ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

THERMAL AND STRESS ANALYSIS OF AALRT WHEEL WHEN BRAKING WITH BLOCK BRAKE

By

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A thesis

Submitted to the school of Graduate studies Addis Ababa University in partial fulfillment of the requirements for the degree of Masters of Science in Rolling Stock Engineering

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December , 2014

DECLARATION

I, the undersigned, declare that this thesis work is my original work, has not been presented for a degree in this or any other universities, and all sources of materials used for the thesis work have been fully acknowledged.

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This thesis has been submitted for examination with my approval as a university advisors.

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ABSTRACT

The study of the thermal and stress condition between wheel and tread block of light rail transit (LRT) trains is so important since it has a direct effect on the wheel material, so that the temperature rise between wheel and tread block with speed and time during stop braking at the stations and also the stress is imposed on the wheel tread due this temperature rise and supporting forces and centrifugal force of the wheel itself assuming the axle to be its axis of rotation. The temperature rise and the loads are formulated analytically. The wheel model is performed using ANSYS soft ware on its sketcher work bench and transient thermal and transient structural analysis is used to generate the output of the research and the output gives appropriate value for two different angle of contact different value of temperature rise, deformation and stress induced on the wheel and their values are 1001.9°C for the contact angle 30^o and 662.8℃ for 45^o for the temperature and the maximum and the minimum resulting deformation are 0.0025887m and 0m respectively for $\theta=30^o$ and for $\theta=45^o$ it is 0.0017109m and 0m respectively. Finally this result is used to recommend the wheel condition for the AALRT case for safe operation for the selected material for wheel and LRT operating condition.

Key word AALRT, heat flux, temperature rise, ANSYS

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R – Wheel radius

P(R) - heat power generated per unit contact area at the radius R

F – Brake force

μ_s - Coefficient of friction between wheel and tread block

A – Contact area between wheel and tread block

P_s – brake pressure cylinder

l - Circumferential length of the block

b- Width of the block

θ – Tread contact angle

F_A - Brake force per contact area

V = train speed

$\dot{Q}$- Rate heat transfer

h - Convection heat transfer coefficient

T_s - Surface temperature

T_∞ - Ambient temperature

ε - Material's emissivity

T_{ref} - Temperature of the surrounding air and Addis Ababa

Re -Reynolds number

Pr - Prandtl number

k_a - Thermal conductivity

ρ_{air} - Air density

μ_{va} - Dynamic viscosity

C_{pa} - Specific heat capacity

ΔE - Change in energy

U – Over all heat transfer coefficient

m – Mass of train

I – Mass moment of inertia

ω_o - Rotational speed

g - Gravitational acceleration

Δh - Change in height

δ – Gradient angle

t_b - Brake time

a - Acceleration/deceleration

R_g - Grade resistance (N)

r_g - Coefficient of grade resistance

R_c- Curve resistance (N)

r_c- Coefficient of curve resistance

r_e - Equivalent Grade

W_{res} - Work by the gradient on the train

R_e - Equivalent resistance in N

S - Braking distance

F_c - Centrifugal force of the wheel

m_w -Mass of the wheel

m_b - Mass of car body

$F_{H,f}$ - Horizontal reaction force on the front center plate

$F_{H,r}$-horizontal reaction force on the rear center plate

R_f - Reaction force on front center plate

R_r - Reaction force on rear center plate

L_h - Length from the center plate to the rail

L_g - Length from the center of gravity of the truck (bogie to the rail)

m_{tr} -weight of the truck

R_{tra} - Weight transfer from the body to the front center plate

L - Length of the wheel base

Fcg - Inertia force on the bogie

$R_{w,f}$ - Reaction force on the front wheel

$R_{w,r}$ - Reaction force on the rear wheel

CHAPTER ONE

INTRODUCTION

1.1 Historicaal Background

In the earliest days of railways, braking technology was primitive. The first trains had brakes operative on the locomotive tender and on vehicles in the train, where "porters" or, in the United States brakemen, travelling for the purpose on those vehicles operated the brakes. Some railways fitted a special deep-noted brake whistle to locomotives to indicate to the porters the necessity to apply the brakes. All the brakes at this stage of development were applied by operation of a screw and linkage to brake blocks applied to wheel treads, and these brakes could be used when vehicles were parked. In the earliest times, the porters travelled in crude shelters outside the vehicles, but "assistant guards" who travelled inside passenger vehicles, and who had access to a brake wheel at their posts supplanted them.

The braking effort achievable was limited, and an early development was the application of a steam brake to locomotives, where boiler pressure could be applied to brake blocks on the locomotive wheels. It was also unreliable, as the application of brakes by guards depended upon them hearing and responding quickly to a whistle for brakes.

In the UK, the Abbots Ripton rail accident in January 1876 was aggravated by the long stopping distances of express trains without continuous brakes, which -it became clear- in adverse conditions could considerably exceed those assumed when positioning signals. This had become apparent from the trials on railway brakes carried out at Newark in the previous year, to assist a Royal Commission then considering railway accidents. In the words of a contemporary railway official, these showed that under normal conditions it required a distance of 730 to 1097 meters to bring a train to rest when travelling at 73 to 78Kmph, this being much below the ordinary travelling speed of the fastest express trains. Railway officials were not prepared for this result and the necessity for a great deal more brake power was at once admitted.

During the early 19th century various attempts were made to get away from the concept of vehicle brakes which had to be individually controlled and provide a train brake with one point of control. A scheme of 1840 had a chain which ran along the train to the guard's position at the rear where it was wound round a drum. To apply the brake the drum was lowered until it touched an axle, causing it to rotate and tighten the chain. Levers connected to the chain applied the brakes In addition, as railways developed during the mid 18th century, there were a number of accidents caused by trains becoming uncoupled (a breakaway) or just failing to stop. Sometimes, breakaways ran down a grade and collided with the following train or trains became parted and the second half ran into the front half after the crew had stopped it because they had noticed the uncoupling. The traditional form of wheel tread brake consisting of a block of friction material which could be cast iron, wood or a composition material hung from a lever and being pressed against the wheel tread by air pressure in the air brake or atmospheric pressure in the case of the vacuum brake.

Converting kinetic energy to heat is an old system that is still in use in railway brake systems. Applying wooden blocks to the wheel treads were the oldest design of friction brake systems to stop the initial steam locomotives. It was the driver‘s task to actuate a lever or operate a screw to make a contact between the wooden block and running wheel to apply the brake. This kind of manual system did not match the increasing demand of a suitable brake system, and a system with higher reliability and safety was needed. In the mid-1870s, new systems were introduced replacing the old ones. Compressed air brakes in the US and vacuum brakes in the UK were introduced simultaneously (Elliot, 2006). [1]

Different types of braking systems are used in railway vehicles. Tread (block) braking is the classic type of mechanical (pneumatic) braking system and is still in common use for freight trains. For passenger trains and metros, tread braking is nowadays often used in combination with electrodynamics and/or disc braking. In tread braking, the braking action is obtained by pressing one, two or four brake blocks (brake shoes) against the tread of a wheel. There are different kinds of brake block materials: cast iron, organic composite materials and sinter materials.

The tread braking action on the railway wheel causes frictional heating of both wheel and brake block. The heat partitioning, i.e. the parts of the total generated heat entering the wheel and the block, respectively, is an important aspect. Furthermore, thermo-elastic interaction between wheel and block causes an uneven distribution of heat on the wheel and block surfaces and so-called banding and hot spots will therefore develop. Hot spots are regularly spaced superheated areas often observed on the wheel tread during braking. The elevated temperatures of wheel and blocks are followed by heat transfer through convection and radiation to the surrounding. Heat will also be conducted from the wheel to the rail.

Figure 1 Bogie with shoe brake

The high temperatures due to frictional heat during braking are an important dimensioning factor for railway wheels. Friction, material properties and the subsequent thermal damage of the contacting bodies are also influenced by high temperatures.

The thermal impact in tread braking systems may induce wheel tread damage which is an important issue for wheel manufacturers and train operators. The damage influences the life cycle cost of railway wheels and can also be a safety issue. Repeated stop braking (i.e. the initial speed of the train is reduced substantially, or the train is brought to rest), e.g. in metro applications, may cause both a substantial temperature rise in the wheel rim and the creation of hot spots on the wheel tread. Moreover, excessive drag (downhill) braking (the potential increase in train speed is kept constant by braking with an approximately constant brake force for a relatively long time), may create a risk of a different wheel failure mode: global wheel fracture induced by residual stresses when the wheel cools down.

For a railway to operate efficiently and safely, its locomotives should be powerful enough to accelerate their trains rapidly to the maximum allowed line speed, and the braking systems must be able to bring a train reliably to a standstill at a station or signal, even on an adverse gradient. The frictional heat is generated when two bodies slide against each other with a relative speed and a contact pressure. This frictional heat is generated at sliding or rolling interface of wheel/brake shoe contact.

The thermal aspect around the contact region can be studied by examining the heat transfer between stationary break shoe and a moving body (relative to the heat source). The slip between wheel and braking system causes friction heating of both bodies. When a brake is working, the transformation of kinetic energy of moving masses and the potential energy if there grade into thermal energy takes place.

This energy is dissipated between two bodies and appreciably raises their temperature at the area of the sliding contact. Two aspects are of special importance in this analysis - the nature and distribution of heat partition into each body at the interface and the resultant temperature fields in the two bodies both at the interface and with respect to dept.

Brake shoe is best used on rolling stock materials. Wheel is required first by the driving forces and secondly by braking forces [2] for the second category of claim was necessary to set temperature limits and wear. At automatic brake shoes, force which manifest to the brake cylinder piston rod is amplified by the wheelhouse of brake components and sent to break blocks which press the wheel tread.

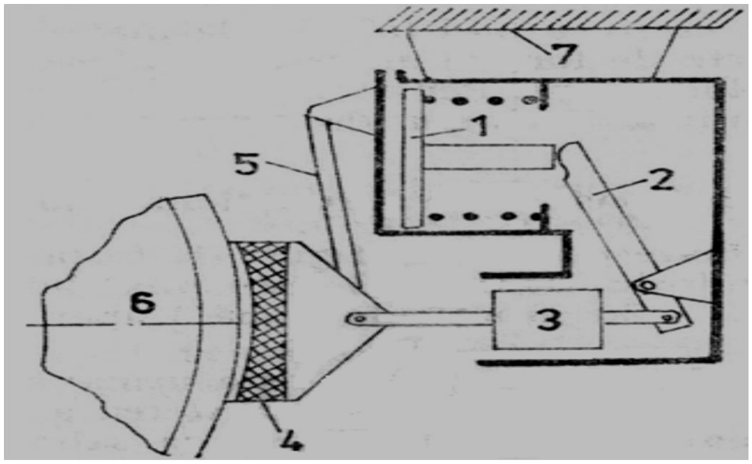

Figure 2 Braking shoes unit's scheme

1-piston; 2- lever amplifiers; 3- automatic regulator wheelhouse; 4- brake shoes; 5- padlock; 6- wheel; 7- bogie frame

1.2 Braking With Brake Shoes For Rail Vehicles

Rail vehicles are characterized by the movement on the railways through wheels auto guided by contact forces between wheels and rails. Braking facility holds the most important role in ensuring the movement of railway vehicles in safety conditions for the traffic.

Brake shoe is best used on rolling stock materials. Wheel is required first by the driving forces and secondly by braking forces. For the second category of claim was necessary to set temperature limits and wear. At automatic brake shoes, force which manifest to the brake cylinder piston rod is amplified by the wheelhouse of brake components and sent to break blocks which press the wheel tread.

During braking friction, the largest share of energy is dissipated as heat absorbed by friction members which is thus subjected to a thermo mechanical intense. In the field of railway braking for example, brakes disk are brought considerable energy to dissipate in a few tens of seconds, several mega joules for a single disc, or even tens of mega joules in the case of high speed.

The severity of such stresses leads to localization phenomena thermal interface, which manifest themselves in different forms, such as the hot bands migrants, or macroscopic hot spots, depending on the severity of braking, temperature reaching locally extreme values. The impact of these phenomena on durability of the braking is of course crucial, and studies show that thermo mechanical approaches alone are not sufficient to describe these phenomena, it is necessary to consider the phenomena of friction and wear at the interface.

1.3 Function Of Braking

Stopping a train requires work. This work equals the change in the train's kinetic energy plus the change in its potential energy (change in height due to the gradient of the track). The 'work' is the energy in decelerating the train over the stopping distance, i.e. the product of the train's mass, the train's acceleration rate (deceleration is negative acceleration) and the stopping distance.
The change in 'kinetic' energy relates to the change in the train's speed i.e. the difference of the speed at which deceleration began and the 'at stop' speed i.e. 0. The change in 'potential' energy relates to the change in height of the train's centre of mass due to the gradient of the track i.e. the difference in height at which deceleration began and the its height at the stopping point.
The braking process is in fact the matter of energy balance. The aim of braking system is to transform mechanical energy of moving vehicle into the some other form, which results by decreasing of the vehicle speed. The kinetic energy is transformed into the thermal energy, by using the dry friction effects and, after that, dissipated into the surroundings.
As is known, braking system is necessary for:

- stopping the train (rail vehicle) within the limits of braking space;
- partial reduction of speed;
- lowering maintenance train on the slopes;
- Train (railway vehicle) immobilization after stop.

The braking action generates heat by transformation of frictional energy. The high interface temperature and a severe thermal environment can affect the tribological characteristics of the contacting surfaces such as friction and wear. Moreover, contact pressure and interface temperature are mutually dependent, a fact which plays an important role in the process of thermoelastic deformations through localized surface effects. Therefore, the contact, the heat conduction and the thermal stress problems show a complex interconnection. In fact, for the heat conduction problem one needs the contact area and contact pressure distribution from the mechanical contact problem solution. For the thermal stress problem, one then uses the temperature distribution from the heat transfer solution and, again, the thermal stress results should be used for solving the total stress and deformation of the wheel.

There are several factors which influence frictional energy transformation at the friction interface of a brake such as the amount of energy generated and partitioned between the mating surfaces and also transferred to the surrounding, the thermophysical properties of the materials, and the distribution of frictional heat generation over the interface, and macroscopic surface effects.[6] Braking performance of a vehicle can be significantly affected by the temperature rise in the brake components. High temperatures during braking may cause brake fade, premature wear, and brake fluid vaporization, bearing failure, thermal cracks and thermally-excited vibration. Therefore, it is important to predict the temperature rise of a given brake system and assess its thermal performance in the early design stage. [7]

1.4 Tread Braking

The traditional form of wheel brake consisting of a block of friction material (which could be cast iron, wood or nowadays a composition material) hung from a lever and being pressed against the wheel tread by air pressure in the air brake or atmospheric pressure in the case of the vacuum brake. Tread (block) braking is still one of the most common braking systems on railway vehicles. Block braking is the ordinarily used system on, often in combination with disc brakes and electrodynamics brakes. The tread braking (or block braking) action is carried out by pressing the brake block(s) against the tread of a wheel. The vehicle can be equipped with four different standard block configurations. Air brake is the most common type of train brake. It uses compressed air to apply the brake block (or pad) to the wheel and to control the operation of the brake along the train. The compressed air is supplied by a motor driven compressor on the locomotive or train.

The brake control is actuated from a "driver's brake valve". This valve is used to feed air to the brake pipe or to allow air to escape from the brake pipe. A fall in brake pipe air pressure causes a brake application on each vehicle whilst a restoration of pressure causes the brake to release.

The total heat generated during braking is partitioned between the wheel and the block at their contact interface. The elevated temperatures of wheel and block will be accompanied by heat transfer through convection and radiation to the surrounding. In addition, heat will be conducted from wheel to rail.

Tread braking systems have both advantages and disadvantages. They are simpler and cheaper than systems with axle- or wheel-mounted disc brakes, and they allow much more flexibility regarding space requirements compared to the other braking systems. Since block braking roughens the wheel tread, it improves the wheel-rail interface friction conditions at traction and braking. Unfortunately, the roughness may also cause an increased level of rolling noise which is typical for trains with cast iron brake blocks. As part of the effort to reduce noise, brake blocks of organic friction materials (composite brake blocks) are increasingly being used. The thermal power distribution between brake block and wheel is less favorable for the wheel when using composite brake blocks. However, the service life of composite brake blocks is much longer than that of cast iron blocks [11] Figure shows tread braked railway wheel. The brake block is located on the left side of the railway wheel.

