Addis Ababa University
Addis Ababa Institute of Technology
School of Mechanical and Industrial Engineering
Graduate Program in Railway Engineering

THERMOMECHANICAL ANALYSIS OF C/SiC BRAKE DISC FOR AALRT BY FINITE ELEMENT METHOD

By

Tekle Asegehegn

A Thesis submitted to the School of Mechanical and Industrial Engineering of Addis Ababa University in partial fulfillment of the requirements for the Degree of Masters of Science in Mechanical Engineering

(Rolling stock Stream)

Advisor: Dr. Daniel Tilahun

June 2016

ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

POSTGRADUATE PROGRAM IN RAILWAY ENGINEERING

DESIGN OF C/SiC COMPOSITE BRAKE DISC FOR AALRT BY FINITE ELEMENT METHOD

By: Tekle Asegehegn

Approved by Board of Examiners

Birhanu Beshah (Dr.)	____________________	________________
Railway Center Head	Signature	Date
Daniel Tilahun (Dr.)	________________	_____________
Advisor	Signature	Date
Mr. Areaya A. (MSC)	____________________	________________
Internal examiner	Signature	Date
Mr. Haileleoul Sahle (MSC)	____________________	________________
External examiner	Signature	Date

I hereby declare that the work which is being presented in this thesis entitled **DESIGN OF C/SiC COMPOSITE BRAKE DISC FOR AALRT BY FINITE ELEMENT METHOD** is original work of my own, has not been presented for a degree of any other university and all the resource of materials used for this thesis are duly acknowledged.

________________________ ______________________

Tekle Asegehegn Date

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

________________________ ______________________

Dr. Daniel Tilahun (Advisor) Date

Addis Ababa

June 2016

Abstract

Gray cast iron is commonly used brake disc material with high density that increase fuel consumption. It also generates heat easily during braking due to its lower specific heat capacity which affects its mechanical properties. The ever increasing demand for high speed trains from passengers and reduction of maintenance costs by operators means a compelling need to develop new disc brake materials with higher friction performance and longer service life. An interesting alternative are C/SiC composite materials characterized by lower wear rate and higher resistance to thermal shock. During braking kinetic energy transforms in to thermal energy resulting to intense heat and high temperature in the brake disc-pad interface. Thus, induced thermal loads determine thermo-elastic behavior of the railway disc brake structure.

This paper is mainly concerned with design analysis of C/SiC composite material to study and evaluate the performance under severe braking conditions and there by assist in brake disc design and analysis. Geometric dimensions of AA LRT train disc are taken on to CATIA where the 3D model is imported to ANSYS for determining the temperature distribution, variation of stresses and deformation produced in the disc brake after applying the boundary conditions. The main boundary and initial condition are the heat flux on the braking surface of the disc and the force of the brake clamps. Two different disc designs are used, one solid and other the ventilated existing one to demonstrate the material response for each variant. The aim is to investigate the structural deformation of the brake disc due to combined effect of thermal expansion when subjected to temperature change during the braking cycle and thereby assist the railway industry in developing optimum and effective disc brake material. The results were found to be satisfactory.

Key words: Brake disc, stress, heat flux, C/SiC composite, reinforcement fibers, transient response, friction coefficient, contact pressure.

Contents

List of Figures 2

List of Tables 3

Acronyms and Notations 4

CHAPTER ONE 7

INTRODUCTION 7

1.1 Background 7

1.2 Statement of the problem 9

1.3 Objective of the research 10

1.3.1 General objective 10

1.3.2 Specific objective 10

1.4 Research Methodology 10

1.5 Significance of the study 11

1.6 Scope of the study 11

CHAPTER TWO 12

LITERATURE REVIEW 12

2.1 Classification of brakes 22

2.2 Disc brake 23

2.2.1 Swinging Caliper Disc Brakes 24

2.2.2 Sliding Caliper Disk Brake: 24

2.3 Materials for Brake Disc and Brake Pad 27

2.3.1 Rotor Disc 27

2.3.2 Candidate Materials for Brake Disc 27

CHAPTER THREE 34

MODEL PREPARATION OF BRAKE DISC 34

3.1 Disc Brake 34

3.1.1 Ventilated Brake Disc 34

3.1.2 Brake Pads 35

3.2 Data Collection, Analysis and Modeling 36

3.2.1 Analytical and numerical Approaches to the Railway Disc Braking Condition 36

3.3 Loading capacity of the rail vehicle 39

3.3.1 Main technical parameters 40

3.3.2 Braking Time and Angular Velocity 40
3.4 Load Determination 42
3.4.1 Load Application 42
3.4.2 Braking Energy of the vehicle 43
3.5 Heat Energy 46
3.5.1 Work of Friction Force 46
3.5.2 Heat Flux Entering the Disc 47
3.6 Calculations in the Railway Disc Brake 53
3.6.1 Determination of the Load Conditions 54
3.6.2 Weight Transfer during Braking 54
3.6.3 Determination of the Physical Model 54
3.6.4 Determination of Friction Surface of Disc 56
3.6.5 Determination of Heat Flux 56
3.6.6 Pressure determination for the brake caliper 58
3.6.7 Centrifugal Load Determination 58
CHAPTER FOUR 60
FEA ANALYSIS OF AA LRT BRAKE DISC 60
4.1 Brake Disc Modeling 60
4.1.1 Geometric Modeling 61
4.2 Assumptions applied for development of physical model of the Brake Disc 62
4.3 Finite element analysis 63
4.3.1 Introduction 63
4.3.2 Definition of Material 65
4.3.3 Geometries and Dimensions of Brake Disk-pad 66
4.4 Preparation of the Geometry and FEM Discretization 66
4.4.1 Methods of Finite Element Analysis 66
4.4.2 Coupled-Field Analyses and Methods 68
4.4.3 ANSYS ICEM CFD Modeling and Mesh Preparation 69
4.4.4 Meshing and Loading Conditions of the Disc 70
4.4.5 Modeling in ANSYS CFD 70
4.4.6 Mesh generation 71
4.5 Definition of the Loads and Boundary Conditions 73
4.6 Thermal Load 76

4.7 Analysis of the stress 80
CHAPTER FIVE 86
CONCLUSION, RECOMMENDATION AND FUTURE WORK 86
5.1 CONCLUSION 86
5.2 RECOMMENDATION 87
5.3 FUTURE WORK 88
REFERENCES 89

List of Figures

Figure 1 a) Cross-section of a simplified disc brake with a fixed caliper. The wheel is attached to the axle flange. (b) Assembling of drum brake 23
Figure 2 Brake force applied equally to both sides of the brake disc 24
Figure 3 Schematic diagram of a brake disc/pad assembly with accessories 27
Figure 4 Structure of the composite brake disc: (a) SEM micrograph of the SiC layer on the C/C composite base and (b) schematic representation of the composite disc. 31
Figure 5 Centrifugal load when travelling on a straight tack. 42
Figure 6 Heat input during braking to a standstill from maximum velocity. 42
Figure 7 Heat input during braking on a downhill tack for maintaining constant velocity and braking to a standstill afterwards. 43
Figure 8 Simplified 60 vane ventilated brake disc geometry 66
Figure 9 3D Geometry of the Disc/Pad assembly in CATIA. 70
Figure 10 Meshed model of the brake disc box enclosure, nodes 50705 and elements 240466.. 71
Figure 11 Meshed model of the brake disc/pad assembly 72
Figure 12 Fluid domain enclosure box of the brake disc CFD CFX fluid flow model 74
Figure 13 Wall heat transfer coefficient distribution on the ventilated disc brake for transient analysis 76
Figure 14 Imported convection coefficient from CFX fluid flow for thermal analysis. 77
Figure 15 Heat flux application for emergency braking from maximum velocity to standstill on a down gradient track after maintaining a constant velocity. 77
Figure 16 Meshed model of the ventilated brake disc 78
Figure 17 Directional heat flux for emergency braking from maximum velocity to standstill on a down gradient track after maintaining a constant velocity. 78
Figure 18 Total heat flux for emergency braking from maximum velocity to standstill on a down gradient track after maintaining a constant velocity. 79
Figure 19 Coupled fluid flow (CFX), transient thermal and structural interaction 79
Figure 20 Temperature distribution of the C/SiC composite ventilated brake disc 80

Figure 21 Load application for emergency braking from maximum velocity to standstill on a down gradient track. 80
Figure 22 Pressure load application for emergency braking from maximum velocity to standstill on a down gradient track. 81
Figure 23 Loads application for emergency braking from maximum velocity to standstill on a down gradient track. 81
Figure 24 Imported body temperature of the ventilated brake disc for the structural analysis 82
Figure 25 Transient temperature of the solid brake disc for structural analysis 82
Figure 26 Equivalent (Von mises stress) of the SiC composite solid brake disc 83
Figure 27 Equivalent (Von mises stress) of the SiC composite ventilated brake disc 83
Figure 28 Directional deformation of the C/SiC composite ventilated brake disc in emergency braking 84
Figure 29 Total deformation of the C/SiC composite ventilated brake disc in emergency braking 84

List of Tables

Table 1 Mechanical properties of some candidate materials for brake disc 35
Table 2 Addis Ababa LRT E-W route station's gradients 36
Table 3 Addis Ababa LRT E-W line distance between adjacent stations 37
Table 4 Addis Ababa LRT N-S route station's gradients 38
Table 5 Number of passengers and weight 40
Table 6 Weights of vehicles 40
Table 7 Data for calculating the heat flux 44
Table 8 Material properties of the brake disc 55
Table 9 Material properties of the brake pad 55
Table 10 Applied heat flux values as function of time per area of the brake disc 58
Table 11 Material properties of the composite brake disc 65
Table 12 Boundary and initial conditions for CFX pre 74
Table 13 Solid disc domain 75
Table 14 Fluid domain 75
Table 15 ANSYS analysis results summary 84

Acronyms and Notations

AA LRT – Addis Ababa Light Rail Transit

AMC – Aluminum metal matrix composite

CAD – Computer aided design

CMC – Ceramic matrix composite

CFD – Computational fluid dynamics

CVI – Chemical vapor infiltration

RDD CVI – Rapid direction diffused chemical vapor infiltration

LPI – Liquid vapor infiltration

ERC – Ethiopian Railway Corporation

FEA – Finite Element Analysis

FEM – Finite Element Methods

TCR – Thermal Contact Resistance

DTV – Disc Thickness Variation

GCI – Gray cast iron

UIC - Union Internationale des Chemins de fer/International Union of Railways

ISO – International Standards Organization

DIN - Deutsches Institut für Normung/ German Institute for Standardization

EN – European Norm/ European Standard

I – Moment of Inertia of rotating parts

g – Gravitational acceleration

h_r – Heat transfer Coefficient

ε – Material surface emissivity

ν – Poisson's ratio

α – Linear thermal expansion

σ – Stefan Boltzmann's constant

E – Elastic modulus

V_o – Initial velocity

ω - Angular velocity

S_b – Braking distance

t_b – Braking time

E_k – Kinetic energy

E_p – Potential energy

r_w – Radius of the wheel

r_d – Radius of the disc

A_b – Area of the brake disc

A_p – Area of the brake pad

E_b – Total mechanical energy

Q_{gen} - Total generated heat

C_p – Specific capacity

F_c – Centrifugal force

F_{disc} – Friction force applied on the disc

v_{disc} – Velocity of the brake disc

T_{∞} - Ambient temperature

K_a – Thermal conductivity

R_e – Reynolds number

μ - Coefficient of friction

ρ – Material density of the brake disc

k – Thermal conductivity

h – Convective heat transfer coefficient

δ – Track gradient

$\frac{dt}{dx}$ – Temperature gradient

CHAPTER ONE

INTRODUCTION

1.1 Background

The railway transport system is one of the most crucial transport systems in the world with higher speed and higher axle loads, higher reliability and safety, large carrying capacity (volume), lower life cycle cost, satisfy environmental demands, higher availability and fewer disturbances compared with roadways. To satisfy such demands, it needs to have a railway transportation system with standard safety, comfort, speed, economy and reliability of the network. Train braking is a very complex process specific to rail vehicles and of great importance by its essential contribution on the safety of the traffic. This complexity results from the fact that during braking occur numerous phenomena of different kinds: mechanical, thermal, pneumatic, electrical, etc. The actions of these processes take place in various points of the vehicles and act on different parts of the train with varying intensities. The major problem is that all must favorably interact for the intended scope to provide efficient, correct and safe braking actions. The main purpose of braking action is to perform controlled reduction in velocity of the train, either to reach a certain lower speed or to stop at a fixed point in a reasonable amount of time by converting the kinetic energy and the potential energy (in case of circulation on slopes) of the train into mechanical work of braking forces which usually turns into frictional heat that dissipates into the environment. [1]

Most brakes use the principle of artificial frictional resistance applied to convert the kinetic and potential energy into heat energy. The brakes must therefore store and dissipate all this heat into the surroundings before subsequent braking stages to have a good braking efficiency. A brake system therefore has to be reliable in order to provide the operator with a better control.

Braking is thus one of the most important and safety critical components of automobile and rail vehicle motion control. For rail and automotive vehicles to operate safely, it is critical to provide a stable and effective brake system that can decelerate or stop the vehicle as required. To achieve adequate brake operation, acceptable temperatures must be maintained during all service conditions. Most of the researches on brakes are engaged on increasing the heat capacity dealing with the brake disc material and modifying the structure using some convection features to facilitate heat dissipation. Brake disc thermal capacity must be sufficient to ensure acceptable temperature rise. If temperatures are allowed to become too high, deterioration of the brake structural integrity and friction performance will take place. For repeated brake applications or drag braking, the brake disc must be able to dissipate the thermal energy that is generated. Adequate brake cooling is necessary to keep the brake temperature within a safe operating range. By design, friction brakes generate thermal energy and can develop very high temperatures. Sufficient brake cooling must be achieved since excessive thermal loading can result in brake fade, surface cracking, judder and high wear of the friction pair. High temperatures can also lead to overheating of brake fluid, seals and other components. The complex design requirements are difficult to satisfy and extensive research, development and testing are required. [2]

Nowadays, with growth of population and increasing trade practices there is high demand of railway transportation services in the world including Ethiopia for a long and medium distance transportation of passenger and goods (freight) due to its low negative environmental impact and higher tonnage capacity. There is a worldwide tendency to improve train vehicles transport performance to develop faster, bigger and more maneuverable machines capable of handling ever higher speeds demand needs efficient brake control mechanism taking traffic safety in to account. Braking technology for such vehicles of today and tomorrow needs to be equally efficient, low maintenance, cost effective and convenient.

Beginning from 2010, Ethiopia is undertaking huge infrastructure projects of a 5000km long standard gauge national railway network radiating from the capital Addis Ababa to be constructed in two phases. Moreover, AA LRT has commenced operation meant to alleviate rampant mass transportation problems in the city with laid out plans for route expansion in the second phase.

A friction material in the form brake pads is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc to stop or slow down the moving disc. The ability of a braking system to provide safe, repeatable stopping action is the key to safe motor vehicle control. To stop rotation of the axle friction material in the form of brake pads mounted on a device called brake caliper is forced hydraulically or pneumatically (hydraulic pressure of 110kpa in case of AALRT) against both sides of the brake disc. The function of the friction brake pairs is to decelerate the railway vehicle by transforming kinetic energy of the vehicle to heat via friction.

Addis Ababa LRT utilizes a combination of an integrated set of braking systems namely regenerative braking, rheostat braking, magnetic track brake and disc brakes on each vehicle. For service brake it starts from regenerative braking to bring down the speed of the train from maximum of 70km/hr. to about 9 km/hr. and then the brake discs will be applied to stop the vehicle completely. But during emergency braking all available brakes are applied simultaneously to stop the vehicle with the shortest distance possible.

To dissipate the heat energy absorbed by brakes into the surrounding atmosphere and stop the rail vehicle, an efficient brake system should satisfy the following requirements.

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

1.2 Statement of the problem

Disc overheating and rapid wear are the main problems that cause a decrease in braking performance. It also causes judder and noise which impose discomfort for the passengers and causes a stress on the rail. These all have a combined undesired effect that rise the maintenance and lifecycle cost of the components. Thus, due to braking capacity limitation the speed of urban trains decreased significantly in order to stop the rail vehicle within a short distance possible safely while it has a smaller kinetic energy. Smaller speeds of urban trains result in poor customer satisfaction and can significantly reduce profit of railway companies. Increasing demand for faster trains with higher acceleration and deceleration demand high performance braking system.

Modelling of bulk thermal and structural effects was found to be most effective for shape optimization and prediction of global disc behavior, with more sophisticated analyses required to determine more detailed design limits. This paper will aim to investigate the coupled thermo-mechanical stress and deformation of a C/SiC ceramic matrix composite as an alternative brake disc rotor material using finite element analysis.

1.3 Objective of the research

1.3.1 General objective

The main objective of the present research is to develop a C/SiC matrix composite material brake disc for Addis Ababa LRT and investigation of thermal and structural deformation during a single braking cycle for maintaining a constant speed and braking to standstill afterwards on a downgrade rail track.

1.3.2 Specific objective

The primary specific objectives of this study are the following:

- To develop a solid and ventilated disc brake model to study the material response in both design variants.
- To perform brake disc fluid flow analysis using CFD CFX fluid flow to simulate the fluid solid interaction.
- To analyze the thermal and structural stresses of the C/SiC composite brake disc for AALRT using finite element method (FEM).
- To conduct transient thermal analysis to investigate temperature distribution across the disc using axis-symmetric elements.
- To conduct sequentially coupled fluid-thermal-structural stress analysis of the brake disc.

1.4 Research Methodology

Geometric 3D modeling was done on CATIA with dimensions taken from AA LRT rail vehicle. Analytical, FEM and CFD analysis of thermal and structural stress of C/SiC composite brake disc under stipulated loads and operating conditions was carried out using the parameters and functions defining thermal and structural stress characteristics to enable find optimum solution at early design stages. The necessary data for this study was gathered from Ethiopian Railways Corporation. Fluid structure interaction was studied using CFX fluid flow sequentially coupled with transient thermal and structural stress analysis to find out the temperature distribution and deformation of the brake disc subjected to severe braking conditions. Recommendation have been suggested to incorporate valuable design variations further at the end.

1.5 Significance of the study

Braking system in a railway vehicle is the main control mechanism to stop or slow down speed whenever the need arises which necessitates the need for safe and reliable brake system. The research will have the following benefits:

- Ability of the brake disc to dissipate more heat energy with less distortion will help manufacturers to produce higher speed vehicles
- The brake disc ability to absorb/dissipate excessive heat developed in its interface will eliminate the risk of failure of train control arising from reduction in braking efficiency.

Therefore, the study will help researchers in academics and industry to further improve design of better disc materials, geometry and improve understanding of ceramic matrix composites to come up with optimum solution for railway and automotive vehicles.

1.6 Scope of the study

The purpose of this research is to design a C/SiC composite brake disc for AA LRT. Thus, thermal and structural analysis for a single braking period was done taking air convection effects in to consideration. Brand new brake disc-pad friction pairs was assumed here while analysis of worn out pairs may require a different approach due to contact stresses and wear effects. The influence of fatigue stresses due to subsequent repeated braking action were not taken in to account.

CHAPTER TWO

LITERATURE REVIEW

This section presents literature survey of some studies pertaining to thermal and structural aspects of brake discs with a summary of important contributions of leading researchers.

The production of safe vehicle is one of the major problem facing automobile and locomotive manufacturing industries today. Since effective braking system is one of the determining factor in vehicle safety enhancement of the brake system's efficiency is a vital problem facing manufacturers. The most important units in the vehicles braking system are the brake discs because these components transform the control force from a brake calipers into a brake torque which is applied to a wheel. Disc brake is one of the most complicated vehicle units with respect to a mathematical description of the processes that take place in them during braking [3].

The braking performance of the train is one of the most important factors that affect the traffic safety and the brake is the key components related to the running safety of the railway vehicle. It was found that about 50% of the observed accident is caused by braking performance of the vehicle. It is mostly due to the thermal and mechanical failure of the brake equipment. The numerical simulation of the temperature field is the main parameter that causes thermal recession and thermal fatigue. The biggest difficulty lies on the analysis of the transient temperature and stress field during the braking process and the establishment of a simplified model of the thermal mechanical coupling together with better thermal and mechanical property of the material used. [4].

The standards associated with railway vehicles are complicated. There are different major railway vehicle's national and international standards development organizations such as International Standards Organization (ISO), International Electro technical Commission (IEC), International Union of Railways (UIC), American Railway Engineering and Maintenance of way Association (AREMA). These organization put forward stringent requirements in place to be met by railway vehicle manufacturers and operators to address potential loss and damage or safety incidents that would otherwise occur.

In recent years, many researches have been done on possibilities of predicting the thermal overload based on the characteristics of trains and railroads especially on long term braking on down grade railroads on which trains run. Having concluded their researches, the European Rail Research

Institute (ERRI) proposed a valuation procedure, with the ultimate goal of making a decision on taking special measures to prevent the occurrence of thermal overload. This implies the application of the regulation and alternating brake or other measures related to regime change and drive train formation. However, the proposed procedure which involves calculating the coefficient for the particular estimation does not provide an efficient and reliable use in all potential cases particularly those relating to the long term braking of the train at a very long fall with the changing slope. Therefore, the research for the improvement of the thermal overload assessment process continues [1].

Recent theoretical and experimental works [5, 6] have been developed to characterize the thermal parameters which govern the heat transfer at the vicinity of a sliding interface. In certain industrial applications, the solids are provided with a surface coating. A recent study has been carried out to analyze the effect of surface coating on the thermal behavior of a solid subjected to the friction process [7].

Increased thermal efficiency and the integrity of materials in high temperature environments is an essential requirement in modern engineering structures in automotive, railway, aerospace, nuclear, offshore, environmental and other industries.

Below are a synopsis of some prominent studies undertaken on disc brake design, analysis and performance.

Jiguang Chen and Fei Gao studied the thermo-elastic phenomenon occurring in disk brakes, the occupied heat conduction and elastic equations were conducted and solved with contact problems. The numerical simulation for the thermo-elastic behavior of the disc brake was obtained in the repeated brake condition. The computational results were presented for the distribution of heat flux and temperature on each friction surface between the contacting bodies. Thermo-elastic instability (TIE) phenomenon (the unstable growth of contact pressure and temperature) was investigated in their study and the influence of the material properties on the thermo-elastic behaviors (the maximum temperature on the friction surfaces) to facilitate the conceptual design of the disc brake system. Based on these numerical results, the thermo-elastic behaviors of the carbon-carbon composites with excellent mechanical properties were also discussed. The thermal distortion of a normally flat surface into a highly deformed state is called thermo-elastic transition. It sometimes occurs in a sequence of stable continuously related states as operating conditions change. At other times, however, the stable evolution behavior of the sliding system crosses a threshold where upon a sudden change of contact conditions occurs as the result of instability. This invokes a feedback loop that comprises the localized elevation of frictional heating, the resultant localized bulging, a localized pressure increases as the result of bulging, and further elevation of frictional heating as the result of the pressure increase.

When this process leads to an accelerated change of contact pressure distribution, the unexpected hot roughness of thermal distortion may grow unstably under some conditions resulting in local hot spots and leaving thermal cracks on the disc. This known as the thermo-elastic instability (TEI) phenomenon occurs more easily as the rotating speed of the disc increases. This region where the contact load is concentrated reaches very high temperatures which cause deterioration in braking performance. Moreover, in the course of their presence on the disc the passage of thermally distorted hot spots moving under the brake pads causes low frequency brake vibration. Their investigation aimed to study the given disc brake rotor of its stability and rigidity (for the thermal analysis and coupled structural analysis) was carried out on a given disc brake rotor and to investigate best combination of parameters of disc brake rotor like flange width, wall thickness and material there by a best combination was suggested. [8]

S. Koetniyom studied thermal stress analysis of automotive disc brakes to develop material properties of gray cast iron brake disc model of the Rover disc subjected to severe thermal cycles using the commercial package ABAQUS. One particular existing brake disc design for a medium passenger car was chosen for the investigation. Due to symmetry, the final model of the disc was a 20° segment of the brake disc and hub meshed using nearly three thousand 20 node solid elements with a quadratic interpolation function. In addition, experimental work was undertaken to derive the rotor material properties in tension and compression as a function of temperature. This data was used to generate suitable FE material model routines which accurately allow for the different temperature dependent yield properties of cast iron in tension and compression. Using the most accurate user developed material subroutine, the thermal response of the back-and front-vented disc designs were compared: the back-vented disc suffers lower thermal distortion but at the expense of higher plastic strain accumulation, particularly near the point of attachment of the vanes. The result indicated that temperatures increase non-uniformly with the braking time and the disc is subjected to maximum temperatures up to around 380 °C at the end of the brake application. Thermal stress result shows maximum von Mises elastic stresses at the neck was 273 MPa and near the inner fillet radius of the long vane 442 MPa due to the constraints applied to the free expansion of the rotor rubbing surfaces. If these stresses arc beyond the proportional limit, plastic strains would occur in the brake disc. [9]

Daniel Das A. studied **t**hermal and stress analysis of disc brake made from grey cast iron under specific loads (driving downhill and braking to standstill). The Finite Element Method was used to carry out the analysis. The analysis dealt with centrifugal load for two cases of braking, braking to a standstill on a flat surface and braking on a downhill track for maintaining constant speed and afterwards braking to a standstill. The main boundary condition in both cases was 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 purpose of the model and analysis was to determine the effect of thermal loads on the temperature field and the effect of centrifugal load in a specific brake disc. To determine the boundary conditions parts of car brake disc calculation were used.

The main goal of the analysis was to define a model for the thermal and centrifugal load. With this model all the necessary parameters (stresses as a consequence of thermal and centrifugal loads) defined by the maker had been calculated.

They observed and concluded from their study that, due to the brake heating up during braking the effect of the heat flux was high. Since the highest values of temperature fields occurred in the worn disc during braking downhill with later stopping on flat surface. They reached 598 °C within the time of simulated braking 136 s. The highest allowable temperatures in the brake pad and consequently, in the disc were 350 °C (long standing). The results of stress analyses show that by considering the centrifugal loads the stress increased by 10 to 20 MPa.

The highest stress value was 201 MPa and was also smaller than the permissible stress for the material of the disc. In all cases the effect of centrifugal loads on the disc was small in comparison to thermal loads. The highest values of stresses of concerned stress fields occurred during braking downhill with stopping on a flat surface due to the fact that the temperature difference between the warmest and coldest part is much higher than in case of braking on a flat surface only.

