DESIGN OF EXHAUST GAS RECIRCULATION SYSTEM (EGR) FOR DIESEL ENGINE

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I certify that the project entitled “*Design of Exhaust Gas Recirculation System (EGR) For Diesel Engine”* is written by *Syahmi Bin Ahmad Ramly*. I have examined the final copy of this project and in my opinion; it is fully adequate in terms of scope and quality for the award of the degree of Bachelor of Engineering. I herewith recommend that it be accepted in partial fulfillment of the requirements for the degree of Bachelor of Mechanical Engineering with Automotive Engineering.

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DESIGN OF EXHAUST GAS RECIRCULATION SYSTEM (EGR) FOR DIESEL ENGINE

SYAHMI BIN AHMAD RAMLY

A thesis submitted in partial fulfillment of the requirements for the degree of
Bachelor of Mechanical Engineering with Automotive Engineering.

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JUNE 2012

SUPERVISOR’S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Automotive.

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STUDENT’S DECLARATION

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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ABSTRACT

The exhaust from vehicles pollutes the environment and contributes to global warming, acid rain, smog, odors, respiratory and other health problem. This thesis aims to design a novel system of exhaust gas recirculation (EGR) system for diesel engine. A full system of EGR was built based on the design to reduce the exhaust temperature back to the combustion chamber in order to reduce the NOx emission. The new technique design consists of pipe, heat exchanger, valve and flow meter. The experiment used a diesel engine Mitsubishi 4D68 operated with diesel fuel and PME100 fuel. A gas analyzer was used to measure the emission level on the diesel engine. The experiment was conducted to identify the effect of the quantity of NOx emissions in diesel engine using EGR, new EGR and without EGR. The results of the emissions level on exhaust gas using EGR, new EGR and without EGR are compared. Result shows that NOx emission level using EGR is lower than using new EGR and without EGR. This is because the exhaust temperature is low when using EGR. The NOx concentration was reduced due to the decreasing exhaust temperature. The new EGR did not reduce enough exhaust temperature compared to the original EGR. However, compared to not using EGR, the new EGR still reduced the NOx emission. Finally, the study conforms to its objective where it provides a new and effective technique of EGR that reduces the NOx concentration in diesel exhaust. This study is conducted in order to design a new heat exchanger and a more effective cooling system made with different materials. The experiment was conducted using various type of fuel at difference engine RPM and engine loading.

ABSTRAK

Pada zaman sekarang perlepasan asap ekzos kenderaan menyebabkan pencemaran alam sekitar dan menyumbang kepada pemanasan global, hujan asid, jerebu, pencemaran bau, masalah kesihatan pernafasan dan lain-lain. Tesis ini bertujuan untuk mereka bentuk satu sistem baru gas ekzos edaran semula (EGR) untuk enjin diesel. Sistem EGR yang baru dibina berdasarkan reka bentuk untuk mengurangkan suhu ekzos untuk kembali ke kebuk pembakaran bagi mengurangkan perlepasan gas NOx. Teknik reka bentuk sistem yang baru terdiri daripada paip, penukar haba, injap dan meter aliran ekzos. Ekperimen ini menggunakan enjin Mitsubishi 4D68 diesel yang beroperasi dengan minyak disel dan bahan api PME100. Penganalisa gas digunakan untuk mengukur tahap pencemaran perlepasan gas ekzos pada enjin diesel. Tujuan ekperimen ini dijalankan adalah untuk mengetahui kesan kuantiti pengeluaran NOx dalam enjin diesel apabila menggunakan EGR, EGR baru dan tanpa menggunakan EGR. Hasilnya nanti akan menunjukkan perbandingan tahap perlepasan gas ekzos menggunakan EGR, EGR baru dan tanpa EGR. Tahap perlepasan NOx adalah lebih rendah apabila menggunakan EGR berbanding dengan menggunakan EGR baru dan tanpa EGR. Ini adalah kerana suhu ekzos adalah lebih rendah apabila menggunakan EGR. kepekatan NOx berkurang disebabkan penurunan suhu ekzos. EGR baru tidak cukup untuk mengurangkan suhu ekzos berbanding dengan EGR asal. Tetapi ia masih boleh mengurangkan perlepasan NOx jika dibandingkan dengan tanpa menggunakan EGR. akhirnya kajian ini menyokong objektif dengan menunjukkan keberkesanan teknik baru EGR untuk mengurangkan kepekatan NOx dalam ekzos enjin diesel. Selain itu, kajian ini dirancang untuk menukar rekabentuk penukar haba yang baru, dengan jenis yang lain, diperbuat daripada bahan lain dan sistem penyejukan yang lebih effektif. Dan kajian ini juga akan dijalankan menggunakan pelbagai jenis minyak pada kelajuan enjin yang berbeza dan beban enjin yang pelbagai.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURE

LIST OF SYMBOL

A	Area
A_s	Surface Area
dT/dx	Temperature gradient
h	Convection heat transfer coefficient
K	Thermal conductivity
K	Kelvin
m	Mass, kg
$\dot{m}$	Mass flow rate, kg/s
M	Molar mass, kg/kmol
Q_{cond}	Rate of heat conduction
Q_{conv}	Rate of convection heat transfer
Ts	Surface temperature
T_∞	Temperature of the fluid
W	Power
ε	Emissivity
σ	Stefan-Boltzmann constant
^oC	Celsius

LIST OF ABBREVIATION

ATDC	After Top Dead Center
BDC	Bottom dead centre
CI	Compression ignition
CO	Carbon Monoxide
CO_2	Carbon Dioxide
Cp	Heat capacity
EGR	Exhaust gas recirculation
N_2	Nitrogen gas
HC	Hydrocarbon
NO	Nitric Oxide
NOx	Nitrogen oxides
O_2	Oxygen
PMs	Particulate Matter
PME100	Palm Methyl Ester (100%)
ppm	Part per million
Rpm	Revolution per minute
SAE	Society of Automotive Engineers
SI	Spark Ignition
SO_2	Sulfur Dioxide
TDC	Top Dead centre
WOT	Wide open throttle

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Exhaust gas recirculation system (EGR) is one of the techniques to reduce a nitrogen oxide emission on internal combustion engines. EGR works by re-circulating a portion of an engine exhaust gas back to the engines cylinders from intake manifold. EGR is effective to reduce nitrogen oxides (NO_x) from diesel engines because it lowers the flame temperature and the oxygen concentration of the working fluid in the combustion chamber (M.Zheng et al., 2003). In a diesel engine, the exhaust gas replaces some of the excess oxygen in the pre-combustion mixture. NO_x forms primarily when a mixture of nitrogen and oxygen is subjected to high temperature. The low temperature in the combustion chamber is caused by EGR because it reduces the amount of NO_x when the combustion generates. EGR is used in modern diesel engine to reduce NO_x emissions.

Diesel engines have inherently high thermal efficiencies because of their high compression ratio and fuel lean operation. The high compression ratio produces the high temperature required to achieve auto-ignition. The result of high expansion ratio makes the engine discharge less thermal energy in the exhaust. The extra oxygen in the cylinder is necessary to facilitate complete combustion and to compensate for non-homogeneity in the fuel distribution (M.Zheng et al.,2003). In fact, partial recirculation of exhaust gas which is not a new technique, has recently become essential, in combination with other techniques, for attaining lower emission level (Abd-Alla 2001).

1.2 PROBLEM STATEMENT

Until the introduction of the EPA 2004 emission standards, the manufacturers of truck diesel engine had successfully avoided using exhaust gas recirculation (EGR) system. However, things changed and from 2007 onward, all diesel engine OEMs used some form of EGR though they did not call it EGR because EGR concerns emissions control. The emissions until today pollute the environment and contribute to global warming, acid rain, smog, odors and respiratory and other health problem (Willard 2003). Diesel engine is a major source for air pollution. It's contains oxides of nitrogen (NO_x) carbon monoxide (CO), hydrocarbon (HC), carbon dioxide (CO_2), oxygen (O_2) and particulate matter (PM).

High combustion temperatures produce nitrogen oxide (NO_x), a constituent of zone. There are two ways to reduce peak combustion temperatures: Spark control system and Exhaust Gas Re-circulation (EGR) systems (Mike and Hynes 2005). From the two ways, EGR system is found to be a better way to control NO_x. EGR systems reduce peak combustion temperatures by diluting the incoming air/fuel mixture with small amount of "inert" (won't undergo a chemical reaction) exhaust gas. Six to 14 percent concentration of exhaust gas, routed from the exhaust system to the intake manifold, mixed with the air/fuel mixture entering each cylinder, and reduced the mixture's ability to produce heat during combustion (Mike and Hynes 2005).

1.3 PROJECT OBJECTIVE

The objective of this project is to design a novel system of EGR for diesel engine.

1.4 PROJECT SCOPE

The scope of this experiment is:

i. Design of the new techniques of EGR system
ii. Fabricate the EGR system
iii. Conduct experimental works

iv. Get the data analysis

1.5 FLOW CHART

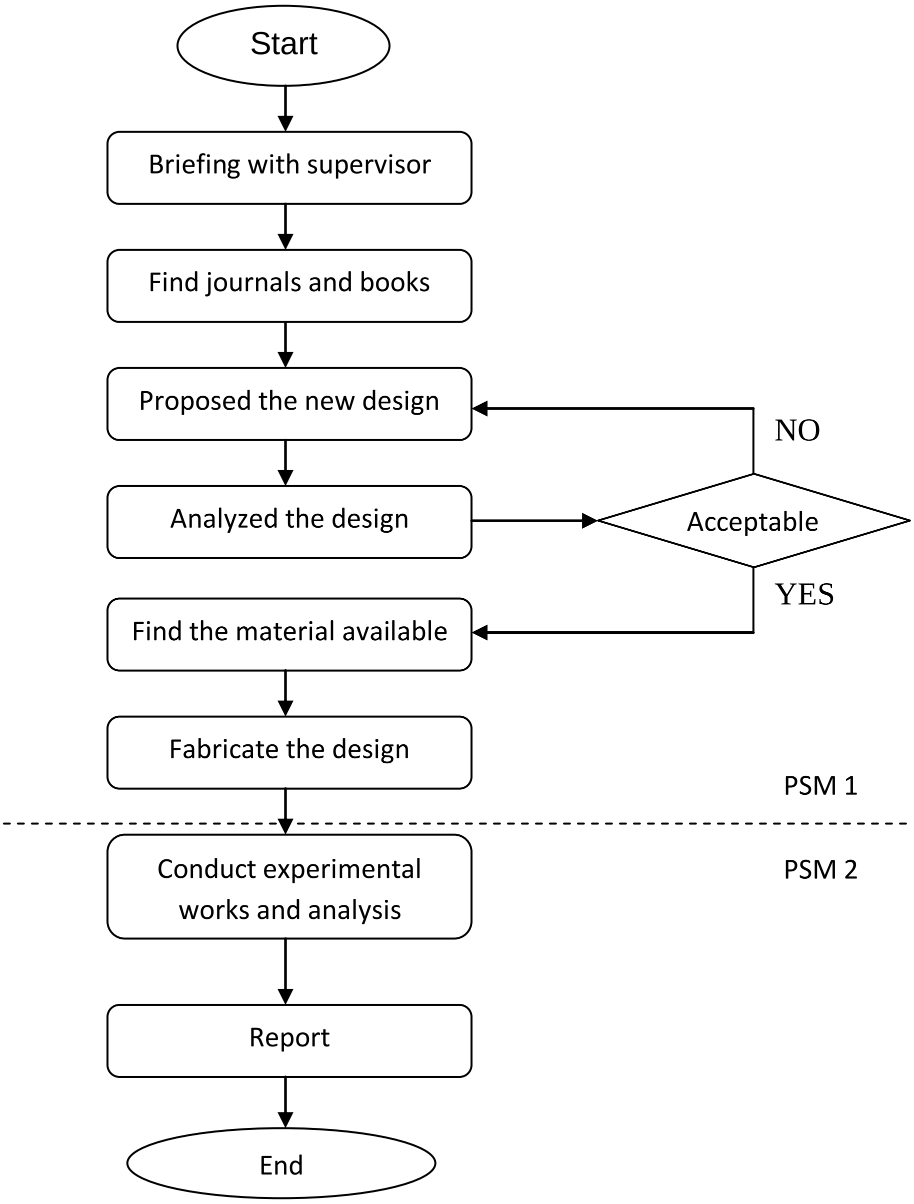


Figure 1.1: Project Flow Chart

CHAPTER 2

LITERATURE RIVIEW

2.1 INTRODUCTION

This chapter presents an introduction of diesel engine, effect of exhaust gas recirculation system on diesel engine and effect of cooling the re-circulated exhaust gas. This chapter also describes the design of heat exchanger.

2.2 DIESEL ENGINES

Martynn Randall (2004) stated that the first commercially – successfully compression – ignition engine was invented by Rudolf Diesel at the end of the 19th century. In 1892, Rudolf Diesel a German engineer perfected the compression-ignition engine and named it after his name, diesel. The diesel engine uses heat created by compression to ignite the fuel. Thus the spark ignition system is not requires. The diesel engine requires compression ratios of 16:1 and higher. Heat of compression happened when incoming air is compressed until its temperature reaches about 1000^{o}F (540^{o}C). When the piston reaches the top of its compression stroke, fuel is injected into the cylinder, where it is ignited by the hot air (Halderman 2009). In diesel engine, air is not controlled by a throttle as in a gasoline engine. Instead, the amount of fuel injected is varied to control power and speed.

Compared to spark ignition engine, diesel engine has the advantages of lower fuel consumption, the ability to used cheaper fuel, the potential to produce higher power output. Over the decades, diesel engine was widely used for stationary and marine applications, but the fuel injection system used was not capable of high –

speed operation. This speed limitation and the weight of air compression needed to operated the injection equipment, made the first diesel engine unsuitable to be used on road- going vehicle.

The air-fuel mixtures of a diesel as lean as 85:1 at idle. It varies to as rich as 20:1 at full load. Diesel is more fuel-efficient than a gasoline engine because of higher air-fuel ratio and increased compression pressure. Besides that, diesel engine does not suffer throttling losses. Throttling losses involve the power needed in a gasoline engine to draw air past a closed or partially closed throttle. Diesel engines have two-stroke and four stroke versions. The most common two-stroke diesels were the truck and industrial engines made by Detroit Diesel (Halderman 2009). In these engines, air intake is through poppet valves in the head. A blower pushes air into the box surrounding liner port to supply air for combustion and to blow the exhaust gases out of the exhaust valves. Diesel engine is also known as compression ignition (CI) engine. The fuel injection into the combustion chamber to the compression stroke was very late in the early period of CI engines.

Due to ignition delay and the finite time required to inject the fuel, combustion lasted into the expansion stroke. This kept the pressure at peak levels well past TDC. This combustion process is best approximated as a constant pressure heat input in an air-standard cycle, resulting in the diesel cycle shown in figure 2.2. The rest of the cycle is similar to the air standard Otto cycle (Willard 2003).

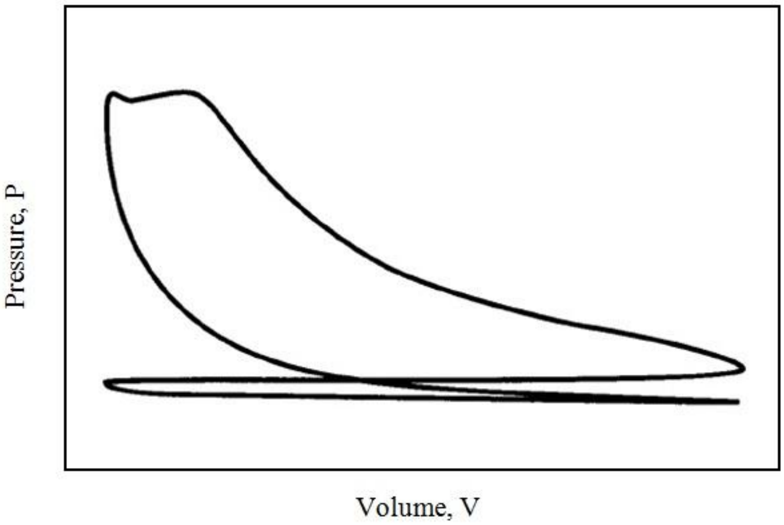
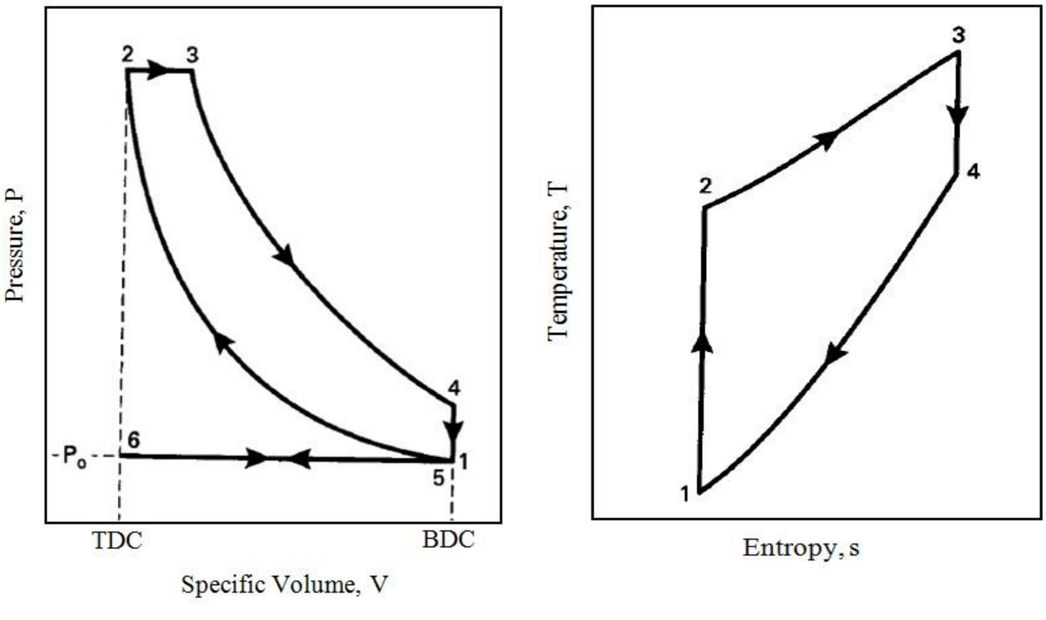


Figure 2.1: Indicator diagram of a historic CI engine operating on an early four stroke cycle



(a) (b)

Figure 2.2: Four stroke cycle on early CI engine on (a) Pressure-specific volume coordinate, and (b) temperature-entropy coordinate.

Source: Willard (2003)

During the suction stroke, the inlet valve opens and air enter the cylinder as the piston moves from TDC to BDC. When the inlet and exhaust valves are closed, the piston compresses the air, and the pressure and temperature rise. When the piston is about to reach the TDC, fuel is injected in a finely divided form into the hot swirling air in the combustion space. Ignition occurs after a short delay, the gas pressure rises rapidly and a pressure wave is set up. Work is done by the gas pressure on the piston as the piston sweeps the maximum cylinder volume. During this expansion or power stroke, the temperature and the pressure of the burn gas will decrease. As the piston approaches the BDC, the exhaust valve opens and the products of combustion are rejected from the cylinder during the exhaust stroke. Near the TDC, the inlet valve opens again and the cycle repeated (Gupta 2006).

The typical valve timing for a 4 stroke CI engine are as follows:

- Inlet valve opens about 30^0 before TDC
- Inlet valve closes about 50^0 after BDC
- Exhaust valve opens about 45^0 before BDC
- Exhaust valve closes about 30^0 after TDC
- Injection of fuel is about 15^0 before TDC

The basic engine cycle for four stroke compression ignition (CI) engine cycle is for the first stroke is known as intake stroke where only air goes into the combustion chamber without adding fuel to it. It is the same as the intake stroke in a spark ignition (SI) engine. For the second stroke or compression stroke, air is compressed to increase the pressure and temperature. For this stroke, fuel is injected directly into the combustion chamber and will mix with the hot air. The fuel then evaporates and self-ignites, thus combustion begins. Combustion is fully developed by TDC and continues at a constant pressure until the fuel injection is complete and the piston has started to move to BDC (Willard 2003). The third stroke is the power stroke where it continues the combustion process end and the piston travel towards BDC. Finally the fourth stroke or exhaust stroke, the exhaust gas exits the engine.

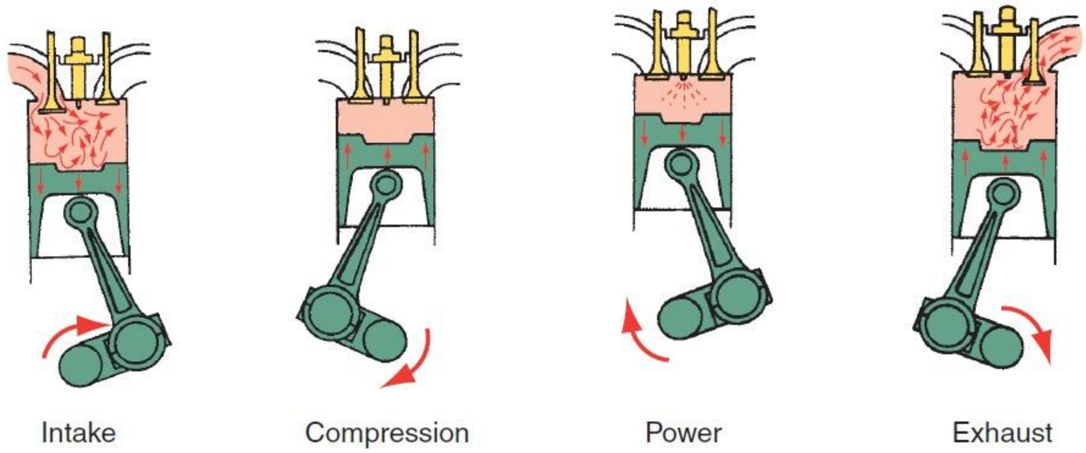


Figure 2.3: A four stroke diesel engine cycle.

Source: Jack Erjavec (2010)

Diesel engine uses heat to ignite the fuel by compressing air in the combustion chamber. The compression ratio of diesel engines is typically three times (as high as 25:1) that of a gasoline engine. The temperature rises up to 700^{o}C to 900^{o}C when the air intake is compressed. A small amount of diesel fuel then is injected into the combustion chamber using the injector spray just before the air is fully compressed. The air will instantly ignite the fuel when it reaches a certain temperature due to the compression. Meanwhile in power stroke, the piston goes down when the combustion increases the heat in the cylinder as a result of the high pressure in the chamber.

