CHAPTER ONE

INTRODUCTION

1.1 Introduction

As the world's largest economies focus on reducing greenhouse gases (GHG), energy efficiency and it is ideally placed to help meet the most pressing challenge of making our planet a cleaner and better place to live and work.

A turbocharger can meet these challenges. Turbocharger or turbo is a turbine-driven forced induction machine that improves an internal combustion engines' efficiency and power output by adding extra air into the combustion chamber. This improvement over a naturally aspirated engine's output are observed because the turbine can force more air, and thus proportionately more fuel, into the combustion chamber than atmospheric pressure alone. Turbocharger is powered by a turbine which is driven by the engine's exhaust gases.

In short turbocharger uses the pressure energy of the exhaust gases and converts into kinetic energy to drive the turbines of turbocharger. There are various types of turbocharger like waste gated turbocharger, variable geometry turbocharger, twin turbocharger.

1.2 Problem Statement

The undesirable emissions generated in the combustion process of automobile and other IC engines. These emissions pollute the environment and contribute to global warming, acid rain, smog, odors, and respiratory and other health problems. The major causes of these emissions are non-stoichiometric combustion, dissociation of nitrogen, and impurities in the fuel and air. The emissions of concern are hydrocarbons (He), carbon monoxide (CO), oxides of nitrogen (NO_X), sulfur, and solid carbon particulates. These emissions should be reduced.

1.3 Research Objectives

(I)Evaluation influence of turbocharger pressure ratio on diesel engine emissions

(II)Reduce greenhouse gases in the environment.

(III)Reduce global warming phenomena.

(IV)Reduce emissions of diesel engine causing environmental pollution and damage to health of living organisms.

1.4 Significant of Study

This research addresses environmental problems in terms of harmful emissions from the diesel engine (greenhouse gases and global warming).

Global warming has led to a hole in the ozone .the radiation reaches the earth through this hole. Greenhouse gases also contribute to acid precipitation which harms the plant and pollutes water and effects on the ecosystem, hence the importance of research to reduce these phenomena by reduce emissions that is done by using turbocharger on engines.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

An engine or motor is a machine designed to convert one form of energy into mechanical energy. Heat engines burn a fuel to create heat, which is then used to create a force. Electric motors convert electrical energy into mechanical motion; pneumatic motors use compressed air and clockwork motors in wind-up toys use elastic energy. [1]

2.2 Types of engines

internal combustion engine

steam engine

electrical engine

steerling engine

Non-combusting heat engines

external combustion engine

Air-breathing combustion engines [1]

2.3 Internal combustion engines

An internal combustion engine (ICE) is a heat engine where the combustion of a fuel occurs with an oxidizer (usually air) in a combustion chamber that is an integral part of the working fluid flow circuit. In an internal combustion engine the expansion of the high-temperature and high-pressure gases produced by combustion applies direct force to some component of the engine. The force is applied typically to pistons, turbine blades, rotor or a nozzle. This force moves the component over a distance, transforming chemical energy into useful mechanical energy. [1]

2.4 Classification of internal combustion engines

Internal combustion engines can be classified in a number of different ways:

2.4.1Types of ignition

- Spark Ignition (SI): An SI engine starts the combustion process in each cycle by use of a spark plug. The spark plug gives a high voltage electrical discharge between two electrodes which ignites the air fuel mixture in the combustion chamber surrounding the plug.

- Compression Ignition (CI): the combustion process in a CI engine starts when the air – fuel mixture self ignites due to high temperature in the combustion chamber caused by high compression. [2]

2.4.2Engine Cycle

-A four- stroke cycle: has four piston movements over two engine revolutions for each cycle.

-Two- Stroke Cycle: A two – stroke cycle has two piston movements over one revolution for each cycle.[2]

2.4.3Valve Location

- Valves in head (overhead valve) also called I Head engine.

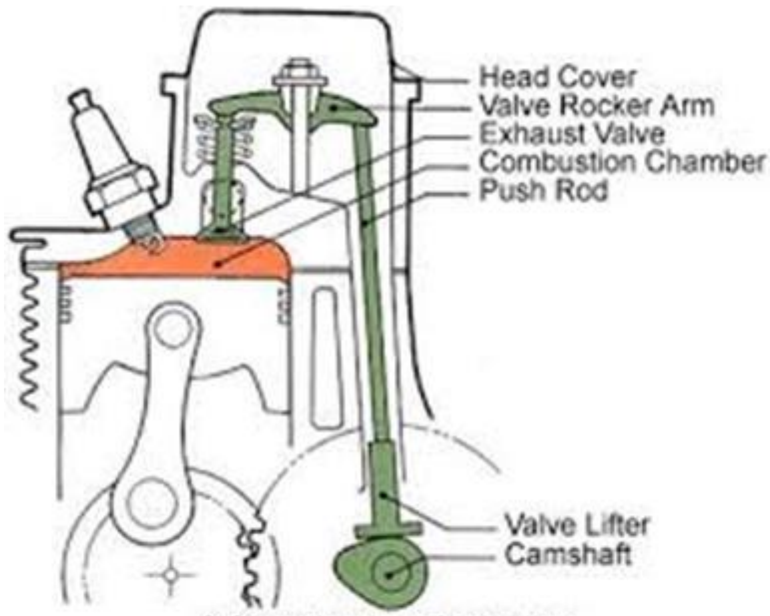


Figure (2-1) Valves in head

- Valves in block (flat head) also called L Head engine: Some historic engines with valves in block had the intake valve on one side of the cylinder and the exhaust valve on the other side.

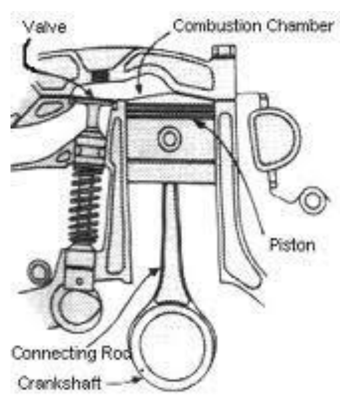


Figure (2-2)Valves in block

- One valve in head (usually intake) and one in block also called F Head engine.[2]

2.4.4 Basic design

-Reciprocating engine: has one or more cylinders in which pistons reciprocate back and forth. The combustion chamber is located in the closed end of the each cylinder. Power is delivered to a rotating output crankshaft by mechanical linkage with the pistons.[2]

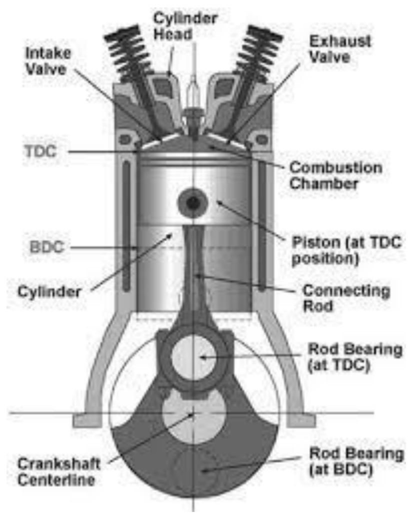


Figure (2-3) Reciprocating engine

-Rotary engine: is made of a block (stator) built around a large nonconcentric rotor and crankshaft. The combustion chamber are built into the nonrotating block. A number of experimental engines have been tested using this concept but the only design that has ever become common in an automobile is the wankel engine in several Mazda models. Mazda builds rotary automobile engines with one two and three rotors.[2]

Figure (2-4) Rotary engine

2.4.5 Position and number of cylinder of reciprocating engines

- Single cylinder engine: has one cylinder and piston connected to the crankshaft.

- In line cylinders : are positioned in a straight line one behind the other along the length of the crankshaft.

- V Engine: Two banks of cylinders at an angle with each other along a single crankshaft allowing for a shorter engine block.[2]

Figure (2-5)V Engine

- Opposed cylinder engine : Two banks of cylinders opposite each other on a single crankshaft. These are common on small air craft and some automobiles with an even number of cylinders from two to eight or more. These engines are often called flat engine.[2]

Figure (2-6) Opposed cylinder engine

- W Engine: Engines of two different cylinder arrangements have been classified as W engines in the technical literature. One types is the same as V engine except with three banks of cylinders on the same crankshaft.

Figure (2-7)W Engine

- Opposed piston engine: Two piston in each cylinder with the combustion chamber in the center between the pistons.

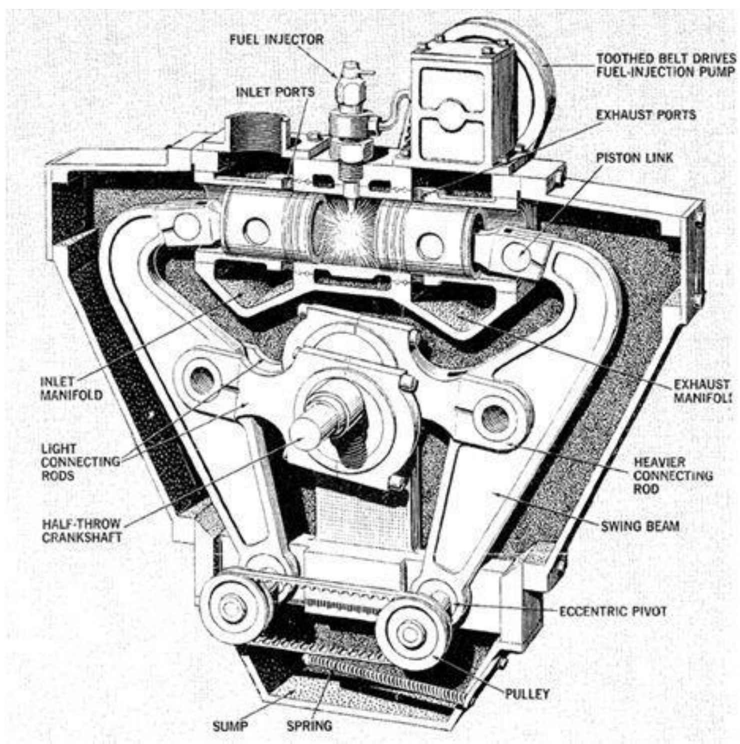


Figure (2-8) Opposed piston engine

- Radial engine: Engine with pistons positioned in a circular plane a round the central crankshaft. The connecting rods of the pistons are connected to a master rod which in turn is connected to the crankshaft.[2]

Figure (2-9) Radial engine

2.4.6 Air intake process

- Naturally air: No intake air pressure boost system.

- Supercharged: Intake air pressure increased with the compressor driven off the engine crankshaft.

- Turbocharged: Intake air pressure increased with the turbine compressor driven by the engine exhaust gases.

- Crankcase compressed: Two stroke cycle engine which used the crankcase as the intake air compressed. Limited development work has also been done on design and construction of four stroke cycle engines with crankcase compression.[2]

2.4.7 Method of fuel input for spark ignition engines

- Carbureted.

-Multipoint port fuel injection: One or more injectors at each cylinder intake.

- Throttle body fuel injection: Injectors upstream in intake manifold.

- Gasoline direct injection: Injectors mounted in combustion chambers with injection directly into cylinders.[2]

2.4.8 Method of fuel input for compression ignition engines

- Direct injection: Fuel injected into main combustion chamber.

- Indirect injection: Fuel injected into secondary combustion chamber.

- Homogeneous charge compression ignition: Some fuel added during intake.[2]

2.4.9 Fuel used

- Gasoline.

- Diesel oil or fuel oil.

- Gas, natural gas, methane.

- LPG(Liquefied petroleum gas)

- Alcohol ethy, methy.

- Dual fuel: There are a number of engines that use a combination of two or more fuels, some usually large, CI engines use a combination of NG and diesel fuel. These are attractive in developing third world countries because of the high cost of diesel fuel. Combined gasoline alcohol fuels are becoming more common as an alternative to straight gasoline automobile engine fuel.

- Gasohol: Common fuel consisting of 90% gasoline and 10% alcohol.[2]

2.4.10 Application

- Automobile, Truck, Bus.

- Locomotive.

- Stationary.

- Marine.

- Aircraft.

- Small portable, Chain saw, Model Airplane.[2]

2.4.11Type of cooling

- Air cooled.

- Liquid cooled, water cooled.[2]

2.5 Diesel Engine

The diesel engine (also known as a compression-ignition or CI engine) is an internal combustion engine in which ignition of the fuel that has been injected into the combustion chamber is caused by the high temperature which a gas achieves (i.e. the air) when greatly compressed (adiabatic compression). Diesel engines work by compressing only the air.

Diesel engines may be designed as either two-stroke or four-stroke cycles. They were originally used as a more efficient replacement for stationary steam engines. Since the 1910s they have been used in submarines and ships. Use in locomotives, trucks, heavy equipment and electricity generation plants followed later. In the 1930s, they slowly began to be used in a few automobiles. Since the 1970s, the use of diesel engines in larger on-road and off-road vehicles in the US increased. According to the British Society of Motor Manufacturing and Traders, the EU average for diesel cars accounts for 50% of the total sold, including 70% in France and 38% in the UK. [3]

2.6 Exhaust Emissions from the Combustion of Diesel and Biodiesel Fuel

Transport for many years has had a significant effect upon air pollution since the inception of the internal combustion engine and especially after major commercialization .The exhaust emissions generally result from the combustion of fossil fuel in vehicle engines. Diesel fossil fuel is toxic and may cause long term adverse effects to the aquatic environment. In diesel engines, the pollutant formation processes are strongly dependent on the fuel distribution and how that distribution changes with time due to mixing (Heywood 1988). In general, diesel engines fuelled with biodiesel emit a lower amount of unburned hydrocarbon (HC), particulate matter (PM) and carbon monoxide (CO) as compared to fossil diesel whereas NO(x) emissions are slightly increased.

The combustion of biodiesel alone provides over a 90% reduction in total unburned hydrocarbons (HC) and a 75-90% reduction in polycyclic aromatic hydrocarbons (PAHs).The results from previous experiments on biodiesel and their blends with ULSD in a single cylinder engine show that an increased proportion of biodiesel blend resulted in higher NO_X ,reduced smoke and increased brake specific fuel consumption .[4]

2.6.1 Oxides of Nitrogen

The major pathway of NO_X formation is thermal NO_X, fuel NO_X and prompt NO_X (Heywood 1988; Kutz 2006).

Thermal NO_X refers to NO_X produced during high temperature oxidation of the diatomic nitrogen found in combustion air and simply derived via the well-known Zeldovich mechanism. The formation rates principally a function of temperature and the exposure period of nitrogen at that temperature (Horn et al. 2007; Keating 2007).

The fuel NO_X formed when the combustion of fuel which contains organic nitrogen compounds occur. During combustion, the nitrogen bound in the fuel is released as a free radical and eventually forms free N2, or NO. High quality gaseous fuel has no organically bound nitrogen which produces an ignorable amount of NO_X through this process. Conversely, fuel NO_X is vital to revision for residual fuel oil, coal or waste fuel used that may contain a considerable amount of organically bound nitrogen.

The prompt NO_X is attributed to the reaction of atmospheric nitrogen, N2, with radicals such as C, CH, and CH2 fragments derived from fuel. The reaction took place in the earlier stage of combustion and produced fixed species of nitrogen such as Nitrogen monohydrate (NH), hydrogen cyanide (HCN), dehydrogenate cyanide (H2CN) and Ciano radical (CN-) which can oxidize to NO. Prompt NOX is commonly important in the low temperature combustion process.

NO_X formed by the combustion of fuel in an internal combustion engine typically consists of nitric oxide (NO) and nitrogen dioxide (NO2) where the nitric oxide is dominant with a small amount of NO2 (Heywood 1988). The formation of NO_X is mostly from nitrogen in the air but some liquid fuels contain nitrogen such as NH3, NC and HCN thus contribute higher potential on producing more NO_X . [4]

2.6.2 Carbon Monoxide

Carbon monoxide (CO) emissions from IC engines are primarily controlled by the fuel/air equivalence ratio (Heywood 1988). The CO concentration in the exhaust tail pipe increases steadily with increasing equivalence ratio. Since CO emissions are closely related to fuel rich combustion, therefore, spark ignition engines produce a significant amount of CO emission compared with diesel engines. The diesel engine is often operating well on the lean side of stoichiometric ratio especially at low load. Lower volumetric efficiency of air intake system resulted to increase the emission of CO in the exhaust gas. This is due to the incomplete combustion in engine cylinder. [4]

2.6.3 Unburned hydrocarbons

Hydrocarbons (HC) are grouped into categories based on their chemical structure such as paraffin (alkenes), olefins (alkenes), acetylenes (alkynes), or cyclic hydrocarbons. Hydrocarbon emissions are the consequence of in complete combustion of the hydrocarbon fuel. The compression ignition engine produces less HC as compared to SI engines due to its operation with an overall fuel lean equivalence ratio. The combustion of diesel fuel in compression ignition engines involved complex heterogeneous processes.

Although the combustion of diesel fuel takes place in fuel-lean conditions, incomplete combustion still occurred. Due to non-homogeneity of the fuel mixture, some local spots in the combustion chamber will be too lean to completely combust. The amount of unburned HC resulting from this over lean regions is dependent on the amount of fuel injected during the ignition delay, the mixing rate with air during this period, and the extent to which prevailing cylinder conditions are conducive to auto-ignition (Heywood 1988). HC emissions are also reported as sensitive to engine temperature where these emissions decrease as the engine temperature increase (Heywood 1988). Fuel also has an effect on HC emissions where the fuel with lower end boiling point produces higher HC (Lilly 1984). [4]

2.6.4 Sulfur oxides

-Produced by fossil fuel burning, industry, biomass burning.

Sulfur dioxide involved in acid deposition; corrodes metals and damages stone, irritates respiratory system. (SO_2 reacts with enzymes, impairing their functions.)

When burning fuels with sulfur, like coal and diesel, all of the sulfur will be oxidized in the flame into SO_2 and SO_3, collectively called $SO_{(x)}$. This poses serious problems because:

Sox dissolve in clouds to form sulfuric acid, which can then be deposited to the earth by rain. This is called “acid rain” and has caused deforestation in Europe and North America and serious damage to structures (monuments, steel buildings).

-Sox is a respiratory irritant and in large concentrations can cause death. Sulfur-rich coal combustion for domestic use (e.g. for cooking or heating) has been responsible for thousands of deaths in London over the past centuries, notably during the "Great London Smog” in December 1952, in which about 4,000 people died. Current emission standards on sulfur oxide emissions are very strict and are met by post combustion treatment of the exhaust gases. [4]

2.6.5 Particulate Matter

Particulate matter is defined as any material other than water, in the exhaust of an internal combustion engine which can be filtered after dilution with ambient air (Lilly 1984). Most of the PM is generated from incomplete combustion of diesel fuel but some comes from lubricating oil components which vaporize and then react during combustion (Heywood 1988).

The individual structure of PM is principally clusters of many small spheres or spherules of carbon. The PM emissions consist of absorbed and condensed high molecular weight organic compounds which include unburned hydrocarbons, oxygenated hydrocarbons (Esters, ethers, organic acids) and poly nuclear aromatic hydrocarbons (Heywood 1988).The combustion of biodiesel produces less PM as compared to fossil diesel.

