Sudan University of science and Technology

College of Post graduate studies

Mechanical Engineering Department (Power)

Effects of Injector Parameters on Diesel Engine Performance and Exhaust Emissions by Using Simulation Software

تأثير متغيرات الحاقن على أداء محرك الديزل وإنبعاثات العادم بإستخدام برنامج محاكاة

Thesis submitted in partial fulfillment of requirement for the degree of M.Sc. in Mechanical Engineering (Power)

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الإستهلال

قَالَ تَعَالَىٰ

آية الكرسي سورة البقرة آية ٢٥٥

صدق الله العظيم

Dedication

This thesis is dedicated to my father, who taught me that the best kind of knowledge to have is that which is learned for its own sake.it is also dedicated to my mother, who taught me that even the largest task can be accomplished if it is done one step at atime.to my friends, who give me the greatest support in my study and life.

Acknowledgement

I would like to express my gratitude to Allah for enabling the guidance to complete this work. I wish to express my gratitude to all those who assisted me in the course of preparing this work. To Sudan University of Science and Technology, Faculty of mechanical engineering and those who increased my knowledge there with this work.

I would also like to extend my gratitude to Super visor DR. Abubaker Yousif Eltayeb, for his intellectual insights, were very helpful in the development of this work.

Finally, I must express my very profound gratitude to my parents and my partner for providing me with unfailing support and continuous encouragement throughout my years of study and through the process of researching and writing this thesis. This accomplishment would not have been possible without them. Thank you.

Author

Eng. Khaled Abdalwhab Omer Fadl Alseed

Abstract

The phenomenon of global warming is one of the most important problems facing us at the present time, which has not been treated so far, and that the root cause of this phenomenon is the greenhouse gases surrounding the globe, and hence the aim was to reduce the rates of these gases and know the causes of the occurrence and attempts To reduce or minimize these causes.

Emissions from motor vehicles and power plants are one of the main reasons for the presence of greenhouse gases, so there has to be a solution that reduces these emissions, and here are the emissions from motor vehicles in general and diesel engine in particular. The engine was tested to reduce its emissions, and the method used was to adjust the fuel injector to reduce emissions and increase mechanical efficiency.

A comprehensive study was conducted for these emissions, their different types, causes and ways to reduce them by studying the amount of emissions and mechanical efficiency of one of the diesel machines and then conducting the same study for the same machine, but by changing the injector variables Where at each stage we change one of the variables of the fuel injector in a certain range and installing other variables, at a certain nominal engine speed and compression ratio until we get the lowest amount of emissions and the highest mechanical efficiency as possible.

And when entering the values of injector parameters obtained into the Diesel-RK program, it is noted that the value of mechanical efficiency increased from 88.01% to 88.8%, the PM emission decreased from 0.17372 ppm to 0.13187 ppm, CO_2 emission decreased from 714.04 g/kWh to 711.67 g/kWh and the value of NOx emission decreased from 1205.2 g/kWh to 805.45 g/kWh.

المستخلص

إن ظاهرة الاحتباس الحراري تعد من أهم المشاكل التي تواجهنا في الوقت الحالي ,والتي لم يتم علاجها الي الان والسبب الأساسي في هذه الظاهره هي الغازات الدفيئه المحيطه بالأرض ومن هنا كان المنطلق إلي تقليل نسب الغازات الدفيئه المحيطه بالأرض ومعرفة أسبابها وإجراء محاولات عديده للحد و التقليل من هذه الأسباب.

ومن أهم الأسباب في وجود الغازت الدفيئه هى الإنبعاثات الصادره من المحطات النوويه ومحطات الطاقه ومن محركات السيارات فكان لابد من وجود حل يقلل من هذه الإنبعاثات. ومن هنا تم اخذ الإنبعاثات الصادره من محركات السيارات عموما ومن محرك الديزل بشكل خاص وأجريت عليها دراسه لتقليل هذا النوع من الإنبعاثات.

تم إجراء دراسة شاملة لهذه الإنبعاثات وأنواعها المختلفة وأسبابها وطرق تقليلها من خلال دراسة مقدار الإنبعاثات والكفاءة الميكانيكية لإحدى ماكينات الديزل ثم إجراء نفس الدراسة لنفس الآلة ، ولكن عن طريق تغيير متغيرات الحاقن حيث نقوم في كل مرحلة بتغيير أحد متغيرات حاقن الوقود في نطاق معين وتثبيت المتغيرات الأخرى عند سرعه ونسبة انضغاط معينه حتى نحصل على أقل كمية من الإنبعاثات وأعلى كفاءة ميكانيكية ممكنة.

وعند إدخال قيم متغيرات الحاقن التي تم الحصول عليها الي برنامج المحاكاه Diesel-RK نلاحظ إن الكفاءه الميكانيكيه قد زادت من 88.01% الي 88.8% وأن انبعاث الجسيمات الدقيقه (PM) قد إنخفض من 0.17372 ppm الي 0.13187ppm وكذالك إنخفض إنبعاث ثاني أكسيد الكربون من 714.04 g/kWh إلي 711.67 g/kWh كما إنخفضت أكاسيد النترجين من 1205.2 g/kWh الي 805.45 g/kWh .

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LIST OF ABBREVIATIONS

EGR	Exhaust Gas Recirculation
ULSD	ultra-low sulfur diesel
ICE	Internal Combustion Engine
SI	Spark Ignition
CI	Compression Ignition
LPG	Liquefied petroleum gas

PAHs	Polycyclic aromatic hydrocarbons
BSFC	Brake specific fuel consumption

CHAPTER ONE

INTRODUCTION

1.1 **Introduction**

Diesel engines have high efficiency, durability, and reliability together with their low-operating cost. These important features make them the most preferred engines especially for heavy-duty vehicles. The interest in diesel engines has risen substantially day by day. In addition to the widespread use of these engines with many advantages, they play an important role in environmental pollution problems worldwide. Diesel engines are considered as one of the largest contributors to environmental pollution caused by exhaust emissions, and they are responsible for several health problems as well. Many policies have been imposed worldwide in recent years to reduce negative effects of diesel engine emissions on human health and environment. Many researches have been carried out on both diesel exhaust pollutant emissions and after treatment emissions control technologies. [1]

For scientists, continuing challenges in engine design include reduced emissions and increased efficiency, the resulting engine systems must also meet the strict emissions and fuel economy targets that have been recently established [2]

The diesel engine is an auto-ignition engine in which fuel and air are mixed inside the engine. The air required for combustion is highly compressed inside the combustion chamber. This generates high temperatures which are sufficient for the diesel fuel to ignite spontaneously when it is injected into the cylinder. Thus, the diesel engine uses heat to release the chemical energy contained in the diesel fuel and to convert it into mechanical force [3].

Carbon and hydrogen construct the origin of diesel fuel like most fossil fuels. For ideal thermodynamic equilibrium, the complete combustion of diesel fuel would only generate CO_2 and H2O in combustion chambers. However, many reasons (the air–fuel ratio, ignition timing, turbulence in the combustion chamber, combustion form, air–fuel concentration, combustion temperature, etc.) make this out of question, and a number of harmful products are generated during combustion. The most significant harmful products are CO, HC, NOx, and PM. [5]

Pollutant emissions have a rate of less than 1 % in the diesel exhaust gas. NOx has the highest proportion of diesel pollutant emissions with a rate of more than 50 %. After NOx emissions, PM has the second highest proportion in pollutant emissions. Because diesel engines are lean combustion engines, and the concentration of CO and HC is minimal. Besides, pollutant emissions include a modicum of SO2 depending the specifications and quality of fuel. It is produced by the sulfates contained in diesel fuel. For the present, there is not any after treatment system like a catalytic converter to eliminate SO2. Nowadays, most of oil distributors and customers prefer ultra-low sulfur diesel (ULSD) for diesel engines to prevent harmful effect of SO2. [4]

The diesel engine is a compression-ignition engine. Ignition of the mixture in these engines occurs due to extreme pressure and temperature during the compression process where the fuel in the spray mixes with the compressed hot air, evaporates, and then the mixture ignites by itself. It has to be pointed out that injection, atomization, spray development, mixture formation, ignition, combustion, and emissions formation processes proceed largely simultaneously and interact with each other. During the combustion process heat is released and both the in-cylinder pressure and the in-cylinder temperature increase. At the end of the expansion phase, the exhaust valve opens and the exhaust phase begins. This phase and the whole cycle end as the exhaust valve closes.

Diesel engines operate in four-stroke or two-stroke cycles, and four-stroke-cycle engines are usually used in automotive application, either naturally aspirated or turbocharged. In a naturally aspirated diesel engine.

Fuel injectors atomize the fuel into very fine droplets, and increases the surface area of the fuel droplets resulting in better mixing and subsequent Combustion Atomization is done by forcing the fuel through a Small orifice under high pressure. [7]
There are two types of injectors in a diesel injection system, mechanical and electronic fuel injectors, these are categorized depending upon how the fuel is injected in the system.

Design of the diesel fuel injector is critical to the performance and emissions of modern diesel engines. Some of the important injector design parameters include injector nozzles bore, nozzles discharge coefficient obtained as a result of test in atmospheric conditions, number of nozzles (if all sprays are identical or not), distance between spray center and Bowl Axis and distance between sprays center and cylinder head plane. These features not only affect the combustion characteristics of the diesel engine, they can also affect the stability of the emissions and performance over the lifetime of the engine and the mechanical durability of the injector. [14]

1.2 **Problem statement**

Investigated the effect of injector parameters on the performance of diesel engine and exhaust emissions influence air pollution, 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 particulate matter (PM), carbon dioxide (CO_2), and nitrogen oxides (NO_x).These emissions should be reduced.

1.3 **Research Objectives**

i) Evaluation influence of injector parameters on diesel engine efficiency and emissions and obtain the best injector variables that give high efficiency and lower emissions to PM, CO_2 and Nox.

1.4 **Scope of Research**

The scopes of this research in the field of Effect of injector parameters on the performance of diesel engines and emissions from diesel combustion.

1.5 **Scientific Study**

The program (DIESEL-RK) was used to simulate a diesel engine where change the values of injector parameters (injector nozzles bore, number of nozzles, nozzles discharge coefficient, distance between spray center and Bowl Axis and distance between sprays center and cylinder head plane) and the results obtained.

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. [6]

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 [24]

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. [23]

2.4 Classification of internal combustion engines

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

2.4.1Types of ignition

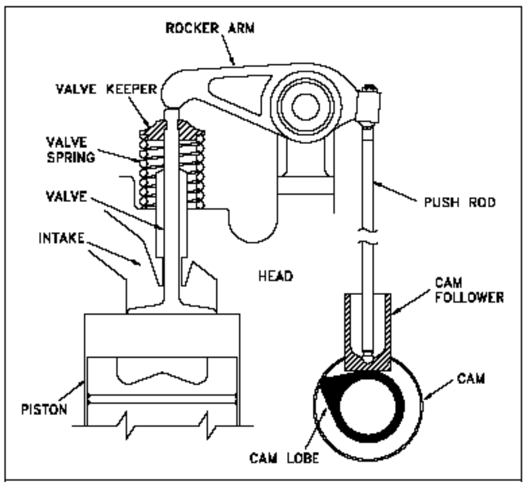
- Spark Ignition (SI): An SI engine starts the combustion process in each cycle by use of a spark plug. The spark plug gives a high voltage electrical discharge between two electrodes which ignites the air fuel mixture in the combustion chamber surrounding the plug. - Compression Ignition (CI): the combustion 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. [7]

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. [7]

2.4.3Valve Location

2.4.3.1 Valves in head (Overhead Valve)

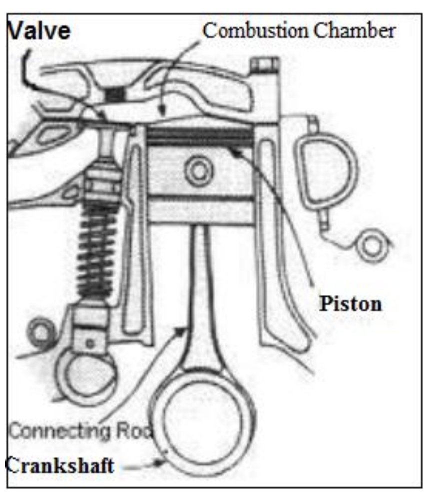


It is also called I Head engine.

Figure (2-1) Valves in head

2.4.3.2 Valves in block (Flat Head)

It is also called L Head engine: Some historic engines with valves in block had the intake valve

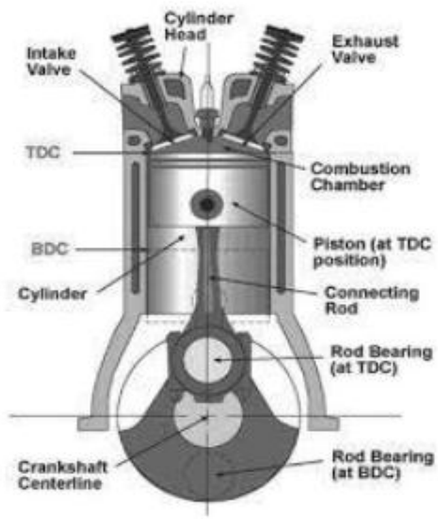


on one side of the cylinder and the exhaust valve on the other side.

Figure (2-2) Valves in block

One valve in head (usually intake) and one in block also called F Head engine. [7]

2.4.4 Basic design



2.4.4.1 Reciprocating engine:

It 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. [7]

Figure (2-3) Reciprocating engine

2.4.4.2 Rotary engine

It 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. [22]

Figure (2-4) Rotary engine

2.4.5 Position and number of cylinder of reciprocating engines

2.4.5.1 Single cylinder engine

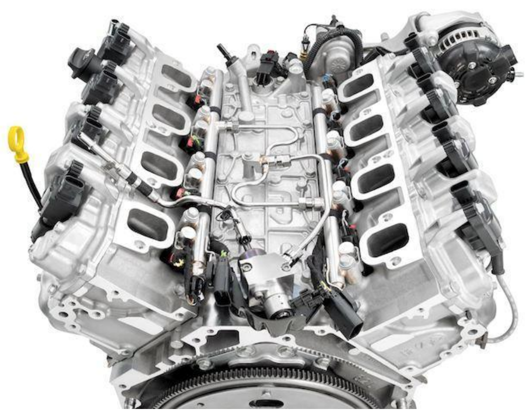
It has one cylinder and piston connected to the crankshaft.

2.4.5.2 Inline cylinders

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

2.4.5.3 V Engine

It has two banks of cylinders at an angle with each other along a single crankshaft allowing for a



shorter engine block. [7]

Figure (2-5) V Engine

2.4.5.4 Opposed cylinder engine

It has 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. [7]

Figure (2-6) Opposed cylinder engine

2.4.5.5 W Engine

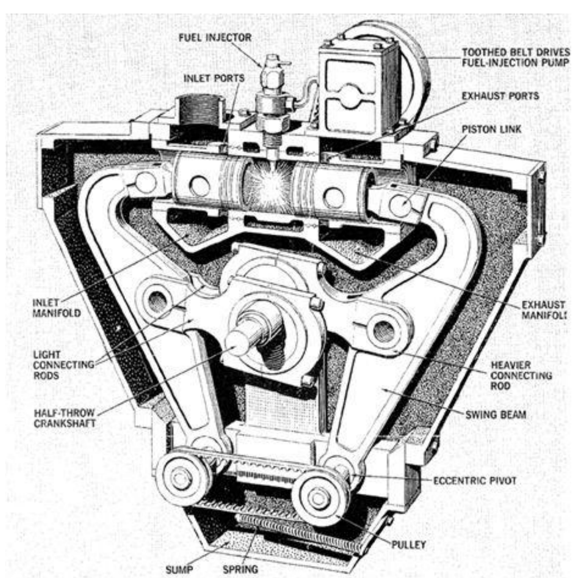
It is engines of two different cylinder arrangements have been classified as W engines in the

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

2.4.5.6 Opposed piston engine

It has two piston in each cylinder with the combustion chamber in the center between the



pistons.

Figure (2-8) Opposed piston engine

2.4.5.7 Radial engine

Radial engine is engine with pistons positioned in a circular plane around the central crankshaft. The connecting rods of the pistons are connected to a master rod which in turn is connected to

the crankshaft. [7]

Figure (2-9) Radial engine

2.4.6 Air intake process

i) Naturally air: No intake air pressure boost system.
ii) Supercharged: Intake air pressure increased with the compressor driven off the engine crankshaft.
iii) Turbocharged: Intake air pressure increased with the turbine compressor driven by the engine exhaust gases.
iv) 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. [7]

2.4.7 Method of fuel input for spark ignition engines

i) Carbureted.
ii) Multipoint port fuel injection: One or more injectors at each cylinder intake.

iii) Throttle body fuel injection: Injectors upstream in intake manifold.

iv) Gasoline direct injection: Injectors mounted in combustion chambers with injection directly into cylinders. [7]

2.4.8 Method of fuel input for compression ignition engines

i) Direct injection: Fuel injected into main combustion chamber.

ii) Indirect injection: Fuel injected into secondary combustion chamber.

iii) Homogeneous charge compression ignition: Some fuel added during intake. [7]

2.4.9 Fuel used

i) Gasoline.

ii) Diesel oil or fuel oil.

iii) Gas, natural gas, methane.

iv) LPG (Liquefied petroleum gas) - Alcohol ethy, methy.

v) 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.

vi) Gasohol: Common fuel consisting of 90% gasoline and 10% alcohol. [7]

2.4.10 Application

i) Automobile, Truck, Bus.

ii) Locomotive.

iii) Stationary.

iv) Marine.

v) Aircraft.

vi) Small portable, Chain saw, Model Airplane. [7]

2.4.11 Type of cooling

i) Air cooled.

ii) Liquid cooled, water cooled. [7]

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 15 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. [8]

2.5.1 Fuel Injection System in Diesel Engine

The performance of diesel engines is heavily influenced by their injection system design. In fact, the most notable advances achieved in diesel engines resulted directly from superior fuel injection system designs. While the main purpose of the system is to deliver fuel to the cylinders of a diesel engine, it is how that fuel is delivered that makes the difference in engine performance, emissions, and noise characteristics.

Unlike its spark-ignited engine counterpart, the diesel fuel injection system delivers fuel under extremely high injection pressures. This implies that the system component designs and materials should be selected to withstand higher stresses in order to perform for extended durations that match the engine's durability targets. Greater manufacturing precision and tight tolerances are also required for the system to function efficiently. In addition to expensive materials and manufacturing costs, diesel injection systems are characterized by more intricate control requirements. All these features add up to a system whose cost may represent as much as 30% of the total cost of the engine.

The main purpose of the fuel injection system is to deliver fuel into the cylinders of an engine. In order for the engine to effectively the fuel must be injected at the proper time, that is, the injection timing must be controlled and The correct amount of fuel must be delivered to meet power requirement, that is, injection metering must be controlled. However, it is still not enough to deliver an accurately metered amount of fuel at the proper time to achieve good combustion.

Additional aspects are critical to ensure proper fuel injection system performance including fuel atomization, bulk mixing and air utilization.

Fuel atomization: ensuring that fuel atomizes into very small fuel particles is a primary design objective for diesel fuel injection systems. Small droplets ensure that all the fuel has a chance to vaporize and participate in the combustion process. Any remaining liquid droplets burn very poorly or are exhausted out of the engine. While modern fuel injection systems are able to produce fuel atomization characteristics far exceeding what is needed to ensure complete fuel evaporation during most of the injection process, some injection system designs may have poor atomization during some brief but critical periods of the injection phase. The end of the injection process is one such critical period.

Bulk mixing: while fuel atomization and complete evaporation of fuel is critical, ensuring that the evaporated fuel has sufficient oxygen during the combustion process is equally as important to ensure high combustion efficiency and optimum engine performance. The oxygen is provided by the intake air trapped in the cylinder and a sufficient amount must be entrained into the fuel jet to completely mixed with the available fuel during the injection process and ensure complete combustion.