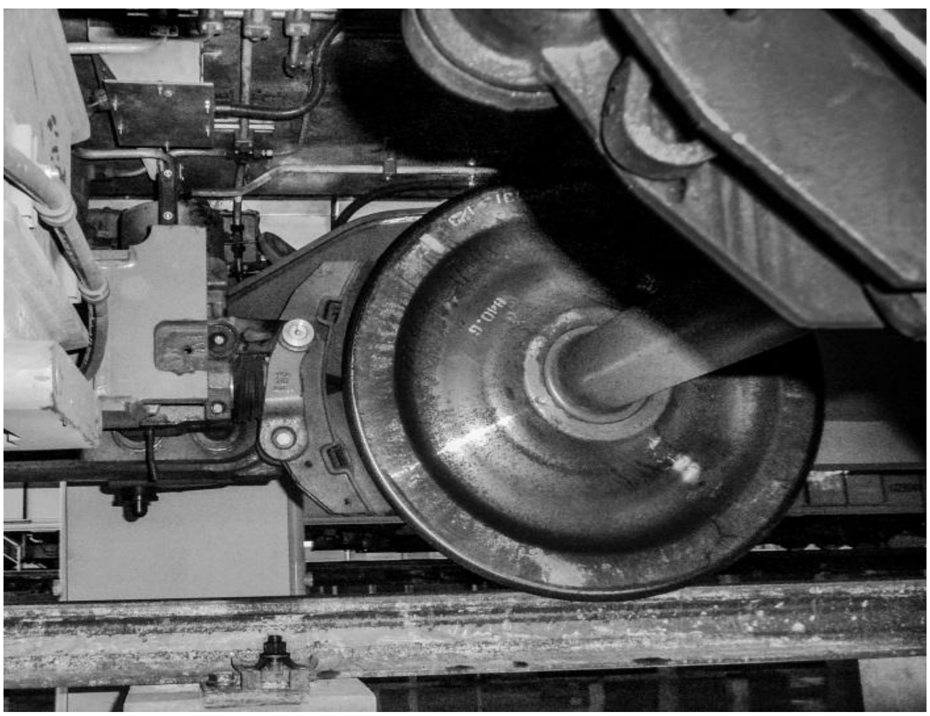

Figure 3 A view of bogie interior

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

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

1.5 Braking Phenomena

The braking action generates heat by transformation of frictional energy. The high interface temperature and a severe thermal environment can affect the tribological characteristics of the contacting surfaces such as friction and wear.[12] Moreover, contact pressure and interface temperature are mutually dependent, a fact which plays an important role in the process of thermo elastic deformations through localized surface effects. Therefore, the contact, the heat conduction and the thermal stress problems show a complex interconnection. In the study of the heat conduction problem one needs the contact area between wheel and thread block and the reaction forces on the truck due to the contact between wheel and rail form the weight transfer condition when braking. For the thermal stress problem, one then uses the temperature distribution from the heat transfer solution and, again, the thermal stress results should be used for solving the contact problem [13].

1.6 Track Line Of AALRT

E-W & N-S lines are 31.025km long, and the ground laying mode is mostly applied. Ground stations are designed on ground line section, including 27 ground stations along the line (13 ground stations on N-S line, and 14 ground stations on E-W line). Three elevated stations are set for the N-S line. One elevated station is set for the E-W line. And five elevated stations are shared by the E-W line and N-S line at their common rail section. [3] The east-west line phase I project starts from Ayat and ends at Torhailoch. The total length is 17.4km. There are 22 stations, among which 5 are elevated stations, 1 underground station and 16 ground stations.

The south-north line phase I project starts from Menelik II Square and ends at Kaliti. The total length is 16.97km. There are 22 stations, among which 9 are elevated stations (5 common stations at the common line), 2 underground station and 11 ground stations. [5]

On each of these train stations braking the train is mandatory to stop the train to load and unload the passenger. While braking at these stations the temperature of the wheel tread increases as the driver operates the driver brake valve and reaches the maximum value when the train stops. This temperature is due to the contact between the brake block and the tread surface of the wheel to stop the train at the station. The rise this temperature should be known in order to operate the vehicle at the limit of the range to operate the train safely and to save the life of the wheel.

1.7 Thesis Objective

The objective of this research is to determine the thermal and stress condition between wheel and tread block while the train is braking from high speed to stopping using the soft ware analysis. Among these objectives;

- To determine the heat transfer during braking from braking material to the wheel and time and speed.
- To determine the temperature rise of wheel in relation to the speed of the train.
- To understand the temperature rise between wheel and tread block during braking the train in relation to the full braking time of the train.
- To understand the thermal and stress condition when train of AALRT operates at its maximum speed.
- To determine the stress induced on wheel due to the centrifugal force, reaction forces and brake pressure of the train.
- To understand the wheel and rail deformation by the thermal and stresses induced during braking and others.

1.8 Thesis Statement

Railway vehicles use different type of braking system for different trains. The performance of these braking systems differs as the braking type differs. Tread braking is one of the braking types of railway brake which tread block to press in to the railway wheel tread. During this phenomenon some engineering parameters should be considered to perform effectively its function.

In this research paper the thermal and stress analysis of the AALRT wheel is studied assuming that if it uses tread brake which is imposed due the braking action on different gradients and some resistance factors. While the train brakes from its maximum speed to stop the total energy gained of the vehicle with different phenomena is converted to heat and most of this heat is dissipate between wheel tread and tread block. This heat can raise the temperature of the tread surface of the wheel above the allowable temperature and cause different failure on it. In addition to this mechanical stress also induced on the wheel tread surface due the reaction forces on the wheel, centrifugal force on the wheel and the brake pressure during braking at different situations. Due to these conditions the thermal and stress condition of AALRT is studied and concludes the behavior of the wheel when under this assumption.

1.8.1 Limitation

During this research work the main problem encountered the proper data from the AALRT project didn’t found at time. Due to this to fulfill my achievement through this work I have taken some data from some standard manuals of some countries from internet.

1.9 Scope Of The Thesis

The scope of this research paper is to study the thermal and stress induced on AALRT wheel when braking with tread brake under different resistances which are the major factor to reduce the train speed and/or do work on the moving train. During this action the thermal and stress condition between the two materials (i.e. wheel and tread block) is to be well studied. The heat flux due to train energy and different loads applied on the wheel are also considered to analyze their effect on the wheel

1.10 Thesis Organization

The report consists of five chapters, first being introduction. In second chapter, literature review of various literatures on the thermal and stress analysis of AALRT wheel during tread braking is carried out. Third chapter depicts the analytical calculation of the energy of the train braking which is dissipated to the surrounding and the mechanical reaction forces on the wheel of the train when braking using the weight transfer method. In chapter four, the FEM of the wheel starting from modeling the wheel to analysis set up and generating the output of the analysis on ANSYS software. Chapter five deals with the results and discussion the analysis done in chapter four and concludes it contains the steps followed in static and spectral analysis of the model of the problem concludes with the conclusion & recommendation of the result.

CHAPTER TWO

LITERATURE REVIEW

2.1 Thermal Stress on Wheel

In Australia railway cars, for both passengers and freight, are quite commonly braked by using blocks which contribute to the thermal load on the rail wheel. There are two kinds of braking: stop braking and drag braking. Stop braking is used for stopping the train at scheduled stops or in an emergency. Drag braking is used to slow the trains over long distances in order to control speed. Brakes that are applied for long periods of time result in prolonged heating of the tread and the rim. Thermal loading has the ability to initiate cracks, increase the crack growth rate and reverse the residual stresses in the rim. During tread braking process the heat generated by frictional forces is distributed either as severe thermal gradients within the near surface or more gradually over the wheel rim and plate, depending on the intensity of the heating source. Block braking of a wagon or a locomotive is performed by pressing the brake block(s) against the tread of the wheel, which also is in rolling contact with the rail. The phenomenon of heat loss from a hot wheel through the contact area into a cold rail is called ‗rail chill. [14]

The thermo-mechanical interaction between brake block and wheel tread during braking has been found to cause thermal crack on the wheel tread. Due to thermal expansion of the rim material, the thermal cracks will protrude from the wheel tread and be more exposed to wear during the wheel/block contact than the rest of the tread surface. The wheel rim is in residual compression stress when is new. After service running, the region in the tread has reversed to tension. This condition can lead to the formation and growth of thermal cracks in the rim which can ultimately lead to premature failure of wheel. [15]

Mircea A. [16] states that during braking friction, the largest share of energy is dissipated as heat absorbed by friction members which are thus subjected to the thermo-mechanical intense and in his journal the centrifugal force of the wheel didn’t considered the gradients and the resistances which reduces the heat energy dissipation by opposing the motion of the train. According to F. de C. Santos [17], objective of braking is to take the kinetic and potential energy out of the train. Most of the energy is turned into heat in the regular brake system. Part of the heat flows to the wheel rim and creates a temperature gradient in the radial direction. The heated rim tends to expand and the plate, still cool, tends to restrain the rim movement. The difference in deformation between these two parts of the wheel is responsible for the rising of radial stresses in the plate. In addition, constraint causes hoop compressive stress in the rim, because expansion in the circumferential direction requires an increase in this part's mean radius. Stresses caused by braking process are called thermal stresses.

J. Ahlstrom and B. Karlsson [18] investigates that the operating temperatures for the wheel vary, but are normally found to be between 100 °C and 300 °C, depending somewhat on the ambient temperature and atmospheric conditions. In some cases, wheel rim surface temperatures have been determined to reach up to 600 °C. These high temperatures are usually associated with an occasionally-occurring flaw known as a wheel flat. This is a flat region on the normally circular rolling surface of the wheel which is created by the wheel sliding on the railroad tracks. The occurrence of sliding is a consequence of many factors such as poorly adjusted or defective brakes, slopes on the terrain that force the driver to apply the brakes while the car is moving and even contamination on the tracks due to fallen leaves, water or snow. When a wheel flat develops, the surface contact between the two metals creates friction, which over time causes the wheel to heat up to high temperatures.

In a study done by Jergeus [19] a total of 140 experiments were performed using a test train that consisted of an electric locomotive, a measuring car containing data sampling equipment, a car equipped for measuring the pulling force, and a test car. New wheels were used and an average of 10 wheel flats were created on each test under controlled conditions, including a brake that was manually controlled from the measuring car, capable of completely locking the test axle. It was determined that temperatures of up to 750 °C were reached by the outside surfaces of the wheel.

As Miloš S. Milošević [20] the thermal analysis of the braking system of railway vehicles requires a precise de-termination of the quantity of heat produced by friction, as well as the distribution of this energy between the railway wheel and the braking blocks. When the braking process occurs, blocks and railway wheel are in a sliding contact. The resulting force resists the movement so the train slows down or eventually stops. The friction between the wheel and blocks always opposes motion and the heat is generated due to conversion of the kinetic energy. However, the whole braking system is exposed to the enlarged air flow for high speed braking and the heat is dissipated.

A FEA thermal analysis is a finite element analysis that looks at how heat affects certain materials and engineering designs. This heat can come in the form of an environmental load such as an ambient temperature of a certain degree affecting a model, or due to friction in a system, effectively converting it into thermal energy. It can also come from processes of conduction through two solids, convection between a liquid and a solid, or radiation such as in space. A thermal analysis is a great way to test a model before the model is built and real world tested in a thermal chamber. It can reduce the time to test a design by weeks, allowing for several redesigns and improvements to be made in the meantime. [20]

M. Ertz, K. Knothe[21] also studies the power dissipation due to tangential stress and sliding velocity between wheel and tread block results in contact temperatures that can exceed 400 ^{o}C under normal operating conditions. They are confined to a very thin surface layer. The wheel of locomotive or wagon is simultaneously heated by the friction due to wheel/ contact and two brake shoe contacts. When a brake is working, the transformation of kinetic energy of moving masses into thermal energy takes place. This kinetic energy is dissipated between two bodies and appreciably raises their temperature at the area of the sliding contact. While the time for reaching the quasi-steady-state conditions can be very short for a moving body, it can be relatively long for a stationary body. While railway wheels are heating by friction in the contact patch and in the contact brakes, there is also heat loss due to conduction through the contact patch into the rail and into brake shoe.[22]

The thermo-mechanical interaction between brake block and wheel tread during braking has been found to cause thermal crack on the wheel tread. Due to thermal expansion of the rim material, the thermal cracks will protrude from the wheel tread and be more exposed to wear during the wheel/block contact than the rest of the tread surface. The wheel rim is in residual compression stress when is new. After service running, the region in the tread has reversed to tension. This condition can lead to the formation and growth of thermal cracks in the rim which can ultimately lead to premature failure of wheel.[23]

In most of the studies resistance due to gradient, curve resistance, air resistance and others also didn't considered. The gradient resistance can increase or decrease the heat dissipation to the environment depending on the direction of slope. But all the other resistances decrease the heat dissipation because they do work on the moving train by opposing the train motion. In this research all these resistances are considered and its output is the cumulative result of all of them except the rolling resistance.

2.2 Stresses on Wheel Due To Tread Blocks

Vahid Monfared1, Mohammadhassan Hassan, Saeed Daneshmand, Farshad Taheran and Amirhossein Monfared [24] studied as the stress distribution is a significant reality or a collection of wheel–shoe brake contact interfaces, which are dependent on geometry and physics of the contact areas, load conditions and material properties of the system of interest. There are different loads applied to the wheel and rail. Among these loads the load due to the brake force between wheel and tread block while the train starts braking to unload the passenger, the load when the train is loaded and other loads due to different track conditions (damage of the track due to) fatigue load, rolling contact damage, corrugation on the rail etc increases the stress of the wheel.

In railroad service the wheel acts as a brake drum in addition to supporting lateral and vertical mechanical loads. When brake shoes are applied to the wheel tread the surface is heated due to friction. The steel at the tread surface gets hotter tries to expand and is constrained by the colder body of the wheel rim and plate. According to Valdunes Dunkerque, Francois Demilly and Cameron Lonsdale [25] if the tread surface is heated to a high enough temperature by braking, the steel will have a reduced yield strength, and plastic deformation caused by expansion and compressive upsetting of steel in the hot zone is possible. After cooling and shrinking, continuity must be maintained between the locally yielded material at the tread surface and the constraining remainder of the wheel. Therefore, the material at the tread surface is now in tension. In severe cases of braking heating, complete stress reversal of the rim could take place. Both of these situations will allow for propagation of service related fatigue cracks in the tensile zone.[26]

Residual stresses in railway wheels can be a serious concern for railway cars mounting block (shoe) brakes. In consideration of the actual operational conditions, shoe braking can induce severe thermomechanical stresses over the tread, which may eventually turn into potentially dangerous hoop tensile stresses in the rim. The stresses built under each single event shall be rarely of considerable magnitude; nevertheless, accumulation of such tensile stresses can eventually bring the wheel rim to a hazard condition. [27]

Contact stresses are due to weight, driving forces and other types of unusual, mainly dynamic, forces. For the simplest model to calculate the stresses, one uses the Hertz Theory and evaluates the effect of the weight only. [28] The static axle loads are determined by the total mass of the vehicle. Depending on the route class, the loads range between 160 KN and 225 KN. Through various influencing factors (planned super-elevation or sub-elevations, track imperfections, one-sided load etc.) loads may be unevenly distributed - and additionally to the vehicle-related factors – they may create a dynamical component. [29]

The weight of the wheel (on average about 300 Kg) together with the weight of other components of the vehicle imposes a static stress on the rail. While moving the wheel and rail are exposed to the dynamic load. The cumulative effect of these loads has also especial effect on the wheel and rail while it is moving. [30]

As stated above most of the research paper on the rail way wheel was focused mainly on the thermal effect when braking but the mechanical effect have to be added on the thermal load to get most appropriate output. In this research paper in addition to the thermal loading, the mechanical load is also considered. Theses mechanical loads are the centrifugal force on the wheel when moving taking the axel as the center of rotation, the pressure on the wheel tread due to the braking action and the reaction force on the wheel due to the weight transfer from the train boby to the wheel/axle when braking .

CHAPTER THREE

ANALYTICAL MODELING OF THERMAL AND MECHANICAL LOADS ON WHEEL

Before working the thermal and stress analysis between wheel rail and tread block, it is necessary to define an analytical model of a thermal and stress analysis that describes the heat transfer of the heat generated by friction at surfaces which are in contact, between a railway wheel and braking blocks, through the wheel and blocks, as well as heat outflow of the whole braking system due to cooling of the surrounding air. For those purposes, an analytical model for analyzing thermal effects in braking systems of LRT cars was utilized and its adopted procedure is presented in this paper for a braking system of LRT railway vehicles.

The thermal aspect around the contact region can be studied by examining the heat transfer between stationary break shoe and a moving body (relative to the heat source). The slip between wheel and braking system causes friction heating of both bodies.

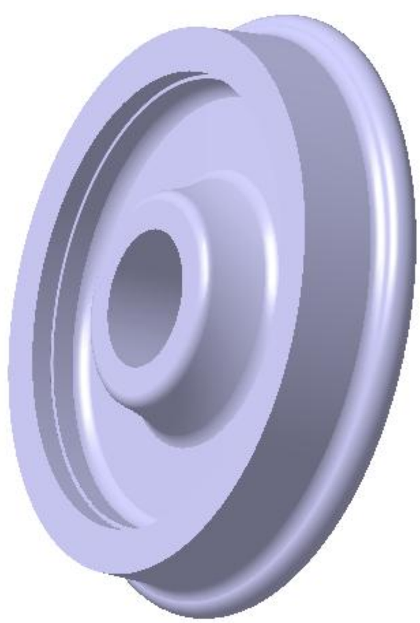

Figure 4 LRT wheel profile

3.1 Mathematical Model for Interface Temperature Rise

The heat flux evacuated of surfaces in contact (between blocks and railway wheel) is equal to the power friction of the wheel and block.[31] In this case the heat dissipation between wheel and rail contact is neglected because it is simply a rolling effect and has no significance effect on the wheel and rail. The heat power due to this condition is related with the braking force, the radius of the wheel and the rotational speed of the train. The heat power generated per unit contact area at the radius *R* of the wheel can be calculated by the following equation:

$$P(R) = -FR\omega \text{ ------------------------------ } 3.1$$

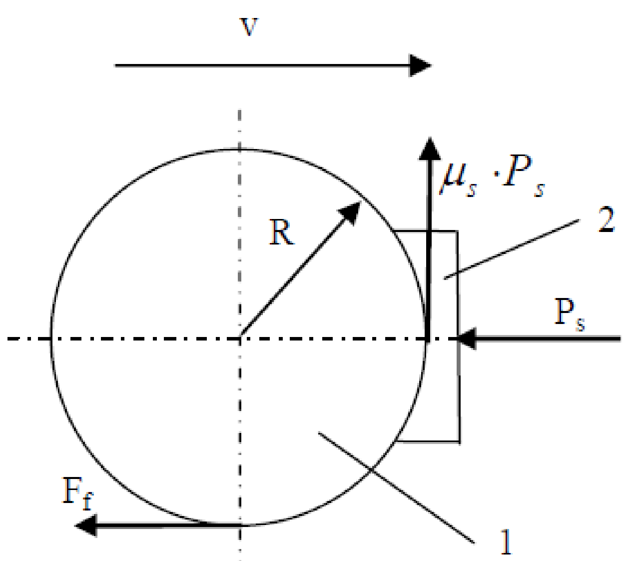

Figure 5 Brake force application on the wheel by the brake shoe

$$F = \frac{\mu_s \; P_s}{A} \text{ ------------------------------ } 3.2$$

Where $\boldsymbol{\mu_s}$ is the friction coefficient between wheel and tread

$\boldsymbol{P_s}$ is the normal pressure on one of the braking blocks and

$\boldsymbol{A}$ is the area of the contact surface between one braking block and the railway wheel.

