ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

GRADUATE PROGRAM IN RAILWAY ENGINEERING

ANALYSIS OF EFFECT OF TRAIN OVERLOAD ON DISC BRAKE OF AALRT

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By

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APPROVED BY BOARD OF EXAMINERS

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DECLARATION

I hereby declare that the work which is being presented in this thesis entitled "Analysis of Effect of Train Overload on Disc Brake of AALRT" is original work of my own, has not been presented for a degree of any other university and all the resource of materials used for this thesis been duly acknowledged.

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

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ABSTRACT

Changes in the train dynamics characteristics such as train weight will affect the train's braking performance and its ability to stop safely in braking situation. This paper studies about the disc temperature and von misses stress of train over loading used in AALRT.

A transient thermal analysis has been carried out to investigate the temperature variation across the disc using ANSYS 14.5. Further structural analysis is also carried out by coupling with thermal analysis.

A more realistic braking disc model for AALRT under various over load percentages, road grade level and maximum travel speed of 70km/h is analyzed. The model of disc brake is done by CATIA software and then simulation was followed using ANSYS 14.5 workbench. Results from this research showed that the temperature and stress on disc brake during service brake at downhill train position and unacceptable load condition has increased by 37.08 ^{o}c and 56 Mpa from acceptable load and the same train position. When a repetitive braking is applied the temperature and von misses stress during train overload has increased by 37.07 ^{o}c and 56 Mpa respectively.

Similarly, when the emergency brake is applied during normal and unacceptable load condition, the temperature and stress on the disc has increased by 41.4 ^{o}c and 51 Mpa in time of downhill track position.

In general, overloading the train has the effect of increasing the temperature and stress induced in to the disc brake rotor during braking.

Keywords: - Disc brake, Temperature, Overloading, Von Mises stress and Light rail transit.

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NOTATIONS

AALRT	Addis Ababa light rail transit
AC	Alternating current
A_d	Disc surface swept by a brake pad
AB	Air Brake
GB	China National Standard
TB	China Railway Industry Standard
DIN	German Industrial Standard
EN	European Standard
ISO	International Organization for Standardization
UIC	International Union of Railways
IEC	International Electro Technical Commission
EMC	Electromagnetic Compatibility
RAMS	Reliability, Availability, Maintainability and Safety
ED	Electric Brake
EHM	Hydraulic Brake of Motor Car
EHT	Hydraulic Brake of Motor Car
MTB	Magnetic Brake
FEA	Finite Element Analysis
FEM	Finite Element Method
PPHPD	Passengers per hour per direction
LRT	Light rail transit
GVW	Gross vehicle weight

CHAPTER ONE

INTRODUCTION

1.1 Background

1.1.1 History of rail transport

Rail transport including systems with man or horse power, and tracks or guides made of stone or wood, the history of rail transport dates to as early as Greek times. Wagon ways were relatively common in Europe (typically in mining) from about 1500 through 1800. Mechanized rail transport systems first appeared in England in the 1820s. [1]

These systems, which made use of the steam locomotive, were critical to the Industrial revolution and to the development of export economies across the world. They have remained the primary form of land transport ever since for most of the world. [2]

1.1.2 Rail transport

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

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

The oldest, man-hauled railways date back to the 6th century B.C, with periander, one of the Seven Sages of Greece, credited with its invention. Rail transport blossomed after the British development of the steam locomotive as a viable source of the power in the 18th and 19th centuries. With steam engines, it was possible to construct mainline railways, which were a key component of the industrial revolution. Also, railways reduced the costs of shipping. Studies have shown that the invention and development of the railway in Europe was one of the most important technological inventions of the late 19th century for the United States.

In the 1880s, electrified trains were introduced, and also the first tramways and rapid transit systems came into being. Starting during the 1940s, the non-electrified railways in most countries had their steam locomotive replaced by diesel-electric locomotives, with the process being almost complete by 2000. During the 1960s, electrified high-speed railway system was introduced in Japan and a few other countries. [From Wikipedia, the free encyclopedia]

1.1.3 Electrification and dieselization

Experiments with electrical railways were started by Robert Davidson in 1838. He completed a battery-powered carriage capable of 6.4 km/h (4 mph). The Gross-Lichterfelde Tramway was the first to use electricity fed to the trains en route, when it opened in 1881. Overhead wires were taken into use in the Modeling and Hinterbruhl Tram in Austria in October 1883. At first, this was taken into use on tramways that, until then, had been horse-drawn horse cars. The first conventional completely electrified railway mainline was the 106 km Valtyellina line in Italy that was opened on 4 September 1902.

During the 1890s, many large cities, such as London, Paris and New York City used the new technology to build rapid transit for urban commuting. In smaller cities, tramways became common and were often the only mode of public transport until the introduction of buses in the 1920s. In North America, interurban became a common mode to reach suburban areas. At first, all electric railways used direct current but, in 1904, the Stubaital Line in Austria opened with alternating current. [3]

Following the large-scale construction of motorways after the war, rail transport became less popular for commuting and air transport started taking large market shares from long-haul passenger trains. Most tramways were either replaced by rapid transit or buses, while high transshipment costs caused short-haul freight trains to become uncompetitive. The 1973 oil crisis led to a change of mind set and most tram systems that had survived into the 1970s remain today. At the same time, containerization allowed freight trains to become more competitive and participate in intermodal freight transport. With the 1964 introduction of the Shinkansen high-speed rail in Japan, trains could again have a dominant position on intercity travel. During the 1970s, the introduction of automated rapid transit systems allowed cheaper operation. The 1990s saw an increased focus on accessibility and low-floor trains. Many tramways have been upgraded to light rail and many cities that closed their old tramways have reopened new light railway systems. [4]

The electrification system provides electrical energy to the trains, so they can operate without a prime mover on board. This allows lower operating costs, but requires large capital investments along the lines. Mainline and tram systems normally have overhead wires, which hang from poles along the line. Grade-separated rapid transit sometimes uses a ground third rail.

Power may be fed as direct or alternating current. The most common DC voltages are 600 and 750 V for tram and rapid transit systems, and 1,500 and 3,000 V for mainlines. The two dominant AC systems are 15kv AC and 25kv AC.

1.1.4 Passenger trains

Passenger trains are part of public transport and often make up the stem of the service, with buses feeding to stations. Passenger trains provide long-distance intercity travel, daily commuter trips, or local urban transit services. They even include a diversity of trains, operating speeds, right-of-way requirements, and service frequency. Passenger trains usually can be divided into two operations: intercity railway and intercity transit. Whereas as intercity railway involve higher speeds, longer routes, and lower frequency (usually scheduled), intercity transit involves lower speeds, shorter routes, and higher frequency (especially during peak hours).

Magnetic levitation trains such as the Shanghai airport train use under-riding magnets which attract themselves upward towards the underside of a guide way and this line has achieved somewhat higher peak speeds in day-to-day operation than conventional high-speed railways, although only over short distances. Due to their heightened speeds, route alignments for high-speed rail tend to be steeper grades and broader curves compared to conventional railways. [5]

High-speed rail are special inter-city trains that operate at much higher speeds than conventional railways, the limit being regarded at 200 to 320 kilometers per hour (120 to 200 mph). High-speed trains are used mostly for long-haul service and most systems are in Western Europe and East Asia. The speed record is 574.8 km/h (357.2 mph). [6], [7]

1.1.5 Rail transport in Ethiopia

Rail transport in Ethiopia consists only of trial services from Djibouti to Dire Dawa on a reconstructed line since September 6, 2013. [8] The rail line continues from Dire Dawa to Addis Ababa, the capital, but is no longer operational. A new rail network is planned.

The Imperial Railway Company of Ethiopia was a firm founded in 1894 to build and operate a railway across eastern Ethiopia from the port of Djibouti to the capital of Addis Ababa. It was founded by Alfred Ilg and Léon Chefneux and headquartered in Paris, France [9]. The firm failed in 1906 when political discord halted construction, and it failed to obtain any new capital. The portion it had completed ran from Djibouti to just short of Harar [10], the principal entre pot for existing commerce in southern Ethiopia. [11]

A group of British investors calling themselves the New Africa Company effectively took control of the firm over several years. They provided a new source of capital and by 1901 had joined with the French investors to form the International Ethiopian Railway Trust and Construction Company a holding firm which essentially controlled the railway and supplied it with further capital. The first commercial service began in July 1901, from Djibouti to Dire Dawa.

This mixture of French and British interests proved volatile, as each group of investors stood increasingly for both national and commercial interests. Both governments became interested in monopolizing Ethiopian trade and conspired to force the other into a minor position. The demands and threats of the two governments led Emperor Menelek in 1902 to forbid the expansion of the railway line to Harar. French negotiations to resume work were blocked by Menelek's growing suspicion of French motives, and the line could not earn enough to pay back the company's debts with such a limited service. The signature of the Entente Cordiale in 1904 reopened the possibility of continued joint Anglo-French investment and development, but there was enough resistance to such proposals on both sides that no progress was made. The firm went formally bankrupt in 1906.

Continued construction

Following the 1906 Tripartite Treaty between Italy, France, and Britain and the 1908 Klobukowski Treaty between France and Ethiopia, Menelek consented to further expansion of the railway, granting the new concession to his personal physician, a black Guadaloupean named Dr. Vitalien, on 30 January 1908.[12] The assets of the former company were then transferred to a new firm, the Franco-Ethiopian Railway, which received a new concession to finish the line to Addis Ababa. After a year of wrangling with the previous financiers and their governments, construction began anew. By 1915 the line reached Akaki, only 23 kilometers from the capital, and two years later came all the way to Addis Ababa itself.

Figure 1.1: Present station of the Djibouti-Ethiopia Railway in Dire Dawa.

Ethiopia's share in the railway was seized by the Italian government in the Second Italo-Abyssinian War, but was regained by Ethiopia after the World War II. Following the independence of Djibouti in 1977, the French share in the railway was transferred to the new nation.

Post-World War II

Workers for the Railway were pioneers in the Ethiopian labor movement. They organized one of the first labor unions in Ethiopia in 1947, the Railroad Workers Syndicate of Dire Dawa, for mutual welfare purposes [13]. Although its leadership co-operated with the Government, an attempted strike in 1949 was brutally suppressed by government troops; at the time, all strikes were seen by government officials as a form of insurrection. [14]

The railway company carried out surveys for extending its line 310 kilometers south from Adama to Dilla between 1960 and 1963. The government formed a Nazareth-Dilla Railway Development Corporation to support this new branch. Although the French government offered a loan to fund this new branch in 1965, and Yugoslav experts had studied and thought the project would be worthwhile, this project was never carried out. [15]

Following the 1977 independence of Djibouti, France transferred its ownership rights to the government of the new country. Around 1982, the railway was subsequently reorganized as the Ethio-Djibouti Railways.

In 2003 the European Commission prepared a grant of EUR 40 million for the rehabilitation of the Djibouti-Ethiopia Railway. Reflecting increases in fuel and steel prices, the commission increased this grant to EUR 50m in 2006. The Djiboutian Ministry of Equipment and Transport and the Ethiopian Ministry of Transportation and Communications chose the South African firm Comazar to administer the railway in March 2006. A 25-year concession was due to be signed in June 2007, but this failed to happen, and negotiations began with Kuwaiti company Fouad Alghanim and Sons Group. [16] The new management was expected to raise the capacity of the railroad from its current average of 240,000 tons to 1.5 million tons. [17]

The governments of Djibouti and Ethiopia signed an agreement with the Italian consortium Consta 29 November 2006; work will begin early 2007 on sections of the line that deteriorated following the Ogaden War. [18] The gauge of the line is 1,000 mm (3 ft 3 $^{3}/_{8}$ in).

New network

In January 2010, it was announced that the Ethiopian government had signed a memorandum of understanding with four companies to build a new rail network. Among these firms was the China Communications Construction, China Railway Group, and an Indian and Russian company. [19]

After all parties except the Chinese firms failed to take further action; the contract was awarded to the CRCG. In August, funding for the project was agreed to come from the Export and Import Bank of China. In September 2010, construction began on the project.

The network will be 5,000 km long, and radiate from Addis Ababa. It will be constructed in two phases, the first phase involving the construction of five lines. [20] The project will provide 300,000 jobs in construction and cost $336 million annually for five years. The system will handle 6 million tons of freight. The government of India has given US$300million for the Mekelle-Djibouti rail line. [21]

1.1.6 LRT

The abbreviation LRT stands for Light Rail Train and this is used to describe a light weight metropolitan electric railway system characterized by its ability to operate single car or short trains along exclusive right of way at street level the train usually powered by overhead electric wires. Light Rail Transit (LRT) also called tram or trolley, the system provides comfortable, environmental friendly, fast and reliable local public transiting service. LRT provides a wide range of passenger transit capacity by a moderate cost. Electrically propelled rail trains operate singly or in trains. [22, 23]

1.1.7 AALRT

Addis Ababa Light Rail transit is a light rail system under construction in Addis Ababa, Ethiopia. It is being contracted by the China Railway Group Limited. The Ethiopian Railway Corporation began construction of the 34.25 kilo meter (21.3 mi) double track electrified light rail transit project in December 2011 after securing funds from the Export-Import Bank of China. Initially, the light rail system will have two lines.

Figure 1.2: on progress AALRT rail project Mexico area.

Of the two rail lines, the east-west line will extend 17.35 kilometers (10.8 mi), stretching from Ayat Village to Tor Hailoch, and passing through Megenagna, Legehar and Mexico Square. The north-south line, which will be 16.9 kilometers (10.5 mi) in length, will pass through Menelik Square, Merkato, Lideta, Legehar, Meskel Square, Gotera and Kaliti. Additionally, two lines will have a common track of about 2.8 km. [24]

The Addis Ababa Light Rail will have a total of 41 stations on its two lines, and each train will have a capacity to carry 286 passengers. This will enable the light rail transit to provide a transportation service to 15,000 passengers per hour per direction (PPHPD) and 60,000 in all four directions. [25]

1.1.8 Train braking

Train braking is a very complex process, specific to rail trains and 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. [26] The actions of these processes take place in various points of the trains 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 action.

