

ADDIS ABABA UNIVERSITY
ADDIS ABABA INSTITUTE OF TECHNOLOGY
SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING
GRADUATE PROGRAM IN RAILWAY ENGINEERING

Analysis of the performance of disc brake system of Addis Ababa light rail transit using temperature and coefficient of friction as a parameter

A thesis submitted to the
School of Graduate Studies of Addis Ababa University
in partial fulfillment of the requirements for the award of the degree of
Master of Science in Mechanical Engineering (Rolling Stock Engineering)

By

Andinet Kumella

Advisor

Dr. Ing Demis Alemu

Co-Advisor

Ato Tsegaye Feleke

September, 2014

Addis Ababa, Ethiopia

Addis Ababa Institute of Technology

School of Mechanical and Industrial Engineering

Analysis of the performance of disc brake system of Addis Ababa light rail transit using temperature and coefficient of friction as a parameter

By

Andinet Kumella

Approved by Board of Examiners

Birhanu Beshah (Dr-Ing)	_____	_____
Head, Railway center	Signature	Date
Demis Alemu (Dr-Ing)	_____	_____
(Advisor)	Signature	Date
Tsegaye Feleke	_____	_____
(Co-advisor)	Signature	Date
Tolossa Debrie	_____	_____
Internal examiner	Signature	Date
Fasil Gessesew	_____	_____
External examiner	Signature	Date

DECLARATION

I hereby declare that the work which is being presented in this thesis entitled as "Analysis of the performance of disc brake system of Addis Ababa light rail transit using temperature and coefficient of friction as a parameter" is original work of my own. The work contained in this thesis has not been previously submitted for a degree or diploma at any other higher education institution. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made and all the resource materials used for this thesis had been accordingly acknowledged.

_____ _____

Andinet Kumella Date

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

_____ _____

Dr.-Ing Demis Alemu Date

(Advisor)

_____ _____

Ato Tsegaye Feleke Date

(Co-Advisor)

ABSTRACT

Most of railroad passenger vehicles in the world use disc brake system which is mounted on the axle but some uses brake shoe directly applied onto the wheels treads. This system causes wheel heating, which in turn gives rise to thermal stresses and loss of mechanical strength. On the contrary, brake disc and pads, do not heat the wheels as shoe brakes do. Braking performance of trains is mainly affected by temperature and μ. This Research work presents the development of a modular tool for the prediction of passengers train braking performance. Particular attention is devoted to the accurate prediction of stopping distance, determination of nodal temperature and friction coefficient, considering different loading and operating conditions, necessary to verifying safety requirements for the movement of trains as a whole. Analysis was done on thermal transient of a brake disc for railway vehicles using finite element analysis method (FEM) simulation software called ANSYS. Performed analysis deals with two cases of braking; the first case considers service braking to a standstill; the second case considers emergency braking to a standstill and maintaining a constant speed. Once the model of the disc brake is done on solid works for simulation purpose, then simulation was followed using Multiphysics Ansys workbench. In general analysis results shows that, the performance of the disc brake is highly affected during emergency brake due to minimum stopping distance and delay time, results the increment value of coefficient of friction; although the nodal temperature is also high compared to service braking condition but it is less than the maximum temperature achieved by the disc in the literature review. However for the case of service brake its performance is good for all stations. Therefore validation results were encouraging using experiments, dynamometer test, and allowed a more accurate identification of the friction factor and temperature influence on braking performance of passenger trains as a whole.

Thanks mom and my sister

ACKNOWLEDGEMENTS

I would like to thank my M.Sc. Thesis Advisors Dr.Ing Demis Alemu and Mr. Tsegaye Feleke for their support and guidance.

I am using this opportunity to express my gratitude to everyone who supported me throughout the course of this M.Sc. Thesis. I am thankful for their aspiring guidance, invaluably constructive criticism and friendly advice during the Thesis work. I am sincerely grateful to them for sharing their truthful and illuminating views on a number of issues related to the thesis.

Finally thanks to you all who provided me with the facilities being required and conductive conditions for my M.Sc. Thesis.

TABLE OF CONTENTS No

Abstract …….....III

Acknowledgements ……V

Table of contents……VI

List of Figures……X

List of Tables……XII

List of Graphs…….....XIII

Notations…….....XIV

CHAPTER- 1 INTRODUCTION……1

1.1 Background……1

1.2 Statement of the Problem…….....5

1.3 Research Questions……6

1.4 Objective of the study……7

1.4.1 General objective……7

1.4.2 Specific objective……7

1.5 Significance of the study……7

1.6 Scope and limitation of the study……7

1.7 Methodology……...8

1.7.1 Data and Data Source…….....8

1.7.2 Method of Data Collection…….....8

1.7.3 Method of Analysis……8

1.7.4 Research Method……8

1.8 Organization of the research……...9

CHAPTER-2 LITERATURE REVIEW…….....10

2.1 Introduction……...10

2.2 Types of train brake systems……12

2.2.1 Air brake……12

2.2.2 Electromagnetic rail brake……..12

2.2.3 Rail (eddy current) braking systems……...13

2.2.4 Rheostatic braking (dynamic braking)……....13

2.2.5 Regenerative braking……14
2.3 Braking capacity…….....15
2.4 Disc brake…….....15
2.4.1 Disc brake components…….....16
2.4.2 Materials for brake disc and brake pad…….17
2.5 Braking distance calculation tool……..19
2.6 Heat flux entering the disc……..……...21
2.7 Finite Element Formulation /Thermal Transient Analysis/ of the Disc Brake……..22
2.8 Related research……23
CHAPTER-3 DATA COLLECTION AND ANALYSIS…….27
3.1 Calculating the slopes and/or gradient for selected Addis Ababa E-W stations……..27
3.2 Addis Ababa lrt E-W rout gradient …….....27
3.3 selected stations……28
3.4 Addis Ababa lrt e-w distance between consecutive stations…….....29
3.5 Loading Capacity/Carrying Capacity of train ……..30
3.5.1 Train weight ……..30
3.6 Equation of motions at CMC1……..31
3.6.1 Calculation of Tractive Force [F_T]……32
3.6.2 Calculation of Drag Force [F_D]…….32
3.6.3 Calculation of Rolling Resistance Force [F_R]……..33
3.6.4 Calculation of Normal component of the force in the direction of the inclined plane [F_G]……..34
3.6.5 Calculation of braking force of the wheel [$F_{B,W}$] ……....35
3.6.6 Calculation of Drag Force [F_D] ……36
3.6.7 Calculation of Rolling Resistance Force [F_R]……...37
3.6.8 Calculation of normal component of the force in the direction of the inclined plane [F_G] …….37

3.6.9 Service and emergency brake for Addis Ababa lrt at station between CMC 2 and CMC 1…….38
3.7 Equation of motion for Ayat 1……...40
3.7.1 Service and emergency brake for Addis Ababa lrt at station: - between Ayat 1 and CMC 2/Meri……..42
3.8 Equation of motion at Ayat 2……...43
3.8.1 Excel file for service and emergency brake for Addis Ababa lrt at station: - between Ayat 2 and Ayat 1…….....44
3.9 Summary of distance vs. speed graph at three stations ……..46
3.10 Heat generated at Ayat 1 during service and emergency brake……..46
3.11 Determination of friction coefficient……..49
3.11.1 Ayat 1 service brake……..49
3.11.2 Ayat 1emergency brake……50
3.11.3 Meri/CMC 2/ service brake……..51
3.11.4 Meri/CMC 2/ emergency brake……52
3.11.5 CMC 1 service brake……53
3.11.6 CMC 1 emergency brake……..54
CHAPTER-4 PHYSICAL MODEL OF DISC BRAKE……...55
4.1 Material properties of grey cast iron……...55
4.2 Disc brake calculation.……56
4.3 Initial temperature for the next braking action ……..58
4.4 Disc brake assembly model in Solid Works……...61
CHAPTER -5 ANSYS RESULTS AND DISCUSSIONS…….....62
5.1 Modeling in Ansys ……62
5.1.1 Material……...62
5.1.2 Mesh Generation……63
5.1.3 Boundary Conditions……64
5.2 Thermal Analysis……...64
5.2.1 Types of Thermal Analysis…….....64
5.3 Results……65

5.3.1 Service brake for station Ayat 1……65
5.3.2 Emergency brake for station Ayat 1…….67
5.3.3 Service brake for station Meri/CMC 2/……68
5.3.4 Emergency brake for Station Meri/CMC 2/…….70
5.3.5 Service brake for Station CMC 1…….71
5.3.6 Emergency brake for Station CMC 1……...73
5.4 Results and Discussions……..74
5.4.1 Results Comparison……..74

CHAPTER -6 CONCLUSIONS AND RECOMMENDATION…….....80
6.1 Conclusion……..80
6.2 Recommendation…….....81
6.3 Future work…….....82

REFERENCES…….....83

APPENDIX……...86
Appendix A…….....86

LIST OF FIGURES

Figure 1.1. Train disc brake…….....5
Figure 1.2. Train shoe brake……..5
Figure 2.1. Train air brake system.……12
Figure 2.2. Train electromagnetic rail brake system…….....12
Figure 2.3. Train rail (eddy current) brake system…….....13
Figure 2.4. Train Rheostatic (Dynamic) brake system…….....14
Figure 2.5. Train Regenerative brake system.……...14
Figure 2.6. Caliper body…….....16
Figure 2.7. Free body diagram showing static force in brake caliper during braking…….....18
Figure 2.8. Forces acting on rotor due to contact with brake pads (variables Renamed from Fig 2.7)…….....18
Figure 2.9. Equation of motion at cmc1 during accelerated motion.……31
Figure 3.0. Equation of motion at cmc1 during decelerated motion.……...35
Figure 3.1. Result page for the brake distance, deceleration and brake time at cmc1……..38
Figure 3.2 Equation of motion for Ayat 1 during accelerated motion…….40
Figure 3.3 Equation of motion for Ayat 1 during decelerated motion…….41
Figure 3.4 Equation of motion at station Ayat 2 during accelerated motion……...43
Figure 3.5 Equation of motion at station Ayat 2 during decelerated motion……...44
Figure 3.6 Train disc brake assembly……..61
Figure 3.7 Disc brake solid work model.……..61
Figure 5.1 Brake disc Ansys model.……62
Figure 5.2 Meshing of the disc.……63
Figure 5.3 Nodal Temperatures at Ayat 1.……65
Figure 5.4 Total Heat Fluxes at Ayat 1.……66
Figure 5.5 Directional Heat Fluxes at Ayat 1.……..66
Figure 5.6 Nodal Temperatures at Ayat 1.……67
Figure 5.7 Total Heat Fluxes at Ayat 1.……67
Figure 5.8 Directional Heat Fluxes at Ayat 1.……..68

Figure 5.9 Nodal Temperatures at Meri……..68
Figure 5.10 Total heat fluxes at Meri……69
Figure 5.11 Directional heat fluxes at Meri……..69
Figure 5.12 Nodal Temperatures at Meri……..70
Figure 5.13 Total heat fluxes at Meri……70
Figure 5.14 Directional heat fluxes at Meri……..71
Figure 5.15 Nodal Temperatures at CMC 1……..71
Figure 5.16 Total heat fluxes CMC 1……...72
Figure 5.17 Directional heat fluxes CMC 1……..72
Figure 5.18 Nodal Temperatures CMC 1……73
Figure 5.19 Total heat fluxes CMC 1……...73
Figure 5.20 Directional heat fluxes CMC 1……..74

LIST OF TABLES

Table 1. Addis Ababa light rail transit east- west rail profile gradient…….27
Table 2. Selected stations……..28
Table 3. Addis Ababa LRT E-W distance between consecutive stations…….29
Table 4. Loading capacity/carrying capacity of train……...30
Table 5. Train Weight……...30
Table 6. Drag coefficient for passenger train……33
Table 7. Accelerated, normal service brake, normal emergency brake and constant motion between station CMC 2 and CMC 1……..38
Table 8. Accelerated, normal service brake, normal emergency brake and constant motion between station Ayat 1 and Meri…….....42
Table 9. Accelerated, normal service brake, normal emergency brake and constant motion between station Ayat 2 and Ayat 1……44
Table 10. Heat generation vs. time for service brake at Ayat1……..47
Table 11. Heat generation vs. time for emergency brake at Ayat1……47
Table 12 Coefficient of friction at Ayat 1 during service brake……...49
Table 13 Coefficient of friction at Ayat 1 during emergency brake…….50
Table 14 Coefficient of friction at Meri during service brake……..51
Table 15 Coefficient of friction at Meri during emergency brake……52
Table 16 Coefficient of friction at CMC 1 during service brake……..53
Table 17 Coefficient of friction at CMC 1 during emergency brake……54
Table 18. Vehicle properties……..55
Table 19. Material properties of grey cast iron……..55
Table 20. Results and comparison for the three stations……...74

LIST OF GRAPHS

Graph 1. Relationship between friction coefficient and sliding speed (a) and Brake-line pressure (b)…….....21
Graph 2. Summary of distance vs. speed graph at three stations for service and emergency brake……...46
Graph 3. Relationships between Qt and time for service brake at Ayat1……48
Graph 4. Relationships between Qt and time for emergency brake at Ayat1……..48
Graph 5 Relationships between μ and V for service brake at Ayat1……..49
Graph 6 Relationships between μ and V for emergency brake at Ayat1……50
Graph 7 Relationships between μ and V for service brake at Meri……51
Graph 8 Relationships between μ and V for emergency brake at Meri……..52
Graph 9 Relationships between μ and V for service brake at CMC 1……53
Graph 10 Relationships between μ and V for emergency brake at CMC 1……..54
Graph 11. Temperature vs. friction coefficient during service brake at Ayat1…….76
Graph 12. Temperature vs. friction coefficient during service brake at CMC1……76
Graph 13. Temperature vs. friction coefficient during emergency brake at Ayat1……...77
Graph 14. Temperature vs. friction coefficient during emergency brake at CMC1……..78

NOTATIONS

F_{TRI}	Tangential force between pad and rotor (inner face)
F_{TRO}	Tangential force between pad and rotor (outer face)
T_B,M	Brake torque
F	resultant force
m	mass
a	acceleration or deceleration
v	final velocity
v_0	initial velocity
S	distance
t	time
l_{brake}	braking distance
g	acceleration due to gravity
μ	coefficient of friction
α	gradient
q_x	conduction heat fluxes in x direction
q_y	conduction heat fluxes in y direction
q_z	conduction heat fluxes in z direction
Cp	specific heat
ρ	specific mass
Q	internal heat generation
T	temperature
K_x	thermal conductivity in x-directions
K_y	thermal conductivity in y-directions
K_z	thermal conductivity in z-directions
Q_s	Surface heat flux
h	convective heat transfer coefficient
T	unknown surface temperature
T_∞	convective exchange temperature
ED	elevation difference

HD	horizontal difference
ω	angular velocity
F_T	tractive force
F_R	rolling resistance
F_D	Drag force
F_G	Normal force component in the direction of the inclined plane.
N	Normal force
T_m	Motor torque
T_{wheel}	torque on the wheel
P	power of the motor
C_D	drag coefficient
ρ	density of fluid
V	velocity of passenger train
A	Characteristics frontal area of the body
f	rolling coefficient of friction
Dfd	disc outer diameter
W	avg. rolling resistance
ηmf	efficiency brake transmission
mfb	brake pad grip
z	Rotational coefficient
i	brake lever ratio
I_h, J	gear drive
Dmin	worn wheel diameter
EW	east west
ERC	Ethiopia railway corporation
LRT	Light Rail Transit
pr	Prandtl number
Nu	Nusselt number
k_a	thermal conductivity coefficient
μ_a	viscosity of air

CHAPTER-1

INTRODUCTION

1.1 Background

Many of the world's large cities grew in conjunction with railways, and today, large cities cannot depend only on motor vehicles for transportation. With worries over global environmental issues, public transportation systems are increasingly seen as an important way to expand and revitalize large cities, while consuming less energy and other resources [1].

Transport is one of the key sectors that play crucial roles in achieving the goals of poverty eradication and sustainable development. The transport sector is very much linked and influences developments in other sectors of the economy.

Road transport is the most dominant mode of motorized transport in Africa, accounting for 80 per cent of the goods traffic and 90 per cent of the passenger traffic on the continent. Road traffic accidents kill 1.2 million people in the world. Of this number, over 225,000, or 19 per cent, were accounted for by deaths on African roads. Moreover, Africa has the highest number of road traffic accidents per capita.

Railway transportation is safer than road mode of transport so development of railway has a big advantage.

In 2005, the continent had a total railway network of 90,320 km or 3.1 km of per 1,000 km2, most of which is disjointed. With the exception of North Africa, railways in Africa generally have a low level of traffic. The railways carry only one per cent of the global railway passenger traffic and two per cent of goods [2].

Ethiopia covers 1.1 M sq. km and it is vast to cover with only road transport. Addis Ababa's Transport problems are diverse aged fleet, chaotic movement of mini-bus taxis, environmentally unacceptable emission, and Unsafe, Hazardous to life and property [3].

Many of transportation problems are caused by their braking system failure. Therefore doing research on vehicles (train) braking system is crucial in order for the safe movement of the passengers, in general as well it has vital applications which are listed in the literature review and this paper as a whole.

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

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

At first, the rather low locomotives power and traction force allowed braking using quite simple handbrakes that equipped locomotives and eventually other vehicles of the train. As the development of rail transport and according to increasing traffic speeds, tonnages and length of trains, it was found that braking has to be centralized, operated from a single location - usually the locomotive driver's cabin and commands have to be correctly transmitted along the entire length of the train.

As a consequence, along the time, for railway vehicles have been developed various brake systems, whose construction, design and operation depend on many factors such as running speed, axle load, type, construction and technical characteristics of vehicles, traffic conditions, etc.

Among various principles and constructive solutions that were developed, following the studies and especially the results of numerous tests, the indirect compressed air brake system proved to have the most important advantages. Therefore, it was generalized and remains even nowadays the basic and compulsory system for rail vehicles.

The braking effort achievable was limited, and an early development was the application of a steam brake to locomotives, where boiler pressure could be applied to brake blocks on the locomotive wheels. However, it was also unreliable, as the application of brakes by guards depended upon them hearing and responding quickly to a whistle for brakes [4].

Brakes have been retuned and improved ever since their invention. The increases in travelling speeds as well as the growing weights of cars have made these improvements essential. The faster a rail car goes and the heavier it is, 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 [5].

During braking the energy absorbed by brakes is dissipated in the form of heat. This heat is dissipated into the surround atmosphere to stop the rail vehicle, so the brake system should have the following requirements.

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

The mechanical brakes is one type of operation brake by which according to the direction of acting force may be sub divided into the following two groups: radial brakes and axial brakes.

- **Radial brakes**

In these brakes the force acting on the brake drum is in radial direction. The radial brake may be subdivided into external brakes and internal brakes. E.g. tread brake/brake shoe/

- **Axial brakes**

In these brakes the force acting on the brake drum is only in the axial direction. E.g. Disc brakes, Cone brakes. Finally this research will focus on the following axial brake systems types listed below [6].

Disc brake

The disc brake is a wheel brake which slows rotation of the wheel by the friction caused by pushing brake pads against a brake disc with a set of calipers. The brake disc (or rotor in American English) is usually made of cast iron, but may in some cases be made of composites such as reinforced carbon–carbon or ceramic matrix composites. This is connected to the wheel and/or the axle. To stop the wheel, friction material in the form of brake pads, mounted on a device called a brake caliper, is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc. Friction causes the disc and attached wheel to slow or stop. Brakes convert motion to heat, and if the brakes get too hot, they become less effective, a phenomenon known as brake fade.

Disc-style brakes development and use began in England in the 1890s. Compared to drum brakes, disc brakes offer better stopping performance, because the disc is more readily cooled. As a consequence discs are less prone to the "brake fade"; and disc brakes recover more quickly from immersion (wet brakes are less effective). Most drum brake designs have at least one leading shoe, which gives a servo-effect. By contrast, a disc brake has no self-servo effect and its braking force is always proportional to the pressure placed on the brake pad by the braking system via any brake servo, braking pedal or lever, this tends to give the driver better "feel" to avoid impending lockup. Drums are also prone to "bell mouthing", and trap worn lining material within the assembly, both causes of various braking problems [6][7].

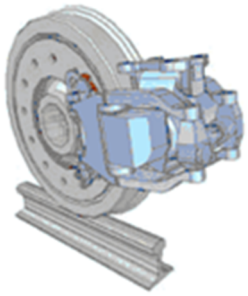


Figure 1.1 Train disc brake

Tread brake/brake shoe

Tread braking uses block-shaped brake shoes made of cast iron or other material. Then these brake shoes are pressed onto the running wheel treads. Then the train's kinetic energy is converted into heat energy by the mechanical friction between wheel treads and brake shoes. The heat generated is then dissipated into the atmosphere and the braking force is created.

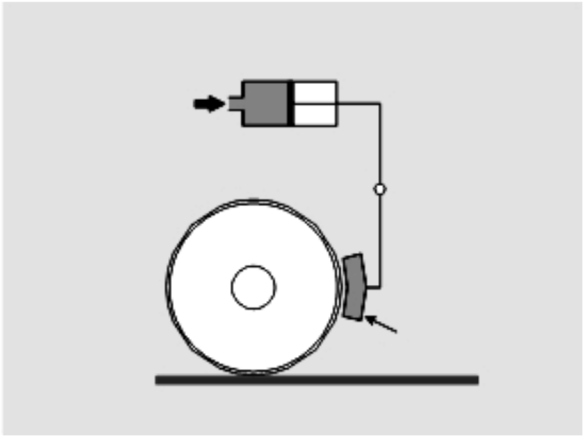


Figure 1.2.Train shoe brake

These several issues, even briefly presented above, reveal not only the importance and complexity of braking systems but here rely on the above idea as a framework the research will focus particularly on disc braking systems used for rail vehicles, and also the necessity of knowledge and understanding of their analysis.

1.2 Statement of the problem

One of the most and frequently happening complex issue involved in the railway transportation is associated with performance of the braking system (disc) for passenger trains as well as freight.

As outlined by [8], transport reporter in Melbourne in November 16, 2006 some of Melbourne's newest passenger trains have had to be withdrawn from service after a spate of braking failures. The problems were connected to trains computerized braking systems so here the movement of the train depends on the performance of manual braking system.

Some researcher argue that disc brake has some problems like scarring, cracking, warping or excessive rusting therefore they choose shoe brake for trains but on the other hand other researchers says disc brake not get too hot they become less effective to a phenomenon known as brake fade and is more efficient [7].

But doing research on analyzing the performance of railroad passenger vehicle disc brake system will have an advantage to solve many problems regarding braking system performance which are listed in this research and different literature's, in addition to saving maintenance cost. Therefore this research will address and clear out the following issues:-

- The way how to analyze the performance of disc braking system.
- How to calculate disc brake system for stopping distance using computer program?
- Is disc brake system providing safe braking action?
- How to calculate disc brake system for delay time of the trains using computer program?
- How to determine the number of discs on the axle by making thermal transient Ansys 14.5 simulation?

1.3 Research Questions

- In what manner does the performance of disc brake system will be analyzed to stop the train for both service and emergency braking conditions?
- Why does the ERC choose disc braking system?
- What are the problems faced by the ERC by imposing this brake system?
- What are the mechanisms that will follow to stop the Addis Ababa LRT if the braking system is disc?
- How disc brake systems provide better stopping distance?

1.4 Objective of the study

1.4.1 General objective

The main objective of the research is to make an analysis on the performance of braking systems of Addis Ababa LRT. This will help us to make further research to improve performance on braking system for the newly constructed Addis Ababa LRT.

1.4.2 Specific objective

The aim at the end of this research is by making transient thermal analysis together with numerical calculation of the performance of disc braking systems on Addis Ababa LRT using simulation software Ansys 14.5, the research analyze the performance of disc brake system by making Nodal temperature and the demand of coefficient of friction, μ as a parameter at selected E-W route line stations. Identifying the station with good disc brake performance and for some stations if there is a scenario of obtained nodal temperature is higher than temperature limit value specified in literature review make appropriate recommendation to improve its performance for the newly constructed A.A LRT.

1.5 Significance of the study

The output of this research firmly believes that it has the following importance:

- The finding of this research will serve as an input to a more advanced and in depth study in the subject for the future.
- Generally this research will be a great help to the ERC in promoting an efficient brake system design considering the prevailing facts that can alleviate the problem.
- The finding of this research would enable the ERC to strive for and facilitate further researches to be done on this research area for better performance improvement of braking system.
- Finally it helps the authority ERC to have documents that is used for the future as a reference for researchers.

1.6 Scope and Limitation of the study

The scopes of the research are:-

- To do finite element analysis on the brake disc.

- Literature review on the working principles, components, standards and theories.
- Construction of 3D model of disc brakes using software Solid work.
- FE model (meshing of geometry model) computer simulation software called ANSYS
- Finite element analysis using ANSYS workbench on thermal transient analysis which shows the temperature distribution of the disc brake rotor.
- Final result and conclusion.

Limitations of the research are:-

Some of the limitation that might be encountered when conducting the research includes.

- Time: data collection, analysis will require more time to find good output. In general, this research may require ample time.
- Data: Finding the required data might not be available in neat and good justification.

1.7 Methodology

1.7.1 Data and Data source

Primary and secondary data from ERC and other sources will be used.

1.7.2 Method of data collection

Employ data collection instrument, which is observation.

1.7.3 Method of Analysis

Quantitative data analysis technique and will make analysis using Simulation software called Ansys will be used.

1.7.4 Research Method

- A lot of paper and journal has been read up and part of it has been considered in this project. Meanwhile, the actual real brake disc dimension has been taken from ERC specification for analysis purpose. Later, the precise dimensions have been used to translate in 3D drawing by using SOLIDWORK.

- Next, the 3D model of disc brake rotor has been transferred to finite element software which is ANSYS software. Thermal analysis will be done on transient

responses. Assigning material properties, load and meshing of the model will be done in ANSYS software. Finally an expected result of thermal analysis will be obtained.

- Finally, the performance of the disc brake will be analyzed based on the Ansys analysis result of thermal nodal temperature and empirically calculated demand of μ at each selected stations. So that from the result, this research checks the performance of disc brake systems and from that the research decided how the speed of the train be for difficult conditions during emergency and/or service brake conditions to keep its performance improve further.

1.8 Organization of the Research

The research is organized in six parts. The first chapter is the introduction part which clearly states the background of the research, statement of problem, and objective, scope and limitation of the study, methodology, schedule, cost and benefits of the research.

Chapter two discusses Literature review of the disc brake system of rail vehicle. Definition, key points, models and analytical approaches will be discussed in this part of the study.

Chapter three shows the data collection and interpretation with data collection results.

The physical model of passenger train disc brake as well as the material and boundary conditions are discussed in detail under chapter four.

Chapter five deals about Ansys result and discussion of the result obtained after the simulation process using Ansys. This is done using numerical approach.

Conclusion and recommendation of the research on railroad vehicle brake system will be discussed and forwarded in chapter six. This will be the final chapter of the research.

CHAPTER-2

LITERATURE REVIEW

2.1 Introduction

Annually 1.2×10^6 persons are killed and about 50×10^6 is traumatized in traffic accidents in the world according to Worldwide Health Organization statistics. [14] On average a braking distance of 2–3m is too short to prevent 70% of traffic accidents. Forecasts show that the number of traffic accidents will increase by approximately 65% in the next 20 years if this problem is not solved. The general costs from traffic accidents and trauma are estimated to be USD 5.18×10^{11} per year worldwide [15].

Therefore, the production of safe vehicles is one of the major problems facing automobile and locomotive makers. The maintenance of effective braking is one of the determining factors in vehicle safety. Enhancement of the brake system's efficiency is a crucial scientific problem facing automotive manufacturers. The most important units in the vehicle's brake system are the wheel brakes because these components transform the control force from a brake gear into a brake torque which is applied to a wheel. Wheel brakes are one of the most complicated vehicle units with respect to a mathematical description of the processes that take place in them during braking [16].

The braking performance of the train is one of the most important factors that affected the traffic safety and the brake is the key components related to the running safety of the automotive. It was found that the 50% is caused by braking according to the analysis of the deviation and sideslip accident analysis. It is mostly due to the thermal fatigue failure cause brake failure caused by traffic accident. The numerical simulation of the temperature field is the main method of solving the brake thermal recession and thermal fatigue. The biggest difficulty is the analysis of the transient and alternating temperature field during the braking process and the establishment of a simplified model of the thermal - structural coupling [17].

Although many researchers have been done previously on brake systems, it was necessary that even further research to be done on this topic to improve the performance of braking systems of vehicles to handle the above listed scenarios.

The braking system is the most important system in trains. If the brakes fail, the result can be disastrous. Brakes are actually energy conversion devices, which convert the kinetic energy (momentum) of the vehicle into thermal energy (heat).When you step on the brakes, you command a stopping force ten times as powerful as the force that puts the car in motion. The braking system can exert high pressure on each of the brakes. In modern systems, the master cylinder is power-assisted by the engine. All newer cars have dual systems, with two wheels' brakes operated by each subsystem. That way, if one subsystem fails, the other can provide reasonably adequate braking power. Safety systems like this make modern brakes more complex, but also much safer than earlier braking systems [18].

