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ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

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Abstract

There are many reasons for affecting of the freight train disc properties. The braking system failures takes lion's share of to affect the brake disc in operation. Under normal condition the disc should have enough thermal capacity so that they can handle the heat developed by friction during normal braking action. However, in any braking components failures the braking force will be uncontrolled. Therefore, this thesis deals with the effects of uncontrolled braking force on thermomechanical propertied of discs.

Particularly this paper emphasis on the effect of uncontrolled brake force on brake disc which is created due to overcharging and losses of air in braking components. The purpose of the study is to show, how the brake disk is affected its performance (thermomechanical properties) due to uncontrolled brake force. At the end, this thesis helps to give a true picture to Ethiopian Railway Corporation (ERC) about pri maintenance on the braking components before any operation. Since any failures of braking system has effects on braking action and it has a consequences on disc performance.

In the analysis a three dimension finite element model of the brake disc is used to investigate the effects of uncontrolled braking force due to service braking. The specimens are modeled by CATIA V5 R19 and analysis is done by ANSYS 12 and by imposing the uncontrolled forces.

The findings of this analysis will observed from ANSIS solution and give conclusion based on the design standards and finally ended with recommendation.

Acknowledgment

First of all, I want to thank God and his mother for all they gave me in my life. Without insisting the almighty God and his mother, I would not have been in a position to finalize successfully accomplished this thesis paper.

I would like to express my greatest gratitude to my advisor Mr. Tsegaye Feleke, his excellent Guidance, valuable suggestions and endless supports were a great power to my work.

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Finally, I express my sincere gratitude to my parents and to my good friends for their constant encouragement and support at all phases of each progress of this thesis.

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Nomenclature

P_o'	$[Pa]$	Absolute pressure in the brake pipe
P	$[Pa]$	Absolute pressure
V	$[m^3]$	Volume
V_{ar}	$[m^3]$	Volume of auxiliary reservoir
r	$[Pa]$	Air pressure reduction in the brake pipe
P_{bc}'	$[Pa]$	The absolute pressure in the brake cylinder
V_{bc}	$[m^3]$	The volume of air in the brake cylinder
L	$[m]$	Length of pip
D	$[m]$	Diameter of pipe
P_{loss}	$[Pa]$	Pressure loses in normal pipe
v	$[\frac{m}{s}]$	Velocity
v_o	$[\frac{m}{s}]$	Initial running velocity of the locomotive
$\omega(t)$	$[\frac{rad}{s}]$	Angular velocity
r_d	$[m]$	Radius of disc
KE	[J]	Kinetic energy
m	[Kg]	Mass
I	$[\frac{J}{m^2}]$	Momenta of inertia
F	[N]	Force
T	[Nm]	Torque
Q_{gen}	[J]	Heat generated
F_{fr}	[N]	Frictional force
a	$[\frac{m}{s^2}]$	Deceleration
t_b	[s]	Braking time
r_{disc}	[m]	Disc radius
r_{wheel}	[m]	Wheel radius
q_{flux}	[W]	Heat flux
Re		Reynolds numb
Pr		Prandlt number
K_a		Thermal conductivity of air
ERC		Ethiopian Railway Corporation

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Chapter one

Introduction

1.1 Background

According to Dr. Michael JT Lewis and Anthony Coulls [1], the eminent scholar of early railways, Railway is defined as ‘a prepared track which so guides the wheels of the vehicles running over it that they cannot leave the track’.

Railways that fulfill the definition of Dr. Lewis existed as far back as 6th century BC. The Greek Diolkos was a railway with a track made of stone, 6 km in length across the Peoloponese, used for transporting ships until the 9th century AD. The ‘Agricolas de re metallica’ dates the extensive use of railways with wooden rails and vehicles to around the 15th century. Although they are of great technical interest their application was very much limited to the mining industry and had short life time. Longer and more complex lines with large wagons and horse haulage were then developed. [1]

Following the early birth of the railway ideas, the birth of modern railway is linked to the innovation of steam engines whereas the enormous development of the railway in the 20th century is very much related to electrification of railway lines. The first electric locomotives were manufactured in the last third of the 19th century, while the first diesel locomotives were built at the beginning of the 20th century. The speeds of diesel locomotives are much lower than electric locomotives. However, in areas where electrification is not available diesel locomotives have played important roles. On the other hand, due to the rising need for speed and environmental issues the electric locomotives are on the forefront of the railway industry throughout the world despite the very high cost of infrastructure in electrifying the lines. [1]

Railway in Ethiopia started in 1901 when the train with Swiss made diesel engines commenced operation on July 22 from Djibouti to Dounle which after a year and half was extended to Diredawa. The bogie wagons were some with four axles weighing10 tons and carrying 22 tons.

1.2 Train Brake

Dr. D.S. Deshmukh and Jha Shankar Madanmohan [2], a moving train contains energy, known as kinetic energy, which needs to be removed from the train in order to cause it to stop. The simplest way of doing this is to convert the energy into heat. The conversion is usually done by applying a contact material to the rotating wheels or to discs attached to the axles. The material creates friction and converts the kinetic energy into heat. The wheels slow down and eventually the train stops. The material used for braking is normally in the form of a block or pad. The vast majority of the world's trains are equipped with braking systems which use compressed air as the force to push blocks on to wheels or pads on to discs. These systems are known as "air brakes" or "pneumatic brakes". The existing air brake system of Railway coach has the following drawbacks due to excessive brake force on the brake disc like thermal cracks on brake disc and pad, brake rigging binding and reduced life of brake block and brake disc. In the opposite manner the low brake force has impacts by increasing braking distance. [2]

The compressed air is transmitted along the train through a "brake pipe". Changing the level of air pressure in the pipe causes a change in the state of the brake action on each vehicle. It can apply the brake, release it or hold it "on" after a partial application.

In the air brake's simplest form, called the straight air system, compressed air pushes on a piston in a cylinder. The piston is connected through mechanical linkage to brake shoes that can rub on the train wheels, using the resulting friction to slow the train. The mechanical linkage can become quite elaborate, as it evenly distributes force from one pressurized air cylinder to 8 or 12 wheels.

The pressurized air comes from an air compressor in the locomotive and is sent from car to car by a train line made up of pipes beneath each car and hoses between cars. The principal problem with the straight air braking system is that any separation between hoses and pipes causes loss of air pressure and hence the loss of air produces the force to applying the brakes. This deficiency could easily cause a runaway train. Straight air brakes are still used on locomotives, although as a dual circuit system, usually with each bogie (truck) having its own circuit. In order to design a system without the shortcomings of the straight air system, Westinghouse invented a system wherein each piece of railroad rolling stock was equipped with an air reservoir and a triple valve, also known as a control valve.

1.2.1 Hand brakes

Which are applied by hand action to a wheel or lever on the vehicle. Nowadays are generally used for securing unattended or unpowered vehicles against unplanned movement, not for braking actions while operating. That is why usually it is known as parking brakes.

1.2.2 Pneumatic brakes

This system uses air pressure variations both to command and to apply the brake blocks or pads, generating braking/releasing actions and forces. The vast majority of trains use compressed air, changing the level of air pressure in the brake pipe determining a change in the state of the brake on each vehicle.

In the case of straight air brake system, the increase of air pressure in the pipe determines the increase of air pressure in the brake cylinders of each vehicle. In the case of indirect air brake system, braking actions are commanded by decreasing the pressure in the train's pipe, generating by special features the increase of air pressure in the brake cylinders of each vehicle.

1.2.3 Electro-pneumatic brakes

That has an electrical command while compressed air is used to increase the pressure in the brake cylinders of each vehicle to apply the brake blocks or pads for generating braking forces.

1.2.4 Rail (track) brakes

Which are usually electric commanded and the braking force is achieved due to strong magnetic forces induced by large electromagnets hung under the vehicle's bogie, over the top surface of the rails. If the braking forces are generated by the frictional forces between the electromagnets and rails it is an electromagnetic rail brake. If the electromagnetic fields generate eddy currents in the rails, creating forces acting in the opposite direction of the movement of the train, it is a linear eddy current brake system.

On the same principle is the rotary eddy currents brake system, the metallic mass being either the wheels, or discs attached to the wheel sets.

1.2.5 Electric braking

Which is based on the reversibility of the electric engine, in particular the electric traction motors are reconnected in such a way that they act as generators which provide braking effort. Practically, the kinetic and/or the potential energy (while running on slopes) are converted into electric energy. If the power generated during braking is dissipated as heat through on-board resistors it is about rheostat braking. [2]

On electric railways, it is also possible to convert the energy of the train back into usable power by diverting the braking current into traction supply line, this being the case of regenerative braking.

1.2.6 Hydraulic brakes

Which act using hydraulic oil and, depending on the achievement of braking forces, there may be hydrostatic, when is due to oil pressure increasing or hydrodynamic when the kinetic energy of the vehicle is converted in the rotor of a hydraulic pump in heat which is dissipated through the oil cooling system.

From the above train braking systems let me discuss on the one braking systems i.e. electro pneumatic braking systems. Because now a day it is widely used across the world they have almost the same components and working principles. [2]

1.3 Electro pneumatic braking systems

Electro-pneumatic braking system is originally designed for subways or metros, and it has more recently been used on main line passenger railways and some specialized freight operations. An electro-pneumatic brake will use electronically actuated valves that will reduce the stopping distance of the vehicle by decreasing the time lag and the response time of the brake system. There are many types of electro pneumatic brake systems is used today and most of them were developed as "add on" the original brake system and ,as a result , incorporate some common principle in their design . [2

Here the main components and function of electro pneumatic braking systems are summarized in the table below.

Table 1 Function of Electro pneumatic with respect to their components.

Braking system components	Function	Braking system components	Function
Electromagnetic valves	For actuating the distributor valve	Service reservoir	Holding air for individual coach
compressor	To compress air	Auxiliary reservoir	Storage of air for braking
Main reservoir	To hold air for the whole train	Distributer valve	To control braking condition
Driver brake valve	For actuating the system	Coupling hose	To connect pipes
Electrical unit	Developing power	Cut off cocks	To cut the passage
Train pipe	Wey of the air	Brake cylinder	Force developing area
Branch pipe	Branching air to each coach	Fluid (air) in the system	Input for the system

The components of electro pneumatic braking system with their arrangements are clearly shown in the diagram below.

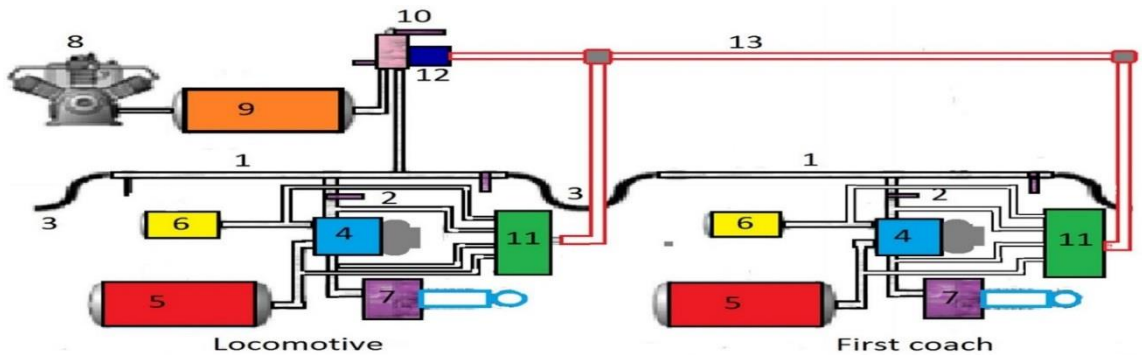

Figure 1 Schematic diagram of train electro pneumatic braking system

1) Train pipe, 2) cut off cock 3) coupling hose 4) distributor valve 5) auxiliary reservoir

6) Service reservoir 7) brake cylinder 8) compressor 9) main reservoir 10) drivers brake valve

11) Electromagnetic valve box 12) electrical unit 13) cable holding pipe

1.4 Working principle of train electro system pneumatic braking

1.4.1 Service brake application position for the locomotive and all coaches

In service brake application when the driver select the drivers brake valve on service application position the service electromagnetic valve located on the locomotive and on all the coaches at different distance from the driver are excited electrically at the same time causing compressed air in the train pipe to exhausts to atmosphere through the service electromagnetic valve at the same time. [3]

Since the pressure in service reservoir and in train pipe makes the distributors main valve piston in equilibrium, up on exhausting the train pipe pressure the main piston of distributor valve moves upward causing the passage between auxiliary reservoir and brake cylinder to open. Compressed air from auxiliary reservoir flows to brake cylinder as a result service braking action is performed.

1.4.2 Release application position for the locomotive and all coaches

In release application position when the driver select the drivers brake valve on release application position the release electromagnetic valve located on the locomotive and on all the coaches at different distance from the driver are excited electrically at the same time causing compressed air from train pipe and service reservoir flow to upper side of distributor main valve piston.[3]

At this time the distributor main valve piston is in equilibrium by the dawn ward pressures of brake cylinder and train pipe and upward pressure of service reservoir. At this time the cumulative downward pressure is greater than up word pressure. As a result the main valve piston displaces downward causing the release valve located at the bottom of distributor valve to open. The compressed air in the brake cylinder exhausts to the atmosphere. [12]

1.4.3 Emergency brake application position for the locomotive and all coaches

In emergency brake application when the driver select the drivers brake valve on emergency brake application position the service electromagnetic valve and emergency electromagnetic valve located on the locomotive and on all the coaches at different distance from the driver are excited electrically at the same time causing compressed air in the train pipe to exhausts to atmosphere through the service and emergency electromagnetic valves at the same time.

Since the pressure in service reservoir and in train pipe makes the distributors main valve piston in equilibrium, up on exhausting the train pipe pressure the main piston of distributor valve moves upward quickly causing the passage between auxiliary reservoir and brake cylinder to open. Compressed air from auxiliary reservoir flows to brake cylinder as a result service braking action is performed. [3]

By observing from the above discussions and diagrams since the command is transferred from the leading locomotive to the last coach by electrical signals the same amount of pressure is developed in the brake cylinder of each coaches at the same time regardless of their position. So the same amount of longitudinal braking force is developed on each coaches causing the whole train set to brake at the same time. This too smooth braking minimizes the coupler and draft gear damage.

1.5 Types of electro pneumatic mechanical brakes

1.5.1 Disc brakes

The applied brake force acting on the brake caliper pushes the brake pads against the rotating disc which in turn constrains the rotation of the wheel.[4]

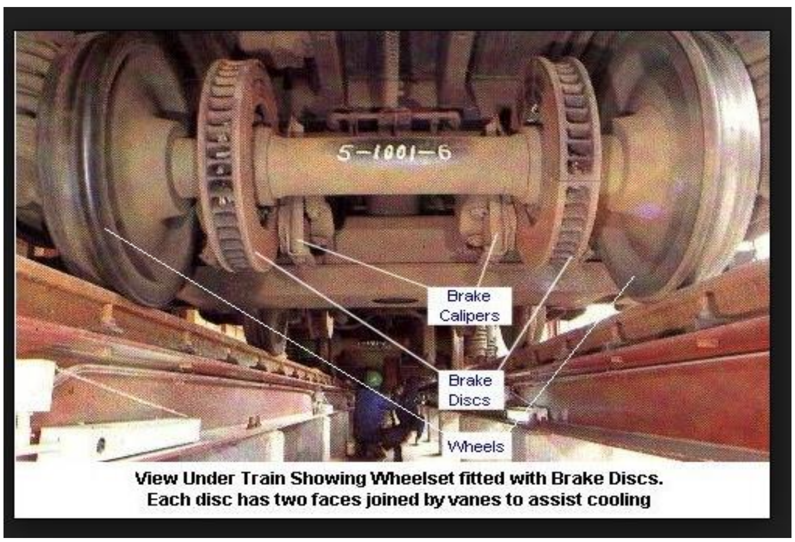

Figure 2 Disc brake assembly [4]

1.5.2 Tread brakes

In this type of braking, block brakes are pressed against the wheel so that the train could either be decelerated or stopped if necessary.

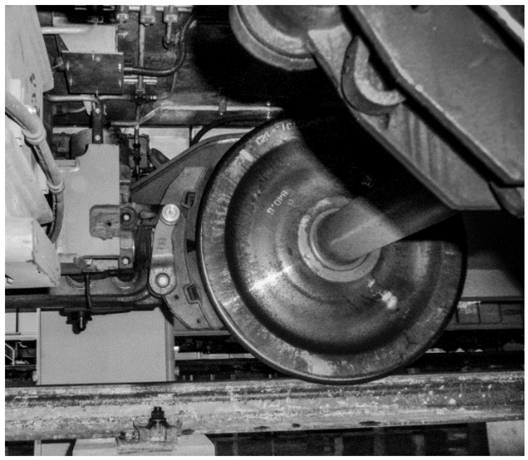

Figure 3 Tread brake assembly [5]

In Railways one of the major maintenance costs is the cost incurred by inspection and replacement of worn out braking materials. The mechanical properties of the braking equipment's like disc pad are highly affected by very high temperatures. The disc pad material is likely to fail due to metallurgical damage, thermal crack or induced tensile residual stress when it is subjected to repeated high temperatures which is caused by the force in brake cylinder.[5]

In recent days many researchers have been conducting researches to study the various failures that brake systems undergo and outline the management of disc pad failures. However the cause of variation in disc pad temperatures is involved due to the complexity in addressing the contributing factors and the multi-disciplinary nature of the problem. This paper will focus on the contribution of the uncontrolled braking force in braking system to the overheating of the disc pad, the variation in braking time and distance, the problem associated with it and the corresponding remedial measures to be taken.

1.6 Possible causes of uncontrolled braking forces of disc braking

The main causes of uncontrolled braking force in electro pneumatic braking system are due to failure of the following factors.[6]

1. Brake power creations system (Pneumatic system)
2. Brake power transmission system (Brake rigging)
3. Human Failure/Negligence
4. Miscellaneous Factors:

1.6.1 Brake power creations system (pneumatic system)

All the components which are available for the passage of compressed air between loco and the brake cylinder come under the power creation system. Any failure in these parts between loco and the brake cylinders results uncontrolled brake force. [6]

The main parts come under power creation system are:

a. Brake pipe,
b. Hose couplings
c. Cut-off angle cock
d. Guard Emergency brake Van valve
e. Dirt collector
f. Auxiliary reservoir
g. Control reservoir
h. Distributor valve
i. Brake cylinder
j. Feed pipe
k. Air hoses

A - Brake pipe

Common Pipe Bracket is permanently mounted on the under frame of a vehicle. One face is mounted with distributor valve along with intermediate piece and other face with control reservoir. The advantage of fitting a common pipe bracket is to remove the distributor valve for repair or replacement without disturbing the pipe connections. [6]

Due to difference in calibration of brake pipe pressure gauges in the locomotives, there is a chance of overcharging/under charging of BP pressure during loco changing, even though pressure gauge indicates 5.0 Kg/Cm^2 Pressure which results in uncontrolled brake force throughout the formation due to differential pressures in the incoming/ outgoing locos. [6]

Whenever the locos are reversed at the junction stations. Sometimes it becomes difficult to charge 5.0 Kg/Cm 2 pressure in rear most vehicle, whereas, AR already remains charged with 5.0 Kg/Cm 2 before the reversal of locomotives. This difference of the pressures causes uncontrolled brake force in the rear portion of the train. [6]

B – Brake pipe coupling

To connect subsequent wagons, the hoses of BP are screwed to coupling and hose nipple by means of stainless steel ‘Bend it type clips. The coupling is designed in the form of palm end and hence also known as palm end coupling. Since a joint is formed at the coupling head, leakage may take place, through it. Therefore, it is necessary to subject the hose coupling of brake pipe to leakage test. The air brake hose coupling are provided in the brake pipe line. [3]

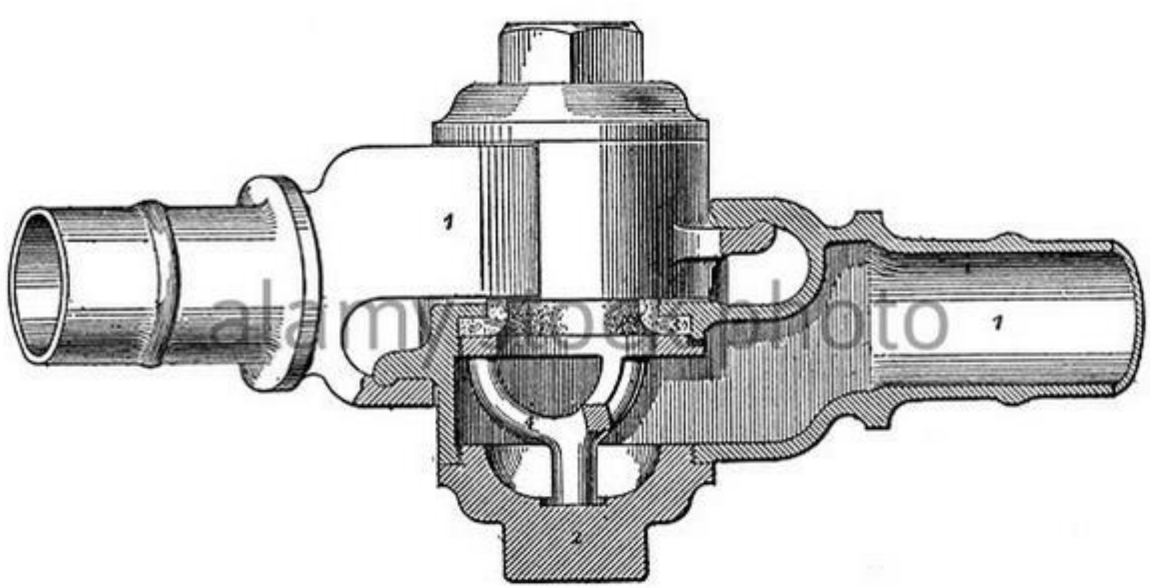

Figure 4 Train pipe coupling hose [6]

C - Cut off angle cock

Cut of angle are provided on the air brake system to facilitate coupling and uncoupling of air hoses. When the handle is parallel to cut off angle cock, the air can easily pass through the cock. The position of the handle is known as open position. When the handle is placed perpendicular to the cock body, thereby closing the passage of air. This position of handle is known as closed position. The cut off angle cock is to be completely dismantled and overhauled during POH or when there is some specific trouble. [6]

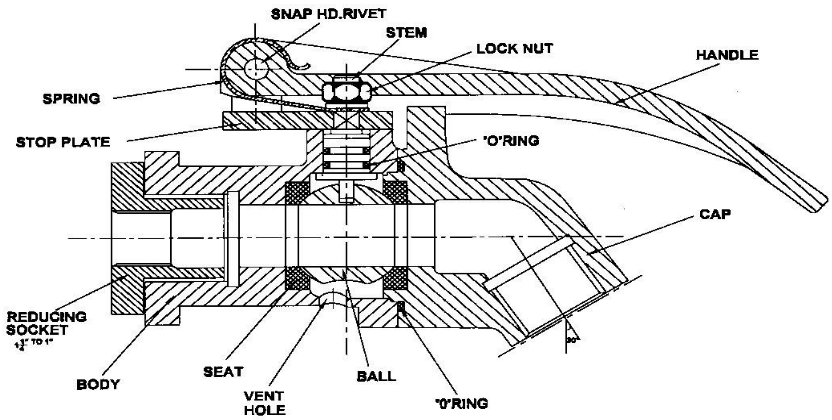

Figure 5 Cut off angle [6]

