but unburned ethanol andacetaldehyde emissions are effective in reducing acetaldehyde emissions; butthe conversion of unburned ethanol is low. Tailpipe emissions of THC, CO and $NO_x$have close relation to engine-outemissions, catalyst conversion efficiency, engine's speed and load, air/fuel equivalence ratio. Moreover, the blendedfuels can decrease brake specific energy consumption.

- Amit Pal, S. Maji, O.P. Sharma and M.K.G.Babu [19] operated a Kirloskar, four stroke, 7.35kW, twin cylinder, DI diesel engine in dual fuel mode (with substitution of up to 75% diesel with CNG). The results of this experiment of substituting the diesel by CNG at different loads showed significant reduction in smoke, 10 to 15 % increase in power, 10 to 15 %reduction in fuel consumption and 20 to 40 % saving in fuel cost (considering low cost of CNG). The most exciting result was about 33% reduction in engine noise which may prolong the engine life significantly and the consequent sound levels of giant diesel engine reduced to that of a similarly sized gasoline engine.

- P. A. Hubballi, and T.P. Ashok Babu [17] investigated experimentally the effect of Denatured spirit (DNS) and DNS-Water blends as fuels in a four cylinder four stroke SI engine. Performance tests were conducted to study Brake Thermal Efficiency (BThE), Brake Power (BP), Engine Torque (T) and Brake Specific Fuel Consumption (BSFC). Exhaust emissions were also investigated for carbon monoxide (CO), hydrocarbons (HC), oxides of nitrogen ($NO_x$) and carbon dioxide ($CO_2$). The results of the experiments reveled that, both DNS and DNS95W5 as fuels increase BThE, BP, engine torque and BSFC. The CO, HC, $NO_x$ and $CO_2$ emissions in the exhaust decreased. The DNS and DNS95W5 as fuels produced the encouraging results in engine performance and mitigated engine exhaust emissions.

- HaiboZhai and H. Christopher Frey [14]evaluated differences infuel consumption and tailpipe emissions of flexible fuelvehicles (FFVs) operated on ethanol 85 (E85) versus gasoline.Theoretical ratios of fuel consumption and carbondioxide ($CO_2$) emissions for both fuels are estimatedbased on the same amount of energy released. Second-bysecondfuel consumption and emissions from one FFVFord Focus fueled with E85 and gasoline were measuredunder real-world traffic conditions in Lisbon, Portugal,using a portable emissions measurement system (PEMS).Results showed that for E85 versus gasoline, empirical ratios of fuelconsumption and CO2 emissions agree within a margin oferror to the theoretical expectations. Carbon monoxide (CO) emissions were found to be typically lower. From thePEMS data, nitric oxide (NO) emissions associated withsome higher VSP modes are higher for E85. From thedynamometer and certification data, average hydrocarbon(HC) and nitrogen oxides ($NO_x$) emission differencesvary depending on the vehicle. The differences of averageE85 versus gasoline emission rates for all vehicle models are 22% for CO, 12% for HC, and 8% for $NO_x$ emissions,which imply that replacing gasoline with E85 reducesCO emissions, may moderately decrease $NO_x$tailpipe emissions, and may increase HC tailpipe emissions. On a fuel life cycle basis for corn-based ethanolversus gasoline, CO emissions are estimated to decreaseby 18%. Life-cycle total and fossil $CO_2$ emissions are estimatedto decrease by 25 and 50%, respectively;

however,life-cycle HC and $NO_x$ emissions are estimated to increaseby 18 and 82%, respectively.

## CHAPTER 3

# BASIC THEORY AND THERMODYNAMICS

### 3.1 Introduction

Thermodynamic cycles can be divided into two general categories: power cycles, which produce a net power output, and refrigeration and heat pump cycles, which consume a net power input. The thermodynamic power cycles can be categorized as gas cycles and vapor cycles. In gas cycles, the working fluid remains in the gas phase throughout the entire cycle. In vapor cycles, the working fluid exits as vapor phase during one part of the cycle and as liquid phase during another part of the cycle. Internal combustion engines and gas turbines undergo gas power cycle.

**Fig-3.1 Overlook of an SI Car Engine**

Internal combustion engines, which are commonly used in automobiles, have two principal types: spark-ignition (SI) engines and compression-ignition (CI) engines.

### 3.2 Internal Combustion Engine Terminology

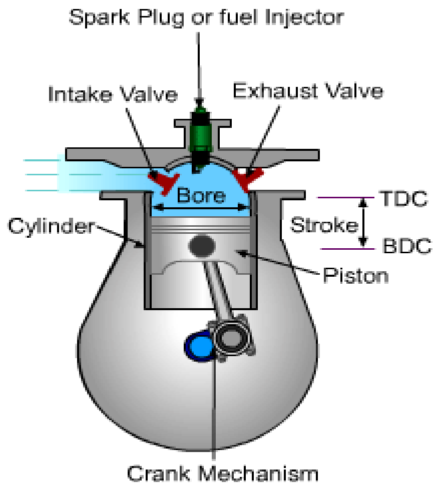



**FIG 3.2-Nomenclature for Reciprocating Engines**

Internal combustion engines are reciprocating engines, which basically are piston-cylinder devices. The sketch of a reciprocating engine is shown above. The sketch is labeled with some special terms.

- The piston is said to be at the top dead center (TDC) when it has moved to a position where the cylinder volume is minimum. This volume is called a clearance volume.
- The piston is said to be at the bottom dead center (BDC) when it has moved to a position where the cylinder volume is maximum.
- The volume swept out by the piston when it moves from TDC to BDC is called the displacement volume.
- The distance from TDC to BDC is called stroke.
- The bore of the cylinder is its diameter.

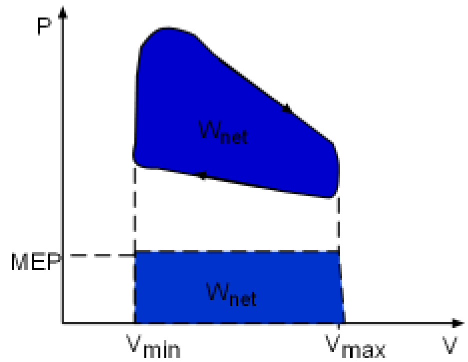



**FIG-3.3 P-Vdiagram**

3.2.1 Definition of MEP

Two other terms frequently used in conjunction with reciprocating engines are compression ratio (r) and mean effective pressure (MEP). The compression ratio is defined as the ratio of the maximum volume formed in the cylinder to the minimum volume (clearance volume).

$$r = \frac{Vmax}{Vmin}$$

The mean effective pressure is a fictitious pressure. It is defined as the pressure that would act on the piston during the entire power stroke, to produce the same amount of net work as that would be produced during the actual cycle.

$$W_{net} = \text{MEP} * \text{Displacement volume}$$

$$MEP = \frac{Wnet}{V_{bdc} - V_{tdc}} \quad \text{..........................} \quad (3.1)$$

### 3.3 Four-stroke Combustion Cycle

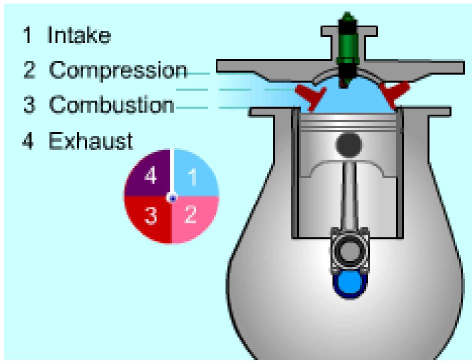



**FIG 3.4 Four-stroke Compression Cycle for SI Engine**

In a spark-ignition (SI) engine, a mixture of fuel and air is ignited by a spark plug. Spark-ignition engines are suited for use in automobiles since they are relatively light and lower in cost. Most cars currently use what is called a four-stroke combustion cycle to convert gasoline into motion. The four strokes are:

- Intake stroke
- Compression stroke
- Combustion stroke (power stroke)
- Exhaust stroke

The piston is connected to the crank shaft by a connecting rod. When the engine goes through its cycle:

- The piston starts at the top, the intake valve opens, and the piston moves down to let the engine take in a cylinder-full of air and gasoline. This is the intake stroke.
- The piston moves back up to compress this fuel/air mixture. Compression makes the explosion more powerful. This is the compression stroke.
- When the piston reaches the top of its stroke (TDC), the spark plug emits a spark to ignite the gasoline. The gasoline in the cylinder explodes, driving the piston down. This is the combustion stroke.
- Once the piston hits the bottom of its stroke (BDC), the exhaust valve opens and the exhaust leaves the cylinder to go out through the tail pipe.This is the exhaust stroke.

Then the engine is ready for the next cycle, so it intakes another charge of air and gas.

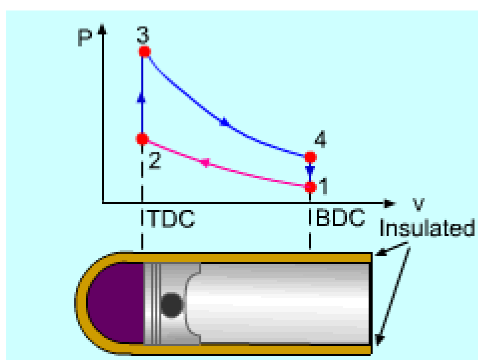
### 3.4 Air-standard Assumption

Internal combustion engine operates on an open cycle since its working fluid is thrown out of the engine at some point instead of being returned to its initial state. That means the working fluid does not undergo a complete thermodynamic cycle. A detailed study of the performance of an actual gas power cycle is rather complex and accurate modeling of internal combustion engines normally involves computer simulation. To conduct elementary thermodynamic analyses of internal combustion engines, considerable simplification is required. To simplify the analysis, air-standard assumptions are made:

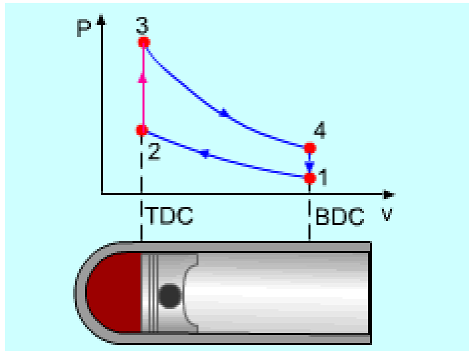
- Gas and air mixture are modeled as air and an ideal gas, which continuously circulates in a closed cycle. Thus, there are no intake and exhaust processes.
- All the processes making up the cycle are internally reversible.
- The combustion process is replaced by a heat-addition process from an external source.
- The exhaust process is replaced by a heat-rejection process and the gas returns to its initial state.

In addition, if specific heats are assumed constants at their ambient temperature, this assumption is called a cold air-standard assumption.

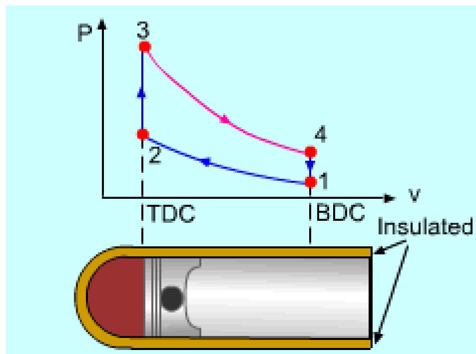
### 3.5 Ideal Otto Cycle - Ideal Cycle for Spark-ignition Engines



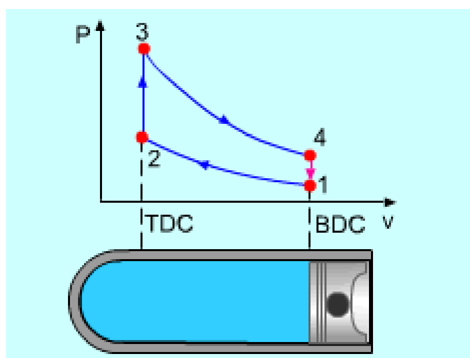
Isentropic Compression (1-2)



Constant Volume Heat Addition (2-3)



Isentropic Expansion (3-4)



Constant Volume Heat Rejection (4-1)

**FIG-3.5 Processes in SI Engine**

The Otto cycle is the ideal cycle for spark-ignition engines, in honor of Nikolaus Otto, who invented it in 1867. In ideal Otto cycles, air-standard assumption is used. The ideal Otto cycle consists of four internal reversible processes:

- 1-2 Isentropic compression

- 2-3 Constant volume heat addition
- 3-4 Isentropic expansion
- 4-1 Constant volume heat rejection

The Otto cycle is executed in a closed system and the working fluid is air according to the air-standard assumption. Also, changes in kinetic and potential energies are negligible. No heat transfer is involved in the two isentropic processes. The energy balances for these two processes are:

$$-w_{12} = u_2 - u_1 \text{.............................. (3.2)}$$

$$-w_{34} = u_4 - u_3 \text{.............................. (3.3)}$$

$w_{12}$ is negative since work is needed to compress the air in the cylinder and $w_{34}$ is positive since air does work to the surroundings during its expansion.

In the constant volume heat addition and heat rejection process, no work interaction is involved since no volume change occurs. The energy balances for these two processes are:

$$q_{23} = u_3 - u_2 \text{.............................. (3.4)}$$

$$q_{41} = u_1 - u_4 \text{.............................. (3.5)}$$

$q_{23}$ is positive since heat is added to the air and $q_{41}$ is negative since heat is rejected to the surroundings.

The thermal efficiency for an ideal Otto cycle is

$$\eta th, otto = \frac{Wnet}{Qin} \text{.............................. (3.6)}$$

According to the analysis above, the net work output is

$$w_{net} = w_{34} + w_{12} = q_{23} + q_{41}$$

$$q_{in} = q_{23}$$

$$\eta th, otto = 1 + \frac{q41}{q23} \text{ .............................. (3.7)}$$

Under the cold air-standard assumption, the thermal efficiency of the ideal Otto cycle is

$$\eta th, otto = 1 - \frac{C_v * (T_4 - T_1)}{C_v * (T_3 - T_2)}$$

$$\eta th, otto = 1 - \frac{T_1}{T_2} * \frac{\frac{T_4}{T_1} - 1}{\frac{T_3}{T_2} - 1} \text{.............................. (3.8)}$$

Process 1-2 and process 3-4 are isentropic. Thus,

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} \text{and} \frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

Since $v_2 = v_3$ and $v_4 = v_1$,

$$\frac{T_1}{T_2} = \frac{T_4}{T_3} \leftrightarrow \frac{T_4}{T_1} = \frac{T_3}{T_2} \text{.............................. (3.9)}$$

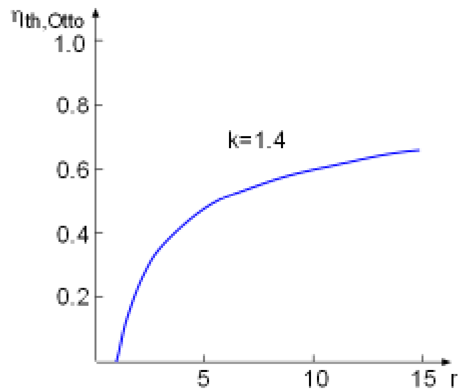



**FIG3.6 Relation between Thermal Efficiency and Compression Ratio with k =1.4**

Considering all the relations above, the thermal efficiency becomes,

$$\eta th, otto = 1 - \frac{T_1}{T_2} = 1 - (\frac{v_2}{v_1})^{k-1} = 1 - (\frac{1}{r^{k-1}}) \text{ ……………………….. (3.10)}$$

Where r is the compression ratio and k is the specific heat ratio.

The expression of thermal efficiency under cold air-standard assumption is only a function of the compression ratio. Thus, a higher r can generate a higher thermal efficiency. But when higher r is used, the temperature of the air-fuel mixture may rise above the auto ignition temperature of the fuel during the compression process, and will cause an early and rapid burn before the spark ignition. This early and rapid burn produces an audible noise, which is called engine knock. Engine knock in spark-ignition engine cannot be tolerated since it hurts performance and can cause engine damage. Thus there is an upper limit of compression ratio for spark-ignition engines.

### 3.6 Combustion

Combustion is a chemical reaction in which certain element of the fuel combine with oxygen and releasing a large quantity of energy causing an increase in temperature of gases. There are many thousands of different hydrocarbon fuel components, which consist mainly of hydrogen and carbon but may also contain oxygen, nitrogen, and/or sulphur, etc. The main combustible elements are carbon and hydrogen; another combustible element often present in fuels, although rather undesirable, is sulphur.

3.6.1 Composition of air

Table 3.1 gives proportion of oxygen and nitrogen by volume as well as by mass of dry air. In combustion, oxygen is the reactive component of air. The properties of air vary geographically, with altitude and with time. It is usually sufficiently accurate to regard air as 21% oxygen and 79% inert gases taken as nitrogen (often called atmospheric nitrogen) by volume.

| Gas | Volume % | Mass% | Molar Mass | Molar Fraction | Molar Ratio |
|---|---|---|---|---|---|
| $O_2$ | 20.95 | 23.16 | 32.00 | 0.21 | 1 |
| $N_2$ | 78.09 | 75.55 | 28.01 | 0.79 | 3.76 |
| Ar | 0.93 | 1.25 | 38.95 | Very less | Very less |
| $CO_2$ | 0.03 | 0.04 | 44.01 | Very less | Very less |
| Air | 100.00 | 100.00 | 28.95 | 1.00 | 4.76 |

**TABLE-3.1 components of air**

For each mole of oxygen in air there are $\frac{1-0.21}{0.21}=3.76$ moles of atmospheric nitrogen

The molar mass of air is obtained as 28.95 (usually approximated by 29) from the equation $M=\frac{1}{n}\sum_i n_i M_i=\sum_i \tilde{x}_i M_i$ where $\tilde{x}_i$ is the mole fraction defined as the number of moles of each component $n_i$, divided by the total number of moles of mixture $n$. Because atmospheric nitrogen contains traces of other species, its molar mass is slightly different from that of pure molecular nitrogen, i.e.

$M_{aN_2}=\frac{28.95-.21\times 32}{1-0.21}=28.14$. The density of dry air can be obtained from equation of state with universal gas constant, $R_o$= 8314.3 J/kmol K and $M$ = 28.95:

$$\rho\left(kg/m^3\right)=\frac{3.482\times 10^{-3}\,p\,(Pa)}{T(K)}.$$

Thus the value for the density of dry air at 1 atmosphere (1.0133 $\times$ $10^5$ Pa) and 25°C is 1.184 kg/m$^3$.Actual air normally contains water vapor, the amount depending on temperature and degree of saturation. Typically the proportion by mass is about 1 percent, though it can rise to about 4 percent under extreme conditions. The relative humidity compares the water vapor content of air with that required to saturate. It is defined as the ratio of the partial pressure of water vapor actually present to the saturation pressure at the same temperature.

### 3.6.2 Stoichiometry

Most IC engines obtain their energy from the combustion of a hydrocarbon fuel with air, which converts chemical energy of the fuel to internal energy in the gases within the engine. The maximum amount of chemical energy that can be released (heat) from the fuel is when it reacts (combust) with a stoichiometric amount of oxygen. Stoichiometric oxygen (sometimes also called theoretical oxygen) is just enough to convert all carbon in the fuel to $CO_2$ and all hydrogen to $H_2O$, with no oxygen left over.

