ADDIS ABABA UNIVERSITY
ADDIS ABABA INSTITUTE OF TECHNOLOGY
SCHOOL OF MULTIDISCIPLINARY ENGINEERING
GRADUATE PROGRAM IN RAILWAY ENGINEERING

OPTIMISATION OF INTERNAL SHAPE OF DISC BRAKE TO INCREASE VENTILLATION

By

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A Research Thesis submitted to the School of Graduate Studies of Addis Ababa University in partial fulfillment of the requirements for the Degree of Masters of Science in Mechanical Engineering

(Railway Engineering Stream)

Advisor

Dr.-Ing. Demiss Alemu

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DECLARATION

I hereby declare that this thesis entitled " OPTIMISATION OF INTERNAL SHAPE OF DISC BRAKE TO INCREASE VENTILATION" is original work of mine and has not been presented for any degree in any university and all the sources of materials used for the thesis have been fully acknowledged.

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Abstract

Brake system has always been one of the most critical active safety systems. Brake cooling is further an important aspect to consider for brake disc durability and performance. In railway transport, braking the train without overheating the discs and losing the braking capacity is a matter of safety. In urban railway transport, like AALRT, the speed of trains is confined to the lower limit because it is difficult to decelerate and stop the train within the short distance of the closely spaced stations. During braking, all the kinetic energy of the train is changed into frictional and finally to heat energy which causes the brake to overheat and lose its braking capacity. To prevent the occurrence of brake heating, brake rotor must be designed such that it ensures sufficient heat dissipation. The special structure for ventilation not only increases the surface area of the disc rotor but also enhances airflow around the surface of the disc to achieve a higher convectional heat transfer. In this research a fluid flow CFX, Thermal and Stress analysis is done on three types of disc brakes; a curved vane and two types of pin vented rotors using finite element method. The fluid flow CFX is used to calculate the cooling effect of the moving air relative to the brake rotor. The pin vented rotors have elliptic cross section with different geometric parameters. Pin vented rotor creates turbulence and gives higher heat transfer coefficient. It has been observed that the pin vented disk brake with elliptic cross section pins which has higher eccentricity has good heat dissipation characteristics. Changing the geometry of the pins increasing the eccentricity of the elliptic cross section creates larger surface area to volume ratio and less obstruction to the air flow which enables to have higher convection heat transfer coefficient. After emergency braking of AALRT with maximum track gradient of 3.15^0 and overload condition, a maximum temperature gradient of 334.95^0C is achieved with one of the pin vented brake which has higher eccentricity while it is 340.93^oC and 346.89^oC for the second type of pin vented brake and the curved vane rotors respectively.

Table of Contents Pages

List of Tables Pages

List of Figures Pages

NOTATIONS

F_{disc}	Resultant force of disc
M	Total mass acceleration or deceleration
V	Final velocity
V_0	Initial velocity
S_b	Braking distance
T_b	Braking time
μ	Coefficient of friction
α	Gradient
C_p	Specific heat
Q	Internal heat generation
T	Temperature
Q_s	Surface heat flux
H	Convective heat transfer coefficient
T	Surface temperature
$T\infty$	Convective exchange temperature
ω	Angular velocity
ρ	Density of fluid
V	Velocity of passenger train
Ω_o	Initial angular velocity the wheel
r_w	Radius of vehicle wheel
I	Polar inertial moment of rotating parts
ε	The material's emissivity,
σ	Stefan-Boltzmann constant and
T_{ref}	Temperature of the surrounding air
CFD	Computational-fluid-dynamics

CHAPTER ONE

INTRODUCTION

1.1 Background

Friction braking systems are still dominating method of retardation of train vehicles. Disc brakes are important components of a vehicle retardation system. They are preferred than other friction braking systems for their good heat dissipation properties.

During braking, a set of pads is pressed against a rotating disc and, due to friction; heat is generated at the disc-pad interface, which causes the disc surface temperature to rise in a short period of time. This heat ultimately transfers to the vehicle and the environment, and the disc cools down.

The field of brake cooling has been a subject of research since the 1950s. Vehicle speeds and weights have steadily increased. These increases lead to braking energy rise resulting in higher brake temperatures, necessitating further research. Most of the researches on brakes are engaged on increasing the heat capacity dealing with the brake disc material and modifying the structure using some features to increase heat dissipation.

Initially, analytical techniques were used to predict brake temperatures during which were limited to steady state analysis and simple geometries. Today, the prevalent way to simulate frictional heating of disc brakes in commercial software is to use the fully coupled FEM.Finite element mesh of a disc rotates relative to a brake pad and, CFX, thermal and mechanical analysis are performed simultaneously.

1.2 Statement of the problem

Disc overheating is the main problem that causes a decrease in braking performance. It also causes judder and noise which disturbs the passengers and causes a stress on the rail. Due to braking capacity limitation the speed of urban trains decreased significantly for a smaller kinetic energy in order to stop the rail vehicle within a short distance. Smaller speeds of urban trains bring about customer dissatisfaction and can significantly reduce profit of railway companies. The following picture shows how the disc is over heated during a test procedure and how the color changed consequently.

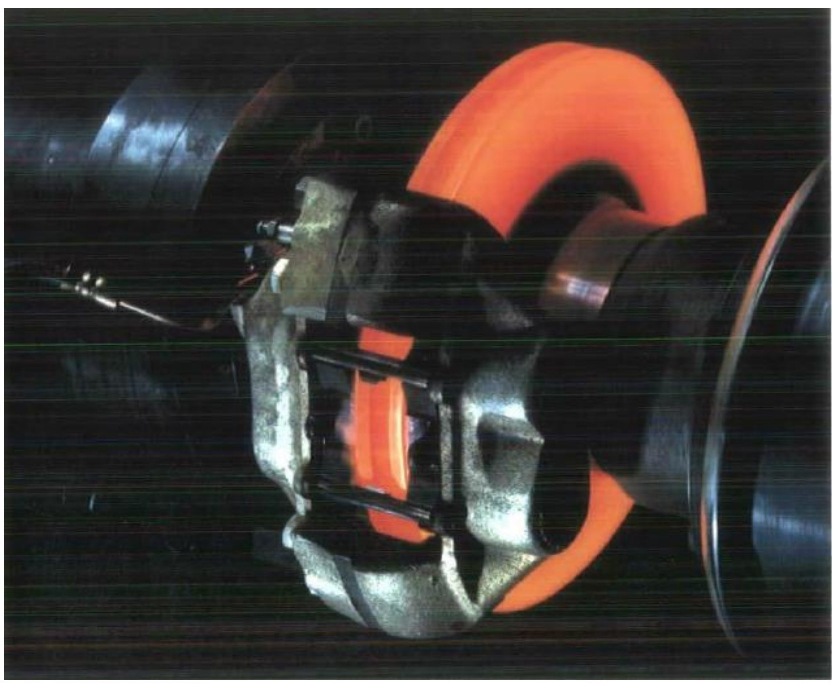

Figure.1.1 Disc overheating during a testing process

1.3 Objective of the study

1.3.1 General Objective

- ✓ To decrease the braking distance and thus increase the operating speed
- ✓ To increase the safety and controllability of trains in emergency and normal braking operations
- ✓ To reduce maintenance cost, increase the service life of breaks lowering the thermal stress on the brakes

1.3.2 Specific Objectives

- ➢ To improve structure of ventilated type disc brake and maximize the heat dissipation in order to keep the disc brake rotor temperature within an acceptable limit which does not degrade the braking capacity during a heavy load braking action. This can be achieved increasing the surface area using some features on the rotor and modeling the rotor in such a way that increases and favors air mass flow for a higher convectional heat transfer.

- ➢ Clearly identify explain and show the features which are important to stabilize the disc overheating by using different rotor models and features.

1.4 Research Methodology

To achieve the objectives of the study the following methods and procedures will be followed.

1. Data collecting and literature survey on similar area of application will be conducted like journals and scientific researches.
2. Analysis of forces acting on the brake which would later input to the software ANSYS
3. Modeling of three different alternatives of the disc using CATIA which would be important for comparison and to identify the important features in terms of their differences.

One will be the curved vane type, the second and third would be a modified pin vented types of disc brake.

4. Application of simulated mechanical and thermal loading conditions of the three disc models on ANSYS after importing the geometry from CATIA.
5. Results and discussion and comparison of results of the three models in terms of temperature and their thermo mechanical stress
6. Conclusion on the finding of the research, the best disc model, and identify best features which help to stabilize disc overheat. To present the importance of the features used to achieve a decrease in braking distance and increase in peak operating speed.

1.5 Significance of the Research

The end result of this study is expected to be a design of ventilated type disc brake with a better heat dissipating quality and performance without compromising the mechanical stress capacity which may be affected while changing the structure of the disc. Being able to reduce the thermal stress, the performance of the brake can be significantly increased. This is because high amount of heat generated on the disc is the main problem that minimizes braking performance.

1.6Scope and Limitations of the Study

I can say that the procedures and the approach followed to conduct this study is scientific, has a logical flow and the results could be dependable except the fact that it is done using a simulation software and no experiments are carried out.

Also it is known that the structural strength of the rotor can be compromised while adding such features as holes and slots but it can be resolved using proper material with an adequate material strength and the disc should be designed with a good safety factor.

1.7 Organization of the Research

To achieve the stated objectives of the research a detailed review of the relevant literature was completed. The structure of the thesis is as follows:

- ✓ Chapter one introduces the research and outlines the aims, objectives and scope of the work.
- ✓ Chapter two provides a preliminary investigation into the theory of disc heat dissipation and related aerodynamics, and includes a detailed review of the relevant background literature.
- ✓ Chapter three contains mathematical modeling of the research that is calculation of different forces and stresses acting on the disc rotor. It includes geometrical modeling of the disc using CATIA and simulating results with ANSYS.
- ✓ Chapter four presents the results and discusses the results.
- ✓ Chapter five states conclusions and major findings of the research, outlines recommendations, and feature work on how the research could be further developed.

CHAPTER TWO
LITERATURE REVIEW

2.1 Introduction

The function of any braking system is to retard the vehicle smoothly, consistently and predictably. It must have the capability to slow the vehicle rapidly if required and from high speeds. It must respond quickly when activated and must maintain stability throughout the stop, having no variation or imbalance across the vehicle that might lead to a loss of directional stability and loss of control over the vehicle by the driver. It must be able to be operated to full effect by a person of any reasonable age or strength. There are many more requirements with which the braking system must comply, including many legal regulations. The braking components, be they based on disc or drum systems, have their own specific requirements. Primarily a braking torque must be provided on the rotating axle. This is achieved by a friction pair, the disc and brake pad. The rotor is mounted on the rotating axle that is to be braked and is often located inside the wheel itself. Its primary function is to transmit the mechanical braking torque caused by the very high frictional forces on the spinning rotor to the axle and subsequently the wheels, slowing the vehicle.

The high frictional forces generate large amounts of heat as the kinetic energy of the moving vehicle is converted to heat energy in the friction pair during braking. The secondary function of the rotor/brake pad pair is to dissipate this heat energy preventing excessive heat accumulation, high temperatures and the resulting damage and deterioration of surrounding components.

2.2 Brake Types

Historically there are two main methods of achieving this friction pair, both of which are still in common usage; drum brakes and disc brakes. Braking with a drum brake is achieved by applying outward pressure with stationary friction pads (brake shoes) to the interior of a spinning drum. The pads oppose one another but drum distortion occurs. Pad wear distribution depends on the actuating mechanism but it is not always uniform. The friction pair is enclosed within the drum which protects it from road debris but does nothing to encourage effective cooling.

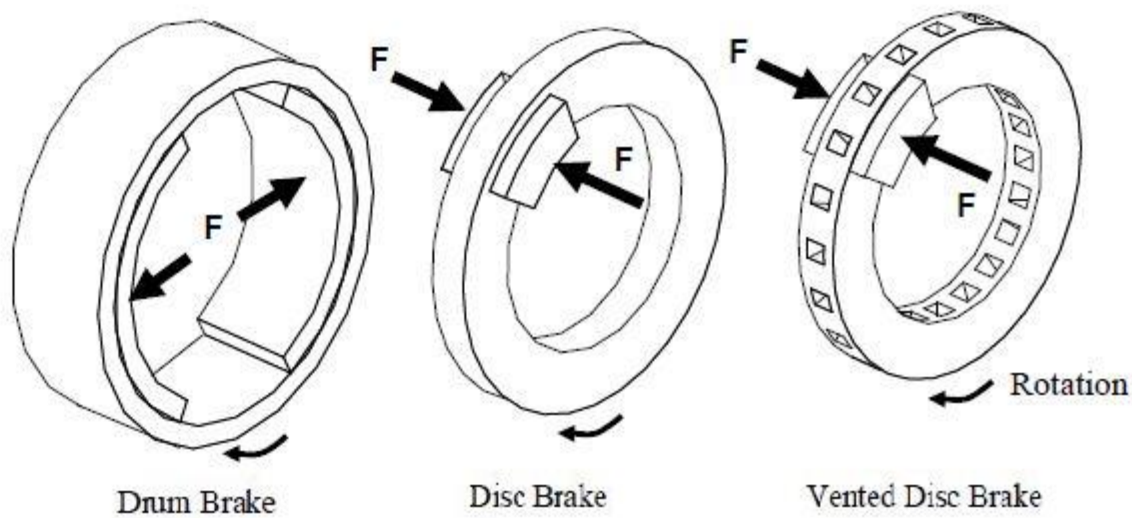

Figure.2.1 Brake types

In the case of the disc brake two stationary pads oppose each other across the thickness of the spinning disc. The disc is much more exposed and better convective cooling is achieved even though the design is prone to the invasion of debris. In addition there is a shorter thermal path for heat to be conducted from the braking faces to the axle/wheel.

There are two major types of brake disc; solid and ventilated. The ventilated disc gives increased cooling ability without a proportional increase in weight, but it is more complex in its design. In a ventilated type brake disc air is able to flow through the air passages between the cooling fins, carrying heat away from the braking surfaces. A solid disc, as its name suggests, has no such air passages and as such has poorer cooling performance.

2.3 Brake Components and Limitations

Disc Brake Caliper Assembly

The caliper/pad is the actuating mechanism for applying force on the brake pads. It is usually a solid casting containing a hydraulic system to provide the force on the back of the disc pads. In this case due to constraints of mounting space inside the wheel the caliper contains only one hydraulic cylinder and this is used to create a reaction force between the pad it directly acts upon and an opposite pad restrained by the opposite side of the caliper. This caliper is required to be mounted on a sliding pin arrangement so that a small amount of lateral movement is available to create equal and opposing forces across the thickness of the disc. It is also common to have a rigidly mounted caliper with two opposing hydraulic cylinders to provide balanced pad forces. The disc pads are removable and need to be replaced periodically due to wear. The pad itself consists of a steel back plate onto which is mounted (usually with special adhesives) a softer friction pad. This pad, a fiber composite, used to be a combination of asbestos and a binding resin but other materials have been developed with specially engineered properties. The pad is forced into contact with the spinning disc and is subjected to very high pressures, temperatures and compressive and shear loads.

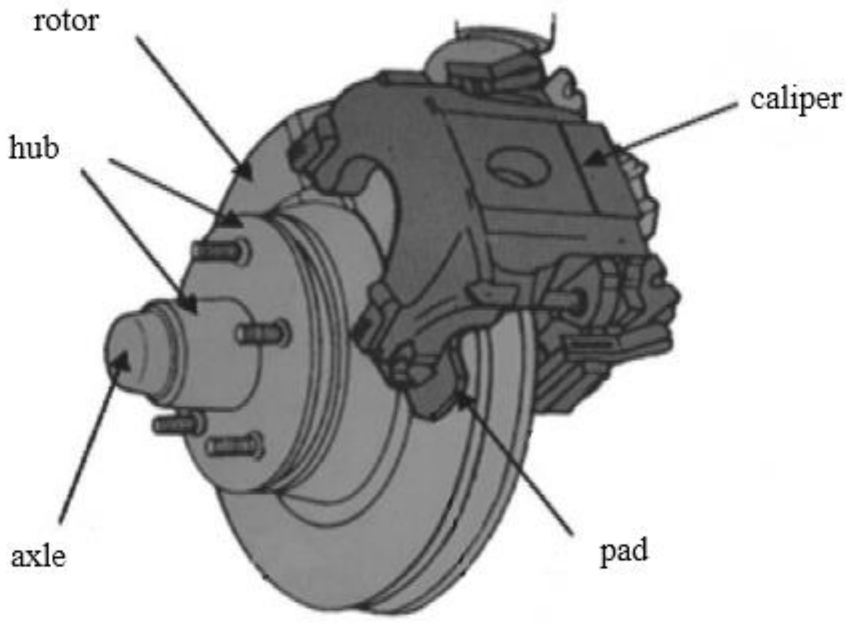

Figure.2.2 Disc brake caliper assembly

Brake Disc

Brake discs subject to three main problems in service; cracking, wear and distortion. Cracking occurs usually because of high stresses (cyclic in nature) caused by temperature differences across parts of the disc. These cracks often appear in the braking face where temperatures are highest and result in severe weaknesses in the disc and increased pad face wear. These cracks can grow rapidly because of the cyclic nature of brake disc operation. It is important to select both disc design and material carefully to ensure that any thermal deformation that takes place is not so great as to put a significant amount of stress on the component. Therefore if brake disc distortion can be reduced it is likely that corresponding stresses and therefore potential cracking will be reduced. Disc wear is caused by the rubbing of the pad on the rotating disc.

