Heat Transfer of Combustion Chamber Wall of a Diesel Engine

By

Muhammad Afieq Zakwan bin Azhar

13640

Dissertation submitted in partial fulfillment of

the requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

MAY 2014

Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

Heat Transfer of Combustion Chamber Wall of a Diesel Engine

By

Muhammad Afieq Zakwan bin Azhar

13640

A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
In partial fulfillment of the requirement for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

Approved by,

____

(Dr. Meor Azman Mior Said)

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
MAY 2014

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in references and acknowledgments, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

____

(MUHAMMAD AFIEQ ZAKWAN BIN AZHAR)

ABSTRACT

An experimental investigation has been undertaken with its main goal is to validate an innovative experimental facility and to establish a methodology to evaluate whether there is difference between the heat flux of thermal distribution across the intake and exhaust side of internal combustion chamber of an air-cooled diesel engine. Spatial and time-averaged temperature measurements have been performed in order to estimate the heat flux of heat conductivity on the engine. Temperatures data were measured under a wide range of speed and applied load at eight locations in the engine cylinder block employing conventional Type-K thermocouple arranged to obtain one-dimensional metal thermal gradients and subsequently deduce the corresponding conductive heat flux from the wall surface temperature across the engine block. Results which were recorded throughout this work had revealed a realistic trend in wall surface temperature as function of engine load for a given engine speed. It also consistent with the theory that in-cylinder gas temperature will decrease across the engine stroke as combusted gas expanded. Result analysis lead the proposal of an initial concept that proposed heat loss in the mode of conductive heat transfer are more concentrated on exhaust side of engine combustion chamber compared to the intake side.

ACKNOWLEDGMENT

First and foremost, I would like to express my deepest gratitude towards Allah S.W.T., for Him who had guided me throughout my studies in Universiti Teknologi PETRONAS, I would be able to completer my Final Year Project successfully and beneficially. I would also like to extend my appreciation towards individuals that have given their full support and assistance throughout the beginning till the completion of this work. Within the Mechanical Engineering Department of UTP, special gratitude is Dr. Meor Azman Mior Said, my supervisor, upon his support and encouragement in assisting my learning process throughout conducting this work. Not to be forgotten the supportive technicians of Block 15, Mr Mahfuz, Mr Fahmi, and Mr Khairul, because of their guidance and assistance on setting up the engine test bed, I am able to complete the project within the two semester of my final year study. On the other hand, I would also like to thank the coordinators and examiners for Final Year Project which indirectly had involved by giving other recommendations on improving this project.

Finally, appreciation and many thanks expressed towards my beloved family and friends for their endless support, encouragement and understanding given in making this project a success.

Thank you.

TABLE OF CONTENTS

LIST OF FIGURE

LIST OF TABLE

CHAPTER 1

INTRODUCTION

1.1 General Background

From the beginning of internal combustion engine development, the importance of understanding the heat transfer through the combustion chamber walls has been recognized. Although this phenomenon is difficult to be analyzed because of its complexity, in that involves various elements such as transient three dimensional behavior, rapid temperature swings, piston motion, cooling passages, among others [1], cannot be neglected. Most of the studies performed in this area have been motivated by the influence of heat transfer on engine performance. Torregrosa et. al. (2011) stated that higher heat transfer rates to the combustion chamber walls will lead to a reduction of the mass average in cylinder temperature and pressure, therefore the work per cycle transferred to the piston will be lower. On the other hand, societies nowadays are heavily concern regarding the preservation of the atmospheric environment. This had emphasized the importance of heat transfer knowledge in this suit as changes in gas temperature due to in-cylinder heat losses to the walls strongly affect emissions formulation and after burning processes [2].

Diesel engines have been illustrating remarkable progresses by the fact that sales of diesel vehicles throughout the globe are increasing and even recently have overtaken gasoline car sales in Europe. Freedonia Group (2012) illustrated that the world demand for diesel engines is projected to grow 6.7% per year through 2015 which is valued around $197.5 billion. This scenario will be driven by an increase in the production of motor vehicles, particularly medium and heavy trucks and buses. This progress in diesel engine has been a positive result due to some extent by the introduction of new technologies. Examples of such technologies include high pressure common rail and variable fuel injection strategies including retarded injection for nitrogen oxides (NOx) emission control; exhaust gas re-circulation (EGR); high levels of intake boost pressure provided by turbocharger or supercharger and inter-cooling; multiple valve per cylinder; and advanced engine management systems (Parra, 2008).

1.2 Problem Statement

In the design and development of reciprocating internal combustion engines, it is frequently desired to calculate the heat transfer rate from the working fluid to the surfaces which contain it. Based on various methodology that have been developed over past decades, these methods for predicting heat fluxes and operating temperatures of compression-ignition (CI) engines to estimate heat transfer coefficient are often updated in view of more recent experimental data. Due to the fact that the approach of investigating the difference of thermal distribution between the intake and exhaust side of internal combustion chamber were not taken into account when those methods were introduced, it leads to scenes where the application of these methods to the modern CI engines is questionable.

1.3 Objective and Scope of Study

An experimental and analytical will be conducted with the aim of providing improvements towards the prediction methods to estimate whether there is difference between the heat fluxes of thermal distribution across the intake and exhaust side of engine internal combustion chamber. Another objective of this project is also to develop an engine test bed that fully equipped to measure temperature gradient across engine cylinder block. In conjunction with achieving the objective, this research is set to be conducted based on following scopes of studies:

- Engine testing will be conducted on a low speed, single cylinder, compression-ignition (CI) engine.

- Selecting suitable thermocouple probe and fittings for conductive heat transfer measurement.
- Investigating the change of temperature profile of different engine operating condition.
- Performing data analysis on the conductive heat transfer along engine cylinder block of the test engine.

. Together with the application thermodynamics principles and the fundamental equations of heat transfer, this research is perhaps will produce realistic and concrete theories in improvising the existing correlations on estimating thermal operating condition of modern CI engines.

1.4 Relevancy of Project

From this project, the determination of difference on conductivity thermal distribution across engine cylinder can lead to significance as a scientific reference to the engine development industry in improving the previous in-cylinder heat transfer correlation that have introduced by previous works.

1.5 Feasibility of Project

This project is analyzed to be feasible where majority automotive laboratory equipment which is required for this project is all provided in the university. The implementations of the experimental work follow theories, which become the fundamental to pursue the project. The allocated of financial cost is sufficient for the purchasing of some other equipment which is not available in the laboratory. Together with basic experience on handling engine testing equipment and data analysis could ensure the feasibility of this project within the given period.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

All internal combustion engines is in which the heat transfer to the working fluid occurs within the engine itself, usually by the combustion of fuel with the oxygen of air. Internal combustion engines basically can be categorized as spark ignition (SI) engines which take a mixture of fuel and air, compress it, and ignite it using spark plug, whereas compression ignition (CI) engines rely on compression of a fluid. In these engines there is a sequence of processes:

- Compression
- Combustion
- Expansion
- Exhaust / induction

2.2 Mechanism of CI (Diesel) Engine

CI engines differ from SI engines in a various ways but the most obvious one being the way which the air and fuel mixture is ignited. Salazar [3] stated that the presence of high temperature and pressures in the combustion chamber cause a flame to initiate at different sites of the combustion chamber as in CI engine there is no spark to create the flame. By implementing fuel injection systems, diesel engines can be divided into direct and indirect ignition engine which explains how the fuel is introduced into the combustion chamber.

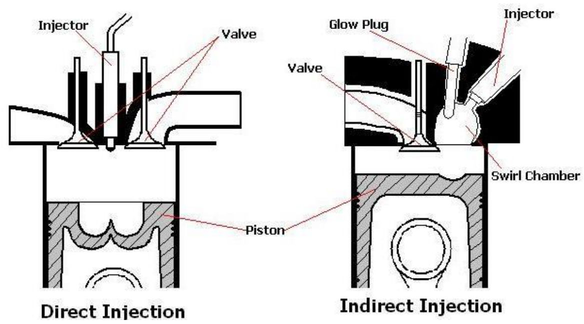


Figure 2.1: Visual Comparison between Direct and Indirect Injection System [27]

Since heat addition process takes place at a compressed state in direct injection diesel engine, Salazar [3] emphasized that high pressure is needed in order for the fuel to be well injected the pressure has to be greater than the one that has been accumulated through compression. This is illustrated as direct injection diesel engine that have been developed since past decade use pressures of up to 1000 bar to inject fuel into the combustion chamber. The other vital element in a diesel engine is the swirl air motion which manages the mixing of air and fuel so that combustion can increase [3].

On the other hand, indirect injection diesel engine system is built up with a pre-combustion chamber where the air to fuel mixture is first stored. Salazar [3] explained that the separate chamber is designed to speed up the combustion process in order to increase the engine output by increasing the engine speed. Turbulence plays in important role in the pre-combustion chamber to increase the combustion speed, while the swirl chamber depends on the fluid motion to raise combustion speed. Due to the mechanism of these divided chambers, the pressure required for indirect injection system is not as high as the pressure required for direct injection system. It is illustrated that the pressure required for both chambers is only up to 300 bar.

2.3 Air-Standard Diesel Cycle

Compared to SI engine which its operation implementing the Otto cycle, CI engines operation are based on diesel cycle with have some similarities with the previous cycle, except that heat addition and rejection occur at different conditions [3].

The diesel cycle is also an ideal cycle meaning that it does not illustrate the exact representation of the actual process.

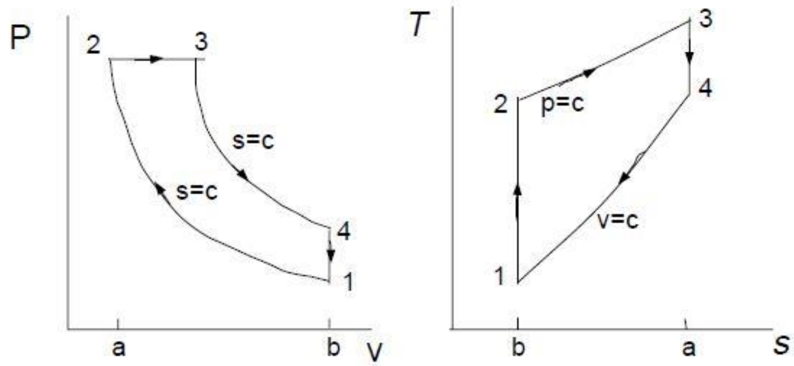


Figure 2.2: P-v and T-s Diagram of Air Standard Diesel Cycle

Figure 2.2 shows the $P - v$ and $T - s$ diagram for a diesel cycle. The cycle consists of four internally reversible processes. Process 1-2 is an isentropic compression. Process 2-3 is a constant pressure heat addition process which illustrates the first part of power stroke (combustion). Process 3-4 is an isentropic expansion, which makes up the rest of the power stroke. Later, process 4-1 finishes the cycle with a constant volume heat rejection with piston a bottom-dead-center (BDC). Compared to SI engine, the compression ratio a CI engine plays a greater significance in determining the performance of the engine which later displayed as the thermal efficiency of a CI engine increases as the compression ratio increases [3].

2.4 Heat Transfer in Engines

Pulkrabek [4] suggested that about 35% of the total chemical energy that enters an engine in the fuel is converted to useful crankshaft work, and about 30% of the fuel energy is carried away from the engine in the exhaust flow in the form of enthalpy and chemical energy. This leaves the remaining energy that must be dissipated to the surroundings by some mode of heat transfer. This relation can be expressed into the form of energy balance expression of:

$$\text{Power generated} = \dot{W}_{shaft} + \dot{Q}_{exhaust} + \dot{Q}_{loss} + \dot{W}_{aux}$$

where

$\dot{W}_{shaft}$ = brake output power of the crankshaft

$\dot{Q}_{exhaust}$ = energy lost in exhaust flow

$\dot{Q}_{loss}$ = energy lost to the surrounding by heat transfer

$\dot{Q}_{aux}$ = power to engine auxiliary equipment (alternator, water pump , etc.)

The amount of energy (power) available for use in an engine is:

$$\dot{W}_{generated} = \dot{m}_f Q_{HV} \eta_c$$

where

$\dot{m}_f$ = fuel flow rate into the engine

Q_{HV} = heating value of fuel

η_c = combustion efficieny

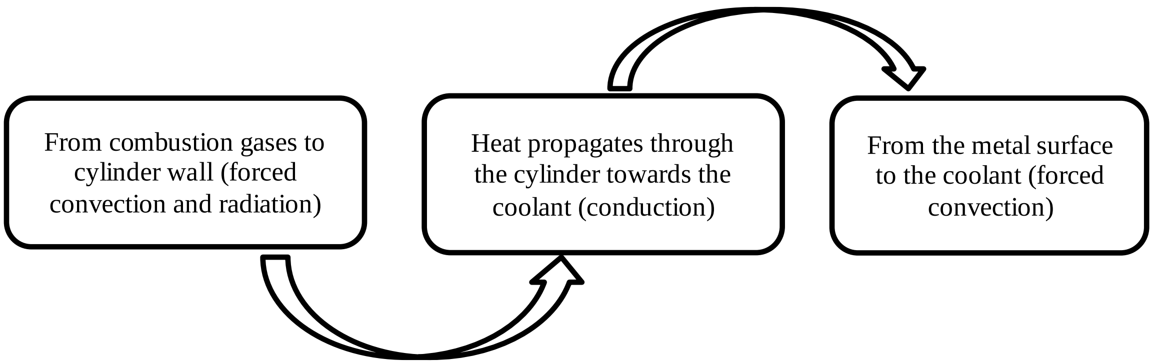
Depending on the size and geometry of an engine, as well as on how it is being operated, the shaft power output is ranging around 20% - 40% of power generated. CI engines are generally on the higher end of this range [4]. Energy lost in exhaust flow is denoted ranging around 20% - 45%, whereas the remaining heat loss sums up between 10% - 35% which is transferred to the surrounding. On the other hand, the heat losses can be subdivided into:

$$\dot{Q}_{loss} = \dot{Q}_{coolant} + \dot{Q}_{oil} + \dot{Q}_{ambient}$$

2.5 Modes of Heat Transfer

Parra [5] suggested that between one quarter to one third of the total energy of combustion is representing the heat transfer from combustion gases to the coolant in reciprocating internal combustion engine. In which, the heat entire heat rejection is depending mainly on engine type and operating conditions. The process plays around with the three primary modes of heat transfer for smooth steady-state operations which are conduction, convection, and radiation. Approximately half of the heat generated is transferred through the cylinder walls and most near the exhaust valve seats [5].