2.3 EXHAUST GAS RECIRCULATION (EGR)

The Exhaust gas recirculation system is designed to reduce the amount of nitrogen oxides (NOx). This NOx is created by the engine during operating periods due to high temperature of combustion. When the combustion temperature exceeds 2500^{o}F, a highly concentrated NOx is formed. The EGR system works by re-circulating a small amount of exhaust gas back to the combustion chamber through the intake manifold where it mixes with the incoming air/fuel charge. The high temperature and the pressure are reduced by diluting the air/fuel mixture under that condition.

The EGR flow has three operating conditions. The first condition is the high EGR flow; where it is necessary during cruising and mid range acceleration. This is a condition where the combustion temperature is very high. Meanwhile the second condition is low EGR flow. Low EGR flow is needed during low speed and light load conditions. Finally the third condition is the no EGR flow condition. When the engine warms up and idle the wide open throttle, no EGR flow should occur during that condition. EGR operations could adversely affect engine operating efficiency or vehicle derivability.

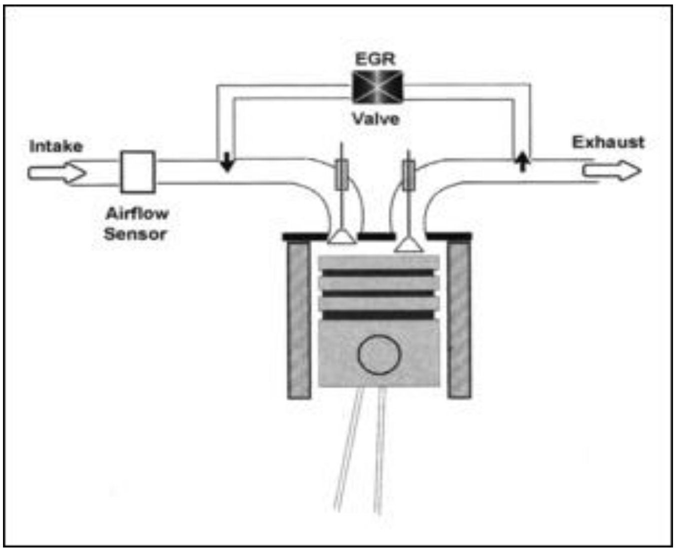


Figure 2.4: Concept of exhaust gas recirculation system

Source: Isuzu Motor (2012)

Exhaust gas recirculation is use to re-circulate the exhaust gas back to the combustion chamber at intake manifold. In other words, to supply exhaust gas to the fresh mixture or to the air sucked into the cylinder. The use of exhaust gas recirculation is needed to control the production of NOx emission for gasoline and diesel engines (Richard 2006). The NOx reduction is primarily caused by the following factors:

- The heat capacity (Cp) of the re-circulated exhaust gas is higher than the heat capacity (Cp) of the air. This makes the low temperature increases for the same amount of energy release by combustion.
- Reduction of the O_2 partial pressure and therefore, lower oxygen mass inside the cylinder, because a portion of the combustion air is replaced by exhaust gas with lower oxygen content.
- Reduction of the combustion speed and therefore lower temperature increase.

When the combustion temperatures are too high it form a nitrogen oxides (NOx). Any measure to decrease NOx and emission lead to reduced the combustion temperature. The use of EGR will increased the soot and other solid paniculate loading of lubricant oil. Re-introduction of the acidic exhaust gas product (sulfuric acid) into the engine will rapidly increase the total acid number (TAN) of the lubricant (S.Aldajah et al. 2006)

Using high exhaust gas recirculation (EGR) rates by increased boost pressure to avoid the negative impact on soot emissions is the one efficient method to control NOx in order to achieve future emissions limits (Hountalas et al. 2006). The combustion noise and the thermal efficiency of the dual fuel engine are found to be affected when EGR is used in the dual fuel engine (Selim 2001).

2.3.1 EGR theory of operation

The purpose of EGR system is to precisely regulate EGR flow under different operating condition. EGR system also has to override flow under conditions which would compromise good engine performance. Like the engine load change, the precise amount of exhaust gas which must be metered into the intake manifold varies significantly. This results in the EGR system operating on a very fine line between good NOx control and good engine performance.

The engine performance will be suffered if too much exhaust gas is metered. The engine may knock and will not meet strict emissions standards if too little EGR flows. The EGR ratio is referred as the theoretical volume of recirculated exhaust gas. The graph in figure 2.4 shows the EGR ratio increases as engine load increases.

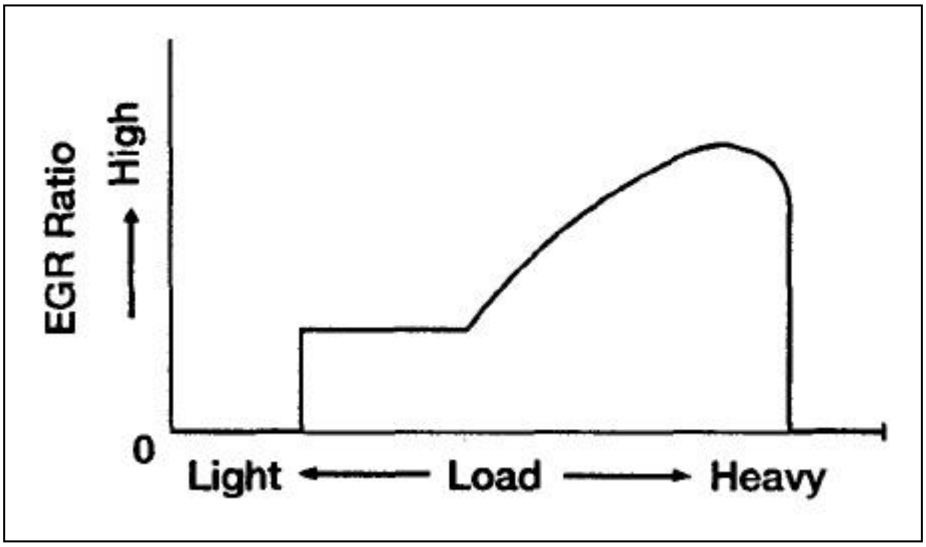


Figure 2.5: Relationship between EGR ratio and engine load

Source: Toyota Motor sales

EGR system also gives impact on the engine control system (ECS). The ECM considers the EGR system as an integral part of the entire engine control system (ECS). The ECM is capable of neutralizing the negative performance aspects of EGR by programming advanced additional spark and decreased the fuel injection duration during periods of high EGR flow. By integrating fuel and spark control with the EGR metering system, engine performance and fuel economy can actually be enhanced when the EGR system is functioning as designed.

To regulate exhaust gas flow from the exhaust to the intake manifold, the EGR control valve is used by means of a pintle valve attached to the valve diaphragm. A ported vacuum signal and calibrated spring on one side of the diaphragm are balanced against atmospheric pressure acting on the other side of the diaphragm. The valve is pulled further from its seat when the vacuum signal applied to the valve increases.

2.3.2 EGR cooling system

The re-circulated exhaust gas can be cooled down to reduce the NOx emissions. Meanwhile, reduction for radiators is possible up to 50% following the design accordingly. Cooling the re-circulated exhaust gas is one of the ways to reduce the emissions that caused pollution; without significant additional fuel consumption. NOx generation can be reduced by lowering gas temperature in the combustion chamber. To achieve this, a specially developed EGR cooler is installed between the EGR valve and the intake manifold entry point for the exhaust gas. It uses water as a cooling medium to reduce the exhaust gases temperature and the amount of pollutants. It is believed that it is more effective than using air as a cooling medium. Exhaust emission NOx decreased but the particulate matter concentration in the exhaust gases increased when cooled EGR rises in ratio (Nidal 2002).

A system with controlled EGR cooling system combined with a controlled engine cooling system shows that it decreased the warm up times for fast warm up of after treatment devices, decreased power consumption, and gave better engine temperature control (Chalgren et al. 2007). Using a cooled EGR system raises the density of the intake air thus, the amount of air entering the combustion chamber increases. A complete combustion then is achieved, thereby reducing the generation of PM.

2.4 HEAT EXCHANGER

Heat exchanger is equipment that transfers heat from one medium to another. It is also a device where two moving fluid streams exchange heat without mixing. Different heat transfer applications require different types of hardware and different configuration of heat transfer equipment. The most basic heat exchanger is double pipe heat exchanger. This type of heat exchanger consists of two concentric pipes of different diameters (Chengel 2006). Heat is transferred from the hot fluid to the cold condition through the wall separating the fluids. Yunus A.Chengel (2007) asserted the conservation of mass principle for a heat exchanger in steady operation requires that the sum of the inbound mass flow rates. The principle can also be expressed as the following, ‘under steady operation, the mass flow rate of each fluid stream flowing through a heat exchanger remains constant’ (Chengel et al. 2007)

Heat exchanger is a device that makes the flow of thermal energy between two or more fluids at different temperatures. Heat exchanger is used in a wide variety of applicants. These include power production, waste heat recovery, manufacturing industries, air-conditioning, refrigerant and many more.
Heat exchanger may be classified according to;

1. Recuperators/ regenerators
2. Transfer process: direct contact and indirect contact
3. Geometry of construction: tubes, plates, and extended surfaces.
4. Heat transfer mechanisms: parallel flow and counter flow.

2.4.1 Recuperation and Regeneration

Recuperator is a conventional heat exchanger where it transfers the heat between two fluidswith heat transfer between two fluids because the hot stream A recovers (recuperates) some of the heat from stream B (Figure 2.6). The heat transfer occurs through a separating wall or through the interface between the streams as in the case of direct contact type of heat exchanger in Figure 2.7.

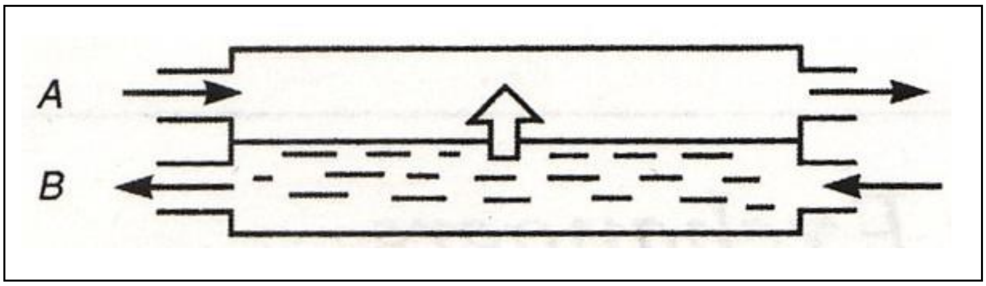


Figure 2.6: Recuperator

Source: Sadic Kakac and Hongton Liu (2002)

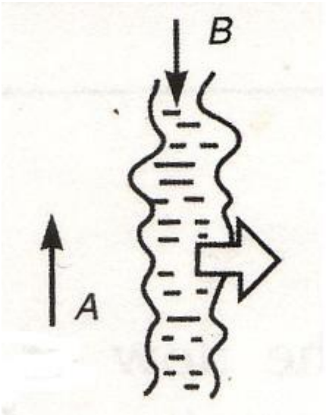


Figure 2.7: Direct contact Heat Transfer across interface between fluids

Source: Sadic Kakac and Hongton Liu (2002)

The same flow passage (matrix) is alternately occupied by one of the two fluids in regenerators or in storage type heat exchanger. The hot fluid stores the thermal energy in the matrix and when the cold fluid flows through the passage after that, the stored energy will be extracted from the matrix. Therefore, thermal energy is not transferred directly through the wall like the usual direct heat transfer in heat exchanger. This cyclic principal is illustrated in Figure 2.8. The solid in the cold stream A loses heat; while in the hot stream B, it receives heat and then it is regenerated.

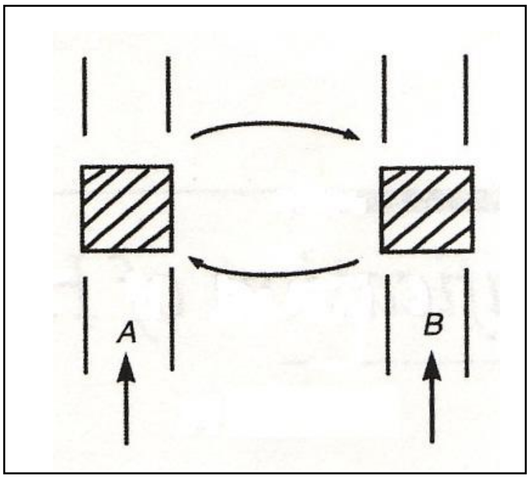


Figure 2.8: Recuperator

Source: Sadic Kakac and Hongton Liu (2002)

2.4.2 Transfer Processes

Heat exchangers are classified into two; direct contact and indirect contact (transmural heat transfer). In direct contact heat exchanger, heat is transferred between the cold and hot fluids through directly between these fluids. There is no wall between hot and cold streams, and heat transfer occurs through the interface between two streams, as shown in figure 2.7. The streams are two immiscible liquids, a gas liquid pair, or a solid particle-fluid combination.

Indirect contact heat exchanger on the other hand is when the heat exchanging process between hot and cold fluid happens through a transfer surface like a wall separating the fluids. The cold and hot fluids flow simultaneously while heat is transferred through the separating wall (Figure 2.8). The fluids do not mix.

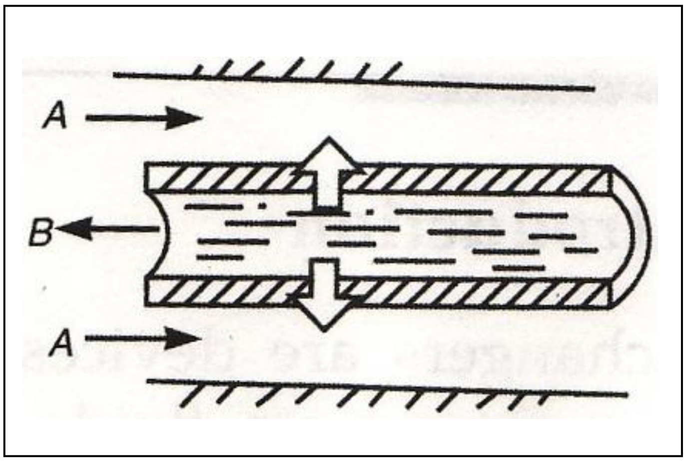


Figure 2.9: Transmural Heat Transfer through walls: fluids not in contact

Source: Sadic Kakac and Hongton Liu (2002)

2.4.3 Geometry of Construction

Direct transfer heat exchanger (transmural heat exchanger) is often describes according to its construction features; tubular, plates and extended surface heat exchanges.

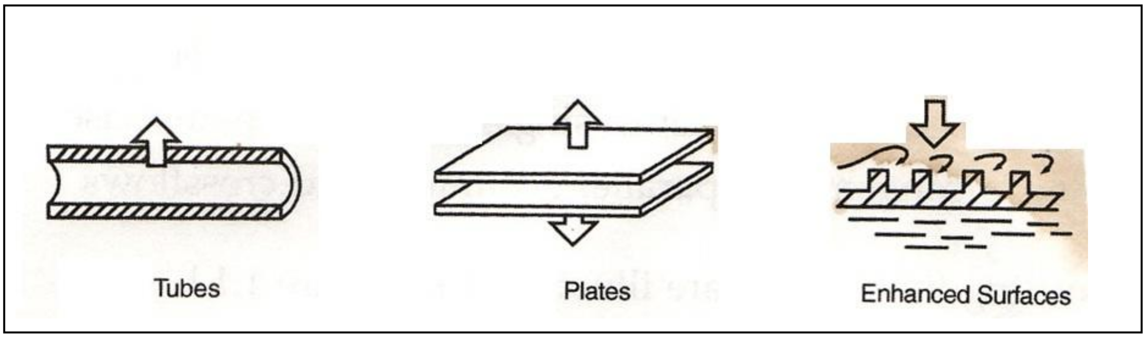


Figure 2.10: The geometry of Construction heat exchangers

Source: Sadic Kakac and Hongton Liu (2002)

1. Tubular Heat Exchanger

    i. Double-Pipe Heat Exchangers

    - A typical double-pipe heat exchanger consists of one pipe placed concentrically inside another pipe with larger diameter and appropriate fittings to direct the flow from one section to the next. Double pipe heat exchanger can be arranged in various series and in a parallel arrangement to meet the pressure drop and mean temperature difference required.

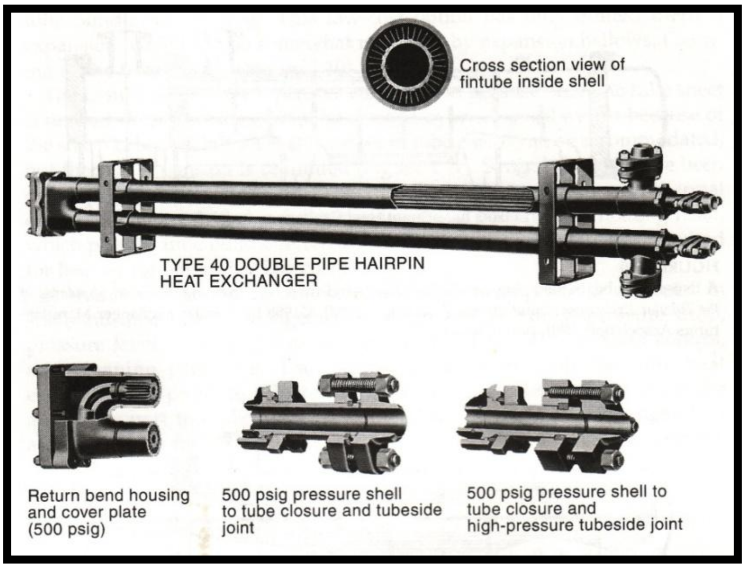


Figure 2.11: Double pipe hair pin heat exchanger.

Source: Sadic Kakac and Hongton Liu (2002)

    ii. Shell And Tube Heat Exchangers

    - One fluid stream flows through the tubes while the others flow on the shell side, across or along the tubes and it is called shell-and-tube heat exchangers. It is made of round tubes mounted in large cylindrical shells with the tube axis parallel to that of the shell.

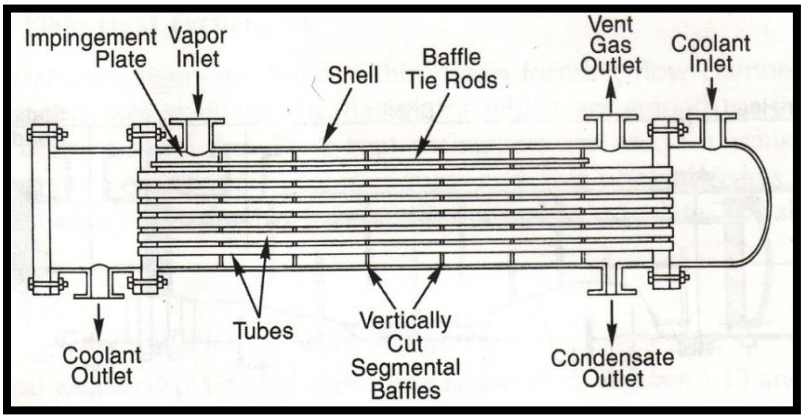


Figure 2.12: Shell and tube heat exchanger as a shell side condenser

2. Source: Sadic Kakac and Hongton Liu (2002)

i. Spiral Tube Type Heat Exchangers
   - Spiral tube type heat exchanger consists of spirally wound coils placed in a shell. Some of the designs usually are designed as coaxial condenser and co-axial evaporators that are used in refrigeration system.

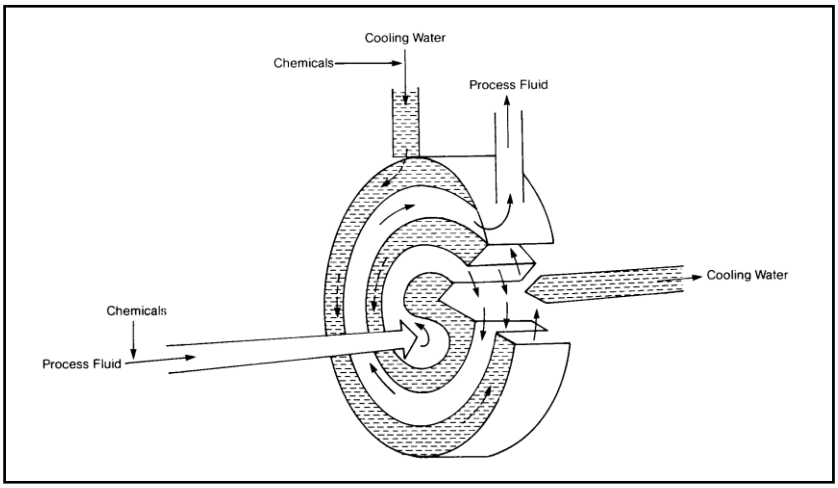


Figure 2.13: Spiral heat exchanger

Source: Sadic Kakac and Hongton Liu (2002)

3. Plate Heat Exchanger

   i. Gasketed Plate Heat Exchangers
      - Consist of series of thin plates with corrugations or wavy surface that separates the fluids. The plates come with the corner parts arranged so that the two media between which heat is to be exchanged flow through an alternate interpolate space.

   ii. Spiral Plate Heat Exchangers
      - Spiral heat exchangers are formed by rolling two long, parallel plates into a spiral using a mandrel. The edges of adjacent plates are welded to form channels. The distance in between the metal surfaces in both spiral channels is maintained by means of distance pins welded to the metal sheet.

   iii. Lamella Heat Exchangers
      - The lamella (Ramen) heat exchanger consist of a set of parallel, welded, thin plate channels or lamellae (flat tubes or rectangular channel) placed longitudinally in a shell, it is a modification of the floating-head type shell-and-tube heat exchanger.

