MODELLING AND ANALYSIS OF PERFORMANCE FOR BIOGAS FUELLED GAS ENGINE USING MATLAB

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MODELLING AND ANALYSIS OF PERFORMANCE FOR BIOGAS FUELLED GAS ENGINE USING MATLAB

ABSTRACT

A wide-ranging performance evaluation of a multi-cylinder gas engine operating wit biogas as a fuel were studied based on its performance parameters such as effective power, torque, SFC, MEP and Thermal Efficiency and its HRR analysis with the aid of modeling software, MATLAB. The analysis and performance study were used based on the thermodynamic model of an Otto cycle, which is based on the gas engines used. Parameters such as cylinder pressure with domain of crank angle and heat addition model (using Wiebe function) were studied to model the and analyze the HRR .Besides that , the contributing factors in the efficiency and performance of the engine were deduced such as fluctuating HRR and increased thermal efficiency determining the performance optimization of the engine . The results obtain from the study were compared using the trend demonstrated with similar gas engine models. This research would set a milestone to extended research on the gas engines used in the industrial scale for performance enhancement and optimization.

Keywords: gas engine, performance study, biogas, thermal efficiency, heat release rate

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LIST OF SYMBOLS AND ABBREVIATIONS

ASTM	American Society for Testing and Materials
B.M.E.P	Brake Mean Effective Pressure
BOD	Biological Oxygen Demand
CH_4	Methane
CI	Compression Ignition
CPO	Crude Palm Oil
CO_2	Carbon Dioxide
COD	Chemical Oxygen Demand
EFB	Empty Fruit Brunch
H_2S	Hydrogen Sulphide
HRR	Heat Release Rate
I.M.E.P	Indicated Mean Effective Pressure
ISO	International Organization for Standardization
kW	kilo Watt
kJ	kilo Joule
l	liter
LPG	Liquid Petroleum Gas
m	meters
m^3	meter cube
MEP	Mean Effective Pressure
MPOC	Malaysian Palm Oil Council
NG	Natural Gas
P	Power
P_{atm}	Atmospheric Pressure
P_{max}	Maximum Pressure

POME	Palm Oil Mill Effluent
RPM	Rotation per meter
SAE	Society of Automobile Engineers
SFC	Specific Fuel Consumption
SI	Spark Ignition
ηv	Volumetric efficiency
η_{TE}	Thermal Efficiency
θ	Crank Angle

CHAPTER 1

INTRODUCTION

1.1 Background

With the spike in global energy demand, the fossil fuel energy sources would result in skydiving trend as the reserves of fossil fuel supply are limited and catering the demand increase would only result in rapid diminish of fossil fuel supply. However, there the emergence of alternative energy (which is also known as Renewable Energy) such as Solar , Hydroelectricity , Biomass and Biogas contributes to the feat to support the energy demand spike .

Malaysia been one of the largest key player of Palm Oil global import and export , there is no wonder that it has abundance of palm waste such as Palm Oil Mill Effluent (POME) and Empty Fruit Brunches (EFB) which is enriched with fuel potentials which contributes to the Biogas and Biomass generation . According to Malaysian Palm Oil Council (MPOC) , it has been reported that Malaysian's Palm Oil exports contributes approximately 39% of the global palm oil production and 44% of world export (MPOC , 2011) which translates to the estimation that the crude palm oil (CPO) production of 19,510,000 metric tonne to have the potential to produce 58.53×10^6 m^3 of POME annually and with the estimation of 35 to 40 % gas engine efficiency, which an approximate estimation of POME has the capacity to generate up to 1,044,760,500 m^3 of

biogas through anaerobic digestion which is equivalent to 4.38 TWh/y of electricity (Kumaran , 2016) . POME is generally treated in open pond and tanks anaerobically which is due to the high Biochemical Oxygen demand (BOD) and Chemical Oxygen Demand) ratio whereby the ideal BOD to COD ratio is approximately 0.5 (Hosseini and Wahid, 2013). Thus, POME becomes an ideal fuel option for biogas.

The biogas composed to about 50 to 70% of Methane (CH_4) and 28 to 40% of Carbon dioxide followed by 0.08 to 1 % of Hydrogen Sulphide (H_2S) and less than 0.01 of Water Vapor traces (Bu et.,2017).

Table 1.1: Biogas Components Breakdown (Archinas et.al 2017)

Components	% of Breakdown by Volume
CH_4	50 to 70%
CO_2	28 to 40 %
H_2S	0.05 to 1.0 %
Water Vapour	≥0.01 %

In line with harnessing the best of biogas potential, a suitable gas engine would be required with optimized efficiency and performance. A typical biogas engine performs with a nominal efficiency ranging from 30 to 50% and close to 85% in cumulative efficiency (cumulative; consists of Thermal efficiency and Electrical Efficiency) depending on the fuel quality.

The use of gas engines dates back as early as 1860, which was the Belgian engineer Étienne Lenoir who built the engine and later was developed by the German engineer Nicolaus Otto circa 1864 with the association with Eugen Laugen. From the extended development of the gas engine, developed the 4 stroke engine which operates under the

Otto cycle principle (Blanning,2003) However, the first historical record of gas engines used in biogas generation facility dates back to during World War 2, where most of the German military vehicles and farm manures were operating under biogas driven engines. In addition to that, the agricultural Biogas plant in Birmingham in the United Kingdom which was the first to conventionally use the gas engine to generate electricity for auxiliary utilization (Chasnyk et al., 2016) .

The present trend of energy utilization, which is heavily dominant on fossil fuels, refrained to be sustained, limiting to two major hurdle : the environmental impact due to the use of fossil fuel and the its depleted reserves . In recent times , the environmental issues such as   pollution and climatic changes   stirred   by   the   consumption   of conventional fossil fuels is becoming a significant attention which concerned the sustainability of the environment which developed global collaboration treaties such as the Paris Agreement and Kyoto Protocol which heavily emphasized on the significance of reductions Greenhouse Gas (GHG) emissions and its impact with respect to the global energy sustainability (Emission Gap Report 2019 , *UN Environment Program* , 2019) .

1.2 Problem Statement

In the scientific researches, only few works study the performance of biogas engines that too using test engines and not machines installed on real biogas plants which operate in the electricity market. Due to the limitation of study of biogas engine performance installed at actual site , we couldn't identify the parameters contributing to the performance of the engine. However there are several questions to be subjected to identify the key problem which would also objectify the research scope and those questions are:

a. How does the parameters such as Specific Fuel Consumption (SFC) , Mean Effective Pressure and Thermal Efficiency contribute to the optimized efficiency ?

b. Does the burned or combusted mass fraction of the air-fuel mixtures proportionate with the stroke stages in the 4-stroke gas engine?

c. How does the Heat Release Rate reflect on the overall performance of the engine?

The performance of a biogas engine drops significantly low as the engine load and speed of the gas engine reduces. Thus, making it less economical for intermediate loads in the means when the biogas fuel supply is in scarcity. According to the article review by Bhandari et.al (2005), which discusses on the performance output in comparison with diesel fuels (LPG) and gasoline-based fuels (NG). The article debated on the physics behind the reasoning on why diesel engine is relatively more efficient compared to gasoline engines in higher output torque production within limited operational speed. This is due to the ability of an engine to pump air which is also known as the volumetric efficiency parameter, whereby when it is reduced, the maximum power generated by the engine reduced. The amount of air accessible to engine relies on the resistance to flow through the engine intake system, cylinder and exhaust system. In that case, when liquid fuel such as diesel, its atomization for combustion, consumes a fraction of small space in the intake system which correlates to not much significant affects on the volumetric efficiency . In the other hand , gasoline fuelled engine such as gas engines which is driven by biogas – the fuel might require approximately 4 to 15% intake passage volume whereby the space occupied by the fuel narrows the volume of air channeled into the engine significantly reducing the power output . The power output losses for LPG fuel and NG fuel are 4% and 9.5% respectively.

Besides that, it is hypothesized that the heat release contributes to the increase of performance drop. The higher heat released by the engine will not only deteriorate the engine performance in the longer run but also results in safety hazards of the engine as well .

1.3 Objectives

The key objectives of this research are as below:

- To investigate the performance of gas engine operating on biogas derived from palm oil effluent (POME) as well as modeling the performance of the engine with the use of MATLAB as a simulation tool .

- To model the rate of heat release analysis based on the performance of the engine with the use on MATLAB as a simulation tool.

- To investigate factors contributing to the performance drop during at lower load and engine running speed capacity.

1.4 Scope Research

The scope of research for this project are defined based on the study of the gas engine performance derived from the preliminary data such as used to fuel caloric value , mass and volume flow rate obtained from the biogas feed supply after it has undergone H_2S filtration such as Scrubbing where the harnessed biogas is channeled into a Scrubber to filter out the H_2S concentration with the aid of biological H_2S fixation mechanism by the Thiobacillus bacteria from the scrubber before directing it to the gas engine(s) . As discussed on the biogas composition in Section 1.1. Biogas constituent to about 50 to 70% of CH_4 which was channeled to the gas engine.

The research project is split into 2 phases; whereby the first part of the research project deals with performance study based on the specific fuel consumption (SFC) , mean effective pressure (MEP) and thermal efficiency in respect to the rotational speed , torque and the load utilized . However, the second part consists on the performance study of the engine based on the data set such Cylinder pressure and indicated mean effective pressure

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(IMEP) in respect to the crank angle (CA). Whereby, the data obtained in part 2 is significant in modelling the heat release of the engine in line with the performance of the engine derived from the engine's operation and combustion. Thus, from the, data and study obtained, we could model the performance with the aid of numerical simulation tool called as MATLAB which simulates the performance in terms of plots.

The scope of this project revolves on the following aspects;

a.) The study on the gas engine(s) and its performance data relative to the step operations of the engines.

b.) Analysis on the data, as well as interpreting the physics behind the performance of the engine.

c.) To model the performance using MATLAB.

1.5 Thesis Outline

This thesis consists of 6 chapters and the description of the chapters are as below.

Chapter One defines the introduction of the research and providing the preliminary understanding to the reader. Besides that, this chapter discusses the problem statement, objective and scope of this research.

Chapter Two furnishes with literatures researches which related to it. However this research doesn't limit to biogas engines only but as well other engine configuration such as Otto cycle engines, Diesel cycle engines, 2-stroke engines, 4-stroke engines and the technical science behind it. The aim of this chapter is to acquaint the reader with the fundamental ideas and theories on the research.

Chapter Three describes the methods employed and the equipment used. Since this research has certain experimental element as well the MATLAB programming. Thus this chapter furnishes the reader with the methodological approach as well as the planning

and execution of the MATLAB Programming; from data collection to analysis

Chapter Four analyses the results of the research as of the outputs from the MATLAB programming encompassing both scopes of research; engine performance and HRR analysis.

Chapter Five discusses the technical justification and the defined hypothesis on the scope of research. Also benchmarking literature references to justify the hypothesis.

Chapter Six deduces conclusion and further potential researches. In this chapter, the list some recommendations for further works / potential.

LITERATURE REVIEW

2.1 Introduction

This chapter deals with the literature studies relating to the research. As mentioned in the Problem Statement, there were limited resources of literatures which aid as the reference and also in benchmarking the research in providing the scientific findings and validation to support it. However, there were several experimental based researches which were taken into references, especially on the conceptual idea and the scientific theory behind it. Besides that, this research did not limit to the fuel type (Diesel based engine) and engine configuration (Direct Fuel Injection, Compression ignition engines). Besides that, since this research deals with real life applications which are commonly used in the industry and in order to process with the research, a commendable ideas and theory on the engines and the physics behind it is highly significant.

2.2 Octane Number Critical Study

Theoretically, the performance of engine are derived from the combustion efficiency and the delivery of the power as the aftermath of the combustion. Thus, for an effective combustion to take place, the working fluid which is the gas fed into the engine shall withstand compression. In other words, the higher the octane number, the greater the fuel

compression can be achieved where how well the gas or working fluid resist pre-ignition firing due to the compression (Sugiarto et.al, 2019).There are 2 types of measuring parameters when it comes to octane number which are known as Random Octane Number (RON; achieved through simulating by running it on a test engine) and Motor Random Number (MON; similar to how RON is being done where the only variation would be pre-heating the fuel) .Diesel engine are not evaluated based on octane number which is due to the fact that it does not initiate ignition based on the fuel to generate power but fuel is added as a medium of air compression to mix with the air .That's why diesel are synergized with biogas or other gaseous fuel to optimize the engine's efficiency and combustion performance. Due to high compression ratio, the likelihood of the working fluid and air mixture to pre-ignite is high especially during the engine knocking. Which also explains why, dual fuel engines such as biogas-diesel have higher octane number which subjects to the high compression ratio (12 to 20) compared to gas fueled which operates with SI engines (Bhandari et.al, 2005).

As to our understanding, CH_4 constitutes about 50% of the fraction of biogas as discussed in Chapter 1 (Table 1.1). However, the octane number differs especially in comparison with raw CH_4 contents in Natural Gas and biogas. Based on the works of Ferozkhan and Ismail, 2017 , which states the octane number biogas as 130 with benchmarking 60% CH4 content for the biogas. However , based on the works of Cho and He, 2009) the octane number of CH_4 is 120 which is used in raw natural gas. Thus, from the octane number it could be deduced, that biogas has a higher tendency to withstand compression compared to raw methane gas. Which in this case, explains that why the premature ignition resulting due from engine knocking is most commonly occurs in natural gas fed engines and has a reduced engine lifespan compared to biogas

fuel which has a greater tendency to withstand pre-ignition due to its higher octane number.

2.3 Internal Combustion Engine

There is a common definition of an Internal Combustion Engine (ICE) is which best described by Heywood (1988), which states that an ICE a type of machine which generates power, derived from the reactant's combustion (blend of oxidizer and fuel) whereby the engine obtains its energy from heat released during the combustion from the non reacted working fluids, the blend of the oxidizer-fuel mixture. The work transfers provide desired output power between this working fluid and the mechanical components within the engine.

Figure 2.1 : Example of Internal Combustion Engine (Gas Engine). *Courtesy image of Caterpillar Inc*.

A internal combustion engine consists of several parts with respective functions and operations which are interdependent to one another parts. The internal combustion parts are consists of 2 types ; stationary and moving parts where the stationary parts remain

static and does not moves and provides structural support whereby the moving parts are dynamic and converting heat energy harnessed into mechanical action. Example of stationary parts are the crankcase, cylinder blocks and its attachments (cylinder, its sleeves, liners), water jackets, core hole plug, intake and exhaust manifold, gaskets and oil pan. As of the moving parts are the piston, connecting rod (and its attachment; bearings, oil and compression ring and snap and wrist pin), crankshaft and camshaft.
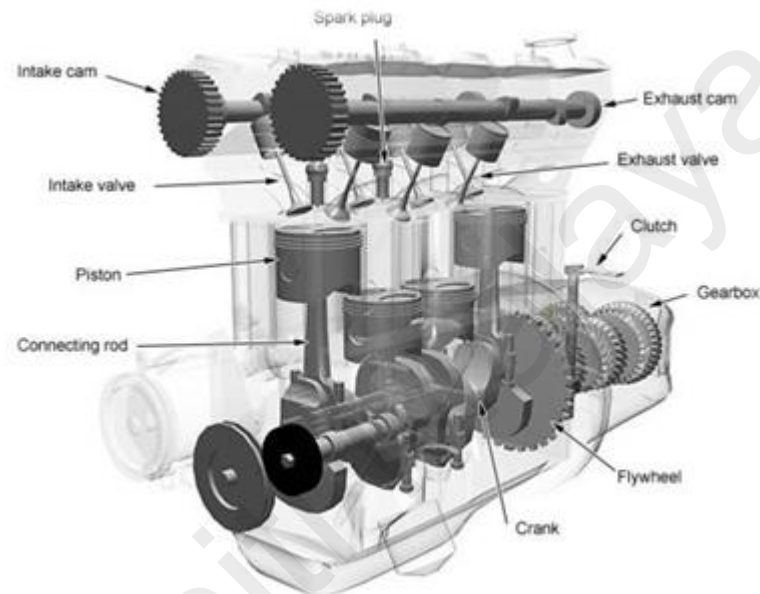
Figure 2.2 Key Components in a Internal Combustion Engine.