There are two types of distortion that can occur during brake operation. The first is due to the high mechanical forces on the disc during heavy braking and the transmission of the braking torque to the axle.

The disc must be stiff enough to withstand these loads without raising internal stresses to a level where they either generate cracks or enter a plastic deformation region. The disc must be designed to remain within its elastic zone at all instances of operation, maintaining dimensional stability. The second, and main, form of disc distortion is thermal deformation arising from large temperature differences across the disc due to heavy braking. Four basic types of thermal deformation occur in discs.

i) Radial expansion - produces a radial displacement of the braking face at the position of the brake pads on the surface of the disc.

ii) Coning - results in an axial deflection at the outer edge of the braking faces producing a conical deformation of the disc. This forces the pads apart slightly, changing the pressure distribution of the pad on the disc and hence creating an uneven heat input to the braking face. Braking effectiveness will be reduced as a result.

iii) Waving - results in torque oscillations as the wheel turns, introducing instability to the braking system and a loss of smooth operation. This torque fluctuation also manifests itself in 'brake judder', a low frequency noise and vibration.

iv) Rippling - takes the form of localized deformation between the ventilation ribs due to the thermal stresses, and resulting strains, induced between the constraints of the ribs around the rotor braking face. This adversely affects pad contact pressure distribution and stability resulting in reduced braking efficiency and possibly in such effects as 'squeal' where a high frequency vibration of the pad is initiated causing loud, high-pitched noise during certain braking conditions.

Of these deformation characteristics, waving and coning are the most important having the largest effect on braking performance. It is important that the disc must be designed to conduct heat to the hub effectively and without incurring large distortions such as those described above. It must also be able to lose heat to its surroundings by forced convection and it is very important that the material and design is able to withstand the continual thermal cycling without loss of performance or integrity.

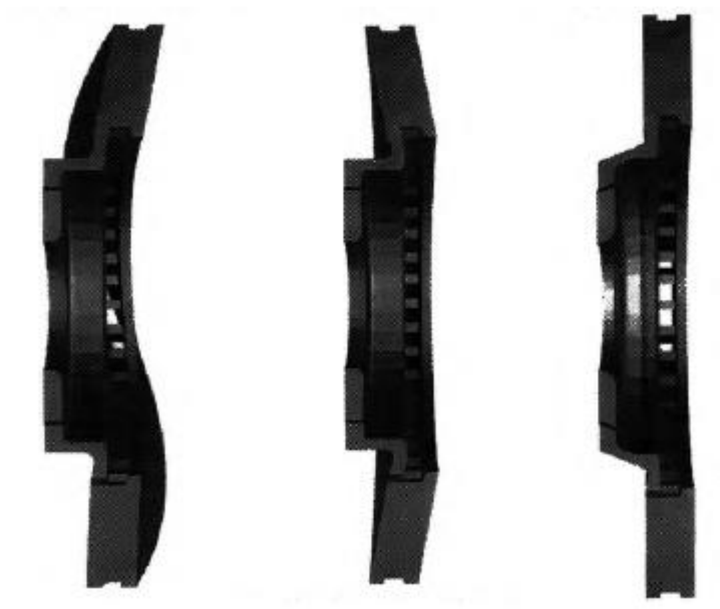

Figure.2.3 Principal distortion types(Waving, Coning and Radial expansion)

Brake Pad

If the pad is poorly designed it will distort under heavy loading resulting in uneven pressure distribution. The resulting uneven heat input into the braking face can result in localized heat buildup with associated localized high temperature regions. The pad back plate stiffness must be carefully determined along with the optimum friction pad material compressibility to give sufficient pad rigidity for caliper piston loads to be transmitted evenly and uniformly over disc/pad contact area. As very high temperatures are developed at the interface it is also vital that the pad friction material has a very low thermal conductivity. Any significant increase in caliper bulk temperature would result in brake fluid degradation and impaired function of the braking system. In addition rubber piston seals would be damaged, considerably reducing the operating life of the components. The wear characteristic of the friction material must be carefully selected to ensure that the proposed pressures, surface speed and temperatures will not cause the material to wear too quickly or to cause too much wear to the disc i.e. the material is not to abrasive.

A friction film is developed in the high temperature/pressure contact region and this film adheres to the disc surface. This has only recently been studied and research is emerging on the characteristics of this film and its importance. If significant disc run-out is evident and the pad material is abrasive then this wear will be localized, causing judder during a brake application that is unrelated to thermal distortion. Pad material compressive properties are important in determining pad pressure distributions especially in the presence of either disc or caliper distortion.

Caliper

The most important attribute of the caliper is stiffness. Any flexure in the component results in it 'opening out' under loading. This inevitably causes uneven loading of the pad back plate and hence uneven pressure distribution at the disc/pad interface. As an unsprung component it is also desirable for the caliper to be as light as possible without adversely affecting its stiffness. Some capability for heat dissipation must be included in its design. There are two types of caliper arrangements.

Fixed caliper

A disc brake system which uses fixed coupled pistons, e.g. one, two or three pistons on each side, is called a fixed caliper braking system. Pistons on both sides are pushing respective brake pad directly against respective side of the brake disc, to create a frictional force which provides a braking momentum on the rotor.

Floating caliper

A floating caliper braking system uses pistons on one side of the brake disc only. These pistons push the brake pad against the inner side of the rotor at the same time as they push the caliper backwards with the same force. The resultant force of the caliper affects the other side of the rotor where the outer brake pad will push the outer rotor side. The most common disc brake type is with a floating caliper braking system. The floating caliper is often preferred over fixed caliper due to both economical and its lower mass/volume. The cost will increase due to the double amount of pistons and the system will take up more volume and mass with a fixed caliper system.

Overheating of Brakes

If the temperatures reached in braking become too high, deterioration in braking may result, and in extreme conditions complete failure of the braking system can occur. It can be difficult to attribute thermal brake failure to motor vehicle accidents as normal braking operation may return to the vehicle when the temperatures return to below their critical level[1]. One of the most common problems caused by high temperatures is brake fade; other problems that may occur are excessive component wear, rotor deterioration, and thermally excited vibration (brake judder). Heat conduction to surrounding components can also lead to damaged seals, brake fluid vaporization, as well as wheel bearing damage. The major problems associated with elevated brake temperatures are outlined below.

1. Brake Fade

Brake fade is a temporary loss of braking that occurs as a result of very high temperatures in the friction material. The high temperature reduces the coefficient of friction between the friction material and the rotor, and results in reduced braking effectiveness and ultimately failure. Generally fade is designed to occur at temperatures lower than the flame temperature of the friction material to reduce the possibility of fire at extreme temperatures. Normal braking will usually return when temperatures drop below their critical level.

2. Brake Fluid Vaporization

Most braking systems are hydraulically actuated. If temperatures reached during braking exceed the boiling point of the hydraulic fluid then brake fluid vaporization will occur. A vapor lock will then form in the hydraulic circuit, and as gas is more compressible than liquid the pedal stroke is used to compress this gas without actuating the brakes. Brake fluid is hydroscopic causing it to absorb water from the atmosphere over time; this may result in a reduced boiling temperature of the fluid. Therefore it is usually recommended by vehicle manufacturers to replace brake fluid periodically.

3. Excessive Component Wear

High temperatures in the braking system can form thermal deformation of the rotors leading to uneven braking, accelerated wear and premature replacement. The life of the friction material is also temperature dependent, at higher temperatures chemical reactions in the friction material may cause a breakdown in its mechanical strength, which reduces braking effectiveness and causes rapid wear.

The wear of frictional material is directly proportional to contact pressure, but exponentially related to temperature; therefore more rapid wear will occur at elevated temperatures.

4. Thermal Judder

On application of the vehicles brakes, low frequency vibrations may occur. This phenomenon is known as ‘judder’. Two types of judder exists; hot (or thermal judder) and cold judder. Cold judder is caused by uneven thickness of the rotor, known as disc thickness variation, this leads to deviations in contact pressure as the pads connect with the rotor. This results in uneven braking or brake torque variation. The second type, thermal judder, occurs at elevated temperatures, and is caused by thermal deformation of the rotor. When a rotor containing a cold disc thickness variation is subjected to braking; the contact pressure in the thicker parts will be much greater than the thinner parts. As a result, the thicker parts become hotter causing uneven thermal expansion of the rotor, which compounds the original disc thickness variation and creates a “self-accelerating instability”[2]. Thermal judder can also be a result of ‘hotspots’ on the rotor surface. Hotspots can be caused by localized contact between the pads and rotor resulting in small areas of very high temperatures, >700 °C, which causes a thermal disc thickness variation. This thermal disc thickness variation may develop into a permanent disc thickness variation due to a phase change from pearlite to martensite[3] when cast iron is cooled rapidly. Martensite occupies a larger volume than pearlite, and therefore a cold disc thickness variation is formed and the problem is again compounded.

2.4 Heat dissipation from Disc Brakes

The rise in temperature of the brake disc in any braking operation will depend on a number of factors including the mass of the vehicle, the rate of retardation, and the duration of the braking event. In the case of short duration brake applications with low retardation, the rotor and friction material may absorb all of the thermal energy generated. As a result very little heat dissipation occurs as the temperature rise in the rotor is minimal. In extreme braking operations such as steep descents or repeated high speed brake applications, sufficient heat dissipation becomes critical to ensure reliable continued braking. As the rotor temperature rises it begins to dissipate heat, at steady-state conditions heat generated through braking equals heat dissipation and no further heating occurs. If the heat generation is greater than the dissipation then the temperature will rise, the rate of this rise will depend of the relative quantities of each.

If sufficient heat dissipation does not occur the temperature of the rotor and friction material can reach critical levels and brake failure may occur. Heat dissipation from the brake disc will occur via conduction through the brake assembly and hub, radiation to nearby components and convection to the atmosphere.

At high temperatures heat may create chemical reactions in the friction material, which may dissipate some of the braking energy. While conduction is an effective mode of heat transfer it can have adverse effects on nearby components. Such effects include damaged seals, brake fluid vaporization, as well as wheel bearing damage. Convection to the atmosphere must then be the primary means of heat dissipation from the brake rotor. Convection is governed by the expression:

$$Q=hAs(Ts-Tr) \qquad (2.1)$$

,also known as Newton's law of cooling.

Where:

Q = the rate of heat transfer (Watts),

h = the convection heat transfer coefficient (W/m2 k),

As = the surface area of the rotor (m2), and

Ts and *T*r are the surface temperatures of the brake rotor and ambient air temperature respectively.

It can be seen from this expression that in order to maximize heat transfer from the rotor(increase Q) and keep the rotor temperature (*Ts*) to a minimum, the value of heat transfer coefficient (*h*), or the surface area *(As)* needs to be increased. As it is required to keep *Ts*to a minimum, improvements must be made through increasing the heat transfer coefficient (*h*) and or the surface area *(As)* of the rotor.

The amount by which the surface area can increase is confined by the diameter of the wheel and the requirements of minimizing unsprang mass1, so improvements in cooling can be made through increased values of the heat transfer coefficient[4]. This heat transfer coefficient is dependent on the boundary layer, which is influenced by surface geometry, the nature of the fluid motion around the rotor, as well as thermodynamic and fluid transfer properties[5].

The use of an internally ventilated rotor will increase both surface area (extra internal area exposed to the atmosphere) and the heat transfer coefficient, due to forced convection created by the internal airflow, with negligible influence on unsprung mass. The material selection and the physical dimensions of the rotor will also have a direct bearing on cooling ability. The steady state surface temperatures decrease with increasing thermal conductivity of the rotor. This was mainly attributed to the ability of high conductive materials to transmit heat into the hub and wheel assembly, which act as a heat sink. However, as discussed previously this is not a particularly advantageous situation, and therefore the more energy dissipated to the surrounding air the better. The greater the volume of air which interacts with the heated elements, the greater the heat dissipation. The study of aerodynamics has long been used to optimize the airflow around vehicles.

2.5Computational Fluid Dynamics

The high financial implications and the long development time associated with wind tunnel development of vehicles shapes has led to look for cheaper viable alternatives. Computational fluid dynamics relies on the use of computers to locally solve the equations that describe the motion of fluids, and can be used to simulate the airflow around vehicles. The mathematical equations that govern the motion of a fluid are based on the principles of conservation of mass, momentum and energy. The majority of CFD software is based on the Navier-Stokes equations; these are a set of second-order non-linear partial differential equations, which (apart from certain specific situations) cannot be solved analytically. Therefore CFD techniques use approximations of the Navier-Stokes equations, which can be solved numerically[6].

These approximations use such techniques as Euler-method, Reynolds Averaged Navier-Stokes(RANS), Large Eddy Simulation (LES), and Direct Numerical Simulation (DNS).Numerical methods such as finite element, finite volume and finite difference are then used to solve these equations. As numerical solutions can only provide results at specific points, the model must first be divided into a number of discrete points or a grid, the approximation equations are then solved at each grid point. The final result is a series of numbers, which are usually displayed as a graphic for ease of understanding. The density of the grid size will have a direct bearing on the accuracy of the results as well as the computational time required to complete the simulation. CFD simulations are useful for predicating flow field trends in the early stages of vehicle development, as shape modification is possible without the need to build the physical model. CFD solutions seldom replace wind-tunnel testing as accurate quantitative data is usually not possible, i t can however ensure time in the wind-tunnel is used more efficiently. It should be noted that CFD is still a developing technology, and is undergoing rapid development.

Krusemann, R. and G. Schmidt [7] used the STAR-CD CFD code to simulate the airflow through the brake disc in order to optimize cooling. CFD analysis gave a new fundamental insight into the behavior of the airflow through the disc. The simulation of geometric modifications is also presented and provides a good estimation of disc cooling potential.

Kubota.M[8] performed parametric studies based on an analysis of airflow through discs. Brake cooling performance is determined by the airflow around the wheel unit and the airflow through the disc vanes. A one-tenth-scale model of a wheel unit incorporating a disc is tested in a water tank. Findings show the air velocity through the ventilation channels increased in proportion with the disc inner radius and alternately arranging fins of two different lengths increased the ventilation channel inlet, which in tum increased velocity. Velocity through the vanes increases with gourd-shaped vanes and short straight vanes. CFD results compared well with the water tank results. It was also found that increasing the number of fins achieves a more uniform stress distribution.

Mass flow rate

One on the most important factors to consider when developing a brake disc design for better internal cooling performance is the mass flow rate through the brake disc passages. When a brake disc is heated due to braking, it should cool down as fast as possible to avoid overheating. A higher mass flow rate through the rotor passage will in this case provide a higher cooling rate. A disc brake rotor could therefore be seen as a centrifugal impeller which main purpose is to translate the energy from the pump/rotor to the air being pumped by accelerating the fluid outwards the center of rotation.

Centrifugal impeller theory

Centrifugal impellers have been used in many different applications all the way back since early 1800's. The most famous application is the simple centrifugal pump. Other research reports that are dealing with development of ventilated brake disc design, are investigating the pumping performance with help from centrifugal impeller theory. A centrifugal impeller is, like a ventilated brake disc, a construction that imparts kinetic energy into the fluid. This translated energy will affect the fluid to flow. The basic theory of this flow phenomenon works basically as; the centrifugal impeller (or the ventilation channels) force the fluid out of the impeller by centrifugal force.

2.6 Friction Braking Development

The problem of safe vehicle braking became apparent at the beginning of self-propelled vehicle development[9]. Highlighted the story of 1769 when the Frenchman, Cugnot made the first self-propelled steam road vehicle. The vehicle had a top speed of about 5 km/h and demolished a wall because of braking system failure. The story goes that hot embers and boiling water spilled everywhere; Cugnot was arrested and sent to prison. Early combustion engine car brake systems were also inadequate, Benz pressed a small block against a drum and Daimler used spoon brakes acting on the rear tires. A reliable and contamination resistant brake system was not developed until 1903 when Daimler introduced the first internal drum brake. In the 1920s cars became much faster, the need for improved braking became apparent and vehicle braking systems evolved to hydraulic brakes on all four wheels. The first car of to use hydraulic brakes at all four wheels was the 1921 Duesenberg of the U.S[10]. The main disadvantage of the drum brake is its non-linear relationship between brake torque and pad/disc friction coefficient, which is exaggerated by the poor cooling characteristics of the design. Temperature rise causes drum expansion increasing pedal travel, soft pedal feel and improper contact between drum and brake shoe lining. The need for improved braking systems led to the development of the disc brake, which has a more linear relationship between brake torque and pad/disc friction coefficient and far superior cooling characteristics.

The first disc brake was patented in 1902 by F. Lanchester. However, interest in the design was not established until it was introduced at the UK 1951 International Motor Show and later when the Jaguar C-Type won the 1953 Le Mans fitted with Dunlop disc brakes. During the 1950s the disc brake became more commonplace on cars. Present day road vehicles (passenger cars and commercial vehicles) are fitted with disc brakes at the front wheel as standard and discs fitted to the rear wheels are increasingly more common.

George Westinghouse invented the first version of the railway air brake in 1869, acting on a tread (wheel) brake. The tread brake method has many disadvantages including limited energy capacity, high wheel tread wear and accelerated tread damage. Since then train speeds have increased dramatically and more advanced disc braking has been employed. Operation at 200 km/h has existed since the 1970s and in 1989 TGV (Train a Grande Vitesse) trains were running at 300 km/h. In 1990 TGV set a new world record travelling at 515.3 km/h. In order to provide the necessary braking energy capacity for the new generation of high-speed trains TGV incorporated four brake discs on each trailer axle and later wheel mounted discs on power cars (replacing tread brakes)[11]. The development of the motorcycle brake was much slower than that of the car. The first drum brake was introduced in 1935 by BMW on the BMW R 32. At the Tokyo Show of 1968 Honda unveiled the CB750F, fitted with the first disc brake.

