THERMAL STRESS ANALYSIS ON DISC BRAKE ROTOR FOR NGV VEHICLE BY USING FINITE ELEMENT ANALYSIS (ANSYS)

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A report submitted

in fulfillment of the requirements for the degree of

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DECLARATION

I declare that this project entitled "Thermal Stress Analysis On Disc Brake Rotor For NGV Vehicle By Using Finite Element Analysis (ANSYS)" is the result of my own work except as cited as reference

Signature :

Name : MUHAMMAD SYAFIQ BIN ABDUL MUNIR

Date : …………………………

APPROVAL

I hereby declare that I have read this project report and in my opinion this report is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering (Plant & Maintenance).

Signature :....................................

Name of Supervisor :....................................

Date :....................................

ABSTRACT

In modern vehicle, braking system is one of the most important systems in order to prevent accidents. Braking system is used to slow or stop the vehicle. When braking, the friction between pads and disc rotor will generate heat and will result on temperature rise. The rise in temperature usually will contribute to disc brake problem such as thermal crack and brake fade. Therefore, controlling the thermal stress of disc brake is a must in order to prevent these problems. This research will focused on thermal stress analysis on gray cast iron disc brake rotor for NGV vehicle for steady state and transient condition. The gray cast iron disc brake for NGV vehicle is designed and modeled by using SolidWorks. Heat flux and convectional heat transfer coefficient was calculated to determine the temperature distribution of disc brake. Thermal stress of disc brake was predicted by using finite element analysis technique in ANSYS and the effect of NGV to thermal stress distribution of disc brake rotor was analyzed to determine whether the disc brake is safe to use. As a result, the highest temperature distribution recorded for NGV vehicle gray cast iron disc brake is still below the maximum service temperature of gray cast iron disc brake and the value of equivalent (von-Mises) stress also below maximum tensile strength.

ABSTRAK

Di dalam kenderaan, sistem brek merupakan salah satu sistem yang paling penting untuk mengelakkan kemalangan. Sistem brek digunakan untuk melambatkan atau menghentikan kenderaan. Apabila brek ditekan, geseran antara pad dan rotor akan menjana haba dan akan mengakibatkan kenaikan suhu. Kenaikan suhu biasanya akan menyumbang kepada masalah cakera brek seperti retakan haba dan brek pudar. Oleh itu, mengawal tekanan haba brek cakera adalah satu keperluan untuk mengelakkan masalah ini. Kajian ini akan memberi tumpuan kepada analisa tekanan haba pada cakera brek besi tuang kelabu untuk kenderaan NGV dalam keadaan statik dan mengikut masa. Cakera brek besi tuang kelabu untuk kenderaan NGV telah direka dan dimodelkan menggunakan SolidWorks. Fluks haba dan pekali pemindahan haba telah dikira untuk menentukan taburan suhu cakera brek. Tekanan haba brek cakera di jangka dengan menggunakan teknik analisa unsur terhingga dalam ANSYS dan kesan-kesan NGV terhadap agihan tekanan haba cakera brek rotor telah dianalisa untuk menentukan sama ada brek cakera selamat untuk digunakan. Sebagai hasil kajian, haba agihan maksimum yang direkodkan oleh kenderaan NGV brek cakera tuangan kelabu masih di bawah suhu servis maksimum brek cakera besi tuangan kelabu dan nilai setara (von-Mises) tekanan juga di bawah kekuatan tegangan maksimum.

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LIST OF ABBEREVATIONS & SYMBOLS

NGV	Natural Gas Vehicle
FEA	Finite Element Analysis
CFD	Computational Fluid Dynamic
CNG	Compressed Natural Gas
LNG	Liquefied Natural Gas
Q	Rate of Heat Transfer
h	Convection Heat Transfer Coefficiant
As	Surface Area of Rotor
Ts	Surface Temperature
T_∞	Ambient Temperature
ε	Emissivity
σ	Stefan Boltzmann's Constant
[K]	Heat Conduction Matrix
{u}	Vector of Unknown Temperature
[R]	Radiation Exchange Matrix
{P}	Vector of Constant Applied of Heat Flow
{N}	Vector of Temperature Dependent Heat Flow
K	Kelvin
$\{\dot{u}\}$	du/dt
MPa	Mega Pascal
°C	Degree Celsius

CHAPTER 1

INTRODUCTION

1.1 Background of Research

The brake disc rotor is the rotating part of a disc brake assembly normally located on the front axle which is one of the most important part in NGV vehicles. The function of the disc brake is to slow or stop the rotation of wheel. To stop the wheel, the brake pad mounted on brake caliper is forced mechanically or pneumatically against both part of the disc. The friction between the brake pad and disc rotor of the NGV vehicle will create heat flux generation that will effect brake performance because the heat is mainly be absorbed by rotor and brake pad. Due to the generation of frictional heat on the interface of the brake pad and disc rotor, there is rise in temperature. The rise in temperature must be effectively dissipated through convection, conduction and radiation to improve braking performance. It is because, when this temperature exceeds the critical value, it leads to catastrophic events such as brake fail, failure of bearing, premature wear, thermal crack or vaporisation of brake fluid. The rate of heat generate during the braking process are depends on the certain criteria such as vehicle mass, velocity and rate of deceleration. Since this research is about a NGV vehicle, the mass of vehicle might be different from the actual vehicle mass and the rate of heat generation also will differ. In this research, finite element analysis using ANSYS will be used to predict the thermal distribution inside the rotor in steady and transient condition.

1.2 Problem Statement

This project concerns about thermal stress on disc brake rotor of a NGV vehicle. Most of the vehicles today have disc brake rotors that are made of cast iron and stainless steel. Both are chosen for its relatively high thermal conductivity, high thermal diffusivity and low cost. But, the disc brake rotor is designed to suit a regular vehicle. Are a NGV vehicle that has much higher mass than a regular vehicle suitable or safe to use the same disc brake as regular vehicles? In this project, analysis on the thermal issues of a NGV vehicle on disc brake rotor are to be done, and to determine the temperature behaviour of the disc brake rotor due to severe braking condition by using Finite Element Analysis (FEA) and effect of NGV in thermal stress distribution to disc brake rotor. Braking performance of a NGV vehicle can be significantly affected by the temperature rise in the brake components as the disc brake of a NGV vehicle require more power for disc brake to stop compared to a regular vehicle. High temperature during braking will caused to: Brake fade, thermal judder, Brake fluid vaporization, Bearing failure, Thermal cracks, Thermally-excited vibration. Therefore, it is important to study and predict the temperature rise of a disc brake rotor of a NGV vehicle and assess its thermal performance in the early design stage. Finite element analysis (FEA) has been preferred and chosen method to analyse the in thermal stress distribution to disc brake rotor during braking operation and compare it with regular vehicles.

1.3 Objective

This research is focus on thermal stress analysis on brake disc rotor during solid state condition and transient condition and to show temperature distribution of disk brake on NGV vehicle. The main objectives of this study are:

1. To study thermal stress distribution in disc brake rotor on NGV vehicle caused by temperature distribution during braking operation
2. To analyze the effect of NGV in thermal stress distribution to disc brake rotor during severe braking operation and compare it with regular vehicles

1.4 Scope Of Project

The scopes of this project are:

1. Literature review on working principle, components and theories
2. Design of 2D and 3D model of disc brake rotor
3. FE model (Meshing of geometry model)
4. Finite element analysis(ANSYS) on steady state and transient state
5. Justification of thermal stress analysis on disc brake rotor for NGV vehicle

1.5 General Methodology

The actions that need to be carried out to achieve the objectives in this project are listed below.

1. Literature review

   Journals, articles, or any materials regarding the project will be reviewed.

2. Calculation

   The calculation related to load analysis and heat transfer.

4. Simulation

   Simulation of disc brake rotor on steady state and transient state for NGV vehicle.

5. Analysis and discussion

   Analysis will be presented on how the thermal stress distribution to disc brake rotor during braking operation. Thermal stress of disc brake will be discuss based on the analysis.

6. Report writing

   A report on this study will be written at the end of the project.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction to Braking System

In a vehicle, one of the most important systems is the braking system. Braking system is used to control the speed of a moving vehicle. During braking process, braking system enable vehicle to stop within a distance. Thus, it provides safety to the passenger during emergency as it prevent vehicle to collide. To ensure the safety of vehicle, the braking system must be efficient in term of braking performance and proper heat dissipation. Braking performance involve the necessary braking torque to be applied to the wheel while heat dissipation involve dissipation of heat at brake components due to the friction between brake pads and rotor.

In this research, disc brake system is being used to analyse the temperature distribution of disc rotor. The temperature of disc brake rotor will increase as the heat generated due to the friction is high. The increasing temperature of disc rotor usually is depends on mass of the vehicle, duration of braking event and rate of retardation. This increasing temperature need to be reduced to ensure the brake efficiency is at it best.

2.2 History of braking system

In the 19th century, the first mechanisms in automotive industry to slow a vehicles momentum and prevent motion were designed and tested. Today, over 200 years later, the design of braking system has evolved into a complex device to adapt to different working conditions. From the simple design such as wooden block brake to modern day discs, braking system has improved safety and reduced the risk of car crashes worldwide. With so many types of brakes that have existed over the century, it is hard to pinpoint the inventor of the original brake system. However, those who are designed these braking systems had a common goal which is to control the speed of vehicle and to stop it. With the goal of creating safer conditions, the designers have come up with new technologies to the braking system and improve the original idea. In all new developments of braking system, the number one priority is to improve efficiency and safety of vehicles. Since the earliest type of automobiles, several methods of braking have been used such as drum. As the vehicle is keep improving, the braking systems also improve in order to catch up with the modern vehicles. There are a few type of braking system that has been used over the years as:

- Wooden block brake - The earliest braking system that applied the physical principles to design brakes today. However, this system consisted of only wooden blocks and a single lever used by the driver to apply the brake. This form was used on vehicles with steel-rimmed wheels, including horse-drawn vehicles and steam-driven automobiles.

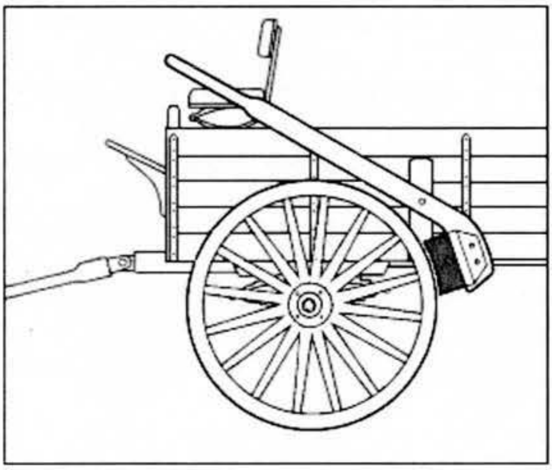

Figure 2.1 Wooden block brake (Source: www.dbrake.com)

- Mechanical drum brakes – This brake is considered to be the foundation of the modern braking system. The mechanical drum brake was first developed in 1902 by French manufacturer Louis Renault, but it had been invented earlier by Gottlieb Daimler. Daimler had theorized that anchoring a cable-wrapped drum to the vehicles chassis could be used to stop momentum, thus creating the first concept of the drum brake.

Figure 2.2 Mechanical drum brakes (Source: Knott Brake)

- Expanding internal shoe brakes - Before the expanding internal shoe brake was invented, all the braking systems had been fastened outside of the vehicle. Those systems were vulnerable to the environment such as collecting dust and water, and being affected by surrounding temperature. All of it made the brake less effective. The internal shoe brake was the first brake to be fixed inside the vehicles frame. Thus, it is an important innovation in the history of braking systems.

- Hydraulic brakes - In 1918, Malcolm Loughead proposed the concept of a four-wheel brake system using hydraulics. This system used fluids to transfer force to the brake shoe when pedal was pressed. This braking system then was adopted in majority of vehicle by the late 1920's.

- Disc brakes - This disc brake system was invented long before it become popular. The design of disc brake was patented by Frederick William Lanchester at Birmingham factory in 1902. The system was not really popular until the automotive industry began to boom in the mid-20th century. The rise of disc brakes popularity is due to the increasing weight and speed capabilities of vehicles, which caused hydraulic brakes to become less efficient in distributing heat. The heat produced when braking is dissipated directly from the surface of disc and it is more efficient than hydraulic drum brake. The first system to use disc brakes integrated both disc and hydraulic functions and was introduced in the Chrysler Imperial. Later, in the modern vehicle, most of the braking system is using disc brake.

2.3 Disc Brake

2.3.1 Introduction

Brake disc is an important component in vehicle system. A brake disc consists of a disc component bolted to the wheel hub and a stationary housing called caliper. The caliper is used to press the pads against a disc. This caliper is located at some stationary part of the vehicle like the axle casing or the stub axle as is cast in two parts each part containing a piston. In between each piston and the disc, there is a brake pad held in position by retaining pins, spring plates. As the brake is applied, pressurised hydraulic pressed fluid is constrained in the chamber pushing the contradicting cylinders and the caliper will squeezes the pad against the disc rotor. The sandwiched disc brake will results in friction. That friction slows the rotation of a shaft to hold it stationary or slow its rotational speed. Also, the friction between brake pad and disc rotor will produce kinetic energy and potential energy and it is transferred into heat which is mainly absorbed by rotor and brake pad. Due to the generation of frictional heat on the interface of the brake pad and disc rotor, there is rise in temperature. The rise of the temperature can affect the braking performance and increase brake fade. Thus, good heat dissipation is needed in order to overcome this problem.

2.3.2 How Disc Brake Function

As the brake pedal is pushed, push rod which is linked between pedal and master cylinder will pushed master cylinder pistons. This piston will slide and push return spring that is located inside bore of master cylinder, and pressure in reservoir tank is generated. Then, primary seal will allow brake fluid in reservoir tank to brake hosepipes. A secondary seal is used to make sure brake fluid does not go to other side. After that, brake fluid from brake hosepipes will enter to cylinder bore of caliper assembly and push caliper piston. Later, caliper piston will move in rolling shape with piston ring. Thus, the moving of piston will push the brake pad against the disc rotor which create friction and slow down the rotor. The friction slows the rotation of brake rotor to hold it stationary or slow its rotational speed. When brake pedal is released, piston ring will push caliper piston back to cylinder bore of caliper until piston ring and caliper piston return to their original shape and will result retraction spring to push brake pad to its original place. Also, master cylinder piston is pushed by return spring in master cylinder assembly to its original position and brake fluid will flow back to reservoir by master cylinder bore and hosepipe.

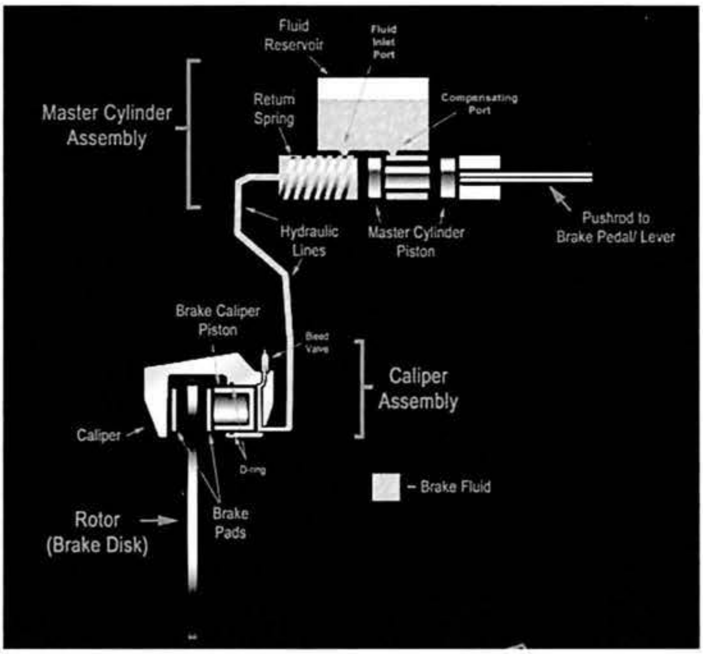

Figure 2.3 How Disc Brake Function (Source: www.bikeadvice.com)

2.3.3 Brake Disc Component

There are a few components that make the brake disc fully function. The basic components of disc brake include rotor, hub, pad and caliper assembly. Each part plays an important role in order to maintain brake performance. The wheel is mounted to the hub by wheel lug nuts and the stud. The hub houses wheel bearings allow wheel to rotate freely.