The reductions of PM could be explained by the increase in oxygen content in the fuel which contributes to complete fuel oxidation even in locally fuel rich zones, and by the lower final boiling point which guarantees a complete evaporation of the liquid Fuel. [4]

2.7 The Impact of diesel engine emissions

2.7.1 Carbon Oxides

Carbon dioxide is a major contributor to the greenhouse effect, which is a term denoting the warming of the atmosphere due to the CO2 absorbing part of the radiation emitted by the earth surface. This may then lead to global warming.. Carbon monoxide is extremely dangerous and can cause death if inhaled in large concentrations. [12]

2.7.2 Hydrocarbons

Important greenhouse gas, so affects global climate change. And Causes photo chemical smog, cancer-causing, respiratory tract irritants, and some mutagens. [12]

2.7.3 Nitrogen oxides

Nitric oxide (NO),nitrogen dioxide (NO2), and nitrous oxide (N2O).Involved: Acid deposition, Global warming, Ozone depletion, Photochemical smog, and Inhibits plant growth, aggravate health problems.

2.8 Technical used to reduce emissions

2.8.1 Reducing NO_X Emission

The presence of NO_X in internal combustion engines exhaust emission is due to high combustion temperature which reacts with nitrogen in the air supplied for combustion. Following are the methods to reduce NO_X emission.[5]

2.8.1.1 Exhaust Gas Recirculation (EGR)

As the name suggests, some amount of engine exhaust gases are send back to the scavenge space to mix up with the air to be supplied to cylinder for combustion. This reduces the oxygen content of the air. [5]

2.8.1.2 Water Injection and Water Emulsion

In this method, water is added to reduce the temperature of combustion leading to low NO_X emission. In water emulsion, fuel is blended with water and in water injection a separate fresh water injector is mounted in the cylinder head which injects water. This method has a drawback of increasing the specific fuel oil combustion. [5]

2.8.1.3 High Scavenge Pressure and Compression Ratio

With high scavenge pressure and compression ratio, large amount of air can be introduced inside the cylinder to lower combustion temperature and NO_X emission. [5]

2.8.1.4 Selective Catalytic Reduction

The SCR is the most efficient method to reduce NO_X emissions. In this method, low sulfur fuel is used and exhaust temperature is maintained above 300^0 C. The exhaust gas is mixed by water solution of urea and then it is passed through catalytic reactor. The only disadvantage of SCR is its expansive installation and operating cost. [5]

2.8.1.5 Two Stage Turbocharger

Two stage turbocharger can reduce the exhaust temperature in the intercoolers and also the NO_X content in the emitted exhaust.

2.8.1.6 Engine Component Modification

It is better to design an engine which has a property to reduce the NO_X formation during combustion process rather than investing on expensive secondary measures. Integration of slide valve type fuel injector with almost zero sack volume eliminates any chance of fuel dripping and after burning, leading to lower cylinder temperature and NO_X formation.

2.8.2 Reducing SO_X Emission

SO_X or sulfur oxides are formed during combustion process in the engine because of presence of sulfur content in the fuel. Following are the methods and technologies used to reduce sulfur emission from engines.

2.8.2.1Use of Low sulfur fuel

It is expensive but most commonly used.

2.8.2.2 Exhaust Gas Scrubber Technology

The exhaust gas from the engine is passed through the scrubber tower where a liquid is showered over it . Fresh water blended with caustic soda (Noah) is used as a scrubbing liquid which reduces the SO_X to 95%. The scrubbing water is then sent to a water treatment effluent emulsion breaking plant after which it can be discharged overboard.

2.8.2.3 Cylinder Lubrication

Good quality cylinder lubrication along with efficient control systems such as Pulse or Alpha lubrication systems can neutralize the sulfur in the fuel and reduce SO_X emissions from the engine.[5]

2.9 Engine Performance Parameters

2.9.1 Brake power of IC Engine

The brake power (briefly written as B.P.) .Of an IC Engine is the power available at the crankshaft. The brake power of an IC engine is usually, measured by means of a brake mechanism (prony brake or rope brake). [6]

In case of prony brake, brake power of the engine:

$$BP=\frac{2\pi NT}{60} \quad \text{(watts)} \qquad (2.1)$$

Where:

T = Brake torque (Nm)

N = Speed of the engine (R.P.M)

2.9.2 Brake specific fuel consumption (BSFC)

Is a measure of the fuel efficiency of any prime mover that burns fuel and produces rotational, or shaft power. It is typically used for comparing the efficiency of internal combustion engines with a shaft output.

It is the rate of fuel consumption divided by the power produced. It may also be thought of as power-specific fuel consumption, for this reason. BSFC allows the fuel efficiency of different engines to be directly compared.

To calculate BSFC, use the formula:

$$\text{BSFC}=\frac{mf}{BP} \qquad (2\text{-}2)$$

Where: m_f is the fuel consumption rate in grams per second (g/s); (Bp) is the power produced in watts. [6]

2.9.3 Brake thermal efficiency

Is defined as brake power of a heat engine as a function of the thermal input from the fuel. It used to evaluate how well an engine converts the heat from a fuel to mechanical energy.

For engines where a fuel is burned there are two types of thermal efficiency: indicated thermal efficiency and brake thermal efficiency.[7]

Calculations**:**

$$\eta b = \frac{\mathrm{BP}}{\text{fuel flow(kg/sec)}*\text{calroific value(kj/kg)}} * 100\% \qquad (2\text{-}3)$$

$$\eta b = \frac{\mathrm{BP}}{mf * LHV} * 100\% \qquad (2\text{-}4)$$

2.10 Effects of Variable Geometry Turbochargers in Increasing Efficiency and Reducing Lag:-

2.10.1The Turbochargers

The turbocharger is a turbine that is driven by exhaust gases that compresses incoming air into the engine. The hot side of the turbo receives its energy from the heat and flow energy of the exhaust system.

The cold side of the turbocharger pressurizes fresh air and forces it into the engine. The pressure generated by the cold side is called the boost. The cold side is driven by a shaft that is connected to the hot side. [8]

2.10.2The drawbacks to turbochargers

The main drawback to a turbocharger, besides cost, is its fixed geometry. The Aspect Ratio (A/R) of a turbo, which is based on its geometry, has a direct relation to both the power increase generated and the motor speed at which the power increase is generated.

A smaller A/R will produce boost pressure at a lower engine speed, but will be unable to provide a high enough flow rate at higher engine speeds. This leads to higher exhaust manifold pressures, lower pumping efficiencies, and lower power output. A larger A/R will create boost at higher engine speeds, and thus create more power, but it will be unable to produce boost at lower engine speeds. So an A/R must be picked to either; produce power at lower engine speeds for quicker acceleration, or for higher engine speeds to produce a greater total power. [9]

2.10.3The Lag

The time it takes for the engine to produce boost between transients is called lag. A large A/R turbo will have a longer lag time than a smaller A/R turbo due its larger requirement of energy from the engine to produce boost. [8]

2.10.4The Variable Geometry Turbocharger (VGT)

Variable Geometry Turbochargers are turbochargers whose geometry and thus effective A/R can be altered as needed while in use. The most common design includes several adjustable vanes around a central turbine. As the angle of the vanes change, the angle of air flow onto the turbine blades changes, which changes the effective area of the turbine, and thus the aspect ratio (A/R) changes.

Figure (2-10) Variable Geometry Turbocharger

2.10.5 The benefits of Variable Geometry Turbochargers

-Reducing Lag Time:

The area between the adjustable vanes works as nozzles. These nozzles are thus varied in size as a function of engine operating conditions. By opening the nozzles at high engine speed or closing them at low speed, effectively changing the A/R with engine speed or demands, the turbo can produce boost from a low speed without restricting flow at higher speed. Since they can produce boost at lower engine speed Lag time is decreased. Also since the vanes are remotely controlled the boost pressure can be altered without changing engine speed. By adjusting the vanes you can increase exhaust manifold pressure during transients (gear changes). Coming out of a transient with a higher exhaust manifold pressure allows this stored energy, in the form of pressure, to be used to drive the turbo to a higher boost level faster. By increasing the boost level faster Lag is once again reduced.

-Increasing Efficiency:

Turbochargers in general are a very good way to improve the efficiency of an engine. By pressurizing the intake manifold, more air, and thus more fuel, is brought into the cylinder every time the intake valve opens. This creates a volumetric efficiency of greater than 1. A volumetric efficiency of even 1 is impossible in any real engine without some kind of forced induction due to friction losses. This improves the overall efficiency of the engine by allowing it to burn more air and fuel on every cycle. The high positive pressure generated also helps to overcome any casting defects in the manifold, such as surface roughness (major losses) or tight corners (minor losses), by providing a larger driving force, or pump head. Fixed geometry turbochargers (FGT) work as any other centrifugal pump and thus have a limited optimal operating range. VGTs have the advantage that many different pressure ratios can be produced at a single engine speed due to the variable vanes changing the effective area and A/R. The vanes can be manipulated to create an optimal boost pressure at any speed. By producing an optimal boost through a larger engine speed range the overall efficiency is increased. [10]

2.10.6 The disadvantages of variable geometry turbochargers

Cost and Reliability: VGTs are very complex and require complicated control systems. The small moving parts, sensors, and controllers make them more expensive to produce than FGT. All the parts are exposed to extreme temperatures of over 1000oF making them wear quickly. Also due to the extreme temperatures they need to be made from exotic materials which increase the cost even more.

-Availability for Gasoline Engines: Typically VGTs are only available for diesel engines. Diesel engines produce much lower exhaust temperatures than gasoline engines. These lower temperatures allow for the use of more common materials and higher reliability. Through the use of newer materials borrowed from the aerospace industry a few VGTs will be hitting the gasoline market soon. The 2007 Porche 911 Turbo has a twin turbocharged 3.6-litre flat six, and the turbochargers used are BorgWarner′s Variable Turbine Geometry (VTGs). [11]

2.11 Previous Simulation Studies

For improving the thermal efficiency of insulated engine, the volumetric efficiency drop is compensated by turbo charging in the present experimental work. This gave the better performance with reduction in emissions. With the turbo, charging the intake boost pressure is raised and its effect on the engine performance is also studied.

To pressurize the inlet air, internally powered turbo charging equipment with closed loop lubrication is fabricated. In the turbo charging the high temperature exhaust gases are expanded in a low-pressure turbine for the power generation and this is further coupled to motor of the compressor. This compressor compresses the inlet air and supplies to the engine at slightly higher pressure. By controlling the inlet air, the engine is turbocharged at different inlet pressures.

On the experimental investigations on an insulated diesel engine under turbo charging conditions: The increase in the intake boost pressure improves the brake thermal efficiency of the engine, For the compensation of drop in volumetric efficiency of the insulated engine 4% intake boost pressure is required for turbo charging, Though the higher temperatures are available in the combustion chamber due to insulation, the increase in exhaust gas temperature is marginal. This is attributed to the higher latent heat of vaporization of alcohol, as the alcohol contains oxygen and more air is available in the turbo charging for combustion, the ignition delay is reduced, Due to the complete combustion of alcohol at higher temperatures the emissions are also marginal.[13]

CHAPTER THREE

METHODOLOGY

3-1 Introduction

The DIESEL-RK is professional thermodynamic full-cycle engine simulation software. On a market, there are few well known thermodynamic engine simulation tools from different developers. These tools cover wide range of practice tasks: from general engine concept analysis up to design engine systems. The kernels of engine simulation models of other programs are focused mainly on non-steady 1D gas dynamic phenomena. DIESEL-RK is focused on advanced diesel combustion simulation and emission formation simulation, one has not such specific functions as analysis of engine transient behavior or analysis of difference between engine cylinders operation. Usage of DIESEL-RK is effective if customer deals with engine combustion optimization and emissions control, port timing, EGR and turbo charging optimization as well. [14]

3-2 The Main Features of DIESEL-RK

Thermodynamic analysis of Diesels fuelled by diesel oil, methanol, bio-fuels and mixtures of bio fuels with diesel oil. HCCI concepts and Dual fuel systems are supported.

Thermodynamic analysis of SI petrol engines and gas engines, including prechamber engines, and engines fuelled by Natural gas (Methane), Pipeline gas (Propane-Butane), Biogas, Wood gas, Syn gas with arbitrary composition (Producer gas), by any gas having arbitrary composition as well.

Thermodynamic analysis of Two- and Four-stroke engines, Junkers engines with opposite pistons; Crank case scavenged engines, etc.

Simulation and optimization of Mixture Formation and Combustion in diesel. Fuel Injection optimization. Optimization of sprayer design and location, injection pressure, injection timing, rate shaping, split / multiple injection strategy. PCCI analysis including Low Temperature Combustion phase, etc. Individual diameters and orientations of nozzles of few injectors, having independent control (own fuel and own injection profiles) are accounted and may be optimized.

Detail Chemistry is simulated for Ignition Delay prediction at HCCI for Diesel Fuel, Methanol and for Bio-Fuel.

Common Rail control algorithm development; Automatic optimization of Injection profile fronts shape.

Effect of Combustion Chamber Geometry modification.

Fuel Sprays Evolution visualization.

Nitrogen Oxides, Soot and Particles formation simulation. Detail Kinetic Mechanism for NO_X formation at large EGR and multiple injection.

Simulation of effects of Turbo charging, Intake and Exhaust Port flows, Bypasses, and EGR.

Valve and Port Timing optimization. VVA optimization with the dwell of the valves.

Multipara metric optimization of engines parameters, Conjoint optimization of NO_X, PM and SFC, including Pareto optimization.

DIESEL-RK solver may be run under the control of other packages:
Simulink, IOSO NM, etc.

The tool is very easy in use. Remote access is provided. [15]

3-3 History of the project

Development of the DIESEL-RK software core has been started in 1981-82 in the department of Internal Combustion Engines (Piston Engines), Bauman Moscow State Technical University. From its very beginning the software was devised as a tool for optimization research and, therefore, a particular emphasis was made on the adequacy and operating speed of the mathematical models and algorithms applied, many of those being original authors' codes. All these years the work has been constantly carried out in touch with engineers and computational research performed was to the order of different manufacturers of Internal Combustion Engines (ICE). In this time the software adequacy has been tested to fit dozens of engines of various types and purposes. Many computational procedures and options were introduced into the software to suit the demand of industrial enterprises software users including biggest engine manufacturers of Russia.[14]

3-4 Applications of DIESEL-RK

DIESEL software has been designed since 1981 - 1982. In those years it was applied in researches of many engines with different sizes, speed and application. All these years the software is improving to grow efficiency and field of application. Results of simulation in every research are compared with experimental data obtained by various authors, mainly obtained by diesels manufacturers.

Main features of program DIESEL-RK are similar to known thermodynamic programs. However, together with conventional features, the

DIESEL-RK has new advanced applications which are absent in other programs. DIESEL-RK is oriented on diesel combustion optimization and ICE analysis and optimization. The assumption about identical work of all engine cylinders allows considerably increase operating speed and one makes it possible to resolve multipara metrical optimization tasks. [15]

3-4-1 Main Applications

-Torque curve and other engine performances prediction.

-Fuel consumption prediction and optimization.

-Combustion and emission analysis and optimization.

-Knock prediction.

-Valve timing optimization.

-EGR system analysis and optimization.

-Turbocharger and bypasses matching and optimization.

-Research and optimization of fuel injection profile including multiple injection, sprayer design and location as well as piston bowl shape optimization.

-Convert of diesel engines into gas engines.

-Dual fuel engine analysis.

3-5 GENERAL ENGINE PARAMETERS

- Cylinder bore

- Piston stroke

- Compression ratio

- Number of cylinders

- Nominal engine speed (rpm)

- Mechanism of transformation of a movement of the piston.

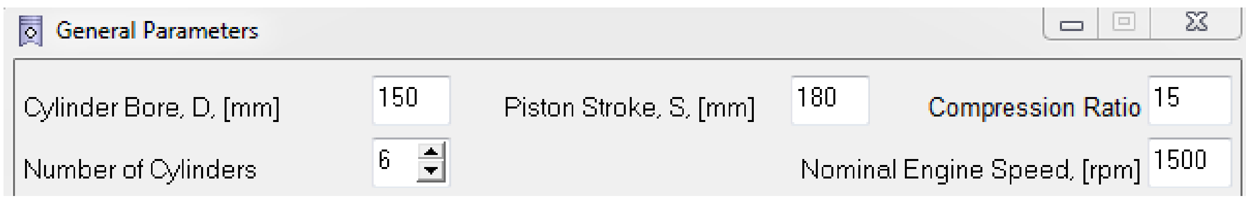


Figure (3-1) General Parameters

-Length of the connecting-rod. For Junkers engine specify length of the both connecting rods and angular offset.