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2.7Aerodynamics and Brake Cooling

The primary means of heat dissipation from a brake rotor is by convection. Increasing either the surface area of the rotor *(As)* or the heat transfer coefficient *(h)* will improve convection. In order to maximize cooling by convection, a sufficient supply of cooling air must interact with the heated elements. Clearly cooling can be improved by guiding air towards the appropriate parts, many examples of this can be seen in motor racing however this may have a negative effect on the vehicles drag. Consequently a compromise must be achieved between effective cooling and minimum drag. Generally this airflow is extremely unsteady and turbulent, and is influenced by the local geometry and vehicle velocity. The airflow is further affected by the pumping action of internally vented rotors. The nature of the flow in and around the brake rotor also means that a single Reynolds number cannot fully describe the flow in this region. The flow may be laminar on the face of the rotor and turbulent within and exiting the vanes.

The selection of wheel will also have an influence on the aerodynamics in the vicinity of the brake components, a more open wheel (large open area) will allow more cooling flow to reach the rotor. Fabijanic, J[12]showed the flow within the wheel arch to be very unsteady, and alternated from entering the wheel arch to exiting at the same position with no obvious periodicity. Most aerodynamic research in the area of brake cooling has focused on the airflow through internally ventilated rotors in isolation, and has not considered the effect on the flow under normal driving conditions.

The need for increased cooling of disc brakes led to the development of vented rotor discs. The primary advantage of vented rotors is increased heat dissipation from internal pumping of air, however under slow speeds the pumping action of the vanes is minimal and only becomes significant as rotor speed increases. At higher speeds the airflow flowing around the disc as a result of the forward movement of the vehicle, tends to prevent effective pumping of air through the vanes.

Early work by Limpert[4] indicated that the heat dissipation from internal ventilation amounted to about one third of the total heat dissipation from the rotor, but suggested it was the larger surface area and not the pumping action that made the major contribution to cooling. While vented rotors do provide a larger surface area for heat dissipation, and extra airflow from the pumping effects of the vanes, it must be remembered that they will usually have a reduced mass than their equivalent solid rotor and therefore have less capacity to store thermal energy.

One approach has been the use of curved vanes instead of straight vanes ones. Curved vanes can create additional cooling in two ways; firstly an increase in surface area of up to 30% can be achieved, and secondly optimization of the vanes can improve the pumping efficiency. However curved vane rotors add to manufacturing complexity and introduce the need for different rotors for each side of the car.

Many attempts have been made to improve the cooling ability of straight vane ventilated rotors. Zhang [13] proposed a redesign of vented rotors to include an optimized number of flow passages, improved rounding on inlet vanes, and a long-short alternative vane configuration. This design contains twice the number of outlet vanes as inlet vanes, in order to reduced inlet blockages and guide to flow though the exit more easily. The configuration was modeled on CFD software and a 42% increase in flow through the vanes is claimed, however no experimental verification is given.

A similar technique was used by Daudi[14] to develop a rotor with three times the number of outlet vanes as inlet vanes, providing 35% more flow through the vanes when tested on a model in still air. Hudson and Ruhl[15] looked at ways of increasing the centrifugal pumping of existing vented rotors by using traditional turbo-machinery techniques to improve flow condition at the inlet to the vanes. Airflow from the modified rotor were measured at the outlet of the vanes using an adapted three hole pitot static tube at the outlet of the vanes. The modifications were designed to reduce flow separations and decrease losses while keeping the symmetry of existing vented rotors.

Experimental on-vehicle testing of the modified rotor resulted in temperatures of between 3-6% lower than the production rotor at the end of a ten stop fade test. Daudi[16] also concluded from CFD modeling that increased airflow through a vented rotor could be achieved by allowing air to enter the vents from both the inboard and outboard side of the rotor.

Another development has been the cross-drilled or axial cooled rotor; this increases the surface area presented to the atmosphere and can potentially improve cooling, however the rotor mass and friction area are reduced. Analytical work by Limpert[17] showed that the addition of axial cooling passages reduced cooling effectiveness; this is mainly attributed to the increased heat flux. Visualization of the flow through ventilated rotors by Kubota[18] using scale models submerged in water, indicated significant regions of separated flow within the vanes of the rotor, which would indicate that increased flow could be achieved with optimized designs. Measurement of the airflow through the vanes of a vented rotor is difficult while on the vehicle, so it is usually done in isolation.

$$V_{ave} = \omega D_O \sqrt{-0.0201 + (0.2769 \times D_I) - (0.0188 \times D_I^2)} \tag{2.2}$$

Where

V_{ave} = average velocity of airflow through vanes (m/sec)

ω = angular velocity (rads/s)

D_O = outer diameter of rotor (m)

D_I = inner diameter of rotor (m)

Similar work by Limpert[17]also produced an expression for the average velocity through a rotor:

$$V_{ave} = \frac{V_{in}}{2}\left(1 + \frac{A_{in}}{A_{out}}\right), \qquad V_{in} = 0.052N(D_O^2 - D_I^2)^{1/2} \tag{2.3}$$

Where

N = revs per minute of rotor

V_{in} = air velocity at inlet (m/sec)

A_{in} = vent area inlet (m^2)

A_{out} = vent area outlet (m^2)

It should be noted that neither equation caters for the number of vanes in the rotor, as both authors found that the number of vanes did not affect the amount of air pumped, provided that the flow areas were the same.

However Zhang [20] concluded that a square cross section provides the lowest wetted perimeter and therefore lowest viscous friction losses, as it has the shortest perimeter than any other rectangular shape. Thus the optimum number of vanes could be calculated using the formula:

$$Z = \frac{\pi D}{w + t} \tag{2.4}$$

Where: Z = number of vanes,

D = the mid diameter of the rotor,

w= the flow passage width,

t= the thickness of the vanes.

Hudson and Ruhl[15] used turbo machinery techniques to improve the airflow through a typical ventilated rotor, and compared experimental findings with values expected from both Limpert and Sisson's equation. They suggested three improvements to the existing brake rotor including a rounded inlet chamfer, a rounded inlet horn and a pre impeller, all of which reduces the number of sharp edges the flow interacts with.

The results of these experiments are compared to results obtained analytically from both Sisson's and Limpert's equations. Sisson's relationship yields values of average velocity higher than Limpert's, although both appear to give reasonable correlation with the average velocity recorded by Hudson and Ruhl. The pumping power is proportional to the component of the absolute velocity, which is the velocity of whirl in the direction tangential to the vane circumference.

Another approach to the problem is to use a vane/pillar combination that pumps air onto pillars, the pillars reduce pumping power but still provide good cooling by increasing the surface area. Other manufactures use somewhat different designs based on the same principle (avoiding vanes). To aid the design of large high speed ventilated discs a method to compare the efficiency of ventilation designs with regards to thermal and pumping efficiency an efficiency ratio has been developed. The ventilation system of a ventilated disc affects only the convective component of thermal power loss (radiation is enclosed within the vane channel and conduction is minimal).

The ventilation efficiency ratio *'1*$_v$has been defined as the ratio of convective power dissipation (Q_{conv})and pumping power (Q_{pump}),by the following equation:

$$\eta_v = \frac{Q_{conv}}{Q_{pump}} \tag{2.5}$$

The ratio *'lv* increases with increase in convective thermal power dissipation and reduces with increase in pumping power. That is to say, the higher the ratio the more energy efficient the ventilation system is in dissipating heat. The thermal and pumping power are functions rotational speed, the thermal power dissipation is also a function of disc surface temperature. The ratio *'1v* for a particular ventilated disc is thus a function of rotational speed and temperature.

2.8 Disc/Pad Interface Energy Distribution

Most friction brake applications are performed in the thermo elastic instability regime, which causes variation of interface pressure and heat generation over the nominal disc/pad contact area. Irregularities in the surface will cause the pressure distribution to be non-uniform. Barber[21]showed experimentally the formation and movement of these areas are determined by the thermo-mechanical process of thermal deformation and wear. The highest part of the surface will carry the greatest pressure, reach the highest temperature and consequently expand more than the surrounding surface. Thus the thermal expansion tends to exaggerate the initial irregularity of the surface. The wear at the interface has an opposite effect, but under suitable conditions the process can be unstable.

The higher temperatures in these areas cause the material to expand and rise above the level of the surrounding surface, which increase pressure and thermal loading. High thermal stresses produced in these areas can cause surface cracking. Work on various brake types has shown that hot spots are a very common occurrence in disc brakes.

Fec and Sehitoglu[22] examined the thermal-mechanical damage in train wheels due to hot spotting; constraints and thermal expansion cause stresses that eventually initiate and propagate fatigue cracks. Anderson and Knapp[23]studied various types of hot spots in detail and described different types of hot spots found in automotive brakes. It is suggested that the most critical operating conditions involve high sliding speeds, low bulk temperatures, long friction contacts, thick materials and low material wear rates.

In the study of brake interface pressure distributions by Tirovic and Da[24] the effects on brake performance are discussed describing how the pressure and contact at the brake friction interface can be broken down into three levels. The first relates to large scale pressure variation over the full rubbing surface, induced by bulk deformations and application of actuating forces. The second relates to 'macroscopic' interface pressure variation, arising from localized deformations or distortion of the rubbing surfaces.

High peaks of interface pressure cause high rates of localized frictional heat flux generation, which initiate heat spotting, banding and ultimate cracking, crazing and failure. The third level represents frictional contact on the microscopic scale, fundamental to the study of friction and wear.

Lee and Barber[25] have investigated TEl experimentally showing that the onset of instability is clearly identifiable through the observations of non-uniformities in temperatures measured using imbedded thermocouples. Hot spots from TEl are usually anti-symmetric Hartsock, Hecht et[26]and will not occur below a critical speed. This speed is dependent on the disc and pad material properties, pad/disc relative size, pad thickness, disc thickness, and the coefficient of friction. Results show that critical speed is a weak function of the number of hot spots per revolution, which means the number of hot spots obtained in a practical brake system will be sensitive to initial conditions.

Kao, Richmond et al[3], has demonstrated experimentally and theoretically that in addition to the thermal growth the thermoelastic unstable warping or buckling induces excitation to hot spotting and disc cracking, as well as torque variation and hot judder. The non-uniform heat distribution caused by hot spots result in higher expected surface temperatures effecting convective and radiative heat dissipation.

The problems of adequate brake cooling are associated with all brake types and heat dissipation has attracted much investigation and research. The experimental and theoretical results were limited with the tools available at the time. Most of the recent published work has been fragmented and the efforts concentrated on the specific problem, design or heat transfer mode. Conduction is the least studied mode of heat dissipation from brakes. From published research on TCR at the interface of two mating components, the effect of interface condition has been shown but not directly related to brake components. Brake interface conditions vary due to corrosion and dismantling and re-assembly procedures. Different material combinations also complicate the issue. Published research, dealing with brake thermal modeling, takes a simplified approach to conductive heat dissipation, applying general heat transfer coefficients to brake areas conducting heat to adjacent parts.

A number of published papers deal with convective cooling from specific disc areas. Research has also been focused on the heat transfer in the channels of ventilated discs. Comparison of the suggested approaches show that although similar heat transfer coefficients can be obtained in some cases, in other cases differences can be substantial. More recent approaches have been published using CFD methods to predict airflow around the vehicle's front ventilated disc brakes. Most of the published work relates to automotive applications. Available data is insufficient to offer a reliable guide regarding the heat transfer coefficients that should be used for railway brake thermal analyses. Limited published data shows that ventilated railway discs can cause substantial aerodynamic losses, and a compromise with cooling must be achieved.

Radiative cooling has often been neglected in published brake thermal analyses, even though at higher temperatures radiation is a very significant mode of heat dissipation. It is interesting to note the substantial differences of published cast iron disc surface emissivity; values range from 0.15 to 0.9. Again, available data is very limited and insufficient for adequate modeling of radiative heat losses. It is particularly important to accurately model this mode of heat dissipation, because of its significant influence at the high temperatures reached by new disc and pad materials and high performance designs.

T.P. Newcomb has made research, both theoretical and experimental, into the behavior of discrete brake components. He published four papers[27, 28, 29, and 30] between 1958 and 1960, all concerned with the development of transient temperatures and employing Laplace transformation techniques in producing analytical solutions to various simplified problems. He produced solutions for drums, discs and dry friction clutch transmissions operating under various conditions and also suggested methods for sizing components, a vital step in producing research of significant value to the brake design engineer.

Although the solutions contained, were often based on infinite or semi-infinite slab theory the proposed results were important in the development of analytical theory in the field and have been repeatedly referenced since. In 1965,Ferodo, Newcomb and Millner[31] experimented on disc and drum cooling.

Drag tests were performed on vehicles and then cooling rates of the components were obtained with thermocouples. They produced an empirically derived formula relating cooling to vehicle speed or different sizes of drum/discs. However this equation could not account for the differences in cooling air flow over the discs caused by different vehicle body designs. Ventilated discs were compared to solid. It was estimated that cooling rates for front brakes are 20% higher than for rear brakes and that disc brakes cool 25% faster than drums of an equivalent capacity.

Abbas, Cubitt and Hooke[32] studied the distortion effect of coning and using a computer program, produced a method of calculating stresses in a solid brake disc from an initial known temperature distribution. This method, based on finite numerical methods, could be used to produce a disc less susceptible to this form of deformation by careful design of its section. Abbas[33] followed this with a further paper, relating experimental work to confirm the previous theoretical research, using a static test rig to heat the disc and then allow it to cool, whilst thermocouples and strain gauges recorded the data. Good agreement was found.

El-Sherbiny and Newcomb[34] derived a numerical method in 1976, following Abbas'work using finite difference techniques to analyse temperature distributions in dry clutches under a variety of different conditions.

Numerical methods were being used because the increased complexity of problems and original assumptions that had now been found to be unacceptable could not be contained only with the analytical solution. Joining Ashworth, El-Sherbiny and Newcomb[35] proceeded to apply their method to brake drums. In both papers such complex situations as incomplete contact between the friction pair were examined, even though they only had access to limited computing power and a relatively simple Finite Element method. Distortion was able to be modeled for a variety of contact cases and modifications to the drum made for improved cooling and therefore thermal distortion.

Inoue from the Toyota Motor Corporation published an important paper [36] combining disc rotor transient thermal measurements and finite element work. He started by devising a system for measurement of ventilated disc temperatures during a stop, recording them for later display and analysis. Several interesting phenomena were documented. Firstly after a single stop on a dynamometer some plastic deformation was found to occur in the form of a double wave around the circumference of the brake disc. A corresponding torque fluctuation was measured. In addition 'hot spots' were encountered on the braking faces of the disc.

Hot rings had previously been observed but when the ability to view the temperatures frozen on the disc during part of a braking application was available then localized high temperatures were found. These hot spots did not remain stationary and grew and disappeared at apparently arbitrary positions, although in a cyclic manner, over a period of many brake applications. The measurements did show that not only were the temperatures not uniform in a circumferential direction around the disc but they were also uneven in a radial direction over the disc surface. The first assumption here was universal among previous papers predicting disc behavior and it was usually assumed that the contact pressure had no initial variation with radial position. Inoue then went on to create a finite element model, subjecting it to the results of the dynamometer tests as thermal input. Again a waveform deformation was observed in the results of the simulation. In addition Inoue related initial disc run out to peaks of temperature on the disc braking face. The paper concluded by proposing a new form of disc, the 'inside-out' configuration, specially designed to combat coning and reduce brake judder. Its ability was demonstrated in further computer simulations.

2.9 Manufacturing of brake disc

In modern days, the use of metal is vast and there are various methods of manufacturing a product from only use of pure molten metal or from any other state of metal as well. When considering the different methods of manufacturing, most popular methods used in large industries are casting, metal cutting, forming and shaping, fabrication and welding. The above mentioned are few that are used by industries to produce different products that could make up a machine and electronic components or other day to day tools. The disc brake system is currently used in automotive industries, locomotives and in jumbo jets as well. The disc brake system is an assembly product and each parts are manufactured separately with different procedures to one another. When the disc ring (brake disc) has a perfect circular shape, most of the time these brake discs (disc ring) are made of cast iron, which has good anti wear properties and it is cheap as well. But in certain other cases such as high performance vehicles, these brake discs are not up to their standards of high performance because the cast iron brake discs are heavy in weight and so reduces the vehicle's performance to a particular extent. In case such as this, ceramic composite brake discs are used, that are processed and used at high temperatures. These ceramic composite brake discs are known to be heat resistant and able to withstand large compressive loads at higher temperatures.

In earlier days, these brake discs were made straight from the molten brittle ceramic materials but researchers have found that short carbon fibers would be a solution for the brittleness of ceramic materials and hence for the manufacturing of the discs, materials are short carbon fibers, carbon powder, heat molded resin used. When the brake disc shape is obtained by heating the mixture of above materials cooling down, another ceramic material known as silicon is added to harden the brake disc, forming a new material called silicon carbide.

