EFFECT OF EXHAUST GAS RECIRCULATION TO PERFORMANCE AND EMISSION OF DIESEL ENGINE

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I certify that the project entitled “Effect of Exhaust Gas Recirculation to Performance and Emission of Diesel Engine” is written by Mustaqim Bin Mohamad. I have examined the final copy of this project and in our opinion, it is fully adequate in terms of scope and quality for the award of the degree of Bachelor of Engineering. I here with recommend that it be accepted in partial fulfilment of the requirements for the degree of Bachelor of Mechanical Engineering.

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Thesis submitted in fulfilment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering with Automotive

Faculty of Mechanical Engineering
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DECEMBER 2010

SUPERVISOR'S DECLARATION

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I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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Dedicated,

with love to my loving mother, father and family. Also to all lectures and friends who have been supported and give encouragement.

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ABSTRACT

Direct injection (DI) diesel engine is well design today as a main power train solution for trucks and others relevant heavy duty vehicles. However, at the same time emission legislation, mainly for oxides of nitrogens (NOx) and particulate matter (PM) becomes more obvious, reducing their limit to extremely low values. One efficient method to control NOx in order to achieve the future emission limit are the rather high exhaust gas recirculation (EGR) rates accompanied by increased boost pressure to avoid the negative impact on soot emissions. EGR is one of the most effective means of reducing NOx emissions from compression ignition (CI) engines and is widely used in order to meet the emission standards. In the present work, experimental investigation has been carried out to make an analysis the NOx reduction characteristics and the effect to the engine performance by using exhaust gas recirculation between two different fueled engine using biodiesel (palm oil methyl ester B5) and light diesel. This experiment was carried out using a four-cylinder DI diesel engine Mitsubishi 4D68. The purpose of this experiment was conducted is to know the effect of EGR on diesel engine performance and the quantity of NOx emissions in diesel engine’s exhaust. The performance of the diesel engine and the quantity of NOx emissions in exhaust gas with using EGR in this experiment will be compared with the performance of the engine and the quantity of NOx emissions in the exhaust gas which not using EGR during the experiment. The results obtained by experiments showing that the EGR has caused the engine performance decreased as compared with no use of EGR. EGR is the best solution for reduced the production of NOx in the exhaust.

ABSTRAK

Pada zaman ini enjin diesel yang menggunakan sistem minyak suntikan langsung telah dibina dengan sebaiknya dan telah menjadi salah satu nadi penggerak utama bagi lori dan kenderaan besar yang lain. Pada masa yang sama, undang-undang pembebasan pencemaran ke udara sekeliling telah diketatkan, terutamanya bagi gas nitrogen oksida (NO_x) dan bendasing seperti jelaga hitam dari ekzos kenderaan yang menggunakan enjin diesel, undang-undang telah ditetapkan supaya kuantiti pembebasan asap dan jelaga ini dikurangkan kepada tahap yang paling rendah. Oleh itu, salah satu cara yang paling berkesan untuk mengurangkan kuantiti pembebasan gas NO_x ini ialah dengan menggunakan sistem resikulasi gas ekzos (EGR) dan disertai dengan meningkatkan tekanan yang tinggi dalam enjin bagi mengelakkan kesan negatif dalam pembebasan jelaga. EGR adalah salah satu cara yang paling berkesan untuk mengurangkan pembebasan gas NO_x dari enjin diesel dan digunakan secara meluas dalam rangka memenuhi piawaian pembebasan gas ekzos enjin diesel. Bagi membuktikan kenyataan ini, satu eksperimen telah dijalankan untuk menganalisis pengurangan pembebasan NO_x dalam asap ekzos dan kesan EGR ini terhadap prestasi enjin dengan menggunakan dua jenis minyak yang berbeza, iaitu minyak diesel biasa dan minyak biodiesel (minyak kelapa sawit B5). Eksperimen ini telah dijalankan dengan menggunakan enjin diesel mitsubishi 4D68. Tujuan eksperimen ini dijalankan ialah untuk mengatahui kesan penggunaan EGR pada prestasi enjin diesel dan kuantiti pembebasan gas NO_x dalam asap ekzos enjin diesel. Prestasi enjin diesel dan kuantiti pengeluaran gas NO_x yang menggunakan EGR dalam eksperimen ini akan dibandingkan dengan prestasi enjin dan kuantiti pengeluaran gas NO_x yang tidak menggunakan EGR semasa eksperimen dijalankan. Keputusan yang diperolehi berdasarkan eksperimen menunjukkan bahawa EGR telah menyebabkan prestasi enjin menurun berbanding dengan eksperimen yang tanpa menggunakan EGR. Namun begitu EGR adalah jalan penyelesaian terbaik untuk mengurangkan pengeluaran NOx dalam asap ekzos.

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LIST OF SYMBOLS

τ	Torque
π	pi
$\dot{w}$	Power
$\emptyset$	Equivalent Ratio

LIST OF ABBREVIATION

BDC	Bottom Dead Center
CI	Compression Ignition
CO_2	Carbon dioxides
DI	Direct Injection
EGR	Exhaust gar recirculation
HC	Hydrocarbon
NOx	Nitrogen oxides
O_2	Oxygen
PM	Particulate matter
r_c	Compression ratio
TDC	Top Dead Center

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Nowadays, people have come with new discovered source of fuel for the diesel engine named biodiesel. Biodiesel is an alternative choice of fuel to replace the crude oil from the industry. Biodiesel fuel offers a potential reduction of carbon dioxides (CO_2) and hydrocarbon (HC) emissions due to its higher content of oxygen (O_2). Many studies of biodiesel engine have found that exhaust from biodiesel fuel has higher nitrogen oxide (NO_x) emissions while HC and lower particulate matter (PM) than conventional diesel fuel (Yoshimoto, Onodera et al., 1999)

EGR is used to solve the problem of excessive NO_x emission from the biodiesel exhaust (Santoh, Zhang et al., 1997). EGR is operated by re-circulating the gas produced by the diesel engine exhaust back to the engine cylinder, so that the exhaust gas which is re-circulating replaces some of the excess O_2 in the pre-combustion mixture. At higher temperature which is about 1371^0C, the formation of NO_x will be faster. Chemical gases are formed from the chemical reaction between nitrogen (N_2) and O_2 in the combustion chamber. When these gases react with HC with the presence of sunlight, a black haze will appear in the skies known as smog. The EGR reduces the amount of NO_x in the exhaust gas emission by re-circulating it into the intake manifold where it mixes with air-fuel-ratio charge. The result of the mixing of inlet air with re-circulated gas is that peak combustion temperature and pressure are reduced by diluting the mixture of air-fuel-ratio at these conditions.

Generally, EGR flow is divided into three conditions. Firstly is high EGR flow. High EGR flow is necessary during cruising and mid-range acceleration and it happens when the combustion temperature is very high. Secondly is low EGR flow which is necessary at low speed and light load condition. Lastly is no EGR in which this condition occurs when EGR operation could adversely affect engine operating efficiency or vehicle drivability, such as when engine is warmed up, idling, and when the throttle is wide open.

The operation of the EGR will affect the performance of the diesel engine. Theoretically, EGR system has to precisely match with the various EGR flows and to over ride flow under condition which would give the best performance to the engine. A precise combination of the amount of EGR into the intake manifold and the load change will give a good result to the engine performance. The engine performance will be reduced if the exhaust gas is too much metered into the intake manifold. While in the other hand, if too little EGR flow goes into the intake manifold, it will also give a negative effect in which the engine might suffer from knocking and thus end up not meeting strict emission standards.

The effectiveness of EGR to reduce the emission contained in the exhaust gas depends on the flow of EGR. When the flow is too slow, it will cause detonation and emissions failure for excessive NO_x, because EGR tends to reduce the vitality of the air fuel charge. Otherwise, when too much EGR, an excessive flow for driving condition will cause stumble, flat spot, hesitation, and surging. This problem happens because EGR dilutes the air-fuel charge. When the flow of EGR is too much compared to the engine demand, misfiring happens.

1.2 OBJECTIVES

To performed this experiment smoothly, several objective are aimed. The objectives of this experiment are:

i. To analyze the significance of exhaust gas recirculation (EGR) system in reducing nitrogen oxide in the exhaust gas emissions
ii. To study the effects of the EGR system towards the performance of the engine
iii. To compare the effects of using biodiesel and diesel fuel to the performance and emissions of a diesel engine.

1.3 SCOPES OF THE PROJECT

The scope of this experiment covers the aspects as stated below:

i. Install the engine with sensors such as thermocouples and pressure transducer.
ii. Calibrate the data acquisition system for data correction.
iii. Test the engine performance and gas emissions with and without EGR.

1.4 PROBLEMS STATEMENT

The problem statements of this experiment are shown below:

i. The advantages of using EGR in solving the problem to reduce the amount of nitrogen oxide in exhaust gas emission and its effects to the performance and emissions of the diesel engine.
ii. The difference between using diesel fuel and biodiesel fuel in its effects to the amount of nitrogen oxide emissions in the exhaust gas.
iii. The difference between the diesel engine performances when using diesel fuel and biodiesel fuel.

1.5 FLOW CHART FOR THIS PROJECT

This flow chart is about the flow of the experiment which was carried out to test the effect of the EGR towards the performance and emissions of a diesel engine.

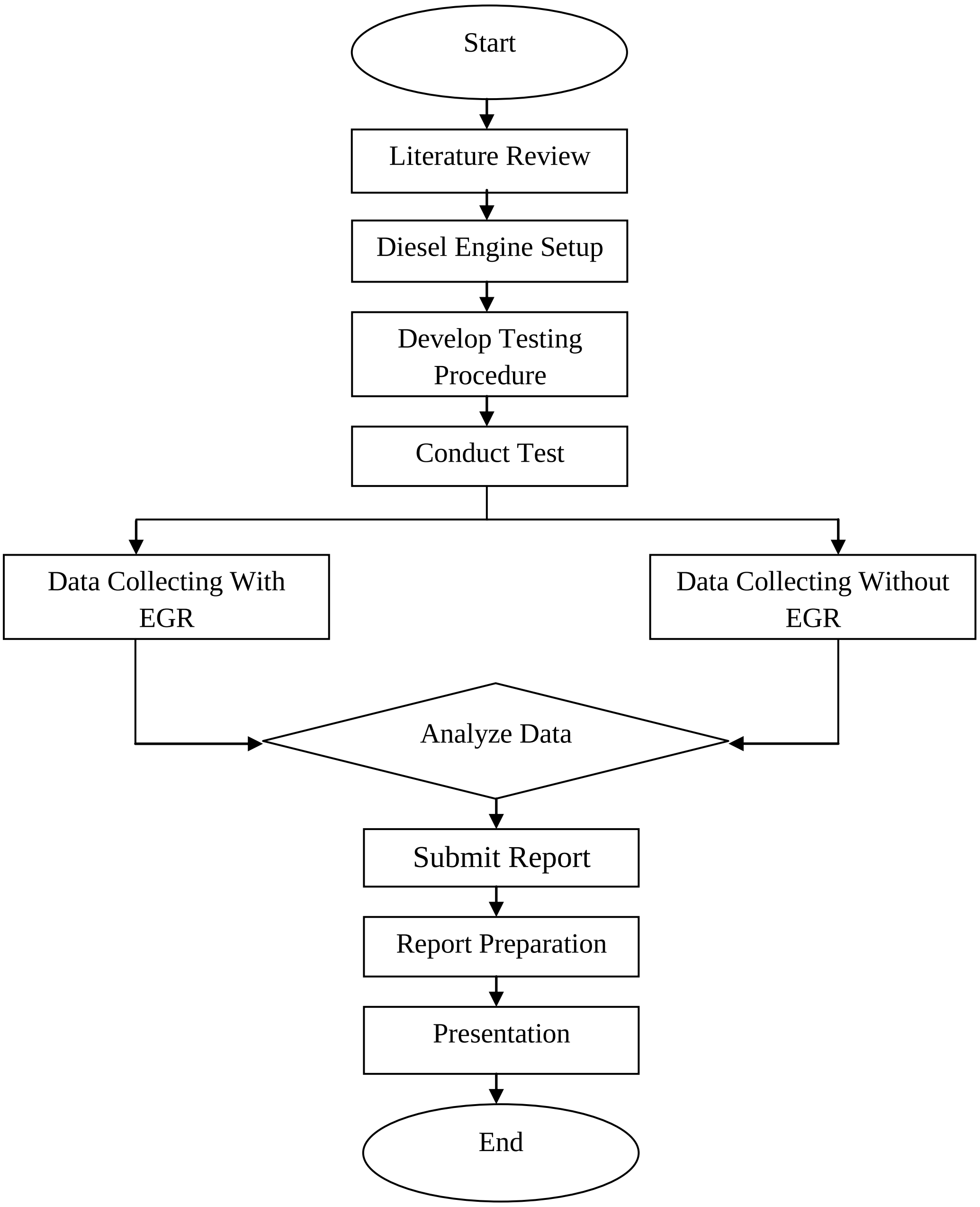


Figure 1.1 : Flow Chart for his project

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter consists of reviews about the related content or knowledge with regards to the effect of EGR towards the performance and emissions of a diesel engine.

2.2 DIESEL ENGINE

Diesel engine is like a gasoline engine, it is an internal combustion engine that converts chemical energy in fuel to mechanical energy which makes the piston moves up and down in the cylinders. To change the motion of the piston in the engine from linear motion to rotational motion, the pistons are connected to the crankshaft. This motion is needed to drive the vehicle’s wheel. The primary difference between the diesel engine and the gasoline engine is in the way the explosion in the cylinder occurs. In diesel engine operation, the fuel ignites on its own. Air heats up when it is compressed. Therefore, the spark plug and carburetor are replaced by a fuel injector in diesel engine. When the piston approaches the Top Dead Center (TDC), the fuel injection process in diesel engines will start and this process will continue during the first part of the power stroke. Therefore, the combustion process will happen over a long interval. The combustion process in the ideal diesel cycle is approximated as a constant-pressure heat addition process.

Process 1-2 is an isentropic compression, while 3-4 is an isentropic expansion and 4-1 is a constant heat-volume rejection (Cengel and Boles, 2007). Hence diesel engine is also known as compression ignition engine, it also has a high thermal efficiency because of its high compression ratio and fuel lean operation. To achieve the auto-ignition, high compression ratio is required as it will produce high temperature, so that the high expansion ratio will make the engine discharge less thermal energy in the exhaust. To complete and compensate the combustion for homogeneity in the fuel distribution, the extra oxygen (O_2) is necessary to facilitate it. However, locally stoichiometric air will cause the high flame temperatures predominate, so in such heterogeneous combustion process the fuel ratios will prevail (BormanGL and GaglandKW, 1998)

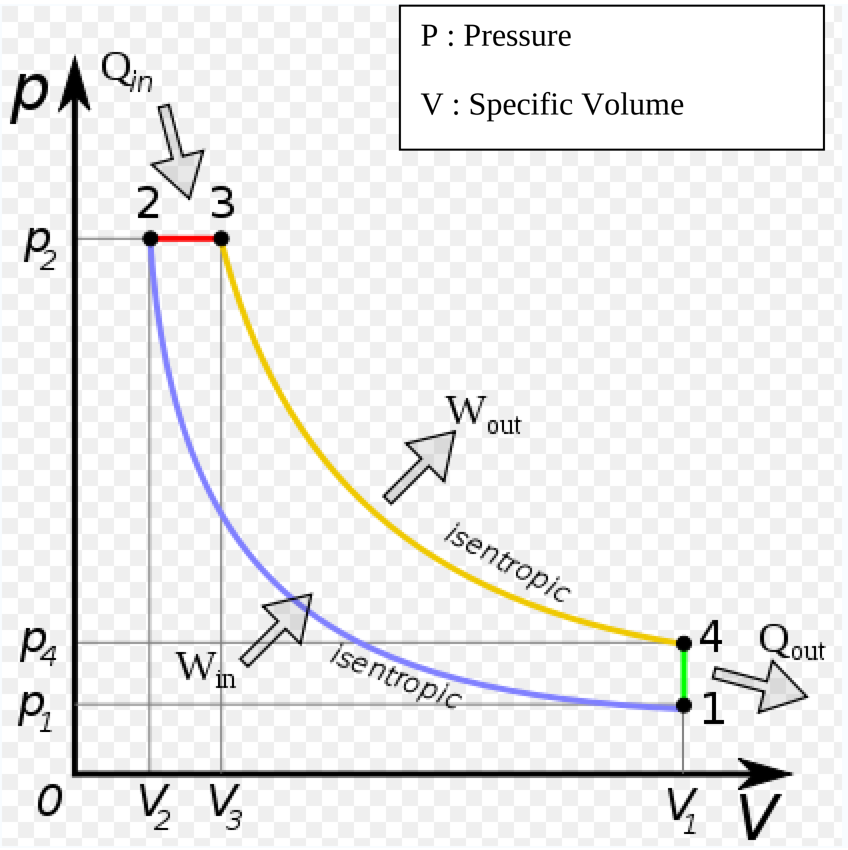


Figure 2.1 : Ideal p-v diagram for diesel cycle

Source : From the thermodynamics text book – an engineering approach sixth edition (SI units), Yunus A. Cengel, 2007

2.2.1 Compression Ratio

Compression ratio (r_c) is important in diesel engine. It is because, when air is compressed, the collision between molecules in cylinder will produce heat that ignites the diesel fuel. Compression ratio is the parameter to measure how much the air is compressed.

$$\text{Compression Ratio} = \text{swept volume} + \frac{\text{clearance volume}}{\text{sweapt volume}} \qquad (2.1)$$

- Swept Volume – the volume of the cylinder transverse by the piston travel from top dead center (TDC) to bottom dead center (BDC).
- Clearance Volume – combustion chamber volume.

The diesel engine needs a very minimum compression ratio at about 15:1. This is at the cold starting condition. When the compression ratio is higher, for example 16 or 17:1, the benefits are that the starting becomes easier and less exhaust smoke is produced. A compressor in the mode of turbocharger and supercharger will raise the effectiveness of the compression ratio (Dampsey, 2008).