D -Brake cylinder

Brake cylinder is provided for actuating the brake rigging for the application and release of brakes. The brake cylinder receives compressed air from auxiliary reservoir after bring regulated by the distributor valve and develops mechanical brake power by outward movement of its piston assembly. [6]

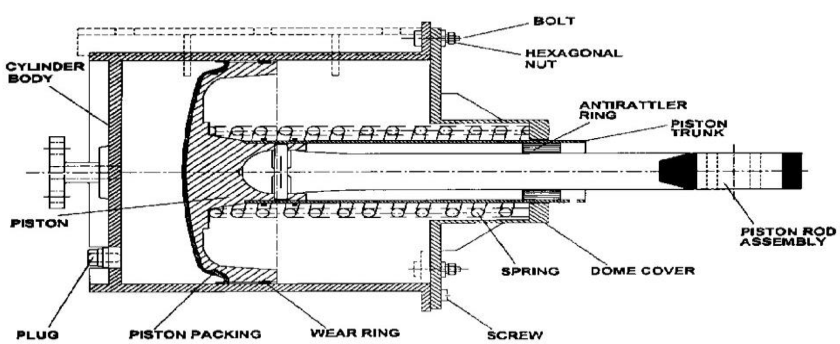

Figure 6 Brake cylinder [6]

E -Dirt collector

Dirt collectors are provided at the junction of the main pipe and branch pipe in both feed pipe and brake pipe. These are meant for removing dust, moisture and scale particles from air before it enters the DV & AR. [6]

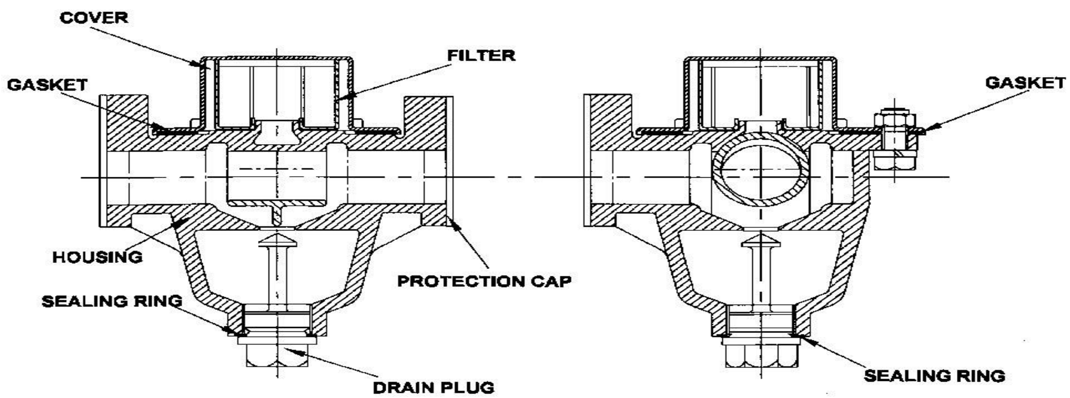

Figure 7 Dirt collector [6]

F- **A**uxiliary reservoir

It is a pressure vessel and its function is to feed dry compressed air to the brake cylinder for application of brakes. AR is charged through brake pipe as well as feed pipe in coaching. Pressure in AR for goods single pipe – 5 kg/cm^2 and for Coach Twin pipe 6 kg/cm^2. Capacity of AR in Goods and Coaches are 100 letters and 200 letters, respectively. [6]

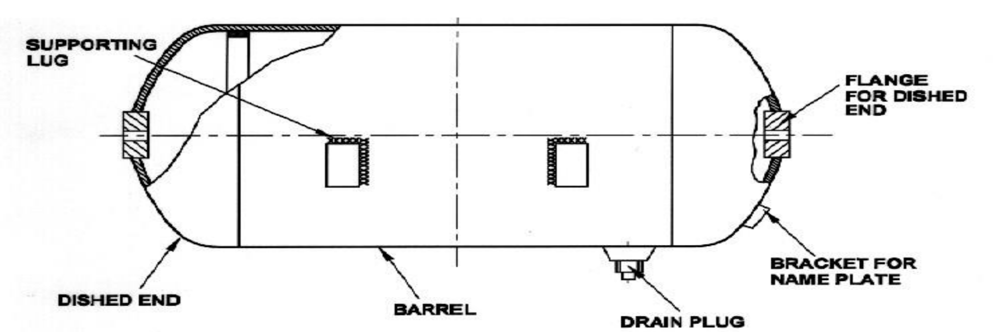

Figure 8 Auxiliary reservoir [6]

G- Guard's emergency brake valve

GEBV is provided in the Guards compartment. This valve provide a facility to the guard initiate brake application in case of any emergency. [6]

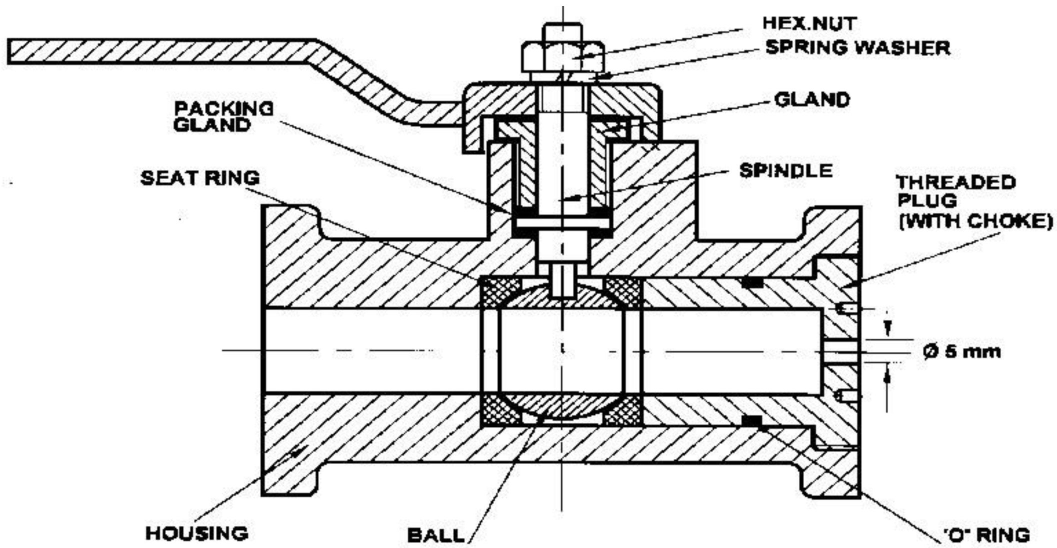

Figure 9 Guard's emergency brake valve

H - Control reservoir

The control reservoir is mounted on the under frame of coach very near to the DV and connected through a small chock and a pipe with the DV. During braking operation and releasing operation through recharging operation, the air pressure inside the control reservoir remains unaltered. Only during manual release the air of control reservoir escapes to the atmosphere through release valve of DV. Capacity of CR in per wagon and Coach are 6 liter and 9 liter respectively. [6]

I - Distributor valve

DV is most important functional component of Air Brake System and is also some time referred to as the Brain of air brake system. It is connected to brake pipe AR & BC. It senses the BP pressure variations & work automatically to provide brake application as well as brake release. The distributor valve assemble consists of a valve body, a common pipe bracket and CR.

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In general the above brake power creation system components have many impacts on disc since they created uncontrollable brake force due to air pressure leakages. The main components that creates air leakages are listed below,

a. Leakage through Brake pipe
b. Leakage through Cut- off Angle Cock
c. Leakage through Dirt Collector
d. Leakage through DV joints
e. Leakage through AR
f. Leakage through Brake cylinders
g. Defective Air Hoses.

How breakages in the pipe connection between DV and the Brake cylinder causes uncontrolled brake force? Breakage in the pipe connection between BP, DV and the brake cylinder leads to excessive leakages of air from AR after the brake application. This excessive leakage drops the MR pressure abruptly. Once MR pressure drops, it is not possible to restore the BP pressure to 5.0 Kg/Cm 2 during release, results in excessive braking force which mainly cases of brake binding in the entire formation.[6]

In case of single pipe, auxiliary reservoir is charged through brake pipe. Excessive leakage in the A.R due to working out of drain plug or corrosion and foreign body hitting on the AR, prevents recharging of BP to 5.0 Kg/Cm 2 results in brake binding on the entire formations.[3]

1.7 Statement of the Problem

This is to address to the operator and maintainable division that uncontrolled brake force has different effects on braking equipment's due to braking operation. The effects mainly appeared in different manner like,

- ✓ Overheating of the disc pad,
- ✓ Lesser braking force generation by brake cylinder.
- ✓ Higher maintenance cost.
- ✓ Discomforts to the passengers
- ✓ Difficulties to control the train

1.8 Objective

1.8.1General Objective

The overall aim of this research is to develop a fully coupled thermo-mechanical FE model on brake disc and pad to investigate brake system stability due uncontrolled brake force and to show the structural deformation of the brake components when subject to uncontrolled brake forces.

1.8.2 Specific Objectives

The specifics objectives of the research are as follows:

1. To develop a validated 3-dimensional finite element model of a typical Brake disc assembly by conducting CATIA modal analysis in order to validate the finite element model by correlation of ANSYS analysis.
2. To analyze the thermo-mechanical contact effects of brake disc and pad due to uncontrolled brake force losses air in pipe and brake cylinder.
3. To simulate the effects of material properties and geometrical changes of the brake components on the occurrence of uncontrolled brake force.

1.9 Research methodology

1.9.1 Research Design

The research design started from data collection, producing modeling and making analysis of numerical and finite element analysis and it ended with giving recommendation.

1.9.2 Data collection

Gathering and collecting data's from international standards and Parameters of brake component geometry, material properties and design operation condition to establish the importance of these parameters with respect to different brake force condition. This is very useful to compare the brake disc condition due to normal brake force and uncontrolled brake force condition.

1.9.3 Modeling

The modeling methodology, proposed through this thesis is integrated by FE approach is described. This uses a fully coupled thermo-mechanical instability simulation technique to study the dynamic behavior of a typical rail brake system during different brake force within a braking event. The finite element approaches starting from geometry modeling using CATIA software and ANSYS workbench.

1.9.4 Numerical And finite Element Analysis

The applied braking force and effective braking distance numerically calculated at different operation condition with regarding to both normal and uncontrolled braking condition. This helps us to prepare input parameter for ANSYS finite element analysis and to compare the finding to the standards.

A thermo-mechanical contact analysis investigates the disc-pad transient thermal and contact pressure distribution under specific braking operating conditions. The effect of contact pressure distribution under heating and cooling conditions is reported for different contact interface properties. The results (for example, contact condition) are presented as a function of time. This chapter also discusses the effect of disc deformation during the heating-up process. A thorough study gives an insight into the extent of disc coming during the braking process. A correlation between the FE model prediction and numerical results is presented.

1.9.5 Research Area

This study concerns railway operation in Ethiopia mainly focusing on freight train in ERC.

1.9.6 Significant of the study

As usual braking is applied in each train activates. Also a problem occurred in a train due to each breaking application. This leads to lessen the profitability of the railway cooperation and reduced safety. Hence, conducted “analyze of the effects of uncontrolled braking force” will benefit both the customer and railway operating organization based on protecting the life of brake equipment’s and reducing maintenance costs.

1.9.8 Conclusion and Recommendation

Conclusions are drawn from the overall findings of this research along with recommendations for future work and making awareness for operator and responsible persons.

Chapter Two

Literature review

Dr. D.S. Deshmukh & Jha Shankar Madanmohan [7] developed research about "Design Evaluation and Material Optimization of a Train Brake" the aim of the project is to overcome the drawbacks of braking components by reducing the effective brake force on the brake blocks without affecting the existing designed (Braking Function) requirements.

To validate the strength of train brake, Structural and Modal analysis were done on the train brake. In structural analysis, ultimate stress limit for the design is found and in modal analysis, mode shapes of the train brake for number of modes can be analyzed. The analysis is done by applying two different materials Cast Iron and High Carbon Steel for train brake.

According to the existing air brake system of Railway coach the brake force applied per one brake block is 2.187tons. The following drawbacks due to existing brake force on the brake blocks - thermal cracks on wheel tread, brake binding and reduced life of brake block. A modification is done in the project to overcome the above said troubles by reducing the minimum effective brake force without affecting the existing designed requirements.

After modification, the brake force applied per one brake block is 1.653tons.The analysis is done for two forces and by using two materials Cast Iron and High Carbon Steel. The maximum stress induced in the brake block by the application of modified brake force (1.653 N) is 2.765 N/mm^2 which is less as compared with stress induced in the brake block by the existing brake force (2.187N) 3.67 N/mm^2. With the application of modified minimum brake force, the brake block is safe. Hence the modification carried out in this project work is justified. The following finding was observed. [7]

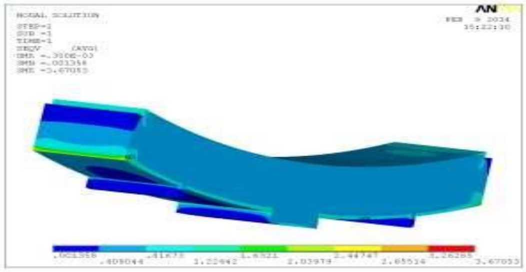


Figure 10 High Carbon Steel Stress analysis

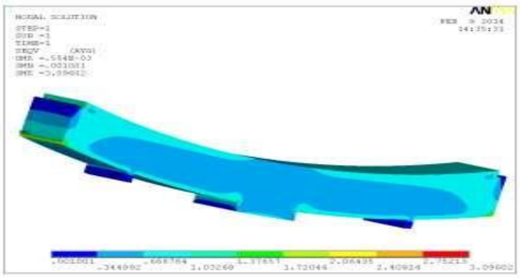


Figure 11 Cast Iron Stress Analysis

Table 2 High Carbon Steel by applying 2.187 tons

	Results	**Permissible**
Displacement (mm)	$0.300e^{-3}$	
Stress (N/mm^2)	3.67	295

Table 3 High Carbon Steel by applying 1.653 tons

	Results	**Permissible**
Displacement (mm)	$0.22e^{-3}$	295
Stress (N/mm^2)	2.765	295

Table 4 Cast Iron by applying 2.187 tons

	Results	**Permissible**
Displacement (mm)	0.554e-3	
Stress (N/mm^2)	3.09	520

Table 5 Cast Iron by applying 1.653 tons

	Results	**Permissible**
Displacement (mm)	$0.418e^{-3}$	
Stress (N/mm^2)	2.33	520

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

Above images illustrates the stress and model analysis plots for the existing and modified train brake system. From the plots illustrated above, conclusions were made, by comparing the results of two materials due to different modified braking forces they concluded that different stresses and deformation are developed and they choices high carbon steel for train brake system with different braking condition.

Maharajpur, Gwalior – 474020, [8] states about brake continuity test is one of the most important test of air brake system. This test is done to check the continuity of the brake pipe through-out the train. It ensures that the brake pipe pressure gets released/created, during the brake/released, simultaneously in brake pipes of all wagons throughout the length of the train.

The underline principle of the test is that first the brake pressure is created from the locomotive. The pressure so created in brake pipe is then destroyed from the brake van/last vehicle and in doing so, the brake pipe pressure in locomotive should also become zero.[8]

The brake pipe pressure is once again created from the locomotive and the build-up of brake pipe pressure is checked from the brake van/last vehicle. In case of any discontinuity and/or blockage in the brake pipe the rise/fall of brake pipe pressure in loco shall not cause rise/fall of brake pipe pressure in brake van and vice versa.[8]

The brake continuity test must be carried out on train in the following circumstances.

a) Fresh locomotive or additional locomotive is attached to the front of the train.
b) Fresh locomotive or additional locomotive is attached to the rear of a fully fitted train.
c) Vehicle is attached at any position in the fitted portion of the train.
d) Vehicles in the fitted portion of the train are detached from other than extreme rear end.
e) After any brake defect or irregularity which has affected the continuity of the brake system has been rectified.

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

Whenever the problem of low brake pipe pressure is experienced, conduct leakage test as per procedure detailed below to check whether the leakages in train air brake system are within permissible limits and to take corrective actions, if required. The writer described the steps how to test leakages in the air braking system. [8]

Step 1: Ensure that brake pipe of all wagons are coupled and brake pipe angle cocks on all the wagons are in open position.

Step 2: Place the Driver's Automatic Brake valve handle in release position. Ensure that the brake pipe pressure has stabilized in the locomotive and rear most vehicle to the level indicated below.

Table 6 Brake pipe pressure for different wagons

Length of the train	Loco	Last wagon
Up to 56 BOXN wagons	5.0 Kg/cm2	4.8 Kg/cm2
More than 56 BOXN wagons	5.0 Kg/cm2	4.7 Kg/cm2

Step 3 : Move the driver's automatic brake valve towards the application position to reduce the brake pipe pressure from 5 Kg/cm^2 to 4kg/cm^2.

Step 4: After the brake pipe pressure has stabilized, close the brake pipe isolating cock provided between additional relay valve and brake pipe on the locomotive.

Step 5: Wait for 60 seconds for temperature and gauge settlement and then note the drop in pressure in the brake pipe pressure gauge in the locomotive for five minutes.

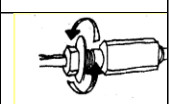
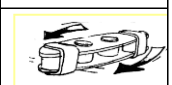
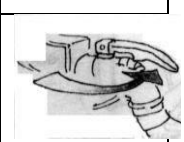
Step 6: The drop in brake pipe pressure gauge shall not be more than 1.25kg/cm^2 in 5 minutes (i.e. rate of drop should be less than 0.25 kg/cm^2/min.).

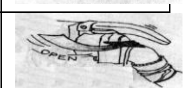
Step 7: If the leakage rate is more than the value indicated in step 6,check for leakage's on individual wagons.

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

[Maharajpur, Gwalior - 474020, [8] states and identified the en-route trouble shooting which shows the defects and remedial action on air brake train system and summarized as the following table,

Table 7 En-route trouble shooting on air brake trains

No	Trouble	Defects	Remedial Action	
1	Brake Binding	Leakage from Coupling head due to displaced work out MU washer.	Reset/Replace MU washer	
		Leakage from angle cock brake pipe joints drains plugs of auxiliary & control reservoir.	Control Leakage by Tightening loose joints.	
		Hand brake may be partially „ON‟	Release hand brakes fully.	
		Setting of empty load device handle may be disturbed.	Reset it properly.	
		Sleeve nut of empty tie rod may be in tampered condition.	Adjust and lock properly.	
		Horizontal live lever jamming against its guide brackets.	Ensure smooth operation & lubricate.	
2	Poor Brake power	Lack of through brake pipe connection from loco to brake van.	Ensure full open position of all Angle cocks –i.e., all angle cock handles in parallel Position.	
		Inadequate air pressure level in engine and brake van.	Ensure specified BP pressures are maintained in loco (5 kg/cm^2) and in	

			brake van (4.8 kg/cm^2 for 56 wagons).	
		In operative Pistons	Check & Ensure that isolating cock of DV are in "ON" position of wagons with in operative piston.	
		Empty load box change over handle of loaded wagon is kept in empty position.	Keep the handle in correct position.	
3.	Brake pipe is not charging	Cut off angle is being closed position.	Move the handle to open position i.e. move it to parallel with the pipe line.	
4	Un-Coupling of Air hose on run.	Air hose too short and not forming cradle shape.	Replace air hose with proper length (i.e. 660 □ 6 mm)	
		Working out of air hose from nipple or palm end.	Replace MU washer & Ensure proper fitment of MU washer and re-coupled properly.	
		Air hose not coupled properly.	Re-couple loco and first wagon BP hoses correctly.	
5.	BP Pressure in last Vehicle more than 5 kg/cm^2.	Loco feed pipe coupled to brake pipe.	Re-couple Loco and first wagon BP hoses correctly.	

H. Belghazi, M. El Ganaoui, J.C. Labbe [9] conducted a transient thermal and structural analysis of a brake disk to evaluate its performance under severe braking conditions and there by assist in disk rotor design and analysis. The usage of new materials was investigated which aims at improving the braking efficiency and providing greater stability to the vehicle. This study was done using ANSYS 11 software to analyze the temperature distribution, variation of the stresses and deformation across the disk brake profile. The new materials under study were Aluminum base metal matrix composite and High Strength Glass Fiber composites. These materials have a promising friction and wear behavior as a brake disk. The transient thermo elastic analysis of disks in repeated brake applications was performed and the results were compared to that of cast iron disk. In the study conducted by Zhang et al.

A non-linear FEM model of brake discs of high-speed trains with thermal-mechanical coupling has been established in ANSYS. The simulation and analysis of 3-D transient temperature field and stress field of brake discs have been carried out for the braking process. According to typical imperfections of brake discs, some imperfection models of brake discs have been developed and the thermal stresses of different models are obtained through simulation and analysis. [9]

2.2 Conclusion

From the above literature review and brake test book have discussed the effects of different brake force in train brake equipment’s and also they described how different braking force are created in the route of air train braking system.

Therefore in the paper will elaborate all the factors which are the case of different braking force (uncontrolled braking force) in the air train brake system and here will analyses the effects of such uncontrolled barking force on braking equipment’s specifically on the brake disc

Chapter 3

Numerical analysis of pressure reduction in brake power creation components

3.1 Relationships between Air pressure reduction in the brake pipe and in brake cylinder

In automatic air brake systems the brake pipe pressure has to be reduced in order to braking is applied. However, not all pressure reductions in the brake pipe cause brake application. The pressure reduction in the brake pipe should reach a certain minimum amount for the brake cylinder to move and progressively press the brake pad against the disc. In addition, there exists some maximum amount of brake pipe pressure reduction beyond which does not cause any increase in brake cylinder pressure.

The succeeding sub-sections lay the foundation for the minimum amount of brake pipe pressure leakages that is capable of causing unintentional brake application.

3.1.1 Valid range of pressure reduction in the brake pipe

When pressure is reduced in the brake pipe significantly, the piston in the triple valve will be pushed by the auxiliary reservoir air pressure and air will flow to the brake cylinder till the pressure in the brake cylinder and the auxiliary reservoir are the same.

Values of the valid range of pressure reduction in the brake pipe are obtained based on the following assumptions.

- The air in the auxiliary and brake cylinder obeys ideal gas laws
- No air is retained within the brake cylinder Prior to brake application
- The reaction time of the triple valve is so small that no back flow occurs from the auxiliary reservoir to the brake pipe
- The stroke (piston travel) of the brake cylinder is approximately 200mm.

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The air in the main reservoir is cooled down to the atmospheric temperature by the heat exchanger after being pressurized by the compressor to the auxiliary reservoir and through the brake pipe.