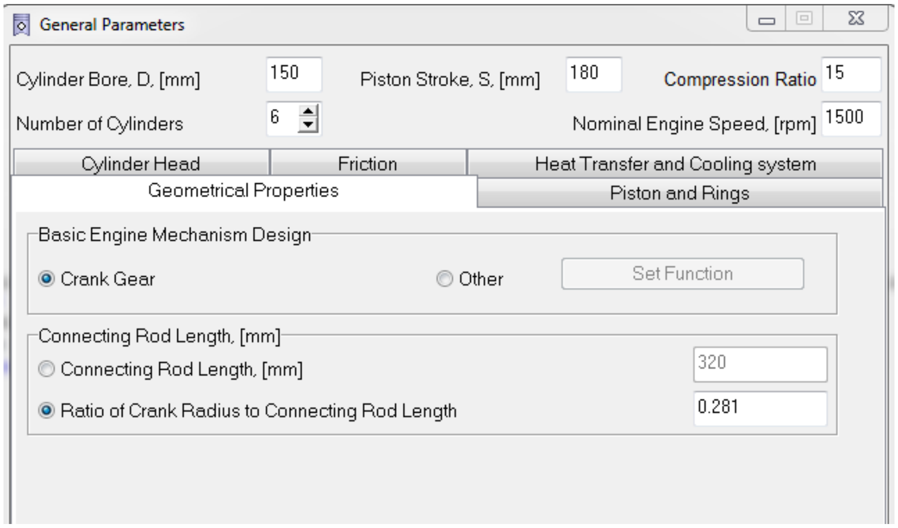


Figure (3-2) Geometrical Properties

3-5-1 Design of engine

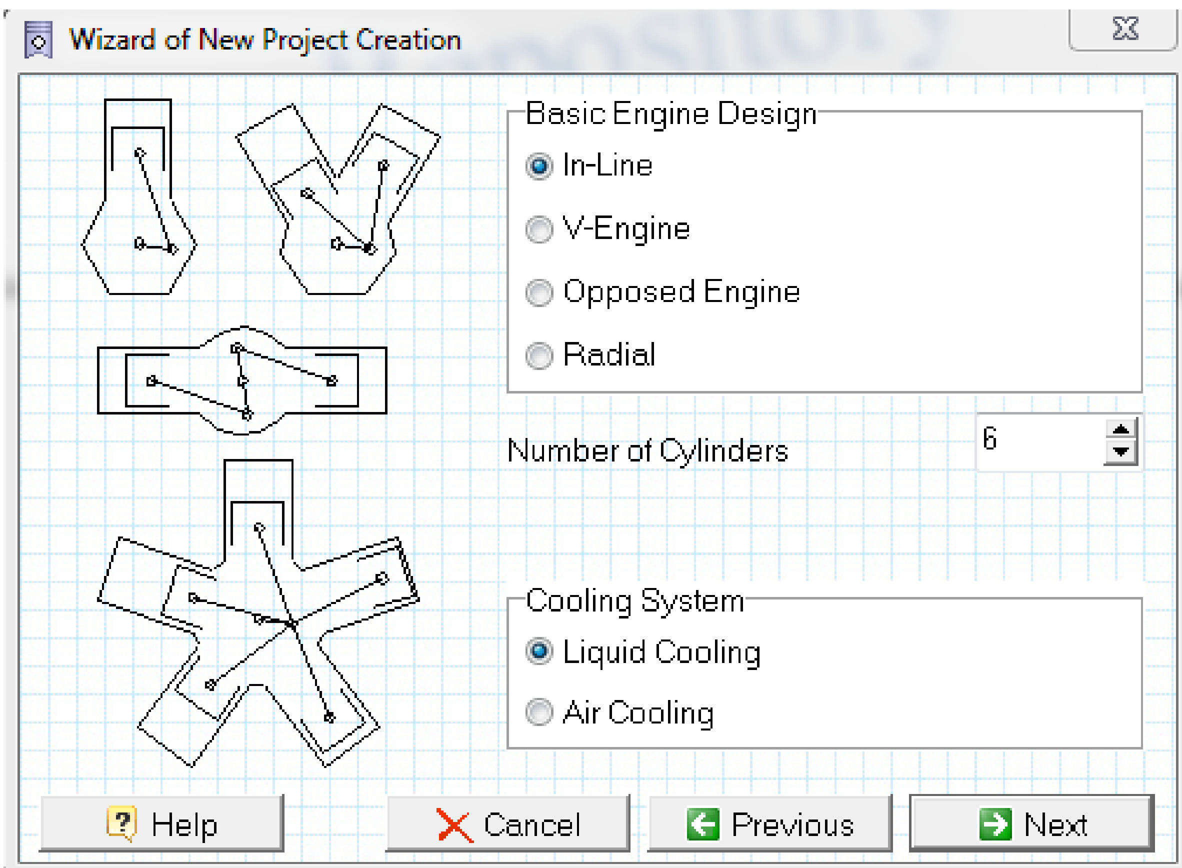


Figure (3-3) basic engine design – cooling system – number of cylinders

3-5-2 Material of the piston or piston crown

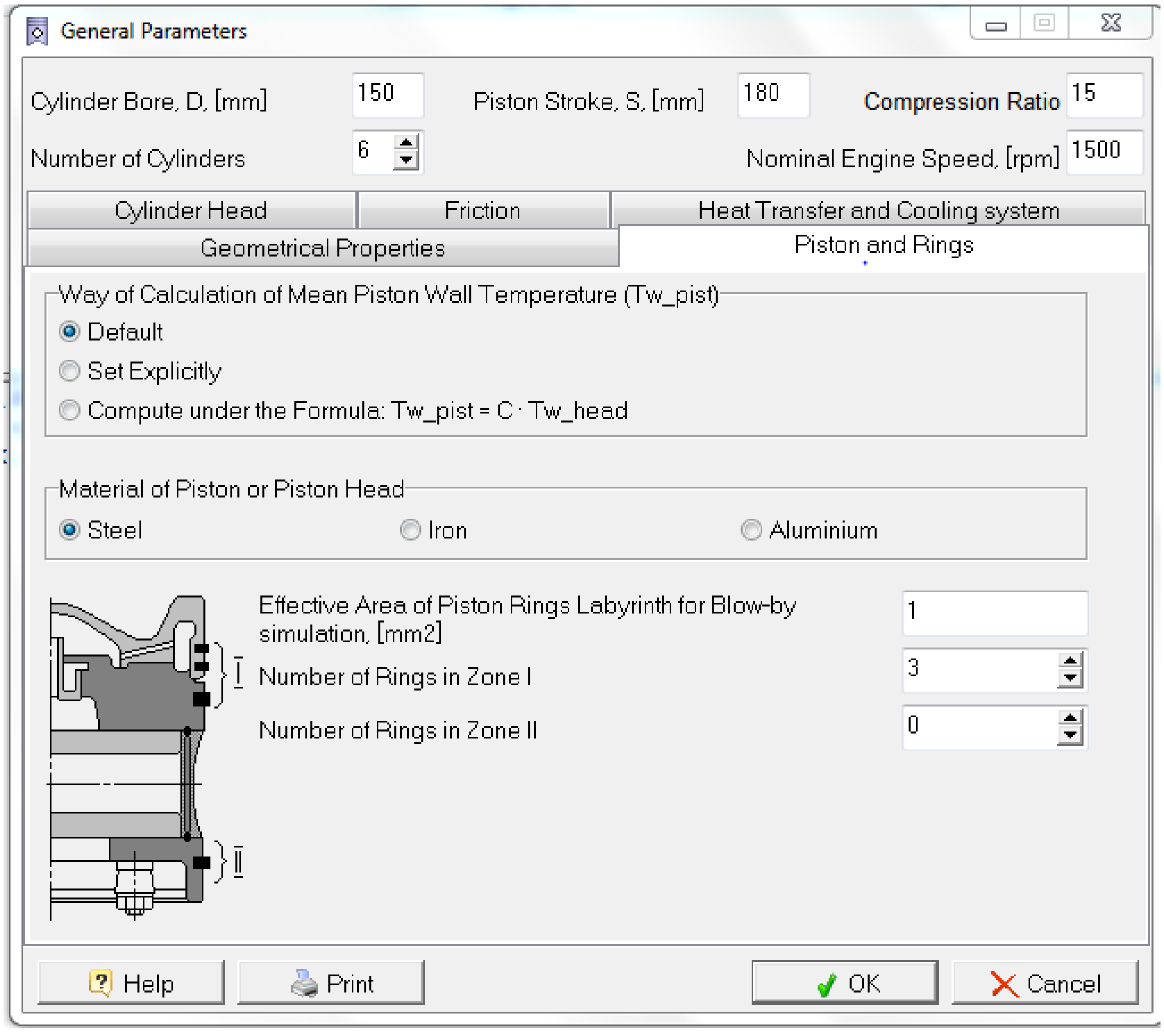


Figure (3-4) piston and rings

3-5-3 Material of the Cylinder Head and Average thickness of the cylinder head wall

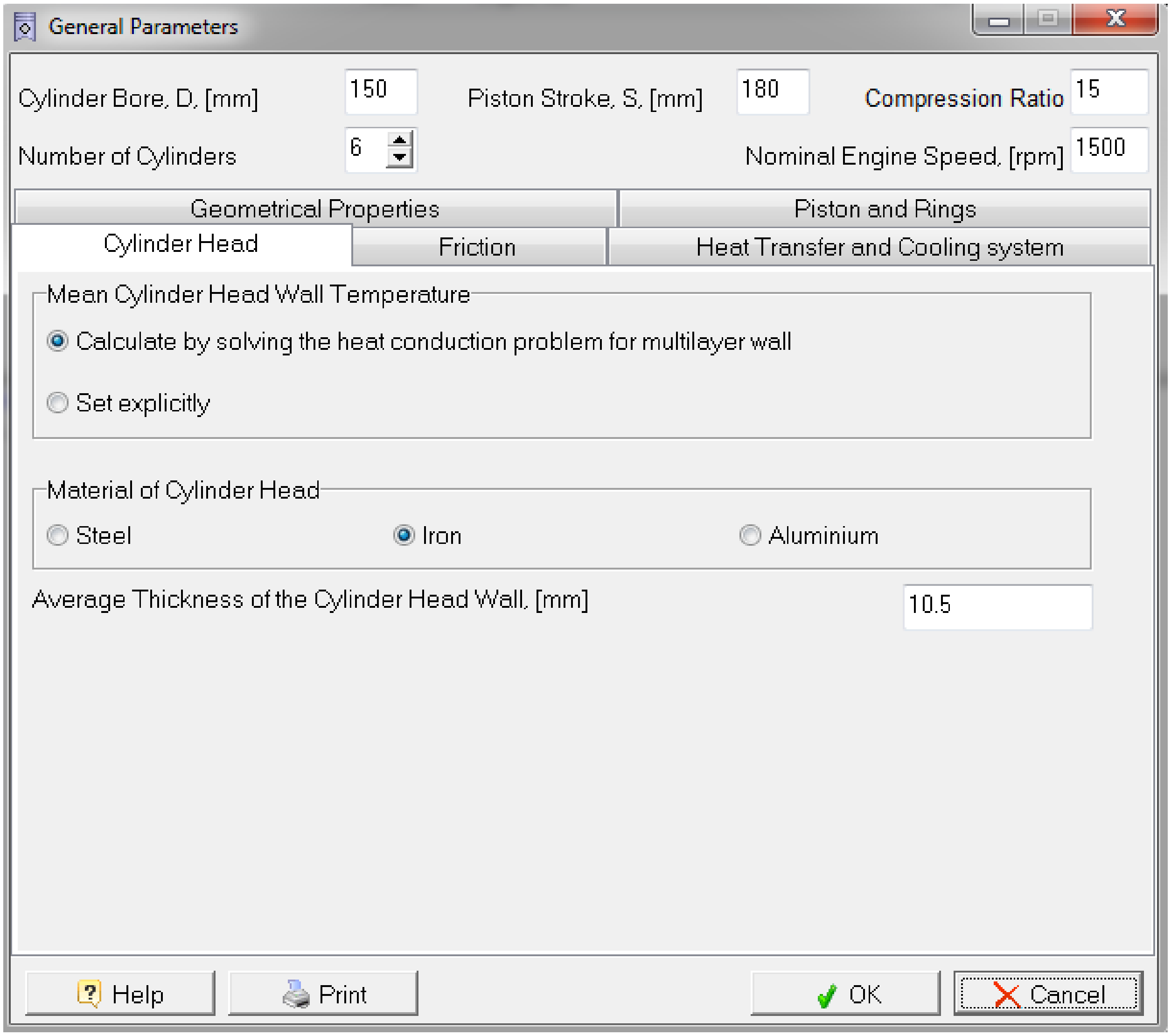


Figure (3-5) cylinder head

3-5-4 Type of engine cooling (liquid-cooling)

Parameters of a fur layer and casting skin on a cooled wall of water cooling system:

- Thickness

-thermal conductivity [W/ (m K)] [16]

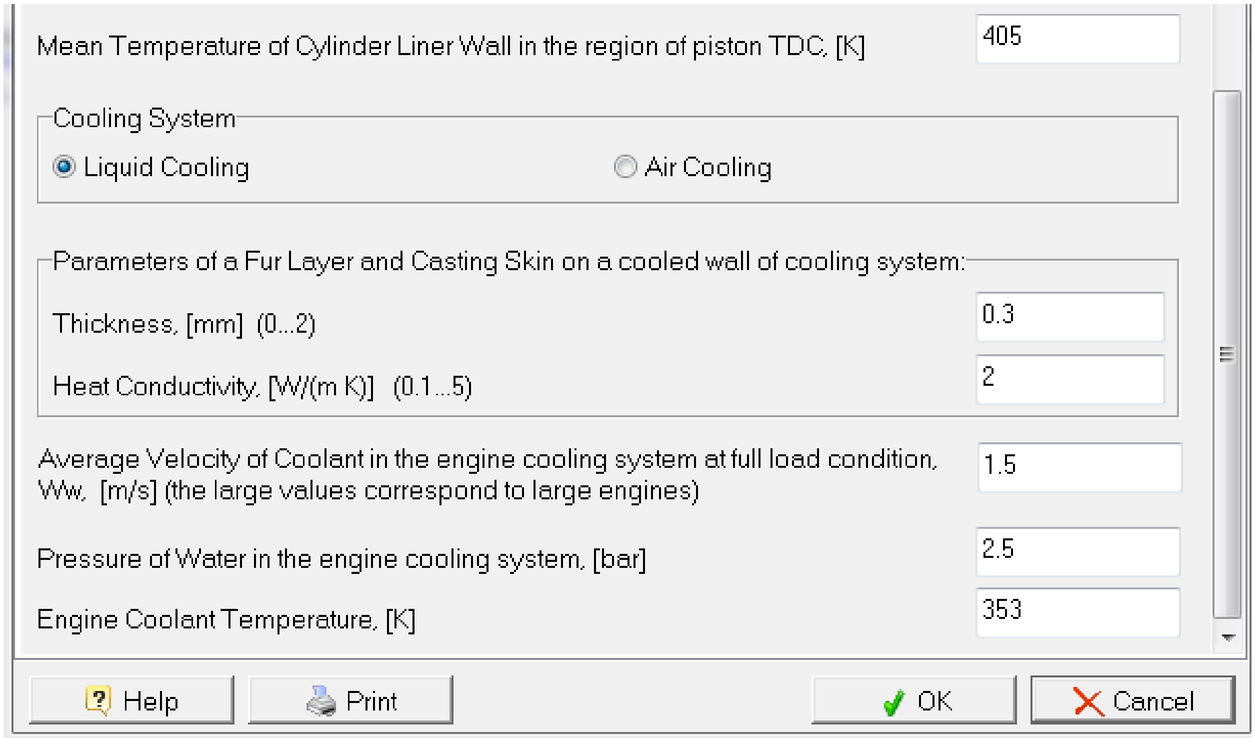


Figure (3-6) type of cooling system

3-5-5 Fuel:

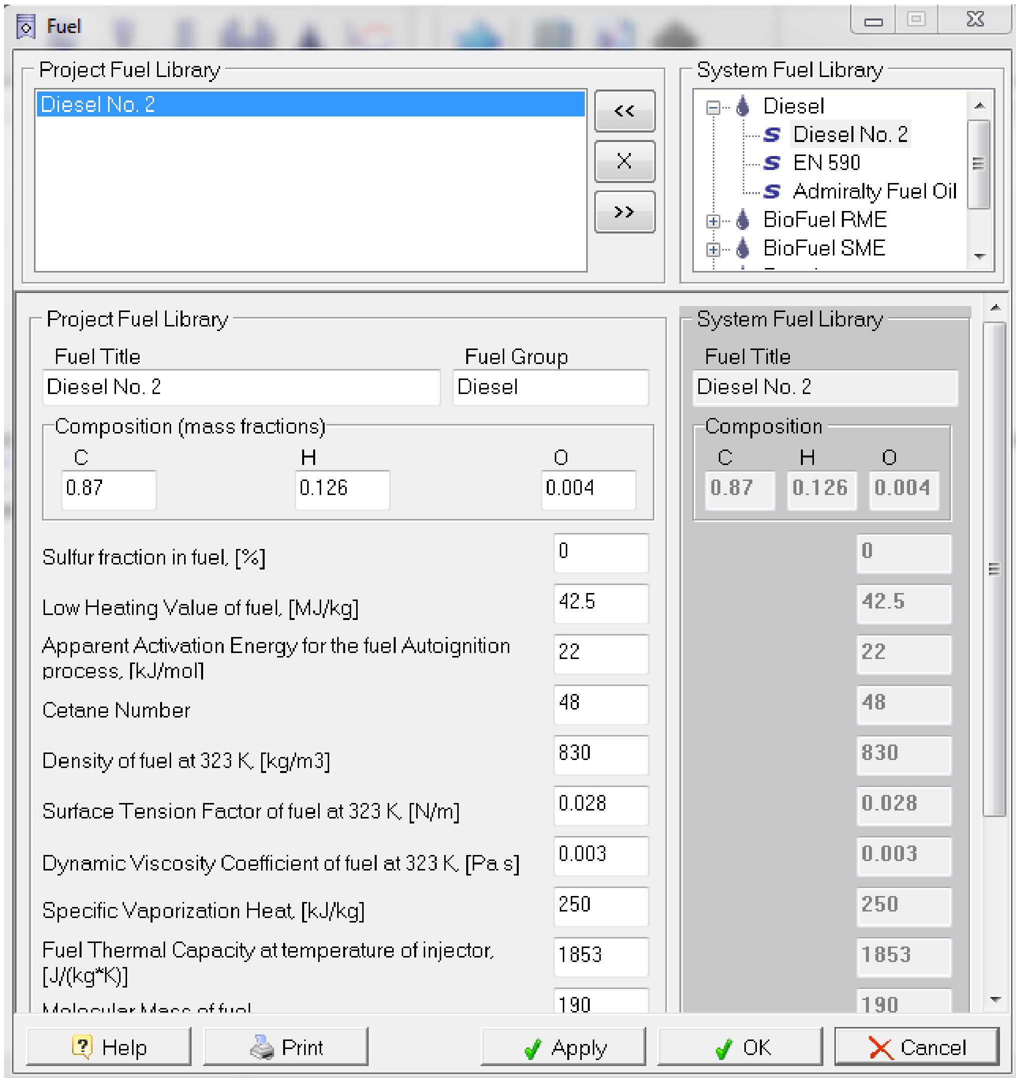


Figure (3-7) fuel

3-5-5-1 Fuel Properties

The following fuel properties are necessary if the fuel is not the diesel oil.

Composition of fuel (C, H, O), [%].

Low Heating Value, [MJ/kg].

Density of fuel at 323 K, [kg/m3].

Specific Vaporization Heat, [J/kg].

Fuel Thermal Capacity at temperature of injector, [J/kg/K].

Molecular Mass (or weight).

Sulfur Fraction in the fuel, S, [%], is used for SO_2 emission calculation.

Calculation in diesel.

Cetane Number is used for auto ignition delay calculation in diesel.

Surface Tension Factor of fuel at 323 K (50°C), *sf*, [N/m].

Dynamic Viscosity Coefficient for different fuels at 323 (50°C) mf, [Pa.s].

Diffusion factor at atmospheric condition, [m/s].

Critical temperature of evaporation, [K].

Saturated vapor pressure at critical temperature, [bar].

Small temperature where saturated vapor pressure is known, [K].

