MODELING AND ANALYSIS OF DISC BRAKE PERFORMANCE

by

Hafifi bin Mohd Yusop

Dissertation submitted in partial fulfillment of
the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

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CERTIFICATION OF APPROVAL

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Mechanical Engineering Programme
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Approved by,

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UNIVERSITY TEKNOLOGI PETRONAS
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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 concept as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or person

HAFIFI BIN MOHD YUSOP

ABSTRACT

In automotive industry, disc brake is a device that plays a vital role in slowing or stopping the rotation of a wheel. A brake disc, which is usually made of cast iron or ceramic, is connected to the wheel or the axle. To stop the wheel, friction material in the form of brake pads (mounted in a device called a brake caliper) is forced mechanically, hydraulically, pneumatically or electromagnetically against both side of the disc. Friction causes the disc and the attached wheel to decelerate or stop. Current disc brake has weaknesses either from the design and the application, which causes the damages to the brake. This research attempts to investigate a ventilated disc brake with the aim to improve the stopping time, material used for disc caliper and brake pad and the optimization of the brake disc position.

The project will employ the method of simulation in ANSYS software, which is a software used to analyze the air flow and dynamic inside the brake rotor vane, as well as thermal stress analysis of ventilated brake disc. The data are gathered from trusted resources such as from internet, books, journal and dissertation thesis to be used in the simulation. The result from the simulation using the detail in Proton Waja's disc brake will be a reference to improve the design.

This research aims mainly to find an optimum braking performance and to reduce the percentage of damage modes reoccurrence. If the braking time and damage modes are reduced using the best material chosen, then this project has succeeded to achieve its goal.

ACKNOWLEDGEMENT

In the name of Allah; The Most Gracious and The Most Merciful.

With humble heart, thanks to god; The Almighty for giving me the abundant strength to complete this Final Year Project. Without His nourishing guidance, this project would not be a success.

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LIST OF ABBREVIATIONS

I	=	Mass moment of inertia of rotating parts (kgm^2)
m	=	Vehicle mass (kg)
V_1	=	Velocity at begin of braking (ms^{-1})
V_2	=	Velocity at end of braking (ms^{-1})
k	=	Correction factor for rotating masses
R	=	Tyre radius (m)
X_1	=	Proportion of braking at the front
Y_1	=	Proportion of heat distribution to rotor
g	=	Gravitational Constant (ms^{-2})
a	=	Thermal diffusivity of rotor (m^2h^{-1})
k_c	=	Thermal conductivity of rotor (Nm/hm^2)
$q''_{(0)}$	=	Heat Flux into rotor surface (Nm/hm^2)
t	=	Braking time (h)
T_i	=	Initial Temperature (K)
t_s	=	Time vehicle stops (h)
c	=	Specific heat of rotor material (Nm/kgK)
L_c	=	Characteristic Length (m)
ρ	=	Rotor Density (kgm^{-3})
P_{bav}	=	Average braking power (Nmh^{-1})
A_s	=	Swept area of brake rotor (m^2)
Nu	=	Nusselt number
Re	=	Reynolds number
h_R	=	Convective heat transfer coefficient (Nm/hKm^2)
L_d	=	Hydraulic Length or diameter (m)
k_a	=	Thermal conductivity of air (Nm/hmK)
V	=	Vehicle speed (ms^{-1})
ρ_a	=	Air density (kgm^{-3})
m_a	=	Mass flow rate of air (m^3s^{-1})
c_a	=	Specific heat of air (Nm/kgK)
l	=	Length of cooling vane (m)
V_{in}	=	Velocity of inlet (ms^{-1})
V_{out}	=	Velocity of outlet (ms^{-1})

A_{in}	=	Inlet area (m^2)
A_{out}	=	Outlet area (m^2)
d	=	Inner diameter (m)
D	=	Outer diameter (m)
ω	=	Rotor rotation speed in revolution per minute (rpm)
E	=	Elastic Modulus of rotor (Nm^{-2})
ΔT	=	Temperature difference (K)
α_T	=	Thermal expansion coefficient (m/Km)
v	=	Poisson's ratio
v_r	=	Rotor volume (m^3)

CHAPTER 1
INTRODUCTION

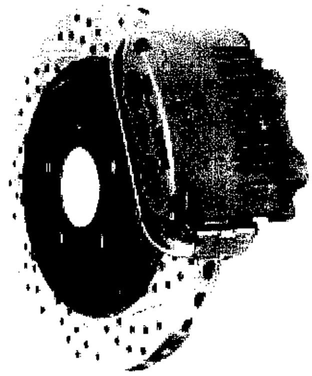
1.1 BACKGROUND OF STUDY

Disc brake technology has begun in late 1890 but, was not adopted until the year 1902 [1]. Since then disc brake technology have been developed and widely used in car. The innovation and development of disc brake design are not specifically implemented in car or automotive but; disc brakes are in demand for vehicle that requires brake performance. This research is focused on automotive section though not specific to a type of car.

This section explains the disc brake performance in design and damage modes that affect the performance of disc brake. The section starts with the description of the disc brake emphasizing on the design of current brake disc followed by commonly encountered damage modes of disc brake that affect the disc brake performance.

In automotive industry, there are 2 main criteria involved in analysis of disc brake performance for different use, which are the design of disc brake and the damage modes that occur at the disc brake. In the design, some of the disc brakes are made of solid cast iron, while others are hollowed out with fin and joined together the disc's two contact surfaces in casting process, some are ventilated disc produced from milling process on a solid cast iron, cross-drilling from the tapping process (varies size according to the application) and some are slotted. Figure 1.1 shows the disc brake with the different designs.

(a) Solid cast iron



(b) Combination design of ventilated, cross drilling and slotted disc brake

Figure 1.1: Disc brake designs [1]

Disc brake damage modes can be categorized as follows:

1. Warping
2. Cracking
3. Squeal
4. Judder

1.1.1 Warping

Warping is the change in shape of a disc brake often caused by excessive heat. The cause of warping is due to excessive heat when the disc is overheated during continuous braking and at the same time moving until the vehicle come to rest. When the brake is applied, friction is generated and heats up the area which the brake pads are in contact with the disc. As a result of the contact, the expose area of the disc will cool at a higher rate as compared to the surface in contact with the brake pads. This will cause uneven cooling of the disc and leads to warping [1] [2].

1.1.2 Cracking

Cracking is limited mostly to drilled discs, which get small cracks around outside edges of the drilled holes near the edge of the disc due to the disc's uneven rate of expansion in severe duty environments [1][3].

1.1.3 Squeal

Loud noise or high pitch squeal occurs when the brakes are applied. Most brake squeal is produced by vibration (resonance instability) of the brake components [4].

1.1.4 Judder

Judder occurs when a driver decelerates from speeds of around 120 km/h to about 60 km/h, which results in severe vibrations being transmitted to the driver through the chassis during braking [1].

1.2 PROBLEM STATEMENT

The time for effective braking in conjunction with thermal analysis when using ventilated disc brake is used. Thermal problem, stress on rotor due to brake power in single stop with the percentage of reoccurrences of the damage modes.

1.3 OBJECTIVES OF STUDY

The project aims to achieve the following objectives:

1. To model a portion of disc brake and brake pad action in ANSYS software.
2. To analyze the thermal stress analysis of a ventilated disc brake.
3. To propose a solution to reduce the reoccurrence of damage mode at disc brake.

1.4 SCOPE OF STUDY

This project was implemented in two parts. Part 1 (FYP-semester 1) focused on the analysis of current design and material use for braking performance. This was achieved by literature search about disc brake system and performance, damage modes, desirable properties for brake and suitable design are needed. These informations were gathered from trusted resources such as from internet, books, journal and dissertation thesis.

This project used the method of simulation in ANSYS. The data gathered was used in simulation using the ANSYS software. The results from the simulation using the detail in Proton Waja's disc brake will be used as a reference to improve the design.

The simulation provided analysis of flow inside the rotor vane (ventilated disc). The results were in term of thermal analysis and thermal stress analysis. The theoretical analysis carried out in literature review using the equation in brake safety design to acquire the product of dimensionless numbers and also the calculated heat transfer coefficient. Using the dimensionless parameters and a few assumptions suggests from text book, journal paper and past research, the simulation will be conduct in ANSYS.

CHAPTER 2
LITERATURE REVIEW

2.1 COEFFICIENT OF FRICTION

Mean coefficient of friction (COF) are the most important parameter in analysis of a disc brake performance. It is a dimensionless quantity used to calculate the force of friction (static or kinetic) [5]. Both static and kinetic COF depend on the pair of surfaces in contact. Schmid and Duncan [6] suggested that the COF-evolution during a stop braking cycle (in-stop behaviour) provides more information about the surface states at the pad/disc interface. COF is often stated as 'material property', but it is better recognized as a 'system property'. The COF for two materials depends on system variables like temperature, atmosphere as well as on geometric properties of the interface between the material. High value of COF will give the shorter braking time and improve braking performance.

Below are the desirable properties for friction material for brake [7].

1. Two material and contact must have a high coefficient of friction.
2. The material in contact must resist wear effects.
3. The friction value should be constant over a range of temperatures and pressures.
4. The material should be resistant to the environment. (Moisture, dust, pressure).
5. The material should possess good thermal properties, high heat capacity, good thermal properties, high heat capacity, and good thermal conductivity, withstand high temperature.
6. Able to withstand high constant pressures.
7. Good shear strength to transferred friction forces to structure.
8. Should be safe to use and acceptable for the environment.

2.2 BRAKE PADS

Brake pads convert the kinetic energy of the car to thermal energy by friction. When a brake pad is heat up, it starts to transfer small amounts of friction material to the disc or pad. The brake rotor and disk will then be in contact with each other to provide stopping power. The friction of the pad against the disk is however responsible for the majority of stopping power. Two brake pads are positioned in the caliper on the inboard and outboard sides of the rotor. Fundamentally, a brake pad is a steel plate with a friction material lining bonded or riveted to its surface as shown in Figure 2.1 [8].

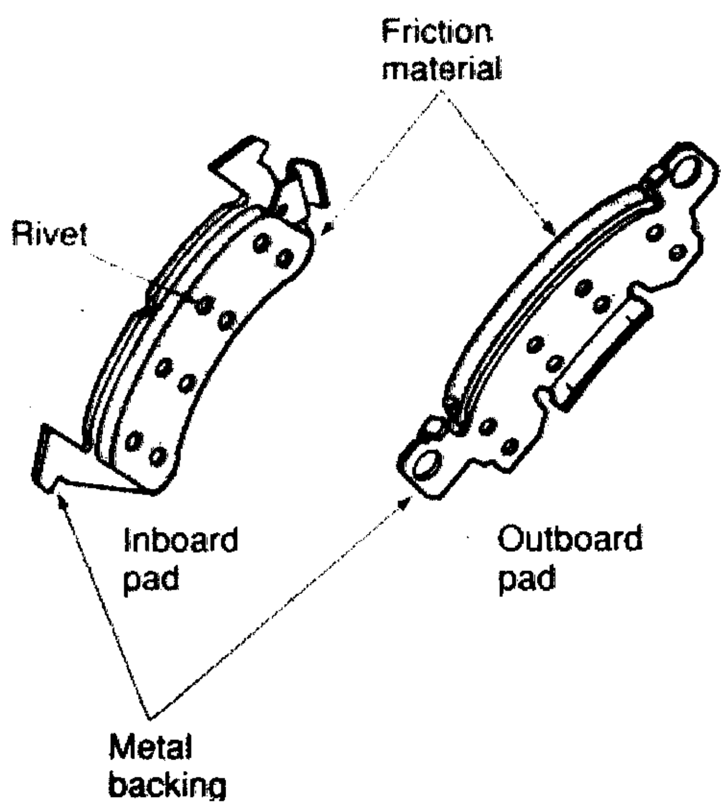


Figure 2.1: The parts of disc brake pads [8]

2.3 DAMAGE MODES

These are the common damage modes occur at disc brake that affects the braking performance:

1. Warping
2. Cracking
3. Brake squeal
4. Brake judder
5. Brake Dust

After reading the past research paper [2], the warping and cracking in damage modes are the problem that disc brake suffers the most and relevant to this project. As they involve in temperature analysis.