Hence, the expansion of the air into the brake cylinder is an isothermal process following Boyle's law of ideal gas equation.

$$PV = C \dots\dots (3.1)$$

Where:

P= absolute pressure

V=Volume

C=constant

Therefore, upon initiation of braking action by the triple valve the air in the reservoir is governed by the equation:

$$P'_{bc}V_f = (P'_o - r) * V_{ac} + P'_{bc} * V_{bc} \dots\dots (3.2)$$

Where:

P'_o = the rated absolute pressure in the brake pipe

V_{ac} = the volume of auxiliary reservoir

r = the amount of air pressure reduction in the brake pipe

P'_{bc} = the absolute pressure in the brake cylinder

V_{bc} = the volume of air in the brake cylinder

Solving for the absolute pressure in the cylinder gives:

$$P'_{bc} = \frac{V_{ac}}{V_{bc}} * r \dots\dots (3.3)$$

The relative pressure in the brake cylinder is then:

$$P'_{bc} = (\frac{V_{ac}}{V_{bc}} * r - 100)Kpa \dots\dots (3.4)$$

Considering the effect of leakage and taking the piston travel to be 200 mm. The value of $\frac{V_{ac}}{V_{bc}}$ for railway application is set to be 3.25.[10]

Hence the above equation reduces to:

$$P'_{bc} = (3.25 * r - 100)Kpa \dots\dots (3.5)$$

The minimum valid brake pipe pressure reduction

For the brake pad to be practically pushed against the disc the resistance force of the release spring at the back of the brake cylinder needs to be overcome. The pressure exerted by release spring and riggings at the back of the brake cylinder piston is approximately 35Kpa.

From equation (3.5) Solving for the pressure reduction in the brake pipe r gives:

$$(3.25 * r - 100) = 35\text{kpa}$$

$$r = 41.5384\text{Kpa}$$

$$r \approx 40\text{kpa} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.6)$$

For the sake of safety and convenience we take, therefore any leakage or reduction that brings about a brake pipe pressure reduction greater than or equal to 40kpa create braking action.

3.2 Cause of air pressure reduction in brake system

3.2.1 Friction along through in pipe

Friction is the main cause of energy loss in fluid power system. The energy losses due to friction is transfer in to heat, which is given off the surrounding air.

Friction loss

$$P_f = \frac{C*L*Q^{2}}{CR*d^{2}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.7)$$

Where; P_{loss} = pressure loss

C = experimentally determined coefficient

L = length of pipe

CR = compression ratio = $\frac{\text{pressure in pipe}}{\text{Atmospheric pressure}}$

d = inside diameter of pipe

For commercial pipe the experimental data, determined as following

$$C = \frac{0.1025}{d^{0.31}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.8)$$

Where: C = experimental data

d = pipe diameter

Substituting equation …..(3.8) in to (3.7)

$$P_f = \frac{0.1025 * L * Q^2}{CR * d^{5.31}} \quad (3.9)$$

Table 8 Value of d^5.31 for several pipe size

Nominal pipe size	Inside Diameter (d)	$d^{0.31}$	Nominal pipe size	Inside diameter	$d^{5.31}$
$\frac{3}{8}$	0.493	0.0234	$1\frac{1}{2}$	1.610	12.538
$\frac{1}{2}$	0.622	0.0804	2	2.067	47.256
$\frac{3}{5}$	0.824	0.3577	$2\frac{1}{2}$	2.469	121.419
1	1.049	1.2892	3	3.068	384.771
$1\frac{1}{4}$	1.380	5.5304	$3\frac{1}{2}$	3.548	832.550

3.2.2 Pressure reduction in brake pipes fitting

The frictional loss in pneumatic fittings can be computing the Harris formula if the equivalent length of the fitting are known. The term L in the Harris formula would then represent the total equivalent length of the pipeline including its fitting. Figure below gives equivalent length value in feet for various type of fitting.

Table 9 Equivalent length of various fitting

FITTING	NOMINAL PIPE SIZE						
	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2
Gate valve (fully open)	0.30	0.35	0.44	0.56	0.74	0.86	1.10
Globe valve (fully open)	14.0	18.6	23.1	29.4	38.5	45.2	58.0
Tee (through run)	0.50	0.70	1.10	1.50	1.80	2.20	3.30
Tee (through branch)	2.52	3.30	4.20	5.30	7.00	8.10	10.44
90^0 elbow	1.40	1.70	2.10	2.60	3.50	4.10	5.20
45^0 elbow	0.50	0.78	0.97	1.23	1.60	1.90	2.40

Thus the total pressure loss in pipe, valves and fitting is 46% greater than the pressure loss only in pipe. This shows that fitting also be adequately sized to avoid excessive pressure lose.

Therefore;

$$P_{fitting} = 0.46\ P_f \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3.10)$$

Where

$P_{fitting}$ = pressure loses in fitting

P_f = pressure loses in normal pipe

3.2.3 Brake power creations and the leakages modeling

All braking time when the driver select the driver valve to braking position, the train pipe pressure exhaust to the atmosphere but because of different reasons like pipe line breakages, vents and holes, failure of cut off angle brake pipe friction…etc. Because of those reasons the reduction or exhaust amount of air either decrease or increase, as a result pressure difference is occurred between service reservoir and train pipe on the distributer valve so the main valve piston is displaced to the brake position causing compressed air to pass from auxiliary reservoir to the brake cylinder.

Due to small reduction of air (exhaust) due to high friction in pipe in air brake pipe, it cause the much time duration to create pressure different as a result the time taken to move the main piston in the triple valve is longer and it causes the compressed air to pass slow speed from auxiliary reservoir to the brake cylinder. At the system the created braking force in braking cylinder will be small and maximum braking force happened after design braking time, as result of this time the braking distance will be maximum.

Due to excess reduction because of pipe coupling hose, cut off angle cock defection, etc. in air brake pipe it cause small time duration to create pressure difference , as a result the time taken to move the valve in main piston shorter and causes the compressed air pass high speed from auxiliary reservoir to the brake cylinder . At this time the created braking force in brake cylinder will be excess either short period of time as a result of the excess braking force the brake disc and pad will affected.

So analysis will elaborate the effects in ANSYS programs to show the effects .The processes will pointed due to full service brake application.

Due to full service brake application analyze the effects of created excessive braking force due to air leakages and overcharging through pipe, cut off angle cock, coupling hose by taking assumption and comparing to normal condition. Finally thermo mechanical effects will anayse using ANSYS software.

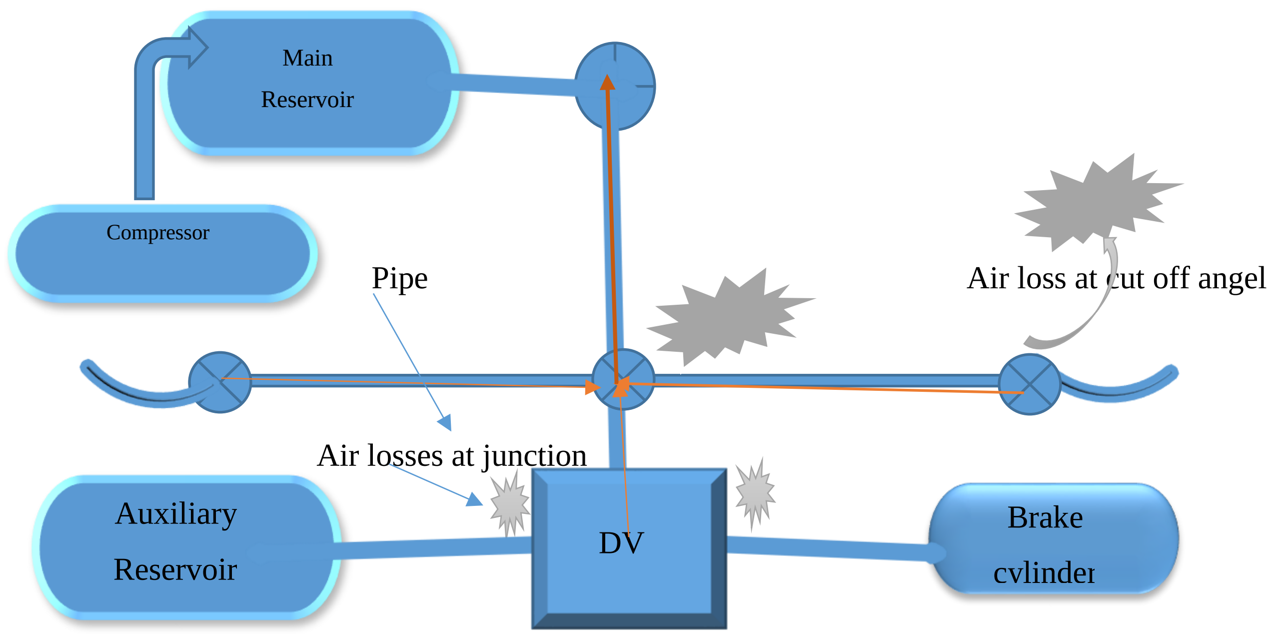

Figure 12 Sample of losses at brake components

3.3 Numerical Analysis of the uncontrolled braking force

The specific loading condition in this analysis is braking of the locomotive running with a maximum velocity of 120 km/h to standstill with the temperature of the surrounding is 20°C. The assumed temperature is average environmental temperature of the operating route of the locomotive in the region.

The weight distribution of the locomotive considered is equally distributed between on both the front and rear bogies. Each bogie consists of two wheel sets. Every wheel set is equipped with two mechanical disc brakes on locomotive bogie. And one mechanical disc brake on motor bogies axle. Here, is only 1/8 of the whole brake force is applied to one disc from the locomotive.

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

$$v(t) = v_o \quad \text{(3.11)}$$

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

$$\omega(t) = \frac{v(t)}{r_d} \quad \text{(3.12)}$$

$$\omega(t) = \frac{33.33}{0.36} \frac{rad}{s}$$

$$\omega(t) = 92.58 \frac{rad}{s}$$

Where:

$v(t)$ Velocity of the locomotive

v_o= Initial running velocity of the locomotive

$\omega(t)$ = angular velocity of the disc at any time

r_d= the radius of disc

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 locomotive as follows.

$$v_{disc} = \frac{r_{disc}}{r_{wheel}} v(t) \quad \text{(3.13)}$$

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

$$s_b = v_o t_b - \frac{1}{2} a t_b \quad \text{(3.14)}$$

$$t_b = \frac{v_o}{a} \quad \text{(3.15)}$$

$$t_b = \frac{33.33}{1.4} \text{ s} = 23.8\text{s}$$

3.3 Heat Transfer coefficient

The heat dissipation from the free surfaces of the disc and pad to the surrounding air is described by both convection and radiation.

To calculate the convective film coefficient as a function of the railway vehicle velocity v, the following formula should be used.

$$h = \frac{0.037K_a}{2r_{disc}} * Re^{0.8}Pr^{0.33} \text{.......................... (3.16)}$$

Where

h Heat transfer coefficient

Re Reynolds number

Pr Prandlt number

K_a Thermal conductivity of air

But the Reynolds number Re and Prandtl number Pr are given as the following formula

$$Re = \frac{2\rho_a r_{disc}\, V}{\mu_{va}}$$

$$Pr = \frac{C_{pa}\mu_{air}}{K_{air}}$$

Here, properties of air from table (3.5)

ρ_{air} Density of air ($\rho_{air} = 1.205\ \frac{Kg}{m^3}$)

r_{disc} Is the radius of the disc ($r_{disc} = 0.360m$)

V Is the train speed$V = 33.33\,\frac{m}{s}$

μ_{air} Dynamic viscosity of air ($\mu_{va} = 15.11 * 10^{-6}\,\frac{m^2}{s}$)

K_{air} Thermal conductivity of air ($K_a = 0.02441\,\frac{w}{m\,K}$)

C_{pa} Specific heat capacity of air, ($C_{pa} = 1.005\ \frac{KJ}{Kg\,K}$)

Therefore;

$$Re = \frac{2\rho_a r_{disc}\, V}{\mu_a}$$

$$Re = \frac{2*1.205\,\frac{Kg}{m^3}*0.360m*33.33\frac{m}{s}}{15.11*10^{-6}\frac{m^2}{s}}$$

$$Re = 1.9075 * 10^6$$

$$\mathrm{Pr} = \frac{C_{pa}\mu_{air}}{K_{air}}$$

$$\mathrm{Pr} = \frac{1.005\,\frac{KJ}{Kg\,K}15.11*10^{-6}\frac{m^2}{s})}{0.02441\frac{w}{m\,K}}$$

$$\mathrm{Pr} = 0.6221$$

Finally heat transfer coefficient h is equals to

$$h = \frac{0.037K_a}{2R} * Re^{0.8}Pr^{0.33}$$

$$h = \frac{0.037*0.02441\frac{w}{m\,K}}{2*0.360m} * (1.9075 * 10^6)^{0.8}0.6221^{0.33}$$

$$h = 77.78\frac{w}{m^2k}$$

3.4 Braking energy

The total energy of the train which is to be converted to heat while braking should be determined. Here the change in rotational and translational kinetic energy that the train undergoes during braking should be precisely determined.

Assuming that the train slows down or stops at the end of brake application, the change in translational kinetic energy expressed as:

$$\Delta KE = \frac{1}{2}m{v_o}^2 - \frac{1}{2}m{v_f}^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.17)}$$

Where

KE , Kinetic energy [J]

m, Mass of train [kg]

v_o , Initial speed [m/s]

v_f , Final speed [m/s]

The change in rotational kinetic energy of all rotating parts is given by:

$$\Delta Ek_r = \frac{1}{2}I{\omega_o}^2 - \frac{1}{2}I{\omega_f}^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{(3.18)}$$

Where

Ek_r , Rotational kinetic energy [J]

I, Momenta of inertia $[\frac{J}{m^2}]$

ω, Angular speed $[\frac{rad}{s}]$

The total energy to be converted to heat while braking is the sum of the above three energies

$$E_b = \Delta KE + \Delta Ekr \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.19)$$

$$E_b = (\frac{1}{2} m{v_o}^2 - \frac{1}{2} m v_f^2) + (\frac{1}{2} I{\omega_o}^2 - \frac{1}{2} I\omega_f^2)$$

The moment of inertia of the rotating parts can calculated as

$$I = \sum_{i=1}^{n} m_i{r_i}^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.20)$$

Assuming that all the rotating parts have the same radius of rotation as the wheel, the expression for the braking energy could be rewritten as:

$$E_b = \frac{1}{2} m{v_o}^2 - \frac{1}{2} m v_f^2 + \frac{1}{2} {r_w}^2 \sum_{i=1}^{n} m_i\, ({\omega_o}^2 - \omega_f^2) \ldots\ldots\ldots\ldots (3.21)$$

In most analysis the contribution of the rotating masses of the wheel set and discs is taken to be 10% the tare weight of the axle load of the locomotive.

Therefore; $E_K = \frac{1}{2} m v_o^2 + \frac{1}{2} m\omega_o^2$

$$E_K = \frac{1}{2} m v_o^2 \left[1 + \frac{I}{m r_o^2}\right] \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \text{Since; } \omega_o^2 = \frac{v_o^2}{r_o^2}$$

$$E_K = \frac{1}{2}(1.1) m v_o^2$$

Generally, the total braking energy will be

$$E_K = \frac{1}{2}(1.1) M v_o^2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.22)$$

Assumption: All the rotating parts like discs shafts etc. have the same radius of rotation as the wheel.

The rotating parts mass are 10 % of the tare weight.

The train travel at level root condition

This energy is characterized by a total heating energy which is induced to 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°C to 800°C.

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.

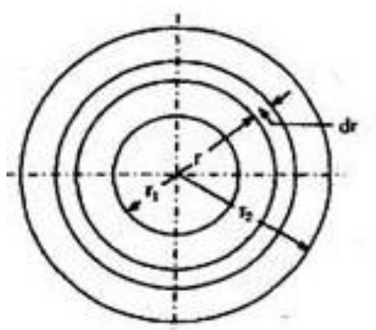

Figure 13 Brake disc

Let P = normal pressure, pa

The total axial load, F

$$F = \int_{r_1}^{r_2} 2\pi r dr P \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.23)$$

And total friction torque, T

$$T = \int_{r_1}^{r_2} 2\mu\pi r^2 dr P \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3.24)$$

The above two equation can be integrated using either of the assumption.

- The intensity of pressure is uniform, or P= constant
- The rate of wear is uniform, p_r= constant

Uniform pressure (p = constant)

When the two surface have perfect contact, the pressure P is uniform over the entire surface. The intensity pressure becomes;

$$P = \frac{F}{\pi({r_2}^2 - {r_1}^2)}$$

$$F = P\pi\,({r_2}^2 - {r_1}^2) \quad \text{..} \quad (3.25)$$

Therefore the total frictional torque, T

$$T = \int_{r_1}^{r_2} 2\mu\pi P r^2 dr$$

$$= \int_{r_1}^{r_2} 2\mu\pi P r^2 dr$$

$$= \frac{2}{3}\mu F \frac{(r_2^3 - r_1^3)}{(r_2^2 - r_1^2)}$$

$$T \quad = \quad \frac{2}{3}\mu F \frac{(r_2^3 - r_1^3)}{(r_2^2 - r_1^2)} \text{..} \quad (3.26)$$

3.4.3 Braking Force

Considering a vehicle having M mass, the following relation gets particular expressions for the case of being equipped with brake shoes.

The total work done during the whole brake cycle is equal with the total heat generated. That is

$$Q_{gen} = E_b = \int_0^{t_b} p(t)\,dt$$

$$E_K = \frac{1}{2}(1.1)Mv_o^2 \quad = \quad \int_0^{t_b} p(t)\,dt$$

$$= \quad \int_0^{t_b} 2F_{fr}V(t)\,dt$$

$$= \quad 2F_{fr}\int_0^{t_b} V(t)\,dt$$

$$= \quad 2F_{fr}\frac{r_{disc}}{r_{wheel}}\left(v_o t_b - \frac{1}{2}at_b^2\right)$$

The brake force(F_{disc}),that work on the disc to retard the locomotive is that;

$$F_{disc} = \frac{\frac{1}{2}MV_O{}^2(1.1)}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} \text{ ………………....………..………. (3.27)}$$

3.4.4 Heat generated

The total heat generated during the whole brake cycle and induced to disc pad is equal to.

$$Q_{gen} = F_{fr}\frac{r_{disc}}{r_{wheel}}\left(v_o t_b - \frac{1}{2}at_b^2\right) \text{………………………..…… (3.26)}$$

3.4.5 Heat flux

Heat Flux is defined as the amount of heat transferred per unit effective area, from or to a surface. Heat flux entering the Disc 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. Generally,the thermal conductivity of material of brake pads is smaller than that of disc/(kp< kd).

$$\mathbf{q_{flux}} = \frac{Q_{gen}}{\pi(r_{max}^2 - r_{min}^2)} \text{………………………………………... (3.27)}$$

3.5 Technical parameters and specification of freight train

Table 10 Technical parameters of the locomotive

Item	**Parameter**
Purpose	Freight transport
Gauge	1435mm
Wheel diameter	1000mm
Starting tractive effort	520 KN (23t), 570 KN (25t)
Continuous rated speed of locomotive	100 Km/h (23t), 120 Km/h(25t)
Wheel Base	2250mm＋2000 mm
Locomotive Total Wheel Base	16260 mm
Maximum height (upon pantograph dropping):	4,760mm
Power supply system	AC 25 kV, 50 Hz
Mechanical brake	Disc brake
Maximum continuous power at the wheel rim	7,200 kW
Maximum electric braking force	370KN(23t), 400 KN(25t)
Ambient temperature	22°C

Table 11 Material properties of brake disc

Material Properties	**Disc**
Thermal conductivity, k (W/m k)	5
Density, ρ (kg/m3)	3700
Mass of the brake disc, (Kg)	33.038
Specific heat, C (J/Kg k)	650
Poisson's ratio, v	0.13
Thermal expansion, α (10-6 / K)	0.15
Elastic modulus, E (GPa)	254
Tensile yield strength, (Mpa)	314
Ultimate tensile strength, (Mpa)	410

Table 12 Properties of Air

Temperature - t (^{o}C)	Density - ρ - (kg/m^3)	Specific heat capacity -c_p (kJ/kg.K)	Thermal conductivity - k (W/m.K)	Kinematic viscosity - ν - x 10^{-6} (m^2/s)	Expansion coefficient - b - x 10^{-3} (1/K)
20	1.205	1.005	0.0257	15.11	3.43
40	1.127	1.0049	0.0271	16.97	3.20

Table 13 Data for calculating heat flux

Item	**Values**
Mass of the vehicle – M [kg]	70,000
Start velocity – v_0 [m/s]	33.33
Deceleration – a [m/s^2]	1.4
Braking time – t_b [s]	23.8
Effective radius of the braking disc – r_d [m]	0.2475
Radius of the wheel – r_w [m]	0.5
Friction coefficient disc/pad – μ [/]	0.30
Surface of the braking pad A_p – [cm^2]	225

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.[10]

However, this paper did not consider tribological parameters of braking processes. According to DIN EN ISO1183 the value μ = 0.30 of the friction coefficient for materials of the disc and the pad for locomotives is taken as the recommended value. [10]

3.5.1 Determination of friction surface of disc

According to UIC 541-3 Standard, a Sinter material brake pad and Grey Cast Iron Disc with dimension tabulated in table 8 is used to calculate friction area swept on the disc by the brake pad.

Table 14 Dimension of disc and pad for ventilated disc brake

Description	**Disc**	**Pad**
Outer radius (R_o), [m]	0.320	0.320
Inner Radius (R_i)	0.175	0.175
Bolts hole Radius(ϕ13*9)		
Inner bore radius [m]	0.118	
Friction face Thickness (L)[m]	0.022	0.0265
Effective friction radius (r_{disc})[m]	0.2475	-
Friction surface (A_b, A_P)[m^2]	0.225	0.225

All technical parameters summarized in the above tables are taken from the specifications of electric locomotives and Rolling stocks obtained from the Ethiopian Railway Corporation.