The heat molded resin is a material that binds all other materials together in that mixture of the brake disc, and once this material is hardened by molding, it can't be softened by any process. The equipment and tools used in this process will be talked more about in the manufacturing details section of this report but some of the main tools used in as follows:

- Furnaces and a press
- Permanent molds
- Crucibles
- Drilling machines
- Computer guided machines

The manufacturing method used in production of brake discs is the metal casting process and to be more specific, it is the permanent mold casting process that takes place in the production of these brake discs that usually gives out a good surface finish for the end product. To begin this process short carbon fibers, carbon powder and heat molded resin must be mixed together. Then using an automated machine this mixture is poured into a permanent aluminum mold cavity which is in shape of a brake disc (disc ring), until it is half full. Once it is half full, the mold is removed and workers have to insert aluminum cores into a belt with gaps around the mold that allows the cores to be inserted into the mold.

These cores will form a ventilation cavity in the disc ring (brake disc) to prevent the disc from overheating. The mold again moves back into the automated machine to fill the other half of the mold cavity with the rest of the mixture that was poured into first half of the cavity. Once the cavity is full it is leveled using a roller and then using the cover or the other half of the permanent mold, it is covered and is pressed lightly to compact the contents inside. Then the fully covered mold is sent to a large press which applies 20 tons of pressure and heating to almost of 400^{o}F. This heat and pressure compact the carbon fiber and resin into plastic and makes it stronger.

Once the mold is cooled down to be handled, submerge it in cold water for 5-8 minutes which cools the disc ring completely, enabling them to pull out the cores that were inserted for the ventilation purposes. After all the cores have been removed, remove the cover of the mold and pull out the disc ring from the mold. Then using the computer guided machines smooth out all the rough edges on the disc ring and drill tiny ventilation holes. They then put the disc ring into an oven and on over two days it gradually heats it to 1800^{o}F. This would then cause the chemical change which transform plastic into carbon. Then take a crucible which is a high heat resistant container and place five mounts inside so that it can hold the disc ring on them without having the disc ring to touch the base of the crucible. Once the disc is mounted on the crucible, place a funnel at the center of the disc ring and fill it with a fine silicon powder.

Then they load the crucible into a furnace for 24hours and allows it to gradually heat the disc ring to temperature of 3000^{o}F until the silicon is melted completely. This liquid silicon is then drawn into the disc ring by the pores of the framework of the disc ring and forms a completely new material called silicon carbide which makes the disc ring exceptionally hard. After it is removed from the furnace, a drill machine bores the mounting holes on the disc ring. And then the disc ring goes to a chamber to receive a coat of protective paint. This paint is used to shield the carbon and disc ring from oxygen and this process is very critical since at high temperatures, oxygen burns carbon. Hence this anti oxidation process increases the lifetime of the disc ring. After the protective player is applied, with the help of a computer guided robot arm, moves the disc and polishes the entire disc surface. After all the polishing has been completed, a computer guided machine thoroughly inspects the disc ring surface by taking high definition photographs to further examine the molecular and crystal structures to detect any defects.

CHAPTER THREE

ANALYSIS AND MODELING

3.1 Slopes or gradient for selected Addis Ababa stations

The braking force needed is dependant on the track gradient. In this research the down hill track gradiant is used for investigating the limits of the design parameters for the brake disc.The following calculation shows the gradient of the rail using profile of the Addis A baba routeline obtained from ERC, the station rail gradient will be calculated with the sa me procedure listedbelow.

$$\text{Slope}=\frac{\text{Elevation difference}}{\text{Horizontal distance}} \tag{3.1}$$

According to Addis Ababa light rail transit (AALRT) office the Maximum gradient should be ≤ 55 0/00 which is 0.055; slope=tan-1 0.055=3.15 °. This paper will be considering this maximum gradient 3.153 ° as the other values are less than 3.15 °.

No	Station name	Gradient/slope	Station type
1	Ayat	0.0333 [1.905^0]	Ground station [-down]
2	CMC	0.0483 [2.767^0]	Ground station [+ up]
3	St.Michael church	0.0185 [1.058^0]	Ground
4	Civil service college	0.0178 [1.019^0]	Ground station
5	Megenagna	0.018 [1.031^0]	Under Ground station
6	Mexico square	0.0278 [1.591^0]	Elevation station
7	Coca cola	0.0392 [1.302^0]	Ground station
8	Urael church	0.0243[1.394^0]	Ground station

Table3.1Slope [Source: Addis Ababa (E-W& N-S) rout light rail transit project from ERC,September 2009).

3.2 Load Determination

Braking from the maximum velocity of 70km/h to a standstill with the temperature of thesurrounding area constant at 25 °C,

Main technical parameters

Average brake deceleration (maximum vehicle operation speed: 70km/h) of vehicles frommaximum operation speed to stop on straight and dry track under seating capacity.

Average deceleration for service brake: ≥ 1.1 m/s^2

Average deceleration for emergency brake: ≥ 2.0 m/s^2

Loads	Car body weight	Passenger weig	Total weight
Empty vehicle (t)	44	0	44
Seating capacity	44	15.24	59.24
Overload capacit	44	19.02	63.02
Axle load	≤11（1+3%）t		

Table 3.2 Number of passengers and weight of vehicles.

Note: Take 60 kg as average weight of each passenger; tare weight of vehicles ≤44t.

Angular velocity

The railway vehicle running with initial velocity (v_o) is supposed to stand still with co nstantdeceleration (a). Its linear translational velocity as function of time (t) is given by:

$$vo = vo - at \tag{3.2}$$

The angular velocity (ω) of the wheel set can be determined using:

$$\omega\,(t) = \frac{v(t)}{r_w} \tag{3.3}$$

at v_o, ω_o=58.922 rad/s

Where,

v = velocity of the vehicle

v_o= Initial running velocity of the vehicle

ω_o= Initial angular velocity the wheel

ω = angular velocity of the wheel at any time

r_w = the radius of vehicle wheel

During the application of brake, the brake force acts at the effective radius (r_{disc}) of the disc rotor.The translational velocity (v_{disc}) of this frictional force (F_{disc}) could be expressed inte rms of thetranslational velocity *v(t)* of the vehicle as follows.

The total braking time and distance can also be calculated by the formula:

$$S_b = v(t) - \frac{1}{2}at^2 \tag{3.4}$$

Hence, angular velocity (ω) at any time (t) could also be determined from:

$$T_B = \frac{v_o}{a}, \tag{3.5}$$

$$\omega = \omega o\left(1 - \frac{t}{t_b}\right) \tag{3.6}$$

Where:

t_b = brake time to stop the vehicle

S_b = total brake distance

Braking Energy of the Vehicle

During braking process, available energy of the vehicle that is transformed into frictional heat, need to be determined accurately. Kinetic as well as the potential (while running on slopes) energy of the locomotive that is dissipated by the work of the braking. Rail vehicles have important masses in rotation. Therefore, the contribution of rotational kinetic energy is taken in to account.

The initial kinetic energy imposed in to the locomotive is given by the sum total oftranslat ionaland rotational.

$$E_k = \frac{1}{2}Mv_o^2 + \frac{1}{2} I\, \omega_o^2 \tag{3.7}$$

Where:-

E_k= Initial Kinetic Energy of the locomotive just before braking starts

M = Mass of the locomotive

I = Polar inertial moment of rotating parts

And, the moment of inertia of the rotating wheel set and discs can be calculated by using theequation below.

$$I = \sum mr^2 \quad (3.8)$$

Where:-m_i=masses of the wheel set and brake discs

r_i= rotation radius of the rotating parts from the center of rotation

The contribution of the rotating masses of the wheel set and discs is taken to be 10% the tareweight of the axle load of the locomotive. Therefore the kinetic energy of the vehicle would be;

$$Ek = \frac{1}{2}(1.1)Mv_o{}^2 \quad (3.9)$$

The potential energy of the locomotive depends on the track gradient δ [mm/m]and o n thetravelled distance S_b[m].The definition of gradient being $i=\tan\alpha$,the potential energy of the vehicle is

$$E_P = Mg\sin\alpha \quad (3.10)$$

Where, α is angle of inclination, g = 9.81m/s^2

The total available braking energy of the locomotive is the sum of the kinetic, rotationaland potential energyfor emergency brake at the gradient.

$$E_b = \frac{1}{2}(1.1)Mvo^2 + Mg\sin\alpha \quad (3.11)$$

E_b =13,132,834.93J

The following parameters are used for calculating heatflux.

Item	Values
Mass of the vehicle – *M* [kg]	63020
Start velocity – v_0[Km/h] or [m/s]	70 or 19.44
Deceleration – m/s^2	1.1 for service brake
	2 for emergency brake
Braking time – t_b[s],	17.67 for service brake
	9.72 for emergency brak
Effective radius of the braking disc – r_d[m]	0.243
Radius of the wheel – r_w[m]	0.33
Incline of the track – δ [‰]	55
Friction coefficient disc/pad – μ [/]	0.36
Surface of the braking pad A_p– [m]	0.02
Thickness of the disc Th-[mm]	60
Outer radius of the disc [mm]	230
Inner radius of the disc [mm]	110
Surface area of the braking disc A_d– [m]	0.2057
Pad inner and outer radius [mm]	120 & 240

Table 3.3 Data for calculating the heat flux

The total Braking energy is distributed on the number of the discs of the axles. Since the railvehicle has 8 disc brakes. Heat flux and force of the disc is calculated from thetable above.Force on the brake disc is calculated as;

$$F_{disc} = \frac{\left[\frac{1}{2}(1.1)Mv_o{}^2+Mgsin\alpha\right]/8}{2({}^{rd}/{}_{rw})[v(t)-\frac{1}{2}at^2]} \tag{3.12}$$

F_{disc}= 16,856.69N

The thermal conductivity of material of the brake pads is smaller than that of disc (*kp*<*kd*). We consider the energy from brake application is converted into heat and transferred to the disc and pad approximately 90% and 10% respectively. This ratio is called theproportion of heat transferred to disc (γ = 0.90). The rate of heat generation is:

$$Q_{(t)}= \gamma 2(F_{disc})v_{disc}=0.9F_{disc}\,\frac{r_d}{r_w}(\text{vo}-\text{at}) \tag{3.13}$$

Convective heat transfer between the air and the brake rotor surfaces is the primary mean o f heatrejection. When locomotive begins to brake, the train lost power and will stop by frictional forces. The imposed mechanical energy theoretically transformed to frictional heat. The brake disc rotor has got two friction surfaces swept by the brake pads on both sides. Two friction forces act on both faces of the disc at an effective radius (r_{disc}). These forces retard the movement of the locomotive.

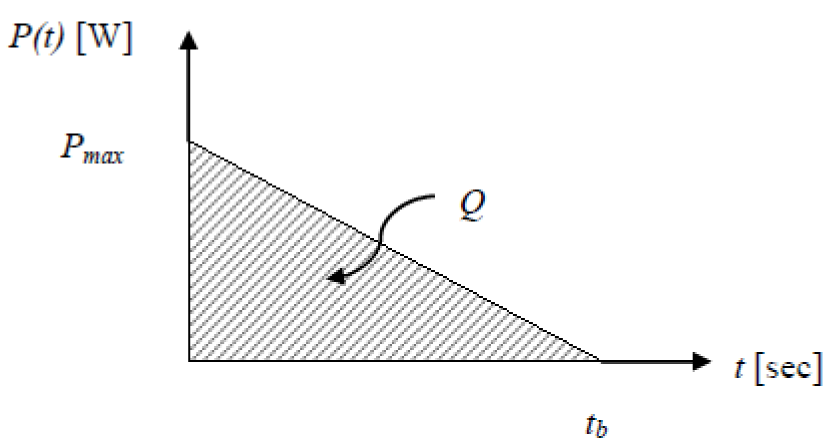


Figure.3.1 Instant heats flux entering on the side of the braking disc:

The distribution of heat flux with time is

$$Q_{(t)} = \gamma 2(F_{disc})v_{disc} = 0.9F_{disc}\frac{r_d}{r_w}(\text{vo} - \text{at}) \quad (3.14)$$

$$= 2730340 - 280836.25t$$

Braking distance

A lower braking time and distance can be found appliying a greater force and heat flux on the brake rotor. To lower the braking distance we need to increase the heat flux applied on the brake disc which gets the disc temperature higher. Inorder to compare results of different brake models in terms of brake distance and braking time, the heat flux equation(3.14) is solved for different time values. Now simulating with ANSYS each values calculated we can find the braking time at specific temperature value selected for comparison. In this case the maximum temperature of curved vanes disc rotor is used so that we can observe how much shorter the train stops for the pin vented rotor with similar disc temperature.

Time[s]	Braking force(N)	Heat flux[W/m^2]
9.72	16856	2.730e+006
9	16957	2.803e+006
8	17409	2.872e+006
7	18297	3.009e+006
6	19758	3.233e+006

Table 3.4heat flux, braking force and time

Pressure Determination on the Disc

The surface pressure between the disc and pad, on behave of the calculated force applied to thedisc, needs to be determined. Hence, brake caliper pressure is:

$$P_{pad} = \frac{F_{disc}}{2\mu Ap} = 1939942 \quad (3.15)$$

$$P_{rotor} = \frac{F_{disc}}{2\mu Ar} = 188618.64 \quad (3.16)$$

Where: P_{pad} is pressure of pad,

Ap is surface area of pad

P_{rotor} is pressure of disc rotor

Ar is surface area of rotor

μ is friction coefficient disc/pad

Centrifugal Load Determination

Due to the rotation of the wheels the brake experiences additional type of force, in the centrifugal direction. It is calculated by

$$F_C = M_d \omega^2 \mathrm{R} \quad (3.17)$$

Where; F_C– Centrifugal force

M_d-Mass of the brake disc

ω- Angular velocity

R-Radius of the brake disc

The angular velocity at any time t is:

$$\omega = \omega(1- \frac{T}{T_b}) \quad (3.18)$$

$\omega = (67.33 - 4.028)^2$

M_d= 33kg

ω =67.33

R = 170mm or 0.17m

$F_C = 0.17x33(67.33 - 4.028)^2$

= 25,431.9 N

Finite element formulation for heat conduction

The unsteady heat conduction equation of each body for an axisymmetric problem described in the cylindrical coordinate system is given as follows:

$$\varrho c \frac{\partial T}{\partial t} = \frac{1}{r} \partial\partial r \left(r k_r \frac{\partial T}{\partial r} \right) + \frac{\partial}{\partial z} \left(k_z \frac{\partial T}{\partial z} \right) \tag{3.18}$$

with the boundary conditions and initial condition

$T = T^*$ on $\Gamma 0$, $qn = h(T - T\infty)$ on $\Gamma 1$, $qn = q^*$, n on $\Gamma 2$, $T = T0$ at time = 0,

Where ϱ, c, kr, and kz are the density, specific heat and thermal conductivities in the r and z directions of the material, respectively. Also, T ∗is the prescribed temperature, h is the heat transfer coefficient, q∗, n is the heat flux at each contact interface caused by friction, T∞ is the ambient temperature, T0 is the initial temperature and Γ0, Γ1 and Γ2 are the boundaries on which temperature, convection and heat flux are imposed, respectively. Using the Galerkin's approach, a finite element formulation of unsteady heat equation(3.18)can be written in the following matrix form as

$CT\ \dot{T} + KHTT = R$ (3.19)

where CT is the capacity matrix, KHT is the conductivity matrix, T and R are the nodal temperature and heat source vector, respectively. The most commonly used method for solving Eq(3.19) is the direct integration method based on the assumption that the temperature Tt at time t and the temperature Tt+Δt at time t+ΔT have the following relation:

$$Tt{+}\Delta t = Tt + [(1 - \beta)\ \dot{T}t + \beta\ \dot{T}t{+}\Delta t]\Delta t \tag{3.20}$$

Eq. (3.20) can be used to reduce the ordinary differential equation (7) to the following implicit algebraic equation:

$(CT + b1KHT)Tt{+}\Delta t = (CT - b2KHT)Tt + b2Rt + b1Rt{+}\Delta t$ (3.21)

where the variable b1 and b2 are given by

$b1 = \beta\Delta t$, $b2 = (1 - \beta)\Delta t$ (3.22

Heat Transfer Coefficient

The heat transfer coefficient depends on the surface condition and texture of the body over which a fluid flows. As the air approaches the front side of the disc, air pressure rises from the free stream value to the stagnation point value. The high pressure forces the air to move along the rotor surface and boundary layers develop on both sides. The pressure force is counteracted and the air separates from both sides of the rotor and forms two shear layers. The innermost part of the shear layers are in contact with the rotor surface and moves slower than the outermost part. As a result, the shear layers roll up. The flow pattern is dependent on the Reynolds number. Analytical estimation of fluid flow and heat transfer parameters doesn't give accurate results as transient analysis. The results at different points of the fluid domain are different.

. $\nu = \mu / \rho$ (3.23)

Where:

ν = kinematic viscosity (m2/s)

μ = absolute or dynamic viscosity (N s/m2)

ρ = density (kg/m3)

The characteristic length for a cylinder or sphere is the external diameter, D, The Reynolds number is calculated as:

$$V_{ave} = \omega D_O \sqrt{-0.0201 + (0.2769 \times D_I) - (0.0188 \times D_I^2)} \quad (3.24)$$

= 10.8 m/s

Re= (VxDo)/v (3.25)

= 5.703 x 105

Where: Re, Reynolds number

V_{av}, rotor velocity

Do, outer diameter of the rotor

v, Kinematic viscosity of air

Now, for Re ≥ 2.4x105 a turbulent fluid flow equation is applicable for the estimation of heat transfer coefficient(h). The heat transfer coefficient associated with the flow characteristics for ventilated brake discs the heat transfer coefficient for ventilated brake discs is calculated below.

