Temperature Distribution on a Car Disc Brake

by

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Dissertation submitted in partial fulfilment of
the requirements for the
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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
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Approved:

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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 the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

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KAMARUL BAHRIN B. MOHD KAMIL

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ABSTRACT

The analysis of temperature distribution on a car disc brake can be approached by considering the contact surface between disc and brake pads of a car when a car decelerating by applying braking force to the brake pedal. The braking force will be transferred from the brake pedal via brake fluid to the brake pads to generate the friction area between disc brake and brake pad. Heat flux, or frictional heat, will be produced at the frictional contact area by two mating parts with difference materials properties will decrease with time and it may result in increasing the temperature of the disc brake. On top of that, this may lead to false thermal elastic distortions and unrealistic contact conditions. The objective of this study is to investigate on temperature distribution in a car brake disc of small vehicle in the course of brake application. An analytical model will be used in this study for the determinations of the heat flux dissipated by the disc brake on the working surface. The three dimensional analysis using the finite element method is conducted for consecutively repeated brake application. The braking is done by two brake pads and the heating and cooling occur alternatively in the brake disc during the brake application. A severe repeated brake application is taken into account for the wheel load of 1.629 kN and the braking is applied from a speed of 120 km/h till to stop with a deceleration of 0.3*g*. After the first braking is completed, the vehicle is accelerated again and it takes 2 minutes which is considered as the cooling period before the next brake application is conducted. The temperature distributions within the brake disc for three times brake application are then plotted and discussed. It is found that the maximum temperature is obtained at the mid-time of brake application.

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Nomenclature

E_{in}	heat energy enters to the system, kJ
E_{out}	heat energy leaves to the system, kJ
g	gravitational acceleration, 9.81 m/s^2
U_{Tire}	initial velocity on tire, m/s
V_{Tire}	final velocity on tire, m/s
D	deceleration, in unit of g
a	acceleration, m/s^2
D_{Tire}	tire diameter, m
D_{Disc}	disc diameter, m
ΔU	internal heat energy, kJ
T	kinetic energy, kJ
m	mass of the car, kg
B_{front}	braking at front wheel
B_{rear}	braking at rear wheel
v	velocity, m/s
$F_{Friction}$	friction force, kN
ΔQ_{absorb}	heat absorbed, kJ
q_{1-2}	heat flux from point 1 to point 2, unit of MW/m^2
$\dot{E}$	rate of energy generation
$\dot{q}$	thermal energy generation rate

CHAPTER 1
INTRODUCTION

1.0 Background of Study

Thermal expansions and deformations of the mating parts in disc brakes result in loss in contact area [1] which will further increase the temperature. This may result in temperature flashes [2]. These flashes can cause the thermal cracking of gray cast iron disc and disintegration of friction material, which makes brake performance unpredictable, poor and ultimately leading to brake failure. Disc brakes respond reliably well even with higher energy loadings (MW/m 2) on friction pads with lesser distortions and thermal deformations. Hence with improved friction pad materials, disc brakes accept higher thermal loadings.

Thus as part for checking brake performance variation with high thermal loadings, analysis for transient disc temperature is needed. During braking major share of heat energy (about 95%) is dissipated through brake disc while rest of it goes into brake pads. This major heat energy diffuses through conduction within the disc is dissipated by convection and radiation from the outer surfaces of disc. Temperatures within the disc are transient in nature and are difficult to log correctly and immediately through conventional methods. The performance of the brake depends mainly on the brake pad material and its behaviour.

The behaviour of the brake pad is depends on speed, normal and drag load pressure and temperature at which it is rubbing with the disc. The heat generated during the severe brake application rises at the disc/pad interface temperature and this temperature rise may vary the pad performance. Generally, this may result in variation of coefficient of friction of the brake pad and thus in fade of the pad. (The fade in a brake is defined as the drop in the output torque for the given constant input). In view of the smaller size of the wheels and consequential larger dissipation rates of the energy in brake system, it is essential to know the peak temperatures attained during repeated cycles to make sure of the positive braking action.

1.1 Problem Statement

Temperatures rising and thermal stresses which occurred in a car disc brake during braking may cause undesirable effects on the material of the disc that eventually lead to the over heated disc brake and initiation of a crack. These effects are small for a small car, but they can cause big impact especially when the car is moving. The predominant trend in the development of vehicles nowadays is toward to higher vehicle speed and a heavier load, which considerably increases the energy, required stopping the vehicle. The maximum heat is created and few amount of heat be able to be dissipated by the disc brake system.

1.2 Significant of the Study

The study of temperature distribution in a car disc brake is preliminary stage toward further study on initiation of crack occurred at disc brake surface due to excessive thermal stress. The weakness of the brake is due to material failure which might cause fatal to the car passenger and the driver as well. The design of the disc brake should be optimized and therefore, it appears to be very important to examine the transient temperature distribution in the disc brake to acknowledge the significant points of brake for improvement.

1.3 Objectives

The objectives of this project are as stated below.

- To investigate the temperature distribution of a car disc brake
- To analyse and simulate the temperature distribution on a car disc brake consecutively during periodical brake applications.

1.4 Scope of study

A type of brake pad and disc are considered to be analyzed in this study. The disc brake undergoes heating and cooling during the brake application, which figured as clock mechanism as shown at the Figure 1.1. The transient temperature in the brake disc will be analyzed under the three severe braking conditions, which means the brake is applied when the car is in its maximum load, specific speed and subjected to high deceleration. Although, it assumed that the frictional heat created at the contact region is constant and equal along the surface.

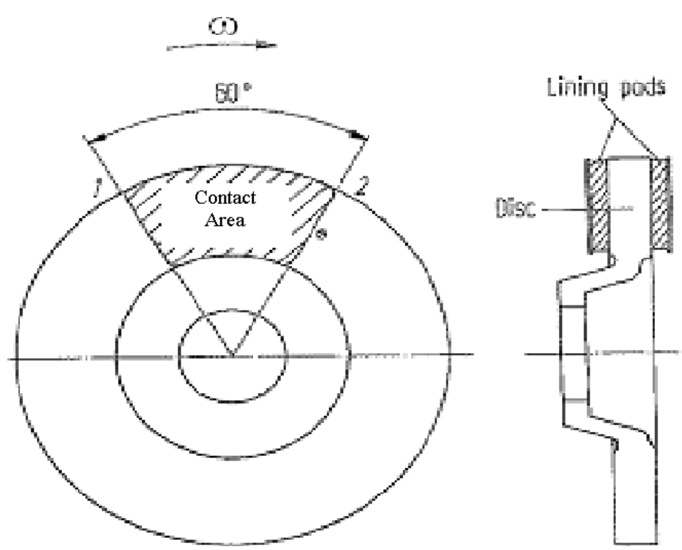


Figure 1.1: Clock Mechanism heating and cooling

Several parameters are set to define these three severe braking conditions and as stated in Table 1.1. Also, a specific disc material is selected for this analysis that is **Gray Cast Iron A48 Class 40**. The thermal properties of this material are:

- Specific heat, C_p = 550 J/kg ^{o}C
- Thermal conductivity, k = 51 W/m ^{o}C
- Density, ρ = 7114 kg/m^3
- Film coefficient of convection = 50 W/m ^{o}C

Table 1.1: Considerable parameter of a car disc brake analysis

Parameter	Value
Car initial velocity at tire, U_{Tire}	120 km/h or 33.333 m/s
Car final velocity at tire, V_{Tire}	0 km/h or 0 m/s
Car initial velocity at disc, U_{Disc}	46.73 km/h or 12.981 m/s
Car final velocity at disc, V_{Disc}	0 km/h or 0 m/s
Car constant deceleration, a (or d)	0.3g or 2.943 m/s^2
Average passager mass, m_P	75 kg per person
Average luggage mass, m_L	100 kg
Car total mass including 5 passengers at front tire (60% at front), m_{Total}	0.5535 tons or 553.5 kg per tire
tire diameter, D_{tyre}	0.5752 m (tyre size: 175/70 R13)
Average rubbing diameter, $D_{friction}$	0.1825 m
Pad or contact area angle, Ɵ	60 degrees
Pad width, W_{pad}	0.0171 m
Pad contact area, $A_{friction}$	0.004 m^2

Analysis is come out with three different braking cases or conditions. Generally, the simulation of heat distribution is done on the contact disc and pad as described in Figure 1.1 while the contact surface area is as equal as friction area of the disc for all braking conditions. Then, different heat fluxes from the calculation are applied to the friction surface according to the car speed and deceleration for the first condition. When the car meets the final velocity of analysis, the brake is unclamped or released to let the disc cool naturally in three minutes and this lies to the second condition. Third condition is second braking force is applied when car at the speed of 120km/h to stop. Fourth condition is the same as the second condition when the disc is cooled naturally. Last but not least, the final condition is to stop the car from the speed of 120km/h at 0.3g deceleration.

CHAPTER 2
LITERATURE REVIEW

2.0 Concept of Energy Conversion during Braking

Heat, friction and mechanical combination are the factors that every of vehicle depend on, in order to get it moving and stop from moving. In order to get the vehicles moving, the heart of the vehicle which is the engine will generates heat energy in the combustion chamber where the heat energy being converted into mechanical energy to the power-train and transmitted to the tire via the axles. Although these energies are fundamental to the vehicle faction, friction resistance will be the final issue where the tires and road interfaced mutually and generate traction to accelerate the vehicle. The more traction is produced; the enhanced grip on the road surface we can get.

When the heat energy is converted into mechanical energy, the mechanical energy will returned to heat energy during the vehicle condition is moving to stop. Heat energy to stop the vehicle is achieved when two area between brake pads and disc are in contact. When the brake pedal was pushed or pressed, the pressure will develop in the hydraulic or air brake system which means the brake pad is pushed by the piston cylinder in the brake calliper adjacent to the both disc surface. The contact between to surfaces will produce the heat Friction which then it will convert from mechanical energy into the heat energy. Consequently it will result to slowing down the vehicle. Heat generated by two mating surfaces will be absorbed by the disc and as well as the brake pads. About 95% of heat produced will be transferred immediately to the disc brake and the remaining is transferred to the brake components. The Transferred heat will then dissipated into the environment, as a result cooling the disc and brake components.

The theory of energy conversion in a control volume has been proven through this phenomenon. The heat energy enters and leaves during the control volume surface, will denoted as $\boldsymbol{E_{in}}$ and $\boldsymbol{E_{out}}$. Therefore, the inflow and outflow terms are related wholly with the processes taking place at the control surface are proportional to the surface area. There are stated heat transfer condition involved in energy inflow and outflow are conduction and convection as described in Figure 2.1. The general form of energy conservation expressed at any instant time within the time interval Δt as:

$$E_{in} - E_{out} = 0 \qquad (1)$$

In this research, assumption has been made where no thermal energy is produced and stored inside the control volume.

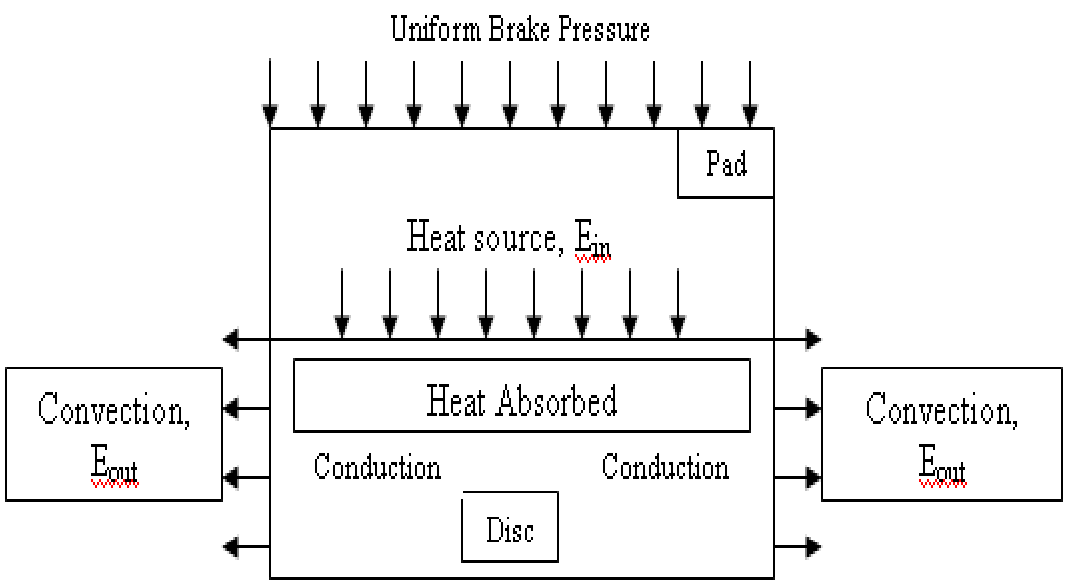


Figure 2.1: Heat balance in a brake disc during the brake application

2.1 Car and Disc Brake Dynamic Theory

2.1.1 Equation of Rigid Body Rectilinear Motion

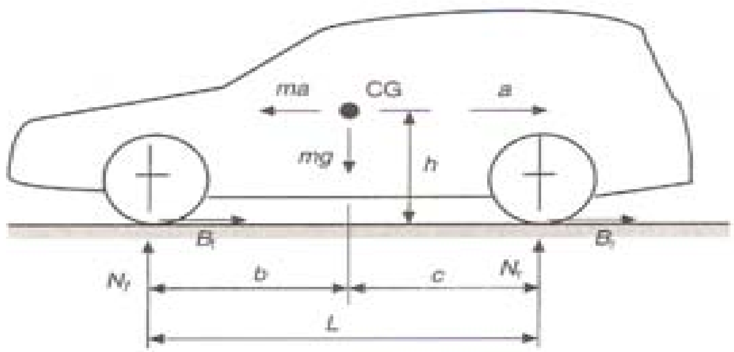


Figure 2.2: Free body diagram of a vehicle during braking

In view of the car is move at initial speed of U_{Tire}, and attain to the final speed of V_{Tire} after braking force applied. The car will become slower at constant deceleration, *D* or *a*, according to the uniform braking force applied on the brake pedal as shown on Figure 2.2. Therefore, the relationship of velocity and acceleration of the car in linear motion can be defined as

$$v = u + at \tag{2}$$

or

$$v^2 = u^2 + 2a(s - s_0) \tag{3}$$

Where; v = final velocity (m/s)
u = initial velocity (m/s)
t = time (s)
s = distance (m)

According to this relation equation, we can find or determine the time taken between two velocities and acceleration. In addition, the distance travelled by the car while changing the velocity can be calculated as well as from the equation (2). In order to get

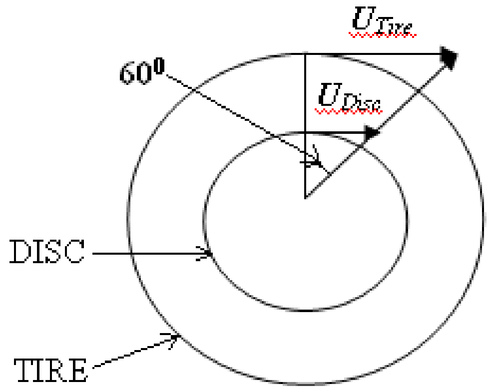


Figure 2.3: Velocity analysis diagram

the actual tangential velocity of disc which is proportional to the tire velocity, the diameter of the tire, D_{Tire} and the disc, D_{Disc} are required as well as the brake pad angle. Regarding to these parameters, the tangential velocity of the disc can be find using trigonometry concept when the radius of the tire and disc are known as well as pad angle refer to Figure 2.3.

