ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY (AAiT)

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

GRADUATE PROGRAM IN RAILWAY ENGINEERING

IMPROVEMENT OF TREAD BRAKE HEAT DISSIPATION USING FORCED AIR COOLING

A Thesis Submitted to the School of Graduate Studies of Addis Ababa University in Partial Fulfillment of the Requirement for the Degree of Masters of Science

Master of Science

In

Mechanical Engineering

(Under Railway Engineering)

By: Berhe Hishe

Advisor: Dr. Ir. Demiss Alemu

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ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF GRADUATED STUDIES

POSTGRADUATE PROGRAM IN RAILWAY ENGINEERING

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APPROVED BY BOARD OF EXAMINERS

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DECLARATION

I hereby declare that the work which is being presented in this thesis entitled **IMPROVEMENT OF TREAD BRAKE HEAT DISSIPATION USING FORCED AIR COOLING** is original work of my own, has not been presented for a degree of any other university and all the resource of materials used for this thesis been duly acknowledged.

______________________ ____________________

Berhe Hishe Date

This is to certify that the above declared made by the candidate is correct to the best of my knowledge.

______________________ _____________________

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June/2015

AKNOWLEDGEMENT

First of all, I am very glad to praise the Almighty God who has enabled me complete this study.

I express my deep sense of gratitude towards my advisors; **Dr. Ir. Demis Alemu**. I would like to thank my examiners **Dr.Tessefaye Dama** and **Mr. Tsegaye Feleke** whose guidance and contributions of knowledge to this research led me towards completion.

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I express my cordial gratitude to my parents for their unlimited support. I know that there is no way to repay my debt except make them proud of me.

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ABSTRACT :

Sufficient heat dissipation is crucial to the effective operation of friction based braking systems. Such cooling is generally provided by ensuring a sufficient supply of cooling air to the heated components, hence the aerodynamics in the region of the brake components is extremely important. The objective of the research was to develop an understanding of how forced air vented from the brake pipe and main reserviore could be used to improve the cooling of train tread brake.

The thermal analyses of tread brake during emergency braking at a speed of 100 km/h considering forced air cooling were investigated using finite element (FE) and computational fluid dynamics (CFD) methods. The results suggested that the higher convection coefficients achieved with forced air cooling will not only reduce the maximum temperature in the braking but also reduce the thermal gradients, since heat will be removed faster from hotter parts of the tread brake.

Forced air cooling should be effective to reduce the risk of hot spot formation and tread wheel thermal distortion. The highest temperature after emergency braking was 432.43 °C and 412.75 °C , 399.13 °C and 384.81°C without and with considering forced air cooling with velocities of (50,100 and 150 m/s) , respectively. The main purpose of this study is to improve the heat dissipation of tread brake by using forced air cooling from the brake pipe and main reservoir.

Key words: heat dissipation, convection heat transfer coefficient, temperature, total heat flux and CFD (computational fluid dynamics).

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CHAPTER ONE

INTRODUCTION

"The modeling of thermal effects has become increasingly important in product design in different transport means, road vehicles, airplanes, railway vehicles, and so forth. The thermal analysis is a very important stage in the study of braking systems, especially of railway vehicles, where it is necessary to brake huge masses, because the thermal load of a braked railway wheel prevails compared to other types of loads. In the braking phase, kinetic energy transforms into thermal energy resulting in intense heating and high temperature states of railway wheels. Thus induced thermal loads determine thermo mechanical behavior of the structure of railway wheels. In cases of thermal overloads, which mainly occur as a result of long-term braking on down-grade railroads, the generation of stresses and deformations occurs, whose consequences are the appearance of cracks on the rim of a wheel and the final total wheel defect"[1].

Transient thermal analyses determine temperatures and other thermal quantities that vary over time. The variation of temperature distribution over time is of interest in many applications such as cooling of electronic packages or quenching analysis for heat treatment. Also of interest are the temperature distribution results in thermal stresses that can cause failure. In such cases, the temperatures from a transient thermal analysis are used as inputs to a structural analysis for thermal stress evaluations. Heat generation controlling is a pre-requisite for qualitative weld creation during the friction stir welding process, and it is important to have an adequate mathematical model that is capable of estimating heat generation with satisfying accuracy [1].

Freight car wheels have the toughest job among all the components of a freight car. The wheel must support the weight of the car and steer it on the rails. The wheel must withstand tremendous amounts of abuse from extreme thermal and mechanical stresses caused by such factors as brake shoe friction and high dynamic loading.

In railroad service the wheel acts as a brake drum in addition to supporting lateral and vertical mechanical loads. When brake shoes are applied to the wheel tread, the tread surface is heated due to friction. Severe thermal input into a wheel occurs when a loaded train descends a grade for an extended period of time. Also, the failure of the brake mechanism may keep the brake locked on the tread of the wheels. On these occasions, the wheel rim is heated to a high temperature for an extended period of time. The steel on the tread surface and in the rim under the tread surface gets hotter and tries to expand. At the same time, the expansion of the hotter part of rim is constrained by the colder body of the wheel rim and plate and is in compression. The steel in the hotter part of the wheel rim will have a reduced yield strength and the compressive thermal stress will be higher than the yield strength. Therefore, the hotter part of the wheel rim will yield. However, during the wheel cooling, the situation is reversed. The tread surface and top part of the rim get cooler and shrink first. They are constrained by hotter lower part of rim and are in tension. After cooling and shrinking, the material in the top wheel rim could now be in tension due to the locally yielded material in the rim. As the result, the residual stress could be altered to tensile from compressive that is generated during the heat treatment.

Thermal fatigue due to repeated brake applications may initiate and propagate thermal-cracks. The presence of a thermal crack in a wheel with tensile residual stress in the rim may lead to failure by brittle mode rim fracture initiated from the thermal crack. Therefore, it is important to know the residual stress change in a wheel.

Braking performance of a train can be significantly affected by the temperature rise in the brake components. High temperature during braking may cause brake fade, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally excited vibration. Therefore, it is important to predict the temperature rise of a given brake system and assess its thermal performance in the early design stage. Recently, brake fluid vaporization has been suspected as a possible cause of some collisions and a proper inspection procedure has been recommended.

Thermal analysis is involved in almost every kind of physical processes and it can be the limiting factor for many processes. Therefore, its study is of vital importance, and the need for powerful thermal analysis tools is virtually universal. Furthermore, thermal effects often appear together with, or as a result of, other physical phenomena.

The modeling of thermal effects has become increasingly important in product design including areas such as electronics, automotive, aerospace, railway (e. g. wheel and rail rolling contact, braking systems, and so on), medical industries, etc. Computer simulation has allowed engineers and researchers to optimize process efficiency and explore new designs, while at the same time reduce costly experimental trials [1].

The thermal effect in wheel introduced by reference: (Technical Specialist Rolling Stock Performance Engineering Standards) [2]. Wheels with defective wheels may cause damage to both the track and the vehicle or lead to derailment.

1.1 Brake Fade

Brake fade is a temporary loss of braking that occurs as a result of very high temperatures in the friction material. The high temperature reduces the coefficient of friction between the friction material and the wheel tread, and results in reduced braking effectiveness and ultimately failure. Generally fade is designed to occur at temperatures lower than the flame temperature of the friction material to reduce the possibility of fire at extreme temperatures. Normal braking will usually return when temperatures drop below their critical level.

1.2 Brake Fluid Vaporization

Most braking systems are hydraulically actuated, with the exception of heavy-duty trucking. If temperatures reached during braking exceed the boiling point of the hydraulic fluid then brake fluid vaporization will occur. A vapor lock will then form in the hydraulic circuit, and as gas is more compressible than liquid the pedal stroke is used to compress this gas without actuating the brakes. Brake fluid is hydroscopic causing it to absorb water from the atmosphere over time; this may result in a reduced boiling temperature of the fluid, (Hunter et al. 1998). Therefore it is usually recommended by vehicle manufacturers to replace brake fluid periodically.

1.3 Excessive Component Wear

High temperatures in the braking system can form thermal deformation of the rotors leading to uneven braking, accelerated wear and premature replacement. The life of the friction material is also temperature dependent, at higher temperatures chemical reactions in the friction material may cause a breakdown in its mechanical strength, which reduces braking effectiveness and causes rapid wear. The wear of frictional material is directly proportional to contact pressure, but exponentially related to temperature, (Day and Newcomb 1984); therefore more rapid wear will occur at elevated temperatures.

1.4 Thermal Judder

On application of the vehicles brakes, low frequency vibrations may occur, these vibrations can be felt by the driver as body shake, steering shake and in some cases an audible drone, (Kao et al. 2000). This phenomenon is known as 'judder'. Two types of judder exists; hot (or thermal judder) and cold judder. Cold judder is caused by variable thickness of the brake foundation, known as brake foundation thickness variation, this leads to deviations in contact pressure as the pads/shoe connect with the tread/disc. This results in uneven braking or brake torque variation. The second type, thermal judder, occurs at elevated temperatures, and is caused by thermal deformation of the tread/rotor. When a tread/rotor containing a cold disc/tread thickness variation is subjected to braking, the contact pressure in the thicker parts will be much greater than the thinner parts. As a result, the thicker parts become hotter causing uneven thermal expansion of the tread/rotor, which compounds the original disc/tread thickness variation and creates a "self-accelerating instability", (Little et al. 1998). Thermal judder can also be a result of 'hotspots' on tread/rotor surface. Hotspots can be caused by localized contact between the pads/shoe and rotor/tread resulting in small areas of very high temperatures, which causes a thermal tread/disc thickness variation. This thermal disc thickness variation may develop into a permanent disc thickness variation due to a phase change from pearlite to martensite, (Kao, et al. 2000), when cast iron is cooled rapidly. Martensite occupies a larger volume than pearlite, and therefore a cold disc thickness variation is formed and the problem is again compounded.[3]

1.5 Thermal cracks

Thermal cracks are the result of alternate heating and cooling of the wheel tread and rim area, and originate from metallurgical changes in the wheel material. Thermal cracks are the most severe form of wheel defect.

Heating from brake blocks when braking frequently produces a fine network of fine, shallow, superficial lines or "checks" running in many different directions on the wheel tread surface. Because of its similarity to the type of fine cracks found in pottery glazes, it is sometimes referred to as surface crazing. This should not be confused with true thermal cracking, and if found on its own causes no problems.

Thermal cracks are usually transverse, across the wheel tread, and, if allowed to grow without corrective action, can develop to the point where the wheel will fracture. Many shallow thermal cracks can be removed by machining but extra care must be used to ensure that the crack has been completely eliminated in the operation [2].

If thermal cracks are found on a wheel, then the vehicle's brake system should be checked for evidence of dragging brakes (sticking brakes).

The following two figures are type of thermal crake cause

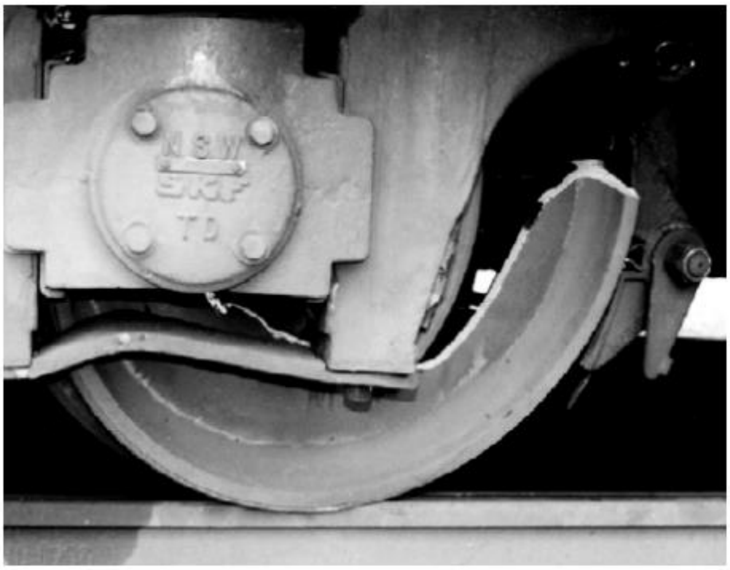

Figure 1.Wheel failure

Figure 2.Wheel crack

1.6 Project motivation

As explained in previous chapter one, most of the complications and failures with a tread brake is due to problems with heat distribution and heat level absorbed by the tread. This absorbed energy in form of heat is dissipated through convection, conduction and radiation. The easiest heat dissipation area is convection. To be able to start modifying existing designs and constructions for better convective cooling performance, more knowledge of the flow behavior has to be clarified. This project will therefore focus on cooling affected flow behavior which exist model and if it can be simplified for a more effective and user-friendly evaluation tool.

1.6.1 Statement of the problem

One of the most and frequently happen complex issue involved in the railway transportation is associated with how maximize the heat dissipation of the brake system and convert to usable manner , which is produce from the huge kinetic energy of the vehicle converted to the thermal energy (heat). In the braking phase, kinetic energy transforms into thermal energy resulting in intense heating and high temperature states of railway tread brake. Thus induced thermal loads determine thermo mechanical behavior of the structure of rail vehicle tread brake. The importance to precisely determine the temperature distribution caused by the transfer process of the heat generated during braking due to the friction on contact surfaces of the braking system makes it a challenging research task. The heat induce in the tread brake during braking, should be extracted as fast as possible for long life service. Therefore this research will address and clear out the following issues:

- How will be extracted the excesses heat develop in the tread brake?
- What method will use to fast heat dissipation of the tread brake?
- What factors should be considered at emergency downhill train brake?
- How much power converting in emergency brake time at downhill?
- How much wall temperature of tread brake at the end of emergency brake at downhill when air coolant use?

1.6.2 Objectives

1.6.2.1 General objective:
The main objective of the research is to improve the heat dissipation of the tread brake from the currently functioning train

freight (box car) of ERC (Ethiopian Railway Corporation) by using air coolant**.**

1.6.2.1 Specific objective:

The study will be

- Minimize of the effect of high temperature during braking
- Analysis the total heat flux entering to the tread brake using ANSYS
- Calculate the wall temperature of wheel tread by using FEM in both forced air cooled and spontaneous heat dissipation
- Calculate the convection heat transfer coefficient of the fluid by using computational fluid dynamics (CFD).

1.6.3 Significance of the study:

The brake system in rail vehicle is the main control method to stop, to slow down speed and in dawn hill. So that it need good safety and durable brake system. The research has alto of significance for rail way transport as:

- **Economic benefit**: when the heat capturing of the tread and block are minimized the fading, wearing, cracking, scaring and excessive rusting will be minimizing. so that minimize the cost of maintenance and replacing tread and block
- **Safety:** when excessive heat develops in the tread and blocks the brake system reduce braking efficiency. Because the friction material wear out and face to the severely derailment of the train.
- **Decrease the brake distance:** when the tread and block are wear and fad out the braking efficiency reduce, then brake distance increase.

1.6.4 Scope of the study:

The scope of the study was to analyze the heat dissipation improvement with regarding of air cooling. Else, analysis the thermal effect of tread brake and estimate thermal result by the application software ANSYS. To define convection heat transfer coefficient using computational fluid dynamics (CFD) of the coolant. Additionally by using CATIA software to model the profile of tread wheel.

CHAPTER TWO

ANALYSIS OF HEAT FLUX AND CONVECTION HEAT TRANSFER COEFFICIENT

2.1 ANALYSIS OF HEAT FLUX

This paper focused on the emergency brake train at downhill. At the emergency brake time when the train is at downhill the total energy converging is through friction of the tread and block contacts. At that time over heating of tread occurred.

2.1.1 Assumptions of heat analysis (heat flux)

To analysis the heat flux entered to the tread brake we should be justify the input values and consideration. In this work, the following assumptions are considered

As mentioned above the braking condition was emergency brake at downhill with grad 1.5%. At that condition the train exposed to variable forces/loads .such is:

- Inertia of the train (transfer load)
- Kinetic energy of the train
- Rolling resistance of between tread wheel and track(rail) (F_r)
- Gradient resistance of train because of the grad (F_g)
- Drag force resistance because of Aerodynamics atmospheric air on the frontal area of the train (F_d)

2.1.2 Heat flux

During braking, the kinetic and potential energies of a moving vehicle are converted into thermal energy through friction heating between the tread brake and the block. Frictional heat is generated on the surface of the tread brake and brake block. In the present work, considering the amount of heat generation by wear is very small relative to the heat generated by friction, so the effect of material wear is neglected [8]. The friction heat flux generated in the interface of tread and block can be expressed

$$\dot{q_o} = \frac{P_t}{nA_f} \quad (1)$$

Where:

P_t is total power induced by the brake treads

A_f is the friction area of the brake tread

n is the number of contact surface of tread brake

q_o is heat flux entered to single tread brake

Table 1. Train data

The vehicle data	quantity
Vehicle mass with carried total load (t)	100
Maximum Vehicle velocity (km/hr.)	100
Deceleration at emergency time (m/s^2)	1.4
The ratio of load distribution b/n front and rear wheels	3/2
Number of wheel in train	8
The maximum gradient of the track % (0.8594^o)	1.5
Rolling resistant coefficient (μ) of wheel and rail	0.09
Area of friction contact b/n tread and brake block (m^2)	0.1315574
The friction coefficient brake tread and brake block (*f*)	0.15
Wheel diameter (mm)	840

This data is from the train freight box car of Ethiopian railway corporation (ERC) proposed in mar.5th, 2003 (procurement reference number: OS ESTS=RS-01/12). The general specification of the train was as following [15].

