

ADDIS ABABA UNIVERSITY

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

OPTIMUM THERMAL STRESS MATERIAL ANALYSIS OF TRAIN VENTILATED DISK BRAKE

A RESEARCH SUBMITTED TO THE SCHOOL OF GRADUATE STUDIES OF ADDIS ABABA UNIVERSITY IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE AWARD OF MASTER OF SCIENCE IN ROLLING STOCK ENGINEERING

BY

ASRAT AIRHANU

August 2016

Addis Ababa, Ethiopia

Declaration

I, under signed, Declare That This Research is My Original Work, Has Not Been Presented For A Degree In This Or Other University And That All Sources Of Materials Used For This Have Been Acknowledged.

Asrat Birhanu _____ _____

Name Signature Date

Place Addis Ababa University, Institute of Technology, Addis Ababa, Ethiopia

August 2016

ADDIS ABABA UNIVERSITY

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

Optimum Thermal material Analysis of Train Ventilated Disk Brake

By

Asrat Birhanu Gebre Marriam

August 2016

Approved By Board of Examiners

1. Dr.Ing:- Demis Alemu	_____	_____
Advisor	Signature	Date
2. Dr, Ing. Yilma	_____	_____
Internal Examiner	Signature	Date
3. Yidnekachewu Mesele (M.Sc)	_____	_____
Internal Examiner	Signature	Date

Dedication

To my beloved brother for his immeasurable believes encouragement and patience

Acknowledgement

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

I express my deep sense of gratitude towards my advisors Dr. Ing. Demis Alemu whose guidance and contributions of knowledge to this research led me towards completion. I am greatly indebted to him for his constructive suggestions and criticism from time to time during progress of my work.

I would like to express my gratitude to Ethiopian Railway Corporation for providing me the opportunity to study Railway Engineering as well as furnishing the necessary materials to perform this research.

Finally, I want to thank my classmate students and instructors at the department. I am also thankful to all my friends and the staff members of school of Mechanical and Industrial Engineering and to all my well-wishers for their inspection and help.

Abstract

The brake disk is an important component in the braking system, which is expected to withstand and dissipate the heat generated during the braking event. Hence, stresses arising from thermal effects are much higher than mechanical stresses and are more likely to give concerns over the disk integrity. In the present work, an attempt has been made to investigate the most acceptable material, which has better performance. In order for this to be achieved, the brake rotor must be designed such that it ensures sufficient uniform heat dissipation and thermal capacity. In high-demand braking applications, vented disks consisting of two rubbing surfaces separated by straight radial vanes are normally employed, as they utilize a greater surface area to dissipate heat. Within this paper, the heat transfer coefficient has been determined using ANSYS CFX.

Also, thermo elastic instability (TIE) phenomenon (the unstable growth of contact pressure and temperature) is investigated in the present study, and the influence of the material properties on the thermo elastic behaviors (the maximum temperature on the friction surfaces) is investigated to facilitate the conceptual design of the disk brake system. Based on these numerical results, the thermo elastic behaviors of the four brake materials are discussed and ALMMC with excellent mechanical and thermal properties is selected for design purpose.

Key words: brake disk, Heat transfer, HTC, ANSYS CFX & CFD, thermal & structural analysis.

Table of contents page

CHAPTER ONE

INTRODUCTION 1
1.1 Background.....1
1.2 Principle of disk brake….....5
1.3 Problems concerning disk brake.....6
1.4 Scope of the study.....7
1.5 Objective of the study.....8
1.4.1 General objective….....8
1.4.2 Specific objective….....8
1.6 Significance of the study.....8
1.7 Limitation of the study.....9
1.8 Research Method 9
1.9 Organization of the study.....9

CHAPTER TWO

LITERATURE REVIEW.....11

CHAPTER THEE

3. Data analysis and thermal distribution analysis of disk brake.....15
3.1 Disk surface failure due to craking.....15
3.2 Wight of the disk.....16
3.3 Improvement on disk brake.....16
3.4 Material optimization.....17
3.5 Promising materials for disk brake.....17
3.6 Remarkable material for brake pad.....18
3.7 Determining numerical model of disk brake.....19
3.7.1 Braking time and angular velocity.....19
3.7.2 Braking energy of the train.....20
3.7.3 Heat energy.....22
3.7.4 Heat Flux.....22

3.7.5 Calculation of boundary conditions for numerical analysis.....23
3.7.6 Determination of the physical model of the train.....24
3.7.7 Determination of centrifugal force of the disk.....25
3.7.8 Determination of heat flux.....26
3.7.9 Determining of the caliper brake presure.....27
3.8 Thermal distribution analysis.....27

CHAPTER FOUR

4. ANSYS ICEM AND CFX MODELING.....29
4.1 Presentation of computing code ANSYS.....29
4.2 ANSYS ICEM CFD modeling and mesh preparation.....30
I. Create and import geometry.....30
II. Defining the volume or BODY.....31
III. Creating parts.....32
IV. Blocking the geometry.....32
V. Generating mesh.....33
VI. Checking the mesh and cleaning up.....35
VII. Creating the file for CFX.....35
4.3 Modeling with ANSYS CFX.....35
a) Physical model.....37
b) Definition of the model.....37
c) Definition of the boundary condition and application.....38
d) Temporary conditions.....41
e) The calculation result of heat transfer coefficient.....42

CHAPTER FIVE

5. FEA ANALYSIS OF VENTILATED DISK BRAKE USING ANS.....43
5.1 Introduction to FEA software ANSYS.....43
5.2 General analysis procedure in ANSYS.....43
I. Geometrical Definition.....43
II. Model generation.....43

III. Define material properties.....44
IV. Generate mesh.....44
V. Finite element generation.....44
VI. Boundary condition and loading.....45
VII. Solution.....45
VIII. Post processor.....45
5.3 Analysis of train ventilated disk brake.....46
5.3.1 Considerable assumption.....46
5.3.2 Geometrical model.....47
5.3.3 Candidate material definition.....48
5.3.4 Meshing.....49
5.3.5 Definition of the loads and boundary conditions.....49
A. In a case of thermal analysis, the train-ventilated Disk brake is subjected to the following loads and boundary conditions.....50
B. In a case of structural analysis, the train-ventilated Disk brake is subjected to the following loads and boundary conditions.....50
53.6 Soluion.....52

CHAPTER SIX

6. RESULT AND CONCLUSION.....53

6.1 Simulation Results.....53
6.1.1 Thermal result.....53
6.1.2 Structural result.....58
6.2 Result Discution.....66
6.3 Conclusions.....69
6.4 Recommendation.....70
6.5 Scope for further study.....70
6.5 References.....71

LIST OF FIGURES

Figure 1.1 Disk brake system assembly parts.....5
Figure 3.1.Physical model of ventilated disk brake of a train.....16
Figure 4.1 Imported solid ventilated disk brake (i.e. one twelfths of the full disk) and the fluid domain air.....31
Figure 4.2 Final mesh result quality.....34
Figure 4.3 Simulation steps with CFX ANSYS 12.....36
Figure 4.4 Brake disk model after all the domain & BCs loaded.....41
Figure 4.5 Wall heat transfer coefficient of brake disk.....42
Figure 5.1 Geometric Models of ventilated disk brake.....47
Figure 5.2 Meshed Model of ventilated disk brake.....49
Figure 5.3 thermal boundary conditions and constrains.....51
Figure 5.4 structural boundary conditions and constrains.....51
Figure 6.1 Nodal temperature field contours throughout simulation of ALMM disk.....54
Figure 6.2 Nodal temperature field contours throughout simulation of Grey cast iron disk.....55
Figure 6.3 Nodal temperature field contours throughout simulation of Cast steel disk.....56
Figure 6.4 Nodal temperature field contours throughout simulation of C/C Composite disk.....57
Figure 6.5 Total deformation field contours throughout simulation of ALMMC disk.....58
Figure 6.5 Von-mises stress field contours throughout simulation of ALMMC disk.....59

Figure 6.6 Total deformation field contours throughout simulation of Grey cast iron disk.....60
Figure 6.6 Von-mises stress field contours throughout simulation of Grey cast iron disk.....62
Figure 6.7 Total deformation field contours throughout simulation of

Cast steel disk.....62
Figure 6.7 Von-mises stress field contours throughout simulation of Grey cast steel disk.....63
Figure 6.8 Total deformation field contours throughout simulation of C/C composite disk.....64
Figure 6.8 Von-mises stress field contours throughout simulation of C/C Composite disk.....66
Figure 6.9 Temperature distribution gradient of the four disk material throughout braking time 16 seconds.....68

LIST OF TABLES

Table3.1 The potential materials for brake disk and pad.....18

Table 3.2 Input parameters.....25

Table 4.1 Boundary condition in ANSYS CFX.....36

Table 4.2 Definition of the Solid domain and Fluid domain.....37

Table 4.3 Boundary Physics for CFX.....37

Table 5.1 Candidate materials property of ventilated disk brake and brake pad.....48

Table 6.1 Results of candidate disk materials.....67

NOMENCLETURES

M	Mass
a	Acceleration or deceleration
V_f	Final velocity
v_o	Initial velocity
S	Distance
t	Time
s_b	Braking distance
g	Acceleration due to gravity
μ	Coefficient of friction
q_x	Conduction heat fluxes in x direction
q_y	Conduction heat fluxes in y direction
q_z	Conduction heat fluxes in z direction
c_p	Specific heat
Q	Internal heat generation
T	Temperature
Q_s	Surface heat flux
h	Convective heat transfer coefficient
T	Unknown surface temperature
T_∞	Convective exchange temperature
H_D	Horizontal difference
ω	Angular velocity
N	Normal force
T_m	Motor torque
T_{wheel}	Torque on the wheel
P	Power of the motor
ρ	Density of fluid
V	Velocity of passenger train
D_{oD}	Disk outer diameter
η_o	Efficiency of brake transmission

CHAPTER ONE

INTRODUCTION

1.1 Background of the study

In the early years of vehicle development people realized that to be in full control of a machine, one have to be able to start it, but also to stop it. At that time, as still today, the braking system was considered to many people to be the most important system involved in the operation of a vehicle [1].

However, it took quite a long time until vehicles, drawn by animals, were equipped with true brakes. With various systems, as pole lever brakes, bands and wooden shoes rubbing against the wheel, people tried to slow down cabs and stagecoaches. The first cars had an even greater disadvantage; they had no brakes at all. Therefore, the first steam-driven wheeled vehicle developed by Nicolas Cugnot in 1769 was brought to a halt when the vehicle collided with a wall because of the lack of brakes [2].

The original application of brakes, by means of controlling locomotion, was for mining skips, later they were further improved by the development of railways. However, it was only with the advent of gas as a fuel in combustion engines that true car brakes were developed. In effect, there was an increase in speed but safety was reduced and therefore accidents multiplied. It was Lanchester, who first applied for a patent for the disk brake in 1902. This consisted of a thin metal brake disk fixed to the wheel axle. The disk was gripped by two friction material parts when the brakes were applied. Nevertheless, the results were quite disappointing because of the lack of suitable materials and the inefficiency of the control circuit. One of the major steps in disk brake development happened in 1953 when Dunlop equipped a Jaguar XK 120 with disk brakes. One year later, ventilated disks were used for the first time on an Alfa Romeo. Then, it took almost another ten years until disk brakes were applied on heavy trucks.

The need for a good braking system became more and more relevant as the speed and load of vehicle increased. The amount of energy stored in the vehicles motion increases with the square of the speed and its tonnage but much more dramatic is the increase of power dissipated in case

of an emergency stop. In a case of frictional disk braking operation, the vehicle's kinetic energy is entirely converted into thermal energy by the frictional interaction between brake pads and the rotor.

The major problem in the case of friction brake is the friction between wheel and brake shoes, brake pads and disk respectively, which leads to severe thermal regimes and special thermal fatigue nature efforts, requiring specific constructive and operating standards.

The generated heat is distributed between the rotor and the pads. This partition between pads and disk depends highly on the physical-chemical properties of the two materials. Over 95% of this thermal energy flows into the rotor and is stored, conducted to the hub and the wheel, and for longer brake applications convected to the surrounding air [3]. Since, brake disk rotor is a responsible component from safety point of view; materials used for brake disc should have stable and reliable frictional and wear properties under varying conditions of load, velocity, temperature and environment, and high durability. There are several factors to be considered when selecting a brake disk material. The most important consideration is the ability of the material to withstand high friction and less abrasive wear. Another requirement is to withstand the high temperature that evolved due to friction. Weight, manufacturing process ability and cost are also important factors those are need to be considered during the design phase. The brake disk must have enough thermal storage capacity to prevent distortion or cracking from thermal stress until the heat can be dissipated.

Nowadays transportation is one of the key sectors that play important in achieving the goals of poverty eradication and sustainable development. The transport sector is strongly linked and influences developments in other sectors of the economy. Many of transportation problems are caused by their braking system failure. Therefore doing researches on vehicles (trains), braking system is crucial in order for the safe movement of the passengers. The railway vehicles are the most popular, widely available way and a kind of acceptable level transportation. Therefore, the train must be suitable, reliable and safe.

The most important part of the vehicle is the braking system. Train braking is a very complex process specific to rail vehicles and of great importance by the essential contribution on the safety of the traffic. This complexity results from the fact that during braking occur numerous phenomena of different kinds: mechanical, thermal, pneumatic, electrical, etc. The actions of

these processes take place in various points of the vehicles and act on different parts of the train with varying intensities. The major problem is that all must favorably interact for the intended scope to provide efficient, correct and safe braking actions.

At first, the rather low locomotives power and traction force allowed braking using quite simple handbrakes that equipped locomotives and eventually other vehicles of the train. As the development of railway transport and according to increasing traffic speeds, tonnages and length of trains, it was found that braking has to be centralized and operated from a single location usually the locomotive driver's cabin and commands have to be correctly transmitted along the entire length of the train. As a consequence, along the time for railway vehicles have been developed various brake systems whose construction, design and operation depend on many factors such as running speed, axle load, type, construction and technical characteristics of vehicles, traffic conditions etc.

Among various principles and constructive solutions those were developed, the following studies and especially the results of numerous tests, the indirect compressed air brake system proved to have the most important advantages. Generally, the complementary braking systems provide consistent controllable braking forces permitting speed reductions. These braking systems add braking power without having the thermal capacity limitations of the friction wheel or disk brakes that would necessitate expensive solutions or lead to excessive wear from harder use and to diminish the wear of the friction based braking systems.

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

Thermal and mechanical analysis is involved in almost every kind of physical processes and it can be the limiting factor for many processes. Therefore, its study is of vital importance and the need for powerful thermal and mechanical analysis tools is virtually universal. Furthermore, thermal and mechanical effects often appear together with or because of other physical phenomena [5]. The modeling of thermal effects using computer simulation has allowed

engineers and researchers to optimize process efficiency and explore new designs while at the same time reduce costly experimental trials. It is a great way to test a model before the model built and real world tested in a thermal chamber. It can reduce the time to test a design by weeks allowing for several redesigns and improvements to be made in the meantime [6].

Thermal stress analysis is conducted to investigate how heat affects certain materials and engineering designs. This heat can come in the form of an environmental load such as an ambient temperature of a certain degree affecting a model, or due to friction in a system, effectively converting it into thermal energy. It can also come from processes of conduction through two solids, convection between a liquid and a solid, or radiation such as in space [5, 7].

In recent years, many researchers have been done on possibilities of predicting the thermal overload based on the characteristics of trains and railroads especially on long term braking on train operation. Having concluded their researches, the European Rail Research Institute (ERRI) proposed a valuation procedure with the ultimate goal of making a decision on taking special measures to prevent the occurrence of thermal overload. This implies the application of the regulation and alternating brake or other measures related to regime change and drive train formation. However, the proposed procedure, which involves calculating the coefficient for the particular estimation, does not provide an efficient and reliable use in all potential cases. Therefore, the research for the improvement of the thermal overload assessment process continues [8].

Many research results have confirmed the dominant influence is thermal loads concerning mechanical loads and residual stresses induced by high thermal loads in brake disks. Therefore, it is important to determine with high precision the temperature field of the braking system as well as to emphasize those high thermal loads.

This study is focused on the temperature and thermal stress analysis of candidate materials (aluminum based metal matrix composite, cast iron, cast steel and carbon carbon-composite) of disk brake of train under centrifugal and thermal load in emergency brake applications has examined. The transient thermo elastic analysis of ventilated disk brake has been performed and finally the results are compared. The three-dimensional model is generated using software tool CATIA V5, and the meshing is done using in workbench ANSYS ICEM CFD as a pre-processor

and ANSYS-CFX package used as heat transfer coefficient solver and post processor tool. The analyses carried out on existing brake system. In the load analysis stage, heat flux and convection heat transfer coefficients were determine based on the maximum overall vehicle load. In this stage, the materials properties, load and boundary conditions as well as meshing properties are all assign to the geometrical ventilated disk brake model.

1.2 Principle of disk brake

The main components of disk brake are caliper; piston, brake pads, and rotor (see figure 1.1). The caliper is attached to some stationary part of the vehicle like the axle casing or the stub axle as is cast in two parts each part containing a piston. In between each piston and the rotor there is a friction pad held in position by retaining pins, spring plates etc. passages are drilled in the caliper for the fluid to enter or leave each housing. The passages are also connected to another one for bleeding. Each cylinder contains rubber-sealing ring between the cylinder and piston.

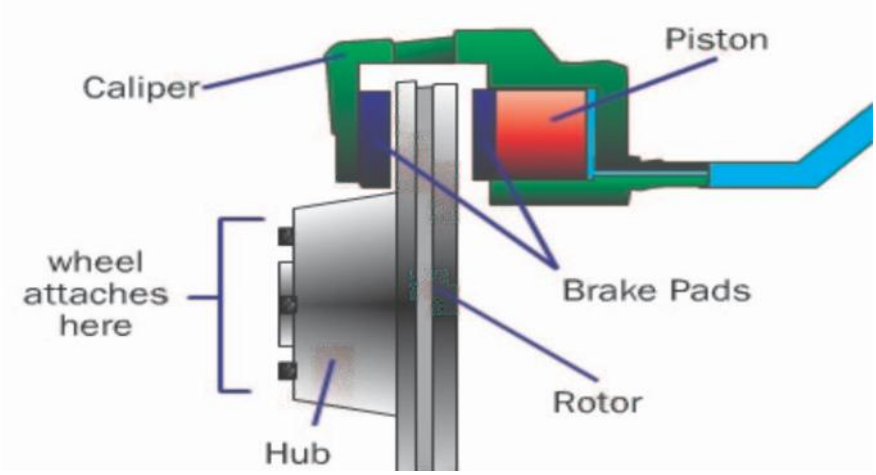


Figure 1.1 Disk brake parts system assembly

The principle of a disk brake is very simple: A metal disk, the rotor is firmly mounted to the rotating axle. When the brake applied, hydraulically actuated pistons move the friction pads in to contact with the rotating disk or wheel, applying equal and opposite forces on the disk or wheel. Due to the friction in between pad surfaces and disc or wheel, the kinetic energy of the rotating wheel is converted into heat, by which locomotive is to stop after a certain distance. On releasing the brakes, the rubber-sealing ring acts as return spring and retracts the pistons and the friction pads away from the disk.

1.3 Problems concerning disk brake

One of the most and frequently happen complex issue involved in the railway transportation is associated with performance of the braking system (disk) for passenger trains as well as freight. As outlined by transport reporter in Melbourne in November 16, 2006, [10], Some of Melbourne's newest passenger trains have had to be withdrawn from service after a spate of braking failures. The problems were connected to trains computerized braking systems so here the movement of the train depends on the performance of manual braking system. Some researcher argue that disk brake has some problems like scarring, cracking, warping or excessive rusting therefore they choose shoe brake for trains but on the other hand other researchers says disk brake not get too hot they become less effective to a phenomenon known as brake fade and is more efficient [9].

During repeated brake application heavy mass vehicle and high-speed vehicle like trains, a great deal of energy is required to stop or decelerate the spinning disks artificially. This process gave birth to the following basic braking phenomena:-

- Stresses and deformations of brake disk material resulted from high temperature.
- Stresses and deformations of brake disk material resulted from centrifugal force.
- Thermal distortion, which result in brake judder.
- Heavy weight of grey cast iron increases the inertia (body force) and unsuspended mass of the vehicle.
- Thermo-elastic instability, because of pressure and temperature growth at the contact surface.
- The frictional heat generated on the rotor surface can influence excessive temperature rise, which in turn leads to undesirable effects.

The main problem of brake application is the great input of heat flux into the disk in a very short time. Because of the high heat flux (temperature difference), the brake disk material is exposed to high stress and deformation [11].

Therefore doing research on analyzing thermal stress performance of railway vehicle disk brake system will have an advantage to solve many problems regarding braking system performance, which are listed in this research and different literatures. In addition to saving maintenance cost.

Consequently this research is designed to come up with solution by investigation into usage of the most acceptable material (highest thermal stress resistance material lower temperature distribution), which improve braking efficiency; provide greater stability to railway vehicle and a great importance by essential contribution on safety of the traffic.

1.4 Scope of the study

In this project, the thermal and mechanical behavior and performance of the candidate brake-disk materials under certain loading conditions has studied. This shall be done by simulating the loading conditions under emergency baking application, the brake-disk is predicted to undergo and analyzing the results. The primary focus of this project is to simulate the performances of the brake-disk in each case of materials using ANSYS consequently to investigate the best material.

Braking disk can be divided into solid or ventilated brake-disk systems. In this study, ventilated brake disk modeled, tested and then thermal and structural stress analyzed. As the braking power of the front brake-disk is more important than the rear disk during hard braking and stops, considering the front brake-disk of each system would be studied.