Air utilization: Effective utilization of the air in the combustion chamber is closely tied to bulk mixing and can be accomplished through a combination of fuel penetration into the dense air that is compressed in the cylinder and dividing the total injected fuel into a number of jets. A sufficient number of jets should be provided to entrain as much of available air as possible while avoiding jet overlap and the production of fuel rich zones that are oxygen deficient.[21]

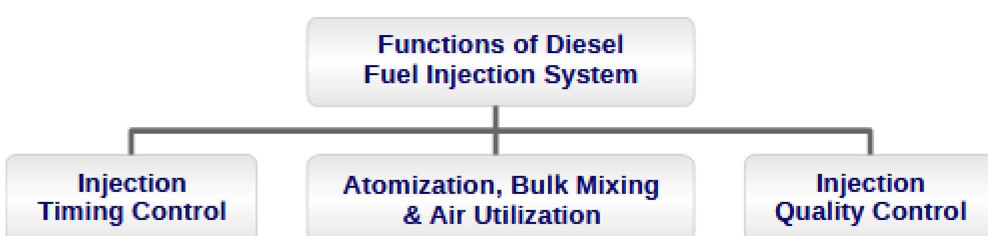


Figure 2-10. Main Functions of Diesel Fuel Injection System

2.5.2 Basic Fuel System Components

With a few exceptions, fuel systems can be broken down into two major component groups:

2.5.2.1 Low pressure side components

These components serve to safely and reliably deliver fuel from the tank to the fuel injection system. Low pressure side components include the fuel tank, fuel supply pump and the fuel filter.

2.5.2.2 High pressure side components

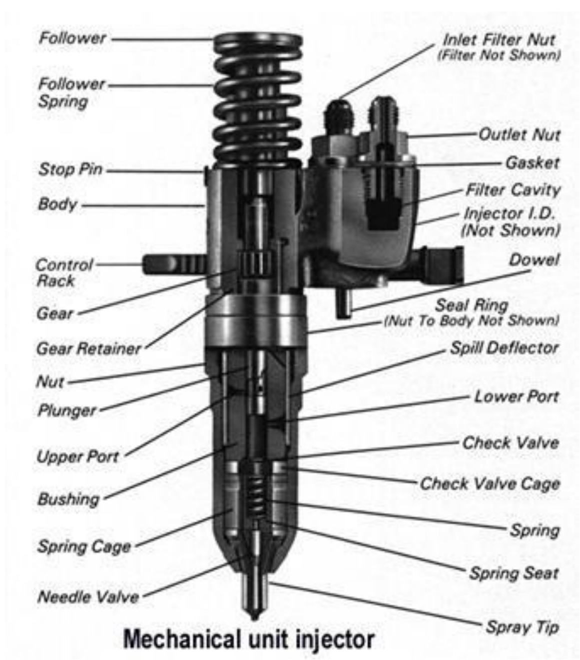
Components that create high pressures, meter and deliver the fuel to the combustion chamber. They include the high pressure pump the fuel injector and fuel injection nozzle. Some systems may also include an accumulator.[20]

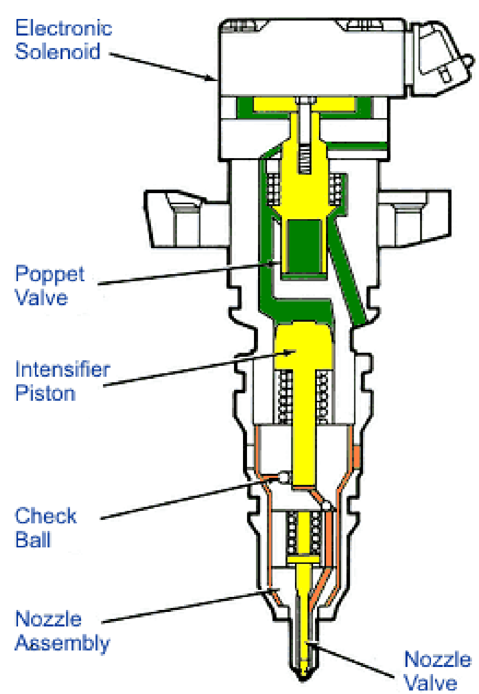
2.5.3 Fuel injector

The injector is a device to deliver fuel intermittently into the engine cylinder at times that are synchronized with the engine camshaft position. The injector needle in an injector acts as a valve to open or close the passage to the injector hole(s). In a fuel pressure-activated injector, one end of the needle is preloaded with a spring force, and the other end is exposed to the fuel pressure. The needle is closed against the seat when the spring force is greater than the force due to fuel pressure.

Fuel injectors atomize the fuel into very fine droplets, and increases the surface area of the fuel droplets resulting in better mixing and subsequent Combustion Atomization is done by forcing the fuel through a Small orifice under high pressure. [7]

There are two types of injectors in a diesel injection system namely mechanical fuel injectors





(Fig.2-3) and electronic fuel injectors (Fig.2-4). Which are categorized depending upon how the fuel is injected in the system.

Figure 2-11 Mechanical fuel injectors Figure 2.12 Electronic fuel injectors

2.5.3.1 The main components of the injector:

The main components of the injector are a needle valve, compression spring, nozzle, and injector body

Fuel supplied by the injection pump exerts sufficient force against the spring to lift the nozzle valve, after injection the spring pressure pushes the nozzle valve back on its seat, mall quantity of fuel is allowed to leak through the clearance between nozzle valve and its guide for proper lubrication, Valve opening pressure is controlled by adjusting the screw (spring tension)

2.5.4 Fuel injector nozzle

Nozzle is that part of an injector through which the liquid fuels sprayed into the combustion chamber, the nozzle should fulfill the following functions

i. **Atomization**: This is a very important function since it is the first phase in obtaining proper mixing of the fuel and air in the combustion chamber.
ii. **Distribution of fuel**: Distribution of fuel to the required areas within the combustion chamber. The factors affecting in distribution of fuel are injection pressure, density of air in the cylinder, and Physical properties of fuel like self-ignition temperature, vapor pressure, viscosity, etc. [25]

2.5.4.1 Types of Nozzle

The most common types of Nozzles are Pintle nozzle, Single-hole nozzle, Multi-hole nozzle, Pentax nozzle.

2.5.4.1.1 Pintle Nozzle:

The stem of the nozzle valve is extended to form a pin or pintle which protrudes through the mouth of the nozzle. The size and shape of the pintle can be varied according to the requirement. It provides a spray operating at low injection pressure of 8-10 MPa. The spray cone angle is generally 600. The spray obtained by the pintle nozzle is hollow conical spray.

Advantages:

i) Self-cleaning type prevents carbon deposition on the nozzle hole.

ii) It avoids weak injection and dribbling

iii) Results in good atomization

Disadvantages: Distribution and penetration is poor hence not suitable for open combustion chambers.

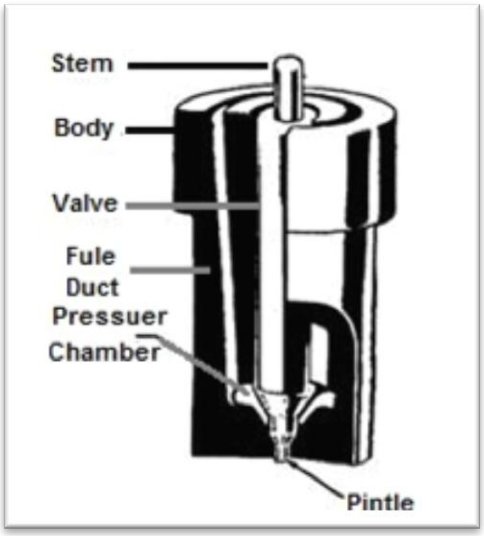


Figure 2-13 Show the Pintle nozzle

2.5.4.1.2 Single-hole Nozzle

This is the simple type of nozzle and used in open combustion chambers. It consists of a single hole bored centrally through the nozzle body and closed by the needle valve. The size of the hole is usually larger than 0.2 mm.

Advantages: It is simple in construction and operation.

Disadvantages:

i) Very high injection pressure is required because whole of the fuel passes through the single hole.

ii) It has tendency to dribble.

iii) As the spray angle is narrow, this does not facilitate good mixing unless higher velocities are provided.

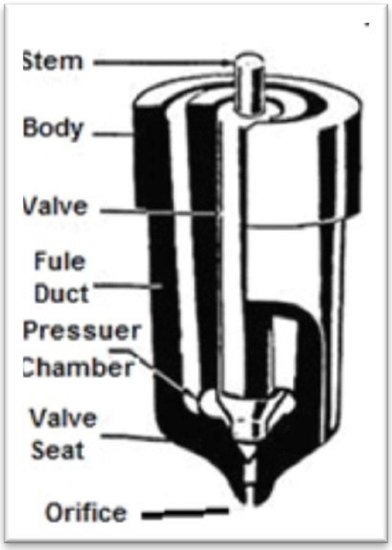


Figure 2-14 Show the Single-hole Nozzle

2.5.4.1.3 Multi-hole Nozzle

This type of nozzle finds extensive use in automobiles particularly with open combustion chambers. It consists of a number of holes bored in the tip of the nozzle. The number of holes varies from 4 to 18 and the size from 0.25 to 0.35 mm. The hole angle lies between 200 to 450. These nozzles operate at high injection pressures of the order of 18 MPa.

Advantages:

i) Gives good atomization
ii) Distribute fuel properly even with lower air motion in open combustion chambers.

Disadvantages:

i) Holes are small and liable to clogging
ii) Dribbling between injections
iii) Very high injection pressures required
iv) Close tolerances in manufacturing hence costly

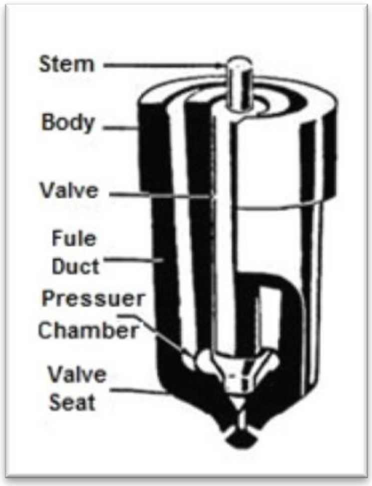


Figure 2-15 Show the Multi-hole nozzle

2.5.4.1.4 Pintaux Nozzle

The stem of the nozzle valve is extended to form a pin or pintle which protrudes through the mouth of the nozzle. The size and shape of the pintle can be varied according to the requirement. It provides a spray operating at low injection pressure of 8-10 MPa. The spray cone angle is generally 600. The spray obtained by the pintle nozzle is hollow conical spray.

Advantages:

i) Self-cleaning type prevents carbon deposition on the nozzle hole.
ii) It avoids weak injection and dribbling
iii) Results in good atomization

Disadvantages: Distribution and penetration is poor hence not suitable for open combustion chambers.

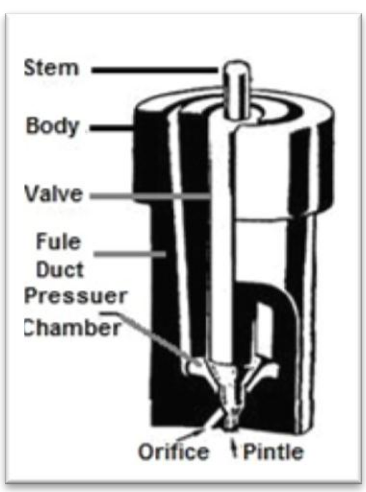


Figure 2-16 Show the Pintaux nozzle

2.6 Common Rail System

Common Rail injection was first used in production by Atlas Imperial Diesel in the 1920s. The rail pressure was kept at a steady 2,000 - 4,000 psi. In the injectors a needle was mechanically lifted off of the seat to create the injection event. Modern common rail systems use very high-pressures. In these systems an engine driven pump pressurizes fuel at up to 2,500 bar (250 MPa; 36,000 psi), in a "common rail". The common rail is a tube that supplies each computer-controlled injector containing a precision-machined nozzle and a plunger driven by a solenoid or piezoelectric actuator. [13]

Since the 1960s, engineers realized that an updated common-rail system, using computer-controlled injectors and ultra-high fuel pressures, offered revolutionary possibilities for the diesel engine. As shown in Fig. (2.2), the modern c-r system employs a remote pump to pressurize a fuel rail, which functions both as a reservoir and as an accumulator. The rail expands to dampen pump pressure peaks and contracts to stabilize pressure when the injectors open. Since rail pressure is almost constant, fuel can be injected at will, independent of pump plunger movement. When coupled with split shot electronic injectors, fuel delivery begins early during compression stroke and can be initiated after combustion to light-off carburized particulate traps. In other words, common-rail was an enabling technology for electronic fuel injection. [14]

First successful use of common rail in a production vehicle, by Two Nippon Denso engineers-Shokei Itoh and Mashiko Miyaki-were responsible Hino Rising Ranger trucks. Meanwhile Fiat and its subsidiary Magneti Marelli Power. Trains were working on c-r for passenger cars and light trucks. By the late 1990s, the prototype was turned over to Robert Bosch GmbH for production.

The first generation 1350-bar (1 bar _14.51 psi) Bosch c-r system made its debut in 1997 on Alfa Romeo and Mercedes-Benz high-speed touring cars.

Subsequent Bosch developments were rapid:

• 1999—First generation, truck 1480-bar system (Renault).

• 2001—Second generation, passenger-car 1600-bar system (Volvo and BMW).

• 2002—Second generation, truck 1600-bar system (MAN).

• 2003—Third generation, passenger-car 1600-bar system (Audi V-6). Piezo injectors reduced emissions by 20%, boosted power 5%, reduced fuel consumption 3%, and engine noise by 3 dB(A).

• 2006—Fourth generation under development with higher pressures and revised injector geometry. [10]

Since the common rail is not protected by patent (low-pressure versions have been around since the 1920s), the technology has become nearly universal for automotive and light truck engines, manufactured by companies as diverse as Hyundai, Cummins, and Mercedes-Benz. Nor is c-r limited to automotive applications: L'Orange GmbH has prototyped a c-r upgrade for marine engines with cylinder bores as large as 500 mm, and Cummins will soon release a common-rail kit for retrofit to locomotives A diesel common rail system is the mainstream fuel injection system for diesel engines. [14]

The fuel, which is highly compressed by the fuel pump, is stored in an accumulator called a common rail.

Then it is sprayed through the electrically controlled injectors into the combustion chambers.

Storing highly compressed fuel in the common rail, not only further increases the pressure of the fuel, but also controls fuel injection pressure and timing without being effected by the engine's rotation speed.

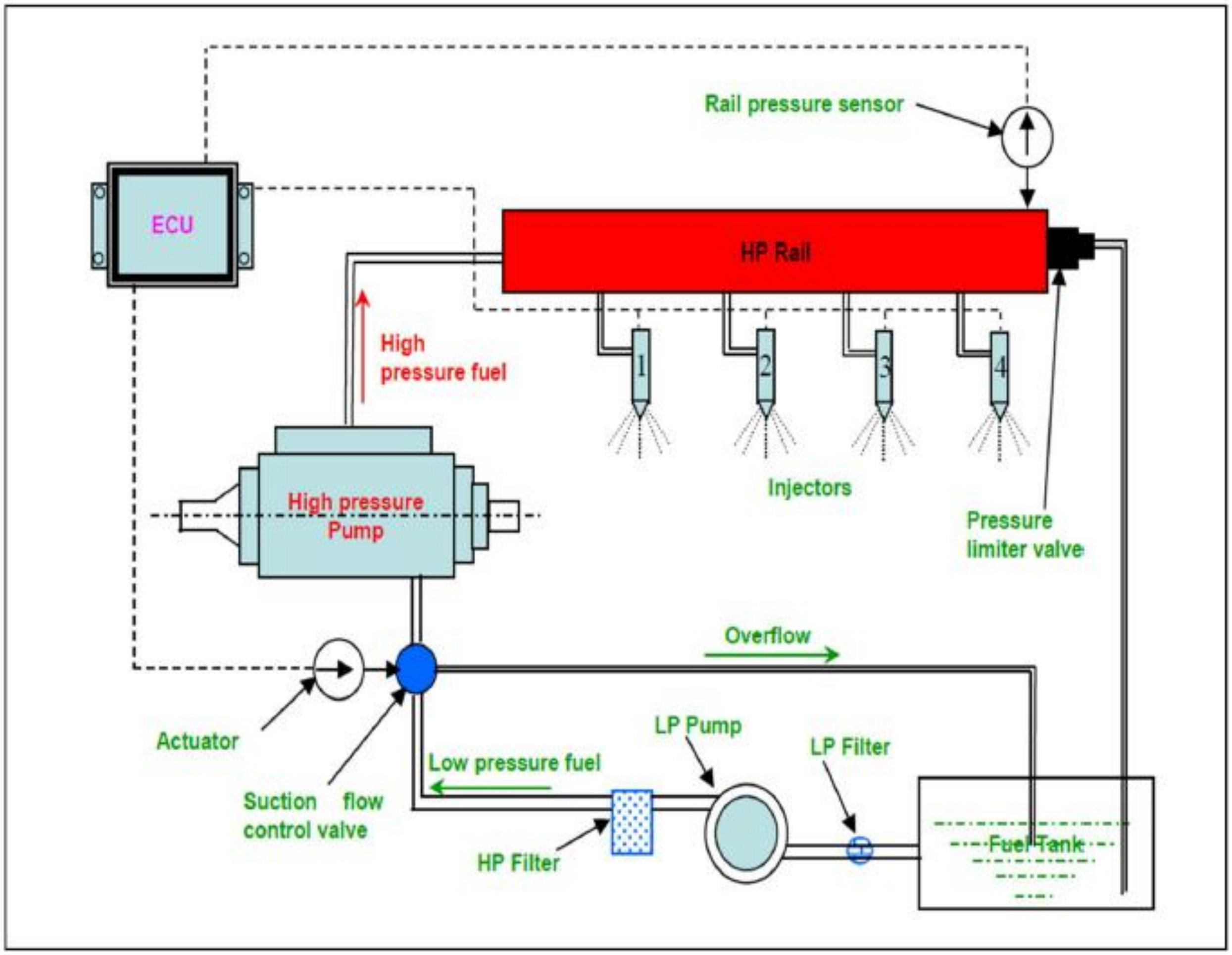


Figure 2.17 Diesel common rail system [14]

2.7 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 NOx, reduced smoke and increased brake specific fuel consumption. [9]

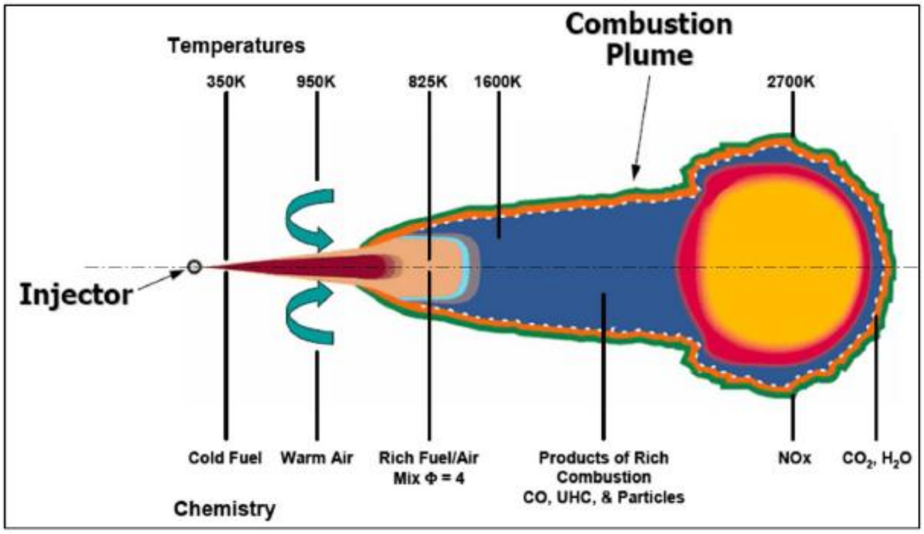


Figure (2.18) Schematic of a diesel flame with temperatures and chemistry [14]

Table (2.1): **Reference Diesel Fuel**

Property	Unit	Specification		Test
		Min	Max	
Cetane Number	-	52	54	ISO 5165
Density @15°C	kg/m3	833	837	ISO 3675
Distillation (vol. % recovered)	°C	-	-	ISO 3405
-50% point		245	-	
-95% point		245	350	
- final boiling point		-	-370	
Flash point	°C	55	-	EN 22719
CFPP	°C	-	-5	EN 116
Viscosity @40°C	mm2/s	2.5	3.5	ISO 3104
Polycyclic aromatic hydrocarbons	% wt.	3.0	6.0	IP 391, EN 12916

Sulfur content	mg/kg	-	300*	ISO/DIS 14596
Copper corrosion	-	-	Class 1	ISO 2160
Conrad son carbon residue(10% DR)	% wt.	-	0.2	ISO 10370
Ash content	% wt.	-	0.01	ISO 6245
Water content	% wt.	-	0.05	ISO 12937
Neutralization (strong acid) number	mg KOH/g	-	0.02	ASTM D974-95
Oxidation stability	mg/ml	-	0.025	ISO 12205`

2.7.1 Oxides of Nitrogen

The major pathway of NOx formation is thermal NOx, fuel NOx and prompt NOx (Heywood 1988; Kutz 2006). Thermal NOx refers to NOx 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 NOx 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 NOx through this process. Conversely, fuel NOx 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 NOx 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. [17]

NOx 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 NOx 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 NOx. [9]

2.2.2 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. [18]

2.7.3 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). [9]

2.7.4 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 emissions standards on sulfur oxide emissions are very strict and are met by post combustion treatment of the exhaust gases. [9]

2.7.5 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. [9]

2.8 The Impact of diesel engine emissions

2.8.1 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 CO_2 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. [15]

2.8.2 Hydrocarbons

Important greenhouse gas, so affects global climate change. And Causes photo chemical smog, cancer-causing, respiratory tract irritants, and some mutagens. [15]

2.8.3 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.