Where "μ" is the so-called adhesion coefficient, whose rolling values for wheel and steel rail is dependent on temperature, humidity, dirt, etc, and particularly on speed. An expression that will enable us to find the value of the adhesion coefficient according to speed is the following:

$$\mu = \mu_o(0.2115 + \frac{33}{42+v}) \quad (2)$$

Where "V" is the speed of the train in Km/h, and "μo" is the adhesion coefficient for zero speed. The value of "μ_o" depends on the atmospheric conditions of temperature, humidity, dirt, etc,[16]. Therefore at μ_o = 0.20275 the value of rolling coefficient is 0.09.

Technical Specification of Box Car: The box car is used for the railway with the gauge 1435mm, to transport various goods which are afraid of the sun, rain and snow erosion and boxed and bagged goods. Key Parameters and Dimensions

- Loading Capacity =70t
- Dead weight ≤25t
- Design Speed 100km/h
- Minimum Curve Radius 145m
- Vehicle Length (at Coupler Connection Point): ≤17400mm
- Height from Center of Coupler to Top of Rail: 880mm

Based on the above data the analysis was calculated.

The effective contact area or the foundation brake heat generated area (A*f*) calculated as

$$A_f = lw\,; \quad (3)$$

$$A_f = 0.1315574m^2$$

>>Where; l= ($2\pi r$ = 2.6385m). The circumference contact of tread with block and w = (0.05m). The width of the contact face of brake block

During the process of braking available energy of the locomotive that must be transfer into frictional heat need to be determine accurately .this study considers the kinetic as well as the potential (while running on slopes) energy of the locomotive that must be dissipated by the work of the braking. The initial kinetic energy imposed into the locomotive is given by the translation and rotational. So; the total power induced (P_t) by the brake tread is calculate as:

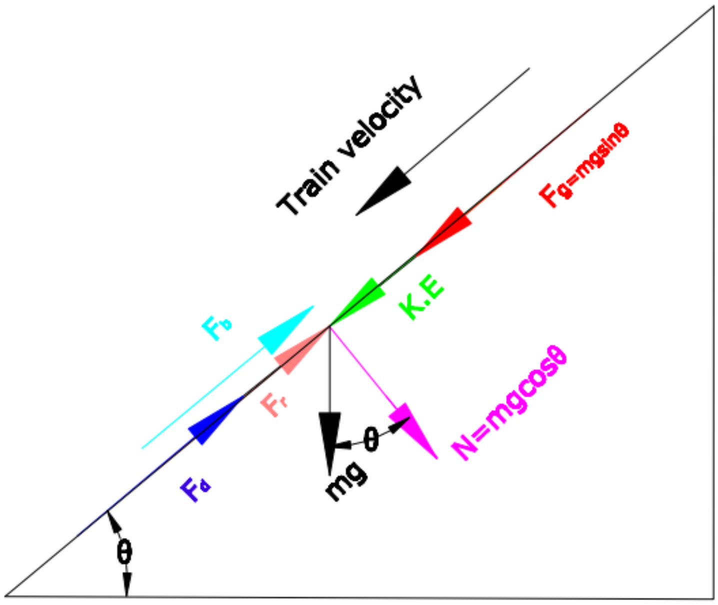

Figure 3. The free body diagram of force component at brake time and train motion at down-hill

$$P_t = \frac{0.5(mv^2+Iw_o^2)+(F_g-F_d-F_r)*S_b}{t_b} \tag{4}$$

Where:

m, is the total mass of train (tera load and carried mass)

v, initial velocity of the train (100km/h)

f_g component force along the vehicle direction induced by the gravitational load

f_d; the resistance of drag force of induced by the atmospheric air

f_r, the resistance of rolling force or friction force

t_b , the brake time of the train

S_b, is the brake distance

W_o; angular velocity of the rotating parts

I; Polar inertia moment of rotating

Since the analysis focused on the emergency at downhill, the energy induced kinetic and potential energy additional there have aerodynamics effect should be considered.

The moment of inertia of the rotating wheel set can be calculated as the following equation:

$$I = \sum_{i=1}^{n} m_i r_i^2 \tag{5}$$

Where r-rotational radius and m- mass of the rotating parts

For our case all the rotating parts are fixed on the axle of the wheel set and the rotating axis could be taken as the tangent line joint the contact point of the rail heads with wheel set of the locomotive. It is the parallel to the axis of the wheel axle. Hence, the rotating radius is equal to the radius of the wheel (r_w) hence

$$E_K = 0.5mv_o^2 + 0.5Iw_o^2 = 0.5mv_o^2\left(1 + \frac{I}{mr_w^2}\right) \cong 0.5(1.1)mv_o^2 \tag{6}$$

$$E_K = 42506200\ J$$

From figure # 4 we understand how the kinetic energy converts to the thermal energy. At the initial braking train have maximum K.E and at the end of train brake time have zero value. This means the K.E convert to thermal energy gradually.

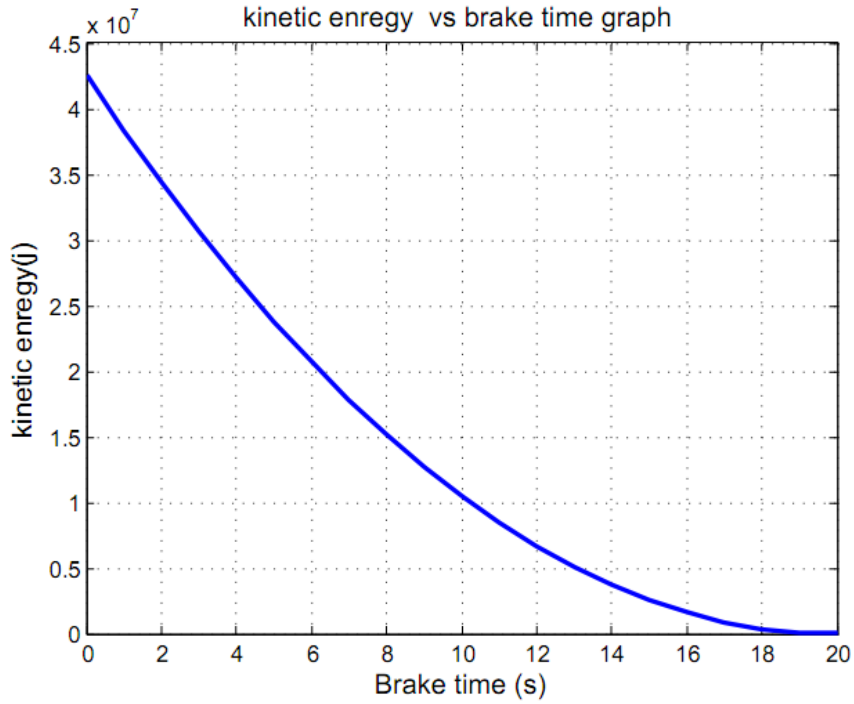

Figure 4. Converting of kinetic energy to heat energy vs. brake time

The resistance forces calculate as:

a. Aerodynamic Resistance Composed of(drag force):

- Turbulent air flow around vehicle body (85%)

The drag force induced by atmospheric air;

Table 2. Train aerodynamics data [17]

No	Distribution	Value
1	C_d the drag coefficient	1.8
2	ρ, the density of air (kg/m^3)	1.2
3	A is the vehicle front body area	(4.433*3.106m= 13.4m^2)

$$F_d = 0.5 C_d \rho A v^2 \quad (7)$$

Where:

C_d the drag coefficient

ρ: the density of air

A is the vehicle front body area (m^2)

V is the velocity of vehicle (m/s)

F_d is the drag force

Note that; the drag force depends on the velocity of train and frontal area of the train. Since the train is gradually decline its speed the drag force is diminish to null. When we take the initial braking speed the value of the drag force is:

$$F_d = 0.5 * 1.8 * 1.2 * 13.4 * (27.8)^2$$

$$F_d = 11184.54\text{N}$$

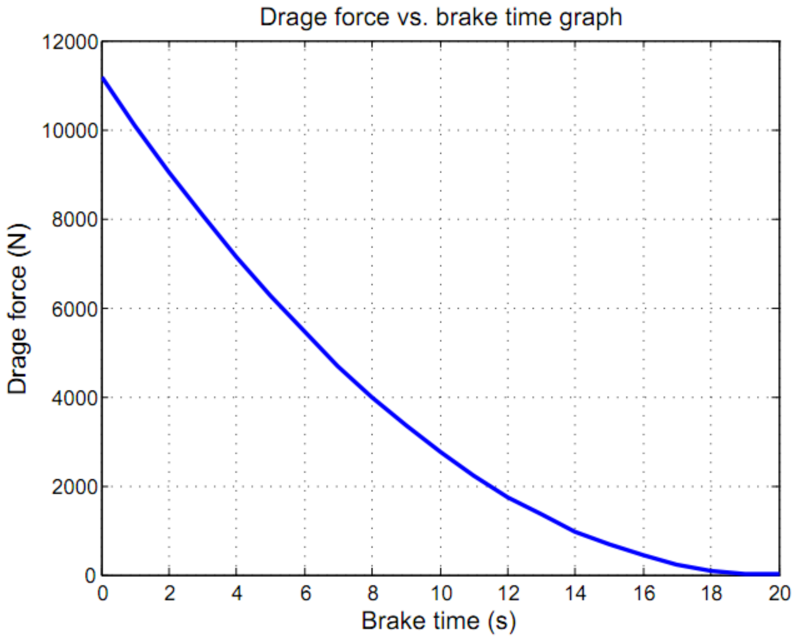

Figure 5. Drag force vs. brake time

b. Grade resistance (Gravitational force acting on the vehicle):

- Increase the tractive force,
- Increase the load on front axle and decrease the load on the rear one.
- Increase the stopping distance when using the brakes

The grade resistance induced because of the geometrical position of the train;

$$R_g = mg\sin\theta \qquad (8)$$

Where:

m, is total mass of the train (tera load and carried load)

g ,is the gravitational acceleration

θ, the gradient of the track angle (0.8594)

$$F_g = 100000 * 9.81 \sin 0.8594$$

$$F_g = 14713.82\text{N}$$

c. Rolling Resistance (R_{rl}) :

Composed primarily of

- Resistance from tread deformation (90%)
- Wheel surface compression (4%)
- Wheel slippage and air circulation around wheel (6%)

The rolling resistance force induced by the friction of rail with wheel;

$$F_r = \mu mg \cos\theta \tag{9}$$

Where; μ is the rolling resistance coefficient between wheel and rail (0.09).

$$F_r = 0.09 * 100000 * 9.81 \cos 0.8594$$

$$F_r = 88280.068N$$

The Brake time of the train;

$$t_b = \frac{v_f - v_o}{a} \tag{10}$$

Where,

t_b is the brake time of the train

V_f is the final velocity of the train (stop velocity (**Vf**) = 0)

V_o is the initial velocity of the train (Vo = 100km/hr.)

a is the deceleration of the train (a =1.4 m/s^2)

$$t_b = \frac{0 - 27.8}{-1.4}$$

$$t_b = 19.857sec$$

The brake distance of the train;

$$s_b = \frac{v_f^2 - v_o^2}{2a} \tag{11}$$

$$s_b = \frac{0^2 - 27.8^2}{-1.4 * 2}$$

$$s_b = 276m$$

Therefor the total power induced by the brake disc from equation (3):

$$p_t = \frac{0.5(mv^2 + Iw_o^2) + \left(F_g - F_d - F_r\right) * S_b}{t_b}$$

$$P_t = \frac{42506200 + (14713.82N - 11{,}184.6N - 88280.068N) * 276m}{19.857sec}$$

$$P_t = 962631.1 \text{ watt}$$

Hence the total energy generated is equal to the heat dissipated then. The total heat flux entered to the tread calculated.

In practical world the load transfer is obvious. The load distribution in front and rear wheels have different because of load transfer. Besides that, the weight distribution of the vehicle was considered .The load transfer approximate by ref. [4]. The weight arrangement was 60/40 in the favor of the front part of the bogie. This means that the front part of the bogie takes 60% of the whole load. In our case only 15% of the whole brake force was applied to one tread from the front part of the bogie. Because of the mentioned weight distribution, only the front part of the bogie was analyzed. Every bogie consists of two axles with two tread brake at front and rear. Therefor the total power distribution at the front single tread brake calculated as

$$p_{front} = 15\% p_t \tag{12}$$

$$p_{front} = 144394.666 \text{ watts } ;$$

>> The power induced at the front single tread; hence it has four tread brakes and for each heat flux calculates as:

$$\dot{q_o} = \frac{P_{front}}{A_f} \tag{13}$$

$$\dot{q_o} = 1097579.205 \frac{w}{m^2}$$

>> The total heat flux entered to each front brake tread.

Based on the first law of Thermodynamics, during braking, kinetic energy is converted to thermal energy. Due to friction between the main components of tread brake (block and tread) the conversion of energy takes place [5].

2.2 CONVECTION HEAT TRANSFER COEFFICIENT

Convection heat transfer takes place when a fluid is in contact with a solid surface that is at a different temperature than the fluid. If the fluid is moving past the solid surface because of an external driving forced, like a pump or blower, then it is called forced convection. If fluid motion is due to density differences caused by temperature variation, then it is called natural convection or free convection [6]. A major component of most convection heat transfer calculations is obtaining a good estimate for a convection heat transfer coefficient.

2.2.1 Forced Convection Heat Transfer

Convection is the mechanism of heat transfer through a fluid in the presence of bulk fluid motion. Convection is classified as natural (or free) and forced convection depending on how the fluid motion is initiated. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect, i.e. the rise of warmer fluid and fall the cooler fluid. Whereas in forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or fan. Convection heat transfer is complicated since it involves fluid motion as well as heat conduction. The fluid motion enhances heat transfer (the higher the velocity the higher the heat transfer rate).

In braking, the major portion of the generated heat flows out to the air, and some goes out into the air by radiation, conducted into the hub and pad, and the rest of the heat is stored in the tread. The convective heat transfer coefficients of the forced convection brake are determined from the experimental formulas [7].

2.2.2 Dimensionless numbers for Convection Heat Transfer

When making convection heat transfer calculations, it is well to keep in mind that estimating values for convection heat transfer coefficients is not an exact science. The value of a convection heat transfer coefficient depends upon the physical configuration.as well as upon several properties of the fluid involved. Empirical correlations are available to estimate heat transfer coefficient for a variety of natural convection.

Those correlations are typically expressed in terms of dimensionless numbers. The dimensionless numbers used for forced convection heat transfer coefficients are the Nusselt number (Nu); Reynolds number (Re) and Prandtl number (Pr).they define as:

i. **Nusselt number(Nu);** Non dimensional heat transfer coefficient where l is the characteristic length, i.e. D for the tube and L for the flat plate. Nusselt number represents the enhancement of heat transfer through a fluid as a result of convection relative to conduction across the same fluid layer.

$$\mathrm{Nu} = \frac{hD}{k} \tag{15}$$

Where:

- ћ is the heat transfer coefficient.
- D, wheel diameter
- *K* is heat conductivity of the flowing fluid.

ii. **Reynolds number(Re);** ratio of inertia forces to viscous forces in the fluid at large Re numbers, the inertia forces, which are proportional to the density and the velocity of the fluid, are large relative to the viscous forces. Thus the viscous forces cannot prevent the random and rapid fluctuations of the fluid (turbulent regime). The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number.

$$\mathrm{Re} = \frac{\mathrm{D}v\rho}{\mu} \tag{16}$$

Where:

- D, wheel diameter
- V, is a characteristic velocity of the problem
- ρ is density of the flowing fluid (air)
- μ is viscosity of the flowing fluid(air)

iii. **Prandtl number (Pr)**: is a measure of relative thickness of the velocity and thermal boundary layer

$$\mathrm{Pr} = \frac{\mu Cp}{k} \tag{17}$$

Where:

- μ is viscosity of the flowing fluid(air)
- C_p is the constant-pressure specific heat of the flowing fluid.

- K is heat conductivity of the flowing fluid.

Using the above non dimensionless numbers we can calculate convection heat transfer coefficient.

The experimental correlation with widespread application for cross flow over a circular cylinder; prepared by Churchill and Bernstein [8]. The following figure indicts the fluid follow cross cylindrical pipe. This correlation is similar with the wheel tread .the air cooling follow through the brake pipe or forced fan along to the longitudinal train direction; this is cross the wheel tread.