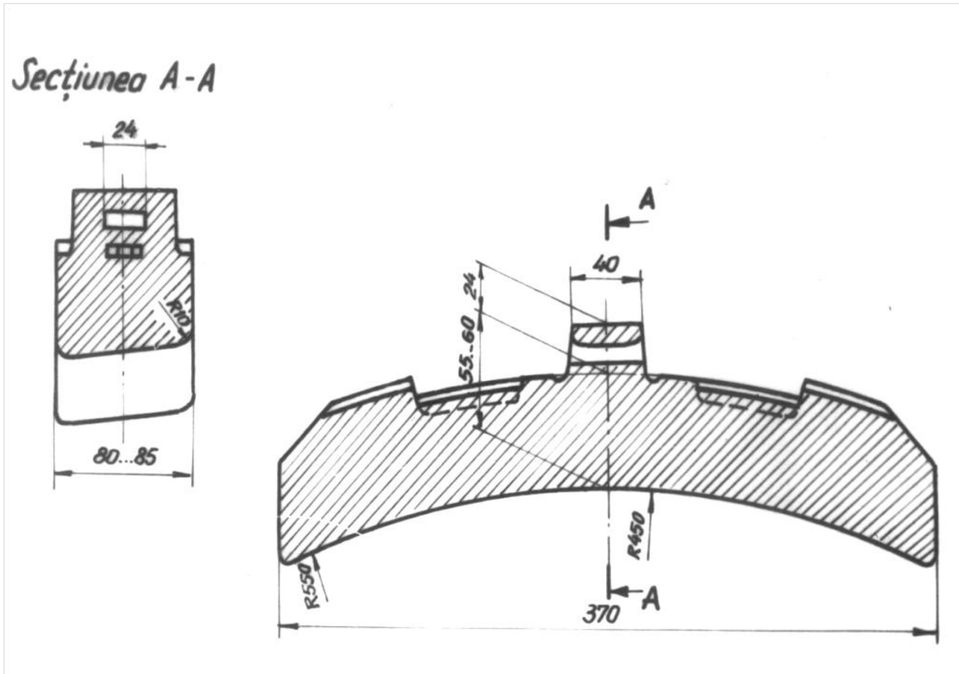

Figure 6 Brake shoe

On experimental basis it has been established that the optimal length of brake shoes are 320 mm for easy mounting in shoe-wear and 250 mm when fitted to double shoe-wear. Width of the brake shoes in all cases is 80 mm [32]. Using this standard the area (A) of contact between railway wheel and block is

$$A = l\,b \text{ ----------------------------- } 3.3$$

But the most accurate way of determining the brake force is using the distribution force due to the pressure in the brake cylinder by the taking the differential contact area between wheel and block. This enables us to determine the appropriate size of the trade block. In addition to this at which type of trade block the temperature of the wheel rises to the maximum value will be obtained so that the wheel will maintained at the permissible value of the maximum temperature. But the most accurate way of calculating the heat power generation is to use the pressure distribution over the wheel tread by the tread block on the wheel tread. This also enables us to determine which size of the tread block is appropriate to use with minimum cost and compact design for the rail way locomotive.

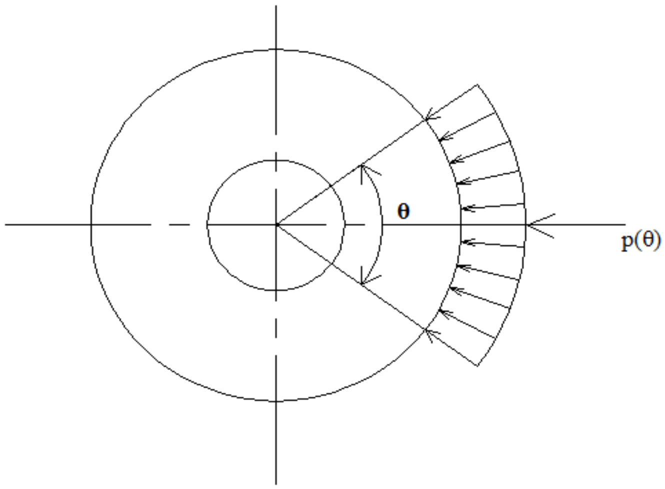

Figure 7 Brake Distribution Force

The normal pressure from the brake cylinder is distributed the surface of tread wheel and the tread block of brake shoe. From the figure given below the differential area of contact between wheel and tread block is dA which related to the differential contact angle and the radius of the wheel will be written as.

$$dA = bRd\theta ---------------------------3.4$$

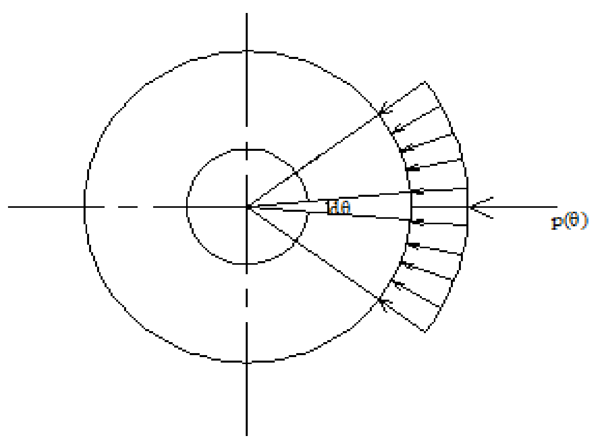

Figure 8 Differential contact angle of the wheel

The surface pressure between the tread block and railway wheel, on behave of the calculated force applied to the tread block, needs to be determined.

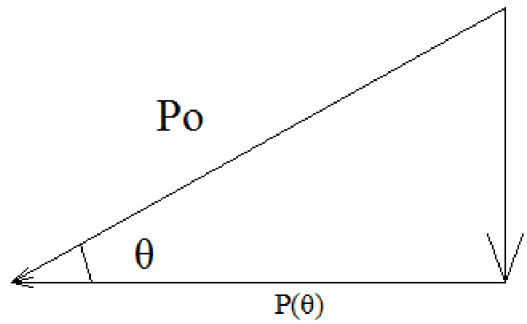

Figure 9 Determination of pressure distribution

From the figure it is shown that the normal pressure is perpendicular to the surface of the wheel. From this the pressure at some specified angle of the contact between wheel and tread block will be calculated as follows.

$$P(\theta) = P_o \cos(\theta) --------------------- 3.4$$

The optimum range of the contact angle of tread block and shoe is between 30^o and 45^o. [43] According to the Indian Railway Institute, Rao mannual the pressure in the brake cylinder ranges from 280 to 350KPa. From this equation the maximum pressure at the two different contact angle will be calculated so that P (30^0) =303KPa and P (45^0) =247.5KPa. The minimum the brake pressure applied on the wheel is the maximum the brake force applied on the wheel at the contact area.

$$dF = P_o \cos(\theta)\, b\, R\, d\theta ------------------------ 3.5$$

$$F = \int_{-\theta}^{+\theta} P_o \cos(\theta)\, b\, R\, d\theta ---------------------- 3.6$$

$$F = 2P_o\, b\, R \sin(\theta) ------------------------ 3.7$$

By integrating the force the total brake force applied on the wheel brake force on the wheel occurs at θ=45^o (i.e. F=12.87KN).

Therefore the maximum pressure which is normal to the force acting on the thread block can be calculated by integrating the above equation with respect to θ.

Since the rotational speed is determined from the speed of the train in relation to the radius of the wheel, the heat power generated per unit contact area is

$$P(R) = -F_A V ----------------------------- 3.8$$

The maximum train speed is 19.44m/s and frictional force per unit area (637.73KN/m^2) so that P (330mm) is 12.397KN/m^2 for θ=30^o and for θ=45^o P(330mm) s about 8.287MW/m^2. No matter how large the value is it is due to the large contact are between the wheel and the block. The heat power generation calculated above correspondent to the heat flux entering the wheel from the block.

3.1.1 Heat Transfer

In order to accurately determine brake temperatures at a particular time (t) in the braking phase, knowledge of the heat transfer coefficient is required. The heat transfer coefficient (h) will depend on the airflow in the region of the brake block and the vehicle speed. It will therefore vary constantly throughout the braking process. It is generally considered extremely difficult to obtain accurate values of the heat transfer coefficient.

Heat dissipation from the brake block will occur via conduction through the brake assembly, radiation to nearby components and convection to the atmosphere. At high temperatures heat may create chemical reactions in the friction material, which may dissipate some of the braking energy. While conduction is an effective mode of heat transfer, it can have adverse effects on nearby components. Radiation heat transfer from the tread block will have its greatest effect at higher temperatures but must be controlled to prevent beading of the tire. It is estimated that the amount of heat dissipation through radiation under normal braking conditions is negligible. Convection to the atmosphere must then be the primary means of heat dissipation from the brake rotor. Convection is governed by the expression:

$$\dot{Q} = hA(T_s - T_\infty) -------------------------- 3.9$$

In most heat transfer problems, we are concerned with solid walls separating liquids or gases from each other. In such cases we usually do not know the temperatures on the wall surfaces, but rather the temperatures of the bulk of fluids on both sides.[33] The heat dissipation from the free surfaces of the wheel and blocks to the surrounding air is described by[34] both convection and radiation:

$$\dot{q} = -h(T_s - T_\infty) - \varepsilon\sigma\left(T_s^4 - T_{ref}^4\right) ------------------ 3.10$$

σ is the Stefan-Boltzmann constant (5.67·10−8)

To calculate the convective film coefficient as a function of the railway vehicle velocity *v*, the following formula [35] should be used:

$$h = \frac{0.037K_a}{2\,R}\,Re^{0.8}Pr^{0.33} --------------------- 3.11$$

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

$$Re = \frac{2\,\rho_{air}R\,v}{\mu_{va}} -------------------------- 3.12$$

$$Pr = \frac{C_p\,\mu_{va}}{K_a} ------------------------------ 3.13$$

Table 1 Properties of Air

No	Properties of materials	Value
1	Thermal conductivity(k_a)	$\mathbf{2.624 * 10^{-5}}\ \boldsymbol{KW}/m$K
2	Density of air at 300K(ρ)	1.177Kg/m^3
3	Dynamic viscosity(μ_{va})	1.983*10^{-5}Kg/ms
4	Specific heat capacity (C_a)	1.0049KJ/KgK
5	Train speed(v)	70Km/hr
6	Radius of the wheel(R)	330mm
7	Ambient temperature (T)	25.5℃

Taking the data from the above table, convective film transfer coefficient according to the equations given above will be, h is 66.58W/m^2 ^{o}c.

From the thermodynamic point of view the total heat energy generated by the moving train is proportional to the total change in mechanical energy of the moving train. During this situation the temperature of the moving component of the train will rise to some initial temperature before braking from which the maximum temperature rise while braking will be calculated.

$$\dot{Q} = \Delta E \text{ ----------------------------- } 3.14$$

The rate of change in hest energy of the train can be expresses as

$$\frac{dQ}{dt} = UA(T_i - T_\infty) \text{ ------------------------- } 3.15$$

And

$$\Delta E = m\Delta TC \text{ -------------------------- } 3.16$$

From equations 3.14, 3.15 ,3.16 and Newton's law of cooling

$$\frac{T - T_\infty}{T_i - T_\infty} = e^{-\frac{(UA)}{mC}t} \text{ ------------------------- } 3.17$$

Rearranging this equation

$$T_i = T_\infty - \frac{\Delta T}{1 - e^{-\beta t}} \text{ ------------------------- } 3.18$$

Where $\Delta T = T - T_\infty$ and $\beta = \frac{UA}{mC}$

3.1.2 Heat Flux

Applications with dry sliding friction between metal parts are common in classical mechanical engineering practice. However, the processes in railway brake systems are quite complex. There are many studies about the steady state friction of such applications, some of them considering even the wear and the thermal effects, yet the real-life braking are typically a transient phenomenon, and the vehicle is usually stopped in seconds or in a few minutes. Since most of the kinetic energy of the train is transferred to heat by the friction between the wheels and the brake blocks, a big amount of heat is developing in a short time. It was found earlier that in the beginning of the braking, the edges of the brake blocks contact the wheel, even when the brake block is worn. This can be explained by the thermal expansion of the brake blocks: at the end of the previous braking, the brake block is worn, so the radii of the wheel and the brake block approximately are equal. Then, the system cools down, and since there is a great decrease in temperature, a significant thermal deformation occurs, which causes the brake block to have a shape that fits the wheel only at the edges.

The distribution of heat between wheel and brake pads is directly dependent thermo physical properties of materials that are built [5]. It was found both in the theoretical calculations and experimental determination that most of the developed heat is taken from the wheel so, in the wheel can occur significant increases of temperature. Both temperature values reached in the wheel and their distribution during braking is of particular importance because it was found that high temperatures that develop thermal stresses rise to very high. In extreme cases (very dangerous situations in terms of traffic safety) can lead to weakening or spins of the bandage's wheel or wheel on the wheel axle.

Rail vehicles are characterized by the movement on the railways through wheels auto guided by contact forces between wheels and rails. Braking facility holds the most important role in ensuring the movement of railway vehicles in safety conditions for the traffic.

During braking friction, the largest share of energy is dissipated as heat absorbed by friction members which are thus subjected to a thermo mechanical intense. In the field of railway braking for example, brakes disk are brought considerable energy to dissipate in a few tens of seconds, several mega joules for a single disc, or even tens of mega joules in the case of high speed.

The severity of such stresses leads to localization phenomena thermal interface, which manifest themselves in different forms, such as the hot bands migrants, or macroscopic hot spots, depending on the severity of braking, temperature reaching locally extreme values.

High thermal loads of railway wheels very often occur as a result of a long-term braking for the maintenance of constant train velocity on down-grade railroads. When the LRT train is broken from the maximum speed to the stop the most of the energy of the train is dissipated as heat between wheel thread and thread block and the little of heat is dissipated to air. The heat energy generated from braking the vehicle between wheel and thread block is then distributed over the thread surface of the wheel as the heat flux. The heat flux over the surface of the wheel rises the temperature of the wheel thread to the maximum.

The temperature of this surface is dependent on the speed of the train from which it is broken, the mass of the train and the geometry of the train (i.e. the track gradient and the curvature on the station in which the train is braked down). As these factors increases the heat flux interning the wheel thread increases and then the temperature of the wheel and thread block interface. And it affects the mechanical and physical properties of the wheel and the thread block.

3.2 Energy Generated By Train When Braking

Heat generation due to friction in the sliding contact of two bodies influences friction and wear characteristics of brake systems. We note that numerous experimental evidences suggest that the contact area is generally circular, e.g. tread broke railway wheels exhibit circular thermally affected zones on the surface. [36]

When a brake is acted upon the moving train, the kinetic energy of the moving mass transforms into thermal energy. The heat thus dissipated [37] between the contacting bodies. In addition to this when the train is moving on the gradient both the kinetic energy and the potential energy developed by the train is converted to the thermal energy between the brake block and the wheel of the train. This in turn raises the temperature of the wheel at the contact surface.

In railroad service the wheel acts as a brake drum in addition to supporting lateral and vertical mechanical loads. When brake shoes are applied to the wheel tread, the tread surface is heated due to friction. Severe thermal input into a wheel occurs when a loaded train descends a grade for an extended period of time. Also, the failure of the brake mechanism may keep the brake locked on the tread of the wheels. On these occasions, the wheel rim is heated to a high temperature for an extended period of time.

When braking the train the heat energy developed generates the heat flux over the surface of the wheel. The heat flux entering the wheel surface can be taken in two braking conditions.

1. While braking the train on a flat track and
2. While braking the train over the inclined track

During braking longitudinal conduction has the effect of heating up a rail section ahead of the wheel but due to its effect is minimum longitudinal conduction can be negligible. [38]

3.2.1 Braking On A Straight Track

When braking on a flat surface the train has only kinetic energy because the train potential energy is neglected as the gradient of the train is set to be zero. Due to this kinetic energy the heat is developed between wheel and brake block due to friction. The heat energy during this time is calculated from the kinetic energy.

$$Q = m\,V_o^2 \text{ ------------------------------------ } 3.19$$

Where Q is heat energy developed due to the kinetic energy of the train

The train has also many rotating components and so that it has a rotational kinetic energy due to rotating parts. Then the energy equation will be

$$Q = \frac{1}{2} m\,V_o^2 + \frac{1}{2} I \omega_o^2 \text{ ------------------------------ } 3.20$$

Using this equation the total energy of the wheel during braking over the flat track is obtained. But the train has many rotating parts such as wheels, motor armature, gears have to be accelerated in an angular direction therefore weight is multiplied by a factor K which is taken as a 1.2, 1.1-1.15 and 1.03 for locomotive, EMU and freight train respectively.[39] The determination of moment of inertia for all these rotating elements is complex so that taking their masses in to account and using acceleration resistance factor (K) the equation is simplified to

$$Q = \frac{1}{2}\, m_e V^2 \text{ ----------------------------- } 3.21$$

Where m_e (k*m) is combined mass, taking K=1.2 for the locomotive, the heat energy developed as the kinetic energy while the train braking.