At present, the most common types of engines utilized widely would the 2-stroke and 4-strokes engines. The significant difference between a 2-stroke engine and 4-stroke engines would be that in a 2-stroke engine, all five functional operations of the cycle are done within two strokes of the piston (or in other words; it takes one revolution of the crankshaft) whereby in a 4-stroke engine, the five functional operation require four strokes of the piston (or two revolutions of the crankshaft) (Garcia,2018). According to Heywood (1988), to distinguish 2 stroke engines and 4 stroke engines would be the operational sequences. To begin with, a 4-stroke engine (which consists multiple cylinders, ideally 4 cylinders) – each cylinder requires 4 piston strokes correlating to 2 crankshaft revolutions in completing the event sequences generating a power stroke. The

sequence are as below:
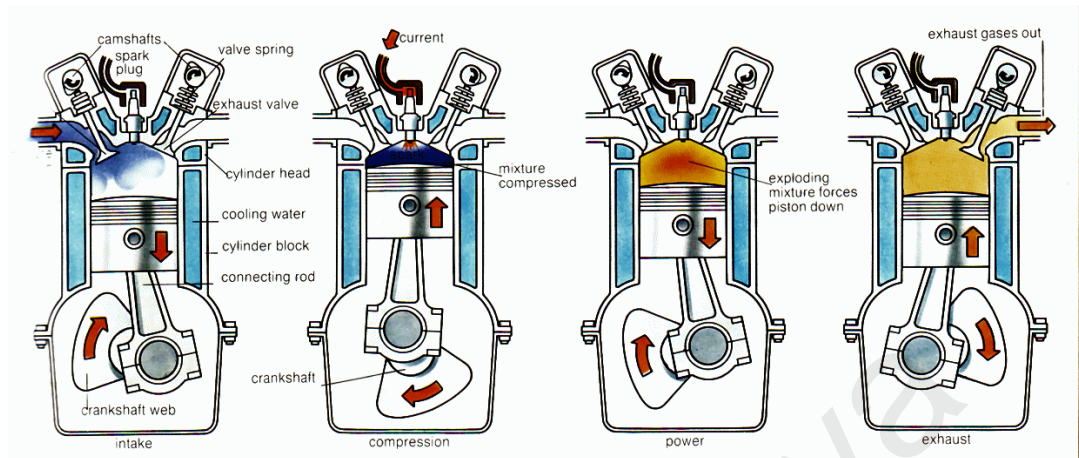
Figure 2.3 : Sequence of 4 stroke engine

1. ***INTAKE STROKE*** – Where the piston from its initial point (at the Top Dead Centre , TDC) moving towards Bottom Dead Centre (BDC) which draws new mixture of air and fuel into the cylinder . The inlet valve opens partially to increase the mass of mixture inducted before initiating the stroke and closes upon completing it .

2. ***COMPRESSION STROKE*** – Whereby the both inlet and exhaust valves were closed and the air and fuel mixture in the cylinder will be compressed to smaller fraction of its initial volume. Combustion initiated as the pressure increases rapidly.

3. ***POWER STROKE*** – Also known as the expansion stroke. Similar to intake stroke, the piston moves from TDC to BDC due to the impulse generated through the high temperature, escalated pressure and gases. This motion , will be achieved with approximately 3 to 5 times as much of work done impacted to the piston during the power stroke in comparison with the compression stroke which also significantly drop the cylinder pressure drop through opening up of the exhaust valve .

4. *EXHAUST STROKE* – burned residual gas mixtures were exhausted from the engine cylinder due to the elevated exhaust pressure sweeping the piston towards TDC, sequentially initiating the inlet valve to open.

In the other hand, 2 stroke engines it takes a revolution of the crankshaft to incorporate 2 or more functions (such as power and expansion strokes) within one piston stroke . Thus , the 2 important sequence in a 2 stroke engines are as below :
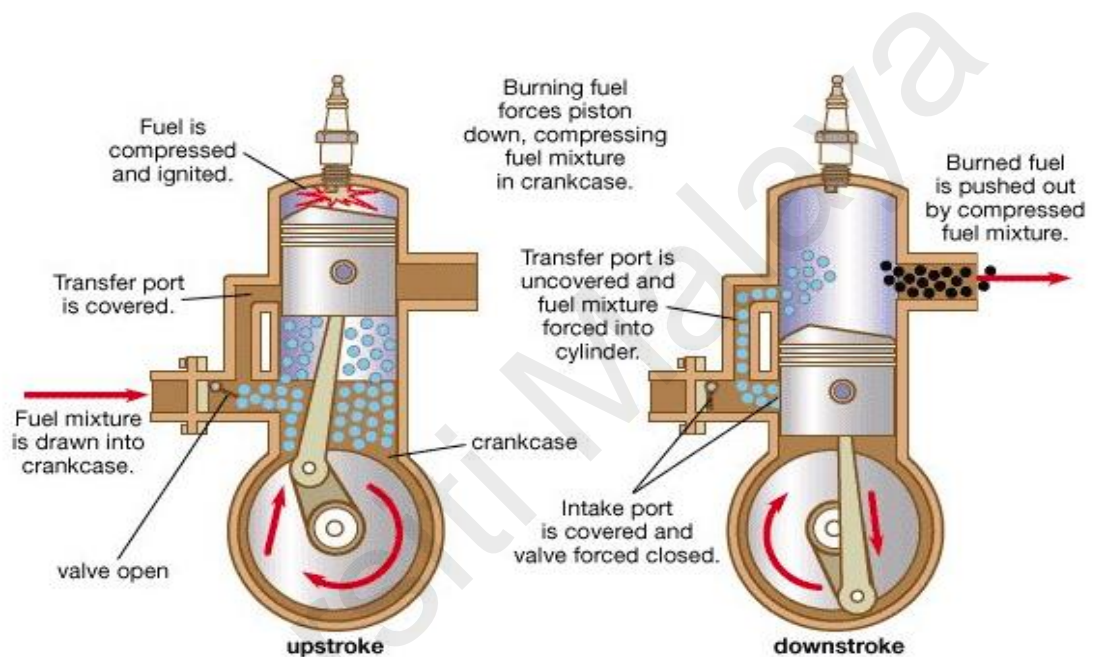
Figure 2.4: Sequence of 2 stroke engine

1. **COMPRESSION STROKE** – initiates via the closing of inlet and exhaust port where the compression in the cylinder would draw new charge into the crank case. Therefore, combustion will be initiated as it approaches the TDC.

2. **POWER / EXPANSION STROKE** – the exhaust port will open up gradually as the piston travels towards BDC and then followed by the inlet port whereby most of the residual burnt gases will be purged from the cylinder due to the exhaust blowdown. Followed by the opening up of the inlet port, fresh charge is drawn into the cylinder via compression due to the crankcase flow.

Each engine cycle with a power stroke is completed with one crankshaft revolution.

However over the recent years , gas engines manufacturers such Catepillar (CAT) are in the frontline of developing 6 stroke engines and several research as well as investigation were conducted on studying the performance and emissions of the engine . According to William.R (et al. 2014) on the research on compression ignition 6 stroke cycle investigation, whereby the findings were based on the comparison between 4 stroke engines and the prototype 6 strokes engines. From the comparative investigation, it was deduced that the 6-stroke cycle demonstrated lesser emissions at stoichiometric operation. Hence, elevated NO_x emissions were observed under some lean conditions. The ratio of fuel burned at the initial combustion sequence to the second exhibited significant impact on engine's performance, heat released and lost with respect to the emission which has the potential to be a milestone breakthrough for the internal combustion engine's development.

2.4 Gas Engine vs Diesel Engine

An internal combustion is normally divided into 2 types; which are the spark ignited (SI) engine and the compression ignition (CI) engine. The significant difference between SI engines and CI engines cycles will be the method of igniting the fuel. Spark ignition engines (operates in function to the Otto cycle) use a spark plug to ignite a pre-mixed air fuel mixture introduced into the cylinder. Compression ignition engines (operates in function to the Diesel-cycle) compress the air introduced into the cylinder to a high pressure, raising its temperature to the auto-ignition temperature of the fuel that is injected at escalated pressure. The fuel is channeled through a fuel injector whereby the air channeled via the inlet port.
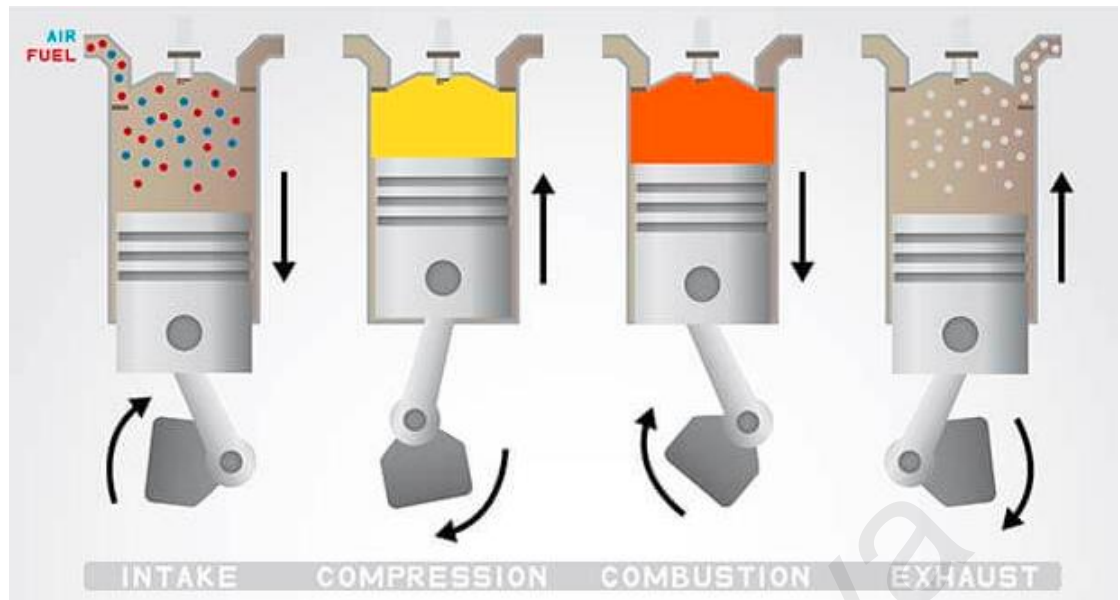
Figure 2.5: Otto cycle sequence (NG Engine).
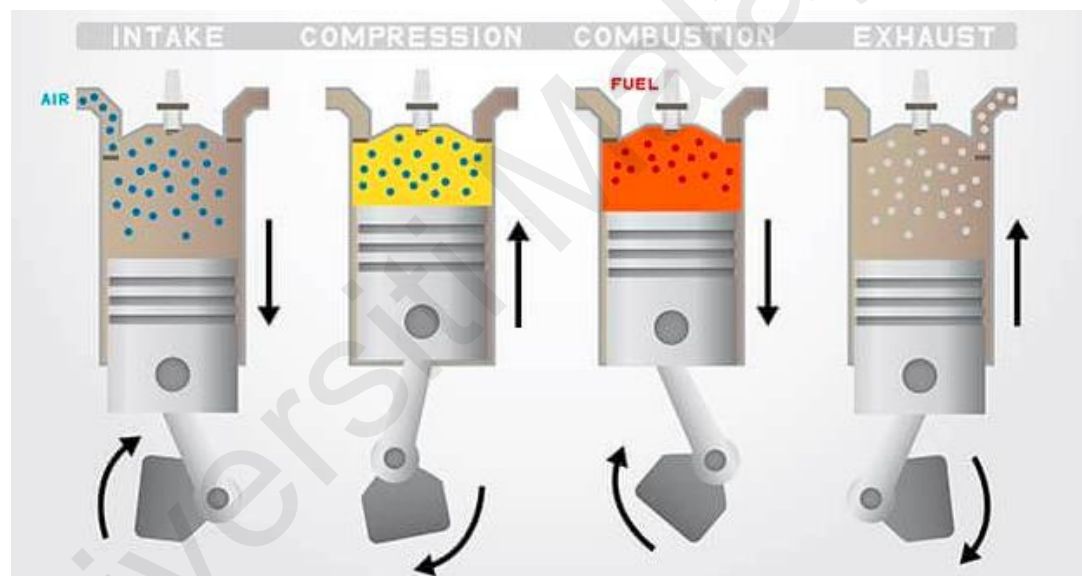
Figure 2.6: Diesel cycle sequence (Diesel Engine)

The Otto cycle defines the occurrence of mass of gas (consists of the fuel and air) as it is affected to pressure change, temperature and displaced volume with respect to addition and losses of heat. Constraining the mass of gas within the cylinder as a system (closed system) which the effect will be to generate sufficient net work from the system . The Otto cycle provides energy for most of the gasoline engine. The Otto cycle can be best described with a Pressure-volume (Pv) diagram to define the overall process relative pressure and volume changes by the fuel and air mixture due to the

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combustion which correlates with the piston movement from TDC to BDC and vice versa , generating heat as a byproduct of the combustion . The expansion (which results in augmented volume displaced in the cylinder) whereby the piston motions were subjected by the thermal energy is released from combustion process which also induce work being done by the gas translated to the piston motion . The piston displacement,x is denoted as ;

$$x = r.cos\theta + \sqrt{l^2 - (r^2.sin\theta)^2} \qquad (2.1)$$

Where the instantaneous crank angle, θ is represented by degrees (o) and crank radius,r which is represented as the following relationship :

$$r = 2s \qquad (2.2)$$

Where s is the stroke length (measured in m) . However, the scenario is inverted during compression whereby the piston does the work   significantly reducing the displaced volume in the cylinder. Figure 2.7 illustrates the Pv relationship of an Ideal Otto Cycle in relative to the cycle processes.
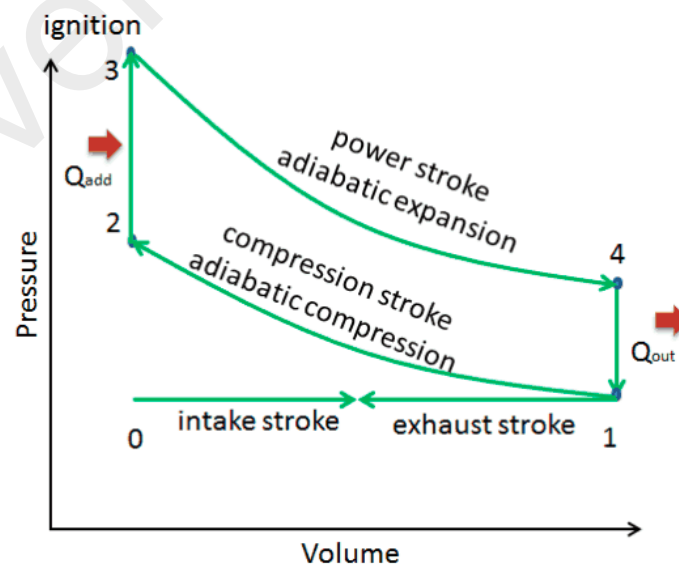
Figure 2.7 : Pv relationship of an Ideal Otto Cycle in relative to the Cycle Processes

Air and gas are drawn into the cylinder via the intake valve at a constant pressure, approximate to the atmospheric pressure, during the intake stroke. Following to that, the air and gas mixture is then compressed at the top of the TDC where the clearance volume is in, as the piston moves from BDC to TDC during the compression stroke. Combustion happens between points 1 and 2. At point 2, the ignition is initiated of the compressed air and gas mixture with the aid of the spark plug at the top of the engine cylinder which instantaneously increases the pressure to its peak at point 3 which also forces the piston to move downwards from TDC to BDC, resulting in rotation of the crankshaft with effect of the power stroke. However, at point 4, the exhaust valve gradually opens. At idyllic conditions, the pressure in the cylinder instantly returns approximately to atmospheric pressure at point 1. Finally, the residual by-products of combustion drawn out from the cylinder via the exhaust valve during the effect of exhaust stroke. According to the works on Saris and Phillips (2005), the fluctuations of pressure with function of volume can be represented by equations below relative to the crank angle, θ .

$$P = 0.9P_{atm} \quad , \quad (\text{ at } \theta<0) \quad\quad (2.3)$$

$$P = \frac{P_{atm}(V_d+V_c)}{V} \quad , \quad (\text{at } \pi \le \theta < 2\,\pi) \quad\quad (2.4)$$

$$P = P_{max}(\frac{V_c}{V})^k \quad , \quad (\text{at } 2\,\pi \le \theta < 3\,\pi) \quad\quad (2.5)$$

$$P = 1.1P_{atm} \quad , \quad (\text{at } 3\pi \le \theta < 4\,\pi) \quad\quad (2.6)$$

Whereby the coefficient of expansion is represented as below ;

$$k = \frac{C_p}{C_v} \quad\quad (2.7)$$

Which is also represented by the ratio of Specific Heat capacity at constant pressure , C_p and Specific Heat Capacity at constant volume , C_v (Ebrahimi,2011). Where in the other hand the cylinder volume varies in domain of the crank angle .

$$\frac{V}{V_d} = \frac{1}{CR-1} + \frac{1}{2}(l + \frac{l}{r} - cos\theta - \sqrt{\left(\frac{l}{r}\right)^2 - (sin\theta)^2} \quad\quad (2.8)$$

$$CR = \frac{V_D + V_C}{V_C} \qquad (2.9)$$

$$\frac{\delta V}{\delta \theta} = \frac{V_d}{2} sin\theta \left(1 + \left(\frac{cos\theta}{\left(\sqrt{\left(\frac{l}{r}\right)^2 - sin^2\theta}\right)}\right)\right) \qquad (2.10)$$

Where CR is the compression ratio , a dimensionless ratio of engine's cylinder volume V_D with respect to combustion chamber volume , V_C . As mention earlier , that the gas engine (8 to 12) has a lower CR compared to diesel engine (CR ranges from 12 to 20) .