The analysis and comparison study will be done through modeling and simulation using the respective software ANSYS. After building three-dimensional model of the brake-disks, the models would be simulated using ANSYS under transient thermal analysis and residual stress simulation will be performed.

In order to obtain accurate and reliable results for the simulations, in this case, the meshing of the brake-disks has been properly carried out using ASYS ICEM CFD. Heat transfer coefficient of the ventilated disk would be solved using ASYS CFX.

In ANSYS, simulation tests can be either carried out in two or three dimensions. In this case, three-dimensional simulation techniques shall be studied and used to carry out the various tests in attempt to achieve more results that are accurate. Various simulation methods, such as techniques to input load, boundary conditions and assumption also considered in the study. These would enable a more accurate simulation of the “real-life” conditions that the brake-disks actually experience. After the various tests are carried out, the performance of brake-disk analyzed and compared the results. The ability of the each material of brake-disk to withstand

these loading conditions would be examined, and their weaknesses and strengths have been discussed. Particular attention paid to thermal stress simulation brake-disk where a potential problem may arise.

Conclusion have discussed on the investigating the new potential brake disk material while exploring possible advantages. Finally, in order to achieve a higher accuracy of results and a better understanding of the performance of the brake-disk the recommendation has been suggested to consider various aspects of its design.

1.5 Objective of the study

1.5.1 General objective

The main objective of the present study is using ANSYS software to conduct thermal stress analysis of the disk brake in emergency brake application and an attempt has made further structural analysis carried out by thermal analysis, from analysis result, to suggest the most suitable material, which yields a low temperature variation, less deformation, and minimum von misses stress possible. Moreover, Based on the study of their thermal and mechanical performances, further improvements has recommended to the design of the new brake-disk.

1.5.2 Specific objective

The specific objectives of the present study are:

- To develop, brake disk model and simulation using ANSYS software package.
- To perform, the brake disk fluid flow analysis.
- To analyze the temperature and stress distribution on disk for each of identified materials.
- To investigate the effect of air ventilation on vented brake disk

1.6 Significance of the study

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

Finally, it helps the authority ERC to have documents that is used for the future as a reference for researchers.

1.7 Limitation of the study

In this study, Ethiopian Railway Corporation offered the primary required data of the study. However, it was not in neat and good justification. And less simulation capacity performance of my computer.

1.8 Research method

A lot of paper and journal has been read up and part of it considered in this study. Meanwhile, the actual real brake disk dimension has been taken from ERC specification for analysis purpose. Later, the precise dimensions have been used to generate three-dimensional CAD model using software tool CATIA V5.

- Next, the 3D model of disk brake rotor has been transfer to finite element software, which is ANSYS software. Thermal analysis would be done on transient responses. Assigning the materials properties, load and meshing of the model has done in ANSYS software. Finally, all results of thermal analysis obtained will be compiled in a table.
- The performance of the disk brake would be analyzed based on the ANSYS analysis result of thermal nodal temperature and empirically calculated demand of at each selected stations. So that from the result, this study suggests the suitable material vent disk brake train be for difficult conditions during emergency and/or service brake conditions to keep its performance improve further.

1.9 Organization of the research

The research is organized in six parts. The first chapter is the introduction part, which clearly states the background of the research, statement of problem, and objective, scope and limitation of the study, methodology and benefits of the research. Chapter two, discusses Literature review of the disk brake system of rail vehicle. Similar past studies had been reviewed under this chapter. In addition, Definition, key points, models and analytical approaches has been discussed in this chapter of the study.

Chapter three, covers the data collection and interpretation. Available resources are gathered, functional requirements are isolated and arranged in such a way that they could be used in the data analysis section of the study. In addition, thermal distribution analysis has been represented. Under chapter four, the physical model of train disk brake as well as the material and boundary conditions are discussed in detail. In addition, ASYSYS CFD modeling and prepared meshing result. Using ANSYS-CFX package determination of heat transfer coefficient has been performed and preset result of heat transfer coefficient.

Chapter five, deals about preparing, applying boundary conditions and starting thermal and structural simulation. This is done using numerical approach. Conclusion and recommendation of the research on brake system will be discussed and forwarded in chapter six. This has been the final chapter of the research.

CHAPTER TWO

LITERATURE REVIEW

As we have seen from the previous chapter, one of the most important components in a road vehicle is its braking system. There are three types of mechanical stresses subjected by the disk brake. The first one is the traction force created by the centrifugal effect due to the rotational of the disk brake when the wheel is rotating and no braking force is applied to the disk. During braking operation, there are another two additional forces experienced by the disk brake. Firstly, compression force is created as the result of the force exerted by the brake pad pressing perpendicular onto the surface of the disk to slow it down. Secondly, the braking action due to the rubbing of the brake pad against the surface of the disk brake is translated into frictional or traction force on the disk surface which acts in the opposite direction of the disk rotation.

The frictional heat generated on the interface of the disk and the pads can cause high temperature. Particularly, the temperature may exceed the critical value for a given material, which leads to undesirable effects, such as brake fade, local scoring, thermo elastic instability, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally excited vibration.

Long repetitive braking, such as one which occurs during a mountain descent, will result in a brake fluid temperature rise and may cause brake fluid vaporization. This may be a concern particularly for passenger cars equipped with aluminum calipers and with a limited airflow to the wheel brake systems. Braking performance of a vehicle can be significantly affected by the temperature rise in the brake components. High temperature during braking may cause brake fade, premature wear, brake fluid vaporization, bearing failure, thermal cracks, and thermally excited vibration. Therefore, it is important to predict the temperature rise of a given brake system and assess its thermal performance in the early design stage. Recently, brake fluid vaporization has been suspected as a possible cause of some collisions and a proper inspection procedure has been recommended [12, 13].

Formation of hot spots as well as non-uniform distribution of the contact pressure is an unwanted effect emerging in disk brakes in the course of braking or during engagement of a transmission

clutch. If the sliding velocity is high enough, this effect can become unstable and can result in disk material damage, frictional vibration, wear, etc. [14].

Gao and Lin, (2002), had presented an analytical model for the determination of the contact temperature distribution on the working surface of a brake. To consider the effects of the moving heat source (the pad) with relative sliding speed variation, a transient finite element technique had used to characterize the temperature fields of the solid rotor with appropriate thermal boundary conditions. Numerical results shows that the operating characteristics of the brake exert an essentially influence on the surface temperature distribution and the maximal contact temperature.

Faramaz Talati and Salman Jalalifar, (2009), they had derived the governing heat equations for the disk and the pad are extracted in the form of transient heat equations with heat generation that is dependent to time and space. In the derivation of the heat equations, parameters such as the duration of braking, vehicle velocity, geometries and the dimensions of the brake components, materials of the disk brake rotor and the pad and contact pressure distribution had been taken into account. The problem is solved analytically using Green's function approach. Finally, the study show that the heat generated due to friction between the disk and the pad should be ideally dissipated to the environment to avoid decreasing the friction coefficient between the disk and the pad and to avoid the temperature rise of various brake components and brake fluid vaporization due to excessive heating.

David et al. (2003) and Mc.Phee et al., (2008), carried experimental work emphasizing on determining the characteristics of airflow for a vented brake rotor using Particle Image Velocimetry techniques. He attempted to determine the air velocity in as many areas in and around the rotor as possible. It was noticed that CFD has been applied successfully to disk type vented rotors and hardly any literature available for drum-type vented rotors.

Thilak VMM et al., (2011), conducted a transient thermal and structural analysis of a brake disc to evaluate its performance under severe braking conditions and then to assist in disc rotor design and analysis. The usage of new materials was investigated which aims at improving the braking efficiency and providing greater stability to the vehicle. This study was done using ANSYS 11 software to analyze the temperature distribution, variation of the stresses and deformation across the disk brake profile. The new materials under study were aluminum base metal matrix

composite and High Strength Glass Fiber composites. These materials have a promising friction and wear behavior as a brake disk. The transient thermo elastic analysis of disks in repeated brake applications was performed and the results were compared to that of cast iron disc.

Yildiz and Duzgun, (2010), analyzed the stress of ventilated brake disks by the finite element method, three different ventilated brake disks were designed and manufactured and their performances of braking force were investigated experimentally in addition to those of a full disk. Afterwards, stress analyses were performed by FEA. In these analyses, a different approach, the variable loading on brake pads, was also introduced. It was found that this approach remarkably reduced the stresses on ventilated discs. Hence, this unique approach could be an improvement for eliminating crack formations in ventilated brake discs.

A. Belhocine, A.R. Abu Bakar and M. Bouchetara, (2004), they had presented a numerical simulation of the thermal behavior of a full and ventilated disc in transient state. By means of the computer code ANSYS 11, they were able to study the thermal behavior of a gray cast iron. In addition to the influence of the ventilation of the disk, on the thermal behavior of the discs brake, the numerical simulation shows that radial ventilation plays a very significant role in cooling of the disc in the braking phase; also studied the influence of the braking mode on the thermal behavior of the disk brake. Through the numerical simulation, it could be noted that the quality of the results concerning the temperature field is influenced by several parameters such as:

- Technological parameters illustrated by the design.
- Numerical parameters represented by the number of elements and the step of time.
- Physical parameters expressed by the types of materials.
- Braking mode implemented.

Ali Belchocine & Mostefa Bouchetara, (2012), presented paper on thermal analysis of a solid brake disk **[9]**. The objective of this study is to analyze the thermal behavior of the full and ventilated brake discs of the vehicles using computing code ANSYS. In this analysis approach is to create the model CFD that contains the fields to be studied in ANSYS Workbench. Three different grade of cast iron is chosen (FG 25 AL, FG20, and FG15). The numerical simulation shows that radial ventilation plays a very significant role in cooling of the disc in the braking phase. The variation in temperature between a full and ventilated disc having same material is

about 60 degree at the moment1.8839 s from application of brake The obtained results are very useful for the study of the thermo mechanical behavior of the disc brake (stress, deformations, efficiency and wear).

It can be seen that the literatures focused on the thermal and structural analysis, material property, and disk brake configuration (solid or ventilated disk brake) in order to reduce the exceeding temperature at the frictional contact of the disk brake and brake pad during braking process. Because this temperature primarily cause fatigue, wear, judder, vibration, deformation and stresses. Therefore, the research still proceeds to avoid the problem.

Many research results have confirmed the dominant influence is the thermal loads concerning mechanical loads and residual stresses induced by high thermal loads in brake disks. Therefore, it is important to determine most promising disk material with high precision temperature field of the braking system, as well as to emphasize that high thermal loads, in other words, overloads, of disk very often occur because of long-term braking on straight railroads or unwanted locking of wheels and braking to standstill.

This research is designed to come up with solution by investigating the temperature, structural and thermal analysis of ALMMC, Grey cast iron, Cast steel and carbon-carbon composite made vented brake disk under centrifugal load and thermal load considering airflow cooling and to suggest the best one based on simulation results using ANSYS Mechanical APDL.

CHAPTER THREE

DATA ANALYSIS AND THERMAL DISTRIBUTION ANALYSIS OF DISK BRAKE

In the case of braking operation, frictional heat is dissipated mostly in to pad and disk. An occasional uneven temperature distribution on the components could induce sever thermo elastic distortion of the disk.

Disks are usually damaged in one of four ways: scarring, cracking, warping or excessive rusting. Scarring can occur if brake pads are not changed promptly when they reach the end of their service life and are considered worn out. Cracking is limited mostly to drilled disks, which may develop small cracks around edges of holes drilled near the edge of the disc due to the disc's uneven rate of expansion in severe duty environments. The disks are commonly made from cast iron and a certain amount of what is known, as "surface rust" is normal. Sometimes a loud noise or high-pitched squeal occurs when the brakes are applied. Brake squeal is a phenomenon of dynamic instability that occurs at one or more of the natural frequencies of the brake system. Most brake squeal is produced by vibration (resonance instability) of the brake components, especially the pads and disks (known as force-coupled excitation). This type of squeal should not negatively affect brake-stopping performance [23].

3.1 Disk surface failure due to cracking

The high amount of energy to be handled by the rotor can lead to disk failure. In particular, it is realized that repeated severe brake application under high speed driving may lead to thermal cracking of brake disks due to inelastic cyclic strain accumulation [23]. The reason lays in the large amount of energy that has to be dissipated by the disk. Due to the short braking period, the conduction and convection are not large enough to take it all away. Thus, the surface temperature rises much more rapidly than the main body of the disk and undergoes higher expansion.

If the temperature gradient is steep enough, compressive plastic strain are generated at the surface and while cooling afterwards tensile residual stresses are induced which may be great

enough to cause surface failure in one cycle. If cracking does not occur after one cycle, the fatigue damage caused by repeated cycles of compressive to tensile stress may cause surface rupture in an unacceptably low number of cycles. If the temperature changes at the surface are of sufficient magnitude, heat cracks generally oriented in a radial direction will develop.

3.2 Weight of the disk

Spark ford N.Y., (1998), showed that there is a dependency between the disk thickness and the surface temperature. For instance if the rotor thickness is doubled from 4 mm to 8 mm a decrease of the maximum surface temperature from 600 ℃ down to 300 ℃ could be observed. So surface cracking problems, as described in the previous section, could be retarded by thicker rotors and therefore heavier discs. However, recent trends in especially in automotive industry go more towards lightweight components.

3.3 Improvement on disk brake

Based on calculations and experience, several improvements were made [24]. To help the disk cooling down the heat exchange surface was increased, as in the case of the ventilated disk. Similar to compressors air is sucked through the vanes by the rotational motion of the disk (see Fig. 3.1). This leads to a much higher heat exchange, compared to the single solid disk and therefore lowers surface temperatures. Heat dissipation can further be increased by holes in the disk. Several investigations were done to come up with lighter brake disks. In the present study, ventilated type of disk brake is investigated.

Figure 3.1 Physical model of ventilated disk brake of a train



3.4 Material optimization

Looking at the brake system the rotor disc seems to be the limiting factor concerning lighter systems with equal or better thermal performance. A lot of research has been done in the fields of transient thermal analysis, optimization of the brake disc design, and development of better materials. So far, however, these investigations concentrate on brake discs with a single uniform material. Hence, the structural materials used in brake disc should possess some combination of properties such as good compressive strength, higher friction coefficient, wear resistant, lightweight, good thermal capacity and economically viable [25].

3.5 Promising materials for disk brake

The following section describes the potential candidate materials accustomed to the study (see Table 3.1), based on the critical requirement of lining or disk brake materials tends to close the characteristics listed below:

- It should have high coefficient of friction with minimum fading. In other words, the coefficient of friction should remain constant with change in temperature.
- It should have low wear rate.
- It should have high heat resistance.
- It should have high heat dissipation capacity.
- It should have adequate mechanical strength and not be affected by moisture and oil.

Grey cast Iron: Considering its cost, relative ease of manufacture and thermal stability this cast iron (particularly grey cast iron) is actually a more specialized material for brake applications particularly the material of choice for almost all automotive brake disks.

Aluminum-Metal Matrix Composite (ALMMC): Aluminum alloy based metal matrix composites with ceramic particulate reinforcement have shown great promise for brake rotor applications. These materials having a lower density, higher thermal conductivity and weight reduction of up to 50-60% in brake systems. Wilson Et. Al. studied the abrasive wear resistance of the AA6061 with 20-vol percentage SiC reinforced composite in short sliding distance testing (about 20m). Adding 20 vol. percentage SiC particulate greatly enhanced the wear resistance, raised room temperature strength and stiffness and improved high temperature strength [26].

Carbon carbon-composites: C/C composites exhibit not only high specific strength and modulus for making structural parts at high temperature but also their functional features such as thermal insulation, superconductivity and anti-friction characteristics. The application of C/C composites to the brake disc can increase the braking effect, decrease the weight of brake system, and raise the service life. Therefore, many advanced and commercial vehicles have employed C/C composite brake disks.

Cast steel (G42crmo4): Cast steel used for making disks brake have the following qualities: very good mechanical characteristics at high and low temperature; a good resistance to hot deformation, a good ductility, a good resistance to wear (have very elevated hardness) and a good conduct in service (at thermal fatigue). Therefor cast steel used friction linings with a special geometric configuration (ventilated disc) from very competitive materials in terms of coefficient of friction and the maximum allowable temperature.

3.6 Remarkable materials for brake pad

Carbon/carbon (C/C) composites exhibit not only high specific strength and modulus for making structural parts at high temperature but also their functional features such as thermal insulation, superconductivity and anti-friction characteristics. The application of C/C composites to the brake pad can increase the braking effect, decrease the weight of brake system, and raise the service life. Therefore, many advanced and commercial vehicles have employed C/C composite brake pads as well it remarked for this study.

Table3.1 the potential materials for brake disk and pad

Brake components	Potential materials			
Brake disk	Case 1	Case 1	Case 1	Case 1
	Grey cast Iron	Aluminum-Metal Matrix Composite (AMC 1)	Carbon carbon-composites	Cast steel (G42crmo4)
Brake pad	Carbon carbon-composites	Carbon carbon-composites	Carbon carbon-composites	Carbon carbon-composites

3.7 Determining numerical model of disk brake

In braking process, the temperature field changes input heat flux and heat exchange conditions. The input heat flux is mainly relevant to friction coefficient disc, the angular velocity of brake disc while heat exchange is connected with the friction pair materials and external environmental factors. Analytical model for analyzing thermal and mechanical stress effects in braking systems of railway vehicles is utilized and its adopted procedure is presented in this research [27].

The specific loading condition under this analysis is braking from maximum velocity of 80km/hr. to standstill on a flat track with the initial temperature of the surrounding air is 22°C. In this loading, emergency braking application on a horizontal track to a standstill, the analysis was carried out in 16 seconds. The goal is to find out how the temperature and stress are distributed on the whole construction of the disk by braking to standstill on a straight track. The deceleration factor is 1.4 m/s^2. In this case, the effect of the humidity in the air and the heat transfer with radiation is not considered.

3.7.1 Braking time and angular velocity

A train running with initial velocity (v_0) is supposed to standstill with constant deceleration (a). Its linear translational velocity as function of time (t) is given by:

$$v(t) = v_o - at \quad \text{.....3.1}$$

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

$$\omega(t) = \frac{v(t)}{r_w} \quad \text{.....3.2}$$

$$\omega_o = \frac{v_o}{r_w} \quad \text{.....3.3}$$

Where, v = velocity of the locomotive

v_o = Initial running velocity of the locomotive

ω_o = Initial angular velocity the wheel

$\omega(t)$= angular velocity of the wheel at any time`

r_w = the radius of locomotive wheel

The brake force acts at the effective radius (r_{disk}) of the disc and the translational velocity (v_{disk}) of this frictional force (F_{disc}) could be expressed in terms of the translational velocity $v(t)$.

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

$$S_b = v_o t_b - \frac{1}{2} a t_b^2 \quad \text{.....3.4}$$

$$t_b = \frac{V_0}{a} \quad \text{.....3.5}$$

Hence, angular velocity (ω) at any time (t) can also be determined as follow:

$$\omega = \omega_0 \left(1 - \frac{t}{t_b}\right) \quad \text{.....3.6}$$

Where, t_b = brake time to stop the locomotive

S_b = total brake distance

3.7.2 Braking energy of the train

During the process of braking, available energy that transformed into frictional heat, need to be determined accurately. This study considers the kinetic as well as the potential (while running on slopes) energy of the train that dissipated by the work of the braking. Rail vehicles have important masses in rotation. Therefore, the contribution of rotational kinetic energy is also must be taken in to account [28].

Thereby initial kinetic energy imposed on braking is given by the total sum of translational and rotational energy.

$$E_k = \frac{1}{2} M v_0^2 + \frac{1}{2} I \omega_0^2 \quad \text{.....3.7}$$

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

M = Mass of the train

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 below.

$$I = \sum_{i=1}^{n} m_i r_i^2 \quad \text{.....3.8}$$

Where, m_i -represents the masses of the wheel set and brake disks

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 is the parallel to the axis of the wheel axle. Hence, the rotating radius is equal to the radius of the brake disc (r).

$$E_k = \frac{1}{2}Mv_0^2 + \frac{1}{2}I\omega_0^2 = \frac{1}{2}Mv_0^2\left[1 + \frac{I}{Mr_d^2}\right] = \frac{1}{2}(1.1)Mv_0^2 \quad \text{.....3.9}$$

Considering the contribution of the rotating masses of the wheel set and disks is taken to be 10% the tare weight of the axle load of the locomotive.

The term $[\frac{1}{M(r_w)^2}]$ accounts to the rotational masses involved and its value equals 0.1 [38].Hence the total kinetic energy adapted as follow:

$$E_k = \frac{1}{2}[0.1](1.1)Mv_0^2 \quad \text{.....3.10}$$

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

$$E_p = M.g.S_b\frac{\delta}{1000} \quad \text{.....3.11}$$

Where, g = 9.81m/s^2

Thus, the total available braking energy of the train is determined from the equation below:

$$E_b = \frac{1}{2}MV_0^2[1.1] + M.g.S_b\frac{\delta}{1000} \quad \text{.....3.12}$$

3.7.3 Heat energy

This energy is characterized by a total heating of the disk and the pads during the braking phase. The energy dissipated in the form of heat can generate rise in temperature ranging from 250°C to 800°C [29]. The heat energy generated in contact area is the result of plastic micro deformation generated by the friction forces between disc rotor and brake pad. Assuming the total heat generated (Q_{gen}) in the brake system equals with the total mechanical energy lost from the train. Hence, considering the energy balance that could be equals with the total braking force (E_b).

$$Q_{gen} = E_b \quad \text{.....3.13}$$

3.7.4 Heat flux

Heat flux, which is the total work done on brake system. When a train begins to brake, it lost power and will stop by frictional forces. The imposed mechanical energy theoretically transformed to frictional heat. The brake disc rotor has two friction surfaces ($2A_b$) swept by the brake pads on both sides. Two friction forces ($2F_{disk}$) act on both faces of the disc at an effective radius (r_{disk}). These forces retard the movement of the train.