Saturated vapor pressure at noted above small temperature, [bar]. [16]

3-6 Specification of the engine used in the study

Table (3-1) Engine Parameters

Manufacturer	Mitsubishi
Called	4M40
Engine Type	In line – 4Cylinder
Fuel Type	Diesel
Displacement	2835cc
Cylinder Bore	95mm
Piston Stroke	100mm
Cylinder Block	Aluminum Alloy
Cylinder Head	Cast Iron Alloy
Pressure Ratio	21:1
Cooling System	Water-Cooling

3-7 Properties of fuel used in the study

Table (3-2) Fuel Properties

Cetane Number	Minimum 40
Cloud Point	40 degrees Celsius (maximum)
Water and Sediment	0.02% (maximum volume)
Fame Content	7%
Density	0.832 Kg/L
Sulfur Content	500ppm
Heating Value (Calorific)	45.5 MJ/Kg
Flash Point	40 degrees Celsius (minimum)
Viscosity (at 40 degrees Celsius)	(2.5-3.5) mm^2/s
Carbon residue	0.1% (maximum mass)
Ash	0.01%(maximum)
Oxidation Stability	Up to 0.025 mg/ml

3-8 Steps to work on the Diesel-RK software

Step (1)

Click on the program icon to show the main interface of program. And click (create new project):

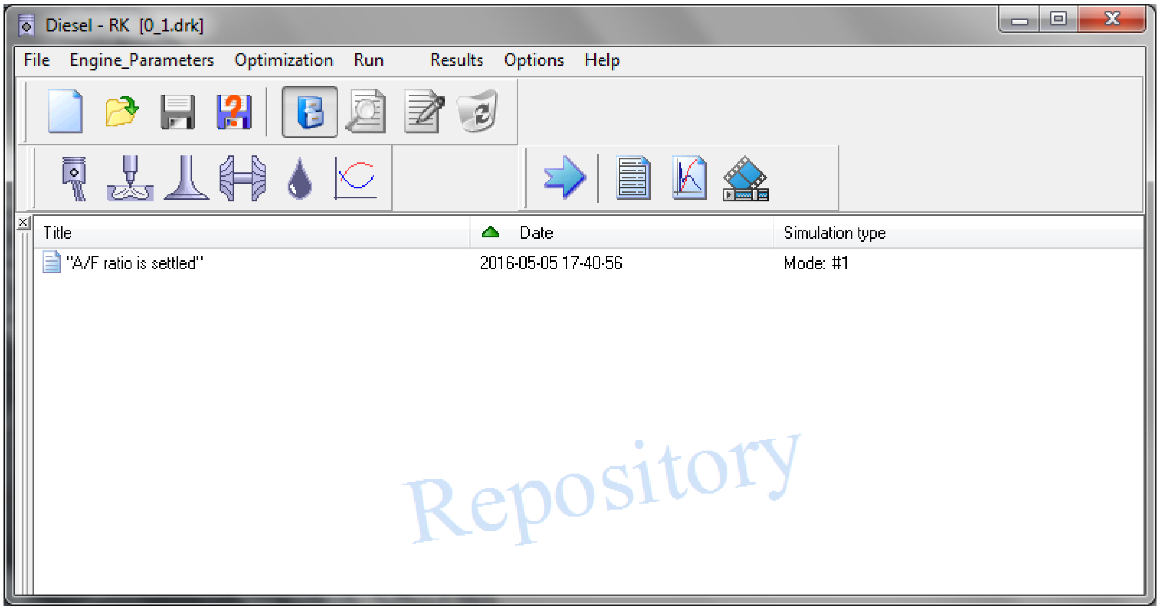


Figure (3-8) main interface of Diesel-RK

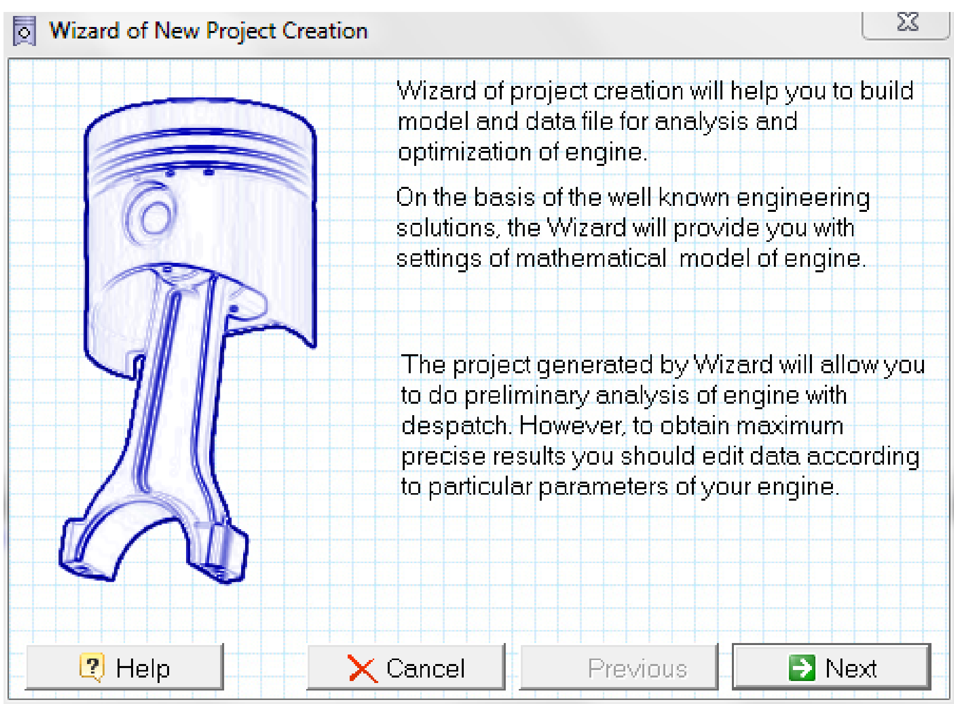


Figure (3-9)new project creation

Step (2)

The second step press (next) to get this information and inter the technical specification of engine.

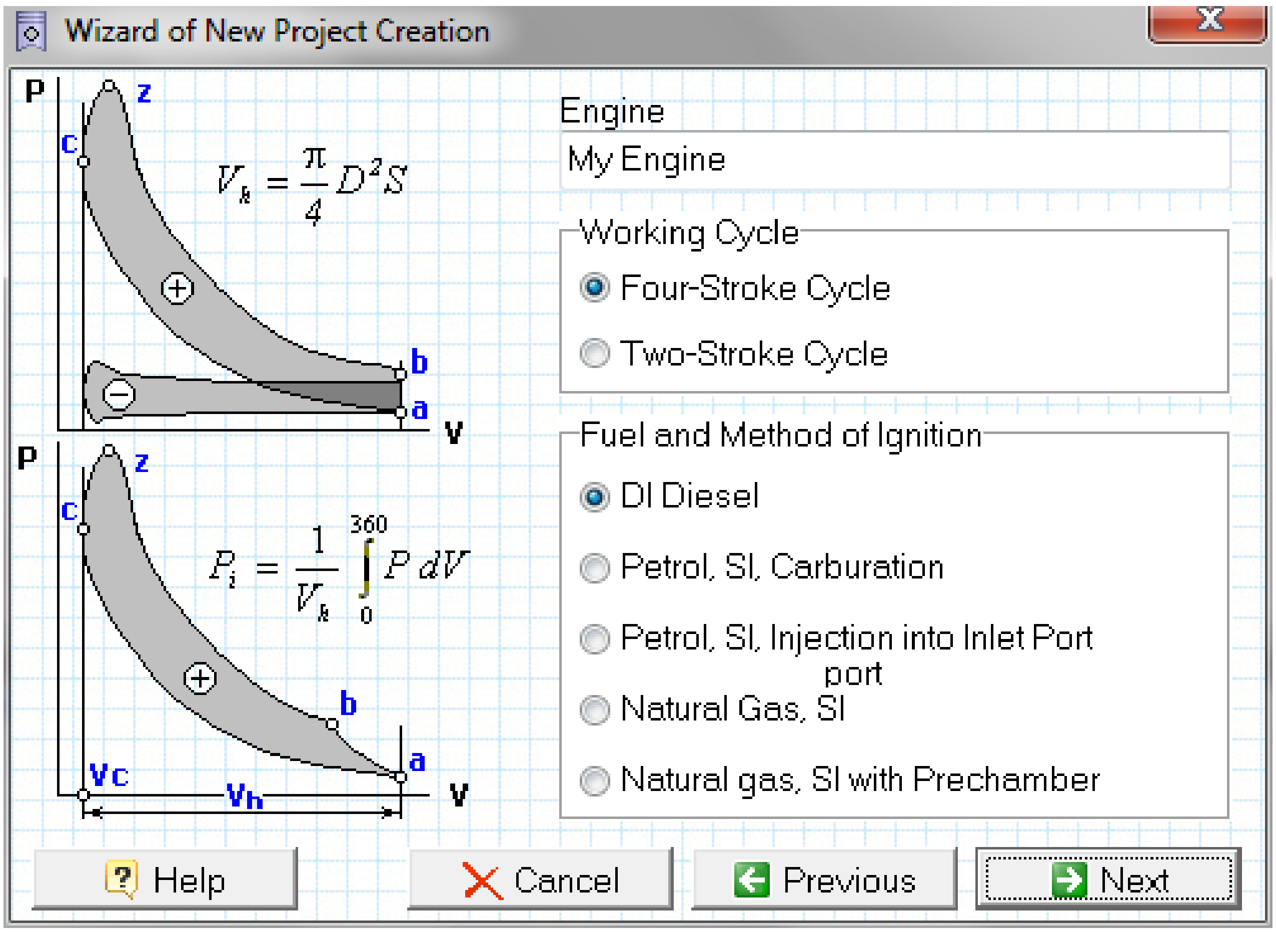


Figure (3-10)specification of engine (a)

And then by clicking on (Next) each time until the intervention all specifications of engine and engine speed.

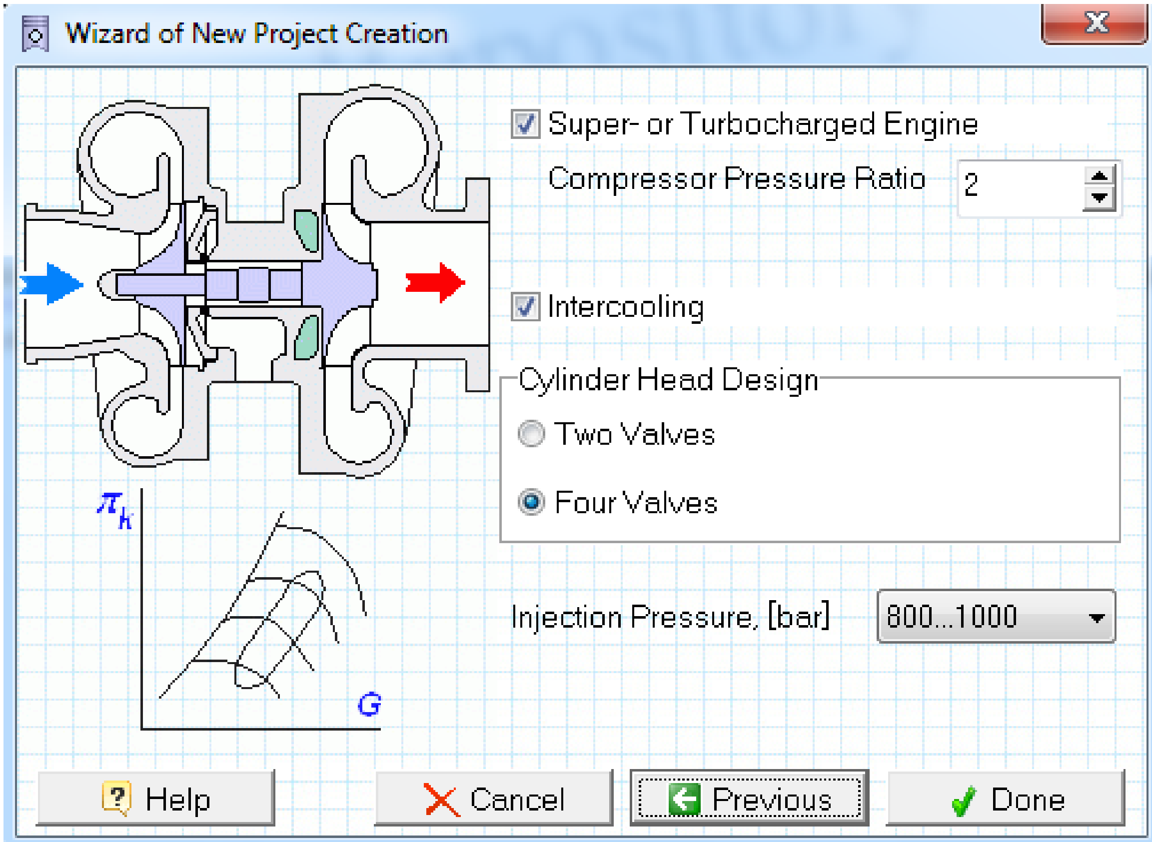


Figure (3-11) specification of engine (b)

Step3:

Enter dimension of engine in the general parameters

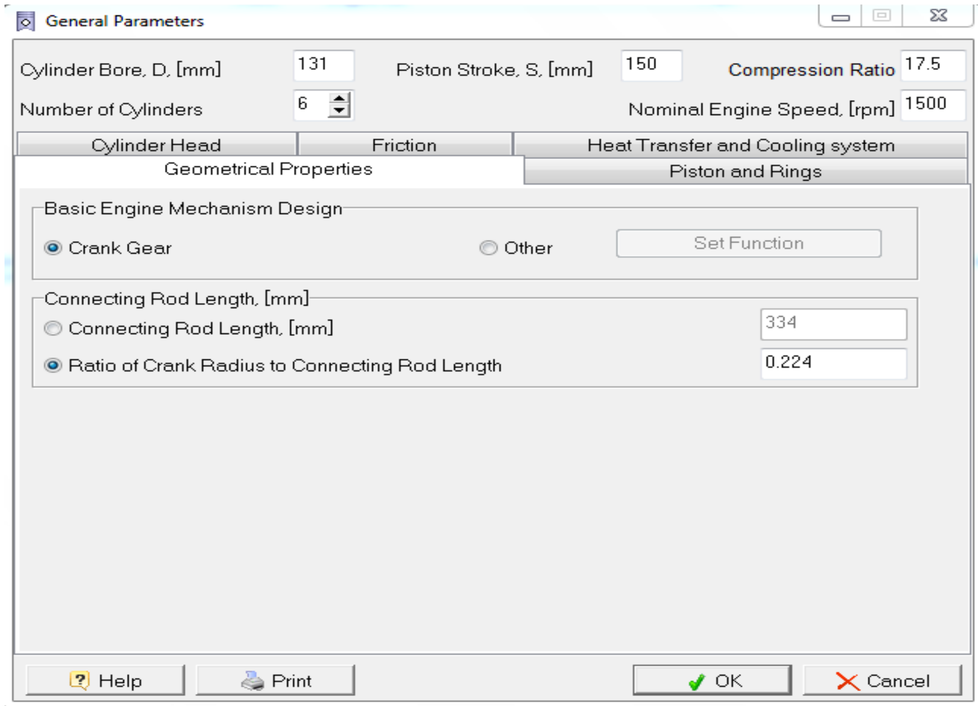


Figure (3-12) General parameters

Step 4:

Selection type of fuel used (Diesel Sudan)

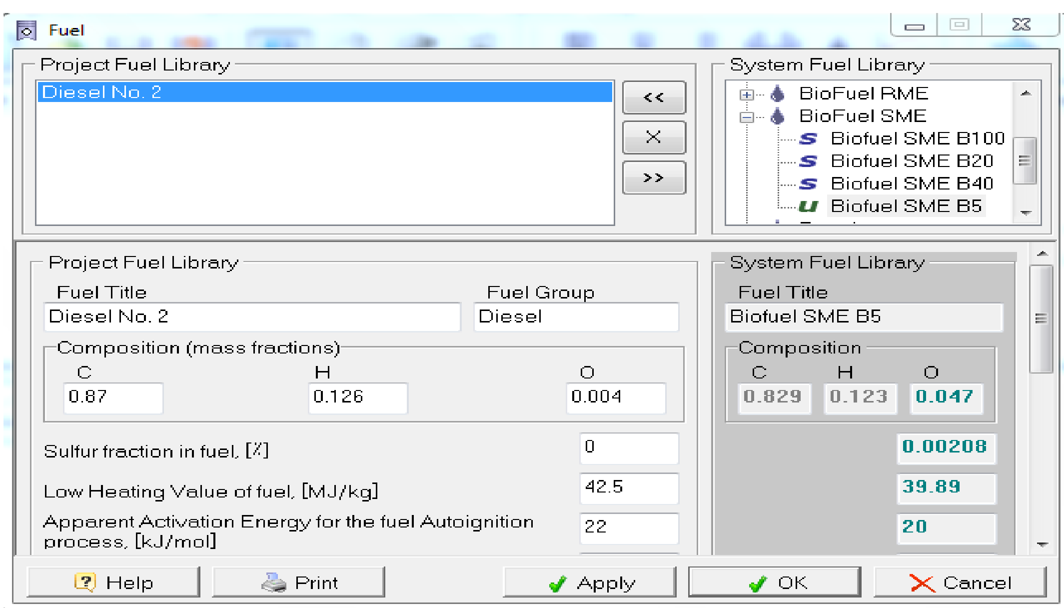


Figure (3-13) fuel

Step 5:

Pre-run must be connected to the internet and you have a username in the program which get it by sends a message to the work of the program team and then click run and press ICE simulation.