In the other hand , the compression ignition engines predominantly applies Diesel cycle . The thermodynamic science behind the compression-ignition engines- when air is being compressed at elevated pressure and temperature beyond the automatic ignition temperature of the fuel with a compression ratio ranging between 12 and 20 .Then the fuel is injected concurrent with the combustion being occurred . The Diesel cycle is best described with the Pv diagram in Figure 2.8 .
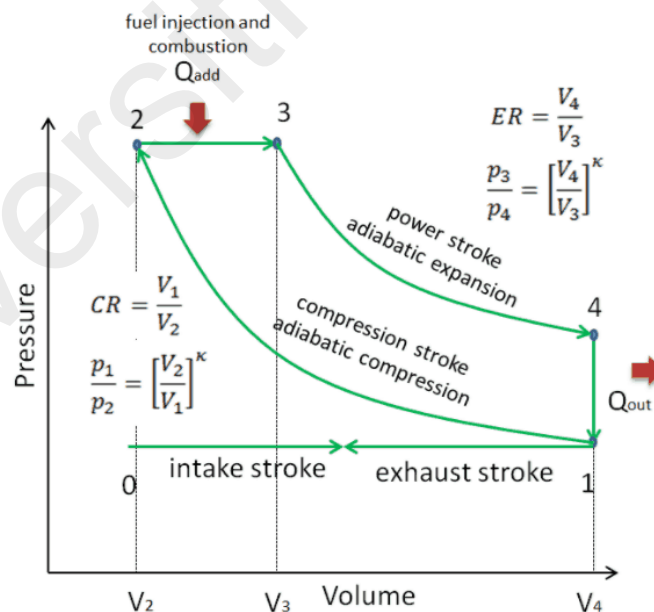
Figure 2.8: Pv relationship of an Diesel Cycle in relative to the Cycle Processes

The four strokes for a CI engine is the same as SI engine. To begin with the intake stroke the piston moves from the TDC towards BDC which allows the air to draw into the cylinder in sync with the intake port gradually open . The phenomenon takes place

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from point 0 to point 1 . Followed by the compression stroke , the piston moves up (from BDC to TDC) whereby the air is compressed to higher than the auto ignition temperature of the fuel. The compression stroke subjects to adiabatic condition whereby the process is isothermal , in other means no heat or mass transferred between the system and its transferred to its surrounding – may it be transferring heat through radiation or convection of heat . Resulting in exponential increase of pressure from point 1 to point 2 . Then followed by the combustion / power stroke - the piston approaches the TDC to complete 1 stroke , fuel were to be injected from the fuel injector and the instantaneously the combustion were to be occurred , propelling the piston to go downwards. Fuel is injected during the first part of the power stroke, resulting in a longer combustion interval. As well the heat is being added . This process occurs in a constant pressure from point 2 to 3 .For the power stroke , the result in decline of pressure exponentially defined in point 3 and 4.. Hence , the heat is being released to the surrounding due to the aftermath of the power stroke defined from point 4 to point 1 . Finally , the exhaust stroke – whereby once the piston reaches the BDC , the exhaust port opens drawing the residual and combusted gas out of the cylinder subjecting the piston to return to its original position and the cycle continues .

The Diesel cycle is identical to the Otto cycle whereby both are closed cycles for an internal combustion engines model . As mentioned previously , the variation between them is that the Diesel cycle is a compression-ignition cycle compared to Otto cycle which functions in a spark ignition cycle . Diesel fuels are infected with a fuel injector , to be mixed in such away to initiate efficient combustion at the proper thermal state so that Diesel engines runs fine.

Thus technically , the compression ratio of Diesel cycle and Otto cycle differs significantly , whereby the compression ratio for Otto cycle typically ranges somewhere from 8 to 11 and for Diesel cycle were ranged somewhere 12 to 20.

2.5 Fundamentals of Biogas Fuelled Gas Engines

With the present escalated energy demand of energy , renewable energy sources such biogas , biomass , solar and hydro were synergized to reconcile with fossil fuel and coal energy sources , accommodating the energy demand spike . In that case , countries such as India , Germany , Italy , Brazil ,Indonesia and etc , are exploring the use of biogas for its daily energy utilization and auxiliary use due to its highly appreciated potential as a gaseous fuel source (Santos et.al , 2016) . Biogas are reach with Methane (CH_4) constituent – depending on the type of sources used to harness the biogas. In tropical countries such as Malaysia and Indonesia , where palm oil industry is a major agricultural player . With abundance of palm oil production and its useful by-product from the production of the palm oil such as empty fruit brunch (EFB) and palm oil mill effluent (POME) – which yields high potential of biogas energy . As discussed in Section 1.1 , biogas is enriched with approximately 50 to 70 % of CH_4 . Which the gas is later desulphurized (using scrubber) and dried (using chiller), fed to the gas engines as fuel . However , the gas engine used to generate the energy is sensitive to the moisture content of the CH_4 . In other means , CH_4 being a naturally derived gas which has also high moisture content consisting of 60 to 70% . Thus , the gas is dried to eliminate the moisture content before to be fed to the gas engines (Saetor et.al, 2017). Figure 2.9 illustrates an example of overall process whereby the POME is converted to biogas in order to generate electricity . The design flow below are the standard process flow applied in the palm oil and renewable energy (biogas) industry. The process defines the stages and properties requisite of the POME substrate to useful biogas generation and utilization .
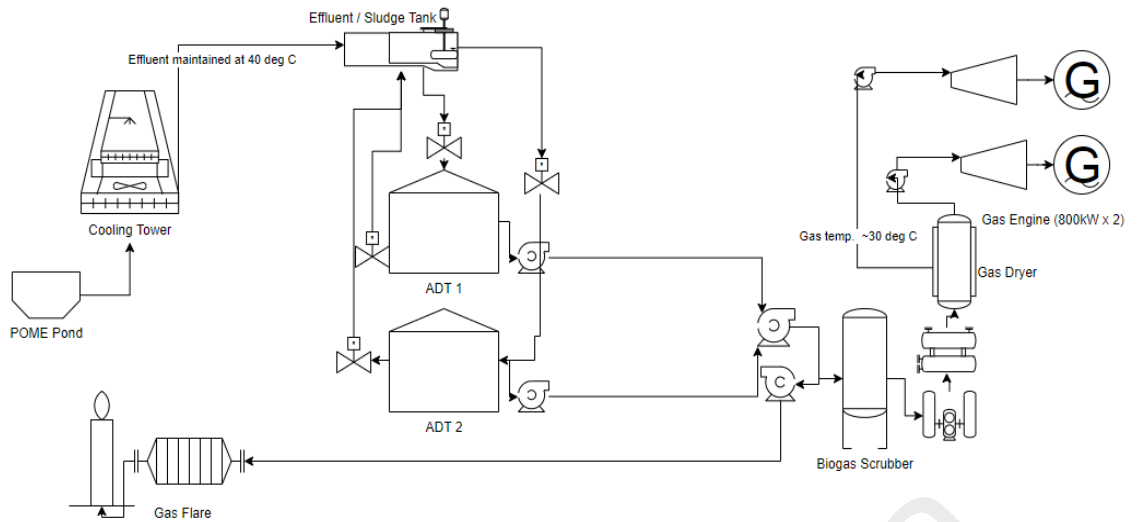
Figure 2.9 : Process Flow of a POME feed Biogas Powerplant

The POME is derived from the POME deposit pond , which were channelled to the Cooling Tower to maintain the POME substrate to 55 to 60 °C . As we know the , Hydraulic Retention Time (HRT) of a POME substrate in order to generate biogas would be 15 to 20 days (depending on the availability of the biogas captured in the Anaerobic Digester Tanks (ADT) . The POME substrate were later deposited into the effluent tank to till it achieves the desired (and recommended   HRT) . The POME which were undergone retention , were channelled to the ADT to trap the biogas via the Methanogenosis method – with the aid of methanogenic bacteria such as *Methanosarcina* (Arif et.al , 2020) . The trapped biogas is than channelled to the scrubber for desulphurization . The desulphurization is achieved via biological fixation by eliminating (or even minimizing) the H_2S content in the gas with the aid of microbes such as *Thiobacillus* (Dumont ,2015) . The desulphurized biogas is than sent to a chiller to dry to gas as to eliminate the high moisture content due to the organic reactions from the biogas generation and biological fixation and followed by drying the biogas and maintained the temperature from 25 to 30 °C then it is channelled to the gas engine to generate the electricity for the utilities or even auxiliary use of the plant or mill . However , at circumstances where the gas could not be used – the gas is flared due to limitation of containment of the biogas .

However , from the existence of the potential to use biogas as a source of fuel , the question which boggles the minds of engineers and scientist would be on why is biogas used in gas engines for combustion compared to be used in turbines or micro-turbines ? Biogas uses gas engines instead of turbines or micro-turbines to generate electricity due to its capacity to improve combustion efficiency and optimize its NO_x, greenhouse gases (GHG) and smoke emission (Ferozkhan and Ismail , 2017). In contrary to that , based on a recent case study research , whereby micro-turbines can be used as Combined Heat Power (CHP) application to reuse the residual heat from the engine combustion of the biogas engine which ultimately boosts the generating systems cumulative efficiency and capacity as well reduces the NO_x and GHG gases to the environment (Chang et.al,2019)

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2.5.1 Theory of Combustion in Gas Engines

The operating principles of gas engine functions according to the Otto cycle . Predominantly the gas engines operates with 4 stroke cycle application which is due to the increased temperature enabling the fuels to effectively evaporate , making it ideally suitable for such as gasoline based fuel . In addition to that , it increases the efficiency of the engine. Higher compression ratios also means that the distance that the piston can push to produce power is greater (which is called the expansion ratio) (Cho and He , 2009)

.

As mentioned in the previous section most gas engines are spark ignited engines , however there some engines which are compression ignited engines due to enhanced efficiency yield demanded. Commonly those engines are also known as hybrid engines , which operates with dual fuel combination system such as diesel and natural gas (Surata et.al, 2014) .
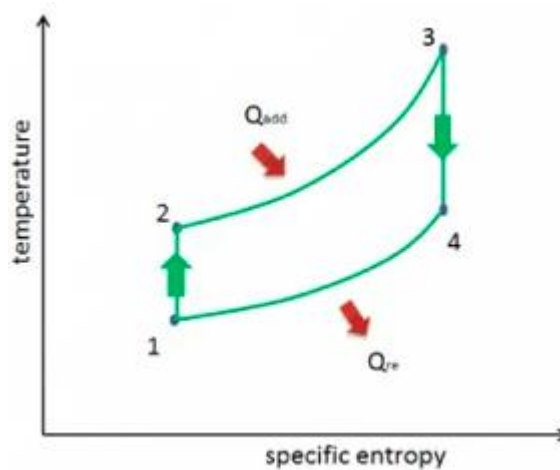
Figure 2.10 : Ts diagram of Otto cycle

The combustion science of the gas engines can be theorized using the Ts diagram of an Otto cycle. To begin with , at the intake stroke (point 1 to point 2); constant pressure is maintained as the air and fuel mixture is drawn into the cylinder in variance of increase of displaced volume. Allowing a significant increase in temperature. Followed by the intake, the compression stroke were to occur . During the compression stroke (point 2 to point 3), adiabatic compression condition will be employed whereby thermodynamic compression occurs without channeling heat and mass between the closed system and its ambience. In addition to that, the pressure gradually increases as the displaced volume reduces. The pressure increase of pressure is subjected to the van der Waals force acting upon the air and gas mixture increases relative to the reduction of displaced cylinder. In other words, as the volume reduces, the pressure increases due to the van der Waals force acting upon the particles of air and fuel mixture increases proportionally with the increased change of the entropy. This condition initiates pre-combustion conditions whereby heat is being added in the form of thermal energy released by the air and fuel mixture particles as the increase in van der Waals forces and rapid vibration within the particles . Therefore , ignition takes place subjecting to a combustion aided by the spark plug as the piston reaches at the TDC . The combustion will result to a power stroke (from point 3 to point 4) enabling the piston to move

downwards with no significant change in the entropy during this process . Thus , the power stroke results in reduction of temperature as the displaced volume of the cylinder increases enabling the particle of the combustion products to disperse as the volume increases . The thermal energy is then stored in the combustion products gases in the form of heat dispersed as the volume increases resulting during in the adiabatic expansion resulting in a drop of pressure. Finally , the combustion product gases is then purged out of the cylinder as the piston forces the gas channelling to the exhaust port by gradually opening the exhaust valve up (at point 4 to 1) . And the thermal energy stored in the combustion / exhaust gas in the form of heat is released which is indicated in the reduced temperature. A constant pressure is maintained during the exhaust.

2.5.2 Engine Performance Fundamentals

The performance of an engine is determined by several parameters such as the effective power (P) , Specific Fuel Consumption (SFC) , Thermal efficiency (η_{th}) and Mean Effective Pressure (MEP) which is correlated to parameters such as engine's Effective Torque and engine's rotational speed (RPM). The effective power (P) is relatively influenced by the engine's rotational speed and torque (Arjuna et.al , 2018) .

$$P = \frac{2\pi N\tau}{60000} \qquad (2.11)$$

Where the unit of P is in terms of kW.The τ is denoted as the engine's effective torque (unit : Nm) N is defined as the rotational engine speed (RPM) .The torque were to be generated at the crankshaft on each cam rod journal, every time the piston is in the power stroke which contributes to the delivery of the effective power (Garcia , 2018). The SFC is the fraction of fuel that were consumed over a unit power in generating hourly operations, which in other words indicates the engine's efficiency producing

power derived from the fuel combustion. the equation below denotes the SFC of an engine;

$$\text{SFC} = \left(\frac{\dot{m}}{3600}\right) \times \left(\frac{1}{P}\right) \times 1000 \qquad (2.12)$$

Which the ṁ is denoted as the fuel's mass flow rate (kg/s) . The SI unit defined by the SFC is (g/kW.h) . The lowest SFC, is directly proportional to the engine's rotational speed (RPM) for that. Functioning with a low RPM reduces friction in the engine and enhances the air intake capacity (volumetric efficiency).