In order to reduce stresses and to improve construction of the disc brake, they recommended two improvements:

- Selection of another material with better mechanical properties,
- Modification of transition and radii, where stress concentrations occur. [7]

T.Hogskolan studied simulation of thermal fatigue stresses in a disc brake by taking as an input the heat flux produced from the friction between a disc and pad system for a number of repeated braking cycles. He used the finite element analysis (FEA) to determine the temperatures profile in the disc and to analyze the stresses for the repeated braking, which could be used to calculate the fatigue life of a disc. Sequentially coupled approach was used for thermomechanical problem and the problem was divided into two parts, heat analysis and thermal stress analysis. The heat analysis was obtained by including frictional heat and adopting an Eulerian approach. The thermal stress analysis, which was the main focus of his thesis, was followed using ABAQUS. The plasticity theory as background for stress analysis was discussed in detail and temperature independent material properties were considered throughout the thesis work. The linear kinematic hardening model with rate independent elastic-plastic plasticity was used for benchmark and real disc-pad model. The results of the benchmark model and the real model were observed to be similar in terms of plasticity theory. First brake application stays for 6 second, and the result shows maximum temperature of 220^{o}C at 2.5 seconds. After first brake application mean equivalent stress of 180Mpa was shown. [10]

Dufre´noy proposed a macro-structural model of the thermo-mechanical behavior of the disk brake, taking into account the real three-dimensional geometry of the disk– pad couple. Contact surface variations, distortions and wear were taken into account. Real body geometry and thermo-elastic-plastic modeling of the disk material were specially introduced. Such a model aims to give predictions of the thermal gradients varying with time and of the thermo-mechanical response of the components. Predictions of the temperature distributions are compared with experimental measurements obtained by thermographs and thermocouples. Such a model seems to be a suitable base for the study of the thermal dissipation and the thermo-mechanical behavior and for the introduction of local friction effects. [11]

Yu Liang, Jiang Yan-Li, Lu Sen-kai, Luo Kun, Ru Hong-qiang investigated thermal and structural analysis of vented and normal disc brake rotors. Steady state and transient responses have been conducted through the heat transfer analysis to predict the worst case scenario and temperature behaviors of disc brake rotor. In their study, finite element analysis approached has been conducted in order to identify the temperature distributions and behaviors of disc brake rotor in steady state and transient responses. ANSYS has been used as finite elements software to perform the thermal analysis on both responses. Both results have been compared for better justification. Thus both results provide better understanding on the thermal characteristic of disc brake rotor and assist the automotive industry in developing optimum and effective disc brake rotor. The heat generated on the surfaces of disc brake rotor when brake is applied. Materials of disc brake rotor usually are made from cast iron, spheroidal graphite cast iron or cast steel. It was chosen as a rotor material due to low cost of material and performs high thermal resistance. This type of material normally suit to normal passenger vehicle but not for high performance car.

Once brake pads contacts to rotating rotor, there will be huge amount of heat generated to stop or slow down the vehicle. The rotor temperature can exceed 350ºC for normal cars and 1500ºC for race cars. Disc brake rotor is a crucial part in the brake system where the main role of the rotor is to reduce the heat generated by dissipating all of the heat. In that case, ventilated disc brake rotor is much better than solid rotor where more airflow from the surrounding area to dissipate produced heat.

The internal vanes allow air to circulate between two friction surfaces of the rotors. Their study can provide a useful design tool and improve the brake performance of disc brake system. Hence the brake disc design was safe based on the strength and rigidity criteria. Comparing the different results obtained from analysis they concluded that ventilated type disc brake was the best possible for that specific application. [12]

Khong Keng Leng, AM34 studied transient thermal and structural analysis of the rotor disc of disc brake aimed at evaluating the performance of disc brake rotor of a LRT under severe braking conditions and there by assist in disc rotor design and analysis. An investigation into usage of new materials was required which improve braking efficiency and provide greater stability to vehicle. This investigation can be done using ANSYS software a dedicated finite element package used for determining the temperature distribution, variation of the stresses and deformation across the disc brake profile. In the their work, an attempt has been made to investigate the suitable hybrid composite material which is lighter than cast iron and has good young’s modulus, yield strength and density properties. High strength glass fiber composites have a promising friction and wear behavior as a disc brake rotor. The transient thermo-elastic analysis of disc brakes in repeated brake applications has been performed and the results were compared. The suitable material for the braking operation is S2 glass fiber and all the values obtained from the analysis were less than their allowable values. Hence the brake disc design was safe based on the strength and rigidity criteria. By identifying the true design features, the extended service life and long term stability was assured.

The investigation into usage of new materials is required which improves the braking efficiency and provide greater stability to vehicle. The suitable hybrid composite material which is lighter than cast iron and has good young’s modulus, yield strength and density properties. The low weight, the hardness, the stable characteristics also in case of high pressure and temperature, the resistance to thermal shock and the ductility provide long life time of the brake disc and avoid all problems resulting of loading which are typical for the classic grey cast iron brake discs.

The transient thermo-elastic analysis of disc brakes in repeated brake applications has been performed using ANSYS software applied to the thermo-elastic contact problem with frictional heat generation.

To obtain the simulation of thermo-elastic behavior appearing in disc brakes, the coupled heat conduction and elastic equations were solved with contact problems. The effects of the friction material properties on the contact ratio of friction surfaces has been examined and the larger influential properties were found to be the thermal expansion coefficient and the elastic modulus. They observed that the orthotropic disc brakes can provide better brake performance than the isotropic ones because of uniform and mild pressure distributions.

Their study can provide a useful design tool and improve the brake performance of disc brake system. They claimed that S2 glass fiber was the suitable material for the braking operation and all the values obtained from the analysis were less than their allowable values. Hence the brake disc design was safe based on the strength and rigidity criteria. [13]

Oder G., Reibenschuh M., Lerher T., Šraml M., Šamec B., Potrč I. Conducted a study to investigate the temperature fields and structural fields of a solid disc brake during short and emergency braking with four different materials. The finite element simulation for two-dimensional model was preferred due to the heat flux ratio constantly distributed in circumferential direction. It takes the value of temperature, friction contact power, nodal displacement and deformation for different pressure condition using analysis software with four materials namely cast iron, cast steel, aluminum and carbon fiber reinforced plastic. Presently the disc brakes are made up of cast iron and cast steel. With the value at the hand, the paper tried to determine the best suitable material for the brake drum with higher life span along with detailed parts drawings**.**

By observing the structural analysis and thermal analysis results using aluminum alloy and carbon reinforced polymer the stress values were within the permissible stress value. So using aluminum alloy and carbon reinforced polymer is safe for disc brake. By observing the frequency analysis, the vibrations were less for aluminum alloy than other two materials since its natural frequency is less. And also weight of the aluminum alloy reduced almost 3 times when compared with alloy steel and cast iron since its density is very less thereby mechanical efficiency will be increased. But the strength of carbon reinforced material is more than aluminum alloy since the thermal analysis of carbon reinforced material is also within permissible limits. By observing analysis results, carbon reinforced polymer is best material for disc brake. [4]

Ali Belhocine * and Mostefa Bouchetara studied the thermo-mechanical behavior of dry contact between the brake disc and pads during the braking phase. The simulation strategy was based on the computer code ANSYS11. The modeling of transient temperature field in the disc brake is actually used to identify the factor of geometric design of the disc to install the ventilation system in vehicles. The thermal/structural analysis is then used to couple the deformation established, the Von Mises stresses in the disc and the contact pressure distribution in pads.

At their conclusion remarks, the thermo-mechanical behavior analysis of the dry contact between the disc and brake pads during the braking process was presented. They have shown that ventilation system plays an important role in cooling discs and provides a good high temperature resistance.

The analysis results illustrated that, a temperature and stress fields in the braking process phase were fully coupled. The temperature, the von Mises stress and the total deformations of the disc and the contact pressure distribution in the pads increases as the thermal stresses are additional to mechanical stress which causes the crack propagation and fracture of the bowl and wear of the disc and brake pads. Regarding the calculation results, they were found to be satisfactory compared to the common earlier literature investigations. They then recommend thermo-mechanical problem of the disc brakes to be solved with an experimental study to validate the numerical results. [14]

Mijuca et al proposed the temperature distribution due to friction process necessitates a good knowledge of the contact parameters. The interface is always imperfect due to the roughness from a mechanical and thermal point of view. Nowadays, the finite element method is used regularly to obtain numerical solutions for heat transfer problems. The most common choice when using finite elements is the standard Galerkin formulation. [15]

Nakatsuji et al conducted a study on the initiation of hair-like cracks formed around small holes in the flanges of one-piece disks during overloading conditions. The study showed that thermally induced cyclic stress strongly affects the crack initiation in the brake disks. In order to show the crack initiation mechanism, the temperature distribution at the flange was measured first. The temperature distribution under overloading was analyzed by using the finite element method. Based on the experimental and calculated results, the crack initiation mechanism for one-piece brake disks at the very severe braking condition was explained. In addition, effective methods were suggested for reducing the initiation of tiny cracks around the holes. [16]

Miloš S. Milošević, D. S. investigated the temperature and thermal stress in ventilated disc/pad brake during single brake. The brake disc was decelerated at the initial speed with constant acceleration until the disc comes to a stop. The ventilated brake pad and disc brake assembly was built by a 3D model with a thermo-mechanical coupling boundary condition and multi body model technique. To verify the simulation results, an experimental investigation was carried out.

After the completion of their study, it was concluded that the non-axisymmetric contact pressure distribution in the friction interface of disc/pad from the distribution, it was found that the maximum contact pressure occurred at the entrance and descends towards the exit of friction region. The contact pressure distribution at the mid stage of braking showed that the maximum contact pressure lies at the center of contact region due to the thermo-mechanical coupling behavior.

Friction heating, thermal distortion, and elastic contact affects the contact pressure and temperature on the contact surface.

This thermo-mechanical coupling behavior due to a relative high sliding speed that exceeds the critical sliding speed can be unstable, leading to localized high temperature contact regions called “hot spots” on the sliding interface. This shows the temperature field with the distortional hot spot in the ventilated brake disc in the different stages of braking operation. The temperature distribution is not uniform field, since the frictional heat generated in the contact region and the rest region dissipates the heat by the convection and heat conduction of the disc. The maximum temperature occurred around the middle circle of the contact surface at the exit of contact region, since the middle circle region is where the maximum contact pressure region is generated.

Temperature in the surfaces of vanes is lower than the temperature in the work surface, because of the conductive behavior in the disc and higher convection in the vanes. The temperature field in the pad presents no uniformity characteristics, and presents approximately axis-symmetric at the end of braking, similar to the temperature field in the disc. Temperature field affects the thermal expansion and leads to variation of contact pressure distribution.

Due to the heat generation ratio decrease, conduction in the disc and convection on the surface of brake disc temperature field presents approximately axis-symmetric in the end stage of the braking operation. The node temperature on the work surface in the disc present fluctuation. The thermal strain and thermal stress distributions were in accordance with the temperature distribution. [1]

Noyes and Vickers predicted the temperature response on the rubbing surfaces of a brake disc, using the assumption of a uniform heat flux. The computational results were compared with the temperatures measured at the rotor surface on entry to and exit from the pad area. It was found that the measured temperature on exit from the pad was higher than the temperature calculated using the assumption of uniform heat flux by approximately 55°C. However, for many applications, the effect of this circumferential temperature variation, the so-called "rotating heat source" effect, can be ignored. [17]

J. G. Balotin studied analysis of the influence of temperature on the friction coefficient of friction materials. He tried to verify the behavior of the friction coefficient when the friction material is submitted to high temperature braking (stage also known as Fade). The experiments were performed in three distinct stages: bedding-in, characterization and fade. Three sets of twelve braking were performed for friction characterization. He found that the fade stages degrade the phenolic resins of the materials and affect the performance of the friction material.

The test results of his paper show that brake pads with different formulations have a variation on the performance of friction coefficient with braking at high temperatures (fade) that is decrease of coefficient of friction with increase of temperature which leads to brake fade. [18]

Babukanth and Vimal studied transient analysis of disk brake using ANSYS Software by choosing element type solid 90, a higher order version of the 3-D eight node thermal element (Solid 70). The element has 20 nodes with single degree of freedom, temperature, at each node. The 20-node elements have compatible temperature shape and are well suited to model curved boundaries. The 20-node thermal element is applicable to a 3-D static or transient thermal analysis. If the model containing this element is also to be analyzed structurally, the element should be replaced by the equivalent structural element (Solid 95). [19]

The literatures reviewed so far focused on the thermal and mechanical analysis, material property and disc brake configuration (solid or ventilated disc brake) to reduce the exceeding temperature at the frictional contact of the disc brake and pad during braking process. Because it is this temperature that primarily cause fatigue, wear, judder, deformation and stresses, therefore the research still proceeds to completely avoid the existing problem.

An overview of past researches showed that brake discs are mostly tested for thermal loads and their effects (thermal cracks, thermal deformation). The purpose of this model and analysis was to determine effect of thermal loads on temperature field and influence of centrifugal and axial clamping load in a C/SiC composite brake disc.

2.1 Classification of brakes

The mechanical brakes according to the direction of acting force may be divided into the following two groups:

- Radial brake
- Axial brake

Radial brake

In these type of brakes the force acting on the brakes drum is in radial direction. The radial brakes may be subdivided into external brakes and internal brakes.

Axial brake

In these brakes the force acting on the brake drum is only in the axial direction i.e. Disk brakes, cone brakes.

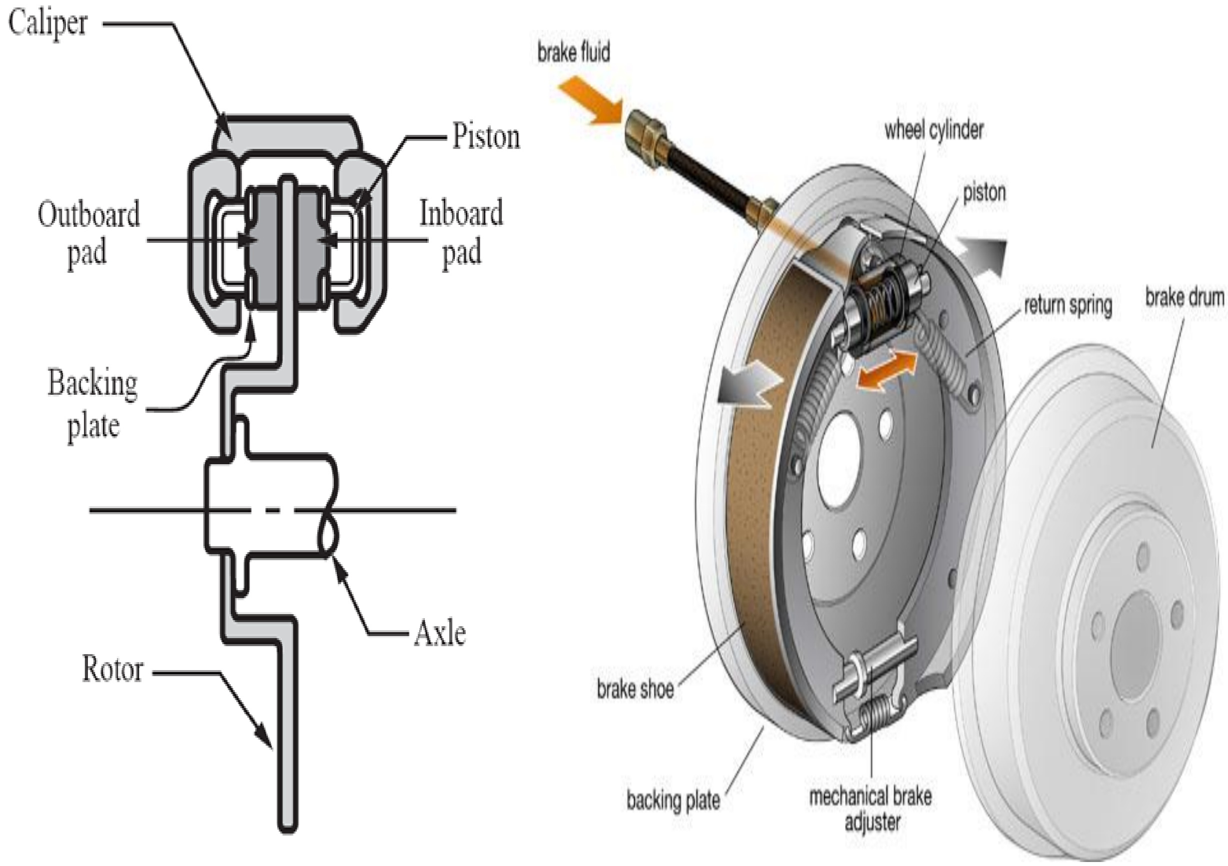


Figure 1 a) Cross-section of a simplified disc brake with a fixed caliper. The wheel is attached to the axle flange. (b) Assembling of drum brake

2.2 Disc brake

A brake disk consists of a cast iron, structural steel or composite disk bolted to the wheel hub 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 as is cast in two parts each part containing a piston. In between each piston and the disk 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. The passages are also connected to another one for bleeding. Each cylinder contains rubber-sealing ring between the cylinder and piston.

The brake disc is a brake system that slows rotation of the wheel by the friction caused by pushing brake pads against a brake disc with a set of calipers.

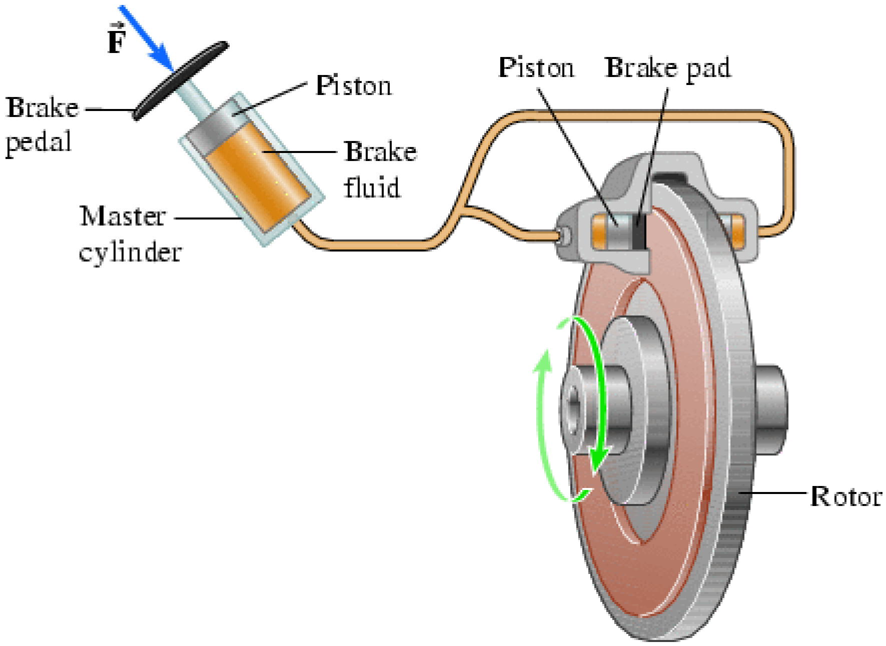


Figure 2 Brake force applied equally to both sides of the brake disc

When the brakes are applied, hydraulically or pneumatically actuated pistons move the friction pads in to contact with the rotating disk, applying equal and opposite forces on the disk. Due to the friction in between disk and pad surfaces, the kinetic energy of the rotating wheel is converted into heat which enables the vehicle to stop after a certain distance. On releasing the brakes the rubber-sealing ring acts as return spring and retracts the pistons and the friction pads move away from the disk.

2.2.1 Swinging Caliper Disc Brakes

The caliper is hinged about a fulcrum pin and one of the friction pads is fixed to the caliper. The fluid under pressure presses the other pad against the disc to apply the brake. The reaction on the caliper causes it to move the fixed pad inward slightly applying equal pressure to the other side of the pads. The caliper automatically adjusts its position by swinging about the pin.

2.2.2 Sliding Caliper Disk Brake:

There are two pistons between which the fluid under pressure is sent that press one friction pad directly on to the disc whereas the other pad is passed indirectly via the caliper.

Brakes have been retuned and improved ever since their invention. The increases in travelling speeds as well as the growing weights of vehicles have made these improvements essential. The faster a rail car goes and the heavier it is and the harder it is to stop. An effective braking system is needed to accomplish this task with challenging term where material need to be lighter than before and performance of the brakes must be improved [4].

2.2.2 Brake shield

Almost all car manufacturers today are using brake shield today, they're placed on the inside of the brake disc. Its main purpose is to protect the brake disc from all dirt and splash that comes up with the flow underneath the vehicle. These dirt particles will interfere with the contact patch between the brake disc and the brake pads and will further decrease the brake friction force. The unwanted dirt will also damage both the brake disc and the brake pads which will decrease the life time of the products.

Railway vehicles are the most popular and widely available way of transportation. Therefore, the train must be suitable, reliable and safe. The most important part of the vehicle is the braking system. Thermal and mechanical analysis is involved in almost every kind of physical processes and it can be the limiting factor for many processes. Therefore, its study is of vital importance and the need for powerful thermal and mechanical analysis tools is virtually universal. Furthermore, thermal and mechanical effects often appear together with or as a result of other physical phenomena [20].

Thermal and mechanical analysis is conducted to investigate how heat affects certain materials and engineering designs. This heat can come in the form of an environmental load such as an ambient temperature of a certain degree affecting a model, or due to friction in a system, effectively converting it into thermal energy. It can also come from processes of conduction through two solids, convection between a liquid and a solid, or radiation such as in space [4, 21].

Thermal and mechanical analysis is a great way to test a model before it is built and real world tested enabling costly reductions in time, material and improve safety of critical components.

Precise prediction of the maximum temperature is needed for the design of many systems as well as braking systems especially for both discs and linings. The thermal structural coupled analyses is an appropriate tool to study how to handle the high spinning speed of the brake disc.

Transient thermal analyses determine temperatures and other thermal quantities that vary over time. The variation of temperature distribution over time is of interest in many applications such as cooling of electronic packages or quenching analysis for heat treatment. Also of interest are the temperature distribution results in thermal stresses that can cause failure.

In such cases, the temperatures from a transient thermal analysis are used as inputs to a mechanical analysis for thermal stress evaluations. Thermal analysis is the primary stage in the study of braking systems because the temperature determines the thermo-mechanical behavior of the structure. In the braking phase, kinetic energy transforms into thermal energy resulting in intense heating of the railway brake discs. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks on brake discs and eventually fractures of the whole brake disc [22].

These unique characteristics provide the mechanical engineer with design opportunities not possible with conventional monolithic (unreinforced) materials. Composites are important materials that are now used widely not only in the aerospace industry but also in a large and increasing number of commercial mechanical engineering applications such as internal combustion engines, machine components, thermal control and electronic packaging, automobile, trains, aircraft structures and mechanical components such as brakes, drive shafts, flywheels, tanks and pressure vessels, dimensionally stable components and process industries equipment requiring resistance to high temperature corrosion, oxidation and wear, offshore and onshore oil exploration and production, marine structures, sports and leisure equipment and biomedical devices [20,23].

Latest electric trains usually decelerate almost to a stop depending only on electric brakes: the drive motors are used as generators, and the braking force is obtained by converting the train's kinetic energy into electric energy and consuming the generated electric energy in the train itself or using it somewhere else. The Addis Ababa LRT uses a combination of regenerative, dynamic, magnetic track and disc brake braking system, where the non-adhesion brakes decelerate the vehicle to a lower speed and the brake disc bring it to a complete stop. All systems can be simultaneously applied during an emergency brake. Since braking is of vital importance, railway vehicles must have a mechanical brake system capable of bringing them to a stop by itself.

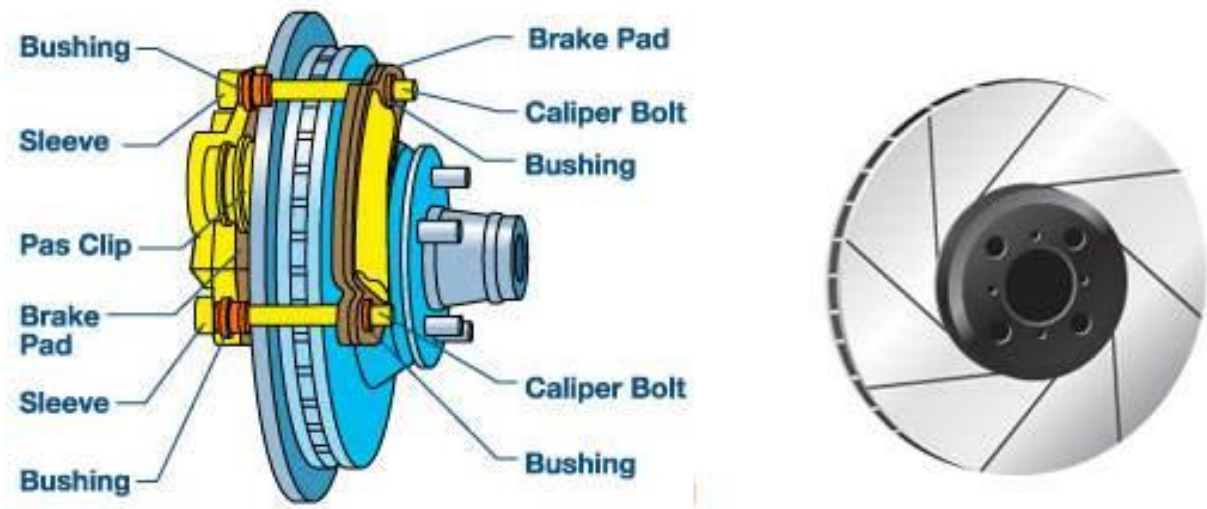


Figure 3 Schematic diagram of a brake disc/pad assembly with accessories

2.3 Materials for Brake Disc and Brake Pad

2.3.1 Rotor Disc

The braking system is a vital safety component of ground based transportation systems. Hence the structural materials used in brakes should possess some combination of properties such as good compressive strength, higher friction coefficient, wear resistant, light weight, good thermal capacity and economically viable [24].

2.3.2 Candidate Materials for Brake Disc

Traditional material for brake rotor is the cast iron. The specific gravity or density of cast iron is higher which consumes much fuel due to high inertia. Following section will describe some of the potential candidate materials that can be used for brake rotor application.

Cast Iron: Metallic iron containing more than 2% dissolved carbon within its matrix (as opposed to steel which contains less than 2%) but less than 4.5% is referred to as grey cast iron because of its characteristic color. Considering its cost, relative ease of manufacture and thermal stability this cast iron (particularly grey cast iron) is actually a more specialized material for brake applications particularly the material of choice for almost all automotive brake discs. To work correctly, the parts must be produced at the foundry with tightly monitored chemistry and cooling cycles, to control the shape, distribution and form of the precipitation of the excess carbon. This is done to minimize distortion in machining, provide good wear characteristics, damped vibration and resist cracking in subsequent use. [25]

Titanium alloys: Titanium alloys and their composites have the potential to reduce weight of the brake rotor disc component which is about 37% less than a conventional cast iron with the same dimensions and offering good high temperature strength and better resistance to corrosion. [26]

Aluminum-Metal Matrix Composite (Al-MMC): Aluminum alloy based metal matrix composites (MMCs) with ceramic particulate reinforcement have shown great promise for brake rotor applications. These materials having a lower density and higher thermal conductivity as compared to the conventionally used grey cast irons are expected to result in weight reduction of up to 50-60% in brake systems. The repeated braking of the AMC brake rotor lower the friction coefficient μ and cause significant wear of the brake pad. The friction properties of the AMC brake disc are thus remarkable poorer than those of conventional brake disc. After increasing hard particles content the result showed that the repeated braking operations did not lower the friction coefficient. Wilson et al. [27] studied the abrasive wear resistance of the Al6061 with 20 vol. % SiC reinforced composite in short sliding distance testing (about 20m).

Adding 20 vol. % SiC particulate greatly enhanced the wear resistance, raised room-temperature strength and stiffness, and improved high temperature strength [28]. Three major problems exist with this aluminum composite rotor. First, because of the density difference between aluminum and SiC, segregation or inhomogeneous distribution of SiC particles during solidification cannot be avoided. Also, adding SiC particles in an aluminum matrix dramatically reduces the ductility of the material, resulting in low product liability. The third problem is a lack of a solid lubricant, such as graphite. The lack of graphite in the system results in low braking efficiency, adhesive wear, and galling. In a cast iron rotor, graphite is always present in the iron. As the brake wears, the graphite is freed from the iron matrix to be used as a solid lubricant on the wear surface. An aluminum SiC and graphite composite brake rotor was developed and reported as having better wear resistance than a cast iron rotor. [29]

This material contains 10 vol. % SiC particulate and 5 vol. % nickel-coated graphite particulate reinforcement in an aluminum-silicon alloy matrix. These types of rotors were produced by casting. Although incorporating graphite particulate improved the wear resistance, the wear resistance and frictional performance of Al-Cu alloys reinforced with SiC particles are superior to those of cast iron brake rotors. [28]

Ceramic matrix composites (CMC): These materials have attracted a great deal of attention from researchers in many fields including aerospace, defense, energy, biomaterials and engineering among others in the recent past decades.

