Addis Ababa University

Addis Ababa Institute of Technology

School of Mechanical and Industrial Engineering program in Railway Engineering

Analysis the effect of unequal force application on A.A LRT disc brake during emergency brake

By

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(Railway stream)

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Abstract

A train disc brake system is used to perform three basic functions, i.e. to reduce speed of a vehicle, to maintain its speed when travelling downhill and to completely stop the vehicle. During these braking events, the disc brake may suffer of temperature, deformation and stress issues. It is quite sometimes that the disc brake components fail structurally and/or having severe deformation on the pad. Thus, this paper aims to examine stress concentration, structural deformation and contact pressure of brake disc and pads during braking event by employing commercial finite element software, ANSYS and CATIA for modeling purpose with considering the effect of unequal pressure application on the two opposite side of the disc which are developed due to brake line corrosion, caliper sticking, failure of piston seal, variation of rotor thickness and also failure of caliper bolts.

The main results of this analysis is consider factors which are relate about disc brake structure and contact behavior such as deformation, Von Misses stress and temperature which rise when breaking action with considering the unequal heat flux which generate on opposite side of disc when unequal pressure applied on two opposite side of the disc and define from the analysis result there is a high deformation, stress and temperature difference between equal pressure and unequal pressure applied on two opposite side. I.e. when unequal pressure applied on two opposite side, there are greater temperature, heat flux, deformation and stress as compare with equal pressure applied on two opposite disc side.

KEY WORDS: Finite element (FEM), pressure, heat flux and AALRT, disc brake and unequal pressure.

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Notations

q_s	*The specified surface temperature*
T_s	*The unknown surface temperature*
h	*convective heat transfer*
k	*Thermal conductivity*
B_F	*Total braking force*
α	*Gradient*
NS	*North To South*
EW	*East To West*
s_b	*Braking distance*
t_b	*Braking time*
COM	*change in center of mass*
Q_d	*Heat dispassion of disc*
Q_p	*Heat dispassion of pad*
k_d	*Thermal conductivity of disc*
k_p	*Thermal conductivity of pad*
μ	*Coefficient of friction*
ω_o	*initial angular velocity*
R	*Radius of the disc*
A_b	*Swept area covered by brake pads*
ri	*The radius from the center of mass*
mi	Total mass of wheel set and disc

CHAPTER ONE

1.1) Introduction

The braking component represents one of the most fundamental and safety-critical system which is used in modern railway vehicles. The ability of the braking system to bring a vehicle to rest and safe controlled to stop is absolutely essential for preventing accidental vehicle damage and any personal or equipment loses. Therefore, the braking system of any railway vehicle is very important, especially in slowing down or stopping the rotation of a wheel by pressing brake pads against rotating wheel discs. An interaction between a brake disc and friction material of a disc brake is characterized by a number of dry contact phenomena. These phenomena are influenced by brake operation conditions such as, applied pressure, speed, and brake interface temperature and also material characteristics of a friction couple which is used to make the disc rotor and the pad [4] .Therefore the coefficient of friction should be relatively high and keep a stable level irrespective of temperature change, humidity, degree of wear and corrosion, presence of dirt and dust spraying from the rail. In short the braking performance of a railway vehicle can significantly be affected by the temperature rise in the brake components, coefficient of friction which is found due to the rotor and pad contact, and also the disc friction force which is used to slow down or stop the vehicle.

The frictional heat generated at the interface of the disc and the pads can cause a high temperature during the braking process. Particularly, the temperature may exceed the critical value for a given material, which leads to undesirable effects, such as brake fade, local scoring, thermo elastic instability, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally excited vibration.

Generally the function of the braking system is to retard the speed of the moving vehicle or bring it to rest in a shortest possible distance whenever required. Brakes are either mechanical devices or electro-mechanical which is used for increasing the frictional resistance that retards the turning motion of the vehicle wheels. It absorbs either kinetic energy or potential energy or both while remaining in action and this absorbed energy appears in the form of heat. While moving down a steep gradient the vehicle is controlled by the application of brakes. In this case brakes remain in action for a longer period making it imperative to dissipate the braking heat to atmosphere as rapidly as much as possible.

Most of the time the LRT railway vehicle are fitted with three type brakes, such as, the service, emergency and safety or magnetic brakes. The service brake is used to control the speed of the vehicle and to stop it when and where desired, by the application of force on the brake pad. The emergency brake, applied by a lever, is used to keep the vehicle from moving when parked and the safety or magnetic brake is used to stop when the case is very difficult.

1.2) Statement of the problem

During the braking operation, the pressure pushes the pads into the disc and therefore pads and disc brake are sliding contact. Clamping forces resist the movement of the train slows down or eventually stops. But at the braking condition, the brake disc exposed to extreme temperature due to unequal pressure distribution or due to unbalance force applied on two opposite sides of disc. This unequal pressure or force has its own thermo-mechanical effect such as temperature, stress and deformation on the disc brake. This unequal pressure developed in the case of disc effective area which are not equal on opposite side, Uneven wear of pad in opposite side, variation of disc thickens ,corrosion of brake pipe etc. [2].The main statement of this paper is analyzing the effect of the unequal pressure and compare the result with the normal or equal pressure effect when applied on two opposite side of disc. This paper focus on the following point.

- In what manner does the temperature of disc brake system will be rise up during braking?
- At what condition, unequal pressure applied on two opposite side of disc braking system?
- What are the problems faced by applied unequal pressure in disc brake system?
- What are the methods that will follow to avoid unequal pressure applied on opposite side of disc?
- How the disc brake systems provide equal braking force as compare with drum brake?

1.3) Objective of the research

1.3.1) General objective

The general objective of this research is to analysis the effect of unequal pressure as compare with the normal or equal pressure applied on two opposite sides in the disc braking system on the thermo-mechanical property of disc brake.

1.3.2) Specific objective

The specific objective of this research.

- To estimate the external load which exert on the disc rotor.
- To analysis the effect of unequal pressure applied on the two opposite side and disc brake thermo-mechanical effect.
- To consider the effect of the unequal pressure on temperature, stress and deformation on the disc brake. To estimate the temperature generated during unequal pressure braking event and compare it the normal or equal pressure application.
- To estimate the deformation and stress which influence the disc brake service life.
- To explain the cause which develop unequal pressure applied on two opposite disc side.

1.4. Research Methodology

In this paper, consider the primary and secondary data source which has good contribution in order to acquire reliable information, numerical analysis for the purpose of an input to the software, CATIA and ANSYS for modeling and analysis purpose, interview questions whom it concerned and use time, cost and effort saving method in order to going deduce conclusion and easily understandable for user about the effect and cause of unequal pressure applied on two opposite side of disc brake.

1.5) Data collection /Research materials

For the purpose of this research and in order to achieve the objective listed in the subtopic of specific objective of the research the data will be collected through secondary data collection method.

- By browsing different published papers and journal.

- By browsing different disc braking related books and
- By visiting the Addis Ababa LRT manufacturing work shop found in leg hare in order to gathers information if there is any document which have explain about disc braking structural contact analysis.
- By distribute questions which have to get reliable idea and discus directly face to face with a person whom it may concern.

1.6) Research Procedure

In this subtopic the overall procedure of the research work is stated as clear as possible as the following manner. Sequentially having in mind that specific question to answer.

- Frist collect the data which have relate issue about disc brake.
- Understanding clearly the existing disc brake structure and contact area.
- Reading other recent related research works that have been done before by other researchers.
- Analyzing appropriate tools to solve the complexity stated in the statement of the problem subtopic like.
- Understanding the cause which develop this unequal pressure on opposite side disc brake.
- Train appropriate software such as CATIYA AND ANSYS

  CATIYA: to simply show the complete modeling diagram.

  ANSYS: for modeling and analyzing purpose.

1.7) Research questions

- What are the reason LRT used disc braking system
- What are the advantage disc brake as compare to the drum brakes?
- What are the main cause for uneven wear of brake pad in opposite side of disc brake?
- What are the main cause for different disc active area present on two opposite side of disc?
- What method use LRT in order to protect unequal pressure or load applied on two opposite side of disc brake?

1.8) Significance of Study

Any research have its own advantage and benefit for the next researcher and use as a pioneer when the next generation wants to study about that problem or issue. This research deals relate to disc brake structure, contact analysis and related causes which must be solved and used as essential for the disc stability and express the cause which are developed unequal pressure on disc brake opposite side. Normally this research also has its own significance and advantage. Some of them are listed like:

- To compare the thermo-mechanical effects when equal pressure and unequal pressure applied on two opposite sides of disc brake.
- To understand the main cause which develop the unequal pressure on the two opposite side?
- To minimize the disc brake defect or failures and increase the disc service life and also create the riding comfort by identifying the main cause.
- To estimate the effect of unequal pressure applied on two opposite side of disc brake

1.9) Limitation

It is difficult to get reliable information or material from whom it concerned for the support one and there is a problem regarding with the internet connection to browsing the material. And it is difficult to get the material lab test which is used to relate with experimental result.The time is also be a major constraint or limitation of this research .In order to contribute this and others factors, should use the simplified method which used to the research will became clear for users and estimate the delamination which it is comfort to perform the research in the short time as much as possible and select the appropriate material used to support instead of the main parameter.

CHAPTER TWO

Literature rivew

2.1) Introduction

A disk brake consists of a cast iron disc which is the most common material and bolted to the wheel hub and a stationary housing which used to press the rotor pad is known as caliper. The caliper is connected to some stationary part of the railway vehicle like the axle casing mounted or the wheel set mounted as is cast in two parts each part containing a piston. In between each piston and the disc there is a friction pad which used to held the rotor in the position by retaining pins, spring plates etc. passages are drilled in the caliper for the purpose of the fluid to enter or leave each housing? The passages are also connected to another one for bleeding. Each cylinder contains rubber-sealing ring between the cylinder and piston which is used to retract the spring properly. The working principle is comprises two opposing pistons each faced with a pad of lining material which is against the rotating motion of the disc. When the hydraulic pressure is increased the pads are forced against the rotating metal friction disc, exerting a normal force at each contact [16]. The two normal forces cancel one another axially but cause additive tangential friction forces which oppose the disc's motion and decelerate it.

2.2) General over-view for improvement of brake system

Train braking is a very complex process, specific to rail vehicles and of great importance by the essential contribution on the safety of the traffic. This complexity results from the fact that during braking occur numerous phenomena of different kinds - mechanical, thermal, pneumatic, electrical, etc. The actions of these processes take place in various points of the vehicles and act on different parts of the train, with varying intensities. The major problem is that all must favorably interact for the intended scope, to provide efficient, correct and safe braking actions[38].

The purpose of braking action is to perform controlled reduction in velocity of the train, either to reach a certain lower speed or to stop to a fixed point. In general terms, this happens by converting the kinetic energy of the train and the potential one - in case of circulation on slopes - into mechanical work of braking forces which usually turns into heat, which dissipates into the environment.

At first, the rather low locomotives power and traction force allowed braking using quite simple handbrakes that equipped locomotives and eventually other vehicles of the train. As the development of rail transport and according to increasing traffic speeds, tonnages and length of trains, it was found that braking has to be centralized, operated from a single location - usually the locomotive driver's cabin and commands have to be correctly transmitted along the entire length of the train.

2.2.1) Brake characteristics

Brakes are subject to three main problems in service; cracking, wear and distortion. Cracking occurs usually because of high stresses (often 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 an inordinate amount of stress on the component [33].

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 or wheel set. 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.

2.2.3) Thermal deformation of disc brake

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, which are most common due to different factors[40]

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

- **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.
- **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 transmitted to the driver through the suspension, chassis and steering gear of the vehicle.
- **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.

2.2.4) Classification of brakes (depend on transformation of energy)

- Mechanical brakes.
- Electric brakes.
- Hydraulic brakes.

The mechanical brakes also may be sub divided into the following two groups depend on the direction of acting force when the braking action applied.

- Radial brakes.
- Axial brakes.

1) Radial brakes

In these brakes the force acting on the brake drum is in radial direction. The radial brake may be subdivided into external brakes and internal brakes.eg tread or shoe brakes

1.1) Tread brake (brake shoe)

Tread braking uses block-shaped brake shoes which is made of cast iron or other material. Then these brake shoes are pressed onto the running wheel treads. Then the train's kinetic energy is converted into heat energy by the mechanical friction between wheel treads and brake shoes. The heat generated is then dissipated into the atmosphere through convection and the braking force is developed [21].

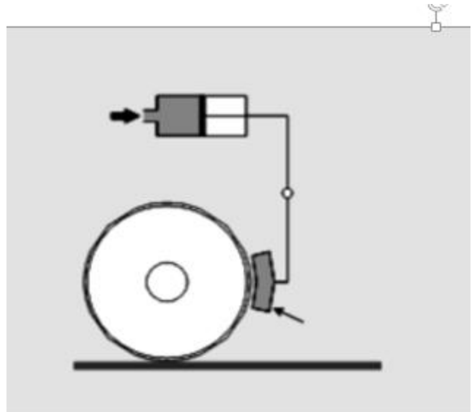


Figure 1.1:-Train shoe brake

2) Axial brakes

In these brakes the force acting on the brake rotor is only in the axial direction. e.g. Disc brakes, Cone brakes.

2.1) Disc brake

A disc brake consists of a cast iron disc bolted to the wheel hub and a stationary housing called caliper. The caliper is connected to some stationary part of the vehicle, like the axle casing or the stub axle and is cast in two parts, each part containing a piston. In between each piston and the disc, there is a friction pad held in position by retaining pins, spring plates etc. passages are drilled in the caliper for the fluid to enter or leave each housing[22]. These passages are also connected to another one for bleeding. Each cylinder contains rubber-sealing ring between the cylinder and piston.

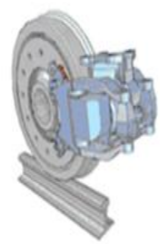


Figure 1.2:- Railway disc brake

2.2.5) Working principle of the hydraulic brake control unit

The main function of the hydraulic unit is to produce and regulate the forces needed for the continuous variable, passive and service brakes, emergency brake and safety brake. When the device is powered off and the pipeline pressure is removed, the brake applies the maximum braking force. In order to be able to realize application and release of brakes, the braking force is continuously regulated according to the power brake demand signal received from the external brake control device [7].

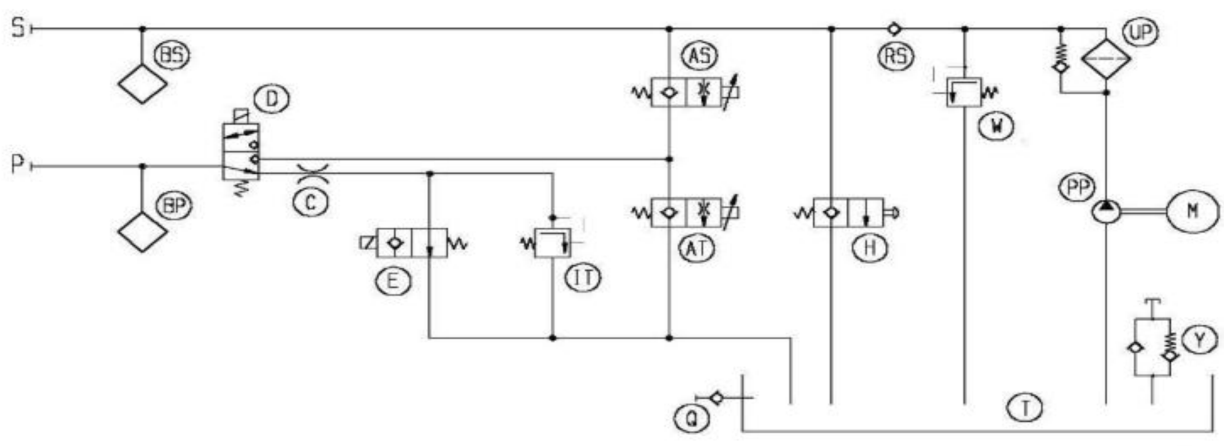


Figure 1.3:-Schematic diagram of hydraulic unit

AS Control valve, AT Control valve ,BP Pressure sensor for brake pressure, BS Pressure sensor for accumulator pressure ,C safety brake throttle D, Safety brake valve ,E Parking brake valve ,H Hand valve, IT Pressure limiting valve ,M Motor ,P Hydraulic port for the brake pipe, PP Pump, RS Check valve ,S Hydraulic port for the hydraulic accumulator, UP Delivery filter and check valve, W Pressure limiting valve ,Y Breathe valve (oil tank) ,Q filling pipe.

2.3) Types of train braking system

There are different types of braking system according to working mechanism and which is comfort for controlling purpose. Some of them are [24].

- Air (pneumatic) brake
- Electrically controlled (air) pneumatic brakes
- Regenerative Brakes
- Rail (eddy current)Brake
- Electromagnetic rail brake

2.3.1) Air brake

The air brake uses the compressed air as the source of brake force and controls the brake force by changing the pressure intensity of compressed air and the most common are two types which are sensitively used in railway vehicles in most countries but has its own disadvantage because of air is compressible and not reliable function.

2.3.1.1) Direct Air Brake (Straight Air brake)

In the direct or straight air brake, the unique feature is the brake pipe connects brake cylinder directly as shown into the following figure

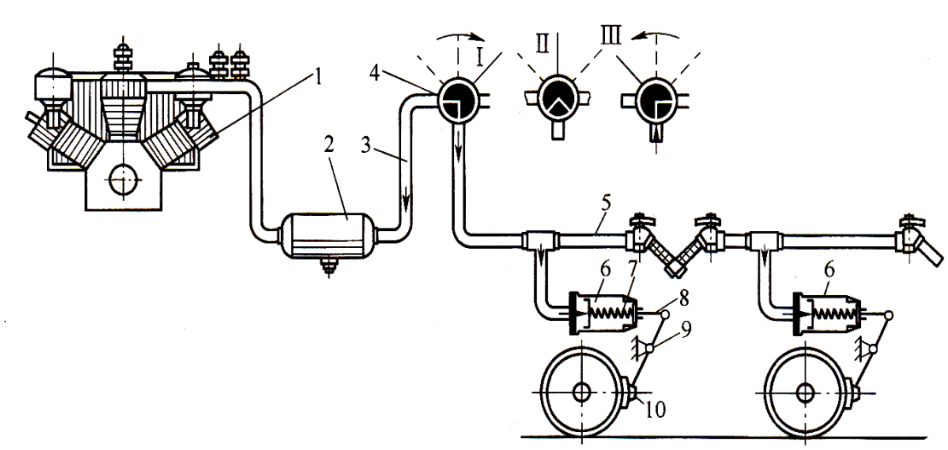


Figure 2.1:- Features of Straight air brake [28]

When pressure intensity in brake pipe is increased, the pressure intensity in brake cylinder is also increased, resulting in the brake application. When brake pipe discharges its compressed air, brake cylinder also discharges its compressed air, resulting in the release of train. The composition of brake system is simple and has the function of gradual brake and gradual release, so it is easy for the driver to control the train. There is a big time difference between front and rear vehicles when the train is braked or released, making it unsuitable for the train with long length. The train will lose its braking ability when it is separated into several parts (at least two parts), making it replaced by the automatic air brake

2.3.1.2) Automatic air brake

When compared the automatic air brake with straight air brake system, a triple valve and an auxiliary reservoir are added on each vehicle in automatic air brake system. The triple valve connects Brake pipe, Auxiliary reservoir and Brake cylinder.

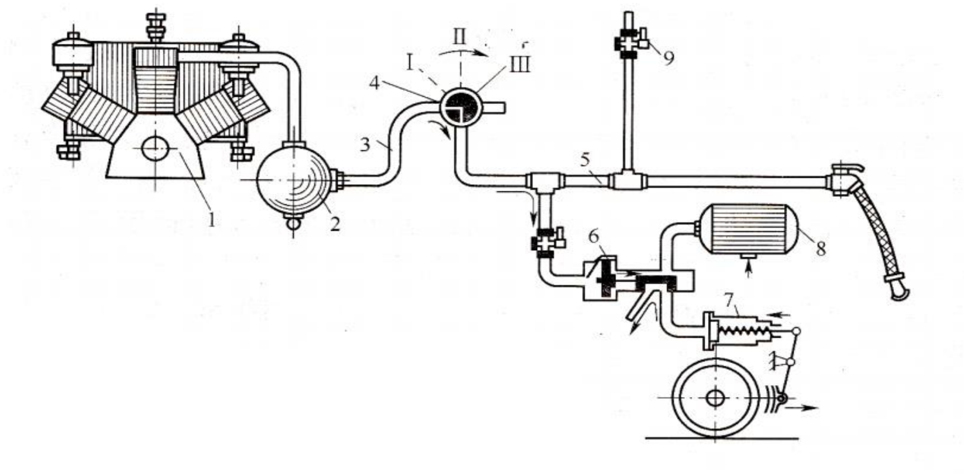


Figure 2.2:- Diagram of automatic air brake [6]

2.4) Electrically controlled pneumatic brake

The introduction of electric traction and multiple unit control was the spur which eventually produced electrically controlled air brakes. The rise of rapid transit operations in cities, with their high volume and frequent stops and starts, meant that quick responses to brake commands and accurate stopping at stations was an essential ingredient in getting more efficiency. If the electric control is out of order, the brake system can still be controlled pneumatically. [13]

2.5) Electromagnetic rail brake

The electromagnetic rail brake operation is based on developing electromagnetic attractive forces towards the rail which causes a normal application force acting on their contact surfaces that are in relative displacement. This leads to friction forces between the magnetic brakes and rails, opposing the vehicle's direction of motion, which generate braking forces. The electromagnetic brake is used as additional wheel-rail adhesion braking system [34].