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.[27]

The main goal in modeling and analyzing of train brake is to provide the necessary braking capacity appropriate to the type and specific running speed of the train according to the traffic safety. Braking capacity is a significant feature of any railway 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. [28]

The braking capacity of a railway train depend on numerous factors and some of the most important are train running speed, train weight, type of brake, constrictive and functional characteristics of the brake rigging, brake characteristics, thermal phenomena, etc. [29] so, it is important for those responsible for the maintenance and operation of railway infrastructure to monitor and prevent train overloading. Otherwise a direct and consistent assessment of braking capacity is difficult to achieve with train overload.

The addition weight carried by overloaded train accelerates the deterioration of the road way, leading to rutting, fatigue crack, and in certain cases structural failure (Sharma, 1995; CSIR, 1997; Bushman et al., 2003; Santero et al., 2005). Straus and Semmens (2006) conducted a study to quantify state highway damage on the basis of the impact of overweight trains. Each year, millions of dollar of damage associated with life span, design and maintenance of state highways and structures are attributed to train that exceed state weight limit they found that for every dollar invested in motor carrier enforcement efforts, there would be $4.50 in pavement damage avoided. Three performance measures were used to assess and characterize train performance, namely, dynamic stability, braking and handling downhill.

Without modification of engines and driver trains, increase truck weight would lead to greater speed reduction on upgrades and greater difficulty for trucks to merge, weave and change lanes on freeways. Other things being equal, increased gross weight may also increase the probability of brake overheating on long step downhill runs. Any one of these situations can have adverse traffic (delays and congestion) and accident implications (Transportation Research Board, 1990“Estimating the Cost of Overweight Vehicle Travel on Arizona Highways”).

Train overloading can be of two types:

1 – More people than the car can accommodate and

2 – Dead weight in cars weight.

If the mass of a train are not controlled, overloading decreases the efficiency of train braking performance. Consequently, overloaded trains become a traffic hazard, especially regarding the train braking system and additional braking distance involved. This situation is aggravated by steep downhill slopes and sharp curves. [30]

1.2 Problem Statement

Impact of overload rail traffic includes economic, social and environmental losses. Many countries are confronted with this problem. Overloaded rail traffic induces extreme harm to the economy of an entire country. Thus economy impact always is the major concern for Government.

The major impact induced by overloaded rail is unexpected overheat on the brake thread or disc. Because brake design is based on normal traffic load and expected heat that generated in time of braking. As a result, the braking capacity is in sufficient to retard or to stop the train.

Most of the time accident occurs by lowering the effect of overloading the train on the safety side of the transport. If the train is overloaded it has the effect to decrease the braking performance of the train. If the braking performance of the train decreases the train expose to a danger condition and its impact includes social and economic losses for the country.

1.3 Objective

1.3.1 General objective

The main objective of this research is, analyzing the effect of train overloading on the thermo-mechanical condition of the disc by taking AALRT as a case study.

1.3.2 Specific objective

The specific objective of this research is analyzing the train overloading effect on the brake disc with recommended and assumed overloading percentage on horizontal and downhill train position of Addis Ababa LRT using simulation software Ansys 14.5, the research analyze the thermal and mechanical stress of the brake disc by applying the recommended and assumed loading of the train and interpret the result and give detail information about the thermal and structural effect of train overloading on disc brake rotor.

1.4 Significant of the research

If train dimensions, number of axles, and other aspects of the train and component designs were unchanged, Substantial increase in gross train weight would lower rollover resistance in steady turns for all rails, which may lead to more rollover accidents. Increased weight would also downgrade the rearward amplification behaviors, which may increase the probabilities of rear-trailer overturns during obstacle avoidance or sudden lane change maneuvers.

This research has a benefit to Ethiopia railway corporation operation managements are clearly understand and give attention for the overloading problem and implement it in the new Ethiopia railway transport to prevent economic and consequential other damage of the company and make it profitable and a very safe transportation system.

1.5 Scope of the research

The scope of this research is analyzing the effect of train overloading on train disc brake by finite element method using ANSYS work bench transit thermal and structural analysis which shows the temperature distribution and von Mises (equivalent) stress of the disc brake rotor.

1.6 Limitation of the research

1. As per this research proposal have the aim of analyzing the overloaded effect of rail train on its braking performance on AALRT and give a scientific evidence for Ethiopia Railway Corporation managements and operation staff members but the corporation has no real recorded data on previous percentage of the train overload because it is new transportation system for the country. So the research depends on the other transport organization sample in the country (Addis Ababa Anbesa city bus).

2. The required data’s are not available in primary data’s form and good justification in the other transport system of the country.

1.7 Research methodology

The research method for analyzing of effect of train overload on brake disc has consider the complexity and dynamics associated with the problem that has been dealt in this research, FEM is consider to be an appropriate tool for modeling the system structure.

- Analyze the problem and data collection.
- Modeling and simulation are done by applying different overloads percentage and train braking from maximum operational speed to stop and interprets the simulating result.
- Discussing the results of the analysis.
- Make conclusion and recommendation.

1.7.1 Analyze the problem and data collection

This is the most important for modeling and simulating process. In this phase, the problem is identified, defined and clearly stated. Showing and clearly defining the problem by collecting and using data's which shows the problem of overloading the train from different sources (journals, articles, researches and interview with some relative transport company's (Anbesa city bus)).

1.7.2 Modeling, applying different overloads and simulating

Modeling, applying different overload percentage and simulating are done based on the data collected and calculated values, there are three main steps involved. Which are:-

1.7.2.1 Modeling

Virtual 3D model of disc brake rotor with pad are modeled by CATIA software and transfer to finite element software which is ANSYS. Thermal analysis will be done by assigning material properties, meshing of the model and apply load will be done in ANSYS software.

1.7.2.2 Simulate the model with its input parameter

Simulation was carried out under acceptable and unacceptable train load condition and maintaining maximum forward velocity, 70km/h before the brakes are applied.

1.7.2.3 Data generation and interpretation

Model that can reflect an actual AALRT disc brake situation is important to determining more realistic effect. Thus, in this research, the disc and pad model and its specification for train have been developed in accordance to train class available on the new Ethiopian railway.

1.7.3 Evaluation, Recommendation and Conclusion

Evaluation of generated data depends on the analysis and simulation on unacceptable overload results are compared with the acceptable load condition. This step helps to conclude the effect of overloading on the train disc brake. Recommendation for Ethiopian Railway Corporation will be provided based on the results of the analysis.

1.8 Organization of the Research

The research is organized in six chapters. The first chapter being the introduction part which clearly states the background of the research (history of rail transport, electrification and dieselization, passenger trains, rail transport in Ethiopia, LRT and AALRT), problem statement, objective, significant, scope, limitation of the research and research methodology.

Second chapter, literate review of the braking system and disc brake system of rail train definition, parts and types are discussed in this chapter, with literature review of various literatures on thermal and mechanical analysis of disc brake.

Chapter three shows the data collection, unacceptable load generate for the input of the analysis, and interpretation the data collection result.

Chapter four has physical and analytical calculation of energy of the train braking which is dissipating in to the surrounding, initial heat flux and mechanical force or pressure on the rotor in time of braking are modeled.

The FEM of the disc brake physical model starting from modeling the disc, material property, boundary conditions and all inputs for ANSYS are applied.

Chapter five contains the results and discussion of the analysis.

In Chapter six conclusion, recommendation and future works are forwarded.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

The analysis of thermal effects has become increasingly important in product design in different transport means, road trains, airplanes, railway trains, and so forth. The thermal analysis is a very important stage in the study of braking systems, especially of railway trains. The thermal load of a braked railway disc brake rotor prevails compared to other types of loads. In the braking phase, kinetic energy transforms into thermal energy resulting in heating and high temperature states of disc rotor. Thus induced thermal loads determine thermo mechanical behavior of the structure of disc brake rotor. In cases of thermal overloads, which is mainly occur as a result of long-term braking on down-grade railroads and emergency braking. Train overloading has its effect on the generation of high heat on the brake rotor and pads, these consequences are the appearance of initiate cracks on the rotor, decrease the rotor braking properties which decreases the train’s braking performance.

2.2 Braking

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 a fixed point. This action 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.[31]

The rather low locomotives power and traction force allowed braking using quite simple handbrakes that equipped locomotives and eventually other trains 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 trains 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 trains, 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. [32] Therefore, it was generalized and remains even nowadays the basic and compulsory system for rail trains. It is still to notice that, regarding the classical systems used for railway trains, there are also several major challenges that may affect the braking capacity. These aspects must be very well known and understood, so as to find appropriate solutions in such a manner that the problems to be overcome by applying different constructive, functional, operational and other kinds of measures Reliability and Safety in Railway.

2.2.1 Brake Principle

Of all the systems that make a train, the brake system might just be the most important. Its function is to retard or stop the motion of the train which determine the safety of the driver, passenger and also pedestrian. The average driver uses the brakes about 75,000 times a year, making the brakes one of the most important (and overworked) parts of the car. [33]

In the olden days it was also one of the simplest mechanisms in the train. Over the years as improvements have been made, the system that has evolved is not simple anymore. When the brakes are applied, the pads or shoes that press against the brake drum or rotor convert kinetic energy into thermal energy via friction. The cooling of the brakes dissipates the heat and the train slows down. This is all to do with the First Law of Thermodynamics, sometimes known as the law of conservation of energy.[34]

This law states that energy cannot be created nor destroyed; it can only be converted from one form of energy to another. In the case of brakes, it is converted from kinetic energy to thermal energy. Typically, there are two types of brake that are implemented in today's car; Drum brake and disc brake. To stop the wheel, friction material in the form of brake pads or shoe is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc or drum which cause the wheel to slow or stop.[35]

When the brakes are used rapidly, the complete brake assembly will stay hot and get no chance to cool off. The brake cannot absorb much more heat because the brake components are already so hot. The braking efficiency is reduced thereby brake failure could result in road accident.

2.2.2 Details of Train Brake Operation

In train braking operation each power unit (locomotive) has an air compressor that supplies air for the entire trains braking system. A feed valve in the locomotive regulates the desire pressure that is supplied to reminder the train. This pressure must be at least 482.6 Kpa (although most modern system uses 620.5Kpa). A "brake pipe" runs the full length of the train. The pipe carries the compressed air from the control unit to the rest of the train. Unlikely truck brakes (and passenger trains brakes for that matter) this single source of air carries both the air that powers the brakes as well as the single to control them.

The brake component includes the brake cylinder, brake shoes, a dual reservoir, and a control or AB valve. The AB valve is used to route air from the reservoirs (auxiliary and emergency) to the brake cylinder. The brake cylinder is connected through rods, levers and slack adjusters to the brake shoes. While these components are similar to track brakes their operation is very different. Unlike track brakes, train brakes are normally off, or unapplied. The spring in the brake chamber is used as a return device to pull the brake shoe away from the wheel and allow the wheel to roll freely. So, in order to apply the brakes, air must be ported from the reservoir to the brake chamber. There are several ways the engineer can apply braking to the train. According to the nature of stop braking system is divided in to two.

2.2.2.1 Service Brake

This is the type of brake application normally used for braking. This level of braking is achieved with a 41.37kpa to a 179.3kpa reduction in the brake pipe pressure. When the AB valve senses the difference in pressure air is ported from the reservoir to the brake chamber. Air pressure acts against the diaphragm and brakes are applied. Braking the service brake offers up to 75% of a train's Emergency Brake capability.

2.2.2.2 Emergency Brake

This is all the brake capability that a train has. It is utilized, as implied, when there is an emergency. Application of Emergency opens the brake pipe to atmosphere on all cars and units sequentially from front to rear. As a result, the AB Valves ports pressure from the Emergency reservoir to the brake chamber and all brakes slow the train. This type of brake use applies the brakes as fast as possible.

2.2.3 Braking Performance

The braking performance of a train is consistent controllable braking forces on the entire traffic speed domain, permitting speed reductions; stop at fixed point and to maintain the train standstill on slopes, usually being also automated. [36]

The braking performance of a train depends up on many factors such as braking material property of friction and adhesion, loading condition of the train, type of braking system etc. The friction between brake pads and disc leads to severe thermal regimes and special thermal fatigue nature efforts, requiring specific constructive and operating standards. More than that, due to the air compressibility and to the length of trains, the pneumatic commands propagates with limited speed in the brake pipe and, as a result, there is always a delay in the braking of neighboring trains [37].

As a consequence, the rear trains are running into the front ones, producing large dynamic longitudinal reactions in buffers and couplers. The induced compression and tensile forces can reach significant levels affect the braking performance even conducting to deteriorations of safety operation of the trains [38].

Railway high speed operations also determined more severe requirements for braking systems, given to the necessity for develop higher braking forces and to dissipate larger amounts of energy in a short time, not to mention the problem of wear in the case friction brakes. In that case, complementary systems whose performance and reliability are safety relevant were developed to enhance the braking performance.

Brakes have been returned and improved ever since their invention. The increases in traveling speed as well as the growing weight of trains have made these improvements essential. The faster a rail car goes and the heavier it is, the harder to stop. An effective braking system is needed to accomplish this task. Regarding the construction of the braking systems components, especially those mounted under the train chassis or on the bogie, they must have low mass and such size and shapes to fit the rolling stock gauge. The mobile mechanical element of the brake installation and mainly the brake riggings have two function correctly and optimally transmit forces regarding of the fact that the train empty or loaded at maximum capacity and as long as the wear of all constrictive elements of the train are within the limit allowed by regulations. It is also very important that the thermal regime developed during braking to remain within acceptable limits, without affecting the braking capacity [39].

2.3 Disc brake

A disc brake is a wheel brake that slows rotation of the wheel by the friction caused by pushing brake pads against a brake disc with a set of calipers. The development and use of disc type brakes began in England in the 1890s. The first caliper-type automobile disc brake was patented by Frederick William Lanchester in his Birmingham, UK factory in 1902 and used successfully on Lanchester cars. Compared to drum brakes, disc brakes offer better stopping performance, because the disc is more readily cooled. As a consequence disc brakes are less having a tendency to brake fade, and recover more quickly from immersion (wet brakes are less effective).

Reliable caliper-type disc brakes were developed in the UK by Dunlop and first appeared in 1953 on the Jaguar C-Type racing car. 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" caused when brake components overheat; and disc brakes recover more quickly from immersion.[40]

For railway trains, the braking technology get fast pace worldwide, even the main brake method is electricity or eddy current brake but frictional brake is a redundancy backup and reliable way to stop the trains. Most form of frictional brakes on the railway locomotives is disc brake type mounted on wheels or axels to dissipate brake energy.