Ceramics are compounds between metallic and nonmetallic elements; they are most frequently oxides, nitrides and carbides. Some of the common ceramic materials include Carbon (C), aluminum oxide (Al_2O_3), silicon dioxide (or silica, SiO_2), silicon carbide (SiC), mullite (Al_2O_3–SiO_2) fibers and silicon nitride (Si_3N_4). With regard to mechanical behavior, ceramic materials are relatively stiff and strong comparable to metals. In addition, ceramics are typically very hard. On the other hand they are extremely brittle. Ceramics are very good insulators of heat and electricity and are more resistant to high temperatures and harsh environments than metals and polymers.

One of the most important properties of ceramics is that they are very good insulators of heat, which means they can hold very high temperatures without any failure. This fact is one of leading advantages of ceramic composite brake discs compared to the conventional brake discs made of grey cast iron. In the comparison experiment two test specimens, one made of grey cast iron and another made of ceramic–carbon composite are used.

A composite material is a material consisting of two or more physically and/or chemically distinct phases. The reinforcing component is distributed in the continuous or matrix component. Composites are classified into three categories depending on the matrix material, namely polymer matrix composites (PMCs), ceramic matrix composites (CMCs) and metal matrix composites (MMCs). MMCs have different property combinations and processing procedures as compared to either PMCs or CMCs. MMC remains a vastly used material for aerospace, automotive, medical, sports equipment and other engineering fields due to its several advantages.

They have a combination of superior properties to an unreinforced matrix such as increased strength, increased hardness, higher Young's modulus, improved wear resistance, high thermal conductivity, wear resistance, resistance to corrosion, processing flexibility and lower cost. [30]

Carbon/Carbon Composite (C/C)

There are other advanced materials for disc brake which have been in the commercial market for a few decades. The second principle used for gray cast iron disc brake requires a larger diameter of disc brake to fulfil the challenging requirement for modern luxury and sports cars' brake performance. An increase in the size of gray cast iron discs is also accompanied by an increased mass in the wheel suspension [31]. This has an impact on the weight of the car as this increases inertial forces. Weight efficiency is vital in Formula 1 race cars. The race car teams spend much time trying to have components of the absolute minimum weight because this allows them to redistribute weight around the car. Therefore there is an incentive to use weight-efficient materials wherever possible [32].

Carbon/Silicon Carbide Composite (C/SiC)

C/SiC brake was first developed by British engineers working in the railway industry in 1988. The advantages of C/SiC composites are similar to that of C/C composites, which include low density, good high-temperature resistance, high strength and low wear rate. Its thermal properties and mechanical properties are specifically tailored.

These composite disc brakes are able to reduce judder due to its low coefficient of thermal expansion and low wear. The low Young's modulus is thought to help promote uniform contact and reduces thermal DTV and hot spots [33]. C/SiC composites overcome the disadvantage of C/C composites which was an unstable coefficient of friction. In the early 1990s, at the German Aerospace Center in Stuttgart started researching on the development of C/SiC composites for high performance automobile applications. As of now, C/SiC brakes have been successfully embedded in Porsche, Ferrari, Daimler Chrysler and other high-end performance cars [34].

Carbon/Carbon-Silicon Carbide Composite (C/C-SiC)

Since the early 2000s, C/C-SiC composite has been gradually developed as advanced promising braking materials. Research at Stuttgart University and German Aerospace Center have developed C/C-SiC brake lining for 911 Turbo of Porsche . Other than that, it has been developed for clutch facings. This composite consists of a carbon/carbon-core material with a thin surface layer of SiC. In comparison with gray cast iron or carbon/carbon composite, this composite acquires better coefficient of friction (0.38) [35]. In addition, it also exhibits low wear rate.

Currently, the main preparation methods of C/C-SiC composite involves: (1) a gas route, which is also referred to as chemical vapour infiltration (CVI); (2) a liquid phase route, involving polymer impregnation/pyrolysis (PIP) and liquid silicon infiltration (LSI), also called reactive melt infiltration (RMI); (3) a ceramic route, which is a method to combine the impregnation of the reinforcement with a slurry and sintered at high temperature and high pressure.

The high cost is also a drawback for the RMI process. The carbon fibers of this composite are sensitive to the high pressure sintering process. Hence the preparation method has become an obstruction on the development of C/C-SiC braking composites. In order to overcome this, warm compaction and in situ reaction process (WCISR) have been developed [36]. As a result, C/C-SiC composites (fabricated by WCISR) have been effectively integrated into magnetic levitation vehicles, high speed trains and high-end performance cars.

Despite the fact that these available composite disc brakes are able to operate at substantially higher temperatures, this also leads to a need for heat shielding around the brakes.

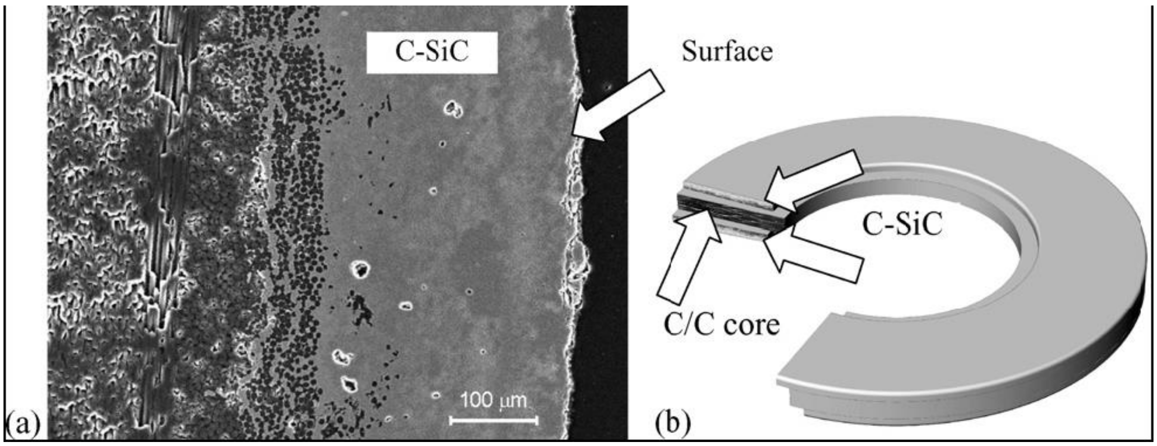


Figure 4 Structure of the composite brake disc: (a) SEM micrograph of the SiC layer on the C/C composite base and (b) schematic representation of the composite disc.

Carbon-Ceramic brake discs offer a tremendous weight advantage (about 50% -65%) over traditional grey-cast iron components, because of their low densities. This reduces the weight of unsprung masses (the mass that includes the weights of the suspension, wheels, wheel spindles, wheel bearings, tires, and a portion of the weight of driveshaft, springs, shock absorbers, and suspension links). In general, the speed and agility of a car will increase as its weight is reduced. The influence of weight is multiplied if a component is rotating. That means the weight save in unsprung mass is very crucial for a high-performance car. If weight can be saved in these areas, the benefits in terms of handling are immense; resulting in noticeably improved handling and agility and of course performance. [37]

Functionally graded materials (FGM)

FGM is a composite material with microscopic inhomogeneous character. The continuous changes in their microstructure result in gradients in the properties of FGM.

Functionally Graded Materials (FGMs) have demonstrated advantages beyond mechanical applications extending to electronic, optical, nuclear, biomedical, and other fields. Ceramic materials are excellent materials due to their high hardness, corrosion resistance and ability to operate under extreme conditions as high temperatures. The unique idea of an FGM is suitable combination of layers with different compositions will yield improved mechanical response with respect to that exhibited by each individual layer. [38]

The fabrication of the Functional Graded Materials ceramics composites involves two different processes; (a) Reaction-synthesis of Al_2O_3 and Al_2TiO_5 with graded interfaces and (b) heat-treatment of reaction synthesized Al_2O_3 and Al_2TiO_5.

Commercial rutile (TiO_2) and alumina (Al_2O_3) are used as base materials. Al_2TiO_5 is created by properly synthesizing the rutile and alumina ($Al_2O_3 + TiO_2 \rightarrow Al_2TiO_5$). MgO is added in as an addictive for improving the thermal stability of Al_2TiO_5, whereas SiO_2 is added in to stabilize its mechanical properties for Al_2TiO_5. [39]

Carbon–carbon composites are materials with high maximum operating temperatures which can be allowed to run much hotter than current designs and so lose significant amounts of heat by radiation as well as more moderate amounts by conduction/convection.

Carbon/carbon (C/C) composites exhibit not only high specific strength and modulus for making structural parts at high temperature but also their functional features such as thermal insulation, superconductivity and anti-friction characteristics. The application of C/C composites to the brake pad can increase the braking effect and decrease the weight of brake system and raise the service life. Therefore, many advanced commercial vehicles have employed C/C composite brake pads. Carbon/carbon (C/C) composites are prepared using rapid direct diffused chemical vapor infiltration (RDDCVI) processes. CVI technology is the most common method to prepare C/C composite brake pad and the discs made are more uniform and contain fewer defects in microstructure. However, the traditional CVI technology would often take over a thousand hours of furnace heat time to densify samples. A new rapid direction diffused chemical vapor infiltration (RDD CVI) method is developed in rapid densification of C/C composites but like other categories of C/C composites, brake pad characteristics of RDD CVI C/C composites are also much influenced by the processing and operating conditions. Therefore, research of the effect of these factors on brake features of C/C composites prepared using RDD CVI processes is significant in their fabrication and proper application as well as in the friction theory. [40]

The material used for the brake pad/lining should have the following characteristics.

- It should have high coefficient of friction with minimum fading. I.e. the coefficient of friction should remain constant with change in temperature.
- It should have low wear rate
- It should have high heat resistance
- It should have high heat dissipation capacity
- It should have adequate mechanical strength and
- It should not be affected by moisture and oil. [41]

Based on their mechanical properties, potential candidate materials for brake disc are selected as:

- Grey cast iron
- Ti-alloy
- 7.5 wt% WC and 7.5 wt% TiC reinforced Ti-composite
- 20% SiC reinforced Al-composite
- 20% SiC reinforced Al-Cu alloy
- 30%C reinforced SiC composite

CHAPTER THREE

MODEL PREPARATION OF BRAKE DISC

3.1 Disc Brake

Here heat transfer fundamental theories and numerical methods available used in brake analysis and performance prediction are discussed. This chapter deliberates on an understanding of the physical phenomena when a vehicle's kinetic and potential energy is converted to thermal energy through friction on the thermal and thermo-elastic aspects of disc brake operation.

There are two types of disc brake mounting. Wheel mounted disc brake and axle mounted disc brake. The wheel mounted disc brake design consists of two rings bolted on to the wheel hub, one on either side of the wheel. In the axle mounted type the disc brake is fixed on to the axle periphery. Brakes convert motion to heat and if the brakes get too hot, they become less effective a phenomenon known as brake fade. [42]

The friction surface is invariably displayed to the air checking good heat dissipation minimizing brake fade. It also allows for self-cleaning as dust and water is thrown off for diluting friction difference. Unlike drum brakes, disc brakes have limited self-energizing action making it necessary to apply greater pneumatic pressure to find enough braking force. This is achieved by increasing the size of the caliper clamp force. The simple design helps easy maintenance and pad replacement. [43]

Disc brakes could be solid or ventilated according to the required performance and condition of the braking. But in this research ventilated type of disc brake will be investigated.

3.1.1 Ventilated Brake Disc

There are different brake disc design forms available in practice. Some are simple solids and some are ventilated cast level but others are provided with holes and slots on outer surface with fins or vanes joining together the disc's two contact surfaces (usually included as part of a casting process) to allow faster brake cooling.

The ventilated disc is lighter than the solid disc and additional convective heat transfer occurs on the surface of the vent hall. Thus the ventilated disc can control its temperature rise and minimize the effects of thermal problems such as the variation of the pad friction coefficient, brake fade and wheel lock [44, 45].

The ventilated disc however, may increase judder problems by inducing an uneven temperature field around the disc. Also the thermal capacity of the ventilated disc is less than that of the solid disc and the temperature of the ventilated disc can rise relatively faster than that of the solid disc during repetitive braking [46]. Therefore, thermal capacity and thermal deformation should be carefully considered when modifying the shape of the ventilated disc.

3.1.2 Brake Pads

Different brake design applications require different kinds of friction materials. Several considerations are weighted in development of brake pads. The coefficient of friction must remain constant over a wide range of temperatures, the brake pads must not wear out rapidly nor should they wear the disc rotors, should withstand the highest temperatures without fading and it should be able to do all this without any noise. Therefore, the material should maximize the good points and minimize the negative points.

Table 1 Mechanical properties of some candidate materials for brake disc [28, 29]

Material	**Properties**				
Name	Compressive strength(*MPa*)	Friction Coefficient(μ)	Wear rate (10^{-6}mm^3 /Nm)	Specific heat C_p(*KJ/Kg.K*)	Specific gravity(*g*/cm^3)
GCI	1293	0.41	2.36	0.46	7.2
Ti-6Al-4V	1070	0.34	246.3	0.58	4.42
C/SiC	1204	0.34	1.9	0.70	1.9

In this study C/SiC matrix composite material is selected due to their superior physical and thermo-mechanical characteristics while extensive research on their manufacturing processes present production alternatives of low cost, light weight products.

3.2 Data Collection, Analysis and Modeling

In various research articles different modeling approaches for thermal and structural analysis of disc brakes can be found. The models range from one dimensional, two dimensional to too complex three dimensional analysis.

The three dimensional axis-symmetrical model of transient heat transfer has been chosen in this study for better accuracy. It should be noted that the geometry of a disc brake rotor is axis-symmetrical.

In this case the assumption of axis-symmetric heat flux on the friction surface means only time averaged boundary conditions over a rotor revolution is applied.

3.2.1 Analytical and numerical Approaches to the Railway Disc Braking Condition

This section presents the fundamental theories and laws applied in this research. In the process of braking of railway vehicles, it is necessary to define the model for thermal analysis that describes the heat transfer of the heat generation by friction at surfaces which are in contact between a railway brake disc and pads through the disc-pads interface as well as heat outflow of the whole braking system due to cooling effect of the surrounding air.

For these purposes, an analytical approach for modeling thermal and mechanical stress effects in the braking systems of railway vehicles will be considered.

Technical parameters and operating conditions of AA LRT assumed for the study are presented below.

Table 2 Addis Ababa LRT E-W route station's gradients [Source: Addis Ababa (E-W and N-S) route light rail transit project from ERC, September 2009]. [47]

No	Stations	Gradient	$\sin\delta$
1	EW1(Ayat)	0.435	0.0076
2	EW2(Meri)	0.894	0.0156
3	EW3(CMC)	-0.286	-0.005
4	EW4 (St. Michael)	-1.861	-0.0325
5	EW5(Civil service college)	-0.6588	-0.0115
6	EW6(Management institute)	-0.0974	-0.0017
7	EW7(Gurd shola 1)	0	0

8	EW8(Gurd shola 2)	-0.0859	-0.0015
9	EW9((Lem Hotel)	-0.802	-0.014
10	EW10(Haya Hulet 1)	-0.544	-0.0095
11	EW11(Haya Hulet 2)	-0.0593	-0.001
12	EW12(St. Urael)	1.528	0.0265
13	EW13(Bambis)	0.573	0.01
14	EW14(St. Estifanos)	1.088	0.019
15	EW15(Stadium)	0.802	0.014
16	EW16(Leghar)	-0.172	-0.003
17	EW17(Mexico)	-0.0859	-0.0015
18	EW18(Tegbared)	0.0859	0.0015
19	EW19(St. Lideta)	1.547	0.027
20	EW20(Coca Cola)	-1.547	-0.047
21	EW21(Tor hayloch)	1.747	0.0325

Table 3 Addis Ababa LRT E-W line distance between adjacent stations

No	Stations	Station distance
1	EW1(Ayat to Meri)	1.30km
2	EW2(Meri to CMC)	1.03km
3	EW3(CMC to St. Michael)	0.896km
4	EW4 (St. Michael to Civil service college)	0.865km
5	EW5(Civil service college to Management institute)	0.785km
6	EW6(Management institute to Gurd Shola 1)	0.906km
7	EW7(Gurd Shola 1 to Gurd Shola)	0.922km
8	EW8(Gurd Shola 2 to Lem Hotel)	1.004km
9	EW9((Lem Hotel to Haya Hulet 1)	1.035km
10	EW10(Haya Hulet 1 to Haya hulet 2)	0.838km
11	EW11(Haya Hulet 2 to St. Urael)	0.694km
12	EW12(St. Urael to Bambis)	0.875km

13	EW13(Bambis to St. Estifanos)	0.570km
14	EW14(St. Estifanos to Stadium)	0.569km
15	EW15(Stadium to Leghar)	0.700km
16	EW16(Leghar to Mexico)	0.865km
17	EW17(Mexico to Tegbared)	0.554km
18	EW18(Tegbared to St. Lideta)	0.936km
19	EW19(St. Lideta to Coca Cola)	0.848km
20	EW20(Coca Cola to Tor hayloch)	1.149km

Table 4 Addis Ababa LRT N-S route station's gradients [Source: Addis Ababa (E-W and N-S) route light rail transit project from ERC, September 2009]. [47]

No	Stations	Gradient	$\sin\delta$
1	NS6(Kality)	-0.716	-0.0125
2	NS7(Abo junction)	-0.802	-0.014
3	NS8(Saris)	-1.26	0.022
4	NS9(Adey Ababa)	-1.66	-0.029
5	NS10(Nefas silk 1)	-1.7756	-0.031
6	NS11(Nefas silk 2)	0.5156	0.009
7	NS12(Lancha)	-0.487	-0.0085
8	NS13(Temenja Yazh)	-1.432	-0.025
9	NS14(Meshwalekya)	-2.205	-0.0385
10	NS15(Stadium)	-0.401	-0.007
11	NS16(Leghar)	-0.172	0.014
12	NS17(Mexico)	-0.0859	-0.003
13	NS18(Tegbared)	0.0859	-0.0015
14	NS19(St.Lideta)	1.547	0.0015
15	NS20(Darmar)	-1.547	0.027
16	NS21(Abnet)	-2.163	-0.03776
17	NS22(Sebategna)	-3.641	-0.06364
18	NS23(Autobus Tera)	-1.856	-0.0324

19	NS24(Gojam Berenda)	0.138	0.0024
20	NS25(Atikilt Tera)	2.336	0.0408
21	NS26(Minilik II square)	0.4297	0.0075

According to Addis Ababa light rail transit (AALRT) office the maximum gradient should be ≤ 55 $^0/_{00}$ which is 0.055; slope = $\tan^{-1} 0.055=3.15°$. This study will consider this maximum gradient 3.153 ° as the other values are less than 3.15 °.

3.3 Loading capacity of the rail vehicle

Brake system for trams with 70% low floor incorporates advanced, well-proven and reliable products already installed for practical applications. Such brake system is suitable for mounting and operation on vehicles with 70% low floor. Train brake system complies with standard EN13452 for relevant level.

Brake system incorporates modular design with simple interface to meet requirements of modern brake system. It is equipped with electric brake, hydraulic disc brake and track brake, which are independent of each other.

Components are concentrated with sufficient spacing in-between for easy replacement and removal of components. Components should be replaced without removing adjacent units. Components requiring frequent maintenance should be mounted close to vehicles. Test connections and interfaces for hydraulic and electrical applications should be easily accessible. Their connection should not require removal of pipes or lines on the train.

Train brake system is microprocessor based analog hydroelectric brake system incorporating monitor terminals for self-diagnosis and failure recording. Such system can offer service brake (including blended brake for electric brake and hydraulic brake), emergency brake, safety brake, stopping brake, substitute brake in case of failure of electric brake and parking brake. Each bogie is equipped with an independent brake control device that will receive brake command from train control unit.

Service brake is applied by electric brake and hydraulic friction brake. Safety brake is applied by hydraulic friction brake and magnetic track brake. Emergency brake is applied by electric brake, hydraulic friction brake and magnetic track brake.

Brake rigging is disc brake, partially for parking brake. It can conveniently execute manual release of parking brake (including hand pump release and mechanical release).

3.3.1 Main technical parameters

Average brake deceleration (maximum vehicle operation speed: 70km/h) of vehicles from maximum operation speed to stop on straight and dry track under seating capacity.

Average deceleration for service brake: ≥1.1 m/s^2

Average deceleration for emergency brake: ≥2.0 m/s^2

Table 5 Number of passengers and weight [47]

Condition	No of passengers	Approximate car body weight (t)	Passenger weight (t)	Total(t)
AW1	0	44	0	44
AW2	64	44	3.84	47.84
AW3	254	44	15.24	59.24
AW4	317	44	19.02	63.02

Vehicle weight

Table 6 Weights of vehicles [47]

Loads	Passenger weight	Car body weight (t)	Total(t)
Empty vehicle(t)	0	44	44
Seating capacity(t)	15.24	44	59.24
Overload capacity(t)	19.02	44	63.02
Axle load(t)	≤11 （1+3%） t		

3.3.2 Braking Time and Angular Velocity

Brake application is defined as an application of the brake effort that results in a brake force being applied to the vehicle. Brake force is the force applied to the brake disc-pad braking surface interface. [48]

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

$$\boldsymbol{v(t)} = \boldsymbol{v_o} - \textbf{at} \quad \textbf{(3.1)}$$

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

$$\boldsymbol{\omega}(t) = \frac{\boldsymbol{v(t)}}{\boldsymbol{r_w}} \quad \textbf{(3.2)}$$

$$\boldsymbol{\omega_o} = \frac{\boldsymbol{v_o}}{\boldsymbol{r_w}} \quad \textbf{(3.3)}$$

$$\omega_o = \frac{19.44m/s}{0.33m} = 58.9224\ rev/s$$

Where,

v = velocity of the vehicle

v_o= Initial running velocity of the vehicle

ω_o= Initial angular velocity the wheel

ω = angular velocity of the wheel at any time

r_w= the radius of vehicle wheel

During the application of brake, the brake force acts at the effective radius (r_{disc}) of the disc rotor. The translational velocity (v_{disc}) of this frictional force (F_{disc}) could be expressed in terms of the translational velocity $\boldsymbol{v(t)}$ of the vehicle as follows.

$$v_{disc} = \frac{v_o}{r_w}\boldsymbol{v(t)} \quad \textbf{(3.4)}$$

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

$$S_b = v_o t_b - \frac{\mathbf{1}}{\mathbf{2}}\mathbf{a}\boldsymbol{t_b^2} \quad \textbf{(3.5)}$$

$$t_b = \frac{v_o}{\boldsymbol{a}} \quad \textbf{(3.6)}$$

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

$$\boldsymbol{\omega} = \omega_o\left(\mathbf{1} - \frac{\mathbf{1}}{\boldsymbol{t_b}}\right) \quad \textbf{(3.7)}$$

Where,

t_b = brake time required to stop the vehicle

S_b = total brake distance

3.4 Load Determination

Braking from the maximum velocity of 70km/h to a standstill with the temperature of the surrounding ambient air constant at 30 °C,

3.4.1 Load Application

a) Centrifugal Load at Constant Temperature of the Surrounding Air

The figure below represents application of the centrifugal load on the A.A. LRT vehicle when travelling on a straight track (TEGBARED to MEXICO a distance of 554m).

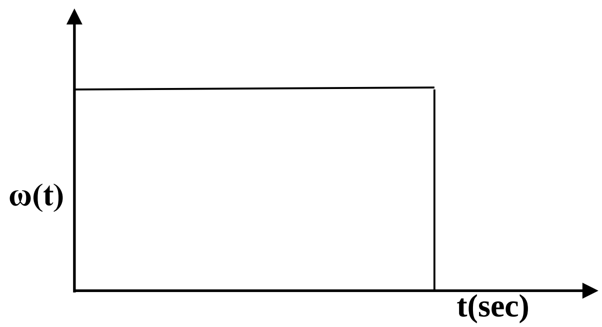


Figure 5 Centrifugal load when travelling on a straight tack.

b) Braking from Maximum Velocity to Standstill on a Flat Track

The figure below represents the application of the braking force on the AA LRT vehicle from maximum velocity *(70*km*⁄h*r*)* to standstill when travelling on a straight track (TEGBARED to MEXICO a distance of 554m).

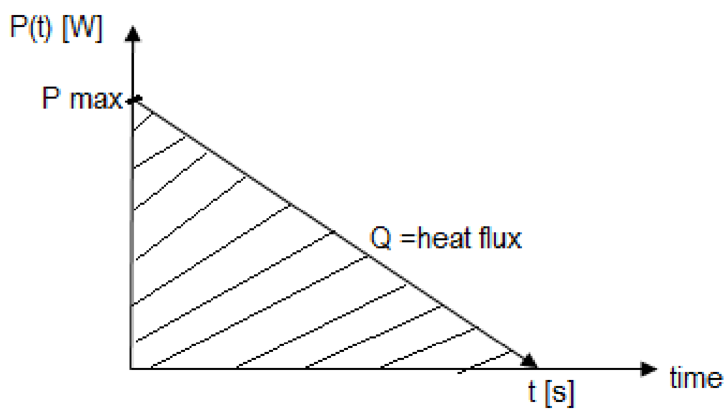


Figure 6 Heat input during braking to a standstill from maximum velocity.

c) **Braking on a downhill track for maintaining constant velocity and braking to standstill afterwards**.

The figure below represents the application of the braking force on the A.A. LRT vehicle to maintain a constant velocity when travelling on a downhill track from BAMBIS to ESTIFANOS a distance of 570m.

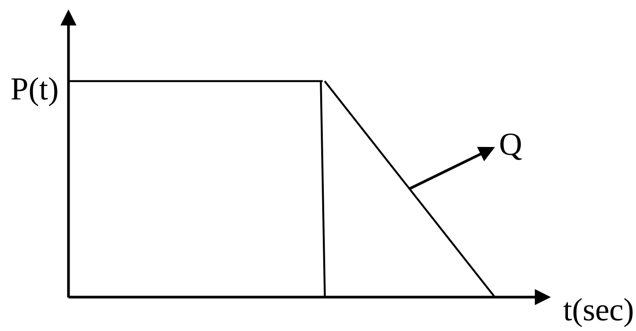


Figure 7 Heat input during braking on a downhill tack for maintaining constant velocity and braking to a standstill afterwards.

3.4.2 Braking Energy of the vehicle

Braking on a straight track

During the process of braking, available energy of the locomotive that must be transformed into frictional heat, need to be determined accurately. Both the kinetic as well as the potential (while running on slopes) energy of the locomotive that must be dissipated by the work of the braking should be considered. Rail vehicles have important masses in rotation. Therefore, the contribution of rotational kinetic energy is taken in to account. [49]

The initial kinetic energy imposed in to the locomotive is given by the sum total of translational and rotational.

$$E_k = \frac{1}{2}\mathbf{M}\boldsymbol{v_o^2} + \frac{1}{2}\mathbf{I}\boldsymbol{\omega_o^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \mathbf{(3.8)}$$

Where E_k= Initial Kinetic Energy of the locomotive just before braking starts

M= mass of the rolling stock

I= moment of inertia of rotating parts

ω_o= angular velocity of rotating parts

The moment of inertia of the rotating wheel set and discs can be calculated by using the equation below.

$$\mathbf{I} = \sum_{i=1}^{n} \boldsymbol{m_i\, r_i^2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \mathbf{(3.9)}$$

Where,

$\boldsymbol{m_i}$ -represents the masses of the wheel set and brake discs

$\boldsymbol{r_i}$ -represents the rotation radius of the rotating parts from the center of rotation

Therefore weight is multiplied by a factor which is taken as a 1.2, 1.1-1.15 and 1.03 for locomotive, EMU and freight train respectively. In most analysis the contribution of the rotating masses of the wheel set and discs is taken to be 10% the tare weight of the axle load of the locomotive.

$$\boldsymbol{E_k = \frac{1}{2}Mv_o^2 + \frac{1}{2}I\omega_o^2 = \frac{1}{2}Mv_o^2\left[1 + \frac{1}{mr_i^2}\right]} \text{ .... } \mathbf{(3.10)}$$

In some literatures the contribution of the rotating masses of the wheel set and discs is taken to be 10% the tare weight of the axle load of the vehicle [50]. The term $\left[\frac{1}{mr_d^2}\right]$ accounts to the rotational masses involved and its value equals 0.1 and it is adopted here.

$$\boldsymbol{E_k = \frac{1}{2}(1.1)\, Mv_o^2} \text{ .... } \mathbf{(3.11)}$$

Braking on a gradient track

The potential energy of the locomotive depends on the track gradient δ [mm/m] and on the travelled distance $\boldsymbol{S_b}$ [m]. The exact definition of gradient is $i = \tan\alpha$ where α is angle of inclination. According to EN 14531- 6 [50], for calculation of external forces that result from gradients in railway applications, the simplification sin α = tan α is commonly used.