The total flux is equal to:

$$\text{Total heat flux} = \frac{Q_{gen}}{2A_b} = \frac{E_b}{2A_b} \quad \text{.....3.14}$$

Where, A_b - The area swept by the brake pad at one face of the disk.

E_b- The braking energy during the whole braking time

Generally, the thermal conductivity of material of the brake pad is smaller than the disk ($k_p < k_d$). We consider the heat energy generated in brake system, transferred to the disc and pad. The brake disc assumes the most part of the heat, usually 95% [30]. This ratio normally is called the proportion of heat transferred to disc ($\gamma = 0.95$). Hence the rate of heat generation is:

$$\dot{Q}(t) = \gamma(2F_{disc})v_{disc} = (0.95)(2F)\frac{r_d}{r_w}(v_o - at) \quad \text{.....3.15}$$

In the present work, considering the amount of 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 the transient heat flux can be determined as follow.

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

$$Q_{gen} = E_b = \int_0^{t_b} P(t)dt \quad \text{.....3.16}$$

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

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

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

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

$$F_{disk} = \frac{\frac{1}{2}MV_0^2[1.1]+M.g.S_b\frac{\delta}{1000}}{2\frac{r_{disk}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]} \quad \text{.....3.17}$$

Where, n_b the number of disk brakes

3.7.5 Calculation of boundary conditions for numerical analysis

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

On behave of analytical calculation of the boundary conditions for numerical analysis, it is necessary to determine centrifugal force of the disk, a braking energy of train, a pressure of brake pads and a heat flux into the disc in dependence of braking time.

3.7.6 Determination of physical model of the train vehicle

On application of brakes on the vehicle, inertia forces are set up. The braking force is applied at lesser height whereas the vehicle load is at a higher height. This creates a couple to act on the body of the vehicle which ultimately resulting an additional loading on leading bogie as compared to trailing bogie. Therefore the ratio of weight distribution between the front and rear bogie is considered to be 60/40 in favor of the front bogie. Besides that, the weight distribution of the vehicle is considered. The weight arrangement is 60/40 [31] in the favor of the front part of the carriage. This means that the front part of the carriage takes 60% of the whole load. In this case, only 10 % of the whole brake force is applied to one disc from the forward part of the carriage.

Because of the mentioned weight distribution, only the front part of the carriage is analyzed. Every bogie is consistent of two axles with three brake discs attached to each axle. The kinetic energy for one wheel considering constant deceleration is:

$$\dot{Q}(t) = (2F_{disk})v_{disk} = 2F_{disk}\frac{r_{disk}}{r_{wheel}}(v_o - at) \quad \text{.....3.18}$$

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

According to UIC 541-3 Standard, a brake pad (Carbon/carbon (C/C) composites) and brake disk (Cast iron, aluminum based metal matrix composite, cast steel and carbon carbon-composite), the loading corresponds to the heat flux on the disk surface. The dimensions and the parameters used in the thermal calculation with the possible swept area on the disk by the brake pad and other input parameter data used for the analysis are tabulated in Table 3.2.

Table3.2 Input parameters

Description	Value
Inner disc diameter ,mm	220
Outer disc diameter ,mm	460
Disc thickness ,mm	60
Effective disc radius, mm	205
Rate distribution of the braking forces ϕ, ‰	10
Factor of charge distribution of the disk ε_p	0.5
Track inclination δ , 0	11
Surface disc swept by the pad $A_{c,}$ m m^2,	128890
Wheel diameter, mm	660
Deceleration a, m/s2	1.4
Permissible Line Velocity Vo, km/h	80
Vehicle mass m, kg	64000

The effective radius is the intermediate friction surface radius of the disk r_{disk}= 0.17m. The calculated contact surface area between the disk rotor and braking pad is A_b= 0.12889m^2. However, the friction surface of the pad is a given standard value.

3.7.7 Determination of centrifugal force of the disk

The disc rotating on the rail imposes the centrifugal force on its own due to its weight. Due to the centrifugal load, the pressure/stress is applied to the surface of the disk width, which is

dependent on the density, rotational speed and the radius of the rotating disk. According to the UIC S1002 standard, the radius of the disk could have been 230 mm. The rotational speed of the wheel is calculated from the speed of the vehicle in relation to the radius of the disk. Therefore, the centrifugal force on the wheel is expressed as follow:

$$F_c = m_{disk}\,\omega^2 \quad \text{.....3.19}$$

Where; F_c – centrifugal force

m_{disk}-mass of the brake disk

ω - Angular velocity

R-radius of the brake disk

Hence, Mass of the disk (m_{disk}) = $\rho_{disk} v_{disk}$

Where; ρ_{disk} -density of the disk materials

v_{disk} - Volume of the disk

Since the maximum speed of train is 80Km/hr. and the rotational speed of the disk is to be about 96.6 rad/sec and the volume of the disc is approximately 0.01m3. The density of the candidate materials: Grey cast iron, aluminum based metal matrix composite, cast steel (G42CrMo4) and carbon-carbon composite are 7400kg/m^3, 2765.2kg/m^3, 7850kg/m^3 and 1750kg/m^3 respectively.

3.7.8 Determination of heat flux

According to DIN EN ISO1183 the value μ = 0.32 of the friction coefficient for materials of the disk and the pad for the train is taken as the recommended value. In the present case, emergency braking on a flat track, the value of the friction force which work on the brake disk is calculated below:

$$F_{disk} = \frac{0.12 * \frac{1}{2} M V_0^2 [1.1]}{2\frac{r_{disk}}{r_{wheel}}\left[v_o t_b - \frac{1}{2} a t_b^2\right]} = 8580.848\ \text{N}$$

The instant heat flux entering brake disk:

$$\frac{\dot{Q}(t)}{2A_b} = \frac{\gamma(2F_{disc})v_{disc}}{2A_b} = \frac{(0.95)(2F_d)\frac{r_d}{r_w}(v_o - at)}{2A_b} = 979473.15 - 61712.99t[^{W}/_{m^2}]$$

Hence, Heat $flux_{disc\ at\ t=0} = 1405331.1[^{W}/_{m^2}]$

3.7.9 Determination of the caliper brake pressure

The surface pressure between the disk and pad, on behave of the calculated force applied to the disk need to be determined. Hence the brake caliper pressure that applied on the disc is:

$$P = \frac{F_{disc}}{\mu A_p} = \frac{8580.848}{0.32*.12889} = 208046.784 \text{ Pa.}$$

All the above-calculated values are used throughout the finite element simulations.

3.8 Thermal distribution Analysis

The dissipated energy converted into heat is specified as all the mechanical energy is converted into thermal energy. Energy dissipated as heat between the surfaces and the distributions are equal between the two interacting surfaces. Heat is generated on the surfaces between the rotor and pad when the rotor rotates [36]. This could be expressed as:

$$Q = \mu VP \quad \text{.....1}$$

Where μ is the friction coefficient, V is the sliding velocity of the rotor disk and P is the contact pressure at the interface, Q is the amount of heat generated by friction. For other regions on the rotor and pad exposed to the environment, it is assumed that the heat exchange is transferred through convection process. Therefore, convection surface boundary condition is applied [37]. This can be expressed as:

$$-k\frac{\partial T}{\partial X} = h[T_\infty - T(0,t)] \quad \text{.....2}$$

Where h is convection heat transfer coefficient, T_∞ is atmosphere temperature and $T(0,t)$ is the current temperature of the node.

In addition, the study supposed a transient thermal problem with two boundary conditions:

- a heat flux entering the disk localized in the contact zone disk-pad in both sides
- a heat transfer by convection on all the free surfaces of the disk of which the exchange coefficient h depends on time because rotational speed of the disk varies with time.

CHAPTER FOUR

ANSYS ICEM CFD AND CFX MODELING

ANSYS is software program, created in 1970 in the United States; general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces called elements. The software implements equations that govern the behavior of these elements and solves them all; creating a comprehensive explanation of, how the system acts as a whole. These results then can be presented in tabular or graphical forms. This type of analysis is typically used for the design and optimization of a system far too complex to analyze by hand.

The ultimate purpose of a finite element analysis is to recreate mathematically the behavior of an actual engineering system. In other words, the analysis must be an accurate mathematical model of a physical prototype. In the broadest sense, this model comprises all the nodes, elements, material properties, real constants, boundary conditions, and other features that are used to represent the physical system. In this study, the strategy of numerical approaches is based on the transient thermal problem with initial temperature and two boundary conditions.

4.1. Presentation of computing code ANSYS

ANSYS modules are software programs that implement the finite element method to solve models previously discretized. The modules used for this study are:

- ANSYS Workbench: This platform offers a different approach in the construction of model using the original computer code ANSYS 12. It is particularly adapted to handling cases with complex geometry and to the unconfirmed users;
- ANSYS ICEM CFD: It is mesh generation software for applications in fluid mechanics and mechanical structure.
- ANSYS CFX: This software is designed to perform simulations in fluid mechanics;
- ANSYS Metaphysics: This product contains all modules of ANSYS simulation code.

4.2. ANSYS ICEM CFD modeling and mesh preparation

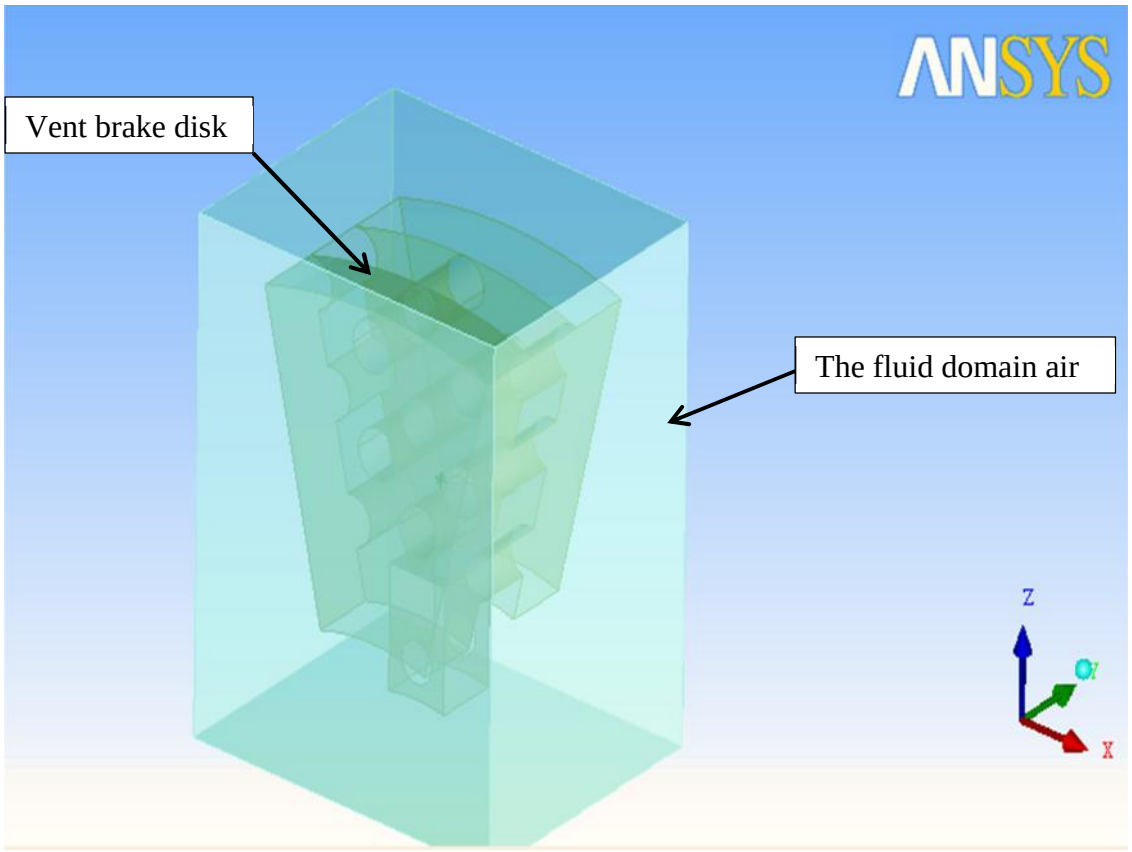
ANSYS ICEM CFD: It is meshing software; used especially in engineering applications such as computational fluid dynamics (CFD) and structural analysis. It provides a direct link between geometry and analysis. In ANSYS ICEM CFD, geometry can be input from just about any format, whether from a commercial CAD design package, third party universal database, scan data or point data. Beginning with a robust geometry module, which supports the creation and modification of surfaces, curves and points, ANSYS ICEM CFD'S open geometry database offers the flexibility to combine geometric information in various formats for mesh generation. The resulting structured or unstructured meshes, topology, inter-domain connectivity and boundary conditions are then stored in a database where they can easily be translated to input files formatted for a particular solver.

In the present study, I have performed the following procedures for the developing partial part of 3D vent disk brake and meshing with ANSYS ICED CFD in workbench using ANSYS 12.

I. Create and import geometry

The three-dimensional model of vent disk rotor and its vane passage chosen for the numerical analysis are shown in figure 4.1, which is one twelve of the full disc brake. The dimensions for the baseline study with an outer diameter 470 mm, inner diameter 220mm, thickness 60mm and 151(length 30mm and diameter 20mm) vanes. The fluid domain consisting of air has been built around the disc rotor and the vanes to study heat convection, because the outside surface of the vanes and the disc, thus interface and connection between the air and the ventilated solid disc on which the heat flow, happened. The domain has represented by rectangle (300*90*130mm). The geometric model has created using CATIA V5 software package and then 'save as' a file type of IGS. The geometry has created in units of millimeters and then scaled to meters at the output step. Come into being ANSYS ICEM CFD in workbench phase by selecting "Geometry" from the option tabs and select import geometry from the menu having a file type of IGS.

Figure 4.1 Imported solid ventilated disk brake (i.e. one twelfths of the full disk) and the fluid domain air.



II. Defining the volume or 'body'

- Select 'Create Body' from the option buttons. The new sidebar 'Create Body' will be opened.
- From within the sidebar select the 'Material Point' button and fill in both the 'Part' and 'Name' as 'VENT DISC'. Leave the default setting of 'Centroid of 2 points' in the 'Location' choice.
- Within the sidebar, click on the 'Select location(s)' button.
- Now, in order the see the points that are to be selected, turn off (unclick) Surfaces under Geometry in the model tree.
- Note that the way I defined this body it is in the center of the FLUID domain and that any two, opposite vertices have been chosen with the same result.
- Now turn on the surfaces again in the model tree.

III. Creating parts

- Now that the full geometry has been developed, it is helpful to assign meaningful names to the different surfaces. These names will be used when applying boundary conditions to surfaces in CFX. Because I will be selecting only surfaces to create parts, uncheck Points, Curves, and Bodies under Geometry in the model tree.
- Assigning names to surfaces is done by creating a part for each surface. In the geometry tree right click the word 'Parts' and select 'Create Part' from the menu that appears. The new sidebar 'Create Part' will be opened.
- Click **Apply** so that **FLUID** appears under **Parts** in the display control tree.
- Once all of the parts are created, press 'Dismiss' in the sidebar.
- Save the geometry file (VENT DISC BRAKE-geometry. Tin). File > Geometry > Save Geometry As...

IV. Blocking the geometry

The blocking strategy for the 3D BRAKE DISC geometry involves creating two blocks from the initial block one each for each half part, forming a rectangle shaped configuration. You need to create an O-Grid to improve the mesh quality. The blocking functionality in ANSYS ICEM CFD provides a projection based mesh generation environment. All block faces between different materials are projected to the closest CAD surfaces. Block faces within the same material may also be associated to specific CAD surfaces to allow for the definition of internal walls.

In general, there is no need to perform any individual face associations to underlying CAD geometry that reduces the time for mesh generation. In the present case I perform the following producers creating the initial blocking, splitting the initial block in sub-blocks in order to have two blocks for vent disc beak and boundary fluid domain Air, projection edges to the curves, to optimize mesh quality, the vertices should be spaced to minimize the average deviation of the edges from the curve. To achieve the most even distribution with this BRAK DISK cylinder case, place the vertices approximately 60 degrees apart (180° half circle/3 edges = 60° per edge).

Finally, I save the blocking (**3D-VENT BRAKE DISC-geometry.blk**).

File > Blocking > Save Blocking As...

The blocking file can be reloaded later, using the **File > Blocking > Open Blocking...**menu item.

V. Generating the mesh

The hex- blocking file (.blk) is different from the ANSYS ICEM CFD geometry file (.tin). You can set the entity mesh parameters at any point before or after blocking. The **Update Sizes** command is a quick and easy way to translate the entity parameters from the geometry to the blocking. Mesh counts are propagated through a mapped mesh. Hence, the smallest size across any index is used.

- Select 'File' from the main menu, then selects 'Blocking' and 'Save Unstruct. Mesh' in the menus that appear.
- Name the file something meaningful like ‘duct’ and press save.
- Select 'File' from the main menu, then select 'Mesh' and 'Load from Blocking' in the menus that appear. Wait a moment while the mesh is generated.
- You can now see your mesh in the view window. When you zoom in there will be a grid around the geometry.

This stage consists in preparing the mesh of the fluid field and Vent brake disc. The final mesh after all steps has shown in the figures 4.2 (a) and 4.2 (b). The finite elements used for the meshing were Free triangular. The total numbers of generated tetrahedral elements in the mesh have been was 727243.

Figure 4.2(a). The final ventilated disk brake mesh and mesh information

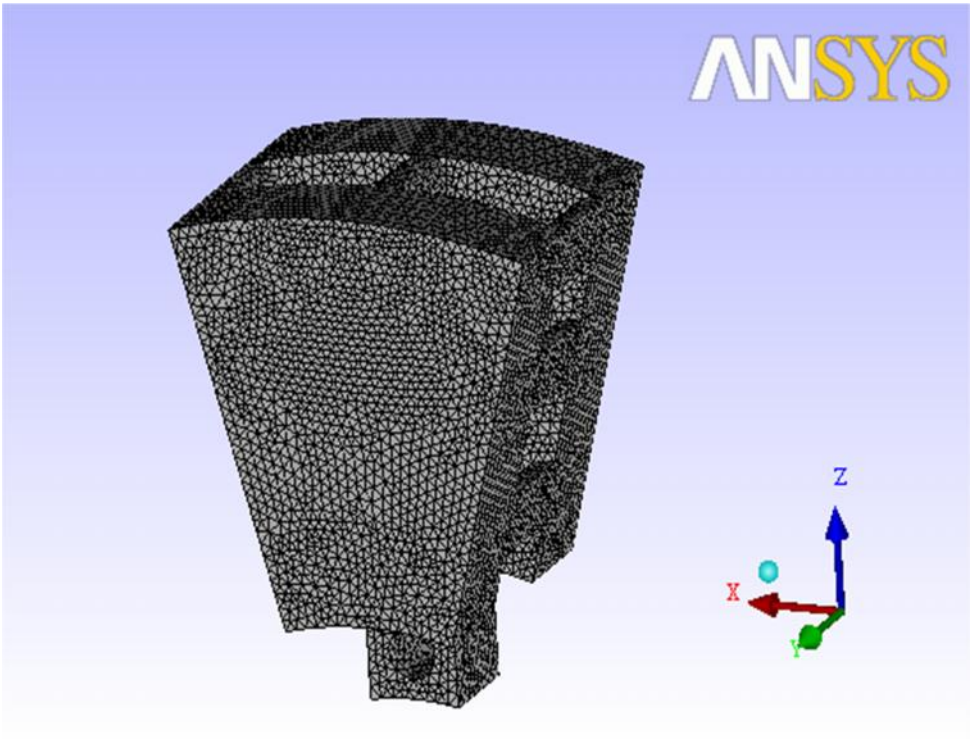
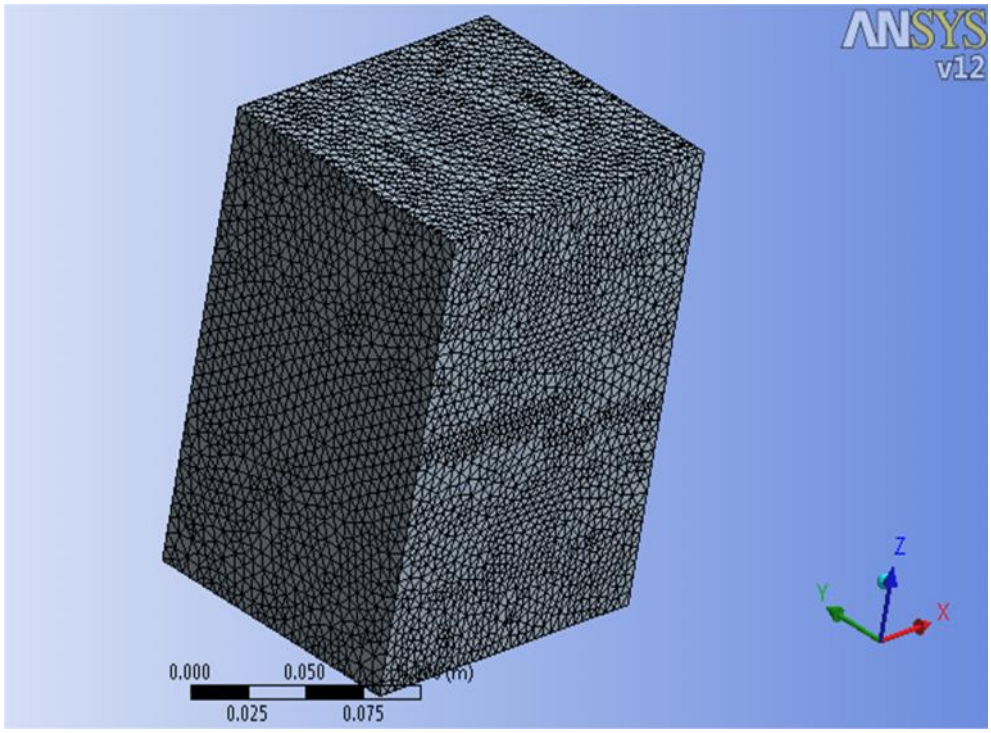


Figure 4.2(b). The final ventilated disk brake and surrounding boundary Air mesh



VI. Checking the mesh and cleaning up

In this sequence, check the mesh for any errors that may cause problems during the analysis. The mesh will be checked for errors and then improved by a Linear-to quadratic transformation. After examining the mesh quality, quality metrics criterion: Skew has been [Min 0.28 Max 1].