Disc brakes are widely used because it is simple, powerful, easy maintenance and allow new materials of disc and pad to be brought into use to provide effective friction and minimize the wear. The premature wear and thermal cracking of brake discs are attributed to high thermal stress during brake process, more heat need to be distributed to disc rotor and brake pad friction pair, which cause higher thermal stress.[41]

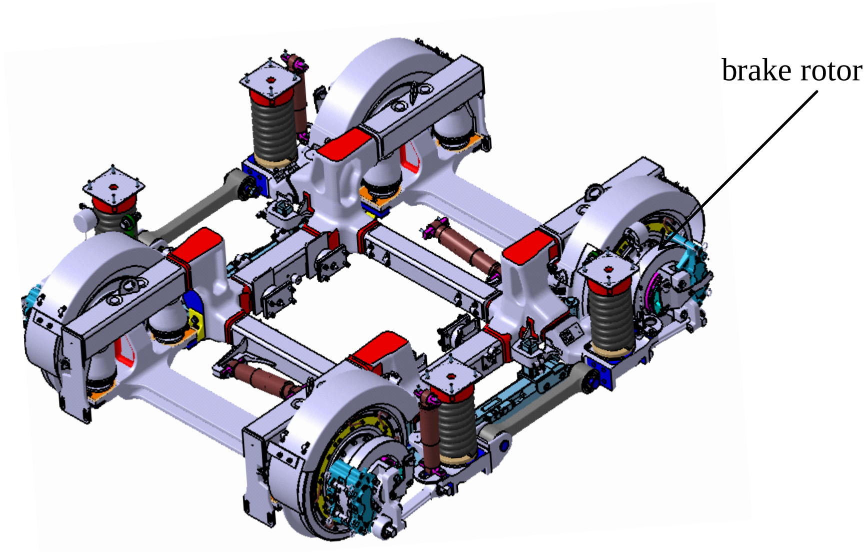


Figure 2.1: wheel mounted disc brakes of unpowered bogie.

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 break fade [42].

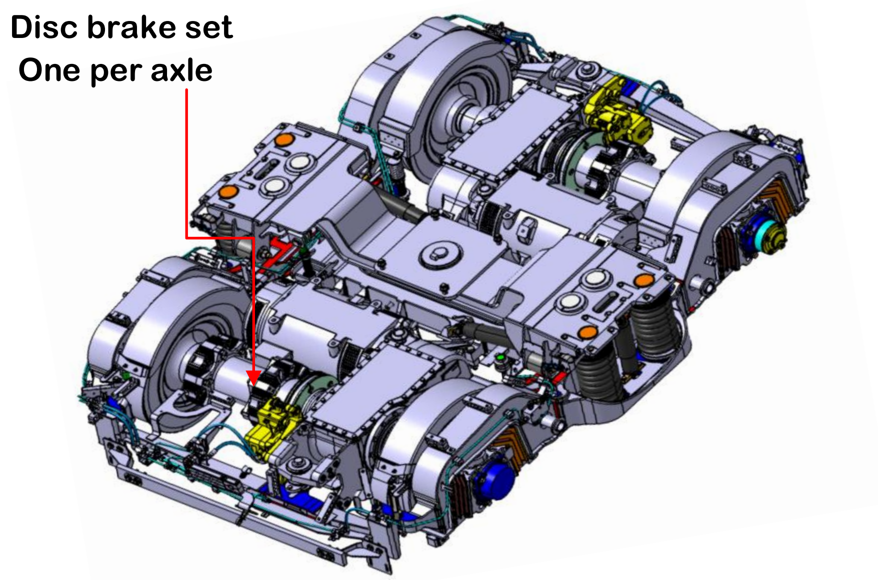


Figure 2.2: axle mounted disc brakes of powered bogie.

2.3.1 Working principle of disc brake

The principle used is the applied force (pressure) acts on the brake pads, which comes in to contact with the moving disc. At this point of time due to friction the relative motion is constrained.
When the brakes are applied, hydraulically actuated pistons move the friction pads in to contact with the disc, applying equal and opposite on the later. On releasing the brakes the rubber-sealing rings acts as return spring and retracts the pistons and the friction pads away from the disc. [43]

2.3.2 Types of brake disc

Each time we stop on the brakes, the friction causes the temperature of the brake system parts are spike up. This heat speeds up wearing and tearing of the brake parts aside from that, abrupt stopping can also wear out brake part really fast. One part that takes brunt of the abuse is the brake disc. This day’s two major types of brake discs are used in most cars.

2.3.2.1 Drilled brake disc

Drilled brake disc have holes that make harmful element such as water gas and heat easy to remove from the discs surface.

2.3.2.2 Slotted brake disc

Slotted brake disc have slots engraved on the discs flat surface. These slots like the hole in drilled brake discs help remove unwanted elements from the disc surface. The most commonly used brake discs nowadays are the drilled type, because these are tougher and they do not wear out easy. But we are in to competitive racing, using slotted brake disc in the best choice. Slotted disc are the most resilient to breakage, a common racing brakes problem.

2.3.3 Disc Brake Components

There are many different designs of disc brake but all contain the following basic parts: disc rotor, pads and caliper.

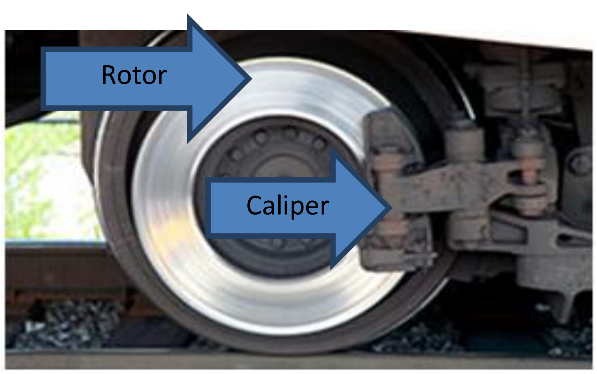


Figure 2.3: wheel mounted train disc brake setup.

2.3.3.1 Caliper

The brake caliper is the assembly which houses the brake pads and pistons. The pistons are usually made of plastic, aluminum or chrome-plated steel.

Calipers are of two types, floating or fixed. A fixed caliper does not move relative to the disc and is thus less tolerant of disc imperfections. It uses one or more single or pairs of opposing pistons to clamp from each side of the disc, and is more complex and expensive than a floating caliper.

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; a piston on one side of the disc pushes the inner brake pad until it makes contact with the braking surface, then pulls the caliper body with the outer brake pad so pressure is applied to both sides of the disc.

2.3.3.2 Disc Brake Rotor

Rotor and its type is described in 2.3.2

2.3.3.2 Brake pads

Different brake design applications require different kinds of friction materials. Several considerations are weighted in development of brake pads; the coefficient of friction must remain constant over a wide range of temperatures, the brake pads must not wear out rapidly nor should they wear the disc rotors, should withstand the highest temperatures without fading and it should be able to do all this without any noise.[44]

Figure 2.4: typical pads used for disk brake systems. [45**]**

2.4 Train overload

Exceeding a train’s maximum permissible weight is not only a danger to the driver and other road users; it is an unacceptable offence which carries with it a range risk and penalties, from fixed fines to prison sentences. Overloading trains significantly increases fuel consumption. Wheels are more prone to wear, steering becomes more difficult to control and trains take longer to react to braking, overload makes the train more unstable and more difficult to stop in an emergency. Drivers and operators alike should be aware of the possibility of prosecutions for dangerous driving, health and safety offence and corporate manslaughter. Overloaded trains damage our track, bridges and underground service producing a massive reduction in predicted road life expectancy and dramatically increasing maintenance costs.

2.4.1 Impact of overloaded train on road safety

As the degree of over loading increase, major safety issues are raised in addition to non-recovery from the road user of damage to the infrastructure.

These issues include:

- Increase severity of accidents when overloaded trains are involved.
- Reduced grade climbing capability and acceleration.
- Greater loss of lateral stability especially when cornering.
- Increased braking distance required for overloaded trains.
- Increased train emissions, noise and ground-borne vibration.

The severity of road accident in Africa is extremely high - estimated to be some 30to 50 times higher than the United Kingdom or the USA. Many of these accidents are caused by overloaded commercial trains. The cost of overloading is estimated to consume some 1 to 2 percent of GNP in Africa. [46]

2.5 Related Researches Review

Mohamed Rehan Karim [47]**,** the purpose of the article was to understand and establish the extent to which train overloading is happening in a developing country like Malaysia by Comparison between actual overloading data for 2- axle, 3-axle and 4-axle trucks and the stopping distance. Because of the author understands the problem of train overloading in developing countries may not fully realized to enable appropriate and effective mitigation measures to be employed. The article also determines the implications of train overloading on safety, simulation data on the stopping distance of different truck categories traveling at different speeds and GVW (Gross Train Weight) was generated using the MSC ADAMS software. Through analysis of the train weight data, namely the GVW was performed to determine the train overloading characteristics at the study location. According to the results from this study Data on GVW violations amongst the different truck types revealed that 50% of the 3-axle trucks are overloaded and the degree of overloading reaches 101% More than a third of the 4-axle trucks (37%) are also overloaded and degree of overloading reaches 84% of the legal weight limit. As such, the 3-axle and 4-axle trucks may be considered as the main contributors to truck overloading occurrences in Malaysia. Furthermore, even though 9% of the 2-axle trucks are overloaded. And the Malaysian government has spent a large portion of the yearly infrastructure budget on road network and bridge maintenance that causes by overloading. A significant amount of the total allocated budget for road maintenance could be saved if road damage caused by overweight trains can be avoided or at least minimized. The damage on road pavements would be accelerated as the volume of overweight train increases.

Finally the author concludes Significant GVW violation involving overweight commercial trains is observed in Malaysia, The frequency and degree of overloading in heavy commercial trains is very significant and alarming. Not only does overloading accelerate pavement damage (which in turn may contribute to accidents), overloaded heavy trains would be hazardous to other road users. Monitoring and enhancing enforcement of weight limits of heavy trains may be a step in the right direction. And comprehensive and continuous data is needed, especially at critical locations in the road network.

In this study the researcher investigates only the happening of train overload and how much the percentage of it in the country but he does not seen the effect of overloading the train on its braking performance and the safety side of the condition

Libo Tango and Guoshun Wang[48]**,** investigated simulation analysis of train disc brake temperature field of train disc brake system in disc brake under 500r/min fixed speed condition and in continuous braking the method they performed the study is carried out by using the software ABAQUS. In the simulation process, heat flux distribution adopts heat flux distribution coefficient method, heat flux distribution proportion between brake disc and brake pad is 8.2, friction coefficient is obtained by actual measurement method, and friction coefficient of 500r/min is 0.41; the result obtained from the study shows the heat quantity generated by braking firstly forms tailed area using friction radius as center; with continuous development of braking process, shaped like a ring temperature distribution is formed, and finally, surface temperature area of brake disc displays spotted. The temperature reaches extreme with extension of braking process. In general the result obtained from this study is good and useful. But the researchers study the thermal analysis with the case of continuous braking only. They do not see in case of emergency braking and overloading condition.

S. Milosevic, S. Staminokovic, P.Milojevic And M.Tomic [49], researches on Modeling Thermal Effects In Braking System Of Railway, using analytical and numerical modeling of thermal effect during long term braking for maintaining a constant speed on a down grade rail road by using finite element method (ANSYS). The thermal analysis was performed for braking a 444 class locomotive of the national railway operator Serbian Railway that run with three different velocities (20km/h, 40km/h and 60km/h) the duration of braking was t=300s for all cases. Conclusion they obtain from the thermal result are the existing models for the estimation of the amount of the heat generated during friction process assume that the energy delivered to bodies in the frictional contact transformations in to heat only due to the friction losses on contact surfaces.

Pyung Hwang and Xuan Wuin [50],the "Investigation of Temperature And Thermal Stress In Ventilated Disc Brake Based On 3D Thermo-Mechanical Coupling Model" investigators wants to investigate the temperature and thermal stress in the ventilated disc-pad brake during single brake. The model is symmetrical about the central crossing plain of ventilated disc. The cover angle of the pad is 60^o. The brake pressure 2.38Mpa is applied on the pad to generate the brake force. Based on multibody technique, the convectional heat transfer coefficient is applied to the surface of brake disc. An experimental investigation is carried out to determine the temperature distribution in the brake disc. The disc temperature is burnished to the initial temperature, and accelerated to 100kph, and then the brake is applied with constant deceleration 0.6g until the disc comes to a stop. The brake is operated three times. the comparison of experimental result and simulation result the temperature fluctuation is due to heat generated in the contact region of the disc work surface and the cooling effect in the rest of the region the temperature at non-contact node is relatively slow, because there is no heat generated, only conductivity effect during brake there is no cooling on the contact surface of the pad temperature field affect the thermal expansion and lead to variation of contact pressure distribution. Due to the heat generation ratio decreases, conduction in the disc and convection in the surface of brake disc. The node temperature on the work surface in the disc is presenting fluctuation. The thermal stress distributions are in accordance with the temperature distribution.

Zaid, M.A. [51]**,** an investigation of disc brake rotor by finite element. The author has conducted a study on vented disc brake rotor of normal passenger train with full load of capacity. This study is concern of heat and temperature distribution on disc brake rotor. The researcher use finite element method analysis approach for conducted in order to identify the temperature distribution and behavior of disc brake rotor in transit response. ABAQUS/CAE has been used as finite elements software to perform the thermal analysis. The author concludes the maximum temperature of transit response is 543.9^{o}c and where the temperature range of gray cast iron which is between -150^{o}c and 550^{o}c, so the author expected that all findings in this study will assist automotive designer in developing optimum and effective disk brake rotor.

Summary of the reviewed literatures

This paper discusses the effect of train overloading on disc brake by taking our country loading trend from the nearest way of transportation to train.

CHAPTER THREE

DATA COLLECTION AND ANALYSIS

The following data are the main parameter of the line, train power supply, basic train technical condition, train dimension, loading capacity, acceleration and deceleration rate of train which are obtained from AALRT.

3.1 Main parameter of lines

Train line parameters are given in the following table.