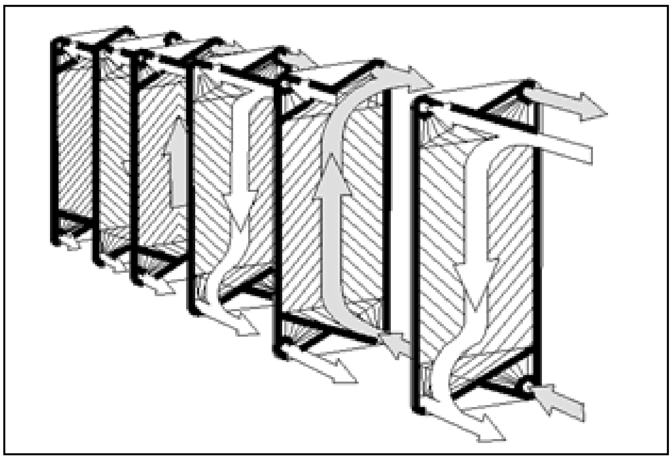


Figure 2.14: Gasketed plate heat exchangers

Source: Sadic Kakac and Hongton Liu (2002)

4. Extended Surface Heat Exchangers

    i. Plate-Fin Heat Exchanger

        - The plate fin heat exchangers are primarily used for gas to gas applications and tube fin exchangers. In most of the applications (i.e. in truck, cars, and airplane), mass and volume reductions are particularly important.

    ii. Tubular-Fin Heat Exchanger

        - These heat exchangers are used in gas to liquid exchangers. The heat transfer coefficient on the gas side is generally much lower than those on the liquid side and fins are required on the gas side. A tubular-fin (or tube-fin) heat exchanger consists of an array of tubes with fins fixed on the outside.

2.4.4 Heat transfer mechanisms

Two types of heat transfer mechanisms in heat exchanger are parallel flow and counter flow. Parallel flow is when both hot and cold fluids enter the heat exchanger at the same time and move in the same direction. For the counter flow, the hot and cold fluid enters the heat exchanger at opposite ends and flow in opposite direction (Chengel 2006).

Conduction

Conduction heat transfer energy is energy transported due to molecular motion and interaction. Conduction heat transfer through solids is due to molecular vibration. It is the transfer of energy from the more energetic particles of a substance to the adjacent less energetic ones as a result of interactions between the particles (Cengel 2006). It can take placed in solid, liquid or gas. The Fourier determined that Q/A, the heat transfer per unit area (W/m^2) is proportional to the temperature gradient dT/dx. The constant of proportionality is called the material thermal conductivity k. The thermal conductivity k depends on the material, for example various materials used in engines have the following thermal conductivities (W/m.K):

Table 2.1: Thermal Conductive of Common Materials

Material	**k**
Copper	400
Aluminum	240
Cast Iron	80
Water	0.61
Air	0.026

Source: Chengel (2006)

Fouriers equation:

$$\dot{Q}_{cond} = -kA\,\frac{dT}{dx} \qquad [2.1]$$

Convection

Convection heat transfer through gases and liquids from a solid boundary results from the fluid motion along the surface. Convection heat transfer is energy transport due to bulk fluid motion. Convection is the mode of energy transfer between a solid surface and the adjacent liquid or gas that is in motion and it involves the combined effects of conduction and fluid motion. Newton determined that the heat transfer/area, Q/A, is proportional to the fluid solid temperature difference T_s-T_f. The temperature difference usually occurs across a thin layer of fluid adjacent to the solid surface. This thin fluid layer is called a boundary layer. The constant of proportionality is called the heat transfer coefficient, h.

Newton's Equation:

$$\dot{Q}_{conv} = hA_s\,(T_s - T_\infty) \qquad [2.2]$$

The heat transfer coefficient depends on the type of fluid and the fluid velocity. The heat flux, depending on the area of interest, is the local or area averaged. The various types of convective heat transfer are usually categorized into the following areas:

Table 2.2: Convection heat transfer coefficients

Convection Type	**Description**	**Typical value of h ($W/m^2.K$)**
Natural Convection	Fluid motion induced by density differences	10 (gas) 100 (liquid)
Forced Convection	Fluid motion induced by pressure differences from a fan or pump	100 (gas) 1000 (liquid)
Boiling	Fluid motion induced by a change of phase from liquid to vapor	20,000
Condensation	Fluid motion induced by a change of phase from vapor to liquid	20,000

Source: Sadic Kakac and Hongton Liu (2002)

Radiation

Radiation is the energy emitted by matter in the form of electromagnetic waves (or photons) because of changes in the electronic configurations of the atoms or molecules. Radiation heat transfer is energy transport due to emission of electromagnetic waves or photons from a surface or volume. The radiation does not require a heat transfer medium, and can occur in a vacuum. The heat transfer by radiation is proportional to the fourth power of the absolute material temperature. Unlike conduction and convection, the transfer of heat by radiation does not require the presence of an intervening medium (Chengel 2006). The radiation heat transfer also depends on the material properties represented by e, the emissivity of the material.

$$\dot{Q}_{emit} = \varepsilon\sigma A_s T_s^4 \qquad [2.3]$$

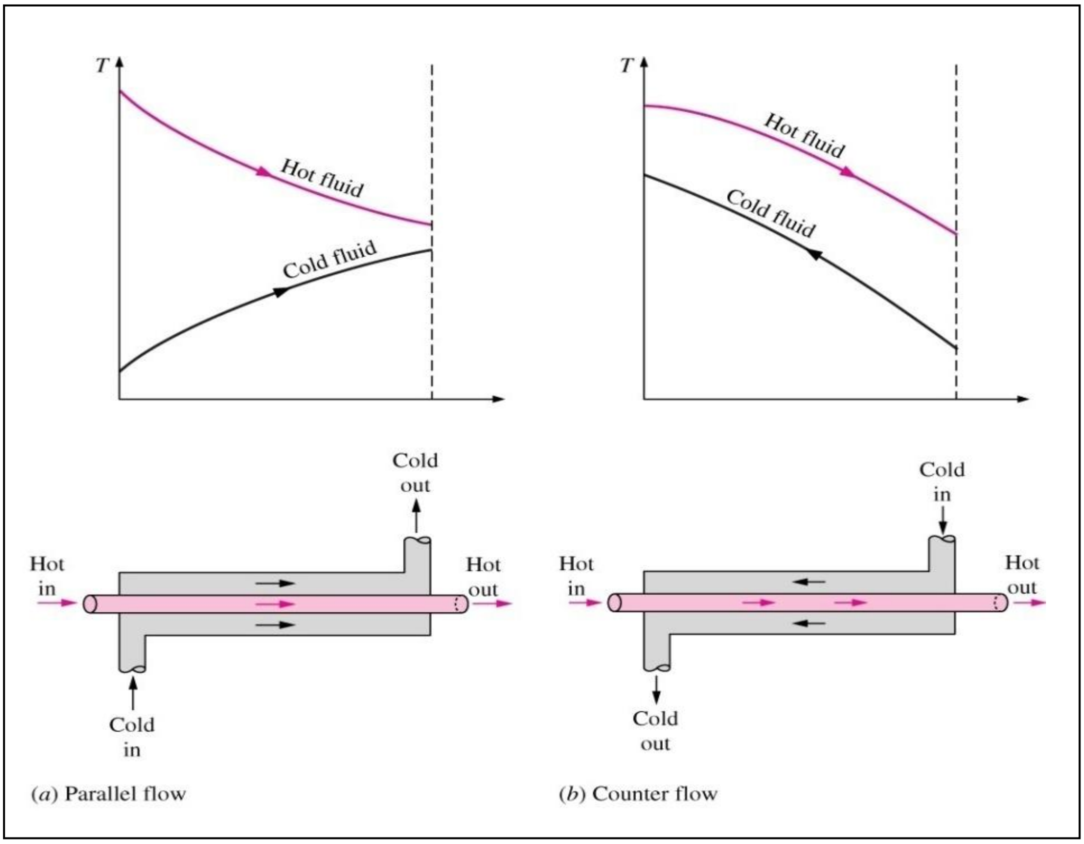


Figure 2.15: (a) Parallel flow heat exchanger (b) counter flow heat exchanger

Source: Yunus A.Chengel (2006)

2.5 EXHAUST EMISSION

Exhaust emission from vehicle is one of the major problems that cause air pollution. Diesel engine exhaust can cause health problem, air pollution, global warming, acid rain, smog, and odors. The exhaust emissions generally result from the combustion of fossil fuel inside the engine. Environmentalist, engineers and technicians give extra attention to automotive emissions like hydrocarbon (HC), carbon monoxide (CO), oxides of nitrogen (NOx), and Oxygent (O_2) (Erjavec 2010).

These emissions are caused by the non-stoichiometric combustion, dissociation of nitrogen, and impurities in the fuel and air (Willard 2003). Diesel engine runs with a very lean fuel-air mixture, therefore the concentration of CO and HC are much lower than the typical SI engine levels in the exhaust (Guptha 2009).

The particulate matter from diesel's exhaust consists primarily of soot with some additionally absorbed hydrocarbon materials. Overall, the engines and fuels can be developed and generated in such a way where less harmful emissions will be emitted. These emissions can be exhausted to the surroundings without harming the environment. The after treatment for the exhaust gasses is important since it can reduce the emissions; however, the latest technology nowadays is yet to reach this level. Nowadays the technology mainly focuses on the use of thermal or catalytic converters and particulate traps (Willard 2003). The exhaust temperature of CI engine is lower due to their greater expansion ratio and generally in the range from 200^0C to 500^0C (Gupta 2006). The heat transfer of exhaust gas affects the emissions that burnt up in the exhaust system. It influents turbocharger performance and it contributes significantly to the engine requirements.

Willard W. Pulkrabek (2003) stated that until in the middle of the 20th century, the number of internal combustion engines in the world was small that the pollution they emitted was tolerable, and the environment, with the help of sunlight, stayed relatively clean. As the world population grew, power plants, factories and an ever increasing number of automobiles began to pollute the air to the extent that it was no longer acceptable. In 1940s, air pollution was first recognized as a problem in the Los Angeles basin in California. Two causes of this were the large population density and the natural weather conditions of the area. The large population created many factories and the power plants, as well as one of the largest automobile densities in the world.

Smoke and other pollutants from many factories and automobiles combined with the fog that was common in this ocean area resulted to smog. In 1950s, the smog problem increased along with the population density and automobile density. It was recognized that automobile was one of the major contributors to the problem, and by 1960s, the emission standards was enforced in California. It was due to the exhaust emissions that were found out containing pollutant gasses which are hydrocarbon (HC), carbon monoxide (CO), oxide of nitroge (NOx), and non-pollutant gasses; oxygen (O_2) and carbon dioxide (CO_2) (Barry Hollembeak, 2005).

In the next few decades after that, emission standards were adopted in the rest of the United States and in Europe and Japan. By making engines more fuel efficient, and with the use of exhaust after treatment, emissions per vehicle of HC, CO and NOx were reduced by 95% in 1970s and 1980s. Lead, one of the major air pollutants, was phased out as a fuel additive in 1980s. More fuel-efficient engines were developed, and by 1990s the average automobile consumed was less than half of the fuel used in 1970s. However, the number of automobile usage increased rapidly during that time thus the fuel usage did not decrease at all. In 1999, petroleum consumption in the Unites States amounted to 16500 L/sec, a large percentage of fuel which was for internal combustion engine.

2.5.1 Oxides of Nitrogen

NOx refer to oxides of nitrogen. Nitrogen gas occurs naturally in the atmosphere as an inert gas in the air. It is approximately 78% N_2 by volume. NOx generally includes nitrogen monoxide, also known as nitric oxides (NO) and nitrogen dioxide (NO_2). They may also include Nitrous Oxide (N_2O), as well as other less common combinations of nitrogen and oxygen such as nitrogen tetroxide (N_2O_4) and nitrogen pentoxide (N_2O_5) (Baukal 2003).

The oxides of nitrogen tend to settle on the hemoglobin in the blood. The most undesirable toxic effect oxides of nitrogen is their tendency to join with the moisture in the lungs form diluted nitric acid. NOx is one of the primary causes of photochemical smog (smoke + fog). Smog is form by the reaction below:

$$NO_2 + \text{energy from sunlight} \longrightarrow NO + O + \text{smog} \qquad [2.4]$$

Monatomic oxygen reacts with O_2 to form ozone (O_3) as follow:

$$O + O_2 \longrightarrow O_3 \qquad [2.5]$$

Ozone is harmful to the lungs and other biological tissues. It is harmful to crops and trees too. It reacts with rubber, plastics, and other materials causing damage to it.

Most of oxides of nitrogen comprise nitric acid (NO), a small amount of nitrogen dioxide (NO_2) and traces of other nitrogen oxides. These are all grouped together and the group is called NOx. NOx is mostly formed from atmospheric nitrogen. There are a number of possible reactions that form NO. NO form in both the flame front and the post flame gases. Some of the NO forming reactions are:

$$\begin{aligned} O + N_2 &\longrightarrow NO + N \\ N + O_2 &\longrightarrow NO + O \\ N + OH &\longrightarrow NO + H \end{aligned} \qquad [2.6]$$

N, O, OH is formed from dissociation of NO_2, O_2 and H_2O vapor at high temperatures that exit in the combustion chamber (2500 – 3000 K). The higher combustion reaction temperature, the more diatomic nitrogen (N_2) will dissociate to monatomic nitrogen (N) and NOx will be formed. At low temperatures, a very small quantity of NOx is created.

The flame temperature is maximum at the stoichiometric equivalence ratio ($ø = 1.0$) but maximum NOx is formed slightly at a lean equivalence ratio ($ø = 0.95$). At this condition the flame temperature remains high but excess oxygen helps in the formation of more NOx. The most important engine variables that affect NOx emission are the fuel/air equivalence ratio, the burn gas fraction, and combustion duration within the cylinder.

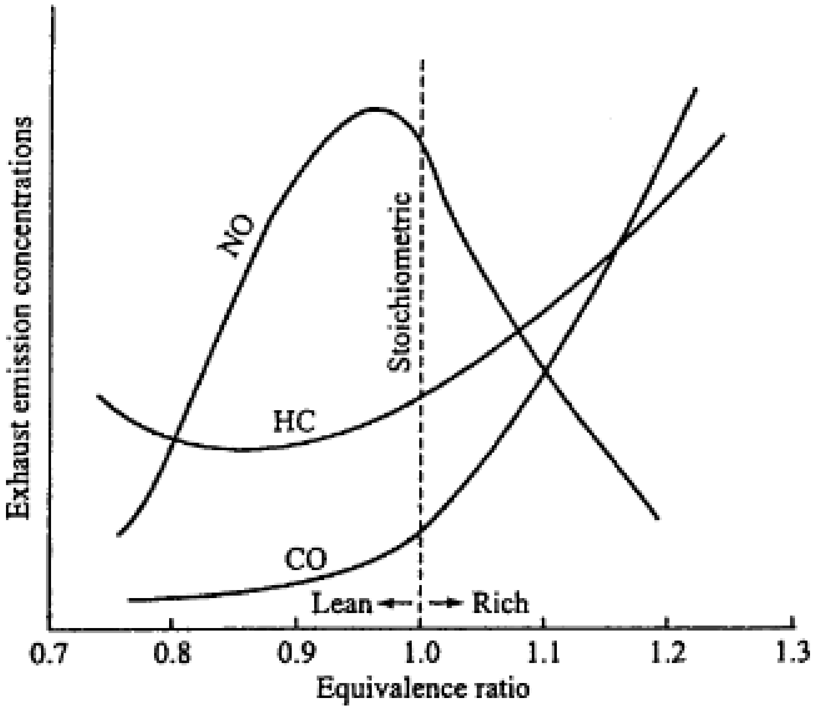


Figure 2.16: Variation of CO, NO, HC concentration in the exhaust of an SI engine versus equivalence ratio.

Source: H.N. Gupta (2009)

2.5.2 Carbon Monoxide

Carbon monoxide, a colorless, odorless, poisonous gas, is produced in an engine when it is operated with fuel – rich equivalence ratio. When they're not enough oxygen to convert all carbon to CO_2, some fuel does not get burn and some carbon ends up as CO. Typically the exhaust of an SI engine will be about 0.2% to 5% carbon monoxide. Not only CO is considered as an undesirable emission, but it also represents lost chemical energy that was not fully utilized in the engine. CO is a fuel that can be combusted to supply additional thermal energy.

$$CO + \tfrac{1}{2} O_2 \longrightarrow CO_2 + \text{heat} \qquad [2.7]$$

Maximum CO is generated when an engine runs rich, such as when starting or when accelerating under load. Even when the intake of air – fuel mixture is stoichiometric or lean, some CO will be generated in the engine. Poor mixing, local rich regions, and incomplete combustion will create some CO.

2.5.3 Carbon Dioxide

Carbon dioxide is one of the two exhaust gases that are not considered as pollutant reported by Barry Hollembeak, (2005). CO_2 is a by – product of perfect combustion. It is formed when one atom of carbon bonds with two atom of oxygen from the air during combustion. An essentially harmless gas, it is present at levels of 14 percent to 15 percent in the exhaust of properly running engine. Carbon dioxide is also produced when carbon monoxide is oxidized in the catalytic converter. The amount of CO_2 in the exhaust is directly related to the air – fuel ratio. As the fuel mixture approaches stichoimetric, the level of CO_2 peaks. It decreases when the mixture become richer or lean. This fact makes CO_2 in the exhaust an excellent reference that can help to determine how efficient the engine is in combusting its fuel. Higher CO_2 reading means higher efficiency of combustion.

2.5.4 Oxygen

Oxygen was stated by Barry Hollembeak, (2005), is not the pollutant gases that come out from the exhaust. Oxygen reacts with the fuel for combustion during combustion process. If the combustion chamber provides too much fuel to the amount of oxygen, all the available oxygen is used up and will result in low O_2 contents in exhaust. As the air – fuel mixture move toward lean, the amount of oxygen in the exhaust steadily increase. When the air – fuel mixture is rich or lean, the levels of oxygen and carbon monoxide opposite one another where when the O_2 high, CO is low and vice versa. At the stoichiometric air –fuel ratio, the level of O_2 and CO in the exhaust approximately equal.

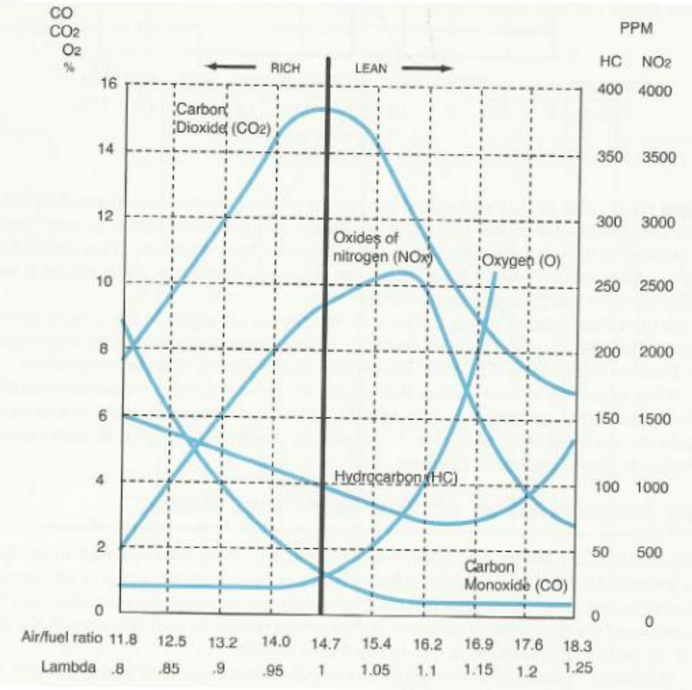


Figure 2.17: Total Effect of the Air – Fuel Ratio on Exhaust Gases

Source: Hollembeak (2005)

2.5.5 Hydrocarbon

Hydrocarbon consists of hydrogen and carbon atom in various combinations. The CI engine operate with an overall fuel – lean equivalent ratio, therefore they emit only about one – fifth of the hydrocarbon emission of an SI engine (Gupta, H.N. 2009). The following are major causes for hydrocarbon emissions in the exhaust of CI engine:

1. Diesel fuel contains components of higher molecular weight on average than those in a gasoline fuel, resulting in higher boiling and condensing temperatures. This causes some hydrocarbon particle to condense on the surface of the solid carbon soot generated during combustion. Most of this is burned as mixing continues and the combustion process proceeds but a small amount is exhausted out the cylinder.

2. The air – fuel mixture in a CI engine is heterogeneous with fuel still being added during combustion. It causes local spot to range from very high to very lean and many flame front exist at the same time unlike the homogenous air – fuel mixture of an SI engine that essentially has one flame front. Incomplete combustion may caused by undermixing or overmixing. With undermixing, in fuel - rich zone some fuel particle do not find enough oxygen to react with, while in fuel - lean zone some local spot will be too lean for combustion to take place properly. With overmixing, some fuel particles may be mixed burned gases and it will therefore lead to incomplete combustion.

3. A small amount of liquid fuel is often trapped on the tip of injector nozzle even when injections stop. This small volume of fuel is called sac volume. This sac volume of liquid fuel is surrounded by a fuel – rich environment and therefore it evaporates very slowly causing hydrocarbon emission in the exhaust.

4. CI engine also has hydrocarbon emission for some of the same reasons as SI engine. For example: flame quenching, crevice volume, oil- film and deposits on the cylinder wall, misfiring.

2.5.6 Particulates

The exhaust of CI engine contains solid carbon soot particles that are generated in the fuel – rich zones within the cylinder during combustion. It is seen as exhaust smoke and it is an undesirable odorous pollution. Maximum density of particulate emissions occur when the engine is under load at WOT. At this condition maximum fuel is injected to supply maximum power, resulting in the rich mixture and poor fuel economy. Soot particles are clusters of solid carbon spheres. This spheres have diameter of 10 nm to 80 nm (1 nm = 10^{-9} m), with most within the range of 15 – 30 nm. The spheres are solid carbon with HC and traces of other component absorbed on the surface. A single soot particle will contain up to 4000 carbon spheres.