Prandtl number is given as

Pr = cμ /K

= 0.707(3.26)

where ;Pr is the Prandtl number

c is the specific heat capacity

K is the thermal conductivity of surrounding air.

The following correlations proposed by Churchill and Bernstein[37] Average Nusselt number, Nu, for cross flow forced convection of air over a cylinderis calculated by

$$Nu = 0.3+\frac{0.62*Re^{\frac{1}{2}}Pr^{\frac{1}{3}}}{[1+\left(\frac{0.4}{Pr}\right)^{\frac{2}{3}}]^{\frac{1}{4}}}[1+\frac{Re}{282000}^{\frac{5}{8}}]^{\frac{4}{5}}(3.27) \qquad (3.28)$$

= 801.9

h = K Nu /r, (3.29)

= 90.65 W/(m^2K)

3.3 Modeling of the Brake Disc

3.31 CATIA V5 part Modeling

The modeling of the disc geometry is done with CATIA V5. The CATIA V5 is a three dimensional solid modeler with both part and assembly modeling capabilities. CATIA V5 is used to model piece parts and then combine them into more complex assembly. With CATIA V5 a part is designed by sketching its components shapes and defining their size shape and interrelationships. Since CATIA V5 has parametric features, you can change one feature and all related features are automatically updated to reflect the change and its effects throughout the part.

It can be used to create angular shaped part, to which 3D surface can be applied to create hybrid parts consisting of mixture of angular and curved shapes. This provides the ability to create model designs with shapes of varying types. Assemblies can be created from parts combined individually or grouped in subassemblies. CATIA V5 builds these individual parts and subassemblies into an assembly in a hierarchical manner according to relationships defined constrains. As in part modeling, the parametric relationship allows you to quickly update an entire assembly based on a change in one of its parts. And save the model as IGS profile for importing to the ANSYS work bench for finite element analysis.

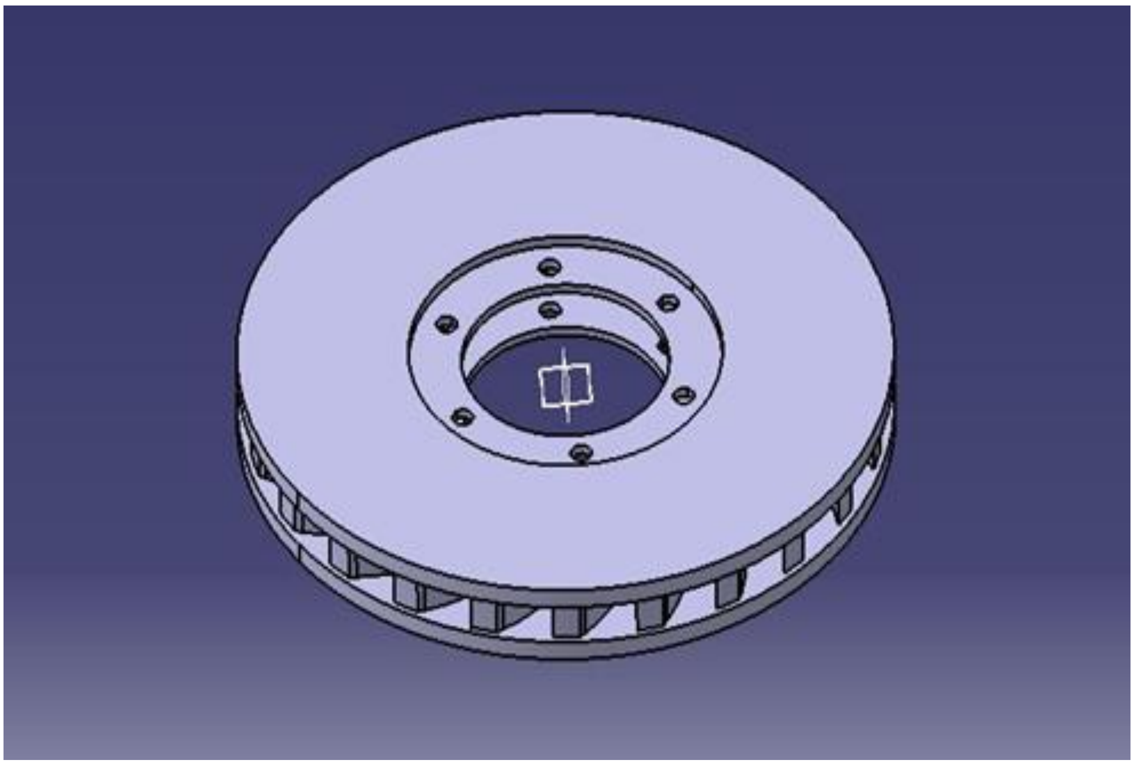


Figure. 3.2CATIA V5 brake disc model

The wire framed geometry of a curved vane disc brake is shown below to show the internal shape and arrangements of the vanes. Curved vane disc brakes have higher heat dessipating characteristics as it enhances the out flow of the air inside and acts as a centrifugal pump.

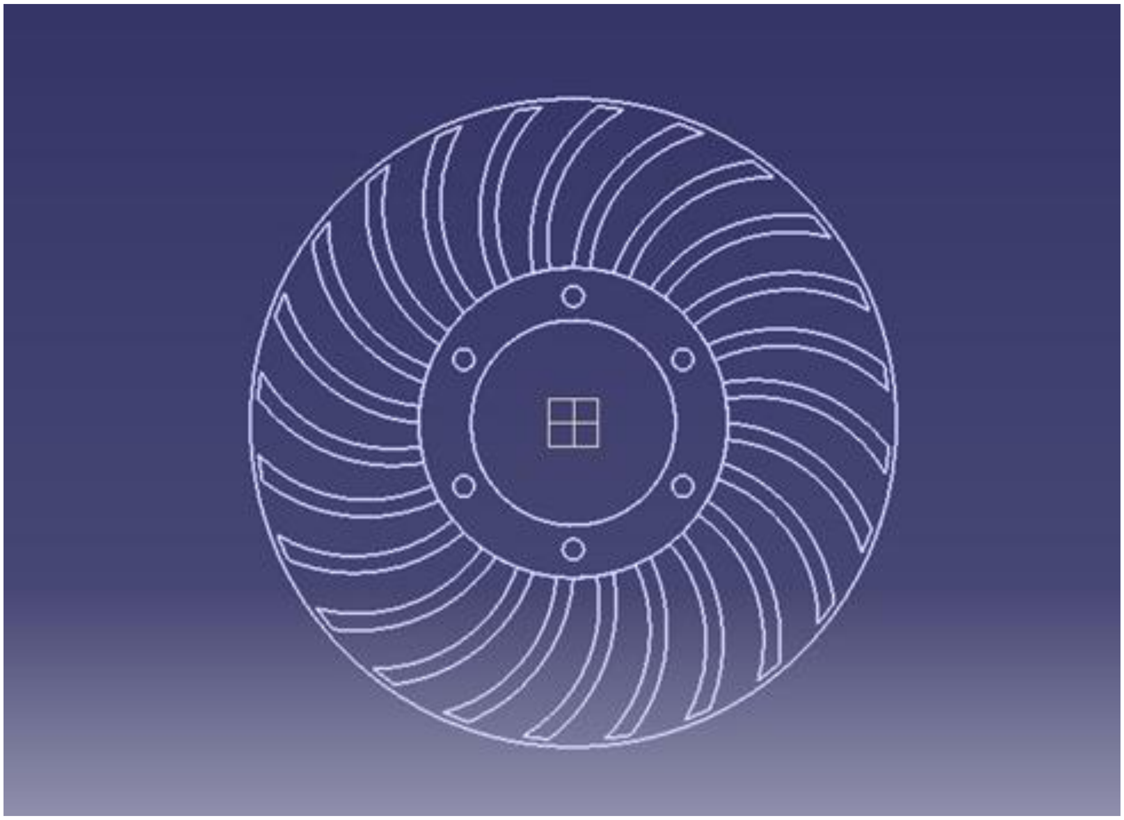
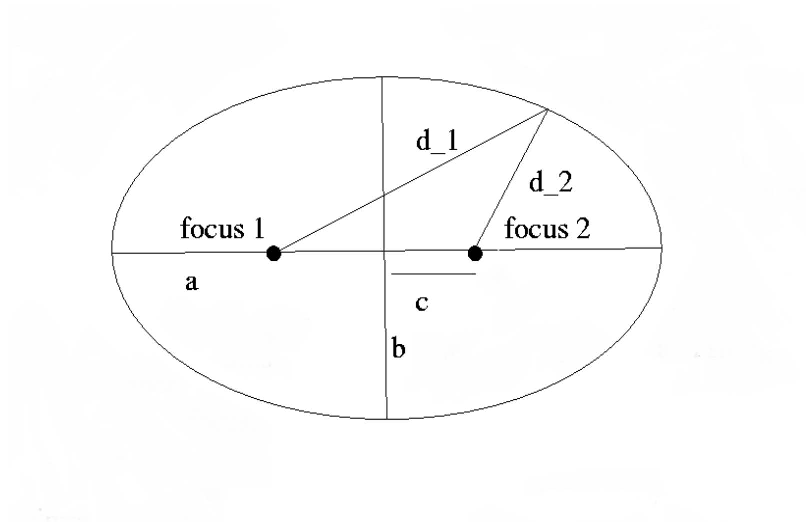


Figure.3.3 Curved vane disc rotor wireframe view

The wire framed geometry of two types of pin vented disc brakes is shown below. The cooling pins have an elliptic cross section of varying geometries. The elliptic pin section of the geometry shown in figure(3.4) has an eccentricity of 0.74 with minor diameter of 20mm and major diameter of 30mm while the one shown in figure(3.5) has eccentricity of 0.9 with 15mm minor diameter and 35mm major diameter. Lower eccentricity value represents an ellipse that resembles a circle. A circle is an ellipse in which the eccentricity is zero. Increased eccentricity of an ellipse with the same area means a greater surface area to volume ratio of the pins which adds to the convection heat transfer coefficient. Eccentricity of the ellipses geometry is calculated by the following formula.



e = f/a(3.29)

Where f is focal point distance, a is major diameter and b is minor diameter.

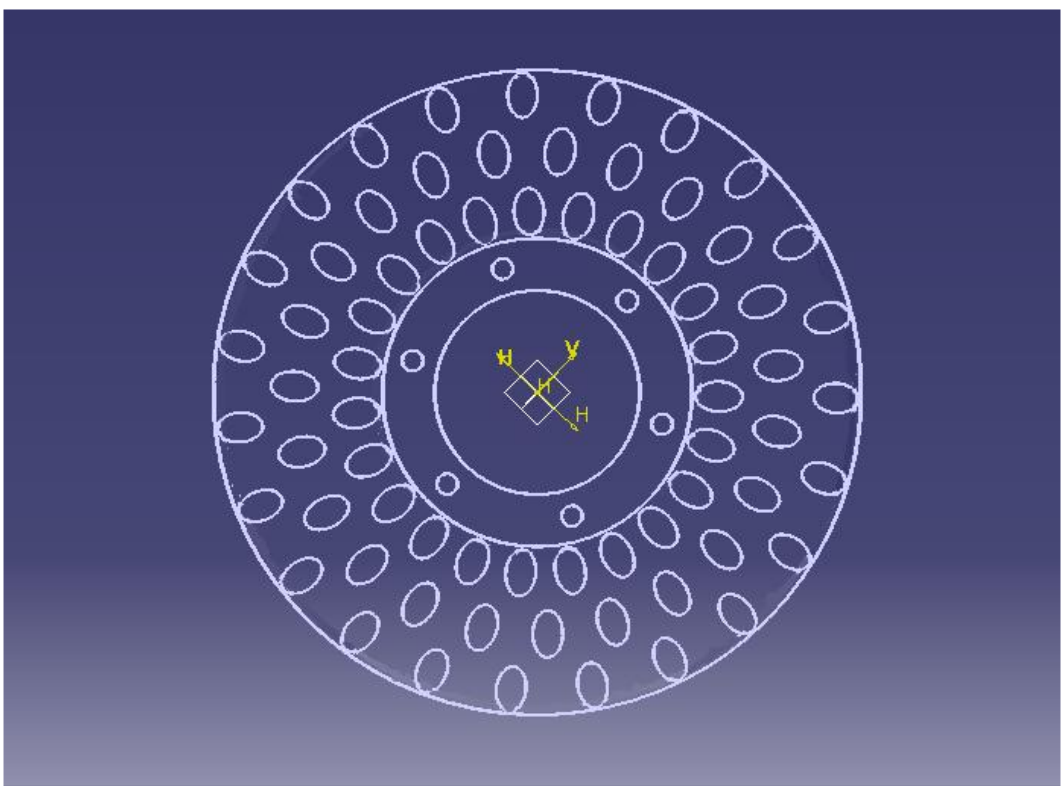


Figure.3.4Pin vented(1) brake disc wire framed view

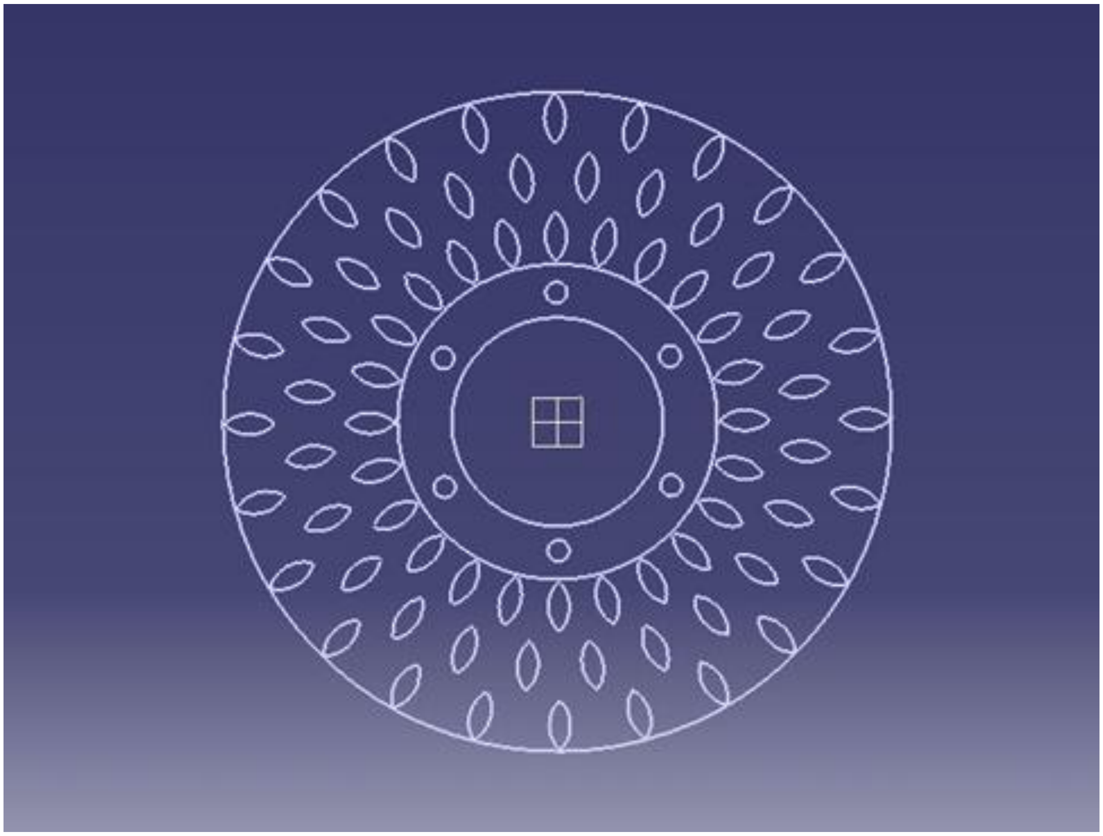


Figure.3.5Pin vented(2) brake disc wire framed view

3.32 Assumptions Used for the Physical Model of Brake Disc

To simplify the analysis, the following assumptions has been made

- Only convection form of heat transfer is considered. The heat transfer due to radiation can be neglected since it has small amount of contribution for the heat transfer (5 % to 10 %). The heat radiation only plays an important role at high temperature and low speeds.
- The disc material is considered as homogeneous and isotropic because young's modulus, Poisson's ratio and the thermal expansion coefficient are assumed to be constant for isotropic material.
- The disc is stress free at ambient temprature before the application of brake. The ambient temperature and initial temperature has been set to 25 °C.

- All other possible disc brake loads are neglected for example, the effects of wear or other atmospheric factors.
- Uniform pressure distribution by the brake pad onto the disc brake surface. Uniform pressure distribution in the contact region is often valid when the pad is new.

3.33 Definition of Material

The material of locomotive disc material is gray cast iron (GFC) ISO standard with high carbon content, with good thermo physical characteristics and the brake pad has an isotropic elastic behavior, whose thermo-mechanical characteristics adopted in this simulation in the transient analysis is recapitulated in table below.

Material Properties	Grey Cast Iron
Thermal conductivity - *k* [W/m ^{o}C]	52
Density – ρ [kg/m^3]	7200
Specific heat - *c* [J/Kg ^{o}C]	447
Poisson's ratio *v* [/]	0.28
Elastic modulus - *E* [G Pa]	1.52×10^{11}

Table 3.4 Material Properties of Grey Cast Iron Disc

3.4 Finite element Analysis

3.41 Introduction

Finite Element Method is a numerical procedure for solving continuum mechanics of problem with accuracy acceptable to engineers. Finite Element Method is a mathematical modeling tool involving discretization of a continuous domain using building-block entities called finite elements connected to each other by nodes for once and moment transfer.