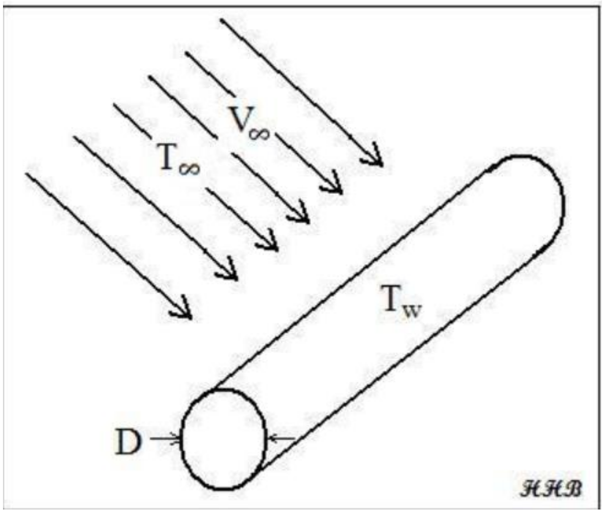

Figure 6. Air Cooling flow direction

Where, D is the outer diameter of tread wheel, T_∞ temperature of the forced air cooling, V_o velocity of air flow to the tread and T_w is the wall temperature generated by brake.

In this work two type of convection heat transfer coefficient. The first one because of the speed of train without forced air cooling. The 2nd; forced air cooling from the brake pipe or external fan drive.

Table 3. Thermal property of air

No	Item	Value
1	Radius of wheel (m)	0.42

2	Train speed (km/hr)	100
3	K_a(heat conductivity of air)(w/m.c^o)	0.026
4	Air density (ρ_a) (kg/m^3)	1.2
5	C_p (specific heat capacity of air) (J/kg .c^o)	1007
6	Viscosity of air (μ_a)	1.8x10^{-5}

Figure.7 shows the mechanisms of cooling brake tread and brake system. When the train is applied braking system the air in brake pipe flow to tread brake for cooling purpose. If the air is not sufficient to cool the brake, another option to supply sufficient air has to be devised. As we see from the diagram the main air brake pipe connect to the exhaust direct for cooling purpose. The connection is by the T type pipe connection for both from the brake pipe and direct from main compressed air reservoir. The air velocity from the brake pipe and main reservoir is 100 m/s.

When estimate the air supply power consume and change of pressure we can calculate as:

$$\mathrm{Pe} = \frac{\mathrm{Q}\Delta \mathrm{p}}{\eta} \tag{20}$$

Where: Q, air flow rate

Δp , change of pressure required

η , power efficiency

Since the air flow depends of the air velocity (v) and cross section area (A) (5.0

X10^{-4} m^2) of the brake pipe (the standard diameter of the pipe is 25.4mm) we calculate as:

$$Q = vA \tag{21}$$

Else the change of pressure depends with the air velocity (v) (100m/s) and density of the air (ρ) (1.2 kg/m^3). We can calculate as:

$$\Delta p = \frac{\rho v^2}{2} \tag{22}$$

Therefore the results with power efficiency of 95% has gotten 0.32kwatt and change of pressure 6kpa estimated.

The following two graphs show that how the convection heat transfer coefficient related with the train speed and the external forced air cooling was constant through the braking time because the source is from the compressed air reservoir and brake pipe.

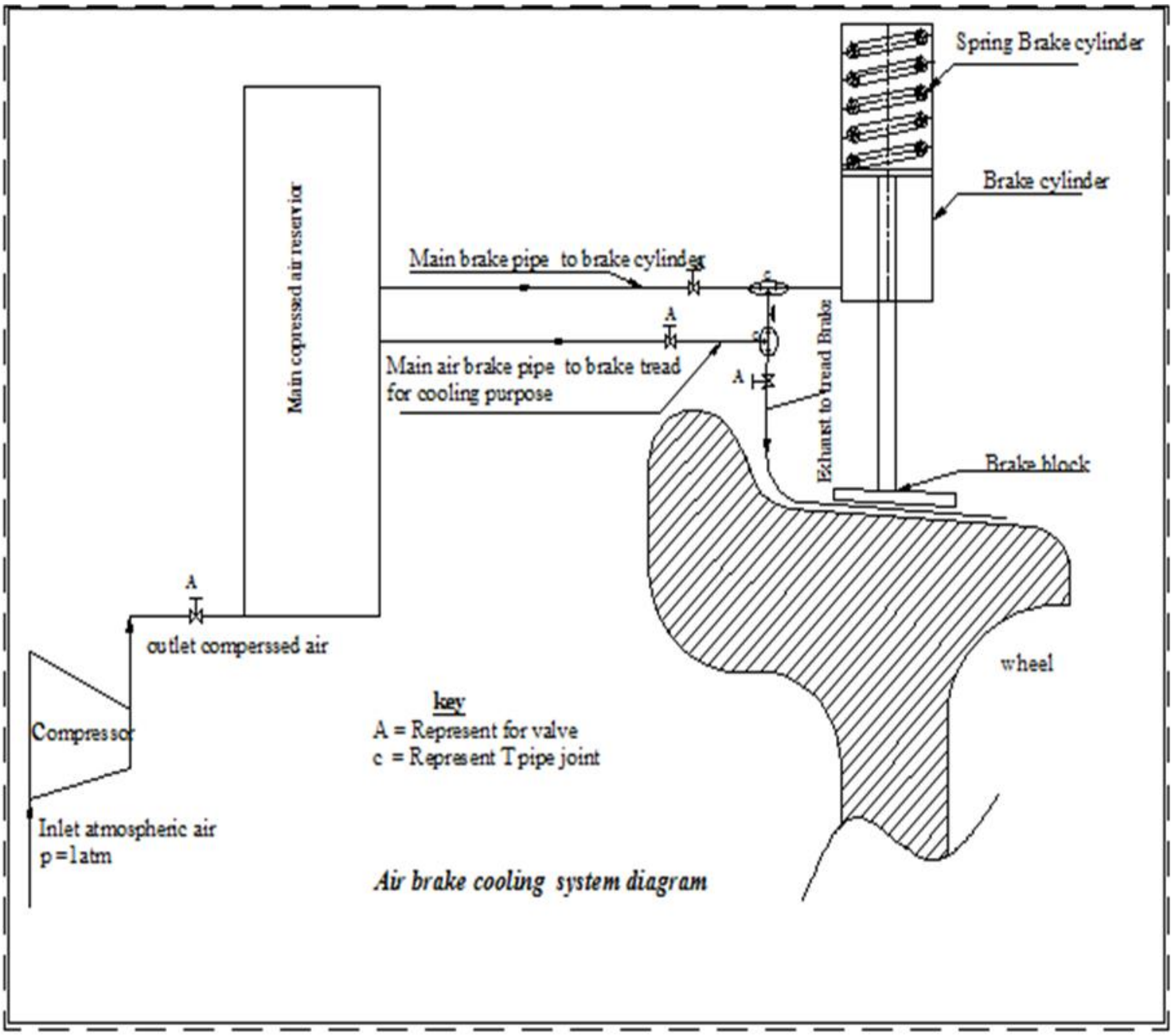

Figure 7.Air brake system diagram

Figure.8 shows that convection heat transfer coefficient without air cooling. From the figure, it is clear that the convective heat transfer coefficient declines with time as train speed drops.

Figure.9 indicts the convection heat transfer coefficient using the air cooling /with air from brake pipe or from main reservoir. As it can be seen from the figure, the convection heat transfer coefficient gradually declines. As we discussed in figure.9 when the speed decreases the heat transfer coefficient will decrease slightly.

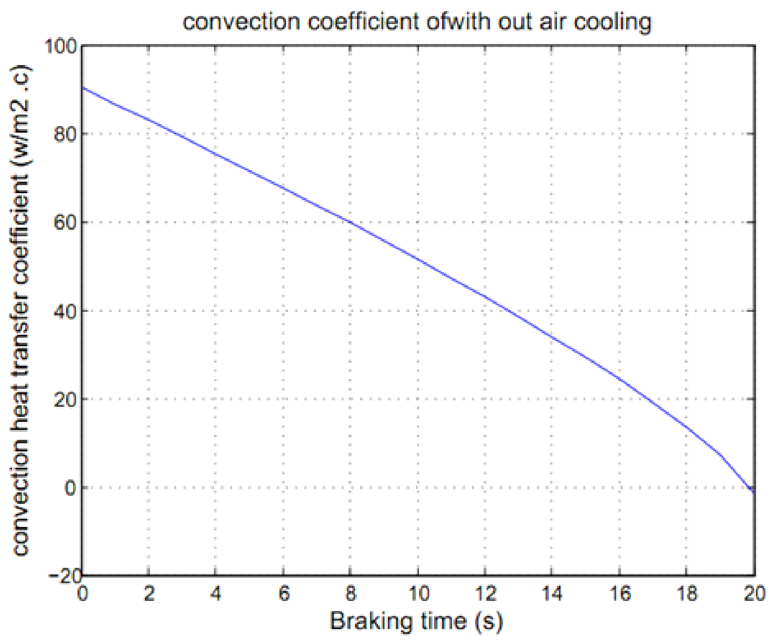

Figure 8. Convection heat transfer coefficient without air cooling

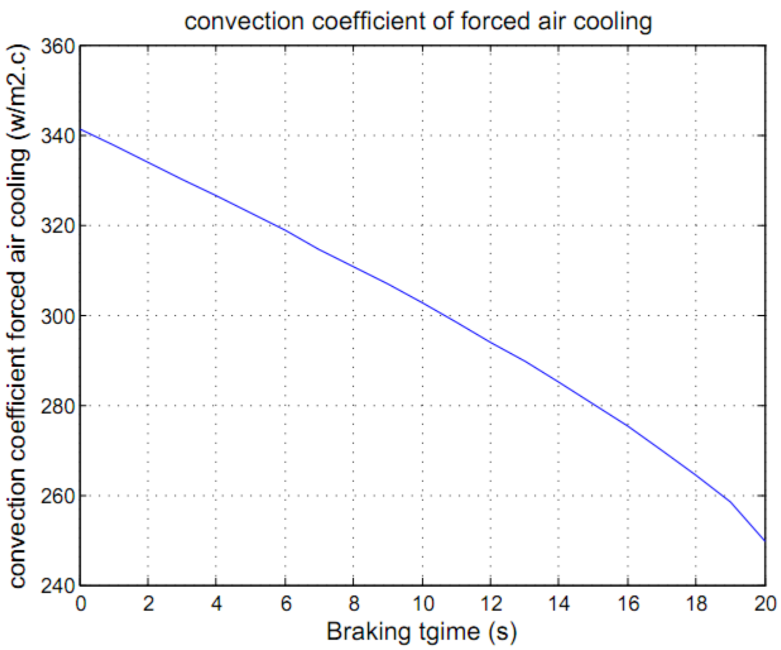

Figure 9. Forced convection heat transfer graph

CHAPTER THREE

NUMERICAL ANALYSIS USING ANSYS

Most numerical approaches on tread brakes preset a brake time or deceleration rate, in practice, the block brake press on the tread wheel by pneumatic cylinders; the constant deceleration rate can be maintained.

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 tread wheel.

This study will use the ANSYS14.5 version FEA software to carry out transient thermal analysis and the CFD (computational fluid dynamics) of a railway tread wheel during emergency brake at downhill conditions will be determined.

During the braking process, the temperature field changes input heat flux and heat exchange conditions. The input heat flux is mainly dependent on the friction coefficient and angular velocity of the tread brake, while the heat exchange is connected with the friction pair materials and external environmental factors like, the natural convection film coefficient and forced convection film coefficient.

3.1 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 result to get other information such as values of principal stresses, heat fluxes, temperature distribution, convection heat transfer coefficient, speed flow , pressure disruption etc.

The goal was to find out the distribution of temperature in the tread brake considering airflow cooling. The initial temperature of the brake tread and the surrounding was 25 °C. The part of the braking energy that transfers to the surrounding air was not considered. The heat flux was applied in the calculation equation 13.

3.2 Transient Thermal Analysis wall tread brake without air coolant

3.2.1 Geometrical Model

The analysis is carried out for three dimensional solid of the tread wheel (Fig. 10) (the tread wheel is supposed to be symmetrical). In this modeling of thermal effects was performed according to the following assumptions. in this work to simplified the simulation tread brake wheel ,it is used different modeling software like CATIA, MATLAM , ANSYS 14.5 work bench module. The following geometry modeled in CATIA and converted to compatibility of ANSYS14.5. igs format. The geometry is symmetry of the wheel to simplify analysis even if not relative accurate than the whole wheel analysis.

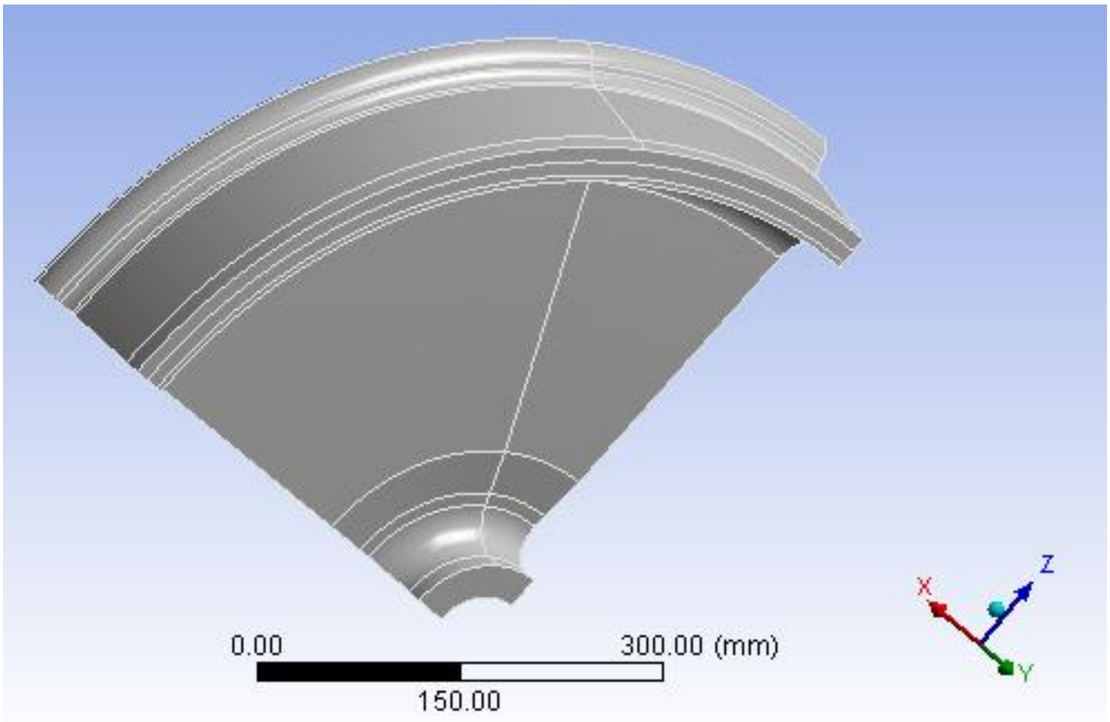

Figure 10. Geometrical of tread brake

3.2.2 Definition of Material

The material of the railway wheel is steel DIN 40Mn4 (AISI 1039, JIS S40C, Č3130) and of the braking blocks is gray cast iron P10. Table 4 summarized the thermal properties of these materials; the density of air at a reference temperature of 25^{o}c was calculated using the thermal air [11]. Else the wheel steel chemical composition summarized in table 4

Table 4. Thermal material property of brake foundation

Property	**Railway wheel**	**Braking block**	**Air**
ρ [kgm $^{-3}$]	7872	7200	1.2
Cp [J kg^{-1}C $^{-1}$]	481	165	1007
k [W m $^{-1}$C $^{-1}$]	45	83	0.026
μ_{va} [Pa·s]	--	--	1.8x10^{-5}

Wheel	**Content of element, %**							
Chemical	C	Mn	Si	S	P	Cr	Ni	Cu
%	0.58	0.74	0.34	0.025	0.011	0.14	0.17	0.21

3.2.3 MESHING

Meshing is discretizing the solid object to finest parts to perform the analysis to get the precise value at each and every elements of the meshed object. Since the wheel a 3D element the appropriate meshing method will be volume mesh so that all the volume of the wheel is discretized to the smallest part of the wheel. The resulting volume mesh is automatic and the output of the mesh consists of 73899 nodes and 36576 tetrahedral elements. After this the object is ready to be analyzed by setting the analysis on the ANSYS work bench.