Carbon spheres are generated in the combustion chamber in the fuel – rich zones where there is not enough oxygen to convert all carbon to CO_2 as expressed in Eq. (2.8).

$$CxHy + zO_2 \longrightarrow CO_2 + bH_2O + cCO + dC(s) \qquad [2.8]$$

Then, as turbulence and mass motion continue to mix the component in the combustion chamber, most of these carbon particles find sufficient oxygen to further react and are consumed to CO_2 write in Eq. (2.9).

$$C(s) + O_2 \longrightarrow CO_2 \qquad [2.9]$$

Over 90% of carbon particle originally generated within an engine are thus consumed and never get exhausted. If CI engine would operate with an overall stoichiometric air – fuel mixture, instead of overall lean as they do particulate emissions in the exhaust would far exceed acceptable levels. Up to 25% of the carbon in the soot comes from lubricating oil components which vaporize and then react during combustion. The rest are the fuel and the amount to 0.2 – 0.5% of the fuel. Because of the high compression ratio of CI engine, a large expansion occurs during the power stroke, and the gases within the cylinder are cooled by expansion cooling to a relatively low temperature. This cause the remaining high-boiling point component found in the fuel and lubricating oil to condense on the surface of the carbon soot particle. This absorbed portion of the soot particle is called soluble organic fraction (SOF), and the amount is highly dependent on cylinder temperature. At low cylinder temperature (200 ^{0}C), SOF can be 50% of the total mass soot, while at moderate temperature, SOF can be as low as 3%. SOF consists mostly of hydrocarbon component with some hydrogen, SO_2, NO, NO_2, and trace amount of sulfur, calcium iron, silicon, phosphorus, zinc, and chromium. Diesel fuel contains sulfur, calcium, iron, silicon, and chromium. For lubricating oil additives, it contains zinc, phosphorus, and calcium.

2.5.7 Sulfur

Many fuel in CI engine contain small amount of sulfur that when exhausted; contribute to acid rain. Unleaded gasoline generally contains 150 – 600 ppm sulfur by weight. Some diesel fuel contain up to 5000 ppm by weight. At high temperature, sulfur combines with hydrogen to form H_2S and with oxygen to form SO_2 as expressed in Eq. (2.10).

$$\begin{gathered} H_2 + S \longrightarrow H_2S \\ O_2 + S \longrightarrow SO_2 \end{gathered} \qquad [2.10]$$

Engine exhaust can contain up to 20 ppm of SO_2. SO_2 then combines with oxygen in the air to form SO_3 as written in Eq. (2.11).

$$2SO_2 + O_2 \longrightarrow 2\,SO_3 \qquad [2.11]$$

These then combine with water vapor in the atmosphere to form sulfuric acid (H_2SO_4) and sulfurous acid (H_2SO_3), which are the ingredients in acid rain as in Eq. (2.12).

$$\begin{gathered} SO_3 + HzO \sim HzSO_4 \\ SOz + HzO \sim HzSO_3 \end{gathered} \qquad [2.12]$$

The amount of sulfur in natural gas can range from little (sweet) to large (sour) amount. This can be a major emission problem when this fuel is used in a CI engine or any other combustion system.

2.6 BIODIESEL

2.6.1 Introduction of Biodiesel

Biodiesel is a liquid that is gold and dark brown in color depending on the production feedstock. It is practically immiscible with water, has high boiling point and low vapor pressure. The process for making fuel from biomass feedstock used in the 1800s is basically the same as the one used today. The history of biodiesel is more political and economical rather than technological. Gasoline powered automobiles made into history in the early 20th century. Oil companies were obliged to refine so much crude oil to supply gasoline that were left with a surplus of distillate, which is an excellent fuel for diesel engines and less expensive than vegetable oils. Producing biodiesel from vegetable oils is not a new process. The conversion of vegetable oils or animal fats into monoalkyl esters or biodiesel is known as transesterification. On the other hand, resource depletion has always been a concern with regard to petroleum. The farmers also have always sought new markets for their products. Consequently, work has continued on the use of vegetable oils as fuel. Transesterification of triglycerides in oils is not a new process. Duffy and Patrick conducted Transesterification as early as 1853. Life for the diesel engine begins in 1893, when the famous German inventor Dr. Rudolph Diesel published a paper entitled “The theory and construction of a rotational heat engine”. The paper described a revolutionary engine which air would be compressed by a piston to a very high pressure, thus produces high temperature. Dr. Diesel designed the original diesel to run on vegetable oil. Dr. Diesel was educated at the predecessor school to the Technical University of Munich in Germany. In 1878, he was introduced to the work of Sadi Carnot, who theorized that an engine could achieve much higher efficiency than the steam engines of the day. Diesel them sought to apply Carnot’s theory to the internal combustion engine.

The efficiency of the Carnot cycle increases with the compression ratio-the ratio of gas volume at full expansion to its volume at full compression. Nicklaus Otto invented an internal combustion engine in 1876 which is the predecessor to the modern gasoline engine. Otto’s engine mixed fuel and air before their introduction to the cylinder and a flame or spark was used to ignite the fuel air mixture at the appropriate time. However, air gets hotter as it is compressed and if the compression ratio is too high, the heat of compression will ignite the fuel prematurely. The low compression needed to prevent premature ignition of the fuel air mixture limited the efficiency of the Otto engine. Dr. Diesel wanted to build an engine with the highest possible compression ratio. He introduced fuel only when combustion was desired and allowed the fuel to ignite on its own in the hot compressed air. Diesel's engine achieved efficiency level higher than that of the steam engine. Diesel received a patent in 1893 and demonstrated a workable engine in 1897. Today, diesel engines are classified as "compression-ignition" engines, and Otto engines are classified as "spark-ignition" engines. Dr. Diesel used peanut oil to fuel one of his engines at the Paris Exposition of 1900 (Nitschkae and Wilson, 1965). Because of the high temperature, the engine was able to run a variety of vegetable oils including hemp and peanut oil. At the 1911 World's Fair in Paris, Dr. Diesel ran his engine on peanut oil and declared "the diesel engine can be fed with the vegetable oils and will help considerably in the development of the agriculture of the countries with use it".

One of the first uses of transesterified vegetable oil was powering heavy duty vehicles in South Africa before World War II. The name "biodiesel" has been given to transesterified vegetable oil to describe its use as diesel fuel (Demirbas, 2002). Vegetable oils were used in diesel engines until 1920s. In 1920s, diesel engine manufactures altered their engines to utilize the lower viscosity of petrodiesel, rather than vegetable oil. The use of vegetable oils as an alternative renewable fuel competing with petroleum was proposed in the early 1980s. The advantages of vegetable oils as diesel fuel are its portability, ready availability, renewability, higher heat content (about 88% of No. 2 petroleum diesel fuel), lower sulphur content, lower aromatic content and biodegradability. The energy supply concerns of the 1970s renewed interest in biodiesel but commercial production did not begin until late 1990s. Dr. Diesel believed that the engine running on plant oils had potential and these oils could one day be as important as petroleum based fuels. Since 1980s, biodiesel plants have opened in many European countries and some cities have run buses on biodiesel or blends of petro and biodiesels. Most recently, Renault and Peugeot have approved the use of biodiesel in some of their truck engines. Recent environmental and domestic economic concerns have prompted resurgence in the use of biodiesel throughout the world. In 1991, the European Community (EC) proposed a 90% tax deduction for the use of biofuels, including biodiesel. Biodiesel plants are now being built by several companies in Europe; each of these plants will produce up to 1.5 million gallons of fuel per year. The European Union accounted for nearly 89% of all biodiesel production worldwide in 2005.

2.6.2 Biodiesel Concept

Energy is used mainly for transport, and partly for services, industry and commerce. Today the world's demand for energy is fulfilled by fossil fuels, basically by oil, petroleum and natural gas. Petroleum-based fuels are limited reserves concentrated in certain regions of the world. The scarcity of conventional fossil fuels, growing emissions of combustion generated pollutants and their increasing costs will make biomass sources more attractive (Sensoz, 2000). Petroleum-based fuels are limited reserves concentrated in certain regions of the world. On the other hand, biomass which is gaining people interest lately has the properties of being a biomass source and a carbon-neutral source (Dowaki, 2007). In the future, the price will increase along with interest and needs when there is reduction in oil availability thus looking for other energy resources is really a necessity. Against this background, new energy sources are needed to fulfill the demand. Experts suggest that current oil and gas reserves would suffice to last only for a few more decades. To meet the rising energy demand and replace reducing petroleum reserves, fuels such as biodiesel and bioethanol are in the forefront of alternative technologies. Accordingly, the viable alternative for compression-ignition engines is biodiesel. Biodiesel is briefly defined as the monoalkyl esters of vegetable oils or animal fats. Diesel fuel can also be replaced by biodiesel made from vegetable oils but biodiesel is the best candidate for diesel fuels in diesel engines. Biodiesel burns like petroleum diesel as it involves regulated pollutants. In fact, biodiesel probably has better efficiency than gasoline. Biodiesel exhibits great potential for compression-ignition engines. It is now mainly being produced from soybean, rapeseed and palm oils. The Higher Heating Values (HHVs) of biodiesel are relatively high. The HHVs of biodiesel (39 to 41 MJ/kg) are slightly lower than those of gasoline (46 MJ/kg), petrodiesel (43 MJ/kg), or petroleum (42 MJ/kg), but higher than coal (32 to 37 MJ/kg).

Biodiesel is pure or 100%, biodiesel fuel. It is referred as B100 or “neat” fuel. A biodiesel blend is pure biodiesel blended with petrodiesel. Biodiesel blends are referred to as BXX. The XX indicates the amount of biodiesel in the blend (*i.e.*, a B80 blend is 80% biodiesel and 20% petrodiesel). The majority of energy demand is fulfilled by conventional energy sources like coal, petroleum, and natural gas. Petroleum-based fuels are limited reserves concentrated in certain regions of the world. These sources are on the verge of reaching their peak production. The scarcity of known petroleum reserves will make renewable energy sources more attractive (Sheehan, 1998). World energy demand continues to rise. The most feasible way to meet this growing demand is by using alternative fuels. One such fuels that exhibits great potential is biofuel, in particular biodiesel (Fernando, 2006). The term biofuel can refer to liquid or gaseous fuels for the transport sector that are predominantly produced from biomass (Demirbass, 2006). In developed countries there is a growing trend towards using modern technologies and efficient bioenergy conversion using a range of biofuels, cost wise; it begins to compete with fossil fuels (Puhan, 2005). It is well known that transportation is almost totally dependent on fossil, particularly petroleum based fuels such as gasoline, diesel fuel, liquefied petroleum gas (LPG), and natural gas (NG). An alternative fuel to petrodiesel must be technically feasible, economically competitive, environmentally acceptable, and easily available. The current alternative diesel fuel can be termed biodiesel. Using biodiesel may improve emissions levels of some pollutants and deteriorate others. However, for quantifying the effect of biodiesel it is important to take into account several other factors such as raw material, driving cycle and vehicle technology. Use of biodiesel will allow a balance to be sought between agriculture, economic development, and the environment (Meher, 2006).

2.6.3 Definition of Biodiesel

Biodiesel is methyl or ethyl ester of fatty acid made from used vegetable oils (both edible and non edible) and animal fats. Biodiesel is an alternative fuel for diesel engines that is produced by chemically reacting vegetable oil or animal fat with an alcohol such as methanol. Chemically, biodiesel is defined as the monoalkyl esters of long-chain fatty acids derived from renewable biolipids. Biodiesel is typically produced through the reaction of a vegetable oil or animal fat with methanol or ethanol in the presence of a catalyst to yield methyl or ethyl esters (biodiesel) and glycerine (Demirbas, 2002). Fatty acid methyl esters or biodiesels are produced from natural oils and fats. Generally, methanol is preferred for transesterification because it is less expensive than ethanol (Graboski and McCormick, 1998). In general term, biodiesel may be defined as a domestic and renewable fuel for diesel engines derived from vegetable oil which meets the specification of ASTM D 6751 (Fukuda, 2001).

Biodiesel consists of alkyl esters, which are produced from the transesterification reaction between triglycerides and alcohol. In experimental studies, the final product is term as fatty acid alkyl esters or fatty acid methyl esters (FAME) instead of biodiesel unless it meets the specification of ASTM D6751 (Lois, 2007). Biodiesel, in application as an extender for combustion in CIEs (diesel), possesses a number of promising characteristics, including reduction of exhaust emissions (Dunn, 2001). Chemically, biodiesel is referred as the monoalkyl esters, especially methyl ester of long chain fatty acids derived from renewable lipid sources via a transesterification process.

Table 2.3: Technical properties of biodiesel

Technical Properties	**Description**
Common name	Biodiesel (bio-diesel)
Common chemical name	Fatty acid (m)ethyl ester
Chemical formula range	$C_{14} – C_{24}$ methyl ester or $C_{15} – C_{25} H_{28-48} O_2$
Kinematic viscosity range (mm^2/s, at 313K)	3.3-5.2
Density range (kg/m^3, at 288K)	860 – 894
Boiling point range (K)	>475
Flash point range (K)	430 – 455
Distillation range (K)	470 – 600
Vapour pressure (mm Hg, at 295K)	<5
Solubility in water	Insoluble in water
Physical appearance	Light to dark yellow, clear liquid
Odor	Light musty or soapy odor
Biodegradability	More biodegradable than petroleum diesel
Reactivity	Stable but avoid oxidation agents

Source: Demirbas (2003)

Biodiesel is a mixture of methyl esters of long-chain fatty acids like lauric, palmitic, stearic, oleic, *et cetera*. Typically examples are rapeseed oil, canola oil, soybean oil, sunflower oil, palm oil and their derivatives from vegetable sources. Beef and sheep tallow and poultry oil from animal sources and cooking oil are also sources of raw materials. The chemistry of conversion into biodiesel is essentially the same. Oil or fat reacts with methanol or ethanol in the presence of sodium hydroxide or potassium hydroxide catalyst to form biodiesel, methyl esters and glycerine.

2.6.4 Biodiesel as an Alternative to Diesel Fuel Engine

Biodiesel is a processed fuel that can be readily use in diesel-engine vehicles, which distinguishes biodiesel from the straight vegetable oils or waste vegetable oils used as fuels in some modified diesel vehicles. In general, the physical and chemical properties and the performance of ethyl esters are comparable to those of the methyl esters. Methyl and ethyl esters have almost the same heat content. The viscosities of ethyl esters are slightly higher and the cloud and pour points are slightly lower than those of the methyl esters. Engine test have demonstrated that methyl esters produce slightly higher power and torque than ethyl esters. Biodiesel is a clear amber-yellow liquid with a viscosity similar to that of petrodiesel. Biodiesel is non-flammable and in contrast to petrodiesel is non-explosive, with a flash point of 423K for biodiesel as compared to 337K for petrodiesel. Unlike petrodiesel, biodiesel is biodegradable and non-toxic and it significantly reduces toxic and other emissions when burned as a fuel. Currently, biodiesel is more expensive to produce than petrodiesel, which appears to be the primary factor in preventing its more widespread use.

Current worldwide production of vegetable oil and animal fat is not enough to replace liquid fossil fuel use (maximum replacement percentage: 20% to 25%) (Bala, 2005). Methyl esters of vegetable oils (biodiesels) have several outstanding advantages among other new-renewable and clean-engine fuel alternatives. Methanol as a monoalcohol is generally used in the transesterification reaction of triglycerides in the presence of alkali as a catalyst (Clark, 1984). Methanol is a relatively inexpensive alcohol. Several common vegetable oils such as sunflower, palm, rapeseed, soybean, cottonseed and corn oils and their fatty acids can be used as the sample of vegetable oil. Biodiesel is easier to produce and cleaner with equivalent amount of processing when starting with clean vegetable oil. Tallow, lard and yellow grease biodiesels require additional processing at the end of the transesterification process due to the presence of high free fatty acids. Diesel derived from rapeseed oil is the most common biodiesel available in Europe while soybean biodiesel predominates in the United States.

Biodiesel is a plant-derived product and contains oxygen in its molecules, making it a cleaner-burning fuel than petrol and diesel (Sastry, 2006). Biodiesel is a clean burning alternative fuel produced from domestic, renewable resources that are much more efficient to produce and use than gasoline. The development history of biodiesel is more political than technological. The actual process for making biodiesel was originally developed in the early 1800s and has basically remained unchanged. It was the political and economic influences of industrial leaders during the 1920s and 1930s that caused the fuel trends to favour the use of petroleum-based fuels as opposed to agricultural fuels.

CHAPTER 3

EXPERIMENTAL SETUP

3.1 INTRODUCTION

This chapter shows the engine and instrumentation used to conduct the experimental work and the progress picture for fabrication work. It also contains the experiment setup and the startup procedure to run the experiment.

3.2 ENGINE AND INSTRUMENTATION

3.2.1 Engine

The experiment used Mitsubishi diesel engine 4D68, 4 in line number of cylinder and using single type of exhaust gas recirculation system (EGR) diaphragm type. This direct injection diesel engine has a bore of 82.7 mm and stroke 93 mm. The injection pressure is more or less set to 206 bars and a compression ratio of 22.4:1. It is an air-cooled, low-speed and the maximum power was 64.9 kW at 4500 rpm.

Figure 3.1: Diesel Engine Mitsubishi 4D68

Source: Engine performance test lab

Table 3.1: The specification of Diesel Engine 4D68

Description	**Specifications**
Type	Diesel Engine
Number of cylinder	4 in line
Bore (mm)	82.7
Stroke (mm)	93
Total Displacement (dm^3)	1.998
Compression Ratio	22.4
Cooling System	Water cooled
EGR Type	Single type
Lubrication System	Pressure

Source: Mitsubishi engine 4D68 workshop manual

3.2.2 Dynamometer

Dynamometer is used to measure engine power and vehicle speed under various operating conditions. When engine started, the shaft would drive the driven wheel to rotate the roller and the torque generated and it will be absorbed by the chassis dynamometer. The vehicle remained static while the engine and other components operated as if on road. Eddy-Current 150kW dynamometer model ECB-200F from Dynalec Controls was used for this experiment.

Figure 3.2: Eddy Current 150kW dynamometer

3.2.3 Pressure Sensor

A pressure sensor is used to measure the pressure inside the compression ratio by removing the glow plug at the first cylinder and fixed the pressure sensor. This experiment also used Kistler Model 6041 A pressure sensor. The pressure sensor needs a cooling system to reduce the heat because it fixed into the combustion chamber. It produces a high heating and can affect the sensitivity then will damage the transducer.

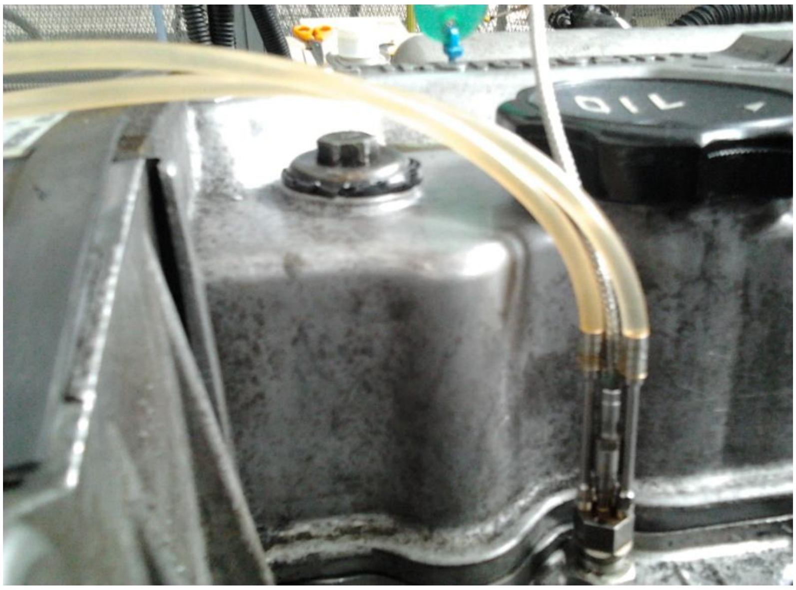


Figure 3.3: Pressure Sensor Kistler Model 6041A

Table 3.2: Pressure transducer specification

Description	**Unit**	**Range**
Range	bar	0 – 250
Overload	bar	300
Sensitivity	P C/bar	≈ 19/89
Operating Temp-range	^{0}C	-50 ~ 350
Weight (sensor only)	g	10

Source: Pressure Sensor Kistler Model 6041A technical data

3.2.4 Thermocouple

There are various types of thermocouple to measure temperature. Thermocouple made of two dissimilar metals that are joined and it creates two junctions. The junction is called ‘sensing junctions’. It is use to measure the temperature differences. Electric current will flow through the circuit and the temperature is measure in milivolt. A signal converter will make the milivolt reading and compare the milivolt value of the reference junction at a specific temperature.

For this experiment the type of thermocouple is type-K. It is the most popular due to the interchangeability advantage, provided with standard connectors and can measure a wide range temperatures.

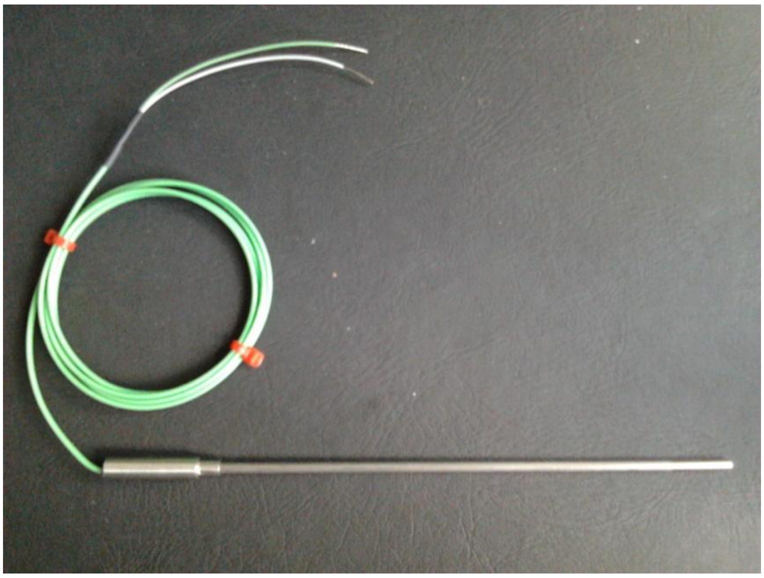


Figure 3.4: Thermocouple K-Type

Table 3.3: Thermocouple Specification

Description	**Value**
Wire Material	Chrome (+ve), Alumel (-ve)
Range	-328 to 2300 F
ANSI Standard Limits of Error	±4.0 F or 0.75%

Source: Thermocouple a quick guide

3.2.5 Data Acquisition System

The Data acquisition system is used to measure the temperature and the reading on cylinder pressure from the engine while conducting the experiment. Dewetron mode Dewe-800series is used in this experiment.