Heywood [6] also stated that there is also some heat transferred indirectly from the combustion product to the coolant by the lubricating oil, and also heat dissipated by friction between the bore and piston rings and skirts. From the gases to the metal, heat is transferred mainly by forced convection with a contribution by radiation. Stone [7] stated that due to the formation of highly radiative soot during combustion, thus the contribution of radiation heat transfer is more significant in CI engines than in SI engines. In addition, a diesel engine heat radiation might account for more than one fifth of the in-cylinder heat transfer.



Over the last decades, several empirical correlations have been developed to estimate heat fluxes from the combustion chambers of internal combustion engine. There are many other proposed formulas that will be briefly discussed in this chapter.

2.6 Instrumentation

2.6.1 Thermocouple

Discovered back in 1821 by Thomas Johann Seebeck, he experimentally determined that a voltage exists between the two ends of a conductor when the conductor’s ends are at different temperature [8]. His discovery soon became the basis of “the thermocouple”, which today is one of the most popular and cost effective temperature sensors.

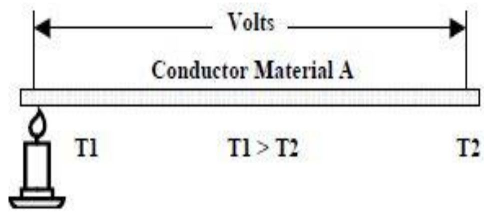


Figure 2.3: Original Discovery of Thermocouple

Pyromation Inc. [9] defined that a thermocouple is a type of temperature sensor, which is made by joining two dissimilar metals at one end. The joined end is referred to as the hot junction, while the other end is referred to as the cold end or cold junction. The cold junction is actually formed at the last point of thermocouple material. Certain combinations of metals must be used to make up the thermocouple pairs. The voltage created by a thermocouple is extremely small and is measured in terms of millivolts (mV). To establish a means to measure temperatures with thermocouples, a standard scale of millivolt outputs was established by the mean of implementation of amplifier with cold junction compensation [9].

There several type of thermocouple that are available which are named as [10]:

- Type E (Nickel-Chromium)
- Type J (Iron/Constantan)
- Type K (Nickel-Chromium/Nickel-Aluminum Silicon)
- Type R (Platinum-Rhodium/Platinum)
- Type T (Copper-Constantan)

Figure above showed the example of heat probe design of the Berdensky [11] type surface thermocouple which was used by Overbye [12], Bennethum [13], Ebersole [14], and Lefeuvre [15] in their work to measure the instantaneous wall temperature. The thermocouple used was in the form of a 2-56 screw (in term of standard thread size). The author emphasized that since the major portion of the thermocouple probe was iron, the disturbance to the heat transfer pattern was kept to minimum. When flushed mounted, the thermocouple junction temperature was considered to be the true surface temperature.

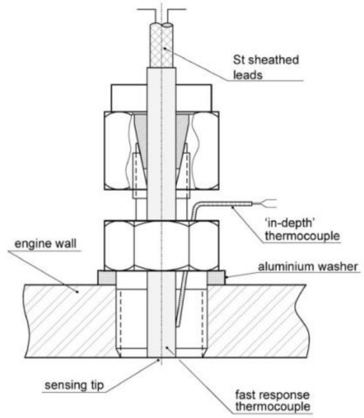


Figure 2.5: Example of Thermocouple by Rakopoulus et al [16]

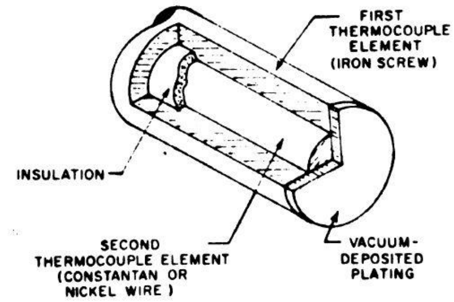


Figure 2.4: Example of Surface Thermocouple by Berdensky [11]

Rakopoulus et al [16] have been using two types of thermocouples on his work which were type K and type J thermocouple which were custom designed and manufactured for the needs of the experimental installation. This heat flux probes were installed in the engine cylinder head and the exhaust manifold, for measuring the heat flux losses at the respective positions. The authors stated that both heat flux sensors types were designed throughout extensive preliminary testing procedures performed during a period of several years before reaching the final mature state for the present application.

2.7 Heat Transfer Correlations used in Internal Combustion Engines

A number of empirical correlations have been developed to estimate heat fluxes from the combustion chambers of internal combustion engines. Numerous models of varying complexity in representing the heat flux variation have been proposed to quantify heat fluxes arising from combustion. Accordingly, there are correlations to predict time-averaged heat flux; correlations to predict instantaneous spatially-averaged heat flux; and correlations to predict the instantaneous local heat flux.

Time-averaged heat transfer coefficient is useful to estimate component temperatures and the overall heat given up by the gases, which is used in heat balance calculations. Instantaneous spatially-averaged heat transfers which consist of instantaneous coefficients are used in predicting power output, efficiency and exhaust emissions. For which the temporal variation of heat flux is needed, while the spatial distribution is less important. Time dependent heat losses are necessary for the calculations of net heat release, which is used for emission analysis. Likewise, the prediction of surface temperature within the chamber is also affected by the heat flux as well as coolant heat transfer coefficient. On the other hand, the requirement of determining local heat flux had introduced the correlations to predict these heat fluxes for thermal analysis of engine components as well as engine modeling and they need to be accurately predicted. The thermal evaluation, in particular, demands a detailed distribution of instantaneous as well as time-averaged local heat fluxes, in order to prevent thermal failure at critical locations. An accurate prediction of the temperature distribution in the engine block and cylinder head is required to improved engine designs in terms of bore distortion; this means less friction losses and blow-by, with consequently reduced fuel consumption and improved durability.

2.7.1 Example of Correlation for the Time-Averaged Coefficient (Taylor and Toong)

They carried out tests on nineteen commercial engines of various sizes and configurations within three different types; water-cooled CI engines, water-cooled SI engines, and air-cooled SI engines. Taylor and Toong [17] proposed a relationship between Nu and Re based on heat transfer measurement taken from those engines, which later commented by Heywood [6] and by Annand [18]. An expression of $Nu = aRe^m Pr^n$ was reduced to $Nu = aRe^m$ where Prandtl number, Pr is negligible since it varies little in gases. This expression where its effect can be included in the coefficient a; Nu and Re are defined as:

$$Nu = \frac{\dot{Q}B}{k_g A_p \left(T_g - T_w\right)}$$

$$Re = \frac{\dot{m}_g B}{\mu_g A_p}$$

A correlation for a time-averaged overall heat flux was derived which can be used to predict gas-side engine heat transfer providing the air-fuel ratio and mass flow are known or can be estimated. It also requires values of T_g as a function of the air-fuel ratio, which are also difficult to be estimated.

$$\frac{\dot{Q}}{A} = 10.4\frac{k_g}{B}\left(T_g - T_w\right)(Re)^{0.75}$$

2.7.2 Example of Correlation for the Instantaneous Spatially-Averaged Coefficient (The Eichelberg and Pflaum formulae)

This is the first attempt on applying direct measurement of instantaneous heat transfer rate of an operating engine [19]. These measurements of conducting his work were obtained on a large, low-speed, two-stroke diesel engine, which later proposed a formula that was intended to fuse both convective and radiant transfer;

$$\frac{\dot{Q}}{A} = 8.06 \times 10^{-5}\left(V_p\right)^{1/3}\left(pT_g\right)^{1/2}\left(T_g - T_w\right)$$

2.7.3 Example of Correlation for the Instantaneous Local Coefficient (The work of Lefeuvre et al.)

The prediction of heat fluxes at specific locations in the combustion chamber some knowledge of the local conditions, either from theoretical considerations or by experience. Lefeuvre et al. [15] proposed an equation to predict the instantaneous wall heat flux on the cylinder head or piston at different radial distance from the bore axis.

$$\frac{\dot{Q}}{A} = a\frac{k_g}{r}Re^{0.8}Pr^{0.33}\left(T_g - T_w\right)$$

2.8 Computation of Temperature Gradients, Wall Surface Temperatures, Heat Fluxes and Heat Transfer Coefficients

The instrumentation of the engine with fixed thermocouples at key locations along the engine cylinder block will be explained on following chapter. The thermocouples of every set at all locations were arranged to allow the calculation of steady-state temperature gradients in one direction. In its simplest form the temperature gradient in one direction can be calculated by using the following equation:

$$\frac{\Delta T}{\Delta x} = \frac{T_1 - T_2}{x_1 - x_2}$$

This is the equation to obtain the one-dimensional temperature gradient through a flat wall, which was applied to estimate the gradient in the direction of the cylinder axis in the cylinder block, where two arrays of thermocouples for each set will be placed at along the cylinder block. The temperature gradient can then be used to find the surface temperature of the cylinder block on the gas-side and coolant-side by linear extrapolation, in accordance with the equation:

$$T_x = \frac{T_1 - T_2}{x_1 - x_2}(x - x_1) + T_1$$

The one-dimensional temperature distribution can then be used to calculate the corresponding heat flux through the cylinder wall, with the heat flux normal to a particular surface of radius r, $\dot{q}_r$. The equation is given by Fourier's law of heat conduction:

$$\dot{q}_r = -k\frac{m}{r}$$

In compliance with the first law of thermodynamics, the conductive heat flux at the gas surface must be equal to the heat flux rising arising from the combustion chamber. Similarly, the conductive heat flux at the coolant-side of the wall must be equal to the convective heat flux between the wall surface and the coolant. According to Newton's law of cooling, the equations for the convective heat fluxes are:

$$\dot{q}_g = h_g\left(T_g - T_{wg}\right)$$

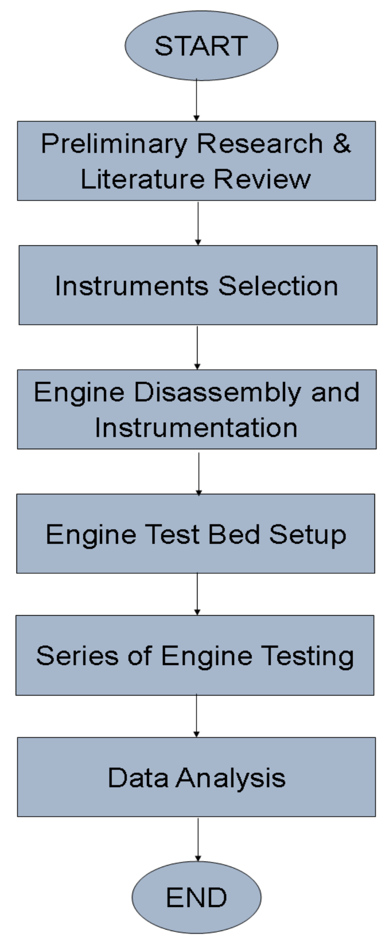
$$\dot{q}_c = h_c(T_{wc} - T_c)$$

Subsequently, it is possible to estimate the local time-averaged convective heat transfer coefficient on the gas-side, h_g, and on the coolant-side, h_c of the combustion chamber walls, providing the estimates of mean gas temperature, T_g, and mean coolant temperature, T_c can be determined. The heat transfer on the gas-side is particularly complicated compared to the coolant side, since T_g and consequently h_g change significantly during the engine cycle. It is worth to mention that the one-dimensional approach used to estimate steady-state temperatures and heat fluxes in this section is particularly suitable to the heat transfer process in the cylinder walls of the engines, where the dominant temperature gradients occur in the radial direction. Therefore, at any particular location in the cylinder walls the one-dimensional heat flux predicted by this method can be approximated to the total heat flux.

CHAPTER 3

METHODOLOGY

In conjunction with the main objective of evaluating the heat flux behavior of internal combustion chamber, a fully equipped engine test cell will be designed with the aim of reproducing thermodynamic conditions in a modern diesel engine. In addition, intensive cares need to be taken not only in the configuration and control of variables but also on the selection of suitable and reliable measurement equipment. On the other hand, necessary modifications onto original engine will be made as essential equipment such as thermocouples and pressure transducers need to be installed. This step need to done in order to carry out the measurement of combustion behavior in term of temperature and pressure at different locations along the combustion chamber which later will illustrate the conductive heat transfer model.



3.1 Equipment and Instrumentation

3.1.1 Internal Combustion Engine (ICE) Selection

The selected engine for the experimental work was as single cylinder air-cooled diesel engine with four valves (two intake valves & two exhaust valves) as diesel engine operation plays around with all three modes of heat transfer and easy to be instrumented.

The engine is readily available in the Automotive Laboratory at Block 15. Preliminary inspection and trial run was conducted in order to roughly determine the engine condition before initiating this experimental. Table 3.1 contains the main characteristics of the engine.

Robin Industrial Engine	
Model	DY23-2D
Type	Air-cooled, 4-cycle, Overhead Valve, Single vertical cylinder, Diesel engine
Bore × Stroke	70 × 60 mm (2.76 × 2.36 in.)
Displacement	230 cc (14.04 cu.in.)
Compression Ratio	12:1
Max Power Output	3.1kW @ 3600 rpm
Max Torque Output	10.5Nm @ 2200 rpm
Combustion system	Direct injection type

Table 3.1: Test Engine Specification

3.1.2 Portable Engine Test Rigs

The portable engine test rig basically consisted of a hydraulic dynamometer and an instrumentation unit. The instrumentation unit is designed to stand beside the engine under test. In addition to house the instruments necessary for measuring the engine performance, it holds the fuel system and the airbox/viscous flow meter used to measure the consumption of air. Engine speed is measured electronically by a pulse counting system. An optical head mounted on the dynamometer chassis contains a infrared transmitter and receiver. A strong disc with radial slots is situated between the optical source and sensor. The resulting pulse trains is electronically processes to provide a readout of engine speed. Unfortunately, the torque

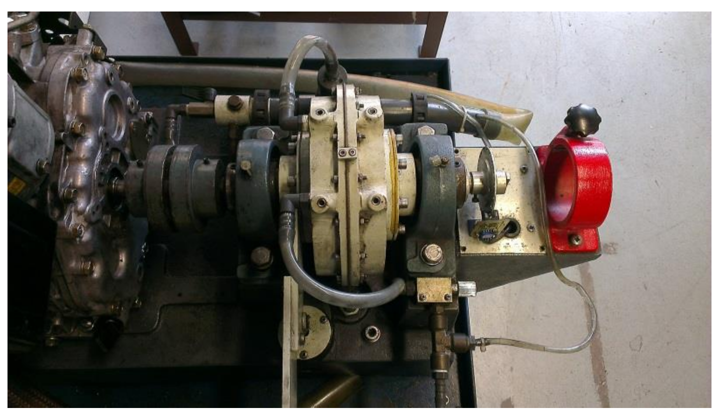


Figure 3.1: Hydraulic (water-brake) Dynamometer

measurement unit is longer operational due to dismantle work were done on the dynamometer unit by others previous works in the Automotive Laboratory.