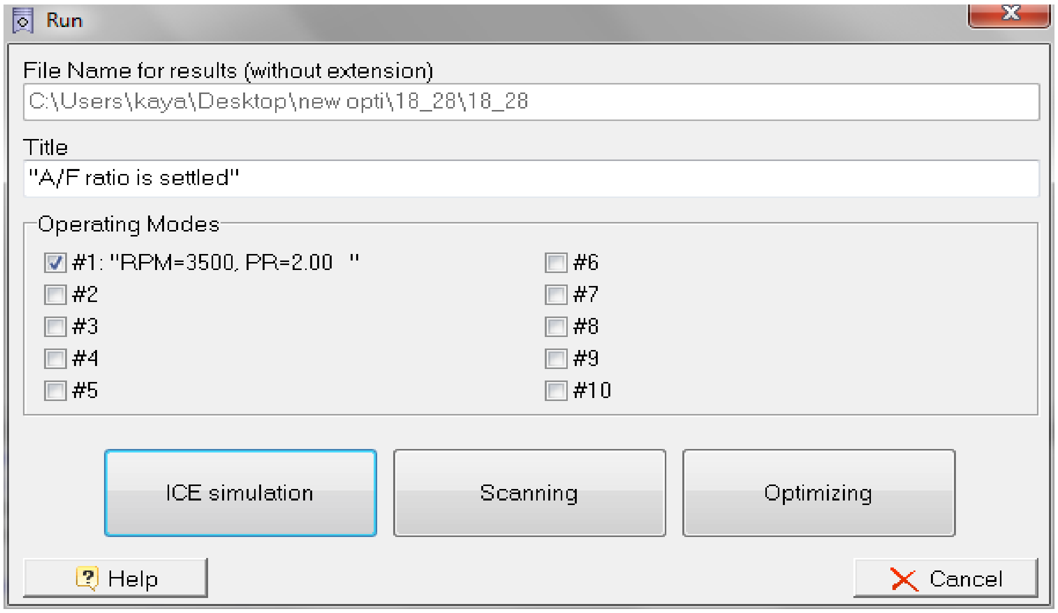
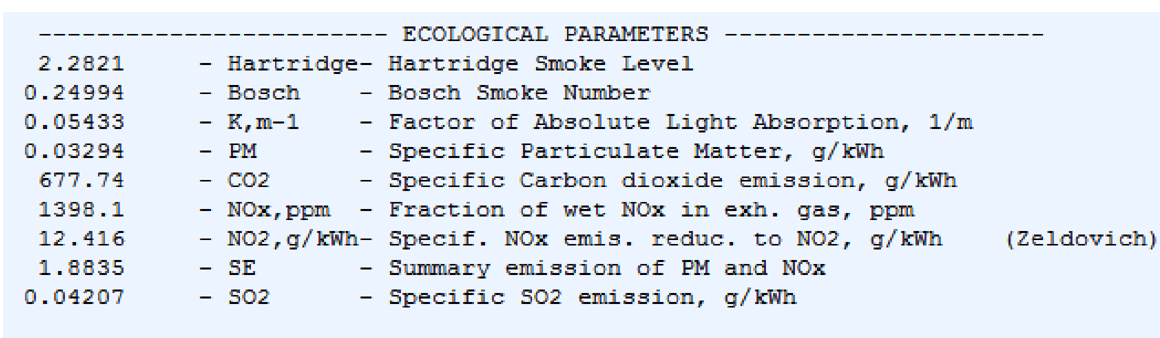


Figure (3-14) run-ICE simulation

Step6:

Getting a result:-



```
------------------------ ECOLOGICAL PARAMETERS ----------------------
 2.2821      - Hartridge- Hartridge Smoke Level
0.24994      - Bosch     - Bosch Smoke Number
0.05433      - K,m-1     - Factor of Absolute Light Absorption, 1/m
0.03294      - PM        - Specific Particulate Matter, g/kWh
 677.74      - CO2       - Specific Carbon dioxide emission, g/kWh
 1398.1      - NOx,ppm   - Fraction of wet NOx in exh. gas, ppm
 12.416      - NO2,g/kWh- Specif. NOx emis. reduc. to NO2, g/kWh    (Zeldovich)
 1.8835      - SE        - Summary emission of PM and NOx
0.04207      - SO2       - Specific SO2 emission, g/kWh
```

Figure (3-15) result

CHAPTER FOUR

RESULTS AND DISCUSSION

4-1 The results

4-1-1 without a turbocharger

Table (4-1) the amount of exhaust emissions in the absence of a turbocharger

Engine Speed[rpm]	PM [g/kwh]	CO_2 [g/kwh]	NO_X [ppm]	NO_2 [g/kwh]	SO_2 [g/kwh]
700	0.19873	830.38	1698.1	18.514	0.05154
1000	0.19976	827.04	1685.6	18.289	0.000
1500	0.13426	723.58	1506.9	14.317	0.04491
2000	0.14339	688.39	1179.9	10.654	0.000
2500	0.19058	688.30	938.40	8.4665	0.000
3000	0.20721	701.09	884.47	8.1256	0.000
3500	0.23323	721.93	893.63	8.4570	0.000
4000	0.27700	753.47	910.04	9.0003	0.000

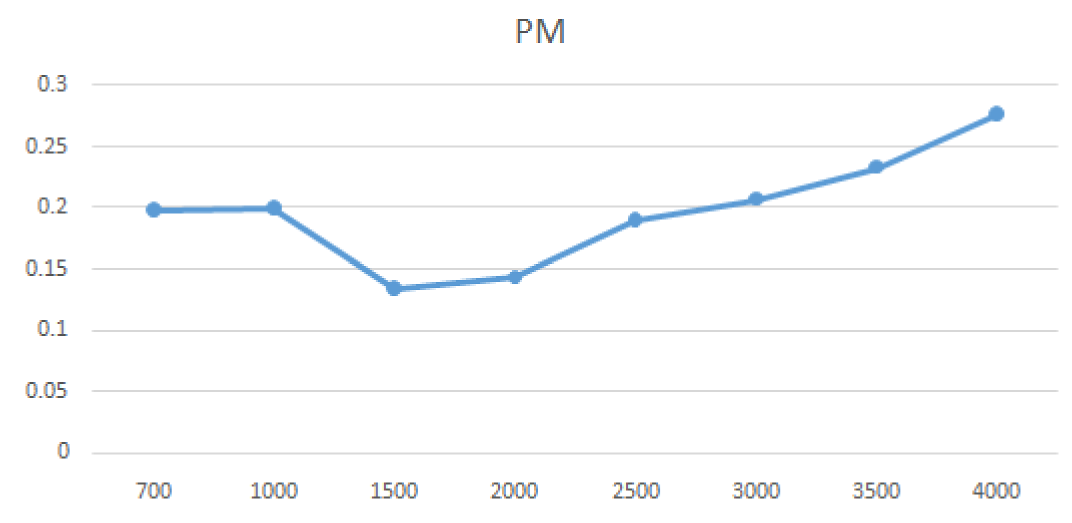


Figure (4-1) Emission of PM at different speed of engine

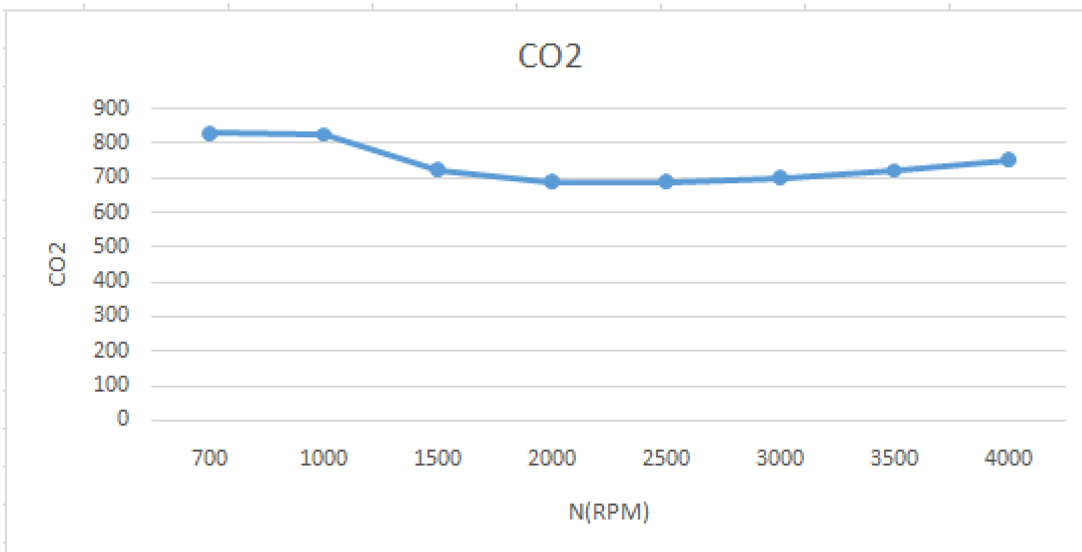


Figure (4-2) Emissions of CO_2 at different speed of engine

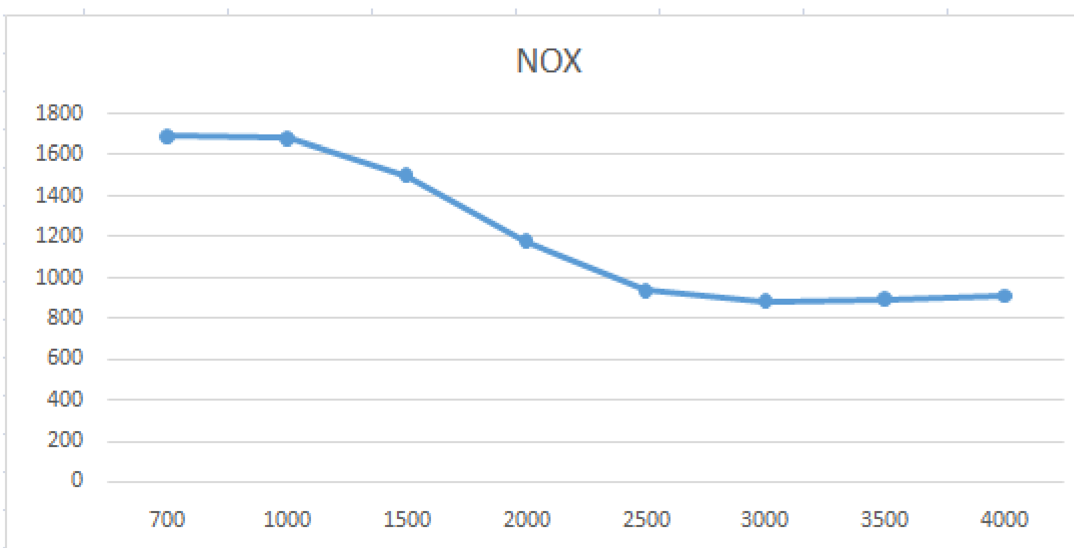


Figure (4-3) Emissions of NO_X at different speed of engine

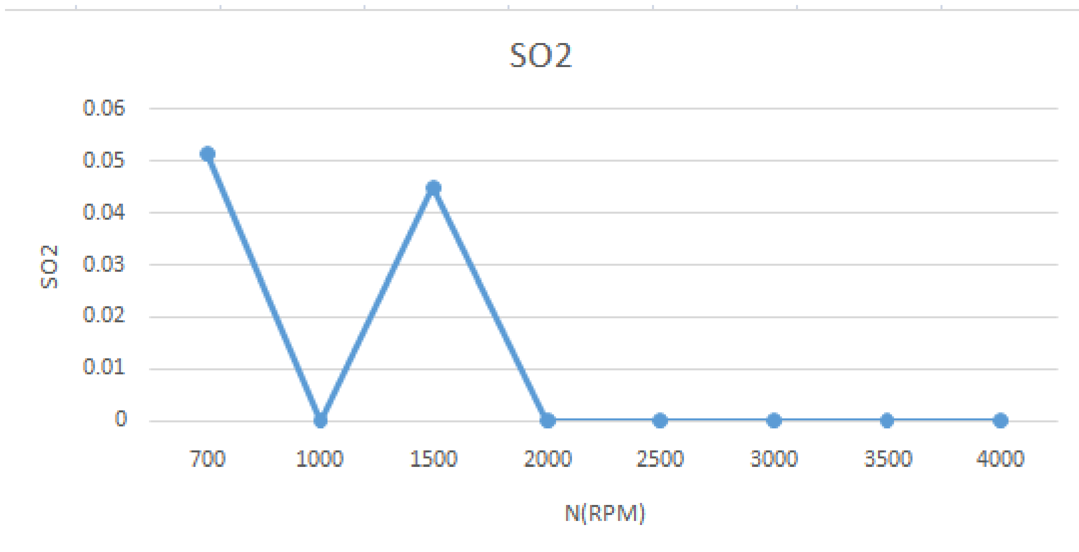


Figure (4-4) Emissions of SO_2 at different speed of engine

4-1-2 at pressure ratio (PR=1.5)

Table (4-2) the amount of exhaust emissions at pressure ratio (PR=1.5) for a turbocharger

Engine Speed[rpm]	PM [g/kwh]	CO2 [g/kwh]	NO(x) [ppm]	NO2 [g/kwh]	SO2 [g/kwh]
700	0.09044	1296.3	1917.3	32.793	0.0550
1000	0.03990	862.51	2139.0	24.279	0.0536
1500	0.02186	729.91	9852.5	17.750	0.0453
2000	0.02317	688.77	1617.0	14.605	0.04275
2500	0.03294	677.74	1398.1	12.410	0.04207
3000	0.02731	683.68	1336.8	11.976	0.04245
3500	0.06354	699.07	1236.7	11.327	0.04339
4000	0.11044	723.92	1167.6	11.087	0.04493

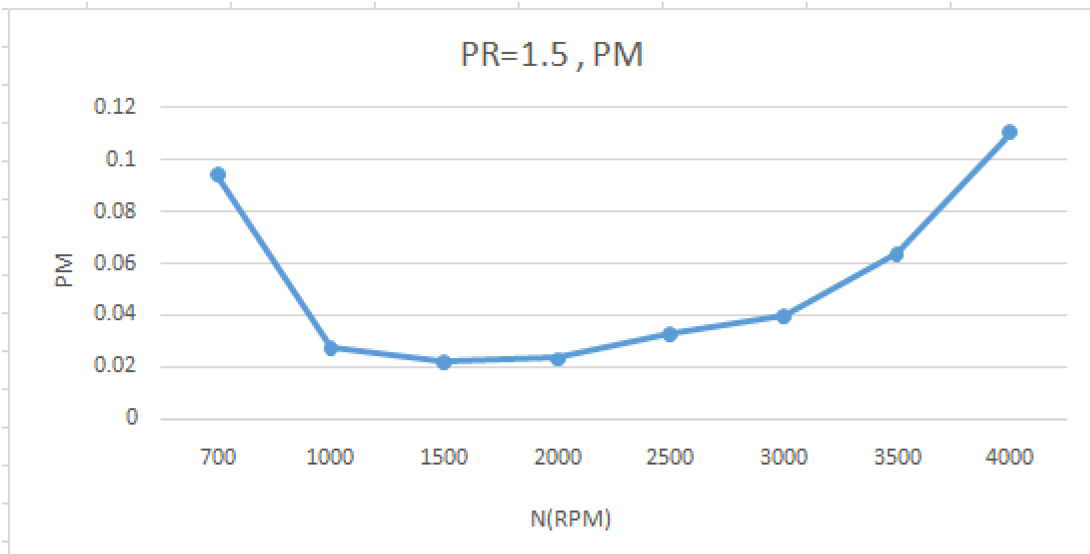


Figure (4-5) Emission of PM at PR=1.5

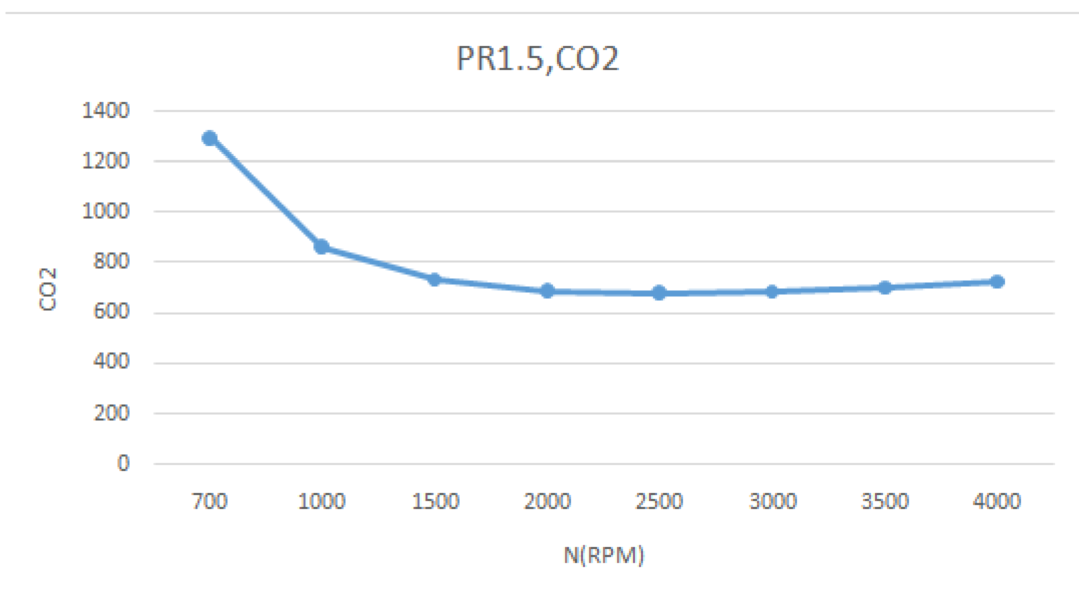


Figure (4-6) Emission of CO2 at PR=1.5

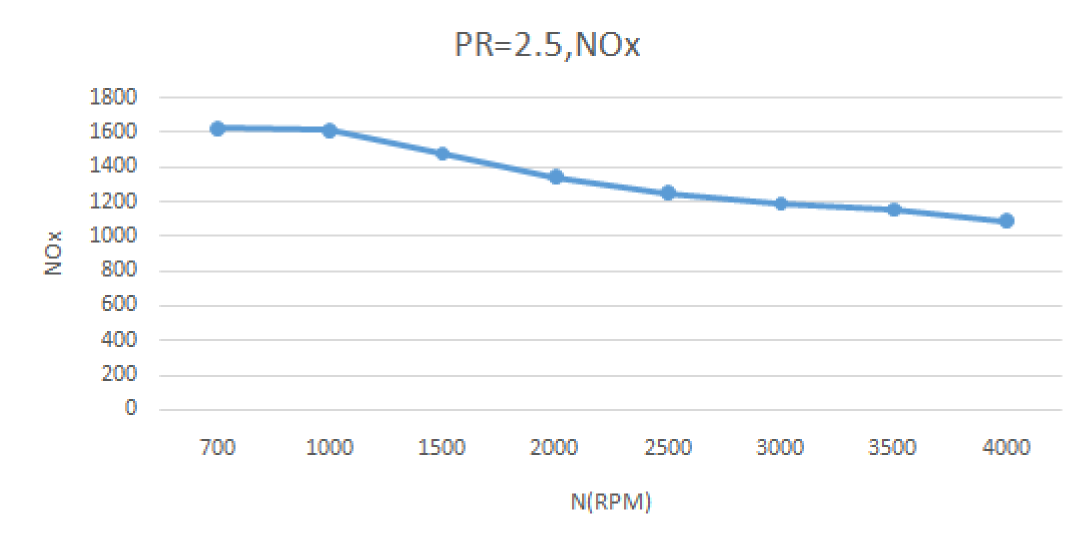


Figure (4-7) Emission of NO_X at PR=1.5

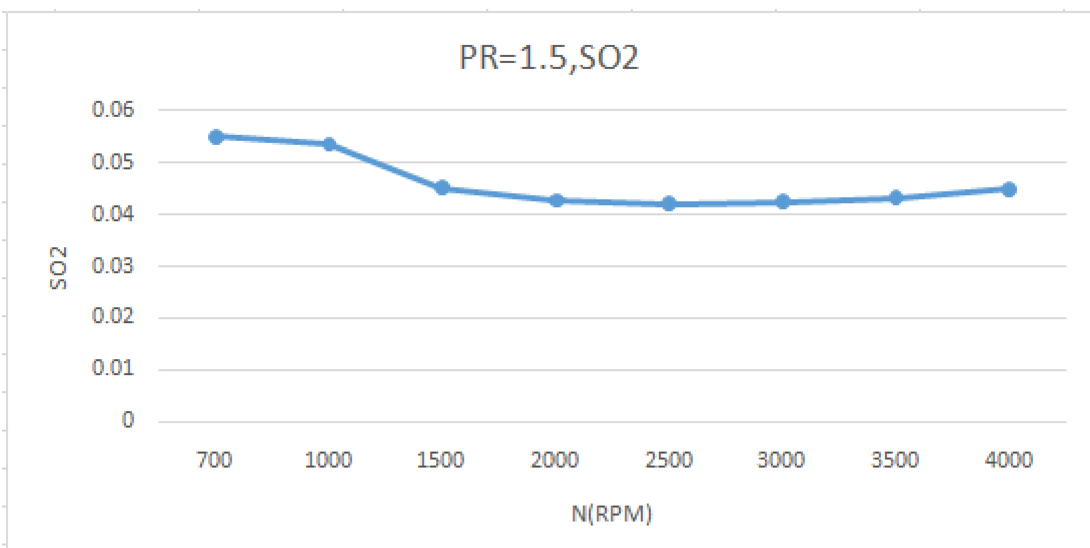


Figure (4-8) Emission of SO_2 at PR=1.5

4.1.3At pressure ratio (PR=2)