Therefore the potential energy is expressed as:

$$\boldsymbol{E_p = M*g*S_b\,\frac{\delta}{1000}} \text{ .... } \mathbf{(3.12)}$$

Where, g = 9.81m/s^2

Table 7 Data for calculating the heat flux. [47]

Item	Values
Mass of the vehicle – M [kg]	63020
Start velocity – v_o[km/h] or [m/s]	70 or 19.44
Deceleration – a [m/s^2]	1.1 for service brake
	2 for emergency brake

Braking time – t_b[s], calculated from equation [3.6]	17.673 for service brake
	9.72 for emergency brake
Effective radius of the braking disc – r_d[m]	0.170
Radius of the wheel – r_w[m]	0.33
Slope of the track – δ [‰]	55
Friction coefficient of disc/pad – μ [/]	0.34
Surface of the braking pad A_p– [m^2]	0.035
Thickness of the disc Th-[mm]	60
Outer radius of the disc [mm]	230
Inner radius of the disc [mm]	110
Surface area of the braking disc A_b– [m^2]	0.139
Disc inner and outer radius [mm]	110 & 230
Pad inner and outer radius [mm]	150 & 230

The effective radius is the intermediate friction surface radius of the disc r_{disc}= 0.17m. The calculated contact surface area between the disc rotor and braking pad is A_b= 0.12889m^2. But the brake pad friction surface is given standard value. Therefore, the total available braking energy of the locomotive is determined by:

$$\boldsymbol{E_b = \frac{1}{2}(1.1)\, Mv_o^2 + M*g*S_b \frac{\delta}{1000}} \quad \textbf{(3.13)}$$

Depending on the gradient the force may tend to accelerate or decelerate the train. The inclination of the track also slightly reduces the normal reaction between the wheels and the rails the effect depend on the inclination angle α.

3.5 Heat Energy

In braking system, the mechanical energy is transformed into heat energy. This energy is characterized by a total heating of the disc and the pads during the braking phase. The energy dissipated in the form of heat can generate rise in temperature ranging from 300ºC to 800ºC.The heat quantity in contact area is the result of plastic micro-deformation generated by the friction forces between disc rotor and brake pad. The total heat generated ($\boldsymbol{Q_{gen}}$) in the brake system equals with the total mechanical energy ($\boldsymbol{E_b}$) lost from the locomotive. Hence, considering the energy balance;

$$\boldsymbol{Q_{gen} = E_b} \quad \textbf{(3.14)}$$

3.5.1 Work of Friction Force

When a locomotive begins to brake, the train losses power and will stop by frictional forces.

The imposed mechanical energy theoretically transforms to frictional heat. The brake disc rotor or wheel trade has got friction surfaces ($\boldsymbol{A_b}$) swept by the brake pads or block. Friction forces (F) act on the disc or tread at an effective radius. These forces retard the movement of the locomotive.

The brake disc rotor has got two friction surfaces ($2\boldsymbol{A_b}$) swept by the brake pads on both sides. Two friction forces ($2\boldsymbol{F_{disc}}$) act on both faces of the disc at an effective radius ($\boldsymbol{r_{disc}}$). These forces retard the movement of the train. The total heat flux is equal to:

$$\text{Total heat flux} = \frac{\boldsymbol{Q_{gen}}}{\boldsymbol{A_b}} = \frac{\boldsymbol{E_b}}{\boldsymbol{A_b}} \quad \textbf{(3.15)}$$

Where,

$\boldsymbol{A_b}$ = *is* the area swept by the brake pad at one face of the disc

$\boldsymbol{S_{disc}}$= distance traveled by $\boldsymbol{F_{disc}}$*F*disc during the whole braking time

The total work of friction force during the whole brake cycle equals with the total heat generated.

$$\boldsymbol{Q_{gen} = E_b = \int_0^{tb} p(t)d(t)} \quad \textbf{(3.16)}$$

$$\tfrac{1}{2}\mathbf{(1.1)\ M}\boldsymbol{v_o^2} + \mathbf{M*g*}\boldsymbol{S_b}\,\frac{\boldsymbol{\delta}}{\mathbf{1000}} = \int_0^{tb} \boldsymbol{p(t)d(t)}$$

$$\tfrac{1}{2}\mathbf{(1.1)\ M}\boldsymbol{v_o^2} + \mathbf{M*g*}\boldsymbol{S_b}\,\frac{\boldsymbol{\delta}}{\mathbf{1000}} = \int_0^{tb} \boldsymbol{p(t)d(t)}$$

$$\tfrac{1}{2}\mathbf{(1.1)\ M}\boldsymbol{v_o^2} + \mathbf{M*g*}\boldsymbol{S_b}\,\frac{\boldsymbol{\delta}}{\mathbf{1000}} = \mathbf{2}\boldsymbol{F_{disc}} \int_0^{tb} \boldsymbol{v_{disc}d(t)}$$

$$\tfrac{1}{2}\mathbf{(1.1)\ M}\boldsymbol{v_o^2} + \mathbf{M*g*}\boldsymbol{S_b}\,\frac{\boldsymbol{\delta}}{\mathbf{1000}} = \mathbf{2}\boldsymbol{F_{disc}}\,\frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}\left[\boldsymbol{v_o t_b} - \tfrac{1}{2}\,\mathbf{a}\boldsymbol{t_b^2}\right]$$

The friction force that works on the disc to retard the rail vehicle is:

$$\boldsymbol{F_{disc}} = \frac{\frac{1}{2}(\mathbf{1.1})\ \mathbf{M}\ \boldsymbol{v_o^2} + \mathbf{M}{*}\mathbf{g}{*}\boldsymbol{S_b}\frac{\boldsymbol{\delta}}{\mathbf{1000}}}{\mathbf{2}\frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}\left[\boldsymbol{v_o t_b} - \frac{\mathbf{1}}{\mathbf{2}}\,\mathbf{a}\boldsymbol{t_b^2}\right]} \quad \text{.. } \mathbf{(3.17)}$$

In the present work, considering the amount of heat generation by wear is very small relative to the heat generated by friction, so the effect of material wear is neglected. Therefore, using the value of the frictional force obtained above the transient heat flux can be determined.

The heat power generated at the effective radius ($\boldsymbol{r_{disc}}$) of the disc can be expressed as:

$$\dot{Q(t)} = \mathbf{2}\boldsymbol{F_{disc}}\ \boldsymbol{v_{disc}} = \mathbf{2}\boldsymbol{F_{disc}}\ \frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}[\boldsymbol{v_o} - \mathbf{at}] \text{..............................} \mathbf{(3.18)}$$

3.5.2 Heat Flux Entering the Disc

The analytical solutions given assume the heat flux varying linearly with time applied at each of its two faces of the disc geometry. Calculating heat loss from a component to the environment is affected by many variables. These relationships are functions of air velocity, surface finish, surface optical properties and temperature differences.

The thermal analysis of the braking system of railway vehicles requires determination of the quantity of heat produced by friction as well as the distribution of this energy between the railway vehicle disc and the braking pads.

Generally, the thermal conductivity of material of the brake pads is smaller than that of disc ($K_p < K_d$). We consider the energy from brake application is converted into heat and transferred to the disc and pad approximately 95% and 5% respectively [51]. This ratio normally is called the proportion of heat transferred to disc (γ = 0.95). The rate of heat generation is:

$$\dot{Q}_d(t) = \gamma \mathbf{2}\boldsymbol{F_{disc}}\ \boldsymbol{v_{disc}} = \mathbf{(0.95)2}\boldsymbol{F_{disc}}\ \frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}[\boldsymbol{v_o} - \mathbf{at}] \text{..............................} \mathbf{(3.19)}$$

a) **Heat conduction**

Though conduction is an effective mode of heat transfer it can have adverse effects on nearby components such as damaged seals, brake fluid vaporization and wheel bearing damage. Most of heat generated is characterized by the heating of the disc. The heat conduction in the disc is governed by the classic heat equation:

$$\boldsymbol{\rho C_p \dot{T}} = \frac{\mathbf{1}}{\boldsymbol{r}}\frac{\boldsymbol{\partial}}{\boldsymbol{\partial r}}\left(\mathbf{r}\boldsymbol{K_r}\frac{\boldsymbol{\partial T}}{\boldsymbol{\partial r}}\right) + \frac{\boldsymbol{\partial}}{\boldsymbol{\partial z}}\left(\mathbf{r}\boldsymbol{K_z}\frac{\boldsymbol{\partial T}}{\boldsymbol{\partial r}}\right) \text{..} \mathbf{(3.20)}$$

Where,

ρ -the density for material of the disc

K-represents the thermal conductivity,

$\boldsymbol{C_p}$- Specific heat capacity,

There are two paths of heat conduction from the discs, one through the bearing assembly (which should be avoided) and another through the wheel carrier, which is the major conductive path. Heat transfer within brake components obeys Fourier's Law of heat conduction as in Equation 3-21. This is based on the heat transfer rate (q), the applied area (A), the material thermal conductivity (K), and the temperature gradient at the direction of heat flow $\boldsymbol{\frac{dt}{dx}}$ [52] as follows**:**

$$\boldsymbol{Q_{cond} = - KA\frac{dt}{dx}} \quad \textbf{.. (3.21)}$$

The small area A and very low temperature difference (T_h -T_c) limits the amount of power dissipated by conduction. Therefore, the heat conduction can become negligible in brakes.

b) **Convection heat transfer**

The major aim of designing brake discs is to improve the convection dissipation capability of disc braking systems. In operations of braking systems, convection is the most important mode of heat transfer, dissipating the highest proportion of heat to surrounding air.

The quantity q_{conv} represents heat loss from the surface at temperature T by convection with a heat transfer coefficient h into an external ambient at a temperatureT_∞. Cooling from exposed surfaces follows Newton's Law of cooling as in Equation 3-22, where the rate of heat transfer (or cooling) from an exposed surface depends on the convection heat transfer coefficient (h), the area of the surface (A), and the temperature difference between the surface (T) and the surroundings (T_∞,). [53]

$$\boldsymbol{q_{conv} = \frac{q}{A} = h(\mathrm{T} - T_\infty)\, W/m^2} \quad \textbf{.. (3.22)}$$

From the above expression to maximize heat transfer from the rotor (increase $\boldsymbol{q_{conv}}$) and keep rotor temperature T to a minimum value, the value of heat transfer coefficient (h) or the surface area needs to be increased. Heat transfer coefficient h varies with the type of flow (laminar, turbulent, etc.), the geometry of the body and flow passage area, the physical properties of the fluid, the average temperature and many others.

The thermal resistance of a component ($R_{thermal}$) is obtained from the material thermal conductivity (k), length or thickness (Δx) and the component characteristic area (A), which depends on the direction of heat flow. [53]

$$R_{thermal} = \frac{\Delta x}{kA} \quad \text{……} \quad (3.23)$$

Heat transfer (cooling) coefficient (h)

The heat transfer coefficient depends on the type of air flow (laminar or turbulent) around the surfaces. This can be identified based on Reynolds number, and it is given from Equation 3-24 specifically for the rotor face of the disc brake [54]. Reynolds number depends on the velocity ($V_{rotor\ avg}$) and perimeter (L_p) of the rotor mean diameter and the kinematic viscosity of air ($\nu = \frac{\mu}{\rho}$).

$$R_e = \frac{V_{rotor\ avg} * L_p}{\nu} \quad \text{……} \quad (3.24)$$

For a laminar flow, $R_e < 2.4\ x\ 10^5$and Equation 3-25 [54] is applicable for the estimation of the heat transfer coefficient (h). The thermal conductivity of air is represented by K_a , and the rotor outer diameter by r_o.

$$h_r = 0.7.\left(\frac{K_a}{r_o}\right) R_e{}^{0.55} \quad \text{……} \quad (3.25)$$

For a turbulent flow, $R_e > 2.4\ x\ 10^5$and Equation 3-26 [54] is applicable for the estimation of heat transfer coefficient (h_r).

$$h_r = 0.04.\left(\frac{K_a}{r_o}\right) R_e{}^{0.8} \quad \text{……} \quad (3.26)$$

Thermal contact resistance (TCR) due to surface roughness

Thermal contact resistance can exist between two surfaces because the real contact area is just a small fraction of the apparent contact area [55, 56 and 57]. This is a result of the surface roughness of materials [56] which in many cases can only be seen on a microscopic scale. TCR is expected to reduce with increased contact pressure [58] as a result of the increase in real contact area at the microscopic level.

c) **Radiation heat transfer**

The quantity q_{rad} represents the heat loss from the surface by radiation in to an ambient at an effective temperature T, and is given by

$$q_{rad} = \varepsilon\sigma\ (T^4 - T_\infty^4) \quad \text{……} \quad (3.27)$$

At the contact surface between the railway disc and the braking pads, the braking process produces heat. The heat dissipation from the free surface of the disc to the surrounding air is described by both convection and radiation:

To develop the boundary condition, we consider the energy balance at the surface as
Heat supply = heal loss

Since all the friction generated energy has to be dissipated to reach thermal equilibrium in the disc,

$$Q_{diss} = q_{conv} + q_{rad}$$

$$Q_{diss} = -h(T - T_{\infty}) - \varepsilon\sigma(T^4 - T_{\infty}^4) \text{ .... } \mathbf{(3.28)}$$

Where,

h= the convective film coefficient,

ε = the material's surface emissivity,

σ= Stefan Boltzmann constant$[5.6697 * 10^{-8} W/m^{2}°C^{4}]$ and

T_{∞}=Temperature of the surrounding air

Convective heat transfer between the air and the brake rotor surfaces is the primary means of heat rejection. Radiation heat transfer is less important especially for the low surface temperatures and therefore its effect is neglected.

The evaluation of the convective heat transfer coefficient (h) is one of the most difficult tasks in the thermal analysis of a brake system. It varies over the brake surface and is a complex function of many factors such as geometry, air flow conditions, wheel angular velocity, surface temperature etc.

Heat Partition Coefficient and Energy Input

Microscopic model: It is where rate of frictional heat is equal to friction power. Some of this frictional heat is absorbed by the disc and the rest is absorbed by the pad. According to Majcherczak Dufrenoy and Nait-Abdelaziz (59) two kinds of thermal contact descriptions are usually used in analyses.

Perfect contact: It considers equal surface temperature of the disc and the pad.

Imperfect contact: Consider a heat resistance between the disc and the pad due to the formation of third body film (interface tribo layer) constituted by detached debris of particles.

As contact surfaces of the disc and pad is not the same, they don't have perfect contact. Majcherczak, Dufrenoy and Nait-Abdelaziz (59) focused on the thermal aspects of disc brakes in order to understand the way by which frictional heat is generated and obtain indications of the surface temperatures of brake discs and pads. Considering imperfect contact they presented a heat partition equation, which assumed constant heat flux partition.

$$\varphi_{disc} = \frac{\xi_d S_d}{\xi_d S_d + \xi_p S_p} \quad \dots \dots \dots (3.29)$$

Where φ_{disc} is the heat flux partition coefficient (to the disc side), $\xi_p,\ \xi_d$ are the thermal effusivity of the pad and disk which define the materials properties with regards to exchanging thermal energy with the surroundings and $S_p,\ S_d$ are the frictional contact surface area of the pad and disc respectively. Thermal effusivity is obtained from the following equation.

$$\xi = \sqrt{k\rho c} \quad \dots \dots \dots (3.30)$$

Where k = the thermal conductivity, ρ = density, c = specific heat of the brake material

Denoting Q_d and Q_p the heat quantities assumed by the disc and the braking pads respectively, the heat partition ratio can be expressed in the following manner

$$\frac{\mathrm{Qd}}{\mathrm{Qp}} = \frac{\sqrt{\rho\mathrm{d.Kd.Cd}}}{\sqrt{\rho\mathrm{p.Kp.Cp}}} \quad \dots \dots \dots (3.31)$$

To calculate the frictional heat generation at the contact zone of the two components of the brake system, the friction coefficient between two sliding components, relative sliding velocity, geometry of the disc and the pad and the pressure distribution at the sliding surfaces is required. Here we consider two assumptions for pressure distribution.

Macroscopic model: Brakes are generally a mechanism to convert the kinetic energy of vehicle moving with certain speed to thermal energy. The cooling of the brake dissipates the heat and the vehicle slows down. It obeys the first law of thermodynamics, sometimes known as the conservation energy which states that energy can't be created neither destroyed, it can only be converted from one form to another.

Pressure distribution:

Contact pressure between either a soft or hard contact is defined as a function of the over closure (negative clearance) of the surfaces. We will consider the two practical cases, a new pad and a worn-in pad. A new brake pad when subjected to a force through its center of area generates uniform pressure over the surface of the pad. But as the sliding velocity increases with radius the wear rate increases from the center out. After some operational time a state of uniform wear will be established, where pressure on the pads will decrease inversely with radius, to maintain a uniform wear over the interface.

- Uniform pressure: Pressure is assumed to be uniform over the surface area.

$P = P_{max}$

p = constant for new brakes

When the two brakes have perfect contact, the pressure p is uniform over the entire surface. The intensity of pressure becomes,

$$\boldsymbol{P = \frac{F}{\pi(r_o^2 - r_i^2)}} \text{ ………………………………………………………… } \mathbf{(3.32)}$$

Therefore, the total frictional torque, $T = \int_{r_i}^{r_o} 2\mu\pi p r^2 dr$

$$= 2\mu\pi p \left[\frac{r^3}{3}\right]_{r_i}^{r_o} = 2\mu\pi \frac{F}{\pi(r_o^2 - r_i^2)} * \left(\frac{r_o^3 - r_i^3}{3}\right)$$

$$= \frac{2}{3}\mu F \frac{(r_o^3 - r_i^3)}{(r_o^2 - r_i^2)}, \text{ Nm………………… } \mathbf{(3.33)}$$

- Uniform wear: After a certain amount of initial wear has taken place, the pressure distribution will change so as to permit the wear to be uniform. The intensity of pressure should vary inversely proportional to the elemental area. pr is constant for worn brakes. Below are the relations for the pressure and total frictional torque.

The constant wear rate R_w is assumed to be proportional to the product of pressure p and velocity$v$.

$R_w = pv$ = const.

The velocity at any point on the face of the disc is $v = r * \omega$

Combining these equations and assuming a constant angular velocity ω

pr = constant = C

The largest pressure P_{max} must then occur at the smallest area,r_i

$C = P_{max}\, r_i$

Hence pressure at any point in the contact region

$$p = \frac{P_{max}\, r_i}{r} \text{ and } F = \int_{r_1}^{r_2} 2\pi r dr\, p$$

$$F = 2\pi C \int_{r_i}^{r_o} dr = 2\pi C(r_o - r_i) \text{ ………………………………… } \mathbf{(3.34)}$$

$$C = \frac{F}{2\pi(r_o - r_i)}, \text{ and } T = \int_{r_i}^{r_o} 2\mu\pi r\, r^2 dr p \text{ ………………………… } \mathbf{(3.35)}$$

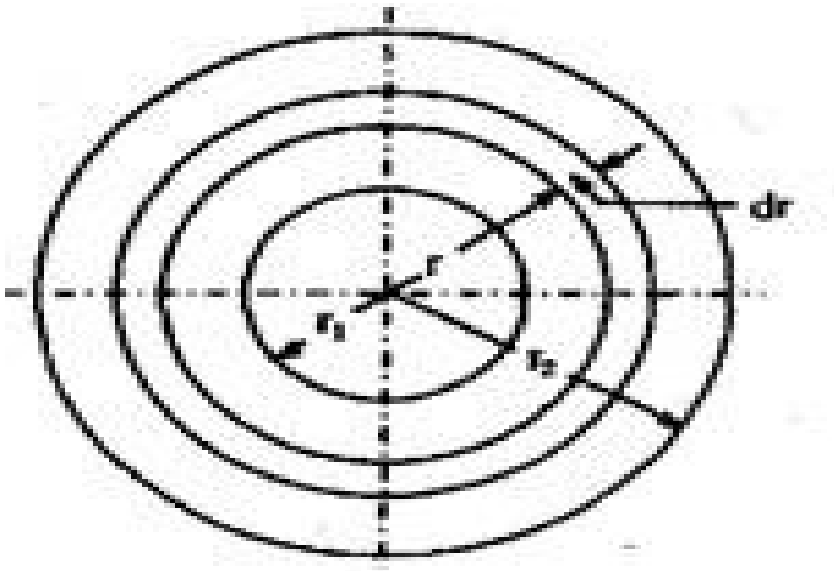
Total frictional torque becomes,

$$T = 2\mu\pi C \int_{r_i}^{r_o} r dr = \mu\pi C\left(r_o^2 - r_i^2\right) = \mu F \frac{(r_o + r_i)}{2} = \mu FR, \text{ Nm…………… } \mathbf{(3.36)}$$

Where R is the mean radius of the friction surface = $\frac{(r_o+r_i)}{2}$

$\eta = kpr$ = const.

Where η is the wear rate, P_{max} is the maximum distributed pressure in the pad and P is the pressure at the radial position.



Let p be the normal pressure, Pa

The total axial load, **F** = $\int_{r_1}^{r_2} \mathbf{2}\pi \boldsymbol{r\,dr\,p}$ and

The total frictional torque, **T** = $\int_{r_1}^{r_2} \mathbf{2}\mu\pi \boldsymbol{r^2\,dr\,p}$

The above two equations can be integrated using either of the assumptions $\int_{r_1}^{r_2} 2\pi r dr\, p$ and

i. The intensity of pressure is uniform, or p = constant

ii. The rate of wear is uniform, pr = constant

The brake disc-pad couple were assumed to be new where the pressure is uniform over the contact surface area.

3.6 Calculations in the Railway Disc Brake

The main problem of braking and stopping a heavy train system is the great input of heat flux into the disc in a very short time. Because of high temperature difference the material is exposed to high stress resulting in a heat shock. The problem can be solved only by applying a non-stationary and numerical calculation.

3.6.1 Determination of the Load Conditions

The specific loading condition considered for the braking analysis is from max velocity of 70km/hr. to standstill on a flat track (**TEGBARED** to **MEXICO** a distance of **554m**) and braking on a downhill track to maintain a constant velocity of 70km/hr. and braking to standstill afterwards (from **ESTIFANOS** to **BAMBIS,** a distance of **569m**) where the temperature of the ambient air is 30°C. The assumed temperature is the average natural environmental temperature of the operating route in Addis Ababa region.

3.6.2 Weight Transfer during Braking

When the train stands still on a straight track, its weight is uniformly distributed on all axles. On application of brakes on the vehicle, inertia forces are set up as the braking force is applied at lesser height whereas the vehicle load is at a higher height. This creates a couple to act on the body of the vehicle which ultimately result an additional loading on leading bogie as compared to trailing bogie. Therefore the ratio of weight distribution between the front and rear bogie is considered to be 60/40 in favor of the front bogie. On the front bogie there are five discs located on the two axle and one disc could able to take only 12% (0.6/5*100%) of the whole weight of the vehicle.

3.6.3 Determination of the Physical Model

Analysis of railway train braking to stop on a track derives from the physical model for determination of the heat transfer along the brake pairs in response to the variation in braking time. The weight distribution of the train considered is distributed between the front and rear bogies while each bogie consists of two wheel sets.

Hence, only 12 % of the whole brake force is applied to one disc from the forward part of the carriage. The braking energy for one wheel considering constant deceleration is:

$$(\mathbf{0.12})\ \left[(\mathbf{1.1})\frac{1}{2}\mathbf{M}\boldsymbol{v_o^2} + \mathbf{M} * \mathbf{g} * \boldsymbol{S_b}\,\frac{\boldsymbol{\delta}}{\mathbf{1000}}\right] = \mathbf{2}\boldsymbol{F_{disc}}\int_0^{tb} \boldsymbol{v_{disc}}\,\boldsymbol{d}(\boldsymbol{t}) \quad \ldots\ldots\ldots\ldots\ldots\ldots \ (3.27)$$

This change of energy is equal to the heat flux on the surface of the disc. This ratio is used to calculate the thermal load on the brake disc.

The material property of C/SiC ceramic composite brake disc with their physical and thermo-elastic material characteristics of the rotor disc and pad taken from literature is summarized in Table (8) and (9).

Table 8 Material properties of the brake disc

Material properties	Disc
Thermal conductivity, k(W/m ºC)	15
Density, ρ(kg/m^3)	2100
Mass of the brake disc, kg	18
Specific heat, C(J/kgk)	700
Poison's ratio, ν	0.25
Thermal expansion, α (10^{-6}/K)	3
Elastic modulus, E(GPa)	60
Allowed temperature(ºC)	800
Tensile yield strength, *(Mpa)*	190
Ultimate tensile strength, *(Mpa)*	300

Table 9 Material properties of the brake pad

Material properties	Pad
Thermal conductivity, k (W/m ºC)	26
Density, ρ(kg/m^3)	1800
Specific heat, C(J/kgK)	880
Poison's ratio, ν	0.3
Thermal expansion, α (10^{-6}/K)	0.3
Elastic modulus, E(GPa)	50.2

The coefficient of friction has significant effects on the form of the thermal field as well as on temperature distribution. The proper definition of the coefficient of friction is very important for perception of temperature field qualities for analyzing thermal effects in braking systems at railway discs. The value of the coefficient of friction depends on a number of tribological parameters such as speed of relative movement of parts in contact, the contact surface roughness, quality of materials of a tribo-pair, temperature of contact parts and so on. [60]

However, this research did not consider tribological parameters of braking processes. According to DIN EN ISO1183 the value μ = 0.34 of the friction coefficient for materials of the disc and the pad for AA LRT is taken as the recommended value.

3.6.4 Determination of Friction Surface of Disc

According to UIC 541-3 Standard, a Carbon/carbon (C/C) composites brake pad and C/SiC composite of brake disc with dimensions given in table 7 is used to calculate friction area swept on the disc by the brake pad.