3.6 Analyze the braking force and its effect on braking action at normal operation condition

At normal operation condition, the required maximum braking force during the whole braking cycle is calculated from equation (3.27)

$$\mathrm{F_{disc}} = \frac{\frac{1}{2}\mathrm{MV_O}^2(1.1)}{2\frac{\mathrm{r_{disc}}}{\mathrm{r_{wheel}}}\left[\mathrm{v_o t_b} - \frac{1}{2}\mathrm{a t_b^2}\right]}$$

$$\mathrm{F_{disc}} = \frac{\frac{1}{2}*70{,}000\mathrm{Kg}*\left(33.33\frac{\mathrm{m}}{\mathrm{s}}\right)^2(1.1)}{2*\frac{0.36}{0.50}\left[33.33\frac{\mathrm{m}}{\mathrm{s}}*23.8 - \frac{1}{2}*1.4\frac{\mathrm{m}}{\mathrm{s}^2}*(23.8\mathrm{s})^2\right]}$$

$$\mathrm{F_{disc}} = 74.861\mathrm{KN}$$

Also the maximum brake cylinder pressure at maximum braking condition is 435KPa

We Know that

$$\mathrm{F_{disc}} = P_{bc} * A_{bc} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. (3.28)$$

$$A_{bc} = \frac{\mathrm{F_{disc}}}{P_{bc}}$$

$$A_{bc} = \frac{\mathrm{F_{disc}}}{P_{bc}} = \frac{74.861\mathrm{KN}}{435KPa}$$

$$A_{bc} = 0.1721m^2$$

Friction force ($\mathrm{F_{fr}}$)

The corresponding friction force created on disc can be calculated using equation

$$\mathrm{F_{fr}} = \mu\mathrm{F_{disc}}$$

$$\mathrm{F_{fr}} = \mu * P_{bc} * A_{bc}$$

$$\mathrm{F_{fr}} = 0.30 * 435\mathrm{KPa} * 0.1721\mathrm{m}^2$$

$$\mathrm{F_{fr}} = 22.459\mathrm{KN}$$

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

Braking power or heat generated **(**Q_{gen}**)**

$$Q_{gen} = F_{fr}\frac{r_{disc}}{r_{wheel}}\left(v_o t_b - \frac{1}{2}at_b^2\right)$$

$$Q_{gen} = 22.459KN\frac{0.36}{0.50}\left(33.33\frac{m}{s} * 23.8s - \frac{1}{2}1.4\frac{m}{s^2} * (23.8s)^2\right)$$

$$Q_{gen} = 6415.586KW$$

The total heat generated is distributed to each disc of the locomotive which has 8 disc in locomotive. Therefore;

$$Q_{gen/disc} = \frac{6415.586KW}{8}$$

$$Q_{gen/disc} = 801.948KW$$

Heat flux

Heat flux induced in to the disc is calculated as follows

$$q = \frac{Q_{gen/disc}}{\pi({r_{max}}^2 - {r_{min}}^2)}$$

$$q = \frac{801.948KW}{\pi(0.32^2 - 0.175^2)m^2}$$

$$q = 3556.312\frac{KW}{m^2}$$

r_{max} Maximum effective radius of the disc (m)

r_{min} Minimum effective radius of the disc (m)

This heat flux is imposed in to two faces of the disc, therefore each face of the disc will imposed half of the total heat flux.

$$q/_{face} = \frac{3556.312\frac{KW}{m^2}}{2}$$

$$q/_{face} = 1778.156\frac{KW}{m^2}$$

Table 15 Summary resulting braking force and its effects due to normal braking condition

Normal braking condition			
Cylinder pressure [KPa]	435	Braking time (s)	23.8
Braking force [KN]	74.861	Speed ($\frac{\mathrm{m}}{\mathrm{s}}$)	33.33
Friction force [KN]	22.459	Deceleration ($\frac{\mathrm{m}}{\mathrm{s}^2}$)	1.4
Braking power [W]	801,948	Angular speed ($\frac{\mathrm{rad}}{\mathrm{s}}$)	92.58
Heat flux $\frac{W}{m^2}$	1,778,156	Brake cylinder area (m^2)	0.1721

3.7 Analyze the uncontrolled braking force and its effect on braking action at different condition

Because of the complexity of air charging and discharging in the system, and also operator negligence in operation, it is difficult to calculate the specific amount of air charging and discharging due to brake application. So that in this paper I take my assumption and figure out for two specific case which are overcharging and air leakages in the pipe as follows.

3.7.1 Numerical Analysis of uncontrolled Braking Force Due to Air Overcharging

The standard brake pipe pressure for a freight train is 600 KPa. Due to negligence in operation or improper inspection before coupling cars, the brake pipe might get overcharged.

But the complexity of the air charging in the system, it is difficult to calculate the specific the amount of overcharging air in the system. So that I take my assumption and I tried to quantify the pressure difference in the system by assuming 2%, 4%, 6%, 8% and 10% brake pipe overcharge.

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

Assuming 2% brake pipe pressure overcharge, the corresponding brake cylinder pressure is obtained as follows:

$$\mathrm{P_{bc,o}} = \left(\frac{\mathrm{V_f}}{\mathrm{V_z}} * \mathrm{r_{max}} - 100\right) + \mathrm{P_o} \quad \mathrm{KPa}$$

Where

$\mathrm{P_{bc,o}}$, brake cylinder pressure duo to overcharging

$\mathrm{P_o}$, Overcharging brake pipe pressure

$\mathrm{r_{max}}$, Max air pressure reduction =165KPa

$$\mathrm{P_o} = 0.02 * 600\mathrm{KPa} = 12\mathrm{KPa}$$

$$\mathrm{P_{bc}} = ((3.25 * 165 - 100) + 12\ )\mathrm{KPa} = 448.25\mathrm{KPa}$$

Assuming 2% overcharged brake pipe the corresponding braking force can be calculated as following

Braking force (F)

$$\mathrm{F_{disc}} = P_{bc} * A_{bc}$$

$$\mathrm{F_{disc}} = 448.25KPa * 0.1721m^2$$

$$\mathrm{F_{disc}} = 77.144KN$$

Friction force ($\mathrm{F_{fr}}$)

The corresponding friction force created on disc can be calculated using equation

$$\mathrm{F_{fr}} = \mu \mathrm{F_{disc}}$$

$$\mathrm{F_{fr}} = \mu * P_{bc} * A_{bc}$$

$$\mathrm{F_{fr}} = 0.30 * 448.25KPa * 0.1721m^2$$

$$\mathrm{F_{fr}} = 23.143\mathrm{KN}$$

Braking power or heat generated **($\mathrm{Q_{gen}}$)**

$$\mathrm{Q_{gen}} = \mathrm{F_{fr}} \frac{\mathrm{r_{disc}}}{\mathrm{r_{wheel}}} \left(\mathrm{v_o t_b} - \frac{1}{2}\mathrm{a t_b^2}\right)$$

$$\mathrm{Q_{gen}} = 23.143\mathrm{KN} * \frac{0.36}{0.50}\left(33.33\frac{\mathrm{m}}{\mathrm{s}} * 23.8\mathrm{s} - \frac{1}{2}1.4\frac{\mathrm{m}}{\mathrm{s}^2} * (23.8s)^2\right)$$

$$\mathrm{Q_{gen}} = 6611.005\mathrm{KW}$$

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

The total heat generated is distributed to each disc of the locomotive which has 8 disc in locomotive.

Therefore,

$$Q_{gen/disc} = \frac{6611.005KW}{8}$$

$$\mathrm{Q_{gen/disc}} = 826.376\mathrm{KW}$$

Heat flux

Heat flux induced in to the disc is calculated as follows

$$q = \frac{\mathrm{Q_{gen/disc}}}{\pi({r_{max}}^2 - {r_{min}}^2)}$$

$$q = \frac{826.376KW}{\pi(0.32^2 - 0.175^2)m^2}$$

$$q = 3664.637\frac{KW}{m^2}$$

Where

q Heat flux $\frac{W}{m^2}$

$\mathrm{Q_{gen/disc}}$ Heat generated per disc (W)

r_{max} Maximum effective radius of the disc (m)

r_{min} Minimum effective radius of the disc (m)

This heat flux is imposed in to two faces of the disc, therefore each face of the disc will imposed half of the total heat flux.

$$q/_{face} = \frac{3664.637\frac{KW}{m^2}}{2}$$

$$q/_{face} = 1832.318\frac{KW}{m^2}$$

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

Assuming 4% brake pipe pressure overcharge, the corresponding brake cylinder pressure is obtained as follows:

$$P_{bc,o} = \left(\frac{V_f}{V_z} * r_{max} - 100\right) + P_o \quad KPa \ldots\ldots\ldots\ldots\ldots\ldots \text{(a)}$$

Where

$P_{bc,o}$, brake cylinder pressure duo to overcharging

P_o , Overcharging brake pipe pressure

r_{max} , Max air pressure reduction =165KPa

$$P_o = 0.04 * 600KPa = 24KPa$$

$$P_{bc,o} = ((3.25 * 165 - 100) + 24\,)KPa = 460.25KPa$$

Braking force (F)

$$F_{disc} = P_{bc} * A_{bc}$$

$$F_{disc} = 460.25KPa * 0.1721m^2$$

$$F_{disc} = 79.209KN$$

Friction force (F_{fr})

The corresponding friction force created on disc can be calculated using equation

$$F_{fr} = \mu F_{disc}$$

$$F_{fr} = \mu * P_{bc} * A_{bc}$$

$$F_{fr} = 0.30 * 460.25KPa * 0.1721m^2$$

$$F_{fr} = 23.763KN$$

Braking power or heat generated **(**Q_{gen}**)**

$$Q_{gen} = F_{fr}\frac{r_{disc}}{r_{wheel}}\left(v_o t_b - \frac{1}{2}at_b^2\right)$$

$$Q_{gen} = 23.763KN * \frac{0.36}{0.50}\left(33.33\frac{m}{s} * 23.8s - \frac{1}{2}1.4\frac{m}{s^2} * (23.8s)^2\right)$$

$$Q_{gen} = 6787.987KW$$

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

The total heat generated is distributed to each disc of the locomotive which has 8 disc in locomotive.

$$Q_{gen/disc} = \frac{6787.987KW}{8}$$

$$\mathrm{Q_{gen/disc}} = 848.499\mathrm{KW}$$

Heat flux

Heat flux induced in to the disc is calculated as follows

$$q = \frac{\mathrm{Q_{gen/disc}}}{\pi({r_{max}}^2 - {r_{min}}^2)}$$

$$q = \frac{848.499KW}{\pi(0.32^2 - 0.175^2)m^2}$$

$$q = 3762.742\frac{KW}{m^2}$$

Where

q Heat flux $\frac{W}{m^2}$

$\mathrm{Q_{gen/disc}}$ Heat generated per disc (W)

r_{max} Maximum effective radius of the disc (m)

r_{min} Minimum effective radius of the disc (m)

This heat flux is imposed in to two faces of the disc, therefore each face of the disc will imposed half of the total heat flux.

$$q/_{face} = \frac{3762.742\frac{KW}{m^2}}{2}$$

$$q/_{face} = 1881.371\frac{KW}{m^2}$$

Table 16 Summary different braking force and its effects from brake system overcharge

Extent of overcharge	P_o, Overcharging brake pipe pressure[kPa]	Cylinder pressure [KPa]	Braking force [KN]	Friction force [KN]	Braking power [KW]	Heat flux [$\frac{KW}{m^2}$]
2%	12	448.250	77.147	23.143	826.376	1832.318
4%	24	460.250	79.209	23.762	848.498	1881.371
6%	36	472.250	81.274	24.382	870.621	1930.424
8%	48	484.250	83.339	25.002	892.743	1979.476
10%	60	496.250	85.405	25.621	914.867	2028.529

3.7.2 Numerical Analysis of uncontrolled Braking Force Due to Air Leakages

According to CSX Air brake and train Handling rule book, CSX transportation**,** 2010 the maximum tolerable brake pipe leakage is 5Psi=34.4737KPa per minute. Besides, as it is shown earlier, the minimum valid brake pressure reduction is 40KPa and the maximum valid brake pressure reduction is 165KPa. There is 125KPa range between the minimum and maximum valid brake pipe pressure reduction.

Here will analyze the effect of force when the brake pipe leakages increase with some extent from the standards by taking assumption and I tried to quantify the pressure difference in the system by assuming 5%, 10%, 15%, 20% and 25% air leakages in the system.

Table 17 Air pressure reduction due to air leakages

Assumption brake pipe leakage	New brake pipe pressure reduction (KPa) r_{new}
0%	40
5%	40+0.05*600 =70
10%	40+0.1*600 = 100
15%	40+0.15*600 =130
20%	40+ 0.2*600 =160
25%	40+.25*600 = 190

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

Assuming 5% air leakages in the system, the corresponding brake cylinder pressure and braking forces are obtained as follows

$$\mathrm{P_{bc}} = \left(\frac{\mathrm{V_f}}{\mathrm{V_z}} * r_{new} - 100\right) \mathrm{KPa}$$

$$\mathrm{P_{bc}} = 3.25 * 70 - 100 = 127.5\mathrm{KPa}$$

Braking force (F)

$$\mathrm{F_{disc}} = P_{bc} * A_{bc}$$

$$\mathrm{F_{disc}} = 127.5KPa * 0.1721m^2$$

$$\mathrm{F_{disc}} = 21.943KN$$

Friction force ($\mathrm{F_{fr}}$)

The corresponding friction force created on disc can be calculated using equation

$$\mathrm{F_{fr}} = \mu \mathrm{F_{disc}}$$

$$\mathrm{F_{fr}} = \mu * P_{bc} * A_{bc}$$

$$\mathrm{F_{fr}} = 0.30 * 127.50KPa * 0.1721m^2$$

$$\mathrm{F_{fr}} = 6.583\mathrm{KN}$$

Braking power or heat generated **(**$\mathrm{Q_{gen}}$**)**

$$\mathrm{Q_{gen}} = \mathrm{F_{fr}} \frac{\mathrm{r_{disc}}}{\mathrm{r_{wheel}}} \left(\mathrm{v_o t_b} - \frac{1}{2} \mathrm{a t_b^2}\right)$$

$$\mathrm{Q_{gen}} = 6.5836\mathrm{KN} * \frac{0.36}{0.50} \left(33.33 \frac{\mathrm{m}}{\mathrm{s}} * 23.8\mathrm{s} - \frac{1}{2} 1.4 \frac{\mathrm{m}}{\mathrm{s}^2} * (23.8s)^2\right)$$

$$\mathrm{Q_{gen}} = 1880.430\mathrm{KW}$$

The total heat generated is distributed to each disc of the locomotive which has 8 disc in locomotive.

$$Q_{gen/disc} = \frac{1880.430KW}{8}$$

$$\mathrm{Q_{gen/disc}} = 235.054\mathrm{KW}$$

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

Heat flux

Heat flux induced in to the disc is calculated as follows

$$q = \frac{Q_{gen/disc}}{\pi({r_{max}}^2 - {r_{min}}^2)}$$

$$q = \frac{235.054KW}{\pi(0.32^2 - 0.175^2)m^2}$$

$$q = 1042.367\frac{KW}{m^2}$$

Where

q Heat flux $\frac{W}{m^2}$

$Q_{gen/disc}$ Heat generated per disc (W)

r_{max} Maximum effective radius of the disc (m)

r_{min} Minimum effective radius of the disc (m)

This heat flux is imposed in to two faces of the disc, therefore each face of the disc will imposed half of the total heat flux.

$$q/_{face} = \frac{1042.367\frac{KW}{m^2}}{2}$$

$$q/_{face} = 521.183\frac{KW}{m^2}$$

Assuming 25% air leakages in the system, the corresponding brake cylinder pressure and braking forces are obtained as follows

$$P_{bc} = \left(\frac{V_f}{V_z} * r_{new} - 100\right) \text{KPa}$$

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

$$\mathrm{P_{bc}} = 3.25 * 190 - 100 = 517.5\mathrm{KPa}$$

Braking force (F)

$$\mathrm{F_{disc}} = P_{bc} * A_{bc}$$

$$\mathrm{F_{disc}} = 517.5KPa * 0.1721m^2$$

$$\mathrm{F_{disc}} = 89.062KN$$

Friction force ($\mathrm{F_{fr}}$)

The corresponding friction force created on disc can be calculated using equation

$$\mathrm{F_{fr}} = \mu \mathrm{F_{disc}}$$

$$\mathrm{F_{fr}} = \mu * P_{bc} * A_{bc}$$

$$\mathrm{F_{fr}} = 0.30 * 517.5KPa * 0.1721m^2$$

$$\mathrm{F_{fr}} = 26.719\mathrm{KN}$$

Braking power or heat generated **(**$\mathrm{Q_{gen}}$**)**

$$\mathrm{Q_{gen}} = \mathrm{F_{fr}} \frac{\mathrm{r_{disc}}}{\mathrm{r_{wheel}}}\left(\mathrm{v_o t_b} - \frac{1}{2}\mathrm{a t_b^2}\right)$$

$$\mathrm{Q_{gen}} = 26.719\mathrm{KN} * \frac{0.36}{0.50}\left(33.33\frac{\mathrm{m}}{\mathrm{s}} * 23.8\mathrm{s} - \frac{1}{2}1.4\frac{\mathrm{m}}{\mathrm{s^2}} * (23.8s)^2\right)$$

$$\mathrm{Q_{gen}} = 7632.337\mathrm{KW}$$

The total heat generated is distributed to each disc of the locomotive which has 8 disc in locomotive.

$$Q_{gen/disc} = \frac{7632.337KW}{8}$$

$$\mathrm{Q_{gen/disc}} = 954.042\mathrm{KW}$$

Heat flux

Heat flux induced in to the disc is calculated as follows

$$q = \frac{Q_{gen/disc}}{\pi({r_{max}}^2 - {r_{min}}^2)}$$

$$q = \frac{954.042KW}{\pi(0.32^2 - 0.175^2)m^2}$$

$$q = 4230.785\frac{KW}{m^2}$$

r_{max} Maximum effective radius of the disc (m)

r_{min} Minimum effective radius of the disc (m)

This heat flux is imposed in to two faces of the disc, therefore each face of the disc will imposed half of the total heat flux.

$$q/_{face} = \frac{4230.785\frac{KW}{m^2}}{2}$$

$$q/_{face} = 2115.393\frac{KW}{m^2}$$

Table 18 Summary of calculated excessive brake pipe leakage and effects on braking system

% in to valid range	New pressure reduction [KPa]	Brake cylinder pressure[KPa]	Braking force[KN]	Friction force [KN]	Braking power [KW]	Heat flux [$\frac{KW}{m^2}$]
5%	70	127.5	21.857	6.577	234.132	519.140
10%	100	225	38.723	11.617	414.801	919.736
15%	130	322.5	55.502	16.651	594.58	1318.288
20%	160	420	72.282	21.685	774.295	1716.840
25%	190	517.5	89.062	26.719	954.042	2115.393

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

In both condition the created pressure in between in contact surface of pad and disc are calculated and summarized as below.

$$P_{disc} = \frac{F_{disc}}{A_P}$$

Where P_{disc} pressure in between disc and pad

A_P Area of pad = $0.255m^2$

Due to normal braking action

$$P_{disc} = \frac{F_{disc}}{A_P}$$

$$P_{disc} = \frac{74.861KN}{0.255m^2}$$

$$P_{disc} = 293.582KPa$$

Due to 2% of overcharging air in the system

$$P_{disc} = \frac{F_{disc}}{A_P}$$

$$P_{disc} = \frac{77.147KN}{0.255m^2}$$

$$P_{disc} = 302.525KN$$

Due to 10% of overcharging air in the system

$$\mathrm{P_{disc}} = \frac{\mathrm{F_{disc}}}{\mathrm{A_P}}$$

$$P_{disc} = \frac{85.405KN}{0.255m^2}$$

$$P_{disc} = 334.921KN$$

Due to 10% of air leakages in the system

$$P_{disc} = \frac{F_{disc}}{A_P}$$

$$P_{disc} = \frac{38.723KN}{0.255m^2}$$

$$P_{disc} = 151.854KN$$

Due to 25% of air leakages in the system

$$P_{disc} = \frac{F_{disc}}{A_P}$$

$$P_{disc} = \frac{89.062KN}{0.255m^2}$$

$$P_{disc} = 349.262KN$$

ANALYZE THE EFFECT OF UNCONTROLLED BRAKING FORCE ON BRAKE DISC DUE TO SERVICE BRAKEING

The above calculated value of pressure between disc and pad is summarized for both overcharging, leakages and normal condition in case of each assumption are below

Table 19 Created pressure on disc duo to different braking system condition

Different braking condition	Assumptions (%)	Force on the disk [KN]	Created pressure between disc and pad [KPa]
Due to brake pipe overcharge	2%	77.147	302.447
	4%	79.209	310.624
	6%	81.274	318.721
	8%	83.339	326.820
	10%	85.405	334.92
Due to excessive leakage	5%	21.857	85.713
	10%	38.723	151.855
	15%	55.502	217.655
	20%	72.282	303.067
	25%	89.062	349.263
Due to normal braking		74.861	293.357

Chapter 4

Finite Element Analysis and Modeling

For analyzing the process of braking, it is very useful to use FEM simulations based on the adopted analytical model for defining heat sources at surfaces which are in contact with disc rotor. This study will use the FEA software ANSYS 13.5 version to carry out transient thermal and structural analysis of a railway brake disc during different loading conditions.

The analysis will conducted in each three braking condition which are normal, overcharging and air leakage conditions and the numerically value is analyzed in chapter three and summarized as below

4.1 Assumptions and input parameters of analysis

- Material properties are isotropic and do not vary with environmental temperature.
- The convective heat transfer coefficient is constant throughout the braking time.
- The radiative heat transfer is neglected as the braking time is comparatively short and the magnitude is very small.
- Pressure in the contact area between brake block and wheel as well as between rail and wheel is uniform.
- The induced pressure to the disc and pad over the contact area is uniform.
- The wear on the contact surfaces is negligible

4.2 Defining Boundary conditions and loads

The boundary conditions set for this analysis are ambient temperature, initial wheel temperature, the convective heat transfer coefficient (film coefficient) and support conditions.

Whereas the loads to be applied to the disc are the heat flux, pressure at the contact areas angular velocity and gravitational acceleration. Hence, the boundary conditions and the load which are numerically calculated in chapter 3 applied to this analysis are listed in the following tables.

Table 20 Boundary condition

Ambient temperature (c^0)	20
Initial wheel temperature (c^0)	*30*
Film coefficient ($\frac{\mathrm{w}}{\mathrm{kgc}^0}$)	77.78
angular velocity ($\frac{\mathrm{rad}}{\mathrm{s}}$)	92.58

Table 21 Pressure loads imposed to disc at different braking system

Different braking condition	Assumptions(%)	Created pressure between disc and pad [KPa]
Due to brake pipe overcharge	2%	302.447
	6%	318.721
	10%	334.92
Due to excessive leakage	5%	85.713
	15%	217.655
	25%	349.263
Due to normal braking		293.357

Table 22 Heat flux imposed to disc at different braking system

Different braking condition	Assumptions	Induced Heat flux $\frac{KW}{m^2}$
Due to brake pipe overcharge	2%	1,832.318
	6%	1,930.424
	10%	2,028.529
Due to excessive leakage	5%	519.140
	15%	1,318.288
	25%	2,115.393
Due to normal braking		1,778.156

4.3 Modeling

Modeling of disc brake rotor is the primary action before doing of any type of analysis. CATIA software is the standard in the 3-D product design. The figure below is model of the disc brake in CATIA. The disc brake rotor designed by using the disc brake parameters data obtained from Addis Ababa Railway corporation train specification document and actual measuring of the disc brake rotor. The three dimensional model of the disc is then saved as igs file format for later use in the ANSYS workbench.