According to original specifications, a constant supply of waterflow is required, which come from a header tank that installed on the test rig instrumentation unit. The delivery pressure will be generated from the water supply head ranging between 6 to 12 cm which will flow at 4 litres/min into the dynamometer casing in order to develop load onto engine rotation. Since this method of supplying water into the dynamoter casing in no longer applicable as a lot of original parts are missing, other approach is implemented by connecting the water intake port directly water tap in order to supply water which then generate load onto the engine. The dynamometer drain pipe is simply placed at the laboratory drainage to channel the water drain out of the system. The drawback of this method is that torque developed by the engine can no longer able to be measured by the instrumentation unit since the original methods is not longer applicable.

Figure 3.2: Test bed instrumentation unit. Consisted of speed meter, torque meter, exhaust temperature meter, air flow meter, fuel tank, and water tank

3.1.3 Thermocouples

In attempting to measure the heat conduction process along the engine combustion chamber, Type-K thermocouples were used. These probes are commercially available and can be purchased with the necessary amplification and logging system. As stated by the manufacturer, RS, this thermocouple is mineral insulated thermocouple to IEC 584 which the international standard of thermocouple calibration for all type of thermocouple reference. The manufacturer has also claimed that it possessed a good corrosion and oxidation resistance to suit wide range of processes, satisfactorily operates in sulphur bearing atmosphere.

Sensor Type	Type 'K' (Nickel Chromium/Nickel Aluminum)
Dimension	Probe with 1.0mm in diameter and 150mm in length.
Construction	310 stainless steel sheath. Flexible mineral insulated probe with plain pot seal and & 1 meter extension cable.
Element/Hot junction	Single element, junction from sheath (offer protection against spurious electrical signals)
Termination	1 meter 7/0.2mm PFA Teflon® insulated flat pair cable, colour coded in accordance with IEC 584
Probe temperature range	-40˚C to +1100˚C
Pot seal rating	200˚C

Table 3.2: Thermocouple specification

3.1.4 Data logger

A portable handheld data logger (OM(Omega)-DAQPRO-5300) is used to record the data provided by the thermocouple to illustrate the temperature variation across the engine combustion chamber. It is an eight-channel portable data acquisition and logging system with graphic display and built-in analysis functions and is capable of sampling, processing and displaying measurement without being connected to a computer. The 16-bit, high resolution data logger can also be used with a wide variety of applications as it is capable to measure voltage, current and temperature in real-time. The input ports use pluggable screw terminal blocks for easy connection. An internal clock and calendar keeps track of the data and time of every sample measured which will assist in tracking the logged data across the total period of conducting the engine testing.

3.2 Engine Disassembly and Instrumentation

Engine disassembly is conducted for the purpose of conducting a thorough examination on each component (internal and external) of the engine to conform its health before instrumentation is done onto the engine. Upon components of the engine are fully dismantle, they are clean thoroughly to remove any oil leftover or contaminants before engine dimensioning is conducted, as any presence of impurities on the engine can affect the dimension value across the engine parts.

Figure 3.3: Engine components upon disassembly

Engine dimensioning is then conducted through coordinate-measurement machine (CMM), which is based on computer numerical control (CNC) technology to automate measurement of Cartesian coordinates using a touch probe, contact scanning probe or non-contact probe. This method of dimensioning is implemented in order to produce a dimensioning result that is controlled under a tolerance of ± 0.1mm, which the accuracy is a vital factor before instrumentation of thermocouple along the engine cylinder block is done. The CMM machine had determined the dimension of engine cylinder block as measurement is took a various spot along the intake side and exhaust of the engine, which later concluded that the cylinder wall thickness is averaged about 10mm.

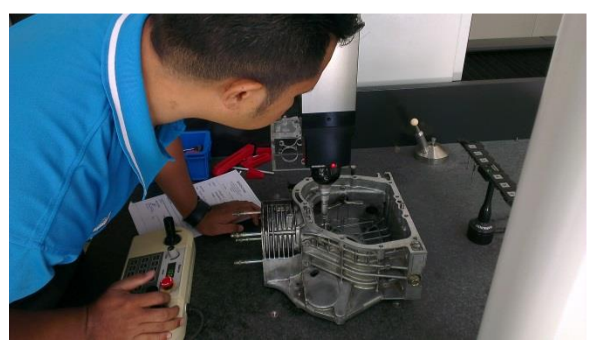
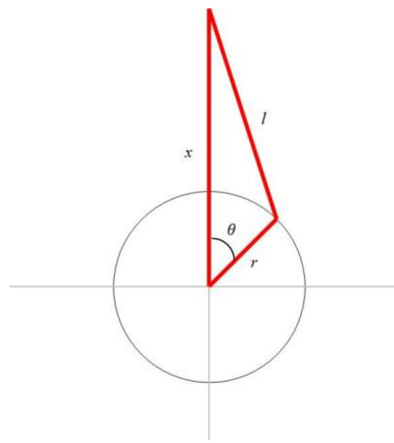


Figure 3.4: CMM dimensioning on engine block

Thermocouple placement along engine cylinder block is determined based on the piston motion equation that has been outlined with respect to various elements such as crank angle, connecting rod length, and also crank radius. The equation that follows describes the reciprocating motion of the piston with respect to crank angle.



$$x = r\cos\theta + \sqrt{l^2 - r^2\sin^2\theta} \qquad [33]$$

Engine instrumentation is then carried out by firstly determining the location of thermocouple on the engine cylinder block. As per the limitation of data logger input channel, four locations which two on each intake side and exhaust side is determined where an arrays of two thermocouples are installed onto each location. Hole drilling on each thermocouple spot is carried out carefully by experienced personnel in order maintain the thermocouple insertion’s accuracy across the engine cylinder block. A hole is drill for 8mm depth which will result a 2mm thickness from gas side cylinder wall surface, and the other hole is drill for 3mm depth which will a 7mm thickness from the gas side cylinder wall surface. Two sets of thermocouples along the intake side are located at 18mm and 54mm from the combustion chamber top deck, whereas the other two sets of thermocouples along the exhaust side are located at 22mm and also 54mm from the combustion chamber top deck. The top section thermocouples along both sides are placed with intention to provide the view of thermal value during the initial phase of heat conduction after gas combustion and also to provide the data whether there are huge significant of thermal gradient as combusted gases expanded for a short distance (4mm in this case). On the other hand, the other two sets of thermocouples that located at the same height from the top deck are intended to illustrate whether there is any difference in thermal distribution on the intake and exhaust side as gas expansion is approaching the end of its stroke. The designated thermocouples are labeled as follows.

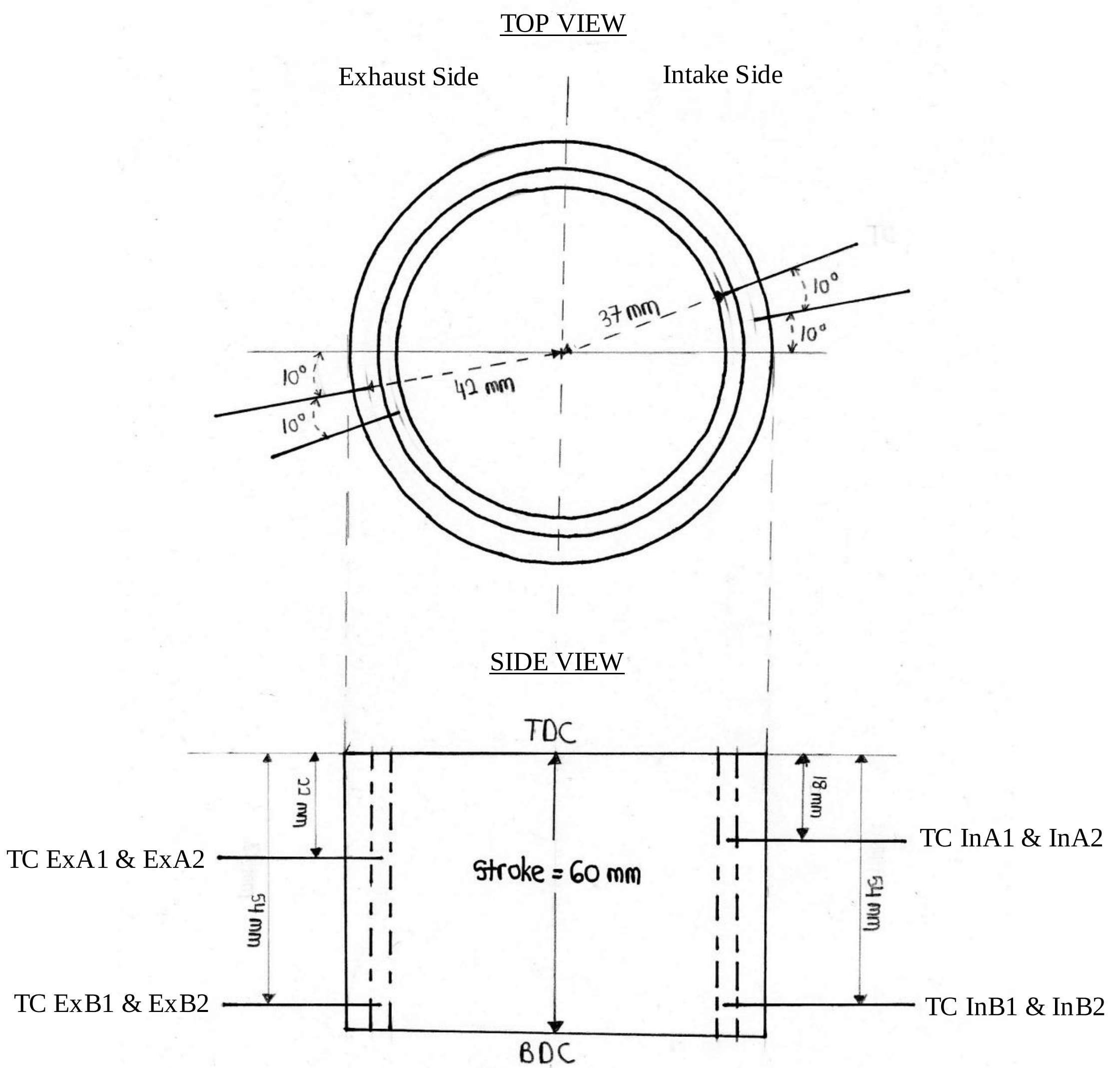


Figure 3.5: Thermocouple location across engine cylinder wall (Not to scale)

TC InA1:	Intake side surface temperature at 18mm from TDC.
TC InA2:	Intake side cylinder wall temperature at 18mm from TDC after conduction across 5mm thickness from InA1.
TC ExA1:	Exhaust side surface temperature at 22mm from TDC.
TC ExA2:	Exhaust side cylinder wall temperature at 22mm from TDC after conduction across 5mm thickness from ExA2.
TC InB1 & ExB1:	Intake side and exhaust side surface temperature at 54mm from TDC.
TC InB2:	Intake side cylinder wall temperature at 54mm from TDC after conduction across 5mm thickness from InB1.
TC ExB2:	Exhaust side cylinder wall temperature at 54mm from TDC after conduction across 5mm thickness from ExB1

Figure 3.6: View of engine upon thermocouples installation (Left image: Intake side, Right image: Exhaust side)

Once the thermocouples were in place, they are simply secured with thermosetting epoxy resin to the outer wall as there are no precaution of oil leakage arise the engine block wall does not have any oil passage. Thermosetting epoxy resin adhesive is chosen as it has ability to withstand high temperature without being soften or melted. The method proved to be successful since it strongly held the thermocouples onto their places as it cured across time.

After all procedures of engine dimensioning and instrumentation were done, the engine is mounted back onto the test bed to all test bed equipment is arranged to prepare the test bed for engine testing.

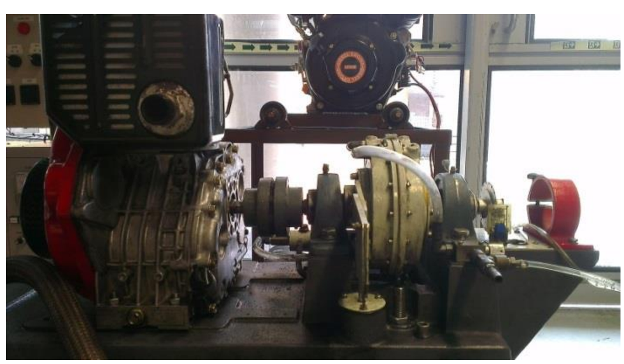


Figure 3.7: Test bed setup

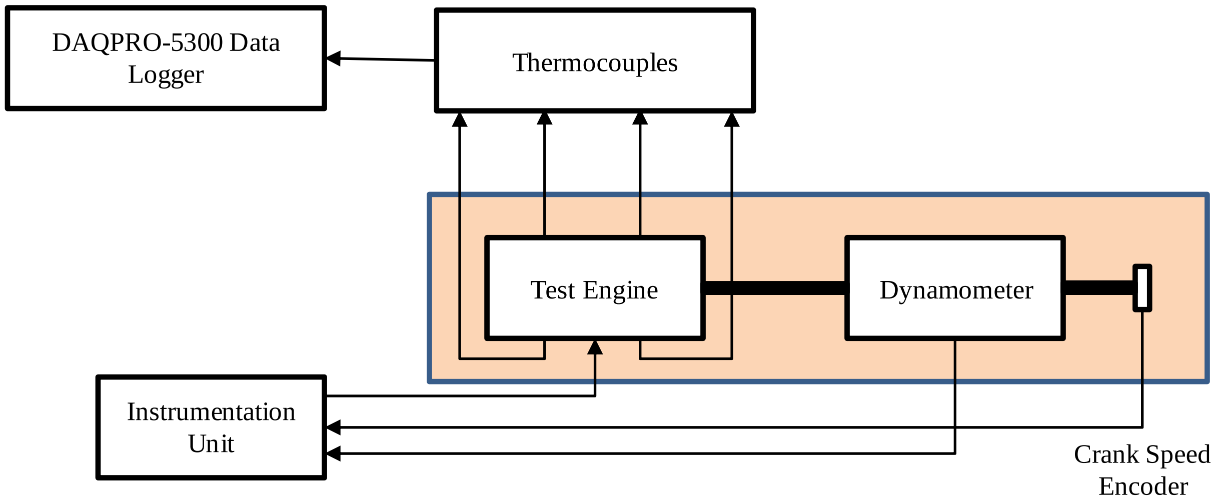


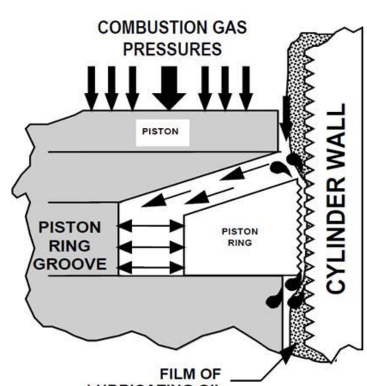
Figure 3.8: Basic schematic layout of engine test bed setup

3.3 Series of Engine Testing

3.3.1 Engine Break-In

It is procedure of conditioning a new piece of equipment by giving it an initial period of running, usually under light load, but sometimes under heavy load or normal load. It is generally a process of moving parts wearing against each other to produce the last small bit of size and shape adjustment that will settle them into a stable relationship for the rest of their working life. A new engine break-in follows specific driving guidelines during the first few hours of its use. The focus on breaking in an engine is on the physical mating of the engine's piston rings to its corresponding cylinder wall, where the new piston rings are needed to wear into the cylinder wall until a compatible seal between the two is achieved.

