ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

THESIS ON

'THERMO-MECHANICAL AND LIFE TIME ANALYSIS OF AA-LRT DISC BRAKE UNDER DIFFERENT SPEED AND LOAD DURING EMERGENCY BRAKING'

A Thesis Submitted to the School of Graduate Studies of Addis Ababa University in Partial Fulfilment of the Requirements for the Degree of Masters of Science in Mechanical Engineering

(Railway stream)

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DECLARATION

I, the undersigned, declare that this thesis entitled ‘THERMO-MECHANICAL AND LIFE TIME ANALYSIS OF AA-LRT DISC BRAKE UNDER DIFFERENT SPEED AND LOAD DURING EMERGENCY BRAKING’is original work of mine and has not been presented for any degree in any university and all the sources of materials used for the thesis have been duly acknowledged.

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ACKNOWLEDGMENT

I owe my deepest gratitude to my advisor, Mr. Tsegaye Feleke. It was he who led me into the field of Braking research and helped me tremendously throughout the final. I feel very fortunate to study under his guidance.

I am grateful for the great opportunity I had to conduct Thesis in the field of Railway Engineering. I would also like to express my thanks to all AAIT Lecturers from whom I have taken courses and learned knowledge.

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ABSTRACT

The main purpose of this study is to analyze the thermo-mechanical behavior of the dry contact between the brake disc and pads under different speed and load of the AA-LRT train during emergency braking. CATIA software was used for the modeling purpose. The thermo-mechanical analysis of the disc brake was carried out using ANSIS workbench. In this work, a transient thermal and structural analysis was carried out to investigate the temperature variation across the disc, total heat flux using ANSYS software. The thermal-structural analysis is then used to determine the deformation established and stresses in the disk. And also the life time of the disc was analyzed using ANSIS workbench. In all cases as vehicle speed and load of the vehicle increases both temperature and Total heat flux increases. But For a constant speed of vehicle as the load increases both the Equivalent stress and Total deformation increases. But if speed of the vehicle increases both Equivalent stress and Total deformation increases for all loads. As the load and speed of the vehicle increases the life time of the vehicle decreases. Moving the train with speed of 22.342 m/s shortens the life disc brake.

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NOMENCLATURE

k_x= thermal conductivity in x-direction

k_y = thermal conductivity in y-direction

k_z= thermal conductivity in z-direction

Cp= specific heat

ρ = specific mass

Q= internal heat generation rate per unit volume

T = temperature that varies with the coordinates as well as the time

h =convective surface heat transfer coefficient.

T_1= specified surface temperature

q_s =specified surface heat flux

T_s = unknown surface temperature

T_∞ =convective exchange temperature

ε =material's emissivity

σ= Stefan-Boltzmann constant

$M = mass\ of\ the\ vehicle$

K= correction factor for rotating parts

λ= proportion of heat entering the brake disc

s_b= braking distance of the vehicle

F_T ==Tractive force

F_D= Drag force

F_R = Rolling resistance force

F_G = Total weight component in the direction of the inclined plane

a_T = train acceleration

a= deceleration of the train

T_m= motor torque

$P = power$

ω= angular velocity

T_W= $torque\ on\ the\ wheel$

$\eta = mechanical\ efficiency\ of\ transmission$

τ = gear ratio of transmission

R_W = wheel radius

α = gradient angle of the rail track

μ_R= rolling resistance coefficient

g = acceleration due to gravity

p_{ab}= absolute pressure

$R_{specific}$= Specific gas constant for dry air

t_b=braking timerequired to stop the vehicle

A_d= contact swept area

r_{op}= Outer pad radus

r_{ip}= inner pad radus

Re=Reynolds number

Pr=Prandtl number

k_a=thermal conductivity

μ_{va}=dynamic viscosity

C_{pa}=specific heat capacity

P=clamping pressure

F_{rotor}=rotor force

$F_{caliper}$=caliper force

V1= @speed of 40$^{Km}/_{h}$

V2=@speed of 70$^{Km}/_{h}$

V3=@speed of 80.431$^{Km}/_{h}$

M1= mass of 44000Kg

M2= mass of 59240Kg

M3= mass of 63020Kg

CHAPTER ONE

INTRODUCTION

1.1BACKGROUND

Rail transport is a means of conveyance of passengers and goods, by way of wheeled vehicles running on rails. It is also commonly referred to as train transport. In contrast to road transport, where vehicles merely run on a prepared surface, rail vehicles are also directionally guided by the tracks on which they run. Track usually consists of steel rails installed on ties (sleepers) and ballast, on which the rolling stock, usually fitted with metal wheels, moves. However, other variations are also possible, such as slab track where the rails are fastened to a concrete foundation resting on a prepared subsurface. [1]

Rolling stock in railway transport systems generally has lower frictional resistance when compared with highway vehicles and the passenger and freight cars (carriages and wagons) can be coupled into longer trains. The operation is carried out by a railway company, providing transport between train stations or freight customer facilities. Power is provided by locomotives which either draw electric power from a railway electrification system or produce their own power, usually by diesel engines. Most tracks are accompanied by a signaling system. Railways are a safe land transport system when compared to other forms of transport. Railway transport is capable of high levels of passenger and cargo utilization and energy efficiency, but is often less flexible and more capital-intensive than highway transport is, when lower traffic levels are considered.

The oldest, man-hauled railways date back to the 6th century B.C, with periander, one of the seven sages of Greece, credited with its invention. Rail transport blossomed after the British development of the steam locomotive as a viable source of the power in the 18th and 19th centuries. With steam engines, it was possible to construct mainline railways, which were a key component of the industrial revolution. Also, railways reduced the costs of shipping, and allowed for fewer lost goods, compared with shipping, which faced occasional sinking of ships. The change from canals to railways allowed for "national markets" in which prices varied very little from city to city. Studies have shown that the invention and development of the railway in Europe was one of the most important technological inventions of the late 19th century for the United States, without which, GDP would have been lower by 7.0% in 1890.

In the 1880s, electrified trains were introduced, and also the first tramways and rapid transit systems came into being. Starting during the 1940s, the non-electrified railways in most countries had their steam locomotives replaced by diesel-electric locomotives, with the process being almost complete by 2000. During the 1960s, electrified high-speed railway systems were introduced in Japan and a few other countries. Other forms of guided ground transport outside the traditional railway definitions, such as monorail or maglev, have been tried but have seen limited use.

The Imperial Railway Company of Ethiopia was a firm founded in 1894 to build and operate a railway across eastern Ethiopia from the port of Djibouti to the capital of Addis Ababa during Emperor Menelek II. It was founded by Alfred Ilg(advisor to Emperor Menelek II) and Léon Chefneux and headquartered in Paris, France. The firm failed in 1906 when political discord halted construction, and it failed to obtain any new capital. The portion it had completed ran from Djibouti to just short of Harar, the principal entrepôt for existing commerce in southern Ethiopia.Its terminus evolved into the city of Dire Dawa, today a larger city than Harar itself.

When Menelek acceded to the throne in 1889, negotiations began anew and a decree was granted on February 11, 1893, to study the construction of rail line. Ilg, a Swiss citizen, and a number of French associates put together a firm and received a royal charter on March 9, 1894, enabling them to start work. Menelek resisted personally putting any funds into the venture, but did grant a 99-year lease to Ilg and his associates in return for a number of shares in the firm and half of all profits in excess of 3,000,000 francs. Furthermore, the firm was obliged to construct a telegraph line along the route.

It took until 1897 before the necessary permission from French authorities was received, by which time significant opposition in Ethiopia had materialized.

Despite the shortfall, construction began in October 1897 from Djibouti, a hitherto minor port that expanded primarily to serve the railway and the first commercial service began in July 1901, from Djibouti to Dire Dawa.After a long time construction began anew. By 1915 the line reached Akaki, only 23 kilometers from the capital, and two years later came all the way to Addis Ababa itself. But the ETHIO-DJIBUTI line did not give services for a long period of time.

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.

The network will be 5,000 km long, and radiate from Addis Ababa. It will be constructed in two phases, the first phase involving the construction of five lines. Among the five lines Addis Ababa light rail transport (AALRT) is one of them.

Railway vehicle

Railway vehicle has a lot of systems. The braking system represents one of the most fundamental safety-critical components in train vehicles. The ability of the braking system to bring a vehicle to safe controlled stop is absolutely essential in preventing accidental vehicle damage and personal injury [2].Therefore, the braking system of a vehicle is undeniably important, especially in slowing down or stopping the rotation of a wheel by pressing brake pads against rotating wheel discs.

For high speed train, the braking technology get fast pace worldwide especially in China, even the main brake method is electricity or eddy current brake but frictional brake is a redundancy backup and reliable way to stop the train in emergency. Most form of frictional bakes on the train is disc brake type mounted on the train axels to dissipate brake energy.
The ability of a braking system to provide safe, repeatable stopping is the key to safe motoring. To stop the wheel, friction material in the form of brake pads is forced hydraulically, against

both sides of the disc. The purpose of friction brakes is to decelerate a vehicle by transforming the kinetic energy of the vehicle to heat, via friction, and dissipating that heat to the surroundings. So friction based braking systems are still the common device to convert kinetic energy in to thermal energy, through friction between the brake pads and the disc faces.

Disc-style brakes development and use start at England in the 1890’swhichis the first ever automobile disc brakes were patented by F.W.Lanchester [3].
ItwaspatentedatBirminghafactoryin1902, though it took another half century for the innovation to be widely adopted.
The first designs resembling modern style disc brakes began to appear in Britain in the late 194 and early 1950.ThefirstappearedonthelowvolumeCrosleyHotshotin1949,althoughithadtobe discontinued in 1950duetodesignproblems.Modernstylediscbrakesofferedmuchgreater stopping performance than comparable drum brakes, including much greater resistance to "brake fade “which is caused by the overheating of brake components.Meanwhile,fromthelate1990 to present, North American automotive industry accelerated the pace on brake research and application to catch up with Japanese quality performance. It has been more tailored towards American vehicle brake designs which often have more challenges to balance between the brake performance sand quality[4].
Disc brakes are widely used for it simple, powerful and easy maintenance and allow new materials of disc and pad to be brought into use to provide effective friction and minimize the wear.

The premature wear and thermal cracking of brake discs are attributed to high thermal stress, as the speed of train getting faster and during brake process, more heat need to be distribute to brake disc and pad friction pair, which cause higher thermal stress.

In this study, analysis of the thermo-mechanical behaviour of the dry contact between the discs of brake pads under different speed and load during emergency braking will be made; the strategy of calculation is based on the software ANSYS workbench.

1.2 VEHICLE BRAKING

A brake is a device by means of which artificial frictional resistance is applied to moving train vehicle, in order to stop the motion of the train vehicle. In the process of performing this function, the brakes absorb either or both kinetic energy or potential energy of the moving vehicle. The energy absorbed by brakes is dissipated in the form of heat. This heat is dissipated in the surrounding atmosphere to stop the vehicle, so the brake system should have the following requirements:-

- The brakes must be strong enough to stop the vehicle with in a minimum distance in an emergency.
- The driver must have proper control over the vehicle during braking and the vehicle must not skid.
- The brakes must have well anti fade characteristics i.e. their effectiveness should not decrease with constant prolonged application.
- The brakes should have well anti wear properties.

Classification of brakes (based on transformation of energy)

- Hydraulic brakes.
- Electric brakes.
- Mechanical brakes.

The mechanical brakes according to the direction of acting force are divided in to two groups.

Radial brakes

In these brakes the force acting on the brake drum is in radial direction. The radial brake may be subdivided into external brakes and internal brakes. Example, Tread brake (brake shoe)

Axial brakes

In these brakes the force acting on the brake drum is only in the axial direction. Example, Disc Brakes, Cone brakes..

A more advanced form of the disc brake is the ventilated or vented disc, where internal cooling is achieved by air flowing through radial passages or vanes in thedisc.

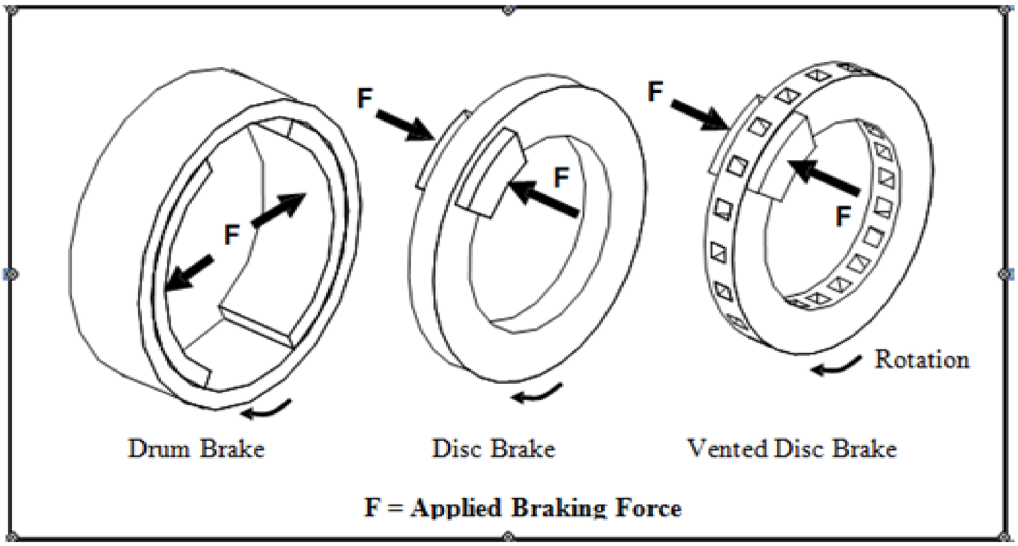


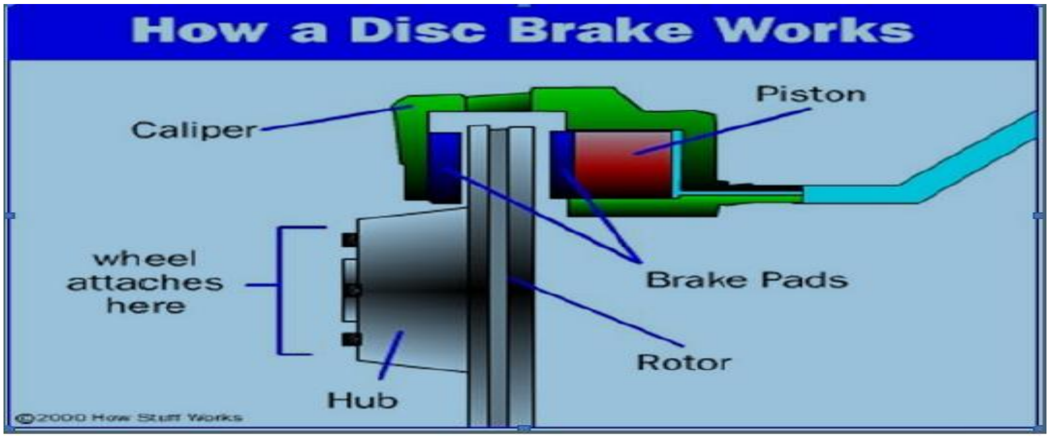
Figure1.1SchematicView of Drum and Disc Brakes [24]

Disc brakes:-

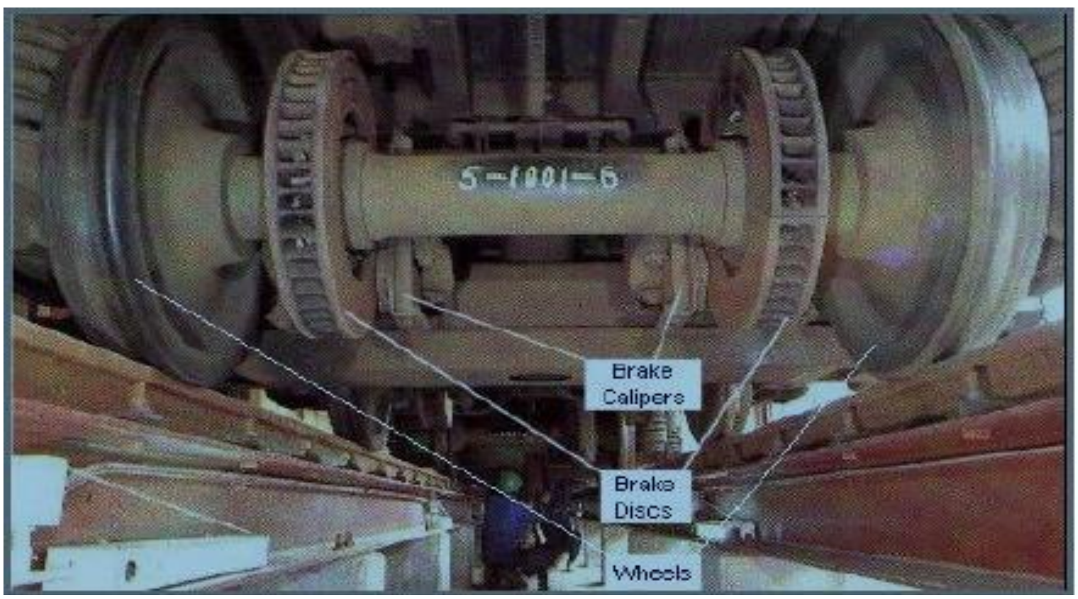
A disc brake consists of a cast iron disc bolted to the wheel hub or fitted (mounted) to the axle and a stationary housing called caliper. The caliper is connected to some stationary part of the vehicle, like the axle casing or the stub axle and is cast in two parts, each part containing a piston. In between each piston and the disc, there is a friction pad held in position by retaining pins, spring plates etc. passages are drilled in the caliper for the fluid to enter or leave each housing. These passages are also connected to another one for bleeding. Each cylinder contains rubber-sealing ring between the cylinder and piston. A schematic diagram is shown in the figure.

Principle:-

The principle used is the applied force (pressure) acts on the brake pads, which comes into contact with the moving disc. At this point of time due to friction the relative motion is constrained.



(a)



(b)

Fig.1.2 working principle of disc brake bolted to the wheel (a) and fitted (mounted) to the axle (b) [4, 5]

Working principle of disc brake:-

When the brakes are applied, hydraulically actuated pistons move the friction pads in to contact with the disc, applying equal and opposite forces on the later. On releasing the brakes the rubber-sealing ring acts as return spring and retracts the pistons and the friction pads away from the disc.

The main components of the disc brake are:-

- The Brake pads
- The caliper, which contains the piston
- The Rotor, which is mounted to the hub

1.3 PROBLEM STATEMENT

Most of railway vehicles are using disc brakes for braking. The main problem of braking and stopping a heavy railway vehicle is great heat load into brake disc in a very short time. The friction forces generated during braking between brake pads and discs produce high thermal gradients on the rubbing surfaces. These thermal gradients may cause braking problems such as brake fade, premature wear or hot spotting. An overview of past research showed that brake discs for rail vehicles are mostly tested on thermal loads and their effects. Long repetitive braking leads to temperature rise of various components of vehicle brake system that reduce the performance of braking. There were so many researches done on thermal structural analysis of disc brake, but most of the researches are concentrated on analysis of disc brake with respect to normal speed and operating braking. But, this thesis is done at different speed and mass during emergency braking conditions both analytical and finite element simulations.

High temperatures during braking may cause premature wear, bearing failure, thermal cracks and thermal excited vibration. Since the condition of braking is very much severe, then it is important to analyze the thermal-mechanical analysis of the disc brake with respect to over speeding and emergency braking. and also the life time of disc was carried out.

1.4 OBJECTIVE OF THE RESEARCH

1.4.1 GENERAL OBJECTIVE

The objective of this work is to analyse the thermo-mechanical behaviour of the disc brake and life time of disc brake under different speed and load during emergency braking condition using ANSYS software.

1.4.2 SPECIFIC OBJECTIVE

- Analyze the thermal and mechanical behavior of the disc brake numerically.
- 3DModelingof the discs brake using CATIA.
- Analyze the temperature distribution of the disc brake for both normal and over speed with emergency braking,
- Analyze the deformation simulation of the stress of the disc brake.
- Analyze the life time of the disc.
- Give conclusions and propose recommendations on the basis of the numerical and simulation results.

1.5. SIGNIFICANCE OF THE RESEARCH

This work shows the thermo-mechanical analysis in a disc brake of a rail vehicle during braking. As the brakes slow the vehicle, they transform its kinetic energy into thermal energy, resulting in intense heating of the brake disc. If the discs overheat, the brake pads stop working and, in a worst-case scenario, can melt. In order to protect this problem this thesis gives information on the thermo-mechanical behavioural approach and life time analysis of the disc brake during both normal and over speed with emergency braking. Andalso this research area provides a unique opportunity for future research contributions that will be done in ERC. It also helps for the train masters as reference material during training and operation.

1.6. SCOPEAND LIMITATION OF THE RESEARCH

1.6.1 SCOPE:-

The scope of this research will cover:-

- Modeling and analysis of the disc brake is presented.
- Various approaches that have been investigated in the literature to enhance the thermo-mechanical analysis and their issues are highlighted to substantiate this research.
- By employing mathematical-simulation models, the temperature distribution ,stress and deformations will be discussed.
- Finally concludes with a summary of the contributions. Following that, the direction of further studies required to continue this research is highlighted.

1.6.2 LIMITATION:-

Experimental analysis is accurate as far as instruments are fully available, sufficient knowledge of using instruments, and good conditions to apply instruments. But due to in sufficient time and material unavailability this thesis is not tested experimentally.

1.7. RESEARCH METHODOLOGY

To fulfill the objectives of the study the following methodology is used

I. Survey of books, journal articles, proceedings of international conferences, auto manufacturer catalogues, and other relevant literature is done.
II. Data Collection, data regarding the disc brake of locomotive vehicle is collected from journals and ERC.
III. Modeling: The finite element model development as well as the corresponding thermal and stress analysis is performed using ANSYS software.
IV. Conclusions, Recommendations and future work.

1.8 ORGANIZATION OF THE RESEARCH (THESIS STRUCTURE)

The research will be organized in six parts. The first chapter will be the introduction part which clearly states the background of the research, statement of problem, and objective, scope and limitation of the study, methodology, clearly states the vehicle braking types, component, and working principles.

Chapter two discusses Literature review of the disc brake system of rail vehicle.

Chapter three shows Thermal Analysis of disc brake Rotors with an Analytical Method.

Chapter four describes modeling and Analysis of disc brake using finite element analysis of ANSYS Workbench.

Chapter five discusses the major results of the research.

Chapter six gives conclusions, recommendations and future work.

CHAPTER TWO

LITERATURE REVIEW

2.1 INTRODUCTION

In recent times, disc brakes have increasingly become more popular and gradually found their ways in many different types of vehicles, ranging from light motorcycles to heavy trains. They offer several advantages over drum brakes, including better stopping performance (disc cooled readily), easier to control (not self-applying), less prone to brake fade, and quicker recovery from immersion, which largely contributed to their popularity. However, after long repetitive braking, brake fade could still set in and compromise the performance of a disc brake due to the change of friction characteristics caused by temperature rise and overheating of brake components. Overheated components could further lead to more problems such as thermal cracks, plastic deformation, and premature wear, and the life span of a disc brake could be shortened as a result. Due to the application of brake on the train disk brake rotor, heat generation takes place due to friction and this thermal flux has to be conducted and dispersed across the disk rotor cross section and pad. Due to this great input of heat flux into the disc in a very short time, high temperature difference is produced on the material. Because of high temperature difference the material is exposed to high stress-the consequence is thermal (heat) shock. Rapid aging and fatigues are the result. In addition to these thermal stresses, mechanical stress also generated on the disc due to rotor force.

So, the condition of braking is very much severe and thus the thermal and structural analysis has to be carried out.

2.2 RELATED LITERATURE REVIEWS

Taking the above in to consideration, many researchers have done brake disc thermo-mechanical coupling analysis, some of them are as follows.

