Addis Ababa University

Addis Ababa Institute of Technology

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EFFECT OF MALFUNCTIONING OF THE BRAKING SYSTEM ON THERMO-MECHANICAL PROPERTIES OF FREIGHT TRAIN WHEELS

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A thesis submitted to the School of Graduate Studies of Addis Ababa University in partial fulfillment of the Degree of Masters of Science in Mechanical Engineering under Railway Engineering.

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Abstract

There are many causes for the overheating of freight train wheels. The braking system takes the lion's share of the heat induced to the wheels while the train is in operation. Under normal conditions the wheels should have enough thermal capacity so that they can handle the heat developed by friction during any sort of braking. However, as in any mechanical systems malfunctioning of the braking system is inevitable. This thesis deals with the overheating of freight train wheels caused by malfunctioning of the train's braking system.

Mainly, the effects of friction fade, overcharged brake pipe and excessive leakages from the brake pipe on the thermo-mechanical properties of the wheels are modelled and analyzed using finite element analysis. The effect of friction fade is investigated by assuming the brake blocks on two wheels and four wheels had entered friction fade during previous sever braking action. This assumption is drawn from previous researches and derailment investigation reports. Investigation regarding the effect of brake pipe overcharge and excessive brake pipe leakage are made on assumptions backed by the train brake inspection rules of some railways and the generally accepted valid range of brake pipe pressure reduction.

In all the three malfunctioning cases, the result from the finite element analysis revealed that the malfunctions made considerable contribution to the overheating and overstress conditions of the wheel of the freight locomotive.

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Nomenclature

a_x	Longitudinal acceleration of the rolling stock
B	Width of the brake block
E_b	Braking energy
E_b'	Braking energy per wheel
ε_1	Heat partitioning parameter between wheel surrounding
ε_2	Heat partitioning parameter between wheel and brake block
$F_b=F_N$	Braking force
F_{fr}	Friction force
h	Convective heat transfer coefficient
I	Moment of inertia of the wheel
K	Thermal conductivity of the wheel
K_a	Thermal conductivity of air
k	Thermal diffusivity
L	Pelcet number
M	Mass of the locomotive
N_w	Wheel load
P_z	Relative pressure in the brake cylinder
P_z'	Absolute pressure in the brake cylinder
Q	Heating power per unit volume
dQ/dt	Heat flux induced to the wheel
q_{diss}	Heat dissipated by convection and radiation
q_{gen}	Heat generated during braking
R	Radius of the wheel
r	Brake pipe pressure reduction
S_b	Braking distance
T_{br}	Braking torque
T_{ad}	Adhesion torque
T_{reff}	Surrounding temperature

T_{avg}	Average temperature between wheel and brake block
T_{max}	Maximum temperature between wheel and brake block
T_b	Braking time
U	Velocity field
V_f	Volume of air in the auxiliary reservoir
V_z	Volume of air in the brake cylinder
V_s	Sliding velocity
μ_{wr}	Coefficient of friction between wheel and rail
μ_{br}	Coefficient of friction between wheel and brake block
μ'	Equivalent coefficient of friction between wheel and brake block
ΔEK	The change in translational kinetic energy
ΔEK_r	The change in rotational kinetic energy
ΔEP	The change in potential energy
TAZ	Temperature affected zone
AAR	Association of American Railways
FEA	Finite element analysis
ERC	Ethiopian Railway corporation

Chapter 1

Introduction

1.1. Background

According to Dr. Michael JT Lewis in (1974), 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 [22].

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 viable 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[22].

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 Dire dawa. The bougie wagons were some with four axles weighing10 tons and carrying 22 tons[21].

In Railways one of the major maintenance costs is the cost incurred by inspection and replacement of worn out and broken wheels. The mechanical properties of the wheel material are highly affected by very high temperatures. The wheel material is likely to fail due to metallurgical damage, thermal crack or induced tensile residual stress when it is subjected to repeated high temperatures.

In recent days many researchers have been conducting researches to study the various failures that hot wheels undergo and outline the management of wheel failures. However the cause of variation in wheel temperatures is seldom 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 braking system to the overheating of the wheels, the variation in wheel temperatures, the problem associated with it and the corresponding remedial measures to be taken.

Sara Caprioli , Tore Vernersson and Anders Akberg, (on journal of Rail Rapid Transit) in 2012 concluded that sever drag braking due to malfunctioning brakes cause very deep cracking that runs radially from the rim to the center of the wheel [3].

In addition, it is clearly stated by Schobel Andrias on ‘ report on Analysis of derailment causes, impact and prevention assessment’ in 2012, that one of the causes of of derailment is the wheel failure caused by overheating due to blocked brakes and locked wheels [4].

These and many other researchers indicate that the temperature of some wheels in a train could reach as high as 600°C after faulty braking action, which is enough to lead to a disastrous consequence. Whereas other wheels on the same train may still be too cold to be suitable even for braking action.

This research will try to identify the contribution of the malfunctioning of the braking system to the overheating of the wheels and to the overstressed condition of the wheels of the train. Moreover, it will suggest means of maintaining more or less uniform temperature of the wheels to distribute the thermal burden and extend the life of the wheels in the train. The two main malfunctioning of the braking systems that contribute a great share of the high temperature in wheels are drag braking caused by stuck tread brakes and wheel slides caused by the locking of wheels.

1.2. Types of pneumatic mechanical brakes

1.2.1. 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.

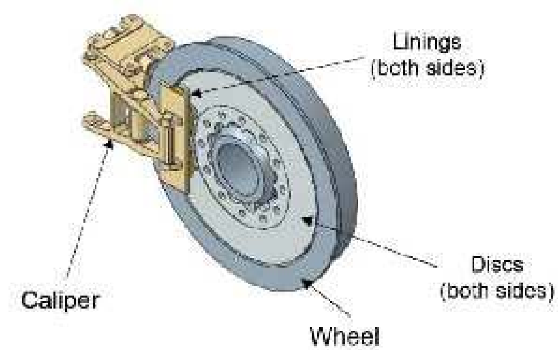

Figure 1-1 Disc brake assembly[26]

1.2.2. 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.

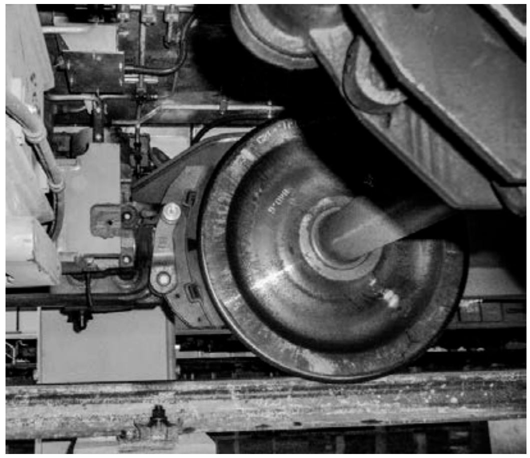

Figure 1-2 Tread brake assembly [7]

1.3. Possible causes of malfunctioning of tread braking

1.3.1. Causes of unintended drag braking

Stuck brake riggings could be the result of faulty guidance of the brake beam in the bogie and specifically by the reaction in the brake beam guides in the side frame. In addition, inadvertent reduction of brake pipe pressure caused by air leakage somewhere along the pipe could be another root cause of stuck brakes which in turn brings about unintended drag braking.

1.3.2. Causes of locked wheels

The wheel locks and slides over the rail when the braking torque is greater than the adhesion torque between wheel and rail. The possible root causes of this are:

- The presence of insufficient adhesion force between wheel and rail due to contaminants like wet rail, leaves and dust etc causing variable friction conditions.
- Slope on the terrain that forces the driver to apply brake while the car is moving fast.
- Irregularities along the brake rigging assembly that causes increased braking force.

1.4. Statement of the problem

During braking and curving the kinetic energy of the train is converted into heat energy thereby increasing the temperatures of the wheels. If this temperature is so high, it will affect the mechanical properties of the wheel material and may eventually lead to failure. However, this is not true for all wheels in a train. Some wheels may still be much colder than others. In other words, when the kinetic energy of the train is converted to heat energy and absorbed by the wheels, some wheels absorb much greater proportion of this huge heat than others. Consequently, these wheels suffer from the resulting high temperature and fail either abruptly or slowly due to metallurgical damage, fatigue thermal crack or induced tensile residual stress. High temperature of wheels also aggravates the circumferential degradation by spalling, shelling, flat spot and out of round.

According to the findings released by WDPRC (wheel defect prevention consortium) in 2008, almost 50% of the wheel replacements made in 2006 were tread damage related causes, most of which are caused by overheating of wheels. The cost of replacement of these wheels was estimated to be 350 million USD. Moreover, the trains would often be subjected to stopping for inspection, maintenance and replacements as hot wheels continue to manifest thereby significantly affecting productivity. Hence, this gives us a good picture of the economic impact hot wheels may incur to the operation of the train. Furthermore, the WDPRC also strongly suggested that investigations should be made to find out the real causes of temperature variation in the wheels of a train within cars and between cars [1].

In addition, the findings of P. Laczko and P.J. Mutton of the institute of railway technology, Monash University[2], the methods in use for the assessment of overheated wheels(such as sectioning, discolouration and paint blistering) have not been found to provide a completely reliable indications of the extent of damage, thereby allowing at- risk wheels to be returned to service or removing wheels that might function well for additional service life. Hence, in areas where more advanced inspection techniques (non-destructive residual stress measuring techniques) are not implemented, efforts to avoid the frequency of occurrence of overheated wheels are indispensable [2].

Hot wheels might also be causes of derailment in cases when fracture of wheels following thermally initiated crack occur under high dynamic loading conditions as it is clearly indicated by Schöbel Andreas and other researchers on 'Report on analysis of derailment causes, impact and prevention assessment' in 2013 [4].

One of the root causes of overheating is sever adhesion braking as a result of inadvertent drag braking caused by stuck brake riggings, and wheel sliding during wheel locking events. These conditions result in the overheating of the wheel tread thereby causing either reversal of residual stresses within the tread (compressive to tensile) or wheel-set gauge change and eventually causing either of the problems illustrated above

1.5. Objectives of the research

1.5.1. General objective

- To analyse the contribution of malfunctioning of the braking system to overheating of wheels using finite element analysis and suggest remedial measures to minimize their effects.

1.5.2. Specific objective

- To identify the causes of hot wheels in relation to the malfunctioning of the braking system.
- To analyse the contribution of tread brake malfunctions to overheating of the wheels in a train.
- To suggest ways of minimizing the malfunctioning of tread brakes with regards to the air brake lines and brake riggings.

1.6. Scope and limitations

This research will identify the causes of overheating of wheels in relation to the braking system by analyzing the conditions in faulty braking actions. The thermal load contribution of blocked brakes and locked wheels will be thoroughly analyzed using the finite element analysis. Legitimate and careful assumptions will be made to simplify the analysis and see only the effect of abnormal braking on the variation in temperature within the wheels of the train. Finally, the results of the FEA will be collected as tables and diagrams of temperatures and stresses of the wheel corresponding to each case of braking malfunctioning.

It would be far better if a miniature model of single car was manufactured and the effects of wheel locking and drag braking were analyzed in the workshop. However, the time frame set for this thesis does not allow us to do that.

As the research is not aimed to analyze the different forms of failures of the wheels caused by overheating, it does not entertain deep into the details of these wheel failure modes.

1.7. Methodology

At this stage all necessary data regarding braking system, wheel and rail material, axle and wheel loads and truck conditions are collected from Railway Corporation and literatures and the finite element analysis will be conducted for the cases of a wheel with drag braking and wheel locking.

1.7.1. Primary data collection

- The type of braking system and its operation principles will be observed and recorded.
- The wheel and rail materials and useful quantities describing their mechanical properties will be recorded.
- The axle load and corresponding wheel load on a particular freight car on the Addis

Ababa-Djibouti main line will be recorded.

- A portion of the main line that may present the worst braking conditions will be selected.
- Major possible causes that bring about wheel locking and drag braking will be analyzed

1.7.2. Analytical solutions for temperature in the wheel

The heat energy induced to the wheel will be calculated from the kinetic energy given to the train along the portion of the main line during brake application. In addition, the maximum temperature to be attained by the wheel will be calculated and the results will be recorded for later use in comparing it with the result obtained from the finite element analysis. This calculation will be carried out for the cases of normal braking, drag braking and wheel locking.

1.7.3. Finite element analysis

The finite element model of the wheels will be constructed using CATIA V-5 and imported to the ANSYS 14.5 workbench for further analysis. The primary data collected in the previous steps will be an input to the finite element analysis.

To minimize the uncertainty arising from overlooked factors in the assumptions, the following care will be taken

- Material for the FEA will be made to correspond to the wheel material used in the freight car
- The sever running condition considered in the selected portion of the Addis Ababa Djibouti main line will be analyzed and the worst case scenario will be identified.
- The superimposed effect of structural and thermal loads will be considered for all cases.

The finite element analysis will be conducted and the results from all possible braking irregularities analyzed will be recorded as follows.

- The constructed finite element model will be imported to ANSYS workbench.
- Material will be selected as described above and the wheels will be meshed properly
- The thermal and structural loads corresponding to the running conditions will be applied on each wheel of the car
- The temperature and stress distributions resulting from the superimposition of the structural and thermal analysis will be collected from ANSYS 14.5 workbench outputs as tables, graphs and diagrams.
- These procedures will be repeated for normal braking, drag braking and wheel locking cases over the portion of the main line under consideration.

1.7.4. Analysis of the results obtained

The results from the finite element analysis will be analyzed and the contribution of each set of malfunctioning of the braking system to overheating of the wheels and to variation in temperature between the wheels will be investigated.

The result of this analysis is then reviewed to see if there is any inconsistency and mismatching with the hypothetically predicted in the beginning of the research. In addition, the result of this analysis will be evaluated and compared with the results of similar studies conducted by other researchers.

The outcome of the whole research will then be thoroughly analyzed with regards to the effect of the locked wheels and drag braking and briefly synthesized for recommendations to be made on how to improve the effect of malfunctioning of the braking system on overheating of the wheels and variation in the temperature of the wheels.

Finally, the result of this analysis will be stated briefly so that it can be a reliable source to be used later in the conclusion stage.

1.7.5. Discussion on findings

The Results from the finite element analysis will be presented in tables, graphs and diagrams obtained from the superimposition of the transient structural and thermal Ansys workbench analysis.

1.8. Significance of the research

First and for most the risk of derailment arising from hot wheels should be minimized significantly and the human causality and property loss associated with it should be avoided. Tread brakes transform the moving energy into heat as a result of the thermal stress on the running surface and the resulting formation of residual stresses during cooling , a wheel breakage may happen. If a wheel breaks because of overheating it will not be able to offer guidance anymore and derailment will be inevitable. Hence, the public at large will be the beneficiary if the risk of derailment associated with overheating of wheels is mitigated.

Furthermore, the maintenance and replacement costs will be significantly minimized and the Ethiopian railway corporation will be benefited by implementing the recommendations and suggestions made at the end of this research. Moreover, the results of this research can be used in the future studies that might be conducted under Ethiopian Railway Corporation regarding overheating of the wheels of freight train.

Chapter2

Literature review

Railway wheels suffer from the enormous heat generated during sever braking action. When locking of wheels occur or during sever drag braking events the high friction generated between the brake block and the wheel or between wheel and rail raises the temperature of the wheel. This rise in temperature brings about the failure of the wheel as a result of thermal crack, spalling or tensile residual stresses.

2.1. Failure of wheels as a result of overheating

1) Thermal cracks

Intense brake heating initiates small radial cracks that are confined to the tread or flange of the wheel which potentially progresses towards the web and lead to wheel failure. Many shallow thermal cracks can be removed by machining, but care must be taken to make sure that the cracks have been completely eliminated.

2) Spalling

Spalling develops as a result of repeated wheel sliding, skidding or slipping over the rail. These service conditions produce brief intense heating of wheel tread surface in rail contact area leading to the development of small transverse thermal cracks. Once these cracks are formed repeated contact with the rail under load leads to progress of fracture beneath the layer affected by heat and portion of the tread surface falls away.

3) Tensile residual stresses

A very important category of wheel failure is brought about by the presence of tensile residual stresses within the rim of the wheel due to overheating following sever drag braking event. At the end of wheel manufacturing compressive residual stresses are induced to the rim of the wheel by heat treatment. These compressive residual stresses are very useful for the wheel to withstand fatigue stresses from the repeated rolling contact with the rail. However, the overheating caused by Sevier drag braking event reverses it to tensile stress. In the presence of tensile residual stresses, small thermal cracks may be sufficient to cause the sudden fracture of the wheel

Many researchers have been conducting researches focused on improving the performance of the wheels of freight trains under sever braking conditions. The efforts made by the researchers cover wide range of issues related to the disc and tread braking systems, thermo-mechanical behavior of the wheels, improved design of railway wheels for a better residual stress distribution, influence of friction materials on thermo mechanical properties of railway wheels etc.

2.2. Thermal capacity of Railway wheels

Shahab Teimorimanesh, Roger lunden and Tore Vernersson, in 2010, have discussed the influence of high braking temperatures on the thermo-mechanical properties of wheels of trains.In their research titled 'braking capacity of railway wheels state-of-the-art survey', several aspects of tread braked wheels were discussed. The temperature, termo-mechanical aspects, wheel design and block brake were the main issues addressed [7].

The influence of heat partitioning between brake block and wheel as well as between wheel and rail on the temperature of the wheel for the cases of drag and stop braking cycles are discussed. In addition, factors like wheel geometry, brake block material together with temperature and residual stresses in the wheel rim were discussed as to how they are used in dimensioning Railway wheels. Furthermore, the effects of the high heat generated during sever drag braking on the reversal of the useful compressive residual stresses in the wheel rim to tensile stress was shown [7].

The friction material of the brake block was another important issue that was discussed in this research. The friction material should always maintain coefficient of friction within a specified range at all service conditions. The fact that the damage it incurs to the wheel tread should be minimum and it shouldn't wear out so easily are the other criteria set out for the brake block in the discussion. Furthermore, for newly innovated brake block friction materials, it is clearly indicated that the temperature of the wheel should not exceed the limit set in UIC standard during brake rig testing. According to the UIC-45 standards, during brake rig testing, temperature 9mm below the tread is accepted if it is less than 600○C or below 700○C if the duration is not greater than 25 minutes or greater than 700○C is also accepted if the duration is not greater than 5 minutes [7].

In this research, one freight and another metro wheels are studied. The freight wheel is considered to study the braking capacity of the wheel in case of sever drag braking and the metro wheel is for analyzing the effect of frequent stop braking cycles. The freight wheel has got a parabolic web and is adapted to withstand sever drag braking for prolonged duration and the metro wheel has got a straight web and is adapted for frequent stop braking cycles. However, the metro wheel is also tested for drag braking to account for the possible unintended brake application (malfunctioning of the braking system) [7].

An axis symmetric finite element model was constructed and the finite element analysis conducted for freight and metro train. The drag braking consists of heating (when the block is applied) and cooling phases (when the train moves at constant speed) and the stop braking cycles are stopping from a specified speed, holding at stationary position for some specified time and accelerating to some specified maximum speed from rest [7].