2.9 Technical used to reduce emissions

2.9.1 Reducing NOx Emission

The presence of NOx 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 NOx emission. [10]

2.9.1.1 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. [10]

2.9.1.2 Water Injection and Water Emulsion

In this method, water is added to reduce the temperature of combustion leading to low NOx 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. [10]

2.9.1.3 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 NOx emission. [10]

2.9.1.4 Selective Catalytic Reduction

The SCR is the most efficient method to reduce NOx emissions. In this method, low sulfur fuel is used and exhaust temperature is maintained above 300 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. [10]

2.9.1.5 Two Stage Turbocharger

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

2.9.1.6 Engine Component Modification

It is better to design an engine which has a property to reduce the NOx 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 NOx formation.

2.9.2 Reducing SO_X Emission

SOx 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.

2.9.2.1 Use of Low sulfur fuel

It is expensive but most commonly used.

2.9.2.2 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 SOx to 95%. The scrubbing water is then sent to a water treatment effluent emulsion breaking plant after which it can be discharged overboard.

2.9.2.3 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 SOX emissions from the engine. [10]

2.10 Engine Performance Parameters

2.10.1 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), in case of prony brake, brake power of the engine:

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

Where:

T = Brake torque (Nm)

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

2.10.2 Brake specific fuel consumption (BSFC)

It 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:

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

Where: mf is the fuel consumption rate in grams per second (g/s); (Bp) is the power produced in watts. [11]

2.9.3 Brake thermal efficiency

It 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. [12]

Calculations**:**

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

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

CHAPTER THREE

METHODOLOGY

3-1 Introduction

This chapter is interested in Diesel RK software steps and the engine used in the present investigation is a four-cylinder, four-stroke, turbo-charged, water-cooled and direct-injection CI engine and also explains how to insert fuel injector variables and how to obtain efficiency and emissions.

3-2 DIESEL-RK Software

The DIESEL-RK is vocational thermodynamic full-cycle engine simulation software. On a market, there is little well recognized thermodynamic engine simulation material from different contractor. These tools cover wide range of practice tasks: from general engine connotation 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. [14]

DIESEL-RK is concentrate on advanced diesel combustion simulation and emissions formation simulation, one has not such specific functions as analysis of engine passing 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. The major concentrates of DIESEL-RK are below 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 contractor. These tools cover wide range of practice tasks: from general engine concept analysis up to design engine systems. [19]

The kernels of engine simulation models of other programs are concentrating fundamentally on non-steady 1D gas dynamic phenomena. DIESEL-RK is concentrate on advanced diesel combustion simulation and emissions formation simulation, one has not such specific functions as analysis of engine passing 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. [21]

3.3 The Main Features of DIESEL-RK

Thermodynamic test of Diesels fueled by diesel oil, methanol, bio-fuels and mixtures of bio fuels with diesel oil, HCCI / PCCI concepts and Dual fuel systems are supported.

Thermodynamic test of SI petrol engines and gas engines, including pre chamber engines, and engines fueled 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 test of Two- and Four-stroke engines, Junkers engines with reverse pistons; Crank case scavenged engines, etc. [21]

3-4 Simulation and optimization of Mixture Formation and Combustion in diesel

Fuel injection optimization of sprayer design and location, injection pressure, injection timing rate shaping, split / multiple injection strategy PCCI test including Low Temperature Combustion phase, etc. personal diameters and direction of nozzles of few injectors, having separate control (own fuel and own injection profiles) are accounted and may be optimized.

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

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

2- Effect of Combustion Chamber Geometry modification.

3- Fuel Sprays Evolution visualization.

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

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

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

7- Multipart metric optimization of engines parameters, Conjoint optimization of NO_X, PM and SFC, including Pareto optimization.

DIESEL-RK is full cycle thermodynamic engine simulation software. One is designed for simulating and optimizing working processes of two and four stroke internal combustion engines with all types of support. The program can be used for modeling the following types of engines:

1 - DI Diesel engines, including PCCI and engines fueled by bio-fuels.

2- SI petrol engines.

3- SI gas engines including pre chamber systems, and engines fueled by different gases: Methane, Propane-Butane, Biogas, Wood gas, Syn gas, etc.

4- Two-stroke engines with unflow and loop scavenging, opposed piston engines (OP or Junkers engines) and OPOC engines.

5- Dual fuel engines (engines having few independent fuel injection systems for different fuels).

The DIESEL-RK is thermodynamic software: engine cylinders are considered as open thermodynamic systems.

3-5 Representative applications include

1- Fuel consumption prediction and optimization.

2- Torque curve and other engine performances predictions.

3- Combustion and emissions analysis, including PCCI.

4- Dual fuel engine mixture formation and combustion analysis.

5- Knock prediction.

6- Valve timing optimization, including VVA optimization for every operating mode.

7- EGR analysis and optimization.

8- Turbocharger and bypasses matching and optimization.

9- Conversion of diesel engines into gas engines.

10- Cooperation with different modeling tools: Simulink, IOSO NM, etc. DIESEL-RK solver can be run under the control of other applications [22-25].

In order to run DIESEL-RK kernel under the control of external codes intended for optimization or for simulation of vehicle where the engine has been used, the special interface is developed.

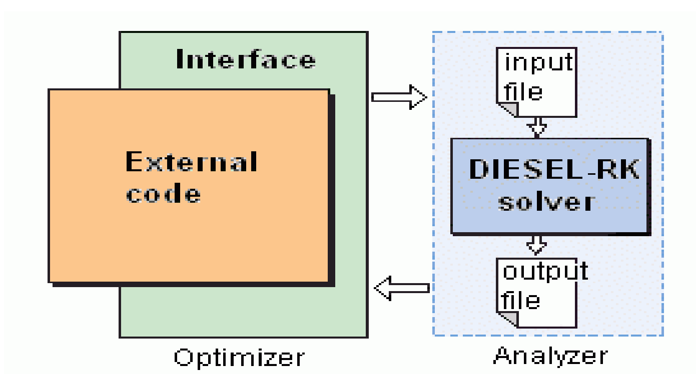


Figure 3.1 Selection of engine models used in the DIESEL-RK

The interface includes text files with input data and output data. The DIESEL-RK solver may be run by external code via batch file

Selection of engine models used in the DIESEL-RK is stipulated by the requirements of a high accuracy of results, high rate of calculation and generality. The last condition is a reason of refusal from empirical equations, which are correct only in narrow boundaries. Authors have preferred frequently laborious methods which consider the physical nature of phenomena in engines. A number of calculation methods was developed by authors of this project

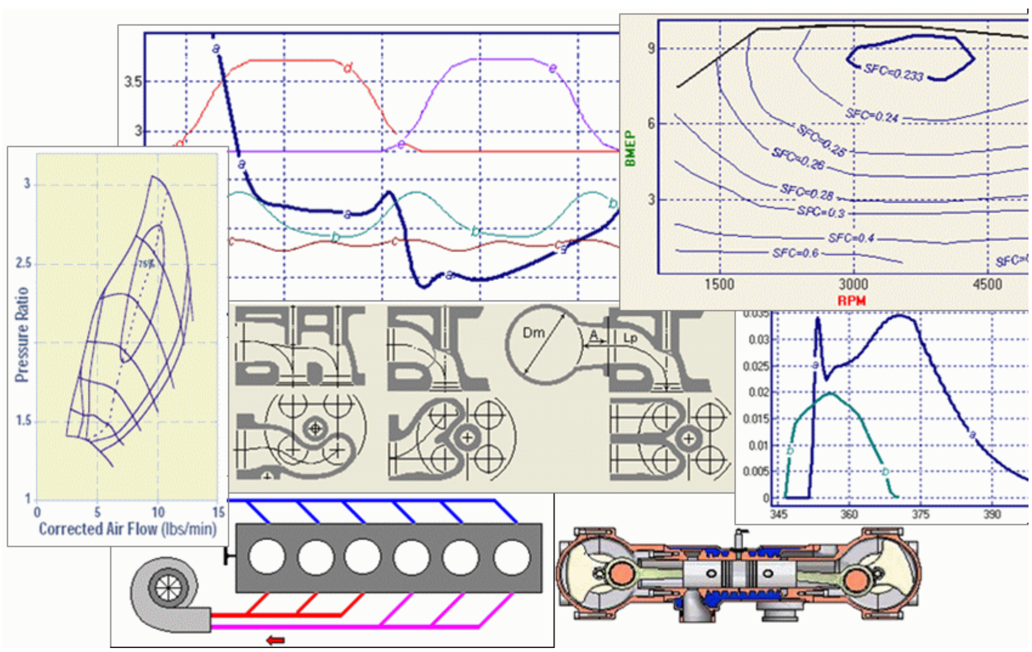


Figure 3.2 DIESEL-RK with other Simulation Tools

3-6 History of the program

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.

The software is intended primarily for ICE developers and researchers; hence much attention is given to provide the product of convenient use for an engineer. The input and output data therefore are organized in a way to suit the industrial demand.

The first software version with a convenient interface and functions of multipara metrical optimization was issued in 1991 owing to financial support of A.Surin, to whom the project leader is truly grateful. The first DIESEL software was in use among leading motor manufacturers of Russia because of its convenient interface and core with the realized and at that time modern combustion model of the compression ignition engine.

In 1993-1994 based on the novel computational code of mixture formation and combustion in diesels, developed by professor of the Kharkov Polytechnic Institute N.Razleytsev, a new software generation was developed and issued: DIESEL-4t software and its modification DIESEL-2t for two-stroke engines. Being a DOS application, the programs were equipped with a window interface reminding a Window application by appearance. For visualization of the mixture formation process and combustion in a diesel Fuel Jet Visualization code was included as part of DIESEL-4t software. In late 90s remote access to the software via INTERNET was organized. The software was intensively used not only within industrial enterprises but in the educational process at the department of ICE (Piston Engines), Bauman University, as well.

The year 2002 gave start to the development of the new Window NT-oriented DIESEL-RK software. The software core was considerably improved and included the up-to-date procedure for calculating toxic emissions with account of the EGR system, multiphase injection research, what provides and expands the potentialities of the diesel performances calculation. As a result of hard work of the workers' team the first version of DIESEL-RK software has been issued in 2004.

As a professional product, DIESEL-RK software can be successfully used by the beginners: students and PhD students of higher schools. To make the process of numerous input data setting (being at times very time-consuming) easier, and empirical coefficients setting easier, the software comprises specific tools settings wizards. The wizards will create data files automatically on the basis of most general information on the engine to be studied, using common technical solutions accepted in the motor-building industry. Thus, the process of input data setting is considerably simplified, and the most elaborate stage of computational research – calculation model calibration – is simplified as well. The latter fact is special importance to students having little experience, time and experimental date for customizing the software with the engine, and also for the needs of researchers performing quick examination of a particular engine design.

Multipart metrical optimization used in the software ensures a radically increased efficiency of numerical research aimed at improving the technical level of engines.

Over the course of its development DIESEL software has been always using advanced mathematical models of combustion in a diesel. In the present software version the RK-model is realized, taking into account specific injection features and fineness of fuel spraying, dynamics of fuel sprays evolution, interaction of sprays with air swirl and with the walls as well as orientation of sprays in the piston bowl. In this case the model accounts for conditions of each fuel spray evolution and thus generated near-wall flows as well as their interaction.

Calculation of NOx emission is realized by the latest techniques: with Zeldovich scheme and with Detail Kinetic Mechanism.

The software includes “Fuel Spray Visualization” code. This code allows a user in pictorial form to analyze the animation picture of fuel sprays evolution, their interaction with the piston bowl walls as well as with swirl and among themselves. The code is helpful in designing the piston bowl shape and in making a proper choice of diameter, number and directions of injector nozzles for a particular fuel supply characteristic and swirl intensity.

3-7 DIESEL-RK Customers

A lot of research projects was carried out with codes DIESEL under the contracts with many companies-manufacturers and researchers of engines in Russia and Euro Union.

Table (3.1) below show that the company using diesel-rk and their country

Country	Customers	Russan Customers
GERMANY	Robert Bosch GmbH *	JSC "Kolomna plant" (Kolomna) *
ITALY	LOMBARDINI srl *	JSC "GAZ" (N.Novgorod) *
HUNGARY	ESPA AGENT Kft.*	JSC "KamAZ" (Naberezhnie Chelny) *

UK	WDL Ltd.*	JSC "ZIL" (Moscow) *
USA	General Motors *	JSC "Zavolzhsky Engine Plant" (Zavolzhie) *
UK	PTL Power train Technology *	JSC "Vladimir plant" (Vladimir) *
GERMANY	HEINZMANN GmbH *	JSC "Auto diesel" (Yaroslavl) *
GERMANY	Astremo Power train AG *	JSC "Altai Precision Components Plant" (Barnaul) *
NETHERLANDS	RDA *	JSC "RUMO" (N. Novgorod)
SWEDEN	FT Engineering AB *	JSC " Penz diesel mash" (Penza)*
FINLAND	Aumet OY *	JSC "Ufa Motor" (Ufa) *
ITALY	Istitu to Motor I -CNR *	TsAGI (Moscow)
UK	Sir Joseph Swan Centre for Energy Research	JSC "Rybinsk Motors" (Rybinsk)
	Newcastle University *	JSC "Liulka-Saturn" (Moscow) *
SWISS	WARTSILA *	JSC "OKB Sukhoy" (Moscow)
FINLAND	VTT Technical Research Centre of Finland *	JSC "Zvezda" (St. Petersburg)
JAPAN	National Maritime Research Institute	

3-8 Calculation models used in the DIESEL-RK

Selection of engine models used in the DIESEL-RK is stipulated by the requirements of a high accuracy of results, high rate of calculation and generality. The last condition is a reason of refusal from empirical equations, which are correct only in narrow boundaries. Authors have preferred frequently laborious methods which consider the physical nature of phenomena in engines.

3.8.1 Specification of the engine used in the study Table (3-2) Engine Parameters

Table 3-2 Parameters Specification

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

3.8.2 Properties of fuel used in the study

Table (3-3) 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	15 ppm
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

3-9 Engine Parameter Test in Diesel RK

Click  button and create a new project using Wizard of New Project Creation.

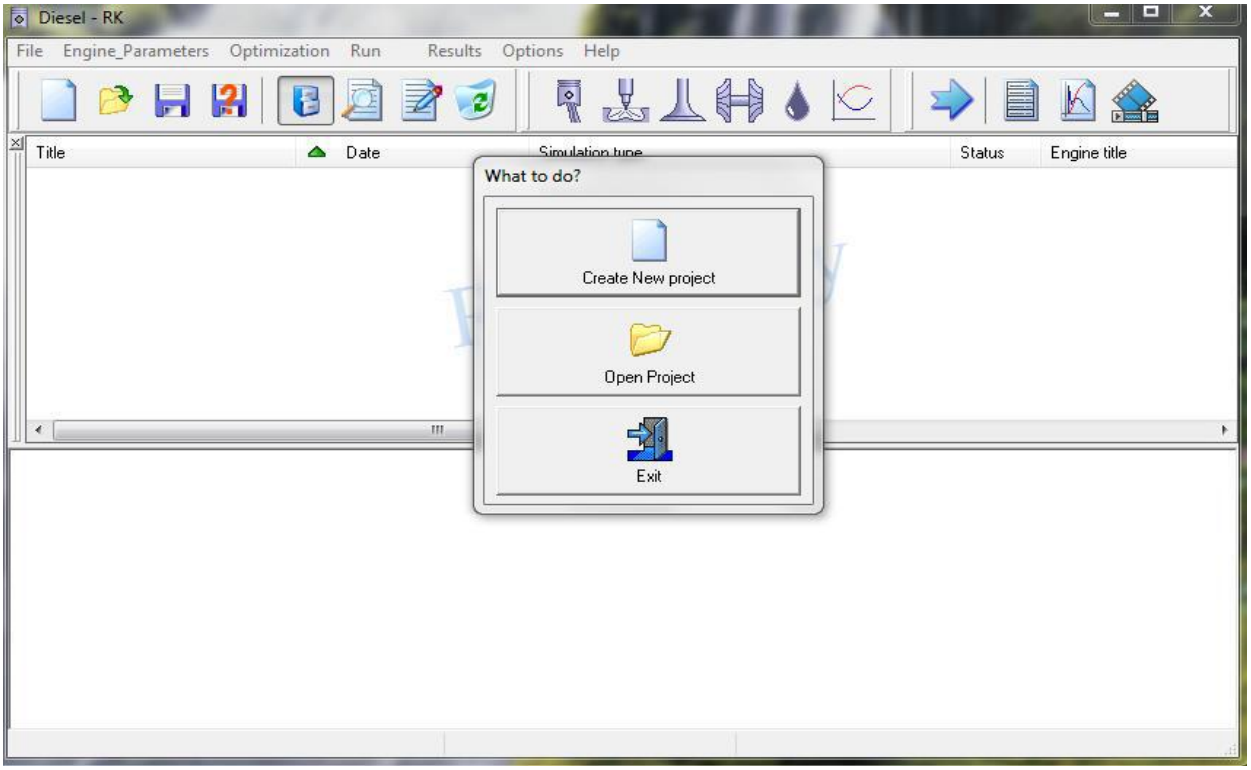


Figure 3.3 show the select of new project step 1

Another window appears, and then chooses (NEXT).