This process includes Finite Element Modeling and Finite Element Analysis. In displacement based FEM, stiffness of the entire structure (Part or assembly) is assembled from stiffness of individual elements. Loads and boundary conditions are applied at the nodes and the resulting sets of the simultaneous equations are solved using matrix methods and numerical techniques. In short, FEM is a numerical method to solve ordinary differential equations of equilibrium. Starting with simple linear static stress and heat transfer analysis, complex simulations involving highly non-linear, fluid flow and dynamic events can be successfully analyzed with ANSYS. ANSYS is general purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user-designated size) called elements. The software implements equations that govern the behavior of these elements and solves them all; creating a comprehensive explanation of how the system acts as a whole. These results then can be presented in tabular or graphical forms. This type of analysis is typically used for the design and optimization of a system far too complex to analyze by hand. In practice, a finite element analysis usually consists of three principal steps;

1. **Preprocessing:** Create and discretize the solution domain into finite elements. This involves dividing the domain into sub-domains, called 'elements', and selecting points, called nodes, on the inter-element boundaries or in the interior of the elements. Assume a function to represent the behavior of the element. This function is approximate and continuous and is called the "shape Function". Develop equations for an element. Assemble the elements to represent the complete problem. Apply boundary conditions, initial conditions, and the loading.

2. **Analysis:** Solve a set of linear or nonlinear algebraic equations simultaneously to obtain nodal results, such as displacement values, or temperature values, depending on the type of problem.

3. **Post processing:** This stage involves processing the nodal data to get other information such as values of principal stresses, heat fluxes, temperature distribution, etc.

3.42 General Analysis Procedure in ANSYS

Similar to solving problems analytically, it needs to define: the solution domain, the physical the model, the boundary conditions and the physical properties. Then solving the problem and present the results. In numerical methods, the main difference is an extra step called mesh generation. This is the step that divides the complex model into small elements that become solvable in an otherwise too complex situation. Below describes the procedure that need to be followed to carry out analysis using ANSYS software.

1. Geometrical Definition:

There are four different geometric entities in preprocessor namely key points, lines, areas and volumes. These entities can be used to obtain the geometric representation of the structure. All the entities are independent of other and have unique identification labels.

2. Model Generation:

Two different methods are used to generate a model: Direct generation and Solid modeling with solid modeling we can describe the geometric boundaries of the model, establish controls over the size and desired shape of the elements and then instruct ANSYS program to generate all the nodes and elements automatically. By contrast, with the direct generation method, we determine the location of every node and size, shape and connectivity of every element prior to defining these entities in the ANSYS model. Although, some automatic data generation is possible (by using commands such as FILL, NGEN, EGEN etc.) the direct generation method essentially a hands on numerical method that requires us to keep track of all the node numbers as we develop the finite element mesh. Solid modeling is usually more powerful and versatile than direct generation and is commonly preferred method of generating a model.

3. Define Material Properties

All elements are defined by nodes, which have only their location defined. In the case of plate and shell elements there is no indication of thickness. This thickness can be given as element property. Property tables for a particular property set have to be input. Different types of elements have different properties. For instance, Beams: Cross sectional area, moment of inertia etc, Shells: Thickness Springs: Stiffness Solids. The user also needs to define material properties of the elements. For linear static analysis, modules of elasticity and Poisson's ratio need to be provided. For heat transfer, coefficient of thermal expansion, densities etc are required. They can be given to the elements by the material property set.

4. Generate Mesh

In the finite element analysis the basic concept is to analyze the structure, which is an assembly of discrete pieces called elements, which are connected, together at a finite number of points called Nodes. Loading boundary conditions are then applied to these elements and nodes. A network of these elements is known as Mesh.

5. Finite element generation

The maximum amount of time in a finite element analysis is spent on generating elements and nodal data. Preprocessor allows the user to generate nodes and elements automatically at the same time allowing control over size and number of elements.. The elements developed by various automatic element generation capabilities of preprocessor can be checked element characteristics that may need to be verified before the finite element analysis for connectivity, distortion-index, etc. Generally, automatic mesh generating capabilities of preprocessor are used rather than defining the nodes individually. If required, nodes can be defined by defining the allocations and translating the existing nodes.

6. Boundary conditions and loading

After completion of the finite element model one has to constrain and load have to be applied to the model. User can define constraints and loads in various ways. All constraints and loads are assigned. This helps the user to keep track of load cases.

7. Solution

The solution phase deals with the solution of the problem according to the problem definitions. All the tedious work of formulating and assembling of matrices are done by the computer and finally displacements and stress values are given as output.

Some of the capabilities of the ANSYS are linear static analysis, non-linear static analysis, transient dynamic analysis, etc.

8. **Post-Processor**

It is a powerful user-friendly post-processing program using interactive color graphics. It has extensive plotting features for displaying the results obtained from the finite element analysis. One picture of the analysis results (i.e. the results in a visual form) can often reveal in seconds what would take an engineer hour to asses from a numerical output, say in tabular form. Important aspects of the results that could be easily be missed in a numerical data are shown. Generally it facilities viewing of:

- Contours of stresses, displacements, temperatures, etc.
- Deform geometric plots
- Animated deformed shapes
- Time-history plots
- Solid sectioning
- Hidden line plot
- Light source shaded plot
- Boundary line plot etc.

The entire range of post processing options of different types of analysis can be accessed through the command/ menu mode there by giving the user added flexibility and convenience.

3.43 Modeling ANSYS CFX

The geometry is imported from CATIA in igs form and an enclosure applied to input the boundary conditions of the moving air around the disc.

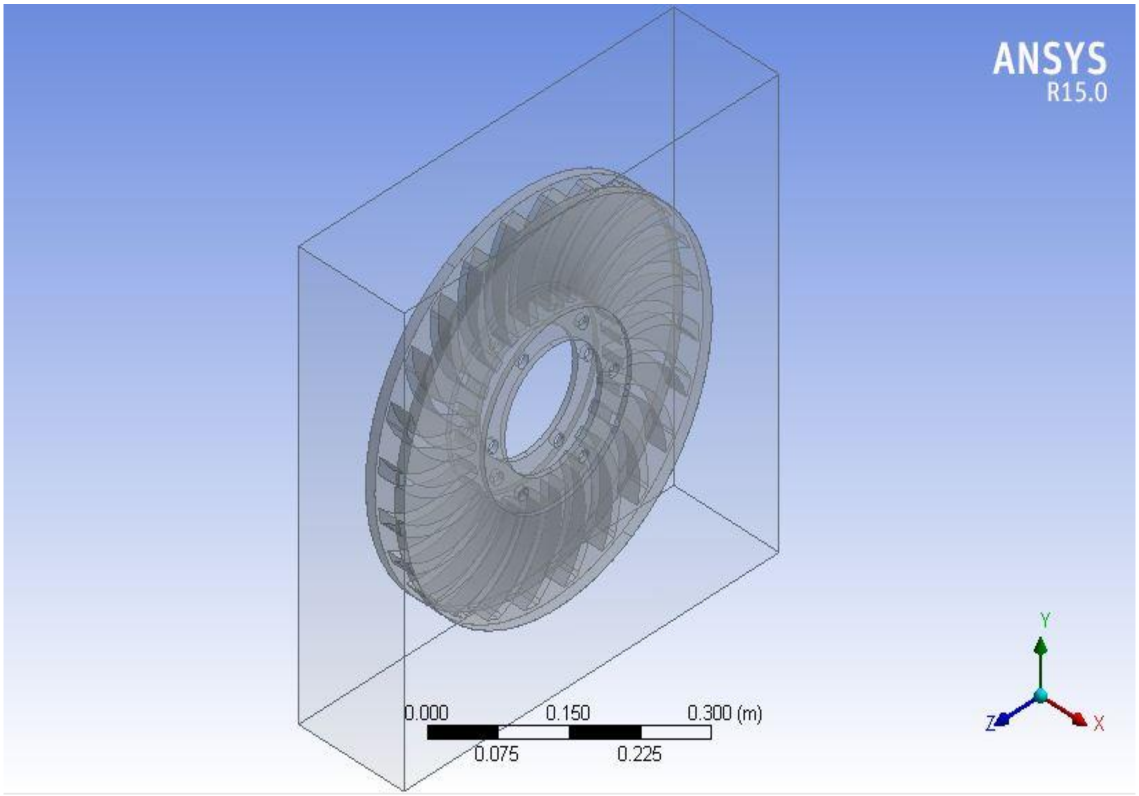


Figure.3.6Imported CATIA V5 geometry inside an enclosure

Mesh Generation

The finite element mesh is the heart of the Finite Element Method. It is therefore very important that it is well designed. The number and type of elements (and therefore the number of nodes) and the grading of the mesh (the variation of the element size throughout) are all crucial and guidelines exist for the determination of these values. Quadratic elements contain a mid-side node on all element edges, whereas linear elements only possess corner nodes, and their use results in greater modeling accuracy (more 'Degrees of Freedom' per element).

The greater the number of nodes and elements, the more accurate the results the mesh will yield. This is analogous to approximating the area under a curve with a number of trapeziums, the greater the number of trapeziums, the more accurate the estimation of the area. However the effect also has a diminishing return, such that less and less advantage is gained by increasing nodes (until error actually increases because of the rise of a particular calculation error). An increase in number of degree of freedom results in a corresponding increase in computation time and therefore to maintain an efficient solution system the number of degree of freedoms has to be carefully chosen. Other errors can be incurred due to the excessive distortion of the prismatic elements (usually brick or triangular) as they are employed in the mesh.

In general it is advisable to use larger elements where stress concentrations are low and smaller elements in a denser mesh where stress concentrations are high for combination of solution efficiency and accuracy. Whilst defining this variation in mesh size it is also advisable to maintain as smooth as possible the rate of change of size from one element to the next. A linear tetrahedral element type is used. In order not to weigh down calculation, an irregular mesh is used in which the mesh is broader where the gradients are weaker (non uniform mesh).

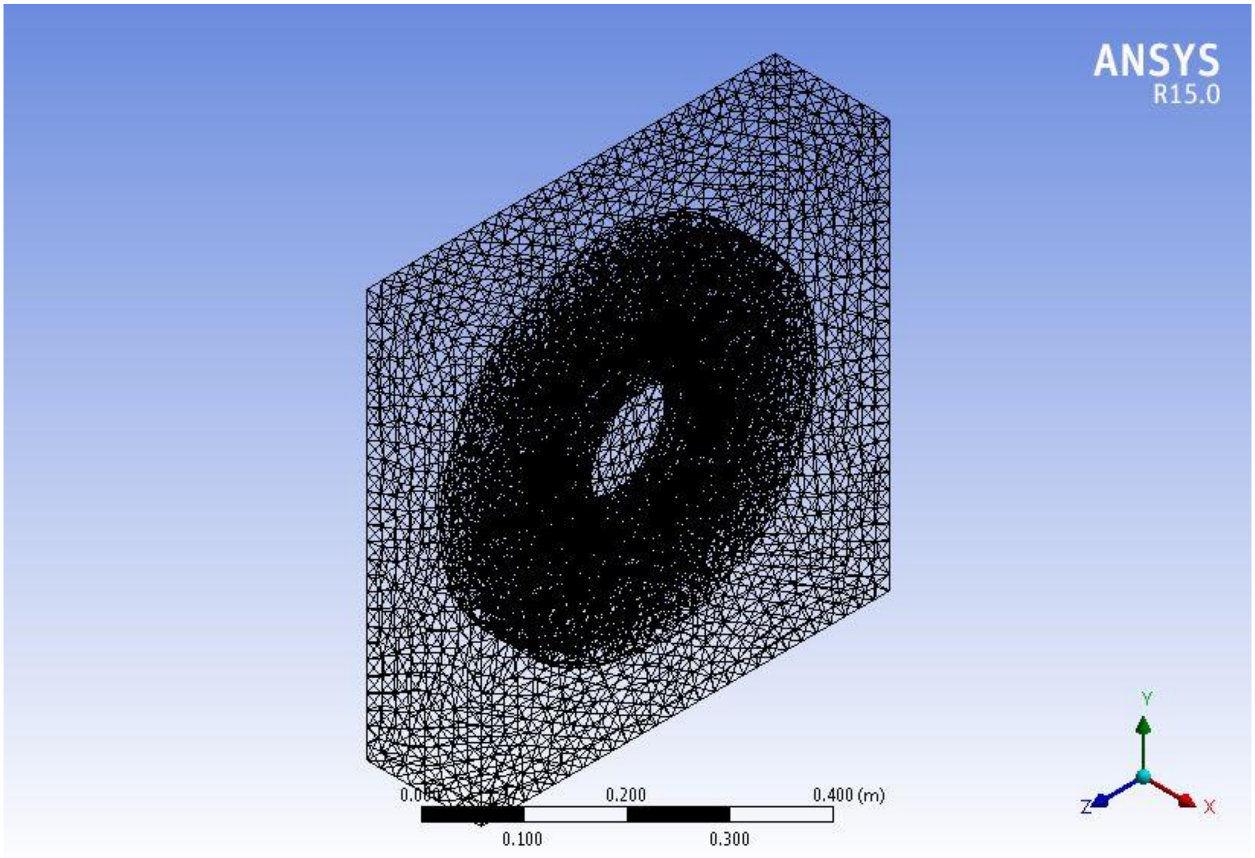


Figure.3.7Meshing the geometry

The mesh has 202298 elements and 48692 nodes with average skewness of 0.295(000143 minimum and 0.863 maximum).

Definition of the Loads and Boundary Conditions

After completion of the finite element model, it was necessary to apply constraints and loads to the model. The analyzed railway disc brake is subjected to the following loads.

- ✓ The initial temperature of the disc and the pads is 25 °C.
- ✓ The surface convection condition is applied at all surfaces of the disc with the values of the convection coefficient that would be imported from the CFX
- ✓ The heat flux into the brake disc during braking is calculated to be 2.73Mwatt

- ✓ The pressure on the rubbing contact surfaces of the disc and brake pad of value 2.4Mpa.
- ✓ Gravity and rotational inertial loads are also incorporated in the boundary conditions

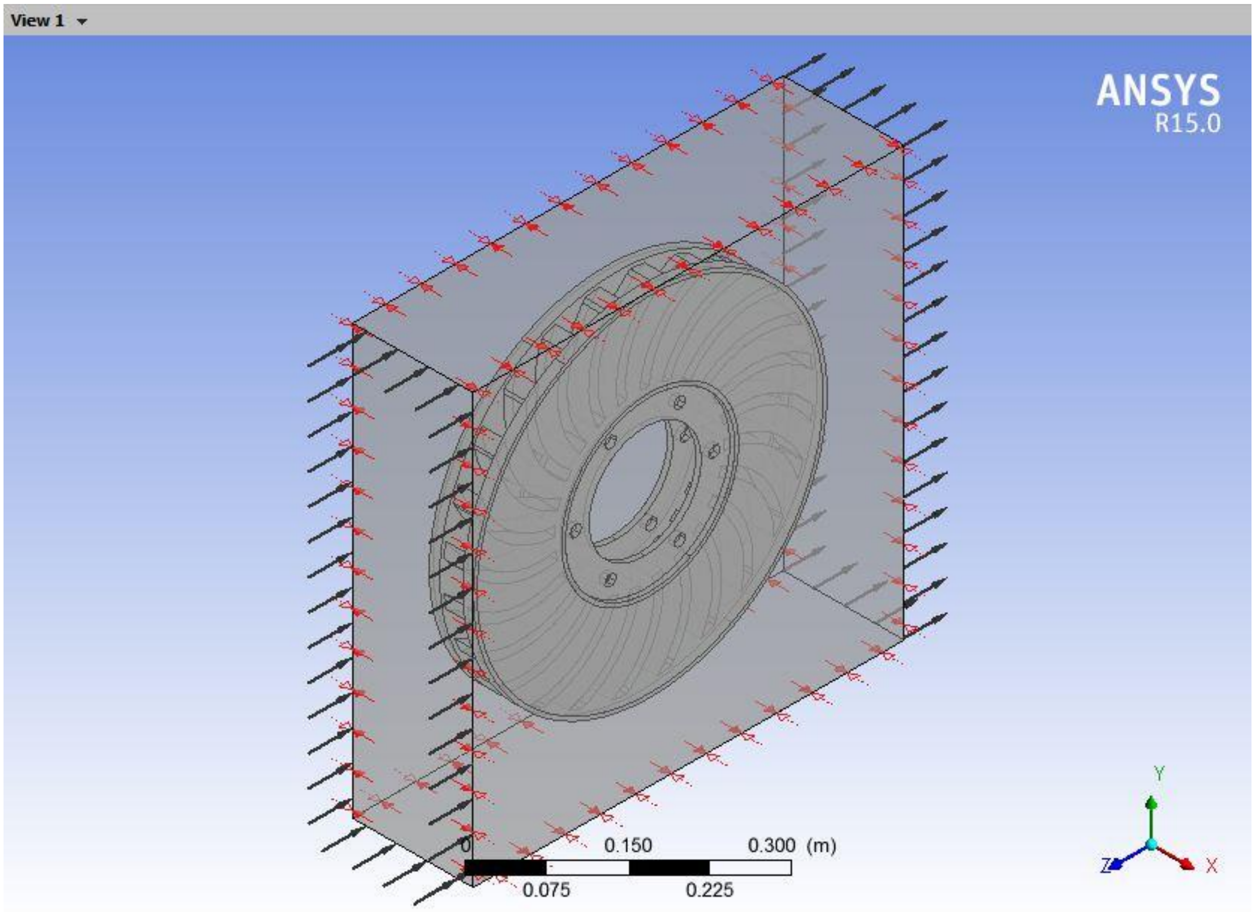


Figure.3.8Boundary conditions

Domain - fluid	
Type	Fluid
Location	B558
Materials	
Air at 25 C	
Fluid Definition	Material Library
Morphology	Continuous Fluid
Settings	
Buoyancy Model	Non Buoyant
Domain Motion	Stationary
Reference Pressure	1.0000e+00 [atm]
Heat Transfer Model	Total Energy
Turbulence Model	k epsilon
Turbulent Wall Functions	Scalable
High Speed Model	Off
Domain - solid	
Type	Solid
Location	B183
Settings	
Domain Motion	Rotating
Angular Velocity	5.8922e+01 [radian s^-1]
Axis Definition	Coordinate Axis
Rotation Axis	Coord 0.1

