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ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Thermo-mechanical analysis of friction brake block for freight train A.A-Djibouti main line

A thesis Submitted to the Graduate School of Addis Ababa University in Partial Fulfillment of the Requirements for the Degree of Masters of Science in Railway Engineering (Rolling Stock)

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I, the undersigned, declare that this thesis with **Thermo-mechanical analysis of friction brake block for freight train AA-Djibouti main line** is my original work and has not been presented for any masters program (degree) in any university and all the sources of materials used for the thesis have been duly acknowledged.

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Abstract

The main purpose of this study is to analyze the thermo-mechanical behavior of the contact between the brake block and wheel during the braking time and at two different contact angle brake block. The simulation strategy is based on the commercially available FEM software packages, ANSYS and CATIA modeling. The model is simulated using a 3D brake block in order to observe the surface temperature distributions profile for applied braking condition using brake block materials. The temperature rise and the loads are formulated analytically. The 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 brake block. Therefore, the thermal analysis of a block-braked material on a freight train of the national railway operator Ethiopian Railways using analytical and numerical modeling of thermal effects during braking and in one wheel side have two brake blocks the highest temperature are after emergency braking was 459.1℃ contact surface area 30^o and 245.2℃ at contact surface area of 45^o, which occurred at 9.48second and service braking, was 412.86℃ contact surface area 30^o and 222.45℃ at contact surface area of 45^o, which occurred at 12.1second. In one wheel side have one brake block the highest temperature after emergency braking was 899.49℃ contact surface area 30^o and 471.68℃ at contact surface area of 45^o, which occurred at 9.48second and service braking, was 793.66℃ contact surface area 30^o and 423.44℃ at contact surface area of 45^o, which occurred at 12.1second.

The maximum deformed and strain to a part of the brake block (shoe) result from cumulative effect of the thermal and pressure load applied on one wheel side have two brake blocks during emergency brake the deformation are 0.0000339m for $\theta=30^0$ and 0.0000204m for $\theta=45^0$. During service brake the deformation are 0.0000283m for $\theta=30^0$ and 0.0000188m for $\theta=45^0$. In one wheel side have one brake block the maximum deformed and strain to a part of the brake block (shoe) during emergency brake the deformation are 0.0000690m for $\theta=30$ and 0.0000439m for $\theta=45$ and the maximum strain for $\theta=30^o$ is 0.01024 and for $\theta=45^o$ is 0.00673. During service brake the deformation are 0.0000568m for $\theta=30^0$ and 0.0000412m for $\theta=45^0$ and the maximum strain for $\theta=30^o$ is 0.00842m and for $\theta=45^o$ is 0.0060m. In one wheel side have two brake blocks the maximum equivalent (Von mises) stress induced on the brake block during emergency brake is about $5.3568*10^8 N/m^2$ (535.68Mpa)for $\theta=30^0$ and $3.4325*10^8 N/m^2$ (343.25Mpa) for $\theta=45^0$ respectively and during service brake is about $4.5954*10^8 N/m^2$ (459.54Mpa)for $\theta=30^0$ and $3.0735*10^8 N/m^2$ (307.35Mpa)for $\theta=45^0$. In one wheel side have one brake block the maximum equivalent (Von mises) stress induced on the brake block during emergency brake is about $1.1264*10^9 N/m^2$ (1126.4Mpa) for $\theta=30^0$ and $7.402*10^8 N/m^2$ (740.2Mpa)for $\theta=45^0$ respectively and during service brake is about $9.2645*10^8 \mathbf{N}/m^2$(926.45Mpa)for $\theta=30^o$and $6.6075*10^8$ (660.75Mpa)for $\theta=45^o$.

Key words: rail way, brake block, temperature rise

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Nomenclature

$v(t)$ - velocity of the freight car at function of time

v_o - Initial running velocity of the freight car

ω_o - Initial angular velocity the wheel

ω - angular velocity of the wheel at any time

t_b - braking time

S_b - total brake distance

a -Acceleration

ΔE_p- Potential energy

m - Mass of the vehicle

g - Gravitational acceleration

h – Height

E_b- Total braking energy

F_t - Total braking force

n_1 - Initial angular velocity

n_2 - Final angular velocity

t_b - Braking time

μ - coefficient of friction between wheel and brake block (shoe)

A – Contact area between wheel and brake block

l - Circumferential length of the block

b- Width of the block

θ – Brake block contact angle

$p(\theta)$- Pressure per contact area

F_A- brake force per contact area

R- radius of wheel

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

Q- Rate heat transfer

h - Convection heat transfer coefficient

T_s - Surface temperature

T_∞- Ambient temperature

$\mathcal{E}$- 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

U - Over all heat transfer coefficient

q_{diss} - Heat dissipation

q_{cond} - Heat conduction between wheel and brake block

Chapter-One

1.1Background

1.1.1History of railway

By the standards of most modern industries railways have unusually deep historical roots. Railways that fit Lewis's definition existed as far back as the 6th century BC; the Greek *Diolkos*wasa railway with a track made from stone, 6km in length across the Peloponnese, usedfortransportingshipsuntilthe9thcentury AD–an extraordinarily long period. Works such as Agricola's *De Re Metallica* date the extensive use of railways with wooden rails and vehicles to around the 15thcentury.Although of great technical interest, individual systems had short lives and were of no significance as anything other than adjuncts to the mining industries. By the 18th century, however, wooden railways began to be used for larger loads and more diverse purposes. Railways developed from mine tracks where people pushed four- wheel trucks of coal, stone, or ore in to longer and more complex lines with large wagons and horse haulage. Late in the same century, the change was made in many places to iron rails and wheels. Wooden and stone rail ways did not immediately disappear, however; indeed, in Britain, the HayTor Tram road was built with stone 'rails' at the late date of 1820 [1].

Leaving aside these very early lines, we can date the mechanically worked railway to the first two decades of 19th century England and Wales. These short isolated routes, just a few miles (kilometers) long, were still usually conceived, financed, built, and operated with the needs of a small number of extractive and primary industries in mind. They shared little beyond a very basic technical similarity with today's rail ways. However, they rapidly developed in length, volume of traffic, technical sophistication, and financial and managerial requirements. Most historians agree that with the opening in1830 of the Liverpool &Manchester Railway in the north-west of England, the prototype of the 'modern railway' had arrived: a combination of specialized track, the accommodation of public traffic, the conveyance of passengers as well as freight, mechanical traction, and some measure of public control.

(Robbins1998).Conceived principally as a competitor to the carriage of goods by canal, the line tapped a hit her to unsuspected demand for passenger travel. Profits were very consider able and in the generally vibrant economic conditions of the world's first industrial nation, the railway swept all before it as a means of in land transport over distances of any length.By1850, Britain enjoyed the benefits of a national network linking most of the Centre's of population and industry. British engineers rapidly gained employment across Europe, building many of the continent's earliest and most important railways (Gourvish1996; Channon 1996; Ambler 1999). By 1907 there were about 200,000 miles (320,000km) of railways in Europe (Robbins 1998) [1].

The place of the railway in the history of industrialization is assured. Economic historians might disagree over the precise contribution that railways made to economic growth in the industrializing nations of the 19th century, but all recognize the steam railways' critical role as the dominant form of inland transport for any but the shortest of journeys (e.g Szostak 1991; Ville 1990). The railways' advantages of speed, capacity, and economy made them more than mere instruments of industrial and business development, however. Culturally their impact was huge. In particular the sensibilities of societies that had never known travel at speeds above that of a galloping horse were irrevocably changed by the coming of steam locomotion. The first known electric locomotive was built in 19th century (1837) by chemist Robert Davidson, while the first diesel locomotives were built at the beginning of the 20th century [1].

1.1.2History of Ethiopian railway

Addis Ababa owed much of its importance to its geographical position on the Horn of Africa, and, in particular, to its relative ease of access to and from the French-occupied Gulf of Aden Port of Djibouti, which handled a significant portion of Red Sea and Indian Ocean trade. It was indeed on account of this access that Menilek granted his first railway concession on 11 February 1893. It was very different in scope from the line which eventually emerged. It envisaged the establishment – and the grant to Menilek's Swiss technical aide and diplomatic adviser Alfred Ilg - of a 99 year monopoly for a railway, conceived as operating on three separate but related stretches of line: The first was to run from the French-administered port of Djibouti to Harar, along the principal commercial centre of eastern Ethiopia. The second stretch was to go from Harar to Entoto, the country's former capital which had only been replaced as capital by Addis Ababa a few years earlier. The third stretch was to go from Entoto to the White Nile, which Menilek then claimed as his Western frontier.

Construction work on the railway began in October 1897, with the erection of the Djibouti railway station, offices, stores, staff-housing, and other buildings. This coincided with the shipping in of rails, sleepers, telegraph poles and cement. The sleepers, sometimes referred to in the trade as Menilek's, were for reasons of economy no more than a metre long and were unusually made of iron to protect them from being eaten by termites There was also a large influx of both skilled and unskilled labor, from many countries. This included local Oromos, Afars and Somalis, as well as Arabs, Armenians and Indians, besides Europeans: Greeks, Italians, and Frenchmen.

Many transport camels were also acquired. The first commercial service began 1901, from Djibouti to Dire Dawa.By 1915 the line reached Akaki, only 23 kilometers from the capital, and two years later came all the way to Addis Ababa itself. It has finally deteriorated after serving for nearly a century due to lack of maintenance, poor management and lack of commercial focus.

In January 2010, it was announced that the Ethiopian government had signed a memorandum of understanding with four companies to build a new rail network. Among these firms was the China Communications Construction, China Railway Group, and an Indian and Russian company. After all parties except the Chinese firms failed to take further action, the contract was awarded to the CRCG. In August, funding for the project was agreed to come from the Export and Import Bank of China. In September 2010, construction began on the project.

1.2 Problem statement

Brakes is such a critical system in stopping the rail vehicle on all moving stages including braking during high speed, sharp cornering, and downgraded rail road. All of those braking moments give a different value of temperature distribution and thermal stress. Good performance of brake block comes from good material with better mechanical and thermal properties. Good designs of brake block are varying across the range of the vehicles. There are different design and performance of brake block if compared between passenger and freight train.

This project concerns of the temperature distribution and stress analysis of brake block. Most of the freight train currently have brake block that are made of gray cast iron. The investigation are to determine the temperature behavior and stress analysis of the brake block during braking by using Finite Element Analysis (FEA) in freight trains of AA-Djibouti line.

The main problem of braking and stopping a rail way vehicle is great heat load in to foundation brake components in very short time and leads to temperature rise [2].

Hence, it is important to study and predict the temperature rise of a given brake component and assess its thermal performance by using Finite element analysis (FEA) or ANSYS software.

1.3 Objective

- **General objective**

The main objective of this research is to analyze the temperature rise and the thermo-mechanical analysis and grey cast iron P10 friction brake block for freight train of AA-Djibouti line during service and emergency braking.

- **Specific objective**

To predict the temperature rise and thermal behavior of brake block by performing the following specific objectives.

- To model (simulate) of brake block.
- To perform load analysis by using heat flux and convectional heat transfer coefficient.
- To show the result of transient structure and transient of thermal analysis of brake block made of grey cast iron P10 material.
- To make the necessary conclusion and recommendation based on the results of the research.

1.4 Scope of the study

As it is seen in the problem statement main the problem during braking and stopping a rail way vehicle is great heat load in to foundation brake components in very short time which leads to temperature rise that reduce the performance of braking. The effects are analyzed by a 3D modeling and finite element method (FEM) using ANSYS software thus the scope will be achieved by predict the temperature rise of a given brake component and assess its thermal performance. This paper does not address fabrication and manufacturing process of brake block.

1.5 Significant of the study

As a research, the primary merits of the study goes to ERC future research center. Since there are a lot has to be done studies in the area, it will give a comprehensive starting point for more advanced researches. Students will be motivated for further investigation of the brake block thermal stress and material optimization. The benefit of this research can also help to discover on time the conditions for appearing temperature rise and the thermal regime developed during braking on brake shoe to remain within acceptable limits, without affecting the braking capacity or other elements of the vehicle.

1.6 Research of methodology

This research thesis use different type of research methodology in order to meet the objectives and achieved by:

Data collection

Primary as well as secondary data which is needed for the study are collected using different data collection techniques. These are:

Primary Data

- Collecting data from technical manual of ERC.
- Standard block shape and material.

Secondary Data

- Searching for thesis, published paper, journal, books, etc. on different web sites.

Analysis

Begin with a literature review, published paper, journals and books and part of it has considering in this project and understanding the CATIA software in depth to perform modeling tools. After organizing the collected data and dimensions have used to translate in 2D and 3Dmodelby using CATIA, load analysis will be done by using heat flux and convectional heat transfer coefficients. Load analysis is calculated based on the full load of the train after that the value of load analysis will be applied on finite element and then 3D model brake block has transfer to finite element software which is ANSYS work bench. Thermal Analysis will be done on steady state and transient responses. Assigning materials properties load and meshing of the model will be done in this stage. Then, completed meshing model will be submitted for analysis. Finally an expected result from steady state and transient responses of thermal analysis has obtained. A flow charts, tables and modeling thermal effects of the braking process at brake block rail way vehicle show a better understanding of a perdition of thermal effect and thermal behavior of brake block. By comparing the result, write conclusion, which is at the same time a recommendation for further work (if any) to investigate further more by other researchers later on. At this stage brief summery or abstract of the research is written after completing the research work. Acknowledge for the materials used to prepare the research or for any one who helps to facilitate your work. Finally edit the overall paper to submit.

Chapter-Two

Literature review

2 Introduction

Braking is to restrain the movement of the moving body artificially, including making its speed decreased, arresting its movement or its acceleration trend. 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 [3]. The vast majority of the world's trains are equipped with braking systems which use compressed air as the force to push blocks on to wheels or pads on to discs. These systems are known as "air brakes" or "pneumatic brakes". The compressed air is transmitted along the train through a "brake pipe". Changing the level of air pressure in the pipe causes a change in the state of the brake on each vehicle. It can apply the brake, release it or hold it "on" after a partial application. The system is in widespread use throughout the world.

In the air brake's simplest form, called the direct air (straight air) system, compressed air pushes on a piston in a cylinder [4]. The piston is connected through mechanical linkage to brake shoes that can rub on the train wheels, using the resulting friction to slow the train [4]. The mechanical linkage can become quite elaborate, as it evenly distributes force from one pressurized air cylinder to 8 or 12 wheels.

The pressurized air comes from an air compressor in the locomotive and is sent from car to car by a train line made up of pipes under each car and hoses between cars. The principal problem with the straight air braking system is that any separation between hoses and pipes causes loss of air pressure and hence the loss of the force applying the brakes. This deficiency could easily cause a runaway train. Straight air brakes are still used on locomotives, although as a dual circuit system, usually with each bogie (truck) having its own circuit.

In order to design a system without the shortcomings of the straight air system, Westinghouse invented a system where in each piece of railroad rolling stock was equipped with an air reservoir and a triple valve, also known as a control valve [5]. This allows the driver to control the brakes on all the vehicles. When the air is under pressure, the brake is released, and in the event of a leak the brakes are locked and secured.

2.1 Tread braking

Tread (block) braking is still one of the most common braking systems on railway vehicles. Block braking is the ordinarily used system on freight wagons and is also commonly used on passenger trains, often in combination with disc brakes and electro-dynamic brakes [6]. The tread braking (or block braking) action is carried out by pressing the brake block(s) against the tread of a wheel [6]. The vehicle can be equipped with four different standard block configurations. Furthermore, a brake block is generally manufactured of cast iron, organic composite or sinter materials. 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. Since block braking roughens the wheel tread, it improves the wheel-rail interface 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 [7].

A railway vehicle faces, in principle, two different types of braking modes called stop braking and drag braking. Stop braking means that the initial speed of the train is reduced. Substantially, or the train is brought to rest [8]. In the drag braking (downhill braking) case a potential increase in train speed is detained by applying the brake to the train wheel at about a constant for design and technical approval of railway wheels. The level of braking power and braking energy introduced from the friction between brake block and running surface of the wheel are different for the two modes.

On modern trains, the use of mechanical (pneumatic) braking system (block braking/disc braking) is avoided to a high degree by employing electrodynamics braking. This means that braking energy is transformed to electric energy by the traction motor and is often circulated back to power grid. This gives environmental advantage since it both reduces the required consumption energy of the train and also reduces wear of the mechanical braking systems.

2.2 Brake blocks

Brake shoe is best used on rolling stock. 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 braking shoes, forces which manifest to the brake cylinder piston rod is amplified by the wheel house of brake components and sent to break blocks which press the wheel tread.

Brake shoe is the main body of which depends largely on braking effect. Quality brake characterized by the length of braking road depends among other things by the shape, sizes and quality of the material being manufactured brake shoe [9]. The influence of these-factors is contained in the amount of friction's coefficient between shoe and wheel. Figure 1 shows schematically the formation of the braking force on the action of the brake shoe pressure, where F_f is the friction force per unit contact area, Ps is the normal braking force on the shoe, Q is the load on wheel rail, Fa normal contact force on the point of contact, $\mu_S$representing the coefficient of friction for the case of wheel and brake blocks, and R radius of the wheel. After executing a braking action, the kinetic energy of the locomotive (and possibly potential energy-flow on the slope) is consumed by the work of the drag forces and braking forces, leading to slow down.