Table 3.4: Data Acquisition System specification

Model	**Dewe – 800 series**
Hard Disk	250GB
Data Throughput	Type 70 MB/s^2
Processor	Intel Celeron M1.8GHz
Ram	2GB
Operating System	Window Xp
Total PCI-Slots	2 full/ 1 half length
Slots for DAQ and DAD modules	16
Software	Dewesoft

Source: DEWETRON Technical Reference Guide

3.2.6 Gas Analyzer

Gas analyzer is the measuring device that is used to measure the emission on the exhaust gas. It can be measured and calculated all the parameters and display the percentage. The model is Kane Automotive Gas Analyzer.

Table 3.5: Gas Analyzer Specification

Parameter	**Resolution**	**Accuracy**	**Range**
Carbon Monoxide	0.01%	± 5% of Reading	0-10%
Oxygen	0.01%	± 5% of Reading	0-21%
Carbon Dioxide	0.1%	± 5% of Reading	0-16%
Oxides of Nitrogen	1 ppm	± 4% of Reading	0-5000 RPM

Figure 3.5: Gas Analyzer

3.2.7 Temperature Indicator and Flow rate meter

The temperature from the water in and water from the heat exchanger is measured using temperature indicator. Meanwhile the flow rate meter is used to measure volume flow rate from the water in source from the pipe. These two devices are important to measure the heat.

Figure 3.6: Temperature Indicator

Figure 3.7: Flow Rate Meter

3.2.8 Heat Exchanger

The design of heat exchanger is done using the Shell and Tube Heat Exchanger, the simplest form of a heat exchanger. Parallel flow heat exchanger, the two fluid streams enter together at one end, flow through in the same direction, and leave together at the other end. One fluid stream flows through the tubes while the other flows on the other side of the shell across or along the tubes. Figure 3.8 shows the diagram of the shell and tube heat exchanger. The heat exchanger is made of steel. The diameter for the shell is 60mm and diameter for tube inside the shell is 20mm. The overall length of heat exchanger is 350mm. All parts to merge the heat exchanger were done using arc welding. The overall heat transfer coefficient is 25.85 W/m^{2}.^{o}C. Since gases have very low thermal conductivity.

For the new exhaust gas recirculation (EGR) system, it has been attached on the original EGR system. It requires a large space for new EGR because of the heat exchanger. The cooling system in heat exchanger using water is passed directly from the tap then enter to the heat exchanger and flows to the outside. There are having four thermocouples to measure the temperature at the inlet and outlet of water and heat exchanger. Cooling water also has flow rate meter to measure a volume flow rate.

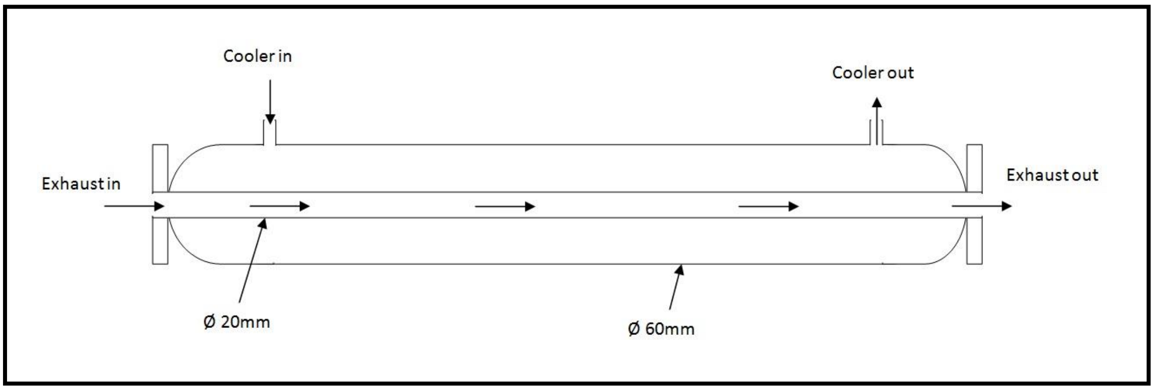


Figure 3.8: Shell and tube heat exchanger system

3.2.8 The fabrication work of heat exchanger

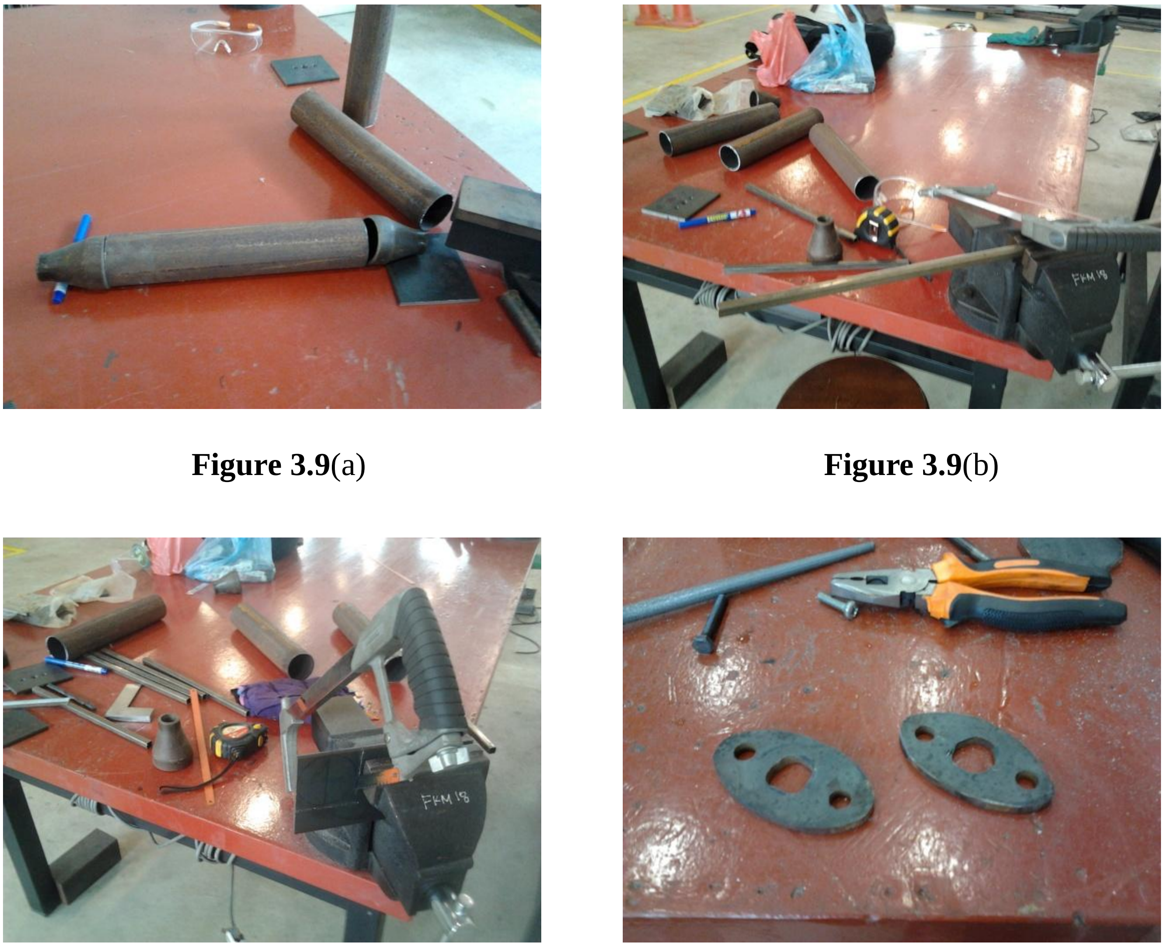


Figure 3.9(a)

Figure 3.9(b)

Figure 3.9(c)

Figure 3.9(d)

Figure 3.9(a), 3.9(b), 3.9(c) and 3.9(d): Cut the raw material and fabrication work

Figure 3.10(a) **Figure 3.10**(b)

Figure 3.10(a) and 3.10(b): Heat Exchanger before welding

Figure 3.11(a) **Figure 3.11**(b)

Figure 3.11(a) and 3.11(b): Use arc welding

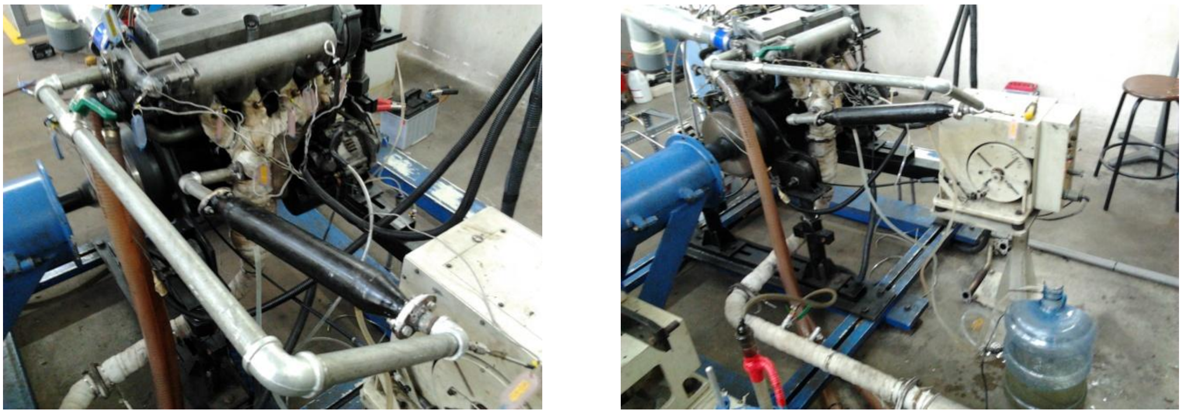


Figure 3.12(a) **Figure 3.12**(b)

Figure 3.12(a) and 3.12(b): New EGR system attach to the engine.

3.3 EXPERIMENTAL SETUP

The experimental study was carried out using a Mitsubishi 4D68 direct injection 2.0L four stroke engine. This engine used diaphragm type EGR. An exhaust gas analyzer *KANE Automotive* system completed with a 3-meter sampling probe was used for emission measurement. The sampling probes of smoke meter and gas analyzer were mounted at the centre of the engine exhaust pipe. The water jacket was used around the particle trap cylinder to cool the exhaust gas and caused the particle to fall at the bottom of the trap. After that, clean exhaust gas entered the exhaust analyzer and the gas content was analyzed. The characteristic and amount of content was measured instantly by reading the exhaust analyzer display unit. The engine speed was set at 1000, 1500, 2000, 2500, 3000 and 3500 rpm using original EGR, new EGR with heat exchanger and without EGR at each speed. Besides that the experiment was conducted at engine speed 2000 rpm too, and the emissions were measured using gas analyzer with variable EGR opening valve at 100%, 75%, 50% and 25% and difference water flow rate.

For crank angle measurement and combustion characteristic, the engine was mounted with Kistle CAM crank angle encorder type 2613B connected to Kistler signal conditioner type 2613B2. From the signal conditioner, crank angle encoder is connected to PC DEW-800 with connecting cable, 2613B3. Water cooed piezoelectric pressure transducer (kistler 6041A) to measure combustion pressure was bolted at the cylinder No. 1 in order to measure the temperature it used, Thermocouple RS K-Type, and reading by using temperature indicator. The engine was directly coupled to an eddy-current brake ECB dynamometer equipped with a Dynalec load controller.

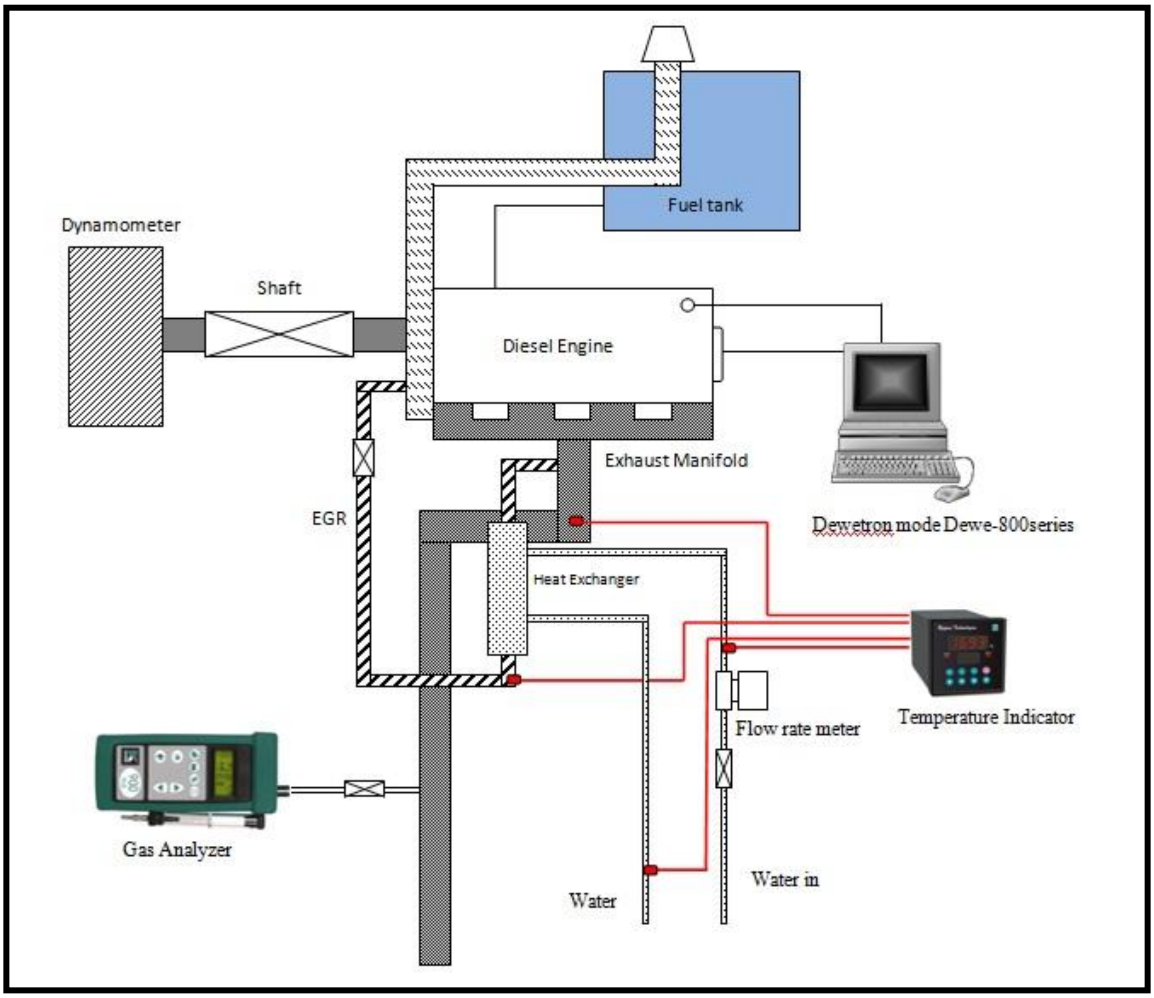


Figure 3.13: Experiment setup

3.4 ENGINE STARTUP PROCEDURE

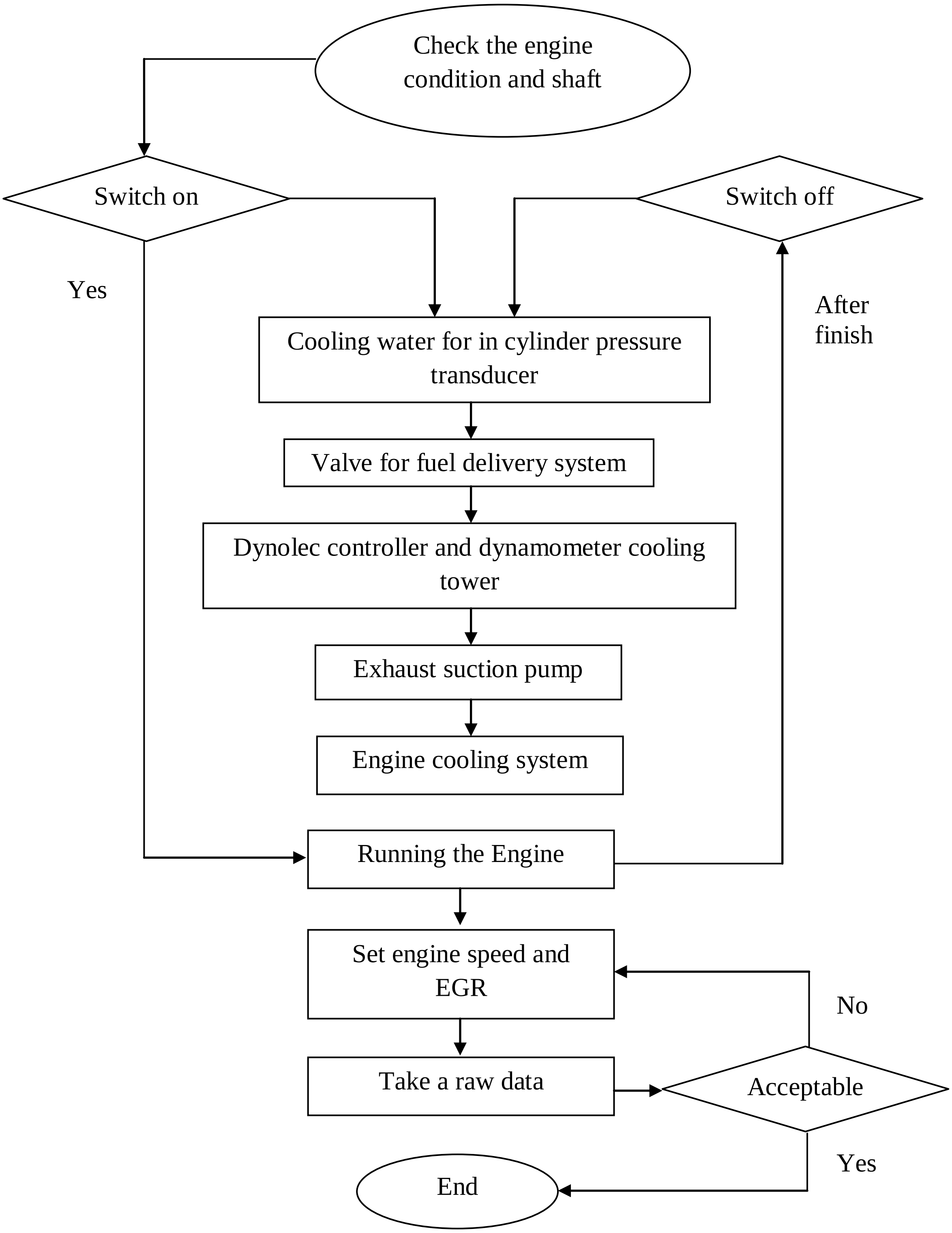


Figure 3.14: Engine start up flowchart

Flow chart is the simplest way to show the overall steps from beginning until the end of the experiment. First, check the engine condition including shaft. The most important thing is to check the engine oil level. Shaft from engine to dynamometer must be fixed to their placed to avoid damage to the shaft.

After the cooling water system for cylinder pressure transducer is switched on, the valve and also the fuel pump are shut for the fuel delivery system. From this valve, the researcher will know which fuel he needs to use. Then shut on the dynolec controller and the dynamometer cooling tower, on the exhaust suction pump.

Next, the engine is started and switched on the cooling system, the pump and the fan few minutes after that. The engine speed then is set according to the experiment test condition. Exhaust gas analyzer is installed and data is collected. The experiment will be continued to the next condition if the result is acceptable. Switch off the engine and the equipment when experiment is done.

3.5 LITERATURE ANALYSIS

Literature analysis is a combination of idea and elaboration of the literature that have been collected from various sources such as articles, journals, books, and other result analysis from previous research as guide for the project. All information related to the project came from valid and reliable sources. The objective is to design a novel system of exhaust gas recirculation system (EGR) in diesel engine. The literature review covers on the exhaust gas recirculation (EGR) system, diesel engine, heat exchanger and the emissions. The methods used by previous study were explained and were used to collect data for the project as well after consulting the project’s supervisor.

3.5.1 Journals

Journal provides vast information and knowledge for this project. There are a lot of journals which are easily accessible for student like journal from Society of Automotive Engineering (SAE), Japan Society of Mechanical Engineering (JSME) and other online journal and researches that can be referred to. Journals provide articles, theses and dissertations complete with the data and result analysis that are useful for comparison and guideline for the project. Thousands of journals can be accessed and downloaded using the university's server at library.

3.5.2 Previous Research

By reviewing previous researches and study done on the particular subject or somehow related, researcher will gain more insights and perspectives in completing his project. Results, data and methods from previous research can be applied again in the current research. There are many research papers available online and in fact, it can be downloaded online as well. However, validity and reliability of papers retrieved online are questionable.

3.5.3 Books

Referring to various textbooks is the most common method in literature review. Book like Internal Combustion Engine book, diesel engine book, emission book and also the heat exchanger book really helps a lot in the process of completing the project. Apart from going to the library, books can be obtained from the lecturer or modules used in class.Detail information such as concept, analysis, and calculation method can easily be found in the source.

3.5.4 Project Supervisor

A project supervisor does not only supervise student's work but to guide and facilitate the student as well. It is important to communicate frequently with the supervisor to make sure the progress of the project is smooth. So, it is a role of the project supervisor to help in understanding the information and give the opinion to make sure the gathering knowledge is really useful to the project and also for written in the final project report. In other hand, discussion with the supervisor which is expert in the related field gives clear guideline to conduct a smooth project's progress.

CHAPTER 4

RESULT AND ANALYSIS

4.1 INTRODUCTION

This chapter presents the result and discussion for the experiment about the emission gas using gas analyzer, also the engine performance and the analysis from the result. Data is analyzed from the result in graph.

4.2 RESULT AND DISCUSSION

Table shows the fuel properties for palm methyl ester and diesel fuel correspondingly. The data shows that PME100 has higher density compared to diesel

Table 4.1: Fuel properties for diesel and PME100

Property	**Diesel**	**PME100**
Heat value (MJ kg-1)	45.28	41.3
Cloud point (°C)	18	13
Density @ 15°C (kg/m^3)	853.8	867
Flash point (°C)	93	165
Pour point (°C)	12	7.0
Cetane Number	54.6	67

4.2.1 Effect on Emission Characteristics

The formation levels of NOx emission are highly dependent on in-cylinder temperature, the oxygen concentration and residence time for the reactions to take place. These listed factors provide some convenience answers in how to reduce the emission levels of NOx. As an example, the in-cylinder temperature can be manipulated by lowering its temperature using recirculate exhaust gas to minimize the oxygen content in the cylinder.