A.R. Abu Bakar, A. Belhocine and M. Bouchetara [6, 7] studied Numerical Modeling of Disc Brake System in Frictional Contact to investigate and analyze the temperature distribution of rotor disc during braking operation using ANSYS Multiphysics. One particular existing brake disc middle class car design parameter was chosen for the investigation. The work uses the finite element analysis techniques to predict the temperature distribution on the full and ventilated brake disc bybuilding3D model and to identify the critical temperature of the rotor by holding account certain parameters such as; the material used, the geometric design of the disc and the mode of braking. The analysis also gives the heat flux distribution for the two discs. The result shows that the geometric design of the disc is an essential factor in the improvement of the cooling process of the discs. For the short braking time of t=2s the fully disc reaches a temperature of around 400 ^{O}C whereas the ventilated disc reaches around 335 ^{O}C.
It also shows the Influence of braking mode for both full disc and ventilated disc. The first one (mode 1) is driving cycle at successive (repeated) braking and the second one (mode 2) is at Cycle braking with phase of idles after each braking. During Mode1 the disc maximum temperature was1244.2 ^{O}C at t= 131.72 s whereas at mode2 maximum temperature was708.79 ^{O}C at t= 130.45 s which is a reduction of the temperature of approximately535 °C is 45.19 % compared with the first cycle. The braking mode with a cooling phase influences the heat transfers in the disc very positively, which involves a reduction in the maximum temperature of the interface, which causes cracking and mechanical wear. Finally, the results of the numerical simulation were not validated experimentally but recommended to validate experimentally on brake test benches.

Singh and Shergill [8] presented the heat generation and dissipation in a disc brake of an ordinary car during panic braking and the following release period by using computer aided engineering software for three different materials of rotor disc. The work investigates and analyzes the temperature distribution of rotor disc during operation using COMSOL MULTIPHYSICS. Modeling and Mesh was carried out using COMSOL software. The element used for meshing is of tetrahedral shape.

The work used the finite element analysis techniques to predict the temperature distribution on the brake disc and to identify the critical temperature of the brake rotor disc. The heat transfer mode was analyzed. It was concluded that the results obtained from the analysis shows that different material on the same retardation of the car during panic braking shows different temperature distribution. Thus, a comparison was made between three different materials (cast iron, aluminum alloy and ceramic Al2O3) used for brake disc and the best material (cast iron) for making brake disc based on the rate of heat dissipation have been suggested. The researchers did not validate the result with other researches or did not check the results with experiment.

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G. M. Nathi, T. Charyul, K.Gowtham, P.Reddy [9] studied the coupled structural/thermal analysis of car disc brake. The work used the dedicated finite element package of ANSYS package for determining the temperature distribution, variation of stresses and deformation across the disc brake profile. A transient thermal analysis was carried out to investigate the temperature variation across the disc using axisymmetric elements. Further structural analysis was also carried out by coupling thermal analysis**.** The time for thermal analysis was taken as 4, 5 and 6 seconds of braking. The thesis was concluded as follows.

- Transient thermal analysis was carried out using the direct time integration technique for the application of braking force due to friction for time duration of 4,5 and 6 seconds.
- The maximum temperature obtained in the disc is at the contact surface and is observed to be 240.161°C material of steel at 4 seconds.
- Static structural analysis is carried out by coupling the thermal solution to the structural analysis and the maximum Von Misses stress is observed to be 115MPa which is far below the working stress i.e. 375 M pa according to the manufacturer’s specifications.
- Comparing the different results obtained from the analysis, it is concluded that disc brake of material Cast Iron was the Best one with temperature distribution of 134.23, deformation of 0.120e-03 and a von misses stress of 115 M Pa.

The researcher's results were not validated ether experimentally or analytically but it was validated with manufacturer's specifications. Heat flux due to only kinetic energy with the mass of vehicle m=1400kg was used in the Analysis.

The research was done using constant speed of v=70 mph ,did not give sense whether it was normal or over speed and also did not specified whether it was service braking or emergency braking. The researchers also did not consider heat flux due to potential energy and did not make analysis using different vehicle speed.

P. Hwang and X. Wu [10] investigated the temperature and thermal stress in the ventilated disc-pad brake during single brake. The ventilated pad-disc brake assembly was built by a 3D model with a thermo-mechanical coupling boundary condition and multi-body model technique. The simulation results were verified using an experimental investigation. The analysis was carried out with vehicle weight of1900 kg under initial velocity of 100 kph and initial temperature was 40℃.The brake pressure 2.38MPa was applied on the pad to generate the brake force.

The maximum temperature of disc measured in the experiment was 204.56℃ whereas the maximum temperature of disc measured in the finite element simulation was 205.56℃. Therefore the simulation results were in good agreement with the experimental values.

J. Lan , J.Yan-li, Y. Liang, S. Nan, D. You-dong[11,12] were investigated the thermal and stress analyses of SiCn/Al brake disk during emergency braking at a speed of 300 km/h considering airflow cooling using finite element (FE) and computational fluid dynamics (CFD)methods. All three modes of heat transfer (conduction, convection and radiation) were analyzed along with the design features of the brake assembly and their interfaces.Formodeling the emergency braking phenomena, the brake disc was accelerated with the constant value of 0.8 $m/s^2$and reached the velocity of 300 km/h, then the braking started and caused constant deceleration with the rate of1.117 m/s^2 applying emergency braking. The initial temperature of the brake disk and the surrounding was 50 °C. The highest temperature after emergency braking was 461 °C and 359 °C without and with considering airflow cooling, respectively but the highest allowed temperature of the disk was 600 °C under the long-term condition. The equivalent stress could reach 269 MPa and 164 MPa without and with considering airflow cooling, respectively which are smaller than the permitted value of314 MPa.

The researchers did the experiment to validate the simulation result. It is observed that the temperature obtained by simulation reaches the maximum of 461 ºC at 107 s while the experimental data comes up to the maximum of 532 °Cat 93 s without considering airflow cool. However, the simulated temperature reaches the maximum of 359 °Cat 105 s, while the experimental data reach up to 392 °Cat 98 s considering airflow cooling.

M.Reibenschuh,G.Oder,F.cus,I.Potrc[13] have calculated the thermal and stress analysis of disc brakes under specific loads (driving downhill and braking to a standstill). The FEM (Finite Element Method) was used to carry out the analysis. The analysis was dealt with centrifugal load for two cases of braking, braking to a standstill on a flat surface and braking downhill, maintaining constant speed and afterwards braking to a standstill. The main boundary condition in both cases was the entered heat flux on the braking surface of the disc and the force of the brake clamps. Two different discs were used, one brand new (unused) and one with permitted wearing. The temperature of the surrounding area was constant at 50 °C and the Mass of the vehicle was 70 000kg. A physical model, considering the heat flux in dependence of the braking time, was used to determine the brake influence. In case of braking on the horizontal track, the surface temperatures of the new disc after 30s are 174 °C. For the worn disc with equal load after 38s the highest temperatures amounted to 211 °C.

In case of braking downhill the highest temperature on the new disc after expiration of 104 s was equal to 154 °C, namely on the disc - brake pad contact surface. During further braking to a standstill on the straight track the brake disc warmed up to the highest temperature of 251 °C within 132 s after the start of braking. For the worn disc the highest temperatures after 104 s of maintaining constant speed during braking downhill amounted to 166 °C. By further stopping on flat surface the disc temperature increased to the maximum temperature of 298 °C within 32 s after the start of stopping on a flat surface. The highest allowable temperatures in the brake pad and, consequently, in the disc were 350 °C.

In case of braking on flat surface, by considering the centrifugal loads in the new disc, the highest stresses amounted to 185MPa and the highest stresses for worn disc were equal to 174 MPa.In case of braking downhill and later stopping on a flat surface the highest stresses amounted to 201 MPa for the new disc and stresses were equal to 198 MPa. The result was not validated experimentally but it was validated with manufacturer's specification. The stresses themselves are high, but in comparison with permissible stresses amounting to 210 MPa by considering the 1.5 safety factor the analyzed disc is adequately dimensioned.

J.Chen and F.Gao[14] have studied the distribution of temperatures gradient and thermal stress of brake disc by FEM code. The FEM models build upon LS-DYNA® thermo-mechanical code and contact algorithm. Parametric analysis for disc brakes was carried out by comparison of grouped brake applications conform to UIC code. The parameters considered to be normal forces, initial angular speed under the given moment of inertia. Nine applications were chosen for comparison.

All initial temperature was 0°C (273 K) and moment of inertia Izz is 800 which have an equivalent mass of 4000 kg.

It concludes that the main factor causes the high temperature gradient and thermal stress of brake disc was brake force and its initial speed. Normal force increase disc temperature and higher initial kinetic energy will cause higher temperature, which means more energy need to be transformed. The sample 3C maximum stress can reach 530 MPa with maximum temperature of 484 ^{0}C.The higher train speed will transform more kinetic energy to heat and cause higher thermal stresses, the series 3 samples results extreme thermal stresses which will cause fatigue quickly and series 2 samples also have high stresses compare to series 1 samples at low train speed, so life time of disc is mainly affected by the train speed. The researcher did not validate the result experimentally.

Summary of the reviewed literatures:-

A literature reviews conducted above to investigate the past research that has been done in many areas related to this work are reviewed. In addition, description, methods, results of the literatures are discussed. The limitations and strengths of the research (validation of the results) were analyzed. Most of the literatures listed in this thesis focus only the thermo-mechanical analysis of disc brake due to heat flux generated on the disk either from kinetic energy of the vehicle, moment of inertia, or centrifugal load of the vehicle. In this thesis the thermo-mechanical analysis of disc brake due to both kinetic energy and potential energy during emergency braking for both normal and over speed will be analyzed.

Although the objectives of the above-mentioned study differ from the objective of this project, similarities can be drawn in the simulation set-up. For instance, the proportion of heat entering the disc as well as the convective heat transfer coefficient can be used as a reference for this thesis.

The focus of this paper differs from the objectives of the above research literatures is, in this paper, the main focus is on the thermo-mechanical analysis, in which the transient-state temperature profile will be used to predict the thermal distortion in a brake rotor. Although the objectives differ, the idea of using a PC based program to calculate heat inputs is novel and can be used in this thesis to calculate the thermal parameters as well.

CHAPTER THREE

THERMO-MECHANICAL ANALYSIS AND IN PUT DATA COLLECTIONS

3.1. Thermo-mechanical Analysis of disc brake Rotors with an Analytical Method.

3.1.1Thermal Transient Analysis of the Disc Brake

For an isotropic material with internal heat generation, the governing equations for 3D heat conduction equation under transient conditions in Cartesian and Cylindrical coordinate systems are as follows:-

$$k_x \frac{d^2T}{dX^2} + k_y \frac{d^2T}{dy^2} + k_z \frac{d^2T}{dz^2} + Q = \rho C_p \frac{dT}{dt} \quad \ldots\ldots 3.1$$

Wherek_x, k_y and k_z are thermal conductivity in x, y and z-directions, respectively, Cp isthe specific heat, ρ is the specific mass, Q is internal heat generation rate per unit volumeand T is the temperature that varies with the coordinates as well as the time t.

The heat conduction in any direction within the material is governed by Fourier's law of heat conduction:-

$$\mathrm{q} = -\mathrm{kA} \frac{dT}{dx} \quad \ldots\ldots 3.2$$

Transient thermal analysis requires material properties which include thermal conductivity, specific heat and density. An isotropic material has the same properties in every direction. The boundary conditions are specified by the heat flux entering the model and dissipated from the model by convection or radiation. The initial conditions are specified by the initial nodal temperature value. The heat transfer rate of convection for the disc brake dominates the cooling process specified by Newton's law of cooling:

$$q_s = -kh(T_s - T_\infty) \text{Where } T_s = T_1(x, y, z) \quad \ldots\ldots 3.3$$

Where$h$ is the convective surface heat transfer coefficient (also known as the film Conductance, T_1 is the specified surface temperature; q_s the specified surface heatflux (positive into a surface); T_s the unknown surface temperature, and T_∞ the convective exchange temperature.

Heat transfer through radiation is calculated from the Stefan-Boltzmann law as:-

$q = \varepsilon\sigma(T_s^4 - T_\infty^4)$……………………………………………………………………………..3.4

Where ε is the material's emissivity,σis Stefan-Boltzmann constant,

T_∞ is the temperature of the surrounding air.

Heat transfer by radiation can be neglected as it amounts only to 5 % to 10 %. So in this thesis heat transfer through radiation is not considered.

3.1.2. HEAT FLUX ENTERING IN TO THE DISC

A time-varying heat flux q(t)is applied to the model to simulate the action of the brake pad on the disc. The procedure to determine the time varying heat flux is as follows: -

Braking energy,

$E_b = \frac{1}{2}MV_o^2 + \frac{1}{2}I\omega^2$……………………………………………………………….3.5

Where$\frac{1}{2}MV_o^2$ is energy due to kinetic energy of the vehicle and $\frac{1}{2}I\omega^2$is energy due rotational kinetic energy of rotating parts.

Using$V_o = R\omega$, then

$E_b = \frac{M}{2}\ \ 1 + \frac{I}{R^2M}\ \ V_o^2 = (\frac{kMV_o^2}{2})$……………………………………………………………...3.6

In this thesis, k (correction factor for rotating parts) is estimated to be about 1.1.

$E_b = (\frac{kMV_o^2}{2})$………………………………………………………………………………….3.7

Braking power,

$P_b = \frac{E_b}{t_b}$……………………………………………………………………………………….3.8

$P_b = \frac{kMV_o^2}{2t_b}$……………………………………………………………………………………3.9

Heat flux, Q(t) is obtained by dividing the braking power by the swept area of the brake rotor. i.e.

$$Q\ t\ =\frac{kMV_o^2}{2A_d t_b}\ldots\ldots 3.10$$

This heat flux is the averaged heat flux generated during the braking cycle.

Furthermore, not all of the heat enters the brake disc.The proportion of heat entering the brake disc is λ. In this thesis, λ is estimated to be about 0.83 [15]. Therefore, the heat flux function should be modified to be:

$$Q\ t\ =\lambda\frac{kMV_o^2}{2A_d t_b}\ldots\ldots 3.11$$

Equation (10) is for moving a vehicle on a flat track. But heat flux for vehicle moving on inclined track is

$$Q\ t\ =\lambda[\frac{kMV_o^2}{2A_d t_b}\pm\frac{s_b Mg\sin\alpha}{At_b}]\ldots\ldots 3.12$$

Where s_b is the braking distance of the vehicle which is$s_b=\frac{{V_o}^2}{2a}$.

With vehicle deceleration of a.Substituting this Eq. into Eq. 3.12 and rearranging the Equation and will the following Equation:

$$Q\ t\ =\lambda[\frac{kMV_o^2}{2A_d t_b}\pm\frac{{V_o}^2 Mg\sin\alpha}{2aAt_b}]$$

$$Q\ t\ =\frac{\lambda {V_o}^2 M}{2A_d t_b}[k\pm\frac{g\sin\alpha}{a}]\ldots\ldots 3.13$$

3.1.3 ANALYSIS OF DISC ROTOR FORCE

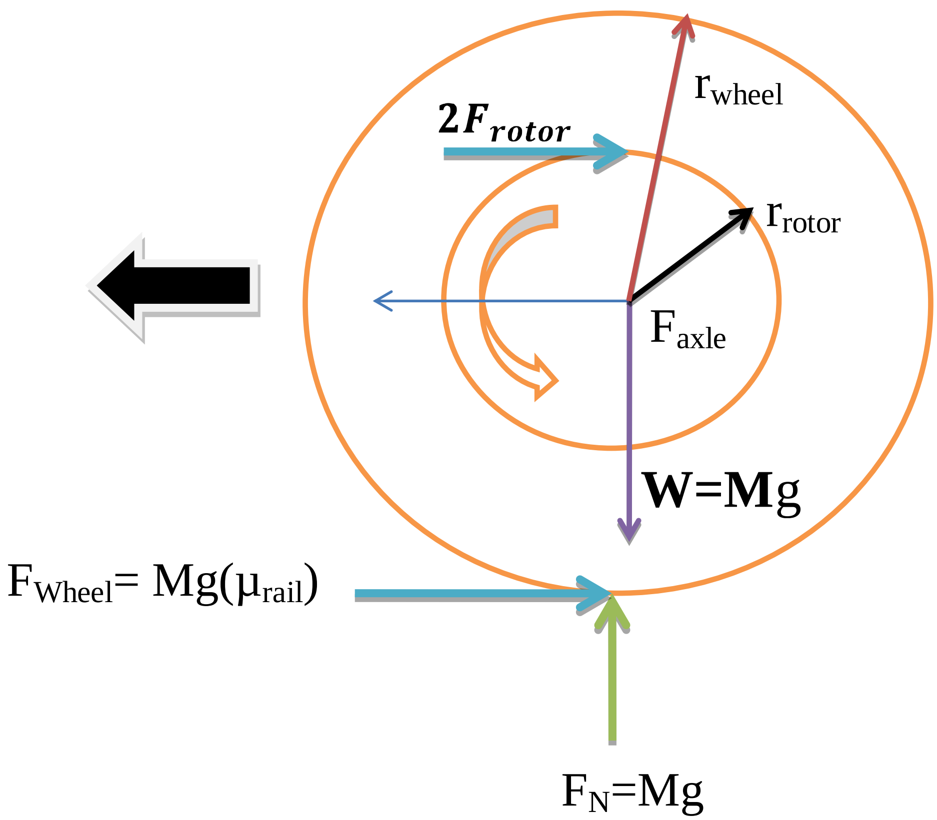


Fig.3.1 free body diagram of a front wheel-rotor system

A free body diagram of a front wheel-rotor system, fig 3.1, is used to drive the equation of equilibrium. Since large amount of the braking load is born by the front bakes, that amount of kinetic energy and potential energy in to a single disc is given by

$$E_{\text{dissipated}} = \frac{1}{2}k\text{MV}_o^2 + S_b Mg\sin\alpha \qquad \text{3.14}$$

But $S_b = \frac{V_o{}^2}{2a}$

$$E_{\text{dissipated}} = \frac{1}{2}k\text{MV}_o^2 + \frac{\text{V}_o^2}{2a}Mg\sin\alpha$$

The power dissipated by each rotor face is equal to the heat flux in to the rotor face.

$$E_{dissipated} = \int P_{dissipated}(t)\,dt = \int 2F_{rotor}\,v_{rotor}(t)dt$$

$$\frac{1}{2}kMV_o^2 + \frac{V_o^2}{2a}Mg\sin\alpha = KE_{dissipated} = \int P_{dissipated}(t)\,dt = \int 2F_{rotor}\,v_{rotor}(t)dt \quad \ldots 3.15$$

But from kinematic relationships.

$$V_{vehicle}(t) = V_o - at\ ,$$

$$a = \frac{V_o}{t_{stop}},$$

$$\frac{V_{vehicle}(t)}{r_{wheel}} = \omega(t) = \frac{v_{rotor}}{r_{rotor}}$$

$$v_{rotor} = \frac{r_{rotor}}{r_{wheel}}(V_o - \{\frac{V_o}{t_{stop}}\})$$

F_{rotor}Is constant with respect to time, and v_{rotor} varies only linearly with time so the energy balance equation becomes:

$$\frac{1}{2}kMV_o^2 + \frac{V_o^2}{2a}Mg\sin\alpha = 2F_{rotor}\int_0^{t_{stop}} v_{rotor}(t)dt = 2F_{rotor}\frac{r_{rotor}}{r_{wheel}}\left(V_O t_{stop} - \frac{1}{2}\left(\frac{V_O}{t_{stop}}\right)t_{stop}^2\right) \quad \ldots 3.16$$

$$F_{rotor} = \frac{\frac{1}{2}kMV_O^2 + \frac{V_O^2}{2a}Mg\sin\alpha}{2.\frac{r_{rotor}}{r_{wheel}}.(V_O t_{stop} - \frac{1}{2}\{\frac{V_O}{t_{stop}}\}t_{stop}^2)} \quad \ldots 3.17$$

3.2 DATA COLLECTION AND ANALYSIS

3.2.1 Addis Ababa Light Rail Transit

Addis Ababa Light Rail Transit (AA-LRT) Project consists of E-W & N-S lines, with the total length of main line 31.025km. These two lines operate on the same rail in downtown area for a total length of 2.662km.The ground line mode is mostly applied in rail laying, while elevated line and underground line are also applied in some sections.

The entire line has 39 stations, consisting of 9 elevated stations (including 5 in the section that uses the same rail, 1 on E-W line, and 3 on S-N line), 2 underground stations, 1 semi-underground, and Ground stations are designed on ground line section, including 27 ground stations along the line (13 ground stations on N-S line, and 14 ground stations on E-W line).

The east-west line phase I project starts from Ayat and ends at Torhailoch. Thetotal length is 17.4km. There are 22 stations, among which 5 are elevated stations, 1 underground station and 16 ground stations. The depot locates at the west ends of the project.

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

For ensuring the traction power supply on the main line as well as the power supply for the station equipment, 18substations are set up along the line, including 17 traction and decompression substations, and 1 decompression substation.

Table1.Slope/Gradient of AA-LRT Station (E-W Route) [16]

No	Station name	Station symbol	Slope/gradient	Station types
1	Ayat 2	EW1	0.0333[1.905°]	Groundstation[- down]
2	Ayat 1	EW2	0.00143[0.0818°]	Ground station[level]
3	Meri/CMC 2	EW3	0.0016[0.0917°]	Ground station[level]
4	CMC 1	EW4	0.0483[2.767°]	Ground station[+ up]
5	St.Michael Church	EW5	0.0185[1.058°]	Ground station
6	Civil Service College	EW6	0.0178[1.019°]	Ground station
7	SahliteMhiret Church	EW7	0.0138[0.79°]	Ground station
8	GurdSholla	EW8	0.0022[0.127°]	Ground station
9	Megenagna/Adwa square	EW9	0.018[1.031°]	Underground station
10	Lem Hotel	EW10	0.0063[0.358°]	Ground station
11	Mazoria/Traffic Police HQ.	EW11	0.0086[0.491°]	Ground station
12	Chemical Corporation	EW12	0.0253[1.247°]	Elevated station
13	Urael Church	EW13	0.0243[1.394°]	Ground Station
14	Yordanose Hotel	EW14	0.0148[0.849°]	Ground Station
15	Estifanose/Meskel Square2	EW15	0.0027[0.155°]	Ground Station
16	Stadium/Meskel Square 1	EW16	0.0016[0.093°]	Elevated station
17	La Gare	EW17	0.008[0.458°]	Elevated station
18	Road Authority	EW18	0.0018[0.102°]	Ground station
19	Mexico Square	EW19	0.0278[1.591°]	Elevated station
20	Lideta	EW20	0.0392[1.247°]	Elevated station
21	Coca Cola	EW21	0.0227[1.302°]	Ground station
22	Tor Hailoch	EW22	0.0088[0.504°]	Ground station

3.2.2 VEHICLE BRAKING ON A RAIL TRACK

3.2.2.1 BRAKING ON A STRAIGHT TRACK

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

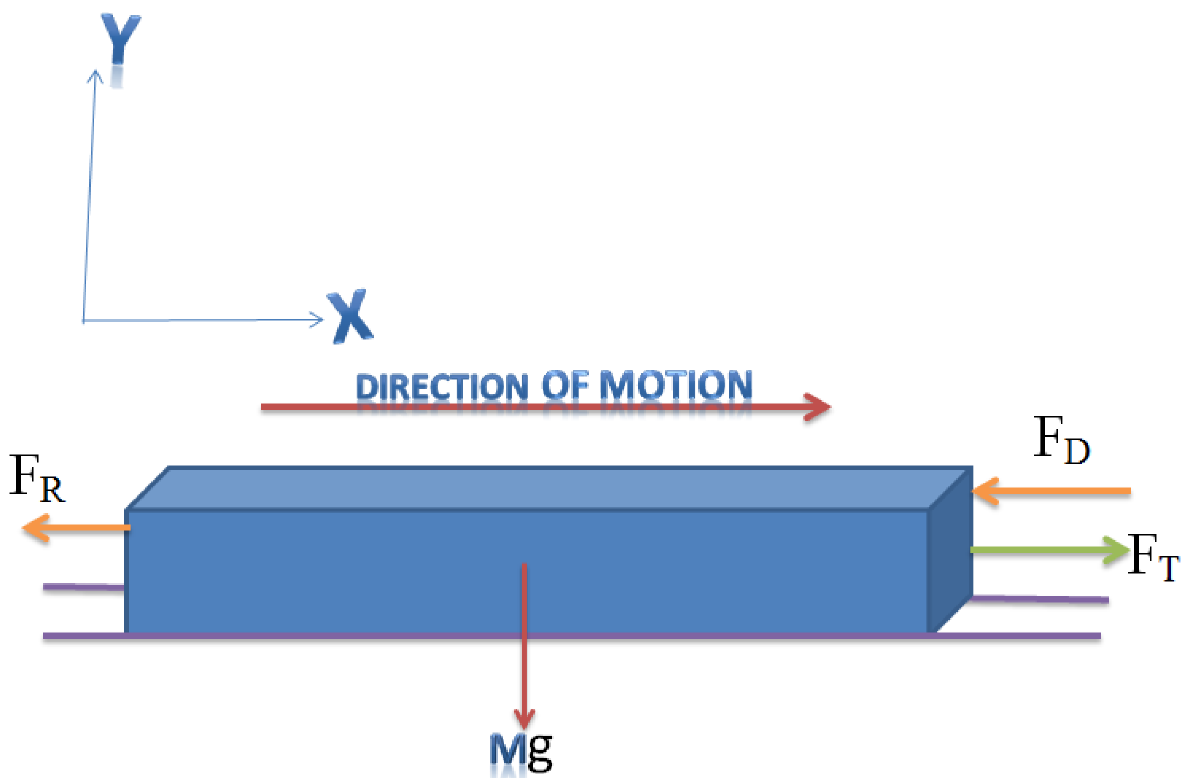


Fig 3.2 braking on a straight/flat track

Using Newton's second law of motion the equation of motion when a vehicle moving along a straight track is

$Ma = F_T - F_D - F_R$……………………………………………………………………….3.18

Where M=Total mass of the vehicle including passenger weight

F_T =Tractive force

F_D =Drag force

F_R = Rolling resistance force

The heat flux generated when the vehicle moving on straight track during the braking cycle is:

$Q\ t_b = \lambda[\frac{kMV_o^2}{2At_b}]$………………………………………………………………………….3.19

3.2.2.2 BRAKING ON INCLINED TRACK

The alignment of the rail way track is not only straight but also has a different gradient and curves. Now the train has both kinetic energy and potential energy due to its motion and its inclination /gradient which intern increases the heat energy dissipation from the contact surface to the surrounding. The gradient increases and decreases the dissipation of heat energy depending on the direction of the movement. When the vehicle moving along the downhill the gradient increases the heat energy while when it moves along the uphill the gradient decreases the heat energy. The total energy will be the summation of all the three energies.