Table 1: Addis Ababa LRT line main parameters

No	Parameters	Description	value
1	Track gauge		1435mm
2	Minimum radius of horizontal curve.	-Main line between sections. -Yard line	50m 30m
3	Minimum radius of vertical curve		1000m
4	Maximum downhill		5.5%
5	Axle load		≤11（1+3%）t

3.2 Basic technical conditions and overloading assumptions of the train.

3.2.1 Main train dimension.

Main train dimensions are listed in table below

Table 2: Addis Ababa LRT train main parameter

No	Parameters	Description	Value
1	Length of car body		≤30000 mm
2	Maximum width of car body		2650mm
3	Height of train floor from top of rail	low floor area, new wheels and empty load	≤380 mm
4	Height of train floor from top of rail	exit and entry areas, new wheels and empty load	≤350 mm
5	Height of train floor from top of rail	raised floor area, new wheels and empty load	≤900 mm
6	Wheelbase	power bogie unpowered bogie	1900 mm 1800 mm

3.2.2 Train type and train formation

The trains shall be 70% low-floor articulated 6-axle modern trams, consisting of three modules, bi-directional driving. Two tramcars shall be able to operate with double heading.

Train formation: Mc+Tp+Mc

Mc module: motor car with driver’s cab

Tp module: trailer without driver’s cab and with pantograph

+: articulation device

-: Hidden folding coupler

3.2.3 AALRT train Seating capacity and overloading assumption.

Data collected from the Addis Ababa city Anbesa bus enterprise officers, users, operators (drivers and ticket sealers) and technicians about 18 meter articulated bus loading capacity and overloading trend is the bench mark for forecasting how much the Addis Ababa light rail train will be overload.

Figure 3.1: Addis Ababa 18 meter articulated bus.

The Addis Ababa city bus enterprise has large coverage in and around the city of Addis Ababa. To satisfy its customer the enterprise buys 18 meter articulated bus. It has the loading capacity of 48+1seats and 112 passengers are stand on the floor of the bus. But the information gathered shows that during the pick hour the bus is overloaded up to 25 % of its loading capacity. This means above ten stand passengers per meter square. It has about fourteen meter square area for stand passengers. If ten passengers are stand per meter square (28 passengers) the other overloaded passengers are added on the seat or other area which is not recommended for passengers seat or stand.

So the research takes this unacceptable loading trained for the new Addis Ababa light rail train overloading analysis, and indicating the possible overloading area and conditions inside the train.

Figure 3.2: Addis Ababa light rail transit train.

3.2.4 Assumption areas for Addis Ababa light rail transit train overload.

3.2.4.1 The areas which are not considered to passenger seat or stand

It is assumed from the Addis Ababa city bus, that the area behind and around the driver is not included for passenger seat or stand. But in the time of overloading, this area is taken by the passengers. Like to this the area behind the locomotive operator or train master is restricted for passenger but this is one of the assumption areas during train overloading.

Figure 3.3: areas inside the locomotive behind the driver.

3.2.4.2 Area that passenger exit or inter in to train.

In the Addis Ababa city bus, the area where passengers inter in to and exit from are not considered for passenger seat or standing areas but those areas are taken by the passengers in time of overloading.

Figure 3.4: area inside the locomotive passenger inters and exit

3.2.4.3 Ten passengers per m^2 and over seated passenger are assumed for train overloading.

This unacceptable load condition is based on the Addis Ababa city bus overloading trend. There is another unacceptable load condition based on the new Addis Ababa city locomotive train which has three five-in-one (five passengers seat on one long chair as indicated on figure 3.5) and twenty seven two- in-one chairs. One passenger seat Chair has 50cmwidth. The average human being width is 36cm. So in time of overloading it is assumed that three passenger seat on two chairs of the train and eight passengers seat on five in one chairs. Ten passengers are assumed also to stand per m^2.

Figure 3.5: areas inside the locomotive for passenger seat and stand

3.2.4.4 Passenger luggage

Passenger luggage is not considered in the train's weight but for this research, this is one of train unacceptable loading condition. The data collected from the LRT interior interview with Addis Ababa Anbesa city bus users and operators shows the average passenger luggage weight is 2 kg per person.

Figure 3.6: Passenger board on overloaded commuter train in Kenya's capital Nairobi, January 5, 2010(AFP) photo.

Passenger loading capacity and unacceptable load of a locomotive is summarized in the following table.

Table 3: seating and standing passenger loading value of a train

Number of passengers	Seated	Standing	Total
Seats (AW_1)	65	0	65
Seating capacity (AW_2) (standing:6 persons/m^2)	65	189	254
Recommended Overload capacity (AW_3)(standing: 8 persons/m^2)	65	252	317
Unacceptable overloading capacity(AW_4)(standing: 10 persons/m^2)	101	319	420

3.2.5 Train weight

Table 4: Total weight of a locomotive [AALRT]

Loads	Car body weight (ton)	Passenger weight (ton)	Luggage (ton)	Total weight (ton)
Empty train	44	0	0	44
Seating capacity	44	15.24	0	59.24
Recommended Overload capacity	44	19.02	0	63.020
Overload due to more passenger and luggage on board	44	25.2	0.82	70.02

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

3.3 Main technical indicator

3.3.1 Speed

Maximum operation speed: 70 km/h

Average travelling speed: ≥20km/h (average dwelling time of 30 seconds at each station)

3.3.2 Average acceleration

Under rated load and rated voltage on straight and dry track, with half-worn wheels, average acceleration is as below:

Train speed from 0km/h to 40 km/h $\geq 1 m/s^2$

Train speed from 40km/h to 70 km/h $\geq 0.5 m/s^2$

3.3.3 Average braking deceleration

Under rated load on straight and dry track, with half-worn wheels, average deceleration from maximum train operation speed of 70 km/h until stop is as below:

Maximum service brake deceleration: $\geq 1.1 m/s^2$

Emergency brake deceleration: $\geq 2.0 m/s^2$

3.4 AALRT Train Brake System Description

Table 5: type of brake and mounting place [AALRT]

Abbreviation	**Type**	**Mounting place**
ED	Electric brake, Generally depending on load	M1 + M2 Motor bogie for M1 + M2
EHM	Hydraulic Brake of Motor Car	M1 + M2 Motor bogie for M1 + M2
EHT	Hydraulic Brake of Trailer Car	TB1 Trailer bogie for TB1
MTB	Magnetic track brake	M1 + M2 + TB1
Sanding	Sanding device	M1 + M2 Motor bogie for M1 + M2

Note: M1 indicate train motor bogie 1 and M2 for motor bogie 2.

Table 6: type of brake and way of application [AALRT]

Brake system	Service brake (Maximum)	Emergency brake	Safety brake	Stopping brake	Holding brake	Parking brake
ED-Brake	X	X	-	-	-	-
EHM-Brake	-	X	X	X	X	X
EHT Brake	X	X	X	X	X	-
MTB Brake	-	X	X	-	-	-
Sanding	-	X	X	-	-	-

Note: X indicates way of brake application in such brake mode.

3.4.1 AALRT Train Service Brake

Service brake will be activated when driver controller is in brake position. Brake set point is linearly proportional to handle position. ED brake is major brake type for motor car and takes precedence over EH brake. Regenerative brake is applied preferably as electric brake. When train brake cannot be satisfied by ED brake under all speed and load conditions, EH brake of trailers will furnish extra brake based on blended functions. At low speeds (up to 5km/h), electric brake will descend while hydraulic brake will ascend. However, total braking force remains unchanged. Service brake is limited to load correction and jerk.

3.4.2 AALRT Train Emergency Brake (ED + EH + MG brake)

For emergency brake, ED brake for motor car and EH brake and magmatic brake for trailer will operate simultaneously. Emergency brake is not limited to jerk after load correction.

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

CHAPTER FOUR

PHYSICAL AND NUMERICAL MODEL OF DISC BRAKE

4.1 Numerical model of disc brake

Brake Application is defined as an application of the brake that results in a brake force being applied to the train. Brake force is the force applied to the brake disc / pad / braking surface interface. [52]

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

$$\mathrm{v(t) = v_o} \tag{1}$$

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

$$\omega(\mathrm{t}) = \frac{V(t)}{r(w)} \tag{2}$$

$$\omega_o = \frac{Vo}{rd} \tag{3}$$

Where:

v= velocity of the locomotive
v_o= Initial running velocity of the locomotive
ω_o= Initial angular velocity the wheel
ω = angular velocity of the wheel at any time
r_d = the radius of disc

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

$$\mathrm{v_{disc}} = \frac{\mathrm{r_{disc}}}{\mathrm{r_w}}\mathrm{v(t)} \tag{4}$$

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

$$\mathrm{S_b = v_o t_b - \frac{1}{2} a t_b^2} \tag{5}$$

$$\mathrm{t_b = \frac{V_0}{a}} \tag{6}$$

Braking time for service brake =19.44 / 1.1 = 17.67sec and for emergency braking 19.44 / 2 = 9.72sec.

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

$$\omega = \omega_0 \left(1 - \frac{t}{t_b}\right) \tag{7}$$

Where:

t_b = brake time to stop the locomotive

S_b = total brake distance

Initial velocity and mass are belongs to the factor which affects the total braking distance and braking time. The braking distance and time increase with velocity. This is explained by the relation between kinetic energy and velocity. The greater the velocity the greater kinetic energy the brakes have to transfer. Mass of the vehicle is also related to kinetic energy in relationship. The greater the mass the greater kinetic energy, thus a heavier train will require a longer braking distance and time.

Both the braking time and braking distance changes with if the velocity of the train changes. The train going uphill gravity assists and reduces the braking time. Similarly, gravity works against when train are descending and will increase the braking time give a constant deceleration. This paper takes the same velocity of a train (70km/h) on level and gradient track position in time of emergency and service brake and constant deceleration(for service brake $1.1 m/s^2$ and emergency brake $2 m/s^2$)in both track positions and the braking force on the downhill is more than that of the horizontal train position. That is why the braking time in gradient and level track condition in the calculation of this paper is equal.

4.1.1 Train braking energy at level or horizontal track

When the train is braked on the flat surface the brake force have only oppose kinetic energy of the train because the train potential energy is neglected as the downhill of the train is set to be zero. Due to this kinetic energy heat is develop between disc and pads. Braking energy during this time is equals to the train kinetic energy ($K_E = E_b$).

$$K_E = \frac{1}{2} m v^2 \tag{8}$$

The train has also many rotating components and so that it has a rotational kinetic energy due to rotating parts. Then the heat energy equation will be

$$K_E = \frac{1}{2} m v_0^2 + \frac{1}{2} I \omega_0^2 \tag{9}$$

Where:

K_E = Initial Kinetic Energy of the locomotive just before braking starts

m = Mass of the locomotive

I = Polar inertial moment of rotating parts

The moment of inertia of the rotating wheel set and discs can be calculated by using the equation 10.

$$I = \sum_{i=1}^{n} m_i r_i^2 \tag{10}$$

Where:

m_i - represents the masses of the wheel set and brake discs
r_i - represents the rotation radius of the rotating parts from the center of rotation

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

$$K_E = \frac{1}{2}Mv_0^2 + \frac{1}{2}I\omega_0^2 = \frac{1}{2}Mv_0^2\left[1 + \frac{I}{M{r_w}^2}\right] = \frac{1}{2}(1.1)Mv_0^2 \qquad (11)$$

The term $\left[\frac{I}{M{r_w}^2}\right]$ accounts to the rotational masses involved and its value equals 0.1.
In Some literatures, the contribution of the rotating masses of the wheel set and discs is taken to be 10% the tare weight of the axle load of the locomotive [53].

The total braking energy when the trains travel in horizontal track is

$$K_E = \frac{1}{2}(1.1)Mv_0^2 \qquad (12)$$

4.1.2 Brake Force at level or horizontal track

When locomotive begins to brake, the train lost power and will stop by frictional forces. The imposed mechanical energy theoretically transformed to frictional heat.

The brake disc rotor has got two friction surfaces ($2A_b$) swept by the brake pads on both sides. Two friction forces ($2F_{disc}$) act on both faces of the disc at an effective radius (r_{disc}). These forces retard the movement of the locomotive.

$$\text{The total heat flux is equal to} = \frac{\text{Total heat generated}}{\text{braking time} * \text{twise theprojected area}}$$

$$\text{Total heat flux} = \frac{Qg}{2A_{b*t_b}} = \frac{E_b}{2A_{b*t_b}} \qquad (13)$$

Where:

A_b = is the area swept by the brake pad at one face of the disc
The total work of friction force during the whole brake cycle equals with the total heat generated.

$$Q_g = E_b = \int_0^{t_b} P(t)dt \qquad (14)$$

$$\frac{1}{2}Mv_0^2[1.1] = (2F_{disc})\int_0^{t_b} v_{disc}(t)dt \qquad (15)$$

$$\frac{1}{2}Mv_0^2[1.1] \ = (2F_{disc})\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right] \quad (16)$$

The friction force that work on the disc to retard the locomotive in train horizontal track position is:

$$F_{disc} \ = \frac{\frac{1}{2}MV_0^2[1.1]}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} \quad (17)$$

4.1.3 Heat generated during braking at downhill

In braking system the applied mechanical energy is transformed in to heat energy. The generated heat energy range between the pad and disc is determining the braking efficiency of the system. The energy dissipated in the form of heat can generate in temperature range from300^{o}C to 800^{o}C [54].

The total heat generated (Q_g) in the brake system equals the total braking energy (E_b) lost from the locomotive during braking.

$$Q_g = E_b \quad (18)$$

4.1.4 Brake Force at train downhill position

When locomotive begins to brake, the train lost power and will stop by frictional forces. The imposed mechanical energy theoretically transformed to frictional heat.

The brake disc rotor has got two friction surfaces ($2A_b$) swept by the brake pads on both sides. Two friction forces ($2F_{disc}$) act on both faces of the disc at an effective radius (r_{disc}). These forces retard the movement of the locomotive.

$$\text{The total heat flux is equal to} = \frac{\text{Total heat generated}}{\text{twise Pad and disc friction area} * \text{braking time}}$$

$$\text{Total heat flux} \ = \frac{Qg}{2A_{b} * tb} \quad (19)$$

Where:

A_b Area swept by the brake pad on one face of the disc

t_b Braking time

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

$$Q_g = E_b = \int_0^{t_b} P(t)dt \quad (20)$$

$$\frac{1}{2}Mv_0^2[1.1] + M.g.S_b\frac{\delta}{1000} = (2F_{disc})\int_0^{t_b} v_{disc}(t)dt \quad (21)$$

$$\frac{1}{2}Mv_0^2[1.1] + M.g.S_b \frac{\delta}{1000} = (2F_{disc})\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right] \qquad (22)$$

The friction force that work on the disc to retard the locomotive on train downhill position is:

$$F_{disc} = \frac{\frac{1}{2}MV_0^2[1.1]+M.g.S_b\frac{\delta}{1000}}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} \qquad (23)$$

In the present work, considering the amount of heat generation by wear is very small relative to the heat generated by friction, so the effect of material wear is neglected.