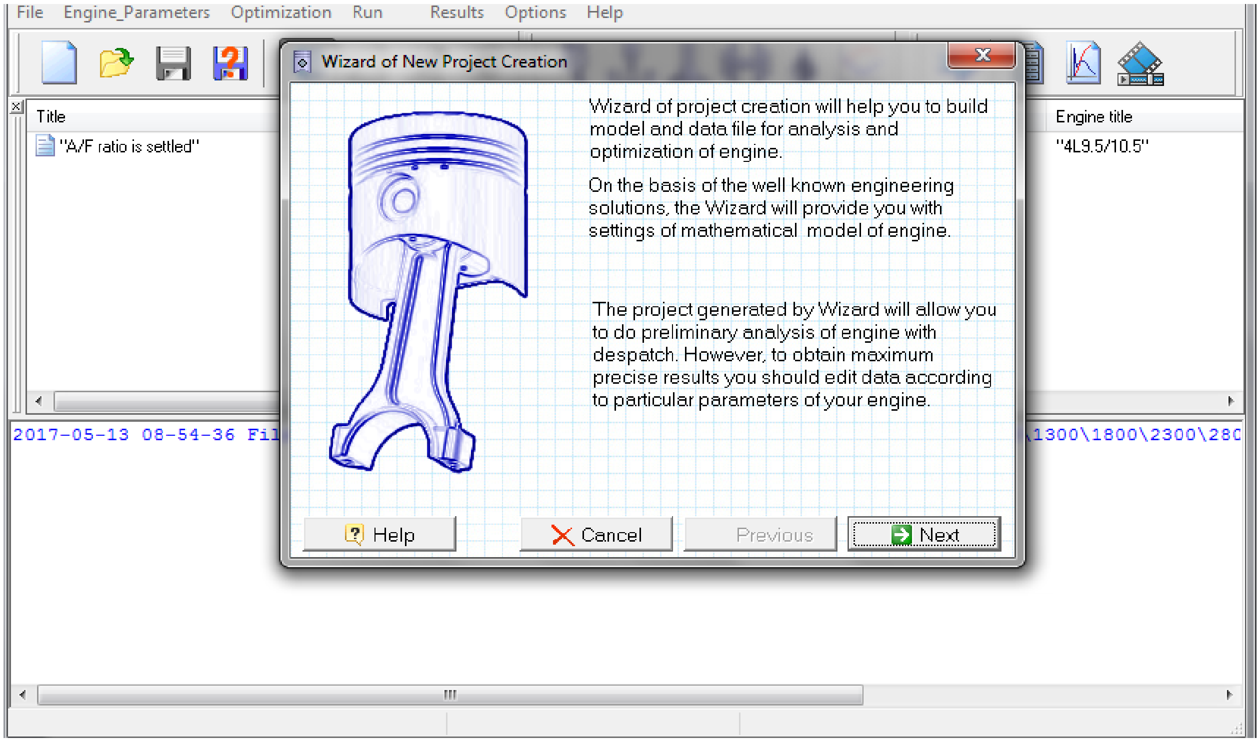


Figure 3.4 show the cylinder shape step 2

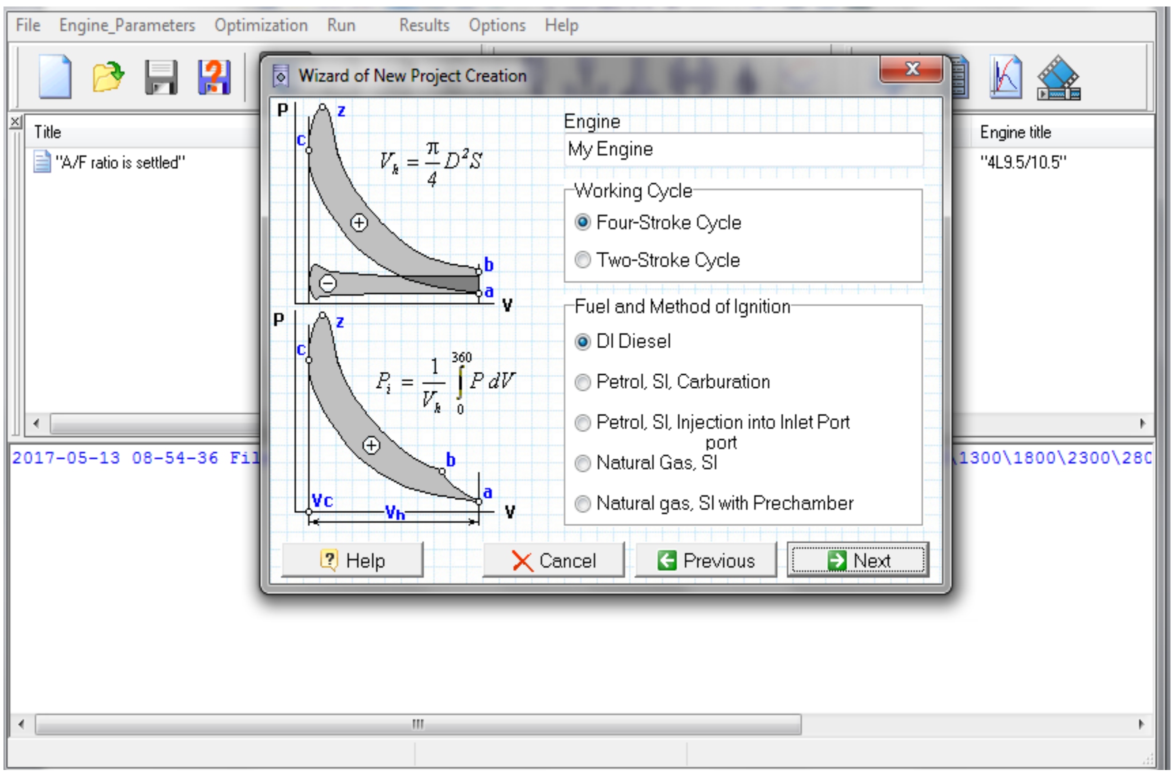


Figure 3.5 show define the type of basic engine design step 3

In the new window we select the name of the engine (a project name) and then determine the type of working cycle choose (Four-Stroke Cycle) after that we choose the fuel and method of ignition (DI Diesel) then press (NEXT)

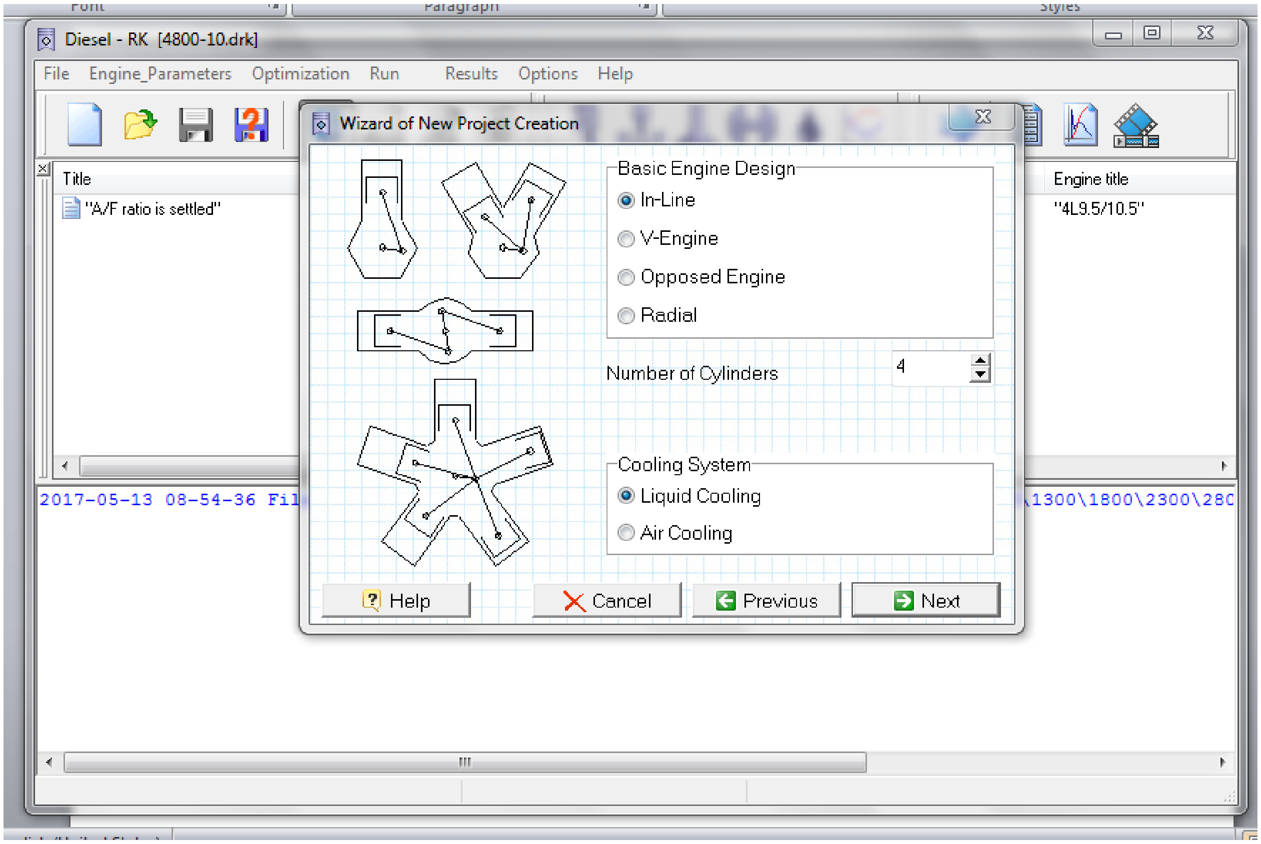


Figure 3.6 show step 4.

In the new window define the type of basic engine design we choose (In-Line) then we determine number of cylinder choose (4), and cooling system (Liquid Cooling) then press (NEXT)

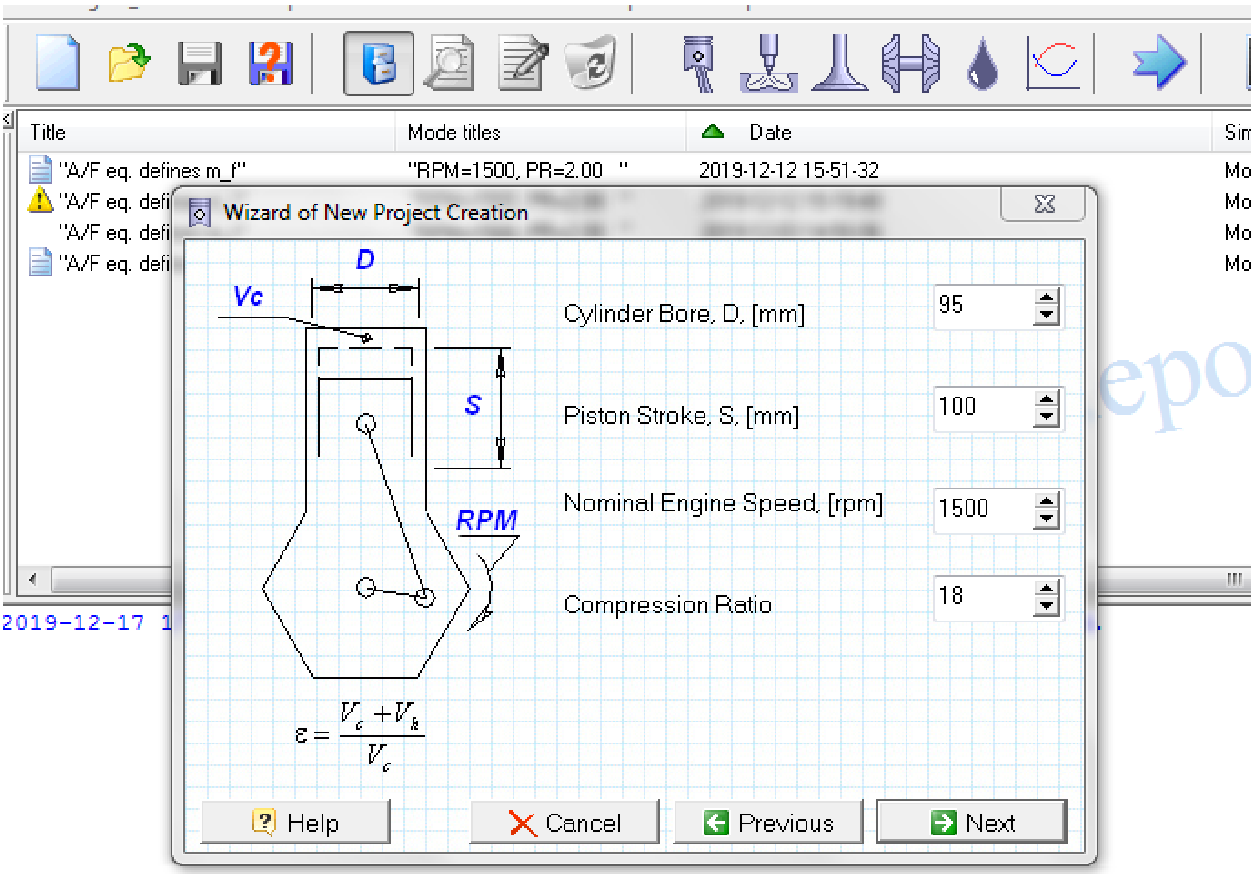


Figure 3.7 show step 5.

In the new window we determine the cylinder bore (95 mm) and piston stroke (100mm) then determine the nominal engine speed (1500 rpm),and compression ratio (18) and number of cylinder then press (NEXT)

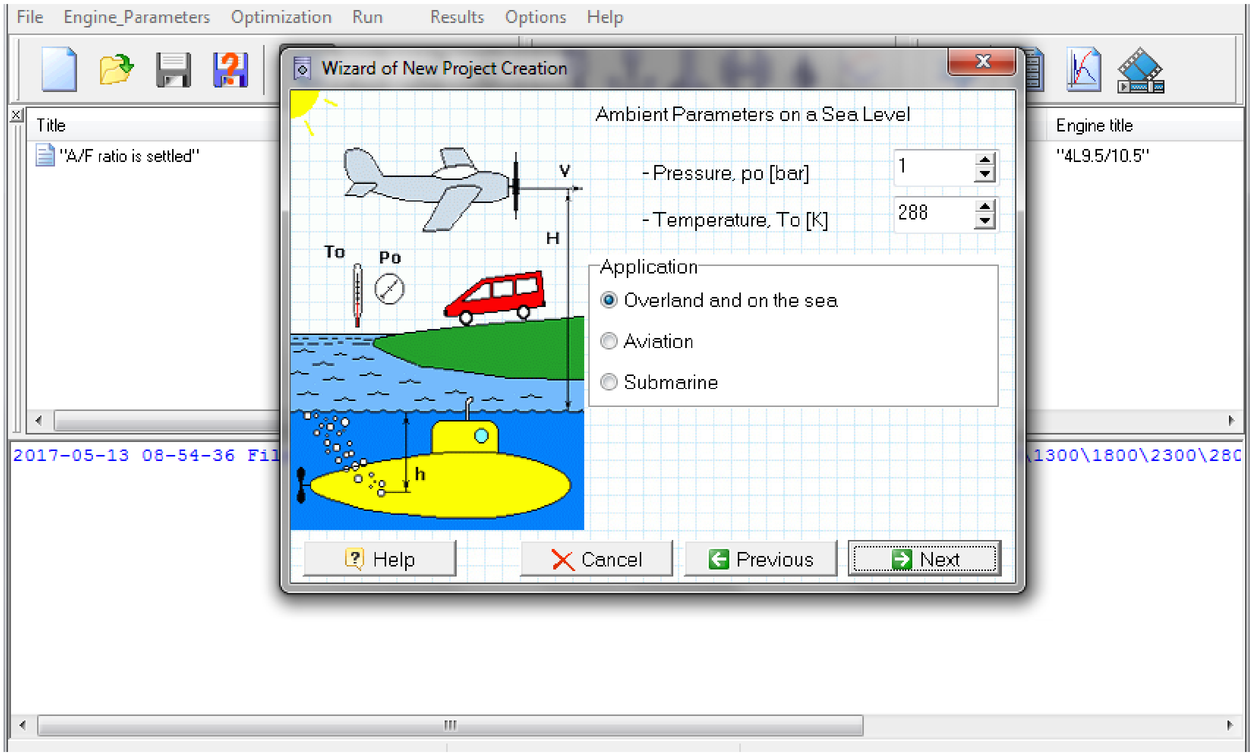


Figure 3.8 show (step 6)

Ambient parameter on sea level window appears then choose the pressure value (1 bar), and temperature value (288 K) and determine the application type so we choose (Over land and on the sea), then press (NEXT)

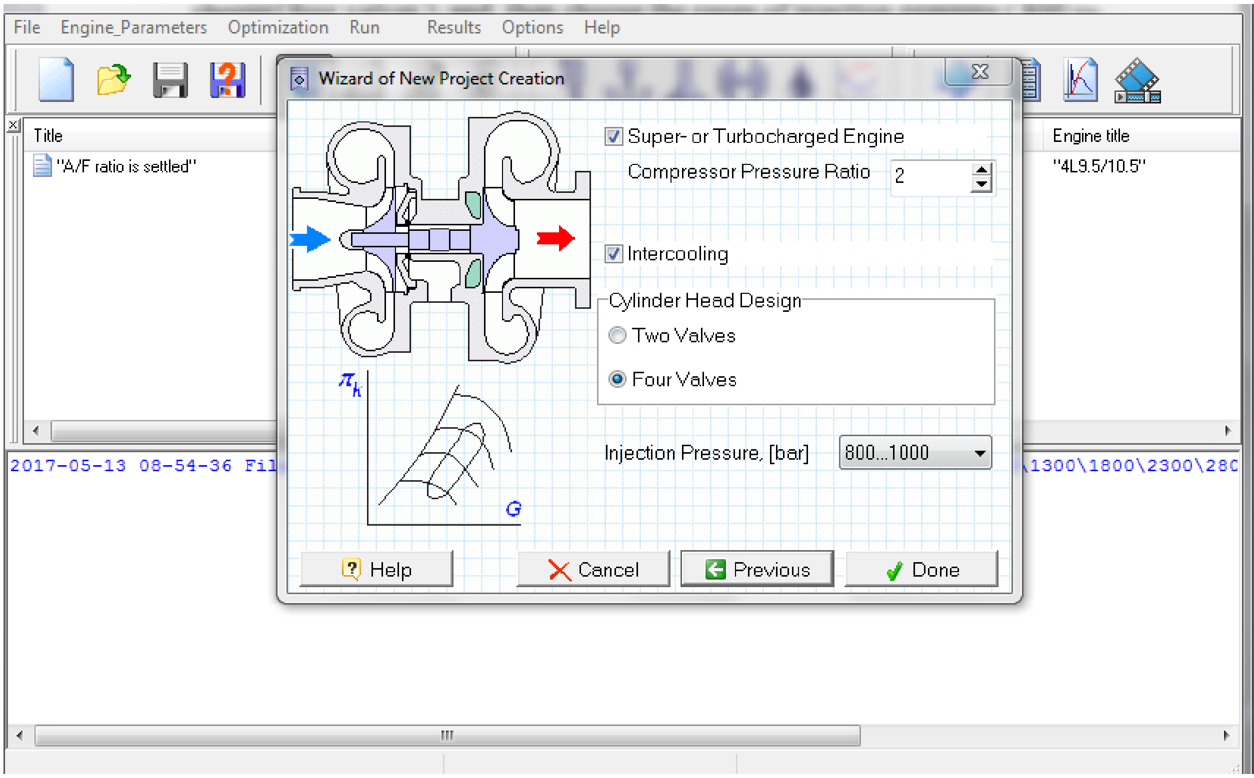


Figure 3.9 Show chose the turbo charger (step 7)

In new window we choose super – or turbocharged engine , then choose Inter cooling , according to cylinder head design we determine the number of valves we choose (four valves), and then choose the range of injection pressure (800 --- 1000 bar) ,then press (Done).

