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Effect of Injector Nozzle Holes on Diesel Engine Four Stroke Performance

(تأثير فتحات الحاقن على أداء محرك ديزل رباعي الأشواط)

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وَقُل رَّبِّ زِدْنِي عِلْمًا

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 freindes, who give me the greatest support in my study and life.

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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.

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Author

Abubaker Faisal Musa Abdelgaliel

Abstract

Diesel engine model simulation designed for four-stroke direct-injection diesel engine requires the use of advanced analysis to carry out of the direct-injection diesel engine model performance effect focuses on fuel nozzle multi holes geometries. The computational model simulation development was use the commercial computational fluid dynamics of GT-POWER 2016 software were specially development for internal combustion engines performance simulation. The research concentrated on the one-dimensional model and focuses on fuel nozzle multi holes geometries variation developed from all of the engine components size measurement of the original selected diesel engine. All of the measurements data input to the window engines component menu for running input data in the model. Results of the diesel engine fuel nozzle multi holes geometries model simulation running is in GT-POST. The model performance shows in engine cylinder and engine crank-train on software window output. The performance analysis effect of the model investigated of fuel in-cylinder engine, indicated specific fuel consumption, indicated torque and indicated power of engine modeled. The simulation result was shown that the seven holes nozzle provided the best for indicted power, indicated torque and indicated specific fuel consumption in any different engine speed in simulation.

المستخلص

محاكاة نموذج محرك ديزل مصمم للحقن المباشر رباعي الأشواط يتطلب استخدام التحليل المتقدم لتنفيذ الحقن المباشر. يركز تأثير أداء طراز محرك الديزل على فتحات الوقود ذات الفتحات المتعددة الهندسية.

تم تطوير محاكاة النموذج الحسابي باستخدام ديناميات الموائع الحسابية التجارية لبرنامج GT-POWER 2016 والتي تم تطويرها بشكل خاص لمحاكاة أداء محرك الاحتراق الداخلي.

يركز البحث على نموذج الأبعاد ويركز على التباين الهندسي متعدد الثقوب في فوهة الوقود والذي تم تطويره من قياس حجم مكونات المحرك لمحرك الديزل الأصلي المحدد. جميع بيانات القياسات أدخلت في نافذة قائمة مكونات المحرك لتشغيل بيانات الإدخال في النموذج. نتيجة تشغيل محرك الديزل لمحرك الديزل ذو فوهة متعددة الثقوب في برنامج .GT-POST يظهر أداء الطراز في أسطوانة المحرك وجسم المحرك على نافذة البرنامج. تأثير تحليل الأداء للنموذج الذي تم فحصه من الوقود في محرك الأسطوانة، وأشار إلى استهلاك الوقود المحدد، وعزم الدوران المحدد، وقوة المحرك المحددة. أظهرت نتائج المحاكاة أن فوهة الثقوب السبعة توفر أفضل قدرة محددة، وعزم الدوران المحدد، وتشير إلى استهلاك محدد للوقود في أي سرعة مختلفة للمحرك في المحاكاة.

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LIST OF ABBREVIATIONS

GT	Gamma Technology
EGR	Exhaust Gas Recirculation
EPA	Environmental Protection Agency
CRS	Common-Rail Systems
UIS	Unit Injector Systems
ISFC	Indicated Specific Fuel Consumption
HAP	Hazardous Air Pollutants
DPF	Diesel Partic Filter
DOC	Diesel Oxidation Catalyst
VGT	Variable Geometry Turbine
PM	Particulate Matter
POM	Polycyclic Organic Matter
DPM	Diesel Particulates
PPM	Parts volume Per Million

CHAPTER ONE

INTRODUCTION

1.1 Background

Design of the diesel fuel injector nozzle is critical to the performance and emissions of modern diesel engines. Some of the important injector nozzle design parameters include details of the injector seat, the injector sac and nozzle hole size and shape. 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. [10]

All nozzles must produce a fuel spray that meets the requirements of the performance and emissions goals of the market for which the engine is produced regardless of details of the fuel system design (i.e., regardless if the fuel system is of the common rail, unit injector, unit pump or pump-line-nozzle type). Additionally, specific requirements for injection nozzles can also depend on the fuel system type:

- Common rail—nozzle operates under more demanding tribological conditions and must be better designed to prevent leakage.
- Unit injector/unit pump—pressure pulsing conditions create more demanding fatigue strength requirements.
- Pump-line-nozzle—hydraulic dead volume must be minimized.

1.2 Problem statement

Investigated the effect of injector nozzle holes number on the performance of diesel engine such as indicated power, indicated torque, fuel consumption and fuel in-engine cylinder.

The investigation is using GT-POWER computational model and explored of four-cylinder diesel engine performance effect based on engine rpm. GT-POWER is the leading engine simulation tool used by engine and vehicle makers and suppliers and is suitable for analysis of a wide range of engine issues.

1.3 **Research Objectives**

This research is focused on the following objectives:

1- Reduce emissions from heavy vehicles.
2- Reduce running cost in diesel engine.
3- The simulation results on engine performance effect of injector fuel nozzle holes number and geometries in indicated power.

1.4 **Scope of Research**

The scopes of this research in the field of injection holes in the performance of diesel engines and the assessment of emissions from diesel combustion.

1.5 **Scientific Study**

The program (GT-power) was used to simulate a diesel engine where the values of the holes in the injector (2-9) were changed and the results obtained.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

The diesel engine, invented in the late 19th century by Dr. Rudolf Diesel, is the most energy efficient power plant among all type of internal combustion engines known today. This high efficiency translates to good fuel economy and low greenhouse gas emissions. Other diesel features that have not been matched by competing energy conversion machines include durability, reliability, and fuel safety. The downsides of diesels include noise, low specific power output, NOx and PM (particulate matter) emissions, and high cost. [11]

2.2 Diesel Engine & Emissions

The diesel engine is the most efficient power plant among all known types of internal combustion engines. Heavy trucks, urban buses, and industrial equipment are powered almost exclusively by diesel engines all over the world and diesel powered passenger cars are increasingly popular. For the foreseeable future, the world's transportation needs will continue to rely on the diesel engine and its gasoline counterpart. However, both engine technologies are evolving at an ever increasing pace to meet two major challenges: lower emissions and increased energy efficiency. [11]

Internal combustion engines are significant contributors to air pollution that can be harmful to human health and the environment. In response, clean diesel technologies with near-zero emissions of NOx and (PM) particulate matter have been developed and introduced in regions with the most stringent emission standards: North America, Europe and Japan. While new clean diesel engines are gradually replacing the population of older diesel engines in these regions, older engines already in service are being retrofitted with clean diesel technologies to hasten emissions reductions. As this trend spreads to other parts of the world, the environmental focus has shifted to climate changing emissions and energy efficiency. The environmental benefit of low greenhouse gas emissions, traditionally associated with the diesel engine, is no longer sufficient. To meet future greenhouse gas and fuel economy regulations, new technologies are being developed—low temperature combustion, waste heat recovery, powertrain electrification, to name a few—that further increase the efficiency not only of the diesel engine powertrain but the entire vehicle as well. Under low-carbon regulatory policies, the scope for potential improvements is no longer limited to engines and vehicles, but also includes life cycle effects of fuel production and vehicle manufacture. [9]

2.2.1 Diesel combustion and emission formation

In a diesel engine the fuel is injected into a highly compressed gas volume. The temperature and pressure of the gas causes the fuel to auto ignites. Some residence time is required for ignition as the thermos chemical reactions involved do not take place instantaneously. Therefore the initial phase of the combustion event is premixed since some fuel has had time to mix with air during the ignition delay. After the premixed phase the combustion continues with fuel being burnt in mixing controlled diffusion flames. [15]

2.2.2 DIESEL ENGINE

Most modern diesel engines use the conventional cylinder and piston arrangement operated with a slider crank mechanism common to other internal combustion engines such as the gasoline engine. Considering this basic mechanism, there is very little difference between the basic structure of diesel and gasoline engines. [12]

Conceptually, diesel engines operate by compressing air to high pressure/temperature and then injecting a small amount of fuel into this hot compressed air. The high temperature causes the small amount of highly atomized injected fuel to evaporate. Mixing with the hot surrounding air in the combustion chamber, the evaporated fuel reaches its auto-ignition temperature and burns to release the energy that is stored in that fuel . [12]

The definition of the diesel engine has evolved over the years. For example, in the early 20th century, a distinction was made between a “true Diesel Engine” and one that shared some aspects of the diesel cycle but did not encompass all aspects considered to be part of the diesel cycle as then envisioned. One early definition of a “true Diesel Engine” is one having the following features:

1. Compression sufficient to produce the temperature requisite for spontaneous combustion of the fuel.
2. Injection of fuel by a blast of compressed air.
3. A maximum cycle pressure (attained during combustion) not greatly exceeding the compression pressure, i.e., absence of pronounced explosive effect. [17]

While the first point of the above characteristics is consistent with the modern diesel engine, the last two are not. Over the course of the 1920s and 1930s, the other two characteristics lost their significance. [6]

Solid-type fuel injection started to emerge around 1910, but it wasn't until the end of the 1920s that it started to rapidly gain acceptance. It is interesting to note that Diesel himself chose air blast injection more out of necessity than choice. Diesel envisioned a solid-injection type of fuel system rather than an air-blast system. [6]

Diesel was quite strict about adhering to constant pressure combustion, item 3. This, however, was only possible in the large relatively slow speed Diesel engines that were common prior to the 1920s. In the smaller high-speed engines that emerged in the 1920s, practical considerations meant that combustion was closer to a constant volume process as in the Otto cycle rather the constant pressure as in the Diesel cycle.[6]

2.2.3 Health and Environmental Effects:

Diesel emissions include a number of biologically active substances. In this group, diesel particulates and the associated organic phase became a major health concern. From the environmental perspective, the advantages of diesels are low "greenhouse gas" and hydrocarbons emissions; their drawback is high NOx emission. [15]

2.2.3.1 Toxic Compounds in Diesel Exhaust:

The principal toxic gas compounds found in diesel exhaust include carbon monoxide (CO), nitric oxide (NO), nitrogen dioxide (NO2), and sulfur dioxide

(SO_2). Biological activity and toxic characteristics of these compounds have been studied for years and are relatively well understood.

In recent years, emission of diesel (PM or DPM) has become one of the major health concerns among all diesel emissions. Medical research on health effects of PM is still in the initial phase of exploring this new area of human knowledge. There are many controversial opinions and many questions have not been answered, awaiting the results of ongoing and future studies. These uncertainties as to the effects of PM and its components are also reflected in the lack of a precise, universal definition of diesel particulates. While practically all of the public health/engine emission regulations define PM as a mix of solids, organics, and sulfates, such definitions as total carbon (i.e., excluding sulfates), or elemental carbon (i.e., excluding sulfates and organics) have been proposed and/or implemented by various occupational health regulations. [3]

Diesel emissions contain numerous other compounds that are present in Diesel emissions in smaller quantities, but still may be posing health threat to humans. The most important substances in this group include Poly nuclear aromatic hydrocarbons (PAH), nitro-PAHs, aldehydes, and selected other hydrocarbons and their derivatives. In their pure state, several of these species have been classified as human carcinogens Even though their concentrations in diesel exhaust are Orders of magnitude lower in comparison to the principal diesel Pollutants, they are still seen as a potential serious health hazard. In the USA, the Environmental Protection Agency (EPA) included "polycyclic Organic matter" (POM) in the list of urban hazardous air pollutants (HAP). The POM, defined as compounds with more than one benzene ring and a boiling point of 100°C and higher, includes practically all of the diesel PAH material. [9]

Most of the heavy organic compounds, such as PAHs, are found in the Particulate phase of diesel emissions. This association, in combination with their very low concentrations, makes it rather difficult to differentiate between the health's effects of the solid DPM fraction and the particular organic species. A common approach is to study the effects of DPM as a whole. Diesel particulates, including both the solid and organic phase, have been identified as a toxic air contaminant in California. [6]

Some health studies take an even more simplistic approach, investigating the effects of "whole diesel exhaust", which includes both gaseous pollutants and particulates. Diesel particulates are frequently used in these studies as an indicator of the diesel exhaust exposure, but no effort is made to analyze which components of the exhaust gases are responsible for particular health effects. From the perspective of diesel emission control, this approach is not practical. Blaming the entire diesel exhaust for adverse health effects is not useful in setting emission control targets or selecting suitable control technologies. After all, the diesel exhaust gas is composed in over 99% of non-toxic materials, including nitrogen, oxygen, water vapor, and carbon dioxide. A consensus in newer publications is that the particulate phase in diesel exhaust, including solid inorganic carbon and the associated organic material, has the greatest effect on health. [6]

2.3 Engine Emission Standards:

1- North America.
2- Europe.
3- Asia.
4- Australia.
5- South America.
6- International.