In general, there are three main functions of a brake system, i.e., to control a vehicle's speed when driving downhill, to minimize a vehicle's speed when necessary and to keep a vehicle stationary when in parking [19].

Therefore, a brake is a device for slowing or stopping the motion of a train or vehicle, or alternatively a device to restrain it from starting to move again. Brakes of some description are fitted to most wheeled vehicles, including automobiles of all kinds, trucks, trains, motorcycles, and bicycles. Some airplanes are fitted with wheel brakes on the undercarriage. Some aircraft also feature air brakes designed to slow them down in flight. Baggage carts and shopping carts may have them for use on a moving ramp. For trains somehow it different from the normal brake system there may two or more brake system may involve and there exist disc and /or wheel-tread brake for each wheel. Friction brakes on cars store the heat in the rotating part (drum brake or disc brake) during the brake application and release it to the air gradually. This research will focus on train brake systems.

2.2 Types of train brake systems

2.2.1 Air brake

The air brake uses the compressed air as the source of brake force and controls the brake force by changing the pressure intensity of compressed air.

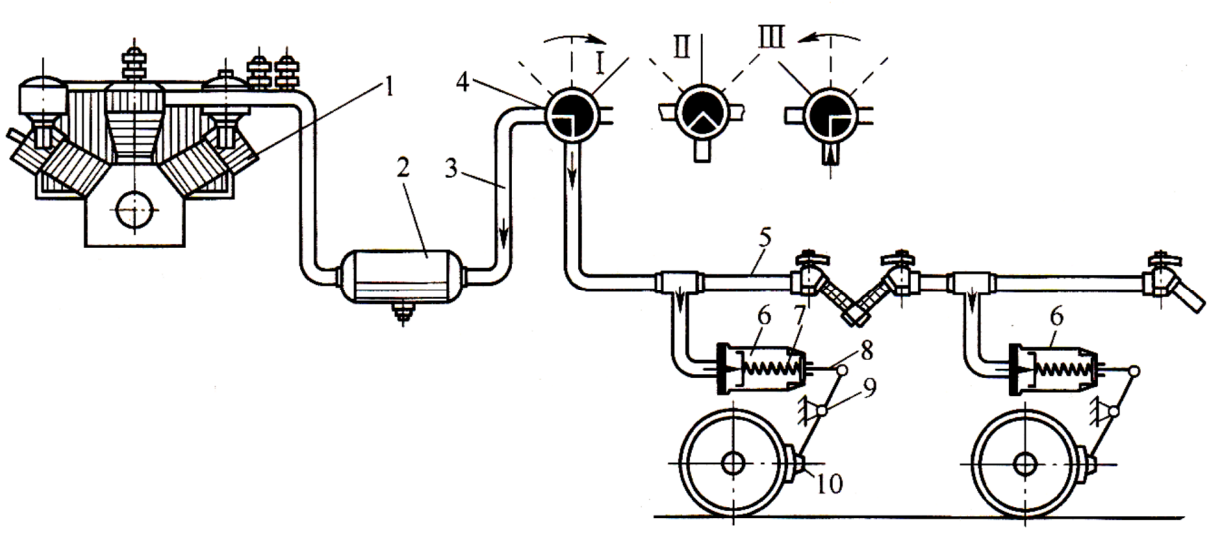


Figure 2.1 Train air brake systems

2.2.2 Electromagnetic Rail brake

The electromagnetic rail brake is based on developing electromagnetic attractive forces towards the rail which causes a normal application force acting on their contact surfaces that are in relative displacement. This leads to friction forces between the magnetic brakes and rails, opposing the vehicle's direction of motion, which generate braking forces.

1 – Coil; 2 – yoke; 3 – rail...

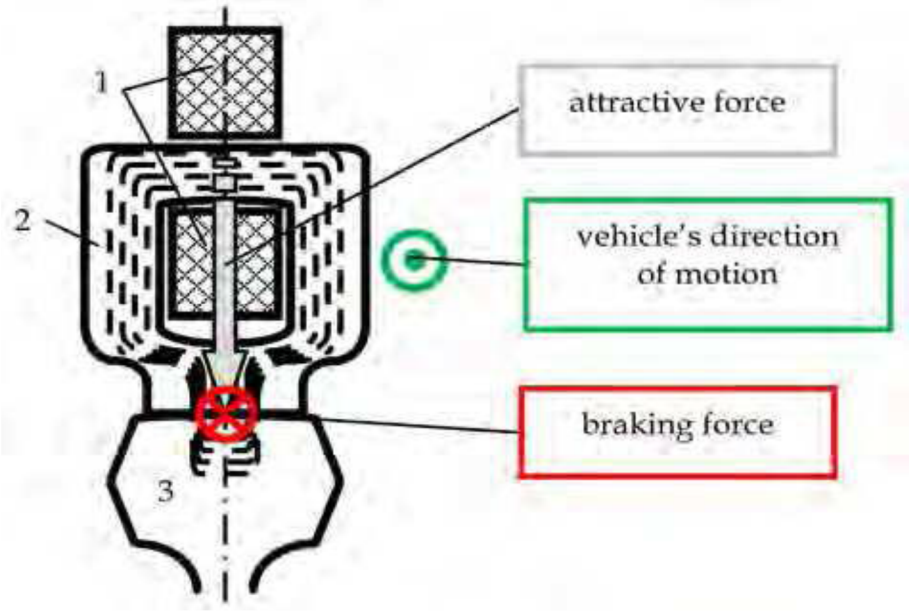


Figure 2.2 Train electromagnetic rail brake system

2.2.3 Rail (eddy current) braking systems

The rail brake on the bogie is connected to south poles forming magnetic fields between the poles. The magnetic fields generate eddy currents in the top surface of the rails, creating force acting in an opposite direction to the movement of the train, in other words, a braking force.

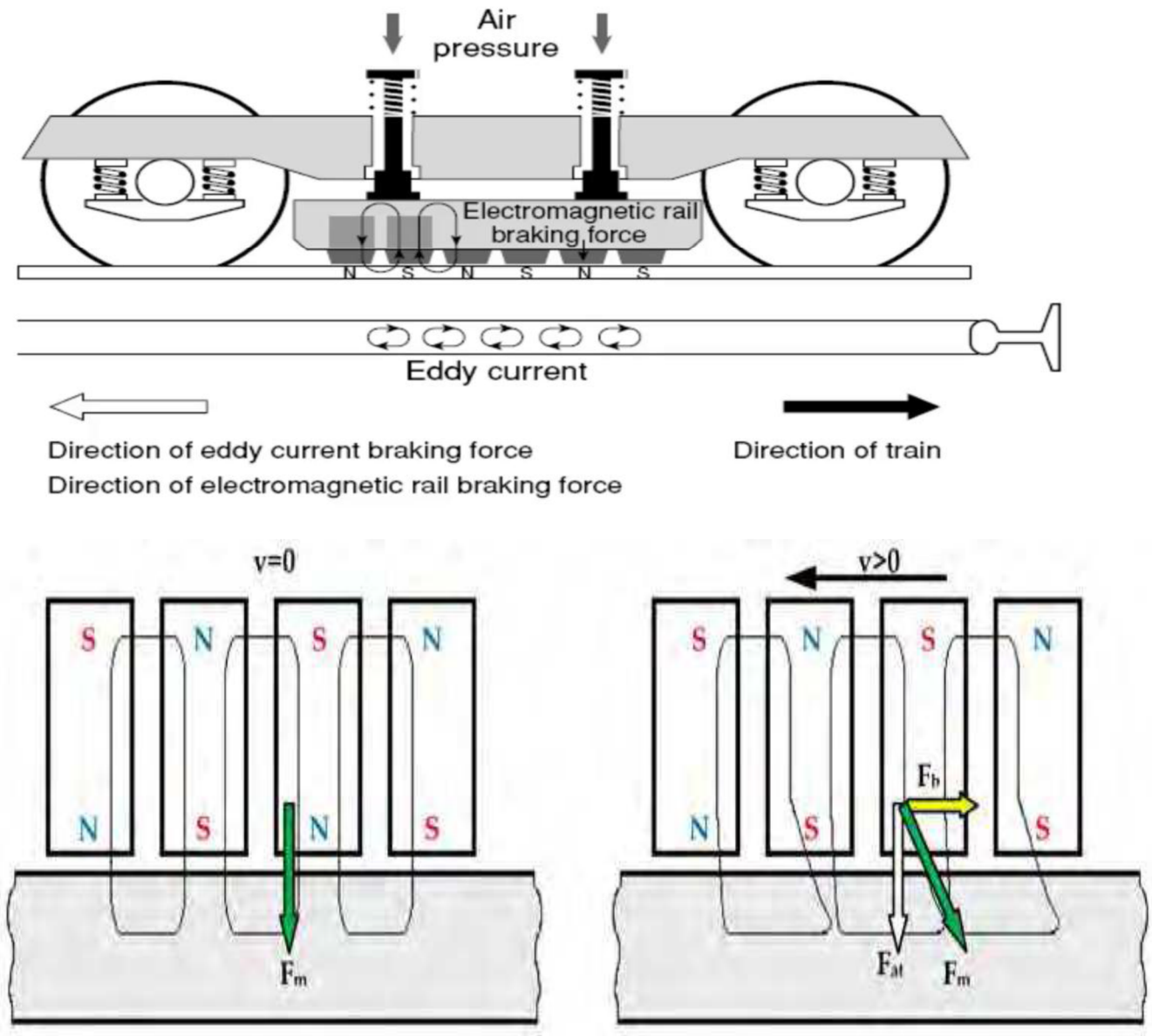


Figure 2.3 Train rail (eddy current) brake system

2.2.4 Rheostatic braking (dynamic braking)

A braking system used by locomotives and trains fitted with reuse but electric traction motors where the motors become generators and the current developed is fed to on-board

resistor grids. The energy is dissipated as heat as the grids cool. Some grids have to be force-ventilated to dispose of the heat quickly enough.

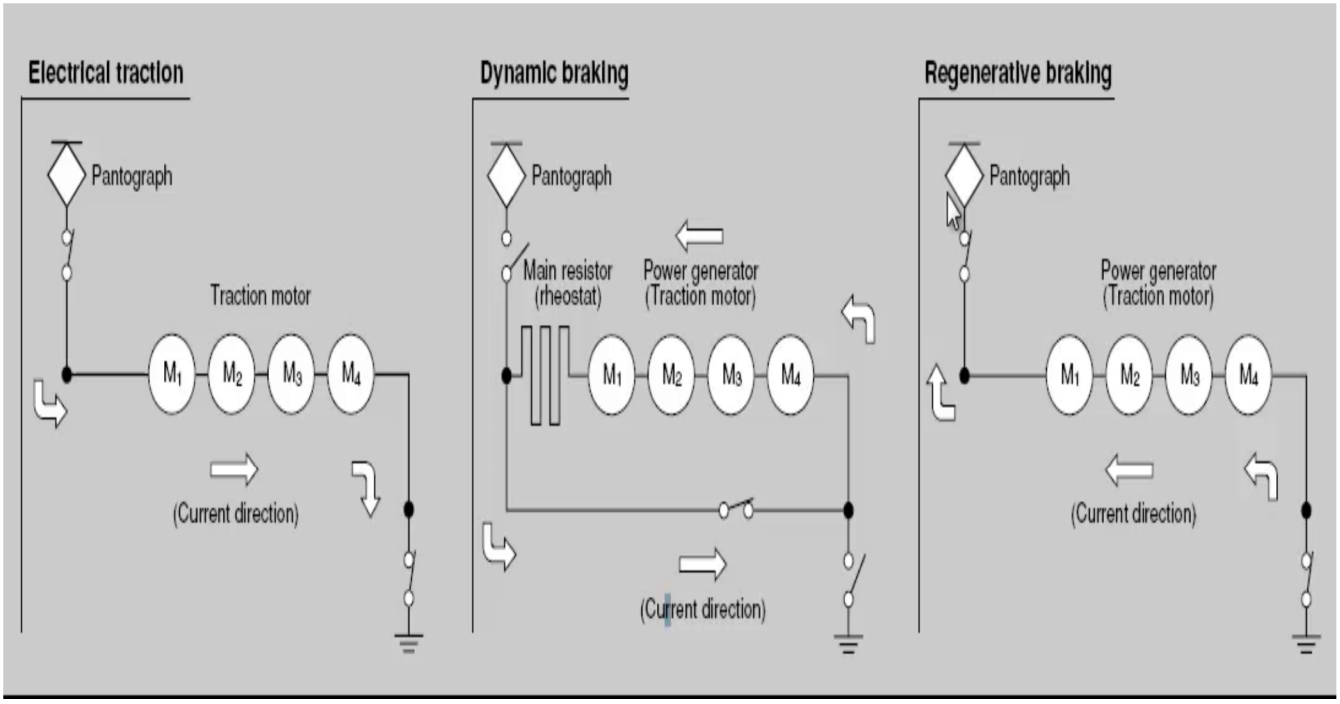


Figure 2.4 Train Rheostatic (Dynamic) brake system

2.2.5 Regenerative braking

In this braking system used by locomotives and trains fitted with electric traction motors where the motors become generators and the current developed is fed into the overhead line or third rail for use by other trains or for return to the supplier.

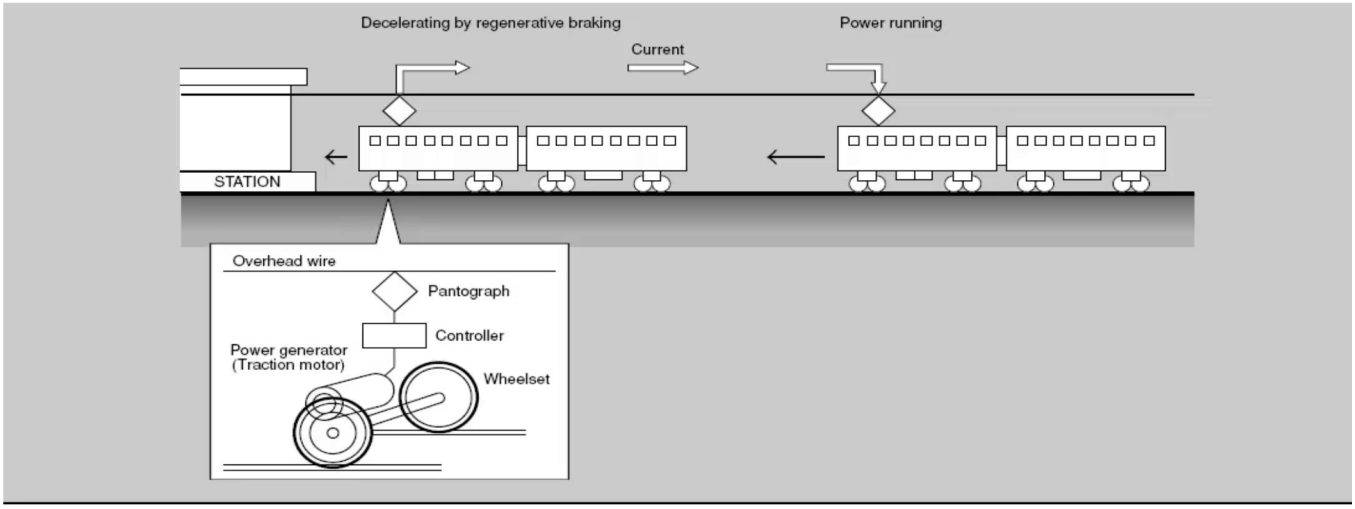


Figure 2.5 Train Regenerative brake system

2.3 Braking capacity

The main goal in designing and operating of to provide the necessary braking capacity appropriate to the type and specific running speed of the vehicles/trains according to the traffic safety.

Braking capacity is a significant feature of any railway vehicle and train which states the overall design and functional capability to stop from a maximum running speed according to the maximum braking forces developed during an emergency braking.

The braking capacity of a railway vehicle depends on numerous factors and some of the most important are: running speed, weight, type of brakes, constructive and functional characteristics of the brake rigging, braking characteristics, thermal phenomena, etc. In assessing the braking capacity of a train, additional parameters are involved: the train type, composition and length, the braking wave propagation characteristics, etc. Therefore, a direct and consistent assessment of braking capacity is difficult to achieve [20].

Most of trains are using the above listed braking systems but in our country the type of braking system that we use for the newly constructed Addis Ababa light rail transit trains is disc brake therefore in this research we will focus on this type of brake system.

2.4 Disc Brake

The disc brake is a wheel brake which slows rotation of the wheel by the friction caused by pushing brake pads against a brake disc with a set of calipers. The disc brake is usually made of grey cast iron, structural steel, but may in some cases be made of composition such as reinforced carbon-carbon or ceramic matrix composition. This is associated to the Wheel or the axle. To stop the wheel, frictional material in the form of brake pads, mounted on a device called a brake caliper, is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc. Friction causes the disc and attachment wheels pad to slowdown and Stop. Brakes convert motion to heat, and if the brakes get too hot, they become less effective and this phenomena is known as brake fade [21].

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, diluting friction difference. Unlike drum brakes, disc brakes have limited self-energizing action making it necessary to apply greater hydraulic pressure to find enough braking force. This is achieved by increasing the size of the caliper piston. The simple design helps easy maintenance and pad replacement [22].

2.4.1 Disc Brake Components

There are many different designs but all contain the following basic parts:

Caliper body

Foundation of any disc brake is the caliper body: the U-shaped casting that wraps around the rotor (like a hydraulic C-clamp) is usually made of Nodular cast iron but some are aluminum for lighter weight.

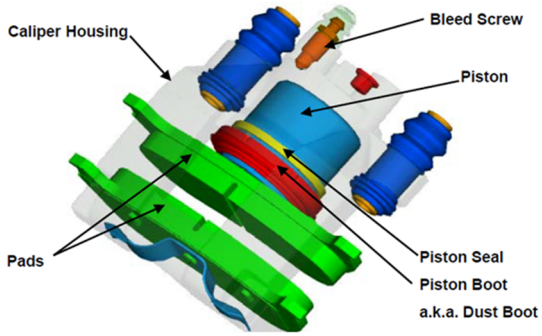


Figure 2.6 caliper body

Disc Rotor

There are various designs of disc rotors. Some are simply solid and ventilated cast level, but others are hollowed out with fins or vanes joining together the disc's two contact surfaces (usually included as part of a casting process).

Caliper

The brake caliper is the assembly which houses the brake pads and pistons. The pistons are usually made of aluminum or chrome plated steel. There are two types of calipers: floating or fixed. A fixed caliper does not move relative to the disc and is, thus, less

tolerant of disc imperfections. A floating caliper (also called a “sliding caliper”) moves with respect to the disc, along a line parallel to the axis of rotation of the disc.

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.

2.4.2 Materials for brake disc and brake pad

Rotor disc (Grey cast iron)

High thermal conductivity and specific heat capacity of grey cast iron are two major factors often considered to make cast iron disc brake rotors.

Brake pad

The material use for the brake lining should have the following characteristics

- It should have high coefficient of friction with minimum fading. In other words, 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 [23].

Force involved in brake torque variation

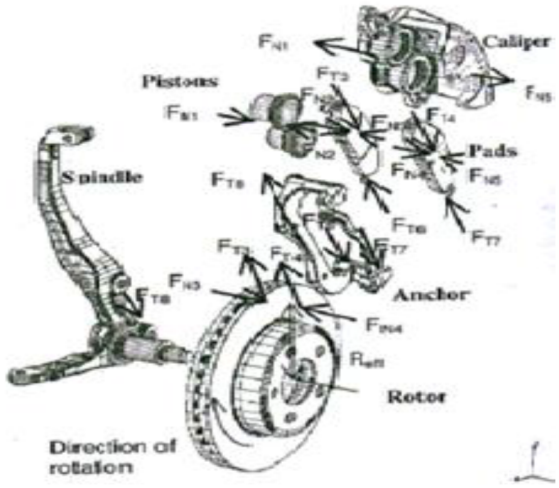


Figure 2.7 Free body diagrams showing static force in brake caliper during braking

The forces shown in Figure 2.7 were divided into two main categories: normal to the rotor faces and tangential to the rotor faces (e.g., F_{N3} and F_{T3} respectively). For example, the subscript 3 refers to the interface between the inner pad and the rotor. The forces of F_{N3} and F_{T3} were applied to the rotor by the pad and hence equal and opposite forces were applied to the pad by the rotor.

Tangential force

The forces acting on the inner and outer rotor faces due to contact with brake pads are shown in Figure2.

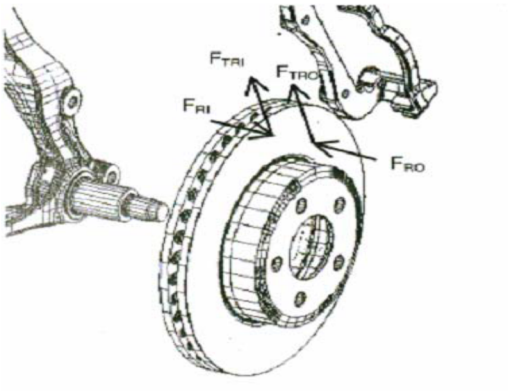


Figure 2.8 Forces acting on rotor due to contact with brake pads (variables renamed from Fig 2.7)

Tangential force between pad and rotor (inner face)

$$F_{TRI} = \mu_1 \times F_{RI} \quad \text{.....1}$$

Tangential force between pad and rotor (outer face)

$$F_{TRO} = \mu_o \times F_{RO} \quad \text{.....2}$$

Brake torque

The expression for brake torque was developed as a function of the two tangential forces and the effective radius of the pad/rotor interface:

$$T_B = (\mu_1 \times F_{RI} + \mu_0 \times F_{RO}) \times R_{eff} \quad \text{.....3}$$

Brake torque in case of pad and rotor same material

With the assumption of equal coefficients of friction and normal forces FR on the inner and outer faces [23]:

$$T_B = 2\mu \times F_R \times R_{eff} \quad \text{.....4}$$

2.5 Braking Distance Calculation Tool

The tool is used when required to calculate train braking distances. There are two versions of the tool; one for the Microsoft Windows®2 environment, the other for the IBM OS/2®3 environment. Both versions are written using the C++ language. The two versions have been designated as “ISAAC” and “NEWTON”. From the user’s perspective, each version of the tool operates in the same way: here in this research to calculate brake distance, brake deceleration and brake time there involves a computer program written using c# code, Newton second law [24][25].

$$F = m \times a \quad \text{.....5}$$

$$S = (v^2 - v_o^2) / (2 \times a) \quad \text{.....6}$$

$$t = (v - v_o) / a \quad \text{.....7}$$

Where: F is the resultant force

S is brake distance and

t is brake time

The braking forces are essentially influenced by the friction coefficients involved, their

dependence on different parameters having important role on braking characteristics of the vehicle. There are many factors determining the evolution of friction coefficients, among them the most important proved to be the running speed, the clamping forces, the surface contact pressure and temperature.

Sill the main problem is that friction interface between the brake pads and disc must not exceed temperatures of 800 ^{o}C, else severe and sudden wear of pads occurs. Practically, the thermal regime determines the necessity of several brake discs mounted on the wheel set, even four in the case of high speed vehicles [30].

Now the performance of disc brake system can be defined using the coefficient of friction here the braking distance is dependent on speed, coefficient of friction and the roadway grade by applying the following formula coefficient of friction can be determined:

$$\mu = \frac{V^2}{(2 \times g \times \text{lbrake} \times 3.6^2)} - \alpha \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots 8$$

l_{brake} = braking distance (m)

V = speed (km/h)

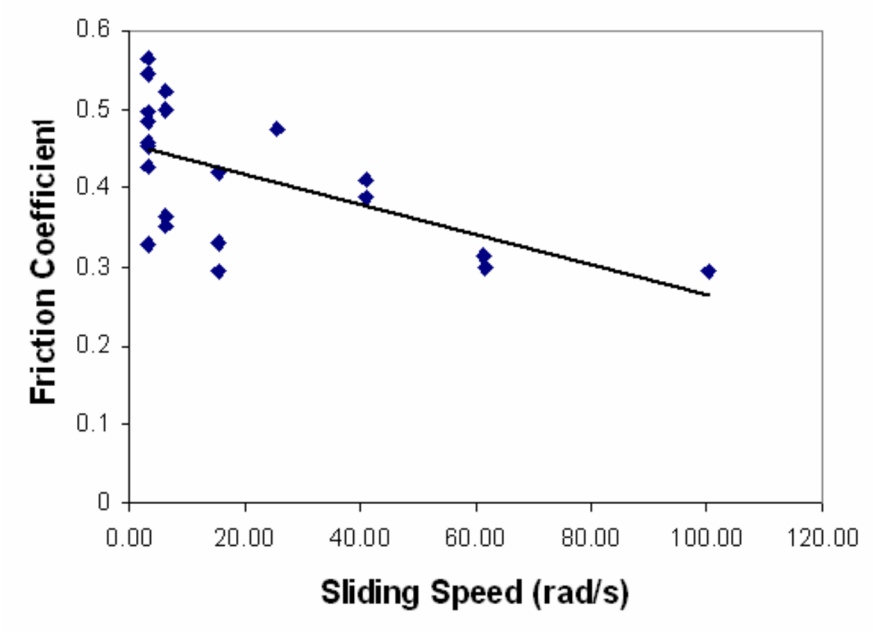
g = acceleration due to gravity (9.81 m/s^2)

μ = coefficient of friction

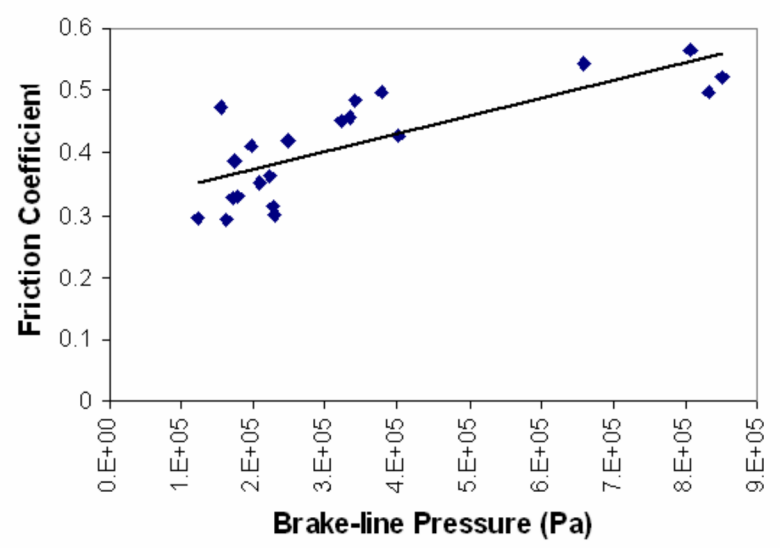
α = gradient

As a rule, the coefficient of friction is not constant, in that it increases during braking as speed diminishes. In determining the coefficient of friction, a mean value is therefore applied for the given speed [26].

It is also worthwhile to see how friction coefficients vary with sliding speeds and Brake - line pressures. Figure 6.1 shows that friction coefficient decreases as sliding speed increases. This may lead to a term called negative μ −n slope and consequently will produce negative damping.



(a) Friction coefficient Vs. sliding speed



(b) Friction coefficient Vs. brake line pressure

Graph 1. Relationship between friction coefficient and sliding speed (a) and Brake-line pressure (b)

[27].

2.6 Heat Flux entering the disc

The heat flux evacuated of this surface is equal to the friction power. The initial heat flux q_0 entering the disc is calculated by the following formula:

Step 1: Kinetic Energy (K.E) = ½× m×υ^2……..9

The above said is the Total Kinetic Energy induced while the vehicle is under motion.

Step 2: The total kinetic energy = the heat generated

Step 3: The area of the rubbing faces, $A = 2\Pi \times (r_o^2 - r_i^2)$……10

Step 4: Heat Flux = Heat Generated / Time / rubbing area

The analysis is done by taking the distribution of braking torque between the front and rear axle [28].

$$Q_t = \frac{\frac{1}{2}\times\beta\times M\times(V^2-V0^2)}{\Delta t} \pm \frac{M\times g\times \sin\alpha\times s}{\Delta t} \quad \text{……11}$$

Once the model of disc brake prepared using solid works different loads are applied using the above listed formulas and then finite element will be developed for the rotor.