Heat-spotted rotor shows hard and slightly raised spot on the braking surface with uneven wear. Heat (or hot) spots are caused by localized high-temperature causing material changes of the structure. Bluish, discolored swept braking surfaces are caused by excessive heat exceeding 400°C to 500°C [10]; possibly by overheating in severe fade combine with sudden cooling from the air flow will cause warping.

Normal braking operation will cause fine, hairline cracks laterally across the swept surface of rotor. This is normal condition caused by constant low-energy and cooling of the braking surface. What makes this damage modes worst in the ventilated disc are the thickness and the vane which is the essential criteria for the ventilated disc.

2.4 DESIGN

The designs of disc brakes are application specific. For example, a solid cast iron disc brake and slotted disc brake are common use for racing car (high load and hard use of braking). While, a ventilated disc and cross drilling are suitable for heavily loaded front disc and standard vehicles.

Ventilated disc from Proton Waja are choose as the Proton Waja are the most common and reliable car Proton ever produce. The project criteria are the thermal analysis and thermal stress analysis which are relevant to the ANSYS fluid simulation and to understand the flow inside the rotor vane in ventilated disc. As the thermal analysis and thermal stress analysis in ANSYS structural simulation are more relevant to brake drum.

2.5 THERMAL DESIGN MEASURES

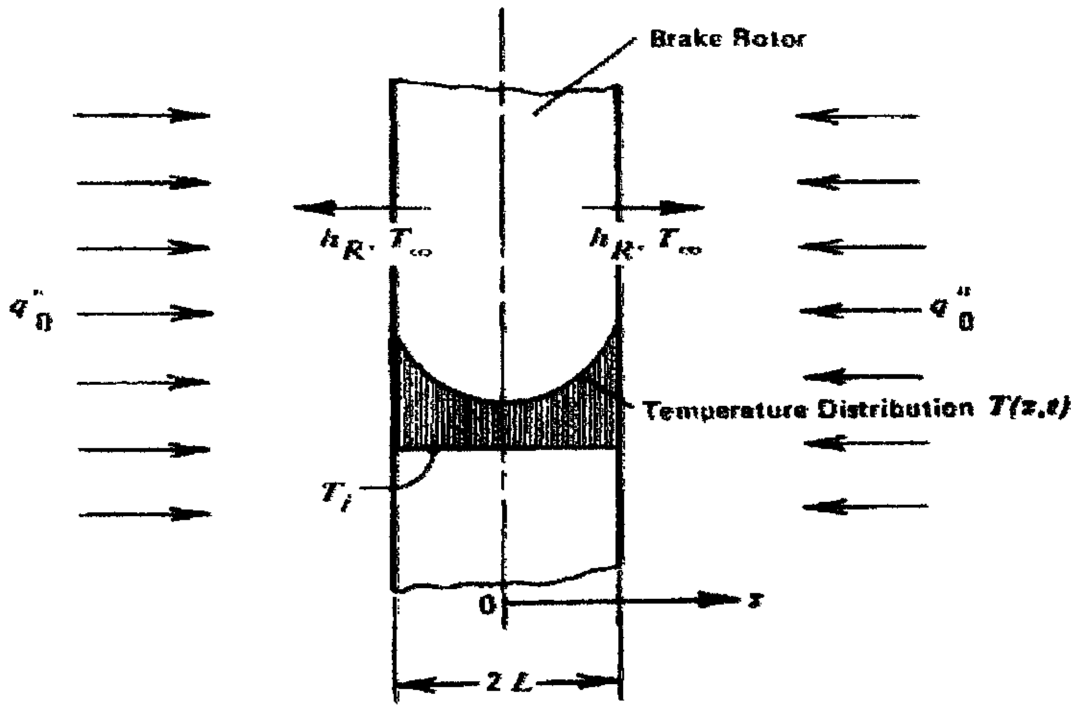


Figure 2.2: Thermal system representing brake rotor [10]

Energy flow from high temperature to lower will cause temperature difference. A study of heat transfer is concerned with determination of the instantaneous rates of energy flow in all situations as shown in Figure 2.2. In the thermal design measure, we need to consider the braking power and the braking energy that are in the temperature analysis. In this case, the simulation was conducted base on the dimensionless parameters that is Reynolds number, Prandtl number and Nusselt number. Based on these dimensionless parameters, we can theoretically calculate the convective heat transfer coefficient. As in theory, the kinetic and potential energies of a moving vehicle are converted into thermal energy through friction in the brakes.

Braking power is P_b, is equal to braking energy devided by time during which braking occur [10]. In this project, assumptions are made; the vehicle comes to a complete stop and the braking are in continued braking.

$$P_b = \frac{d(E_b)}{dt} \qquad [\frac{Nm}{s}]$$

Basic formulae for braking energy, E_b is

$$E_b = \left(\frac{m}{2}\right)(V_1^2 - V_2^2) + \left(\frac{I}{2}\right)(\omega_1^2 - \omega_2^2) \qquad [Nm]$$

Where m is the vehicle mass, V_1 is the initial velocity of the car, V_2 is the final velocity of the car, I is the mass moment of inertia of rotating parts, $\omega_1$is the initial angular velocity of rotating rotor and ω_2 is the final angular velocity of rotating parts.

Since the vehicle comes to a complete stop, the equation becomes

$$E_b = \left(\frac{m}{2}\right)\left(1 + \frac{I}{R^2 m}\right)V_1^2 \approx \frac{kmV_1^2}{2} \qquad [Nm]$$

Where R is the tyre radius and k is the correction for rotating masses. The average braking power of the entire vehicle P_{bavg} is computed as below

$$P_{bavg} = agE_b \qquad [\frac{Nm}{s}]$$

Where a is the thermal diffusivity of rotor and g is the gravitational constant. Then, the average braking power absorbed per hour by one side of one front brake P_{bav} is

$$P_{bav} = \frac{3600 P_{bavg} X_1 Y_1}{4} \qquad [\frac{Nm}{h}]$$

The X_1 is the proportion of braking at the front and Y_1 is the proportion heat distribution to rotor.

As for the surface temperature, T as a function of time, the analysis are based on the assumption that the vehicle went to complete stop while the brake are continued apply. The equation is

$$T(L,t) - T_i = \left(\frac{5}{4}\right)^{\frac{1}{2}}\left(\frac{q''_{(0)}}{k_c}\right)(at)^{\frac{1}{2}}\left(1 - \frac{2t}{3t_s}\right) \qquad [K]$$

Where T_i is the initial surface temperature, $q''_{(0)}$ is the heat flux into the rotor surface, k_c is the thermal conductivity of rotor, t is the braking time and t_s is the time take for the vehicle to stop.

The formulae were based on the assumption made that is linearly braking power. This equation are the basic temperature analysis for braking energy and braking power considering the vehicle decelerating on a level surface from higher velocity V_1 to a lower velocity V_2.

The thickness of the disc brake rotor has an influence on rotor failure. A rotor material will have improved thermal performance with higher thermal endurance strength, higher thermal conductivity, decreased elastic modulus, and decreased thermal expansion coefficient. Thicker rotors tend to exhibit higher potential for thermal stress will result in rotor failure or what we call here damage mode. Disc brake rotors generally have sufficient thermal endurance when the heat flux is limited to a value computed by [10].

$$q''_{R,allowable} = 28.8(439 - 046T)/L$$

$q''_{R,allowable}$ is the maximum heat flux into the rotor disc where T is the surface temperature and L is the length of the vane. The velocity and deceleration used for the heat flux computing the heat flux produced should be the maximum speed and deceleration attainable. If the heat flux produced exceeds the heat flux allowable, then the disc brake rotor will exhibit a limited endurance relative to thermal cracking.

The dimensionless parameter, Nusselt number is used for estimating the heat transfers rate q" (0). It is in function of Reynolds number, Prandtl Number, heat transfer constant for geometric shape and boundary condition. The convective cooling of ventilated disc brakes can be presented by the product of dimensionless number raised to some power [10].

$$\mathrm{Nu} = 0.037\mathrm{Re}^{0.8}\mathrm{Pr}^{0.33}$$

The constant of heat transfer 0.037 is a function of geometry of the brake and assumes different value for disc, drum and ventilated rotor. The value of 0.037 indicates uniform surfaced temperature while 0.053 indicates it is uniform heat flux.

Power value of 0.8 and 0.33 are the function of the type of flow and the thermal properties of the surrounding air respectively.

The dimensionless parameters are:

$$\mathrm{Nu} = h_R l_c \qquad \text{(Nusselt Number)}$$

$$\mathrm{Re} = \frac{V_{avg}\rho_a L_c}{m_a} \qquad \text{(Reynolds Number)}$$

$$\mathrm{Pr} = \frac{c_a m_a}{k_a} \qquad \text{(Prandtl Number)}$$

Where h_R is the convective heat transfer coefficient, l_c is the length of the cooling vane, V_{avg} is the average velocity of air from inlet and outlet, ρ_a is the density of the air, L_c is the characteristic length, m_a is the air flow rate, c_a is the specific heat of air and k_a is the thermal conductivity of air.

The cooling effectiveness are associated with the internal vanes tends to decrease the temperature flow (special characteristic of ventilated disc brake) for higher speeds due to the increased stagnation pressure of the air.

For estimating purposes, the following relationship may be used to obtain the heat transfer coefficient inside the vanes of the brake rotor [10].

$$h_R = 0.023\left[1 + \left(\frac{d_h}{l}\right)^{0.67}\right]\mathrm{Re}^{0.8}\mathrm{Pr}^{0.33}\mathrm{x}\left(\frac{k_a}{d_h}\right) \qquad [\frac{Nm}{hKm^2}]$$

This equation are valid for $\mathrm{Re}>10^4$ which mean turbulent flow. The hydraulic diameter, d_h is defined as the ratio of four times the cross sectional flow area (wetted area) devided by the wetted perimeter.

The equation for convection heat transfer coefficient of low values velocity (laminar flow $\mathrm{Re}<10^4$) is [10]

$$h_R = 1.86(\mathrm{RePr})^{\frac{1}{3}}\left(\frac{d_h}{l}\right)^{0.33}\mathrm{x}\left(\frac{k_a}{d_h}\right) \qquad [\frac{Nm}{hKm^2}]$$

The velocity associated with the Reynolds number is the air flow velocity existing in the vanes which is not identical to the forward speed of the vehicle. The detail data used in calculation is in the methodology.

The average velocity through the cooling vanes can be computed by

$$V_{avg} = \frac{(V_{in} + V_{out})}{2} \qquad [\frac{m}{s}]$$

$$V_{in} = 0.0158\omega(D^2 - d^2)^{\frac{1}{2}} \qquad [\frac{m}{s}]$$

$$V_{out} = V_{in}\left(\frac{A_{in}}{A_{out}}\right) \qquad [\frac{m}{s}]$$

Where V_{in} is the velocity of air at the inlet, V_{out} is the velocity of air at the outlet of the vane rotor, ω is the rotor rotation speed in revolution per minute (rpm), D is the outer diameter of the rotor disc, d is the inner diameter of the rotor disc, A_{in} is the inlet area of the rotor vane and A_{out} is the outlet area of the rotor vane.