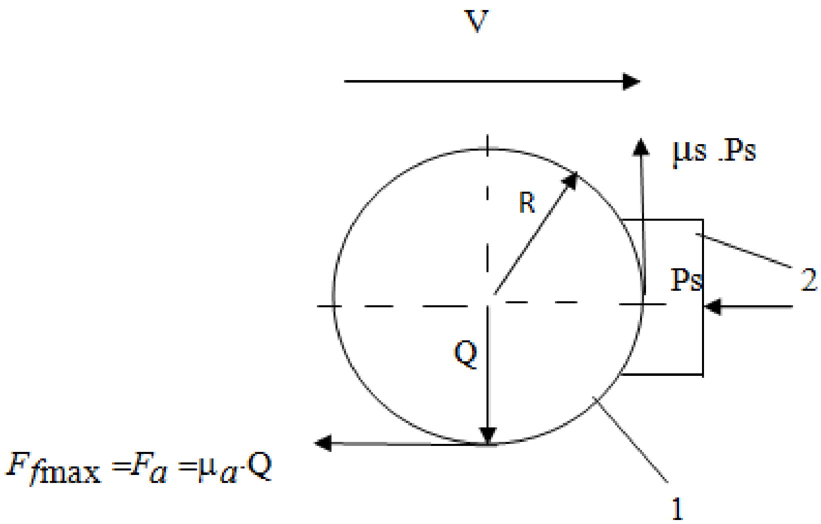


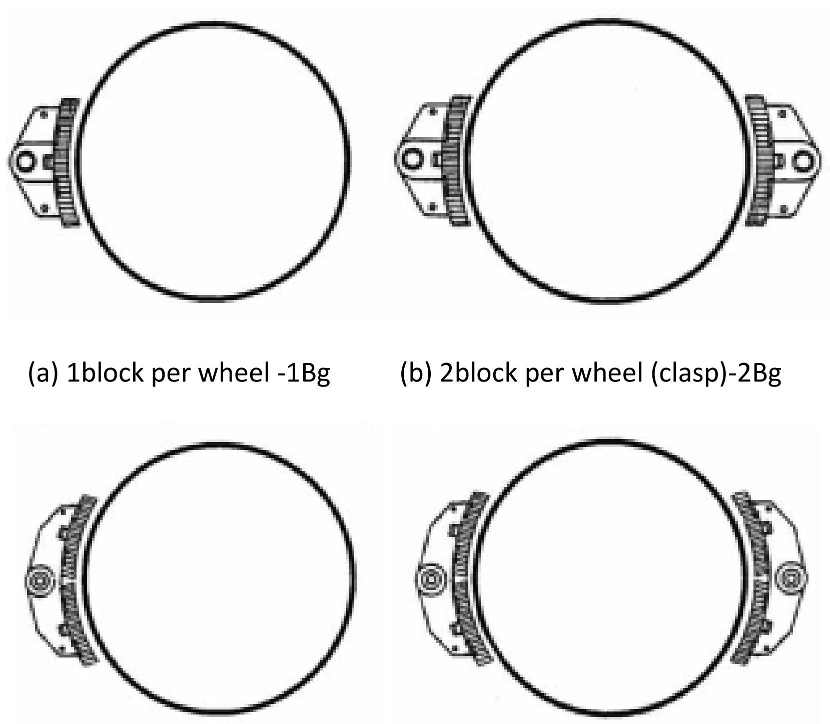
Figure 2.1 braking with shoes: 1-wheel 2- brake block (shoe).

$$F_f = \frac{\mu_s P_s}{A} \quad \text{......................................(1)}$$

$$F_{f\max} = F_a = \mu_a.Q \quad \text{.......................(2)}$$

Brake block are commonly manufactured from cast-iron, organic composite material or sinter materials. The development of friction materials over the last two decades has enabled introduction of new braking materials in the railway industry. One of the major issues faced over the last decades, which has influenced a change of material in brake blocks, is environmental constraints. For example, asbestos has been extensively used as filler for brake blocks but is hazardous to health and has therefore been restricted because of its fibrous nature [10], in [11] Vokolov and Lamizini observe results such as frictional properties, wear rate and heat resistance of railway block on freight car for different composites materials with minimal asbestos content.

They show that the composite brake blocks produced by slag pumice and various types of graphite as heat-conductive additives can fulfill the required braking properties. In addition, research has been performed on the influence of various types of block materials regarding brake performance and noise emissions [12-14].Accordingly, new standards [15] and new regulations to replace cast iron blocks with brake blocks of composite materials are being developed.



(a) 1block per wheel -1Bg (b) 2block per wheel (clasp)-2Bg

(c) 2blocks per wheel (tandem) -1Bgu (d) 4blocks per wheel -2Bgu

Figure 2.2 Four common brake block arrangement.

The present study is motivated by the fact that block braking still one of the most common braking systems on rail way vehicles. Block braking is the ordinarily used system on freight wagon and is also used on passenger trains often in combination with disc brakes and electro dynamic brakes [16]. The most commonly used block material is cast iron, but also sinter and composition material are in use.

2.2.1 Cast iron

Cast iron was firstly used as the material of brake shoe. Blocks made of cast iron are relatively cheap to manufacture. They have also a positive influence on the adhesion between wheel and tread. An increased adhesion has a positive influence on braking as well as on traction [17].

The amount of phosphorus contained in cast iron has great influence on brake shoe friction coefficient. Normally, the friction coefficient increases with the increase of the amount of phosphorus. However, if the amount of phosphorus is too high, the brake shoe is quite easy to crack [18].

2.2.2 Composite

Composition brake blocks are already in use for passenger rolling stock and was proven solution of both general applications to the railway system and acoustically, giving a reduction noise. Distinction is made between K blocks (composite blocks) of which several are available and LL blocks which are composites blocks with a similar friction characteristic to cast iron and could therefore be used as a simple retrofit in existing wagons [19]. The brake shoe made of composition material has the following advantages: The composition of material could be adjusted according to the requirement, making the friction coefficient much higher and insensitive to the variation of train's speed.

- ✓ High wear resistant
- ✓ Material has level of high uniformity and small porosity
- ✓ No spark occurs when brake is applied
- ✓ Weight is low, being only about 1/3 of that of cast iron brake shoe

2.3 Finite Element method and thermo-mechanical aspects of the brake shoe

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 mean time [20].

The power dissipation due to tangential stress and sliding velocity between wheel and tread block results in contact temperatures that can exceed 400 ℃ under normal operating conditions. They are confined to a very thin surface layer [21]. 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 in to 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 which can ultimately lead to premature failure of wheel [23].

Thermal analysis is involved in almost every kind of physical processes and can be the limiting factor for many processes. The finite element method (FEM) has become the preferred method in performing thermal analysis on a many systems and processes in recent years [24]. A FEA thermal analysis is a finite element analysis that looks at how heat affects certain materials and engineering designs. Thermal analysis and precise prediction of the maximum temperature is needed for the design of many systems, for example braking systems, especially for both discs and brake block (shoe), thermal analysis on brake block based on a combination computer based thermal model and FEM (ANSYS) based techniques to provide reliable method to calculate the temperature rise under a given brake schedule [25]. The thermal analysis is a primordial stage in the study of the braking systems, because the temperature determines thermo-mechanical behavior of the structure. In the braking phase, kinetic energy transforms into thermal energy, resulting in intense heating of the railway wheel. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks on treads of wheels and finally fractures of wheels and wear the brake shoe [26].

In this paper, the FEM model of a real brake thread assembly is developed and simulated using the commercially available FEM software packages, ANSYS and CATIA modeling. The model is simulated using a 3D brake block in order to observe the surface temperature distributions profile for different applied braking condition using gray cast iron brake block material.

2.4 Related Research

Milos Milosevic, Dusan Stamenkovic, Misa Tomic, Andrija Milojevic and Miroslave Mijajlovic [27], studies induced thermal loads determine thermo-mechanical behavior of the structure of railway wheels. In cases of thermal overloads, the generation of stresses and deformations are occurred whose consequences are the appearance of cracks on the rim of a wheel and the final total wheel defect. The material of the railway wheel is steel DIN 40Mn4 (AISI 1039, JIS S40C, C3130) and the braking blocks is gray cast iron P10.The presented modeling thermal effects of the braking process at block-braked railway vehicles is the base for the following stress analysis of the appropriate stress states as consequence of analyzed thermal loads in the simulated operation conditions. That approach could significantly decrease the probability of appearing of cracks caused by the thermal loads. This research can also help to discover on time the conditions for appearing these cracks and represents the final part of the process of solving problems concerning the solid wheel fracture due to thermal loads. In general the results obtained by the using analytical and numerical modeling of thermal effects during braking, until locomotive stops, and are good compared with those of the specialized literature. But the paper doesn’t compare with the experimental analysis of tread brake.

Shahab Teimourimaneh, Roger Lunden, Tore Vernersson [28], studies the tread braking systems with special focus on the braking capacity of the wheels, several aspects of the tread braking system, important for the dimensioning of railway wheels, are assessed, such as brake block, residual stresses and temperature gradients through wheel rim and disc. The influence of heat partitioning between wheel, brake block and rail on wheel temperature for different braking cycles is considered. The numerical results are on thermo-mechanical behavior of tread braking systems are given regarding the freight and metro applications. Limitation to this paper is it only limited to show dimensioning of rail way wheels considering the most common aspects such as wheel geometry and brake block material together with temperatures and residual stresses in the wheel rim. It does not show the optimization of material of brake block and focus on the braking of wheel.

M.Timur, H.kuscu [29], studies the friction of brake pads which is one of the factors affecting automobile performances with brake disc, heat energy comes in to being. This heat energy causes brake pads to expose to excessive temperature. For this reason, the braking acts of the pads decreases in time and some problems (decrease in the performance of brakes, malfunction, quick pad wear and sound) occur. In order to eliminate these negative effects emerging on the brake pads, a range of tests were made on pads. To analyze the temperature states of the pads, Solid Works Simulation Solution packet with finite element method was used. In this program, four different pad materials continually underwent braking act during300s process. In terms of revealing the situation of the pads occurring due to wear, two different thicknesses of the same pad we real so taken in to consideration in the analyses. The limitation of this research is shown thermal analysis of pad material by using different thickness but not compare with experiment result and not clearly describe the numerical analysis in detail.

Summary of the reviewed researches

In the above some of the researchers studied about numerical results on the thermo-mechanical behavior of tread braking systems with special focus on the braking capacity of the wheel, while the other researcher studied about the friction of brake pad is modeled using solid works software packet with finite element method to determine the effects of thermal and mechanical characteristics of various materials developed as a pad material on pad.

In this research the thermo-mechanical analysis of friction brake block for freight train A.A-Djibouti main line will be modeled using CATIA and analyzed using finite element method to determine thermo-mechanical effects and to show the surface temperature distributions profile for different applied braking condition using brake block material.

Chapter-Three

Analytical modeling of thermo-mechanical loads on brake shoe

To simulate a process of braking of railway vehicles it is necessary to define an analytical model of a thermal analysis that describes the heating 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 out flow of the whole braking system due to cooling of the surrounding air. For that purposes an analytical model for analyzing thermal effects in braking systems of freight train main line of A.A-Djibouti line was utilized and its approved procedure is presented in this paper for braking system of national freight railway vehicles.

The thermal analysis of the braking system of railway vehicles requires a precise determination of as the distribution of this energy between the railway wheel and the braking blocks. When the braking process occurs, blocks and rail way wheel are in a sliding contact. The resulting force resists the movements of the train slows down and eventually stops. The friction between the wheel and blocks always opposes motion and the heat is generated due to conversion of the kinetic energy [30].

3.1 Ethiopian railway national main line Freight Car specification

The below table show that the general technical parameters and condition of freight car of the national railway of Ethiopia [Source: Ethiopian Railways Corporation, Rolling Stocks Specifications].

Table3.1 Technical parameters of freight cars for national railway of Ethiopia

Loading capacity	22t
Dead weight	37.4t
Design speed	100Km/hr
Minimum curve radius	145m
Vehicle length(At coupler connection)	26066mm
Ambient temperature	25℃
Deceleration at emergency brake	1.4m/s
Deceleration at service brake	1.1m/s

3.1.1Bogie

This bogie is the three-piece cast steel structure and takes the separated swing bolster and lower cored disk structure, and the lower cored disk is mounted with the non-metallic cored wear plate, -dual-role elastic side bearing and variable friction damper. The swing bolster and side frame take the Chinese standard- compliant B+ grade cast steel, the compact double-row tapered roller bearing and the overall wheel with the diameter 840mm. The material is the Chinese standard- compliant CL60 steel, and the axis material is the Chinese standard-compliant LZ50 steel. The spring takes the Chinese standard-compliant 60Si2CrVAT material and the brake shoe with the high friction, and it takes the supporting brake head and brake bolt.

Table 3.2 Technical parameters and dimension of the bogie

Fixed Wheel Base	1830mm
Side bearing Center Distance	1520mm
Height from Cored Disk to Rail Surface(Empty Vehicle)	80mm
Railway Gauge	1435mm
Wheel Diameter	840mm
Axle load	25t
Design Speed	100Km/h

3.2 Freight car Braking and Angular Velocity

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 the maximum gradient in the Sebeta-Adama (SA) line occurs at the gradient down grade -18.5‰.

Table 3.3 Sebeta-Adama line gradient

No	stations	gradient(δ)	Sinδ
1	SA1	0	0
2	SA2	0.607	0.0106
3	SA3	0	0
4	SA4	0.6875	0.012
5	SA5	1.037	0.01810
6	SA6	1.060	0.01850
7	SA7	1.037	0.01810
8	SA8	1.060	0.01850
9	SA9	1.048	0.01830
10	SA10	1.060	0.01850
11	SA11	1.037	0.01810
12	SA12	1.060	0.01850
13	SA	0.487	0.0085
14	SA	-0.200	-0.035
15	SA	-0.6016	-0.0105
16	SA	-0.338	-0.0059
17	SA	0	0
18	SA	0.3437	0.006
19	SA	-0.3437	-0.006
20	SA	-1.0142	-0.01770
21	SA	-1.060	-0.01850
22	SA	-1.0142	-0.01770
23	SA	-0.4584	-0.0080
24	SA	0.0745	0.0013
25	SA	0.0286	0.0005
26	SA	0.5729	0.010
27	SA	0	0

3.3Total energy dissipated during braking

Brake is a device by means of which artificial frictional resistance is applied to moving machine member, in order to stop the motion of a machine. The energy of the freight train is transformed in to heat while the braking is applied. Assume at maximum gradient the driver hold emergency brake the total braking energy becomes the change in rotational and translational kinetic energy and the change in potential energy that the train undergoes during braking should be precisely determined.

The change in translational kinetic energy is given by:

$\Delta KE = \frac{1}{2} m{v_o}^2$……………………………………………………………………3.1

Where,

ΔKE = Initial Kinetic Energy of the freight car just before braking starts

m = Mass of the vehicle

v_o = velocity of the vehicle

m = loading capacity + dead weight

m= 22000+37400 = 59400K.g

By substituting the value of mass of the vehicle and velocity of the vehicle **ΔKE is 23MJ**

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

$\Delta Ek_r = \frac{1}{2} I{\omega_o}^2$……………………………………………………………………3.2

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

$$I = \sum_{i=1}^{n} m_i {r_i}^2$$

$\Delta Ek_r = \frac{1}{2} \sum_{i=1}^{n} m_w {r_i}^2 \, {\omega_o}^2$…………………………………………………………..3.3

Where,

ΔEk_r= in rotational kinetic energy of all rotating parts

ω_o = initial angular velocity the wheel

m_w = the masses of the wheel set

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

For this case, all the rotating parts are fixed on the axle of the wheel set and the rotation axis could be taken as the tangent line joining the contact point of the rail heads with wheel set of the freight car. It is the parallel to the axis of the wheel axle. Hence, the rotating radius is ($\boldsymbol{r_i}$) and equal to the radius of the wheel ($\boldsymbol{r_w}$).

The freight box car bogie is the three-piece cast steel structure and by substituting the values of **m_i (m_w)=391k.g**, **r_i (r_w)=0.42m and ω_o =66.1rad/sec** the ΔEk_r **is 1.2MJ.**

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

$\Delta E_p = mgh$……………………………………………………………………………….3.4

Where,

ΔE_p= potential energy

m = the mass of the vehicle

m = 59400K.g

g = gravitational acceleration

h = height

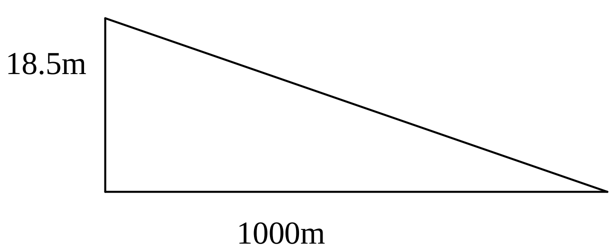


Figure 3.1 Maximum gradient

By substituting the value of m, g, and taking the maximum gradient from the table 3** sebeta – adama line 18.50/00 the potential energy becomes $\Delta \boldsymbol{E_p}$ ***is 10.78MJ.***

Thermal properties of wheel and shoe are listed in table. The specific heat of shoe is about 2 times larger than that of wheel while conductivity of shoe is one ninth of that of wheel and thermal energy 10% conduct in to brake shoe and 100% conduct in to wheel. As compared with wheel, brake shoe cannot transfer thermal energy to the neighborhood cell [31].

Table 3.4 Thermal properties of wheel and shoe

	Density(K.g/m^3)	Specific heat(J/K.g K)	Conductivity(W/mK)
Wheel	7753	486	36
Shoe	2010	1040	4.22

Since the brake block thermal energy 10% and 90% wheel of the thermal energy creates between wheel and brake block .The total braking energy is given by:

$$E_b = 0.1(\Delta KE + \Delta Ekr + \Delta Ep) \ldots\ldots 3.5$$

$$E_b = 0.1(\frac{1}{2}mv_o{}^2 + \frac{1}{2}\sum_{i=1}^{n} m_i r_i{}^2\, \omega_o{}^2 + mgh)$$

By adding the rotational, transitional and potential energy the total energy becomes

$\boldsymbol{E_b}$ **is 3.5MJ**

3.4Braking time and distance at Emergency brake

Brake Application is defined as an application of the brake that results in a brake force being applied to the vehicle. Brake force is the force applied to the brake block surface interface.

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

$$v(t) = v_o - at \qquad (3.6)$$

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

$$\omega_w = \frac{v(t)}{r_w} \qquad (3.7)$$

$$\omega_o = \frac{v_o}{r_w} \qquad (3.8)$$

By substituting the value of v_o and r_w the **ω_o is 66.11S^{-1}**

Where,

v(t) = velocity of the freight car at function of time

v_o = Initial running velocity of the freight car

ω_o = Initial angular velocity the wheel

ω = angular velocity of the wheel at any time

r_w = the radius of freight car wheel

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

$$s_b = v_o t_b - \frac{1}{2} a t_b^2 \qquad (3.9)$$

$$t_b = \frac{v_o}{a} \qquad (3.10)$$

Where,

t_b = brake time to stop the locomotive

S_b = total brake distance

Given

v_o= 100km/hr(27.77m/sec)

a = 1.4m/s^2

By substituting in the above formula we have got **t_b=20sec** *and* **S_b=275m**

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

$\omega = \omega_o(1 - \frac{t}{t_b})$……………………………………………………………….3.11

3.4.1Total braking force during emergency brake

$F_t = \frac{E_b}{\frac{1}{2}(n_1+n_2)*t_b*r_w*\mu}$ ……………………………………………………….....3.12

Where,

F_t= total braking force

E_b = the total braking energy

n_1 = initial angular velocity

n_2 = final angular velocity

t_b = braking time

μ = coefficient of friction between wheel and brake block (shoe)

r_w = radius of the wheel

By substituting the above parameters and the high friction coefficient between wheel and brake block is **μ = 0.4,** the total braking force **F_t is 31.5KN.** Since the freight box cars has two bogies and in one wheel side have two brake blocks applying brake force for one brake block is**1.97KN.** For one wheel side one brake block applying brake force is **3.937KN.**

3.4.2Mathematical Model for Interface Temperature rise at Brake-Block during emergency brake

UIC blocks are the most common blocks for Air Brakes and the optimal length of brake shoes are UIC320mm for easy mounting in shoe-wear andUIC250mm when fitted to double shoe-wear. Width of the brake shoes in all cases is 80 up to 85mm [32].The area (A) of contact between railway wheel and brake block is given by

A=lb……………………………………………………………3.13

Where: l is the circumferential length of the brake block

b is the width of the brake block.