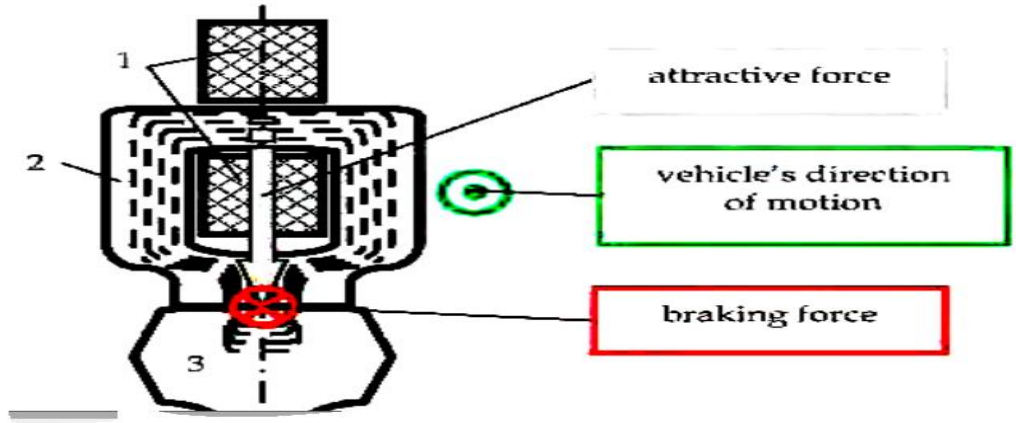


Figure 2.3:- Electro- magnetic brake [35]

2.6) Rail(eddy current) braking system

The brake consists of a magnetic yoke with electrical coils underneath that are positioned along the rail and magnetized with alternating north and south poles. When the coils are put under current while the brake is not in motion a symmetrical magnetic field is generated that includes the rail head and exerts a vertical magnetic force (F). When the magnet is moved along the rail it induces a non-stationary magnetic field in the rail head and generates electrical tension that causes eddy currents. These disturb the magnetic field in such a way that the magnetic force (F) is diverted against the direction of travel. The horizontal component of this magnetic force is used as braking force (FB).

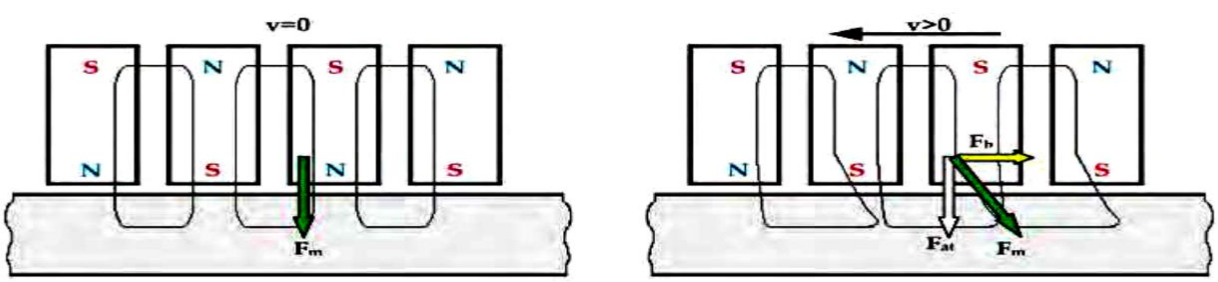


Figure 2.4 :-Eddy current braking system

2.7) Brake Pads Material

To accommodate this, brake friction materials have evolved significantly over the years. They've gone from asbestos to organic to semi-metallic formulations. Each of these materials has proven to have advantages and disadvantages regarding environmental friendliness, wear noise and stopping capability. Asbestos pads caused health issues and organic compounds can't always meet a wide range of braking requirements. Unfortunately the steel strands used in semi-metallic pads to provide strength and conduct heat away from rotors also generate noise and are abrasive enough to increase rotor wear. Since they were first used on a few original equipment applications in 1985, friction materials that contain ceramic formulations have become recognized for their desirable blend of traits. These pads use ceramic compounds and copper fibers in place of the semi-metallic pad's steel fibers [37]. This allows the ceramic pads to handle high brake temperatures with less heat fade, provide faster recovery after the stop, and generate less dust and wear on both the pads and rotors. And from a comfort standpoint, ceramic compounds provide much quieter braking because the ceramic compound helps dampen noise by generating a frequency beyond the human hearing range. Another characteristic that makes ceramic materials attractive is the absence of noticeable dust. All brake pads produce dust as they wear. The ingredients in ceramic compounds produce a light colored dust that is much less noticeable and less likely to stick to the wheels. Consequently, wheels and tires maintain a cleaner appearance longer. Ceramic pads meet or exceed all original equipment standards for durability, stopping distance and noise. According to durability tests, ceramic compounds extend brake life compared to most other semi-metallic and organic materials and outlast other premium pad materials by a significant margin - with no sacrifice in noise control, pad life or braking performance [19].

2.8) Related research about disc brake

Choi and Lee [17] presented a paper by using finite element method to analysis of transient thermo elastic behaviors or effects of disc brakes. The main objective of study is transient analysis for thermo elastic contact problem of disc brakes with frictional heat generation is performed using the finite element method. To analyze the thermo elastic phenomenon occurring in disc brakes, the coupled heat conduction and elastic equations are solved with contact problems. The results of the research are computational results which presented for the distributions of pressure and temperature on each friction surface between the contacting bodies.

.**Talati and Jalalifar[11]** Presented a paper on analysis of heat conduction in a disc brake system analytically by using greens function approach method. The objective or the problem of the study is analysis the rate of heat conduction and governing heat equation for the disc and the pad are extracted in the form of transient equations, taken into account, parameters such as the duration of braking, vehicle velocity geometries and the dimensions of the brake components, materials of the disc brake rotor and the pad and contact pressure distribution. The results is concluded that the heat generated due to friction between the disc and the pad should be ideally dissipated in the environment to avoid decreasing the friction coefficient between the disc and the pad and to avoid the temperature rise of various brake components and brake fluid vaporization due to excessive heating.

Naji et al [9] presented a paper on mathematical model to describe the thermal behavior of a brake system which consists of the shoe and the drum. The model is solved analytically using Green's function method for any type of the stopping braking action. The objective of the study is to investigate thermal behavior for three specified braking actions which are the impulse, the unit step and trigonometric stopping actions. Thermal response of disc brake systems to different materials used for the disc–pad couple has been studied by many researches. The results are express the thermal effects of disc brake at different braking action such as impulse, unit step and trigonometric.

2.8) Factors which are develop unequal pressure/force on two opposite side of a disc brake

There are some factors or conditions which advance to develop unequal pressure /uneven pressure on a disc brake pad contact system. Some of them are:

2.5.1) Caliper slides

There are groves located in the calipers that hold the brake pads and slide in as when push on the brake pedal and out when let off but sometimes the brake pad shims get stuck in the groves or just get corroded or debris built up in them [34]. This will cause the pads to not be able to slide in and out correctly.

2.5.2) Corrosion of brake line

The brake line may be affected by correction or rust in the case contaminated fluid in the reservoir due to the vaporized of the brake fluid and also due to mixed of any bad material which comes from outside the fluid reservoir. For example let us say the one side of the pedal piston already damaged by corrosion. Since the piston pedal which is damaged cannot exert equal force or pressure as the normal one.so because of this there will be unbalance/equal force application on the disc brake**.**

2.5.3) Blocked the pipe

The brake pipe/hose is the way which is used to leave the brake fluid or the air in the system. But at this time, the brake line may be covered by any bad material which is inserted from external environment or from internal fluid reservoir due to the contaminated of fluid. Again the brake pedal can't applied equal force/pressure on the brake rotor.

2.5.4) Wear of disc brake

"Metal-to-Metal" contact is a condition where the shoe rubs directly against the rotor when the lining wears down to the metal brake of pad. Its cause severe damage and loss of braking efficiency due to non-uniform pressure distribution on the disc and pad contact. At One point the rotor may show greater wear than the other point . This indicates that the outer pad is continue to presses against the rotor after the caliper has released .The caliper guide pins and bushing may be seized.it is also possible that the pad may be seized in its mounting bracket, due to corrosion[8]

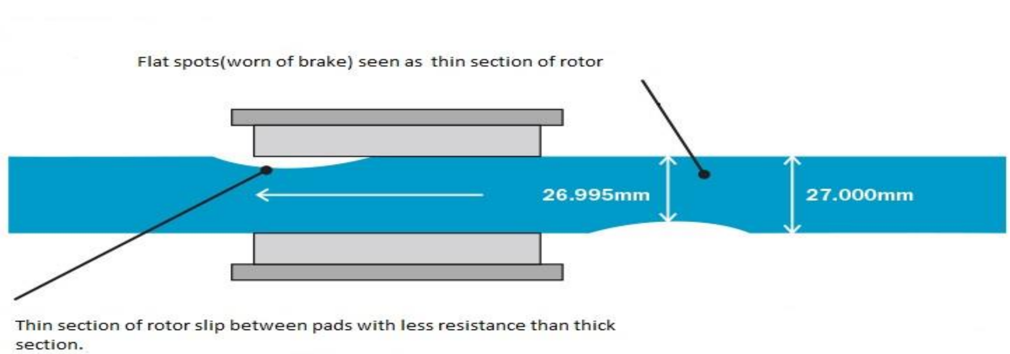


Figure 2.5:- The rotor thickness difference cause of variation clamping force [15]

2.5.5) Failure of piston seal

The piston seal will be fail due to the repeated load of brake application, in the case of lubrication system and also the reaction force applied by the piston. When the brake piston seal became damaged, there is will be the leakage around the piston area because of this there is an inadequate brake pressure in the abnormal side and does not have equal application of force or pressure on the disc rotor [45].

2.5.6) Inner pad wear

Usually occurs when the piston can't retract properly. The piston may be binding in a scored cylinder. The piston seal may be distorted or just plain worn-out. Both of this indicate the caliper teardown and rebuild

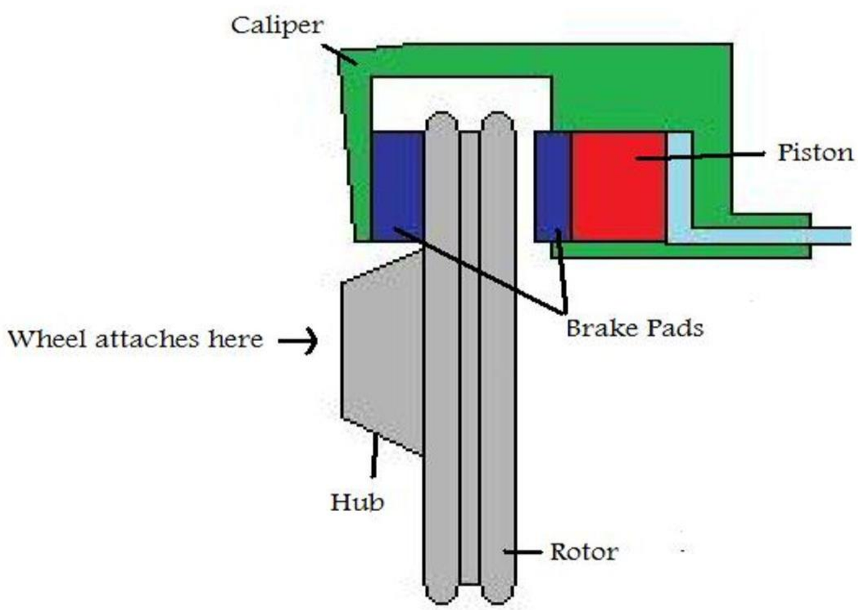


Figure 2.6:- The inner pad wear that cause unequal clamping pressure [42]

2.5.7) Outer pad wear:

Outer pad will wear prematurely, if the caliper bracket/the caliper pin are corroded or if they are lubricated with the wrong lubricant.

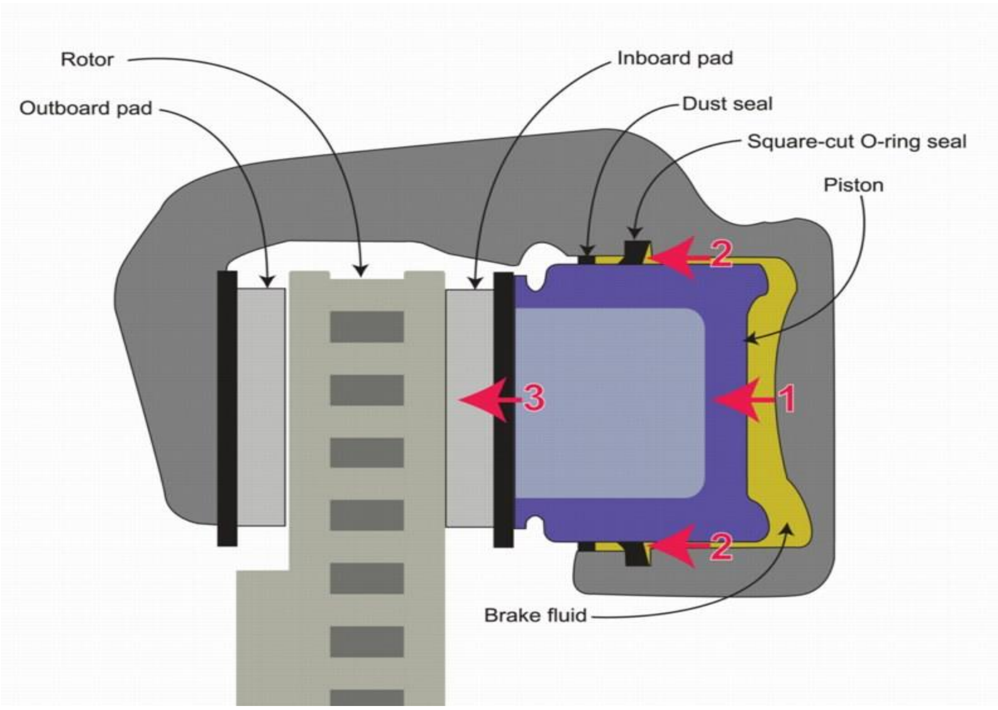


Figure 2.7:- The outer pad wear that cause of clamping pressure variation [43]

2.5.8) Both pads on one side thinner than pads on opposite side

This generally caused by a hydraulic problem, also may be caused by a sticky piston. More often the cause is a extraction in the brake hose on the opposite side. The side with premature wear may also have a brake hose with internal damage that acts just like check valve, preventing the release of the brake fluid. It is also possible that there is a hydraulic restriction higher up line than the brake hose on the side with the pad wear. For example, Abs modulator may not allow the release/return of pressure on that side. The other possible is air in the hydraulic line on the side opposite the pad wear [12].

2.5.9) The rotor thickness difference

The difference in the rotor thickness causes the variation of clamp force/pressure that cause also the variation of torque/rotation

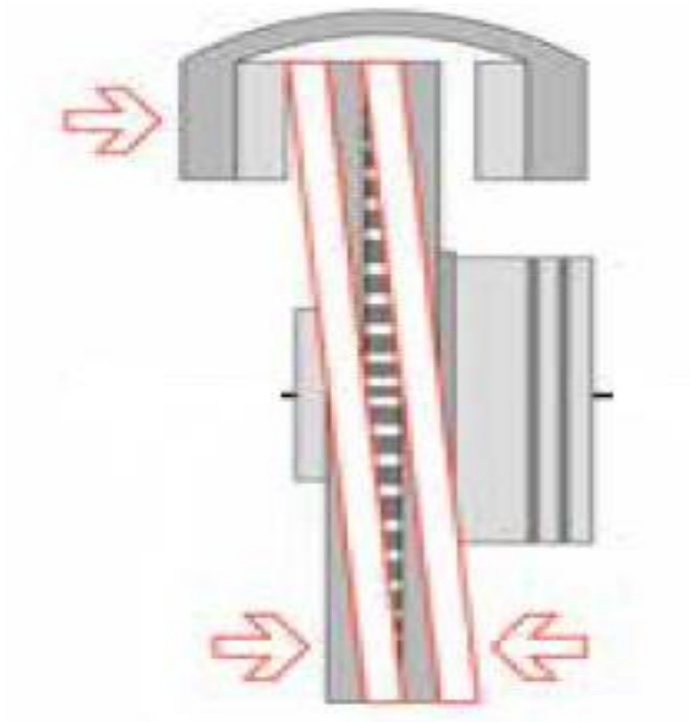


Figure 2.6:- Variation of disc thickness cause of unequal clamping pressure [46]

CHAPTER THREE

Numerical Modeling to analysis the thermo–mechanical effect of disc brake

3.1 Theoretical Approach to Railway Vehicle Disc Braking Condition

The railway disc brake is affected by external or internal factors such as temperature, stress, deformation, von-miss stress, heat flux and also heat flow which limit the working capacity of the disc brake and service life. In this paper will consider the thermal and mechanical property in order to analysis the effect of unequal pressure which is applied on opposite sides of the disc due to the case which is explain in chapter two.

And also on the disc brake, at the braking condition, mainly there are thermal and mechanical effects applied on the disc rotor due to temperature distribution and surface pressure which is applied by the pedal. Due to this reasoning, in the process of braking of railway vehicles, it is necessary to define the model for thermal analysis that describes the heat transfer of the heat generation by friction at surfaces which are in contact between a railway disc and braking pads through the disc and pads as well as heat outflow of the whole braking system due to cooling effect of the surrounding air and the mechanical stress that describes the deformation of disc rotor and pad. For those purposes, an analytical model for analyzing thermal and mechanical stress effects during braking systems of railway vehicles is utilized and its adopted procedure is presented in this research [47].

3.2) Finite Element Formulation of the Disc Brake

Transient heat conduction in three dimensional heat transfer problem is governed by the following differential equation.

$$\left(\frac{\partial qx}{\partial x} + \frac{\partial qy}{\partial y} + \frac{\partial qz}{\partial z}\right) + \mathrm{Q} \quad = \quad \rho C_P \frac{\partial T}{\partial t} ---------------------- 3.1$$

Where:

q_x, q_y and q_z are conduction heat fluxes in x, y and z-directions, respectively, $c_p$is the specific heat, ρ is the specific mass, Q is internal heat generation rate per unit volume and T is the temperature that varies with the coordinates as well as the time t. The conduction heat fluxes can be written in the form of temperature using Fourier's law. Assuming constant and uniform thermal properties, the relations are:

$$q_x = k_x \frac{\partial T}{\partial x}, \qquad q_y = k_y \frac{\partial T}{\partial y}, \qquad q_z = k_z \frac{\partial T}{\partial z} ---------------3.2$$

Where:

k_x, k_y and k_z are thermal conductivity in x, y and z-directions, respectively. Heat transfer boundary conditions consist of several heat transfer modes that can be written in different forms. The boundary conditions frequently encountered are as follows [15].

$$T_S = T_1(X, Y, Z, T) ---------------------------3.3$$
$$q_s = h(T_S - T_{amb}) ---------------------------3.4$$

Where:

T_1 is the specified surface temperature; q_s the specified surface heat flux (positive into a surface);

h the convective heat transfer coefficient; T_s the unknown surface temperature, and

T the convective exchange temperature.

So now using the formulation of finite element for the disc, ANSYS thermal transient simulation will follows using the following equation.

$$[\mathrm{c}]\left\{\frac{\partial T}{\partial x}\right\} + [K]\{T\} = \{Q\} ----------------------3.5$$

3.3) Determination of braking forces and coefficient of friction

3.3.1 Braking force

The braking force is an important which affected by the friction coefficients determined between rotor-pad contact, and it dependence on different parameters having important role on braking characteristics of the railway vehicle. There are many factors determining the value of braking force, among them the most important proved to be the running speed, the wear condition, the clamping forces, the surface contact pressure and temperature [41].

Usually the main problem is that friction interface between the brake pads and disc must not more than temperatures of 800 $°_C$ rather than severe, sudden wear and non-uniform distribution of pressure on the disc occurs. Usually there are two types of mounting method which are axle mounted wheel set mounted. Normally, thermal analysis determines the necessity of several brake discs mounted on the wheel set, even for in the case of high speed vehicles due to vibration and uniform load distribution [18]. In order to analysis the thermo-mechanical property of disc brake the braking force is essential.

3.3.2) Coefficient of friction

Coefficient of friction is a value that shows the relationship between two objects and normal force between the objects. It is a value that used in physics sometimes to find an objects normal force or frictional force. The ability or capacity to slow down the rail way vehicle disc brake system is depend on the coefficient of friction, the braking distance, the vehicle on speed, and the rail gradient[31]. Depend on these parameters the coefficient of friction can be determined by using the following formula.

$$\mu = \frac{v^2}{2gs_b 3.2^2} \text{------------------------} 3.6$$

Where:

$s_b =$ Braking distance (m)

V = velocity (km/h)

g = acceleration due to gravity (9.81 m/s)

μ = coefficient of friction

Normally, the coefficient of friction and braking speed are inversely proportional, in that it increases during braking as speed decreases. In determining the coefficient of friction, a mean value is therefore applied for the given speed.

It is also worthwhile to see how friction coefficients vary with sliding speeds and Brake - line pressures. Logically as the friction coefficient decreases as sliding speed increases.

3.4) Railway Vehicle Disc Braking and Angular Velocity

Braking is defined as an application of the brake that results in a braking force which is being applied to the vehicle. Brake force is the force which applied to the brake disc / pad / braking surface contact or interface .