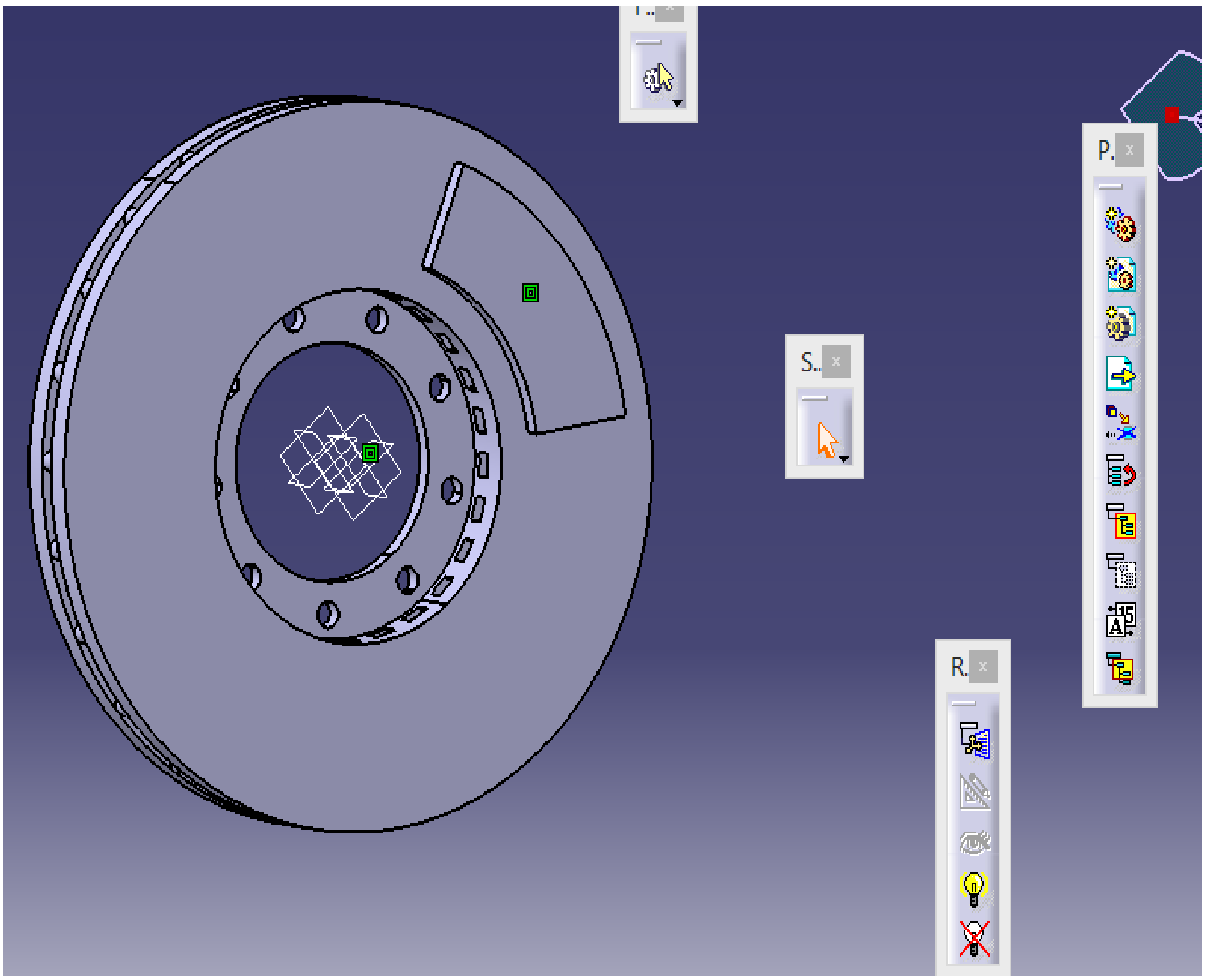

Figure 14 3D model of disc brake

The train brake disc are imposed for both thermal and structural loads at the same time, therefore to analyze both of thermal and structural effects, the ANSYS is integrated as the following.

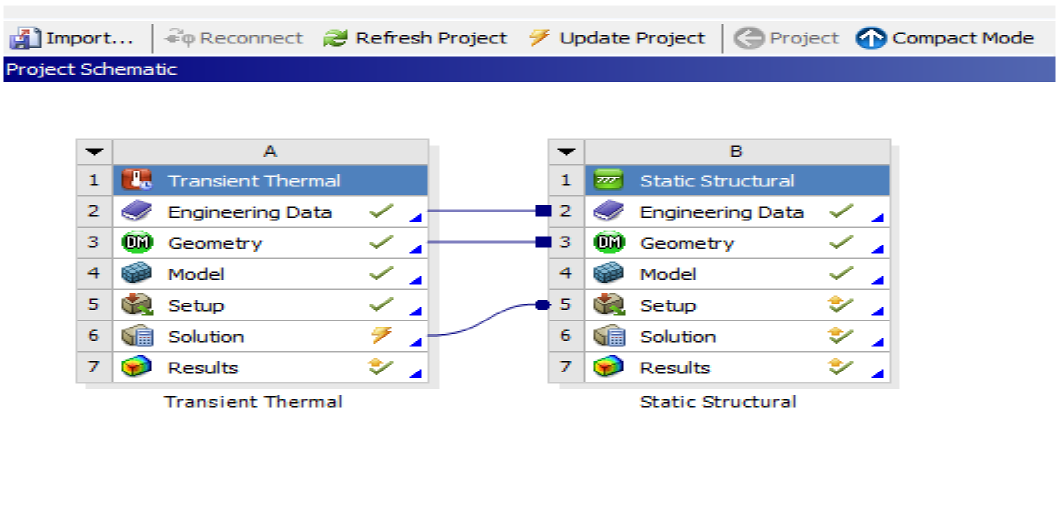

Figure 15 Integration transient thermal and structural

4.4 Mesh generation

The finite element model needs to be divided into small building blocks called elements that are connected at finite points called nodes. In this particular analysis the model is divided into many free triangular elements using the automatic mesh and sing generation command. The meshed disc is shown in fig (4.3) below.

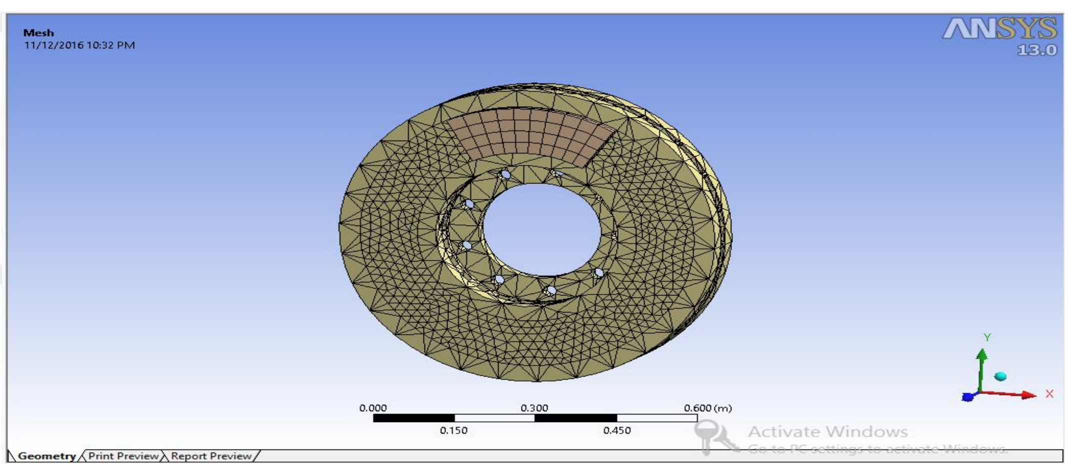

Figure 16 Meshing

4.5 Initial boundary condition of the disk

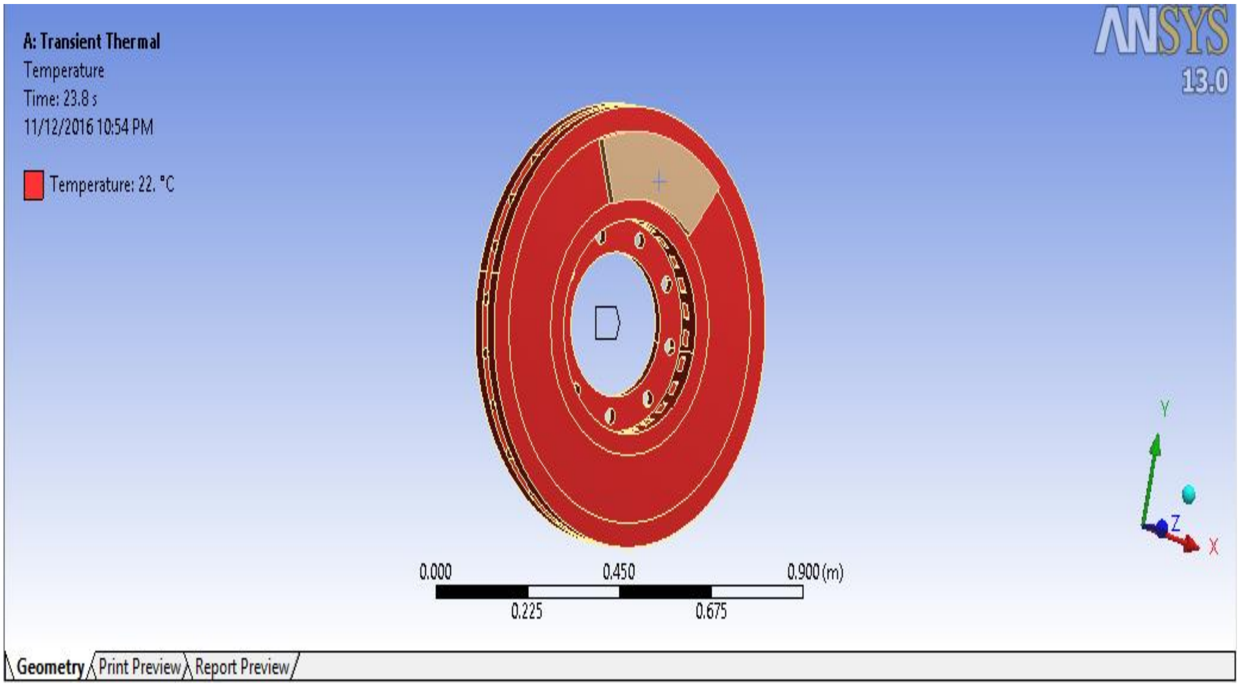

Figure 17 Initial disc temperature

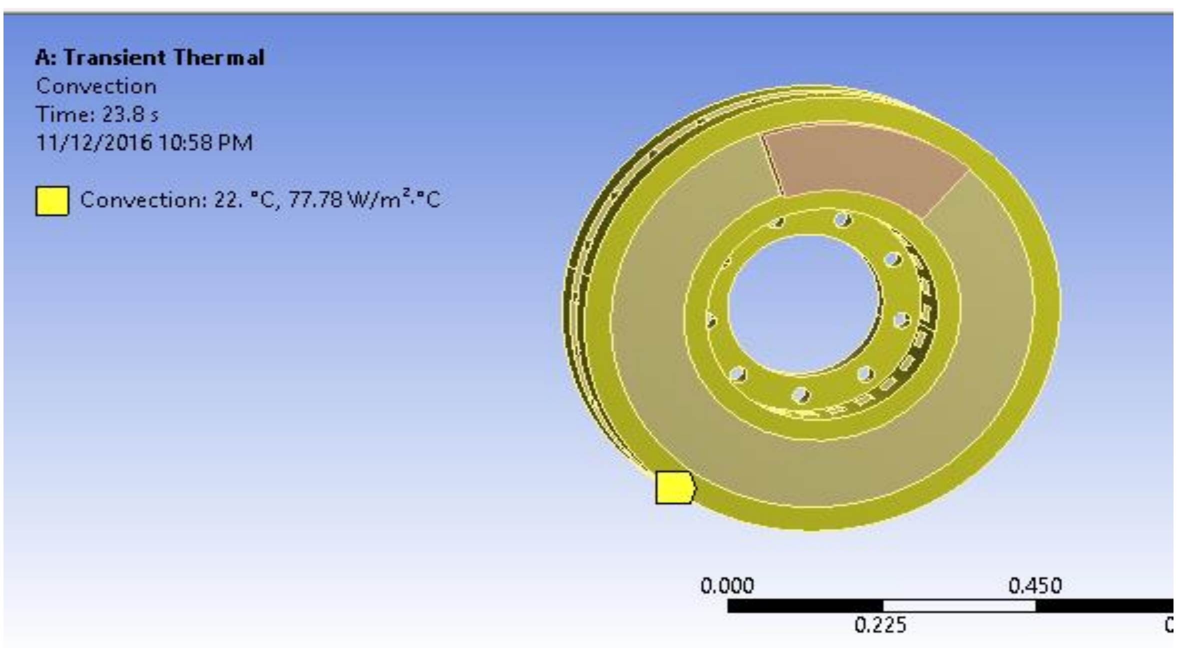

Figure 18 Heat coefficient

Chapter 5

5.1 Results Analysis and discussion

The results obtained from the combined transient thermal and structural analysis for each of braking system condition are presented and analyzed as follows. For all cases, the thermal and structural boundary conditions and loads are applied as illustrated in the figures below.