2.1.2 Equation of Rigid Body Motion about a Fixed Axis

Disc brake is revolving about a fixed axis which at the midpoint of the tire diameter. The pads are actually static in the pad mounting at a brake calliper. In this research, the pads are considered rotate along a fixed axis at both disc sides. Otherwise, the disc brake in this study is assumed to be fixed or not move. The pads are move from one point to another point and this movement is carry on until the car is reached to the final stopping velocity, V_{Tire}. Those points are shown in Figure 2.3. Assuming the tangent velocity, v at outer point of car tire is equal to car speed and the deceleration is also equal to tangent deceleration, D or a, of that point. Then, the initial angular velocity, ω and the constant angular deceleration, α of the disc brake can be figure out using the equations below;

$$v = r\omega \tag{4}$$

$$a = r\alpha \tag{5}$$

Where; r = tire outer radius (m)

ω = initial angular velocity (rad/s),

α = angular deceleration (rad/s^2)

Throughout braking application, the angular velocity of the car will reduce as well as the angular velocity of the disc at constant angular deceleration, *D* or *a* as the pads moves from point 1 to point 2 as described in Figure 2.4.The angular velocity of disc at the next point, ω_{2}, can be determined using plug in the values of angular velocity at the point before, constant angular deceleration and angular displacement of the pads, $\Theta = 60^0$, into the equation below;

$$\omega_2^{\ 2} = \omega_1^{\ 2} + 2\alpha(\theta - \theta_0) \tag{6}$$

Where; ω = angular velocity of pad movement (rad/s)

α = angular deceleration (rad/s^2)

Θ = angular displacement (radian)

The following are been using in order to find the time interval of each brake pads movement;

$$\omega_2 = \omega_1 + \alpha\ \Delta t \tag{7}$$

Where; Δt = time interval (s)

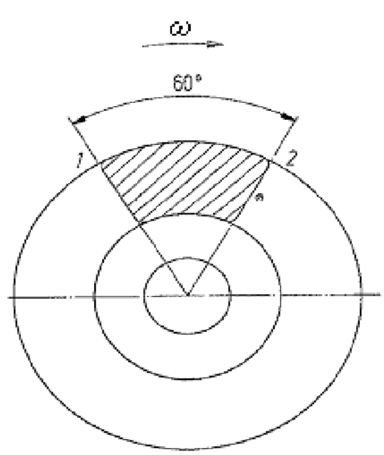


Figure 2.4: Direction of pads movement from 1 to 2

2.2 Braking Work and Energy Theory

2.2.1 The Work of a Constant Force along a Straight Line

A force, *F*, perform work on a particle only when the particle undergoes a displacement. Same thing occur to the moving car when the car need to stop or slow down it speed, a force is required might constant. Besides that, in opposite direction of the car, the movement turn into slower and decelerates at constant rate until it is reach to the required speed or velocity. Work of force is define as

$$U_{1-2} = F_c \cos\theta(s_2 - s_1) \tag{8}$$

Where; *U* = work done (J),

F = constant force magnitude (N)

Θ = force direction (degree)

s = distance (m)

In this case, the grade is 0% which means the car assumed to move laterally. Therefore, *Θ* is 0 since force is acting on same direction as car movement which is in the horizontal direction. The initial distance, s_1 is also 0. So, the equation is simplified to

$$\Delta U = Fs = T \tag{9}$$

When the car is moving, the kinetic energy, *T*, will be store on the tires. Kinetic energy terms are defined as;

$$T = ½\, mv^2 \tag{10}$$

Where; *m* = mass of the car (kg),

v = velocity or speed of the car (m/s)

This term is always in positive scalar. Theoretically, initial kinetic energy of the object adding with the work done by all the forces acting on the object as it moves from initial to the final position is equal to the final kinetic energy, *T*, of the object. It often represent in the form;

$$T_1 + \sum U_{1-2} = T_2 \tag{11}$$

2.2.2 Work of Friction Caused by Sliding pads

Assuming that the value of force needed to brake the car when its moving horizontally is equal to total frictional force required in all applied disc brakes. Generally, the ratio of the braking force between front and rear wheels is not the same. For a typical car, front tires requires 60% of braking force and rear tires requires the remaining of the braking force is about 40%. Also, assume that the deceleration force is equally distributed to 4 tires. Then, the frictional force required by one disc brake is;

$$F_{friction} = F_{car} = 1/2\,\frac{mv^2}{2s} \qquad 12)$$

Where; m = load mass (kg),

v = velocity (m/s),

s = distance (m)

Brake force at front tires as shown in Figure 2.5 is higher to the rear tires because due to the weight transfer during the braking condition and the static load at the front tires which is the load at the front compartment (side) is 60% higher compared to the rear compartment.

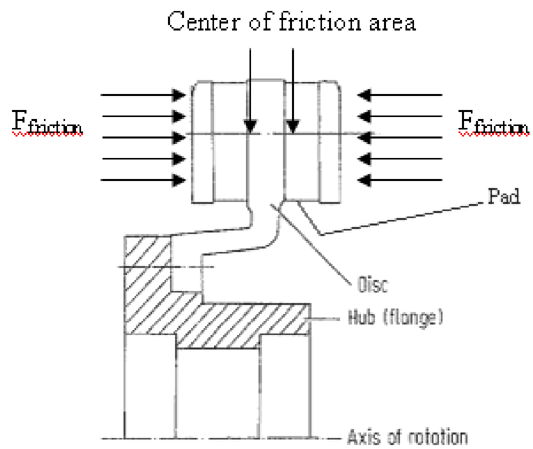


Figure 2.5: Axle layout of disc brake when Brake force applied

Therefore, the load for single front tire is distributed about 30% of the total 60% of front tires load each. Due to the high concentrated load, the front brake disc will undergo the more severe temperature distribution compared to the rear. Therefore, the frictional forces at a front tire equation as below;

$$F_{friction} = 0.3F \tag{13}$$

Since there is force acting on that point and it is moving in certain distance, work is done and the work of friction caused by sliding. Assuming that the generation of heat energy, U, is equal to friction work, but there is only 95% of heat energy will be absorbed by the disc brake. So, the heat energy absorbed, Q, due to friction from point 1 to 2 is

$$\Delta Q_{absorb} = 0.95\Delta U_{friction} = 0.95F_{friction}s_{1-2} = 0.95T \tag{14}$$

Where; ΔQ_{absorb} = heat energy absorbed (kJ)

$U_{friction}$ = heat energy (kJ)

$F_{friction}s_{1-2}$ = friction work (Nm)

T = Kinetic energy (kJ)

Assume that the heat energy generated is equal to kinetic energy and only 95% of heat energy is absorbed by the disc. In this study, we can see that there are 2 surfaces are as frictional surface because of the clamping forces to the brake pads. As a result, the heat absorbed need to be divided to two. Therefore, the heat flux, q_{1-2}, applied to disc surface from point 1 to 2 is

$$q_{1-2} = 0.95\left(\frac{\Delta Q_{absorb}}{2\Delta t_{1-2} A_{shoe}}\right) \tag{15}$$

2.2.3 Clamping force

The clamping force of a caliper in pounds is the brake line pressure multiplied by the total piston area of the caliper in a fixed caliper and two times the total piston area in a floating design. To increase the clamping force it is necessary to either increase the line pressure or the piston area. Increasing the pad area or the coefficient of friction will not increase clamping force. What does this mean is that your clamping force of your caliper and brake subsystems has nothing to do with pad design or makeup nor does it have anything to do with the type of disc used.

2.3 Heat Diffusion Equation

A major objective in a conduction analysis is to determine the temperature field in a medium. That is, it is wished to know the temperature distribution, which represents how temperature varies with position in the medium. One this distribution is known, the conduction heat flux at any point in the medium or on its surface may be computed from Fourier's Law.

2.3.1 Cartesian Coordinates

Consider a homogeneous medium within which there is no bulk motion (advection) and the temperature distribution *T(x,y,z)* is expressed in Cartesian coordinates. If there are temperature gradients, conduction heat transfer will occur across each of the control surfaces as shown in Figure 2.6. The conduction heat rates at the opposite surfaces can then be expressed as a Taylor series expansion where,

$$q_{x+dx} = q_x + \frac{\partial q_y}{\partial x} dx \tag{16.a}$$

$$q_{y+dy} = q_y + \frac{\partial q_y}{\partial y} dy \tag{16.b}$$

$$q_{z+dz} = q_z + \frac{\partial q_z}{\partial x} dx \tag{16.c}$$

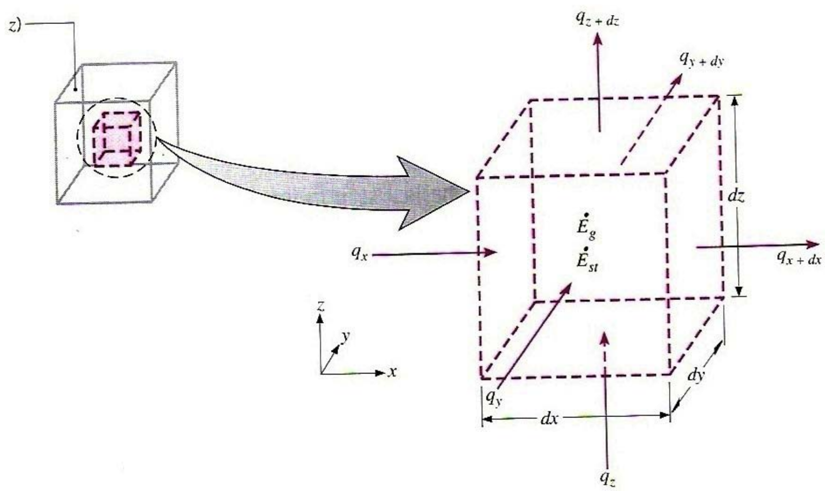


Figure 2.6: Differential control volume for conduction analysis in Cartesian Coordinates

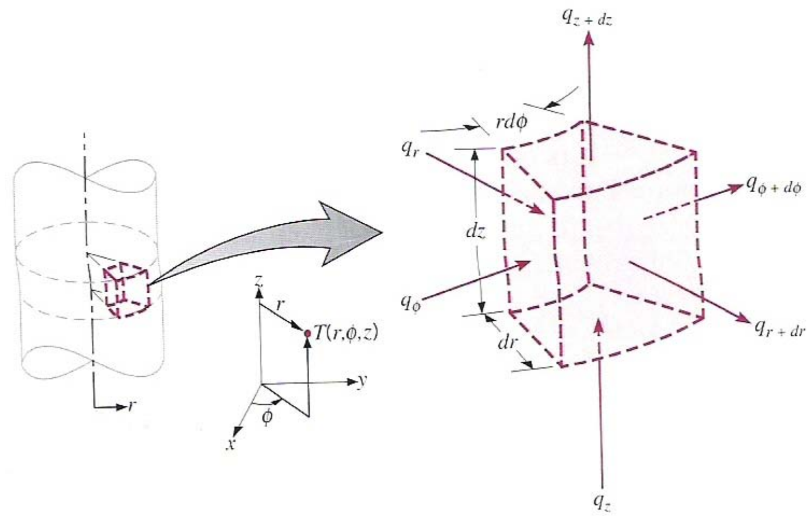


Figure 2.7: Differential control volume for conduction analysis in Cylindrical coordinates

In words, Eq. (16) simply states that the x component of the heat transfer rate at $x+dx$ is equal to the value of this component at x plus the amount by which it changes with respect to x time to dx. Within the medium, there may also be energy generation term represented by

$$\dot{E} = \dot{q}\, dxdydz \tag{17}$$

In addition, there changes may occur in amount of internal thermal energy stored. If the material not experiencing phase change, latent energy effects are not pertinent, then energy storage term expressed by

$$\dot{E} = \rho \; c_p \frac{\partial T}{\partial t} dxdydz \tag{18}$$

On the rate basis, the general form of conservation of energy requirement is

$$\dot{E}_{in} + \dot{E}_g - \dot{E}_{out} = \dot{E}_{st} \tag{19}$$

Substituting all Eq. (16), Eq. (17) and Eq. (18) into Eq. (19),

$$-\frac{\partial q_x}{\partial x}dx-\frac{\partial q_y}{\partial y}dy-\frac{\partial q_z}{\partial z}dz+\dot{q}\,dxdydz=\rho c_p\frac{\partial T}{\partial t}dxdydz \tag{20}$$

Evaluating from Fourier's Law, the final equation is formed

$$\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial t}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial t}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial t}\right)+\dot{q}=\rho c_p\frac{\partial T}{\partial t} \tag{21}$$

Eq. (21) is the general form, in Cartesian coordinates, of heat diffusion equation. This equation provides a basic tool for heat conduction analysis. From its solution, the temperature distribution *T(x,y,z)* as a function of time can be obtained. In word, it states that at any point in the medium the rate of energy transfer by conduction into a unit volume plus the volumetric rate of thermal energy generation must equal the rate of change of thermal energy stored within the volume.

2.3.2 Cylindrical Coordinates

The general form of the heat flux vector of Fourier's Law in cylindrical coordinates expressed as

$$q''=-k\nabla T=-k\left(i\frac{\partial T}{\partial r}+j\frac{\partial T}{\partial \phi}+k\frac{\partial T}{\partial z}\right) \tag{22}$$

Applying an energy balance to the differential control volume in Figure 2.7, the following general form heat equation is obtained

$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial}{\partial \phi}\left(k\frac{\partial T}{\partial \phi}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{q}=\rho c_p\frac{\partial T}{\partial t} \tag{23}$$

2.3.3 Adiabatic process

In thermodynamics, an adiabatic process or an isocaloric process is a thermodynamic process in which no heat is transferred to or from the working fluid. The term "adiabatic" literally means impassable (from Greek, not-through-to pass), corresponding here to an absence of heat transfer. For example, an adiabatic boundary is a boundary that is impermeable to heat transfer and the system is said to be adiabatically (or thermally) insulated; an insulated wall approximates an adiabatic boundary. Another example is the adiabatic flame temperature, which is the temperature that would be achieved by a flame in the absence of heat loss to the surroundings. An adiabatic process that is reversible is also called an isentropic process. Additionally, an adiabatic process that is irreversible and extracts no work is in an isenthalpic process, such as viscous drag, progressing towards a nonnegative change in entropy.

One opposite extreme allowing heat transfer with the surroundings, causing the temperature to remain constant is known as an isothermal process. Since temperature is thermodynamically conjugate to entropy, the isothermal process is conjugate to the adiabatic process for reversible transformations.

A transformation of a thermodynamic system can be considered adiabatic when it is quick enough that no significant heat is transferred between the system and the outside. At the opposite extreme, a transformation of a thermodynamic system can be considered isothermal if it is slow enough so that the system's temperature remains constant by heat exchange with the outside.

2.3.4 Convection

Convection is one of only three heat transfer mechanisms. Conduction and radiation are the other two. Convection is the transfer of heat by fluid flow. Air can be considered to be a fluid in a thermal model of a brake system when it is moving and is contact with the heated surfaces of the disc or drum. In the case of a solid disc the air moving over the surface of the disc is a very random and turbulent, but still function to provide some cooling. In the case of a ventilated disc, by the pressure of a forced air duct or by induced flow that is a result of the centrifugal acceleration of the air already in the vent of a rotating disc, air flows through the vents. The air absorbs thermal energy along the vent path. In this way, the heat generated by the braking system of an automobile is transferred to the moving air stream and away from the brake disc.

As far as cross drilled rotors go, the advent of carbon metallic friction materials with their increased temperatures and thermal shock characteristics ended the day of the drilled disc in professional racing. They are still seen (mainly as cosmetic items) on motorbikes and some road going sports cars. Typically in original equipment road car applications these holes are cast then finished machined to provide the best possible conditions by which to resist cracking in use. Properly designed, drilled discs tend to operate cooler than non-drilled ventilated discs of the same design due the higher flow rates through the vents from the supplemental inlets and increased surface area in the hole. That's right, inlets, the flow is into the hole and out through the vent to the *OD* of the disc. If discs are to be drilled, the external edges of the holes must be chamfered (or, better yet, radiused) and should also be peened.

2.4 Finite Element Method

A finite element is a discrete spatial region that is a subdivision of a continuum. A finite element method (FEM) is a mathematical procedure for satisfying a partial differential equation in an average sense over a finite element. Various methods exist. All of them require that an integral representation of a partial differential equation be constructed. Classical finite element methods for structural mechanics are based on variation principles. Variation principles also apply to steady-state diffusion and conduction processes. However, for transient diffusion and conduction and for convective transfer processes, it is necessary to use more general procedures, such as a method of weighted residuals.

A physical problem solution by a finite element method follows a well-defined sequential process. At first, the physical region is discretized into elements. The number, type and allocation of elements are often a matter of judgment. Second, interpolation or shape functions are selected for the elements. The interpolation functions represent the assumed form of spatial solution in the elements and are related to the number of nodes in the elements. Third, the matrix equations for an individual element are formulated using integral statement for the element as a guide. Fourth, the matrix equations for the overall system, consisting of all elements are assembled. Finally, global equations that are of the same form as the element equations but are of larger dimension are solved.

Figure 2.8 shows the finite element model of disc brake with boundary conditions. The inner radius, outer radius, thickness and ventilation thickness of a disc are *0.0705*, *0.112*, *0.007* and *0.003* m, respectively. The thickness of a pad is *0.010* m. The hydraulic pressure is applied to the boundary along the radius of the piston side pad and the immobility condition in the axial direction is applied to the boundary along the radius of the finger side.

The heat finite element model of disc brakes with boundary conditions is shown in Figure 2.9. The convective boundary conditions are imposed on all boundaries to consider more realistic heat conditions. The initial temperature is T_0 = 30 $^\circ$C in this study.