3.6.5 Determination of Heat Flux

The value of the friction force which work on the brake disc is calculated below as:

3.7.5.1 Braking on a Flat Track

$$F_{disc} = \frac{(0.12)\frac{1}{2}(1.1)\,Mv_o^2}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} = \mathbf{6{,}564.20N} \text{ for service brake}$$

$$F_{disc} = \frac{(0.12)\frac{1}{2}(1.1)\,Mv_o^2}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} = \mathbf{11{,}936.996N} \text{ for emergency brake}$$

The instant heat flux entering the brake disc for service brake is:

$$\frac{\dot{Q(t)}}{2A_b} = \frac{\gamma(2F_{disc})v_{disc}}{2A_b} = \frac{(0.95)(2F_{disc})\frac{r_{disc}}{r_{wheel}}(v_o - at)}{2A_b} = \mathbf{607{,}882.5\text{- }34{,}396.648t}\ \left[W/m^2\right]$$

The instant heat flux entering the brake disc for emergency brake is:

$$\frac{\dot{Q(t)}}{2A_b} = \frac{\gamma(2F_{disc})v_{disc}}{2A_b} = \frac{(0.95)(2F_{disc})\frac{r_{disc}}{r_{wheel}}(v_o - at)}{2A_b} = \mathbf{1{,}105{,}434.2 - 113{,}727.78t}\ \left[W/m^2\right]$$

3.7.5.2 Braking on a Downhill Track

a) Braking on downhill track for maintaining constant velocity

$$Q_{disc} = \frac{0.95\, M.g.v_o.\sin\delta}{12} = \mathbf{18{,}077.552W}$$

$$F_{disc} = \frac{\mu Q(t)}{2v_o\frac{r_{disc}}{r_{wheel}}} = \mathbf{7{,}626.57N}$$

The heat flux entering the brake disc:

$$\frac{\dot{Q(t)}}{2A_b} = \frac{\gamma Q(t)}{2A_b} = \mathbf{57{,}743.375}\ \left[W/m^2\right]$$

b) Braking to stand still after maintaining constant velocity

$$\boldsymbol{F_{disc}} = \frac{\frac{1}{2}(\mathbf{0.12})\ (\mathbf{1.1})\ \mathbf{M}\boldsymbol{v_o^2}+\boldsymbol{Mgv_o}\ \mathbf{sin}\ \boldsymbol{\delta}}{2\frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}\left[\boldsymbol{v_o t_b} - \frac{1}{2}\boldsymbol{a t_b^2}\right]} = \mathbf{7{,}515.286N} \text{ for service brake}$$

$$\boldsymbol{F_{disc}} = \frac{\frac{1}{2}(\mathbf{0.12})\ (\mathbf{1.1})\ \mathbf{M}\boldsymbol{v_o^2}+\boldsymbol{Mgv_o}\ \mathbf{sin}\ \boldsymbol{\delta}}{2\frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}\left[\boldsymbol{v_o t_b} - \frac{1}{2}\boldsymbol{a t_b^2}\right]} = \mathbf{19{,}049.86N} \text{ for emergency brake}$$

The instant heat flux entering the brake disc for service brake:

$$\frac{\boldsymbol{Q(\dot{t})}}{\boldsymbol{2A_b}} = \frac{\boldsymbol{\gamma(2F_{disc})v_{disc}}}{\boldsymbol{2A_b}} = \frac{(\mathbf{0.95})(\mathbf{2}\boldsymbol{F_{disc}})\frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}(\boldsymbol{v_o - at})}{\boldsymbol{2A_b}} = \mathbf{695{,}958.52\text{-}39{,}380.37t}\left[W/m^2\right]$$

The instant heat flux entering the brake disc for emergency brake:

$$\frac{\boldsymbol{Q(\dot{t})}}{\boldsymbol{2A_b}} = \frac{\boldsymbol{\gamma(2F_{disc})v_{disc}}}{\boldsymbol{2A_b}} = \frac{(\mathbf{0.95})(\mathbf{2}\boldsymbol{F_{disc}})\frac{\boldsymbol{r_{disc}}}{\boldsymbol{r_{wheel}}}(\boldsymbol{v_o - at})}{\boldsymbol{2A_b}} = \mathbf{1{,}764{,}126.13\text{-}181{,}494.46t}\left[W/m^2\right]$$

At initial time t = 0s, $\frac{\boldsymbol{Q(\dot{t})}}{\boldsymbol{2A_b}} = 1{,}764{,}126.13 \, {}^{W}/_{m^2}$

t = 1s, $\frac{\boldsymbol{Q(\dot{t})}}{\boldsymbol{2A_b}} = 1{,}582{,}631.67 \, {}^{W}/_{m^2}$

t = 2s, $\frac{\boldsymbol{Q(\dot{t})}}{\boldsymbol{2A_b}} = 1{,}401{,}137.21 \, {}^{W}/_{m^2}$

….. … ……….

t = 9.72s = 0 ${}^{W}/_{m^2}$

S. No.	Time increment [s]	Emergency brake heat flux
1	0	1,764,126.151
2	1.08	1,568,112.113
3	2.16	1,372,098.096
4	3.24	1,176,084.08
5	4.32	980,070.063
6	5.4	784,056.046
7	6.48	588,042.029

8	7.56	392,028.01
9	8.64	196,013.996
10	9.72	0

Table 10 Applied heat flux values as function of time per area of the brake disc

3.6.6 Pressure determination for the brake caliper

The surface pressure between the disc and pad can be determined from the force applied to the disc for each braking condition. Hence, brake caliper pressure is calculated as:

a) Braking on a flat track from maximum velocity to standstill;

$$p = \frac{F_{disc}}{\mu A_p} = \frac{11{,}936.996}{0.0119} = \mathbf{1{,}003{,}108.908}\ \text{Pa}$$

b) Braking on a downhill track for maintaining constant velocity

$$p = \frac{F_{disc}}{\mu A_p} = \frac{7{,}626.57}{0.0119} = \mathbf{640{,}888.24}\ \text{Pa}$$

c) Braking on a downhill track to standstill after maintaining constant velocity

$$p = \frac{F_{disc}}{\mu A_p} = \frac{19{,}049.86}{0.0119} = \mathbf{1{,}600{,}828.57}\ \text{Pa}$$

Where, P_p is pressure of pad and A_p is surface area of pad

P_r Protor is pressure of disc rotor and A_r is surface area of rotor

μ is friction coefficient disc/pad.

3.6.7 Centrifugal Load Determination

$$F_c = m_d \omega^2 R$$

Where; F_c – centrifugal force

m_d -mass of the brake disc

ω - Angular velocity

R -radius of the brake disc

The angular velocity at any time t is:

$$\omega = \omega_o \left(1 - \frac{t}{t_b}\right)$$

$$\omega = 58.9224 - 6.062t\ rad/s$$

$\omega^2 = (58.9224 - 6.062t)^2$

m_d= 18kg

R = 110mm = 0.11m

$F_c = 1.98(58.9224 - 6.062t)^2$

Thus at t = 0, F_c = **6874.26N**.

In the process of braking of railway vehicles, it is necessary to define a model of a thermal analysis that describes the heating transfer of the heat generated by friction at surfaces which are in contact, between a railway disc and braking pads, through the disc and pads, as well as heat outflow of the whole braking system due to cooling of the surrounding air.

CHAPTER FOUR

FEA ANALYSIS OF AA LRT BRAKE DISC

Brake systems involve a high degree of technical risk. Thus, simulation of thermal and structural systems is useful for predicting their performance early during the design stage which is essential to the improvement of the design. Understanding the thermal fluid-structure interaction helps to predict the performance, improve the design, and avoid possible future failures of the system.

Friction is a complex process involving simultaneous appearance of wear, thermal, strain fatigue etc. This coupled phenomena can influence the behavior of the tribological assemblies over time. The stress and overall temperature distribution caused by the friction generated heat acting on the contact surface as part of the mechanical energy (kinetic and potential energy) is transformed to heat will be studied using Finite Element Method and Computational Fluid Dynamics.

For braking process analysis, it is worth investigating trends associated with composite material response using finite element method simulations based on the defined analytical model and heat sources at surfaces in contact with the disc rotor.

In this work, an assessment tool of the thermo-mechanical analysis on a broad method that deals with complex thermos-fluidic and structural systems which involve multiple physics interactions is presented.

The CFD method is first applied to the brake disc model where heat transfer coefficients are determined using CFX to obtain the convection terms necessary for the FEM thermal analysis. The thermal analysis provides the temperature fields where transient structural stress behaviors are estimated.

This study utilizes ANSYS15 finite element analysis software to carry out induced transient thermal and coupled thermo-mechanical stress analysis of a railway brake disc under different braking conditions. It will consider single thermal cycle of severe braking and cooling under the established driving conditions and further accumulation of plastic strain due to repeated braking conditions will not be analyzed.

Since the elastic stresses and strains are completely reversible and non-cumulative, the brake disc model was investigated for one cycle of the braking and cooling period only.

4.1 Brake Disc Modeling

There are many commercial general purpose modeling software packages available some of which are: Solid works, ABAQUS, SIMULIA, FEMLAB, MSC NASTRAN, CATIA, AUTODESK inventor, Pro/Engineer, COSMOS and ANSYS. These packages usually cover a wide spectrum of possibilities (e.g. materials properties, loads and supports, and heat sources varying with time) to solve general partial differential or integral equations representing the field problem.

ANSYS mechanical is a self-contained analysis tool incorporating pre-processing including creation of geometry and meshing, solver and post processing modules in a unified graphical user interface.

CATIA V5 R19 a rigorous multi-platform package developed by Dassault systemes will be employed to model the geometry in this project while ANSYS being used for the finite element simulation. In order to conduct heat transfer and structural analysis of the brake disc with reasonable accuracy a 3-D model of the brake disc and pad has been developed.

4.1.1 Geometric Modeling

CATIA V5 R19 a 3-D parametric solid modeler with part and assembly modeling capabilities, where a part is designed by sketching the component shape and defining dimensions, shape and their interrelationships, will be used to model the geometry. CATIA V5 R19 has parametric features, that allow change of one feature and all related features are automatically updated to reflect the change and its effects throughout the part. It can be used to create angular shaped parts, to which 3D surface can be applied to create hybrid parts consisting of mixture of angular and curved shapes. This provides the ability to create model designs of varying shapes.

Assemblies can be created from parts, either combined individually or grouped in subassemblies. CATIA V5 R19 allows to build these individual components and subassemblies into an assembly in a hierarchical manner according to predefined relationships and constraints. For part modeling, the parametric relationship allows one to quickly update an entire assembly based on a change in one of its parts. The model is then saved as Initial Graphics Exchange Specification (IGS/IGES) file vector graphics standard format to enable file exchange with ANSYS work bench for finite element analysis. This format exports the geometry as just curves giving better control while creating the structured mesh with required mesh distribution.

We can apply two types of symmetry in modeling disc: axis-symmetry (the symmetry about a central axis) and symmetry in material properties and loading.

In axis-symmetry, an axisymmetric model is based on the assumption that both the geometry and loading do not vary in the circumferential direction which implies that the measurements (temperature, stress, strain etc.) are also invariant in the circumferential direction.

The second symmetry condition is symmetry in material properties and loading. Once symmetry in geometry is observed, the same symmetry plane or axis should also be valid for the material properties and loading (heat flux, temperature, etc.).

4.2 Assumptions applied for development of physical model of the Brake Disc

To simplify the analysis, the following assumptions have been made:

- All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux. In other words, a high amount of kinetic energy is converted into heat energy at interfaces according to the first law of thermodynamics during the slipping period and the heat generated between contact surfaces will be dissipated by convection to the environment.
- The heat transfer involved for this analysis is conduction and convection process. The heat transfer due to radiation can be neglected in this analysis because of its small amount and contribution which is 5 % to 10 % [41]. In fact, the heat radiation only plays an important role at high temperature and low speeds.
- The disc material is considered as homogeneous and isotropic because young's modulus, Poisson's ratio and the thermal expansion coefficient are assumed to be constant for isotropic material.
- The domain is considered as axis-symmetric. The temperature field is symmetric with respect to the central plane of the brake disc.
- Inertia and body force effects are negligible during the analysis. Here a transient analysis calculates the effects of thermal loading on a structure while ignoring inertia and damping effects.
- The disc is stress free before the application of brake. Due to the application of brakes on the rotor, heat generation takes place due to friction and this temperature so generated has to be conducted across the disc across section.
- In the analysis, the ambient air initial temperature has been set to 30 °C.
- All other possible disc brake loads are neglected for example, the effects of wear, humidity, dust or other atmospheric factors.
- Only certain parts of disc brake rotor will apply with convection heat transfer such as cooling vanes area, outer ring diameter area and disc brake surface on which, each surface

of the rotor was subjected to different values of convection heat transfer coefficient obtained from this calculations.

- The nominal surface of contact between the disc brake and the pad in operation is equal to the apparent surface in the sliding motion.
- Uniform pressure distribution by the brake pad onto the disc brake surface. Uniform pressure distribution in the contact region is often valid when the pad is new.

4.3 Finite element analysis

4.3.1 Introduction

Mathematical Models

Mathematical models are differential equations with a set of corresponding boundary and initial conditions. There are two common classes of numerical methods. [61]

1. **Finite difference method**

With finite difference method the differential equation is written for each node and the derivatives are replaced by difference equations. This method is difficult to apply to problems with complex geometries or complex boundary conditions and non-isotropic properties.

2. **Finite element method**

In contrast, the Finite element method uses integral formulation rather than difference equations to create a system of algebraic equations. An approximate continuous function is assumed to represent the solution for each element. The complete solution is then generated by connecting or assembling the individual solutions allowing for continuity at the inter-elemental boundaries.

Seven Steps in the Finite Element Method

Step 1 – Idealization

Assume a solution that approximate the behavior of an element. The real problem is idealized, assumptions are made to simplify the problem by:

- Reducing the dimensions (all real problems are 3D, but may be idealized with1D, 2D or 3D models),
- Idealizing the support conditions,
- Suppressing details, such as small holes and fillets, that are insignificant from the analysis point of view, but which complicate matters during mesh[2] generation.

Step 2 – Discretization

Discretize the solution domain in to finite elements. The problem domain is subdivided into a collection of simple nodes and elements.

Step 3 - Choice of the type of element

There are multitude of software available on the market that offer a lot different types of elements.

Step 4 – Assembly of the discrete elements

The element equations for each element in the FEM mesh are assembled into a set of global equations that model the properties of the entire system.

Step 5 - Application of Boundary Conditions and loads

Solution cannot be obtained unless boundary conditions are applied. They reflect the known values for certain primary unknowns. Imposing the boundary conditions modifies the global equations.

Step 6 - Solve for Primary Unknowns

The modified global equations are solved for the primary unknowns at the nodes.

Step 7 - Calculate Derived Variables

Calculated using the nodal values of the primary variables.

The mechanical response of an element is characterized in terms of a finite number of degrees of freedom. These degrees of freedoms are represented as the values of the unknown functions at a set of node points (displacements, temperature, flow etc.)

Finite element method is a numerical procedure for solving continuum mechanics of problem with accuracy acceptable to engineers. Finite element method is a mathematical modeling tool involving discretization of a continuous domain using building block entities called finite elements connected to each other by nodes for once and moment transfer. This process includes finite element modeling and finite element analysis. [4]

Finite Element Method is a numerical method to solve ordinary differential equations of equilibrium starting with simple linear static stress and heat transfer analysis to complex simulations involving highly non-linear, fluid flow and dynamic events. It gives a discreet approximate solution for continuous equations of physical problems. The approximation can be improved by grading/refining the mesh where field gradients are high and more resolution is required. ANSYS analysis involves three principal stages:

Preprocessing: Preprocessing stage involves the preparation of geometric model, element type, real constant, material property and discretization. Create and discretize the solution domain into finite elements. A fully continuous field is represented by a piecewise continuous field defined by finite number of nodal quantities and simple interpolation within each element. This involves dividing the domain into sub-domains, called 'elements', and selecting points called nodes on the inter-element boundaries or in the interior of the elements. Assume an approximate and continuous shape function to represent the behavior of the element. Develop equations for an element and assemble the elements to represent the complete problem. Apply boundary conditions, initial conditions and the loading.

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

Post processing: This stage involves obtaining results from the solution, processing the nodal data to get other information such as values of principal stresses, displacements, heat fluxes, temperature distribution etc. and presentation of the results in tabular or graphical forms.

In general there are several approaches to formulating finite element problems:

- Direct formulation
- The minimum total potential energy formulation and
- Weighted residual formulation

4.3.2 Definition of Material

The material of the proposed ceramic composite brake disc is 30%C/SiC matrix, with good thermo-physical characteristics and the brake pad assumed to have an isotropic elastic behavior, whose thermo-mechanical characteristics is as recapitulated in table 10 below.

Table 11 Material properties of the composite brake disc

Material properties	Disc
Thermal conductivity, k(W/m ºC)	15
Density, ρ(kg/m^3)	2100
Mass of the brake disc, kg	8.075
Specific heat, C(J/KgK)	700
Poison’s ratio, ν	0.25
Thermal expansion, α (10^{-6}/K)	3
Elastic modulus, E(GPa)	60
Allowed temperature(ºC)	800
Tensile yield strength, *(Mpa)*	190
Ultimate tensile strength, *(Mpa)*	460

4.3.3 Geometries and Dimensions of Brake Disk-pad

Dimensions of the geometry used in this study was kept the same as that used in the actual AA LRT train, allowing for direct comparison of results. Brake disc and pads used for development of the 3-D topology with radius and thickness as well as other auxiliary parameters are taken as an input for the analysis.

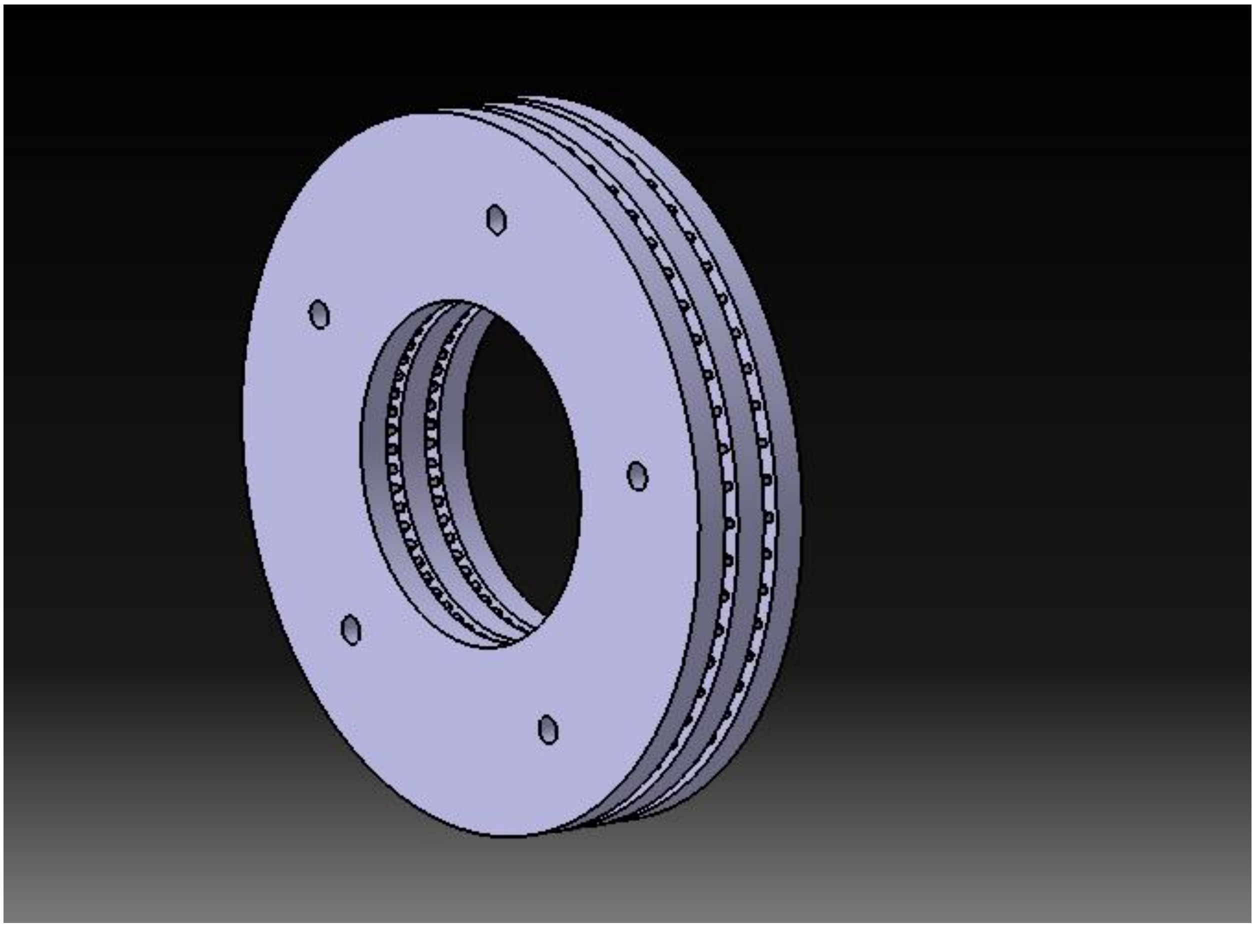


Figure 8 Simplified 60 vane ventilated brake disc geometry

4.4 Preparation of the Geometry and FEM Discretization

4.4.1 Methods of Finite Element Analysis

Before the advent of computers, analytical methods were used as engineering tools for determining the integrity of a design. Mathematical equations for conduction of heat in an isotropic solid could be used to investigate the temperature response of brake disc designs by simplifying complicated parameters such as temperature dependent material properties, real brake disc geometry and complex boundary conditions.

However, for real problems involving complex material properties and boundary conditions, a numerical method of analysis is more suitable. The most popular of the various numerical methods that have been developed is the finite element method. Two types of finite element analysis are widely used in brake design: heat transfer analysis to determine transient temperature distributions and thermal structural stress analysis to determine stresses and strains due to these non-uniform temperature distributions.

Mechanical components in the form of simple bars, beams, etc., can be analyzed quite easily by basic methods of mechanics that provide closed-form solutions. Actual components, however, are rarely so simple, and the designer is forced to less effective approximations of closed-form solutions, experimentation, or numerical methods. There are a great many numerical techniques used in engineering applications for which the digital computer is very useful. In mechanical design, where computer-aided design (CAD) software is heavily employed, the analysis method that integrates well with CAD is finite-element analysis (FEA).

Mechanical component is a continuous elastic structure. Finite Element Analysis divides the structure into small but finite, well-defined, elastic substructures called elements. By using polynomial functions, together with matrix operations, the continuous elastic behavior of each element is developed in terms of the element's material and geometric properties. Loads can be applied within the element, on the surface of the element, or at the nodes of the element. The element's nodes are the fundamental governing entities of the element, as it is the node where the element connects to other elements, elastic properties of the element are eventually established, boundary conditions are assigned and forces (contact or body) are ultimately applied. A node possesses degrees of freedom. Degrees of freedom are the independent translational and rotational motions that can exist at a node. At most, a node can possess three translational and three rotational degrees of freedom. Once each element within a structure is defined locally in a matrix form, the elements are then globally assembled through their common nodes into an overall system matrix. Applied loads and boundary conditions are then specified and through matrix operations the values of all unknown displacement degrees of freedom are determined.

Once this is done, it is a simple matter to use these displacements to determine strains and stresses through the constitutive equations of elasticity.

4.4.2 Coupled-Field Analyses and Methods

A coupled-field analysis is a combination of analyses from different engineering disciplines (physics fields) that interact to solve a global engineering problem, hence, we often refer to a coupled-field analysis as a multi-physics analysis. When the input of one field analysis depends on the results from another analysis, the analyses are coupled [62]. Examples of coupled-field analysis are thermal-stress analysis, thermal-electric analysis, and fluid-structure analysis. The procedure for a coupled field analysis depends on which fields are being coupled, but two distinct methods can be identified: direct (mutual) and sequential (load transfer).

The direct method usually involves just one analysis where each field influences the other that uses a coupled-field element type containing all necessary degrees of freedom. Coupling is handled by calculating element matrices or element load vectors that contain all necessary terms. An example of a direct method coupled-field analysis is a piezoelectric analysis using the PLANE223, SOLID226, or SOLID227 elements.

The sequential method involves two or more sequential analyses, each belonging to a different field. You couple the two fields by applying results from the first analysis as loads for the second analysis in which only one field influence the other. An example of this is a sequential thermal-stress analysis where nodal temperatures from the thermal analysis are applied as "body force" loads in the subsequent stress analysis.

With a physics file-based load transfer, one must explicitly transfer loads using the physics environment. An example of this type of analysis is a sequential thermal-stress analysis where nodal temperatures from the thermal analysis are applied as "body force" loads in the subsequent stress analysis. The physics analysis is based on a single finite element mesh across physics. You create physics files that define the physics environment; these files configure the database and prepare the single mesh for a given physics simulation. The general process is to read in the first physics file and solve. Then read in the next physics field, specify the loads to be transferred, and solve the second physics.

Some of the applications in which coupled-field analysis may be required are pressure vessels (thermal-stress analysis), fluid flow constrictions (fluid-structure analysis), induction heating (magnetic-thermal analysis), ultrasonic transducers (piezoelectric analysis), magnetic forming (magneto-structural analysis), and micro-electro mechanical systems (MEMS).

The term sequentially coupled refers to solving one physics simulation after another. Results from one analysis become loads for the next analysis. If the analyses are fully coupled, results of the second analysis will change some input to the first analysis. The complete set of boundary conditions and loads consists of the following:

- **Base physics loads,** which are not a function of other physics analyses. Such loads also are called nominal boundary conditions.
- **Coupled loads,** which are results of the other physics simulation.

Typical applications you can solve with ANSYS include the following:

- Thermal stress
- Induction heating
- Induction stirring
- Steady-state fluid-structure interaction
- Magneto-structural interaction
- Electrostatic-structural interaction
- Current conduction-magneto statics

The ANSYS program can perform multi-physics analyses with a single ANSYS database. A single set of nodes and elements will exist for the entire model. What these elements represent are changes from one physics analysis to another, based on the use of the physics environment concept.

There are basically two methods of coupling distinguished by the finite element formulation techniques used to develop the matrix equations: Strong (also matrix, simultaneous, or full) coupling, and Weak (also load vector or sequential) coupling.

In the braking process there is a link between the thermal and mechanical effects, which makes the use of a coupled temperature-displacement analysis necessary. This means that the elements used in FE simulations must have both temperature and displacement degrees of freedom.

4.4.3 ANSYS ICEM CFD Modeling and Mesh Preparation

ANSYS ICEM CFD is a mesh generation software for applications in fluid mechanics and mechanical structures. ANSYS CFX is designed to perform fluid flow simulations.

The geometry will be created in units of millimeters and then scaled to meters at the output step.

Select import geometry from the menu having a standard file format of IGS that enables the geometry data developed in CATIA exchange with ANSYS.

The brake disc is rigidly constrained at the bolt holes. For the pads, the leading and trailing edge are rigidly constrained in the circumferential direction.