5.1.1 ANSYS input when the braking system is under overcharging condition

Due to 2% overcharging condition

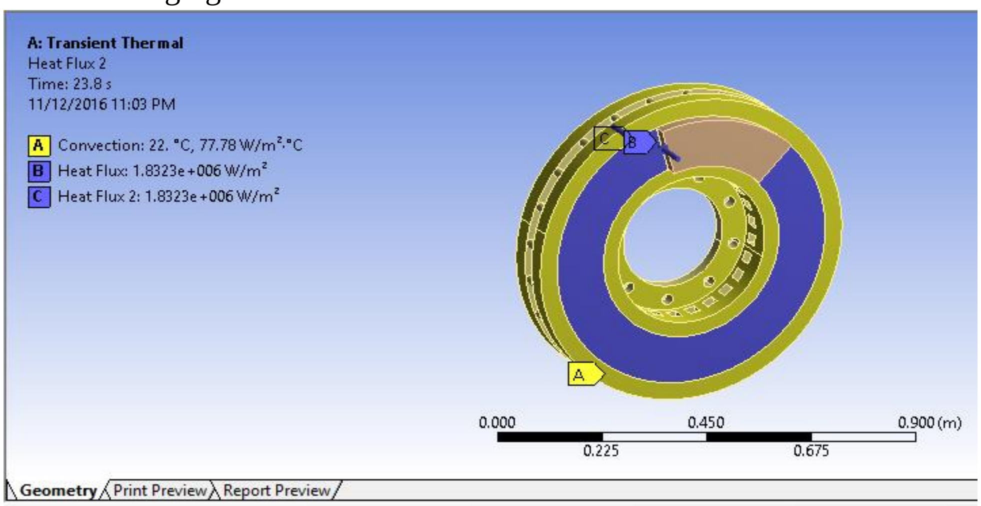

Figure 19 Imposed heat loads due to 2% overcharging

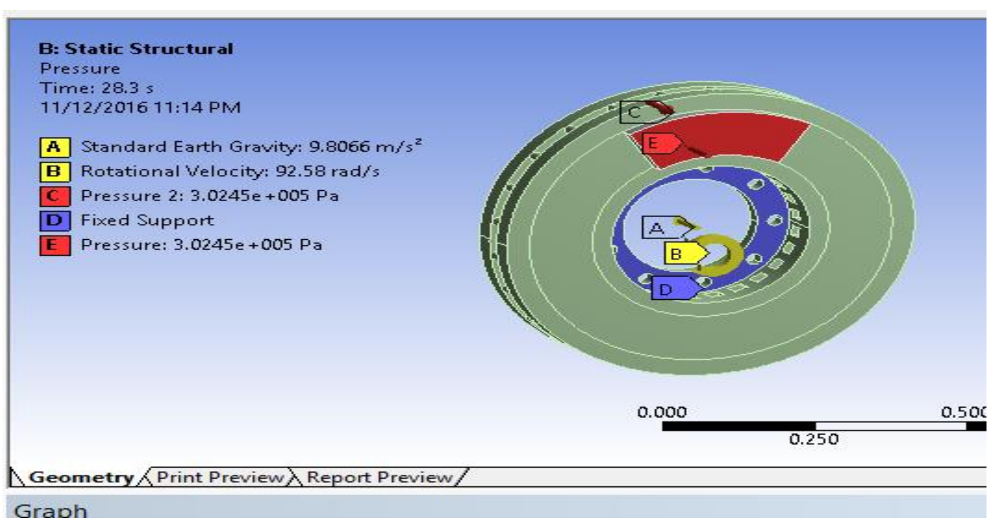

Figure 20 Imposed structural loads due to 2% overcharging

Due to 10% overcharging condition

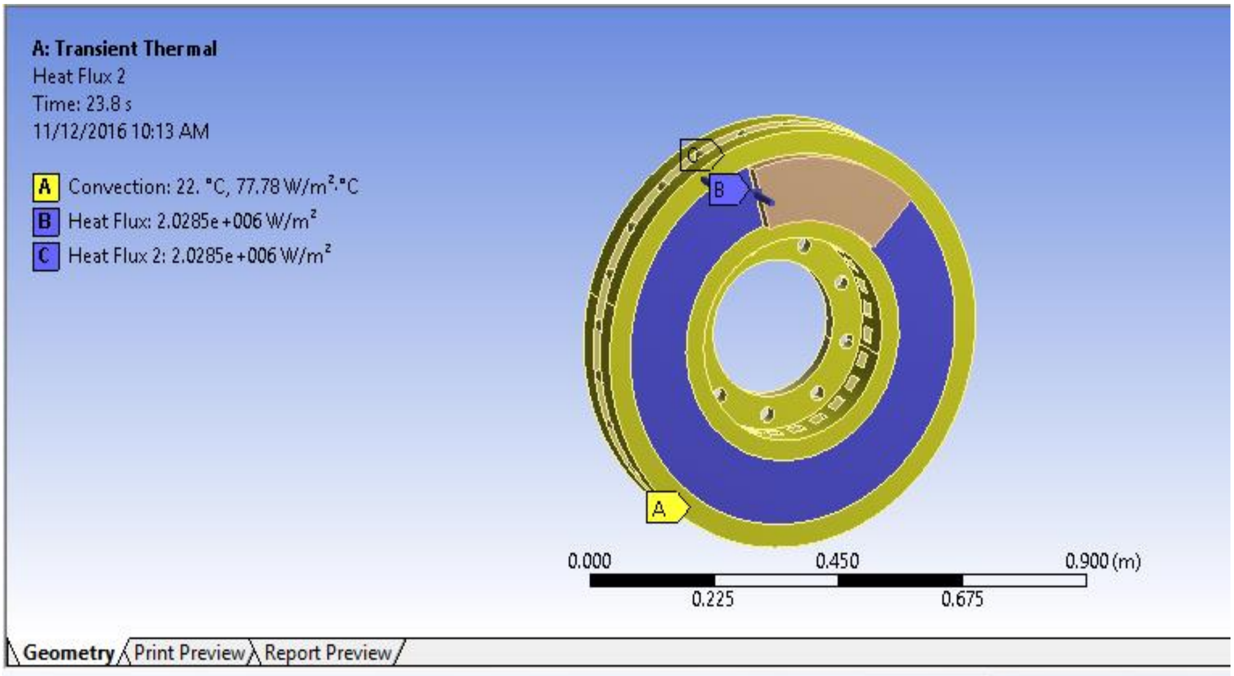

Figure 21 Imposed heat loads due to 10% overcharging

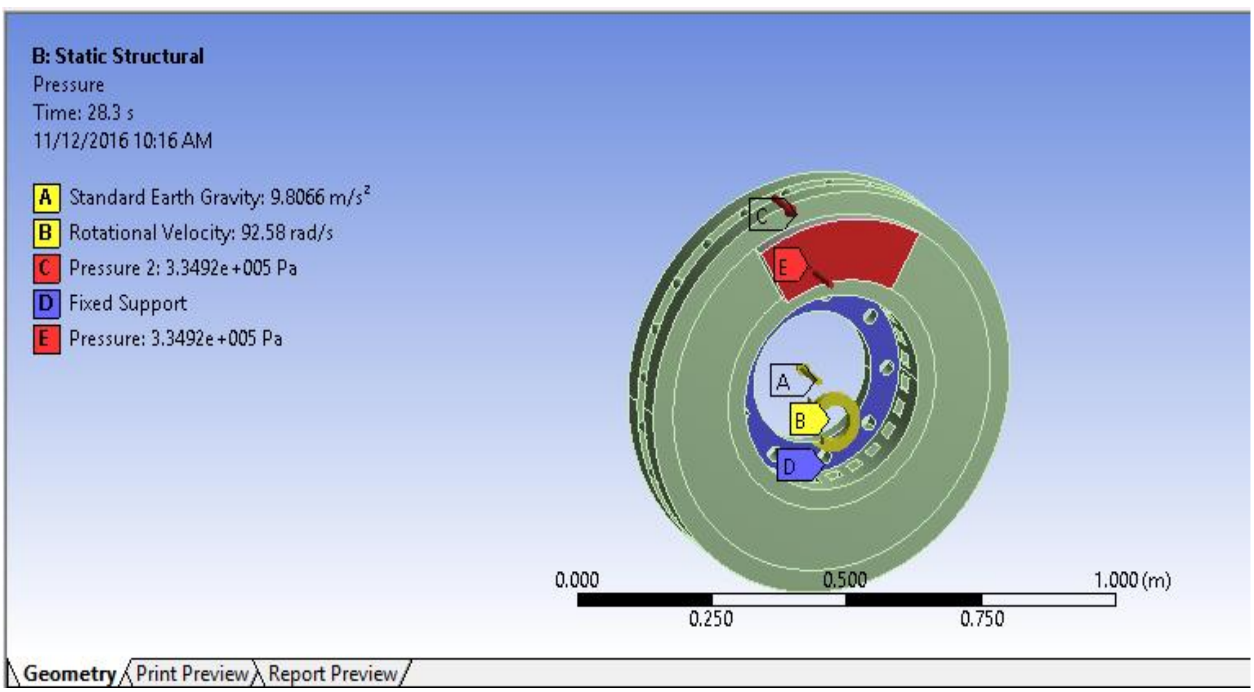

Figure 22 Imposed structural loads due to 10 % overcharging

5.1.2 Results and discussion when the braking system is under overcharging condition

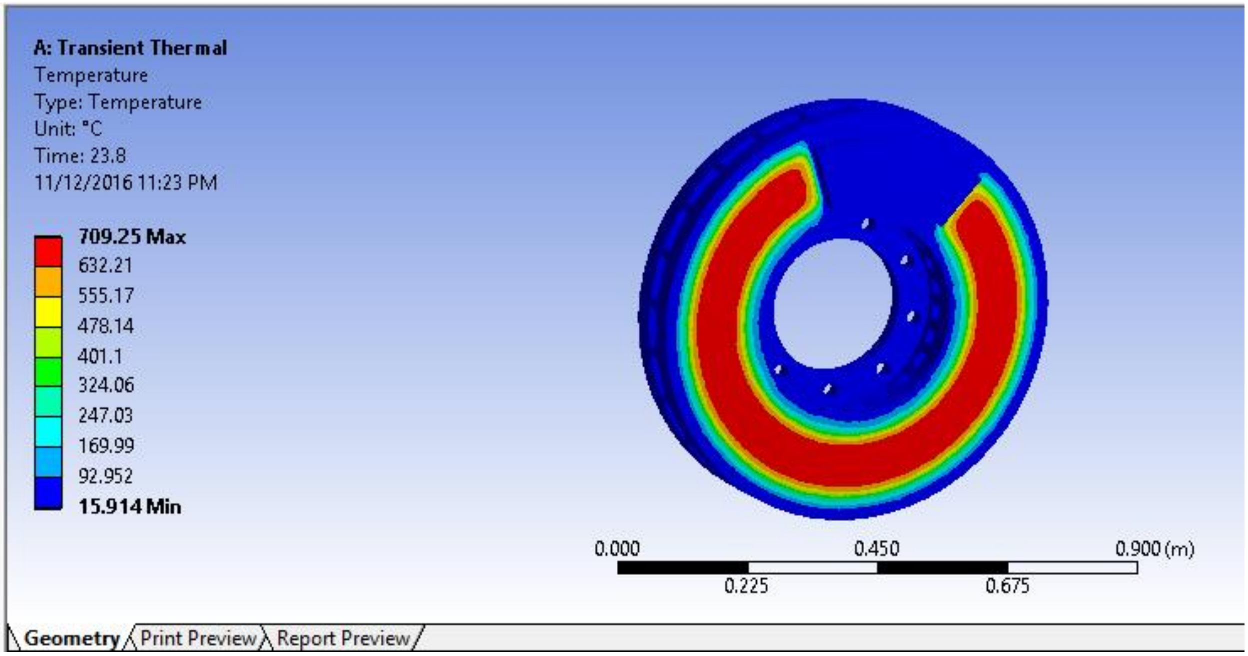

Figure 23 Temperature due to 2% of air overcharging

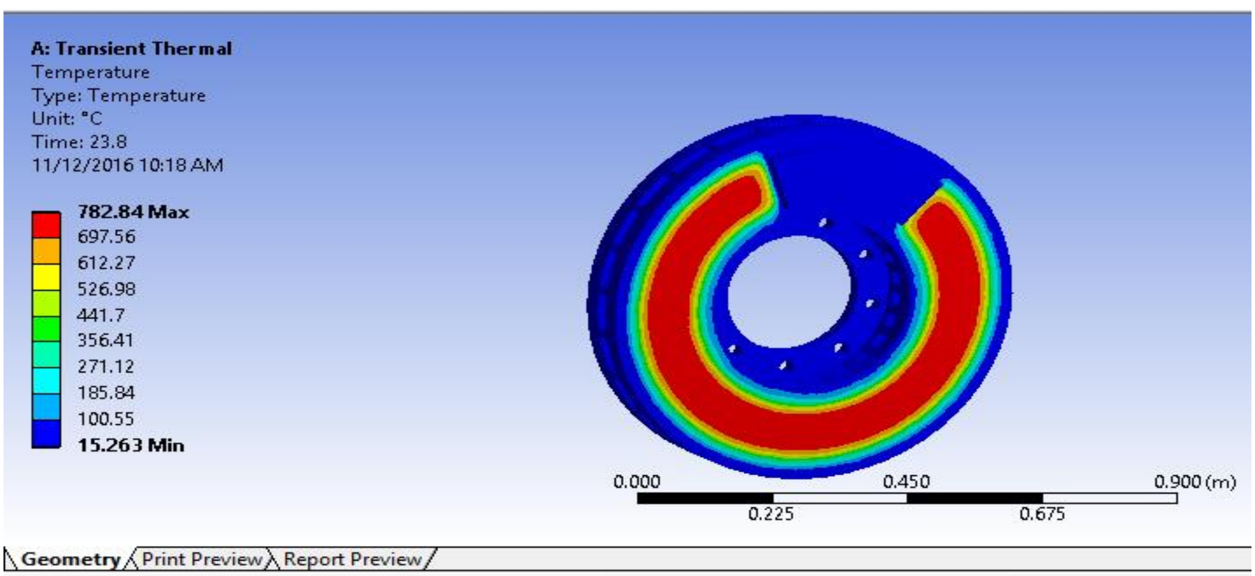

Figure 24 Temperature due to 10 % of air overcharging

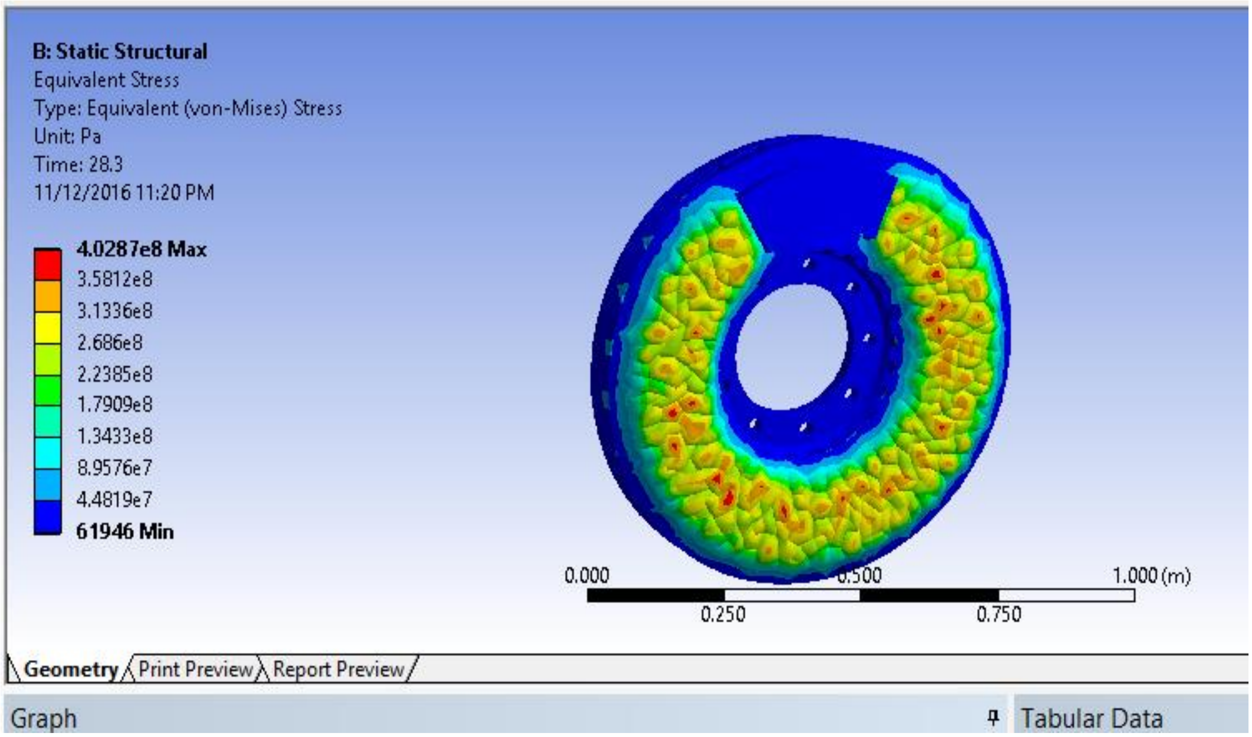

Figure 25 Stress due to 2% of air overcharging

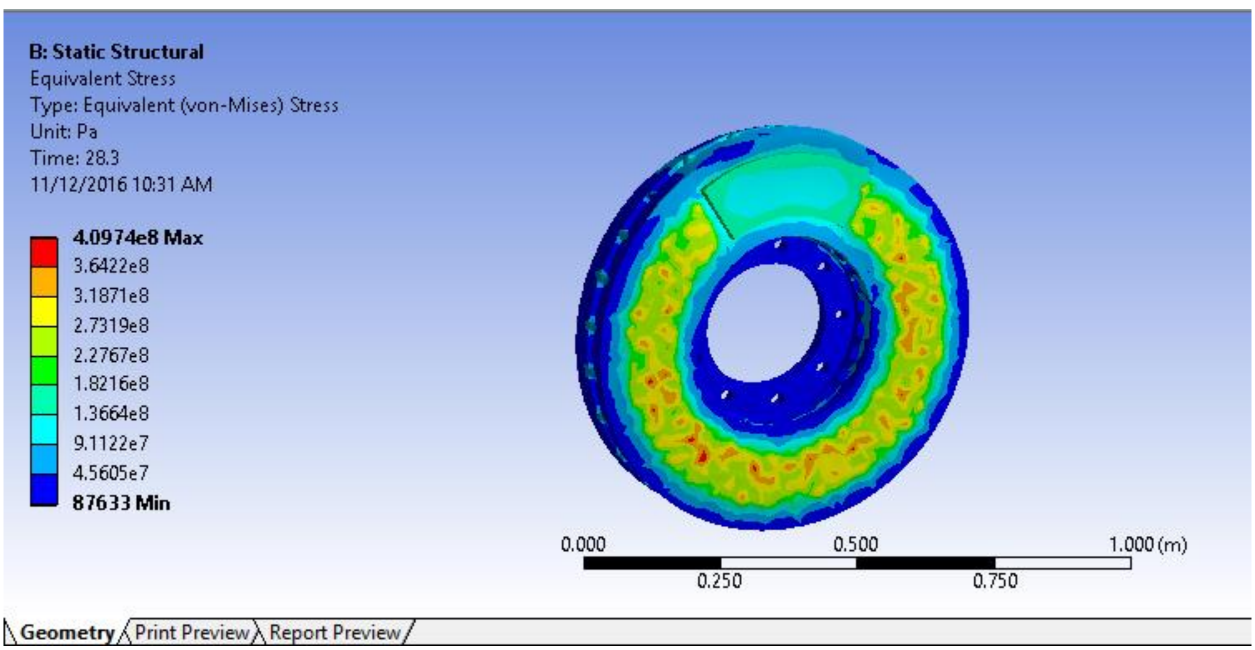

Figure 26 Stress due to 10% of air overcharging

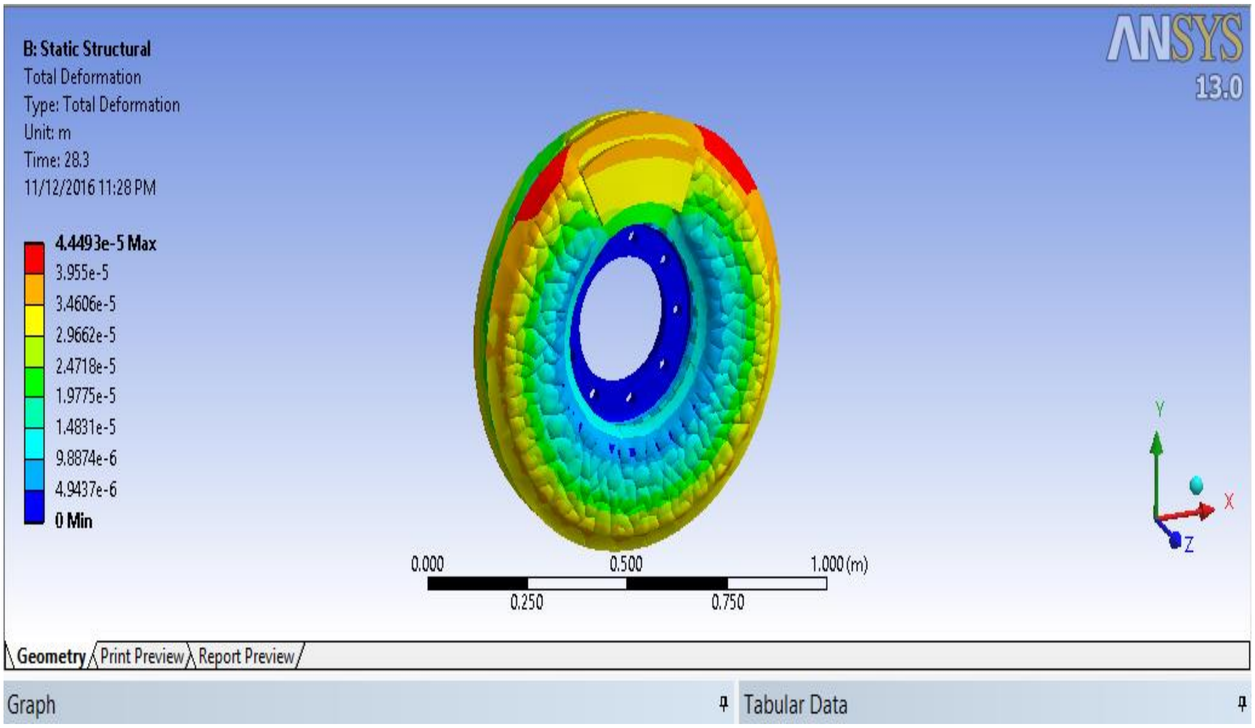

Figure 27 Deformation due to2% of air overcharging

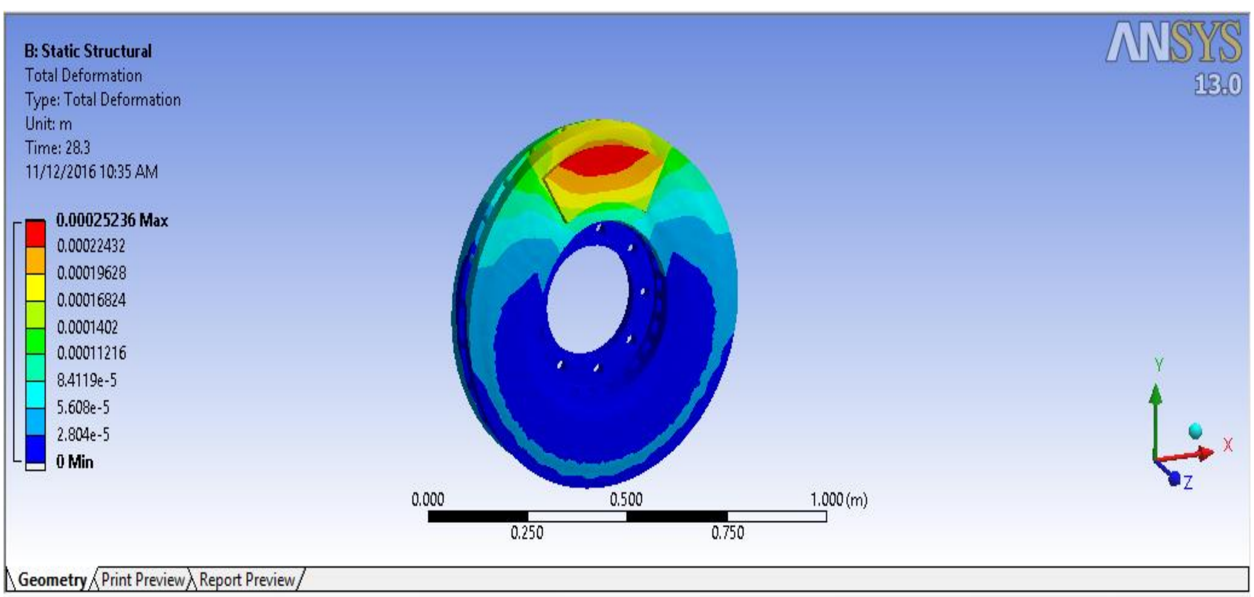

Figure 28 Deformation due to 10% of air overcharging

5.1.3 Discussion when the braking system is under overcharging condition

The above figures (5.5),(5.6) ,(5.7) ,(5.8) (5.9) and (5.10) are shows that temperature, stress and deformation developed on the train disc due to air overcharging braking condition at 2% and 10% assumption. The temperature and stress developed in the disc is increased from 709.25c^{o} to 782.84c^{o}and 402.87MPa to 409.74 MPa repesctively.

Table 23 Summery of ANSYS results for air overcharging condition

Air Overcharging assumption	Temperature	Stress	Deformation
2%	709.25c^{o}	402.87 MPa	0.000044493m
10%	782.84c^{o}	409.74 MPa	0.000025236m