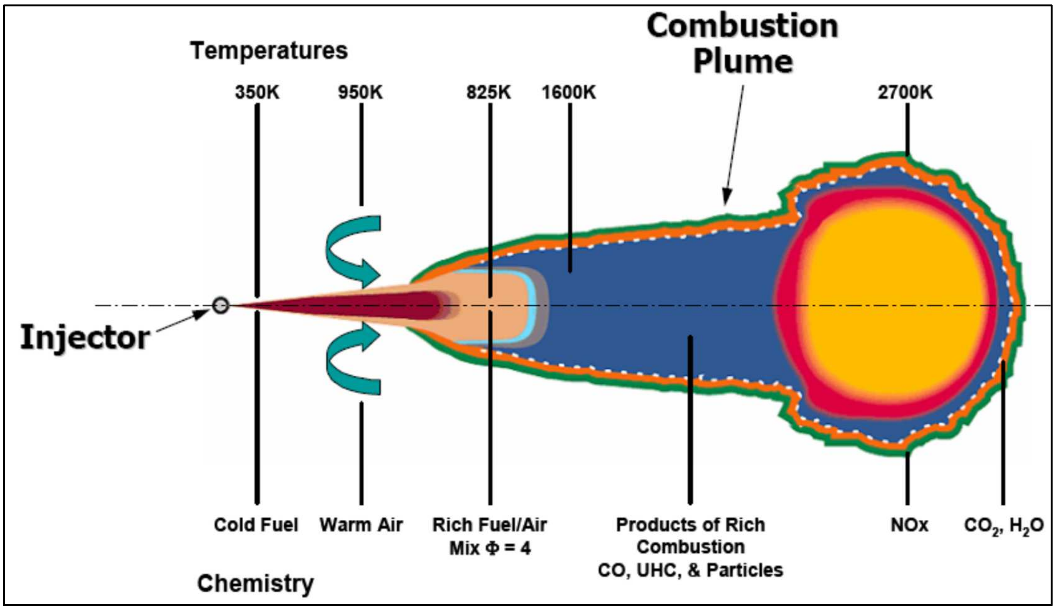

Figure (2.1): Schematic of a diesel flame with temperatures and chemistry [10]

Table(2.1): Reference Diesel Fuel

Property	Unit	Specification		Test
		Min	Max	
Cetane Number	-	52	54	ISO 5165
Density @15°C	kg/m^3	833	837	ISO 3675
Distillation (vol. % recovered)	°C	-	-	ISO 3405
- 50% point		245	-	
- 95% point		345	350	
- final boiling point		-	370	
Flash point	°C	55	-	EN 22719
CFPP	°C	-	-5	EN 116
Viscosity @40°C	mm^2/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

2.4 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. [15]

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. [10]

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. [5]

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. [10]

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.

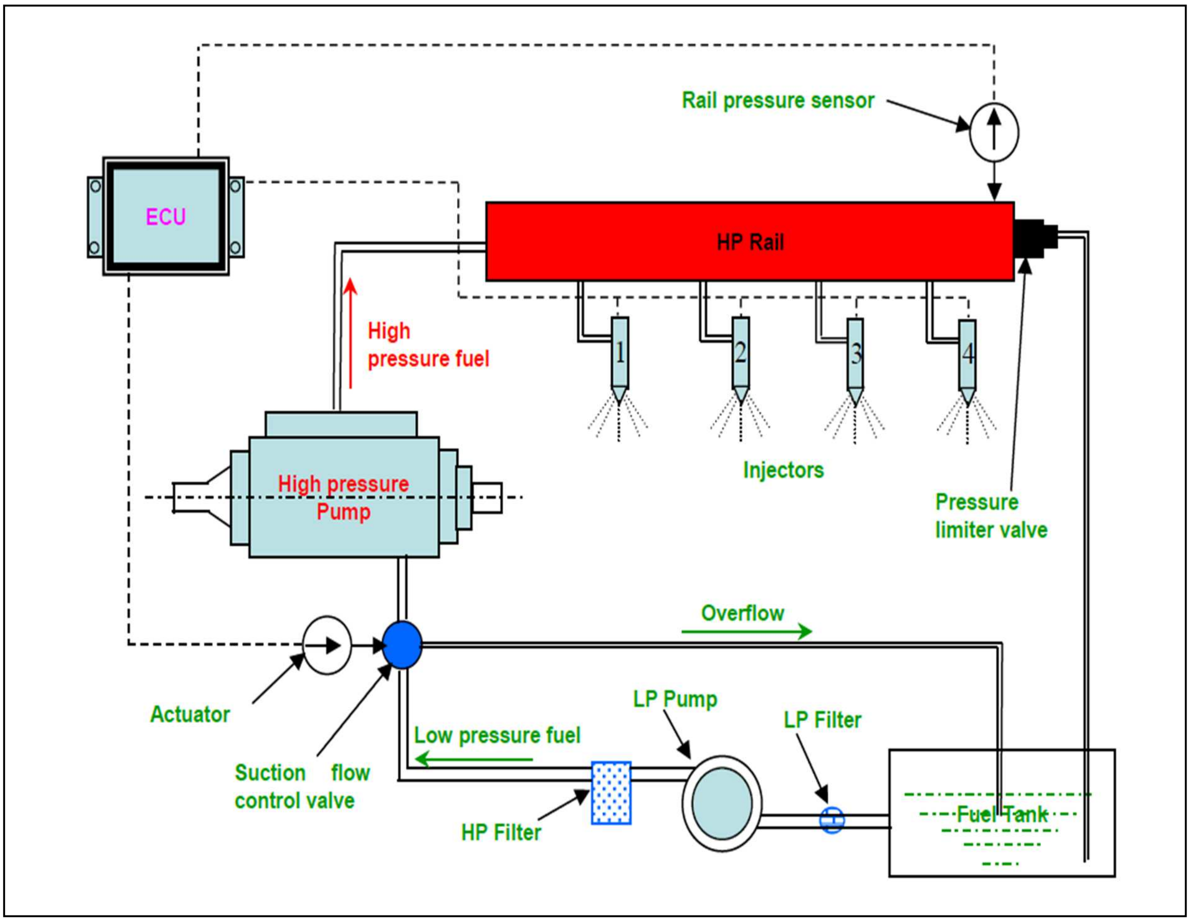

Figure (2.2): Diesel common rail system [10]

CHAPTER THREE

METHODOLOGY

3.1 Nozzles and nozzle holders

In a diesel-engine fuel-injection system, the nozzles connect the injection pump to the engine. Their functions are to:

- Assist in metering the fuel.
- Process the fuel.
- Define the rate-of-discharge curve.
- Seal off the combustion chamber.

In systems with separate injection pumps (inline, distributor and plug-in pumps) the nozzles are integrated in the nozzle holders. On unit injector systems (UIS) and common-rail systems (CRS) show in Figure (3.1), the nozzles are integral parts of the injectors.

Diesel fuel is injected at high pressure. Peak diesel fuel injection pressure can range as high as 2000 bar, a figure which will become even higher in the future. Under these conditions the diesel fuel ceases to behave as a solid, incompressible fluid, and becomes compressible. During the brief delivery period (in the order of 1ms), the injection system is locally "inflated." For a given pressure, the nozzle cross section is one of the factors determining the quantity of fuel injected into the engine's combustion chamber.

The length and diameter of the nozzle spray hole (or orifice), the direction of the fuel jet and (to a certain degree) the shape of the spray hole affect mixture formation, and thus the engine's power output, fuel consumption, and emission levels.

Within certain limits, it is possible to achieve the required rate-of-discharge curve through optimal control of the injector's aperture (defined by the needle's stroke) and by regulating the injector needle's response.

Finally, the injection nozzle must be capable of sealing the fuel-injection system against the hot, highly-compressed combustion gases with temperatures up to approx. 1000 °C. To prevent backflow of the combustion gases when the injection nozzle is open, the pressure in the injection nozzle's pressure chamber must always be higher than the combustion pressure. This requirement becomes particularly relevant toward the end of the injection sequence (when a stark reduction in injection pressure is accompanied by massive increases in combustion pressure), where it can only be ensured by carefully matching the injection pump, the injection nozzle and the nozzle needle for mutually satisfactory operation. [19]

3.1.1 Designs:

Diesel engines with divided or two-section combustion chambers (rechamber and whirl- (or turbulence) chamber engines) require nozzle designs differing from those used in single-section chambers (direct-injection engines).
In rechamber and whirl-chamber engines with divided combustion chambers, throttling-intel nozzles are used which feature a coaxial spray pattern and which are generally equipped with needles which retract to open.
Direct-injection engines with single-section combustion chambers generally require hole-type nozzles. [19]

3.1.2 Throttling-intel nozzles:

One injector (Type DN.SD.) and one injector holder (Type KCA for threaded socket installation) represent the standard combination for use with prechamber and whirl-chamber engines. The standard nozzle holder features M24 x 2 threads and uses a 27 mm wrench fitting. DN O SD. nozzles with a needle diameter of 6 mm and a spray aperture angle of 0° are usually used; less common are nozzles with a defined spray dispersal angle (for example 12° in the DN12SD..).
Smaller holders are used when only limited space is available (e.g., KCE holders). [19]

3.1.3 Hole-type nozzles:

A wide range of nozzle-and-holder assemblies (DHK) is available for hole-type nozzles. In contrast to throttling Intel nozzles, hole-type nozzles must generally be installed in a specific position to ensure correct alignment between the orifices (which are at different angles in the nozzle body) and the engine combustion chamber. For this reason, lugs or hollow screws are usually employed to attach the nozzle-and-holder assemblies to the cylinder head while a locating device ensures the proper orientation. [19]

3.1.4 Hole Thickness:

The length or thickness of the orifice hole for injectors, the ratio of the length to the orifice diameter (L/D) generally ranges from:

• 5.0 - 7.0 for injectors used in automotive and small diesel engines.

• 3.5 - 5.0 for injectors used on large diesel engines. [19]

3.2 Standard nozzle holders:

The basic injector nozzle-and-holder assembly comprises the nozzle and the holder show in Figure (3.3). The injector nozzle consists of two sections: the body and the needle. The nozzle needle moves freely within the body's guide bore while at the same time providing a positive seal against high injection pressures.

At the bottom of the needle is a conical seal, which the nozzle spring presses against the bodies correspondingly shaped sealing surface when the nozzle is closed. These two opposed conical surfaces exhibit a slight mutual variation in aperture angle, providing linear contact with high dynamic compression and a positive seal.

The diameter of the needle guide is greater than that of the seat. The hydraulic pressure from the injection pump acts against the differential surface between the needle diameter and the surface covered by the seat. The injection nozzle opens when the product of sealing surface and pressure exceeds the force of the nozzle spring in the holder. Because this process produces a sudden increase in pressurized surface area – with the seat suddenly joining the needle – a sufficiently high delivery rate will result in the injection nozzle snapping open very rapidly. It does not close again until the system has dropped from its opening pressure to below the (lower) closing pressure. This hysteresis effect is of particular significance when designing hydraulic stability into fuel-injection systems.

The opening pressure of a nozzle-and-holder combination (approx. 110...140 bar for a throttling Intel nozzle and 150...250 bar for a hole-type-nozzle) is adjusted by placing shims under the compression spring.
Closing pressures are then defined by the injection nozzle's geometry (ration of needle diameter to seat diameter). [19]

3.2.1 Nozzle Hole Diameter:

Diameter of an individual nozzle hole.