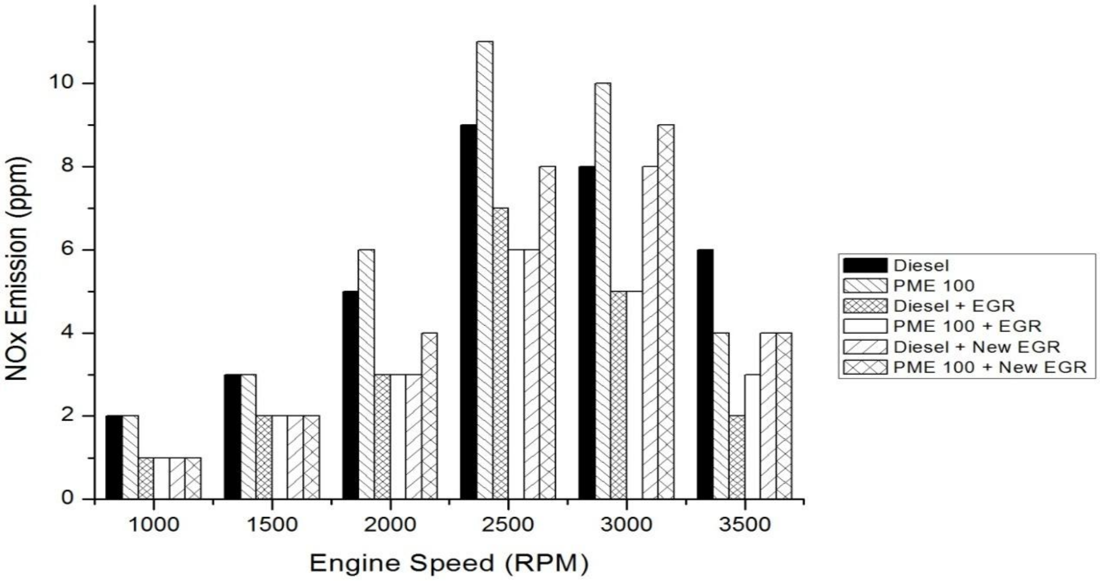


Figure 4.1: Formation levels of NO_x emission with engine speed

Figure 4.1 shows the variation of oxides of nitrogen NOx emission with engine speed for fuels, diesel and PME 100 operating with or without EGR and with new EGR. It is found that PME 100 has higher formation levels of NO_x emission as compared to diesel. This mainly attributed to the higher oxygenated nature content in PME 100 which leads to highly-oxidized flammable in the cylinder as well increase the exhaust temperature.

Generally the NOx emission tends to reduce significantly when using EGR. It’s found that content of NOx emissions increase due to the increasing engine speed and reduce after 2500 rpm. The NOx emission using EGR is lower than NOx emission without using EGR. It is the same to both fuels. Comparing the original

EGR and the new EGR; the result shows that the original EGR is lower NOx emission compared to the new EGR. This is because the exhaust temperature that recirculated back to the combustion chamber is lower than the new EGR. The production of NOx is influenced by the concentration of oxygen and the maximum charge in the cylinder. While the O_2 concentrated and the maximum temperature are low, the reduction of the NOx was produced due to poor combustion.

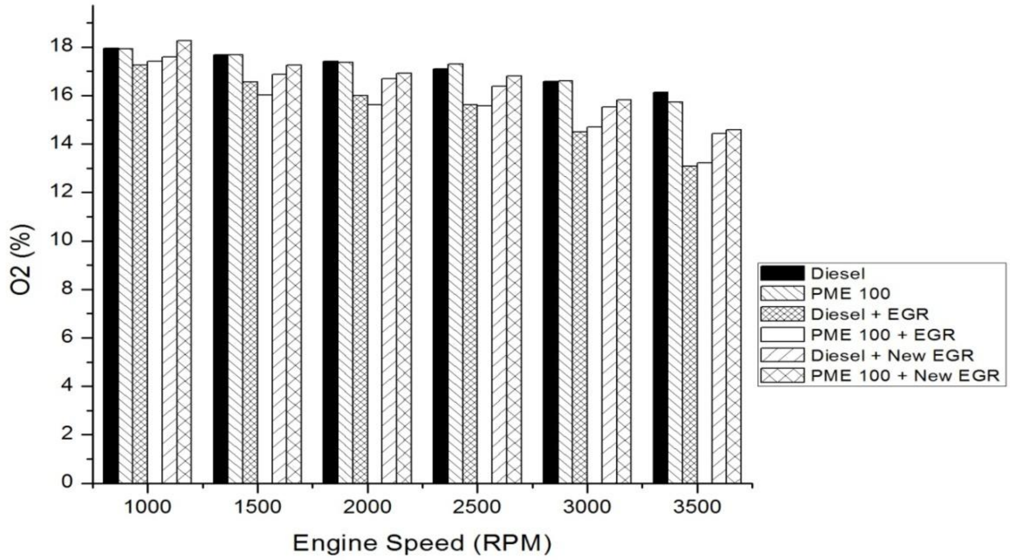


Figure 4.2: Oxygen emission with engine speed

Figure 4.2 shows difference of O_2 emissions when using EGR, new EGR and without using EGR with engine operates using diesel fuel and PME100. The quantity of O_2 emissions when operating using EGR is lower to the quantity of O_2 using new EGR and without EGR. This is because when engine operates with EGR, the CO_2 gas which recirculated from exhaust replaced the amount of O_2 which enters from the intake manifold. From that situation, the amount of O_2 using EGR is lower. The EGR has their own heat exchanger to reduce the temperature so it can reduce the oxygen concentration in the cylinder. For the new EGR, result shows the O_2 emissions value is between using EGR and without EGR. This is because the heat exchanger is not cool enough to reduce the O_2 concentration in the cylinder.

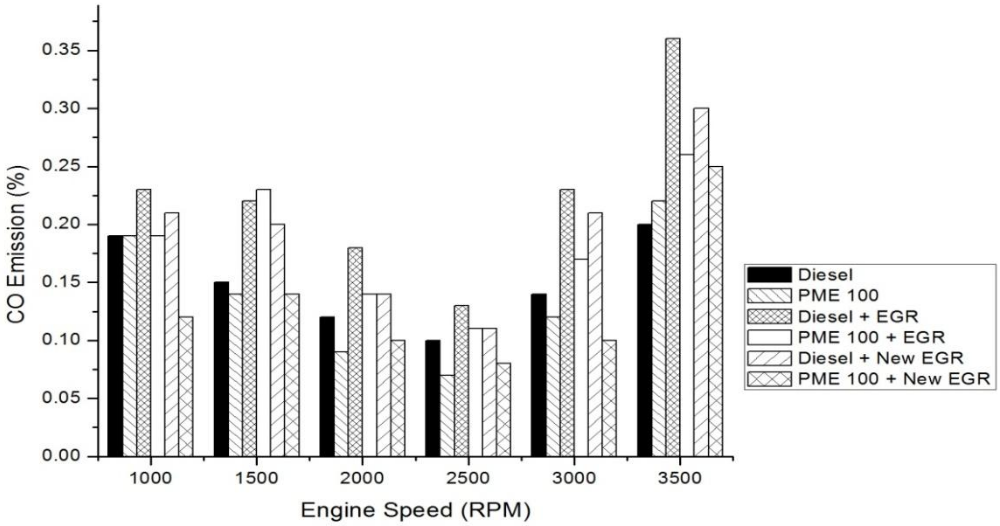


Figure 4.3: Emission level for carbon oxide with engine speed

Figure 4.3 indicates the emission level for carbon oxide. It will be decreased due to engine speed until 2500 rpm. Generally, CI engine operates with lean mixtures and hence the CO emission would be low. CO usually formed due to the lack of oxygen. CO emission is toxic and must be controlled. CO emission level will be increase start from engine speed at 2500rpm until 3500 rpm. It can be seen that using EGR increased the CO emissions. With the use of EGR the intake charge temperature decreased. Because of this, the cylinder charge temperature decreased and led to poor combustion, and then the CO emission will be increased.

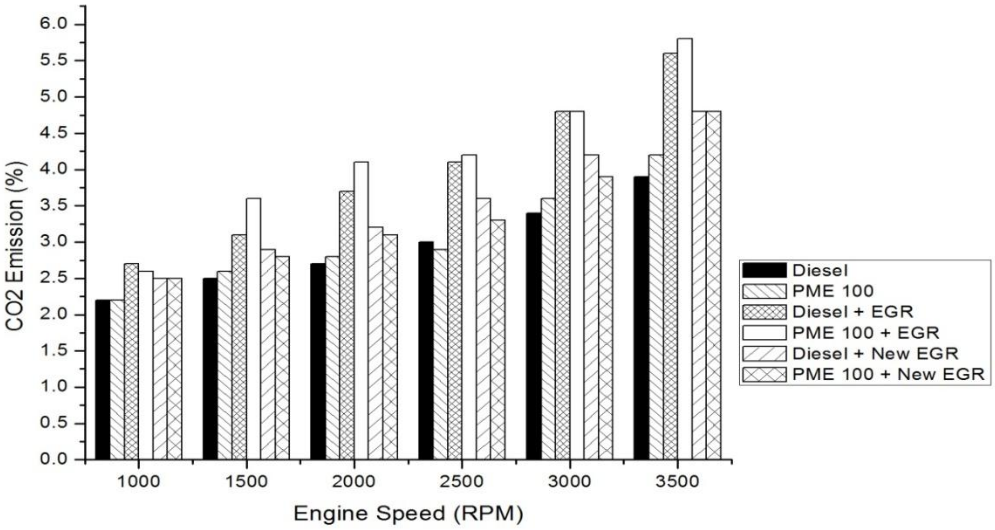


Figure 4.4: Variations of carbon dioxide (CO_2) content with engine speed

Figure 4.4 shows the variations of carbon dioxide (CO_2) using original EGR, new EGR and without EGR operating by using diesel fuel and PME100 fuel. The CO_2 emission was increased due to engine speed. From the graph above, at the starting engine speed 1000 rpm, the quantity of CO_2 emission for the engine by using EGR is higher than the CO_2 emission for the engine without using EGR. This is because the air fuel ratio mixture which operates using EGR is richer than the air fuel ratio mixture which operates without using EGR. The graph also shows that engine operate with the new EGR using heat exchanger produced low CO_2 emission compared to the engine operated using original EGR. This is because using the heat exchanger decreased the intake charge temperature and the maximum temperature in the cycle. From that it might freeze the CO oxidation process to CO_2 as the burned gas temperature fell.

4.2.2 Exhaust Temperature

The exhaust temperature is the comparison between the original EGR and the new EGR.

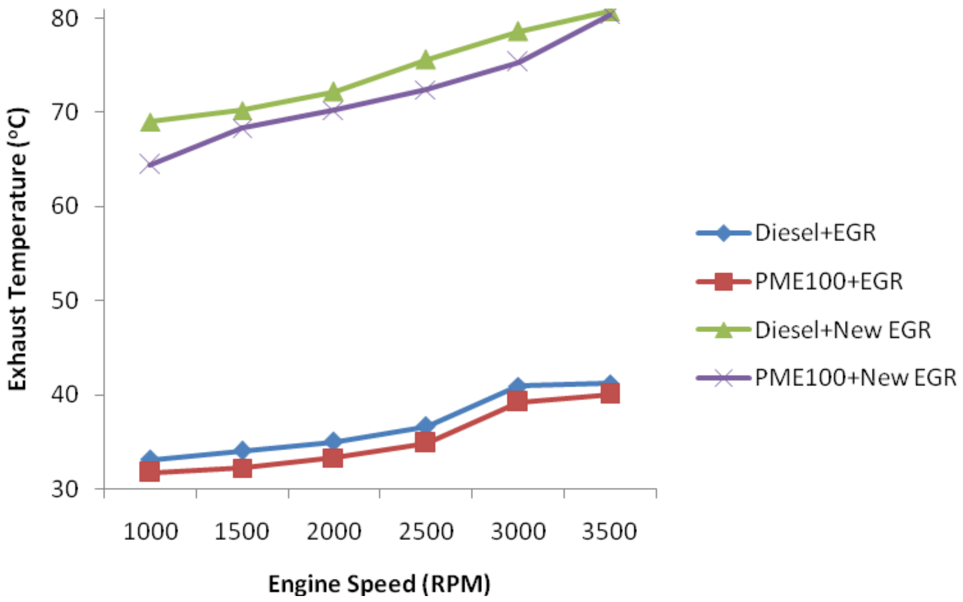


Figure 4.5: Graph exhaust temperature versus engine speed (RPM) at variable RPM

Figure 4.5 shows the exhaust temperature when engine is operated using original EGR and new EGR. The engines operated with diesel fuel and PME100 fuel. The graph shows the exhaust temperature using the original EGR is lower than the temperature using new EGR. The range of exhaust temperature is about 40^{o}C for diesel fuel and 35 ^{o}C for PME 100. This is because the new EGR used the heat exchanger shell and tube type cooling by water but does not enough to reduce the exhaust temperature. From the graph, it can be seen that the exhaust temperature using diesel fuel is higher than using the PME 100 fuel. The temperature different is about 1 ^{o}C until 5 ^{o}C.

4.2.3 Energy transfer by heat

In another experiment, engine operates using diesel fuel and PME100 and engine speed is set at 2000 rpm and zero loading. Experiment was conducted to measure the heat transfer rate of cooling water the heat exchanger. Besides that, this experiment was done to measure the gas emissions. Data collected is the emissions level at difference water volume flow rate and variable percentage of EGR open by using flow rate meter. This experiment used a flow rate meter to measure the flow rate of water from the tab through the heat exchanger. It is also used thermocouple to measure the temperature inlet and outlet of water and exhaust.

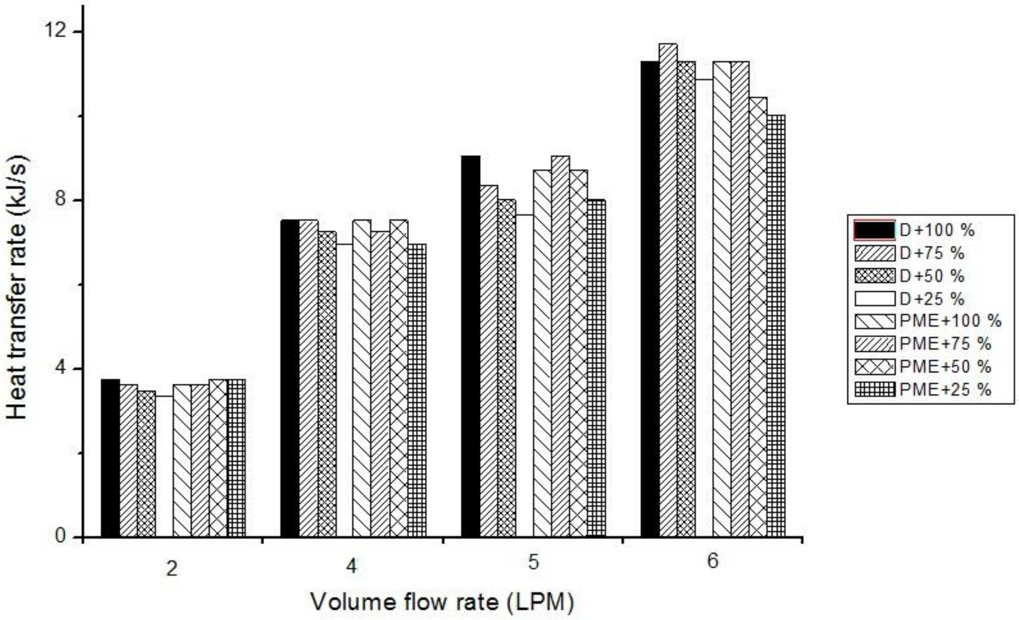


Figure 4.6: The heat transfer rate with the volume flow rate

Figure 4.6 shows the heat transfer rate with volume flow rate at variable conditions of EGR percentage. This experiment was conducted with engine operated using diesel fuel and the PME100 with engine speed is 2000 rpm with zero loading. It can be seen that heat transfer rate increased due to the increasing flow rate of water. The amount of heat transferred per unit time is called heat transfer rate, when the flow rate of water is slow, the heat transfer rate is low too. It affects the rate of heat transfer by conduction. In conduction, heat is transferred from a hot location to a cold location. Heat will continue transferring as long as there is a difference in temperature between the two locations. The rate of conductive heat transfer between two locations is affected by the temperature difference between the two locations.

4.2.4 Emission level for the new EGR system

This experiment was conduct using new exhaust gas recirculation system. The EGR was using the heat exchanger to cool the exhaust gas, flow rate to measure the exhaust flow rate going into the EGR, water flow rate meter to measure the water flow rate of cooling water system, thermocouple indicator to measure the inlet and out let of water and the exhaust temperature. The CI engine operates at 2000rpm with 0% loading by using diesel fuel and PME100 fuel.

Gas analyzer was used to determine the volumetric content of emission. The emission is carbon monoxide (CO), carbon dioxide (CO_2), oxides of nitrogen (NO_x) and oxygen (O_2). The exhaust gas sample taken from the exhaust manifold and was analyzed using the pipe. The EGR percentage is measured using the flow meter and it controls the valve opening. The full opening of valve is 100% of EGR and the flow rate is 23m/s. second is 75% of EGR, 17.25m/s, then 50% of EGR, 11,.5m/s and last 25% of EGR 5.75m/s.

Figure 4.7 (a), 4.7(b), 4.7(c) and 4.7(d) show the emission level of nitrogen oxide (NO_x), carbon monoxide (CO), carbon dioxide (CO_2) and oxygen (O_2) with volume flow rate measured at 100%, 75%, 50% and 25% of EGR valve opening. Diesel fuel and PME100 fuel were used for the engine. All the experiment used EGR, so it can be seen that the emission level for all gas is low compared to not using EGR. Figure 4.7(a) on the other hand elucidates that NOx emission is higher when engine is operated using PME100 compared to diesel fuel. From the literature review, it stated that NOx emit from biodiesel and its level is higher when compared to diesel (Lapuerta, 2008). Meanwhile, carbon monoxide emission shown in the graph is between 0.01% and 0.09%, Figure 4.7 (b). Figure 4.7(c) shows the CO_2 emissions from fuel diesel and PME100 where the level varied between 0.8% until 2.4%. Finally in the last figure 4.7(d), the oxygen concentration is in range of 16%

to18%. The overall result shows that using EGR at diesel operated engine or PME100 engine with the engine speed is at 2000rpm and 0% loading, the emission level is lower.

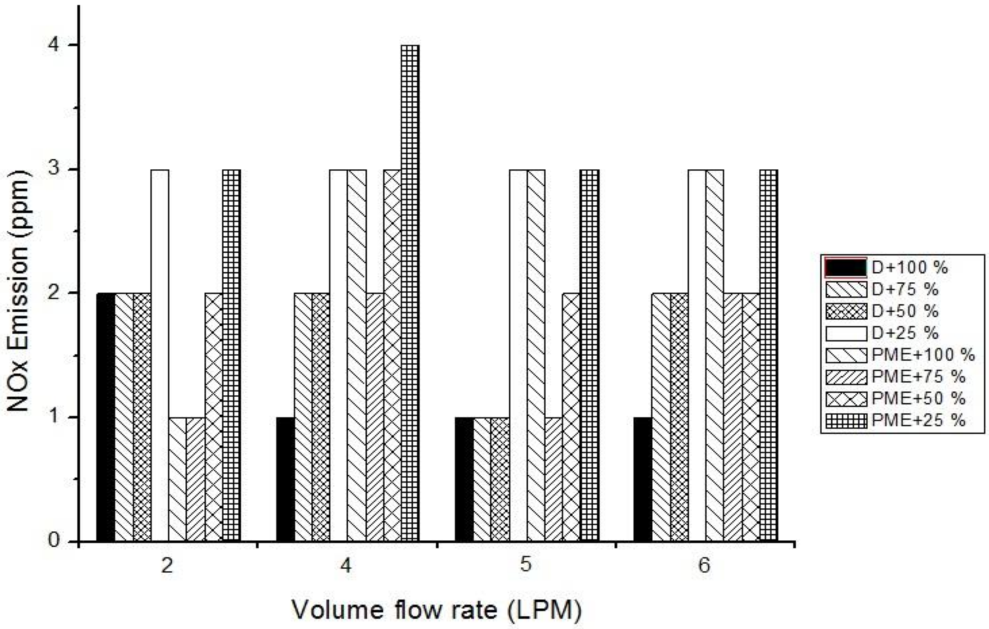


Figure 4.7(a): Nitrogen oxide emission level with volume flow rate at the different EGR percentage using diesel and PME100

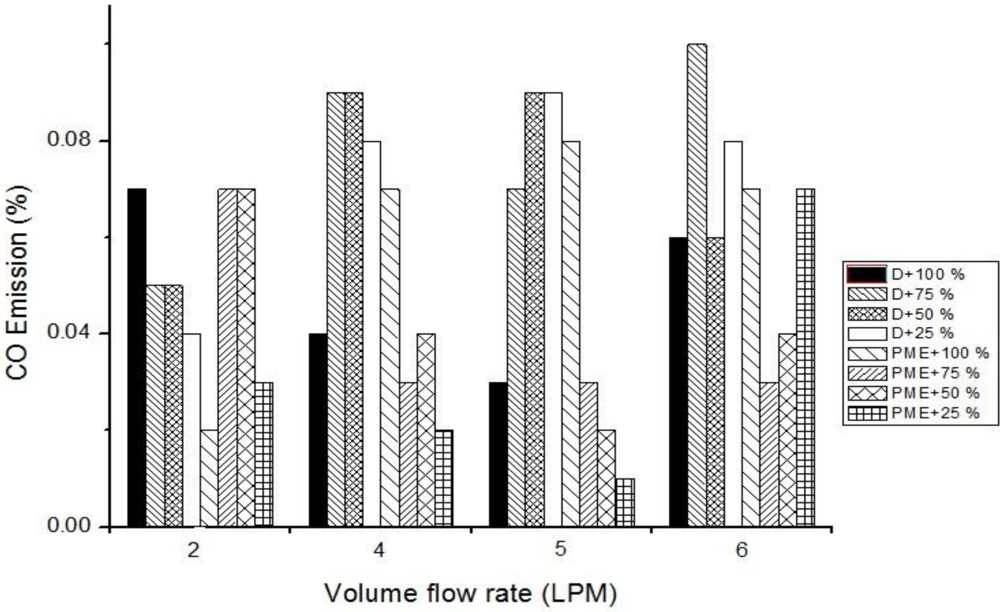


Figure 4.7(b): Carbon monoxide emission with volume flow rate at the different EGR percentage using diesel and PME100