2.7 Finite Element Formulation /Thermal Transient Analysis/ of the Disc Brake

Transient heat conduction in three dimensional heat transfer problem is governed by the following differential equation.

$$- \left(\frac{\partial \dot{q}x}{\partial x} + \frac{\partial \dot{q}y}{\partial y} + \frac{\partial \dot{q}z}{\partial z}\right) + \dot{Q} = \rho C_p \frac{\partial T}{\partial t} \quad \text{……12}$$

Where q_x, q_y and q_z are conduction heat fluxes in x, y and z-directions, respectively, Cp is the specific heat, ρ is the specific mass, Q is internal heat generation rate per unit volume and T is the temperature that varies with the coordinates as well as the time t. The conduction heat fluxes can be written in the form of temperature using Fourier's law. Assuming constant and uniform thermal properties, the relations are:

$$\dot{q}_x = -k_x \frac{\partial T}{\partial x},\ \dot{q}_y = -k_y \frac{\partial T}{\partial y},\ \dot{q}_z = -k_z \frac{\partial T}{\partial z} \quad \text{……...13}$$

Where k_x, k_y and k_z are thermal conductivity in x, y and z-directions, respectively. Heat transfer boundary conditions consist of several heat transfer modes that can be written in different forms. The boundary conditions frequently encountered are as follows.

$T_s = T_1$ (x, y, z, t)

$$-\dot{q}_s = h\,(T_s - T_{amb}) \quad \text{……14}$$

Where T_1 is the specified surface temperature; qs the specified surface heat flux (positive into a surface); h the convective heat transfer coefficient; Ts the unknown surface temperature, and T_∞ the convective exchange temperature [29].

So now using the formulation of finite element for the disc, Ansys thermal transient simulation will follows using the following equation.

$$[C]\{\frac{\partial T}{\partial x}\} + [K]\{T\} = \{\dot{Q}\} \quad \text{.....15}$$

2.8 Related Research

Ali Belhocine and Mostefa Bouchetara, studies temperature distribution in the disc brake used in heavy load vehicles and the results obtained are compared with those of specialized literature. It gives a brief account of ANSYS CFX-CFD to find the convection heat transfer coefficient. The modeling of the temperature distribution in the disc brake is used to identify all the factors and entering parameters concerned at the time of braking operation such as type of braking ,the geometric ,design of the disc and the used material. Limitation to this paper is for accurate determination of the brake time brake deceleration and braking distances computer programs is not used. The results obtained by the simulation are satisfactory compared with those of the specialized literature [29].

A. LEE, AND J. R. BARBER, discussed about the various modes of heat transfer in disc brakes .The braking performance of a vehicle can be significantly affected by the temperature. The limitation to this research paper is it only limited to show its brake performance with respect to temperature, it does not show with respect to coefficient of friction vs. sliding speed. Poor heat dissipation may cause brake fade, premature wear, brake fluid vaporization, bearing failure, thermal cracks and thermally excited vibration. The disk brake and the pads have to own a high thermal conductivity and a high surface to volume ratio and high mechanical resistance to wear. Frictional heat generation in the sliding contact of two bodies influences friction and wear characteristics of brake systems. The use of an internally ventilated rotor will increase both surface area and the heat transfer coefficient, due to forced convection created by the internal airflow, with negligible influence on unsprung mass. A common problem that may happen to disk

brake system is that after several braking when the vehicle is stopped. There is no air flow in the wheel assembly then conduction is an effective mode of heat transfer but it can have adverse effect on nearby components. This causes problems like brake fluid vaporization that happened because of heat concentration phenomena and is intensified when the vehicle is parked [30].

Ameer Fareed Basha et al, studied about the model of a disc brake used in vehicles. Coupled field analysis (Structural+Thermal) is done on the disc brake. The material used was Cast Iron. Analysis is also done by changing the design of disc brake. Actual disc brake has no holes; design is changed by giving holes in the disc brake for more heat dissipation. This research tries to show the coupled field analysis of disc braking system but it doesn't show the performance analysis of this brake system at all with no parameters such as temperature coefficient of friction between the disc pad and the disc rotor /disc/ to have such advantage which are listed throughout this research paper. In general the results obtained by the simulation are good compared with those of the specialized literature [31].

KENG LENG KHONG (2009), investigated thermal stress analysis on heavy truck brake disc rotor for steady state and transient condition. The heat dissipation along the brake disc surface during the periodic braking via conduction, convection and radiation is studied. In order to get the stable and accurate result of element size. Time step selection is very important and all of these aspects are discussed in this paper .This research provide a useful design tool to improve the brake performance of disk brake system [32].

Šamec, B.; Oder, G; Lerher, T. & Potrč, I., studied the following A certain number of railway brake discs, made of gray cast iron, shown the presence of small cracks only after a few thousand kilometers. Investigate main causes of a brake disc failure; numerical analysis was done, using ABAQUS software. Here again coupled analysis of disc brake is done with Numerical analysis resulted from a physical model of heat flux in dependence of braking time. Physical model was applied considering all demands and presumptions given by industry representatives [33].

Akara Sawardsuk, Saiprasit Koetniyom the sirindhorn, Railway is a mass transportation which can cause enormous damage to life and property. Several causes of accident such as dilapidated rail, the railway drive high speed, and the wheel tire takes off from the wheel set under the brake application etc. One cause of such a problem might come from using shoe brake, which pushes against the wheel tire. This situation creates high temperature and wearing on the wheel tire resulting in splitting off the wheel set. For these reasons, the aim of this paper is to develop a railway brake system with considerable retrofit the original brake system with the disc brake system wherewith design of brake disc dimensions are based on the torque requirements from Thai railway locomotive at the initial speed before brake application. The paper Uses mathematical method for the brake disc temperature is investigated based on the conditions in which the brake application and cooling period are considered between the stations. The mathematical method based on the brake application and cooling period is used to investigate the maximum and gradient temperature responses between the stations. Finally in this paper the analysis led to the following point's Maximum speed of locomotive affects high temperature response of brake disc even though high speed causes high cooling performance of brake disc, in brake disc, heat conduction and convection occur at the same time. But, program can predict the temperature response only in one dimension. To increase more accuracy in temperature analysis, two or three dimension temperature analysis is suggested to predict more accurately, the effects of station distance and acceleration conditions dominate the traveling time and cooling period and the braking period and One dimension heat transfer can be used to design brake disc geometry and determine temperature responses of brake disc under various load conditions and requirement conditions from State Railway of Thailand [34].

A. Floquet et al, determined of temperature distribution and comparison of simulation results and experimental results in the disc by 2D thermal analysis using axis-symmetric model. The disc brake used in the automobile is divided into two parts; a rotating axis-symmetrical disc, and the stationary pads. The friction heat, which is generated on the interface of the disc and pads, can cause high temperature during the braking process. The influence of initial velocity and deceleration on cooling of the brake disc was also investigated. The thermal simulation is used to characterize the temperature field of the disc with appropriate boundary conditions. A Finite-element method was developed for determining the critical sliding speed for thermo elastic instability of an axis-symmetric brake [35].

In this work study about a performance analysis using Ansys transient analysis of the thermo elastic problem for disk brakes with frictional heat generation, was done using the finite element analysis (FEA) method using ANSYS software. The computational results are presented for the dispersion of the temperature on the friction surface between the contacting bodies (the disk and the pad).

CHAPTER-3

DATA COLLECTION AND ANALYSIS

3.1 Calculating the slopes and/or gradient for selected Addis Ababa E-W stations

The following shows calculation on the gradient of the rail using profile of the Addis Ababa e-w route line obtained from ERC, the other station rail gradient will be calculated in the same procedure listed below.

1.) Slope = $\frac{\text{ELEVATION DIFFERENCE}}{\text{HORIZONTAL DISTANCE}}$, at station EW4

Elevation difference = |2382.1 – 2379.2| = 2.9 m

Horizontal difference = [1.2×10^{-2} m×5000 m] /1 m = 60 m

Slope = $\frac{2.9m}{60m}$ = 0.0483

$\tan^{-1} 0.0483$ = 2.767°

Slope = 0.0483 = 2.767°

Using similar procedure to the above equation the slope for station EW 1, EW 2 and EW 3 are 0.01580 = 0.91°, 0.0143 = 0.8° and 0.016 = 0.92° respectively.

3.2 Addis Ababa lrt E-W rout gradient

Table 1. Addis Ababa light rail transit east- west rail profile gradient.

No	Station name	Gradient/slope	Station types
1	Ayat 2	0.01580 [0.905°]	Ground station[- down]
2	Ayat 1	0.0143[0.818°]	Ground station[level]
3	Meri/CMC 2	0.0160[0.917°]	Ground station[level]
4	CMC 1	0.0483[2.767°]	Ground station[+ up]
5	St.Michael Church	0.0185[1.058°]	Ground station

6	Civil Service College	0.0178[1.019°]	Ground station
7	Sahlite Mhiret Church	0.0138[0.79°]	Ground station
8	Gurd Sholla	0.0022[0.127°]	Ground station
9	Megenagna/Adwa square	0.018[1.031°]	Underground station
10	Lem Hotel	0.0063[0.358°]	Ground Station
11	Mazoria/Traffic Police HQ.	0.0086[0.491°]	Ground Station
12	Chemical Corporation	0.0253[1.247°]	Elevated station
13	Urael Church	0.0243[1.394°]	Ground Station
14	Yordanose Hotel	0.0148[0.849°]	Ground Station
15	Estifanose/Meskel Square2	0.0027[0.155°]	Ground Station
16	Stadium/Meskel Square 1	0.0016[0.093°]	Elevated station
17	La Gare	0.008[0.458°]	Elevated station
18	Road Authority	0.0018[0.102°]	Ground station
19	Mexico Square	0.0278[1.591°]	Elevated station
20	Lideta	0.0392[1.247°]	Elevated station
21	Coca Cola	0.0227[1.302°]	Ground station
22	Tor Hailoch	0.0088[0.504°]	Ground station

[Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

3.3 Selected stations

These stations are selected based on the geographical locations. Station Ayat 2 located on the down slope but EW 2 and EW 3 have very small angle of elevation and depression respectively. Therefore considered level stations and the last station at CMC 1 is ground station with positive slope. Therefore the analysis procedure for the rest 18 stations will follow the same as it is done for these selected stations.

Table 2. Selected stations

No	Station name	Station types	Gradient/slope
1	Ayat 2	Ground station[- down]	0.0158[0.905°]
2	Ayat 1	Ground station[level]	0.0143[0.818°]
3	Meri/CMC 2	Ground station[level]	0.016[0.917°]

4	CMC 1	Ground station[+ up]	0.0483[2.767°]

[Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

3.4 Addis Ababa lrt E-W Distance between consecutive stations

Table 3. Addis Ababa LRT E-W distance between consecutive stations.

No	Addis Ababa LRT E-W rout Station name	Addis Ababa LRT E-W rout stations symbol	Station distance
1	Ayat 2---------- Ayat 1	EW1----------EW2	1.30km
2	Ayat 1----------Meri/CMC 2	EW2----------EW3	1.034km
3	Meri/CMC 2-------CMC 1	EW3----------EW4	0.896km
4	CMC 1---------- St.Michael Church	EW4---------- EW5	0.865km
5	St.Michael Church---- Civil Service College	EW5---------- EW6	0.785km
6	Civil Service College----- Sahlite Mhiret Church	EW6----------EW7	0.906km
7	Sahlite Mhiret Church-----Gurd Sholla	EW7----------EW8	0.922km
8	Gurd Sholla---------Megenagna/Adwa square	EW8----------EW9	1.004km
9	Megenagna/Adwa square--Lem Hotel	EW9----------EW10	1.035km
10	Lem Hotel--------Mazoria/Traffic Police HQ.	EW10----------EW11	0.838km
11	Mazoria/Traffic Police HQ.--Chemical Corporation	EW11---------- EW12	0.694km
12	Chemical Corporation---Urael church	EW12----------EW13	0.875km
13	Urael Church------ Yordanose Hotel	EW13---------- EW14	0.570km
14	Yordanose Hotel---Estifanose/Meskel Square2	EW14---------- EW15	0.569km
15	Estifanose/Meskel Square2----Stadium/Meskel Square 1	EW15---------- EW16	0.700km

16	Stadium/Meskel Square 1----La Gare	EW16---------- EW17	0.865km
17	La Gare----------Road Authority	EW17----------EW18	0.555km
18	Road Authority--------Mexico Square	EW18----------EW19	0.554km
19	Mexico Square----------Lideta	EW19---------- EW20	0.936km
20	Lideta----------Coca Cola	EW20---------- EW21	0.848km
21	Coca Cola----------Tor Hailoch	EW21---------- EW22	1.149km

[Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

3.5 Loading Capacity/Carrying Capacity of train

Table 4. Loading capacity/carrying capacity of train

Status	Seat	Standing Capacity	Total
Rated capacity (AW3)(Standing capacity:8persons/m^2)	64	190	254

[Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

3.5.1 Train Weight

Table 5. Train Weight

Status	Appr. Weight of train body (t)	Passenger Weight (t)	Total (t)
Rated capacity (AW3)(Standing capacity:8persons/m^2)	48.76	15.24	64

Note: passenger weight 60kg/person

[Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

3.6 Equation of motions at CMC1

- **Accelerated motion**

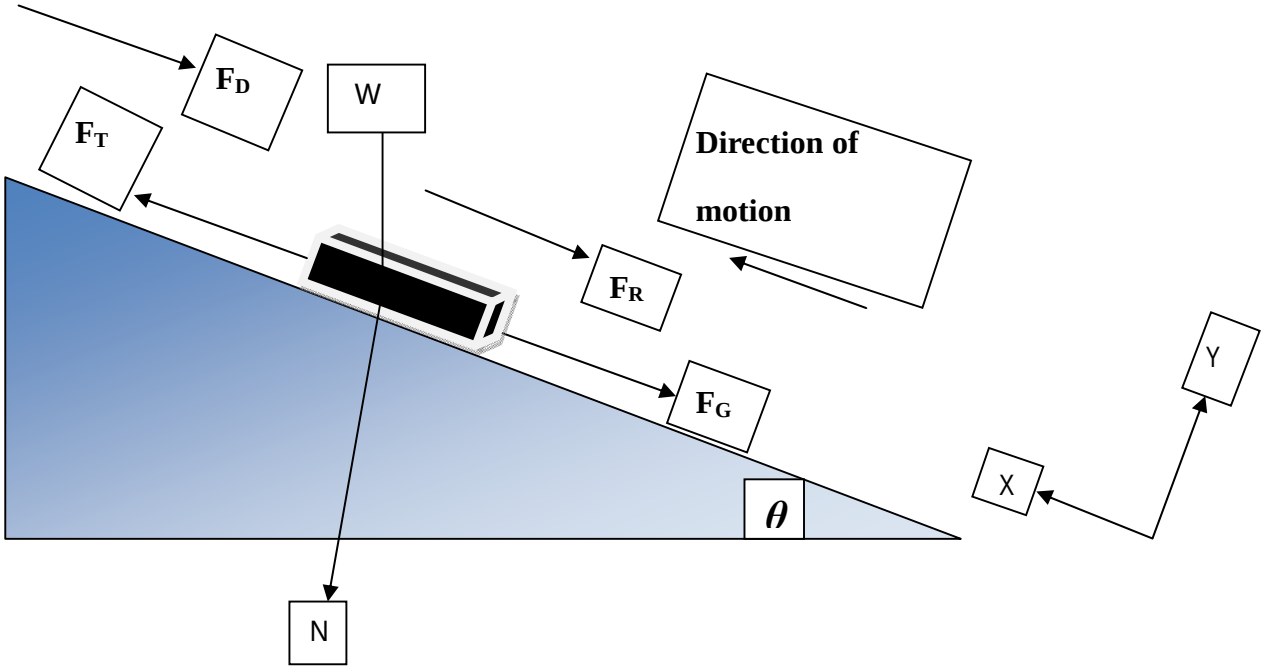


Figure 2.9 equation of motion at cmc1 during accelerated motion

Where:

W=total weight of the rail road vehicle including passenger weight

F_T= Tractive force

F_R=Rolling resistance

F_D=Drag force

F_G=Normal force component in the direction of the inclined plane.

N=Normal force

Appling Newton's second law of motion results

$ma = F_T - F_D - F_R - F_G$……16

3.6.1 Calculation of tractive force [F_T]

In railway engineering, the term tractive effort is often used synonymously with tractive force to describe the pulling or pushing capability of a locomotive. In automotive engineering, the terms are distinctive: tractive effort is generally higher than tractive force by the amount of rolling resistance present, and both terms are higher than the amount of drawbar pull by the total resistance present (including air resistance and grade).

$$\text{Motor torque} = \frac{power\ of\ the\ motor}{angular\ velocity\ of\ the\ motor}$$

$$T_m = \frac{P}{w} \quad \ldots\ldots 17$$

$$= \frac{117\times10^3 w}{1200 rpm} = \frac{117\times10^3 w}{40\pi\ rad/sec} = 932.08\text{Nm}$$

$$T_{wheel} = T_m \times \tau \quad \ldots\ldots 18$$

$$= 932.08\text{Nm} \times 7.43$$

$$= 6925.33\text{Nm}$$

$$F_T = \frac{\text{T wheel}}{R\ wheel} \quad \ldots\ldots 19$$

Diameter of wheel motor = 660mm

$$= \frac{6925.33\text{Nm}}{330\text{mm}}$$

$$= 20985.85\text{N}$$

Since there are 8 traction motors, therefore the total traction motor will be become

$$F_T = 20985.85\text{N} * 8$$

$$F_T = 167886.82\text{N}$$

3.6.2 Calculation of drag force [F_D]

In aerodynamics, aerodynamic drag is the fluid drag force that acts on any moving solid body in the direction of the fluid free stream flow. From the body's perspective (near field

approach), the drag comes from forces due to pressure distributions over the body surface.

$$F_D = ½ \times C_D \times \rho \times A \times V^2 \quad \text{.....20}$$

Where:-

C_D = drag coefficient

ρ = density of fluid ($1.2kg/m^3$ for air at NTP)

V = velocity of the train = 80km/hr

A = Characteristics frontal area of the body

Frontal Area = vehicle width × vehicle height

= 2650mm × 3700mm = $9.805m^2$.

[Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

Table 6. Drag coefficient for passenger train.

Type of object	**Drag coefficient-C_D-**	**Frontal-Area[m^2]**
Passenger Train	1.8	9.805

[Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

$F_D = 0.5 \times 1.8 \times 1.2kg/m^3 \times 9.805m^2 \times (22.22)^2$ m/s

= 5228.29 N

3.6.3 Calculation of rolling resistance force [F_R]

Rolling resistance, sometimes called rolling friction or rolling drag, is the force resisting the motion when a body (such as a ball, tire, or wheel) rolls on a surface. It is mainly caused by non-elastic effects; that is, not all the energy needed for deformation (or movement) of the wheel, roadbed, etc. is recovered when the pressure is removed.

$$F_R = f \times m \times g \times \cos\alpha \quad \text{.....21}$$

Where:-

f= 0.16, rolling coefficient of friction [Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009]

m= mass of the train including passenger live load

g=gravitational force of attraction

α [x, y] = gradient of the rail

$F_R = 0.16 \times 64 \times 10^3 \times 9.81 \times \cos\alpha$

$= 100454.4 \times \cos\alpha$

3.6.4 Calculation of normal component of the force in the direction of the inclined plane [F_G]

$F_G = m \times g \times \sin\alpha$……..22

$= 64 \times 10^3 \times 9.81 \times \sin\alpha$

$= 627840 \times \sin\alpha$

Therefore,

$$a = \frac{[F_T - F_D - F_R - F_G]}{m}$$

$$a = \frac{[167886.82 - 5228.29 - 100454.4 \times \cos\alpha - 627840 \times \sin\alpha]}{66000}$$

$$\mathbf{a = \frac{[162658.53 - 100454.4 \times \cos\alpha - 627840 \times \sin\alpha]}{64000}}$$……23

❖ **Decelerated motion**

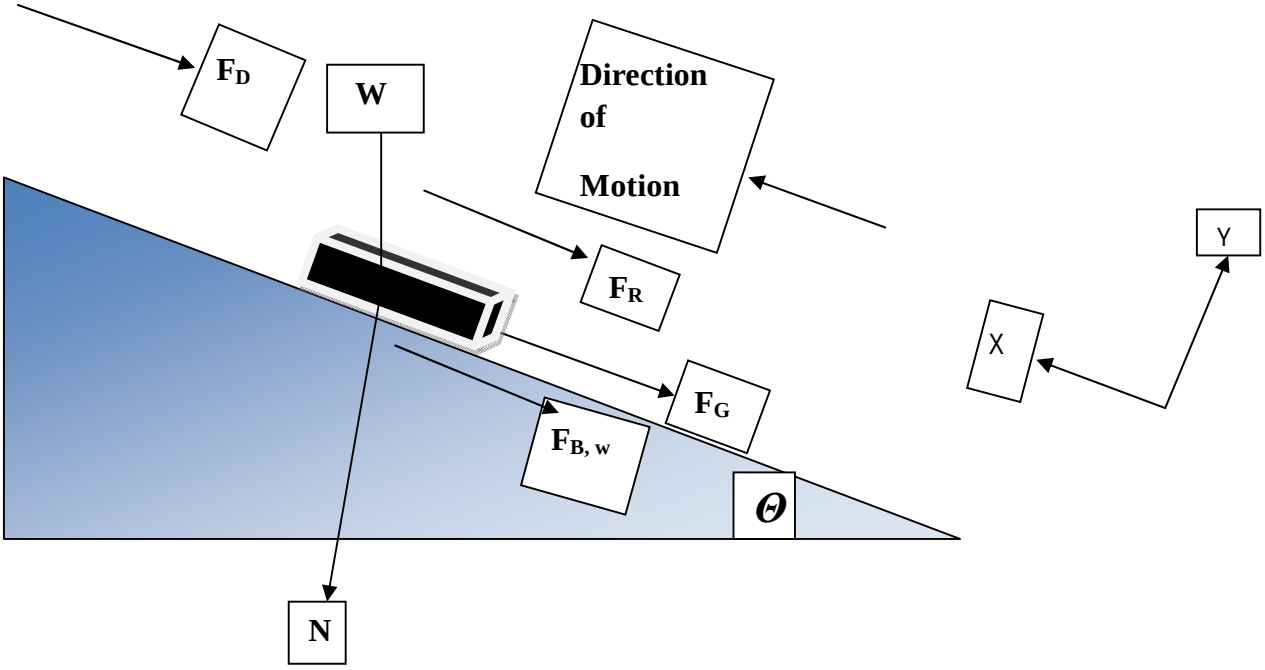


Figure 3.0 equation of motion at cmc1 during decelerated motion

Where:

W=total weight of the rail road vehicle including passenger weight

$F_{B,W}$= Brake force of the wheel

F_R=Rolling resistance

F_D=Drag force

F_G=Normal force component in the direction of the inclined plane.

N=Normal force

Appling Newton's second law of motion results

$ma = F_{B,w} + F_D + F_R + F_G$……24

3.6.5 Calculation of braking force of the wheel [$F_{B,w}$]

$ma = [F_{B,w} + F_D + F_R + F_G]$

$F_{B,w} = ma - F_D - F_R - F_G$

Using standard data from Addis Ababa lrt, service brake deceleration value of 1m/s² and 1.5 m/s² for emergency brake respectively the braking force on the wheel will be as follows.

$F_{B,w}$ for service brake at station CMC1 becomes

$F_{B,w}$ = [(64000×1)-(5228.29N) + (117543.42)-(100454.4×cosα) - (627840×sinα)]

For cos (2.767°) = 0.9988 and sin (2.767°) = 0.0483

$F_{B,w}$ = 189430.24N

$F_{B,w}$ for emergency brake at station CMC1 becomes

$F_{B,w}$ = [(64000×1.5)-(5228.29N)-(100454.4×cosα) - (627840×sinα)]

For cos (2.767°) = 0.9988 and sin (2.767°) = 0.0483

$F_{B,w}$ = -39886.82N

3.6.6 Calculation of drag force [F_D]

$F_D = ½ \times C_D \times \rho \times A \times V^2$

Where:-

C_D = drag coefficient

ρ = density of fluid (1.2kg/m^3 for air at NTP)

V = velocity of the train = 80km/hr

A = Characteristics frontal area of the body

Frontal Area = vehicle width *vehicle height

= 2650mm * 3700mm = 9.805m^2.

F_D = 0.5 x 1.8 x 1.2kg/m^3 x 9.805m^2 x $(22.22)^2$ m/s

= 5228.29N

3.6.7 Calculation of rolling resistance force [F_R]

F_R = f x m x g x cosα

Where:-

f= rolling coefficient of friction

m=mass of the train including passenger live load

g=gravitational force of attraction

α [x, y] = gradient of the rail

F_R = 0.16 x 64 x 10^3 x 9.81 x cosα

= 100454.4 x cosα

3.6.8 Calculation of normal component of the force in the direction of the inclined plane force [F_G]

F_G = m x g x sinα

= 64 x 10^3 x 9.81 x sinα

=627840 x sinα

Decelerated equation of motion for service and emergency brake respectively will become

$$a = \frac{[F_{B,W} + F_D + F_R + F_G]}{m}$$

$$\mathbf{a = -\frac{[194658.53 + 100454.4\ x\ cos\alpha + 627840\ x\ sin\alpha]}{64000}} \quad \text{.....25}$$

$$\mathbf{a = -\frac{[-34658.53 + 100454.4\ x\ cos\alpha + 627840\ x\ sin\alpha]}{64000}} \quad \text{.....26}$$

Brake distance, s for both braking conditions

$$S = \frac{[(V_f{}^2) - (V_i{}^2)]}{2a} \quad \text{.....27}$$

Braking time, t

$$t = \frac{(Vf - Vi)}{a} \quad \text{.....28}$$

Figure 3.1 is essential to calculate accurately the braking distance and braking time for the newly constructed Addis Ababa light rail transit. Upon filling the required data on the space provided such as the initial and final travelling speed of the train, the gradient using cosine and sine values and the status weather the train is traveling uphill or downhill will calculate accurately the required braking distance and braking time of the passenger train.

As we know that software's like Ansys works based on finite element analysis, Mat lab works based on matrix. But this Microsoft visual studio program called c# works based on simple mathematical equations.

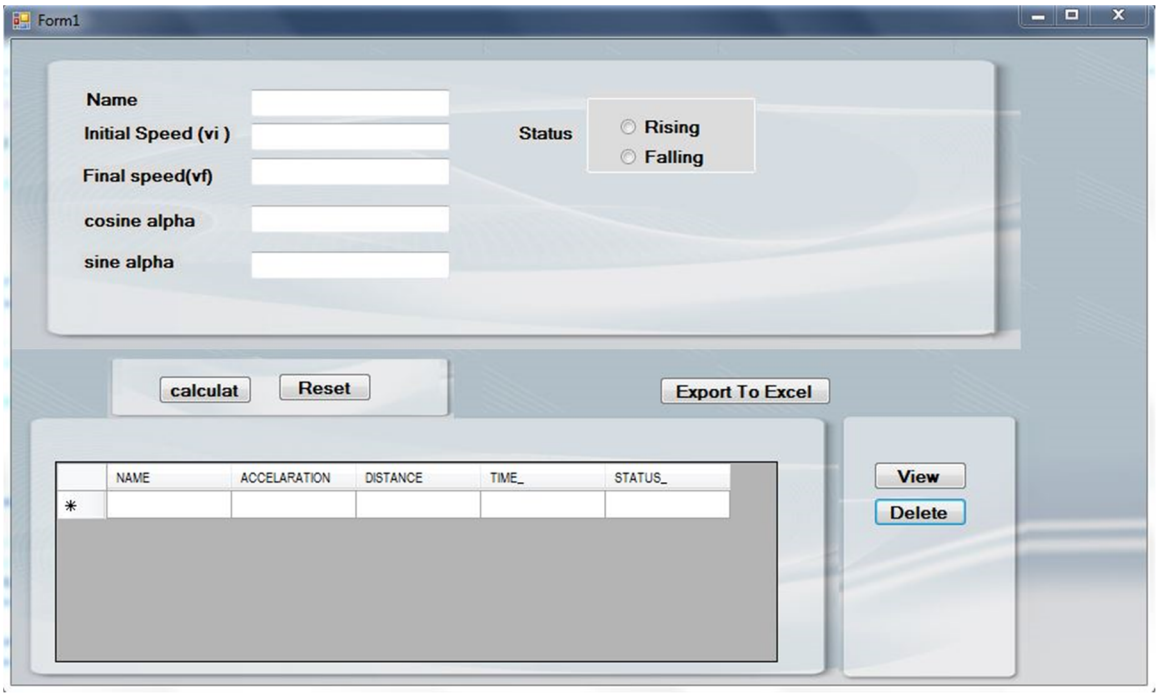


Figure 3.1 result page for the brake distance, deceleration and brake time

3.6.9 Service and emergency brake for Addis Ababa lrt at station: -

Table 7. Accelerated, normal service brake, normal emergency brake and constant motion between station CMC 2 and CMC 1.