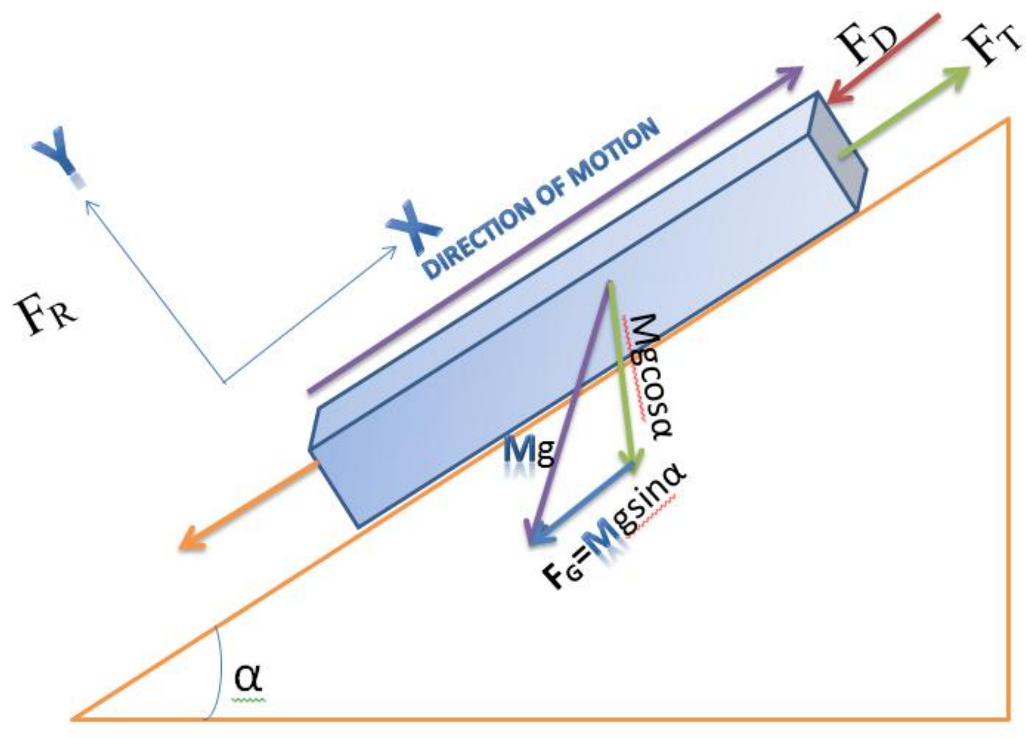


Fig 3.3Vehicle moving along the uphill (moving upward)

Using Newton's second law of motion the equation of motion when a vehicle moving in the upward inclined plane track is

$\mathrm{Ma_T} = \mathrm{F_T} - \mathrm{F_D} - \mathrm{F_G} - \mathrm{F_R}$……………………………………………………………….3.20

Where M=Total mass of the vehicle including passenger weight

$\mathrm{F_T}$ =Tractive force

$\mathrm{F_D}$ =Drag force

$\mathrm{F_G}$ = Total weight component in the direction of the inclined plane.

$\mathrm{F_R}$ = Rolling resistance force

$\mathrm{a_T}$ = train acceleration

The heat flux generated when the vehicle moving on uphill track during the braking cycle is:

$\mathrm{Q\ \ t} = \frac{\lambda {V_o}^2 M}{2At_b}[k - \frac{g\sin\alpha}{a}]$…………………………………………………………………3.21

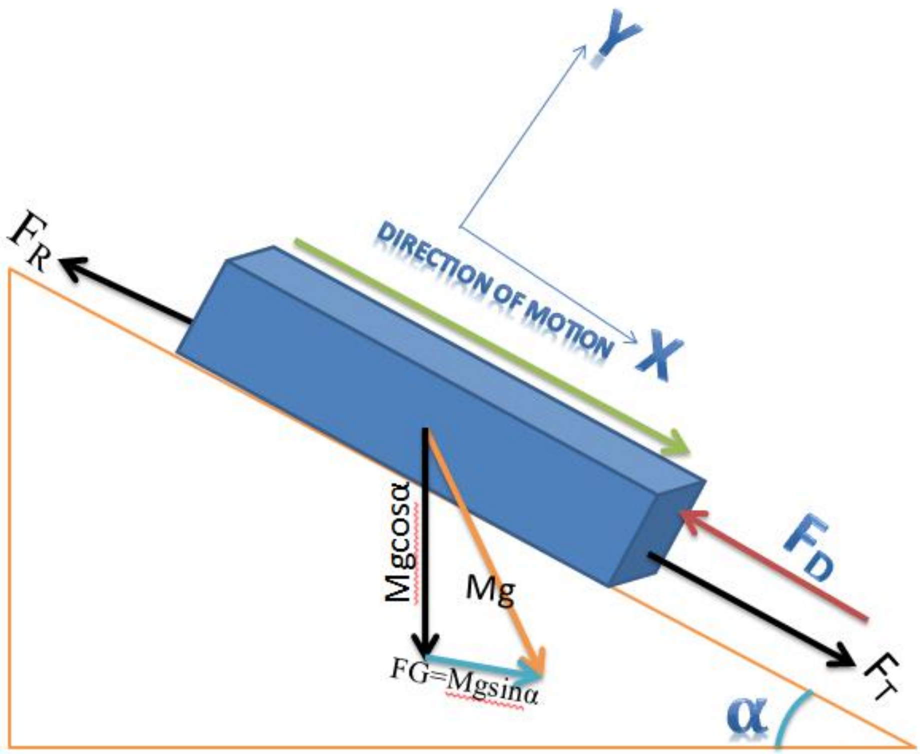


Fig.3.4Vehicle moving along the downhill (moving downward)

Using Newton's second law of motion the equation of motion when a vehicle moving in the upward inclined plane track is

$Ma = F_T + F_G - F_D - F_R$……………………………………………………………………3.22

Where M=Total mass of the vehicle including passenger weight

F_T =Tractive force

F_D =Drag force

F_G = Total weight component in the direction of the inclined plane.

F_R = Rolling resistance force

$a_T = train\ acceleration$

The heat flux generated when the vehicle moving on downhill track during the braking cycle is:

$Q\ \ t\ = \frac{\lambda {V_o}^2 M}{2A_d t_b}[k + \frac{g \sin \alpha}{a}]$……………………………………………………………………3.23

From equations 3.19, 3.21, 3.23, the heat flux entering the disc rotor when the vehicle moving up the gradient is less that of moving on a straight track and down the gradient. Maximum heat flux is occurred when the vehicle moving in the downhill direction of the track having maximum gradient of 1.905° which is Ayat2 (EW1) station.

So the thermo-mechanical analysis of the disc brake under three different speed and load during emergency braking will be done for this station.

3.2.3 SEATING AND LOADING CAPACITY OF THE TRAIN VEHICLE

Table 2: Seating capacity of the train [17]

Number of passengers (persons)	Seated	Standing	Total
Seats (AW_1)	65	**0**	65
Seating capacity (AW_2) (standing: 6 persons/m^2)	65	189	254
Overload capacity (AW_3)(standing: 8 persons/m^2)	65	252	317

Table 3: Weights of the train [17]

Loads	Car body weight	Passenger weight	Total weight
Empty vehicle (t)	44	0	44
Seating capacity (t)	44	15.24	59.24
Overload capacity (t)	44	19.02	63.02

Note: Take 60kg as average weight of each passenger.

3.3 CALCULATION OF DESIGN SPEED OF THE TRAIN AT AYAT2 (EW2) STATION

General Description of Vehicle Movement

Figure 3.4shows the forces acting on a vehicle moving down a grade. The tractive effort,F_T,in the contact area between wheel of the driven wheels and the rail surface propels the vehicle forward. It is produced by the power plant torque and is transferred through transmission and final drive to the drive wheels. While the vehicle is moving, there is resistance that tries to stop its movement. The resistance usually includes wheel rolling resistance, aerodynamic drag, and uphill resistance. According to Newton's second law, vehicle acceleration can be written as:

$$Ma = F_T + F_G - F_R - F_D \quad \ldots 3.24$$

Where M = mass of the vehicle

F_T =Tractive force

F_D =Drag force

F_G = Total weight component in the direction of the inclined plane.

F_R = Rolling resistance force

a_T = train acceleration

Calculation of tractive force (F_T)

Electric motor power is transmitted to the wheels through power transmission unit, driveshaft and axle in a vehicle. The force required to overcome the total running resistance at a given driving condition on wheels is called the traction force of a vehicle. The size of traction force on a wheel is proportional to the torque generated at the train motor. The motor torque is:

$$T_m = \frac{P}{\omega} \quad \ldots 3.25$$

WhereT_m = motor torque,

P = power,

ω = angular velocity

Input data for AALRT

$P = 130\text{kW}$ And $\omega = 1800$ rpm

$T_m == 690.02\text{Nm}$

The torque on the driven wheels, transmitted from the power plant (electric motor), is expressed as

$T_W = T_m * \eta * \tau$……………………………………………………………………………3.26

Where T_W = torque on the wheel = 690.02Nm

η = mechanical efficiency of transmission = 0.98[18]

τ = gear ratio of transmission = 7.43 [19]

$T_W = 5024.312\text{Nm}$

The tractive force of the power train is calculated by

$F_T = \frac{T_W}{R_W}$……………………………………………………………………………………3.27

Where F_T = Tractive force

T_W = torque on the wheel

R_W = wheel radius,

$R_W = \frac{660}{2}\text{mm} = 330\text{mm}$=0.33m

F_T =15225.188N

Since there are four traction motors, therefore the total traction force will become

$F_T = 4 * 15225.188\text{N} = 60900.752\text{N}$

Calculation of Gradient resistance force (F_G)

When a vehicle goes up or down a slope, its weight produces a component, which is always directed to the downward direction, as shown in Figure 3.4

This component either opposes the forward motion (grade climbing) or helps the forward motion (grade descending). It experiences gravitational resistance due to its weight and it is called gradient resistance of the vehicle. The grading resistance, from Figure 3.4 Can be expressed as:

$F_G = Mg \sin\alpha$……………………………………………………………………………3.28

Where α is gradient angle of the rail track

$\alpha = 1.905^0$

Substitute this value in to Eq.3.30 and will get the F_G result for the over load vehicle as follows.

$F_G = 20{,}551.321\text{N}$

Calculation of Rolling resistance force(F_R)

Rolling Resistance is the force necessary to propel a vehicle over a particular surface. The friction resistance between rail and wheel surface is defined as rolling resistance of a vehicle. It is clearly affected to the surface roughness of rail but not to the vehicle speed. From the principle of physics, the rolling resistance of a running vehicle can be obtained as follows:

$$F_R = \mu_R * Mg\cos\alpha \quad \ldots 3.29$$

Where F_R = Rolling resistance force

μ_R = rolling resistance coefficient, $\mu_R = 0.002$ for passanger rail car

M = vehicle mass = 63,020 Kg for overload capacity

g = acceleration due to gravity = $9.81\ \frac{m}{s^2}$

α = gradient of the rail = 1.905^0

Substitute these values in to Eq.3. 29 And will get the F_R result for the over load vehicle as follows.

$F_R = 1{,}235.769N$

Calculation of drag force or Aerodynamic resistance force (F_D)

As a vehicle runs on a rail track, the relative air movement occurs opposite to the driving direction of the vehicle even with no wind in air. Because of this air flow, the vehicle experiences aerodynamic force such as drag and lift on the body. The

Aerodynamic drag force generated on the frontal and rear side of the body acts on the vehicle as a driving resistance. From the analytical equation in Eq. (3.30), the aerodynamic drag force can be estimated as follows:

$$F_D = \frac{1}{2} C_D * A_V * \rho * {V_D}^2 \quad \ldots 3.30$$

Where C_D = drag coefficient, $C_D = 1.8$ [20]

A_V = projected frontal area of the vehicle

ρ = Density of air ($\frac{Kg}{m^3}$)

V_D = Design velocity of the train

Calculation of density of air

The density of dry air can be calculated using the ideal gas law, expressed as a function of temperature and pressure.

$$\rho = \frac{p_{ab}}{R_{specific} * T} \qquad \text{3.31}$$

Where p_{ab} = absolute pressure(Pa)=100000

R_{specific} = Specific gas constant for dry air $(\frac{\text{J}}{\text{kg} * \text{K}}) = 287.058$

T = temperature K

Average daily highest temperature in years of Addis Ababa city:

25.5℃=25.5+273.15=298.65K [17]

$T = 298.65$ K

Therefore the density of air for Addis Ababa will be:

$$\rho = \frac{p_{ab}}{R_{specific} * T} = 1.17 \, {}^{Kg}\!/_{m^3}$$

The projected frontal area (A) of a vehicle is normally obtained by

$$A_V = vehicle\ width * vehicle\ height$$

The maximum width of the car body =2650mm and vehicle height=3700mm [17].But since it is not full rectangle, the actual dimension will be

$$A_V = 2650\text{mm} * 1000 + 2115 * 2000 + (700 * 2650)\text{mm} = 8735000 mm^2 = 8.735 m^2$$

Substitute these values in to Eq.3.32and will get

$$F_D = 0.5 * 1.8 * 1.17 * 8.735 * {V_D}^2$$

$$F_D = 9.2V^2$$

Substitute the above values in to Eqn.3. 26 at acceleration $a_T = 1.2 \, {}^{m}\!/_{s^2}$ will get the result as follows:

$$63{,}020\ Kg * 1.2 \, {}^{m}\!/_{s^2} = 60900.752N + 20{,}551.321N - 1{,}235.769N - 9.2V^2$$

$$\text{V}_\text{D} = 22.342 \, {}^{\text{m}}\!/_{\text{s}} = 80.431 \, {}^{\text{km}}\!/_{\text{h}}$$

Table 4.Geometrical Dimensions of disc and pad [actual measured dimension]

Item	Disc	Pad
Inner radius, mm	110	110
Outer radus, mm	180	180
Thickness, mm	60	20
Effective rotor radius,r_{rotor},mm	145	
Surface disc swept Area by the pad A_d,mm^2	127484	
Pad area,A_p,mm^2	12247.33	
coefficient of friction, μ	0.365	

Tabe5. Vehicle properties [17]

Item		Quantity
Mass of the vehicle(M),kg	Empty load(vehicle mass)	44000
	Normal load	59240
	Over load	63020
Vehicle speed (V_o),$\frac{km}{h}$	Normal speed	40
	Maximum speed	70
	Over speed	80.3
Deceleration(a),$\frac{m}{s^2}$		2
Radius of the wheel, mm		330

3.4 CALCULATION OF HEAT FLUX GENERATION OF AYAT2 (EW2) STATION

Figure 3.4 shows the forces acting on a vehicle moving down a grade. The heat flux generated when the vehicle moving on AYAT2 during the braking cycle is:

$Q\ t\ = \frac{\lambda V_o{}^2 M}{2A_d t_b}[k + \frac{g\sin\alpha}{a}]$……………………………………………………………………3.32

CALCULATION OF HEAT FLUX AT EMPTY LOAD WITH DIFFERENT SPEED

$Where\ M = Total\ mass\ of\ vehicle\ = 4400kg$

$V_o = initial\ speed\ of\ the\ vehicle$

$a = deceleration\ of\ the\ vehicle = 2\,{}^{m}\!/_{s^2}$

$t_b = braking\ timerequired\ to\ stop\ the\ vehicle@Emergency\ braking$

$t_b = \frac{V_o}{a}$…………………………………………………………………………………3.33

$A_d = \text{contact swept area} = 2\pi\ r_{op}{}^2 - r_{ip}{}^2$

$\text{where } r_{op} = \text{Outer pad radius} = 180\text{mm and } r_{ip} = \text{inner pad radius} = 110\text{mm}$

$A_d = 2\pi\ r_{op}{}^2 - r_{ip}{}^2\ = 2\pi\ 180^2 - 110^2\ = 127548mm^2 = 0.128\text{m}^2$

$k = \text{correction factor for rotating parts} = 1.1$

$\lambda = \text{proportion of heat entering the brake disc} = 0.83$

$g = \text{acceleration due to gravity} = 9.81 = {}^{\text{m}}\!/_{\text{s}^2}$

The heat flux generated when the vehicle moving at normal speed of $V_o = 40\,{}^{\text{km}}\!/_{h} =$

$\frac{40*1000m}{60*60\ s} = 11.11\,{}^{m}\!/_{s}\quad @\ t_b = \frac{V_o}{a} = \frac{11.111}{2} = 5.556\ sec.$ [17]

Substitute all the above variables in to Eq.3.32and will give the following result:

$$Q\ t\ = \frac{\lambda V_o{}^2 M}{2A_d t_b}\ k + \frac{g\sin\alpha}{a}$$

$$Q\ t\ = 4004431.032\ {}^{W}\!/_{m^2}$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore,

$$Q\ t\ = 500553.879\,W/m^2$$

But each disc brake consists of two pads; one pad on each surface of the rotor disc face .Then for the heat flux for one side of the disc is calculated as follows:

$$Q\ t\ = 250276.94\,W/m^2$$

This heat flux is for one side the disc brake and is used for importing in to ANSYS workbench for each surface of disc,

The heat flux generated when the vehicle moving at maximum speed of

$V_o = 70\ \mathrm{km}/h = \frac{70*1000m}{60*60\ s} = 19.444\ m/s\ \ @t_b = \frac{V_o}{a} = \frac{19.444}{2} = 9.722\ sec.$

Substitute all the above variables in to Eq.3.32and will give the following result:

$$Q\ t\ = \frac{\lambda V_o{}^2 M}{2A_d t_b}\ k + \frac{g\sin\alpha}{a}$$

$$Q\ t\ = 7008982.38\,W/m^2$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore $Q\ t\ = 876122.798\,W/m^2$

But each disc brake consists of two pads; one pad on each surface of the rotor disc face. Then for the heat flux for one side of the disc is calculated as follows:

$$Q\ t\ = 438061.399\,W/m^2$$

The heat flux generated when the vehicle moving at over speed of

$V_o = 80.431\ \mathrm{km}/h = \frac{80.431*1000m}{60*60\ s} = 22.342\ m/s\ @\ t_b = \frac{V_o}{a} = \frac{22.342}{2} = 11.171\ sec.$

Substitute all the above variables in to Eq.3.32and will give the following result:

$$Q\ t\ = \frac{\lambda V_o{}^2 M}{2A_d t_b}\ k + \frac{g\sin\alpha}{a}$$

$$Q\ t\ = 8050302.156\,W/m^2$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore

$$Q\ t\ = 1006287.77\,{}^{W}\!/_{m^2}$$

But each disc brake consists of two pads, one pad on each surface of the rotor disc face. Then for the heat flux for one side of the disc is calculated as follows:

$$Q\ t\ = \mathbf{503143.885}\,{}^{W}\!/_{m^2}$$

CALCULATION OF HEAT FLUX AT NORMAL LOAD WITH DIFFERENT SPEED

$Where\ M = Total\ mass\ of\ vehicle\ including\ passanger\ mass = 59240kg$

$a = 2\,{}^{m}\!/_{s^2}$ [17]

$A_d = 0.128\,m^2$

$k = 1.1$

$\lambda = 0.83$

$g = 9.81\,{}^{m}\!/_{s^2}$

The heat flux generated when the vehicle moving at normal speed of

$V_o = 40\,{}^{km}\!/_{h} = \frac{40*1000m}{60*60\ s} = 11.11\,{}^{m}\!/_{s}$ @ $t_b = \frac{V_o}{a} = \frac{11.111}{2} = 5.556\ sec.$

Substitute all the above variables in to Eq.3.32And will give the following result:

$$Q\ t\ = \frac{\lambda {V_o}^2 M}{2 A_d t_b}\ k + \frac{g \sin\alpha}{2a}$$

$$Q\ t\ = 5391420.325\,{}^{W}\!/_{m^2}$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore,

$$Q(t) = 673927.541\ W/m^2$$

But each disc brake consists of two pads; one pad on each surface of the rotor disc face .then for the heat flux for one side of the disc is calculated as follows:

$$Q(t) = \mathbf{336963.771}\ W/m^2$$

The heat flux generated when the vehicle moving at maximum speed of

$V_o = 70\ \mathrm{km}/h = \frac{70*1000m}{60*60\ s} = 19.444\ m/s\ \ @t_b = \frac{V_o}{a} = \frac{19.444}{2} = 9.722\ sec.$

Substitute all the above variables in to Eq.3.32and will give the following result:

$$Q(t) = \frac{\lambda {V_o}^2 M}{2A_d t_b}\left(k + \frac{g \sin\alpha}{a}\right)$$

$$Q(t) = 9436639.005\ W/m^2$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore

$$Q(t) = 1179579.876\ W/m^2$$

But each disc brake consists of two pads; one pad on each surface of the rotor disc face .then for the heat flux for one side of the disc is calculated as follows:

$$Q(t) = \mathbf{589789.938}\ W/m^2$$

The heat flux generated when the vehicle moving at over speed of

$$V_o = 80.431\ \mathrm{km}/h = \frac{80.431 * 1000m}{60 * 60\ s} = 22.342\ m/s\ \ t_b = \frac{V_o}{a} = \frac{22.342}{2} = 11.171\ sec.$$

Substitute all the above variables in to Eq.3.32 and will give the following result:

$$Q(t) = \frac{\lambda {V_o}^2 M}{2A_d t_b}\left(k + \frac{g \sin\alpha}{2a}\right)$$

$$Q(t) = 10838634.084\ W/m^2$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore

$$Q\ t\ = 1354829.261 \, W/m^2$$

But each disc brake consists of two pads, one pad on each surface of the rotor disc face. Then for the heat flux for one side of the disc is calculated as follows:

$$\mathbf{Q\ t\ = 677414.631 \, \boldsymbol{W/m^2}}$$

CALCULATION OF HEAT FLUX AT OVER LOAD WITH DIFFERENT SPEED

Where $M =$ Total mass of vehicle including passanger mass $= 63020\text{kg}$

$a = 2 \, \mathrm{m}/s^2$

$A_d = 0.128 \, m^2$

$k = 1.1$

$\lambda = 0.83$

$g = 9.81 \, \mathrm{m}/s^2$

The heat flux generated when the vehicle moving at normal speed of $V_o = 40 \, \mathrm{km}/h = \frac{40*1000m}{60*60\ s} = 11.111 \, m/s$ @ $t_b = \frac{V_o}{a} = \frac{11.111}{2} = 5.556 \, sec.$

Substitute all the above variables in to Eq.3.32and will give the following result:

$$Q\ t\ = \frac{\lambda {V_o}^2 M}{2A_d t_b}\ k + \frac{g \sin \alpha}{2a}$$

$$Q\ t\ = 5735437.355 \, W/m^2$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore,

$$Q\ t\ = 716929.669 \, W/m^2$$

But each disc brake consists of two pads, one pad on each surface of the rotor disc face. Then for the heat flux for one side of the disc is calculated as follows:

$$\mathbf{Q(t) = 358464.835\ W/m^2}$$

The heat flux generated when the vehicle moving at maximum speed of

$V_o = 70\,\text{km}/h = \frac{70*1000m}{60*60\ s} = 19.444\,m/s\ \ @t_b = \frac{V_o}{a} = \frac{19.444}{2} = 9.722\ sec.$

Substitute all the above variables in to Eq.3.32 and will give the following result:

$$Q(t) = \frac{\lambda V_o{}^2 M}{2A_d t_b}\left(k + \frac{g\sin\alpha}{a}\right)$$

$$Q(t) = 10038774.309\ W/m^2$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore

$$Q(t) = 1254846.789\ W/m^2$$

But each disc brake consists of two pad, one pad on each surface of the rotor disc face .then for the heat flux for one side of the disc is calculated as follows:

$$\mathbf{Q(t) = 627423.395\ W/m^2}$$

The heat flux generated when the vehicle moving at over speed of

$$V_o = 80.431\,\text{km}/h = \frac{80.431 * 1000m}{60 * 60\ s} = 22.342\,m/s\ \ t_b = \frac{V_o}{a} = \frac{22.342}{2} = 11.171\ sec.$$

Substitute all the above variables in to Eq.3.32 And will give the following result:

$$Q(t) = \frac{\lambda V_o{}^2 M}{2A_d t_b}\left(k + \frac{g\sin\alpha}{a}\right)$$

$$Q(t) = 11530228.224\ W/m^2$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole heat flux is applied to one disc from the forward part of the carriage.