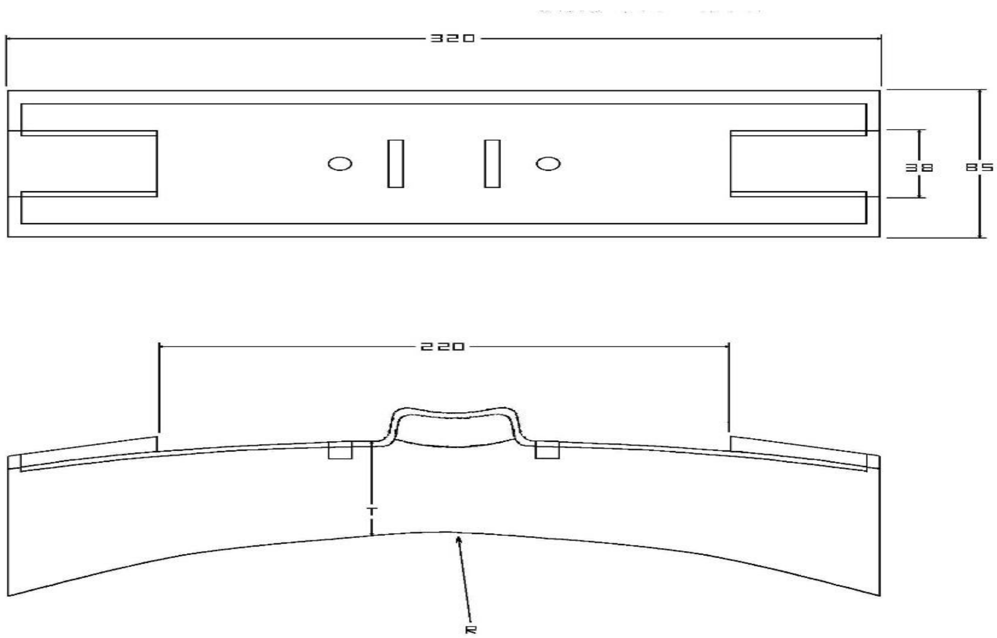


Figure 3.2 brake block (shoe)

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. 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 railway locomotive.

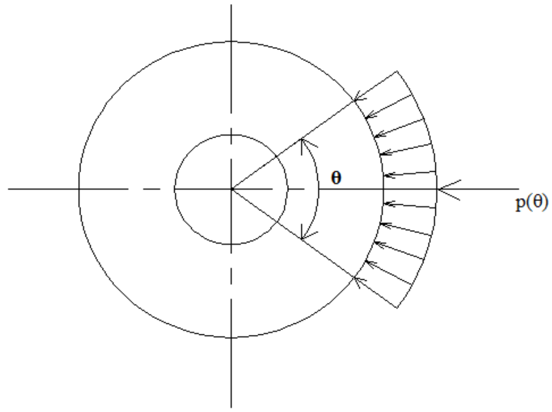


Figure 3.3 Brake Distribution Force

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

$$dA = bRd\theta \quad \text{.......} 3.14$$

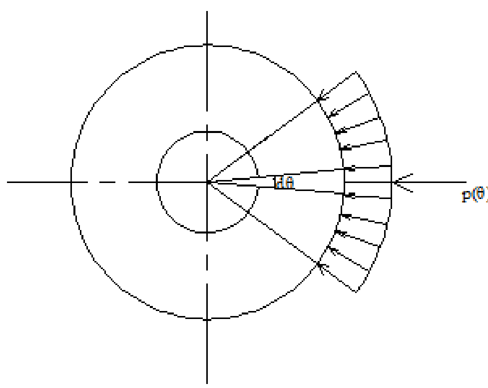


Figure 3.4 Differential contact angle of brake block (shoe) on the wheel

By integrating area of contact is given by

$$A = b*R*\theta$$

For A_1R-Radius of the wheel is 420mm, b-width of the brake block is 85mm and θ_1 is 30^{o}

$$A_1 = b*R*\theta_1$$

$$A_1 = 85*420*\frac{\pi}{6}$$

$$A_1 = 18692.47 mm^2$$

For A_2 R-Radius of the wheel is 420mm; b-width of the brake block is 85mm and θ_2 is 45^o

$A_2 = b*R*\theta_2$

$A_2 = 85*420*45^o$

$A_2 = 85*420*\frac{\pi}{4}$

$A_2 = 28038.71mm^2$

The surface pressure between the brake block and railway wheel using to the calculate force applied to the tread block is given as figure below.

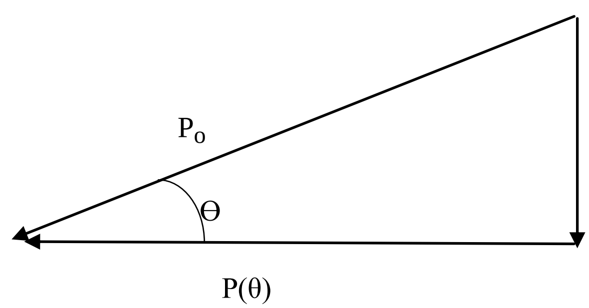


Figure 3.5 Determination of pressure distribution

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 brake block will be calculated as follows.

$p(\Theta) = p_O cos(\Theta)$ ………………………………..……3.15

The pressure at some specified angle of the contact between wheel and brake block will be calculated **(p_O)**. In one wheel side have two brake blocks from braking force applying on the one brake block **F_w = 1.97KN.** For one wheel side one brake block applying brake force is **F_w = 3.9375KN.**

$P_o = \frac{F_W}{A}$...3.16

Where,

P_o= Pressure between wheel and brake block

F_w = braking force on the one brake block

A = area of contact between wheel and brake block

Different angles are selected for this analysis. For the simplicity two angles are selected for the determination of the pressure (i.e.30^{o} and45^{o}).Thus angles are used throughout this research paper in all calculations. Hence on one wheel side have two brake blocks the pressure between wheel and brake block at this selected two angles will becomes **P_o @ 30^{o} is 105Kpa andP_o@45^{o} is 70Kpa** and For one wheel side have one brake block the pressure between wheel and brake block at this selected two angles will becomes **P_o@ 30^{o} is 210.65Kpa andP_o@45^{o} is 140Kpa.**

From above equation 3.15 on one wheel side have two brake blocks the maximum pressure at the two different contact angle will be calculated so that **P(30)=91KPa and P (45)=49KPa** and on one wheel side have one brake block the maximum pressure at the two different contact angle are **P(30)=182KPa and P (45)=99KPa** The minimum the brake pressure applied on the brake shoe is the maximum the brake force applied on the wheel at the contact area.

$dF_A = p_O cos(\Theta) bRd\Theta$ 3.17

$F_A = \int_{-\Theta}^{+\Theta} 2p_o cos(\Theta)\, bRd\Theta$ 3.18

$F_A = 2p_o bRsin(\Theta)$ 3.19

From above on one wheel side have two brake blocks brake force applied on the brake shoe occurs at for θ=30^{o} is **3.24KN** and θ=45^{o} is **2.47KN** and on one wheel side have one brake block brake force applied on the brake block occurs at for θ=30^{o} is **6.497KN** and θ=45^{o} is **5KN.**

3.4.2.1 Heat flux during emergency brake

The thermal analysis of the braking system of railway vehicles requires determination of the quantity of heat produced by friction, as well as the distribution of this energy between the railway wheel and the brake block. The heat flux evacuated of surfaces in contact (between blocks and railway wheel) is equal to the power friction of the wheel and block [20].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 generated per unit contact area at the radius *R* of the wheel can be calculated by the following equation:

$$P(R) = -F_f v(t) \quad \text{......3.20}$$

$$P(R) = -F_f(\omega_o R - at)$$

Where: P(R) = heat power generated per unit contact area

F_f =is the friction force per unit contact area

$v(t)$ = velocity of the freight car at function of time

v_o = Initial running velocity of the freight car

a = deceleration

t = function of braking time

R = radius of wheel

ω = initial angular velocity

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

$$F_f = \frac{\mu_s F_A}{A} \quad \text{......3.21}$$

Where μ_s is the friction coefficient between wheel and tread, F_A– the normal braking force on one of the braking blocks, and **A**–the area of the contact surface between one braking block and the rail way wheel.

The normal braking force on brake block when on one wheel side have two brake blocks is for $\theta=30^o$ is **3.24KN** and $\theta=45^o$ is **2.49KN** and when on one wheel side have one brake block is for $\theta=30^o$ is **6.497KN** and $\theta=45^o$ is **5KN** coefficient of friction of high friction of brake block is **0.4** and inserting the above values in to the above equation frictional force per unit area when on one wheel side have two brake blocks for $\theta=30^o$ will be **69kN/m^2 (0.069N/mm^2)** and for $\theta=45^o$ will be **35kN/m^2 (0.035N/mm^2).** and when on one wheel side have one brake block for $\theta=30^o$ will be **139kN/m^2 (0.139N/mm^2)** and for $\theta=45^o$ will be **71kN/m^2 (0.071N/mm^2).**

The heat power generation per unit contact area can calculated from the above equation 3.20 correspondent to the heat flux entering the brake block (shoe) from the wheel.

Where ω is angular velocity [**66.11rad/s**] , radius of the wheel is **420mm** , braking time[**20sec**] divide by 2sec interval and by taking the values of braking force on brake block and F_f is the frictional force per unit area so that for $\theta=30^o$and $\theta=45^o$ heat flux P**(R)** is given in the table form as follow .

Table 3.5 Heat flux at different time using the contact area at emergency brake when wheel side have two brake blocks.

Time (sec)	**Heat flux P(R) at 30^o (W/m^2)**	**Area of contact at 30^o (m^2)**	**Heat flux P(R) at 45^o (W/m^2)**	**Area of contact at 45^o (m^2)**
2	1722667.8	0.018692	873817	0.028038
4	1529467.8	0.018692	775817	0.028038
6	1336267.8	0.018692	677817	0.028038
8	1143067.8	0.018692	579817	0.028038
10	949867.8	0.018692	481817	0.028038
12	756667.8	0.018692	383817	0.028038
14	563467.8	0.018692	285817	0.028038
16	370267.8	0.018692	187817	0.028038
18	177067.8	0.018692	89817	0.028038
20	0	0.018692	0	0.028038

Table 3.6 Heat flux at different time using the contact area at emergency brake when a wheel side has one brake block.

Time (sec)	**Heat flux P(R) at 30^o (W/m^2)**	**Area of contact at 30^o (m^2)**	**Heat flux P(R) at 45^o (W/m^2)**	**Area of contact at 45^o (m^2)**
2	3470300	0.018692	1772600	0.028038
4	3081100	0.018692	1573800	0.028038
6	2691900	0.018692	1375000	0.028038
8	2302700	0.018692	1176200	0.028038
10	1913500	0.018692	977400	0.028038
12	1524300	0.018692	778600	0.028038
14	1135100	0.018692	579800	0.028038
16	745900	0.018692	381000	0.028038
18	356700	0.018692	182200	0.028038
20	0	0.018692	0	0.028038

3.5Braking time and distance at service brake

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

$$v(t) = v_o - at \quad \text{......3.22}$$

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

$$\omega_o = \frac{v_o}{r_w} \quad \text{......3.23}$$

By substituting the value of v_o and r_w the $\boldsymbol{\omega_o}$**is 66.11S^{-1}**

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

$$s_b = v_o t_b - \frac{1}{2} a t_b^2 \quad \text{......3.24}$$

$$t_b = \frac{v_o}{a} \quad \text{......3.25}$$

Where,

t_b= brake time to stop the locomotive

S_b = total brake distance

Given

v_o= 100km/hr (27.77m/sec)

a = 1.1m/s^2

By substituting in the above formula we have got **t_b=25sec** *and* **S_b=350.5m**

3.5.1Total braking force during service brake

$$F_t = \frac{E_b}{\frac{1}{2}(n_1+n_2)*t_b*r_w*\mu} \quad \ldots\ldots 3.26$$

By substituting the above parameters and the high friction coefficient between wheel and brake block is **μ = 0.4,** the total braking force **F_t is 25KN.** Since the freight box cars has two bogies and in one wheel side have two brake blocks applying brake force for one brake block is **1.56KN.** For one wheel side have one brake block applying brake force is **3.12KN.**

3.5.2Mathematical Model for Interface Temperature rise at Brake-Block during service brake

The pressure at some specified angle of the contact between wheel and brake block will be calculated as follows

$$p(\Theta) = p_O cos(\Theta) \quad \ldots\ldots 3.27$$

The pressure at some specified angle of the contact between wheel and brake block will be calculated **(p_0)**. In one wheel side have two brake blocks from braking force applying on the one brake block **F_w = 1.56KN.** For one wheel side one brake block applying brake force is **F_w = 3.12KN.**

$$P_o = \frac{F_w}{A} \quad \text{..........} 3.28$$

Where,

P_o= Pressure between wheel and brake block

F_w = braking force on the one brake block

A = area of contact between wheel and brake block

Different angles are selected for this analysis. For the simplicity two angles are selected for the determination of the pressure (i.e.30^o and45^o). Hence on one wheel side have two brake blocks the pressure between wheel and brake block at this selected two angles will becomes **P_o@ 30^o is 83Kpa andP_o@45^o is 55.6Kpa** and For one wheel side have one brake block the pressure between wheel and brake block at this selected two angles will becomes **P_o@ 30^o is 167Kpa andP_o@45^o is 111Kpa.**

From above equation 3.15 on one wheel side have two brake blocks the maximum pressure at the two different contact angle will be calculated so that **P(30)=72KPa and P (45)=39KPa** and on one wheel side have one brake block the maximum pressure at the two different contact angle are **P(30)=144.6KPa and P (45)=78.48KPa**.The minimum the brake pressure applied on the brake block is the maximum the brake force applied on the wheel at the contact area.

$$dF_A = p_O cos(\Theta) bRd\Theta \quad \text{..........} 3.29$$

$$F_A = \int_{-\Theta}^{+\Theta} 2p_o cos(\Theta)\, bRd\Theta \quad \text{..........} 3.30$$

$$F_A = 2p_o bRsin(\Theta) \quad \text{..........} 3.31$$

From above on one wheel side have two brake blocks brake force applied on the brake block occurs at for θ=30° is **2.57KN** and θ=45° is **1.97KN** and on one wheel side have one brake block brake force applied on the brake shoe occurs at for θ=30° is **5.1KN** and θ=45° is **3.96KN.**

3.5.2.1 Heat flux during service brake

The heat power generated per unit contact area at the radius R of the wheel can be calculated by the following equation:

$$P(R) = -F_f\, v(t) \qquad \text{3.32}$$

$$P(R) = -F_f(\omega_o R - at)$$

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

$$F_f = \frac{\mu_s F_A}{A} \qquad \text{3.33}$$

Where $\mathbf{\mu_S}$ is the friction coefficient between wheel and tread, $\mathbf{F_A}$– the normal braking force on one of the braking blocks, and **A**–the area of the contact surface between one braking block and the rail way wheel.

The normal braking force on brake block when on one wheel side have two brake block is for $\theta=30^o$ is **2.57KN** and $\theta=45^o$ is **1.97KN** and when on one wheel side have one brake block is for $\theta=30^o$ is **5.1KN** and $\theta=45^o$ is **3.96KN** coefficient of friction of high friction of brake block is **0.4** and inserting the above values in to the above equation frictional force per unit area when on one wheel side have two brake block for $\theta=30^o$ will be **55kN/m^2 (0.055N/mm^2)** and for $\theta=45^o$ will be **28kN/m^2 (0.028N/mm^2).** and when on one wheel side have one brake block for $\theta=30^o$ will be **109kN/m^2 (0.109N/mm^2)** and for $\theta=45^o$ will be **56.5kN/m^2 (0.0565N/mm^2).**

The heat power generation per unit contact area can calculated from the above equation 3.20 correspondent to the heat flux entering the brake block (shoe) from the wheel.

Where ω is angular velocity [**66.11rad/s**] , radius of the wheel is **420mm** , braking time[**25sec**] divide by 2.5sec interval and by taking the values of braking force on brake block and F_f is the frictional force per unit area so that for $\theta=30^o$and $\theta=45^o$ heat flux P**(R)** is given in the table form as follow.

Table 3.7 Heat fluxes at different time interval using the contact area at service brake when wheel side have two brake blocks.

Time (sec)	**Heat flux P(R) at 30^o (W/m^2)**	**Area of contact at 30^o (m^2)**	**Heat flux P(R) at 45^o (W/m^2)**	**Area of contact at 45^o (m^2)**
2.5	1375891	0.018692	700453.6	0.028038
5	1224641	0.018692	623453.6	0.028038
7.5	1073391	0.018692	546453.6	0.028038
10	922141	0.018692	469453.6	0.028038
12.5	770891	0.018692	392453.6	0.028038
15	619641	0.018692	315453.6	0.028038
17.5	468391	0.018692	238453.6	0.028038
20	317141	0.018692	161453.6	0.028038
22.5	165891	0.018692	84453.6	0.028038
25	0	0.018692	0	0.028038

Table 3.8 Heat fluxes at different time interval using the contact area at service brake when wheel side has one brake block.

Time (sec)	**Heat flux P(R) at 30^o (W/m^2)**	**Area of contact at 30^o (m^2)**	**Heat flux P(R) at 45^o (W/m^2)**	**Area of contact at 45^o (m^2)**
2.5	2726765.8	0.018692	1413415.3	0.028038
5	2427015.8	0.018692	1258040.3	0.028038
7.5	2127265.8	0.018692	1102665.3	0.028038
10	1827515.8	0.018692	947290.3	0.028038
12.5	1527765.8	0.018692	791918.3	0.028038
15	1228015.8	0.018692	636540.3	0.028038
17.5	928265.8	0.018692	481165.3	0.028038
20	628515.8	0.018692	325790.3	0.028038
22.5	328765.8	0.018692	170415.3	0.028038
25	0	0.018692	0	0.028038

3.6 Heat transfer

Heat transfer is used accurately to 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 air flow 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 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:

$$Q = hA\,(T_s - T_\infty) \qquad 3.34$$

Where:

Q =Rate heat transfer (W)

h = convection heat transfer coefficient

A =surface contact area (m^2)

T_S=surface temperature (^oc)

T_∞= ambient temperature (^oc)

At the contact surface between the wheel and the blocks, the braking process produces heat according to heat flux. The heat dissipation from the free surfaces of the wheel and blocks to the surrounding air is described by both convection and radiation [29].

$q_{diss} = -h\left(T - T_{ref}\right) - \varepsilon\sigma(T^4 - T_{ref}^4)$..3.35

In this equation, h equals the convective film coefficient; ε is the material's emissivity, σ–the Stefan-Boltzmann constant ($5.67*10^{-8}$), and *Tref*–the temperature of the surrounding air and Addis Ababa–dire dawa main line 25^0c.

To calculate the convective film coefficient as a function of the railway vehicle velocity *v*, the following formula [20] should be used:

$h = \frac{0.037K_a}{r_w} R_e^{0.8} P_r^{0.33}$ 3.36

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

$R_e = \frac{2\rho_a Rv}{\mu_{Va}}$ 3.37

$P_r = \frac{C_a \mu_{Va}}{K_a}$ 3.38

Here, material properties: thermal conductivity k_a ,density ρ_a, dynamic viscosity μ_{va} and specific heat capacity C_{pa},are given for the surrounding air, v is the train speed which is 100Km/hr and R is the radius of the wheel which is 420mm.These material properties greatly affect the properties of the brake block at the operating speed.