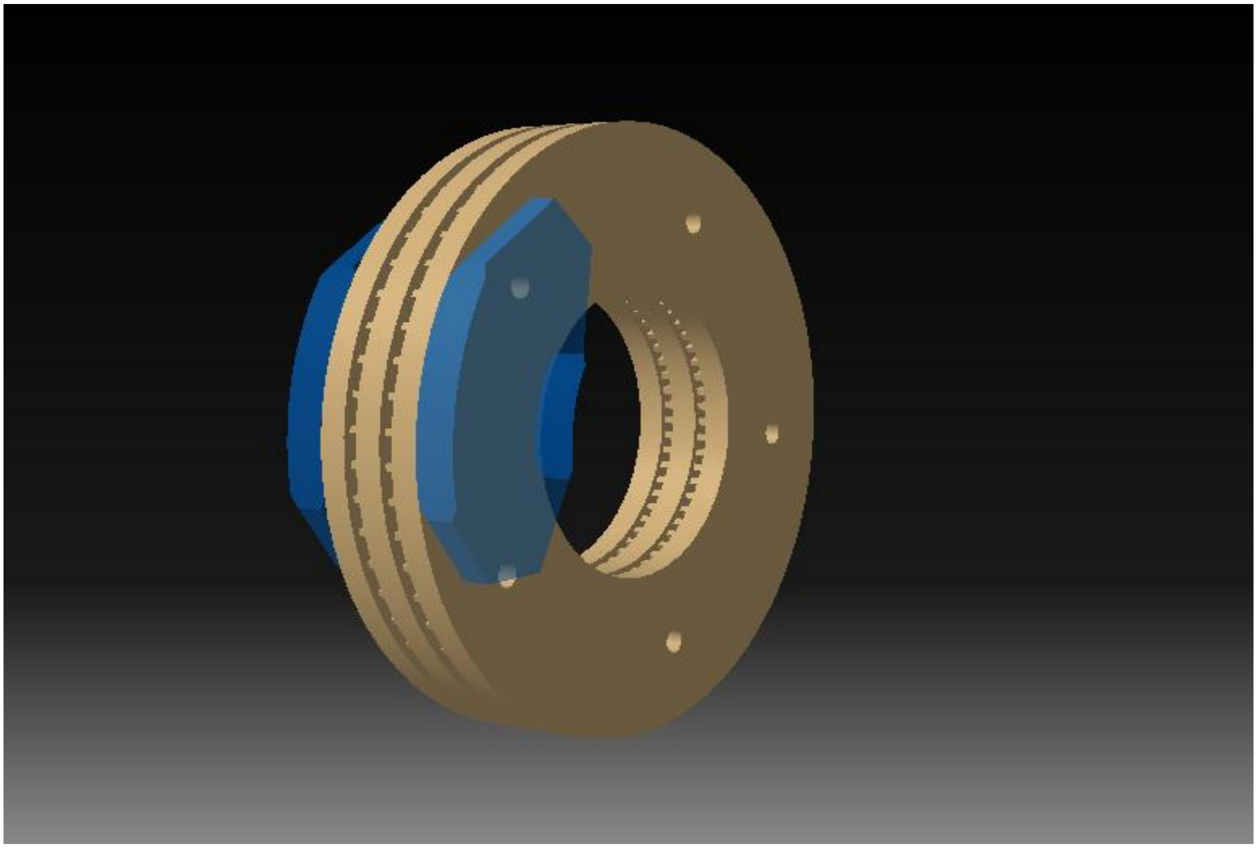


Figure 9 3D Geometry of the Disc/Pad assembly in CATIA.

4.4.4 Meshing and Loading Conditions of the Disc

The elements used for meshing of the full and ventilated disc are tetrahedral three dimensional elements (SOLID90) which is a higher order version of the 3-D eight node thermal element (SOLID70) both thermally and structurally. The element has 20 nodes with a single degree of freedom, temperature, at each node. The 20-node elements have compatible temperature shapes and are well suited to model curved boundaries.

4.4.5 Modeling in ANSYS CFD

The rotating brake disc is enclosed in a box domain of still air where the effect of cooling convection is be determined.

The finite volume method consists of three stages: the formal integration of the governing equations of the fluid flow over the entire (finite) control volumes of the solution domain. Then, discretization, involving the substitution of a variety of finite-difference-type approximations for the terms in the integrated equation representing flow processes such as convection, diffusion, and sources. This converts the integral equation into a system of algebraic equations, which can then be solved using iterative methods. The first stage of the process, the control volume integration, is the step that distinguishes the finite volume method from other CFD methods.

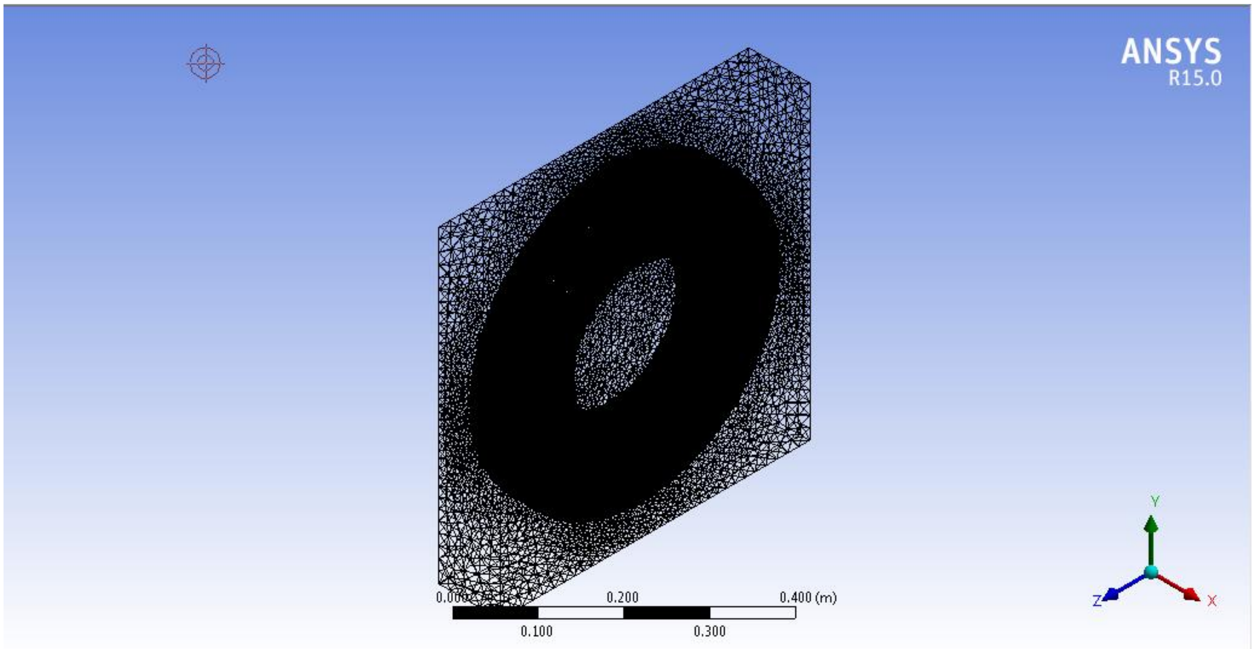


Figure 10 Meshed model of the brake disc box enclosure, nodes 50705 and elements 240466

4.4.6 Mesh generation

Mesh generation is a process of generating finite volumes or elements with elements that are hexahedral, tetrahedral, prismatic (wedges), pyramids or any combination of them which can either be a surface mesh or volume mesh.

The heat dissipation and thermal performance of ventilated brake discs strongly depends on the aerodynamic characteristics of the air flow through the rotor passages. In this thesis, the thermal convection will be analyzed using CFX fluid flow and the wall heat transfer coefficient and temperature contours are determined.

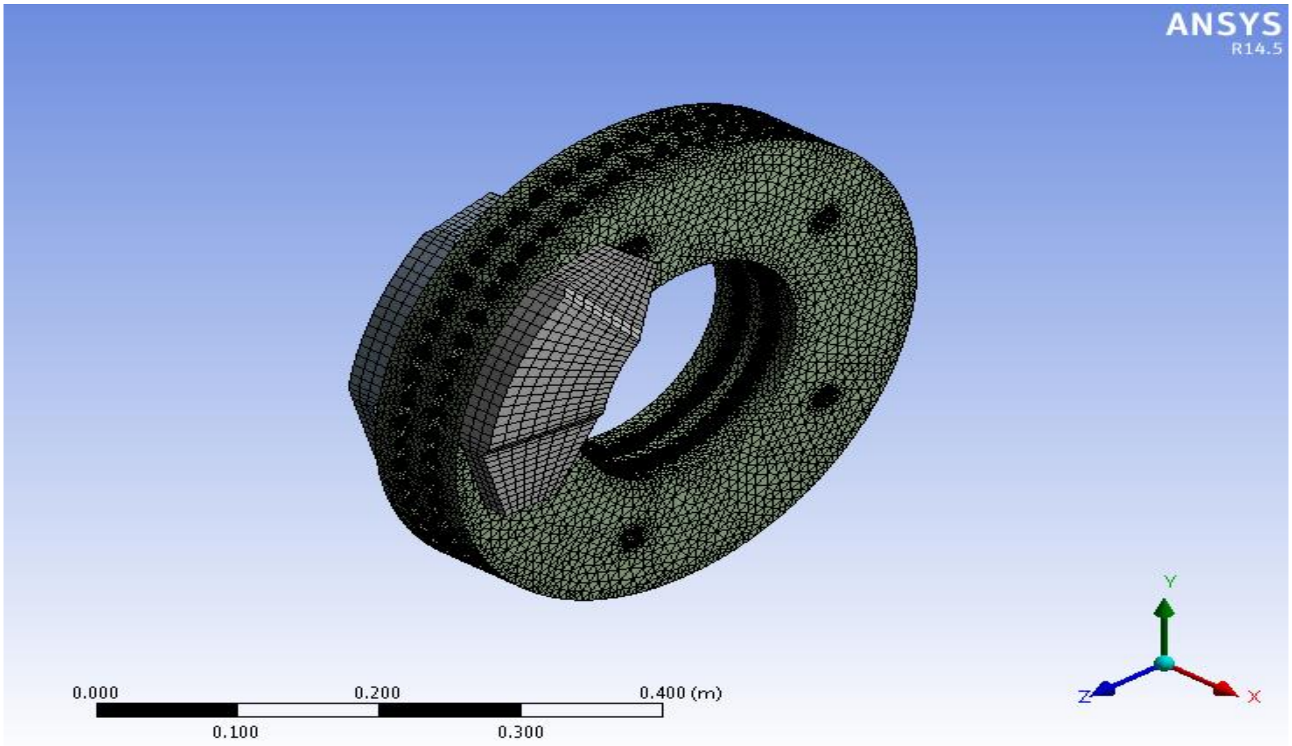


Figure 11 Meshed model of the brake disc/pad assembly

Proper specification of boundary conditions need to be taken care of as incorrect boundary conditions will lead to incorrect results. Wall boundary conditions are used to bind the fluid and solid regions of the brake disc.

To create mesh for the brake disc CFD analysis, the brake disc is imagined to be inserted in an imaginary box enclosure (domain) which contains the field to be studied to simulate the surroundings.

It is important to have the right domain definition to avoid any unnecessary computational effort due to a larger domain and to avoid any interaction between exit jet and the domain boundaries which happens in a smaller domain.

Nestor [63] has performed domain study to evaluate the influence of the domain size on the computed solutions. After wide range of domain configurations were investigated it was suggested that the appropriate domain size is as given below:

Domain diameter = 4.3x (diameter of the brake disc rotor) **……………………... (4.1)**

Domain height = 2.7x (diameter of the brake disc rotor) **………………………. (4.2)**

At the time of the braking process, a part of the frictional heat escapes into the ambient air by convection and radiation. Thus, determination of the heat transfer coefficients is essential.
Their exact calculation is rather difficult because these coefficients depend on the location and the construction of the braking system, the speed of the vehicle and consequently on the air circulation. Since the process of heat transfer by radiation is insignificant at such low temperatures, we will determine only the wall heat transfer/convection coefficient (*h*) of the disc using ANSYS 15 CFX Workbench.

4.5 Definition of the Loads and Boundary Conditions

After completing the finite element model, it is necessary to set up initial conditions and boundary constraints to the model. The AA LRT railway brake disc is subjected to the following loads.

- The initial temperature of the disc and pads is 25 °C.
- The surface convection condition is applied at all surfaces of the disc except the friction contact surface area of the disc and the pad with the values of the convection coefficient (h) obtained from ANSYS CFX simulation result.
- The heat flux applied into the brake disc during braking can be calculated by the formula described in section 3.6.

  a) On a straight track is **1,105,434.2 – 113,727.78t** (W/m^2)

  b) On a downhill track for maintaining constant velocity is **7,626.57** (W/m^2)

  c) On a downhill track braking to standstill after maintaining constant velocity is **1,764,126.13 – 181,494.46t** (W/m^2)
- The pressure on the rubbing contact surfaces of the disc and brake pad applied for three cases of brake applications is.

  a) Pressure application for braking on a straight track has a value of **1,003,108.908*Pa.***

  b) Pressure application for braking on a downhill track for maintaining a constant velocity has a value of **640,888.24 *Pa.***

  c) Pressure application for braking on a downhill track to a standstill after maintaining a constant velocity has a value of **1,600,828.57*pa.***
- Angular velocity of the disc rotor is given **58.9224**$rev/sec.$
- Translational acceleration of the rotor disc is $1.1 m/s^2$ for service brake and $2 m/s^2$ for emergency brake.

- Fixed support is applied at the holes and recess where the bolts are located and cylindrical support at the inner surface of the hub.

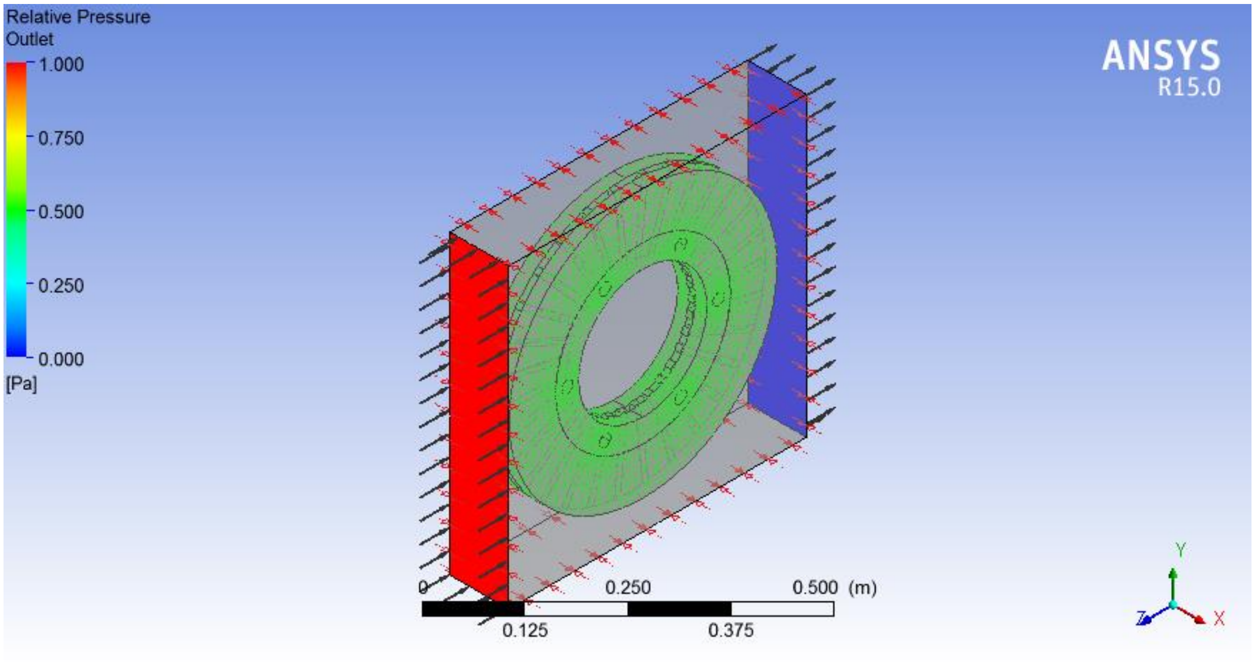


Figure 12 Fluid domain enclosure box of the brake disc CFD CFX fluid flow model

Table 12 Boundary and initial conditions for CFX pre

Boundary conditions	Parameter values
Inlet	
Flow regime	Subsonic
Mass and momentum Normal speed	19.44m/s
Turbulence	Intensity and eddy viscosity ratio
Fractional intensity	0.05
Eddy viscosity	10
Heat transfer: Static temperature	303K
Outlet	
Flow regime	Subsonic
Mass and momentum	Static pressure
Relative pressure	0 [pa]
Symmetry	Symmetry
Wall	
Mass and momentum	No slip wall
Wall roughness	Smooth wall
Heat transfer	Adiabatic

The disc is related to two adiabatic surfaces and two surfaces of symmetry in the fluid domain whose ambient air temperature is taken as equal to 28°C as well as air flow inlet and outlet surfaces.

Table 13 Solid disc domain

Domain	Brake disc zone
Type	Solid
Location	Brake disc
Domain motion	Rotating
Angular velocity	58.9224 rev/s
Rotation axis	Global X axis
Morphology	Continuous solid
Disc wall boundary	
Boundary type	Wall
Frame type	Rotating
Heat transfer: Heat flux	1,764,126.15 W/m^2

Areas of interfaces located between the interactions regions called as solid-fluid interface are commonly used to create connection or linkage areas.

Table 14 Fluid domain

Domain	Brake disc zone
Type	Fluid
Location	Fluid zone
Domain motion	Stationary
Angular velocity	58.9224 rev/s
Solid definition	Material Library(air)
Morphology	Continuous fluid
Heat Transfer Model	Thermal Energy
Turbulence Model	k epsilon
Turbulent Wall Functions	Scalable

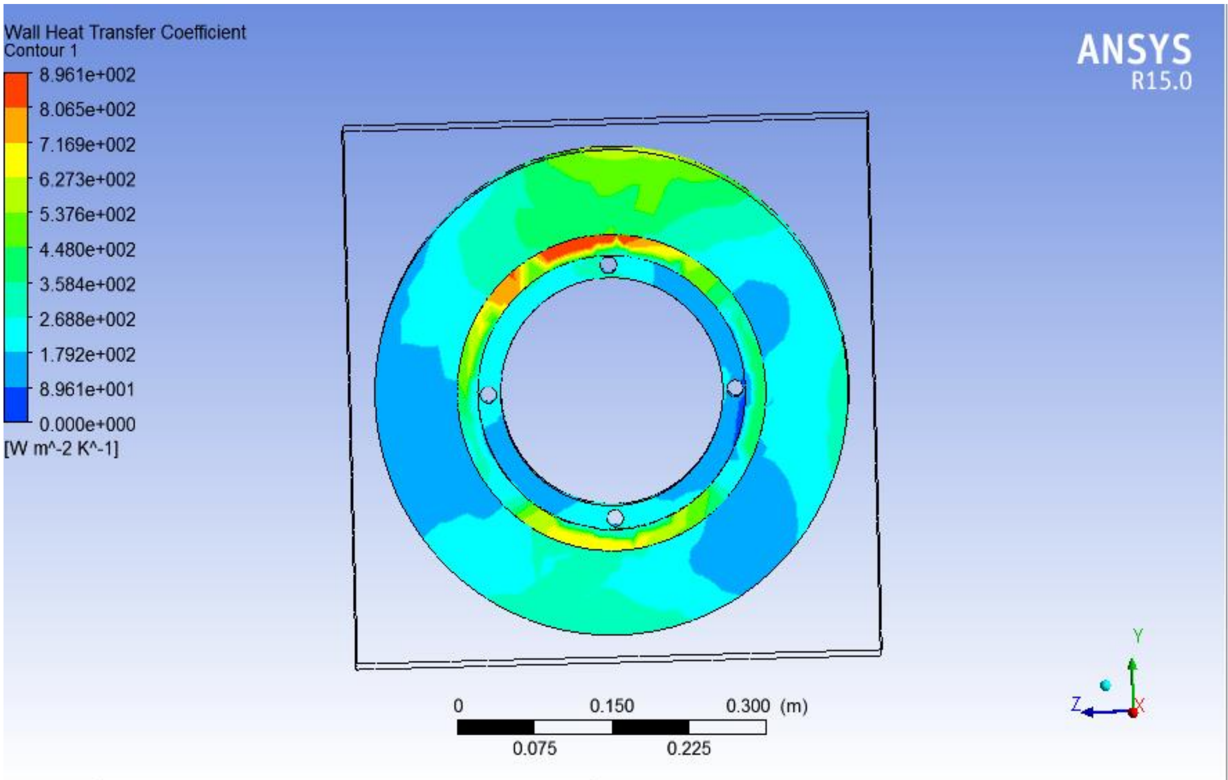


Figure 13 Wall heat transfer coefficient distribution on the ventilated disc brake for transient analysis

4.6 Thermal Load

Thermal load application for braking from maximum velocity to standstill on a downhill track after maintaining a constant velocity.

The thermal loading is characterized by the heat flux entering the brake disc through the real contact area (two sides of the disc). We consider heat flux during application of brake (deceleration period) only and we consider convection when the vehicle accelerates only. The initial and boundary conditions are introduced into ANSYS. The thermal calculation is carried out by choosing the transient state and by introducing physical properties of the materials.

- Total emergency brake time, t_b= 9.72s
- Increment of initial time =0.972s
- Initial time =0s
- Number of time steps =10
- Initial temperature of the disc= 28 ^{0}C
- Material of the disc : 30% C/SiC ceramic composite matrix

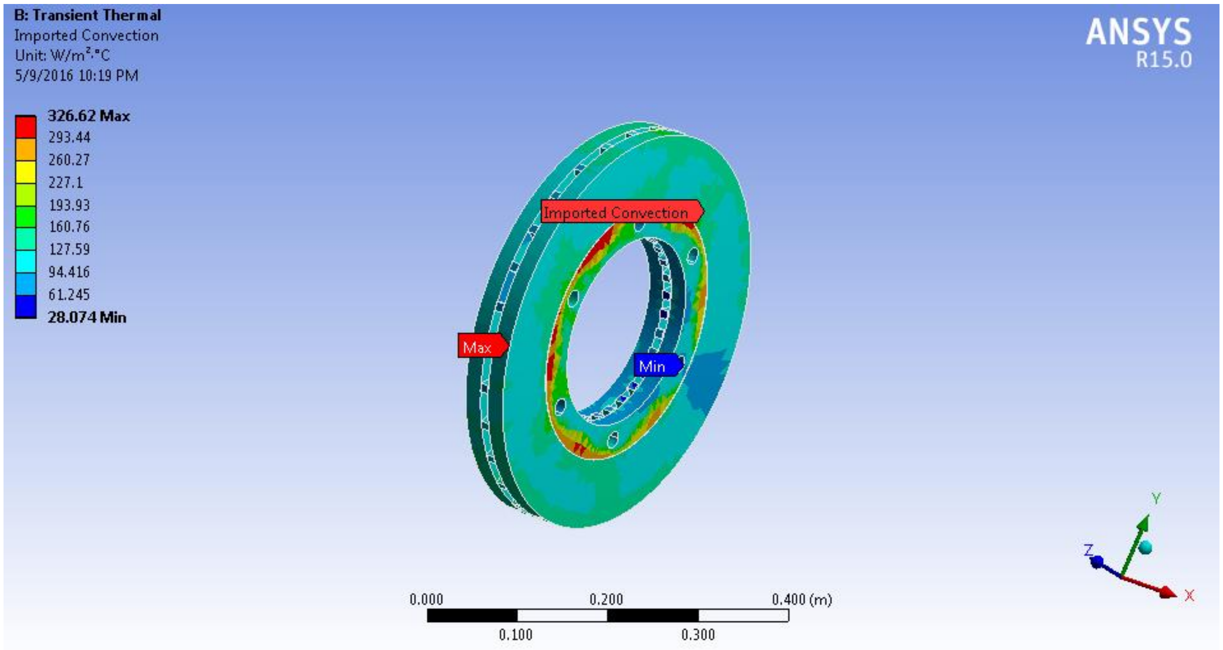


Figure 14 Imported convection coefficient from CFX fluid flow for thermal analysis.

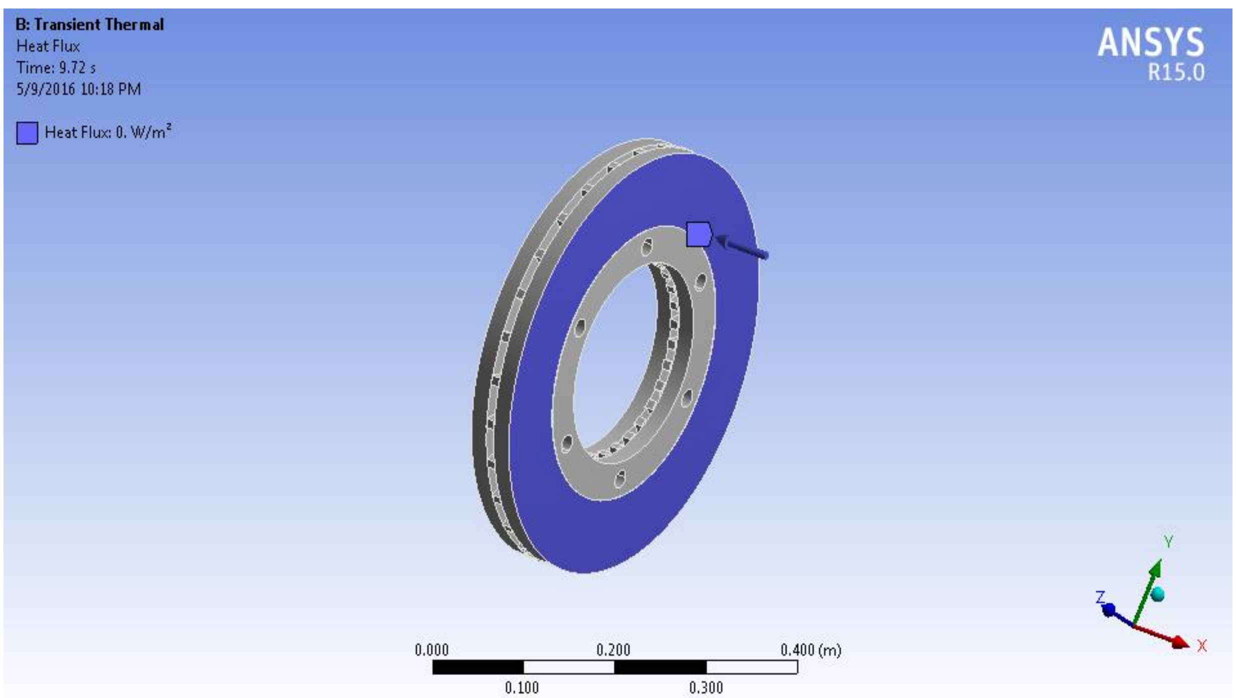


Figure 15 Heat flux application for emergency braking from maximum velocity to standstill on a down gradient track after maintaining a constant velocity.