### 3.6.3 Stoichiometric Reaction

A stoichiometric reaction is defined such that the only products are carbon dioxide and water. The components on the left side of a chemical reaction equation which are present before the reaction are called reactants, while the components on the right side of the equation which are present after the reaction are called products or exhaust.

Chemical equations are balanced on a basis of the conservation of mass principle (or the mass balance), which can be stated as follows: The total mass of each element is conserved during a chemical reaction. That is, the total mass of each element in the products must be equal to the total mass of that element in the reactants even though the elements exist in different chemical compounds in the reactants and products. Also, the total number of atoms of each element is conserved during a chemical reaction since the total number of atoms of an element is equal to the total mass of the element divided by its atomic mass. The total number of moles is not conserved during a chemical reaction.

In chemical reactions molecules react with molecules, so in balancing chemical equations, molar quantities (fixed number of molecules) are used and not mass quantities. It is convenient to balance combustion reaction equations for one kmole of fuel. The energy released by the reaction will thus have units of energy per kmole of fuel, which is easily transformed to total energy when the flow rate of fuel is known.

One kmole of a substance has a mass in kilograms equal in number to the molecular mass (molar mass) of that substance. Mathematically, $m= NM$[kmole] [kg/kmole], where: $m$= mass[kg], $N$ = number of moles[kmole], $M$ = molecular mass[kg/kmole], 1 kmole = $6.02 \times 10^{26}$ molecules.

Very small powerful engines could be built if fuel were burned with pure oxygen. However, the cost of using pure oxygen would be prohibitive, and thus is not done. Air is used as the source of oxygen to react with fuel. Nitrogen and argon are essentially chemically neutral and do not react in the combustion process. Their presence, however, does affect the temperature and pressure in the combustion chamber.Nitrogen usually enters a combustion chamber in large quantities at low temperatures and exists at considerably higher temperatures, absorbing a large proportion of the chemical energy released during combustion. When the products are at low temperature the nitrogen is not significantly affected by the reaction. At very high temperatures a small fraction of nitrogen reacts with oxygen, forming hazardous gases called $NO_x$.

3.6.4 Stoichiometric Air/Fuel Ratio

Stoichiometric (or chemically correct or theoretical) proportions of fuel and air are calculated from the stoichiometric reaction on molar basis. The stoichiometric Air to fuel ratio on molar basis is $4.76\varepsilon = 4.76\,(\alpha + 0.5\beta)$.

The stoichiometric air/fuel $(A/F)_s$ or fuel/air $(F/A)_s$ ratios depend on fuel composition. Stoichiometric air/fuel $(A/F)_s$ on mass basis can also be calculated.

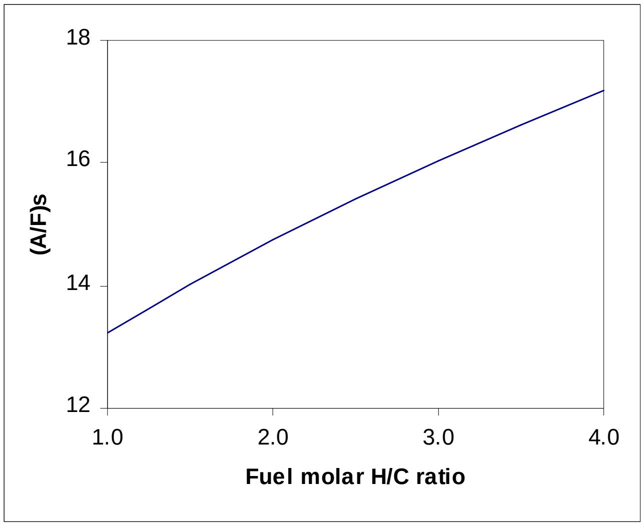



**FIG3.7 Dependency of Air Fuel ratio on fuel composition**

Figure 3.7 shows the variation in $(A/F)_s$ as the ratio of hydrogen to carbon ($y$ = H/C) varies from 1 (e.g. benzene) to 4 (methane).

### 3.6.5 Lean or Rich Mixture Reactions

Fuel-air mixtures with more than or less than the stoichiometric air requirement can be burned. Combustion can occur, within limits, i.e., the proportions of the fuel and air must be in the proper range for combustion to begin. For example, natural gas will not burn in air in concentrations less than 5 percent or greater than about 15 percent. With excess air or fuel-lean combustion, the extra air appears in the products in unchanged form. With less than stoichiometric air requirement, i.e., with fuel-rich combustion, there is insufficient oxygen to oxidize fully the fuel C and H to $CO_2$ and $H_2O$. The products are a mixture of $CO_2$ and $H_2O$ with carbon monoxide CO and hydrogen $H_2$ (as well as $N_2$). Carbon monoxide is a colourless, odourless, poisonous gas which can be further burned to form $CO_2$. It is produced in any combustion process when there is a deficiency of oxygen. It is very likely that some of the fuel will not get burned when there is a deficiency of oxygen. This unburned fuel ends up as pollution in the exhaust of the engine.Because the composition of the combustion products is significantly different for fuel-lean and fuel-rich mixtures, and because the stoichiometric fuel/air ratio depends on fuel composition, the ratio of the actual fuel/air ratio to the stoichiometric ratio (or its inverse) is a more informative parameter for defining mixture composition. Various terminologiesis used for the amount of air or oxygen used in combustion. 80% stoichiometric air = 80% theoretical air = 80% air = 20% deficiency of air; 120% stoichiometric air = 120% theoretical air = 120% air = 20% excess air.

3.6.6 Fuel/Air Equivalence Ratio

For actual combustion in an engine, the fuel/air equivalence ratio is a measure of the fuel-air mixture relative to stoichiometric conditions. It is defined as: $\phi = \frac{(F/A)_{act}}{(F/A)_{stoich}} = \frac{(A/F)_{stoich}}{(A/F)_{act}}$ where: $F/A = m_f / m_a$ = fuel-air ratio; A/F = $m_a / m_f$ = air-fuel ratio; $m_a$ = mass of air; $m_f$ = mass of fuel

3.6.7 Relative Air/Fuel Ratio

The inverse of $\phi$, the relative air/fuel ratio $\lambda$, is also sometimes used. $\lambda = \phi^{-1} = \frac{(F/A)_{stoich}}{(F/A)_{act}} = \frac{(A/F)_{act}}{(A/F)_{stoich}}$ For fuel-lean mixtures: $\phi < 1$, $\lambda > 1$, oxygen in exhaust

For stoichiometric mixtures: $\phi = \lambda = 1$, maximum energy released from fuel

For fuel-rich mixtures: $\phi > 1$, $\lambda < 1$, CO and fuel in exhaust.

3.6.8 Combustion Efficiency

Even when the flow of air and fuel into an engine is controlled exactly at stoichiometric conditions, combustion will not be "perfect," and components other than $CO_2$, $H_2O$, and $N_2$ are found in the exhaust products. One major reason for this is the extremely short time available for each engine cycle, which often means that less than complete mixing of the air and fuel is obtained. Some fuel molecules do not find an oxygen molecule to react with, and small quantities of both fuel and oxygen end up in the exhaust. In practice, the exhaust gas of an internal combustion engine contains incomplete combustion products (e.g., CO, $H_2$, unburned hydrocarbons, soot) as well as complete combustion products ($CO_2$ and $H_2O$).

Under lean operating conditions the amounts of incomplete combustion products are small. Under fuel-rich operating conditions these amounts become more substantial since there is insufficient oxygen to complete combustion.

Because a fraction of the fuel's chemical energy is not fully released inside the engine during the combustion process, hence combustion efficiency is defined as the fraction of the fuel energy supplied which is released in the combustion process. Figure shows how combustion efficiency varies with the fuel/air equivalence ratio for internal combustion engines.

SI engines have combustion efficiency in the range of 95% to 98% for lean mixtures. For rich mixtures, where there is not enough air to react all the fuel, the combustion efficiency steadily decreases as the mixture becomes richer. Combustion efficiency is little affected by other engine operating and design variables, provided the engine combustion process remains stable

CI engines always operate lean overall and typically have combustion efficiencies of about 98%.

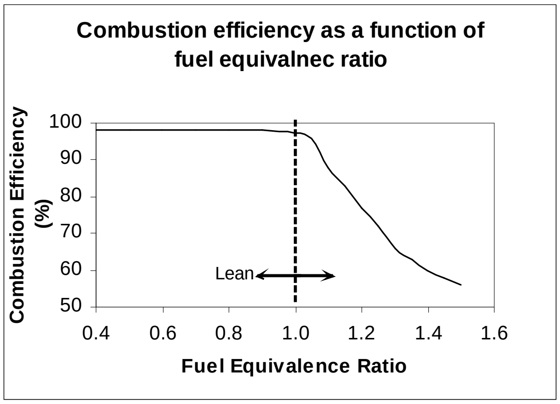



**Fig3.8 Combustion efficiency as a function of fuel equivalence ratio**

3.6.9 <u>Self ignition temperature</u>

Bringing a fuel into intimate contact with oxygen is not sufficient to start a combustion process. The fuel must be brought above its ignition temperature (also called self-ignition temperature) to start the combustion. If the temperature of an air-fuel mixture is raised high enough, the mixture will self-ignite without the need of a spark plug or other external igniter. This is the basic principle of ignition in a compression ignition engine. The compression ratio is high enough so that the temperature rises above self-ignition temperature (SIT) during the compression stroke. Self-ignition then occurs when fuel is injected into the combustion chamber. On the other hand, self-ignition (pre-ignition, or auto-ignition) is not desirable in an SI engine, where a spark plug is used to ignite the air-fuel at the proper time in the cycle. The compression ratios of gasoline-fueled SI engines are limited to about 11:1 to avoid self-ignition. Table gives minimum ignition temperatures of various substances in atmospheric air.

When self ignition does occur in an SI engine higher than desirable, pressure pulses are generated. These high pressure pulses can cause damage to the engine and quite often are in the audible frequency range. This phenomenon is often called knock or ping. Figure shows the basic process of what happens when self-ignition occurs. If a combustible air-fuel mixture is heated to a temperature less than SIT, no ignition will occur and the mixture will cool off. If the mixture is heated to a temperature above SIT, self-ignition will occur after a short time delay called ignition delay (ID). The higher the initial temperature rise above SIT, the shorter will be ID.Ignition delay is generally on the order of thousandths of a second. The values for SIT and ID for a given air-fuel mixture are ambiguous, depending on many variables which include temperature, pressure, density, turbulence, swirl, fuel-air ratio, presence of inert gases, etc. Ignition delay is generally a very small fraction of second. During this time, pre ignition reactions occur, including oxidation of some fuel components and even cracking of some large hydrocarbon components into smaller HC molecules. These pre ignition reactions raise the temperature at local spots, which then promotes additional reactions until; finally, the actual combustion reaction occurs.

| Fuel | SIT (°C) |
|---|---|
| Hydrogen | 580 |
| Methane | 630 |
| Ethane | 515 |
| CO | 610 |
| Propane | 480 |
| Gasoline | 260 |
| Kerosine | 210 |
| Diesel | 210 |
| Ethanol | 365 |
| Methanol | 385 |

**Table 3.2 Self ignition temperature of fuels**

3.6.10<u>Self Ignition</u>

Figures 3.9 a, b, c show the pressure-time history within a cylinder of a typical SI engine.

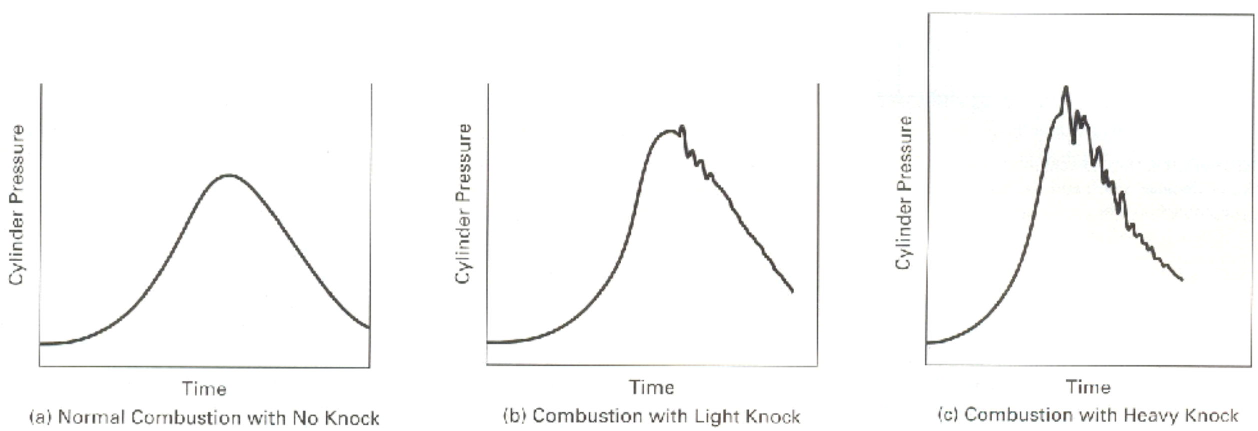



**Fig 3.9 P-Time curves of SI Engine**

With no self-ignition the pressure force on the piston follows a smooth curve, resulting in smooth engine operation (Fig. a). When self-ignition does occur, pressure forces on the piston are not smooth and engine knock occurs (Fig. b & c).The combustion process is a fast exothermic gas-phase reaction (where oxygen is usually one of the reactants). A flame is a combustion reaction which can propagate sub sonically through space; motion of the flame relative to the unburned gas is the important feature. The reaction zone is usually called the flame front.

# EXPERIMENTAL SET UP

Fig 4.1 Diagram of Setup used

## 4.1 DESCRIPTION

The setup consists of fourcylinder, four stroke, Petrol(MPFI) engine connected toeddy current typedynamometer for loading. It isprovided with necessaryinstruments for combustionpressure and crank-anglemeasurements. These signalsare interfaced to computerthrough engine indicator forPθ−PV diagrams. Provision isalso made for interfacingairflow, fuel flow,temperatures and loadmeasurement. The set up hasstand-alone panel boxconsisting of air box, fueltank, manometer, fuelmeasuring unit, transmittersfor air and fuel flowmeasurements, processindicator and engine indicator.Rotameters are provided forcooling water and calorimeterwater flow measurement.

The setup enables study of engine performancefor brake power, indicated power, frictionalpower, BMEP, IMEP, brake thermal efficiency,indicated thermal efficiency, Mechanicalefficiency, volumetric efficiency, specific fuelconsumption, A/F ratio and heat balance.Windows based Engine Performance Analysissoftware package “Enginesoft” is provided for online performance evaluation.

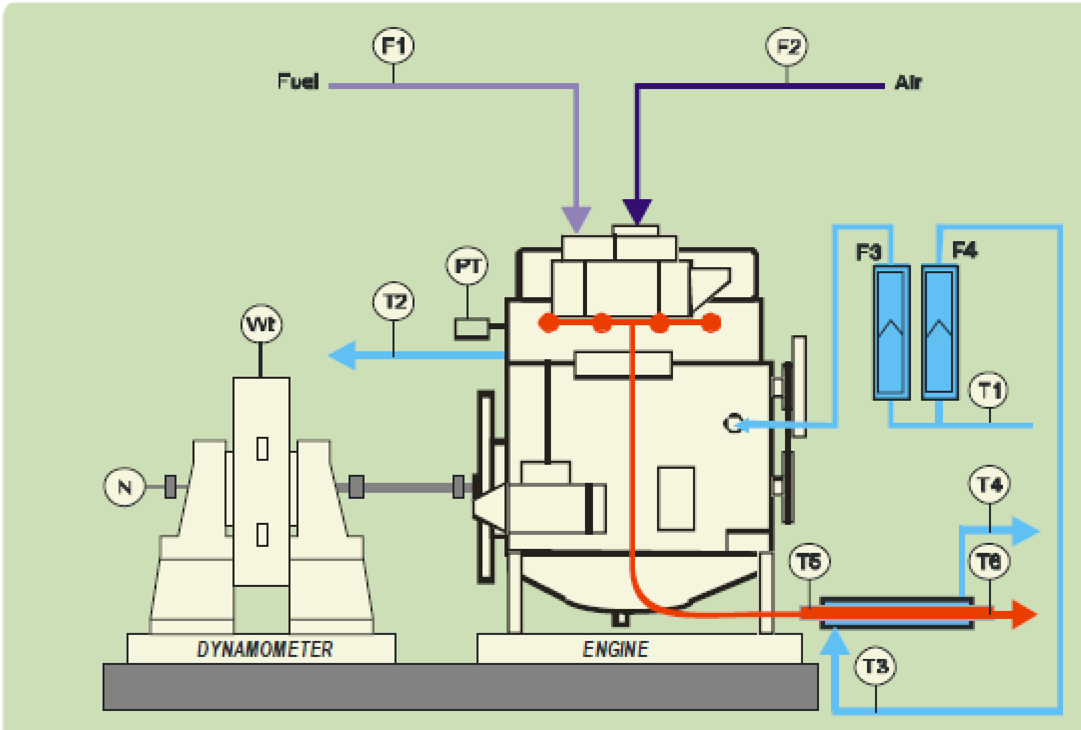



**Fig 4.2 Diagram showing Temperature at different points**

4.1.1 Specification

| | |
|---|---|
| **Product** | Engine test setup 4 cylinder, 4 stroke, Petrol(Computerized) |
| **Product code** | 233 |
| **Engine** | Make Maruti, Model Wagon-R MPFI, Type 4 Cylinder, 4Stroke, Petrol (MPFI), water cooled, Power 44.5Kw at6000 rpm, Torque 59 NM at 2500rpm, stroke 61mm,bore 72mm, 1100 cc,CR 9.4:1 |
| **Dynamometer** | Type eddy current, water cooled, with loading unit |
| **Propeller shaft** | With universal joints |
| **Air box** | M S fabricated with orifice meter and manometer(Orifice dia 40 mm) |

| | |
|---|---|
| **Fuel tank** | Capacity 15 lit with glass fuel metering column |
| **Calorimeter** | Type Pipe in pipe |
| **Piezo sensor** | Range 5000 PSI, with low noise cable |
| **Crank angle sensor** | Resolution 1 Deg, Speed 5500 RPM with TDC pulse |
| **Engine indicator** | Input Piezo sensor, crank angle sensor, No of channels 2, Communication RS232 |
| **Digital milivoltmeter** | Range 0-200mV, panel mounted |
| **Temperature sensor** | Type RTD, PT100 and Thermocouple, Type K |
| **Temperature transmitter** | Type two wire, Input RTD PT100, Range 0–100 Deg C, Output 4–20 mA and Type two wire, Input Thermocouple, Range 0–1200 Deg C, Output 4–20 mA |
| **Load indicator** | Digital, Range 0-50 Kg, Supply 230VAC |
| **Load sensor** | Load cell, type strain gauge, range 0-50 Kg |
| **Fuel flow transmitter** | DP transmitter, Range 0-500 mm WC |
| **Air flow transmitter** | Presure transmitter, Range (-) 250 mm WC |
| **Rotameter** | Engine cooling 100-1000 LPH; Calorimeter 25-250 LPH |
| **Pump** | Type Monoblock |
| **Add on card** | Resolution12 bit, 8/16 input, Mounting PCI slot |
| **Software** | “Enginesoft” Engine performance analysis software |
| **Overall dimensions** | W 2000 x D 2750 x H 1750 mm |

**Table 4.1 Specification of Equipment**

### 4.1.2 Commissioning

- Fill lubrication oil in the engine and fuel in the fuel tank.
- Remove air from fuel line connecting fuel measuring unit to fuel transmitter.
- Lower jack bolts under dynamometer for free movement.
- Provide electric supply to panel box
- Adjust crank angle sensor for TDC matching.
- Confirm all temperatures are correctly displayed on process indicator
- Confirm load signal displayed on process indicator
- Fill water in the manometer up to “0” mark level.
- Keep “Load” knob on loading unit is at minimum position.
- Load the Add on card driver on the computer from Driver CD.
- Insert add on card in computer motherboard slot.
- Connect signal cable from engine panel to add on card in computer.
- Load “Enginesoft” software package on the same computer.
- Ensure water circulation through engine, calorimeter, dynamometer and piezoadaptor. Start the Engine.
- Check engine operation at various loads and ensure respective signals oncomputer.