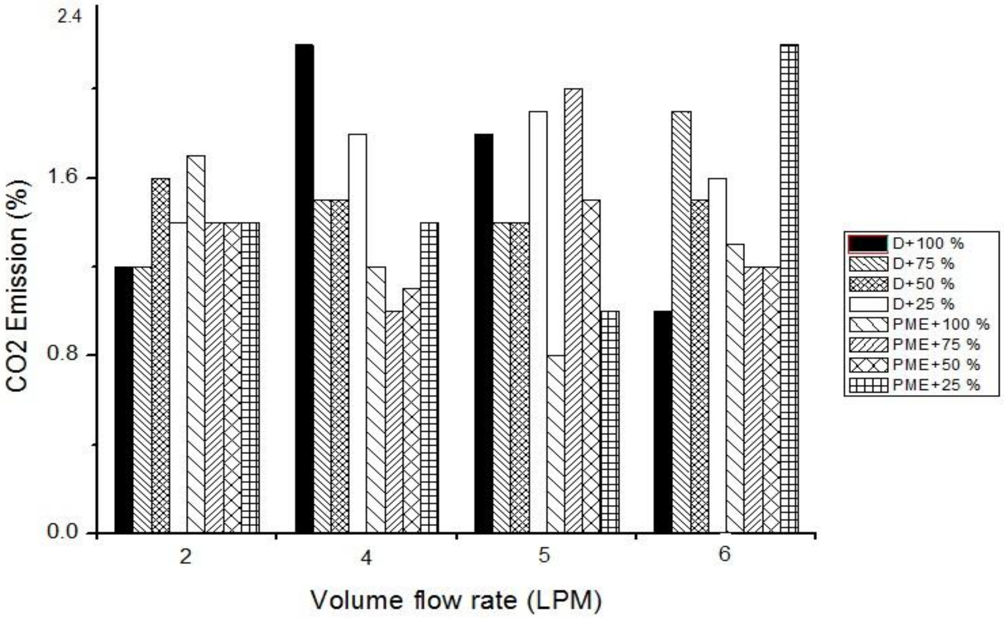


Figure 4.7(c): Carbon dioxide emission with volume flow rate at the different EGR percentage using diesel and PME100

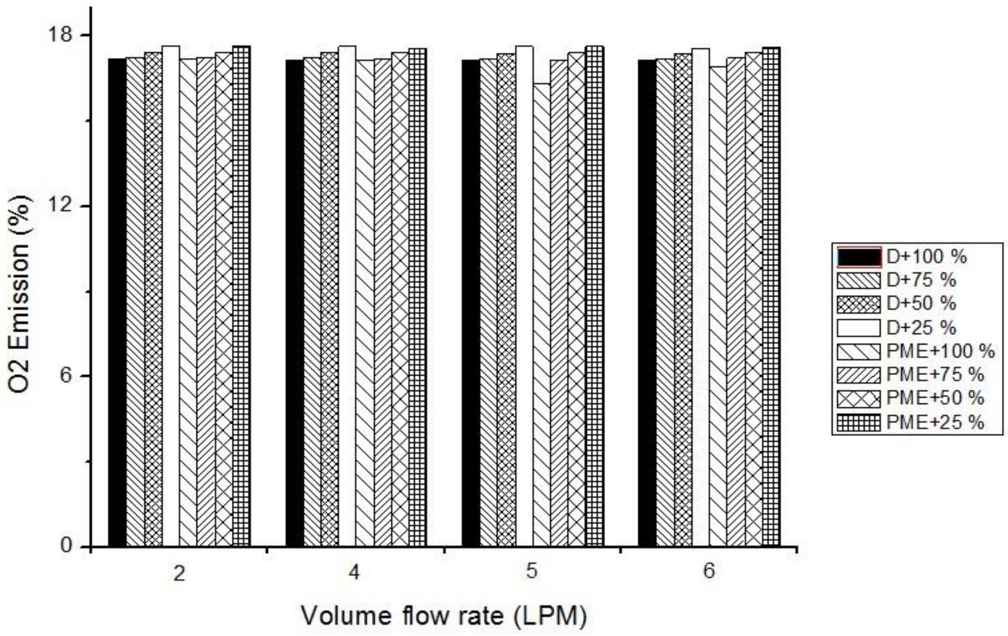


Figure 4.7(d): Oxygen concentration with volume flow rate at the different EGR percentage using diesel and PME100

CHAPTER 5

CONCLUSION

5.1 Introduction

This chapter presents the conclusion based on the result from research and result with analysis.

5.2 Conclusion

EGR is a very useful technique to reduce NOx. An experimental study was conducted on the design a novel system of exhaust gas recirculation for diesel engine. The experiment measured the exhaust emissions in a DI diesel engine operating with diesel and PME 100. Engines operated with original EGR, new EGR and without EGR were used. Lastly the experiment was done to measure the heat transfer rate of water to heat exchanger, also to measure the temperature and the emission level in that condition.

The new design of exhaust gas recirculation system consist of the heat exchanger in order to reduce the exhaust temperature and the flow rate meter to measure the exhaust flow rate in the EGR system. Heat exchanger was made of steel and using shell and tube heat exchanger. The overall heat transfer coefficient for this heat exchanger is 25.85 $W/m^2.^oC$. From the result and analysis comparing the temperature differences between the new EGR with the original EGR is the original EGR, it is better to reduce the temperature. It shows that the heat exchanger at the new EGR was not enough to reduce the temperature. It can be seen that the emission level using the new EGR is between the emission without EGR and the original

EGR. When the NOx is reduced, particulate matter (PM) will increase resulting from the lower oxygen concentration.

5.3 Recommendation

In the future, the design for heat exchanger can be done bu changing the type of heat exchanger. The Shell and tube heat exchanger are not suitable for use in automotive and aircraft applications because of their relatively large size and weight (Yunus 2006). This study also can be conduct with variable parameter such as variable fuel, variable engine speed with the different load.

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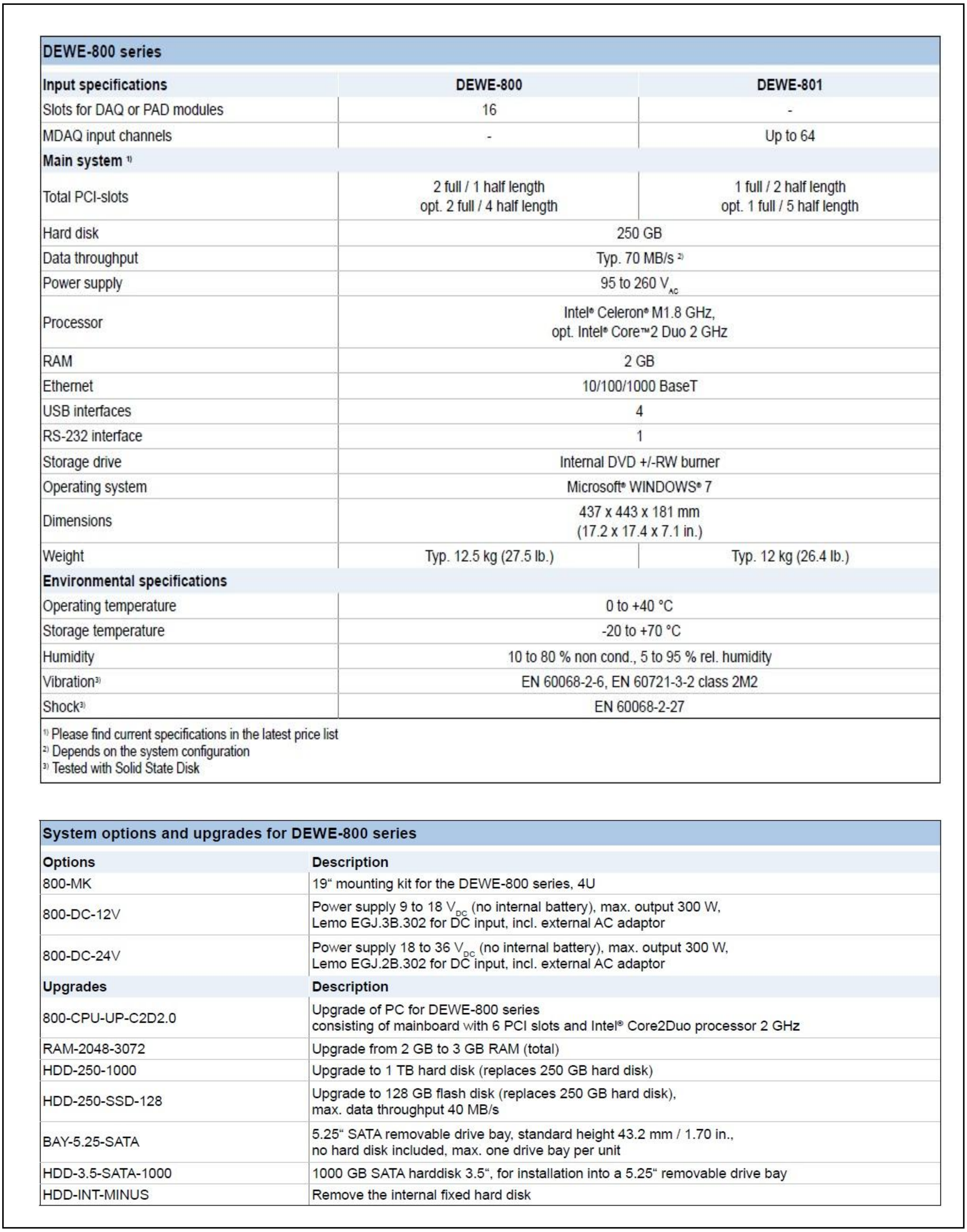
APPENDIX A

Specification of Exhaust Analyzer

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

APPENDIX B

Data Acquisition System specification



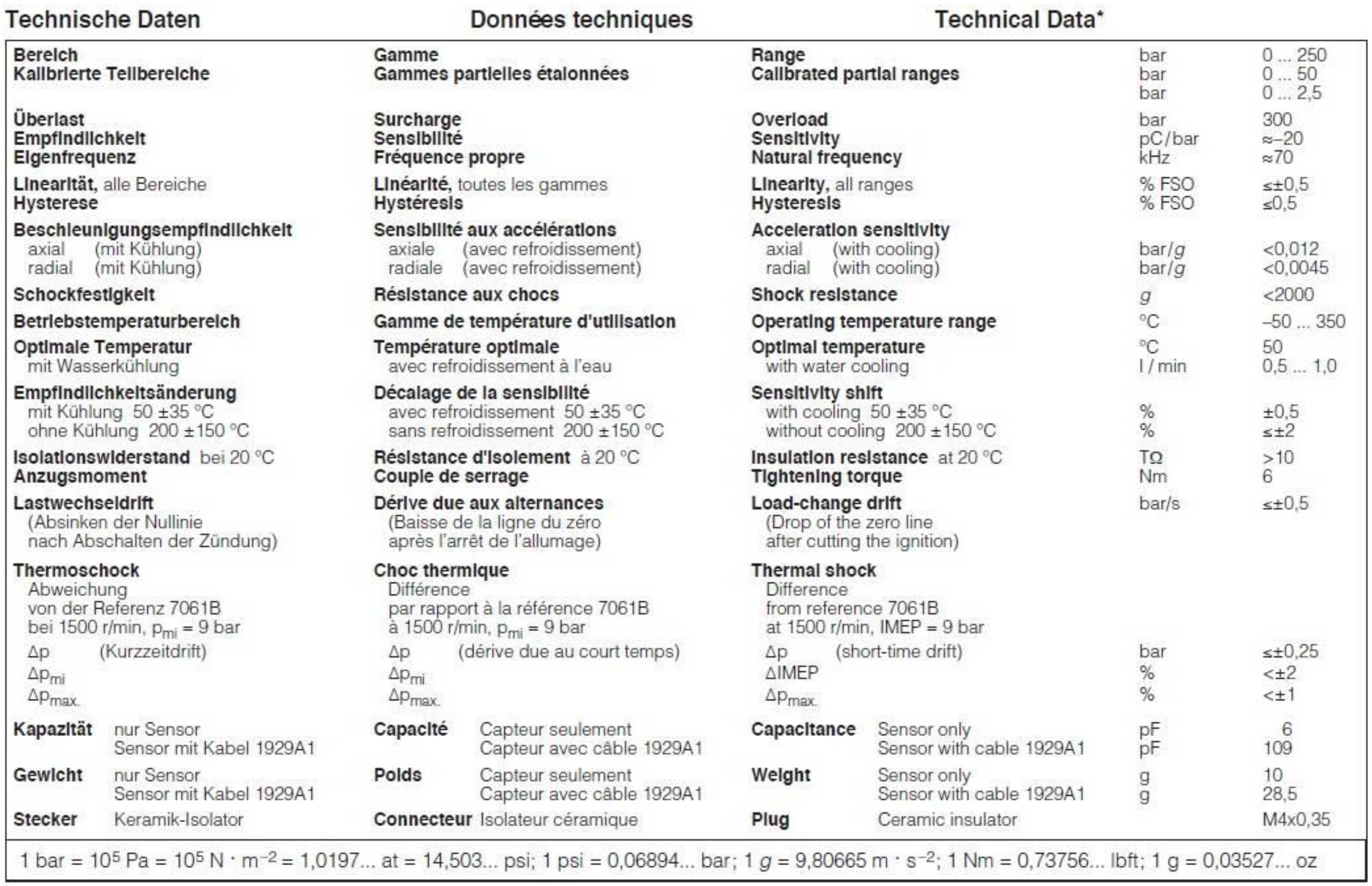
DEWE-800 series		
Input specifications	**DEWE-800**	**DEWE-801**
Slots for DAQ or PAD modules	16	-
MDAQ input channels	-	Up to 64
Main system [1]		
Total PCI-slots	2 full / 1 half length opt. 2 full / 4 half length	1 full / 2 half length opt. 1 full / 5 half length
Hard disk	250 GB	
Data throughput	Typ. 70 MB/s [2]	
Power supply	95 to 260 V_{AC}	
Processor	Intel® Celeron® M1.8 GHz, opt. Intel® Core™2 Duo 2 GHz	
RAM	2 GB	
Ethernet	10/100/1000 BaseT	
USB interfaces	4	
RS-232 interface	1	
Storage drive	Internal DVD +/-RW burner	
Operating system	Microsoft® WINDOWS® 7	
Dimensions	437 x 443 x 181 mm (17.2 x 17.4 x 7.1 in.)	
Weight	Typ. 12.5 kg (27.5 lb.)	Typ. 12 kg (26.4 lb.)
Environmental specifications		
Operating temperature	0 to +40 °C	
Storage temperature	-20 to +70 °C	
Humidity	10 to 80 % non cond., 5 to 95 % rel. humidity	
Vibration[3]	EN 60068-2-6, EN 60721-3-2 class 2M2	
Shock[3]	EN 60068-2-27	

[1] Please find current specifications in the latest price list
[2] Depends on the system configuration
[3] Tested with Solid State Disk

System options and upgrades for DEWE-800 series	
Options	**Description**
800-MK	19“ mounting kit for the DEWE-800 series, 4U
800-DC-12V	Power supply 9 to 18 V_{DC} (no internal battery), max. output 300 W, Lemo EGJ.3B.302 for DC input, incl. external AC adaptor
800-DC-24V	Power supply 18 to 36 V_{DC} (no internal battery), max. output 300 W, Lemo EGJ.2B.302 for DC input, incl. external AC adaptor
Upgrades	**Description**
800-CPU-UP-C2D2.0	Upgrade of PC for DEWE-800 series consisting of mainboard with 6 PCI slots and Intel® Core2Duo processor 2 GHz
RAM-2048-3072	Upgrade from 2 GB to 3 GB RAM (total)
HDD-250-1000	Upgrade to 1 TB hard disk (replaces 250 GB hard disk)
HDD-250-SSD-128	Upgrade to 128 GB flash disk (replaces 250 GB hard disk), max. data throughput 40 MB/s
BAY-5.25-SATA	5.25“ SATA removable drive bay, standard height 43.2 mm / 1.70 in., no hard disk included, max. one drive bay per unit
HDD-3.5-SATA-1000	1000 GB SATA harddisk 3.5“, for installation into a 5.25“ removable drive bay
HDD-INT-MINUS	Remove the internal fixed hard disk

APPENDIX C

Pressure Sensor Kistler Model 6041A technical data

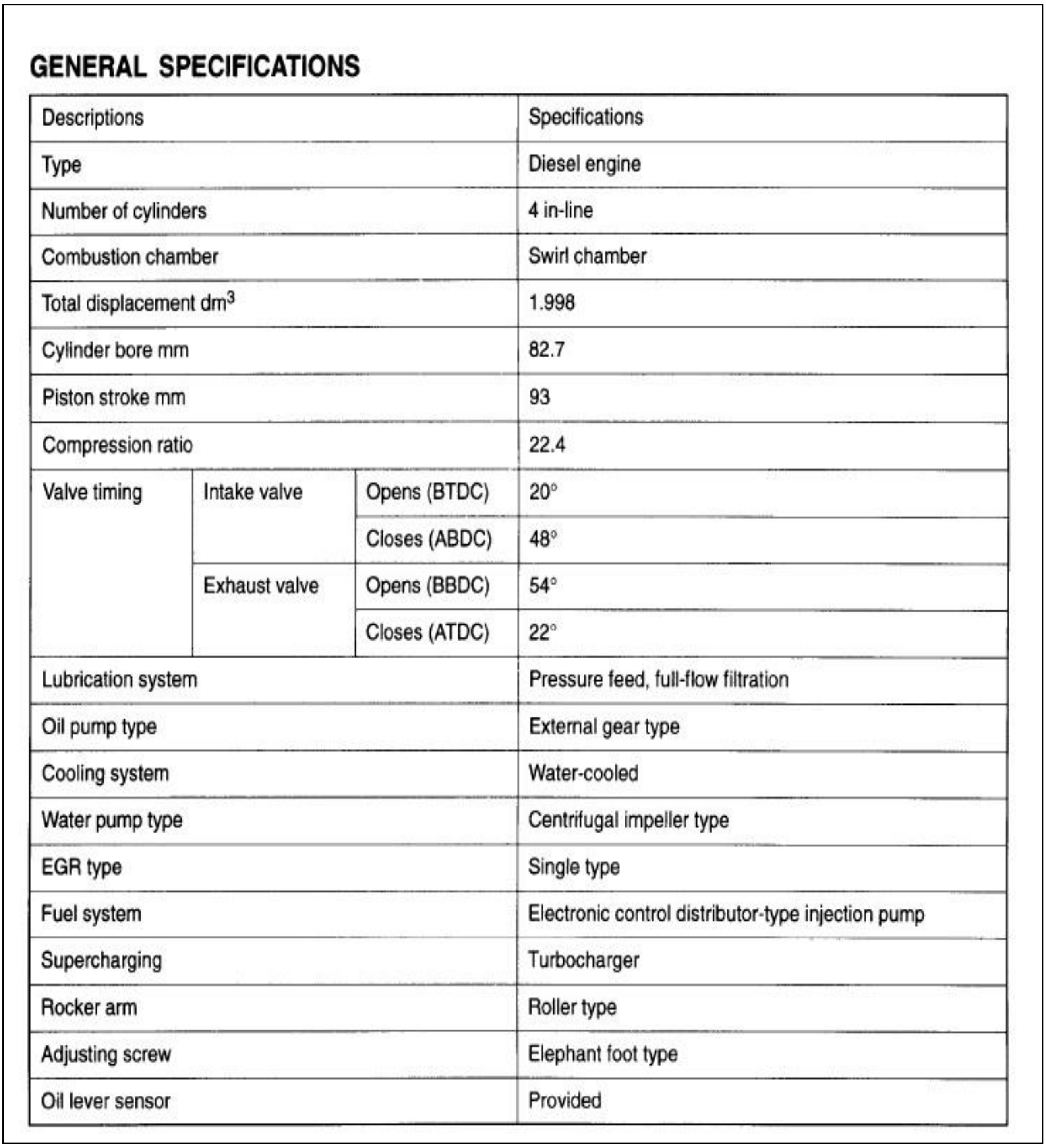


Technische Daten	Données techniques	Technical Data*		
Bereich	**Gamme**	**Range**	bar	0 ... 250
Kalibrierte Teilbereiche	**Gammes partielles étalonnées**	**Calibrated partial ranges**	bar bar	0 ... 50 0 ... 2,5
Überlast	**Surcharge**	**Overload**	bar	300
Empfindlichkeit	**Sensibilité**	**Sensitivity**	pC/bar	≈–20
Eigenfrequenz	**Fréquence propre**	**Natural frequency**	kHz	≈70
Linearität, alle Bereiche	**Linéarité,** toutes les gammes	**Linearity,** all ranges	% FSO	≤±0,5
Hysterese	**Hystéresis**	**Hysteresis**	% FSO	≤0,5
Beschleunigungsempfindlichkeit	**Sensibilité aux accélérations**	**Acceleration sensitivity**		
axial (mit Kühlung)	axiale (avec refroidissement)	axial (with cooling)	bar/g	<0,012
radial (mit Kühlung)	radiale (avec refroidissement)	radial (with cooling)	bar/g	<0,0045
Schockfestigkeit	**Résistance aux chocs**	**Shock resistance**	g	<2000
Betriebstemperaturbereich	**Gamme de température d'utilisation**	**Operating temperature range**	°C	–50 ... 350
Optimale Temperatur	**Température optimale**	**Optimal temperature**	°C	50
mit Wasserkühlung	avec refroidissement à l'eau	with water cooling	l / min	0,5 ... 1,0
Empfindlichkeitsänderung	**Décalage de la sensibilité**	**Sensitivity shift**		
mit Kühlung 50 ±35 °C	avec refroidissement 50 ±35 °C	with cooling 50 ±35 °C	%	±0,5
ohne Kühlung 200 ±150 °C	sans refroidissement 200 ±150 °C	without cooling 200 ±150 °C	%	≤±2
Isolationswiderstand bei 20 °C	**Résistance d'isolement** à 20 °C	**Insulation resistance** at 20 °C	TΩ	>10
Anzugsmoment	**Couple de serrage**	**Tightening torque**	Nm	6
Lastwechseldrift (Absinken der Nullinie nach Abschalten der Zündung)	**Dérive due aux alternances** (Baisse de la ligne du zéro après l'arrêt de l'allumage)	**Load-change drift** (Drop of the zero line after cutting the ignition)	bar/s	≤±0,5
Thermoschock Abweichung von der Referenz 7061B bei 1500 r/min, p_{mi} = 9 bar	**Choc thermique** Différence par rapport à la référence 7061B à 1500 r/min, p_{mi} = 9 bar	**Thermal shock** Difference from reference 7061B at 1500 r/min, IMEP = 9 bar		
Δp (Kurzzeitdrift)	Δp (dérive due au court temps)	Δp (short-time drift)	bar	≤±0,25
Δp_{mi}	Δp_{mi}	ΔIMEP	%	<±2
$\Delta p_{max.}$	$\Delta p_{max.}$	$\Delta p_{max.}$	%	<±1
Kapazität nur Sensor	**Capacité** Capteur seulement	**Capacitance** Sensor only	pF	6
Sensor mit Kabel 1929A1	Capteur avec câble 1929A1	Sensor with cable 1929A1	pF	109
Gewicht nur Sensor	**Poids** Capteur seulement	**Weight** Sensor only	g	10
Sensor mit Kabel 1929A1	Capteur avec câble 1929A1	Sensor with cable 1929A1	g	28,5
Stecker Keramik-Isolator	**Connecteur** Isolateur céramique	**Plug** Ceramic insulator		M4x0,35