Table (4-3) amount of exhaust emissions at pressure ratio (PR=2) for a turbocharger

Engine Speed[rpm]	PM [g/kwh]	CO2 [g/kwh]	NO(x) [ppm]	NO2 [g/kwh]	SO2 [g/kwh]
700	0.05531	887.90	1600.9	21.145	0.05511
1000	0.03112	762.86	1553.3	17.615	0.04735
1500	0.02309	692.04	1402.9	14.438	0.04295
2000	0.02370	672.33	1263.2	12.628	0.04173
2500	0.02773	670.34	1162.8	11.618	0.04161
3000	0.03898	680.63	1073.0	10.856	0.04225
3500	0.05316	694.78	1028.6	10.627	0.04312
4000	0.09666	719.68	936.32	10.033	0.04467

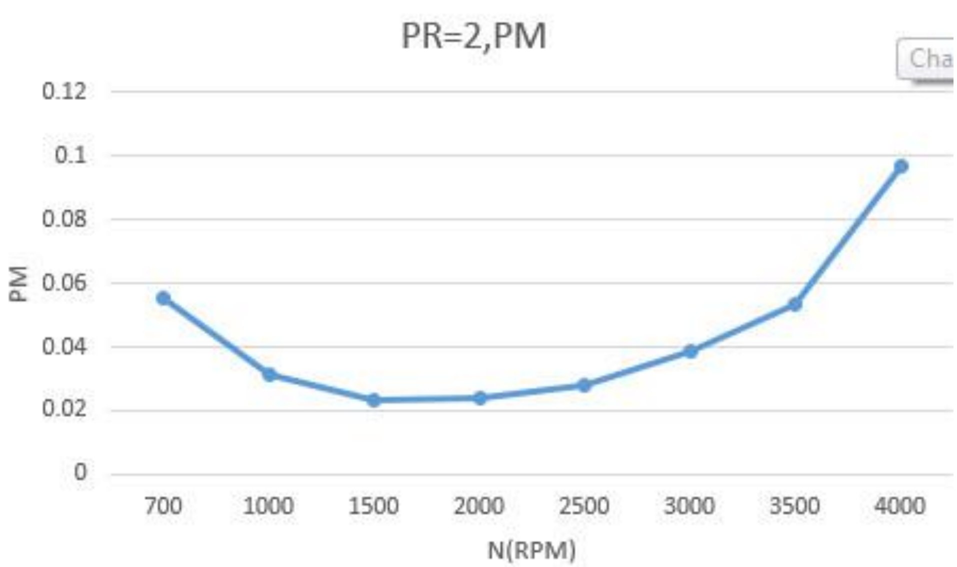


Figure (4-9) Emission of PM at PR=2

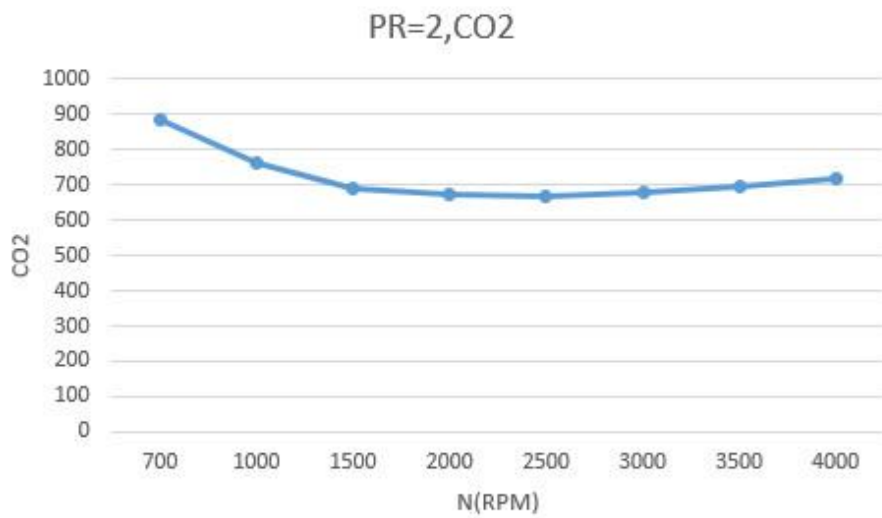


Figure (4-10) Emission of CO_2 at PR=2

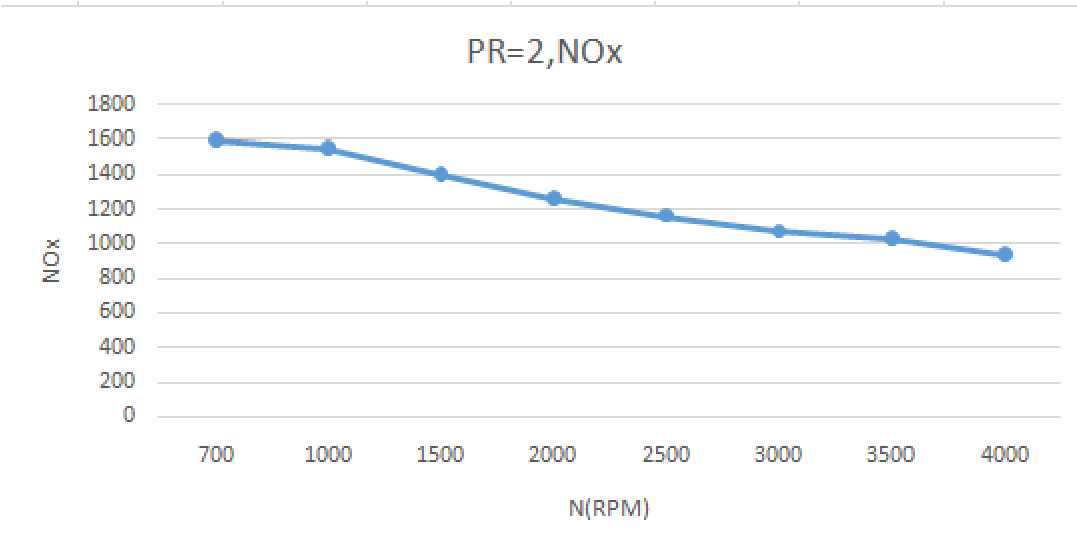


Figure (4-11) Emission of NO_X at PR=2

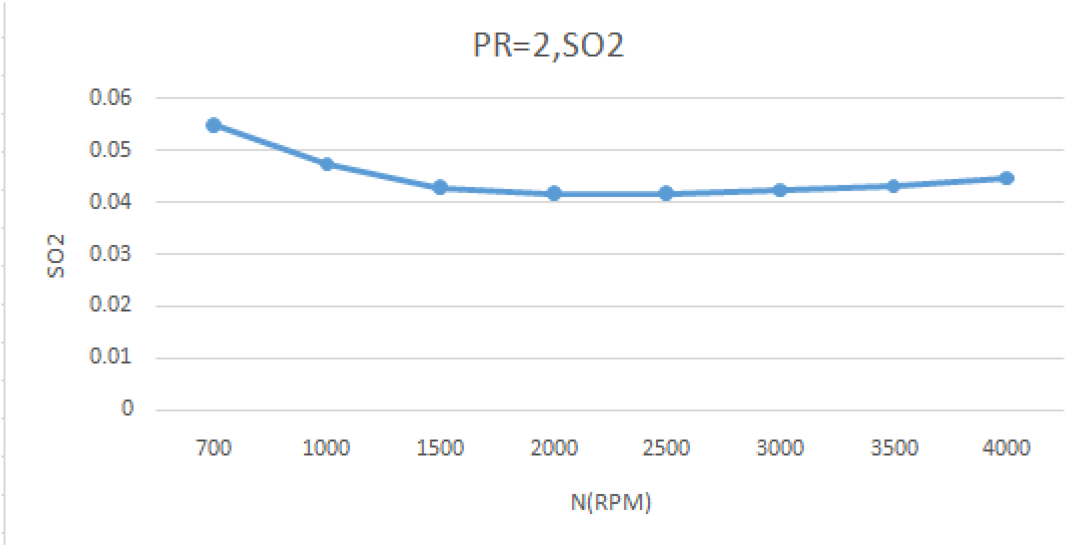


Figure (4-12) Emission of SO_2 at PR=2

4-1-4 at pressure ratio (PR=2.5)

Table (4-4) amount of exhaust emissions at pressure ratio (PR=2.5) of a turbocharger

Engine Speed[RPM]	PM [G/KWH]	CO_2 [G/KWH]	NO_X [PPM]	NO_2 [G/KWH]	SO_2 [G/KWH]
700	0.01916	936.48	1632.3	22.822	0.05813
1000	0.00915	780.56	1616.0	18.783	0.04845
1500	0.00671	699.26	1483.1	15.436	0.04340
2000	0.00753	693.47	1348.4	13.514	0.04180
2500	0.00995	668.55	1258.3	12.514	0.04150
3000	0.01755	674.17	1191.1	11.943	0.04184
3500	0.03772	684.52	1159.9	11.810	0.04249
4000	0.07104	701.25	1091.6	11.397	0.04353

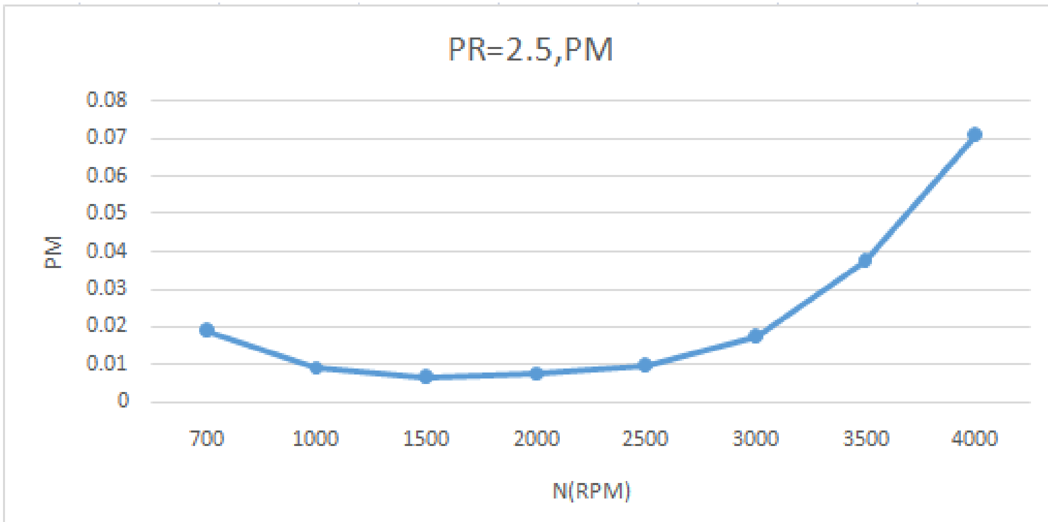


Figure (4- 13) Emission of PM at PR=2.5

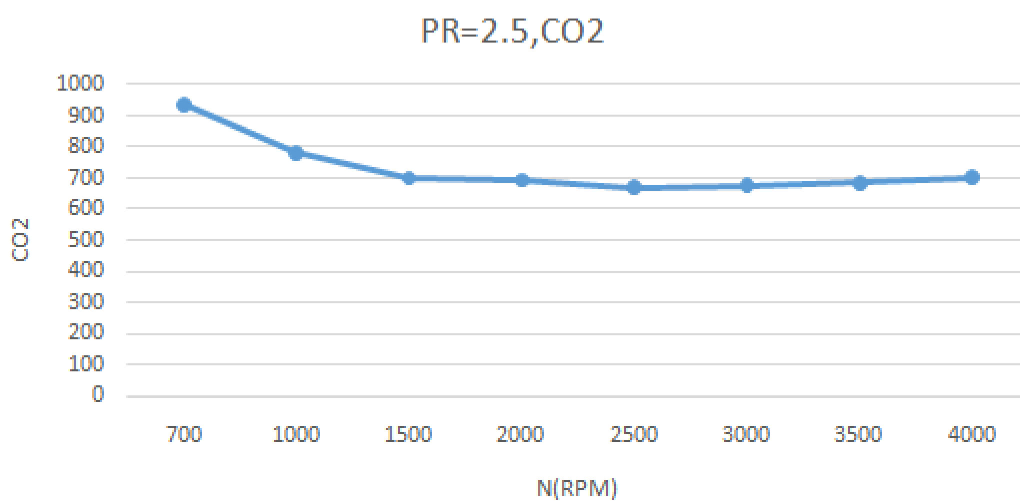


Figure (4-14) Emission of CO2 at PR=2.5

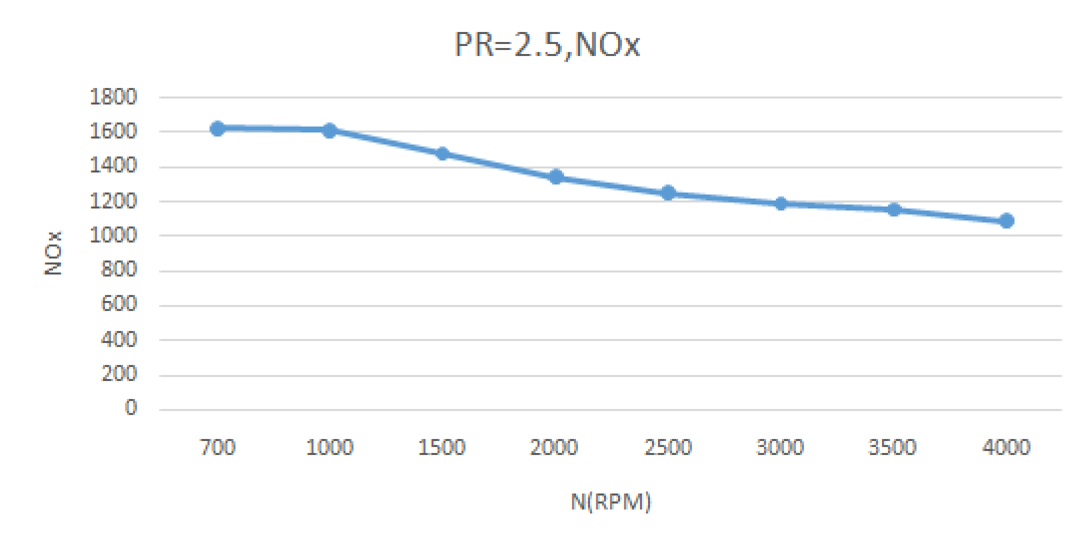


Figure (4-15) Emission of NO_X at PR=2.5

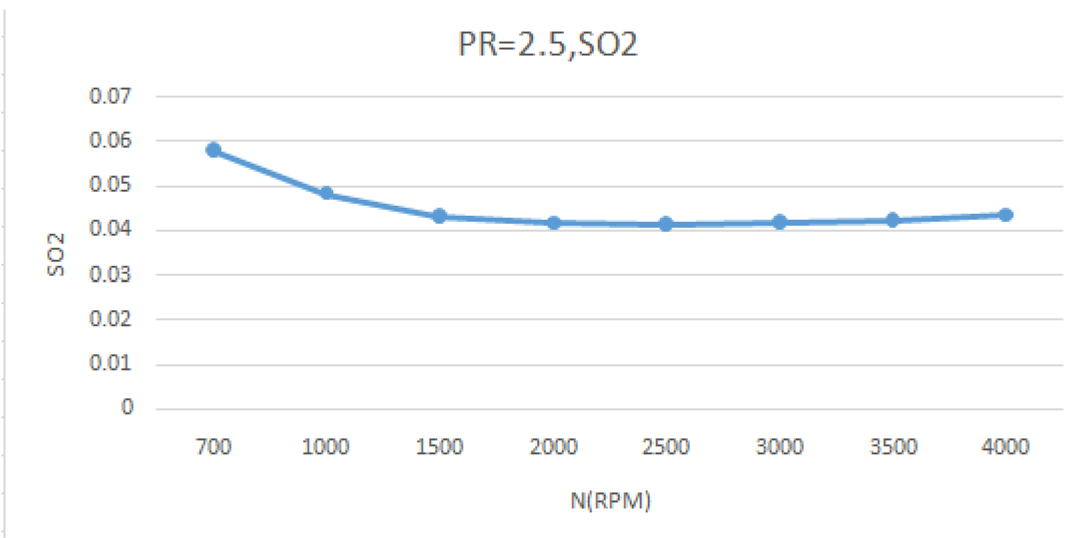


Figure (4-16) Emission of SO2 at PR=2.5

4-1-5 at pressure ratio (PR=3)

Table (4-5) amount of exhaust emissions at pressure ratio (PR=3) of a turbocharger

Engine Speed(RPM)	PM(G/KWH)	CO_2(G/KWH)	NO_X (G/KWH)	SO_2(G/KWH)
700	0.00730	991.06	1639.1	0.06151
1000	0.00313	803.02	1645.6	0.04984
1500	0.00182	709.45	1560.6	0.04403
2000	0.00288	679.23	1420.9	0.04216
2500	0.00735	668.51	1368.3	0.04149
3000	0.02146	670.64	1317.8	0.04163
3500	0.04618	680.32	1253.7	0.04223
4000	0.07122	696.65	1136.2	0.04324