Therefore

$$Q\ t = 1441278.528\, W/m^2$$

But each disc brake consists of two pads, one pad on each surface of the rotor disc face. Then for the heat flux for one side of the disc is calculated as follows:

$$\mathbf{Q\ t = 720639.264\, W/m^2}$$

Table 6.Summary of heat flux at different train speed and load

Velocity (m/s)	Vehicle mass(Kg)	Heat flux (W/m^2) at one side of the disc
11.111	44000	250276.94
	59240	336963.771
	63020	358464.835
19.444	44000	438061.399
	59240	589789.938
	63020	627423.395
22.342	44000	503143.885
	59240	677414.631
	63020	720639.264

3.5 CALCULATION OF FORCE ACTING ON THE DISC BRAKE

From Eq.3.17 the braking force on the disc is equal to

$$F_{rotor} = \frac{\frac{1}{2}kMV_0^2 + \frac{V_0^2}{2a}Mg\sin\alpha}{2.\frac{r_{rotor}}{r_{wheel}}.(V_O t_{stop} - \frac{1}{2}\{\frac{V_O}{t_{stop}}\}t_{stop}^2)}$$

Braking force on the disc at normal speed of

$V_o = 40\, km/h = 11.111\, m/s$ at a mass of $M = 44000$ kg

$F_{rotor} = 111780.980N$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$F_{rotor} = 13972.623N$

Braking force on the disc at maximum speed of

$V_o = 70 \, {}^{km}/_{h} = 19.444 \, {}^{m}/_{s}$ And at a vehicle mass of M=44000 Kg

$$F_{rotor} = 111790.045N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$$F_{rotor} = 13973.756N$$

Braking force on the disc at over speed of

$V_o = 80.431 \, {}^{km}/_{h} = 22.342 \, {}^{m}/_{s}$ and at a vehicle mass of M=44000 Kg

$$F_{rotor} = 111789.671N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$$F_{rotor} = 13973.709N$$

Braking force on the disc at normal speed of

$V_o = 40 \, {}^{km}/_{h} = 11.111 \, {}^{m}/_{s}$ and at a vehicle mass of $M = 59240$ kg

$$F_{rotor} = 150497.853N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$$F_{rotor} = \frac{150497.853}{8} = 18812.232N$$

Braking force on the disc at maximum speed of

$V_o = 70\,{}^{km}/_{h} = 19.444\,{}^{m}/_{s}$ and at a vehicle mass of M=59240 Kg

$$F_{rotor} = 150510.052N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$$F_{rotor} = 18813.757N$$

Braking force on the disc at over speed of

$V_o = 80.431\,{}^{km}/_{h} = 22.342\,{}^{m}/_{s}$ and at a vehicle mass of M=59240 Kg

$$F_{rotor} = 150509.548N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$$F_{rotor} = 18813.694N$$

Braking force on the disc at normal speed of

$V_o = 40\,{}^{km}/_{h} = 11.111\,{}^{m}/_{s}$ and at a vehicle mass of $M = 63020\,kg$

$$F_{rotor} = 160100.856N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$$F_{rotor} = 20012.607N$$

Braking force on the disc at maximum speed of

$V_o = 70\,{}^{km}/_{h} = 19.444\,{}^{m}/_{s}$ and at a vehicle mass of M=63020 Kg

$$F_{rotor} = 160113.833N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$F_{rotor} = 20014.229N$

Braking force on the disc at over speed of

$V_o = 80.431\,{}^{km}/_{h} = 22.342\,{}^{m}/_{s}$ and at a vehicle mass of M=63020 Kg

$$F_{rotor} = 160113.297N$$

The weight distribution of the vehicle considered is equally distributed between on the front and rear bogies. The vehicle is equipped with eight mechanical disc brakes. Hence, only 1/8 of the whole brake force is applied to one disc from the forward part of the carriage.

$$F_{rotor} = 20014.162N$$

Table 7 Summary of force application at different train speed and load

Velocity $({}^{m}/_{s})$	Vehicle mass(Kg)	Force(N) at one side of the disc
11.111	44000	13972.623
	59240	18812.232
	63020	20012.607
19.444	44000	13973.756
	59240	18813.757
	63020	20014.229
22.342	44000	13973.709
	59240	18813.694
	63020	20014.162

CALCULATION OF CALIPER PRESURE

The total rotor force applied to both the inboard and out board rotors can be used to calculate the caliper clamping force required to stop the vehicle. The magnitude of the clamping force is found using the columbic friction, where μ is about 0.365:

$F_{caliper} = \frac{F_{rotor}}{\mu}$ [21]

For a vehicle moving at normal speed of

$V_o = 40\,{}^{km}/_{h} = \frac{40*1000m}{60*60\ s} = 11.11\,{}^{m}/_{s}$ and mass of M=44000 Kg

$F_{caliper} = 38281.162N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = \frac{60}{360} * 0.1634 = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 1407395.662\ {}^{N}/_{m^2}$

For a vehicle moving at normal speed of

$V_o = 70\,{}^{km}/_{h} = \frac{70*1000m}{60*60\ s} = 19.444\,{}^{m}/_{s}$ and mass of M=44000 Kg

$F_{caliper} = 38284.264N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 1407509.706\ {}^{N}/_{m^2}$

For a vehicle moving at normal speed of

$V_o = 80.431\,{}^{km}/_{h} = 22.342\,{}^{m}/_{s}$ and mass of M=44000 Kg

$F_{caliper} = 38284.136N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 1407505 \, {N}/{m^2}$

For a vehicle moving at normal speed of

$V_o = 40 \, {km}/{h} = \frac{40*1000m}{60*60\,s} = 11.11 \, {m}/{s}$ and mass of M=59240 Kg

$F_{caliper} = 51540.362$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 1894866.25 \, {N}/{m^2}$

For a vehicle moving at normal speed of

$V_o = 70 \, {km}/{h} = \frac{70*1000m}{60*60\,s} = 19.444 \, {m}/{s}$and mass of M=59240Kg

$F_{caliper} = \frac{18813.757N}{0.365}$

$F_{caliper} = 51544.542N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

A=$\frac{60}{360} * 0.1634 = 0.0272$

$P = \frac{F_{caliper}}{A} = \frac{51544.542N}{0.0272m^2}$

$P = 1895019.926 \, {N}/{m^2}$

For a vehicle moving at normal speed of

$V_o = 80.431 \, {km}/{h} = 22.342 \, {m}/{s}$and mass of M=59240 Kg

$F_{caliper} = \frac{18813.694N}{0.365}$

$F_{caliper} = 51544.368N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 1895013.53\ {}^{N}\!/_{m^2}$

For a vehicle moving at normal speed of

$V_o = 40\ {}^{km}\!/_{h} = \frac{40*1000m}{60*60\ s} = 11.11\ {}^{m}\!/_{s}$ and mass of M=63020 Kg

$F_{caliper} = 54829.064N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 2015774.412\ {}^{N}\!/_{m^2}$

For a vehicle moving at normal speed of

$V_o = 70\ {}^{km}\!/_{h} = \frac{70*1000m}{60*60\ s} = 19.444\ {}^{m}\!/_{s}$and mass of M=63020Kg

$F_{caliper} = 54833.506N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 2015937.72\ {}^{N}\!/_{m^2}$

For a vehicle moving at normal speed of

$V_o = 80.431\ {}^{km}\!/_{h} = 22.342\ {}^{m}\!/_{s}$ and mass of M=63020 Kg

$F_{caliper} = 54833.32N$

This force generates a clamping pressure. Brake pads cover a 60^0angle along the rotor. This results in a contact area of

$A = 0.0272$

$P = \frac{F_{caliper}}{A}$

$P = 2015930.882\,{}^{N}/_{m^2}$

Table8.Summary of force pressure at different vehicle speed and load

Velocity $({}^{m}/_{s})$	Vehicle mass(Kg)	Pressure $({}^{N}/_{m^2})$
11.111	44000	1407395.662
	59240	1894866.25
	63020	2015774.412
19.444	44000	1407509.706
	59240	1895019.926
	63020	2015937.72
22.342	44000	1407505
	59240	1895013.53
	63020	2015930.882

CALCULATION OF COEFFICIENT OF HEAT TRANSFER

To calculate the convective film coefficient (coefficient of heat transfer) as a function of the railway vehicle velocity *v*, the following formula [22] should be used:

$h = \frac{0.037k_a}{2R} Re^{0.8} Pr^{0.33}$……………………………………………………………………….3.34

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

$Re = \frac{2\rho_a RV}{\mu_{va}}$…………………………………………………………………………………...3.35

$Pr = \frac{C_{pa} * \mu_{va}}{k_a}$……………………………………………………………………………………3.36

Here, material properties: thermal conductivity $k_a = 0.02601\ W/mK$, density $\rho_a = 1.17\,{}^{Kg}/_{m^3}$, dynamic viscosity $\mu_{va} = 1.847 * 10^{-5}\,{}^{Ns}/_{m^2}$ and specific heat capacity $C_{pa} = 1006.284 J/kgK$, are given for the surrounding (ambient air) air at surrounding temperature of T=298.65 K, v is the train speed and R is the radius of the rotor disc which is 180mm.

Therefore the convective film coefficient for three different velocity of the train will be as follows:

For $v = 11.111 \ {}^{m}/_{s}$,

$h = 50.336 \ {}^{w}/_{k.m^2}$

For $v = 19.444 \ {}^{m}/_{s}$,

$h = 78.760 \ {}^{w}/_{k.m^2}$

For $v = 22.342 \ {}^{m}/_{s}$,

$h = 88.019 \ {}^{w}/_{k.m^2}$

CHAPTER FOUR

MODELING AND ANALYSIS USING FINITE ELEMENT METHODE

4.1 INTRODUCTION AND PROBLEM DEFNITION

Due to the application of brakes on the train disc brake rotor, heat generation takes place due to friction and this thermal flux has to be conducted and dispersed across the disc rotor cross section. The condition of braking is very much severed and thus the thermo-mechanical analysis has to be carried out. 3D analysis, which is an exact representation, is carried in this thesis.

4.2 THERMAL AND STRUCTURAL ANALYSIS

THERMAL ANALYSIS

A Thermal analysis calculates the temperature distribution and related thermal quantities in a system or component. A transient thermal analysis determines the temperature distribution and other thermal quantity under conditions that vary over a period of time which is this thesis focuses.

STRUCTURAL ANALYSIS

Using the result obtained from thermal and applied pressure Transient structural analysis will analyze in this thesis.

To simplify the analysis, several assumptions were also been made as follows:

- All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux.
- The heat transfer involved in this analysis takes place only by conduction and convection.

Heat transfer by radiation can be neglected as it amounts only to 5 % to 10 %.

- The disc material is considered as homogeneous and isotropic.
- Inertia and body force effects are negligible during the analysis.

- The disc is stress free before the brake application
- In this analysis, the ambient temperature and initial temperature is set to 25.5 °C.
- Uniform pressure distribution generated by the brake pad onto the disc brake surface is Considered.
- No wear

4.3 MATERIALS USED FOR ROTOR DISC, PAD AND THEIR PROPERTIES

Cast iron usually refers to grey cast iron for disc and Ceramic fiber Al_2O_3 for the pad have selected for this thesis.

Table 9.Properties of the materials of disc and pad

PROPERTIES	DISC	PAD
	Grey Cast Iron BS grade 180	Ceramic Fiber Al_2O_3
DENSITY(Kg/m3), ρ	7150	3800
YOUNGS MODULUS(GPa),E	140	325
THERMAL CONDUCTIVITY(W/m.K), k	52	30
SPECIFIC HEAT capacity (J/Kg.K),Cp	423	700
POSSION'S RATIO, υ	0.24	0.22
COEFFICIENT OF EXPANSION($1/k$), α	1.1E-5	8.4E-6

4.4 GEOMETRICAL MODELING OF DISC AND PAD USISG CATIA SOFTWARE

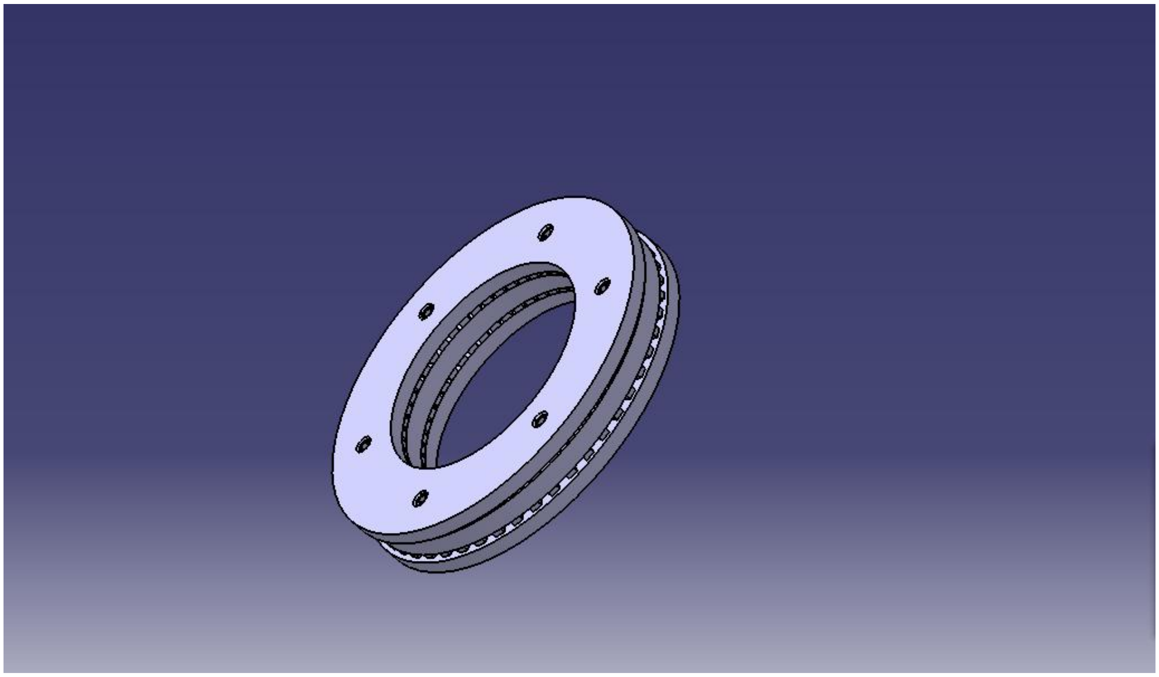


Fig.4.1.3D model of disc brake

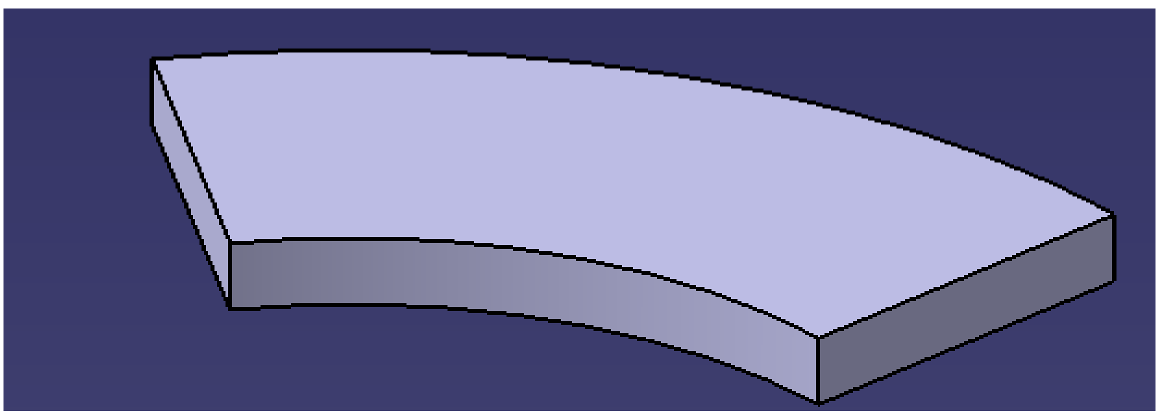


Fig. 4.2.3D model of pad

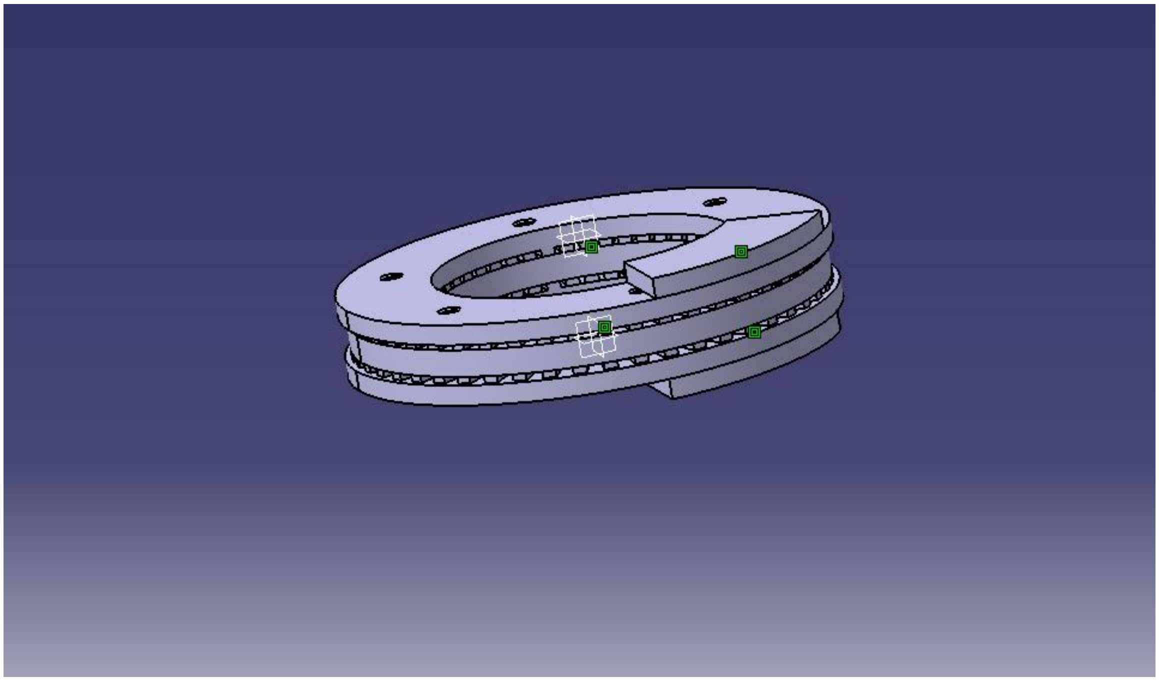


Fig.4.3. Disc and pad assembling model

4.5 CREATING A FINITE ELEMENT 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 disc and pad are 3D element the appropriate meshing method will be volume mesh so that all the volume of the disc-pad assembly is discretized to the smallest part of the disc-pad assembly and refined the contact area of the disc and pad . The output of the mesh consists of 128879 nodes and 73186 elements. After this the object is ready to be analyzed by setting the analysis on the ANSYS work bench.

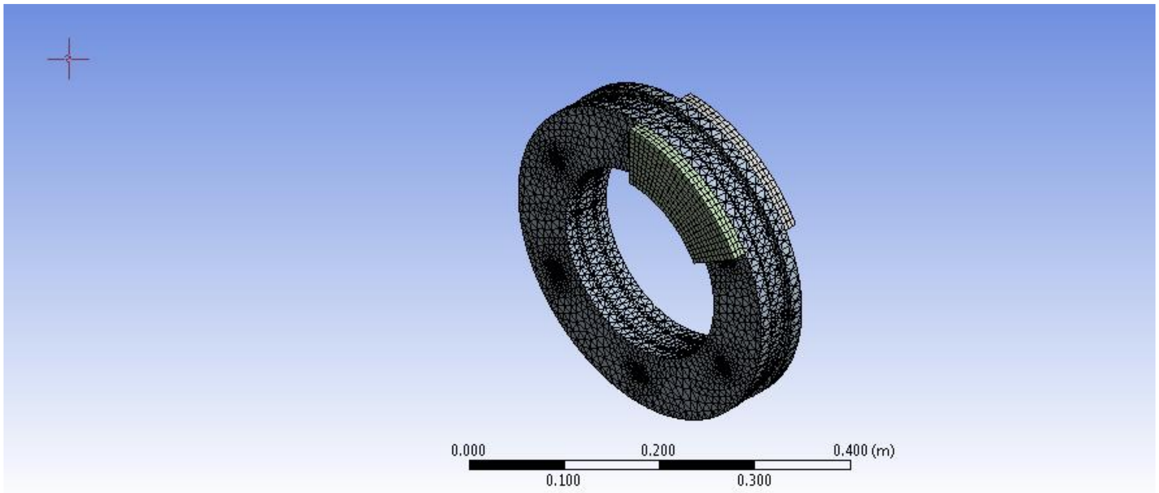


Fig.4.4. meshing of disc-pad assembly

4.6 APPLYING LOADS AND BOUNDRY CONDITIONS

In thermal and structural analysis of disc brake, we have to apply thermal and boundary conditions on 3-D disc model of the disc brake.

4.6.1 Thermal loads and boundary conditions

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

- The initial temperature of the disc and the pads is 25.5 °C.
- The surface convection condition is applied at all surfaces of the disc with the values of the convection coefficient (h) which was calculated before.
- The heat flux into the brake disc during braking can be calculated.