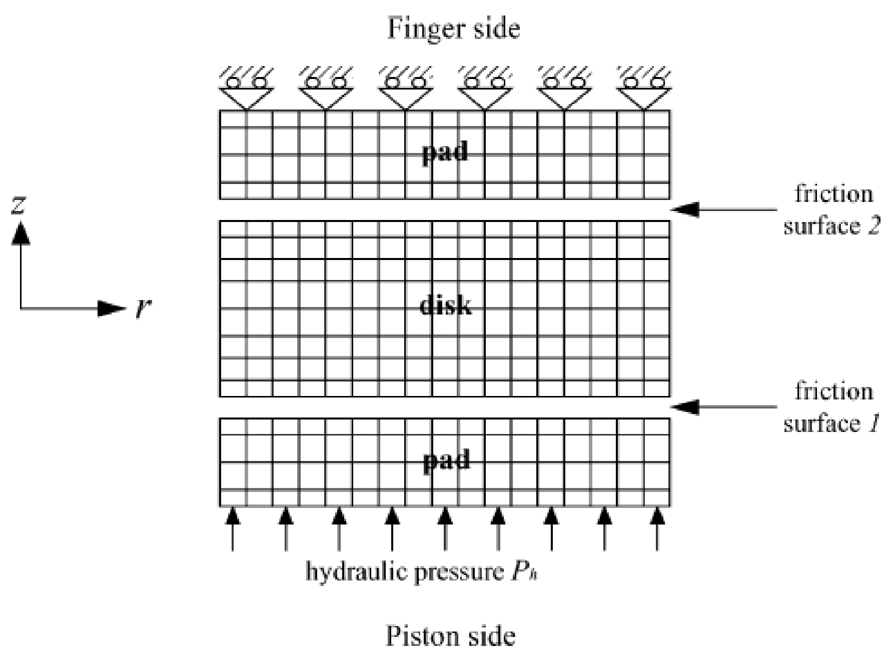


Figure 2.8: Finite element model for the transient temperature analysis

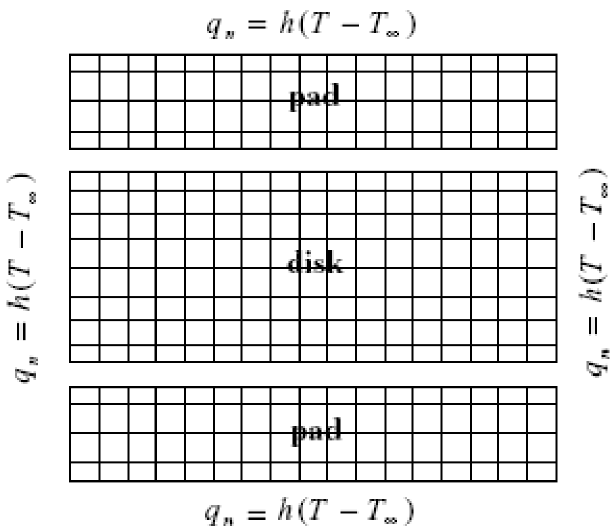


Figure 2.9: Heat finite element models for the transient temperature analysis

2.4.1 Practical Implementation

A wide variety of finite elements can be employed, along with different interpolation and weighting functions, spatial distribution of elements and element dimensions according to the inherent flexibility in FEM. Different approaches can be used to generate, assemble and solve governing system of algebraic equations. The results obtained are stored and processed using interpolation functions to obtain the desired field variables and relevant derived quantities in the form of graphs, tables, and correlating equations.

While the preceding aspects give rise to considerable flexibility and versatility in FEM solutions, they also make it desirable to use available software, whenever possible, to simplify code development for the practical implementation of the method. A typical finite element program consists of the following 3 main parts:

1. ***Preprocessor.*** At this point, the input data, pertaining to the geometry and dimensions of the computational domain, boundary conditions, governing

differential equations, and finite element mesh, are read. The type, number, dimensions and coordinates of the elements are read or generated. Information on the interpolation and weighting functions is also read or generated. The input data may be printed and the mesh plotted to show coverage of the computational region.

2. ***Processor.*** This is main, or central, portion of the program. Here, the element coefficient matrices and column vectors are determined. The boundary conditions are numerically imposed. The element equations are assembled. The system of algebraic equations is solved to yield the values of the various field variables such as temperature, pressure and concentration at the nodes.

3. ***Postprocessor.*** When the field variables are obtained at the nodes, the results at points other than the nodes are computed by using interpolation function and the available value at the nodes. Derived variables and quantities, such as stream function, heat transfer rate, heat transfer coefficient, as needed, are obtained from the calculated variables. The output data are then processed to present the results in desired format like graphs and contour plots.

2.4.2 Finite Element in ANSYS

SOLID90 is a higher order version of the 3-D eight node thermal element (SOLID70). Each element has twenty nodes with a single degree of freedom, temperature, at each node. The twenty node elements have compatible temperature shapes and are well suited to model curved boundaries. The twenty node thermal element is applicable to a 3-D, steady-state or transient thermal analysis.

The geometry, node locations, and the coordinate system for this element are shown in Figure 2.10: "SOLID90 Geometry". The element is defined by *20* node points and the material properties. A prism-shaped element may be formed by defining duplicate K, L, and S; A and B; and O, P, and W node numbers. A tetrahedral-shaped element and a pyramid-shaped element may also be formed.

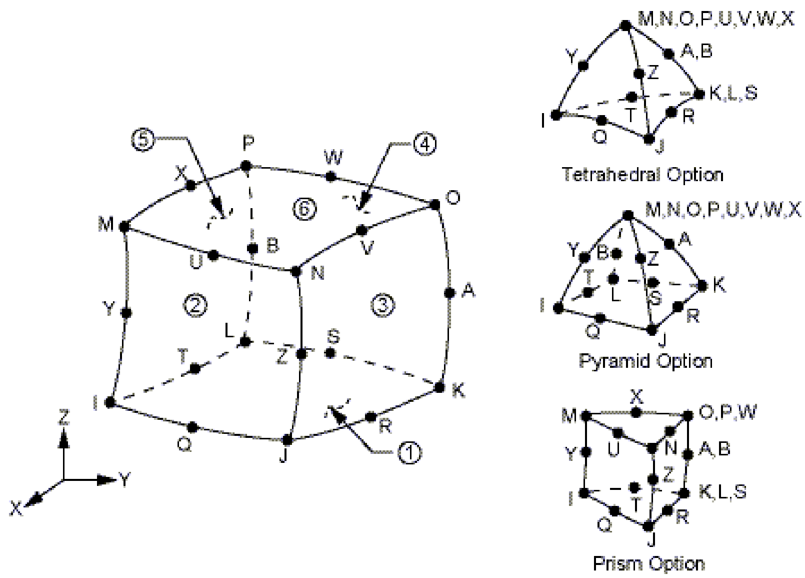


Figure 2.10: "SOLID90 Geometry"

CHAPTER 3
METHODOLOGY

3.0 Procedure Identification

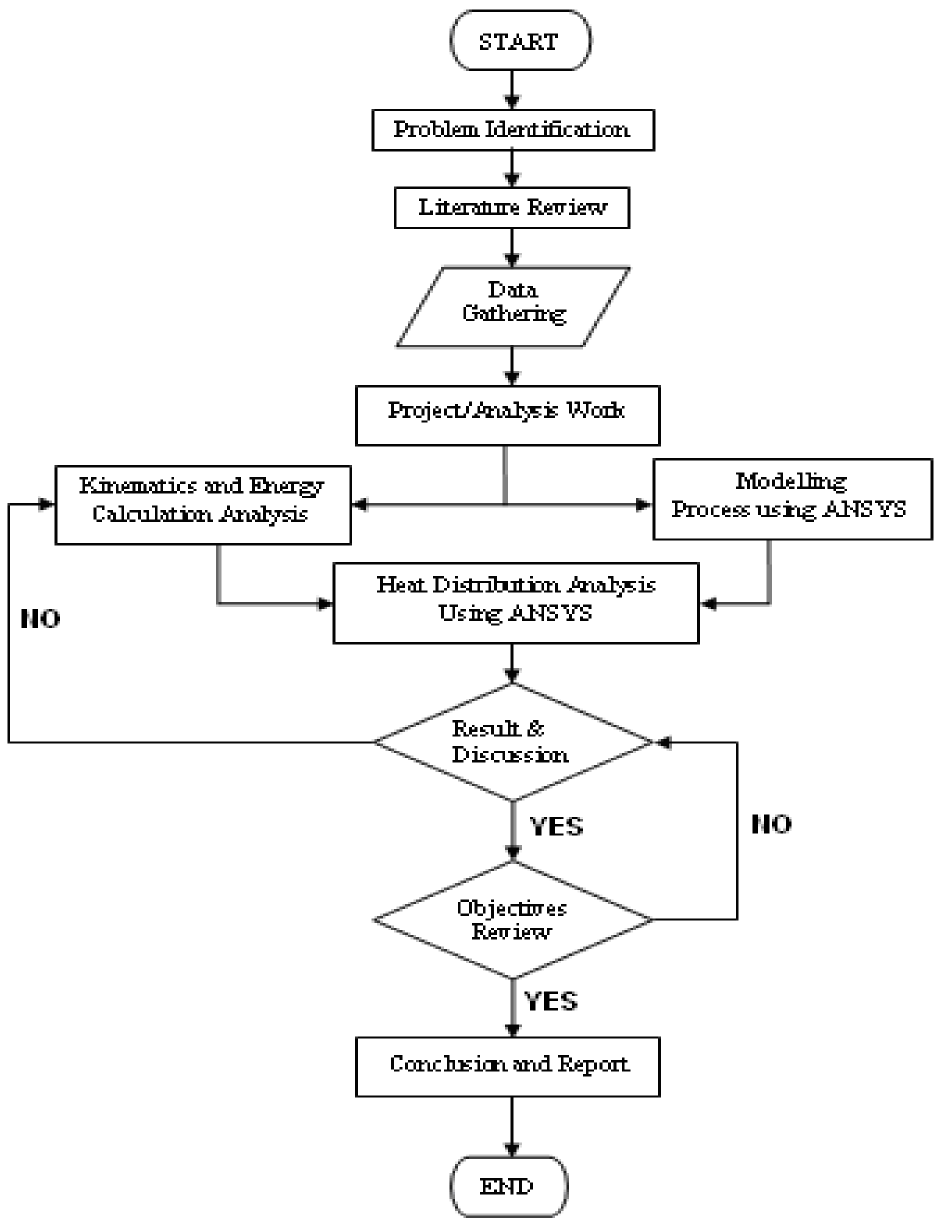


Figure 3.1: Project phases or flow chart

3.1 Literature Review

In order to understand further about the problem that going to be study, some literature review has been done. Most of the information is come from the book and discussion with the supervisor and knowledge full friends. Otherwise, to touch up the unclear information other sources has been use such as journal. In addition, few of information are taking from the internet sources or internet search engine.

3.2 Data Gathering

The data of the disc brake and brake pads specification has been found from the parts itself. First of all, the car specification data has been tabulated into Table 3.1 and the car considerable parameter is tabulated inside Table 1.1. By taking a typical disc brake and brake pad used by one of the national car as shown on Figure 3.2 and Figure 3.3. Determination of the measurement can be found by using reverse engineering process. The specific measurement of both items has been taken using vernier calliper and all the reading has been written down into table form as described on Table 3.1 and Table 3.2. All the readings are in millimetres (mm) and few in inches (inch) unit but its still need to be converted to millimetres unit.

Table 3.1: specification data for a typical car

No.	Parameter	Measurement	Unit
1	seating capacity	5	person
2	Gross vehicle weight (G.V.W)	1370	kg
3	brake type (ventilated disc)	9(1995)	inc
4	tyre size	175/70R13	inc
5	wheel size (steel)	13 x 5.0J	inc

Figure 3.2: Picture of ventilated disc (sample of study)

Figure 3.3: Picture of brake pads

Table 3.2: Parameters and measurement of typical disc brake

No.	Parameter	Measurement	Unit	Resource
1	Outer disc dia	224	mm	stock
2	Inner disc dia	141	mm	stock
3	disc thickness by layer	7	mm	stock
4	ventilation size (big /small)	D8 / D4	mm	stock
5	ventilation thickness	5	mm	stock
6	surface area	24×10^3	mm^2	stock
7	inlet Area of Cooling Vanes, A_{in}	56	mm^2	stock
8	Outlet Area of Cooling Vanes, A_{out}	191	mm^2	stock

Table 3.3: Parameters and measurement of typical brake pad

No.	Parameter	Measurement	Unit	Resource
1	big dia, D_o	224	mm	stock
2	small dia, D_i	140	mm	stock
3	overall length, L	116	mm	stock
4	pad support thickness, ∂s	5	mm	stock
5	pad thickness, ∂p	10	mm	stock
6	pad surface area, Ap	4×10^3	mm^2	stock

The braking parameters to be justified of severe braking conditions are getting from the discussion with the supervisor. The data or parameters involved for these conditions analysis such as the initial velocity, *U*, of the car, final velocity, *V*, of the car, cooling time of the disc brake, car deceleration and etc. By having all the basic parameters, the other subs parameters can be determine by plug it into the related formulas. The sub parameters are such as time interval for car to decelerate, angular velocity of the car and etc.

3.3 Simulation Input Calculation

The kinematics movement of the disc brake during braking is analyzed to identify several important subjects such as total time taken to stop, total distance taken to stop, total pad movement, angular deceleration, angular velocity, time interval of each pad movement, heat flux and total heat energy absorbed by disc. The calculation of each subject is performed at each pad movement, which is from one edge to another edge with displacement of *60 degree*. Several assumptions are made to perform the calculation:

- Disc is in stationary while the brake disc is rotating in counter-clockwise direction
- Brake distribution is 60% on front, 40% on rear
- Force distributed on one brake disc in is equal the total frictional force applied on rubbing surface and
- Kinetics energy is 100% converted to heat energy and only 95% of heat is absorbed by the disc and the rest 5% is absorbed by pad

In calculation procedures, first of all the total braking time and distance taken to stop the car is identified from Eq. (2) and Eq. (3) respectively. From it, the value of angular deceleration is obtained from Eq. (5) and then, the pad angular velocity for each instantaneous movement is calculated using Eq. (6) by knowing the constant angular deceleration and angular displacement. Next, the time interval of each movement is determined in Eq. (7). The total force required to brake the car is determined through Eq. (12). Then, the maximum frictional force exerted on rubbing surface is calculated using Eq. (13) due to the weight transfer ratio and from that amount of heat energy absorbed by the disc at that particular time is calculated using Eq. (14). Finally, the amount of heat flux applied for each pad movement is obtained by computed it using Eq. (15). The value of heat flux and accumulative time interval of each pad movement obtained will be as an input in ANSYS solution. All of the calculation is automatically done by Microsoft Excel by programming particular equation. Next, the graphs of calculated subjects are plotted against time to observe the value changes along the braking period.

3.4 Modeling Process

3.4.1 Using ANSYS V.9

Modelled of 3-dimensional disc are done in ANSYS environment in accordance to actual dimension obtained from actual disc measurement. Before that one assumption is made which is the temperature rise will be same if disc is assumed axisymmetric according to the same brake pressure applied. Therefore, modelling will be done for one side of disc brake only as shown in Figure 3.4. First of all, the 2-dimensional disc cross-sectional profile is drawn and area is created within the combination of lines. Next, these areas are revolved with 6 volume divisions and about a fixed axis which is located at center of the disc brake as described in Figure 3.5. The explanation of dividing the volume into 6 sections is that the pad can make a contact on surface of 60 degrees angle and leave none contact angle at 300 degrees angle. Thus, one contact region consists of one section of disc where as each section is 60 degree angle as shown in Figure 3.6. Every movement of the disc, the contact region would change according to the current location of the brake pad. Heat flux will be applied on these sections and no thermal load is applied on remaining sections. At the disc ventilation cut areas, adiabatic condition is applied which mean no heat transfer in and out occurs during the braking and acceleration.