The result from the finite element analysis showed that the temperature of wheel tread gradually increased after subsequent stop braking cycles and the tread temperature is higher in the freight wheel than the metro wheel. Moreover, average residual stress is calculated from sequentially coupled thermal stress analysis and it is so negligible for stop braking cycle and considerably high for drag braking [7].

2.3. Investigation of effects of different braking systems on rail wheel spalling

S.S Deshpande, S.Srikart, V.K Banthis, J. Jagandeesh and N. Chowdhary, in their publication tiltled ‘ investigation of effects of different braking systems on rail wheel spalling’ have shown that very high temperatures are rapidly developed when railway wheels slide over the rail followed by rapid cooling resulting in martensite micro structure formation in the thermally affected zones (800-1300°C). This in turn makes the wheel brittle and may fail by fracture and thermal cracking .In their research they have used tread and disc brakes and analyzed the wheel slide situation. Analytical solutions for sliding velocity, maximum temperature within the wheel and the heat flow within the wheel and the rail are obtained. The temperature field within the wheel was analyzed using FEA with the corresponding finite element model for both tread and disc brake conditions [8].

TABLE 2-1 Wheel lock up conditions in tread and disc brakes

Braking System	Braking Force per axle(N)	Normal Load on axle(kN)	Braking Torque(N m)	Adhesive Torque (N m)	Wheel Sliding
Tread	8000	162	3800	3845	No
Disc	14225	162	6757	3845	yes

For the specified conditions since no wheel lock up was observed for the tread braked wheels, they continued their analysis with the disc brakes.

They conducted the analysis for sliding velocities of 41.31m/s, 25 .82 m/s and12. 91 m/s with sliding durations of 3, 5 and 10 seconds.

Finally they presented the results by time–temperature transformation curves and the out puts from the finite element analysis. For the specified conditions they also concluded that disc braked wheels are more likely to slide than tread braked ones. The result has shown that the depth of the temperature affected Zone has increased with increase in sliding velocity and sliding duration. Moreover, it was observed that with the sliding velocities of 41.31m/s and 25.82m/s the

TAZ has shown microstructure change from pearlite to martensite. Hence for this two sliding velocities the occurrence of spalling to the wheel treads is inevitable [8].

2.4. An improved Railway wheel to minimize tensile residual stress

Mr. Sayed M. Badruddin, Concordia University, Canada, in his research titled 'An improved Railway wheel to minimize tensile residual stresses after sever drag braking' has proposed three new wheel designs by modifying the outside hub and the inside rim fillets of a parabolic plate wheel profile [9].

Reduction of tensile residual stresses within a wheel by improving fillet profiles will improve the fatigue life of the wheel. The fillet profiles can be represented by the polynomial:

$$Z = ar^3 + br^2 + cr + d$$

Where Z and rare coordinates and a, b, c and d are constants. No weight increment made in the new designs #1 and #2 but there is a 4% weight increase in design #3. By applying 30hp brake force for 60 minutes, the new wheel designs are analyzed elasto-plastically and residual stresses are obtained for each of the three alternative new designs [9].

Among the three new wheel designs, the one with the minimum residual tensile stress is to be selected as an improved design. We could have a reduction of tensile residual stress at one point in the new designs, but an increase of this stress could also happen at another point within the wheel. For this reason a comparison between the four wheel designs is made by using penalty function by giving different weights to 15 different locations on the wheel.. The weights assigned to these points are according to the sensitivity of the location to residual tensile stresses [9].

Finite element model was constructed for all four designs (one old and three new) and the overall result of the FEA suggested that new design #2 is the best design with the minimum tensile residual stress after sever drag braking event [9].

2.5. Summary of the literature review

In all the researches referred above some important aspects of the effect of the braking system on the thermo-mechanical properties of the wheels are discussed. In the first part (braking capacity of Railway wheels) lots of useful issues are addressed except that it lacks the consideration of wheel slide situation, which could also have detrimental effect on the thermo-mechanical properties of the wheel. In addition, the inadvertent drag braking seems to have limited attention and consequently no suggestions were produced to improve the malfunctioning caused drag braking.

In the second research the study is entirely limited to the reaction of the wheels to an event of wheel during an event wheel slides. This research tried to point out that the sever heat developed in the interface of wheel and rail during wheel lock up conditions will directly lead to the micro- structural change of the wheel tread material. It does not include the effect of drag braking event on the thermo-mechanical properties of the wheel material.

The geometry of the wheel profile has a direct influence on the tensile residual stress distribution in the wheel. In the third research the problem of tensile residual stress development within the rim of the wheel is reduced by a new wheel design by modifying the outer –rim fillet and the inner-hub-fillet.

In this thesis the effect of the malfunctioning of the tread braking system on the thermo-mechanical properties of freight wheels will be analyzed using finite element analysis and the sources of wheel locking and inadvertent drag braking will be investigated. In addition, recommendations on how to minimize these malfunctioning of the braking system will be made.

In particular, the effects of drag braking caused by stuck brakes and wheel sliding caused by wheel locking resulting from malfunctioning of freight tread braking on the thermo-mechanical properties of freight train wheels will be thoroughly analyzed.

Chapter 3

Modelling thermo-mechanical effects of braking system malfunctioning

Defining an analytical model the thermal analysis that describes the transfer of the heat generated during the braking to the wheel, the brake block and the surrounding air is very important to simulate the process of drag braking in railway wheels.

The two major events resulting from the malfunctioning of tread braking systems in freight trains are drag braking and wheel slide. The analytical modeling of the thermal effects of these two events of braking system malfunctioning are to be dealt with separately. Drag braking could be the outcome of malfunctioning of tread brakes whereby blocks remain stuck for a considerably long time. Hence, the corresponding analytical modeling quite resembles that of drag braking intentionally applied while descending a slope. Whereas the modeling for wheel slide will be outlined considering the wheel locking caused sliding of the wheel over the rail.

3.1. Freight train braking system malfunctions leading to unintentional braking

There are so many causes of tread brake malfunction that result in unintentional braking. The severity of these unintentional braking depends on the nature of the malfunction itself. For instance, brake shoes friction fade may result in higher braking force on other brake shoes in the next braking action. Malfunction of a brake control valve, on the other hand, leads to application of unintentional emergency braking just as in the case of completely broken brake pipe.

Stuck brakes are mostly sources of unintentional drag braking that induces great heat to the wheels and brake shoes. The possible root causes of stuck brakes are overcharged brake pipe, excessive brake pipe leakage, brake rigging binds or fouls, defective control valve or hand brakes applied.

Even though there could be several causes of malfunctioning of the freight train braking system, the conditions of freight train brake shoe friction fade and the fluctuation in brake pipe pressure are to be considered as the effects of other malfunctions on freight train wheels are more or less the same.

3.1.1. Brake shoe friction fade (heat fade)

During automatic brake application the amount of heat generated is proportional to brake horse power, which in turn depends on the speed of the train and brake retarding force. At higher brake shoe force and wheel speeds, heat fade becomes significant. Higher brake shoe force and wheel speeds generate higher heat within the brake shoe-wheel tread interface than the wheel and brake shoe handle, causing deterioration of the wheel and brake shoe. Deterioration of the brake shoe and flow of metal on the tread of the wheel will diminish the frictional resistance between the wheel and the block brake [10].

The Association of American Rail roads has conducted a braking test on an intermodal freight train descending 2.2% grade to study the temperature dependence of the coefficient of friction between wheel and brake block. According to the results of the test the coefficient of friction between the wheel tread and the brake block is a function of brake shoe force and wheel temperature. It showed that at brake shoe force less than 1000 pounds the coefficient of friction is independent of temperature. However, if the brake shoe force is greater than 1500 pounds friction coefficient decreases with increase in temperature [10].

Heavier cars and locomotives need a higher retarding force for speed control when descending grades and therefore generate higher temperature. In some cases the heat generated while descending grades might exceed the thermal capacity of the wheel tread material and the brake block. The coefficient of friction between the wheel tread and brake shoe friction material will be lowered as the thermal capacity of the brake shoe is exceeded for a sufficiently long time. At this state the brake shoe cannot provide the required retarding force and is said to enter friction fade[11].

For a train to descend long grades safely the necessary amount of drag braking force should be applied at the wheels. However, if sever braking occurred in previous events and one or more brake shoes entered friction fade phenomena, they cannot provide the necessary retarding force of their share. Hence, these retarding forces must be compensated by the remaining brake shoes so that the required total retarding force is generated and the train can descend down long grades safely. Consequently, greater braking force than usual will be applied to the properly working brake shoes to obtain the necessary retarding force. This in turn brings about additional quantity of heat to be tolerated by the remaining wheels.

Here we will try to quantify the additional heat incurred to the wheels due to the occurrence of friction fade at some of the brake blocks in previous sever braking actions and investigate its effect on the thermos-mechanical properties of the wheel.

From AAR investigation reports, four of the 8 brake shoes on a freight locomotive entered the friction fade after a previous severe braking action. However, the train was operating in a very mountainous area [11].

Taking the above as a model, we will carry on our analysis by assuming that 8 of the 24 brake shoes on a locomotive of the freight train running on the Ethio-Djibouti main line, entered friction fade in previous sever braking event. Hence, the calculations of the braking force per wheel will be made by dividing the total retarding force to the number of the properly working tread shoe-wheel tread pairs.

3.1.2. Malfunction of the Automatic Air brake system

There could be many reasons for the unintentional brake applications in freight trains. On one hand, operational factors like misjudgment of the type of braking needed by the operator, giving longer recharging time so that the brake pipe and main reservoir end up overcharged and luck of skill and experience in controlling the train’s speed down grade (cycle braking) contribute a lot to the malfunctioning of the air brake system.

On the other hand, technical problems arising from excessive leakage along the brake pipe, errors in the braking sensitivity and release stability while manufacturing the triple valves, triple valve manufacturing errors regarding the sensitivity of emergency braking and stability of service braking and failure of one or more brake control valves or retaining valves could also be other factors that cause unintentional braking action.

For the purpose of quantifying the unintended braking force applied on the wheels and the resulting heat incurred, we will analyze the effects of overcharged brake pipes and leakage along the train's air brake pipe.

1) Overcharged brake pipes

Unregulated flow of pressurized air from the main reservoir to the brake pipe occurs when the automatic brake valve is in the release position. This could be very useful in charging a long dry trainline in a relatively short time. However, the brake pipe could become overcharged if the valve is left at this position for too long [12].

When there exists an overcharged brake pipe condition, the air pressure in the brake pipe and in the auxiliary reservoir on each car is higher than the setting of the locomotive feed valve. The over charged brake pipe will gradually leak down to the feed valve's set pressure when the automatic brake valve is set at the running position. This will eventually result in the reduction of brake pipe pressure and subsequent application of brakes. This results in situations where brakes are applied and brake pipe is at its fully charged state, leaving no pressure differential with which to release the brakes. Consequently, the brake shoes will remain pressed against the wheel tread till a means is found to bring the pressure within the pipe and auxiliary reservoir to their normal working condition [12].

2) Leakage along the brake pipe

Whatsoever is the cause, brake pipe pressure reduction in automatic air brake systems will lead to the application of brakes depending on the rate at which the brake pipe pressure is decreasing.

There always exist micro leakages along the brake pipe that tend to decrease the brake pipe pressure. There cannot be 100% sealed brake pipes in reality. However, there are standards set for the tolerable brake pipe pressure reduction rates caused by these micro leakages. For instance, according to the AAR standards, during brake pipe leakage test rate of reduction in brake pipe pressure that exceeds 5Psi per minute (35kpa/min) is not tolerable. It will probably trigger braking action somewhere down the line otherwise [13].

Nonetheless, there could still be leakages caused by mechanical failure along the brake pipe that initiate unintentional braking action. The Automatic brake valve being at its release position the brakes could be applied to the wheels unintentionally due to excessive leakage somewhere along the brake pipe line [13].

According to the generally accepted standard, the critical values of air pressure reduction rate, required by release stability and braking sensitivity, are 0.5~1.0 kPa/s (30~60kPa/min) and 5~10 kPa/s respectively. The transition zone between these two critical values is at least 4kPa/s according to above standard. Regulating uniform brake pipe pressure is very important as any pressure reduction greater than or equal to 35kpa/min is almost certain to result in sticking brake shoe condition [14].

This unintentional brake application could be aggravated if there is a triple valve with much higher braking sensitivity and/or higher sensitivity of emergency braking than the standard. Besides, this could even be more aggravated if not enough transition zone is set between the standard ranges of pressure reduction rates corresponding to release stability and braking sensitivity.

3.2. Relationships between pressure in the brake pipe and in brake cylinder

In automatic air brake systems the brake pipe pressure has to be reduced so that brakes are 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 shoe against the wheels. 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.

3.3. 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. The air expansion from the auxiliary reservoir into the brake cylinder could be considered to be an adiabatic and iso-thermal thermodynamic process.

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
- No service quick brake is applied
- The stroke (piston travel) of the brake cylinder is approximately 200mm.

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.

$$PV = C \tag{3.1}$$

Where:

P=absolute pressure

V=Volume of the air

C=constant

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

$$P_o'V_f = (P_o' - r) * V_f + P_z' * V_z \tag{3.2}$$

Where:

P_o'=the rated absolute pressure in the brake pipe

V_f=the volume of auxiliary reservoir

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

P_z'=the absolute pressure in the brake cylinder

V_z=the volume of air in the brake cylinder

Solving for the absolute pressure in the cylinder gives:

$$P_z' = \frac{V_f}{V_z} * r \tag{3.3}$$

The relative pressure in the brake cylinder is then:

$$P_z = (\frac{V_f}{V_z} * r - 100)Kpa \tag{3.4}$$

Considering the effect of leakage and taking the piston travel to be 200 mm . The value of $\frac{V_f}{V_z}$ for railway application is set to be 3.25.

Hence the above equation reduces to:

$$P_z = (3.25 * r - 100)Kpa \tag{3.5}$$

3.3.1. The minimum valid brake pipe pressure reduction

For the brake shoe to be practically pushed against the wheel 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.

Hence, the brake cylinder relative pressure should at least be equal to35kpa for the brake shoe to be pressed against the wheel.

$$(3.25 * r - 100) = 35kpa$$

Solving for the pressure reduction in the brake pipe r gives:

$$r = 41.5384Kpa$$

$$r \approx 40kpa$$

For the sake of safety and convenience we take.

Therefore any leakage that brings about a brake pipe pressure reduction greater than or equal to 40kpa ends up with unintentional braking action.

3.4. Analytical modeling

So many researchers have contributed a lot in the modeling and analysis of the thermo-mechanical effects of different braking systems on the wheels of freight cars. Be it a disc brake or a shoe brake, their effect on thermo-mechanical properties of the wheels of freight cars have been analyzed using two dimensional and three dimensional thermal models by different researchers.

In this part we will simulate the effects of the malfunctioning of tread brake systems in freight trains here in Ethiopia and find analytical and mathematical models for the resulting thermo-mechanical properties of the wheels. While modeling, Inadvertent drag braking (stuck brakes) and wheel slide events will be taken into consideration and their effects on thermo-mechanical properties of the wheels will be modelled.

During the whole analysis in this thesis the inadvertent drag braking resulting from stuck brakes will be treated same as that of the intentionally applied drag braking and the corresponding analytical and numerical modelling is done same way. For both drag braking and wheel sliding cases the transient thermal model that takes the effect of heat loss through convection and radiation is adopted.

3.4.1. Technical and operating conditions of the locomotive

The Ethiopian Railway use the Chinese HXDC1high power Ac drive (7200KW) electric locomotives for freight transport on the main lines. These locomotives are suitable for long distance, steep gradient and heavy load operation.

HXDC1 has a total weight of 138/150t in working order and its axel load changes freely between 23T and 25T. Independent auxiliary inverters with a rated output capacity of 248 KVA are supplied to these locomotives. The Network control system adopts distributed control technology. Eurotrol or CCBII air brake systems and a regenerative electric braking system are used on these locomotives. The car body can bear compressive load of 3000 KN and tension load of 2500KN in the vertical direction [15].

TABLE 3-1 Technical parameters of the locomotive

Item	Parameter	Item	Parameter
Purpose	Freight transport	Locomotive Total Wheel Base	16260 mm
Gauge:	1435mm	Locomotive length (Distance between front and rear coupler centers):	22670 mm
Electric transmission method	AC-DC-AC transmission	Maximum width of locomotive body	3,100 mm
Bogie type	SF 6N	Maximum height (upon pantograph dropping):	4,760mm
Axle arrangement	Co' - Co’	Double heading traction	The two locomotives can be subject to double heading traction and has LOCOTROL system interface reserved
Wheel diameter	1000mm	Power supply system	AC 25 kV, 50 Hz
Locomotive service weight	138T(without additional weight) 150T(with additional weight)	Braking control	Electric braking, auto braking, separate electric pneumatic braking
Starting tractive effort	520 KN (23t), 570 KN (25t)	Mechanical brake	tread brake
Continuous rated speed of locomotive	100 Km/h (23t), 120 Km/h(25t)	Maximum continuous power at the wheel rim	7,200 kW (traction and regenerative braking)
Continuous Tractive Effort of Locomotive	Axle load at 23t 370KN Axle load at 25t 400KN	Maximum electric braking force	370KN(23t), 400 KN(25t)
Minimum radius of curvature negotiable	125 m under the speed of 5 Km/h	Ambient temperature	22^{o}C
Wheel Base:	2250mm＋2000 mm	Transmission Ratio:	6.235(106/17)

All technical parameters summarized in the above table are taken from the specifications of electric locomotives and Rolling stocks obtained from the Ethiopian Railway Corporation except that the mechanical pneumatic brakes are assumed to be tread brakes.

Operational parameters:

- Locomotive's maximum operating speed: 120 km/hr
- Constant deceleration during of stop braking : 1.2 m/s^2
- Maximum gradient of the line : 18‰

3.4.2. Braking force

The dimensions of the brake block are limited by the cylinder pressure and braking force. Hence, careful analysis of the data regarding the cylinder pressure and the braking force will enable us to determine the width and length of the brake block.

A more precise determination of the braking force in the block-wheel contact area can be made by integrating the product of the cylinder pressure and the differential area with respect to the differential angle through which this braking force acts on the given area. The following analysis is based on the figure below.

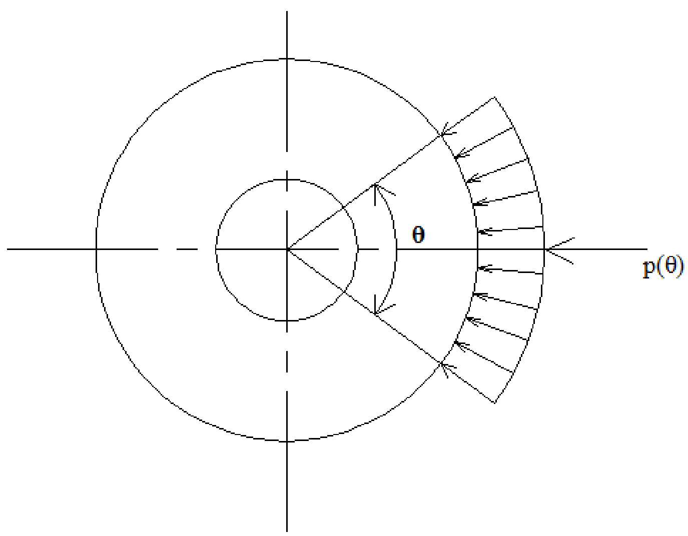

Figure 3-1 Pressure distribution on the wheel from the brake block

The differential length of the brake block is:

$$dl = r d\theta \tag{3.6}$$

Where:

r=the radius of the block brake an

dθ= the differential angle considered

The differential area on which the braking force acts can be calculated as:

$$dA = B(dl) = Brd\theta \tag{3.7}$$

Where B= the width of the block brake and dl is the differential length of the block brake

.