Table 3.9 Material properties of air

No	Properties of materials	Value
1	Thermal conductivity(k_a)	0.026W/Mk
2	Density of the material(ρ)	1.170Kg/m^3
3	Dynamic viscosity(μ_{va})	$1.8*10^{-5}$Kg/ms
4	Specific heat capacity (C_a)	1100J/KgK
5	Ambient temperature (T)	25^oc

Taking the data from the above table, convective film transfer coefficient according to the equations given above will be, h is **177W/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.

$\dot{Q} = E_b$ ………………………………………………………...3.27

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

$\frac{dQ}{dt} = UA(T_i - T_\infty)$……………………………………………..3.28

$E_b = m\Delta TC$ ………………………………………………...3.29

From above equation and Newton's law of cooling

$\frac{T - T_\infty}{T_i - T_\infty} = e^{-(\frac{UA}{mC})}$ ………………………………………………….3.30

Rearranging this equation

$T_i = T_\infty + \frac{\Delta T}{e^{-\beta t}}$ ………………………………………………….3.31

Where U is the overall heat transfer coefficient, C is the specific heat capacity of the material, m is mass of the brake block , T_i initial temperature, T_∞ ambient temperature and

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

By using the total braking energy of conducts on the brake shoe **E_b =3.5MJ**, mass of the brake shoe **m_s** = **16Kg** and the specific heat capacity of the wheel **C_s=510J/K.g K** we calculate the change of temperature.

$$E_b = m_s \Delta T C_s$$

$$\Delta T = 156\ ^0C$$

From the above Newton's of cooling formula the initial temperature is given by

$$\Delta T = T_i - T_\infty$$

$$T_i = \Delta T + T_\infty$$

Where: the change temperature ΔT =156 ^{0}C and ambient temperature T_∞ = 25^{0}C

$$T_i = 181^0\text{C}$$

From the equation 3.23 the heat dissipation from the free surfaces of the wheel and blocks will be calculate by.

$$q_{diss} = -h(T_i - T_{ref}) - \varepsilon\sigma({T_i}^4 - T_{ref}^4)$$

Heat dissipation due to radiation is neglected because the value is very small and film coefficient h =177W/m$^{2\,0}$C, T_i= 181^{0}C and T_{ref}=25^{0}C.

$$q_{diss} = -h(T_i - T_{ref})$$

$$q_{diss} = 27612W/m^2$$

The heat conduction wheel and brake block (shoe) is given by

$$q_{cond} = q_{flux} - q_{diss} \quad \text{………………………………………………………..3.32}$$

By taking heat dissipation from the above $q_{diss} = 27612W/m^2$ and when on one wheel side have two brake shoe is for 30^0 and 45^0 angle of contact the maximum heat flux is from the table for emergency brake $q_{flux@30^0}$ =1.915MW/m^2 and is $q_{flux@45^0}$ = 0.971 MW/m^2 and for service brake $q_{flux@30^0}$ = 1.527MW/m^2 and is $q_{flux@45^0}$ = 0.777MW/m^2 . when on one wheel side have one brake shoe is for 30^0 and 45^0 angle of contact the maximum heat flux is from the table for emergency brake $q_{flux@30^0}$ =3.47MW/m^2 and is $q_{flux@45^0}$ = 1.77 MW/m^2 and for service brake $q_{flux@30^0}$ = 2.726MW/m^2 and is $q_{flux@45^0}$ = 1.41MW/m^2 The heat conduction at different of angle of contact of the brake shoe will be obtained as follow.

For emergency brake when one wheel side have two brake block

$$q_{cond@30^0} = 1.887MW/m^2 \text{ and } q_{cond@45^0} = 0.94MW/m^2$$

For service brake when one wheel side have two brake block

$$q_{cond@30^0} = 1.5MW/m^2 \text{ and } q_{cond@45^0} = 0.75MW/m^2$$

For emergency brake when one wheel side have one brake block

$$q_{cond@30^0} = 3.44MW/m^2 \text{ and } q_{cond@45^0} = 1.74MW/m^2$$

For service brake when one wheel side have one brake block

$$q_{cond@30^0} = 2.698MW/m^2 \text{ and } q_{cond@45^0} = 1.38MW/m^2$$

Chapter-Four

FEA of Freight car brake block (shoe) using ANSYS

For analyzing the process of braking, it is very useful to use FEM simulations based on the adopted analytical model for defining heat sources at surfaces which are in contact with brake shoe.

This study will use the FEA software ANSYS12 version to carry out transient thermal analysis and the thermo - mechanical stresses of a railway brake shoe during stop braking conditions will be determined.

4.1 Introduction to Finite Element Analysis

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

In displacement based FEM, stiffness of the entire structure (Part or assembly) is assembled from stiffness of individual elements. Loads and boundary conditions are applied at the nodes and the resulting sets of the simultaneous equations are solved using matrix methods and numerical techniques. In short, FEM is a numerical method to solve ordinary differential equations of equilibrium. Starting with simple linear static stress and heat transfer analysis, complex simulations involving highly non-linear, fluid flow and dynamic events can be successfully analyzed on a personal computer using a host of popular software like ANSYS. In practice, a finite element analysis usually consists of three principal steps.

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

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

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

4.2 Introduction to FEA Software ANSYS

ANSYS is general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user-designated size) called elements. The software implements equations that govern the behavior of these elements and solves them all; creating a comprehensive explanation of how the system acts as a whole. These results then can be presented in tabular or graphical forms. This type of analysis is typically used for the design and optimization of a system far too complex to analyze by hand.

4.3 brake block (shoe) modeling

The solid modeling of the brake block material is mandatory using solid modeling software before performing any type of analysis. It is possible to use more software's 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 brake block shoe of the freight box car train and the analysis is done in the same way on this work bench on different window.

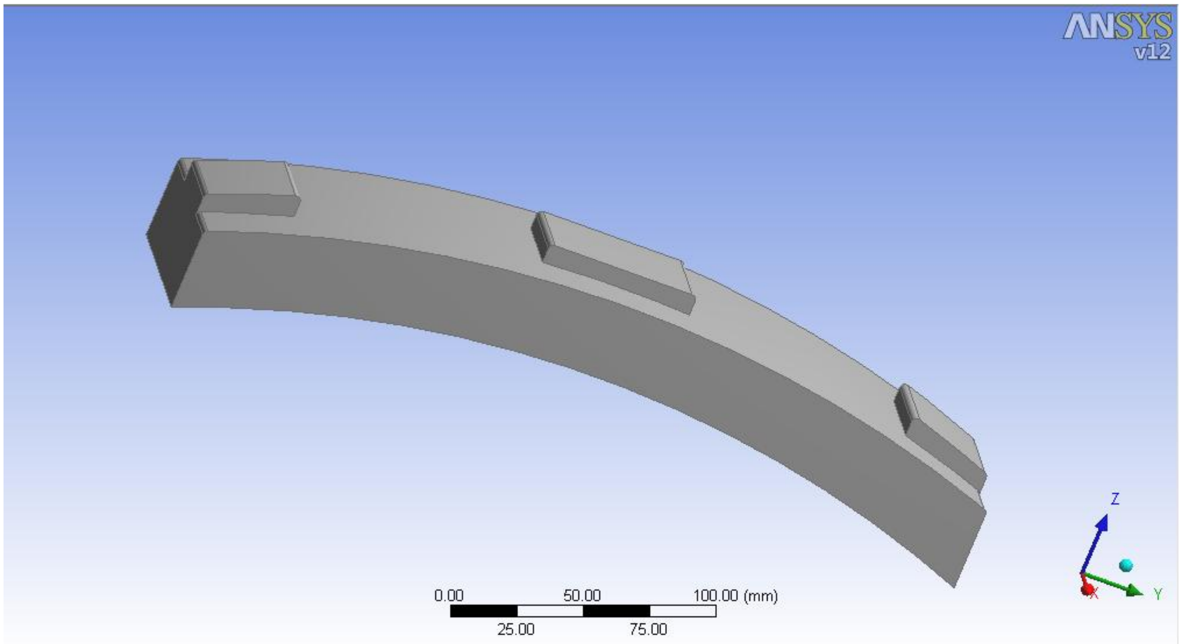


Figure 4.1 Brake block (shoe) model using ANSYS work bench

4.4 Meshing

In the finite element analysis the basic concept is to analyze the structure, which is an assemblage of discrete pieces called elements, which are connected, together at a finite number of points called nodes. A network of these elements is known as a mesh.

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 brake shoe a 3D element the appropriate meshing method will be volume mesh so that all the volume of the brake shoe is discritised to the smallest part of the brake shoe. The resulting volume mesh is automatic and the output of the mesh consists of 28476nodes and 17114 tetrahedral elements each approximately 2.4 mm length. After this the object is ready to be analyzed by setting the analysis on the ANSYS work bench

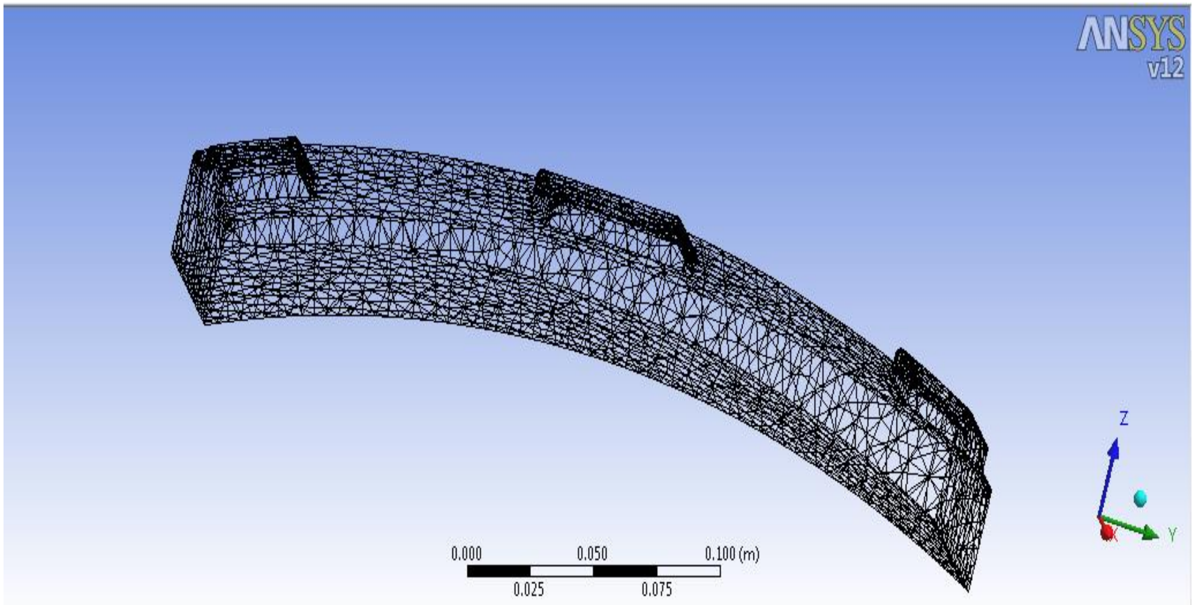


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Figure 4.2 Meshed brake block (shoe)

4.5Analysis 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 brake shoe and the second is to set the structural properties of the brake shoe. 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 brake shoe. Then since the train is on motion the structural transient analysis is used to solve the structural analysis of the brake shoe. The general analysis setting is shown in the following figure.

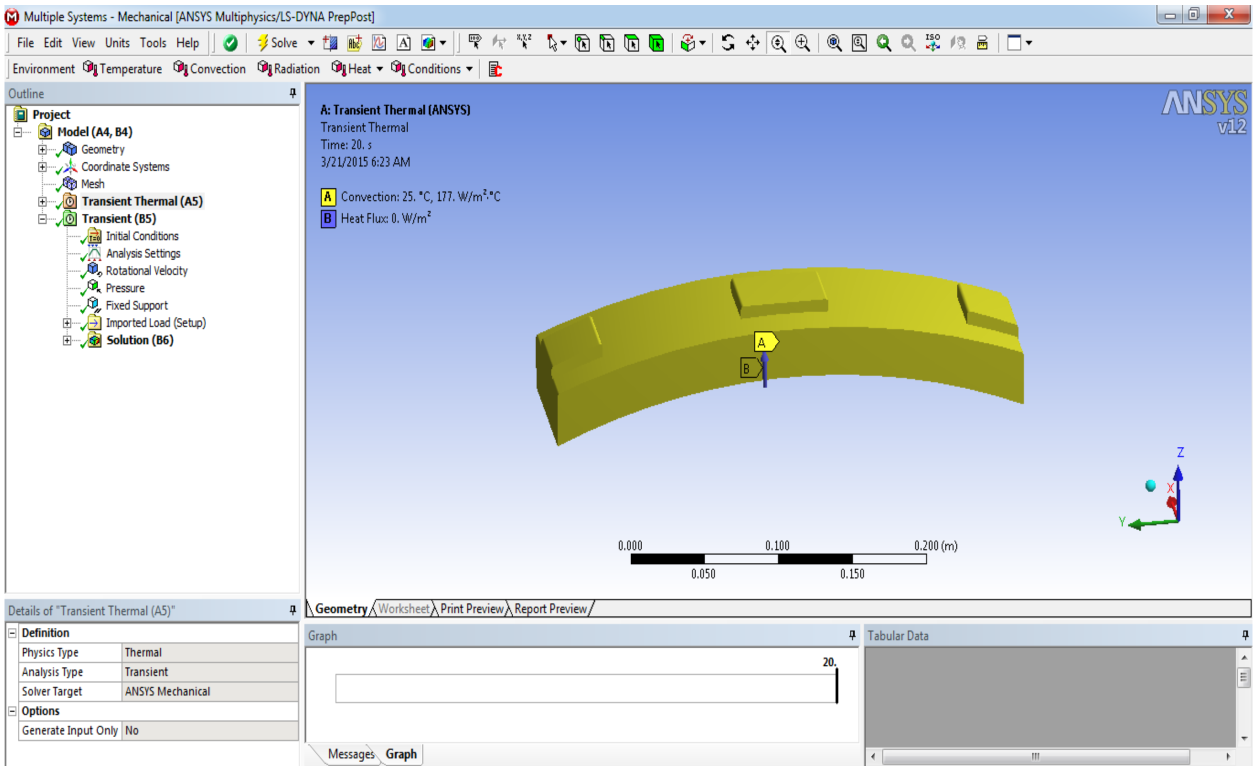

Figure 4.3 Analysis setting

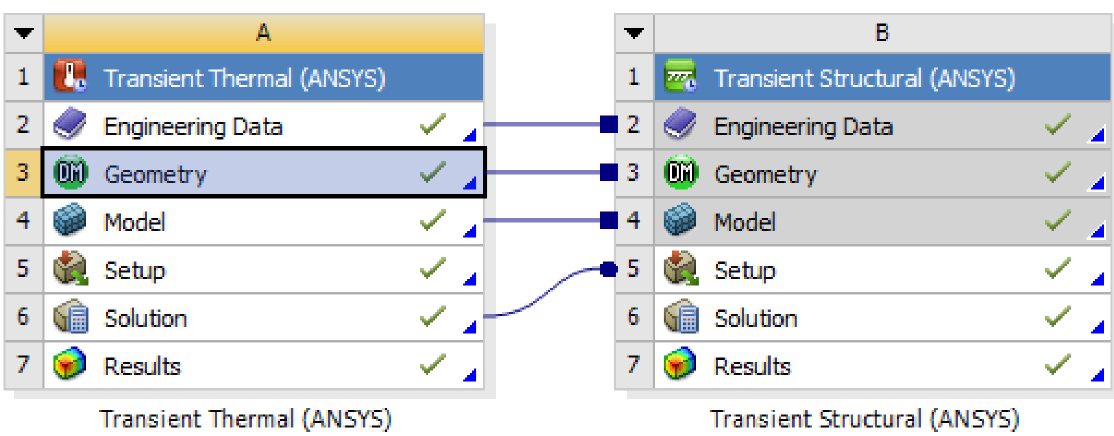

Figure 4.4 Stepwise solving of ANSYS

Chapter five

Results and Discussions

5.1 Material Properties of grey cast iron P10

The material of the freight box train of brake block is grey cast iron P10 with high carbon content [26], with good physical characteristics and thermal properties of materials for transient analysis of the brake block are given.

Table 5.1 Material properties of grey cast iron P10

No	Properties of materials	Value
1	Thermal conductivity(k)	42W/MK
2	Density of the material(ρ)	7200Kg/m^3
3	Specific heat capacity (C)	510J/Kg K
4	Compressive ultimate strength	820Mpa
5	Poisson's ratio *v* [/]	0.28
6	Young modulus (E)	110Gpa

5.2 The input data for the ANSYS

The data needed from the above analytical for this ANSYS analysis and to obtain the correct result. These data are different for the two contact angle of the brake block.

Table 5.2 Input data at emergency brake when on a wheel side have two and one brake block

No	Input	Value	
1	Convective film coefficient	177W/m^2/^{o}C	
2	Acceleration	1.4m/s^2	
3	Braking time	20sec	
4	Rotational speed	66.11rad/sec	
5	Gravitational acceleration	9.81m/s^2	
6	Pressure due to brake force on one wheel side have two brake shoe	91KPa for $\theta=30^o$	49KPa for $\theta=45^o$
7	Pressure due to brake force on one wheel side have one brake shoe	182KPa for $\theta=30^o$	99KPa for $\theta=45^o$

Table 5.3 Heat flux at different time intervals during emergency brake when on a wheel side have two brake blocks.

Time (sec)	Heat flux P(R) at 30^o (W/m^2)	Heat flux P(R) at 45^o (W/m^2)
2	1722667.8	873817
4	1529467.8	775817
6	1336267.8	677817
8	1143067.8	579817
10	949867.8	481817
12	756667.8	383817
14	563467.8	285817
16	370267.8	187817
18	177067.8	89817
20	0	0

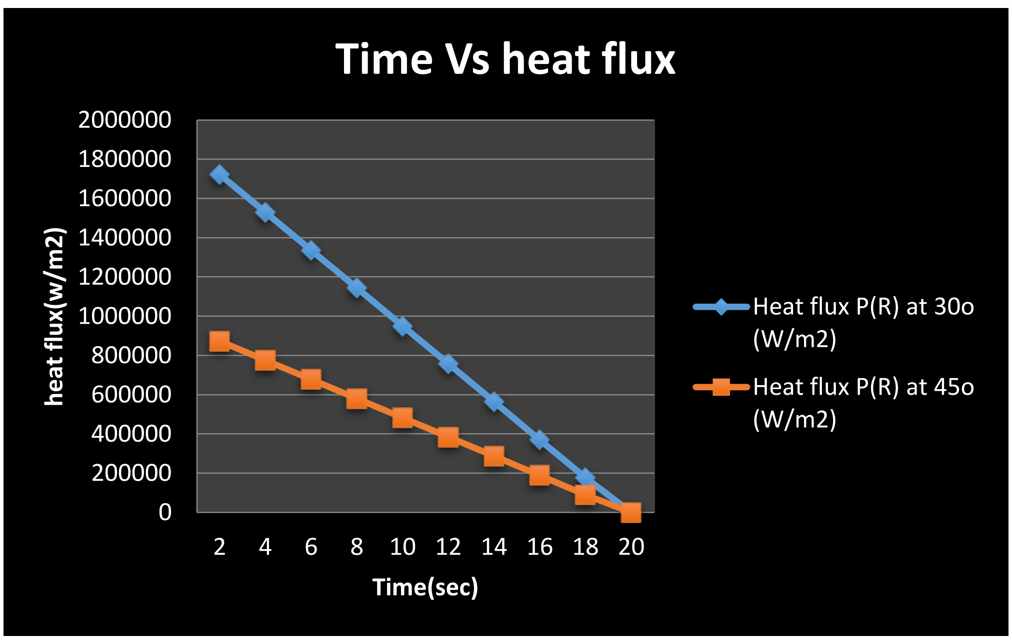

Figure 5.1 Time Vs heat flux graph during emergency brake on a wheel side have two brake blocks

Table 5.4 Heat flux at different time intervals during emergency brake when on a wheel side have one brake block.