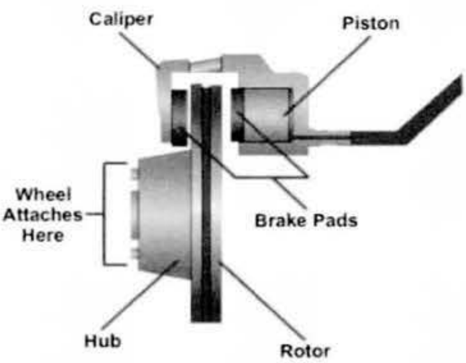

Figure 2.4: Brake Disc Components (Source: www.howstuffworks.com)

2.3.3.1 Rotor

A round disc that is able to turn as the wheel turns. The friction between rotor and brake pad will slow down the rotation of wheel and stop it.

2.3.3.2 Brake Pad

Caliper contains two brake pads which are clamped on both sides of the disc and cover much of the disc. Also called friction pads, they feature a metal portion called a shoe and a lining that is attached to the shoe. Linings are made of different materials and fall into three categories which are organic, semi-metallic and ceramic. The material chosen will impact the length of brake life and the amount of noise heard when the brakes are applied. It is because, the brake pad operate under the most extreme condition in the braking system and are subjected to high temperature when in contact with the rotor.

2.3.3.3 Caliper

Caliper plays a crucial role in braking system. Caliper is usually mounted on torque plate or steering knuckle. Caliper also acts as a housing for hydraulic and friction component. Types of brake caliper include two types which are floating caliper and fixed caliper.

- Floating caliper - Also similar as sliding caliper but sliding caliper slide on the surfaces that have been machined. Floating caliper moves in a path within its support with respect to movement of brake rotor. Floating caliper also has only one piston on one side and brake pads on the other side. Thus, it is a combination of two brake pads and one piston. It using the concept of Newton's third law of motion where "to every action there is always an equal and opposite reaction". As brake pedal is pushed, hydraulic pressure will forces piston to inner brake pad where inner brake pads will push brake rotor. At this point, the reaction force will force outer brake pads towards caliper body and caliper will generate friction on both side of rotor. Also, brake fluid is directly go into caliper by hydraulic passage inside caliper.

- Fixed caliper – Fixed caliper do not move with the rotation of brake rotor. Fixed caliper require a fixed mounting adapter and it is rigidly mounted to the steering knuckle. Both side of fixed caliper has piston. So, it is a combination of two piston two brake pads. When brake pedal is pushed, hydraulic pressure pushes both piston which will push both brake pads towards rotor. Thus, brake pads will squeeze brake rotor and create friction which stop the rotor from moving. Fixed caliper mechanism will always create equal pressure from each side. Inside fixed caliper, brake fluid is rooted through crossover line or steel tubing that is placed outside caliper housing. They are directed in to caliper halves with sealed O rings. A fixed caliper may have 2, 4, 6 or 8 pistons and there is also has three pistons which has two small pistons on the one side and one big piston on the other side depending on the requirement of vehicle.

2.3.3.4 Piston

A piston role is to push brake pads towards the brake rotor. There is a few number of piston and size and it is chosen depend on the braking force needed and the requirement of vehicle. Also, the type of caliper used can affect the number and size of piston. For example, a disc brake may have only two pistons or it may have multiple pistons in order to improve the stopping time of a vehicle. The braking force of one large piston is also equal to braking force of two small pistons.

2.3.4 Brake Disc Advantages and Disadvantages

Each type of brake has its own advantages. The advantages of disc brake are it requires less brake torque to stop the vehicle while in the same time it generates much less heat compared to the drum brake. Thus, it can cool down in a short time. Besides, disc brake requires less maintenance compared to the drum brake as disk brake is outside the wheel rim. For example, if worn out brake shoes are not being change at proper time; it can cut the brake drum in drum brake. Also, disc brake is much safer than drum brake in hard braking condition. For example, the probability of disc brake to skid is less than drum brake. Also, drum brake sometimes can lock up the rear wheel.

Despite of the advantages, disc brake also has a few disadvantages that can play a crucial role in selecting type of braking system. One of the most disadvantages is disc brake is more expensive compared the disc drum. It is because, disc brake system has more component and much more complex system. Thus, it will require extra effort in doing maintenance front like changing brake pads, rotor and brake fluid. Also, more skills is required to operate and maintaining disc brake as it is more complex compared to drum brake that is more simple. Besides, the performance of disc brake can be worst if any air is remains in disc brake system as the brake will not work effectively.

2.4 Brake Disc Rotor

Brake rotor is one of the important components in the brake system. Brake rotor is where the brake pad squeezes to slow or stop the wheel. The friction produce between brake pads and rotor surface will produce heat energy and the brake pads and rotor will absorb the heat before disburse into atmosphere. Like other brake components, there is several type of brake rotor available in market. Brake rotor can be classified into two types which are solid or ventilated. The solid types of rotor usually can be found in earlier model of vehicle or low performance vehicle as it slower cooling ability. Besides, solid rotor also can be found on the rear of four wheel disc brake system because the cost of solid rotor is much cheaper than ventilated rotor.

The other type of brake rotor is ventilated. Ventilated rotor looks like two disc of metal with ribs in between that contains of two spaced apart annular disc which each having inner and outer diameter with a locus radius. A series of vanes is distributed between the annular discs. The ventilated rotor is used in this research has cooling fins between the braking surfaces in order to increase cooling ability by increasing the air circulation. The ventilated rotor is usually found in the front wheel of a vehicle and in high performance vehicle as it has better heat dissipation and cooling ability. The proper cooling is needed to prevent fading and increase brake pads and rotor life. According to Manjunath and Suresh (2013), a ventilated cast iron disc reduction in stresses, temperature and deformation are by 22.5%, 31.47% and 8% respectively than the solid disc.

Also, there are two type of surface of brake rotor which is drilled and slotted. The drilled brake rotor has holes drilled in the rotor. The drilled rotor make it easy for heat and unwanted material to quickly moved away from the rotor surface and keep the brake performance increase as there is less surface area of the rotor. Slotted rotor use slots carved into the flat metal surface to move heat and unwanted material. It is usually found on the high driving performance vehicle.

2.5 Heat Transfer Finite Element Theory

2.5.1 Introduction

In this research, heat transfer is used to predict the energy transfer in the brake disc rotor. As this research is using a NGV vehicle, it requires the repeated high torque brake application which may lead to the rises of rotor temperature. The increasing temperature of brake disc rotor is depends on a few factor such as mass of the vehicle, the rate of retardation and the duration of braking event. This increasing temperature of rotor needs sufficient heat dissipation to avoid further heating which may lead to brake failure. If the heat generation inside the brake disc rotor is higher than heat dissipation then the temperature will rise and the rate of this rise will depend of the relative quantities of each.

Figure 2.5: Heat transfer modes (Source: Pulugundla, 2008)

Based on the figure above, the heat transfer from the brake disc rotor is dissipated in three different modes which are by conduction, convection and radiation. Heat dissipated via conduction through the brake hub and assembly while convention is dissipated to the atmosphere and radiation through nearby components. Sometimes, at high temperature heat may create chemical reaction in the friction material and may dissipate some of the braking energy. Besides, there is some previous research on the contribution of each different modes. According to Voller (2003) at temperature 6000°C and at 150 rpm, conduction contributes 18%, convection 39% and radiation 43%. Then at speed of 450 rpm, the contribution of convection increase to 57% and conduction and radiation are essentially independent of speed. However, according to Limpert (1975) contribution of radiation is less than 5% of the total heat transfer at normal braking condition. Polansky (2003) has performed CFD studies on the contribution of different modes of heat transfer. The result shows that convection was the major contributor while conduction and radiation has lower contribution. Conduction heat transfer contributes 11%, convection 82% and radiation was 7%.

2.5.2 Conduction

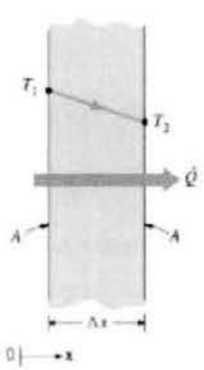

Figure 2.6: Visualization of theory of thermal conduction (Source: Pulugundla, 2008)

Heat conduction or thermal conduction is described as the energy transfer of heat in a solid between particles. It occurs when there is a temperature different between its body and surrounding medium. The higher temperature flows towards lower temperature. Based on the figure, heat flows across the plane from T1 to T2 where T1 is greater than T2. Thermal conductivity is used as the material property for transferring heat and it is defined as λ. That means, a material with higher thermal conductivity transfer at higher rated compared to material that has lower thermal conductivity.

The conduction on in brake disc rotor is when heat flows from hot part of brake disc to cold part due to the electron in the hot part has higher kinetic energy compared to those in cold part. Also, conduction is used by the designer to move the heat from the disc pad interface to the vanes. However, the excessive heat conduction can lead to brake failure. According to Limpert (1975) heat conduction to the components around the brake can lead to brake fluid vaporization, damaged seals and wheel bearing damage, while heat radiation can cause damage at tyre at temperature as low as 93°C.

2.5.3 Convection

Convection or known as convective heat transfer can be described as a physical behavior of heat transfer by moving air or fluid that transports heat energy from one place to another. For brake rotor, convection to the atmosphere is the primary heat dissipation because according to Sheridan (1988), total heat dissipation by convection to surrounding is approximately 90% in all braking condition. The convection of the brake rotor occurs when moving air flow through the heated surfaces of rotor. The air will absorb thermal energy along the vent of the rotor. Thus, the heat generated during braking process is transferred to the moving air. This will contributed to the total heat loss and can be expressed using Newton's law of cooling

$$Q=hAs\ (Ts-T\infty) \tag{2.1}$$

Where Q is the rate of heat transfer (Watts), h is the convection heat transfer coefficient (W/m^2k), As is the surface area of the rotor (m^2) and Ts and T∞ are the surface temperature of the brake rotor and ambient temperature.

From the equation, in order to maximize heat transfer from the rotor and keeping the rotor temperature to minimum, the value of heat transfer coefficient or the surface area needs to be increased. As to keep Ts minimum, improvements on heat transfer coefficient and surface area of rotor is needed. According to Limpert (1975), the surface area can be increase by increasing the diameter of the wheel and minimizing the unsprung mass, so the improvements in cooling can best be made through increased value of heat transfer coefficient. Also, according to Incropera and DeWitt (1996), heat transfer coefficient is dependent on boundary layer, which is influenced by the nature of the fluid motion around the rotor, surface geometry as well as fluid and thermodynamic properties.

2.5.4 Radiation

Besides conduction and convection, radiation also contributes to the total heat loss of the brake disc rotor. According to Noyes and Vickers (1969) and Limpert (1975), it is estimate that amount of radiation that contribute to heat dissipation under normal braking conditions is less than 5% of the total heat dissipation. Although radiation did not contribute much in heat dissipation and some might say that amount of heat dissipation through radiation under normal braking condition can be negligible, uncontrolled radiation heat transfer might affect braking system because according to Limpert (1975) radiation heat transfer from rotor will give greater effect at higher temperature but must be controlled to prevent beading of the tyre. Radiation can be explained by the Stefan-Boltzmann law as given in equation

$$Q = \varepsilon A \sigma (T_{obj}^4 - T_{env}^4) \quad (2.2)$$

Where ε is the emissivity ($0 \leq \varepsilon \leq 1$), σ is Stefan Boltzmann's constant ($5.67 \text{ x } 10^8$), A is the surface area, T_{obj} is the temperature of the object and T_{env} is the temperature of the surroundings.

2.5.5 Steady State Analysis

In this research, the steady state analysis is used to determine the temperature distribution of disc brake rotor under steady state loading condition. For steady state analysis, heat storage effect change over period of time can be ignored. The general form of steady state heat balance equation is:

$$[K]\{u\}+[R]\{u + Tabs\}^{2} = \{P\} + \{N\} \qquad (2.3)$$

Where [K] is the heat conduction matrix, {u} is the vector of the unknown temperature, [R] is radiation exchange matrix, Tabs is temperature offset from absolute required for radiation calculation, {P} is vector of constant applied of heat flow and {N} is vector of temperature dependent heat flow.

2.5.6 Transient Analysis

In this research, the transient thermal analysis is used to determine temperature distribution of disc brake rotor and other thermal quantities that varies over time. The general form of transient heat balance equation is:

$$[B]\{\dot{u}\} + [R]\{u + Tabs\}^{4} = \{P\} + \{N\} \qquad (2.4)$$

Where [B] is heat capacity matrix, $\{\dot{u}\}$ is the du/dt , [R] is radiation exchange matrix, Tabs is the temperature offset from absolute required for radiation calculation, {P} is vector of constant applied heat flow and {N} is vector of temperature dependent heat flow.

2.6 Finite Element Analysis

2.6.1 Introduction

Before computer is widely used across the world, analytical method is used by the engineer to determine the integrity of a design. Today, finite element analysis has become a powerful tool to determine the numerical solution of wide range of engineering aspect. This analysis is a numerical technique to find approximate solution of integral equation and partial differential equation. The generality of this analysis fits the analysis requirement of complex engineering system and design where solution of governing equilibrium equation is not available. Finite element analysis uses a complex system of point called nodes which generated a grid called mesh. This mesh is to define how the structure will react to loading condition. Thus, increase the accuracy of the analysis.

2.6.2 ANSYS Software

The ANSYS is a general purpose finite element analysis software package. The software operates on most major computer and operating system. It also a self-contained analysis tool incorporating pre-processing such as solver and post processing modules or creation of geometry and meshing. This software implements equations that control the reaction of the element and solve it. Thus, it create a comprehensive explanation of how the system act as whole. The results of the ANSYS can be presented in graphical forms or tabulated. Besides, this software is used for design and optimizes a system that is complex to analyze by hand. Hence, the researcher or engineer can analyze a design or product that is too complex due to their scale, geometry and equation and can reduce the number of costly prototype.

2.6.3 Application

Researcher and designer are using finite element analysis in designing or improving a product by including specific content such as fluid, thermal and structural working environment. The result from finite element analysis can shows the detail visualization of a structure whether it is bend or it can support its load or not and it also can indicates the distribution of temperature, displacement and stresses. The accuracy of the results and computational time requirement can be managed simultaneously to cope with the design. Besides, this analysis can accelerate design process and improve it by decreasing the time taken for product development through prototype. For this research, finite element analysis can analyze temperature distribution on disc brake rotor without have to do a prototype. Thus, it can decrease time taken for analysis and cost while in the same time it can increase accuracy.

2.6.4 Finite Element Analysis Stage

First of all, steady state and transient condition must be setup and materials properties for disc brake rotor such as density, poison ratio, thermal conductivity, specific heat and other properties are applied. Next, auto mesh is use to generate mesh to the model due to the complexity of brake rotor. Auto mesh is use because to suit the computer performance as the higher number of nodes will increase the computer processing time. Also, heat flux are applied and are assigned to the elements within the contact zone. It is because, the number of elements can be vary from one surface to another due to the different length and dimension. Location of the contact zone is to be determine by relative position of brake pads and rotor from the calculation of the deceleration rate of disc rotor. Heat flux are being applied to disc brake rotor to perform thermal stress analysis. In order to predict the temperature distribution of disc brake during braking, steady state and transient condition of heat transfer analysis is carried out.

2.7 Natural Gas Vehicle (NGV)

A natural gas vehicle (NGV) is an alternative fuel vehicle that is used in automotive use. In order to reduce the dependency of petroleum, some vehicle is using natural gas as an alternative fuel. NGV in Malaysia mostly from bi-fuel vehicle which the petrol vehicle is converted to NGV and the vehicle is allowed to run with petrol or natural gas. The form of natural gas inside a NGV is either compressed natural gas (CNG) or liquefied natural gas (LNG). The different between CNG and LNG is its form. CNG is compressed to around 3600 psi while LNG is cooled to negative 162 degree Celsius. However, most of the NGV used in Malaysia is using CNG. The compressed natural gas is usually is being stored in a cylinder that is placed in car boot. The size of the cylinder may be vary according to its storage capacity. There are a lot of NGV advantages such as fuel cost saving, lower operating cost, cleaner environment and extended travel range.

In this research, type of NGV vehicle that is being used is Naza Ria. Naza Ria is being chose in this research as a lot of Naza Ria is converted into NGV vehicle due to its higher fuel consumption. The effect of NGV in thermal stress distribution on disc brake rotor of Naza Ria will be analyze. The actual dimension for Naza Ria and its brake disc rotor is used. The result of the analysis is then being compared to the normal Naza Ria vehicle. As the NGV Naza Ria have slightly different mass and performance, the result should be slightly different. Type of NGV that is been used in this research is based on CITY NGV SDN BHD where its component weight is given as table below.