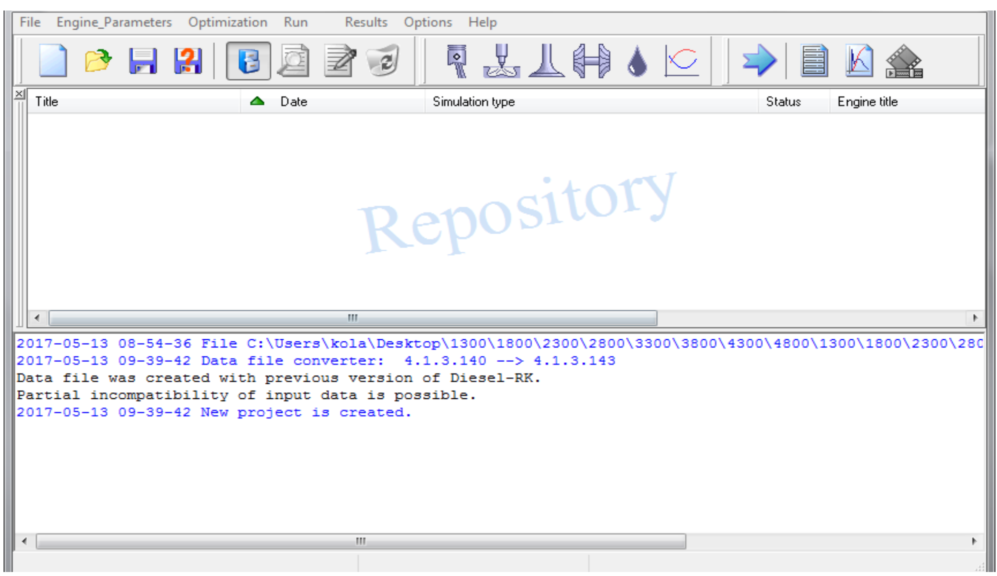


Figure 3.10 show how to save the result file (step 8).

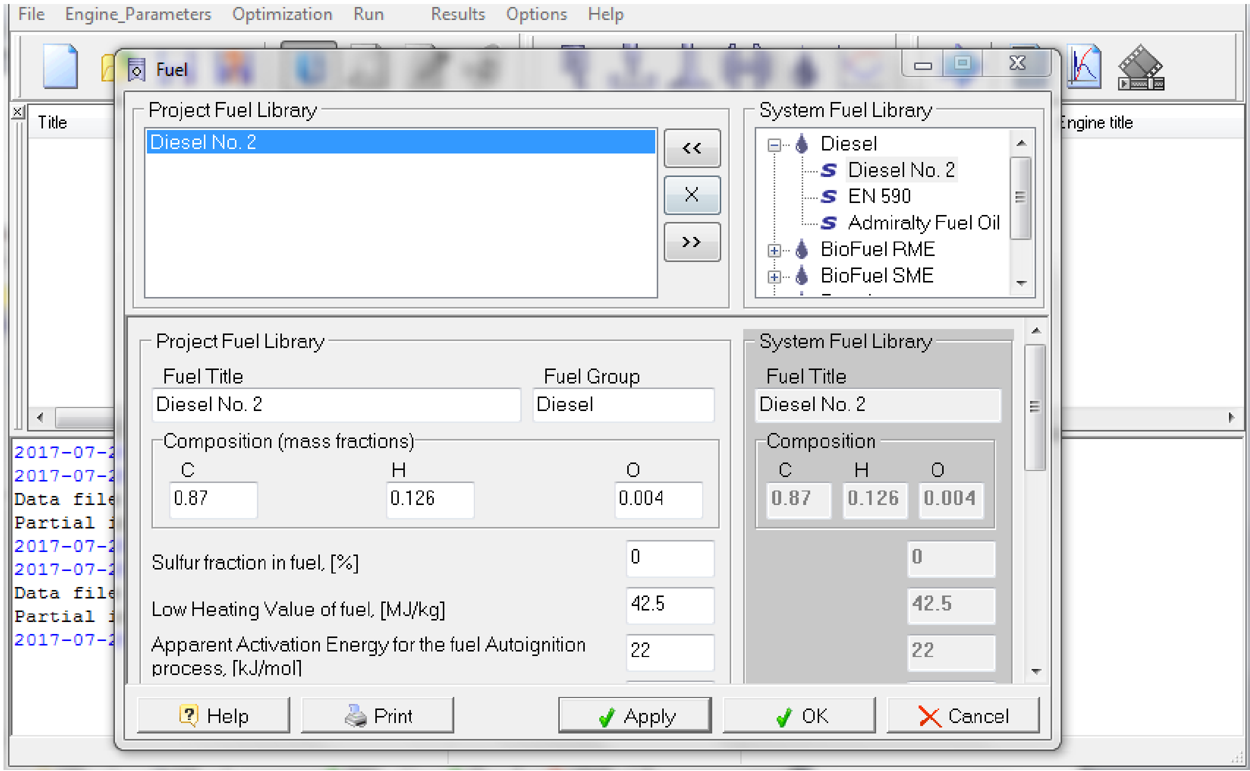


Figure 3.11 show the other fuel input (Step9)

In this stage, we will enter the type and specifications of the fuel that will be used in the machine, and we will use diesel No.2

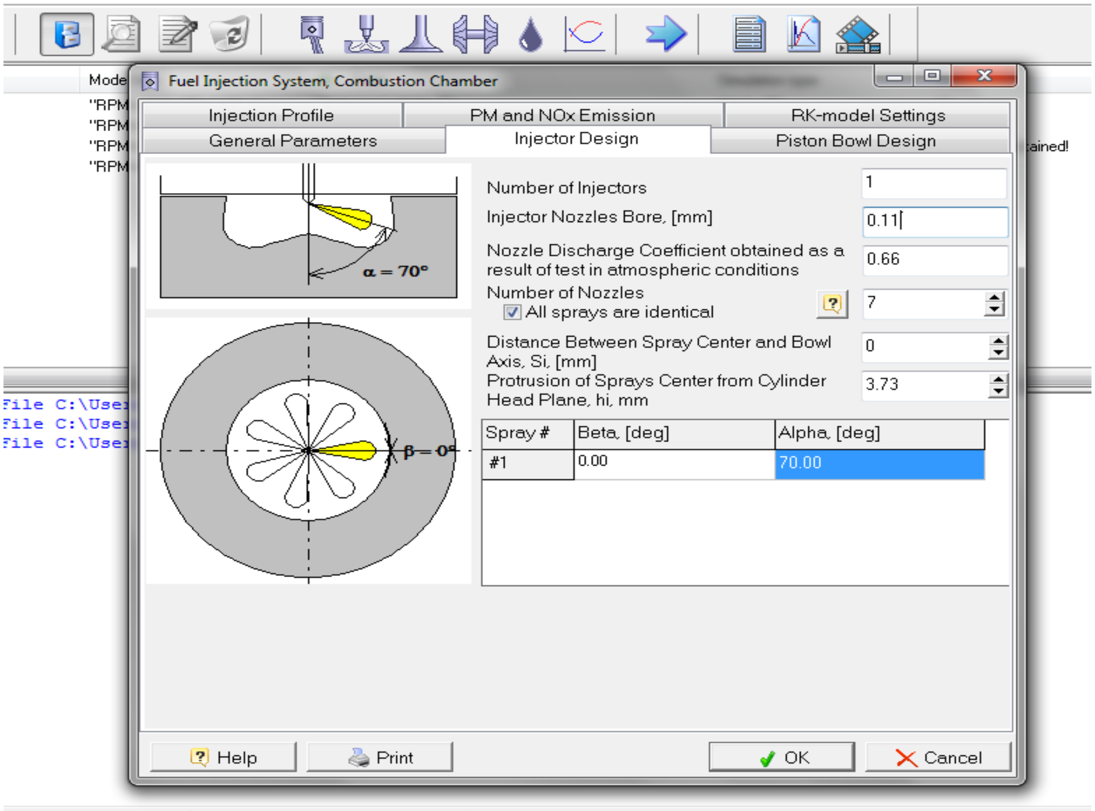


Figure 3.12 show the injector variables (Step10)

In this step, we will enter the parameters of the injector, as this step is the most important in this research as we change these variables in order to obtain lower emissions and higher efficiency.

Number of Injector: here we enter the number of injectors that each cylinder contains

Injector nozzles bore (mm): here we enter the diameter of the injector nozzle in millimeters

Nozzles discharge coefficient : is the ratio of the mass flow rate at the discharge end of the nozzle to that of an ideal nozzle which expands an identical working fluid from the same initial conditions to the same exit pressures is the ratio of the actual discharge to the theoretical discharge,**,** It is entered as a fraction and ranges from 0 to 0.9.

Distance between spray center and Bowl Axis (mm):

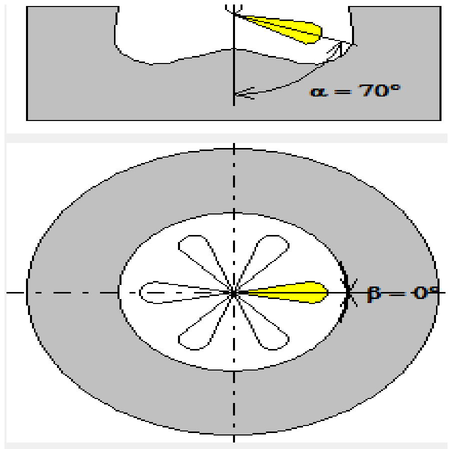


Figure (3-13) shows the Distance between spray center and Bowl Axis=0 mm

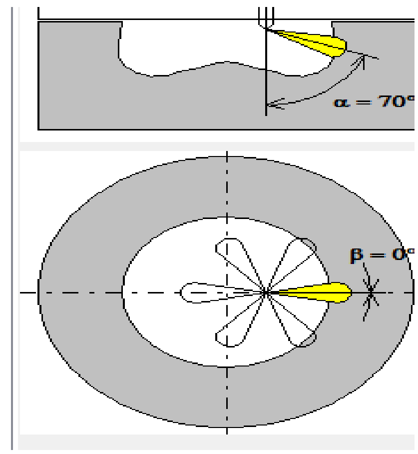


Figure (3-14) shows the Distance between spray center and Bowl Axis= 9 mm

Protrusion between sprays center and cylinder head plane:

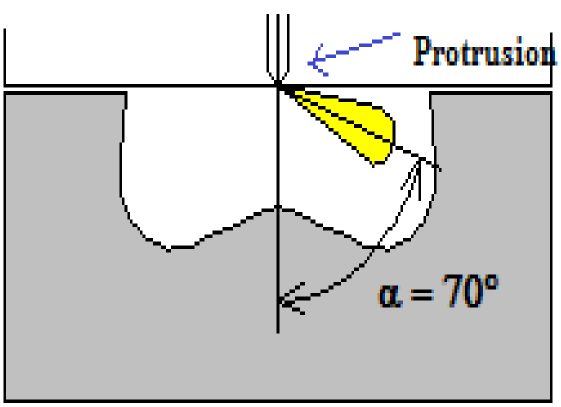


Figure (3-15) Shows the distance between the Protrusion of sprays center and cylinder head plane = 0 mm

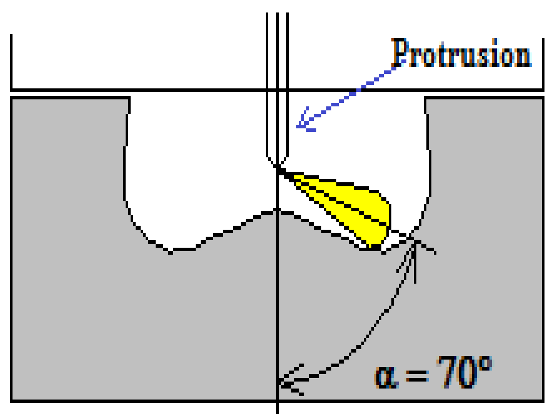
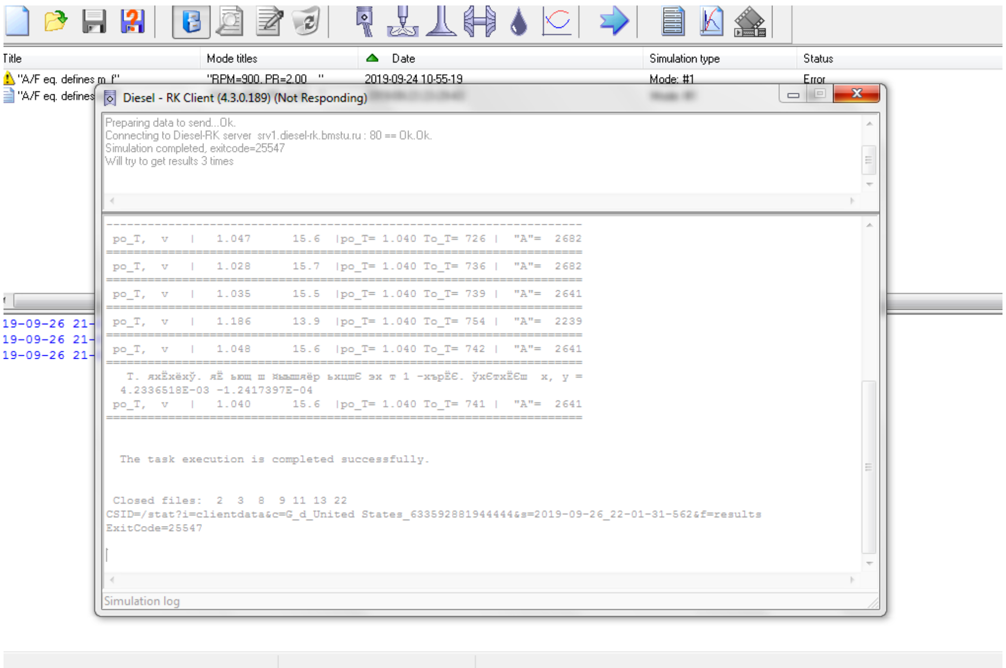
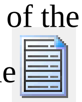


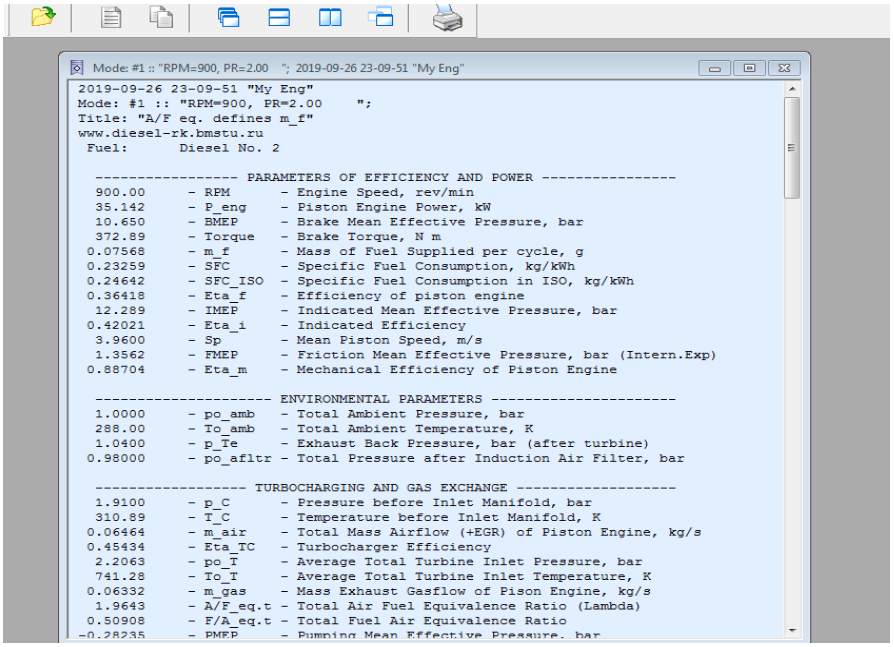
Figure (3-16) Shows the distance between the Protrusion of sprays center and cylinder head plane = 10 mm



In the last step we click to run the program

Figure 3.17 shows a run of the program (Step11)

After the calculation is complete, we click on the  to see the results (mechanical efficiency and emissions)



Mode: #1 :: "RPM=900, PR=2.00   "; 2019-09-26 23-09-51 "My Eng"

```
2019-09-26 23-09-51 "My Eng"
Mode: #1 :: "RPM=900, PR=2.00    ";
Title: "A/F eq. defines m_f"
www.diesel-rk.bmstu.ru
 Fuel:      Diesel No. 2

  ----------------- PARAMETERS OF EFFICIENCY AND POWER ----------------
  900.00     - RPM      - Engine Speed, rev/min
  35.142     - P_eng    - Piston Engine Power, kW
  10.650     - BMEP     - Brake Mean Effective Pressure, bar
  372.89     - Torque   - Brake Torque, N m
 0.07568     - m_f      - Mass of Fuel Supplied per cycle, g
 0.23259     - SFC      - Specific Fuel Consumption, kg/kWh
 0.24642     - SFC_ISO  - Specific Fuel Consumption in ISO, kg/kWh
 0.36418     - Eta_f    - Efficiency of piston engine
  12.289     - IMEP     - Indicated Mean Effective Pressure, bar
 0.42021     - Eta_i    - Indicated Efficiency
  3.9600     - Sp       - Mean Piston Speed, m/s
  1.3562     - FMEP     - Friction Mean Effective Pressure, bar (Intern.Exp)
 0.88704     - Eta_m    - Mechanical Efficiency of Piston Engine

  --------------------- ENVIRONMENTAL PARAMETERS ----------------------
  1.0000     - po_amb   - Total Ambient Pressure, bar
  288.00     - To_amb   - Total Ambient Temperature, K
  1.0400     - p_Te     - Exhaust Back Pressure, bar (after turbine)
 0.98000     - po_afltr - Total Pressure after Induction Air Filter, bar

  ------------------ TURBOCHARGING AND GAS EXCHANGE -------------------
  1.9100     - p_C      - Pressure before Inlet Manifold, bar
  310.89     - T_C      - Temperature before Inlet Manifold, K
 0.06464     - m_air    - Total Mass Airflow (+EGR) of Piston Engine, kg/s
 0.45434     - Eta_TC   - Turbocharger Efficiency
  2.2063     - po_T     - Average Total Turbine Inlet Pressure, bar
  741.28     - To_T     - Average Total Turbine Inlet Temperature, K
 0.06332     - m_gas    - Mass Exhaust Gasflow of Pison Engine, kg/s
  1.9643     - A/F_eq.t - Total Air Fuel Equivalence Ratio (Lambda)
 0.50908     - F/A_eq.t - Total Fuel Air Equivalence Ratio
-0.28235     - PMEP     - Pumping Mean Effective Pressure, bar
```

Figure 3.18 shows result of simulation

CHAPTER FOUR

RESULTS AND DISCUSSION

4-1 Introduction

This section is considered the most important because it illustrates the impact of the injector design on the efficiency and emissions obtained from the program and illustrated graphically.

4-2 The mechanical efficiency and amount of emissions of 4M40 Mitsubishi engine before changing the injector variables

Table (4-1) shows the specifications of the 4M40 Mitsubishi Injector

Number of Injector	1
Injector nozzles bore(mm)	.11 mm
Nozzles discharge coefficient obtained as a result of test in atmospheric conditions, (all sprays are identical)	0.66
Number of Nozzle	7
Distance between spray center and Bowl Axis (mm)	0
protrusion between sprays center and cylinder head plane	3.73 mm

Table (4-2) shows the mechanical efficiency and emissions of PM, CO_2 and NO_X from 4M40 Mitsubishi engine at nominal engine speed 1500 rpm and compression ratio = 18

	Mechanical efficiency	**PM emissions (PPM)**	**CO_2 emissions g/kWh**	**NO_X emissions g/kWh**
4M40 Mitsubishi engine	88.01%	0.17372	714.04	1205.2

4-3 Effect of injector variables on efficiency and emissions of PM, CO_2 and NO_X

In each stage of this part, we will change one of the injection variables in a specific range and install the other variables, and then we record the results of emissions for PM, CO_2 and NO_X and mechanical efficiency at a nominal engine speed 1500 rpm and compression ratio = 18

4-3-1 Effect of injector nozzle pore on mechanical efficiency and emissions (PM, CO_2, NO_x)

Table (4-3) shows the values of mechanical efficiency and amount of emissions calculated by DIESEL-RK when change the injector nozzles pore from 0.05 to 0.32 mm and install the other variables.

