**ADDIS ABABA UNIVERSITY**

**SCHOOL OF MECHANICAL AND INDUSTRIAL**

**INSTITUTE OF TECHNOLOGY (SMIE)**

# THERMAL ANALYSIS OF VENTED DISC AND OPTIMIZATION OFTHE SHAPE OF THE VENTED CURVES IN AALRT

**A Thesis Submitted to the School of Graduate studies Addis Ababa University in partial fulfillment of the requirements for the Degree of Masters of Science in Rolling Stock Engineering**

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**May, 2015**

## ABSTRACT

The disc brake is a device for slowing or stopping the rotation of a wheel. Braking of the vehicle leads to heat generation during each braking event. Transient thermal analysis of the Rotor Disc of Disk Brake is aimed at evaluating the performance of ventilated disc brake rotor of a rail vehicle under emergency braking conditions and there by assist in optimization the curves of the shape of ventilated disc to identified the thermal analysis effect during braking. Here Disc brake modeled with CATIA software and its analysis is done using ANSYS workbench 12 with considering CFD-cfx package software. The main purpose of this study is to analyzing the thermal behavior of the ventilated disc brake during the braking phase by optimization the shapes of the curves of the vented disc and compare results. The coupled thermal-structural analysis is used to determine the Von Mises stress established in the disc for ventilated disc with the same materials and other data from ERC to enhance performance of the rotor disc.

The results from the simulation with changing shapes of the vane like vane number, vane angle and vane curve radius; as vane number 50 ,60, 70 and 80 , then temperature generated reduced from 311.76-302.67[°C] until vane number 70. But at vane number 80 it is 308.94 [°C].And with changing the straight vane number 50 to 5, 10 and 20 angles its values of temperature generating at a time of 5.2488 [s] is 302.9, 239.86 and 302.24 respectively and at vane numbers 60 and 70 at angle 5 and 20 their values are 310.44, 310.19[°C] at time 5.2488 [s] and 309.86, 309.72 [°C] at time 5.0868 [s]respectively. At changes of vane curve radius from straight to 120 to 60mm its generating temperature is reduced 312.03 to 309.83 [°C] at time 5.2488[s]. Generally from simulation of emergency brake at braking time 9.72[s], reduced generating temperature at vane number 70 and at vane angle 5 with vane number 50 are 302.67 and 302.9[°C] respectively which preferable at emergency brake-1, when simulated at tentative with applying initial temperature induced at emergency brake-1, induced temperature at emergency brake-2 increase to 361.99 and 362.55[°C], these are preferable to compare from the current AALRT uses which is induced 383.26[°C] at emergency brake-2.

**KEY WORDS**: Finite element (FEM), CATIA, ERC, Vented disc, ANSYS and Workbench-12, CFD and AALRT

**ACKNOWLEDGEMENTS**

I would like to thank my M.Sc. Thesis Advisor Mr.Tsegaye Feleke for his support and guidance.

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

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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.

**CONTENTS……………………………………………………………………………………PAGES**

# LIST OF TABLES

## LIST OF FIGURES

## NOTATIONS

| | |
|---|---|
| $F_{disc}$ | resultant force of disc |
| M | total mass acceleration or deceleration |
| V | final velocity |
| V0 | initial velocity |
| $S_b$ | brake distance |
| $T_b$ | brake time |
| G | acceleration due to gravity |
| μ | coefficient of friction |
| α | gradient |
| C p | specific heat |
| ρ | specific mass |
| Q | internal heat generation |
| T | temperature |
| Qs | Surface heat flux |
| h | convective heat transfer coefficient |
| T | unknown surface temperature |
| T∞ | convective exchange temperature |
| ω | angular velocity |
| ρ | density of fluid |
| V | velocity of passenger train |
| $\omega_o$ | Initial angular velocity the wheel |
| $r_w$ | Radius of vehicle wheel |
| $I$ | Polar inertial moment of rotating parts |
| $\varepsilon$ | The material's emissivity, |
| $\sigma$ | Stefan-Boltzmann constant and |
| $T_{ref}$ | Temperature of the surrounding air |

CFD…………………………….....Computational-fluid-dynamics

# CHAPTER ONE:

# INTRODUCTION

## 1.1Back ground

There has always been trade, human interaction and transport, for almost no society has ever been purely subsistence in character. Expand trade and political pressure has pushed for improvements, and there have been surges of transport developments associated with the expansion of the empires. [1]

Over land and river routes served the trade of Mesopotamia five million years ago. Roman hegemony was supported by Roman roads just as Persian, Chinese, and new world rulers were supported by their roads .Not much later the grain trade in the Mediterranean flourished along with orient European linkages later, Iberian, Dutch, French , and English empires were based on the transport and trades. [1]

Thorough going changes in transportation accompanied the evolution of the modern world and the economy of the world people society is improved. [1] The time taken for movement from place to place is becomes short .Then after developing different types mode of transportation in the world as well as world society developed still now.

Some of the transportation modes are roads, water, and air etc, from the road transportation railway is the safest and massive transportation and its developments, including systems with man or horse power, and tracks or guides made of stone or wood. The history of rail transport dates to as early as Greek times. Wagon ways were relatively common in Europe (typically in mining) from about 1500 through 1800. Mechanized rail transport systems first appeared in England in the 1820s. These systems, which made use of the steam locomotive, were critical to the Industrial revolution and to the development of export economies across the world. They have remained the primary form of land transport ever since for most of the world. “A good horse on an ordinary turnpike road can draw two thousand pounds, or one ton. [2]

Apart of gentlemen was invited to look upon the experiment, that the superiority of the new road might be established by ocular demonstration. Twelve wagons were loaded with stones, till each wagon weighed three tons, and the wagons were fastened together. A horse was then attached, which drew the wagons with ease, six miles in two hours, having stopped four times, in order to show he had the power of starting, as well as drawing his great load. [2]

Wagon ways (or 'tramways') are thought to have developed in Germany in the 1550s to facilitate the transport of ore tubs to and from mines, utilizing primitive wooden rails. Such an operation was illustrated in 1556 by Georgiou’s Agricola.[3] These used "Hand" carts with un flanged wheels running on wooden planks and a vertical pin on the truck fitting into the gap between the planks, to keep it going the right way.[4] Such a transport system was used by German miners at Cald beck, Cubria, perhaps from the 1560s.[5] The first true railway is now suggested to have been a funicular railway made at Broseley in Shrop shire at some time before 1605.

This carried coal for James Clifford from his mines down to the river Severn to be loaded on to barges and carried to riverside towns. [6] Though the first documentary record of this is later, its construction probably preceded the Wollaston wagon way, completed in 1604, hitherto regarded as the earliest British installation. This ran from strelley to Wollaton near Nottingham.

Wagon ways on less steep gradients could be retarded by allowing the wheels to bind on curves. As the work became more wearing on the horses, a vehicle known as a dandy wagon was introduced, in which the horse could rest on downhill stretches. Because a stiff wheel rolling on a rigid rail requires less energy per ton-mile moved than road transport (with a highly compliant wheel on an uneven surface), railroads are highly suitable for the movement of dense, bulk goods such as coal and other minerals.

The introduction of the Bessemer process, enabling steel to be made inexpensively, led to the era of great expansion of railways that began in the late 1860s. Steel rails lasted several times longer than iron.[7][8][9]Steel rails made heaver locomotives possible, allowing for longer trains and improving the productivity of railroads.[10]The Bessemer process introduced nitrogen into the steel, which made the steel become brittle with age. The open hearth furnace began to replace the Bessemer process near the end of 19th century, improving the quality of steel and further reducing cost, after passing those progresses it developed for the current stages.

Railway in Ethiopia was a firm founded in 1894 to build and operate a railway across eastern Ethiopian from the port of Djibouti to the capital of Addis Ababa. It was founded by Alfred Ilg and Leon Chefneux and headquartered in Paris, France. [11]The firm failed in 1906 when political discord halted construction, and it failed to obtain any new capital. The portion it had completed ran from Djibouti to just short of Harare, [12] the principal entrepôt for existing commerce in southern Ethiopia. [13]Its terminus evolved into the city of Dire-Dawa, today a larger city than Harare itself. Discussion of an Ethiopian railway was initiated by Alfred Ilg, an advisor to Emperor Menelek II. He had attempted to interest the previous emperor and other Ethiopian political figures in the construction of a railway to replace the six-week mule trek between the capital and the French port city, but had no success. When Menelek II acceded to the throne in 1889, negotiations began a new and a decree was granted on February 11, 1893, to study the construction of rail line. Ilg, a Swiss citizen, and a number of French associates put together a firm and received a royal charter on March 9, 1894, enabling them to start work. The demands and threats of the two governments led Emperor Menelek in 1902 to forbid the expansion of the railway line to Harare. At the current time Ethiopian railway is under construction in a new net work planning.

## 1.2 Statement of problem

Brakes are such a crucial system in stopping the rail vehicle on moving stages including braking during high speed, downhill. All of those braking moments give a different value of temperature distribution and thermal stress. Good performance of disc brake rotor comes from good material with better mechanical and thermal properties. Good designs of disc brake rotor are varying across the range of the vehicles. There are different designs and performance of disc brake rotor if compared among passenger train, freight train, commercial and heavy duty vehicle. Where the higher value of temperature during braking could lead to braking failure and cracking of disc brake rotor, so that compared the vented disc and solid disc for thermal analysis as the same material is vented one is better. In this paper, there is to investigate the thermal analysis issues of normal passenger rail vehicle disc brake rotor (ventilated), where the investigation are to determine the temperature behavior of the vented disc brake with comparing for the same disc material and applying force on the interface of the disc and improving the thermal analysis of the vented disc by changing of vane number, shape of the curve and vane angle.

## 1.3 Objective of the study

### 1.3.1. General objective:

The general objective of this study is analyzing thermal analysis of vented disc and optimization of the shape of vented curves during the vehicle braking at the maximum speed, at the maximum load and at the high gradient of the truck.

### 1.3.2. Specific objective of the study:

The specific objective of the study is to understand the temperature rise and achieving these objectives are:

- Modeling the vented disc based on ERC data with CATIA software and thermal analysis with ANSYS 12.
- Determining the temperature rise of disc during emergency braking for vented rotor.
- Identify vented disc induced deformation due to thermal stress during braking with braking time interval.
- Determining the temperature rise, variations of the vented disc by optimization the shape of the curves, by changing the number vanes and vane angles for the same material and dimensions.

## 1.4 Research methodology:

Begin with a literature review, a lot of paper and journal have been related to the disc brake especially related to the vented disc was internalize then after gathering the required inputs and Modeling and thermal analysis of the vented disc brake maximum loads and at high elevation braking conditions were done. The ANSYS workbench was used to carry out the analysis, after 3D drawing was made by using CATIA modeling software, then determining the temperature distribution, variation of the stresses and deformation across the vented disc brake profile.

The necessary data for this study was collected from ERC, all these data/parameters that will be used in load analysis where the heat flux and convectional heat transfer coefficients has been calculated at the maximum load of the rail vehicle. Later, value of load analysis has been applied on finite element analysis to simulation the thermal analysis of the vented disc.

### 1.4.1. Data collection

The purpose of collection data is to achieve the objectives listed in the sub topic of specific objective of the study; it will be collected through secondary data collection method from ERC and

- By browsing different published papers and journals
- By browsing different disc brake related books
- By visiting to the Addis Ababa LRT manufacturing work shop found on Leghar to observe lay out disc brake.

### 1.4.2. Research procedures

This sub topic determined the overall procedures of the research work stated as clear as possible as, 3D model of vented disc brake has transfer to finite element software which is ANSYS for thermal analysis transient responses after simulating wall heat transfer coefficient with CFD-cfx package software. Assigning material properties, load and meshing of the model has been done in this stages. Then, completed meshing model has been submitted for analysis. Finally an expected result from the transient responses of thermal analysis has been obtained. Here following manners sequentially having in minds those specific questions to answer.

- Data collection as stated in the data collection sub topic
- Understanding clearly the working principle of railway disc braking system.
- Analyzing appropriate simulating tools to solve the complexity stated in the statement of the problem sub topic. Like ANSYS: for analysis (simulating purpose) and C ATIA: for modeling the disc, then

Observing the effect or the result of simulation of the vented disc brake rotor at different inputs applying braking force (pressure that arrives at braking pads), and then puts the results in table forms and figures.

- Writing conclusions and recommendations and future development (if any) to investigate further more by other researchers later on.

At this stage brief summery or abstract of the research is written after completing the research work.

## 1.5 Significance of the research:

Significance of this study is beneficially for students as guidance, for researchers to further improve designs, disc materials & maintenance departments, and generally for the Ethiopian corporation of rail way& Automotive engineering.

## 1.6 Scope of the study

Gathering and internalize literature reviews on the working principles, components, standards and assumptions about disc brake and internalize it as input for the study.

- Constructs 3D model of vented using CATIA software using ERC train disc brake dimensions and transient analysis during braking.
- Perform thermal-stress analysis for the 3D modeling using ANSYS 12 for showing the temperature distribution on disc brake and comparing its value on the vented disc with

the same disc material at different time intervals and show the variations when optimize the shape of curves vented disc.

## 1.7 Organization of the Research

The research is organized in five 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 significance of the research.

Chapter two discusses Literature review which includes introduction of disc brake, different types of brake types that are used to stop and decreasing of the speed of rail vehicles and related research to this study.

Chapter three shows the data collection, analysis and physical modeling of passenger train disc brake as well as the material and boundary conditions are discussed in detail.

Chapter four deals about results and discussions of the results obtained after simulating using ANSYS, with various shapes of vane, angle and shape as using the same material.

Chapter five deals Conclusion, recommendation and future work of the research on vented disc brake rotor, this is the final chapter of this research.

# CHAPTER-TWO

# LITERATURE REVIEW

## 2.1 Introduction

Braking performance of the train is one of the most important factors that affected the traffic safety and the brake is the key components related to the running safety of the railway. It is mostly due to the thermal fatigue failure cause brake failure caused by traffic accident. The numerical simulation of the temperature field is the main method of solving the brake thermal recession and thermal fatigue. The biggest difficulty is the analysis of the transient and alternating temperature field during the braking process and the establishment of a simplified model of the thermal - structural coupling. [14].

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

There are two types of disc brake rotor used in passenger Vehicles which is solid disc and ventilated disc [15]. A solid rotor is simply a solid piece of metal with friction surface on each side and this type of rotor is light, simple, cheap and easy to manufacture. A ventilated disc meanwhile refers to a rotor with various opening profiles (holes, grooves etc) which provide better cooling performance (additional heat transfer function) and weight savings as well as aesthetic appearance [16]. Therefore, it is widely used compared to solid disc. The thermal stability of the disc shape is influenced by the quantity of the material and the heat treatment before machining as well as the basic design for the disc rotor. Heat transfer is energy in transit, which occurs as result of a temperature gradient or difference. This temperature difference is thought of as a driving force that causes heat to flow. Heat transfer for a ventilated disc brake rotor occurs by three mechanism or modes: conduction, convection and radiation. Ventilated disc brake generally exhibit convective heat transfer coefficient that is approximately twice as large as those associated with solid discs. [17] Generally there are different brake types and their functions are to control a vehicle's speed when driving downhill, to minimize a vehicle's speed when necessary and to keep a vehicle stationary when in parking.

## 2. 2 Types of train brake system

### 2.2.1 Air brake:

Air brake uses the compressed air as the source of brake force and controls the brake force by changing the pressure intensity of the compressed air. There are two types of air braking system; direct and automatic air brake. Compressed air straight brake system is the simplest continuous brake; both in constructive and functional terms.

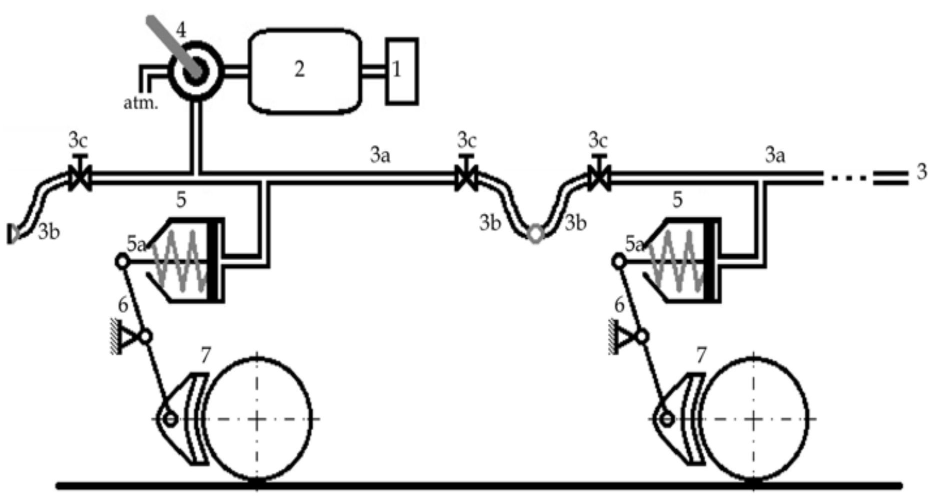



**Fig.2.1. Schematic of straight (direct) compressed air brake system: [35]**

1 –compressor; 2 – main reservoir; 3 – train's brake pipe; 3a – vehicle's brake pipe;
3b – flexible coupled hoses; 3c – angle cock; 4 – driver's brake valve; 5 – brake cylinder; 5a– piston rod; 6 – brake rigging; 7 – brake shoe.

Indirect air brake (automatic) is a continuous brake, basically having the same subsets, with identical functions as for straight brake (see fig. 1.2).