The air flow rate m_a is determined by

$$m_a = 0.00147\omega[(D^2 - d^2)A_{in}]^{\frac{1}{2}} \qquad [\frac{m^3}{s}]$$

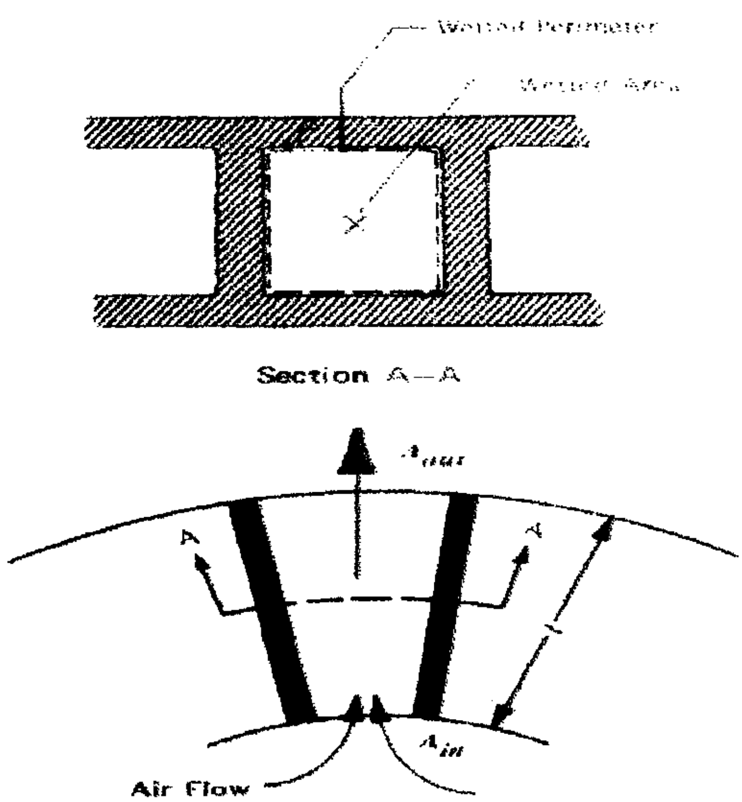


Figure 2.3: The airflow of the ventilated disc

Thermal stresses result from non-uniform temperature distributions. In addition, mechanical stresses may arise from body deformations or body forces. In most practical thermal stress problems, it is permissible to separate the temperature problem from the stress problem and to solve both consecutively.

The approximate compressive stress, σ developed in the surface layer of an infinite flat plate as result of a sudden temperature increase is

$$\sigma = -\left(\frac{E}{1-v}\right)\alpha_T \Delta T \qquad [\frac{N}{m^2}]$$

As stated earlier, the thickness of the disc brake rotor has an influence on rotor failure. A rotor material will have improved thermal performance with higher thermal endurance strength, higher thermal conductivity, decreased elastic modulus and decreased thermal expansion coefficient. Thicker rotors tend to exhibit higher potential for thermal stress giving rotor failure.

The theoretical temperature increase of the swept surface mass of the rotor should not exceed certain limits. The mass is computed by the product swept surface area, rotor thickness and material density.

The braking energy is computed from previous equation. The theoretical temperature increase is computed by an expression below. For example, the theoretical temperature increase $T_{th,F}$ is calculated as

$$T_{th,F} = \frac{(1-\Phi)}{2}\left[\frac{m(V_1^2 - V_2^2)}{2\rho c v}\right] \qquad [K]$$

T_{th} is a theoretical temperature only since heat transfer to the brake pad and all other cooling mechanisms have been ignored. T_{th} however, yields a good comparison measure for the evaluation of a brake system based on the temperature limits stated below. The condition can be divided and described as below:

- For the T_{th} less than 500 K the rotor dimensions is sufficient for most passenger cars.
- For the T_{th} between 500 and 600 K high performance vehicles still have enough brake size.

T_{th} greater than 600 K must be avoided [10].

CHAPTER 3
METHODOLOGY / PROJECT WORK

3.1 MODELLING

Several studies were carried-out indicating the finite element analysis (FEA) technique is the fastest and potentially the most accurate technique for the investigation of brake disc performance [11] [12]. Thermal finite element analysis has been shown to be an efficient and accurate method of estimating the peak disc temperatures during critical vehicle brake test. Sophisticated solution methods and computational fluid dynamics are increasingly used in industry to save cost in simulating sophisticated physical phenomena.

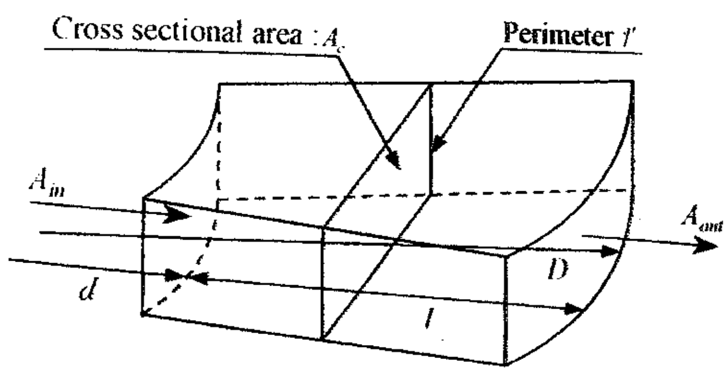


Figure 3.1: The geometry of a brake vane

As shown above in Figure 3.1, the geometry of the vane which is for the fluid region is drawn to give whole idea about the analysis. The data gathered form Waja's specification, analysis is performed to obtain flow and maximum temperature in the fluid region. The heat transfer coefficient (from theory section) and the boundary condition are applied at the faces of the computational domain. The finite element analysis are better than ANSYS fluid as it reduce the computational time significantly and further analysis can be done to predict thermal stresses.

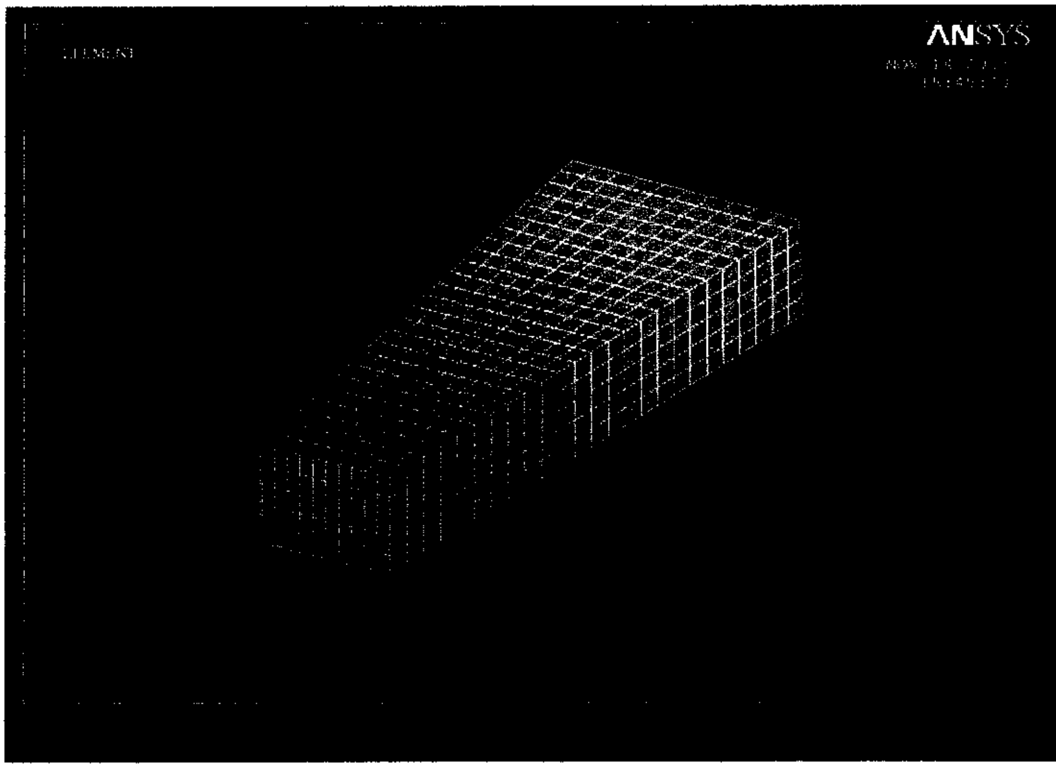


Figure 3.2: Fluid regions in rotor vane

For thermal stress analysis, the computational fluid dynamic and finite element analysis meshes are created in the solid region (the rotor). Then, the compressive stress is calculated and applied uniformly onto the surface of the rotor (swept area).