VII. Creating the file for CFX

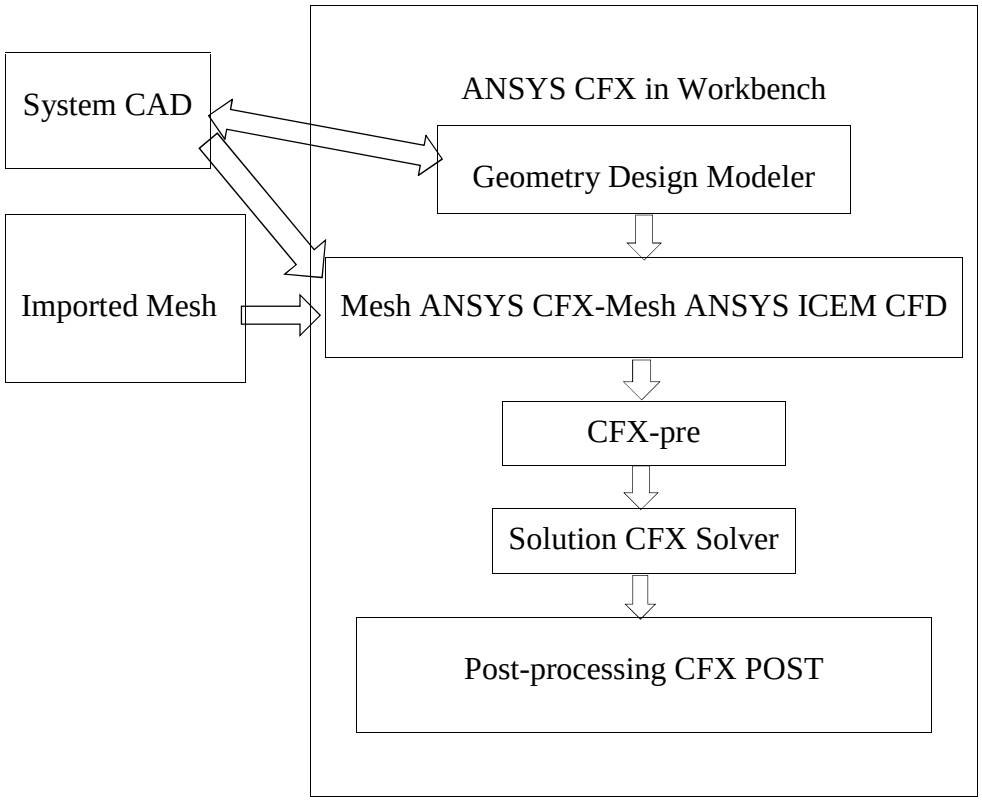
Write the mesh to an input file for CFX Pre

- The final step is to create a CFX readable file from the ICEM mesh. To do this, select 'Output' from the option tabs.
- Select 'Select solver' from the option buttons. The sidebar 'Solver Setup' will be opened.
- In the sidebar, set the 'Output Solver' field to 'ANSYS CFX', the 'Common Structural Solver' field to 'ANSYS'.
- Click 'Apply' and 'Dismiss'.
- Select 'Write input' from the option buttons and finish scaling.

4.3. Modeling with ANSYS CFX

In the present study, the CFD models were constructed and solved using ANSYS-CFX software package [34]. The model applies periodic boundary conditions on the section sides. The first stage is to create the model CFD, which contains the fields to be studied in ANSYS Workbench and then defined the field of the air surrounding this disk. Since, the process of heat transfer by radiation is not too important. I determine using code ANSYS CFX only the convection coefficient (h) of the disc. In addition ANSYS ICEM CFD prepare various surfaces for the two fields in order to facilitate the mesh on which that one will export the results towards CFX using the command "Output to CFX". After obtaining the model on CFX Pre and specified the boundary conditions and define, these physical values come into play on CFX to start calculation. This parameter will be exploited to determine the three-dimensional distribution of the temperature of the disk. Hence, fig. 4.3 illustrates the general simulation stages with ANSYS CFX in workbench.

Figure 4.3 Simulation steps with CFX ANSYS 12



4.3.1. Boundary conditions

Table 4.1 boundary conditions in ANSYS CFX

Boundary	Boundary conditions	Parameters
Inlet	Pressure inlet	Atmospheric pressure and temperature
Outlet	Pressure outlet	Atmospheric pressure and temperature
Domain edge	Symmetry	Symmetry
Disc surface	Wall	298 k

Preparing CFX modeling here are the procedures followed while modeling in **ANSYS CFX.**

a) Physical model

In this step, I declare all of the physical characteristics of the fluid and the solid. After the meshing, are defined all the parameters of the different models to be able to start the analysis.

b) Definition of the domains

Initially, one valid the elaborated models and one activate in the option "Thermal Energy» the calculation of heat transfer «Heat Transfer ".on the outline tree and the domains are loaded as shown in the table.

Table 4.2 Definition of the Solid domain and Fluid domain

Domain - brake disc zone	
Type	Solid
Location	Brake disc
Settings	
Domain Motion	Rotating
Angular Velocity	9.600e+01 [radian s^-1]
Axis Definition	Coordinate Axis
Rotation Axis	Z
Domain - fluid zone	
Type	Fluid
Location	fluid zone
Materials	
Air at 25 C	
Fluid Definition	Material Library(Air)
Morphology	Continuous Fluid

Settings	
Buoyancy Model	Non Buoyant
Domain Motion	Stationary
Reference Pressure	1.0000e+00 [atm.]
Heat Transfer Model	Thermal Energy
Turbulence Model	k epsilon
Turbulent Wall Functions	Scalable

c) Definition of the boundary conditions & application of the interfaces domains

The first step is to select the Inlet and Outlet faces of the heat flux. These options are found in the insertion menu “Boundary Conditions” in the CFX Pre. The boundary conditions concerning the pads will be also defined. One selects the options “Wall” and "Symmetry ", because there will be the possibility of adjusting a certain number of parameters in the boundary conditions such as flux entering the disk.

The areas of interfaces are commonly used to create the connection or linkage areas. Surfaces located between the interactions regions (air disk) are reported as solid-fluid interface and tabulated as follows:

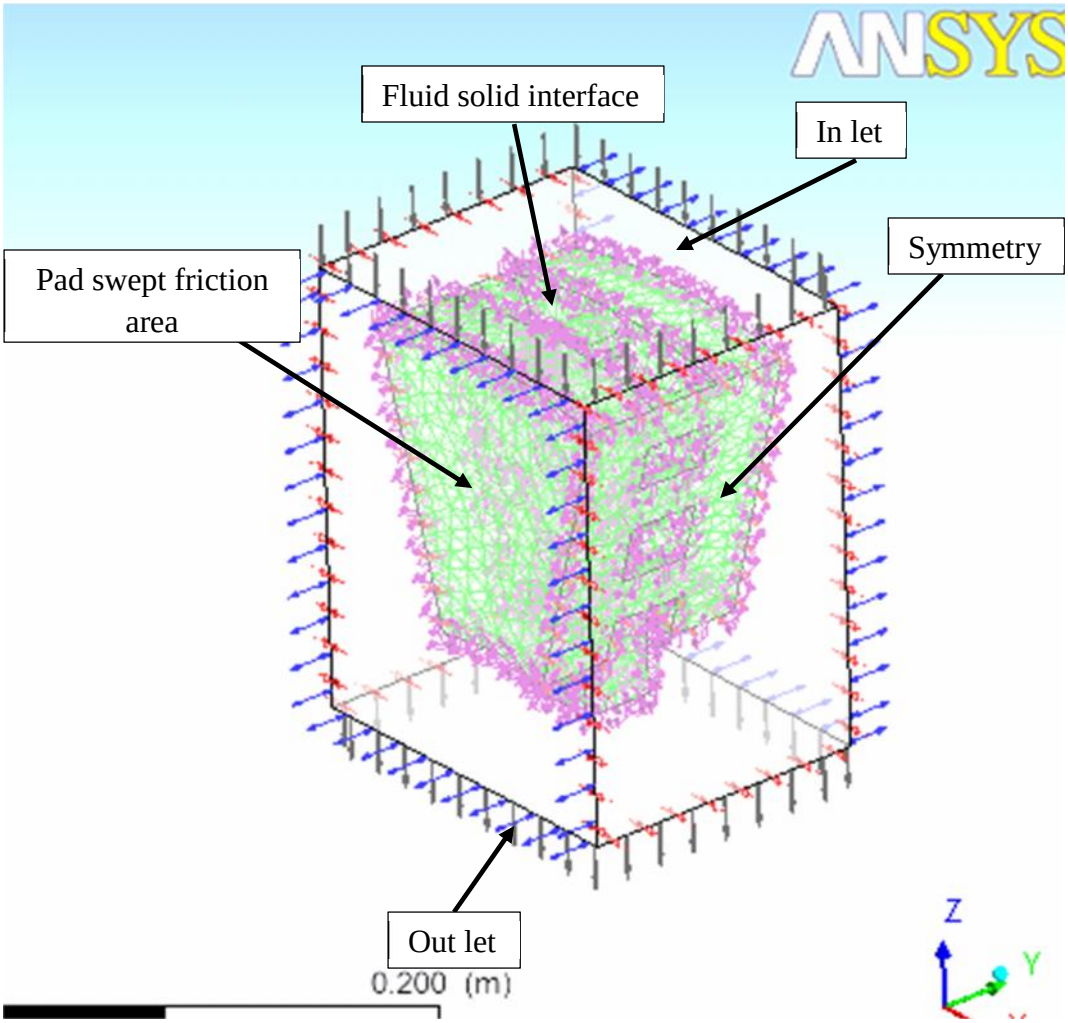
Table 4.3 Boundary Physics for CFX

Domain	**Boundaries**	
Default Domain	**Boundary – radial wall**	
	Type	WALL
	Location	Fixing holes
	Settings	
	Heat Transfer	Adiabatic
	Mass And Momentum	No Slip Wall
	Wall Roughness	Smooth Wall
brake disc zone	**Boundary - fluid solid interface Side 2**	

	Type	INTERFACE
	Location	interface 1 & 2
	Settings	
	Heat Transfer	Conservative Interface Flux
	Boundary - wall heat transfer	
	Type	WALL
	Settings	
	Heat Transfer	Heat Flux
	Heat Flux in	14.05331e+05 [W m^-2]
fluid zone	**Boundary - inlet**	
	Type	INLET
	Location	inflow
	Settings	
	Flow Direction	Zero Gradient
	Flow Regime	Subsonic
	Heat Transfer	Static Temperature
	Static Temperature	2.9800e+02 [K]
	Mass And Momentum	Static Pressure
	Relative Pressure	0.0000e+00 [Pa]
	Turbulence	Zero Gradient
	Boundary - fluid solid interface Side 1	
	Type	INTERFACE
	Location	Air brake disc contacts
	Settings	
	Heat Transfer	Conservative Interface Flux
	Mass And Momentum	No Slip Wall
	Wall Roughness	Smooth Wall
	Boundary - out let	
	Type	OPENING
	Location	outflow
	Settings	

	Flow Direction	Normal to Boundary Condition
	Flow Regime	Subsonic
	Heat Transfer	Opening Temperature
	Opening Temperature	2.7300e+02 [K]
	Mass And Momentum	Opening Pressure and Direction
	Relative Pressure	0.0000e+00 [Pa]
	Turbulence	Medium Intensity and Eddy Viscosity Ratio
	Boundary - wall	
	Type	OPENING
	Location	wall, wall 2
	Settings	
	Flow Direction	Normal to Boundary Condition
	Flow Regime	Subsonic
	Heat Transfer	Opening Temperature
	Opening Temperature	2.9800e+02 [K]
	Mass And Momentum	Opening Pressure and Direction
	Relative Pressure	0.0000e+00 [Pa]
	Turbulence	Medium Intensity and Eddy Viscosity Ratio
	Boundary - symmetry	
	Type	SYMMETRY
	Location	symmetry 1 & 2

Figure 4.4 Brake disk model after all the domain & BCs loaded



d) Temporary condition

Since in this study is to determine the temperature field in a disc brake during the braking phase of a vehicle of average class, we take the following temporal conditions:

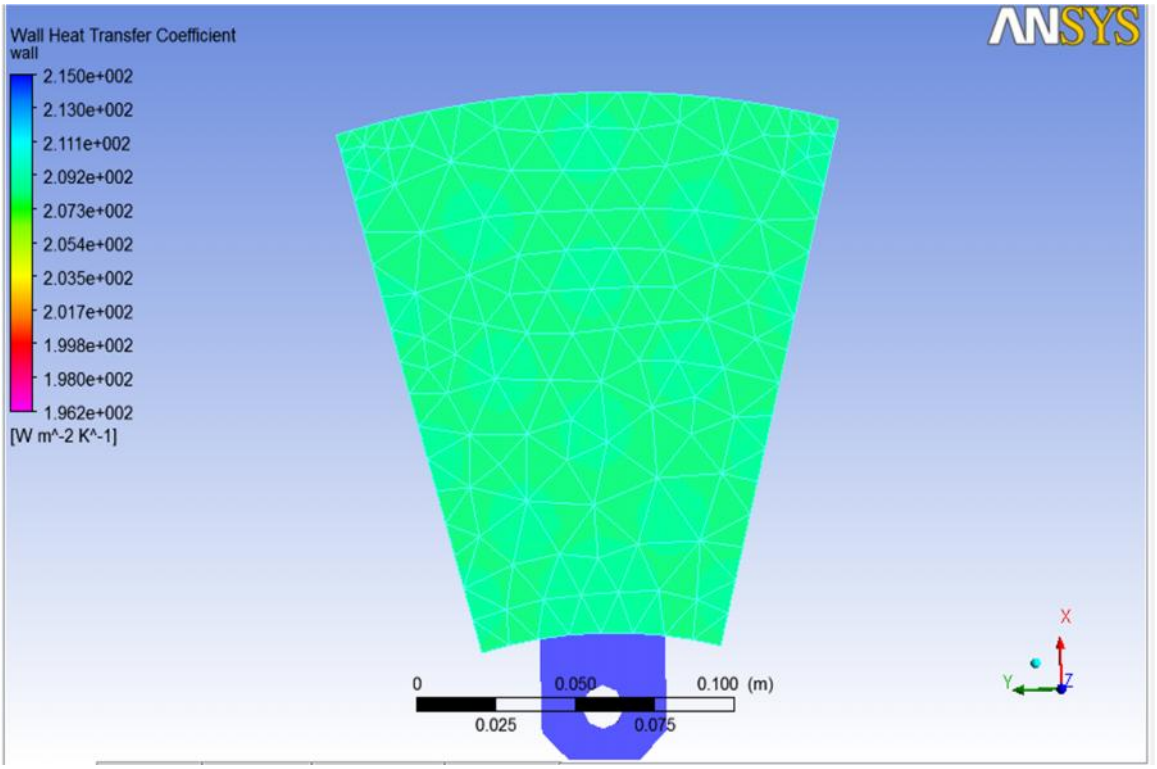
- Braking time= 16 [s]
- Increment time = 1 [s]
- Initial time = 0 [s]

Before starting the calculation and the analysis with ANSYS CFX PRE, it can be ensured that the model does not contain any error. The airflow through and around the brake disc was analyzed using the ANSYS CFX software package. The ANSYS-CFX solver automatically calculates heat transfer coefficient at the wall boundary.

e) The calculation result of the heat transfer coefficient (h)

The heat transfer coefficient is a parameter related to velocity of air and the shape of brake disk, and many other factors. In different velocity of air, the heat transfer coefficient in different parts of brake disk changes with time [32]. Heat transfer coefficient will depend on airflow in the region of brake rotor and vehicle speed; but it does not depend on material. In this simulation, it is determined the value of "Wall heat Transfer Coefficient", variable with time (Figs. 4.5).

Figure 4.5 Wall heat transfer coefficient of brake disk



The airflow through and around the brake disk was analyzed using ANSYS ICEM for meshing & surface preparation and ANSYS CFX for fluid flow simulation as shown on the above figures. The heat transfer coefficients considering convection were calculated and organized in such a way that it could be used as a boundary condition in thermal analysis.

CHAPTER FIVE

FEA ANALYSIS OF VENTILATED DISK BRAKE USING ANSYS

5.1. Introduction to FEA software ANSYS

ANSYS is general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user-designated size) called elements. The software implements equations that govern the behavior of these elements and solves them all; creating a comprehensive explanation of how the, system acts as a whole. These results then can be presented in tabular or graphical forms. This type of analysis is typically used for the design and optimization of a system far too complex to analyze by hand.

5.2. General analysis procedure in ANSYS

Similar to solving problems analytically, it needs to define the solution domain, the physical the model, the boundary conditions and the physical properties. Then solving the problem and present the results. In numerical methods, the main difference is an extra step called mesh generation. This step divides the complex model into small elements that become solvable in an otherwise too complex situation. Below describes the procedure that need to be followed to carry out analysis using ANSYS software.

I. Geometrical definition

There are four different geometric entities in preprocessor namely key points, lines, areas and volumes. These entities can be used to obtain the geometric representation of the structure. All the entities are independent of other and have unique identification labels.

II. Model generation

Two different methods are used to generate a model: Direct generation and Solid modeling. With solid modeling, we can describe the geometric boundaries of the model, establish controls over

the size and desired shape of the elements and then instruct ANSYS program to generate all the nodes and elements automatically.

By contrast, with the direct generation method, we determine the location of every node and size, shape and connectivity of every element prior to defining these entities in the ANSYS model. Although, some automatic data generation is possible (by using commands such as FILL, NGEN, EGEN etc.) the direct generation method essentially a hands on numerical method that requires us to keep track of all the node numbers as we develop the finite element mesh. This detailed bookkeeping can become difficult for large models, giving scope for modeling errors. Solid modeling is usually more powerful and versatile than direct generation and is commonly preferred method of generating a model.

III. Define material properties

All elements are defined by nodes, which have only their location defined. In the case of plate and shell elements, there is no indication of thickness. This thickness can be given as element property. Property tables for a particular property set have to be input. Different types of elements have different properties. For instance, Beams: Cross sectional area, moment of inertia etc., Shells Thickness Springs: Stiffness Solids: None.

The user also needs to define material properties of the elements. For linear static analysis, modules of elasticity and Poisson's ratio need to be provided. For heat transfer, coefficient of thermal expansion, densities etc. are required. They can be given to the elements by the material property set.

IV. Generate mesh

In the finite element analysis, the basic concept is to analyze the structure, which is an assemblage of discrete pieces called elements, which are connected, together at a finite number of points called Nodes. Loading boundary conditions are then applied to these elements and nodes. A network of these elements is known as Mesh.

V. Finite element generation

The maximum amount of time in a finite element analysis is spent on generating elements and nodal data. Preprocessor allows the user to generate nodes and elements automatically at the

same time allowing control over size and number of elements. There are various types of elements that can be mapped or generated on various geometric entities. The elements developed by various automatic element generation capabilities of preprocessor can be checked element characteristics that may need to be verified before the finite element analysis for connectivity, distortion-index, etc. Generally, automatic mesh generating capabilities of preprocessor are used rather than defining the nodes individually. If required, nodes can be defined easily by defining the allocations or by translating the existing nodes. In addition, one can plot, delete, or search nodes.

VI. Boundary conditions and loading

After completion of the finite element model, it has to constrain and load has to be applied to the model. User can define constraints and loads in various ways. All constraints and loads are assigned set 1D. This helps the user to keep track of load cases.

VII. Solution

The solution phase deals with the solution of the problem according to the problem definitions. All the tedious work of formulating and assembling of matrices are done by the computer and finally displacements and stress values are given as output. Some of the capabilities of the ANSYS are linear static analysis, non-linear static analysis, transient dynamic analysis, etc.

VIII. Post-processor

It is a powerful user-friendly post-processing program using interactive color graphics. It has extensive plotting features for displaying the results obtained from the finite element analysis. One picture of the analysis results (i.e. the results in a visual form) can often reveal in seconds what would take an engineer hour to asses from a numerical output, say in tabular form. The engineer may also see the important aspects of the results that could be easily missed in a stack of numerical data. Employing state of image enhancement techniques, facilities viewing of:

- Contours of stresses, displacements, temperatures, etc.
- Deform geometric plots
- Animated deformed shapes
- Time-history plots

- Solid sectioning
- Hidden line plot
- Light source shaded plot
- Boundary line plot etc.