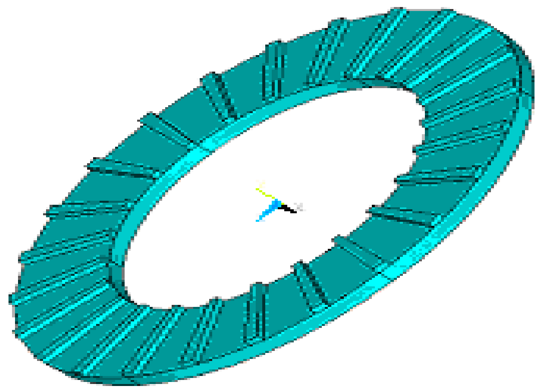
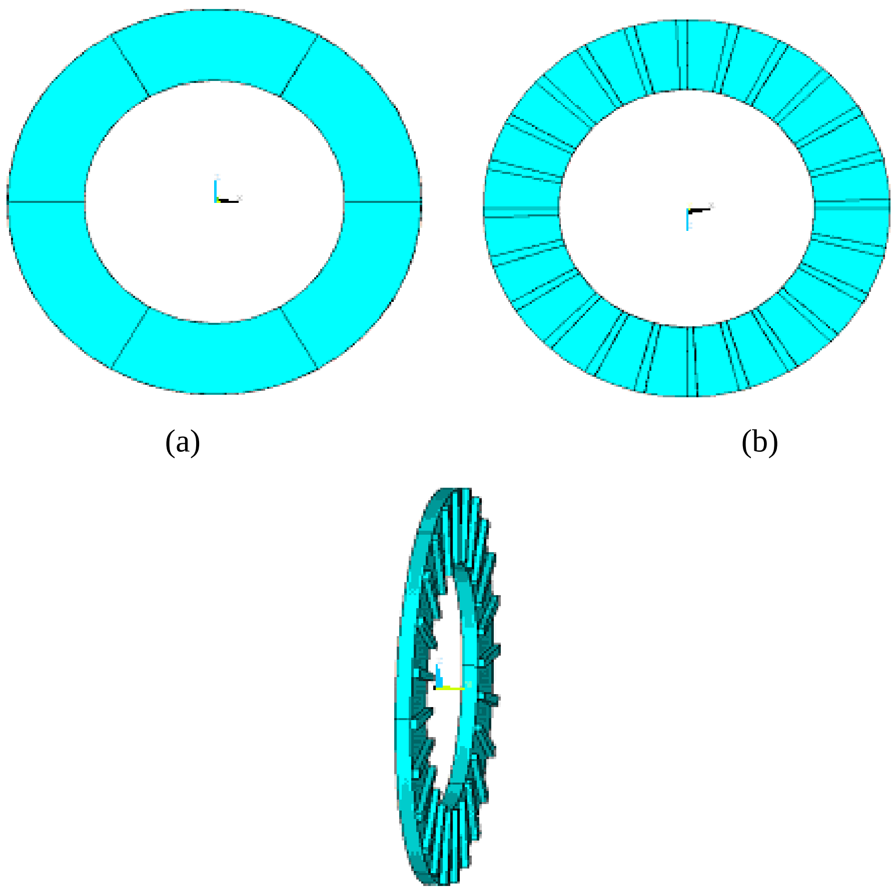


Figure 3.4: Three-dimensional model of disc brake with cutting sections of ventilated wall



(a) (b)

(c)

Figure 3.5: Picture of disc volume modelling in ANSYS, (a) Top View, (b) Bottom View, (c) Isometric View

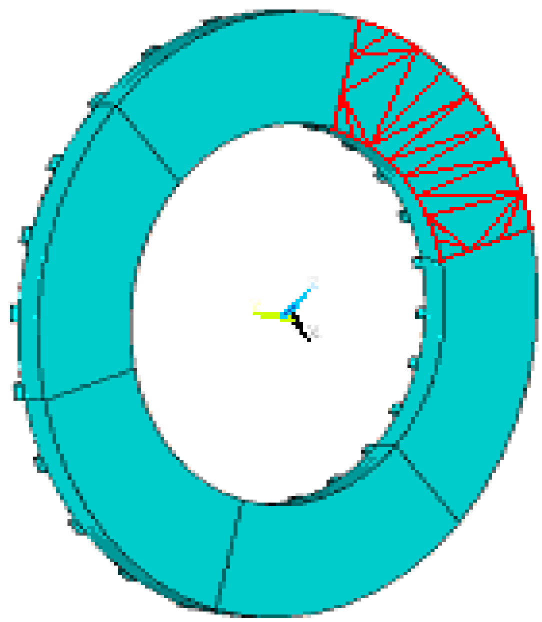


Figure 3.6: Sections highlighted represent pad contact area where heat flux is applied

Since the simulation is done in 3-dimensional model, SOLID90 is chosen as the element type. Main degree of freedom in this study is temperature. In that case, the model is meshed using wedged cubic as element shape. Element length appropriately is identified using try and error method until it provides best and most consistent meshed shape. This process is give big significant since the element shape also contribute to the accuracy of the simulation result. Finally, it is come out with very fine length element determined by the software and the software gave the best mesh for model volume which is shown in Figure 3.7 and Figure 3.8.

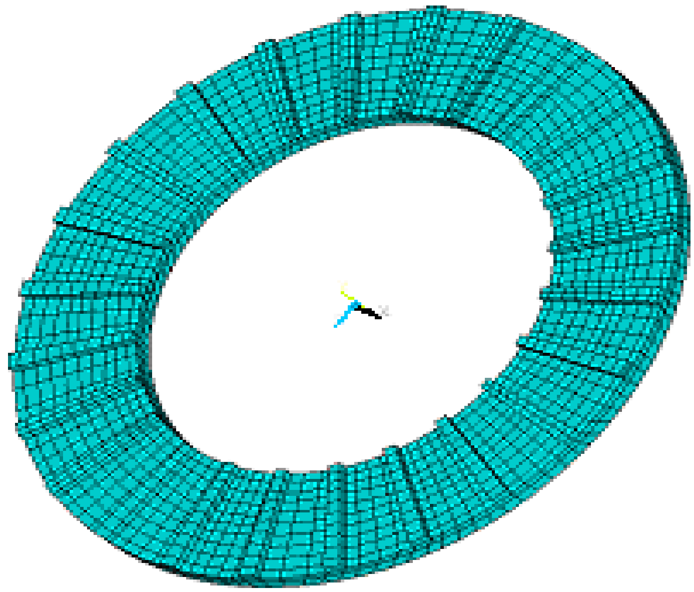


Figure 3.7: Mashed pattern on the surface volume of the disc brake

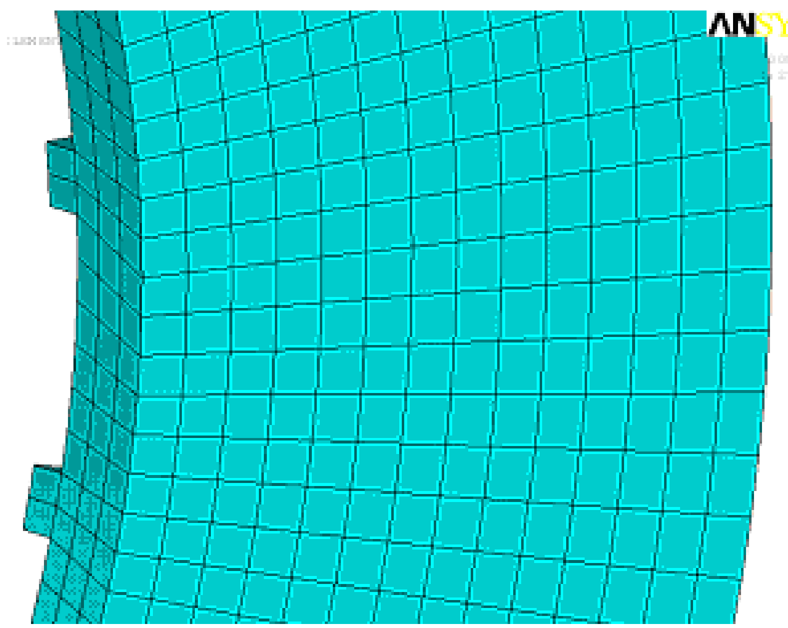


Figure 3.8: Mashed pattern at a closer view

3.5 Simulation Solving

The material properties are defined after the model is completed. Full transient analysis is selected with the initial disc temperature, T_0, of 30°C. In applying the thermal load to the model, it is assumed that:

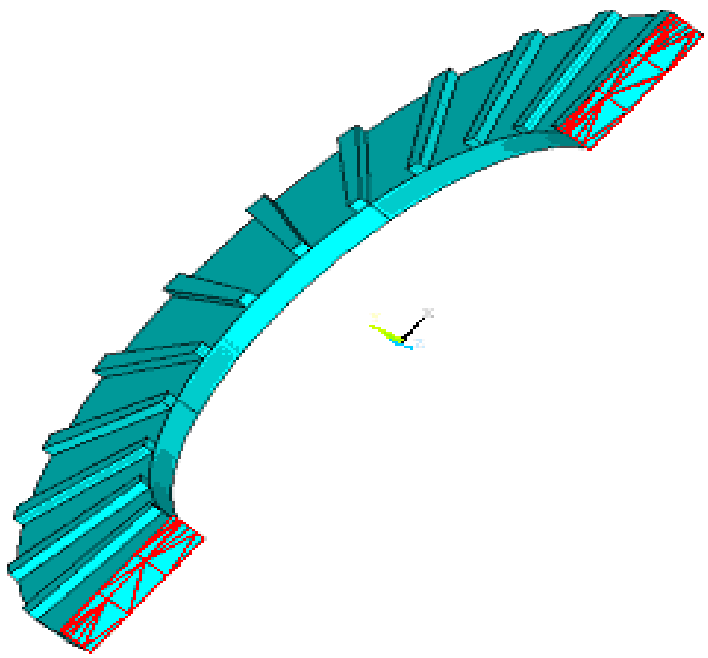
- Thermal properties are invariant with temperature
- Coefficient of friction remains constant during braking
- Heat flux applied is constant along the contact surface
- Film coefficient of convection is remains constant at all time.
- Result obtain will be the same for the other half of disc brake

Different amount of heat flux and surface location is applied for each brake pad movement. Therefore, to simulate different thermal values and applied areas, load step definition is used which represent the brake pad movement. Each load step will be solved subsequently without resetting the previous result. In other word, once a load step is solves, ANSYS will continue solves next load step by refers to previous result as initial condition and applies the load in current load step. For each load step, the value of heat flux is taken from the result of kinematics and energy calculation performed in Microsoft Excel.

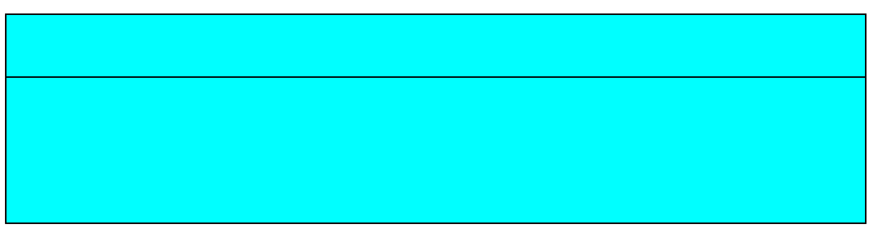
However, the applied areas are determined by reselecting the contact surface for current brake pad contact position. This step is repeated until total 95 of 60 degree brake pad movement when the disc angular velocity is reached zero. For each load step, time step needed to be set. The time step will limit the solution time in time interval defined for each brake pad movement. Since ANSYS used explicit to implicit solution, appropriate time step should be calculated for stability criterion.

3.6 Obtaining the Simulation Result

Result of temperature developed and distribution is observed when simulation is done. There are 2 ways of observing the result, which are read it on disc surface and inside the disc volume. For temperature on disc surface, the value can be directly obtained from the 3D model. However, for temperature inside the disc volume, the disc is cut-cross to see the inner side. The cut section is illustrated in Figure 3.9. All the required result data obtained is tabulated and necessary graphs are plotted for better understanding of the study. Since the ventilated disc have same size of contact area, therefore the result obtain is same like the other side and this condition is called axisymmetric analysis.



(a)



(b)

Figure 3.9: Disc cut section displayed in red lines (a) and

(b) Closer view of disc cut section

CHAPTER 4
RESULT AND DISCUSSION

4.0 Dynamic and Energy calculation Analysis

Analysis of dynamic and energy calculation of a disc brake is done. Therefore, the dynamic behaviour of the disc brake and brake pads can be seen during the braking applications at periodical braking condition. Total time of kinetic energy is converted to heat energy during brake application until the car is fully stopped is about 4.4107 seconds. Regarding to the calculation analysis, there was found that the total number of brake pad movement at both sides is 95 simultaneously in order to get the car fully stopped. Therefore, graphs consist of total number of pad movements against time can be plotted. By looking at the Figure 4.1, shown the angular velocity is decreasing against time linearly from 45 rad/s to 0 rad/s. This is occurred because of the constant deceleration at lateral direction is assumed during braking application as much as 0.3g.

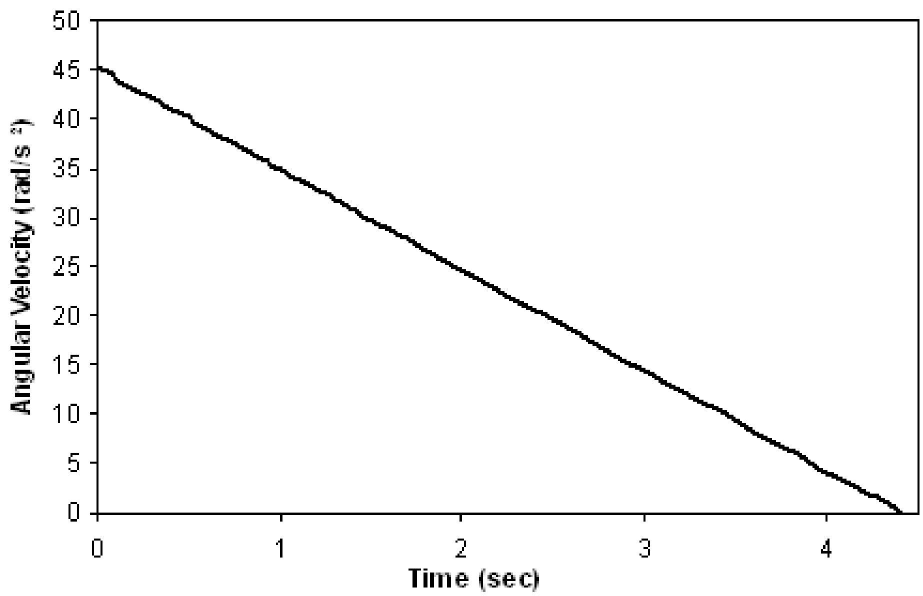


Figure 4.1: Graph of angular velocity against time

Thus, from the result of disc brake velocity, the amount of heat energy generated due to kinetic energy of the car is computed. Therefore, until 4.4107 seconds, the total amount of heat absorbed by disc brake of a car during 1st, 2nd and 3rd braking conditions is about 44.28kJ each. The total mount of heat generated during the braking application is remain constant because of the constant deceleration has been assumed during the braking and constant speed reduction from 120 km/h to 0 km/h are taken into account. From Figure 4.2 shows that, the graph of heat energy absorbed in the disc brake is exponential curve graph for these three braking applications. The lines for three braking conditions are lies on the same curve line because of same amount of heat energy absorbed during brake application according to the theory. Therefore, the different of these three conditions can be observe in the temperature analysis or graph when cooling process is apply during the acceleration.

Therefore, no matter how many times the brake was applied, the total heat amount will remain the same due to the constant deceleration of the car. Meanwhile, the curve of heat energy graph is become leaner is because of every time, when the car is likely slowing down at the desired velocity, the rotational velocity is reduced and it will lower the frictional heat generated. When heat generated is lower, the amount of heat transfer to the environment will increase in order to cool down the disc called cooling process. Hence, the less heat energy will absorb, the leaner of the heat energy of the graph will be constructed.