Followed by that, the next parameter determining the engine's performance would be the thermal efficiency . The thermal efficiency has an significant indication on the engine's performance through the effective power delivered with respect to the heat released . In other means , the thermal efficiency of an engine is defined as the ratio between the output energy which were derived from the effective power done by the crankshaft and chemical energy of the fuel (whereby in this case it is the raw methane from the biogas) (Ma et.al, 2015) . The equation of the η_{th} is denoted as :

$$\eta_{th} = \frac{W}{\dot{m}\eta_c Q_{hv}} \qquad (2.13)$$

Where the η_c is the combustion efficiency were taken into standard assumption of 0.97 (97%) and the Q_{hv} is the caloric value of the fuel (in this it is derived from the biogas, 21500 kJ/kg) . The MEP is an ideal parameter to gauge the engines performance relative to the design integrity and output due to the engine's dimension (size and volume) and rotational speed . The MEP is split into Indicated Mean Effective Pressure (IMEP) and Brake Mean Effective Pressure (BMEP) whereby the IMEP is the pressure subjected on the piston in a full stroke which provides the work output (Fiorini et.al,2019).

The BMEP is the averaged mean pressure acts upon the piston full strokes which can also be defined as pseudo-MEP and it's a theoretical parameter . The MEP is denoted as in the equation below ;

$$MEP = \frac{4\pi(SFC)}{V_d} \times 1000 \qquad (2.14)$$

Where the V_d , volume displaced in the cylinder(s) (measured in m^3) . The unit of measurement for MEP is kPa .

2.5.3 Heat Release Analysis

Heat release rate analysis (HRR) and cylinder pressure are excellent parameters to determine performance of an engine relative to the combustion and thermal efficiency especially in developing and tuning engines (Shehata , 2010) . In thermodynamics perspective , the conservation of mass and energy is used for assessing the closed system model assumptions simulated by the engine cylinders in reference to the first law of thermodynamics where in thermodynamics point of view , the model can be considered as single zone (0 Dimensional) and multizone . As our emphasis is on single zone , where the scope of research limits with the cylinder as in Figure 2.11 .
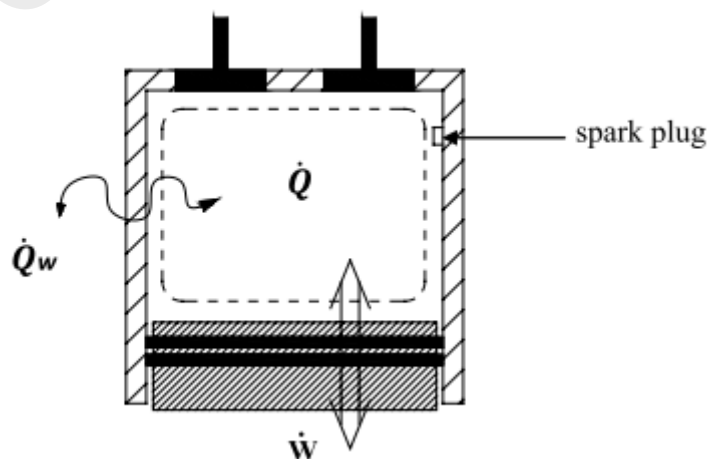
Figure 2.11 : Closed cylinder system model for thermodynamic analysis .

The heat transfer through the cylinder walls plays a significant function in engine's combustion and performance behaviors which is sub-due to the hypothesis where wall

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temperatures are relatively lesser than the maximum temperatures of the burned gases in the cylinder. As we are aware, heat travels downscale gradient whereby the high temperature to lower temperature. There are two main methods which are often referred to as "burn rate analysis" and "heat release analysis". Burn rate analysis are used for obtaining the burn angles in gasoline engines in this case and is used to attain the mass fraction burned (MFB) which is a scaled quantity of 0 to 1 (synonymous to 0 to 100%). The mathematical equation which denotes the MFB or f are as in Equation (2.9):

$$f = 1 - e\left(-a\left(\frac{\theta - \theta_{soc}}{\Delta\theta}\right)^{m+1}\right) \tag{2.15}$$

Equation (2.15) is also known as the Wiebe equation . Where a and m values are fixed to 5 and 2 respectively . The θ_{SOC} is the crank angle which the start of combustion (units are in degrees, °) and θ as the instantaneous crank angle (measured in degrees , o) followed by $\Delta\theta$ which is the duration of combustion .

The mathematical equations of the HRR , in general, a derivative form a set of ordinary differential equations with an independent domain , which are the time or the crank angle (Wallson , 2018) ;

$$\frac{dQ}{d\theta} = \left(\frac{\gamma}{\gamma-1}\right) P \left(\frac{dV}{d\theta}\right) + \left(\frac{1}{\gamma-1}\right) V \left(\frac{dP}{d\theta}\right) + \left(\frac{dQ_{wall}}{d\theta}\right) \tag{2.16}$$

$$\gamma = \frac{C_p}{C_v} \tag{2.17}$$

Where γ is with respect to the cylinder(s) temperature (measured in Kelvins, K) ; represented by the ratio of Specific Heat capacity at constant pressure , C_p and Specific Heat Capacity at constant volume , C_v. Thus , similarly as HRR equation (2.16) used , the First Law of Thermodynamics law governed by Equation (2.18) through simplifying the entire engine's system in a cylinder as a closed system and neglecting the effects of thermal dissipation on the wall of the engine's cylinder by resolving the energy which goes in and out of the close system barrier .(Cengel and Ghaffar ,2020).

$$\frac{\delta Q}{\delta\theta} + \left(1 + \frac{C_v}{R}\right) P \frac{dV}{d\theta} = \frac{C_v}{R} V \frac{dP}{d\theta} \tag{2.18}$$

The R is the gas constant of gas and fuel mixture which is denoted as kJ/kg.K and derived from the mixture of gas (CH_4) and air which is pre-mixed during the intake and combustion process.

The volumetric efficiency – or also known as the ' breathability' of the engine is defined as the proportional ratio of mass of air-fuel mixture drawn into the engine's cylinder(s) relative to the atmospheric pressure subject to the volume of air drawn via the intake manifold . The volumetric efficiency is governed by the equation as below :

$$\eta_v = \frac{m_{fc}}{\rho V_d}$$ (2.19)

Where η_v is the volumetric efficiency and its dependent variables m_{fc} (is the mass of fresh charge during compression in the cylinder , kg) and ρ and V_d are air density (kg/m^3) and volume displaced (m^3) respectively (Luka , 2013).

2.6 Summary

For this research , the engine performance and HRR analysis will be modeled via MATLAB whereby an industrial scaled gas engine with a capacity of 800 kW will be evaluated based on the data available from the site as well using the equations in this chapters . Since this engine is Otto cycle based, therefore we could observe significant increase in the pressure in the cylinder at the compression stroke which results in the thermal efficiency of the engine whereby compensated with the MEP's fluctuation . According to the literature which was reviewed for this research the HRR analysis could be evaluated using the Wiebe function analyzing the change in heat with respect to crank angle. Besides that, the fuel economy of the fuel (which is represented by SFC parameter) influences the engine's performance. whereby the SFC compensates when operating at elevated RPM (or even running load) to prevent the overflow of the fuel. Besides that, this research would enables us to study the contributing factors influencing the performance and HRR distribution. From the modeling of this research , potential

recommendations can be deduced on optimizing the parameters of the gas engines such as MEP , Torque and Effective Power as well design components such as Bore and Stroke length and Displacement volume for more efficient and enhanced performance gas engines which operates with biogas .

CHAPTER 3

METHODOLOGY

3.1 Introduction

Chapter 2 discussed on the fundamental concepts and literatures of biogas energy , internal combustion engine fundamentals and the advancements , as well as developments undergone by internal combustion engines especially in the modernization of those engines to support biogas as fuels . These literatures plays a significant role for the background research of this project and would be a reference especially incorporating the key concepts translating into MATLAB codes and to analyze the physics and rationale behind the performance study.

However, for this chapter we would explore the research and programming methods employed to fulfil the objectives of the research. It would also define the data collection methods and its alternatives approach which would translate into MATLAB codes at the later part of the research. Also, this chapter discusses on the set up and configuration of the research whereby since it's a case study real life application. In this case, the data were obtained in operational biogas plant in Teluk Intan , Malaysia . However, extension of the data obtained, especially on the gas engine performance were based on calculations derived from the basic data obtained from the site.

In addition to that, the limitation and uncertainties were discussed here as part to support the challenges faced during this research project. Thus, the outline of this chapter as in following order;

(a) Research Design

(b) Background Research on Biogas Energy and Fuelled Gas Engines

(c) Data collection and development

(d) Data validation and Review

(e) MATLAB Programming

(f) Data Analysis and Interpretation

However , certain information which were considered as part of the propriety of the design were amended to avoid any breach of intellectual property rights and to withhold the sensitivity of the propriety . The timeline of the research are tabulated in the Gantt chart attached in the Appendix (A.4.0) .

3.2 Research Design and Methodology Sequence

The research design was made sure to follow in a such rational techniques in order to obtain a legitimate result without deviating the scope and objective of the research. In addition to that, there questions which would guide on the research design correlating with the Problem Statement and amongst it would be:

(a) How does the parameters such as Specific Fuel Consumption (SFC), Mean Effective Pressure and Thermal Efficiency contribute to the optimized efficiency?

(b) Does the burned or combusted mass fraction of the air-fuel mixtures proportionate with the stroke stages in the 4-stroke gas engine?

(c) How does the Heat Release Rate reflect on the overall performance of the engine?

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However, the research design shall reflect on these question and a structure of research methods were employed for obtaining the results as well as to analyze those data. The research design consists of combination of Secondary Data Analysis and Partial Experimental techniques. However, there were several limitations and hurdles faced when planning the research design. Amongst it was the crank angle encoder were not installed in the engine which also serve as a data logger to gauge the pressure and temperature in the engine cylinders as well as the torque of the crankshaft. Therefore, the crank angle encoder which were not install limits our capability of obtaining the actual data such as the pressure and temperature in the engine cylinder(s). Thus, as an alternative the parameters were obtained through a series of iterative calculation to obtain it and makes it difficult to validate the data obtained through calculation. Besides that, the limited research on real life industrial application with similar (or approximate) capacity are scarce which also difficult to validate the data and its analysis. However , the data were benchmarked with similar or approximate case such as ; benchmarking research on engine performance for diesel engine or even hybrid one (gas engine with auxiliary diesel feed) . Through this, we could benchmark the science and justification behind the performance study in respect to the research conducted by me.

The Figure 3.1 below illustrates the methodological approaches and step phases for this research right from kicking off to concluding the research. As an initial approach , the study on the biogas power station and generation were executed as part of the background research .The objective of this background research is to obtain fundamental theory on the generation of biogas and as well as the gas engine to generate it . In addition to that, several literatures such as journal articles, thesis and even books were taken into reference as part of the background research. A general study on the effect of biogas generated gas as a source of fuel to feed in to the gas engines.

Background Research on the Gas Engines

•A general background research were done prior to collect the data at site .
•The scope of the background research consists of the engine specification and technical datas such as mechanical , electrical and thermal efficiency as well as the mean effective pressure .

In Situ Data Collection

•Data collected from the site chosen for the research
•Raw Data such as the Biogas flowrate with respect to RPM selection and capacity

Data Development

•The data available will further be developped using calculation tools and equation to obtain the performance parameters such as the internal cylinder volume , pressure in the cylinder as well as the Specific Fuel Consumption (SFC) , Thermal Efficiency and Mean Effective Pressure with respect to Engine loads and RPM .

MATLAB Programming and Simulation

•Based on the data developped a series of MATLAB codes were programmed to simulate the results

Data Analysis

•Based on the MATLAB modelling , the data is analyzed to deduce a conclusion .

Figure 3.1 : Methodologies employed for the research

Hence , the specification of the gas engine were analyzed as well which also appears to be a benchmarked reference to support the data developed as part of the study on the gas engine performance which were tabulated at the Equipment Specification .

Followed by the background research, the data was collected which were the data(s) applied at site. This data would be the building blocks of the data development, modelling and analysis. However, there are several data which are able to be collected which are the gas flow rates with the function of engine capacity load and RPM. However there were certain constraint identified when obtaining those data which were to be discussed in the In Situ Data Collection section.

With the data obtained, a series of calculations were employed to develop to obtain other parameters such as the MEP , TE , SFC , Cylinder Pressure , Volume displaced , mass fraction with respect to Crank Angle . Which than contributes to the HRR model for a single cylinder and all 16 cumulative cylinders.

Prior to the data developed, MATLAB codes were programmed to model and simulate the performance parameters which would define the trends of the parameters in line with the operational parameters from the data .

Upon completing the MATLAB programming, data analysis was done especially on the models generated from the programming developed. The scope of analysis would subject within the performance parameters and its contributing factors justified by the references and theories benchmarked. The analysis would guide , on whether or not the research were fulfilled through the modelling of the performance parameters and would recommend for further development in the near future .

3.3 Test apparatus / Equipment

The gas engine used for this research is the Catepillar's CG132(g)-16 gas engines where this engine is powered using natural gas derived from biogas . With approximately 25,200 hours of running since its first Initial Operating Date (IOD) on February 2017.

Table 3.1: Specifications of the test gas engine

Gas Engine Specification	
Engine Model	CG132b-16
Manufacturer	Caterpillar
No. of Cylinder (Type of Cylinder)	16 (V)
No of Strokes	4
Bore (m)	0.132m
Stroke (m)	0.16m
Displacement (m^3, for 16 cylinders)	$0.0349995m^3$
Nominal Capacity	800 kW
RPM	1500RPM
Thermal Efficiency	42.30%
Electrical Efficiency	42.80%
Total Efficiency	85.10%
Mean Effective Pressure (bar)	18.9

The engine's detailed specification are attached in the Appendix at A.3.0 .

3.4 In-Situ Data Collection

Due to the travel restrictions during the peak of the Covid-19 pandemic, the data collection were challenging. Thus, with the available data from the site were used as it was consistent and reliable. The data obtain at site, was the flow rate of the biogas derived methane to the engine as well in relative with the engine running load . However, for the cylinder pressure and HRR versus with crank angle data were not able to retrieve due to the unavailability of the Crank Encoder and Cylinder Pressure Sensor . Hence, data development were required to develop the data such as to obtain the Torque of the Engine, SFC , MEP and TE . Follow up to it , whereby the Phase 2 of the research which deals with obtaining the mass fraction of air-fuel mixture , cylinder pressure and volume displaced with respect to crank angle and the HRR .

3.5 Data Development and Modeling

Due to the constraint of data obtained at site , it was necessary to use numerical models to develop further data parameters which also serves as the building blocks of the MATLAB programming for the modelling of the gas engine's performance .. The flowchart were designed keeping in mind the data which were obtained at site and based on those data logical sequences of parameters were developed in function with the respective equations defined in the flowchart . Upon that , the flowchart were translated into MATLAB code sequences which will later on models the gas engine's performance and the HRR analysis . There are several simulation programs such as OCTAVE , SciLab , ANALYTICA and many more which could model or even numerically simulate the performance of en engine and even model the HRR of an engine and amongst all those simulation tools and software MATLAB is regarded the most efficient and reliable . What makes MATLAB a better option for modeling solutions such as in our case in modeling the engine's performance and HRR ? Relatively in comparison with the other competitive

numerical analysis software mentioned above , MATLAB has a better user friendly interface and does not require sophisticated programming understanding to model a solution . Secondly , in the case of engine performance and HRR modeling which does not limit to a specific capacity , say for an instance modeling an engine based on 800kW capacity with the aid of features such as Artificial Neural Network (ANN) in capturing the model in the form of neural network sequence for any capacity of engine for similar researches . All would require is to amend the key parameters such as the engine capacity and design parameters such as the Bore length , Connecting Rod length , and Displacement Volume (Kianmehr,2014) .