### 4.1.3 Precautions

- Use clean and filtered water; any suspended particle may clog the piping.
- Piezo Sensor Handling:
- Ensure cooling water circulation for combustion pressure sensor.

- Diaphragm of the sensor is delicate part. Avoid scratches or hammering onit.
- A long sleeve is provided inside the piezo adapter. This sleeve is protectingthe surface of the diaphragm. While removing sensor from the adapter thissleeve may come out with the sensor and fell down or lose during handling.
- Status of the sensor is indicated on the engine indicator.
- Damages to the electronic parts of the sensor or loose connection are
- indicated as "open" or "short" status on engine indicator.

➢ Circulate dynamometer and piezo sensor cooling water for some time aftershutting down the engine

4.1.4 Components Used

| **Components** | **Details** |
|---|---|
| **Engine** | Make Maruti, Model Wagon-R MPFI, Type 4 Cylinder,4 Stroke, Petrol (MPFI), water cooled, Power 44.5Kwat 6000 rpm, Torque 59 NM at 2500rpm, stroke61mm, bore 72mm, 1100 cc,CR 9.4:1 |
| **Dynamometer** | Make Saj test plant Pvt. Ltd., Model AG80, Type Eddycurrent |
| **Dynamometer Loadingunit** | Make Cuadra, Model AX-153, Type variable speed,Supply 230V AC |
| **Propeller shaft** | Make Hindustan Hardy Spicer, Model 1260, Type A |
| **Manometer** | Make Apex, Model MX-104, Range 100-0-100 mm,Type U tube, Conn. 1/4‘‘ BSP hose back side,Mounting panel |
| **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 |
| **White coaxial Teflon Cable** | Make PCB piezotronics, Model 002C20, Length 20 ft,Connections one end BNC plug and other end 10-32 Micro |
| **Crank angle sensor** | Make Kubler-Germany Model 8.3700.1321.0360 Dia:37mm Shaft Size: Size 6mmxLength 12.5mm, SupplyVoltage 5-30V DC, Output Push Pull (AA,BB,OO),PPR: 360, Outlet cable type axial with flange 37 mmto 58 mm |
| **Engine indicator** | Make-Cuadra, Model AX-104, Type Duel channel |
| **Temperature sensor** | Make Radix Type K, Ungrounded, SheathDia.6mmX110mmL, SS316, Connection 1/4"BSP (M)adjustable compression fitting |
| **Temperature sensor** | Make Radix, Type Pt100, Sheath Dia.6mmX110mmL,SS316, Connection 1/4"BSP(M) adjustablecompression fitting |
| **Temperature transmitter** | Make Wika, model T19.10.3K0-4NK-Z, InputThermocouple (type K), output 4-20mA, supply24VDC, Calibration: 0-1200deg.C. |
| **Temperature transmitter** | Make Wika, Model T19.10.1PO-1 Input RTD(Pt100),output 4-20mA, supply 24VDC, Calibration: 0-100°C |
| **Load sensor** | Make SensotronicsSanmar Ltd., Model 60001,Type Sbeam, Universal, Capacity 0-50 kg |
| **Load indicator** | Make Selectron, model PIC 152–B2, 85 to 270VAC,retransmission |

| | |
|---|---|
| | output 4-20 mA |
| **Power supply** | Make Meanwell, model S-15-24, O/P 24 V, 0.7 A |
| **Digital voltmeter** | Make Meco, 3.1/2 digit LED display, range 0-20 VDC,supply 230VAC, model SMP35 |
| **Fuel flow transmitter** | Make Yokogawa, Model EJA110-EMS-5A-92NN,Calibration range 0-500 mm H2O, Output linear |
| **Air flow transmitter** | Make WIKA, Model SL-1-A-MQA-ND-ZA4Z-ZZZ,output 4-20 mA, supply 10-30 Vdc, conn. Range -25 - 0 mbar |
| **Rotameter** | Make Eureka Model PG 5, Range 25-250 lph,Connection ¾" BSP vertical, screwed, Packingneoprene |
| **Rotameter** | Make Eureka, Model PG 9, Range 100-1000 lph,Connection 1" BSP vertical, screwed, Packingneoprene |
| **Pump** | Make Kirloskar, Model Mini 18SM, HP 0.5, Size 1" x1", Single ph 230 V AC |
| **Add on card** | Make Dynalog, Model - PCI1050, 12-Bit |
| **Battery** | Make Exide, Model MHD 350 06687, 12 V DC |

**Table 4.2 Components of Set up**

## 4.2 Theory

### 4.2.1 Terminology

- **Engine Cylinder diameter (bore) (D):** The nominal inner diameter of theworking cylinder.

- **Piston area (A):** The area of a circle of diameter equal to enginecylinder diameter (bore). A = $\pi / 4\times D^2$
- **Engine Stroke length (L):** The nominal distance through which a workingpiston moves between two successive reversals of its direction of motion.
- **Dead center:** The position of the working piston and the moving parts, whichare mechanically connected to it at the moment when the direction of the piston motion is reversed (at either end point of the stroke).
- **Bottom dead center (BDC):** Dead center when the piston is nearest tothe crankshaft. Sometimes it is also called outer dead center (ODC).
- **Top dead center (TDC):** Dead center when the position is farthest from thecrankshaft. Sometimes it is also called inner dead center (IDC).
- **Swept volume (VS):** The nominal volume generated by the working pistonwhen travelling from one dead center to next one, calculated as the product ofpiston area and stroke. The capacity described by engine manufacturers in ccis the swept volume of the engine. $Vs=A\ L=\pi / 4\times D^2\ L$
- **Clearance volume (VC):** The nominal volume of the space on the combustion sideof the piston at top dead center.
- **Cylinder volume:** The sum of swept volume and clearance volume. $V = V_s+V_c$
- **Compression ratio (CR):** The numerical value of the cylinder volume dividedby the numerical value of clearance volume. $CR = V / V_c$

### 4.2.2 <u>Four stroke cycle engine</u>

In four-stroke cycle engine, the cycle of operation is completed in four strokes of the piston or two revolutions of the crankshaft. Each stroke consists of 1800 of crankshaft rotation and hence a cycle consists of 7200 of crankshaft rotation. The series of operation of an ideal four-stroke engine are as follows:

**1. Suction or Induction stroke:** The inlet valve is open, and the piston travels down the cylinder, drawing in a charge of air. In the case of a spark ignition engine the fuel is usually pre-mixed with the air.

**2. Compression stroke:** Both valves are closed, and the piston travels up the cylinder. As the piston approaches top dead centre (TDC), ignition occurs. In the case of compression ignition engines, the fuel is injected towards the end of compression stroke.

**3. Expansion or Power or Working stroke:** Combustion propagates throughout the charge, raising the pressure and temperature, and forcing the piston down. At the end of the power stroke the exhaust valve opens, and the irreversible expansion of the exhaust gases is termed 'blow-down'.

**4. Exhaust stroke:** The exhaust valve remains open, and as the piston travels up the cylinder the remaining gases are expelled. At the end of the exhaust stroke, when the exhaust valve closes some exhaust gas residuals will be left; these will dilute the next charge.

### 4.2.3 Two stroke cycle engine

In two stroke engines the cycle is completed in two strokes of piston i.e. one revolution of the crankshaft as against two revolutions of four stroke cycle engine.The two-stroke cycle eliminates the separate induction and exhaust strokes.

1. **Compression stroke:** The piston travels up the cylinder, so compressing the trapped charge. If the fuel is not pre-mixed, the fuel is injected towards the end of the compression stroke; ignition should again occur before TDC. Simultaneously under side of the piston is drawing in a charge through a spring loaded non-return inlet valve
2. **Power stroke:** The burning mixture raises the temperature and pressure in the cylinder, and forces the piston down. The downward motion of the piston also compresses the charge in the crankcase. As the piston approaches the end of its stroke the exhaust port is uncovered and blowdown occurs. When the piston is at BDC the transfer port is also uncovered, and the compressed charge in the crankcase expands into the cylinder. Some of the remaining exhaust gases are displaced by the fresh charge; because of the flow mechanism this is called 'loop scavenging'. As the piston travels up the cylinder, the piston closes the first transfer port, and then the exhaust port is closed.

4.2.4 Performance of I.C.Engines

- **Indicated thermal efficiency ($\eta_t$):** Indicated thermal efficiency is the ratio of energy in the indicated power to the fuel energy.

$$\eta_t = \frac{\text{IndicatedPower}}{\text{fuel energy}} \qquad \text{(4.1)}$$

- **Brake thermal efficiency ($\eta_{bth}$):** A measure of overall efficiency of the engineis given by the brake thermal efficiency. Brake thermal efficiency is the ratio ofenergy in the brake power to the fuel energy.

$$\eta_{bth} = \frac{\text{Brake Power}}{\text{fuel energy}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.2)$$

- **Mechanical efficiency ($\eta_m$):** Mechanical efficiency is the ratio of brake horse power(delivered power) to the indicated horsepower (power provided to the piston).

$$\eta_{bth} = \frac{\text{Brake Power}}{\text{Indicated Power}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (4.3)$$

- Frictional power = Indicated power – Brake power.

Figure 4.3 gives diagrammatic representation of various efficiencies,

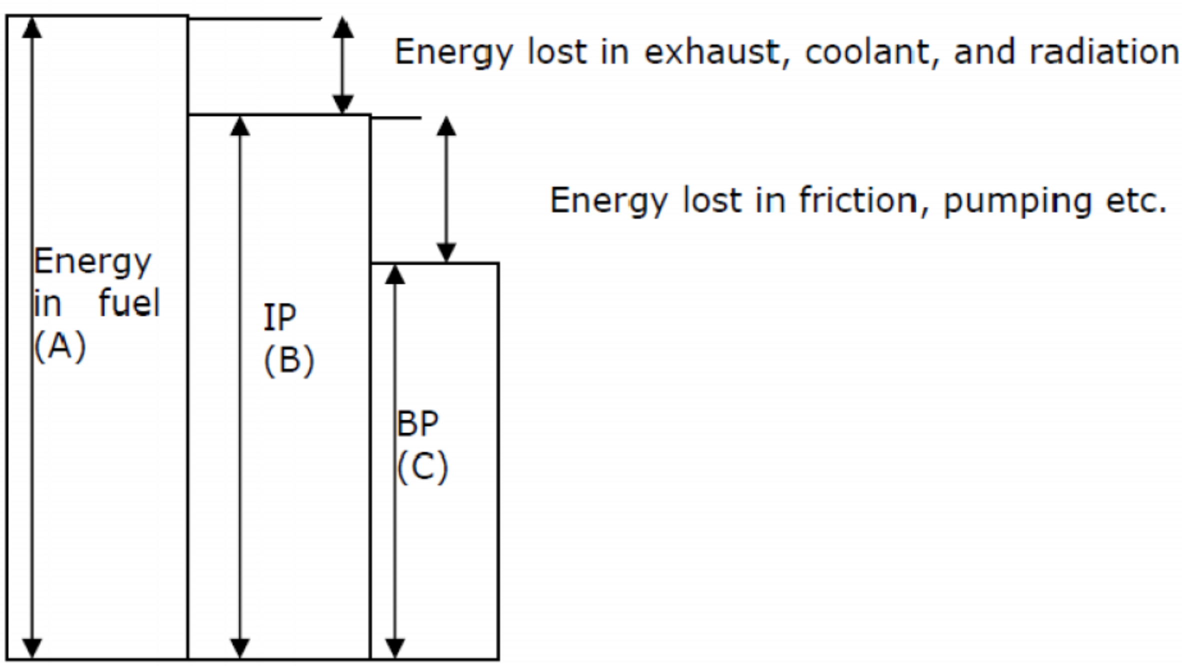



**Fig 4.3 Various Efficiencies**

Indicated thermal efficiency = B/A

Brake thermal efficiency = C/A

Mechanical efficiency = C/B

- **Volumetric efficiency ($\eta_v$):** The engine output is limited by the maximumamount of air that can be taken in during the suction stroke, because only acertain amount of fuel can be burned effectively with a given quantity of air.Volumetric efficiency is an indication of the 'breathing' ability of the engine andis defined as the ratio of the air actually induced at ambient conditions to theswept volume of the engine. In practice the engine does not induce a completecylinder full of air on each stroke, and it is convenient to define volumetricefficiency as:

$$\eta_v\,(\%) = \frac{\text{mass of air induced}}{\text{mass flow of air to fill swept volume at atm condition}}$$

- **Air flow:**

For air consumption measurement air box with orifice is used.

$$\text{Air Flow}\left(\frac{\text{kg}}{\text{hr}}\right) = C_d * \frac{\pi}{4} * D^2 * \sqrt{2 * g * h_{water} * W_{den}/A_{den}} * A_{den} * 3600 \ldots\ldots (4.4)$$

Where Cd = Coefficient of discharge of orifice

D = Orifice diameter in m

g = Acceleration due to gravity ($m/s^2$) = 9.81 $m/s^2$

h = Differential head across orifice (m of water)

Wden = Water density (kg/$m^3$) =@1000 kg/$m^3$

Wair = Air density at working condition (kg/$m^3$) = p/RT

Where

p= Atmospheric pressure in kgf/m2 (1 Standard atm. = 1.0332X104 kgf/$m^2$)

R= Gas constant = 29.27 kgf.m/kg$^0$k

T= Atmospheric temperature in $^0$k

**Specific fuel consumption (SFC):** Brake specific fuel consumption and indicatedspecific fuel consumption, abbreviated BSFC and ISFC, are the fuel consumptionson the basis of Brake power and Indicated power respectively.

**Fuel-air (F/A) or air-fuel (A/F) ratio:** The relative proportions of the fuel and airin the engine are very important from standpoint of combustion and efficiency ofthe engine. This is expressed either as the ratio of the mass of the fuel to that ofthe air or vice versa.

**Calorific value or Heating value or Heat of combustion:** It is the energyreleased per unit quantity of the fuel, when the combustible is burned and theproducts of combustion are cooled back to the initial temperature of combustiblemixture. The heating value so obtained is called the higher or gross calorific valueof the fuel. The lower or net calorific value is the heat released when water in theproducts of combustion is not condensed and remains in the vapour form.

**Power and Mechanical efficiency:** Power is defined as rate of doing work andequal to the product of force and linear velocity or the product of torque and angular velocity.

Thus, the measurement of power involves the measurement offorce (or torque) as well as speed.

The power developed by an engine at the output shaft is called brake power andis given by

Power = NT/60,000 in kW…………………… (4.5)

Where T= torque in Nm = WR

W = 9.81 * Net mass applied in kg.

R= Radius in m

N is speed in RPM

**Mean effective pressure and torque:** Mean effective pressure is defined as ahypothetical pressure, which is thought to be acting on the piston throughout thepower stroke.

Power in kW = (Pm LAN/n 100)/60 in bar

where Pm = mean effective pressure

L = length of the stroke in m

A = area of the piston in $m^2$

N = Rotational speed of engine RPM

n= number of revolutions required to complete one engine cycle

n= 1 (for two stroke engine)

n= 2 (for four stroke engine)

Thus we can see that for a given engine the power output can be measured in terms of mean effective pressure. If the mean effective pressure is based on brake power it is called brake mean effective pressure (BMEP) and if based onindicated power it is called indicated mean effective pressure (IMEP).

$$\mathrm{BMEP(bar)} = \frac{\mathrm{BP(KW)*60}}{\mathrm{A*L*\left(\frac{N}{n}\right)*No\ of\ cycle*100}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4.6)}$$

$$\mathrm{IMEP(bar)} = \frac{\mathrm{IP(KW)*60}}{\mathrm{A*L*\left(\frac{N}{n}\right)*No\ of\ cycle*100}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4.7)}$$

Similarly the friction means effective pressure (FMEP) can be defined as

FMEP=IMEP-BMEP…………………… (4.8)

4.2.5<u>Basic measurements</u>

The basic measurements, which usually should be undertaken to evaluate theperformance of an engine on almost all tests, are the following:

**1 Measurement of speed**

Following different speed measuring devices are used for speed measurement.

1 Photoelectric/Inductive proximity pickup with speed indicator

2 Rotary encoder

**2 Measurement of fuel consumption**

**I) Volumetric method**: The fuel consumed by an engine is measured bydetermining the volume flow of the fuel in a given time interval and multiplying it bythe specific gravity of fuel. Generally a glass burette having graduations in ml is usedfor volume flow measurement. Time taken by the engine to consume this volume ismeasured by stopwatch.

**II) Gravimetric method:** In this method the time to consume a given weight of thefuel is measured. Differential pressure transmitters working on hydrostatic headprinciples can be used for fuel consumption measurement.

**3 Measurement of air consumption**

Air box method: In IC engines, as the air flow is pulsating, for satisfactorymeasurement of air consumption an air box of suitable volume is fitted with orifice. The air box is used for damping out the pulsations. The differential pressure acrossthe orifice is measured by manometer and pressure transmitter.

**4 Measurement of brake power**

Measurement of BP involves determination of the torque and angular speed of theengine output shaft. This torque-measuring device is called a dynamometer.

The dynamometers used are of following types:

**I) Rope brake dynamometer:** It consists of a number of turns of rope woundaround the rotating drum attached to the output shaft. One side of the rope isconnected to a spring balance and the other to a loading device. The power isabsorbed in friction between the rope and the drum. The drum therefore requirescooling.

Brake power = $\Pi DN\ (W-S)/60{,}000$ in kW

where D is the brake drum diameter, W is the weight and S is the spring scalereading.

**II) Hydraulic dynamometer:** Hydraulic dynamometer works on the principal ofdissipating the power in fluid friction. It consists of an inner rotating member orimpeller coupled to output shaft of the engine. This impeller rotates in a casing, dueto the centrifugal force developed, tends to revolve with impeller, but is resisted bytorque arm supporting the balance weight. The frictional forces between the impeller and the fluid are measured by the spring-balance fitted on the casing. Heatdeveloped due to dissipation of power is carried away by a continuous supply of theworking fluid usually water. The output (power absorbed) can be controlled byvarying the quantity of water circulating in the vortex of the rotor and statorelements. This is achieved by a moving sluice gate in the dynamometer casing.