Table 2.1: NGV components weight (Source: CITY NGV (M) SDN BHD)

Item	Weight (kg)
NGV 65 Litre Tank	56
System	8

2.8 Review of Previous Research

There are a lot of researches that have been conducted in order to analyze the thermal stress on brake disc rotor. Each research contains valuable information that is used as a reference in this project. The thermal stress analysis of brake disc rotor has a lot of complicated step of analysis by considering air flow and heat flux. S. Koetniyom (2000) studied about thermal stress analysis of automotive disc brakes on the Rover brake disc rotor using ABAQUS. Because of the brake disc rotor is symmetry, the model of the rotor is 20° segment of the rotor and it is meshed using 3020 noded solid element with a quadratic interpolation function. The rotor material properties in tension and compression were used to generate FE material model routines which can allow different temperature dependent yield properties of cast iron in tension and compression. The result shows that the temperature increase non-uniformly with the braking time and the maximum temperature is around 380 °C at end of brake application. Thermal stress result indicate the maximum Von Mises elastic stress was 273 MPa at the neck and 442 MPa at the inner fillet radius of long vane. If this stress is beyond the proportional limit, plastic strain would happen. Nakatsuji et. al.(2002) did a study on initiation of hair-like cracks which is formed around small holes in the flange of one piece disc under overloading condition. The temperature distribution at the flange under overloading is measured in order to show the crack initiation mechanism. The results that thermally induced cyclic stress strongly affects the crack initiation in the brake disc. Hogskolan (2012) did a study on simulation of thermal fatigue in a disc brake by using input of heat flux produced from the friction between rotor and pad in repeated braking cycles. He used finite element analysis (FEA) to determine the temperature distribution in the rotor and the stress for repeated braking to calculate fatigue life of a disc rotor. The thermo-mechanical problem is divided into two part which is heat analysis and thermal stress analysis. Heat analysis was done by including frictional heat and adopting an Eulerian approach. Thermal stress analysis part was done by using ABASQUS. The linear kinematic hardening model with rate independent elastic-plastic plasticity was used in this research. The results of the model and real model were observed to be similar in term of plasticity theory. First brake application stay for 6 second. The result show maximum temperature of 220 °C at 2.5 seconds and mean equivalent stress of 180 MPa was shown.

CHAPTER 3

METHODOLOGY

3.1 Introduction

This chapter will describe about the action that need to be carried out in order to achieve the objective of this project. This project starts by studying the braking system specializes in disc brake rotor and thermal stress analysis by using finite element analysis. The data such as the dimension of disc brake rotor is collected and recorded. Then, a ventilated disc brake rotor is modeled using SolidWorks and design input for rotor will be calculated. After that, finite element modeling and simulation will be carried out using ANSYS in order to determine the thermal stress on disc brake rotor. The flow chart in Figure 3.1 will describe in detail about the methodology for this project.

3.2 Overall Process

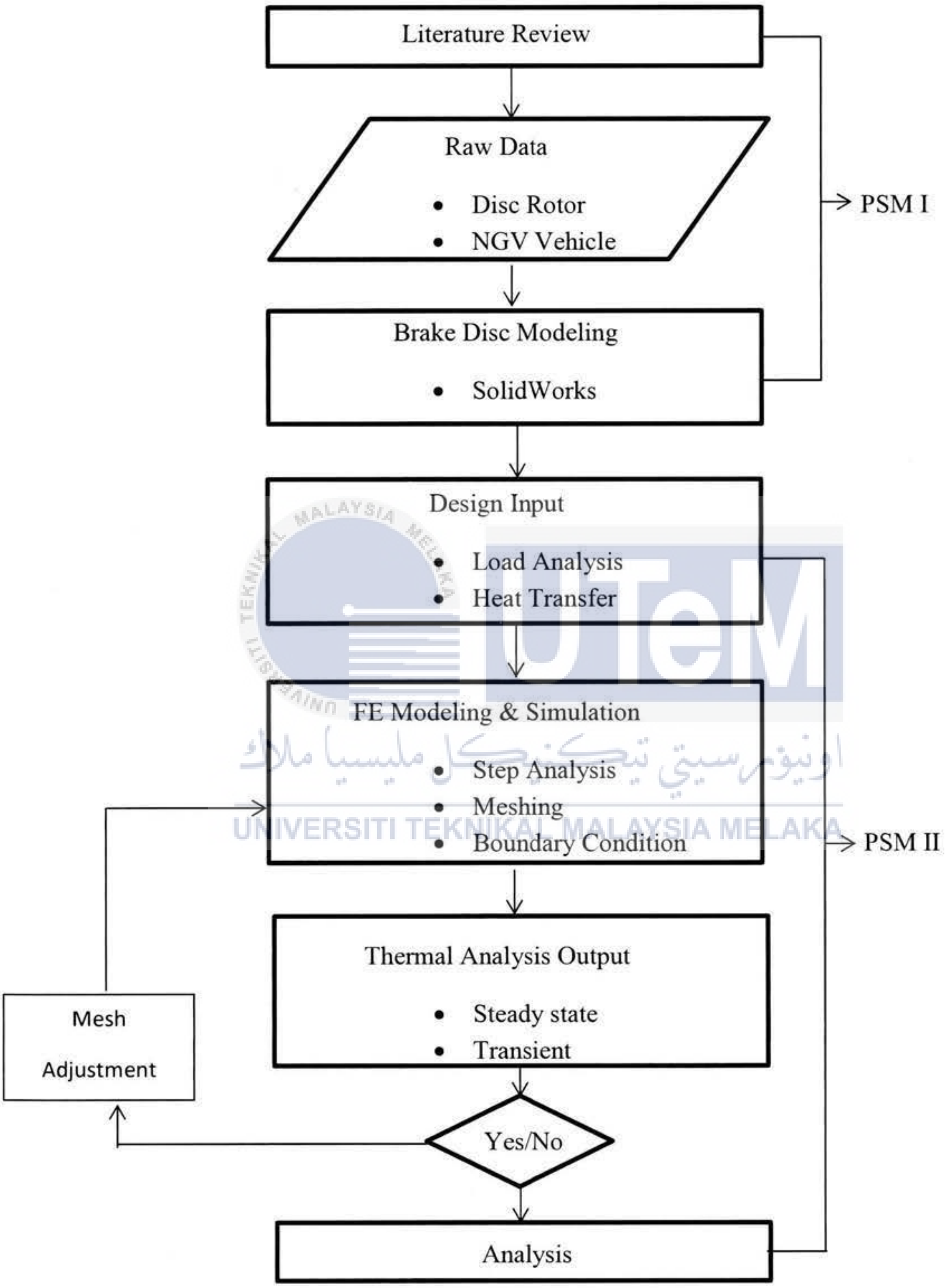

Figure 3.1: Flow chart of the methodology

Based on the flow chart in Figure 3.1, overall process that is involve in this project was stated. All the process need to be carried out in order to achieve the objective of this project. This project starts in the literature review section and end in analysis section. In literature review, all theory involves in this project such as braking system, heat transfer and finite element analysis was stated in order to understand this project. Then, raw data for disc brake rotor, Naza Ria and material properties for disc brake rotor is collected. The data is used in the rotor modeling and finite element modeling. After data was collected, disc rotor was modeled using SolidWorks.

Design input for disc rotor will be calculated. Design inputs include load analysis such as braking power and heat transfer for rotor will be calculated. Next, finite modeling and simulation will be carried out using ANSYS. This part is the most important part in this project whether to determine the objective is achieved or not. This part includes step analysis, meshing and boundary condition of the disc rotor. Lastly, analysis will be carried out based on the result generated from the simulation. If the output of simulation is not accurate, adjustments of mesh need to be carried out.

3.3 Raw Data Selection

3.3.1 NGV Vehicle & Specification

Figure 3.2: Naza Ria (Source: www.otofacts.com)

Naza Ria is a multi-purpose vehicle (MPV) that is widely used in Malaysia. In this research, type of NGV vehicle that is being used is Naza Ria. Naza Ria is being chose in this research as a lot of Naza Ria is converted into NGV vehicle due to its higher fuel consumption. Naza Ria use front vented disc brake and rear drum brake. This research will only focus on front brake. The details of Naza Ria specification is stated in table below:

Table 3.1 Naza Ria Specification (Source: www.otofacts.com)

No	Type	Details
2	Displacement (cc)	2,497
6	Front Suspension	Macpherson strut with stabilizer
7	Rear Suspension	Coupled Torsion Beam Axle
9	Front Brake	Ventilated Disc
11	Tyre Size	P21 5 60R 16
12	Wheel Size	6.55JJ x 16"
19	Wheel Base (mm)	2,905
20	Wheel thread (front/rear) (mm)	1,625/1,600
21	Overhang (front/rear) (mm)	980/1040
22	Curb Weight (kg)	1903
23	Towing Capacity Brake (kg)	2000
24	Unbrake (kg)	570

The gross weight for Naza Ria car is:

Gross Weight = Curb weight + Load of passenger (3.1)

Where;

Curb weight = Weight of body with interior, fuel tank, gearbox, and engine system

Load of passenger = 4 passenger with luggage

According to Reimpell (2002), the ideal load of passenger is 4 people with luggage and the weight data is given in table below.

Table 3.2: Weight Data

4 people with luggage	300 kg
NGV 65 liter tank system	64 kg
Curb weight	1903 kg

Gross Weight = Curb weight + Load of 4 passenger with luggage + NGV system

= 1903 + 300 + 64

= 2267 kg

3.3.2 Brake Disc Rotor

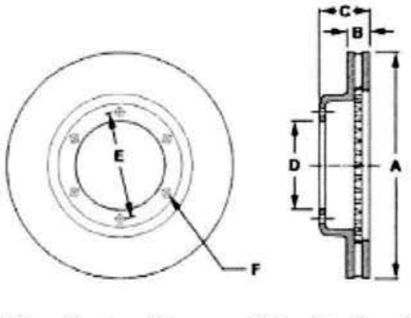

Figure 3.3: Disc Rotor (Source: Disc Brake Australia)

Based on the information from Disc Brake Australia, the actual dimension for Naza Ria Disc rotor is stated at Table 3.3 below:

Table 3.3: Naza Ria Disc Brake Rotor Dimension (Source: Disc Brake Australia)

Criteria	Value
Rotor style	HAT type
Diameter [mm]	274
Height [mm]	48
Thickness [mm]	26
Minimum Thickness [mm]	24
Inner Diameter [mm]	72
Number of Holes	5
Weight [kg]	7.2
Disc Vent	V

3.3.3 Disc Material

The most common material for disc brake rotor in vehicle is grey cast iron. Grey cast iron has many advantages compared to other type of material. The most important advantage is grey cast iron has good strength during high temperature of braking and it does not wrap after severe thermal cycling. Based on Kajela (2013) grey cast iron provide good wear resistance with high thermal diffusivity, high thermal conductivity and lower production cost compared to other disc brake rotor such as ceramic based composite and carbon composite. This research will use grey cast iron rotor as the typical material for Naza Ria rotor is grey cast iron. Based on Kajela (2013) the properties of grey cast iron are 7200 kg/m^3 for density, 125 GPa young's modulus, 0.25 poison's ratio, 54.5 w/mK thermal conductivity, 586 J/kgK specific heat and 0.2 coefficient of friction.

3.4 Brake Disc Modeling

This project use SolidWorks 2013 as a software to do modeling for brake disc rotor. The rotor is modeled based on the actual dimension of Naza Ria rotor from Disc Brake Australia. The model of the rotor is shown in Figure 3.4 below:

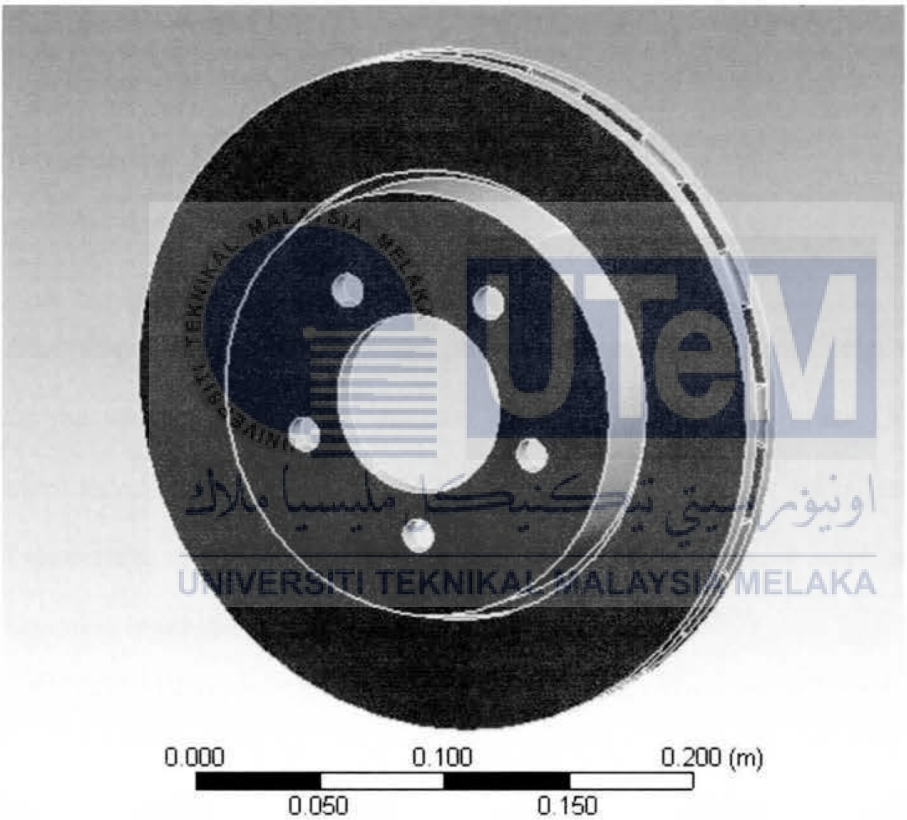

Figure 3.4: Model of Naza Ria rotor using SolidWorks

CHAPTER 4

LOAD ANALYSIS

4.1 HEAT FLUX ANALYSIS

4.1.1 Introduction

According to Kajela (2013), based on Vehicle Research & Test Center of Huanghai SUV car, the average of stopping distance with fully loaded disc brake (30°C ambient temperature) travelling at a speed of 130 km/hour (36 m/s) under the roller test conditions, required an average of 81 stopping distance with the deceleration rate 8 m/s^2 in 4.5 second. This information is adapted in this analysis.

4.1.2 Braking Energy and Braking Power

When the NGV vehicle is decelerate from the higher velocity to lower velocity, the braking energy is:

$$E_b = \left(\frac{m}{2}\right)(v_1^2 - v_2^2) + \left(\frac{I}{2}\right)(w_1^2 - w_2^2) \quad (4.1)$$

Where:

M = Vehicle mass

I = Mass moment of inertia

v_1 = Initial speed

v_2 = End speed

w_1 = Angular velocity of the rotating parts at start braking

w_2 = Angular velocity of the rotating parts at end braking

Since the NGV vehicle testing end at complete stop, v_2 and $w_1 = 0$ and the equation become:

$$E_b = \left(\frac{mv_1^2}{2}\right) + \left(\frac{Iw_1^2}{2}\right) \quad (4.2)$$

All the rotating parts are expressed relative to revolution of the wheel, V=RW and the equation become:

$$E_b = \left(\frac{m}{2}\right)\left(1 + \frac{I}{R^2 m}\right) v_1^2 \sim\sim \frac{kmv_1^2}{2} \tag{4.3}$$

Where;

K= Correction factor for the rotating mass, $1 + \frac{I}{R^2 m}$

R = tire radius

The average value of k of a vehicle is 1. The braking power is equal as the braking energy divided by time t during braking occur

$$P_b = \frac{d(E_b)}{dt} \tag{4.4}$$

Since the deceleration is constant during testing, the velocity becomes:

$$V\ (t) = V_1 - at \tag{4.5}$$

Where;

a= deceleration

t = time

Equation yield the brake power

$$P_{avg} = \text{kma}\ (V_1 - \text{at}) \tag{4.6}$$

The braking power is not constant during braking process. Time for the vehicle to stop is:

$$t_s = \frac{V_1}{a} \tag{4.7}$$

The average braking power, P_{avg} for the vehicle to stop is:

$$P_{avg} = \frac{\text{kma}V_1}{2} \tag{4.8}$$

$$= \frac{(1)(2267)(8)(36)}{2}$$

$$= 326.448 \text{ kW}$$

Based on Limpert(1999), average braking power transferred to the front axle is 0.55. Thus, the average braking power is:

$$P_{avg} = (326448)(0.55)(0.5)(0.5) \quad (4.9)$$

$$= 44886.6 \text{ W}$$

Based on Limpert (1999), the maximum brake power, P_b occurs at the onset of braking is equal to:

$$P_b = 2(P_{avg}) \quad (4.10)$$

$$= 2\ (44886.6 \text{ W})$$

$$=89\ 773.2 \text{ W}$$

4.1.3 Heat Flux Per Unit Area

Heat flux per unit area is the total braking power produced during braking process on the both surface of disc brake.

$$A_{bs} = \left(2\ \frac{\pi D_o^2}{4} - \frac{\pi D_i^2}{4}\right) \tag{4.11}$$

Where;

D_o = Outer diameter

D_i = Inner diameter

$$A_{bs} = \left(2\ \frac{\pi D_o^2}{4} - \frac{\pi D_i^2}{4}\right)$$

$$= \left(2\ \frac{\pi(0.274)^2}{4} - \frac{\pi(0.173)^2}{4}\right)$$

$$= 0.0944228\ m^2$$

Heat flux per unit area on the disc brake surface:

$$q_t = \frac{P_b}{A_{bs}} \tag{4.12}$$

$$= \frac{89773.2}{0.0944228}$$

$$= 950\ 757.656\ \mathrm{W}m^{-2}$$

According to Limpert (1972), heat flux generated on the inboard disc brake surface is 88%

$$q_{inboard} = (0.88)\ (q_t) \tag{4.13}$$

$$= (0.88)\ (950\ 757.656\ \mathrm{W}m^{-2})$$

$$= 836\ 666.7373\ \mathrm{W}m^{-2}$$

While, heat flux generated on the outboard disc brake surface is 91.4% from the heat flux generated on the inboard

$$q_{outboard} = (0.914)\ (q_{inboard}\ ) \tag{4.12}$$

$$= (0.914)\ 836\ 666.7373\ \mathrm{W}m^{-2})$$

$$= 764\ 713.3979\ \mathrm{W}m^{-2}$$

4.2 BOUNDARY CONDITION

4.2.1 Introduction

In this project, convection heat transfer are considered because the convection heat transfer of the brake rotor occurs when moving air flow through the heated surfaces of rotor. Since this project use ventilated disc brake rotor, the convective heat transfer coefficient is approximately twice as large as solid disc. The type of convective heat transfer that will be considered in this project are disc brake surface, inner ring surface, outer ring surface.