However to validate the data and plots generated through via the MATLAB programming ,it was first worked with the Microsoft Excel to obtain the simulated data using the equations listed in the Equation list . Upon completion of the calculation using Microsoft Excel and the data were generated from it , the plots were later compared with the plots which were to be generated by the MATLAB simulation (Fiorini et.al,2019).

Besides that , several literature references such as relevant text ("Internal Combustion Engine Fundamentals " by John B.Heywood) and several journal articles related to the research project were referred to validate the theory and calculations.

For the engine's performance study, the approach undertaken when it comes to model , simulate and code in MATLAB is straightforward . With the data of biogas flowrate relative to the engine running capacity (operational load) and rotational speed (RPM) , the key performance parameters such as Power , SFC , Thermal Efficiency and MEP are derived from Equation (2.11) , (2.12) , (2.13) and (2.14) respectively .Since the gas engine has torque regulator modes where the torque operates according to the running capacity (operational load). In other words , when the engine runs in 100% capacity , the torque employed 5092N.m (which is also the maximum torque)with respect to 1500rpm and for 50% the torque is regulated to 50% of the maximum torque . Followed by that , the

equations were translated into MATLAB codes where the performance were modeled .
A flowchart were attached at the Appendix A.0 which defines the simulation sequence
based on the parameters and equations which is also serves as the MATLAB
programming guide.

However, for the study on the HRR analysis, each parameters were evaluated and
analyzed based on taking in specific assumptions and derivatives of equation taken into
consideration. For instance, the temperature in the cylinder varies with as tolerance of +/-
50°C in actual case. However, to simplify the analysis average cylinder temperature were
made into assumption to standardized the analysis. Besides that , due to the insufficient
cylinder pressure vs crank angle data were not available with no crank encoder installed
at the site – Thus , Ideal Otto Cycle state were taken into assumption . As defined
(Mauro,2018) , the cylinder pressure function its significance in deriving into the HRR
analysis . Therefore, the cylinder pressure model was based using the Ideal Otto Cycle
according to Equation (2.3), (2.4), (2.5) and (2.6) relative to the instantaneous crank
angles. With then data on the cylinder pressure and volume Equation (2.7)), with respect
to its derivatives – Thus, MFB (f) and $\delta f/\delta \theta$ were able to be evaluated and apparently
would model the HRR of the engine cylinder and later translated into 16 cylinders, as the
engine consists of 16 cylinders. The engine specification will be taken into reference
when modelling and developing the data as parameters such as the bore , stroke ,
displacement volume , speed and mean effective pressure were used frequently especially
in the Heat Release Rate (HRR) analysis where the displaced volume of the engine
cylinder were used to derive into the Weibe Model function in Equation (2.15) .

3.6 Error Analysis

The error analysis were done for engine performance with reference to the
manufacturers specification. As mentioned earlier, that the key parameters such as the

Power, SFC, Thermal Efficiency and MEP were modelled and calculated based on the Equation (2.7) , (2.8) , (2.9) and (2.10) respectively . Then, the values calculated were compared with the manufacturer's specification listed on Table 5.1 to perform error analysis. The equation below were used to calculate the error percentage of the performance parameters.

$$\% \, ERROR = \frac{(Calculated \, Value) - (Actual \, Value)}{Actual \, Value} \qquad (3.1)$$

The error were benchmarked to +/- 5% allowable threshold where higher than the threshold value would be subject to discussion in Chapter 5 .

3.7 Limitations

The limitations to be factored when conducting the research are as following:

(a) There were no crank angle encoder installed in the engine , which makes it difficult to obtain real-time data such as the pressure in the cylinder with respect to crank angle (Bueno et.al, 2012)

(b) There was frequent temperature fluctuation in relative to time with higher deviation (+/- 50°C). Thus, during the analysis, the average temperature in the cylinder were taken into consideration for standardized calculation and analysis.

(c) The analysis, were concentrated to per cylinder to provide a focused analysis on the engine's performance. However, the parameters such as the HRR were translated into 16 cylinders.

(d) For the analysis and obtaining the data and results, thermodynamically quasi dimensional engine cycles were factored in.

CHAPTER 4

RESULTS

4.1 Introduction

This chapter deals with the results and observation obtained from the research which also explains the fundamental trend of the modeled plots . As mentioned earlier , the research were split into 2 phases of scope which are ; the gas engine's performance study (which consists of study on performance parameters such as SFC , MEP , TE and Effective Power) , and the second phase of scope deals with heat release analysis of the gas engine , (which incorporates a more focused approach) . Due to the unavailability of the data from the manufacturer, the comparison of the data generated from the numerical and MATLAB modelling . Thus parameters such as the MEP , TE and Effective Power were compared using the data obtained from the manufacturer's datasheet .   The program codes and results tables were attached in Appendix A.1.a and A.2 respectively .

4.2 Scope 1 : Modeling of Gas Engine Performance

The graph plotted in Figure 4.1 defines the RPM of an engine in domain of the engine running load (or also known as functional load). The relationship trend demonstrated by RPM and the running load is linear. However this is to the effect of the safety feature employed, which is the RPM regulator which functions and regulates the engine's RPM based on the engine's running speed. This feature also serves as a safety feature of the engine from overloading when it is running in a lower running load . Hence the fuel economy of the gas engine were to maintained. The maximum nominal rotational speed is 1500 rpm . However, in ideal cases , the RPM is maintained in maximum so that engine does not trip-off.
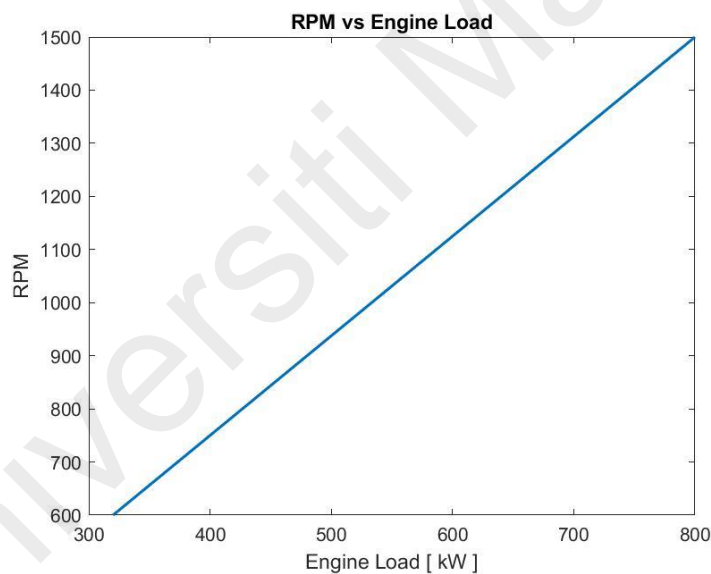
Figure 4.1 : Graph on Engine's RPM vs Engine Load

The graph plotted in Figure 4.2defines the Engine's Torque in domain of the engine running load . Similar to the plot RPM vs Running load, the plot in Figure 4.2 also demonstrates a linear relationship which also due to the RPM regulator . Higher RPM results to more fuel burnt in concurrent and more power generated in domain of time. Thus more torque is also generated . Since the regulator controls the RPM according to

the engine running load , therefore the torque is regulated according to the RPM . The maximum torque achieved at its functional running load which is 5092.874 Nm , which is attained at its highest functional capacity of 800 kW (equivalent to 100% running load) .
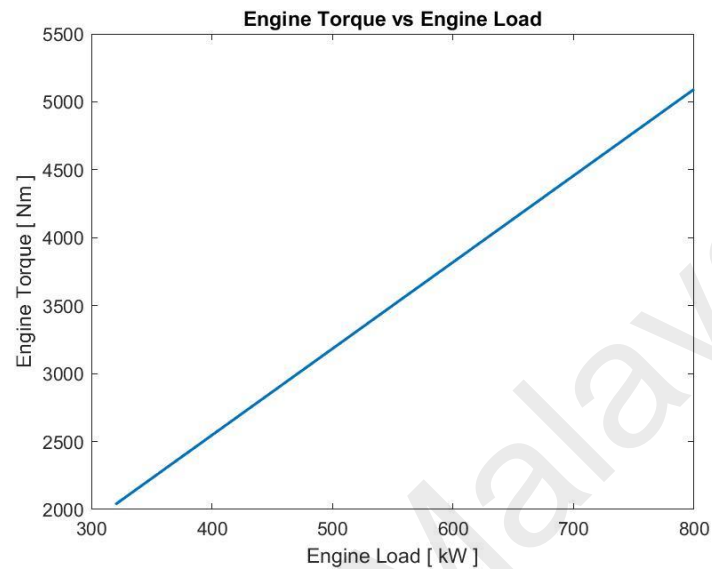
Figure 4.2 : Graph on Engine Torque vs Engine Load

The Figure 4.3 , defines the trend of the effective power in domain to the engine running load which appears to be in a partially linear trend with a slight trace of curvature . The maximum effective power were attained at the highest running load of 800 kW, which is 799.581kW . Since the RPM and engine load is linearly related which also makes it that the effective power is linearly related to the effective power .
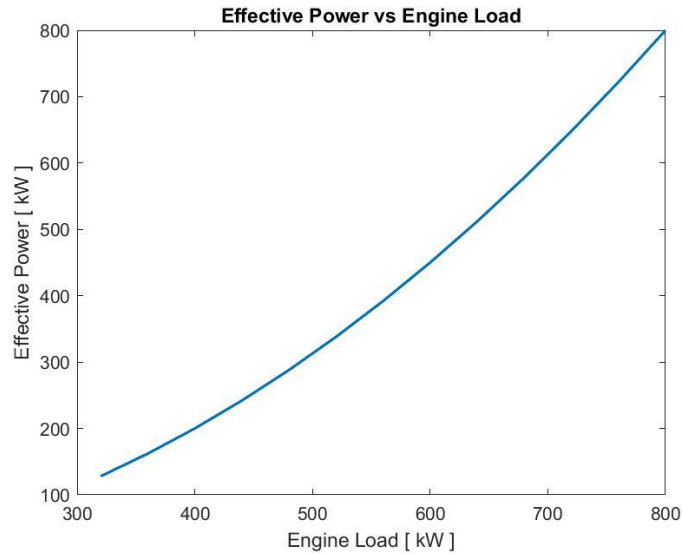
Figure 4.3 : Graph on Effective Power vs Engine Load

In contrary to the engine's effective power delivery , the SFC parameter is defined to a inversely exponential trend with respect to the running load domain . Whereby the highest SFC value is obtained when the engine is operating under the lowest engine's running load (300kW) which is 0.1354 g/(kW.h) . However for the maximum running load of 800kW , the engine's SFC is recorded as 0.0377 g/(kW.h) .
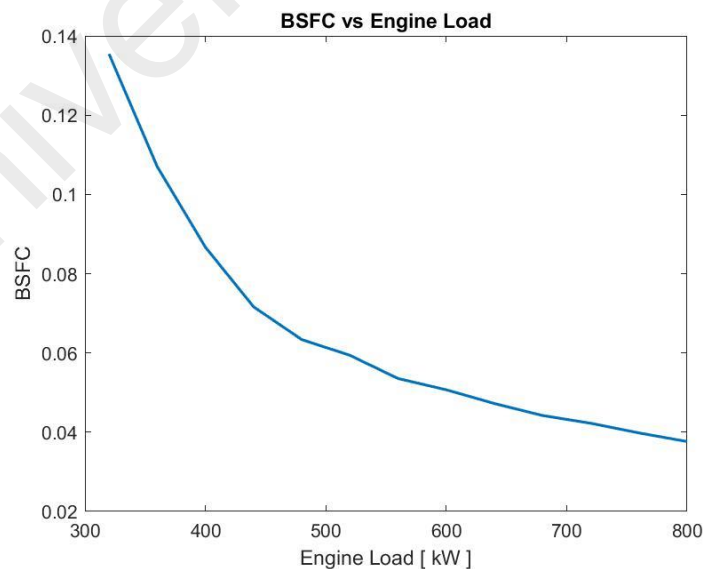
Figure 4.4 : Graph on SFC vs Engine Load

The thermal efficiency of the gas engine is proportional to the running load as in Figure 4.5. The highest thermal efficiency is achieved to be at 800kW running load where the thermal efficiency is 49.32% . The thermal efficiency were regulate with respect to the turbocharger and the water coolant jacket as the engine operates at elevated capacity via increasing the air pressure intake to be drawn into the engine cylinder(s) increasing thermal efficiency .
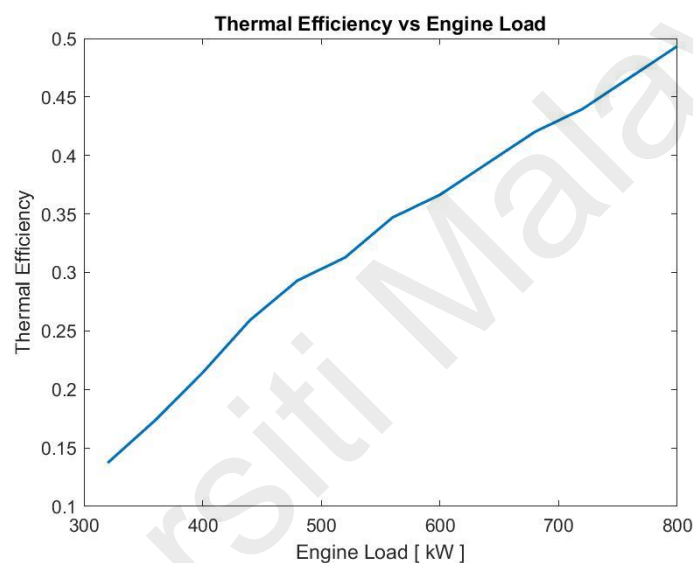
Figure 4.5 : Graph on Thermal Efficiency vs Engine Load

The final parameter of the engine performance would be on the MEP . As in Figure 4.6 , the MEP appears to be directly proportional to the running load . The maximum MEP is achieved at its highest running load capacity , which is 1824.153 kPa (equivalent to 18.2 bar) . As discussed on the turbocharger's role in regulating the thermal efficiency , the MEP is compensated through increasing the air pressure intake .
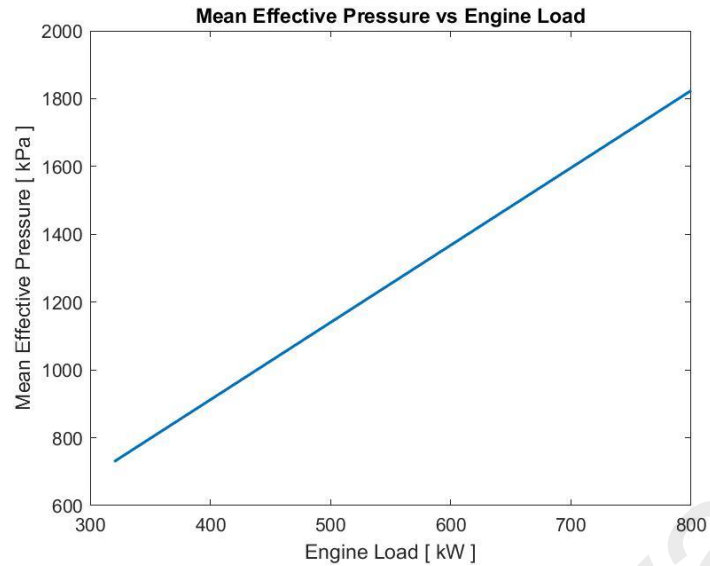
Figure 4.6 : Graph on Mean Effective Pressure vs Engine Load