The railway vehicle running with initial velocity (v_o) is consider to stand still with constant deceleration (a). Depend on this concept,its linear translational velocity as function of time (t) is calculated by[23]

$$v(t) = v_o - at \text{ ---------------------------- } 3.7$$

$$\omega(t) = \frac{v(t)}{r_w} \text{ ---------------------------- } 3.8$$

$$\omega_o = \frac{v(o)}{r_w} \text{ ----------------------------- } 3.9$$

Where

ω_o = initial angular velocity of the wheel

ω = angular velocity of the wheel at any time

v(t) = velocity of the railway vehicle

$v_{o=}$ = Initial running velocity of the railway vehicle

r_w= the radus of railway vehicle wheel

During the application 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 in terms of the translational velocity v (t) of the railway vehicle as follows[25].

$$v_{disc} = \frac{v(t) r_{disc}}{r_w} \text{ -------------------------- } 3.10$$

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

$$s_b = t_b . v_o - \frac{a}{2} (t_{b)})^2 \text{ ------------------------- } 3.11$$

$$t_b = \frac{vo}{a} \text{ ------------------------------- } 3.12$$

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

$$\omega = \omega_o \left(1 - \frac{t}{t_b}\right) ------------------------3.13$$

Where:

t_b = brake time to stop the railway vehicle.

s_b = total brake distance

3.5) Numerical analysis of braking energy

A braking energy is an energy which used to slow dawon the vehicle speed as well as to became rest by disppitating the heat into the brake disc and pad as well as the envronment at the braking condition. The enrgy which loses during braking condition is almost equal to the heat which disspitates into the pad and disec as well as environment. There will be different type of energy systems in rail way vechile at brking stuation such as kinetic ,potential and rotiotional in the cause of motion, elevation and cener of mass respectively[3]. But in the present work, the paper consider the kinetic energy and potential energy because of the rotiotional energy is almost consder under kinetic energy. The initial kinetic energy is the sum of translasional and rotational

$$KE = \frac{M(V_O)^2}{2} + \frac{I(w_O)^2}{2} ---------------------3.14$$

Where:

KE = initial kinetic energy of a vehicle

M = total vehicle mass

I = polar movement of inertia for rotating body

And, the moment of inertia of the rotating parts such as wheel set and discs can be calculated by using the following equation below.

$$I = \sum_{i=1}^{n} mi(ri)^2 -------------------------3.15$$

Where:

M_i. = total mass of wheel set and disc

r_i = the radius distance from the center mass.

In this condition, all rotating parts are fixed on the wheel set axle and the rotation axis could be taken as the tangent line joining the contact point of the rail heads with wheel set of the disc. It is parallel to the axis of the wheel axis. In this case the rotating radius is equal to the radius of the wheel (r_{w}). Hence

$$E_K \quad = \frac{M(V_O)^2}{2} + \frac{I(W_O)^2}{2} = \frac{M(V_O)^2}{2}\left[1 + \frac{1}{M(r_w)^2}\right] = \frac{1.1M(V_O)^2}{2} ------------3.16$$

In some sources or literature the effect of the rotating mass of the wheel set and the disc is taken to be 10% the tire weight of the axle load of the railway vehicle. The term $[\frac{1}{M{r_w}^2}]$ refers to the rotational mass and its value is equal to 0.1. The potential energy of the rail way vehicle is depend on its track gradient and the distance traveled $\delta[\frac{mm}{m}]$, s_b respectively [1]. According to some literature, for simplification to calculate the external forces to railway application take sin α = tan α. Therefore the potential energy of the railway vehicle will be:

$$P_E = Mgs_b \frac{\delta}{1000} ---------------------------- 3.17$$

Where: g = acceleration due to gravity =9.81 m/s^2

Therefore, the total braking energy of railway vehicle can be determined in the following equation

$$E_B = 1.1M\frac{(V_O)^2}{2} + Mgs_b \frac{\sigma}{1000} ------------------- 3.18$$

3.6) Heat to be dissipated during braking

The energy absorbed by the brake and transformed into heat must be dissipated to the surrounding air in order to avoid excessive temperature rise of the brake lining. The temperature rise depends upon the mass of the brake disc, the braking time and the heat dissipation capacity of the brake [36].

Since, the energy absorbed (or heat generated) and the rate of wear of the brake lining at a particular speed are depend on the applied pressure between the braking surface. Therefore it is an important considering in the design of brakes. The permissible normal pressure between the brake surface depends upon the energy is to be absorbed.

3.7) Heat energy

In braking system, the mechanical energy is transformed into heat energy. This energy is characterized by a total heating of the disc and the pads during the braking phase. The energy dissipated in the form of heat can generate rise in temperature ranging from 300℃ to 800℃. The heat quantity in contact area is the result of plastic micro-deformation generated by the friction forces between disc rotor and brake pad. The total heat generated (Q_{gen}) in the brake system equals with the total mechanical energy (Q_{gen}) lost from the vehicle [10]. Hence, considering the energy balance;

$$E_b = Q_{gen} \text{ --------------------------------- } 3.19$$

3.8) Heat conduction

There are two paths of heat conduction from the discs, one through the bearing assembly (which should be avoided) and another through the wheel carrier, which is the major conductive path. The heat flow can be estimated by Fourier's law of heat conduction as follows:

$$Q_{cond} = -KA\frac{dt}{dx} \text{ ----------------------------- } 3.20$$

The small area A and very low temperature difference limits the amount of power dissipated by conduction. Therefore, the heat conduction can become negligible in brakes.

3.9) Convection heat transfer

The major aim of designing brake discs is to improve the convection dissipation of disc braking systems. In operations of braking systems, convection is the most important mode of heat transfer, dissipating the highest proportion of heat to surrounding air. The current research focuses on heat convection of disc rotors. Convection is related to the heat flux by used of Newton's law of cooling [14].

$$\frac{q}{A} = h\,(T_s - T_f) \text{ -- } 3.21$$

Where:

$\frac{q}{A}$ = is heat flux out of the face

h= film coefficient

T_s= temperature of the face

T_f =bulk fluid temperature

3.10) Work of Friction Force

Due to the low of conservation of energy when the vehicle begin to come slow down its velocity and already become to rest by transfer of frictional or mechanical energy into the heat energy to control the motion of the vehicle properly. i e When the vehicle 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 ($2A_b$) swept by the brake pads on both sides. Two friction forces ($2F_{disc}$) act on both faces of the disc at an effective radius (r_{disc}).

The total flux is equal to:

$$\text{Total heat flux} = \frac{Q_{gen}}{2A_b} = \frac{E_b}{2A_b} ------------------3.22$$

Where:

A_b = the area swept by the brake pad

Table 1 AA LRT N-S ROUT Stations

No	**stations**	**gradient**	**sinδ**	**Station type**
1	**NS1**	**0.473**	**0.0082**	**Under ground**
2	**NS2**	**-0.682**	**-0.6303**	**Ground**
3	**NS3**	**0.934**	**0.0163**	**Ground**
4	**NS4**	**0.457**	**0.008**	**Elevated**
5	**NS5**	**-1.27**	**-0.0223**	**Ground**
6	**NS6**	**-0.645**	**-0.6012**	**Elevated**
7	**NS7**	**-0.765**	**-0.6925**	**Elevated**
8	**NS8**	**0.850**	**0.7513**	**Elevated (common)**
9	**NS9**	**0.954**	**0.815**	**Elevated (common)**
10	**NS10**	**1.25**	**0.949**	**Elevated (common)**

11	**NS11**	**0.870**	**0.7643**	**Elevated (common)**
12	**NS12**	**0.730**	**0.6670**	**Elevated (common)**
13	**NS13**	**0**	**0**	**Ground**
14	**NS14**	**-0.972**	**-0.8260**	**Ground**
15	**NS15**	**-0.354**	**-0.3466**	**Ground**
16	**NS16**	**0.53**	**0.5055**	**Ground**
17	**NS17**	**0.45**	**0.4350**	**Ground**
18	**NS18**	**0.760**	**0.6889**	**Ground**
19	**NS19**	**0.460**	**0.4439**	**Ground**
20	**NS20**	**0.432**	**0.4186**	**Ground**
21	**NS21**	**0.123**	**0.1227**	**Ground**
22	**NS22**	**0.323**	**0.3174**	**Ground**

Table 2 AA LRT N-S distance between consecutive stations

No	**AA LRT NS route station name**	**AA LRT route station name**	**Station distance[KM]**
1	**Minilik square……Atkelt Tera**	**NS1 ……NE2**	**0.743**
2	**Atekelt tera………Gogam Berenda**	**NS2 …….NS3**	**0.945**
3	**Gojam bereda……Autobius Tera**	**NS3 ………NS4**	**0.60488**
4	**Autobius tera…….Sebategna**	**NS4 ……….NS5**	**0.607**
5	**Sebategna…………Abenet**	**NS5 ………NS6**	**0.8127**
6	**Abenet……………..Darimar**	**NS6 ………NS7**	**0.739**
7	**Darimar…………..St.Lideta**	**NS7 ………NS8**	**0.591**
8	**St.lideta……………Tegibarid**	**NS8 ………NS9**	**0.735**
9	**Tegidarid…………. Mexico**	**NS9 ……..NS10**	**0.688**
10	**Mexico……………...Legehar**	**NS10 …….NS11**	**0.560**
11	**Legehar…………….Stadium**	**NS11 ……NS12**	**0.445**
12	**Stadium…………….Mesualekia**	**NS12 …….NS13**	**0.908**

13	**Meshualekia………...Riche**	**NS13 …….NS14**	**0.48112**
14	**Riche ….. …………...Temenja yazh**	**NS14 …….NS15**	**0.610**
15	**Temenjayazh ….. …..Lancha**	**NS15 …..NS16**	**0.555**
16	**Lancha ……………...Nifaseslik 2**	**NS16 ……NS17**	**1.9716**
17	**Nifasesilk 2…………..Nifaseslik 1**	**NS17 ……NS18**	**0.861**
18	**Nifasesilk 1….. ……...Adeye Abeba**	**NS18 …….NS19**	**0.995**
19	**Adeye abeba …………Saris**	**NS19 ……NS20**	**0.535**
20	**Saris ………………….Abo junction**	**NS20 ……NS21**	**0.845**

SOURCE: Addis Ababa LRT (NS and EW) project from ERC, September 2009

3.11) Heat flux entering the disc

In the case of disc brake, the effective friction processes between the pads and the disc are extremely complex due to the fact that the present time brake pads, due to their composite structure, do not have constant chemical-physic proprieties, the organic contained elements being subject of a series of transformations under the influence of temperature increase. The heat distribution between the brake disc and the friction pads is mostly dependent on material characteristics, among whom a major influence is due to the density ρ(d p) [kg/m3], the thermal conductivity k(d p) [W/m.K] and the specific heat C(d,p) [J/kg.K] of the disc's (index d) and braking pad's(index p) materials respectively. Denoting Qd and Qp [J] the heat quantities assumed by the disc and the braking pads respectively, one could be expressed in to the following manner [26]

$$\frac{Q_d}{Q_p} = \frac{\sqrt{\rho_d k_d c_d}}{\sqrt{\rho_p k_p c_p}} ---------------------- 3.23$$

The brake disc assumes the most part of the heat, usually about 95% , through the effective contact surface of the friction coupling. Considering the complexity of the problem and average data processing limited, one replaced the pads by their effect, represented by an entering heat flux. The thermal analysis of the braking system of railway vehicles requires determination of the quantity of heat produced by friction, as well as the distribution of this energy between the railway disc and the braking pads. The disc material is gray cast iron (FG15) with high carbon content, with good thermo physical characteristics and the brake pad has an isotropic elastic behavior whose thermo mechanical characteristics adapted in this simulation given in table below. Generally, the thermal conductivity of material of the brake pads is smaller than that of disc ($k_p < k_d$). We consider the energy from brake application is converted into heat and transferred to the disc and pad approximately 95% and 5% respectively [20]. This ratio normally is called the proportion of heat transferred to disc ($\gamma = 0.95$). The rate of heat generation is:

$$\boldsymbol{Q_d(t) = \gamma(2F_{disc})V_{disc} = \quad 0.95(2F_{disc})\frac{r_d}{r_w}(v_o - at) ---------- 3.24}$$

Table 3 Data for heat flux calculating

Input parameter	**value**
Mass of the vehicle-M[kg]	64,000
Mass of the disc m(kg)	62.2
Starting velocity –Vo[km/hr]	20
Deceleration – a[m/s^2]	1.5, for emergency brake
Braking time - t_b[s]	14 ,for emergency brake
Effective radius of the braking disc -r_d[m]	0.112
Radius of the wheel - r_w[m]	0.330
Incline of the track –δ[%°]	11
Friction coefficient disc/pad –μ [/]	0.2
Surface of the brake pad -A_p [cm^2]	200

The coefficient of friction has significant effects on the form of the thermal field as well as on temperature distribution. The proper definition of the coefficient of friction is very important for perception of temperature field qualities for analyzing thermal effects in braking systems at railway discs. The value of the coefficient of friction depends on a number of tribological parameters such as speed of relative movement of parts in contact, the contact surface roughness, quality of materials of a tribo-pair, temperature of contact parts and so on However, in this research did not consider tribological parameters of braking processes. According to DIN EN ISO1183 the value μ = 0.2 of the friction coefficient for materials of the disc and the pad for railway vehicle disc is taken as the recommended value **[2].**

3.12) Load application

On the railway vehicle (train) there are different types of load application. For this analysis and to simplify the task used two types load application:

a) Braking of the vehicle running with a maximum velocity to standstill on a straight track with the temperature of the surrounding is 25°C.

The assumed temperature is average environmental temperature of the operating route of the railway vehicle in the LRT region. The figure bellow referees the application force on Addis Ababa LRT train with maximum velocity (70 km/hr.) to standstill on a flat track (**MESHUALEKIYA TO RICHE, NS13 to NS14** at the distance of 610m). There are 14 steps, 1 second for each step for emergency braking [table 1].

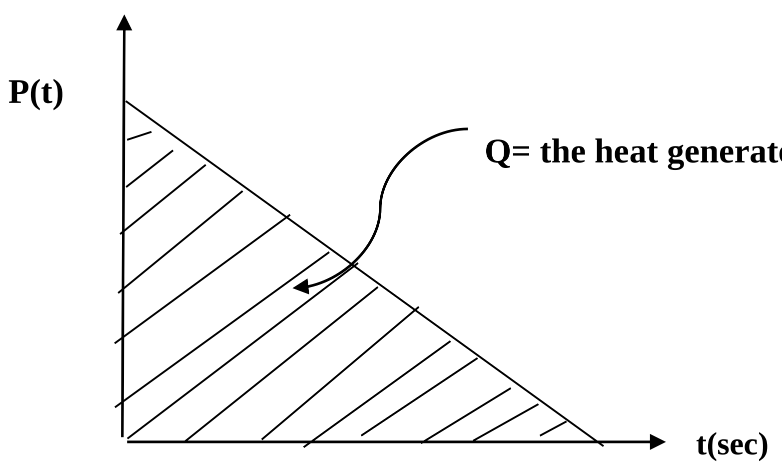


Figure 3.1:- *braking load on straight track*

b) Braking by the maximum speed on downhill track for maintaining constant speed and braking to standstill afterwards

The following figure describe the application of braking force at Addis Ababa LRT train to maintain a constant velocity which covers 555 meters and braking from maximum velocity on a downhill track from **LANCHA TO NIFASILK NS16 TO NS17** at the distance of 1971.66meters. There are 14 steps for emergency brakes [table 2].

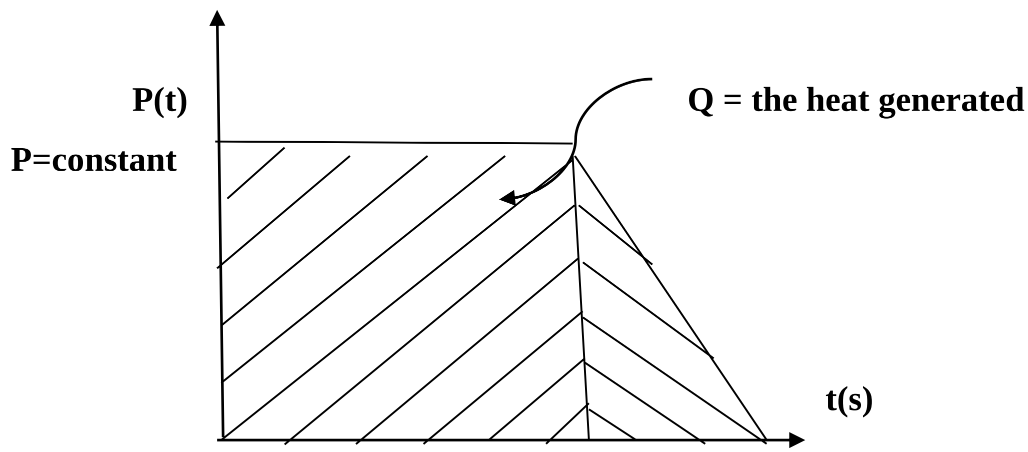


Figure 3.2:- braking load on a downhill track for maintain constant speed and afterwards

The total work of friction during the full braking cycle must be equal with the total heat which dissipated or generated

$$\boldsymbol{Q_{gen} = E_b = \int_0^{t_b} p(t)dt} \text{ --------------------------- } \mathbf{4.25}$$

$$\boldsymbol{\frac{(1.1)M(v_o)^2}{2} + Mgs_b\frac{\delta}{1000} = 2F_{disc}\int_0^{t_b} v_{disc}(t)dt} \text{ ------------- } \mathbf{4.26}$$

$$\boldsymbol{\frac{(1.1)M(v_o)^2}{2} + Mgs_b\frac{\delta}{1000} = 2F_{disc}\frac{r_{disc}}{r_{wheel}}\left(t_b v_o - \frac{a(t_b)^2}{2}\right)} \text{ -------- } \mathbf{4.27}$$

Therefore: the required braking force or the friction force of the vehicle which is generated on the disc will be

$$\boldsymbol{F_{disc}} = \frac{\boldsymbol{(1.1)M(v_o)^2 + Mgs_b\frac{\delta}{1000}}}{\boldsymbol{n_b 2(\frac{r_{disc}}{r_{wheel}}\left[t_b v_o - a\frac{(t_b)^2}{2}\right])}} - - - - - - - - - - - - \boldsymbol{4.28}$$

Where:

n_b = number of disc brake

3.13) Centrifugal force

Centrifugal force can be increased by increasing either (1) the speed of rotation, (2) the mass of the body, or (3) the radius, which is the distance of the body from the center of the curve. Increasing either the mass or the radius increases the centrifugal force proportionally, but increasing the speed of rotation increases it in proportion to the square of the speed; that is, an increase in speed of 10 times, say from 10 to 100 revolutions per minute, increases the centrifugal force by a factor of 100. The disc rotating on the rail imposes the centrifugal force on its own due to its weight. Due to the centrifugal load the pressure/stress is applied to the surface of the disc width which is dependent on the density of the disk, rotational speed of the disk and the radius of the rotating disc. According to UIC s 1002 and other railway standard parameters the radius of the disc is selected to be 300 mm and density of gray cast iron is 7250 kg/m^3 and the rotation speed of the disc is calculated from the speed of the vehicle in relation to the radius of the disc.The Chinese standard is almost relate with UIC standard Therefore the centrifugal force/effect will be as follows:

$$F_C = Rm_{disc}\omega^2 - 3.29$$

Where:

F_C = Centrifugal force of the disc

m_{disc} = the density of disc brake

ω = the rotational speed of the disc

R = the radius of the disc

3.14) Temperature for next braking action

When the train moves along the track detects due to different phenomena which affects the disc temperature. The temperature drop of a brake disc or wheel occurs due to the effect of station stopping time of the train, accelerated and constant traveling time of the train. Therefore for the next braking action the initial temperature of the disc or wheel will rise above from its ambient temperature and will be computed according to Newton's law of cooling.

$$T = T_{amb} + \left[(T_{max} - T_{amb}) * exp\left\{\frac{hA}{mc} * t\right\}\right] - - - - - - - - - - - - 3.30$$

Where

T=initial temperature

Tmax=maximum temperature

T_{amb} =ambient temperature

A= disc brake contact area

h= heat transfer coefficient

t= delay time, station stopping, accelerated and constant travel time

M= mass of disc

C= specific heat capacity of grey cast iron

3.15) Maximum temperature for the single braking action

The heat/ thermal capacity of disc brake is depend on the temperature which rise during braking action. The heat which leave into the outside or output of the model is composed of radiation and convection. But almost the heat which decapitate due to the convection method incase the radiation is almost negligible The temperature drop of a brake rotor occurs due to the effect of station stopping time of the passengers train, accelerated and constant traveling time of the train. Therefore for the maximum temperature for next braking action the initial temperature of the disc will rise above from its ambient temperature [42].

3.16) Load and weight transfer during Braking

On application of brakes on the vehicle, inertia forces are set up. The braking force is applied at lesser height whereas the vehicle load is at a higher height, resulting a couple to act on the body of the vehicle. This results an additional loading on leading bogie as compared to trailing bogie. Weight transfer and load transfer are two expressions used somewhat confusingly to describe two distinct effects the change in load borne by different wheels of even perfectly rigid vehicles during acceleration, and the change in center of mass (CoM) location relative to the wheels because of suspension compliance or cargo shifting or sloshing.[5]

When railway vehicle is standing still on straight track, its weight is uniformly distributed on all bogies. On application of brakes on the vehicle, inertia forces are set up. The braking force is applied at lesser height whereas the vehicle load is at a higher height. This creates a couple to act on the body of the vehicle which ultimately resulting an additional loading on leading bogie as compared to trailing bogie. Therefore the ratio of weight distribution between the front and rear bogie is considered to be 60/40 in favor of the front bogie [32].