Assuming unit width of the brake shoe the differential braking force acting on the differential area is then calculated as:

$$df = P(\theta) * dA = P_o Cos\theta * rd\theta cos\theta \tag{3.8}$$

Where:

Po= the cylinder pressure

Hence, the brake force can be obtained by integrating the above equation as follows.

$$\int df = \int_{-\theta}^{\theta} P_o Cos\theta * rd\theta cos\theta$$

$$F_N = P_o r \int_{-\theta}^{\theta} cos^2\theta d\theta$$

$$F_N = \frac{p_o r}{2}(\sin(2\theta) + 2\theta) \tag{3.9}$$

Whereas the friction force on the differential area is:

$$F_{fr} = \mu F_N = \mu \int_{-\theta}^{\theta} P_o cos\theta\, rd\theta = 2\mu pr\, sin(\theta) \tag{3.10}$$

The equivalent coefficient of friction can be obtained by:

$$\mu' = \frac{F_{fr}}{F_N} = \frac{2\mu\, p_o r\, sin(\theta)}{\frac{p_o r}{2}(sin(2\theta) + 2\theta)} = \frac{4\mu sin\theta}{\sin(2\theta) + 2\theta} \tag{3.11}$$

3.4.3. Angular velocity of the wheel

While applying brake the longitudinal velocity of the vehicle decreases and eventually it either comes to rest or starts moving with constant velocity (the case with down a slope). Whichever maybe the case the translation of the locomotive can be expressed by the following equation assuming constant deceleration.

$$V(t) = v_o - at \tag{3.12a}$$

The angular velocity of the wheel at any time can also be expressed as:

$$\omega(t) = \frac{v(t)}{r_w} \tag{3.12b}$$

3.4.4. 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 and the change in potential 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:

$$\Delta KE = \frac{1}{2}m{v_o}^2 - \frac{1}{2}m{v_f}^2 \tag{3.13a}$$

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

$$\Delta Ek_r = \frac{1}{2}I{\omega_o}^2 - \frac{1}{2}I{\omega_f}^2 \tag{3.13b}$$

The change in potential energy can be calculated from the gradient as:

$$\Delta E_p = mg\,s_b\,\frac{\delta}{1000} \tag{3.14}$$

The total energy to be converted to heat while braking is:

$$E_b = \Delta KE + \Delta Ekr + \Delta Ep \qquad (3.15)$$

$$E_b = (\tfrac{1}{2}\, m{v_o}^2 - \tfrac{1}{2} m v_f^2) + (\tfrac{1}{2} I{\omega_o}^2 - \tfrac{1}{2} I \omega_f^2) + mgs_b \frac{\delta}{1000} \qquad (3.16)$$

The moment of inertia of the rotating parts can be calculated from:

$$I = \sum_{i=1}^{n} m_i {r_i}^2 \qquad (3.17)$$

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:

$$E_b = \tfrac{1}{2} m{v_o}^2 - \tfrac{1}{2} m v_f^2 + \tfrac{1}{2}{r_w}^2 \sum_{i=1}^{n} m_i\,({\omega_o}^2 - \omega_f^2) + mg\,s_b \frac{\delta}{1000} \qquad (3.18)$$

3.4.5. Heat energy generated

Assuming that all the braking energy is converted to heat at the end of the braking process the heat energy generated is equal to the braking energy.

Hence,

$$q_{gen} = E_b \qquad (3.19a)$$

The heat flux to the wheel is then obtained by using appropriate heat partitioning parameters as:

$$\dot{Q} = \frac{\varepsilon_1 \varepsilon_2 F_{fr}(v_o - at)}{A_b} \qquad (3.19b)$$

3.4.6. Analytical solution for maximum wheel temperature

The maximum wheel temperature can be approximately computed using Newton's law of cooling. According to this law the change in the wheel's heat energy due to convection and radiation heat transfer to the surrounding is directly proportional to the difference in temperature between the wheel and the surrounding.

The differential equation governing the cooling of the wheel through radiation and convection is:

$$\dot{q} = UA\Delta T = UA(T - T_{\infty}) \qquad (3.20)$$

Where:

U= the overall heat transfer coefficient

A= the surface area of the wheel exposed to the surrounding

T= the temperature of the wheel

T_{∞}= the surrounding temperature

The above first order differential equation can be integrated to get the change in heat energy of the wheel during cooling through the convection and radiation to the surrounding air as follows.

$$\int_0^{t_1} dq = \int_0^{t_1} UA(T - T_{\infty})dt \qquad (3.21a)$$

The limit of integration limit (0—t_1) refers to the braking time. The left hand side of this equation equals the braking energy computed in the previous subsection. Hence, the above equation can be rewritten as:

$$\Delta E = \int_0^{t_1} UA(T - T_{\infty})dt \qquad (3.21b)$$

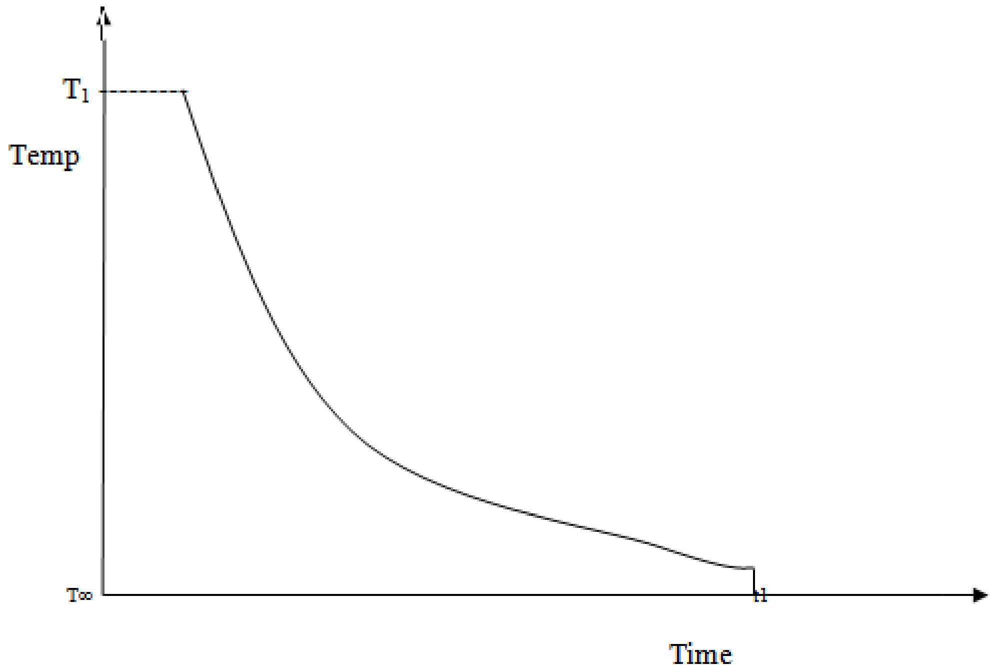

Figure 3-2 Maximum wheel temperature versus time

The ratio of temperature difference between surrounding and wheel at any time t to the maximum temperature difference is Given by:

$$\frac{T-T_{\infty}}{T_1-T_{\infty}} = e^{\frac{-UA}{mc}t} \tag{3.22}$$

Where:

T= temperature of the wheel at any time

T_1= the maximum temperature in the braking event

T_{∞}= the temperature of the surrounding air

U=overall heat transfer coefficient

M= mass of the wheel

A=surface area of the wheel exposed to the surrounding air

C=specific heat capacity of the wheel

Hence, using the above relation equation 3.21a can be rewritten as:

$$\Delta E = \int_0^{t_1} UA(T_1 - T_{\infty})\, e^{\frac{-UA}{mc}t} dt \tag{3.23}$$

The change in heat energy within the wheel is given by:

$$q = \Delta E = cm\Delta T \tag{3.24}$$

Substituting equation 3.23 into 3.22 and solving for T_1 gives:

$$T_1 = T_\infty + \frac{\Delta T}{1 - e^{\frac{-UA}{mc}t_1}} \qquad (3.25)$$

3.4.7. Analytical Model for drag braking

Precise determination of the heat generated during friction, its distribution to the wheel and brake block and the surrounding air are of importance in the thermal analysis of the drag braking event in freight trains **[16].**

The heating power generated between the contacting surfaces of the tread of the wheel and the brake block is given by:

$$q(r) = f_f r \omega \qquad (3.26)$$

Where:

$f_f = the\ frictional\ force\ per\ unit\ contact\ area$

$\omega = the\ angular\ velocity\ of\ the\ wheel$

$r = the\ radius\ of\ the\ wheel$

The frictional force per unit contact area can be calculated as:

$$f_f = \frac{\mu F_N}{A} \qquad (3.27)$$

Where:

$F_N = the\ normal\ braking\ force$

$\mu = the\ friction\ coefficient$

$A = contact\ surface\ area\ between\ the\ wheel\ and\ the\ brake\ block$

The heat conduction through blocks and wheel is a time dependent phenomena that can be computed using transient heat transfere equations as follows.

$$\rho c_p \left(\frac{\partial T}{\partial t}\right) + \nabla(-k\nabla T) = Q - \rho c_p u \nabla T \qquad (3.28)$$

Where the material properties for the wheel and block are:

$\rho = density\ of\ wheel\ or\ block\ material$

$k = Thermal\ conductivity\ of\ wheel\ or\ block$

$c_p = specific\ heat\ capacity\ of\ wheel\ or\ block$

$u = velocity\ field$

$Q = heating\ power\ per\ unit\ volume$

The heat dissipated by convection and radiation can be calculated as:

$$q_{diss} = -h\left(T - T_{reff}\right) - \varepsilon\sigma\left(T^4 - T_{reff}{}^4\right) \qquad (3.29)$$

Where:

$h = the\ convective\ film\ coefficient$

$\sigma = Stefan\ Boltzman\ constant (5.67 * 10^{-8})$

$T_{reff} = surrounding\ temperature$

The convective film coefficient can be obtained from:

$$h = \frac{0.037 k_a}{2r} R_e{}^{0.8} P_r{}^{0.33}$$

$$h = \frac{0.037\ k_a}{2r} * \left(\frac{2\rho_a r v}{\mu_{va}}\right)^{0.8} * \left(\frac{c_{pa}\mu_{va}}{k_a}\right)^{0.33} \qquad (3.30)$$

Where:

$k_a = thermal\ conductivity\ of\ the\ air$

$\rho_a = density\ of\ the\ air$

$\mu_{va} = viscosity\ of\ the\ air$

$c_{pa} = specific\ heat\ capacity\ of\ the\ air$

3.4.8. Modeling for thermal effects of wheel slide

When the braking torque is higher than the adhesion torque the wheel slides over the rail. Insufficient adhesion between wheel and rail caused by the presence of contaminants in the wheel and rail interface, reduced weight of wagon or higher braking force due to malfunctioning of the braking system will ultimately result in unwanted wheel slide phenomena [17].

During wheel sliding large amount of heat is generated as a result of friction between the wheel and rail. The thermal analysis of wheel sliding resulting from malfunctioning of the braking system is here carried out by calculating the heat partitioning parameters for the wheel and rail respectively.

3.4.9. Analytical solution for sliding velocity.

The sliding velocity of the wheel is determined considering two brake bocks per wheel. The analysis is based on figure 3-3 below.

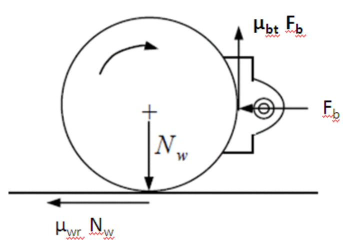

Fig. 3-3 Forces acting on the wheel while braking

From the above figure the braking torque can be calculated as:

$$T_{br} = F_b \mu_{bt} R_b \tag{3.31a}$$

Where:

$F_b = braking\ force$

$\mu_{bt} = coefficient\ of\ friction\ between\ wheel\ treadand\ brake\ block$

$R_b = perpendicular\ distance\ to\ the\ wheel\ rail\ contact\ point$

On the other hand the adhesion torque is given by:

$$T_{ad} = N_w \mu_{wr} R \tag{3.31b}$$

Where:

$N_w = wheel\ load$

$\mu_{wr} = coefficient\ of\ friction\ between\ wheel\ and\ rail$

$R = radius\ of\ the\ wheel$

Assumptions made for calculation of sliding velocity.

- Wheels roll at the centerline.
- Track irregularities are negligible
- During sliding coefficient of friction and braking force do not change

The necessary condition for wheel slide is:

$$T_{br} > T_{ad}$$

$$F_b \mu_{bt} R_b > N_w \mu_{wr} R$$

Hence the net torque on the wheel is then given by:

$$T_N = T_{br} - T_{ad} = 2F_b \mu_{bt} R_b - N_w \mu_{wr} R$$

The angular acceleration is then given by:

$$\dot{\alpha} = \frac{2F_b \mu_{bt} R_b - N_w \mu_{wr} R}{I} \tag{3.32}$$

The deceleration of the train in the longitudinal direction can be obtained as follows.

$$a_x = \frac{F_{xt}}{M} \tag{3.33}$$

Where

$F_{xt} = longitudinal\ tractive\ force$

$M = \frac{N_W}{g} = mass\ of\ the\ rolling\ stock$

Finally, the sliding velocity of the wheel is given by:

$$V_s = V - a_x t \tag{3.34}$$

Where:

$V = velocity\ of\ the\ rolling\ stock$

$a_x = longitudinal\ acceleration\ of\ the\ rolling\ stock$

3.4.10. Mathematical model for interface temperature

According to Martin Ertz and Claus Knothe A dimensionless parameter, *L*, called “Peclet” number, which is the ratio of the surface speed to the rate of diffusion of heat, is used to analyse the condition where velocities of wheel and rail with respect to contact patch are different. Parameter *L* has to be taken into account for heat conduction problem.[18]

$$L = \frac{av}{2k} \tag{3.35}$$

Where:

a=semi-major axis length of the contact elliptic area.

v=speed of the moving heat sourse

k=Thermal diffusivity

Thermal diffusivity (k) can be obtained from:

$$K = \frac{\lambda}{\rho c} \tag{3.36}$$

Where:

λ= thermal conductivity

ρ=density

c=specific heat capacity

According to knothe and Ertz if L is greater than 10 the heat conduction occurs only perpendicular to the contact patch and the transverse and lateral heat conductions can be neglected[18].

The analytical solution for interface temperature will be calculated based on the following assumptions.

- Pure sliding occurs in the whole of the contact patch.
- Coefficient of friction between wheel and rail is constant.
- Contact patch size is constant and does not vary with pressure and temperature
- The effect of elastic deformation is neglected
- The kinetic energy of the train is completely transformed into heat

According to Knothe the maximum wheel-rail interface temperature can be calculated as:

$$T_{max} = \frac{1.276\varepsilon\mu V_s P_o}{\beta_w}\sqrt{\frac{a}{V_s}} \tag{3.37}$$

And the average temperature in the contact patch is:

$$T_{avg} = \frac{0.853\varepsilon\mu V_s P_o}{\beta_w}\sqrt{\frac{a}{V_s}} \tag{3.38}$$

Where:

ε = heat partitioning parameter

μ = coefficientof friction between wheel and rail

V_s = sliding velocity

P_o = maximum hertz contact pressure

β_w = thermal penetration coefficient of wheel

a = semi – major axis of the elleptical contact area

The heat generated at the interface of wheel and rail is then conducted to the wheel and rail and the remaining will be dissipated to the surrounding air through convection and radiation. After deducting the heat dissipated from the total heat generated the heat flow to the wheel and the rail will be governed by heat partitioning parameter (ε) as follows.

The heat to be conducted to wheel and rail is calculated as:

$$\dot{q}_{con} = \dot{q}_f - \dot{q}_{diss} \tag{3.39}$$

Where:

q_{con} =heat to be conducted to wheel and rail

q_f=heating power generated at the interface

The heat partitioning parameter can be obtained from

$$\varepsilon = \frac{t^{0.5}}{t^{0.5}+0.863\,(\frac{a}{v_s})^{0.5}} \tag{3.40}$$

Then the heat flow to wheel and rail respectively are calculated as:

$$\dot{q}_w = (1-\varepsilon)\dot{q}_{con} \tag{3.41}$$

$$\dot{q}_r = \varepsilon * \dot{q}_{con} \tag{3.42}$$

3.5. Determination of Heat flux due to malfunctioning of the braking system

The heat flux induced to the wheel of the freight train wheels as a result of the malfunctioning of the braking system will be calculated below. The heat fluxes due to previous friction fade in some of the brake shoes, due to overcharged brake pipe and due to excessive leakage in the brake pipe are to be determined.

While calculating the heat flux induced to the tread in case of friction fades, average velocity of the locomotive is taken into account. Whereas in cases of the heat flux caused by brake pipe overcharge and excessive brake pipe leakage constant velocity of the locomotive is considered as these are unintentional braking applied to some wheels in the locomotive.

3.5.1. Heat flux resulting from the previous friction fade

The locomotive of the freight train under consideration has 12 wheels and 24 brake shoes (two brake shoes per wheel. Assuming that 8 of the 24 brake shoes entered friction fade in the previous sever braking event, the total heat flux generated in the next braking action on the remaining brake shoes-wheel interfaces will be calculated as follows.

TABLE 3-2 Quantities determining running condition

Item	Values
Mass of the vehicle – M [kg]	150,000
Start velocity – v_0[m/s]	33.33
Deceleration – a [m/s^2]	1.2
Braking time – t_b[s]	27.78
Effective area of the brake shoe[m^2]	362*87=31494*10^{-6}
Radius of the wheel – r_w[m]	0.500
Maximum Gradient of the track – δ [‰]	18
Friction coefficient brake shoe– μ [/]	0.25
Central angle	45^o

The brake block dimension and coefficient of friction are obtained from suppliers catalogue and the maximum gradient is obtained from the Addis - Mojo main line profile delivered by ERC. The rest are calculated, assumed and technical parameters in the previous and succeeding sections.