4.3 Scope 2 : Modelling of Performance Study and Heat Release Analysis

For the scope 2 of the research , the analysis and evaluation is focussed to per cylinder . Figure 4.7, illustrates the graph on the trend of the Cylinder pressure vs crank angle . From the figure below, the pressure achieves its peak of 1266.563 kPa (equivalent to 12.66 bar) at $360^{o.}$ At this point , is where the combustion takes place which pre-induces the power stroke .
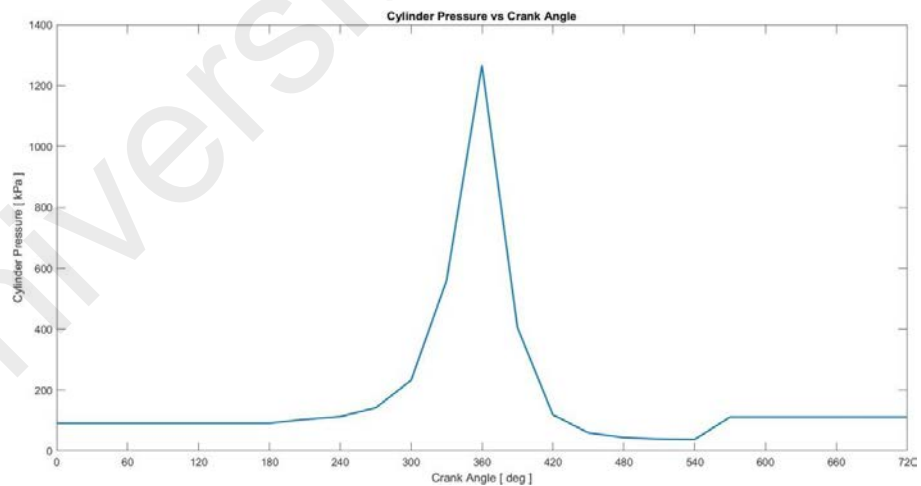
Figure 4.7 : Graph on Cylinder Pressure vs Crank Angle

At Figure 4.8, a graph on comparison between the cylinder pressure and IMEP vs the crank angle were plotted . For the IMEP plots , it fluctuates from 300° to 420° which the highest peak is 122.486 kPa (equivalent 1.17 bar) to 1at 330° . The IMEP readings are

smaller than Cylinder pressure which is due to the pressure acting on the piston during pre-combustion and combustion process . The IMEP appears to be almost 0 , which becomes the initiation point where the work output of the piston being enforced to the piston . Thus , from 360° to 420° the pressure begins to rise and drop due to the combustion and power stroke to occur respectively.
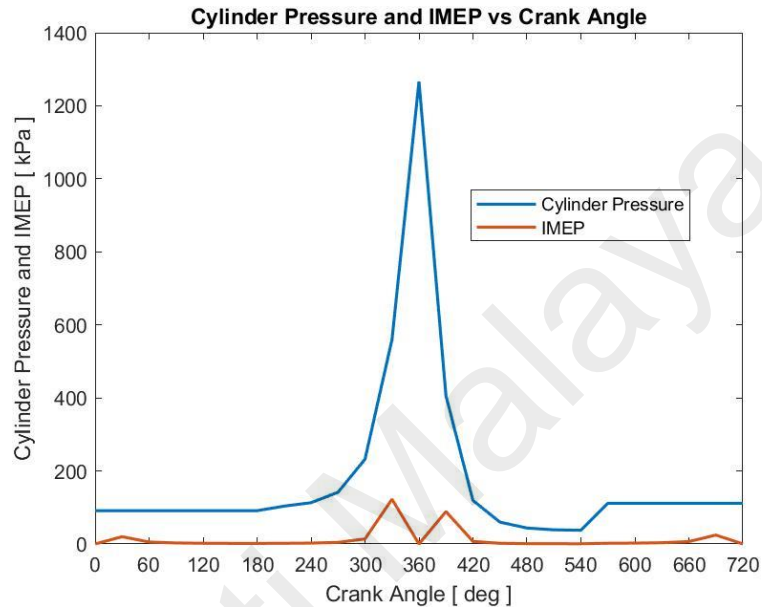
Figure 4.8 : Graph on Comparison of Cylinder Pressure and Indicated Mean Effective Pressure with respect to Crank Angle.

Figure 4.9 , illustrates the graph on the change of volume with respect to the crank angle . The change of volume with respect to the crank angle appears to have a rapid fluctuation which is due to the motion of the piston , but the scattered fluctuation may result in the anomalies of the readings obtained through the calculation .
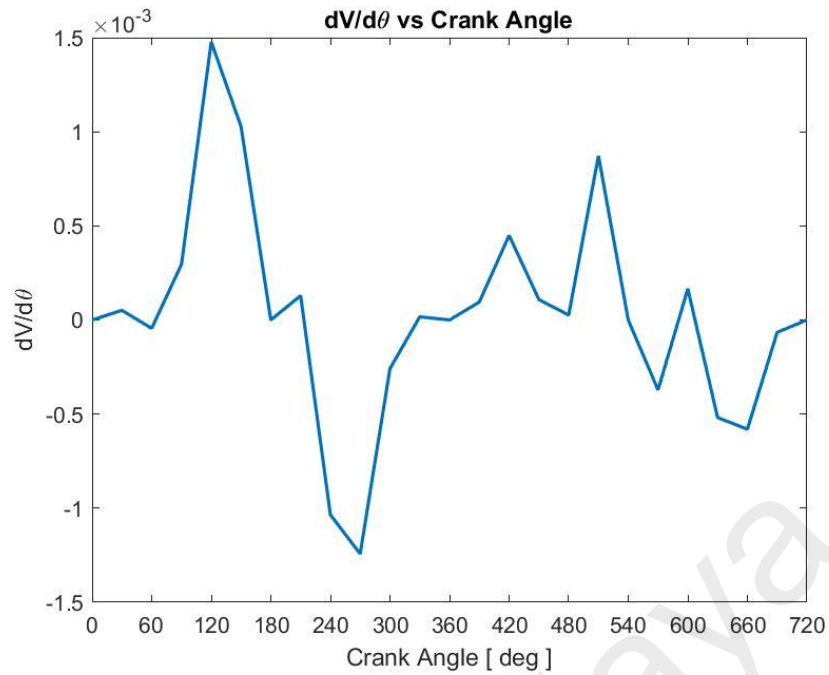
Figure 4.9 : Graph on Displaced Volume changes over Crank Angle domain vs

Crank Angle

Figure 4.10 , illustrates the graph on the fraction of heat added with respect to crank angle . The heat is began to add at the 240° crank angle mark with gradually increasing until it settles to 1 at 540° . The fraction of heat added indicates the heat added to the cylinder , and since at crank angle 240° the combustion begins to take place .
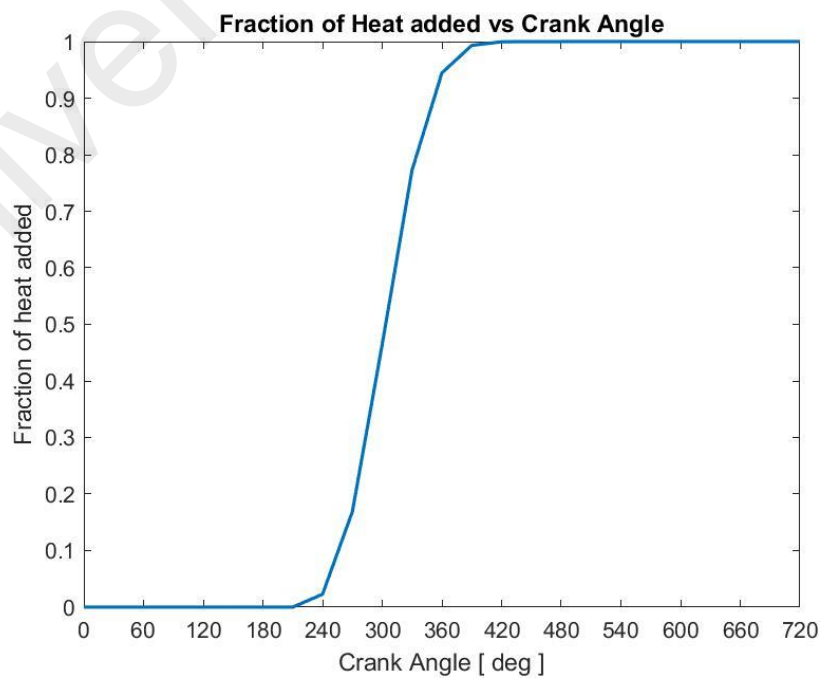
Figure 4.10 : Graph on Heat Added Fraction vs Crank Angle

Figure 4.11 and 4.12 illustrates the graph on the HRR of a cylinder and later translated to 16 cylinders . Both graph demonstrates a fluctuating trend in rhythm with the changes in volume (represented in domain of chang angle position) . This fluctuation also correlates with each stroke processes .
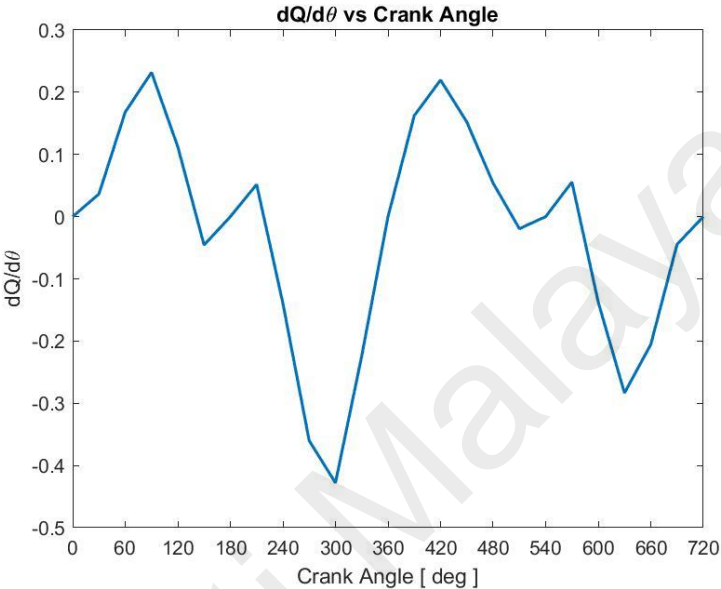
Figure 4.11 : Graph on Change on Heat Release vs Crank Angle
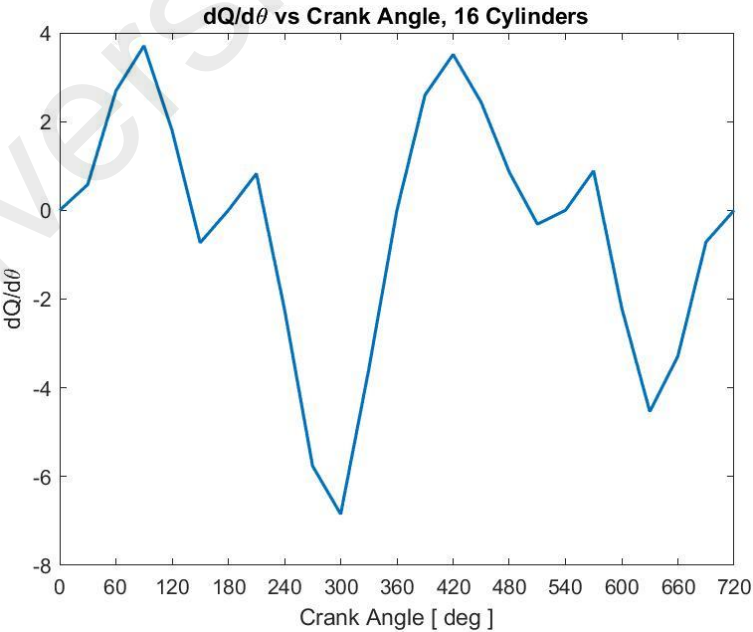
Figure 4.12 : Graph on Change on Heat Release vs Crank Angle for all 16 Cylinders
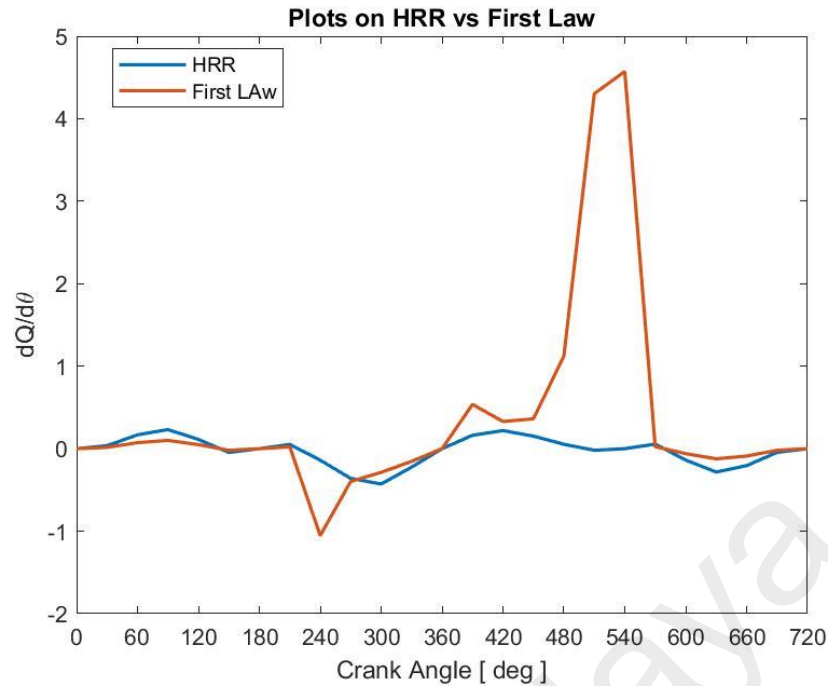
Figure 4.13 : Graph on HRR In Comparison of Conventional Heat Release Equation and First Law of Thermodynamics vs Crank Angle for all 16 Cylinders

Figure 4.13 defines the comparison of HRR obtained from the conventional heat release equation (Equation 2.16) and First Law of Thermodynamics (Equation 2.18) . Based on the comparison from both the plots in the graph in Figure 4.13 deduces a similar trend in fluctuation . However , there were a sudden spike on heat released in obtained at crank angle 510º to 540º which results in a peak of 4.30W/deg and 4.57W/deg respectively . The HRR obtained through conventional heat release equation indicates a steady fluctuation .The plots using HRR obtained through conventional heat release equation is represented in Figure 4.12 .

From Figure 4.14 , the relationship of Volumetric efficiency (also known as the "breathability" of the engine) were analyzed to study the relationship of the volumetric efficiency respective to different RPM . The plot generated were based on theoretical value which were calculated based on the parameters such as the displaced volume and the mass flow of air measured which is 0.00117 kg/s . From the plot , its to be deduced

that the lower the RPM the higher its higher its "breathability" . The peak of "breathability" of the engine of 1500 rpm , 1000 rpm and 500 rpm are 3.8 , 5.7 and 11.2 respectively .
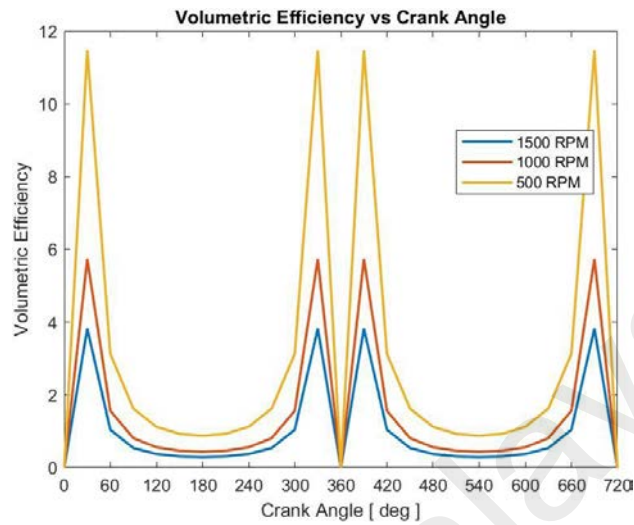
Figure 4.14 : Graph on Volumetric Efficiency of Invariance of Engine's RPM vs Crank Angle.