3.17) Dependence of weight transfer

From the expression above, it can be concluded that weight transfer depends on

- Centre of gravity of the wagon. Higher the center of gravity more will be its effect. This clearly implies higher weight transfer on empty wagon.
- Greater the distance between bogie centers, less the effect of weight transfer
- Weight transfer effect is reduced by adoption of low traction bars which reduces the value of height.

3.18) Thermo- elastic properties of disc brake

In this analysis the disc material is gray cast iron (GF15) with high carbon content, with good thermo physical characteristics and the brake pad has an isotropic elastic behavior whose thermo-mechanical characteristics adopted in the analysis of the two parts. There are three reasons why rotors are made of the cast iron[30]

- It is relatively hard and resists wear.
- It is cheaper than steel or aluminum.
- It absorbs and dissipates heat well and helps in cooling of the brakes.

Table 4 thermo-physical properties of disc and pad material

Material properties (value)	disc	pad
Thermal conductivity k (w/m.℃)	57	5
Density ,$\rho(\frac{kg}{m^3})$	7250	1400
Specific heat, c(J/kg.℃)	460	1000
Poisson’s ratio,v	0.28	0.25
Thermal expansion,$\alpha(10^{-6}/℃)$	10.85	10
Elastic modulus(Gpa)	138	1
Coefficient of friction, μ	0.2	0.2
Yield strength [Mpa]	400	320
Tensile strength [Mpa]	430	345
Linear expansion coefficient [1/k]	0.000056	0.0000035

3.19) Determination of the Physical Model

The railway vehicle (train) stop braking on track derives from the physical model for determination of the heat transfer and mechanical deformation are depend on the braking time. The load distribution of the railway vehicle (train) considered is distributed between the front and rear bogies or carriage. Each carriage or bogie consists of two wheel sets. Hence, according to some literature source the whole brake force is applied from the forward part of the carriage [27].

$$\left\{\frac{1}{2}[1.1]M(V_O)^2 + Mgs_b\frac{\delta}{1000}\right\} = \quad (2F_{disc})\int_0^{t_b} v_{disc}(t)dt --------4.31$$

The change of energy is equal to the heat flux on the surface of the disc. The ratio is used to calculate the thermal load on the brake disc.

A) Carrying capacity of the A.A LRT train

Table 5 *Carrying capacity of LRT*

Status	No set	Standing capacity	Total
Rated capacity	65	192person	257person

B) A.A LRT Train load or weight

Table 6 *LRT train weight*

Status	Appr. Weight of train body[t]	Passenger weight[t]	Total[t]
Rated capacity	47.86	16.142	64.2

Source: Addis Ababa (N-S and E-W) rout light rail transit project from ERC, September 2009

3.20) Determination of the disc friction force

According to, A.A LRT Standard, material brake pad and the gray cast iron (GF15) material of the brake disc, with dimension tabulated in table below is used to calculate disc friction force which area swept on the disc by the brake pad. The reason why this material used is because of it low cost, easy of manufacturing, strength and resistance to thermal loading.

Table 7 A.A LRT vehicle specification

parameters	value
Vehicle mass[kg]	64200
Maximum load per axle[kg]	32000
Number of axle per vehicle	2
Number of disc per axle	2
Start velocity [m/s^2]	19.44

Deceleration [m/s^2]	1.5 for emergency
Braking time[s]	14 for emergency
Bolt holes and radius[mm]	($\emptyset^{12*9)}$
Diameter of the wheel r_w [m]	0.66
Outer disc diameter[m]	0.46
Inner disc diameter[m]	0.22
Disc thickness [m]	0.06
Effective radius of the disc brake r_d [m]	0.112
Surface swept by the pad[m^2]	0.02
Friction coefficient by disc/pad	0.2

Source A.A LRT PROJECT FROM NS and EW in 2009

The value of the friction force which work on the brake disc is calculated by inserting the values which are given: Hence the analysis will focus on emergency brake: i.e. at the same constant deceleration rate, braking time but at different load application i.e. On straight track, downhill track and to standstill across the track afterwards.

3.20.1) Braking on the flat/straight track

$$F_{disc} = \frac{0.1 * \frac{1}{2} M(V_O)^2[1.1]}{2\frac{r_{disc}}{r_{wheel}}\{v_o t_b - \frac{1}{2} a(t_b)^2}$$

$$= \frac{0.1\frac{1}{2}64000(20)^2[1.1]}{2\frac{0.112}{0.23}[20*14-\frac{1}{2}1.5*14)^2}$$

$$= \quad 15557.53[N]$$

Hence the instant heat flux entering the disc

$$\frac{\dot{Q}(t)}{2Ab} = \gamma F_{disc} v_{\frac{disc}{2Ab}}$$

$$= \gamma(2F_{disc}) \frac{\frac{r_{disc}}{r_{wheel}}(v_o - at)}{2Ab}$$

$$\dot{Q}(t) = 0.95(15557.53\frac{0.112}{0.23}(20\text{-}1.5*t)$$

$$\dot{Q}\ (t)\ = 100645.0\text{-}\ 815048.375t = 100645w/m^2@t=0$$

This is for equal pressure application on opposite side of the disc.

Heat flux when the unequal pressure application on opposite side of disc.

At this condition will rise the following points.

1) Heat flux is depend on the amount heat generated
2) Heat flux is also depend on the disc friction force
3) Heat flux is also depend on the disc effective area

So all of this condition also depend on the application pressure on opposite side of the disc brake. Consider the above cases calculate the unequal heat flux. Let us take the value of disc friction force on opposite sides have different value due to the variation of the radius of the disc on opposite side of disc.

Now if there is pressure difference on two opposite side, there is also heat generated difference on two opposite side. So heat flux is depend on the amount of heat generated and per unit area**.** The disc friction force will vary due to the disc effective radius on one side. let assume the 30% disc effective radius variation.

$$F_{new} = \frac{0.1 * \frac{1}{2} M(V_O)^2 [1.1]}{2\frac{r_{new}}{r_{wheel}}\{v_o t_b - \frac{1}{2} a(t_b)^2}$$

$$= \frac{0.1\frac{1}{2}64000(20)^2[1.1]}{2\frac{0.0784}{0.23}[20*14-\frac{1}{2}*1.5*(14)^2]}$$

$$= 18347.72[N]$$

Hence the instant heat flux entering the disc

$$\frac{\dot{Q}(t)}{2Ab} = \gamma F_{new} v_{\frac{disc}{2Ab}}$$

$$= \gamma(2F_{disc})\frac{\frac{r_{new}}{r_{wheel}}(v_o - at)}{2Ab}$$

$$\dot{Q}(t)new = 0.95(18347.72\frac{0.0784}{0.23}(20-1.5*t)$$

$$\dot{Q}\ (t)new\ = 1200774 - 125935t = 1200774w/m^2 @t=0$$

The new heat flux will be **1200774w/m^2** on one side. Following the same assumption to downhill track for maintain constant speed and standstill.

3.20.2) Braking downhill track:

A) For maintaining constant velocity

$$Q_{Disc} = 0.95 * \frac{Mgv_o sin\delta}{12}$$

$$= 0.95*\frac{64000*20*0.0.505}{12}$$

$$=502507.44[w]$$

In the case of inclined track the disc friction force will be

$$\boldsymbol{F_{disc}} = \frac{\boldsymbol{Q_{disc*\mu*r_{disc}}}}{\boldsymbol{2v_{o*r_{wheel}}}}$$

$$= \frac{\mathbf{502507*0.2*0.112}}{\mathbf{2*20*0.23}} = 852.74 \text{ [N]}$$

Heat flux= Q_{disc}/effective area

=502507/ 2*0.2474

= 902591 w/m2

This is for equal pressure application on opposite side of the disc.

Heat flux when the unequal pressure application on opposite side of disc.

At this condition will rise the following points.

1 Heat flux is depend on the amount heat generated
2 Heat flux is also depend on the disc friction force
3 Heat flux is also depend on the disc effective area

So all of this condition depend on the application pressure on opposite side of the disc brake. Consider the above cases calculate the unequal heat flux. Let us take the value of disc friction force on opposite sides have different value, in the case of unequal disc effective area.

Let assume 30% generated heat difference on opposite sides of disc due to the disc effective area variation on two opposite side .The new Q_{disc} will be 653259[N].

$$\boldsymbol{F_{new}} = \frac{\boldsymbol{Q_{new*\mu*r_{disc}}}}{\boldsymbol{2v_{o*r_{wheel}}}}$$

$$= \frac{\mathbf{653259*0.2*0.0784}}{\mathbf{2*20*0.23}}$$

$$= 1113.38 \text{ [N]}$$

$\dot{Q}$ **(t) new** $= Q_{new}$**/effective area**

=653259/ 2*0.1731

= 1083109w/m2

B) Braking to across afterwards i.e. after maintaining constant velocity:

$$F_{disc} = \frac{(1.1)M(v_o)^2 + Mgs_b\frac{\delta}{1000}}{n_b 2(\frac{r_{disc}}{r_{wheel}}\left[t_b v_o - a\frac{(t_b)^2}{2}\right])}$$

Friction force:

$$F_{disc} = \frac{(1.1)M(v_o)^2 + Mgs_b\frac{\delta}{1000}}{n_b 2(\frac{r_{disc}}{r_{wheel}}\left[t_b v_o - a\frac{(t_b)^2}{2}\right])} = \mathbf{25937.78[N]}$$

Hence, the instant heat flux enter into the brake disc:

$$Q_d(t) = \gamma(2F_{disc})V_{disc} = \quad 0.95(2F_{disc})\frac{r_d}{r_w}(v_o - at)$$

= 954520-25088.90t[w] =954520w/m^2@t=0

Heat flux when the unequal pressure application on opposite side of disc.

At this condition will rise the following points.

1) Heat flux is depend on the amount heat generated

2) Heat flux is also depend on the disc friction force

3) Heat flux is also depend on the disc effective area

So all of this condition depend on the application pressure on opposite side of the disc brake. Consider the above cases calculate the unequal heat flux. Let us take the value of disc effective area on opposite sides have 30% different value**.**

$$F_{new} = \frac{(1.1)M(v_o)^2 + Mgs_b\frac{\delta}{1000}}{n_b 2(\frac{r_{new}}{r_{wheel}}\left[t_b v_o - a\frac{(t_b)^2}{2}\right])}$$

Friction force

$$F_{new} = \frac{(1.1)64000(20)^2+64000*9.81*133\frac{0.11}{1000}}{12*2\left(\frac{0.0784}{0.23}\left[14*20-1.5\frac{(14)^2}{2}\right]\right)}$$

= 32562 .56 [N]

Hence, the instant heat flux enter into the brake disc:

$$Q_d(t)new = \gamma(2F_{new})V_{disc}$$

$$= \quad 0.95(2F_{new})\frac{r_{new}}{r_w}(v_o - at)$$

$$0.95(2*32562.56)\frac{0.0784}{0.23}(20\text{-}1.5*t)$$

= 1145424-35432.67t[w] =1145424w/m^2@t=0

3.21) Pressure calculation for the brake caliper

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

$$P = \frac{F_{Disc}}{\mu A_P}$$

3.21.1) For braking on straight track

Note: the analysis of each braking condition will make by considering equal and unequal pressure in both sides of disc brake which applied due to caliper failure or other brake lining materials.

$$P=\frac{F_{disc}}{\mu A_p}= 18795 \text{ [pa]}$$

3.21.2) Braking on downhill track:

A) For maintaining constant velocity:

$$P=\frac{F_{disc}}{\mu A_p}= 21318\text{[pa]}$$

B) For braking to standstill afterwards i.e. that after maintaining constant velocity:

$$P = \frac{F_{Disc}}{\mu A_P} = \frac{19829.04}{0.2*200*10^{\wedge -4}} = 68000[pa]$$

3.21) Numerical modeling for unequal pressure applied on disc brake on two opposite side of disc brake

Friction torque is calculated usually on the basis of two assumptions. Each assumptions lead to a different value of Torque. In one case it is assumed that the intensity of pressure on the surface is constant, where as in the second case it is the uniform wearing of the disc and pad surface. Under the first assumption, pressure is assumed to be uniform over the surface area. For Uniform Wear over an area, the intensity of pressure should vary inversely proportional to the elementary area, that is it should decrease with the increase of the elementary area and vice versa..

a) Uniform pressure

p = constant, New brakes

When the two surface have perfect contact the pressure p is uniform over the entire surface. The intensity of pressure become

$$P = \frac{F_{disc}}{\mu\pi({r_2}^2 - {r_1}^2)} ------------------------- 4.32$$

But when the two surface have imperfect contact p is non uniform or not equal over the entire two opposite surface. And the intensity of pressure is depend on the disc outer and inner radius of disc brake i.e. when the outer radius decrease and the inner radius increase in one side of disc due to high pressure concentration on a coincide area . So the intensity of pressure is greater than with the normal pressure.

$$p_{new} = \frac{F_{new}}{\mu\pi({r_2}'^2 - {r_1}^2)} ------------------------- 4.33$$

If there is not uniform pressure distribution on the entire surface, the maximum pressure applied on near the outer radius area. Hence the outer radius became wear out and became decreasing the length.

b) Uniform wear

pr = constant, worn brakes.

The constant-wear rate Rw is assumed to be proportional to the product of pressure p and velocity V, Rw= PV= constant. And the velocity at any point on the face of the disc is V = rxω. Combining these equation, assuming a constant angular velocity ω,

pr = constant = C.

The largest pressure p_{max} must then occur at the smallest radius ri,C= p_{max} ri Hence pressure at any point in the contact region p = p_{max}ri / r

But when there is non uniform wear rate the maximum pressure not occur on the smallest radius. Because it various from surface to surface and the pressure depend on the ratio of inner and outer radius.

$$p_{new} = p_{max}(\frac{r'_1}{r}) ------------------------ 4.34$$

Hence the outer radius decrease and the inner radius increase when the wear rate is not uniform.

Therefore: - p_{new} is greater than p due to non-uniform wear and non-uniform pressure distribution on the two opposite side of disc brake.

The applied pressure on the disc has contribution for heat generated and the heat flux. Because heat flux mean the heat generated per unit area.

So depend on these two cases calculate the new pressure value by inserting the new disc friction force which calculated from above.

A) For straight track

Assume the outer radius decrease with 15% from the original one

$$p_{new} = \frac{Fnew}{\mu({r_2}'^2 - {r_1}^2)} = \frac{18347.72}{\mu\pi(0.184^2 - 0.11^2)}$$

= 22853[pa]

B) For downhill track to maintain constant velocity

Assume the inner radius of the disc worn i.e. its radius increase with 15%

$$p_{new} = \frac{Fnew}{\mu({r_2}^2 - {r_1}'^2)} = \frac{1113.38}{\mu\pi(0.23^2 - 0.1265^2)}$$

=265634[pa]

C) For downhill track after maintaining constant velocity

When the is non uniform pressure distribution, either the inner radius increase or the outer radius increase

Let the outer radius decrease with 15%

$$p_{new} = \frac{Fnew}{\mu\pi({r_2\prime}^2 - {r_1}^2)} = \frac{32562.56}{\mu\pi(0.184^2 - 0.11^2)}$$

$$= 1024762[pa]$$

All the above calculated values are an input for finite element analysis for emergency brake.

3.22) Modeling and preparing the 3D model of railway disc brake

It is very difficult to exactly model the brake disc, as there are still investigations going on to find out transient thermal behavior of disc brakes during braking. In this case, to model a complex geometry, some simplifications are always necessary. These simplifications are made, keeping in mind the difficulties involved in the theoretical calculation and the importance of the parameters that are taken and those that are ignored. In modelling, we usually ignore the things of less importance and with little impact on the analysis [39]. The assumptions are always made depending upon the details and accuracy required in the modelling. In order to know the effect of unequal pressure applied on two opposite side of the disc brake, the analysis should consider the two cases which are unequal pressure and equal pressure applied on the opposite side of disc as a function to compare and contrast the result between them.

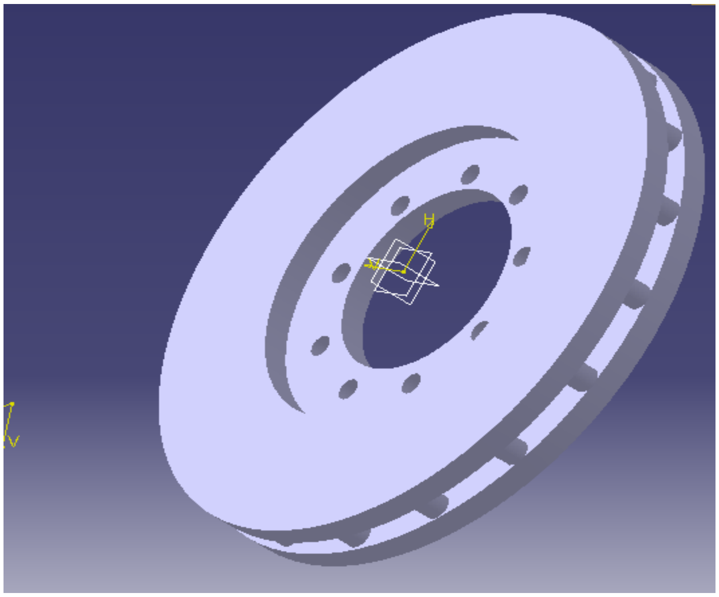


Figure 3.3:- disc brake with CATIA modeling

CHAPTER FOUR

Finite element method for modeling and analyzing the problem

4.1) Analysis the disc braking of LRT using ANSYS 15.0

The analysis is carried out which has three dimensional ventilated disc brake for used to the proper heat dissipation. In the region of temporary contact of the pad and disc, the thermal flux is assigned which is depends on the area of disc and the deceleration rate at different instant of braking time corresponding to the components of the intensity of heat flux product .Transient thermal and mechanical analysis which are indicated the stress, deformation and heat flux are carried out both for braking from maximum velocity to standstill on a straight track and braking in a downhill track to maintain a constant velocity and braking to standstill afterwards for both equal and unequal pressure which is applied on disc brake and for both equal and unequal heat flux on opposite side of disc brake. To simplify the analysis, several assumptions were also been made as follows.

4.1.1) Assumptions

- All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux.
- The heat transfer involved in this analysis takes place only by convection. Heat transfer by conduction and radiation can be neglected as it amounts only to 5 % to 10 % [24]
- The disc material is considered as homogeneous and isotropic.
- The domain is considered as axis-symmetric.
- The disc is stress free before the brake application.
- In this analysis, the ambient temperature and initial temperature is set to 25.
- All other possible disc brake loads are neglected.
- Only certain parts of disc brake rotor experience convection heat transfer such as the cooling vanes area, the outer ring diameter area and the disc brake surface.
- Uniform pressure distribution generated by the brake pad onto the disc brake surface is considered on one side.
- Assume there is the uniform centrifugal force around the disc during rotation.

4.3.2) Specification of the Loads and Boundary Conditions

- After complete the finite element model, should apply initial conditions and loads on the model.
- The initial stage temperature of the disc and the pads is 25 °C.
- The surface convection condition is applied at a surfaces of the disc except the friction contact surface area of the disc and the pad
- The heat flux which is inter into the brake disc during braking action can be calculated by the above formula described and also consider as unbalance heat applied in opposite sides of the disc.

  a) for the straight track : **1000645 – 815048t** when equal heat flux and **12000774-125935t** four unequal heat flux applied

  b) for the downhill track for maintaining constant velocity : **902591w/**m2 both sides equal and 1083109 w/m2 for unequal heat flux on two sides.

  c) for the downhill track braking to standstill after maintaining constant velocity is: **934518.76-75088.90t** both sides equal and 1145424-35432.67t for unequal heat flux.
- The pressure which is applied on the contact surfaces of the rotor and brake pad for three cases of brake pressure applications.

  a) The value of pressure applied which for braking on a straight track is **18795 pa** to equal pressure **and 22853 pa** for unequal pressure.

b) The value of pressure applied which for braking on a downhill track to maintaining the constant velocity is **21318 pa** to equal and **265634 pa** for unequal**.**

c) The value of pressure for applied braking on downhill track to standstill after maintaining a constant velocity is 68000pa to equal pressure and **1024762pa** unequal pressure on two sides.

- The angular velocity of the disc rotor is given ***72.54***rad/s**.**
- Translational acceleration of the rotor disc is *1.5 m/s2 for emergency braking*
- Fixed support is applied at the holes where the bolts are located

4.3.4) Geometrical model

The three dimensional model of ventilated disc brake which has single surface of its symmetry in axial direction and also it is insulated owing nature of considered phenomenon of heating remover. The disc is fastened on to the hub to fixed support in order to protect the movement of disc in tangential direction. There are nine holes to fix the disc with the hub with eighteen diameter bolt. Fig 5.1 describes the assembled geometry of the disc geometrical model.