Therefore, using the value of the frictional force obtained above, the transient heat flux can be determined.

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

$$\dot{Q}(t) = (2F_{disc})v_{disc} = 2F_{disc}\frac{r_{disc}}{r_{wheel}}(v_o - at) \qquad (24)$$

4.1.5 Heat Flux Entering the Disc

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

Generally, the thermal conductivity of the material of the brake force pads is lower than that of the disc (Kp < Kd). Because the disc material gray cast iron with high carbon content has good thermo-mechanical characteristics. The brake pad has an isotropic elastic behavior so considering 95% of the heat quantity produced will be completely absorbed by the brake disc. [55].

4.1.6 Material properties of gray cast iron

AALRT train disc brake constructed from gray cast iron. For the transient analysis of the passenger train disc brake certain physical properties of gray cast iron are given below

Table 7: material property of gray cast iron [56]

Density [kg/m^3]	7200
Poison's ratio	0.28
Thermal conductivity[W/m-k]	57
Specific heat capacity[J/kg.k]	447
Compressive ultimate strength[Mpa]	820
Tensile ultimate strength[Mpa]	240
Yield stress[Mpa]	400
Thermal expansion[µm/m.k]	10.85

The ratio (ratio of disc and pad) normally is called the proportion of heat transferred to disc ($\gamma = 0.95$). The rate of heat generation is:

$$\dot{Q}_d(t) = \gamma(2F_{disc})v_{disc} = (0.95)(2F_d)\frac{r_d}{r_w}(v_o - at) \quad (25)$$

Now focus on the railway disc rotor. Most of heat generated is characterized by the heating of the disc. The heat conduction in the disc is governed by the classic heat equation:

$$\rho C_p \dot{T} = k\sum_{i=1}^{3}\frac{\partial^2 T}{\partial x_i^2} \quad (26)$$

Where:

- ρ Density for material of the disc
- k Thermal conductivity
- C_p Specific heat capacity

At the contact surface between the brake disc and pads the braking process produces heat. According to the expression below the heat dissipation from the free surface of the disc to the surrounding air is described by both convection and radiation:

$$Q_{disc} = -h(T - T_{ref}) - \varepsilon\sigma(T^4 - T_{ref}^4) \quad (27)$$

In this equation,

- h Convective film coefficient
- ε Material's emissivity
- σ Stefan-Boltzmann constant and
- T_{ref} Temperature of the surrounding air

The effect of radiation heat transfer is not considered because it is less important especially for the low surface temperatures. The primary mean of heat rejection between air and brake rotor surface is done by convection.

The brake disc rotates at high speed, the convection coefficient of the brake disc changes with the disc speeds during braking. For convection heat exchange based on convection heat transfer theory, the convective film coefficient between outer brake surface and air as a function of the railway vehicle velocity, the following formula should be used

$$h = \frac{0.037 \times K_{air}}{2r_b}\mathrm{Re}^{0.8} \times \mathrm{Pr}^{0.33} \quad (28)$$

Reynolds number Re and Prandtl number Pr are given in the following formula

$$\mathrm{Re} = \frac{2\rho_{air} \times V_{rail} \times r_b}{\mu_{air}} \tag{29}$$

$$\mathrm{Pr} = \frac{C_{air}\mu_{air}}{K_{air}} \tag{30}$$

Where:

K_{air} Conductivity of air (0.0241 w/mK)

V_{rail} locomotive speed

μ_{air} Dynamic viscosity of air (12.34×10^{-6} m^2/s)

C_{air} Specific heat capacity

r_b Effective friction radius

The Reynolds and Prandtl number is equals to:

$$\mathrm{Re} = \frac{2\times1.184\times0.1765\times19.44}{12.34\times10^{-6}}$$

$$\mathrm{Re} = 0.658\times10^{6}$$

$$\mathrm{Pr} = \frac{1009\times12.34\times10^{-6}}{0.0241}$$

$$\mathrm{Pr} = 0.517$$

Finally heat convection coefficient is equals to:

$$h = \frac{0.037\times0.0241}{2\times0.1765}\times\left(0.658\times10^{6}\right)^{0.8}\times\left(0.517\right)^{0.33}$$

$$h = 84.36\, w/m^{2}\,{}^{\circ}c$$

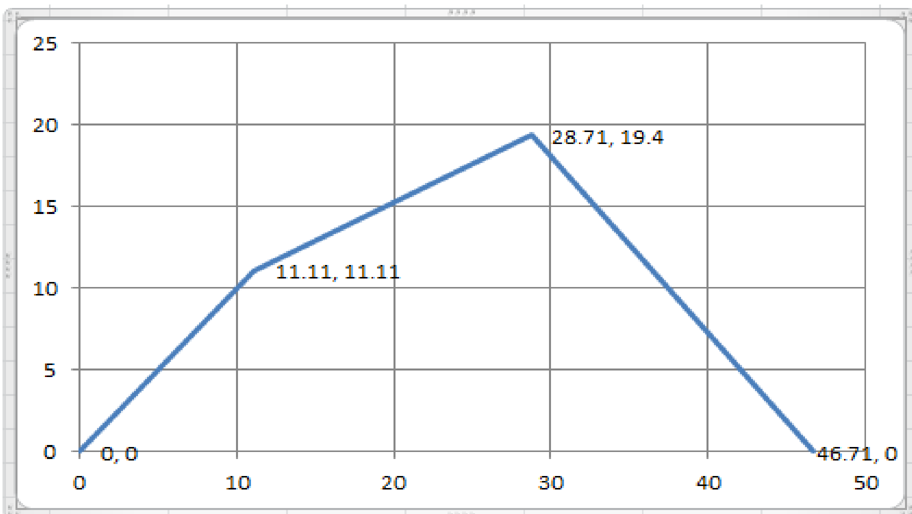
Heat transfer coefficient of the disc is equals to, h= 84.36w/m^2 ^{o}c.

By the end of the first service braking action the disc temperature has reached 400^{o}c as shown on figure 5.4. The train then stops at station for 30 second (average station dwelling time of AALRT) and the train starts to move to the next station with two different acceleration values (a_1= 1.0 m/s^2 and a_2 = 0.5 m/s^2) up to the maximum speed. The first acceleration is 1.0m/s^2 up to a speed of40km/h (11.11m/s) and then up to the final velocity using 0.5m/s^2 acceleration (according to AALRT technical specification). And then the next braking action starts. The total time for cooling of the disc is the summation of the times stopping at station and the time required to reach to the top speed.

$$t = t_s + t_{a1} + t_{a2}$$

Where:

- t total cooling time
- t_s station stopping time
- t_{a1} time for the first acceleration
- t_{a2} time for the second acceleration.



Graph 1: velocity vs. time graph

So the total cooling time of the disc is equal to:

=28.71sec + 30sec = 58.71sec = 59 seconds.

4.1.7 Initial temperature for the next braking

After the first braking, the initial temperature of the disc is rises above from its ambient temperature and it will be computed according to Newton's low of cooling as described below:

$$T = T_\infty + [(T_1 - T_\infty) \times \left[\exp -\frac{UA}{mC} \times t\}\right]] \tag{31}$$

Where:

- T Cooled temperature
- T_1 Maximum temperature
- $T\infty$ Ambient temperature

A Disc brake contact area
U material heat transfer coefficient
t total cooling time
m Mass of disc
c specific heat capacity of gray cast iron

In this research, the initial temperature for the next braking action is calculated by the service brake unacceptable load thermal result obtained (405 ^{o}c).

$$T = 30 + [(405 - 30) \times \left[\exp\{\frac{-5 * 0.0752}{39.4 * 0.447} \times 59\}\right]]$$

$= \underline{57.7\ ^{o}\text{c}}$

4.1.8 Determination of locomotive Load Condition

The specific loading condition in this analysis is braking of the locomotive running with a maximum velocity of 70 km/h to standstill with the temperature of the surrounding is 30°C. The assumed temperature is average environmental temperature of the operating route of the locomotive in the region. In addition, the part of the track with a gradient of 5.5% is also considered.

Hence, the initial temperature of the disc and the surrounding is the same (30°C).The goal is to find out how much temperature distributed on the whole construction of the disc brake rotor by braking on the inclined and horizontal track. The deceleration factor for service brake and emergency brake are 1.1m/s^{2}and 2.0m/s^2 respectively.

In both cases the effect of the humidity in the air and the heat transfer with radiation is not considered.

4.1.9 Weight Transfer during Braking

When a locomotive is standing still on straight track, its weight is uniformly distributed on all axles. On application of brakes on the train, inertia forces are set up. The braking force is applied at lesser height whereas the train load is at a higher height.
Based on the SF6N bogie weight transfer mechanism, the weight transfer efficiency of SF6N (WAG6C) has been 97.4%.

Therefore, equal and uniform weight distribution among the front and rear axles of the bogie is considered in this study.

4.2 physical model of disc brake

The weight distribution of the locomotive considered is equally distributed between on both the front and rear bogies. Each bogie consists of two wheel sets.

Every wheel set is equipped with one wheel mounted mechanical disc brakes on trailer bogie. And one mechanical disc brake on motor bogies axle.

Hence, only 1/8 of the whole brake force is applied to one disc from the trailer part of the carriage and 1/6 on motor bogies disc brake.
The braking energy for one wheel considering train horizontal track position is:

$$\frac{1}{8}\left[\frac{1}{2}Mv_0^2(1.1)\right] = (2F_{disc})\int_0^{t_b} v_{disc}\,dt \tag{32}$$

The braking energy for one wheel considering train deceleration position is:

$$\frac{1}{8}\left[\frac{1}{2}Mv_0^2[1.1] + M.g.S_b\,\frac{\delta}{1000}\right] = (2F_{disc})\int_0^{t_b} v_{disc}\,dt \tag{33}$$

This change of energy is equal to the heat flux on the surface of the disc. This ratio is used to calculate the thermal load on the brake disc. Other operational data used for the analysis are listed in table below.

Table 8: Data for calculating heat flux

Item	**Values**
Mass of the train – M[kg]	63020
Assumed train weight- M[kg]	70020
velocity – v_0[m/s]	19.44
Deceleration for service brake – *a* [m/s^2]	1.1
Deceleration for emergency brake- *a* [m/s^2]	2.0
Service brake braking time – t_b[s]	17.67
Emergency brake braking time - t_b[s]	9.72
Radius of the wheel – r_w[m]	0.330
Incline of the track – δ [%]	5.5

Source: from AALRT and calculating data

Coefficient of friction has significant effect on the thermal field. The value of the coefficient of friction depend on numbers of tribological parameter such as speed of relative movement of parts in contact, the contact surface roughness, quality of materials of a tribo-pair, temperature of contact parts and so on [57].

According to DIN EN ISO1183 the value μ = 0.36 of the friction coefficient for materials of the disc and the pad for locomotives is taken as the recommended value.

4.2.1 Determination of Friction Surface and parameters of Disc and pad.

According to UIC 541-3 Standard, a Sinter material brake pad and Grey Cast Iron Disc with dimension tabulated in table below is used to calculate friction swept area on the disc by the brake pad.

Table 9: Dimension of disc and brake pad for railway locomotives

Description	Disc	Pad
Outer radius (R_o), [m]	0.210	0.210
Inner Radius (R_i) [m]	0.143	0.143
Bolts hole Radius[mm]	(ϕ12*9)	-
Inner bore radius [m]	0.90	-
Friction face Thickness (L)[m]	0.014	0.011
Effective friction radius (r_b)[m]	0.1765	-
Friction surface (A_b, A_P)[m^2]	0.0752	0.0125

Source: from AALRT and calculating data

The effective radius is the intermediate friction surface radius of the disc r_b = 0.1765m. The calculated contact surface area between the disc rotor and braking pad is A_b= 0.0752m^2.

4.2.2 Determination of force on disc.

The value of Force which works for one brake disc in train horizontal position is:

$$F_{disc} = \frac{\frac{1}{8}\{\frac{1}{2}MV_0^2[1.1]\}}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} \tag{34}$$

The value of the friction Force which work for one brake disc at train deceleration position is:

$$F_{disc} = \frac{\frac{1}{8}\left\{\frac{1}{2}MV_0^2[1.1] + M.g.\left[v_o t_b - \frac{1}{2}at_b^2\right]\frac{\delta}{1000}\right\}}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} \tag{35}$$

4.2.3 Pressure determination for the brake caliper

The surface pressure between the disc and pad, on behave of the calculated force applied to the disc, needs to be determined. The pressure force which works for one brake disc is:

$$p = \frac{F_{disc}}{2A_{b*\mu}} \tag{36}$$

4.3 Data for train acceptable load condition

Calculated and collected data about service brake are listed below.

Table 10: calculated and collected service brake data

Service brake			
Brake in horizontal track		**Brake in downhill**	
Description	**Value**	**Description**	**Value**
Braking time – t_b [s]	17.67	Braking time – t_b [s]	17.67
Deceleration – a [m/s^2]	1.1	Deceleration– a [m/s^2]	1.1
Heat flux/disc Q [W]	613578.5	Heat flux/disc Q [W]	886783.7
Pressure/disc P [pa]	138785.6	Pressure/disc P [pa]	200582

Calculated and collected data about Emergency brake are listed below

Table 11: calculated and collected emergency brake data

Emergency brake			
Braking in horizontal track		**Braking in downhill**	
Description	**Value**	**Description**	**Value**
Braking time – t_b [s]	9.72	Braking time – t_b [s]	9.72
Deceleration – a [m/s^2]	2.0	Deceleration – a [m/s^2]	2.0
Heat flux/disc Q [W]	1115425.2	Heat flux/disc Q [W]	1389895.2
Pressure/disc P [pa]	252020.7	Pressure/disc P [pa]	314034.9

4.4 Data for train unacceptable load condition

Data for service brake in train overloading condition are listed below.