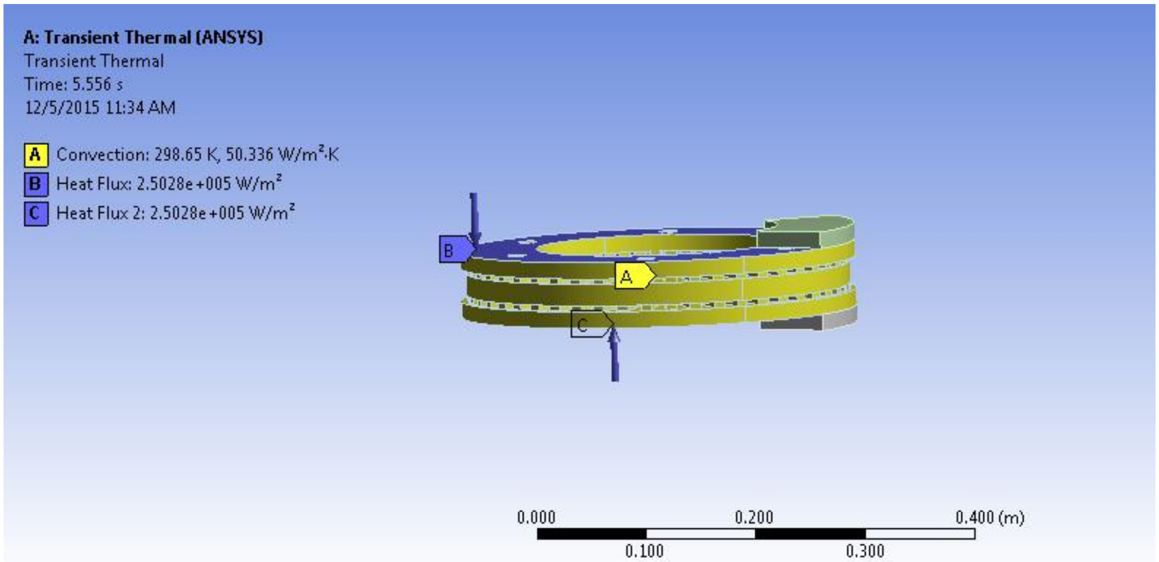


Fig.4.5.Thermal loads and boundary conditions

4.6.2 Structural loads and boundary conditions

All the nodes on the inner disc radius are fixed. So the nodal displacements on the inner disc radius become zero.i.e. in radial, axial, and angular directions. The pads are fixed except in Z-direction.

- The pressure on the rubbing contact surfaces of the disc and brake pad was calculated before.
- Coefficient of friction between disc and pad.
- Fixing the disc on the axle by force fit(press fit)