2.2.2 Induction

In CI engine, the air is compressed first and only then the fuel will be admitted. The injector will be opened to inject the fuel when the piston is at the TDC. The advantage of compressing air rather than the mixture of oil and air to diesel engine is that it can increase the thermal efficiency of the engines. The compression ignition engine dispenses with the throttle plate. This plate’s function is to ensure that the amount of air entering the cylinder at all speed is same. At high speed or under heavy load, the additional fuel supplied drops to a ratio about 20:1. So without this plate, the diesel engine will be able to breathe easily at low speed. For example, a truck can idle for long periods without consuming appreciable fuel.

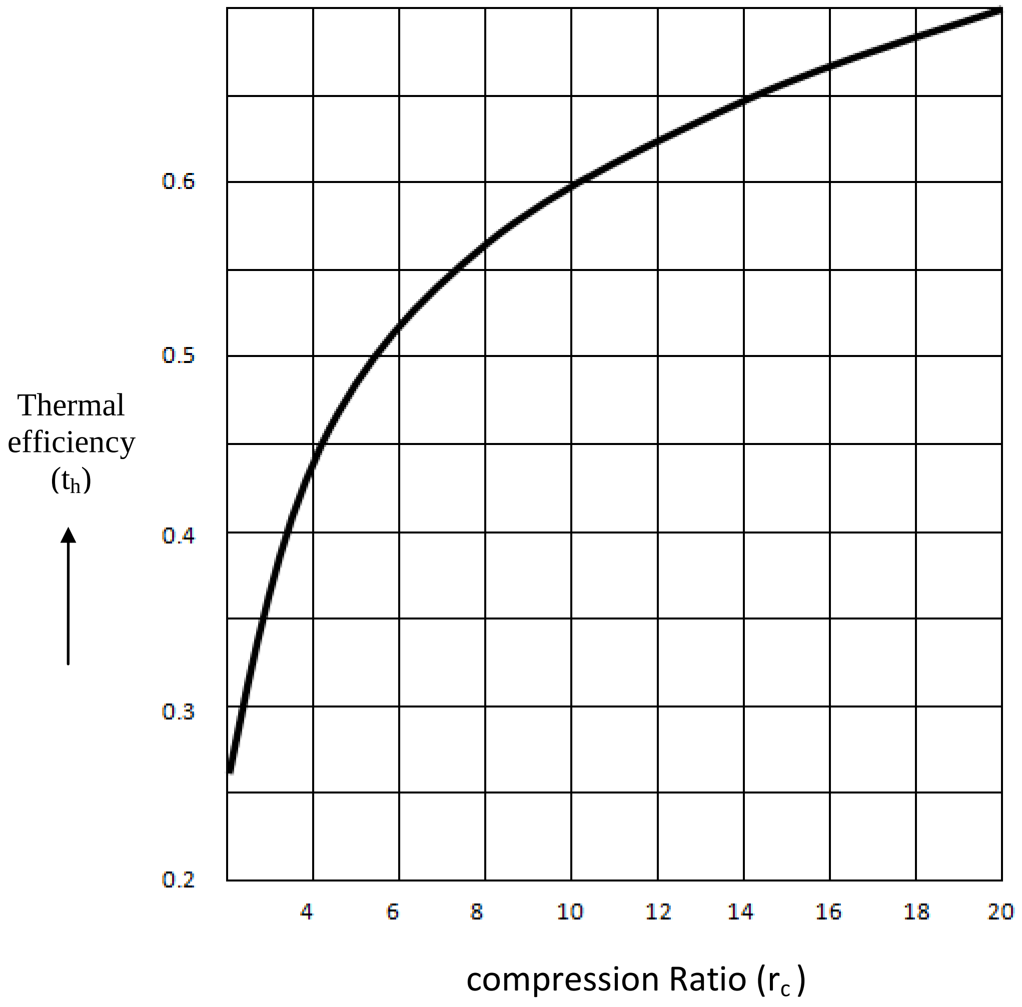


Figure 2.2 : Standard graph of relationship between diesel compression ratio and thermal efficiency.

Source : From the book of Trouble shooting and repairing diesel engine, Dampsey, 2008

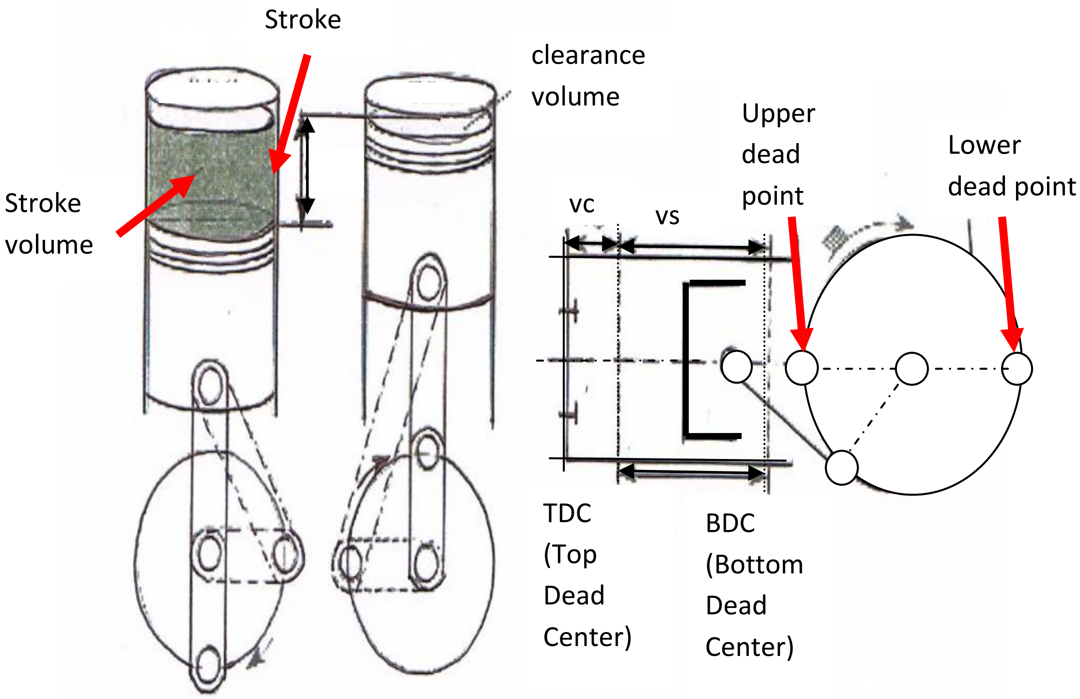


Figure 2.3 : Schematic diagram of diesel engine compression ratio

Source : From the book of Trouble shooting and repairing diesel engine, Dampsey, 2008

2.2.3 Ignition and Combustion

The ignition in diesel engines happens rapidly with accumulated fuel burns corresponding to the temperature and pressure that increase rapidly. At the time rapid combustion happens, the injector is still receiving the fuel and thus a period of controlled combustion follows. The greater the ignition lag, the more violent the combustion as well as greater resulting noise, vibration and harshness (Dampsey, 2008). During the initial starting, at the time where the engine in the cold condition, the ignition lag becomes very bad. The will produce noise, white smoke, and rough combustion as known as ‘diesel detonation’. This phenomenon will disappear when the engine condition is warm.

2.2.4 Two Stroke and Four Stroke Cycle

The operation of a four stroke-cycle of an engine is as shown in the Figure 2.4. At the first cycle, intake valve is opened and air is allowed to enter into the cylinder. Then the intake valve is closed at the second cycle as the piston at the BDC moves up to the TDC thus compressing the volume of air inside the cylinder. At the high compression and temperature, the fuel is injected for ignition. When the fuel is ignited, it drives the piston down as known as expansion cycle and this is third cycle. Then the last cycle, the exhaust valve opened and the piston rises again to reject the spent mixture of gases and fuel. This cycle called the exhaust stroke completes the cycle (Dampsey, 2008). However for the two cycle engine, the intake stroke and the compression stroke are combined to be one cycle. The third and fourth cycle, expansion stroke and exhaust stroke are also combined to become one cycle.

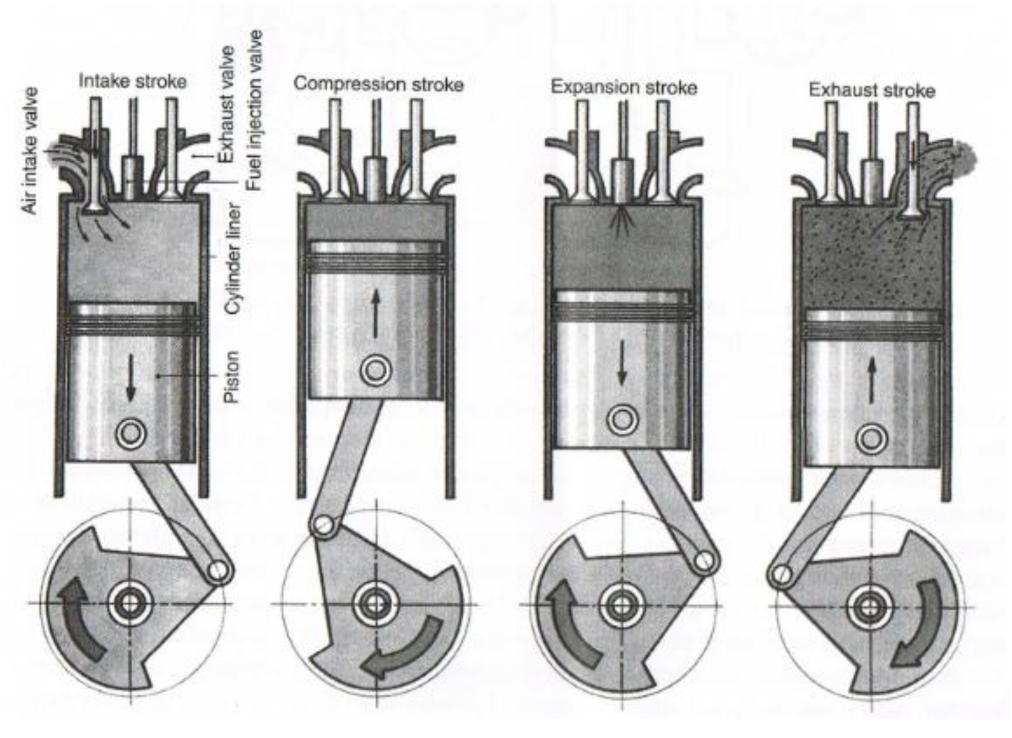


Figure 2.4 : Schematic diagram of Four stroke-cycle

Source : From the book of Trouble shooting and repairing diesel engine, Dampsey, 2008

2.2.5 Power and Torque

Torque is one of the instantaneous twisting forces that is plied to the crankshaft. The torque can also be expressed as :

Horsepower = torque x 2π x rpm (revolution per minute) (2.2)

Torque = displacement x 4π x break mean effective pressure. (2.3)

The break mean pressure is also known as average pressure that is applied to the piston during the expansion stroke. When the value of the break mean pressure is high, it will give advantage to the diesel engine because it makes the engine to have superior torque (Dampsey, 2008).

2.3 NITROGEN OXIDE (NOx)

All the fossil fuel burning processes form nitrogen oxide gas. Both nitric oxides (NO) and nitrogen dioxide (NO_2) are elements in the air pollution. They are the elements in producing NO_x. Comparing these two elements, the NO is a more toxic and irritating compound. The major element in the producing NO_x is NO which contributes 90-95% and NO_x is the rest of the element (Dil Worth, Nations et al., 2003). From the combustion process, the NO will be produced. It arises from high temperature reaction between the N_2 and O_2 in the combustion air and from the oxidation of organically bound N_2 in certain fuels such as coal and oil (Flagan and Seinfeld, 1988). NO_x is a binary compound with chemical reaction between two molecules which is N_2 and O_2. When the combustion chamber exceeds the critical temperature, then the oxides of nitrogen are formed during this combustion. It is because the molecules of N_2 and O_2 are combined (Saleh, 2009). The chemical reaction of N_2 and O_2 is :

$$N_2 + O \longrightarrow NO + N \tag{2.4}$$

$$N + O_2 \longrightarrow NO + O \tag{2.5}$$

The formation of the NO_x usually happens at the high temperature and in the compression zone where there are presence of both N_2 and O_2 (Dilworth, Nations et al. 2003). The NO_x concentration is based on the time, temperature, and concentration history of the combustion. Then their concentration will increase with temperature, the availability of O_2 and the time of the N_2 and O_2 simultaneously exposed to peak flame temperatures (Dilworth, Nations et al. 2003). The diesel engine needs a high compression in the engine cylinder to produce power. At the same time, because of the high flame temperature in the presence of abundance N_2 and O_2, diesel engine combustion generates a large amount of NO_x (AM, JH et al. 1988; P, H et al., 1988.). This large amount of the NO_x comes from the exhaust emissions and this emission will have a negative impact to humanity.

2.4 BIODIESEL

Due to the fast depletion of the petroleum oil, researches about biodiesel fuel begin. The interest of researches towards new energy is not a new thing. Many studies have been made to produce a new energy source that can substitute petroleum. The advantages of this new source are that it may help to lessen our dependence on fossil fuel and also may reduce the net production of CO_2. Therefore in the United State, they use the soybean oil as biodiesel. European countries are concerned with rapeseed oil while at the countries which are covered with tropical climate, coconut oil, hazelnut oil or palm oil is utilized, (Y and YD 2003; C, A et al. 2006; M and F 2007). Other than that, other sources such as animal fat are also included in the source of biodiesel such as salmon oil or waste cooking oil (AS, S et al. 2006; JF and MA 2006; N, S et al. 2006; A, M et al. 2007). In other words, the name ‘biodiesel’ can be applied to any transesterified vegetable oil that makes it suitable for use as a diesel fuel. Biodiesel is technically known as vegetable oil methyl ester, and is composed of long chain fatty acid with an alcohol attached. Transesterification and esterification reaction of the vegetable or waste of oil or animal fats respectively with a low molecular weight of ethanol and methanol in the presence of catalyst will produce biodiesel. Biodiesel is known as methyl ester if the alcohol used is methanol, while will be called as ethyl ester if used ethanol. Methyl ester is cheaper due to the cost of methanol. In 1895 Dr. Rudolf Diesel developed the first diesel engine to run with vegetable oil. In 1900, an engine that runs with the peanut oil was demonstrated in the World Exhibition in Paris. But despite the progress, diesel fuel became the main choice over biodiesel because it was cheap, reasonably efficient and readily available (Vijayalakshmi, Devi et al., 2007).

Table 2.1 are listed the variable of diesel fuel that are used today to generate the diesel engine.

Table 2.1: The List of Biofuel For Diesel Engine

Common Name	Botanical Name
Avocado (Butter fruit)	Persca americana
Babassu palm	Orbignya martiana
Bacuri palm	Platonia insignis
Brazil nut	Bertholletia excelsa
Buffalo ground	Cucurbita foetidissima
Buriti palm	Mauritia flexuosa
Calendula	Calendula officinalis
Camelina	Camelina sativa
Cocoa	Theobroma cacao
Cocnut	Cocos nucitera
Coffee	Coffee Arabica
Coriander	Coriandrum sativum
Corn	Zea mays
Cotton seed	Gossypium hirsutum
crambe	Crambe abyssinica
Euphorbia	Euphorbia lagascae
Gopher plant	Euphorbia lathyris
Jatropha	Simmondsia chinensis
Jojoba	Hibiscus cannabinus L

Source : From the book of Trouble shooting and repairing diesel engine.

Dampsey, 2008

The use of biodiesel does not requires modifications of the diesel engine (K. Rajan, 2007). Biodiesel is the alternative for diesel engine fuel, replacing petroleum oil. However, biodiesel fuel engine produces higher NO_x emission compared to the diesel oil based on the many researches (Yoshimoto, Onodera et al., 1999) .

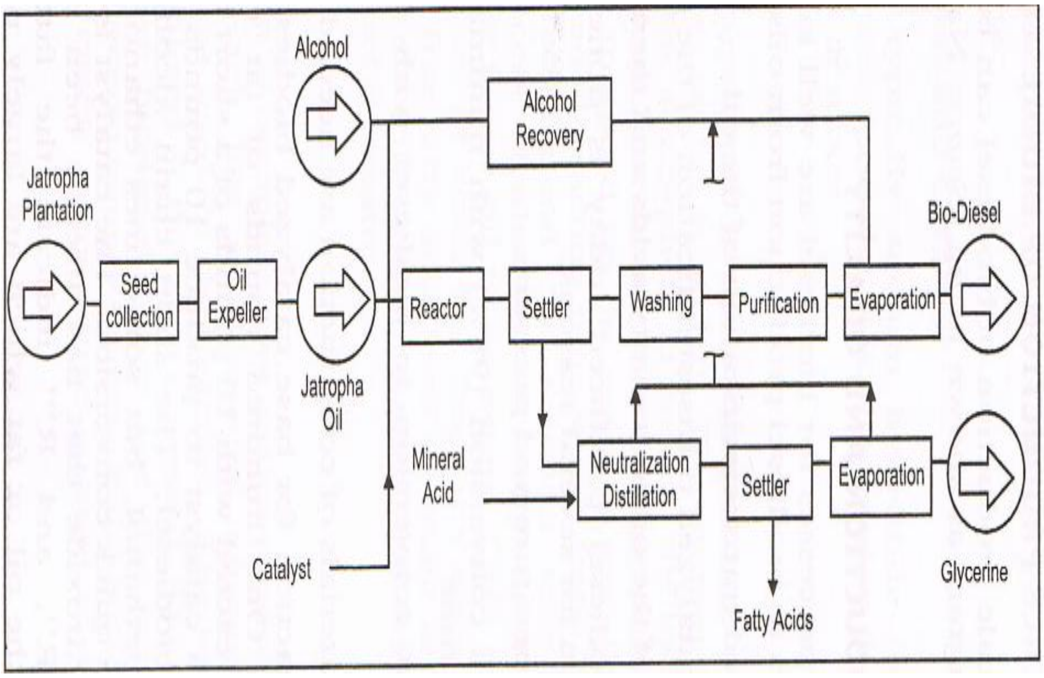


Figure 2.5 : Process to produces biodiesel.

Source : From the book of Fuel and biofuel, vijayalakshmi, 2007

In order to use biodiesel, the efficiency of this fuel must be considered. The followings are some of the characteristics for the efficient biodiesel. These properties are mainly listed on the basis of ASTM standard for biodiesel.