The entire range of post processing options of different types of analysis can be accessed through the command/ menu mode there by giving the user added flexibility and convenience.

5.3. Analysis of train ventilated disk brake using ANSYS 12.1

The analysis is carried out for three-dimensional solid of the disc (Fig. 5.1) (the disk is supposed to be symmetrical). In the zone of temporary contact of the pad and disc, the thermal flux is assigned, which differs in the area of disc at any instant of braking time corresponding to the components of the intensity of heat flux product. The numerical modeling of thermal effects was performed according to the assumptions described before in **ANSYS 12.1 Mechanical APDL (ANSYS)**.

5.3.1. Considerable assumption

It is very difficult exactly to model the brake disc in which there are still undergoing researches on to find out transient thermal behavior of disc brake during braking applications. There is always a need of some assumptions to model any complex geometry. These assumptions are made keeping in mind the difficulties involved in the theoretical calculation and the importance of the parameters that are taken and those, which are ignored. In modeling, always ignore the things that are of less importance and have little impact on the analysis. To simplify analyzing the model several assumptions have performed [33] as follow:

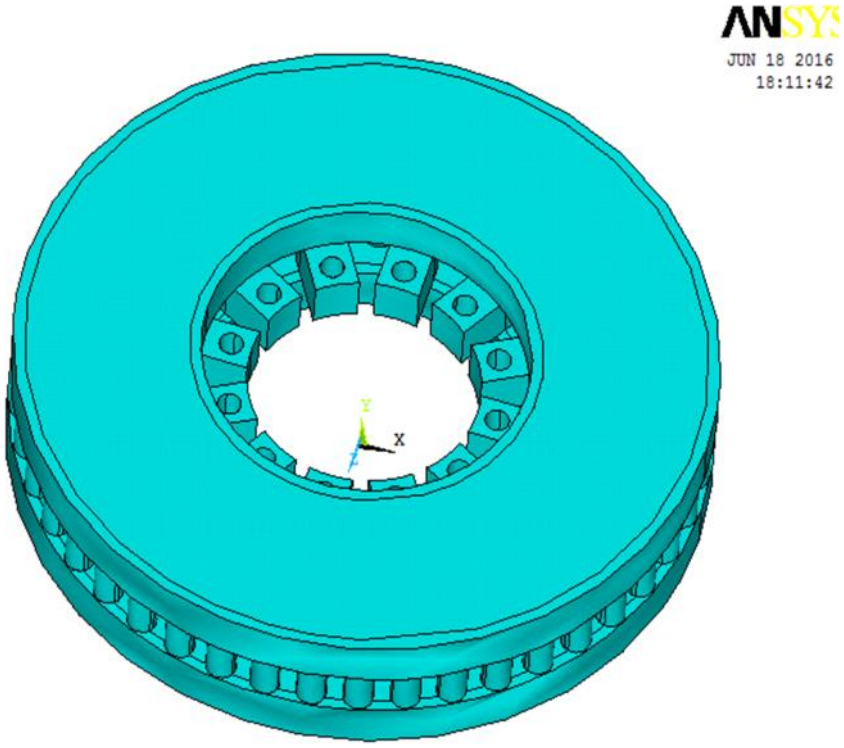
- The nominal surface of contact between the disc brake and the pad in operation is equal to the apparent surface in the sliding motion.
- The contact pressure is uniformly distributed over all friction surfaces hence the heat generation of the mid plane is considered as symmetric.
- The average of the intensity of heat flux into disc on the contact area is equal.
- Radiation is neglected by virtue of short braking time and hence relatively low temperature.

- Heat transfer through conduction is also neglected.
- The wear on the contact surface is neglected.
- Displacement in axial direction on flange is constrained in one side of the disc.
- Brake is applied on the entire railway vehicle.
- The brake disc is stress free before the application of the brake.
- Braking response time is neglected.
- The thermal conductivity of the material is constant throughout; it does not change with temperature.
- The specific heat of the material is constant throughout; it does not change with temperature.

5.3.2. Geometrical model

In the three-dimensional model of solid disc, single surface of its symmetry in axial direction is insulated owing nature of considered phenomenon of heating. The disk is screwed on the hub. It has twelve holes to fasten bolts is illustrated in fig 5.1 below.

Figure 5.1 Geometric Models of ventilated disk brake



5.3.3. Candidate materials definition

Due to the application of brakes on the disk brake rotor, heat generation takes place due to friction; this thermal flux has to be conducted and dispersed on the disk rotor cross section (disk wall). The condition of braking is very much severe and thus the transient thermal analysis has to be carried out. The study conducted thermal loading as well as the structure is axis-symmetric. In addition, I have performed a three dimensional analysis, which is an exact representation for thermal analysis. In this study, the transient thermal and mechanical analyses have performed for analyzing the stability of the structure. The candidate materials, whose thermo-mechanical properties of disc rotor and brake pad having an isometric elastic behavior adapted in transient thermal analysis are tabulated in Table 5.1.

Table 5.1 Candidate materials property of ventilated disk brake and brake pad

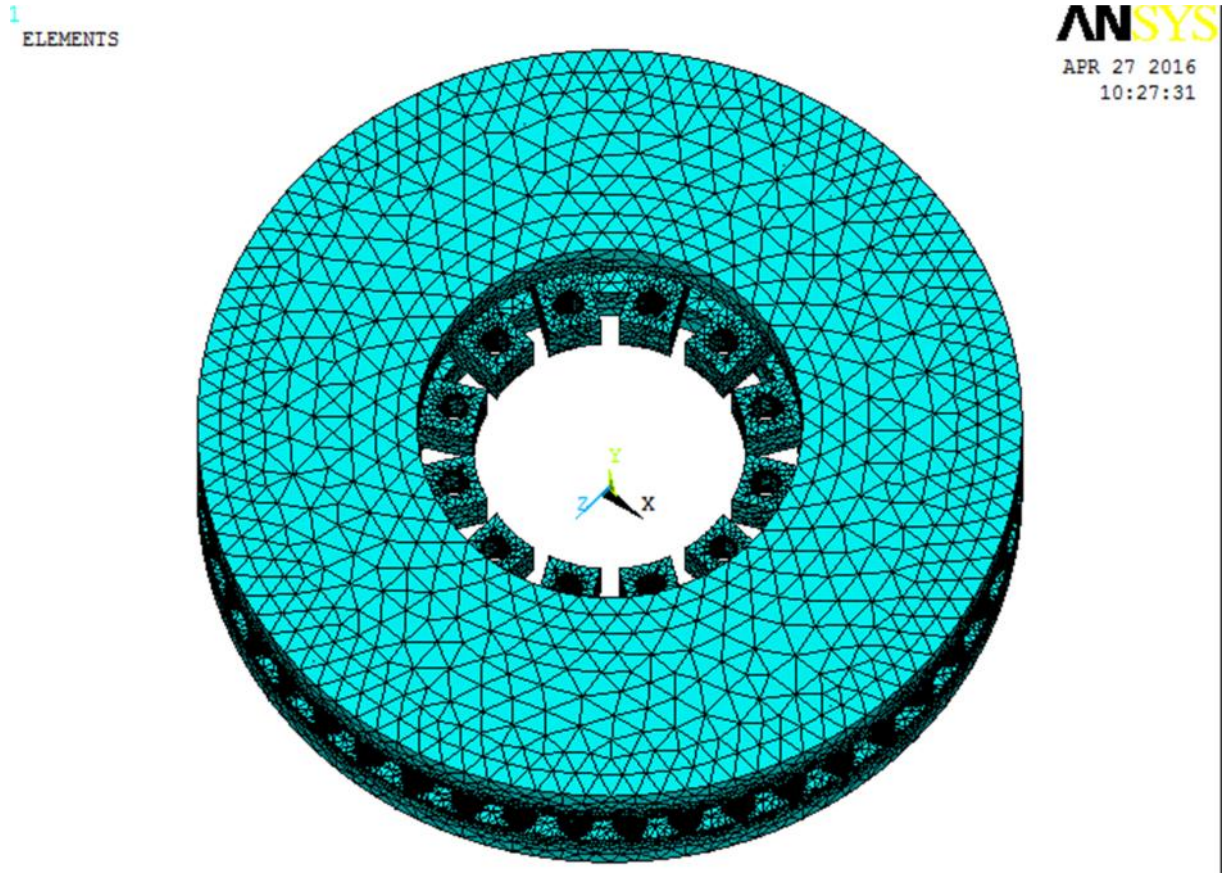
Candidate Materials Properties	Brake disk materials				Brake pad material
	Grey cast iron	ALMMC	Carbon carbon-composite	Cast steel (G42crmo4)	Carbon carbon-composite
Density, ρ [kg/m3]	7400	2765	1750	7850	1750
Thermal conductivity k, [W/mk]	55	181.65	5	45.5	5
Specific heat capacity $C_{P,}$ [J/kgk]	0.46	836.65	1000	461	1000
Poisson's ratio, *v*	0.211	0.33	0.17	0.3	0.17
Modulus of elasticity of material E, [GPa]	152	98.5	40	212	40
Linear expansion coefficient (K-1)	10.85	17.5	0.7	11.5	0.7

5.3.4. Meshing

In the finite element analysis, the basic concept is to analyze the structure, which is an assemblage of discrete pieces called elements, which are connected, together at a finite number of points called nodes. A network of these elements is known as a mesh.

For this analysis, automatic mesh generating capabilities were used in ANSYS rather than defining the nodes individually (see fig. 5.2). The finite elements used for the meshing were Free triangular.

Figure 5.2 Meshed Model of ventilated disk brake



5.3.5. Definition of the loads and boundary conditions

After completion of the finite element model, it was necessary to apply constraints and loads to the model for each case; thermal analysis (Fig. 5.3) and structural analysis (Fig. 5.4).

A. In a case of thermal analysis, the train-ventilated disk brake subjected to the following loads and boundary conditions.

- The initial temperature of the disk and the pads is 22°C.
- The surface convection condition is applied at all surfaces of the disk with the values of the convection coefficient (h) of 209 W/m2.k obtained from ANSYS CFX simulation in chapter 4.
- The heat flux into the brake disk during braking is 1405331 W/m2 that has been obtained the value in section 3.2.7.
- Total emergency braking time used in the analysis is 16 seconds.

B. In a case of structural analysis, the train-ventilated disk brake subjected to the following loads and boundary conditions.

- The pressure on the rubbing contact surfaces of the disk and brake pad of value 208047pa.
- The initial temperature of the disk and the pads is 22°C.
- Considering centrifugal force as inertia applied in **ANSYS 12.1 Mechanical APDL (ANSYS)**.
- Angular velocity of the disk rotor is given 96 rad per second.
- Initially the translational deceleration of the rotor disk is 1.4 meter per Second Square.
- Gravitational acceleration; in negative z-direction
- Fixed support is applied at the holes where the bolts are located and by assuming cylindrical support at the inner surface of the hub.
- Total emergency braking time used in the analysis is 16 seconds.

Figure 5.3 thermal boundary conditions and constrains

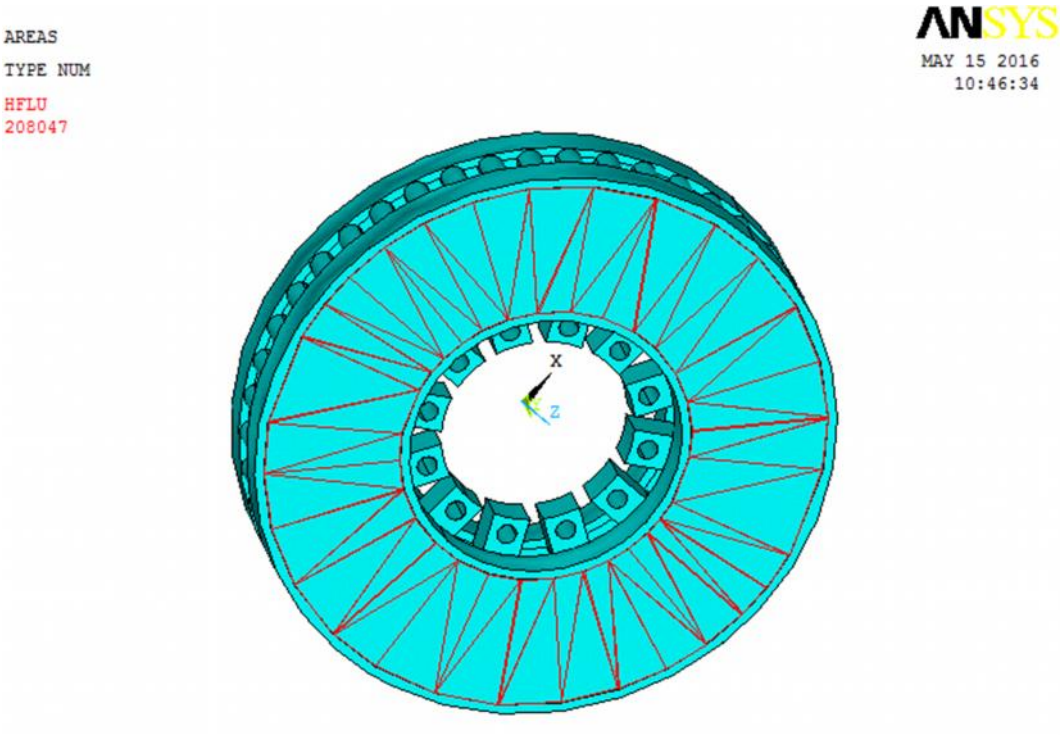
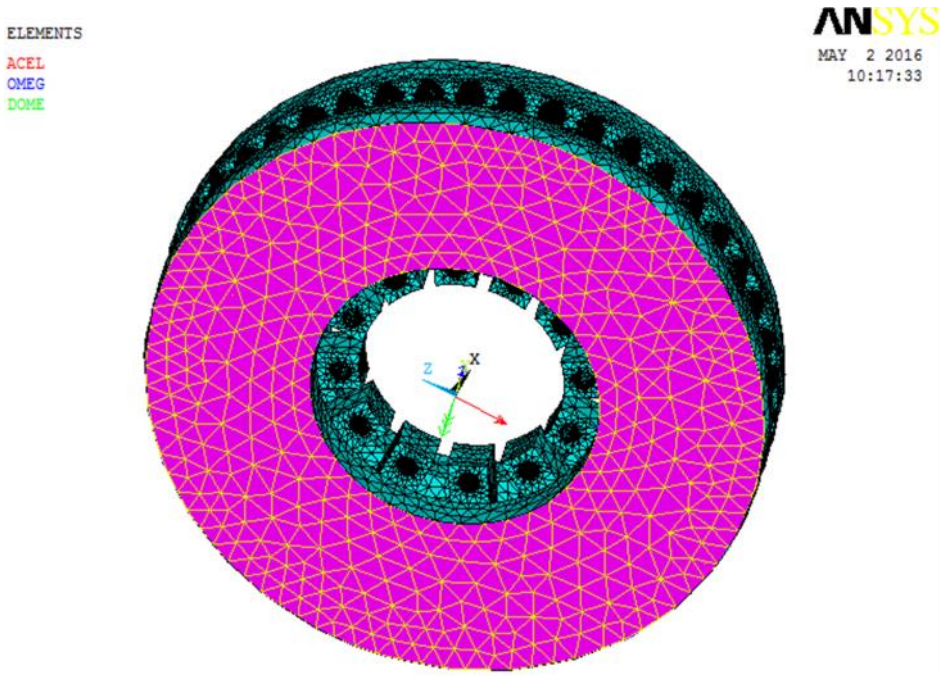


Figure 5.4 structural boundary conditions and constrains



5.3.6. Solution

In the solution procedure, frontal solver is used.

- It involves after the individual element matrices are calculated, the solver reads in the degree of freedom (DOF) for the first element.
- The program eliminates any degrees of freedom that can be expressed in terms of the other DOF by writing an equation to the .**Tri file.** This process repeats for all the elements until all the degree of freedom have been eliminated and a complete triangular zed matrix is left on the .**Tri file.**
- The term frequently used is the frontal solver is wave front. The wave front is the number of degrees of freedom retained by the solver while triangularization of the matrix.
- The nodal solution plots the temperature distribution in thermal analysis.
- Graph of the temperature variation with respect to the time.
- The nodal solution plots stress distribution and deformation in structural analysis.

CHAPTER SIX

RESULT DISCUSSION AND CONCLUSION

To investigate the transient thermo elastic analysis behavior of the four materials of the disk brake, the ANSYS simulation is obtained continuous brake applications using the method general post-processor. The four materials properties are actually considered into ANSYS per-processor.

In the present study, under Emergency braking conditions, the initial braking speed is 80Km/h and the entire braking process lasted 16 seconds, in order to stop the locomotive. **ANSYS 12.1 Mechanical APDL (ANSYS)** compute the temperature and structural stress field of brake disk at any time. The time dependent solver is used for solving the thermal and structural results obtained from ANSYS are deformation, Von-Mises stress and temperature gradient for all the candidate materials. The values are summarized and presented in simulation results below.

6.1 Simulation results

The thermal and structural results obtained from ANSYS software package are temperature gradient, deformation and Von-Mises stress. The values are summarized and presented below for each defined material; ALMMC, Grey cast iron, cast steel (G42crmo4) and Carbon-Carbon composite respectively.

6.1.1. Thermal results

Thermal Gradient - A temperature gradient is a physical quantity that describes in which direction and at what rate the temperature changes the most rapidly around a particular location. In this case, the results show that in case of emergency braking on the flat track.

I. Figures 6.1 (a) and (b) show the temperature gradient obtained throughout the simulation in a case of ALMMC brake disk

Figure 6.1(a) conventional brake disk of ALMM

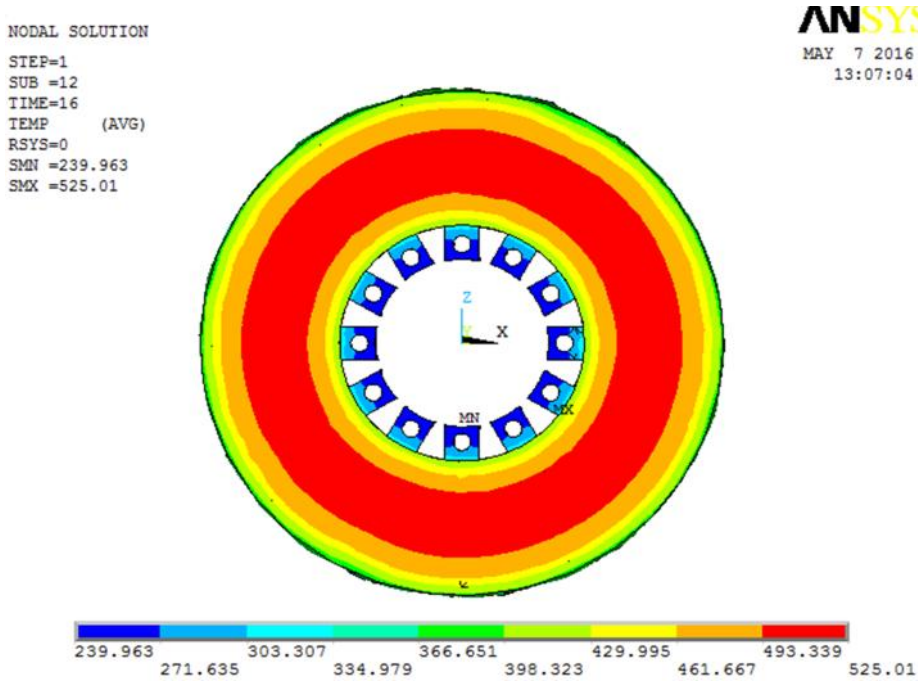
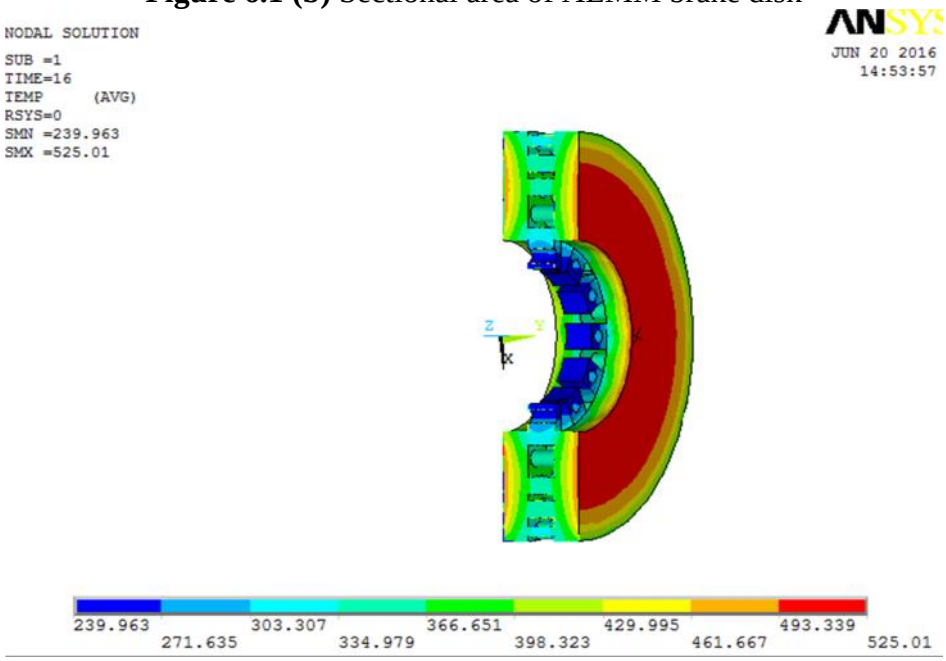


Figure 6.1 (b) Sectional area of ALMM brake disk



Figures 6.1 show that, during the braking process, the maximum temperature of the disk is 525.01 k in the middle of the friction surface and the minimum temperature is 239.963 k.