According to [40] a justifiable assumption that only 90% of heat generated goes into the railway wheel has been used therefore the above equation becomes

$$Q = 0.9\left(\frac{1}{2}\,m_e V^2\right) ---------------------------3.22$$

According to the AALRT Technical Specifications of Vehicles given in table below from AALRT data since the braking time of the train is16 seconds, so that the rate heat power due to the kinetic energy of the train is about 933,611W. Taking three different angle of contact angle between wheel and block three different values are attained for both temperature and heat flux because as contact angle increases the contact area increases, as the area of contact increases the heat flux decreases which in turn decreases the temperature of the contacting bodies increase.

Table 2 AALRT technical data

No	Parameter	Value
1	Ambient temperature	$0 - 29.7$℃
2	Average daily temperature	25.5℃
3	Wind speed	Grade 8 ground 20.7m/s
		Grade9 elevated 24.4m/s
4	Maximum gradient	55%
5	Axle load	$\leq 11tone$
6	Vehicle type	6 axle bidirectional
7	Length of car body	$\leq 30000mm$
8	Height of vehicle floor from top rail	380mm
9	Wheel base	1900mm power bogie
10	Wheel diameter	660mm
11	Searing capacity	59.24tone
12	Maximum train speed	70Km/hr
13	Average acceleration	0-40Km/hr ≥ $1m/s^2$
		0-70Km/hr ≥ $0.7m/s^2$
14	Average braking deceleration	Service brake ≥ $1.1m/s^2$
		Emergency brake ≥ $2m/s^2$

The area of contact between wheel and thread block for θ=30^o is 2.723*10^4 mm^2. Then the heat flux entering the wheel is 34.2MW/m^2. Since the LRT train has three bogies each with four wheels the train has twelve wheels and blocks. The total heat flux on a single wheel is then 2.85MW/m^2. For θ=45^o the contact area between wheel and block is 4.048*10^4 mm^2, the total heat flux entering the wheel is 23.063MW/m^2 and the heat flux at each wheel is 1.92MW/m^2.

3.2.2 Braking On Inclined Track

The alignment of the rail way track is not only straight but also has a different gradient and curves. Now the train has both kinetic energy and potential energy due to its motion and its inclination /gradient which intern increases the heat energy dissipation from the contact surface to the surrounding. The maximum gradient of the tack for AALRT is 5%$_o$. In this case the total energy of the train is the combination of the two energies. Again due to the rotation of the wheel there is also rotational energy. The gradient increases and decreases the dissipation of heat energy depending on the direction of the movement. So the total energy will be the summation of all the three energies. Therefore

$$Q = 0.9\left(\frac{1}{2}m_e V_o^2 - mg\Delta h\right) ------------------------3.23$$

The change in the elevation of the train depends on the geometric survey of the track. The measured horizontal track length and the elevation above sea level create different grade value of the track at each station terminal. Considering the train gradient in to account and from the trigonometric function

$$\Delta h = V_o t_b \sin\delta ---------------------- ------3.24$$

Since the maximum allowable gradient level for the railway track is only about 5%o, $\sin\delta = \delta$ but the calculated maximum gradient for AALRT project is about 3.641%$_{o.}$ So that when the train is moving down the gradient the total heat energy dissipated between wheel and trade block is

$$Q = 0.9\left(\frac{1}{2}m_e\, V_o^2 + mgV_o t_b \delta\right) ----------------------- 3.25$$

So that the heat power generated is 1,534,664.25W and the heat flux entering the wheel during this for θ=30^o is 56.215MW/m^2 and the heat flux entering a single wheel is 4.68MW/m^2. For θ=45^o, the heat flux entering a single wheel is about 3.13MW/m^2.

When the train is moving up the gradient the total heat energy dissipation will be

$$Q = 0.9\left(\frac{1}{2}m_e\, V_o^2 - mgV_o t_b \delta\right) ----------------------- 3.26$$

Where

$$t_b = \frac{V - V_o}{a} -------------------------------- 3.27$$

During this time the heat flux entering a single wheel for θ=30^o is 1.015MW/m^2 and for θ=45^o is about 0.68MW/m^2.This shows that the heat flux entering the wheel when the train is moving up the gradient is less that of moving on a straight track and down the gradient. The operating line has different value of gradient so that at each and every station the value of heat energy is different. The different station designation and its gradient are given as in the table below. As the result in the table shows te maximum gradient in the EW line occurs at the station EW4

Table 3 East-West line gradient

No	**stations**	**gradient(δ)**	**sin δ**
1	EW1	0.435	0.0076
2	EW2	0.894	0.0156
3	EW3	-0.286	-0.005
4	EW4	-1.861	-0.0325
5	EW5	-0.6588	-0.0115
6	EW6	-0.0974	-0.0017
7	EW7	0	0
8	EW8	-0.0859	-0.0015
9	EW9	-0.802	-0.014

10	EW10	-0.544	-0.0095
11	EW11	-0.0593	-0.001
12	EW12	1.528	0.0265
13	EW13	0.573	0.01
14	EW14	1.088	0.019
15	EW15	0.802	0.014
16	EW16	-0.172	-0.003
17	EW17	-0.0859	-0.0015
18	EW18	0.0859	0.0015
19	EW19	1.547	0.027
20	EW20	-1.547	-0.047
21	EW21	1.747	0.0325
22	EW22	0.17	0.003

As the result in the table shows the maximum gradient in the NS line occurs at the station NS22

Table 4 South-North line gradient

No	**stations**	**gradient(δ) in degree**	**sin *δ***
1	NS6	-0.716	-0.0125
2	NS7	-0.802	-0.014
3	NS8	-1.26	0.022
4	NS9	-1.66	-0.029
5	NS10	-1.7756	-0.031
6	NS11	0.5156	0.009
7	NS12	-0.487	-0.0085
8	NS13	-1.432	-0.025
9	NS14	-2.205	-0.0385
10	NS15	-0.401	-0.007
11	NS16	-0.172	0.014
12	NS17	-0.0859	-0.003
13	NS18	0.0859	-0.0015

14	NS19	1.547	0.0015
15	NS20	-1.547	0.027
16	NS21	-2.163	-0.03776
17	NS22	-3.641	-0.06364
18	NS23	-1.856	-0.0324
19	NS24	0.138	0.0024
20	NS25	2.336	0.0408
21	NS26	0.4297	0.0075
22	NS27	1.375	0.024

The contribution of kinetic energy and potential energy for heat dissipation is different for different stations. The kinetic energy and potential energy of the vehicle when the train starts braking before it reaches the station is given in the following graph.

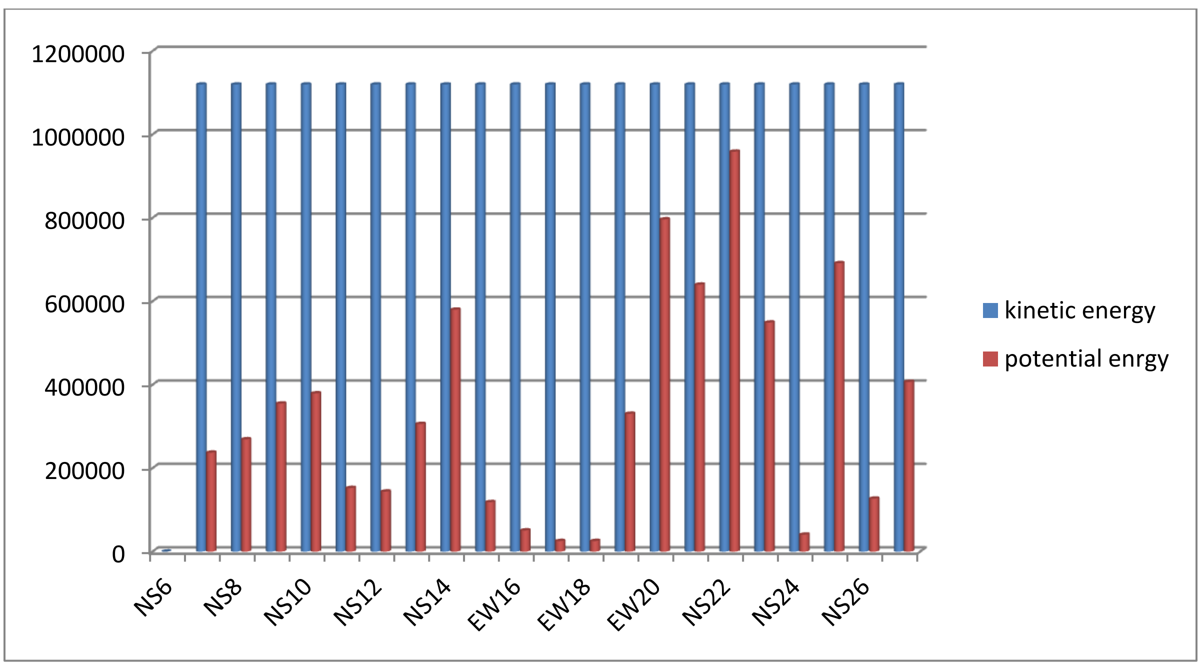

Figure 10 kinetic energy and potential energy of the train

3.2.3 Brake Self-Locking

The brake self-locking is created when the actuating force on the brake is zero and/or the moment created by the normal brake force is lower than the moment created by the frictional force of the tread/block brake.

For tread braking the pivot is located at the point where the tread is symmetrical and the moment of the frictional force is zero. Due to this condition the moment of the normal force is always lower than the moment of the frictional force and so that the self locking is avoided for treads braking. [41, 42]

3.3 Train resistances

3.3.1 Alignment resistance

Alignment resistance includes grade resistance and curve resistance. The former can be computed exactly, whereas the latter is obtained empirically.

3.3.1.1 Grade resistance

Figure 2 shows that the gravity force of a train on a grade can be resolved into two components: the force along the slope and the force normal to the incline. The former creates a resistance to the movement of the train and is defined as grade resistance. From simple trigonometry, it can be shown that grade resistance is

$$R_g = mg\sin\delta ------------------------- 3.28$$

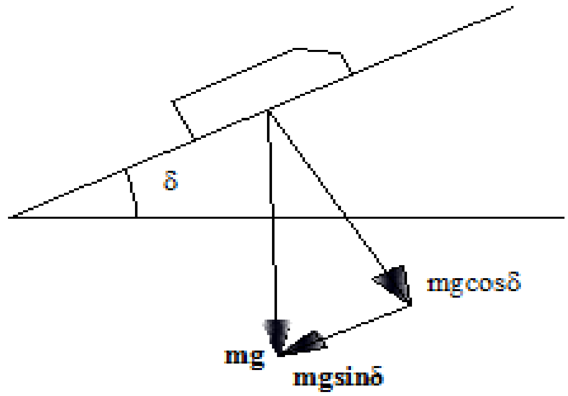

Figure 11 Train moving on the gradient

For railway alignments, θ is usually very small and for such values, $\sin\delta \approx \tan\delta$. Let r_g be the coefficient of grade resistance (‰).

$$R_g = r_g mg \text{ ---------------------------- } 3.29$$

Note that grade resistance is only a general term. It is not necessarily a resistance to train motion. The actual effect depends on the sign of δ. For δ> 0, which indicates an incline, grade resistance is in the opposite direction of train movement. On the other hand, for δ < 0, which represents a decline, grade resistance is in the same direction of train movement and can contribute to train acceleration. If, however, δ= 0 (i.e., a level alignment), then grade resistance is neither an accelerating force nor a decelerating force for the train.

3.3.1.2. Curve resistance

When a train runs in a curve, its instantaneous moving direction is tangent to the curve. Since rail tracks force the train to travel along the curve, the centrifugal force acting on the train and the friction between wheels and tracks produce extra resistance to train movement. The additional resistance is called curve resistance, which can be expressed as

$$R_c = r_c mg \text{ -------------------------- } 3.30$$

The coefficient of curve resistance r_c depends on the friction coefficient, the gauge, the distance between axles in vehicles, and the radius of the curve. It usually increases as the increases of gauge, axle distance, and friction coefficient, but decreases as radius increases. For a specific rail system, the above factors are considered to be known data except radius, which varies with alignment geometry. Hence, r_c can be expressed as a constant *K* multiplied by the reciprocal of the radius γ (m), i.e.,

$$r_c = \frac{K}{\gamma} ------------------------------ 3.31$$

The typical value of *K* in equation (6) ranges from 500 to 800, depending on rail systems. Some of them are listed in Table 4. Note that different *K* values do not make big differences in computing r_c. For example, when the radius of a curve is 300 m (which is very small for rail systems), the difference between the resulting *c r* for *K* = 500 and *K* = 800 is only 1‰.
When the radius is 3000 m, the difference is even as small as 0.1‰!

3.3.1.3. Equivalent grade

Since *g r* and *c r* are independent of train velocity and both are in ‰, it is convenient to add them together. Let r_e denote the sum ofr_c *and* r_g, i.e.

$$r_e = r_g + r_c -------------------------3.32$$

Thenr_e represents the coefficient of total alignment resistance and is called “Equivalent Grade” (Hay, 1982).

These resistance decreases the total energy of the vehicle generated as the train moves with the speed v and the potential energy of the train due to the grade. The total work done by the resistances on the moving train a distance s when braking will be given by

$$W_{res} = R_e\, s -------------------------3.33$$

But from kinematics the braking distance when the train stops will be given by

$$S = \frac{V^2}{2\,a} ----------------------------3.34$$

Then

$$W_{res} = R_e \frac{V^2}{2\,a} ----------------------------3.35$$

.

The Typical Values of *K* for different countries in Computing Curve Resistance Coefficient s give in the table below.

Table 5 Different value of curve resistance constant

No	Rail System	The Value of *K*
1	SNCF	800
2	Japan	800
3	AREMA	700
4	China	700
5	Department of Rapid Transit Systems, Taipei City Government	700
6	Taiwan Railway Administration (TRA)	600
7	IR Universalformel	516
8	RFFS Universalformel	505

Taking the Chinese standard and inserting in to the above equation the train curve resistance will be calculated.

The station NS22 has no curve and has maximum negative maximum heat flux occurs at this station from the whole station in the AALRT line.