3.2.2 Number of Holes per Nozzle:

Number of identical nozzle holes in the injector.

3.2.3 Nozzle Discharge Coefficient:

Nozzle hole discharge coefficient. This attribute is only used when the Pressure or Mass Array Type is set to mass. Typical values are between 0.6 and 0.85.

Table (3.1): Specification of the selected Nozzle

Nozzle Parameters	Object value	Unit
Nozzle Hole Diameter	0.18	mm
Number of Holes per Nozzle	2-9	
Nozzle Discharge Coefficient	0.86	

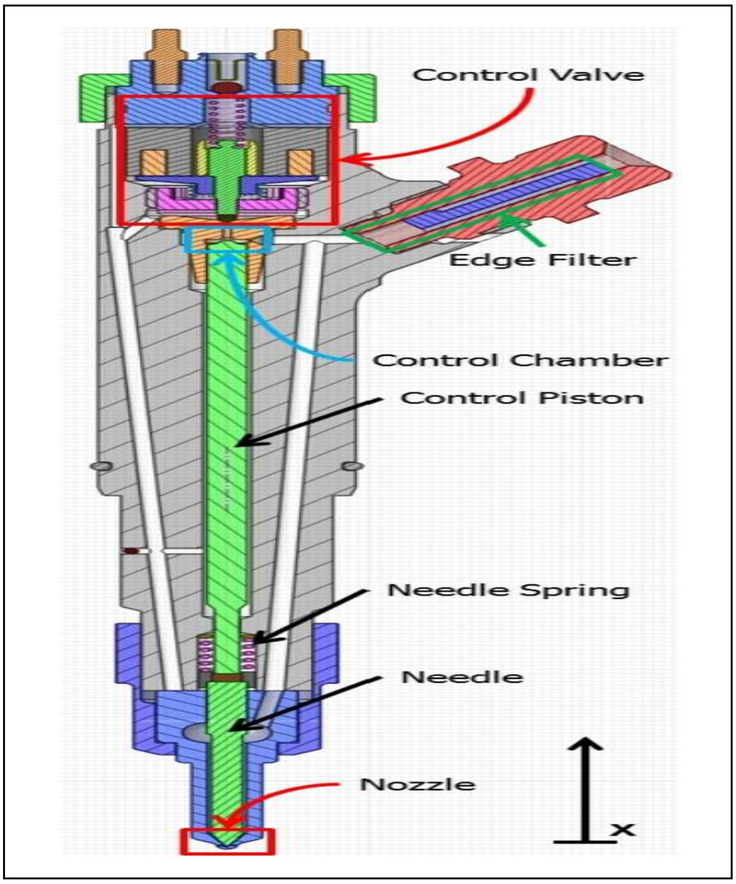

Figure (3.1): Common Rail Fuel Injector [19]

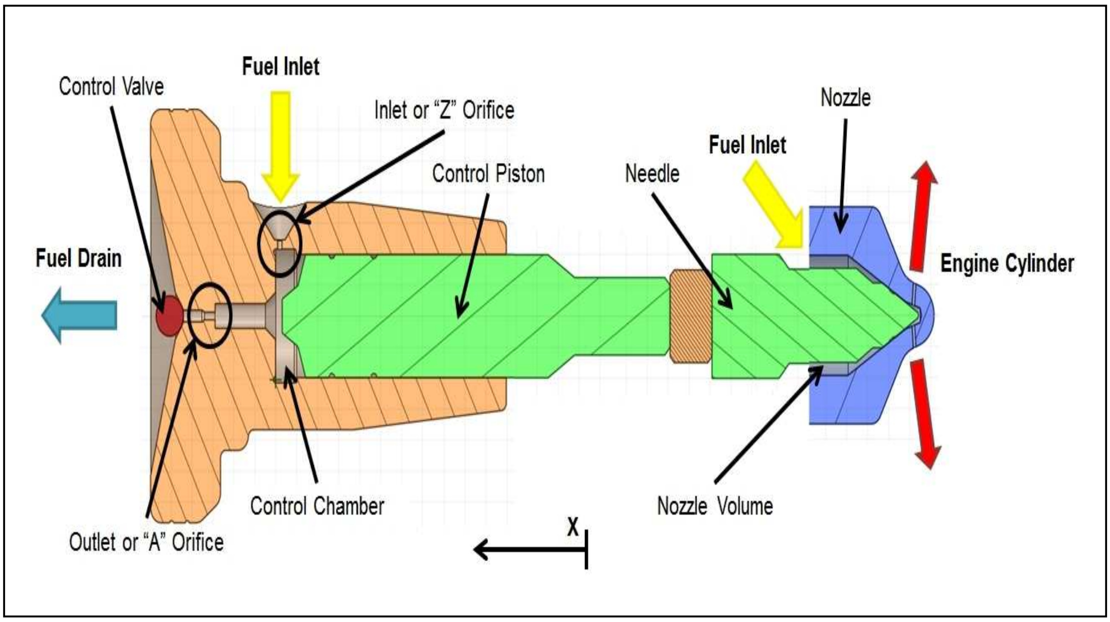

Figure (3.2): Injector model [19]

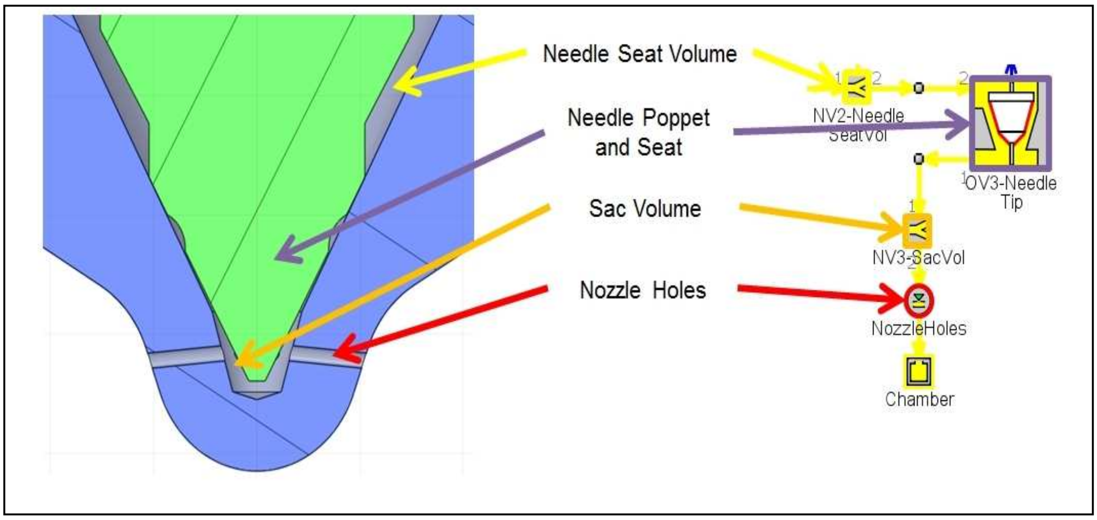

Figure (3.3): Fuel nozzle holes detail [19]

3.3 engine operate:

An engine can operate for its normal function in working time if the sufficient amount of fuel is supplied. Diesel engine obtains the power from the combustion of the mixture of fuel and air in the combustion chamber. But some amount of power obtained from the fuel burning are lost into the cooling system, exhaust system, and moving parts working against the friction force occurred on the contact surfaces. The calorific value of the diesel oil, CV is 44.2MJ/kg. [19]

3.4 Model description:

The example 'Diesel _VGT_EGR' demonstrates several modeling concepts important for automotive diesel engines, including variable geometry turbine, boost control, intercooler, EGR circuit with cooler and EGR rate control, injection limiting for smoke control, and exhaust after treatment device modeling (geometry only, no chemistry). [19]

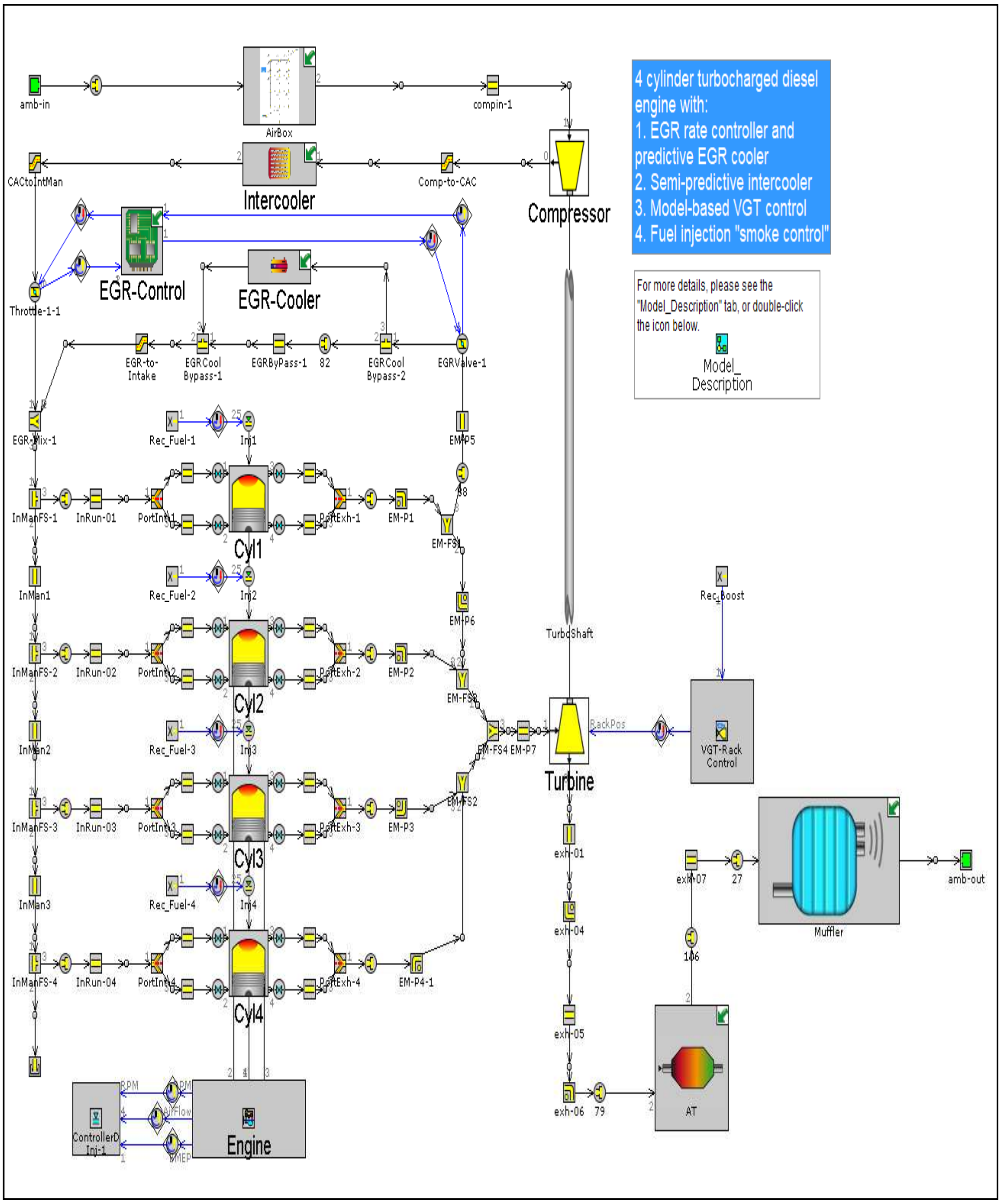

Figure (3.4): Model of VGT _Diesel Engine by using GT-Power [19]

3.4.1 The Engine:

The engine is a 4-cylinder, four-stroke, direct injection, 2L diesel engine. The engine runs in "speed" mode, where the engine speed is entered by the user and the engine torque is calculated. Combustion is modeled by the DI Pulse method, a predictive combustion model using the detailed injection profiles. In-cylinder heat transfer is modeled using the Woschni model. A FE model is used to predict the cylinder, head and piston temperatures. This simulation will run until it reaches steady-state convergence for six different engine speeds at full load, three part load operating points, and an additional idle point (total of 4 cases). [19]

Table (3.2): Specification of the selected diesel engine

Engine Parameters	Object value	Unit
Bore	86	mm
Stroke	86	mm
Displacement	2000	cc
Number of cylinder	4	
Connecting rod length	129	mm
Piston pin offset	2-9	mm
Compression Ratio	17	
TDC Clearance Height	3	mm
Firing order	1-3-4-2	
Intake valve Clearance	0.35	mm
Exhaust valve Clearance	0.4	mm
Intake Valve Diameter	86	mm
Exhaust Valve Diameter	86	mm

3.4.2 The Intake System:

The intake system consists of the air piping upstream of the compressor inlet, and contains a simple model of an air box. The air box is modeled as a group of interconnected flow splits that represent the total volume of the air box. At the inlet and outlet of the air box, "bellmouth" orifice connections are used to model smooth transitions. [19]

3.4.3 The Intercooler:

The intercooler is found downstream of the compressor outlet. This intercooler is an air-to-air intercooler that is modeled using a "semi-predictive" method based on heat exchanger effectiveness. Note that if heat exchanger performance data is available, it is generally preferred to model the heat exchanger using 'HxMaster' and 'HxSlave' parts (which will use the available data to create non-dimensional Nusselt correlations defining heat exchanger performance). This approach will generally extrapolate better than an assumed effectiveness to conditions beyond those present in the heat exchanger data. However, if heat exchanger data is not available (or if the simulation will be run at conditions where HX effectiveness is well known), the semi-predictive effectiveness method is appropriate.