The braking energy required to bring the locomotive from maximum operating speed to a standstill can be calculated using equation 3.16 as:

$$E_b = \frac{1}{2}mv_o{}^2 - \frac{1}{2}mv_f^2 + \frac{1}{2}r_w{}^2 \sum_{i=1}^{n} m_i\,(\omega_o{}^2 - \omega_f^2) + mg\,s_b\,\frac{\delta}{1000}$$

$$I = \sum_{i=1}^{n} m_i r_i^2 = 0.5^2 * (389 * 12) = 1167 kgm^2$$

The braking distance S_b can be calculated as:

$$S_b = v_o t_b - \frac{1}{2} a t_b^2$$

$$t_b = \frac{v_o}{a} = \frac{33.33}{1.2} = 27.78 sec$$

$$S_b = 33.33 * 27.78 - \frac{1}{2} * 1.2 * 27.78^2 = \mathbf{462.87036m}$$

Inserting the above values in equation (1) the total braking energy is calculated as:

$$E_b = \frac{1}{2}(150 * 1000 * 33.33^2) + \frac{1}{2}(1167 * 66.66^2) + 150 * 1000 * 9.81 * 462.87036 * \frac{18}{1000}$$

$$= 83316667.5 + 2592814.69 + 11306515.2 = \mathbf{97215997.4}$$

Since the brake shoes on four of the wheels entered friction fade and could not provide the retarding force of their share, the braking energy obtained above is to be divided by 8 to get the braking energy required from a single wheel.

Hence, the braking energy per wheel is given by:

$$E_b{}' = \frac{E_b}{8} = \frac{97215997.4}{8} = \mathbf{12151999.7J}$$

The corresponding friction force between the brake shoe and the wheel is to be calculated as follows:

$$E_b{}' = 2F_{fr} r\theta$$

$$F_{fr} = \frac{E_b{}'}{2r\theta} = \frac{12151999.7}{2 * 925.9074 * 0.5} = \mathbf{13124.4223N}$$

The heat partitioning parameter used for the conduction of the heat to the wheel and the brake block is $\varepsilon = 0.25$[19]

where :

$q_b = \varepsilon E_b{}'$ and $q_w = (1 - \varepsilon) E_b{}'$

Hence, $q_w = 0.9 * 0.75 * \mathbf{12151999.7} = \mathbf{8202599.8J}$

Braking force

The ratio of friction force between the brake shoe and the wheel tread to the braking force will give the equivalent coefficient of friction.

$$\frac{F_{fr}}{F_N} = \frac{4\mu sin\theta}{\sin(2\theta) + 2\theta} = \mu'$$

$$F_N = \frac{F_{fr}(\sin(2\theta) + 2\theta)}{4\mu sin\theta} = \frac{13124.4223 * (sin90 + \frac{\pi}{2})}{4 * 0.25 * sin45} = \mathbf{47723.0788N}$$

The instant heat flux to the wheel is then calculated from equation 3.19b as:

$$\dot{Q} = \frac{\varepsilon_1 \varepsilon_2 F_{fr}(v_o - at)}{A_b} = \frac{0.9 * 0.75 * 13124.4233(33.33 - 1.2t)}{31494 * 10^{-6}}$$

$$= (\mathbf{9375436.41 - 337549.46}\boldsymbol{t})\frac{\boldsymbol{W}}{\boldsymbol{m^2}}$$

Similar calculations were carried out to get the heat flux if two wheels of the locomotive entered friction fade in the previous sever braking action and the results were summarized below.

TABLE 3-3 Summary of calculated heat flux resulting from friction fade

No of blocks fiction faded	Braking energy per wheel [KJ]	Friction force [KN]	Braking force [KN]	Heat flux [KW/m^2]
8	12152.000	13.124	47.723	9375.436-337.550t
4	9721.600	10.500	38.179	7500.349-270.04t

3.5.2. Determination of convective heat transfer coefficient

The convective heat transfer coefficient is calculated from equation 3.30 taking the quantities for air properties from table 3-4 and the maximum operating speed of 33.33 m/sec.

TABLE 3-4 Material properties of the wheel, brake block and air [15]

Property	Railway wheel	Brake block	Air
Density(ρ) [kg/m^3]	7850	7200	1.17
Specific heat capacity(c) [J/kg ^{o}k]	486	510	1100
Thermal conductivity(k) [W/m ^{o}k]	52	45	0.026
Poisson's ratio (ε)	0.28	0.31	-
Viscousity(μ_{va})	-	-	1.8*10^{-5}

The material properties in the above table are obtained from 'Modelling thermal effects in braking systems of railway vehicles' [15].

Using Equation 3.30 we obtain the value of the convective heat transfer coefficient as follows.

$$h = \frac{0.037k_a}{2r} * (\frac{2\rho_a r v}{\mu_{va}})^{0.8} * \left(\frac{c_{pa}\mu_{va}}{k_a}\right)^{0.33} \approx \mathbf{98.072}\frac{\boldsymbol{W}}{°\text{C}}$$

3.5.3. Heat flux resulting from overcharged brake pipe.

The standard brake pipe pressure for a freight train is 500 KPa. Due to negligence in operation or improper inspection before coupling cars, the brake pipe might get overcharged. Since I couldn't get a guide line to base my assumptions, I tried to quantify the heat flux on the wheel assuming 10% and 15% brake pipe overcharge.

Assuming 10% brake pipe pressure overcharge, the corresponding brake cylinder pressure is obtained from equation 3.5 as:

$$P_z = (\frac{V_f}{V_z} * r - 100)KPa$$

$$r = 0.1 * 500KPa = \mathbf{50KPa}$$

Hence, $P_z = 3.25 * 50KPa - 100KPa = \mathbf{62.50KPa}$

Assuming the central angle of the brake shoe to be 45^o, the corresponding braking force can be calculated using equation 3.9 as:

$$F_{br} = F_N = \frac{P_o r}{2}(\sin(2\theta) + 2\theta) = \frac{62.5 * 0.5}{2}\left(\sin(90^o) + \frac{\pi}{2}\right) = \mathbf{40.186926KN}$$

The corresponding friction force can be calculated using equation 3.11 as:

$$F_{fr} = \frac{4\mu sin\theta}{\sin(2\theta) + 2\theta} * F_N = \frac{4 * 0.25 * \sin(45)}{\sin(90) + \frac{\pi}{2}} * 40.186926KN = \mathbf{11.0518894KN}$$

Braking power

The braking power will be calculated using equation 3.26 as:

$$q = F_{fr} * v = 11.0518894kN * 33.33\ \frac{m}{sec}\ = 368.3595KW$$

Heat flux

Assuming that 10% of the heat is dissipated through radiation and convection and the heat partitioning parameter $\varepsilon = 0.75$, the instant heat flux to the wheel can be calculated from equation 3.19b as:

$$\dot{Q} = \frac{\varepsilon_1 * \varepsilon_2 * q}{A_b} = \frac{0.9 * 0.75 * 368.3595 * 10^3}{31494 * 10^{-6}} = \mathbf{7894921}\frac{W}{m^2}$$

The results for 10 and 15% overcharged brake pipe cases could be summarized as follows.

TABLE 3-5 Summary of calculated heat flux resulting from brake pipe overcharge

Extent of overcharge	Cylinder pressure [KPa]	Braking force [KN]	Friction force [KN]	Braking power [KW]	Heat flux [KW/m^2]
10%	62.5	40.186926	11.0518894	368.3595	7894.921
15%	75	48.20243	13.25625	441.8308	9469.607

3.5.4. Heat flux resulting from brake pipe leakage.

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 140KPa. There is 100KPa range between the minimum and maximum valid brake pipe pressure reduction[20].

As there are no valid data on which to base our assumption, we may assume the leakage brought about 5%, 10% and 15% into the valid brake pipe pressure reduction and quantify the corresponding braking force, friction force, braking power and heat flux induced to the wheel. Hence, we will consider 40+0.05*500=45KPa, 40+0.01*500=50KPa and40+0.15*500=55KPa brake pipe pressure reductions and quantify the corresponding heat fluxes.

Heat flux corresponding to r=45KPa (5% within the valid brake pipe pressure reduction) can be calculated as follows.

The corresponding pressure in the brake cylinder can be obtained using equation 3.5 as:

$$P_z = \left(\frac{V_f}{V_z} * r - 100\right) KPa$$

$$= 3.25 * 45 - 100 = \mathbf{46.25\,KPa}$$

The corresponding braking force exerted on the brake shoe is calculate using equation 3.9 as:

$$F_N = \frac{P_o r}{2}(\sin(2\theta) + 2\theta)$$

$$= \frac{46.25 * 0.525}{2} * \left(\sin(90^o) + \frac{\pi}{2}\right) = \mathbf{29.7248325\,KN}$$

The friction force between the brake shoe and the wheel tread can be obtained from equation 3.11 as:

$$F_{fr} = \frac{4\mu \sin\theta}{\sin(2\theta)+2\theta} * F_N = \frac{4*0.25*\sin 45^O}{\sin(90)+\frac{\pi}{2}} * 29.7248325 = \mathbf{8.1746875\,KN}$$

Braking power

The braking power can be obtained from equation 3.26 as:

$$q = F_{fr} * v = \mathbf{272.462334\,KW}$$

Heat flux

Assuming 10% of the heat is dissipated through radiation and convection and the heat partitioning parameter $\varepsilon = 0.75$, the instant heat flux to the wheel is calculated using equation 3.19b as:

$$\dot{Q} = \frac{\varepsilon_1 \varepsilon_2 * q}{A_b} = \frac{0.9 * 0.75 * 344.315588 * 10^3}{31494 * 10^{-6}} = \mathbf{5839.591} \frac{KW}{m^2}$$

Similar computations will deliver the results corresponding to 10% and 15% within the valid brake pressure reductions and the results are summarized below.

TABLE 3-6 Summary of calculated heat flux resulting from excessive brake pipe leakage

% in to valid range	Brakepipe pressure reduction [KPa]	Brake cylinder pressure[KPa]	Braking force[KN]	Friction force [KN]	Heat flux [KW/m^2]
5%	45	46.25	29.725	8.1746875	5839.591
10%	50	62.5	40.169	11.046875	7891.332
15%	55	78.75	50.613	13.9190625	9943.087

3.5.5. The heat flux resulting from wheel sliding

When the braking torque is higher than the adhesion torque, the wheel slides over the rail and high heat will be produced within a relatively short period of time.

$$\dot{q}_f = \frac{F_f * V_s}{A_c} = \frac{\mu_{wr} F_N * V_s}{A_c}$$

The wheel of the locomotive is likely to slide over the rail when most of the brake blocks in the locomotive enter friction fade.

Let us check the case where 8 brake blocks enter friction fade.

The torque due to the friction between brake block and wheel tread (braking torque) can be calculated using equation 3.31a as:

$$T_{br} = \mu_{br} F_b R_w = F_{fr} * R_w = 13124.4223 * 0.5 = \mathbf{6562.21165 J}$$

Whereas the torque due to the friction between wheel and rail (Adhesion torque) can be obtained using equation 3.31b as:

$$T_{wr} = \mu_{wr} F_N R_w = 0.35 * (12500 * 9.81) * 0.5m = \mathbf{21459.375}\boldsymbol{J}$$

Hence, the braking torque is much less than the adhesion torque resulting in a condition of no wheel slide.

Chapter 4

Finite element analysis

4.1. Introduction

The finite element method is a numerical procedure that can be applied to obtain solutions to a variety of problems in engineering. Steady, transient, linear, or nonlinear problems in stress analysis, heat transfer, fluid flow, and electromagnetism problems may be analyzed with finite element methods. It is a numerical method used to solve complex system of differential equations at equilibrium.

In the Pre-processing of finite element formulation, the process of dividing the big domain into finitely many building blocks (elements) that are connected at their nodes is carried out by defining them by approximate functions called shape functions. These discrete parts are then assembled together to represent the complete problem. Boundary conditions and loads are also applied in the pre-processing stage. Then, analysis follows to solve system of linear or nonlinear algebraic equations and get nodal values of the quantities for the elements. Finally, in the Post processing, the nodal values are processed so that other values of interest at any other location and given time can be obtained depending on the type of the analysis.

4.2. Finite element analysis using Ansys-14.5

ANSYS software is generally used to analyze engineering problems that are too complex to compute by hand in the design and optimization of a system. Its capability to solve the finitely many equations defining the discrete elements created in the meshing process makes it the most suitable mathematical tool to be chosen. It implements the preprocessing, analysis and post processing stages to analyze a given system.

4.2.1. General procedures for the analysis

1) Preprocessing

In this part the geometry will be defined by using the various commands like key points, lines, areas, volumes etc or one can also import a model that is already constructed using catia (iges, igs files). Material selection and meshing are also carried out In this part. Meshing will divide the whole object into many finite parts called elements that are connected at the nodes.

2) Solution

This is a stage where all the constraints and loads are applied and solutions are obtained for the quantities of interest. Choosing the analysis type, defining loads and controlling outputs by using the load step options are to be performed before solving the problem.

3) Post processing

This stage makes ANSYS software the most suitable mathematical tool to obtain the results of the analysis in a user-friendly manner. Stress distribution, temperature gradients, deformation, displacement, etc can be obtained with colored contours that best describe the response of the system to the loading conditions stated in the solution stages. In addition, graphical and tabular outputs are available for the user to analyze and compare outputs corresponding to given set of inputs. Moreover the user is able to make any changes of inputs like material property, loading conditions, etc to obtain another set of outputs in the same project as in the case of optimization.

4.3. Analysis of the freight train wheel using ANSYS -14.5

Ansys version 2012 is used to make the coupled transient structural and transient thermal analysis of the effect of the heat fluxes due to malfunctioning of the braking system on the wheel's thermo-mechanical properties. The finite element model constructed using Catia V-5 is imported to the ansys14.5 workbench and the effects of heat fluxes from friction fade, overcharged brake pipe and excessive brake pipe leakages are analyzed separately. Moreover, the heat fluxes corresponding to the various braking system malfunctioning calculated in the previous chapter are all used in the process so that we can see the extent of thermal load each set of conditions bring to the wheel.

Assumptions

- Material properties are isotropic and do not vary with 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.
- The average of the heat flux to the wheel over the contact area between the brake block and the wheel is uniform.
- Pressure in the contact area between brake block and wheel as well as between rail and wheel is uniform.
- The wear on the contact surfaces is negligible

4.3.1. Geometrical model

CATIA V-5 is used to generate the wheel by using UIC-ERRI standard wheel profile given on the UIC-510-2-2004 leaflet. The contact areas with the brake block and with the rail are also properly placed so that the pressures corresponding to the braking force and the reaction to the wheel load are to be applied during the analysis. The three dimensional model of the wheel is then saved as igs file for later use in the Ansys workbench.

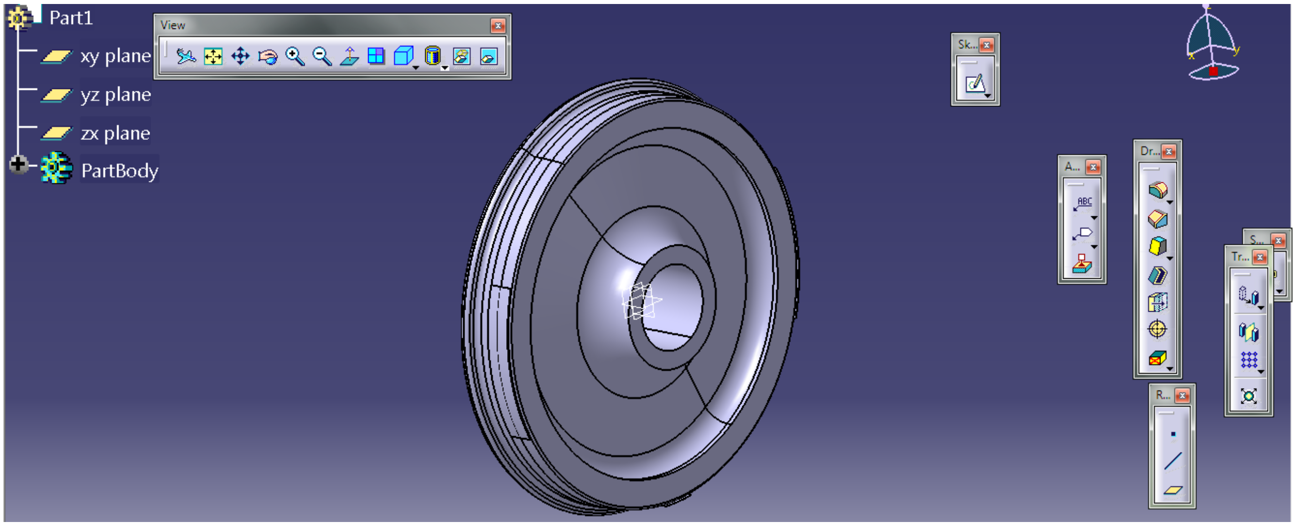

Figure 4-1 Finite element model constructed using CATIA V-5

4.3.2. Wheel material property definition

The wheel to be used for freight application in Ethiopia is a rim treated cast steel that meets the requirements set on the UIC-510-2 standards. The material properties used for the finite element analysis are summarized in table 4.1 as follows.

TABLE 4-1 Material property and geometrical data of the wheel

Mechanical properties of the wheel		**Geometrical data of the wheel**	
Density(ρ)	8750kg/m^3	mass	389kg
Specific heat capacity(c)	486 J/kg °k	Diameter(d)	1000mm
Thermal conductivity(K)	52 w/m °k	Hub- bore diameter	185mm
Poisson's ratio(ε)	0.28	Distance between inside face of hub and rim	55mm
Modulus of elasticity(E)	$2.11*10^{11}$	Web-shape	ss

The geometrical data in the above table are obtained from http//www.wheelsworld.com on 27-02-2015 at 15:02 PM.

4.3.3. 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 small free triangular elements using the automatic mesh generation command. The meshed wheel is shown in fig 4.2 below.

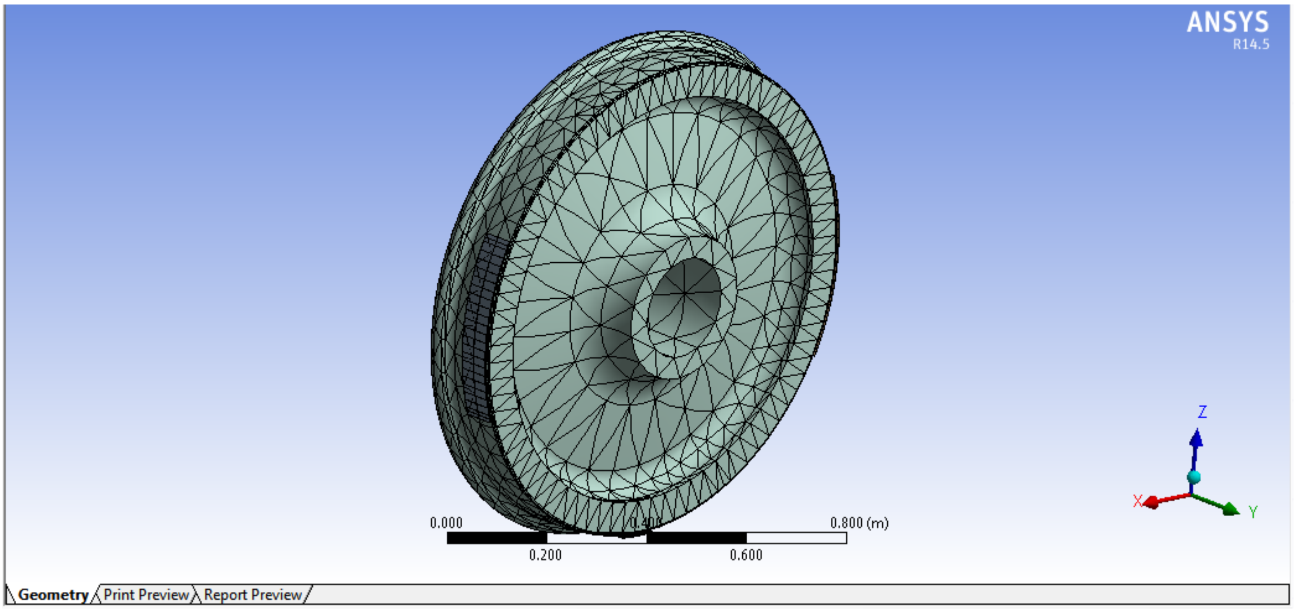

Figure 4-2 Mesh generation

4.3.4. Defining Boundary conditions and loads

After the meshing is completed it is very important first to define the initial boundary conditions so that the forthcoming analysis will have some references to begin with.