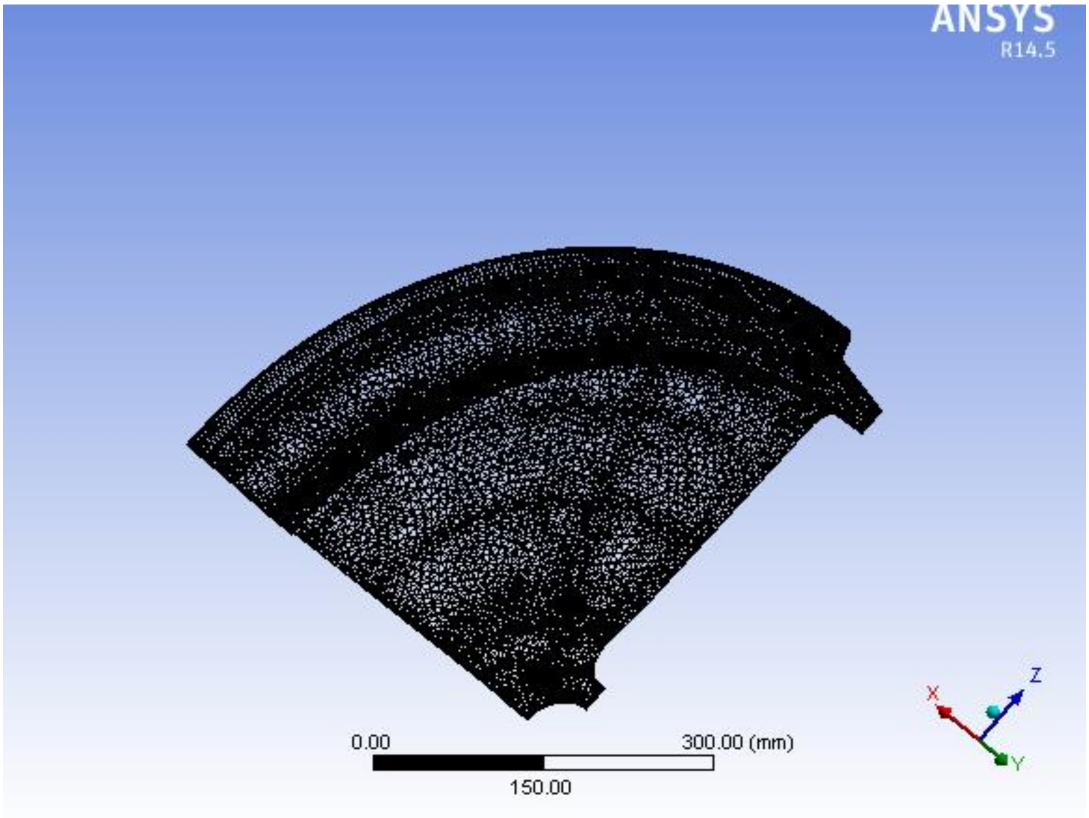

Figure 11 . Mish of the tread brake

3.2.4 Definition of the Loads and Boundary Conditions

After completion of the finite element model, it was necessary to apply constraints and loads to the model. The analyzed railway tread brake is subjected to the following loads.

- The initial temperature of the tread and the block is 25 °C.
- The heat flux entered to the tread brake during braking can be calculated before by the formula described in equation 13. The value of heat flux is 1097579.205 w/m^2.
- The braking time of the train is t =20 second.
- The convection heat transfer coefficient imported from the output of CFD result.

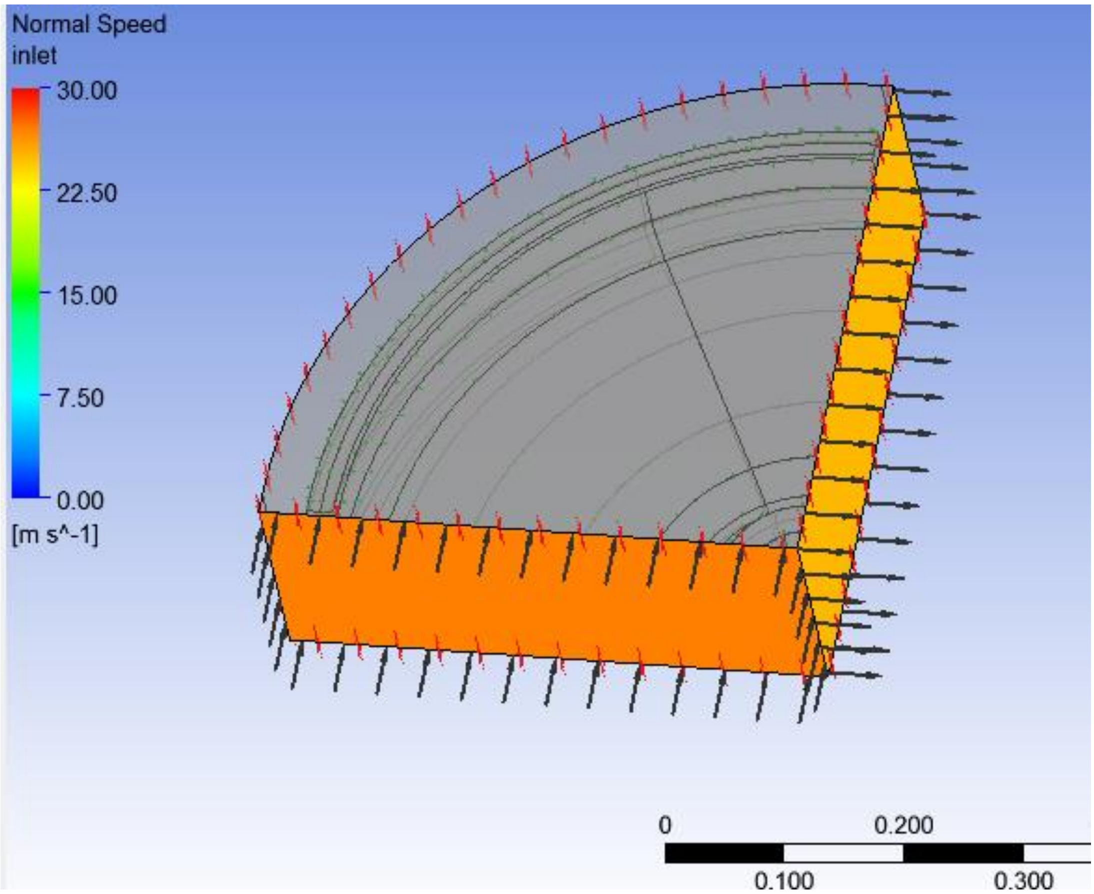

Figure 12. Train velocity at CFD simulation without forced air cooling

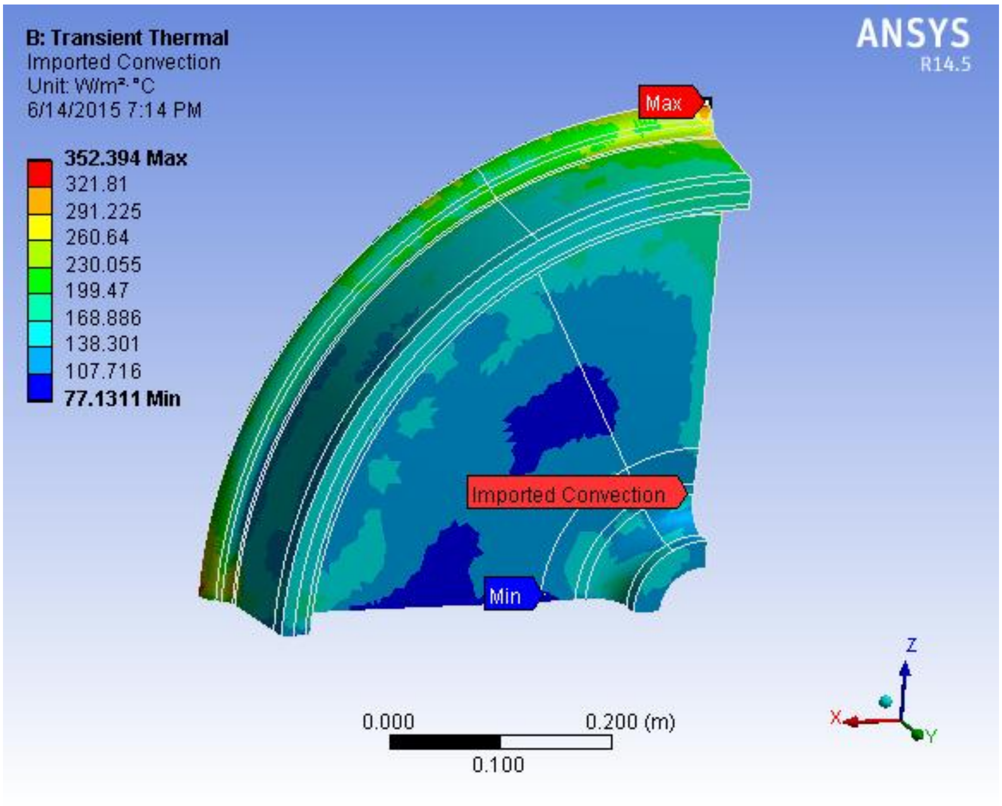

Figure 13. The convection heat transfer coefficient of without forced air cooling

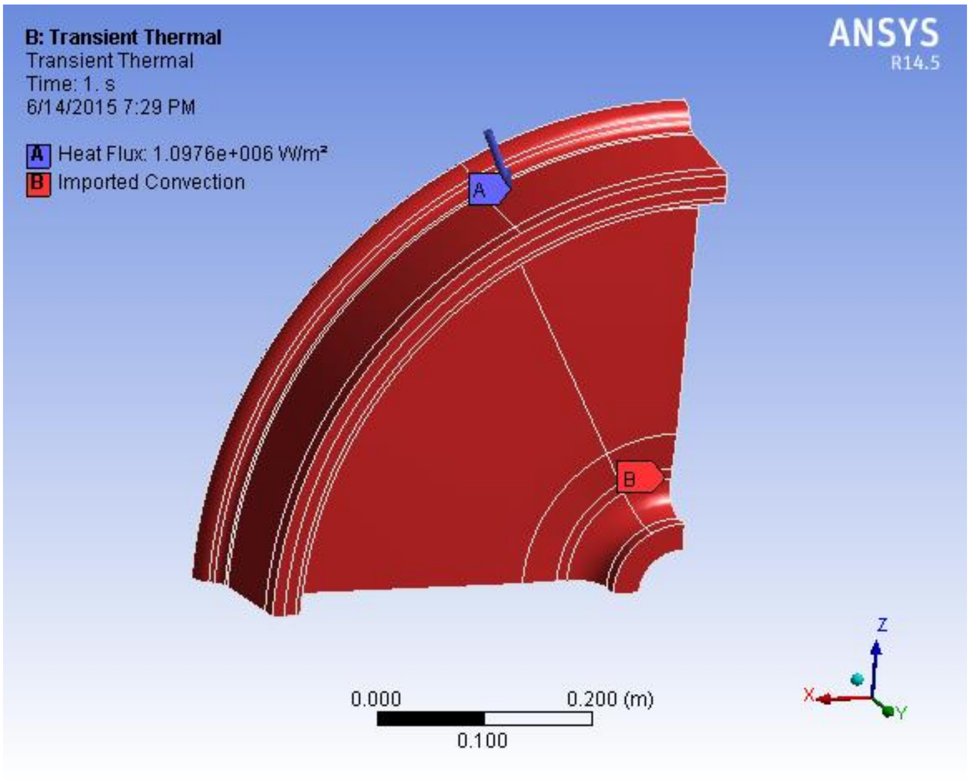

Figure 14. Boundary conditions of the brake tread.

3.2.5 Thermal result and discussion without air cooling braking tread wheel

The thermal result of tread wall temperature and total heat flux are in the following figure-15 and -.16 the value are 432.43 C^o and 1.6502x10^6 w/m^2 respectively. The maximum values of thermal results occurring on the surface of contact tread brake and block brake because of friction. This means that the kinetic energy of the train converted to the thermal energy.

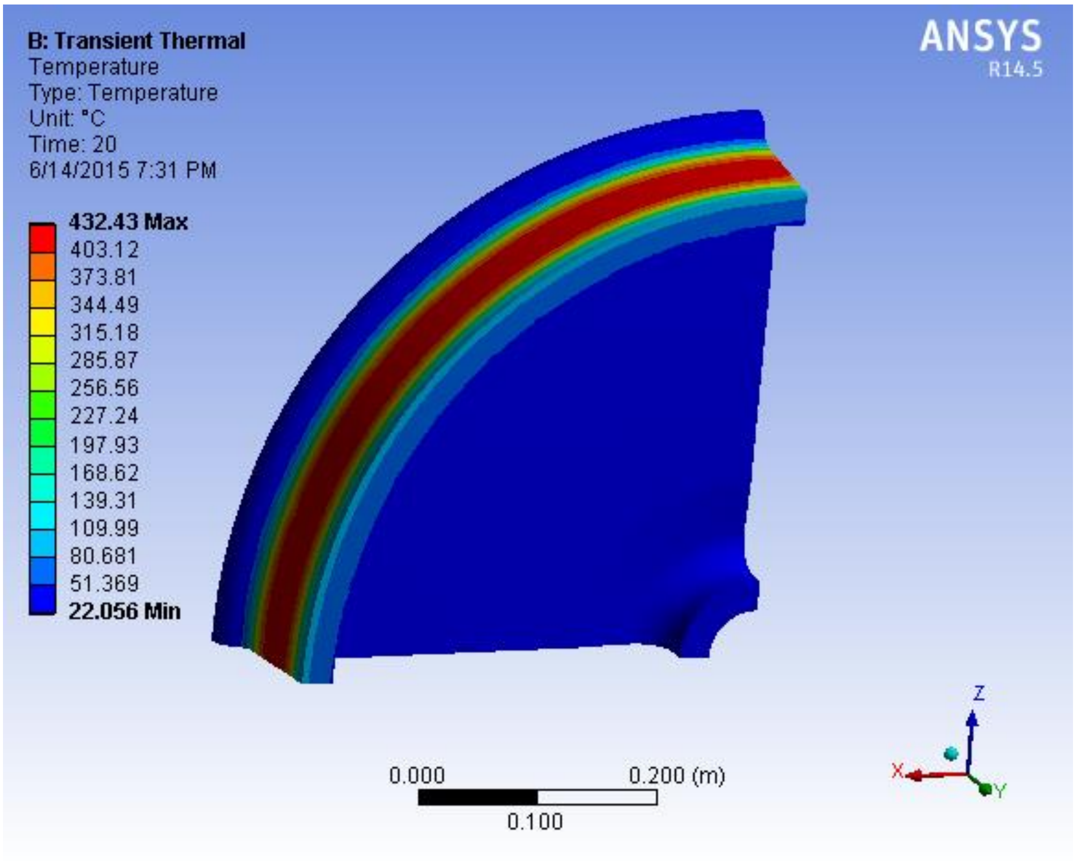

Figure 15. Tread wall temperature result without forced air cooling at the time t=20 seconds

Total heat flux is the summation of heat flux instant through the brake time entered to the wall tread brake continuously. This heat is undesirable energy accumulation on the tread brake. If the flux will not extract as fast as possible the tread face to thermal effect like crack, fade and so on else increase thermal stress and finally fail out the wheel.

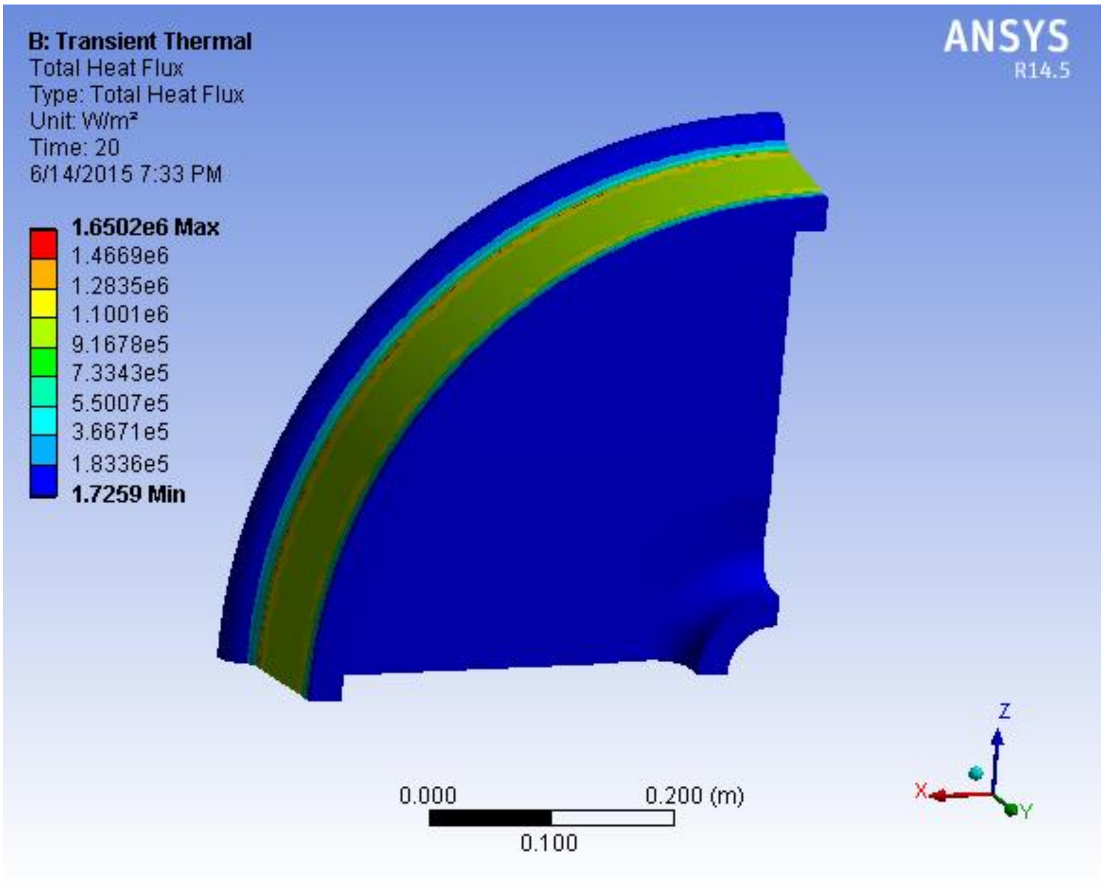

Figure 16. Total heat flux with out air cooling

3.3 Transient Thermal Analyses of tread brake with Air cooling

In air cooling analysis of thermal wall tread brake, it is used the CFD (computational fluid dynamics) to specify the convection heat transfer coefficient. This is input data for transient thermal analysis for cooling of the tread brake. Since it is used the forced ventilation the CFD simulation was mandatory to simulate the air flow and heat exchange. As mentioned above the compressed air in the main reservoir and brake pipe is the source of cooling tread brake.