Table 12: data for train overloading in service braking condition

Service brake			
Brake in horizontal track		**Brake in downhill**	
Description	**Value**	**Description**	**Value**
Braking time – t_b [s]	17.67	Braking time – t_b [s]	17.67
Deceleration – a [m/s^2]	1.1	Deceleration – a [m/s^2]	1.1
Heat flux/disc Q [W]	681732.3	Heat flux/disc Q [W]	985283.9
Pressure/disc P [Mpa]	154201.4	Pressure/disc P [Mpa]	222861.8

Data for Emergency brake in train overloading condition are listed below.

Table 13: data for train overloading in emergency braking condition

Emergency brake			
Braking in horizontal track		**Braking in downhill**	
Description	**Value**	**Description**	**Value**
Braking time – t_b[s]	9.72	Braking time – t_b[s]	9.72
Deceleration – a [m/s^2]	2.0	Deceleration – a [m/s^2]	2.0
Heat flux/disc Q [W]	1239322	Heat flux/disc Q [W]	1544279
Pressure/disc P [pa]	280014.2	Pressure/disc P [pa]	348916.3

The above calculated values are used for the finite element analysis.

4.5 Disc brake modeling in CATIA and FEA using ANSYS

4.5.1 Modeling

Modeling of disc brake rotor is the primary action before doing of any type of analysis. CATIA software is the standard in the 3-D product design. The figure below is model of the disc brake in

CATIA. The disc brake rotor designed by using the disc brake parameters data obtained from Addis Ababa light rail transit train specification document and actual measuring of the disc brake rotor.

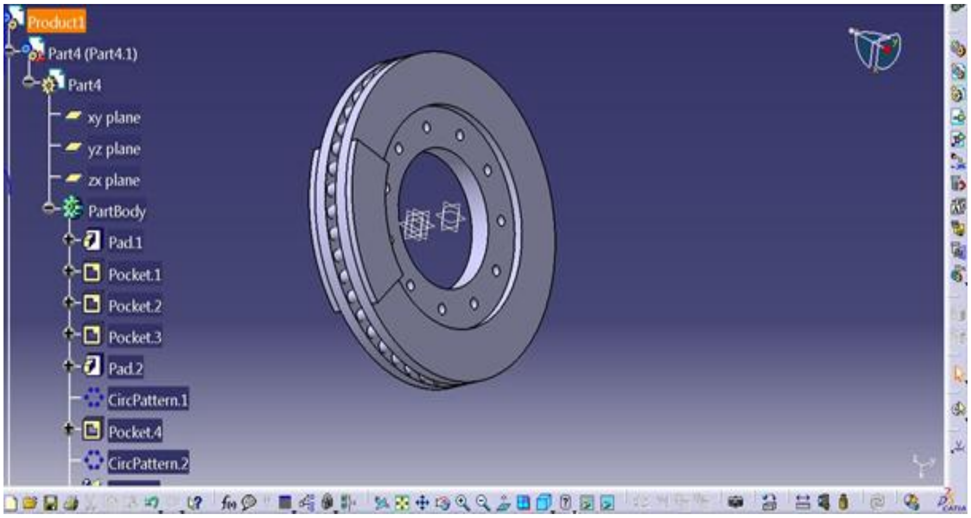
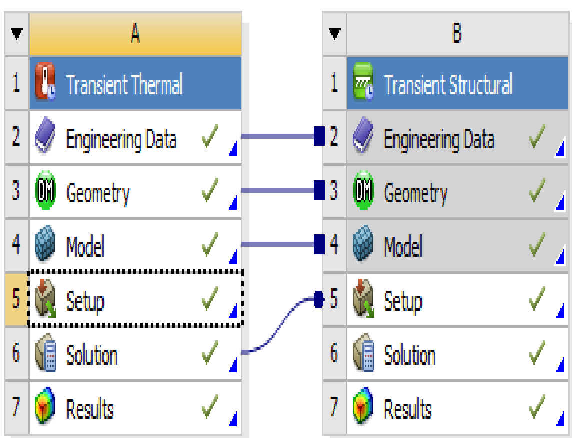


Figure 4.1: disc brake models

4.5.2 FEA USING ANSYS

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

This study will use the FEA software ANSYS14.5 version to carry out transient thermal and structural analysis of a railway brake disc during different loading conditions.



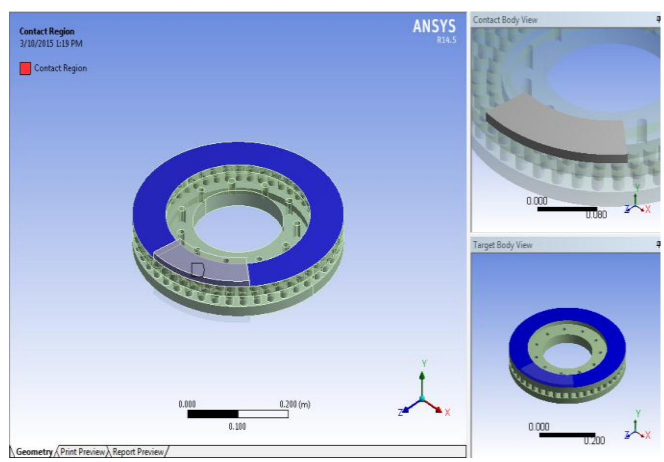


Figure 4.2: General analysis setting.   Figure 4.3: contact body view.

4.5.2.1 Meshing

Meshing is discretizing the solid object to finest parts to perform the analysis for getting the more accurate value at each element of the meshed object. Volume meshed is the method appropriate for discretized the object to the smallest part of the disc rotor. The meshing result consists of 962017 nodes and 615733 tetrahedral elements.

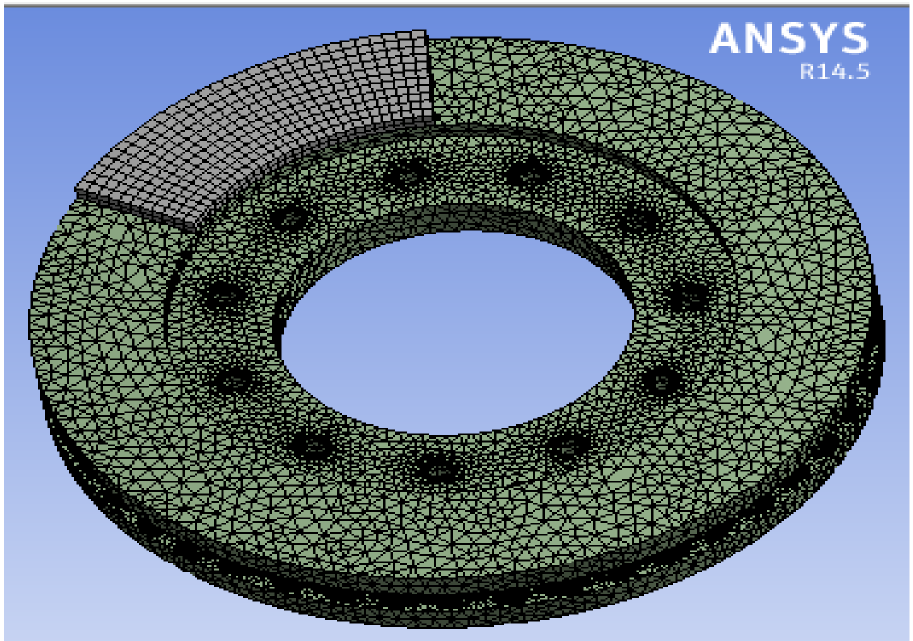
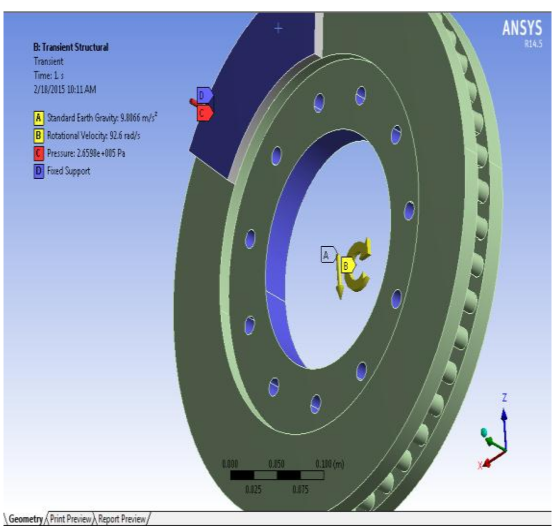


Figure 4.4: meshing of the disc.

4.5.2.2 Loading the essential parameter

Loading in ANSYS terminology for structural and thermal analysis includes boundary condition, force or pressure, velocity, temperature, heat flux and heat convection. After completion the finite element model, loading the terminologies properly to the model is essential.



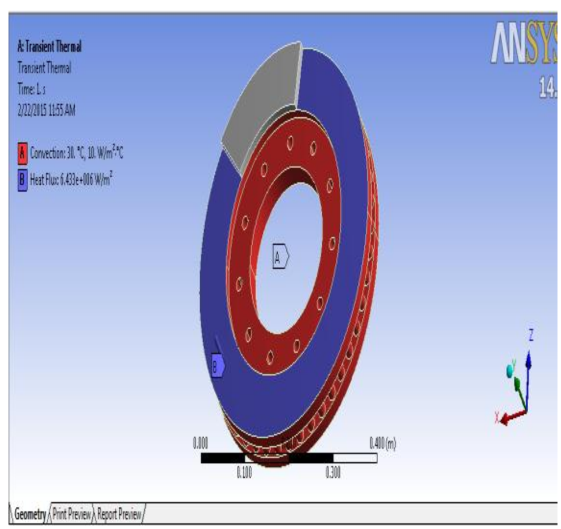


Fig 4.5: structural load imposed to disc and pad. Fig 4.6: thermal load imposed to disc and pad.

4.5.2.3 Boundary Condition

These conditions constitute the initial conditions of our simulation.

- Initial temperature of the disc is 30 ^{o}c.
- Brake disc material: gray cast iron.
- Material heat convection coefficient is calculated by the formula described in equation (28).
- Train braking action is done by hydraulic braking system only.
- Heat flux inter into the brake disc during braking can be calculated by the formula described in equation (19).
- The pressure on the rubbing contact surfaces of the disc and brake pad calculated by the formula described in equation (35)

CHAPTER FIVE

RESULT AND DISCUSSION

5.1 Thermal result

The modeling of temperature in the disc brake will be carried out by taking account of the load variation, type of braking, track position and the choice of disc material. The brake disc is made of gray cast iron with high carbon content; the contact surface of the disc receives an entering heat flux calculated by relation (19).

After applying the necessary loads for transient thermal analysis the following results are obtained from emergency and service brake under acceptable and unacceptable loading conditions in horizontal and downhill train position.

Those pictures below display the surface temperature of brake disc rotor during braking.

5.1.1 Temperature developed on disc in different conditions.

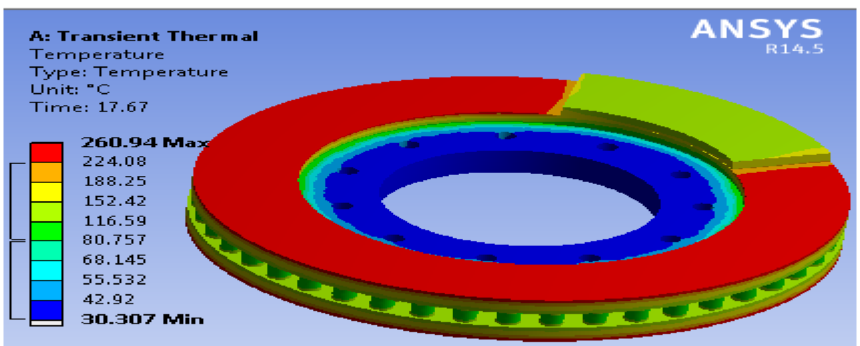


Figure 5.1: Service braking on horizontal track and acceptable loading condition.

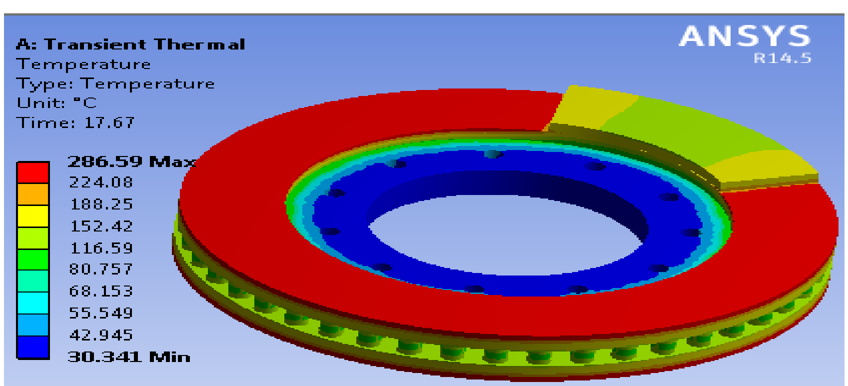


Figure 5.2: Service braking on horizontal track and unacceptable loading condition.

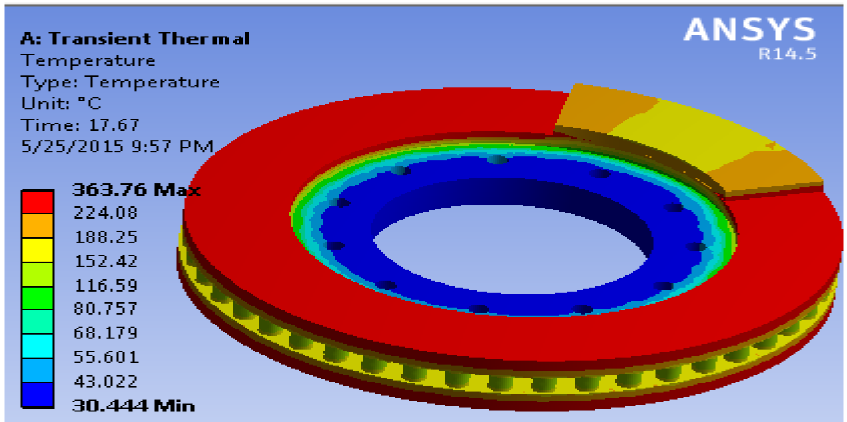


Figure 5.3: Service braking on downhill and acceptable loading condition.