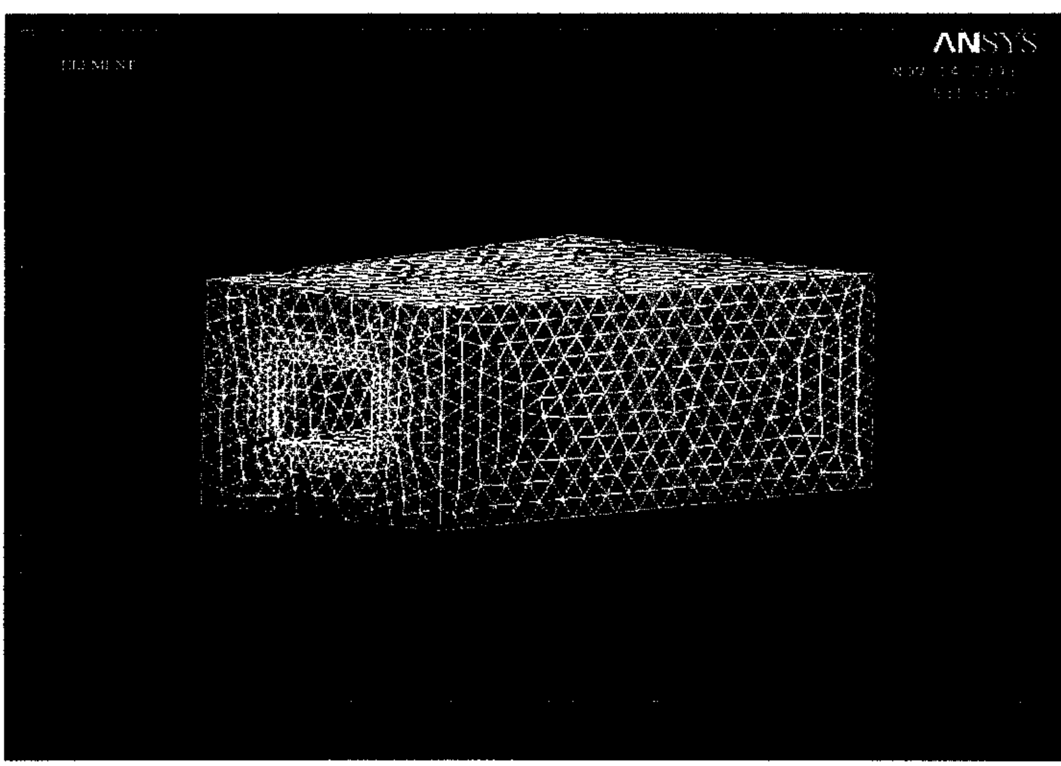


Figure 3.3: Partial of rotor disc used for stress analysis

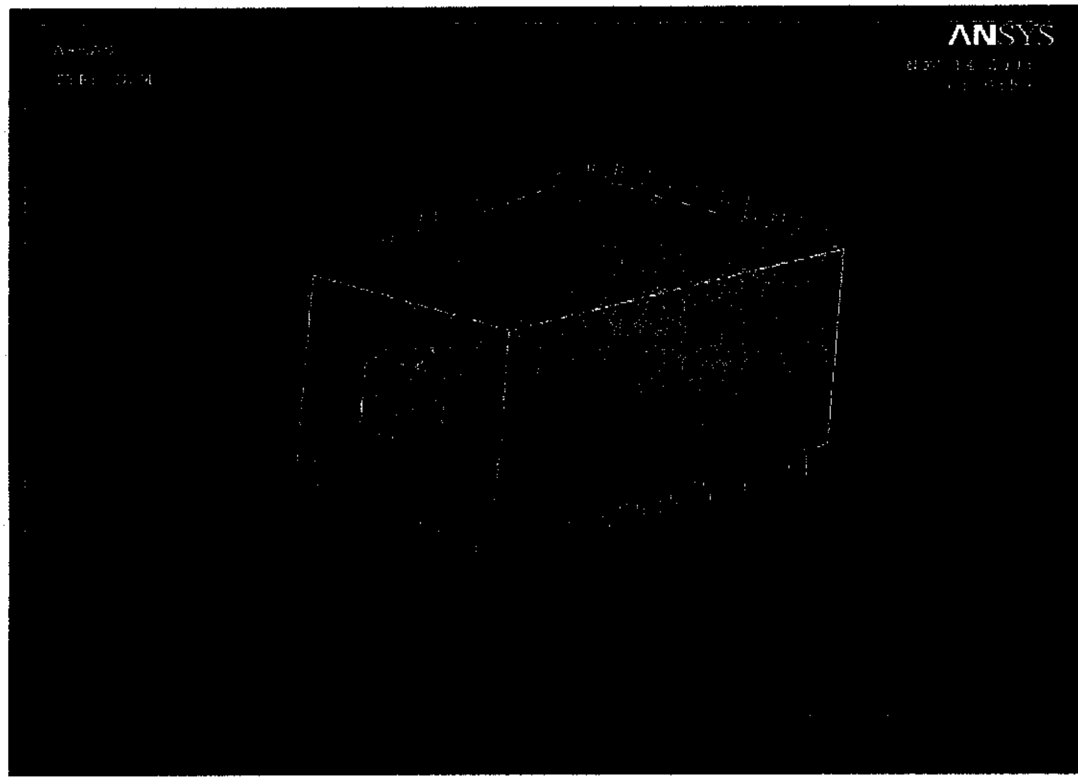


Figure 3.4: Application of Compressive Stress on Surface of The Rotor

Below are the data and specification of the vehicle to be used in calculation:

1. Tyre diameter (*R*) 58 cm
2. Rotor outer diameter (*D*) 25 cm
3. Rotor inner diameter (*d*) 15 cm
4. Mass of vehicle (*m*) 1275 kg
5. Angular velocity (ω) 2122 rpm
6. Proportions of heat distribution 0.9

The data below are the material properties for air and grey cast iron used in the analysis and simulation:

Table 3.1: Material properties for air and grey cast iron

Material properties	Air	Rotor
Density (kgm^{-3})	1.1614	7228
Specific heat (J/kgK)	1007	419
Viscocity (Nsm^{-2})	184.6×10^{-7}	-
Thermal Conductivity (W/mK)	26.3×10^{-3}	48.5×10^{-3}
Thermal diffusivity (m^2h^{-1})	-	57.6×10^{-3}
Thermal expansion coefficient (m/Km)	-	11.36×10^{-3}
Elastic Modulus (Nm^{-2})	-	110.5×10^{9}
Poisson's Ratio	-	0.2

With the data of material properties to be used in the analysis, there are also a few assumption have to be made to make sure this simulation process can be proceeded easily. The assumptions are:

- The rotor is rotating with a constant steady speed.
- The airflow is at steady state.
- Heat transfer coefficient is dependent on the airflow velocity only but does not change with time.
- The frictional heat generated by brake pads is distributed uniformly over the whole surface area of the rotor.
- Flow is incompressible which means the density of air is constant.
- Standard wall functions are used at the wall.
- The heat loss by radiation and conduction is neglected.
- Heat is removed by airflow only (convection only).
- The model only includes a small fraction of rotor.
- The compressive stress is applied uniformly onto surface.

CHAPTER 4
RESULT AND DISCUSSION

4.1 TEMPERATURE ANALYSIS

Based on the calculation and data given, analysis of the temperature and thermal stress analysis in ANSYS can be conducted based on these dimensionless numbers, the product of dimensionless numbers are from theoretical analysis which is carried out during literature review. Also the calculated of heat transfer coefficient is given to summarize the value used during simulation process.

- Reynolds number = 165.45 (laminar flow)
- Prandtl number = 373.32
- Nusselt number = 15.56
- Heat transfer coefficient = 29464.416 Nm/hKm^2

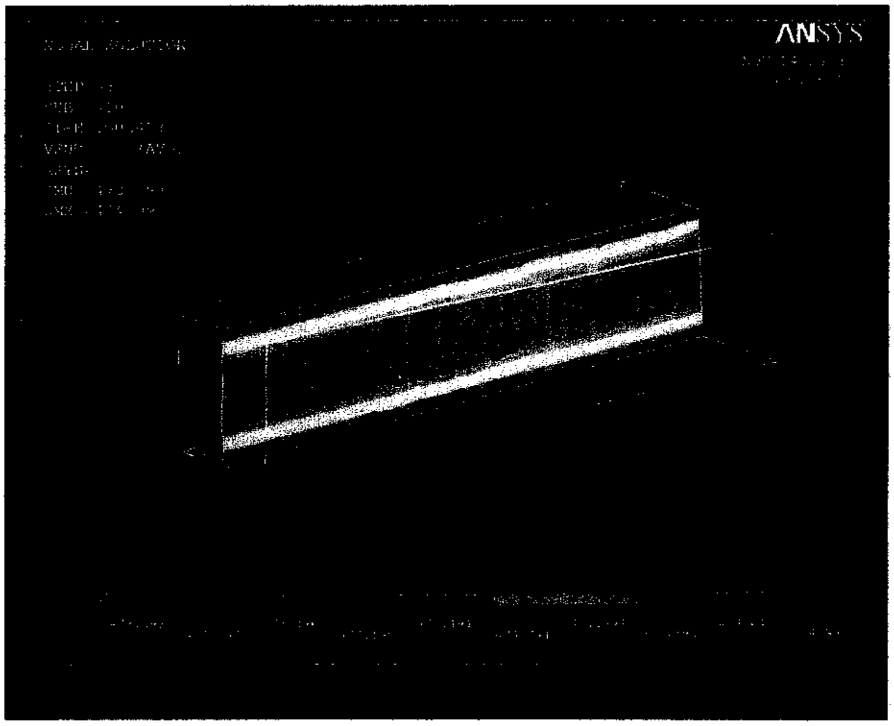


Figure 4.1: Middle cross-section of temperature distribution (horizontally applied)