**III) Eddy current dynamometer:** It consists of a stator on which are fitted anumber of electromagnets and a rotor disc and coupled to the output shaft of theengine. When rotor rotates eddy currents are produced in the stator due to magneticflux set up by the passage of field current in the electromagnets. These eddycurrents oppose the rotor motion, thus loading the engine. These eddy currents aredissipated in producing heat so that this type of dynamometer needs coolingarrangement. A moment arm measures the torque. Regulating the current inelectromagnets controls the load.

While using with variable speed engines sometimes in certain speed zone the dynamometer operating line are nearly parallel with engine operating lines which result in poor stability.

**5 Measurement of indicated power**

There are two methods of finding the IHP of an engine.

**I) Indicator diagram:** A dynamic pressure sensor (piezo sensor) is fitted in the cylinder head to sense combustion pressure. A rotary encoder is fitted on the engine 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.

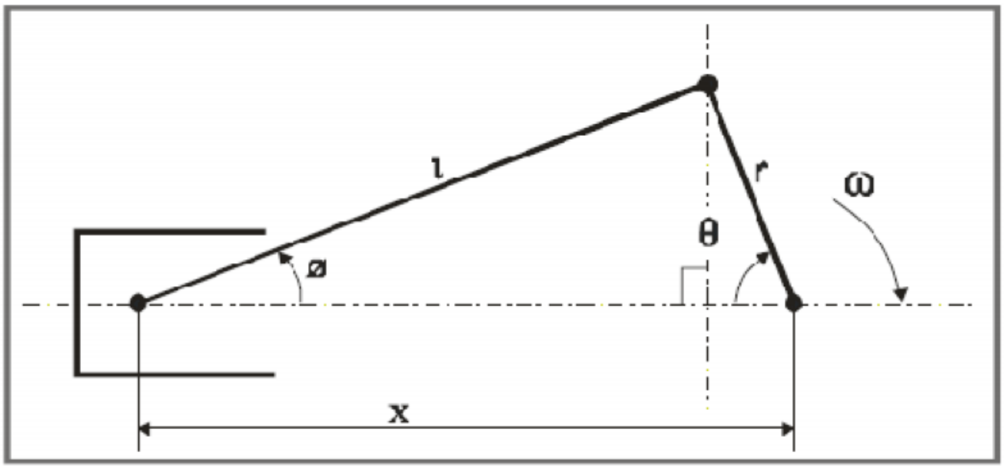



**Fig 4.4 Conversion of pressure crank-angle plot to pressure volume plot**

The figure shows crank-slider mechanism. The piston pin position is given by

$x = r \cos\theta + l \cos\varphi$

From figure $r \sin\theta = l \sin\varphi$ and recalling $\cos\varphi = (1 - \sin^2\varphi)^{1/2}$

$$x = \mathrm{r}\left(\cos\theta + \frac{\mathrm{l}}{\mathrm{r}}\sqrt{\left\{1 - \left(\frac{r}{l}\right)^2 \sin^2\theta\right\}}\right)$$

The binomial theorem can be used to expand the square root term:

$$x = \mathrm{r}\left\{\cos\theta + \frac{\mathrm{l}}{\mathrm{r}}\left|1 - \frac{1}{2}\left(\frac{r}{l}\right)^2 sin^2\theta - \frac{1}{8}\left(\frac{r}{l}\right)^4 sin^4\theta + \cdots\right|\right\} \ldots\ldots\ldots\ldots\ldots\ldots \quad (4.9)$$

The powers of sin θ can be expressed as equivalent multiple angles:

$\sin^2\theta = 1/2 - 1/2\cos 2\theta$

$\sin^4\theta = 3/8 - 1/2\cos 2\theta + 1/8\cos 4\theta$ ………………………………… (4.10)

Substituting the results from equation 4.10 in to equation 4.9 gives

$$x = \mathrm{r}\{\cos\theta + \frac{\mathrm{l}}{\mathrm{r}}\left|1 - \frac{1}{2}\left(\frac{r}{l}\right)^2\left(\frac{1}{2} - \frac{1}{2}cos2\theta\right) - \frac{1}{8}\left(\frac{r}{i}\right)^4\left(\frac{3}{8} - \frac{1}{2}cos2\theta + \frac{1}{8}cos4\theta\right|\right\}$$

The geometry of the engine is such that $\left(\frac{r}{l}\right)^2$ is invariably less than 0.1, in which case it is acceptable to neglect the $\left(\frac{r}{l}\right)^4$ terms, as inspection of above equation shows that these terms will be at least an order of magnitude smaller than $\left(\frac{r}{l}\right)^2$ terms.

The approximate position of piston pin end is thus:

$x = \mathrm{r}\{\cos\theta + \frac{\mathrm{l}}{\mathrm{r}}\left|1 - \frac{1}{2}\left(\frac{r}{l}\right)^2\left(\frac{1}{2} - \frac{1}{2}cos2\theta\right)\right|\}$…………………………………… (4.11)

Where *r* =crankshaft throw and *l* = connecting rod length.

Calculate *x* using above equation; then (*l* + *r* − *x*) shall give distance traversed by piston from its top most position at any angle θ

4.2.6<u>Calculations</u>

- Brake power (kw):

$$\mathrm{BP} = \frac{2\Pi\mathrm{NT}}{60 * 1000}$$

$$BP = \frac{2\Pi N(W * R)}{60 * 1000}$$

$$BP = \frac{0.785xRPMx(Wx9.81)xArmlength}{60*1000} \quad \text{……………………. (4.12)}$$

$$BHP = \frac{T * N}{60 * 75}$$

- Brake mean effective pressure (bar):

$$BMEP = \frac{BP*60}{\frac{\Pi}{4}*D^2*L*\left(\frac{N}{n}\right)*\text{No of cycle}*100} \quad \text{……………………. (4.13)}$$

n = 2 for 4 stroke

n = 1 for 2 stroke

- Indicated power (kw):

From PV diagram

X scale (volume) 1cm =..$m^3$

Y scale (pressure) 1cm =..bar

Area of PV diagram =..$cm^2$

- Work done/cycle/cyl=Area of PVdiagram×Xscalefactor×Yscalefactor×100000

$$IP = \frac{\frac{(\frac{w.d.}{cycle})}{cylinder} * (N/n) * NO\ of\ cycles}{60*1000} \quad \text{……………………. (4.14)}$$

➢ Indicated mean effective pressure (bar):

$$IMEP = \frac{IP*60}{\frac{\Pi}{4}*D^2*L*\left(\frac{N}{n}\right)*No\ of\ cycle*100} \quad \text{……………………. (4.15)}$$

➢ Frictional power (kw):

FP = IP − BP

FHP = IHP − BHP

BHP = IHP – FHP

➢ Brake specific fuel consumption (Kg/kwh):

$$BSFC = \frac{\text{Fuel flow in kg/hr}}{BP} \quad \text{……………………. (4.16)}$$

➢ Brake Thermal Efficiency (%):

$$BTh\ Eff = \frac{BP*3600*100}{Fuel\ flow\ in \frac{kg}{hr}*CV} \quad \text{……………………. (4.17)}$$

- Indicated Thermal Efficiency (%):

$$ITh\ Eff = \frac{IP*3600*100}{Fuel\ flow\ in\frac{kg}{hr}*CV} \text{........................ (4.18)}$$

- Mechanical Efficiency (%):

$$Mech\ Eff = \frac{BP*100}{IP} \text{........................ (4.19)}$$

- Air flow (Kg/hr):

$$Air\ Flow\left(\frac{kg}{hr}\right) = C_d * \frac{\pi}{4} * d^2 * \sqrt{2 * g * h * Wden/Aden} * A_{den} * 3600 \text{.......... (4.20)}$$

- Volumetric Efficiency (%):

$$Vol\ Eff = \frac{\text{Air flow} * 100}{Theorotical\ air\ flow}$$

$$\text{Vol Eff} = \frac{IP*60\text{Air flow}*100}{\frac{\Pi}{4}*D^2*L*\left(\frac{N}{n}\right)*\text{No of cycle}*Aden} \text{........................ (4.21)}$$

➢ Air fuel ratio:

$$A/F = \frac{\text{Air flow}}{Fuel\ flow}$$

➢ Heat Balance (KJ/h):

- Heat Supplied by Fuel = Fuel Flow ×CV
- Heat Equivalent To Useful Work = BP×3600

$$\text{Heat Equivalent To Useful Work In\%} = \frac{\text{Heat Equivalent To Useful Work} * 100}{\text{Heat Supplied By Fuel}}$$

- HeatInJacketCoolingWater= F3 *Cp *W* (T 2- T1)

$$\text{Heat In Jacket Cooling WaterIn\%} = \frac{\text{Heat in jacket cooling water} * 100}{\text{HeatSuppliedByFuel}} \ldots\ldots\ldots\ldots\ldots \text{(4.22)}$$

Heat in Exhaust (Calculate CPex value):

$$\text{CPex} = \frac{F4 * Cp * W * (T4 - T3)}{(F1 + F2) * (T5 - T6)} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(4.23)}$$

Where,

Cpex Specific heat of exhaust gas in kJ/kg$^0$K

Cpw Specific heat of water kJ/kg$^0$K

F1 Fuel consumption kg/hr

F2 Air consumption kg/hr

F4 Calorimeter water flow kg/hr

T3 Calorimeter water inlet temperature $^0$K

T4 Calorimeter water outlet temperature $^0$K

T5 Exhaust gas to calorimeter inlet temp.$^0$K

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

Heat In Exhaust (KJ / h=) (F1 +F2)* Cp ex* (T3 -Tamb)…………………… (4.24)

- Heat to radiation and unaccounted (%)=Heat Supplied By Fuel (100%)-{ Heat Equivalent To Useful Work(%)+Heat In Jacket Cooling Water (%)+Heat To Exhaust(%)}

### 4.3 Experiment

**Aim:**Study of engine performance and exhaust gas emission (Computerized mode)

**Object**

To study the performance and exhaust of 4 cylinder, 4 stroke, Petrol engine connected to eddy current dynamometer in computerized mode.

**Procedure**

- Ensure cooling water circulation for eddy current dynamometer, piezo sensor,engine cooling and calorimeter.
- Start the set up and run the engine at no load for 4-5 minutes.
- Switch on the computer and run “Enginesoft”. Confirm that the Enginesoftconfiguration data is as given below.
- Gradually increase throttle to full open condition and load the enginesimultaneously maintaining engine speed at @ 5000 RPM.
- Wait for steady state (for @ 3 minutes) and log the data in the “Enginesoft”.
- Graduallyincrease the load to decrease the speed in steps of @500 RPM up to@ 2000 rpm maximum and repeat the data logging for each observation.

- Note the reading of Exhaust Gas using Exhaust Gas Analyser at exhaust.
- View the results and performance plots in “Enginesoft”.

**<u>Enginesoft Configuration data</u>**

**Set up constants:**

| | |
|---|---|
| No of PO cycles | 1 |
| Cylinder pressure plot ref | 2010 |
| Fuel read time | 60 sec |
| Fuel factor | 0.096 kg/Volt |
| Orifice diameter | 40 mm |
| Dynamometer arm length | 210 mm |

**Table 4.3 Set up Constant**

**Engine and set up details:**

| | |
|---|---|
| Engine power | 47.7 Kw |
| Engine max speed | 6200 RPM |
| Cylinder bore | 68.5mm |
| Stroke length | 72mm |
| Connecting rod length | 112.5 mm |
| Compression ratio | 9.2:1 |
| Stroke type | Four |
| No. of cylinders | Four |
| Speed type | Variable |
| Cooling type | Water |

| | |
|---|---|
| Dynamometer type | Eddy current |
| Indicator used type | Cylinder pressure |
| Interface type used | PCI-1050 |
| Calorimeter used | Pipe in pipe |

**Table 4.4 Set up details**

**Theoretical constants:**

| | |
|---|---|
| Fuel density | 740 kg/m^3 |
| Calorific value | 44000 kJ/kg |
| Orifice coefficient of discharge | 0.60 |
| Sp heat of exhaust gas | 1.00 kJ/kg-K |
| Max sp heat of exhaust gas | 1.25 kJ/kg-K |
| Min sp heat of exhaust gas | 1.00 kJ/kg-K |
| Specific heat of water | 4.186 kJ/kg-K |
| Water density | 1000 kg/m^3 |
| Ambient temperature | 300C |

**Table 4.5 Theoretical Constants**

**Sensor range**

| | |
|---|---|
| Exhaust gas temp. trans. (Engine) | 0-1200 C |
| Air flow transmitter | -200-0 mm WC |
| Fuel flow DP transmitter | 0-500 mm WC |
| Load cell | 0-50 kg |
| Cylinder pressure transducer | 0-345.5 bar |

**Table 4.6 Sensor Range**

## CHAPTER 5

# RESULTS and DISCUSSION

Gasoline Blendshaving 5%,10%,15% and 20% Ethanol is made and Exhaust emissions, Specific fuel consumption, Brake thermal efficiency and mechanical efficiency curves are plotted firstly at no load and then at constant rpm of 3000 and 4000.Pressure-Crank angle curves are also plotted for the same.

The density and Lower calorific value of blends are first calculated and then fed in the software set up configuration to get the desired results. The results obtained were noted and then curves were plotted as shown below to have a clear understanding of the variations of different parameters by using different blends.

HC exhaust was plotted in parts per million, $O_2$,CO,$CO_2$ were plotted on volume percentage basis. Specific fuel consumption was calculated in Kg/KWhr, Mechanical Efficiency and Indicated Thermal Efficiency were plotted on percentage basis. Lamda is unitless as it just shows the relative air fuel ratio.