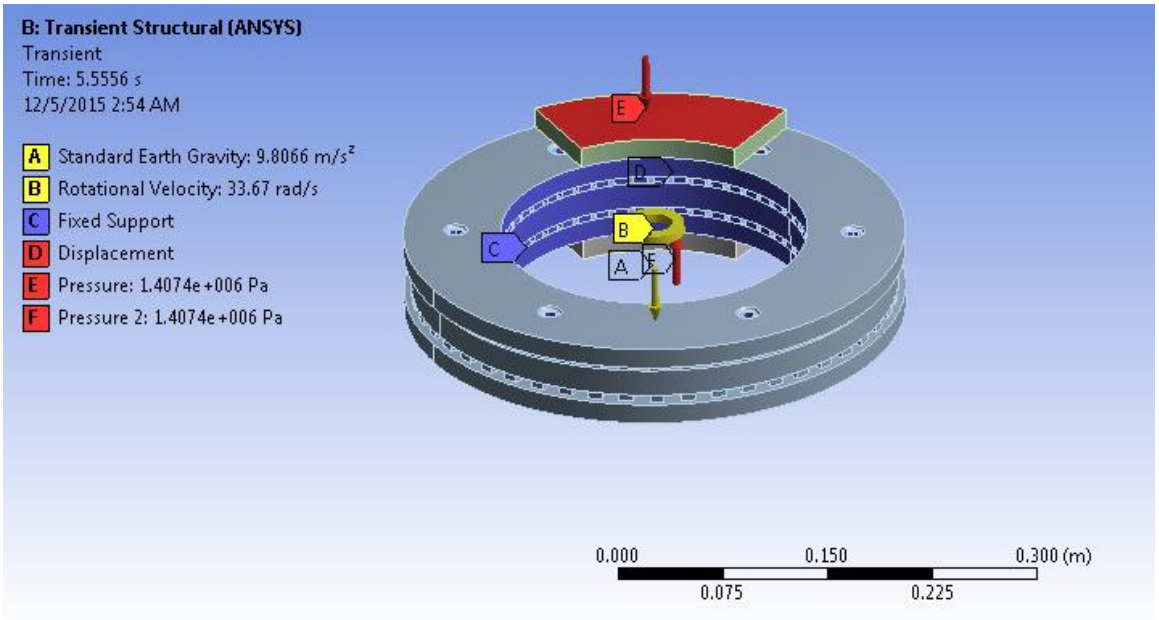


Fig.4.6 Structural loads and boundary conditions

CHAPTER FIVE

RESULTS AND DISCUSIONS

5.1Thermal Analysis Result

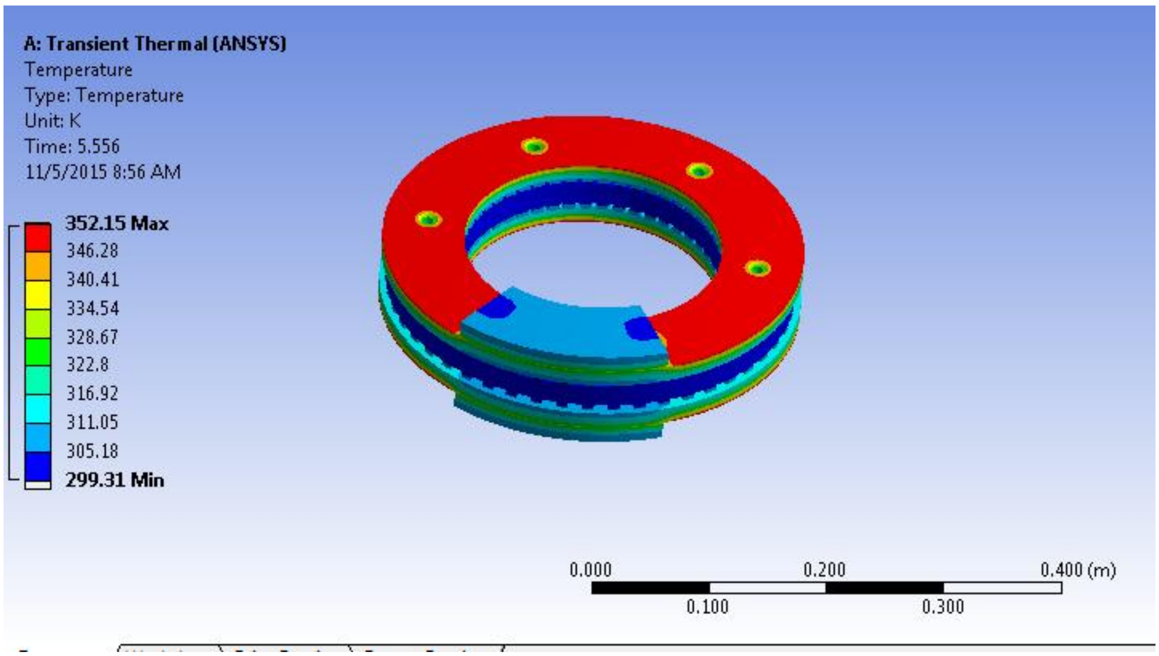


Fig 5.1 Nodal Temperature at V1and M1

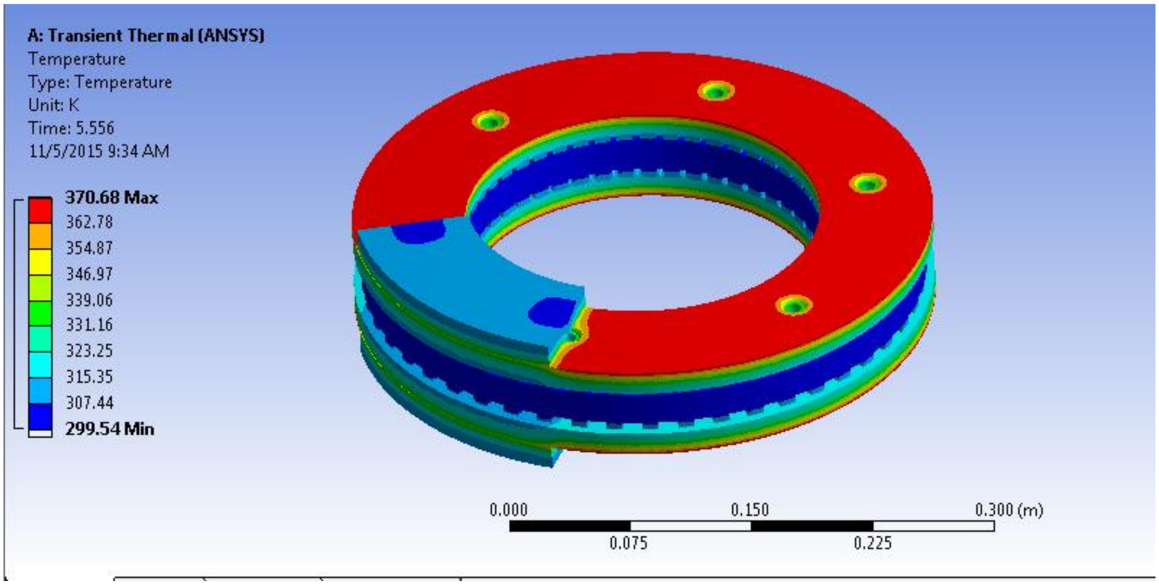


Fig 5.2 Nodal Temperature at V1and M2

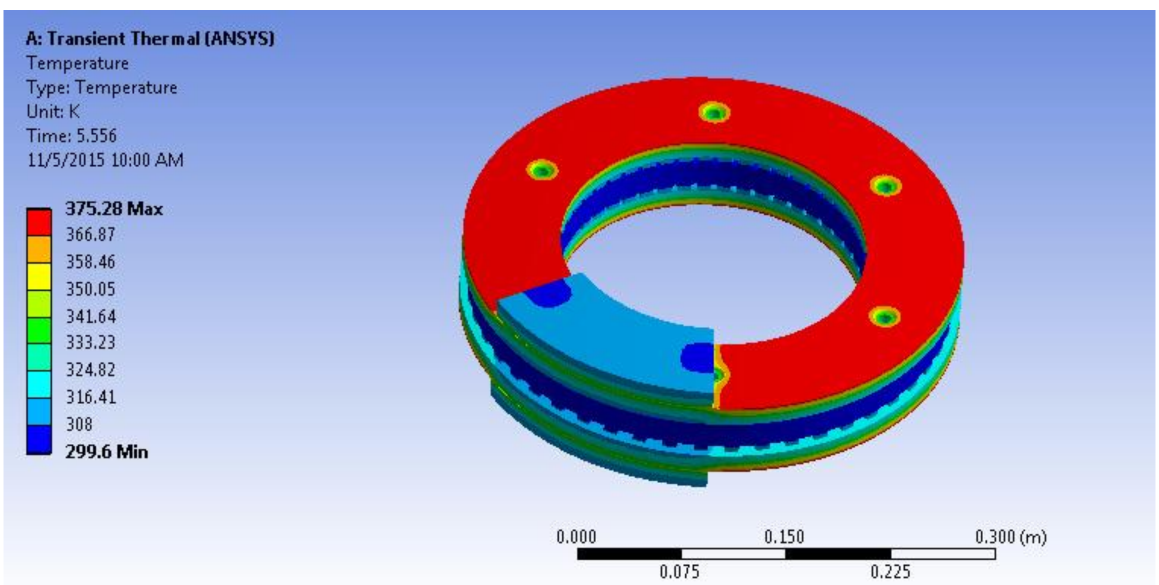


Fig 5.3 Nodal Temperature at V1and M3

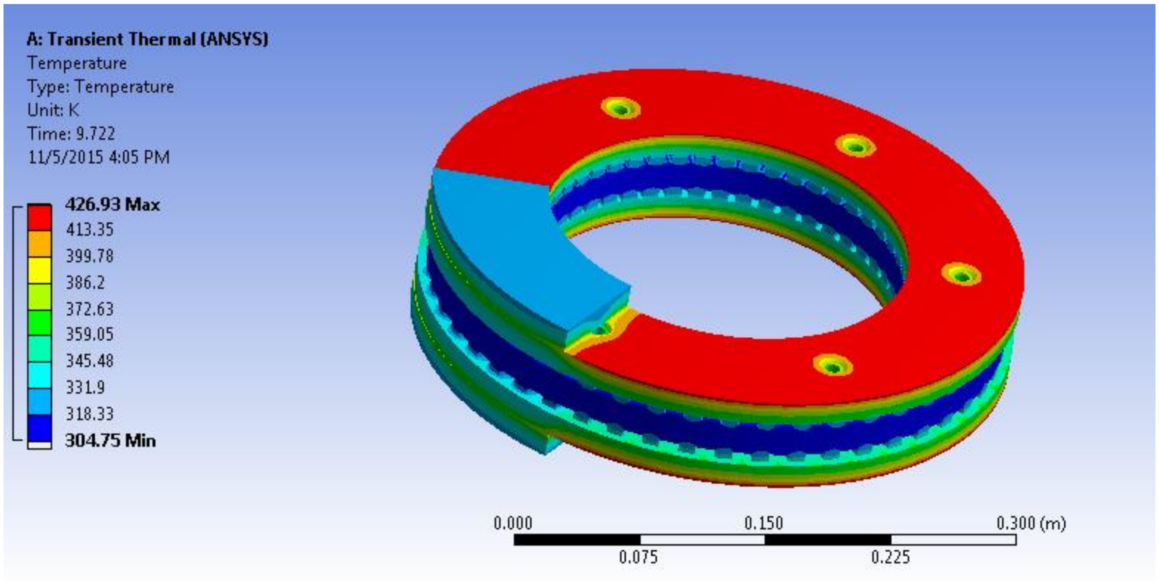


Fig 5.4 Nodal Temperature at V2and M1

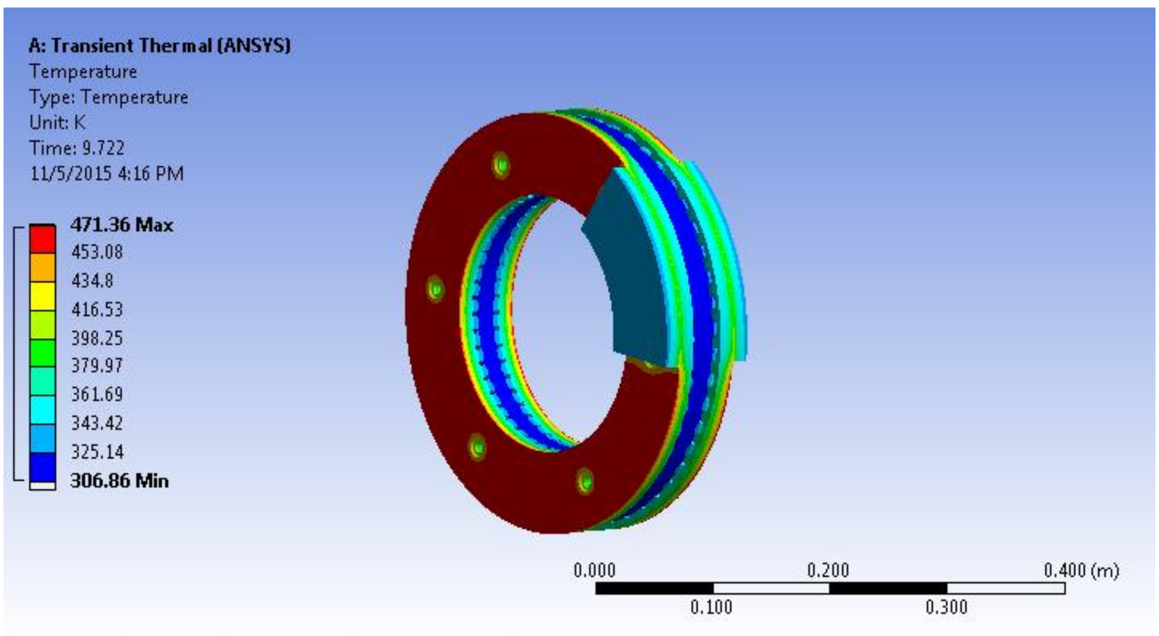


Fig 5.5 Nodal Temperature at V2and M2

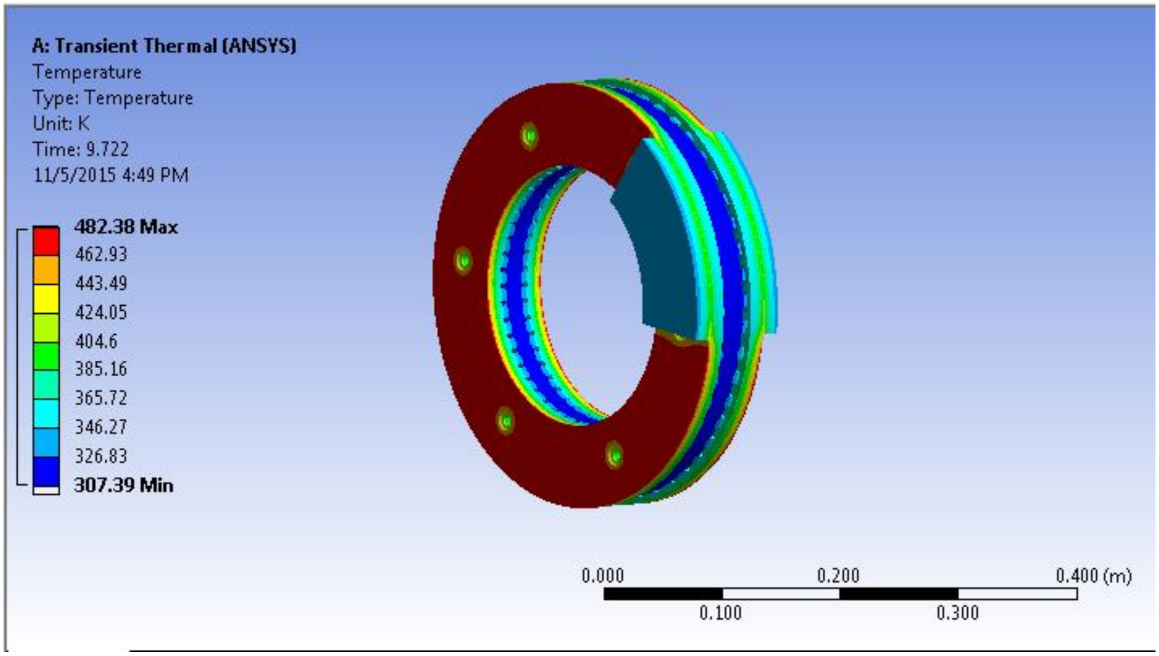


Fig 5.6 Nodal Temperature at V2and M3

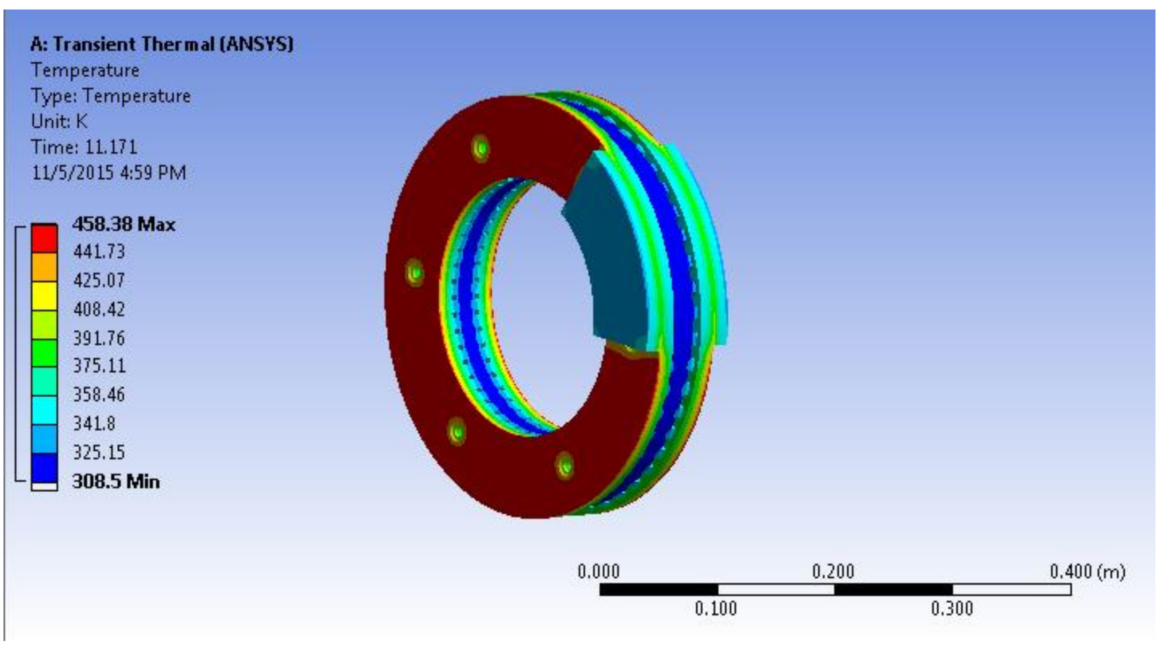


Fig 5.7 Nodal Temperature at V3and M1

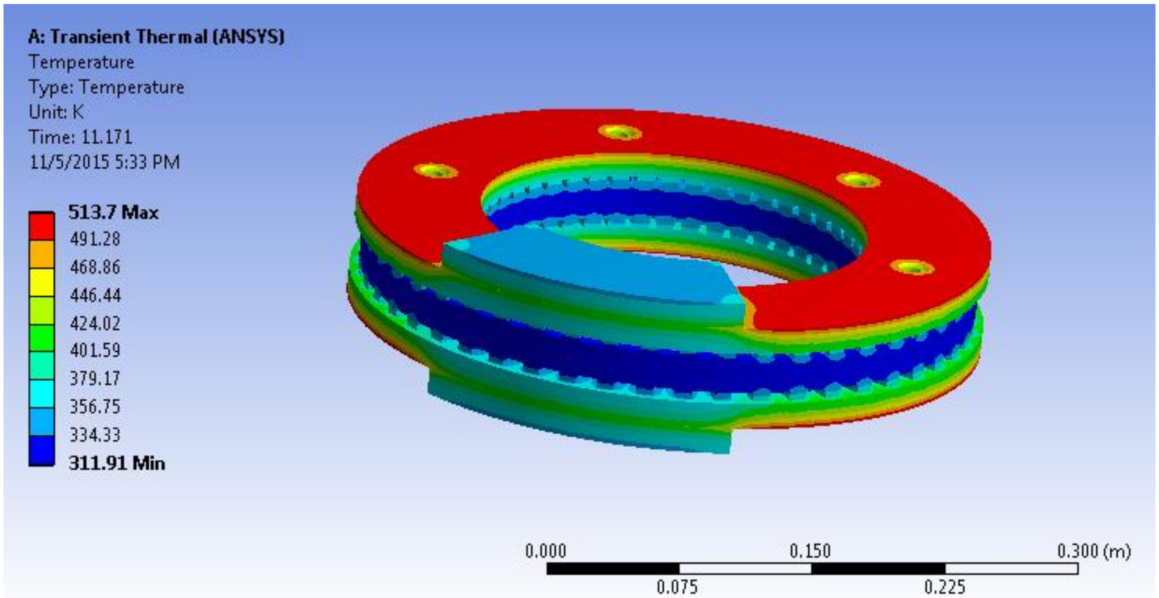


Fig 5.8 Nodal Temperature at V3and M2

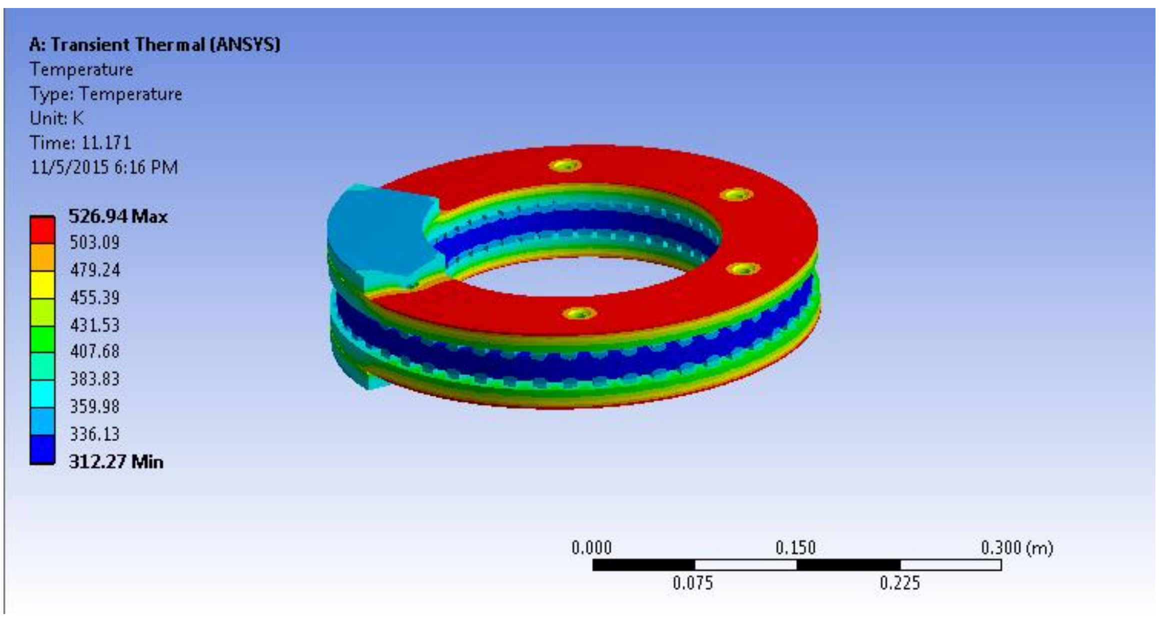
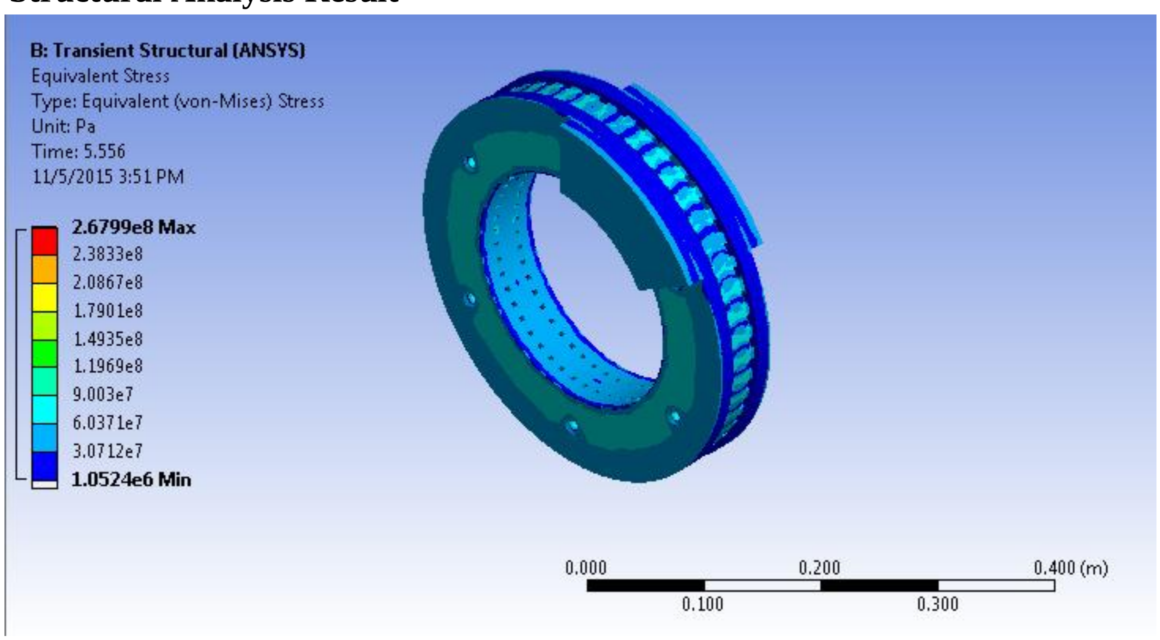


Fig 5.9 Nodal Temperature at V3and M3

Table 10Thermal Analysis Results

Velocity (m/s)	Vehicle mass(Kg)	Temperature(K)	
		Minimum	Maximum
11.111	44000	299.31	352.15
	59240	299.54	370.68
	63020	299.6	375.28
19.444	44000	304.75	426.93
	59240	306.86	471.36
	63020	307.39	482.38
22.342	44000	308.5	458.38
	59240	311.91	513.7
	63020	312.27	526.94

5.2 Structural Analysis Result



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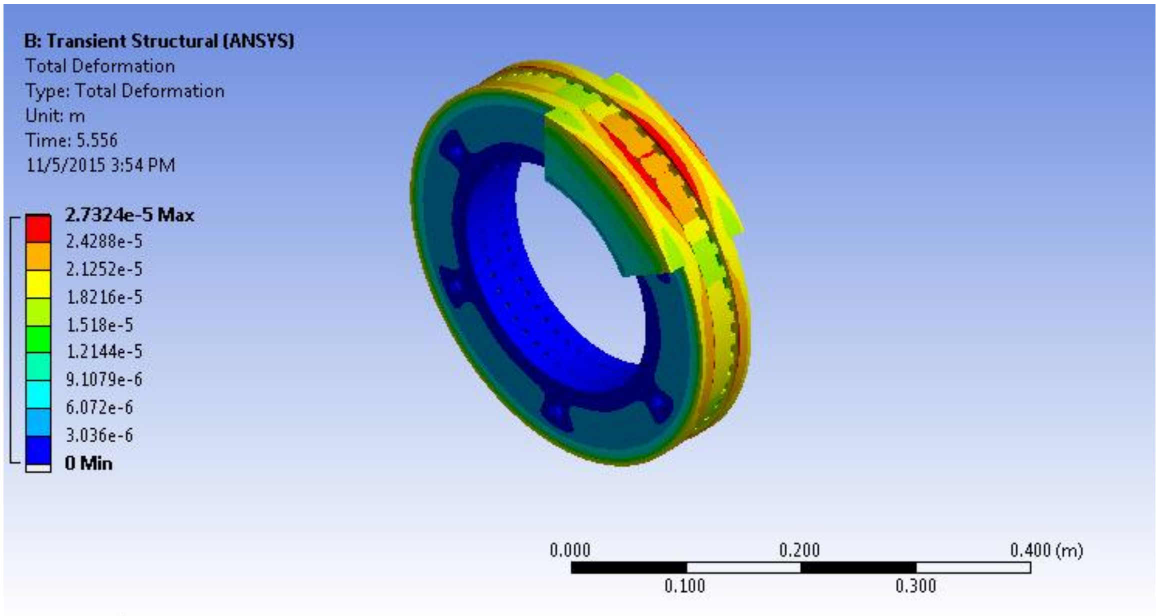
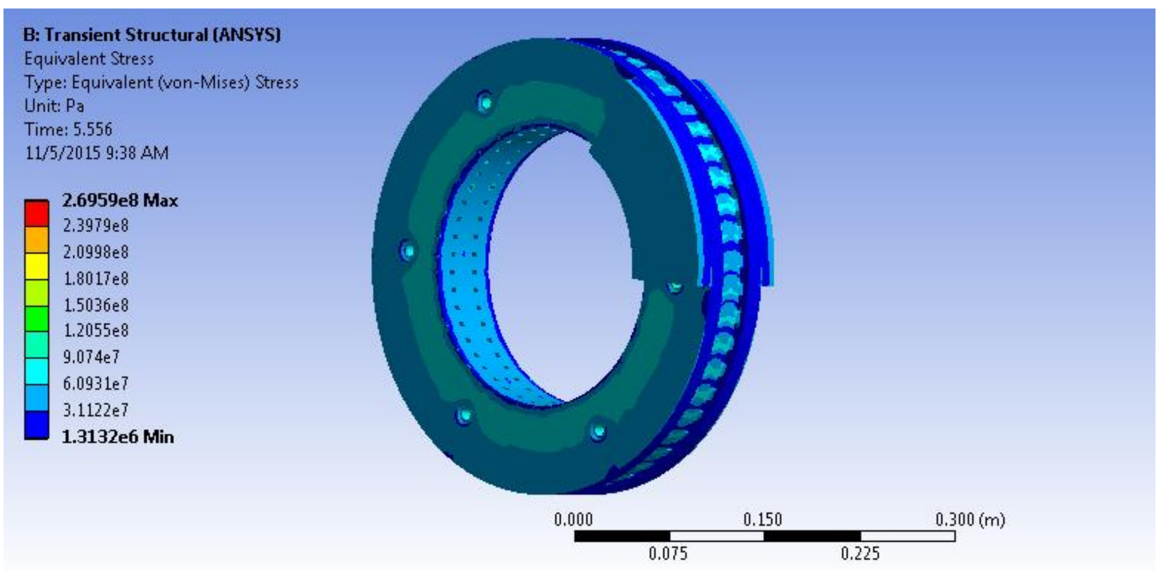


(b)

Fig 5.10 Equivalent stress (a) and total deformation (b) at V1and M1



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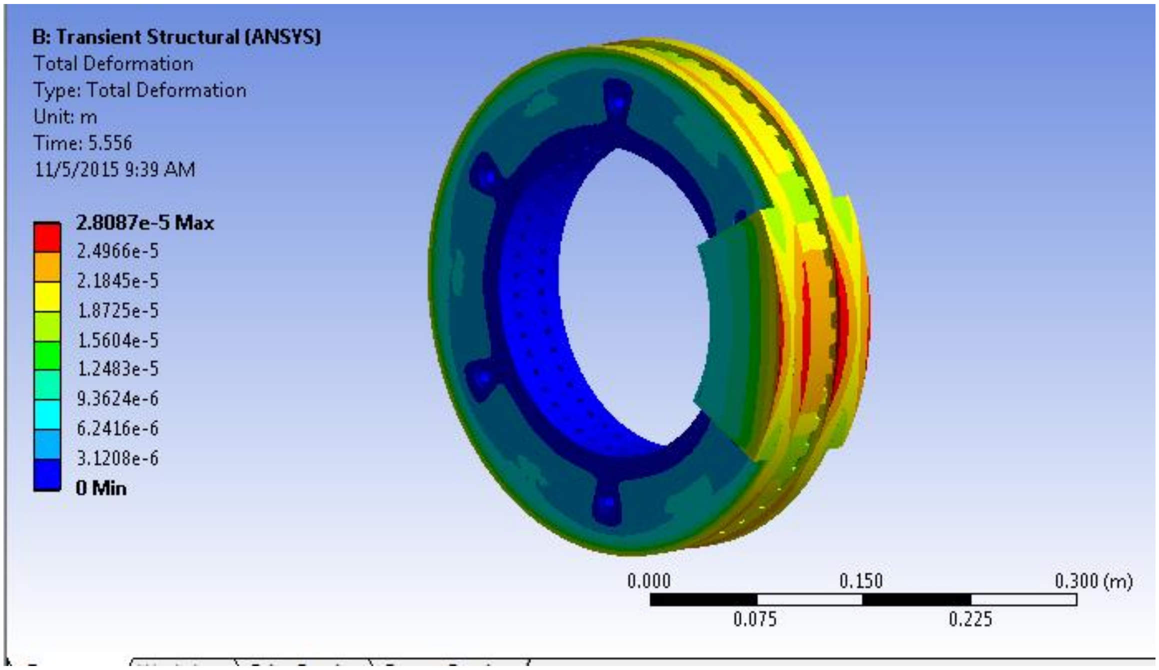
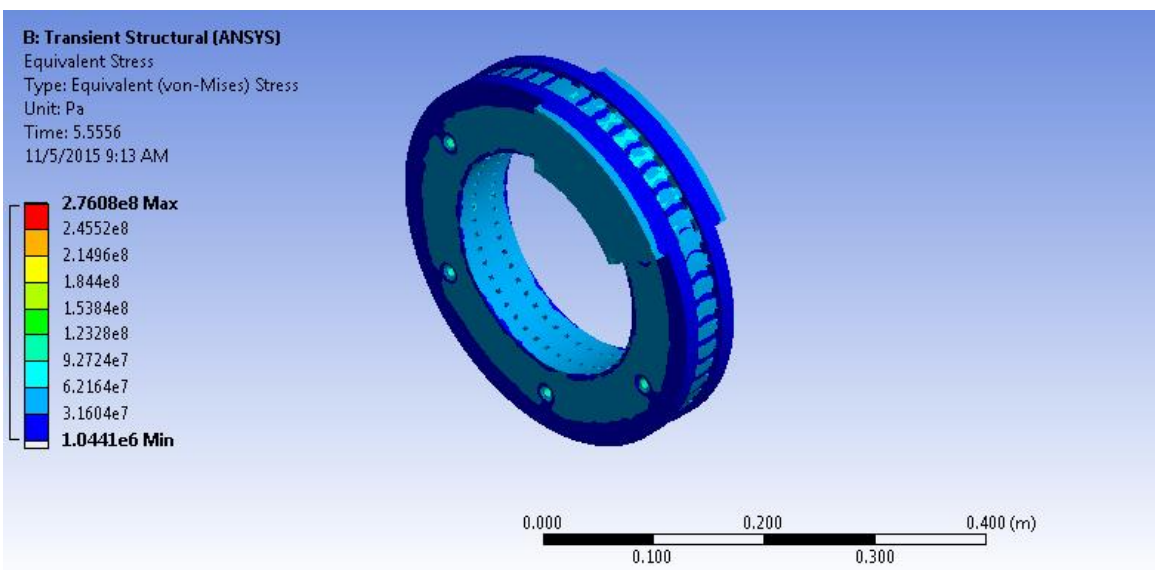


(b)

Fig 5.11 Equivalent stress (a) and total deformation (b) at V1and M2



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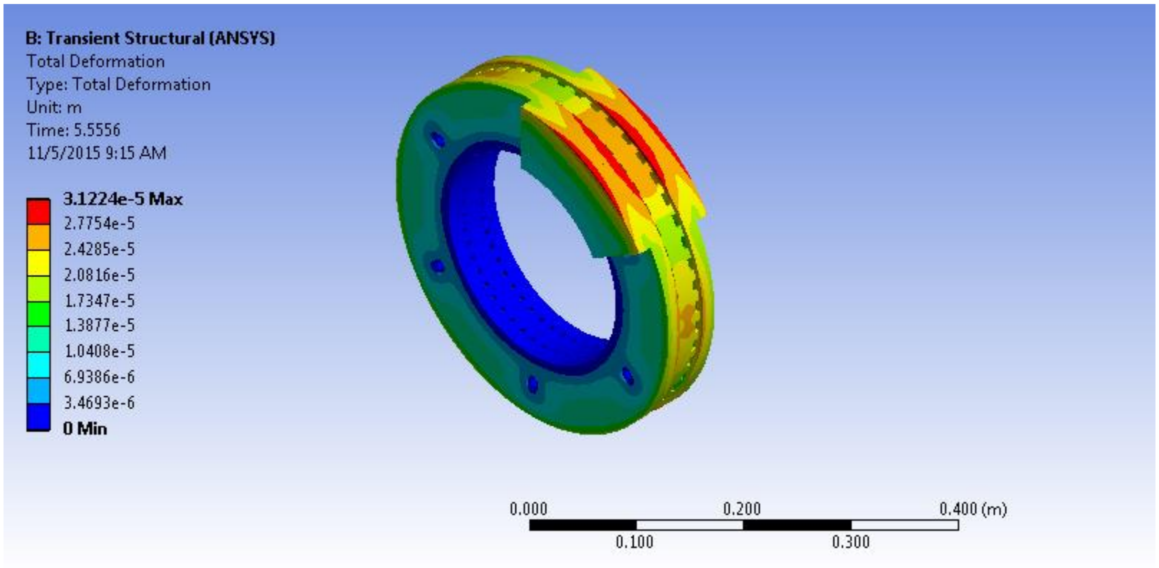
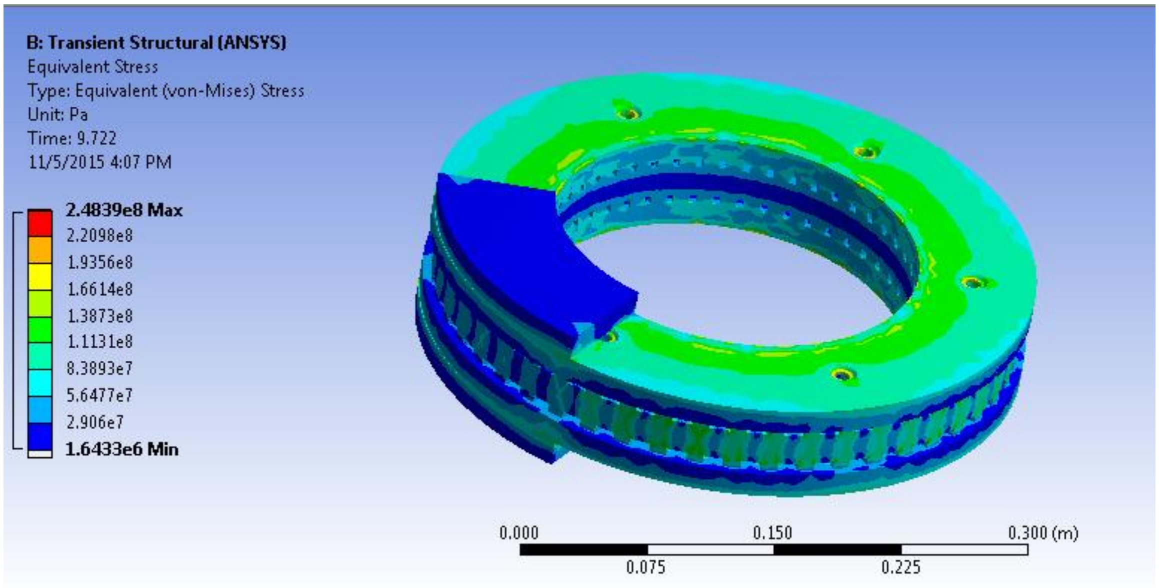


(b)

Fig 5.12 Equivalent stress (a) and total deformation (b) at V1and M3



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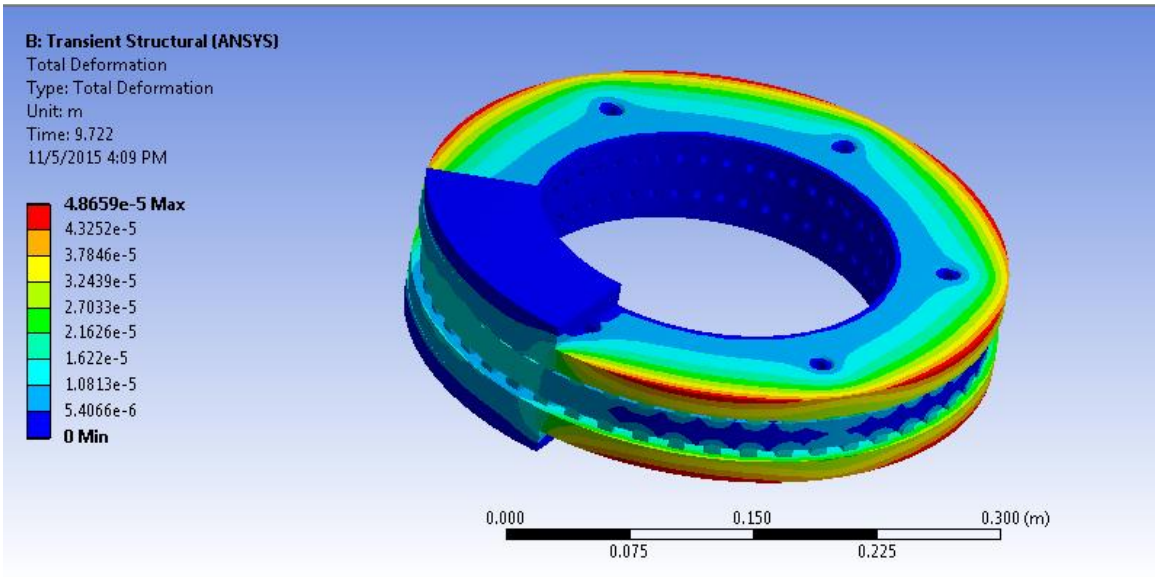
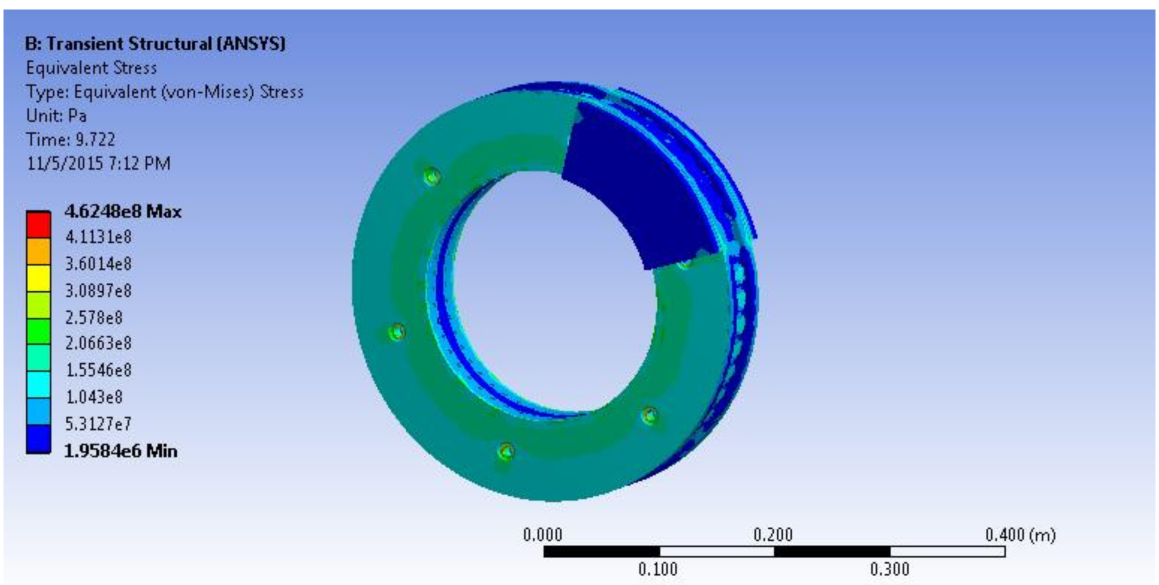


(b)