4.4 Results Table

The results of the analysis were tabulated with Microsoft Excel, and the data were based on the calculation according to the flowchart in Appendix A.0. Table 4.1 is the results obtained tabulated for scope 1 and Table 4.2 is the table tabulated for scope 2. (Refer Next page for the Tabulated Results) .The data for running capacity were blanked as ERRO message displayed (not able to retrieve the data when the engine operates in running capacity lesser and equals to 35% , whereby the engine trips for load running capacity equals or lesser than 35%.

Table 4.1 : Scope 1 (Engine Performance) Results

No	Engine Load ,kW	Running Capacity	ADT BIOGAS VOL , m3	Gas Flow , m3/hr	Density , m3/kg	Mass Flowrate to Engines 1+2 , kg/hr	Mass Flow rate to single unit Engine , kg/hr	Torque of Engine	RPM	Effective Power,kW	BSFC	Thermal Efficiency	Mean Effective Pressure , kPa
1	0	0.00%	0	-		-	-	0	0	0			
2	40	5.00%	1.64	-		-	-		75				
3	80	10.00%	3.28	-		-	-		150				
4	120	15.00%	17.64	-		-	-		225				
5	160	20.00%	32.00	-		-	-		300				
6	200	25.00%	40.96	-		-	-		375				
7	240	30.00%	49.9	-		-	-		450				
8	280	35.00%	107.02	-		-	-		525				
9	320	40.00%	164.14	190		124.83	62.415	2037.1496	600	128.0144809	0.135433897	13.72%	729.6613151
10	360	45.00%	287.07	190		124.83	62.415	2291.7933	675	162.0183273	0.107009499	17.36%	820.8689795
11	400	50.00%	410.00	190	0.657	124.83	62.415	2546.437	750	200.0226264	0.086677694	21.43%	912.0766438
12	440	55.00%	615.00	190		124.83	62.415	2801.0807	825	242.0273779	0.071634458	25.93%	1003.284308
13	480	60.00%	820.00	200		131.4	65.7	3055.7244	900	288.0325819	0.063360887	29.32%	1094.491973
14	520	65.00%	1093.00	220		144.54	72.27	3310.3681	975	338.0382385	0.059386773	31.28%	1185.699637
15	560	70.00%	1367.00	230		151.11	75.555	3565.0118	1050	392.0443476	0.053533484	34.70%	1276.907301
16	600	75.00%	1660.00	250		164.25	82.125	3819.6555	1125	450.0509093	0.05068871	36.65%	1368.114966
17	640	80.00%	1954.00	265		174.105	87.0525	4074.2992	1200	512.0579235	0.047223661	39.33%	1459.32263
18	680	85.00%	2307.00	280		183.96	91.98	4328.9429	1275	578.0653902	0.044199152	42.03%	1550.530295
19	720	90.00%	2442.00	300		197.1	98.55	4583.5866	1350	648.0733094	0.042240592	43.97%	1641.737959
20	760	95.00%	2578.00	315		206.955	103.4775	4838.2303	1425	648.0733094	0.044352621	41.88%	1732.945623
21	800	100.00%	2714.00	330		216.81	108.405	5092.874	1500	799.581218	0.037660339	49.32%	1824.153288

Table 4.2 : Scope 2 (HRR Analysis) Results

Crank Angle(°)	Displaced Volume , Vd, (m3)	**Vc (m3)	V/Vc	δV / δθ	Pcylinder , (kPa)	fraction of heat added ,f	δf/δθ	dP/d(t hetha)	dQ/d(t hetha)	δQ/δθ 16 cylinder	IMEP , kPa	ηv@1500rpm	ηv@1000rpm	ηv@500rpm
0	0		1	0	91.1925	0	0.11342593	0	0	0	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
30	0.000167		2.2635	9.31E-05	91.1925	0	0.0833333	0	0.036499	0.583992	19.96189	3.82567	5.73850648	11.477
60	0.000614		5.4573	0.000429	91.1925	0	0.05787037	0	0.168145	2.690316	5.439341	1.04244	1.56366407	3.1273
90	0.001183		8.9456	0.000592	91.1925	0	0.0833333	0	0.231886	3.710169	2.824284	0.54127	0.81190563	1.6238
120	0.001708		11.207	0.000287	91.1925	0	0.02083333	0	0.112352	1.797626	1.955841	0.37483	0.56225162	1.1245
150	0.002065	0.00019	12.222	-0.00012	91.1925	0	0.00925926	0	-0.04554	-0.72868	1.618013	0.31009	0.46513526	0.9302
180	0.00219		12.5	-6.7E-20	91.1925	0	0.00231481	0	-2.6E-17	-4.2E-16	1.525837	0.29242	0.43863709	0.8772
210	0.002065		12.222	0.000116	103.62286	0	0	1967.166	0.05175	0.828	1.838563	0.31009	0.46513526	0.9302
240	0.001708		11.207	-0.00029	113.01214	0.02288229	-0.00339277	-671.295	-0.13923	-2.22774	2.423816	0.37483	0.56225162	1.1245
270	0.001183		8.9456	-0.00059	141.58353	0.16904961	0.00769399	-235.27	-0.36002	-5.76033	4.384924	0.54127	0.81190563	1.6238

300	0.000614		5.4573	-0.00043	232.0853	0.46473857	0.01115128	-190.096	-0.42793	-6.84687	13.84315	1.0424427	1.56366407	3.1273
330	0.000167		2.2635	-9.3E-05	559.5551	0.7726993	0.00841854	-389.391	-0.22396	-3.58336	122.4857	3.825671	5.73850648	11.477
360	0		1	0	1266.562	0.94461899	0.00320492	0	0	0	#DIV/0!	#DIV/0!	#VALUE!	#DIV/0!
390	0.000167		2.2635	9.31E-05	405.8643	0.99326205	0.0005615	3217.967	0.162446	2.599133	88.84307	3.825671	5.73850648	11.477
420	0.000614		5.4573	0.000429	119.1101	0.99964373	4.0411E-05	439.615	0.219621	3.513929	7.104539	1.0424427	1.56366407	3.1273
450	0.001183		8.9456	0.000592	59.83328	0.99999287	1.0556E-06	289.1775	0.152145	2.434319	1.853071	0.5412704	0.81190563	1.6238
480	0.001708		11.207	0.000287	43.70957	0.99999995	8.7959E-09	743.4644	0.053851	0.861622	0.937456	0.3748344	0.56225162	1.1245
510	0.002065		12.222	-0.00012	38.73464	1	2.0483E-11	2411.64	-0.01934	-0.30951	0.687262	0.3100902	0.46513526	0.9302
540	0.00219		12.5	-2E-19	37.54328	1	1.1661E-14	1.27E+21	-3.2E-17	-5.2E-16	0.628176	0.2924247	0.43863709	0.8772
570	0.002065		12.222	0.000116	111.4575	1	0	0	0.055663	0.890603	1.977572	0.3100902	0.46513526	0.9302
600	0.001708		11.207	-0.00029	111.4575	1	0	0	-0.13732	-2.1971	2.390473	0.3748344	0.56225162	1.1245
630	0.001183		8.9456	-0.00059	111.4575	1	0	0	-0.28342	-4.53465	3.451903	0.5412704	0.81190563	1.6238
660	0.000614		5.4573	-0.00043	111.4575	1	0	0	-0.2055	-3.2881	6.64	1.04244	1.5636	3.1273
690	0.000167		2.2635	-9.3E-05	111.4575	1	0	0	-0.0446	-0.7137	24.39787	3.82567	5.7385	11.477

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Remark :

*DIV! indicates divisible error due to 0 being the denominator . The red columns indicates no data available due to the engine tripping OFF due to low fuel feed .

**V$_c$ defined as the clearance volume above the TDC .

Bore	0.132	m			
air pressure drop	10	bar	1013.25		kPa
Capacity	800	kW			
piston speed	8	m/s			
Speed	1500	rpm	24.99999999		rps
Vd (16 cyl)	0.03504	m3			
Vd (1 cyl)	0.00219	m3			
R					
mass flowrt (air)		kg/s			
mass		kg			
crank angle,r	0.32	m			
connecting rod	0.4801	m			
ratio	1.5003125				
avg temp	388	deg C			

k , expansion coeff CH4	1.299	1.39309
k, expansion coeff Air	1.4	

CHAPTER 5

DISCUSSION AND ANALYSIS

5.1 Introduction

From the conduct of this research , the engine performance and HRR analysis were modelled using a industrial scale gas engine which is widely used in the power generation industry . With the aid of modelling tool such as MATLAB , we could model the engine's performance and the analyse key parameters such as the HRR which could be widely used with invariant to its installed capacity and specification . With the limitation and parameters taken into consideration when modelling the data . For the engine performance , parameters such as MEP , Effective Power , Torque and Thermal Efficiency inherits linear proportional relationship trend with domain to the RPM and Running capacity . However for the SFC , the engine demonstrates an exponentially downwards trend which justifies with the fuel being uses up lesser when running at a higher capacity due to the fuel economy maintained which also justifies the concept that higher volumetric efficiency at lower RPM by drawing more air than fuel resulting in higher air-fuel ratio . The HRR analysis were done through modelling the heat release rate and heat addition (via Wiebe function) in function with the volume and pressure data along with its with its first degree derivative in domain of change in crank angle . As a general outcome , the HRR significantly demonstrates a significantly proportional trend relative to the Otto cycle's HRR spectrum . Besides that , several contributing factors (internal as well as external factors) were studied and analyzed to deduce

5.2 SCOPE 1 : Modelling of Gas Engine Performance

As discussed briefly in Chapter 4 , the safety feature of the engine which prevents the engine from overloading enables a very proportional relationship among the RPM and running load of the engine (refer to Figure 4.1) .

The engine operating load determines the capacity of engine to generate power . Therefore , every time the engine running load increases , the load regulator stabilizes the RPM of the engine proportional to the running load . Hence, the engine trips each time the operational load goes "off sync" with the RPM within exceeds or lesser than the allowable tolerance of 5%. In certain case, when the RPM drop to certain level as the engine's operational load remains high (or vice versa) , the fuel supplied to the engine will be compensated according to the operational load , regulating the RPM . Besides that , the load for the engine's power output is controlled by the fuel altering throttle position at a constant RPM . In most cases , the engine's power is described as the rated power or output power were obtained during the testing of the engine with reference to the method bounded by ISO 1585.

As defined in the previous chapter briefly on the definition on the SFC , whereby it is a performance parameter which indicates the fuel being consumed during combustion process in generating rotational power at the crankshaft . Fundamentally , an internal combustion engine requires the synergy of fuel and air (defined by fuel air ratio - the fraction of fuel air stoichiometric ratio with perfect combustion without any residual excess air) . Where Figure 5.1 shows the fuel economy and SFC (denoted in terms of Fuel Consumption) with respect to air fuel ratio .
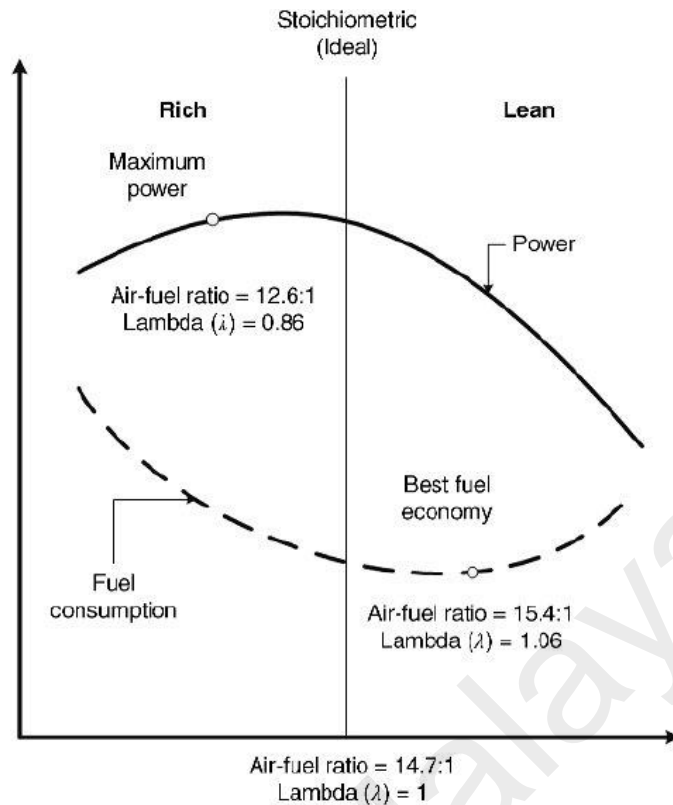
Figure 5.1 : Relationship Plot on Fuel Economy and SFC in domain of Air-Fuel

ratio. *(Image courtesy by Saber Fallah)*

Since the combustion type employed for CG132b-16 is a lean burn principle

whereby the air fuel ratio ranges somewhere around 14.7:1 to 15.5:1 . Thus , from this

trend we could observe that the best fuel economy begins to pickup as the fuel

consumption gradually increases upon transitioning from the rich to ideal

(stoichiometric) process phase . Where during the lean burn principle , the fuel fraction

are relatively lower than air intake in the air-fuel ratio which means it consumes lesser

fuel and takes in more air to combust . Which also comes to the attention that , the

exhaust gas after the combustion generating the power of the engine appears to be hotter

, which also put into argument on its potential to produce Combine Heat and Power

application to enhance the thermal and mechanical efficiency of the engine .

The third parameter of our discussion would be on the MEP . MEP is a theoretical

indicator to gauge the performance of the internal combustion engine. In other words ,

the MEP is also know for being the ratio of net work performed with domain of volume displaced . Mean effective pressure is employed especially for preliminary engine design evaluations, taking into consideration the engine torque and MEP as input parameters, from this approach the engine designer could evaluate the required engine volumetric capacity. How does the MEP affect the engine's performance? As mention in the definition of MEP where it's the theoretical ratio of net work performed by the crankshaft with respect to the displaced volume. In this case, the net work performed by the engine is what influences the performance distribution. Hence, the higher the work produced, where if the heat input derived from the combustion is assumed to be constant – therefore the engine's performance is increased. The relationship of the engine's performance is governed by Equation (4.1).

$$\eta_{TH} = \frac{W}{Q_{in}} \hspace{3cm} (4.1)$$

Where η_{TH} , is the thermal efficiency being the ratio of net work done (W) by the engine relative to the heat being absorbed (Q_{in}). Which also relates to the thermal efficiency of the engine being the crucial engine performance parameter. The thermal efficiency of internal combustion engines relies on numerous factors and the most significant one is the expansion ratio where the work performed extracted from it is proportional to the difference between the initial pressure and the pressure during the expansion phase which is defined in Equation (2.5) in the adiabatic expansion process (power stroke phase) . The higher the expansion ratio, the greater the engine's efficiency.

5.3 SCOPE 2 : Modelling of Performance Study and HRR Analysis

The HRR analysis is the systematic analysis performed to evaluate the performance of an engine specifically during the combustion of the engine which reflects the heat released and absorbed. The HRR models are expressive to γ values

which are a function of the Wiebe Function model in Equation (2.15) (Olanrewaju et.al , 2020) . The value of γ is dependent on temperature during the combustion and the air fuel ratio .