## 5.1No Load Test

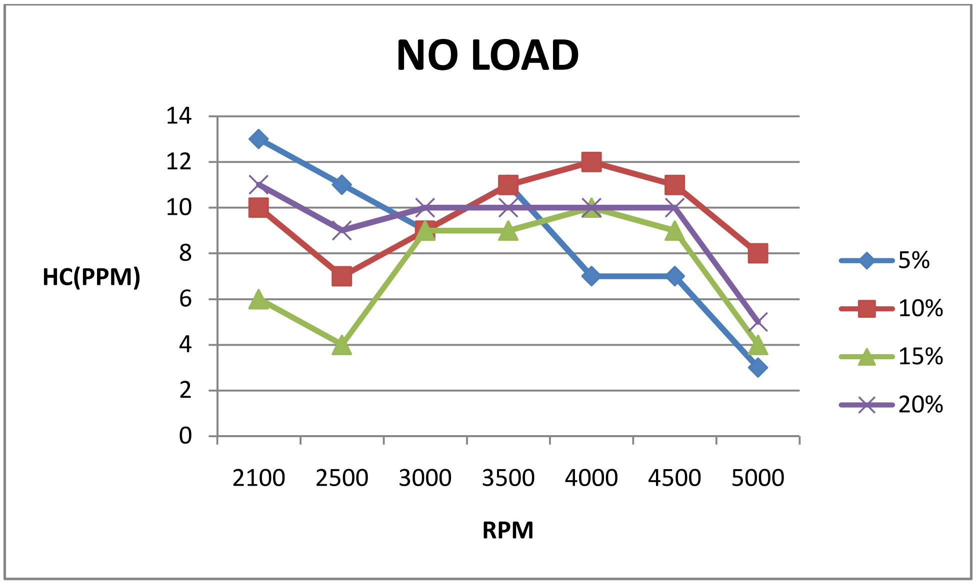



**FIG 5.1 HC exhaust variation with blends at different rpm.**

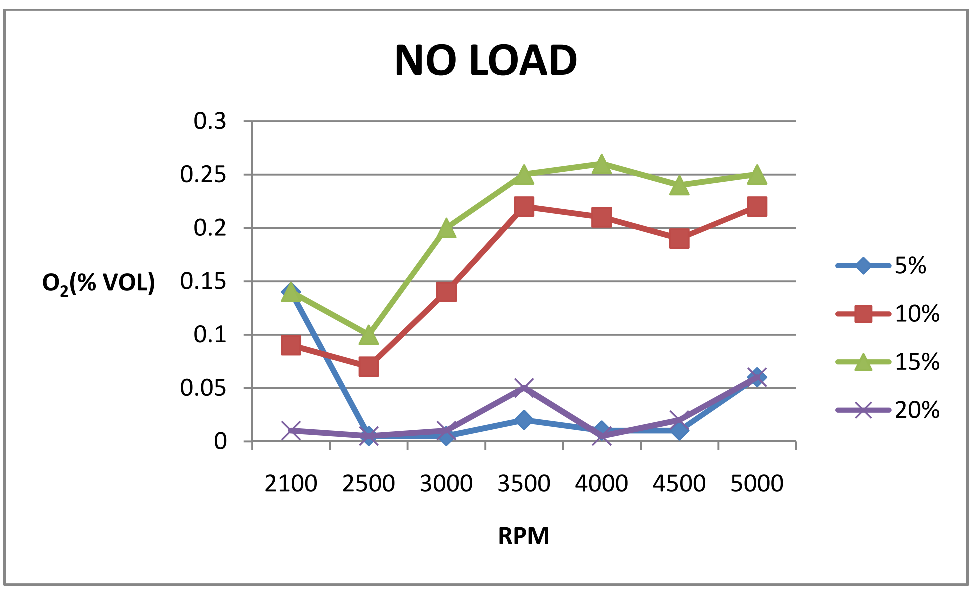



**FIG 5.2 $O_2$ exhaust variation with blends at different rpm.**

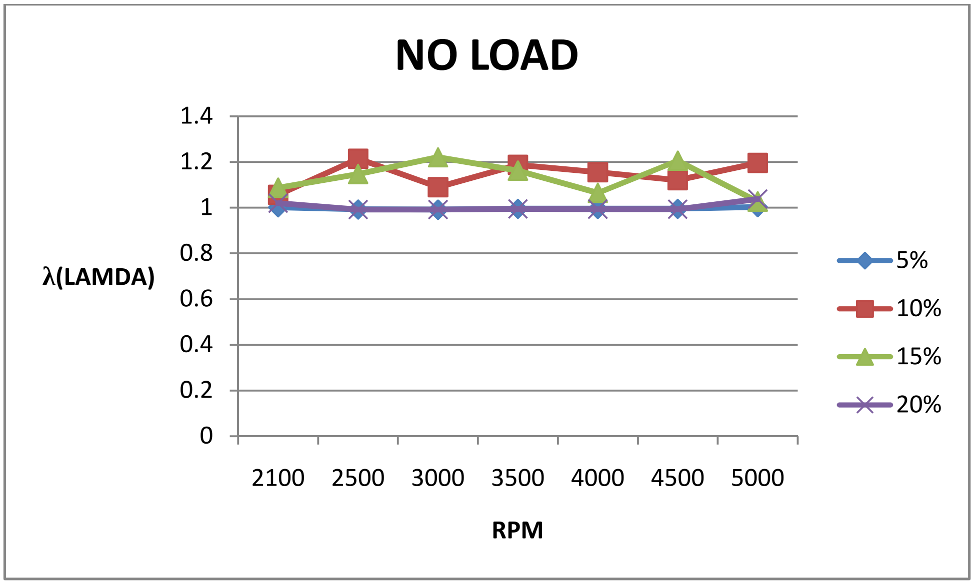



**FIG 5.3 Lamdavariation with blends at different rpm.**

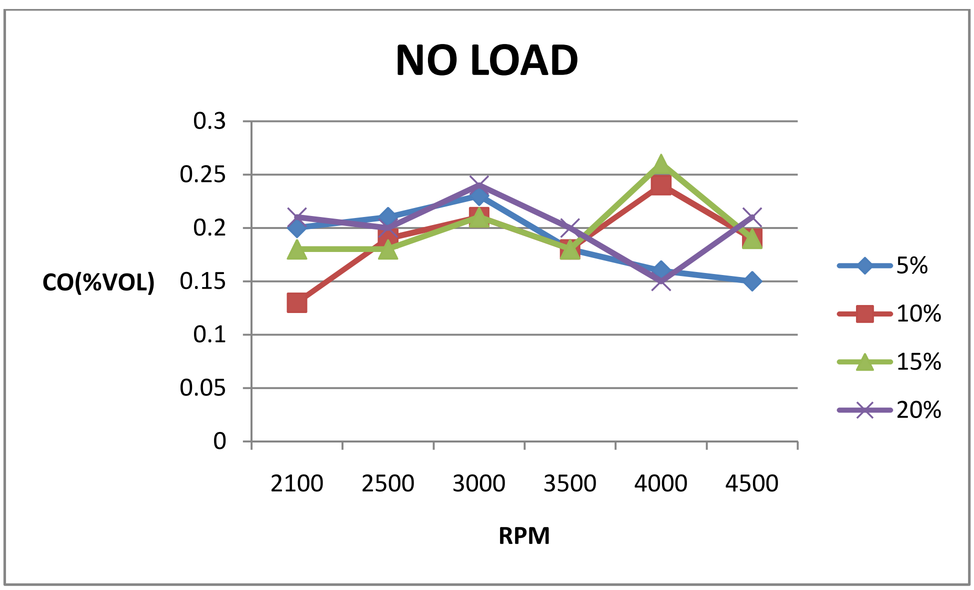



**FIG 5.4 CO variation with blends at different rpm.**

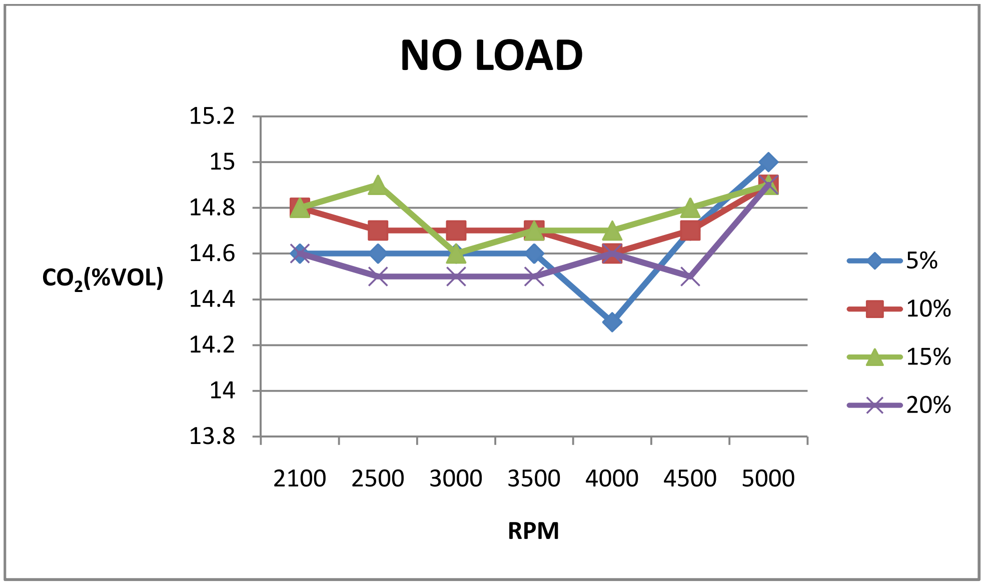



FIG 5.5 $CO_2$variation with blends at different rpm.

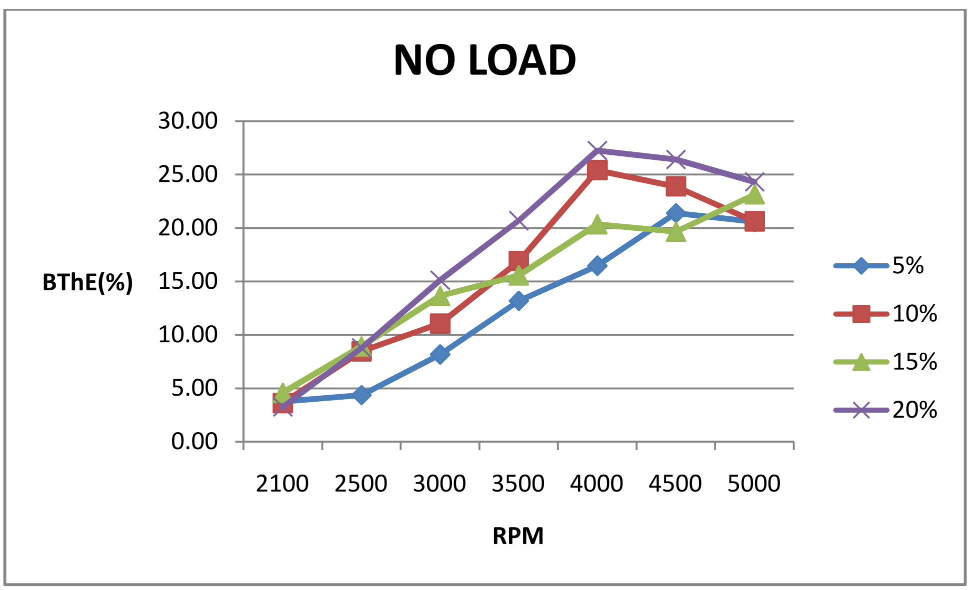



FIG 5.6 Brake Thermal efficiency variation with blends at different rpm.