The intercooler is modeled using a pipe part. The Number of Identical Pipes attribute is used to model a bundle of 25 identical pipes of 7.0 mm diameter each (giving a large surface area for heat transfer). The pressure drop across the intercooler is calibrated using the Friction Multiplier in the in the pipe. Flow splits are required to model the inlet and outlet of the intercooler since pipes cannot connect to pipes with the Number of Identical Pipes attribute greater than one. The desired intercooler outlet temperature is calculated by a simple control system and is imposed by actuating the wall temperature of the intercooler pipe. The Heat Transfer Multiplier in the pipe is increased to a large enough value so that the outlet temperature matches the wall temperature. The desired outlet temperature is calculated in a 'Intercooler Eff' part from the effectiveness table of the intercooler. [19]

3.4.4 The Intake Manifold:

The intake manifold is modeled as aluminum using heat conduction objects to calculate the wall temperatures. This is important in turbocharged models since wall temperatures may change substantially at different operating points, thus influencing the cylinder inlet temperature. The orifices that connect. The initial conditions are different from ambient conditions. The initial temperatures and pressures are specified at the expected boost pressure in order to decrease the convergence time. [19]

3.4.5 The Fuel Injection System:

The fuel is injected directly into the cylinders. Both the injection timing and the injection profile are looked up using an 'RLT Dependence' as a function of engine speed and injection rate. There is a simple control system that determines the total injected fuel mass (mg/stroke). This control system begins with a 'Signal Generator' part that imposes an accelerator pedal position input. The "requested fuel mass" is then looked up as a function of engine speed and pedal position. In some cases, the requested fuel mass may produce excess smoke if the boost pressure is not high enough to provide enough air. Therefore, the maximum allowed fuel mass is looked up as a function of the engine speed and the boost pressure. A 'Min Max' part is used to take the smaller of the requested fuel mass and the smoke limited fuel mass. This value is then imposed on the injectors using an actuator. [19]

3.4.6 The Exhaust System:

The exhaust manifold is modeled as cast iron using heat conduction objects to calculate the wall temperatures. This is very important in the exhaust manifold since wall temperatures change substantially between full load and idle condition, thus influencing the turbine inlet temperatures and turbine power. The orifices that connect the exhaust ports to the runners do not allow heat to be conducted between the pipe walls of the adjacent parts. This is necessary because the wall temperature is calculated in the exhaust pipes, while it has been imposed in the port (the imposed value is artificially high to account for heat transfer from the valve). The initial conditions are different from ambient conditions. The initial temperatures and pressures are specified at the expected backpressure in order to decrease the convergence time. The turbine part references a map file from which map data is read. [19]

3.4.7 The EGR System:

The model includes an exhaust gas recirculation (EGR) system to transport exhaust gases from the exhaust manifold back to the intake manifold. The EGR flow is driven by a positive pressure difference between the exhaust manifold and intake manifold and is controlled by an orifice connection (EGR-valve). The EGR-valve diameter is imposed by an actuator, and is determined automatically by a 'Controller EGR Valve' part to achieve a target EGR fraction. The desired EGR fraction is mapped as a function of engine speed and the requested BMEP. EGR rates in the model are as high as 40% at some operating points. Because of the high EGR rates, it is desirable to cool the EGR gases using a heat exchanger (EGR-cooler). When the pressure drop across the EGR system is not sufficient to achieve the target EGR rate, a throttle in the intake system, controlled by a PID controller, increases the pressure difference. This controller is only active, if the EGR-valve is fully open.

The EGR cooler is modeled using 'HxMaster' and HxSlave' parts. Coolant side boundary conditions are imposed. The heat exchanger model inputs include geometry along with measured heat transfer and pressure drop performance data. During model pre-processing, the measured data is used to generate non-dimensional correlations for heat transfer that allow the model to accurately predict heat transfer rates at any operating condition. The pressure drop data is use to automatically calibrate physical characteristics (friction multiplier, orifice Cd) of the heat exchanger so that the data is well matched. [19]

3.4.8 The Turbocharger and Boost Controller:

The turbocharger in this model contains a fixed geometry compressor and a variable geometry turbine (VGT). For the VGT, there are 5 different sets of turbine map data entered at each of 5 VGT rack positions. The VGT rack position is determined by a control system to achieve a target boost pressure upstream of the intake manifold. The boost pressure is sensed and is then filtered using a 'Moving Average' part to produce an average pressure signal. The model based controller compares the sensed boost pressure to the target boost pressure and then adjusts the VGT rack position accordingly. [19]

3.4.9 The Exhaust System:

A full vehicle exhaust system is modeled downstream of the turbine outlet. The geometry of a Diesel Oxidation Catalyst (DOC) is modeled using a 'Catalyst Brick' part. In this example only the pressure drops across the DOC is of interest, so no chemical reactions are modeled in the DOC. The inlet and outlet of the DOC must be modeled using flow splits because the DOC is effectively a bundle of very small pipes.

Just downstream of the DOC is a Diesel Particulate Filter (DPF) using a 'Diesel Particle Filter' part. In this example, only the geometry attributes are specified so that proper pressure drop across the filter is modeled, but there is no soot accumulation or regeneration modeled. Similar to the DOC, the inlet and outlet of the DPF must be modeled using flow splits.

Downstream of the DOC is a muffler. This muffler was modeled using GEM3D, which enables complex muffler geometry to be entered in a CAD tool and automatically discretized into a GT-POWER model. The muffler in this example is a tri-flow type muffler with two baffles. The center chamber between the two baffles is filled with wool, and various internal pipes and baffles include perforation. The .gem file is included in the directory and can be opened in GEM3D to see the muffler. The output of the GEM discretization is stored within an internal subassembly to keep the main project map clean. [19]

CHAPTER FOUR

RESULT AND DISCUSSION

4.1preface

Whenever a simulation is run, GT-SUITE produces several output files that contain simulation results in various formats. Most of the output is available in the post-processing application GT-POST.
GT-POST is powerful tool that can be used to view animation and order analysis output [2]. After the simulation was finished, report tables that summarize the simulations can be produced. These reports contain important information about the simulation and simulation result in a tabular form. The computational simulation of the engine model result is informed the engine performance. The running simulation result in this research is focuses on the engine performance data based on variation of fuel nozzle material hole diameter size, diameter number and the different engine speed (rpm). The diesel engine model was running on any different engine speed in rpm, there are 1000, 1500, 2000 and 3000.
The variations of fuel nozzle material holes number are multi holes and several number holes, the simulation model there are start from the fuel nozzle 2 – 9 holes.

4.2 Engine Performance Review

The simulation results are shown in every case, such as case 1 is on 3000 rpm, case 2 is on 2000 rpm, case 3 is on 1500 rpm, and case 4 is on 1000 rpm.
Numerous studies have suggested that decreasing the injector nozzle orifice diameter is an effective method on increasing fuel air mixing during injection.
Smaller nozzle holes have found to be the most efficient at fuel/air mixing primarily because the fuel rich core of the jet is smaller.
In addition, decreasing the nozzle hole orifice diameter would reduce the length of the Potential core region. Unfortunately, decreasing nozzle holes size causes a reduction in the turbulent energy generated by the jet. [3]

4.3 Effect of Injector Nozzle Holes on Engine Performance:

The simulation result on engine performance effect of injector fuel nozzle holes number and geometries in indicated power, indicated torque of engine are shown in Figure below. The injector fuel nozzle holes orifice diameter and injector nozzle holes numbers effect on indicated power, indicated torque of direct-injection diesel engine was shown from the simulation model running output. An aerodynamic interaction and turbulence seem to have competing effects on spray breakup as the fuel nozzle holes orifice diameter decreases. The fuel drop size decreases if the fuel nozzle holes orifice diameter is decreases with a decreasing quantitative effect for a given set of jet conditions. [3]

Fuel-air mixing increases as the fuel nozzle holes orifice diameter fuel nozzle holes decreases. Also soot incandescence is observed to decrease as the amount of fuel-air premixing upstream of the lift-off length increases. This can be a significant advantage for small orifice nozzles hole. However, multiple holes orifices diameter required to meet the desired mass flow rate as orifice diameter decreases. In this case, the orifices diameter need to place with appropriate spacing and directions in order to avoid interference among adjacent sprays. The empirical correlations generally predict smaller drop size, slower penetrating speed and smaller spray cone angles as the orifice diameter decreases, however the predicted values were different for different relation. All of the nozzles have examined and the results are shown that the five holes nozzle provided the best results for indicted torque, indicated power and ISFC in any different engine speed in simulation. [2]

4.4 Results obtained

4.4.1 Nozzle 2 hole

The figure(4.1) below indicates the results obtained when using (2) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.941 kw) At engine speed (3000 rpm) and min power (15.53 kw) at engine speed (1000 rpm).

In The figure(4.2) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.6 N-M) At engine speed (2300 rpm) and min torque (148.3 N-m) at engine speed (1000 rpm).