Proper engine break-in will produce an engine that achieves maximum power output with the least amount of oil consumption due to the fact that piston rings have seated properly to the cylinder wall. When the piston rings are broken in or seated, they do not allow combustion gases to escape the combustion chamber past the piston rings into the crankcase section



of the engine which normally referred as ‘blow-by’. This lack of blow-by keeps the engine runs cleaner and cooler by preventing hot combustion gases, which in turn will force normal oil vapors out of the engine’s breather, causing the engine to consume excessive amount of oil.

3.3.2 Engine Heat Flux Test

No Load Engine Operation		
Speed (rpm)	Duration (min)	
	Warm-up	Data Log
1500 rpm	2 min	3 min
2000 rpm	2 min	3 min
2500 rpm	2 min	3 min
3000 rpm	2 min	3 min
3500 rpm	2 min	3 min
With Load Engine Operation		
Speed (rpm)	Duration (min)	
	Warm-up	Data Log
1500 rpm	2 min	3 min
2000 rpm	2 min	3 min
2500 rpm	2 min	3 min
3000 rpm	2 min	3 min
3500 rpm	2 min	3 min

Table 3.3: Procedure of no load and high load engine heat flux test

The procedures of engine testing are derived based on the test engine, Robin Industrial Engine DY23-2D, performance curve. The engine operation is initiated from 1500rpm, which is the engine idle speed and proceeded with increment of 500rpm until it reached the maximum speed around 3500 – 3600 rpm. The warm-up duration of two minutes for each stage is introduced in order to allow the combustion process in the internal combustion chamber stabilized hence it will distribute consistent thermal concentration along the cylinder wall before temperature data is recorded for the next consecutive three minutes. As pointed before, only two engine operations which are no load and high load engine operation are able to be conducted by the engine test bed due to the faulty of torque measurement equipment. Hence, load is simply applied by direct water supply which can provide even higher load onto the shaft rotation as the direct water supply has faster flow rate.

3.4 Gantt Chart and Project Milestone of FYP 1

YEAR	2014													
Period of Study of FYP 1	JANUARY - APRIL													
WEEK NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Selection of Project Topic														
Preliminary discussion with Supervisor regarding background of project														
Preliminary Research Work (Gathering information based on books, journal and research papers)														
Preparing the draft of Extended Proposal														
Extended Proposal Revision by SV														
Extended Proposal modification														
Submission of Extended Proposal														
Proposal Defence														
Progress of Project Works (Equipment Selection, Setting-up Test Cell)														
Submission of Interim Draft Report														
Submission of Interim Report														

Project Work, Milestone

3.5 Gantt Chart and Project Milestone of FYP 2

YEAR	2014													
Period of Study of FYP 2	MAY - SEPTEMBER													
WEEK NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Continuing Research Work (Gathering information based on books, journal and research papers)														
Project Execution -Engine Modification -Shipment of Instrument -Engine Test Bed Preparation -Engine Heat Flux Testing														
Progress Report -Prepare progress report -Submission of progress report														
Pre-Sedex -Poster and presentation														
Dissertation & technical paper														
Submission of dissertation and technical paper														
Oral presentation														
Submission of Hardbound														

Project Work, Milestone

CHAPTER 4

RESULT AND DISCUSSION

In this chapter, steady-state results of cylinder wall surface temperature and heat flux of heat conduction process across engine cylinder block are illustrated and discussed. Within the considerable amount of experimental data produced during this research, a number of results based on engine operation ranging from 1500rpm (idle speed) to 3500 rpm (max speed) is outlined to illustrate the findings of this research.

4.1 Engine Testing Result

The results presented in this section were obtained at all thermocouples location where two arrays of thermocouple were placed at each location. The actual temperature distribution provided by the data logs was affected by the accuracy of the instruments and position of the tips of the thermocouples. Due to the fact that these factors a dispersion of temperature values was expected, provided that all thermocouples were properly calibrated by the manufacturer and the installation was done satisfactorily. All thermocouples had provided a good resolution of temperatures and conductive heat flux in the engine combustion chamber.