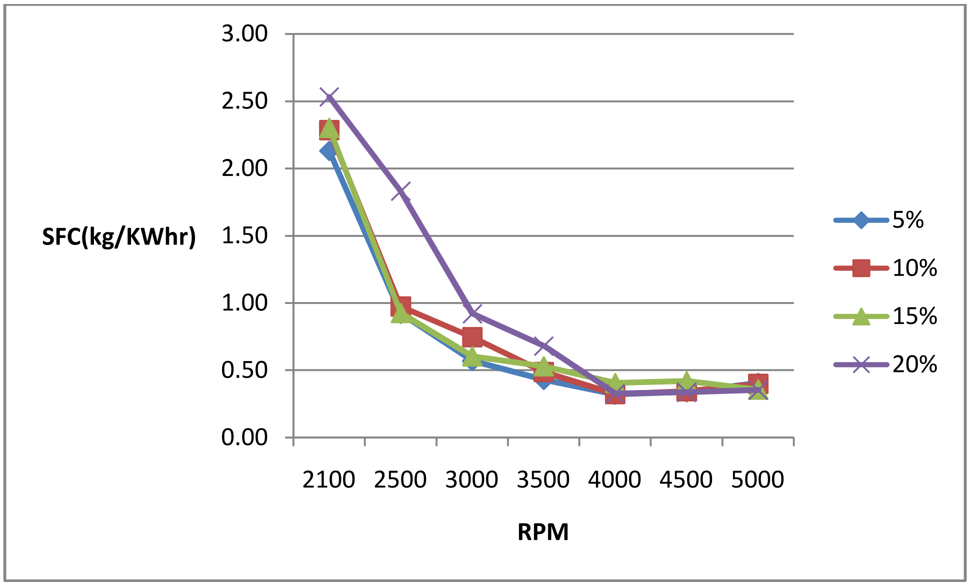



**FIG 5.7 Specific fuel consumption variation with blends at different rpm.**

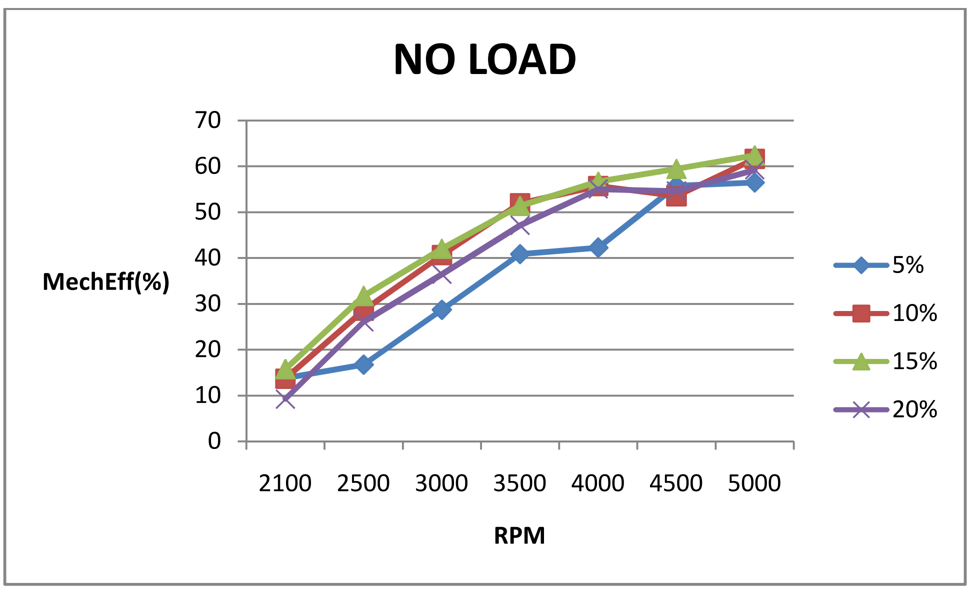



**FIG 5.8 Mechanical efficiency variation with blends at different rpm.**

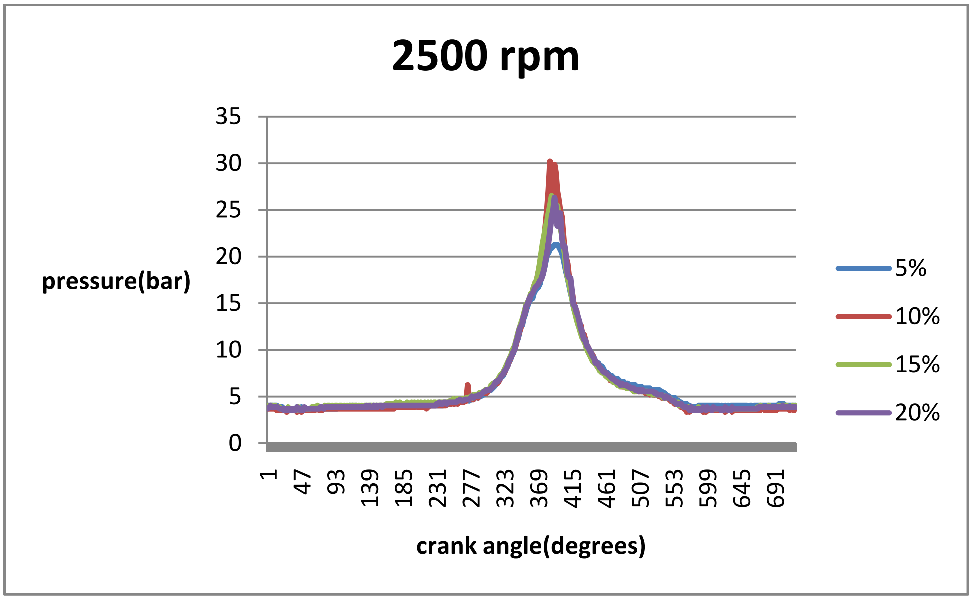



**Fig 5.9 Variation of Pressure with crank angle at 2500 rpm of different blends**

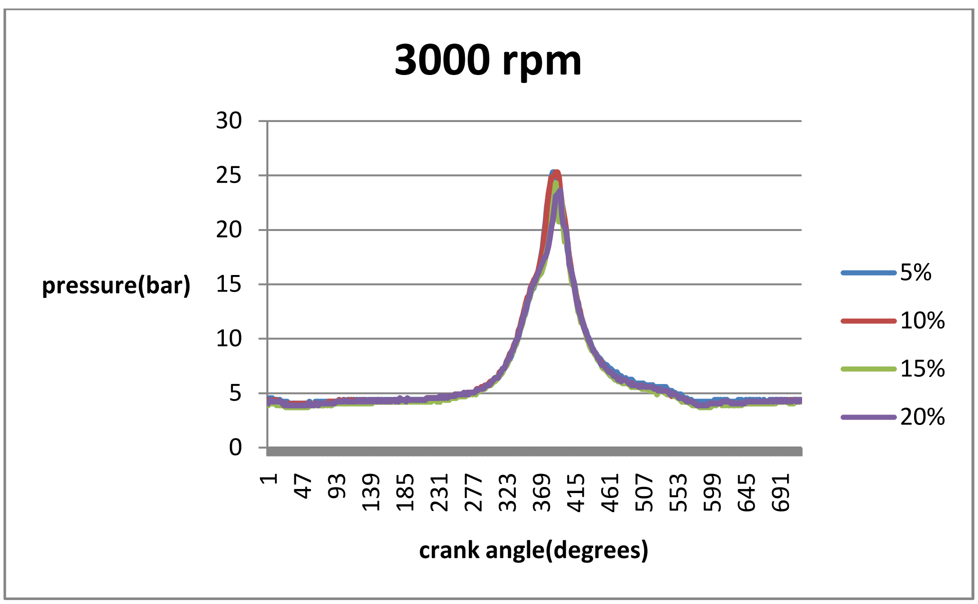



**Fig 5.10 Variation of Pressure with crank angle at 3000 rpm of different blends**

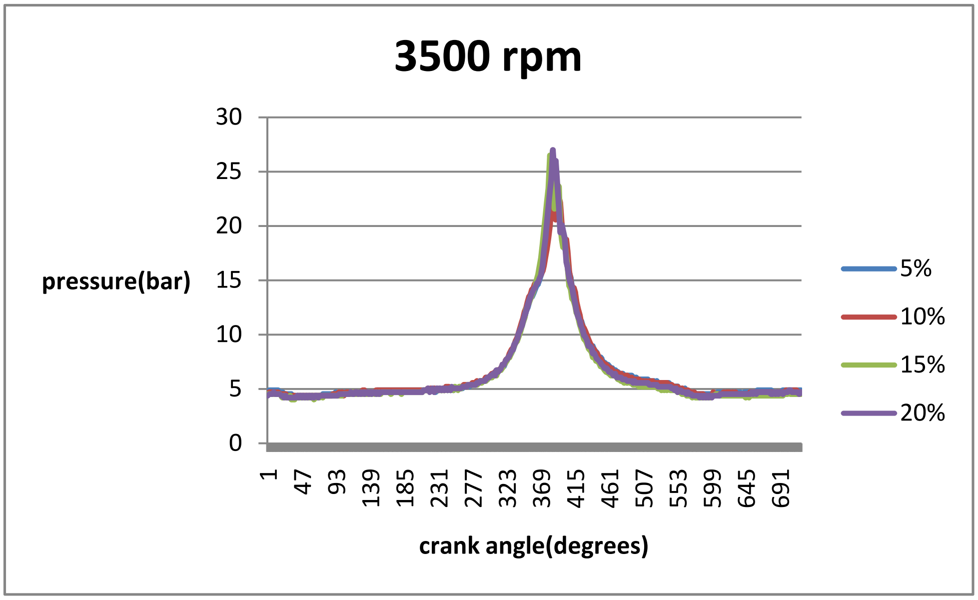



**Fig 5.11 Variation of Pressure with crank angle at 3500 rpm of different blends**

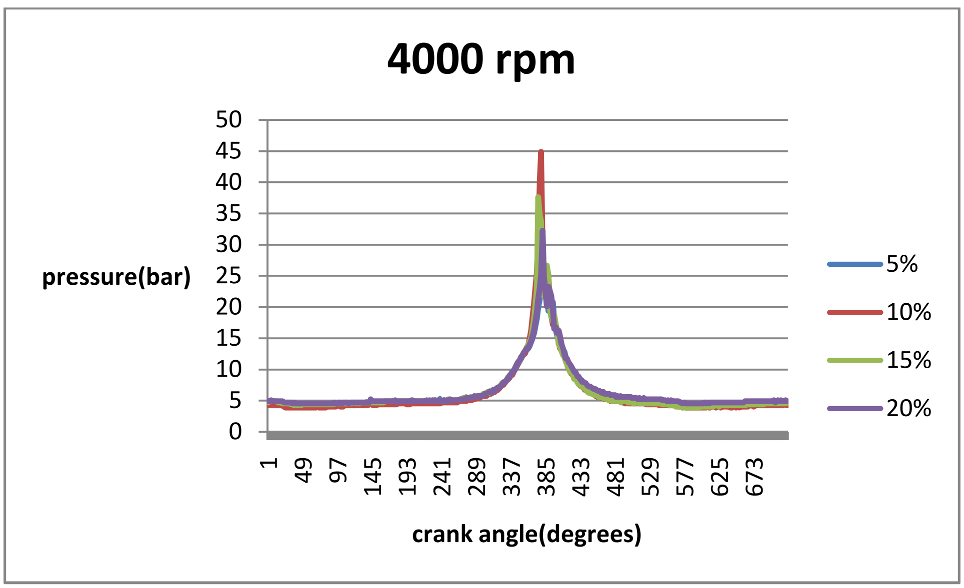



**Fig 5.12 Variation of Pressure with crank angle at 4000 rpm of different blends**

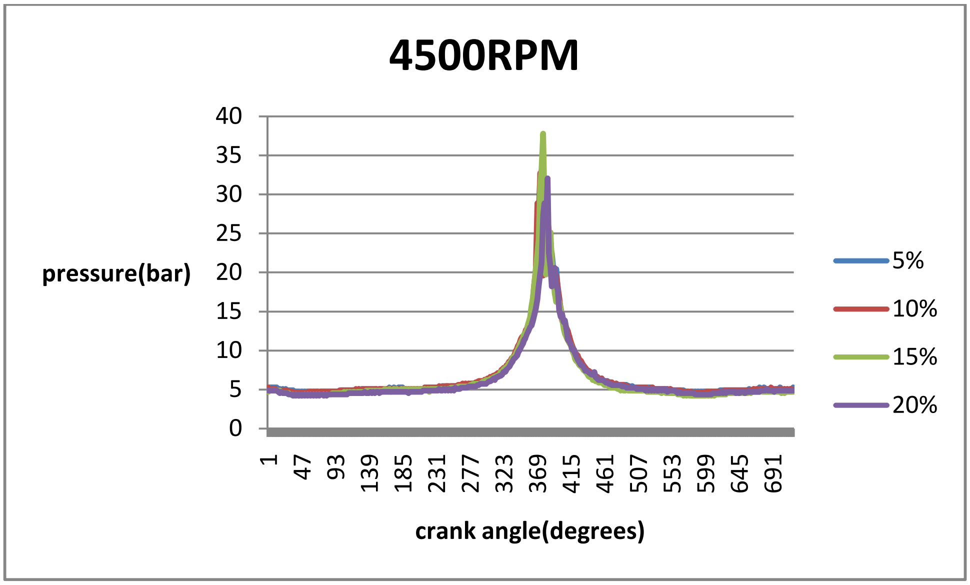



**Fig 5.13 Variation of Pressure with crank angle at 4500 rpm of different blends**

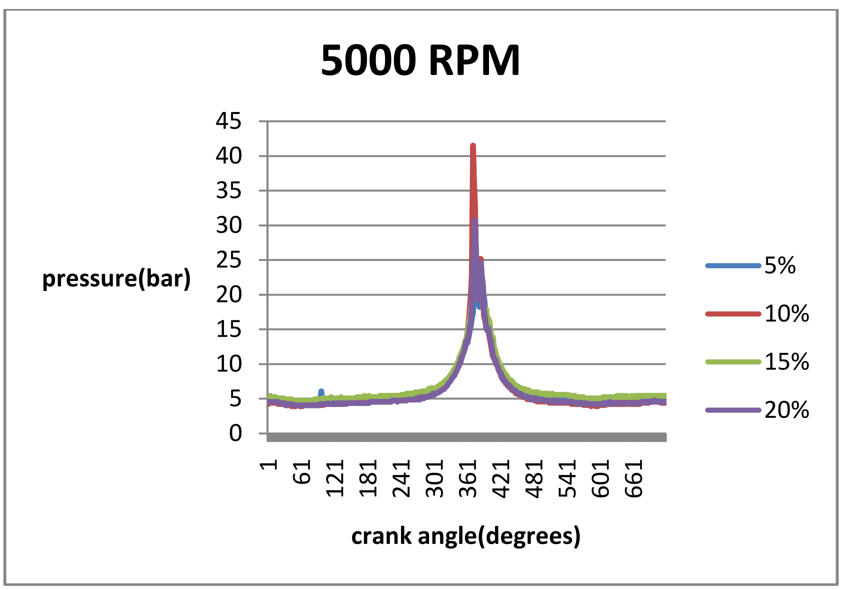



**Fig 5.14 Variation of Pressure with crank angle at 5000 rpm of different blends**

## 5.2 Constant rpm Test

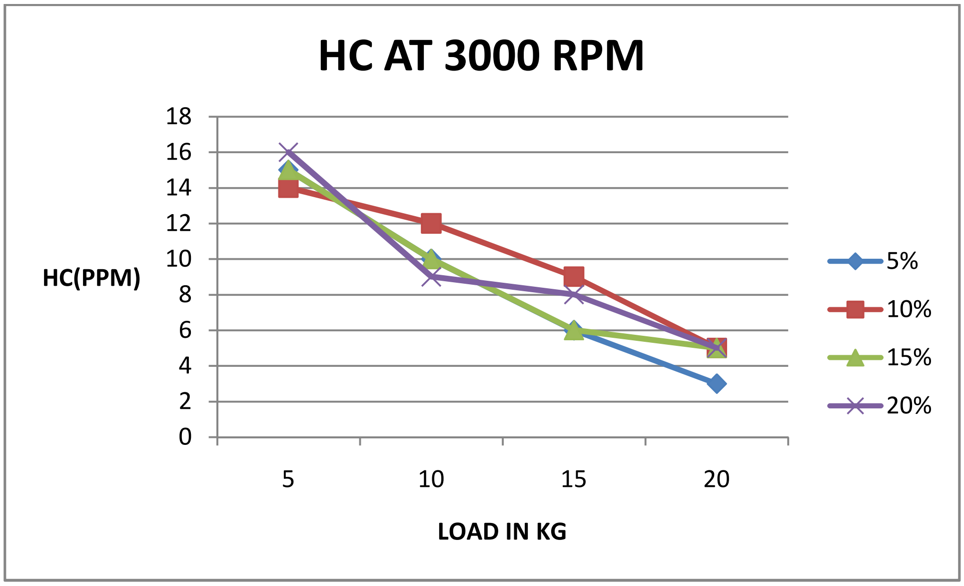



**Fig 5.15 Variation of HC emission with load at 3000 rpm**

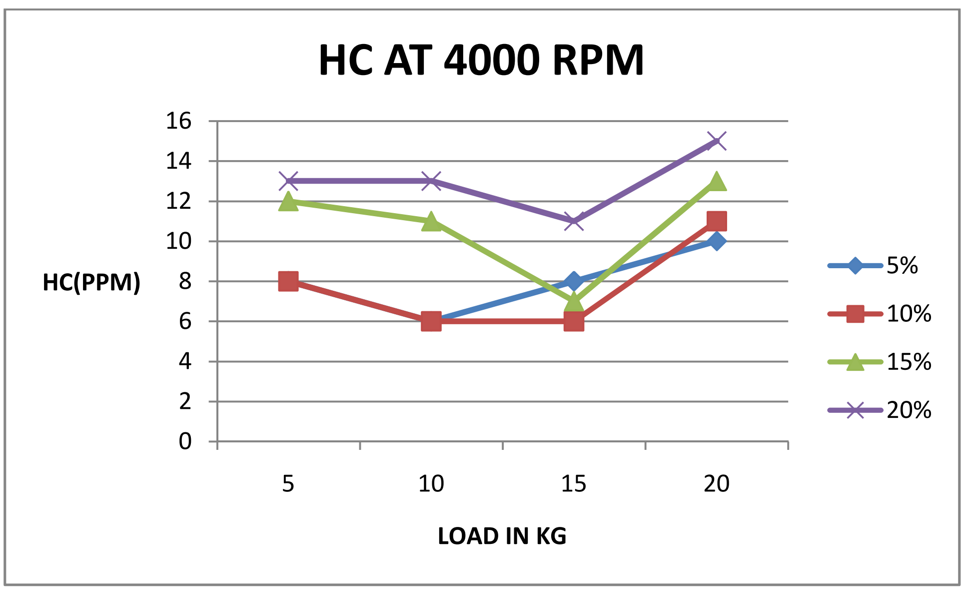



**Fig 5.16 Variation of HC emission with load at 4000 rpm**

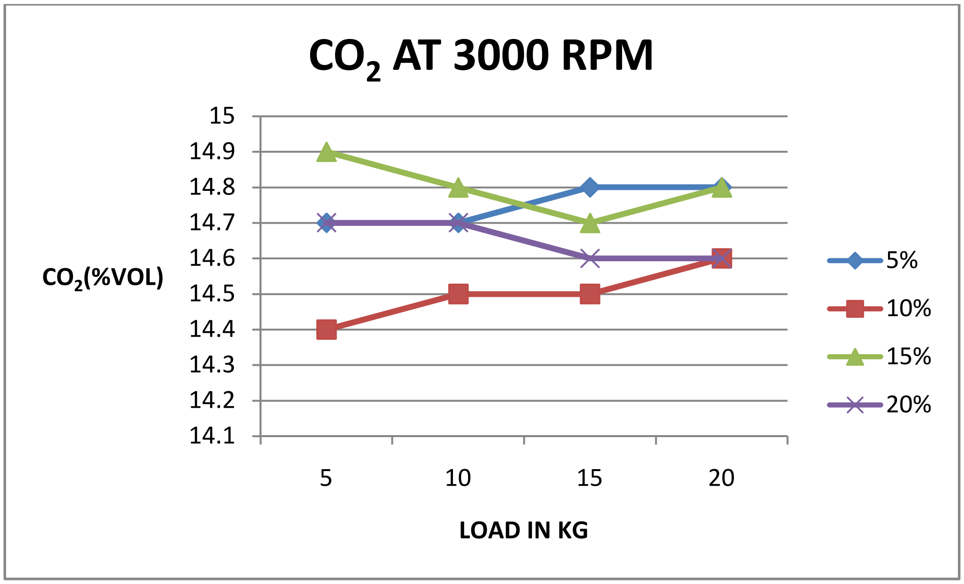



**Fig 5.17 Variation of $CO_2$ emission with load at 3000 rpm**

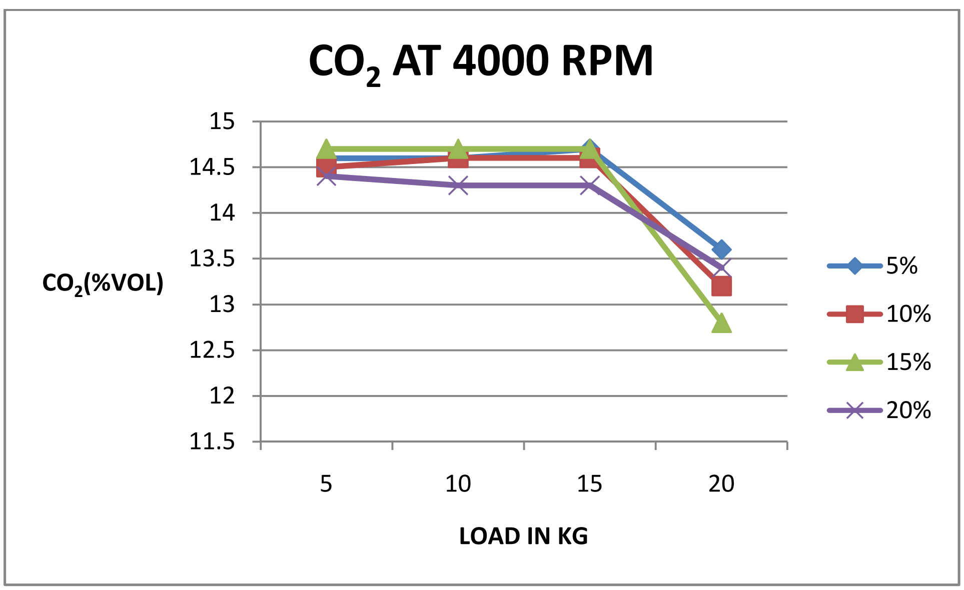



**Fig 5.18 Variation of $CO_2$ emission with load at 4000 rpm**

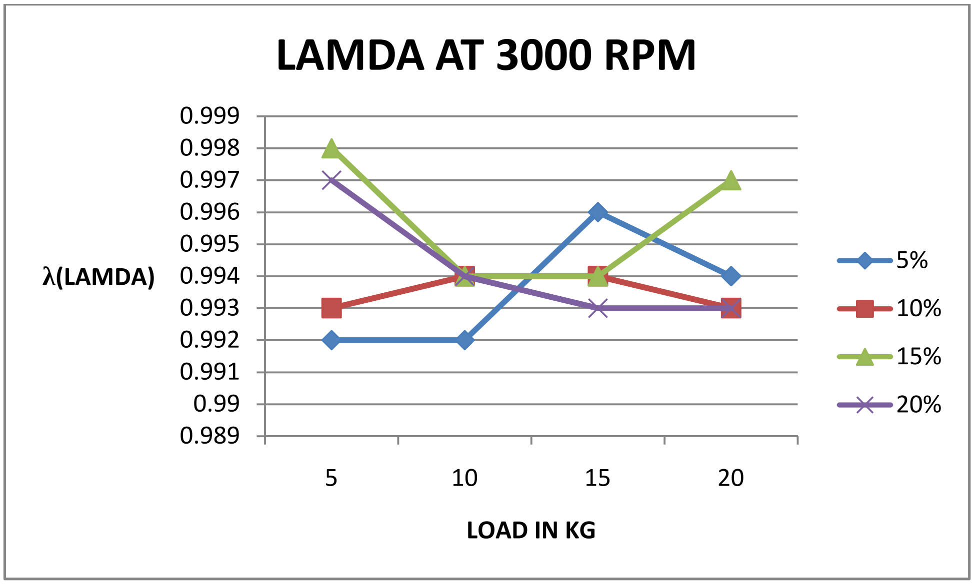



Fig 5.19 Variation of Lamda emission with load at 3000 rpm

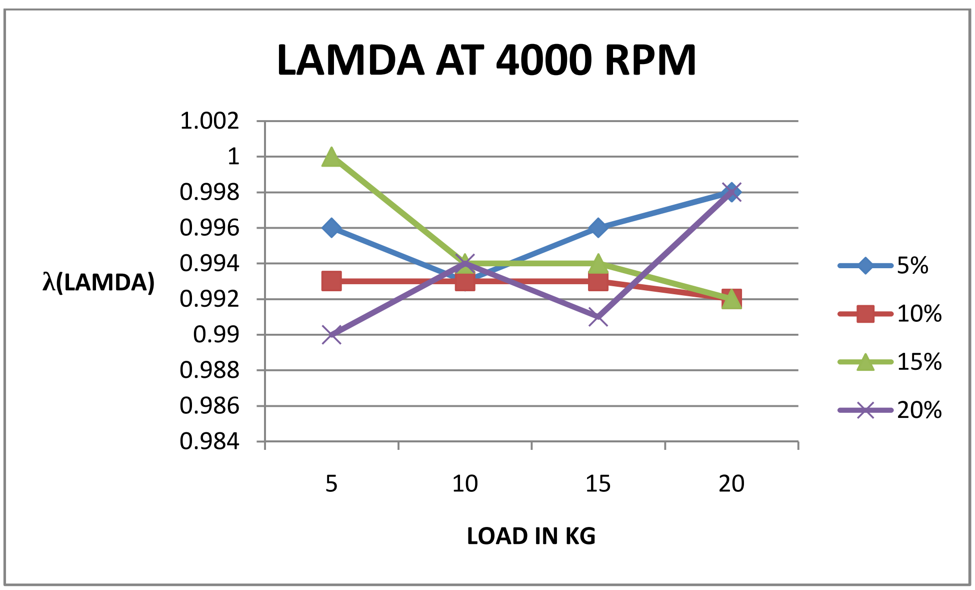



Fig 5.20 Variation of Lamda emission with load at 4000 rpm

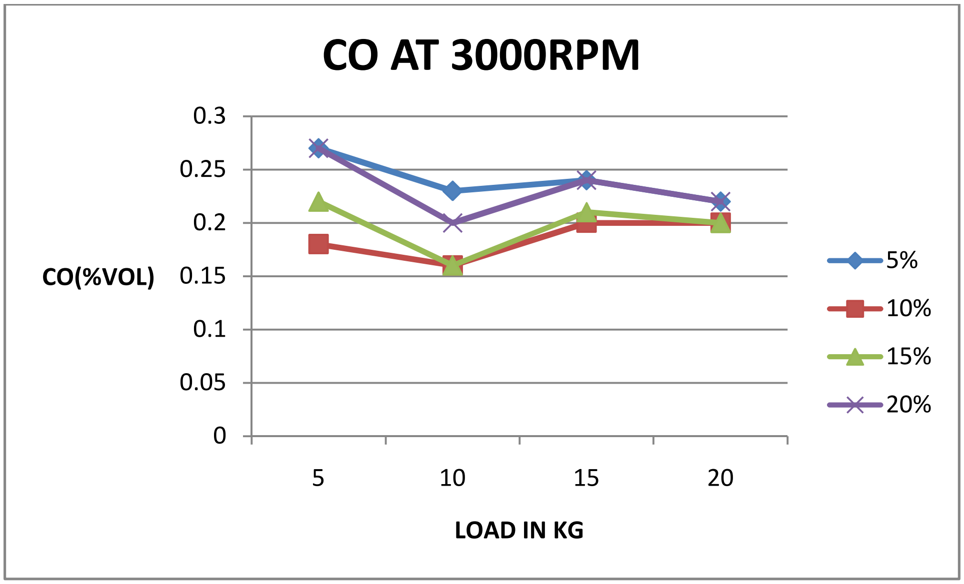



**Fig 5.21 Variation of CO emission with load at 3000 rpm**

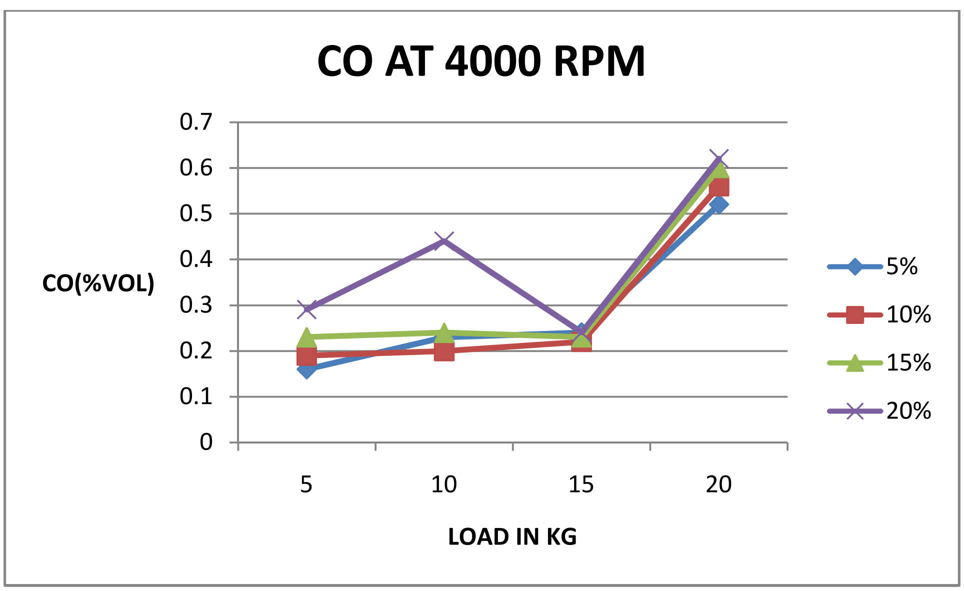



**Fig 5.22 Variation of CO emission with load at 4000 rpm**

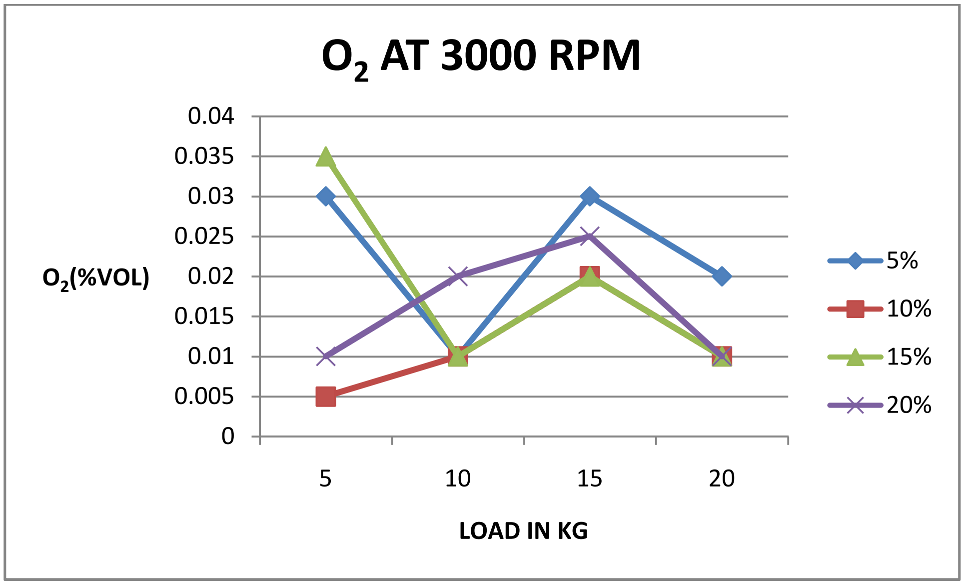



**Fig 5.23 Variation of $O_2$ emission with load at 3000 rpm**

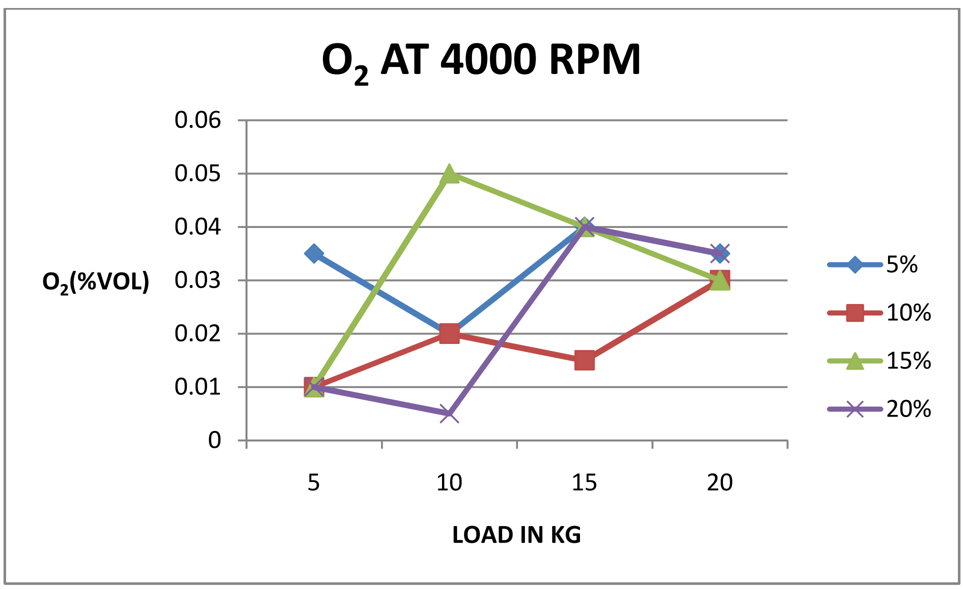



**Fig 5.24 Variation of $O_2$ emission with load at 4000 rpm**

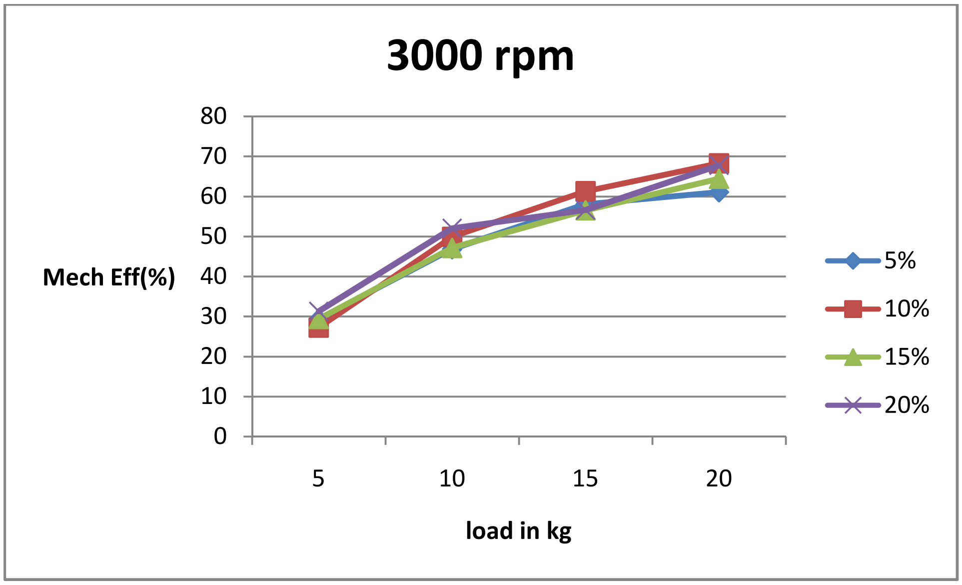



**Fig 5.25 Variation of Mechanical Efficiency with load at 3000 rpm**

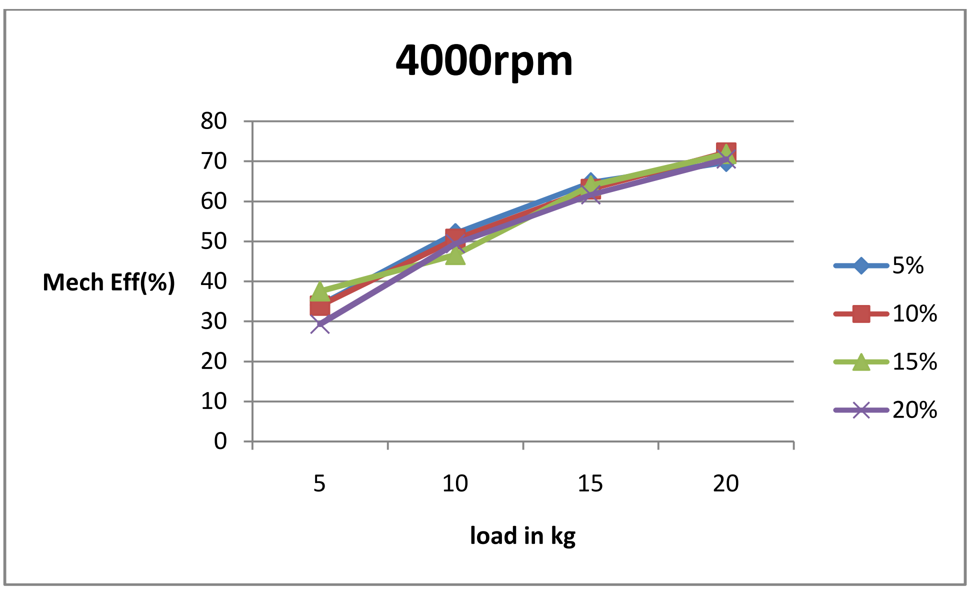



**Fig 5.26 Variation of Mechanical Efficiency with load at 4000 rpm**

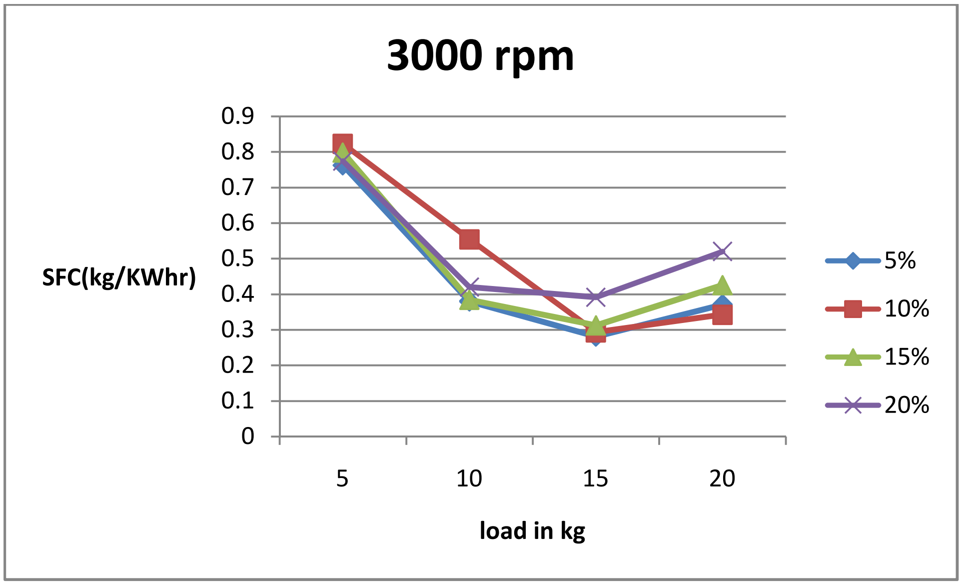



**Fig 5.27 Variation of Specific fuel consumption with load at 3000 rpm**

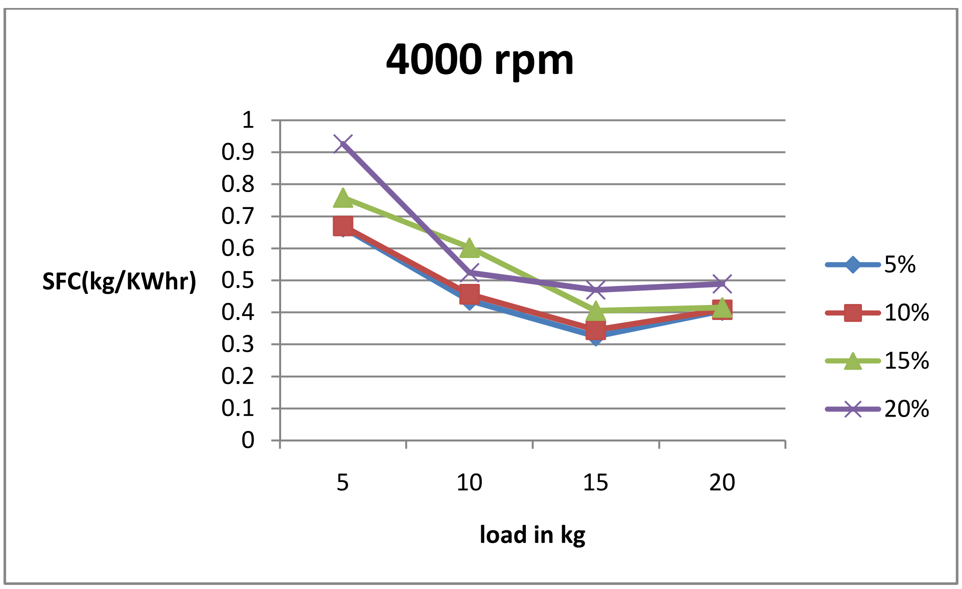



**Fig 5.28 Variation of Specific fuel consumption with load at 4000 rpm**

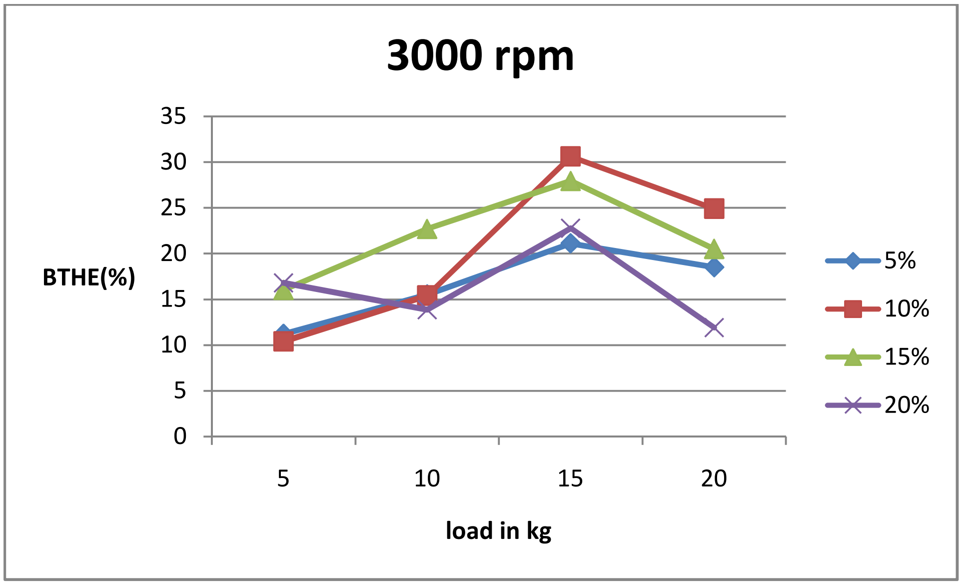



**Fig 5.29 Variation of Brake Thermal Efficiency with load at 3000 rpm**

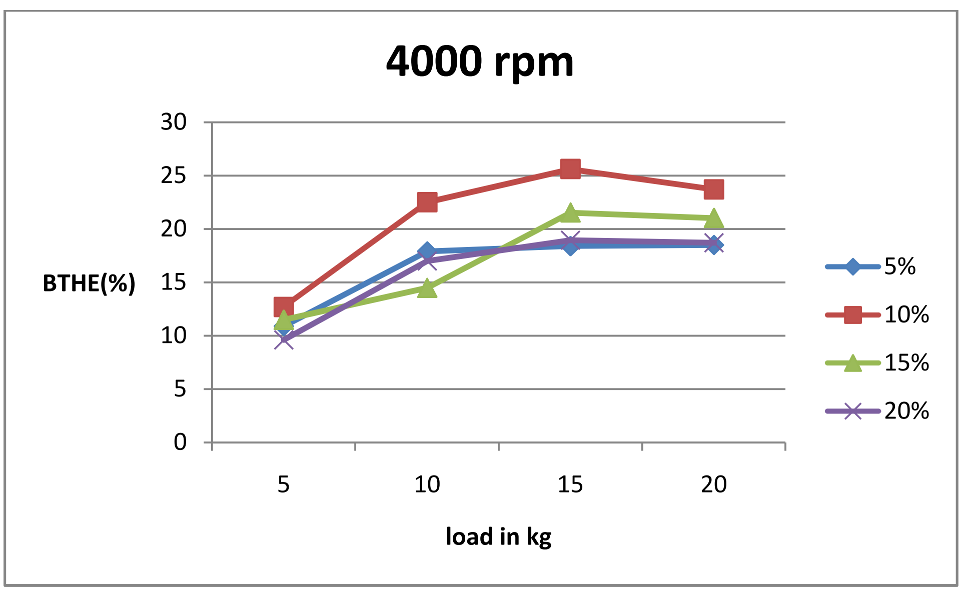



**Fig 5.30 Variation of Brake Thermal Efficiency with load at 4000 rpm**

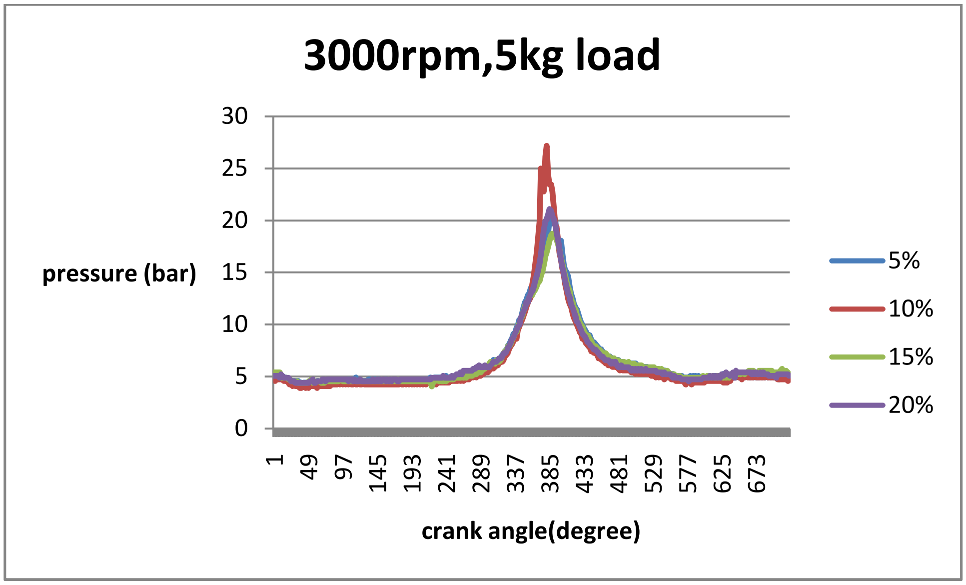



**Fig 5.31 Variation of Pressure with crank angle at 5kg load,3000rpm**

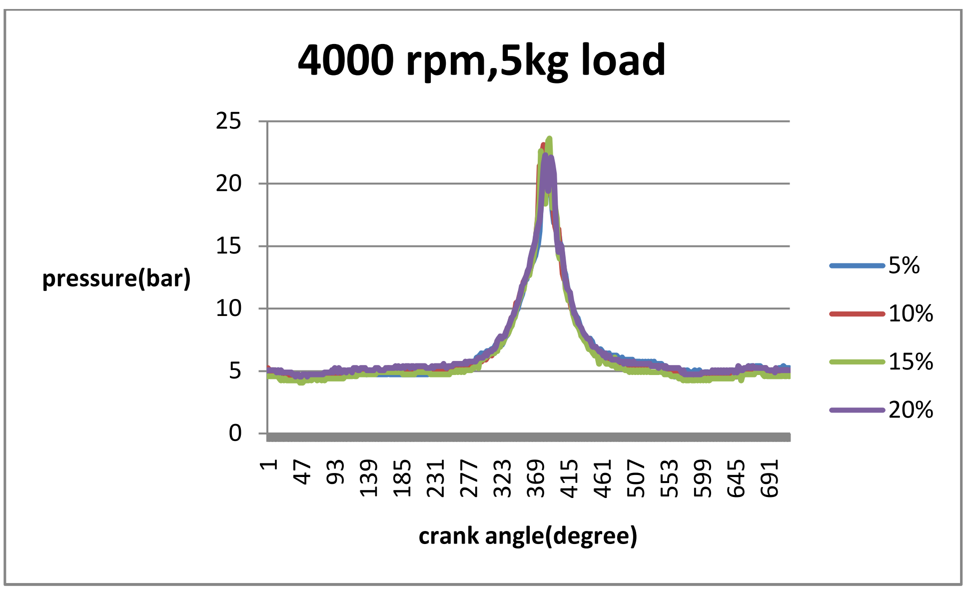



**Fig 5.32 Variation of Pressure with crank angle at 5kg load, 4000rpm**

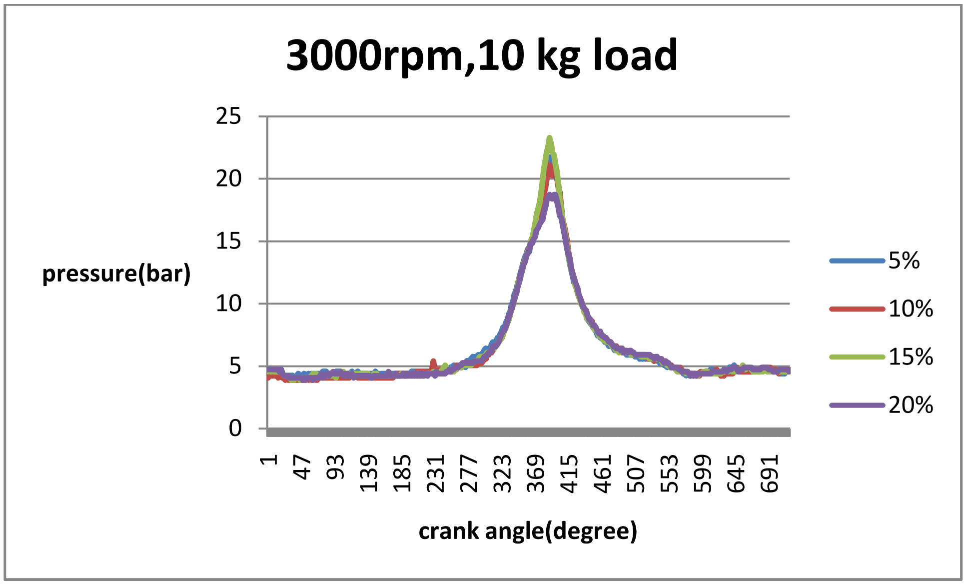



**Fig 5.33 Variation of Pressure with crank angle at 10kg load, 3000rpm**

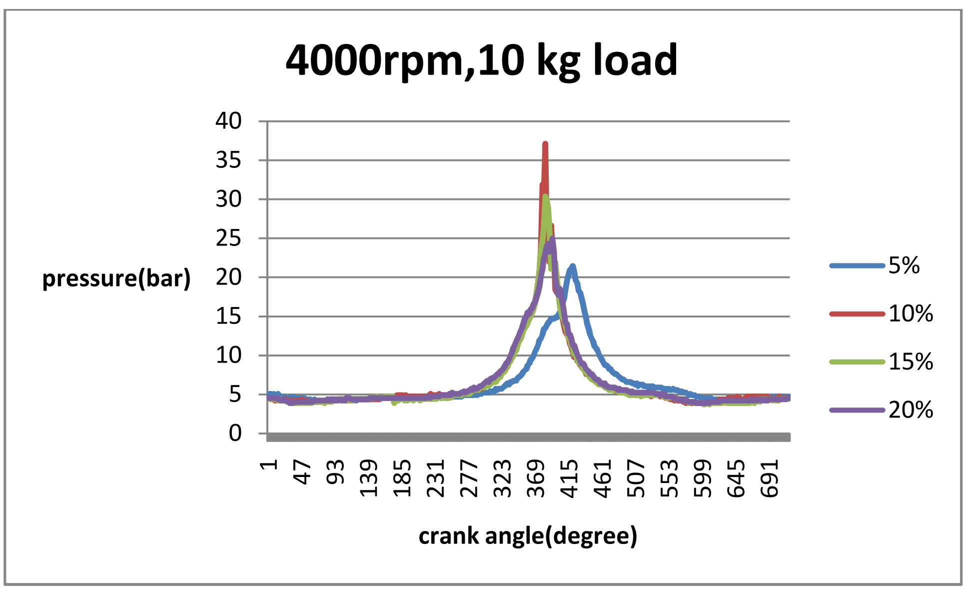



**Fig 5.34 Variation of Pressure with crank angle at 10kg load, 4000rpm**

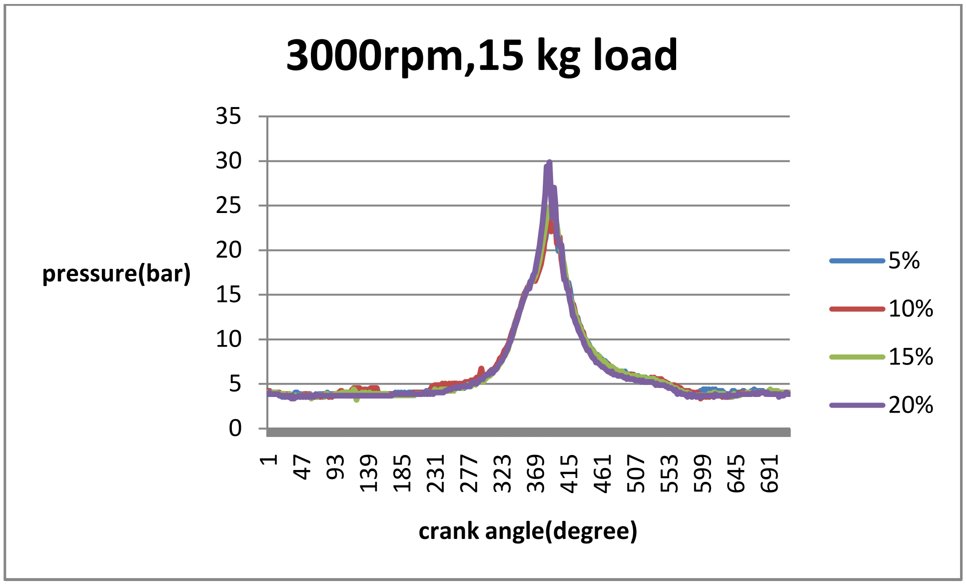



Fig 5.35 Variation of Pressure with crank angle at 15kg load, 3000rpm

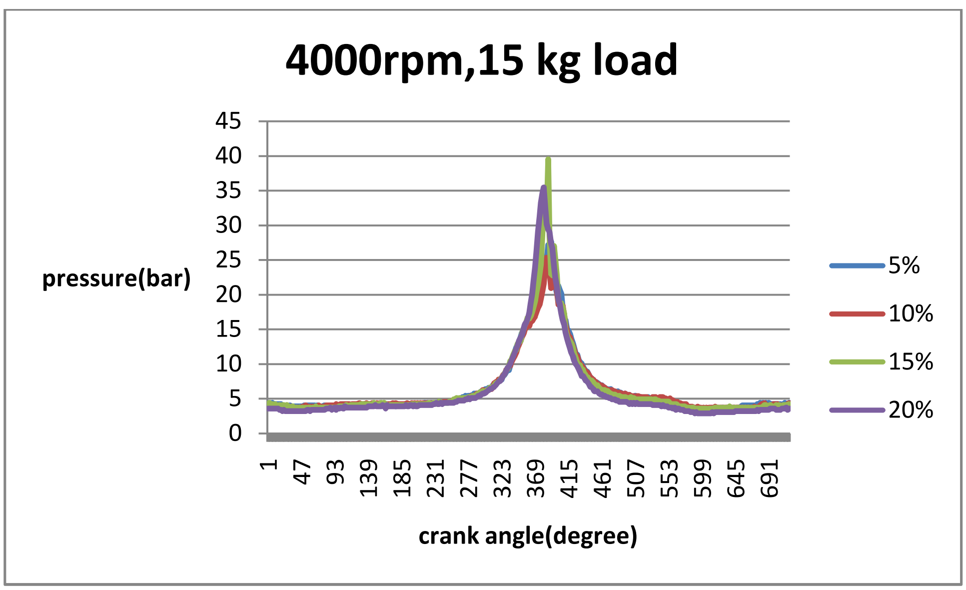



Fig 5.36 Variation of Pressure with crank angle at 15kg load, 4000rpm

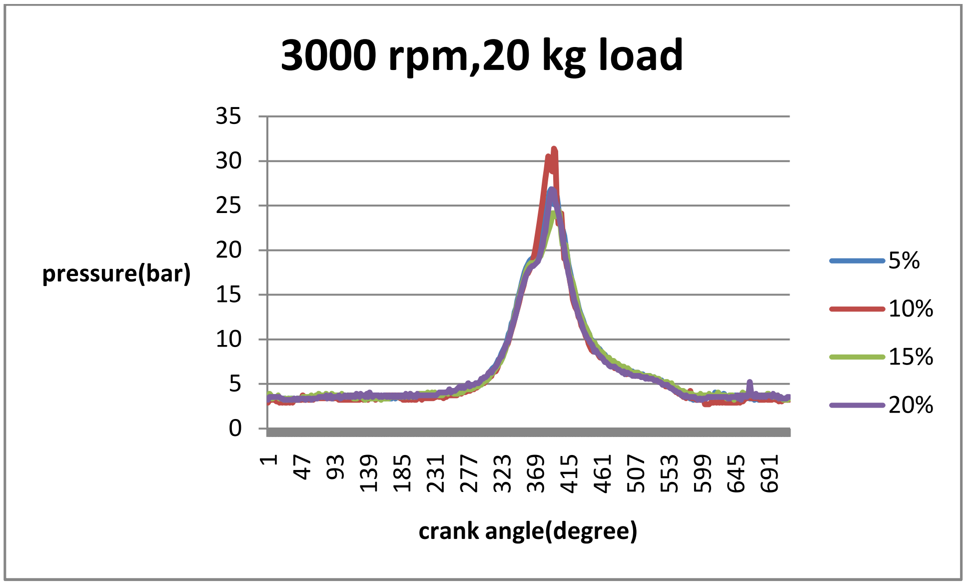



**Fig 5.37 Variation of Pressure with crank angle at 20kg load, 3000rpm**

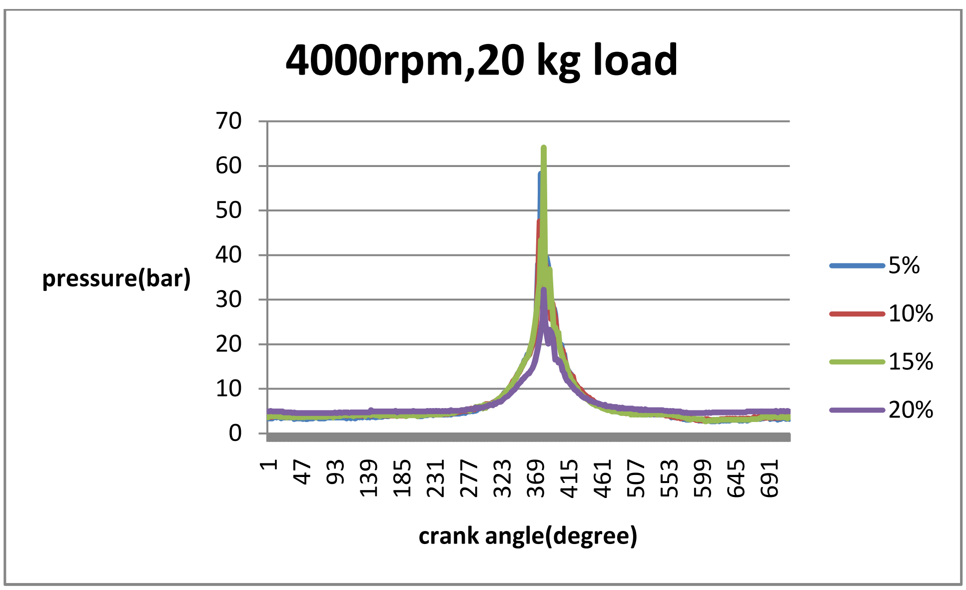



**Fig 5.38 Variation of Pressure with crank angle at 20kg load, 4000rpm**

### 5.3 Discussions

At No Load

- HC emission decreases as blending increases upto 4000 rpm with respect to E5 and is lowest at 2500 rpm. For 10% blend HC emission reduces by 23.08% at 2100 rpm in comparison to commercial Gasoline.
- $O_2$ Percentage increases as blending increases from 5% and is highest between 2500 rpm to 3500 rpm.
- Lamda increases from 1 to 1.2 as blending increased up to 15%.It increases by 22% at 2500 rpm for 10% blend in comparison to commercial Gasoline.