2.4.1 Kinematic Viscosity

Viscosity is the important characteristic for the flow, for example the tendency of the fluid to deform with stress. Then this is the reason for why the vegetable oil is selected as the biodiesel fuel, it is because the viscosity is less, so it will improve fuel flow characteristic. Viscosity is determined by measuring the time taken by the oil to pass through an orifice of a specified size. Viscosity can affect the injector lubrication and fuel atomization. The advantage of the low viscosity fuel is it may not produce sufficient lubrication for the precision fit of fuel injection pump, resulting in leakage or increased wear. Other than that, viscosity also will affect the fuel atomization. Larger droplet in injection will occur with high viscosity of the diesel fuel. This droplet can cause poor combustion, increased exhaust smoke and emissions. The viscosity of vegetable oil is 10 to 15 times greater than diesel fuel viscosity (Vijayalakshmi, Devi et al., 2007)

2.4.2 Density

Density is measured by the weight of the fuel per unit volume. The higher density of oil will give more energy. Density dictates the energy content of fuel where high densities indicate more thermal energy for the same amount of fuel and therefore better fuel economy. Biodiesel is more denser than diesel oil, for example the sample values ranging between 877 kg/m^3 (tallow methyl ester) to 884 kg/m^3 (soy methyl ester) compared with diesel at 835 kg/m^3 . Thus, density of the final product depends mostly on the feedstock used (Vijayalakshmi, Devi et al., 2007).

2.4.3 Caloric Value or Heat of Combustion

The energy generated by the fuel can be determined by the caloric value or heat of combustion. It is the amount of the energy released by the combustion process of a unit value of fuels. The very important thing in heating value is moister contents. Air-dried biomass has about 15-20% of moisture, and in coal the range of the moisture content is 2-30%. By the way, the biomass feedstock’s bulk density is very low even after densification. It is about 10 and 40% only. However biofuel has bigger bulk density compared to the fossil fuels (Vijayalakshmi, Devi et al., 2007).

2.4.4 Flash Point (FP)

Flash point is the temperature at which the fuel becomes a mixture then will ignite when exposed to a spark or flame. The ignition will occur at the minimum temperature. The flash point is the lowest temperature at which contact with a flame causes the vapor of the fuel to ignite under specified condition. This minimum flash temperature is important for the safety and handling in the diesel fuels. The flash point will increase as the percentage of biodiesel increases. Compared to the diesel fuel, the flash point in biodiesel is approximately twice. Based on the sample of the biodiesel, the range is 160^{o}C for tallow methyl ester to 188^{o}C for soy methyl ester. So the biodiesel blends with the petroleum and pure biodiesel is safer to use than conventional diesel fuel. Flash point is a parameter to know the flammability of the material. For transportation and safety purpose, the flash point is set high. It is also to inform the manufacture to remove sufficient alcohol from the process. The poor combustion properties will exist when the residual alcohol reduces the flash point of the biodiesel (Vijayalakshmi, Devi et al., 2007).

2.5 EXHAUST GAS RECIRCULATION (EGR)

EGR has received attention as a potential solution to reduce the emissions of the NO_x in the exhaust gas. Therefore it has been designed to reduce amount of the NO_x in the exhaust gas emission (A, M et al. 2002; T, D et al., 2003). There are two types of EGR, which are internal and external. Internal EGR uses variable timing or other devices to retain a certain fraction of exhaust from a preceding cycle. Besides that, the external EGR uses the piping to route the exhaust gas to the intake system, and this type of EGR is inducted into succeeding cycle. Then for this experiment, the external type is used (Saleh, 2009). The system is operated through recirculation of exhaust gas into the intake manifold by mixing it with the incoming air/fuel charge. By diluting this air fuel mixture, peak combustion temperature and pressure are reduced, thus the amount of NO_x that is produced by exhaust gas also reduced. (Santoh, Zhang et al., 1997). In another case, by using higher fuel consumption and emission penalties in EGR, it will increase the HC, CO and PM emission (Yoshimoto and Tamaki, 2001). While it is very effective in reducing the NO_x in exhaust gas emissions, it also has adverse effects on the engine efficiency and may cause pollution to the lubricating oil and corrosion of inlet manifold and moving parts, as exhaust contains PM (MD, JF et al., 2004).

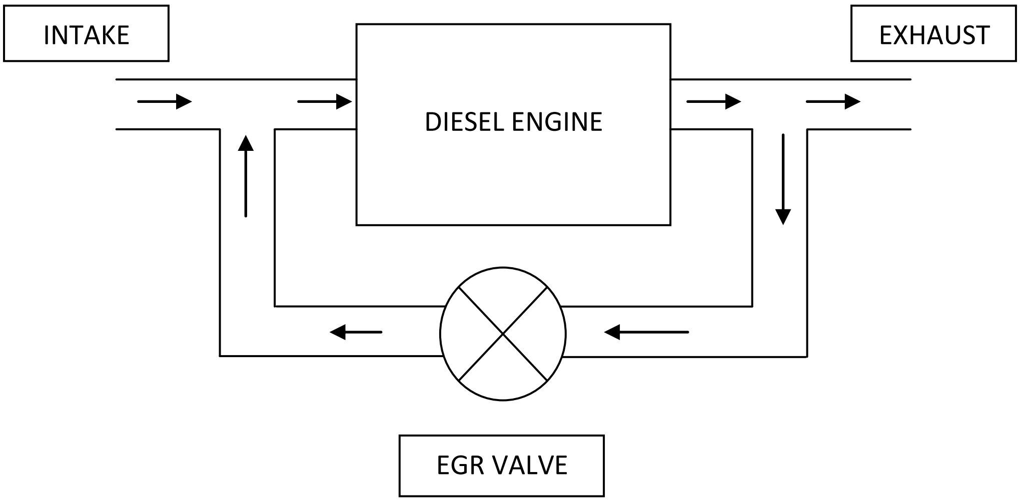


Figure 2.6 : Schematic diagram of Exhaust gas recirculation

Source : Energy Conversion and Management paper - a review on advanced and novel concept, Ming Zheng, 2003

CHAPTER 3

METHODOLOGY

3.0 INTRODUCTION

This chapter is based on the set up of the experimental facilities and apparatus where the four cylinders diesel engine is used for this project. This engine has been modified and run with biodiesel fuel. Some modifications that were done on this engine are several holes that are made at the exhaust extractor system, intake manifold, EGR pipe, engine’s oil tank, and in/out cooling system. This chapter also covers the apparatus and measuring devices, for example exhaust gas monitoring and gas analyzer, fuel, thermocouples system and gas fuel reforming system.

3.1 ENGINE AND INSTRUMENTATION

3.1.1 Engine

Figure 3.1 shows that the engine which was setup to run the experiment of engine performance and gas emission by using different fuel and with EGR or without EGR.

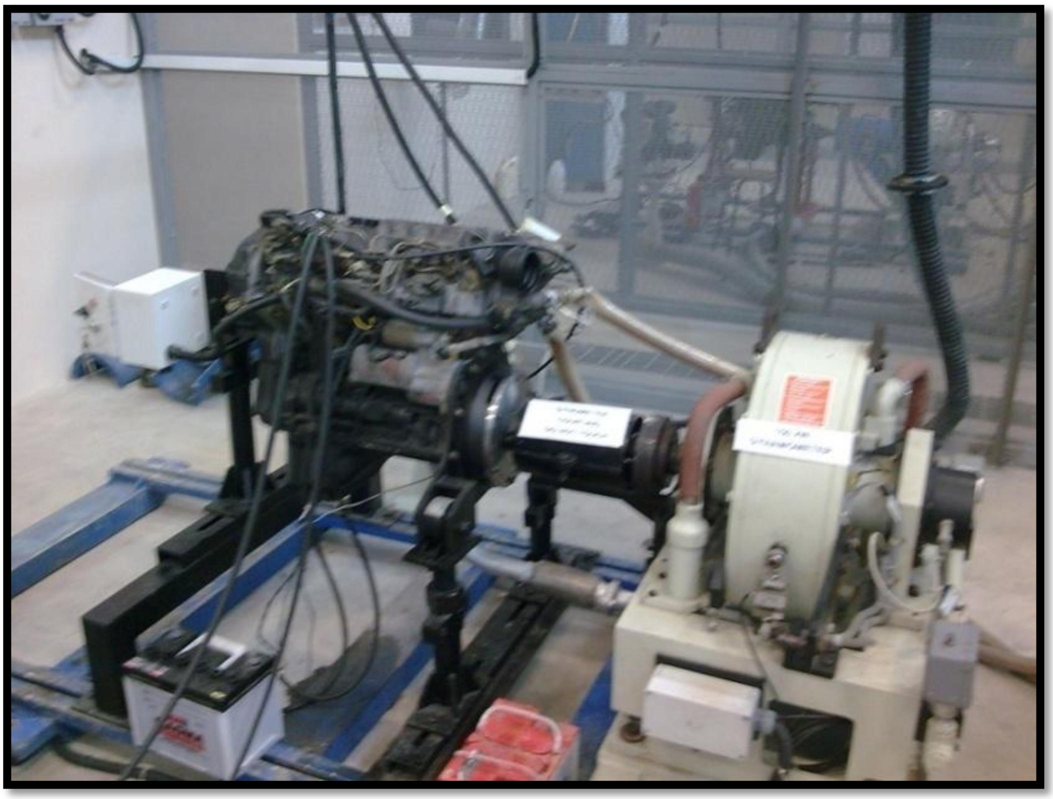


Figure 3.1 : The real 4-cylinder mitsubishi 4D68 engine.

This experiment is conducted by using the 4-cylinder diesel engine with power of 90 kW. Engine characteristics and specifications are shown in Table 3.1. Engine model that is used is mitsubishi 4D68. This engine is equipped with direct injection system of fuel supply. Its exhaust gas recirculation (EGR) system is operated by mechanical function. It does not use the ECU to control the EGR.

All the specifications of mitsubishi engine 4D68 are listed at the table as shown below.

Table 3.1 : The specification of mitsubishi engine 4D68

Description	Specification
Type	Diesel engine
Number of cylinder	4 in-line
Combustion chamber	Swirl chamber
Total diplacement (dm$^{3)}$	1.998
Cylinder bore (mm)	82.7
Piston stroke (mm)	93
Compression ratio	22.4
Lubrication system	Pressure feed, full-flow filtration
Oil pump type	External gear type
Cooling system	Water-cooled
Water pump type	Centrifugal impeller type
EGR type	Single type
Fuel system	Electronic control distributor-type injection pump
Rocker arm	Roller type
Adjusting screw	Elephant foot type
Oil lever sensor	Provided

Source : Data from manual of mitsubishi engine 4D68.

3.1.2 Eddy Current 150 kW Dynamometer

An eddy-current type water-cooled Schenck dynamometer model ECB – 200F from Dynalec Controls was used to load the engine. Main specifications and (absorbed) performance curve for the engine dynamometer are listed and depicted in Table 3.2 and Figure 3.2, respectively. The dynamometer sizing was chosen to cover the full-load engine operation for the whole range of engine speeds. Before performed the experiment, the dynamometer was calibrated in March 2010 using levers and weights that allow simple and precise adjustment and checking of the torque measuring chain, regardless of whether the brake is at standstill or rotating.

Figure 3.2 : 150 kW Eddy current Dynamometer for this engine testing

All the specifications of Eddy current Dynamometer are listed in the table as shown below.

Table 3.2 : 150 kW Eddy current Dynamometer specification

Model	ECB – 200F
Serial No.	617
W (pan) max	40 kg
W (ind) = W (pan) max x	10
Dyno constant	9550

Source : Data from Dynalec Control company

To control the dynamometer, a dynamometer control system series S2000 is used. It is positioned in the control room and is wired by an interconnection loom to the dynamometer. In addition, since the dynamometer is water-cooled type, a flow switch is installed in a cooling water feed line to detect the movement of water and is wired to the controller for safety. Generally, at any revolution speed except idle speed the engine operates on torque-controlled mode by which the user must set the dynamometer controller to be speed-controlled mode. However, at the idle speed when the engine operates on speed-controlled 47 mode, the user must set the dynamometer controller to the torque-controlled mode. This is extremely important because the wrong setting could destroy the engine transmission mechanism as well as the propshaft.

3.1.3 Engine-Dynamometer Shaft Coupling

The universal joint propshaft standard model of Kia lorry shown in Figure 3.3 was supplied by NDE Clarke Transmissions Ltd. The sizing of the propshaft was based on the engine maximum speed and maximum torque which is transmitted to the dynamometer. The compressed length of the propshaft (Lz) is 700 mm and the extension length (La) is 60 mm. Both flange fitting sizes (PCD) are 120 mm in diameter with 4 holes on each and the hole is of 12 mm diameter (φd).

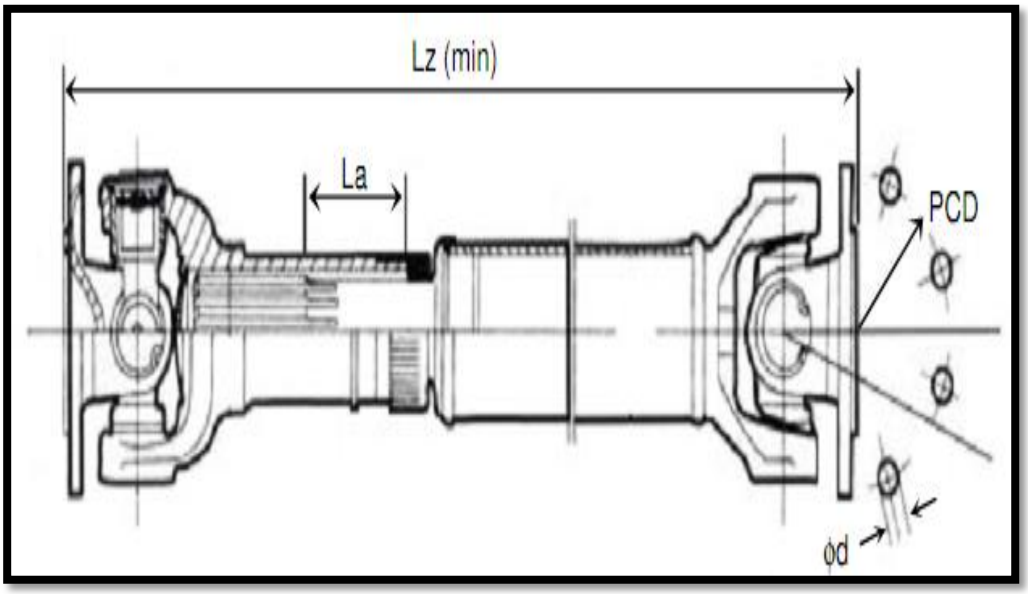


Figure 3.3 : Universal Coupling Shaft to connect the engine and dynamometer

Source: NDE Clarke Transmissions Ltd.

3.1.4 Cooling System and Controller

A cooling system is used to cool the engine during the testing. This cooling system consists of a large radiator, radiator fan, water tank, and water pump which functionalize to absorb the water from water tank and supply it to the engine through the water inlet hose. The water outlet from the engine is connected to the radiator and then to water tank. So the same water will circulate within the system. During the experiment, the temperature of the engine must be controlled to make sure their temperature is constant. The engine temperature must be maintained at 90°C, which follows the standard of engine testing procedure. The flow rate of the water cooling system must also be controlled. To control the water flow rate, a valve controlling is provided at the tank. This valve is manually controlled based on the temperature of engine which is monitored at the control room. The heat exchanger selection was based on a calculation of engine heat loss to cooling water. By using typical values of thermal efficiency (~27% for diesel engines) and heat loss to cooling water (~35% of brake power) (Heywood, 1988) the heat dissipation can be estimated.

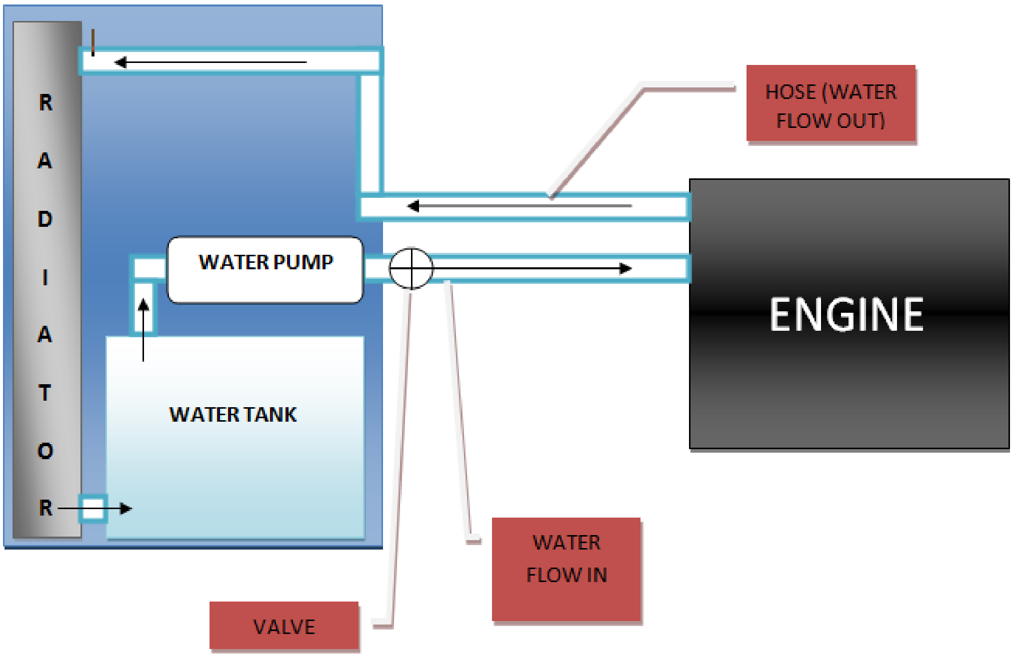


Figure 3.4 : Schematic diagram for the engine cooling system in this experiment

3.1.5 In-Cylinder Pressure Sensor

In-cylinder pressure traces from cylinder 1 were acquired by water cooled ThermoComp 6041A pressure transducers with small dimension, especially suited for small combustion in engines and for thermodynamic investigation in the laboratory. It is fitted with water cooling in a bore M8x0.75 (M8 stands for in-cylinder pressure diameter and 0.75 is their pitch). This pressure sensor cannot be put directly on the glow plug thread, so an additional screw is added as an adaptor to connect the pressure sensor with glow plug hole. This pressure transducer has high sensitivity of pressure, high natural frequency and excellent zero point stability because of integrated water cooling.