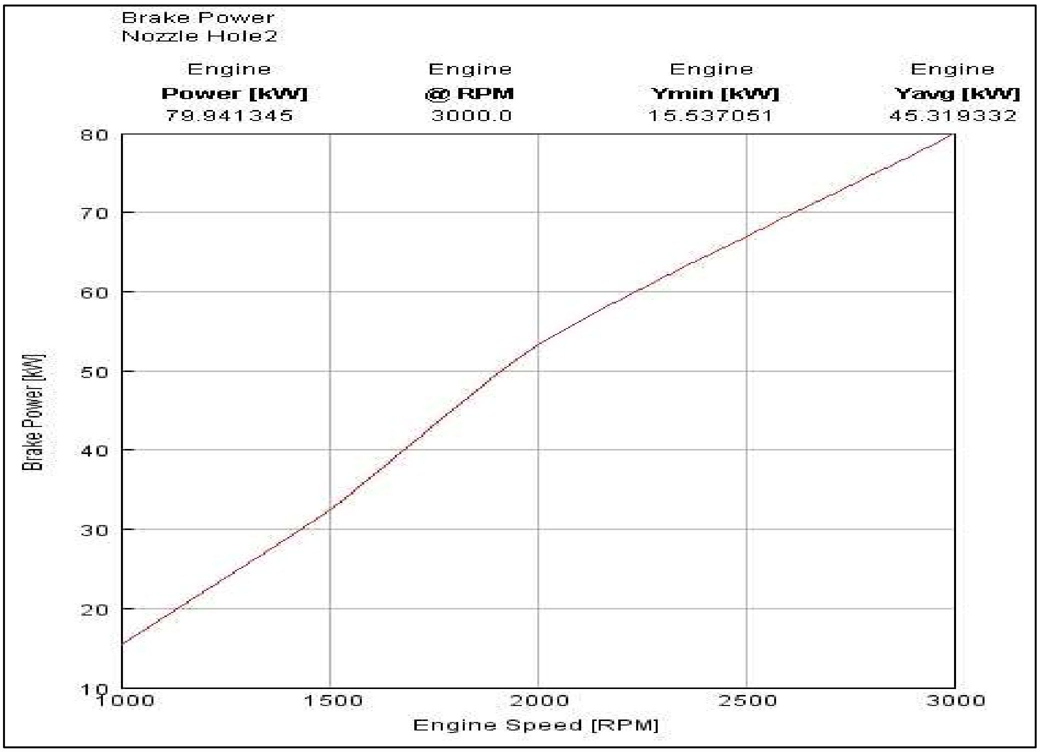

Figure (4.1): Nozzle 2 hole indicated Brake Power of engine

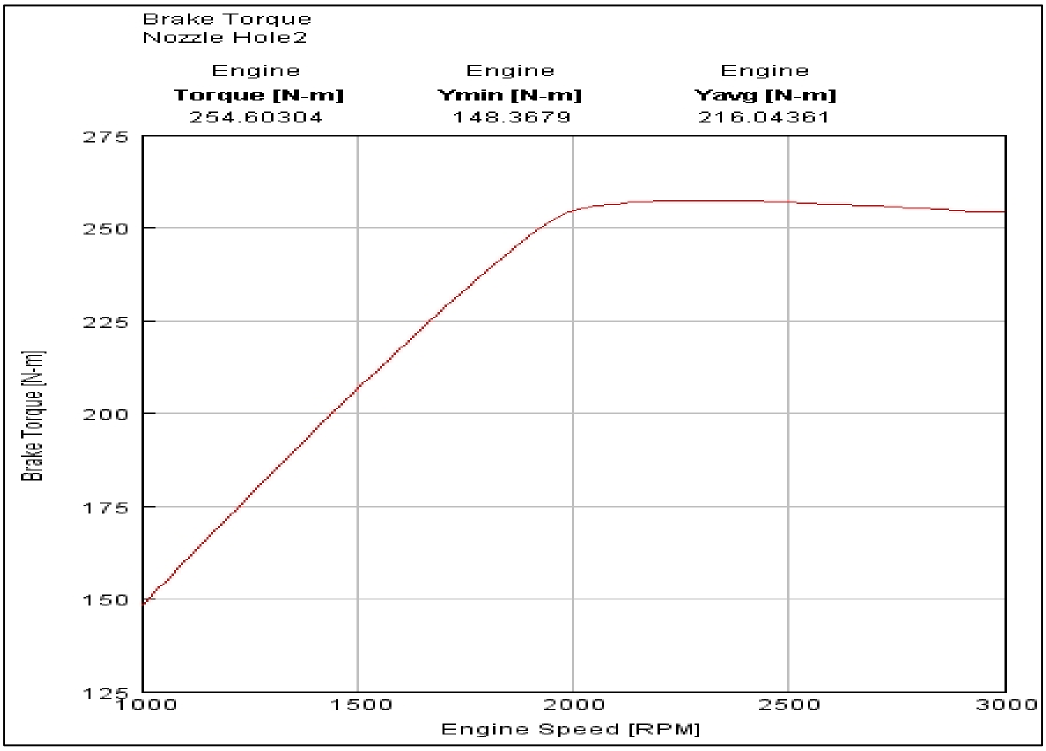

Figure (4.2): Nozzle 2 hole indicated Brake Torque of engine

4.4.2 Nozzle 3 hole

The figure(4.3) below indicates the results obtained when using (3) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.941 kw) At engine speed (3000 rpm) and min power (15.53 kw) at engine speed (1000 rpm).

In The figure(4.4) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.6 N-M) At engine speed (2300 rpm) and min torque (148.3 N-m) at engine speed (1000 rpm).

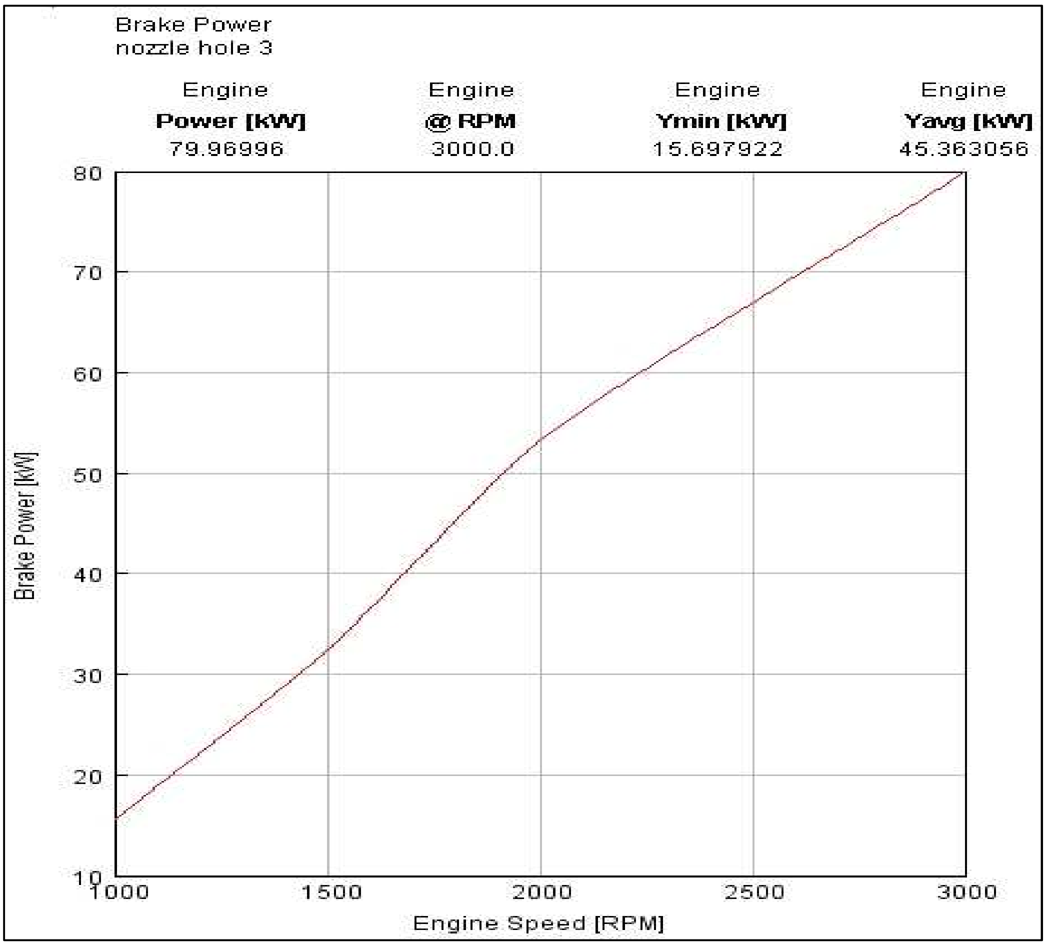

Figure (4.3): Nozzle 3 hole indicated Brake Power of engine

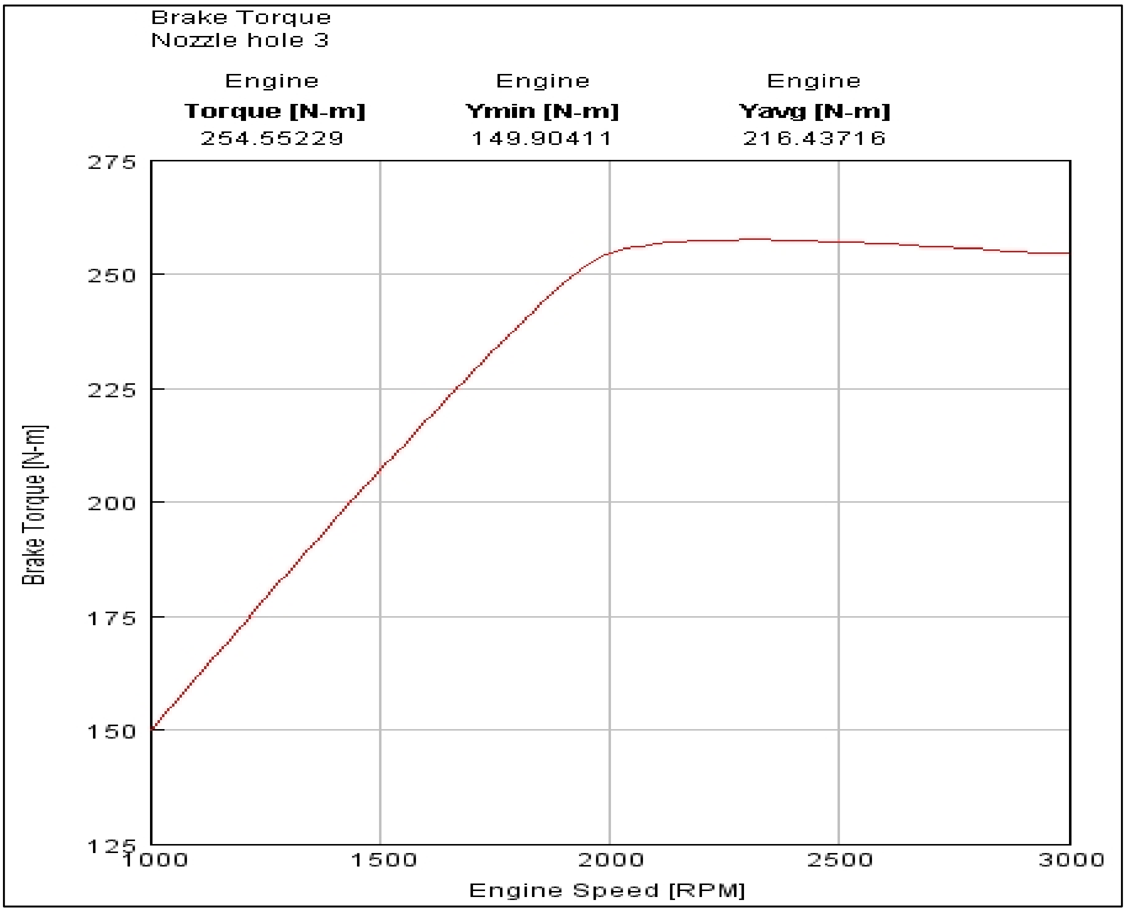

Figure (4.4): Nozzle 3 hole indicated Brake Torque of engine

4.4.3 Nozzle 4 hole

The figure(4.5) below indicates the results obtained when using (4) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.941 kw) At engine speed (3000 rpm) and min power (15.53 kw) at engine speed (1000 rpm).

In The figure(4.6) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.6 N-M) At engine speed (2300 rpm) and min torque (148.3 N-m) at engine speed (1000 rpm).