Time (sec)	**Heat flux P(R) at 30^{o} (W/m^2)**	**Heat flux P(R) at 45^{o} (W/m^2)**
2	3470300	1772600
4	3081100	1573800
6	2691900	1375000
8	2302700	1176200
10	1913500	977400
12	1524300	778600
14	1135100	579800
16	745900	381000
18	356700	182200
20	0	0

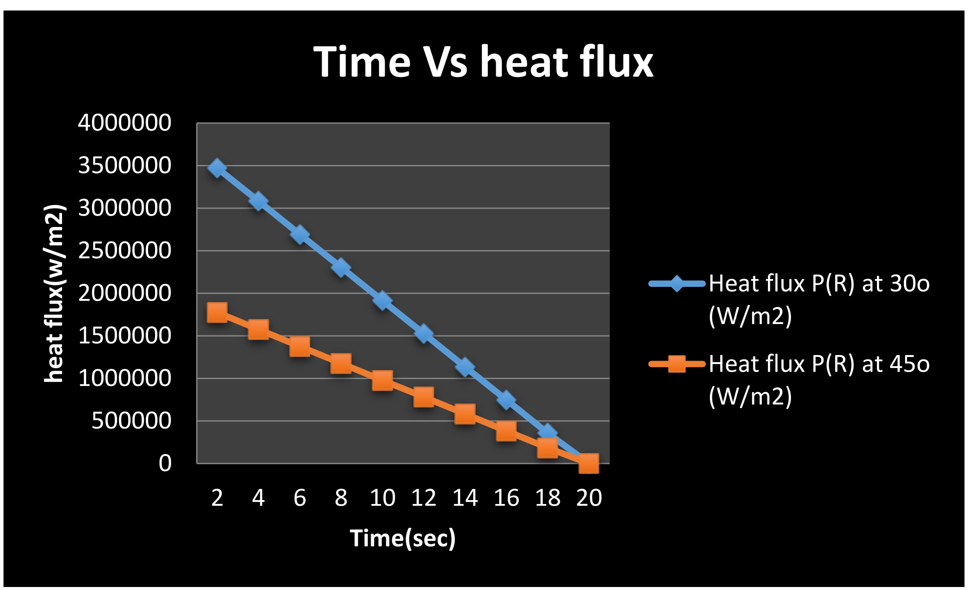

Figure 5.2 Time Vs heat flux graph during emergency brake on a wheel side have one brake block

Table 5.5 Input data at service brake when on a wheel side have two and one brake block

No	Input	Value	
1	Convective film coefficient	$177W/m^2/^oC$	
2	Acceleration	$1.1m/s^2$	
3	Braking time	25sec	
4	Rotational speed	66.11rad/sec	
5	Gravitational acceleration	$9.81m/s^2$	
6	Pressure due to brake force on one wheel side have two brake block	72KPa for $\theta=30^O$	39KPa for $\theta=45^O$
7	Pressure due to brake force on one wheel side have one brake block	144.6KPa for $\theta=30^O$	78.48KPa for $\theta=45^O$

Table 5.6 Heat flux at different time intervals during service brake when on a wheel side have two brake blocks.

Time (sec)	**Heat flux P(R) at 30^o (W/m^2)**	**Heat flux P(R) at 45^o (W/m^2)**
2.5	1375891	700453.6
5	1224641	623453.6
7.5	1073391	546453.6
10	922141	469453.6
12.5	770891	392453.6
15	619641	315453.6
17.5	468391	238453.6
20	317141	161453.6
22.5	165891	84453.6
25	0	0

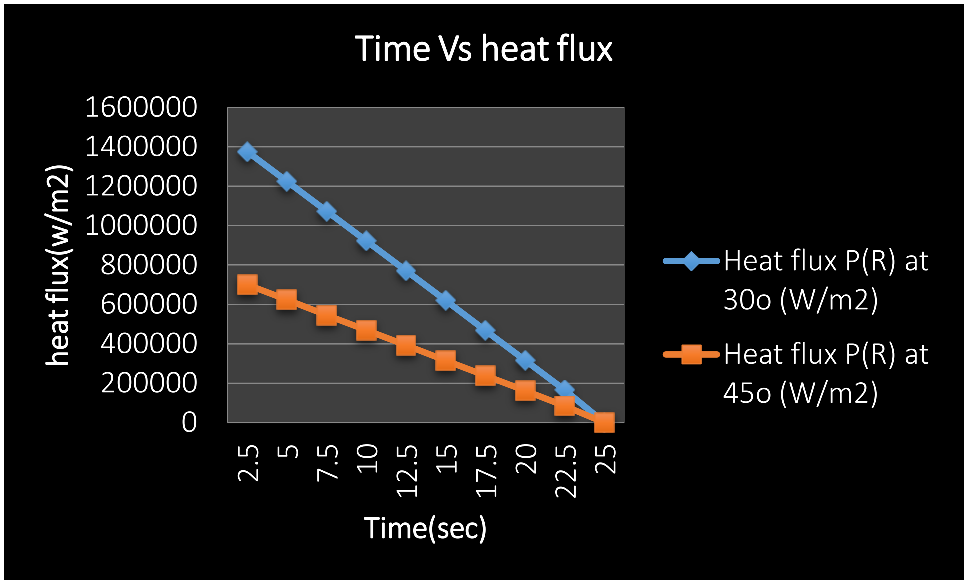

Figure 5.3 Time Vs heat flux graph during service brake on a wheel side have two brake block

Table 5.7 Heat flux at different time intervals during service brake when on a wheel side have one brake block

Time (sec)	Heat flux P(R) at 30^{0} (W/m^2)	Heat flux P(R) at 45^{0} (W/m^2)
2.5	2726765.8	1413415.3
5	2427015.8	1258040.3
7.5	2127265.8	1102665.3
10	1827515.8	947290.3
12.5	1527765.8	791918.3
15	1228015.8	636540.3
17.5	928265.8	481165.3
20	628515.8	325790.3
22.5	328765.8	84453.6
25	0	0

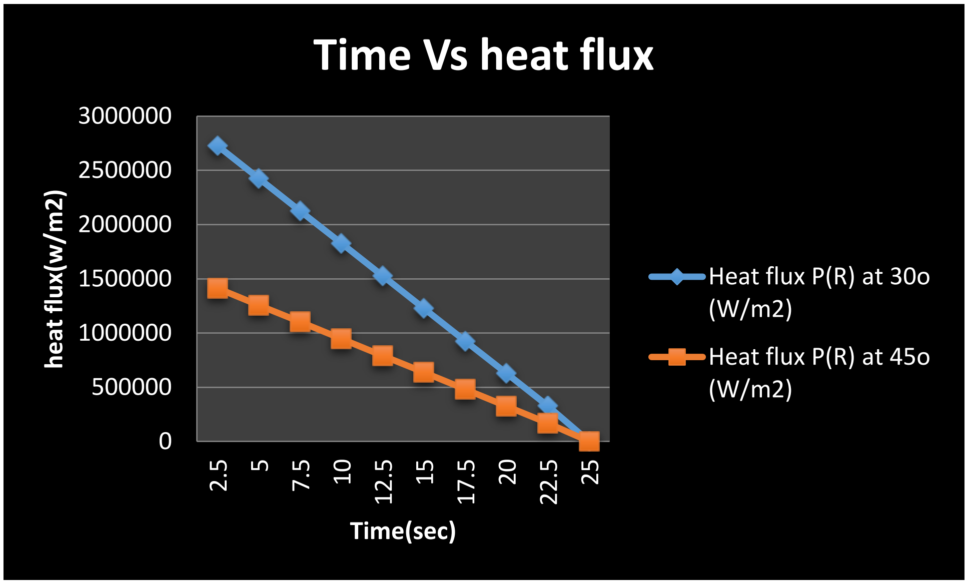

Figure 5.4 Time Vs heat flux graph during service brake on a wheel side have one brake block

5.2.1 Temperature rise at brake block during emergency and service brake on a wheel side have two brake block

The brake shoe profile is circular area and the contact surface area with the wheel the more affected area of the brake shoe. But the maximum temperature rise occurs at the middle of the contact between wheel and the brake 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 sebeta- adama line at the maximum gradient 18.5‰ and maximum temperature rise also occurs at this station. The initial surrounding temperature of the brake shoe (environmental temperature) is 25℃. The temperature of the surrounding is still 25℃. In case of braking the train at this station the highest temperatures during emergency brake reach up to 459.1℃ for the contact angle $30^{O}$and 245.2℃ for 45^{O}.This temperature is reached after a time period of 20 s. the highest temperatures during service brake reach up to 412.86℃ for the contact angle $30^{O}$and 222.45℃ for 45^{O}.This temperature is reached after a time period of 25 s.

The contact area of brake block affect by the temperature during braking. This is also the place where the highest temperatures are achieved. The figures show how the temperatures toward the brake block contact surface. This information is needed to determine the influence of the heat flux on the brake block.

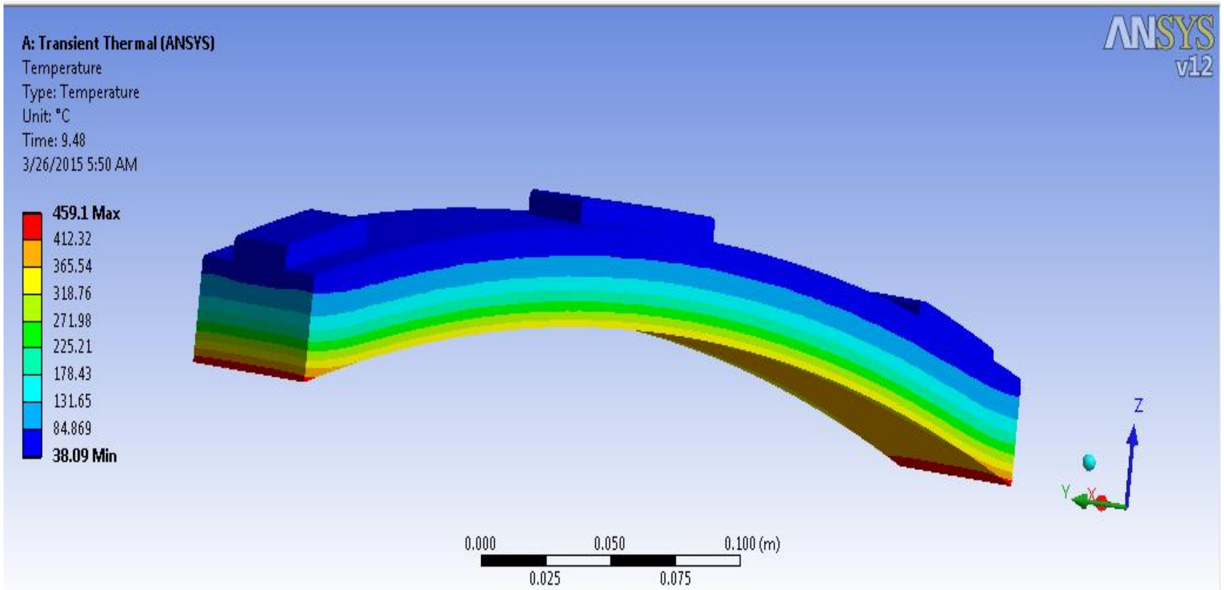

Figure 5.5 Maximum temperature rise at the brake block during emergency brake for $\theta=30^0$

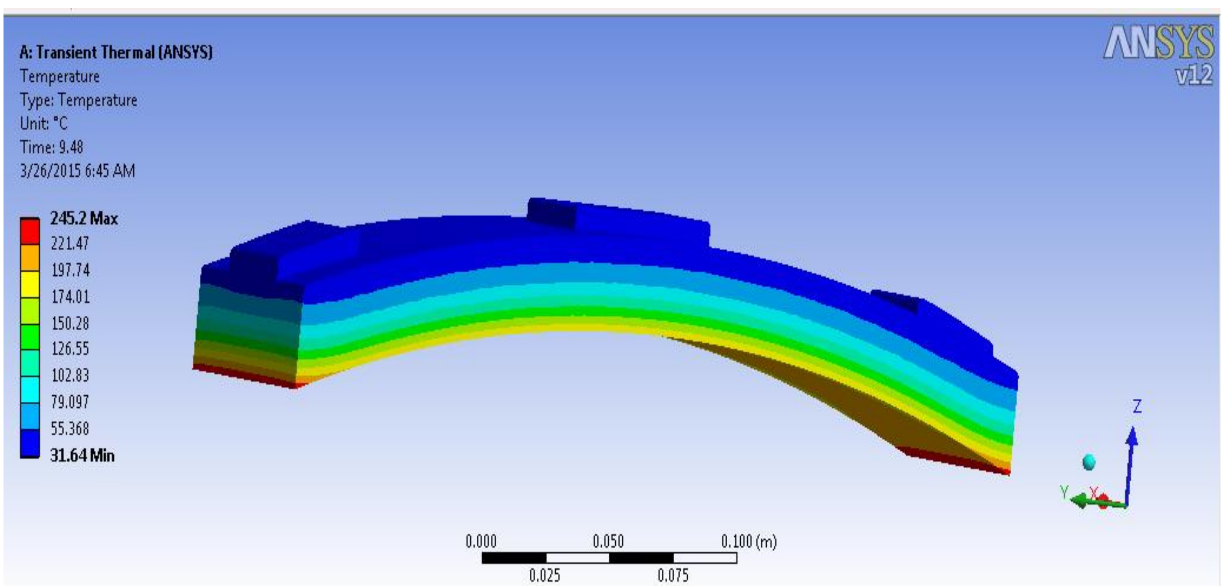

Figure 5.6 Maximum temperature rise the brake block during emergency brake for $\theta=45^0$

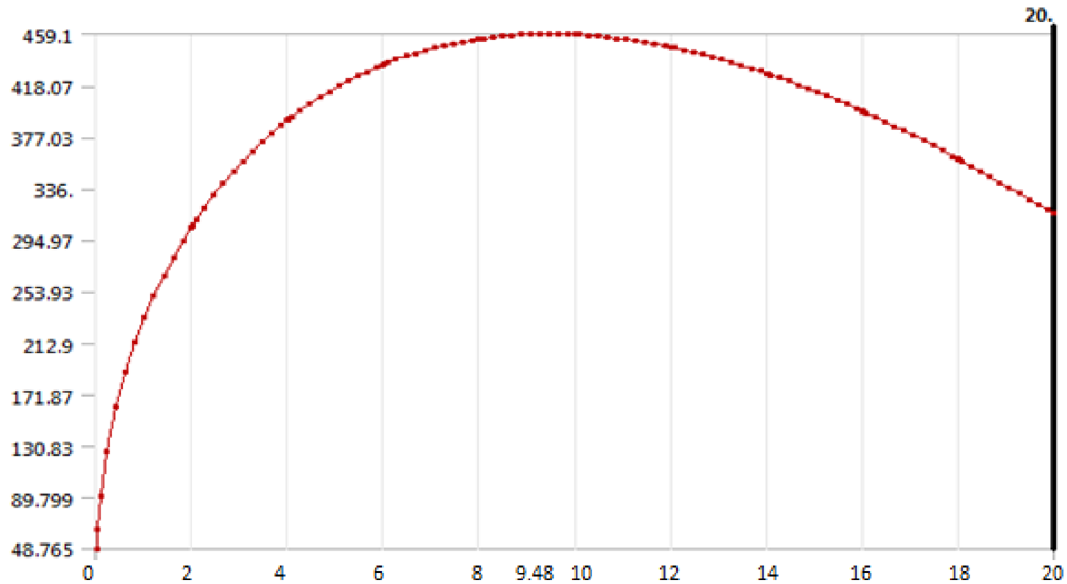

Figure 5.7 Temperature rise graph during emergency brake for $\theta=30^{o}$

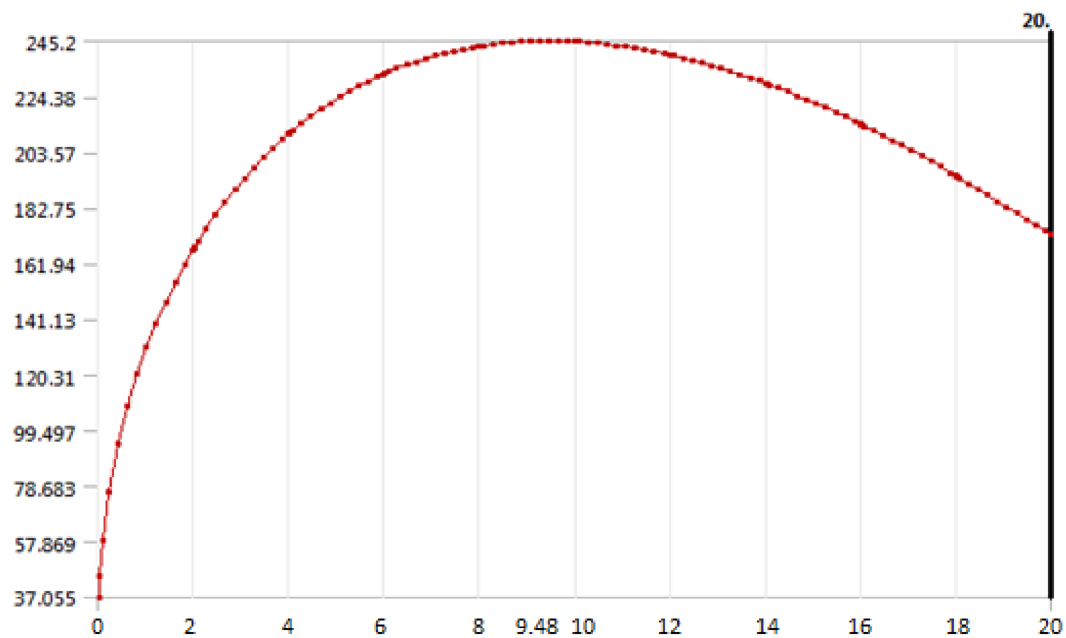

Figure 5.8 Temperature rise graph during emergency brake for $\theta=45^{o}$

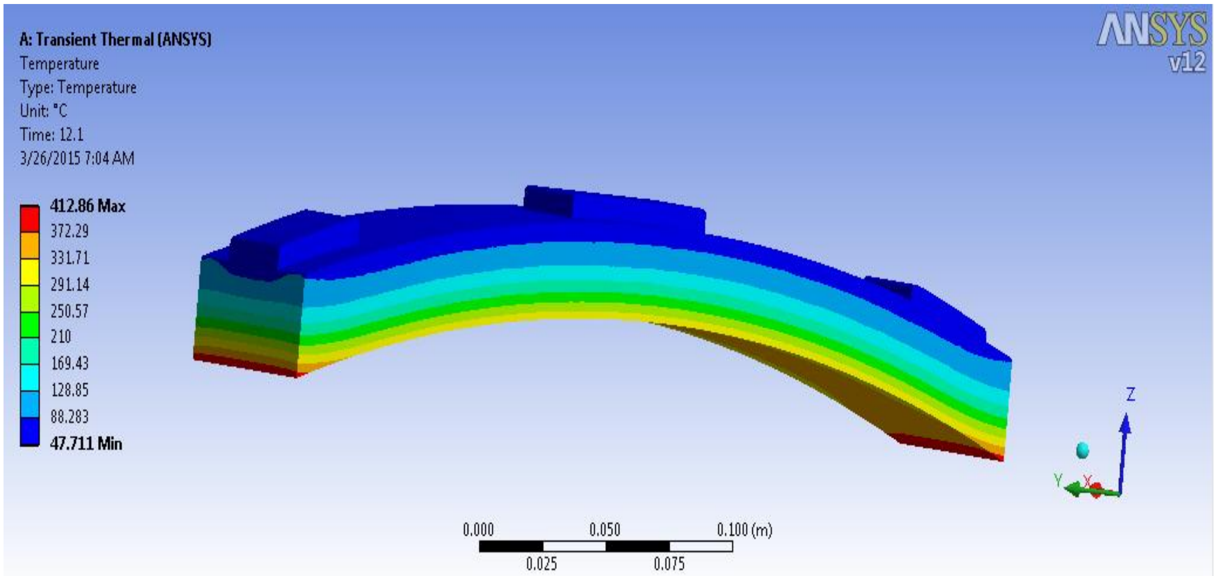

Figure 5.9 Maximum temperature rise at the brake block during service brake for $\theta=30^0$