4.1.1 No Load Engine Operation's Temperature Curve

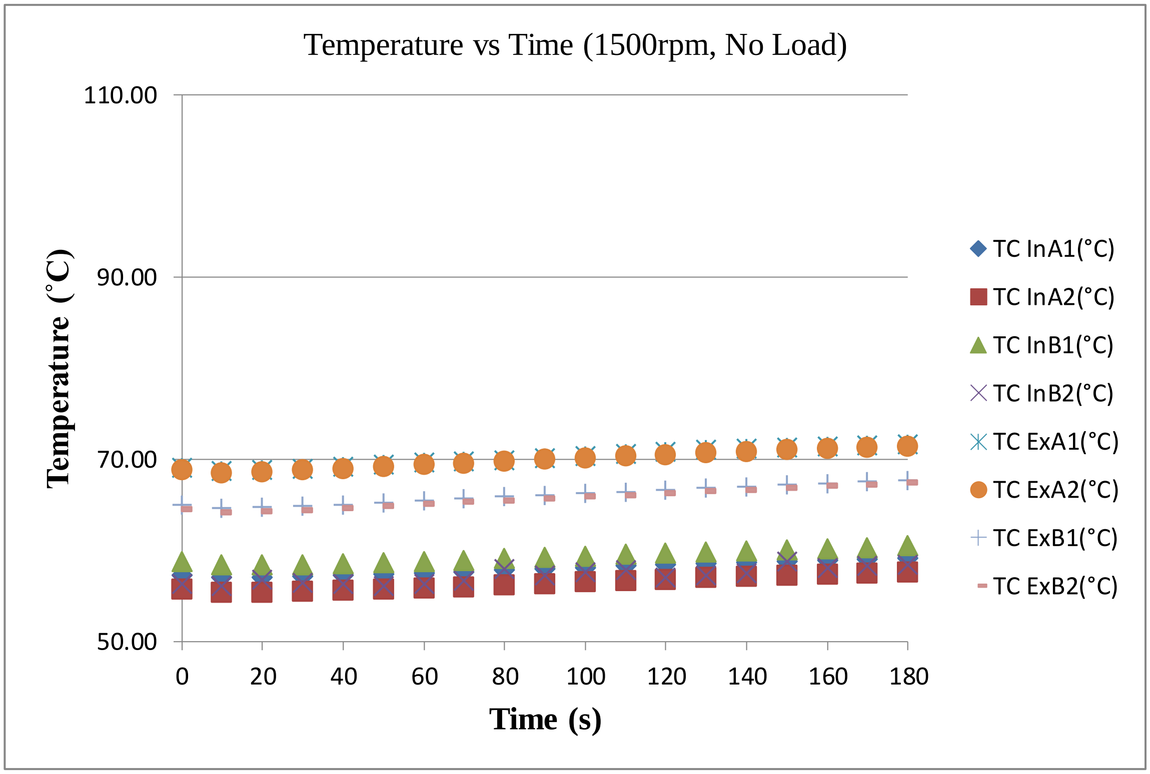


Figure 4.1: Thermal Distribution of Engine Operation at 1500rpm, No Load

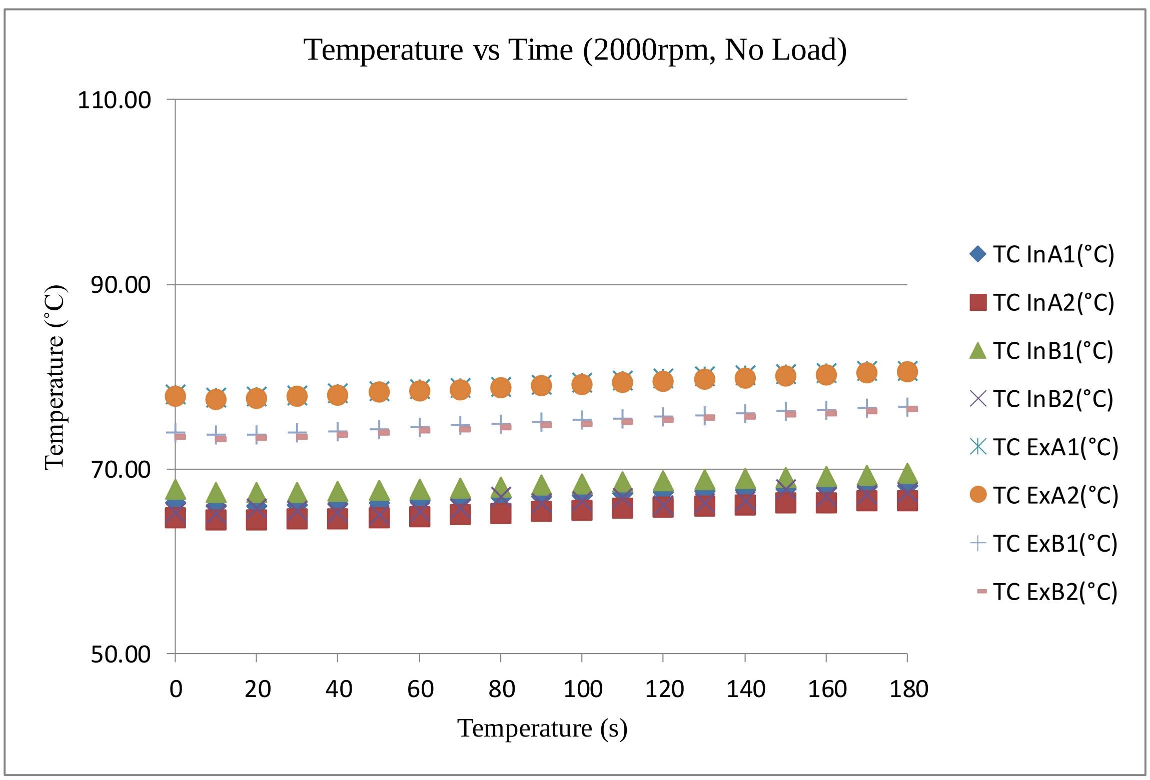


Figure 4.2: Thermal Distribution of Engine Operation at 2000rpm, No Load

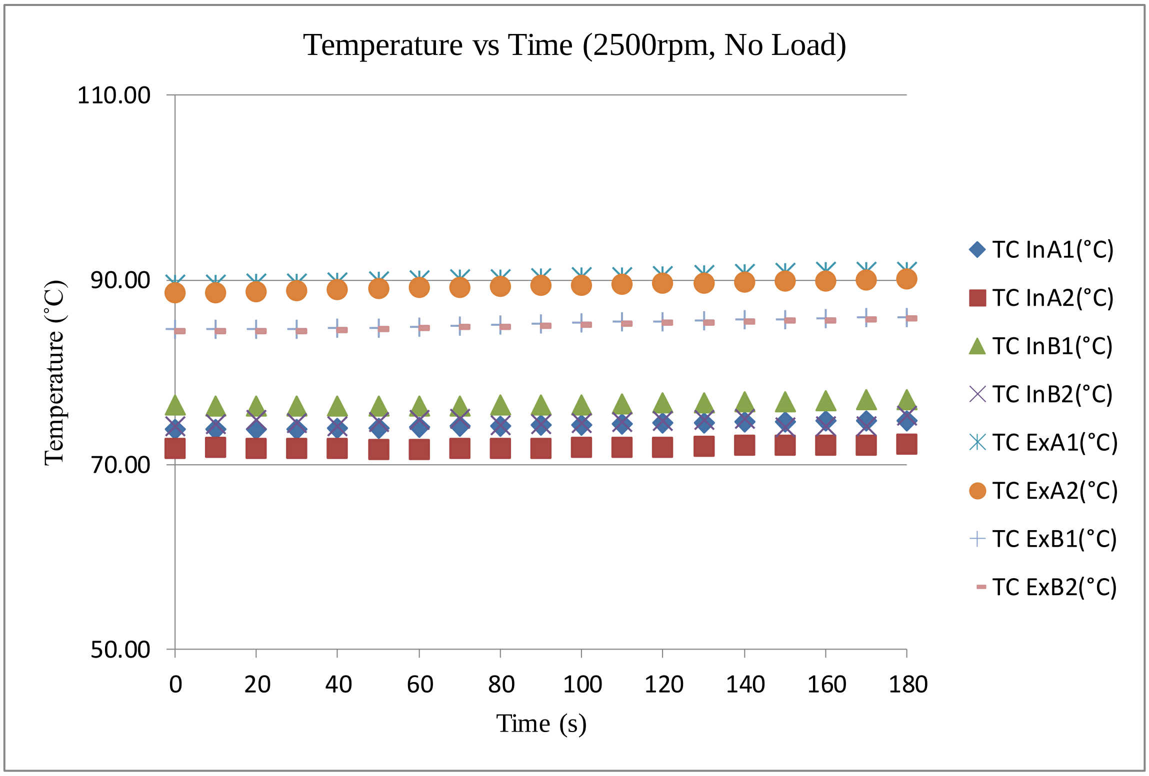


Figure 4.3: Thermal Distribution of Engine Operation at 2500rpm, No Load

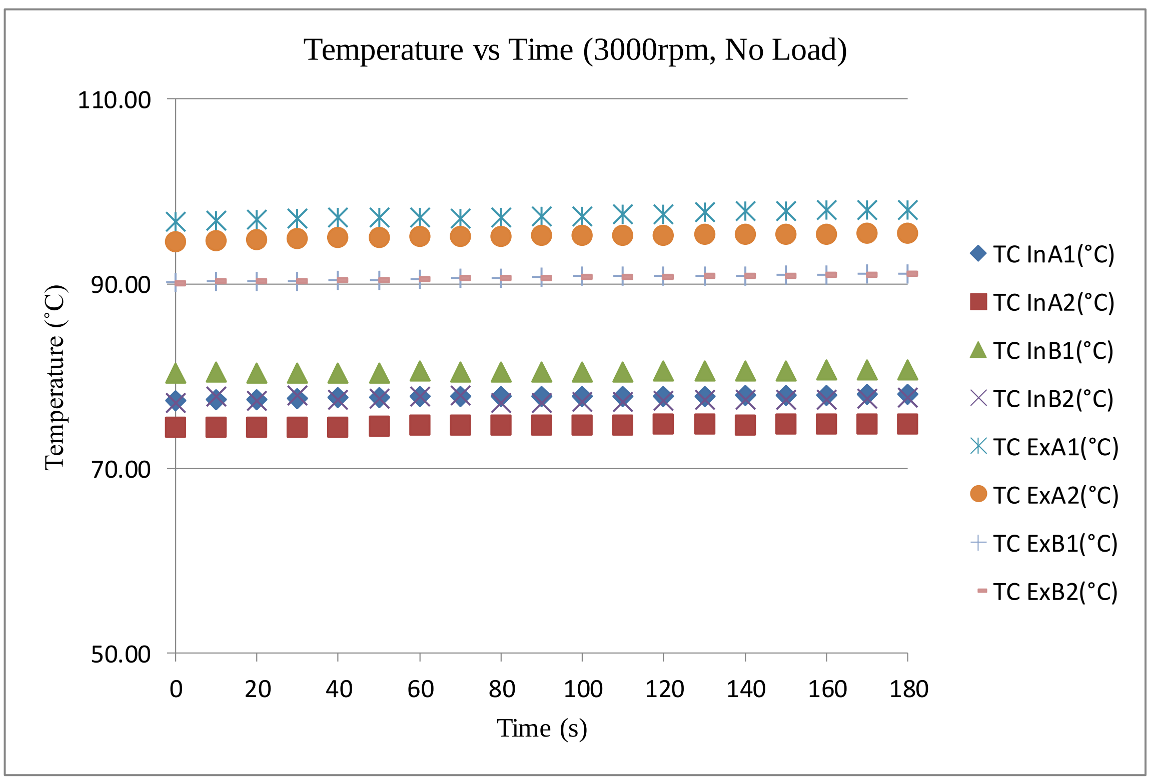


Figure 4.4: Thermal Distribution of Engine Operation at 3000rpm, No Load

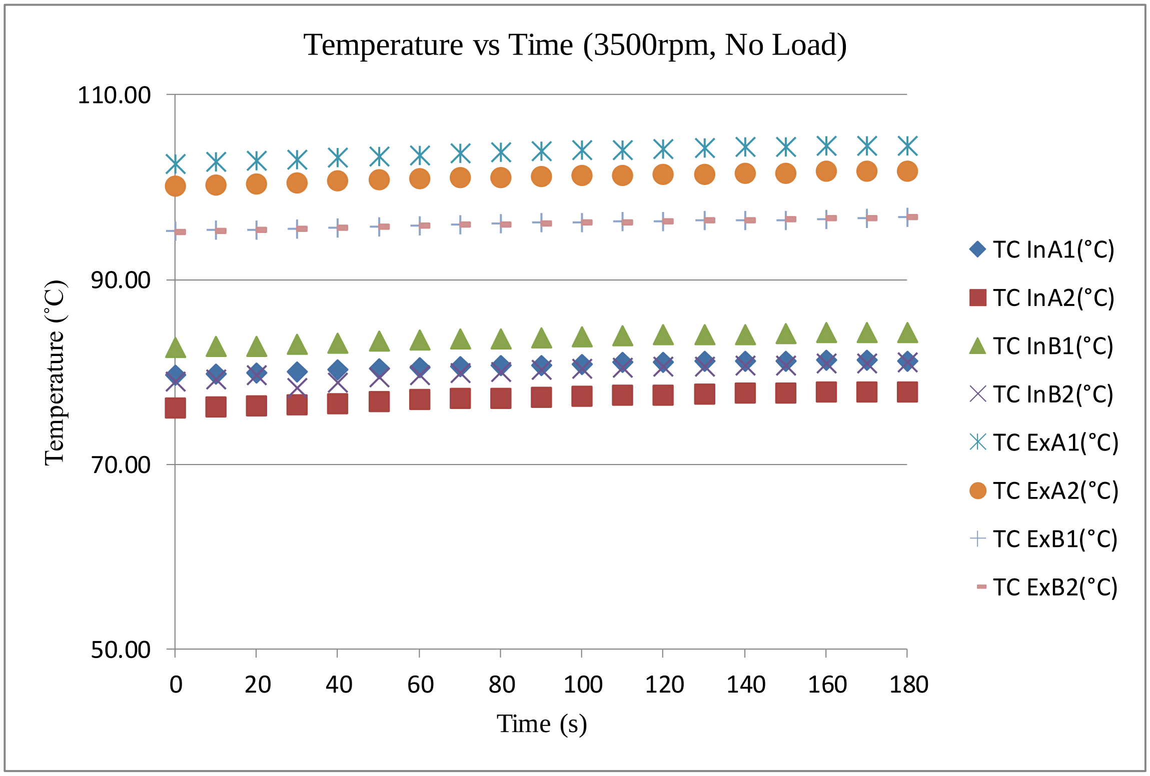


Figure 4.5: Thermal Distribution of Engine Operation at 3500rpm, No Load

Data logs of thermal distribution across the idle speed to the max speed of no load engine operation are summarized into a time-averaged temperature curve that illustrated by Figure 6 below. Basically, obtained data of the no load engine operation has conformed the behavior of engine combustion process as combustion temperature will increase as engine speed increases.

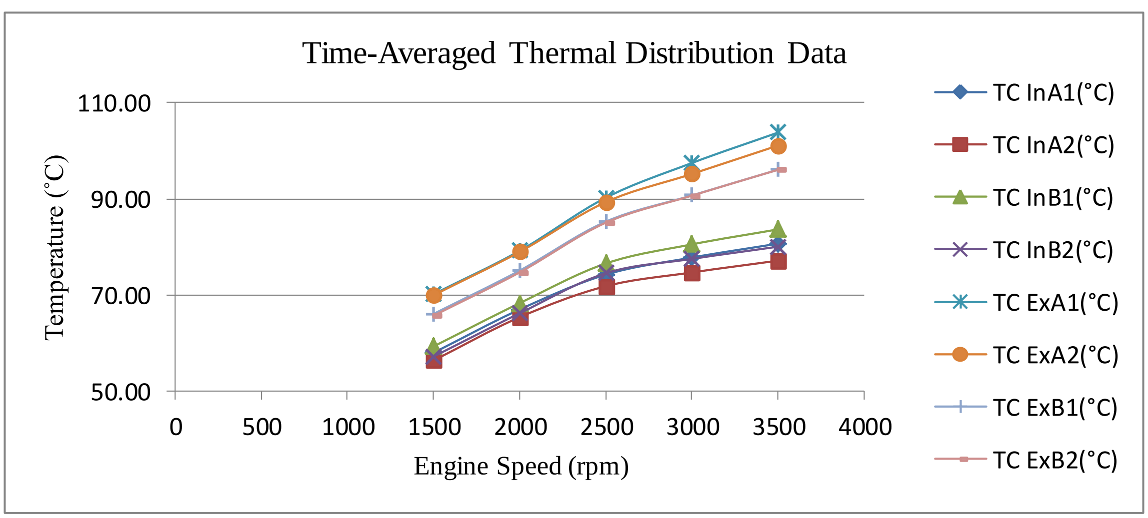


Figure 4.6: Time-Averaged Thermal Distribution throughout the No Load Engine Operation

4.1.2 With Load Engine Operation's Temperature Curve

Due to unfortunate issue that was caused by engine test bed capability, loaded engine operation was able to be conducted only up until an engine speed of 2200rpm. The attempts of increasing engine speed higher than 2200rpm while load is applied had triggered very high vibration onto the engine itself together with the mobile engine test bed. As a mobile engine test bed is simply built up of a water-brake dynamometer that is mounted on a table, it can only manage very small magnitude of vibration. The fact that it is not mounted on any isolation feet, heavy vibration that occurred cannot be passed to the ground thus cause the engine operation's vibration level was unable to be kept to manageable level. To avoid any possibility of damaging the engine and dynamometer components, the engine testing was halted at maximum of 2200rpm.