Name	Initial speed	Final speed	Time	cose_alpha	sine_alpha	Acceleration	Distance
acceleration 1	0	3.33	3.33	0.9999	0.016	0.5	11.108889
acceleration 2	3.33	6.67	0.001	0.9999	0.016	0.5	33.34
acceleration 3	6.67	10	3.334	0.9999	0.016	0.5	55.551111
acceleration 4	10	13.33	6.667	0.9999	0.016	0.5	77.768889

acceleration 5	13.33	16.67	9.999	0.9999	0.016	0.5	100.02
acceleration 6	16.67	20	13.334	0.9999	0.016	0.5	122.21111
acceleration 7	20	22.22	17.778	0.9999	0.016	0.5	93.817284
accelerated motion	**0**	**22.22**	**22.22**	**0.9999**	**0.016**	**0.5**	**493.7284**
Normal service brake							
service brake 1	22.22	20	2.222	0.9999	0.0483	-1	46.908642
service brake 2	20	16.67	3.333	0.9999	0.0483	-1	61.105556
service brake 3	16.67	13.33	3.334	0.9999	0.0483	-1	50.01
service brake 4	13.33	10	3.333	0.9999	0.0483	-1	38.884445
service brake 5	10	6.67	3.333	0.9999	0.0483	-1	27.775556
service brake 6	6.67	3.33	3.334	0.9999	0.0483	-1	16.67
service brake 7	3.33	0	3.333	0.9999	0.0483	-1	5.5544445
decelerated motion	**22.22**	**0**	**22.222**	**0.9999**	0.0483	**-1**	**246.90864**
Normal emergency brake							
emergency brake 1	22.22	20	1.48133	0.9999	0.0483	-1.5	31.272428
emergency brake 2	20	16.67	2.222	0.9999	0.0483	-1.5	40.737037
emergency brake 3	16.67	13.33	2.22267	0.9999	0.0483	-1.5	33.34
emergency brake 4	13.33	10	2.222	0.9999	0.0483	-1.5	25.922963
emergency brake 5	10	6.67	2.222	0.9999	0.0483	-1.5	18.517037
emergency brake 6	6.67	3.33	2.22267	0.9999	0.0483	-1.5	11.113333
emergency brake 7	3.33	0	2.222	0.9999	0.0483	-1.5	3.702963
decelerated motion	**22.22**	**0**	**14.8147**	**0.9999**	0.0483	**-1.5**	**164.60576**
Constant motion for service brake							
constant	22.22	22.22	6.98	0.9999	0.016	0	155

motion							
Constant motion for emergency brake							
Constant motion	22.22	22.22	10.67	0.9999	0.016	0	237

3.7 Equation of motion for Ayat 1

❖ **Accelerated motion**

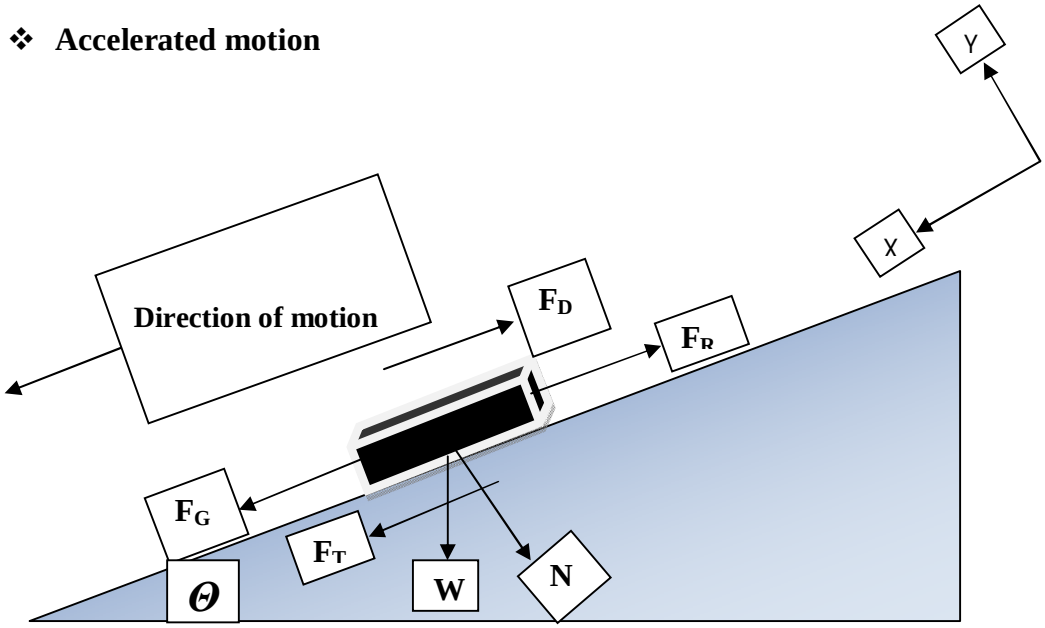


Figure 3.2 equation of motion for Ayat 1 during accelerated motion

Again using equations [16], [17], [18], [19], [20], [21] and [22] the following result is obtained

$ma = F_T + F_G - F_D - F_R$

$$\mathbf{a = \frac{[123654.6 - 100454.4\cos\alpha + 627840\sin\alpha]}{64000}} \quad \text{.....29}$$

❖ **Decelerated motion**

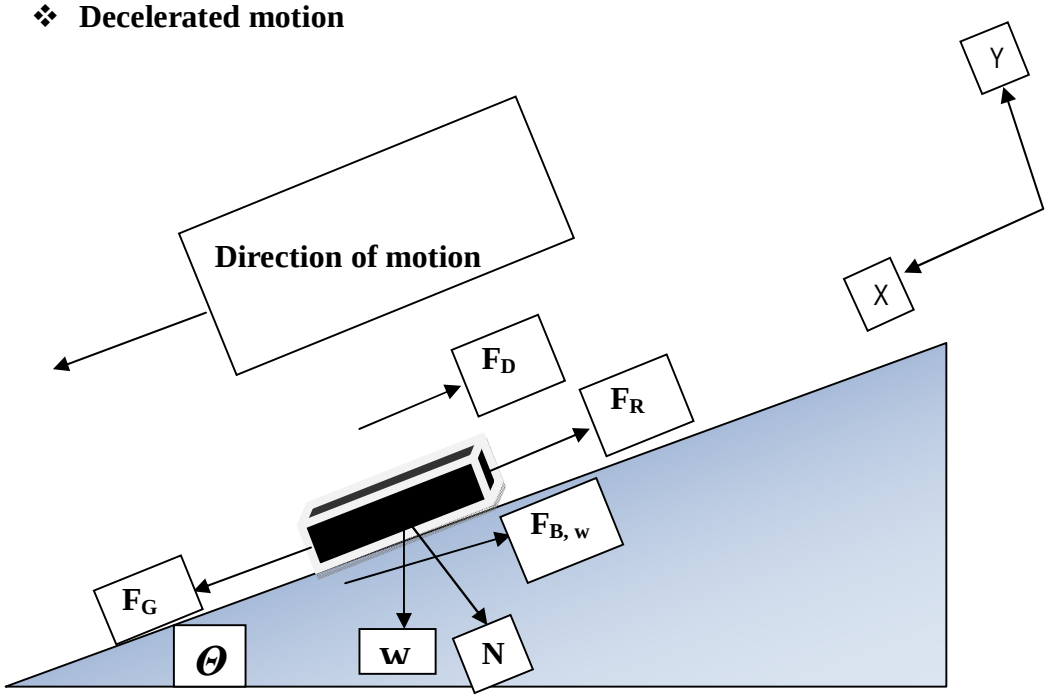


Figure 3.3 equation of motion for Ayat 1 during decelerated motion

Appling Newton's second law of motion results

$ma = F_G - F_D - F_R - F_{B,w}$

$F_{B,W} = -ma + F_G - F_D - F_R$

Using table 4 brake force of the wheel for this case will be

$F_{B,W}$ = -49398.757N for service brake

$F_{B,W}$ = -148398.757N for emergency brake

Decelerated equation of motion for service and emergency brake respectively will become

$$\mathbf{a} = \frac{-[155604.37 - 100454.4\cos\alpha + 627840\sin\alpha]}{64000} \quad \text{.....30}$$

$$\mathbf{a} = \frac{-[187604.37 - 100454.4\cos\alpha + 627840\sin\alpha]}{64000} \quad \text{.....31}$$

3.7.1 Service and emergency brake for Addis Ababa lrt at station: - between Ayat 1 and CMC 2/Meri

Table 8. Accelerated, normal service brake, normal emergency brake and constant motion between station Ayat 1 and Meri.

Name	Initial speed	Final speed	Time	cose_alpha	sine_alpha	Acceleration	Distance
acceleration 1	0	3.33	3.33	0.9999	0.0143	0.5	11.108889
acceleration 2	3.33	6.67	0.001	0.9999	0.0143	0.5	33.34
acceleration 3	6.67	10	3.334	0.9999	0.0143	0.5	55.551111
acceleration 4	10	13.33	6.667	0.9999	0.0143	0.5	77.768889
acceleration 5	13.33	16.67	9.999	0.9999	0.0143	0.5	100.02
acceleration 6	16.67	20	13.334	0.9999	0.0143	0.5	122.21111
acceleration 7	20	22.22	17.778	0.9999	0.0143	0.5	93.817284
accelerated motion	**0**	**22.22**	**22.22**	**0.9999**	0.0143	**0.5**	**493.7284**
Normal service brake							
service brake 1	22.22	20	2.222	0.9999	0.016	-1	46.908642
service brake 2	20	16.67	3.333	0.9999	0.016	-1	61.105556
service brake 3	16.67	13.33	3.334	0.9999	0.016	-1	50.01
service brake 4	13.33	10	3.333	0.9999	0.016	-1	38.884445
service brake 5	10	6.67	3.333	0.9999	0.016	-1	27.775556
service brake 6	6.67	3.33	3.334	0.9999	0.016	-1	16.67
service brake 7	3.33	0	3.333	0.9999	0.016	-1	5.5544445
decelerated motion	**22.22**	**0**	**22.222**	**0.9999**	0.016	**-1**	**246.90864**
Normal emergency brake							
emergency brake 1	22.22	20	1.48133	0.9999	0.016	-1.5	31.272428
emergency brake 2	20	16.67	2.222	0.9999	0.016	-1.5	40.737037
emergency brake 3	16.67	13.33	2.22267	0.9999	0.016	-1.5	33.34
emergency brake 4	13.33	10	2.222	0.9999	0.016	-1.5	25.922963
emergency brake 5	10	6.67	2.222	0.9999	0.016	-1.5	18.517037
emergency brake 6	6.67	3.33	2.22267	0.9999	0.016	-1.5	11.113333

emergency brake 7	3.33	0	2.222	0.9999	0.016	-1.5	3.702963
decelerated motion	**22.22**	**0**	**14.8147**	**0.9999**	0.016	**-1.5**	**164.60576**
Constant motion for service brake							
constant motion	22.22	22.22	13.19	0.9999	0.0143	0	293
Constant motion for emergency brake							
Constant motion	22.22	22.22	16.88	0.9999	0.0143	0	375

3.8 Equation of motion at Ayat 2

❖ **Accelerated motion**

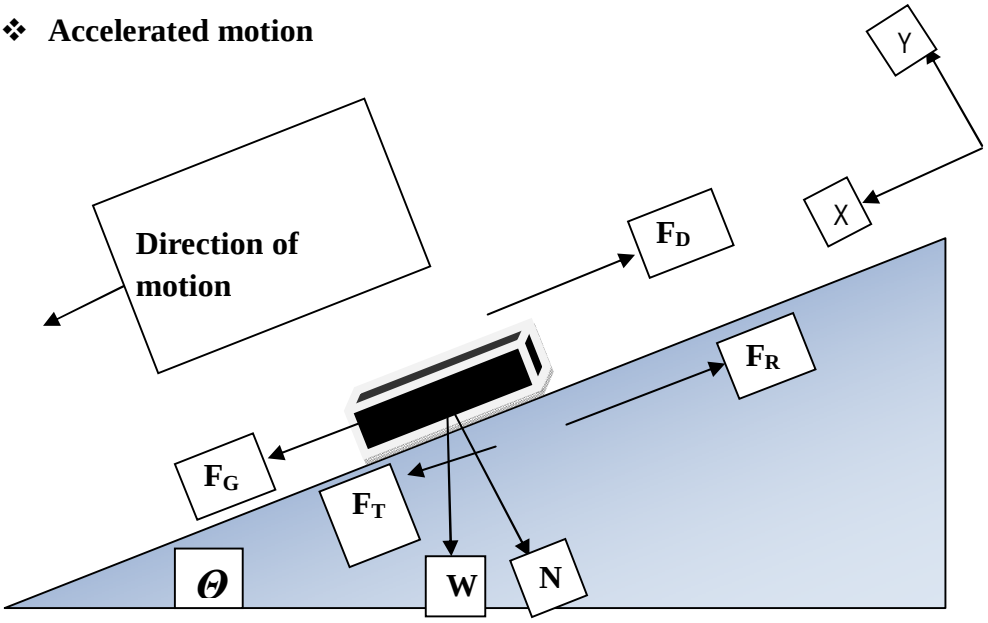


Figure 3.4 Equation of motion at station Ayat 2 during accelerated motion

Similarly using equations [16], [17], [18], [19], [20], [21] and [22] the following result is obtained

$ma = F_T + F_G - F_D - F_R$

$$\mathbf{a} = \frac{[111468.22 - 100454.4\cos\alpha + 627840\sin\alpha]}{64000} \quad \text{.....32}$$

❖ **Decelerated motion**

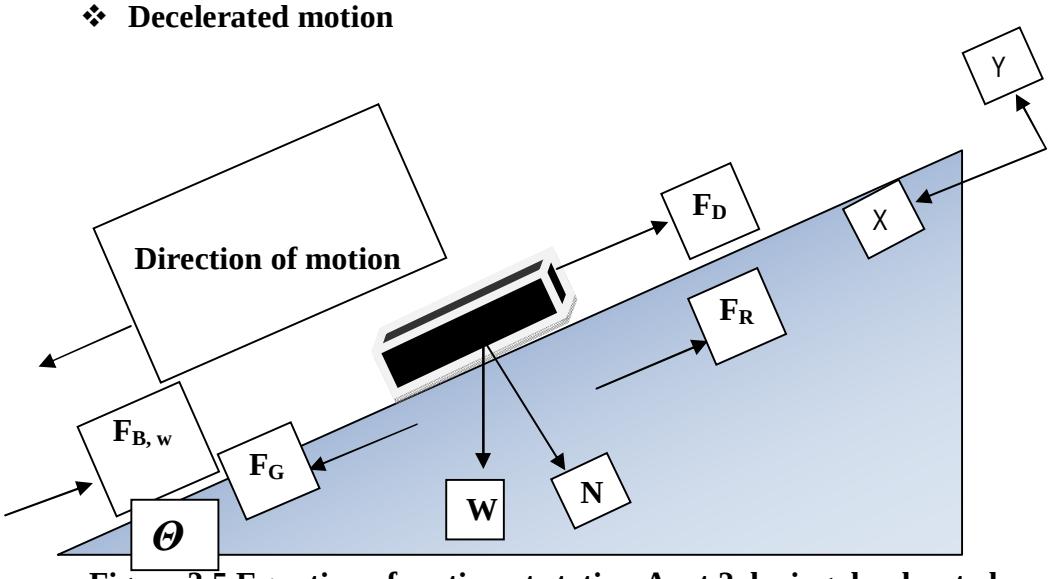


Figure 3.5 Equation of motion at station Ayat 2 during decelerated motion

Appling Newton's second law of motion results

$ma = F_G - F_D - F_R - F_{B,w}$

Decelerated equation of motion for service and emergency brake respectively will become

$$\mathbf{a} = \frac{-[143468.22 - 100454.4\cos\alpha + 627840\sin\alpha]}{64000} \quad \ldots\ldots 33$$

$$\mathbf{a} = \frac{-[175468.22 - 100454.4\cos\alpha + 627840\sin\alpha]}{64000} \quad \ldots\ldots 3$$

3.8.1 Excel file for service and emergency brake for Addis Ababa lrt at station: - between Ayat 2 and Ayat1

Table 9. Accelerated, normal service brake, normal emergency brake and constant motion between station Ayat 2 and Ayat 1

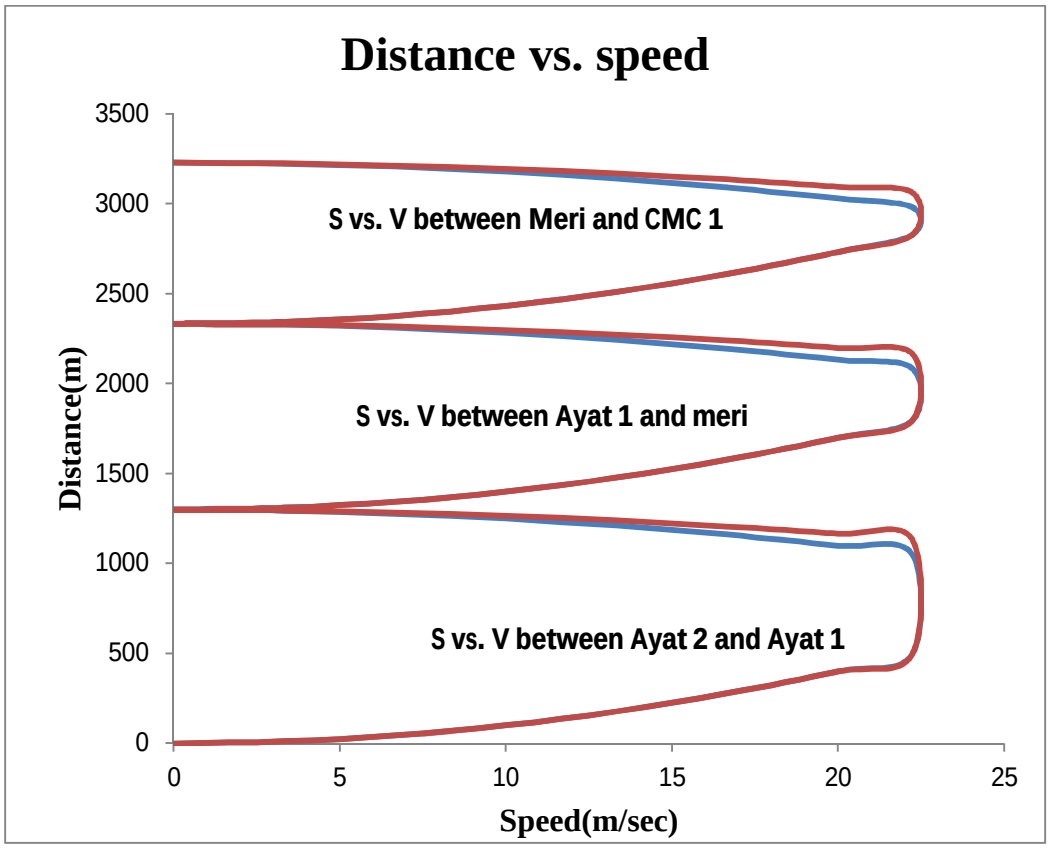
Name	Initial speed	Final speed	Time	cose_alpha	sine_alpha	Acceleration	Distance
acceleration 1	0	3.33	3.33	0.9999	0.0158	0.5	11.108889
acceleration 2	3.33	6.67	0.001	0.9999	0.0158	0.5	33.34
acceleration 3	6.67	10	3.334	0.9999	0.0158	0.5	55.551111
acceleration 4	10	13.33	6.667	0.9999	0.0158	0.5	77.768889
acceleration 5	13.33	16.67	9.999	0.9999	0.0158	0.5	100.02
acceleration 6	16.67	20	13.334	0.9999	0.0158	0.5	122.21111
acceleration 7	20	22.22	17.778	0.9999	0.0158	0.5	93.817284

accelerated motion	**0**	**22.22**	**22.22**	**0.9999**	**0.0158**	**0.5**	**493.7284**
Normal service brake							
service brake 1	22.22	20	2.222	0.9999	0.0143	-1	46.908642
service brake 2	20	16.67	3.333	0.9999	0.0143	-1	61.105556
service brake 3	16.67	13.33	3.334	0.9999	0.0143	-1	50.01
service brake 4	13.33	10	3.333	0.9999	0.0143	-1	38.884445
service brake 5	10	6.67	3.333	0.9999	0.0143	-1	27.775556
service brake 6	6.67	3.33	3.334	0.9999	0.0143	-1	16.67
service brake 7	3.33	0	3.333	0.9999	0.0143	-1	5.5544445
decelerated motion	**22.22**	**0**	**22.222**	**0.9999**	**0.0143**	**-1**	**246.90864**
Normal emergency brake							
emergency brake 1	22.22	20	1.48133	0.9999	0.0143	-1.5	31.272428
emergency brake 2	20	16.67	2.222	0.9999	0.0143	-1.5	40.737037
emergency brake 3	16.67	13.33	2.22267	0.9999	0.0143	-1.5	33.34
emergency brake 4	13.33	10	2.222	0.9999	0.0143	-1.5	25.922963
emergency brake 5	10	6.67	2.222	0.9999	0.0143	-1.5	18.517037
emergency brake 6	6.67	3.33	2.22267	0.9999	0.0143	-1.5	11.113333
emergency brake 7	3.33	0	2.222	0.9999	0.0143	-1.5	3.702963
decelerated motion	**22.22**	**0**	**14.8147**	**0.9999**	**0.0143**	**-1.5**	**164.60576**
constant motion for service brake							
constant	22.22	22.22	25.16	0.9999	0.0158	0	559

motion							
constant motion for emergency brake							
constant motion	22.22	22.22	28.85	0.9999	0.0158	0	641

3.9 Summary of distance vs. speed graph at three stations becomes

From the graph below blue line indicates service brake for all the three stations and red line indicates emergency brakes respectively



Graph 2. Summary of distance vs. speed graph at three stations for service and emergency brake

3.10 Heat generated at Ayat 1 during service and emergency brake

The heat generated between the pad and the disc is vary within time

$$Q_t = \frac{\frac{1}{2}\times\beta\times M\times(V^2-V0^2)}{\Delta t} - \frac{M\times g\times\sin\alpha\times s}{\Delta t}$$

Where: M - mass of the Addis Ababa LRT Train

V and V_0 - are train's final and initial velocities respectively

g - acceleration due to gravity

S and t - braking distance and braking time of the train respectively

α – gradient

To include the rotational effect of different components' the value of β is included with a value of 1.1. [Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009].

Substitute the values

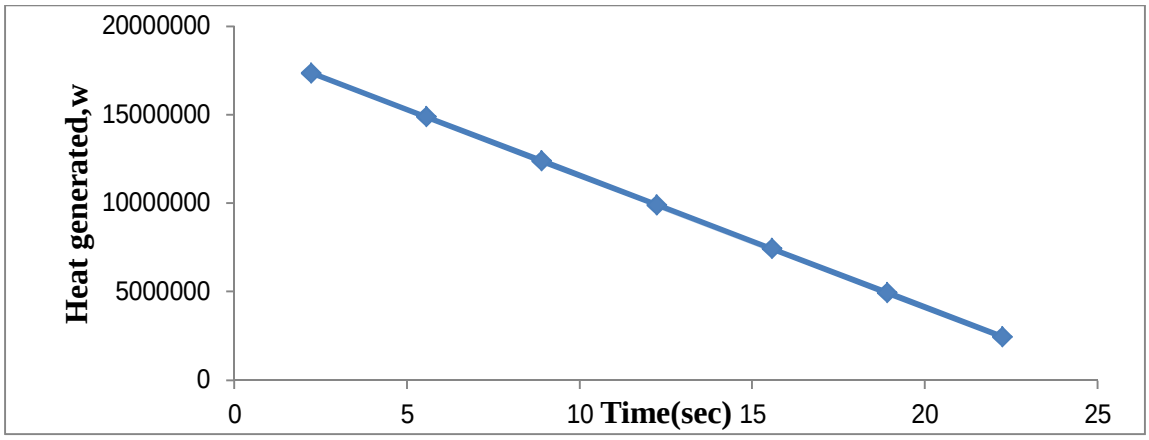
Variation of Q_t with time t becomes as follows.

Table 10. Heat generation vs. time for service brake @ Ayat1

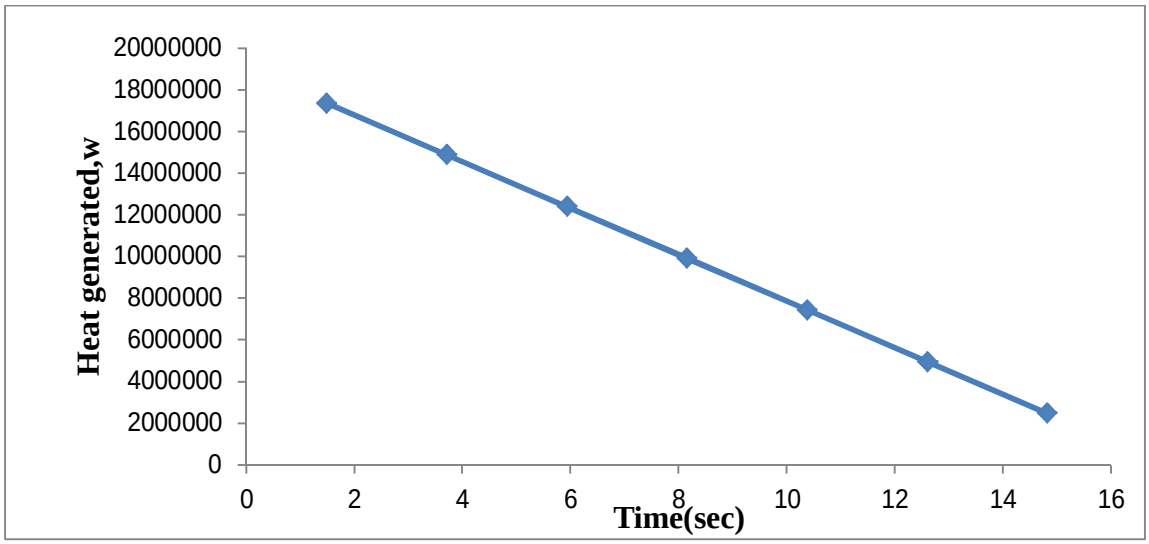
V^2	V_O^2	s	t	sin_alpha	Q
493.7284	400	47	2.22	0.0143	17380000
400	277.8889	61	5.553	0.0143	14897143
277.8889	177.6889	50	8.887	0.0143	12414286
177.6889	100	39	12.22	0.0143	9931429
100	44.4889	28	15.553	0.0143	7448571
44.4889	11.0889	17	18.887	0.0143	4965714
11.0889	0	6	22.222	0.0143	2482857

Table 11. Heat generation vs. time for emergency brake @ Ayat1

V^2	V_O^2	s	t	sin_alpha	Q
493.7284	400	19	1.48133	0.0143	17380000
400	277.8889	24	3.70333	0.0143	14897143
277.8889	177.6889	20	5.926	0.0143	12414286
177.6889	100	16	8.148	0.0143	9931429
100	44.4889	11	10.37	0.0143	7448571
44.4889	11.0889	7	12.59267	0.0143	4965714
11.0889	0	2	14.81467	0.0143	2482857



Graph 3. Relationships between Qt and time for service brake @ Ayat1



Graph 4. Relationships between Qt and time for emergency brake @ Ayat1

Again using the above procedure the heat generated at Meri/CMC 2 during service and emergency brake with maximum value of 17.37×10^6 and minimum value of 2.48×10^6.

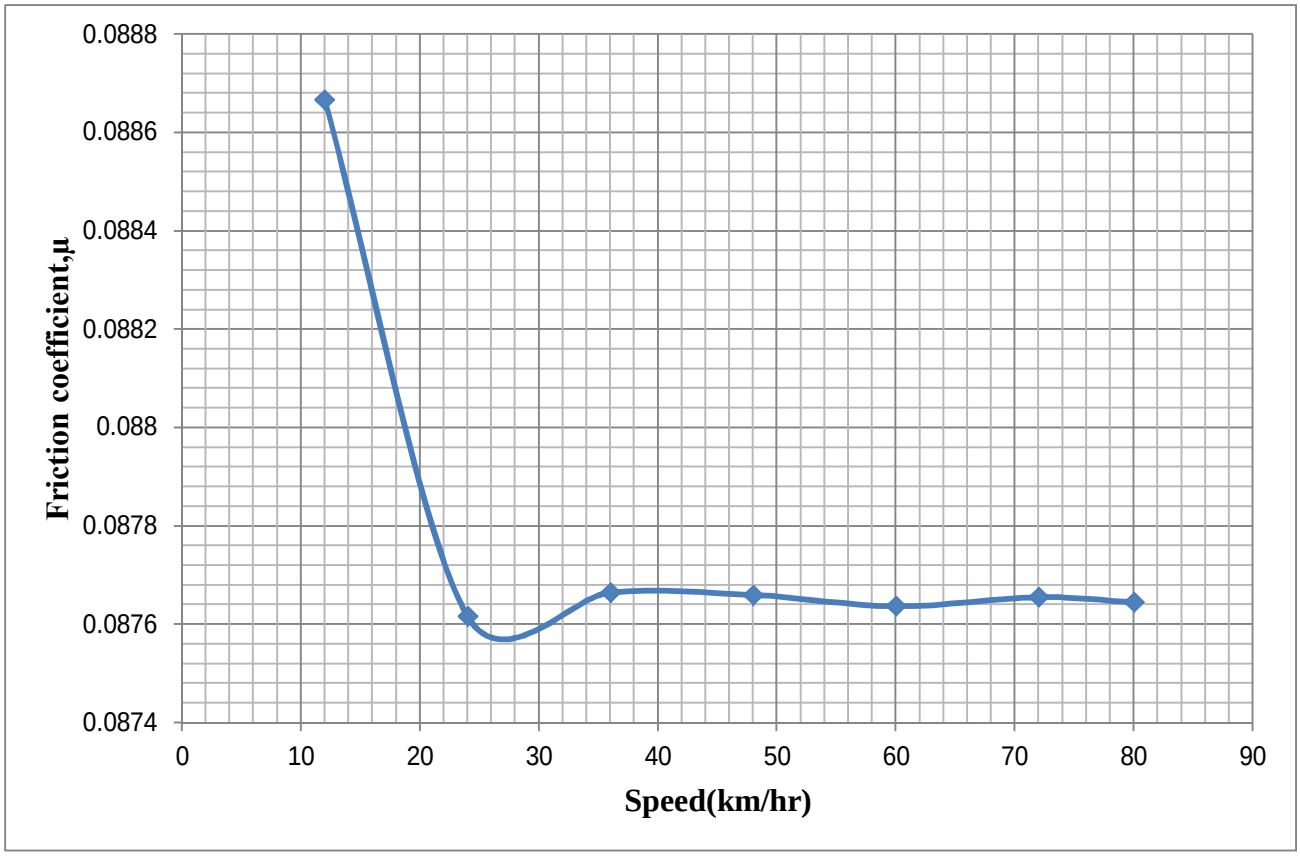
Similarly at station CMC 1 the heat generated value during service and emergency brake will become with maximum value of 17.35×10^6 and minimum value of 2.47×10^6. Using the same braking time for all the three stations. The pattern of resulting graph is resemble for all the three stations and look like with that of heat generated energy at station Ayat 1.