Figure 3.5 : Pressure sensor to measure in-cylinder pressure for experimental diesel engine

Key technical data of the water cooled ThermoComp 6041A pressure transducers are listed in Table 3.3.

Table 3.3 : In-cylinder pressure transducer specification

Measuring Range	0 – 250 bar
Calibrated Range	0 – 50 bar
Overload	300 bar
Sensitivity	-20 pC/bar
Natural Frequency	70 kHz
Operating Temperature Range	-50 -300 ^{o}C
Tightening Torque	6 Nm

3.1.6 Thermocouple

A thermocouple is a junction between two different metals that produces a voltage related to a temperature difference. Thermocouples are a widely used type of temperature sensor for measurement and. There is various type of thermocouple that can be used to measure temperature. Figure 3.6 above shows that the type of thermocouple that is used for this experiment is type-K thermocouple. This type of thermocouple is most popular due to its interchangeability, standard connectors and can measure a wide range of temperatures. Thermocouples are made of two dissimilar metals joined and it creates two junctions and those junctions are called ‘sensing junctions’ which can measure different temperatures. Electric current will flow through the created circuit and the temperature is measured in milivolt (mV). A signal converter will make the milivolt reading and compare the milivolt (mV) value of the reference junction at a known temperature in order to be able to measure the temperature.

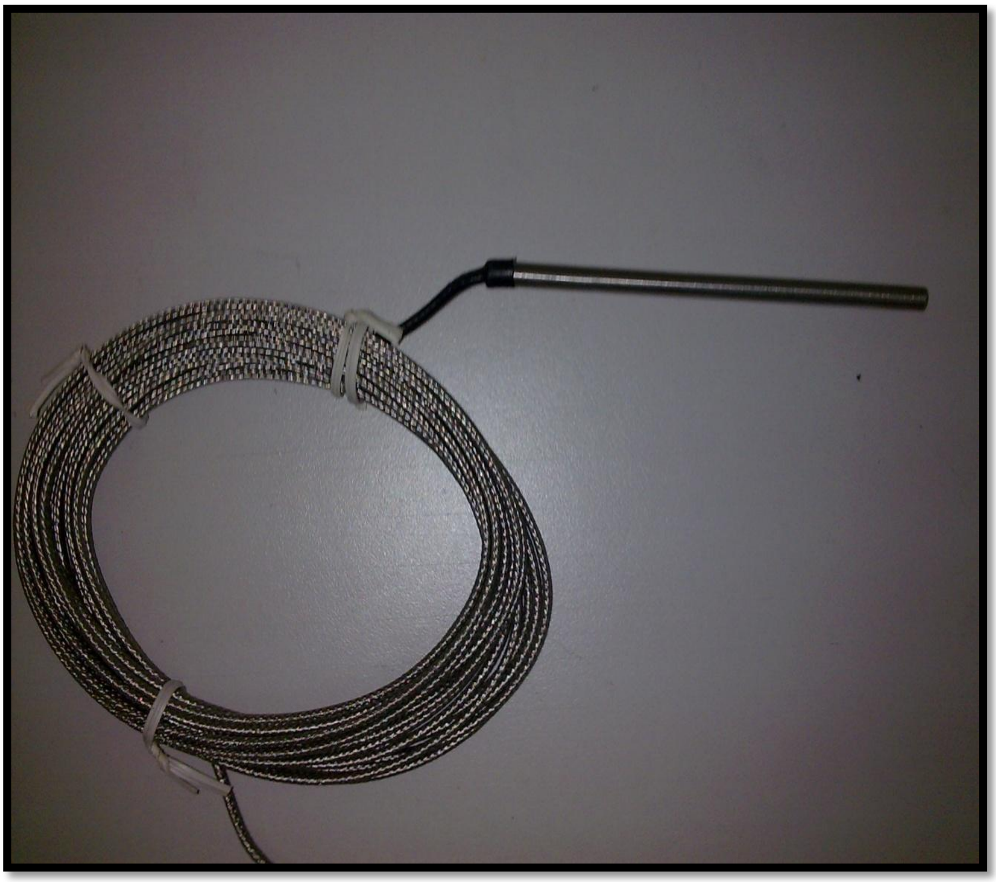


Figure 3.6 : Thermocouple used in this experiment to measure the temperature at exhaust, intake manifold, engine oil, and ambient

The specifications of the K-type thermocouple are shown in table 3.4 below.

Table 3.4 : K-type Thermocouple Specification

Wire Material	Chrome (+ve), Alumel (-ve)
Range	-328 to 2300 ^{o}F
ANSI Standard limits of Error	±4.0^{o}F or 0.75%

3.1.7 Gas Analyzer

Internal combustion engine will produce exhaust gases which is the combustion product. This exhaust gases consists of few hazardous gases such as Hydrocarbons (HC), Carbon Dioxide (CO_2), Carbon Monoxide (CO), Nitric Oxide (NO) and etc. The composition of the gases is measured in percentage (%) or ppm (parts per million) unit. To analyze the exhaust gases composition, a gas analyzer measuring device is used. This device can measure and calculate all the parameters and display the percentage. The model of Gas Analyzer that was used for this experiment is Kane Automotive Gas Analyzer. This analyzer has two mode, it can analyze the gasoline and diesel exhaust gas. Other than HC, CO, $CO_{2,}$ and NO, this device can also read the air fuel ratio.

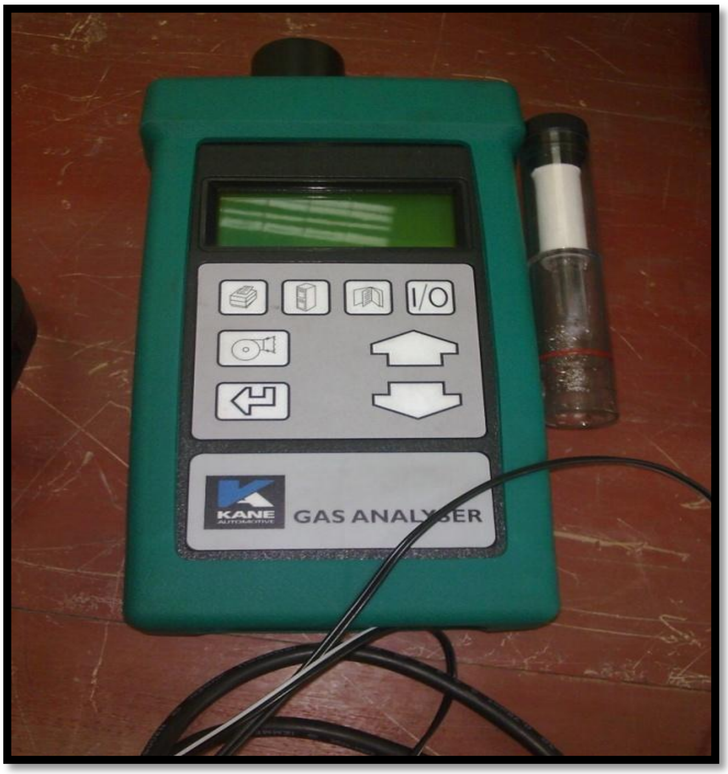


Figure 3.7 : Gas Analyzer used to measure the quantity of CO2, NOx, O2, and HC emissions in exhaust gas

All the specifications of the Gas Analyzer are shown in Table 3.5, which consists of parameter, resolution, accuracy and then range of parameter.

Table 3.5 : Specification of Gas Analyzer

Parameter	Resolution	Accuracy	Range
Carbon Monoxide	0.01%	±5% of reading	0-10%
Oxygen	0.01%	±5% of reading	0-21%
Hydrocarbon	1ppm	±5% of reading	0-5000ppm
Carbon Dioxide	0.1%	±5% of reading	0-16%
Nitric Oxide	1ppm	±4% of reading	0-5000ppm

3.1.8 Data Acquisition System

Data Acquisition system is used to obtain all the temperature readings and the cylinder pressure readings from the engine while conducting the experiment. The data acquisition system that has been used for the experiment is Dewetron model Dewe-800 series.

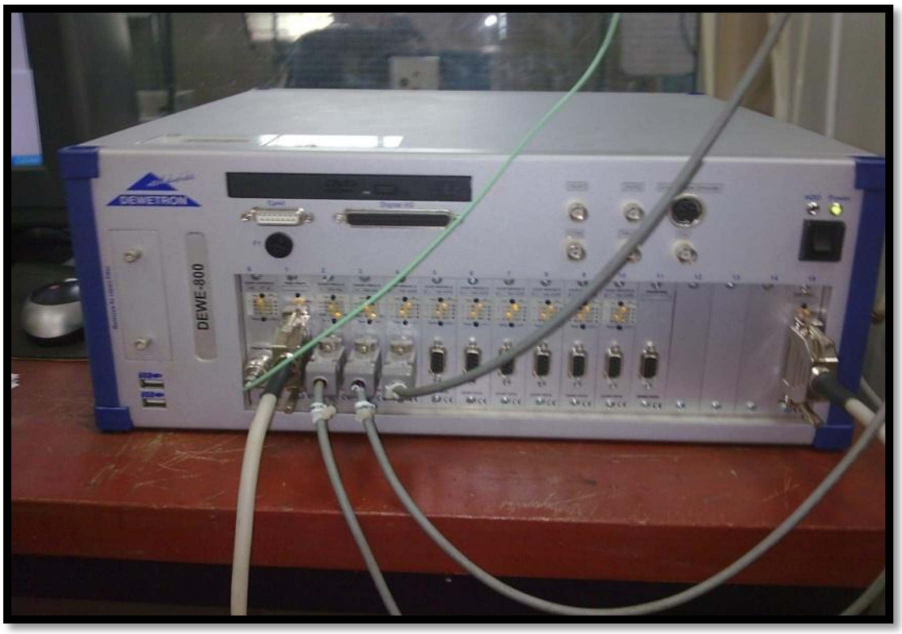


Figure 3.8 : Dewetron (Dewe-800) used to convert data from pressure sensor and thermocouple to digital number then display it at the desktop screen

The specifications of the Dewe-800 series are shown Table 3.6 below and the actual figure is shown in Figure 3.8 above.

Table 3.6 : The data acquisition system specification that used for the experiment.

Model	Dewe-800 Series
Hard Disk	250 GB
Data Throughout	Typ. 70MB/s^2
Processor	Intel Celeron M1.8 GHz
RAM	2 GB
Operating System	Window XP
Total PCI Slot	2 Full / 1 Half-Length
Slot for DAQ and PAD Module	16
Software	DeweSoft

3.2 Experimental Facilities

The Mitsubishi 4D68 diesel engine is used to conduct this experiment. The 4D68 engine is a diesel engine with water cooled, direct injection, EGR, 4-cylinder and 4-stroke. This engine is tested using diesel and biodiesel B5 methyl ester. The 150kW Eddy-Current dynamometer is used to apply load to the engine, then the torque of engine is displayed. To measure the percentage of carbon monoxide, nitrogen oxide, oxygen, and carbon dioxide in the exhaust smoke, the gas analyzer is used. The portable gaz analyzer is used for this experiment.

3.3 Experimental Procedure

The experiment is started with relative humidity checking. After that, all the instruments and experiment utilities like gas analyzer, engine cooling system, dynamometer cooling system, pressure transducer cooling system are ensured to be ready. Then, the engine is started and then left in idling mode. Then experiment is conducted after running the engine until it reaches steady state, with oil temperature at 99^{o}C and cooling water temperature 100^{o}C. The experimental work started with an investigation of the engine running on diesel fuel at various speeds of 1000 rpm, 1500 rpm, 2000 rpm, and 2500 rpm under 25% load in order to determine the engine's operating characteristics and exhaust emission levels thus constituting the ‘baseline’ that is compared with the corresponding cases when using biodiesel fuel. At each speed, the engine was stabilized for 15 minute and then all the measured values are recorded at 5 minute interval. In each test, volumetric fuel consumption and exhaust regulated gas emissions, such as nitrogen oxides, carbon monoxide, total unburned hydrocarbons, exhaust gas temperatures, and torque are measured. The differences in the measured performance and exhaust emission parameters from the base line operation (diesel fuel) and biodiesel fuel are determined and compared (Saleh, 2009). The DeweSoft software is used as a data acquisition system to measure all the values such as exhaust, intake, ambient, and engine temperature. It also used to measure the in-cylinder pressure of the engine. The entire instrument that used is calibrated beforehand, so the data which is displayed is not wrong.

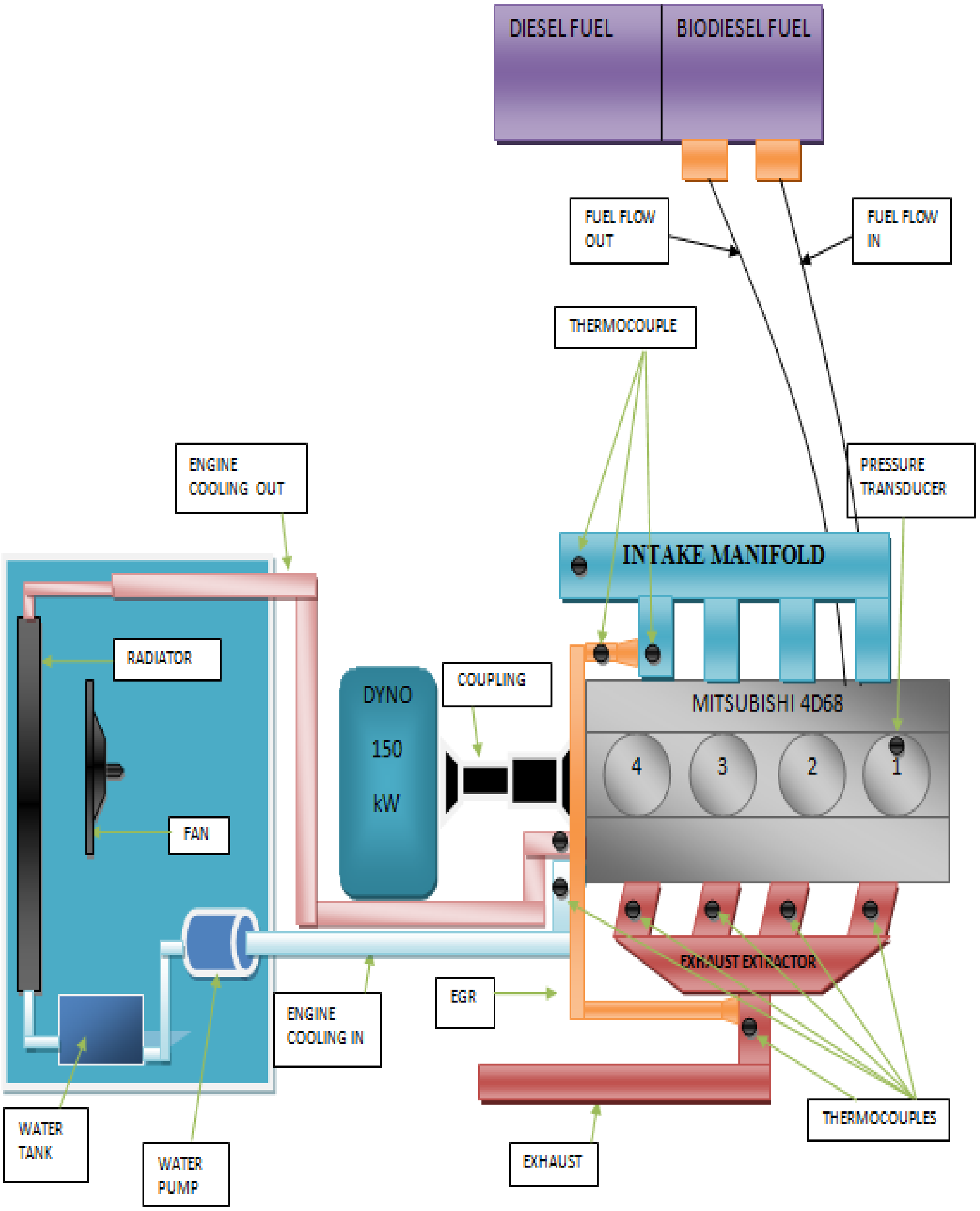


Figure 3.9 : Schematic Diagram of the engine for experimental setup

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

This chapter consist of the result and discussion of the experiment that have been conduct based on the standard of engine testing. The data which get from this experiment have been analyzed in the discussed. The results will be expressed in tables and graphs to provide the reader with a clearer view. The experimental result will then be analyzed and compared. Recommendations will be prepared for future improvements.

4.2 RESULT

4.2.1 Result of Emissions With EGR and Without EGR by Using Diesel and Biodiesel at 50% Throttle.

Graph below shows that the different of CO_2 emission when using EGR system and without using EGR. This experiment was carried out by constant 50% throttle and with increasing of engine speed. From the result, at the starting point of 1000 rpm, the quantity of CO_2 emissions for the engine which operated without EGR is higher than the emissions for the engine which operated with EGR, this is because at lower speed, the operating of EGR was recirculate some of the quantity of CO_2 back to the intake. Then starting at the engine speed of 2000 rpm to 3000 rpm the quantity of CO_2 emission for diesel engine which operated with EGR is higher than the quantity of diesel engine operated without EGR. This is because the air-fuel ratio of diesel engine which operated with EGR is richer than the air-fuel ratio of diesel fuel which operated without EGR. The increasing and decreasing of CO_2 emissions also are related to the decreasing and increasing of O_2 gas emissions.