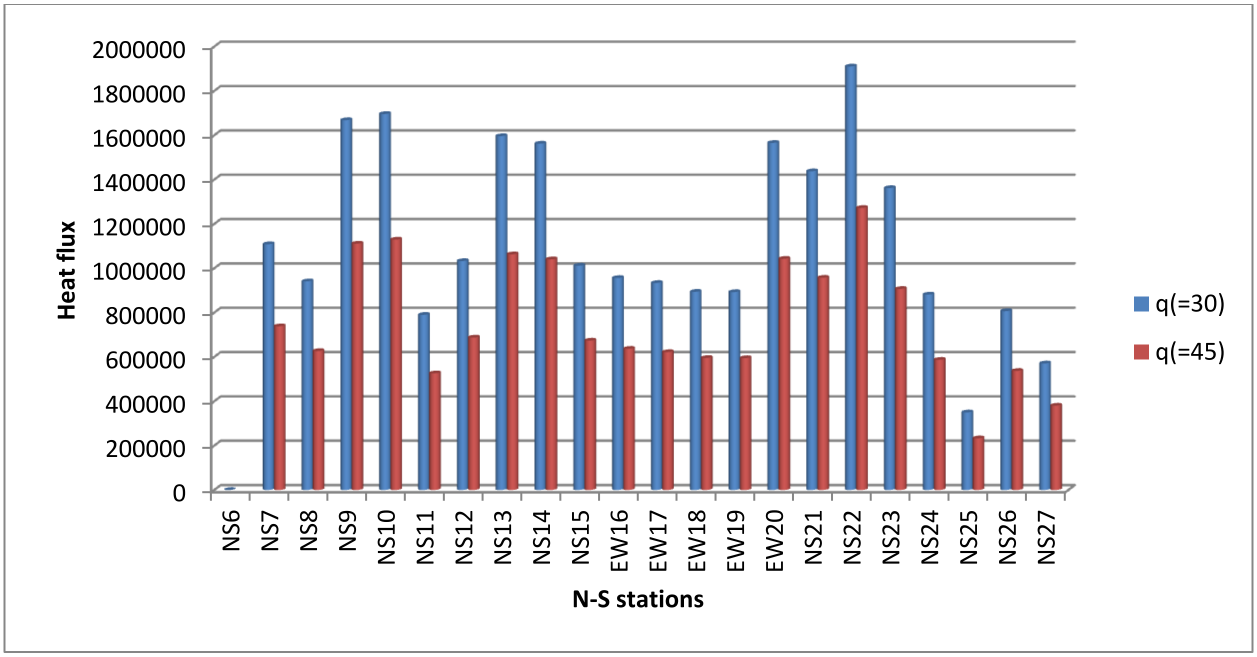

Figure 12 Heat flux distribution for N-S line

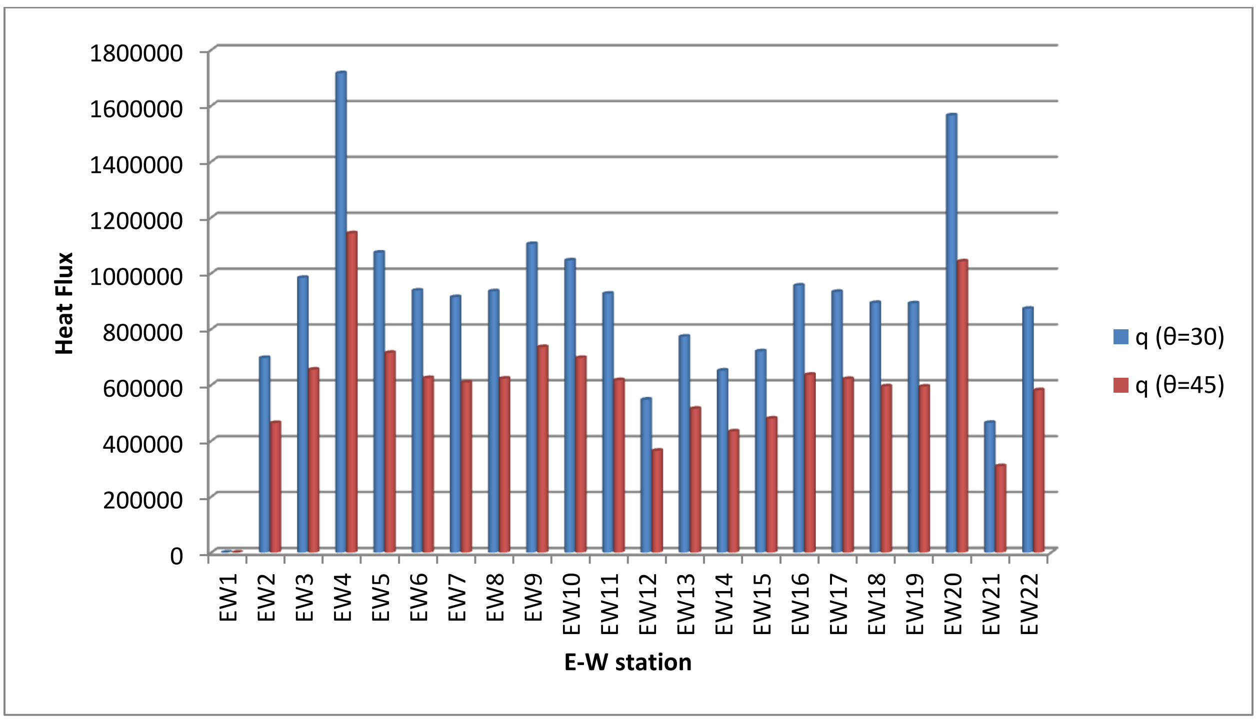

Figure 13 Heat flux distribution for E-W line

3.4 Mechanical Forces on the Wheel When the Train is Moving

When brake s applied to the moving train several forces are applied on the wheel of the moving vehicle. These forces are generated due to the braking action, rolling resistance, curving resistance, grading resistance and other reactions. These forces and the centrifugal force of the wheel impose stress to the thread surface of the wheel. So the study of this stress has a vital role in the stress analysis of the wheel during braking. In this study the effect of the centrifugal force together with the reaction forces due to weight transfer when braking the train are taken in to account to determine the mechanical properties of the wheel.

3.4.1 Wheel influencing parameters

Three influencing parameters in the domain of contact mechanics include material properties, interacting forces and the geometry of contacting bodies. Hence, each of these parameters is described concisely below.

3.4.1.1 Wheel Loading Configurations

Locomotive steel wheels are subjected to a series of normal and tangential forces due to the weight, curving and tractive effort. On the other hand, large braking forces from its contact type, i.e., brake shoes on the wheel tread, and its non-contact type, i.e., auxiliary dynamic braking system could highly affect the wheel surface. Moreover, wheel slip protection system of this specific locomotive, in addition to the application of a new generation of traction motor, i.e., AC type with a higher capability of power transmission, might have enhanced the adhesion coefficient. Hence, such an increased tractive effort could substantially influence material behavior of the wheels.

3.4.1.2 Wheel Material Properties

In order to adapt various large forces within the wheel/rail interface with wheel material properties, the application of a variety of steel wheels have been proposed by the railway regulations.

3.4.1.3 The Effect of Geometry

Wheel dimensions and profile are significant factors to change dynamic related issues and stress configurations. For instance, wheel diameter can vary the size of the contact patch. Furthermore, any change in the diameter of a locomotive wheel could vary the applied torque within the tread region either in the acceleration or the braking process. Wheel profile could considerably affect dynamic behavior of the wheel sets and the related bogie through the equivalent conicity. Also, the shape, size and location of the contact patch are subjected to the wheel profile, and its lateral displacement and gradual variations. As a result, the profile is usually considered by the researchers in the railway field.

3.4.2 Centrifugal force of the wheel

The wheel rotating on the rail imposes the centrifugal force on its own due to its weight. Due to the centrifugal load the pressure/stress is applied to the surface of the wheel thread which is dependent on the mass of the wheel, rotational speed of the wheel and the radius of the rotating wheel. According to the UIC, S1002 and other railway standard the mass and the radius of the wheel are selected to be 230Kg and 325mm respectively and the rotational speed of the wheel is calculated from the speed of the vehicle in relation to the radius of the wheel. Therefore the centrifugal force on the wheel is expressed as

$$F_c = m_w \omega^2 R \text{ --------------------------- } 3.36$$

Since the maximum speed of train is 70Km/hr, the rotational speed of the wheel is to be about 58.9rad/sec. Then the centrifugal load of the wheel is about 263,313N.

3.4.4 Reaction forces on wheel while braking

3.4.4.1 Forces due to the weight transfer during braking

When a locomotive is standing still on straight track, its weight is uniformly distributed on all axles but during traction, reactive forces and turning moments re-distributes the loading on each axle. System for transmission of tractive effort from wheel to axle-bogie-body-draw bar pull results into non-uniform distribution of weight on each axle. [43] On application of brakes on the vehicle, inertia forces are set up. The braking force is applied at lesser height whereas the vehicle load is at a higher height which resulting a couple to act on the body of the vehicle. This results an additional loading on leading bogie as compared to trailing bogie. [44]

3.4.4.2 Reaction force of the car body

Consider the car body given in the figure below moving on a straight track with its center of gravity H, height from the bogie frame h, length from the two bogie center L and length from the center of the leading and trailing axle l.

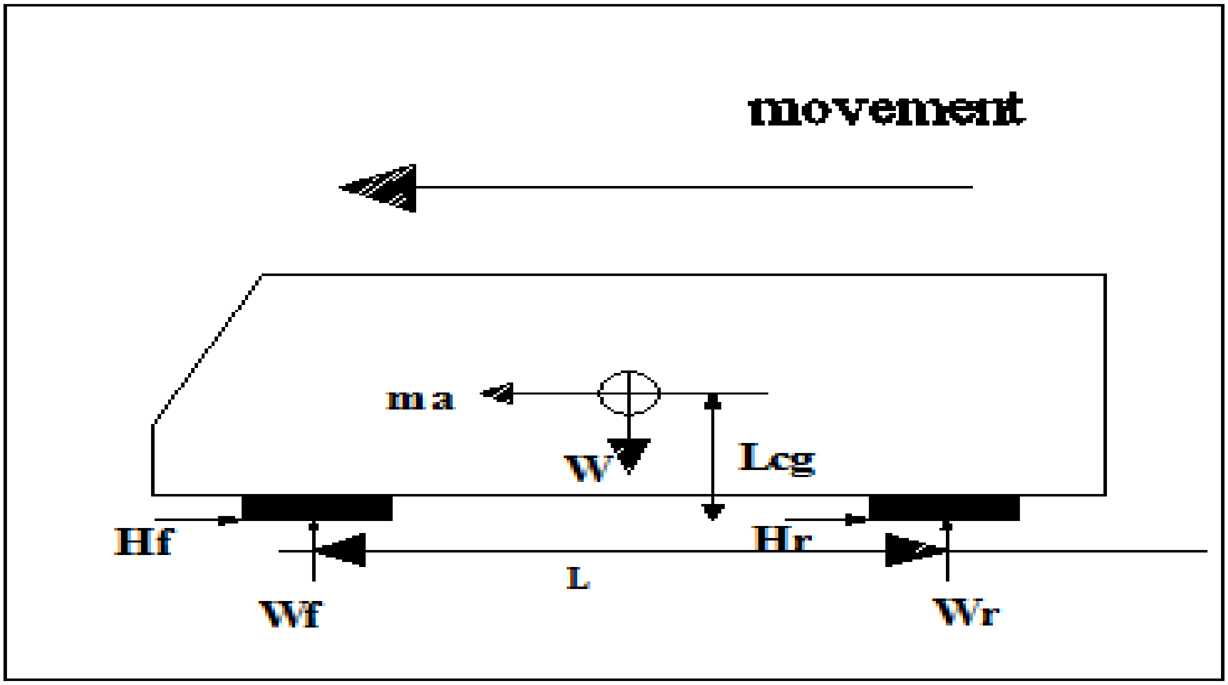

Figure 14 Reaction forces on the train

The reaction force on the horizontal plate is determined from the Newton's second law of motion and ignoring draft gear reaction on a straight track it is calculated as follows

$$F_{H,f} + F_{H,r} = m_b a \text{ --------------------- } 3.37$$

For the train moving on a gradient it will be

$$F_{H,f} + F_{H,r} + m_b \text{ g} \sin\theta = m_b a \text{ ---------------- } 3.38$$

The values of $F_{H,f}$ and$F_{H,r}$ are assumed to be equal and opposite to the inertia force and therefore the average reaction force on the center plate will be

$$F_H = \frac{m_b a}{2} \text{ --------------------------- } 3.39$$

Weight of the bogie is approximately 3.2tone. So that

$$F_H = \frac{(m_t - m_{tru})a}{2} ------------------------3.40$$

So that H is about 40.04KN for straight track and 7.529KN for the maximum gradient.

Taking moment at front center plate where $F_{H,f}$ and R_f meet: the reaction force at the rear plate can be found as

$$R_r\, L_{tr} + (D_f - D_r)L_d + \frac{(m_b a L_{cg})}{g} - \frac{(m_b L_{tr})}{2} = 0 ------------ 3.41$$

But for LRT case there is draft gear so that the second term is neglected and the equation will be

$$R_r L_{tr} + m_b\, a\, L_{cg} - \frac{(m_b\, g\, L_{tr})}{2} = 0 --------------------3.42$$

From equation 3.41 the reaction force on the front center plate is

$$R_r = \frac{(m_b g)}{2} - \frac{(m_b\, a\, L_{cg})}{L_{tc}} ------------------------ 3.43$$

In the same way the reaction force on the front center plate will be

$$R_f = \frac{(m_b g)}{2} + \frac{(m_b\, a\, L_{cg})}{L_{tc}} --------------------- 3.44$$

Different literature defines the weight transfer in percents that the weight transfers on the leading and trailing bogie. According to [43] the weight arrangement is 60/40 [3] in the favor of the front part of the carriage. This means that the front part of the carriage takes 60% of the whole. Because of the mentioned weight distribution, only the front part of the carriage is analyzed. Therefore load of the train body and therefore the total weight transfer on the front bogie /center plate is 174.34KN and on the rear bogie is 69.74KN. From the LRT data L_{wb} and L_{tc} is 1900mm and 15000mm respectively. This implies that the reaction force on the front bogie is higher than that of the rear bogie of the train.

3.4.4.3. Reaction Force of the Wheel

The reaction forces on the wheels of the bogie depend on the weight transfer and weight of the bogie itself. From the bogie figure given below the reaction forces on the wheel can be determined.

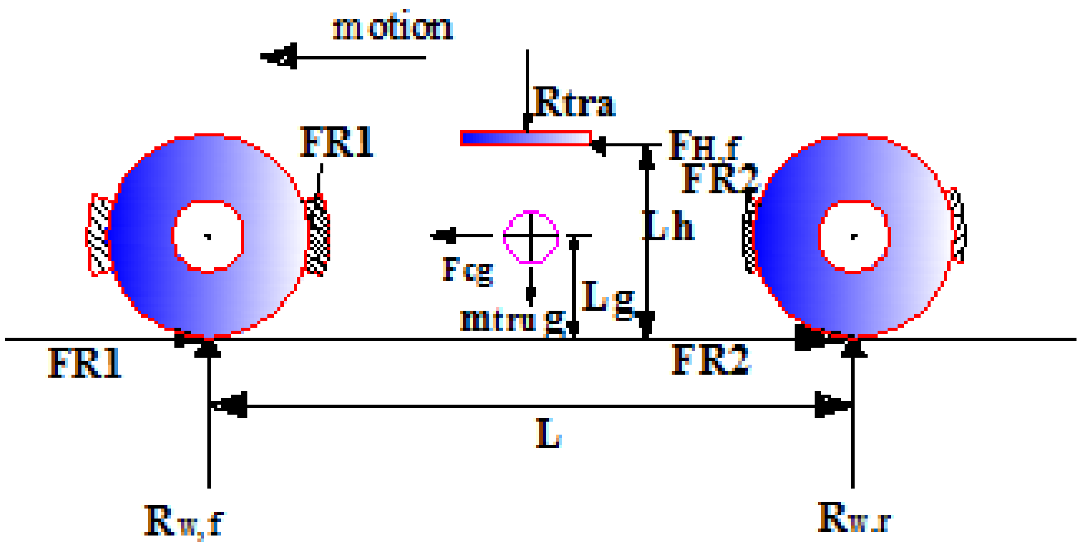

Figure 15 Reaction forces on the wheel on the horizontal track

The sum of the moments about any point of reference must be equal to zero, taking the front and rear contact point on the rail turn by turn and using as reference:

$$\frac{(R_{tra}L_{wb})}{2} + \frac{(m_{tru}\ g\,L_{wb})}{2} + F_{tcg}L_g + F_{H,r}\,L_h - R_{w,f}L_{wb}\ = 0 ------------ 3.45$$

Rearranging equation 45 and simplifying it finally we get the value of the reaction force on the front wheel as follows;

$$R_{w,f} = \frac{(R_{tra} +\ m_{tru}\ g)}{2} + \frac{(F_{tcg}L_g)}{L_{wb}} + \frac{(F_{H,r}\ L_h)}{L_{wb}} ------------------ 3.46$$

This force is the highest force of all the reaction forces on the wheel because of the higher load on the front/ leading bogie. In the same way the next reaction force can be calculated and results as follows

$$R_{w,r} = \frac{(R_{tra} + m_{tru}\, g)}{2} - \frac{(F_{tcg} L_g)}{L_{wb}} - \frac{(F_{H,r}\, L_h)}{L_{wb}} ------------------3.47$$

Comparing equations 46 and 47 the reaction force on the front wheel is higher than the rear force by the amount of the reaction force acting on the center plate and the inertia force of the bogie. The inertia force acting on the bogie due the motion of the train will be calculated based on Newton's second law of motion and for the train moving on the straight track it is expressed as the following;

$$F_{tcg} = m_{tru} a - F_{H,r} ----------------------3.48$$

When the train is moving on the gradient up or down the inertia force equation is changed by the amount of the value of the grade. This value either increase or decrease depending on the grade direction. So that the general equation of the inertia force for this case is

$$F_{tcg} = m_{tru} a - F_{H,r} - (R_{tra} - m_{tru}\, g)\sin\delta ------------3.49$$

The value of these reaction forces is different for all stations due different gradients of each and every station

Since F_H is 12.76KN and weight of the truck is 3.2tone (31.392KN) the inertia force on the bogie is about 23.86KN on the straight track and 28.4KN on the maximum gradient and it is opposite to the direction of the reaction force on the plate.

Then the above equations will be rewritten as follows

$$R_{w,f} = \frac{(R_{tra} + m_{tru}\, g)}{2} + \frac{(m_{tru} a - F_{H,r})L_g}{L_{wb}} + \frac{(F_{H,r}\, L_h)}{L_{wb}} -------------3.50$$

$$R_{w,r} = \frac{(R_{tra} + m_{tru}\,g)}{2} - \frac{(m_{tru}a - F_{H,r})L_g}{L_{wb}} - \frac{(F_{H,r}\,L_h)}{L_{wb}} - - - - - - - - - - - - - - - 3.51$$

When the train is moving on the gradient down ward the reaction force equation changes as

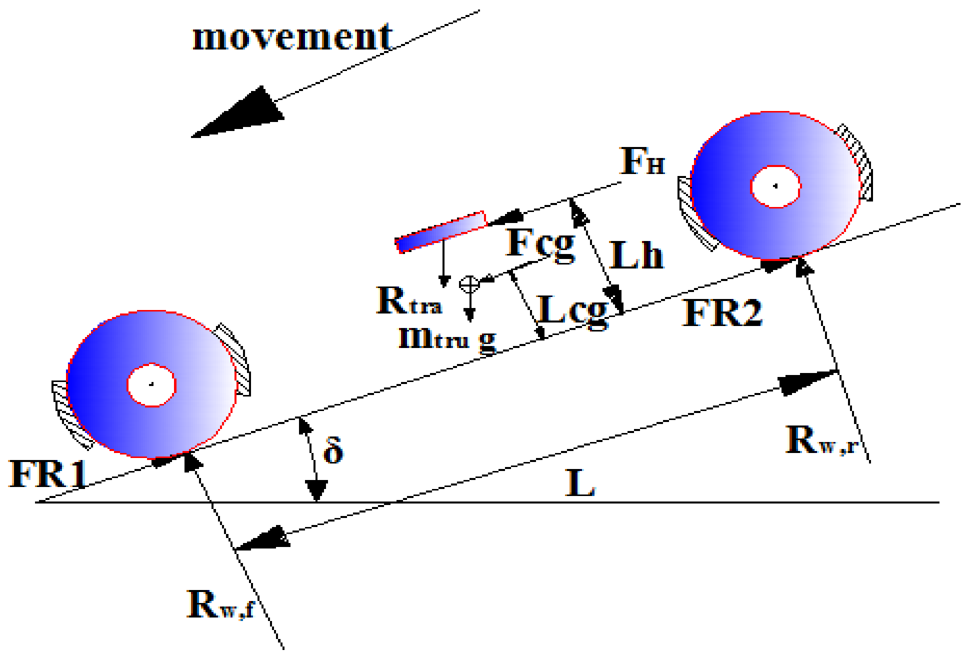

Figure 16 Reaction forces on wheels when moving on the gradient

Taking the moment at the two wheel contacting point at the two points alternatively for reach wheel and equating them to zero, we can get the reaction forces on the two wheels as expressed in the following equations;

$$R_{w.f} = \frac{(R_{tra}L_h - m_{tru}gL_{tcg})}{L_{wb}}\sin\delta + \frac{(R_{tra} + m_{tru}g)}{2}\cos\delta + \frac{F_H\,L_h}{L_{wb}} - \frac{F_{tcg}\,L_{tcg}}{L_{wb}} - - - - 3.52$$

And

$$R_{w.r} = \frac{(R_{tra} + m_{tru}g)}{2}\cos\delta + \frac{(R_{tra}L_h - m_{tru}gL_{tcg})}{L_{wb}}\sin\delta + \frac{F_{tcg}\,L_{tcg}}{L_{wb}} - \frac{F_H\,L_h}{L_{wb}} - - - - 3.53$$

Due to the maximum heat flux entering the wheel occurs on the N-S line at NS22 station the reaction force taken for the analysis of the thermal and stress for the wheel is the force supporting the wheel at this station. Taking the data from AALRT projectL_g is 325mm,L_h is 380mm and $L_{wb}$is 1900mm for power bogie. Then inserting this data in to the above equations the reaction forces on the power bogie are $R_{w,f}$ is 73.472KN and $R_{w,r}$ is 72.605K

CHAPTER FOUR

FINITE ELEMENT ANALYSIS OF THE WHEEL

4.1 Wheel material

The feedback data to be used for the finite element analysis using ANSYS is the one determined in Chapter Two. If these values are determine correctly the output of the analysis of the wheel will be correct, unless the output value will diverge from the expected value of the software. Therefore the first analysis is to gain the correct predetermined value of the wheel heat flux from the energy equations, the ambient temperature of the train after brake which is the residual temperature from the dissipated value when it starts moving from the previous station (i.e. station NS21), the convective heat transfer coefficient, the centrifugal force the wheel moving forward assuming axle as the support of the wheel or axis of rotation and the reaction force from the rail when braking at the station.