- $CO_2$ increases up to 4000 rpm when blending increased from 5% and is highest at 2500rpm. For 10% blend it increases by 0.68% at 2500 rpm in comparison to commercial Gasoline.
- CO decreases as blending is increased and is lowest at 2100 rpm. For 10% blend, it reduces by 35% in comparison to commercial Gasoline.
- Brake Thermal Efficiency increases on blending.Brake Thermal Efficiency reaches a maximum at around 4500 rpm and then starts decreasing. In comparison to commercial Gasoline it increases by 11.6% for 10% blend, 8.1% for 15%blend and 23.37% for 20%blend at 4500 rpm.
- Specific Fuel Consumption increases on blending Gasoline.In comparison to commercial Gasoline, it increases by 7.2% for 10% blend, 8.0% for 15%blend and 18.77% for 20%blend at 2100 rpm.
- Mechanical Efficiency increases on blending Gasoline. In comparison to commercial Gasoline, it increases by 9% for 10% blend, 8.8% for 15%blend and 4.85% for 20%blend at 5000 rpm.

- The maximum Pressure increases on blending and reaches a maximum of 45 bar at 4000 rpm.

At Constant RPM

- HC emission increases with blending and is more at 3000 rpm compared to 4000 rpm for low loads. At 5kg load, it increases by 6.67% for 10%blend at 3000 rpm and increases by 25% for 10% blend at 4000 rpm with respect to commercial Gasoline.
- $CO_2$ generally decreases with blending and is generally more for 3000 rpm as compared to 4000 rpm. At 5kg load, it decreases by 2.04% for 10%blend at 3000 rpm and decreases by 2.94% for 10% blend at 4000 rpm with respect to commercial Gasoline.
- Lamda decreases on blending at high loads and generally lies between 0.992 to 0.996 for 3000 rpm and 4000 rpm.
- CO is less for 3000 rpm as compared to 4000 rpm.At 5kg load, it decreases by 33.33% for 10% blend at 3000 rpm and increases by 18.75% for 10% blend at 4000 rpm with respect to commercial Gasoline.
- $O_2$ Percentage decreases with blending and is less for 3000 rpm.
- Mechanical Efficiency increases with blending and is slightly greater at 4000 rpm. At 20kg load, it increases by 11.85% for 10%blend, 5.5% for 15% blend, 10.99% for 20% blend at 3000 rpm and increase by 3.36% for 10% blend, 2.89% for 15% blend and 1.03%for 20% blend at 4000 rpm with respect to commercial Gasoline.

- Specific Fuel Consumption increases on blending and is generally lower for 4000 rpm.At 20kg load, it increases by 5.66% for 10%blend, 14.55% for 15% blend, 40.16% for 20% blend at 3000 rpm and increase by 0.75% for 10% blend, 2.47% for 15% blend and 20.47 % for 20% blend at 4000 rpm with respect to commercial Gasoline.
- The maximum Pressure increases on blending and is greaterat 4000 rpm as compares to 3000 rpm for high loads.
- Brake Thermal Efficiency increases on blending.It reaches a maximum at 15 kg load and is generally higher for 3000 rpm than 4000rpm.At 20kg load, it increases by 45% for 10%blend, 32.2% for 15% blend, 7.91% for 20% blend at 3000 rpm and increase by 39.1% for 10% blend, 17% for 15% blend and 2.99 % for 20% blend at 4000 rpm with respect to commercial Gasoline.

**CHAPTER 6**

# CONCLUSION AND RECOMMENDATIONS

## 6.1 Conclusion

From the results, it can be concluded that Ethanol blends are quite successful in replacing pure Gasoline in Spark Ignition Engine. Results clearly show that there is a decrease in exhaust emissions, increase in Brake Thermal Efficiency and Mechanical Efficiency. There is an increase in Specific Fuel Consumption because of low calorific Value of Ethanol than Gasoline and also increase in the maximum pressure induced due to blending of Ethanol in Gasoline.

So from the curves it is seen that 10% ethanol blended Gasoline is the best choice for use in the existing Spark Ignition Engines without any modification to reduce exhaust and increase Efficiency. A little consideration has to be taken on material used as maximum pressure inside cylinder is increased by blending. A balance has to be made between Specific Fuel Consumption and Efficiency to take care of users using blend as more fuel will be consumed due to blending of Ethanol with gasoline.

## 6.2 Recommendations for future work

- Blending Ethanol with Gasoline is quite beneficial in terms of exhaust emissions and Efficiency though there is an increase in specific fuel consumption. So a balance between them has to be established so that buyers don't suffer and at the same time Environment is also not polluted.