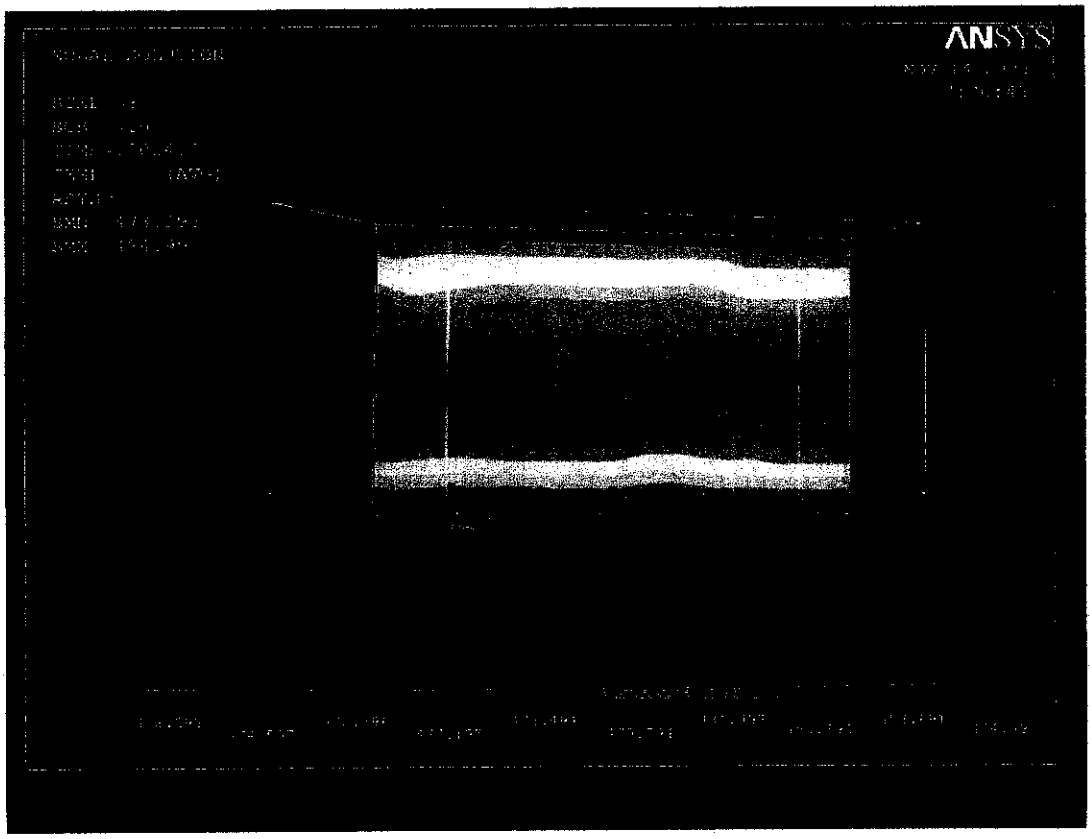


Figure 4.2: Cross-section of temperature distribution at 1/3 from inlet

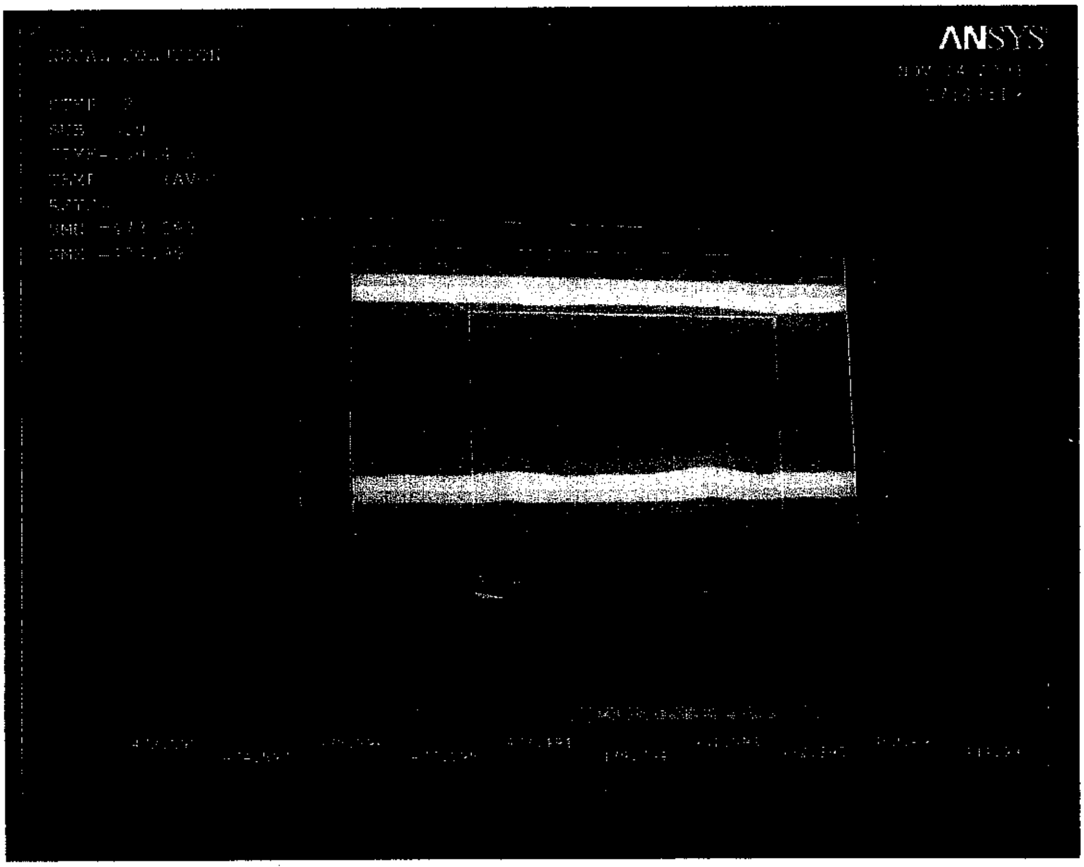


Figure 4.3: Cross-section of temperature distribution at 2/3 from inlet

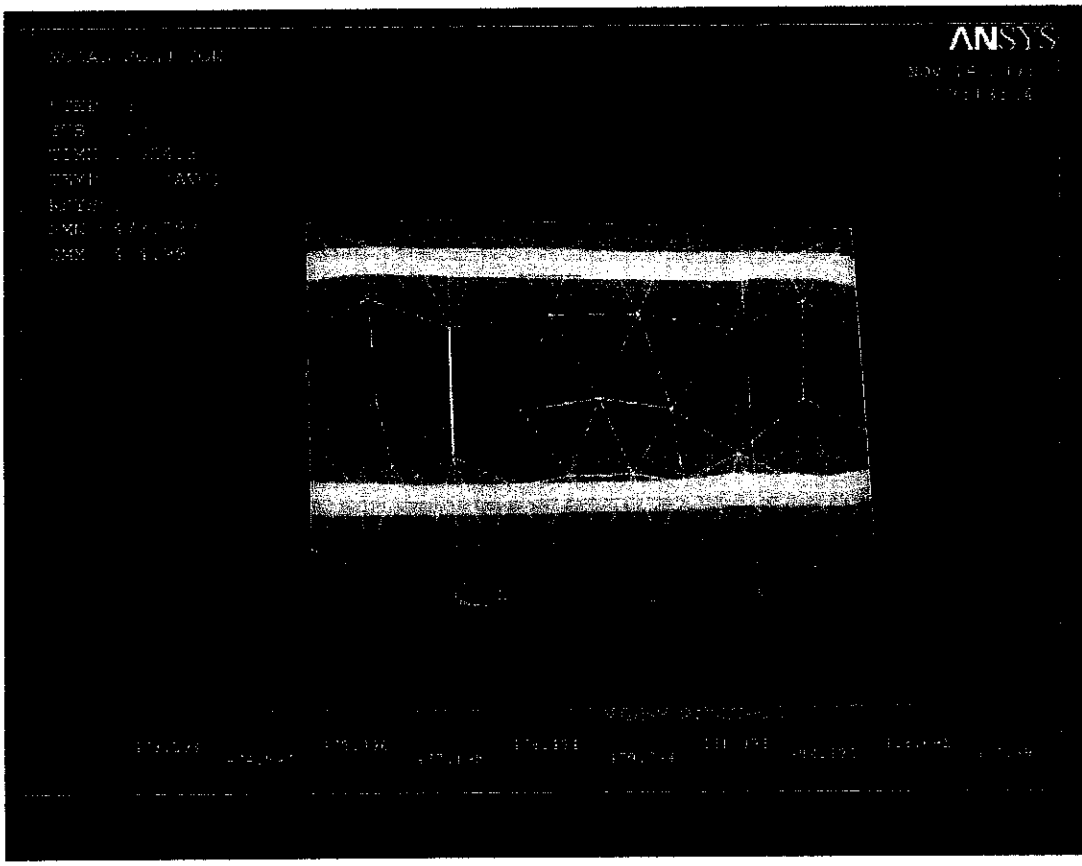


Figure 4.4: Cross-section of temperature distribution at outlet

From the simulation in Figure 4.1 to Figure 4.4, it shows the temperature differences within the vane volume. The maximum temperature achieved is about 485 K which is 212°C. It can be seen that the separation of the temperature is obvious. Also the separation will be more obvious as the airflow approaches outlet. To get better mixing of the airflow, higher rotation of rotor will do the job as increase in Reynolds number produces turbulent flow within the region.

It can be concluded that the thermal capacity of the rotor for Proton Waja has excellent thermal capacity and also excellent in dissipating heat to surrounding. This can be proved by the low maximum temperature for single stopping process.

4.2 THERMAL STRESS ANALYSIS

The next simulations are more related to the stress produced as there is temperature distribution-imbalance within the rotor itself. As seen in the Figure 4.5, the general stress contour especially the for the inlet surface. The compressive stress used in this simulation is 2.725 x 10^8 Nm^{-2}.

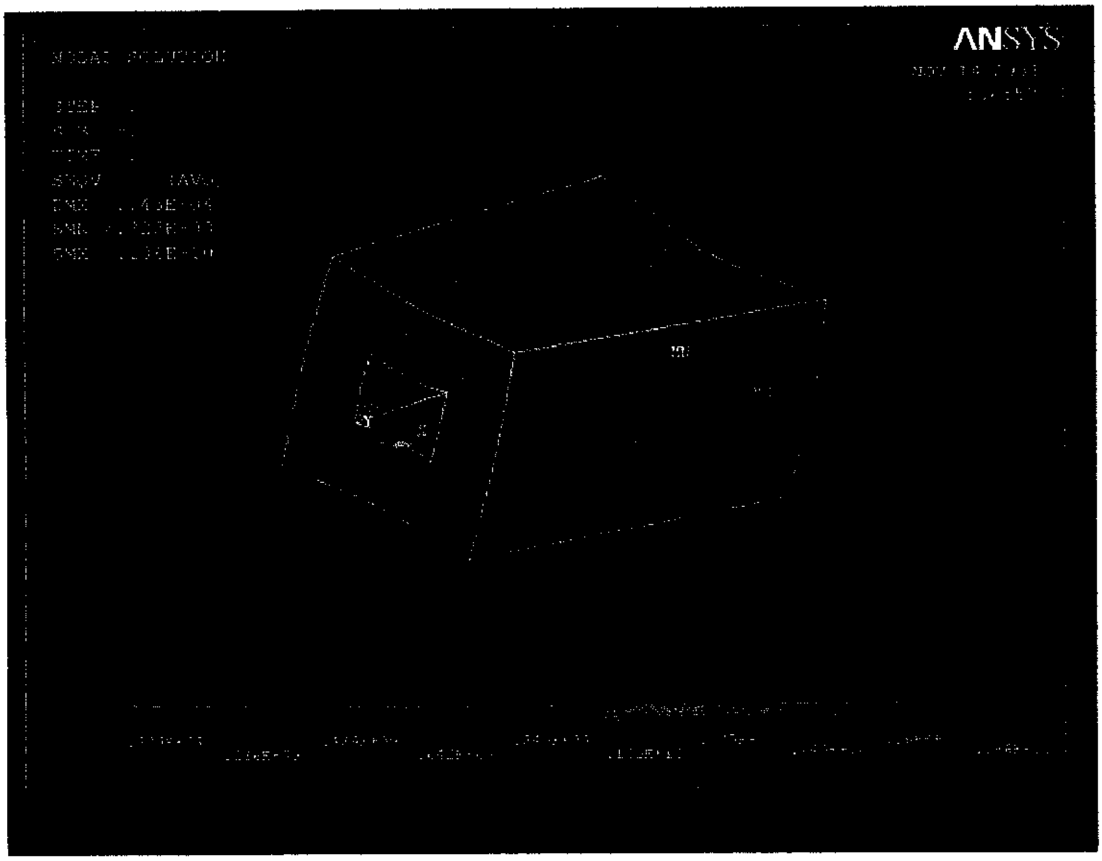


Figure 4.5: Stress distribution at the inlet surface

From Figure 4.5 till Figure 4.8, the variable that leads to the different of the stress distribution is the width of the vane as the area become larger from inlet to outlet. The maximum stress happens to be at the sharp edges of the four corners of the vane at the outlet. For the inlet, the stress is not as high as the outlet.