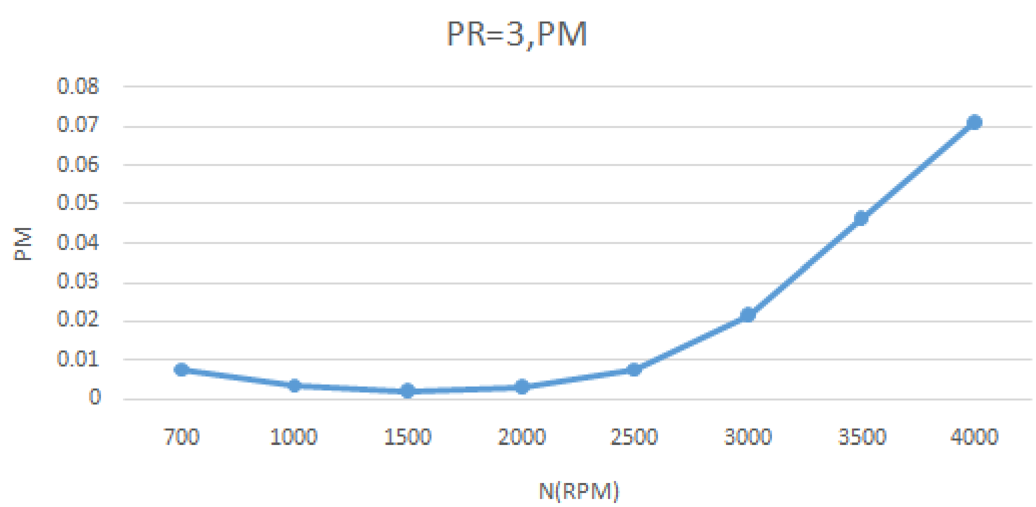


Figure (4-17) Emission of PM at PR=3

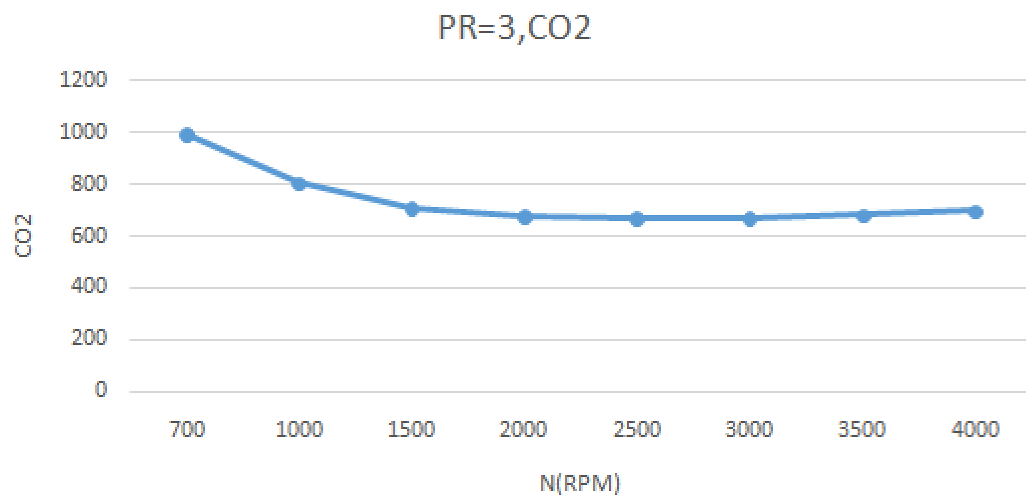


Figure (4-18) Emission of CO_2 at PR=3

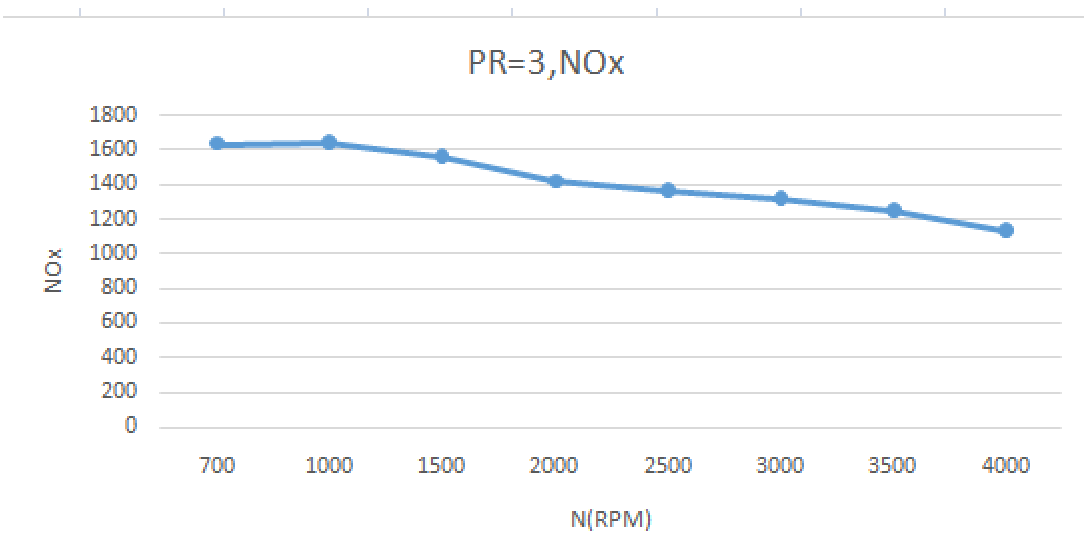


Figure (4-19) Emission NO_X at PR=3

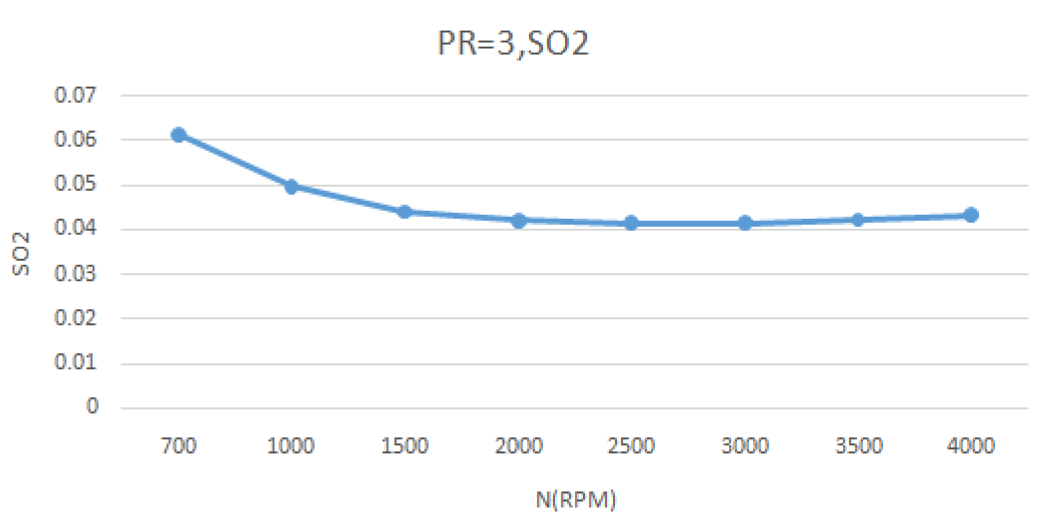


Figure (4-20) Emission of SO_2 at PR=3

4.2 Discussion

4.2.1 Particulate Metter (PM) Emissions

What we can be observed here and especially if the turbocharger in not used at figure (4.1), the value of PM fluctuates at the low speed of the engine .However, when the speed of the engine reaches 1500rpm, here we find that this relationship between the PM and the speed of rotation of the engine is a direct link .if the speed of the rotation increases, the value of PM is increased with value (0.0655G/KWH).

When using a turbocharger : for example a turbocharger with PR=1.5, at figure(4.5) we find that the value of PM at the beginning decreases when the engine speed is increased, but this decrease occurs only at the low speed of the engine .when the speed of the engine reach to 1500rpm here we find that relationship between PM and speed of rotation became a direct link (it is tapped by increasing RPM).we find that the value of the increase in PM is less than that without a turbocharger of (0.0469G/KWH).this indicates that the turbocharger has the most recent significant change in the amount of PM at high speed .

Another example :at PR=3,at figure (4.17) we find that the value of PM at low speed decreases when increasing the speed of the engine when it reach 2500rpm .the value begins to increase but with very little value (0.0250G/KWH).

Hence, when increasing the pressure ratio of the turbocharger, reduce the amount of PM at a high speed of engine. This is observed from the chart below:

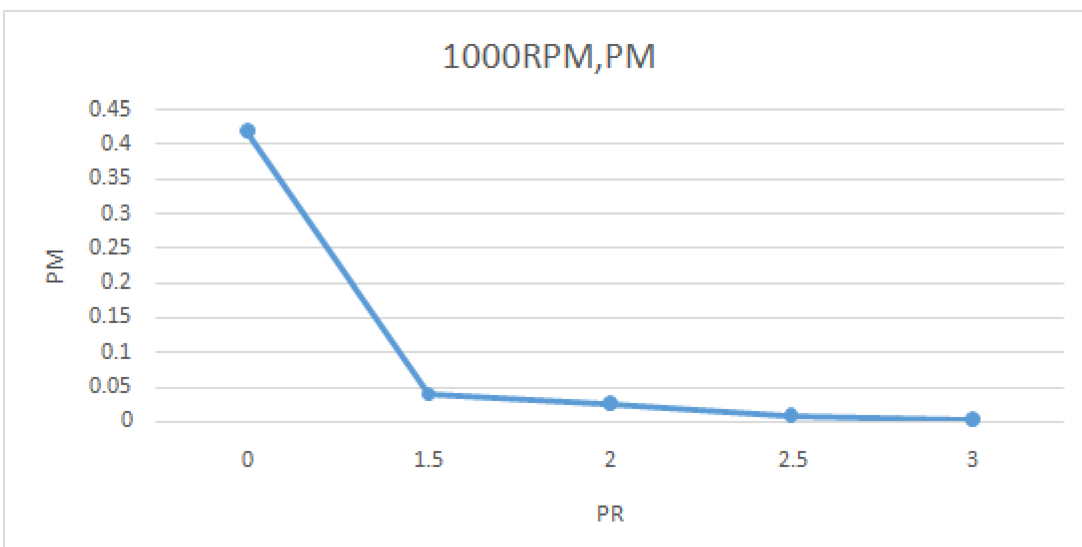


Figure (4-21) Emission of PM at 1000RPM

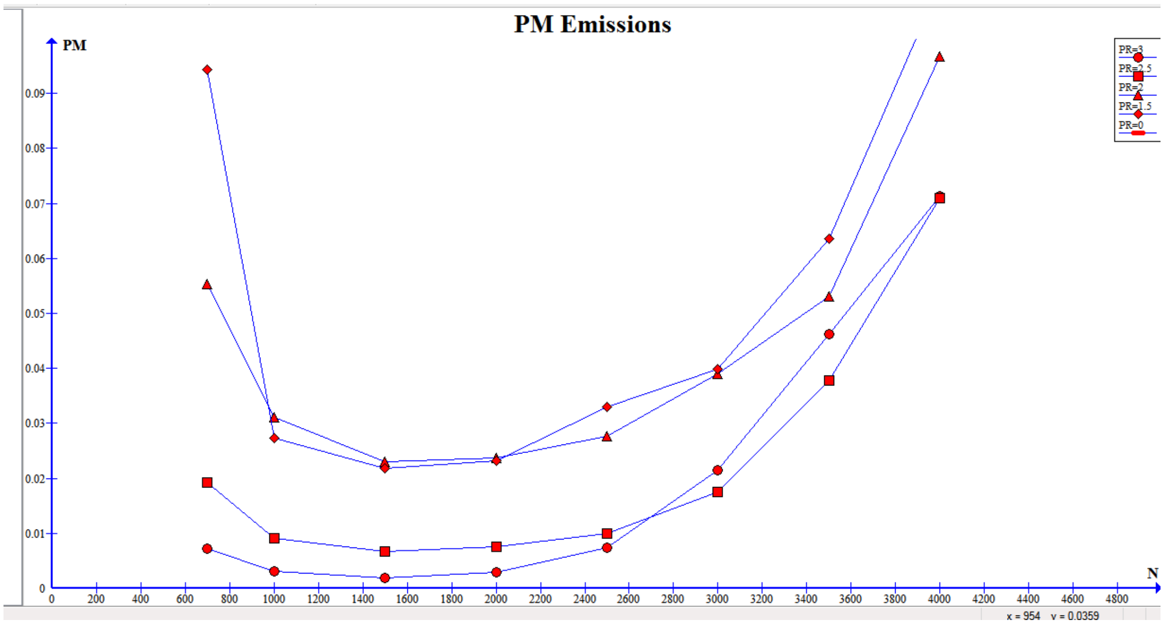


Figure (4-22) PM Emissions

4.2.2 CO_2 Emissions

Figure (4.2) diagram shows the relationship between engine speed and CO2 emissions in the absence of a turbocharger .from this diagram it is clear: the proportion of CO2 emitted from the exhaust decreases when the engine speed increases, but when the engine speed reaches 2500rpm here we find that the proportion of CO2 emitted from the engine has begun to increase but with very little value.

In the case of a turbocharger as shown in: figure(4.6),figure(4.10), figure(4.14),figure(4.18)at PR=1.5, PR=2, PR=2.5, PR=3 :here we can be observed that the CO2 emissions from the exhaust decrease when the engine speed increases but this drop is in a very narrow range .the relationship between the CO2 emissions and the speed of rotation of the engine is almost a linear relationship, especially at speeds of (1500-4000rpm) for the engine .

On the other hand, if we describe the relationship between the emission of CO_2 and the pressure ratio of the turbocharger at a certain speed, for example at the speed of 700rpm of the engine ,at this chart below:

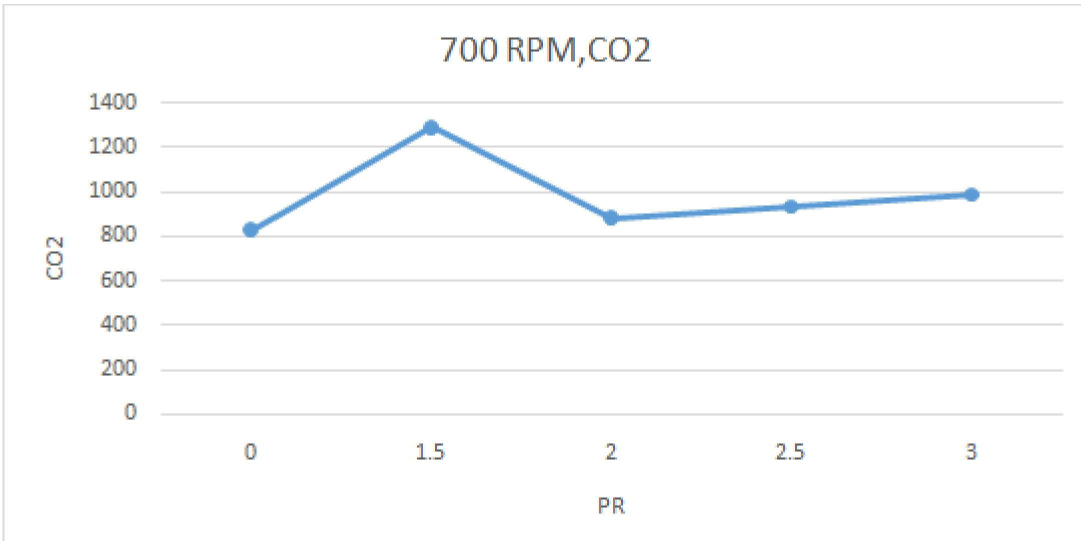


Figure (4-23) Emission of CO2 at 700RPM

here we find that in the case where the turbocharger was not used, the ratio of CO2 emitted from the engine was minimal .once we added a turbocharger at PR=1.5 the value of CO2 emitted from the engine has increased .when we increase the pressure ratio of the turbocharger to PR=2 we find that the value of CO2 emitted from the engine has decreased .this means that if a car travels at a speed of 700rpm, it can't guarantee a fixed value for the amount of CO2 emitted from the engine, from this point of view it was necessary to know a certain speed through which the constant change in the amount of CO2 emitted from the engine can be described at different pressure ratios for the turbocharger .

When taking another example of the speed of rotation of a motor, for example at speed of 2500rpm, as shown in the chart below:

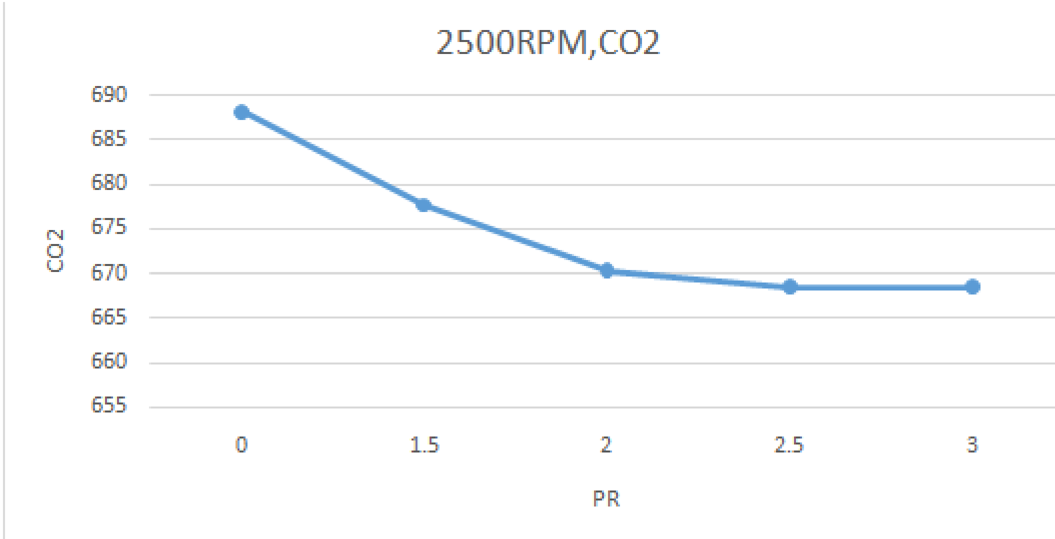


Figure (4-24) Emission of CO2 at 2500RPM

In this chart can be observed that the value of CO_2 emitted in the absence of a turbocharger is the maximum value for it.

When we added a turbocharger at PR=1.5 here the percentage of CO_2 emitted from the engine decreased. As the pressure ratio of the turbocharger increases, we find that the value of CO_2 is decreasing and this is happening consistently .here we can describe the relationship between the CO_2 emitted from the engine and the increase in the pressure ratio of the turbocharger as an inverse relationship. This occurs only when the engine speed is 2500rpm. From here we can say that the ideal speed that the engine should rotate in the case of the use of a turbocharger is the speed of 2500rpm in order to maintain the minimum amount of CO_2.

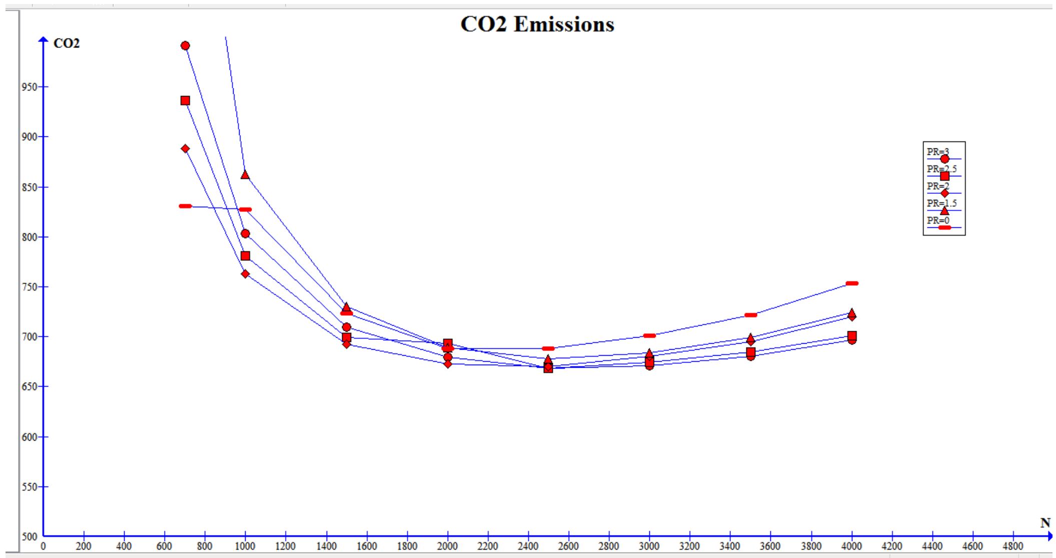


Figure (4-25) CO_2 Emissions

4.2.3 NO_X Emissions

The diagram at figure (4.3) shows the relationship between engine speed and NO_X emitted from the engine. Looking at the chart we can see that the NO_X emission rate decreases gradually as the engine's speed is increased. This happens if the turbocharger is not used.