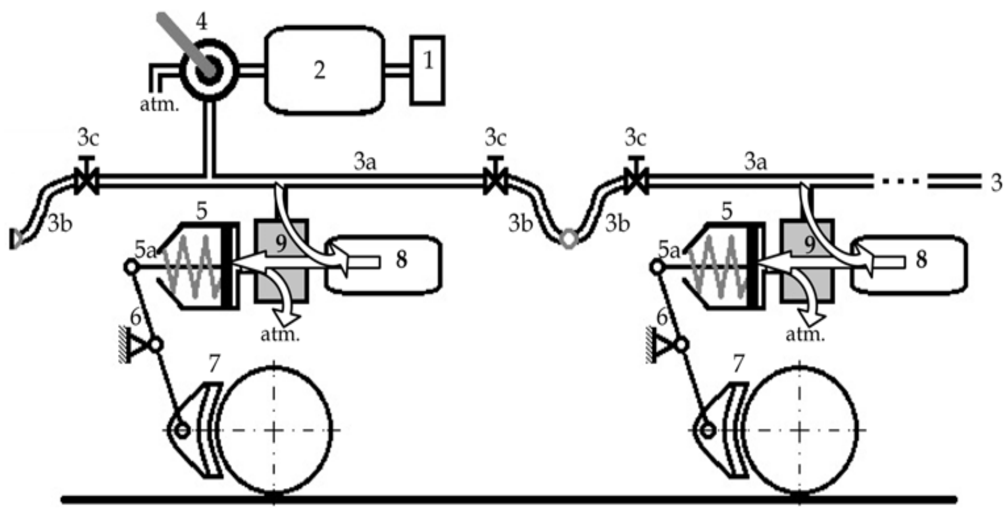



**Fig.2.2. Schematic of indirect (automatic) compressed air brake system: [35]**

1 –compressor; 2 – main reservoir; 3 –train's brake pipe; 3a – vehicle's brake pipe;
3b – flexible coupled hoses; 3c – angle cock; 4 – driver's brake valve; 5 – brake cylinder; rod; 6 – brake rigging; 7 – brake shoe; 8 – auxiliary reservoir; 9 – air brake distributor.

### 2.2.2 Electronically controlled (air) pneumatic brakes

Electro-pneumatic brake is formed by adding some electrically controlled components, such as electro-magnetic valves, on the basis of automatic brake. The electronic signals are used to control the action of brake and release, the original power of brake is compressed air, the electronic control is out of order, the braking system controlled pneumatically.

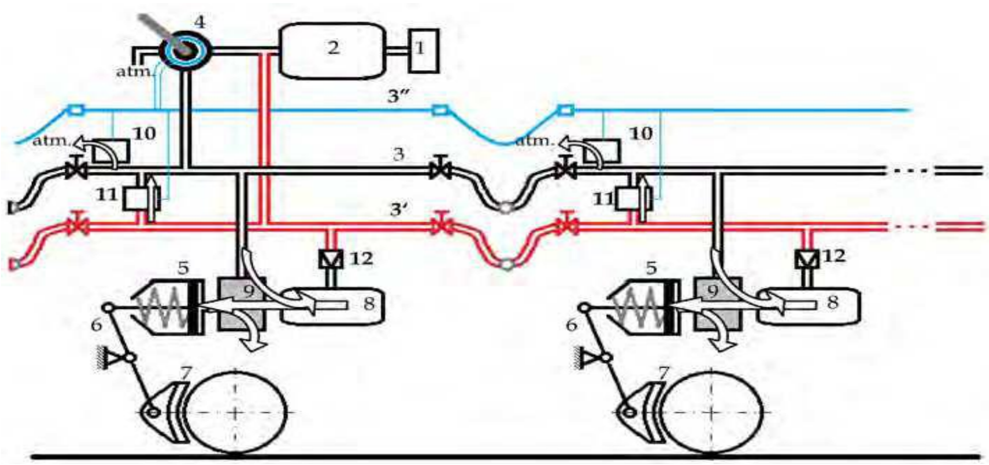



**Fig.2.3. Schematic of electro-pneumatic brake system**: [35] 1 – compressor; 2 – main reservoir;3 –brake pipe; 3' – main reservoir pipe; 3" – electric brake command;4 – driver's brake valve; 5 – brake cylinder; 5a – piston rod; 6 – brake rigging; 7 – brake shoe;8–auxiliaryreservoir; 9 – air distributor; 10 – braking vehicle's electrovalves;11 – releasing vehicle's electro valves; 12 – one-way valve.

### 2.2.3 Tread brake

Tread braking uses block-shaped brake shoes made of cast iron or other materials. Here the brake shoes are pressed on to the running wheel treads, and then the train's kinematic energy is converted into heat energy by the mechanical friction between wheel treads and brake shoes. The heat generated is dissipated into the atmosphere and the braking force is created.

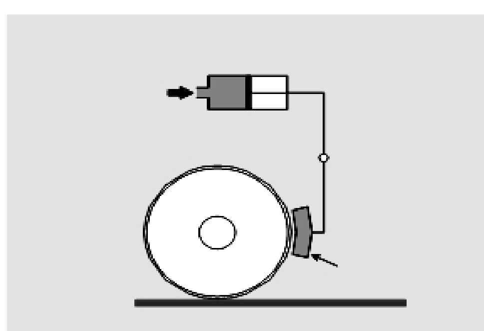


**Fig.2.4. tread brake system [35]**

### 2.2.4 Rail (eddy current) braking system

Adhesion braking systems depend heavily on adhesion between the wheel tread and the rail. In the case of high speed rolling stock, adhesion decreases as speed increase, making it necessary for the train to reduce braking force to avoid wheel sliding. The end result is longer braking distance, and then to solve this problem a rail (eddy current) braking systems developed that does not depend on adhesion.[35]

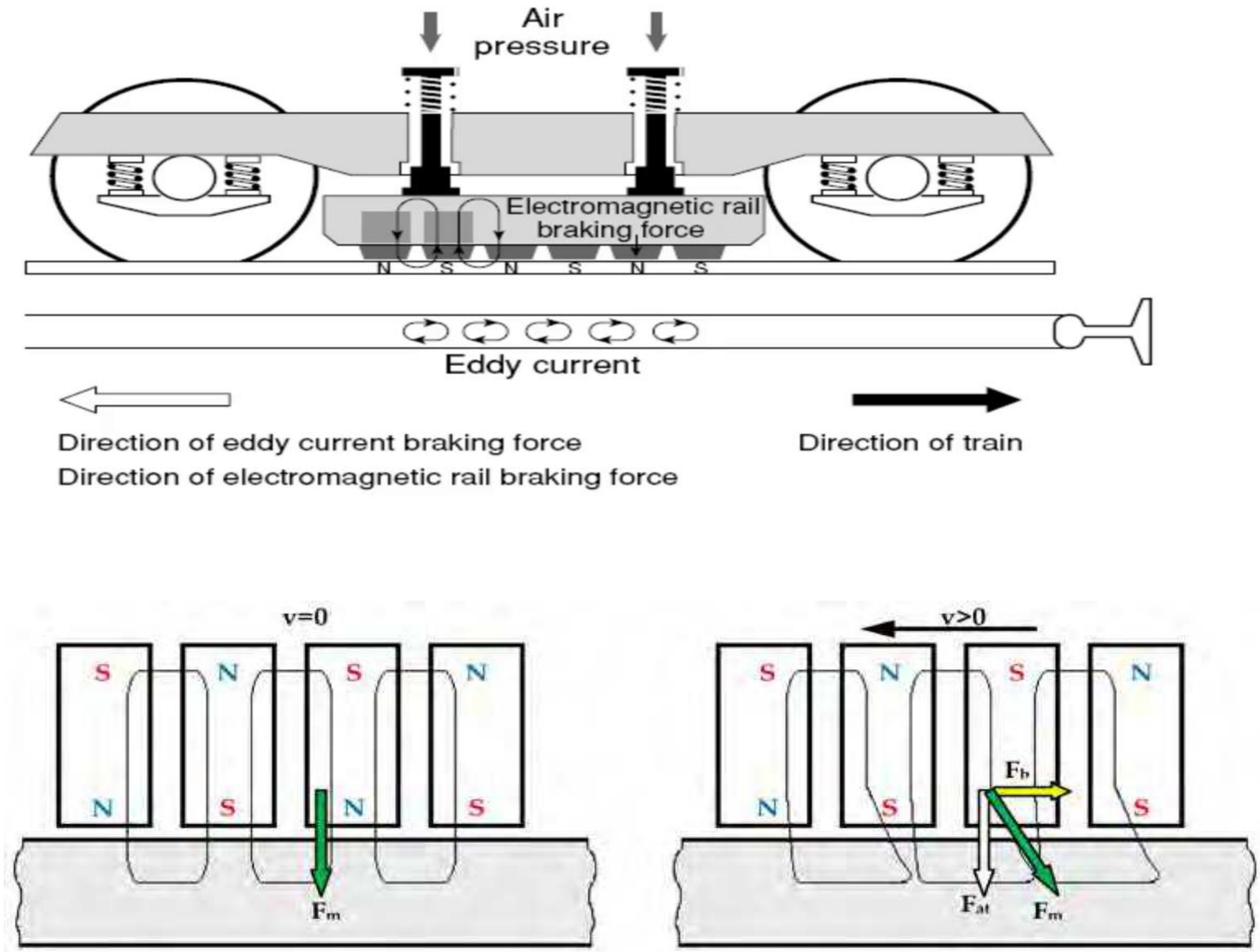



**Fig.2.5. Principle of eddy current brake system (Fm – magnetic force; Fat – attraction force; Fb– braking force**

## 2.3 Disc brake

Disc brake in brief: For high speed train, the braking technology get fast pace worldwide especially in China, even the main brake method is electricity or eddy current brake but frictional brake is a redundancy backup and reliable way to stop the train in emergency. Most form of frictional bakes on the train is disc brake type mounted on the train axels or on wheel plate to dissipate brake energy. Disc brakes are widely used for it is simple, powerful and easy maintenance and allow new materials of disc and pad to be brought into use to provide effective friction and minimize the wear, to slow or stop. Brakes convert motion to heat, and if the brakes get too hot, they become less effective, a phenomenon known as brake fade.

Disc-style brakes development and use began in England in the 1890s. Compared to drum brakes, disc brakes offer better stopping performance, because the disc is more readily cooled. As a consequence discs are less prone to the "brake fade"; and disc brakes recover more quickly from immersion (wet brakes are less effective). [33] The disc is normally made up of cast iron & other materials, at braking the discs are clamped by brake pads normally made up of composite materials on the calipers to impede rotations, creating braking force & converting trains kinetic energy into heat and its advantages are more resistance to brake fade, better cooling, water& dirt resistance, less maintenance and greater surface area for a given weight of brake. Disc brake has three main components; caliper, disc rotor and frictional pads.

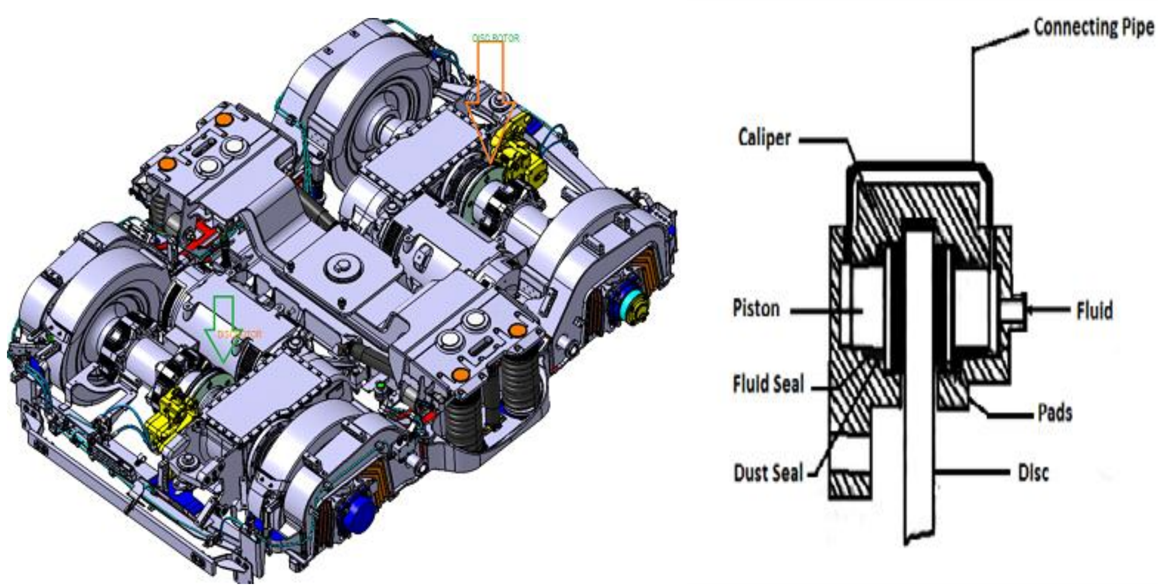



**Fig.2.6. assembly of disc brake system [34, 35]**

### 2.3.1 Disc brake components

A disc brake assembly generally consists of three main components: a brake disc (also called the rotor), a caliper and two brake pads. Brake discs may come in many (e.g. solid or vented). The caliper, which contains a cylinder with a piston (multiple pistons may be used in some types of heavy-duty brakes), holds the two brake pads on either side of the rotor. The movement of the piston is controlled by a hydraulic system. When hydraulic pressure is applied, the piston is pushed forward to press the inner pad against the rotor while the housing is pushed in the opposite direction to press the outer pad against the rotor, hence, generating a braking torque. Brake pads are normally made of complex resin-based, short fiber reinforced composites containing various friction modifiers (some types of brake pad also contain metallic material).

## 2.4 Related Research

**Sung-SooKang, and Seong-Keun Cho** [18] has studied Thermal deformation and stress analysis of disk brakes by finite element method. In this paper order to analyze the thermal characteristics of disk brakes, thermal deformation analysis and thermal stress analysis due to heat transfer was carried out through the finite element analysis for ventilated disk and solid disk. By comparing the maximum temperature in the braking process, the ventilated disk showed a lower temperature than the solid disk and effect of temperature increase and decrease, depending on the vent area generated in the flange part of the disk. The thermal deformation in ventilated disk type occurs in all directions by 0.1162mm, thermal distribution in the circumferential direction showed large deviation, about 0.017 mm due to the vents.

This study compared the thermal deformation and distribution between vented and solid disc brake. While vented disc have been dependent on the area of the flange, the research doesn't optimizing the shape of vented curves, number of vanes, offset vanes etc.

**MesutDuzgun** [19] has studied Investigation of thermo-structural behaviors of different ventilation applications on brake discs. In this study, the thermal behaviors of ventilated brake discs using three different configurations were investigated at continuous brake conditions in terms of heat generation and thermal stresses with finite element analysis, the results were compared with a solid disc.
Heat generation on solid brake discs reduced to a maximum of 24% with ventilation applications. The experimental study indicated finite element temperature analysis results in the range between 1.13% and 10.87%. However, thermal stress formations were higher with ventilated brake discs in comparison to those with solid discs.

**Ali Belhocine et al.,** (2012) [20] studied and analyzed the thermal behavior of the solid and ventilated brake discs of the vehicles using computing code ANSYS. The modeling of the temperature distribution in the disc brake was used to identify all the factors and the entering parameters concerned at the time of the braking operation such as the type of braking, the geometric design of the disc and the material used. The numerical simulation for the coupled transient thermal field and stress field was carried out by sequentially thermal-structural coupled method based on ANSYS to evaluate the stress fields and deformations which were established in the disc with the pressure on the pads. The results obtained by the simulation at braking operation of different time interval there is different values like at 1.8506 seconds its temperature distribution is 345.44 $C^0$at ventilated disc brake and 401.5 $C^0$ at solid disc.

**Zaid,M.A**.[21] Has conducted a study on ventilated disc brake rotor of normal passenger vehicle with full load of capacity. The study is more likely concern of heat and temperature distribution on the disc brake rotor. In this study, finite element analysis approach has been conducted in order to identify the temperature distribution and behaviors of disc brake rotor in transient response. ABAQUS/CAE has been used as finite elements software to perform the thermal analysis on transient response. Thus, this study provides better understanding on the thermal characteristic of the disc brake rotor and assists the automotive industry in developing optimum and effective disc brake rotor. Here finite element method was used to obtain the temperature behavior during braking and non braking conditions. A common finite element model was used in this analysis. Total time of the load cycles is 350 seconds which consists of ten cycles of braking operation and ten cycles without braking (idle).In each cycles has been found different values of temperature; the maximum value of temperature in all ten cycles is 543.9 $C^0$ which is still below the maximum service temperature of the typical grey cast iron where the temperature is approximately 550 $C^0$. Thus the yield maximum temperature is acceptable.

Summary of the related researches have been mentioned in these researches comparison of solid disc with vented one without considering the shape of the curve of vent and number of vanes.

But in this thesis the thermal analysis of vented disc brake and optimization of the shape of the vented curve of disc and also analyzes the effect of the curve of vent, number of vanes and vane angle is analyzed the effect of the shape of vanes on heat convection during emergency braking operation is analyzed.

# CHAPTER-THREE

# DATA COLLECTION, ANALYSIS AND MODELING

## 3.1 Calculating the slopes or gradient for selected Addis Ababa stations

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

Slope=$\frac{\text{Elevation difference}}{\text{Horizontal distance}}$, at station CMC. (3.1)

Elevation difference = $|2382.1 - 2379.2| = 2.9$ m

Horizontal difference = $[1.2\times10^{-2}$ m x 5000 m]/1 m=60 m.

Slope =$\frac{2.9}{60}$= 0.0483, $\tan^{-1} 0.0483=2.767^0$

Slope = 0.0483 = $2.767^0$

Using similar procedure to the above equation the slope of some Addis Ababa's stations having high gradient are illustrated in tabular form.

Table 3.1 slope [Source: Addis Ababa (E-W and N-S) rout light rail transit project from ERC, September 2009). [34]

| No | Station name | Gradient/slope | Station type |
|---|---|---|---|
| 1 | Ayat | 0.0333 [$1.905^0$] | Ground station [-down] |
| 2 | CMC | 0.0483 [$2.767^0$] | Ground station [+ up] |
| 3 | St.Michael church | 0.0185 [$1.058^0$] | Ground |
| 4 | Civil service college | 0.0178 [$1.019^0$] | Ground station |
| 5 | Megenagna | 0.018 [$1.031^0$] | Under Ground station |
| | | | |

| | | | |
|---|---|---|---|
| 6 | Mexico square | 0.0278 [$1.591^0$] | Elevation station |
| 7 | Coca cola | 0.0392 [$1.302^0$] | Ground station |
| 8 | Urael church | 0.0243[$1.394^0$] | Ground station |

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

## 3.2 Loading capacity of the rail vehicle

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

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

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

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

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

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

### 3.2.1 Main technical parameters

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

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

Average deceleration for emergency brake: $\geq 2.0$ m/s$^2$

Table 3.2 Number of passengers and weight [34]

| Condition | Number of passengers | Approximate car body weight (t) | Passenger weight (t) | Total (t) |
|---|---|---|---|---|
| AW0 | 0 | 44 | 0 | 44 |
| AW1 | 64 | 44 | 3.84 | 47.84 |
| AW2 | 254 | 44 | 15.24 | 59.24 |
| AW3 | 317 | 44 | 19.02 | 63.02 |

### 3.2.2 Vehicle weight

Table 3.3 Weights of vehicles [34]

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

Note: Take 60 kg as average weight of each passenger; tare weight of vehicles ≤44t [34].