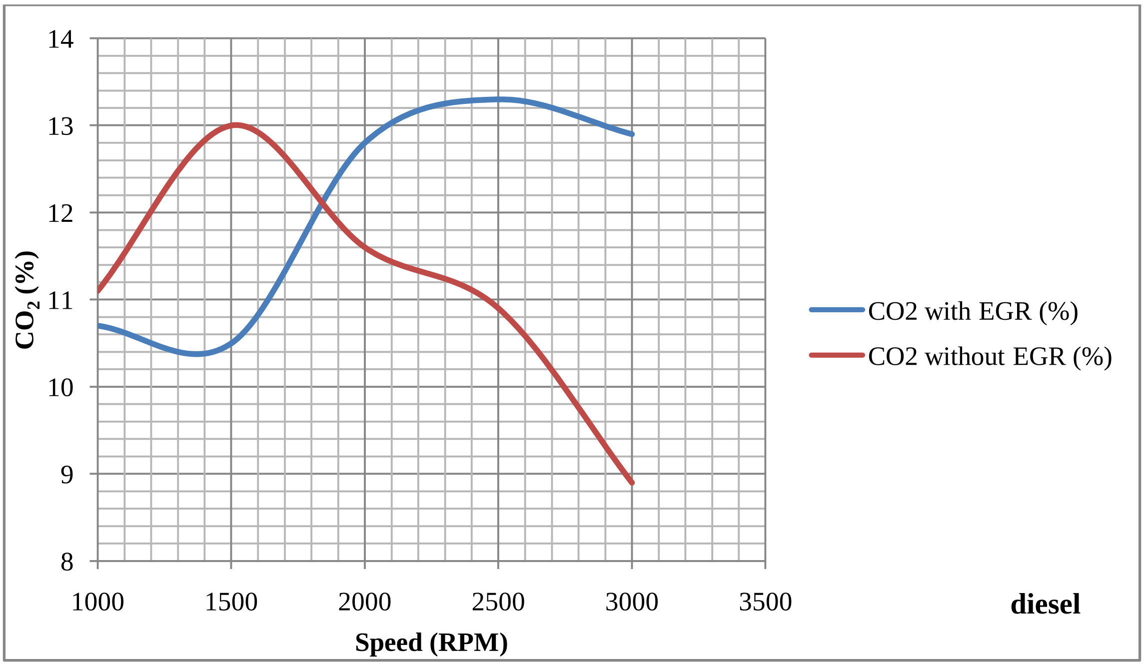


Figure 4.1 : Graph of different CO_2 emissions when using EGR and without using EGR, operating with diesel fuel

Figure 4.2 below shows the graph of different CO_2 emissions when using EGR and without using EGR, operating with biodiesel fuel. This experiment was carried out by constant 50% throttle and with increasing of engine speed. From the result, at the starting point of 1000 rpm, the quantity of CO_2 emissions for the engine which operated with EGR is higher than the emissions for the engine which operated without EGR, this is because the air-fuel-ratio mixture for the biodiesel fuel which operated with EGR is richer than the air-fuel-ratio mixture for biodiesel fuel which operated without EGR. Then because some of the quantity of CO_2 is reciculate into intake, at 1500 rpm to 2000 rpm of engine speed, the quantity of CO_2 for biodiesel fuel which operated without EGR is higher than quantity of CO_2 for biodiesel which operated with EGR. At the high engine speed of 2000 rpm to 3000 rpm the quantity of CO_2 emissions for the engine which operated with EGR is higher than the emissions for the engine which operated without EGR, this is because the air-fuel-ratio mixture for the biodiesel fuel which operated with EGR is richer than the air-fuel-ratio mixture for biodiesel fuel which operated without EGR. The increasing and decreasing of CO_2 emissions also are related to the decreasing and increasing of O_2 gas emissions.

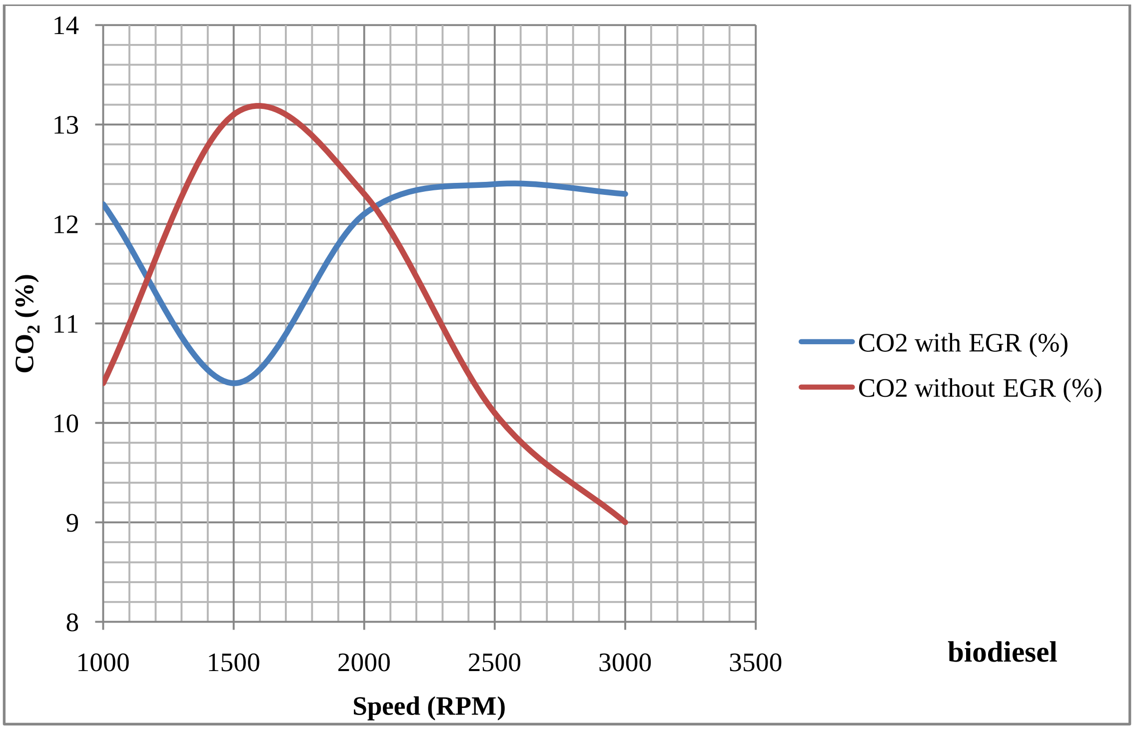


Figure 4.2 : Graph of different CO_2 emissions when using EGR and without using EGR, operating with biodiesel fuel

Graph below shows that the different of HC emissions when using EGR and without using EGR, operating with diesel fuel. Graph of HC emissions for diesel fuel which operating with EGR is increasing at 1000 rpm to 1500 rpm from 64 ppm to 242 ppm, this is because the air-fuel ratio at engine speed of 1500 rpm is rich than at engine speed of 1000 rpm. Then the air-fuel mixture become lean during the increasing of engine speed from 1500 rpm to 3000 rpm. For the graph of HC which operated without EGR, the quantity of HC emissions is constant 1 ppm from engine speed of 1000 rpm to 3000 rpm. From the result, the quantity of HC emissions for diesel fuel which operated with EGR is higher than the HC emissions for diesel fuel which operated without EGR. This is because, the air-fuel ratio for diesel fuel which operated with EGR is richer than the air-fuel-ratio mixture for diesel fuel which operated without EGR. HC emissions will increase when the oxygen content in the combustion chamber is lower, because there is not enough oxygen to burn the excess fuel.

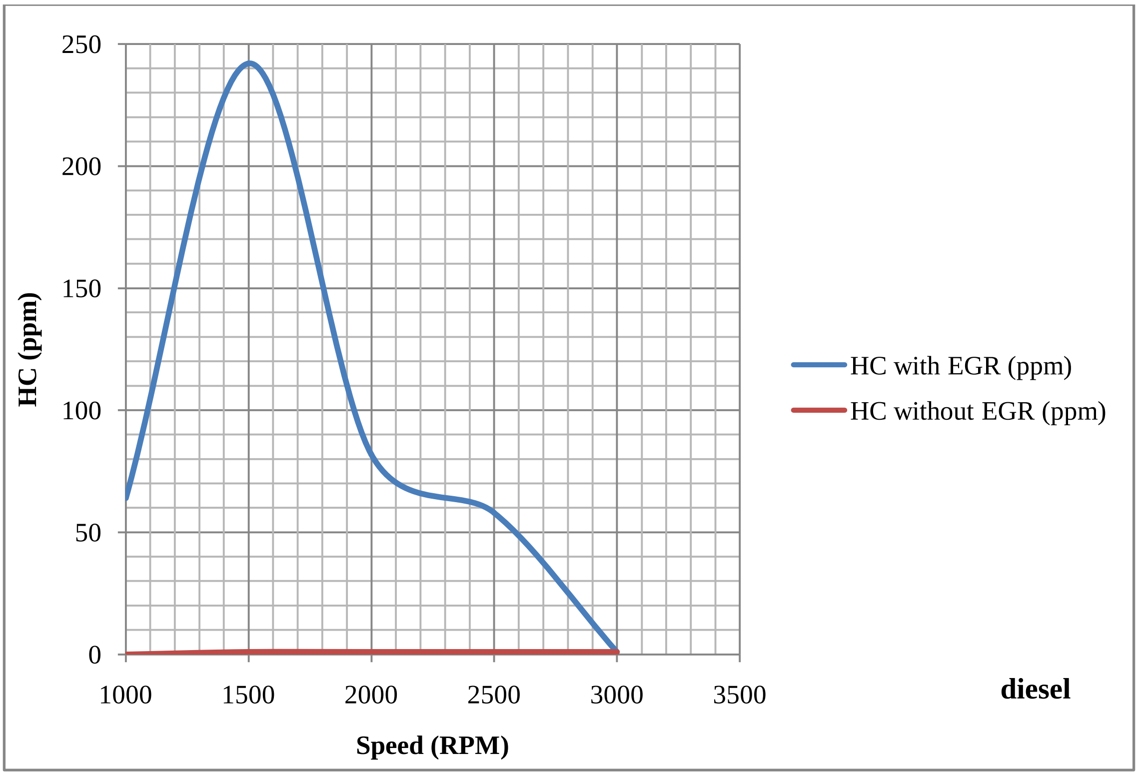


Figure 4.3 : Graph of different HC emissions when using EGR and without using EGR, operating with diesel fuel

Figure 4.4 below shows that the different of HC emissions when using EGR and without using EGR, operating with diesel fuel. Graph of HC emissions for diesel fuel which operating with EGR is increasing at 1000 rpm to 1500 rpm from 20 ppm to 251 ppm, this is because the air-fuel ratio at engine speed of 1500 rpm is rich than at engine speed of 1000 rpm. Then the air-fuel mixture become lean during the increasing of engine speed from 1500 rpm to 3000 rpm. For the graph of HC which operated without EGR, the quantity of HC emissions is constant 1 ppm from engine speed of 1000 rpm to 1500 rpm, then the quantity of HC increase to 15 ppm at engine speed of 2000 rpm and decrease back to 1 ppm at engine speed 2500 rpm, then increase to 6 ppm of HC at 3000 rpm. From the result, the quantity of HC emissions for diesel fuel which operated with EGR is higher than the HC emissions for diesel fuel which operated without EGR. This is because, the air-fuel ratio for diesel fuel which operated with EGR is richer than the air-fuel-ratio mixture for diesel fuel which operated without EGR.

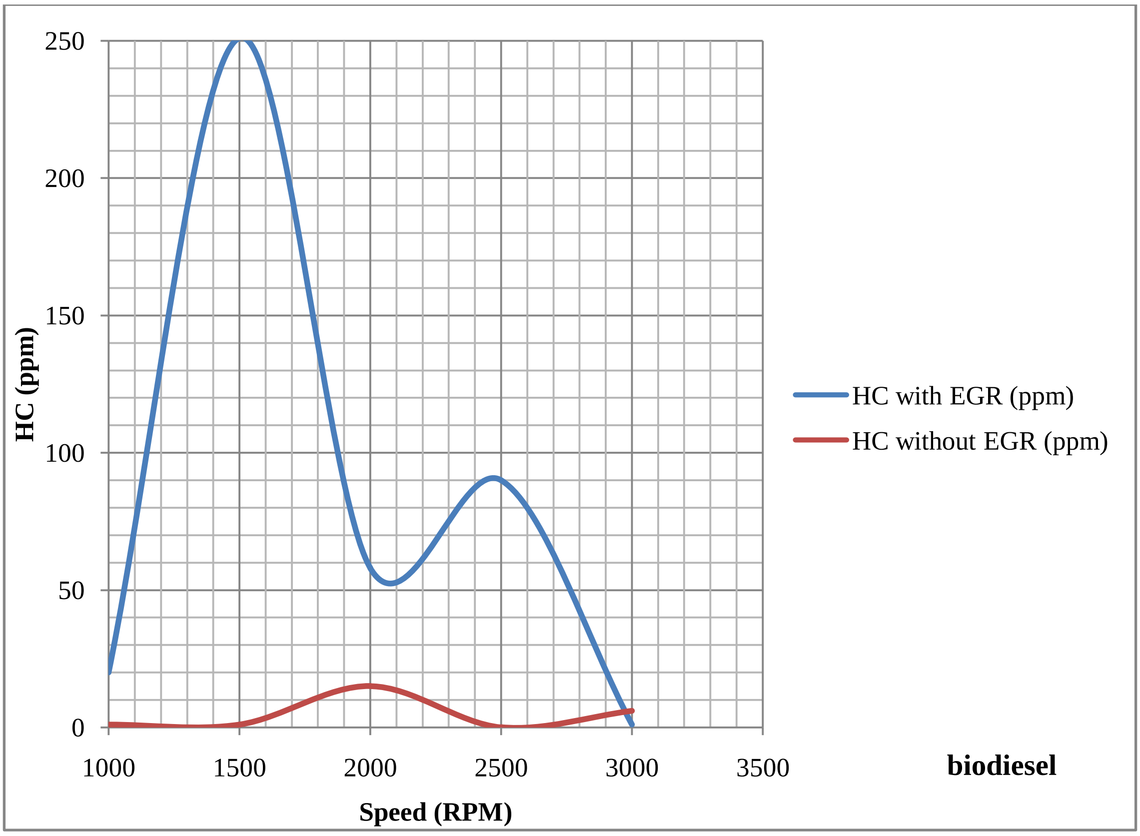


Figure 4.4 : Graph of different HC emissions when using EGR and without using EGR, operating with biodiesel fuel

The different of O_2 emissions when using EGR and without using EGR, operating with diesel fuel are shown below. From the result that was shown below, the quantity of O_2 emissions for the diesel fuel which operated with EGR is lower than the quantity of O_2 emissions for diesel fuel which operated without EGR, This is because when the engine operated with EGR, the CO_2 gas which reirculated from exhaust was replaced the amount of O_2 which enter from intake. So as a result, the quantity of O_2 for diesel fuel which operated with EGR is lower. For the diesel fuel which operated with EGR, at engine speed of 2500 rpm to 3000 rpm the quantity of O_2 emissions is increased. This is because, there is not enough fuel to burn the amount of O_2 which enter the combustion chamber. The fuel cannot be burned bcause lack of fuel due to higher engine speed with constant 50% throttle.

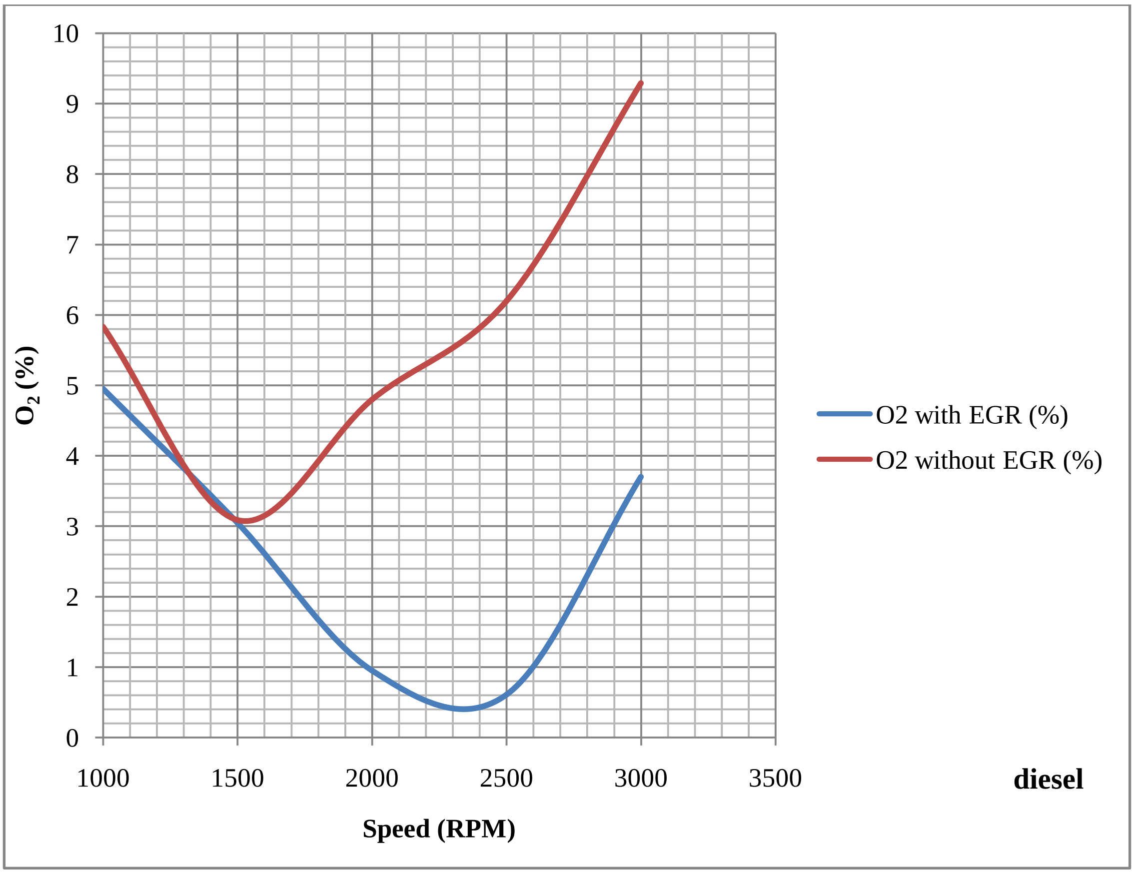


Figure 4.5 : Graph of different O_2 emissions when using EGR and without using EGR, operating with diesel fuel

Figure 4.6 below shows that the graph of different O_2 emissions when using EGR and without using EGR, operating with biodiesel fuel. From the engine speed of 1000 rpm to 1500 rpm, the quantity of O_2 emissions are reduced because of the higer of fuel supply during the low engine speed. Then due to the increasing of engine speed from 1500 rpm to 3000 rpm, the quantity of O_2 emissions also increase. This is because, the quantity of fuel supply into combustion chamber is reduce. This is same situation to the biodiesel fuel which using EGR, the quantity of O_2 increase at the engine speed of 2500 rpm to 3000 rpm. The re-circulated exhaust gas has effected the quantity of O2 emissions in exhaust gas for biodiesel fuel which operated with EGR, so the quantity of O2 emissions for biodiesel fuel which operated without EGR is higher than that.