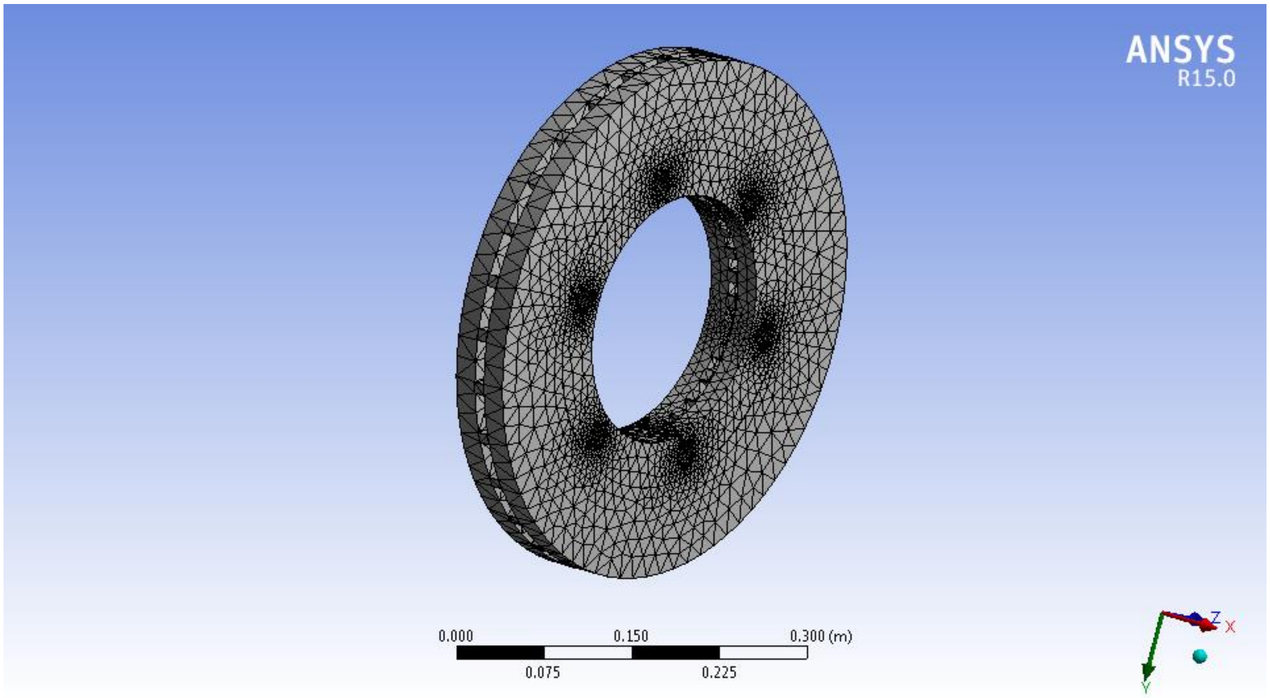


Figure 16 Meshed model of the ventilated brake disc

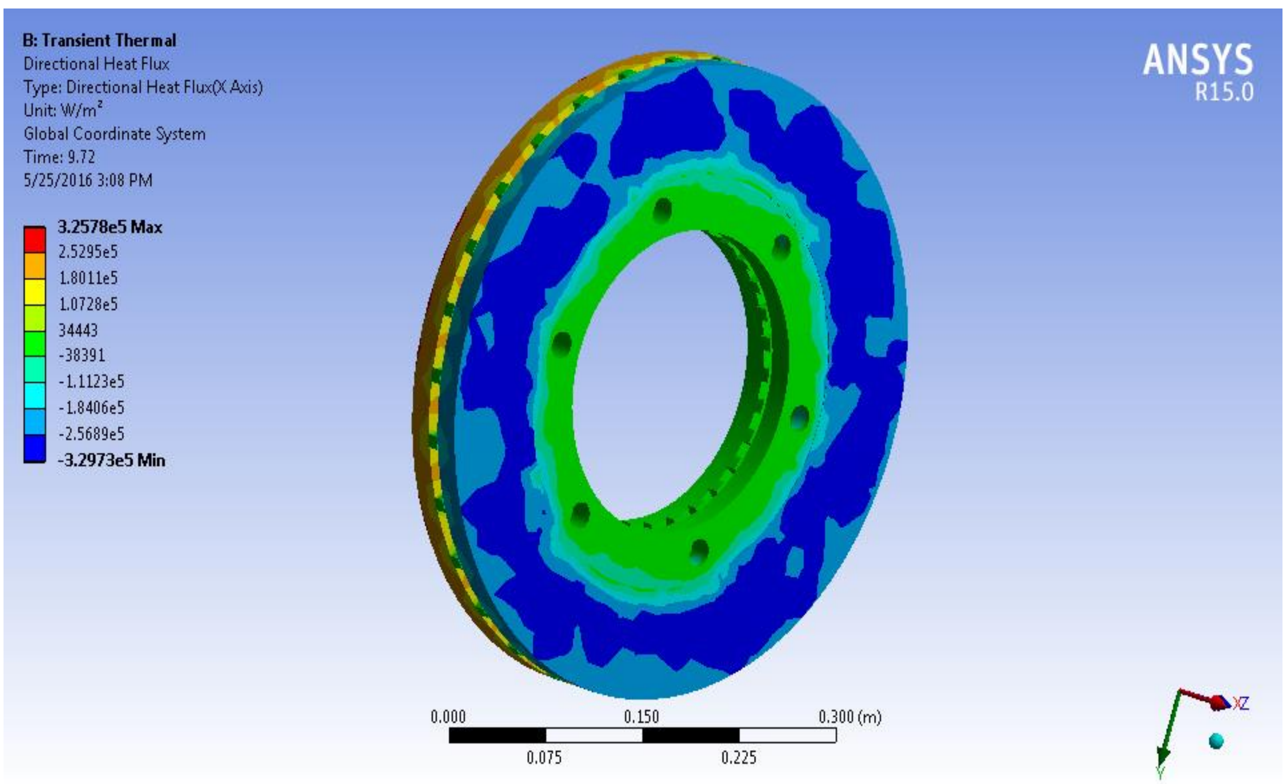


Figure 17 Directional heat flux for emergency braking from maximum velocity to standstill on a down gradient track after maintaining a constant velocity.

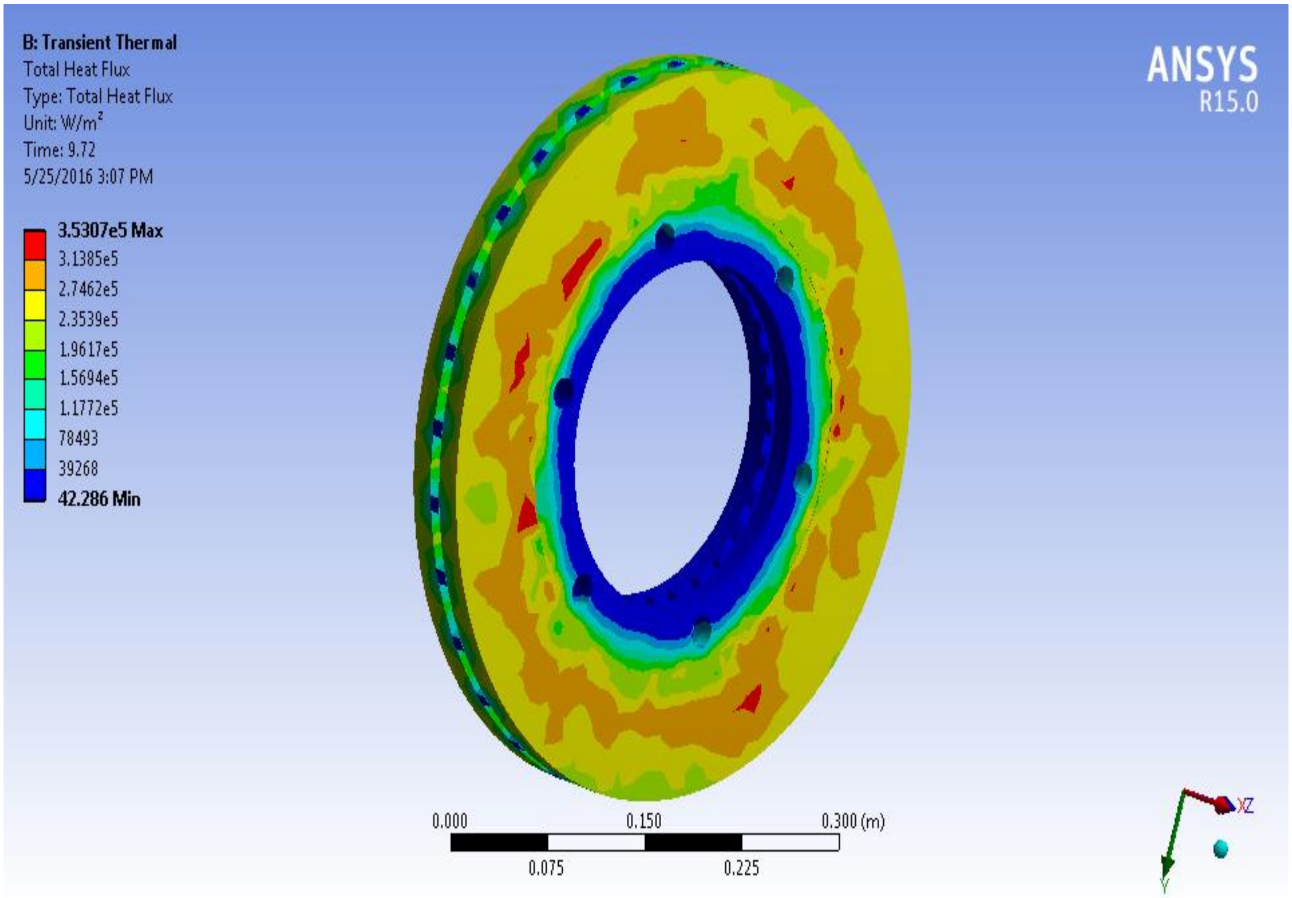


Figure 18 Total heat flux for emergency braking from maximum velocity to standstill on a down gradient track after maintaining a constant velocity.

After applying the necessary boundary and initial conditions defined by the rail vehicles operational parameters of the brake disc during emergency brake in the steep down gradient stations by fully coupling CFX fluid flow with transient thermal and transient structural analysis as follows:

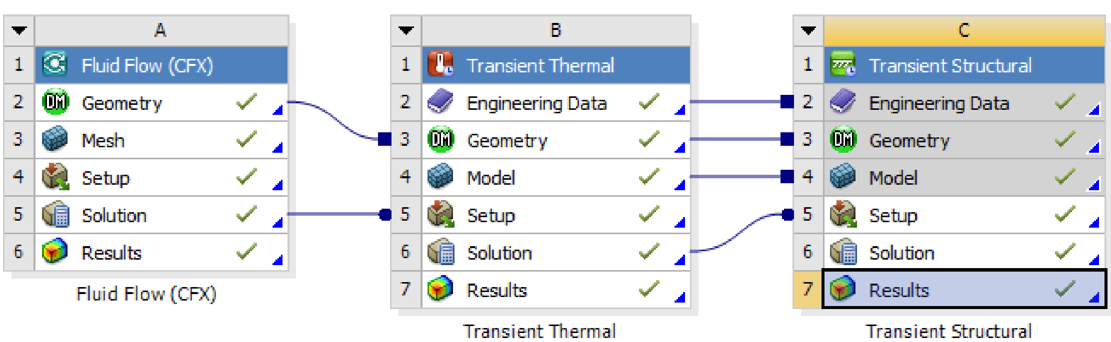


Figure 19 Coupled fluid flow (CFX), transient thermal and structural interaction

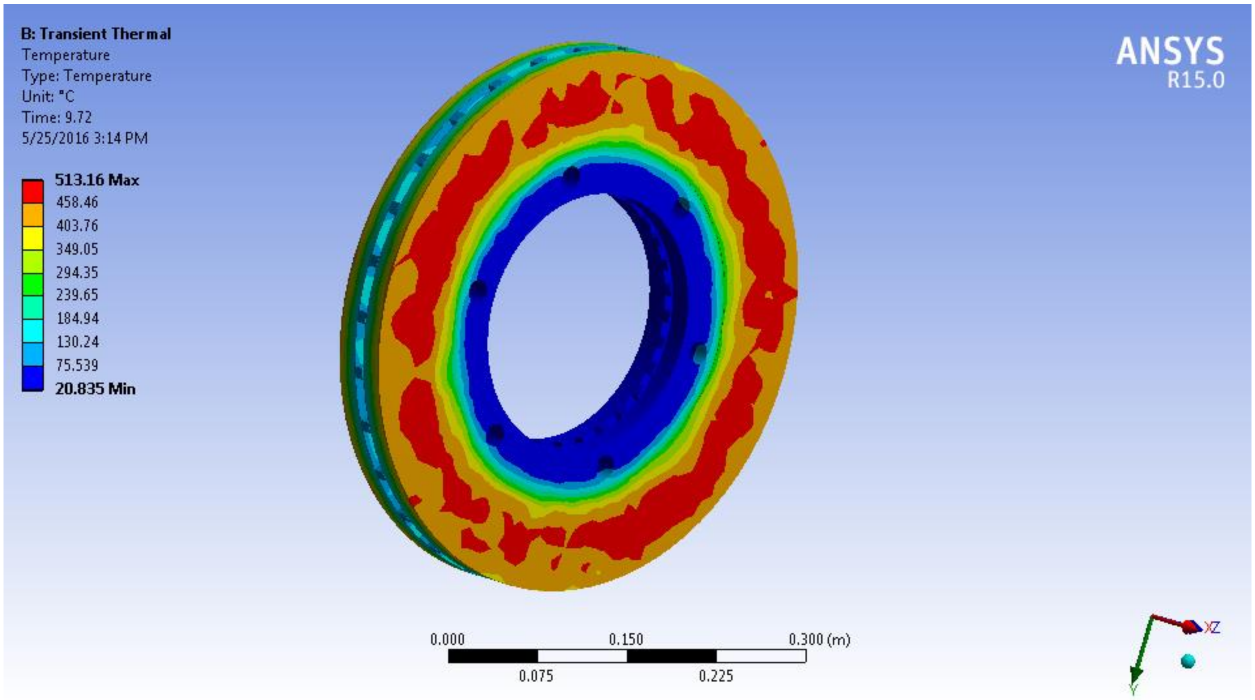


Figure 20 Temperature distribution of the C/SiC composite ventilated brake disc

4.7 Analysis of the stress

Pressure application for braking on a straight track has a value of 1,600,828.57pa.

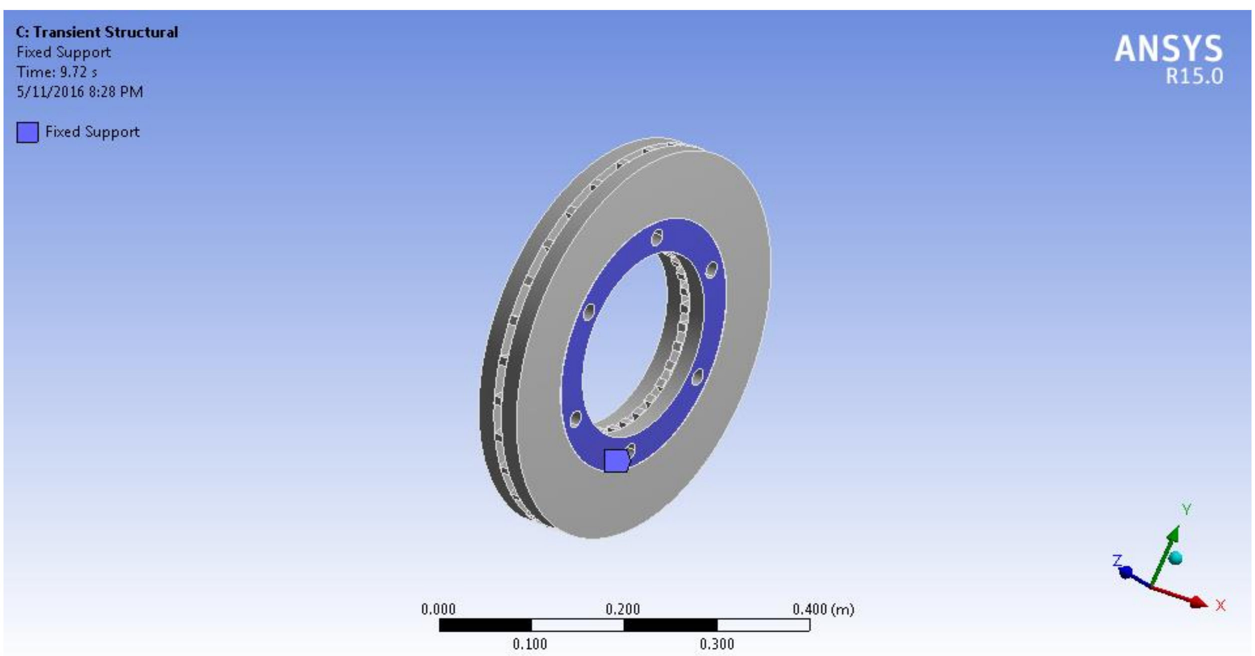


Figure 21 Load application for emergency braking from maximum velocity to standstill on a down gradient track.

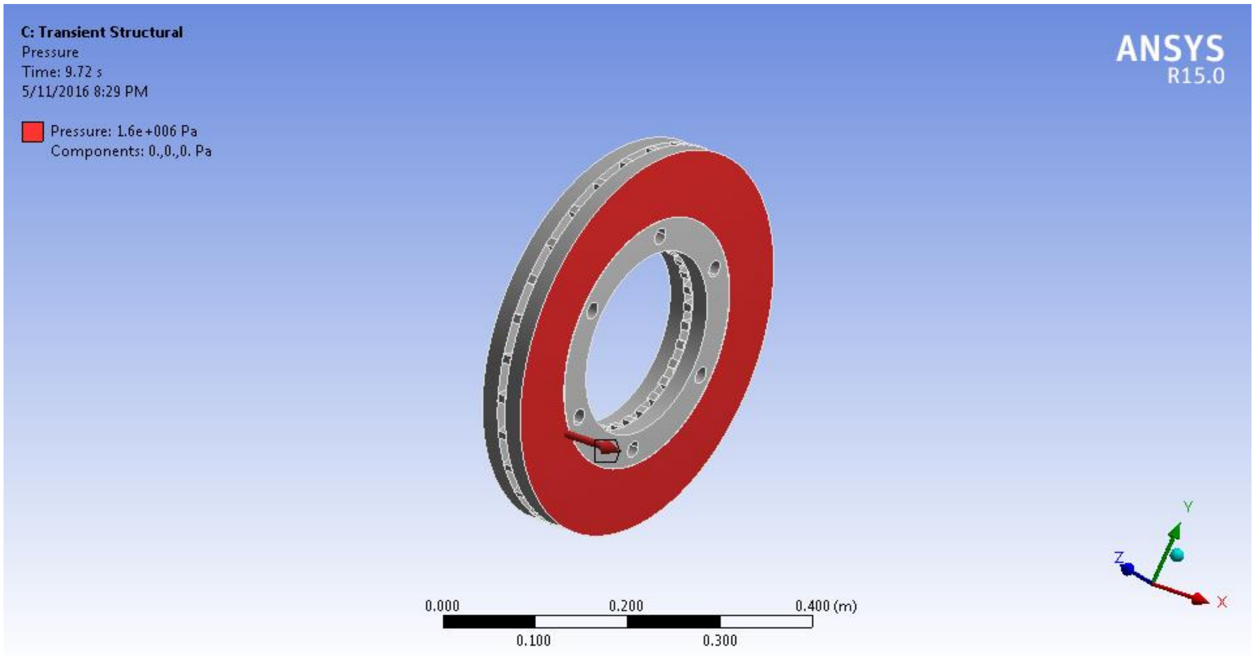


Figure 22 Pressure load application for emergency braking from maximum velocity to standstill on a down gradient track.

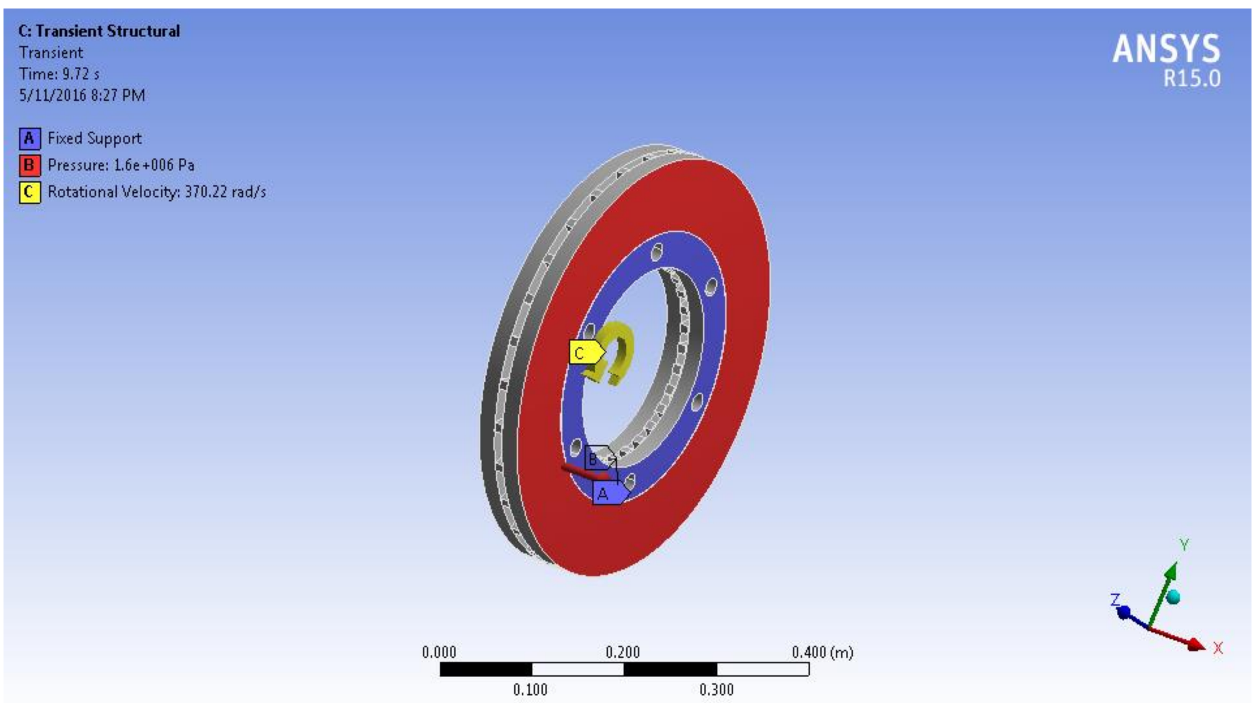


Figure 23 Loads application for emergency braking from maximum velocity to standstill on a down gradient track.

Imported body temperature resulting from thermal analysis of braking for maintaining a constant velocity and braking to standstill afterwards which becomes an input for the structural analysis.

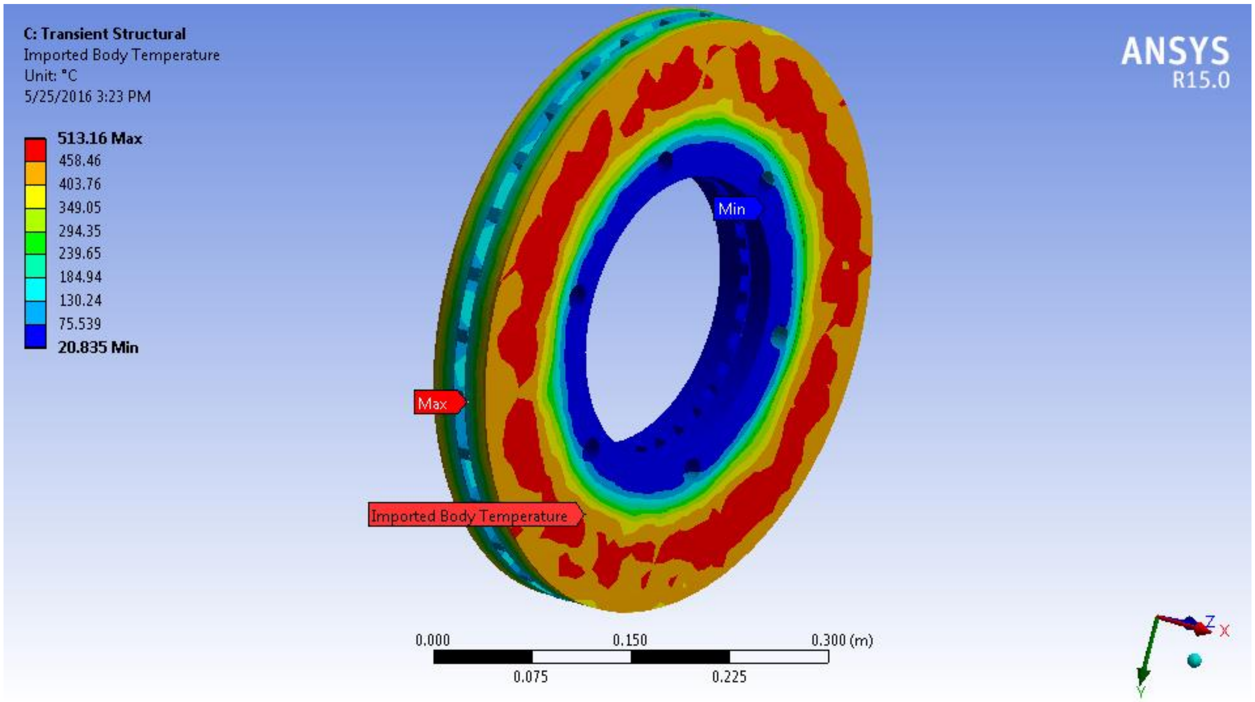


Figure 24 Imported body temperature of the ventilated brake disc for the structural analysis