## 3.3Railway Disc Braking Condition

In the process of braking of railway vehicles, it is necessary to define a model of a thermal analysis that describes the heating transfer of the heat generated by friction at surfaces which are in contact, between a railway disc and braking pads, through the disc and pads, as well as heat outflow of the whole braking system due to cooling of the surrounding air.
For those purposes, an 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 paper. [22]

Brake Application is defined as an application of the brake that results in a brake force being applied to the vehicle. Brake force is the force applied to the brake disc / pad / braking surface interface [23].

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

$$v(t) = v_o - at \tag{3.2}$$

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

$$\omega(t) = \frac{v(t)}{r_w} \tag{3.3}$$

$$\omega_o = \frac{v_o}{r_w} = \frac{19.44\ m/s}{0.33\ m} 58.922 \text{ rad/s} \tag{3.4}$$

Where,

$v$ = velocity of the vehicle

$v_o$= Initial running velocity of the vehicle

$\omega_o$= Initial angular velocity the wheel

ω = angular velocity of the wheel at any time

$r_w$= the radius of vehicle wheel

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

$$v_{disc} = \frac{r_{disc}}{r_w} v(t) \tag{3.5}$$

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

$$S_b = v_o t_b - \frac{1}{2} a t_b^2 \tag{3.6}$$

$$\boldsymbol{t_b = \frac{v_0}{a}} \tag{3.7}$$

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

$$\omega = \omega_0 \left(1 - \frac{t}{t_b}\right) \tag{3.8}$$

Where:

$t_b$ = brake time to stop the vehicle

$S_b$ = total brake distance

## 3.4 Load Determination

Braking from the maximum velocity of 70km/h to a standstill with the temperature of the surrounding area constant at 25 °C,

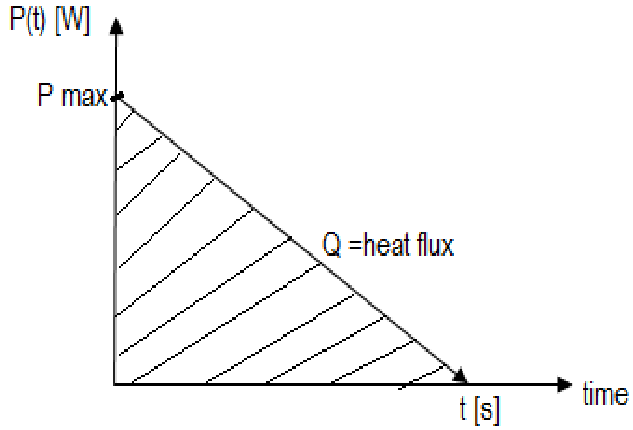



**Fig.3.1 Heat input during braking to a standstill.**

### 3.4.1 Braking Energy of the Vehicle

During braking process, available energy of the vehicle that must be 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 locomotive that must be dissipated by the work of the braking. Rail vehicles have important masses in rotation. Therefore, the contribution of rotational kinetic energy is taken in to account.

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

$$E_k = \frac{1}{2}Mv_0^2 + \frac{1}{2}I\omega_0^2 \quad (3.9)$$

Where:-

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

$M$ = Mass of the locomotive

$I$ = Polar inertial moment of rotating parts

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

$$I = \sum_{i=1}^{n} m_i r_i^2 \quad (3.10)$$

Where:-

$m_{i=}$represents the masses of the wheel set and brake discs

$r_i$= represents the rotation radius of the rotating parts from the center of rotation

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

$$E_k = \frac{1}{2}Mv_0^2 + \frac{1}{2}I\omega_0^2 = \frac{1}{2}Mv_0^2\left[1 + \frac{I}{M{r_w}^2}\right] = \frac{1}{2}(1.1)Mv_0^2 \tag{3.11}$$

The contribution of the rotating masses of the wheel set and discs is taken to be 10% the tare weight of the axle load of the locomotive [25]. The term $\left[\frac{I}{M{r_w}^2}\right]$ accounts to the rotational masses involved and its value equals 0.1.

The potential energy of the locomotive depends on the track gradient $\delta$ [mm/m] and on the travelled distance $S_b$[m].The exact definition of gradient is $i = \tan\alpha$ where $\alpha$ is angle of inclination.

$$E_p = M.\mathrm{g.}\sin\alpha \tag{3.12}$$

Where, g = 9.81m/s$^2$

**Table 3.4 data for calculating the heat flux. [34]**

| Item | Values |
|---|---|
| Mass of the vehicle – $M$ [kg] | 63020 |
| Start velocity – $v_0$[Km/h] or [m/s] | 70 or 19.44 |
| Deceleration – $a$ [m/s$^2$] | 1.1 for service brake |
| | 2 for emergency brake |
| Braking time – $t_b$[s], calculated from equation [3.7] | 17.67 for service brake |
| | 9.72 for emergency brake |
| Effective radius of the braking disc – $r_d$[m] | 0.1475 |
| Radius of the wheel – $r_w$[m] | 0.33 |
| Incline of the track – δ [‰] | 55 |

| | |
|---|---|
| Friction coefficient disc/pad – μ [/] | 0.28 |
| Surface of the braking pad $A_p$– [m$^2$] | 0.032 |
| Thickness of the disc Th-[mm] | 60 |
| Outer radius of the disc [mm] | 185 |
| Inner radius of the disc [mm] | 110 |
| Surface area of the braking disc $A_d$– [m$^2$] | 0.139 |
| Disc inner and outer radius [mm] | 110 & 185 |
| Pad inner and outer radius [mm] | 115 & 185 |

Therefore, the total available braking energy of the locomotive is the sum of the kinetic, rotational and potential for emergency brake at the gradient.

$$E_b = \frac{1}{2} M V_0^2 [1.1] + M.\mathrm{g}.sin\alpha \tag{3.13}$$

$E_b$ =13128667.1J at $\alpha$=3.153$^0$ and $E_b$ =13113903.18J at $\alpha$=1.394$^0$.

The total Braking energy is distributed on the number of the discs of the axles. Since the rail vehicle has 8 disc brakes, 4 at motor cars (MC) one disc for each axle and 4 at the middle there are two discs for each axle. Heat flux and force of the disc is calculated from the table 3.5 above

Force which work on brake disc [30].

$$\mathrm{F_{disc}} = \frac{\left[\left(\frac{1}{2} M V_0^2 [1.1] + M.\mathrm{g}.sin\alpha\right)\right]/8}{2\,\frac{\mathrm{r_{disc}}}{\mathrm{r_{wheel}}}\left[\mathrm{v_o t_b} - \frac{1}{2}\mathrm{a\, t_b^2}\right]} \tag{3.14}$$

F disc= 19429.4N for emergency at $\alpha$=3.153$^0$ and F disc= 19407.5N at $\alpha$=1.394$^0$

In braking system, the mechanical energy is transformed into heat energy. This energy is characterized by a total heating of the disc and the pads during the braking phase.

The total heat generated ($Q_{gen}$) in the brake system equals with the total mechanical energy ($E_b$) lost from the vehicle. Hence, considering the energy balance;

$$Q_{gen} = E_b \tag{3.15}$$

## 3.5 Heat dissipation

### 3.5.1 Heat conduction

There are two paths of heat conduction from the discs, one through the bearing assembly (which should be avoided) and another through the wheel carrier, which is the major conductive path. The heat flow can be estimated by Fourier's law of heat conduction as follows**:**

$$Q_{cond} = -kA\frac{dt}{dx} \tag{3.16}$$

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

### 3.5.2 Convection heat transfer

The major aim of designing brake discs is to improve the convection dissipation of disc braking systems. In operations of braking systems, convection is the most important mode of heat transfer, dissipating the highest proportion of heat to surrounding air. The current research focuses on heat convection of disc rotors. Convection is related to the heat flux by used of Newton's law of cooling.

$$\frac{q}{A} = h(T_s - T_f) \tag{3.17}$$

Where: q/A is heat flux out of the face

h= film coefficient

$T_s$= temperature of the face

$T_f$ =bulk fluid temperature

### 3.5.3 Heat Flux Entering the Disc

In the case of disc brake, the effective friction processes between the pads and the disc are extremely complex due to the fact that the present time brake pads, due to their composite structure **[**26**]**, do not have constant chemical-physic proprieties, the organic contained elements being subject of a series of transformations under the influence of temperature increase. The heat distribution between the brake disc and the friction pads is mostly dependent on material characteristics, among whom a major influence is due to the density ρ_(d,p) [kg/m3], the thermal conductivity k_(d,p) [W/m.K] and the specific heat C_(d,p) [J/kg.K] of the disc's (index d) and braking pad's materials respectively. Denoting $Q_d$ and $Q_p$[J] the heat quantities assumed by the disc and the braking pads respectively, one could be expressed in the following manner [27].

$$\frac{Qd}{Qp} = \frac{\sqrt{\rho d.Kd.Cd}}{\sqrt{\rho p.Kp.Cp}} \qquad 3.18)$$

Because the braking disc is not entirely covered by the friction pads, within computing we have to consider the ratio between the disc surface $Sd$ and the pad surface $S$p. denoting the ratio of heat's division between the disk and pads with:

$\varphi_c = \frac{Qd.Sd}{Qp.Sp} = \frac{\sqrt{\rho d.Kd.Cd}}{\sqrt{\rho p.Kp.Cp}} \cdot \frac{Sd}{Sp}$ and considering $Q$ [J] the heat quantity generated during the friction process, the heat quantities assumed by the pad sand by the disc are:

$$\varphi_d = Q \cdot \frac{\varphi c}{1+\varphi c} \qquad (3.19)$$

$$\varphi_p = Q \cdot \frac{1}{1+\varphi c} \qquad (3.20)$$

The brake disk assumes the most part of the heat, usually more than 90% [28], through the effective contact surface of the friction coupling. Considering the complexity of the problem and average data processing limited, one replaced the pads by their effect, represented by an entering heat flux.

The thermal analysis of the braking system of railway vehicles requires determination of the quantity of heat produced by friction, as well as the distribution of this energy between the railway disc and the braking pads. The disc material is gray cast iron (GFC) with high carbon content [25], with good thermo physical characteristics and the brake pad has an isotropic elastic behavior whose thermo mechanical characteristics adapted in this simulation given in table 3.6.

Generally, the thermal conductivity of material of the brake pads is smaller than that of disc ($k_p$< $k_d$). We consider the energy from brake application is converted into heat and transferred to the disc and pad approximately 90% and 10% respectively. This ratio normally is called the proportion of heat transferred to disc (γ = 0.90). The rate of heat generation is:

$$\dot{Q}_d(t) = \gamma(2F_{disc})v_{disc} = (0.90)(2F_{disc})\frac{r_d}{r_w}(v_o - at) \tag{3.21}$$

Table 3.5 thermo-elastic properties of disc & pad materials

[Take from ANSYS software property of cast iron]

| Material properties | pad | Disc |
|---|---|---|
| Thermal conductivity, $k$ (w/m. °C) | 5 | 52 |
| Density, $\rho$ (kg/m3) | 1400 | 7200 |
| Specific heat, $c$ (J/Kg. °C) | 1000 | 447 |

At the contact surface between the railway disc and the braking pads, the braking process produces heat according to expression (3.15).The heat dissipation from the free surface of the disc to the surrounding air is described by both convection and radiation:

$$Q_{diss} = -h\left(T - T_{ref}\right) - \varepsilon\sigma(T^4 - T_{ref}^4) \tag{3.22}$$

In this equation,

$h$ equals the convective film coefficient,

$\varepsilon$ is the material's emissivity,

$\sigma$ is Stefan-Boltzmann constant and

$T_{ref}$is the temperature of the surrounding air

Convective heat transfer between the air and the brake rotor surfaces is the primary mean of heat rejection. Radiation heat transfer is less important especially for the low surface temperatures and in this paper; its effect is neglected [29].

For this study the value of heat transfer coefficient is calculated with CFD-cfx software package in chapter three, for more accurate.

Instant heats flux entering on the side of the braking disc [30]:

$Q_d(t) = F_{disc}.\ V_{disc}(t) = F_{disc}.\ \frac{rd}{rw}\ (v_0 - a.\ t)$ (3.23)

$\dot{\mathbf{Q}}_{\mathbf{d}}(\mathbf{t}) = \mathbf{303882.8812 - 31263.6791t}\ [\mathbf{W}]$ **,** at F disc= 19429.4N.

$\dot{\mathbf{Q}}_{\mathbf{d}}(\mathbf{t}) = \mathbf{303540.3573 - 31228.43182t}[\mathbf{W}]$ **, at** F disc= 19407.5N.

The heat flux at calculated above as the function of time is distributed to the effective area of the surface of the disc, $A_d = 2\pi(r_0^2 - r_i^2);2\pi(0.185^2 - 0.110^2) = 0.139\ m^2$.

### 3.5.4 Pressure determination for the brake caliper

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

$$p_{pad} = \frac{F_{disc}}{2*\mu\ Ap} = \frac{1429.4}{2x0.28x0.032} = 1.08Mpa \quad (3.24)$$

$$p_{rotor} = \frac{F_{disc}}{2*\mu\ Ar} = \frac{1429.4}{2x0.28x0.139} = 0.25Mpa$$

Where: $p_{pad}$ is pressure of pad, Ap is surface area of pad

$P_{rotor}$ is pressure of disc rotor, $A_r$ is surface area of rotor

$\mu$ is friction coefficient disc/pad.

## 3.6. Modeling software

There are different software's available for modeling some of them are: Solid works, CFD GAMBIT, ANSYS work bench and CATIA etc.

CATIA V5 (computer aided three dimensional interactive application) a multi platform is used as the modeling tool in this project.

### 3.6.1 CATIA V5 part Modeling

The CATIA V5 is a 3-D parametric solid modeler with both part and assembly modeling capabilities. CATIA V5 to model piece parts and then combines them into more complex assemblies. With CATIA V5 a part is designed by sketching its components shapes and defining their size shape and interrelationships. Since CATIA V5 has parametric features, you can change one feature and all related features are automatically updated to reflect the change and its effects throughout the part. It can be used to create angular shaped part, to which 3D surface can be applied to create hybrid parts consisting of mixture of angular and curved shapes. This provides the ability to create model designs with shapes of varying types.

Assemblies can be created from parts, either combined individually or grouped in subassemblies. The CATIA V5 builds these individual parts and subassemblies into an assembly in a hierarchical manner according to relationships defined constrains. As in part modeling, the parametric relationship allows you to quickly update an entire assembly based on a change in one of its parts. And save the model as IGS profile for importing to the ANSYS work bench for finite element analysis.

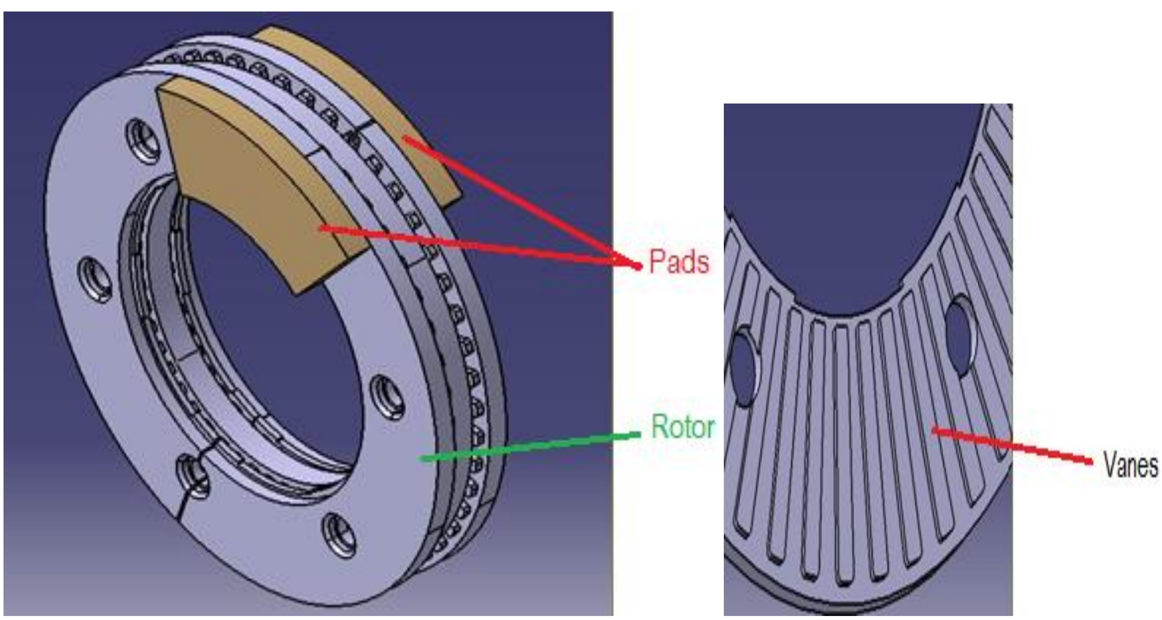



**Fig 3.2 model of disc Rotor and pad assembly**

## 3.7 Finite element analysis

### 3.7.1 Introduction

Finite element method is a numerical method used for solving engineering problems. Because it is the mathematical representation of physical problems& it gives the approximate solution & also applicable even if physical prototype not available [30]. ANSYS is FEA software developed by ANSYS Inc. USA. ANSYS involves three stages preprocessing, solution & post processing for solving problems. Preprocessing stage involves the preparation of FEM model, element type, real constant, material property & discretization. In Solutions stage ANSYS software automatically generates matrices that describe the behavior of each element, assemble them & computes the unknown values of primary field variables such as displacement, temperature etc. [30] Post processing stage involves the presentation of results.