Table 3.5 ANSYS domain physics for CFX

Domain	Boundaries	
fluid	**Boundary - inlet**	
	Flow Regime	Subsonic
	Heat Transfer	Static Temperature
	Static Temperature	3.0300e+02 [K]
	Mass And Momentum	Normal Speed
	Normal Speed	1.9440e+01 [m s^-1]
	Turbulence	Medium Intensity and Eddy Viscosity Ratio
	Boundary - outlet	
	Flow Regime	Subsonic
	Mass And Momentum	Static Pressure
	Relative Pressure	0.0000e+00 [Pa]
	Boundary - symmetry	
	Type	SYMMETRY
	Location	symmetry
	Boundary - fluid Default	
	Type	WALL
	Heat Transfer	Adiabatic
	Mass And Momentum	No Slip Wall
	Boundary - walls	
	Heat Transfer	Adiabatic
	Mass And Momentum	No Slip Wall
solid	**Boundary - disc**	
	Type	WALL
	Heat Transfer	Heat Flux
	Heat Flux in	2.7303e+06 [W m^-2]

Table 3.6 ANSYS boundary physics for CFX

3.44 ANSYS CFX results of Wall heat transfer coefficient

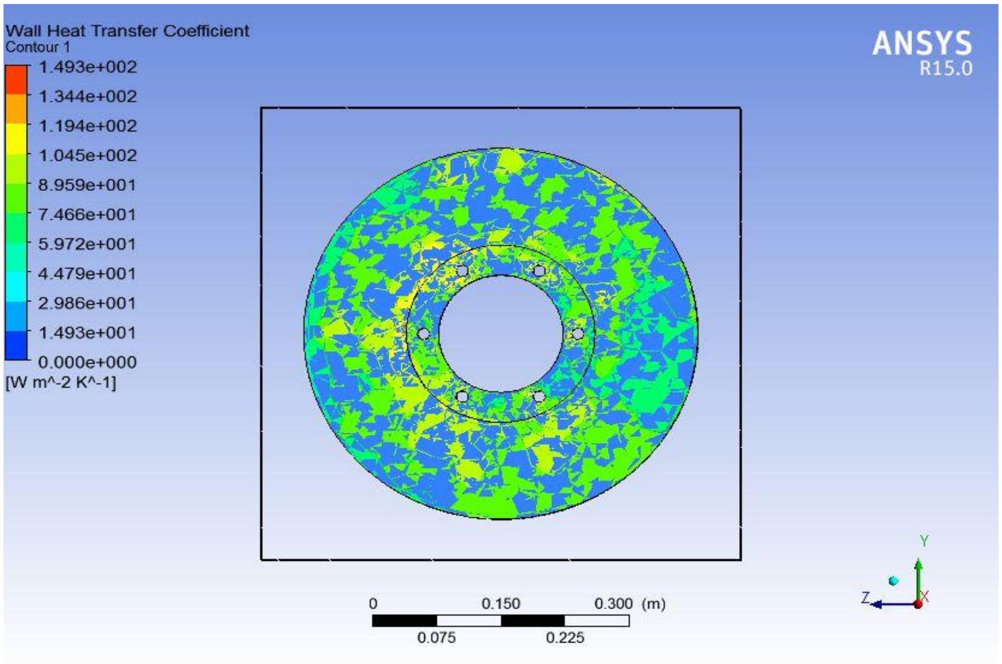


Figure.3.9 Wall heat transfer coefficient of curved vane disc brake

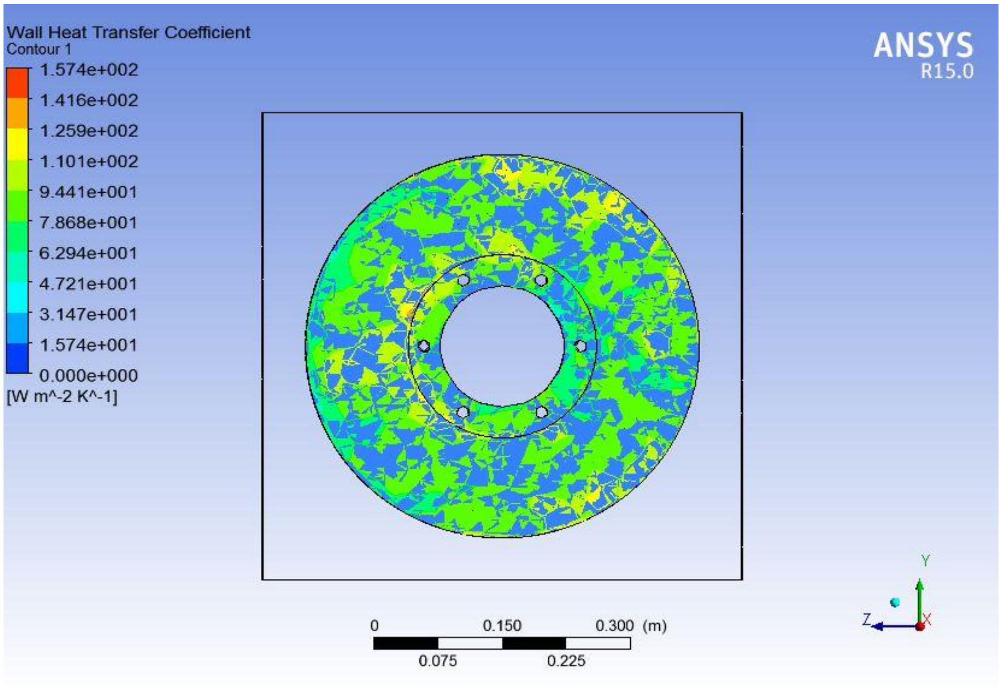


Figure.3.10 Wall heat transfer coefficient of pin vented brake(1)

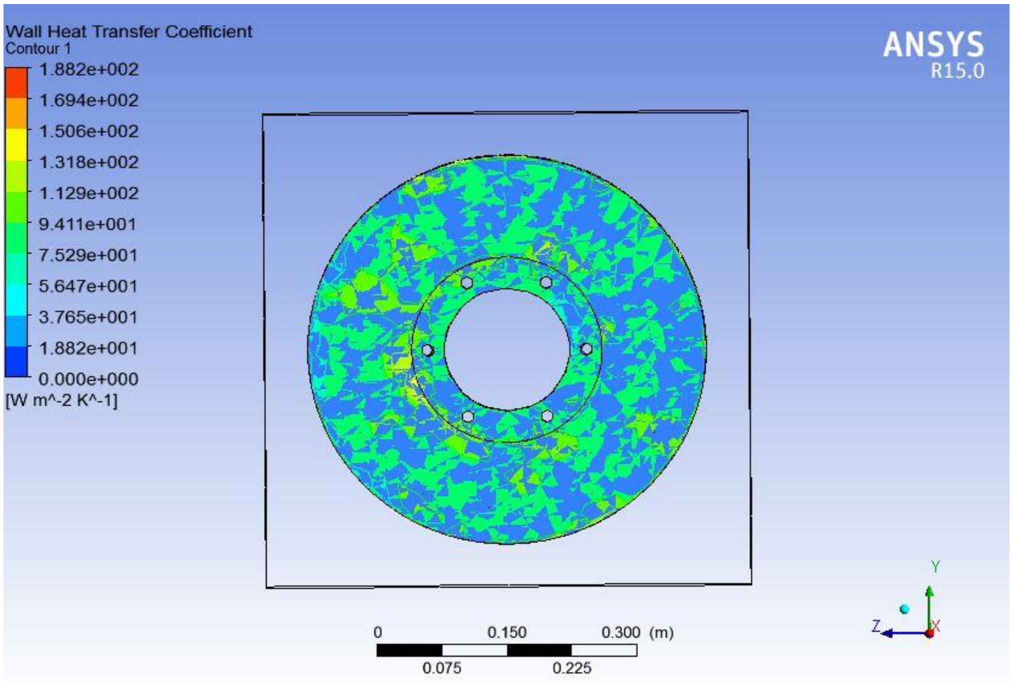


Figure.3.11 Wall heat transfer coefficient of pin ventedbrake(2)

3.45 Thermal Analysis

A Thermal analysis calculates the temperature distribution and related thermal quantities in a system or component. A Transient thermal analysis determines the temperature distribution and other thermal quantities under conditions that vary over a period of time. Therefore rather than steady state thermal analysis, transient thermal analysis is used. Heat flux applied to the brake disc is maximum at the beginning of the braking action and decreases to zero when the vehicle comes to stop. Heat flux at the different time interval is shown in table 3.7.

Steps	Time[s]	Heat flux[W/m^2]
0	0	2.7303e+006
1	1	2.4495e+006
2	2	2.1686e+006
3	3	1.8877e+006
4	4	1.6068e+006
5	5	1.3259e+006
6	6	1.045e+006
7	7	7.6407e+005
8	8	4.8316e+005
9	9	2.0225e+005
10	9.72	0.

Table 3.7 Heat flux distribution vs time

The fluid flow CFX, thermal and structural parts of the ANSYS are combined together to analyze the difference in heat dissipation of each of the models.

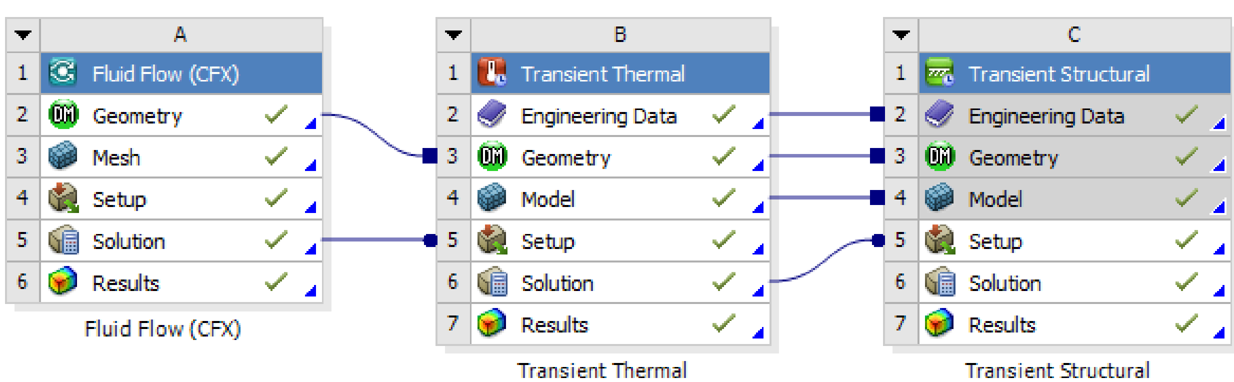


Figure.3.12Combined fluid flow(CFX), thermal and structural

3.46 ANSYS results of temperature and Von Misses stress

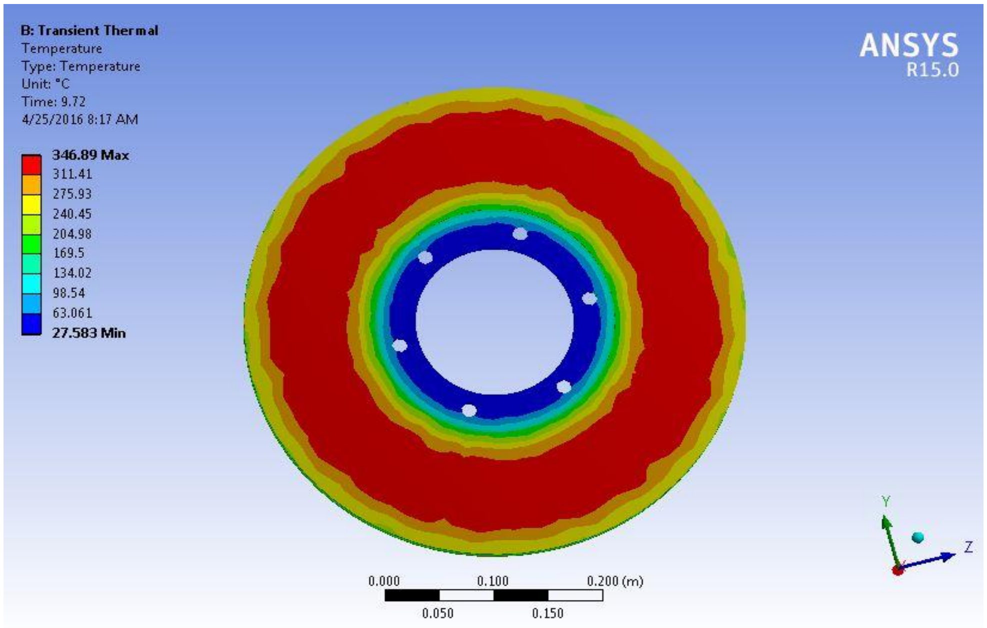


Figure.3.13Temperature distribution of curved vane type brake disc

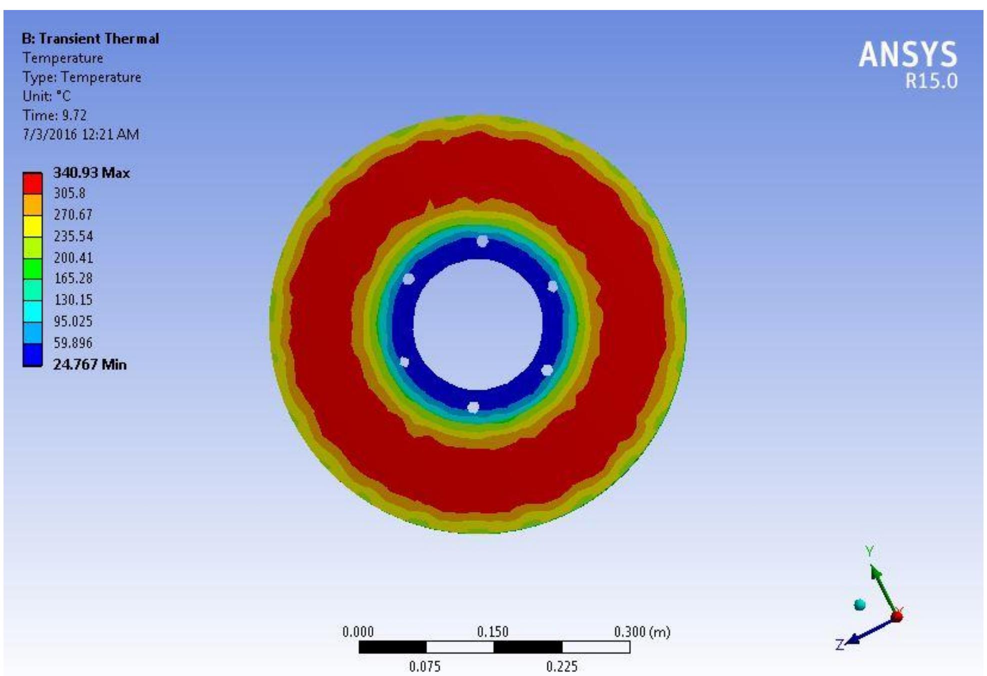


Figure.3.14 Temperature distribution of pin vented brake disc(1)

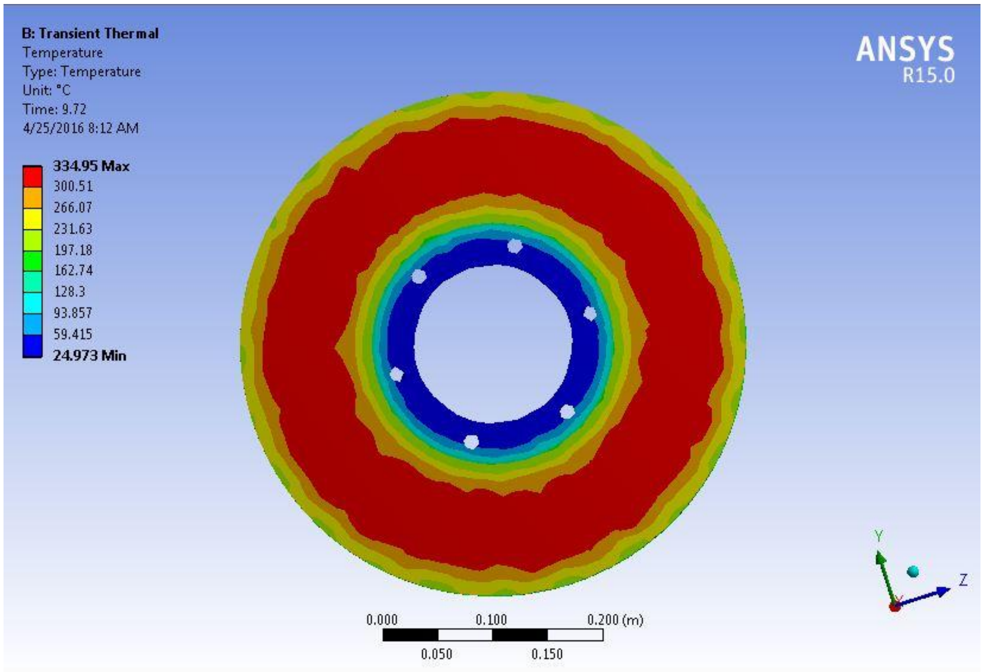


Figure.3.15Temperature distribution of pin vented brake disc(2)

In the transient structural analyses pressure, heat flux, rotational velocity and gravitational and decelerating acceleration of the train are used as an input to calculate stress.