The proper selection of the wheel material is one of the major criteria to get good performance of the material and perfect analysis. Many researchers select the material of the wheel to be steel. Among these the German institute of the railway [45] proposes some standard steel category for manufacturing and its material properties. According to this standard UIC 812-3 and EN 13262 the wheel material of the steel category R6 T, E/ER which are listed in the table below.

Table 6 Wheel steel requirement according to UIC 812-3 and EN 13262

Steel category		Carbon content (%)	Yield strength (N/mm²)	Tensile strength (N/mm²)	Elongation (%)	Notch impact energy (J)	
UIC 812-3	EN 13262	UIC/EN	EN 13262	UIC/ EN	UIC/ EN	UIC 812-3 U-notch (RT)	EN 13262 V-notch (-20°C)
R1 N	-	≤ 0.48	-	600...720	≥ 18	≥ 15	-
R2 N	-	≤ 0.58	-	700...840	≥ 14	≥ 10	-
R3 N	-	≤ 0.70	-	800...940	≥ 10	≥ 10	-
R6 T, E	ER6	≤ 0.48	≥ 500	780...900	≥ 15	≥ 15	≥ 12
R7 T, E	ER7	≤ 0.52	≥ 520	820...940	≥ 14	≥ 15	≥ 10
R8 T, E	ER8	≤ 0.56	≥ 540	860...980	≥ 13	≥ 15	≥ 10
R9 T, E	ER9	≤ 0.60	≥ 580	900...1050	≥ 12	≥ 10	≥ 8

4.2 Wheel modeling

Before performing any type of analysis the solid modeling of the material is mandatory using solid modeling software. It is possible to use more than one software at a time to achieve the simplified operation and to ignore the complexity of the modeling process. But this have a defect on the performing the analysis any solid component. During this work the ANSYS work bench is for modeling the wheel of the LRT train and the analysis is done in the same way on this work bench on different window.

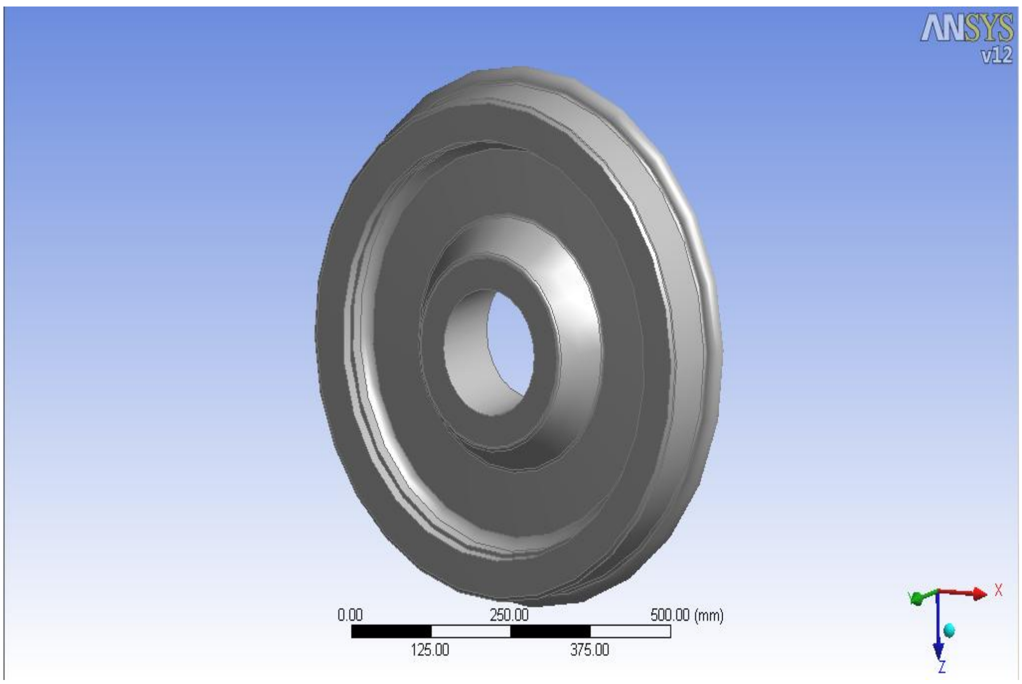

Figure 17 Wheel model using ANSYS work bench

4.3 Meshing

Meshing is discretizing the solid object to finest parts to perform the analysis to get the precise value at each and every elements of the meshed object. Since the wheel a 3D element the appropriate meshing method will be volume mesh so that all the volume of the wheel is discritised to the smallest part of the wheel. The resulting volume mesh is automatic and the output of the mesh consists of 25975 nodes and 14991 tetrahedral elements each approximately 3 mm big. After this the object is ready to be analyzed by setting the analysis on the ANSYS work bech.

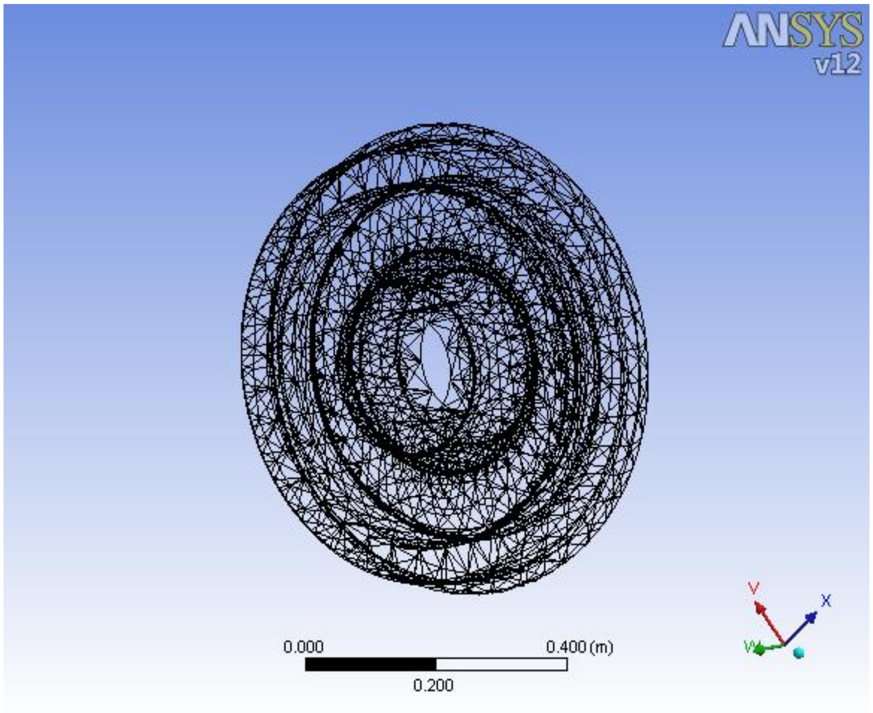

Figure 18 Meshed wheel

4.4 Analysis setting

Two steps are required for solving the ANSYS analysis for thermal and stress analysis. The first is to solve the temperature rise and heat flux distribution over the wheel tread and the second is to set the structural properties of the wheel. The thermal analysis uses transient thermal analysis and the resulting temperature rise from this analysis is used as an input for the total stress on the wheel. Then since the train is on motion the structural transient analysis is used to solve the structural analysis of the wheel. The general analysis setting is shown in the following figure.

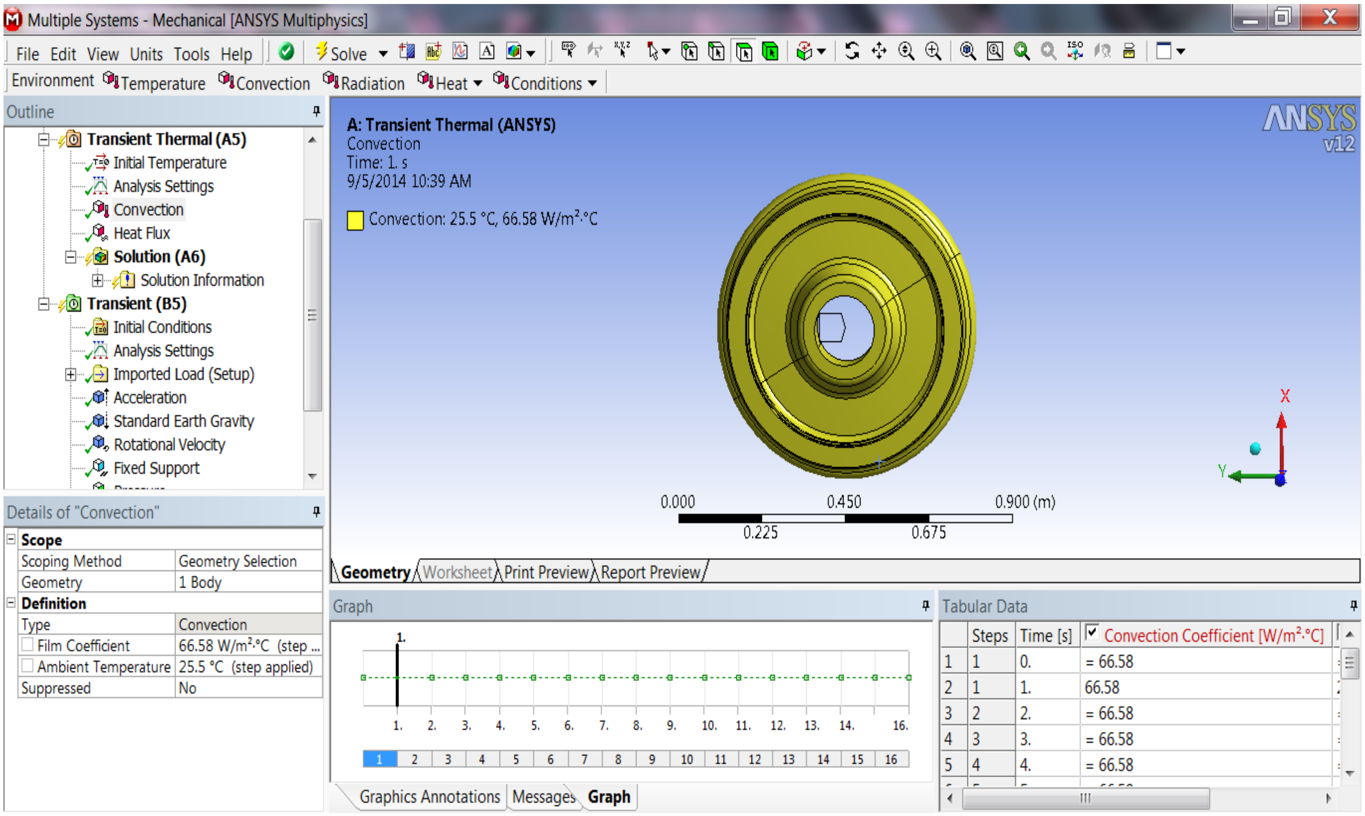

Figure 19 Analysis setting

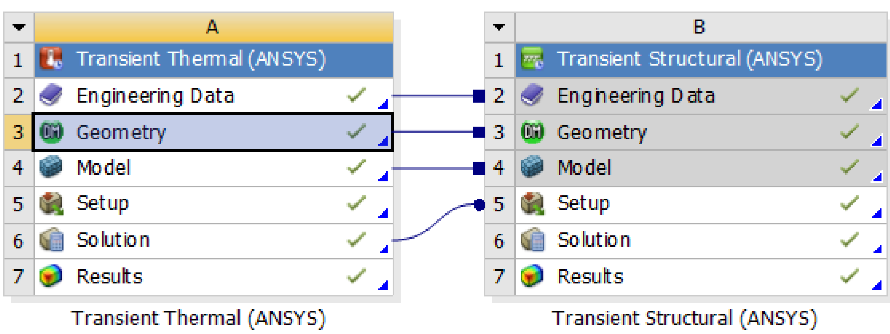

Figure 20 Stepwise solving of ANSYS

CHAPTER FIVE

RESULTS AND DISCUSSIONS

5.1 The input data for the ANSYS

To obtain the correct result the data needed for this analysis should be intruded correctly. These data are different for the two contact angle of the wheel

.

Table 7 Input data

No	Input	Value	
1	Initial temperature	84.566°C	
2	Maximum heat flux	4035685.617W/m^2 for θ=30°	2714716.387W/m^2 for θ=45°
3	Convective film coefficient	66.58W/m^2/°C	
4	Centrifugal force	269100N	
5	Reaction force on the front wheel	124291N	
6	Acceleration	1.5m/s^2	
7	Deceleration	1.1m/s^2	
8	Rotational speed	60rad/sec	
9	Gravitational acceleration	9.81m/s^2	
10	Pressure due to brake force	303KPa for θ=30°	247.5KPa for θ=45°

5.2 Thermal results

5.2.1 Temperature rise at the tread of the wheel

Te profile of the wheel is conical with 1/20 therefore the more affected area of the wheel is the area with the lower radius which is tip of the wheel. But the maximum temperature rise occurs at the middle of the contact between wheel and the tread block due the critical contact area is on center point of the tread of the brake shoe. Since the maximum energy generated by the train is at the station on the north-south line at the station NS22 the maximum temperature rise also occurs at this station. The initial temperature of the wheel and the surrounding (environmental temperature) is 25.5°C. When coming from station NS21 the wheel cools down to557.58℃ and 366.88℃ for the wheel contact of 30° and 45°. The temperature of the surrounding is still 25.5°C. In case of braking the train at NS22 station the highest temperatures reach up to 1001.9°C for the contact angle 30° and 662.8℃ for 45°. This temperature is reached at a time period of 16 s.

The area directly beneath the braking pad carries the main burden. This is also the place where the highest temperatures are achieved. The figures show how the temperatures toward the wheel rim fall. This information is needed to determine the influence of the heat flux on the wheel tread.