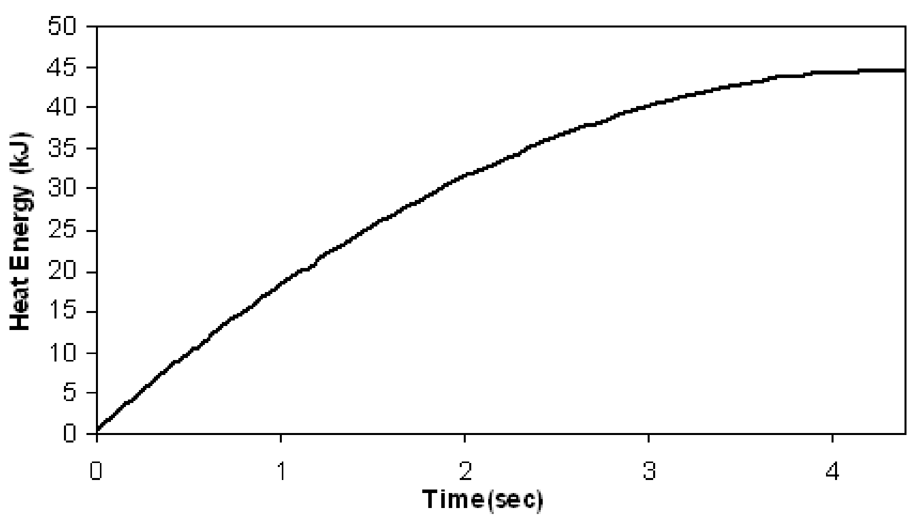
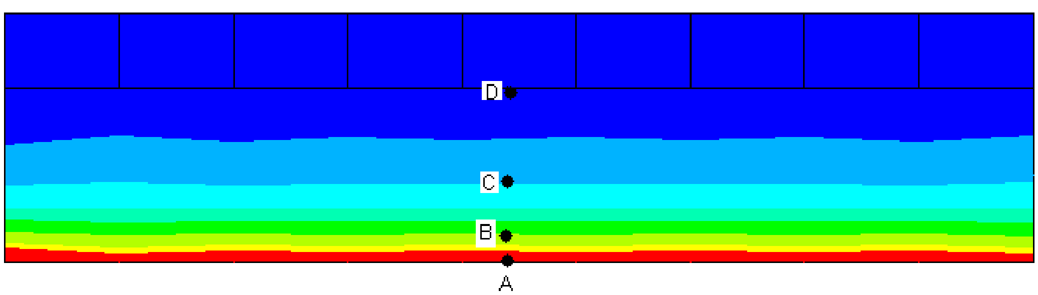


Figure 4.2: Graph of heat energy against time

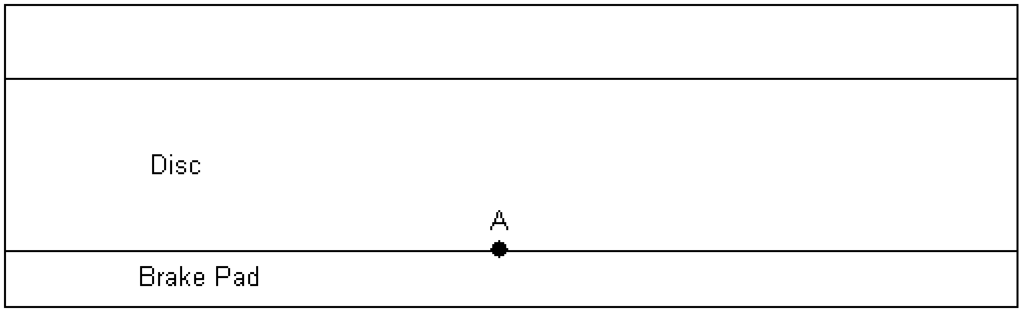
4.1 Temperature Development

Frictional heat created will result in temperature rise on the rubbing surface. At one point, it is subjected to heating and cooling alternately since the cooling is occurred there when the car start to accelerate after first and second brake applications. First of all, 1 node is selected at disc brake cross-section view to investigate the temperature development along 3 braking period as illustrated in Figure 4.3.

The maximum frictional heat flux generated during three brake applications at one friction contact area is 2.5043 MW/m^2. However, the value of heat flux is decreasing at throughout the time as shown in Figure 4.3. The heat flux is defined in term of time and it was depends on rotational deceleration rate. Therefore, the car was assumed to be constant deceleration and result to the constant angular deceleration of the disc brake. Hence, the heat flux created will decreasing at constant rate. The higher time interval for the car to decelerate, the lower heat flux will be created. The operation ambient temperature of 30^{o}C and the coefficient of convection considered is 50 W/m^2.K based on reference from experimental data source.



(a)



(b)

Figure 4.3: Point A on disc brake at figure (a) and (b) shown where the rubbing surface are occurs.

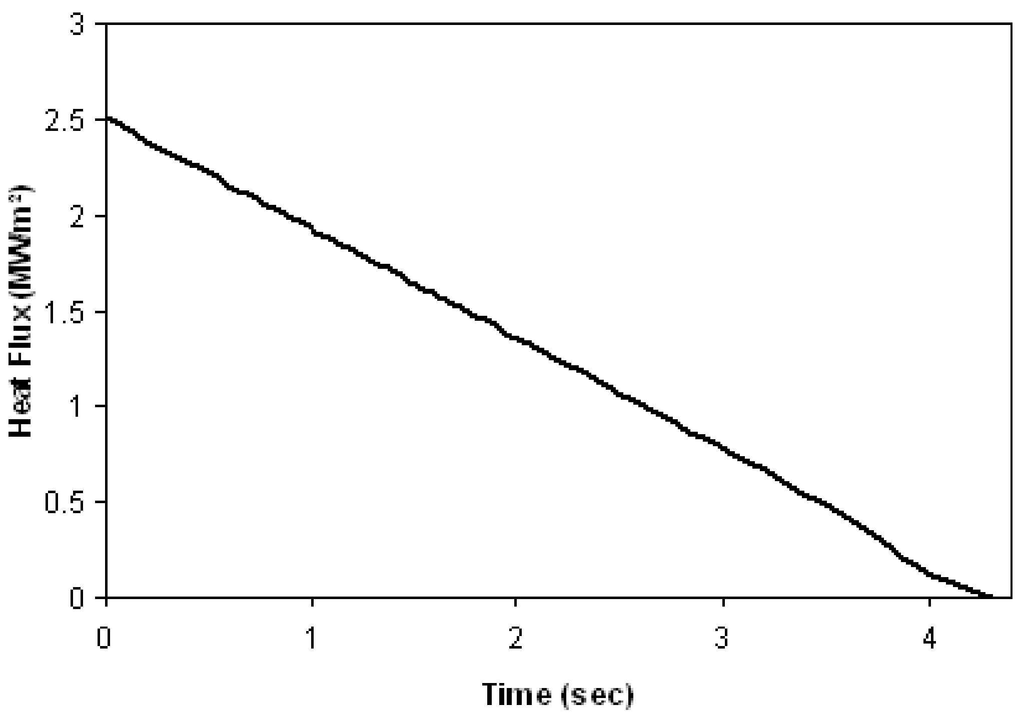


Figure 4.4: Graph of heat flux against time

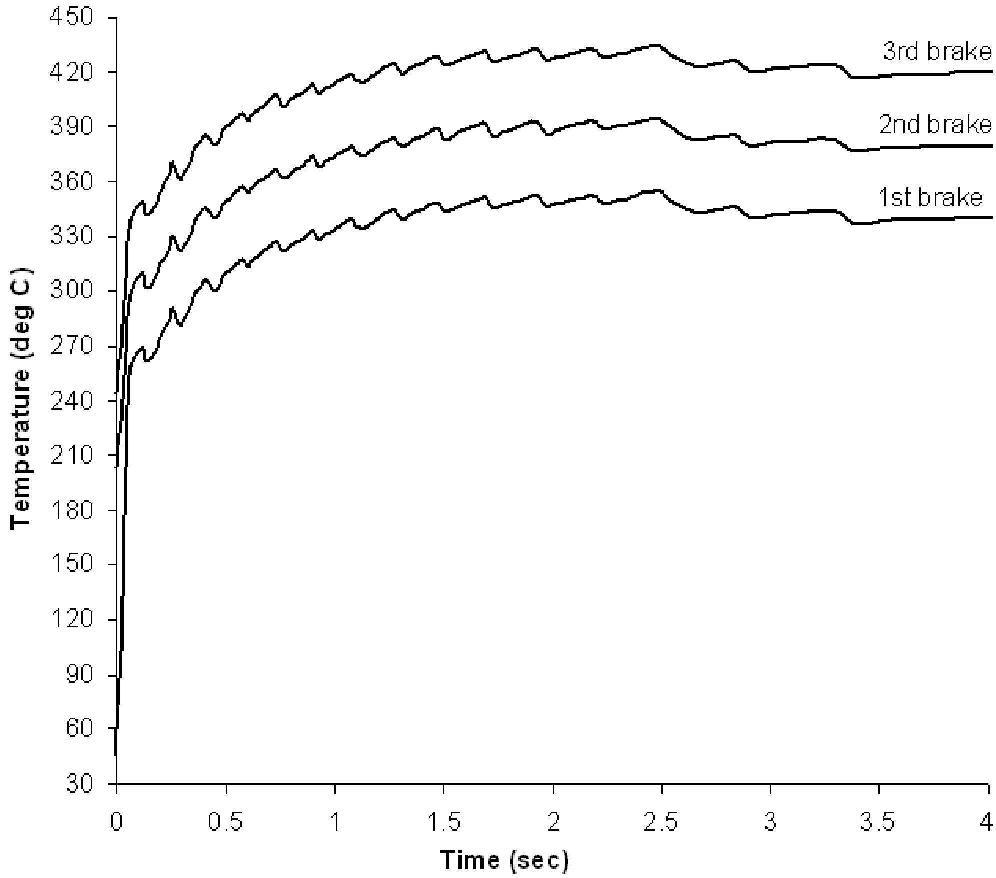


Figure 4.5: Time history temperature along 3 braking periods

Therefore, the result of temperature develop is presented at Figure 4.5. It has shown that temperature at node A where is located on the rubbing surface rise dramatically from the initial temperature of 46^{o}C. The temperature increases is in fluctuating manner since the heating and cooling occur alternately due to the pad movement on the disc. Whenever the pad is rubbing at one surface node, heat flux is occurred in direction from pad to disc surface and then it will rise the temperature of the disc. While the pad is not makes contact, the node is cooled and resulting to temperature drop.

The peak temperature is obtained at 2.50 second with value of 354^{o}C. After 1.90 second, the temperature will drop back to the final temperature of 333^{o}C at the end of first braking application. When the car was stopped at first brake application, in few seconds the car is required to accelerate about 2 minutes in order to reach the experimental velocity at 120 km/h same as first attempt. During acceleration, cooling is occurred on the disc brake as a result of the initial disc temperature is slightly lower compared to the end of first brake application. The initial temperature for the second brake application is about 204^{o}C.

The peak temperature for second brake application shown occurs at 2.50 second and the temperature is 394^{o}C exactly at the same time of first brake application. Similar thing happened on this 2nd application where as the temperature at the end of brake application was drop until 373^{o}C due to the decreasing in velocity. The heating and cooling fluctuation is occurs alternately and it is because of the pad movement on the different disc surface section. The free contact disc surface will experience the cooling process but the friction surface will get the heat flux from the brake pad due to the braking force applied on the brake pedal.

Same goes to the third brake application, when the car reach the experimental velocity at 120 km/h, the initial disc temperature was dropped until 244^{o}C due to the cooling occur when the accelerate. The maximum temperature for this braking condition is 433^{o}C at 2.49 second. The temperature is dropped slowly until 413^{o}C at the end of 3rd brake application due to the elimination of velocity of the car. When velocity is decreasing, the heat flux will decrease alternately.

For node B, C and D, their highest temperature is 308^{o}C, 92^{o}C and 50^{o}C respectively which all of them are reached at the end of braking (4.41 second) as shown at Figure 4.7. However, Node D seems to have more capability to transfer the heat because all the heat is transferred via conduction and the location itself is within the ventilation area. While more heat is dissipated to ambient air via convection at non ventilation area of disc brake which is this method of transferring is less effective than conduction. As the result, temperature at node D becomes lower than temperature at non ventilation area due to fast and effective heat transfer. Node on top of ventilation area supposedly has the lowest temperature because it is located at the furthest thickness as it takes longest time for heat to be transferred there.

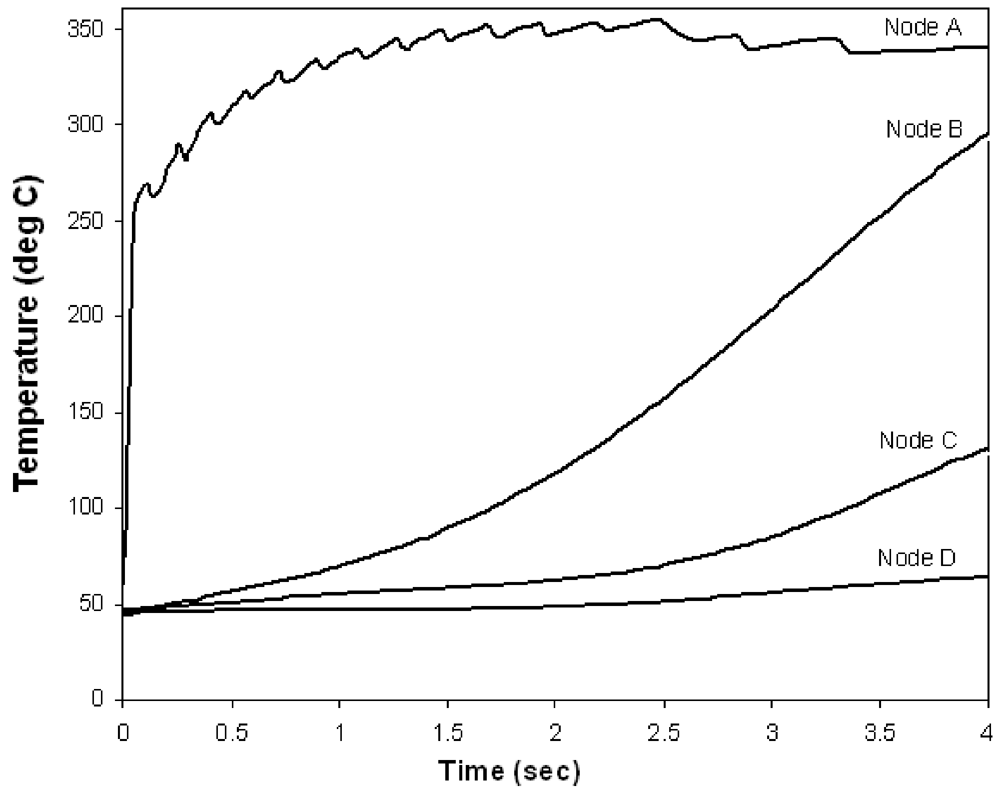


Figure 4.6: Time history temperature along nodes A, B, C and D

4.2 Temperature Distribution

Heat generated on the disc and brake pad contact surface is transferred throughout the disc body. In this case study, the disc brake material is assumed homogenous. Thus, the temperature is expected to be well distributed within the material. Several paths from one node to node are defined to investigate the temperature distribution at one disc brake cross-section. Paths are defined as below and each node is shown in Figure 4.7.

- Path 1 (node 1 to 2) will represent the horizontal path along the friction surface where heating take place.
- Path 2 (node 3 to 4) will represent the horizontal path along the outer disc surface in between the ventilation where convection for cooling take place.
- Path 3 (node 5 to 6) represent the vertical path from friction surface to outer surface.

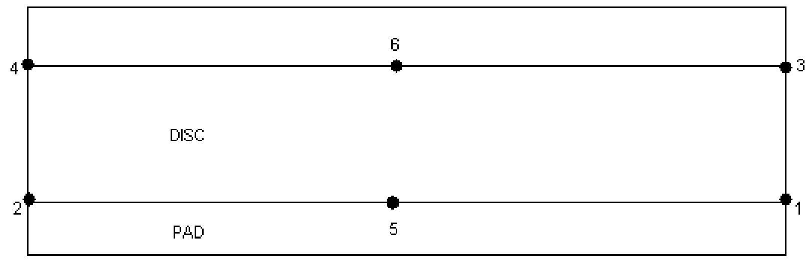
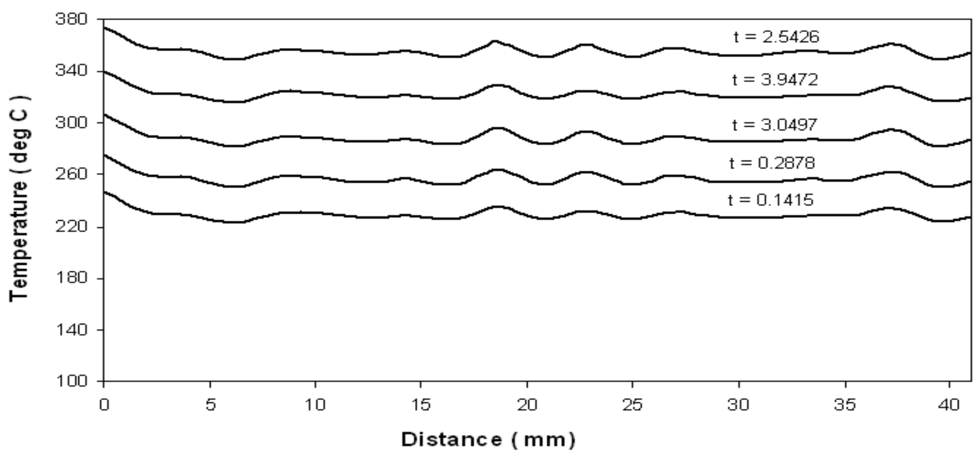
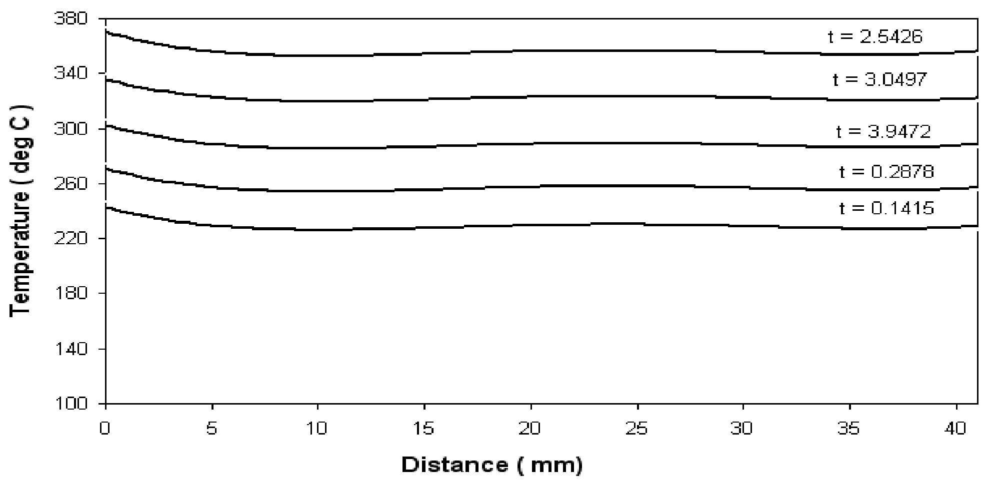