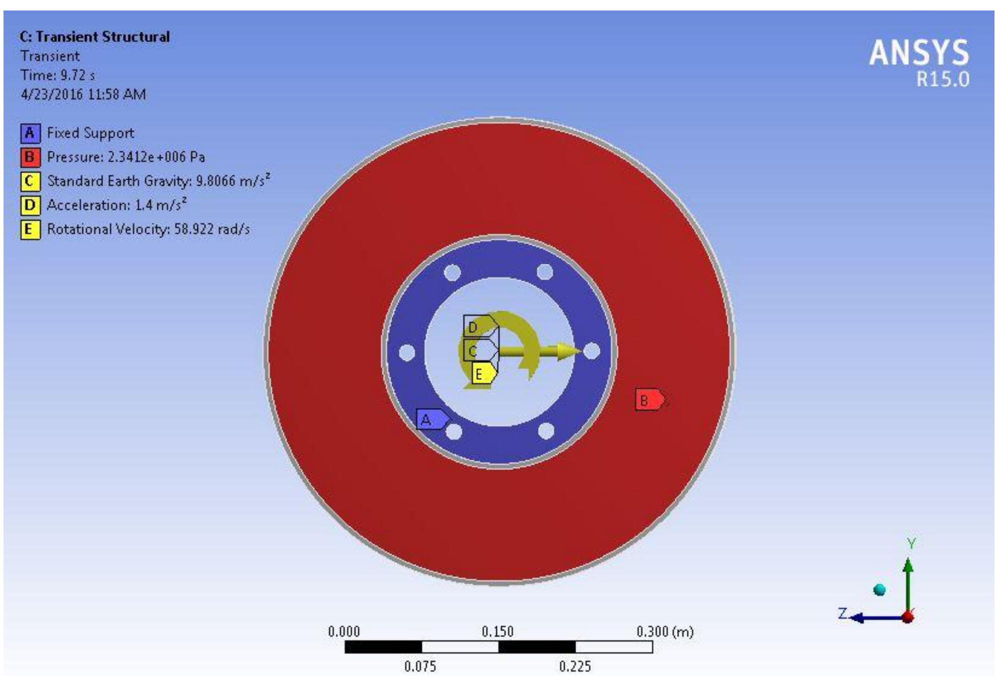


Figure.3.16 Boundary conditions for structural analysis

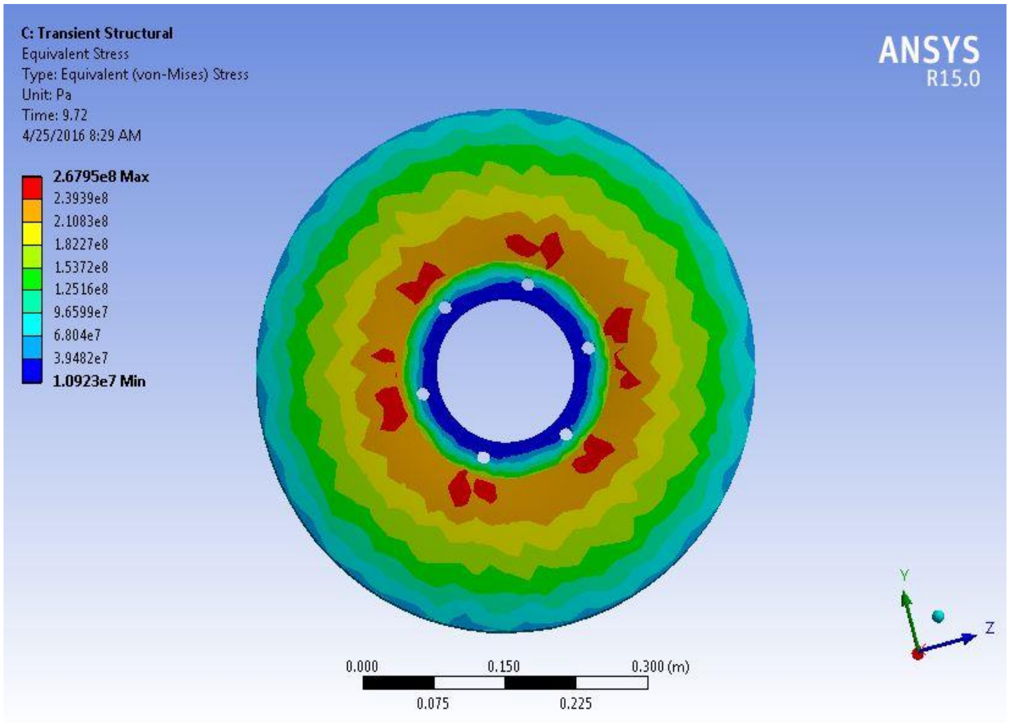


Figure 3.17 Von Mises stress of curved vane disc brake

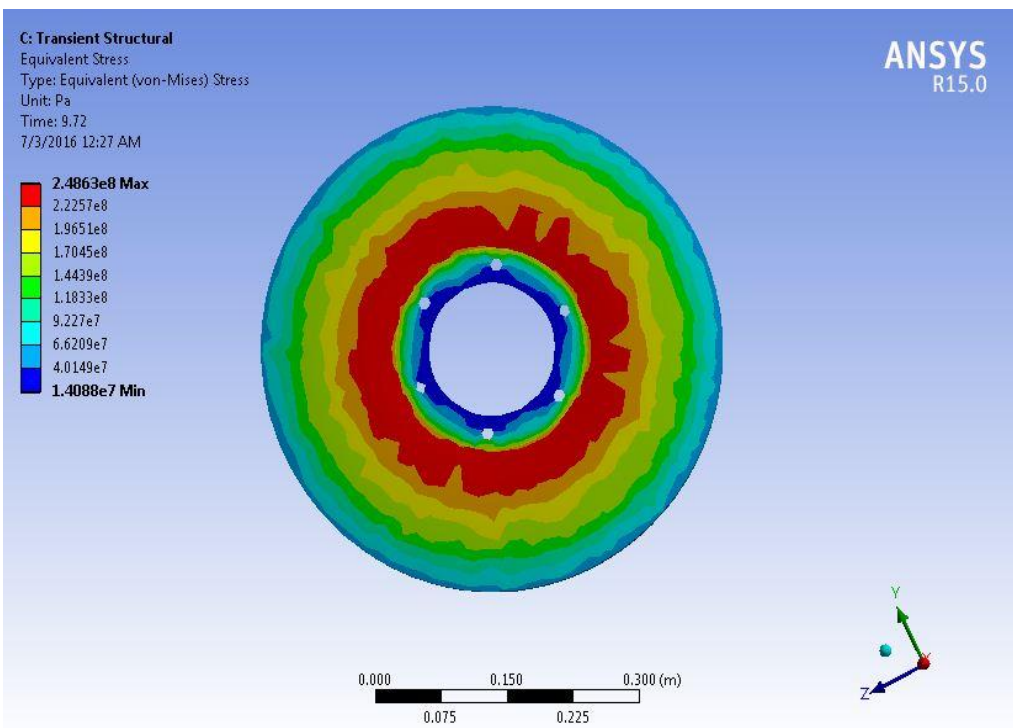


Figure.3.18 Von Mises stress of pin vented brake disc(1)

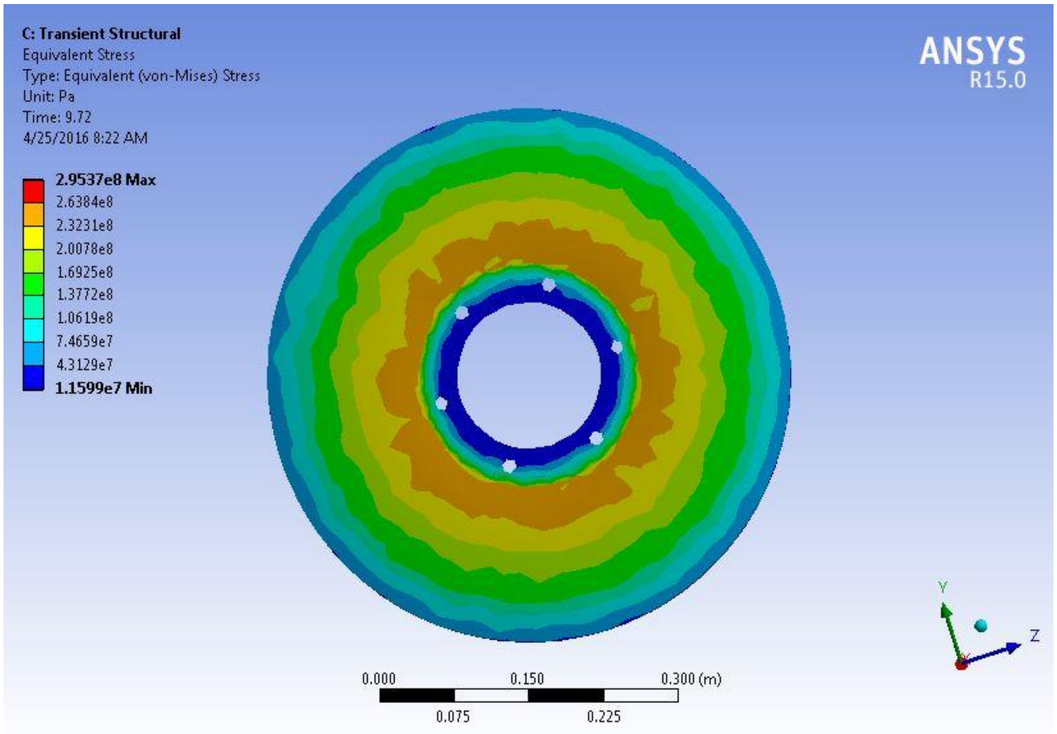


Figure.3.19Von Mises stress of pin vented brake disc(2)

CHAPTER FOUR

RESULT AND DISCUSSION

In this study the effect of changing the internal shape of disc brake is analyzed. Three types of brake rotors; curved vane and two types of pin vented brake rotors have been studied at the emergency braking time of 9.72sec from the maximum operating speed of 70m/s with the highest AALRT track gradient of 3.15^{o}. During a braking action the temperature of disc brake rapidly rises to some maximum value with conduction heat transfer. Then it starts to decrease when convection becomes dominant during extended braking. The maximum value of temperature, deformation and Von Misses stress results at the braking time is tabulated below.

Description	Curved vanes	Pin vented(1)	Pin vented(2)
Convection heat transfer coefficient[W/$m^2$$k$]	149.3	157.4	188.2
Temperature Gradient [^{o}C]	346.89	340.93	334.95
Total Deformation [m]	$392.09e^{-6}$	$394.58e^{-6}$	401.18e-6
Von Misses Stress [Pa]	$267e^{6}$	$248e^{6}$	$295e^{6}$

Table4.1 Summary of results of the disc models

The pin vented disc brake with higher eccentricity has higher convection heat transfer coefficient of 188.2W/m^2*k* and a maximum temperature value of 334.95^{o}C while the curved vane disc brake has shown better structural strength and deformation results with relatively lower convection heat transfer coefficient and higher temperature values. Pin vented rotor creates turbulence inside the passage for better convection. It has been shown that pin profiles affect heat transfer characteristics of a pin vented disc considerably. Comparing the two types of pin vented brakes controlling the eccentricity of elliptic section of pins, it has been possible to increase the convection heat transfer coefficient by 16.36% (30.8W/m^{2}k) and decrease the temperature by 5.98^{o}C and by 11.94^{o}C when compared with curved vanes disc rotor.

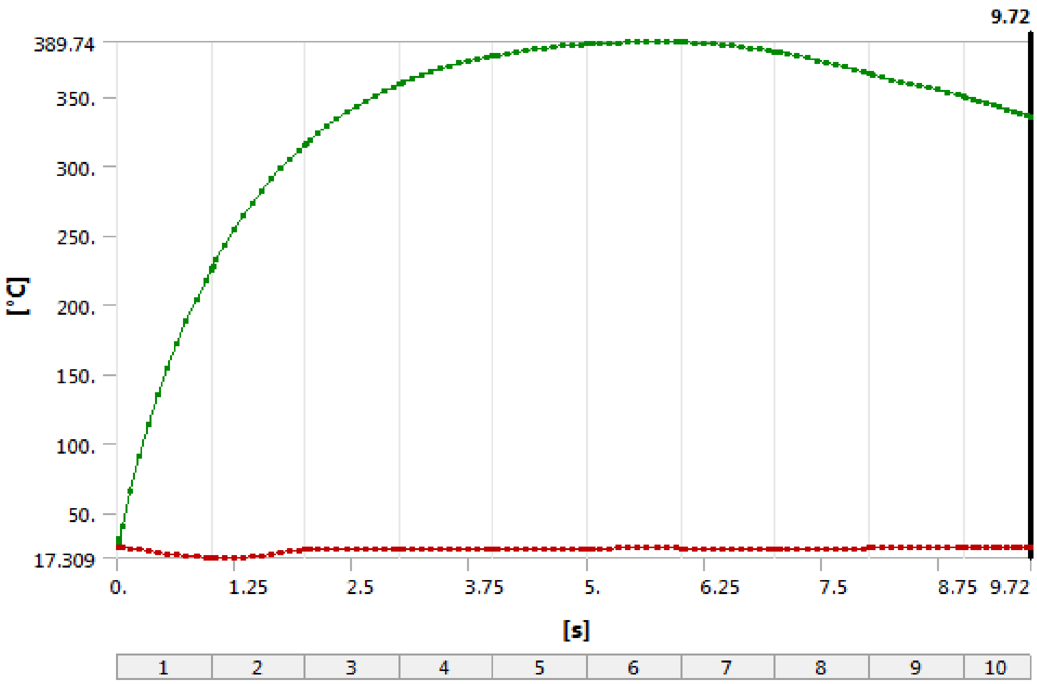


Figure 4.1 Temperature time distribution of the pin vented brake disc

As indicated in the figure the temperature rises during braking to reach maximum value of 389.74^{0}C at 5.74 sec and it starts decreasing as convection the heat flux applied is eased and heat dissipates through convection to the environment. A higher convection heat transfer enables to get a rapid cooling rate and eventually a shorter braking distance.

The Ethiopian AALRT locomotive has a radial vane type ventilated disc brake. With maximum speed and track gradient downhill braking condition, the emergency braking distance is 94.48 meters. The emergency braking distance for the second type of pin vented disc brake is 87.08 meters. Therefore a 7.8% (7.4meters) reduction in the breaking distance is achieved. The pin vented disc brake, having good heat dissipating characteristics, takes a higher amount of heat flux and brakes the train in a shorter distance for the same temperature value of the brake rotor.

CHAPTER FIVE

CONCLUSION, RECOMMENDATION AND FUTURE WORK

5.1 Conclusion

In this study a coupled CFX, thermal and structural analyses of disc brake is made with ANSYS-15 software. The fluid flow CFD/CFX analysis is used to simulate the cooling effect of the moving air as the brake disc rotates. Convection wall heat transfer coefficient is taken from the CFX analysis result which then used as an input to the thermal analyses. The CFD/CFX package calculates discrete values of heat transfer coefficient at different points of the geometry for accurate results. Temperature and heat flux results from the thermal analysis are combined with the structural analysis to demonstrate the effect of temperature on stress distribution and deformation.

Three types of disc models have been analyzed: a curved vane and two types of pin vented brake rotors. It has been observed that keeping all other parameters as volume and mass the same, the temperature decreases while the convection heat transfer increases changing the internal shape of the disc from curved vane to pin vented rotors with different eccentricity of pin sections. After emergency braking of AALRT maximum track gradient of 3.15^0 with overload condition, a maximum temperature gradient of 334.95^0Cis achieved with the pin vented brake disc with higher eccentricity while it is 346.89^oC and 340.93C for the curved vane and the second type of pin vented disc respectively. Controlling the air flow characteristics and heat convection with the internal shape and geometry of the brake rotor, a 7.8% reduction of braking distance is achieved.

5.2 Recommendation

This study is done based on the operating parameters of the AALRT locomotives and track conditions. The AALRT currently uses radial vane type of disc brakes. It is known and has been studied before that curved vane disc brakes have better cooling characteristics than the radial vane brake discs and know pin vented brake disc of specific geometry used in this study has proven to have even better results. Therefore I strongly recommend the implementation of this type of disc model for a safer and economical operation as lower temperature means longer lifecycle time of the brake and also the braking fluids used in the caliper assembly which would have a good economic advantage**.**

5.3 Future work

Future research can be done changing the size and alignment angle of the cooling fins and the analysis can further be done with different materials and braking conditions as braking at an elevated temperature due to prior braking.

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