Computational fluid dynamics is an approximate solution technique used in the prediction of flow fields. The accuracy of results from CFD simulations depends on the quality of the mesh and the user's ability to utilize the correct modeling techniques within the solver. Therefore validation of results obtained from the CFD analysis is essential to make meaningful predictions.

The model applied periodic boundary conditions on the section sides. Airflow around the tread was considered to be 30 °C, and open boundaries with zero relative pressure were used for the upper, lower and radial ends of the domain. For modeling the emergency braking phenomena, the brake tread decelerated with the constant value of 1.4 m/s2 with the velocity of 100 km/h, and then the braking started and caused constant deceleration during emergency brake,

CFD could be a terribly powerful tool used for higher understanding the flow and warmth transfer in air brakes and making new a lot of economical styles.

The object of this study is to grade up the heat dissipation of tread brake by using forced air cooling. So that to simulate the air flow it should be used the CFD application. The output of CFD data is for input of transient thermal analysis. From CFD we got the convection heat transfer coefficient and then export to transient thermal analysis. To start the computational fluid dynamics first specified the domain of fluid and solid. And then it applied the boundary condition for both domains. Else it specified the condition is transient CFD analysis.

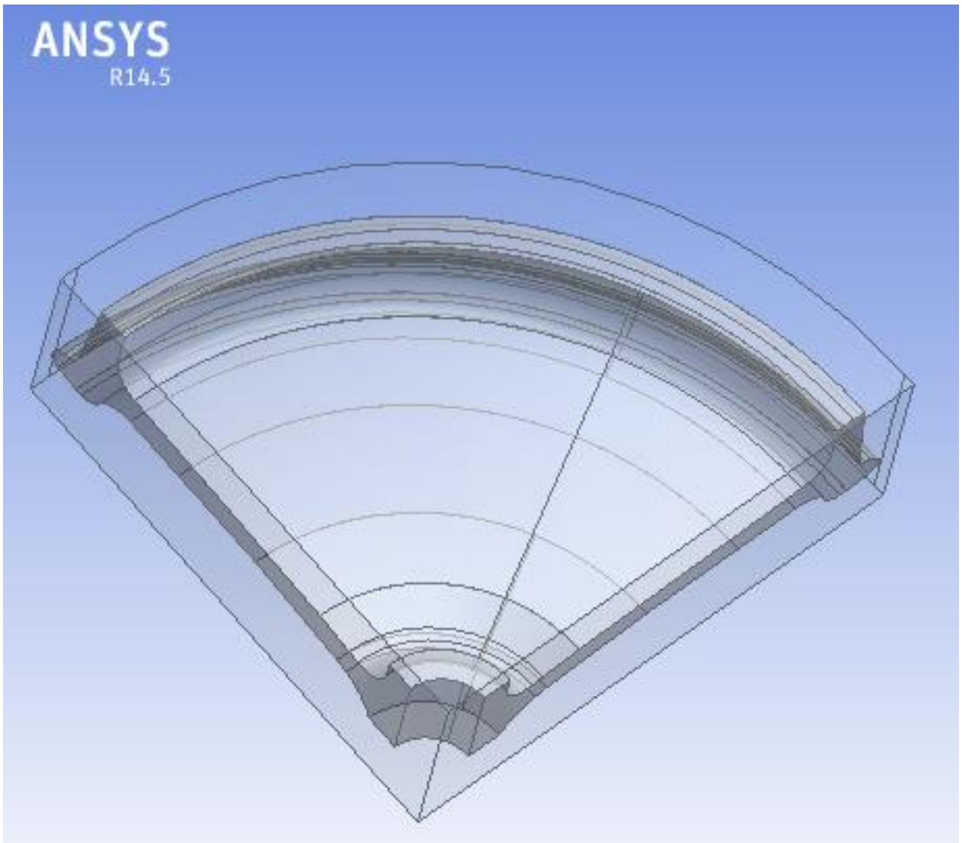

Figure 17. Geometry of CFD analysis

3.3.1 Boundary condition of the CFD

This is the initial step in analysis method. The first purpose of pure mathematics creation is to get a solid that defines region for fluid flow. This section describes creation of pure mathematics. Dimensions and pure mathematics details of existing model was collected. Modeling was done victimization professional-E Wild hearth a pair of.0 and ported in IGES format. Meshing and Domain Specification

3.3.1.1 Making Regions and Meshing:

This step defines creation of regions and pure mathematics. 2nd region is made for outlining body of air and outlet. Creation of regions facilitates to assign stipulation for body of air, outlet and alternative outlined regions. The advantage of applying regions to the pure mathematics is that they're directly associated to the model; if we tend to re-mesh the pure mathematics, they continue to be related to the model. Once a replacement mesh is made, the regions square measure mechanically reassigned. For illustration functions, Fig.16 shows the mesh used for CFD analyses.

Mesh generation is that the method by that spatial discretization of CFD model is accomplished. Meshing relies on polyhedron part discretization. The model is exported in IGES format and is employed in CFX- CFD tool. Surface and volume meshes were generated victimization this tool by shaping the sort of meshing part and mesh part size.

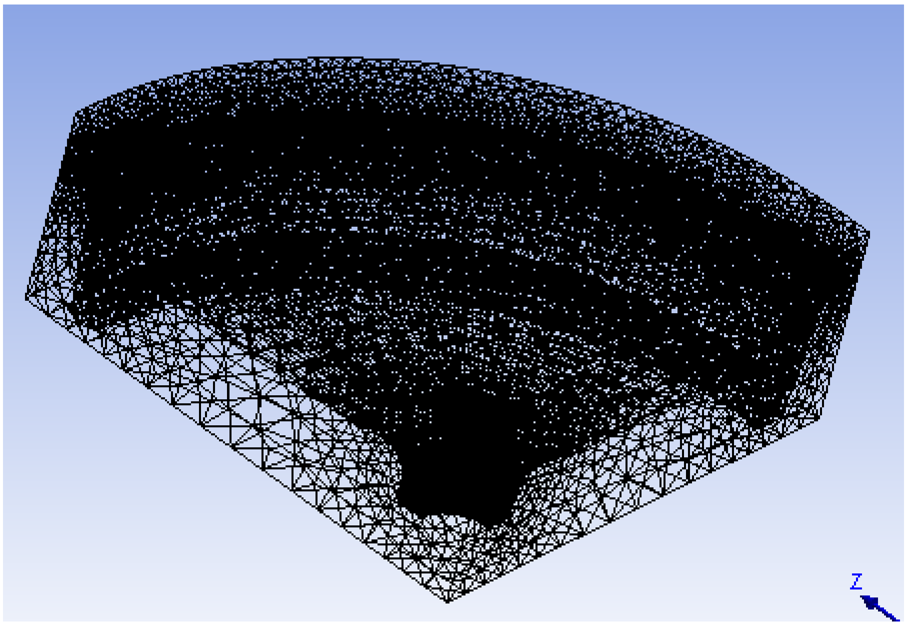

Figure 18. Meshing of the geometry

3.3.1.2 Specification of Domain and Boundary Conditions:

The flow of air through and round the tread brake was analyzed victimization the ANSYS CFX code package. After the warmth transfer coefficients considering convection was calculated and arranged in such the simplest way, that they may be used as a stipulation in thermal analysis. Averaged heat transfer coefficients had to be created for all tread wheel operative rotation speeds and temperatures. Because the brake tread is formed from steel, the surface roughness is taken to be as smooth surface.

The tread wheel surface assumes square measure heated uniformly. Air round the tread is taken into account to be 25 °C and open boundaries with zero relative pressure were used for the higher, lower and radial ends of the domain. A spread of constant angular speeds was sculptured employing a results solely in cases wherever mean pressure gradients square measure small; accuracy has been shown by rotating frame of reference regarding the tread wheel and shaft axis. Material knowledge was taken from ANSYS material knowledge library for air at twenty five °C. Reference pressure was set to be one atm, turbulence intensity low and turbulent model used was k-å. K-å model has been shown to be helpful for free-shear layer flows with comparatively little pressure gradients. Similarly, for wall-bounded and internal flows, the model provides smart experimentation to be reduced for flows containing massive adverse pressure gradients.

Fluid Domain:

- Domain kind: Fluid Domain
- Domain name: compressed air
- Fluid: Air
- Domain Motion: stationary
- Heat Transfer: Thermal Energy

Solid Domain:

- Domain Type: Solid Domain
- Domain Name: tread wheel
- Material: steel
- Domain Motion: Rotating
- Rotational Speed: 66rpm

Domain Interface:

- Interface Type: Fluid and Solid

- Fluid Domain: Atmosphere and compressed air
- Solid domain: tread wheel

Boundary Conditions

Inlet:

- Boundary Type: Body of air
- Air Velocity: 100 m/s
- Turbulence: low (Intensity=1%)
- Temperature: 25 C

Outlet:

- Boundary Type: air body
- Pressure: One bar

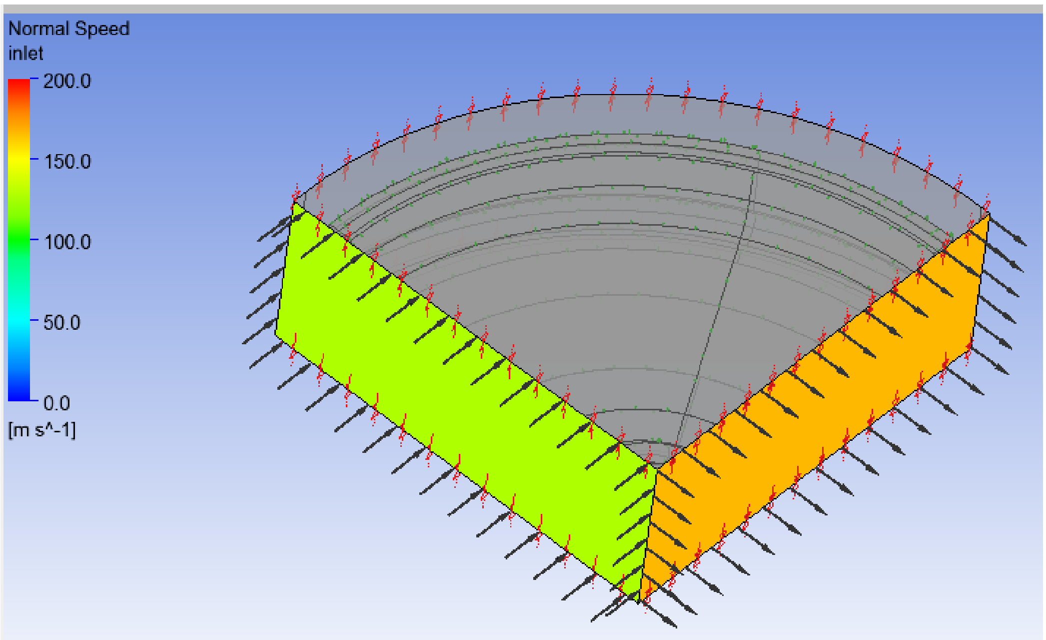

Figure 19. Boundary condition of computational fluid dynamics

Wall Boundary Conditions:

- Heat Transfer = Adiabatic
- Wall Influence on Flow = No Slip
- Wall Roughness = smooth wall

Problem Solver Stage: After shaping all the conditions the modeling analysis should be continuous with CFX. The following problem solver management parameters are per CFX-Pre Module.

Solver Management Parameters:

- Number of iterations: 20
- Time scale management: brake time of the train
- Residual Target: 1e-4

CFD result and discussion: From the computational fluid dynamics we have gotten the convection heat transfer coefficient. The maximum value of convection heat transfer is 1049 $wm^{-2}k^{-1}$. This value exported to the transient thermal analysis to cooling the tread brake.

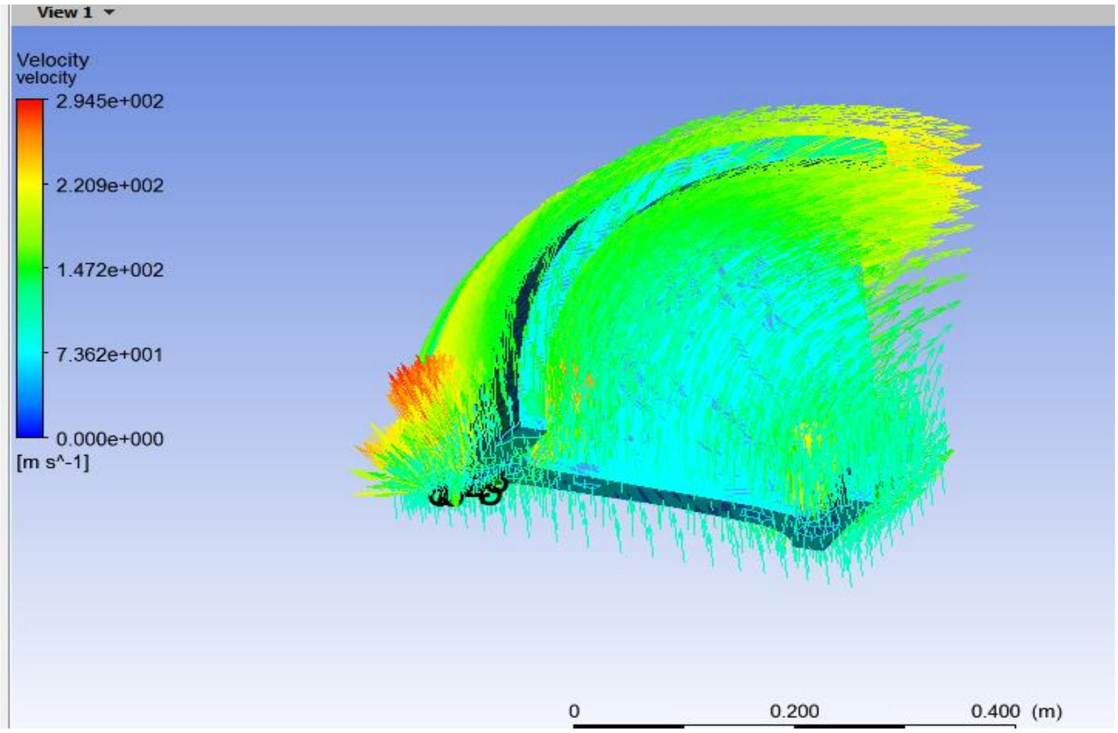

Figure 20 .Velocity of air follow vector on the wheel tread and convection heat transfer coefficient

Figure 21 is the convection heat transfer coefficient of output of the CFD simulation. From the figure we understand the maximum coefficient caused by the increment of air flow in the fluid domain internally. This is may be not the true value because on inlet velocity is maximum value. In practical situation since the out let opened to atmospheric air. But in the model of the CFD fluid domain increase the velocity. That's why the speed of air flow and heat transfer increased. In the another side when figure .19 look carefully the maximum(1249w/m2k) convection heat transfer coefficient on the flange of the tread wheel and the minimum (124.9 w/m^{2}k)value on the wheel tread part which exposed to the high heat energy because of the brake foundation.