4.2.2 Convectional Heat Transfer Coefficient (Braking Surface)

The angular velocity of tire:

$$\omega_{tire} = \frac{v_{truck}}{r_{tire}} \tag{4.13}$$

$$= \frac{36}{0.333}$$

$$= 108.2\ s^{-1}$$

Where;

ω_{tire} = Angular velocity

v_{truck} = Velocity of truck

r_{tire} = Radius tire

The rotational speed of disc brake rotor

$$\omega_{rotor} = \omega_{tire} = \left(2\ \frac{\pi\ N_{rotor}}{60}\right) \tag{4.14}$$

$$N_{rotor} = \frac{108.2\ (60)}{2\pi}$$

$$= 1033.2 \text{ rpm}$$

$$\text{Re} = \frac{(\omega_{rotor})\ (r_o^2)}{v_a} \tag{4.15}$$

$$= \frac{(108.2)(0.137^2)}{1.85\times10^{-5}}$$

$$= 1.097\times10^5$$

Where;

ω_{rotor} = Angular velocity

r_o = Outer rotor radius

V = Kinematics Viscosity

Laminar flow of the braking surface is associated with the convection of heat transfer coefficient. Hence, the convectional heat transfer coefficient can be calculated:

$$h_c = 3.974 \left(\frac{k_a}{D_o}\right) Re^{0.55} \tag{4.16}$$

$$= 3.974 \left(\frac{0.02624}{0.274}\right) (1.097 \times 10^5)^{0.55}$$

$$= 225.272 \text{ W}m^{-2}\, K^{-1}$$

Where;

k_a = Thermal conductivity of air

D_o = Outer diameter of disc rotor

Re = Reynolds Number

4.2.3 Convectional Heat Transfer Coefficient (Upper Inner Ring Surface)

$$\text{Re} = \frac{(\omega_{rotor})\,(r_i^2)}{v_a} \tag{4.17}$$

$$= \frac{(108.2)(0.079^2)}{1.85\times10^{-5}}$$

$$= 36\ 501.416$$

Where;

ω_{rotor} = Angular velocity

r_i = Inner rotor radius

V = Kinematics Viscosity

The Nusselt's number for the inner ring:

$$Nu_D = 0.133\ Re^{\frac{2}{3}}\ \text{Pr}^{\frac{1}{3}} \tag{4.18}$$

$$= (0.133)\ (36\ 501.416)^{\frac{2}{3}}\ (0.708)^{\frac{1}{3}}$$

$$= 130.4375$$

Where;

Re = Reynolds Number

Pr = Prandtl Number = 0.708

Laminar flow of the braking surface is associated with the convection of heat transfer coefficient. Hence, the convectional heat transfer coefficient can be calculated:

$$H_c = \left(\frac{k_a}{D_i}\right) Nu_D \tag{4.19}$$

$$= \left(\frac{0.02624}{0.158m}\right)(130.4375)$$

$$= 21.66 \mathrm{W} m^{-2}\, K^{-1}$$

Where;

k_a = Thermal conductivity of air

D_i = Inner diameter of disc rotor

Nu_D = Nusselt Number

4.2.4 Convectional Heat Transfer Coefficient (Upper Outer Ring Surface)

$$\text{Re} = \frac{(\omega_{rotor})(r_o^2)}{v_a} \tag{4.20}$$

$$= \frac{(108.2)(0.0865^2)}{1.85\times10^{-5}}$$

$$= 43\ 761$$

Where;

ω_{rotor} = Angular velocity

r_o = Outer rotor radius

V = Kinematics Viscosity

The Nusselt's number for the outer ring:

$$Nu_D = 0.133\ Re^{\frac{2}{3}}\ \text{Pr}^{\frac{1}{3}} \tag{4.21}$$

$$= (0.133)\ (43\ 761)^{\frac{2}{3}}\ (0.708)^{\frac{1}{3}}$$

$$= 147.2$$

Where;

Re = Reynolds Number

Pr = Prandtl Number = 0.708

Laminar flow of the braking surface is associated with the convection of heat transfer coefficient. Hence, the convectional heat transfer coefficient can be calculated:

$$H_c = \left(\frac{k_a}{D_o}\right) Nu_D \tag{4.22}$$

$$=\left(\frac{0.02624}{0.173m}\right)(147.2)$$

$$= 22.327 \text{ W}m^{-2}\, K^{-1}$$

Where;

k_a= Thermal conductivity of air

D_i = Outer diameter of disc rotor

Nu_D = Nusselt Number

4.2.5 Convectional Heat Transfer Coefficient (Outer Ring Surface)

$$\mathrm{Re} = \frac{(\omega_{rotor})\,(r_o^2)}{v_a} \tag{4.23}$$

$$= \frac{(108.2)(0.137^2)}{1.85\times10^{-5}}$$

$$= 1.097\times10^5$$

Where;

ω_{rotor} = Angular velocity

r_o = Outer rotor radius

V = Kinematics Viscosity

The Nusselt's number for the outer ring:

$$Nu_D = 0.133\ Re^{\frac{2}{3}}\ \mathrm{Pr}^{\frac{1}{3}} \tag{4.24}$$

$$= (0.133)\ (1.097 \times 10^5)^{\frac{2}{3}}\ (0.708)^{\frac{1}{3}}$$

$$= 271.765$$

Where;

Re = Reynolds Number

Pr = Prandtl Number = 0.708

Laminar flow of the braking surface is associated with the convection of heat transfer coefficient. Hence, the convectional heat transfer coefficient can be calculated:

$$H_c = \left(\frac{k_a}{D_o}\right) Nu_D \tag{4.25}$$

$$= \left(\frac{0.02624}{0.274m}\right)(271.765)$$

$$= 26.026\ \mathrm{W}m^{-2}\ K^{-1}$$

Where;

k_a= Thermal conductivity of air

D_i = Outer diameter of disc rotor

Nu_D = Nusselt Number

4.3 ANALYSIS SETUP

4.3.1 Steady State Analysis Setup

The setups for steady state analysis are stated as below:

I. Steady state thermal analysis system is chosen

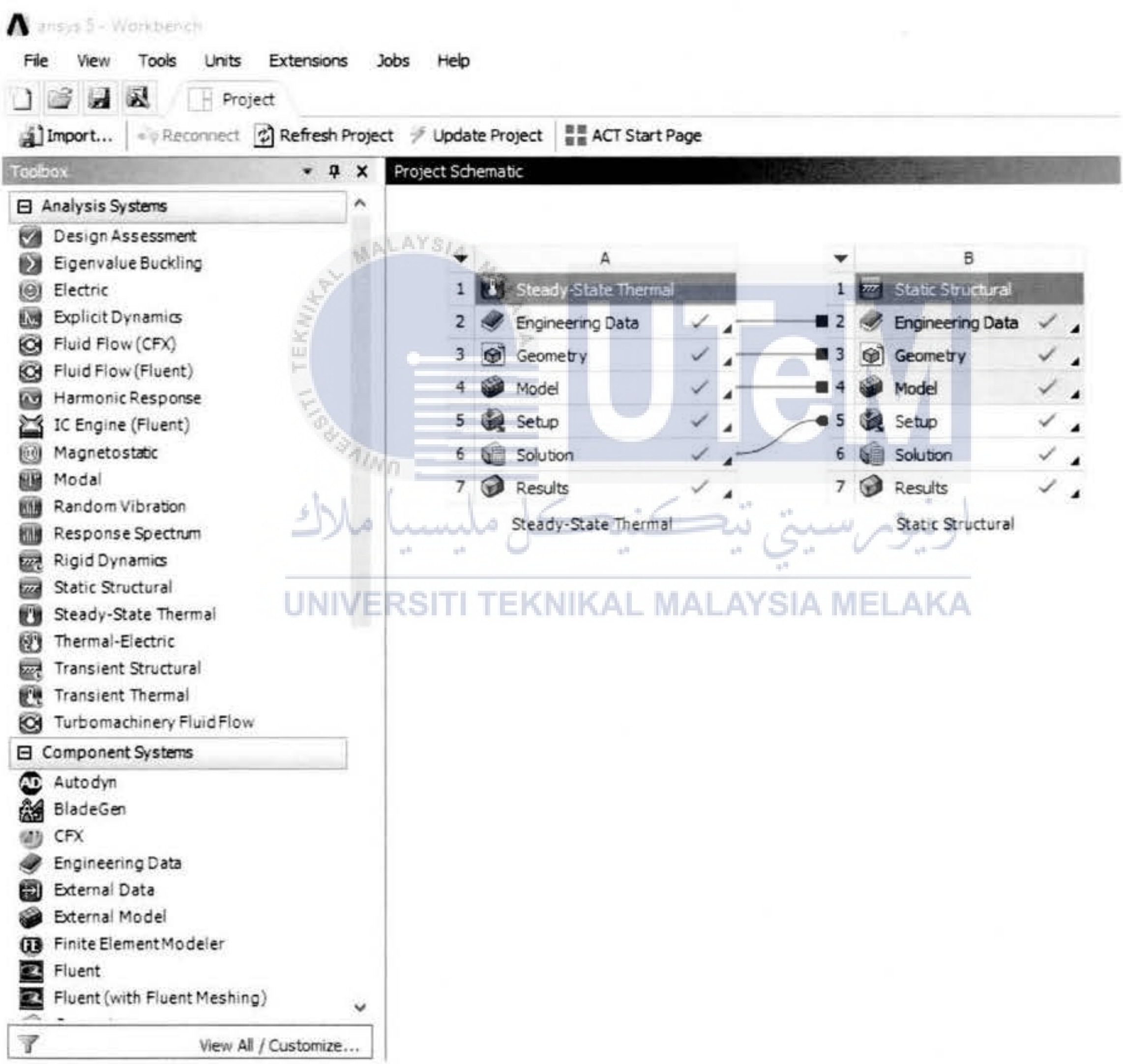

Figure 4.1: Steady state thermal analysis

II. Gray cast iron is selected as the material for this analysis

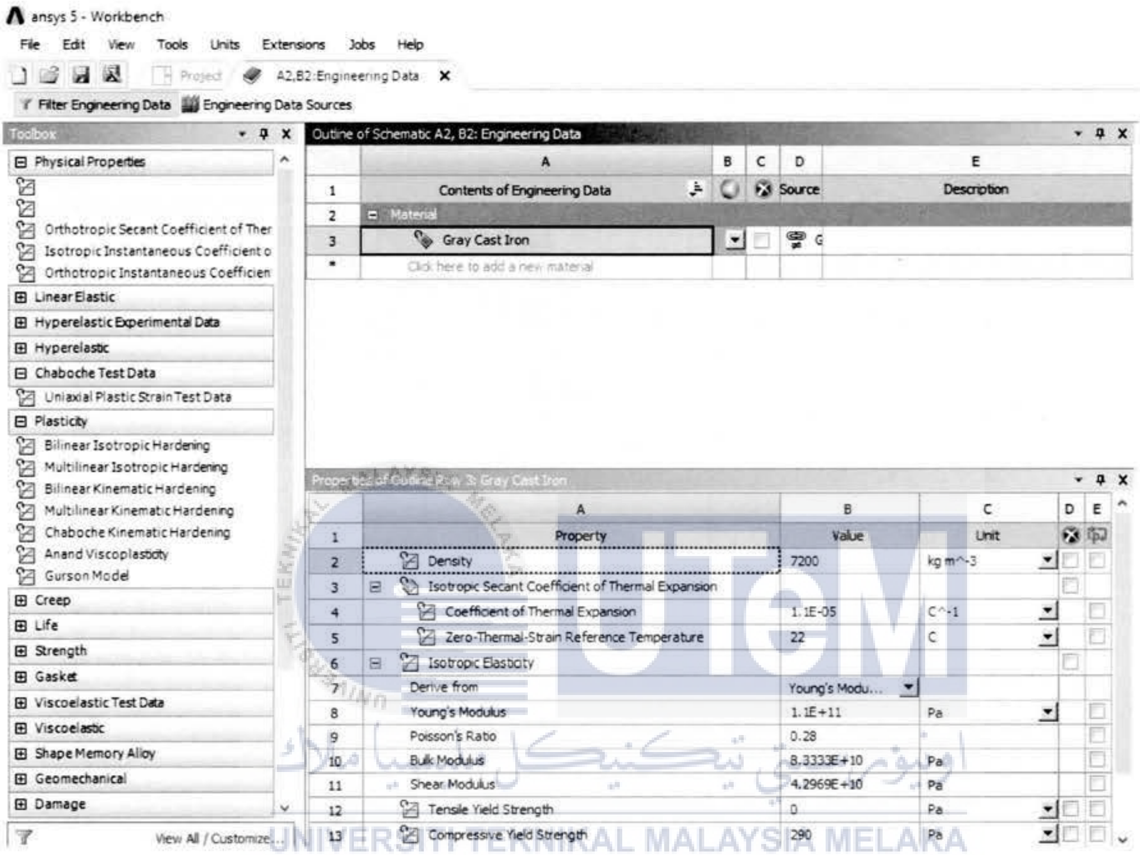

Figure 4.2: Material selection

III. The disc brake design is imported from SolidWorks to ANSYS as shown in figure below

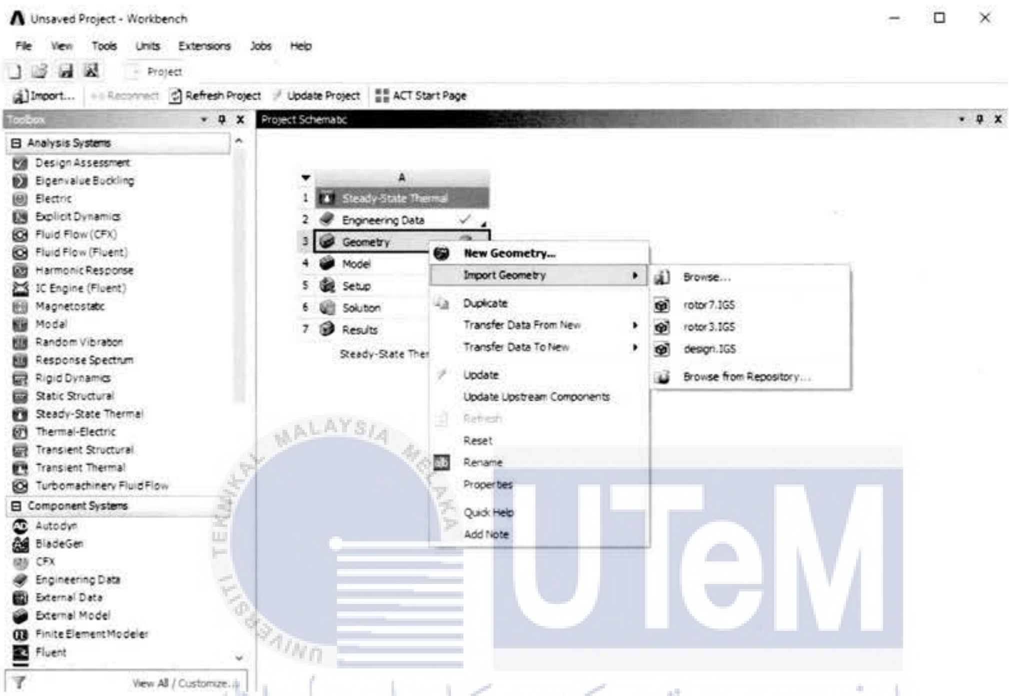

Figure 4.3: Importing geometry

IV. The mesh is generated not exceeding the allowable educational license. The allowable license for nodes and element is 30,000. Hence the mesh is generated at 29,149 nodes and 17,541 elements.