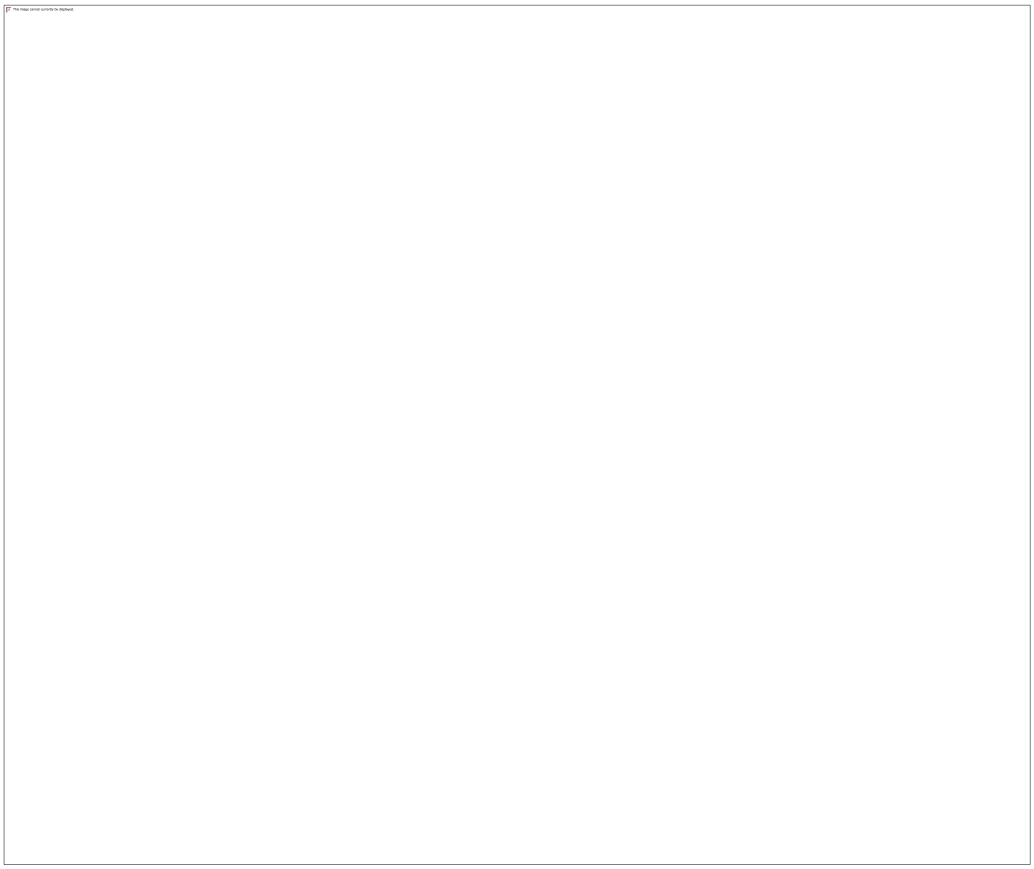


Figure 4.1:- Geometrical model of disc brake

4.3.5) Geometrical mesh

In ANSYS workbench mesh generation is used to divide the geometry into the simplest area as much as possible and used to get the correct solution relatively. There are deferent types of meshing method in the ANSYS work bench such as method, sizing reinforcement etc. For this paper used fine face sizing mesh and it has 0.001 element size to divide the geometry in good condition

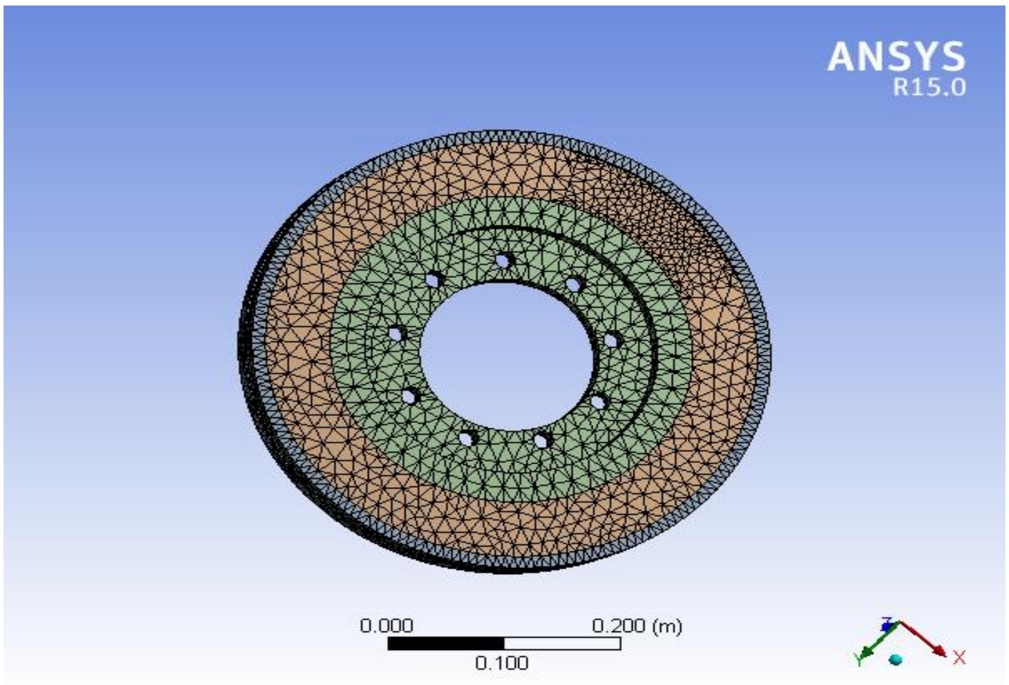


Figure 4.2:- geometrical mesh of disc brake

4.5) Specification the applied loads on two opposite sides of disc brake

According to load specification on the disc brake which is applied due to the caliper pressure and heat dissipation there are two most common loads which are specified as follows. In this paper consider two load such as unbalance heat flux and unequal pressure as explained in chapter two which are mentioned the few factors used to develop just like that condition**.**

4.4.1) Thermal load when equal heat flux is applied on two opposite side of disc

a) Thermal load applied when the vehicle run on stright track at a maximim velocity

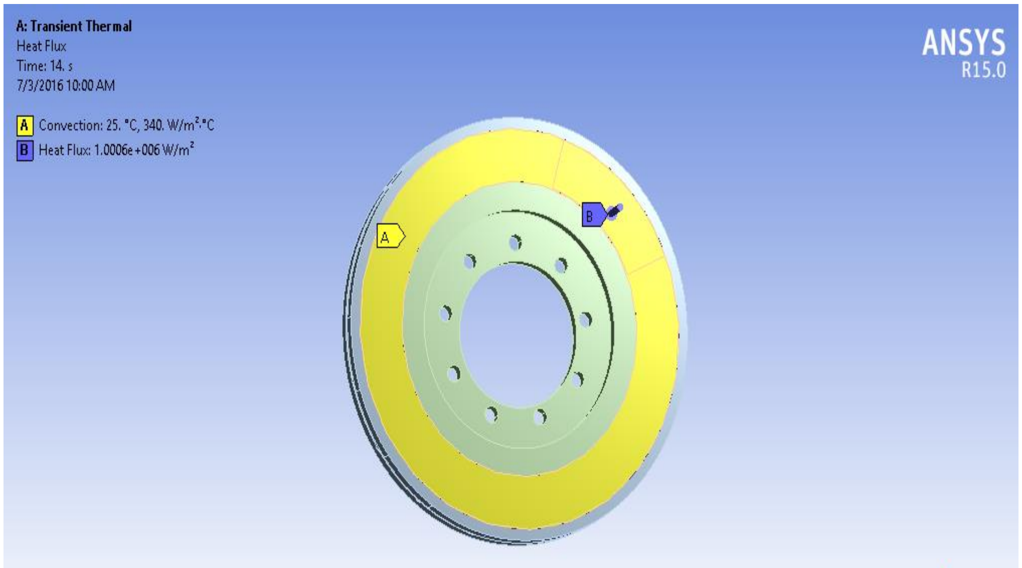


Figure 5.3:- thermal heat flux and convection cofficent on stright track when equal heat flux applied

b)Thermal load application on a downhill track for maintaning a constant velocity

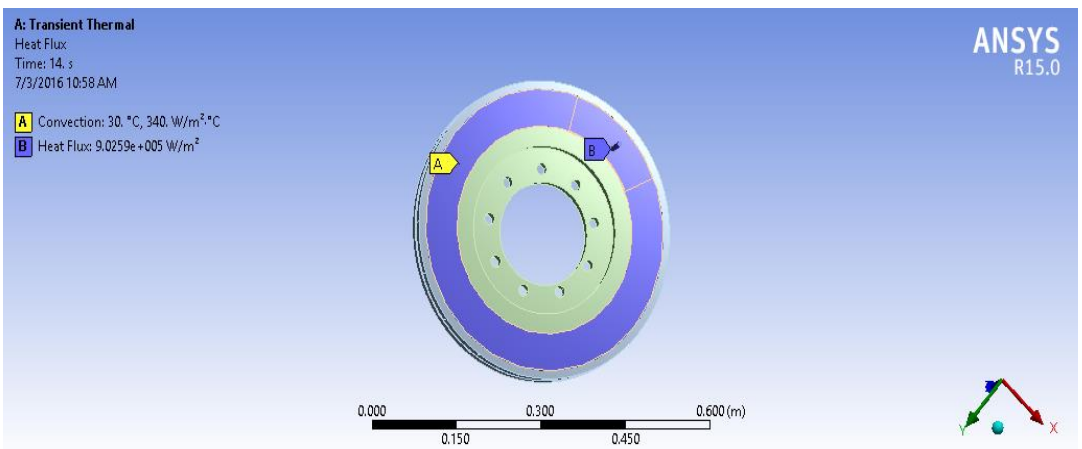


Figure 5.4:- thermal load heat flux and convection cofficent on a daownhill track when equal heat flux applied

4.4.2) Thermal load : when unequal heat flux applied in opposite sides

a) Unequal heat flux applied on stright track braking at a maximim velocity

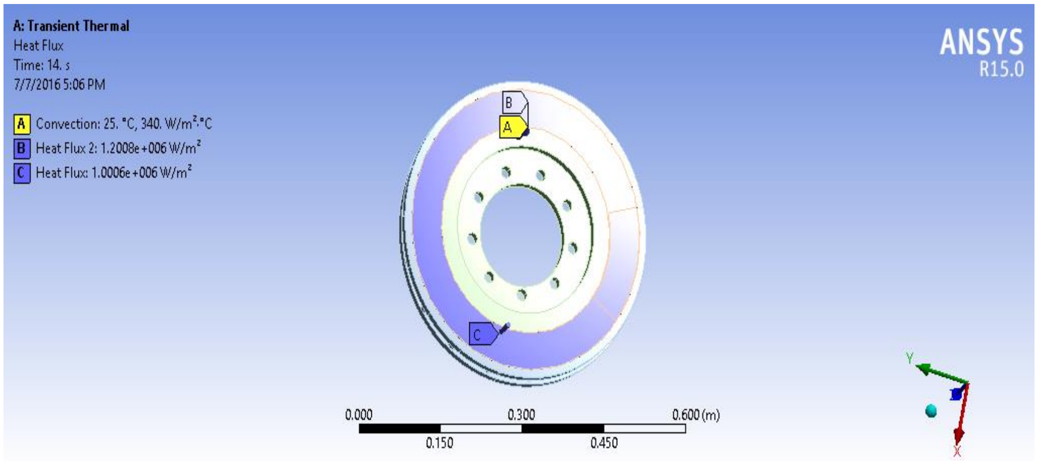


Figure 5.5:- unequal thermal heat flux and cofficent of convection on strsight track

b) Unequal thermal load application on a downhill track for maintaning a constant velocity

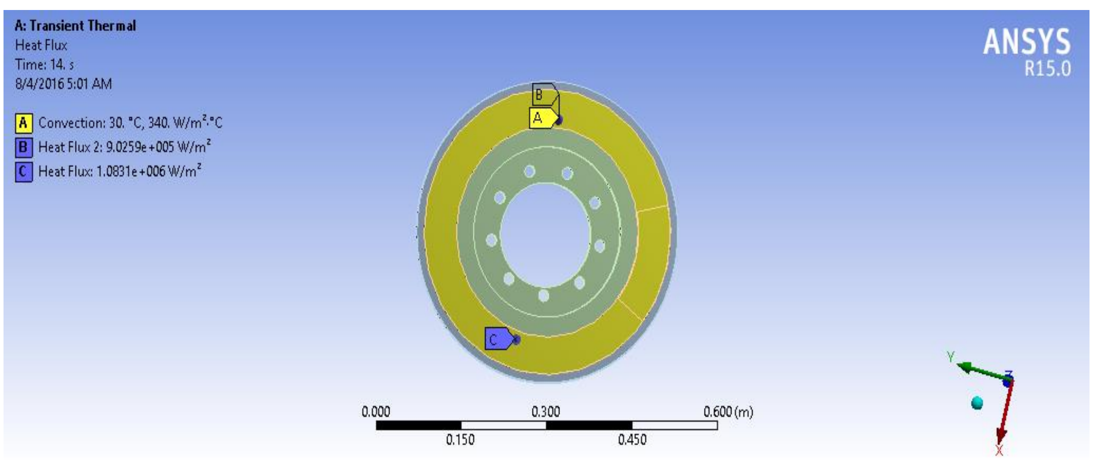


Figure 5.6:- unequal thermal load heat flux and convection cofficent on a daownhill track

4.4.3) Mechanical loads: when equal pressure applied on two opposite sides of disc

a) Pressure application when run at maximum velocity on stright track:

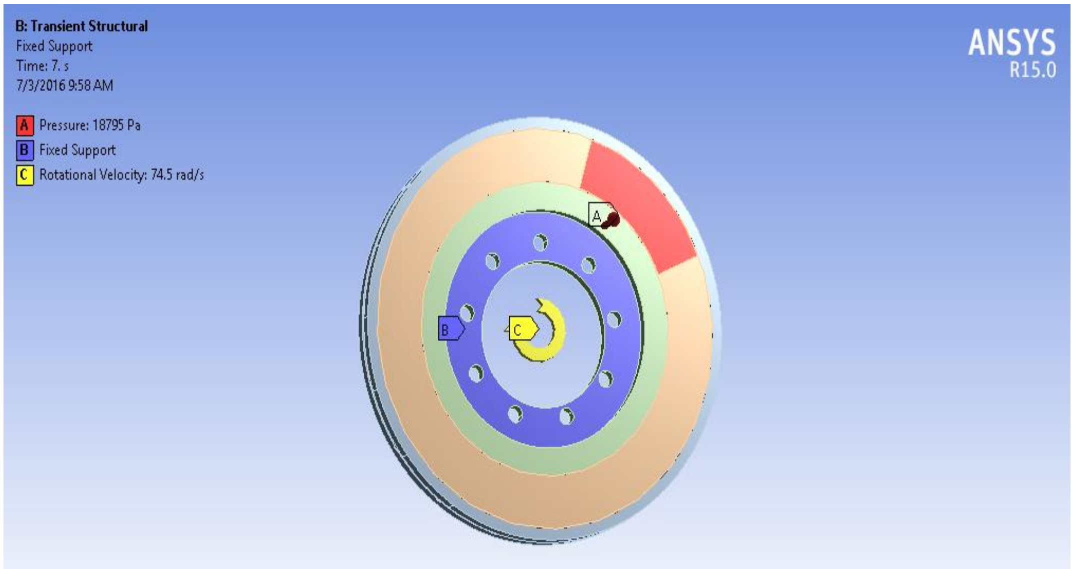


Figure 5.7:- pressure applied on the disc effective area on stright track

b) Pressure applied on adownhill track for mantaining a constant speed

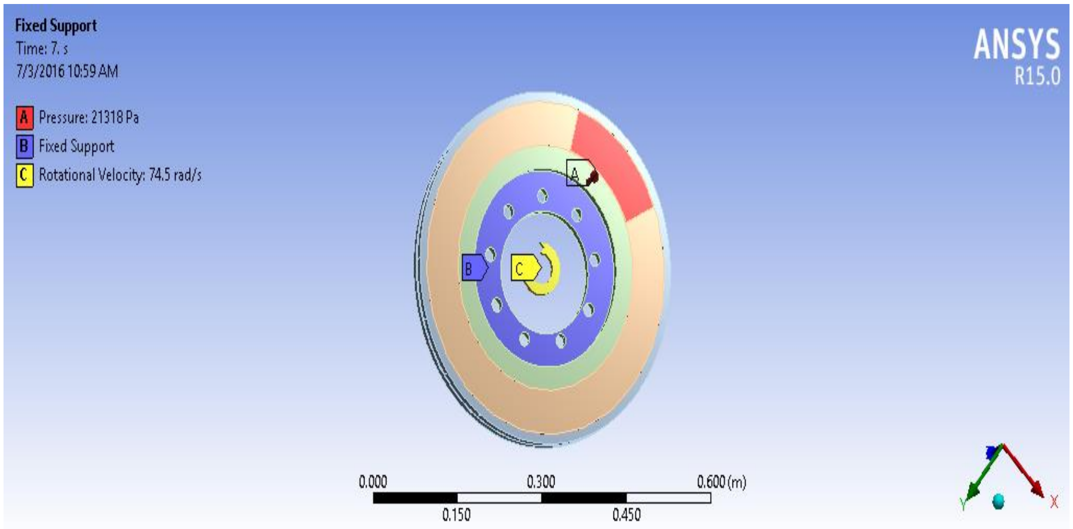


Figure 5.8:-pressure acting on the disc effective area on a downhill track

4.4.4) Mechanical load :when unequal pressure application on two opposite sides

a) Unequal pressure application on stright track at maximum velocity

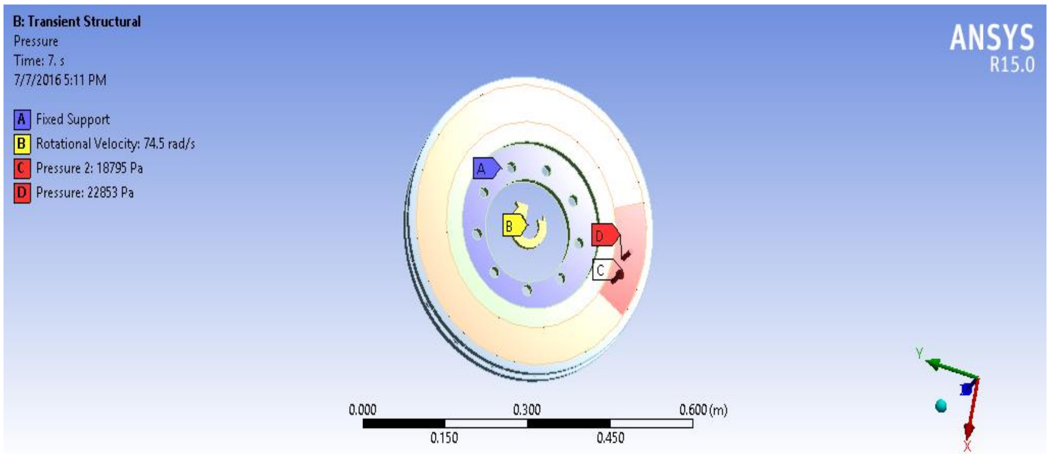


Figure 5.9:- unequal pressure acting on disc effective area

b) unequal presure application on a downhill track for maintaing the constant velocity

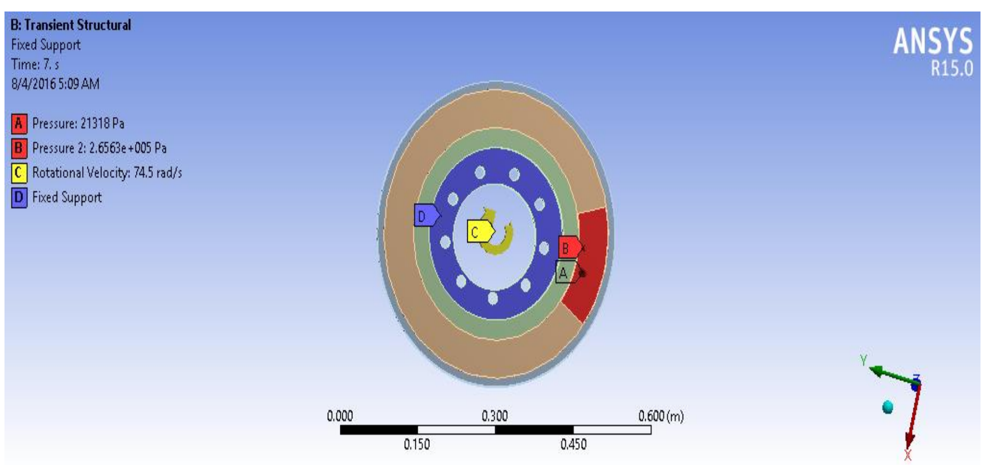


Figure 5.10:- unequal pressure acting on the disc effective area on downhill track

4.4.5) Impotred body temperature: when equal heat flux applied on two opposite sides of disc

It is found from the thermal analysis during braking with maximum velocity to accrose on a stright track, to acrose on the downhill track for maintaining a constant velocity and to stand still on a downhill track after maintaining a constant velocity and the imported value is used as an input for structural analysis. The figures and the procudre are listed bellow

The procudre to import the body temperature

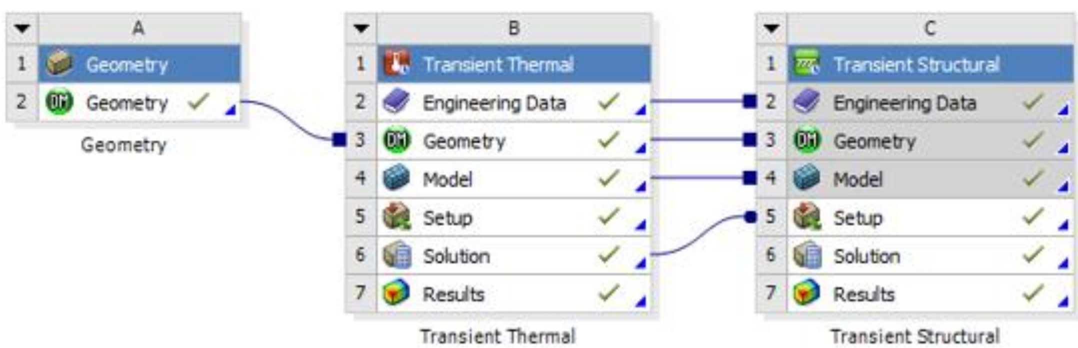
1) Select the disc effective area from the whole modeling

2) Done the trasient thermal analysis

3) Coupling the trasient into structural analysis

4) Done the structural analysis from the thermal input analysis and import the body temperature from importing load.