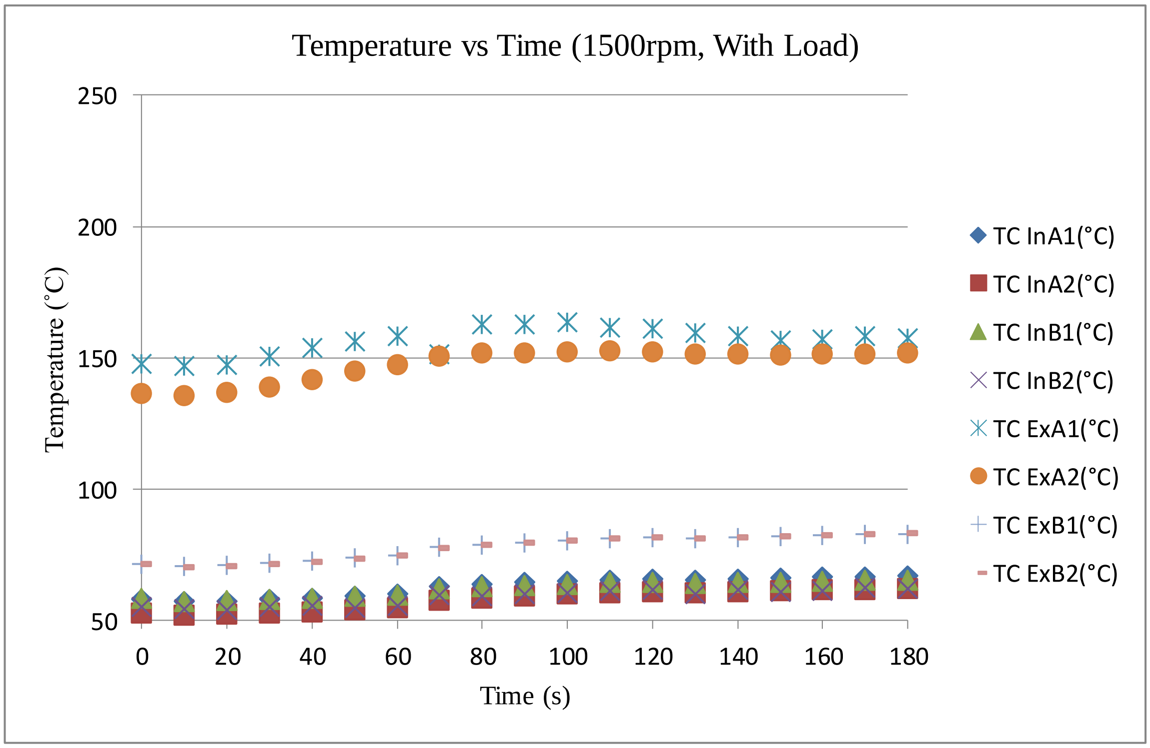


Figure 4.7: Thermal Distribution of Engine Operation at 1500rpm, High Load

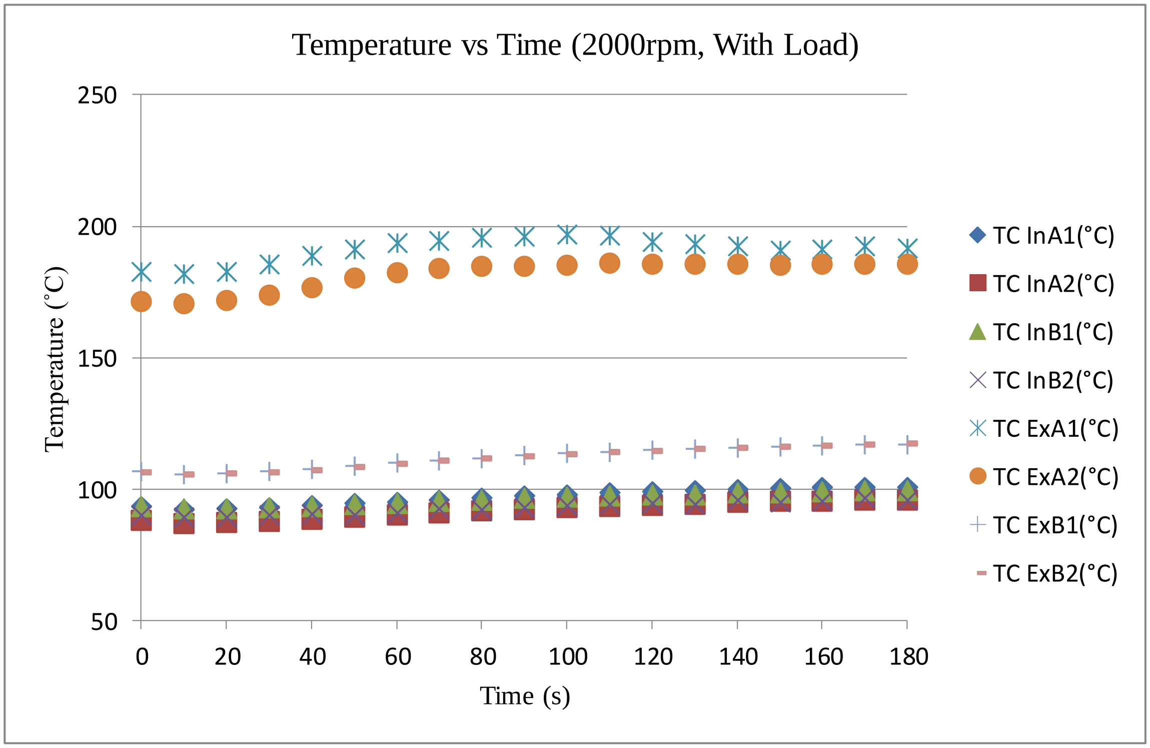


Figure 4.8: Thermal Distribution of Engine Operation at 2000rpm, High Load

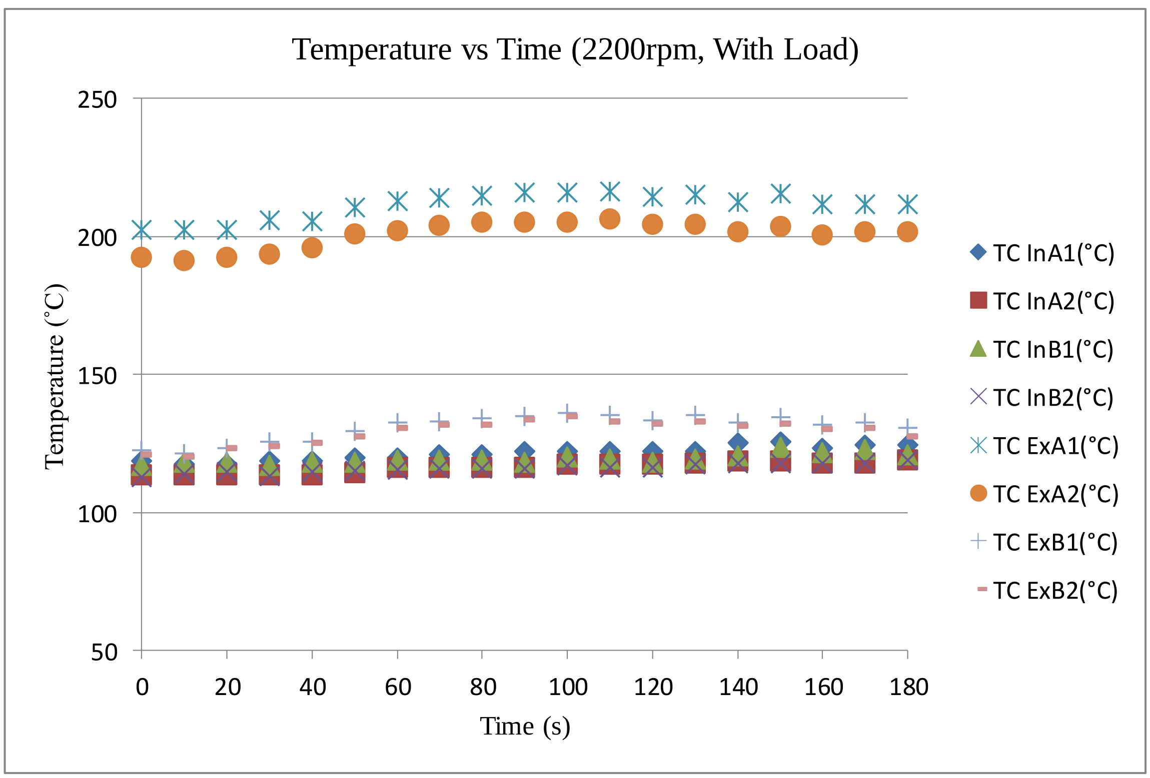


Figure 4.9: Thermal Distribution of Engine Operation at 2200rpm, High Load

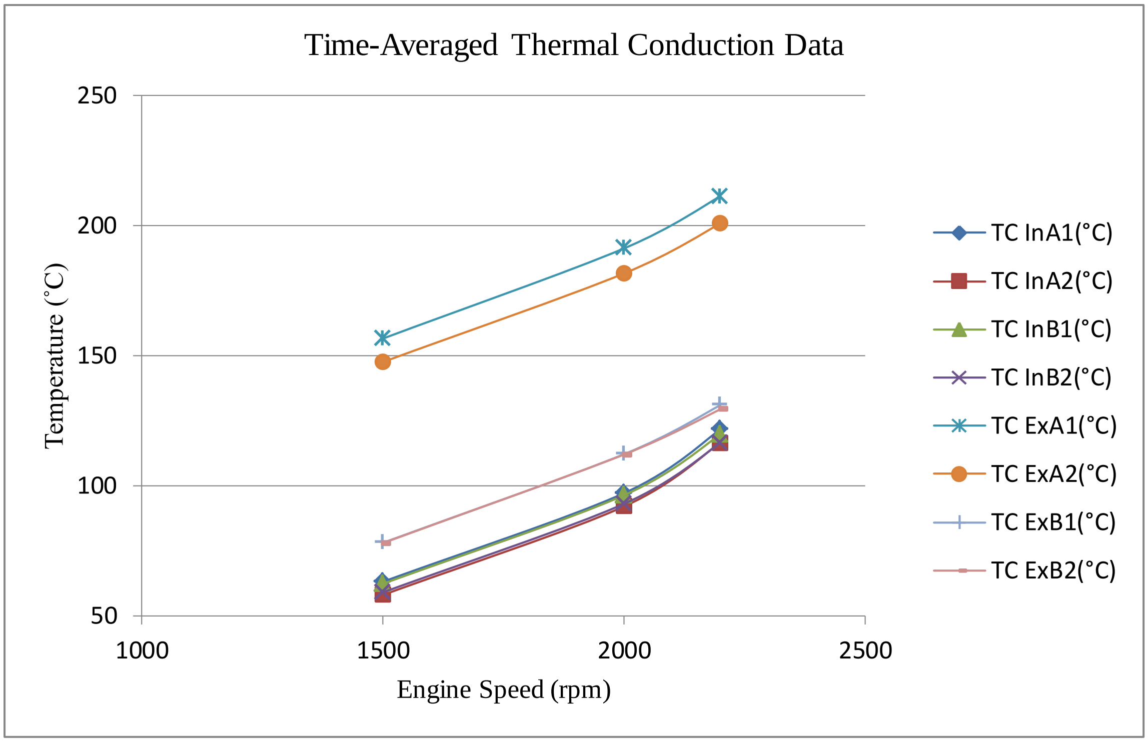


Figure 4.10: Time-Averaged Thermal Distribution throughout the With Load Engine Operation

As illustrated before by Figure 4.6, Figure 4.10 also took the same approach in summarizing the data logs of high load engine operation into a time-averaged temperature curve. The obtained data is able to point out the significant comparisons between the thermal distribution of a no-load engine operation and load engine operation. Figure 4.10 had illustrated that combustion temperature will significantly increase as load is applied onto the engine, due to the fact that more fuel injected into the combustion chamber at respective engine speeds in order to provide the required torque. A high value of difference can be observed on both time-averaged temperature curves as a different around 75˚C to 100˚C are recorded.

4.2 Data Analysis

Experimental results that were outlined based on logged data are analyzed to formulate the model of general tendencies of thermal distribution in surface temperature and conductive heat flux along the intake and exhaust side of the engine internal combustion chamber. They also reveal a realistic trend in surface temperature as a function of engine load for a given engine speed, since rising temperatures are observed as the torque increases.

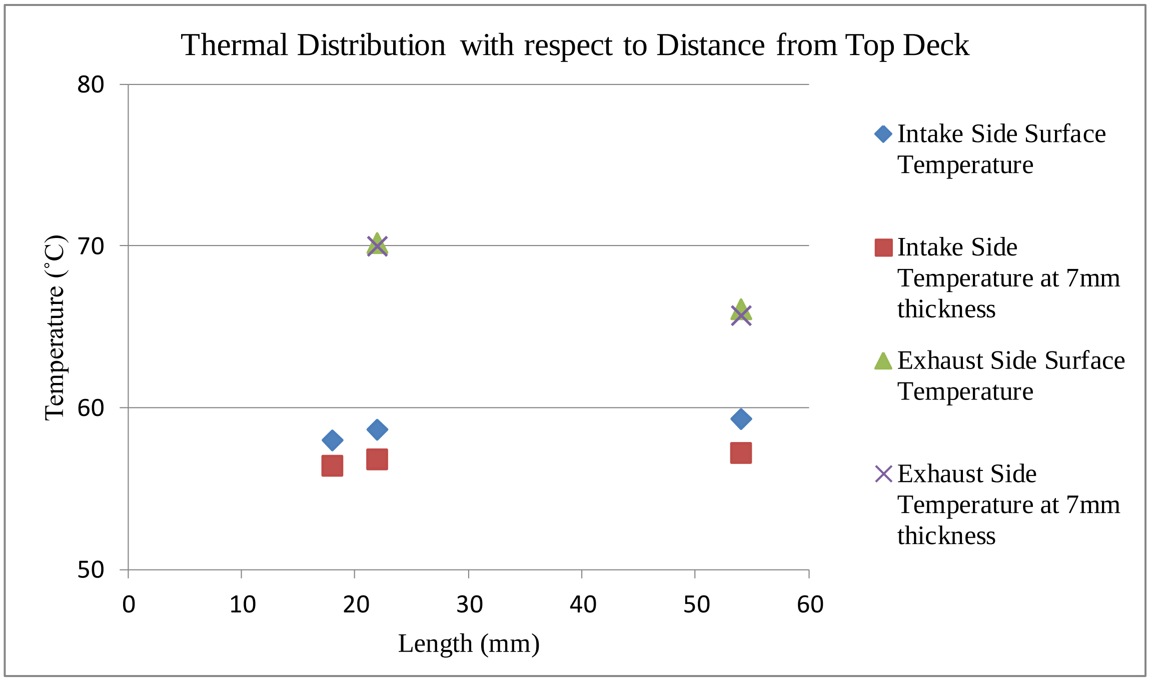


Figure 4.11: Spatial Heat Transfer along Cylinder Wall (1500rpm, No Load)

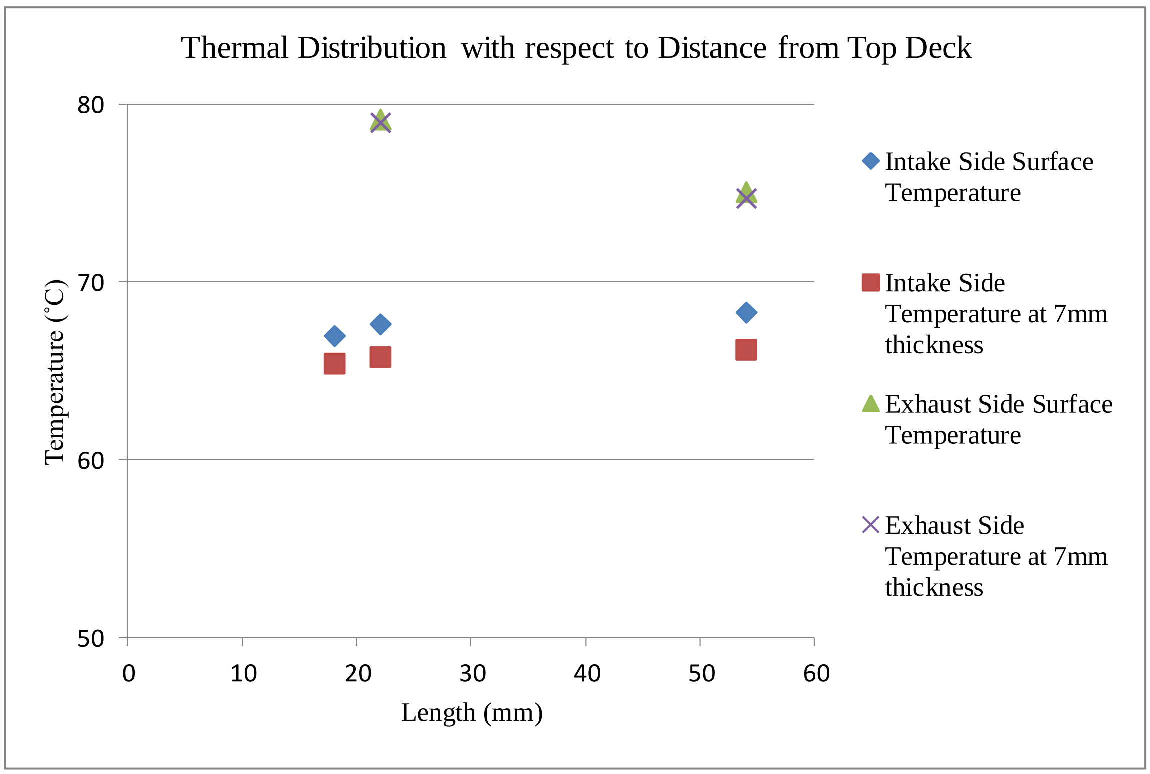


Figure 4.12: Spatial Heat Transfer along Cylinder Wall (2000rpm, No Load)

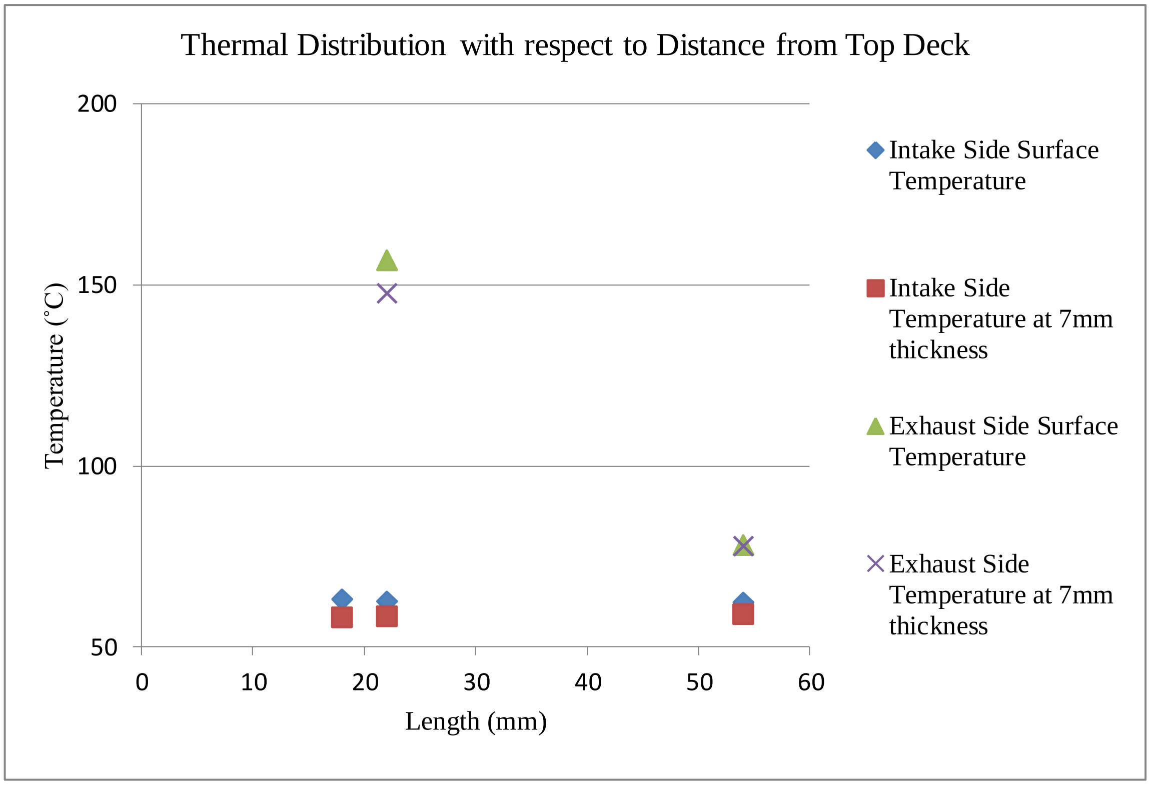


Figure 4.13: Spatial Heat Transfer along Cylinder Wall (1500rpm, With Load)