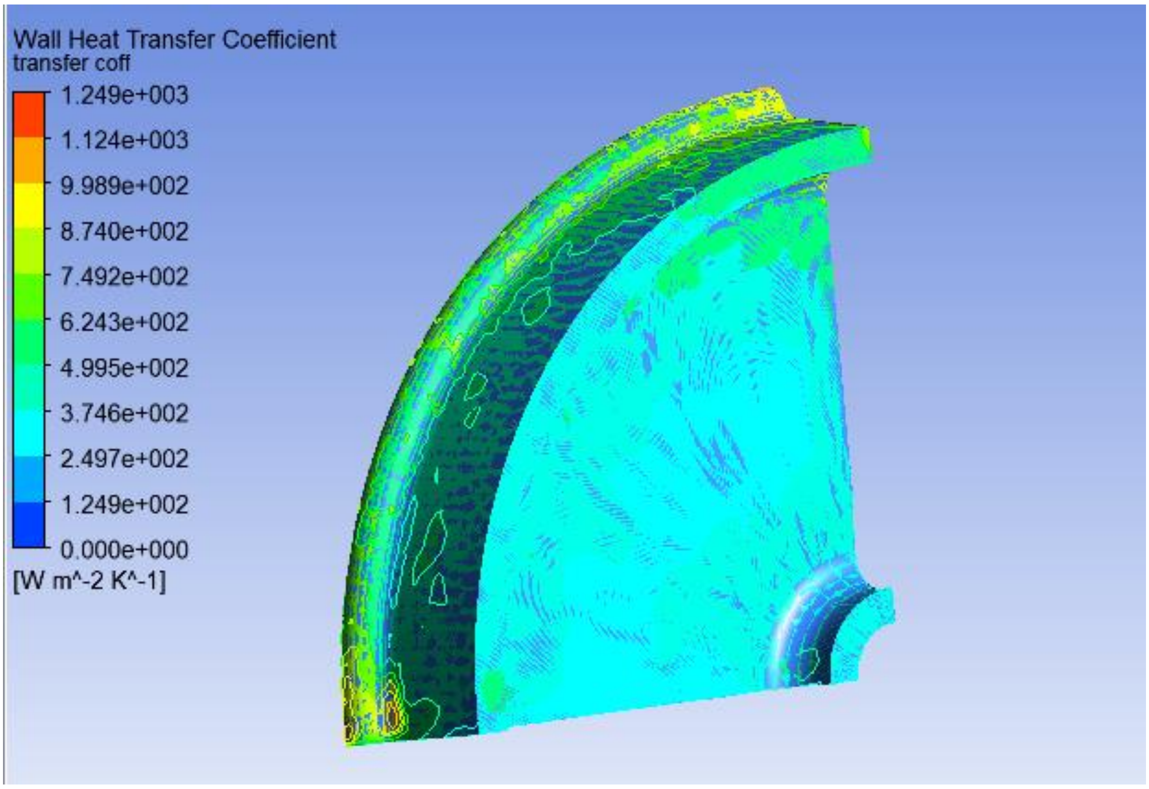

Figure 21. Heat transfer coefficient graph

From CFD output I have gotten convection heat transfer coefficient as input to the transient thermal analysis. This input additional with the free convection heat transfer coefficient because of the train speed calculated by equation.15.

Contours of heat transfer coefficient extracted from the CFD study are presented in Fig. 22 to show the relationship between the flow field characteristics presented in Fig. 21 and the heat transfer properties of the tread brake. The distribution of heat transfer coefficient over the tread wheel is non-uniform, with through in the low-velocity regions and peaks in the high-velocity regions. This is because high-energy dissipation occurs in the high-velocity regions and low-energy dissipation in the low- velocity regions.

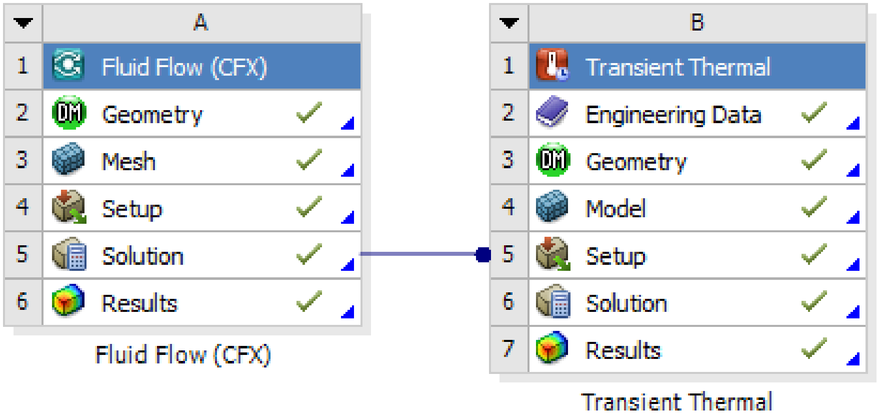

Figure 22. Combine of CFD and transient thermal analysis

Figure.22 show the combination of both CFD and thermal analysis to estimate the surface tread brake temperature. From the computational fluid dynamics we have extracted the convection heat transfer coefficient. This imported to the transient thermal analysis. As discussed in section of thermal analysis of tread brake without air cooling the step was the same.

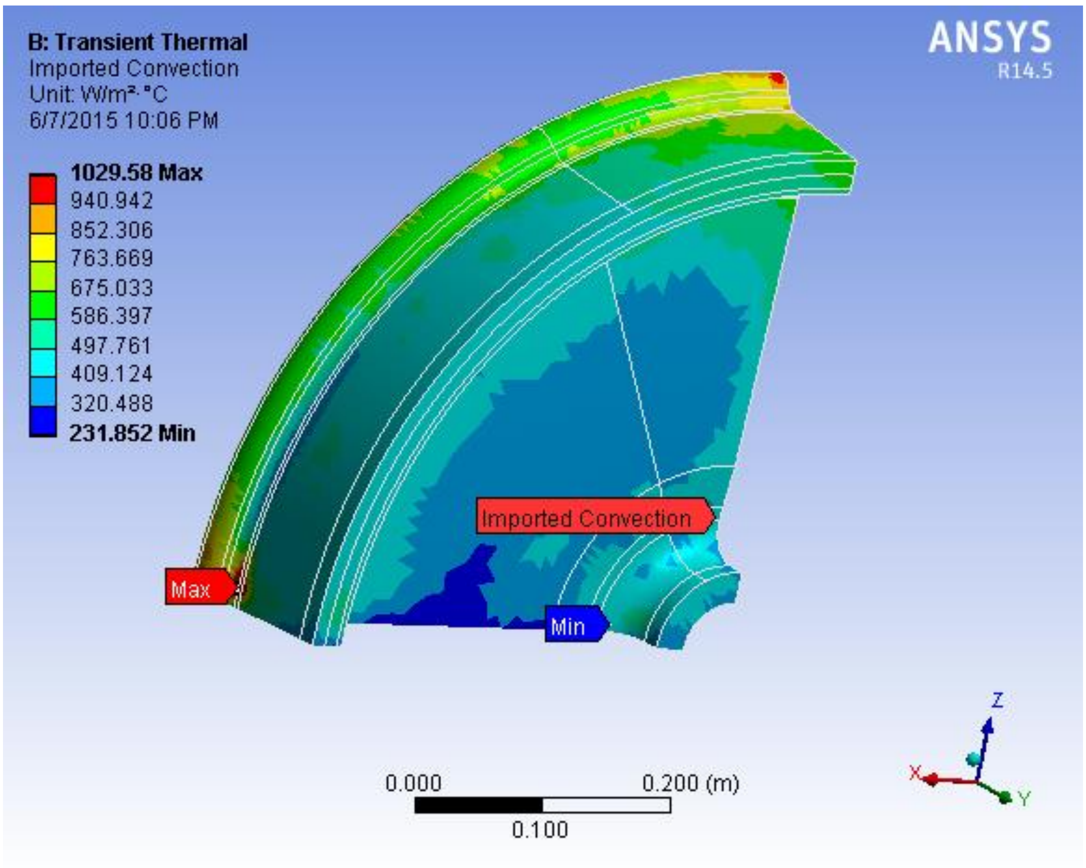

Figure 23. Inported convection heat transfer coefficient from the CFD

The imported convection heat transfer coefficient (figure.23) show that the heat transfer distribution is un-uniform .this is caused by the different factors. Like the quality of the geometry modeling, user knowledge of the CFD and supply air disruption. The maximum value show in the flange of wheel and the minimum value is on both faces (front and back wheel) of the wheel.

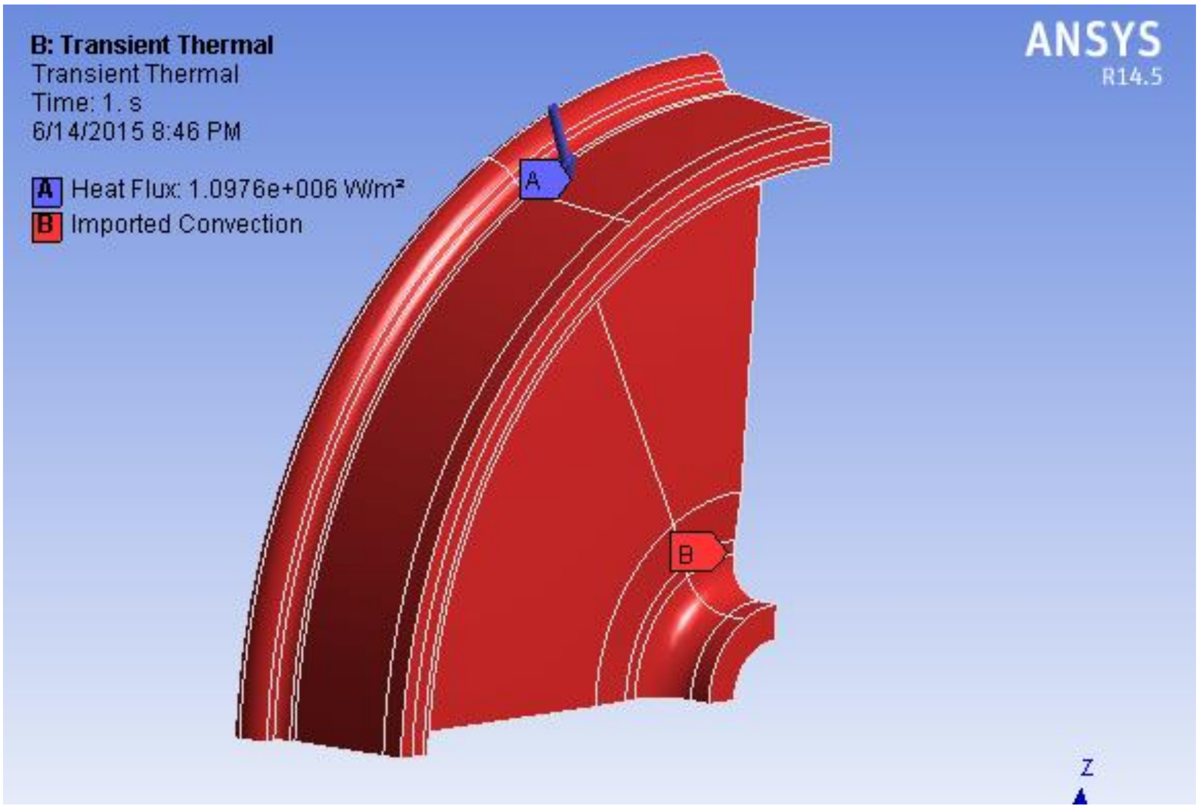

Figure 24. Boundary condition of forced air cooling at air velocity of 100m/s

After imported the convection heat transfer coefficient from the computational fluid dynamics I inserted the input values of free convection heat transfer coefficient at ambient temperature 25^{o}c and the heat flux of calculated with equation .13. Since the analysis is depending with time the brake time is 20 second. The thermal result is the maximum temperature 399.13^{o}c and the total heat flux is 1.6479 x10^6 w/m^2.

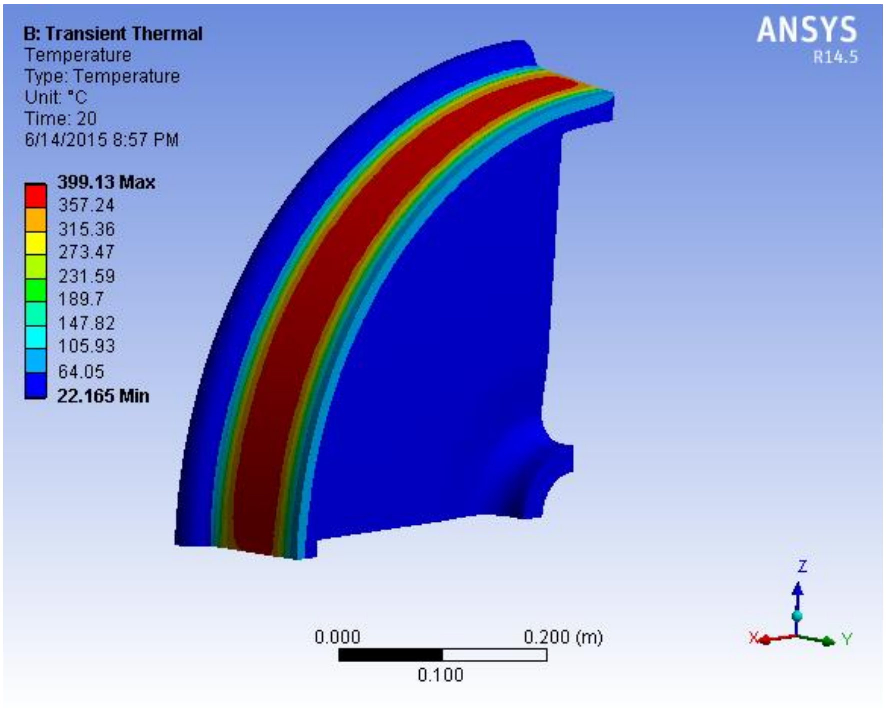

Figure 25. Temperature result with air cooling at air velocity 100m/s

Figure .25 shows the result of forced air cooling wall temperature of tread brake with the brake time 20seconed. Figure .26 shows the maximum total heat flux accumulated with length of 20 seconds.

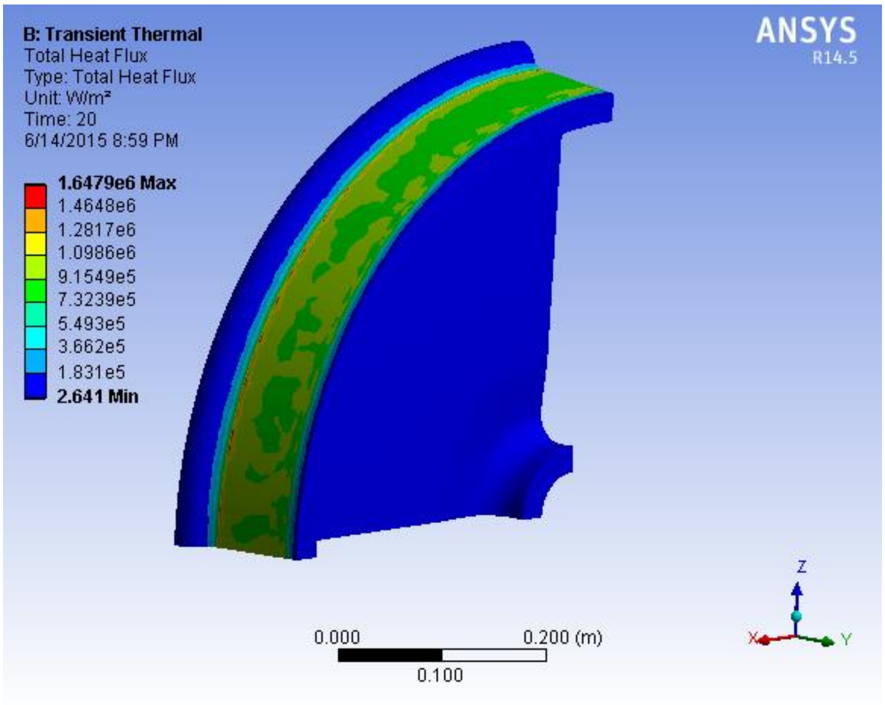

Figure 26. Total heat flux accumulated on the tread brake wall with air cooling @ v=100m/s