This is the general procuder for the coupling analysis of thermo- mechanical behaviour of disc brake



a) Imported body temprature when the train run on downhill track to maintain constant velocity.

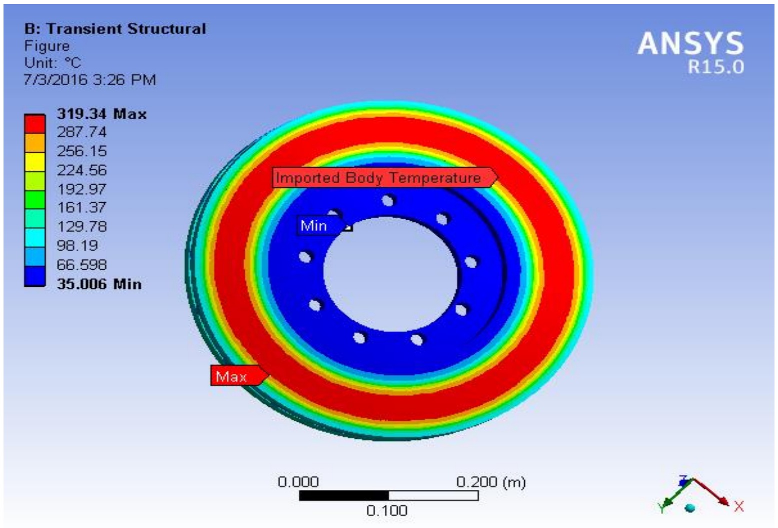


Figure 4.11:- imported body tempreture on downhill track

4.4.6) Imported body temperature: when unequal heat flux applied on opposite side

a) Imported body temprature on downhill track to maintain constant velocity

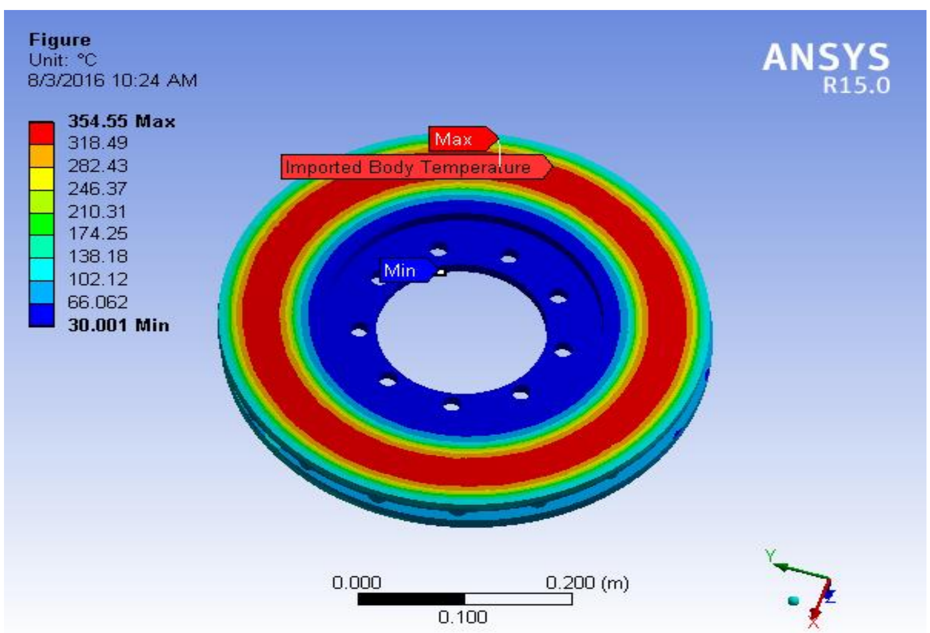


Figure 4.12:-imported body thempreture on downhill when unequal heat flux

CHAPTER FIVE

5.1) Results and Discussion

In the case of the application of pressure and heat on the disc brake rotor, the rotor would be exposed for different effects such as heat disipition,due to friction on the disc effective surface area,tempreture and also the deformation which is happened in the case of repeation pressure application on the disc.Normaly the braking condtion is more sensetive phenomenea for the life service of brake piston seal, caliper and pad. Because of this, the coupled thermo-mechanical analysis shouid be performed for the stability of the disc structure. As menshioned in chapter two the analysing consider the two case which are equal and unequal heat flux and pressure application respectively.

At the end of check up the boundary condition, model and input parameters specification, the analysing is performed for the two main case of brake application area.

1) brake application on the stright track (**from MESHUALEKYA TO RICHE ,NS11 TO NS12**) the braking time to stop the train takes 14 seconds,t_b = 14 seconds
2) Brake application for down hill track(**from LANCHA,NS TO NIFASILK NS 116 TO NS17**) for maintaing a constant velocity and brake application on daownhill track after maintaining a constant veloity. The braking time take t_b =28 seconds for each take 14 seconds to stop for maintaing a constant velocity and to stop after maintaining a constant velocity.

5.2) Results

In this analysis there are two commom result which are most known as thermal and mechanical and also has its own effect such as tempreature , thermal heat flux, stress(von miss) and deformation.The anlaysis is combined termo-mechanical results because of the immported temperature is one of the factor which increase present used as an input for structural analysis.

5 .2.1) Thermal results

5.2.1.1) Thermal results when equal heat flux applied on opposite sides of disc on stright track at maximum velostiy.

Table 8 *ANSYS analysis results for gray cast iron FG15 for stright track*

parameters	minimum	maximum	Time(seconds)
Tempreture(℃)	25.06	323.08	14
Total Heat Flux (w/m^2)	0.63022	1.8544e^6	14

5.1.2.2) Thermal results when unequal heat flux applied on opposite sides of disc brake on stright track at maximum velocity.

Table 9 *ansys results for stright track when unequal heat flux application*

parameters	minimum	maximum	Time(seconds)
Tempreture(℃)	25.001	385.14	14
Total Heat Flux (w/m^2)	0.29908	2.2312e^6	14

Temperature gradient:

It is a dimensional quantity which is expressed interms of degree and also it is aphysical quantity that discribes in which direction and at what rate the temprature changes in the most rapidly speed around a specific space.Generally the temperature gradient refers to the change of temperature with displacement in a given direction from agiven reference point. As the result show from the above two tables,there are different values of temperature have seen when equal and unequal heat flux application on opposite sides of disc rotor on stright track at maximum velocity. As we have seen from the result, when unequal heat applied on the opposite side of the rotor the temperature is greater than with equal heat flux applied on opposite side of disc.

The highest allowed tempreture of the disc is 300-800 ℃ during the long braking condition and the highest tempreture rigions are consider the disc effective area as shown bellow. i.e the temprature will drop along the direction which is in the median plane of the disc.

a)The maximum tempreture at the end of braking time when equal heat flux applied on opposite sides of disc rotor.

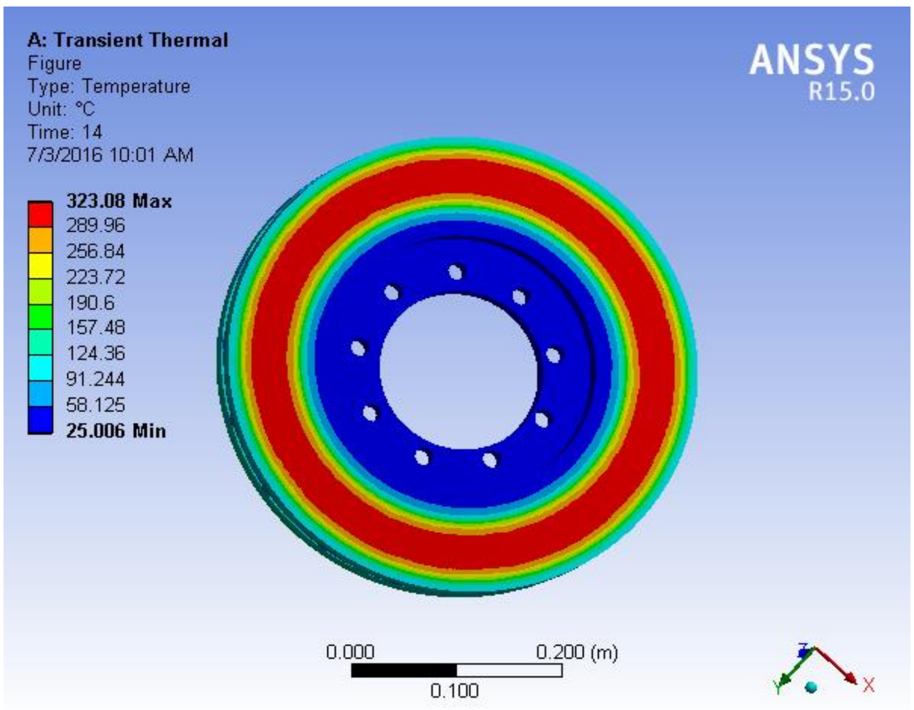


Figure 5.1:- Temperature value of disc brake when equal heat flux on straight track

b) The temperature value at the end of braking time when unequal heat flux

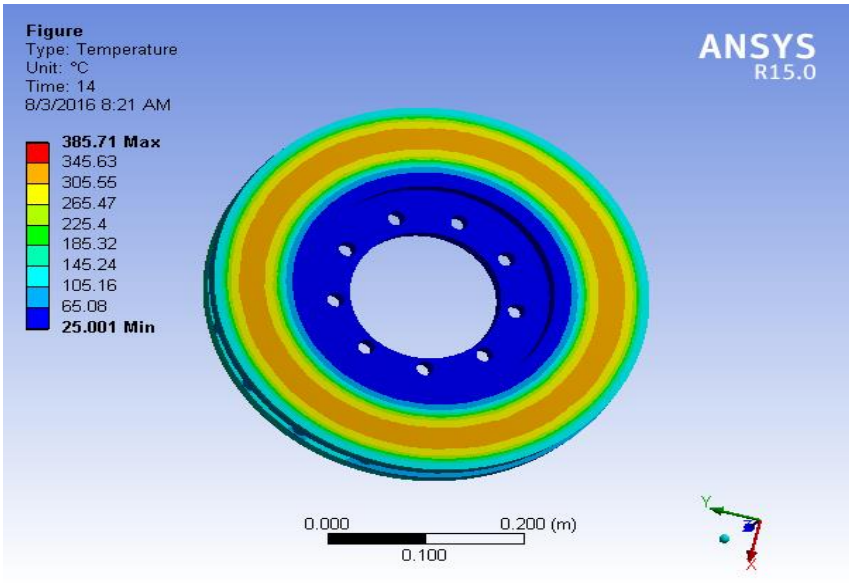


Figure 5.2:- Temperature gradient when equal heat flux on straight track

Thermal heat flux: The rate of heat distribution per unit area

a) The maximum heat flux at the end of braking when equal heat flux applied on opposite side of disc on straight track

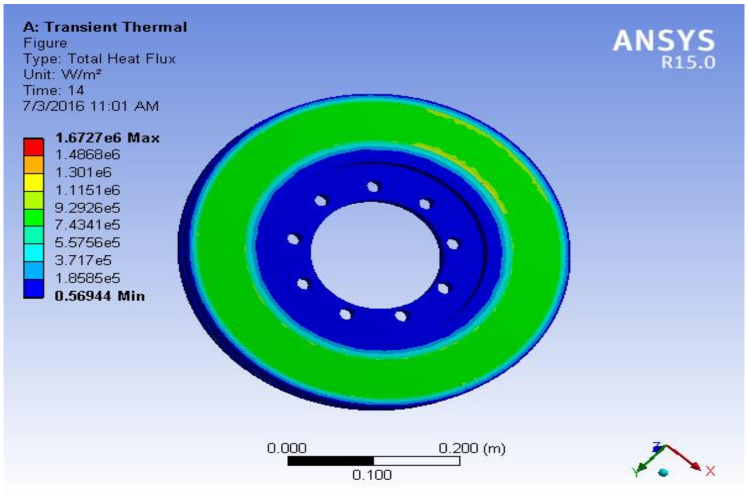


Figure 5.3:-Heat flux at the end when equal heat applied

b) The maximum heat flux at the end of braking when unequal heat flux applied on opposite sides on straight track

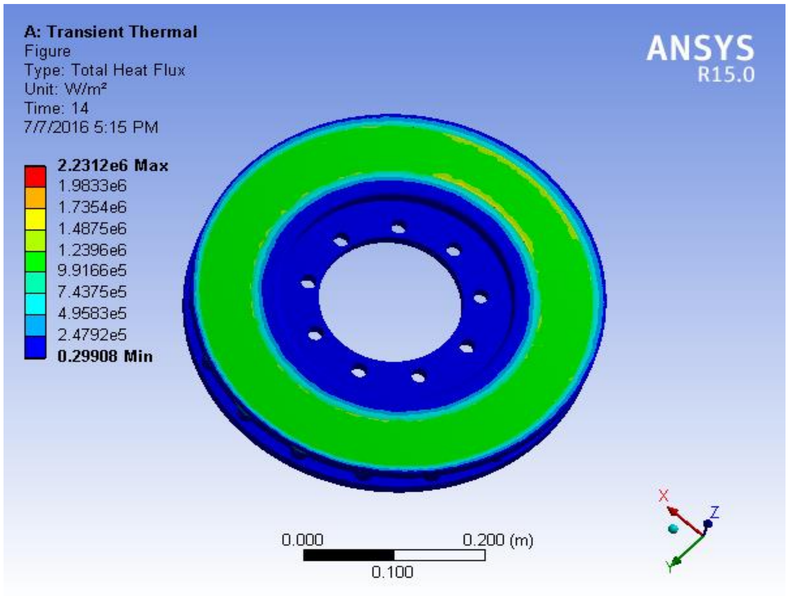


Figure 5.4:- heat flux at the end when unequal heat applied on straight track

5.2.1.3) Braking on a downhill track to maintaining a constant velocity

a) Braking on downhill track for maintaining constant velocity when equal heat flux application on opposite sides of the brake disc

Table 10 ANSYS analysis results of FG15 gray cast iron when heat flux on downhill equal

Parameters	Minimum	maximum	Time (second)
Temperature(℃)	30.006	298.87	14
Total heat flux(w/m2)	0.56944	$1.67727e^{6}$	14

Table 11 ANSYS results of FG15 gray cast iron when unequal heat flux on downhill track

parameters	minimum	maximum	Time(second)
Temperature(℃)	**30.001**	**354.55**	**14**
Total heat flux(w/m2)	**0.22435**	**$1.9947e^6$**	**14**

Thermal gradient value: It is a dimensional quantity which describes the rate of heat distribution at what direction and in what way of mechanisms in specific area or location

a)The maximum temperature at the end of braking time when equal heat flux on downhill

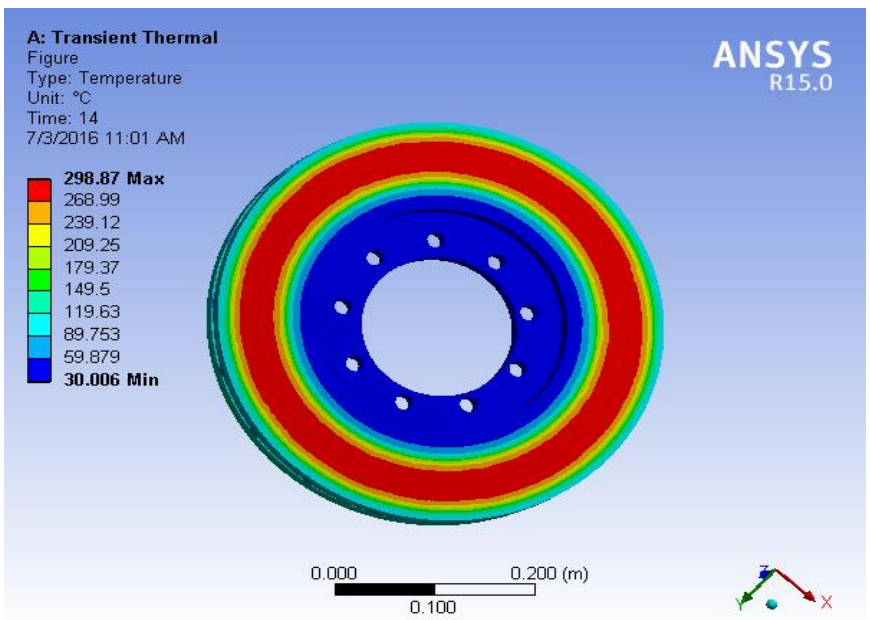


Figure 5.5:- temperature value at end when equal heat flux

b) The maximum temperature at the end of braking time when unequal heat flux

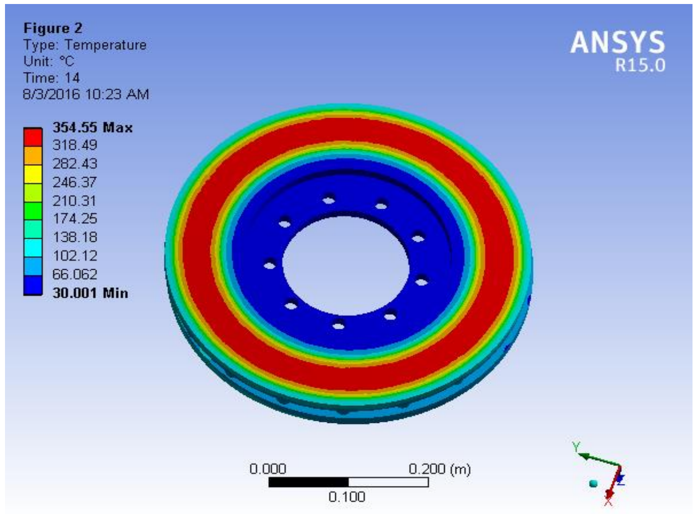


Figure 5.6:-Temperature value at the end when unequal heat flux

Heat flux: The heat distribution per unit area

a) **The maximum heat flux at the end of braking time during balance heat flux**

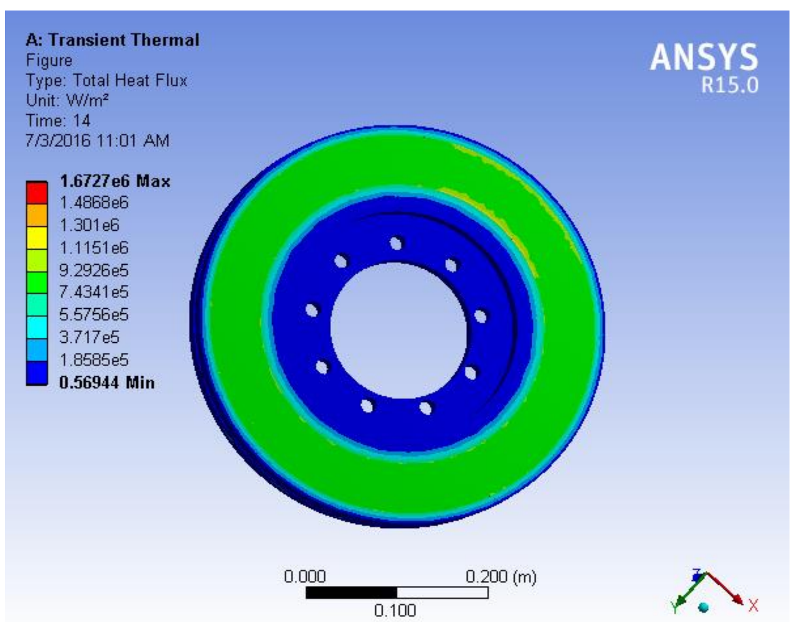


Figure 5.7:-Heat flux when equal heat applied

b) The maximum heat flux at the end of braking time when unequal heat flux applied on opposite sides of disc

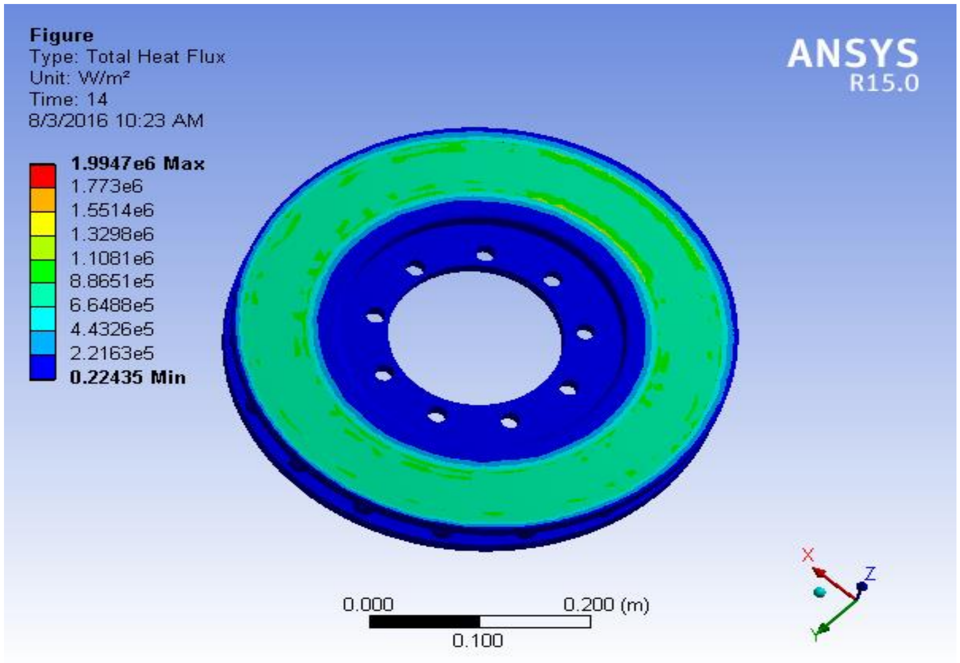


Figure 5.8:- Heat flux when unequal heat applied

5.2.1.4) Braking on downhill track after maintaining constant velocity

a) Braking when heat flux equal applied on opposite side of disc

Table 12 ANSYS analysis results when equal heat flux after maintaining constant velocity

parameters	Minimum	maximum	Time (second)
Temperature (℃)	**35**	**319.34**	**14**
Total heat flux (w/m2)	**0.60264**	**$1.7689e^{6}$**	**14**

b) Braking when unequal heat flux application on opposite sides of disc

Table 13 ANSYS analysis results when unequal heat flux after maintain constant velocity

Parameter	Minimum	Maximum	Time (Seconds)
temperature (℃)	**35.001**	**378.23**	**14**
Total heat flux(w/m2)	**0.24561**	**2.1094e^{6}**	**14**

As we have seen from the above table the effect of heat flux when the train moves on the downhill track after maintaining constant there is a large variation on temperature gradient when the disc exposed the equal and unequal heat flux on opposite sides of disc. Normally can be conclude that at every load application such as straight track, downhill track for maintaining constant velocity and standstill after maintaining constant velocity there is a great temperature difference**.**

Thermal gradient value:

It is the rate of heat distribution per unit area at specific time and at what direction the rate of heat variation with place to place.