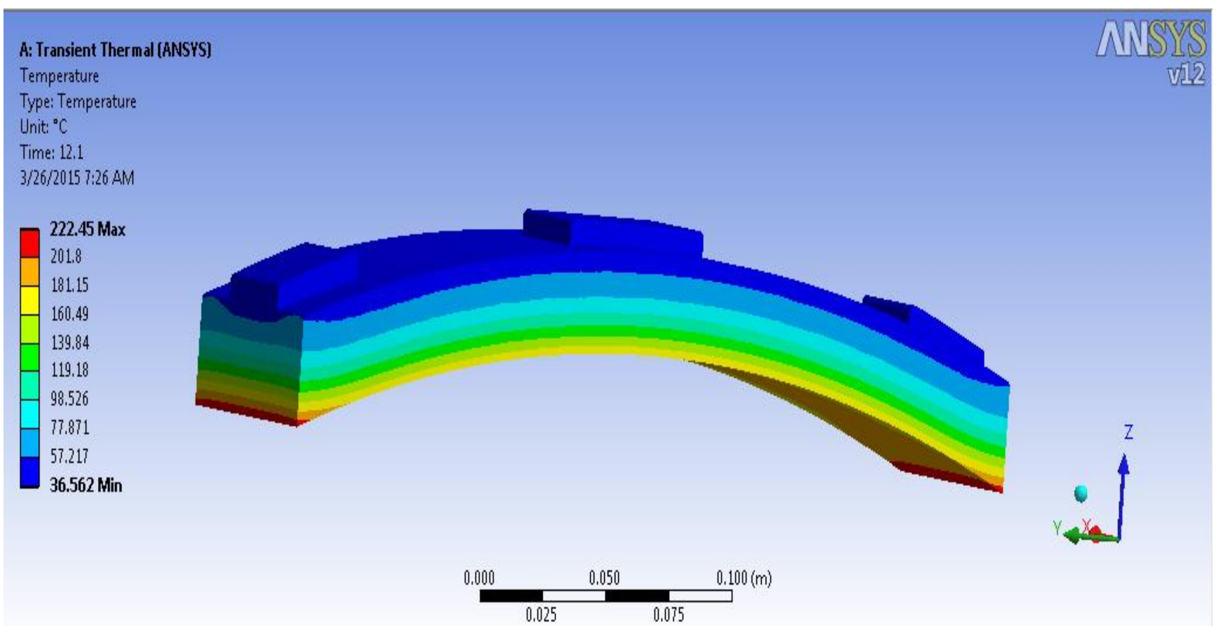

Figure 5.10 Maximum temperature rise at the brake block during service brake for $\theta=45^0$

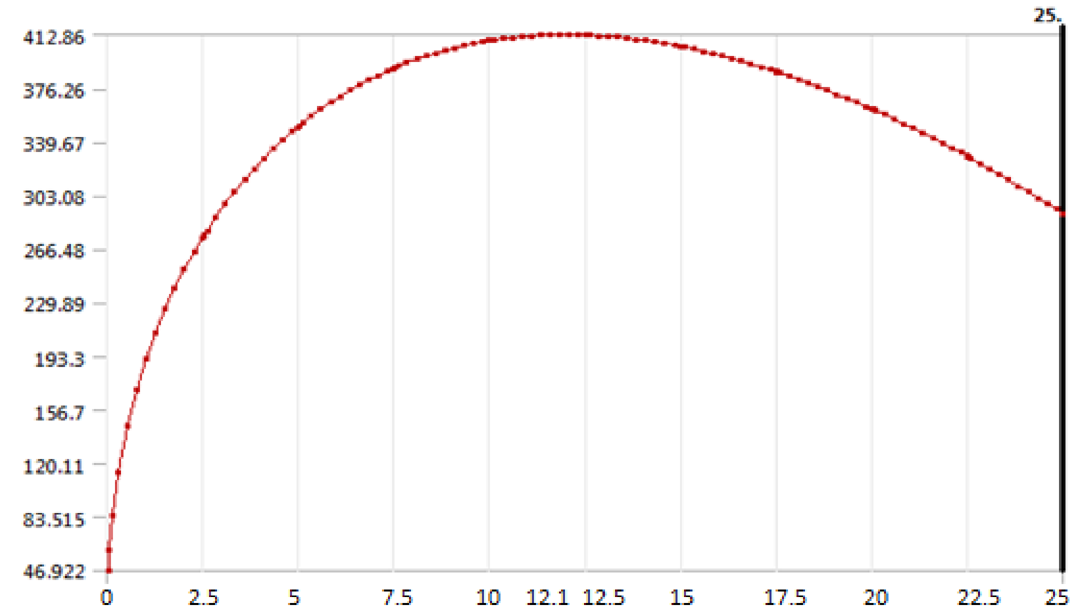

Figure 5.11 Temperature rise graph during service brake for θ=30^{O}

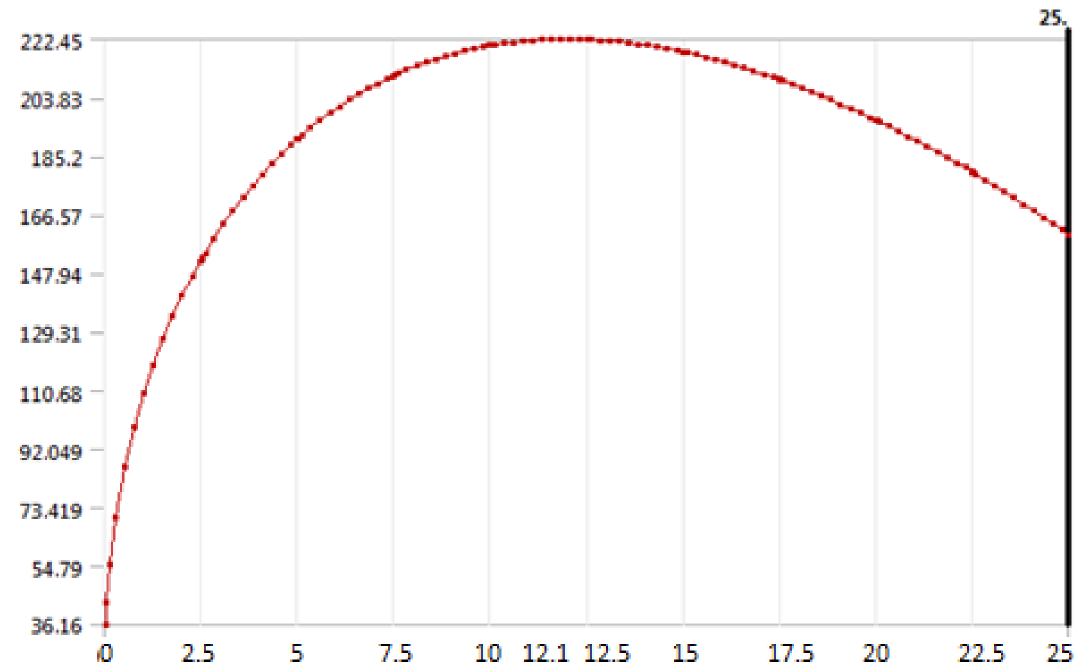

Figure 5.12 Temperature rise graph during service brake for θ=45^{O}

5.2.2 Temperature rise at brake block during emergency and service brake on a wheel side have one brake block.

The same way of calculating temperature rise at brake block for wheel side have one brake block as wheel side have two brake blocks. In case of braking the train at this station the highest temperatures during emergency brake reach up to 899.49℃ for the contact angle $30^{\circ}$and 471.68℃ for 45°.This temperature is reached after a time period of 20 s. the highest temperatures during service brake reach up to 793.66℃ for the contact angle $30^{\circ}$and 423.44℃ for 45°.This temperature is reached after a time period of 25 s.

Table5.8 Temperature rise at brake block on a wheel side have one brake block.

Braking action	**Temperature rise at 30^0 contact surface of the brake block**	**Temperature rise at 45^0 contact surface of the brake block**
Emergency brake	899.49℃	471.68℃
Service brake	793.66℃	423.44℃

5.2.3 Heat flux distribution on the surface of the brake block on a wheel side have two brake block

As in the case of temperature rise the maximum heat flux occurs at the brake block due to the contact between wheel and brake block. The maximum heat flux distribution occurs at the surface of the brake block due to its contact surface area at different angle with diameter of the wheel tread. Since the heat flux is directly related to the temperature the maximum heat flux occurs at the station on the sebeta-adama line at maximum gradient 18‰. Therefore from the above maximum temperature rise graph heat flux on the brake block during emergency brake is about $1.0697*10^6\ W/m^2$ for θ=30° and for θ= 45° reaches about $5.426*10^5\ W/m^2$. During service brake is about $8.4606*10^5\ W/m^2$ for θ=30° and for θ= 45° reaches about $4.3072*10^5\ W/m^2$.

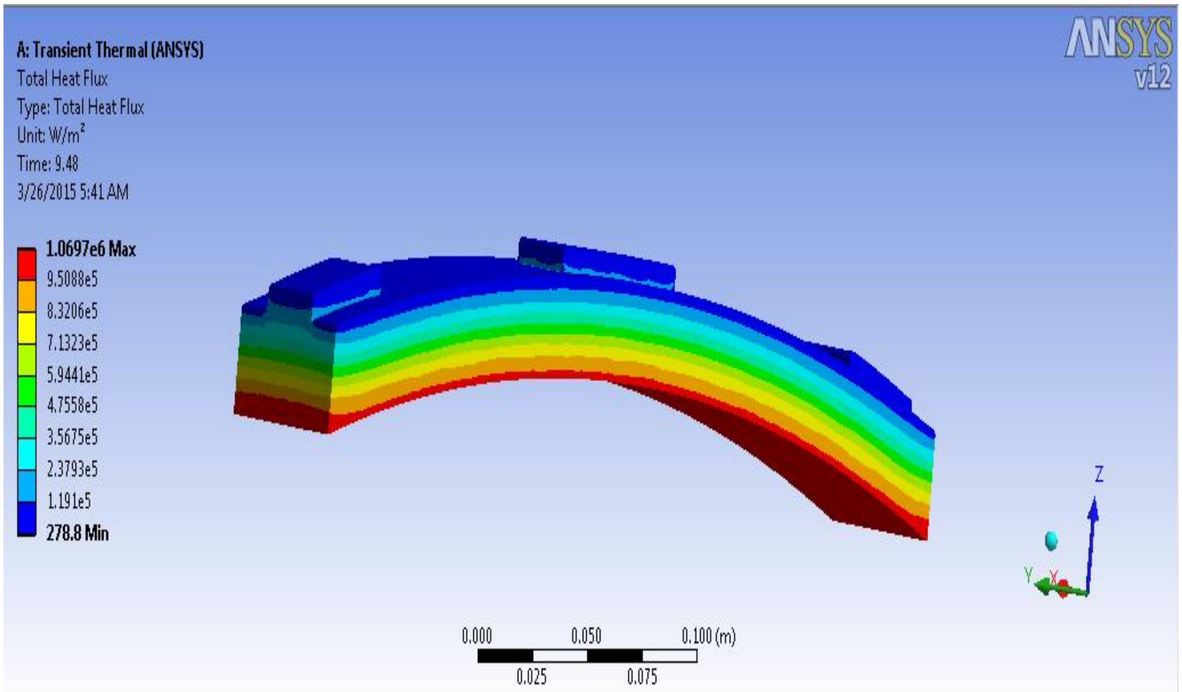

Figure 5.13 Heat flux distribution at the brake block during emergency brake for $\theta=30^{o}$

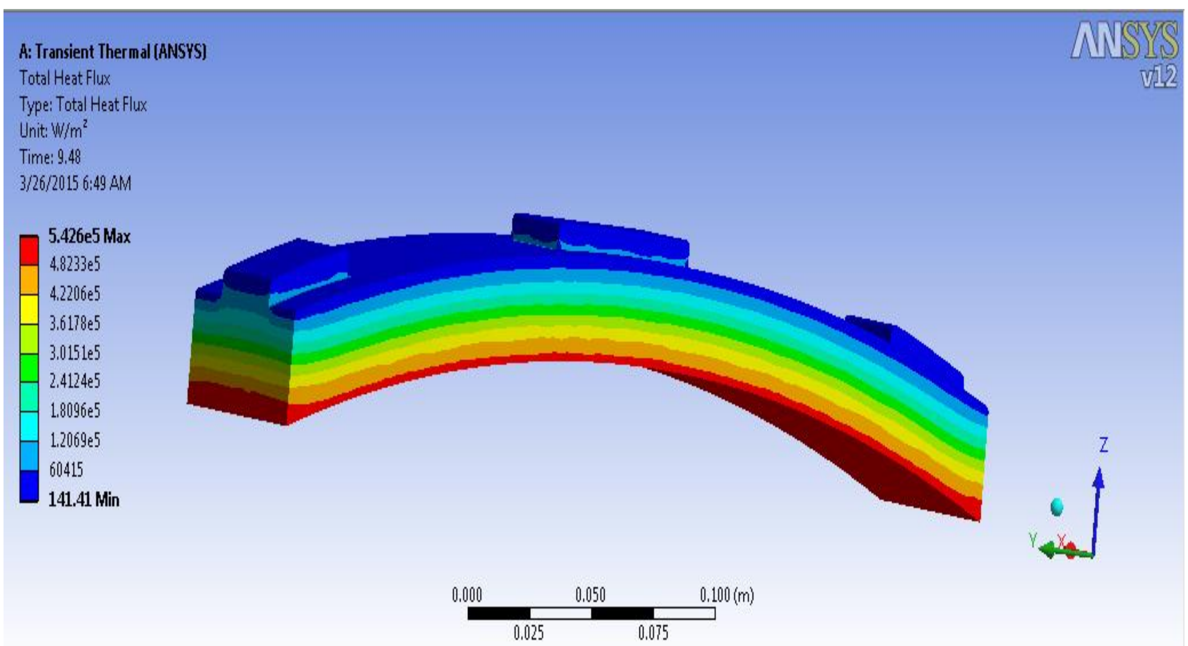

Figure 5.14 Heat flux distribution on the brake block during emergency brake for $\theta=45^{o}$

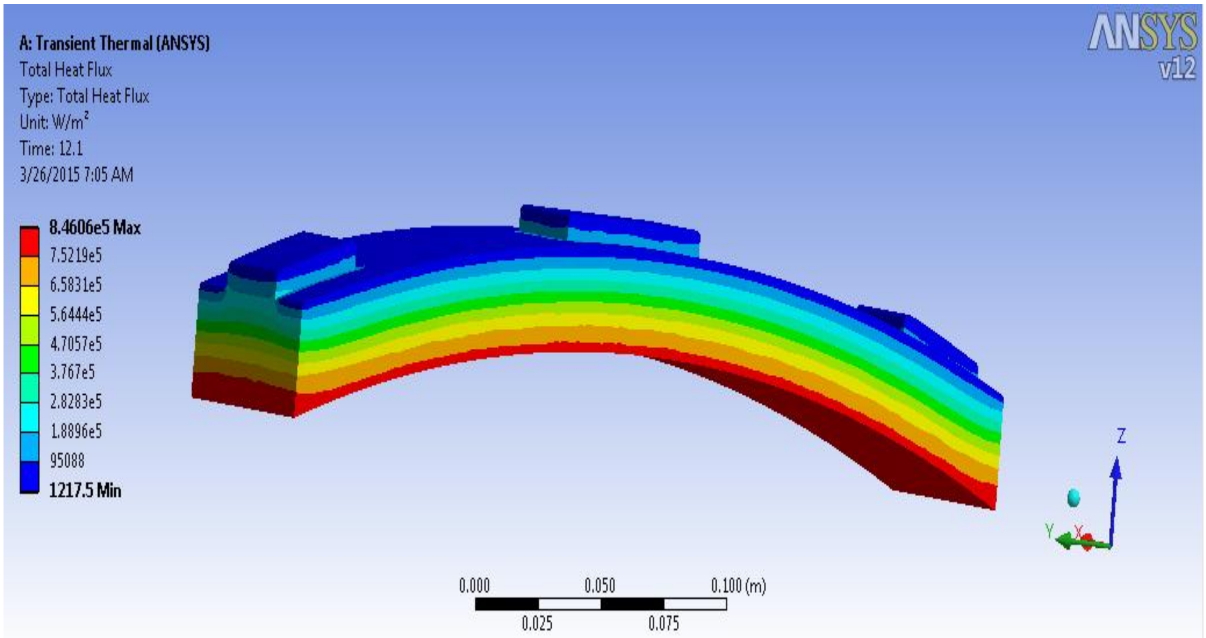

Figure 5.15 Heat flux distribution at the brake block (shoe) during service brake for θ=30^{o}

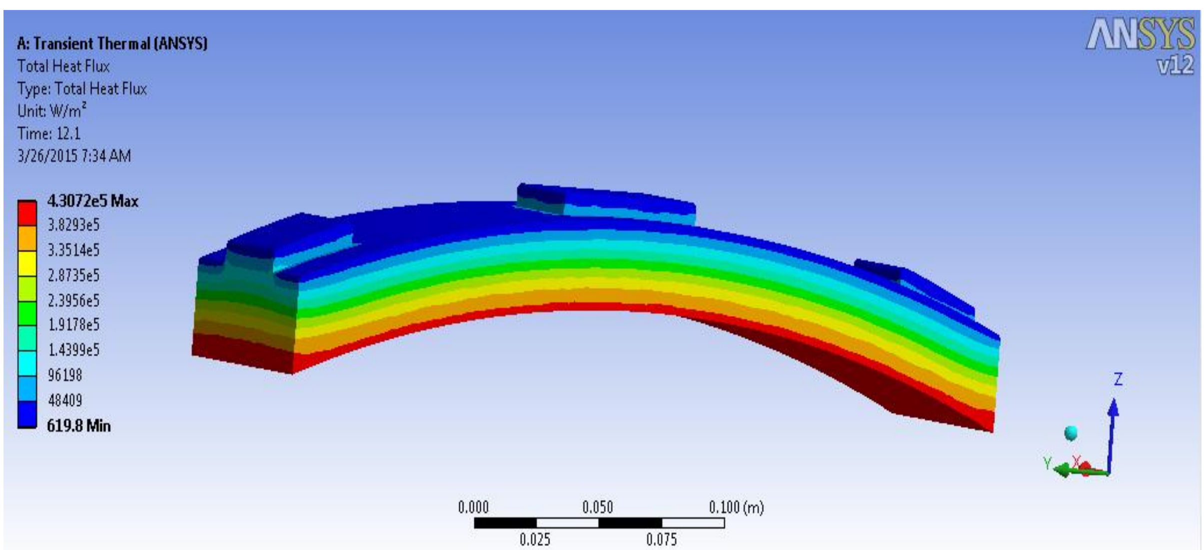

Figure 5.16 Heat flux distribution at the brake block (shoe) during service brake for θ=45^{o}

5.2.4 Heat flux distribution on the surface of the brake block on a wheel side have one brake block

The same way of ANSYS analysis result of heat flux distribution on the surface of brake block for wheel side have one brake block as wheel side have two brake blocks. As in the case of temperature rise the maximum heat flux occurs at the brake block due to the contact between wheel and brake block.