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 wheel are the heat flux, pressure at the contact areas with the brake block and with the rail, angular velocity and gravitational acceleration. Hence, the boundary conditions applied to this analysis are as follows.

- Ambient temperature=22ºC

- Initial wheel temperature=30^{o}C
- Film coefficient=98.072W/m^2
- Fixed support on the inside faces of the hub.

The normal force at the contact with the rail is to be applied as a pressure to the contact area. The braking force from the brake shoe is also to be applied as a pressure on the contact area between brake shoe and wheel. The angular velocity of the wheel is applied as a component about the y-axis and gravitational acceleration is applied in the z-direction. The heat fluxes obtained from the various cases of braking system malfunctioning are applied to the wheels tread turn by turn.

Brake block-wheel contact area=31494mm^2

Tread area exposed to heat flux=219900mm

4.3.5. Summary of the heat fluxes and corresponding brake block pressures

The heat flux to be applied to the tread of the wheel and the pressure exerted by the brake block on the wheel tread calculated in the modelling chapter are summarized in the table below.

TABLE 4-2 Summary of loads on the wheel

Braking system malfunctioning		Heat flux over the tread[KW/m^2]	Pressure by the brake block [KPa]
Friction fade	4 blocks faded	537.096694	385.724389
	8 block faded	671.373287	482.1487047
Extent of Brake pipe overcharge	10 %	1130.70829	406.011315
	15%	1356.23374	486.9916145
Extent of Excessive leakage	5% leakage	836.344137	300.31150232
	10% leakage	1130.19478	405.822314
	15% leakage	1404.04542	511.341207
Normal braking		451.9722486	324.58520105

Chapter 5

Result Analysis and discussion

The effect of malfunctioning of the braking system on thermo-mechanical properties of the freight train wheel is analyzed using ANSYS 14.5 software. The temperature gradient, Von-misses stress distribution and total deformation due to previous friction fade, brake pipe overcharging and excessive leakage from the brake pipe are computed separately. Each analysis is made for a braking time of 28 seconds.

The results obtained from the combined transient thermal and structural analysis for each of the three malfunctioning of the braking system 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.

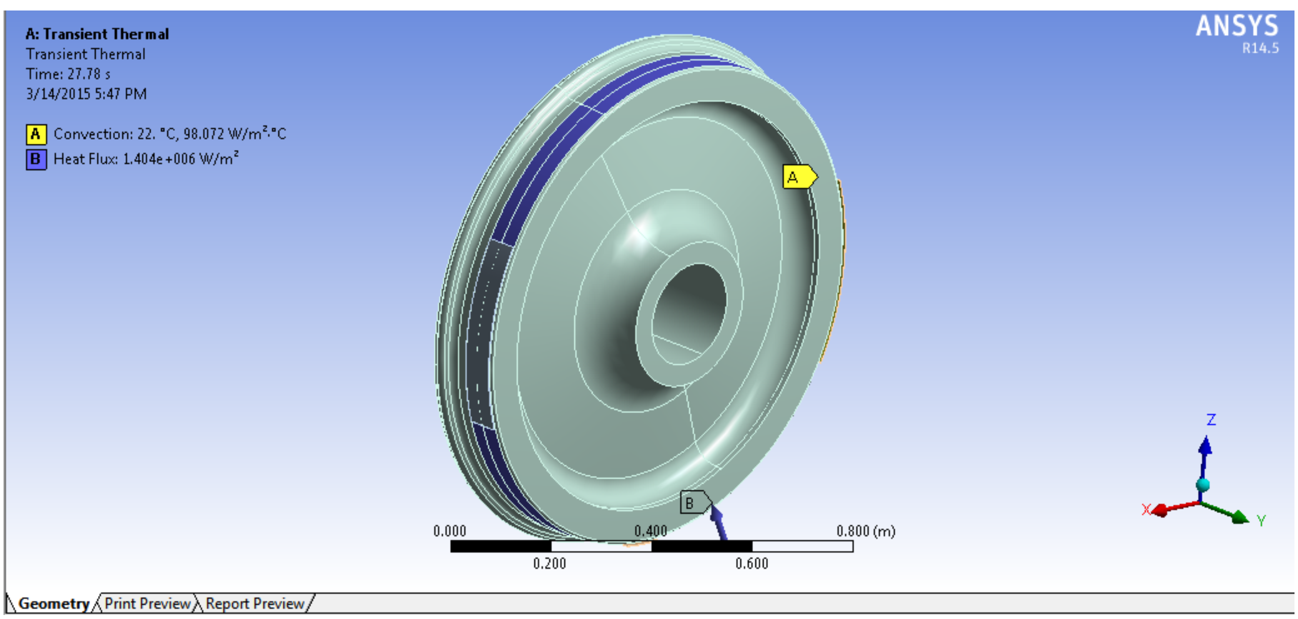

Figure 5-1 Heat flux and convective film coefficient applied on the wheel

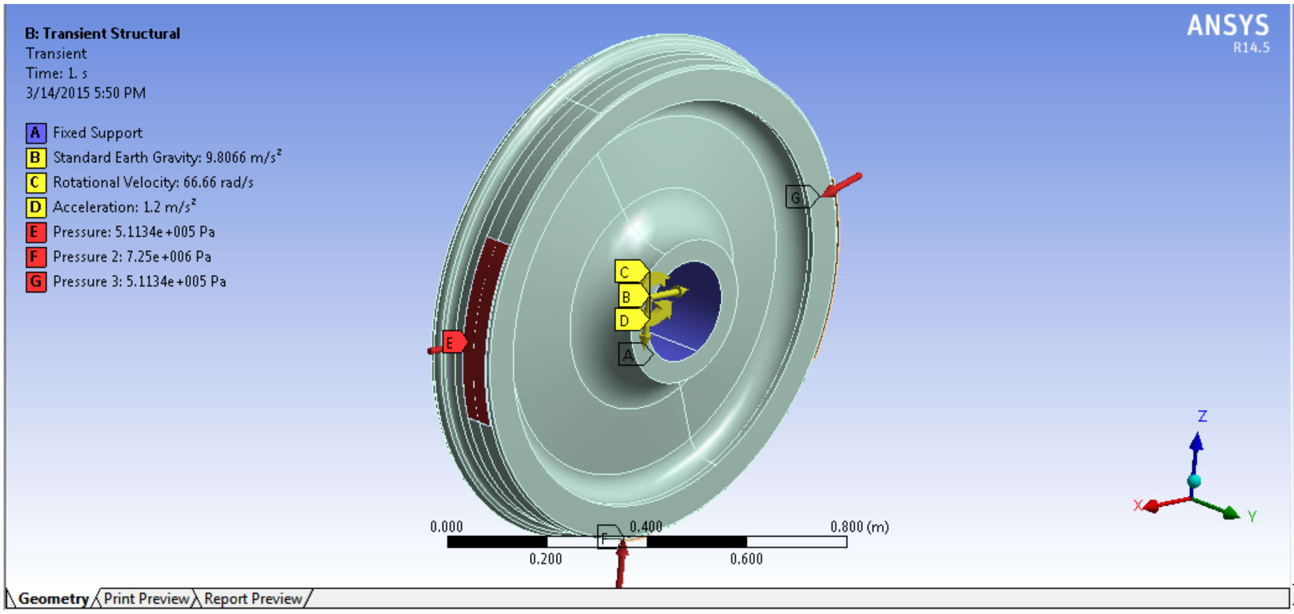

Figure 5-2 Structural loads applied on the wheels

5.1. Results of the normal braking action

First the results from the normal braking action were computed In order that we can compare and contrast the results with those from the malfunctioning of the braking system. The temperature gradient, the Von-misses stress distribution and total deformation resulting from the normal braking action are presented below.

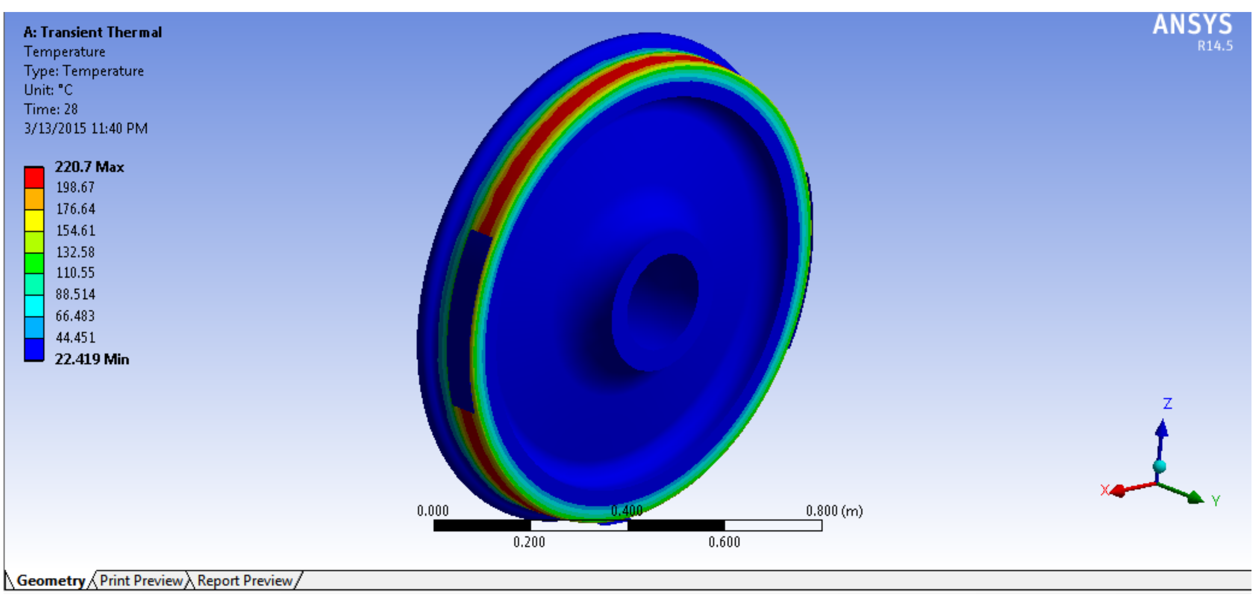

Figure 5-3 Temperature distribution from normal braking action

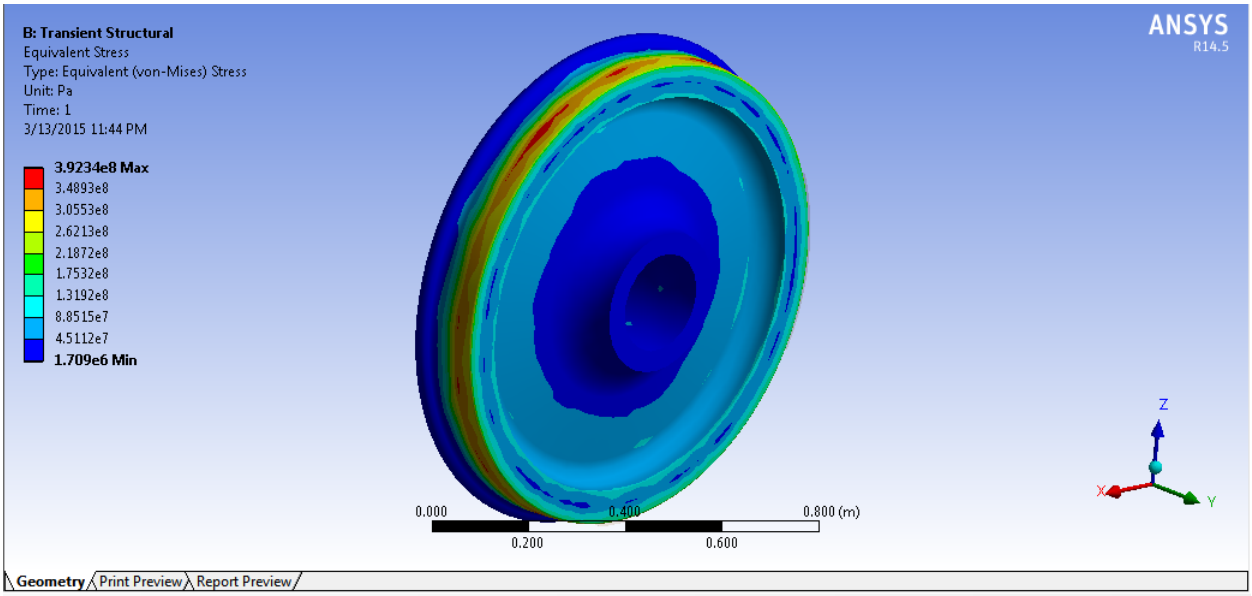

Figure 5-4 Von-misses stress distribution resulting from normal braking action

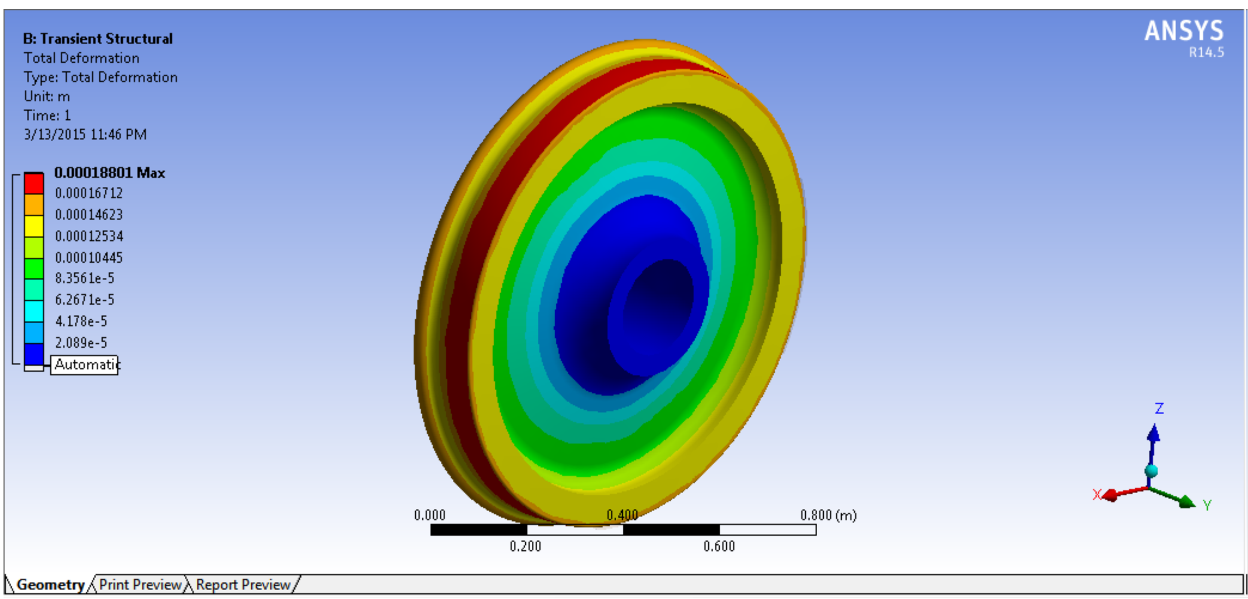

Figure 5-5 Total deformation resulting from normal braking action

TABLE 5-1 Summary of ANSYS results for normal braking action

Maximum Temperature[^{O}C]	Maximum Von-misses stress[Pa]	Maximum Total deformation[m]
220.7	3.923e8	0.00018801

5.2. The effect of friction fades

The analysis was performed for the cases when eight brake blocks and four brake blocks enter friction fade in the previous sever braking action and the results are summarized in the following table and are also presented by the diagrams from ANSYS 14.5 as follows.