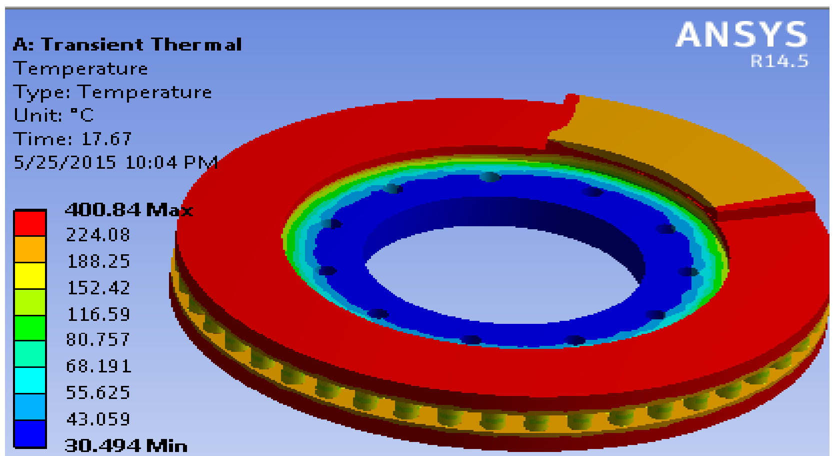


Figure 5.4: Service braking on downhill and unacceptable loading condition.

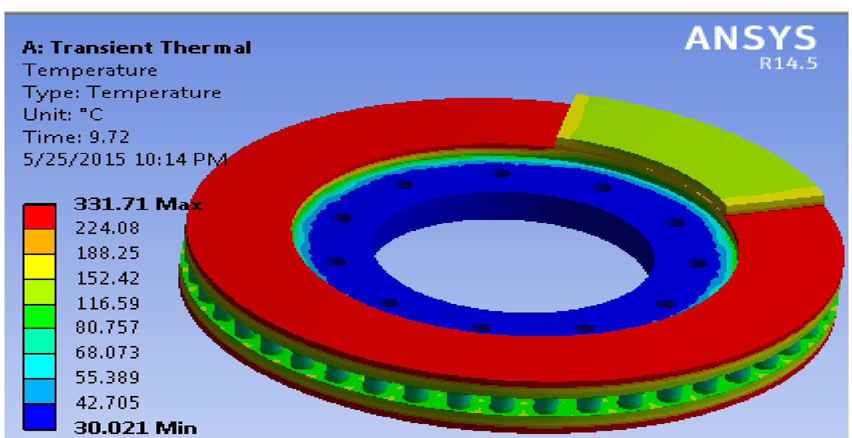


Figure 5.5: Emergency braking on horizontal track and acceptable loading condition.

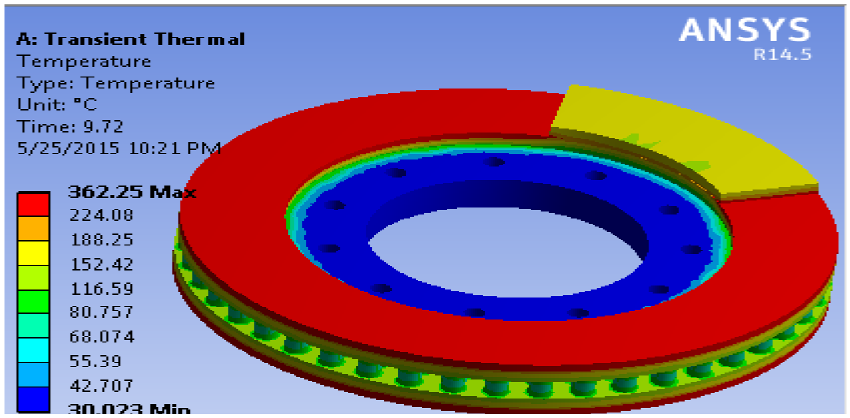


Figure 5.6: Emergency braking on horizontal track and unacceptable loading condition.

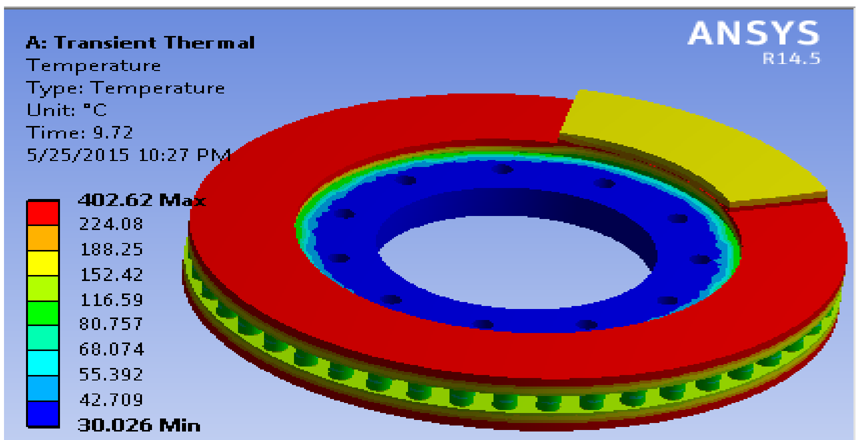


Figure 5.7: Emergency braking on downhill and acceptable loading condition.

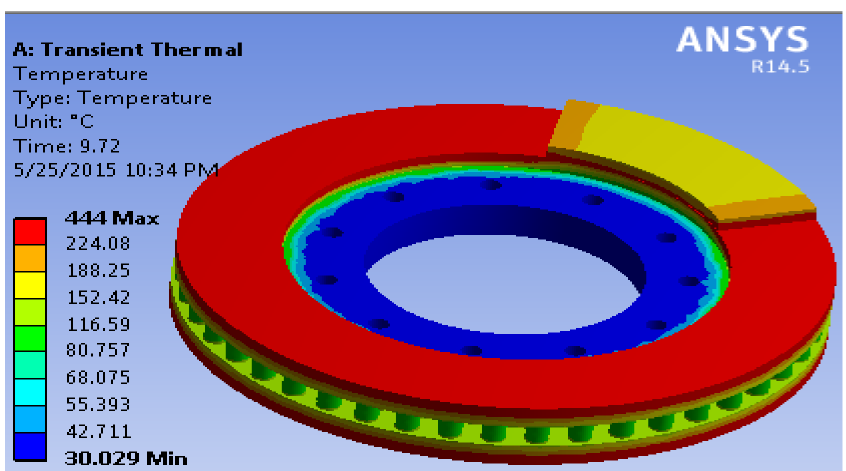


Figure 5.8: Emergency braking on downhill and unacceptable loading condition.

The above figures(from figure 5.1 up to figure 5.8) are shows the temperature developed on the Addis Ababa light rail transit train disc brake rotor at different conditions. The temperature value during acceptable load condition is less than that of the unacceptable. The minimum temperature difference between the two load variations was seen in the condition of service brake and train horizontal position (25.65 ^{o}c) and maximum 41.4 ^{o}c in the condition of emergency braking and downhill train position.

The maximum temperature develops during service brake on train downhill position and unacceptable load condition is 400 ^{o}c and 444 ^{o}c on the same condition of emergency brake.

5.2 Thermal coupled Structural result

5.2.1 Imported temperature

Imported temperature during emergency brake, unacceptable load and downhill train position

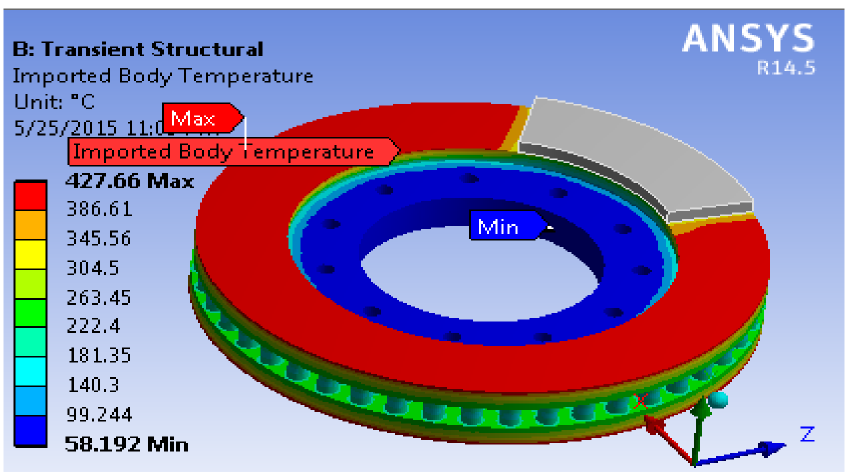


Fig 5.9: Imported body temperature.

5.2.2 Von Mises stress distribution

Von Mises stress is widely used by designer to check whether there design will with stand a given load condition. If the maximum value of von Mises stress induced in the material is less than strength of material it works well most causes. So engineer's duty is to keep the maximum value of von Mises stress induced in the material is less than its strength.

Those figures below shows the distributions of equivalent or von Mises stress distribution of disc brake rotor at different loading, track position and type of braking (service and emergency) brake.

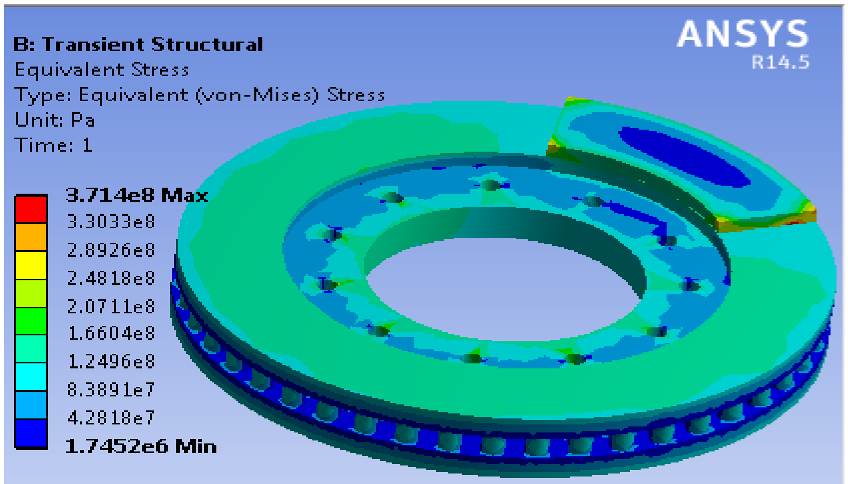


Figure 5.10: Service braking on horizontal track and acceptable loading condition.

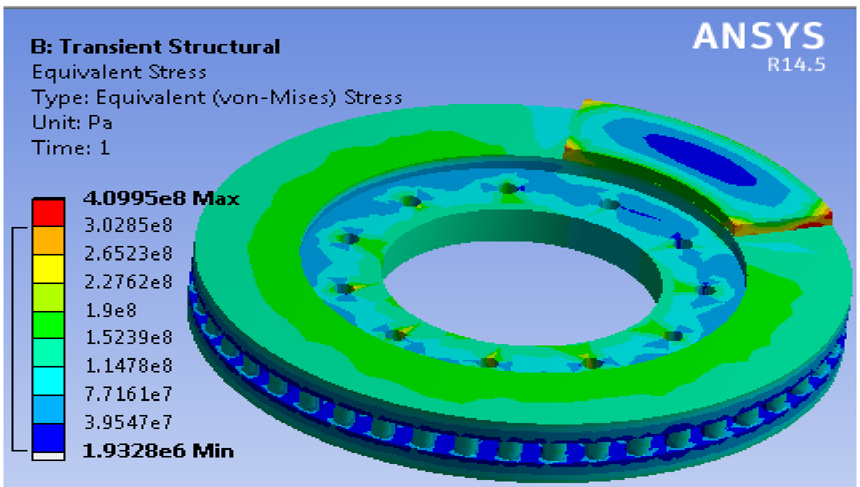


Figure 5.11: Service braking on horizontal track and unacceptable loading condition.

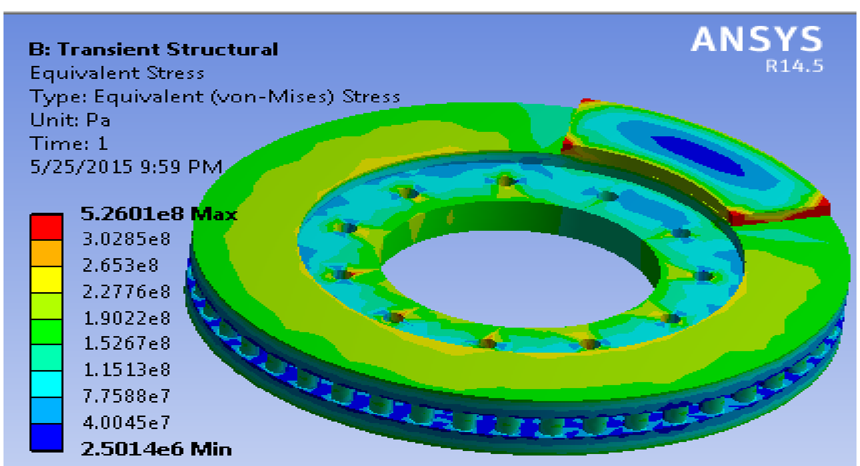


Figure 5.12: Service braking on downhill and acceptable loading.

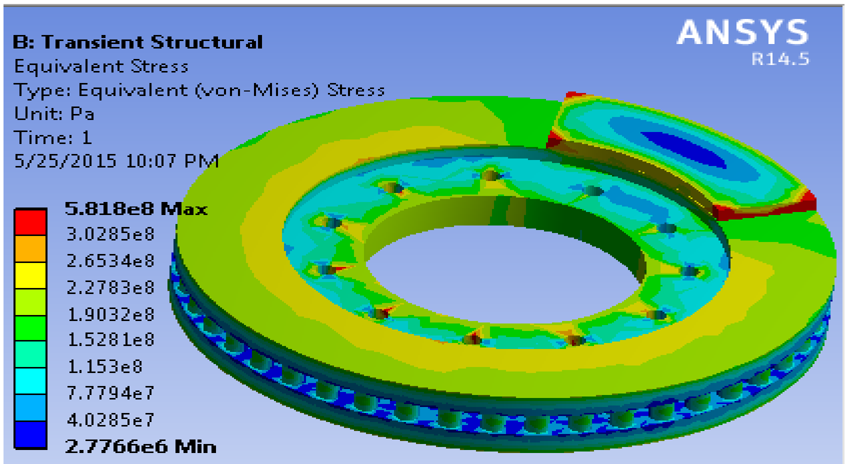


Figure 5.13: Service braking on downhill and unacceptable loading condition.