II. Figures 6.2 (a) and (b) show the temperature gradient obtained throughout the simulation in a case of grey cast iron brake disk

Figure 6.2 (a) conventional brake disk of Grey cast iron

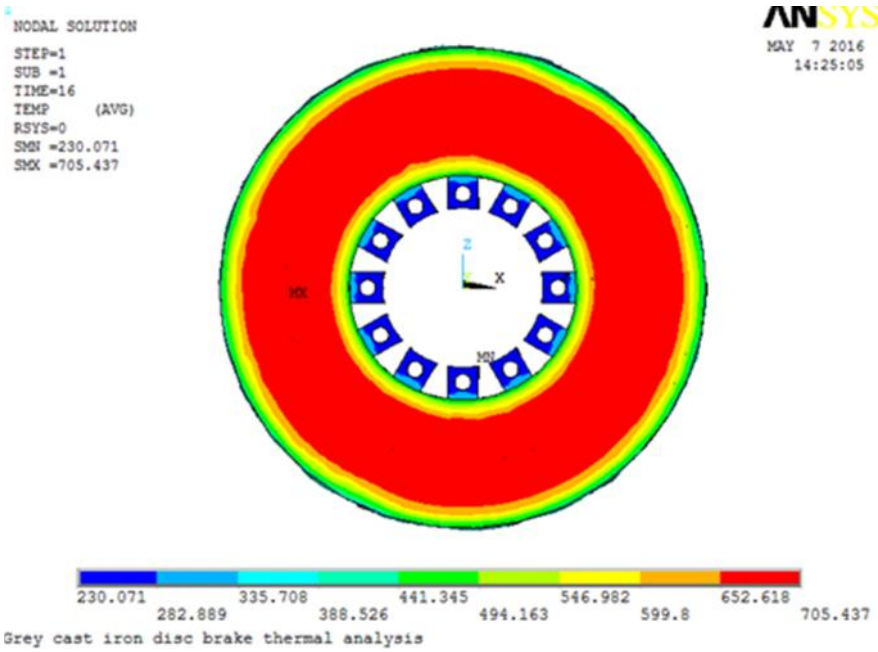
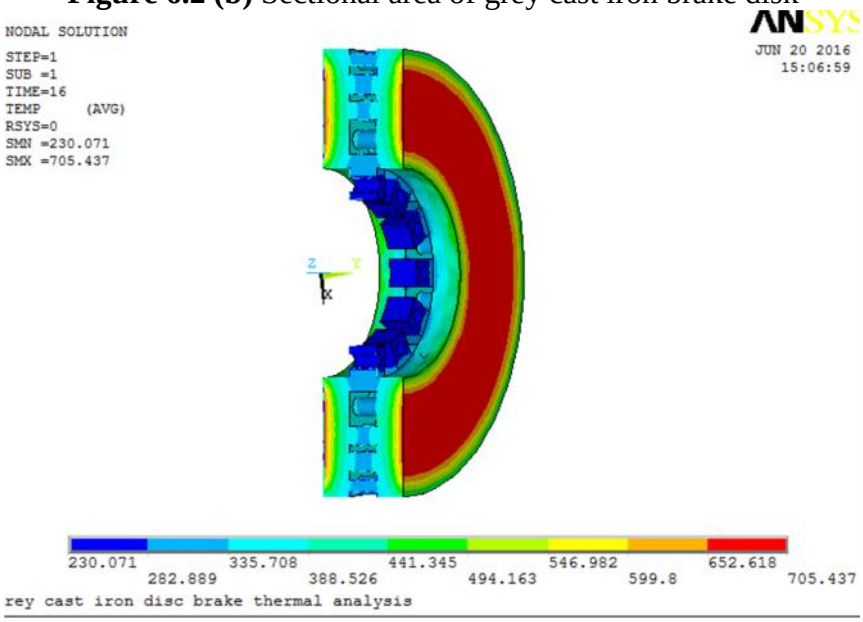


Figure 6.2 (b) Sectional area of grey cast iron brake disk



Figures 6.2 show that, the maximum temperature of the disk is 705.437 k covers wide area in the middle of the friction surface and the minimum temperature is 230.071 k.

III. Figures 6.3 (a) and (b) show the temperature gradient obtained throughout the simulation in a case of cast steel (g42crmo4) disk

Figure 6.3 (a) conventional brake disk of Cast steel

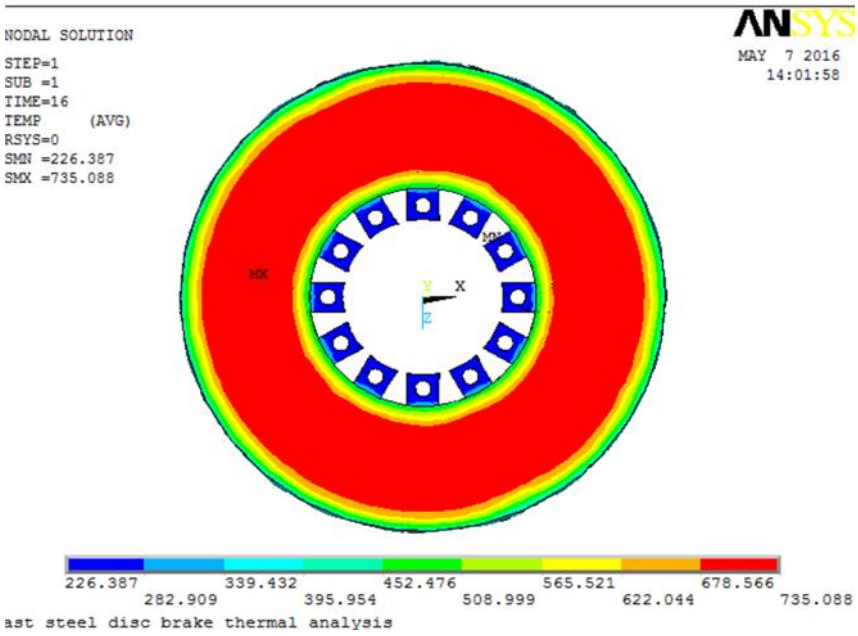
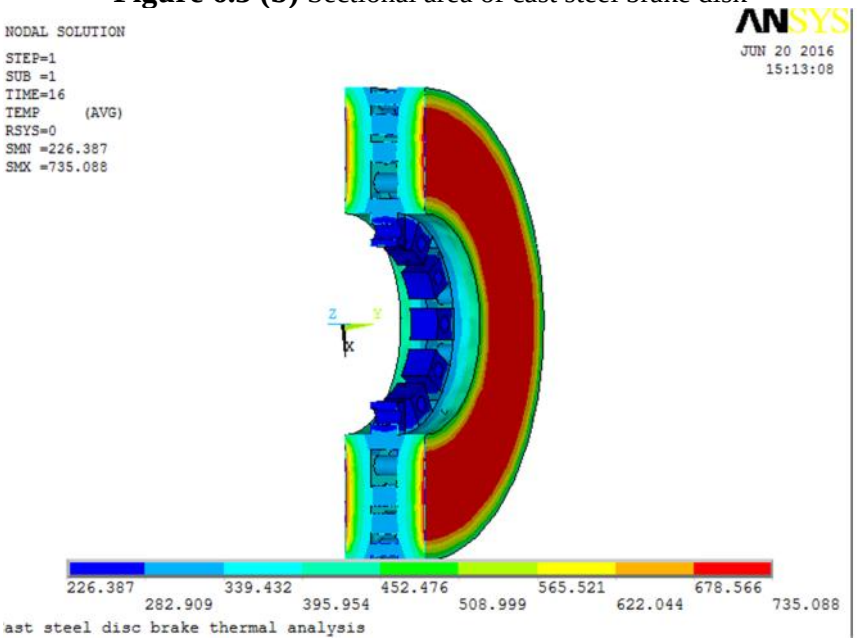


Figure 6.3 (b) Sectional area of cast steel brake disk



Figures 6.3 show that, the maximum temperature of the disk is 735.088 k wide area in the middle of the friction surface and the minimum temperature is 226.387 k.

IV. Figures 6.4 (a) and (b) show the temperature gradient obtained throughout the simulation in a case of carbon-carbon composite disk

Figure 6.4 (a) conventional brake disk of Carbon-carbon composite

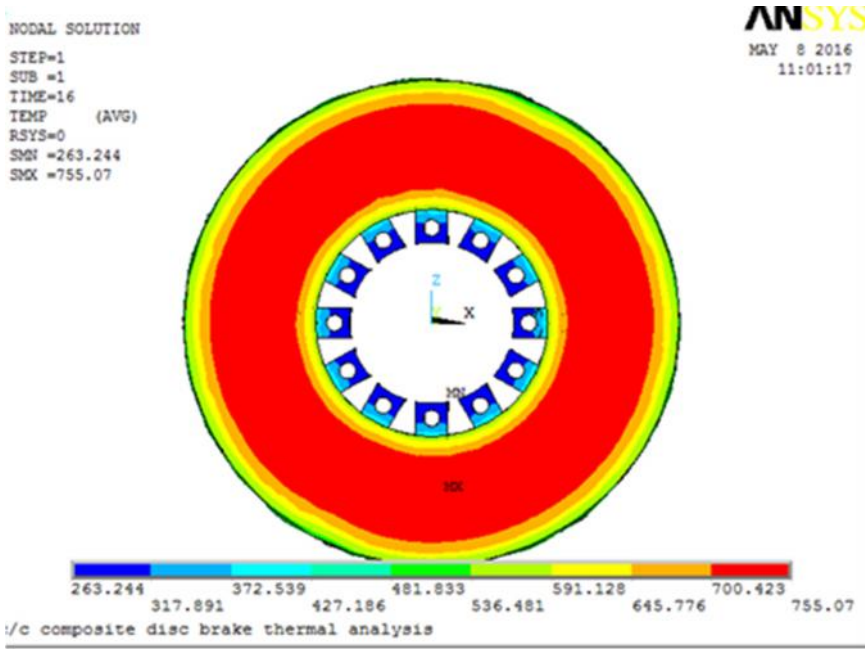
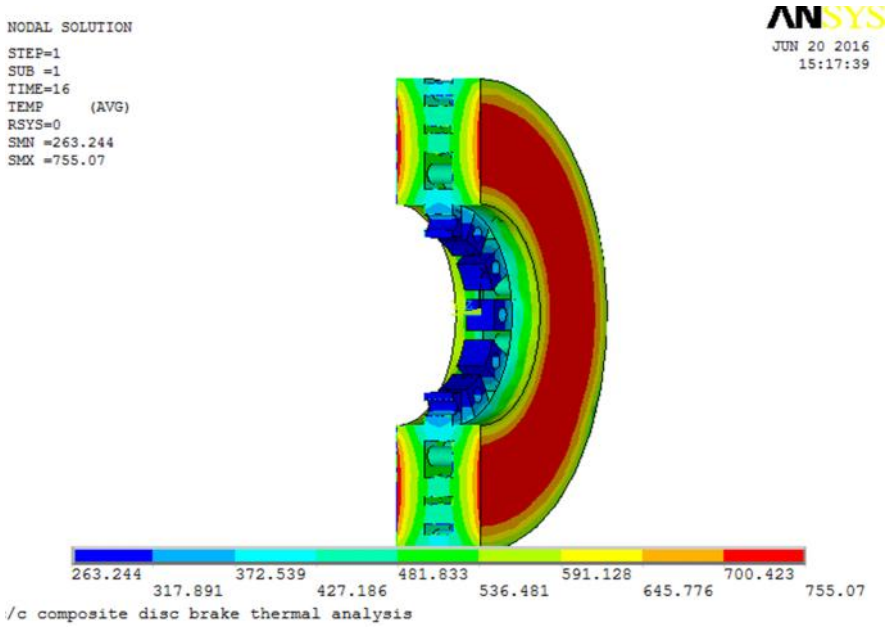


Figure 6.4 (b) Sectional area brake disk of Carbon-carbon composite



Figures 6.4 show that, maximum temperature of the disk is 755.07 k cover wide area in the middle of the friction surface and the minimum temperature is 263.244 k.

6.1.2 Structural results

Deformation - A vector quantity, which refers to the distance which an object has moved in a given direction. It is measured as the length of a straight line between the initial and final positions of a body.

Von-Mises (Equivalent) Stress - The Von-Mises criteria is a formula for combining directional stresses into an equivalent stress, which is then compared to the tensile stress of the material.

I. Figures 6.5, (a) and (b), show the total deformation (c) and (d) von-mises stress obtained throughout the simulation in a case of ALMMC disk

Figure 6.5 (a) conventional disk brake of ALMMC

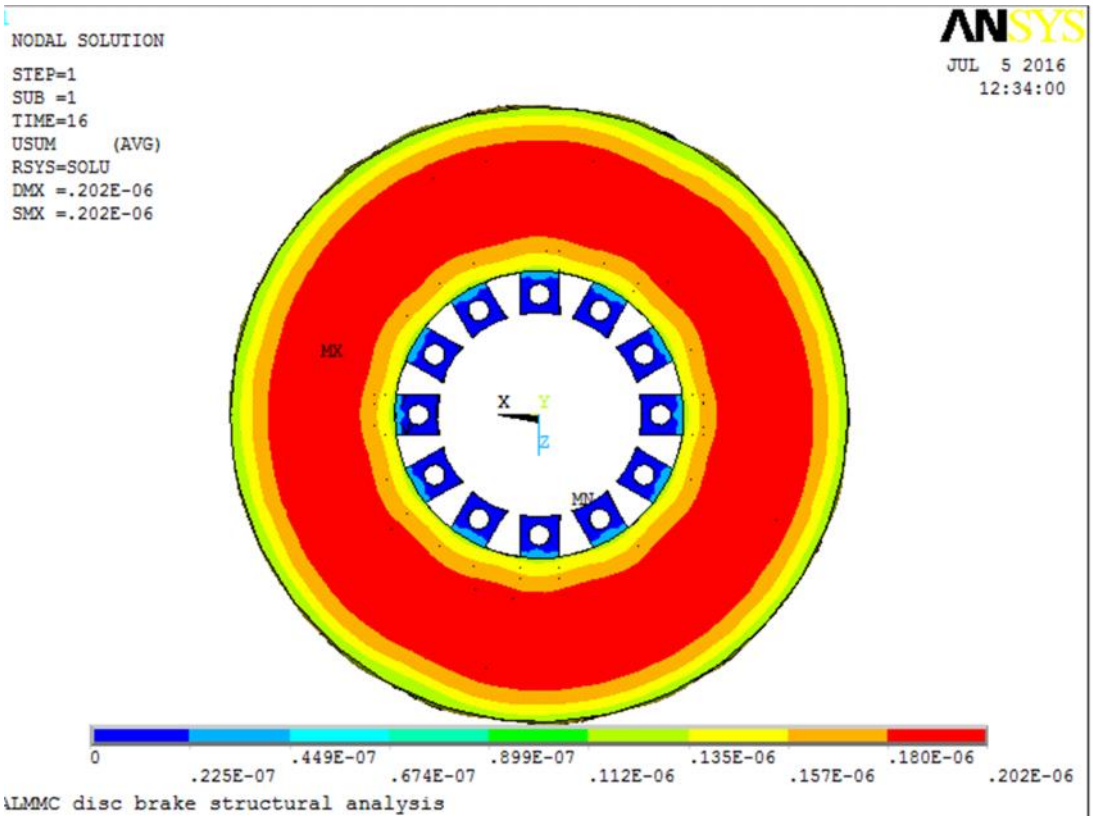
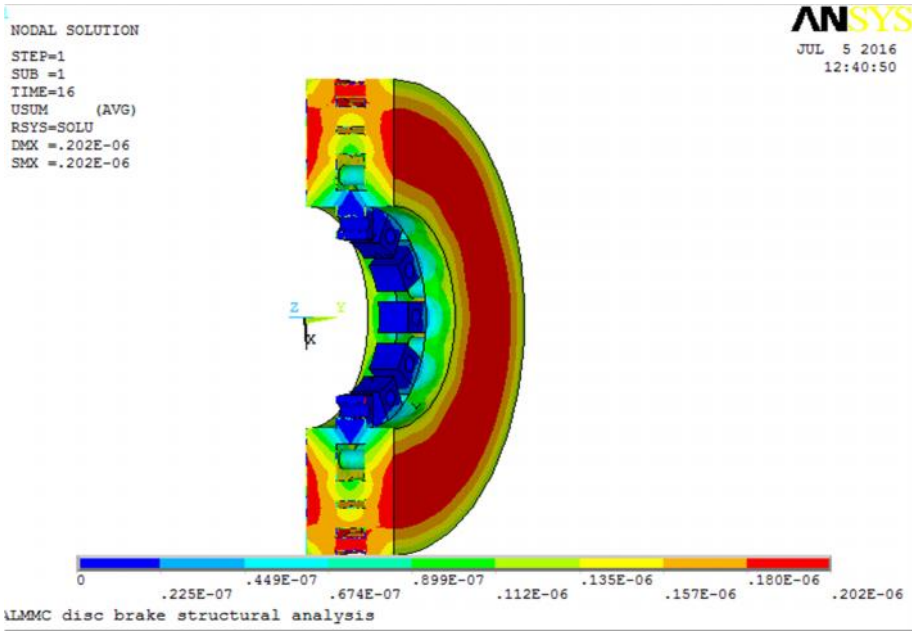


Figure 6.5 (b) sectional area of ALMMC disk



Figures 6.5 (a) and (b) show that, the maximum deformation of the disk is 0.202E-6 m appeared in the middle of the friction surface and ventilation vanes of the disk.

Figure 6.5 (c) conventional disk brake of ALMMC

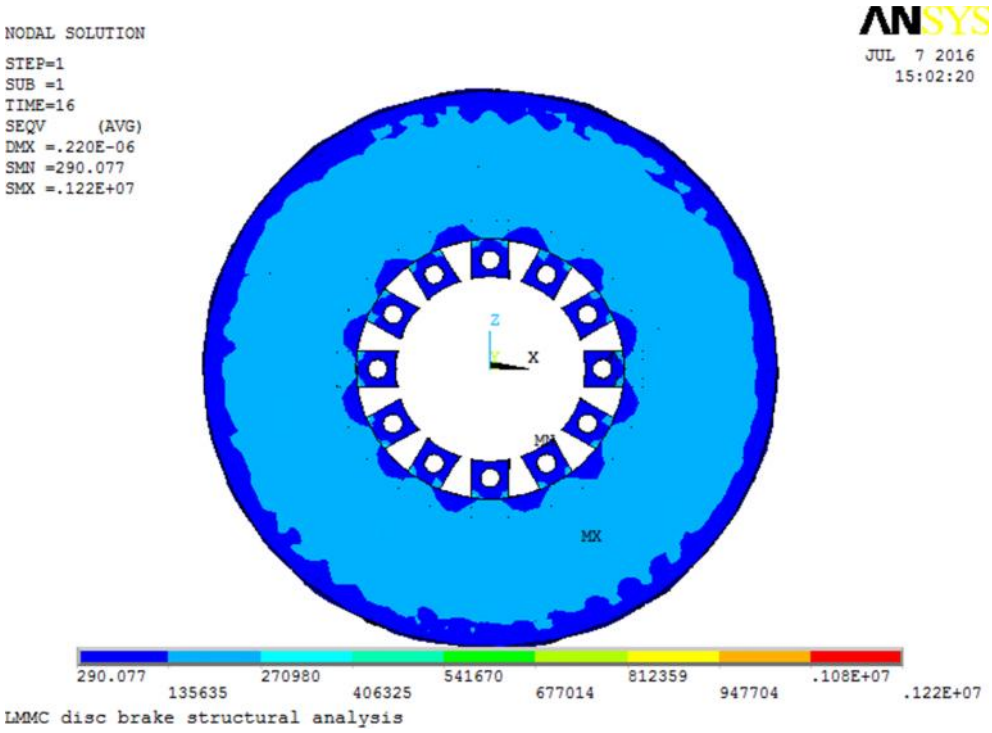
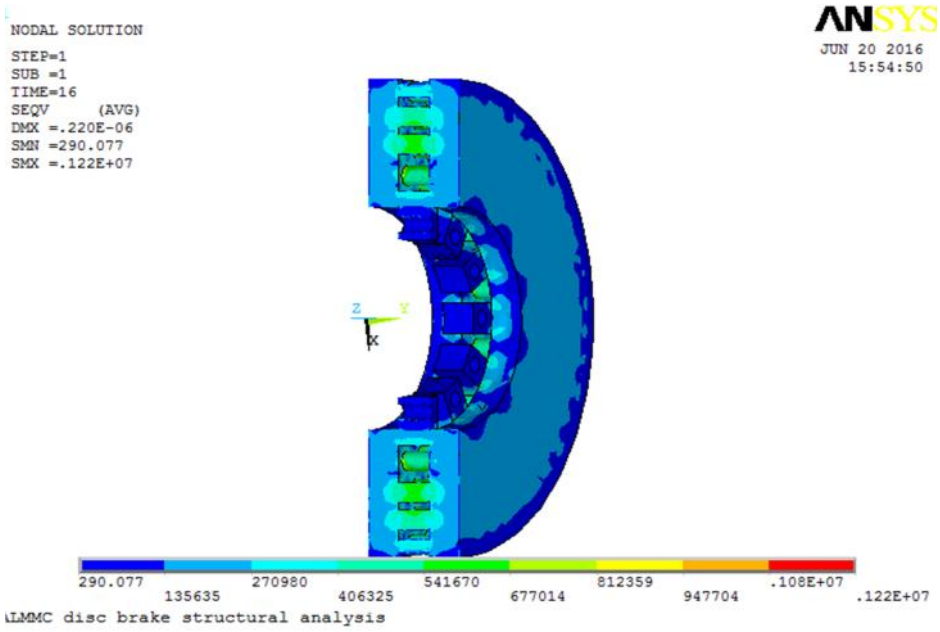


Figure 6.5 (d) sectional area of ALMMC disk



Figures 6.5 (c) and (d) show that, the maximum Von-mises stress of the disk is 0.122E+07 appeared in the middle of the friction surface and minimum value is 290.077.