Figure 4.7: Nodes that defining the path

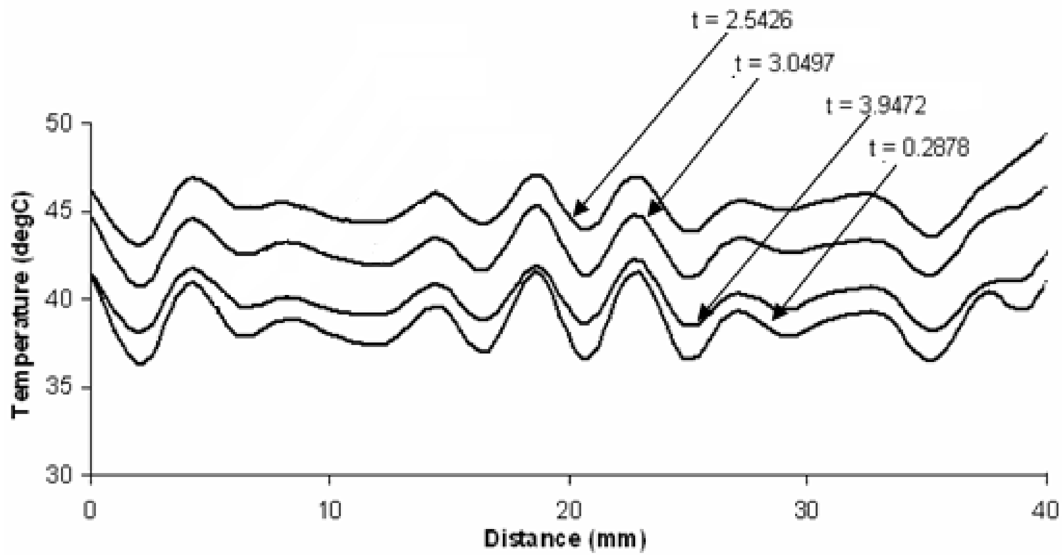


(a)

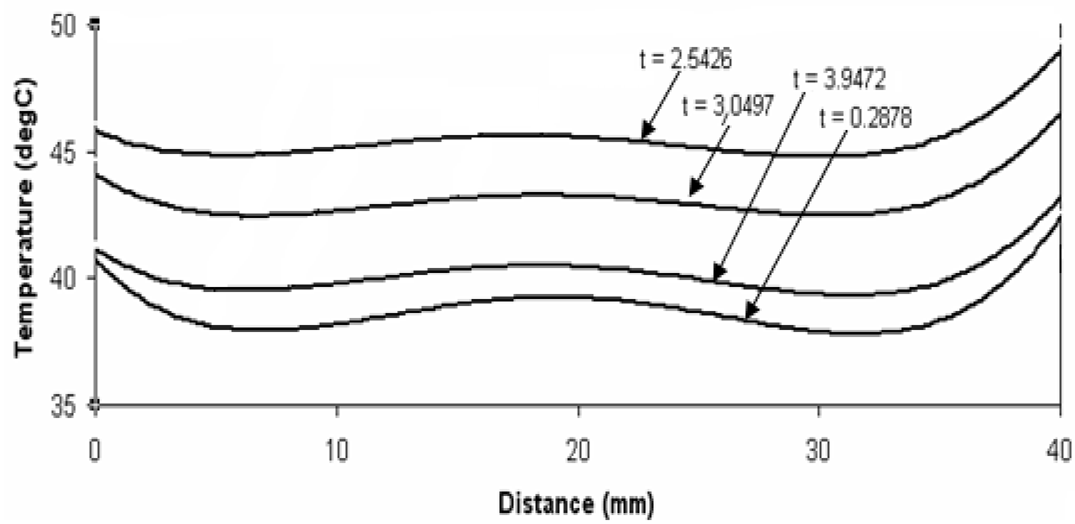


(b)

Figure 4.8: Temperature distribution along Path 1 (a) and (b) the trend line for path 1 for smoother lines



(a)



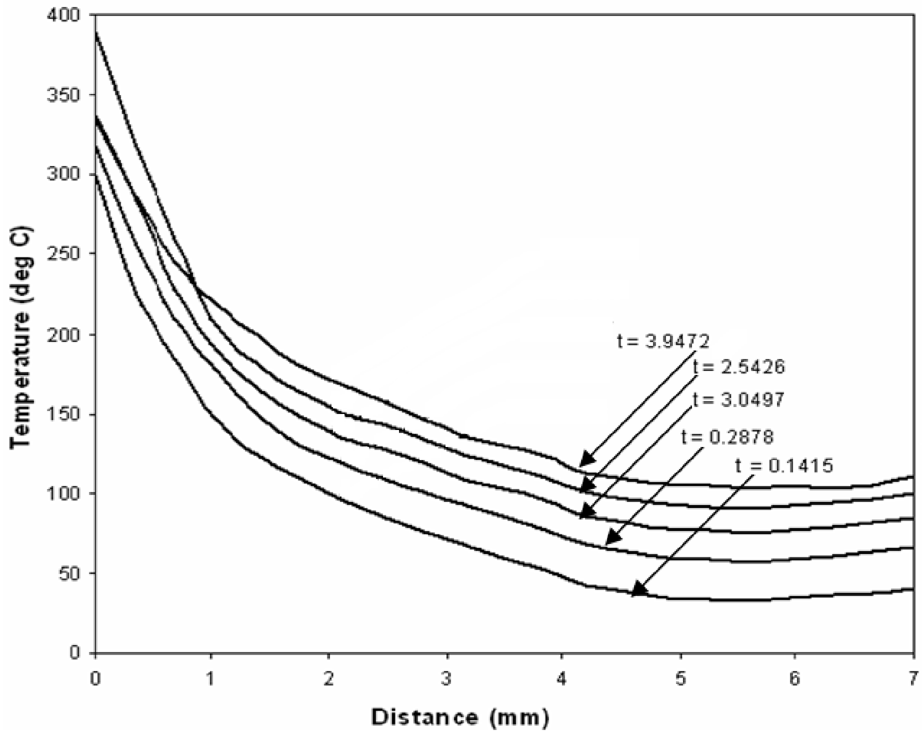
(b)

Figure 4.9: Temperature distribution along Path 2 (a) and (b) the trend line path 2 for smoother lines

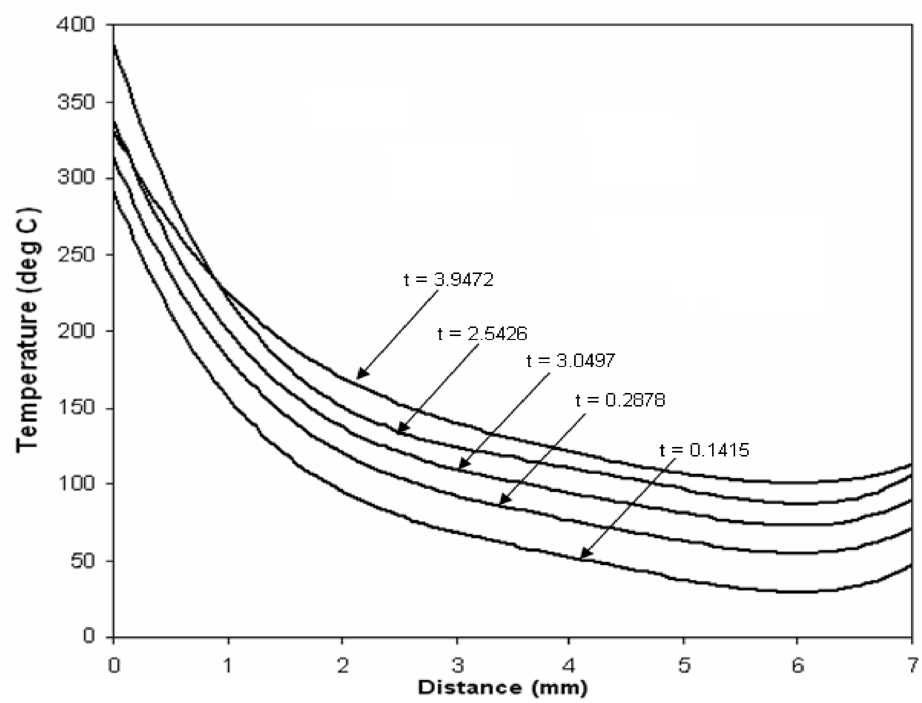
4.2.1 Temperature along Horizontal Path

Path 1 and Path 2 is defined as horizontal paths. Temperature at the specific time is noted along these paths in order to investigate the temperature distribution on the horizontal direction which is at the disc and pad contact surface. Figure 4.8 shows the temperature distribution along Path 1. This event shows that temperature distribution on the rubbing surface is affected by outer geometry of disc brake which can be see at both figures at (a). Using the trend line function we can see that the temperature remains constant along the path, since the material is assumed homogenous at Figure 4.8 (b). On the other hand, at distance of 0.4 meter,

For Path 2, the temperature distribution is as shown in Figure 4.9. At the same thickness, the temperature is constant along the time (refer to Figure 4.9(b)). But, the thickness is increases due to location of ventilation parts. At greater thickness, the temperature rise of node is slower than the one at the lower thickness because more time is taken for heat to be transferred to the outer surface. As the result, the temperature starts to fluctuate slowly at certain distance and then rise back and repeat again until reach the distance of 0.40 meter. The cause of late temperature drop and rise is due to some of the heat is already transferred from hot to cool molecules slightly before the changes of geometry. However, the small differences of temperature fluctuation can be assumed constant as a straight line or constant at the specific thicknesses as described on Figure 4.9(b).



(a)



(b)

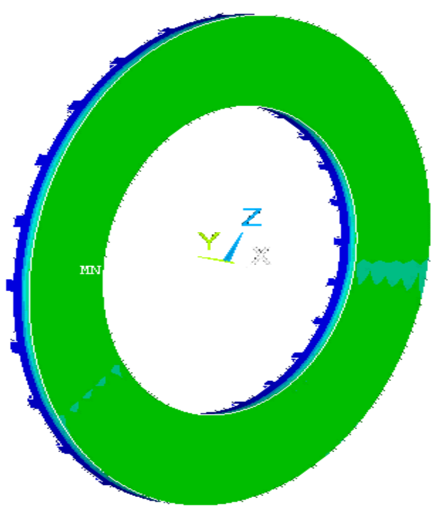
Figure 4.10: Temperature distribution along Path 2 on graph (a) and (b)trend line path 3 for smoother line

4.2.2 Temperature along Vertical Path

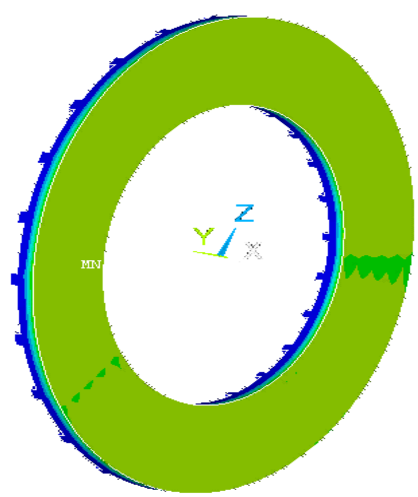
Path 3 is defined as vertical path. The temperature along these paths is investigated to observe its distribution from inner (rubbing surface) to outer surface (ventilation area) of the disc bake. The temperature distribution also is recorded at same specific time as in vertical path in path 3. From Figure 4.10, highest temperature for Path 3 is experienced at distance of inner surface at any time because at this point where the frictional heat flux application takes place. Then, the temperature is gradually drops as the thickness increase. The temperature is far lower at the end of the thickness because the rate of heating at rubbing surface is higher than faster the heat is been transferred. However, the extra thickness on the ventilation point promotes the disc with effective heat transfer via conduction. As a result, the temperature at ventilation is slightly lower than outer disc temperature at the same thickness.

4.3 Temperature Contour

In this case, the temperature contour is plotted according to the various braking times to observe in the temperature distribution in clearer view. The temperature contour on disc surface is shown in Figure 4.11. While the contours at disc brake cross-sectional area is shown in Figure 4.12.



(a)



(b)

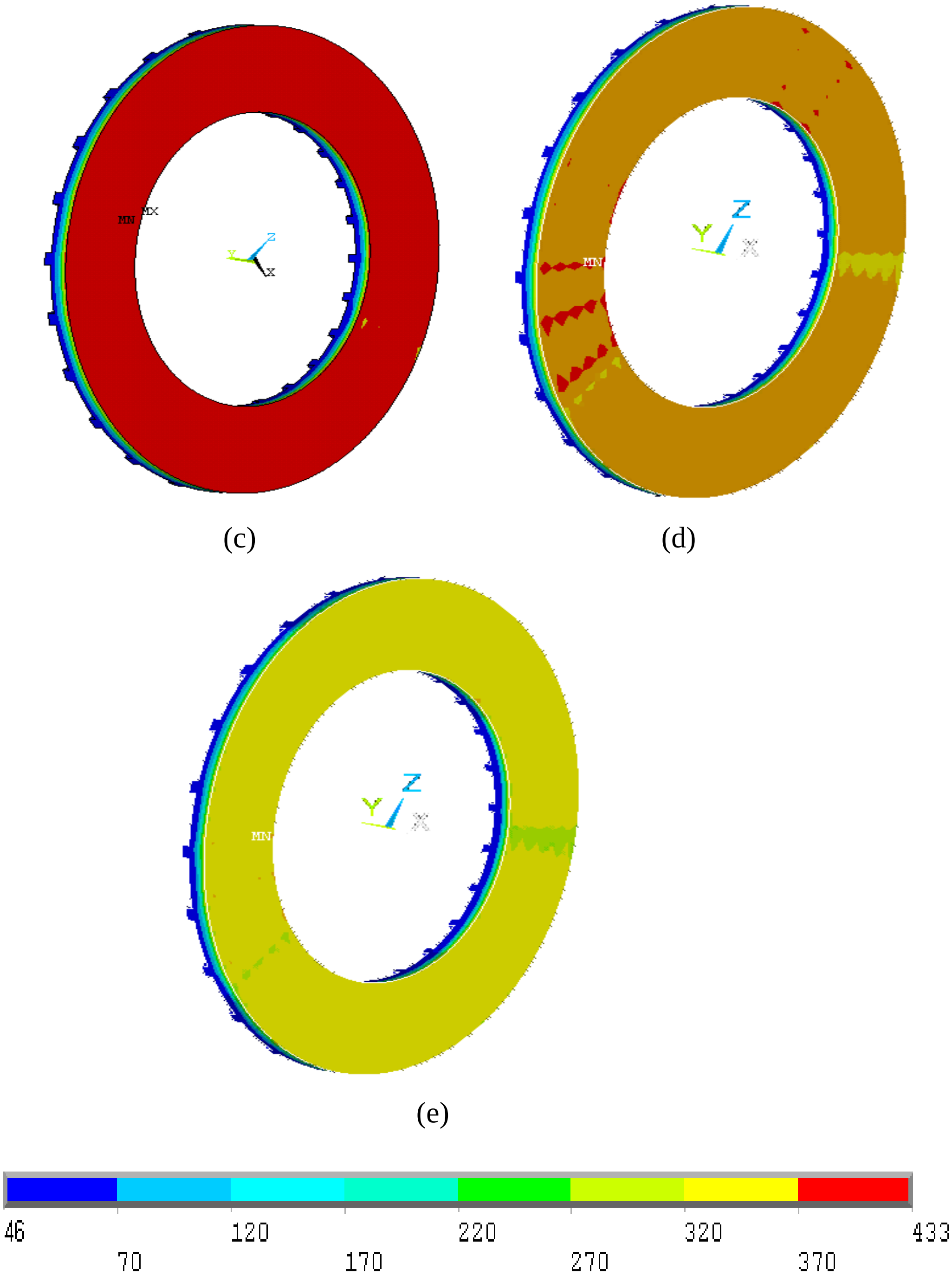


Figure 4.11: Surface temperature contours at (a) 0.1415s, (b) 0.2878s, (c) 2.5476s, (d) 3.0497s and (e) 3.9472 s