5.2.2) Mechanical results

5.2.2.1) Braking from a maximum velocity to across on straight track

a) Braking from a maximum velocity to run on straight track when equal pressure applied on opposite sides of disc

Table 14 analysis results when equal pressure applied on straight track

Parameters	minimum	maximum	Time (second)
Total deformation (m)	**0**	**0.00025458**	**7**
Von-miss stress (pa)	$\mathbf{1.6673e^{6}}$	$\mathbf{3.1693e^{8}}$	**7**

b) Braking from maximum velocity to run on straight track when unequal pressure applied on opposite sides of disc

Table 15 **analysis results when unequal pressure applied on straight track**

parameters	minimum	maximum	Time(second)
Total deformation (m)	**0**	**0.00034732**	**7**
Von-miss stress (pa)	$\mathbf{2.1821e^6}$	$\mathbf{3.0657e^8}$	**7**

Deformation:-It refers to any change in shape or size of an object due to an applied force or pressure which the deformation energy in this case is transferred through work or due to a change temperature which the deformation energy in this case is transferred through heat. Depending on the type of material, size and the force applied the various types of deformation may result but this paper consider the most common types such as total deformation.

a)Total deformation:

1) The maximum total deformation at the end of braking time when equal pressure applied on opposite side of disc**.**

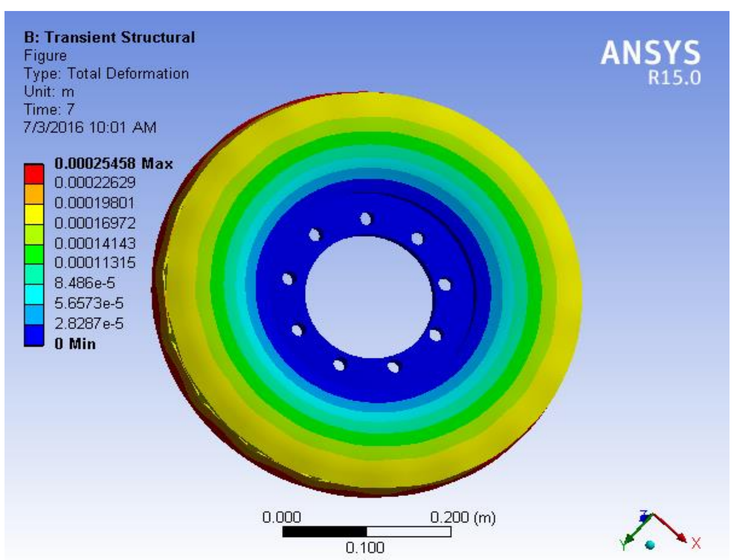


Figure 5.9 total deformation when equal pressure applied for straight track

2) The maximum total deformation when unequal pressure applied on opposite sides of disc

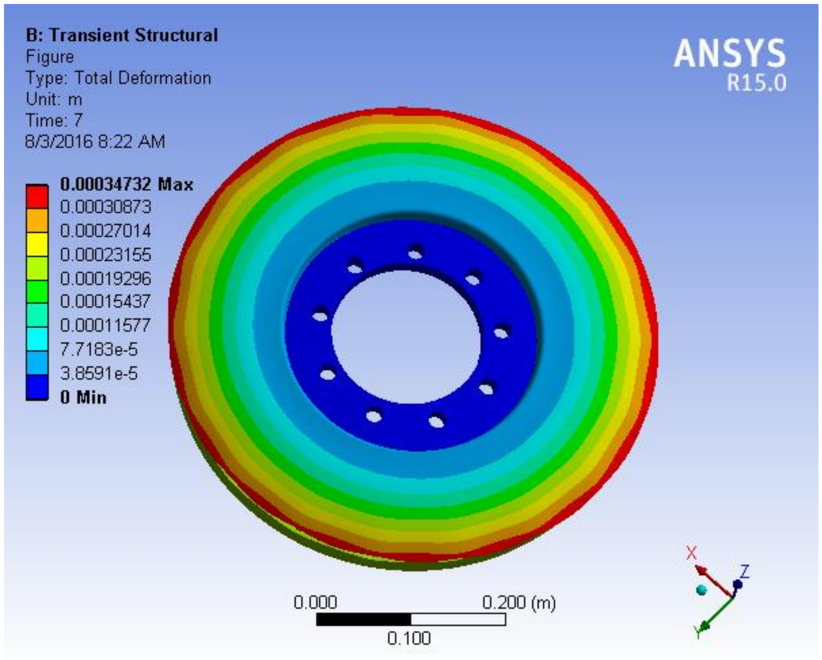


Figure 5.10:- total deformation when unequal pressure applied on straight track

b)Von-miss equivalent stress

It is mainly consider with the designing procedure by comparing the material ultimate stress with experimental analysis which most of the an engineer used to design any machine by considering the ultimate stress of the material with regard to compare the equivalent (von-miss stress) in order to avoid any failure**.**

1) The maximum Von – miss equivalent stress at the end of braking time when equal pressure applied on opposite side

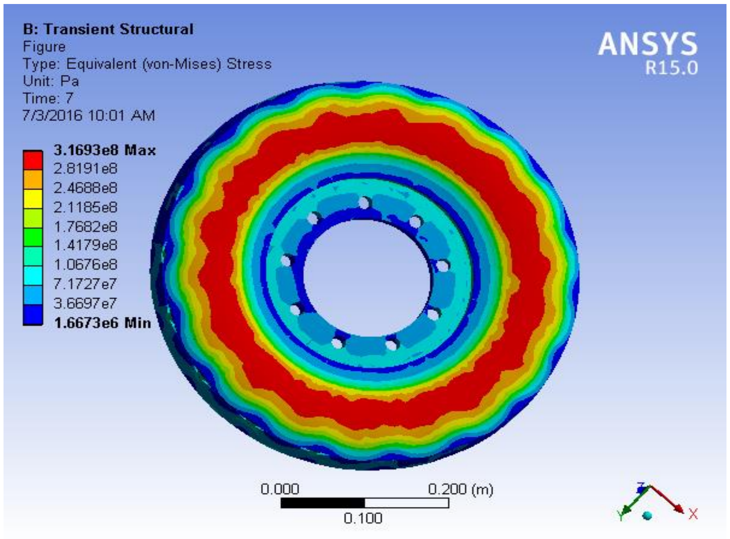


Figure 5.11:-Equivalent stress on straight track when equal pressure applied

2)The maximum von-miss equivalent stress on straight track when unequal pressure applied on opposite side of the disc.

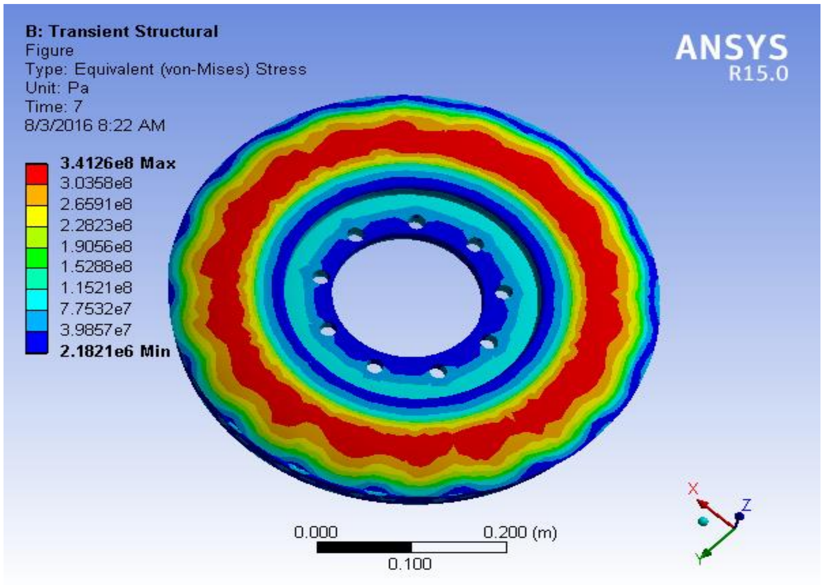


Figure 5.12:- equivalent stress on straight track when unequal pressure applied

5.2.2.2) Braking on downhill track for maintain a constant velocity

Table 16 ANSYS analysis results when equal pressure applied on downhill track

Parameters	Minimum	Maximum	Time (second)
Total deformation[m]	0	0.00021662	7
Equivalent stress (von-miss stress)[pa]	$1.6323e^{6}$	$2.8679e^{8}$	7

Table 17 ANSYS analysis results when unequal pressure applied on straight track

Parameters	Minimum	Maximum	Time(second)
Total deformation[m]	0	0.00024045	7
Von-miss stress[pa]	$1.7266e^{6}$	$3.3788e^{8}$	7

a) Total deformation

1) The maximum total deformation when equal pressure applied on opposite sides on downhill track

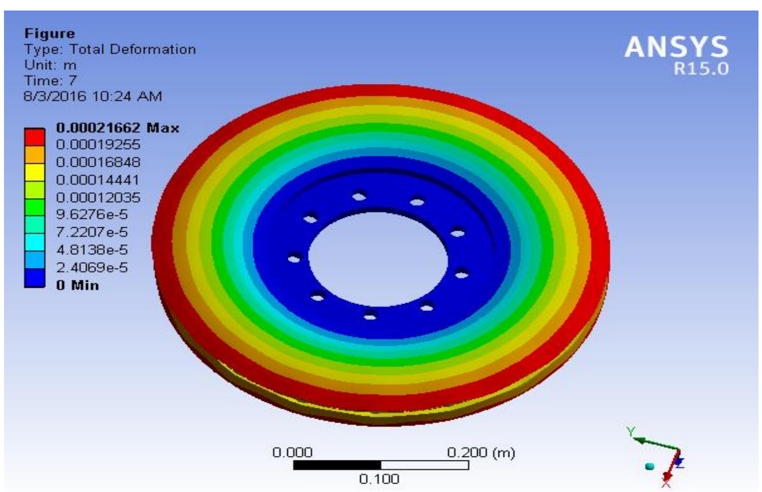


Figure 5.13:-Total deformation when equal pressure applied on downhill track

2) The maximum total deformation when unequal pressure applied on downhill track

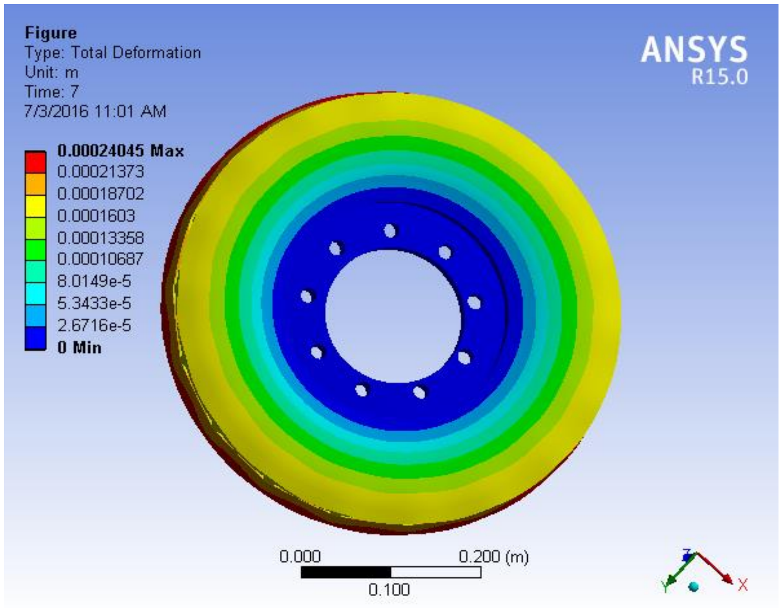


Figure 5.14 Total deformation when unequal pressure applied

a) Von- miss equivalent stress

It refers to the distribution of stress from the fixing area to the outer side of the disc at the various braking time. It various from **1.63Mpa** to **286.79Mpa** and **1.72Mpa** to **337.88Mpa** for equal pressure applied and unequal pressure applied respectively. And at the fixing area it has seen the maximum stress.

1)The maximum equivalent stress when equal pressure applied on opposite two side

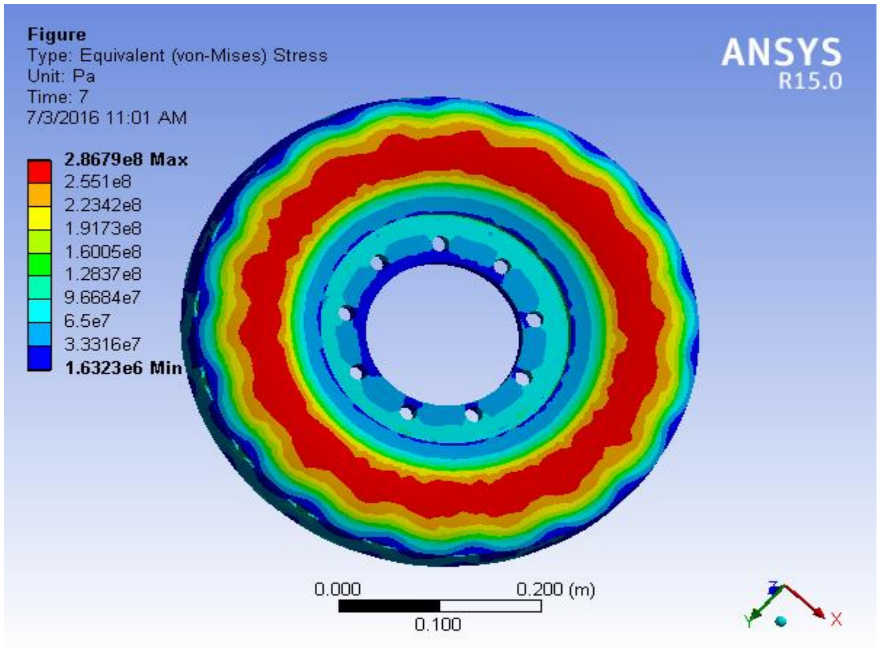


Figure 5.15:- equivalent stress when equal pressure applied on downhill track

2) The maximum von-miss equivalent stress when unequal pressure applied on two opposite side.

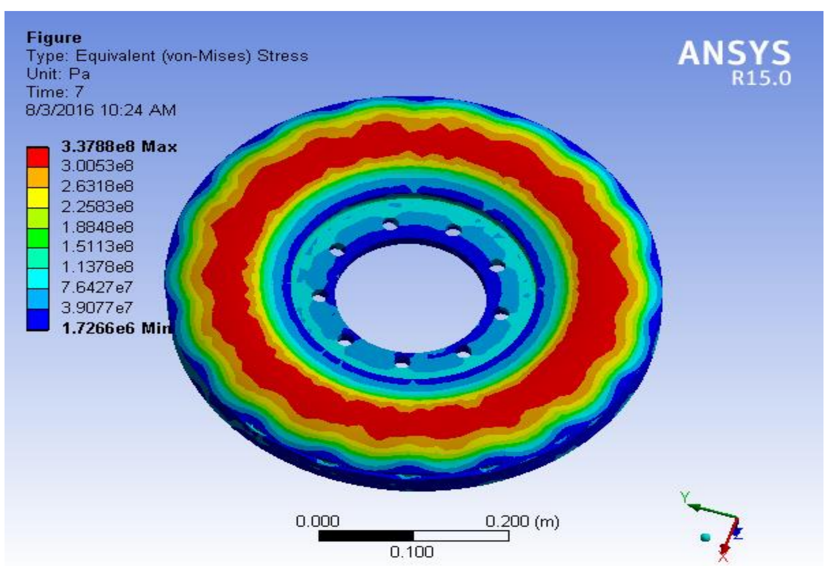


Figure 5.16:- equivalent stress when unequal pressure applied on downhill track

5.2.2.3) Braking on downhill track to run after maintaining constant velocity

Table 18) ANSYS structural results when equal pressure applied on opposite side of disc on downhill track after maintain constant velocity

Parameters	Minimum	Maximum	Time (second)
Total deformation[m]	0	0.00023652	7
Von-miss equivalent stress[pa]	$2.2412e^6$	$3.0657e^8$	7

Table 19 ANSYS structural results when unequal pressure applied on opposite side of disc on downhill track after maintain constant velocity

parameters	Minimum	Maximum	Time (second)
Total deformation[m]	0	0.00026481	7
Von-miss equivalent stress[pa]	$2.9964e^6$	$3.609e^8$	7

According to the Von-miss theory, when the disc temperature will rise up, the thermal stress is also increase. There are some factors which are an input for the development of thermal stress such as the effect of centrifugal load and also the effect of clamped pressure which is consider in the side of mechanical stress. Generally this analysis consider both the thermal and mechanical stress by considering the uniform centrifugal load of the disc brake in the rotation.

5.3) Discussion

In this analysis of disc brake was consider two main load or braking condition by compare and contrast the effect of unequal pressure application on the opposite side of disc with equal pressure application, braking on straight track at maximum velocity, braking on a downhill track for maintaining and to across after maintaining constant velocity as have seen from the simulation results, There are some parameters consider in the analysis such as the initial disc temperature, ambient temperature and disc fixture should be equal to 25℃. Depending on each case, we have got different temperature, deformation and stress value as compare between equal and unequal pressure application on opposite side of disc. Let us see each or individual case as follows.

In the case of straight track, the disc surface contact maximum temperature at the end of 14 seconds, 323.08 ℃ for equal heat flux and 385.14 ℃ for unequal heat flux due to the variation of pressure on opposite side of the disc. The mechanical analysis which are include deformation and the equivalent stress at the end of the braking time are **2.5458e^{-4},361.93Mpa** and **3.4732e^{-4},341.26Mpa** for equal and unequal pressure application on opposite side of disc respectively.

The total deformation and the equivalent stress on braking downhill track for maintaining constant velocity is reached about, **2.1662e^{-4}**, **286.79Mpa** and **2.4045e^{-4}, 3.3788Mpa** for equal and unequal pressure respectively. In the case of downhill track after maintaining constant velocity to across the track, in this case also there is a higher temperature difference, deformation and equivalent stress variation between equal and unequal pressure application at braking time or simulation time.

The maximum temperature is **319.34** and **378.23** for equal and unequal heat flux application at the end of 14 seconds respectively. The total deformation and stress are **2.3652e^{-4}, 306.57Mpa** and 2.648 **e^{-4} ,360.9 Mpa** for equal and unequal pressure respectively.

CHAPTER SIX

Conclusion and recommendation

6.1) Conclusion

The results expressed that the effect of unequal pressure which applied on two opposite side of the disc and the heat flux has large influence on the disc rotor, such as temperature, deformation and stress value. The higher temperature on a downhill track after maintaining constant velocity and braking on straight track when equal heat flux application on two opposite side of the disc rotor after brake application was **323.08** ℃ and **298.87** ℃ which happened at 14 seconds respectively.

But the temperature value when unequal heat flux application on the two opposite side of the disc after the brake application was **385.71** ℃ and **378.23** ℃ braking on a downhill track after maintaining a constant velocity and braking on straight track which occurred at the same 14 seconds respectively. The maximum deformation and stress value on the downhill track for maintaining a constant velocity and braking on straight track during the application of equal pressure has reached **286.79Mpa, 316.93Mpa** and **2.1662mm**, **2.5458mm** after emergency brake application respectively.

But the deformation and the stress value on the downhill track after maintaining constant velocity and on straight track when the application of unequal pressure has reached **337.88Mpa, 341.26Mpa** and **2.4045mm**, **3.4732mm** respectively.