Injector nozzle pore(mm)	Mechanical efficiency	PM emissions (PPM)	CO_2 emissions g/kWh	NO_X emissions g/kWh
0.05	88.21%	0.28031	713.27	1008.3
0.06	88.19%	0.27148	713.33	1041.4
0.08	88.18%	0.25445	713.45	1148.5
0.1	88.17%	0.19986	713.59	1262.3
0.12	88.27%	0.15421	710.67	1186.6
0.14	88.36%	0.12512	709.3	1071.9
0.16	88.42%	0.13184	711.7	872.2
0.18	88.43%	0.21024	718.95	692.02
0.2	88.40%	0.32586	729	567.05
0.22	88.37%	0.47334	743.15	459.4
0.24	88.17%	0.78668	772.34	333.46
0.26	87.30%	1.5077	862.68	192.76
0.28	85.38%	2.4361	1049.2	103.9
0.3	81.98%	3.6124	1390.1	54.841
0.32	76.43%	5.2619	2003.7	29.468

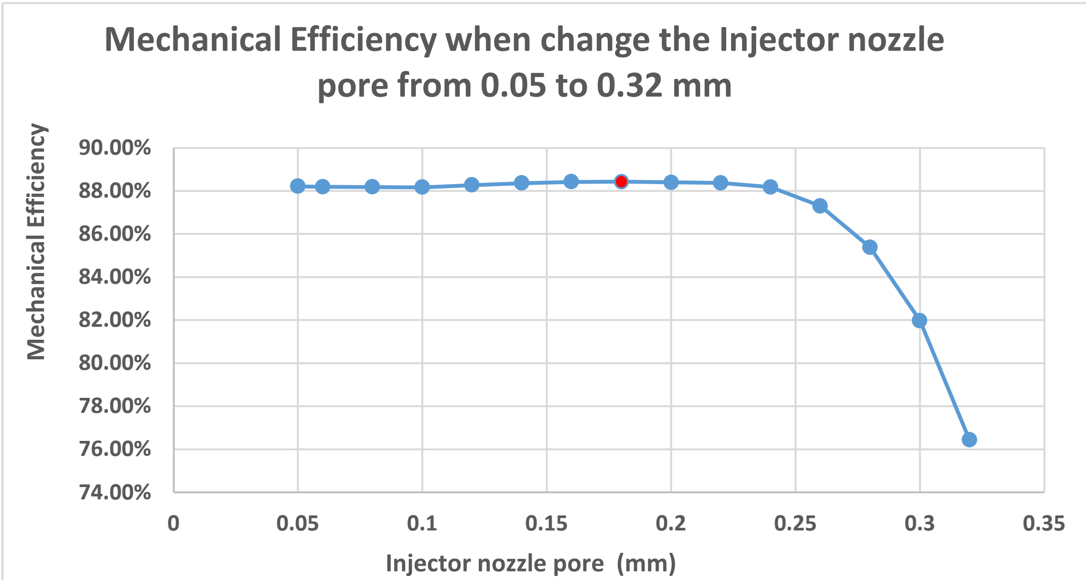


Figure (4-1) Show that the value of Mechanical Efficiency reached its maximum value 88.43% when injector nozzles pore = 0.18 mm and then decrease gradually with the increase in Injector nozzle pore.

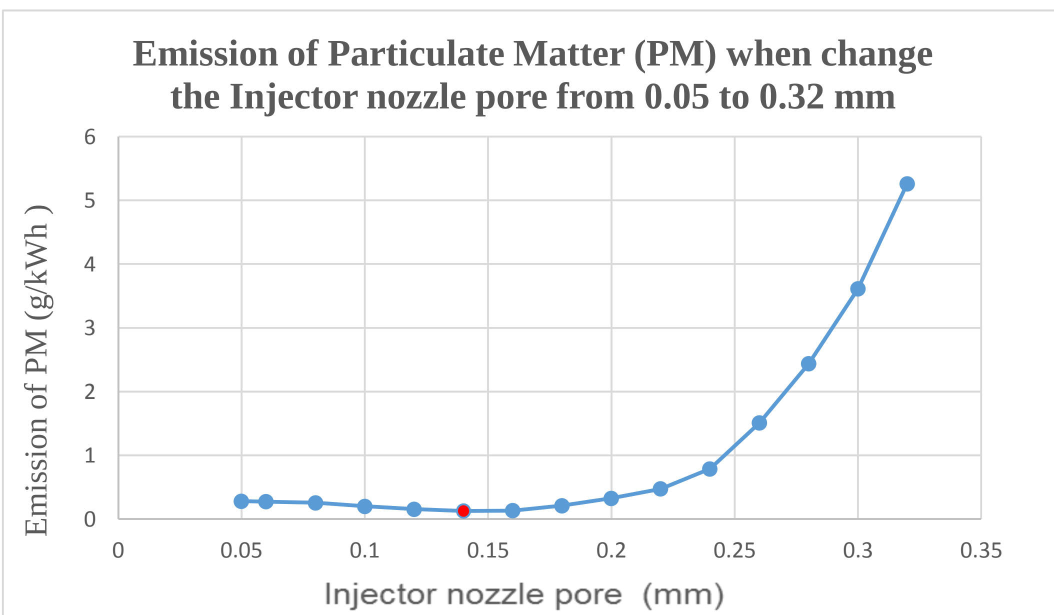


Figure (4-2) Show that the value of PM emissions decrease gradually until reaches to lowest value 0.12512 g/kWh at nozzle pore =.14 mm and then the PM emissions increase gradually with the increase in Injector nozzle pore.

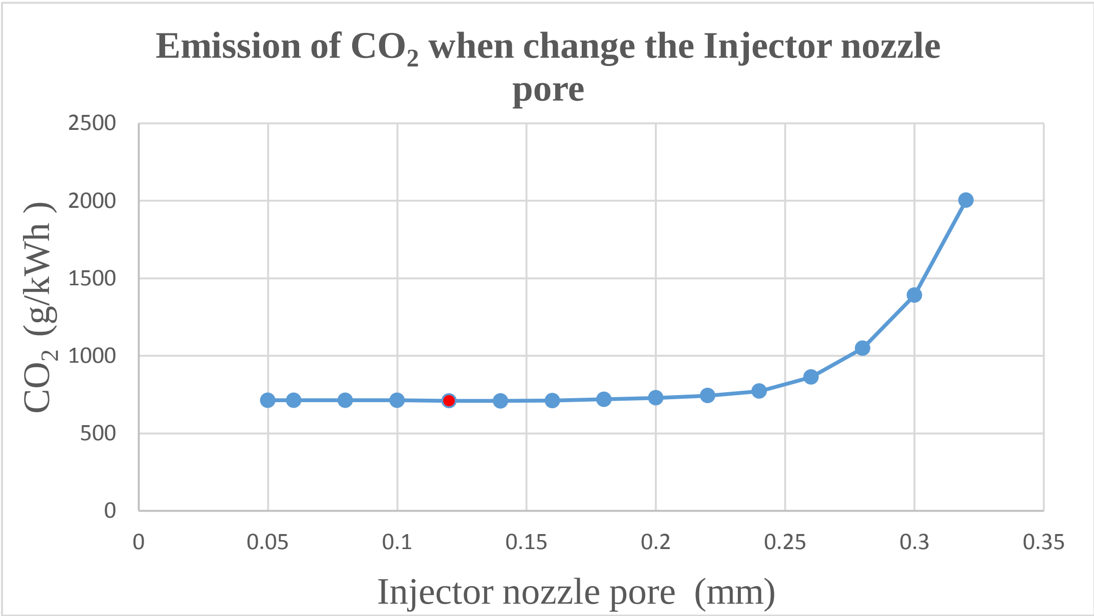


Figure (4-3) Show that the CO_2 Emission is relatively constant until the nozzle pore = 0.2 mm and then the value of CO_2 gradually increases with the increase of the injector nozzle pore.

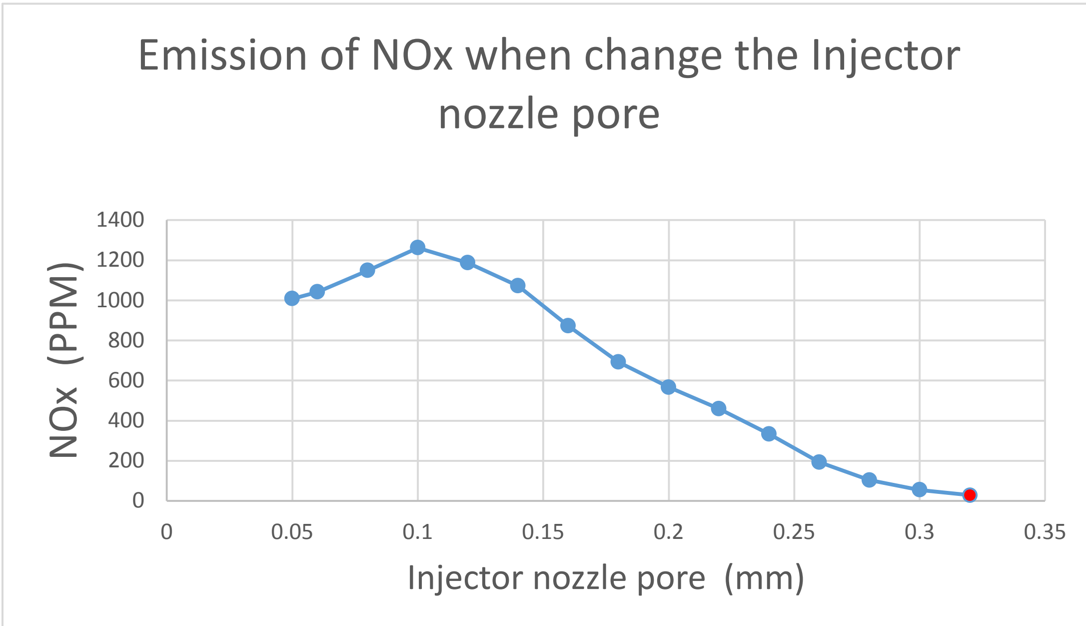


Figure (4-4) Show that the values of NO_X emission increase gradually until reaches to highest value at nozzle pore =.1 mm and then the NO_X emission decrease gradually with the increase in Injector nozzle pore.

4-3-2 Effect of number of nozzles on mechanical efficiency and emissions (PM, CO_2, NOx)

Table (4-4) shows the values of mechanical efficiency and amount of emissions calculated by DIESEL-RK when change the number of nozzles from 0 to 20 nozzles and install the other variables.

No. of nozzle	Mechanical efficiency of	PM emissions (PPM)	CO_2 emissions g/kWh	NO_X emissions g/kWh
1	88.04%	0.25836	710.99	1134.5
2	88.06%	0.25652	712.34	1121.9
3	88.10%	0.25257	712.59	1132
4	88.09%	0.23158	712.1	1169.2
5	88.04%	0.19725	713.09	1238
6	88.02%	0.18102	713.07	1281.6
7	88.01%	0.1737	711.67	1205.2
8	88.19%	0.17413	712.43	1127.8
9	88.18%	0.17575	711.67	1100.4
10	88.09%	0.21079	713.52	960.52
11	88.02%	0.22104	712.14	1033.6
12	88.00%	0.24451	711.67	1010.7

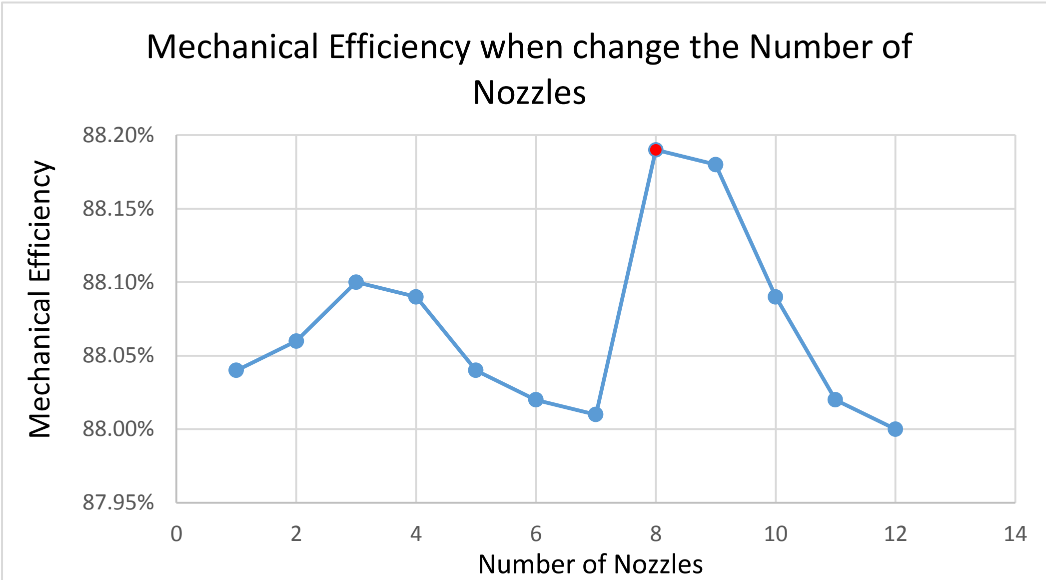


Figure (4-5) shows that there is a slight change in the efficiency value when the number of nozzles changes, reaching a maximum value 88.19% when the number of nozzles = 8 nozzles.

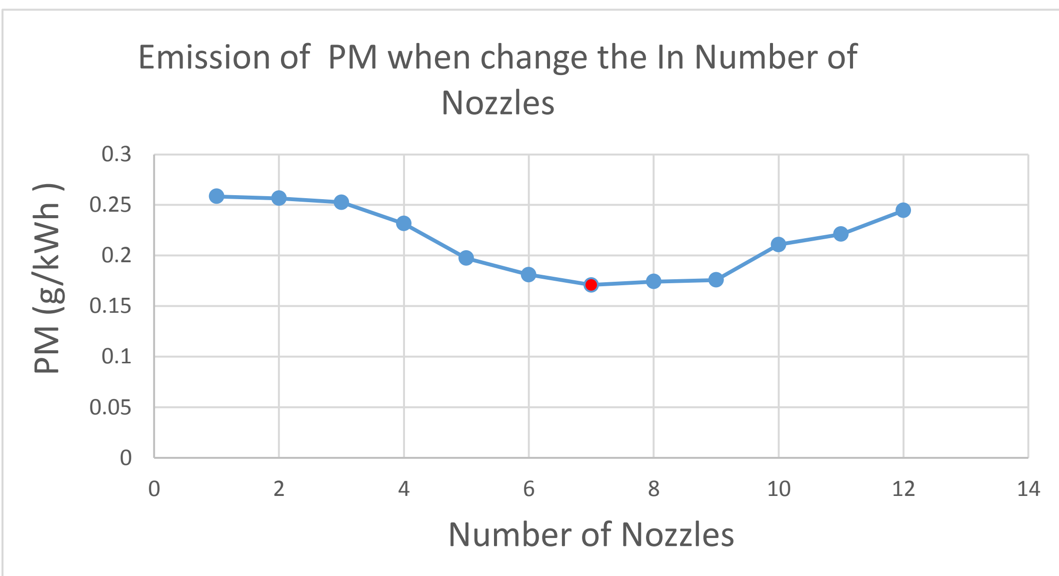


Figure (4-6) shows that the value of PM emissions gradually decreases with increasing the number of nozzles until reach the lowest value when number of nozzles = 7 nozzles, and then the emission of PM continues to increase.

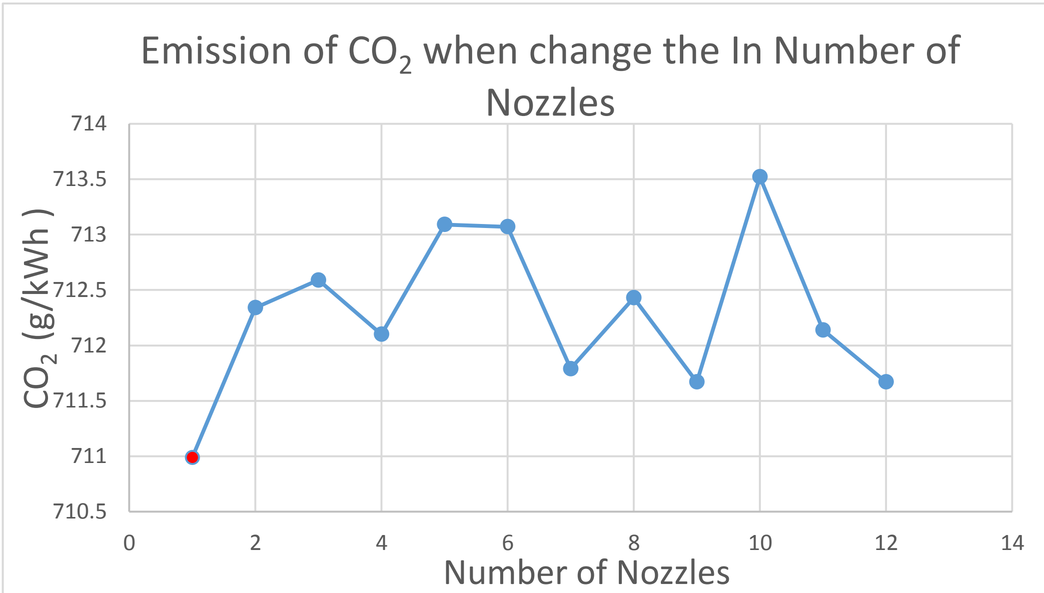


Figure (4-7) shows that there is a slight change in a mount of CO_2 emission when the number of nozzles changes, all values are between 711-713.5 g/kWh, and hence, the impact of the number of nozzles on CO_2 emissions is very small.

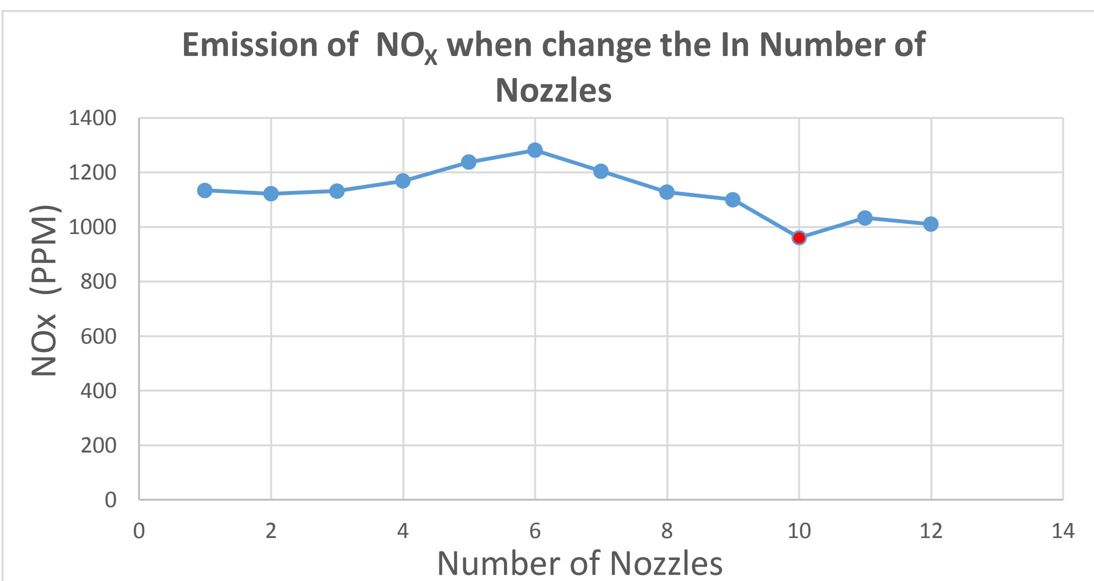


Figure (4-8) Show that the values of NO_X emission increase gradually until reaches to highest value at the number of nozzles =6 nozzles and then the NO_X emission decrease until reaches the lowest value No_X =960.52 ppm at number of nozzles = 10 and then increase gradually.