### 3.7.2 Definition of Material

The material of locomotive disc material is gray cast iron (GFC) ISO standard with high carbon content [31], with good thermo physical characteristics and the brake pad has an isotropic elastic behavior, whose thermo-mechanical characteristics adopted in this simulation in the transient analysis is recapitulated in table 4.1below.

Table 3.6 Material Properties of Grey Cast Iron Disc [31]

| Material properties | Grey Cast Iron Disc |
|---|---|
| Thermal conductivity - *k* [W/m °C] | 52 |
| Density – ρ [kg/m3] | 7200 |
| Specific heat - *c* [J/Kg °C] | 447 |
| Poisson’s ratio *ν* [/] | 0.28 |
| Elastic modulus - *E* [G Pa] | $1.52X10^{11}$ |
| Compressive ultimate strength [Mpa] | 820 |

## 3.8 Modeling ANSYS in CFD

The heat dissipation and thermal performance of ventilated brake discs strongly depends on the aerodynamic characteristics of the air flow through the rotor passages. In this thesis, the thermal convection is analyzed using an analytical method, and the velocity distribution and temperature contours are determined. Then numerical models for different vane shape rotors are generated and numerical simulations are conducted to determine the desired parameters. To analyze more realistic vane rotor models, commercial CFD software packages, Fluent are used to simulate the heat flux rate, air flow rate, velocity distributions, temperature contours, and pressure distributions inside the rotors. Furthermore, sensitivity studies have been performed, to determine the effects of a different number of vanes, curve shape of the vane and various angles of vanes. And numerical simulations based on cfx.

The finite volume method consists of three stages; the formal integration of the governing equations of the fluid flow over all the (finite) control volumes of the solution domain, then discretization involving the substitution of a variety of finite-difference-type approximations for the terms in the integrated equation representing flow processes such as convection, diffusion and sources. This converts the integral equation into a system of algebraic equations, which can then be solved using iterative methods [32]. The first stage of the process, the control volume integration, is the step that distinguishes the finite volume method from other computational fluid dynamics (CFD) methods. The statements resulting from this step express the ‘exact’ conservation of the relevant properties for each finite cell volume. This gives a clear relationship between the numerical analogue and the principle governing the flow. To enable the modeling of a rotating body (in this case the disc) the code employs the rotating reference frame technique. The solution scheme employees the κ-e model with scalable wall function and sequential load steps.

The CFD models were constructed and were solved using ANSYS-CFX software package [33].The model applies periodic boundary conditions on the section sides. As the brake disc is made from sand casted grey cast iron.

The disc model is attached to an adiabatic shaft whose axial length spans that of the domain. Air around the disc is considered to be 25 ° C, and open boundaries with zero relative pressure were used for the upper, lower and radial ends of the domain. Materials data were taken from ANSYS material data library for air at 25 ° C. Reference pressure was set to be 1 ATM, turbulence intensity low and turbulent model used was k-e. (Fig.3.3)

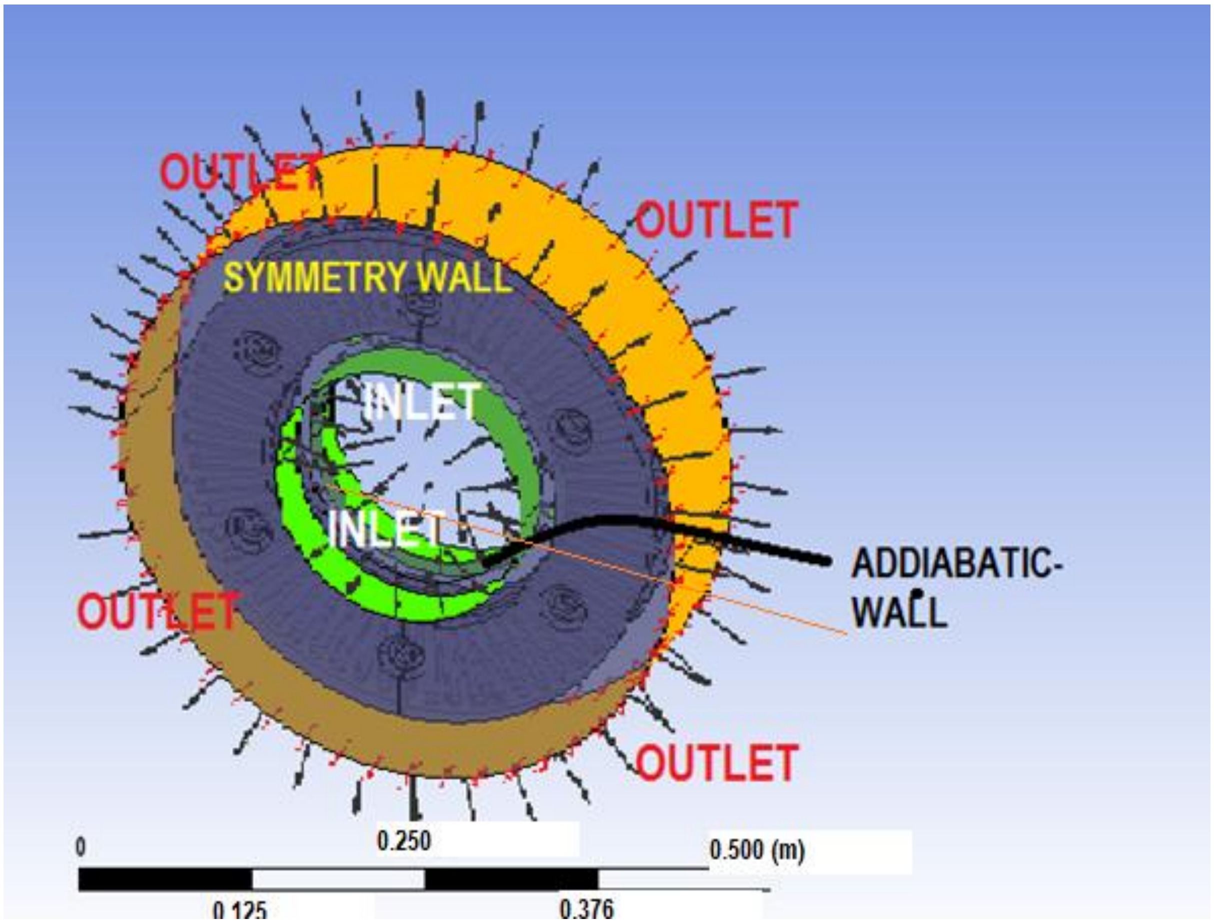



**Fig.3.3 brake disc CFD model**

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. Afterwards the heat transfer coefficients considering convection were calculated and organized in such a way, that they could be used as a boundary condition in thermal analysis. Averaged heat transfer coefficient had to be calculated for all disc using ANSYS CFX Post as it is indicated.

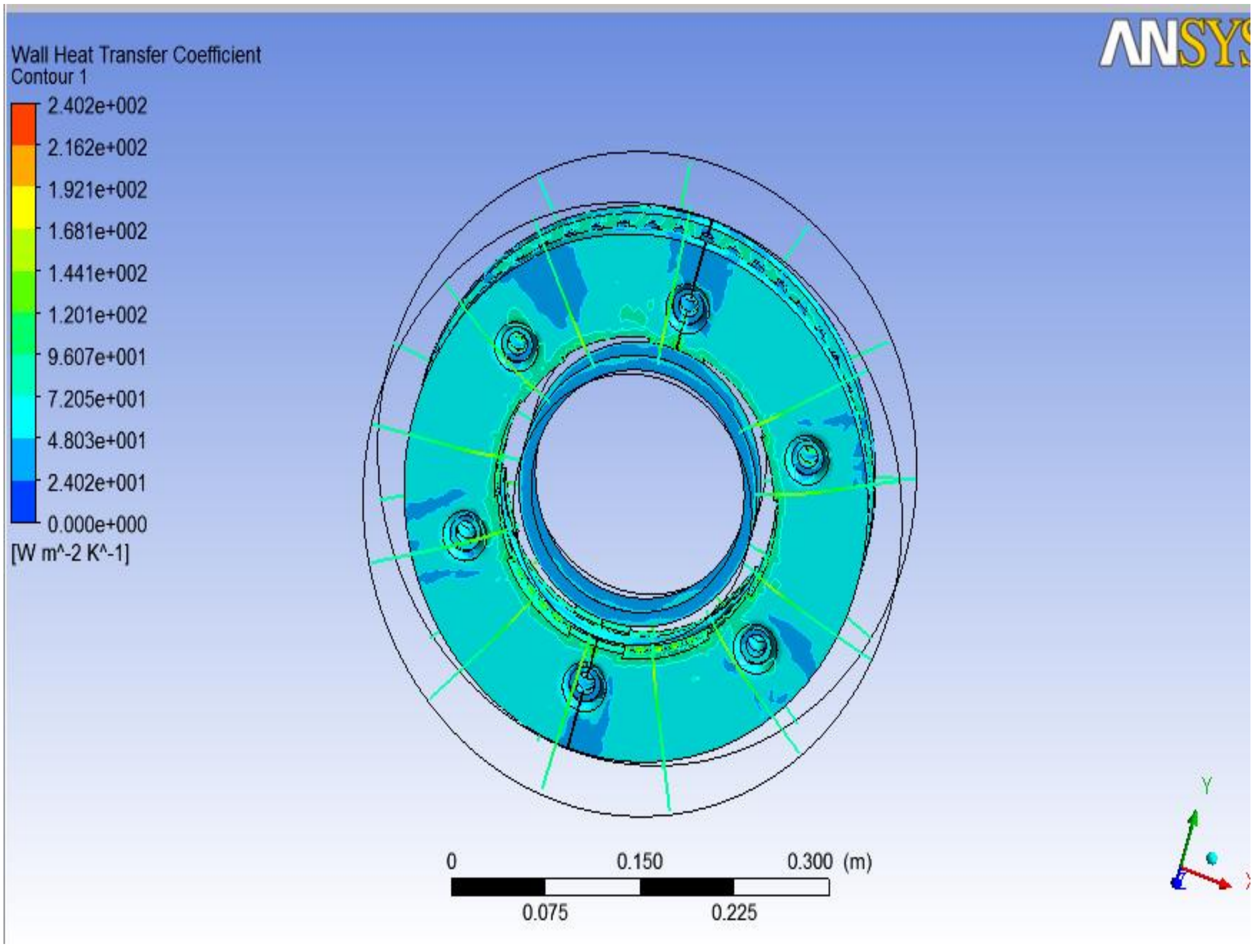



**Fig 3.4 distribution of heat transfer coefficient on a ventilated disc with vane number 50 on transient**

With the same procedure the wall heat transfer coefficients of the disc at different vane number, vane angle and at different curves of the vane is as follow:

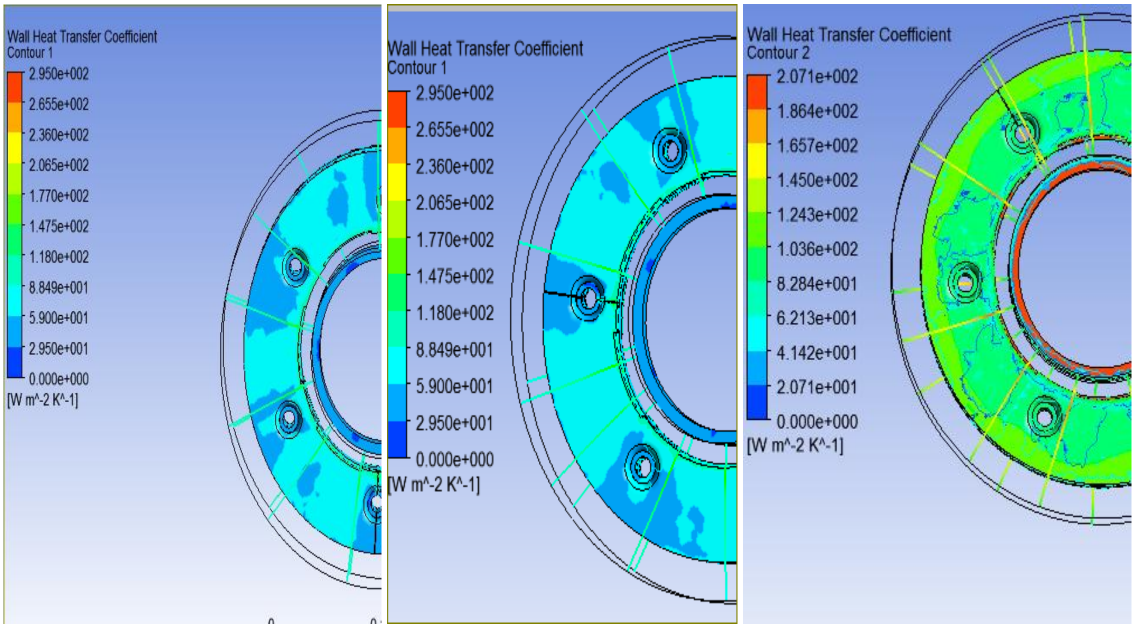



**Fig 3.5 distribution of heat transfer coefficient on a ventilated disc with vane numbers 60 70 and 80 on transients**

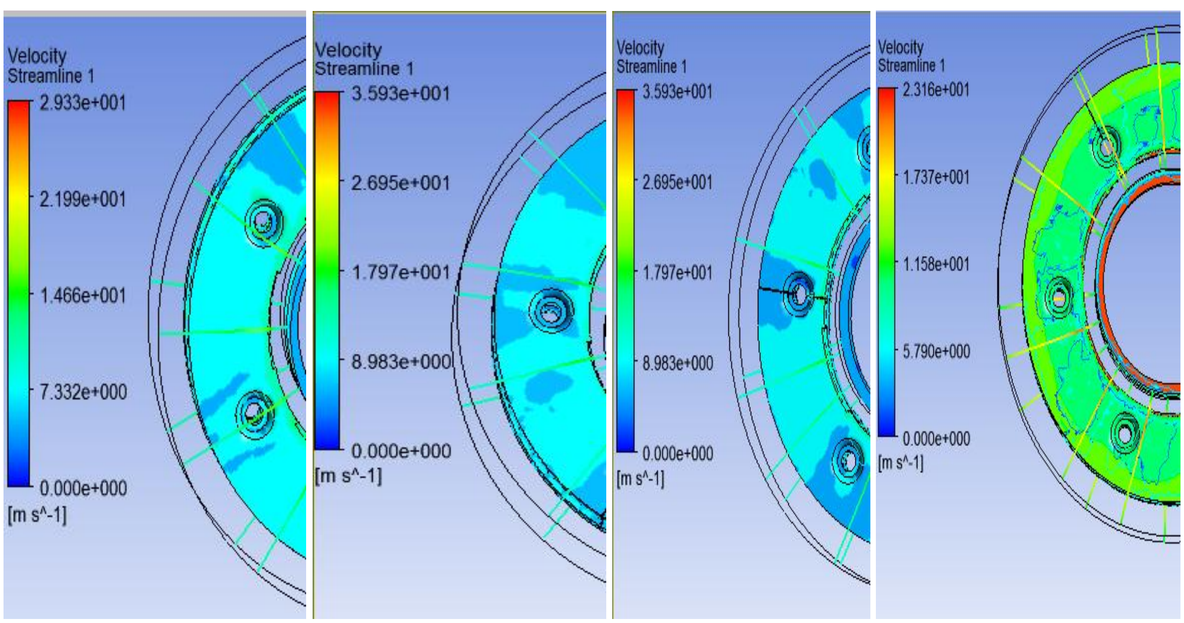



**Fig 3.6 velocity on a ventilated disc with vane numbers 50, 60, 70 and 80 on transients**

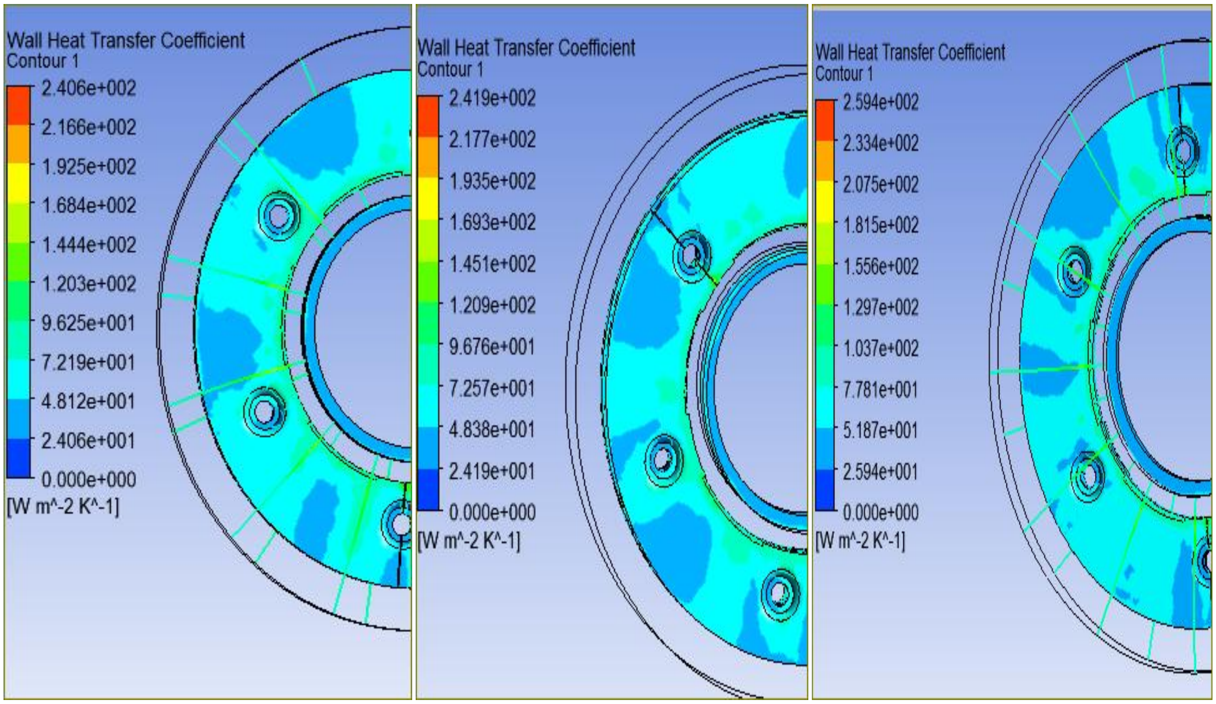



**Fig 3.7 distribution of heat transfer coefficient on a ventilated disc with vane angle 5, 10 and 20 on transients**

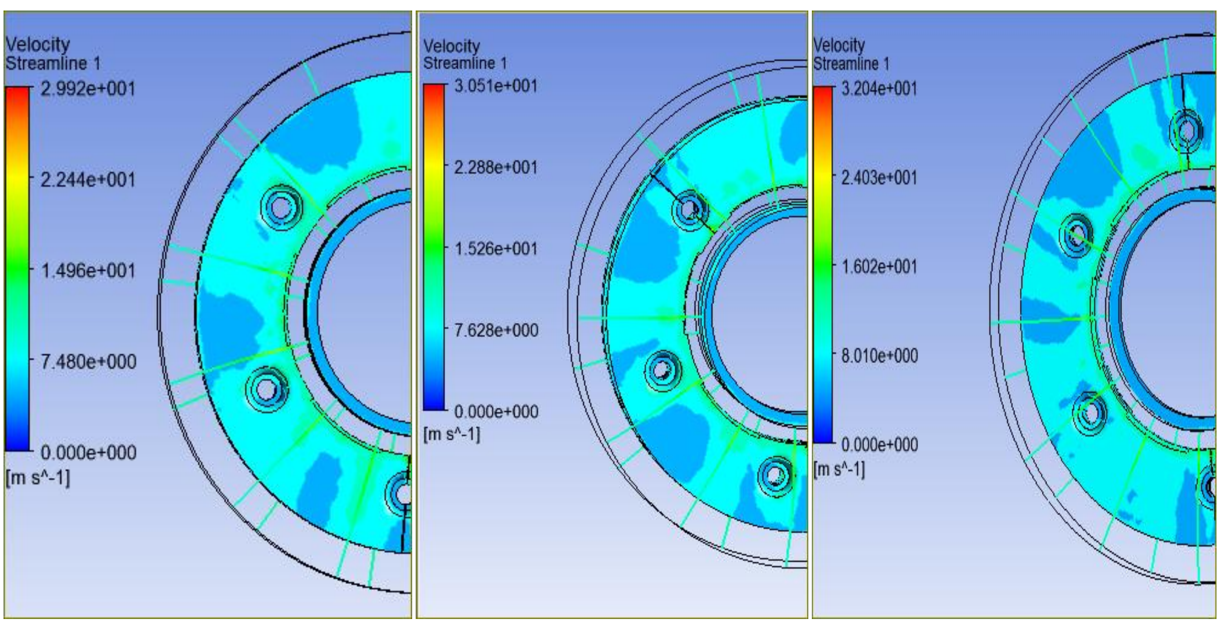



**Fig 3.8 velocity on a ventilated disc with vane angles 5, 10 and 20 on transients**

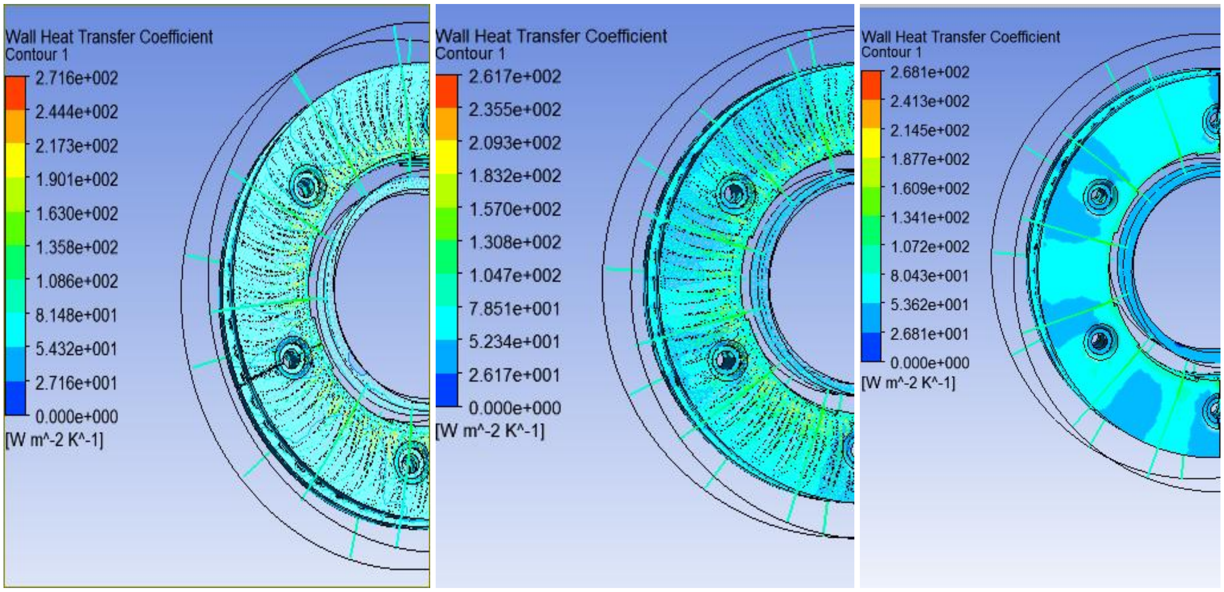



**Fig 3.9 distribution of heat transfer coefficient on a ventilated disc with curve vane radius 80,100 and 120mm on transients**

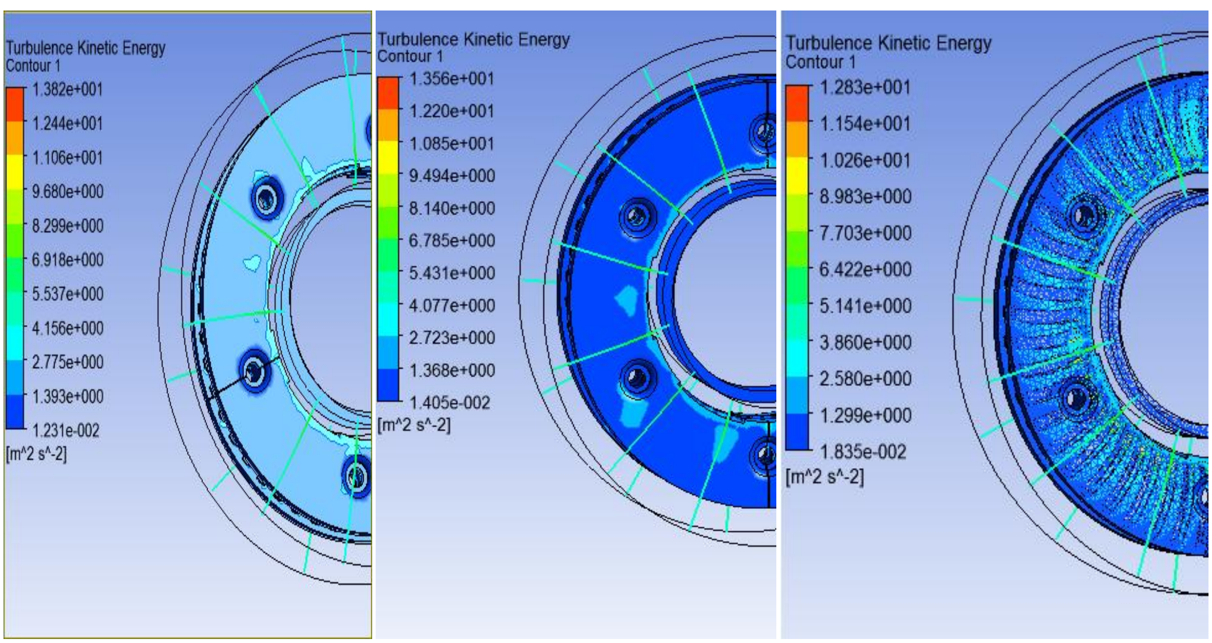



**Fig 3.10 Turbulence kinetic energy on a ventilated disc with curve vane radius 80, 100 and120 on transients**

### 3.8.1 Assumptions

- 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 equals.
- Radiation is neglected by virtue of short braking time and hence relatively low temperature;
- The wear on the contact surface is negligible.
- Consider the disc-pad assembly is open to atmospheric air.
- Assume the enclosure of the model is taking 25mm

# CHAPTER-FOUR

# ANSYS RESULTS AND DISCUSSIONS

## 4.1. 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, the size of the mesh of the model one factor to the accurate results. Here fine type mesh is used. The dimension of the in closure take as assumption 25mm, as the in closure becomes larger and larger the value of the coefficient wall heat transfer is become less and less and as the in closure become less and less the heat at the sounding of the disc high and then if affect to the disc then in this study has been taken 25 mm this is becomes approach to the software ANSYS default value.

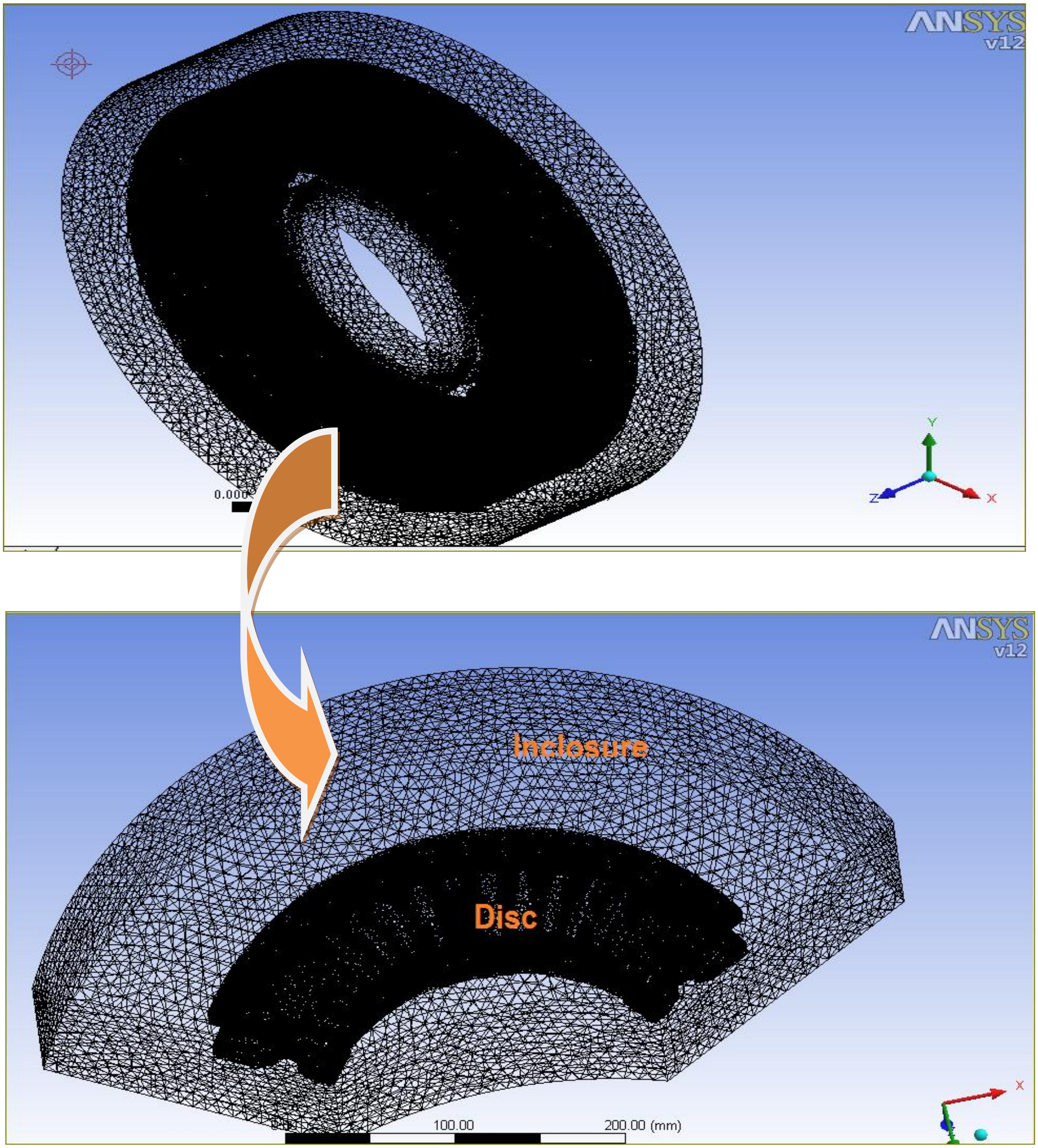



**Fig 4.1 Meshing of vented disc, nodes 374478 and elements1958550**

## 4.2 Boundary Conditions

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

- Total emergency brake time, $t_b$= 9.72 [s]
- Increment of initial time =0.0162[s]
- Minimum initial time =0.00162 [s]
- Maximum initial time =0.162 [s]
- Initial temperature of the disc=$30^0$c
- Material of the disc : Gray cast iron FG 15
- Convection: one introduces the value of the wall heat transfer coefficient (h) obtained for each surface in the shape of the disc with CFX soft ware package.
- Heat flux: has been calculated on table 4.1

## 4.3 Thermal Analysis

A Thermal analysis calculates the temperature distribution and related thermal quantities in a system or component. A Transient thermal analysis determines the temperature distribution and other thermal quantities under conditions that vary over a period of time.

Therefore rather than steady state thermal analysis, for this paper work transient thermal analysis is used. Based on equation [3.17] heat flux at the different time interval is:

Table 4.1 value of heat flux at function of time per applied area of the disc

| **Time in [s]** | **Heat flux in[ W/m$^2$]** | |
|---|---|---|
| | **At emergency brake-1** | **At emergency brake-2** |
| 0 | 2186207.778 | 2183743.578 |
| 1.62 | 1821839.72 | 1819786.315 |
| 3.24 | 1457471.661 | 1455829.052 |
| 4.86 | 1093103.603 | 1091871.789 |
| 6.48 | 728735.5441 | 727914.5259 |
| 8.1 | 364367.4855 | 363957.263 |
| 9.72 | 0 | 0 |

After applying the necessary boundary conditions the following results will be obtain under different circumstances during emergency brake in the high gradient stations with different vane numbers, and different shape of vanes, by combining the fluid flow (cfx) with transient thermal and transient structural as follow:

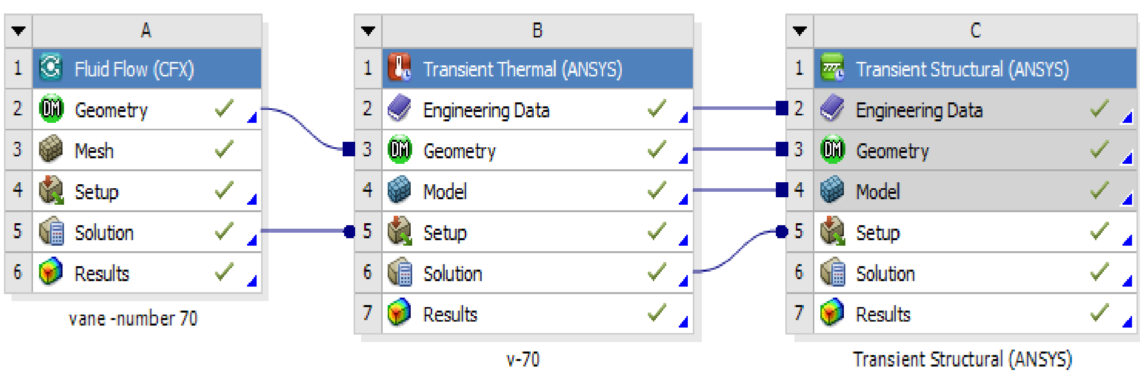
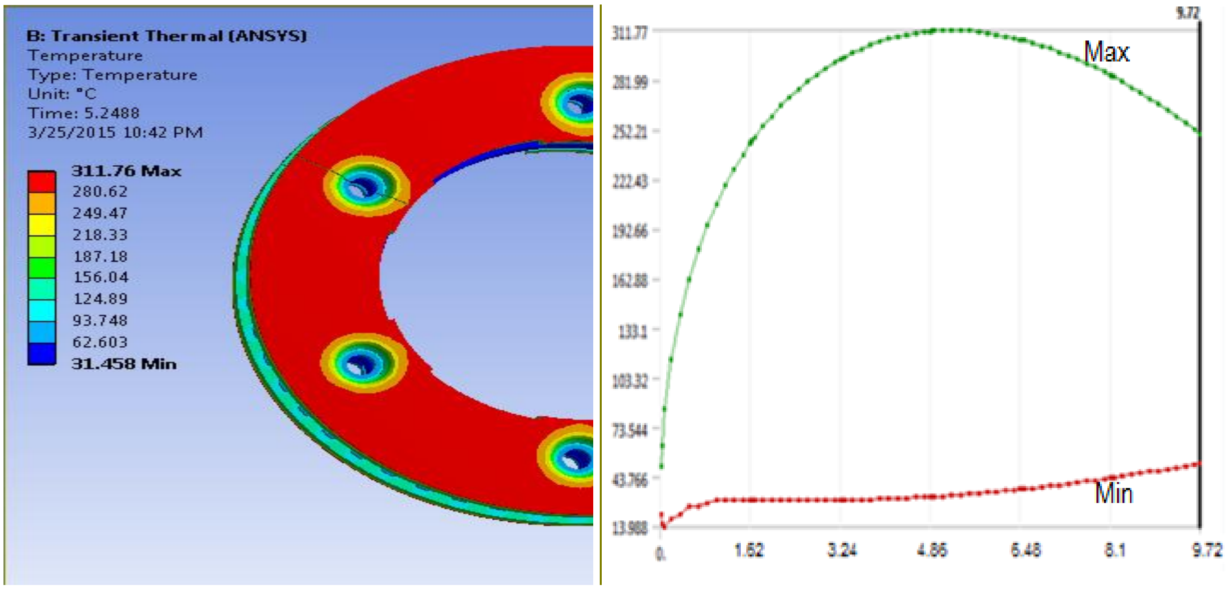
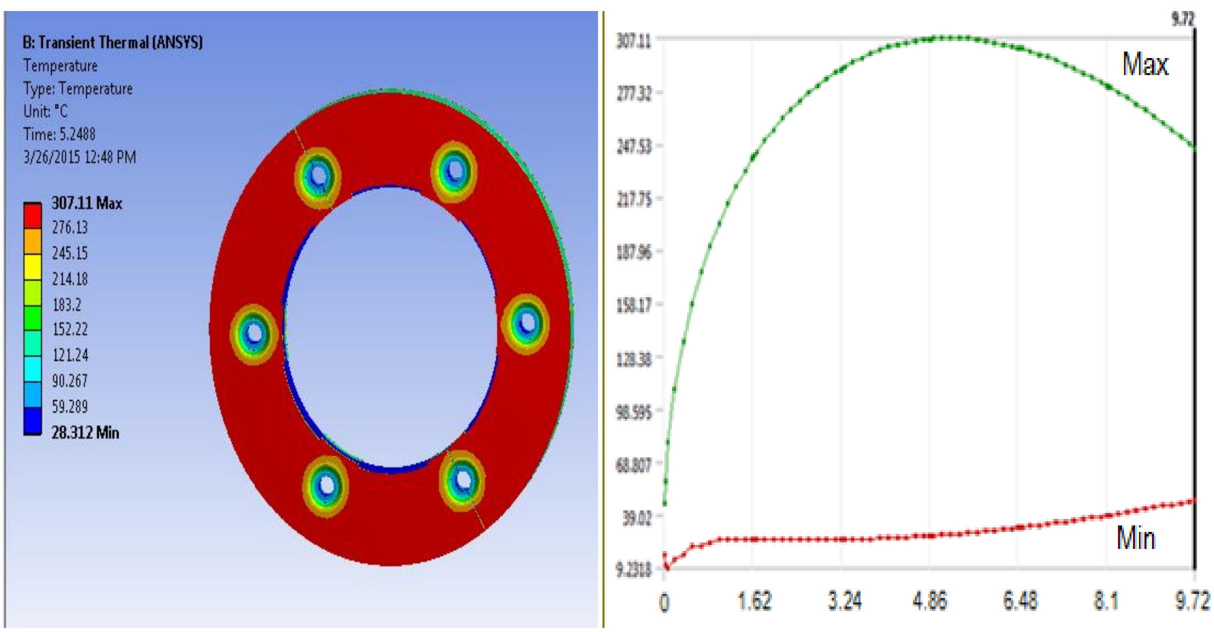