II. Figures 6.6 (a) and (b) show the total deformation; (c) and (d) von-mises stress obtained throughout the simulation in a case of grey cast iron disk

Figure 6.6 (a) conventional brake disk of Grey cast iron

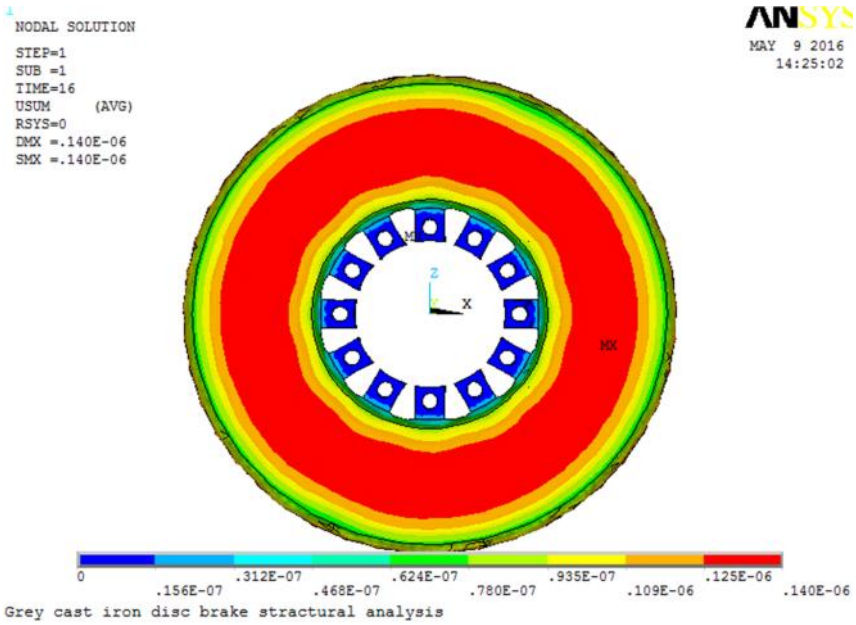
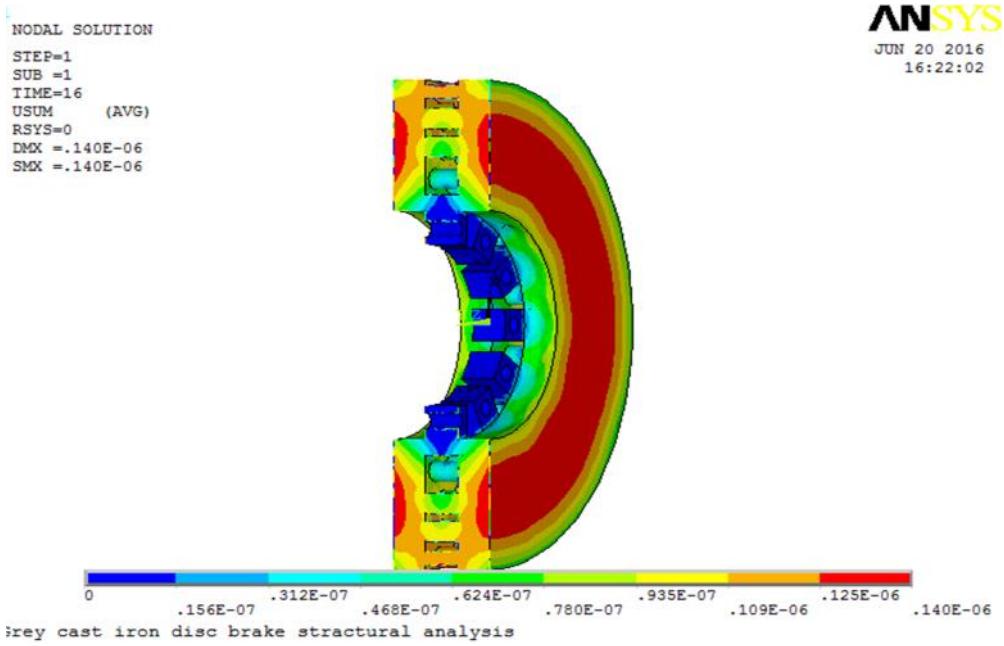


Figure 6.6 (b) sectional area of grey cast iron disk



Figures 6.6 (a) and (b) show that, the maximum deformation of the disk is 0.140E-6 m appeared wide area in the middle of the friction surface and ventilation vanes of the disk.

Figure 6.6 (c) Von-mises stress field contours throughout simulation of Grey cast iron disk

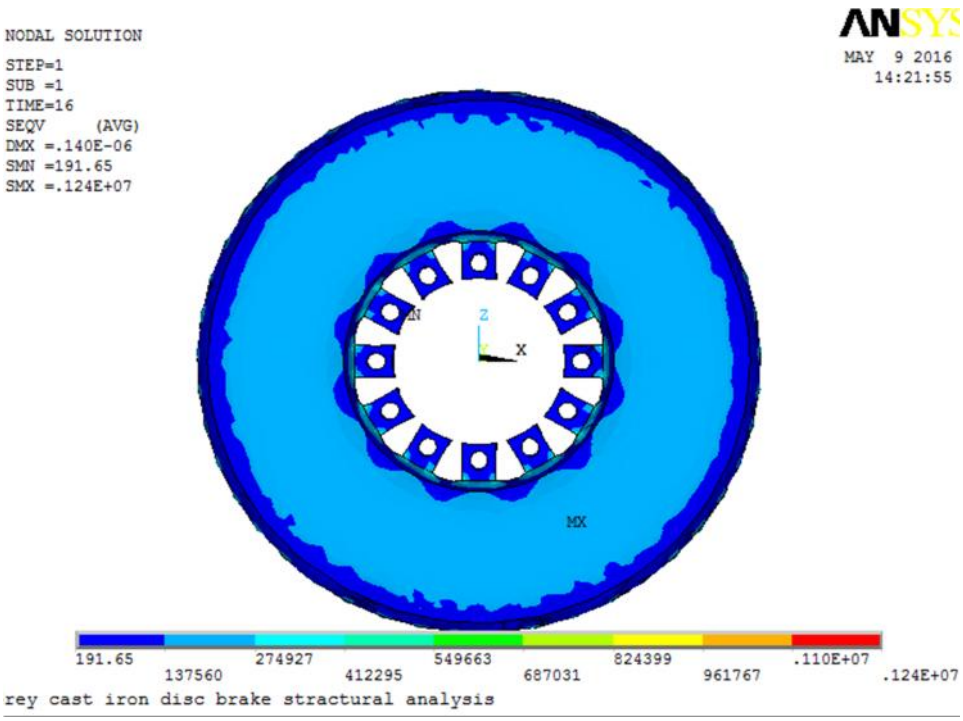
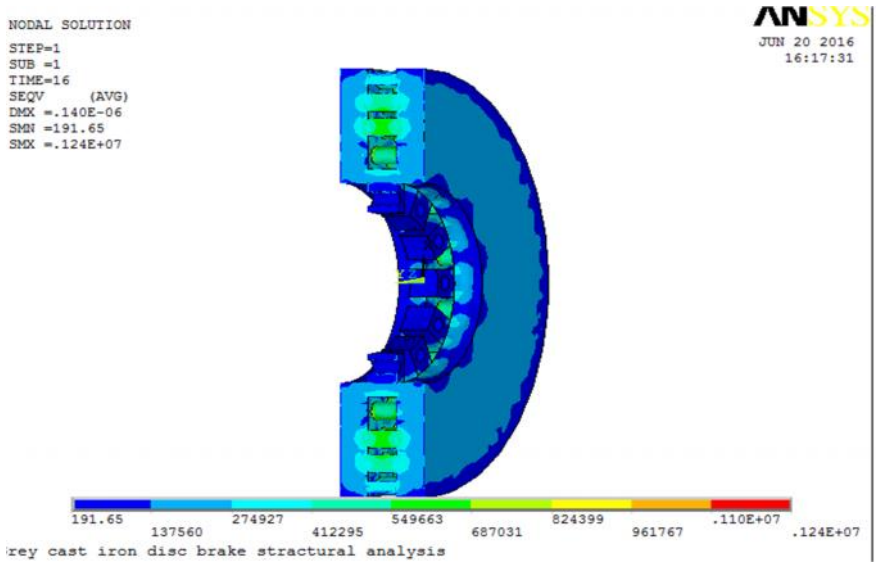


Figure 6.6 (d) sectional brake disk of Grey cast iron



Figures 6.6 (c) and (d) show that, the maximum Von-mises stress of the disk is 0.124E+07 appeared in the middle of the friction surface and minimum value is 191.077.

III. Figures 6.7 (a) and (b) show the total deformation; (c) and (d) von-mises stress obtained throughout the simulation in a case of cast steel disk

Figure 6.7 (a) conventional brake disk of cast steel

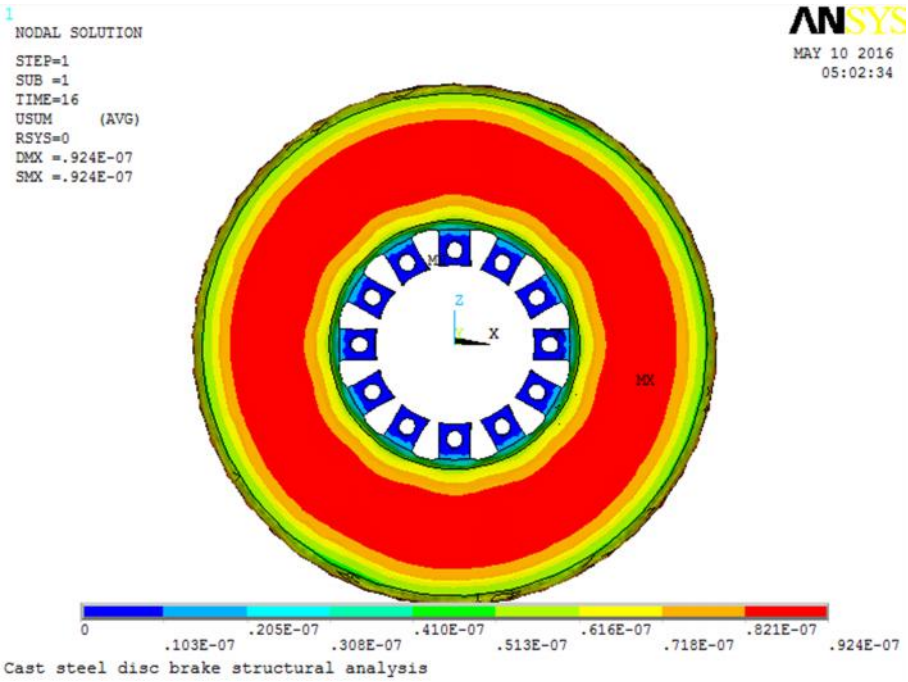
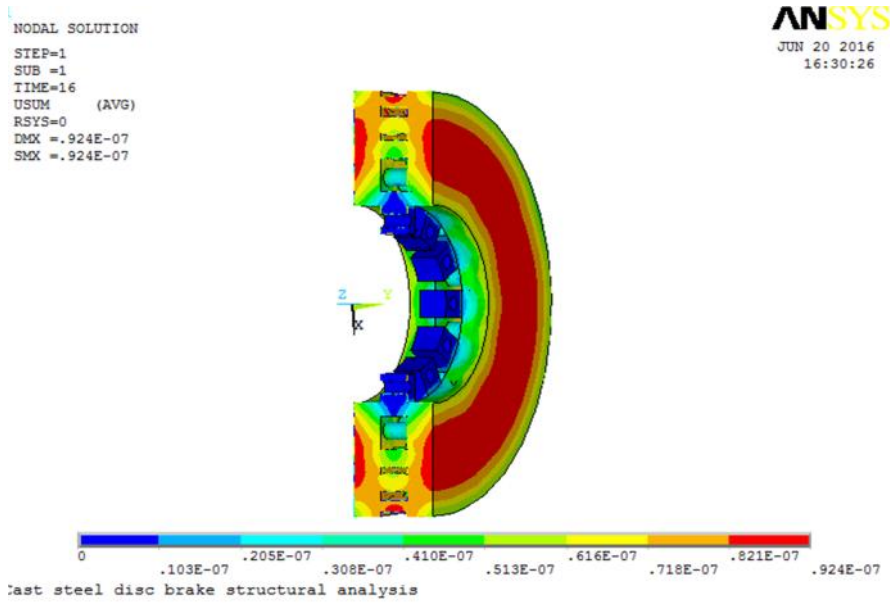


Figure 6.7 (b) sectional area of cast steel disk



Figures 6.7 (a) and (b) show that, the maximum deformation of the disk is 0.924E-7 m appeared wide area in the middle of the friction surface and ventilation vanes of the disk.

Figure 6.7 (c) conventional brake disk area of cast steel

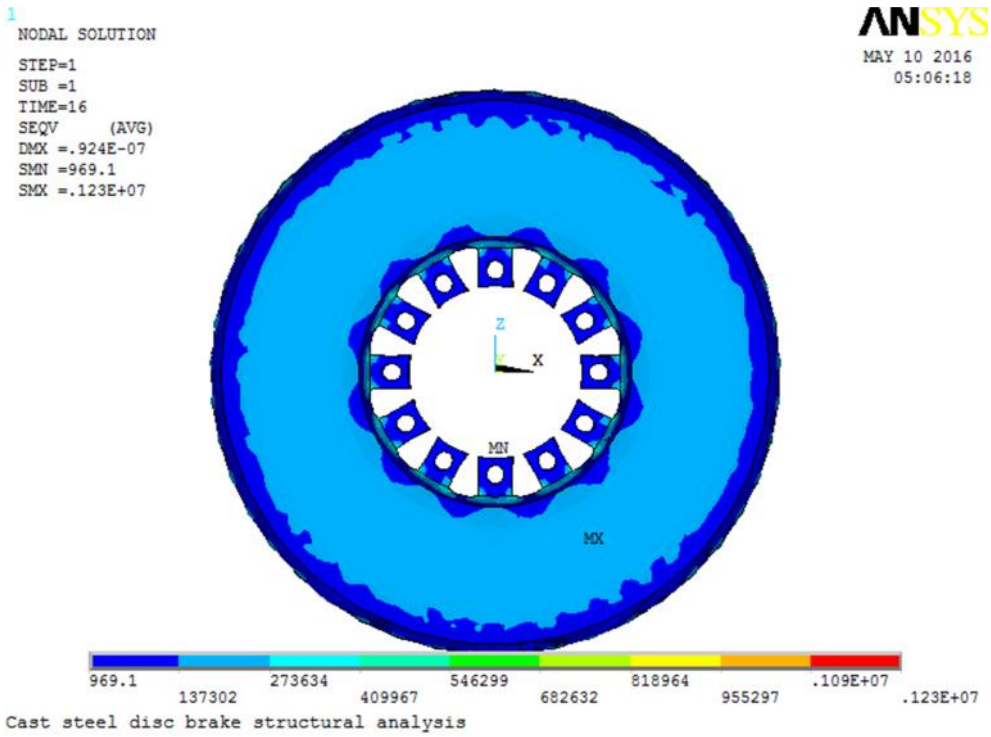
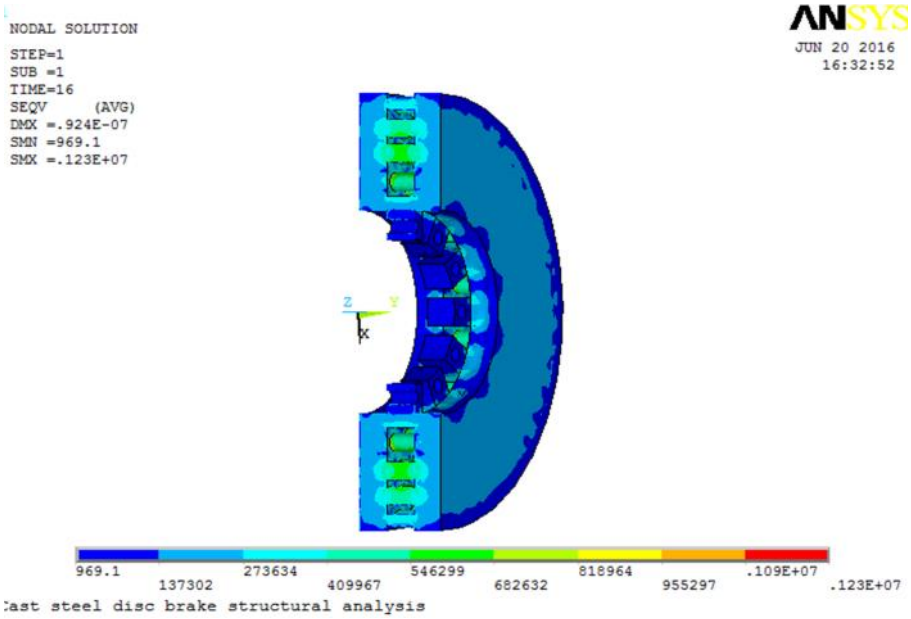


Figure 6.7 (d) sectional area of cast steel disk



Figures 6.7 (c) and (d) show that, the maximum Von-mises stress of the disk is 0.123E+07 appeared in the middle of the friction surface and minimum value is 969.1.

IV. Figures 6.8 (a) and (b) show the total deformation; (c) and (d) von-mises stress obtained throughout the simulation in a case of carbon-carbon composite disk

Figure 6.8 (a) conventional brake disk of c/c composite

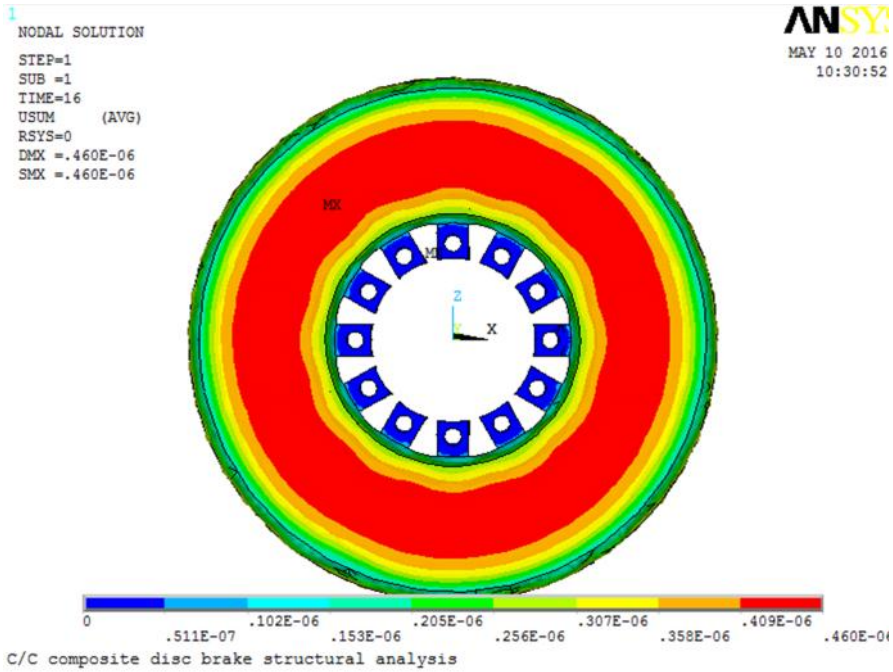
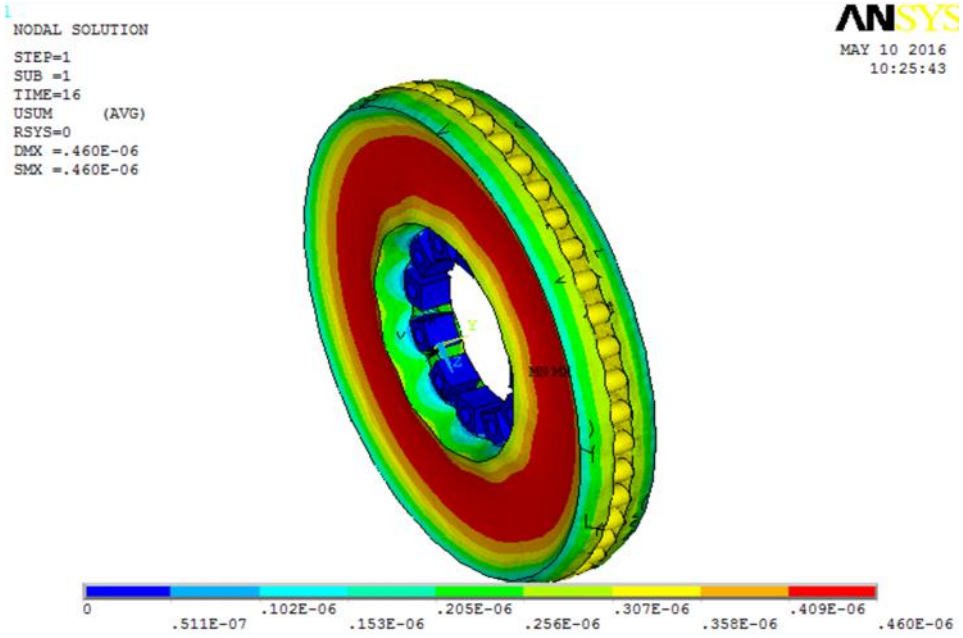


Figure 6.8 (b) isometric view of c/c composite disk



Figures 6.8 (a) and (b) show that, the maximum deformation of the disk is 0.460E-6 mm only appeared wide area in the middle of the friction surface the disk.