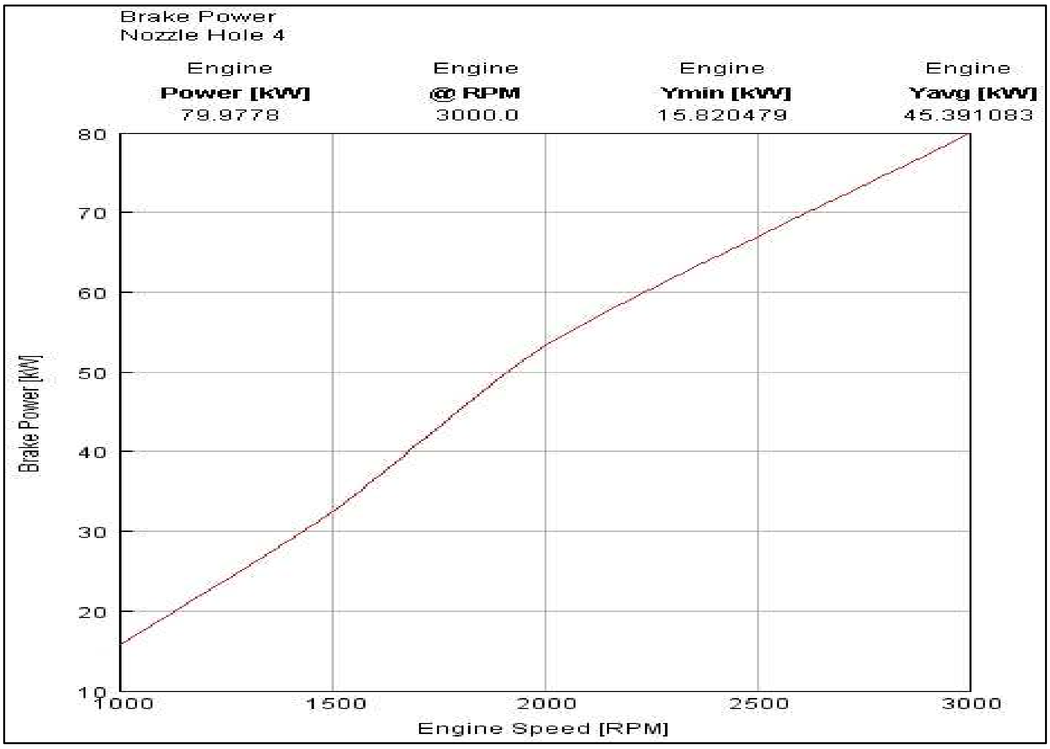

Figure (4.5): Nozzle 4 hole indicated Brake Power of engine

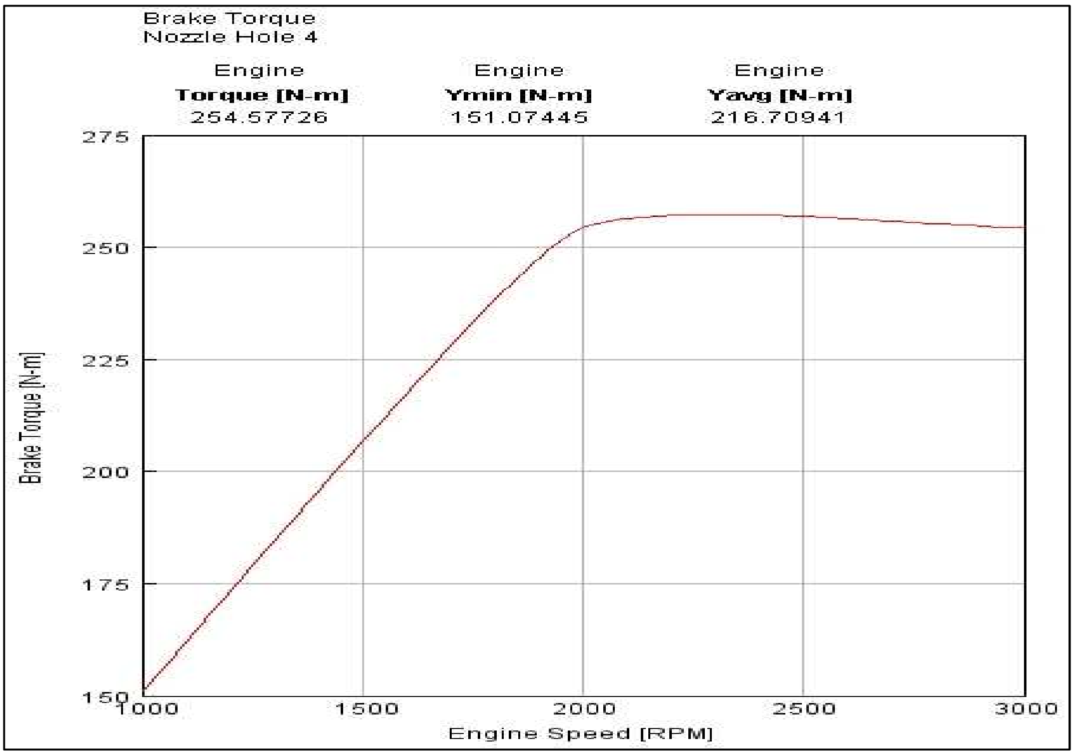

Figure (4.6): Nozzle 4 hole indicated Brake Torque of engine

4.4.4 Nozzle 5 hole

The figure(4.7) below indicates the results obtained when using (5) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.941 kw) At engine speed (3000 rpm) and min power (15.53 kw) at engine speed (1000 rpm).

In The figure(4.8) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.6 N-M) At engine speed (2300 rpm) and min torque (148.3 N-m) at engine speed (1000 rpm).

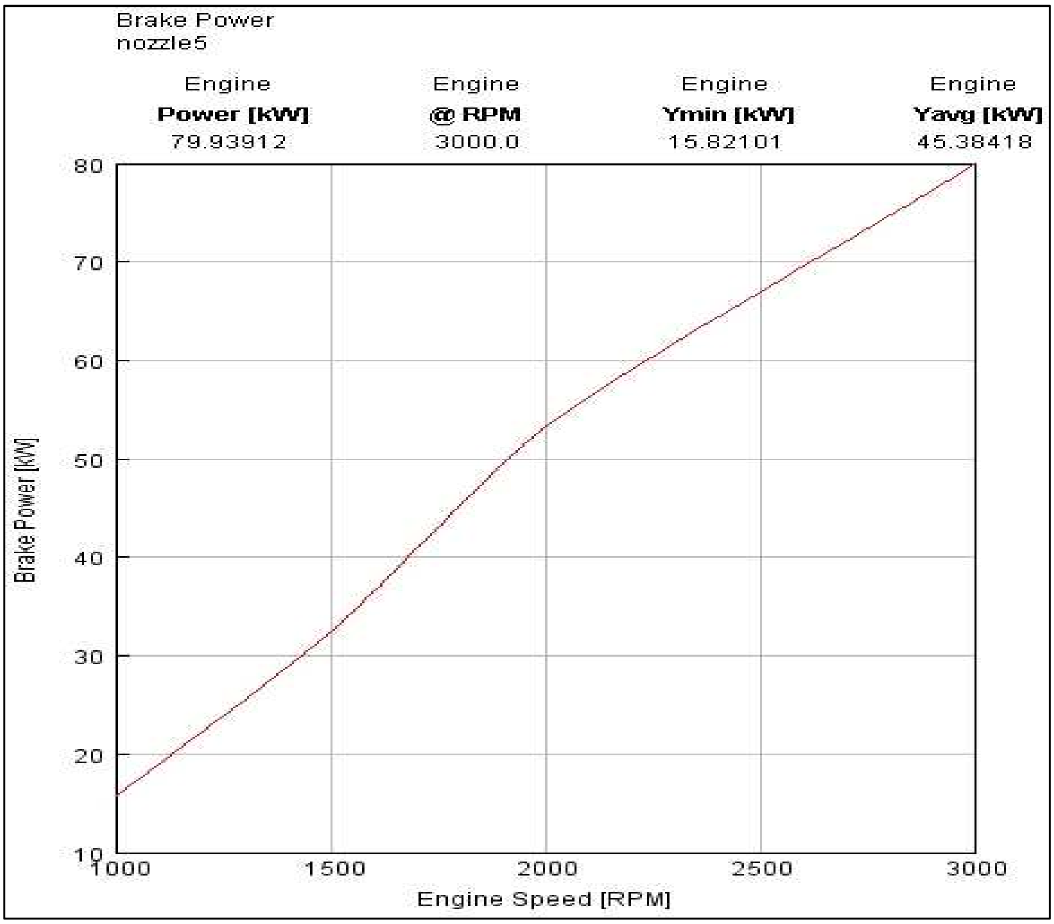

Figure (4.7): Nozzle 5 hole indicated Brake Power of engine

Figure (4.8): Nozzle 5 hole indicated Brake Torque of engine

4.4.5 Nozzle 6 hole

The figure(4.9) below indicates the results obtained when using (6) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.941 kw) At engine speed (3000 rpm) and min power (15.53 kw) at engine speed (1000 rpm).

In The figure(4.10) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.6 N-M) At engine speed (2300 rpm) and min torque (148.3 N-m) at engine speed (1000 rpm).

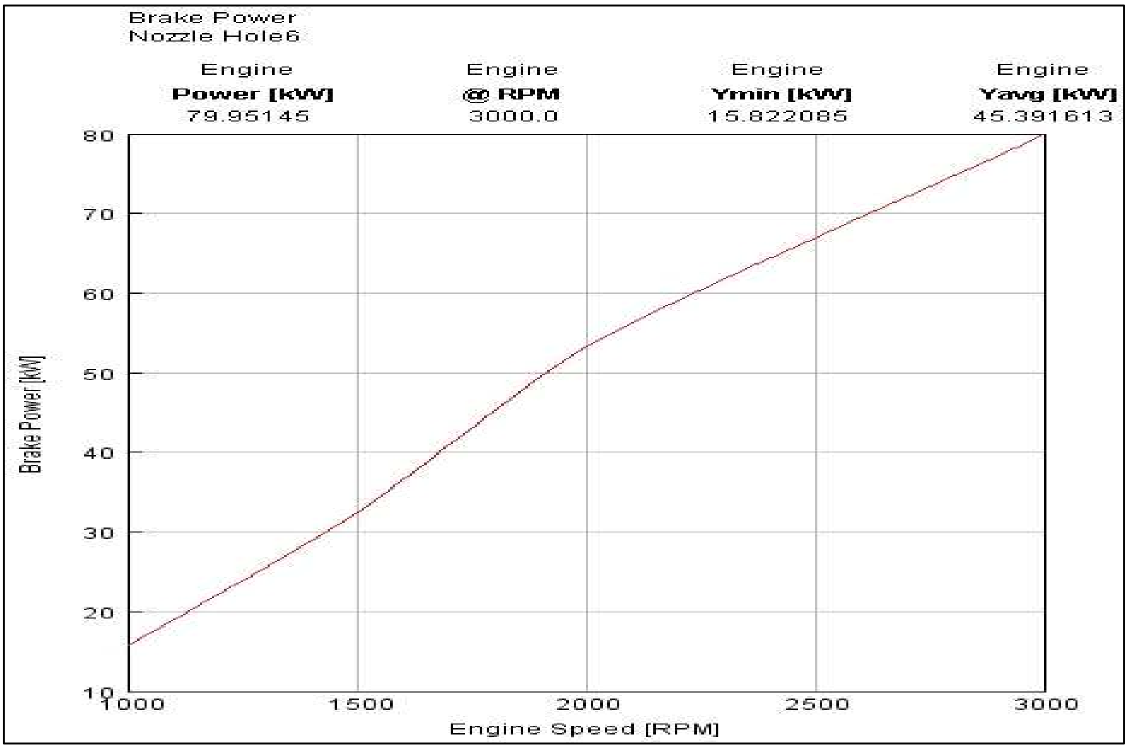

Figure (4.9): Nozzle 6 hole indicated Brake Power of engine

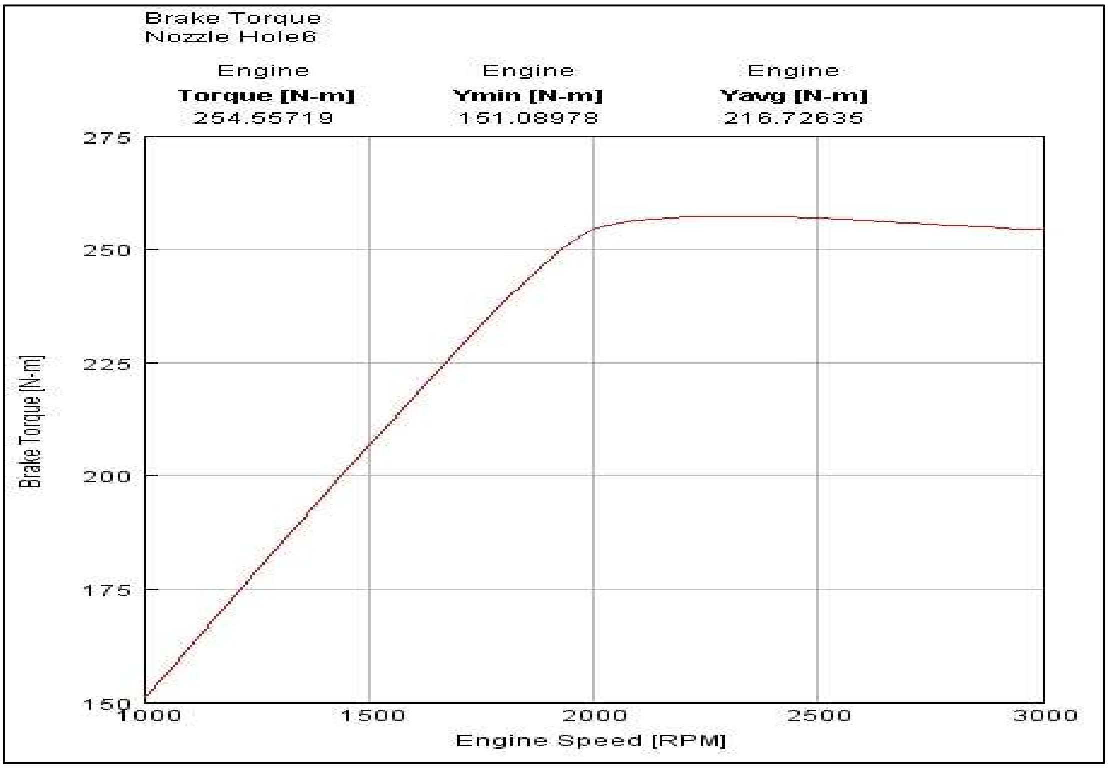

Figure (4.10): Nozzle 6 hole indicated Brake Torque of engine

4.4.6 Nozzle 7 hole

The figure(4.11) below indicates the results obtained when using (7) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.941 kw) At engine speed (3000 rpm) and min power (15.53 kw) at engine speed (1000 rpm).