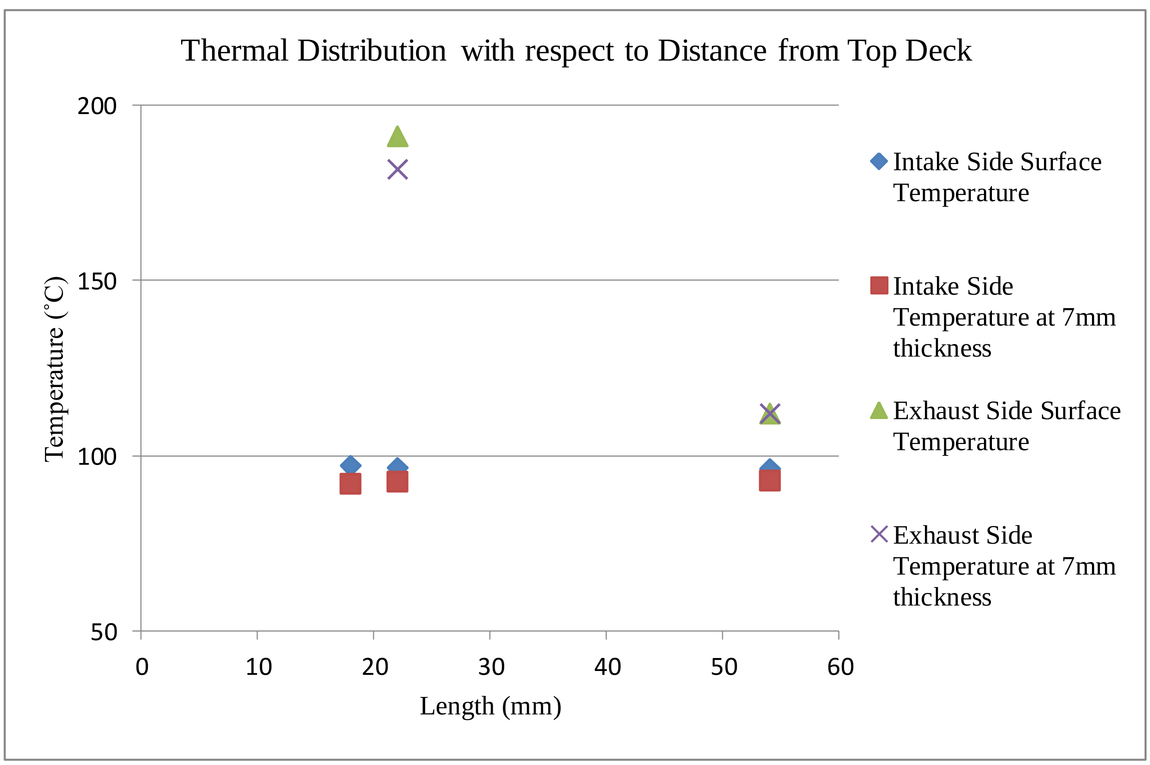


Figure 4.14: Spatial Heat Transfer along Cylinder Wall (2000rpm, With Load)

In Figures 4.11 to 4.14, a set of four graphs of temperature plotted against distance from the top deck is given for each selected locations. Interestingly, no load engine operations produced trending that showed temperature values along the intake side is lower at the top section of the bore, and later become higher as combustion proceeded downward to the bottom section of the bore. This scenario raised some uncertainty as the fact that temperature value should be higher at the top section as this is where combustion is initiated and will decrease subsequently as the combusted gases expanded.

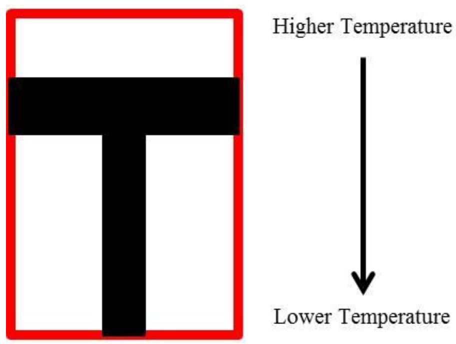


Figure 4.15: Illustration of a normal temperature variation across engine cylinder from top to bottom

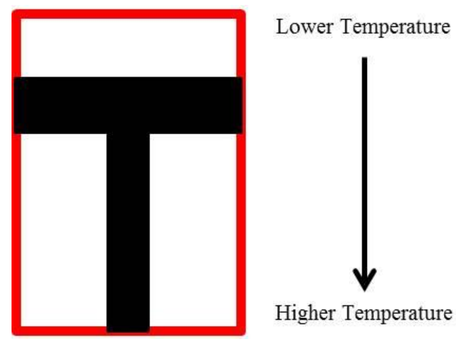


Figure 4.16: Illustration of the behavior of temperature variation across the intake side of engine cylinder from top to bottom during no load engine operation

The situation can be described to be happening as air-fuel mixture that being injected into the combustion chamber in some extent acted to be one of the cooling agents of engine combustion chamber. As mention before on the previous chapter of this paper, a compression-ignition (CI) or diesel engine basically operates by compressing diesel as fuel together with air at later will ignite itself by optimal compression and heat addition from the engine. Since the temperature developed inside combustion chamber is not significantly high when engine is running freely, not under stress, thus the air-fuel mixture had absorbed a fraction of heat the surround the top section of the combustion chamber. The combusted gases will then expand and its temperature will increase as it should and will gradually decrease when piston stroke approaching the bottom dead center (BDC). This behavior was unable to be illustrated by the obtained results as there are only two set of thermocouple arrays there placed along each intake and exhaust side of engine block, as there is no other thermocouple that is placed between these two points.

As engine testing is continued by applying load onto the engine, temperature value along both side of the wall surface showed significant increases. This observation prove the effectiveness of the applied method in analyzing the process of heat conduction across the engine block, as the process aligned with the theory that more fuel is being injected into engine combustion chamber in order to meet the torque requirement to counter the applied load. The temperature values recorded along intake side of wall surface showed a rather constant and stable reading as it does show not any huge temperature difference between the thermocouple arrays located at 22mm from top deck (top section of combustion chamber) and 54mm from top deck (bottom section of combustion chamber). On the other hand, temperature values are higher and more focused on the exhaust side of the engine combustion chamber which indicated that heat is more concentrated along the exhaust side during the high load engine operation. The temperature value increased drastically between the no load engine operation and the high load engine operation as a temperature difference about 90˚C to 100˚C were recorded at top section of engine combustion chamber.

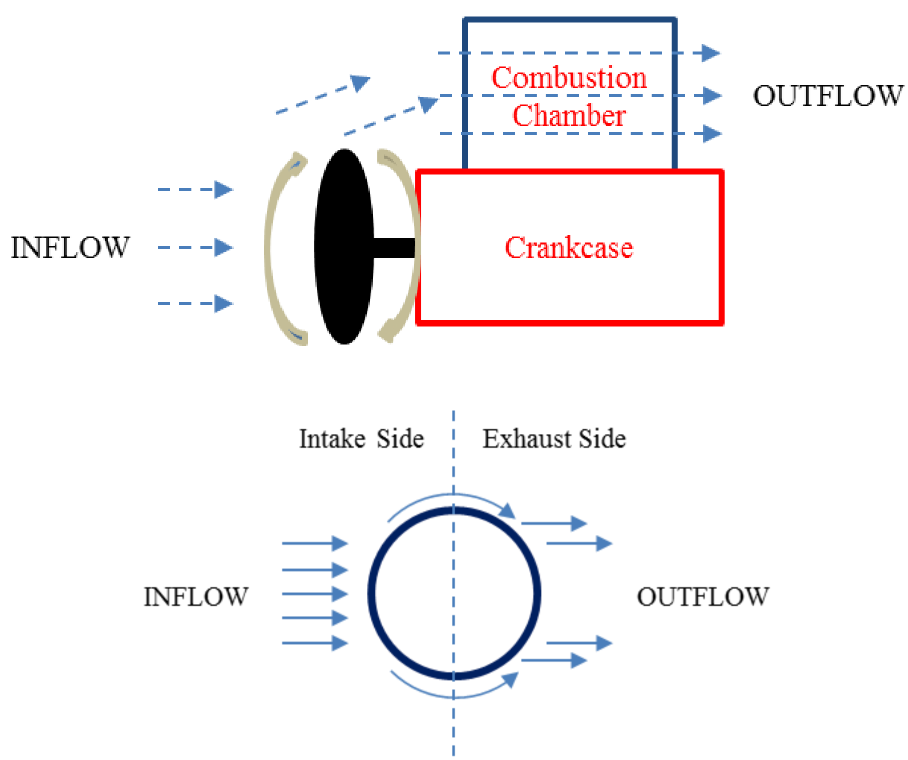


Figure 4.17: Mechanism of Airflow across engine cylinder block (Top image: Side View, Bottom image: Top View)

The difference in concentration of heat on the exhaust side of engine combustion chamber is affected by the presence of airflow on both intake and exhaust side of engine cylinder block since this is an air-cooled engine. The cooling process of an air-cooled engine is basically managed by a blower or fan that connected as auxiliary equipment or directly onto the engine crankshaft for this case. Figure 4.17 illustrated how the installed blower aided in the cooling mechanism of an air-cooled engine, as it will suck surrounding air as it spins and the air is redirected to flow across the engine block in order to cool it (Figure 4.16). Moreover, the engine cylinder block is also designed to have layers of cooling fin that played role in aiding the heat dissipation from the hotter surface (engine block surface) to the cooler surrounding (air).

Airflow will be increased as blower speed increases together with the engine crank speed. Since the blower is installed facing the intake side of engine combustion chamber, it will significantly cooled and maintained the temperature across the intake side from top section to the bottom section of engine combustion engine due to maximum presence of airflow across the intake side surface. As air flowed to the exhaust side of the engine cylinder block, it will unfortunately unable to cover overall engine block surface in order to facilitate the heat dissipation to surrounding air. Only a portion of the surface is able to be cooled by the airflow, as the heat concentrated along the uncovered section will take a longer time to be dissipated to surrounding air thus explained why heat produced by combustion is more concentrated on the exhaust side of engine combustion chamber.

4.3 Heat Flux of Conduction along Cylinder Wall

Figure 4.18 illustrated the averaged heat flux variation across the top section till bottom section of engine combustion chamber. The thermocouples of every set at all locations were arranged to allow the calculation of steady-state temperature gradients in one direction. In its simplest form the temperature gradient in one direction, $\Delta T/\Delta x$, can be found from two temperature values T_1 and T_2, measured by thermocouples located apart a certain distance Δx, by using the following equation:

$$\frac{\Delta T}{\Delta x} = \frac{T_1 - T_2}{x_1 - x_2}$$

This is the equation to obtain the one-dimensional temperature gradient in a flat wall, which was applied to predict the gradient in the direction of a cylinder axis in the engine cylinder wall. In this system, the temperature on the surface of the cylinder wall in contact with the gases is the temperature obtained for a radius of 37mm, which is at the thickness of 2mm from cylinder wall. The one-dimensional temperature distribution is used to calculate the corresponding heat flux along the engine cylinder wall. The equations are given by Fourier's law of heat conduction:

$$\dot{q_x} = -k\frac{\Delta T}{\Delta x}$$

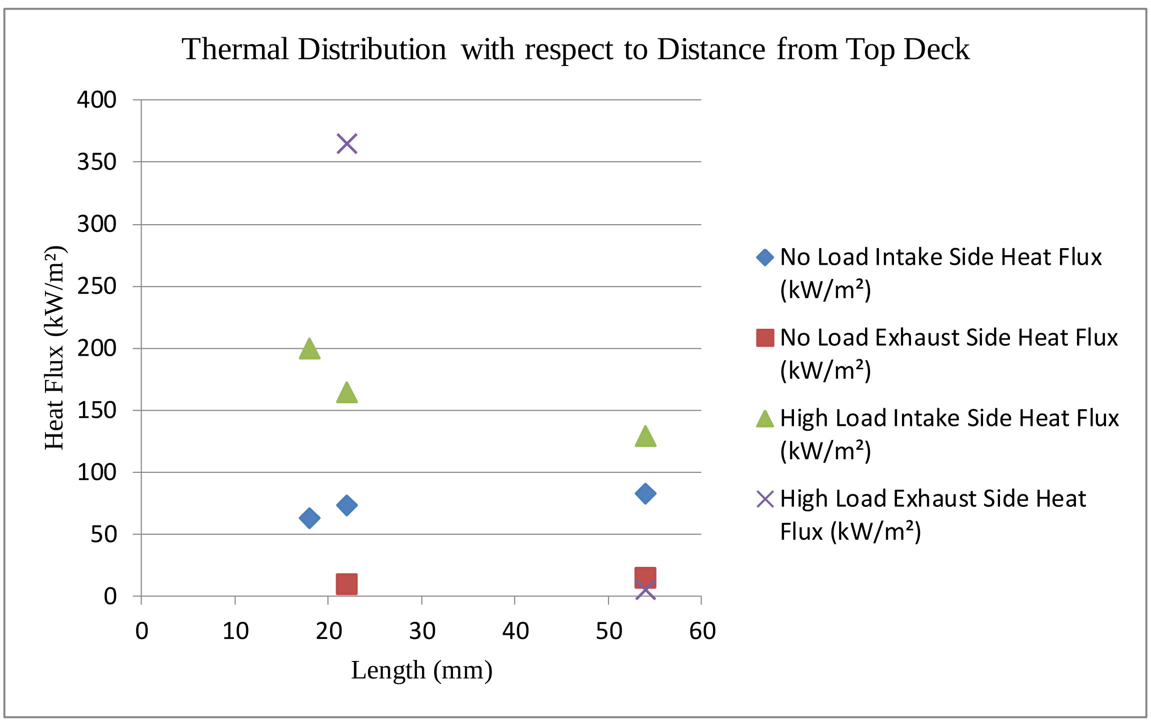


Figure 4.18: Predicted average heat flux (1500rpm & 2000rpm) at different torque condition

The constant *k*, represents the thermal conductivity of the aluminum alloy, which were the material used to build up the engine block through the casting method. The thermal conductivities used in the calculations was 195 W/m·K which is referred to the work of Parra (2008). For the same reasons given in the discussion of temperature distribution along engine combustion chamber, the heat flux was expected to be highest at the top of the bore. The illustrated figure able to highlight that the heat flux occurred at exhaust side of cylinder wall is higher than the intake side, significantly a large different at the top section of the combustion chamber during the high load engine operation. On the other hand, the heat flux at the bottom section of exhaust side combustion chamber had showed an almost the same value between the no load and high engine operation which are approaching to zero heat flux. This can be also be explained for the same reason that is given before, which that the combustion temperature had gradually decrease along the combustion chamber as it expand where it can come to the point where the wall temperature value is becoming close the temperature value that have been conducted through the cylinder block.