TABLE 5-2 Summary of ANSYS results for friction fade

No of brake blocks faded	Maximum Temperatur [ºC]	Maximum Von-misses stress[Pa]	Maximum Total deformation[m]
8	313.29	5.835e8	0.00027023
4	256.63	4.665e8	0.00021712

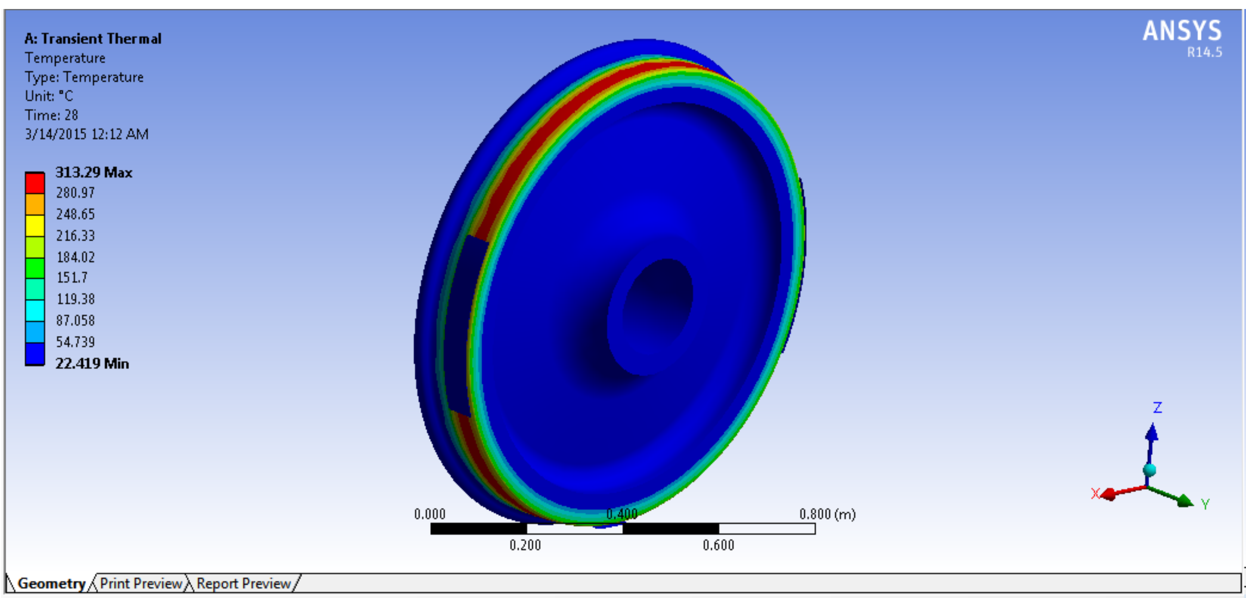

Figure 5-6 Temperature gradient after 8 brake shoes entered friction fade

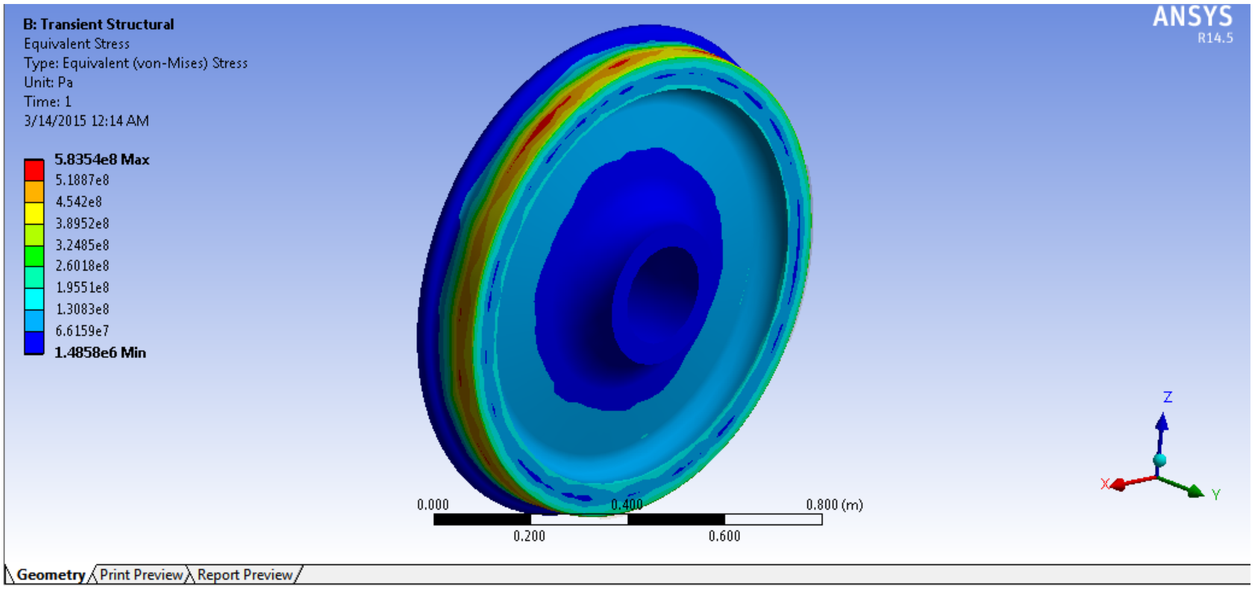

Figure 5-7 Stress distribution after 8 brake blocks entered friction fade

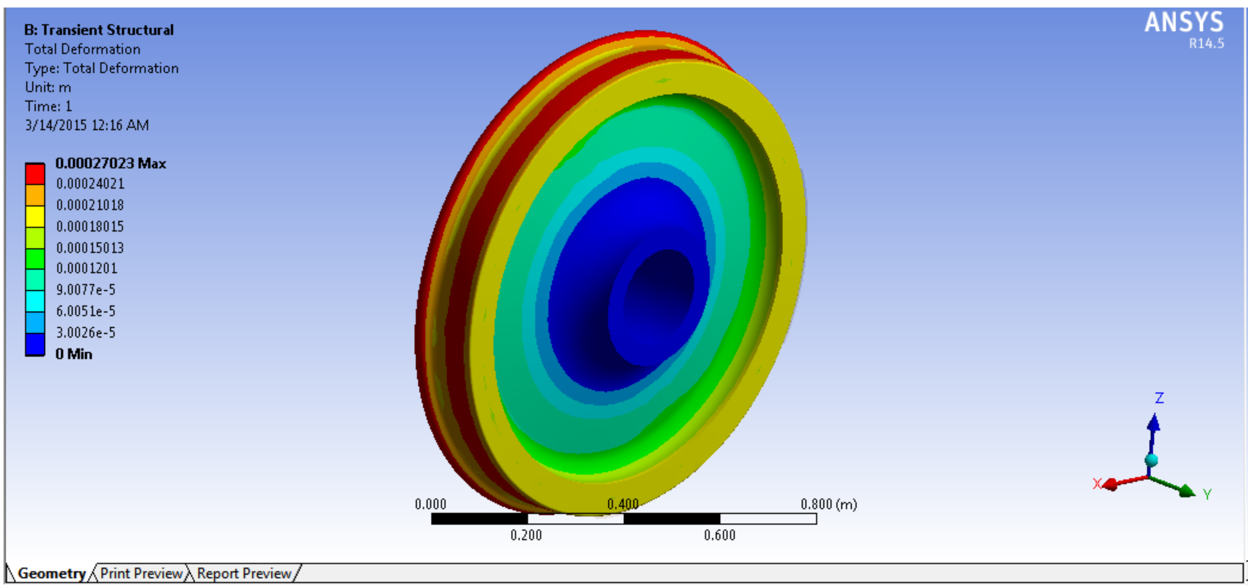

Figure 5-8 Total deformation after 8 brake blocks entered friction fade

(See Appendix A for Figures when 4 block brakes enter friction fade)

Referring to the summary made in table 5.3, the maximum temperature of the wheel while braking with 8 brake blocks friction faded is 313^{o}C and the corresponding maximum von-misses stress is 5.835e8 Pa. This clearly shows that the wheels are getting thermally overburdened as a result of the brake blocks on the four wheels failing to provide friction force of their share. When four brake blocks entered friction fade in the previous braking action, there is still a huge thermal burden on the remaining wheels. Though the maximum temperature of the wheel braking with four blocks faded is relatively small (256.63^{O}C), the maximum von misses stress obtained from the coupled analysis (4.665e8pa) is much greater than the ultimate tensile strength of the wheel material(4e8Pa).

5.3. The effect of brake pipe overcharging

The analysis is carried out for the cases of 10% and 15% brake pipe overcharging and the results following the unintentional braking action are summarized in the following table and the results from 15% brake pipe overcharging are presented by the succeeding figures from ANSYS 14.5.

TABLE 5-4 Summary of ANSYS results for overcharged brake pipe

Extent of overcharge	Maximum Temperature [^{o}C]	Maximum Von-misses stress[Pa]	Maximum Total deformation[m]
10%	507.11	9.8381e8	0.00048606
15%	602.27	1.1803e9	0.00059312

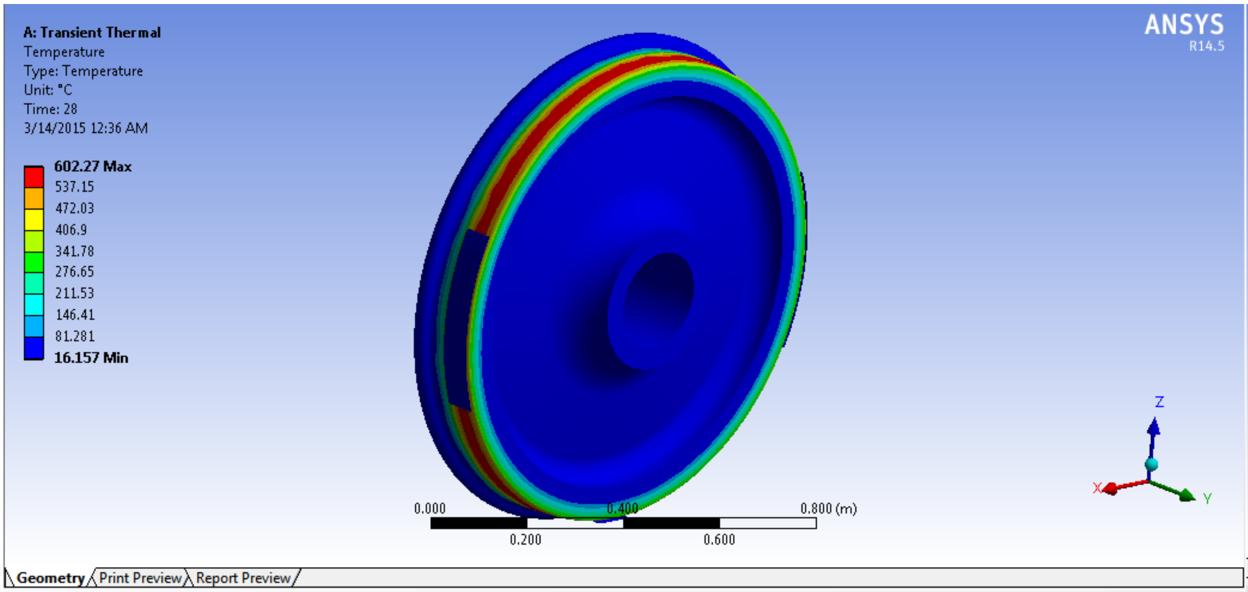

Figure 5-9 Temperature gradient resulting from 15% brake pipe overcharge

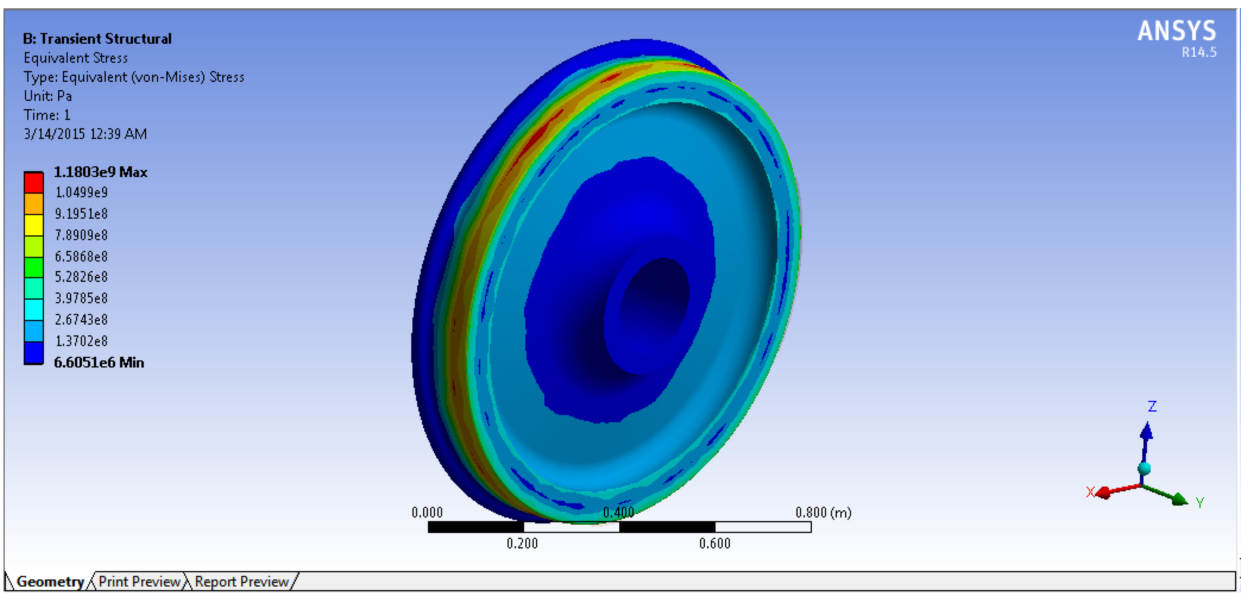

Figure 5-10 Von-misses stress distribution resulting from 15% brake pipe overcharge

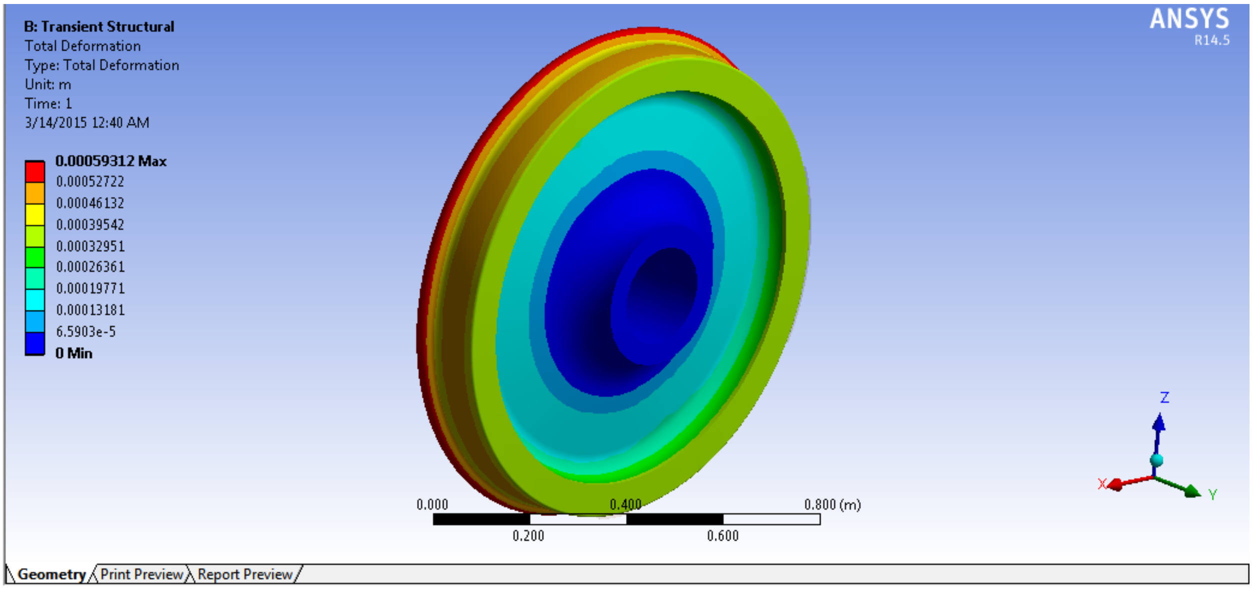

Figure 5-11Total deformation resulting from 15% brake pipe overcharge

(See Appendix B for Figures for 10% brake pipe overcharge)

As we can see from the summary made in table 5.4 the heat induced to the wheel as a result of unintentional braking action following 10% brake pipe overcharge has raised the temperature of the wheel to 507.11°C and when there is 15% brake pipe overcharge, the temperature rises to 602.27°C. As compared to the 220.7°C maximum temperature exhibited during normal braking action, overcharged brake pipe contributes huge amount to the overheating of the wheel. In addition, the maximum stresses induced to the wheel as a result of unintentional braking action following both 10 and 15% brake pipe overcharge are 9.8381e8 and 1.1803e9 respectively, which are far beyond the ultimate tensile stress of the wheel material.

5.4. The effect of excessive brake pipe leakage

The possible cases where 5%, 10% and 15% brake pipe pressure reductions into the valid range of brake pipe reduction were simulated to see the effect of the resulting unintentional braking action following excessive leakage from the brake pipe. The results of the analysis are summarized in the following table and the results for the analysis of pressure reduction 10%into the valid range is shown by the succeeding figures from ANSYS 14.5.

TABLE 5-5 Summary of results for excessive brake pipe leakage

Extent of leakage	Maximum Temperature[°C]	Maximum Von-misses stress[Pa]	Maximum Total deformation[m]
5%	382.9	7.273e8	0.00034733
10%	506.89	9.8336e8	0.00048582
15%	622.45	1.222e9	0.00061584