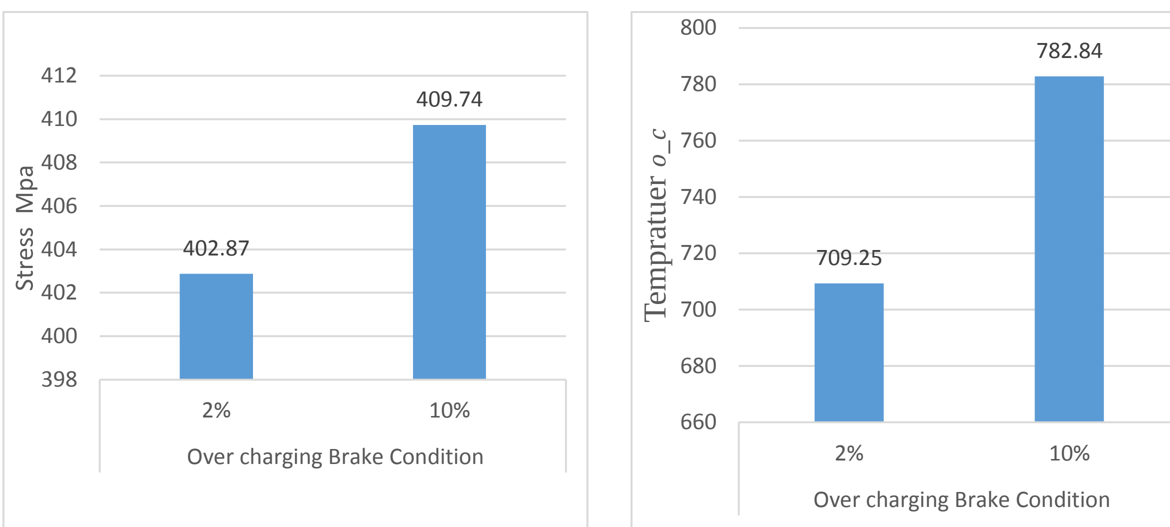

Figure 29 Temperature and stress effects due to overcharging brake condition

5.1.4 ANSYS input when the braking system is under air leakages condition

Due to 15% air leakages condition

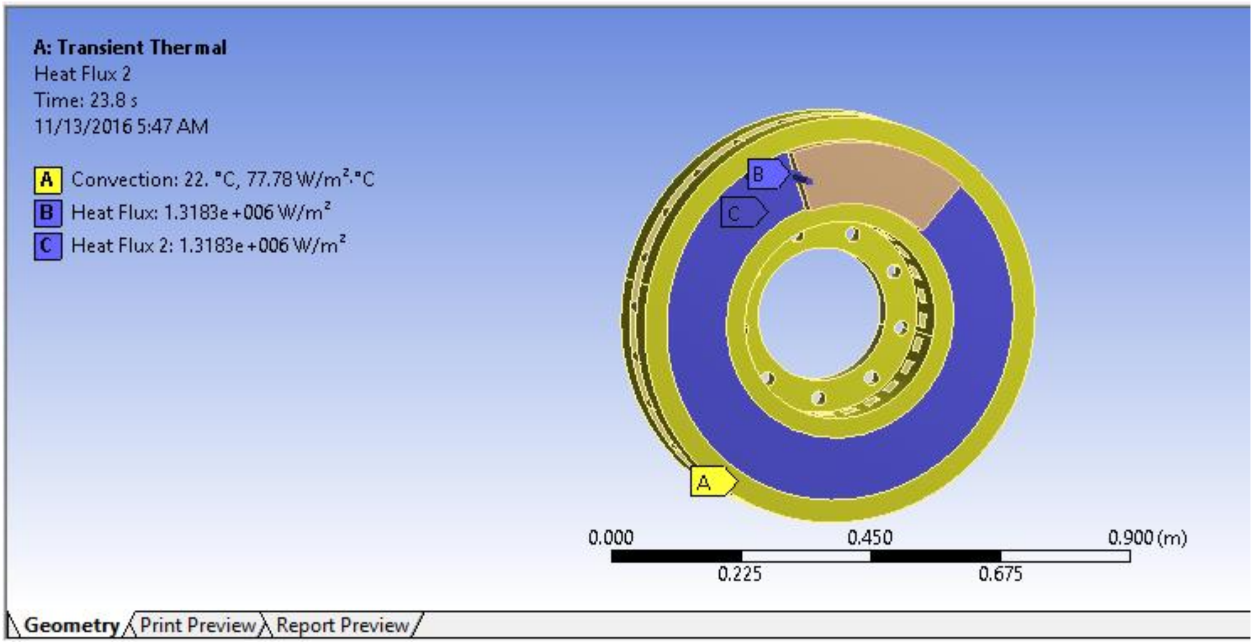

Figure 30 Imposed heat loads due to 15% of air leakages

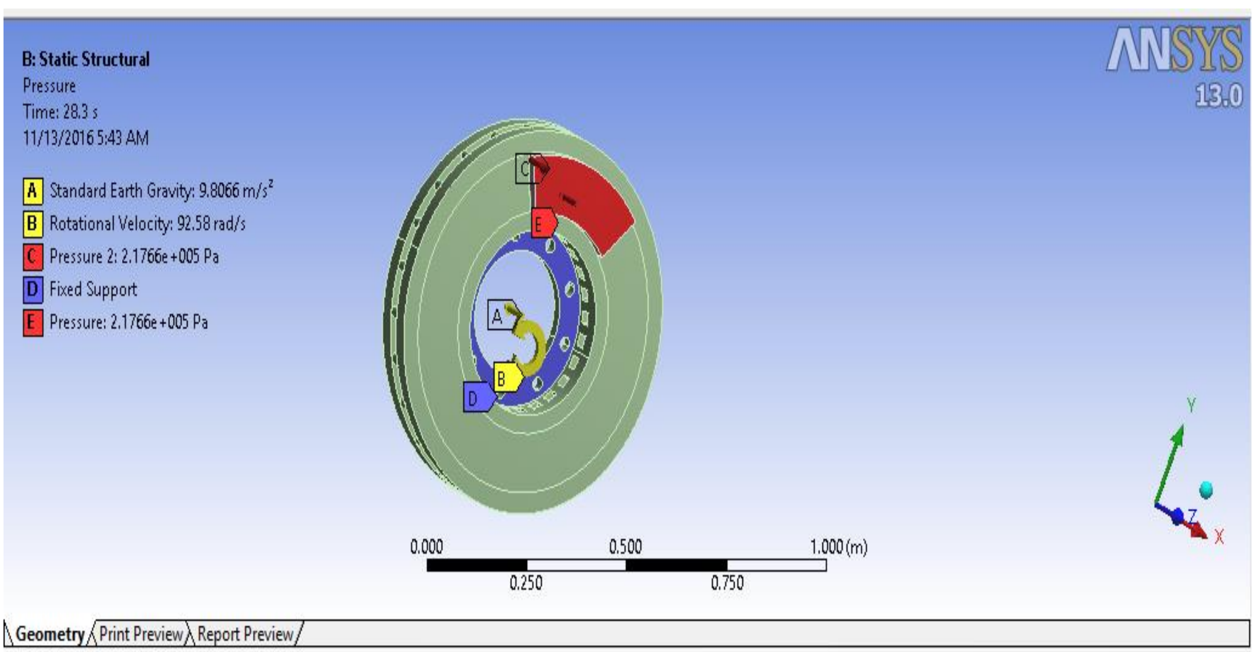

Figure 31 Imposed structural loads due to 15% of air leakages

Due to 25% air leakages condition

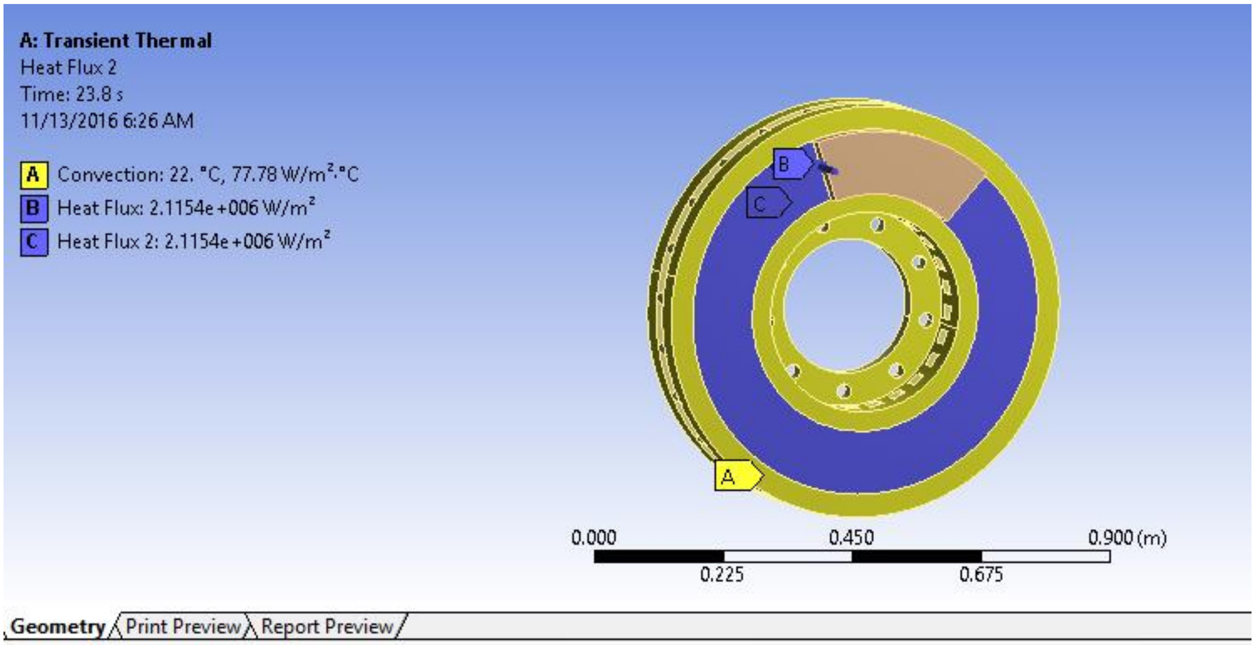

Figure 32 Imposed heat loads due to 25% air leakages

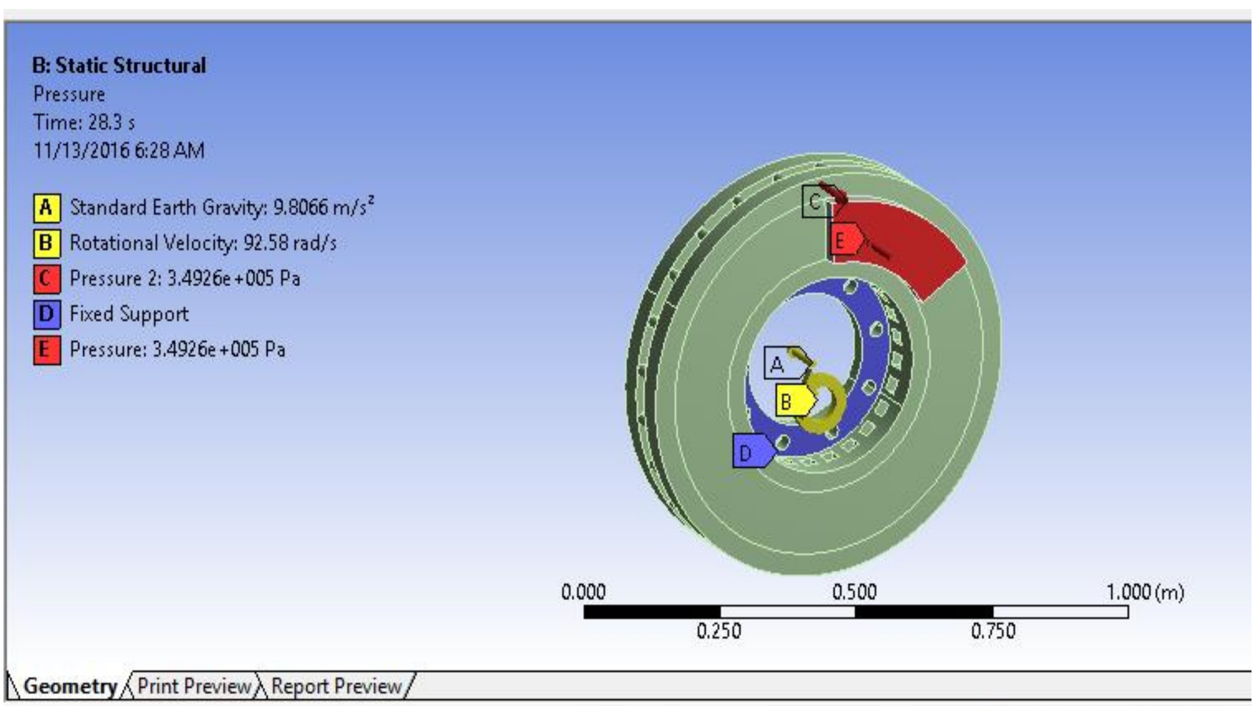

Figure 33 Imposed structural loads due to 25% air leakages

5.1.5 Results and discussion when the braking system is under air leakages condition

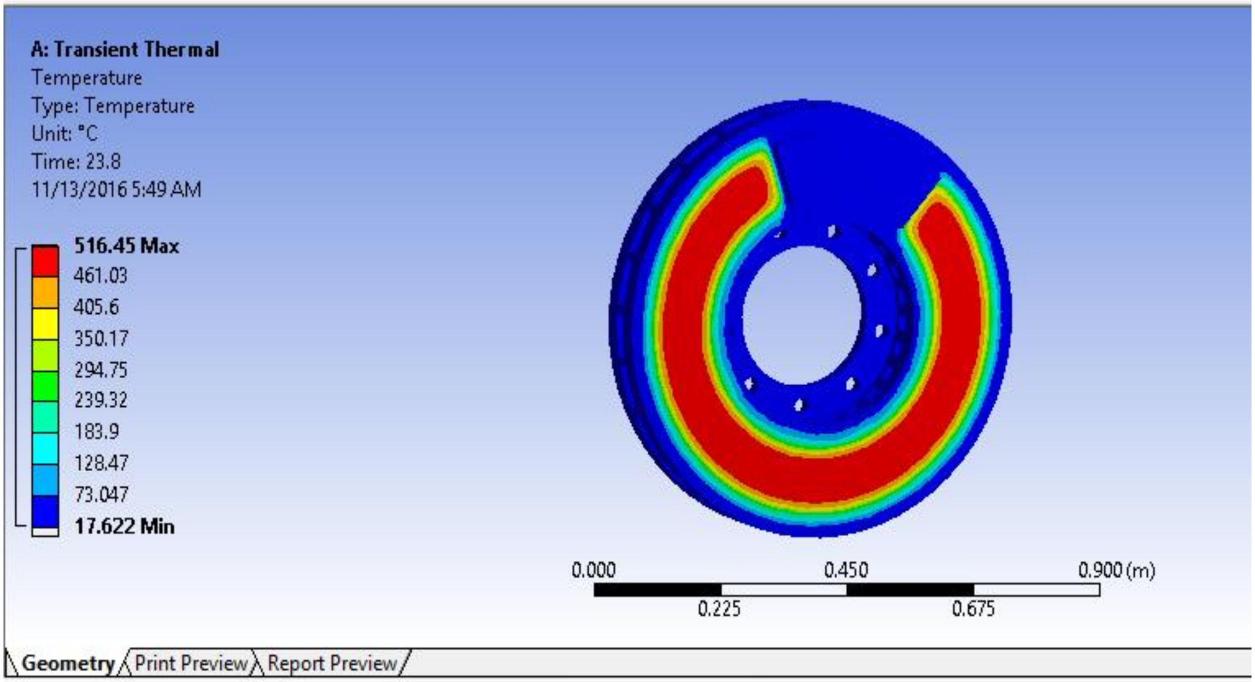

Figure 34 Temperature due to 15% of air leakages

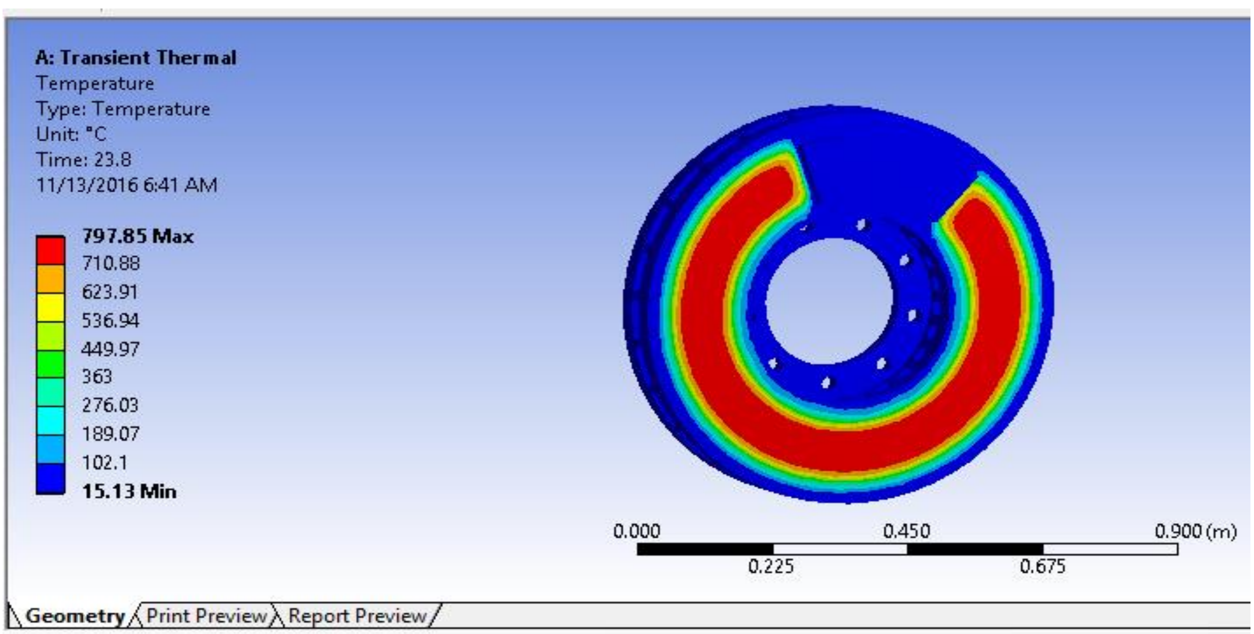

Figure 35 Temperature due to 25% of air leakages

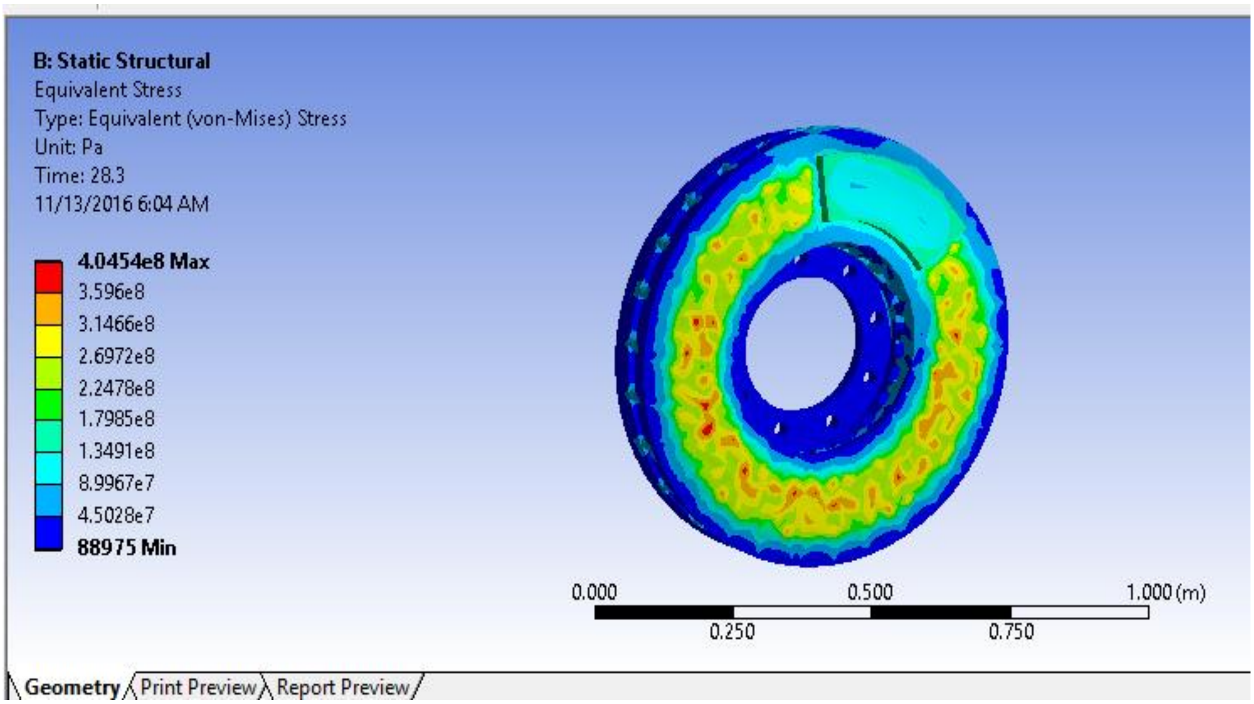

Figure 36 Stress due to 15% of air leakages

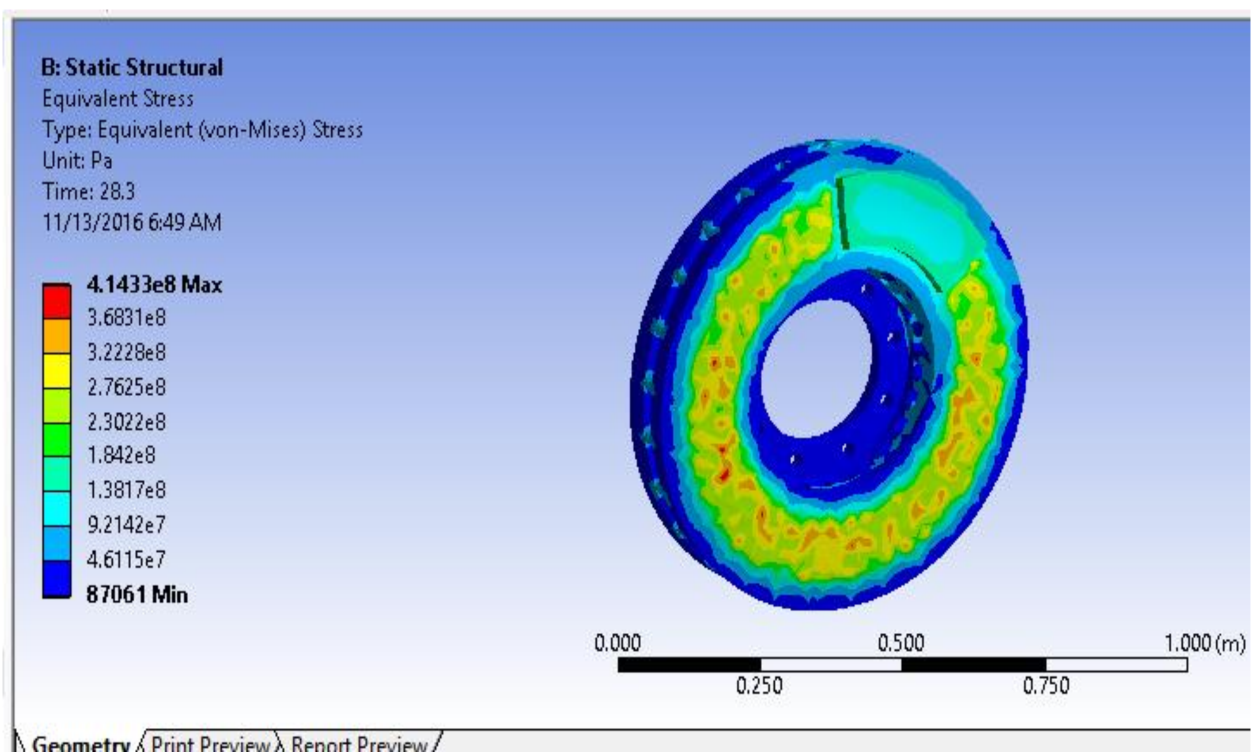

Figure 37 Deformation due to 25% of air leakages

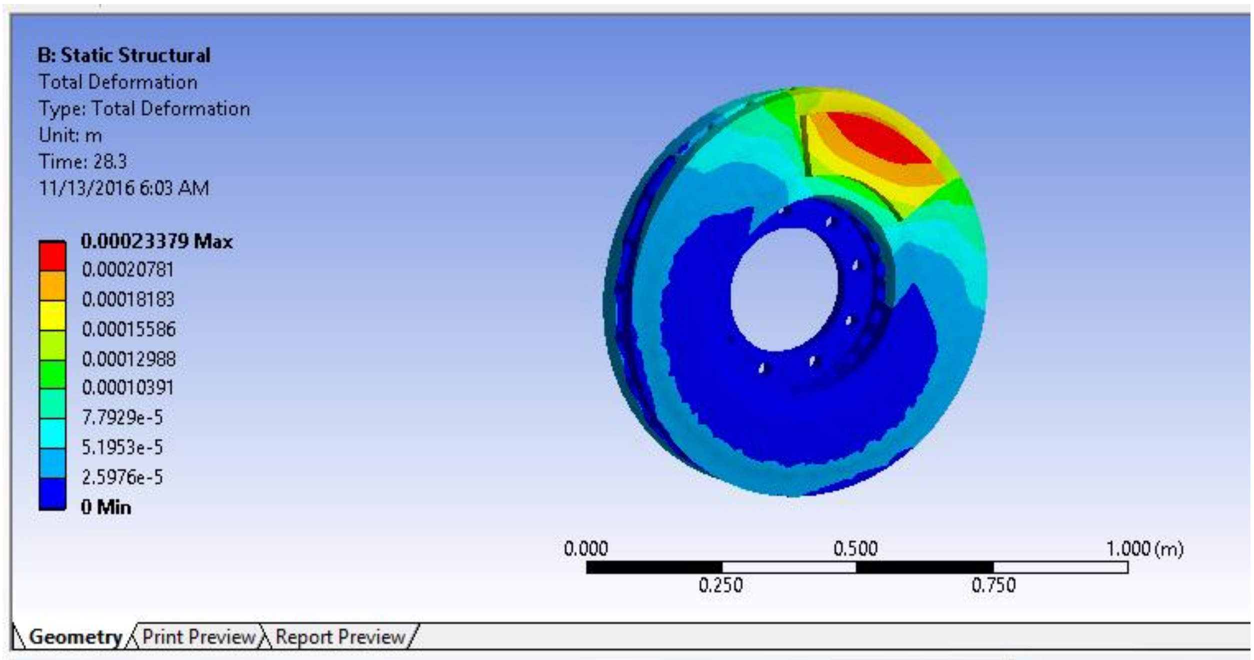

Figure 38 Deformation due to 15% of air leakages

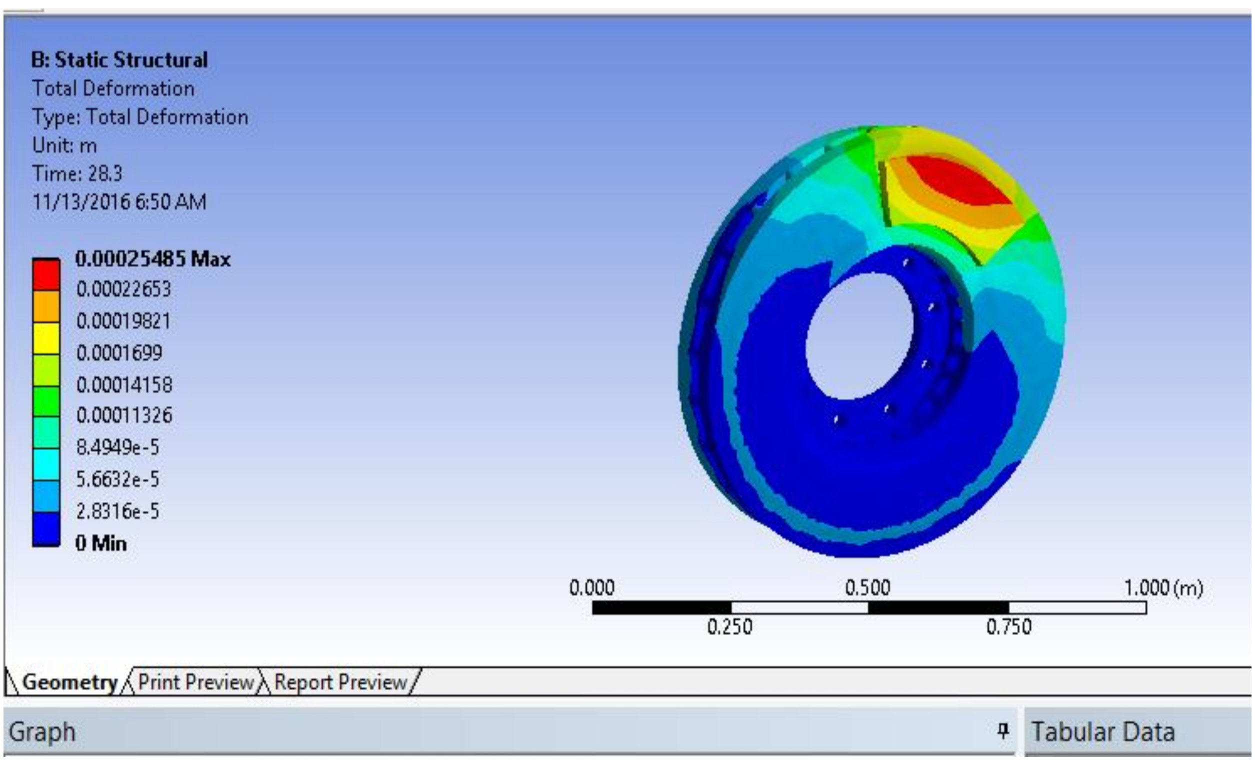

Figure 39 Deformation due to 10% of air overcharging

5.1.6 Discussion when the braking system is under air leakages condition

The above figure (5.15), (5.16), (5.17), (5.17), (5.18), (5.19) and (5.20) are shows that temperature, stress and deformation developed on the train disc due to air leakages braking condition at 15% and 25% assumption. The temperature and stress developed in the disc is increased from 516.45c^{o} to 797.85c^{o}and 404.54MPa to 414.33 MPa respectively.