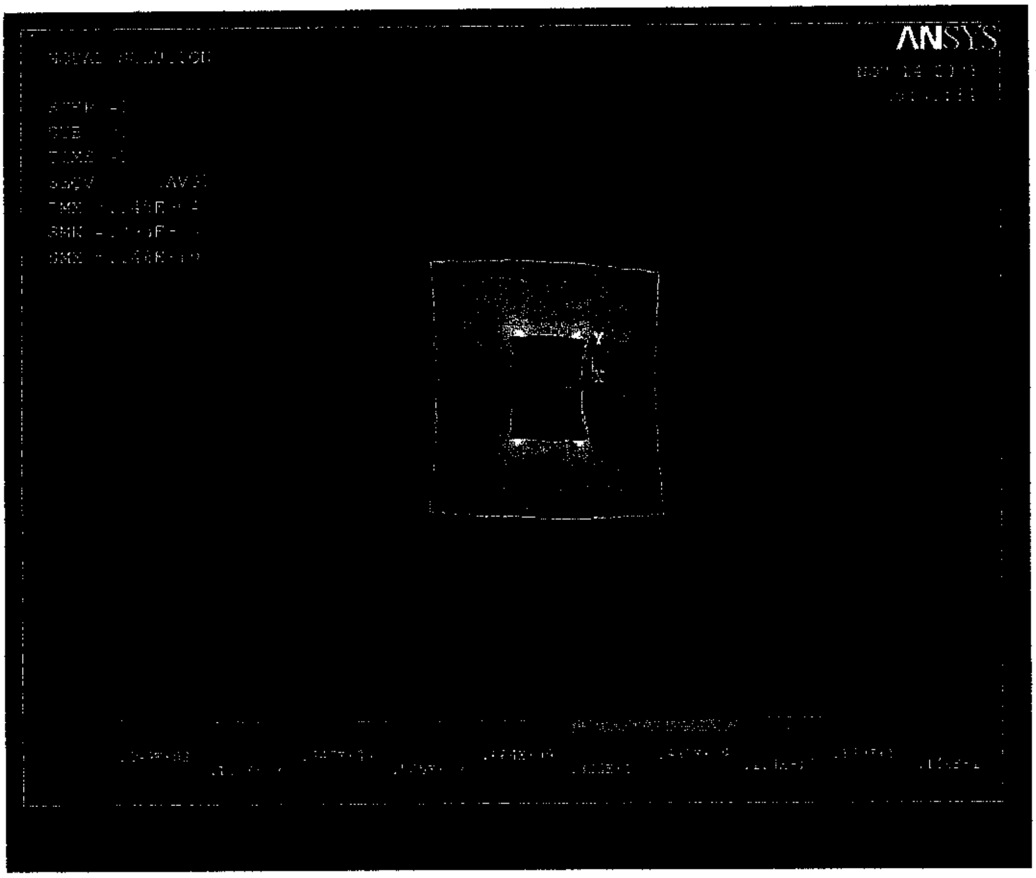


Figure 4.6: Stress distribution on the inlet of the rotor

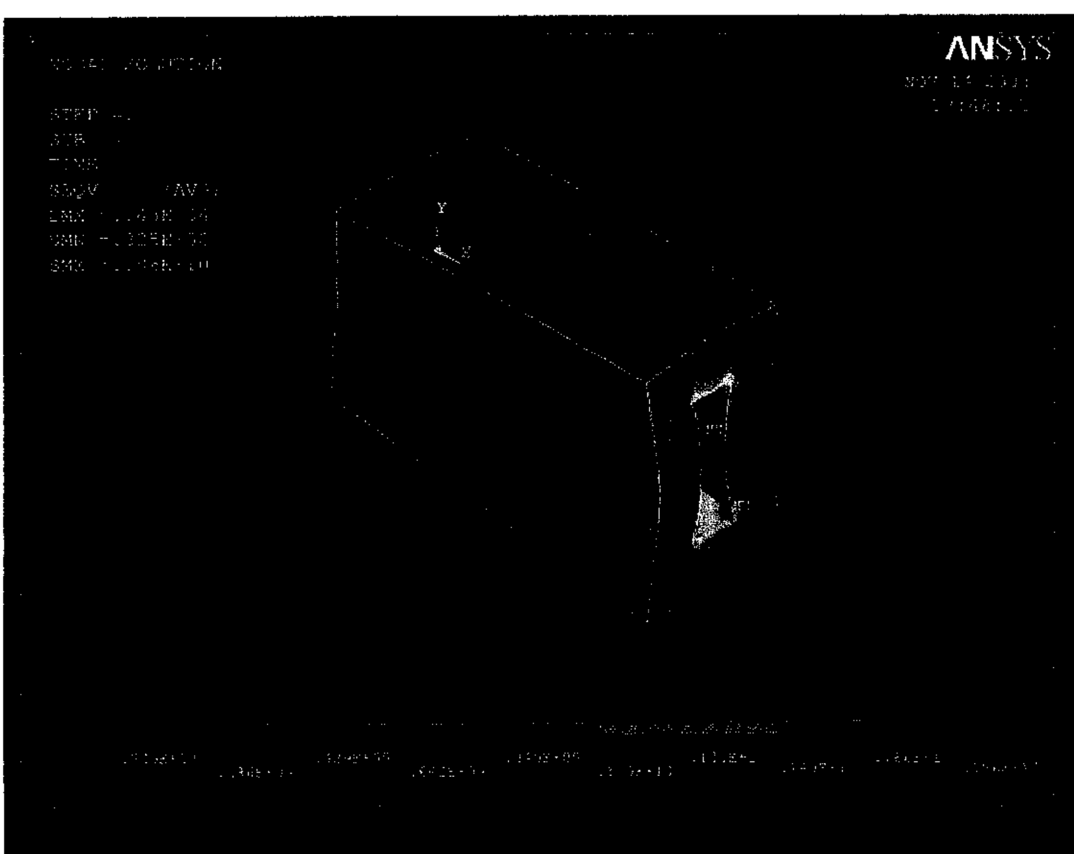


Figure 4.7: Stress Distribution at the Outlet Surface

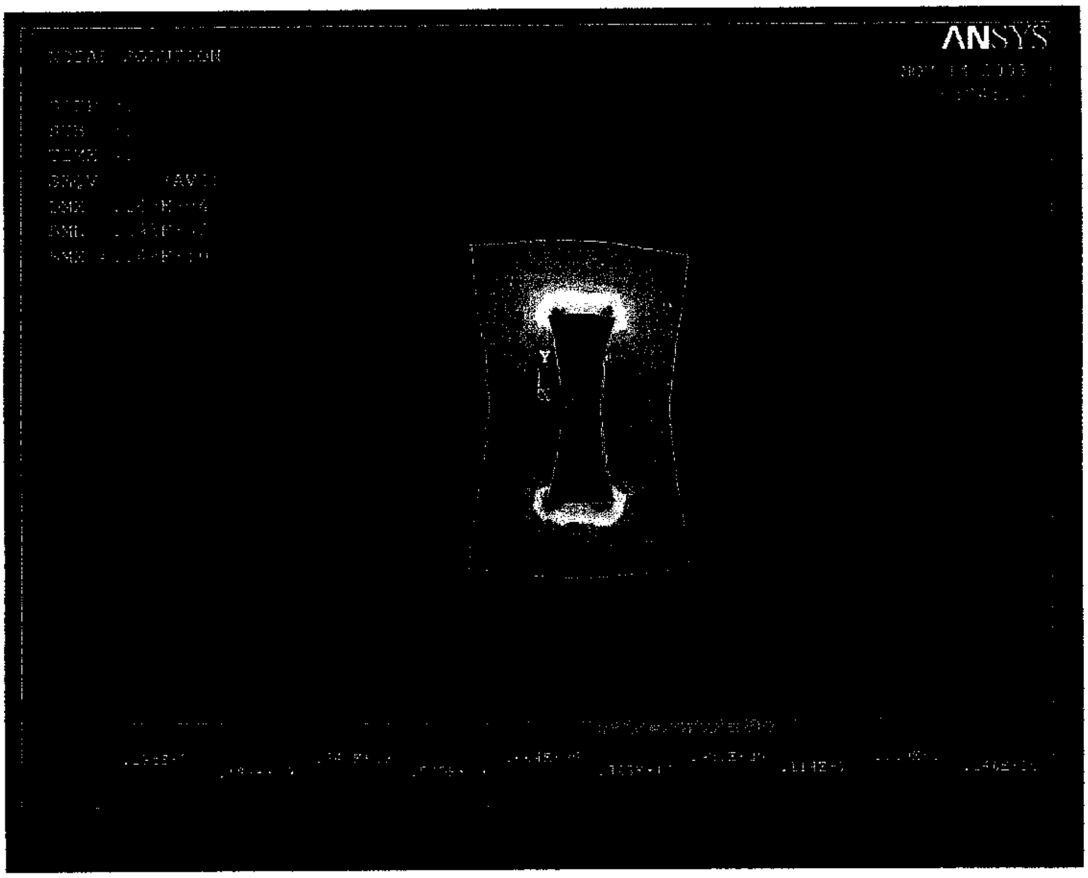


Figure 4.8: Stress distribution on the outlet of rotor

This result is the discussion for the damage mode of warping and cracking (for the cross drilling disc brake). Warping comes from uneven cooling at the area with contact at the pad and the expose area of the disc. The expose area of the disc will cool at a higher rate as compared to the surface in contact with the brake pads. As in the simulation, the ventilated area is the best solution in increasing the cooling rate for the part that is in contact with the disc pad. The ventilated disc with larger inlet area and outlet area reduce the percentage of the warping problem. But there is another variable that still need to be considered to design the disc brake. Thickness of the rotor is the essential factor to avoid cracking. As this simulation is using 2.37cm for the rotor thickness and 0.88cm for the vane thickness, the specification for both thickness are within the limit range of braking safety design.