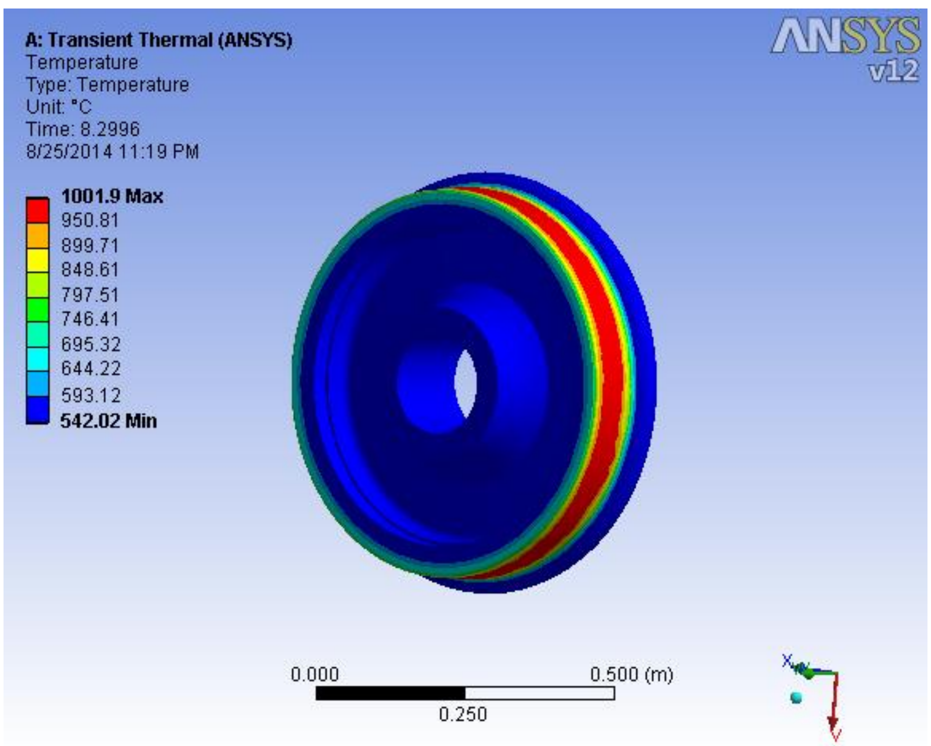

Figure 21 Maximum temperature rise at the tread of the wheel for $\theta = 30°$

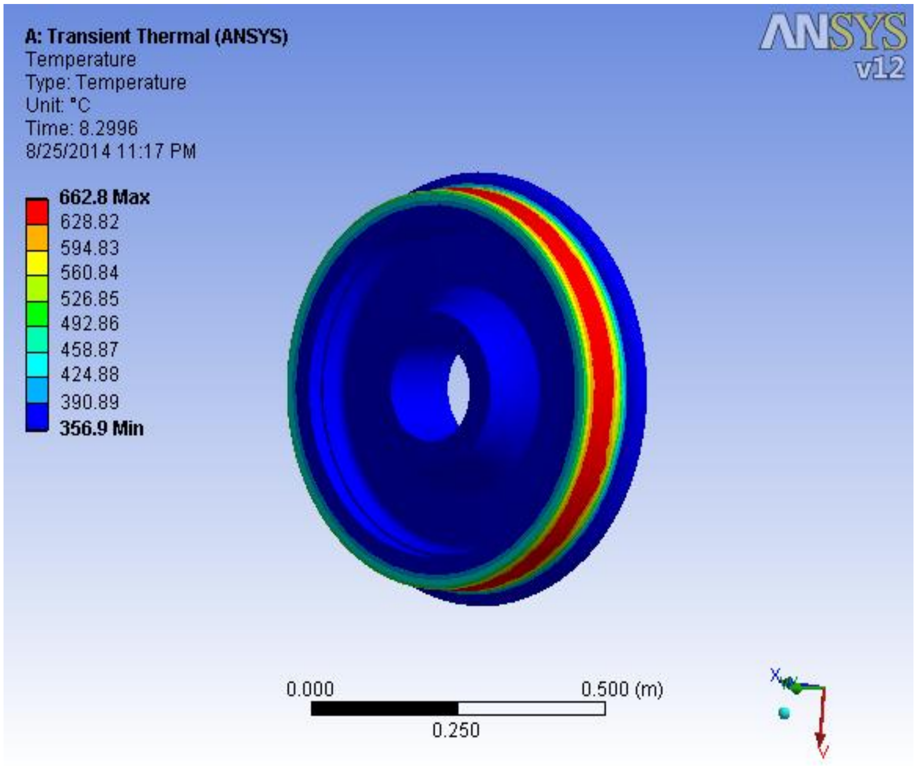

Figure 22 Maximum temperature rise at the tread of the wheel for $\boldsymbol{\theta = 45°}$

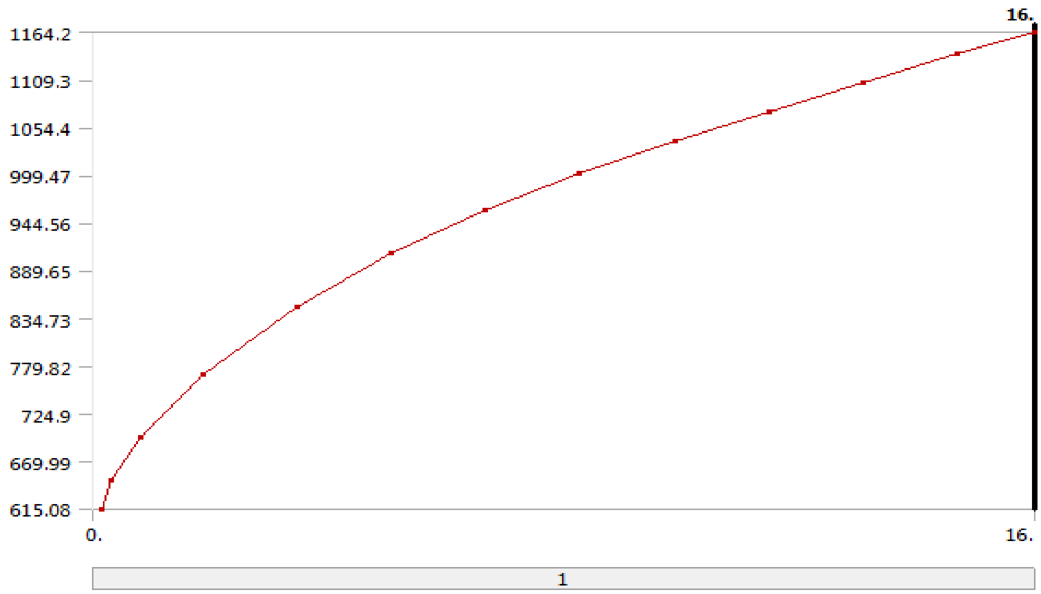

Figure 23 Temperature rise graph for $\boldsymbol{\theta = 30°}$

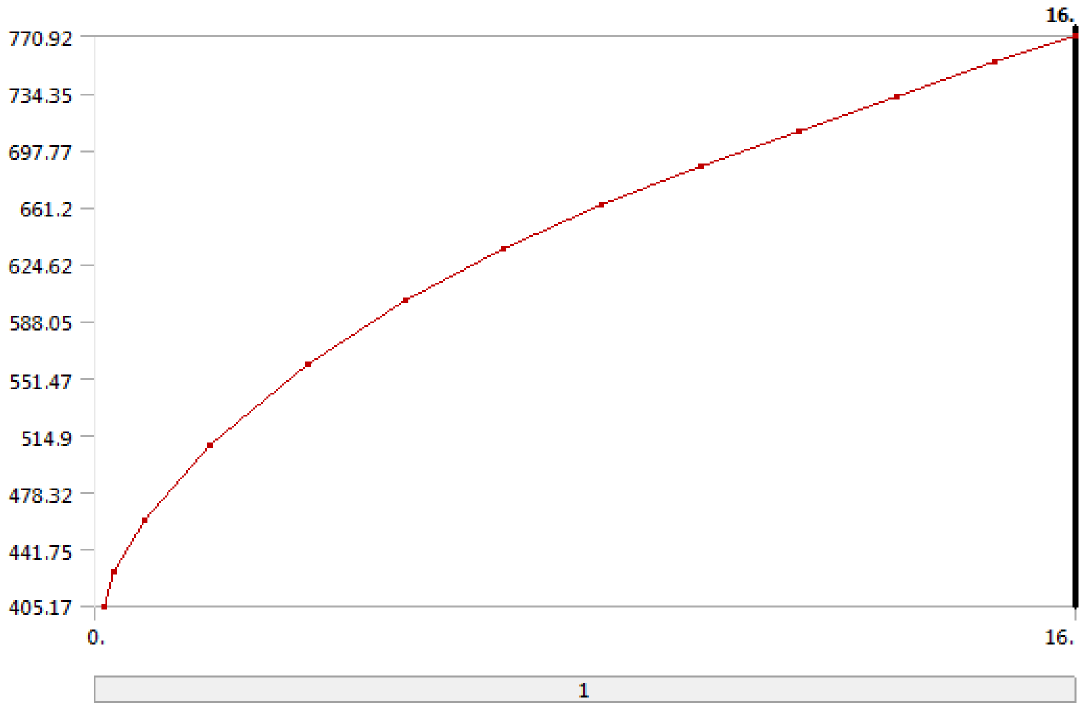

Figure 24 Temperature rise graph for $\boldsymbol{\theta} = \mathbf{45}°$

5.2.2 Heat flux distribution on the surface of the wheel tread

As in the case of temperature rise the maximum heat flux occurs at the tread of the wheel due the contact between wheels and tread block. The difference is the maximum heat flux distribution occurs at the end point of the wheel due to its conical with lower diameter of the wheel tread at this point. Since the heat flux is directly related to the temperature the maximum heat flux occurs at the station on the north-south line at the station NS22. There for from the analysis the maximum het flux on the wheel tread is about $2.1077*10^6 W/m^2$ for $\theta=30^o$ and for $\theta=45^o$ reaches about $1.6504*10^6 W/m^2$.

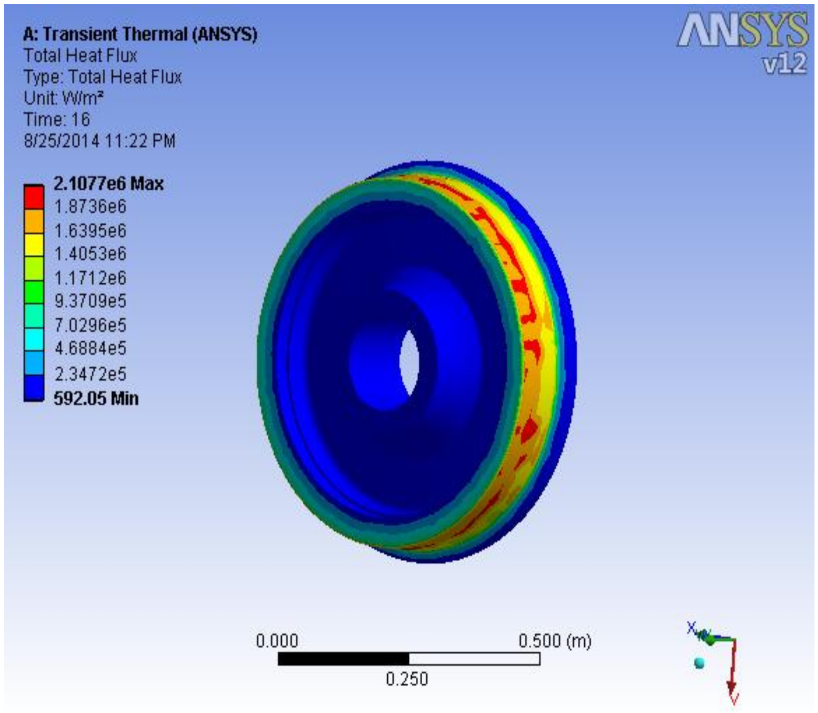

Figure 25 Heat flux distribution on the wheel treads for $\boldsymbol{\theta} = \mathbf{30}°$

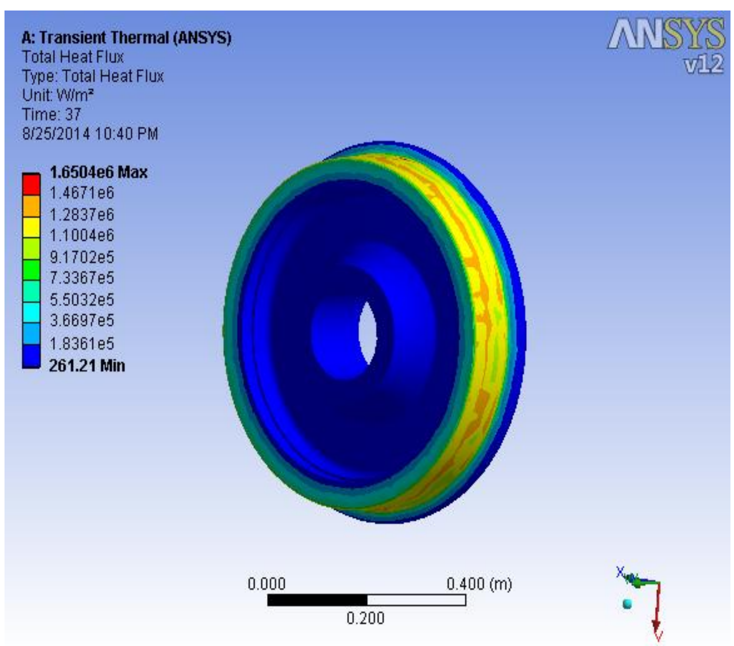

Figure 26 Heat flux distribution on the wheel treads for $\boldsymbol{\theta} = \mathbf{45}\boldsymbol{o}$

5.3 Imported body temperature

The body temperature raised as a result of braking is imported to the structural transient analysis for further use to determine the stress and deformation. Different result of temperature for both cases (i.e. 30^o and 45^o angle) result in different stress and deformation in addition with the loads on the wheel due to braking and weight transfer of the vehicle which is constant throughout the motion of the train.

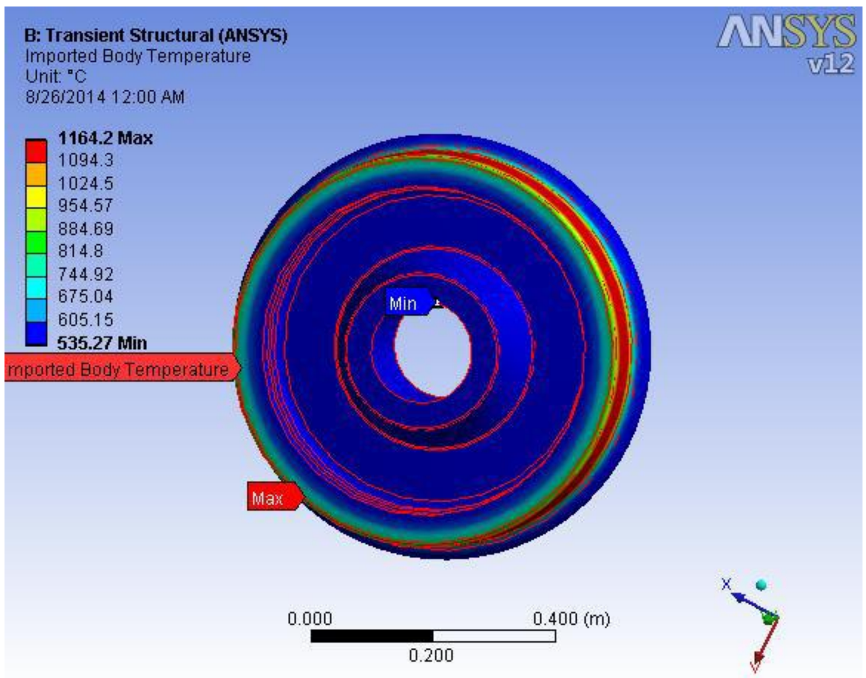

Figure 27 Imported body temperature for $\boldsymbol{\theta = 30°}$

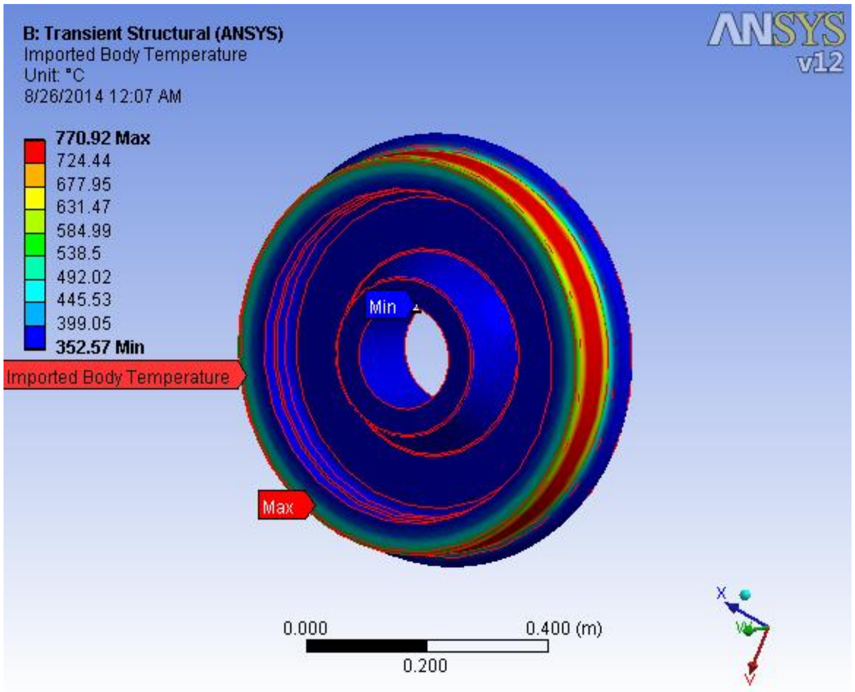

Figure 28 Imported body temperature for $\boldsymbol{\theta = 45°}$

5.4 Mechanical results

5.3.1 Transient structural

Thermal stresses in the wheel appear because the temperatures rise. Beside the thermal stress, the centrifugal load and the holding force of the brake block is also considered. The goal of this analysis is to determine the influence of the centrifugal load in comparison with the thermal load. The comparison stress is given on von Moses. The resulting temperature from the transient thermal analysis is imported to the transient structural analysis. This maximum temperature rose when the train stopped is used together with other mechanical loads calculated in chapter three to determine the maximum equivalent (Von messes) stress imposed on the wheel.

From the output result of the analysis the minimum equivalent (Von messes when the train starts braking is $2.548*10^7$N/m^2 and the maximum equivalent (Von messes) stress induced on the wheel is about $2.6028*10^9$N/m^2 θ=30^o and $1.5044*10^6$ and $1.6675*10^9$ for 30^o respectively.