Therefore from the above maximum temperature rise graph heat flux on a wheel side have one brake block during emergency brake is about 2.1549*10^6 W/m^2 for θ=30^o and for θ= 45^o reaches about 1.1007*10^6 W/m^2. During service brake is about 1.6767*10^6 W/m^2 for θ=30^o and for θ= 45^o reaches about 8.6913*10^5 W/m^2.

Table5.9 heat flux distribution on the surface of the brake block on a wheel side has one brake block.

Braking action	Heat flux distribution at 30^0 contact surface of the brake block	Heat flux distribution at 45^0 contact surface of the brake block
Emergency brake	2.1549*10^6 W/m^2	1.1007*10^6 W/m^2
Service brake	1.6767*10^6 W/m^2	8.6913*10^5 W/m^2

5.3Mechanical results

5.3.1 Transient Structural during emergency and service brake on a wheel side have two brake block

Thermal stresses in the brake block appear because the temperatures rise. Beside the thermal stress, the holding force of the brake block is also considered. The goal of this analysis is to determine the influence of the pressure 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 mises) stress imposed on the brake block. From the output result of the analysis the maximum equivalent (Von mises) stress induced on the brake block during emergency brake is about $5.3568*10^8 N/m^2$(535.68Mpa)for $\theta=30^o$and $3.4325*10^8 N/m^2$(343.25Mpa)for $\theta=45^o$ respectively and during service brake is about $4.5954*10^8 N/m^2$(459.54Mpa)for $\theta=30^o$and $3.0735*10^8$ (307.35Mpa)for $\theta=45^o$.

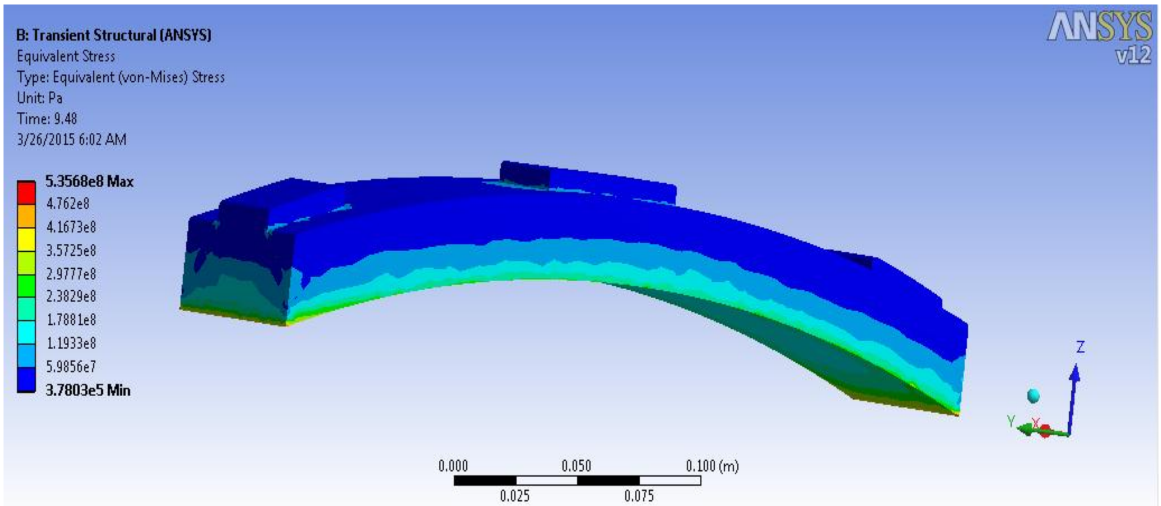

Figure 5.17 Equivalent (Von-mises) stress during emergency brake for $\theta=30^O$

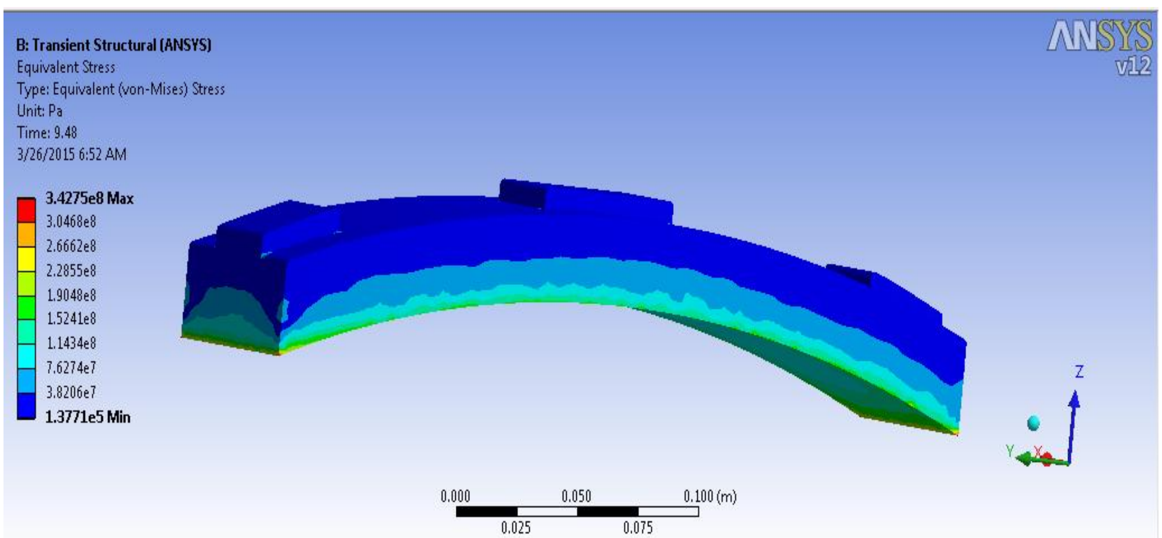

Figure 5.18 Equivalent (Von-mises) stress during emergency brake for $\theta=45^O$

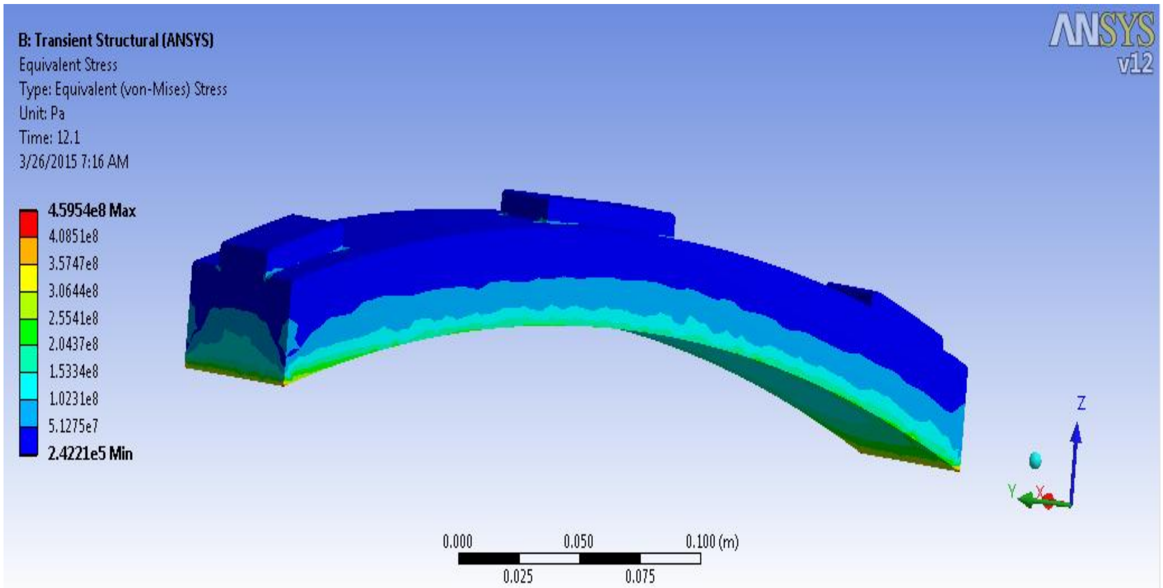

Figure 5.19 Equivalent (Von-mises) stress during service brake for θ=30^{O}

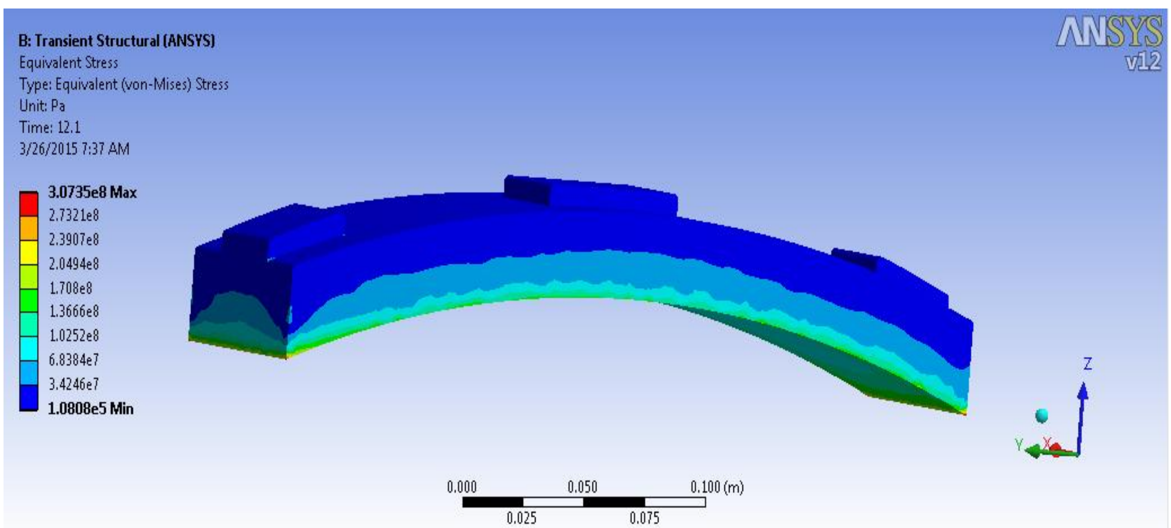

Figure 5.20 Equivalent (Von-mises) stress during service brake for θ=45^{O}

5.3.2 Transient Structural during emergency and service brake on a wheel side have one brake block

The same way of ANSYS analysis result of thermal stresses in the brake block due to the temperatures rise for a wheel side have one brake block as a wheel side have two brake blocks. The goal of this analysis is to determine the influence of the pressure 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. From the output result of the analysis the maximum equivalent (Von mises) stress induced on the brake block during emergency brake is about $1.1264*10^9 N/m^2$(1126.4Mpa)for $\theta=30^o$and $7.402*10^8 N/m^2$(740.2Mpa)for $\theta=45^o$ respectively and during service brake is about $9.2645*10^8 N/m^2$(926.45Mpa)for $\theta=30^o$and $6.6075*10^8$ (660.75Mpa)for $\theta=45^o$.

Table5.10 maximum equivalent (Von-mises) stress induced on the brake block

Braking action	**Max. equivalent (Von mises) stress at 30^0 on the brake block**	**Max. equivalent (Von mises) stress at 45^0 on the brake block**
Emergency brake	1126.4Mpa	740.2Mpa
Service brake	926.45Mpa	660.75Mpa

5.3.3Total Deformation and strain on a wheel side have two brake block

The thermal and mechanical loads on the brake shoe do not induce only stress on the brake shoe but also deforms the brake shoe surface. This deformation may also result in the other defects such as brake shoe crack, wear of the brake shoe surface and other defects. According to the analysis result the maximum and the minimum resulting deformation during emergency brakes are 0.0000339m and 0m respectively for $\theta=30^o$ and for $\theta=45^o$ it is 0.0000204m and 0m respectively. During service brakes are 0.0000283m and0m respectively for θ=30and for θ=45 it is0.0000188mand 0m respectively.

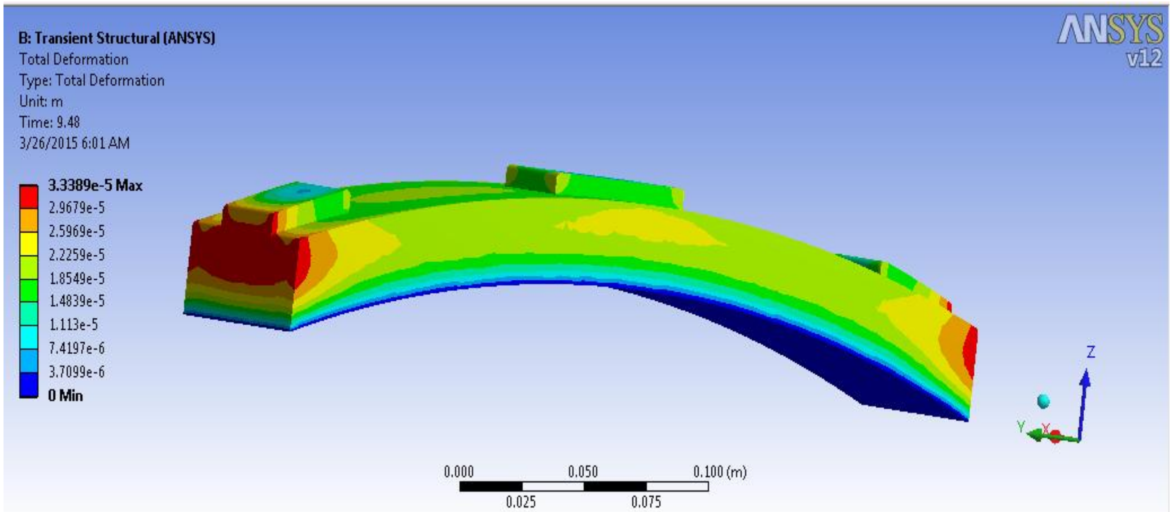

Figure 5.21 Total deformation of the brake block surface during emergency brake for θ=30^{O}

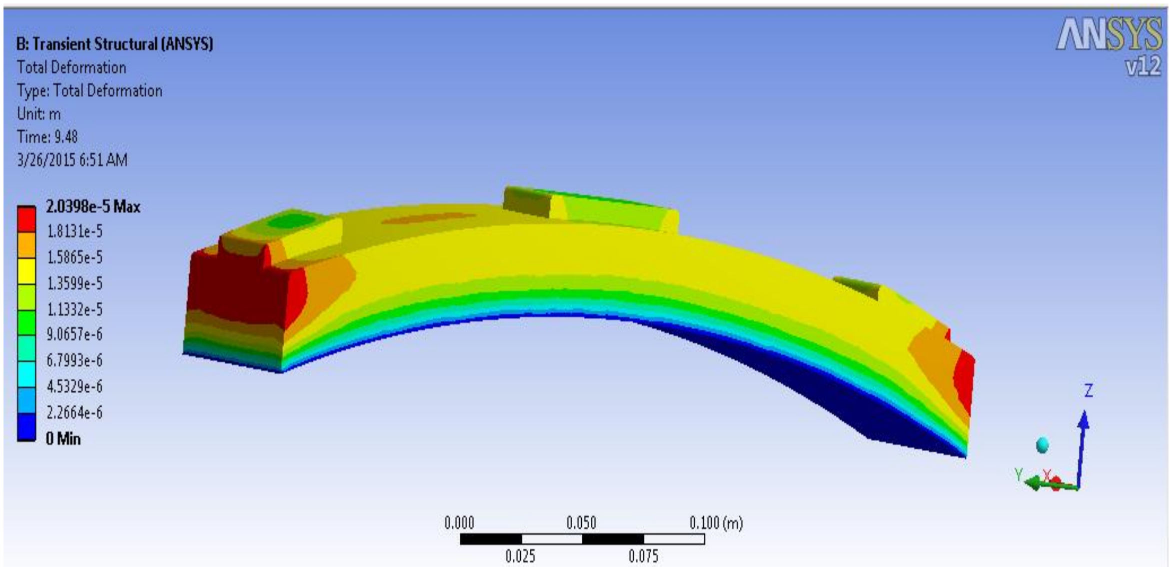

Figure 5.22 Total deformation of the brake block surface during emergency brake for θ=45^{O}