- Gasoline with 10% blend is very much effective and its further use can be studied and tested for practical purpose.
- Using blend increases the pressure generated inside the cylinder, thus material property should also be checked for future use of blends.

## REFERENCES


1. Science fair projects encyclopedia, 2004; launder, 2001.
2. Song, C., M. Klein, B. Johnson, and J.G. Reynolds, "catalysis in fuel processing and environmental protection, an introduction," catalysis today 50, 1 (1999).
3. Huff, peer review comments, December 19, 2001 D.P. Gardiner, et al., “Sub-Zero Cold Starting of a Port-Injected M100 Engine Using Plasma Jet Ignition and Prompt EGR,” SAE Technical Paper 930331, 1993.
4. http://www.afdc.energy.gov/afdc/pdfs/sae_m85.pdf.
5. Rodrigo C. Costa, www.elsevier.com/locate/fuel,fuel 89 (2010) 287–293.
6. BayraktarHakan, experimental and theoretical investigation of using gasoline ethanol blends in spark-ignition engines, renewable energy 30 (2005) 1733–1747.
7. YukselFikret, YukselBedri, the use of ethanol–gasoline blend as a fuel in ansi engine, renewable energy 29 (2004) 1181–1191.
8. Wen Lan-Bin, Xin Chen-Ying, Yang Shyue-Cheng, the effect of adding dimethyl carbonate (dmc) and ethanol to unleaded gasoline on exhaust emission, applied energy 87 (2010) 115–121.
9. Ceviz M.A, Yuksel F., effects of ethanol–unleaded gasoline blends on cyclic variability and emissions in an si engine, applied thermal engineering 25 (2005) 917–925.

10. Aulich Ted R., Xinming He, Ames A. Grisanti, and Curtis l. Knudson, gasoline evaporation-ethanol and nonethanol blends, ISSN 1047-3289 J.Aira waste manage. assoc. 44:1004-1009.
11. GrabysJuozas, influence of composition of gasoline – ethanol blend parameters of internal combustion engines,journal of kones, internal combustion engines 2003, vol. 10, 3-4.
12. Srinivasan C. Ananda and Saravanan C.G,study of combustion characteristics of an SI engine fuelled with ethanol and oxygenated fuel additives, journal of sustainable energy & environment 1 (2010) 85-91.
13. VaghelaJatin h., investigation of performance and emissions of 2- stroke single CylinderSI engine operating on alcohol–gasoline fuel blends using 4-spark plugs a new kind of ignition, international journal of advanced engineering sciences and technologies vol no. 6, issue no. 1, 027 – 030.
14. ZhaiHaibo and Frey H. Christopher, comparison of flexible fuel vehicle and life-cycle fuel, consumption and emissions of selected pollutants and Greenhouse gases for ethanol 85 versus gasoline, air & waste manage. assoc. 59:912–924.
15. SrinivasanC.Ananda and Saravanan C.G, emission reduction in SI engine using ethanol – gasoline blends on thermal barrier coated pistons, international journal of energy and environment, volume 1, issue 4, 2010 pp.715-726.
16. He Bang-Quan,WangJian-Xin studied emission characteristics of an efficient engine with ethanol blended gasoline, atmospheric env 37(2003) 949-957.

17. Hubballi P.A. and BabuT.P.Ashok, effect of aqueous denatured spirit on engine performance and exhaust emissions, SAE 2004-28-0036.
18. AltunSehmus, Hakan F. Oztop, exhaust emissions of methanol and ethanol-Unleaded gasoline blends in a spark-ignition engine, applied energy, 86 (2009), pp. 630–639
19. Pal Amit, Maji S., Sharma O.P. and Babu M.K.G.,“Vehicular Emission Control Strategies for the Capital City of Delhi”, India, January 16-18, 2004, SAE Paper no 2004-28-0051
20. YucesuHuseyinSerdar,effect of ethanol–gasoline blends on engine performance and exhaust emissions in different compression ratios, applied thermal engineering 26 (2006) 2272–2278.
21. http://www.greenfuels.org/ethafood.html
22. http://www.comalc.com/chatham.htm
23. http://www.greenfuels.org/ethaques.html
24. Demain, A. L., “Biosolutions to the energy problem,” Journal of Industrial Microbiology & Biotechnology. 36(3): 319-332, 2009.
25. Festel, G. W., “Biofuels – Economic Aspects,” Chemical Engineering & Technology. 31(5): 715-720, 2008.
26. Wallner, T., Miers, S. A., and McConnell, S., “A Comparison of Ethanol and Butanol as Oxygenates Using a Direct-Injection, Spark-Ignition Engine,” Journal of Engineering for Gas Turbines and Power. 131(3): 2009
27. Smith C.Hammel, Aabakken J., Issues associated with use of higher Ethanol Blends,NREL/TP-510-32206.