Table 24 Summery of ANSYS results for air leakages condition

Air leakages assumption	*Temperature*	*Stress*	*Deformation*
15%	*516.45c^{o}*	404.54 MPa	*0.00023379m*
25%	797.85c^{o}	414.33 MPa	*0.00025485m*

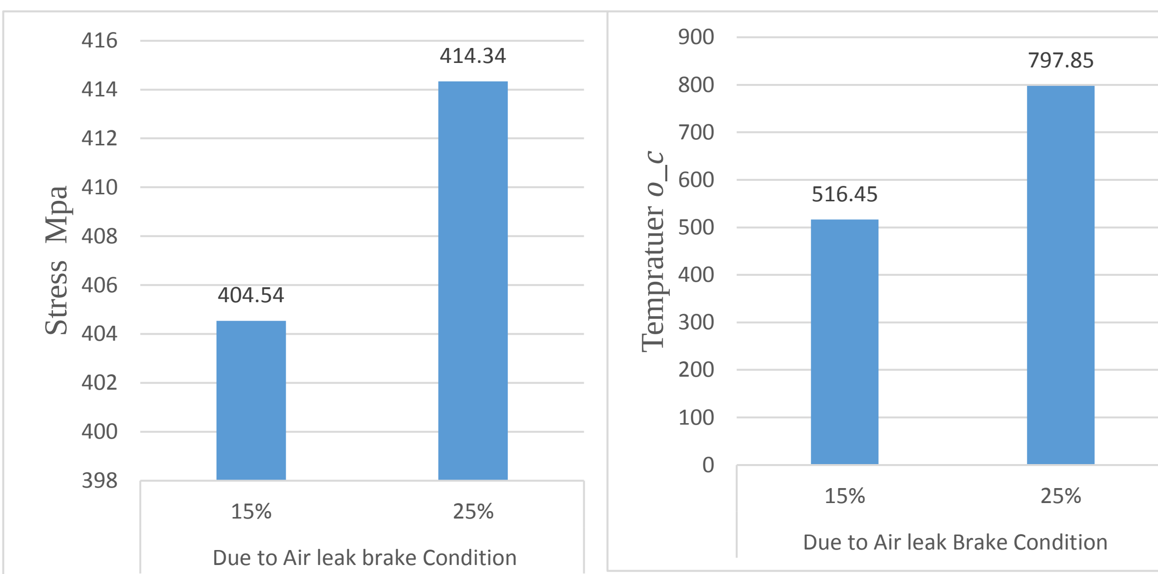

Figure 40 Temperature and stress effects due to air leakages brake condition

5.1.7 ANSYS input when the braking system is under normal operation condition

Due to normal operation condition

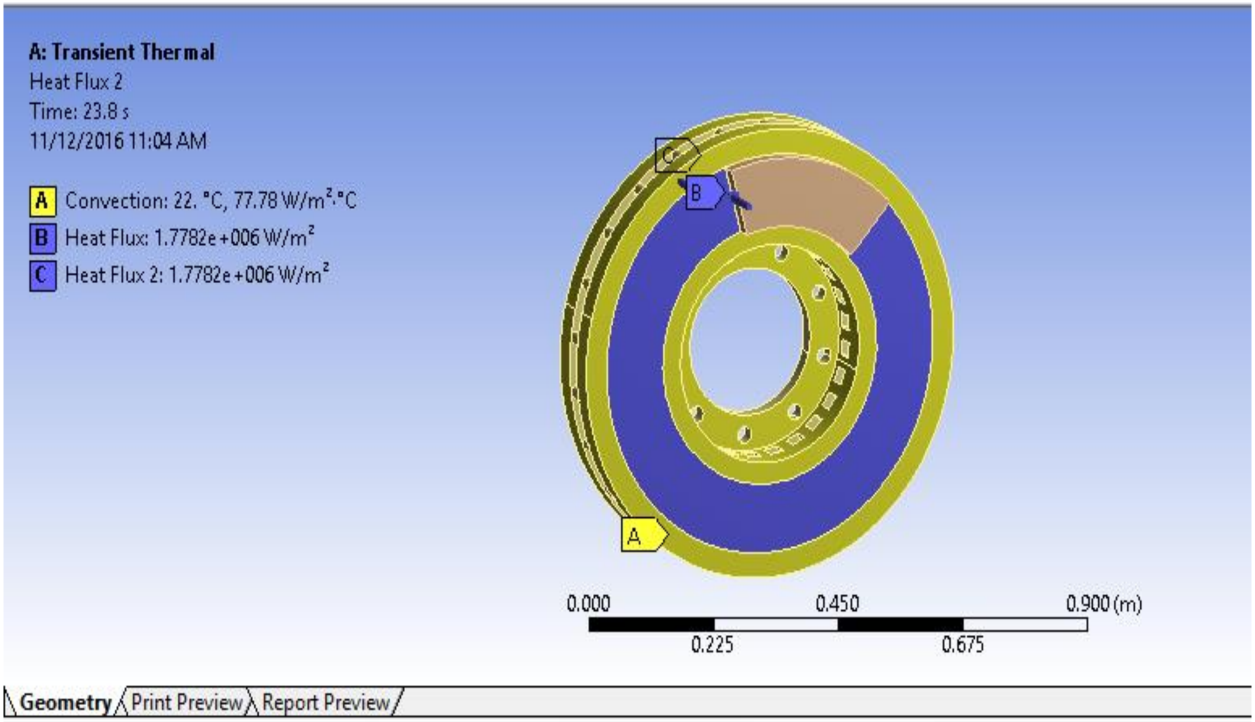

Figure 41 Imposed heat loads due to normal condition

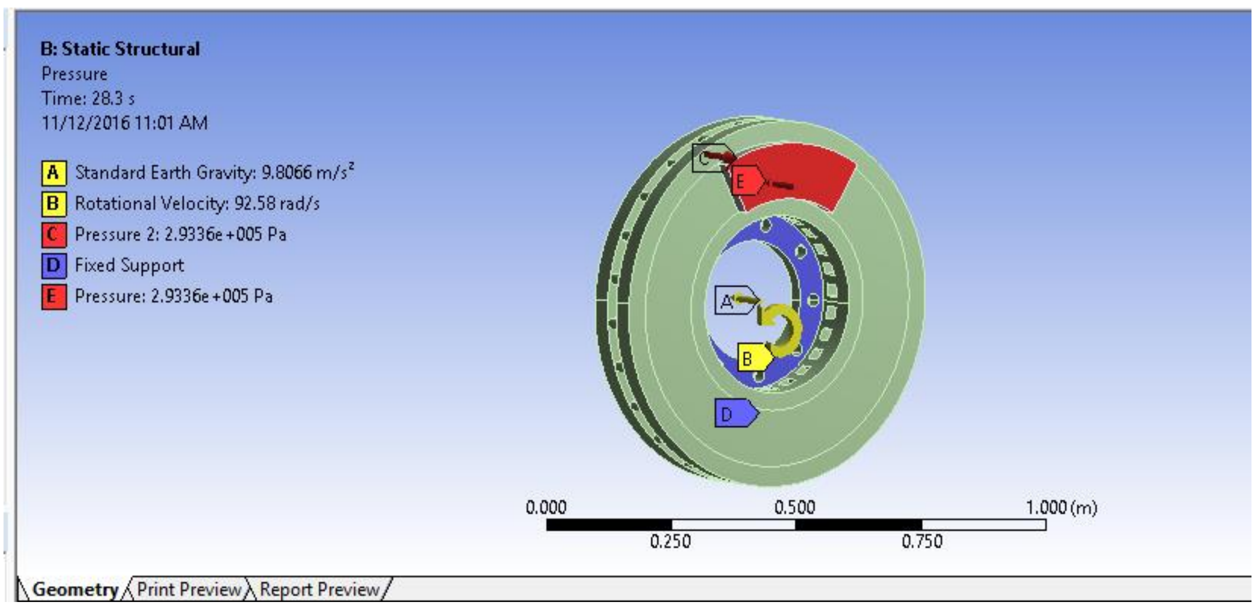

Figure 42 Imposed structural loads due to normal operation conditio

5.1.8 Results and discussion when the braking system is at normal operation

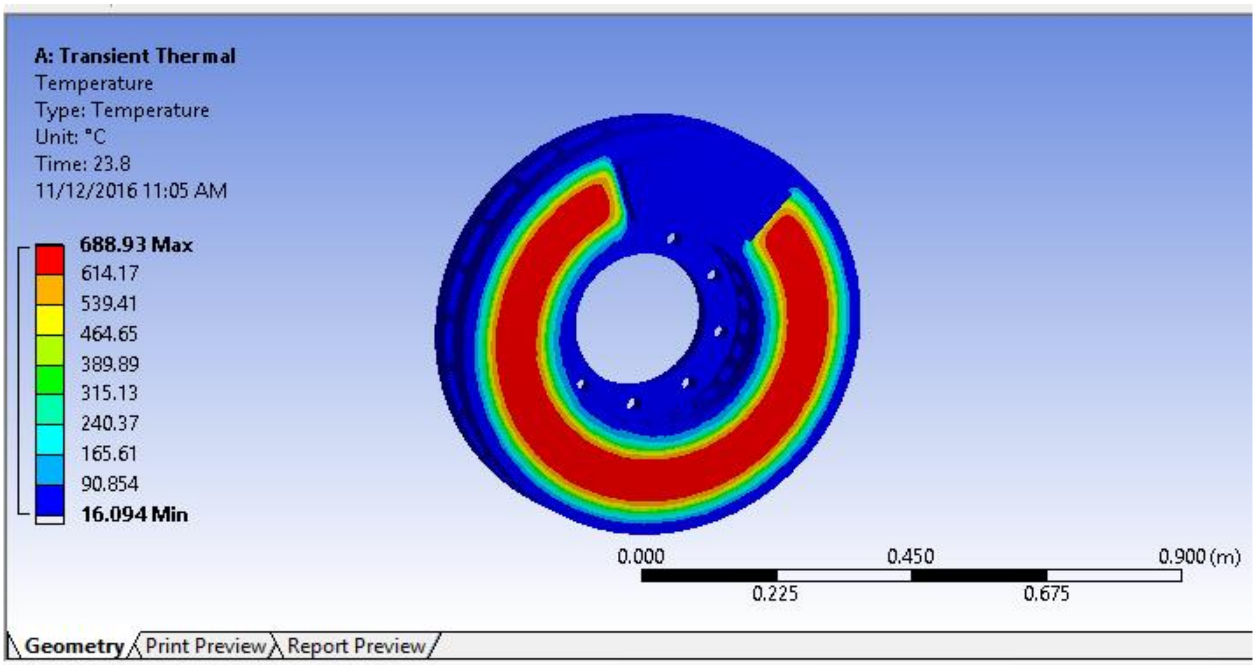

Figure 43 Temperature due to normal operation condition

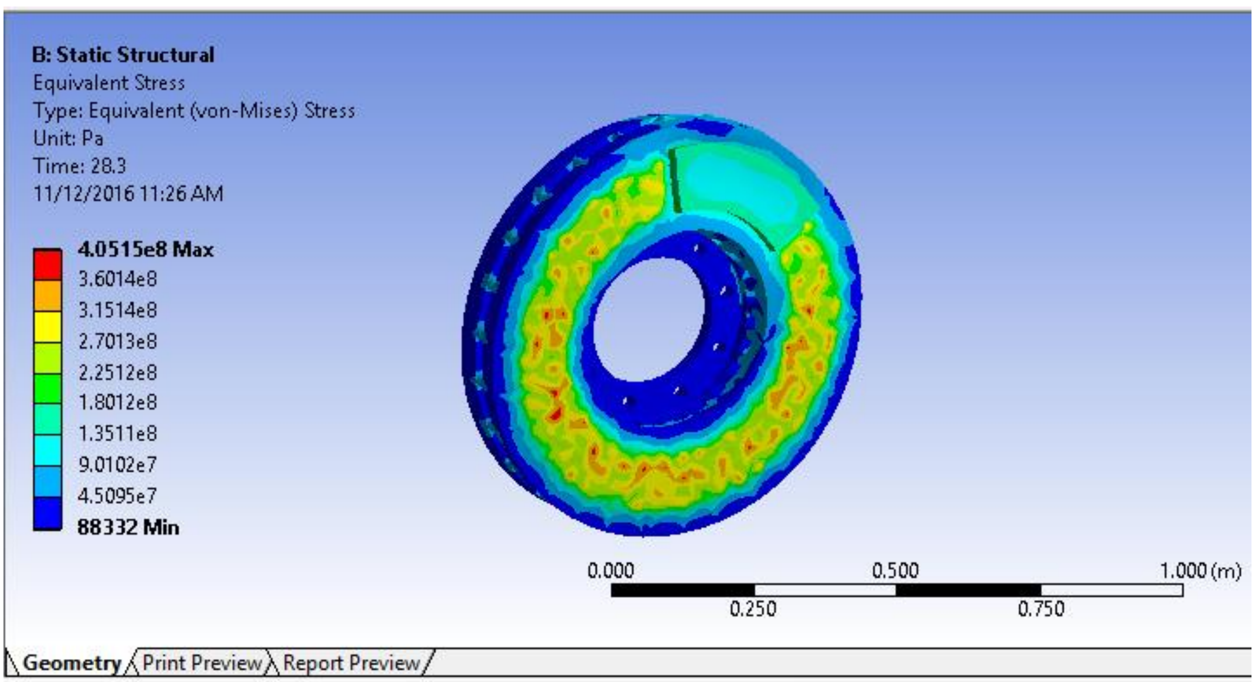

Figure 44 Stress due to normal operation condition

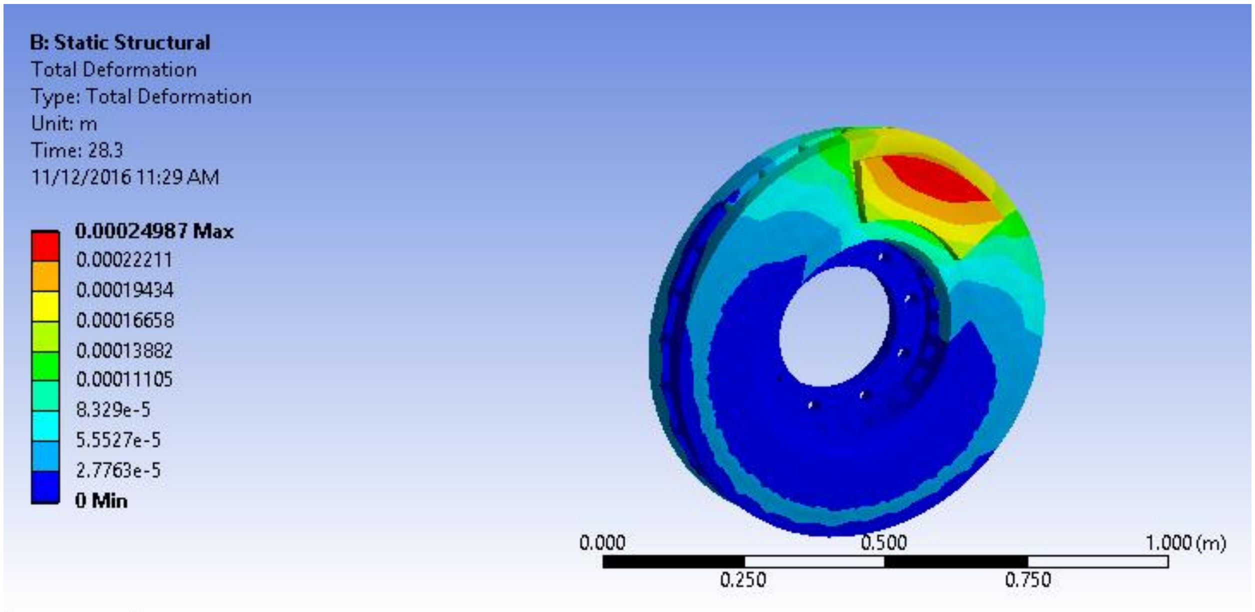

Figure 45 Deformation due to normal operation condition

The above figure (5.23), (5.24) and (5.25) are shows that temperature, stress and deformation developed on the train disc due to normal braking condition. The temperature and stress developed in the disc is 688.93c^{o} and 405.15MPa respectively

Table 25 Summery of ANSYS results for normal operation condition

	Temperature	*Stress*	*Deformation*
Normal operation condition	688.93c^{o}	405.15 MPa	0.00024987m

Chapter 6

Conclusion, Recommendation and Future work

6.1 Conclusion

This paper presented the analysis of the thermo-mechanical behavior of brake disc during controlling braking force. The modeling and analysis is conducted ANSYS AND CATAI soft wares.

From the Analysis it conclude that the maximum temperature of the disc corresponding to 2% and 10%of air overcharging in the braking system is $709.25c^{o}$ and $782.84c^{o}$, respectively. Also the corresponding von misses stress are 404.74 MPa and 409.74 MPa respectively. This has a difference when we compared with to the values resulting from normal brake operation condition, which has temperature of $688.93c^{o}$ and stress 405.15 MPa.

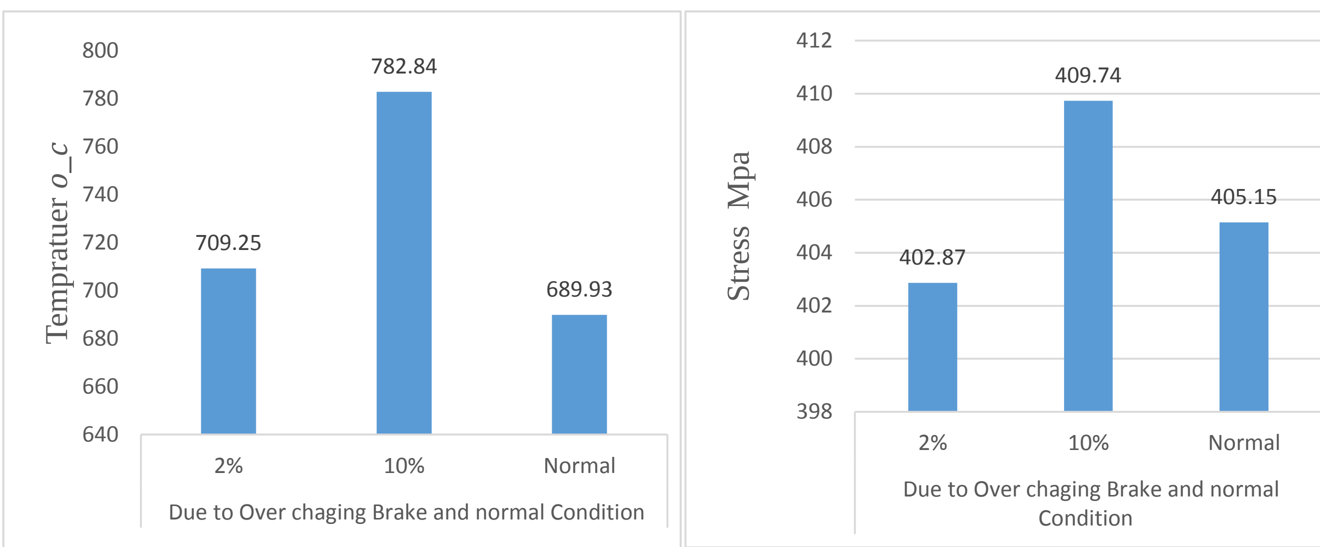

Figure 46 Temperature and stress effects due to overcharging and normal brake condition

Also the maximum temperature of the disc corresponding to 15% and 25%of air leakages in the braking system is 516.45c^{o} and 797.85c^{o}, respectively. Also the corresponding von misses stress are 404.54 MPa and 414.33 MPa respectively. This also has difference when we compared with to the values resulting from normal brake operation condition, which has temperature of 688.93c^{o} and stress 405.15 MPa.

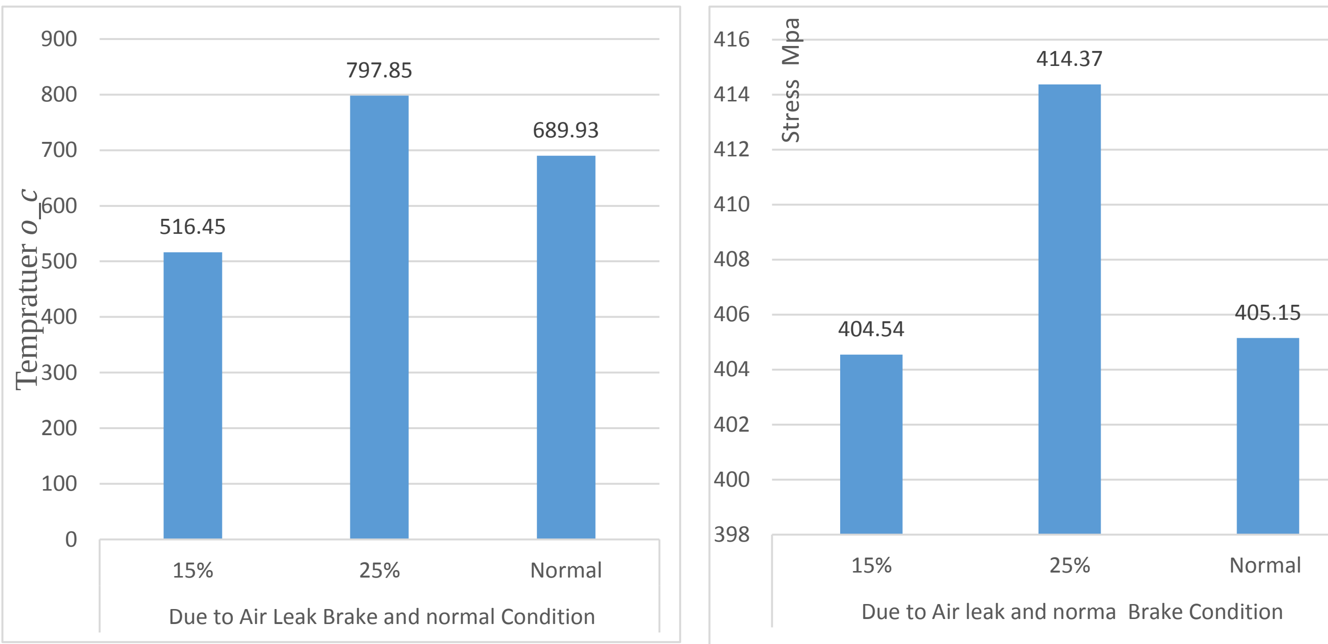

Figure 47 Temperature and stress effects due to air leakages brake and normal condition

Therefore comparing the temperature and stress from normal and uncontrolled brake condition is raised by 108.92c^{o} and 9.18 MPa, which is uncontrolled braking force has a significant contribution to affect the braking disc mechanical properties.

6.2Recommendation

From the analysis result the uncontrolled braking system has many effects, which is a negative consequences on brake equipment’s, therefore ERC should take high attention to control the uncontrolled braking condition.

6.3 Future work

This paper presented the effects of uncontrolled braking force due to only the power creation system, but there are many others factors that creates uncontrolled braking force like Brake power transmission system (Brake rigging) and Human Failure/Negligence, Therefore anyone can continue this paper by including such and similar factors.

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