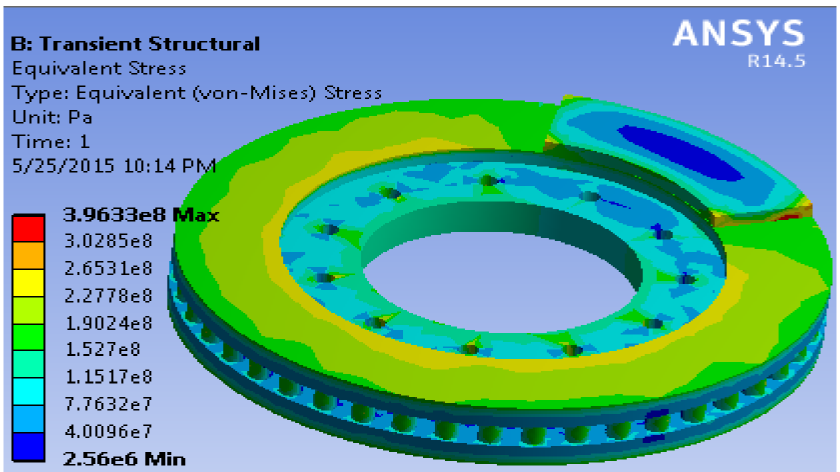


Figure 5.14: Emergency braking on horizontal track and acceptable loading condition

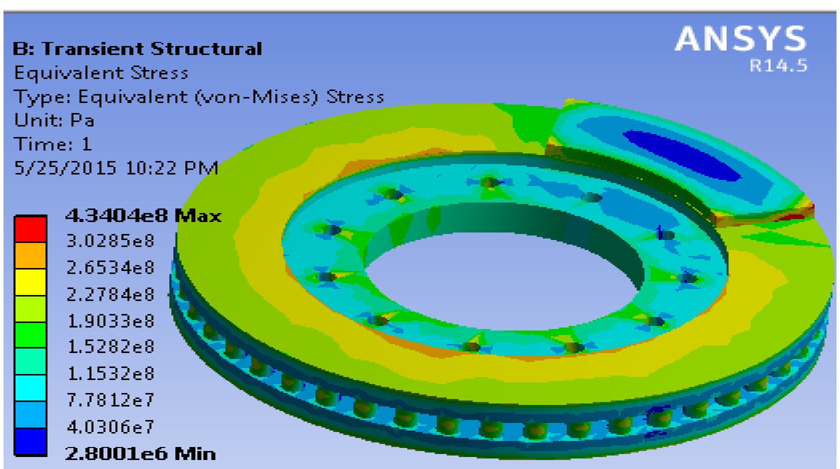


Figure 5.15: Emergency braking on horizontal track and unacceptable loading condition.

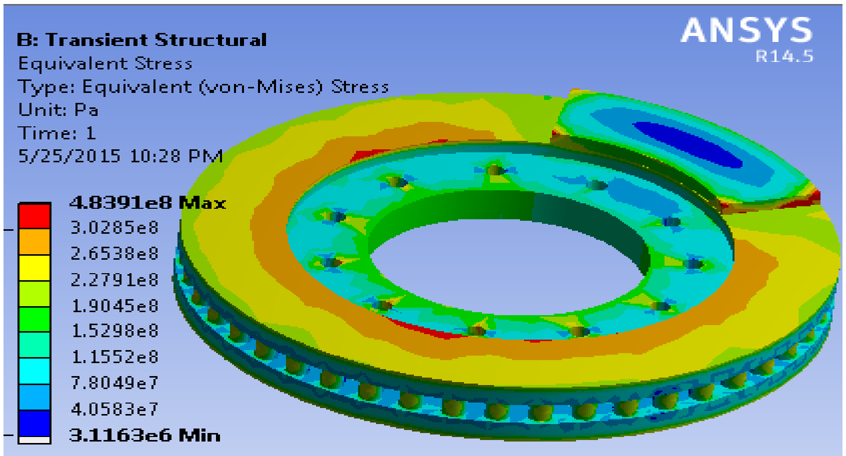


Figure 5.16: Emergency braking on downhill and acceptable loading condition.

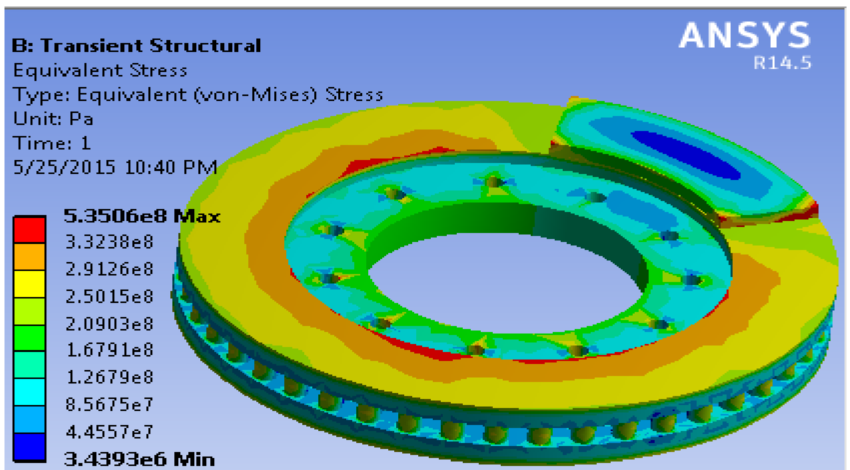


Figure 5.17: Emergency braking on downhill and unacceptable loading condition.

The above figures (from figure 5.10 up to figure 5.17) are shows the difference in von Mises stress distribution on the two braking, two track and two loading conditions.

The equivalent stress value during acceptable train load condition is less than that of unacceptable load. The maximum stress (535 Mpa) appears during emergency brake, train downhill position, and unacceptable loading condition.

5.2.3 Repetitive braking thermal and mechanical stress results

The first braking was simulated with initial disc temperature of 30^{o}c. after standstill in a station; the train acceleration reaches its maximum speed followed by a second braking. Figures below shows the temperature and stress result after applying the residual temperature obtained from expression (31).

5.2.3.1 Repetitive braking thermal results

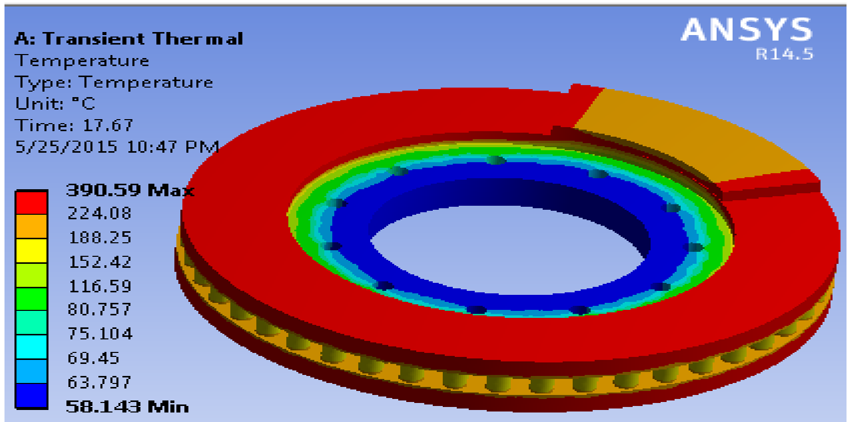


Fig 5.18: Thermal stress on downhill, acceptable loading and repeated brake condition

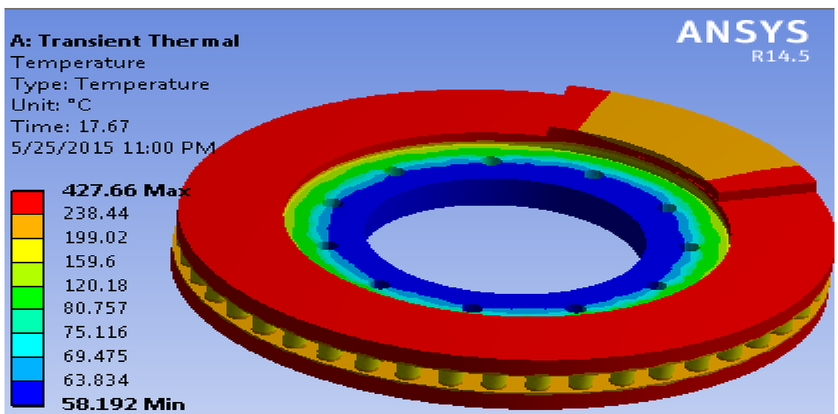


Fig 5.19: Thermal stress on downhill, unacceptable loading and repeated brake condition

5.2.3.2 Repetitive braking mechanical stress results

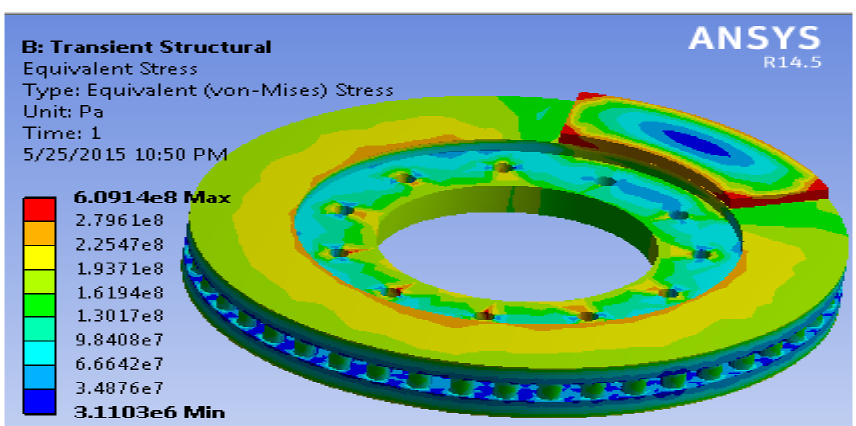


Fig 5.20: Mechanical stress on downhill, acceptable loading and repeated brake condition

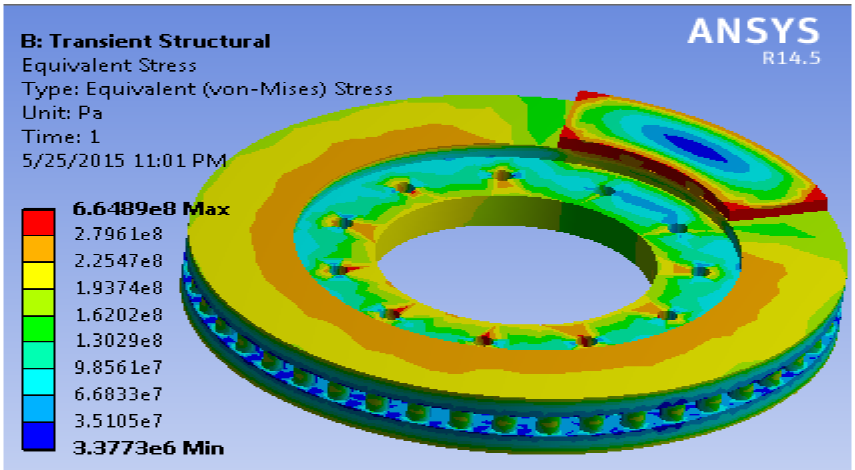


Fig 5.21: Mechanical stress on downhill, unacceptable loading and repeated brake condition

The temperature of the disc after the second braking action has taken is 427.66 ^{o}c which is greater by-27.66 ^{o}c when the brake is applied only for the first time during train overloading.

Similarly the Von Mises stress of the second braking action is 665 Mpa which is greater by 85 Mpa when the brake is applied only for the first time during train overloading.

5.2.4 Summary of results

Table 14: summary of results at different braking condition

No	Braking condition	Acceptable load condition temperature[^{o}c]	Unacceptable load condition temperature [^{o}c]	Acceptable load condition stress[Mpa]	Unacceptable load condition stress[Mpa]
1	Service brake at level	260.94	286.59	371	409.95
2	Service brake at gradient	363.76	400	526	582
3	Emergency brake at level.	331.7	362.25	396	434
4	Emergency brake at gradient.	402.6	444	484	535
5	Repetitive braking using service brake at gradient.	390.59	427.66	609	665

CHAPTER SIX

CONCLUSION, RECOMMENDATION AND FUTURE WORK

6.1 Conclusion

This paper presented the analysis of the thermo - mechanical behavior of brake disc during the braking action; the modeling is based on the ANSYS 14.5. I demonstrate that the effect of train overload on disc brake of AALRT.

From the analysis it can be concluded that the temperature on disc brake during service brake at unacceptable load condition has increased by 36.24 ^{o}c, which has a negative effect on the operation of the disc. Even the condition becomes worst when a repetitive braking is applied. The temperature has increased by 37.07 ^{o}c, which again affects the operation of the disc.

When the temperature of the disc surface increases, there will be a built up of the residual stress for the second braking and it increases the thermal and mechanical stress during repetitive braking.

Stress developed on the disc during normal and unacceptable load condition of service and emergency braking at train downhill position, the stress on the disc has increased by 56 Mpa and 51 Mpa respectively. When a repetitive braking is applied, the stress increased by 56 Mpa. This also affects the operation of the disc.

Generally, adding extra load beyond the capacity of train loading has the effect of developing more thermal and mechanical stress on the disc rotor than that of the acceptable load condition.

6.2 Recommendation

If the temperature and von-misses stress of the disc increases there will a built up of the residual stress. Overheating and over stress eventually leads to increase in deflection of the disc and causing the disc approach to fade. The disc approaches to the fade condition will adversely affect the stopping distance of the train which might even lead to an accident. So the Ethiopian Railway Corporation should give high attention to the overloading conditions of the train to prevent the disc brake from overheating, over stressed and its consequential damage.

6.3 Future work

- Analysis of effect of train overloading on Addis Ababa light rail transit with the actual or operational overloading data after the system starts.
- Experimental study on Effect of overloading on brake disc on test bench and compare the results with the numerical analysis.

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APPENDIX

Interview check list questions used with A.A Anbesa city bus

How much is the normal loading capacity of 18 meter bus?

Seated passenger ____?

Stand passenger_____?

Is there a trend of loading over capacity? Yes/ No

If yes, how much overload during a time?

Which area of the bus was prone to overload the bus by accommodate more passengers?

A. Area behind the driver and around.
B. Loading of passenger stand area above the recommended number.
C. By adding passengers on train seating chairs?
D. In all conditions.

How much average kilogram of luggage has a passenger?