So it can be conclude that the effect of unequal pressure application and unequal heat flux due to the variation of pressure on two opposite side of disc has heavy thermo- mechanical effect Such as temperature, deformation and stress value variation as compare with equal pressure and equal heat flux applied due to failure of brake piston seal, corrosion or brake caliper sticking. One of the most problem in the frictional brakes the formation of wear due to the effect of high heat distribution and temperature. This is also one of the effect of uneven worn out internal and external pads and opposite side of the disc rotor, this also have expenses to the pads and the disc economically.

6.2) Recommendation

The paper or the analysis is not consider many loads which has an effect on the disc brake such as cyclic, the residual and shearing loads which are the most common loads and has an effect on the disc service life time and not consider also the two heat transfer methods, conduction and radiation and according to product and manufacturing cost of gray cast iron FG15 relative to the other material not consider. And also for the accuracy of the analysis it should consider the experimental work to cross chick or compare with the numerical results. It should to be for future work. Generally check the material properties of brake line, calipers, piston seal etc., in order to protect the unequal pressure application on two opposite side of disc which increase temperature, deformation and stress and should maintain properly.

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Apendex

a) Thermal load application to braking from maximum velocity on adownhill track after maintaining a constant velocity

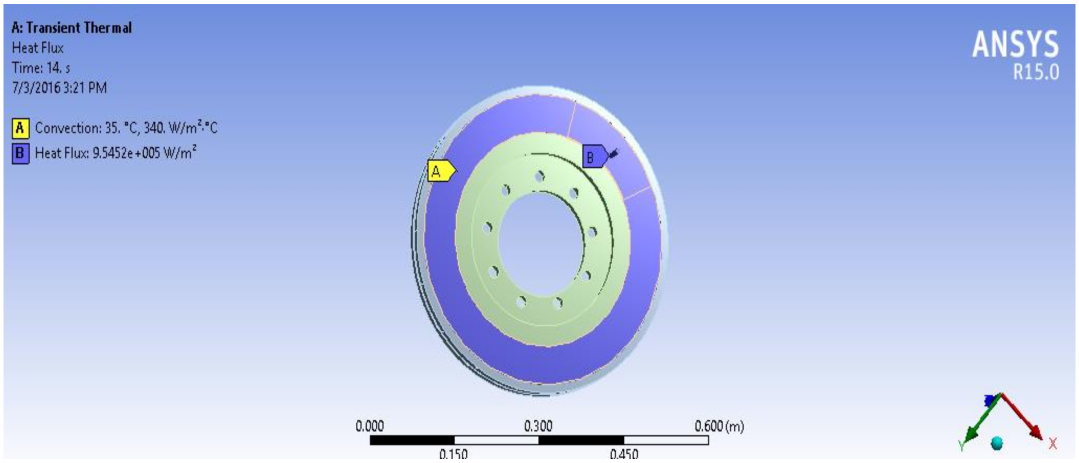

Figure 1.1: thermal load of heat flux and cofficent of convection on downhill track when equal heat

b) unequal thermal load application to braking from maximum velocity to standstill on adownhill track after maintaining a constant velocity

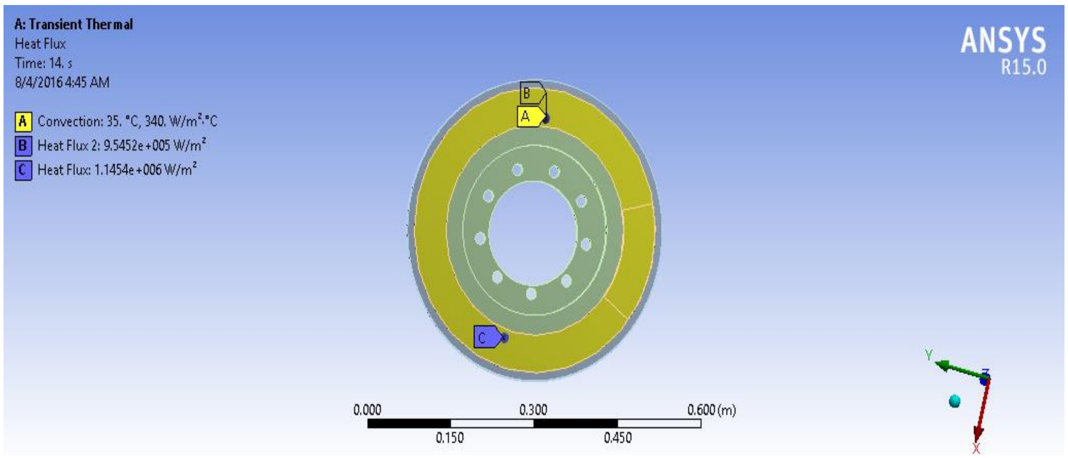

Figure 1.2: *unequal thermal load of heat flux and cofficent of convection on downhill track*

a) Pressure applied on a downhill track after mantaining a constant velocity

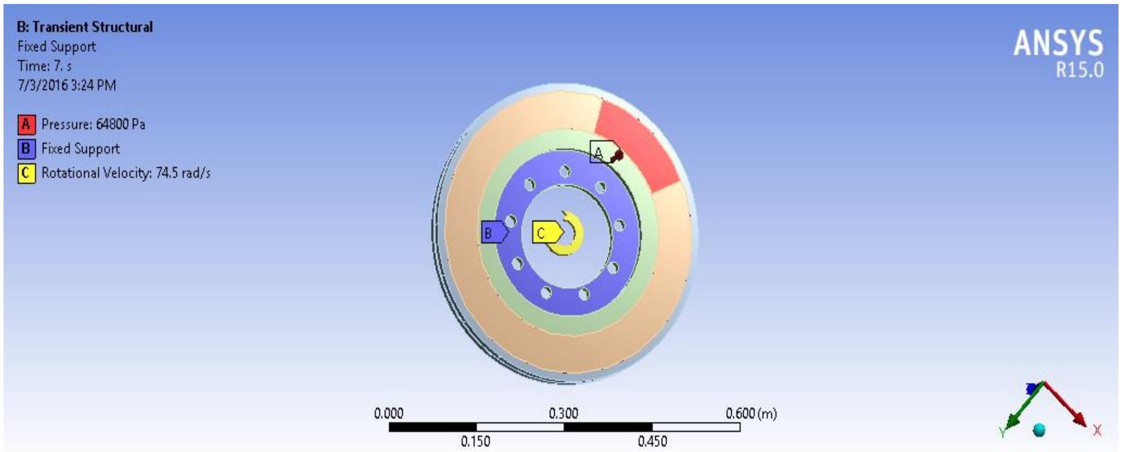

Figure 1.3: *pressure acting on disc effective surface on downhil track*

b) unequal presuure application on a downhill track after maintaining constant velocity

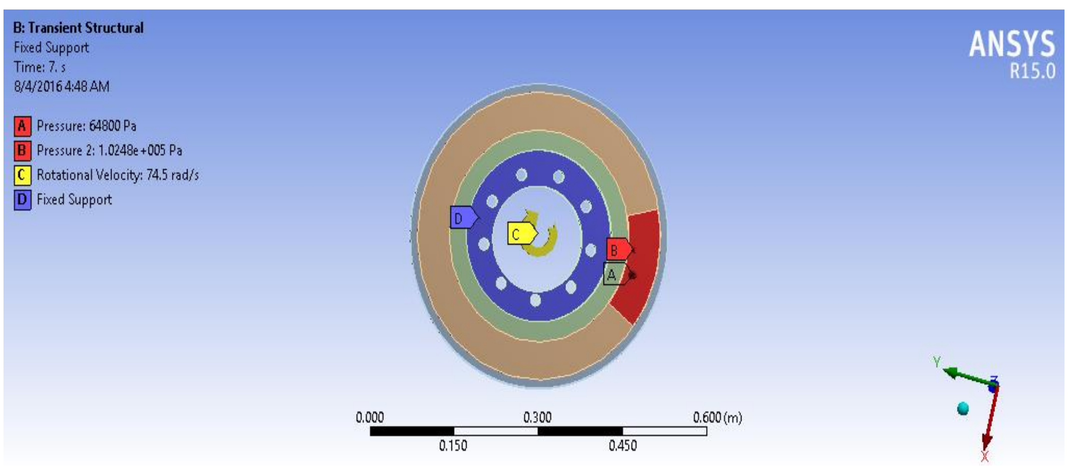

Figure 1.4: *unequal pressure acting on a disc effective area after maintaing constant velocity*

a) The maximum temperature at the end of braking time during balance heat flux applied on opposite sides:

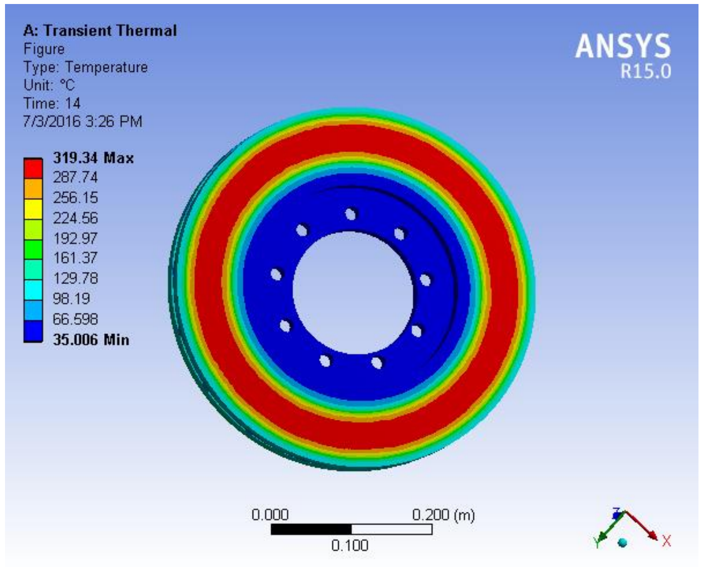

Figure 1.5: temperature value when equal heat flux

b) The maximum temperature at the end of braking time when unequal heat flux application on opposite sides.

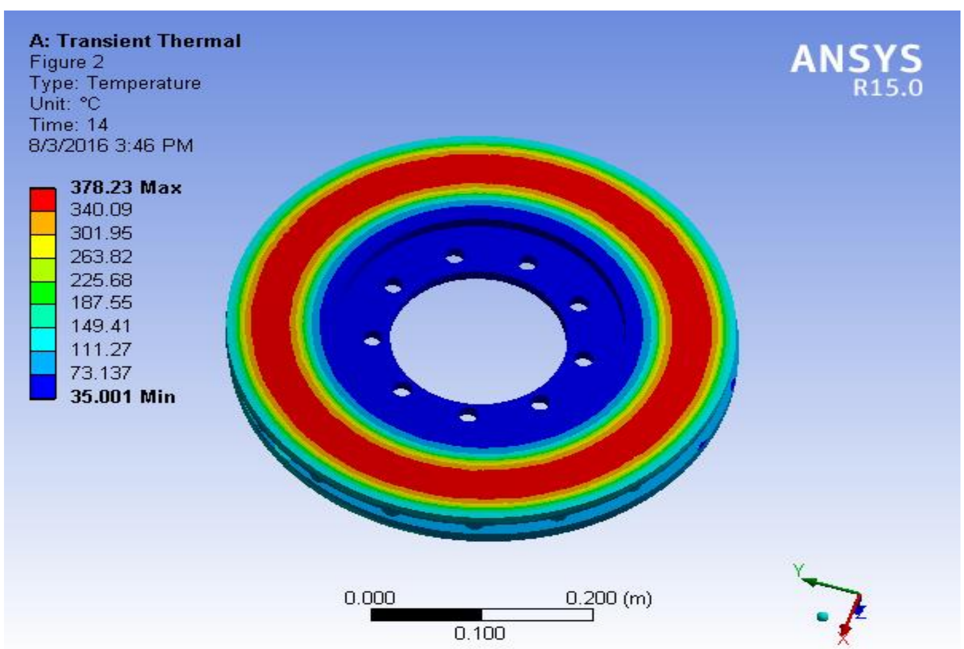

Figure 1.6: Temperature value when unequal heat flux applied

Heat flux: heat dissipation per unit area

a) The maximum heat flux at the end of braking time when equal heat flux applied on opposite sides

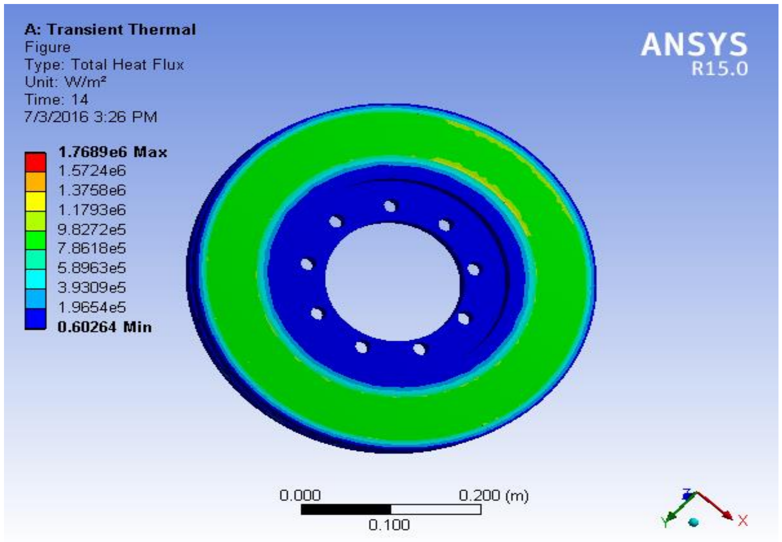

Figure 41 Total heat flux value when equal heat flux applied

b) The maximum heat flux at the end of braking time when unequal heat flux applied on two opposite sides

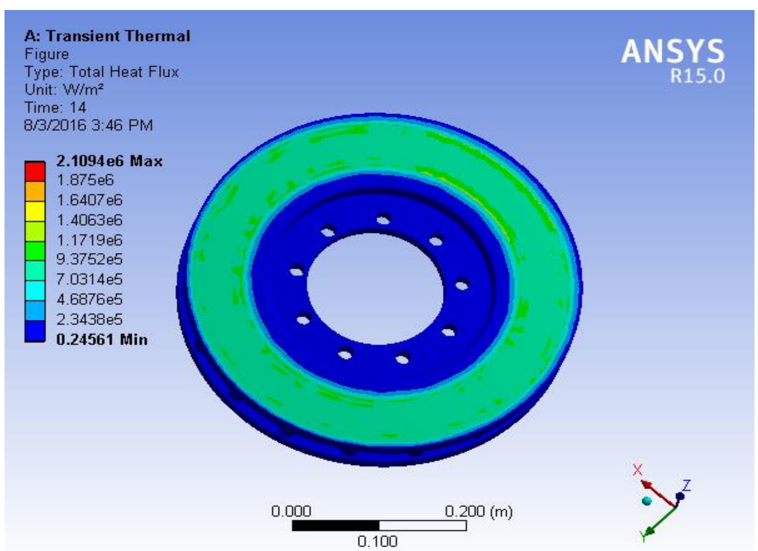

Figure 1.7: total heat flux value when unequal heat flux applied

a) Total deformation

1) Total deformation when equal pressure applied on opposite sides of disc on downhill track after maintain constant velocity

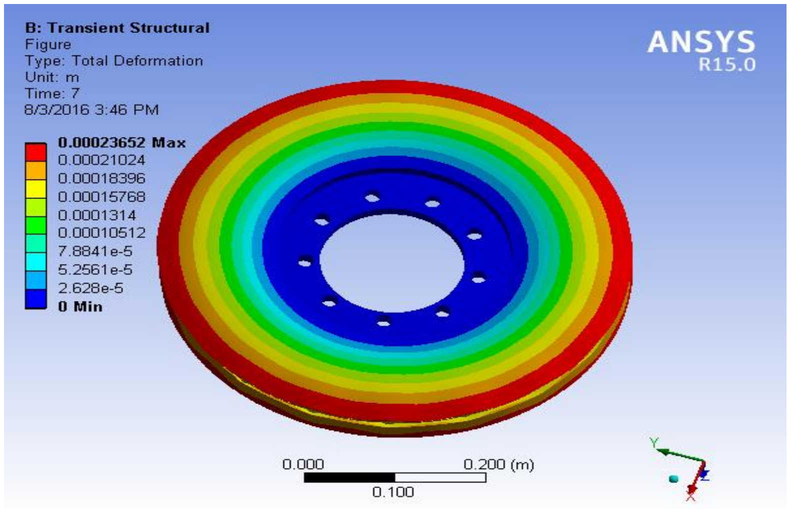

Figure 1.8: *total deformation when equal pressure applied*

2) Total deformation when unequal pressure applied on downhill track after maintain constant velocity

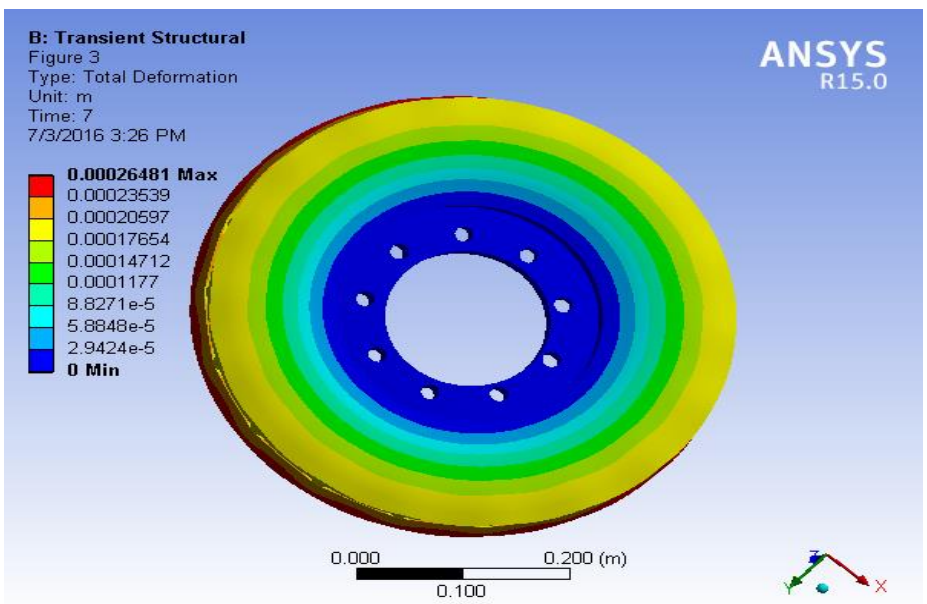

Figure 1.9: total deformation when unequal pressure applied on opposite side of disc downhill track

b) Von- miss equivalent stress

1) **The maximum Von-Miss stress when equal pressure applied on opposite side of disc**

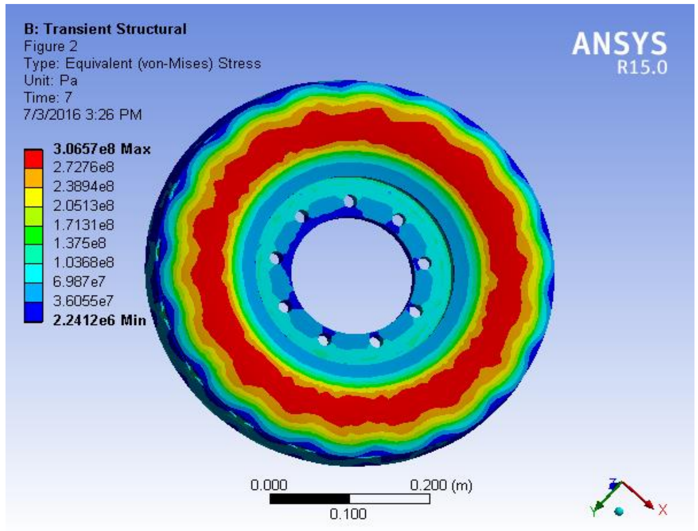

Figure 2.0: equivalent stress when equal pressure applied on opposite side of disc on downhill track

2) The maximum Von-Miss stress when unequal pressure applied on opposite side of disc on downhill track

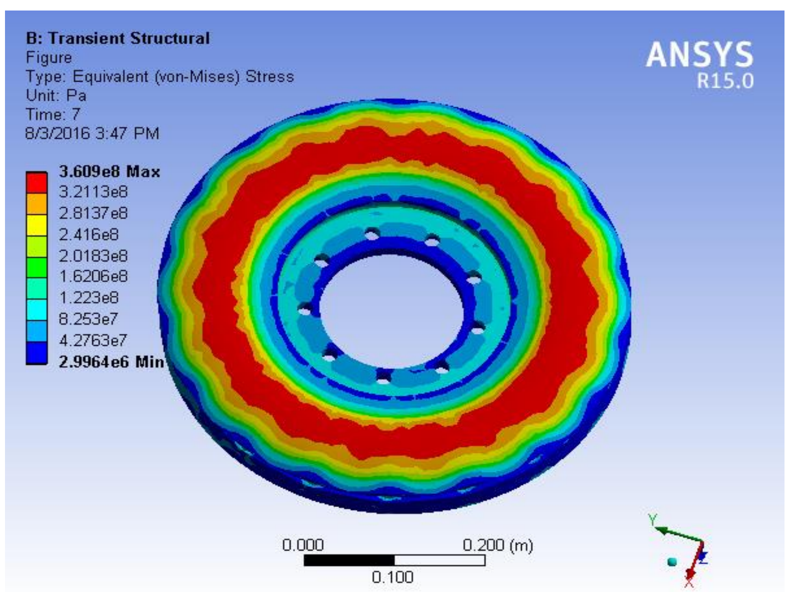

Figure 2.1: equivalent stress when unequal pressure applied on opposite side disc on downhill track after maintaining constant velocity