3.11 Determination of friction coefficient

3.11.1 Ayat 1:-service brake

Table 12 coefficient of friction at Ayat 1 during service brake

V	V^2	V_0	V_0^2	$V^2 - V_0^2$	s	2g	p	α	μ
80	6400	72	5184	1216	46.91	19.62	12.96	0.0143	0.08765
72	5184	60	3600	1584	61.1	19.62	12.96	0.0143	0.087655
60	3600	48	2304	1296	50	19.62	12.96	0.0143	0.087637
48	2304	36	1296	1008	38.88	19.62	12.96	0.0143	0.08766
36	1296	24	576	720	27.77	19.62	12.96	0.0143	0.087665
24	576	12	144	432	16.67	19.62	12.96	0.0143	0.087616
12	144	0	0	144	5.5	19.62	12.96	0.0143	0.088666



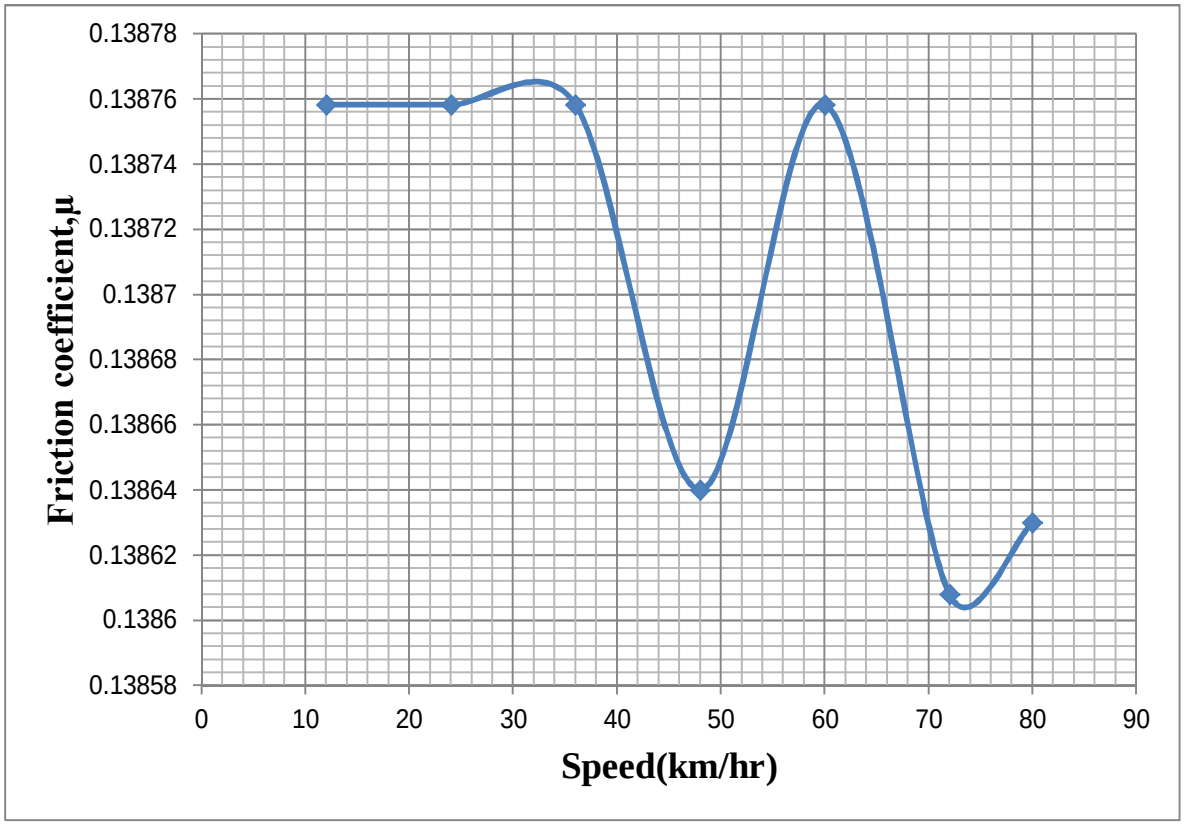
Graph 5 Relationships between μ and V for service brake @ Ayat1

The slope of the above graph is -9×10^{-6} and its y-intercept is 0.0882.

3.11.2 Ayat 1:-emergency brake

Table 13 coefficient of friction at Ayat 1 during emergency brake

V	V^2	V_0	V_0^2	$V^2 - V_0^2$	s	2g	p	α	μ
80	6400	72	5184	1216	31.27	19.62	12.96	0.0143	0.13863
72	5184	60	3600	1584	40.74	19.62	12.96	0.0143	0.138608
60	3600	48	2304	1296	33.3	19.62	12.96	0.0143	0.138758
48	2304	36	1296	1008	25.92	19.62	12.96	0.0143	0.13864
36	1296	24	576	720	18.5	19.62	12.96	0.0143	0.138758
24	576	12	144	432	11.1	19.62	12.96	0.0143	0.138758
12	144	0	0	144	3.7	19.62	12.96	0.0143	0.138758



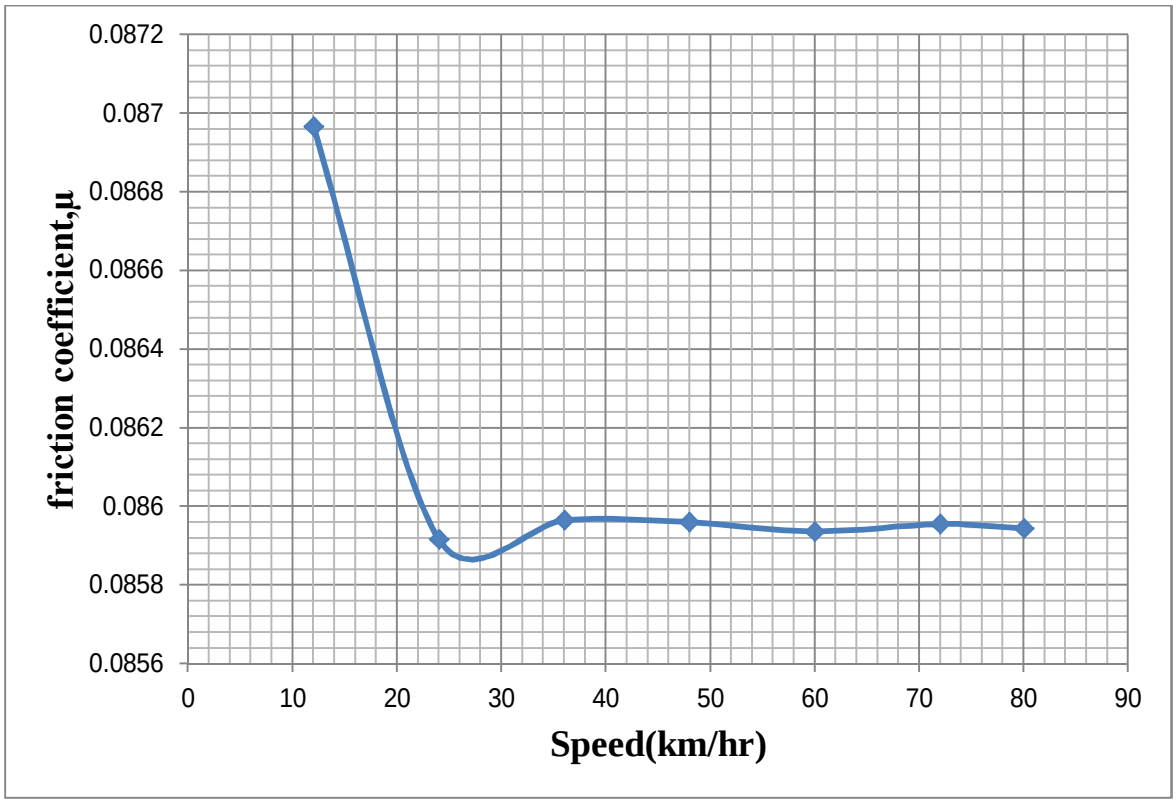
Graph 6 Relationships between μ and V for emergency brake @ Ayat1

The slope of the above graph is -2×10^{-6} and its y-intercept is 0.1388.

3.11.3 Meri/CMC 2/:-service brake

Table 14 coefficient of friction at Meri during service brake

V	V^2	V_0	V_0^2	$V^2 - V_0^2$	s	2g	p	α	μ
80	6400	72	5184	1216	46.91	19.62	12.96	0.016	0.08594
72	5184	60	3600	1584	61.1	19.62	12.96	0.016	0.08596
60	3600	48	2304	1296	50	19.62	12.96	0.016	0.08594
48	2304	36	1296	1008	38.88	19.62	12.96	0.016	0.08596
36	1296	24	576	720	27.77	19.62	12.96	0.016	0.08597
24	576	12	144	432	16.67	19.62	12.96	0.016	0.08592
12	144	0	0	144	5.5	19.62	12.96	0.016	0.08697



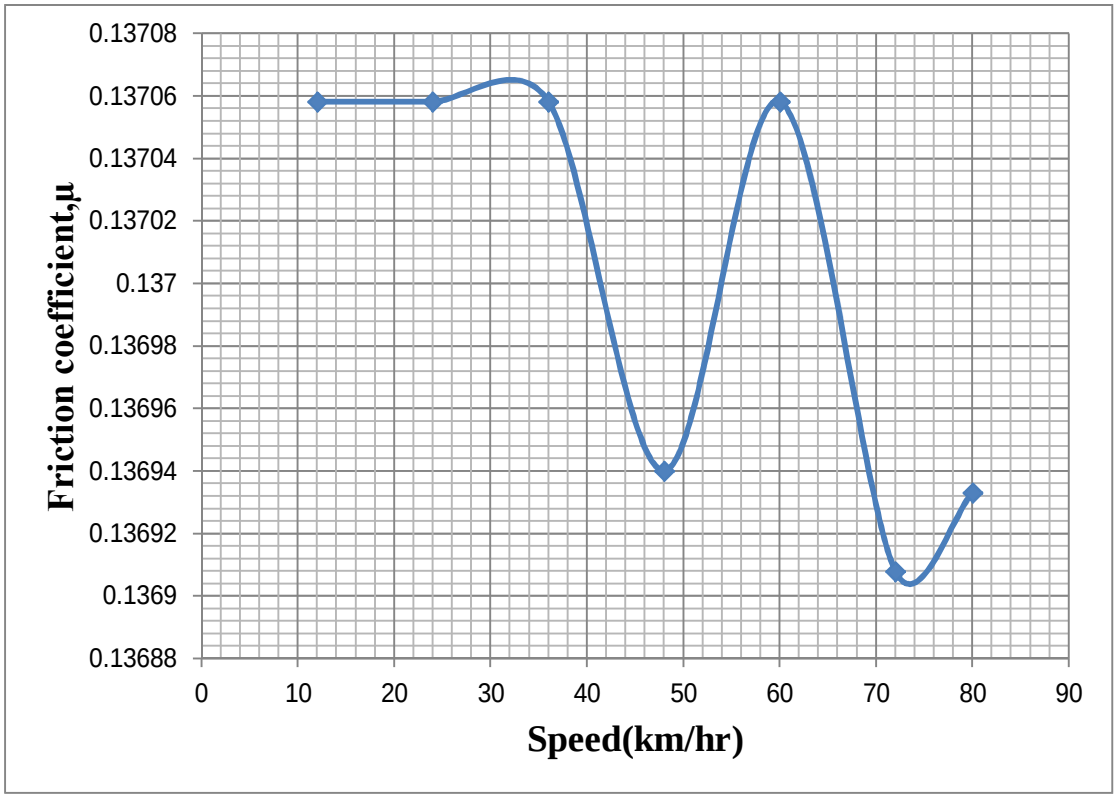
Graph 7 Relationships between μ and V for service brake @ Meri

The slope of the above graph is -9×10^{-6} and its y-intercept is 0.0865.

3.11.4 Meri/CMC 2/:-emergency brake

Table 15 coefficient of friction at Meri during emergency brake

V	V^2	V_0	V_0^2	$V^2 - V_0^2$	s	2g	p	α	μ
80	6400	72	5184	1216	31.27	19.62	12.96	0.016	0.13693
72	5184	60	3600	1584	40.74	19.62	12.96	0.016	0.13691
60	3600	48	2304	1296	33.3	19.62	12.96	0.016	0.13706
48	2304	36	1296	1008	27.77	19.62	12.96	0.016	0.13694
36	1296	24	576	720	18.5	19.62	12.96	0.016	0.13706
24	576	12	144	432	11.1	19.62	12.96	0.016	0.13706
12	144	0	0	144	3.7	19.62	12.96	0.016	0.13706



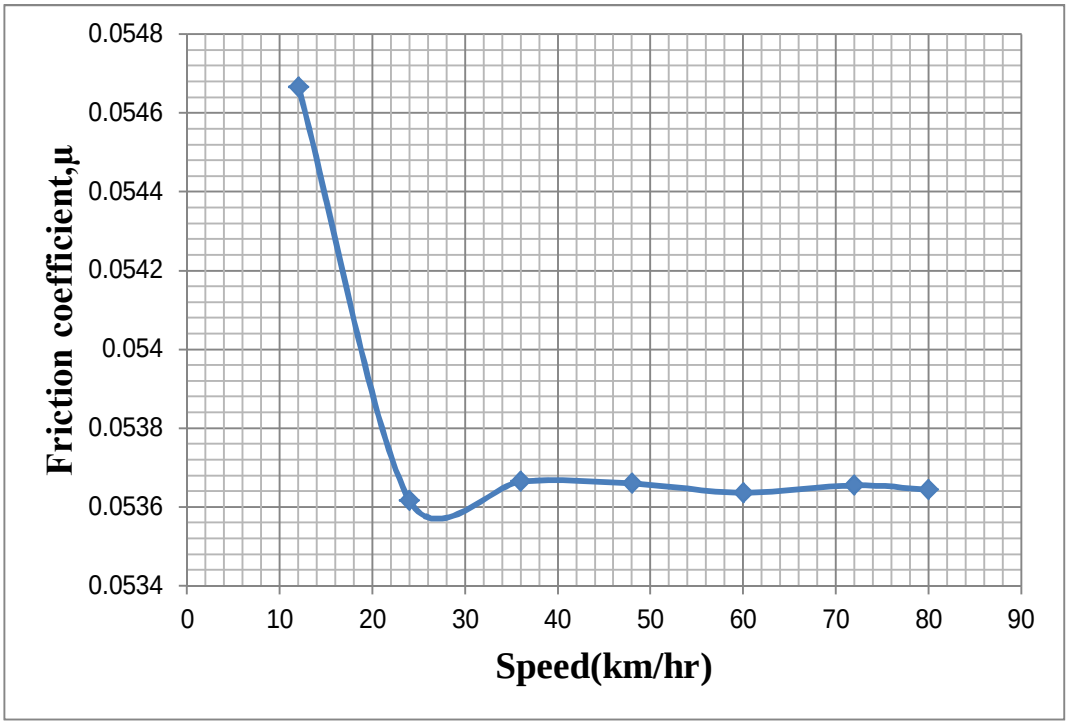
Graph 8 Relationships between μ and V for emergency brake @ Meri

The slope of the above graph is -2×10^{-6} and its y-intercept are 0.1371.

3.11.5 CMC 1:-service brake

Table 16 coefficient of friction at CMC 1 during service brake

V	V^2	V_0	V_0^2	$V^2 - V_0^2$	s	2g	p	α	μ
80	6400	72	5184	1216	46.91	19.62	12.96	0.0483	0.05364
72	5184	60	3600	1584	61.1	19.62	12.96	0.0483	0.05366
60	3600	48	2304	1296	50	19.62	12.96	0.0483	0.05364
48	2304	36	1296	1008	38.88	19.62	12.96	0.0483	0.05366
36	1296	24	576	720	27.77	19.62	12.96	0.0483	0.05367
24	576	12	144	432	16.67	19.62	12.96	0.0483	0.05362
12	144	0	0	144	5.5	19.62	12.96	0.0483	0.05467



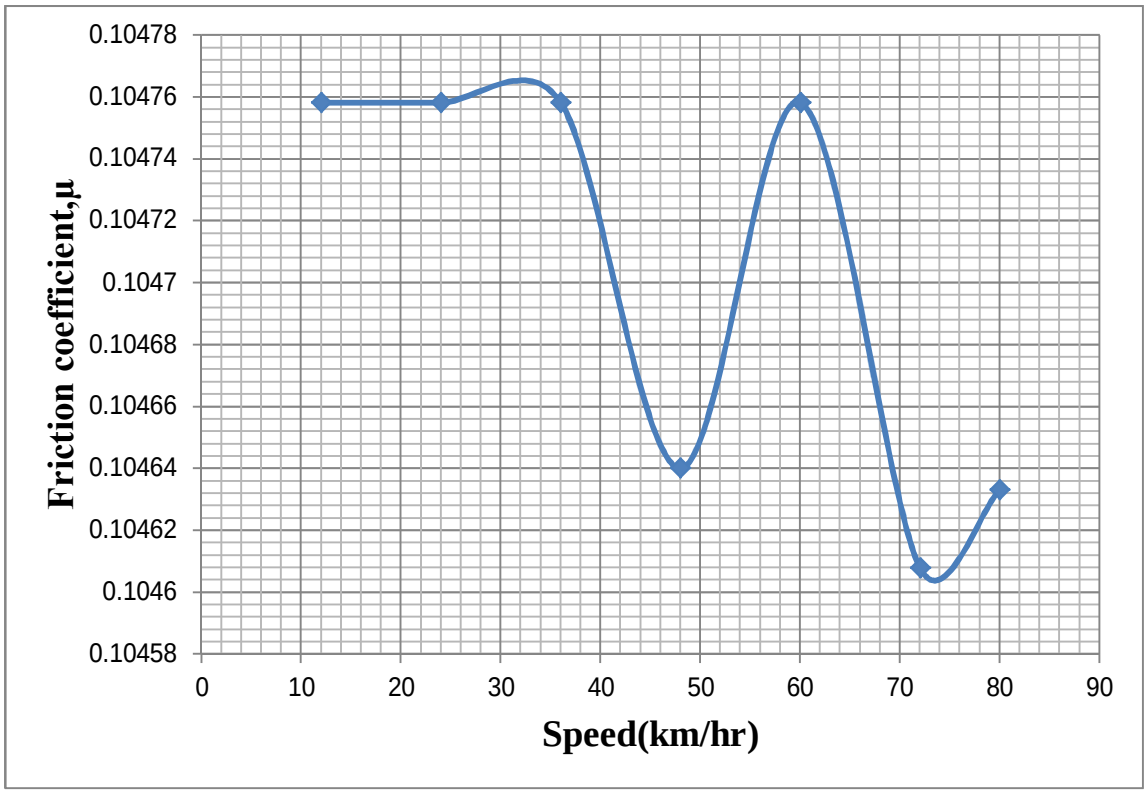
Graph 9 Relationships between μ and V for service brake @ CMC 1

The slope of the above graph is -9×10^{-6} and its y-intercept is 0.0542.

3.11.6 CMC 1:-emergency brake

Table 17 coefficient of friction at CMC 1 during emergency brake

V	V^2	V_0	V_0^2	$V^2 - V_0^2$	s	2g	p	α	μ
80	6400	72	5184	1216	31.27	19.62	12.96	0.0483	0.10463
72	5184	60	3600	1584	40.74	19.62	12.96	0.0483	0.10461
60	3600	48	2304	1296	33.3	19.62	12.96	0.0483	0.10476
48	2304	36	1296	1008	25.92	19.62	12.96	0.0483	0.10464
36	1296	24	576	720	18.5	19.62	12.96	0.0483	0.10476
24	576	12	144	432	11.1	19.62	12.96	0.0483	0.10476
12	144	0	0	144	3.7	19.62	12.96	0.0483	0.10476



Graph 10 Relationships between μ and V for emergency brake @ CMC 1

The slope of the above graph is -2×10^{-6} and its y-intercept is 0.1048.

CHAPTER-4

PHYSICAL MODEL OF DISC BRAKE

Table 18 illustrates some of the vehicle properties of Addis Ababa light rail transit. The data which is going to be used during the Ansys simulation processes are also included which are calculated in the previous chapters of this paper.

Table 18.vehicle properties

Mass of the vehicle-M [Kg]	64000	
Maximal load per axle [Kg]	11000	
Number of axles per vehicle	4	
Number of discs per axle	2	
Start speed- v_o [m/s]	22.22	
Deceleration-a [m/s^2]	Service brake = 1	
	Emergency brake = 1.5	
Braking time - t_s [s]	station at Ayat 1	Service =22.22
		Emergency = 14.82
	station at Meri	Service = 22.22
		Emergency = 14.82
	station at CMC 1	Service = 22.22
		Emergency =14.82
Effective radius of the disc brake-r_{disc} [m]	0.113	
Radius of the wheel - r_{wheel} [m]	0.33	
Friction coefficient disc/pad-μ [/]	0.2	

4.1 Material Properties of grey cast iron

For the transient analysis of the passenger train disc brake certain physical properties of materials of the disc, given in Table 19, were required.

Table 19. Material properties of grey cast iron

Density [kg/m^3]	7250
Modulus of elasticity [GPa]	138
Poison's ratio	0.28
Thermal conductivity [w/m-k]	57

Specific heat capacity [J/kg-k]	460
Linear expansion coefficient [k^{-1}]	0.000006
Tensile strength [Mpa]	430
Yield stress [Mpa]	400
Thermal expansion [μm/m-k]	10.85

[Source: shigley mechanical engineering design, eighth edition budynas- nisbett]

4.2 Disc brake calculation

Step-1:-

Kinetic Energy (K. E_T) = ½×β×M×V^2

= ½×1.1×64000×22.22^2 Joules

= 17.38×10^6 Joules

Potential energy (P.E) = M×g×Sinα

= horizontal rail with no elevation difference at Ayat 1

= 64000×9.81×0.016 = 10,045.44 Joules at CMC 2/Meri

= 64000×9.81×0.0483 = 30,324.672 Joules at CMC 1

Total energy = Kinetic Energy + Potential Energy+ Rotational kinetic energy

= 17.38×10^6 - 10,045.44 Joules

= 17.37×10^6 Joules at CMC 2/Meri

= 17.38×10^6 - 30,324.672 Joules

= 17.35×10^6 Joules at CMC 1

Step-2:-

The total energy= the heat generated Qg = 17.38×10^6 Joules at Ayat 1,17.35×10^6 Joules at CMC 1 and 17.37×10^6 Joules at CMC 2/Meri

Step-3:-

Area of the rubbing faces

$$A = 2\pi \times (r_o^2 - r_i^2)$$

$$= 0.0966\ m^2$$

Step-4:-

Heat Flux = Heat Generated / Time / Twice the projected area

$$= \frac{17.38\times10^6}{22.22\times2\times0.0966}\ \frac{w}{m^2}$$

$= 4.05\times10^6\ \frac{w}{m^2}$……station @ Ayat 1

$= 4.046\times10^6\ \frac{w}{m^2}$……...station @ Meri/CMC 2 } service brake

$= 4.04\times10^6\ \frac{w}{m^2}$……station @ CMC 1

$$= \frac{17.38\times10^6}{15\times2\times0.0966}\ \frac{w}{m^2}$$

$= 6\times10^6\ \frac{w}{m^2}$……..station @ Ayat 1

$= 5.99\times10^6\ \frac{w}{m^2}$……..station @ Meri/CMC 2 } emergency brake

$= 5.98\times10^6\ \frac{w}{m^2}$……..station @ CMC 1

Basic data	W	2,5	(N/kN)	avg. rolling resistance
	η_{mf}	0,75		efficiency brake transmission
	m_{fb}	0,35		brake pad grip
	z	1,15		Rotational coefficient
	s_{m10}	4,5	(m)	prescribed max. stopping distance braked from 10km/h

[Source: Standard data from INVENT Engineering Ltd H- 1086 Budapest, Csobánc u. 11. VI/3]

The analysis is done by taking braking efficiency 75% and heat distributed from front and rear axle is 75:25

Heat Flux = $4.05\times10^6 \times 0.75$

$= 6\times10^6 \times 0.75$

$= 3.04\times10^6 \frac{w}{m^2}$, $3.03\times10^6 \frac{w}{m^2}$, $3.03\times10^6 \frac{w}{m^2}$…….station @ Ayat 1, CMC 2, and CMC1 service brake respectively.

$= 4.5\times10^6 \frac{w}{m^2}$, $4.49\times10^6 \frac{w}{m^2}$, $4.49\times10^6 \frac{w}{m^2}$, $4.48\times10^6 \frac{w}{m^2}$……station @ Ayat 1, CMC 2, and CMC 1 emergency brake respectively.

4.3 Initial temperature for the next braking action

The thermal capacity of disc brake is temperature dependent. The heat output of this model is composed of convection and radiation. The temperature drop of a brake rotor occurs due to the effect of station stopping time of the passengers train, accelerated and constant traveling time of the train. Therefore for the next braking action the initial temperature of the disc will rise above from its ambient temperature and will be computed according to Newton's law of cooling be described as:

$$T = T_\infty + [(T_1 - T_\infty) \times [\exp\{-\frac{UA}{mC} \times t\}]]$$

Where: T:-cooled temperature

T_1:- maximum temperature

T_∞ :-ambient temperature

A: - disc brake contact area

U: - heat transfer coefficient

T: - delay time, station stopping, accelerated and constant travel time

M: - mass of disc

C: - specific heat capacity of grey cast iron

The cooling of the brake disc is assumed independent on the state of the brake pad. This is to simplify the model and is justified by the fact that the disc absorbs almost 100 times more energy than the pad. The disc cools down due to convection, radiation and conduction. While driving, convection is by far the dominant mode of heat loss. Therefore, radiation is assumed negligibly small.

Finally using the effect of time on the cooling of the disc brake before the second braking action starts will be shown below. The temperature drop will be obtained using calculated value of convective heat transfer coefficient. i.e.

Using Thermal properties of air

Pressure: - 1.01 bars

Ambient temperature where the disc operated is: - 31.8°C

Density: - 1.19145kg/m3

Prandtl number: - 0.714104

The convective coefficient h is

$$h = \frac{k_a \times Nu}{2r} = \frac{0.037 \times pr^{0.33} \left(k_{a\,/\,\mu_a^{0.8}}\right)(rw)^{0.8}}{L^{0.2}}$$

$$= \frac{0.037 \times (0.714104^{0.33}) \times \left(\frac{57 \times 10^{-3}}{(15.15 \times 10^{-6})^{0.8}}\right) \times (0.113 \times 67.33)^{0.8}}{(0.226)^{0.2}}$$

$$= 92.40 \frac{w}{m^2 \times ℃}$$

Next substitute the value of h in the above equation to obtain the maximum temperature drop.

$$T = 31.8 + [(268.96 - 31.8) \times \exp\left(\frac{-92.4 \times 0.0966}{40.86 \times 0.46} \times 90\right)]$$

=31.8 °C, cooling temperature during service brake at station Meri using station delay time for free convection

$$T = 31.8 + [(268.96-31.8) \times \exp(\frac{-92.4\times0.0966}{40.86\times0.46}\times35.41)]$$

= 31.8 °C, cooling temperature during service brake at station Meri using accelerated and constant travel time for forced convection

T = (31.8 + 31.8) °C

T = 63.6 °C, initial temperature during service brake at station Meri/CMC 2

Again for emergency brake it will become as follows

T = 63.6 °C, initial temperature during emergency brake at station Meri/CMC 2

Similarly, the cooling temperature during service and emergency brake at station CMC 1 will follow the same procedure as above and will result as follows

T = 63.6 °C, initial temperature during service brake at station CMC 1

T = 63.6 °C, initial temperature during emergency brake at station CMC 1 Therefore, for all stations the cooling temperature is calculated using the above procedure.