**Fig 4.2 combined of fluid flow (cfx) with transient thermal and structural**

### 4.3.1 ANSYS results based on number of vanes




v-50




V-60

**Fig 4.3Temperature distribution of ventilated disc of vane numbers 50 and 60**

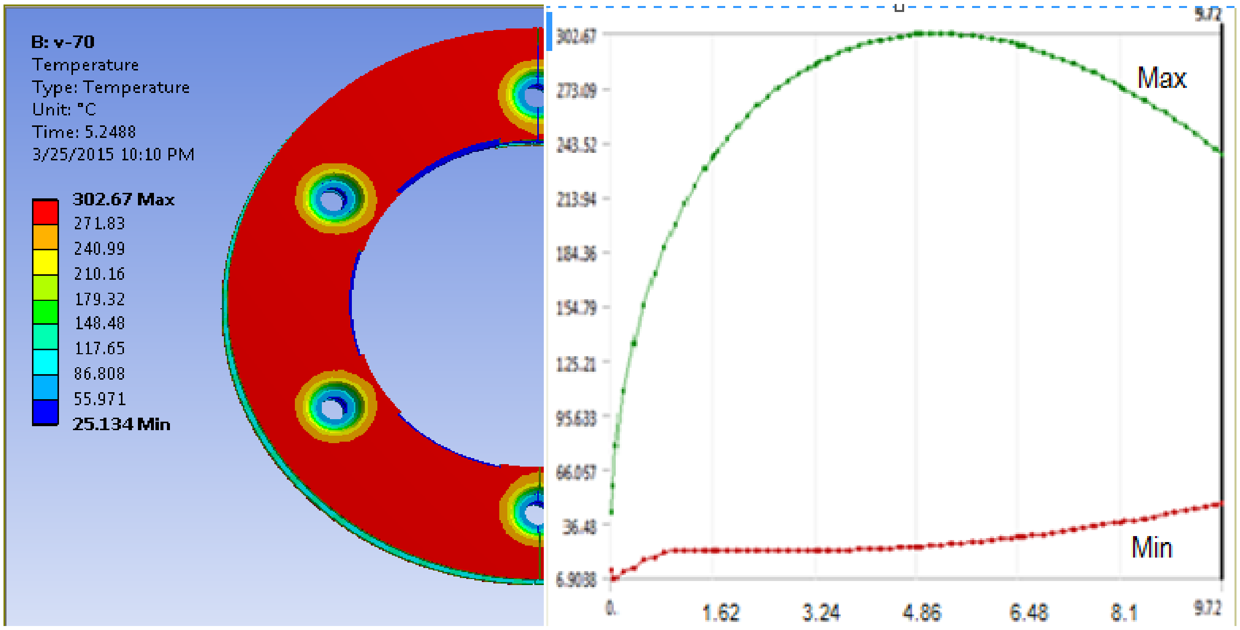



**Fig 4.4 Temperature distribution of ventilated disc of vane numbers 70**

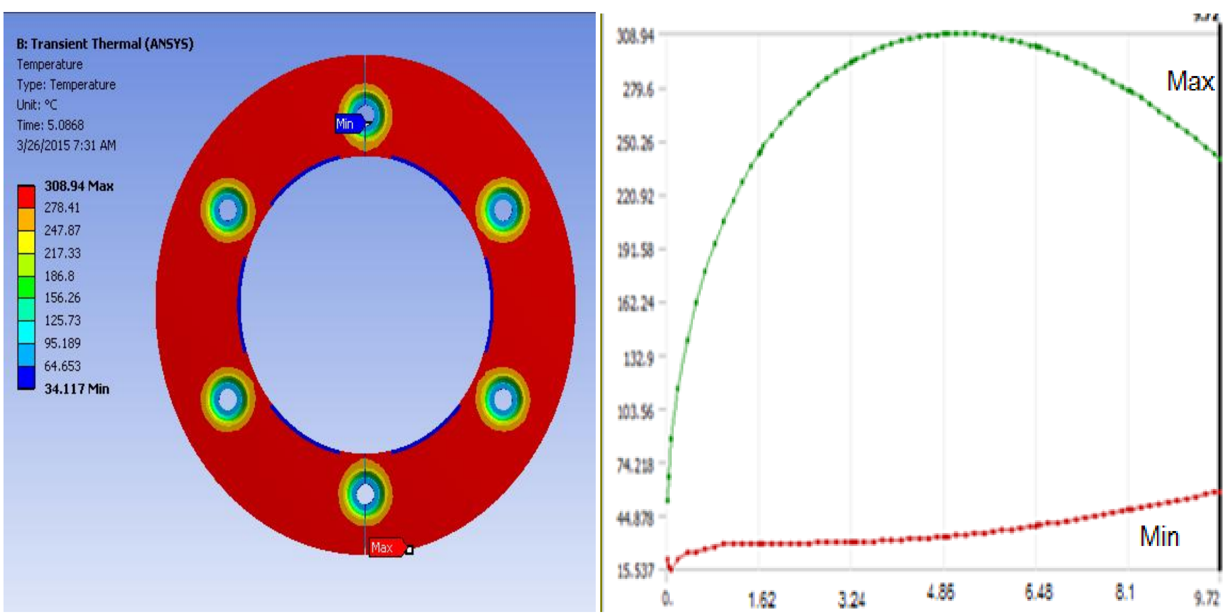



**Fig 4.5 Temperature distribution of ventilated disc of vane numbers 80**

**4.3.2 ANSYS results based on vane angle at the straight vane.**

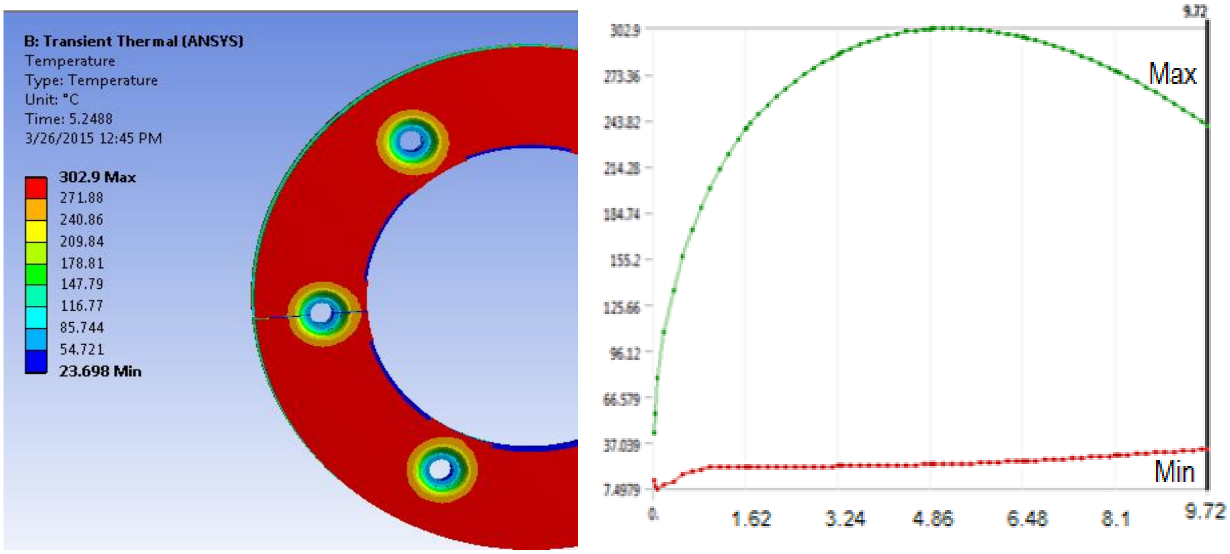



**Fig 4.6 Temperature distribution of ventilated disc at vane angle 5 with vane number 50**

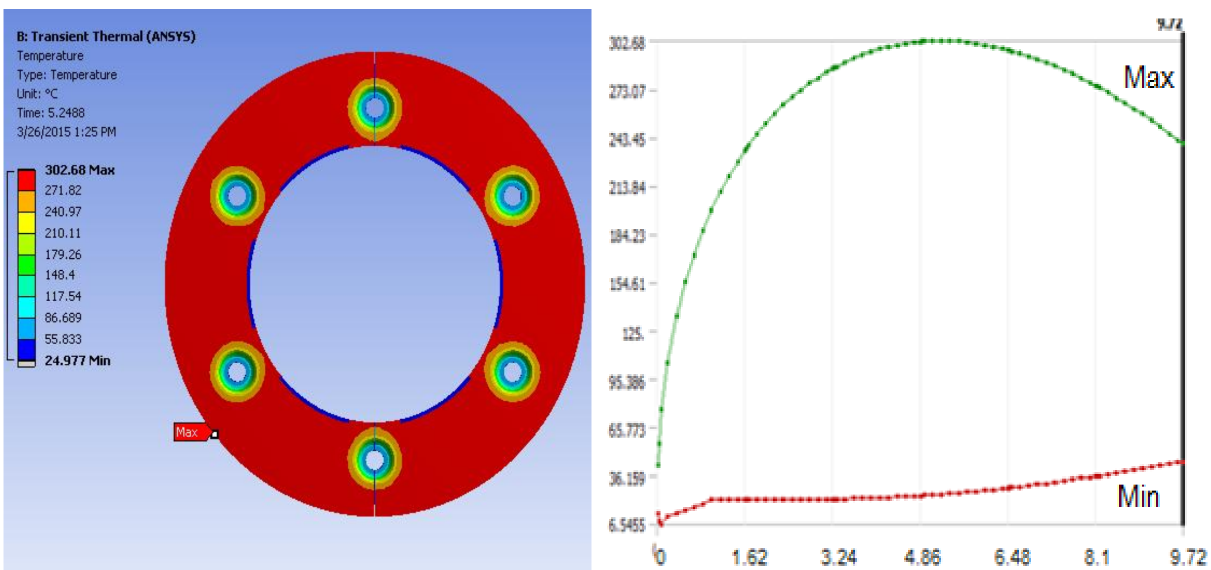



**Fig 4.7Temperature distribution of ventilated disc at vane angle 10 with vane number 50**

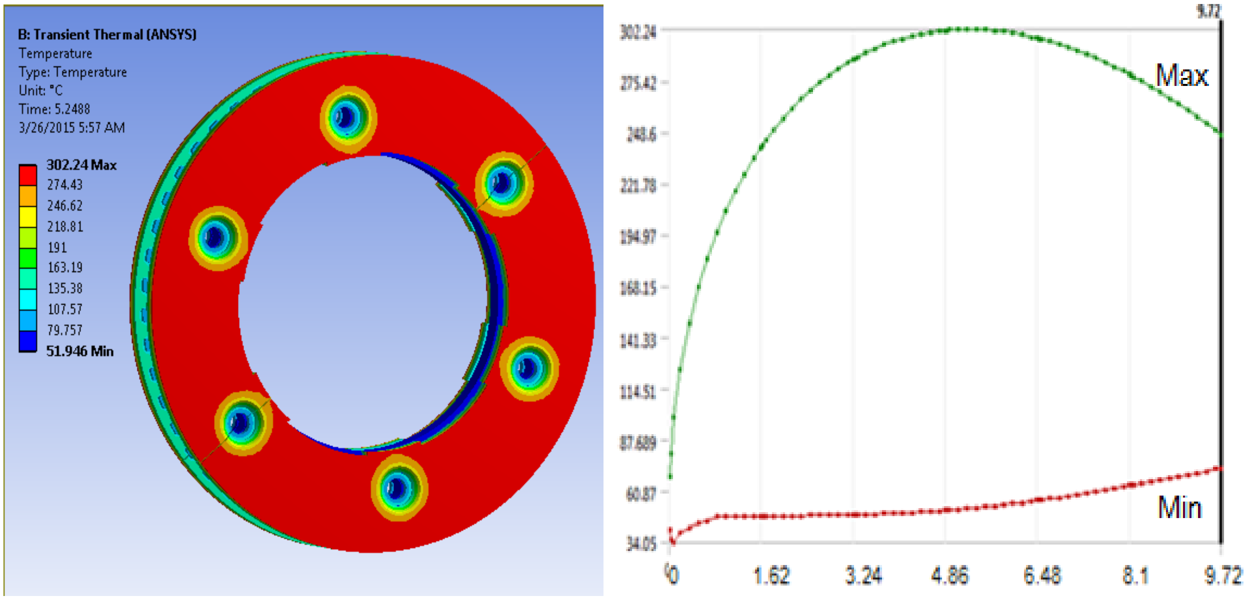



**Fig 4.8 Temperature distribution of ventilated disc at vane angle 20 with vane number 50**

### 4.3.3 ANSYS results based on curve vane radius

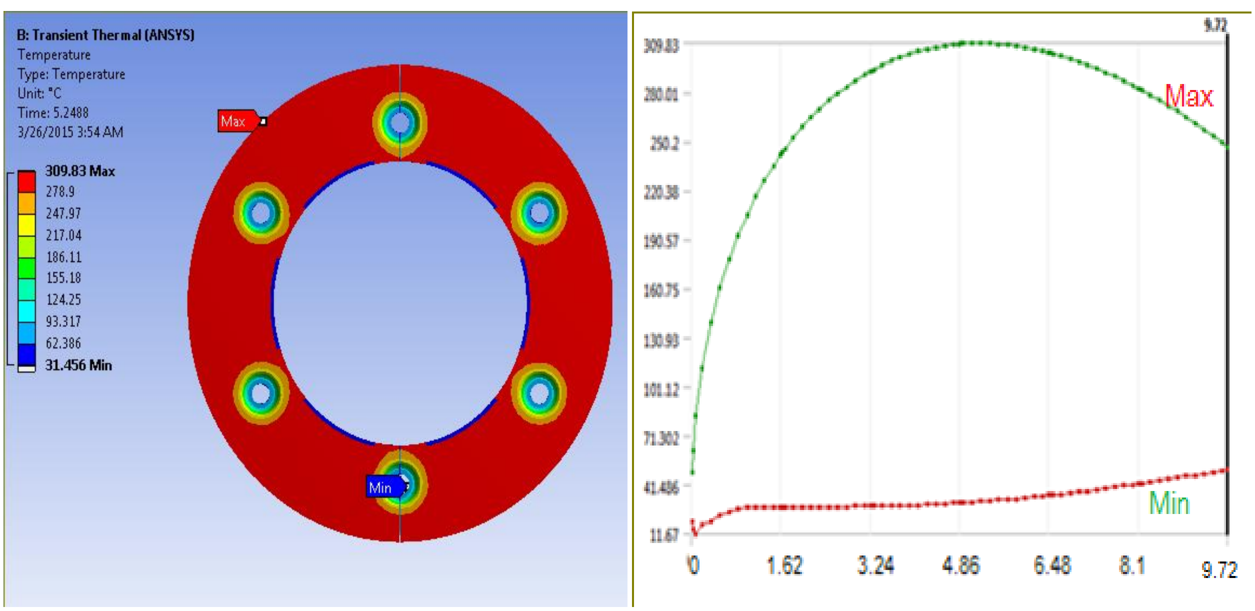



**Fig 4.9 Temperature distribution of ventilated disc at curve vane radius 60 mm**

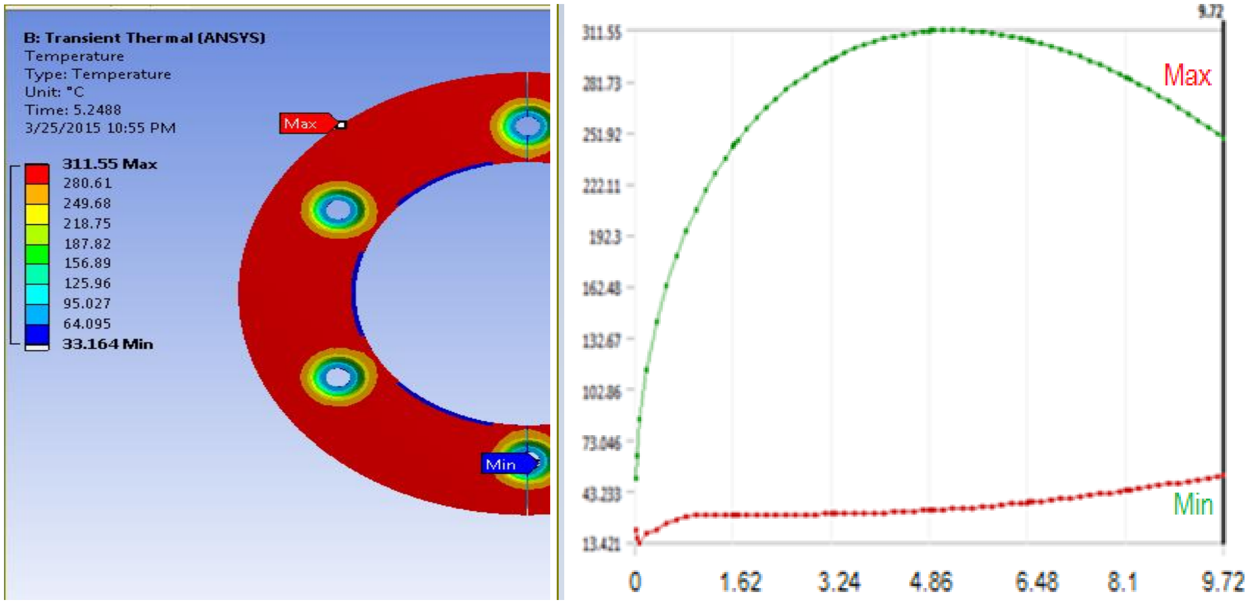



**Fig 4.10Temperature distribution of ventilated disc at curve vane radius 80mm**

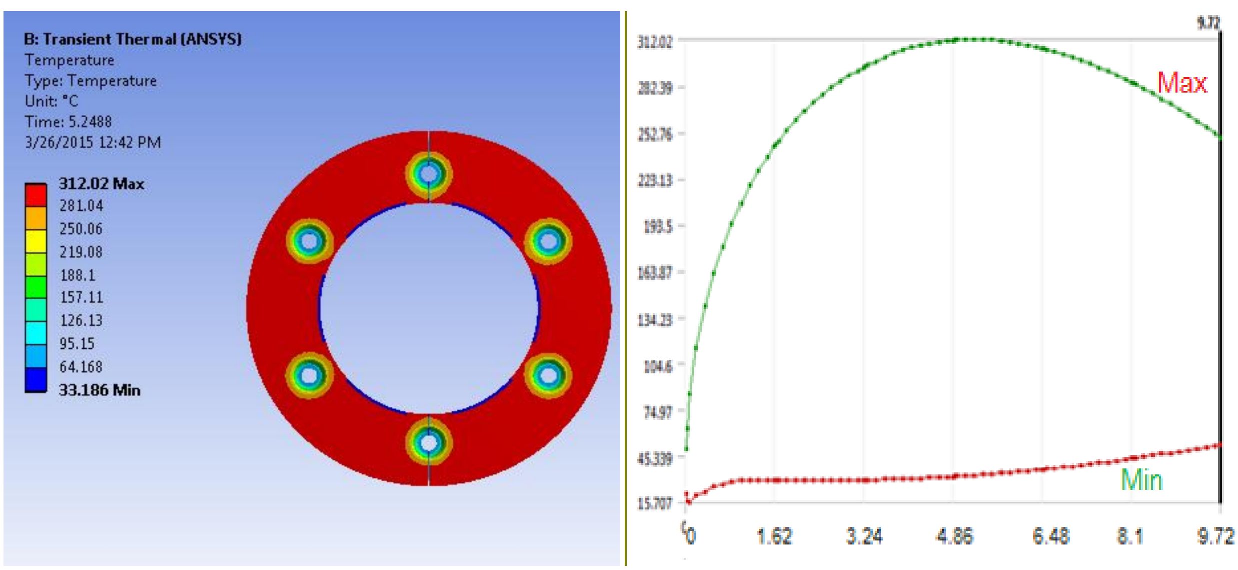



**Fig 4.11 Temperature distribution of ventilated disc at curve vane radius 100 mm**

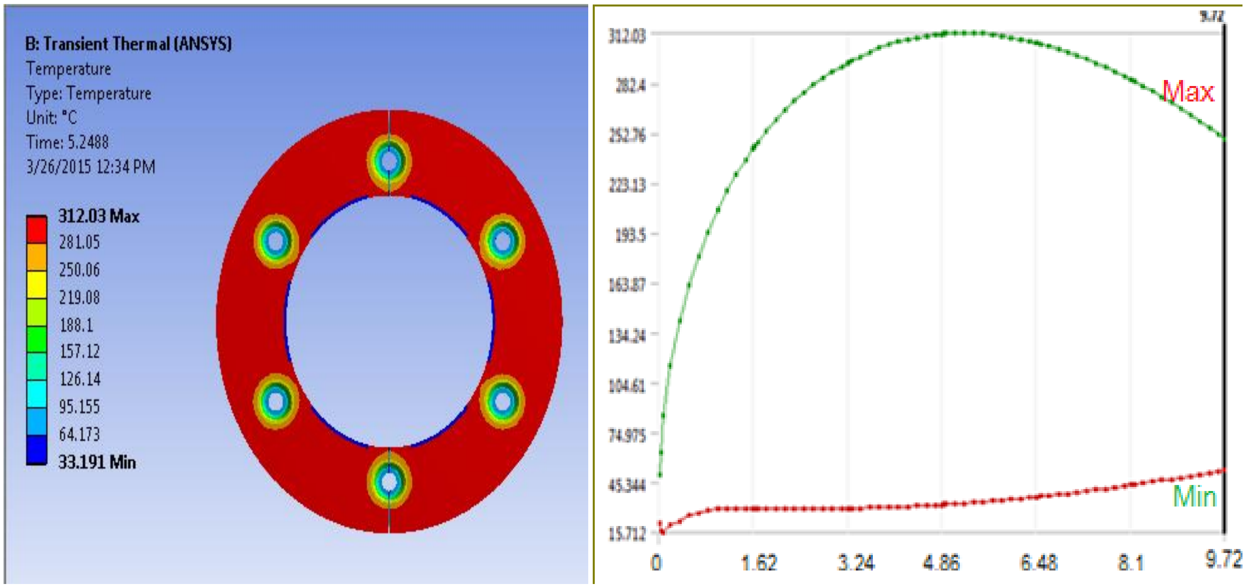