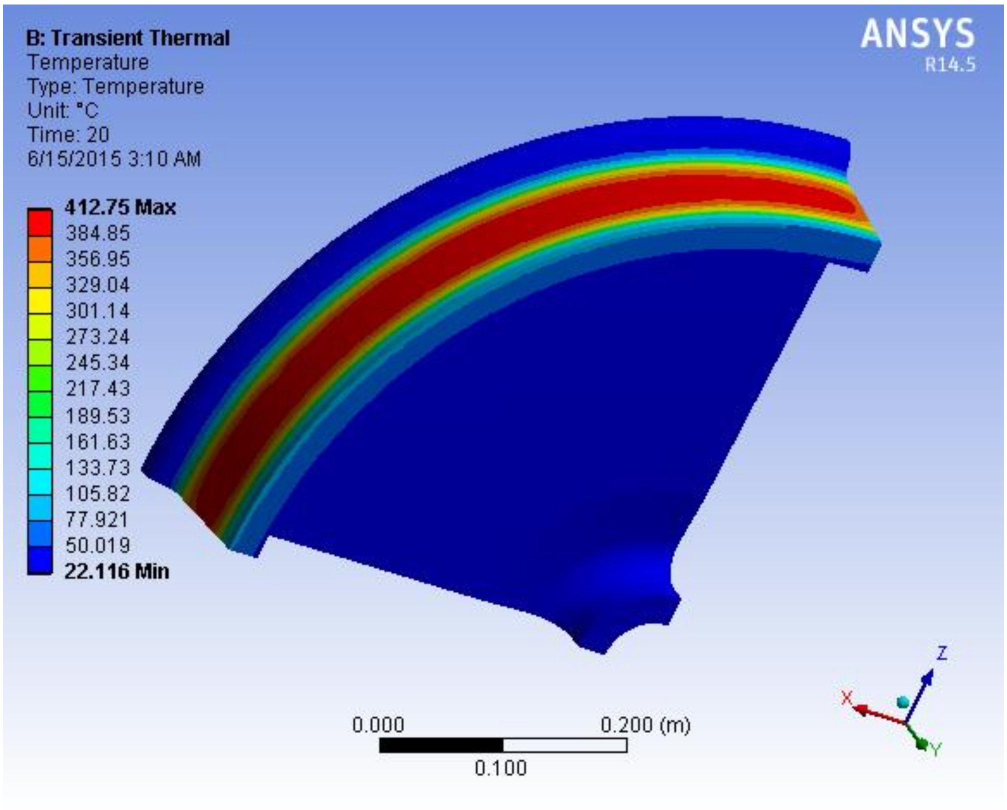

Figure 27. Forced air cooling wall temperature at air velocity of 50m/s

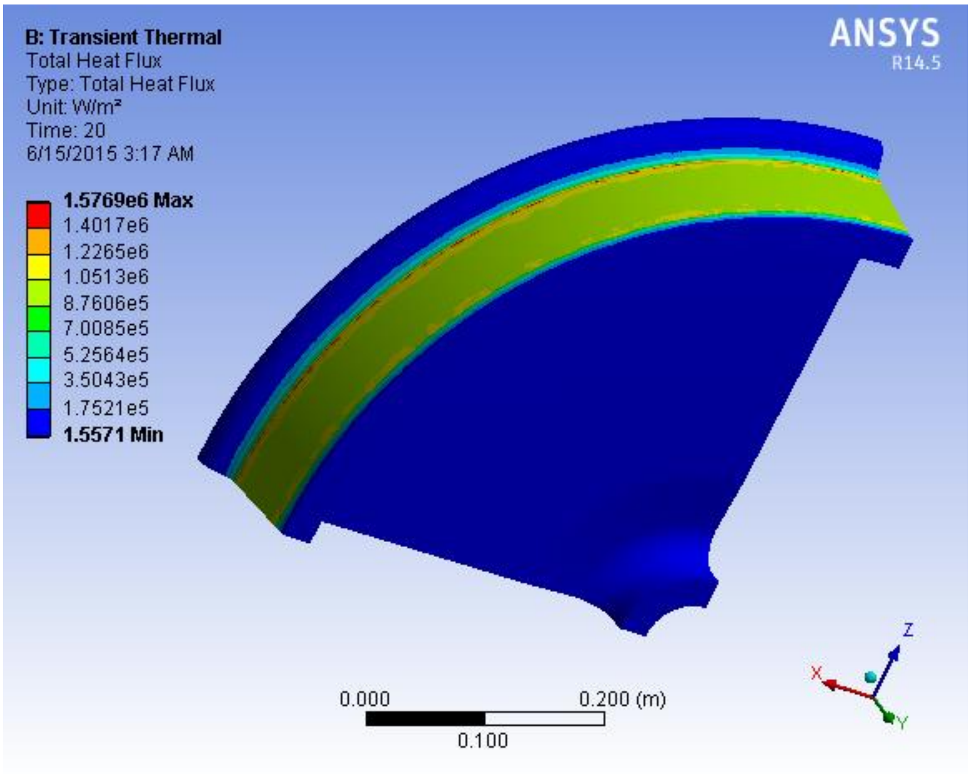

Figure 28. Total heat flux at air v=50 m/s

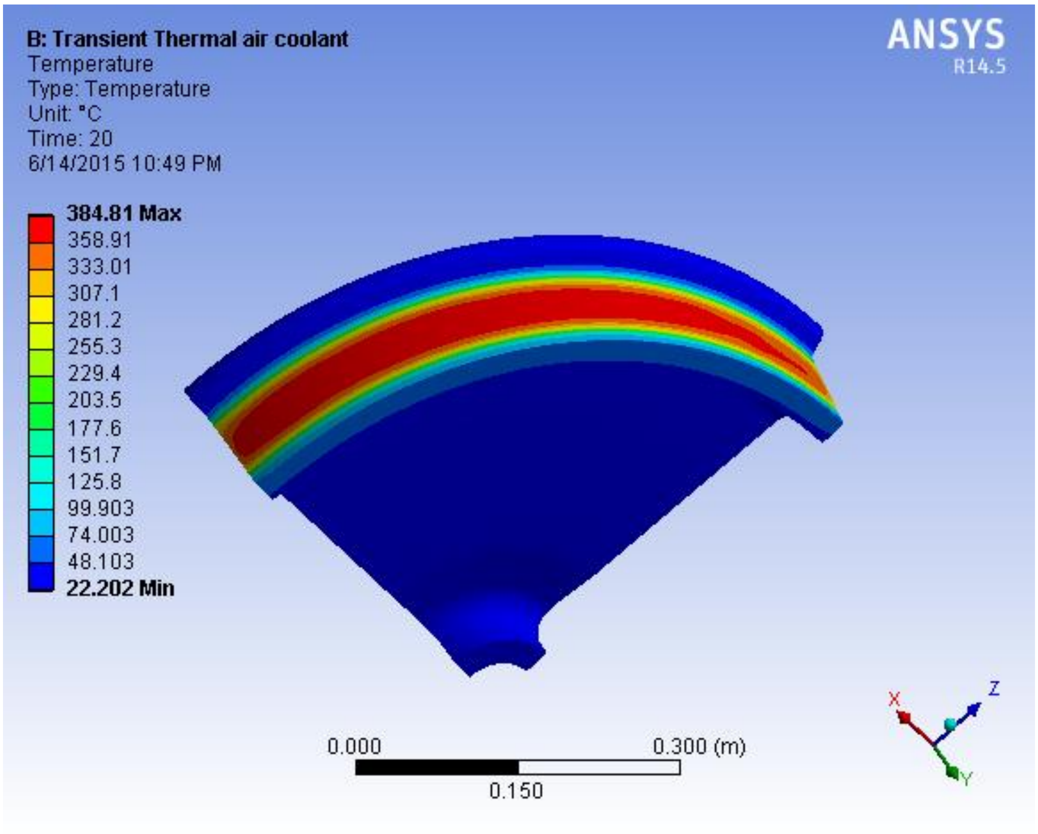

Figure 29. Temperature at air v = 150m/s

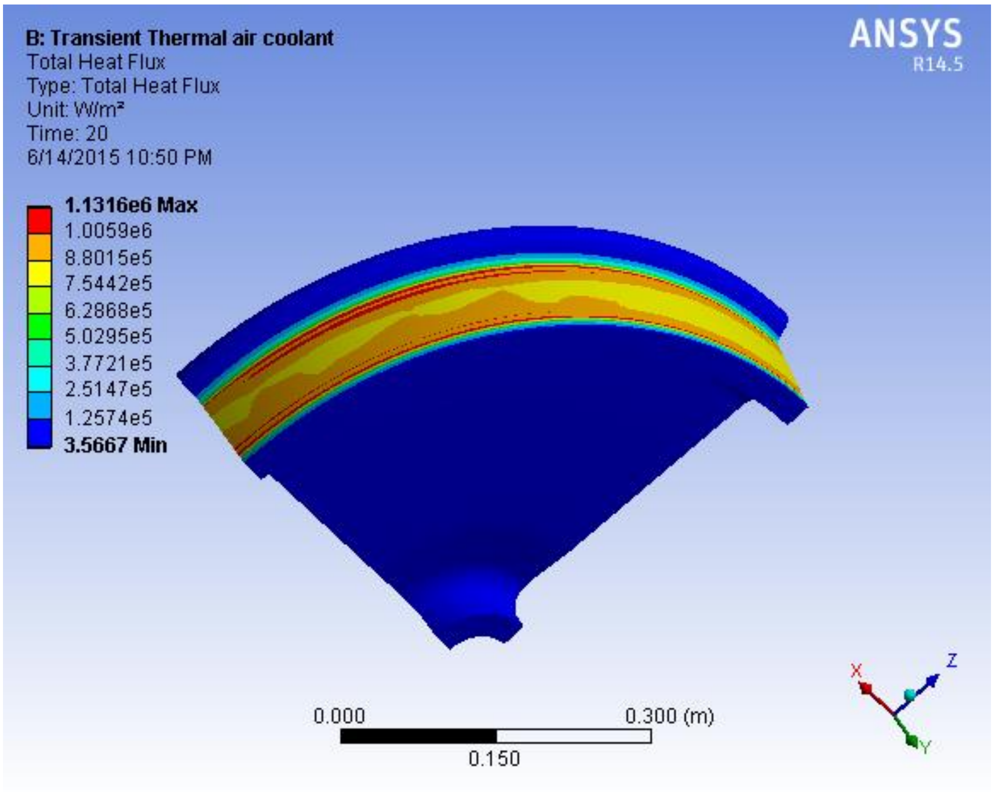

Figure 30. Total heat flux at air v = 150m/s

3.4 Compete and discussion both without and with forced air cooling thermal results

3.4.1 Temperature

From the above results with forced air cooling and without air thermal analysis have gotten difference in temperature and total heat flux. This showed the forced air cooling better than without cooling thermal analysis. Figural compete both without and with air cooling thermal analysis as following.

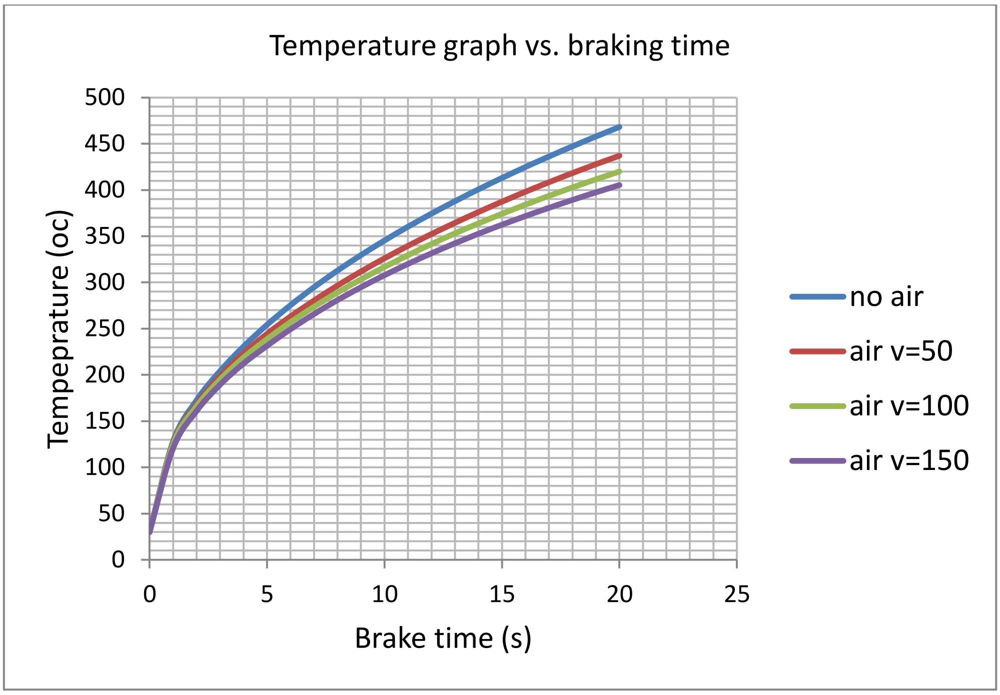

Figure 31. Temperature vs. Brake time

The above graph shows that the temperature graphs in the difference air velocity. T_1 is temperature graph result without air cooling with maximum temperature value of 432.43^{o}c. The $T_{2@v=50\ m/s}$ is the temperature graph with forced air cooling with air velocity of 50 m/s and the maximum temperature value 412.75^{o}c. The $T_{3@v=100m/s}$ is the temperature graph forced air cooling with air velocity maximum temperature value 399.13^{o}c. The $T_{4@v=150m/s}$ is temperature graph with forced air cooling and air velocity of 150m/s. Its maximum temperature is 384.81 ^{o}c.

From the graph when the air velocity is increase the cooling rate is increased. When we take T_1 (wheel wall temperature of without air cooling) as reference the efficient of cooling of the wheel tread is:

$$r = \frac{no\ air - air\ v_i}{no\ air} x100\% \tag{15}$$

Where:

r, is the temperature reduction percentile (%)

no air, is the temperature of wheel without air cooling

air v_i, is the wall temperature of wheel with air cooling at different velocity

Table 5. Maximum temperature difference (ΔT_{max}) and reduction efficiency (r)

No-	**ΔT_{max} (^oC)**	**r (%)**
1	19.68	4.55
2	33.3	7.7
3	47.62	11.012

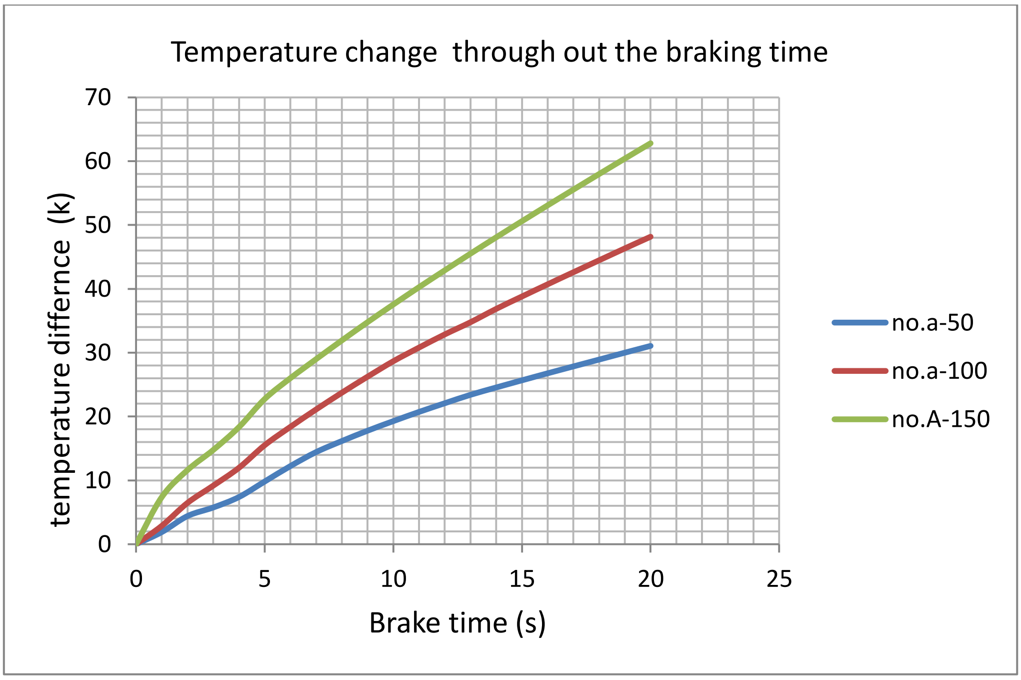

Figure 32. Relative Temperature difference graph

Figure 31. Showed that the temperature in three conditions have significant variations.