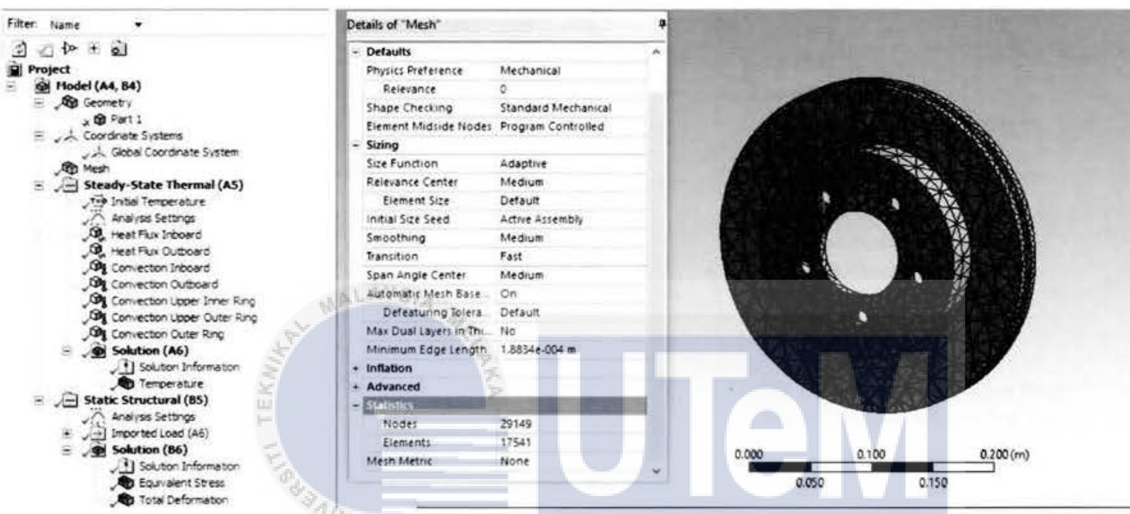

Figure 4.4: Meshing

V. The initial temperature is setup to be 303.15 K and Analysis Setting is set up based on the figure below:

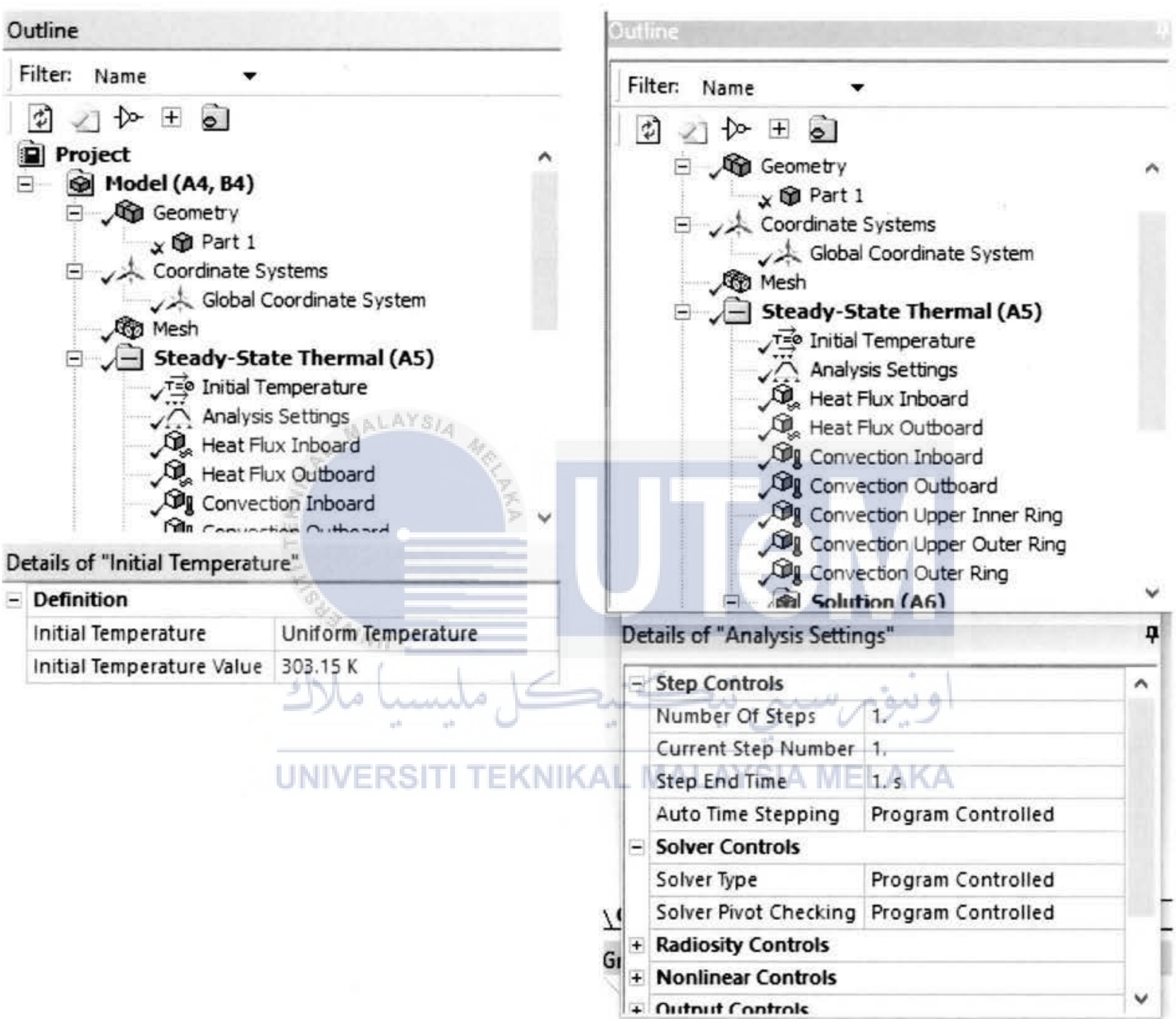

Figure 4.5: Initial temperature and analysis setting

VI. The heat flux and convection heat transfer coefficient is taken from the calculation in chapter 4.1 and 4.2. Table 4.1 below shows the value for each heat flux and convection heat transfer coefficient.

Table 4.1: Heat Flux and Convection Heat Transfer Coefficient

Surfaces	Heat Flux
Inboard	836666.7373 W/m^2
Outboard	764713.3979 W/m^2
Surfaces	Convection Heat Transfer Coefficient
Inboard	225.272 W/m^2K
Outboard	225.272 W/m^2K
Upper inner ring	21.66 W/m^2K
Upper outer ring	22.327 W/m^2K
Outer ring	26.026 W/m^2K

VII. The result for temperature distribution are solved

4.3.2 Transient Thermal Analysis Setup

The setup for transient thermal analysis is stated below:

I. Step 1 until step 4 in steady state analysis is repeated in transient analysis

II. The initial temperature is setup to be 303.15 K and Analysis Setting is set up based on the figure below:

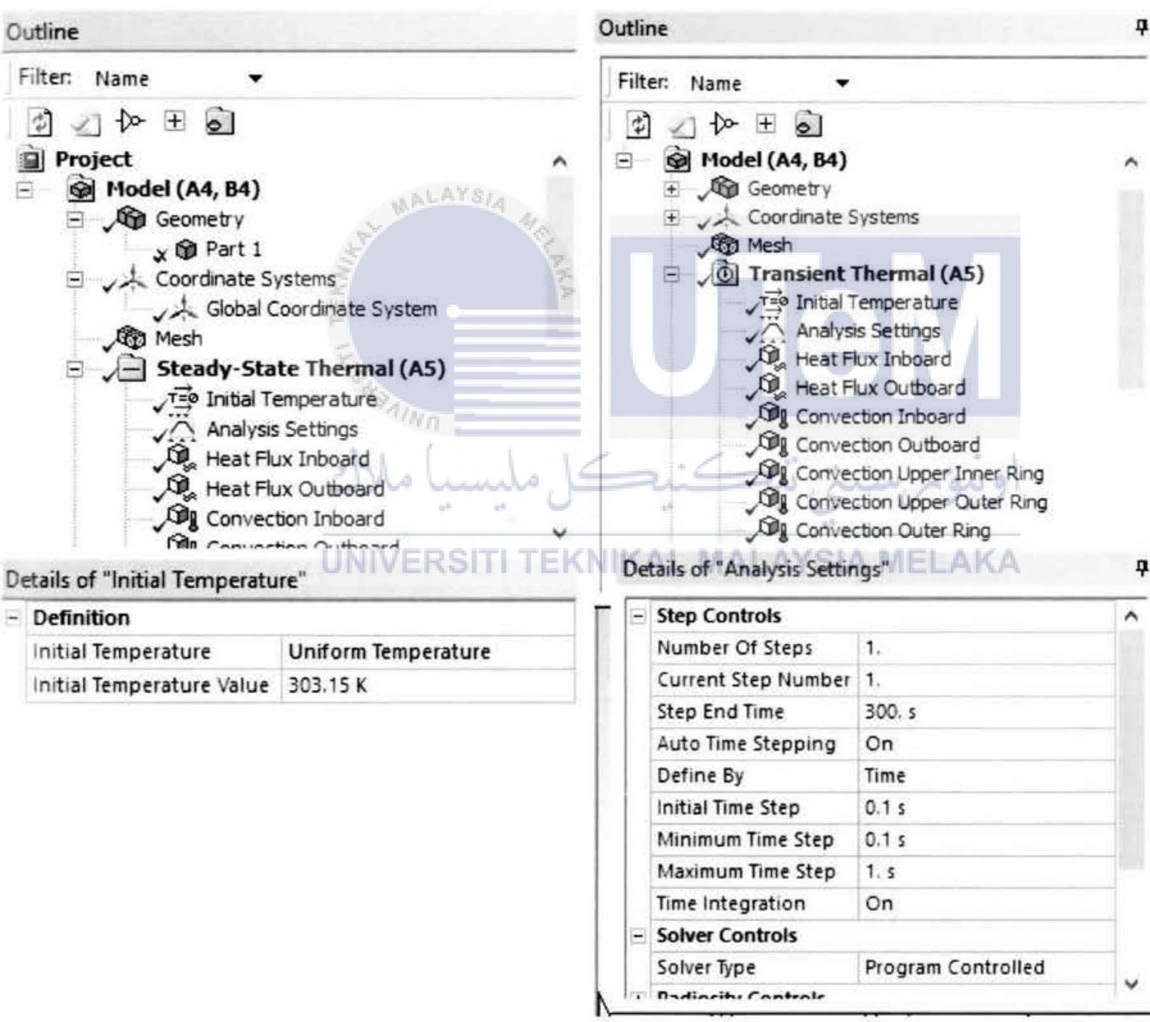

Figure 4.6: Initial temperature and analysis setting

III. The heat flux and convection heat transfer coefficient is taken from the calculation in chapter 4.1 and 4.2 and it is applied to disc brake. Based on Xiaolin and Yijun (2015), the Magnitude in Heat Flux Inboard/Outboard is set to be tabular data since transient thermal involve time. The convection heat transfer coefficient for inboard, outboard, upper inner ring, upper outer ring an outer ring is similar to the steady state. Table 4.2 below shows the value for each heat flux.

Table 4.2: Heat flux inboard and outboard

Load	Braking	Heat Flux Inboard		Heat Flux Outboard	
		Time (s)	W/m^2	Time (s)	W/m^2
1	Heating	0	836666.7373	0	764713.3979
		4.5	836666.7373	4.5	764713.3979
	Cooling	4.6	0	4.6	0
		30	0	30	0
2	Heating	30.1	836666.7373	30.1	764713.3979
		34.5	836666.7373	34.5	764713.3979
	Cooling	34.6	0	34.6	0
		60	0	60	0
3	Heating	60.1	836666.7373	60.1	764713.3979
		64.5	836666.7373	64.5	764713.3979
	Cooling	64.6	0	64.6	0
		90	0	90	0
4	Heating	90.1	836666.7373	90.1	764713.3979
		94.5	836666.7373	94.5	764713.3979
	Cooling	94.6	0	94.6	0
		120	0	120	0
5	Heating	120.1	836666.7373	120.1	764713.3979
		124.5	836666.7373	124.5	764713.3979

	Cooling	124.6	0	124.6	0
		150	0	150	0
6	Heating	150.1	836666.7373	150.1	764713.3979
		154.5	836666.7373	154.5	764713.3979
	Cooling	154.6	0	154.6	0
		180	0	180	0
7	Heating	180.1	836666.7373	180.1	764713.3979
		184.5	836666.7373	184.5	764713.3979
	Cooling	184.6	0	184.6	0
		210	0	210	0
8	Heating	210.1	836666.7373	210.1	764713.3979
		214.5	836666.7373	214.5	764713.3979
	Cooling	214.6	0	214.6	0
		240	0	240	0
9	Heating	240.1	836666.7373	240.1	764713.3979
		244.5	836666.7373	244.5	764713.3979
	Cooling	244.6	0	244.6	0
		270	0	270	0
10	Heating	270.1	836666.7373	270.1	764713.3979
		274.5	836666.7373	274.5	764713.3979
	Cooling	274.6	0	274.6	0
		300	0	300	0

IV. The results for temperature distribution are solved.

4.3.2 Transient Structural Analysis Setup

The setup for transient analysis is stated below:

I. Transient thermal analysis is linked to Transient Structural

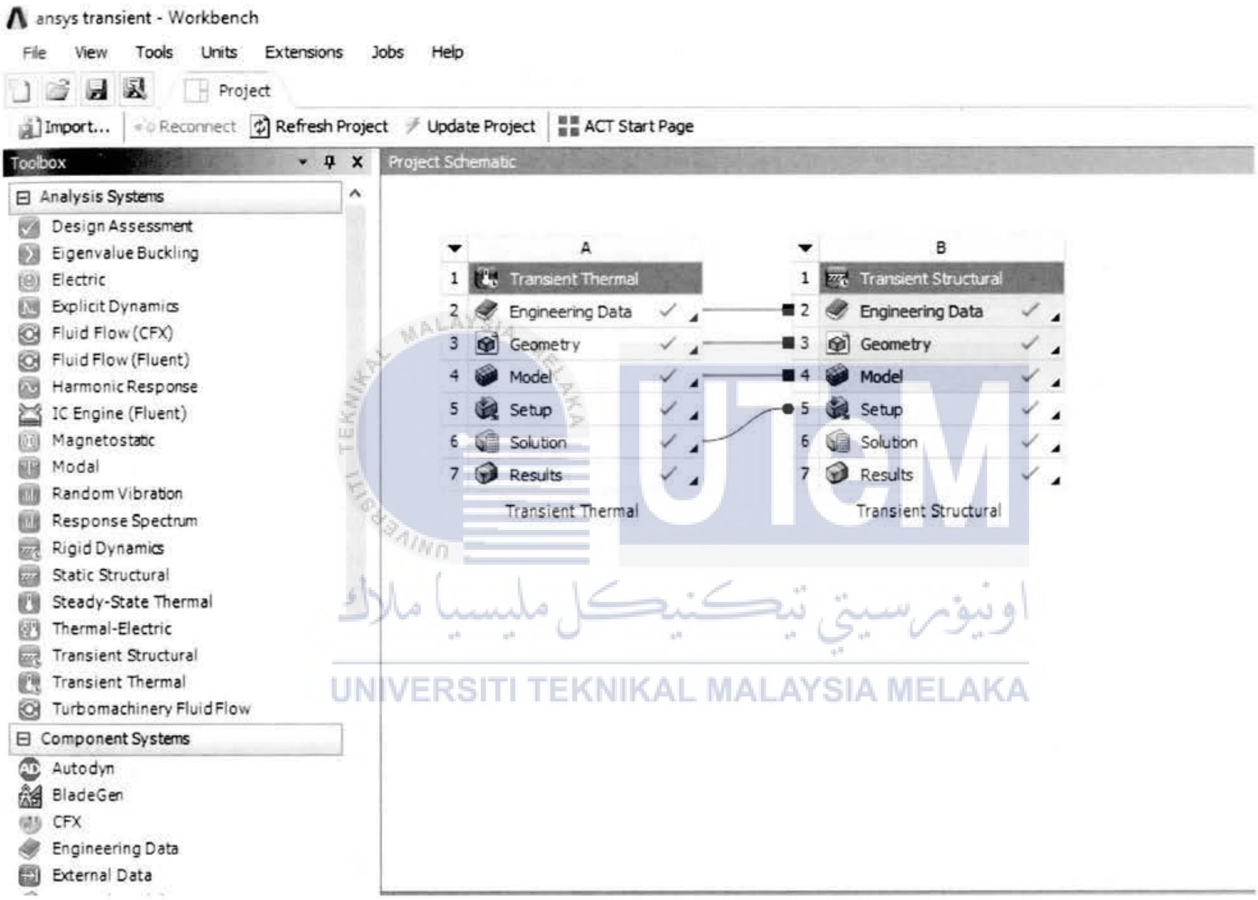

Figure 4.7: Transient Structural

II. The initial temperature is setup to be 303.15 K and Analysis Setting is set up based on the figure below:

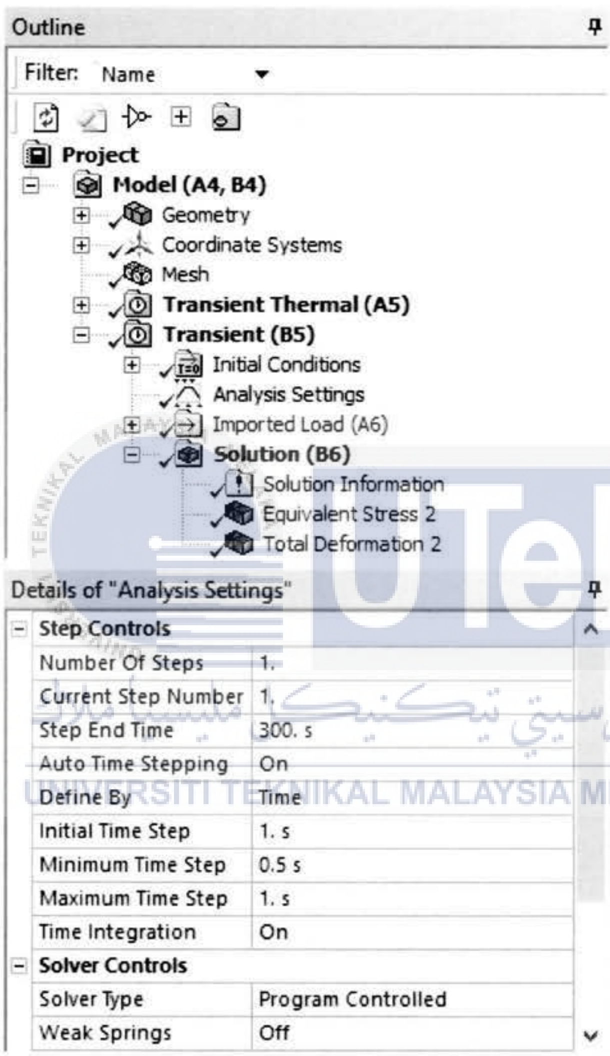

Figure 4.8: Analysis setting

III. Imported body temperature is obtained from transient thermal analysis. The source time and sources environment is set as Figure 4.9 below:

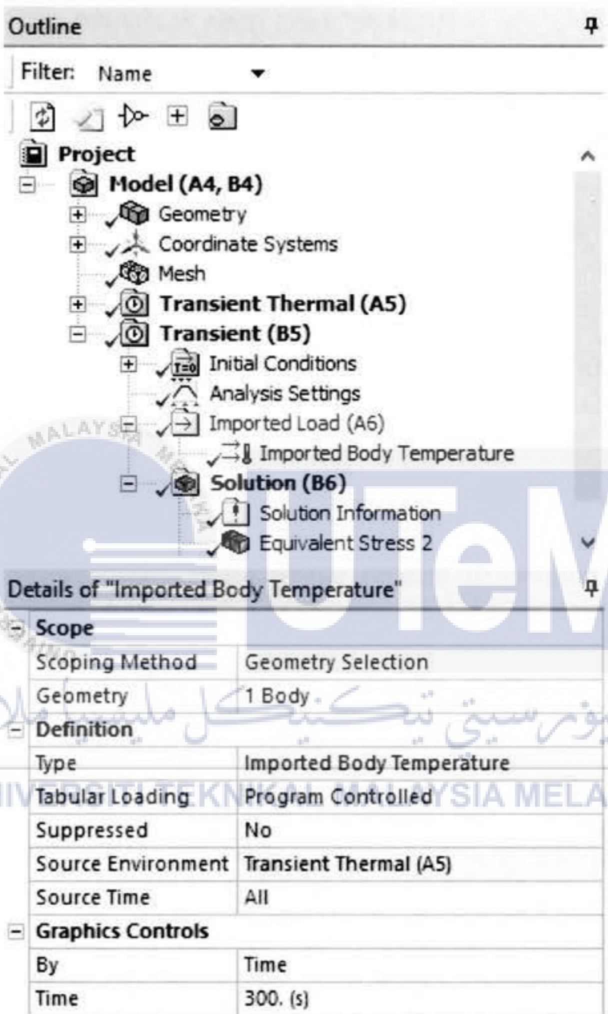

Figure 4.9: Imported body temperature

IV. The results for equivalent (von-Mises) stress and total deformation are solved.

CHAPTER 5

RESULT AND DISCUSSION

5.1 STEADY STATE THERMAL & STRUCTURAL ANALYSIS

In this steady state condition, brake load is applied by means of heat flux on both inboard and outboard surfaces of disc brake continuously and the heat storage effect over period of time is ignored. Forced convection was applied on outboard, inboard, upper inner ring, upper outer ring and outer ring. Cooling process of braking surface is ignored because heat flux applied is constant.

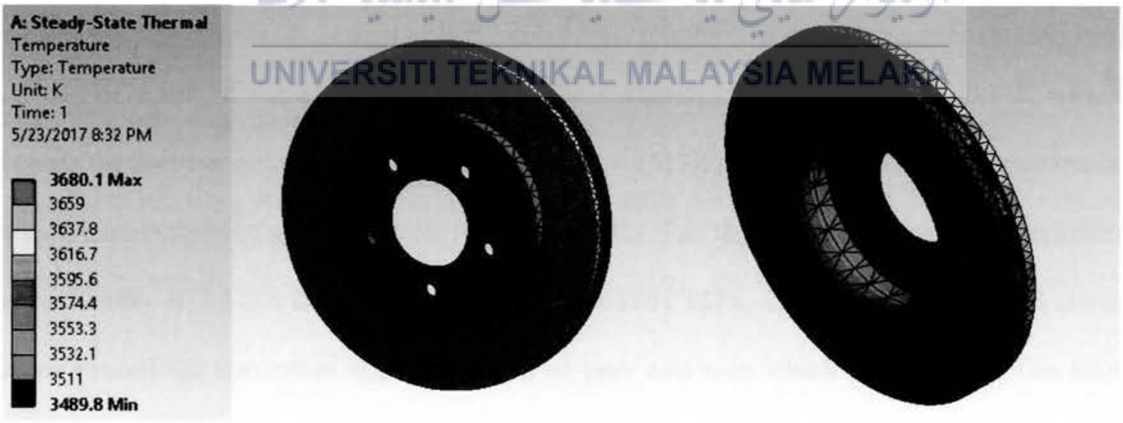

Figure 5.1: Steady state temperature analysis on disc brake

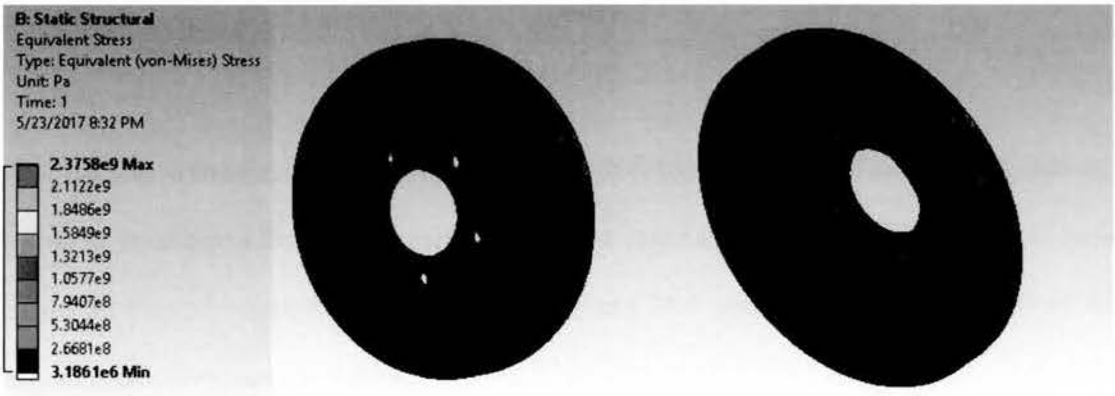

Figure 5.2: Steady state stress analysis on disc brake

Figure 5.3: Total deformation

The result of the analysis shows that the maximum temperature is 3680.1 K which exceeds the melting point of gray cast iron (1403K – 1513K) and it also exceed the maximum service temperature of gray cast iron (623K – 723K). For the stress analysis, the maximum thermal stress is 2.3758 GPa while minimum is 3.1861 MPa. The maximum thermal stress values exceed the maximum tensile strength of gray cast iron which is 240 MPa. The total deformation in steady state is 1.6085×10^5 m. Due to the high temperature and stress recorded in steady state analysis, it is difficult to the researcher to predict the temperature distribution and disc brake stress behavior. Therefore, transient thermal is applied.

5.2 TRANSIENT THERMAL ANALYSIS

In this transient thermal analysis, ten load cycles with total time 300 seconds are applied to investigate the cooling performance and thermal stress of disc brake. Each cycle contain 30 second where 4.5 second of braking and 25.5 second of idle. The result of the analysis is tabulated in table below:

Table 5.1: Result of transient thermal analysis

Cycle	Time (s)	Maximum Temperature (K)
0	0	303.15
1	4.5	439.79
	30	396.32
2	34.5	508.96
	60	438.74
3	64.5	531.63
	90	474.12
4	94.5	582.93
	120	487.09
5	124.5	593.34
	150	504.62
6	154.5	607.94
	180	518.95

7	184.5	620.31
	210	530.5
8	214.5	630.87
	240	539.8
9	244.5	640
	270	547.28
10	274.5	647.97
	300	550.41

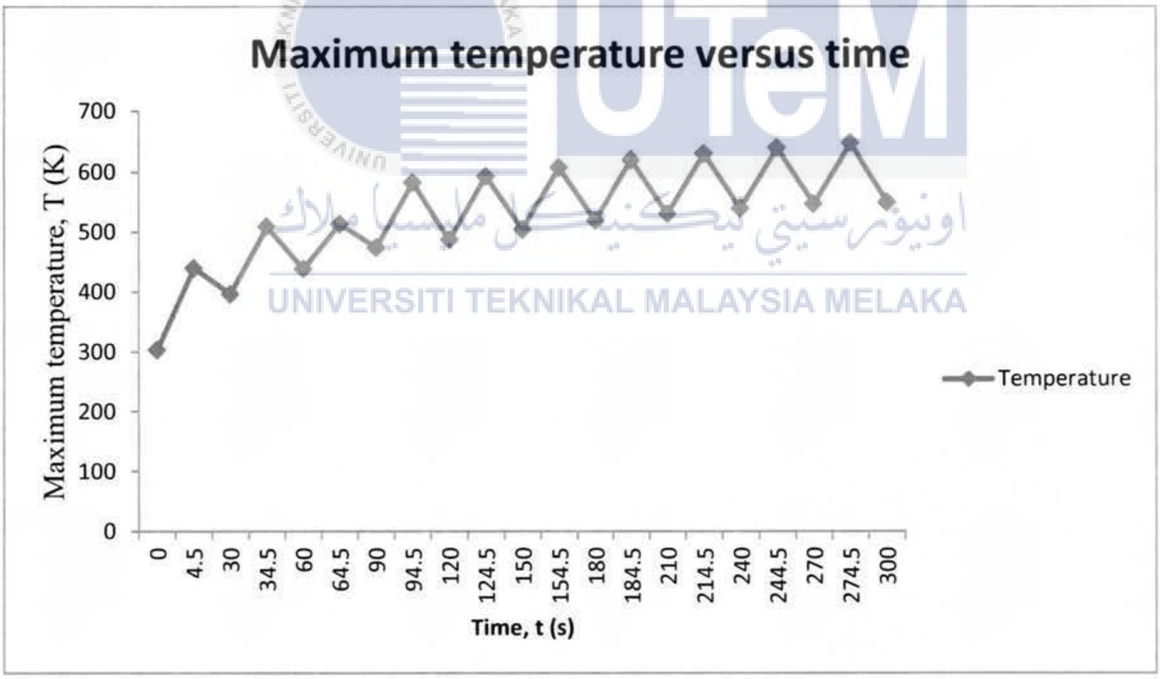

Figure 5.4: Maximum temperature versus time graph

At 1^{st} cycle, from 0 second to 4.5 seconds of braking, the temperature rise from 303.15 K to 439.79K. Later, 25.5 seconds of idling are applied and the temperature drops to 396.32 seconds. The drop in temperature is due to the forced convection through the outboard and inboard surface, upper inner ring, upper outer ring and outer ring surfaces. Figure 5.5 below shows the maximum temperature recorded at 1^{st} cycle where its temperature is 439.79 K while Figure 5.6 shows the temperature distribution after the cooling process.

Figure 5.5: Temperature distribution at 4.5 seconds

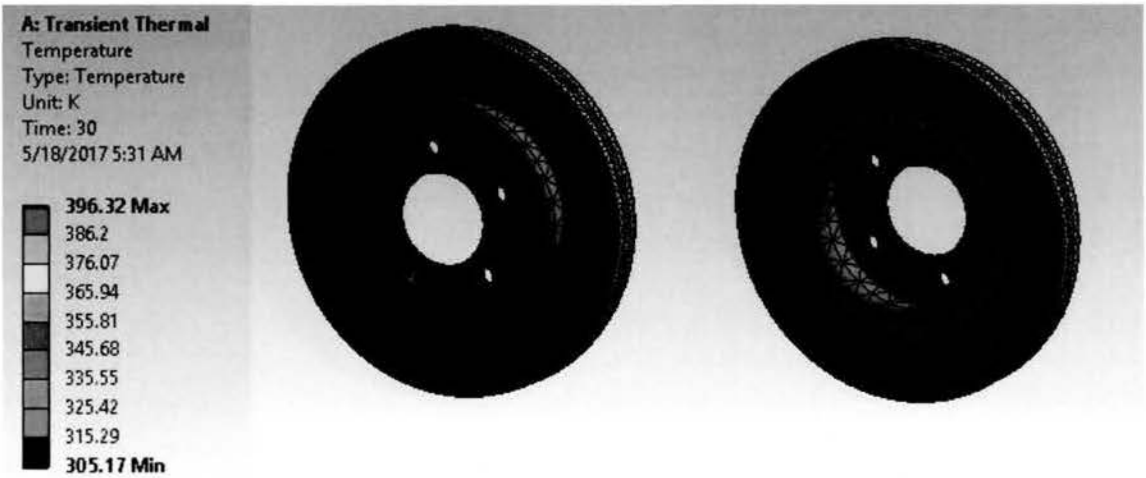

Figure 5.6: Temperature distribution at 30 seconds

At the 5^{th} cycle which is at the middle of load cycle (124.5 seconds), the maximum temperature is at 593.34 K and its minimum temperature is at 362.92 K. Figure 5.7 below shows the maximum temperature recorded at 5^{th} cycle where its temperature is 593.34 K while Figure 5.8 shows the temperature distribution after the cooling process where the maximum temperature during cooling process is 504.62K.