4-3-3 Effect of distance between spray center and Bowl Axis on mechanical efficiency and emissions (PM, CO_2, NOx)

Table (4-5) shows the values of mechanical efficiency and amount of emissions calculated by DIESEL-RK when change the distance between spray center and Bowl Axis from 1 to 20 nozzles and install the other variables.

Distance between spray center and Bowl Axis (mm)	**Mechanical efficiency of**	**PM emission (PPM)**	**CO_2 emission g/kWh**	**NO_X emission g/kWh**
0	88.01%	0.17372	711.3	820.56
1	88.02%	0.17991	712.31	930.48
2	88.03%	0.18766	712.48	1039.8
3	88.02%	0.19142	712.61	1091.6
4	88.02%	0.19569	712.45	1067.4
5	88.01%	0.20072	712.74	1039.8
10	87.90%	0.21799	713.73	965.13
15	87.70%	0.22911	714.06	922.48
20	87.50%	0.24168	714.77	906.56

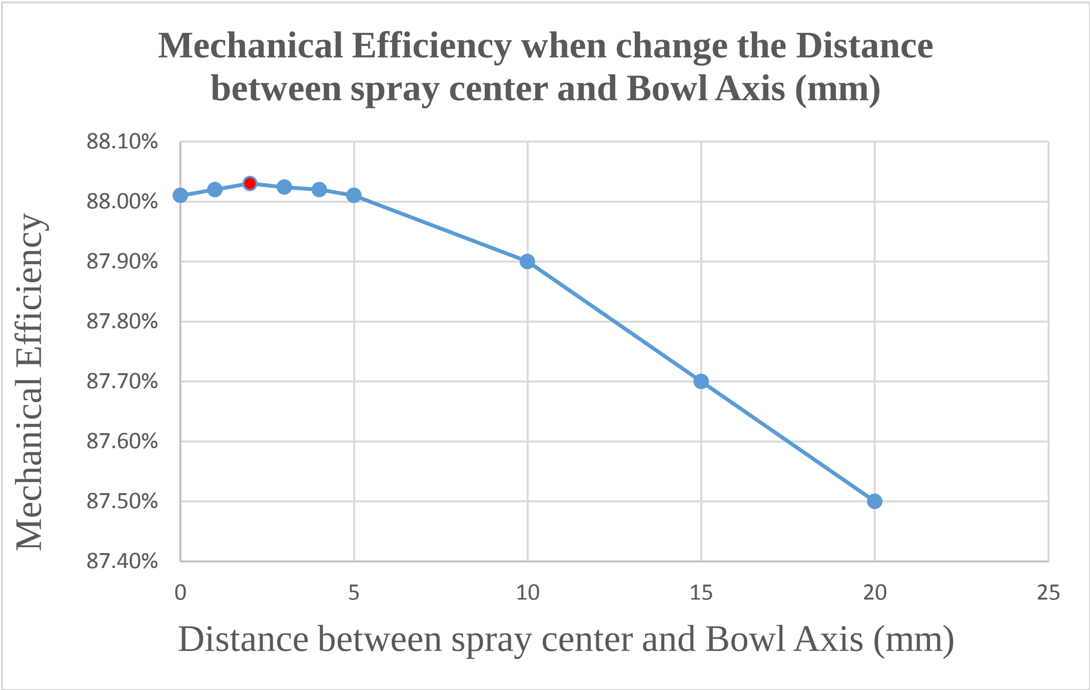


Figure (4-9) shows that there is a slight change in the efficiency value when the distance between spray center and Bowl Axis changes, reaching a maximum value 88.03%% when the distance = 2 mm

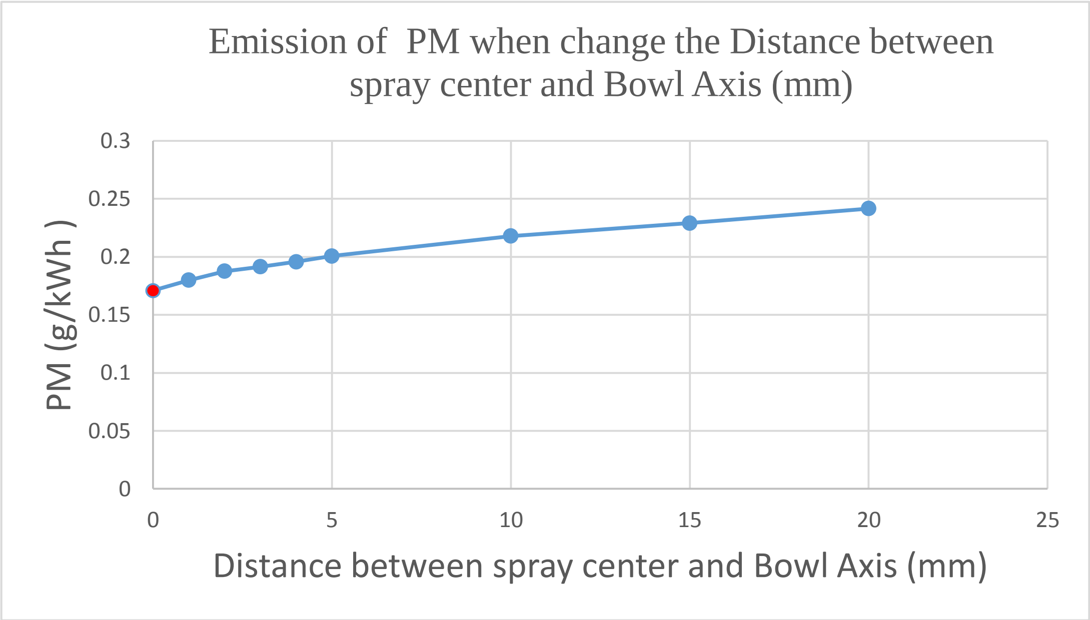


Figure (4-10) Show that the increase in PM emission is directly proportional to the increase in distance between spray center and Bowl Axis, the Minimum value of PM emission = 0.17089 g/kWh at distance between spray center and Bowl Axis = 0 mm.

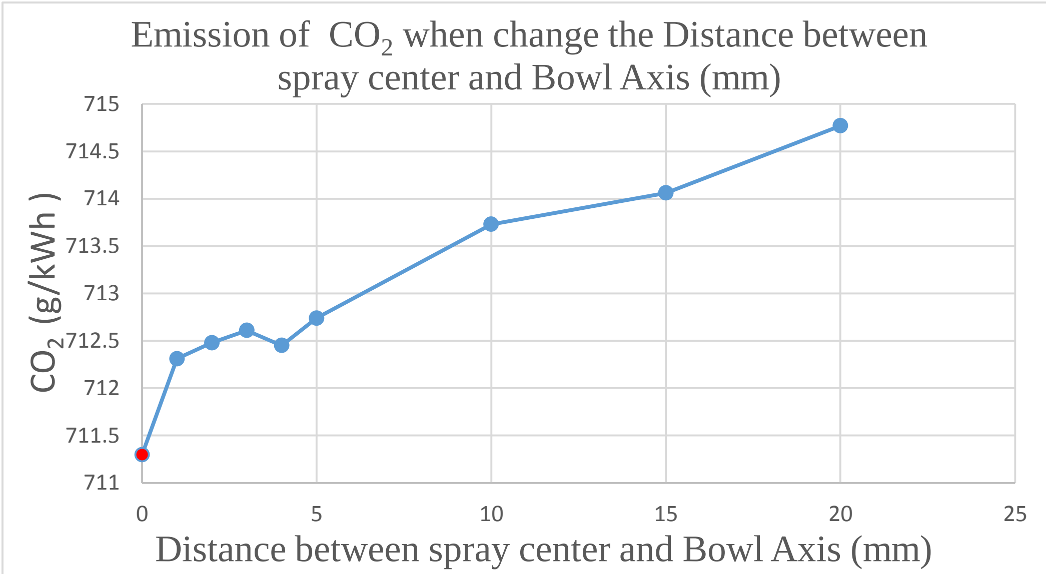
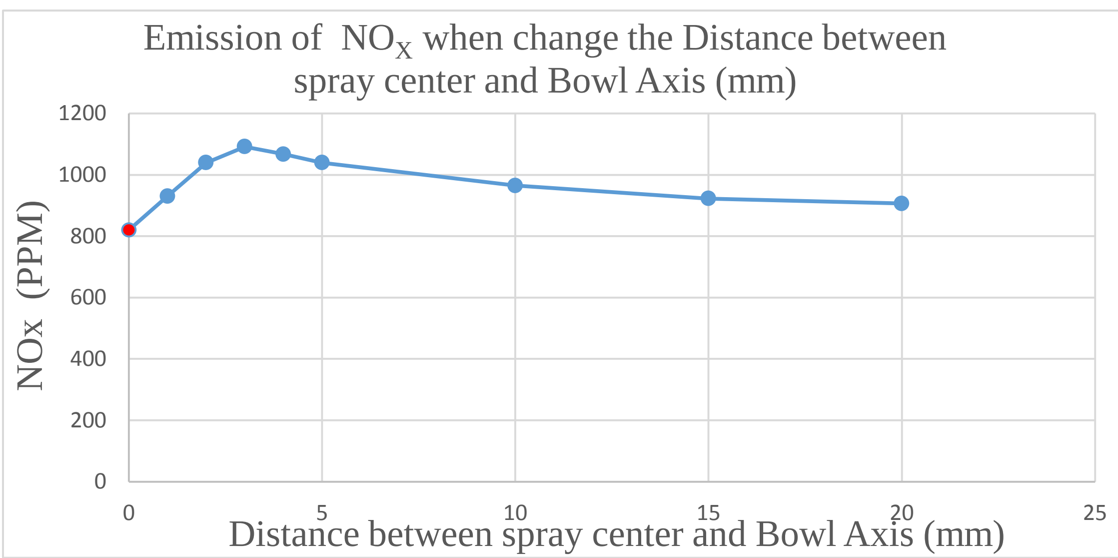


Figure (4-11) shows that there is a slight change in a mount of CO_2 emission when the distance between spray center and Bowl Axis changes, all values are between 711.3-714.7 g/kWh, and hence, the impact of the distance between spray center and Bowl Axis on CO_2 emission is very small, the Minimum value of CO_2 occur when the distance between spray center and Bowl Axis



= 0 mm.

Figure (4-12) Show that the values of NO_X emission increase gradually until reaches to highest value at the distance between spray center and Bowl Axis = 3mm and then the NO_X emission decrease until reaches the No_X =906.56 ,the lowest value of NO_X emission = 820.56 ppm at the distance between spray center and Bowl Axis =0 mm.

4-3-4 Effect of the distance between the protrusion sprays center and cylinder head plane on mechanical efficiency and emissions (PM, CO_2, NOx)

Table (4-6) shows the values of mechanical efficiency and amount of emissions calculated by DIESEL-RK when change the distance between the protrusions of sprays center and cylinder head plane from 0 to 10 mm and install the other variables.

Distance between the protrusions of sprays center and cylinder head plane	Mechanical efficiency of	PM emission (PPM)	CO2 emission g/kWh	NOX emission g/kWh
0	88.00%	0.17129	712.71	1220.4
1	88.00%	0.17112	713.88	1222.1
2	88.00%	0.17132	713.2	1217
3	88.00%	0.17269	714.95	1210.7
3.73	88.01%	0.17372	714.04	1205.2
4	88.02%	0.1746	713.95	1198.4
5	88.03%	0.1724	712.05	1190.4
6	88.04%	0.17815	711.99	1156.6
7	88.04%	0.17559	712.76	1172.2
8	88.04%	0.17564	713.92	1182.3
9	88.04%	0.17564	713.97	1183.5
10	88.04%	0.17984	713.66	1155.2

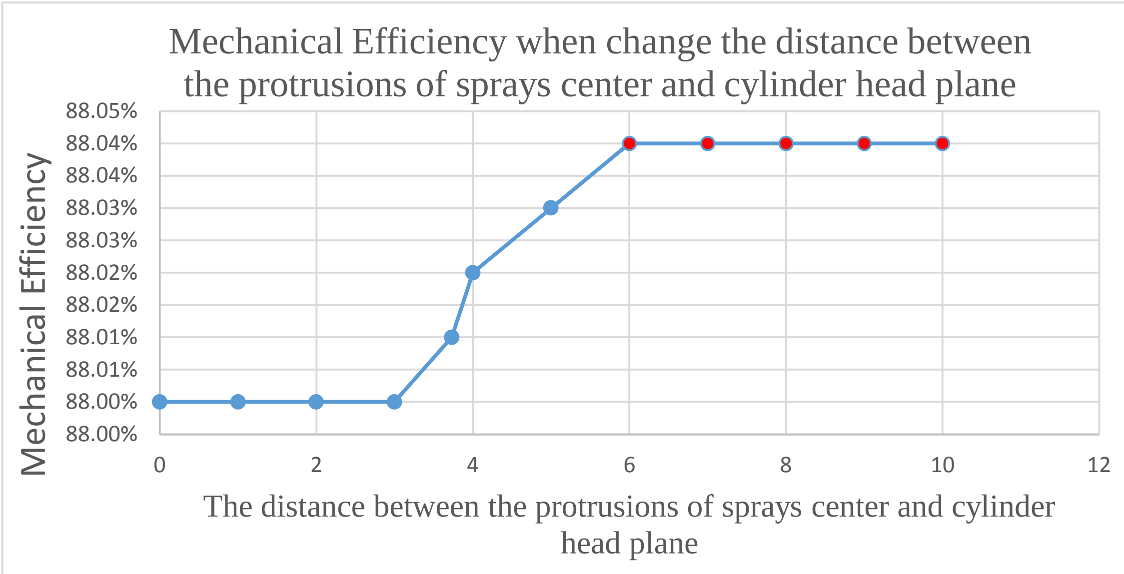


Figure (4-13) shows that there is a slight change in the efficiency value when the protrusion between sprays center and cylinder head plane changes.

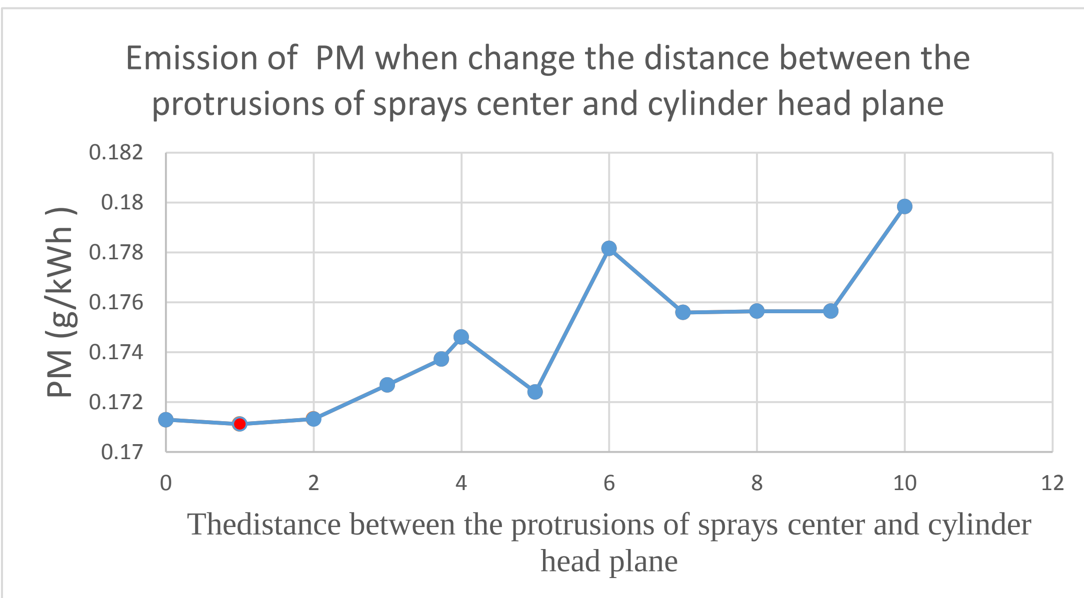


Figure (4-14) Show that the increase in PM emission is directly proportional to the increase the protrusion between sprays center and cylinder head plane , the Minimum value of PM emission = 0.17112 g/kWh at the protrusion between sprays center and cylinder head plane = 1 mm.

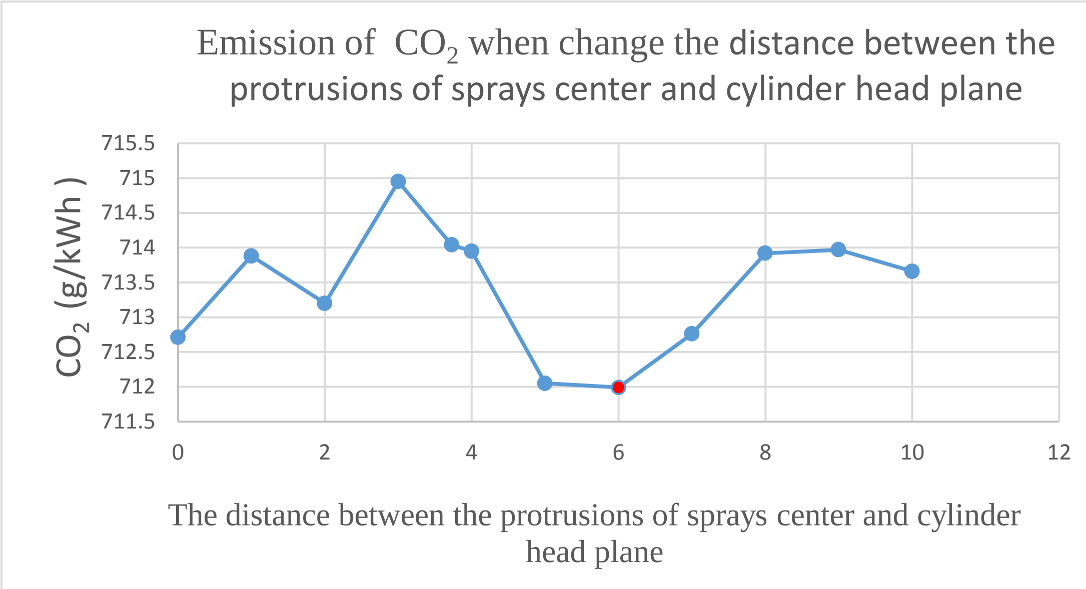


Figure (4-15) shows that there is a slight change in a mount of CO_2 emission when the protrusion between sprays center and cylinder head plane changes, all values are between 711.99-714.95 g/kWh, and hence, the impact of high of the protrusion between sprays center and cylinder head plane on CO_2 emissions is very small.

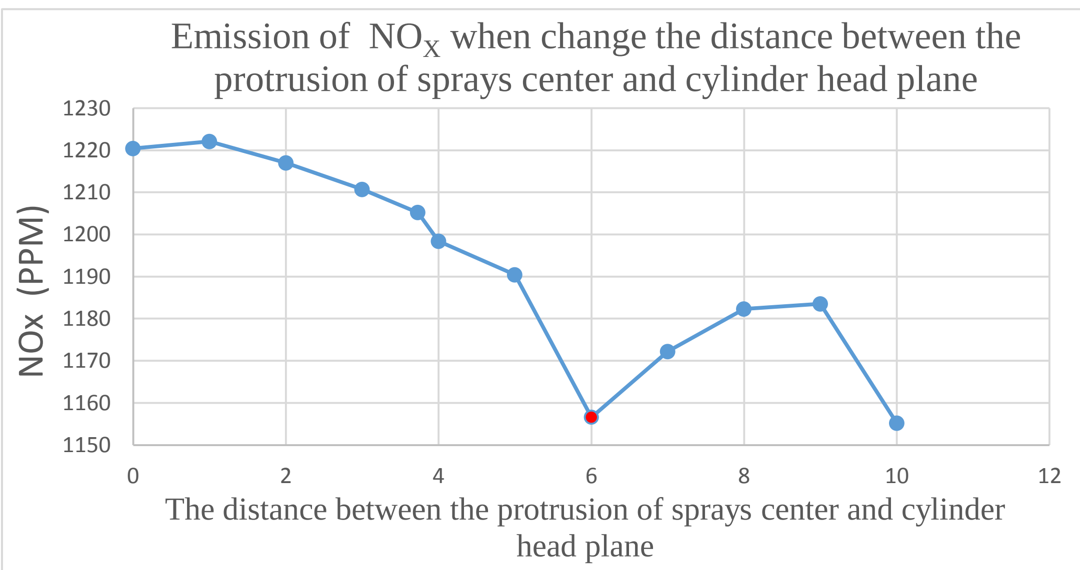


Figure (4-16) Show that the values of NO_X emission inversely proportional to the increase in the protrusion between sprays center and cylinder head plane. The Minimum values of NO_X

emission = 1156.6, 1155.2 at the protrusion between sprays center and cylinder head plane =6 and 10 mm.