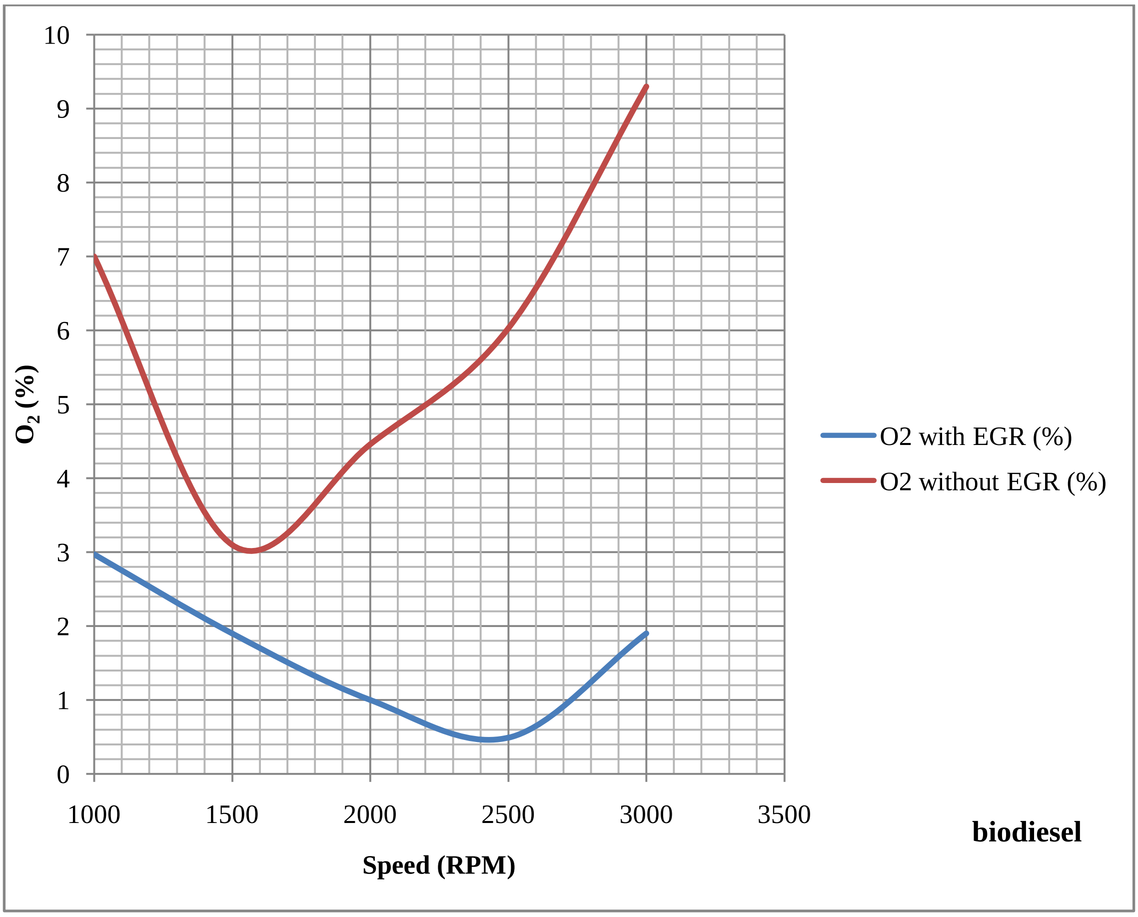


Figure 4.6 : Graph of different O_2 emissions when using EGR and without using EGR, operating with biodiesel fuel

Figure 4.7 shown the differences between with EGR and without EGR for NO_x emissions using diesel fuel with respect to engine speed. The quantity of NO_x emissions for diesel fuel which operated without EGR increased from engine speed of 1000 rpm to 2000 rpm. The increasing of NO_x emissions is because of the increasing of in-cylinder pressure and temperature. Then it reduces at engine speed 2500 rpm and increase back at engine speed 3000 rpm. NO_x emissions for diesel fuel which operated with EGR decrease at engine speed 1000 rpm to 1500 rpm and increased back at 2000 rpm. Then it increased back at 2500 rpm to 3000 rpm. The NO_x emissions for diesel fuel using EGR is lower than NO_x emissions of diesel fuel without using EGR. This is because the effect of the function of EGR system which is can reduce the NO_x emissions.

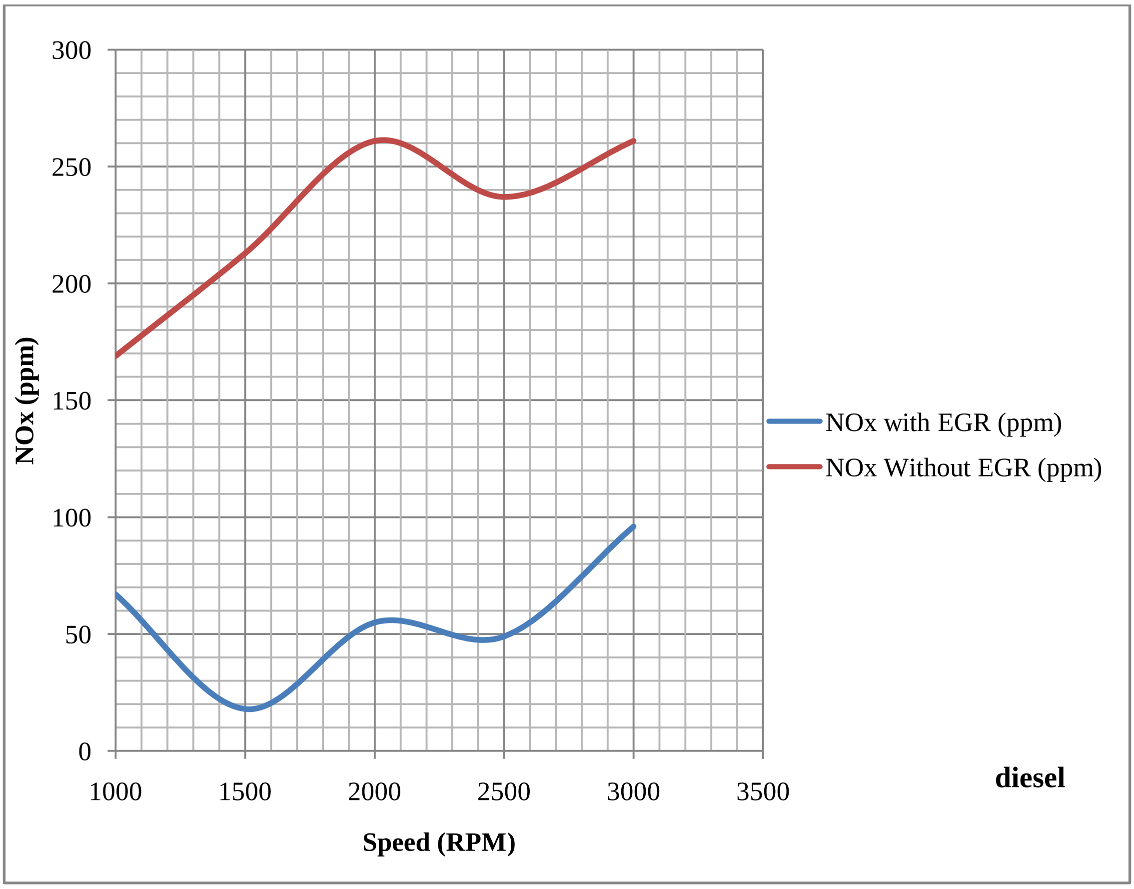


Figure 4.7 : Graph of different NO_x emissions when using EGR and without using EGR, operating with diesel fuel

Graph below shows the different of NO_x emissions when using EGR and without using EGR which operated with biodiesel fuel. The same result have occured for the biodiesel fuel which is the quantity of NO_x emissions produces for biodiesel fuel which operated with EGR is lower than NO_x emissions for biodiesel fuel which operated without EGR. NOx emissions for biodiesel fuel which operated with EGR increased back at the higher engine speed. The NOx emission increased because the EGR system does not work at higher engine speed due to higher engine speed.

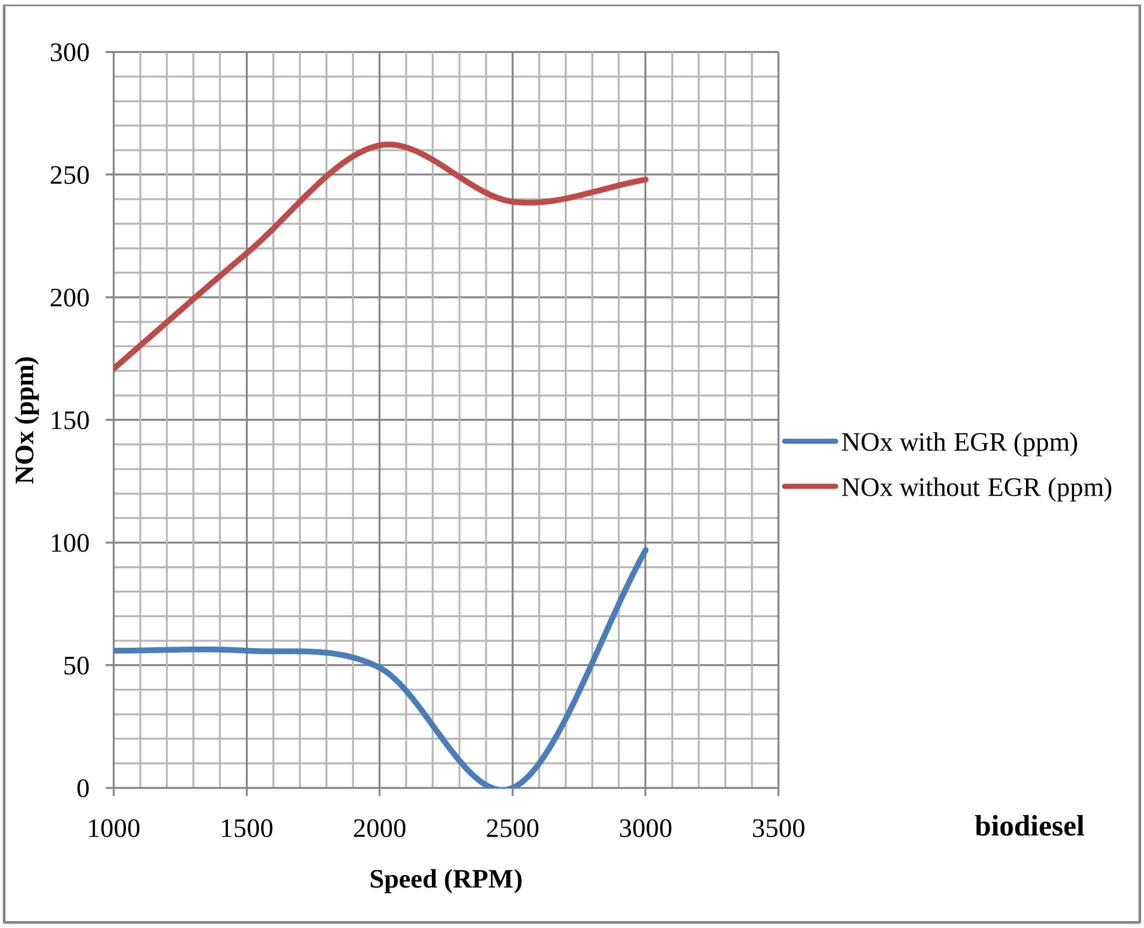


Figure 4.8 : Graph of different NO_x emissions when using EGR and without using EGR, operating with biodiesel fuel

4.2.2 Result of Diesel and Biodiesel fuel Performnace With EGR and Without EGR at 50% Throttle

Table 4.1 above shown that the torque and power (kW) for the diesel engine which operated with diesel fuel at various of engine speed (RPM). This diesel fuel is operated with two different conditions which is with EGR and without EGR.

Table 4.1 : Torque and Engine Performance Operating With Diesel fuel

RPM	**WITH EGR**		**WITHOUT EGR**	
	TORQUE (Nm)	**POWER (kW)**	**TORQUE (Nm)**	**POWER (kW)**
1000	89	9.32	102	10.68
1500	74	11.62	113	17.75
2000	76	15.92	105	22
2500	63	16.49	94	24.61
3000	59	18.54	75	23.56

Table 4.2 above shown that the torque and power (kW) for the diesel engine which operated with biodiesel fuel at various of engine speed (RPM). This biodiesel fuel is operated with two different conditions which is with EGR and without EGR.

Table 4.2 : Torque and Engine Performance Operating With Biodiesel fuel

RPM	**WITH EGR**		**WITHOUT EGR**	
	TORQUE (Nm)	**POWER (kW)**	**TORQUE (Nm)**	**POWER (kW)**
1000	91	9.53	104	10.89
1500	74	11.62	115	18.06
2000	77	16.13	105	21.99
2500	73	16.49	94	24.61
3000	58	18.22	69	21.66

Below are shown some sample calculation of relation between torque and performance. The performance and torque graph are shown at figure 4.10 below

$\dot{w}b = 2\pi N\tau$ where, (4.1)

$\dot{w}b$ - brake power (kW)
N - rev. per second = N/60
τ - torque (N)

$$\dot{w}b = 2\pi N\tau$$
$$= 2\pi(1000\text{rpm}/60)(89\text{Nm})$$
$$= 9.32\text{kW}$$

Table 4.3 : Result for AFR and Equivalent Ratio Operating With Diesel Fuel

RPM	WITH EGR		WITHOUR EGR	
	AFR	**EQUIVALENT RATIO**	**AFR**	**EQUIVALENT RATIO**
1000	18.05	0.8	19.93	0.73
1500	14.99	0.97	17.06	0.85
2000	14.14	1.03	18.8	0.77
2500	16.02	0.91	20.5	0.71
3000	17.55	0.83	25.5	0.57

Table 4.3 above shown that the list result of air-fuel-ratio and calculated equvalent ratio based on the result of air-fuel-ratio. The result is collected during the engine tested at various engine speed, constant 50% throttle, with diesel fuel and with two opeating condition which is with EGR and without EGR. the equivalent ratio values is calculated by using equation as shown below :

$\emptyset = (AFR)_{stoic} / (AFR)_{act}$ (4.2)

Table 4.4 above shown some of the properties of fuel, this table is refer from the text book Engineering Fundamental of the Internal Combustion Engine by Willard W. Pulkrabek. From this table the value of $(AFR)_{stoic}$ for light diesel fuel is 14.5.

Table 4.4 : Properties of Fuel Table

Fuel		Molecular Weight	HeatingValue		stoichiometric		Octane Number	
			HHV (kJ/kg)	LHV (kJ/kg)	AF	FA	MON	RON
Gasoline	C8H15	111	47300	43000	14.6	0.068	80-91	92-99
Light Diesel	C12.3H22.2	170	44800	42500	14.5	0.069	-	-
Heavy Diesel	C14.6H24.8	200	43800	41400	14.5	0.069	-	-

Regarding to this value, the equivalent ration are calculated as shown below :

$$\begin{aligned} \emptyset &= (AFR)_{stoic} / (AFR)_{act} \\ &= (14.5)_{stoic} / (18.05)_{act} \\ &= 0.8 \end{aligned}$$

The equivalent ratio is calculated to consider there is rich mixture or lean mixture of diesel fuel. When :

$\emptyset < 1$ running lean, oxygen in exhaust (4.3)

$\emptyset > 1$ running rich, CO and fuel in exhaust (4.4)

$\emptyset = 1$ stoichiometric, maximum energy release from fuel (4.5)

Figure 4.9 shown that the graph of exhaust 1 (see Figure 3.9) temperature in the increasing of engine speed at constant 50% throttle and operating with EGR. The temperature for used of B5 methyl ester fuel is large than used of diesel fuel. This is because the compression for the B5 methyl ester fuel is more than diesel fuel as shown in Figure 4.10.