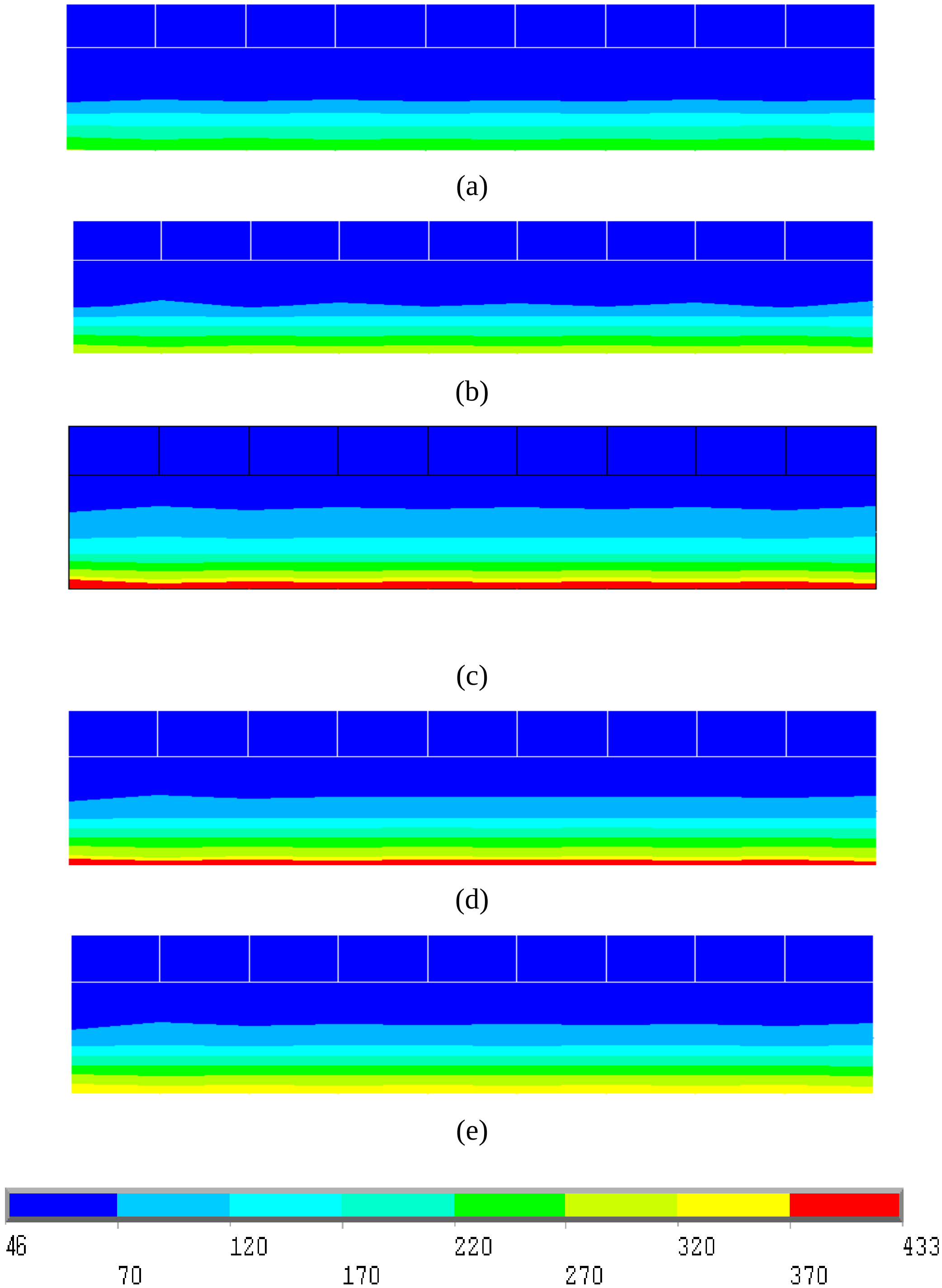


Figure 4.12: Surface temperature contours at (a) 0.1415s, (b) 0.2878s, (c) 2.5476s, (d) 3.0497s and (e) 3.9472 s

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

Based on the theoretical and simulation investigations in this study, the following observations and conclusions are made:

- The approximate time duration to stop the car under horrible braking condition is 4.4107 s. At some stage in this time period, maximum temperature on disc rubbing surface may achieved up to 433^{o}C by 2.49 s after braking started. However, at outer surface, maximum temperature reached at the end of braking with value of 413^{o}C.
- The temperature rise at one point is going up and down alternately along braking period. At the early of braking, the heating gradient is higher than cooling gradient. But, after 2.50 s, the cooling gradient becomes higher than heating gradient.
- The temperature is equally distributed along the heating line or rubbing surface at the end of the braking and it is due to the contact between brake pad and disc brake. At outer surface, the temperature also equally distributed along the line even it is fluctuate little due to the ventilation parts the temperature level will change as there is geometry variant with different thickness.
- The temperature difference between inner and outer surface of disc decrease as the thermal conductivity is increase.
- The temperature curve level increase as the heat capacity is decreasing.

5.2 Recommendation

The following recommendations are presented to further improve the understanding of

Temperature rise and distribution in brake disc.

- In the future, the experiment for observing the temperature development and its distribution should be conducted to validate the simulation results approximation. No data validation and comparison is made in this study due to no experimental data.
- The analysis with thermal properties that varies with temperature, inconsistent brake pressure and coefficient of friction and thermoelastic instability consideration may be useful to improve the approximation to real braking condition.
- Good understanding in real braking phenomenon will helps a lot to undergo in this study.
- More case should be made and more uncertainty should be considered. This really helpful in improving the brake design and performance and enhance the safety.
- The analysis method can be reduced to two-dimensional analysis without affecting the results obtained in three-dimensional method because it less time consuming in two-dimensional analysis.

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APPENDICES

1) Sample Calculation:

Initial velocity , U = 120 km/h

= 120 x 1000 / 3600

= 33.333 m/s^2

Final velocity, V = 0 km/h

Deceleration, D or a = 0.3 g

= 0.3 x 9.81

= 2.943 m/s^2

Tire dia, D_{tire} = (175*0.7/1000+0.5*13*2.54/100) x 2

= 0.5752 m

Time taken for car braking condition, $t = \dfrac{v-u}{a} = \dfrac{0-12.9806}{-2.943} = 4.41 \text{ sec}$

Pad contact area, $A_{friction} = \dfrac{\pi D_{disc}\theta}{180} \times W_{pad} = \dfrac{\pi \times 0.224 \times 60 \times 0.0171}{180} = 0.004 \text{ m}^2$

Initial angular velocity of disc, $\omega_0 = \dfrac{2u}{D_{tyre}} = \dfrac{2 \times 12.9806}{0.5752} = 45.13 \text{ rad/s}$

Angular acceleration of disc, $\alpha = \dfrac{2a}{D_{tyre}} = \dfrac{2 \times -2.943}{0.5752} = 10.233 \text{ rad/s2}$

Angular velocity of next pad movement,

$$\omega_1 = \left(\omega_0{}^2 + 2\alpha\theta\right)^{0.5} = \left(45.13^2 + 2\times 10.233\times\frac{60\eta}{180}\right)^{0.5} = 44.892\ \text{rad/s}$$

Time interval between pad movement, $\Delta t = \dfrac{\omega_1 - \omega_0}{\alpha} = \dfrac{44.892 - 45.13}{-10.233} = 0.0233$ sec

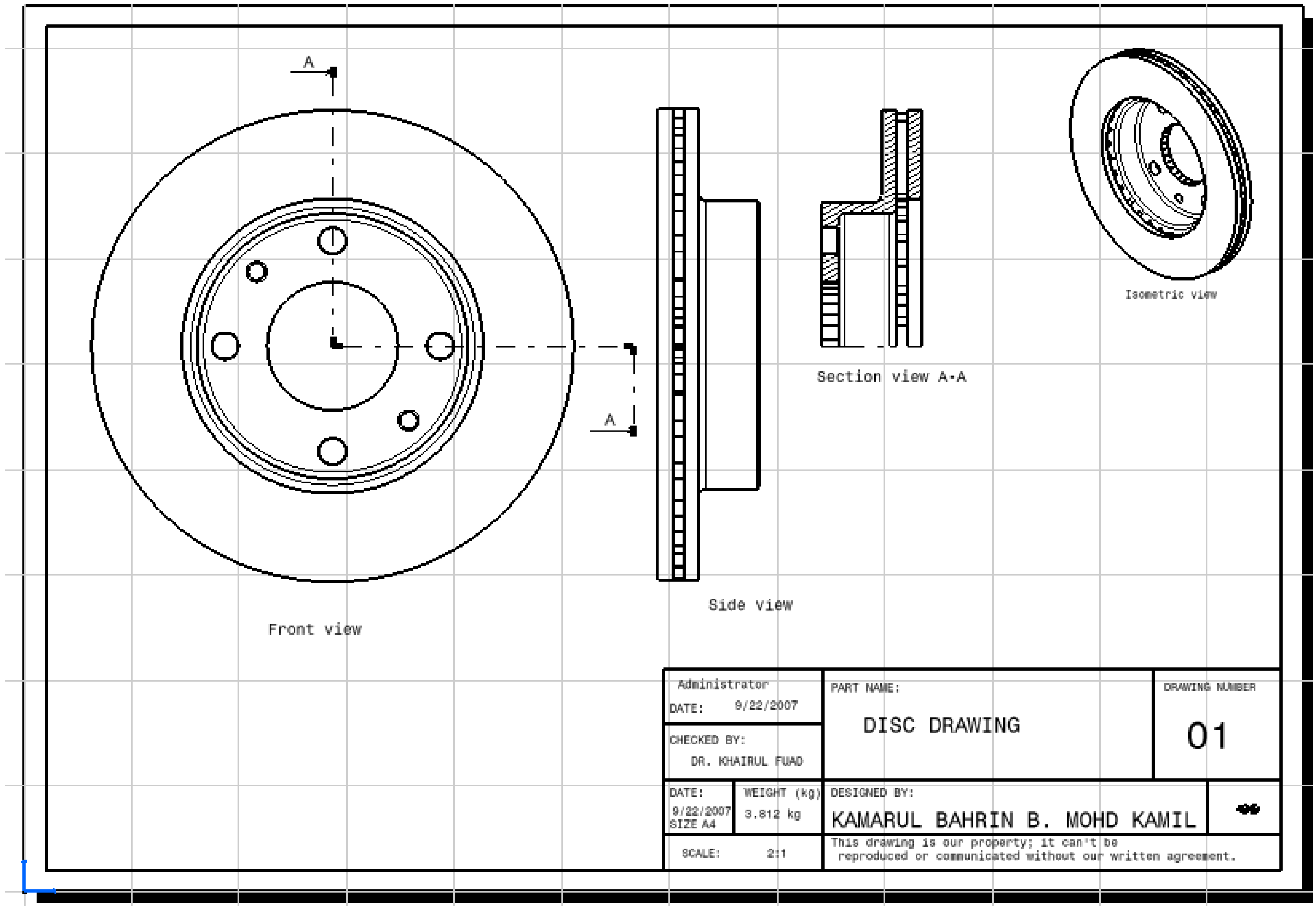
Heat energy absorbed by disc brake,

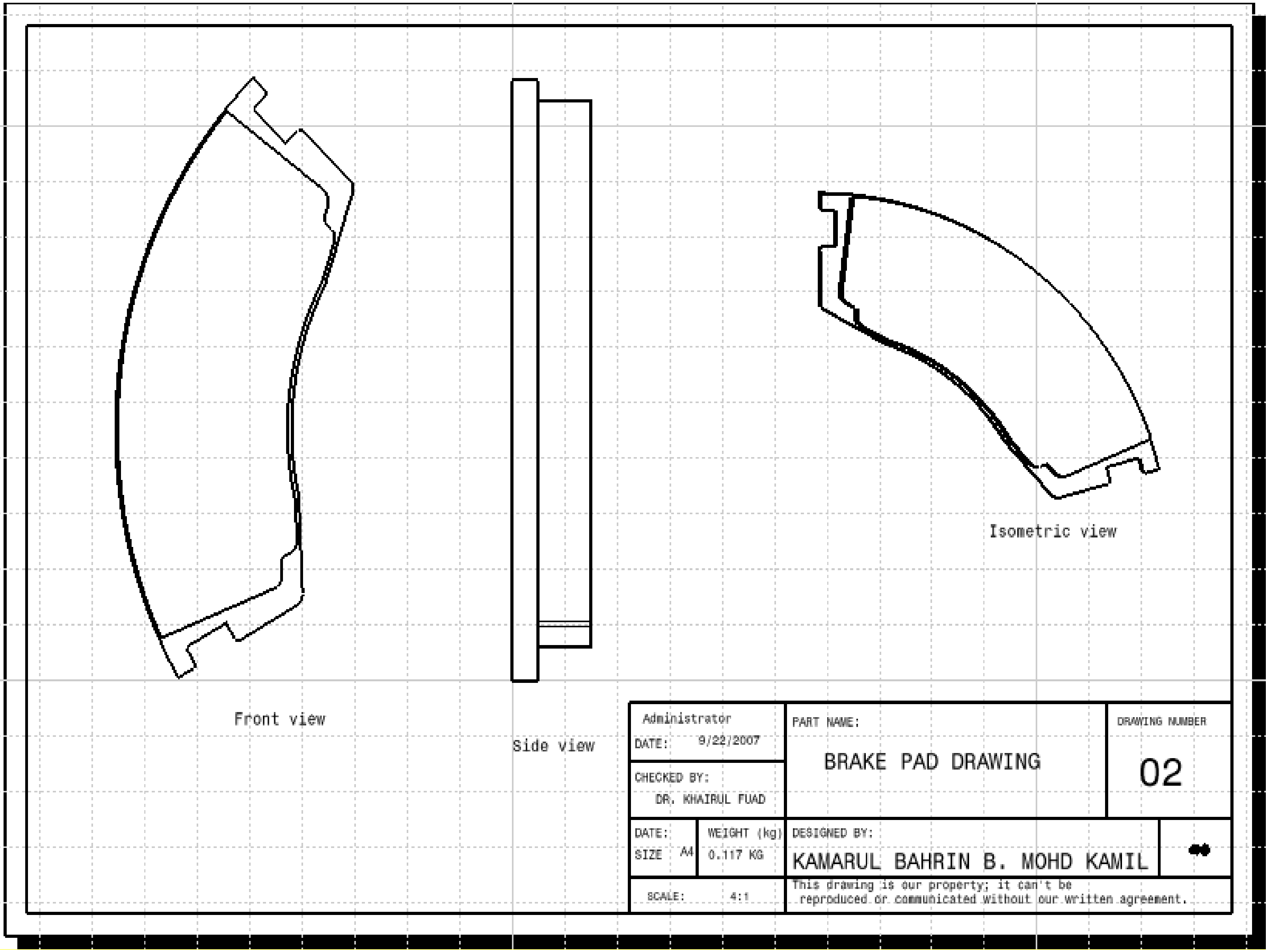
$$\begin{aligned}\Delta Q &= 0.95T = 0.95\times 0.5\times m\times\left(u^2 - v^2\right)\\ &= 0.95\times 0.5\times 0.5535\times\left(12.9806^2 - 12.9121^2\right)\\ &= 0.46607kJ\end{aligned}$$

Heat flux applied on contact area,

$$q = \frac{\Delta Q}{2\Delta t A_{friction}} = \frac{0.46607}{0.0233\times 0.004\times 1000\times 2} = 2.5004\ \text{MW/m2}$$

2) Disc brake and brake pad drawing using CATIA V5 R14





Front view
Side view
Isometric view
Administrator
DATE: 9/22/2007
CHECKED BY:
DR. KHAIRUL FUAD
PART NAME:
BRAKE PAD DRAWING
DRAWING NUMBER
02
DATE:
SIZE A4
WEIGHT (kg)
0.117 KG
DESIGNED BY:
KAMARUL BAHRIN B. MOHD KAMIL
SCALE: 4:1
This drawing is our property; it can't be reproduced or communicated without our written agreement.