Fig 5.13 Equivalent stress (a) and total deformation (b) at V2and M1



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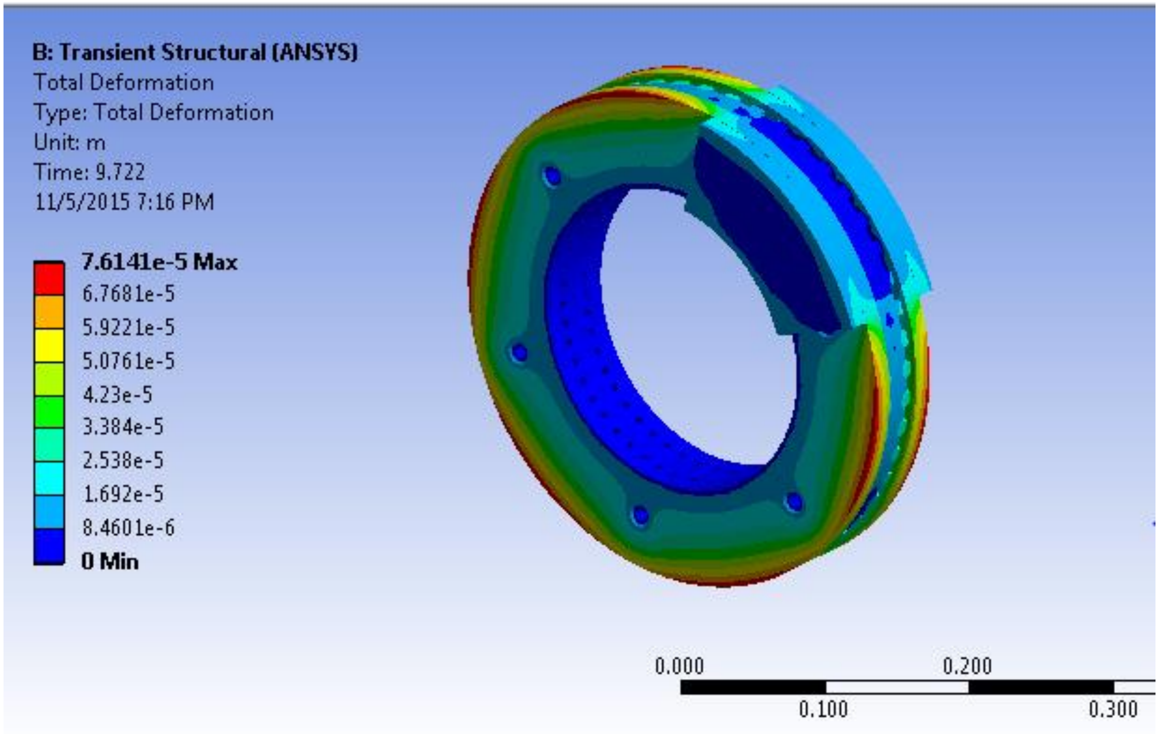
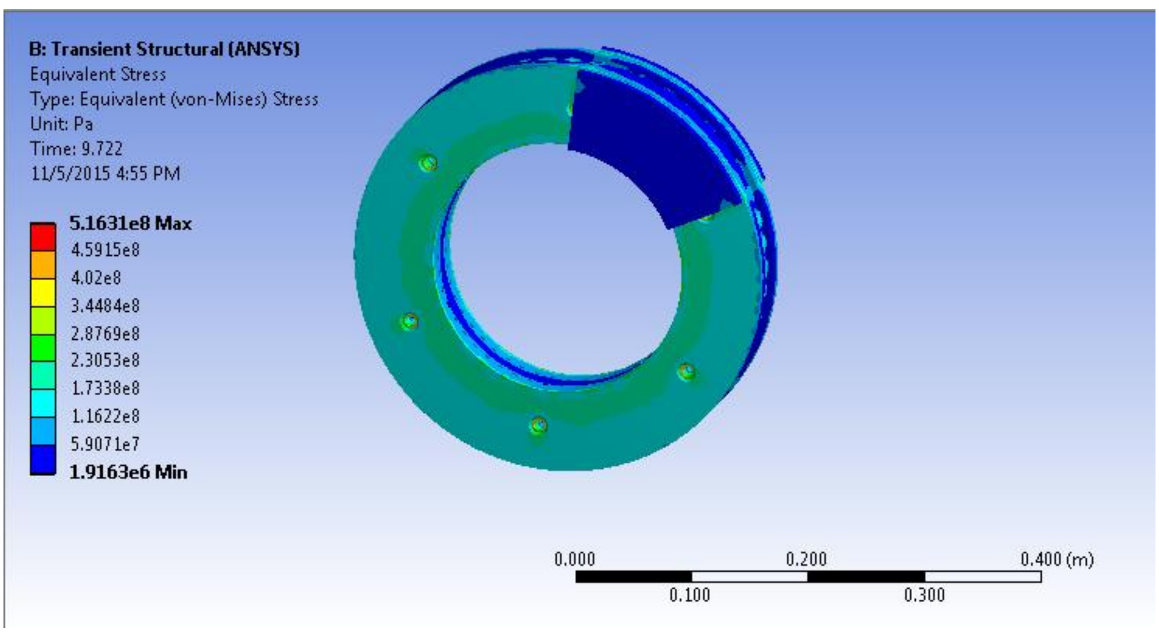


(b)

Fig 5.14 Equivalent stress (a) and total deformation (b) at V2and M2



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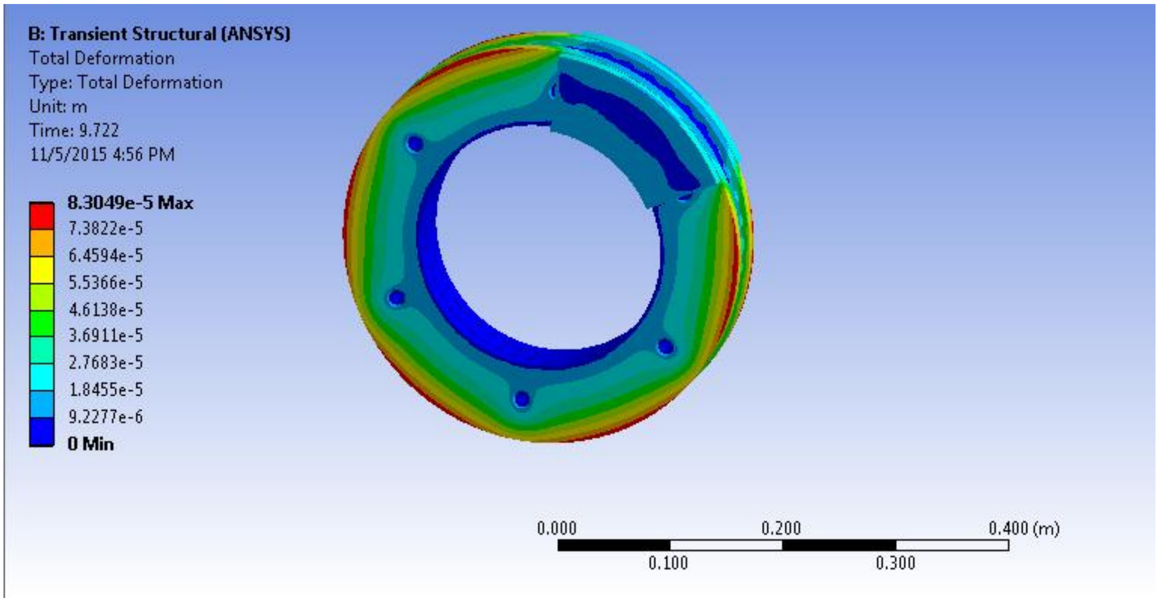
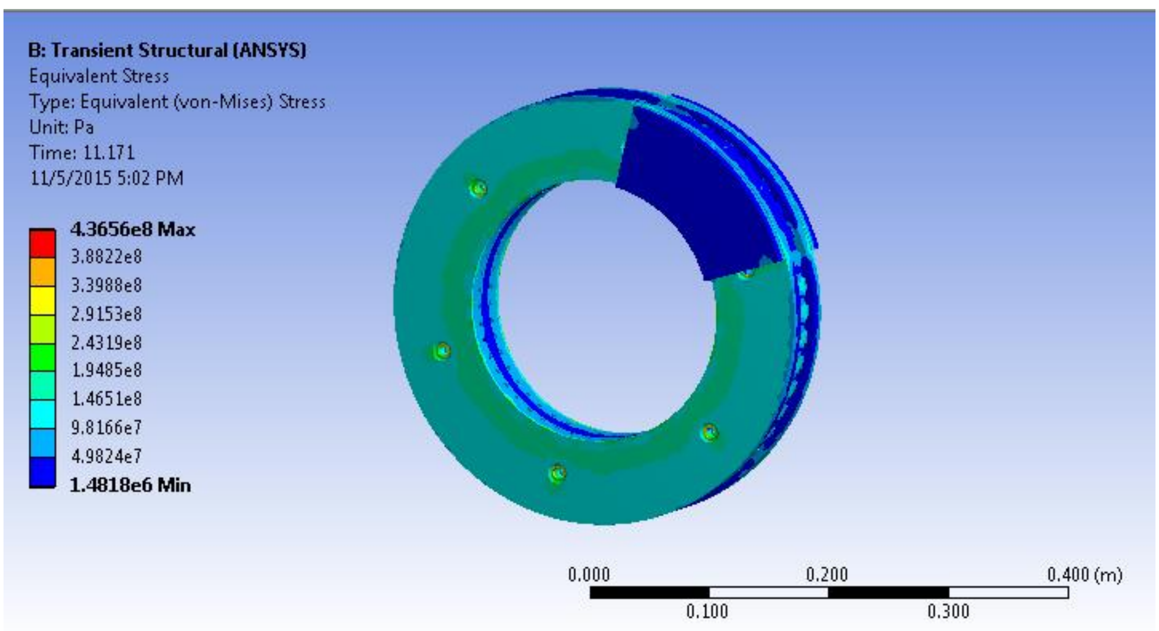


(b)

Fig 5.15 Equivalent stress (a) and total deformation (b) at V2and M3



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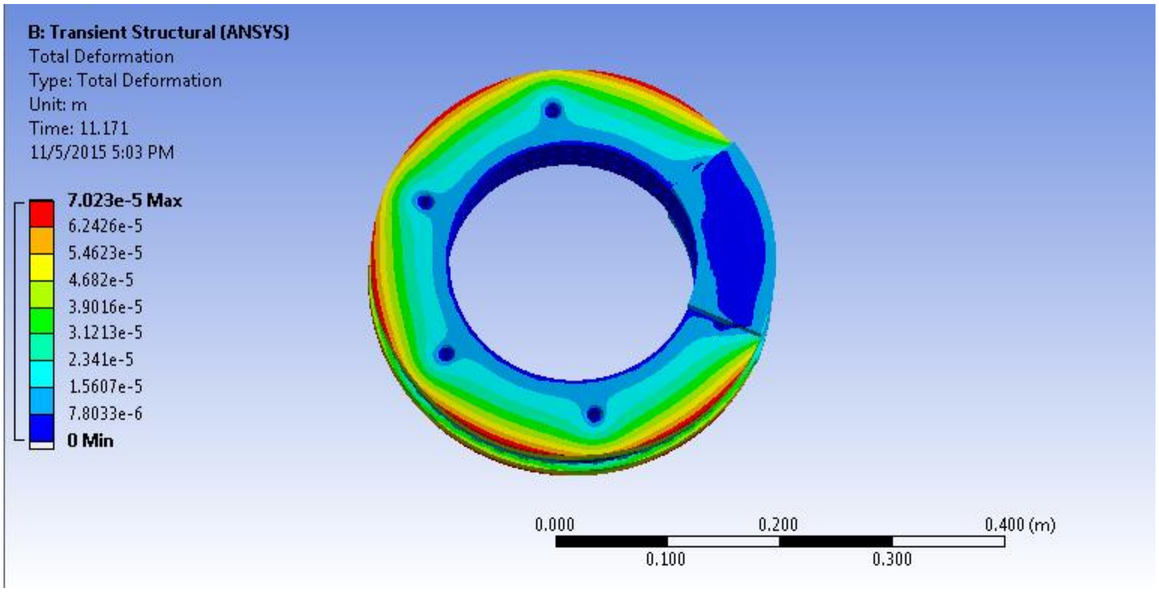
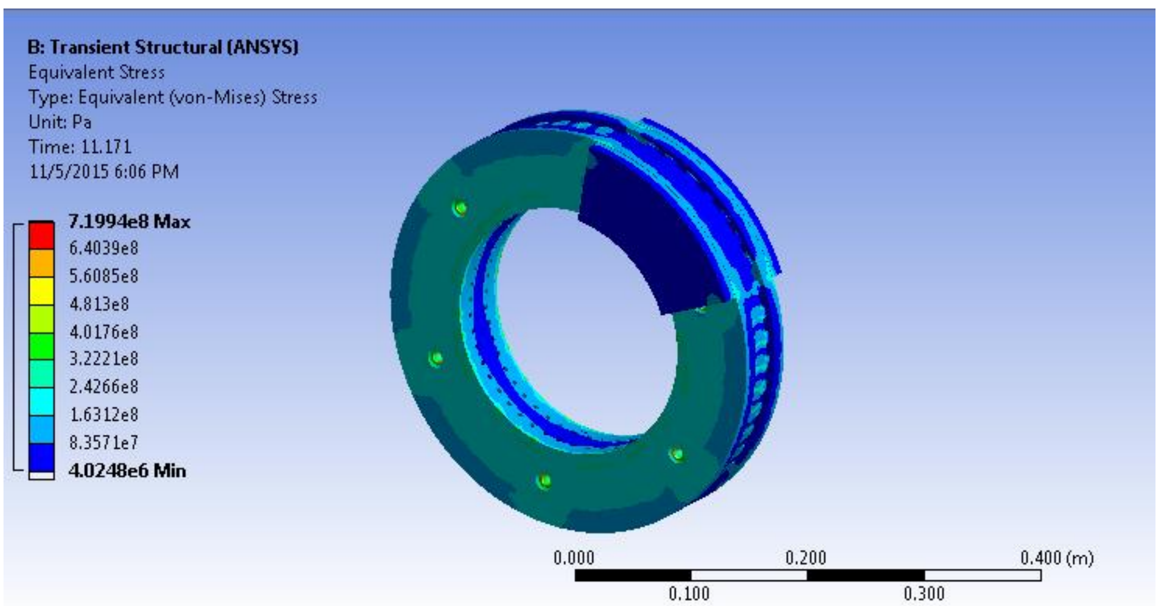


(b)

Fig 5.16 Equivalent stress (a) and total deformation (b) at V3and M1



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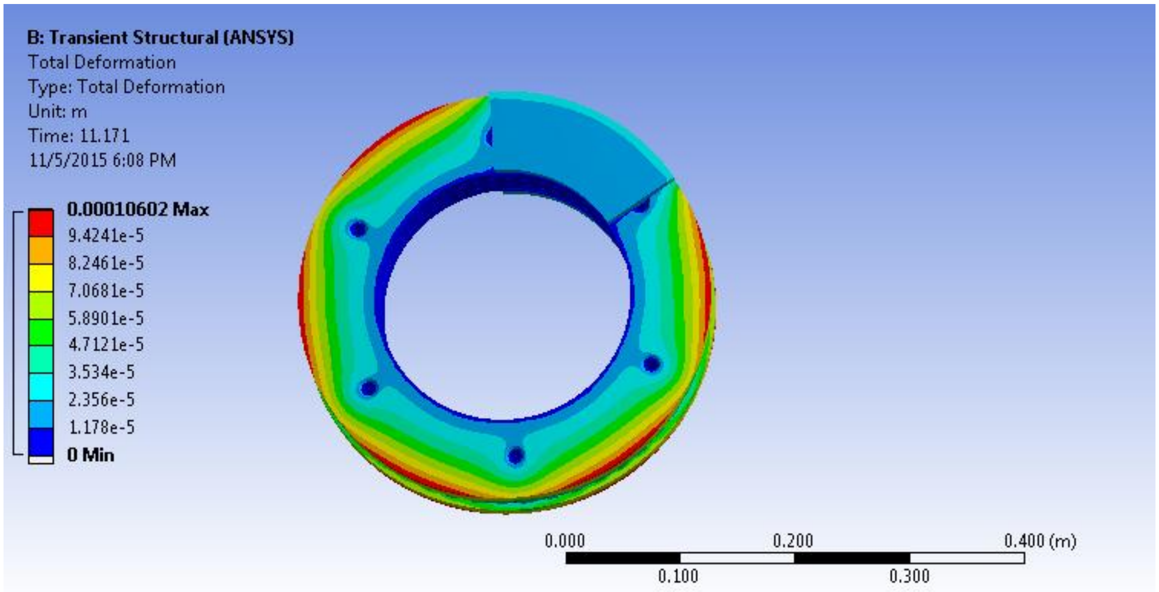
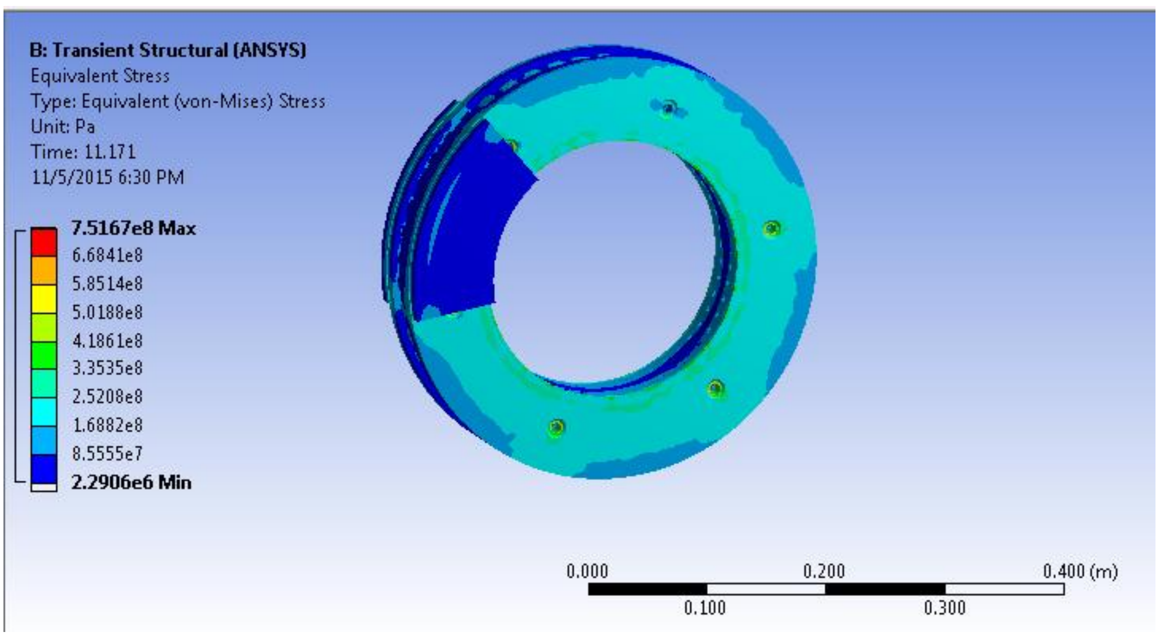
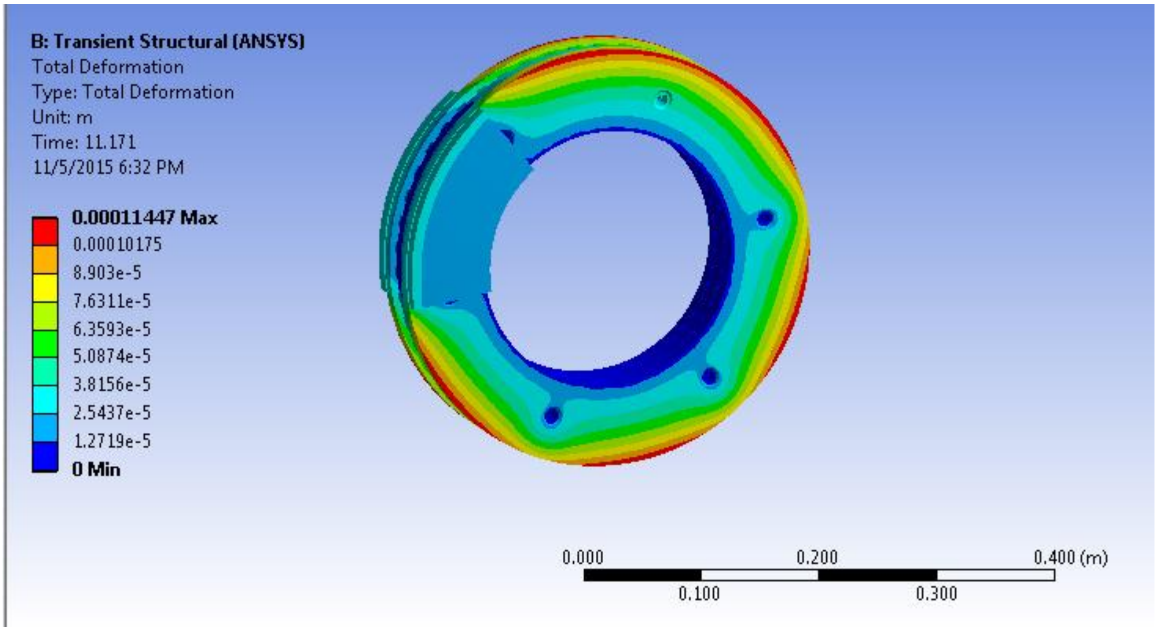


(b)

Fig 5.17 Equivalent stress (a) and total deformation (b) at V3and M2



(a)



(b)

Fig 5.18 Equivalent stress (a) and Total deformation (b) at V3and M3

Table 11.Structural Analysis Results

Velocity (m/s)	Vehicle mass(Kg)	Equivalent stress(MPa)		Total deformation(m)	
		Minimum	Maximum	Minimum	Maximum
11.111	44000	1.0524	267.99	0	2.7324e-5
	59240	1.3132	269.59	0	2.8087e-5
	63020	1.0441	276.08	0	3.1224e-5
19.444	44000	1.6433	278.39	0	4.8659e-5
	59240	1.9584	462.48	0	7.6141e-5
	63020	1.9163	516.31	0	8.3049 e-5
22.342	44000	1.4818	436.56	0	7.023e-5
	59240	4.0248	719.94	0	0.00010602
	63020	2.2906	751.67	0	0.00011447

5.3 LIFE TIME ANALYSIS

Now a day's rail way transportation system is becoming popular in terms of cost and long life operation all over the world including Ethiopia; hence the transportation system should be free of risks that arise due to component failures such as disc brake.one of the major problems that lead to disc brake unknowing the life of the component. As temperature and loading of the vehicle increases the life time of the disc decreases due to high load and temperature applied on it. So it better to know the life time of the material. In this thesis the life time of the was analyzed using ANSYS workbench. The thermo-mechanical results were used as an input to get the life time result.as the load and speed of the train increases, the thermo-mechanical analysis (temperature, stress and deformation) increases resulting in decreasing the life time of the disc. The life was expressed in month.