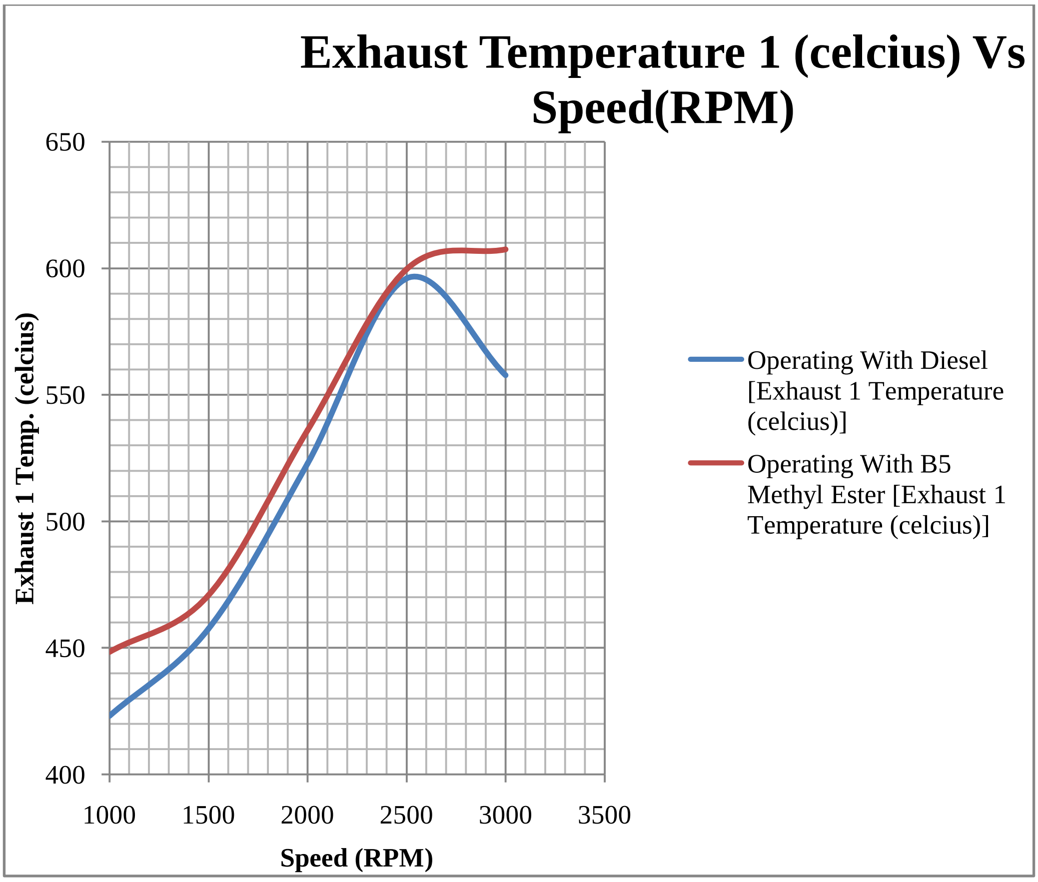


Figure 4.9 : Graph of Exhaust 1 Temperature With EGR against engine speed

Figure 4.10 shown that the graph of average in-cylinder pressure against engine speed of two different fuels, which is between diesel and biodiesel at constant 50% throttle with EGR operating system. Mostly the in cylinder pressure of the biodiesel fuel is higher than the diesel fuel at every increasing of the engine speed, but only at the half transition of engine speed from 2500 rpm to 3000 rpm, the biodiesel fuel in-cylinder pressure drop below than in-cylinder pressure of diesel fuel.

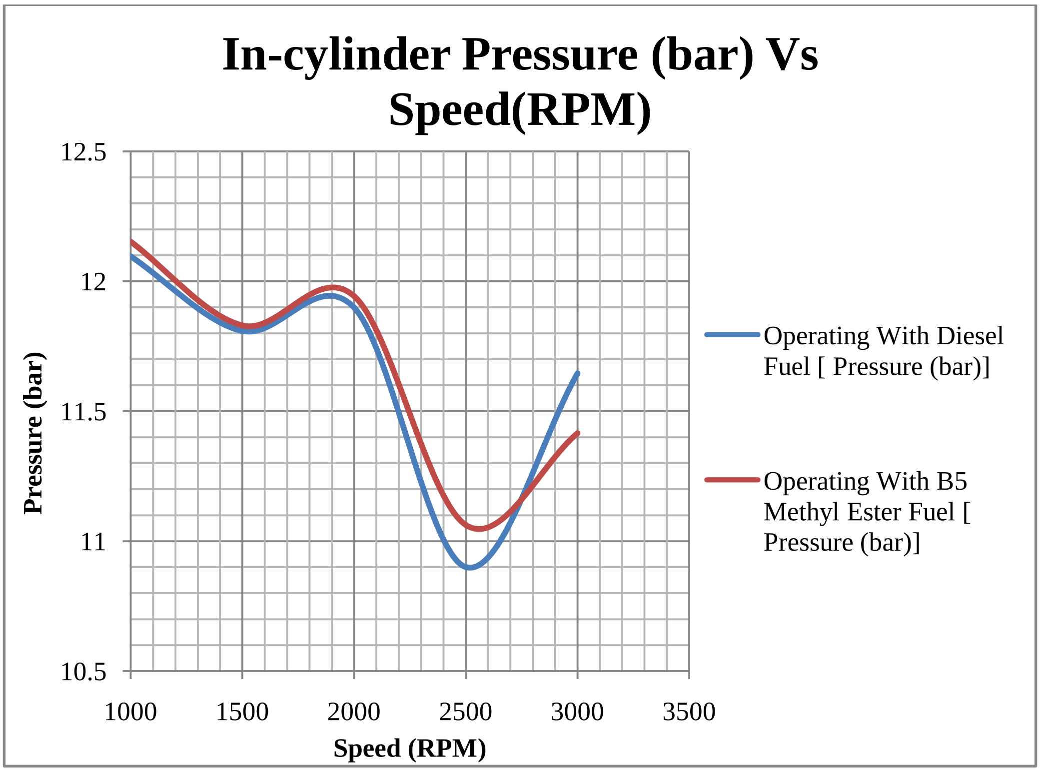


Figure 4.10 : Graph of Average Pressure against Engine Speed With EGR System

Figure 4.11 above shown that the graph of average in-cylinder pressure vs engine speed of two different fuel, which is between diesel and B5 methyl ester at constant 50% throttle without EGR operating system and at the various engine speed. Based on the graph, the in-cylinder pressure for B5 methyl ester fuel is higer than diesel fuel. This situation happen because based on the graph of temperature in Figure 4.9 shown for every increasing and decreasing of the temperature, the in-cylinder also same as temperature graph. The pressure increase because of combustion in cylinder increase and the evidance is temperature increase.

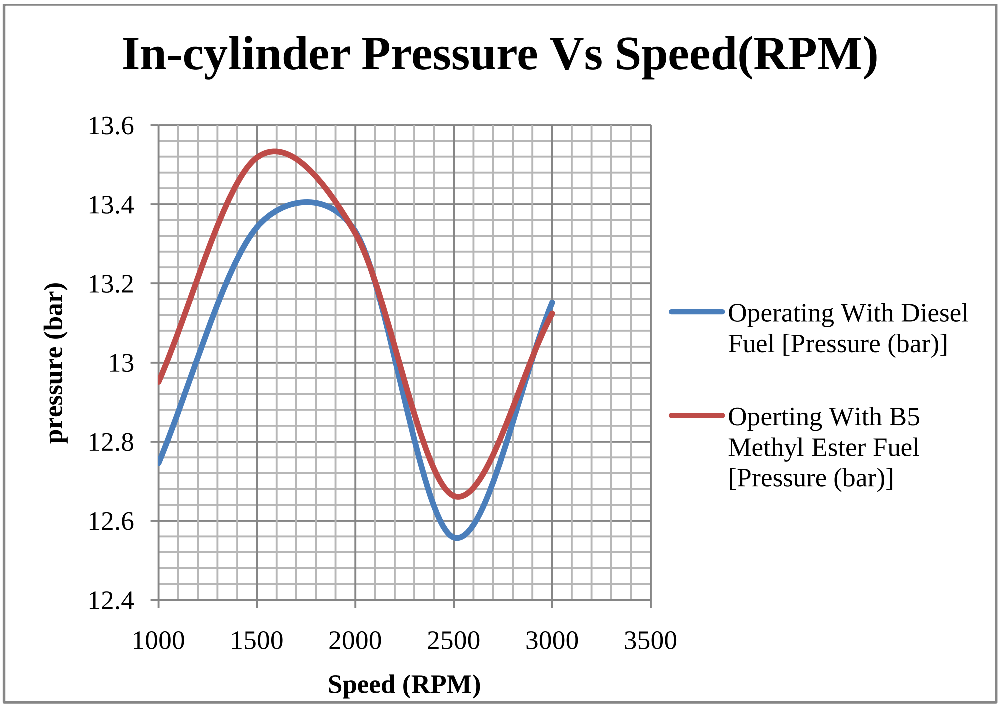


Figure 4.11 : Graph of Average Pressure against Engine Speed Without EGR System

Figure 4.12 shown that the graph of performance (kW) and torque (Nm) of the mitsubishi diesel engine which operated with the diesel fuel and with EGR at constant throttle 50% and at various engine speed (RPM). The graph is ploted based on the result at the Table 4.1. At the first point when the torque at 1000 rpm to 1500 rpm,the torque of the engine is decreasing due to the effect of EGR at lower engine speed, because of inteference from the recirculation gas, which causes incomplete combustion. Then the torque is increase as the engine speed increase, then decrease back as engine speed increase further as shown in Figure 4.12. This is because during the increasing of engine speed the torque is no longer needed. The performance of the engine is increase due to the increasing of the engine speed.

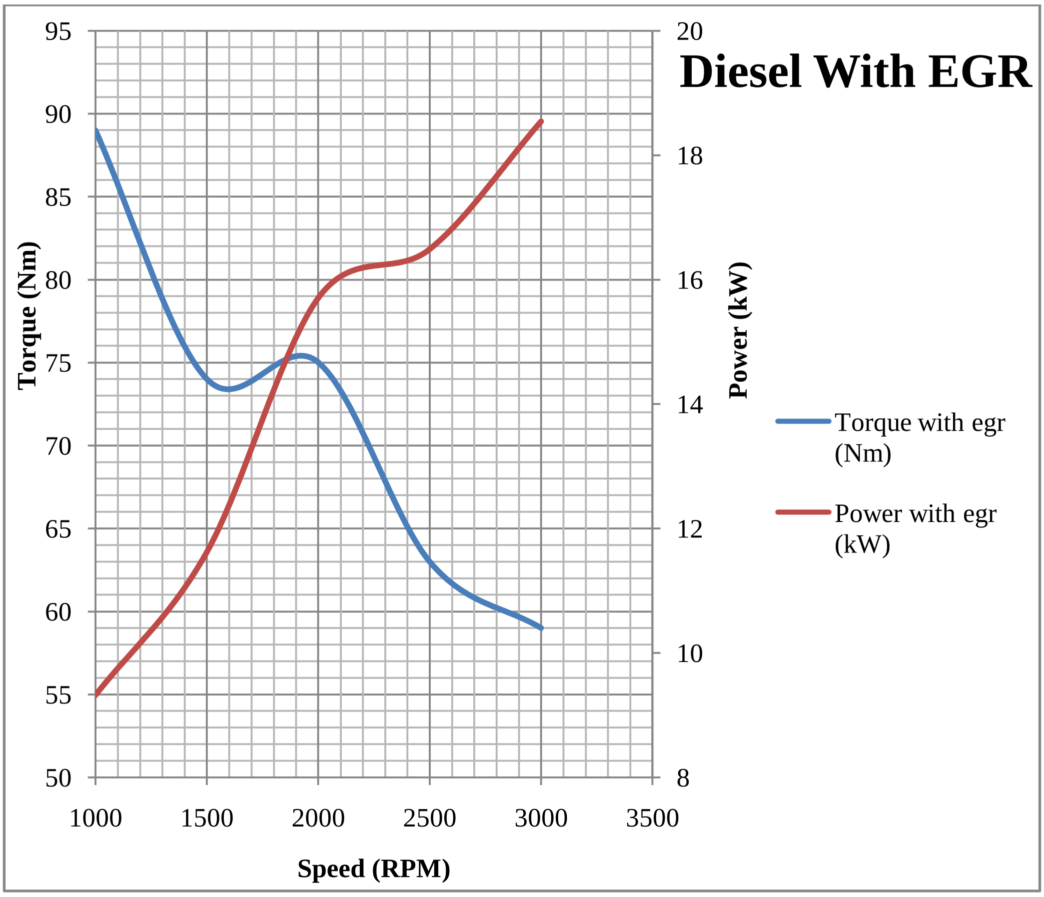


Figure 4.12 : Performance and Torque Graph of Operating Engine With Diesel Fuel against Engine Speed

Based on the Figure 4.13, the performance and torque curve for B5 methyl ester fuel. At the first point when the engine speed is at 1000 rpm the torque of the engine is 91 Nm. At engine speed 3000 rpm the value of torque is 53 Nm. This is the smallest torque between at the others engine speed, the value is smallest because at this engine speed, the fuel mixture is leanest. That mean the ratio of oxygen is more than fuel. The curve of performance is increase because of decreasing of the torque curve. When the torque is decreasing, at the high engine speed, the performance of the engine is increase because of the lower load to the engine.

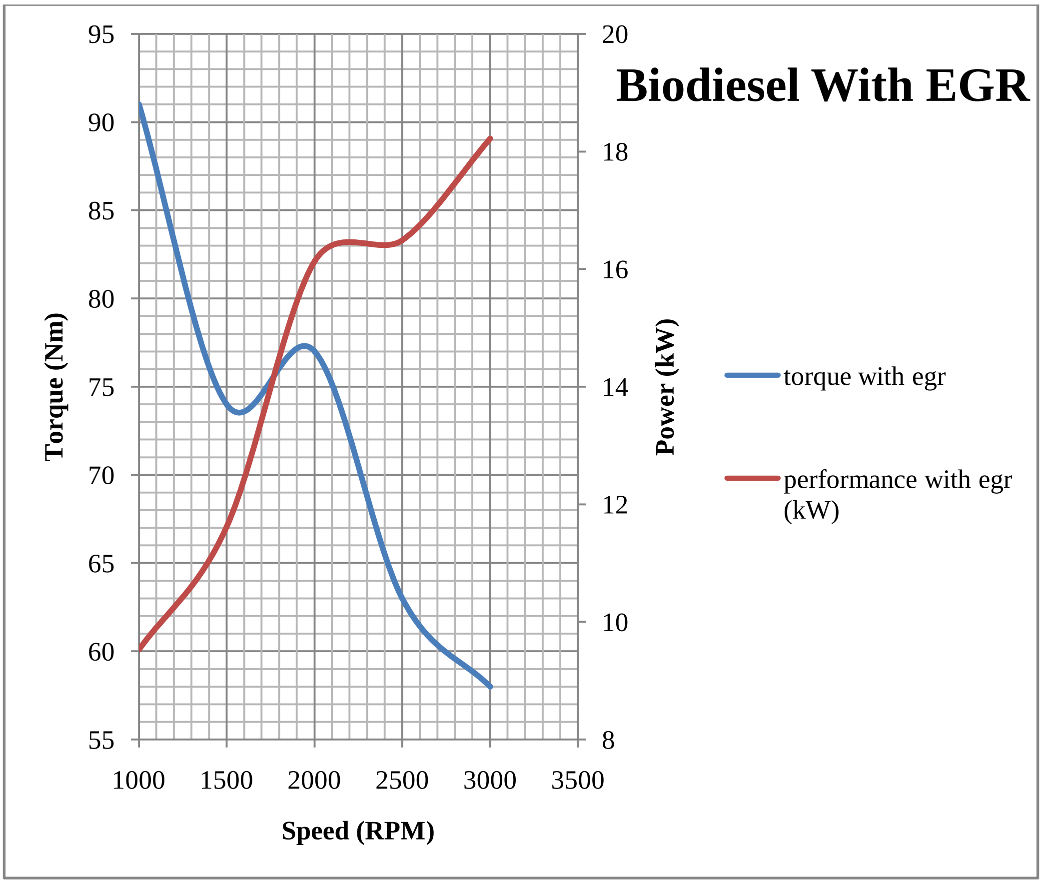


Figure 4.13 : Performance and Torque Graph of Operating Engine With Biodiesel Fuel against Engine Speed

Figure 4.14 and Figure 4.15 shown that the performance graph of diesel engine which is tested for different type of fuel. Both are diesel and B5 methyl ester palm oil biodiesel fuel. At lower speed, torque increase as the engine speed increase, reaches the maximum and then, as the engine speed increases further, torque decrease as shown in Figure 4.14 and 4.15. The torque decrease because the engine is not supplied with enough fuel at high engine speed due to the constant 50% throttle. At 2500 rpm, the different between this two fuel about the performance of engine is, the B5 methyl ester produced highest maximum power than diesel fuel related to constant 50% throttle. Although there is differentat power and torque betwen this two fuel, there are so small and not give high effect to the engine. Maximum torque was obtained at 1500 rpm for each kind of fuel. At 1500 rpm, power and torque of the diesel fuel and B5 methyl ester fuels were almost imperceptible. At higher speeds, the torque delivered with B5 methyl ester was higher approximately 2 Nm on average than the torque delivered by diesel fuel. The significant different is when the engine is operated with same fuel but with using EGR or without using EGR. compared the Figure 4.14 and 4.15, the power produce without using the EGR is more than the power produces with EGR. So EGR is efficient in reducing the NOx but not efficient for performance of the engine, because when using EGR, the engine power will drop than without using EGR.