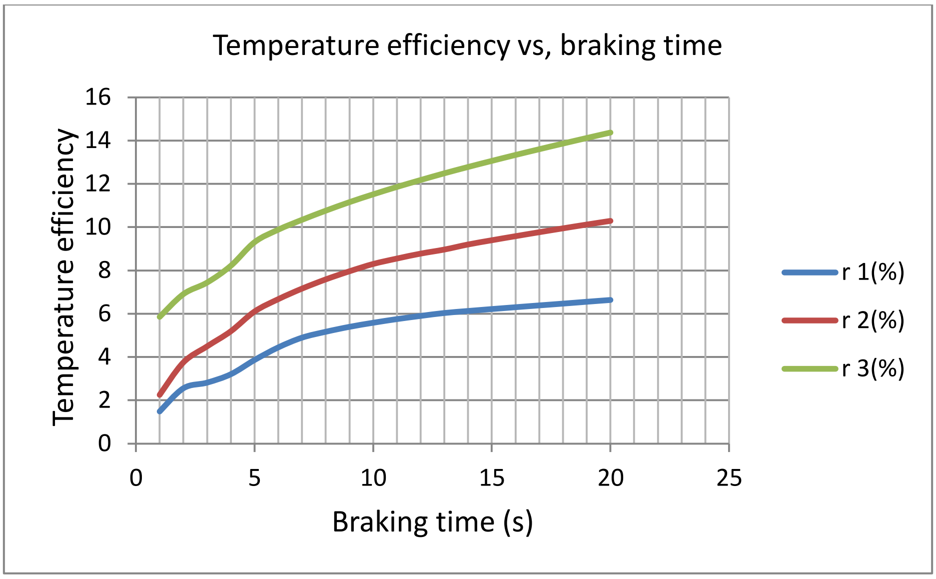

Figure 33. Air cooling efficiency graph

Figure.35. Induct that the cooling efficiency through braking time. Hence we have understood the cooling efficiency depend with air cooling speed. When cooling speed increased, the cooling efficiency parallel increased and the via verse.

Table 6 .Temperature results on four conditions approach

Time	No air	air v=50	air v=100	air v=150	No.A-50	No.A-100	No.A-150	r 1(%)	r2(%)	r 3(%)
0	30	30	30	30	0	0	0			
1	127.96	126.06	125.08	120.58	1.9	2.88	7.38	1.484839	2.250703	5.854355
2	172.55	168.14	166.08	160.94	4.41	6.47	11.61	2.555781	3.749638	6.90496
3	204.13	198.38	194.96	189.37	5.75	9.17	14.76	2.816832	4.492235	7.440266
4	230.63	223.24	218.66	212.29	7.39	11.97	18.34	3.204267	5.190131	8.215374
5	254.19	244.37	238.68	231.45	9.82	15.51	22.74	3.863252	6.101735	9.305561
6	275.4	263.16	257.02	249.39	12.24	18.38	26.01	4.444444	6.673929	9.883721
7	294.82	280.39	273.71	265.83	14.43	21.11	28.99	4.894512	7.160301	10.33917
8	312.8	296.64	289.07	280.88	16.16	23.73	31.92	5.16624	7.586317	10.76052
9	329.58	311.8	303.34	294.8	17.78	26.24	34.78	5.394745	7.961648	11.15459
10	345.35	326.07	316.69	307.8	19.28	28.66	37.55	5.582742	8.298827	11.51593
11	360.25	339.53	329.45	319.99	20.72	30.8	40.26	5.751561	8.549618	11.85757
12	374.37	352.29	341.51	331.47	22.08	32.86	42.9	5.897908	8.777413	12.17747
13	387.82	364.42	353.05	342.31	23.4	34.77	45.51	6.033727	8.965499	12.48834
14	400.67	376.12	363.81	352.6	24.55	36.86	48.07	6.127237	9.199591	12.7805

15	412.98	387.31	374.18	362.39	25.67	38.8	50.59	6.215797	9.395128	13.06189
16	424.8	398.03	384.09	371.72	26.77	40.71	53.08	6.301789	9.583333	13.33568
17	436.18	408.33	393.58	380.64	27.85	42.6	55.54	6.384979	9.76661	13.60174
18	447.17	418.25	402.7	389.19	28.92	44.47	57.98	6.467339	9.944764	13.86252
19	457.8	427.8	411.47	397.4	30	46.33	60.4	6.55308	10.12014	14.11875
20	432.43	412.75	399.13	384.81	19.68	33.3	47.62	4.55	7.7	11

3.4.2 Total heat flux:

Figure.33 shows the total heat flux on the tread brake at the different air velocity and without cooling brake tread. q_1 represent the value of total heat flux without air brake and 2, 3 & 4 are represent the heat flux values of with air cooling at the different air velocity. Therefore we can be understood the value of heat flux accumulated on the tread brake gradually decline through air speed increase.

The Interest point is the improvement of total heat flux on the tread brake is large. When we see the efficiency total heat flux accumulated on the tread brake in the four conditions have been significant values. Since this research main goal is to minimize the risk of thermal fracture and brake fad.

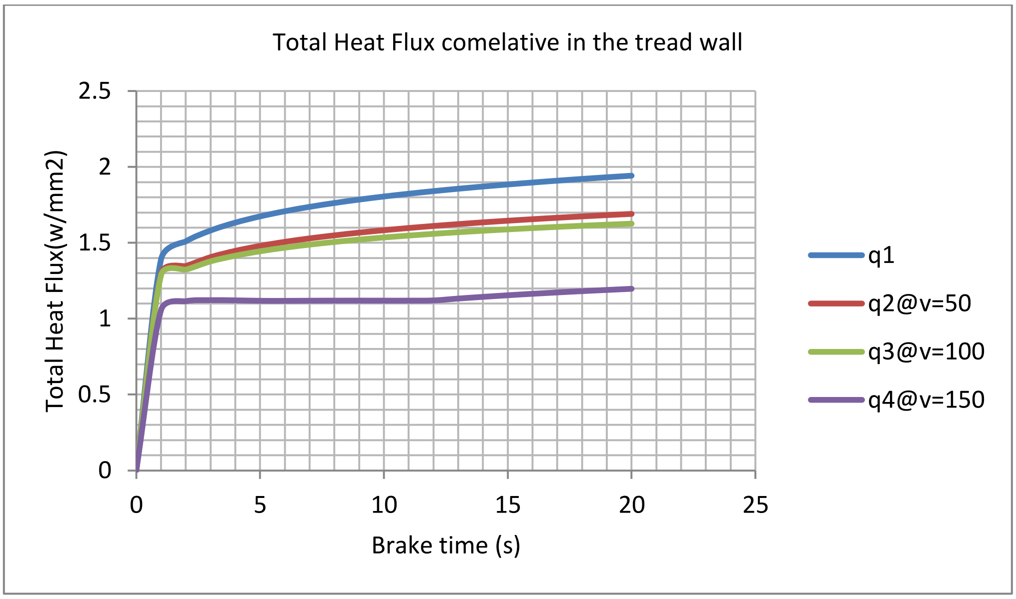

Figure 34 Total Heat Flux vs. Brake time

Generally from the above results we understand that cooling thermal analysis depended with air velocity.

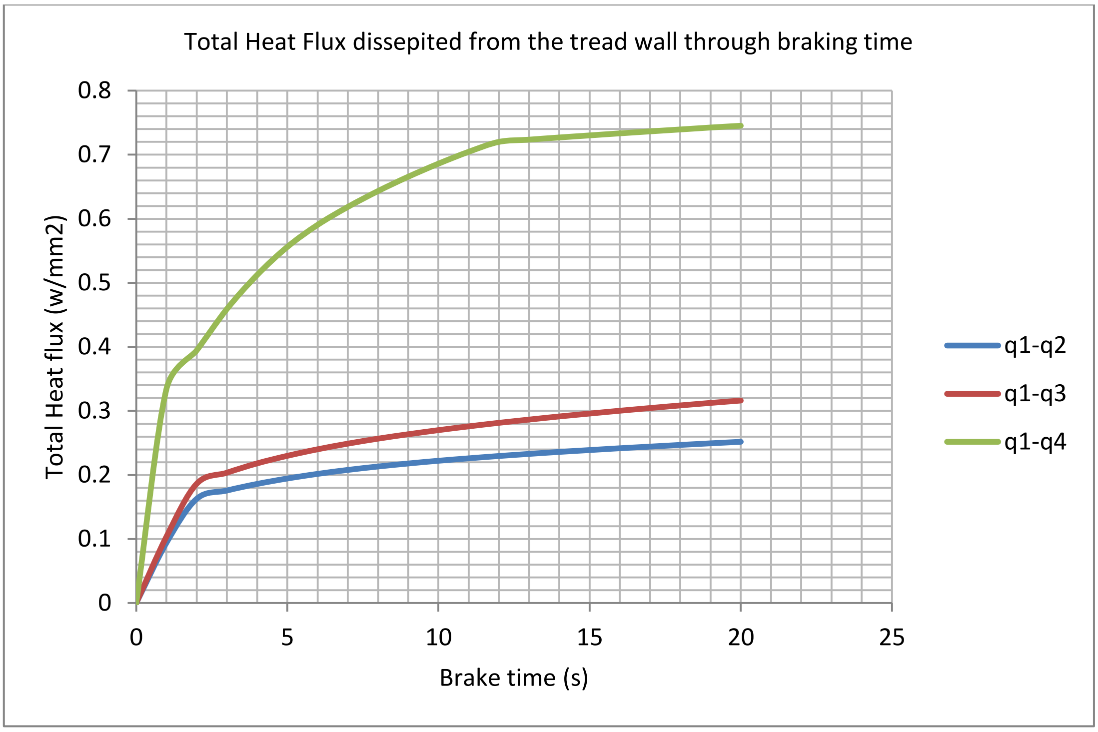

Figure 35 Relative total heat flux difference

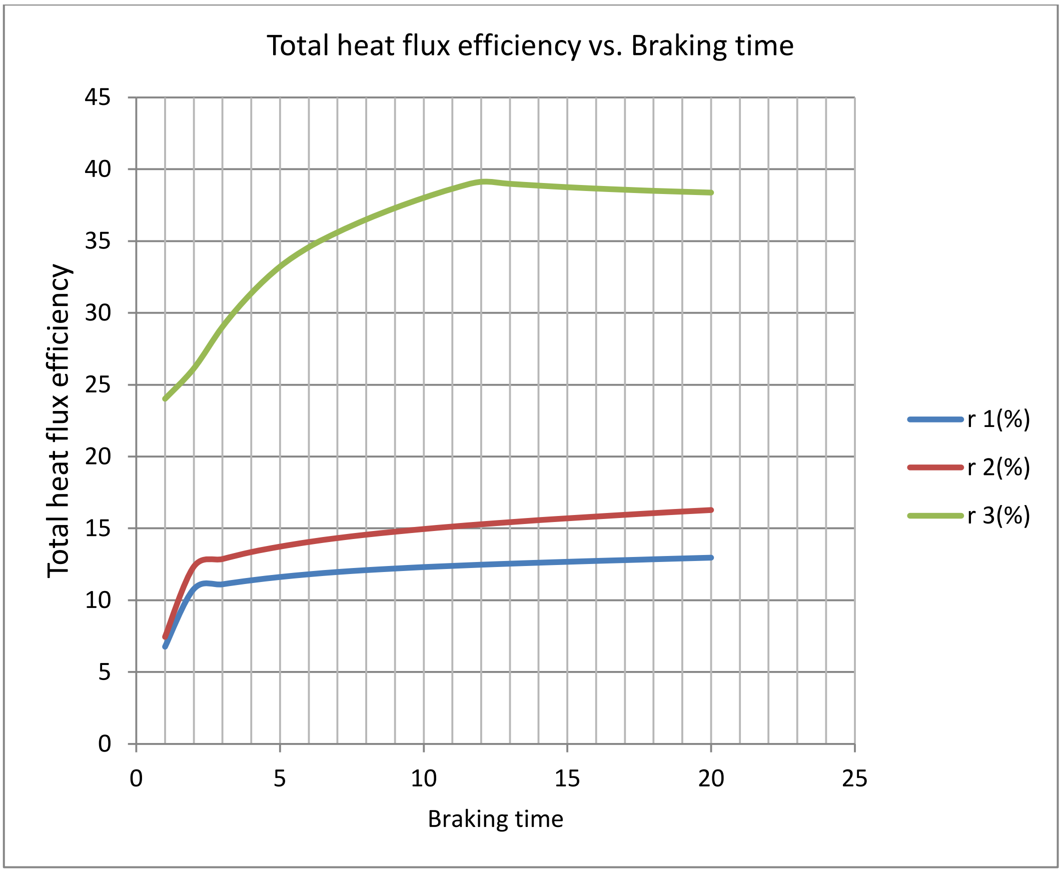

Figure 36 Total heat flux efficiency

Table 7 Total heat flux in the four condition approach

time	q1	q2@v=50	q3@v=100	q4@v=150	q1-q2	q1-q3	q1-q4	r 1(%)	r 2(%)	r 3(%)
0	0	0	0	0.00E+00	0	0	0.00E+00			
1	1.3909	1.2968	1.2873	1.0568	0.0941	0.1036	3.34E-01	6.765404	7.448415	2.40E+01
2	1.5096	1.3469	1.3234	1.1152	0.1627	0.1862	3.94E-01	10.77769	12.33439	2.61E+01
3	1.5813	1.4056	1.3778	1.1222	0.1757	0.2035	4.59E-01	11.11111	12.86916	2.90E+01
4	1.6332	1.4472	1.4151	1.121	0.186	0.2181	5.12E-01	11.38868	13.35415	3.14E+01
5	1.6742	1.4797	1.4443	1.1181	0.1945	0.2299	5.56E-01	11.61749	13.73193	3.32E+01
6	1.7082	1.5065	1.4681	1.1177	0.2017	0.2401	5.91E-01	11.80775	14.05573	3.46E+01
7	1.7372	1.5293	1.4883	1.1187	0.2079	0.2489	6.19E-01	11.96753	14.32765	3.56E+01
8	1.7625	1.5493	1.5058	1.1192	0.2132	0.2567	6.43E-01	12.09645	14.56454	3.65E+01
9	1.785	1.5671	1.5214	1.1193	0.2179	0.2636	6.66E-01	12.20728	14.76751	3.73E+01
10	1.8053	1.5831	1.5353	1.1192	0.2222	0.27	6.86E-01	12.3082	14.95596	3.80E+01
11	1.8237	1.5977	1.5478	1.1191	0.226	0.2759	7.05E-01	12.39239	15.12858	3.86E+01
12	1.8407	1.6111	1.5593	1.1206	0.2296	0.2814	7.20E-01	12.47352	15.28766	3.91E+01
13	1.8564	1.6235	1.5699	1.1328	0.2329	0.2865	7.24E-01	12.54579	15.4331	3.90E+01
14	1.871	1.635	1.5796	1.144	0.236	0.2914	7.27E-01	12.61358	15.57456	3.89E+01
15	1.8846	1.6457	1.5887	1.1544	0.2389	0.2959	7.30E-01	12.67643	15.70094	3.87E+01
16	1.8975	1.6558	1.5972	1.1641	0.2417	0.3003	7.33E-01	12.73781	15.82609	3.87E+01
17	1.9096	1.6653	1.6051	1.1731	0.2443	0.3045	7.37E-01	12.79326	15.94575	3.86E+01
18	1.9211	1.6742	1.6125	1.1816	0.2469	0.3086	7.40E-01	12.85201	16.06371	3.85E+01
19	1.932	1.6826	1.6195	1.1895	0.2494	0.3125	7.43E-01	12.9089	16.17495	3.84E+01
20	1.6502	1.6479	1.5769	1.1316	0.0023	0.0733	0.51816	1.4	10.44	31.5

Where:

q1 = total heat flux on the end of brake time on without air coolant wall tread wheel

q2 = total heat flux on the end of brake time on forced air cooled wall tread wheel at the velocity of air 50 m/s

q3 = total heat flux on the end of brake time on forced air cooled wall tread wheel at the velocity of air 100 m/s

q4 = total heat flux on the end of brake time on forced air cooled wall tread wheel at the velocity of air 150 m/s

CHAPTER FOUR

CONCLUSION AND RECOMMENDATION

4.1 CONCLUSION

The main goal of this study is to improve heat dissipation of tread brake. Unfortunately, the heat dissipation is depends on material property of the tread brake, velocity of air cooling and mass flow of air cooling. The work was focused on the velocity of the air cooling.

The present study can provide a useful design tool to analyze and in order to improve the tread brake heat dissipation performance of tread brake system.

In generally the improvement of heat dissipation of tread brake using air cooling is related with the air velocity flowed to the tread brake. Which is as increase the velocity of air cooling we have gotten good heat dissipation. Because the heat transfer of air with tread wall is increased .Therefor to improve heat dissipation of tread brake we should be provides air cooling. Since the air availability is free from the compressed main reservoir of air brake tank and brake pipe. Thermal analysis from the above detail discussion and graphs introduced briefly. The heat dissipation improvement at the different cooling velocity have different rate.

The total heat flux accumulates on the tread brake without air cooling brake has been `introduced in the above. Using forced air cooling will not only reduce the maximum temperature, but also naturally tend to reduce the thermal gradients.

4.2 RECOMMENDATION

From the above result we can recommend that the tread brake improvement of heat dissipation was satisfactory .The power need to supply cooling air is expect 0.32kwatts. We should be proved the cooling line pipe and automatic pneumatic valve from the main brake pipe and main reservoir tank. Figure.7 (air brake system with air cooling supply) shows the air supply piping and control system.

Since the rail way tread wheel can be exposed to difference forces and thermal effects, the cooling improvement is necessary. The forces effect on rail way wheel treads are compression stress, lateral force, and friction force between rails and tread wheel and braking force. Additionally atmosphere effect has its own roll.

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