4.4 Disc brake assembly model in solid works

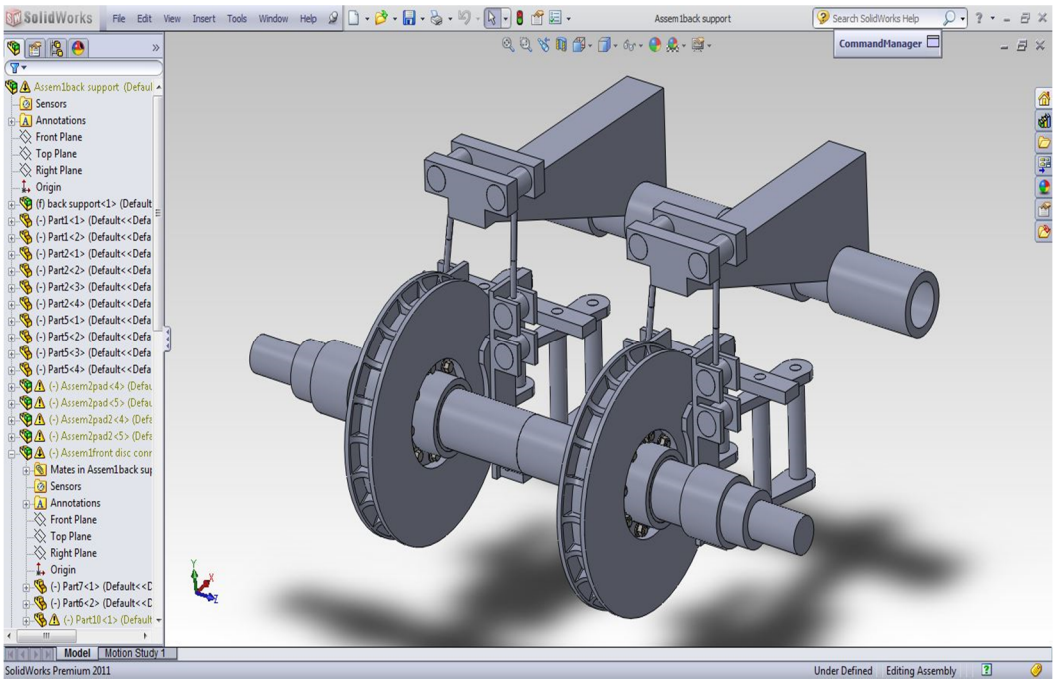


Figure 3.6 Train disc brake solid work assembly

Disc brake

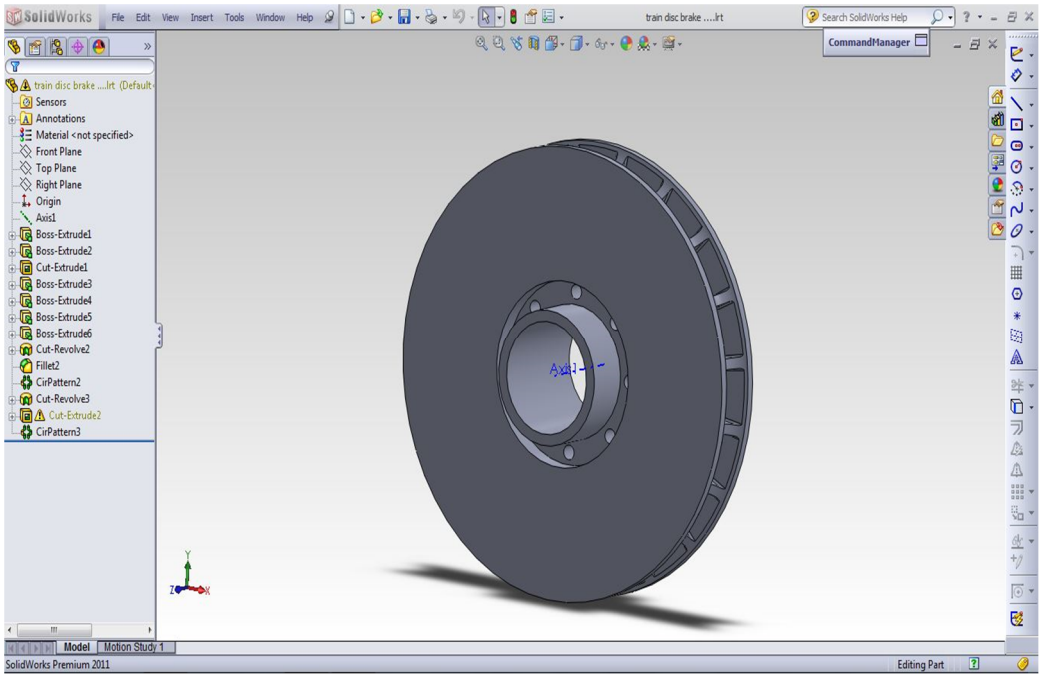


Figure 3.7 Disc brake models

CHAPTER 5

ANSYS RESULTS AND DISCUSSIONS

5.1 MODELING IN ANSYS

The brake disc was analyzed using the ANSYS software package. The Ansys solver automatically calculates temperature at the wall boundary and contact rotor-pad surface. Afterwards the heat transfer coefficients by considering convection were organized in such a way, that they could be used as boundary condition in thermal analysis.

5.1.1 Material

For the analysis of the discs certain physical properties of materials, given in Table 19, were required.

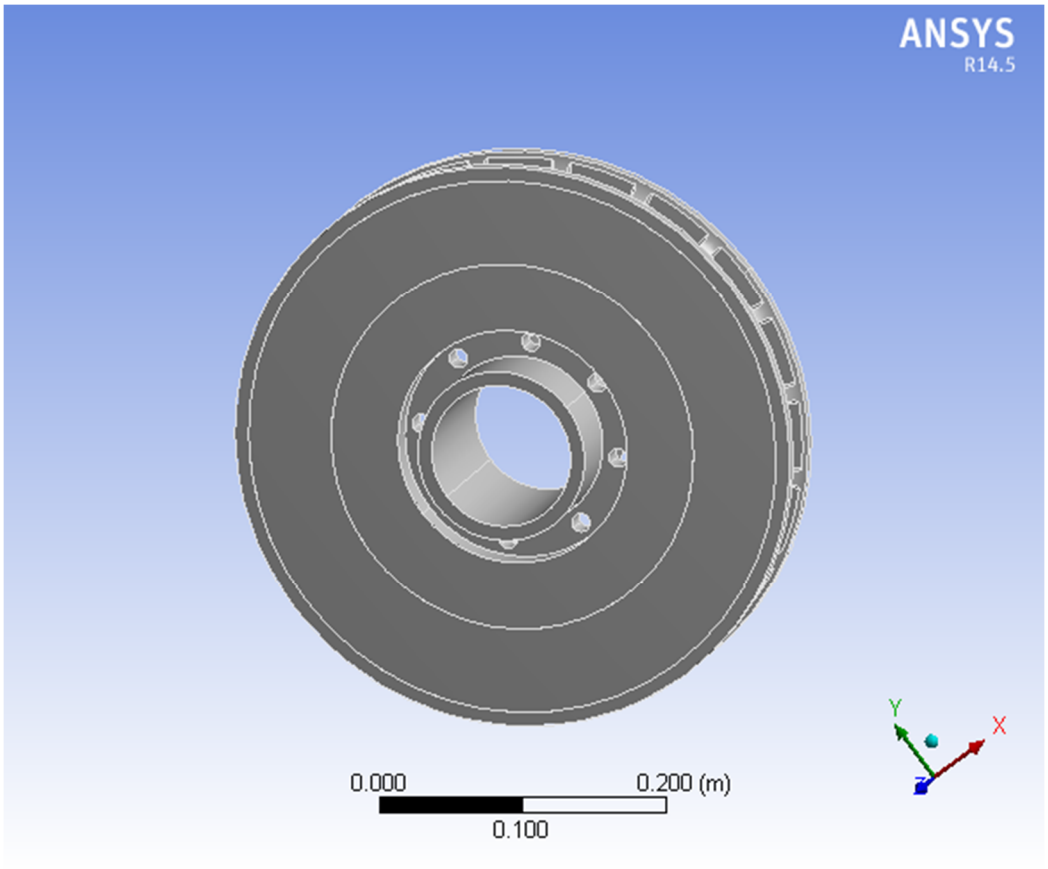


Figure 5.1 brake disc model

5.1.2 Mesh Generation

Once the disc brake model was done then it is crucial to think about to select what type of mesh will be used for the analysis purpose whether a free mesh or a mapped mesh. Compared to the free mesh, a mapped mesh is restricted in terms of the element shape it contains and pattern of the mesh. A mapped mesh contains either only quadrilateral or only triangular element, while a mapped volume mesh contains only hexahedral elements. In addition, a mapped mesh typically has a regular pattern, with obvious rows of elements. For mapped mesh, we must build the geometry as a series of fairly regular volumes and/or areas that can accept a mapped mesh. The type of mesh generation considered here is a mapped mesh automatic generation nodes of 427157 and 254440 elements.

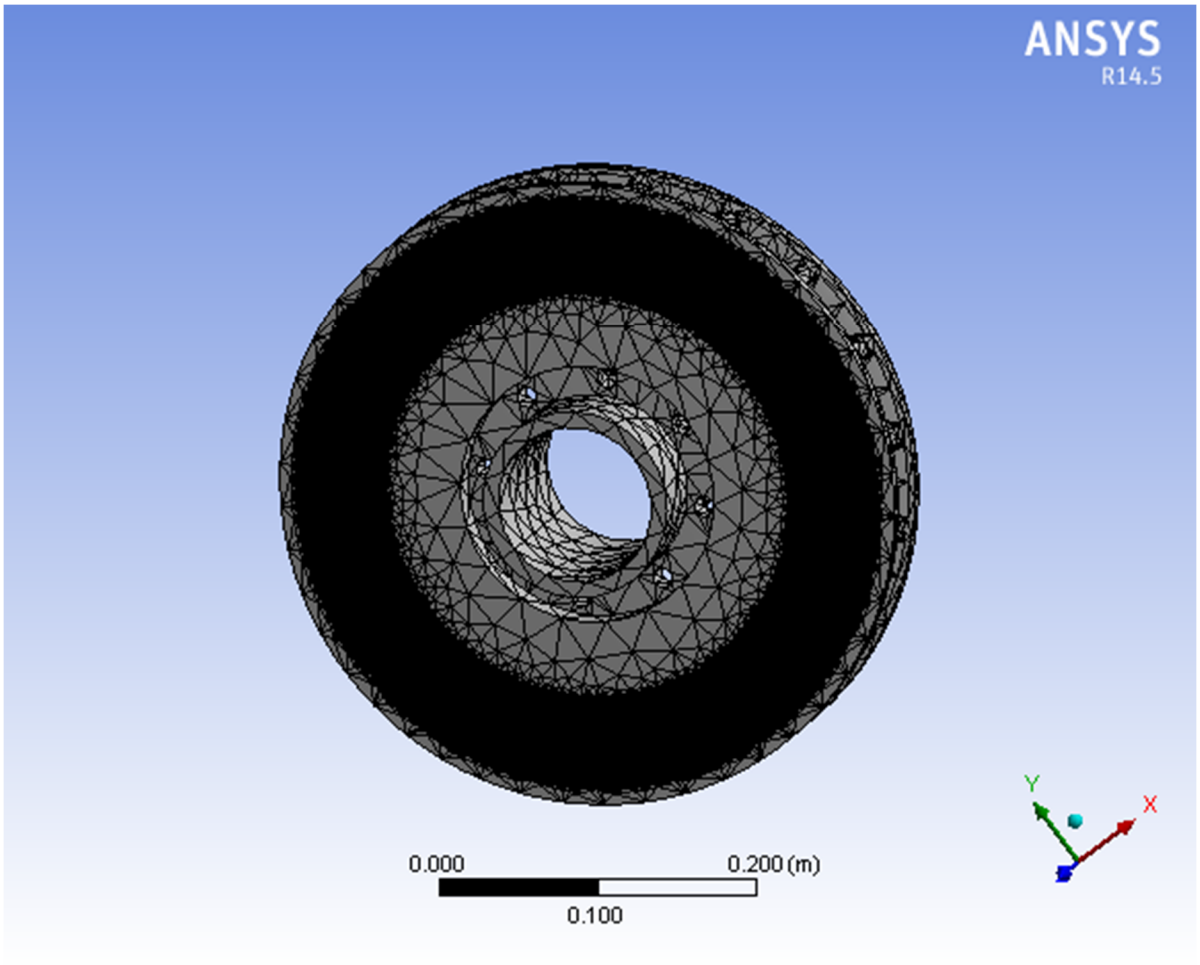


Figure 5.2 Meshing of the disc

5.1.3 Boundary Conditions

The boundary conditions are introduced into module ANSYS Workbench [Multiphysics], by choosing the mode of first simulation like thermal transient, and by defining the physical properties of materials. These conditions constitute the initial conditions of our simulation. After having fixed these parameters, one introduces a boundary condition associated with each surface.

- Initial Temperature of the disc = 31.8 [°C] during service and emergency brake at station Ayat 1
  63.6[°C] during service brake and emergency brake at Meri and CMC 1 respectively.
- Materials: Grey Cast iron FG 15.
- Convection: One introduces the values of the heat transfer coefficient (h) from standard.
- Flux: One introduces the values obtained.
- Geometric Boundary conditions: All the nodes in the hub radius are fixed. So the nodal displacements in the hub bore become Zero i.e. both in radial and axial direction and nodes on flange are also fixed in Y-direction.

5.2 Thermal Analysis

A Thermal analysis calculates the temperature distribution and related thermal quantities in a system or component. Typical thermal quantities are:

1. The temperature distributions
2. The amount of heat lost or gained and
3. Thermal fluxes

5.2.1 Types of Thermal Analysis:

1. A Steady State Thermal Analysis determines the temperature distribution and other thermal quantities under steady state loading conditions. A steady state loading condition is a situation where heat storage effects varying over a period of time can be ignored.

2. A Transient thermal analysis determines the temperature distribution and other thermal quantities under conditions that vary over a period of time.

Since in brake disc simulation the heat flux as well as temperature varies with time. Therefore rather than steady state thermal analysis, for this paper work transient thermal analysis is used. After applying the necessary loads the following result obtained under different circumstances during service and emergency brake in the following stations.