4-3-5 Effect nozzles discharge coefficient on mechanical efficiency and emissions (PM, CO_2, NOx)

Table (4-7) shows the values of mechanical efficiency and amount of emissions calculated by DIESEL-RK when change the Nozzles discharge coefficient from 0.4 - 0.9 and install the other variables.

Nozzles discharge coefficient	Mechanical efficiency of	PM emission (PPM)	CO_2 emission g/kWh	NO_X emission g/kWh
0.4	87.92%	0.22423	716.13	1299.5
0.5	87.93%	0.19625	715.21	1256.4
0.6	87.96%	0.17935	714.83	1230.2
0.65	87.98%	0.17533	714.22	1211.7
0.66	88.01%	0.17372	714.04	1205.2
0.67	88.02%	0.17082	713.42	1212.6
0.7	88.02%	0.16741	712.38	1222.9
0.8	88.06%	0.15779	711.23	1250.2
0.9	88.09%	0.14888	710.03	1291.5

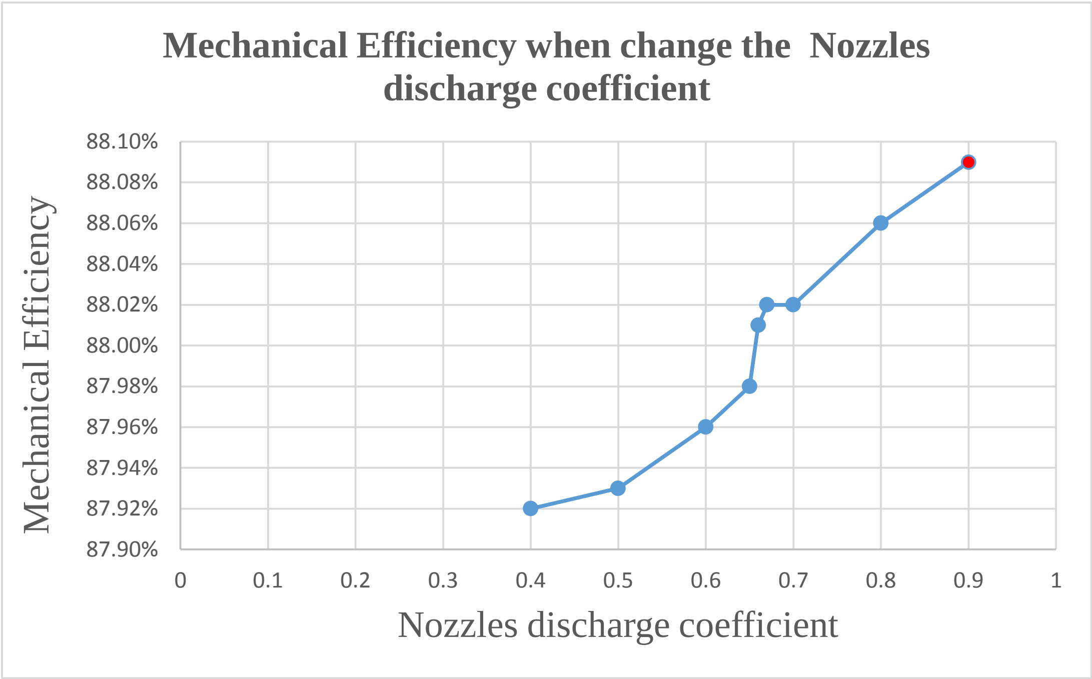


Figure (4-17) shows that there is a slight change in the efficiency value when the Nozzles discharge coefficient changes.

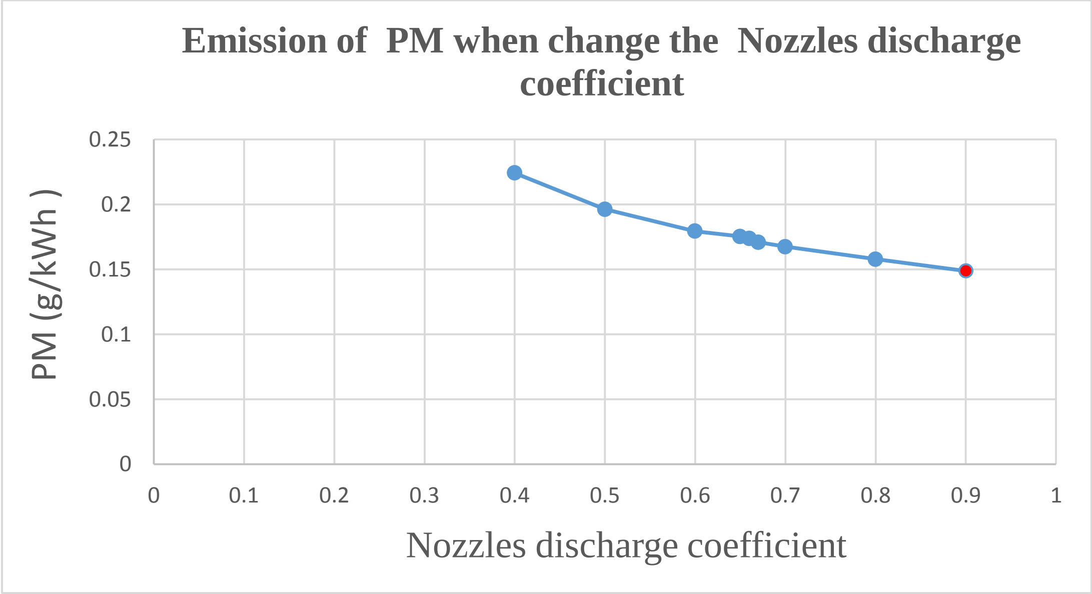


Figure (4-18) Show that the increase in PM emission is inversely proportional to the increase to the increase Nozzles discharge coefficient, the Minimum value of PM emission = 0.1488 g/kWh at the Nozzles discharge coefficient = 0.9

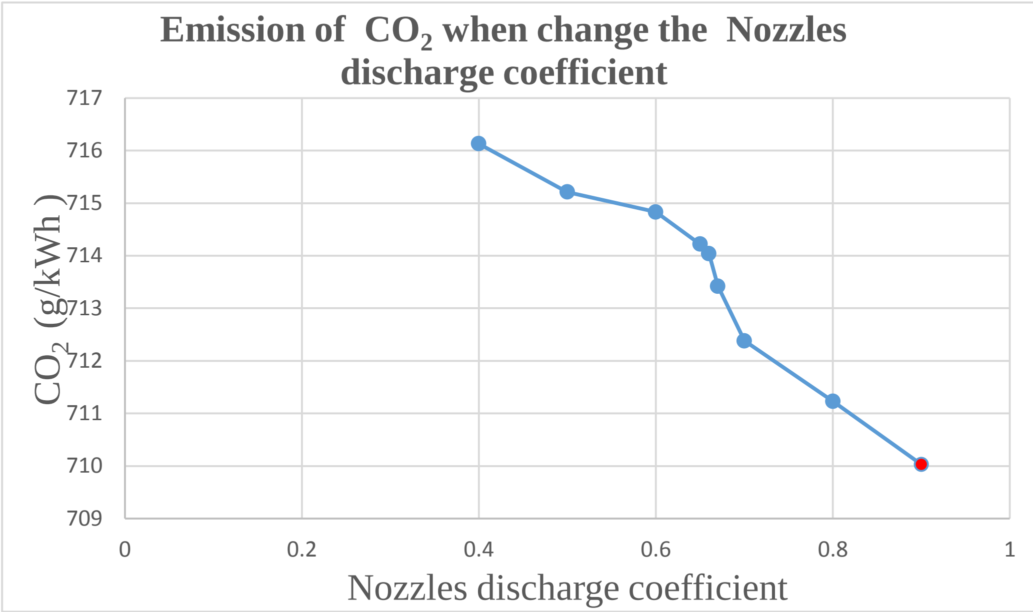


Figure (4-19) shows that there is a slight change in a mount of CO_2 emission when Nozzles discharge coefficient changes, all values are between 710.3-716.13 g/kWh, and hence, the impact of Nozzles discharge coefficient on CO_2 emissions is very small.

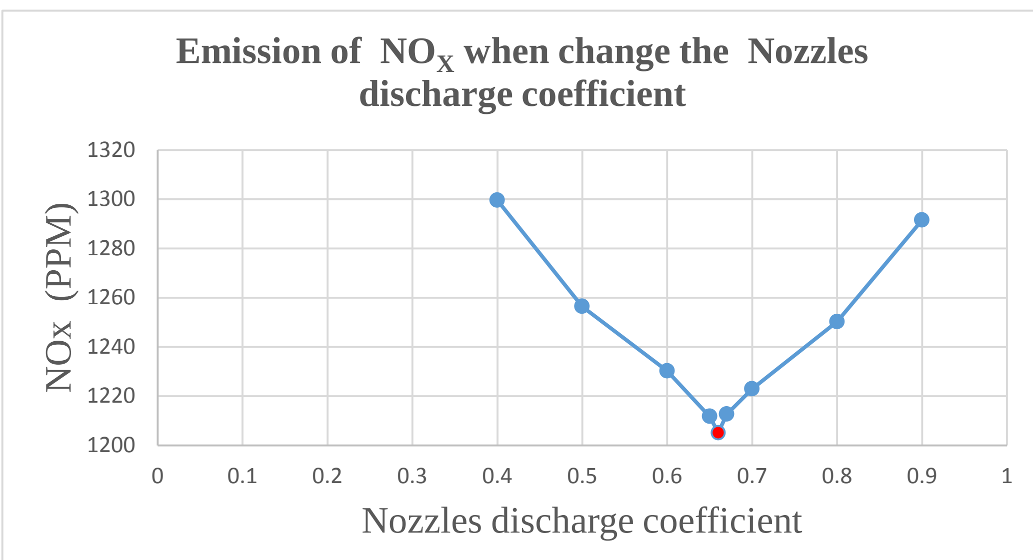


Figure (4-20) Show that the values of NO_X emission is changed with the change of Nozzles discharge coefficient. The Minimum NOx emissions =1205.2 ppm at Nozzles discharge coefficient = 0.66.

4.4 Discussion

4.4.1 Mechanical efficiency and amount of exhaust emissions when change the injector nozzles pore

The simulation results are shown in every case, in each case insert a different value of the injector nozzle pore, where we start from the lowest value can be entered in DIESEL-RK where the nozzle pore=.05 mm and continue in ascending to enter the values of injector nozzle pores and in each case we monitor the emissions and the value of mechanical efficiency, These are the values of injector nozzle pore entered in the program (0.05, 0.06, .08, 0.1, 0.12, 0.14, 0.16, 0.18, 0.2, 0.22, 0.24, 0.26, 0.28, 0.3, 0.32) mm.

From the results obtained from DIESEL-RK in table (4-1) we can note that when the increase in the diameter of the injector nozzle in ascending, at first the impact on mechanical efficiency is very slight until we reach the injector nozzle pore =0.24 mm and then the efficiency decreases significantly until it reaches 76.43% at injector nozzle pore=0.32 mm and from here we can conclude that Using a diameter from 0.05to 0.24 mm gives a high efficiency 88% Figure (4-1).

For the PM emission the lowest values are 0.12512 g/kWh and 0.12512 g/kWh at injector nozzle pore =.14 mm .16 mm Figure (4-2).

For CO_2 emissions, in the beginning at the Injector nozzles bore in range between 0.05 and 0.24 mm the change in amount of CO_2 emissions is very small (709 – 772) g/kWh and after that the CO_2 emission increase until to reach the maximum value 2003.7 g/kWh at the injector nozzle pore = 0.32 mm Figure (4-3).

For NO_X emissions the values of NO_X emission increase gradually until reaches to highest value at nozzle pore =.1 mm where the NO_X emissions = 1262.3 ppm and then the NO_X emission decrease gradually with the increase in Injector nozzle pore Figure (4-4)..

From these observations conclude that the most suitable nozzle pores is 0.16 where it has the highest efficiency 88.4% and the lowest emissions (PM= 0.13184 g/kWh, CO_2=711.7 g/kWh NO_X=872.2 ppm).

4.4.2 Mechanical efficiency and amount of exhaust emissions when change the number of nozzles

In this stage the values of the number of nozzles are entered in DIESEL-RK and these values start from 1 to 12 nozzles and we monitoring the results of efficiency and emissions in each case.

From the results obtained from DIESEL-RK in Table (4-2), we can note that when increasing the number of nozzles, the mechanical efficiency changes very slightly and this indicates that the change in the number of nozzles does not significantly affect the mechanical efficiency Figure (4-5).

It was also observed that the lowest value of PM emissions=0.17093 g/kWh and 0.17413 g/kWh at number of nozzle = 7 and 8 nozzles Figure (4-6).

For CO_2 emissions, the change in the amount of emissions is very small all values range from 711 to713.5 g/kWh, and hence we conclude that the effect of the number of nozzles on CO_2 emissions is very small Figure (4-7).

For NO_X emissions the values of NO_X emission increase gradually until reaches to highest value at the number of nozzles =6 nozzles and then the NO_X emission decrease until reaches the lowest value No_X =960.52 ppm at number of nozzles = 10 and No_X =1127.8 ppm at number of nozzles = 8 then increase gradually Figure (4-8).

From these observations conclude that the most suitable number of nozzles is 8 nozzles where it has the highest efficiency 88.7% and the lowest emissions can be obtained (PM= 0.17093 g/kWh, CO_2=712.43 g/kWh, NO_X=1127.8 ppm).

4.4.3 Mechanical efficiency and amount of exhaust emissions when change the distance between spray center and Bowl Axis

In this stage we enter the values of the distance between spray center and Bowl Axis by millimeter into DIESEL-RK and these values range from 0 to 20 mm and then monitor the results of mechanical efficiency and emissions in each distance.

From the results obtained from DIESEL-RK in Table (4-3), note that when increasing the distance between spray center and Bowl Axis, the mechanical efficiency changes very slightly and this indicates that the change in the distance between spray center and Bowl Axis does not significantly affect the mechanical efficiency Figure (4-9).

For the PM emissions the lowest value of PM emissions = 0.17089 g/kWh at the distance between spray center and Bowl Axis = 0 mm Figure (4-10).

For CO_2 emissions, the change in the amount of emissions is very small, the lowest value of CO_2 emission =711.3 g/kWh at distance between spray center and Bowl Axis=0 mm. Figure (4-11).

For NO_X emissions note that the values of NO_X emission increase gradually until reaches to highest value at the distance between spray center and Bowl Axis = 3mm and then the NO_X emission decrease until reaches the No_X =906.56 ,the lowest value of NO_X emission = 820.56 ppm at the distance between spray center and Bowl Axis =0 mm Figure (4-12).

From these observations conclude that the most suitable distance between spray center and Bowl Axis is 0 mm where it has the highest efficiency 88.7% and the lowest emissions can be obtained (PM= 0.17089 g/kWh, CO_2=711.3 g/kWh, NO_X=820.56 ppm).

4.4.4 Mechanical efficiency and amount of exhaust emissions when change the distance between the protrusion of sprays center and cylinder head plane

In this stage the values of the distance between the protrusion of sprays center and cylinder head plane by millimeter entered into DIESEL-RK and these values range from 0 to 10 mm and then monitor the results of mechanical efficiency and emissions in each distance.

From the results obtained from DIESEL-RK in Table (4-4), note that when increasing the distance between the protrusion of sprays center and cylinder head plane, the mechanical efficiency changes very slightly and this indicates that the change in the distance between the protrusion of sprays center and cylinder head plane does not significantly affect the mechanical efficiency Figure (4-13).

For the PM emissions the minimum value = 0.17112 g/kWh at the distance between the protrusion of sprays center and cylinder head plane = 1 mm Figure (4-14).

For CO2 emissions, the change in the amount of emissions is very small, all values are between 711.99-714.95 g/kWh, and hence we can conclude the impact of high of the distance between the protrusion of sprays center and cylinder head plane on CO2 emissions is very small Figure (4-15).

For NOX emissions note that the values of NOX emission inversely proportional to the increase in The distance between the protrusion of sprays center and cylinder head plane, the Minimum values of NOX emission = 1156.6, 1155.2 at the distance between the protrusion of sprays center and cylinder head plane = 6,10 mm(4-15).

From these observations conclude that the most suitable distance between the protrusion of sprays center and cylinder head plane is 6 mm where it has the highest efficiency 88.04% and the lowest emissions can be obtained (PM= 0.17815g/kWh, CO_2=711.99 g/kWh, NO_X=1156.6 ppm).

4.4.5 Mechanical efficiency and amount of exhaust emissions when change the nozzles discharge coefficient

In this stage we enter the values of the nozzles discharge coefficient into DIESEL-RK and these values range from 0.1 to 0.9 and then monitor the results of mechanical efficiency and emissions in each distance.

From the results obtained from DIESEL-RK in Table (4-5), note that when increasing the nozzles discharge coefficient, the mechanical efficiency changes very slightly and this indicates that the change in the nozzles discharge coefficient does not significantly affect the mechanical efficiency Figure (4-17).

cFor the PM emissions the minimum value = 0.14888 g/kWh at the nozzles discharge coefficient = 0.9 Figure (4-18).

For CO_2 emissions, the change in the amount of emissions is very small, all values are between 710.3-717.13 g/kWh and the minimum value = 710.3 g/kWh at the discharge coefficient = 0.9 Figure (4-19).

For NO_X emissions note that the values of NO_X emission is changed with the change of Nozzles discharge coefficient. The Minimum NOx emissions =1184.7 ppm at nozzles discharge coefficient = 0.66

From these observations conclude that the most suitable Nozzles discharge coefficient is .66 where it has the highest efficiency 88.01% and the lowest emissions can be obtained (PM= 00.17372 g/kWh, CO_2=714.04 g/kWh, NO_X=1205.2 ppm).

CHAPTER FIVE

CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion:

All injector variables were examined by DIESEL-RK, the readings were taken initially without any change in the injector variables and emission values for PM, CO_2, NO_X and mechanical efficiency were obtained, after that, and every time one of the variables is changed in a certain range and the other variables are fixed, and each time we record the emission results.

It was found that the best injector variables obtained which give high efficiency and lower emissions for PM, CO_2 and NO_X are: Number of Injector =1, Injector nozzles bore (mm) = .16 mm, Nozzles discharge coefficient obtained as a result of test in atmospheric conditions = 0.66, Number of Nozzle = 8, Distance between spray center and Bowl Axis (mm) = 0 and the distance between the protrusion of sprays center and cylinder head plane = 6 mm,

After entering these values into the Diesel-RK program, the program gives these results; Mechanical efficiency = 88.8%, PM emission=0.13187 ppm, CO_2 emission=711.67 g/kWh and NOX emission=805.45 g/kWh

It is noted that the value of mechanical efficiency increased from 88.01% to 88.8%, the PM emission decreased from 0.17372 ppm to 0.13187 ppm, CO_2 emission decreased from 714.04 g/kWh to 711.67 g/kWh and the value of NOX emission decreased from 1205.2 g/kWh to 805.45 g/kWh.

5-2 Recommendation

1- This project can be applied experimentally and by other program like GT-POWER, MATLAB… etc.

2- The researcher can use the DIESEL - RK program instead of the experimental test to invest time, 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.

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