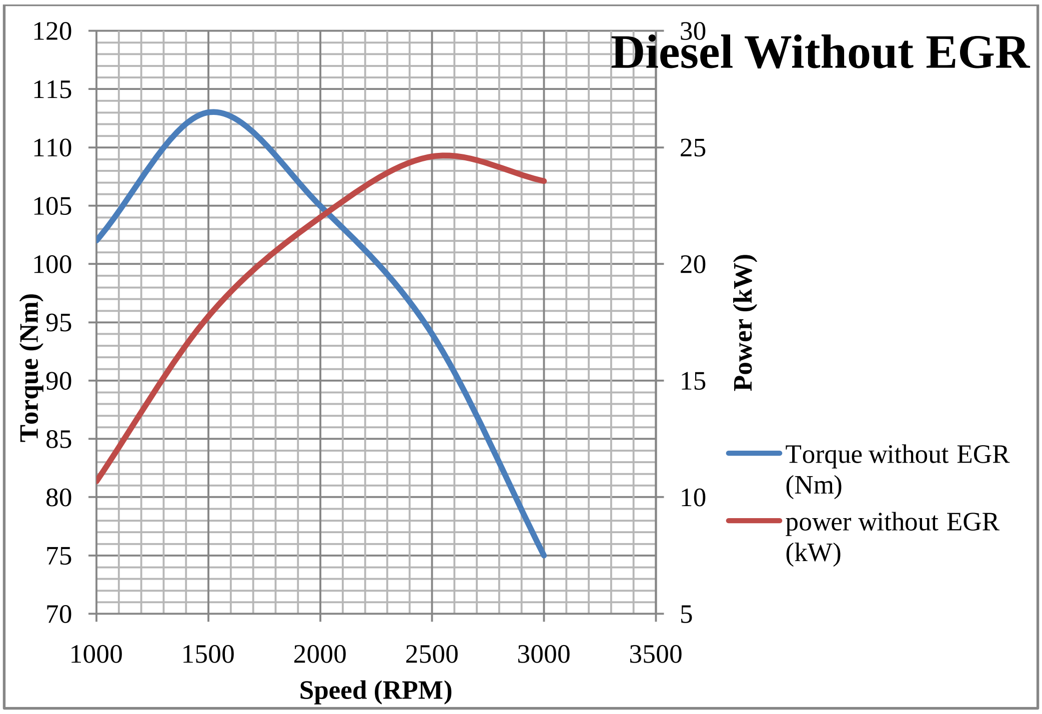


Figure 4.14 : Performance and Torque Graph of Operating Engine Without Diesel Fuel against Engine Speed

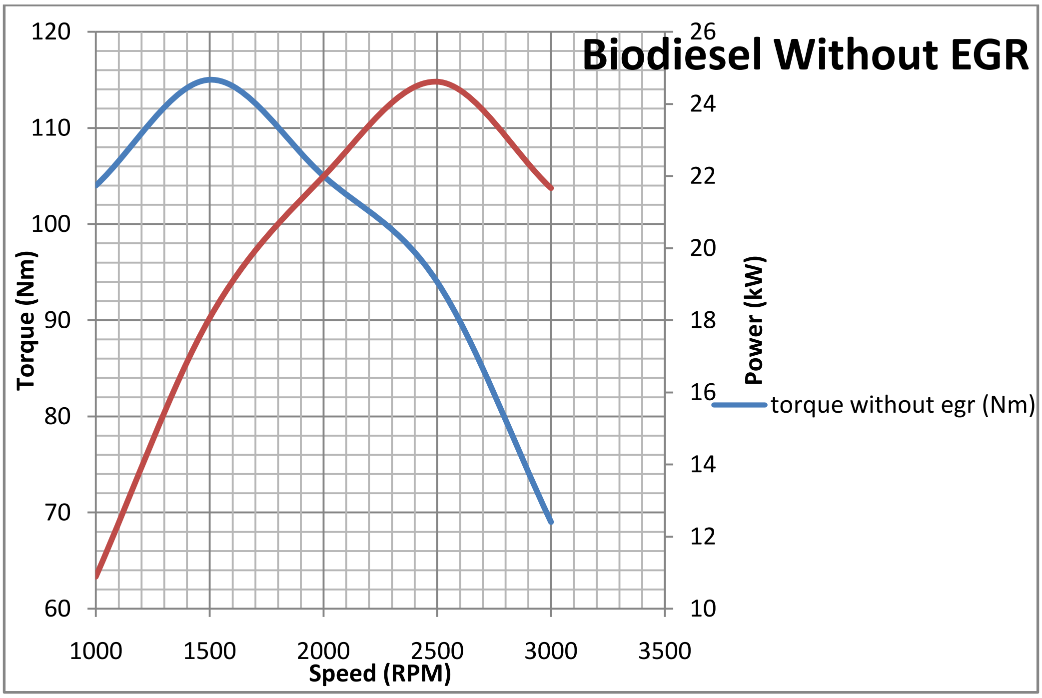


Figure 4.15 : Performance and Torque Graph of Operating Engine Without biodiesel Fuel against Engine Speed

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 INTRODUCTION

This chapter consists of the summary of regarding the whole research. It concludes all the outcomes, observation of results and analysis, and discussion throughout the experiment. Recommendations will also be given on improving future work and studies.

5.2 CONCLUSION

From the experimental result, it can be concluded that the exhaust gas recirculation gave a positive result towards the reduction of NOx in the gas emissions. However, in terms of the engine performance, the use of EGR provides a negative impact. Compared the engine operating without exhaust gas recirculation, the performance of engine using exhaust gas recirculation is poorer. Despite that, the purpose of using EGR to reduce the NOx in gas emission is successfully achieved and therefore, this system can be adopted to keep the environment from being less polluted. Based on the different fuels that are used in the experiment which are biodiesel and diesel, it is observed that the performance of a diesel engine is higher when using biodiesel than using diesel. Either with EGR or without EGR, the biodiesel fuel gives the best performance to diesel engines. However, biodiesel fuel produces more NOx gas compared to diesel fuel. But the difference in NOx production is minor because biodiesel B5 palm oil methyl ester fuel only consists of 5% biodiesel and 95% diesel.

5.2 RECOMMENDATION

1. Use catalyst to reduce the emission of NOx. For example, a copper catalyst is used in the presence of CO since when the value of CO is increase, the NOx emission will be reduced.
2. Use water injection. It has been observed that the specific fuel consumption decreases by a few percent at medium water injection rate. Attempts have been made to use water as a device for controlling the NOx.

REFFERENCE

A, K., G. r. M, et al. 2007. *Biodiesel production from tall oil with Synthesized Mn and Ni based additives: Effects of the additives on fuel Consumption and emissions.* Fuel **86**(1139–43).

A, P., s. M, et al. 2002. *A new approach to boost pressure and EGR rate control development for HD truck engines with VGT.* SAE paper(2002-01-0965; SAE International Congress and Exposition).

Abd-Alla, G. H., H. A. Soliman, et al. 2001. *Effects of diluents and intake air temperature in exhaust gas recirculation of an indirect injection dual fuel engine.* Energy Conversion Management **Vol.42**: 1033-1045.

Agarwal, D., S. Sinha, et al. 2006. *Experimental investigation of control of NOx emissions in biodiesel – fueled compression ignition engine.* Renewable Energy **Vol. 31**: 2356-2369.

AM, K., J. JH, et al. 1988. *A study of the effect of exhaust gas reciculation on heavy-duty Diesel engine emissions*. SAE paper

AS, R., J. S, et al. 2006. *Theoretical modeling and experimental studies on biodiesel-fueled engine.* Renewable Energy **31**(1813-26).

BormanGL and GaglandKW 1998. *Combustion engineering*. WCB/McGraw-Hill.

C, C., E. A, et al. 2006. *Performance and exhaust emissions of a biodiesel engine.* Applied Energy **83**(594-605).

Cengel, Y. A. and M. A. Boles 2007. *Thermodynamics An Engineering Approach Sixth Edition (SI Units)* .

Dampsey, p. 2008. *Troubleshooting and repairing diesel engine* . **4th edition**: 9-21.

DILWORTH, R. L., W. NATIONS, et al. 2003. *AIR POLLUTION CONTROL SYSTEMS FOR BOILER AND INCINERATORS* . **TM 5-815-1/AFR 19-6**.

Flagan, R. C. and J. H. Seinfeld 1988. *fundamental of air pollution in engineering*. 2-3.

Hountalas, D. T., G. C. Mavropoulos, et al. 2008. *Effect of exhaust gas recirculation (EGR) temperature for various EGR rates on heavy duty DI diesel performance and emissions.* Energy Conversion Management **Vol. 33**: 272-283.

JF, R. and S. MA 2006. *Emissions and power of a diesel engine fueled with crude and refinedd biodiesel* from salmon oil. Fuel **85**(174-7).

K. Rajan, K. R. S. *Effect of Exhaust Gas Recirculation (EGR) on the Performance and Emission Characteristic of Diesel Engine with Sunflower Oil Methyl Ester.*

M, C. and Y. F 2007. *The use of hazelnut oil as a fuel in pre-chamber diesel engine.* Applied Thermal Engineering **27**(63-7).

MD, K., T. JF, et al. 2004. *Assessment of corrosivity associated with exhaust gas recirculation in a heavy-duty diesel engine*. SAE paper(2004-05-660).

Mohamed, S. Y. E. 2003. *Effect of exhaust gas recirculation on some combustion characteristics of duel fuel engine*. Energy Conversion Management **Vol.44**: 707-721.

N, N., A. S, et al. 2006. *Improvement of engine emissions with conventional diesel fuel and diesel-biodiesel blends.* Bioresource Technology **97**(372-8).

P, Z., A. H, et al. 1988. *Cooled EGR—a key technology for future efficient HD Diesels*. SAE paper

Saleh, H. E. 2009. *Effect of exhaust gas recirculation on diesel engine nitrogen oxide reduction operating with jojoba methyl ester.* Renewable Energy **34**(2178-2186)

Santoh, K., L. Zhang, et al. 1997. *Relationship between NOx and SM emissions from DI diesel engine with EGR.* Society of Automotive engineers of Japan **Vol.18**: 369-375.

T, j., A. D, et al. 2003. *The impact of exhaust gas reciculation on performance and emissions of a heavy duty diesel engine.* SAE paper(2003-01-1068).

Tsolakis, A., A. Megaritis, et al. 2007. *Engine performance and emissions of a diesel engine operating on diesel-RME (rapeseed methyl ester) blends with EGR (exhaust gas recirculation).* Energy Conversion Management **Vol. 32**: 2072-2080.

Tsolakis.A and Megaritis.A 2007. *Exhaust gas reforming of rapeseed methyl ester for reduced exhaust emissions of CI engines.* Biomass & Bioenergy **Vol.27** 493-505.

vijayalakshmi, m. devi, et al. 2007. *fuels and biofuels*. 87-265.

Y, H. and B. YD 2003. *Study on rapeseed oil as alternative fuel for a single-cylinder diesel engine.* Renewable Energy **28**.

Yoshimoto, Y., M. Onodera, et al. 1999. *Reduction of Nox and smoke and BSFC in a Diesel Engine Fueled by Biodiesel Emulsion with Used Frying oil.* SAE Transaction: 1913-1929.

Yoshimoto, Y. and H. Tamaki 2001. *Reduction of NOx and smoke emissions in a diesel engine fueled by biodiesel emulsion combined with EGR.* SAE Transaction: 467-475.

APPENDIX A

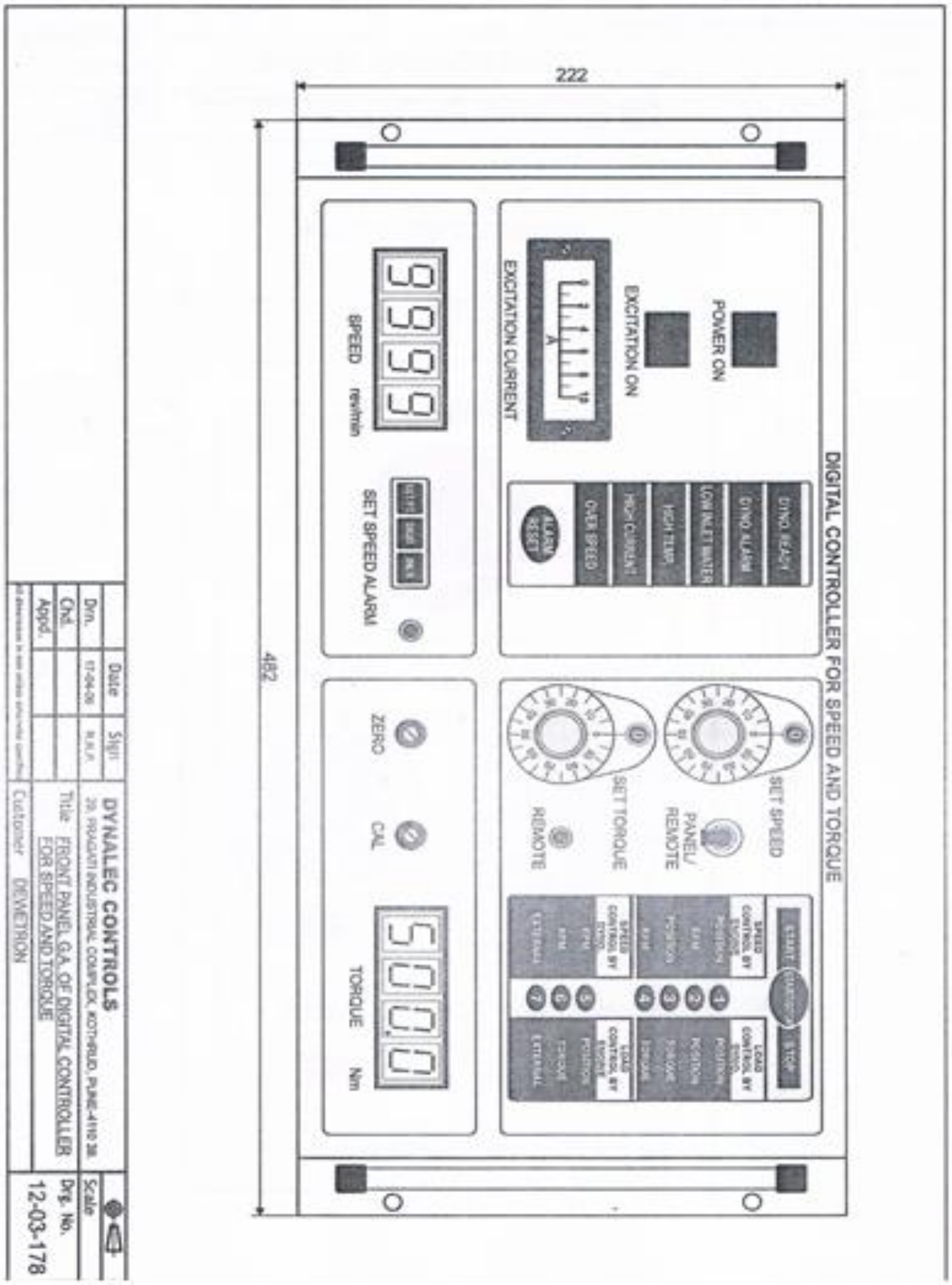
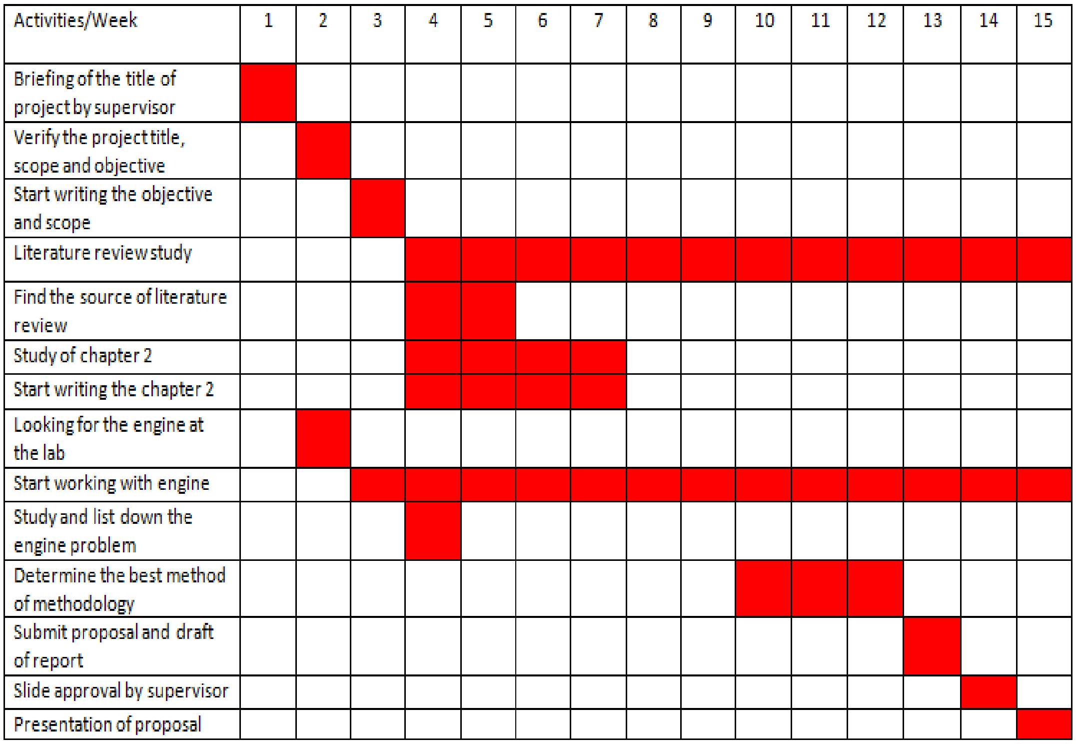


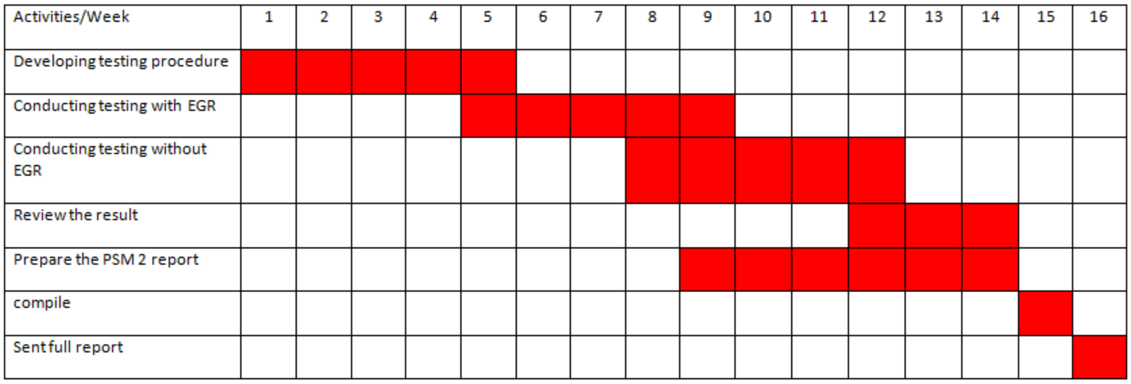
Figure A1 : Dynamometer controller button.

APPENDIX B



Activities/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Briefing of the title of project by supervisor	🟥														
Verify the project title, scope and objective		🟥													
Start writing the objective and scope			🟥												
Literature review study				🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥
Find the source of literature review				🟥	🟥										
Study of chapter 2				🟥	🟥	🟥	🟥								
Start writing the chapter 2				🟥	🟥	🟥	🟥								
Looking for the engine at the lab		🟥													
Start working with engine			🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥	🟥
Study and list down the engine problem				🟥											
Determine the best method of methodology										🟥	🟥	🟥			
Submit proposal and draft of report													🟥		
Slide approval by supervisor														🟥	
Presentation of proposal															🟥

Figure B1 : Gant Chart PSM 1



Activities/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Developing testing procedure	🟥	🟥	🟥	🟥	🟥											
Conducting testing with EGR					🟥	🟥	🟥	🟥	🟥							
Conducting testing without EGR								🟥	🟥	🟥	🟥	🟥				
Review the result												🟥	🟥	🟥		
Prepare the PSM 2 report									🟥	🟥	🟥	🟥	🟥	🟥		
compile															🟥	
Sent full report																🟥

Figure B2 : Gant Chart PSM