Figure 5.7: Temperature distribution at 124.5 seconds

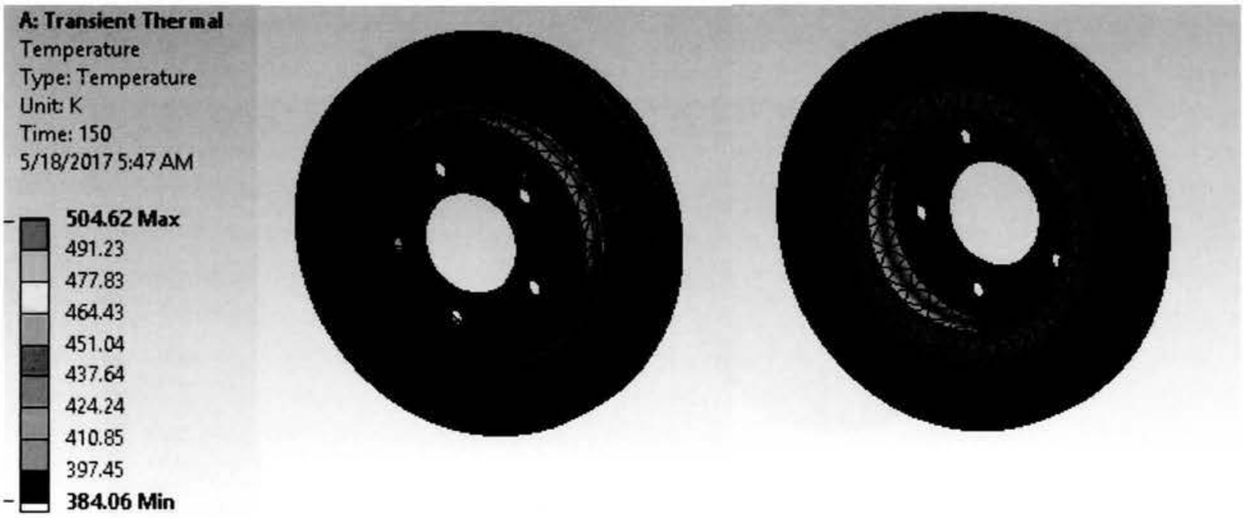

Figure 5.8: Temperature distribution at 150 seconds

At 274.5 seconds or 10^{th} load cycle, the highest maximum temperature of disc brake is recorded. The highest maximum temperature is recorded at the inboard and outboard of disc brake where its value is 647.97 K while its minimum value is recorded at the hub of disc brake where its value is 486.69 K. Figure 5.9 below shows the maximum temperature recorded at 274.5 seconds where its temperature is 647.97 K. At 300 seconds or at the cooling process of 10^{th} load cycle, the maximum temperature recorded is 550.41 K while the minimum temperature recorded is 481.24K. Figure 5.10 shows the temperature distribution after the cooling process.

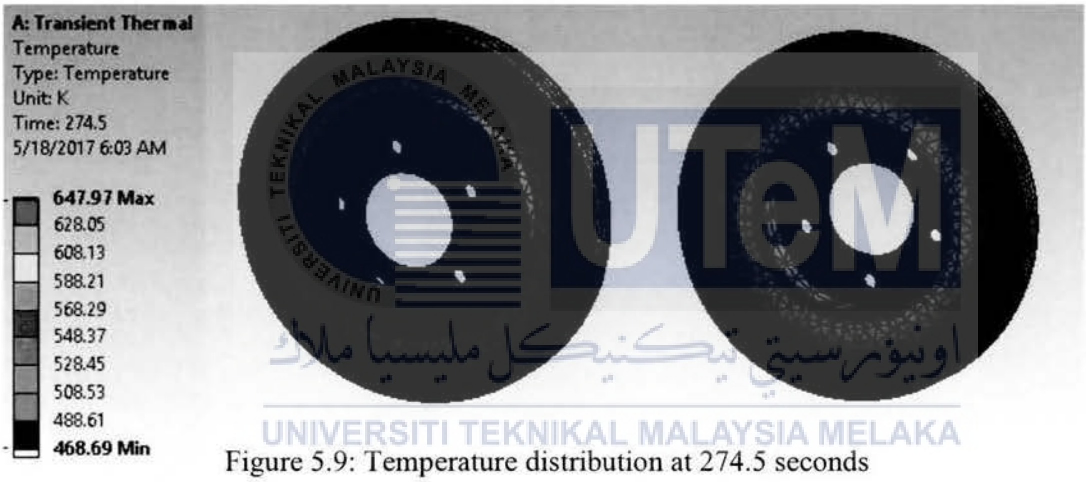

Figure 5.9: Temperature distribution at 274.5 seconds

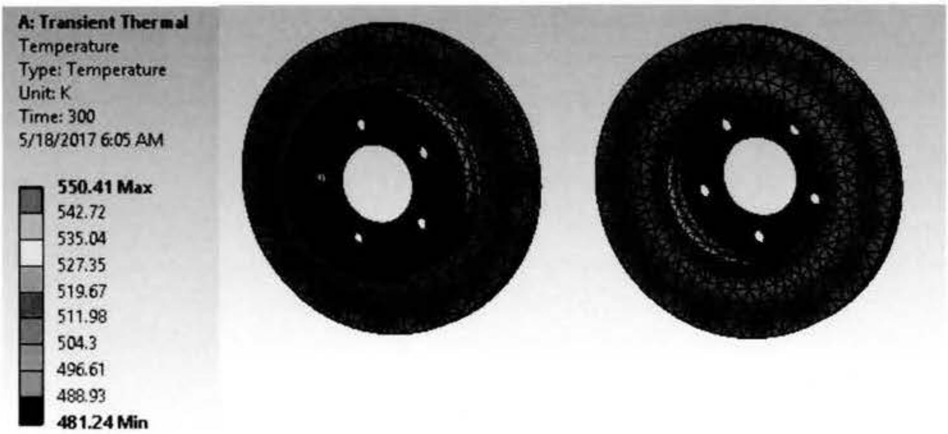

Figure 5.10: Temperature distribution at 300 seconds

In conclusion, since the highest maximum temperature recorded at the inboard and outboard of disc brake are 647.97 K and it is below the maximum temperature service temperature of gray cast iron which is 723.15 K, thus the temperature is acceptable. But, the actual temperature could be less or more precise than the simulation due to the limitation of boundry condition and number of mesh. Besides, the number of heat flux and heat convection also plays an important roles in maximum temperature recorded as the higher the heat flux will produce higher temperature. Since this research is using NGV vehicle where its gross weight is higher than regular vehicle, the heat flux is higher. Thus it should have higher maximum temperature compared to regular vehicle.

5.3 TRANSIENT STRUCTURAL ANALYSIS

5.3.1 Deformation

In this deformation analysis, ten load cycles with total time 300 seconds are applied. Each cycle contain 30 second where 4.5 second of braking and 25.5 second of idle. The body temperature of the disc brake is obtained from transient thermal analysis. The result of the total deformation is tabulated in table below:

Table 5.2: Result of total deformation analysis

Cycle	Time (s)	Total deformation (m)
0	0	0
1	4.5	0.00015843
	30	0.00014496
2	34.5	0.00024374
	60	0.00021796
3	64.5	0.00029398
	90	0.00026393
4	94.5	0.00035681
	120	0.00031261
5	124.5	0.00038834
	150	0.0003519
6	154.5	0.00042637
	180	0.00038676

7	184.5	0.00046019
	210	0.0004207
8	214.5	0.00049383
	240	0.00045371
9	244.5	0.00052518
	270	0.00048852
10	274.5	0.00055918
	300	0.000477

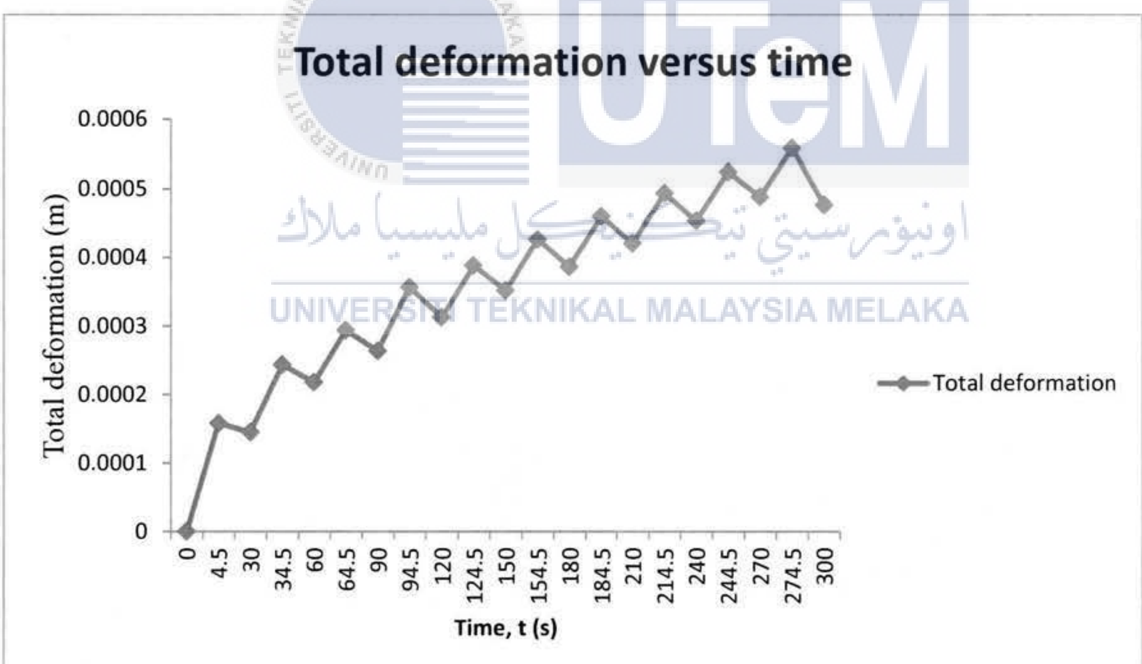

Figure 5.11: Total deformation versus time graph

At 1^{st} cycle, from 0 second to 4.5 seconds of braking, the maximum total deformation is 0.00015843 meter. Later, 25.5 seconds of idling are applied and the total deformation drops to 0.00014496 meter. This drop is due to the cooling effect from forced convection. Figure 5.12 below shows the maximum total deformation recorded at 1^{st} cycle while Figure 5.13 shows the total deformation after the cooling process.

Figure 5.12: Total deformation at 4.5 seconds

Figure 5.13: Total deformation at 30 seconds

At the 5^{th} cycle which is at the middle of load cycle (124.5 seconds), the maximum total deformation is 0.00038834 meter and its minimum total deformation is at 0.0003519 meter. Figure 5.14 below shows the total deformation recorded at 5^{th} cycle while Figure 5.15 shows the total deformation after the cooling process.

Figure 5.14: Total deformation at 124.5 seconds

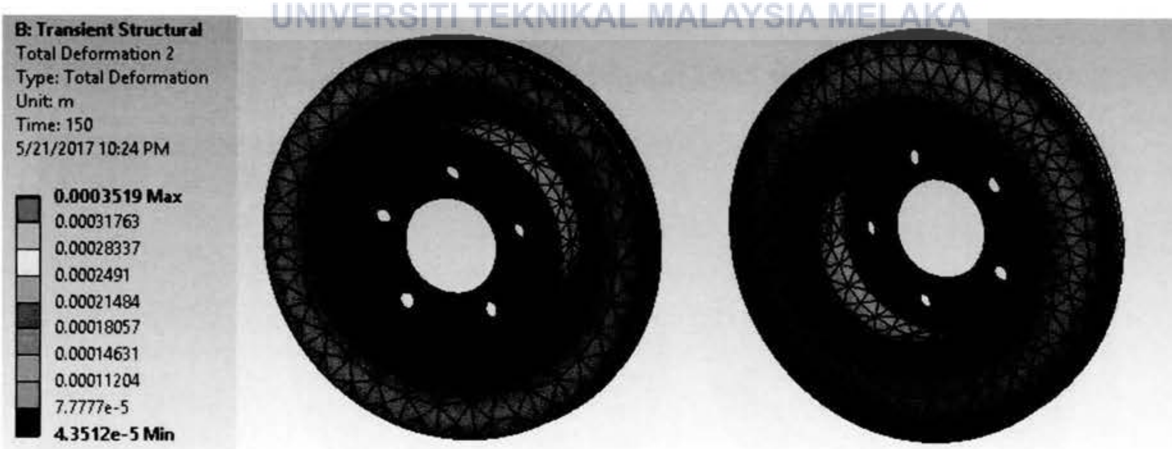

Figure 5.15: Total deformation at 150 seconds

At 274.5 seconds or 10^{th} load cycle, the highest maximum total deformation of disc brake is recorded. The highest maximum total deformation is 0.00055918 meter while its minimum value is 0.000052594 meter. At 300 seconds or at the cooling process of 10^{th} load cycle, the maximum total deformation recorded is 0.000477 meter while the minimum total deformation recorded is 0.00012085 meter. Figure 5.16 below shows the maximum total deformation recorded at 274.5 seconds while Figure 5.17 shows the total deformation after the cooling process.

Figure 5.16: Total deformation at 274.5 seconds

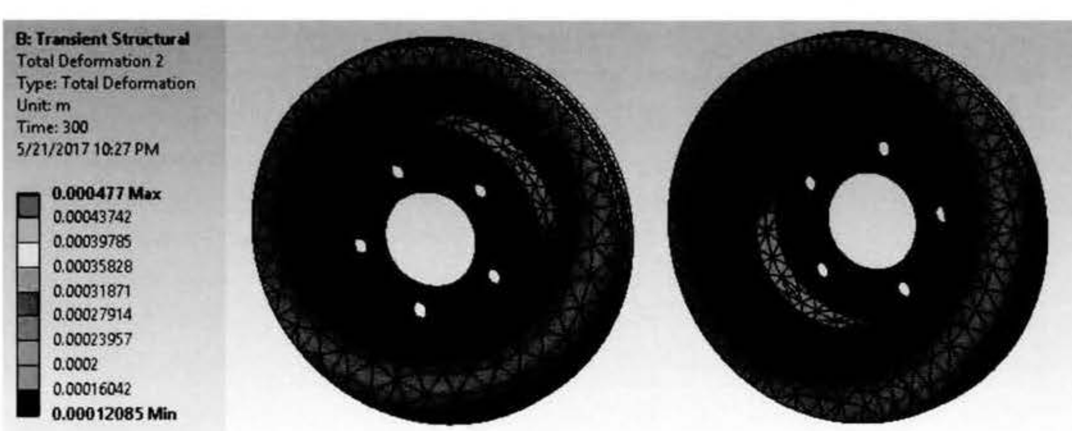

Figure 5.17: Total deformation at 300 seconds

In conclusion, as the temperature increase throughout the cycles, the disc brake begins to deform gradually until it reach highest maximum total deformation of 0.55918 mm at 274.5 seconds. The total deformation shows the increasing in deformation at heating (braking) intervals and suddenly drops during cooling (unbrake) intervals. This shows that, the temperature is directly proportional with the total deformation. The total deformation of disc brake during this analysis can still be considered as small compared to the other research. Although the value of total deformation is small, it cannot be negligible as it may lead to disc coning where the disc brake shape could turn to cone or worst such as buckling. Since this research is using NGV vehicle, the total deformation should be higher than regular vehicle as the maximum temperature is higher.