5.3 Results

5.3.1 Service brake for station Ayat 1

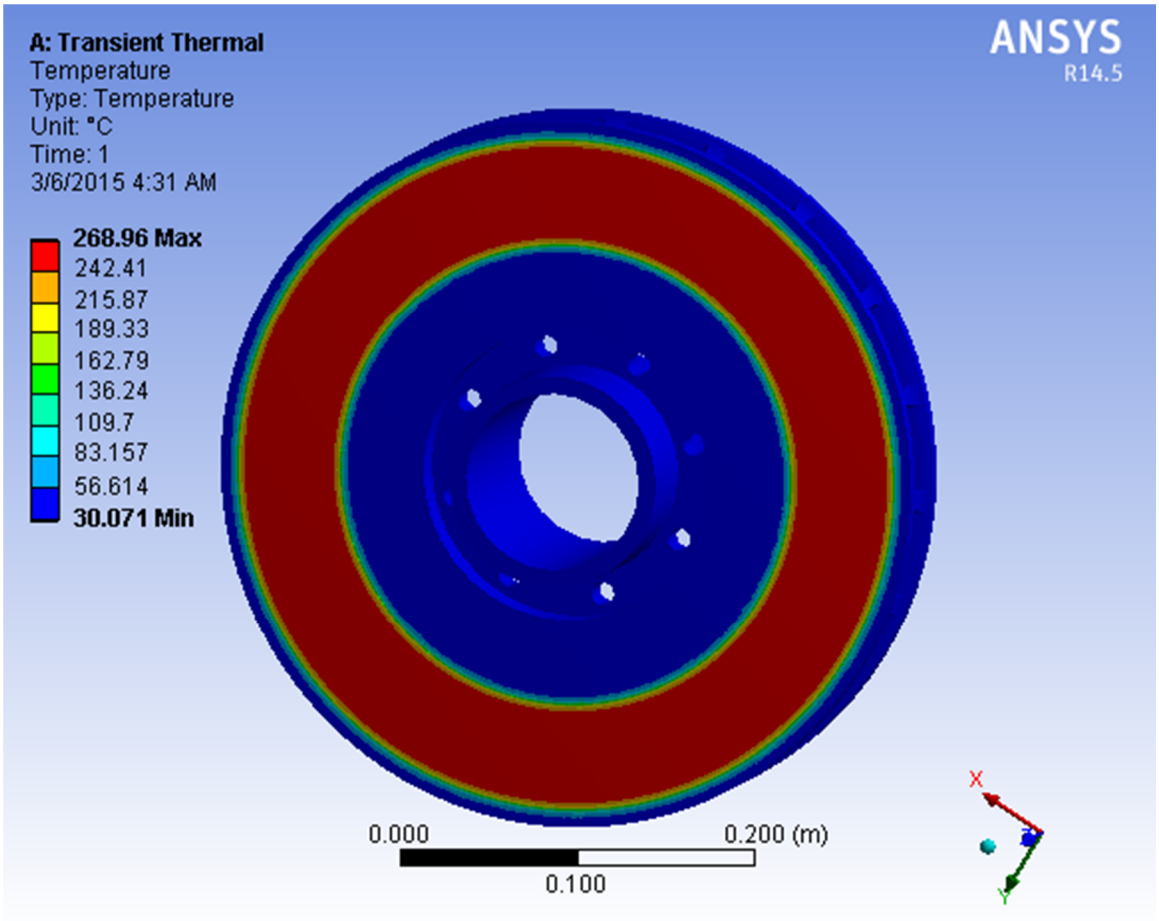


Figure 5.3 Nodal Temperatures at Ayat 1

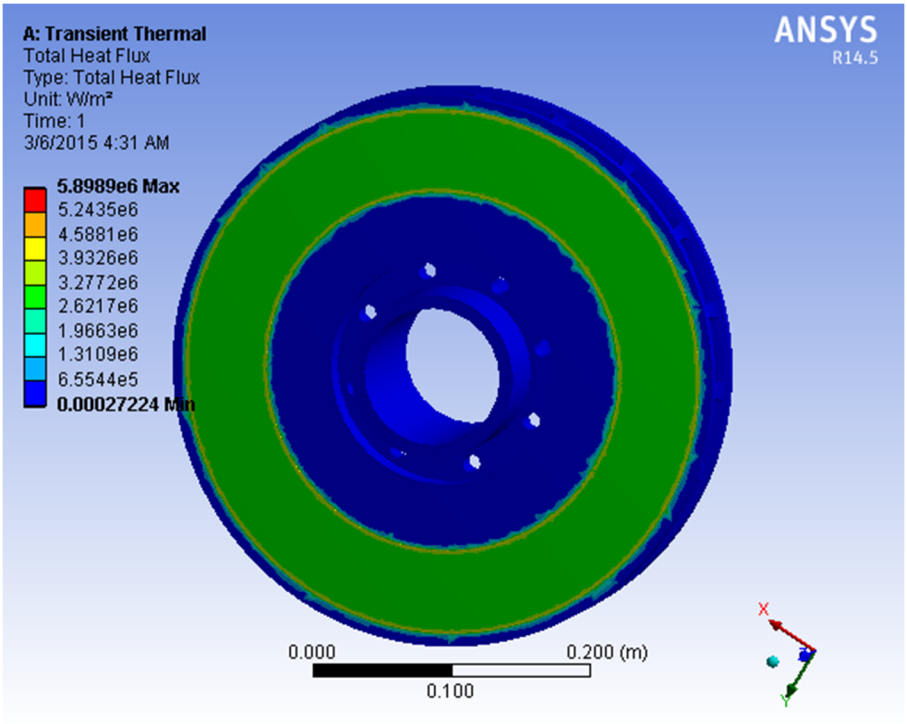


Figure 5.4 Total Heat Fluxes at Ayat 1

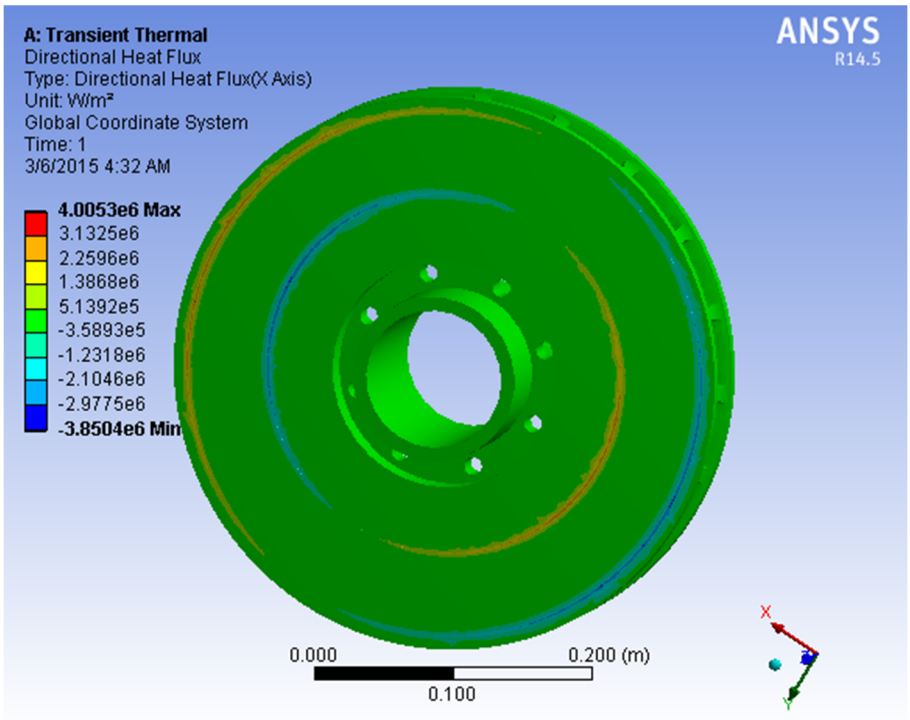


Figure 5.5 Directional Heat Fluxes at Ayat 1

5.3.2 Emergency brake for station Ayat 1

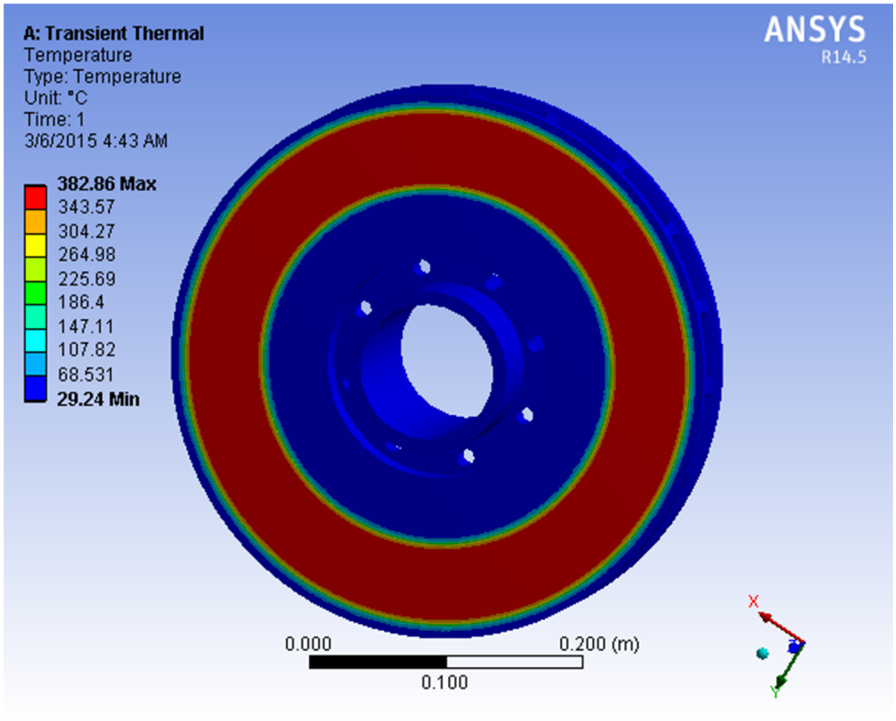


Figure 5.6 Nodal Temperatures at Ayat 1

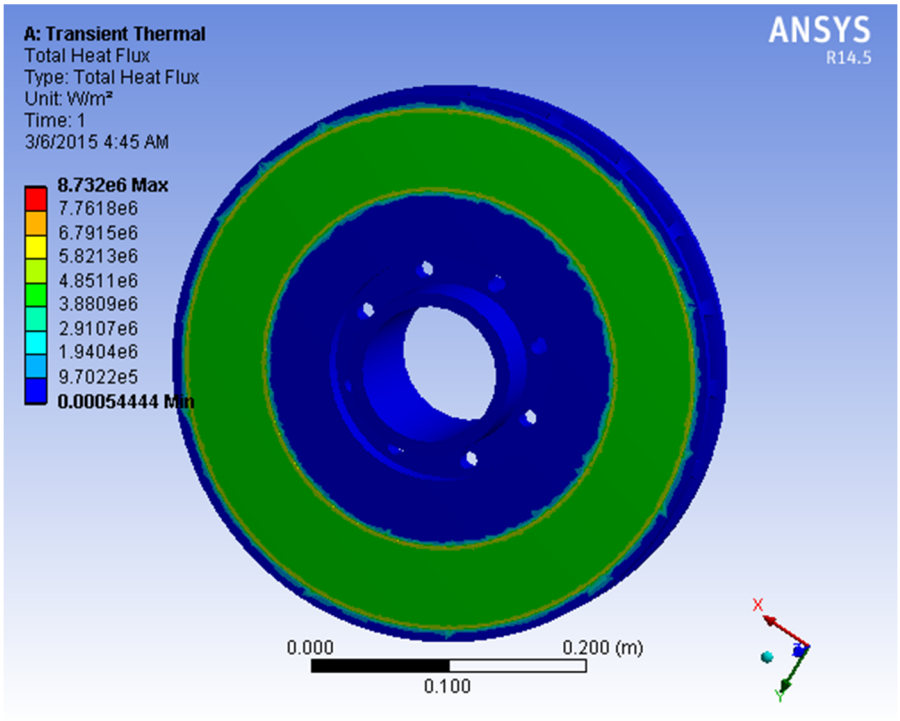


Figure 5.7 Total Heat Fluxes at Ayat 1

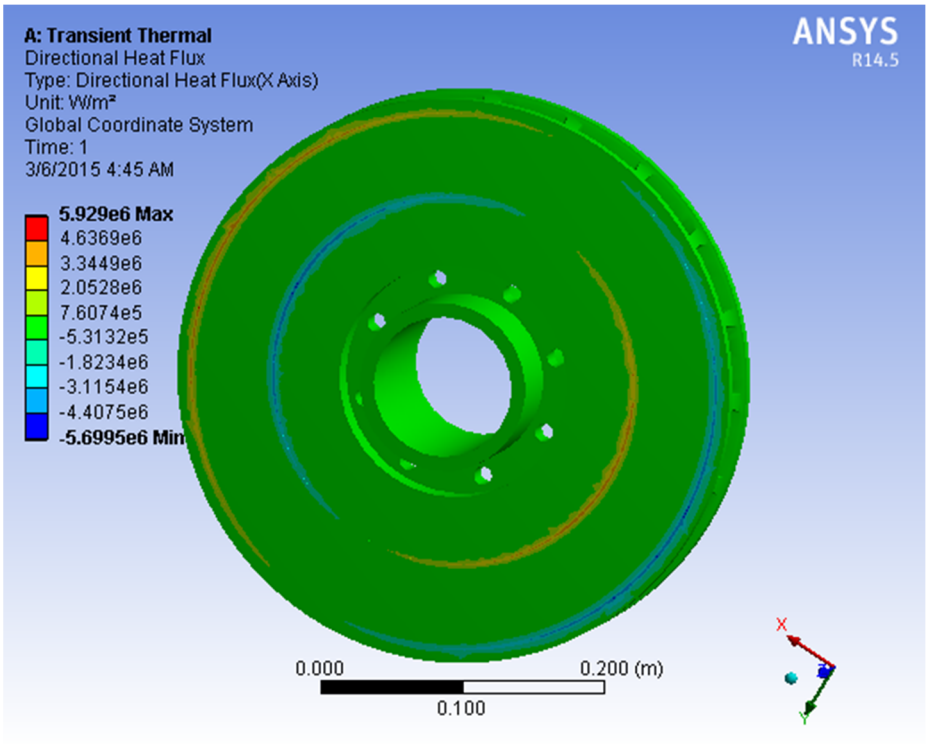


Figure 5.8 Directional Heat Fluxes at Ayat 1

5.3.3 Service brake for station Meri/CMC 2/

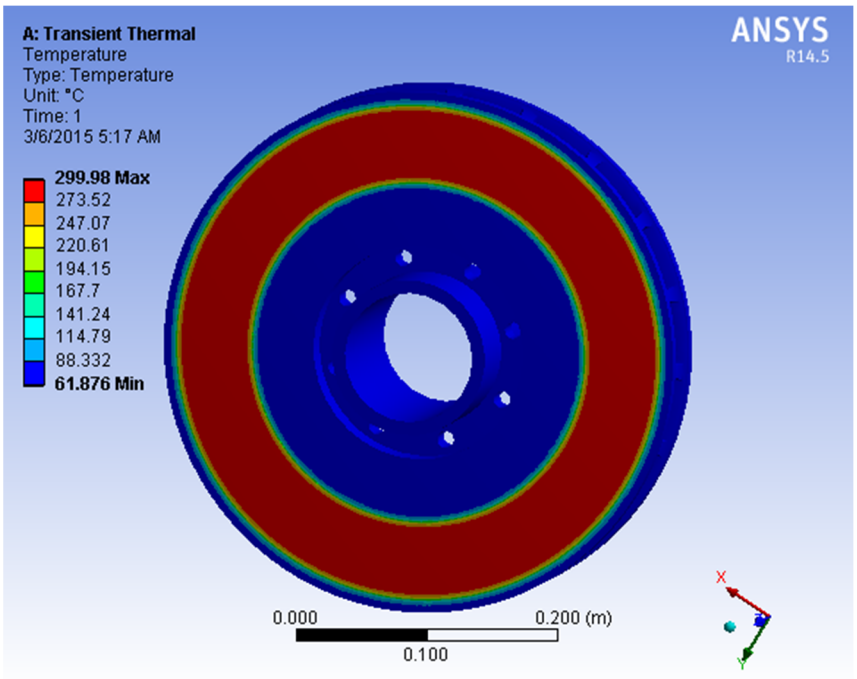


Figure 5.9 Nodal Temperatures at Meri

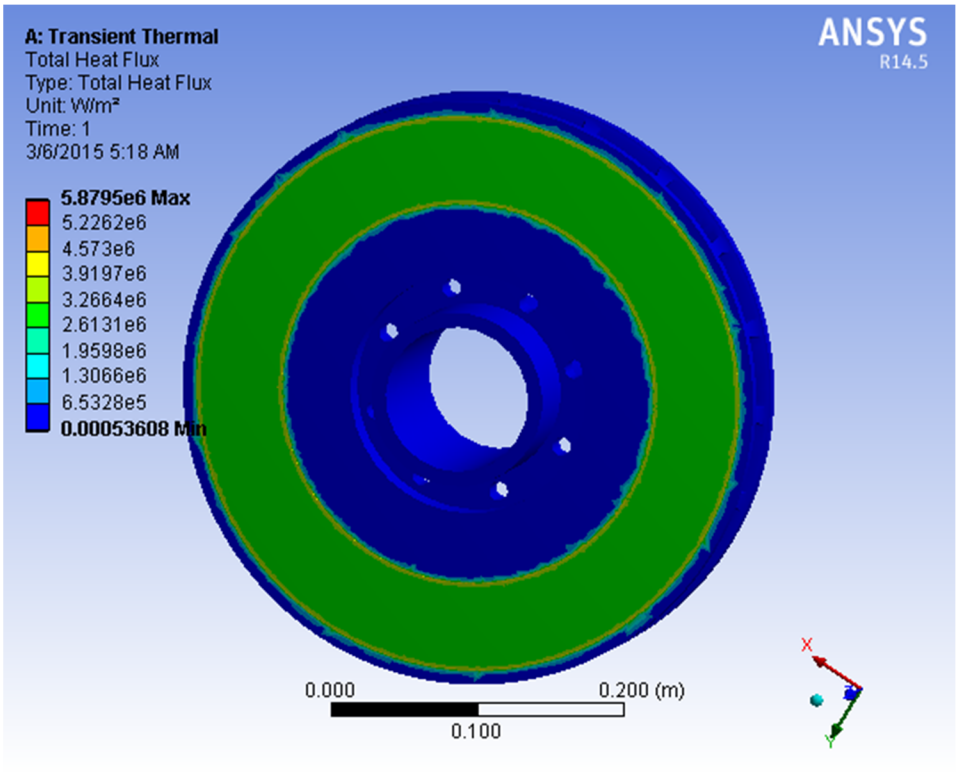


Figure 5.10 Total heat fluxes at Meri

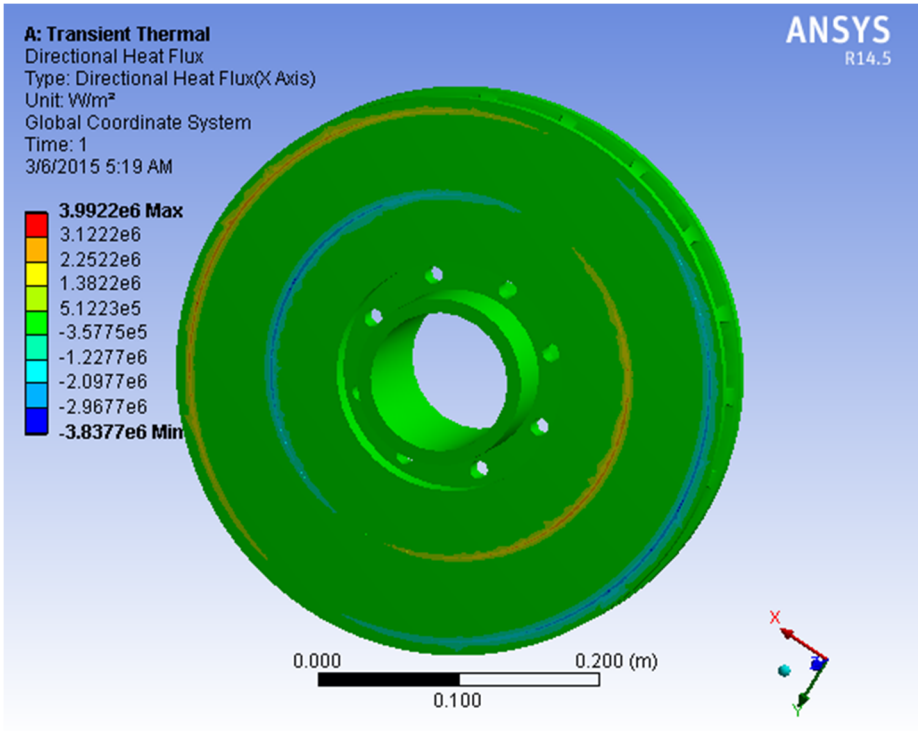


Figure 5.11 Directional heat fluxes at Meri

5.3.4 Emergency brake for Station Meri/CMC 2/

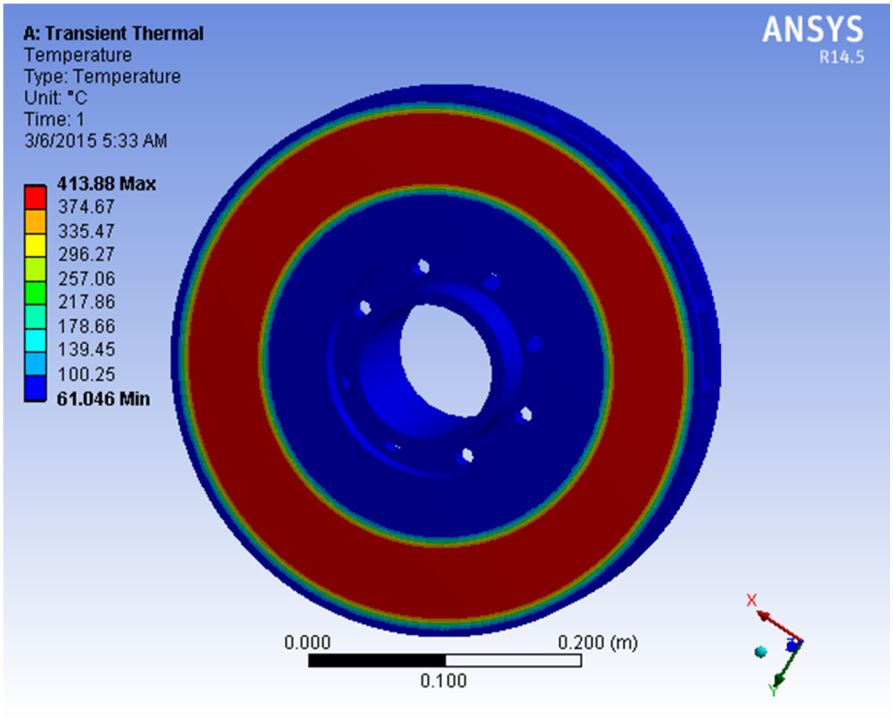


Figure 5.12 Nodal Temperatures at Meri

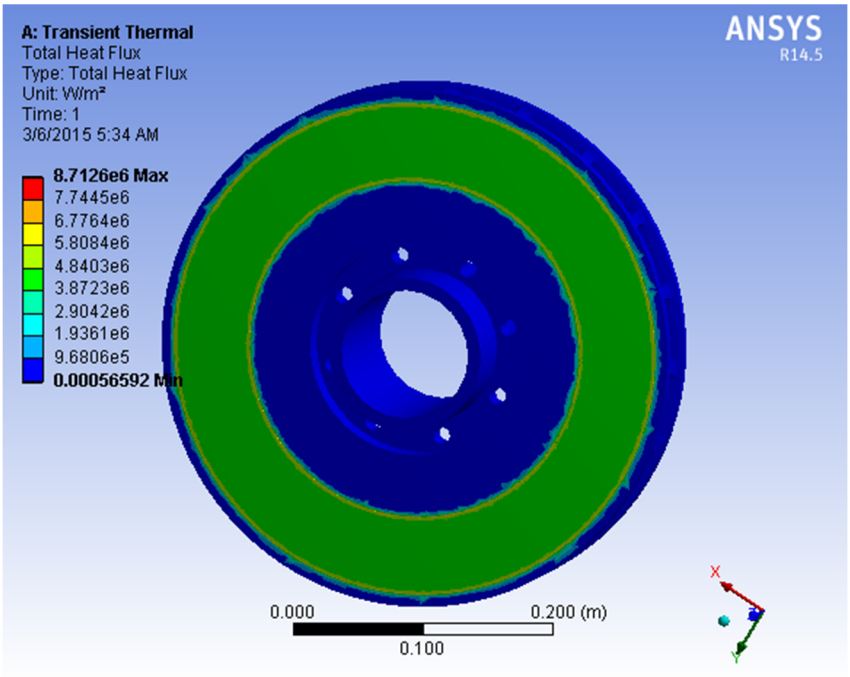


Figure 5.13 Total heat fluxes at Meri

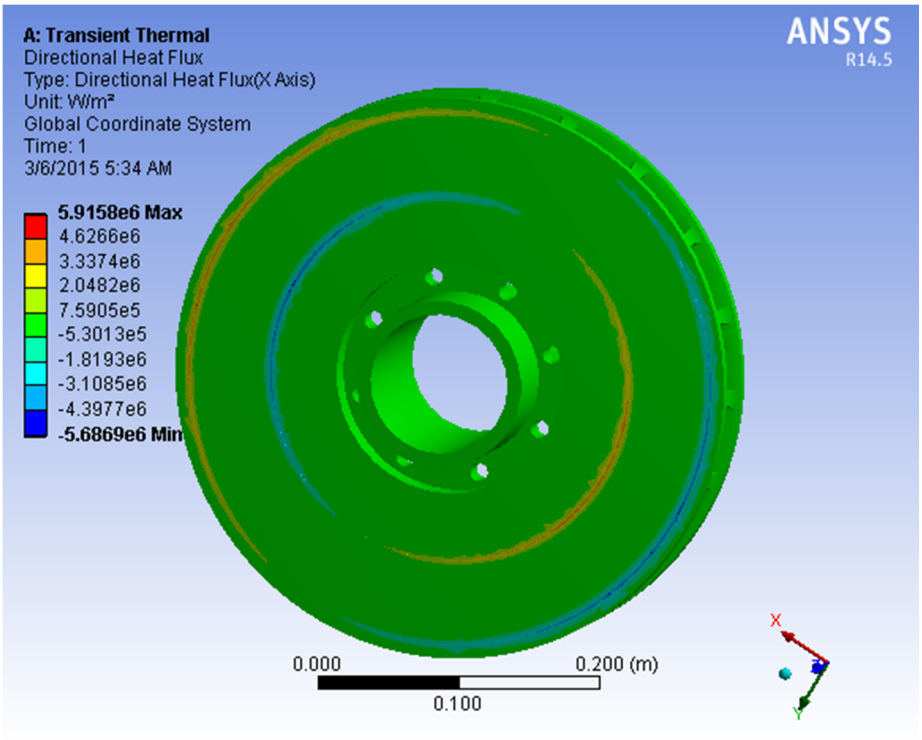


Figure 5.14 Directional heat fluxes at Meri

5.3.5 Service brake for Station CMC 1

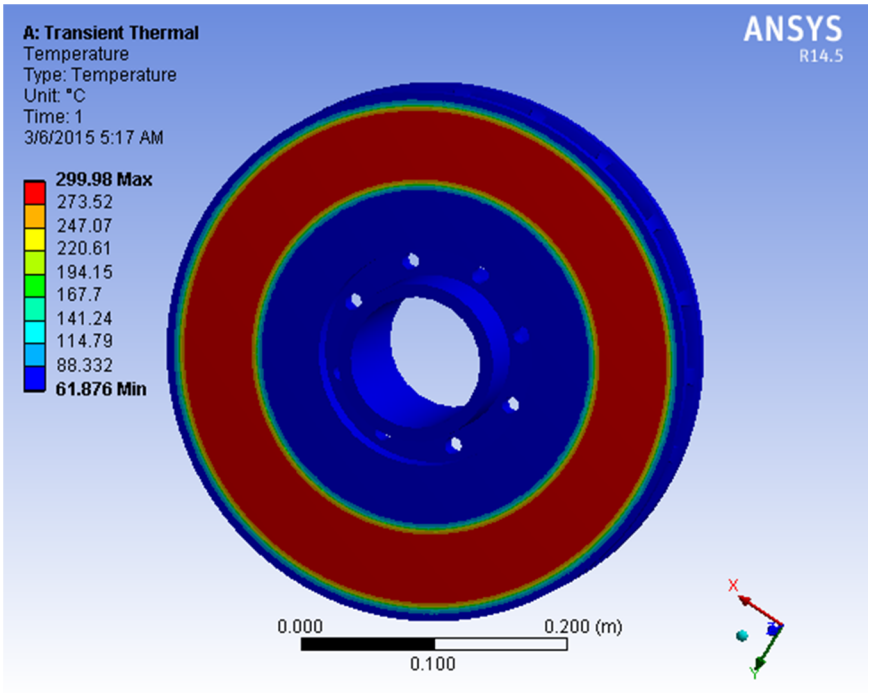


Figure 5.15 Nodal Temperatures at CMC 1

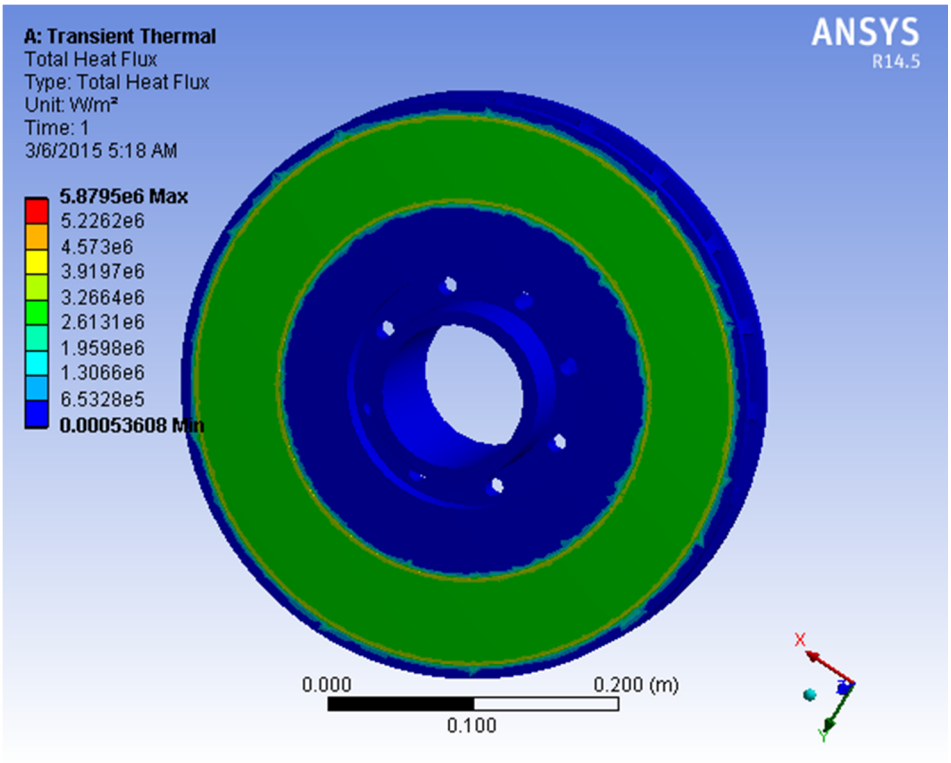


Figure 5.16 Total heat fluxes CMC 1

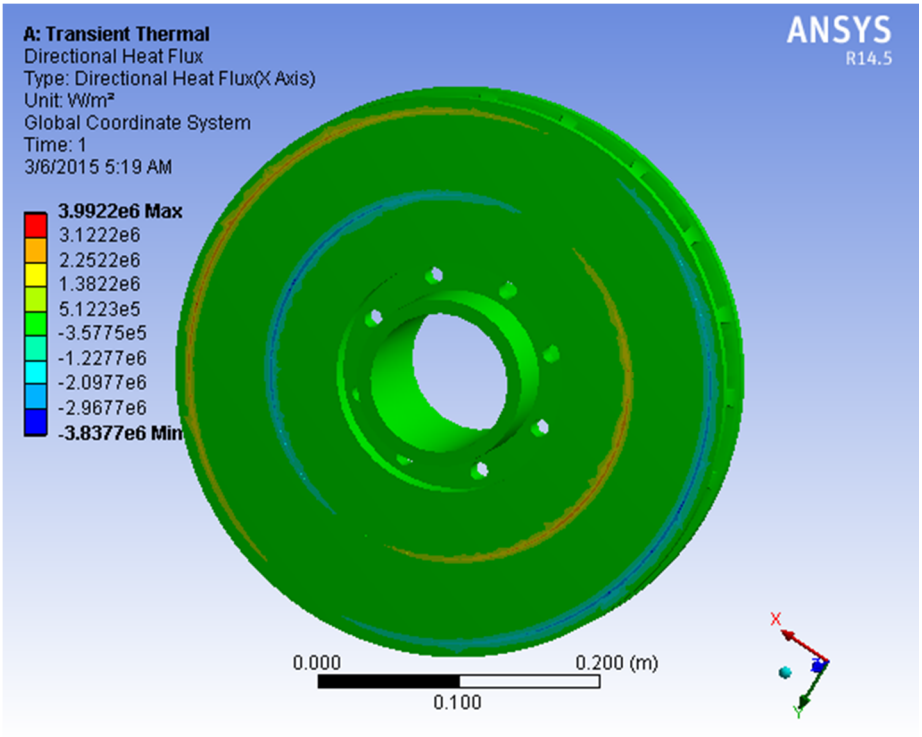


Figure 5.17 Directional heat fluxes CMC 1

5.3.6 Emergency brake for Station CMC 1

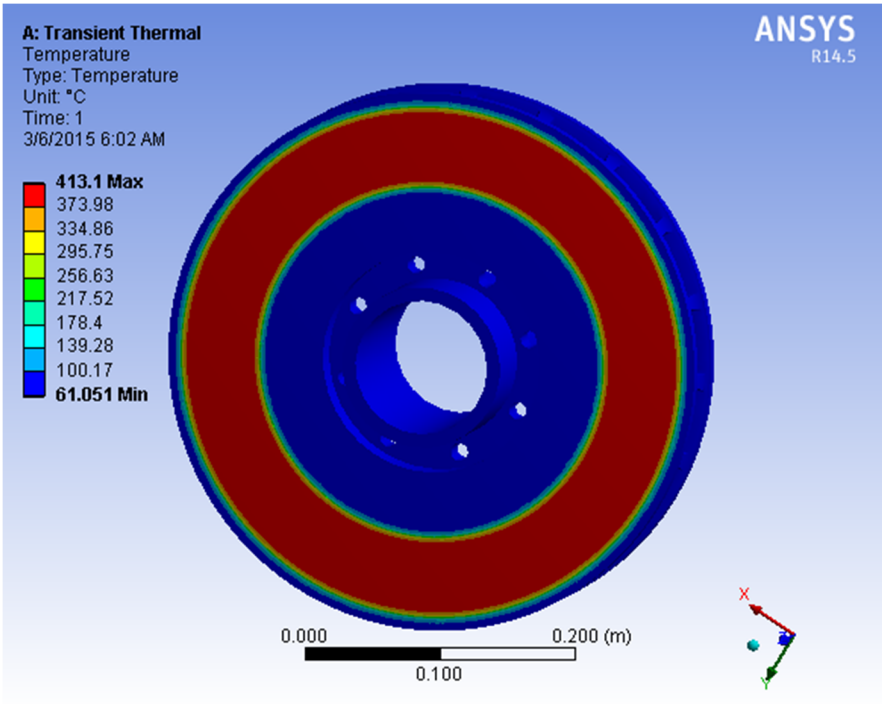


Figure 5.18 Nodal Temperatures CMC 1

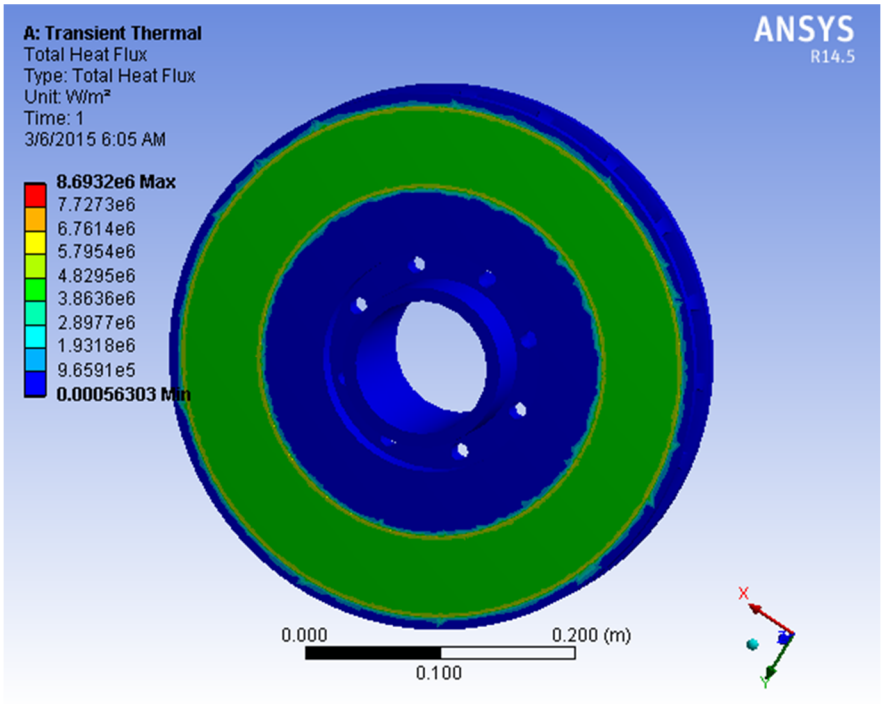


Figure 5.19 Total heat fluxes CMC 1

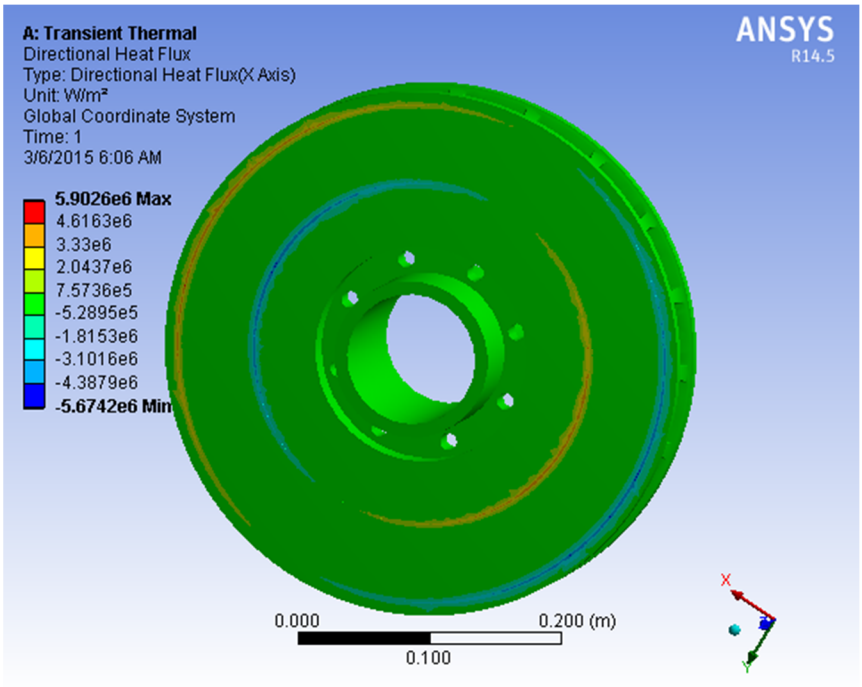


Figure 5.20 Directional heat fluxes CMC 1

5.4 Results and Discussions

The modeling of temperature in the disc brake will be carried out by taking account of the variation in a certain number of parameters such as the type of braking, the cooling mode of the disc and the choice of disc material. The brake discs are made of grey cast iron with high carbon content; the contact surface of the disc receives an entering heat flux calculated by the relation done in the literature [28]

5.4.1 Results Comparison

Table 20.Results and comparison for the three stations

Stations	Temperature[°c]		Total heat flux[w/m^2]		Directional heat flux [w/m^2]		Coefficient of friction, μ	
	Service brake	Emergency brake	Service brake	Emergency brake	Service brake	Emergency brake	Service brake	Emergency brake
Ayat 1	268.96	382.86	5.899×10^6	8.73×10^6	4.01×10^6	5.929×10^6	0.088	0.138

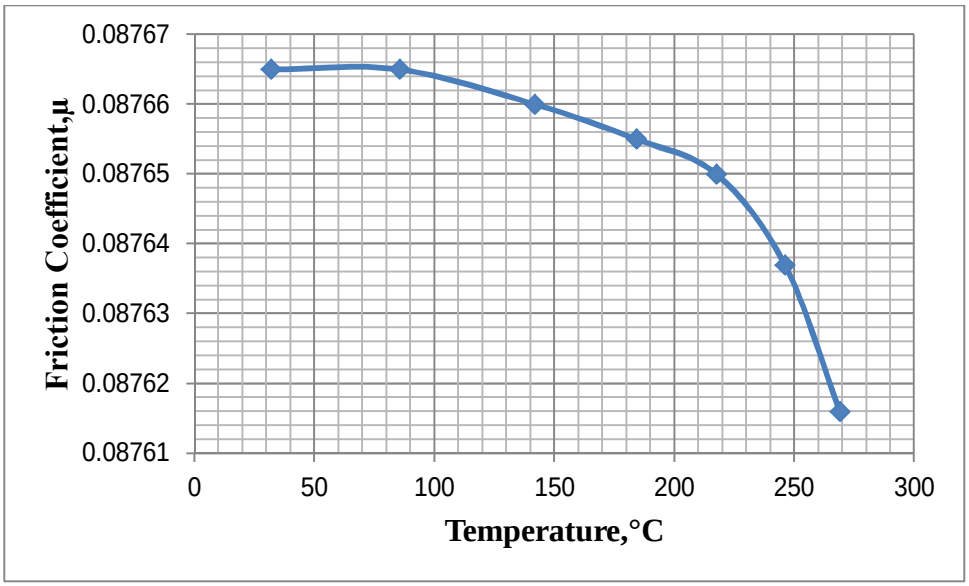
Meri	299.98	413.88	5.879×10^{6}	8.71×10^{6}	3.99×10^{6}	5.9158×10^{6}	0.086	0.136
CMC 1	299.98	413.1	5.879×10^{6}	8.69×10^{6}	3.99×10^{6}	5.903×10^{6}	0.054	0.104

The table gives the results of transient thermal analysis of Addis Ababa light rail transit disc brake at three different stations for the E-W route line. It shows that station Ayat1 gives less nodal temperature value during service brake when the brake force are applied, therefore compared to the rest stations, station at Ayat1 have higher coefficient of friction in both during service and emergency brake and therefore the performance of the disc at this station is good.