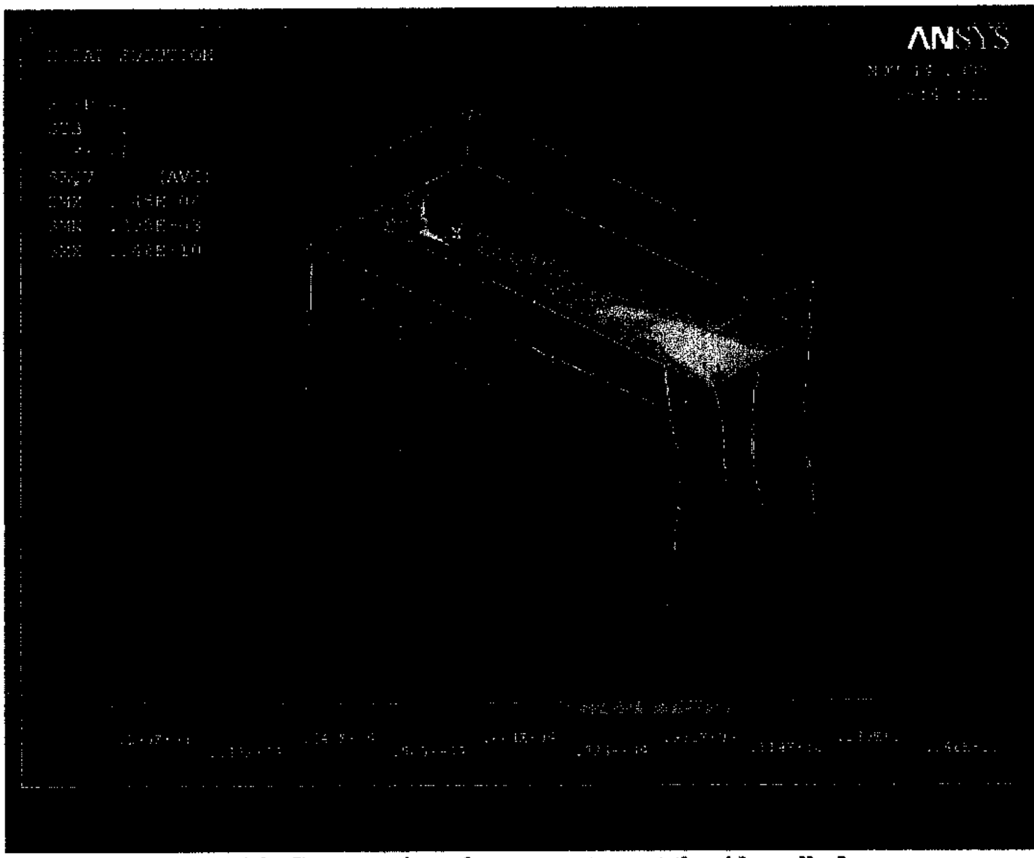


Figure 4.9: Cross-section of stress contour at the side wall of vane

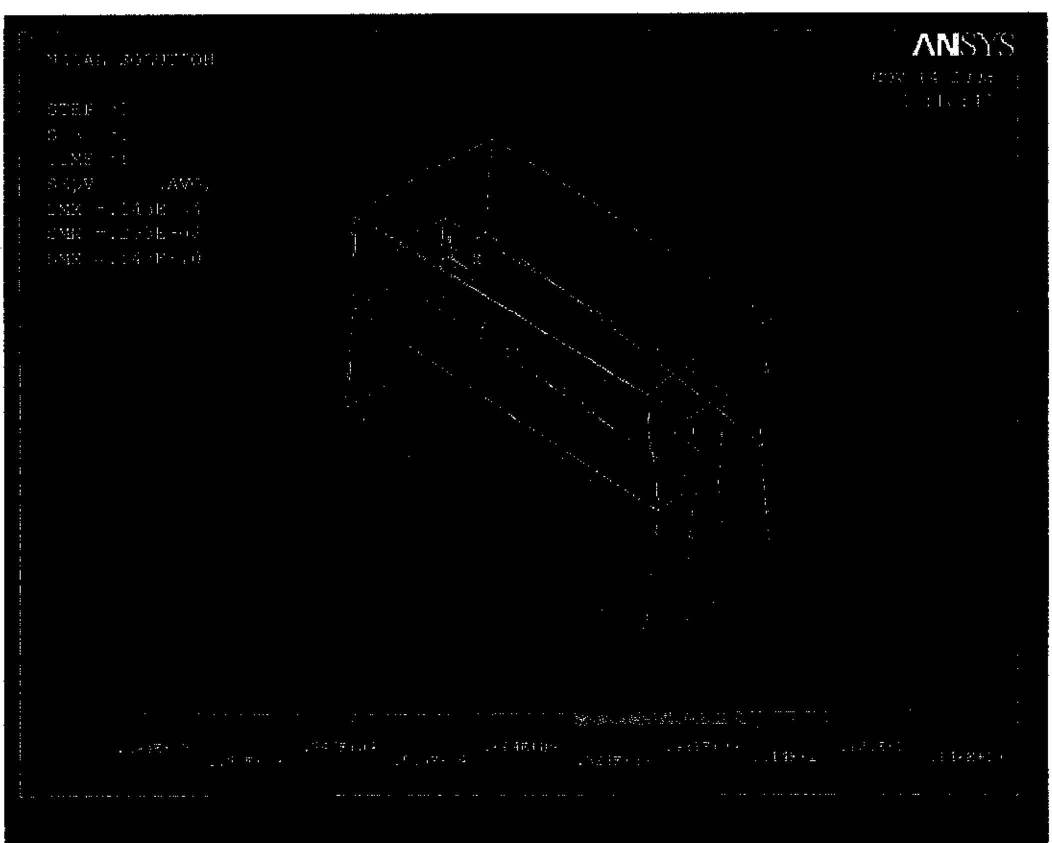


Figure 4.10: Cross-section of stress contour at the mid-point of the vane

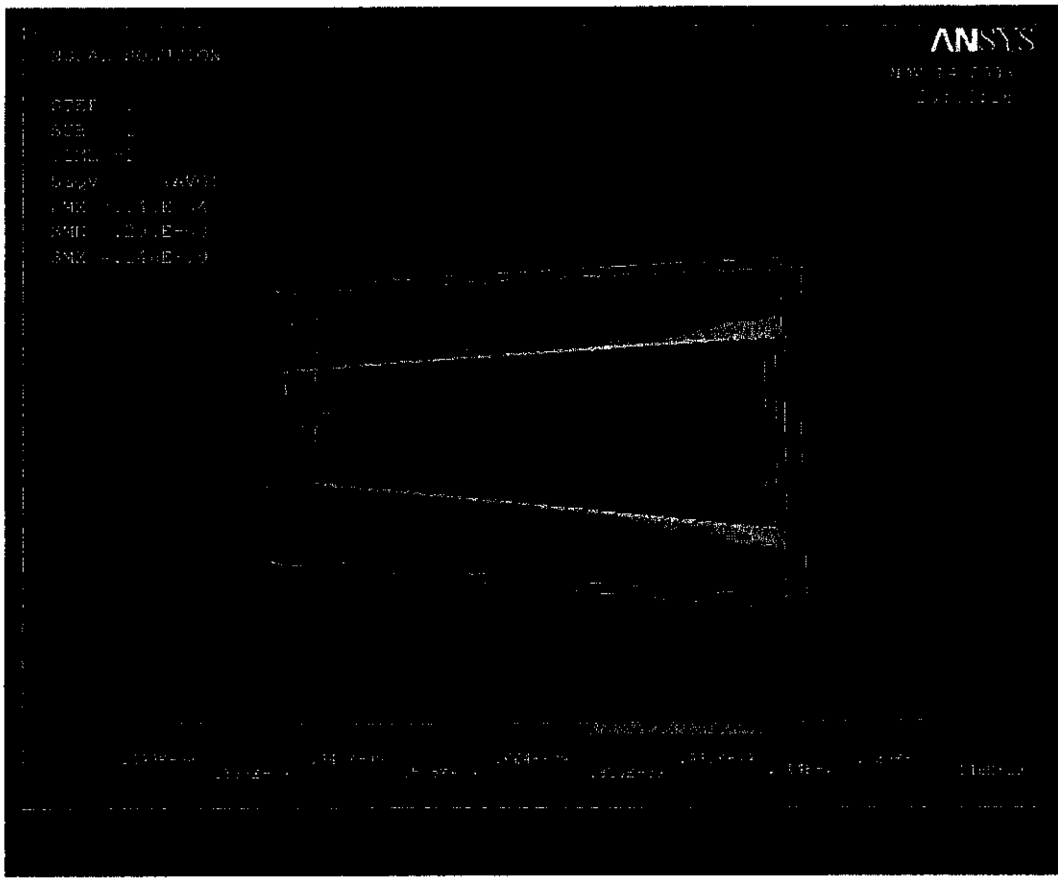


Figure 4.11: Cross-section of stress contour along middle of the vane

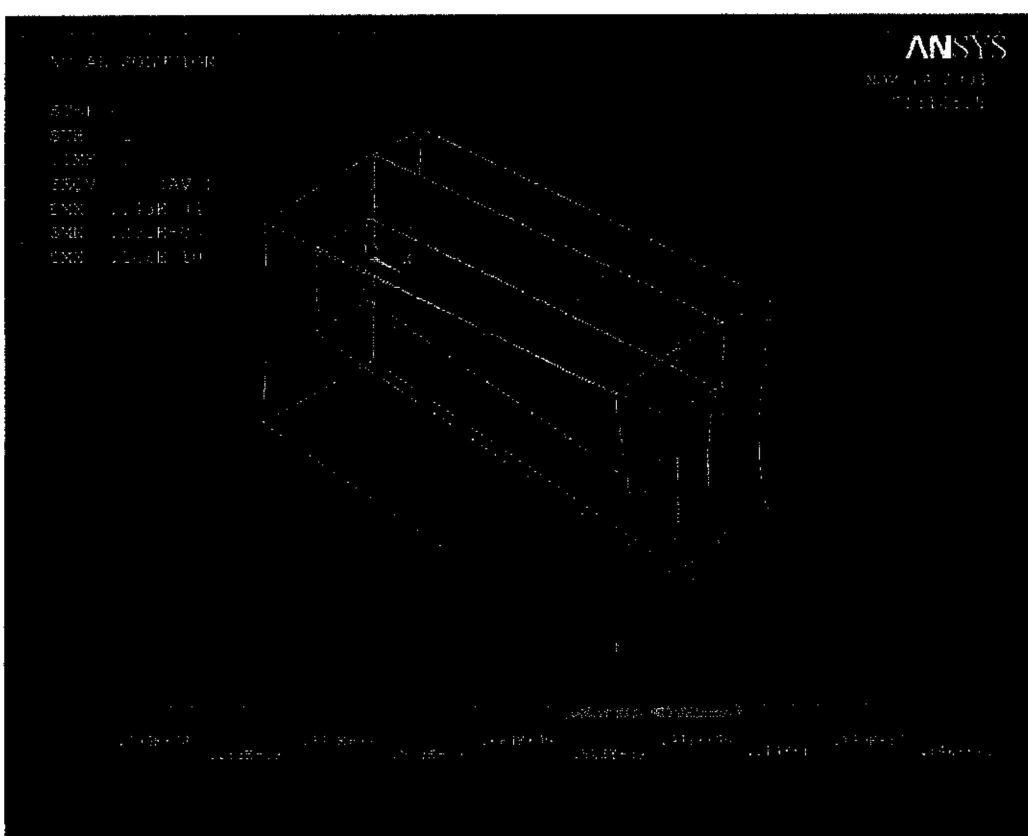


Figure 4.12: Cross-section of stress contour at upper/lower of the vane

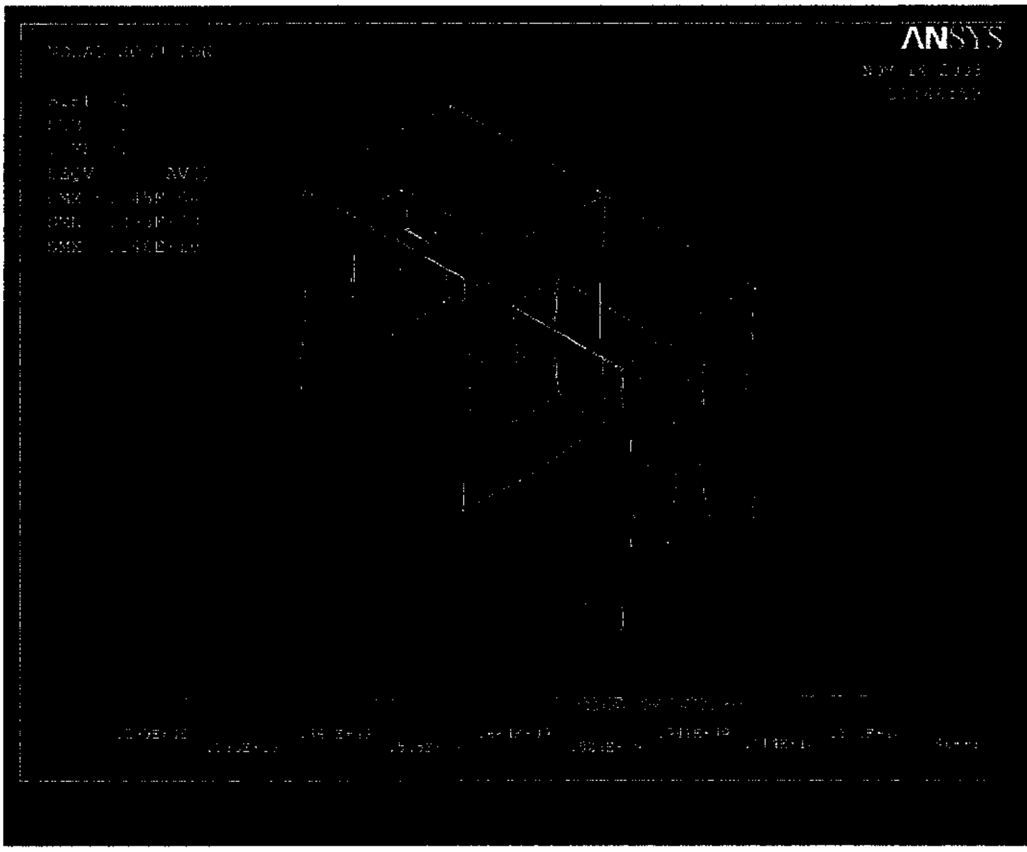


Figure 4.13: Cross-section of stress contour at middle of rotor

As seen in Figure 4.9 till Figure 4.12, there is tendency that the compressive stress becomes larger as the area of the vane increase. This theoretical temperature is only calculating that heat transfer to the brake pad and other cooling mechanisms have been ignored. However, it yields a comparison for the evaluation of a brake system based on temperature limits stated in theory section. The calculation would give an answer of 25.24 K which is less than 500 K. This value is reasonable for commercial car.

From this simulation, it can be concluded that the thermal design measure have been up to standard that the actual thermal performance does not fall below or above safety limits.

CHAPTER 5
RECOMMENDATION AND CONCLUSION

This research is to study performance of the ventilated disc brake and to investigate braking time in conjunction of thermal analysis while in the thermal stress analysis for the damage modes.

This project is basically using the method of simulation in ANSYS software, to analyze the behavior of fluid flow inside a vane. The data are gathered from trusted resources such as from internet, books, journal and dissertation thesis to be used in simulation using the ANSYS software. The result from the simulation using the detail in Proton Waja's disc brake will be used as a reference to improve the design.

The findings of the simulation will be used to suggest a new design to improve braking performance and to reduce the damage modes reoccurrence percentage.

5.1 RECOMMENDATION

Based on this project, there are a few suggestions that may give better result and accuracy. There are:

- More details data and analysis must be achieved as this analysis have many assumptions are made.
- Conduction and radiation needed to be considered in other later research.
- Theoretical analysis such as heat transfer coefficient and surface temperature need to be compared with experimental result to achieve the validity of the value.