3) Kinematics Calculation Using Microsoft Excel

Calculation for three Braking Condition (120 km/h – 0km/h)								
Parameters								
	initial velocity, U =	12.98055556	m/s	total distance,s =	28.62637148	m		
	final velocity, V =	0	m/s	total time, t =	4.410654283	s		
	Deceleration, Dx =	-2.943	m/s^2	total of rotation =	15.84155901	rev		
	wheel load, W =	0.5535	tons	angular acc =	-10.23296245	rad/s2		
	pad area, A =	4.00E-03	m^2	angular disp =	1.047197333	rad		
	tyre dia, Dw =	0.5752	m	pad disp =	0.095556757	m		
	avg rubbing dia , Dr =	0.1825	m	brake force req =	1.6289505	kN		
	pad angle =	60	deg					
no of pad movement	**time interval (second)**	**time (second)**	**angular velocity ω (rad/s)**	**Linear velocity (m/s)**	**Kinetic energy (kJ)**	**Heat absorbed (kJ)**	**total heat energy (kJ)**	**heat flux (MW/m2)**
0	0	0	45.13405965	12.98055556	0.490597461	0.466067588	0.466067588	0
1	0.02326328	0.02326328	44.89600738	12.91209172	0.490597461	0.466067588	0.932135177	2.50E+00
2	0.023387288	0.046650568	44.65668614	12.84326293	0.490597461	0.466067588	1.398202765	2.49E+00
3	0.023513301	0.07016387	44.41607541	12.77406329	0.490597461	0.466067588	1.864270353	2.48E+00
4	0.023641373	0.093805243	44.17415412	12.70448673	0.490597461	0.466067588	2.330337942	2.46E+00
5	0.023771561	0.117576803	43.93090064	12.63452702	0.490597461	0.466067588	2.79640553	2.45E+00
6	0.023903923	0.141480727	43.68629269	12.56417778	0.490597461	0.466067588	3.262473119	2.44E+00
7	0.024038522	0.165519248	43.4403074	12.49343241	0.490597461	0.466067588	3.728540707	2.42E+00
8	0.02417542	0.189694668	43.19292123	12.42228415	0.490597461	0.466067588	4.194608295	2.41E+00
9	0.024314684	0.214009352	42.94410999	12.35072603	0.490597461	0.466067588	4.660675884	2.40E+00
10	0.024456382	0.238465734	42.69384875	12.2787509	0.490597461	0.466067588	5.126743472	2.38E+00
11	0.024600588	0.263066322	42.44211186	12.20635137	0.490597461	0.466067588	5.59281106	2.37E+00
12	0.024747374	0.287813696	42.1888729	12.13351985	0.490597461	0.466067588	6.058878649	2.35E+00
13	0.02489682	0.312710517	41.93410467	12.0602485	0.490597461	0.466067588	6.524946237	2.34E+00
14	0.025049007	0.337759524	41.67777913	11.98652928	0.490597461	0.466067588	6.991013825	2.33E+00
15	0.025204019	0.362963543	41.41986735	11.91235385	0.490597461	0.466067588	7.457081414	2.31E+00
16	0.025361945	0.388325488	41.16033951	11.83771364	0.490597461	0.466067588	7.923149002	2.30E+00
17	0.025522877	0.413848365	40.89916487	11.76259982	0.490597461	0.466067588	8.389216591	2.28E+00
18	0.025686913	0.439535278	40.63631166	11.68700323	0.490597461	0.466067588	8.855284179	2.27E+00
19	0.025854152	0.46538943	40.37174709	11.61091446	0.490597461	0.466067588	9.321351767	2.25E+00
20	0.026024701	0.491414131	40.1054373	11.53432377	0.490597461	0.466067588	9.787419356	2.24E+00

21	0.02619867	0.517612802	39.83734729	11.45722108	0.490597461	0.466067588	10.25348694	2.22E+00
22	0.026376176	0.543988978	39.56744087	11.37959599	0.490597461	0.466067588	10.71955453	2.21E+00
23	0.026557339	0.570546317	39.29568062	11.30143775	0.490597461	0.466067588	11.18562212	2.19E+00
24	0.026742287	0.597288604	39.02202779	11.22273519	0.490597461	0.466067588	11.65168971	2.18E+00
25	0.026931154	0.624219758	38.7464423	11.14347681	0.490597461	0.466067588	12.1177573	2.16E+00
26	0.02712408	0.651343839	38.46888261	11.06365064	0.490597461	0.466067588	12.58382489	2.15E+00
27	0.027321213	0.678665051	38.18930567	10.98324431	0.490597461	0.466067588	13.04989247	2.13E+00
28	0.027522707	0.706187758	37.90766684	10.90224498	0.490597461	0.466067588	13.51596006	2.12E+00
29	0.027728726	0.733916484	37.62391983	10.82063934	0.490597461	0.466067588	13.98202765	2.10E+00
30	0.027939442	0.761855926	37.33801657	10.73841357	0.490597461	0.466067588	14.44809524	2.09E+00
31	0.028155036	0.790010962	37.04990714	10.65555329	0.490597461	0.466067588	14.91416283	2.07E+00
32	0.028375699	0.818386661	36.75953968	10.57204361	0.490597461	0.466067588	15.38023042	2.05E+00
33	0.028601634	0.846988295	36.46686023	10.487869	0.490597461	0.466067588	15.846298	2.04E+00
34	0.028833052	0.875821347	36.17181269	10.40301333	0.490597461	0.466067588	16.31236559	2.02E+00
35	0.029070181	0.904891529	35.87433862	10.31745979	0.490597461	0.466067588	16.77843318	2.00E+00
36	0.029313258	0.934204787	35.57437715	10.23119087	0.490597461	0.466067588	17.24450077	1.99E+00
37	0.029562537	0.963767324	35.27186481	10.14418832	0.490597461	0.466067588	17.71056836	1.97E+00
38	0.029818286	0.99358561	34.96673541	10.05643311	0.490597461	0.466067588	18.17663595	1.95E+00
39	0.030080789	1.023666399	34.65891983	9.967905342	0.490597461	0.466067588	18.64270353	1.94E+00
40	0.03035035	1.054016749	34.34834583	9.878584262	0.490597461	0.466067588	19.10877112	1.92E+00
41	0.03062729	1.084644039	34.03493793	9.788448147	0.490597461	0.466067588	19.57483871	1.90E+00
42	0.030911952	1.115555992	33.71861708	9.697474271	0.490597461	0.466067588	20.0409063	1.88E+00
43	0.031204703	1.146760695	33.39930053	9.605638831	0.490597461	0.466067588	20.50697389	1.87E+00
44	0.031505931	1.178266626	33.07690151	9.512916875	0.490597461	0.466067588	20.97304148	1.85E+00
45	0.031816056	1.210082682	32.75132901	9.419282222	0.490597461	0.466067588	21.43910906	1.83E+00
46	0.032135523	1.242218205	32.42248741	9.324707378	0.490597461	0.466067588	21.90517665	1.81E+00
47	0.032464811	1.274683016	32.09027622	9.22916344	0.490597461	0.466067588	22.37124424	1.79E+00
48	0.032804434	1.307487449	31.75458968	9.132619992	0.490597461	0.466067588	22.83731183	1.78E+00
49	0.033154944	1.340642393	31.41531639	9.035044993	0.490597461	0.466067588	23.30337942	1.76E+00
50	0.033516935	1.374159328	31.07233885	8.936404654	0.490597461	0.466067588	23.76944701	1.74E+00

51	0.033891048	1.408050376	30.72553303	8.836663299	0.490597461	0.466067588	24.23551459	1.72E+00
52	0.034277976	1.442328352	30.37476778	8.735783214	0.490597461	0.466067588	24.70158218	1.70E+00
53	0.034678467	1.47700682	30.01990433	8.633724485	0.490597461	0.466067588	25.16764977	1.68E+00
54	0.035093333	1.512100153	29.66079557	8.530444806	0.490597461	0.466067588	25.63371736	1.66E+00
55	0.035523454	1.547623606	29.2972854	8.425899282	0.490597461	0.466067588	26.09978495	1.64E+00
56	0.035969789	1.583593395	28.9292079	8.320040193	0.490597461	0.466067588	26.56585254	1.62E+00
57	0.036433383	1.620026778	28.55638647	8.212816748	0.490597461	0.466067588	27.03192013	1.60E+00
58	0.036915378	1.656942156	28.17863279	8.104174791	0.490597461	0.466067588	27.49798771	1.58E+00
59	0.037417024	1.69435918	27.79574579	7.994056489	0.490597461	0.466067588	27.9640553	1.56E+00
60	0.037939694	1.732298874	27.40751032	7.882399969	0.490597461	0.466067588	28.43012289	1.54E+00
61	0.0384849	1.770783774	27.01369579	7.769138908	0.490597461	0.466067588	28.89619048	1.51E+00
62	0.039054308	1.809838082	26.61405452	7.654202081	0.490597461	0.466067588	29.36225807	1.49E+00
63	0.039649764	1.849487846	26.20831998	7.537512826	0.490597461	0.466067588	29.82832566	1.47E+00
64	0.040273316	1.889761162	25.79620465	7.418988456	0.490597461	0.466067588	30.29439324	1.45E+00
65	0.040927247	1.930688409	25.37739766	7.298539568	0.490597461	0.466067588	30.76046083	1.42E+00
66	0.041614106	1.972302515	24.95156208	7.176069254	0.490597461	0.466067588	31.22652842	1.40E+00
67	0.042336753	2.014639268	24.51833168	7.05147219	0.490597461	0.466067588	31.69259601	1.38E+00
68	0.043098408	2.057737676	24.07730729	6.924633576	0.490597461	0.466067588	32.1586636	1.35E+00
69	0.043902712	2.101640388	23.62805249	6.795427895	0.490597461	0.466067588	32.62473119	1.33E+00
70	0.044753801	2.146394188	23.17008853	6.66371746	0.490597461	0.466067588	33.09079877	1.30E+00
71	0.045656393	2.192050581	22.70288837	6.529350695	0.490597461	0.466067588	33.55686636	1.28E+00
72	0.046615904	2.238666485	22.22586957	6.39216009	0.490597461	0.466067588	34.02293395	1.25E+00
73	0.047638579	2.286305064	21.73838578	6.251959751	0.490597461	0.466067588	34.48900154	1.22E+00
74	0.048731669	2.335036734	21.23971644	6.108542448	0.490597461	0.466067588	34.95506913	1.20E+00
75	0.049903646	2.38494038	20.72905431	5.961676019	0.490597461	0.466067588	35.42113672	1.17E+00
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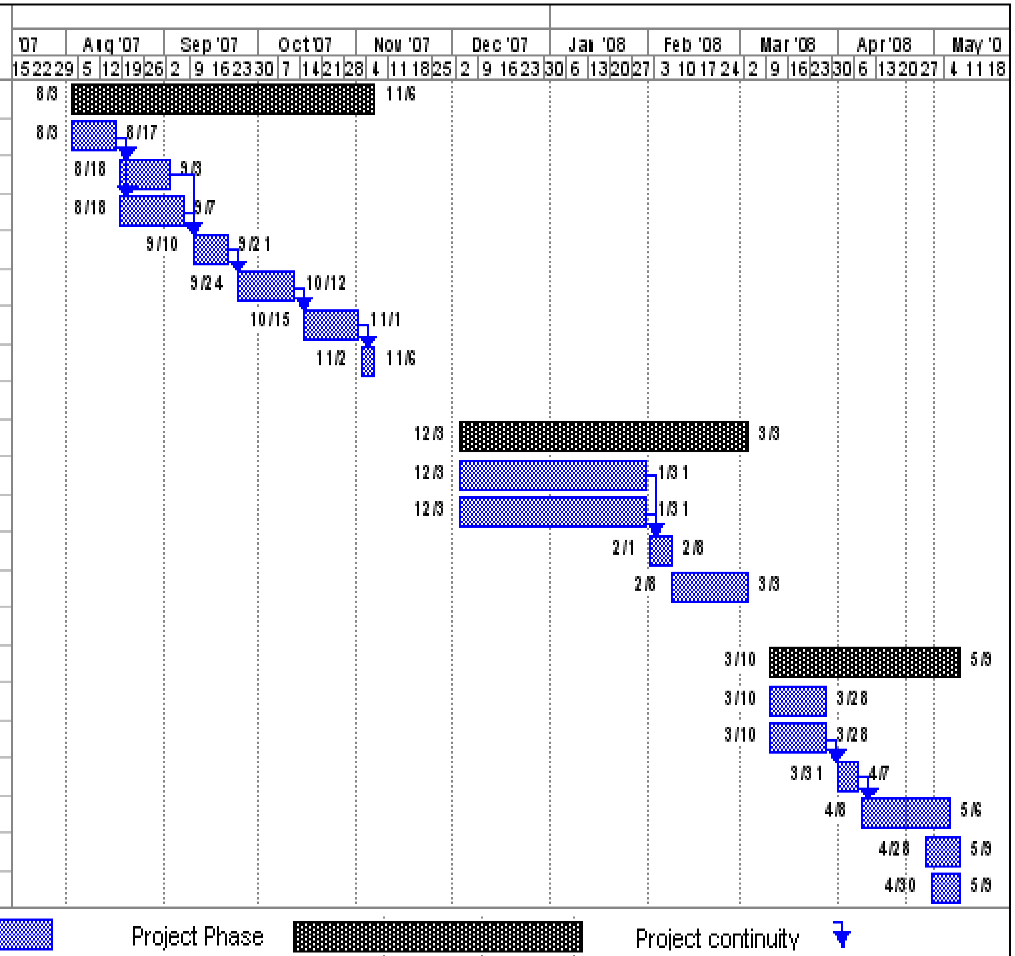
77	0.052526002	2.488630861	19.6679935	5.656514932	0.490597461	0.466067588	36.35327189	1.11E+00
78	0.054002372	2.542633233	19.11538926	5.497585951	0.490597461	0.466067588	36.81933948	1.08E+00
79	0.055610707	2.59824394	18.54632698	5.33392364	0.490597461	0.466067588	37.28540707	1.05E+00
80	0.057371922	2.655615862	17.95924226	5.165078074	0.490597461	0.466067588	37.75147466	1.02E+00

81	0.05931189	2.714927752	17.35230592	4.990523181	0.490597461	0.466067588	38.21754225	9.82E-01
82	0.061463065	2.776390817	16.72335668	4.809637381	0.490597461	0.466067588	38.68360983	9.48E-01
83	0.063866801	2.840257618	16.0698101	4.621677385	0.490597461	0.466067588	39.14967742	9.12E-01
84	0.066576764	2.906834383	15.38853257	4.425741967	0.490597461	0.466067588	39.61574501	8.75E-01
85	0.069664086	2.976498468	14.6756626	4.220720563	0.490597461	0.466067588	40.0818126	8.36E-01
86	0.073225431	3.049723899	13.92634951	4.005218119	0.490597461	0.466067588	40.54788019	7.96E-01
87	0.077396155	3.127120054	13.13435757	3.777441237	0.490597461	0.466067588	41.01394778	7.53E-01
88	0.082372822	3.209492876	12.29143957	3.535018021	0.490597461	0.466067588	41.48001536	7.07E-01
89	0.088454198	3.297947074	11.38629109	3.274697317	0.490597461	0.466067588	41.94608295	6.59E-01
90	0.096121777	3.39406885	10.40268056	2.991810929	0.490597461	0.466067588	42.41215054	6.06E-01
91	0.106214876	3.500283727	9.31578772	2.679220548	0.490597461	0.466067588	42.87821813	5.48E-01
92	0.120368558	3.620652284	8.08406079	2.324975883	0.490597461	0.466067588	43.34428572	4.84E-01
93	0.142366479	3.763018763	6.627229956	1.905991335	0.490597461	0.466067588	43.81035331	4.09E-01
94	0.184213062	3.947231825	4.742184614	1.363852295	0.490597461	0.466067588	44.27642089	3.16E-01
95	0.362978528	4.310210353	1.027838964	0.295606486	#REF!	#REF!	#REF!	1.61E-01

4) FYP Gantt Chart

ID	Task Name	Start	Finish
1	1st Phase	Fri 8/3/07	Tue 11/6/07
2	Problem Identification	Fri 8/3/07	Fri 8/17/07
3	Preliminary Research Work	Sat 8/18/07	Mon 9/3/07
4	Data Gathering	Sat 8/18/07	Fri 9/7/07
5	Preliminary Report	Mon 9/10/07	Fri 9/21/07
6	Drawing Modeling	Mon 9/24/07	Fri 10/12/07
7	Interim I	Mon 10/15/07	Thu 11/1/07
8	Seminar 1	Fri 11/2/07	Tue 11/6/07
9			
10	2nd Phase	Mon 12/3/07	Mon 3/3/08
11	Models Analysis	Mon 12/3/07	Thu 1/31/08
12	Models Simulation	Mon 12/3/07	Thu 1/31/08
13	Progress Report	Fri 2/1/08	Fri 2/8/08
14	Seminar 2	Fri 2/8/08	Mon 3/3/08
15			
16	3rd Phase	Mon 3/10/08	Fri 5/9/08
17	Result and Discussion	Mon 3/10/08	Fri 3/28/08
18	Conclusion	Mon 3/10/08	Fri 3/28/08
19	Poster	Mon 3/31/08	Mon 4/7/08
20	FYP Dissertation Draft	Tue 4/8/08	Tue 5/6/08
21	Oral Presentation	Mon 4/28/08	Fri 5/9/08
22	Dissertation	Wed 4/30/08	Fri 5/9/08



Project: FYP Gantt Chart
Date: Tue 4/22/08