**Fig 4.12 Temperature distribution of ventilated disc at curve vane radius 120 mm**

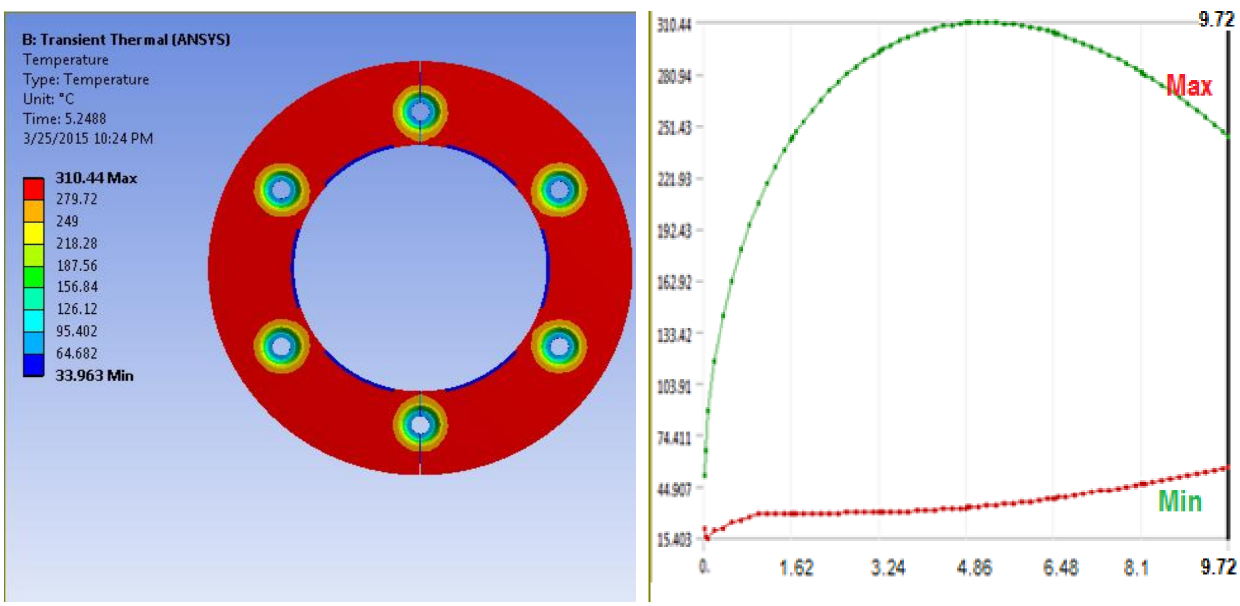



**Fig 4.13 Temperature distribution of ventilated disc vane number 60 at vane angle 5**

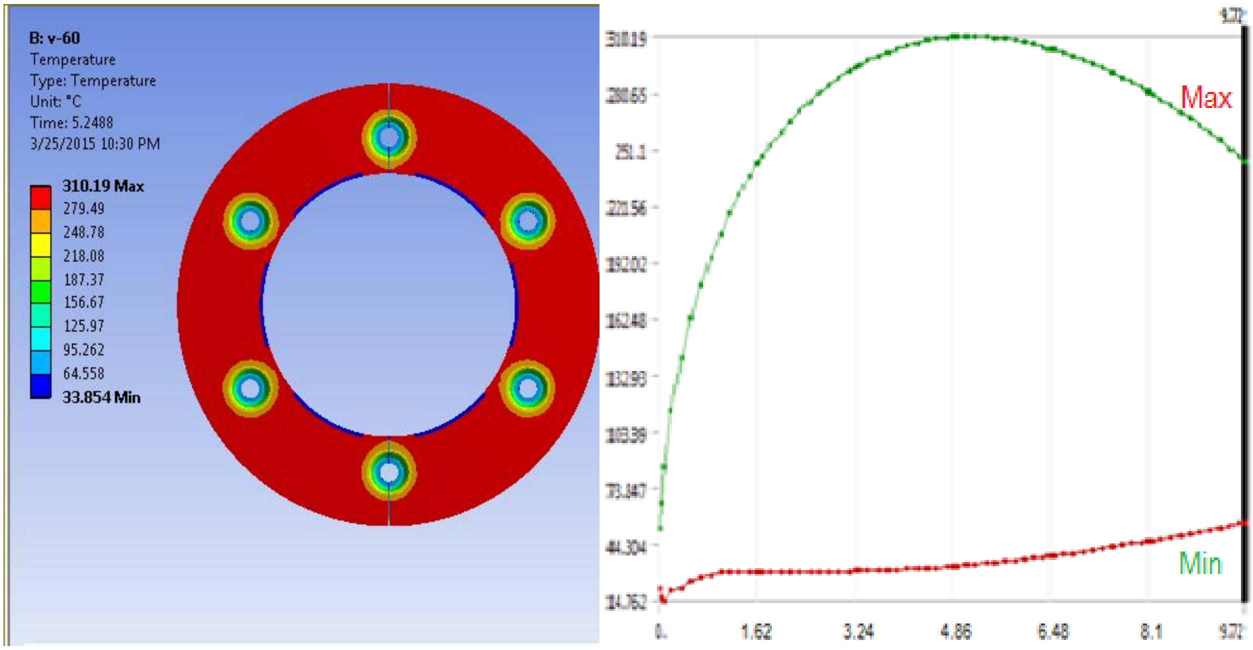



**Fig 4.14 Temperature distribution of ventilated disc vane number 60 at vane angle 20**

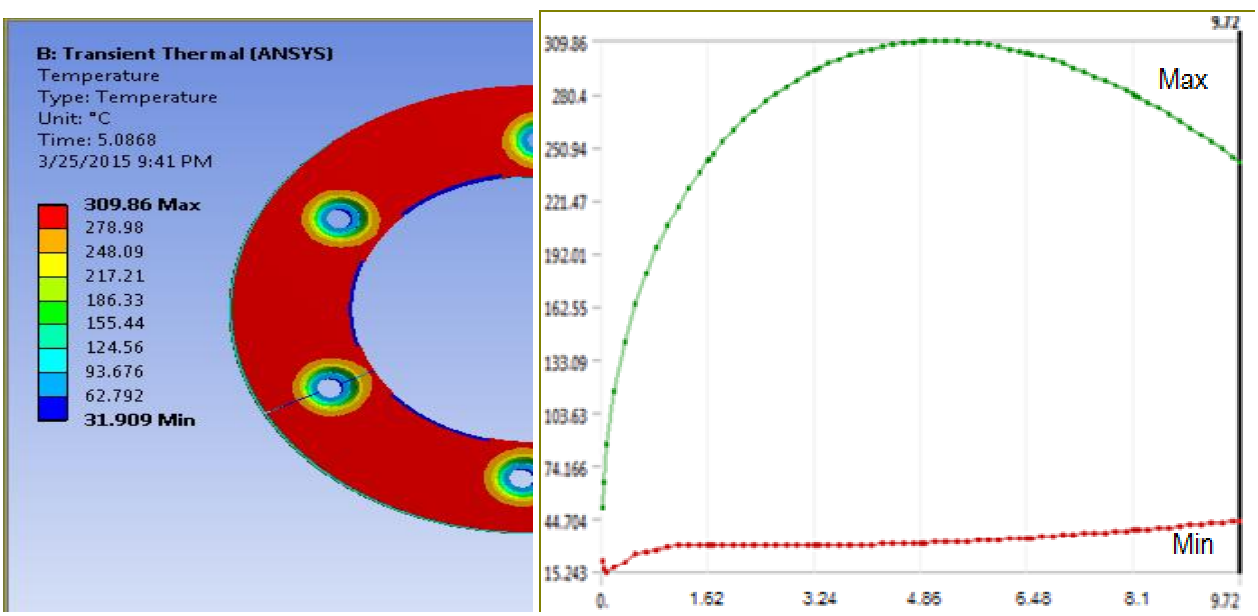



**Fig 4.15 Temperature distribution of ventilated disc at vane number 70 at vane angle 5**

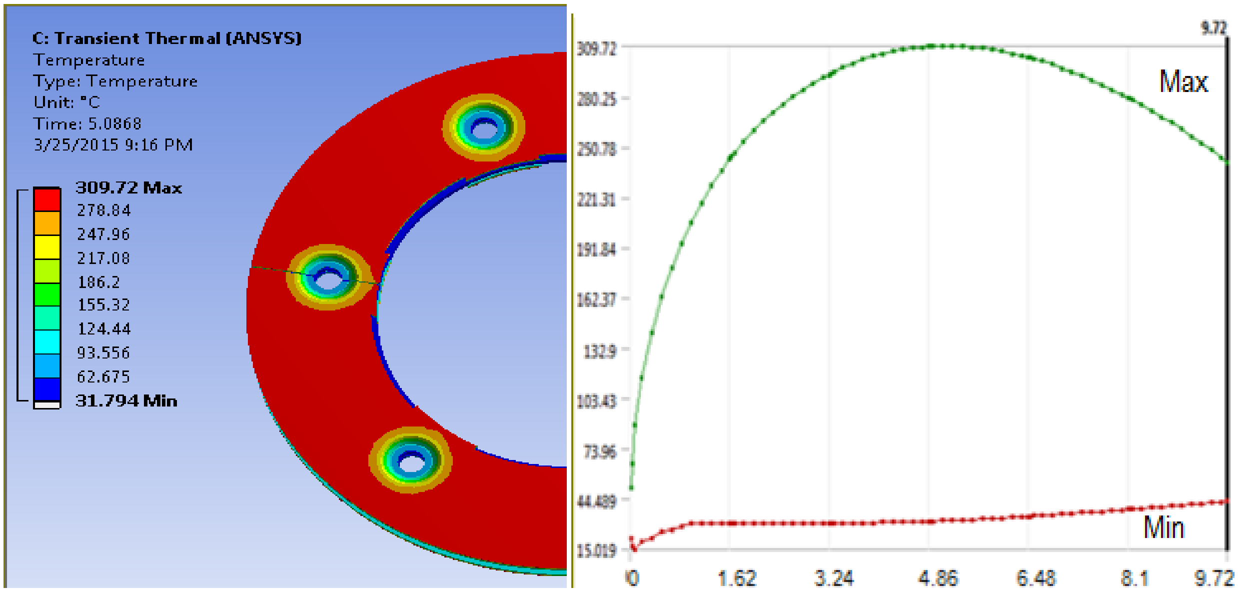



**Fig 4.16 Temperature distribution of ventilated disc vane number 70 at vane angle 20**

Thermal results at the end of the first emergency braking action has reached 30.67$^o$c and302.9$^o$c for 70 vane-numbers and 50 vane-numbers with 5 vane angle disc brake respectively as shown on figures 4.4 and 4.6 which preferable from the other analysis results . According AALRT technical specification 30 second (average station dwelling time of AALRT) and the train starts to move to the next station with two different acceleration values ($a_1$= 1.0m/s$^2$ and $a_2$ = 0.5m/s$^2$) up to the maximum speed. The first acceleration is 1.0m/s$^2$ up to a speed of40km/h (11.11m/s) and then up to the final velocity using 0.5m/s$^2$ acceleration. Distance between consecutive stations, emergency brake-1 to emergency brake-2 ( Urael church) is 0.875 km. The total time for cooling of the disc is the summation of the times stopping at station and the time required to reach to the next station, which is 80 seconds.