According to Figure 4.7 the cylinder pressure demonstrates a steep increase from the crank angle 240° which is due to the fact the compression begins to take place and at the crank angle 360° (where a revolution of the crankshaft have completed) is where the combustion takes place and then forcing the piston to move downwards resulting in an expansion of volume displaced. This motion, results in the power stroke delivering the power in the form of work done to the crankshaft. The drop of the pressure is due to the expansion of the volume displaced and followed by stabilizing the pressure with effect from crank angle 540° till 720° (2 revolutions of the crankshaft is completed) where the combusted gas is exhausted out of the cylinder compensating the cylinder's internal pressure (Kalghatgi and Johansson ,2018) .However , based on the MATLAB simulation which resulted in the plots in Figure 4.7 and Figure 4.8, demonstrates the comparison of the cylinder pressure and comparison of MEP and cylinder pressure indicates a higher pressure which is considered to be beyond the nominal scale of maximum cylinder pressure which is 69 bar(6.99MPa) to 103bar(10.43MPa) .However , based on the works of Yorokiev et.al 2012, which justifies that for an engine with high compression ratio – as the piston approaches the TDC , approximately 90% of the gas mass is in the piston bowl and in the region above this cavity whereby the pressure of this mass fraction is represented as cylinder averaged pressure .Thus , the remaining working fluid mass dwell in the crevice regions between piston and head as well between piston and sleeve, and its pressure may exhibit oscillations of up 10 bar amplitude, which are due to the turbulent in-cylinder flow and to acoustic phenomena resulting from combustion .Therefore, achieving 120bar of cylinder pressure is possible but however it may harm the engine's performance over time. Besides that, the MEP

fluctuates all the way compared to the cylinder's pressure which may due to the fact that the pressure acting on the piston compensates according to the motion travelled by the piston . Besides that , there was a spike at 360° which indicates the pressure acted on the piston were high due to the combustion with being gradually increasing throughout the compression stroke . The MEP builds up at the piston subjecting to the increase in pressure as the volume displaced reduces. Followed by the spike at 360° crank angle, the MEP plunges downwards similar to the cylinder's pressure due the increase of the displaced volume subjecting subdue of expansion and then settles to a fluctuating point (with smaller variance of fluctuation) (Kurtgoz et.al , 2017) .

From the MATLAB modelling of the HRR analysis, it has a fluctuating change of heat respective to the displaced crank angle, $\delta Q/\delta \theta$. Which also meant that the fluctuation corresponds to the change in displaced volume From Figure 4.9 and 4.10 where there was negative trend observed; at 210° to 360° (combustion process) and 570° to 720° (exhaust stroke phase). When the combustion process occurs the heat is removed via being released and dispersed as the volume increases. To prevent extreme heating safety feature such as intercooler cooling is employed cools the engine from overheating which explains the sudden plunge of heat release to negative. Following to the exhaust process, the heat is removed to the exhaust manifold along the end products due to the combustion. Besides in Figure 4.8, the heat addition begins at 210° and settles at 510° indicating complete combustion were taken place before being exhausted.

However , in Figure 4.13 , which the comparison of HRR obtained from the conventional heat release equation (Equation 2.16) and First Law of Thermodynamics (Equation 2.18) demonstrates a similar trend which is a steady fluctuation . As mentioned in Chapter 4 on the sudden spike on heat released in obtained at crank

angle 510º to 540º which results in a peak of 4.30W/deg and 4.57W/deg respectively when obtained HRR using the First Law of Thermodynamics equation. This is justified through the heat being released during the exhaust stroke by purging the heat and the exhaust gas out from the cylinder rather than dissipating most of the heat through cylinder walls (Nguyen and Duv,2018). During the exhaust stroke, the piston travels upwards, squeezing out the exhaust gasses that were by-product of the combustion stroke whereby the heat energy which is translated from the chemical reaction of the air-fuel mixture during the combustion is then synergized with the exhaust gas in the form of exhaust enthalpy (Caton,2018) .Unlike solving the HRR using the conventional heat release equation whereby factoring in the heat being generated and released steadily factoring in the losses of heat released to the ambience through the cylinder wall (Ma et.al, 2015) .

Figure 4.14 demonstrates the trend of volumetric efficiency with respect to crank angle in variant to different RPM of the engine. To begin with discussing on the volumetric efficiency, we shall know what does it means by the term volumetric efficiency and how it relates to the efficiency of the engine. Volumetric efficiency is also known as the " breathability " of an engine – defined as the proportional ratio of mass of air-fuel mixture drawn into the engine's cylinder(s) relative to the atmospheric pressure subject to the volume of air drawn via the intake manifold . Thus, from Figure 4.14, it can be evaluated that higher volumetric is achieved at lower RPM – which differs by the hypothesis proposed by Consuguera et,al ,2019 – where the volumetric efficiency lowers as the RPM reduces . This may however being influenced by the lean burn principle applied by the engine combustion system which uses more air intake rather than fuel to enable a more economic fuel consumption governed by the SFC . In that context, more air is being drawn rather than fuel at lower RPM compared to fuel subject to higher ratio of air over fuel .

5.4 Error Analysis

Due to limited data availability especially for the HRR analysis and benchmarking similar engine configuration , therefore the error analysis were limited to the engine's performance with comparison from the data of the engine specification tabled the manufacturer . Table 5.1 is the error analysis done for the engine performance relative to the parameters such as :

Table 5.1: Error Analysis of Engine Performance Parameters.

PARAMETERS	ACTUAL	EXPERIMENTAL	% ERROR
Power (kW)	800	799.581218	0.0523%
Thermal Efficincy (%)	42.30%	49.32%	-16.60%
Mean Effective Pressure (kPa)	1915.043	1824.153288	4.75%
RPM	1500	1500.944	-0.06%
AVERAGE ERROR %			-2.9669%

Based on the observation on the error analysis in Table 5.1 ,the error percentage of the effective Power , MEP and RPM ; 0.0523%, 4.75% and -0.06% respectively Which appears to be within the acceptable range of +/- 5% . But the Thermal Efficiency demonstrates a higher percentage of error, indicating a -16.60% error. There are several factors contributing to the increased error percentage and amongst that would be the type of fuel used to during the testing done by the manufacturer and the type of fuel used at site. Due to the fact, that the fuel used at site which is the biogas has a higher caloric value (21500 kJ/kg). Thus, the caloric value influence the heat absorption from the combustion of the fuel corresponds to the thermal efficiency of the engine . Whereby the fuel used in the test rigs are Natural Gas which has slightly lower caloric value (52300 kJ/kg) (Arjuna et.al, 2018). Besides that, the methods of evaluation for the thermal efficiency may have been different such as the manufacturer uses high end

analytical tools to model the thermal efficiency at real time basis. However, the methods use for this research is based on calculation derived from the data at site. This explains why are the difference of the thermal efficiency beyond acceptable range. However, in an overall scale the average error obtained were -2.97%, which complies to the acceptable error percentage range.

5.5 Theory of Engine Simulated on MATLAB

As part of the MATLAB modelling, the Pv diagram of the engine were modelled to validate the gas engine process with an ideal Otto cycle. Figure 5.2 are the Pv diagram of an Otto cycle comparing both ideal and practical applications.
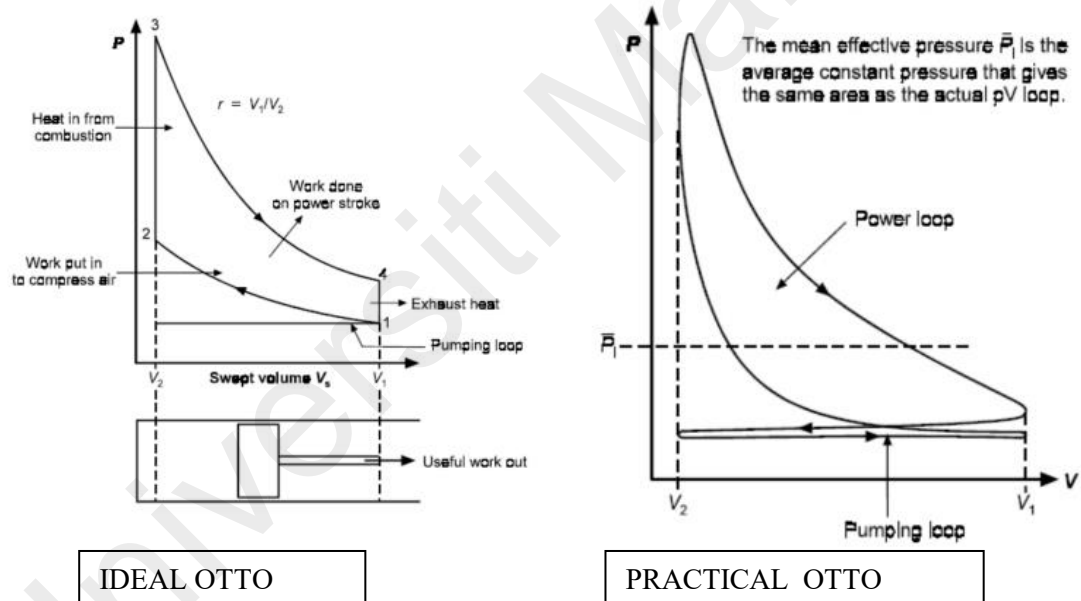

Figure 5.2 : Pv Diagram of Ideal Otto Cycle and Practical Otto Cycle

As explained in Chapter 2, on the Otto cycle processes , the engine undergoes a series of interdependent processes as in following sequence :

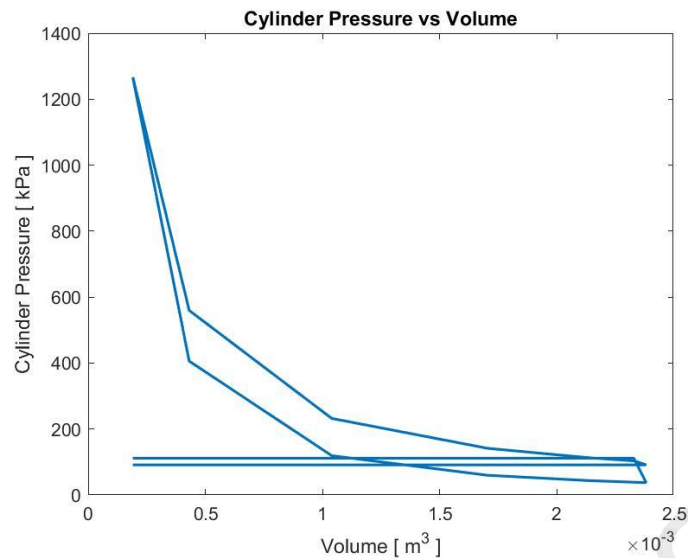**Intake → Compression→Power→Exhaust**

Figure 5.3 : Pv diagram modelled in MATLAB

Thus, Figure 5.3 is the modelled Pv diagram from the MATLAB program coded. Hence, there were similar resembles can be observed in comparison of Pv diagram of the practical Otto cycle and the Pv models from the MATLAB modelling.

5.6 External Contribution factor of the Gas Engine Performance

Through my experience and research on the gas engine's performance there are several external factors which plays its part in contributing to the performance of the engine. Assuming the gas engine as a closed loop system where the engine is the controlled volume. Any factors which act out of the engine boundary significant influences the performance which that factor is known as external contributing factor. The list of contributing factor (s) are;

(a) moisture content of the biogas

(b) frequency of start-stop of the engine

(c) biogas availability

Fundamentally, biogas engine which run in biogas are sensitive due to its sophisticated instrumentation as part of safety measures and conditional monitoring. In that case , the moisture content (or vapor content) of the biogas were undesired . The

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moisture content in the biogas engine does not only deteriorate the engine's performance but also causes harm to the engine parts such as the piston and valves. In most cases the moisture content from the biogas causes the valves to be eroded. In some cases, severe damage of the piston due to moisture erosion. Figure 5.4 shows an example of the piston being damaged due to the corrosion caused by the moisture escaped to the engine cylinders.

Figure 5.4 : Example of a damaged piston due to corrosion caused by escaped moisture content .

This scenarios alarms on why the biogas have been dried before channeling into the gas engine(s). The second factor would be the frequency of start-stop of the engine itself. The frequency of stop-start has only been possible factor as it might result in premature wear and tear as it contributes to the straining of the component compared to be wen its on idle or running . In some case, inducing sudden thermal fatigue may also subject towards in reducing engine performance over time and a measure in overcoming the effects of the thermal fatigue and straining due to the number of starts stop – the engine undergoes a preventive maintenance inspection every 1500 hours as a conditional monitoring measure. The availability of biogas also plays a significant role in contributing to the gas engine's performance. Every gas engine are designed with a specified gas feed and when the gas which is channeled to the engine is lesser than the specified flow rate or volume, the engine's engine begins to drop drastically proportional to the reduction of the gas fed Thus, making it more uneconomical in terms

of fuel consumption due to the scarcity of fuel. As a domino effect, the efficiency drops.

5.7 Technical Recommendations

As part of the research, several recommendations were deduced in order to optimize the gas engines in the near future. Among the recommendations are;

i. Introducing Gas Engines with Miller Cycle to modify the Otto Cycle Engines

- Miller cycle are upgraded or improved Otto cycle where the cylinder pressure in low pressure were supercharged when the air is being drawn into the cylinder increasing the power boost at higher loads and increase fuel economy with high compression ratio . Thus, increased compression ratio, results in the fuel mixture is sufficiently compressed proportionally increasing the thermal efficiency. Therefore, less fuel is required to generate the equivalent amount of energy (Chen et.al, 2011).

ii. Upgrade the Ignition System

- There are 2 ways to upgrade the ignition system of an engine and amongst it which are upgrading the spark plug. Example of upgrades includes rapid firing spark plug which state of the art technology with excellent quality suppressors (measures on prevention of misfiring) or even using enhanced thermal performance spark plugs fouling and premature ignition. Besides that, application of condition monitoring sensors for ignition timing. With this application the firing sequences can be gauged and monitor on a regular basis.

iii. Harness the Potential of CHP

- Since the high temperature exhaust gas which draws out of the cylinder upon is channeled to ambience, its exhaust gas has the potential to be reused into CHP application. The exhaust gas emits temperature at an average about $481^{o}C$.

- The waste heat could be reused for Stage 2 drying the biogas channeled into the engine. Through reheating the excess moisture /vapor can be eliminated and resultant to it increase the thermal efficiency of the engine and its performance.

CHAPTER 6

CONCLUSION AND RECOMMENDATIONS

6.1 Conclusion

From the conduct of this research, the engine performance and HRR analysis were modelled using a industrial scale gas engine which is widely used in the power generation industry. With the aid of modelling tool such as MATLAB, we could model the engine's performance and the analyze key parameters such as the HRR which could be widely used with invariant to its installed capacity and specification. With the limitation and parameters taken into consideration when modelling the data. For the engine performance, parameters such as MEP, Effective Power, Torque and Thermal Efficiency inherits linear proportional relationship trend with domain to the RPM and Running capacity. However for the SFC , the engine demonstrates an exponentially downwards trend which justifies with the fuel being uses up lesser when running at a higher capacity due to the fuel economy maintained which also justifies the concept that higher volumetric efficiency at lower RPM by drawing more air than fuel resulting in higher air-fuel ratio . The HRR analysis were done through modelling the heat release rate and heat addition (via Wiebe function) in function with the volume and pressure data along with its with its first degree derivative in domain of change in crank angle . As a general outcome, the HRR significantly demonstrates a significantly proportional trend relative to the Otto cycle's HRR spectrum. Besides that, several contributing factors (internal as well as external factors) were studied and analyzed to deduce recommendations to optimize the engine's

performance. The performance modelling of the of the engine will be highly significant especially in designing the gas engines in the future.

5.2 Recommendations

The ratio of Maximum pressure and MEP shall always maintain higher in order to increase the firing pressure of the spark ignition upon compression which significantly increases the thermal efficiency to optimize the engine's performance.

Besides that , the SFC shall be maintained at optimum level as the engine maintaining the firing pressure when the load is operated under base load of (~65% of its nominal capacity) as well significantly maintain optimum thermal efficiency .

As mention in the previous chapter that this research had a limitation in terms of obtaining the data (real time cylinder pressure, indicated torque data, heat in/out) due to the unavailability of the crank encoders as well as torque and heat sensors, whereby the value were to be calculated manually.

Thus, an effective measurement system (and applies for conditional monitoring purposes) shall be employed which would be an excellent technique for data logging when engaging periodic performance study in planning schedule and corrective maintenance

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