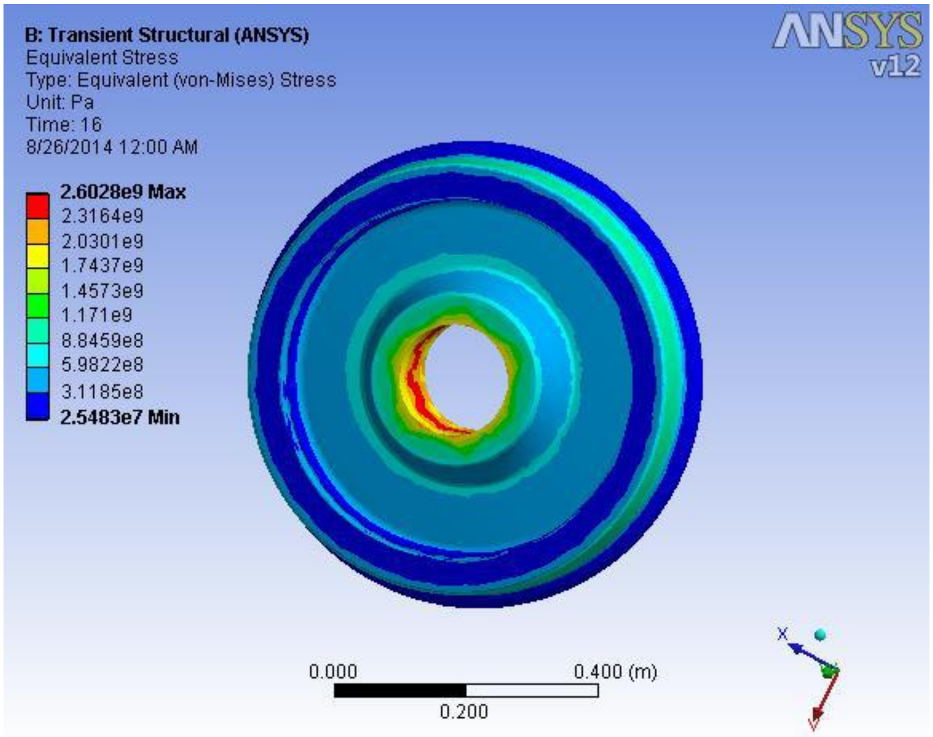

Figure 29 Equivalent (Von-mises) stress for $\boldsymbol{\theta = 30°}$

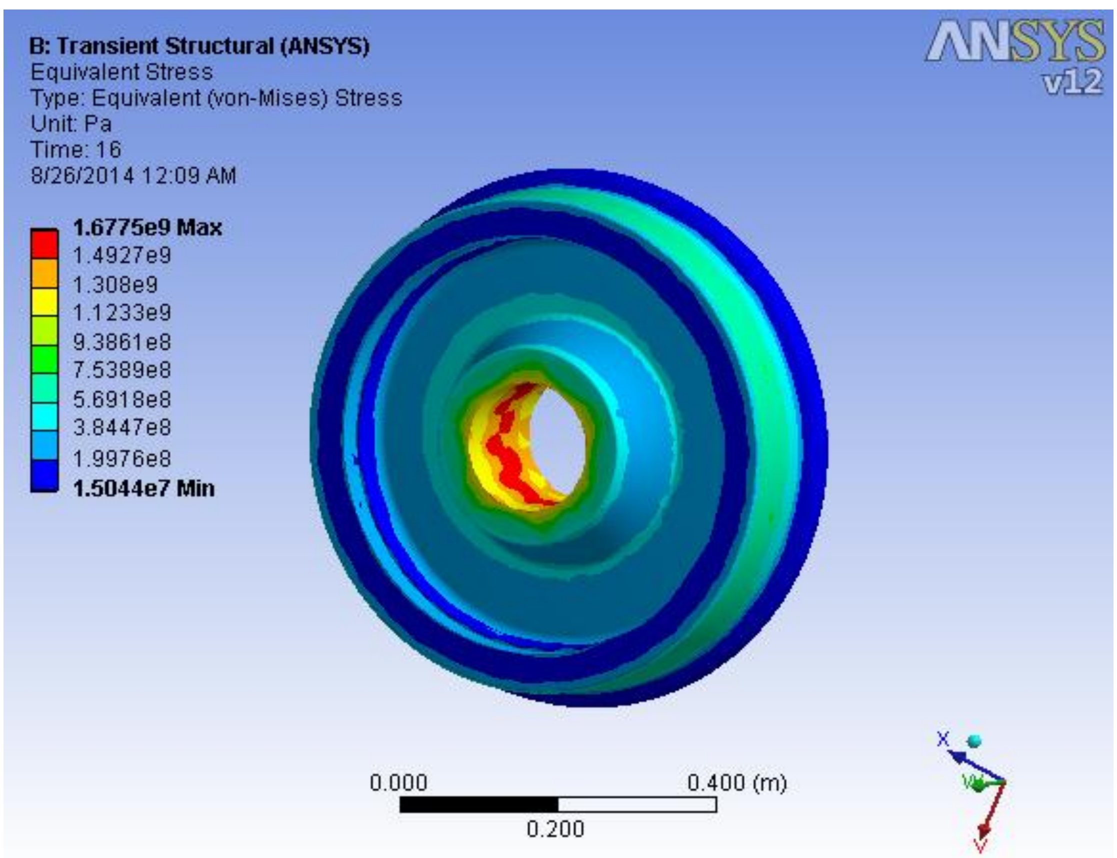

Figure 30 Equivalent (Von-mises) stress for $\boldsymbol{\theta = 45°}$

When the railway train starts moving from rest the environmental or ambient temperature of the train component is the same as the environmental temperature. As the train starts moving the kinetic energy of the train increases and gets maximum at the maximum speed. As the train moves up the grade the its potential energy decreases and vise versa while moving up the gradient. Since the total energy reaches maximum at maximum speed, its temperature reaches some value because there is no heat dissipation from the train but this temperature is the maximum temperature while the train is moving. When the driver starts to brake the train the, heat dissipation from the train to the environment stars to rise and reaches maximum when the train fully stops.

5.3.2 Total deformation and strain

The thermal and mechanical loads on the wheel do not induce only stress on the wheel but also deforms the wheel surface. This deformation may also result in the other defects such as wheel tread crack, wear of the wheel surface and other defects.

According to the analysis result the maximum and the minimum resulting deformation are 0.0025887m and 0m respectively for θ=30^o and for θ=45^o it is 0.0017109m and 0m respectively.

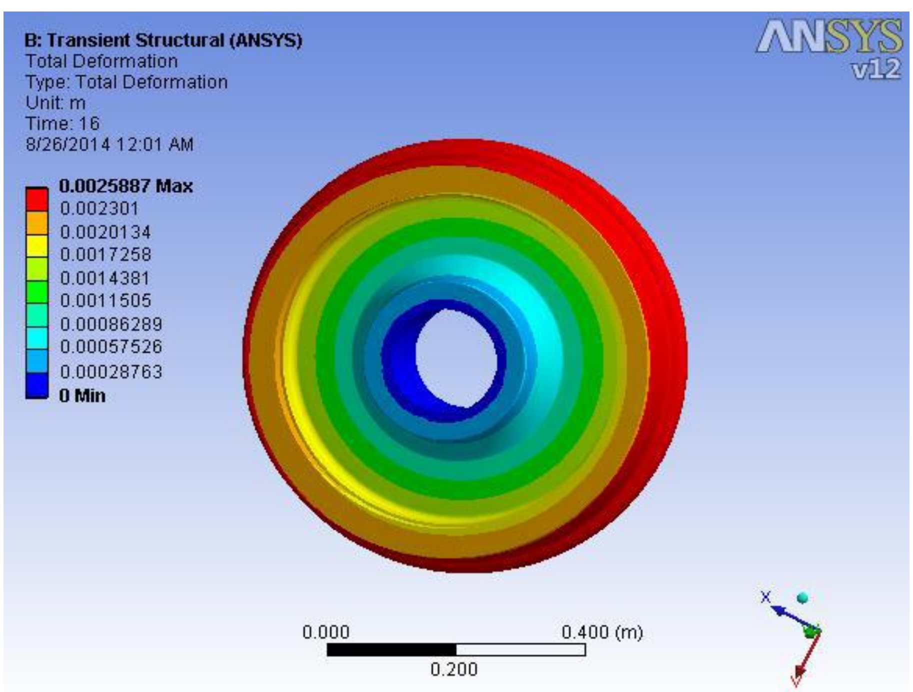

Figure 31 Total deformation of the wheel surface for $\boldsymbol{\theta = 30°}$

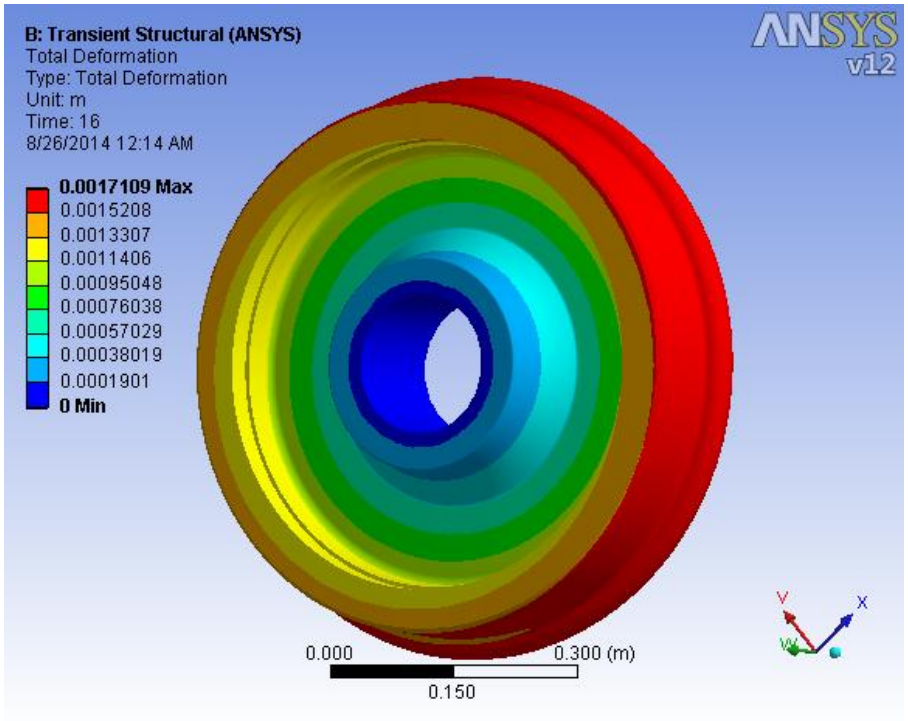

Figure 32 Total deformation of the wheel surface for $\boldsymbol{\theta = 45°}$

The strain of the railway wheel is the ratio of the total stress to the deformation of the wheel. This result given for by the ANSYS result in the figure below. According to the output result of the analysis the maximum and minimum result of the strain for $\theta=45^o$ is 0.013014 and 0.00012742 respectively and for $\theta=30^o$ the resulting total strain is about 0.0083873 and $7.5218*10^{-5}$ respectively.

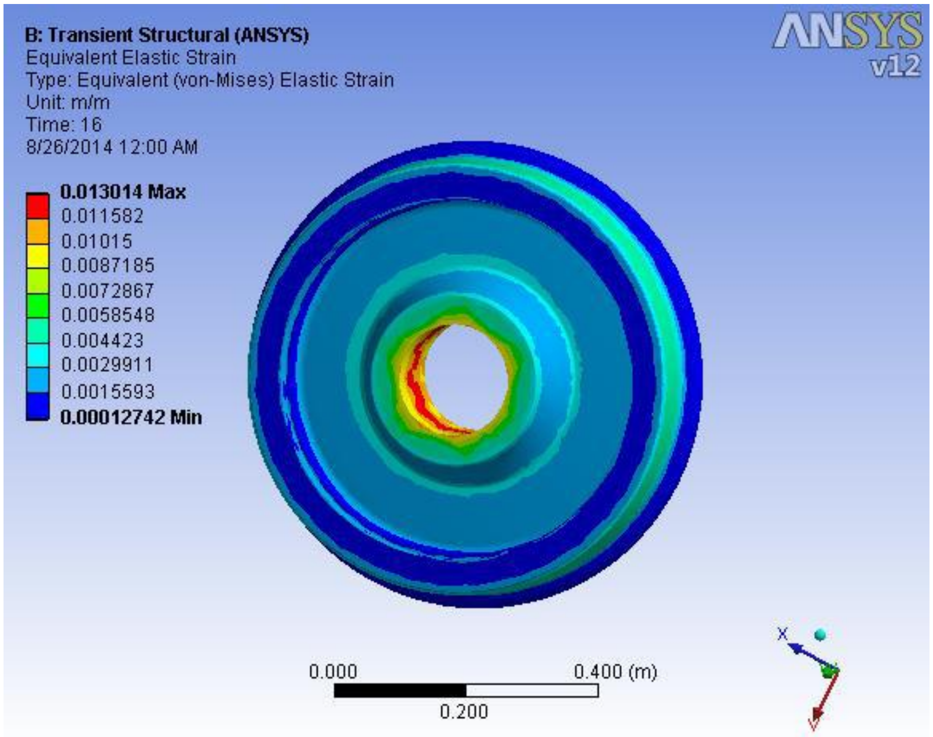

Figure 33 Total strain of the wheel of the wheel for $\boldsymbol{\theta = 30°}$

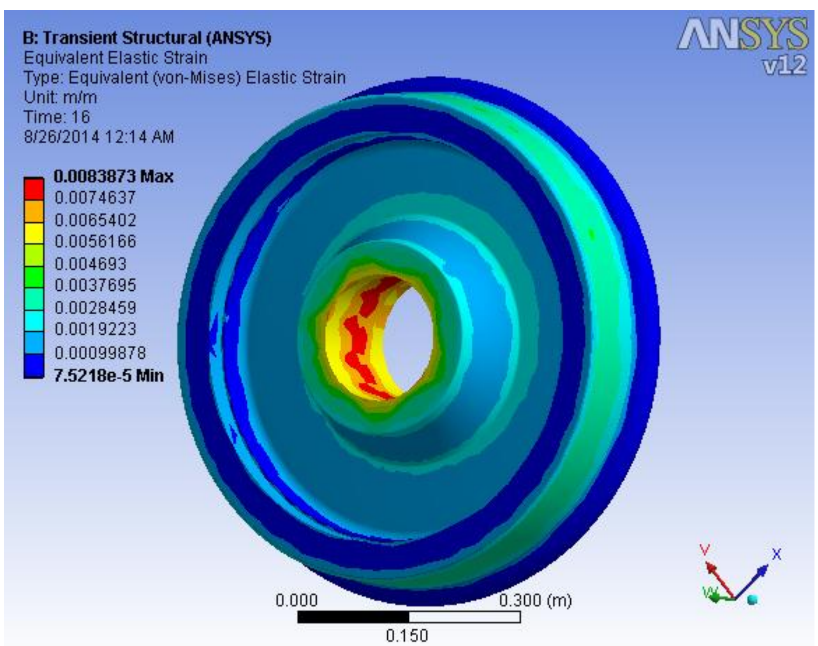

Figure 34 Total strain of the wheel of the wheel for $\boldsymbol{\theta = 45°}$

CHAPTER SIX

CONCLUSION AND RECOMMENDATION

6.1 Conclusion

From this analysis we conclude that as the speed of the train increases the braking time of the train heat flux on the train increases and as a result the rise in the temperature of the tread wheel tread block increases. The increase of contact angle between the tread block/shoe brake and the wheel tread also increases the temperature rise heat flux distribution over the surface of the wheel tread. The gradient of the truck has the negative effect on the heat dissipation from the contacting surface of the wheel tread and brake shoe.

As the reaction forces of the support from the rail on the wheel from below increases the stress induced on the surface of the wheel increases. The maximum deformed and stressed part of the wheel from these input parameters is the flange of the wheel. This result from the cumulative effect of the thermal and reactions loads applied on the wheel tread.

The dimensions of the wheel base distance, the height of the center plate from the top rail and the center of gravity of the truck affect the reaction forces on the wheel tread which increases or decreases the deformation and stress induced on the wheel.

Generally; the longer the distance between truck center, the less the effect on the weight transfer for a given car weight and center of gravity and the lower the deformation and the stress on the wheel tread. The shorter the height of the center of gravity above the center plate the less the effect on the weight transfer of the and the lower the deformation and the stress on the wheel tread of the rail way vehicle.

As stated in [19] the maximum temperature rise at the wheel and tread block interface is about 750℃ and from the analysis the maximum temperature at the interface is662.8℃. So that the result the analysis is valid for this case.

6.2 Recommendation

From the analyses conclusion I recommend the following.

- Even though cost wise using tread brake with lower angle of contact between wheel and tread block is advantageous it has the ability to raise the temperature of the wheel tread. This affects the life of the wheel and the tread block. But since the temperature of the wheel tread decreases with increasing the contact angle it is better to use tread block with higher contact angle.
- The 45^o contact angle is the optimum value to use because the temperature rise as this angle doesn't affect severely the maximum operating temperature of the wheel.
- Minimizing the running speed of train is also on solution for minimizing the temperature rise at the tread of the wheel and the thermal stress induced on the wheel treads due this temperature.
- Using the wheel with larger radius to decrease the heat flux distribution over the tread of the wheel.
- Using disc brake due to non contact between disc and wheel tread while braking the train.
- Using wheel material with high thermal resistivity.
- To decrease the deformation and stress on the wheel is increasing the bogie center distance to reduce the amount of the reaction forces on the wheel.
- Not overloading the train to lower the weight transfer while braking.
- Designing the stations to be with minimum gradient as much as possible.

6.2.1 Future Work

The future work of this paper is that under this research paper the thermal and stress analysis between wheel and tread block is done. In the future the analysis can be performed including the effect of the rail on the wheel on both thermal and stress cases. As soon as the data from AALRT is fully achieved this research paper can updated with this data.

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