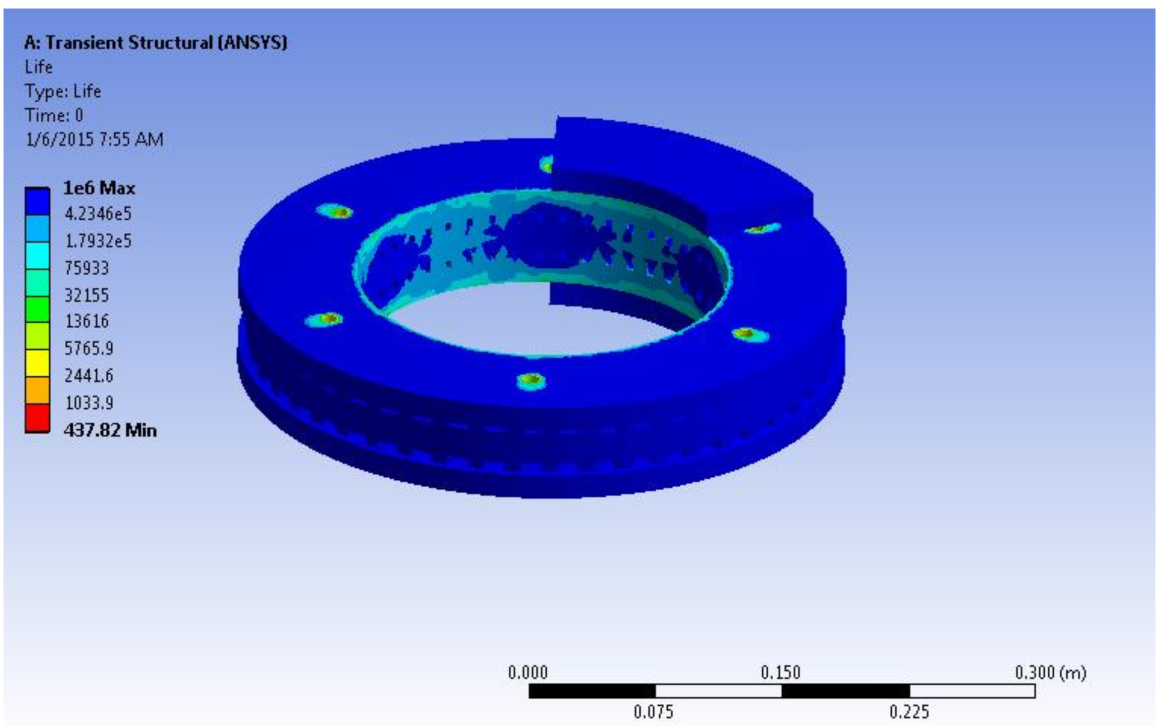


Fig 5.19 Life time at V1 and M1

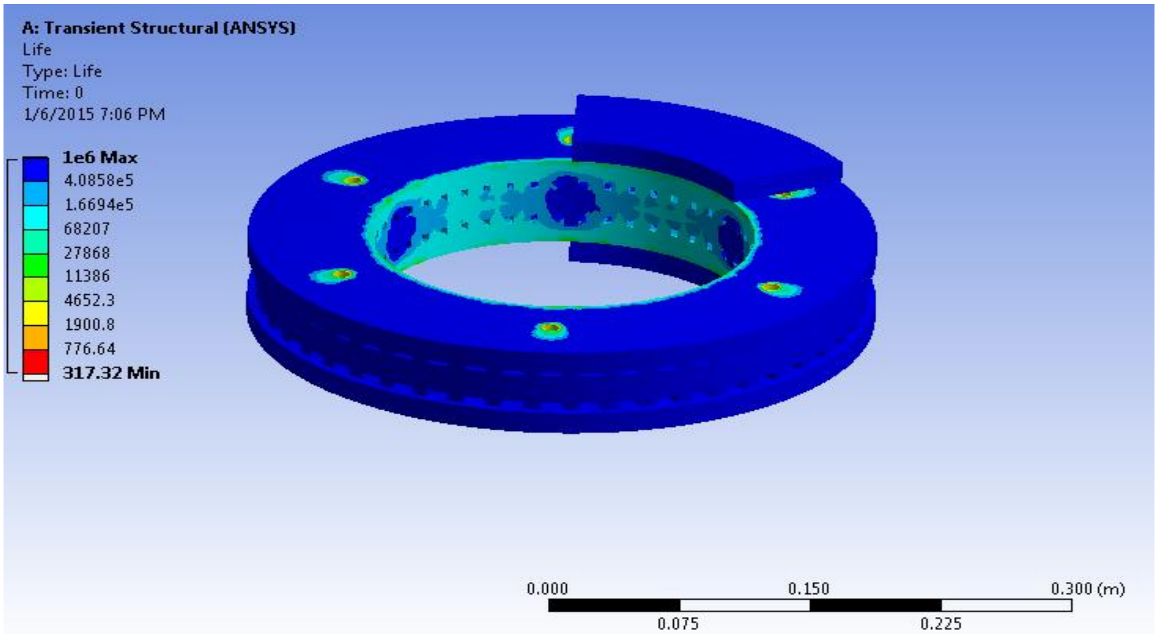


Fig 5.20 Life time at V1 and M2

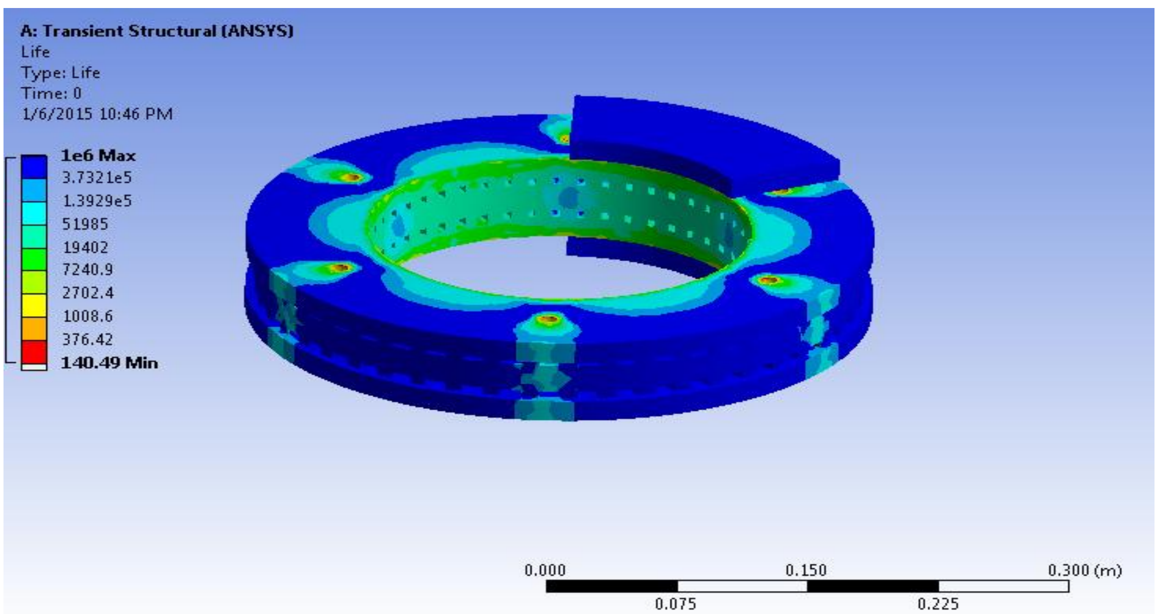


Fig 5.21 Life time at V1 and M3

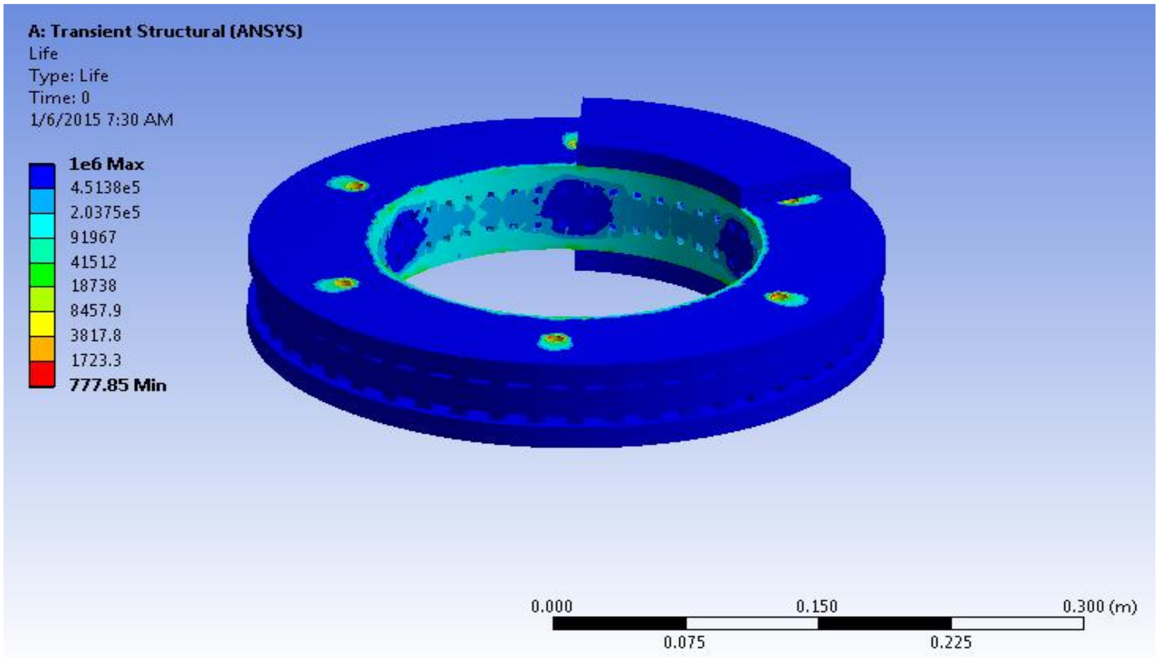


Fig 5.22 Life time at V2 and M1

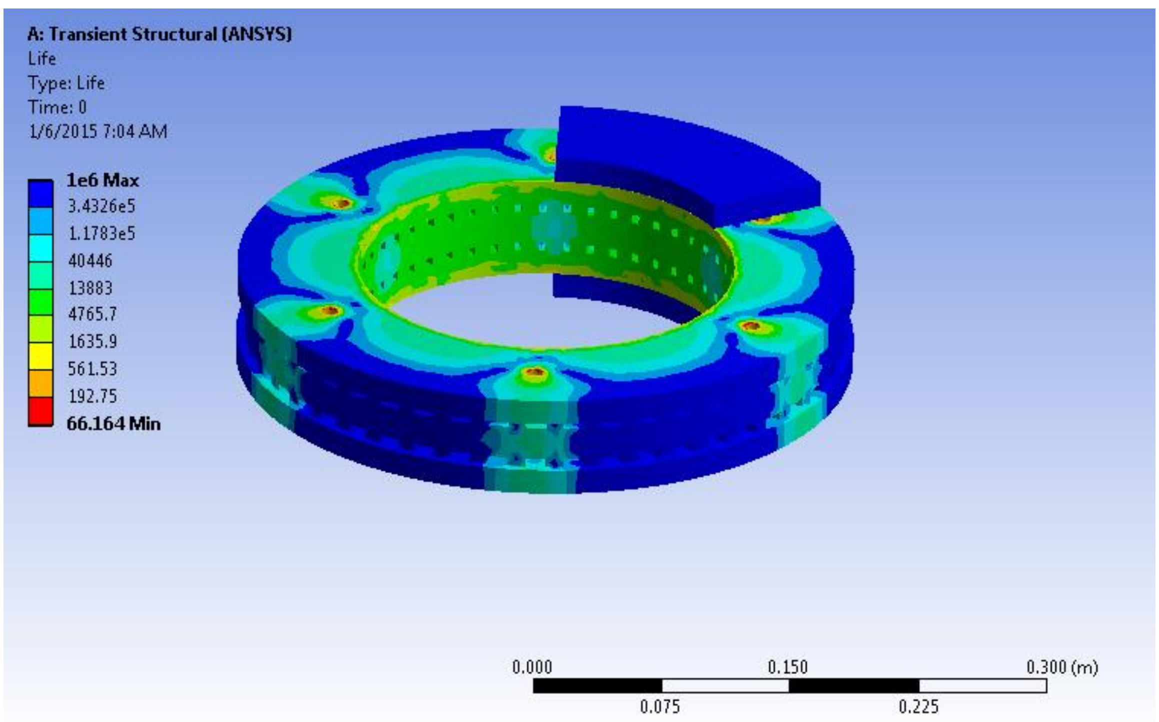


Fig 5.23 Life time at V2 and M2

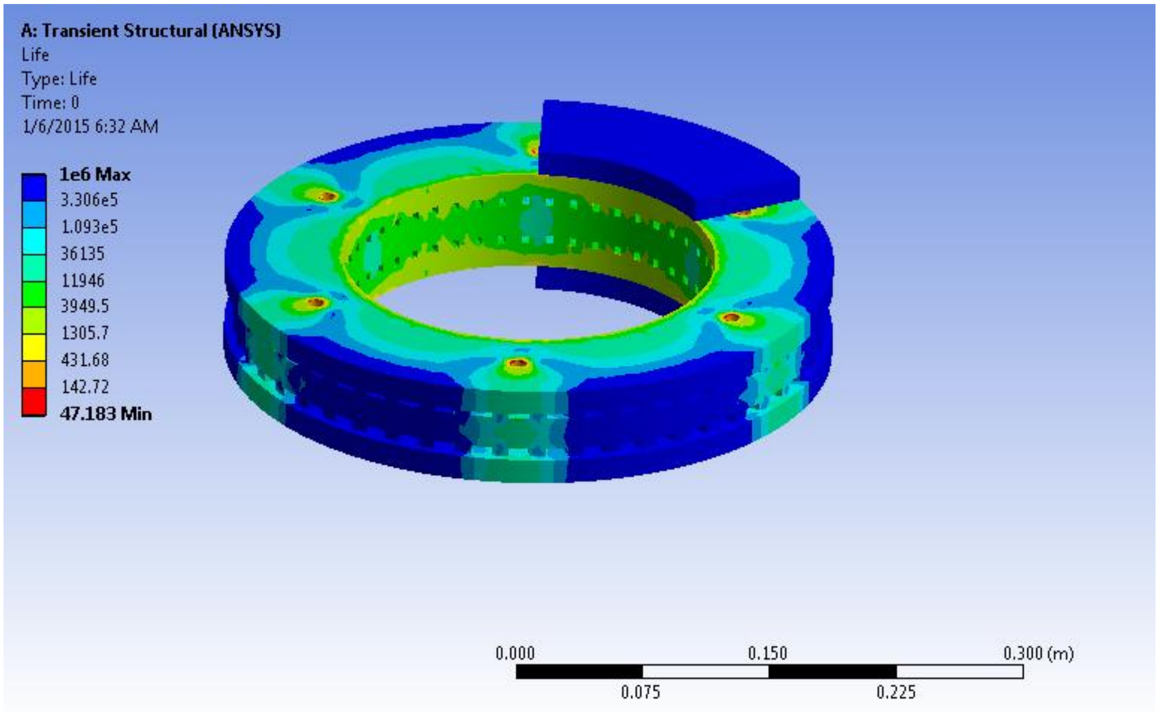


Fig 5.24 Life time at V2 and M3

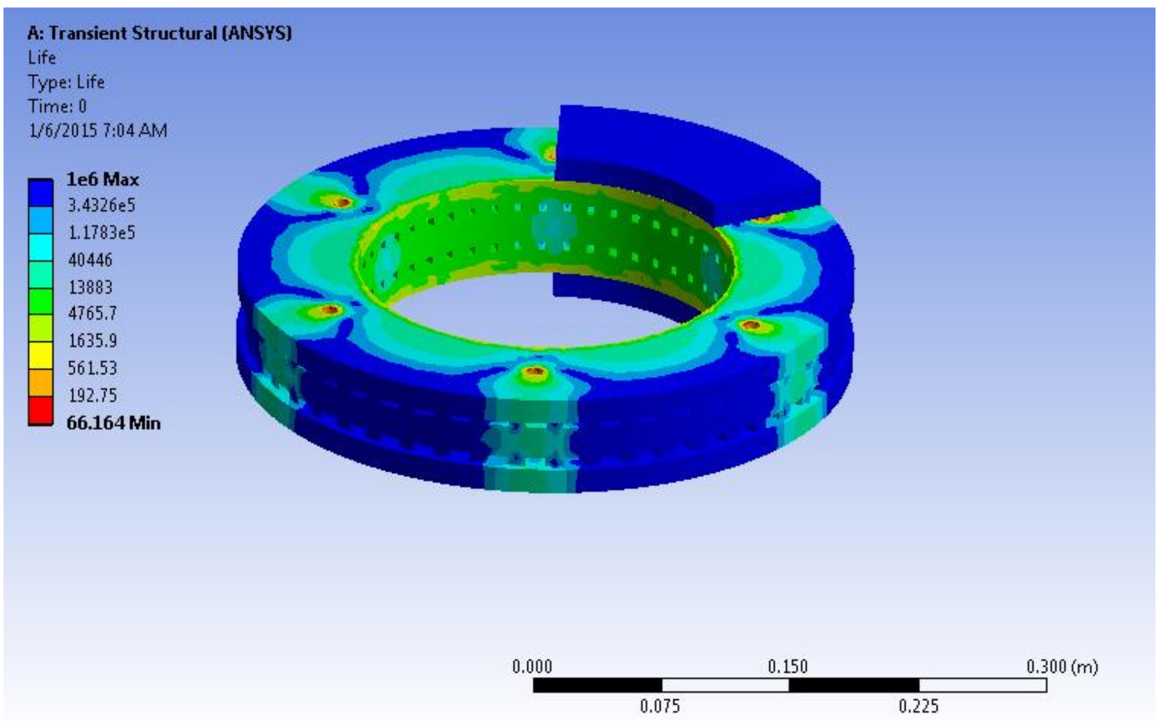


Fig 5.25 Life time at V3 and M1

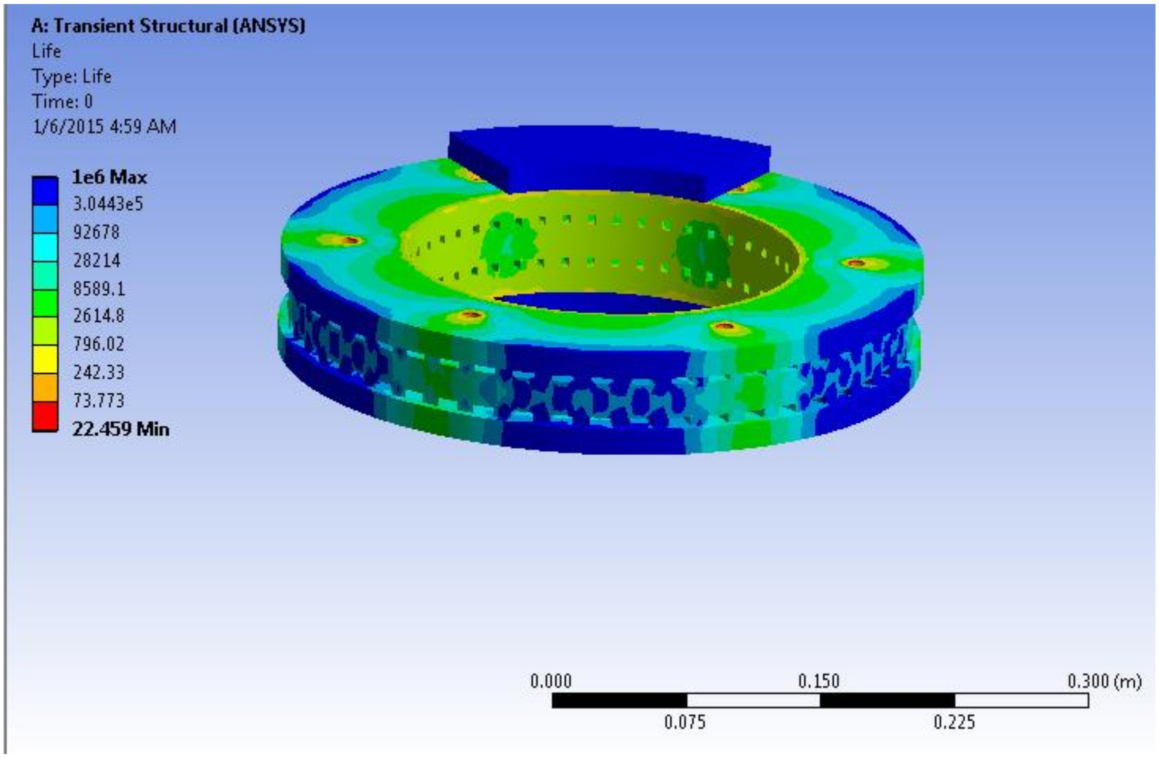


Fig 5.26 Life time at V3 and M2

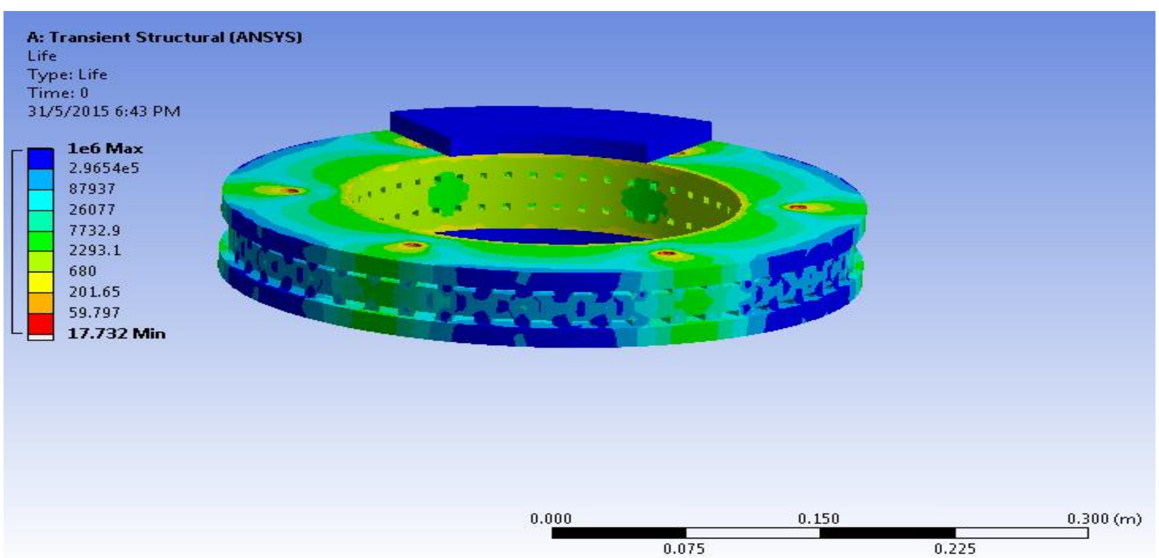


Fig 5.27 Life time at V3 and M3

Table 12.minimum life time Results

Velocity (m/s)	Vehicle mass(Kg)	Life time(month)
11.111	44000	437.82
	59240	317.32
	63020	140.49
19.444	44000	77.785
	59240	66.164
	63020	47.183
22.342	44000	66.264
	59240	22.459
	63020	17.732

5 .4 DISCUSIONS

When the train is engaging braking, disc rotor is contacting brake pads. Due to the heat generation of the rubbing surface, a clear hot annular broadband of heat accumulation is observed in the contact region. The temperature distribution can be observed in the figure 5.1 to 5.9 above.

From figure 5.1 to 5.9 shows the temperature distribution of the disc brake at different speed and loading conditions. As vehicle loading and speed increases the temperature increases. The maximum temperature distribution was occurred when the vehicle was moving with load of 63020 kg and speed 22.342 m/s and its value was 526.94K.In all cases as vehicle speed and load of the vehicle increases the temperature distribution increases.

At it described in the [23] when the nodal temperature between disc and pad exceeds 800 k, severe and sudden wear of pads occurs. However the maximum temperature at station EW 1 which is the maximum gradient about 1.905° during emergency brake of 526.94K which is much less than the limit value. In general as it is shown using transient thermal analysis the performance of this Addis Ababa LRT disc brake system is good.

From figure 5.10 to 5.19 shows the maximum Equivalent (von-misses) stress and Maximum Total Deformation of the disc brake at different speed and loading conditions. Maximum Equivalent (von-misses) stress and Maximum Total Deformation was happened when the vehicle moving at loading condition of 63020 kg and speed of 22.342 $^{m}{}_{s}$ and have the result of 751.67 MPa and 11.447 e-5 m respectively. For a constant speed of vehicle as the load increases both the Equivalent stress and Total deformation increases. But if speed of the vehicle increases both Equivalent stress and Total deformation increases for all loads.

At it described in the [23] the maximum limit of stress for gray cast iron is 820 MPa. However the maximum stress at station EW 1 which is the maximum gradient about 1.905° during emergency brake of 751.67 MPa which is less than the limit value. In general as it is shown using structural analysis the performance of this Addis Ababa LRT disc brake system is good.

From figure 5.19 to 27 the life time of the disc brake was carried out at different loading conditions and vehicle speed. The shortest life of disc was happened when the vehicle was moving with load of 63020 kg and speed 22.342$^{m}{}_{s}$. When both vehicle load and speed increases the life time of the disc decreases due to high thermo-mechanical load applied on the disc during emergency braking.

CHAPTER SIX

CONCLUSIONS, RECOMMENDATIONS AND FURTHER WORK

6.1 CONCLUSIONS

In this thesis work, the thermo-mechanical analysis of disc brake under three different speed and load was observed using finite element analysis (FEA) work bench. CATIA software is also used for modeling the disc and pad parts and then the final assembly.

The life time of the disc at different loading conditions and vehicle speed was carried out.

The present study can provide a useful design tool and improve the brake performance of disk brake system. From the above figures and Table it can conclude that all the values obtained from the analysis are less than their allowable values. Hence the brake disk design is safe based on the strength and rigidity criteria. Even though the maximum temperature, stress and total deformation of the result was with in the design limit ,but moving a vehicle at high speed and load decreases the life time of the disc brake. Long life and short life time was happened when the train moving with speed of 11.111 m/s and 22. 342 m/s respectively.

6.2 RECOMMENDATIONS

From the above result as the mass and speed of the train increases the temperature, stress and deformation increases. So Minimizing the running speed and load of train is a solution for minimizing the temperature rise, stress and deformation of the disc brake.

If the vehicle was loading 59240 kg and 63020kg it is better to move with a speed of 11.11 m/s in order to have high disc life time . So it was recommended that if the vehicle load was 59240 and 63020kg it is better the vehicle to move below 22. 342 m/s since if it moves above 22. 342 m/s its life time was too short. So in order to use the disc brake for a long time it is better to move the vehicle with speed of 11.111m/s as compared 19.444 m/s and 22.342 m/s.

6.3 FURTHER WORK

In this thesis the thermo-mechanical analysis of gray cast iron disc brake was carried out by ANSIS work bench at different load and speed of vehicle. But for future the following points may be studied further in the disc brake.

- Validating the result experimentally
- Methods to reduce temperature gradient will be the future study
- Effect of wear between disc an pad using software's
- Thermo-mechanical analysis of the disc brake using different materials

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