In The figure(4.12) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.6 N-M) At engine speed (2300 rpm) and min torque (148.3 N-m) at engine speed (1000 rpm).

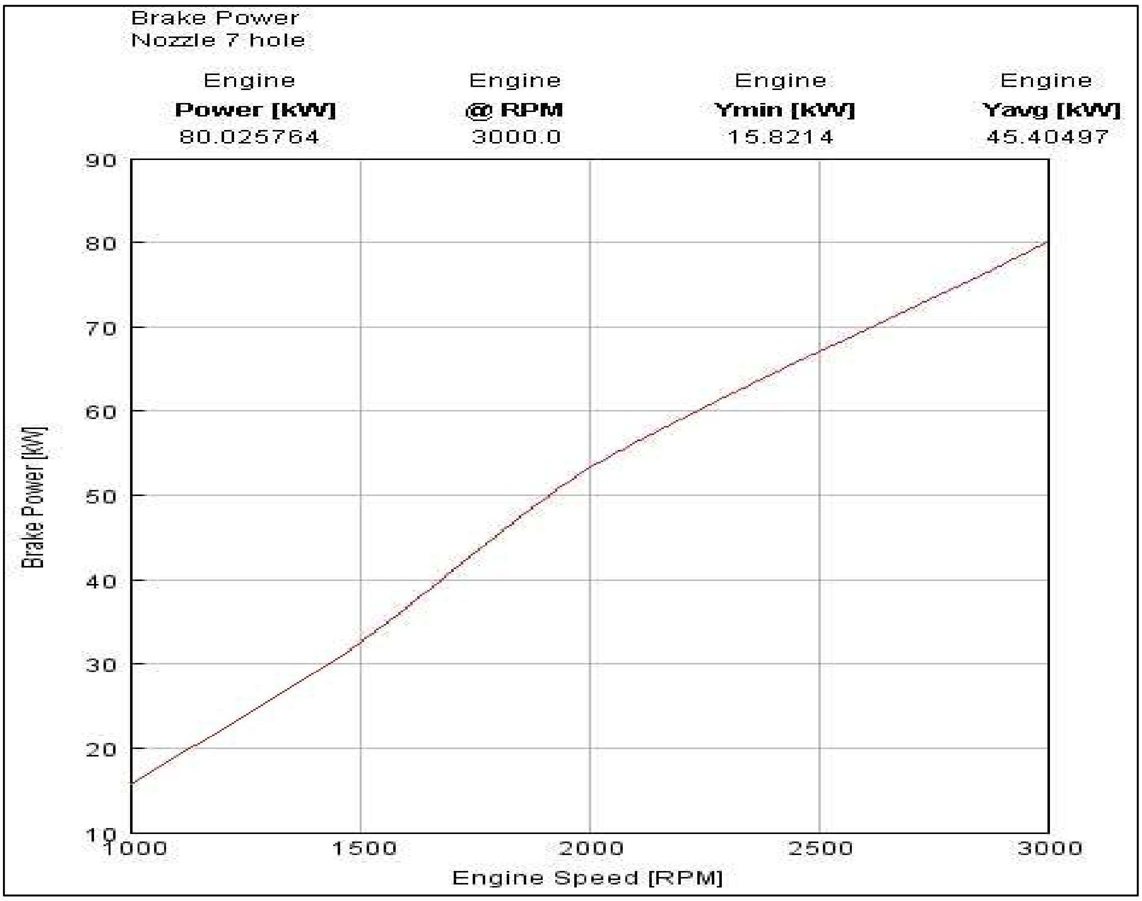

Figure (4.11): Nozzle 7 hole indicated Brake Power of engine

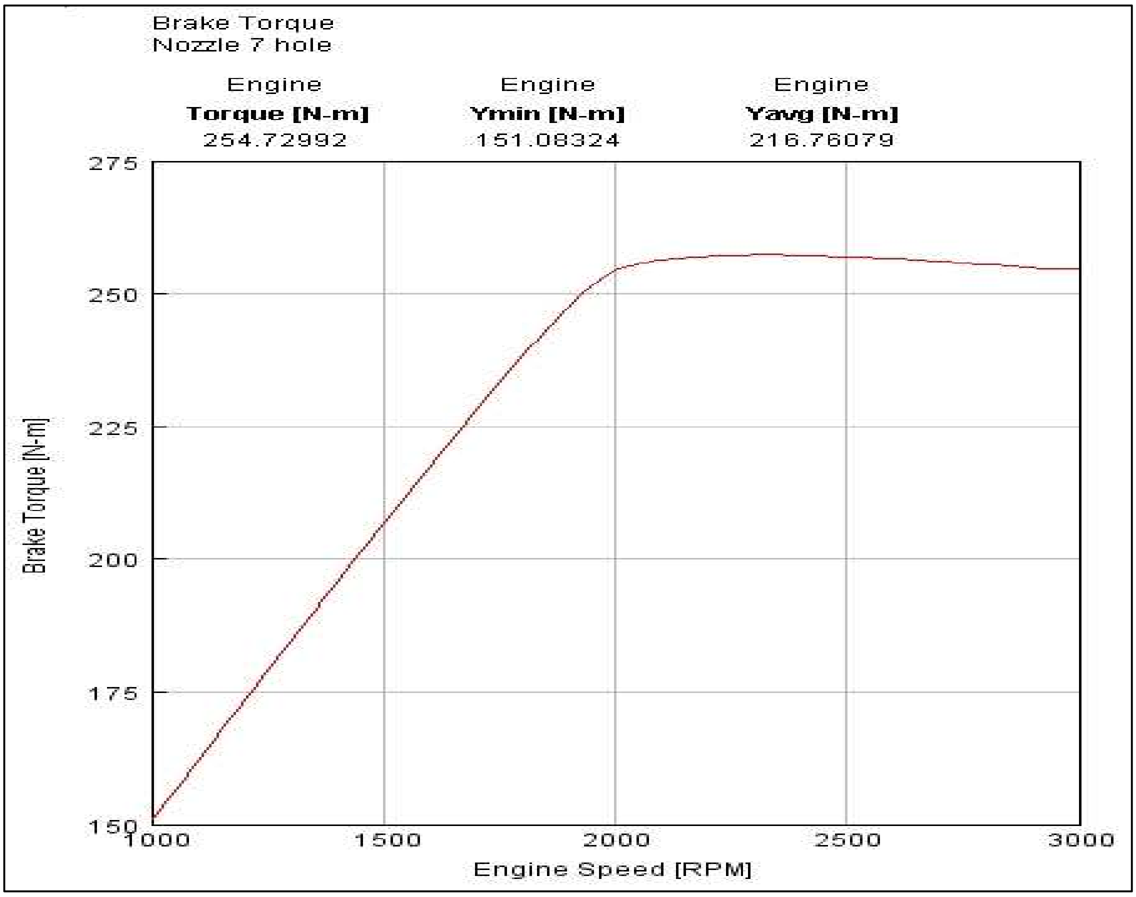

Figure (4.12): Nozzle 7 hole indicated Brake Torque of engine

4.4.7 Nozzle 8 hole

The figure(4.13) below indicates the results obtained when using (8) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.95 kw) At engine speed (3000 rpm) and min power (15.82 kw) at engine speed (1000 rpm).

In The figure(4.14) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.49 N-M) At engine speed (2300 rpm) and min torque (151 N-m) at engine speed (1000 rpm).