But for the case of emergency brake demanded values for stations at Meri and CMC1 are 0.136 and 0.104 respectively. Due to shorter distance and lower service and emergency brake time the nodal temperature is maximum in both conditions during service and emergency brake at CMC 2 with value of 299.98°C and 413.88°C respectively.

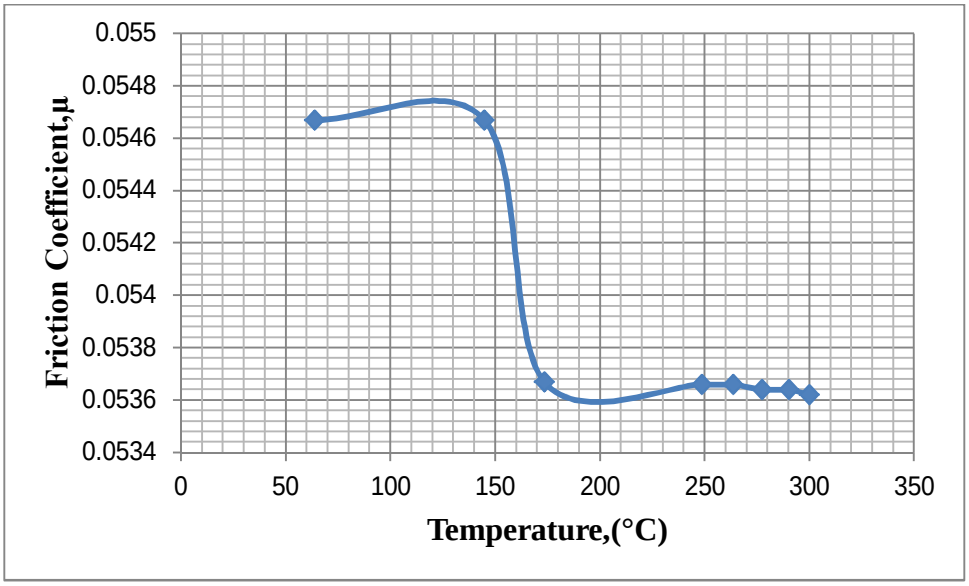
The coefficient of friction of the material is 0.2 therefore during service brake for all stations the demanded value of coefficient of friction is very small about less than 0.1 so, the performance of the disc brake is good. Compared to other stations service brake is good at station CMC1 about 73% and worse by about 56% at Ayat1.

During emergency brake all stations relatively have higher coefficient of friction than the rival's service braking conditions the material property, overall coefficient of friction 0.2. The demanded coefficient of friction during this braking action with maximum value of 0.138 at Ayat 1 and minimum value of 0.104 at CMC 1.Due to minimum braking time the braking distance diminishes that will rise the value of friction coefficient. The performance of the disc is good during emergency brake but the demanded μ is about 31% at Ayat 1 and 48% at CMC1 respectively, so to keep the performance of disc in a safer region the heat generated should be reduced. To do that majorly the kinetic energy should have to be declined. In other words the speed of the vehicle should have to be limited.



Graph 11. Temperature vs. friction coefficient during service brake at Ayat1

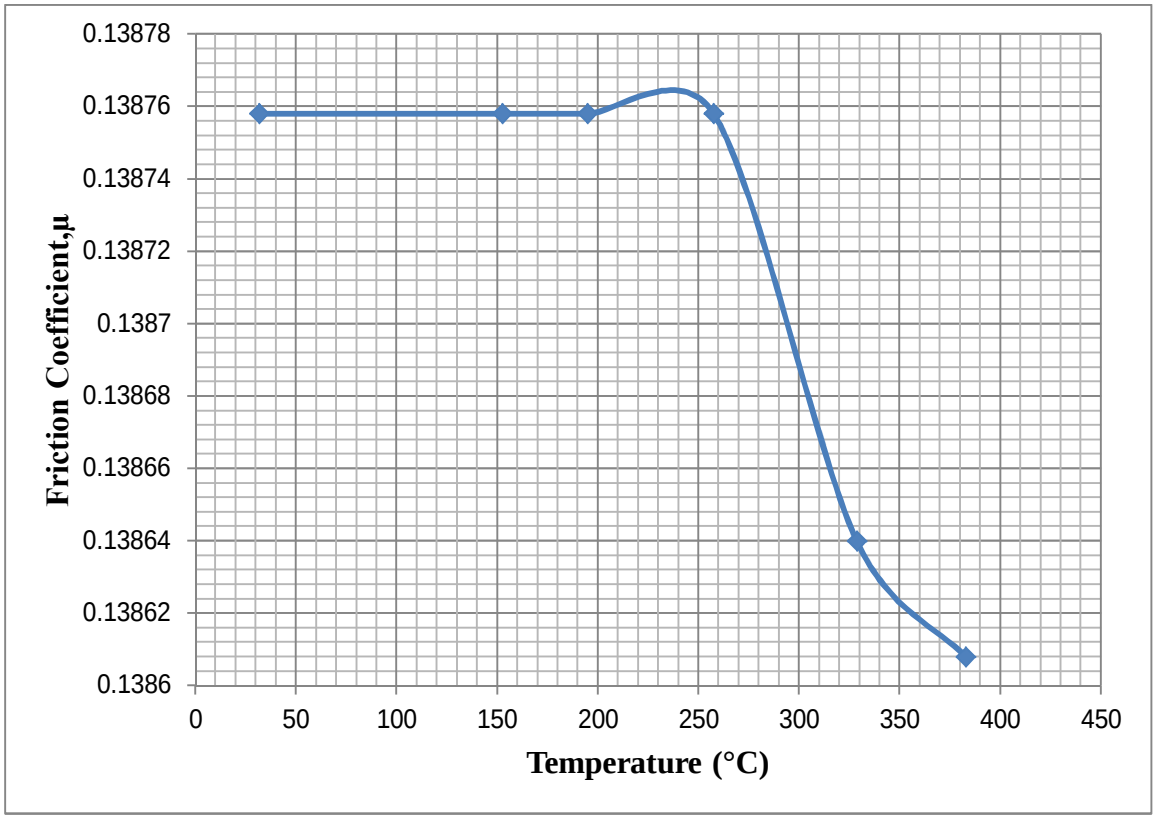
The above graph shows variation of temperature with coefficient of friction at station Ayat 1 during service braking condition. Here the initial temperature is 31.8°C at this time μ value is 0.08767 keeping with this value the temperature raises until 85.419°C after which the graph drops and finally reaches to μ value of 0.08762 when the temperature reaches 268.96 °C.



Graph 12. Temperature vs. friction coefficient during service brake at CMC1

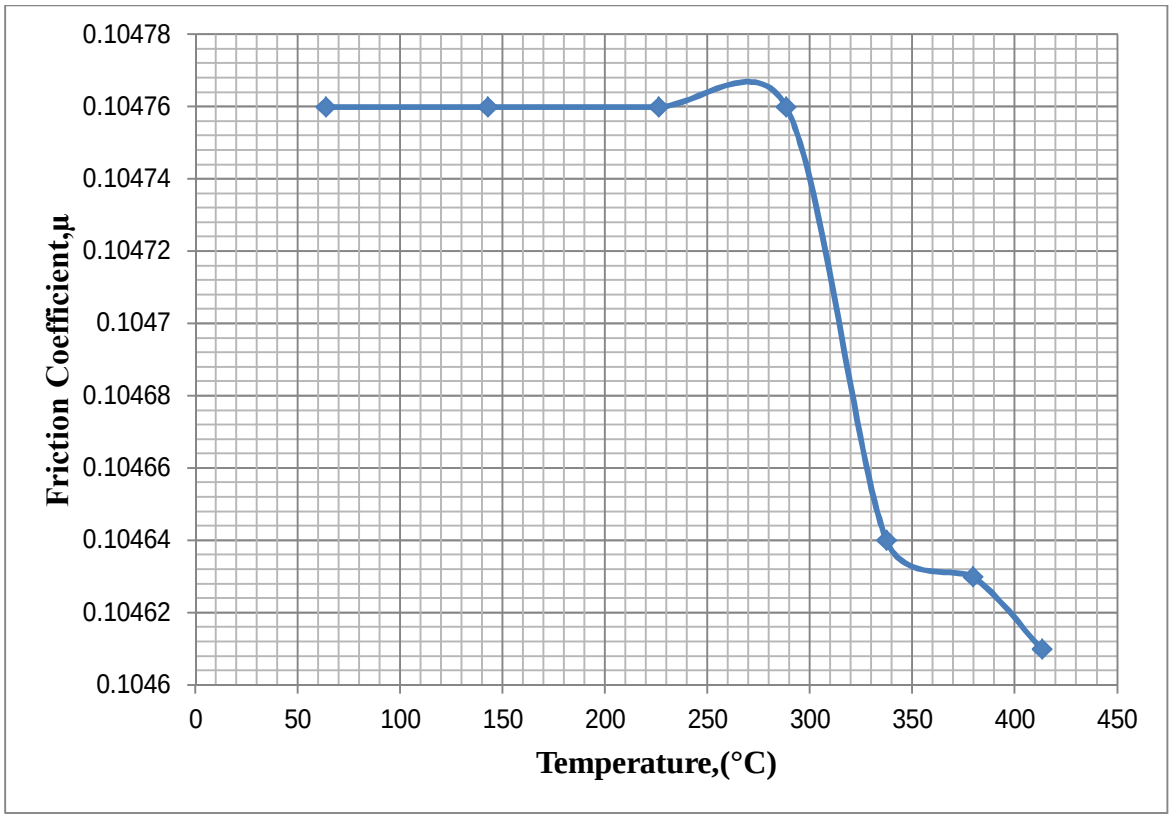
The above graph shows variation of temperature with coefficient of friction at station CMC 1 during service braking condition. Here the initial temperature is 63.6°C at this time μ value is 0.05467 keeping with this value the temperature raises until 173.35°C after which the graph changes and finally reaches to μ value of 0.05362.

The friction coefficient vs. temperature graph during service brake for station at Meri will follow similarly in the above procedure as well.



Graph 13. Temperature vs. friction coefficient during emergency brake at Ayat1

The above graph shows variation of temperature with coefficient of friction at station Ayat1 during emergency braking condition. Here the initial temperature is 31.8°C at this time μ value is 0.13876 keeping with this value the temperature raises until 257.5°C gradually the graph drops and reaches to μ value of 0.13861.

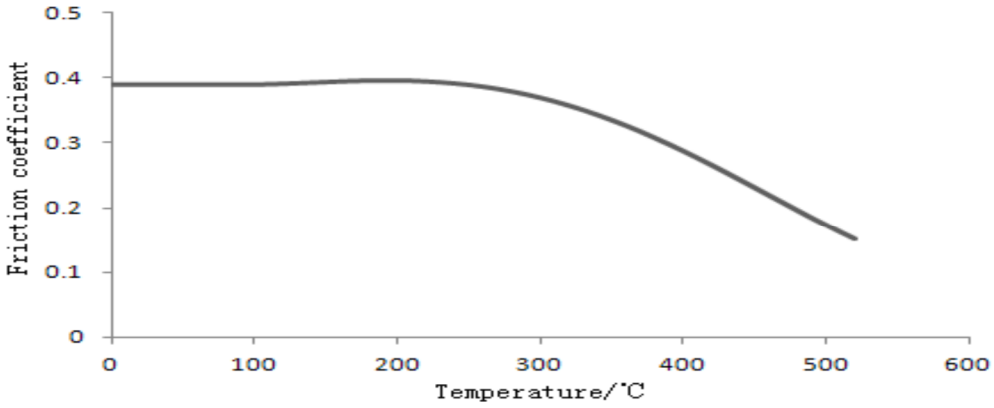


Graph 14. Temperature vs. friction coefficient during emergency brake at CMC1

Here the above graph illustrates the variation of coefficient of friction with time. The initial temperature is 63.6°C at this time μ value is 0.10476 keeping with this value the temperature raises until 288.3°C gradually the graph drops and finally reaches to μ value of 0.10461.

The friction coefficient vs. temperature graph during emergency brake for station Meri will follow in the above procedure in similar manner.

Therefore as it is shown in the graphs above as the temperature between the brake disc and pad increases the friction coefficient moves constant to some temperature value but gradually will fall down, since at the contact interface between the disc and pad the heat generated will result in reducing the performance of grapping ability.



Finally as it shown in the literature review [17] the experimental result for the relationship between coefficient of friction and temperature as the temperature between the brake disc and pad gets rises the coefficient of friction will move constant at some value but gradually the coefficient of friction will move down. The performance of the disc will eventually decline, therefore the empirically calculated value of coefficient of friction results the same output as that of experimentally calculated coefficient of friction. The graphs are also resembled.

CHAPTER-6

CONCLUSION AND RECOMMENDATION

6.1 Conclusion

The present study can provide a useful design tool to analyze and in order to improve the brake performance of disk brake system. From the above Table 20 we can say that all the values obtained from the analysis are satisfactory for braking action at station CMC1. The criteria for safe performance of brake disc is if the μ value produced at braking stage with minimum value or equal to the material friction coefficient at the interface then the performance of the disc is good with that condition. Hence the performance of brake disk is safe based on this performance analysis research criterion. Comparing the different results obtained from analysis. It is concluded that braking action at station cmc1 using disk brake is the best possible for the present application since it will demand less coefficient of friction. By observing analysis results, the performance of the disc brake is relatively highly affected during emergency brake due to the increment value of coefficient of friction; however for the case of service brake its efficiency is good for all stations. To conclude disc brake performance is good at station CMC1, Meri, and finally at station Ayat 1.

During emergency brake the highest value of deceleration 1.5 m/s^2 produces minimum braking distance for the passenger train. Because of minimum delay time the heat flux gets maximum value the generated maximum value of heat flux at the contact surface between the brake pad and rotor results less safe movement of the train as a whole compared to service brake. The μ demanded and obtained using empirical formula results relatively less coefficient of friction than the material property of grey cast iron. For all the 22 E-W stations the μ value is relatively lesser than 0.2, so the performance of disc braking system is minimum for emergency braking conditions compared to service braking action due to the increment value demand of coefficient of friction. With maximum value of demand of 0.138 at station Ayat 1 and minimum value demand of 0.104 at station CMC1.

Due to large station distance and delay time nodal temperature is minimum at station Ayat 1, but the highest value occurs at CMC 2 and CMC 1 during both braking conditions service and emergency.

During service braking condition the deceleration value is 1 m/s^2 the smallest heat flux generated at the interface and large delay time results in maximum braking distance .Therefore as the braking distance gets rises from the empirical relation as it is shown in this paper the friction coefficient reduced its value. For all the 22 E-W stations the μ value is less than 0.1, so the performance of disc braking system is good for service braking conditions in general with maximum value of 0.088 at station Ayat 1 and minimum value of 0.054 at station CMC1.

At it described in the [30] when the nodal temperature between disc and pad exceeds 800 °C, severe and sudden wear of pads occurs. Practically, the thermal transient regime determines the necessity of several brake discs mounted on the wheel set, even four in the case of high speed vehicles to minimize this maximum temperature to the expected value. However the maximum temperature at station CMC 1 which is the maximum gradient about 2.767° during emergency brake of 413.1 °C which is much less than the defined value. In general as it is shown using transient thermal analysis the performance of this Addis Ababa lrt disc brake system is good. Finally the results need to be compared with experimental results, which is also my suggestion for future work.

6.2 Recommendation

From literature review [27] shows the relation of coefficient of friction with respect to velocity of the vehicle, therefore to improve the performance of the disc during emergency brake at station Ayat 1 by 58.12% the train must run with 55km/hr, and not with 80km/hr.

Maximize the gradient by which the train travels by making the travelling distance of the train gets large enough will also improve the μ for safe braking condition of the brake disc.

Increase the brake disc and pad contact area to minimize the braking distance which will enable us to improve the performance of the drake disc by minimizing nodal surface temperature distribution.

6.3 Future works

As it is listed in the literature review making research on braking system performance as a whole have crucial advantages. The performances and parameters matches of the disc brakes directly impact on the vehicle safety. Therefore making research on these points will promote further the comfort and safety of passenger trains in general.

- Analysis the performance of disc braking system of Addis Ababa light rail transit by changing brake rotor with composite material and compare its performance with grey cast iron rotor making friction coefficient and nodal temperature as constraint.

- Performance analysis of passengers trains of Addis Ababa light rail transit by designing its braking system combination of disc and wheel tread by making coefficient of friction and temperature as well as structural deformation as a parameter and making comparison with the existing ones.

- Thermal analysis of disc brakes experimental study to validate the numerical results, for example using dynamometer test benches, in order to show a good agreement between the numerical model analysis of this paper and reality.

REFERENCES

[1] Norihisa Tadakoshi and Takao Okamoto,"Rail transport in the world's major Cities," Japan, October 2000.

[2] UNECA."Africa Review Report on Transport," United Nations Economic and Social Council 27-30, Addis Ababa, Ethiopia, October 2009.

[3] Yehualaeshet Jemere,"Addis Ababa Light Rail Transit Project'" July, 2012.

[4] Perpinya D., Reliability and Safety in Railway, barcelona,spain: In tech, march 2012.

[5] Smith, "an efficient lightweight brake system," 2002.

[6] M. R. 2. D. G. 3. V.Chengal Reddy 1, "modeling and analysis of FSAE car disc brake using fem," vol. 3, no. 9, 2013.

[7] Daniel Das.A, C. R. R. P. B, "Structural and Thermal Analysis of Disc brake in Automobiles," vol. 2, no. 3, 2013.

[8] Moynihan. S, "Rail Crises As Train Brake faile," Melbourn, 2006.

[9] Abiy Zegeye, Alemayehu Worku, Daniel Tefera, Melese Getu, Yilma Sileshi, "introduction to research methodology, "Addis Ababa University, September 2009.

[10] Ameer Fareed Basha Shaik, Ch.Lakshmi Srinivas,"structural and thermal analysis of disc brake with and without cross drilled rotor of race car," Bapatla Engineering.

[11] Design of shoe brake,IIT Kharagpur.

[12] Svante Lindberg,"Heat Distribution When Braking With railbound vehicles-A finite element analysis," May 12, 1999.

[13] Izumi Hasegawa and Seigo Uchida," Braking System,"

[14] Paden M. World report on road traffic injury prevention. Geneva: World Health Organization, 2004, 238 pp.

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

[16] Limpert R. Brake design and safety. 2nd ed. Warren dale, Pennsylvania: SAE International, 1999, 538 pp.

[17] Jiangsu, Zhenjiang, "Research of the Transient Temperature Field and Friction Properties on Disc Brakes," Zhang Jian School of Automobile and Traffic Engineering Jiangsu University, china.

[18] http:// www.thecarguys.net, July 28-2014.

[19] Pratik P. Chaphale1 Dr. S. B. Jaju M. Tech Student," A REVIEW ON THERMAL AND CONTACT STRESS ANALYSIS OF DISC BRAKING SYSTEM," Department of Mechanical Engineering, GHRCE, Nagpur 2, International Journal of Engineering Research and General Science Volume 2, Issue 1, January 2014.

[20] Cătălin Cruceanu (2012). Train Braking, Reliability and Safety in Railway, Dr. Xavier Perpinya (Ed.), ISBN: 978-953-51-0451-3, In Tech, Available from: http://www.intechopen.com/books/reliability-and-safety-inrailway/braking systems-for-railway-vehicles.

[21] 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., Vol. 212, No 4, 245-254.

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

[23] Saran Jintanon S.R.M. Nagar, Kattankulathur,"FORCE AND FRICTION ON DISC BRAKE ANALYSIS," Kanchipuram District- 603 203. June 2009.

[24] David Barney, David Haley and George Nikandros, Calculating Train Braking Distance, Signal and Operational Systems Queensland Rail (in Australia)

[25] Ptv.vic.gov.au/.../VRIOGS-009.3-Train-Dynam, June 12-2014.

[26] Poul greibe trafitec scion – DTU, July 2007, Braking distance, friction and behavior ,www.trafitec.dk

[27] http:// www.fkm.utm, July 19-2014.

[28] Guru Murthy Nathi1, T N Charyulu$_2$, K. Gowtham $_3$, P. Satish Reddy $_4$, "COUPLED STRUCTUAL / THERMAL ANALYSIS OF DISC BRAKE,"

[29] Ali Belhocine, and Mustafa Bouchetara, (JUN., 2013)," Thermo mechanical

Modeling of Disc Brake Contact Phenomena, TRANSACTION ON CONTROL AND MECHANICAL SYSTEMS," VOL. 2, NO. 6, PP. 302-309, JUN., 2013.

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

[31] A. LEE, K. AND BARBER, J. R.

[32] Ameer Fareed Basha et al.

[33] KENG LENG KHONG (2009).

[34] Šamec, B.; Oder, G; Lerher, T. & Potrč, I.

[35] Akara Sawardsuk, Saiprasit Koetniyom the sirindhorn.

[36] A. Floquet et al.

APPENDIX

Appendix A: Braking distance calculation using C# code for station at CMC1

```
using System;
using System.Collections.Generic;
using System.ComponentModel;
using System.Data;
using System.Drawing;
using System.Linq;
using System.Text;
using System.Windows.Forms;
using System.Data.OleDb;
using System.Threading;


namespace WindowsFormsApplication1
{
    public partial class Form1 : Form
    {
        validat validate =  new validat();
        double dist;
        double acc;
        double time;
        String status;

        public Form1()
        {
            Thread t = new Thread(new ThreadStart(splashStart));
            t.Start();
            Thread.Sleep(5000);
            InitializeComponent();
            t.Abort();
        }

        public void splashStart() {
            Application.Run(new SplashScreen());
        }


        private void Form1_Load(object sender, EventArgs e)
        {
            displayResult();
        }

        private void calculate_Click(object sender, EventArgs e)
        {
           if (CheckTxts())
                    return;
                else
                {
            String name = txtName.Text;
            double vf = Convert.ToDouble(txtvf.Text);
            double vi = Convert.ToDouble(txtvi.Text);
            double cos_alpha = Convert.ToDouble(txtcos_alpha.Text);
            double sine_alpha = Convert.ToDouble(txtsine_alpha.Text);
```

```
            //calculate acceleration
           if (radiobtnRising.Checked == true){
     acc =- (194658.53 + (100454.4 × cos_alpha) + (627840 × sine_alpha)) / 64000;

               status = "Rising";

            }

           if (radiobtnFalling.Checked == true)

           {

    acc = - (-34658.53 + (100454.4 × cos_alpha) + (627840 × sine_alpha)) / 64000;

               status = "Falling";


           }
          txtaccelaration.Text = Convert.ToString(acc);

          //calculate distance
           if(acc == 0){
              dist = 0;
               time = 0;
            }else{
              dist = ((vf × vf) - (vi × vi)) / (2 × acc);
              time = (vf-vi)/acc;
           }
           txtdistance.Text = Convert.ToString(dist);
           txttime.Text = Convert.ToString(time);

              /* save calculated Result*/

           try
           {
               string query = "INSERT INTO CALCULATE (INITIAL_SPEED, FINAL_SPEED
, X_COORD, Y_COORD, ACCELARATION, DISTANCE, STATUS_,NAME,TIME_) values ('" + vi +
"', '" + vf + "',  '" + cos_alpha + "' , '" + sine_alpha + "' , '" + acc + "',  '"
+ dist + "', '" + status + "', '" + name + "','" + time + "')";
               OleDbCommand cmd = new OleDbCommand(query, connection);

               if (connection.State == ConnectionState.Closed)
                   connection.Open();
               cmd.ExecuteNonQuery();
               connection.Close();

               groupBoxCalculate.Visible = true;
               MessageBox.Show(" Successfully Save", "Notification Message");
           }
           catch (Exception ex){

           }
       }
       }
       public OleDbConnection connection = new
OleDbConnection(@"Provider=Microsoft.ACE.OLEDB.12.0;Data
Source=D:\calculation.accdb");
```

```
        private void export_Click(object sender, EventArgs e)
        {

            try
            {


                // creating Excel Application
                Microsoft.Office.Interop.Excel._Application app = new
Microsoft.Office.Interop.Excel.Application();

                // creating new WorkBook within Excel application
                Microsoft.Office.Interop.Excel._Workbook workbook =
app.Workbooks.Add(Type.Missing);

                // creating new Excelsheet in workbook
                Microsoft.Office.Interop.Excel._Worksheet worksheet = null;

                // see the excel sheet behind the program
                app.Visible = true;


                // get the reference of first sheet. By default its name is
Sheet1.
                // store its reference to worksheet
                worksheet = workbook.Sheets["Sheet1"];
                worksheet = workbook.ActiveSheet;

                // changing the name of active sheet
                 worksheet.Name = "Exported from Student";

                // storing header part in Excel
                for (int i = 1; i < dataGridView1.Columns.Count + 1; i++)
                {
                    worksheet.Cells[1, i] = dataGridView1.Columns[i -
1].HeaderText;
                }

                for (int i = 0; i < dataGridView1.Rows.Count - 1; i++)
                {
                    for (int j = 0; j < dataGridView1.Columns.Count; j++)
                    {
                        worksheet.Cells[i + 2, j + 1] =
dataGridView1.Rows[i].Cells[j].Value.ToString();
                    }

                    // storing Each row and column value to excel sheet
                    for (i = 0; i < dataGridView1.Rows.Count; i++)
                    {
                        for (int j = 0; j < dataGridView1.Columns.Count; j++)
                        {
                            worksheet.Cells[i + 2, j + 1] =
dataGridView1.Rows[i].Cells[j].Value.ToString();
                        }
                    }
                }
```

```
        }
        catch
        {

        }
    }


    public bool CheckTxts()
    {
        try
        {
            if (validate.IsEmpty(txtvi))
            {
                validate.Txt_message(txtvi, "Initial Speed");
                return true;
            }
            else if (validate.IsEmpty(txtvf))
            {
                validate.Txt_message(txtvf, "Final Speed");
                return true;
            }

            else if (validate.IsEmpty(txtcos_alpha))
            {
                validate.Txt_message(txtcos_alpha, "x coordinate");
                return true;
            }
            else if (validate.IsEmpty(txtsine_alpha))
            {
                validate.Txt_message(txtsine_alpha, "y coordinate");
                return true;
            }
                //Is numeric
            else if (validate.TxtIsLetter(txtvi))
            {
                validate.Numeric_message(txtvi, "Initial Speed");
                return true;
            }
            else if (validate.TxtIsLetter(txtvf))
            {
                validate.Numeric_message(txtvf, "final Speed");
                return true;
            }
            else if (validate.TxtIsLetter(txtcos_alpha))
            {
                validate.Numeric_message(txtcos_alpha, "x coordinate");
                return true;
            }
            else if (validate.TxtIsLetter(txtsine_alpha))
            {
                validate.Numeric_message(txtsine_alpha, "y coordinate");
                return true;
            }
     if (radiobtnFalling.Checked == false && radiobtnRising.Checked == false)
            {
                 validate.Txt_message (txtvf, "Status");
                  return true;
```

```
            }
              else
              {
                  return false;
              }
          }
          catch
          {
              return true;
          }

      }

      private void button1_Click(object sender, EventArgs e)
      {
          displayResult();

      }

      private void btnReset_Click(object sender, EventArgs e)
      {
          txtaccelaration.Text = "";
          txtdistance.Text = "";
          txtvf.Text = "";
          txtvi.Text = "";
          txtcos_alpha.Text = "";
          txtsine_alpha.Text = "";
          radiobtnFalling.Checked =false;
          radiobtnRising.Checked = false;
          groupBoxCalculate.Visible = false;

      }



      public void displayResult()
      {
          this.panel1.Visible = true;
          this.btnexport.Visible = true;
          if (connection.State == ConnectionState.Closed)
              connection.Open();
          DataSet ds = new DataSet();
          DataTable dt = new DataTable();
          ds.Tables.Add(dt);
          OleDbDataAdapter da = new OleDbDataAdapter();

          da = new OleDbDataAdapter("SELECT  NAME,  ACCELARATION,
DISTANCE,TIME_, STATUS_ from CALCULATE", connection);
          da.Fill(dt);

          dataGridView1.DataSource = dt.DefaultView;

          connection.Close();

      }
```

```
        private void btndelete_Click(object sender, EventArgs e)
        {
            this.panel1.Visible = true;
            this.btnexport.Visible = true;
            if (connection.State == ConnectionState.Closed)
                connection.Open();
            DataSet ds = new DataSet();
            DataTable dt = new DataTable();
            ds.Tables.Add(dt);
            OleDbDataAdapter da = new OleDbDataAdapter();

            da = new OleDbDataAdapter("DELETE * from CALCULATE", connection);
            da.Fill(dt);
            dataGridView1.DataSource = dt.DefaultView;
            connection.Close();

            displayResult();
        }

    }
    }
```

The above written code will work for any stations by changing the accelerated and decelerated equation of motion.