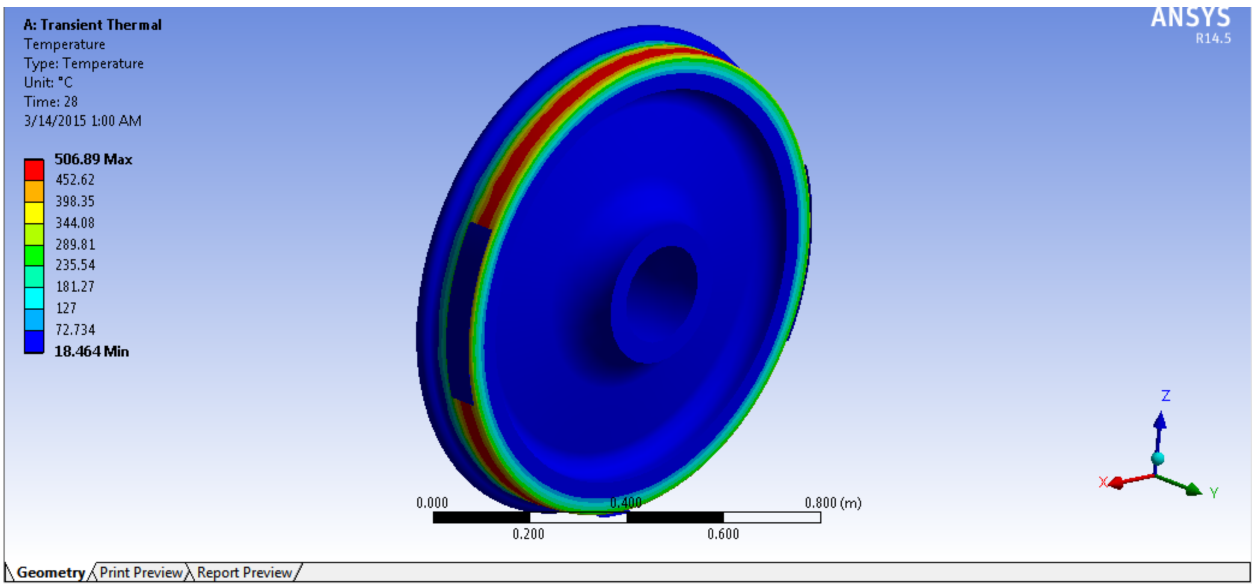

Figure 5-12 Temperature gradient resulting from 10% excessive brake pipe leakage

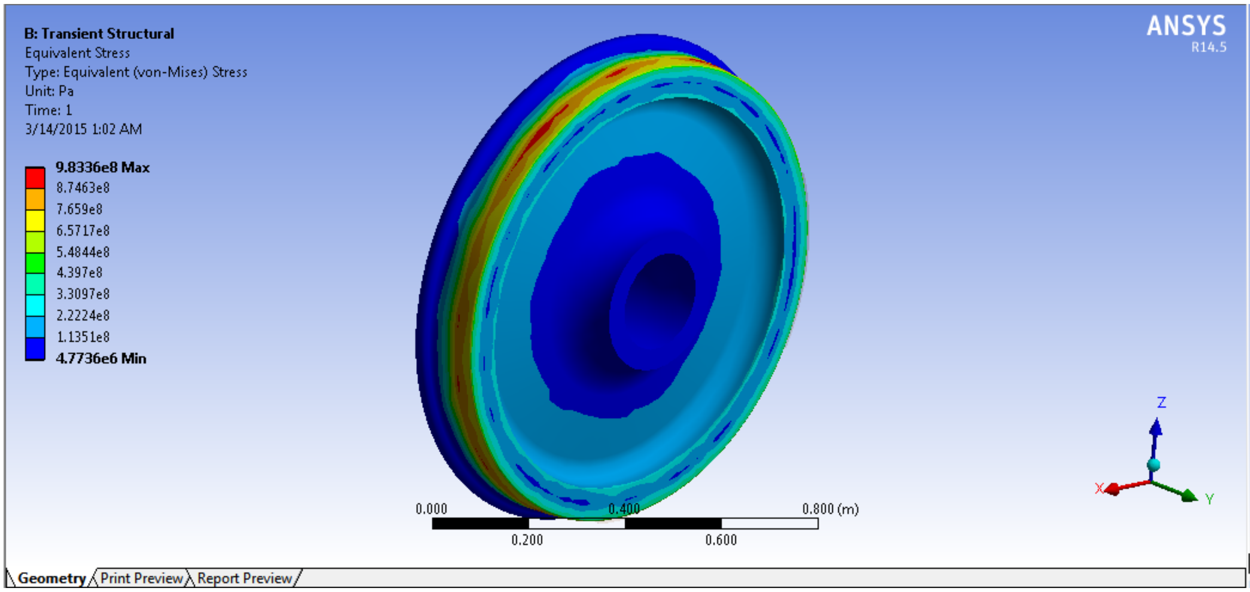

Figure 5-13 Von-misses stress distribution resulting from 10% brake pipe leakage

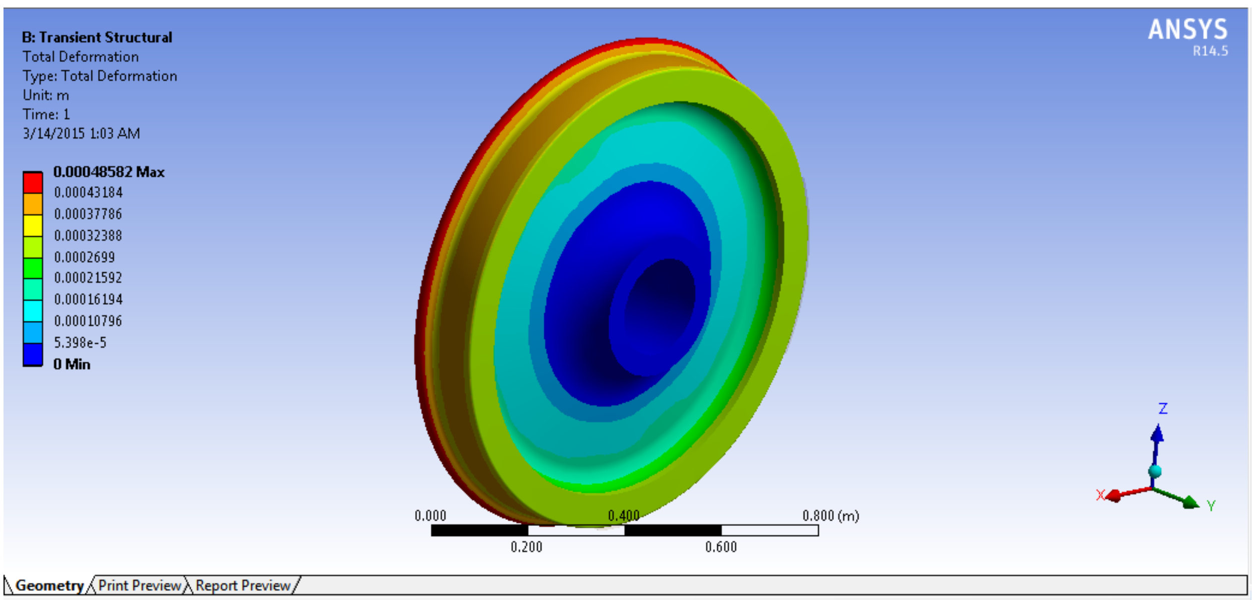

Figure 5-14 Total deformation resulting from 10% brake pipe leakage

(See Appendix C for Figures for 5 and 15% excessive brake pipe leakage)

As a result of 5% brake pipe pressure reduction caused by excessive leakage, the unintentional braking induced 382.9^{o}C temperature to the wheel and a maximum von misses stress of 7.273e8Pa. Though this is the minimum assumed brake pipe leakage, the stress developed at the tread is even way beyond the ultimate strength of the wheel material. As the leakage increases to

10 and 15%, not only it developed an even bigger stress at the tread but it also induced a very large amount of heat to the wheel and raised its temperature to 622.45^{O}C.

Chapter Six

Conclusion, Recommendation and Future work

6.1. Conclusion

This paper deals with the freight train braking system malfunctions exclusively and their independent effects are analyzed to show the extent of their effect on overhearing of the freight train wheel.

When the locomotive is braking with 8 blocks friction faded, the maximum temperature is 313^{O}C. however the resulting Von-misses stress obtained from the coupled analysis is 5.835e8Pa, which is much greater than the ultimate tensile strength of the wheel material. Hence, this shows that friction fade has significant effect on the thermo-mechanical properties of the freight train wheel.

The maximum temperature of the wheel corresponding to 10% and 15% brake pipe overcharge is 507.11^{O}C and 602.27^{O}C respectively. The corresponding von-misses stresses are 9.8381e8Pa and 1.1803e9Pa. There is huge difference when we compare it to the values resulting from normal braking (220.7^{O}C maximum temperature and 3.123e8Pa Equivalent stress). Therefore, brake pipe overcharge has significant contribution to both overheating and mechanical failure due to stress conditions of the freight train wheel.

Although the maximum temperature corresponding to 5% brake pipe leakage is relatively small (382^{O}C), the corresponding von-misses stress (7.273e8Pa) is much higher than the ultimate tensile strength of the wheel material. The maximum temperatures corresponding to the 10% and 15% excessive brake pipe leakages (506.89^{O}C and 622.45^{O}C respectively) are very big compared to the 220.7^{O}C of normal braking. The maximum equivalent stresses corresponding to 10% and 15% are 9.833e8 and 1.222e9 respectively and are both much higher than the ultimate tensile strength of the wheel material. Consequently, it is evident that unintentional braking following excessive leakage from the brake pipe also has very significant contribution to the overheating and overstress conditions of the wheel

To put it in a nut shell, the heat and stress generated during braking when some of the brake blocks had already entered friction fade is very high for the wheels to resist. Moreover, overcharged brake pipe and excessive brake pipe leakage contribute a great deal to the overheating of the freight train wheel. Since the thermal load they incur to the wheel is very big unintentional braking actions following malfunctions cause the wheels to deteriorate fast.

6.2. Recommendations

Continuous assessment on the braking system and the wheels is indispensable to mitigate the problems arising from such malfunctions. Extensive inspection of the brake blocks friction materials as to whether they had entered friction fade will minimize the chance of overheating of the freight train wheel on the next braking action.

Moreover, drivers/operators should always be given proper trainings that enable them to have a very good judgment capability while performing cycle braking downgrades so that the phenomenon of brake blocks entering friction fade and the resulting overheating of wheels are minimized.

The use of bougie mounted braking systems in freight locomotive will also improve the performance of the braking system though there had been wearing of brake blocks in previous sever braking action. The provision of double slack adjusters in BMBS will automatically compensate for the wear and keep the brake blocks working. Hence, the braking force increment on other brake blocks in the next braking will not be so big and this minimizes the chance of overheating of the wheels [28].

Inspecting the pressure states of the bake pipes of cars and locomotives prior to coupling and taking care while charging the brake pipe so as not to leave the automatic brake valve at its release state for too long will prevent the brake pipes from being overcharged.

Moreover, the feed valve for cars to be placed in the rear portion of the freight train should be adjusted at least 69KPa lower than the standard brake pipe pressure so that the chance of sticking brakes and corresponding overheating is minimized [29].

When the train air brakes are used to stop a train, increasing brake pipe reduction to at least 15 Psi after stopping and not releasing until at least 20 seconds after exhaust stops, will insure uniform pressure distribution and avoid overcharging while recharging the brake pipe and the resulting overheating of the wheels is minimized. Whereas, if a running release of train brakes is to be made, if operating conditions permit, increase the brake pipe reduction to at least 69KPa (10 psi) and allow brake pipe exhaust to stop for at least 20 seconds before releasing[29].

In addition, inspecting the whole length of the brake pipe regularly for excessive leakage and taking corrective measures if it exceeds 35KPa (Psi) per minute, as most railway standards state, will prevent the overheating of the freight train wheels [19].

6.3. Future Work

- It is very important to set up workshop experiment by building a single car model and collect first hand temperature data. The model should be made suitable to simulate the malfunctions and collect temperature data resulting from the friction fade, brake pipe overcharge and excessive brake pipe leakage. The results obtained could be compared to those obtained at the end of this thesis.

- Analysis of the micro-structural changes incurred to the wheel tread due to the malfunctioning of the braking system is also another vital investigation that reveals the extent of damage caused by the overheating of the wheel.

- In addition, efforts to observe weather these malfunctions and the resulting overheating have finally caused reversal of the useful compressive residual stress on the rim to tensile, either by destructive or non-destructive methods will add some edges to this thesis.