5.3.2 Equivalent (Von Mises) Stress

In this equivalent (von-Mises) stress analysis, ten load cycles with total time 300 seconds are applied. Each cycle contain 30 second where 4.5 second of braking and 25.5 second of idle. The body temperature of the disc brake is obtained from transient thermal analysis. The result of the equivalent (von-Mises) stress is tabulated in table below:

Table 5.3: Result of equivalent (von-Mises) stress analysis

Cycle	Time (s)	Equivalent (von-Mises) stress (MPa)
0	0	0
1	4.5	113.69
	30	78.877
2	34.5	92.133
	60	87.003
3	64.5	98.103
	90	90.887
4	94.5	111.47
	120	96.833
5	124.5	112.65
	150	92.058
6	154.5	107.28
	180	84.762
7	184.5	99.805

	210	81.296
8	214.5	91.948
	240	78.829
9	244.5	84.393
	270	76.588
10	274.5	79.367
	300	48.308

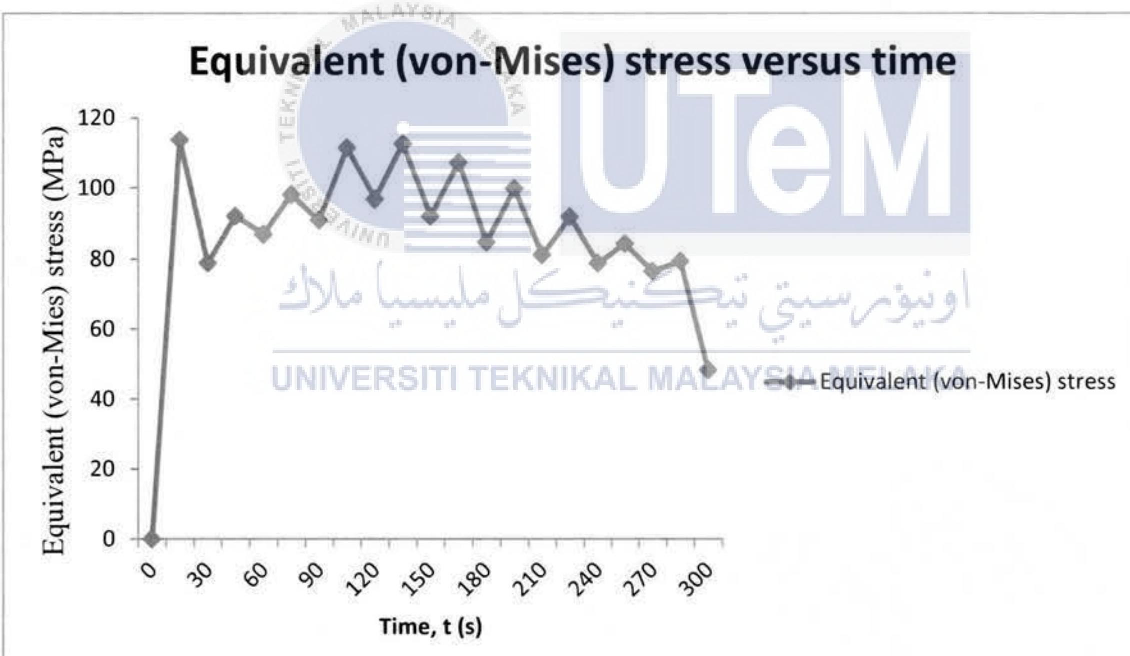

Figure 5.18: Equivalent (von-Mises) stress versus time graph

For the 1^{st} load cycle, the highest maximum equivalent (von-Mises) stress recorded during the 4.5 second braking interval. The highest equivalent (von-Mises) stress is 113.69 MPa while its minimum value is 0.49489 MPa. After 25.5 seconds of idle interval, the value for maximum equivalent (von-Mises) stress is reduced to 78.877 MPa. Although these values for equivalent (von-Mises) stress are high, it is still below the maximum tensile strength of gray cast iron which is 240 MPa. Figure 5.19 below shows the maximum equivalent (von-Mises) stress recorded at 4.5 seconds while Figure 5.20 shows the maximum equivalent (von-Mises) stress after the cooling process.

Figure 5.19: Equivalent (von-Mises) stress at 4.5 seconds

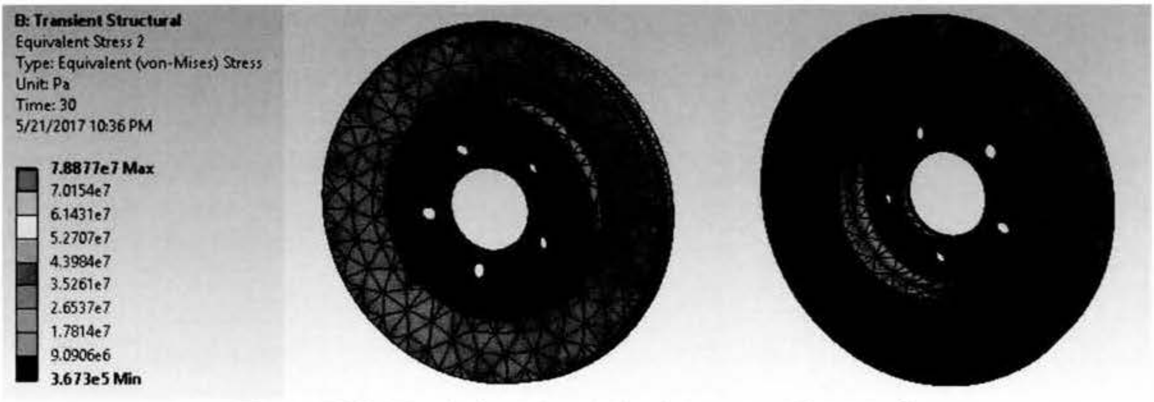

Figure 5.20: Equivalent (von-Mises) stress at 30 seconds

At the 5^{th} cycle which is at the middle of load cycle (124.5 seconds), the maximum equivalent (von-Mises) stress is 112.065 MPa and its minimum equivalent (von-Mises) stress is 0.9794 MPa. Figure 5.21 below shows the maximum equivalent (von-Mises) stress recorded at 124.5 seconds while Figure 5.22 shows the maximum equivalent (von-Mises) stress on after the cooling process.

Figure 5.21: Equivalent (von-Mises) stress at 124.5 seconds

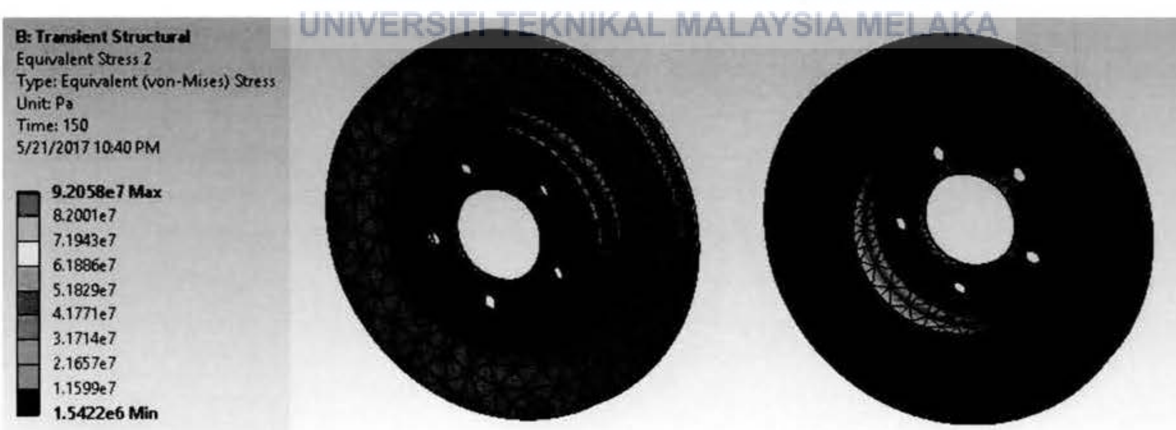

Figure 5.22: Equivalent (von-Mises) stress at 150 seconds

At the 10^{th} cycle which is at the end of load cycle (274.5 seconds), the maximum equivalent (von-Mises) stress is 79.367 MPa and its minimum equivalent (von-Mises) stress is at 1.3051 MPa. Figure 5.23 below shows the maximum equivalent (von-Mises) stress recorded at 274.5 seconds while Figure 5.24 shows maximum equivalent (von-Mises) stress at 300 seconds.

Figure 5.23: Equivalent (von-Mises) stress at 274.5 seconds

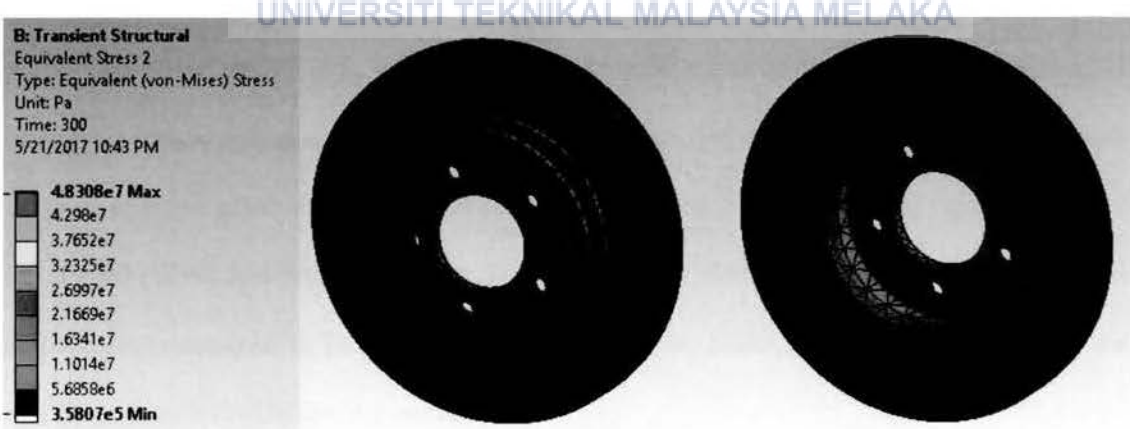

Figure 5.24: Equivalent (von-Mises) stress at 300 seconds

In conclusion, although the highest maximum equivalent (von-Mises) stress recorded at the inboard and outboard of disc brake is 113.69 MPa while its minimum value is 0.49489 MPa, it is still below the the maximum tensile strength of gray cast iron which is 240 MPa, thus the equivalent (von-Mises) stress is acceptable. Thus, the disc brake is still in elastic deformation region where it has tendency to return to its original shape. Although the disc brake equivalent (von-Mises) is nearly half maximum tensile strength, it must be controlled by controlling its temperature distribution as higher temperature will produce higher equivalent (von-Mises) stress.

5.4 VALIDATION OF RESULT

5.4.1 Introduction

There are many ways to validate the analysis result of the thermal stress such as through analytical solution, vehicle disc brake manufacturing data and journal reviews. In this research, analysis result is validated through analytical solution by adapting equation derived by Limpert (1999) and journal reviews. The disc brake is treated as lumped system where the temperature is assumed to be uniform throughout the disc brake rotor, thermal properties and both heat transfer coefficient are constant.

5.4.2 Analytical Result

The lumped equation during the repeated braking adapted from Limpert 1999:

$$\frac{T(t)-T_i}{T_i-T_\infty} = e^{(-h_R A_R t)/(p_R\, c_R v_R)} \qquad (5.1)$$

Where;

A_R = Rotor surface area

h_R = Heat transfer coefficient

t = Cooling time cycle

T_∞ = Ambient temperature

t_s = Brake time

p_R = Rotor density

c_R = Specific heat

v_R = Rotor volume

The average temperature increase per stop is given as:

$$\Delta T = \frac{P_B\, t_s}{p_R\, c_R\, v_R} \qquad (5.2)$$

$$= \frac{(326448)(4.5)}{(7200)(586)(1.275\times10^{-3}}$$

$$=273 \text{ K}$$

The equation of brake disc surface temperature for 2 cycles of braking process is:

$$T_1 - T_\infty = \frac{\left[1-e^{(-n_a h_R\, A_R\, t\,)/(p_R\, c_R\, v_R\,)}\right]\Delta T}{1-e^{(-h_R A_R t)/(p_R c_R v_R)}} \tag{5.3}$$

$$= \frac{\left[1-e^{(-2(225.272)(0.0944228)(300))}/(7200)(586)(1.275\times10^{-3}\,)\right]\Delta T}{1-e^{(-225.272)(0.0944228)(300)}/(7200)(586)(1.275\times10^{-3})}$$

$$= 330.32$$

$$T_1 = 330.32 + 303.15$$

$$= 633.47 \text{ K}$$

Equation (5.3) is used to calculate until ten load cycle temperature and it is tabulate in the table below together with the temperature from the analysis

Table 5.4: Temperature of disc brake rotor

Load Cycle	Analytical Result (K)	Analysis Result (K)
0	303.15	303.15
1	576.3	439.79
2	633.47	508.96
3	657.07	531.63
4	664.27	582.93
5	666.474	593.34
6	667.146	607.94
7	667.37	620.31
8	667.414	630.87
9	667.433	640
10	667.434	647.97

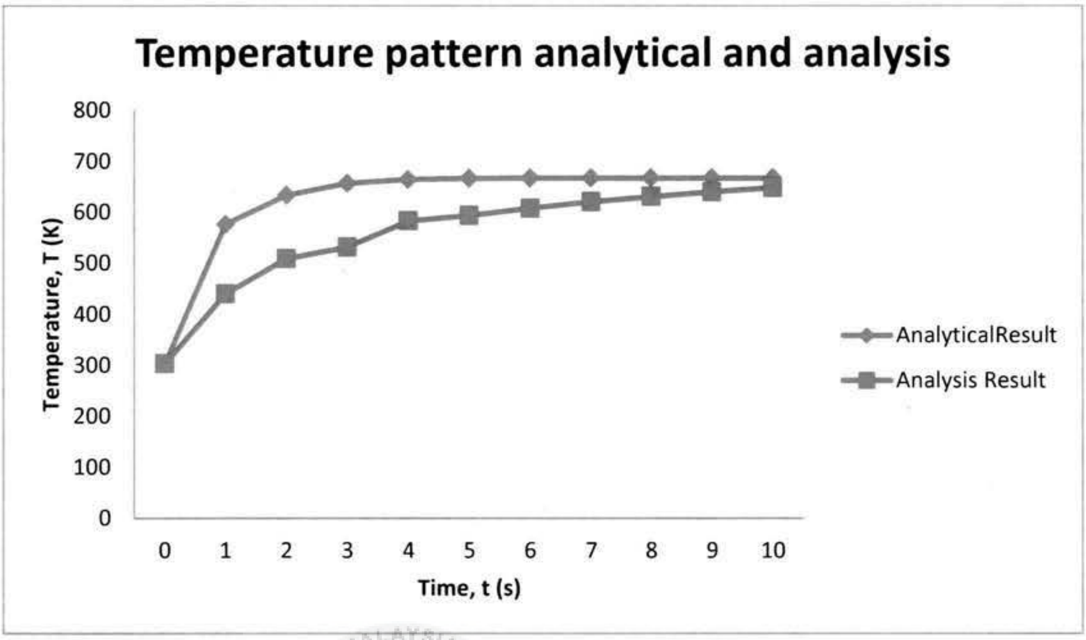

Figure 5.25: Temperature pattern of disc brake rotor

Figure 5.25 shows the graph of temperature pattern of analytical solution versus result of analysis. According to the result of the analysis, the maximum temperature recorded along the disc brake rotor is 647.97 K while the maximum temperature of analytical solution is 667.434. Although the different of the temperature is 19.46 K, it is still accurate and can be accepted since the different between analytical and analysis is small.

5.4.3 Journal Reviews

In order to validate and compare the result of NGV vehicle disc brake rotor analysis, the other research is reviewed. Actually, there are a lot of researches that have been conducted to analyze the thermal stress on brake disc rotor. Each research contains valuable information that is used as a reference in this project. Guru Murthy Nathi (2012) conducted a research about Coupled Structural / Thermal Analysis of Disc Brake using a regular vehicle with mass of vehicle is 1400 kg and 4 seconds time to stop the vehicle. The maximum temperature obtained from the analysis is 408 K and maximum von Mises stress is observed to be 115 MPa. The different between this research and Guru Murthy Nathi research in maximum temperature is 239 K due to the high different in vehicle mass. A.Belhocine (2014) conduct a research about thermal analysis of both ventilated and full disc brake rotor with frictional heat generation using gray cast iron (FG 15) and 1385 kg vehicle mass. The maximum temperature shows 618.44 K as the highest. The 29.53 K different is due to the different of heat flux generated and heat transfer effect.

CHAPTER 6

CONCLUSION AND RECOMMENDATION

6.1 Conclusion

In conclusion, the thermal stress analysis of steady state and transient condition of NGV vehicle disc brake rotor have been successfully done by considering the effect of temperature distribution. The result of analysis using ANSYS software also can be considered as accurate due the temperature different between analytical result and analysis is result is small. However, the actual temperature could be less or more precise than the analysis due to the limitation of boundry condition, number of meshing and also heat transfer effect. The result of this analysis is slightly higher compared to other research as this is research is using NGV vehicle disc brake where its gross weight is higher than regular vehicle, thus the heat flux is higher. The thermal stress due to the heat flux has many influences on the braking performance and excessive thermal stress can lead to brake fade, thermal judder and thermal crack. However, the temperature of the NGV vehicle gray cast iron disc brake is still below the maximum service temperature of gray cast iron disc brake and its equivalent (von-Mises) stress also still below maximum tensile strength. Thus, the NGV vehicle gray cast iron disc brake is still safe to use.

6.2 Recommendation

In this research, the ANSYS software used is the student version and it has limitation on meshing and analysis. Thus, for future work, author would recommend using full version ANSYS software so that the result is more accurate. Besides, the boundary condition in this research only consider disc brake surface, inner ring surface and outer ring surface. The result will be more accurate if more boundary condition is considered and real braking condition such as heat transfer effect from conduction, convection and radiation is adapted. Therefore, the real braking condition of disc brake can be observed. Also, it will be more exciting if the research of thermal stress on NGV vehicle disc brake rotor is conducted when moving downhill or uphill along the highland area. Practically, the thermal stress will be different compared to moving in flat surface. Lastly, author would suggest to analyze the effect of NGV vehicle to disc brake rotor using different parameter beside gross weight.

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