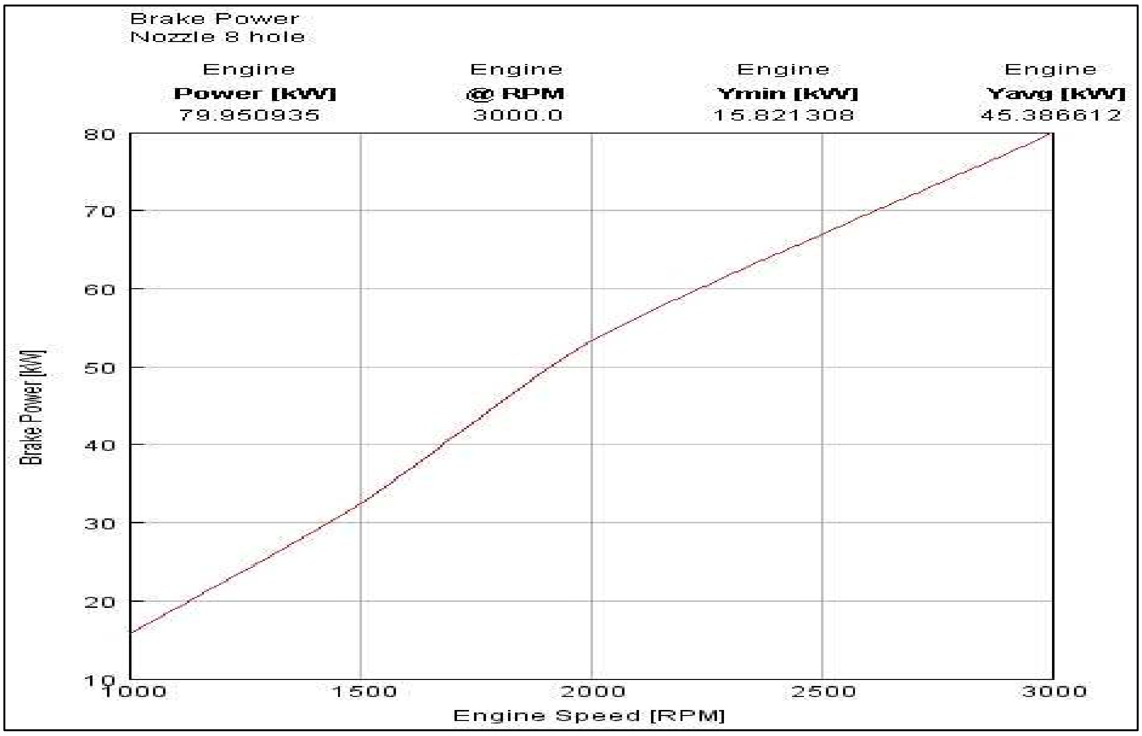

Figure (4.13): Nozzle 8 hole indicated Brake Power of engine

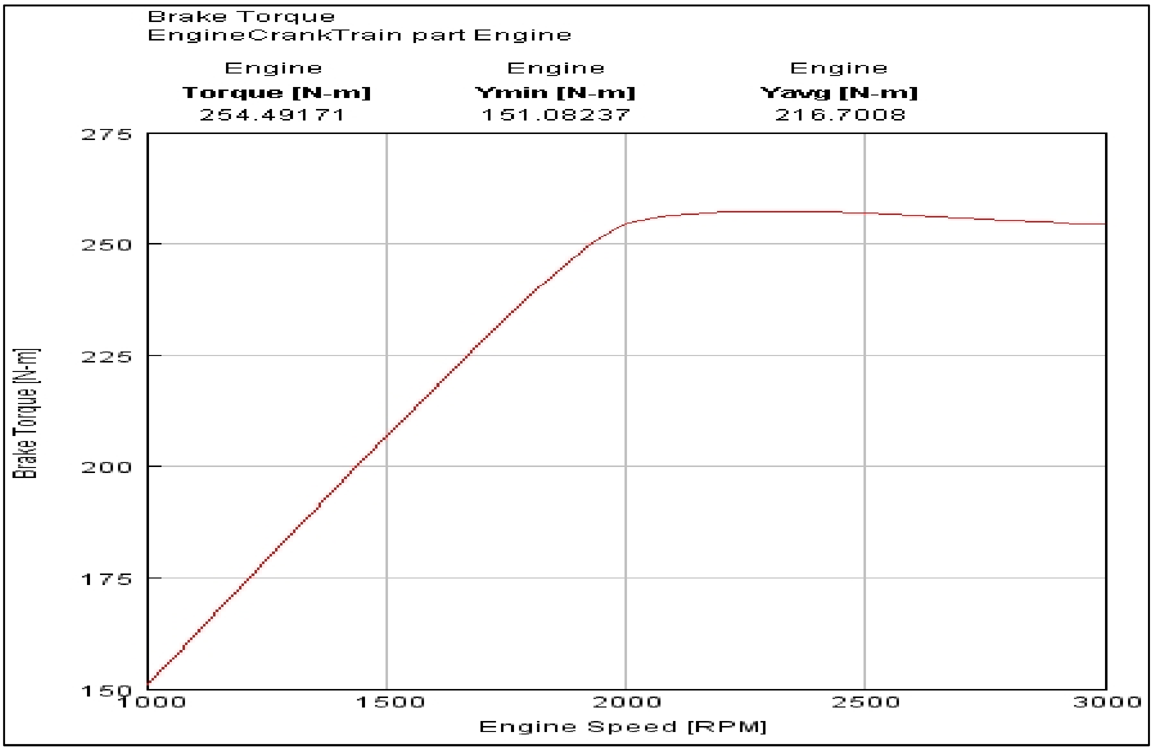

Figure (4.14): Nozzle 8 hole indicated Brake Torque of engine

4.4.8 Nozzle 9 hole

The figure(4.15) below indicates the results obtained when using (9) hole injector that illustrates the relationship between the engine speed (rpm) and the Brake Power(kw) the max power output is (79.98 kw) At engine speed (3000 rpm) and min power (15.82 kw) at engine speed (1000 rpm).

In The figure(4.16) show the relationship between the engine speed (rpm) and the break torque (N-M) the break torque output is (254.61 N-M) At engine speed (2300 rpm) and min torque (151 N-m) at engine speed (1000 rpm).

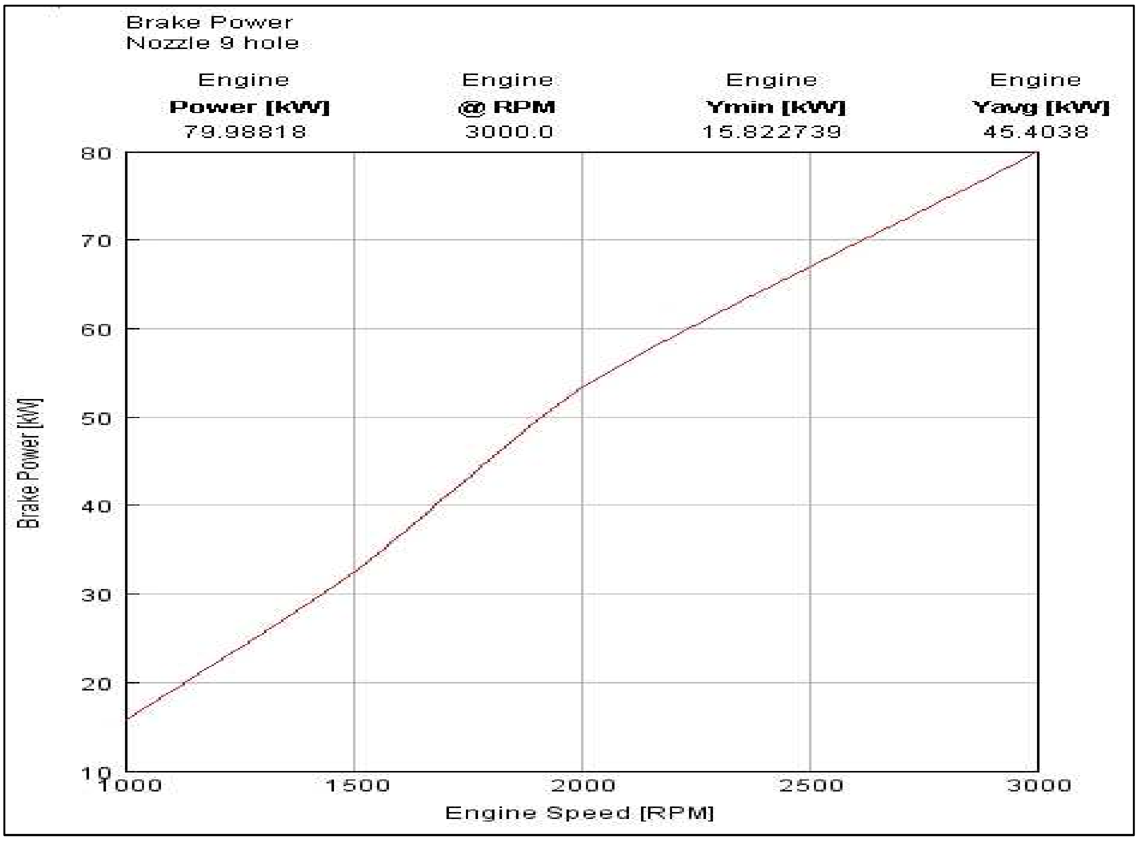

Figure (4.15): Nozzle 9 hole indicated Brake Power of engine

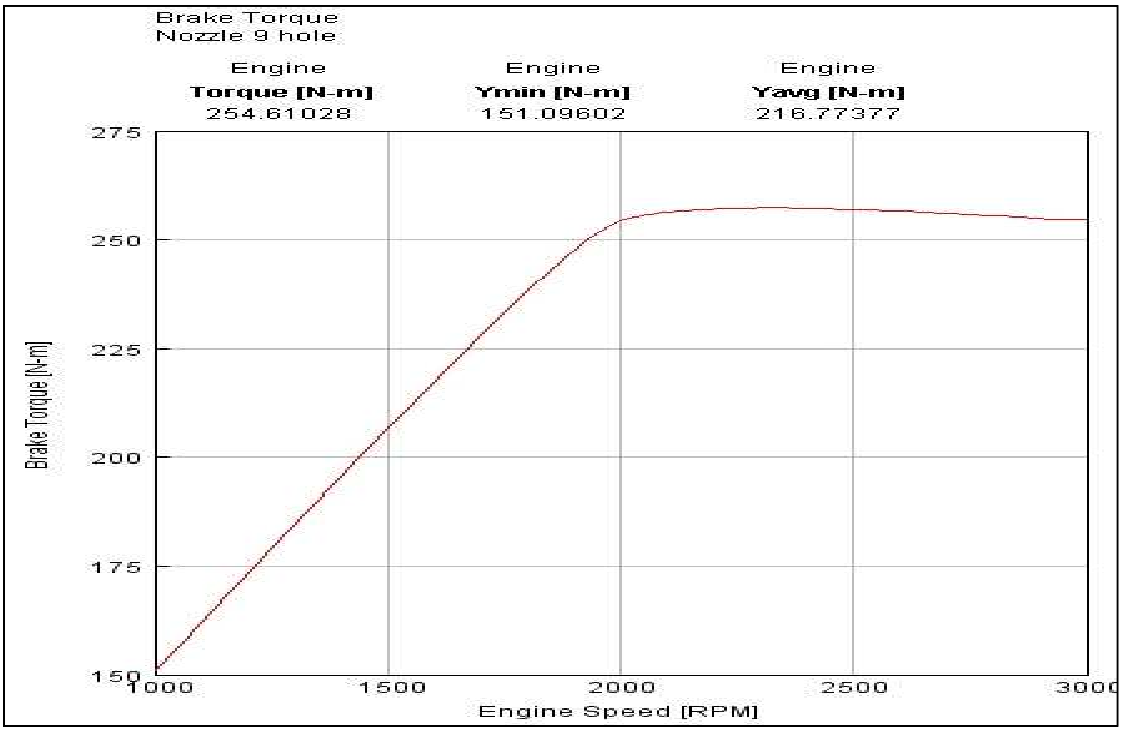

Figure (4.16): Nozzle 9 hole indicated Brake Torque of engine

Table(4.1):Brake Power in all cases

Case / Hole no	1 3000 rpm	2 2000 rpm	3 1500 rpm	4 1000 rpm
2	79.941345	53.323936	32.475	15.537051
3	79.96996	53.298904	32.48543	15.697922
4	79.9778	53.292545	32.473503	15.820479
5	79.93912	53.29928	32.477306	15.82101
6	79.95145	53.314335	32.47858	15.822085
7	80.025764	53.291477	32.481228	15.8214
8	79.950935	53.297997	32.476208	15.821308
9	79.98818	53.318047	32.486244	15.822739

4.5 Total Injected Mass

The amount of fluid injected per cycle or the name of a dependency reference object.

Table(4.2): Mass Flow Rate in all cases

Hole no \ case	max				unit
	1	2	3	4	
2	67.67	67.95	67.30	67.26	mg/s
3	67.68	67.80	67.30	67.26	mg/s
4	67.69	67.67	66.74	66.93	mg/s
5	67.68	67.80	67.80	67.30	mg/s
6	67.62	67.60	67.30	66.97	mg/s
7	67.76	67.61	67.28	66.98	mg/s
8	67.9	67.63	67.27	67	mg/s
9	68.13	67.64	67.49	67.3	mg/s

Table(4.3): Burned Fuel (Mass) in all cases

Hole no \ case	max				unit
	1	2	3	4	
2	46.92	47.45	42.44	32.94	mg
3	46.97	47.29	42.27	32.99	mg
4	46.99	47.15	42.08	32.97	mg
5	47.03	47.04	41.94	32.75	mg
6	47.12	46.46	41.81	32.59	mg
7	47.28	46.89	41.72	32.45	mg
8	47.39	46.87	41.64	32.34	mg
9	47.6	46.89	41.6	32.26	mg

4.6 MassFraction

indicates that the concentration is entered as the mass fraction of unburned fuel (relative to total exhaust mass).

Table(4.4): Burned + Unburned CO, NO, and H2 Mass Fractions

case / Hole no	3000 RPM			unit
	CO	NO	H_2	
2	0.25	0.075	0.003	%
3	0.65	0.072	0.011	%
4	1.18	0.06	0.022	%
5	1.83	0.058	0.033	%
6	2.39	0.049	0.046	%
7	2.87	0.041	0.057	%
8	3.23	0.033	0.071	%
9	3.27	0.027	0.08	%

CHAPTER FIVE

CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion:

All of the nozzles were examined and the result shown that the six holes nozzle provided the best burning results for fuel in-cylinder burned in any different engine speed in simulation and the best burning is in low speed engine. In engine performance effect, all of the nozzles examined and the seven holes nozzle provided the best results for indicted power, indicated torque in any different engine speed in simulation. The following conclusion can be drawn from this research work:

1- The low fuel consumption in use 6 hole.

2- There's a perfect distribution of diesel injections in the combustion chamber when using a 5-6 or 7-hole injector.

3- Mass flow rate values are increases by increasing the nozzle hole number.

4- Burned fuel values are increases by increasing the nozzle hole number and engine speed.

5.2 Recommendations:

The following recommendations are made in reference to this study:

1- When the injector is designed to be considered using the number of hole and calculate the fuel consumption.

2- Future efforts will focus on engine emission studies.

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