Appendix A

Results from ANSYS 14.5 when 4 brake blocks entered friction fade

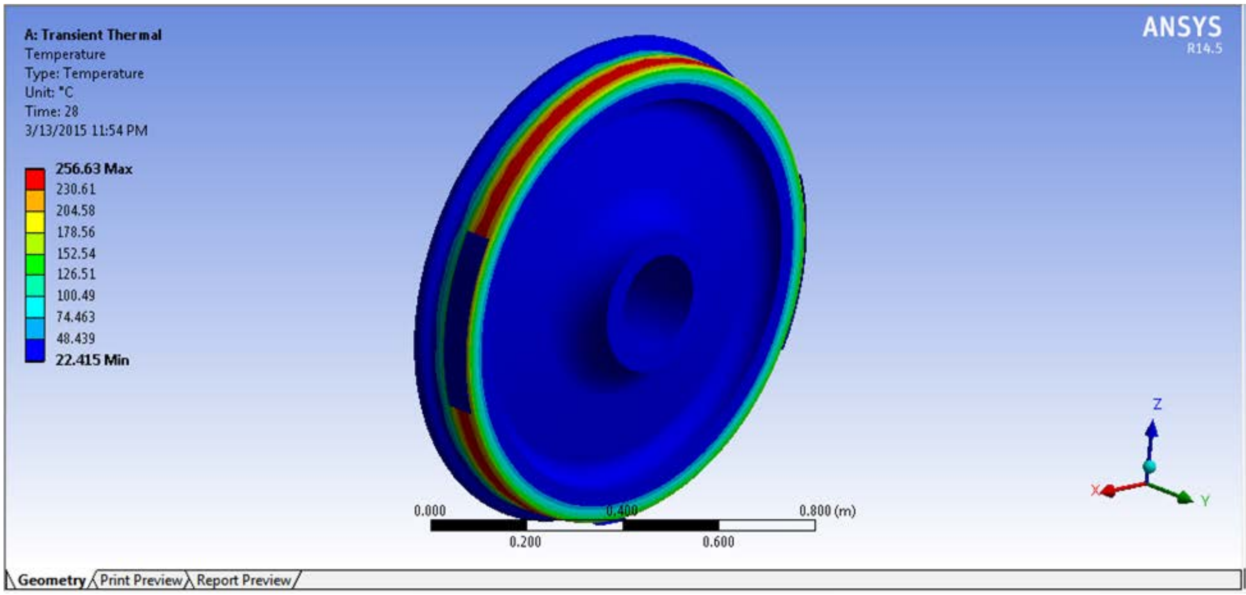

Figure A-1 Temperature gradient after 4 brake blocks entered friction fade

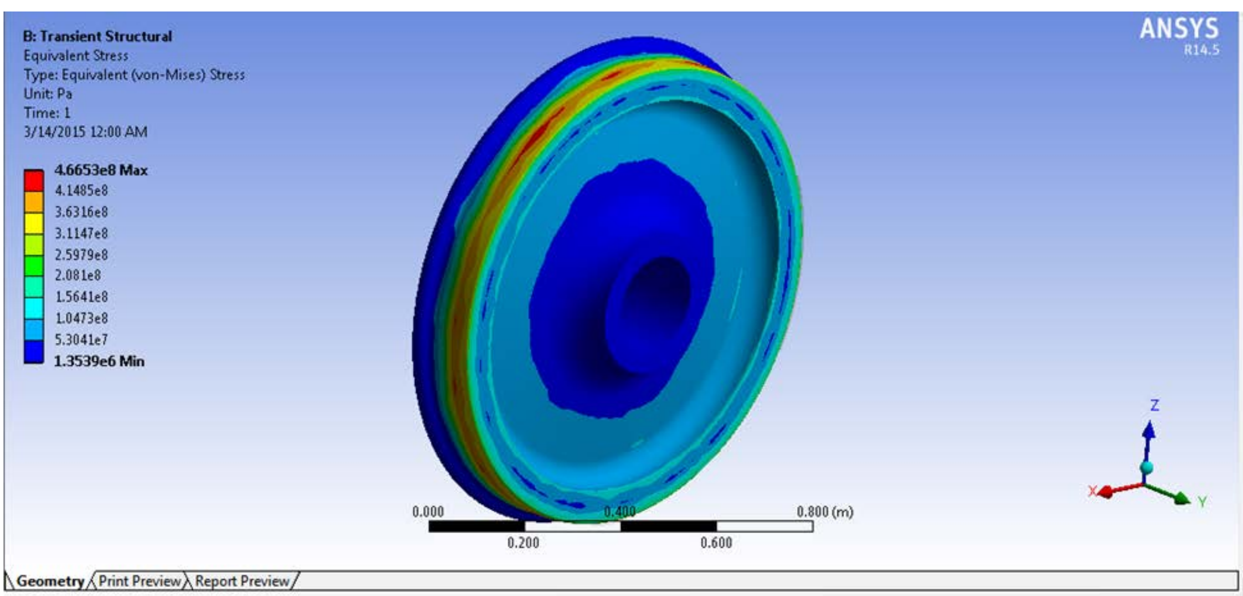

Figure A-2 Von-misses stress after 4 brake blocks entered friction fade

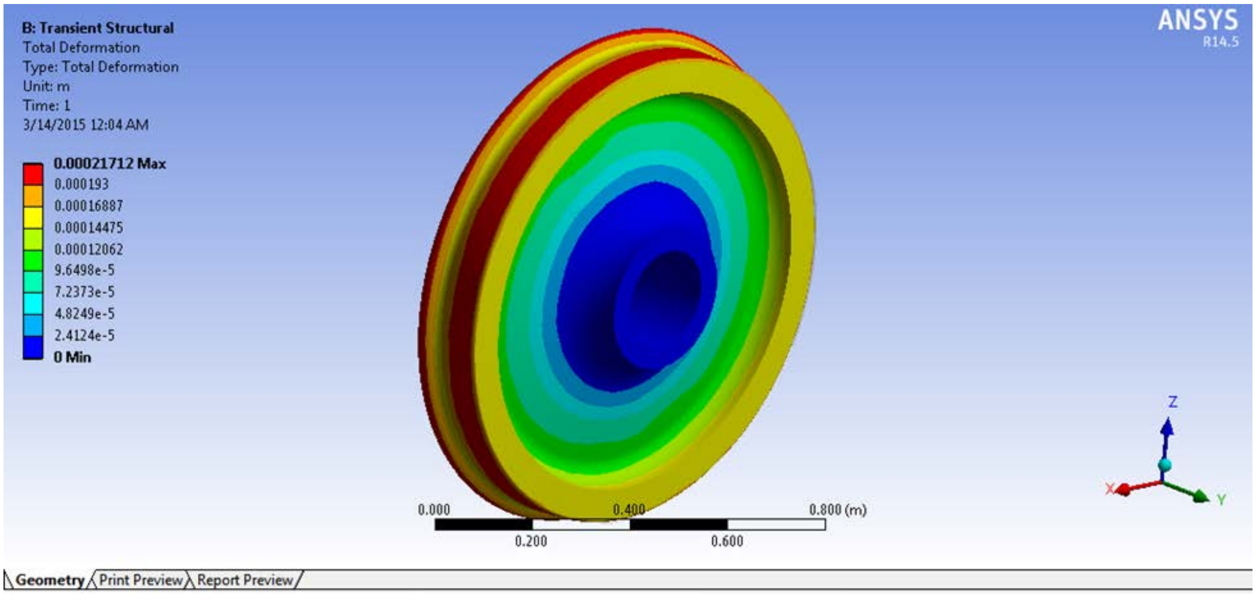

Figure A-3 Total deformation after 4 brake blocks entered friction fade

Appendix B

Results from ANSYS 14.5 for 10% brake pipe overcharge

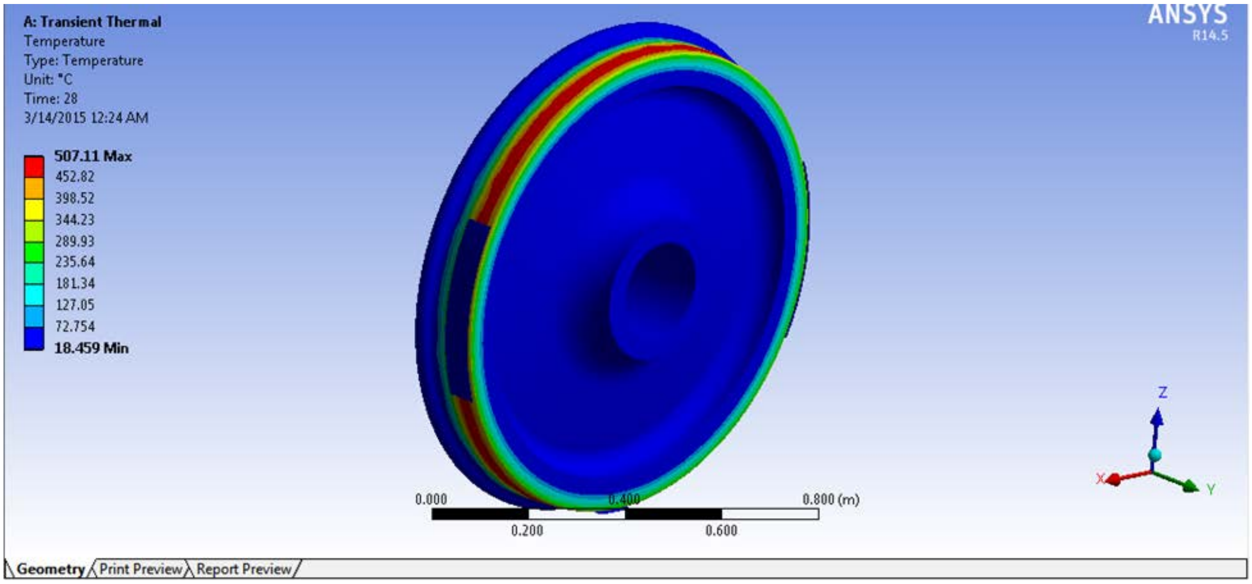

Figure B-1 Temperature gradient resulting from 10% brake pipe overcharge

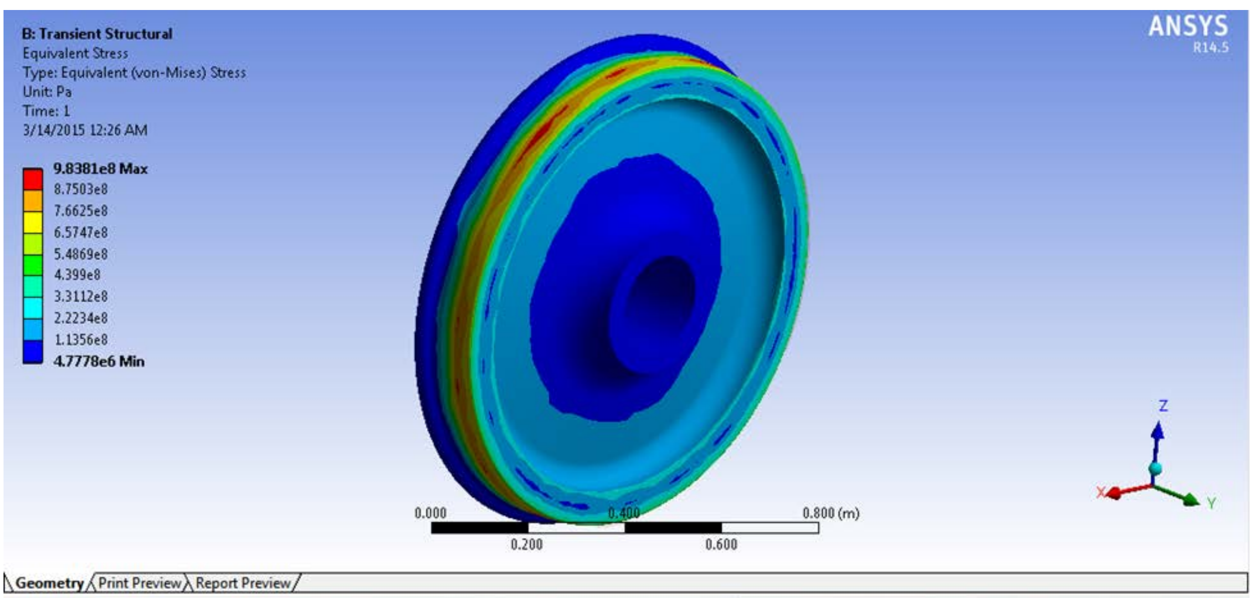

Figure B-2 Von-misses stress resulting from 10% brake pipe overcharge

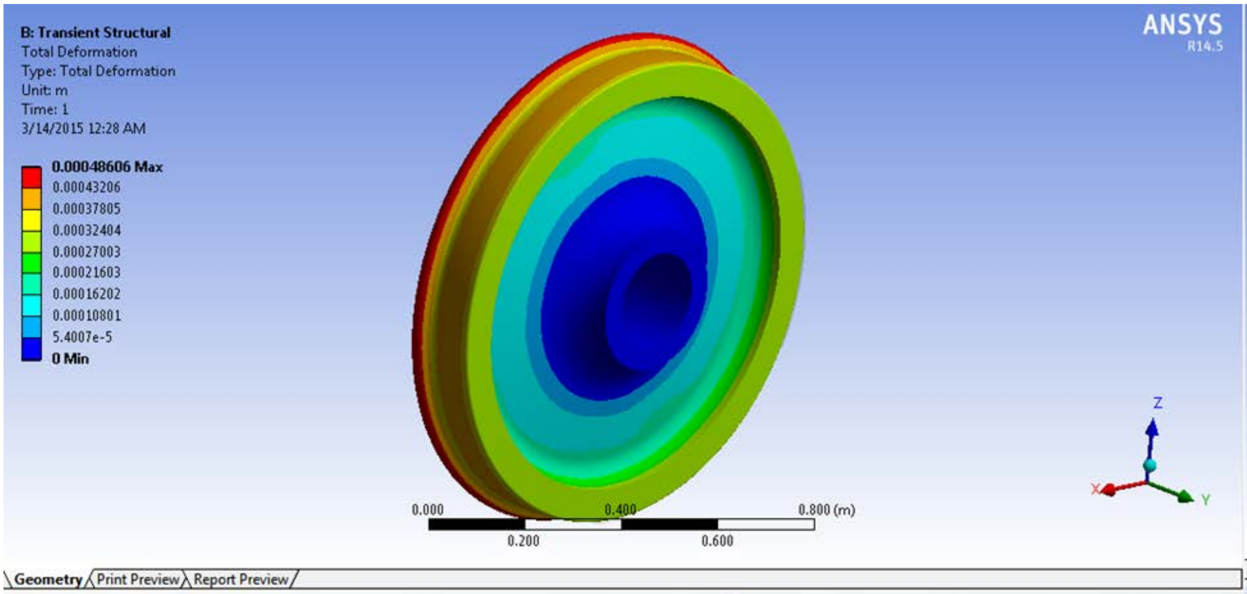

Figure B-3 Total deformation resulting from 10% brake pipe overcharge

Appendix C

Results from ANSYS 14.5 for 5 and 15% brake pipe leakage

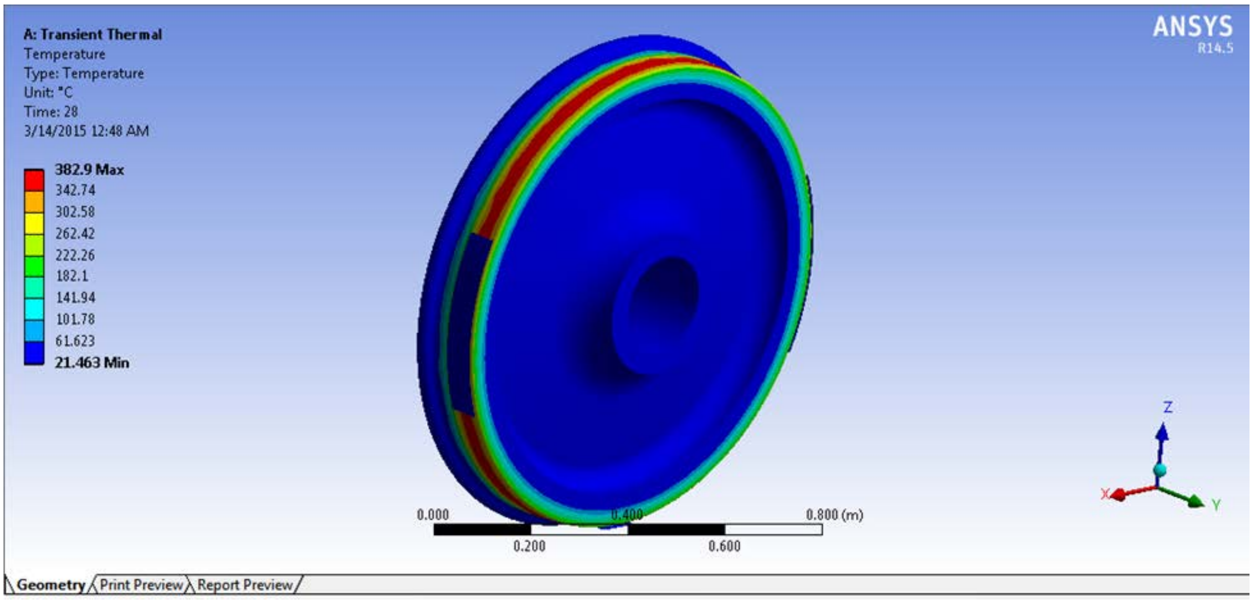

Figure C-1 Temperature gradient resulting from 5% brake pipe leakage

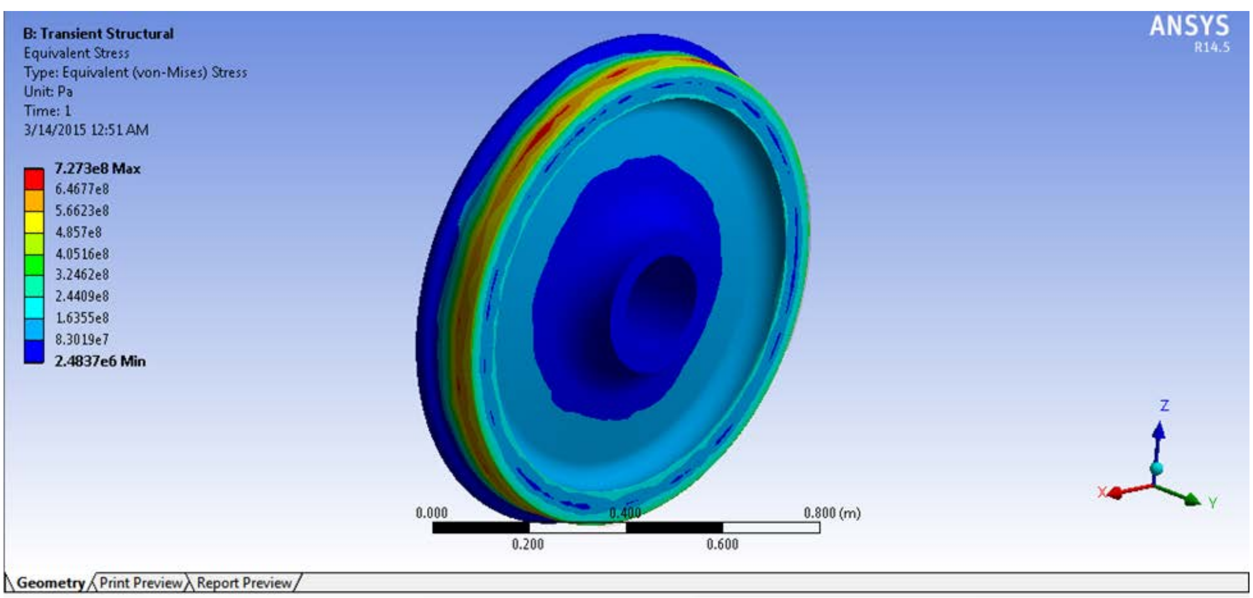

Figure C-2 Von-misses stress distribution resulting from 5% brake pipe leakage

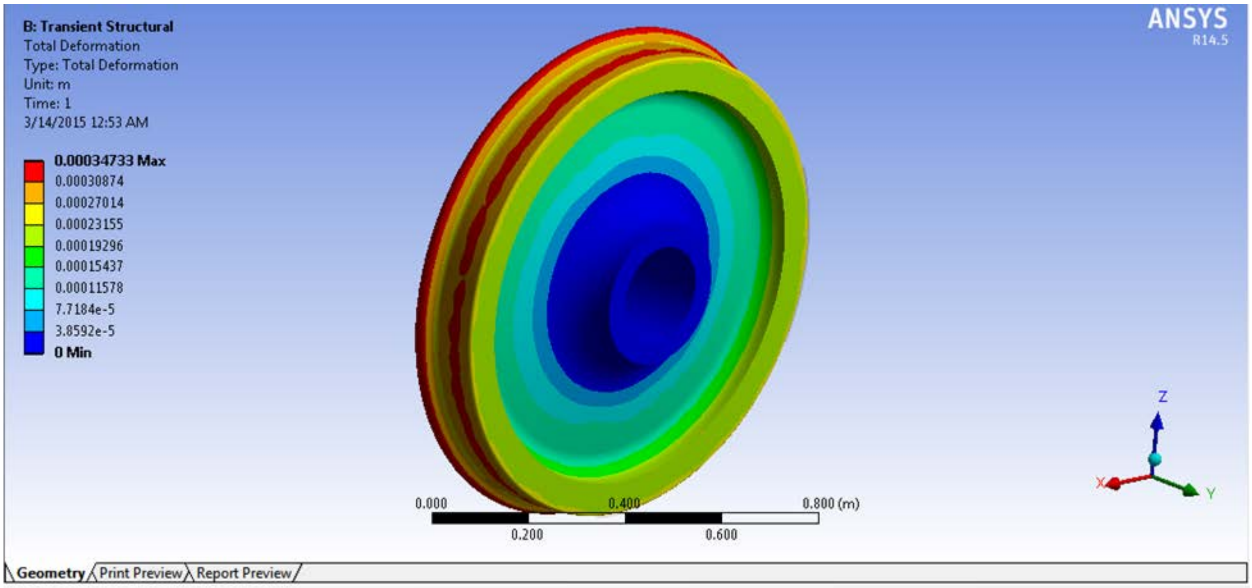

Figure C-3 Total deformation resulting from 5% brake pipe leakage

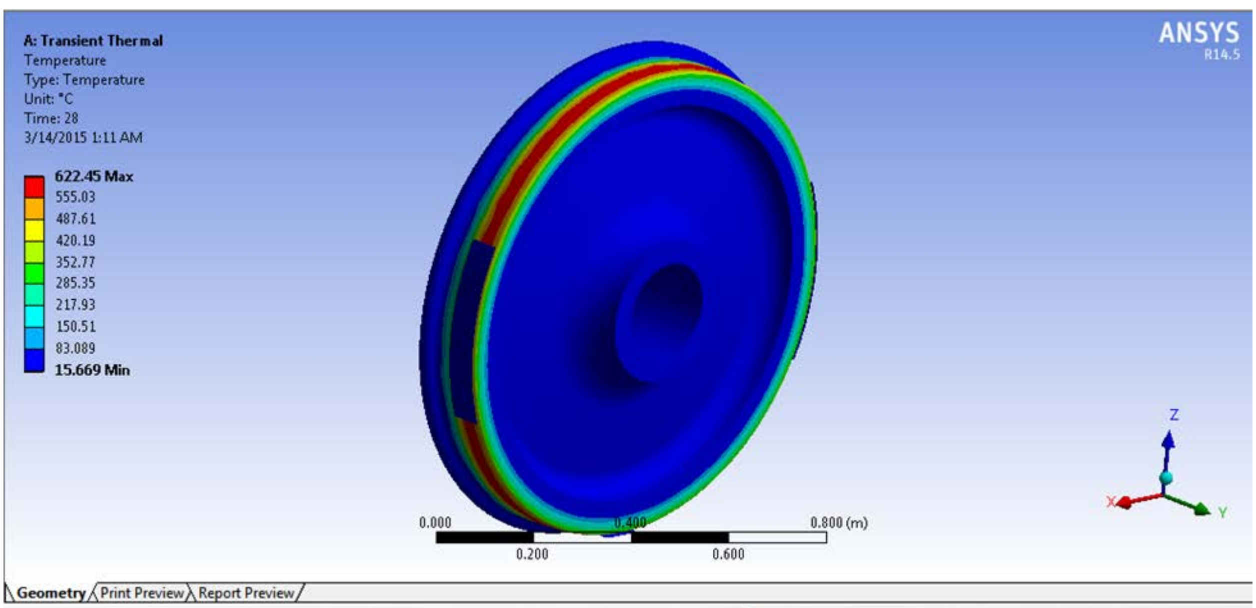

Figure C-4 Temperature gradient resulting from 15% brake pipe leakage

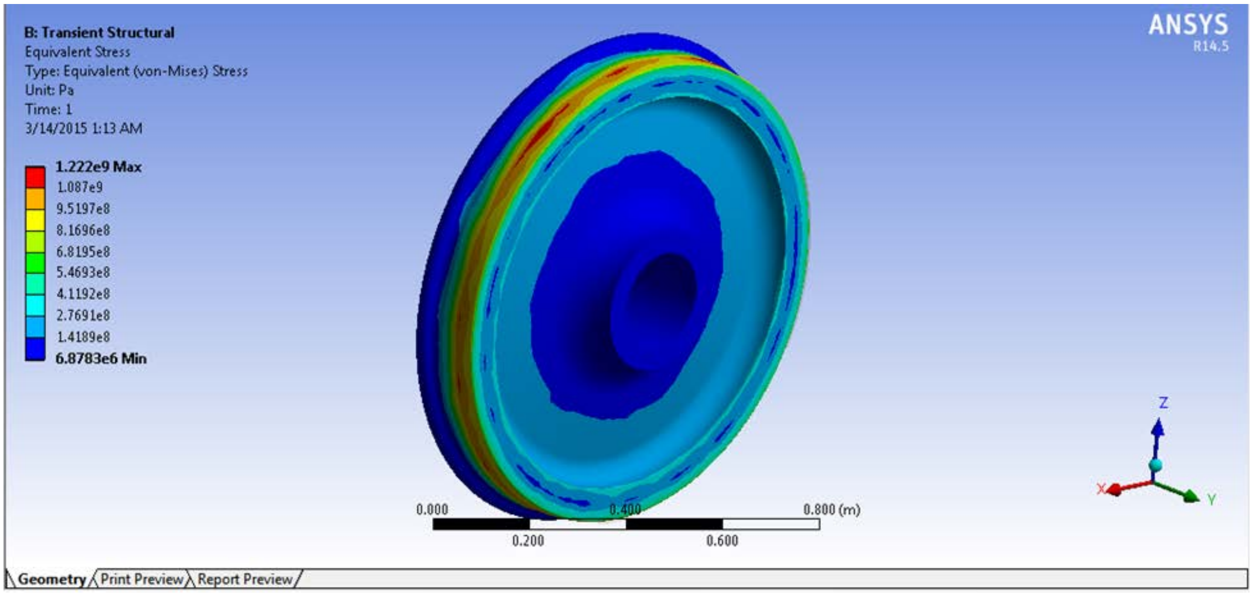

Figure C-5 Von-misses stress distribution resulting from 15% brake pipe leakage

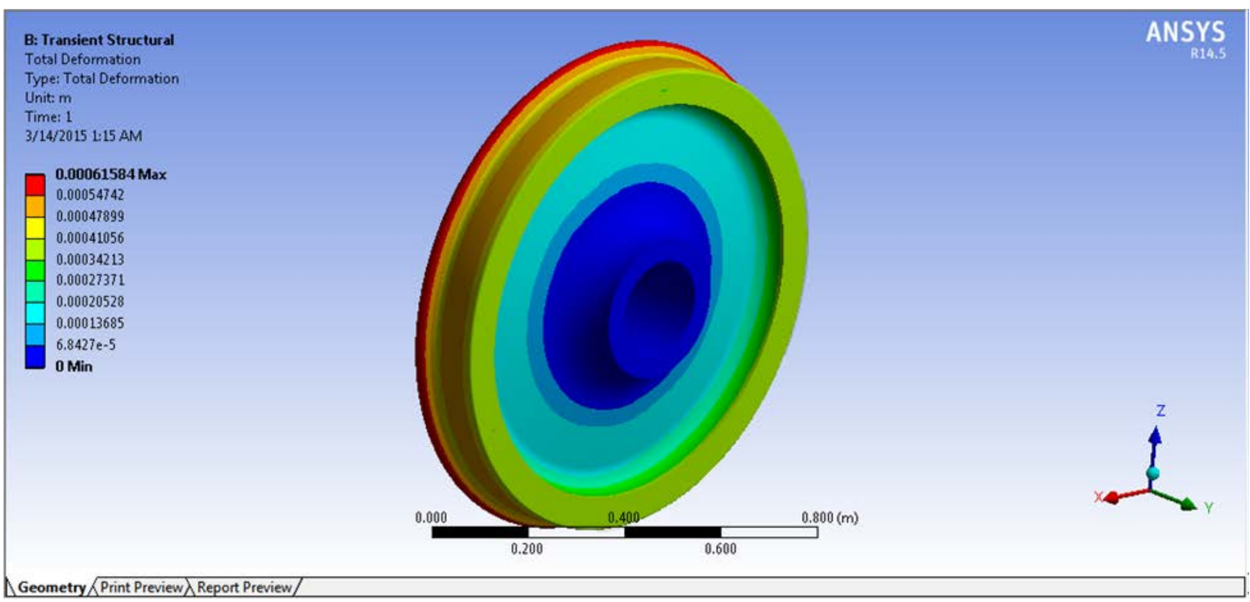

Figure C-6 Total deformation resulting from 15% brake pipe leakage

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