If we use a turbocharger at difference pressure ratio for example, PR = 1.5, PR = 2, PR = 2.5, PR = 3, shown in the figure (4.7),figure (4.11), figure(4.15), figure(4.19). The NO_X ratio of the engine is reduced when the motor speed is increased. The relationship between the speed of rotation of the engine and the ratio of NO_X emitted from the engine as an inverse relationship.

But if we compare the engine in the case of a turbocharger with the same engine but in the absence of a turbocharger from a different perspective by the pressure ratio of the turbocharger on NO_X emission at a specified speed For example, at a speed of 1000rpm at the chart below:

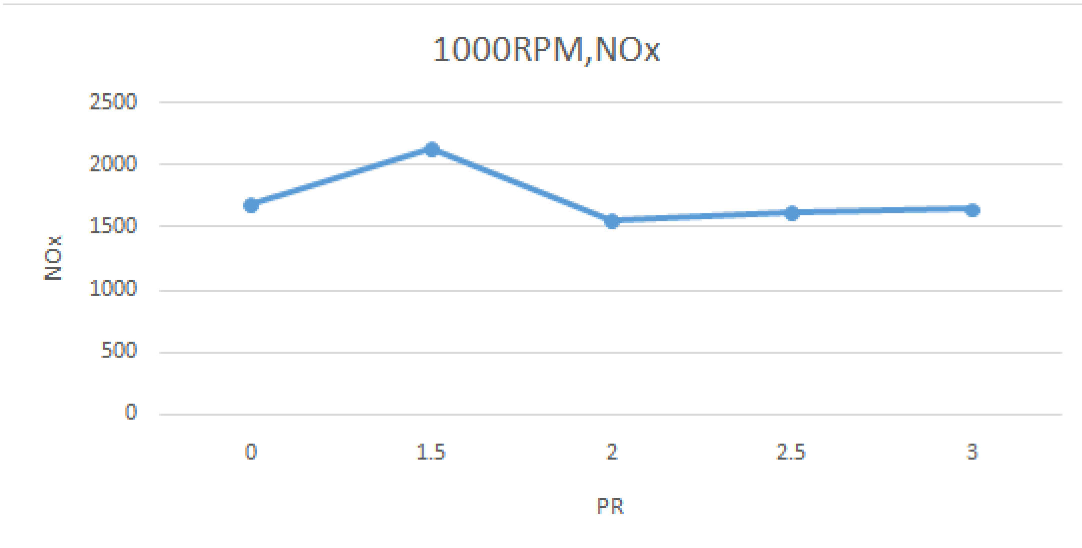


Figure (4-26) Emission of NO(x) at 1000RPM

Here we find that if the turbocharger is not used, the amount of NO_X emitted from the engine is 1685.6. But when adding the turbocharger at a pressure of PR = 1.5 we find that the amount of NO_X has increased. When increasing the pressure ratio of the turbocharger to PR = 2 here, the ratio of NO_X emitted from the motor is reduced as shown in the diagram. Hence, we can say that the amount of NO_X emitted from the engine increases and decreases when the pressure ratios of the turbocharger are changed.

If we take another example at a speed of 1500rpm at the chart below:

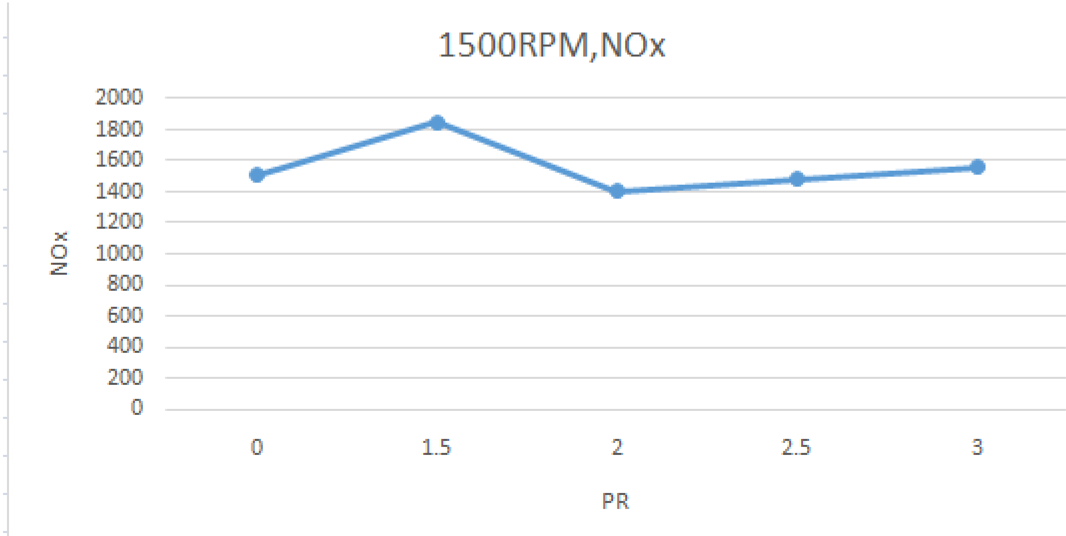


Figure (4-27) Emission of NO(x) at 1500RPM

We find that the amount of NO_X increased in the case of the turbocharger and the pressure of PR = 1.5 and decreased when increasing the pressure ratio to PR = 2 and increased when the ratio of pressure to PR = 2.5

We can be observed is that the minimum amount of NO_X emitted from the engine can be obtained with a turbocharger at PR = 2. Hence, when the objective is to reduce the amount of NO_X emitted, a turbocharger should be used at PR = 2

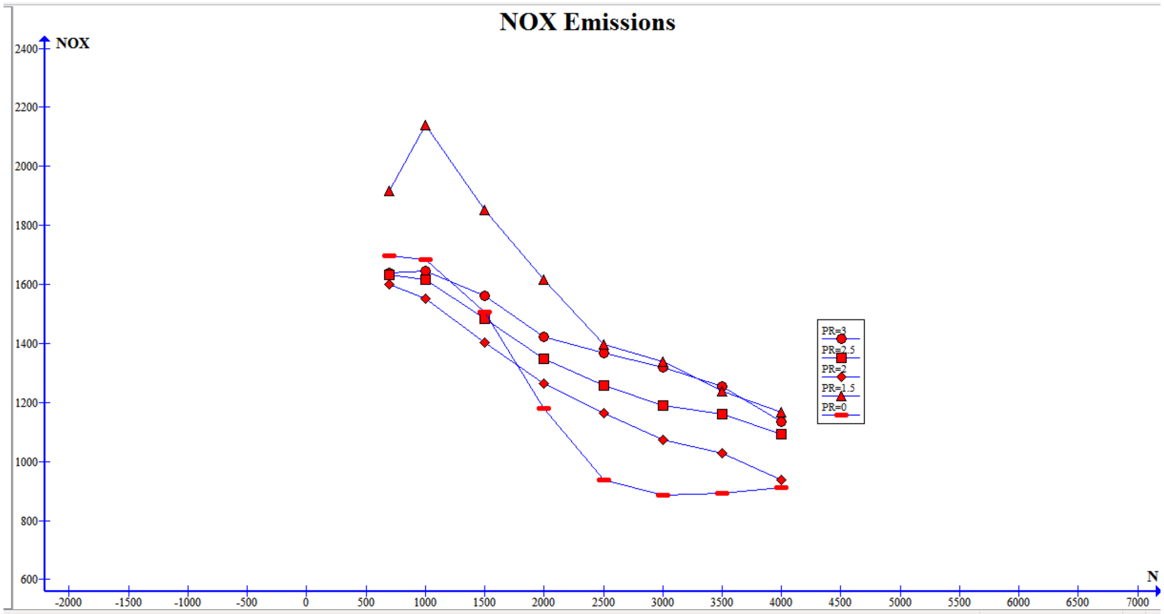


Figure (4-28) NO_X Emissions

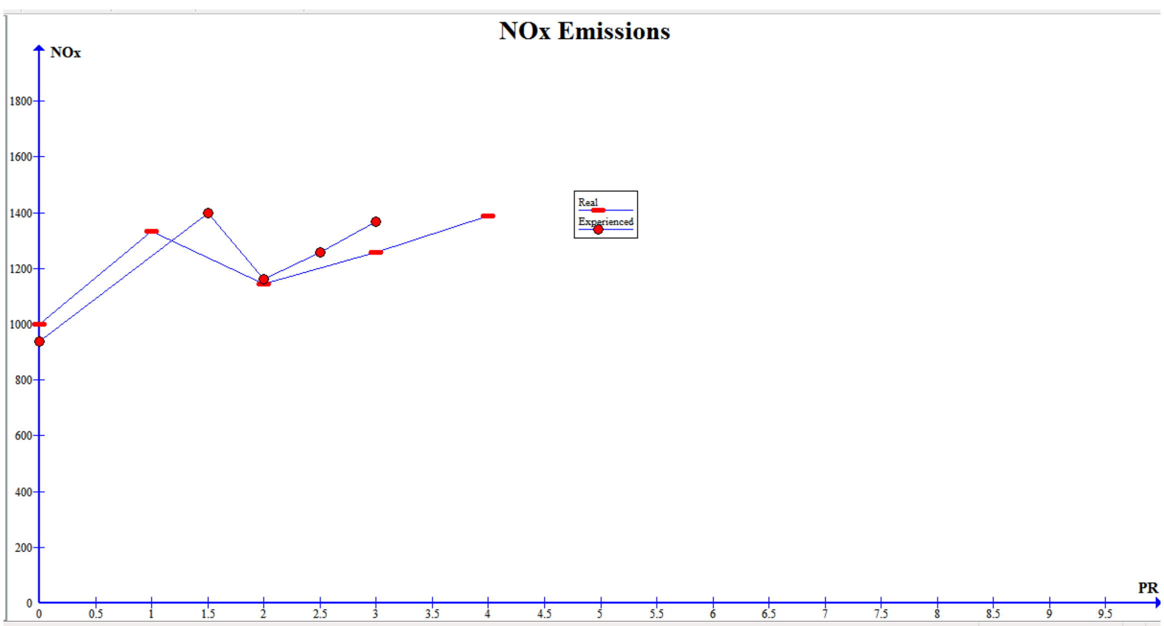


Figure (4-29) compare NO_X Emissions with real data

4.2.4 SO_2 Emission:

The amount of SO2 emitted from the exhaust is very small if a motor is used without a turbocharger and may be absent at high engine speed as shown in figure (4.4) But in the case of the turbocharger we find that the amount of SO2 is very little as shown in figure (4.8) The relationship between the amount of SO2 emitted and the speed of rotation of the engine tends to be a linear relationship. This is because the difference between the amounts of SO2 emitted at a specified speed does not exceed much at the next speed. From here we can say that the amount of SO2 emitted is very simple.

When comparing the emission of SO2 in the case of the turbocharger with the absence of the turbocharger at a specified speed, for example at 2500rpm,as shown in the chart below:

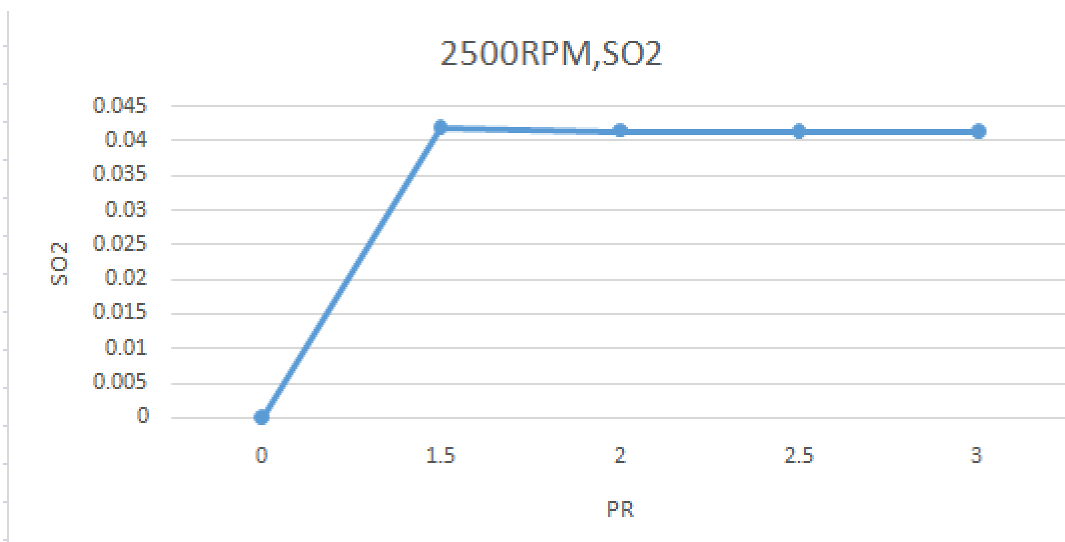


Figure (4-30) Emission of SO2 at 2500RPM

We find that the amount of SO2 is completely absent in the absence of a turbocharger. But when adding the turbocharger we find that the amount of

SO2 has increased but very little. Means that the turbocharger has increased the emission of SO_2.

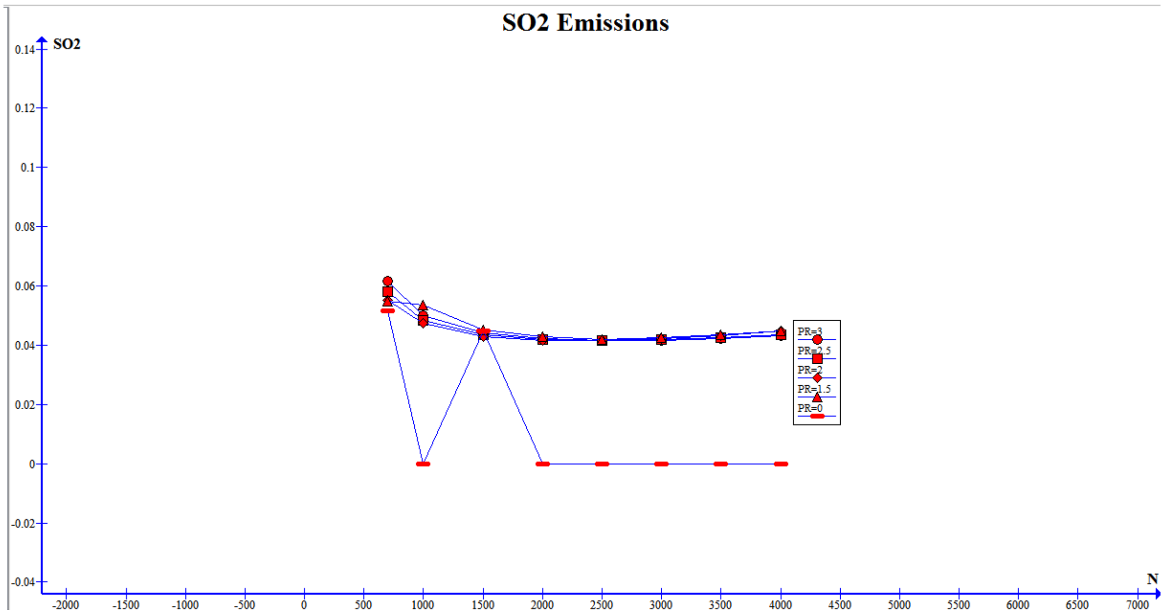


Figure (4-31) SO_2 Emissions

CHAPTER FIVE

CONCLUSION & RECOMMENDATION

5.1 Conclusion

The readings were initially taken without the use of a turbocharger at different engine speeds. Emission values were obtained for pm, co2, so2, and NO_X. Other readings were taken, but if a turbocharger was used at different rates and speeds starting from 700 RPM and finish at 4000 RPM speed.

The readings of the previous emissions were then monitored and the results were compared if the turbocharger was used with the results in the absence of the turbocharger at different pressure rates and at different speeds.

It also noted the lower engine emission and the optimum speed at which the engine should be driven to obtain the lowest emission and found that this speed is 2500 RPM and the optimum pressure ratio for the turbocharger, which was less emitting and found value 1.5

The lowest value for co2 at speed 2500 and the pressure ratio 1.5 is 677.74 grams per kWh and non-charger and at speed 2500 is 688.30 g / kWh

The lowest value for co2 at speed 2500 and the pressure ratio 1.5 is 677.74 grams per kWh and non-charger and at speed 2500 is 688.30 g / kWh

5-2 Recommendation

1- We recommend that this project can applied experimentally and by another program software such as CFD, MATLAB ….etc.

2- We recommend that the researcher to use DIESEL–RK software instead of empirical test to invest the time and money and effort.

3-Enacting strict laws that prevent emissions.

4- The researcher they want to apply the DIESEL–RK software you should to know they have a good network before to starting so as not waste their time because the program is online calculation.

References

[1] https://en.wikipedia.org/wiki/Engine 22/5/2017

[2] FERGUSQ.NC, R, internal combustion engine. New York: Wiley, 1986

[3] https://en.wikipedia.org/wiki/Diesel_engine 22/5/2017

[4] Performance and emission characteristics of an automotive diesel engine using biodiesel fuel with the influence of air intake variables. By Rizalman mamat.

[5] Technologies methods-for-controlling-nox-sox-emissions-from-ships

[6] Srinivasan.C. , M. S. Sayooj, Increasing the Efficiency of an Engine by the use of Variable Geometry Turbochargers, IJIRSET, Volume3, Special Issue 4, April 2014 journal.

[7] ENGINE CHARACTERISTICS (Engine Performance Parameters)

Prof.Dr. Cem Soruşbay.

[8]www.paulton.org/2006/08/16/how-does-variable-turbine-geometry-works

[9] Holset service repair manual – Cummins turbo technology.

[10] Effect of Variable Geometry Turbocharger on HSDI Diesel Engine, Jaehoon Cheong, Sunghwan Cho, Changho Kim, Seoul 2000 FISITA World Automotive Congress journal.

[11] Effect of Variable Geometry Turbine (VGT) on Diesel Engine and Vehicle System Transient Response: ZoranFilipi, Yongsheng Wang and Dennis Assanis journal.

[12] Effect of diesel–microalgae biodiesel–butanol blends on performance and emissions of diesel engine/by G.khan Tüccar , Tayfun .zgür, Kadir Aydın.

[13] Effect of Turbo charging On Volumetric Efficiency in an Insulated Di Diesel Engine For Improved Performance, by S.Sunil Kumar Reddy, Dr. V. Pandurangadu, S.P.Akbar Hussain .

[14] WWW.DIESEL–RK.bmstu.ru.

[15] Software for engine simulation and optimization

[16] DIESEL –RK data list .pdf by Dr Andrey kuleshov