5.2 CONCLUSION

Generally, the specification of brake rotor especially its geometry is up safety standard and reliable enough to with stand repetitive and abusive used of the disc. The simulation have inaccuracy are partly because the neglected use of conduction and radiation heat transfer. When the temperature very high, radiation will take effect as it is known as the one of the main sources of heat dissipation. Last but not least, the analysis using the computational fluid dynamic and finite element analysis are one of the best ways in solving this problem as the techniques are important in engineering design and analysis.

REFERENCES

1. *http://en.wikipedia.org/wiki/Disc_brake*, June 5 2005.

2. Smith, C., 2004, The "Warped" Brake Disc and Other Myths of the Braking System Kalpakjianm, Stoptech LLC.

3. Gunther, B., Klingelhoeffer, H., 2004, A Systematic Study for Fatigue Life Prediction of Grey Cast Iron Disc Brakes. In: Fatigue, p. 397–405.

4. Dunlap, K. B., Riehle M. A., Longhouse R. E., An Investigation Overview of Automotive Disc Brake Noise, SAE Technical Paper Series, 1999.

5. *http://en.wikipedia.org/wiki/Coefficient_of_friction*, February 2008.

6. Schmitt. O. and Duncan, T., 2004 Method for extracting full spectrum of friction materials performance using SAE J2681, *Proceeding of the SAE 22nd Annual Brake Colloqium*. p. 25-31

7. *http://www.roymech.co.uk/Useful_Tables/Drive/Brake_Clutch_mat.htm,* 12 October 2007.

8. Erjavec, J., 2002, Tech one: Automotive Brakes, Thompson, Delmar Learning.

9. Hochlernet, D., Hagedor, P., May 2007, Journal of Sound and Vibration, Institute of Mechanics.

10. Limbert, R., 1999 Brake Design and Safety, McGraw Hill.

11. Grieve, D. G., Barton, D. C., Crolla, D. A., Buckingham, J. T. and Chapman, J., 1995, Investigation of Light Weight Material for Brake Rotor

Applications. In Proceedings of the Symposium on Material for Lean Weight Vehicles, University Of Warwick.

12. Grieve, D. G., Barton, D. C., Crolla, D. A., Buckingham, J. T. and Chapman, J., 1996 Light Weight Disc Brake Materials. In Advances in Automotive Braking Technology.

APPENDICES

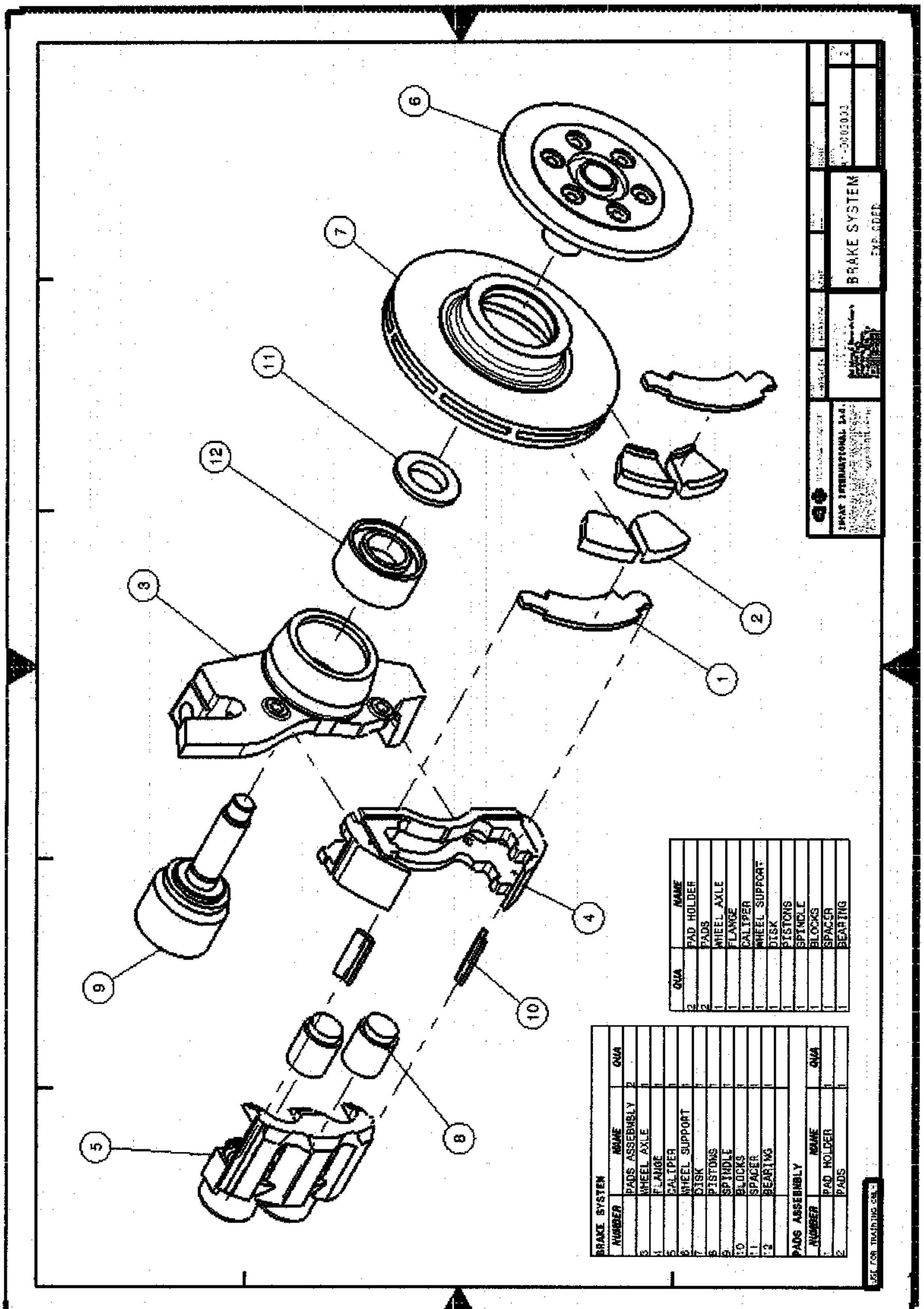
Below are the data and specification of the vehicle to be used in calculation:

1.	Tyre diameter	58 cm
2.	Rotor outer diameter	25 cm
3.	Rotor inner diameter	15 cm
4.	Weight of vehicle	1275 kg
5.	Angular velocity	2122 rpm
6.	Proportions of heat distribution	0.9

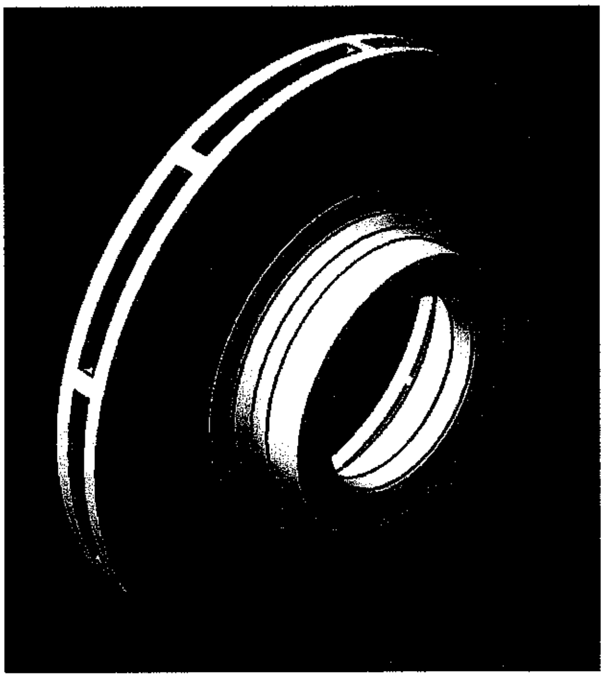
The geometry of rotor vane and the geometric vane relative to the rotor

Material Combination	Coefficient of Friction		Temp.(max)	Pressure (Max)
	Wet	Dry	Deg.C	MPa
Cast Iron/Cast Iron	0,05	0,15-0,20	300	0,8
Cast Iron/Steel	0,06	0,15-0,20	300	0,8-1,3
Hard Steel/Hard Steel	0,05	0,15-0,20	300	0,7
Wood/Cast Iron-steel	0,16	0,2-0,35	150	0,6
Leather/Cast Iron-steel	0,12-0,15	0,3-0,5	100	0,25
Cork/Cast Iron- Steel	0,15-0,25	0,3-0,5	100	0,1
Felt/Cast Iron- Steel	0,18	0,22	140	0,06
Woven Asbestos/Cast Iron- Steel	0,1-0,2	0,3-0,6	250	0,7
Moulded Asbestos/Cast Iron- Steel	0,08-0,12	0,2-0,5	250	1,0
Impregnated Asbestos/Cast Iron- Steel	0,12	0,32	350	1.0
Carbon-graphite/Cast Iron- Steel	0,05-0,1	0,25	500	2.1
Kelvar/Cast Iron- Steel	0,05-0,1	0,35	325	3,0

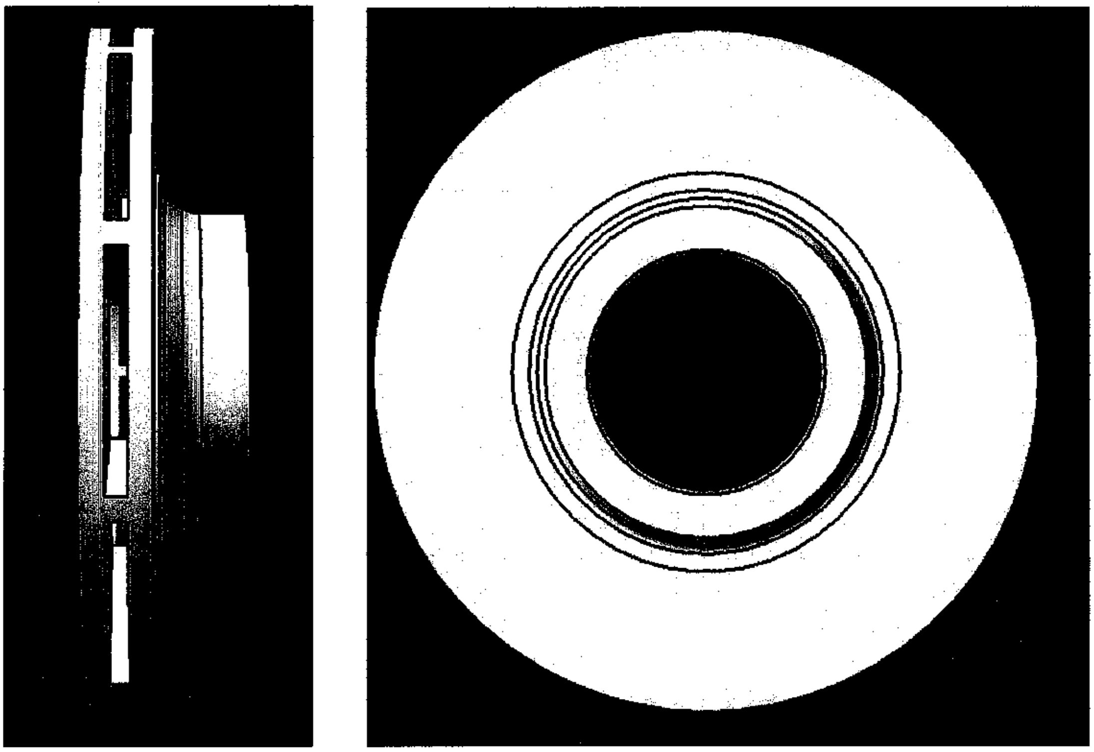
Materials properties for disc brake and brake pads [7]



Disc brake system design



Disc brake system design (disc) – standard isometric view



Disc brake system designs (disc) – front and right view