1 bar = 10^5 Pa = 10^5 N · m^{-2} = 1,0197... at = 14,503... psi; 1 psi = 0,06894... bar; 1 g = 9,80665 m · s^{-2}; 1 Nm = 0,73756... lbft; 1 g = 0,03527... oz

APPENDIX D

Specification of Diesel Mitsubishi Engine 4D68

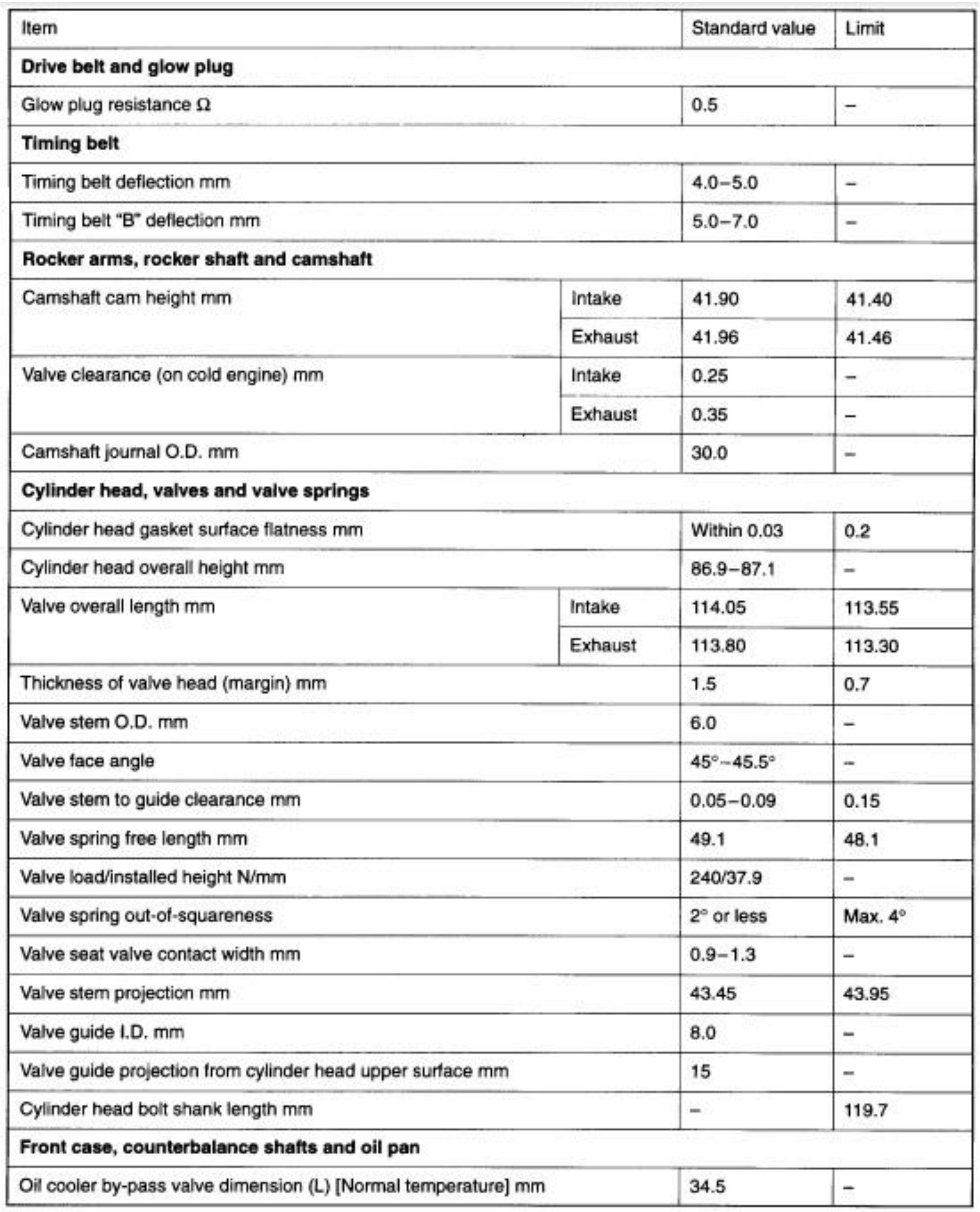


GENERAL SPECIFICATIONS

Descriptions			Specifications
Type			Diesel engine
Number of cylinders			4 in-line
Combustion chamber			Swirl chamber
Total displacement dm^3			1.998
Cylinder bore mm			82.7
Piston stroke mm			93
Compression ratio			22.4
Valve timing	Intake valve	Opens (BTDC)	20°
		Closes (ABDC)	48°
	Exhaust valve	Opens (BBDC)	54°
		Closes (ATDC)	22°
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
Supercharging			Turbocharger
Rocker arm			Roller type
Adjusting screw			Elephant foot type
Oil lever sensor			Provided

APPENDIX E

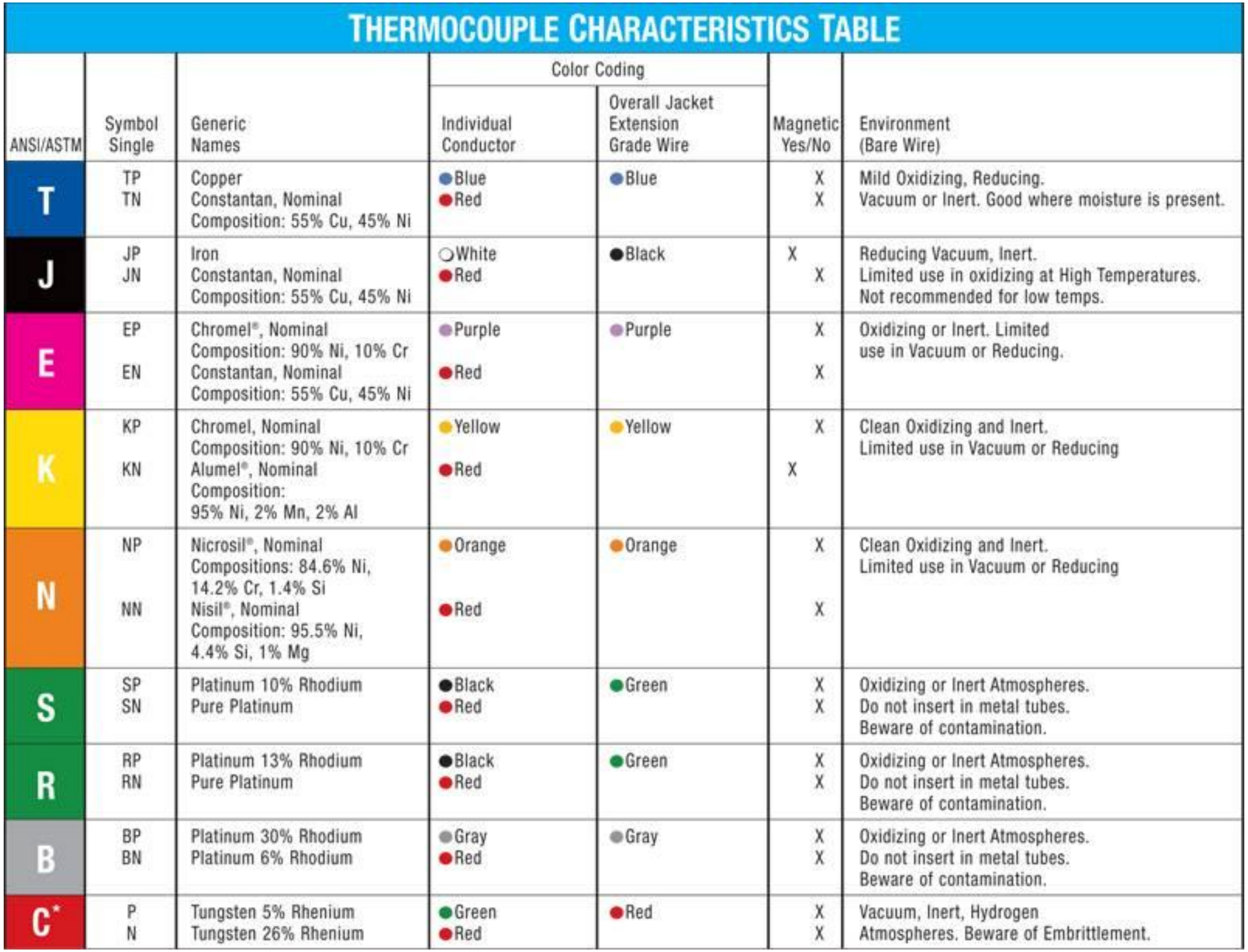
Service Specifications



Item		Standard value	Limit
Drive belt and glow plug			
Glow plug resistance Ω		0.5	–
Timing belt			
Timing belt deflection mm		4.0–5.0	–
Timing belt "B" deflection mm		5.0–7.0	–
Rocker arms, rocker shaft and camshaft			
Camshaft cam height mm	Intake	41.90	41.40
	Exhaust	41.96	41.46
Valve clearance (on cold engine) mm	Intake	0.25	–
	Exhaust	0.35	–
Camshaft journal O.D. mm		30.0	–
Cylinder head, valves and valve springs			
Cylinder head gasket surface flatness mm		Within 0.03	0.2
Cylinder head overall height mm		86.9–87.1	–
Valve overall length mm	Intake	114.05	113.55
	Exhaust	113.80	113.30
Thickness of valve head (margin) mm		1.5	0.7
Valve stem O.D. mm		6.0	–
Valve face angle		45°–45.5°	–
Valve stem to guide clearance mm		0.05–0.09	0.15
Valve spring free length mm		49.1	48.1
Valve load/installed height N/mm		240/37.9	–
Valve spring out-of-squareness		2° or less	Max. 4°
Valve seat valve contact width mm		0.9–1.3	–
Valve stem projection mm		43.45	43.95
Valve guide I.D. mm		8.0	–
Valve guide projection from cylinder head upper surface mm		15	–
Cylinder head bolt shank length mm		–	119.7
Front case, counterbalance shafts and oil pan			
Oil cooler by-pass valve dimension (L) [Normal temperature] mm		34.5	–

APPENDIX F

Thermocouple Characteristics Table

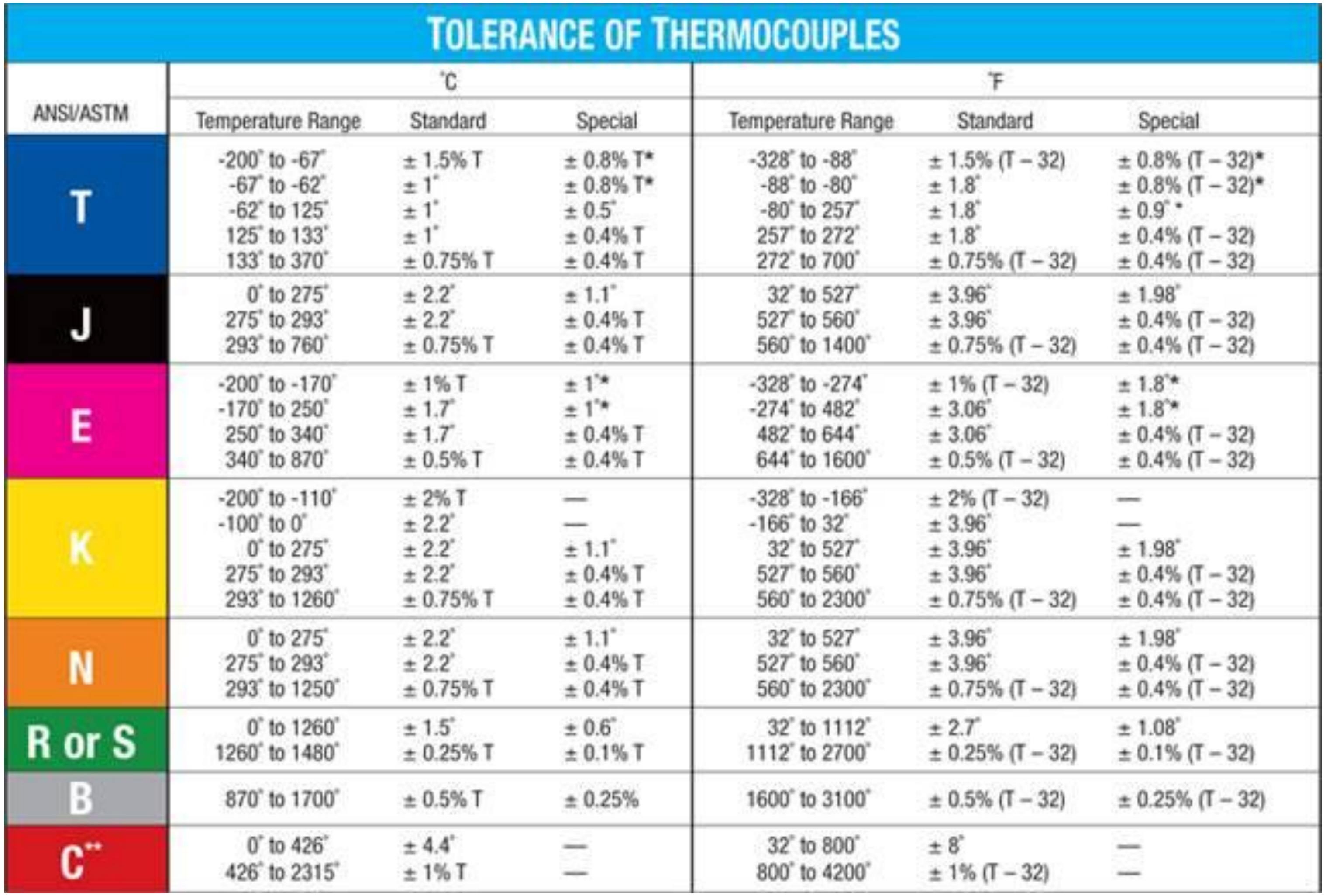


THERMOCOUPLE CHARACTERISTICS TABLE

ANSI/ASTM	Symbol Single	Generic Names	Color Coding: Individual Conductor	Color Coding: Overall Jacket Extension Grade Wire	Magnetic Yes/No	Environment (Bare Wire)
T	TP TN	Copper Constantan, Nominal Composition: 55% Cu, 45% Ni	● Blue ● Red	● Blue	X X	Mild Oxidizing, Reducing. Vacuum or Inert. Good where moisture is present.
J	JP JN	Iron Constantan, Nominal Composition: 55% Cu, 45% Ni	○ White ● Red	● Black	X X	Reducing Vacuum, Inert. Limited use in oxidizing at High Temperatures. Not recommended for low temps.
E	EP EN	Chromel®, Nominal Composition: 90% Ni, 10% Cr Constantan, Nominal Composition: 55% Cu, 45% Ni	● Purple ● Red	● Purple	X X	Oxidizing or Inert. Limited use in Vacuum or Reducing.
K	KP KN	Chromel, Nominal Composition: 90% Ni, 10% Cr Alumel®, Nominal Composition: 95% Ni, 2% Mn, 2% Al	● Yellow ● Red	● Yellow	X X	Clean Oxidizing and Inert. Limited use in Vacuum or Reducing
N	NP NN	Nicrosil®, Nominal Compositions: 84.6% Ni, 14.2% Cr, 1.4% Si Nisil®, Nominal Composition: 95.5% Ni, 4.4% Si, 1% Mg	● Orange ● Red	● Orange	X X	Clean Oxidizing and Inert. Limited use in Vacuum or Reducing
S	SP SN	Platinum 10% Rhodium Pure Platinum	● Black ● Red	● Green	X X	Oxidizing or Inert Atmospheres. Do not insert in metal tubes. Beware of contamination.
R	RP RN	Platinum 13% Rhodium Pure Platinum	● Black ● Red	● Green	X X	Oxidizing or Inert Atmospheres. Do not insert in metal tubes. Beware of contamination.
B	BP BN	Platinum 30% Rhodium Platinum 6% Rhodium	● Gray ● Red	● Gray	X X	Oxidizing or Inert Atmospheres. Do not insert in metal tubes. Beware of contamination.
C*	P N	Tungsten 5% Rhenium Tungsten 26% Rhenium	● Green ● Red	● Red	X X	Vacuum, Inert, Hydrogen Atmospheres. Beware of Embrittlement.

APPENDIX G

Tolerances of Thermocouple



TOLERANCE OF THERMOCOUPLES

ANSI/ASTM	°C			°F		
	Temperature Range	Standard	Special	Temperature Range	Standard	Special
T	-200° to -67°	± 1.5% T	± 0.8% T*	-328° to -88°	± 1.5% (T – 32)	± 0.8% (T – 32)*
	-67° to -62°	± 1°	± 0.8% T*	-88° to -80°	± 1.8°	± 0.8% (T – 32)*
	-62° to 125°	± 1°	± 0.5°	-80° to 257°	± 1.8°	± 0.9° *
	125° to 133°	± 1°	± 0.4% T	257° to 272°	± 1.8°	± 0.4% (T – 32)
	133° to 370°	± 0.75% T	± 0.4% T	272° to 700°	± 0.75% (T – 32)	± 0.4% (T – 32)
J	0° to 275°	± 2.2°	± 1.1°	32° to 527°	± 3.96°	± 1.98°
	275° to 293°	± 2.2°	± 0.4% T	527° to 560°	± 3.96°	± 0.4% (T – 32)
	293° to 760°	± 0.75% T	± 0.4% T	560° to 1400°	± 0.75% (T – 32)	± 0.4% (T – 32)
E	-200° to -170°	± 1% T	± 1°*	-328° to -274°	± 1% (T – 32)	± 1.8°*
	-170° to 250°	± 1.7°	± 1°*	-274° to 482°	± 3.06°	± 1.8°*
	250° to 340°	± 1.7°	± 0.4% T	482° to 644°	± 3.06°	± 0.4% (T – 32)
	340° to 870°	± 0.5% T	± 0.4% T	644° to 1600°	± 0.5% (T – 32)	± 0.4% (T – 32)
K	-200° to -110°	± 2% T	—	-328° to -166°	± 2% (T – 32)	—
	-100° to 0°	± 2.2°	—	-166° to 32°	± 3.96°	—
	0° to 275°	± 2.2°	± 1.1°	32° to 527°	± 3.96°	± 1.98°
	275° to 293°	± 2.2°	± 0.4% T	527° to 560°	± 3.96°	± 0.4% (T – 32)
	293° to 1260°	± 0.75% T	± 0.4% T	560° to 2300°	± 0.75% (T – 32)	± 0.4% (T – 32)
N	0° to 275°	± 2.2°	± 1.1°	32° to 527°	± 3.96°	± 1.98°
	275° to 293°	± 2.2°	± 0.4% T	527° to 560°	± 3.96°	± 0.4% (T – 32)
	293° to 1250°	± 0.75% T	± 0.4% T	560° to 2300°	± 0.75% (T – 32)	± 0.4% (T – 32)
R or S	0° to 1260°	± 1.5°	± 0.6°	32° to 1112°	± 2.7°	± 1.08°
	1260° to 1480°	± 0.25% T	± 0.1% T	1112° to 2700°	± 0.25% (T – 32)	± 0.1% (T – 32)
B	870° to 1700°	± 0.5% T	± 0.25%	1600° to 3100°	± 0.5% (T – 32)	± 0.25% (T – 32)
C**	0° to 426°	± 4.4°	—	32° to 800°	± 8°	—
	426° to 2315°	± 1% T	—	800° to 4200°	± 1% (T – 32)	—

APPENDIX H

Gantt chart for PSM 1

PROGRESS \ WEEK	1	2	3	4	5	6	7	8	9	10	11	12	13
Meeting with supervisor	■	■											
Find journal and book		■	■	■									
Read and understand the title		■	■	■									
Propose the design of EGR system			■	■	■								
Find the design for heat exchanger and flow meter				■	■	■							
Find the material for HE and flow meter							■	■	■				
Cut the material							■	■	■				
Fabrication work									■	■	■	■	
Fabrication work finish													■

APPENDIX I

Gantt chart for PSM 2

Project activities	W1	W2	W3	W4	W5	W6	W7	W8	W9	W10	W11	W12	W13	W14
Experiment testing	■	■	■	■	■									
Report writing			■	■	■	■	■							
Report repair and checking								■	■	■	■			
Prepare for present											■	■	■	
presentation														■

APPENDIX J

Progress Picture 1

APPENDIX K

Progress Picture 2

APPENDIX L

Progress Picture 3