4.4 Other Factors that influencing Engine Heat Transfer

The interactions between combusted gases and the cylinder walls are many and different in nature. Since gas combustion's heat transfer in the combustion chamber is highly three-dimensional, it is a good exercise to consider in isolation the thermal effects of the different flow-features present in engines. The initial flow momentum is given during the induction stroke by the piston motion. The shape of the intake channels induces certain flow-structures (tumble and swirl) [20].With the compression stroke, these structures are deformed or disrupted to create an appropriate flow field at the end of compression. Later the combustion process takes place which however does not occur simultaneously in the entire chamber. During compression the charge the charged air increases its temperature uniformly. However, this is not true during combustion process. In general, the combustion propagates rapidly after fuel injection through the whole charge. However, for a short period of time, different cylinder surfaces are exposed to fluid with very different temperatures, namely the burned and unburned charge [20]. This significantly affects the local heat flux. Later in power stroke, the gases mix and fluid's properties are uniformed once again. In the case of diesel injection, the temperature gradient between and the piston surface is extremely high and, consequently, so is the local heat flux. Although the impinging jet is the event causing the highest heat flux, it is localized and short-lived compared to the rest of fluid-solid interactions. Therefore it is not the largest contributor to engine heat losses.

The heat transfer involved in the intake system occurs when air or an air-fuel mixture comes into the intake manifold. The intake manifold is hotter than the intake air because of its proximity to the engine components or the design of the manifold. The intake manifold can be designed to heat the intake air, so that once the air is charged into the combustion chamber, it can start to vaporize. The most important intake conditions for the intake charge on the engine heat transfer are the intake pressure and temperature [21]. Gas combustion heat transfer rate will increase along with the increasing of the pressure for intake charge with engine speed. On the other hand, there is no any impact for the intake charge temperature on behavior of heat transfer in combustion chamber.

On the other hand, the thermal transport properties of a material are an important issue in high temperature applications, where temperature should be kept low and the thermal gradient with the component capability. Thus uneven thermal conductivity of material across the engine cylinder block could also be the cause of abnormal thermal distribution along the engine cylinder wall. As mention before, the thermal conductivity value, k, is taken based on averaged measurement that have been conducted by the work of Parra [22]. On the contrary of the thermal conductivity value, the value might actually be different with respect across the x-axis, y-axis, and z-axis of a component. The uneven thermal conductivity along the engine cylinder wall later can create inconsistent conductivity heat flux, since temperature gradient will be higher at a specific point with higher conductivity, whereas a lower conductivity point will only allow smaller room for temperature gradient. The influences of microstructure of engine cylinder block's build might be the factor that contributed to uneven thermal conductivity across each axis.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

In the course of this research work, an experimental investigation of study operating temperatures and conductive heat fluxes in the cylinder walls of a modern diesel engine was completed. The study relied on significant amount of data, which, included measurements of metal thermal gradients at many locations along the cylinder wall of the test engine under a wide range of operating conditions. First and foremost, the developed engine test bed configuration had proved that the method was capable to measure and illustrates the distribution of in-cylinder temperature, which later aid in investigating the change of temperature profile of different engine operating condition. They had revealed a realistic trend in surface temperature as function of engine load for a given engine speed. The effectiveness of this method had also proven as the evaluation of obtained result had showed that it consistent with the theory that in-cylinder gas temperature will decrease across the engine stroke which were illustrated the decrease of wall surface temperature as gas expanded from the top section to the bottom section of the engine combustion chamber during engine combustion chamber. The analysis of the experimental data and the application of fundamental concepts of heat transfer lead the proposal of an initial concept that proposed heat loss in the mode of conductive heat transfer are more concentrated on exhaust side of engine combustion chamber compared to the intake side. However, it is suggested that extensive analysis of the experimental data should be pursued in the future to further validate the proposed concept.

There are a lot improvements that can be done in order to perform the in-cylinder heat transfer of gas inside combustion chamber in the future work. Extensive method in analyzing the proposed concept heat transfer can be conducted by including various equipment such modification of engine cylinder head by mounting fast response thermocouple in order to predict the combustion gas temperature which later can provide the illustration of temperature distribution of the gas due to the effect of radiation, and also how it diffuse across the engine cylinder wall via convective heat transfer mode. Thus, it will help on predicting the overall heat loss which contributed by the three mode of heat transfer. With regard to the experimental method, sufficient data were gathered by using fixed set of thermocouples at multiple locations along the engine cylinder block. Although thus technique was laborious in practice, it was economical and fairly reliable. Probably its main disadvantage was the impossibility of replacing faulty thermocouples once they were installed, which was consequence of the need to limit the intrusiveness of the method.

On the other hand, the installation of an angle crank encoder onto either the engine or the dynamometer will definitely aid in predicting the instantaneous heat transfer in engine combustion chamber which it will provide a wide view of the engine operation as data can be obtained at various point of engine crank angle. This data can later be translated to outline the behavior of thermal distribution over each engine stroke either during intake stroke, compression stroke, expansion stroke, and also the exhaust stroke. A more advance data logging system should also be used to replace the usage of a portable data logger as it will provide a better resolution and faster response compared to a portable one due to its capability limitation.

On top of that, engine testing should be done on a better and well equipped engine test rather than a simple engine test that was used in this research work. As pointed out before, the engine test which was used is not able to manage the heavy vibration produced by the engine during the attempt of increasing the engine speed above 2200rpm when load is applied. A better design of engine mountings and test bed foundation is needed in order to mitigate the problem that arise by the use such simple engine test bed configuration.

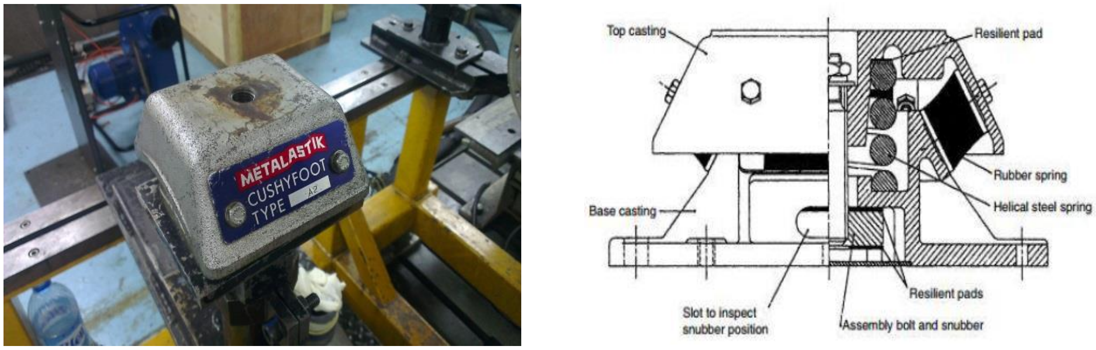


Figure 5.1: Example of combined spring and flexible mount

For example, it is desirable sometimes where the soil condition are suitable, a more modern alternative to the deep seismic block for installation of engine test bed is implemented. If the subsoil is not suitable for such arrangement, the a pit need to be casted to support a concrete block that sits on a mat or pads of a material such as rubber compound which is resistant to fluid contamination, alternatively a cast iron bedplate supported by air springs may be installed.

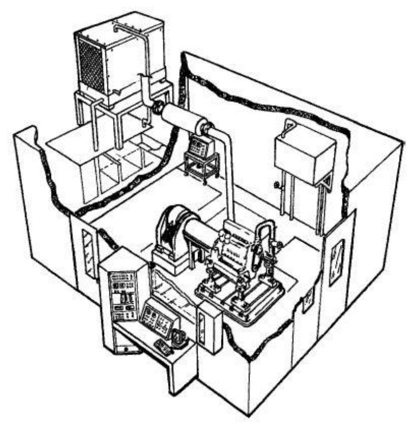


Figure 5.2: Example of a well configured and equipped test cell

Besides that, future engine testing should be done in a closed test cell rather conducting the engine testing in an open environment due to the presence of noise hazard and exhaust gas hazard. This hazard need to be contained especially noise hazard that produce by engine operation and vibration as it will definitely affect the surrounding personnel that operate or work nearby the laboratory. The exhaust gas the produced by the engine should not simply being released to environment, as a diesel engine will produce heavy smoke when it operate under heavy load which some of it can flow back into the laboratory due to wind blow. This can later trigger a safety issue to the personnel in the laboratory, or there is a possibility of triggering false alarm if enough smoke density is concentrated around the laboratory environment as it can alert the smoke sensor. The exhaust gas should also pass through a proper treatment process in order to minimize the release of hazardous exhaust byproduct to the environment such as nitrous oxide and carbon monoxide.

CHAPTER 6

REFERENCES

[1] A. J. Torregrosa, V. Bermudez, P. Olmeda, and O. Fygueroa, "Energy Conversion and Management," *Experimental assessment for instantaneous temperature and heat flux measurement under Diesel motored engine conditions*, pp. 57-66, 2010.

[2] C. D. Rakopoulos, E. G. Giakoumis, and D. C. Rakopoulos, "Cylinder wall temperature effects on the transient performance of a turbocharged Diesel engine," *Energy Conversion and Management 45 (2004)*, pp. 2627-2638, 2003.

[3] F. Salazar, *INTERNAL COMBUSTION ENGINES*. Notre Dame: Department of Aerospace and Mechanical Engineering, University of Notre Dame, 1998.

[4] W. W. Pulkrabek, *Engineering Fundamentals of the Internal Combustion Engine (Second Edition)*. Pearson Prentice Hall, 2004.

[5] C. A. F. Parra, "Heat Transfer Investigations in a Modern Diesel Engine," 2008.

[6] J. B. Heywood, *Internal Combustion Engine Fundamental*. McGraw-Hill International Editions, 1988.

[7] R. Stone, *Introduction to Internal Combustion Engine*, 3rd ed. Macmillan Press Ltd., 1999.

[8] Dataforth Corporation. Introduction To Thermocouples. [Online]. https://www.dataforth.com/catalog/pdf/an106.pdf

[9] Pyromation Inc., "Thermocouple Theory".

[10] Omega Engineering Inc. Thermocouple Introduction and Theory. [Online].

www.omega.com/temperature/z/pdf/z021-032.pdf

[11] D. Berdensky, "A Special Thermocouple for Measuring Transient Temperature," *Mechanical Engineering*, vol. 75, p. 117, 1953.

[12] V. D. Overbye, "Variation of Instantaneous Wall Temperature, Heat Transfer, and Heat Transfer Coefficient in a Spark Ignition Engine," PhD Thesis, University of Wisconsin, 1690.

[13] J. E. Bennethum, "Heat Transfer and Combustion Chamber Deposits in a Spark Ignition Engine," PhD Thesis, University of Wisconsin, 1960.

[14] G. D. Ebersole, P. S. Myers, and O. A. Uyehara, "The Radiant and Convective Components of Diesel Engine Heat Transfer.," in *SAE International Summer Meeting*, Montreal, 1963.

[15] T. LeFeuvre, P. S. Myers, and O. A. Uyehara, "Experimental Instantaneous Heat Fluxes in a Diesel Engine and Their Correlation," 1970.

[16] C. D. Rakopolulos and G. C. Mavropoulus, "Experimental evaluation of local instantaneous heat transfer characteristics in the combustion chamber of air-cooled direct injection diesel engine," *Energy 33 (2008) 1084-1099*, Jun. 2007.

[17] C. F. Taylor and T. Y. Toong, "Heat Transfer in Internal-Combustion Engines," 1957.

[18] W. J. D. Annand, "Proceedings of the Institution of Mechanical Engineers," *Heat Transfer in the Cylinder of Reciprocating Internal Combustion Engines*, pp. 973-990, 1963.

[19] G. Eichelberg, *Temperaturverlauf und Wärmespannung in Verbrennungsmotoren*, 1923.

[20] M. Bovo, "Principles of Heat Transfer in Internal Combustion Engines from a

Modeling standpoint," PhD Thesis, Department of Applied Mechanics, Chalmers University of Technology, Gothernburg, Sweden, 2014.

[21] K. I. Hamada, M. M. Rahman, M. M. Noor, K. Kadirgama, and R. A. Bakar, "Effect of Intake Conditions on Heat Transfer Characteristics for Port Injection Hydrogen Fueled Engine," Universiti Malaysia Pahang Conference Technical Paper, 2010.

[22] C. A. F. Parra, "Heat Transfer Investigations in a Modern Diesel Engine," 2008.

[23] W. Pflaum, "Warmeubergang bei Dieselmaschien mit und ohne Aufladung," 1961.

[24] M. A. Said, D. R. Buttsworth, and T. F. Yusaf, "A Review of Radiation Heat Transfer Measurement for Diesel Engines Using the Two-Colour Method," in *3rd International Conference on Energy and Environment*, Malacca, Malaysia, 2009, pp. 237-242.

[25] N. Watson and M. S. Janota, "Turbocharging the Internal Combustion Engine," 1982.

[26] Freedonia Group, "World Diesel Engines," *Industry Study with Forecasts for 2015 & 2020*, no. Study #2864, p. 478, Apr. 2012.

[27] Nick. (2009, Apr.) How Cars Work. [Online]. http://www.howcarswork.co.uk/modules/articles/article.php?id=7

[28] W. J. D. Annand and T. H. Ma, "Instantaneous Heat Transfer Rates to the Cylinder Head Surface of a Small Compression-Ignition Engine," vol. 185, no. 72/71, 1970-71.

[29] A. I. A. Agrira, "Internal Combustion Engine Heat Transfer - Transient Thermal Analysis," PhD Thesis, University of Southern Queensland, 2012.

[30] Engineers Garage. (2012) Pressure Sensors. [Online]. http://www.engineersgarage.com/articles/pressure-sensors-types-working

[31] Omega Engineering Inc. (2014) Introduction to Pressure Transducers. [Online]. http://www.omega.com/prodinfo/pressuretransducers.html

[32] Honeywell International Inc. (2014) Choosing the Right Pressure Sensors for Engine Test. [Online]. https://measurementsensors.honeywell.com/techresources/appnotes/Pages/choosing_pressure_sensors.aspx

[33] K. C. Weston, "Reciprocating Internal Combustion Engines," in *Enery Conversion*. University of Tulsa, 1992, ch. 6, p. 224.

APPENDICES

Notation

A	:	Instantaneous surface area exposed to heat transfer (m^2)
A_p	:	Piston/bore area (m^2)
B	:	Bore diameter (m)
f	:	Friction coefficient
G	:	Gas mass flow rate per unit of piston area ($kg/m^2 \cdot s$)
h_e	:	Over-all heat transfer coefficient ($W/m^2 \cdot K$)
k	:	Thermal conductivity ($W/m \cdot K$), k_g = combustion gas side
$\dot{m}_g$	:	Gas mass flow rate (kg/s)
Nu	:	Nusselt number
p	:	In-cylinder gas pressure (bar)
$\dot{Q}$	:	Heat transfer rate (heat transfer per unit time), (W)
Re	:	Renolds number
T	:	Temperature (K); T_g = combustion gas side, T_w = cylinder wall side, T_c = coolant side
V_p	:	Mean piston speed (m/s)
μ_g	:	Dynamic viscosity of gas ($kg/m \cdot s$)
σ	:	Stefan-Boltzmann constant (5.67×10^{-8} $W/m^2 \cdot K^4$)

ω : Angular crank speed (rad/s)