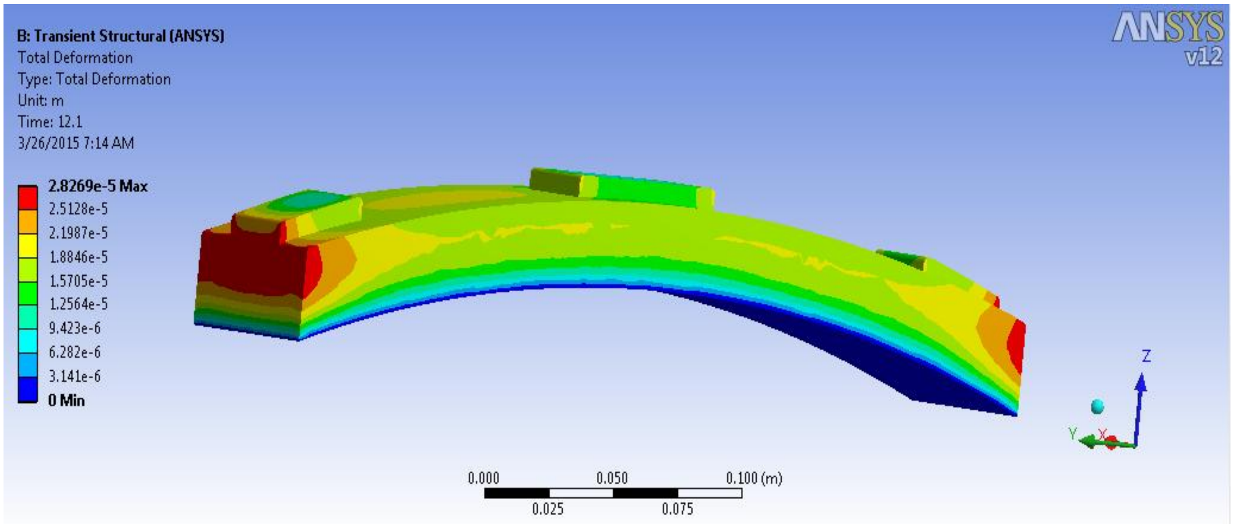

Figure 5.23 Total deformation of the brake block surface during service brake for θ=30^{O}

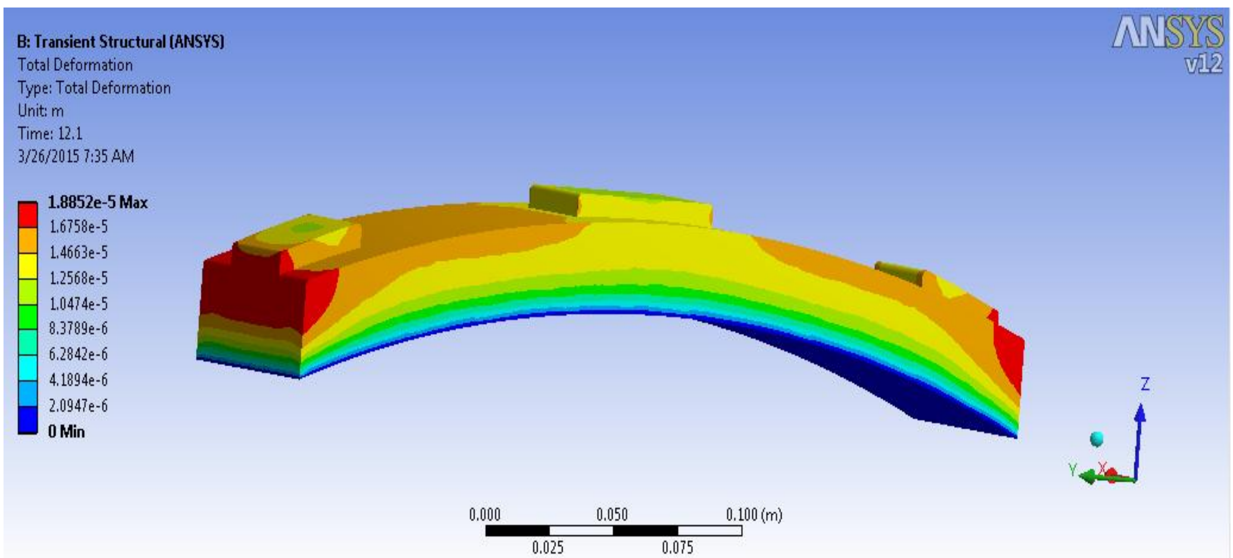

Figure 5.24 Total deformation of the brake block surface during service brake for θ=45^{O}

The strain of the railway brake block is the ratio of the total stress to the deformation of the brake block. 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 during emergency brake for θ=30^{o} is 0.0048698 and $3.4367*10^{-6}$ respectively and for θ=45^{o} the resulting total strain is about 0.0031205 and $1.2519*10^{-6}$ respectively. During service brake for θ=30^{o} is 0.0041776 and $2.2019*10^{-6}$ respectively and for θ=45^{o} the resulting total strain is about 0.0027941 and $9.8254*10^{-7}$ respectively.

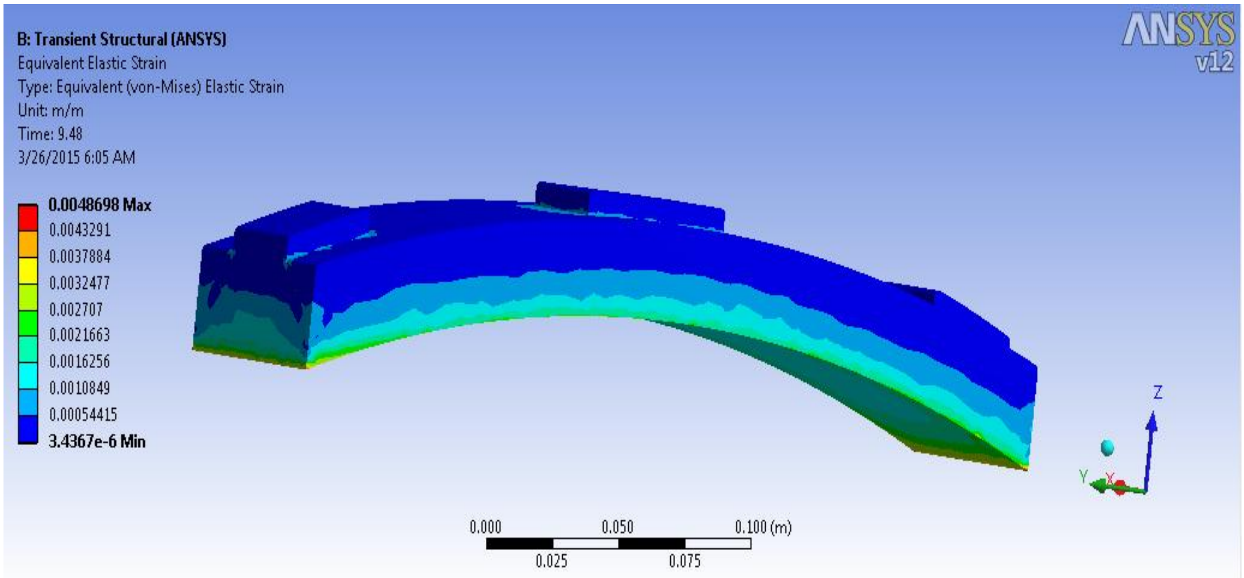

Figure 5.25 Total strain of the brake block (shoe) during emergency brake for θ=30^{o}

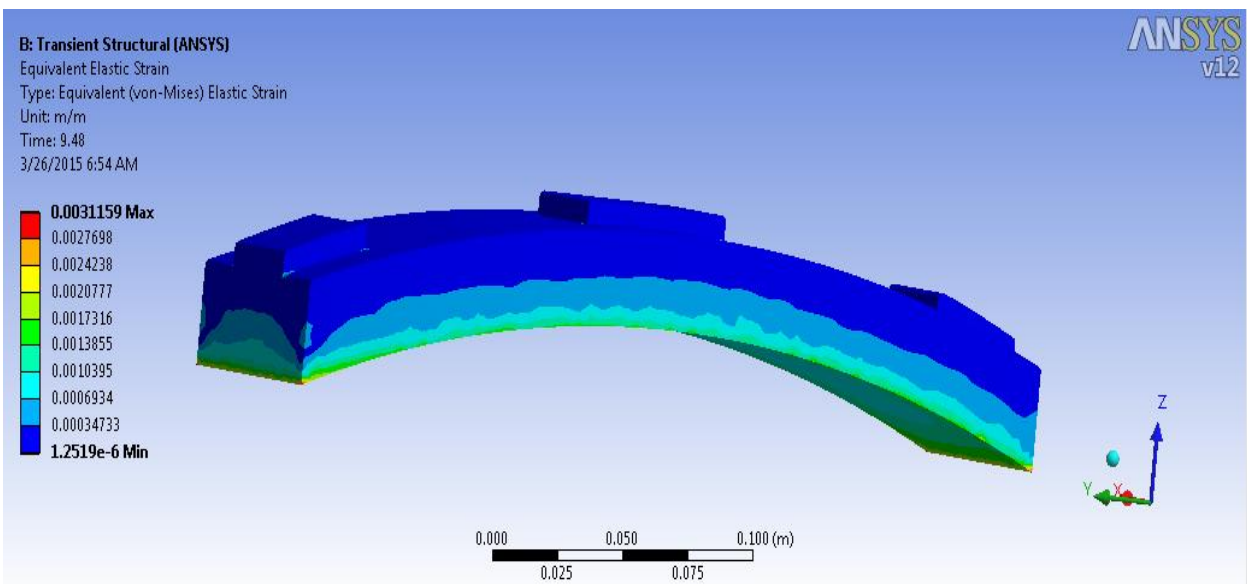

Figure 5.26 Total strain of the brake block (shoe) during emergency brake for θ=45^{o}

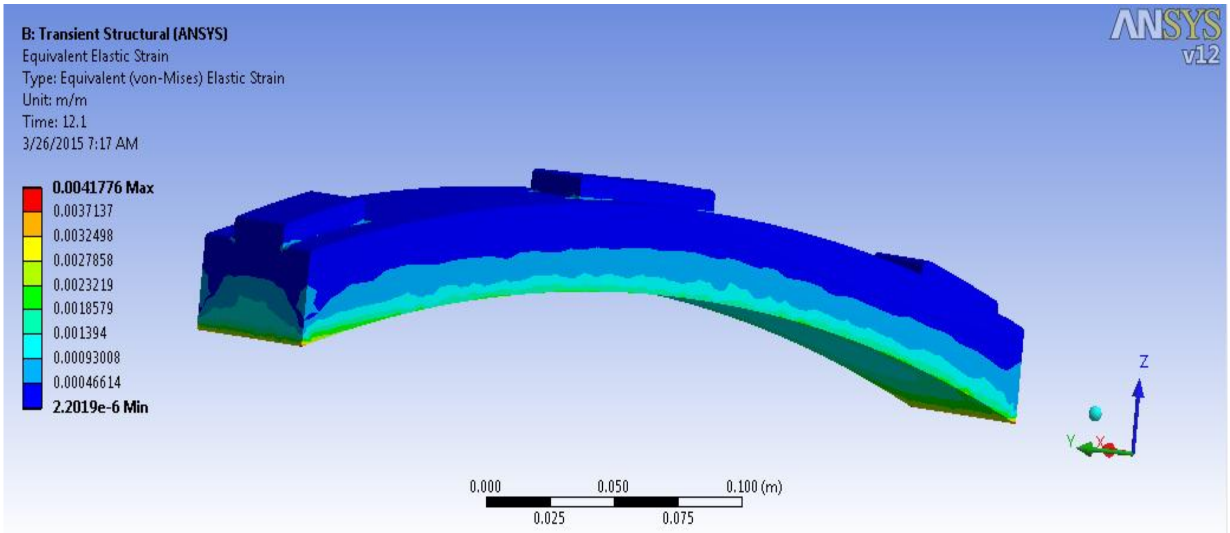

Figure 5.27 Total strain of the brake block (shoe) during service brake for θ=30^{o}

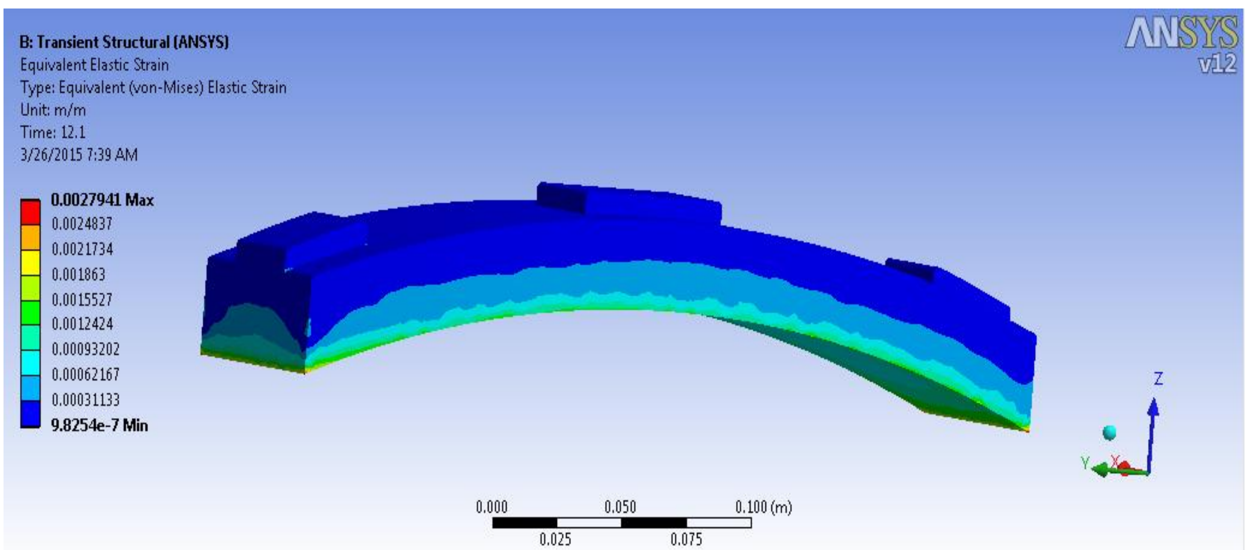

Figure 5.28 Total strain of the brake block (shoe) during service brake for θ=45^{o}

5.3.4Total Deformation and strain on a wheel side have one brake block

The same way ANSYS analysis of total deformation and strain on a wheel side have one brake block as a wheel side have two brake block. The thermal and mechanical load deforms the brake block surface and the strain of the railway brake block is the ratio of the total stress to the deformation of the brake block. According to the analysis result the maximum resulting deformation during emergency brakes are 0.0000690m for $\theta=30^{o}$ and for $\theta=45^{o}$ it is 0.0000439m and. During service brakes are 0.0000568m for $\theta=30$ and for $\theta=45$ it is0.0000412m. The strain result given according the ANSYS output result of the analysis the maximum result of the strain during emergency brake for $\theta=30^{o}$ is 0.01024 and for $\theta=45^{o}$ the resulting total strain is about 0.00673 . During service brake for $\theta=30^{o}$ is 0.00842 and for $\theta=45^{o}$ the resulting total strain is about 0.0060.

Table5.11 Maximum deformation and strain on the surface of brake block

Braking action	**Max. deformation at 30^{0} on the brake block**	**Max. deformation at 45^{0} on the brake block**	**Max. strain at 30^{0} on the brake block**	**Max. strain at 45^{0} on the brake block**
Emergency brake	0.0000690m	0.0000439m	0.01024	0.00673
Service brake	0.0000568m	0.0000412m	0.00842	0.0060

Chapter six

Conclusion, Recommendation and Future work

6.1 Conclusion

The thermal and stress analysis of a brake block for freight rail way vehicles during emergency and service braking at a speed 100km/hr with considering the convective film coefficient and heat flux were studied using FEM. The results revealed brake block can effectively enhance the thermal performance of the brake block; Since the freight box cars has two bogies and in one wheel side have two brake blocks the highest temperature after emergency braking was 316.5℃ contact surface area 30^o and 172.86℃ at contact surface area of 45^o, which occurred at 9.48second and service braking, was 291.07℃ contact surface area 30^o and 160.45℃ at contact surface area of 45^o, which occurred at 12.1second. In one wheel side have one brake block the highest temperature after emergency braking was 899.49℃ contact surface area 30^o and 471.68℃ at contact surface area of 45^o, which occurred at 9.48second and service braking, was 793.66℃ contact surface area 30^o and 423.44℃ at contact surface area of 45^o, which occurred at 12.1second.

The heat flux on the train increases and as a result the rise in the temperature of brake block. The decrease of contact angle between the brake block (shoe) and wheel tread also increase the temperature rise heat flux distribution over the surface of brake block in a dimensioning fashion. In one wheel side have two brake blocks the maximum equivalent (Von mises) stress induced on the brake block during emergency brake is about $5.3568*10^8$N/m^2(535.68Mpa)for θ=30^{o}and $3.4325*10^8$**N**/m^2(343.25Mpa)for θ=45^o respectively and during service brake is about $4.5954*10^8$**N**/m^2(459.54Mpa)for θ=30^{o}and $3.0735*10^8$ (307.35Mpa)for θ=45^o. In one wheel side have one brake block the maximum equivalent (Von mises) stress induced on the brake block during emergency brake is about $1.1264*10^9$N/m^2(1126.4Mpa)for θ=30^{o}and $7.402*10^8$**N**/m^2(740.2Mpa)for θ=45^o respectively and during service brake is about $9.2645*10^8$**N**/m^2(926.45Mpa)for θ=30^{o}and $6.6075*10^8$ (660.75Mpa)for θ=45^o.This is due to the large difference of compressive pressure load between the 30^o and 45^o contact surface area of brake block. In one wheel side have two brake block the maximum deformed and strain to a part of the brake block (shoe) result from cumulative effect of the thermal and pressure load applied on the brake block during emergency brake the deformation are 0.0000339m for θ=30 and 0.0000204m for θ=45 and the maximum strain for $\theta=30^o$ is 0.0048698 and for $\theta=45^o$ is 0.0031205. During service brake the deformation are 0.0000283m for $\theta=30^0$ and 0.0000188m for $\theta=45^0$ and the maximum strain for $\theta=30^o$ is 0.0041776m and for $\theta=45^o$ is 0.002794m. In one wheel side have one brake block the maximum deformed and strain to a part of the brake block (shoe) during emergency brake the deformation are 0.0000690m for θ=30 and 0.0000439m for θ=45 and the maximum strain for $\theta=30^o$ is 0.01024 and for $\theta=45^o$ is 0.00673. During service brake the deformation are 0.0000568m for $\theta=30^0$ and 0.0000412m for $\theta=45^0$ and the maximum strain for $\theta=30^o$ is 0.00842m and for $\theta=45^o$ is 0.0060m.As we have seen from the above analysis result when comparing and contrast on one wheel side have two brake block better than a wheel side have one brake block for friction brake block of freight train AA-Djibouti main line.

6.2Recommendation

In this paper, we have made to analyze brake block of freight railway using analytical and numerical modeling of thermal effects during slope on down-grade rail road. Hence, we recommend that from the analysis as follow.

- At lower angle of contact between the brake block and the wheel tread is good, compact size. But it has the ability to raise the temperature of the brake block. This affects the life of the brake block and wheel tread.
- At higher angle of contact between the brake block and the wheel tread is not a compact size. Since it decreases the temperature rise between brake block and wheel tread while increasing the contact angle, it is better to use it with higher contact angle.

- The 45^0 contact angle is the optimum value to use because the temperature rise as this angle does not affect severely the maximum operating temperature of the brake block and decrease the heat flux distribution over surface area of the brake block.
- The designing the stations to be with minimum gradient as much as possible.

6.3 Future work

In this paper the temperature rise effect on brake block at different contact surface area are studied using FEM. In the future work, it is possible to work the effect of wear and tribological effect by changing the material properties of the brake block and to investigate better material may improve the braking performance.

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