Figure 6.8 (c) conventional disk brake c/c composite

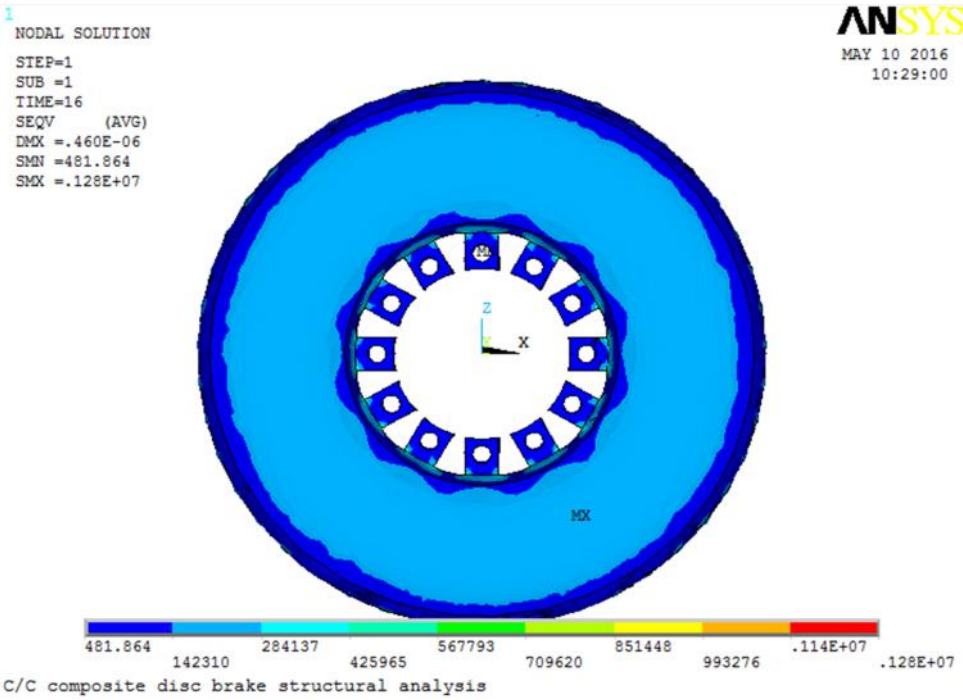
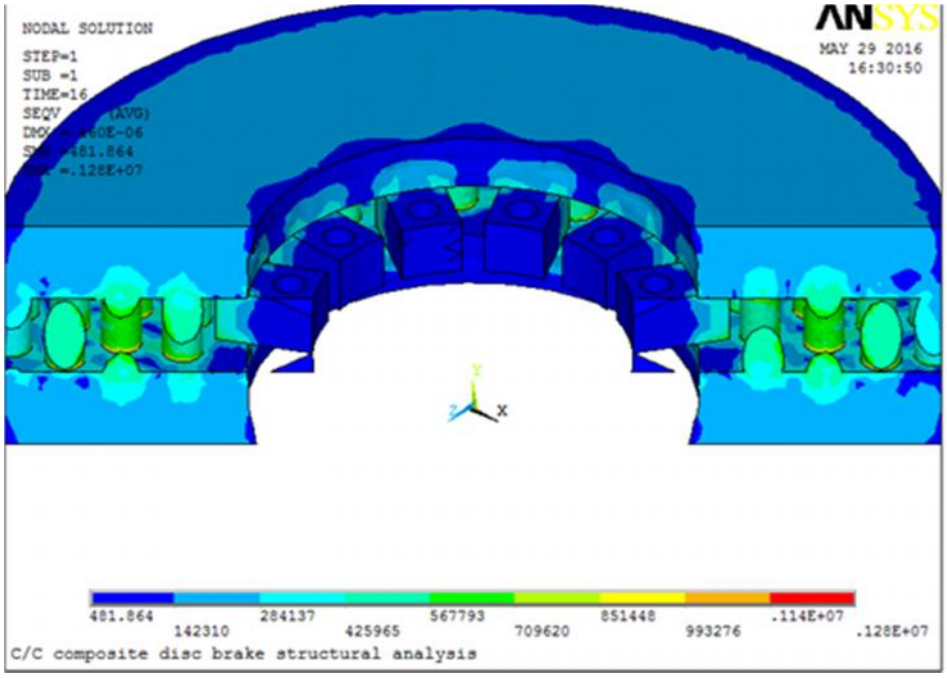


Figure 6.8 (d) sectional area of c/c composite



Figures 6.8 (c) and (d) show that, the maximum Von-mises stress of the disk is 0.128E+07 appeared in the middle of the friction surface and minimum value is 481.864.

6.2 Result discussion

The diagrams of the temperature distribution, total deformation, and Von-Mises for the examined disk of the four different material for the moment $\mathbf{t_b}$ = 16 seconds of the braking process are shown in Figures 6.1, 6.2, 6.3, 6.4, 6.5, 6.6, 6.7 and 6.8. The figure 6.9 shows the temperature distribution verses operating braking time on various friction surfaces of the four candidate materials; ALMMC, Grey cast iron, Cast steel and Carbon-carbon composite of ventilated disk in a case of Emergency braking. In addition, in order to show the maximum and minimum of each quantity, values are tabulated in table 6.1.

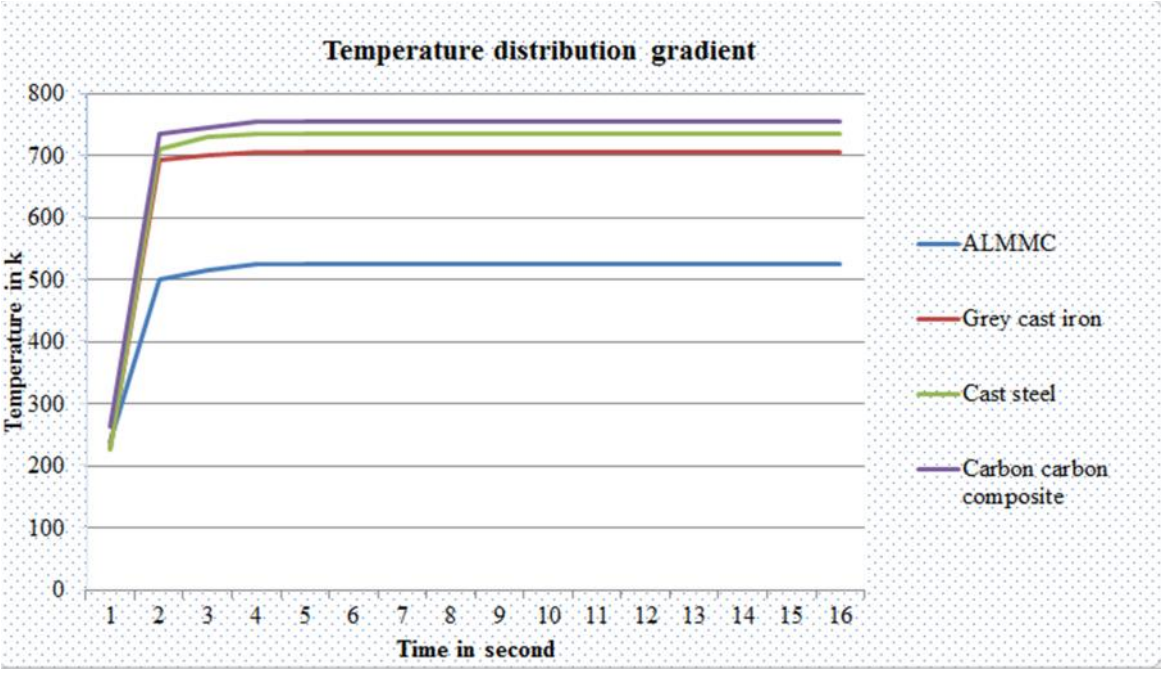
Table 6.1 Results of candidate disk brake materials

Brake disk materials	Descriptions				
	Temperature Gradient [k]		Total Deformation[m]	Von-Mises Stress [Pa]	
	Minimum	Maximum		Minimum	Maximum
ALMMC	239.963	525.01	0.202E-06	290.077	0.122E+07
Grey cast iron	230.071	705.437	0.140E-06	191.077	0.124E+07
Cast Steel	226.387	735.088	0.924E-07	969.1	0.123E+07
Carbon carbon-composites	263.244	755.07	0.460E-06	481.864	0.128E+07

As shown in table 6.1 the maximum temperature rise after braking in ALMMC of disk is 525.01k and the corresponding maximum Von-MISES stress induced is 0.122E+07 pa with total deformation 0.202E-06 m. In case of Grey cast iron disk maximum temperature is 705.437 k and the corresponding maximum Von-Mises stresses induced is 0.124E+7 pa with total deformation 0.140E-06 m. Similarly, in a case of Cast steel, the maximum temperature is 735.088k, and the maximum Von-Mises 0.128+07 pa is induced and the corresponding total deformation is 0.924E-07 m. And also we found that the maximum temperature rise in carbon-carbon composite after transient braking is 755.05 k and maximum Von-Mises stress is 0.128E+07 pa with the corresponding total deformation 0.460E-06 m. So that, we can deduce from thermodynamics consideration in braking system (i.e. the energy dissipated in the form of heat can generate rise in temperature ranging from 300Ċ to 800 Ċ). Carbon-carbon composite disk has slightly highest temperature rise throughout braking. ALMMC has lowest temperature rise. Consequently, from the above temperature distribution results, we arrive at all the materials can be considered suitable for this braking application hence, considering the possible rise temperature ranging (300Ċ to 800Ċ) [2].

In addition, the results show the influence of the material properties of the disk considered to facilitate better conceptual design of the brake system. Figures 6.1, 6.2, 6.3, 6.4, 6.5, 6.6, 6.7 and 6.8 show the effect of material properties in a thermo elastic contact stability problem with one body in contact with a rigid plane surface and further show the relative importance of each parameter on the variation of thermo elastic behaviors. The thermal expansion coefficient and the elastic modulus of the disk materials have a larger effect on the thermo elastic behaviors of disk brakes.

Figure 6.9 Temperature distribution gradient of the four disk material throughout braking time $\mathbf{t_b}$ = 16 seconds



The above figure 6.9 shows the temperature distribution gradient verses operating braking time on various friction surfaces of four different ventilated disk brake materials in a case of Emergency. The numerical simulation shows that radial ventilation plays a very significant role in cooling of the disk in the braking phase. The higher convection coefficients, achieved by airflow cooling will not only reduce the maximum temperature at the friction interface; but also naturally tend to reduce thermal gradients of the cooling ribs and the parts where the disk are bolted to the hub (see fig. 6.1, 6.2, 6.3 and 6.4). From figure 6.9, we can observe that the temperature gradient of the entire candidate disk materials raise rapidly at 2 second and maintain constant value throughout the rest of braking action. This shows that as time increases the

operating temperature does not increase directly. Since heat will be removed faster from hotter parts of the disk. Therefore, the airflow cooling should be very effective to reduce the risk.

6.3 Conclusion

In this final year project, three-dimensional modeling, ANSYS CFX and meshing using the simulation program ANSYS ICEM CFD were successfully implemented. Modeling and simulation successfully modified and improved to using three-dimensional. This has allowed for greater flexibility and accuracy in the results achieved.

In the present study, the thermal and structural stress analyses of the four candidate materials of ventilated brake disk of railway vehicles during emergency braking at a speed of 80 km/h with the airflow cooling has been studied using FE and CFX methods. ANSYS 12.1 software is applied to the thermo elastic contact problem with frictional heat generation. To obtain the simulation of thermo elastic behavior appearing in disk brakes, the coupled heat conduction and elastic equations are solved with contact problems. The effects of the friction material properties on the contact ratio of friction surfaces are examined and the larger influential properties are found to be the thermal expansion coefficient and the elastic modulus. Based on these numerical results, the thermo elastic behaviors of the four different disk brake materials are also investigated.

Comparing the different results obtained from analysis in table 6.1. It is concluded that ALMMC is the best possible disk material for the present application. The results revealed that ALMMC could effectively enhance the thermal performance of the brake disk; the highest temperature after emergency braking is 525.01 k, on a straight track in 16 second. The maximum stress can reach 0.122E+07 Pa with considering centrifugal load and having 0.202E-6 m total deformation. The presented approach is the basis for the stress analysis of the appropriate stress states, as a consequence, of analyzed thermal loads in the simulated operation conditions. The results obtained are satisfactory as compared with research literatures carried out in this area.

This conclusion is in coincidence with other researches. Maleque, Dyuti and Rahman, (2010), Material selection method in design of Automotive brake disk, [35]. In their recent work, the aims were to develop the material selection method and select the optimum material for the application of brake disk system. Two methods were introduced for the selection of materials,

such as cost per unit property and digital logic methods. Material performance requirements were analyzed and alternative solutions were evaluated among cast iron, aluminum alloy, titanium alloy, ceramics and composites. Mechanical properties including compressive strength, friction coefficient, wear resistance, thermal conductivity and specific gravity as well as cost, were used as the key parameters in the material selection stages. The analysis led to aluminum metal matrix composite as the most appropriate material for brake disk system.

In addition, Properties of the AMC rotor shows improved cooling effect due to its high thermal conductivity when compared to conventional cast iron properties. The high heat dissipation rate of the AMC can influence the temperature rise and distribution, which in turn minimizes the thermal elastic instability (TEI), brake fluid vaporization (BFV) and thermally excited vibrations (TEV) and subsequently prolong the service life of the rotor. Both the experimental and simulated results for the AMC brake rotor are in good agreement (37) and similarly [36].

6.4 Recommendation

In order to achieve a higher accuracy of results and a better understanding of the performance of the brake-disk, the whole brake system (including the hub and wheel) should be incorporated for simulation. This task will be facilitated by the three-dimensional modeling and simulation techniques already implemented in this project.

For a more accurate understanding of the ductile and brittle failure behavior of the brake-disk, the prototype of the brake-disk should be manufactured and tested. It is only through actual experiments and testing that the yielding and cracking behavior of the brake-disk be better understood.

6.5 Scopes for further study

The present work intention is to contribute in analyzing optimum material of train in the ventilated disk brakes caused by the thermal and mechanical loading. ANSYS software analyzes the investigation of thermal analysis and thereby considering convection cooling. As future work, the analysis can be taken in a case of repeated braking action and including cost analysis.

6.6 Reference

1. Rusnak R.M., Schwartz H. W., Coleman W. P.; A compari-son by Thermal Analysis of Rotor Alloys for Automobile disk Brakes ; Research Laboratories, The Bendix Corp. 700137.
2. Sparkford N.Y., Sparkford Nr Yeovil; Automotive Disk Brake Manual: Produced in Collaboration with Brembo SpA, Haynes Publishing, 1998.
3. Thomas W. B.; Automotive Braking Systems, 2nd edition, Harcourt Brace College Publishers, 1995 TL269.B57.
4. Perpinya D., Reliability and Safety in Railway, barcelona, Spain: In tech, march 2012.
5. Smith, "an efficient lightweight brake system," 2002.
7. C. R. R. P. B. Daniel Das. A, "Structural and Thermal Analysis of Disc brake in Automobiles," vol. 2, no. 3, 2013.
8. LU S K, LIU H Y, YU L, JIANG Y L. 3D FEM simulations for the homogeneity of plastic deformation in aluminum alloy HS6061-T6 during ECAP [J]. Procedia Engineering, 2011, 12: 35−40
9. Milo {evi}, M. S., et al.: Modeling Thermal Effects in Braking Systems of Railway Vehicles, THERMAL SCIENCE, Year 2012, Vol. 16, Suppl. 2.
10. Moynihan. S, "Rail Crises As Train Brake faile," Melbourn, 2006.
11. LU S K, LIU H Y, YU L, JIANG Y L. 3D FEM simulations for the homogeneity of plastic deformation in aluminum alloy HS6061-T6 during ECAP [J]. Procedia Engineering, 2011, 12: 35−40.
12. Lee K (1999) Numerical prediction of brake fluid temperature rise during braking and heat soaking. SAE, International Congress and Exposition Detroit, Michigan, 1–4 March. URL: http:// delphi.com/pdf/techpapers/1999-01-0483.
13. Valvano T, Lee K (2000) An analytical method to predict thermal distortion of a brake rotor. SAE, World Congress, Detroit, Michigan, 6–9 March. URL 2000-01-0445.
14. Voldrˇich J (2006), Frictionally excited thermo elastic instability in disc brakes-Transient problem in the full contact regime, Int. J. Mech Sci 49(2):129 –137, DOI: 10.1016/j. ijmecsci.2006.08.008.
15. Gao CH, Lin XZ (2002), Transient temperature field analysis of a brake in a non-axisymmetric 3-dimensional model, J. Mat Proc.Tech 129:513–517, DOI:10.1016/S0924-0136(02)00622-2.

16. Faramaz Talati and Salman Jalalifar, (2009), Analysis of heat conduction in a disc brake system, Heat mass transfer 92009) 45:1047-1059, DOI: 10.1007/s00231-0476-y.
17. David, A. J., Bryan, A.S., Ross, G., (2003), "Analysis of the Flow through a vented Automotive Brake Rotor," Journal of Fluids Engineering, Vol. 125, pp 979-986.
18. McPhee, A.D., Johnson, D.A. (2008), "Experimental heat transfer and flow analysis of a vented brake rotor" International Journal of Thermal Sciences 47, pp 458–467.
19. Thilak V.M.M., Krishnaraj R.: Transient thermal and structural analysis of the rotor disc of disc brake, International Journal of Scientific & Engineering Research Volume 2, Issue 8, August-2011, ISSN 2229-5518, (2011).
20. Yildiz Y., Duzgun M.: Stress analysis of ventilated brake discs using the finite element method.International Journal of Automotive Technology, Vol. 11, No. 1, pp. 133-138 (2010).
21. A. Belhocine, A.R. Abu Bakar and M. Bouchetara: Numerical Modeling of Disc Brake System in Frictional Contact, Department of Mechanical Engineering, USTO Oran University , L.P 1505 El-Mnaouer, USTO 31000 Oran, Algeria and Department of Automotive Engineering, Universiti Teknologi Malaysia, 81310 UTM Skudai, Malaysia, Vol. 36, No. 1 (2014) 49-66(2004).
22. Ali Belchocine, 2012 "Thermal analysis of a solid brake disc", Applied thermal engineering 32(2012) 59-67.
23. Koetniyom S., Brooks P. C., Barton D. C.; The development of a material model for cast iron that can be used for brake system analysis; The University of Leeds; Proc Instn Mech Engrs Vol 216 Part D: J Automobile Engineering, 2002.
24. Sparkford N.Y., Sparkford Nr Yeovil; Automotive Disk Brake Manual: Produced in Collaboration with Brembo SpA, Haynes Publishing, 1998.
25. M. R. 2. D. G. 3. V.Chengal Reddy 1, "modeling and analysis of FSAE car disk brake using FEM. "vol. 3, no. 9, 2013.
26. R.M. Wang and M.K. Surappa, Microstructure and interface structure studies of SiCp reinforced Al (6061) metal-matrix composites Materials Science and Engineering. A, 254 (1-2), (1998), 219-226.
27. Ameer Fareed Basha Shaik, Ch.Lakshmi Srinivas,"structural and thermal analysis of disc brake with and without cross drilled rotor of race car," Bapatla Engineering, April 22, 2009.

28. Ali Belhocine, and Mustafa Bouchetara, (August., 2012),” Investigation of temperature and thermal stress in ventilated disk brake based on 3D thermo mechanical coupling model,” August, 2012.
29. Ali Belhocine, and Mustafa Bouchetara, (JUN. 2013), ” Thermo mechanical Modeling of Disc Brake Contact Phenomena, transaction on control and mechanical systems,” vol. 2, no. 6, pp. 302-309, jun., 2013.
30. Cruceanu, C., Brakes for Railway Vehicles (in Romanian), Matrixrom, Bucuresti, 2007, ISBN 978-973-755-200-6.
31. Mackin, T.J.: Thermal cracking in disc brakes. Engineering Failure Analysis (2002), no. 9, str.63-76.
32. Carlos H. Galindo - Lopez. Evaluating new ways of conducting convective heat dissipation experiments with ventilated brake discs" Cranfield University, Bedfordshire, MK43 OAL, UK.
33. Saifullah, A. B. M., Masood, S. H., Finite Element Thermal Analysis of Conformal Cooling Channels in Injection Moulding, 5th Australasian Congress on Applied Mechanics, ACAM 2007, Brisbane, Australia, 2007, pp. 337-341.
34. ANSYS V.11 user’ Manual guide.
35. M.A Malaque, S.Dyuti and M.M. Rahman, (2010), Material selection method in design of Automotive brake disk, vol. WCE 2010, LONDON, UK.
36. Al-Bahkali, E. A. and Barber, J. R. (2006). Nonlinear Steady State Solution for a Thermo elastic Sliding System Using Finite Element Method. Journal of thermal Stresses, 29 (2): 153-168.
37. A.A. Adebisi, M.A. Maleque and Q.H. Shah, (2011), Surface temperature distribution in a composite brake rotor. International Journal of Mechanical and Materials Engineering (IJMME), Vol.6 (2011), No.3, 356-361.