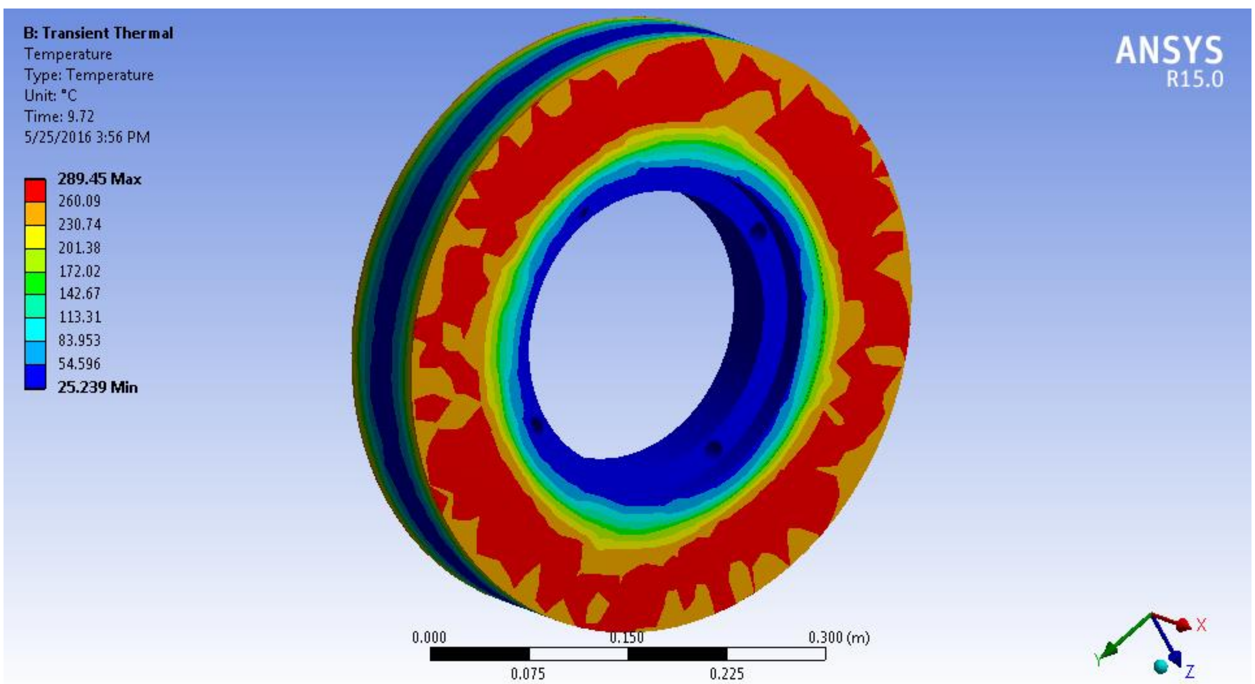


Figure 25 Transient temperature of the solid brake disc for structural analysis

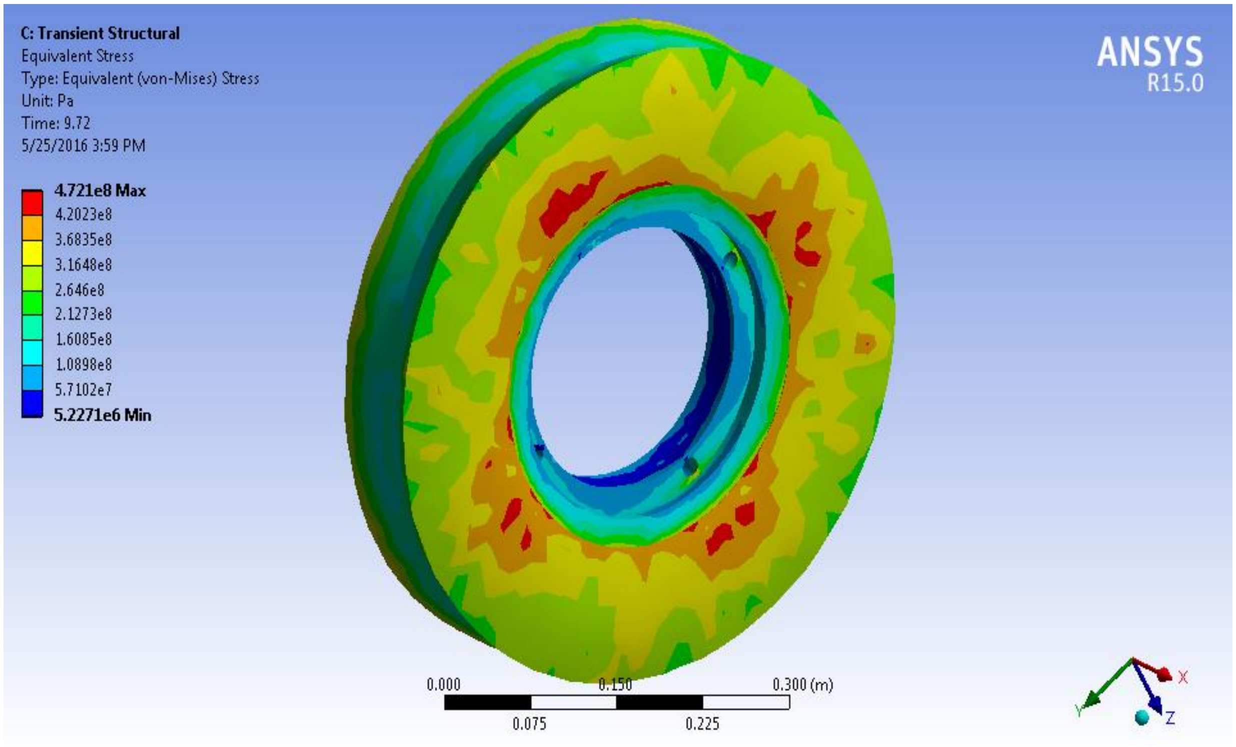


Figure 26 Equivalent (Von mises stress) of the SiC composite solid brake disc

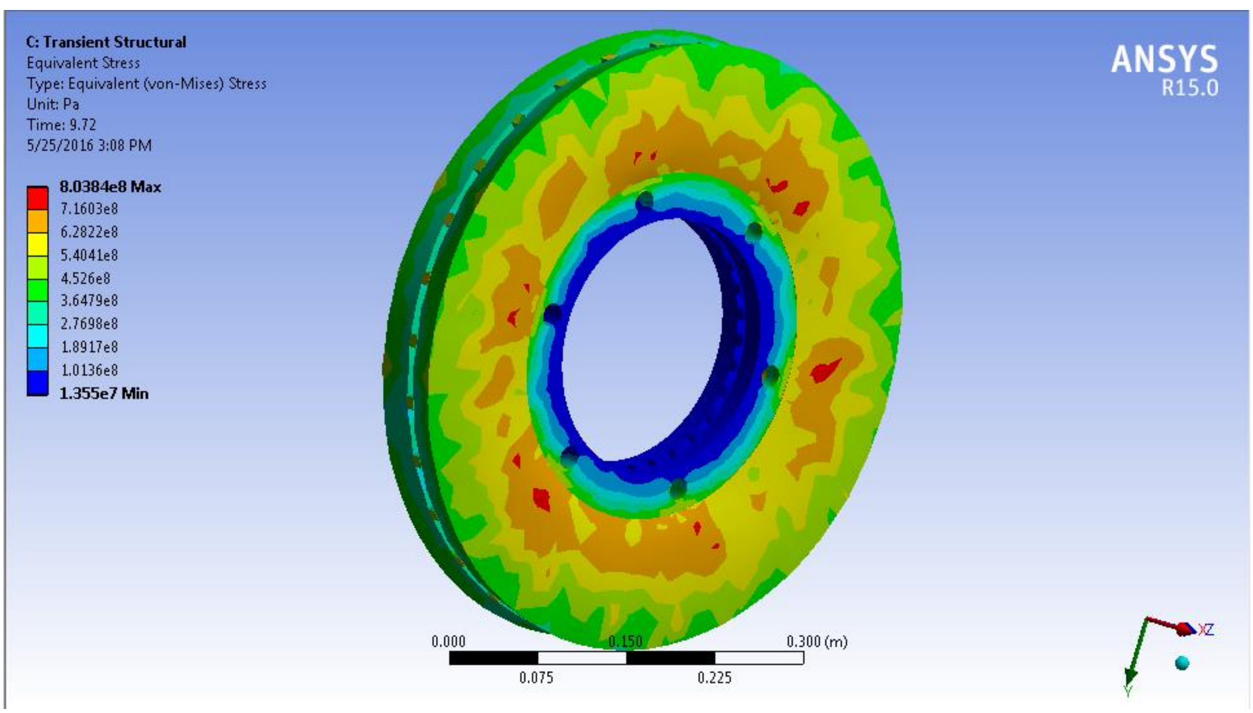


Figure 27 Equivalent (Von mises stress) of the SiC composite ventilated brake disc

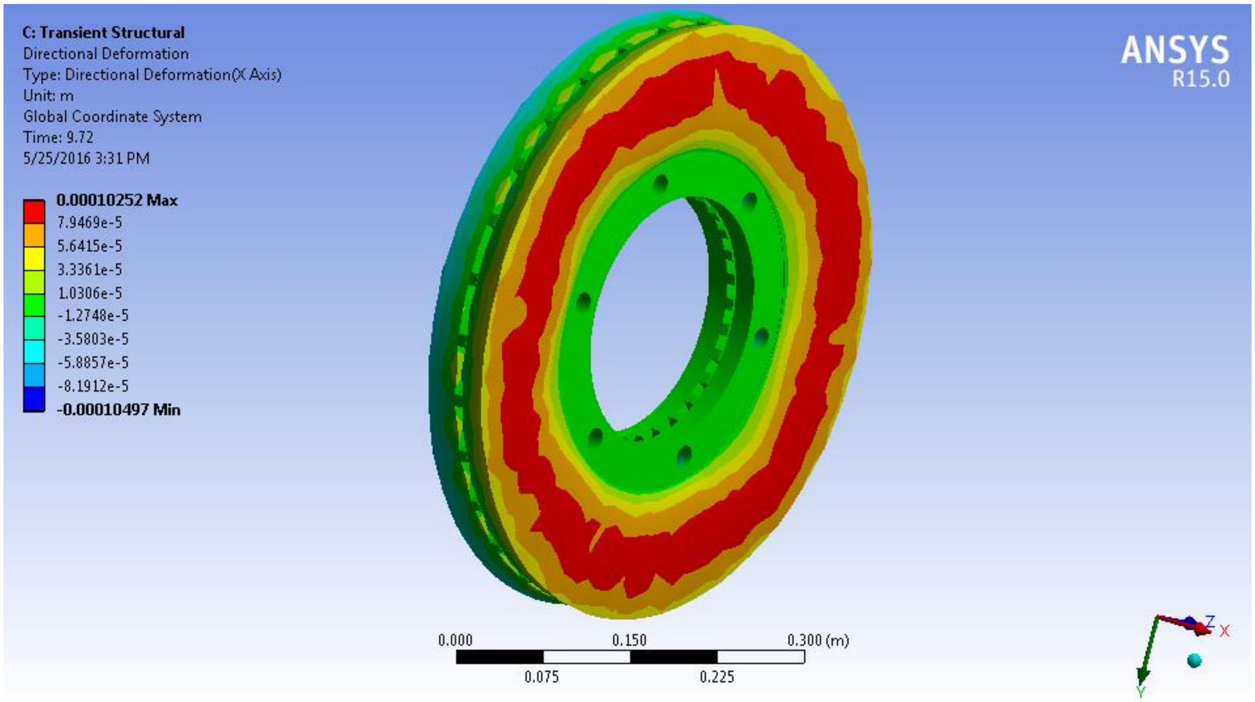


Figure 28 Directional deformation of the C/SiC composite ventilated brake disc in emergency braking

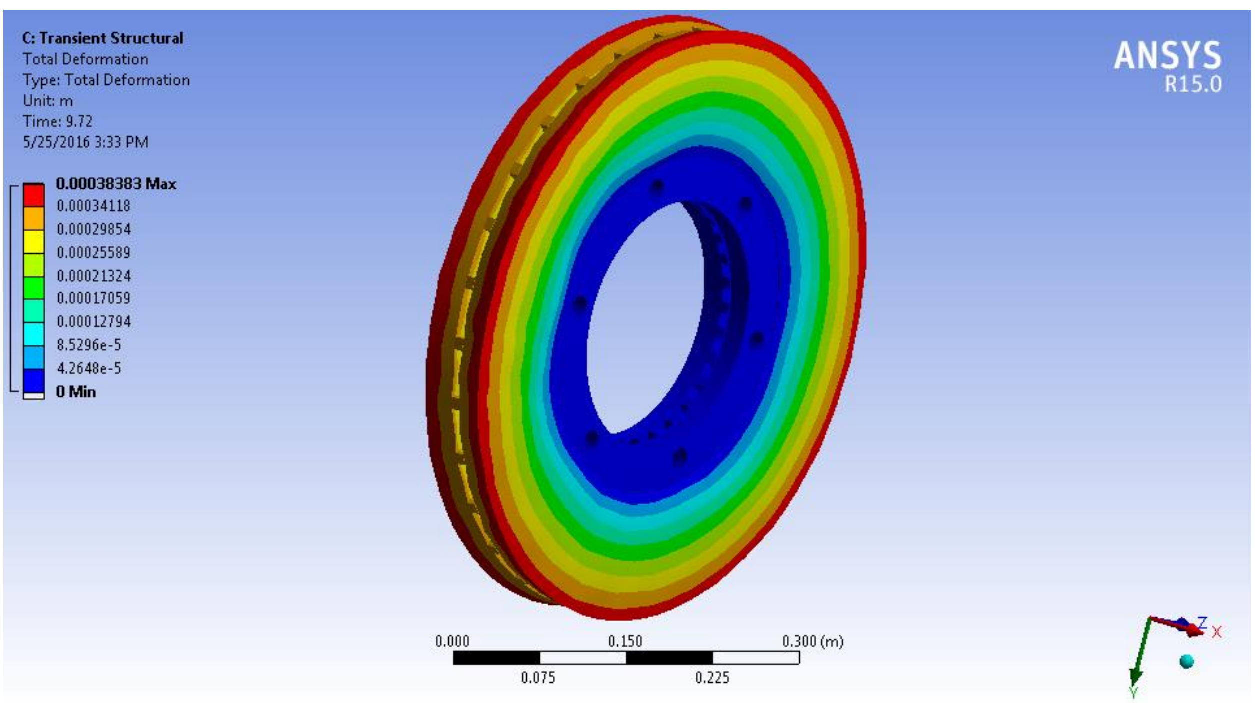


Figure 29 Total deformation of the C/SiC composite ventilated brake disc in emergency braking

The figures above show the variation of temperature, stress and deformation with time during the total time simulation of braking for both the full and ventilated discs.

Table 15 ANSYS analysis results summary

Wall heat transfer coefficient	Max: 8.961x 10^2W/m^2K Min: 0 W/m^2K
Imported temperature from CFX fluid flow for thermal analysis	Max:326.62℃ Min: 28℃
Directional heat flux	Max: 3.258x10^5W/m^2 Min: -3.297x10^5W/m^2
Total heat flux	Max: 3.53x10^5W/m^2 Min: 42.286
Ventilated disc temperature	Max: 513.16℃ Min: 20.835℃
Solid disc temperature	Max: 289.45℃ Min: 25.239℃
Equivalent(Von mises) stress for solid disc	Max: 472.1MPa Min: 5.227MPa
Equivalent(Von mises) stress for ventilated disc	Max: 803.84MPa Min: 13.55MPa
Directional deformation	Max: 0.03838mm Min: 0mm
Total deformation	Max: 0.01025mm Min: -0.010497mm

CHAPTER FIVE

CONCLUSION, RECOMMENDATION AND FUTURE WORK

5.1 CONCLUSION

This study attempted to investigate thermal and structural stress analysis of C/SiC composite brake disc without altering shape and dimensions of current AALRT brake disc geometry under existing operating parameters and boundary conditions. Coupled thermal and structural stress analysis of C/SiC composite brake disc for AALRT railway vehicle during emergency brake from speed of 70km/hr. with air flow cooling was studied using numerical and computational methods. To observe the material response in emergency braking with braking time 9.72s subjected to severe operating conditions (stopping on a down gradient from max. operating speed of 70km/hr.). In an emergency stop as most of the kinetic energy is transmitted to the discs as heat to be dissipated, the disc reaches a temperature of 513.16˚C.

Investigation on application of new materials is required to improve braking efficiency and provide vehicle stability. The C/SiC composite brake disc unlike grey cast iron has low density, good young's modulus, yield strength and thermal stability at elevated temperatures means longer service life of the brake disc. In this study, mass of the brake disc made from 30%C/SiC composite is 18.3 kg and mass of the grey cast iron for the current single AA LRT brake disc equals 62.332kg, thus 44kg mass of material will be reduced per single disc. Significant cost of energy and material will be saved though other factors such as life cycle cost and manufacturing should be known to make full comparison.

In the present study, analysis of C/SiC composite material during an emergency brake was done. The fluid structure interaction effect of still air convection on the rotating disc was analyzed in CFD CFX fluid flow of ANSYS 15.0 Workbench under stipulated conditions to obtain wall heat transfer coefficient.

A solid disc and straight vane ventilated disc were taken to determine temperature distribution, stress and displacement response of the brake disc during a single braking event. The results showed that maximum temperature obtained for solid disc was 289.45˚C and for the straight vane ventilated disc was 513.16˚C at the end of simulation time (9.72s). Moreover, the equivalent stress obtained for the solid disc was well below allowable yield strength of the material while for the ventilated disc it goes beyond the material strength. As simple radial straight vane was taken for the analysis here it doesn't indicate that full disc is better than ventilated one merely because shape optimization and structural modification was not considered. However, from the analysis results it was observed that convection has almost no effect for heat dissipation at such low speed and short braking time. Thus the analysis show promising results for the use of these materials.

Therefore, an extension of the design process to optimize aero-thermal performance modification can be done further for more accurate prediction of ventilation effect.

5.2 RECOMMENDATION

In this paper an effort was made to analyze thermal and structural effects of a solid and straight vane ventilated brake disc C/SiC composite material for AALRT emergency braking during decent on a down grade track from **St. ESTIFANOS** to **BAMBIS** stations from maximum speed of 70km/hr. to a complete stop using analytical and numerical approaches.

The main motive was to understand the C/SiC composite material stress and displacement behavior induced by thermal and mechanical loads in solid and straight vane ventilated forms. The results obtained are satisfactory where the stresses and deformations are well within the limits of the material strength. From the results of the study we recommend the Ethiopian Railway Corporation, other transport sectors and brake disc manufacturers to opt for C/SiC composite material for their brake disc application.

5.3 FUTURE WORK

The present analysis has not considered the effects of conduction and radiation heat transfer, humidity, shearing force, material tribology, wear analysis, manufacturing methods and production cost, residual stresses and the effect of subsequent periodic loads during brake disc life cycle and brake disc geometry profile variations together with its component and brake disc/pad interfaces to fully represent the real physical phenomenon. So as to better understand the characteristic behavior of C/SiC material for rail vehicle brake disc application the following works can be considered as an extension of this paper for further research.

- Conduct experimental investigation to ascertain validity of the analysis/simulation results correlation on a suitable reduced or full scale test rig.
- Life cycle cost analysis for optimum material selection among the brake disc rotor materials available.
- Study of better manufacturing routes to further reduce incumbent production costs.
- Wear, fatigue and fracture stresses shall also be studied to understand the material's response to such loads with different brake pad pair combinations.
- The material behavior response to corrosion, judder and brake squeal noise under assumed operating conditions.
- Study influence of different environmental conditions (wet, oil and dust contamination) on the brake disc.
- To study response of the brake disc material for residual stresses due to frequent brake application on each route stations.

REFERENCES

[1] Miloš S. Milošević, D. S. (2012). Modeling Thermal Effects in Braking Systems of Railway Vehicles, Thermal science, 16 (2).

[2] G. P. Voller. (2003). Analysis of heat dissipation from railway and automotive friction brakes, PHD thesis, 1-5.

[3] Coelingh E. (2007). Collision warning with auto brake – a real life safety perspective. Washington DC National Highway Traffic Safety Administration. 20th enhanced safety of vehicle conference, Lyon, France, 18–21, 07- 0450, 9.

[4] Oder G, Reibenschuh M, Lerher T, Sraml M, Samec B, Potrc I.(2009). Thermal and stress analysis of brake disks in railway vehicles [J]. Journal of Advanced Engineering, 3(1), 95−102.

[5] Asim Rashid, Simulation of Thermal Stresses in a Brake Disc, Linkoping Studies in Science and Technology, Licentiate Thesis No. 1603.

[6] Thundil karuppa raj.R, Ramsai.R and Mathew.J,Soniya.G.(2014). Numerical investigation of fluid flow and heat transfer characteristics on the aerodynamics of ventilated disc brake rotor using CFD, Institute of Technology, Vellore, Tamil Nadu, India, 18(2), 667 - 675.

[7] Daniel Das.A., (2013). Structural and Thermal Analysis of Disc brake in Automobiles. International Journal of Latest Trends in Engineering and Technology, 2(3), 18-19.

[8] Jiguang Chen and Fei Gao, Thermo-Mechanical Simulation of Brake Disc Frictional Character by Moment of Inertia, Dalian Jiao tong University, Liaoning, 116028, China.

[9] S. Koetniyom. (2010). Thermal Stress Analysis of Automotive Disc Brakes. PhD Thesis, the University of Leeds School of Mechanical Engineering.

[10] T. Hogskolan**.** (2012). Simulation of thermal stresses in a disc brake. Product Development and Materials Engineering. MSc Thesis, School of Engineering in Jönköping.

[11] Dufre´noy P. (2004). Two/three-dimensional hybrid model of the thermo-mechanical behavior of disc brakes. J Rail Rapid Transit, Part F 218, 17–30.

[12] YU Liang, JIANG Yan-li, RU Hong-qiang, LIU Jiang-tao, LUO Kun. (2011). Microstructures of co-continuous SiC/Fe-2Cr13 composite fabricated by vacuum-pressure casting and infiltration processes [J]. Advanced Materials Research, 239−242, 1661−1664.

[13] Khong Keng Leng, AM34. (2007/2008). Simulation of Temperature Distribution in Brake Discs, National University of Singapore.

[14] Ali Belhocine, Mostefa Bouchetara. (2012). Simulation of fully coupled thermo-mechanical analysis of disc brake rotor, University of Sciences and the Technology of Oran, WSEAS Transactions on applied and theoretical mechanics, USTO 31000 Oran Algeria, 7(3), 169- 181.

[15] Mijuca, D. M., iberna A. M., Medjo B. I. (2005). A new multi-field finite element method in steady state heat analysis. Therm. Sci. 9(1), 111-130.

[16] Nakatsuji, T. Okubo, K. Fujii.T, Sasada, M. Noguchi, Y. (2002). Study on Crack Initiation at Small Holes of One - piece Brake Discs. SAE Technical Paper, Inc., Humble, 2 002-01-0926.

[17] Noyes, R. N. and Vickers P. T. **(**1969). Prediction of Surface Temperatures in Passenger Car Disc Brakes, SAE Technical Paper Series 690457.

[18] J. G. Balotin. The influence of temperature on the friction coefficient of friction materials ABCM Symposium Series in Mechatronics, 4, 898-906.

[19] G. Babukanth & M. Vimal Teja. Transient Analysis of Disk Brake By using ANSYS Software, 23-24.

[20] LU S K., LIU H Y, YU L., JIANG Y L. (2011). 3D FEM simulations for the homogeneity of plastic deformation in aluminum alloy HS6061-T6 during ECAP [J]. Procedia Engineering, 12, 35−40.

[21] Yang Y, Zhou J M. (2007). Numerical simulation study of 3-D thermal stress field with complex boundary [J]. Journal of Engineering Thermo physics, 27(3), 487−489.

[22] Baughn J W., Shimizu S. (1989). Heat transfer measurements from a surface with uniform heat flux and an impinging jet [J]. ASME Journal of Heat Transfer, 111, 1096−1098.

[23] Skirl S., Hoffman M., Bowman K., Wiederhorn S., Rodel J. (1998). Thermal expansion behavior and macro strain of 3D-SiC/Al composites with interpenetrating networks [J]. Acta Mater, 46(7), 2493−2499.

[24] M. R., D. G., V.Chengal Reddy (2013). Modeling and analysis of FSAE car disc brake using FEM, 3, (9).

[25] Nitesh Ahuja, (2002). Some Studies on Modeling and Simulation of a Disc Brake System for Squeal Prediction, Bangalore University.

[26] Saran Jintanon S.R.M. Nagar, Kattankulathur. (2009). Force and friction on disc brake analysis. Kanchipuram District- 603 203.

[27] Zhang C Q, Zhang L T, Zeng Q F, Fan S W, Cheng L F.(2011). Simulated three dimensional transient temperature field during aircraft braking for C/SiC composite brake disc [J]. Materials & Design, 32(5): 2590−2595.

[28] R.M. Wang and M.K. Surappa. (1998). Microstructure and interface structure studies of SiCp reinforced Al (6061) metal-matrix composites Materials Science and Engineering. A, 254 (1-2), 219-226.

[29] Yu Liang, Jiang Yan-li, Ru Hong-qiang, Liu Jiang-tao, LUO Kun. (2011). Microstructures of co-continuous SiC/Fe-2Cr13 composite fabricated by vacuum-pressure casting and infiltration processes [J]. Advanced Materials Research, 239−242: 1661−1664.

[30] Miracle, D. B. (2005). Metal matrix composites – from science to technological significance. Composites Science and Technology 65: 2526-2540.

[31] Lim, D. W., Kim, T. H., Choi, J. H., Kweon, J. H. & Park, H. S. (2008). A Study of the Strength of Carbon–Carbon Brake Disks for Automotive Applications. Composite Structures 86 (1-3): 101-106.

[32] Savage, G. 2009. Sub-Critical Crack Growth in Highly Stressed Formula 1 Race Car Composite Suspension Components. Engineering Failure Analysis 16 (2): 608-617.

[33] Jacobsson, H. (2003). Aspects of Disc Brake Judder. Proceedings of the Institution of Mechanical Engineers Part D 217: 419-430.

[34] Krenkel, W. & Berndt, F. (2005). C/C-SiC composites for space applications and advanced friction systems. Material Science Engineering A412: 177-81.

[35] Zhuan, L. I., Peng, X., Xiang, X., & Su-hua, Z. H. U. (2008). Tribological characteristics of C / C-SiC braking composites under dry and wet conditions. Transactions of Nonferrous Metals Society of China, 18(5): 1071–1075.

[36] Xiao, P., Li, Z., & Xiong, X. (2010). Microstructure and tribological properties of 3D needle-punched C/C–SiC brake composites. Solid State Sciences, 12(4): 617– 623.

[37] K.W. Hee and P. Filip. (2005). Performance of ceramic enhanced phenolic matrix brake lining materials for automotive brake linings**,** Wear, 259, (7-12), 1088-1096.

[38] Bueno, S., Micele, L., Melandri, C., Baudin, C. & De Portu, G. (2011). Improved Wear Behavior of Alumina–Aluminum Titanate Laminates with Low Residual Stresses and Large Grained Interfaces. Journal of the European Ceramic Society 31 (4): 475-483.

[39] Fu, Ping Rong Kimberly. (2014). Study and development of novel composite materials for the application of car brake rotor. M.Phil. Curtin University, Department of Mechanical Engineering, 40-42.

[40] Figure 30 Structure of the composite brake disc: (a) SEM micrograph of the SiC layer on the C/C composite base and (b) schematic representation of the composite disc.

[41] Bakar Abu, Abd. R. et al. (2010). Thermal Analysis of a Disc Brake Model Considering a Real Brake Pad Surface and Wear, Int. J. Vehicle Structures & Systems, 2(1), 20-27.

[42] Grieve D. G., Barton D. C., Crolla D. A., Buckingham J. T. (1998), Design of a lightweight automotive brake disc using finite element and Taguchi techniques, Proc.Instn. Mech. Engrs., 212(4), 245-254.

[43] Floquet, A. and Dubourg, M.-C. (1994). Non Axis Symmetric Effects for Three Dimensional Analyses of a Brake, ASME J. Tribology, 116, 401-407.

[44] http://www.railelectrica.com

[45] UIC 541-3, Brakes — Disc brakes and disc brake pads — General conditions governing bench tests.

[46] http://www.csrgc.com.cn

[47] (September 2009). Addis Ababa LRT N-S route station's gradients Addis Ababa (E-W and N-S) route light rail transit project, Ethiopian Railways Corporation.

[48] Oder, G.; Reibenschuh, M.; Lerher, T.; Šraml, M.; Šamec, B.; Potrč, I. (2009). Thermal and Stress Analysis of Brake Discs in Railway Vehicles, Advanced Engineering, 3(1), ISSN 1846-5900.

[49] Ali Belhocine and Mustafa Bouchetara. (August, 2012). Investigation of temperature and thermal stress in ventilated disc brake based on 3D thermo-mechanical coupling model.

[50] Adam Adamowicz, Piotr Grzes. (2011). Analysis of disc brake temperature distribution during single braking under non-axisymmetric load. Applied Thermal Engineering, Elsevier, 31 (6-7), pp.1003.<10.1016/j.applthermaleng.2010.12.016 >.< hal-00723969>

[51] Ali Belhocine, and Mustafa Bouchetara. (June, 2013). Thermo- mechanical modeling of Disc Brake Contact Phenomena, transaction on control and mechanical systems, 2,(6), 302-309.

[52] Burmeister, L.C. (1993). Convective Heat Transfer. 2nd ed., New York: Wiley-Inter science.

[53] Holman, J.P. (1997) Heat Transfer. 8th ed., London: McGraw-Hill.

[54] Apte, A.A. and H. Ravi. (2006). FE Prediction of Thermal Performance and Stresses in a Disc Brake System, in Commercial Vehicle Engineering Congress and Exhibition, SAE International.

[55] Tian, X. and J.F.E. Kennedy. (1993). Contact Surface Temperature Models for Finite Bodies in Dry and Boundary Lubricated Sliding. Journal of Tribology. 115(3), 411-418.

[56] Day, A.J. (20080 Friction and Friction Materials, in Braking of Road Vehicles 2008, A.J. Day and B.R. Shilton, Editors, University of Bradford.

[57] Madhusudana, C.V. (1996). Thermal Contact Conductance. Mechanical Engineering Series, ed. F.F. Ling, New York: Springer.

[58] Tirovic, M. and G.P. Voller. (2005) Interface pressure distributions and thermal contact resistance of a bolted joint. Proceedings of the Royal Society a-Mathematical Physical and Engineering Sciences, 461(2060), 2339-2354.

[59] Majcherczak, D., P. Dufrenoy, and M. Nait-Abdelaziz. (2005) Third body influence on thermal friction contact problems: Application to braking. Journal of Tribology-Transactions of the ASME, 127(1): p. 89-95.

[60] Daoud A, Abouel-Khair M T. (2010). Wear and friction behavior of sand cast brake rotor made of A359-20vol%SiC particle composites sliding against automobile friction material. Tribology International, 43, 544−553.

[61] Saeed Moaveni. (1999). Finite Element Analysis theory and application with ANSYS, Minnesota state university, Mankato, Prentice-Hall, Upper Saddle River, New Jersey, p.2-6, ISBN0-13-785098-0.

[62] ANSYS 15 users' manual guide.

[63] Nestor, C. (2004).CFD investigation of the aerodynamics of ventilated automotive brake discs, MSc Thesis, School of Engineering, Cranfield University.