The thermal capacity of disc brake is temperature dependent. The heat output of this model is composed of convection. The temperature drop of a brake rotor occurs due to the effect of station stopping time of the passengers train, accelerated and constant traveling time of the train. Therefore for the next braking action the initial temperature of the disc will rise above from its ambient temperature for 70 vane-numbers and 50 vane-numbers with 5 vane angle disc brake will be computed according to Newton's law of cooling is described as:

$$\mathbf{T = T\infty + [(T_1\text{-}T\infty) \times [exp\ \{\text{-}\frac{UA}{mc} \times t\}]]} \quad (4.1)$$

Where: T:-cooled temperature

T1:- maximum temperature

**T**∞:-ambient temperature

A: - disc brake contact area

U: - heat transfer coefficient

**t**: - delay time, accelerated and constant travel time

M: - mass of disc

C: - specific heat capacity of grey cast iron

$$\mathbf{T = 25 + [(302.9\text{-}25\times\ [exp\ \{\text{-}\frac{5.2*0.032}{19.06*0.447} \times 80\}]]}$$

**T=83.35**°C**,** the initial disc temperature for second emergency brake and the results are analysis at vane number 50 , 70 and 50 vane number with vane angle $5^{o}$ are as follow.

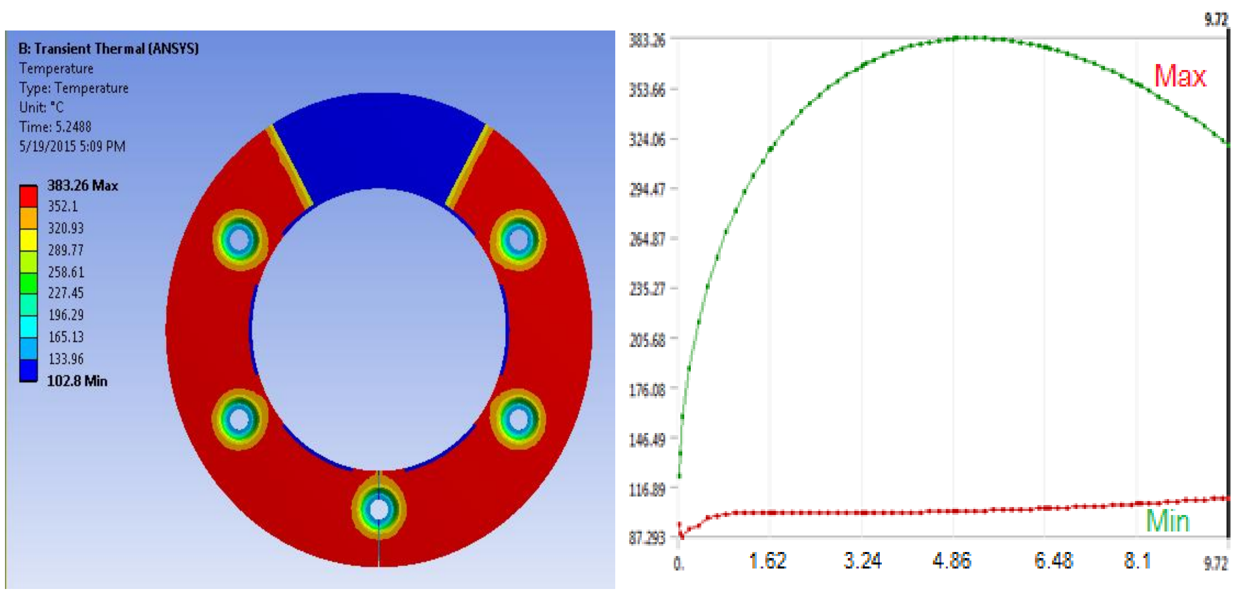



**Fig 4.17Temperature distribution of ventilated disc of vane numbers 50 at emergency brake-2**

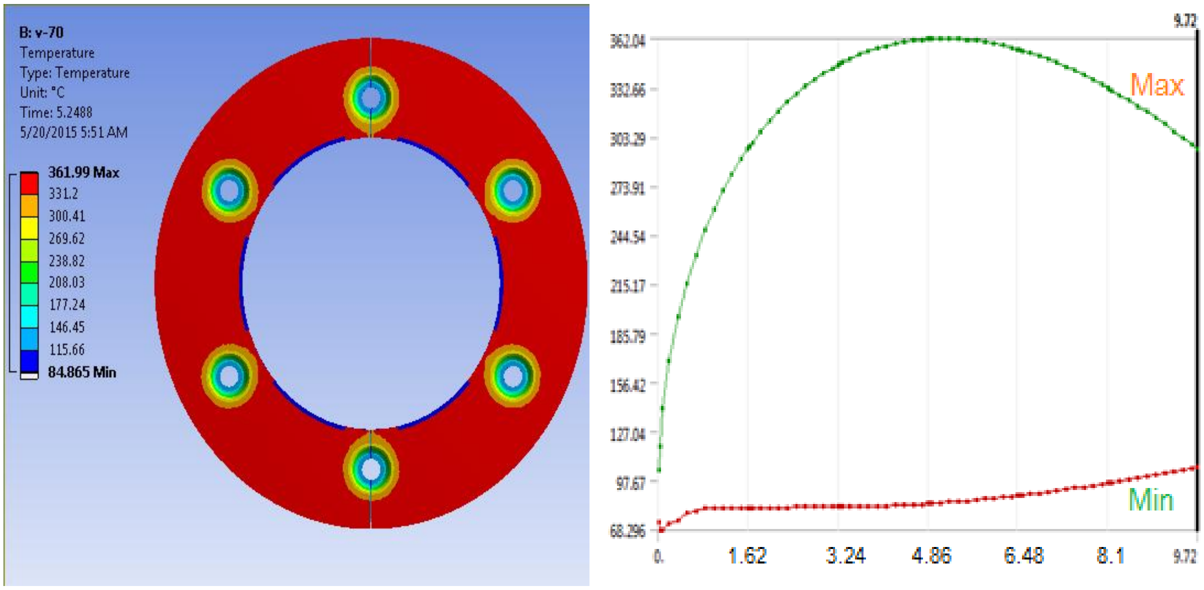



**Fig 4.18Temperature distribution of ventilated disc vane numbers 70 at emergency brake-2**

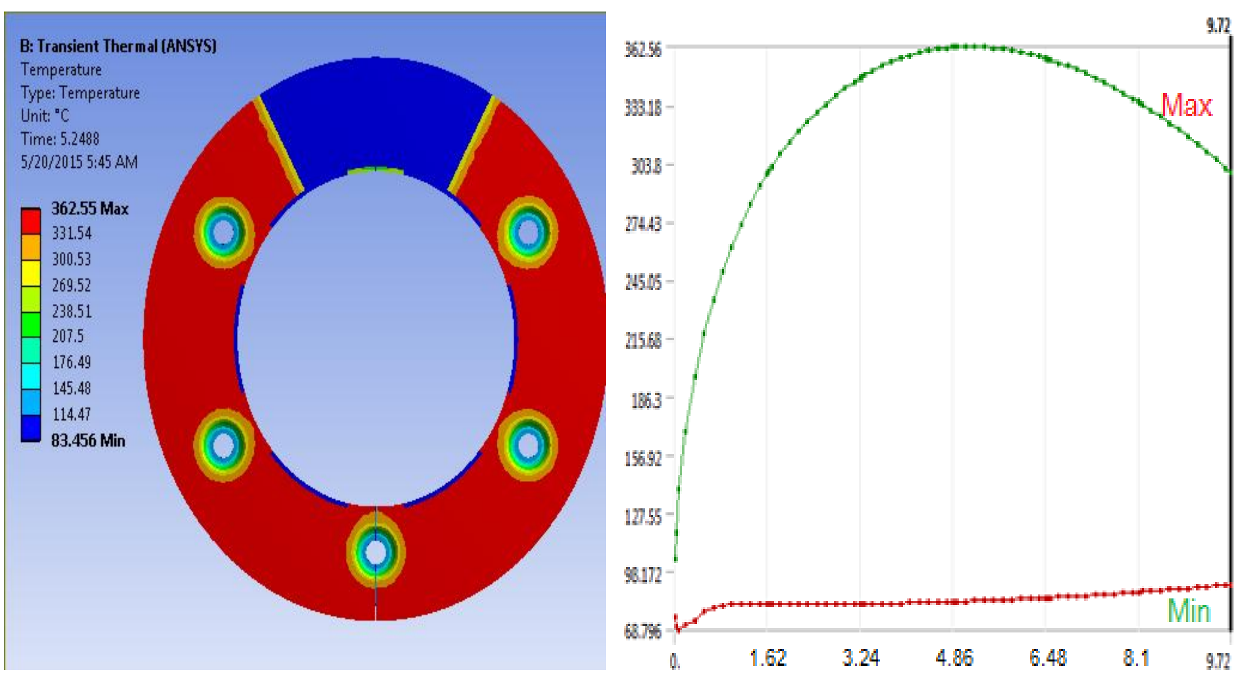



**Fig 4.19Temperature distribution of ventilated disc of vane numbers 50 with vane angle 5$^{o}$ at emergency brake-2**

## 4.4 Analysis of the stress

Thermal stress in the disc appear because during the braking time the temperature rise from the initial material temperature due to load applied to the rotating disc rotor for the purpose stopping the rotation of the disc and then stopping the rail vehicle or to decrease its speed. Stress analysis is to determine the influence of the applied load in the higher thermal load generate during emergency braking time 9.72[s], the comparison stress is gives on von mises at vane number 50, 70 and vane angle 5,50 vane number and at vane number 70 with vane angle 5 and 20 are given as follow :

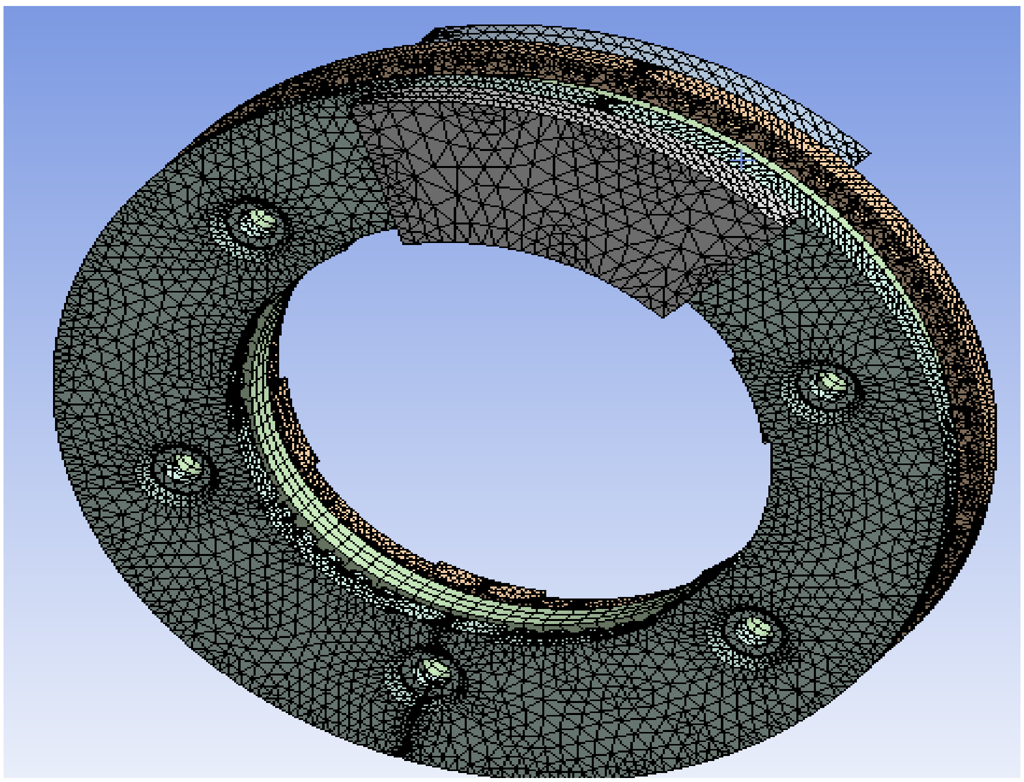


**Fig 4.20 Meshing vented disc of vane number 50 with nodes 112760 and elements 53417**

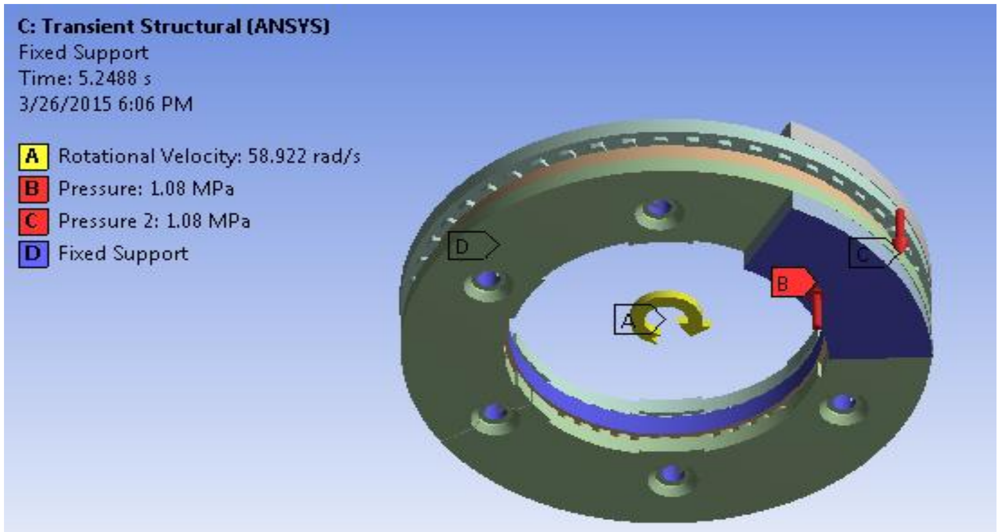



**Fig 4.21 Loading and boundary condition of vented disc at vane number 50**

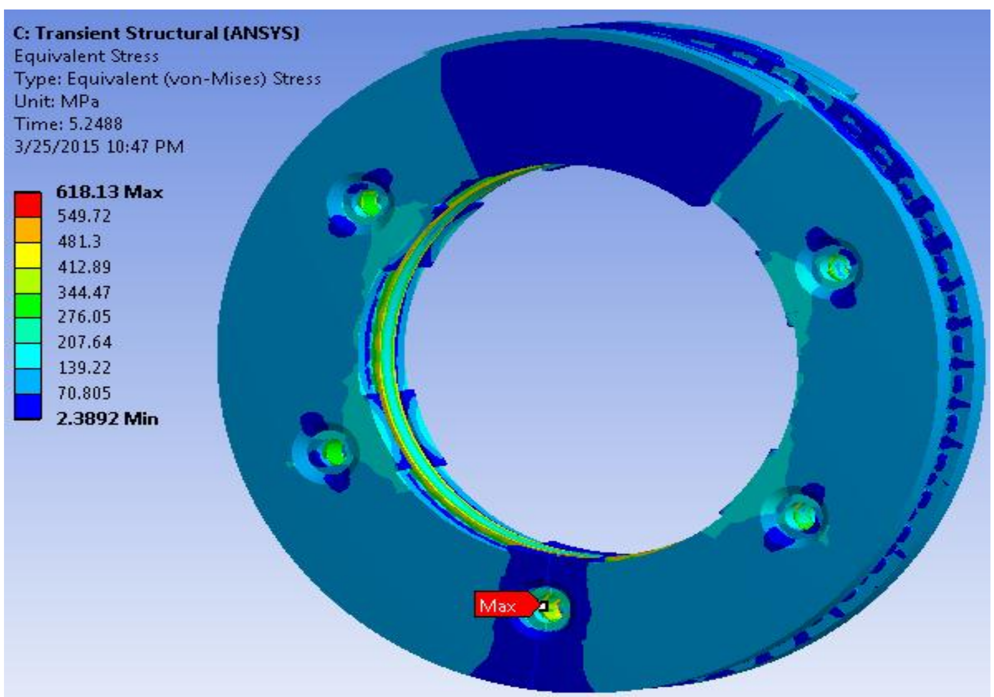



**Fig 4.22 Von mises stress of ventilated disc at vane number 50**

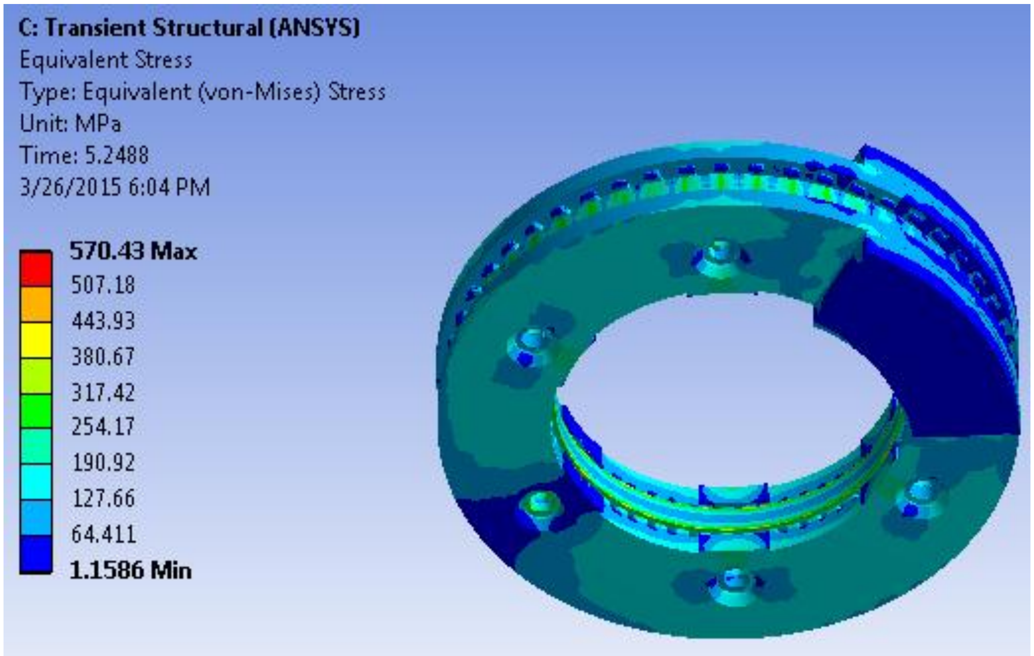



**Fig 4.23 Von mises stress of ventilated disc vane number 50 at vane angle 5**

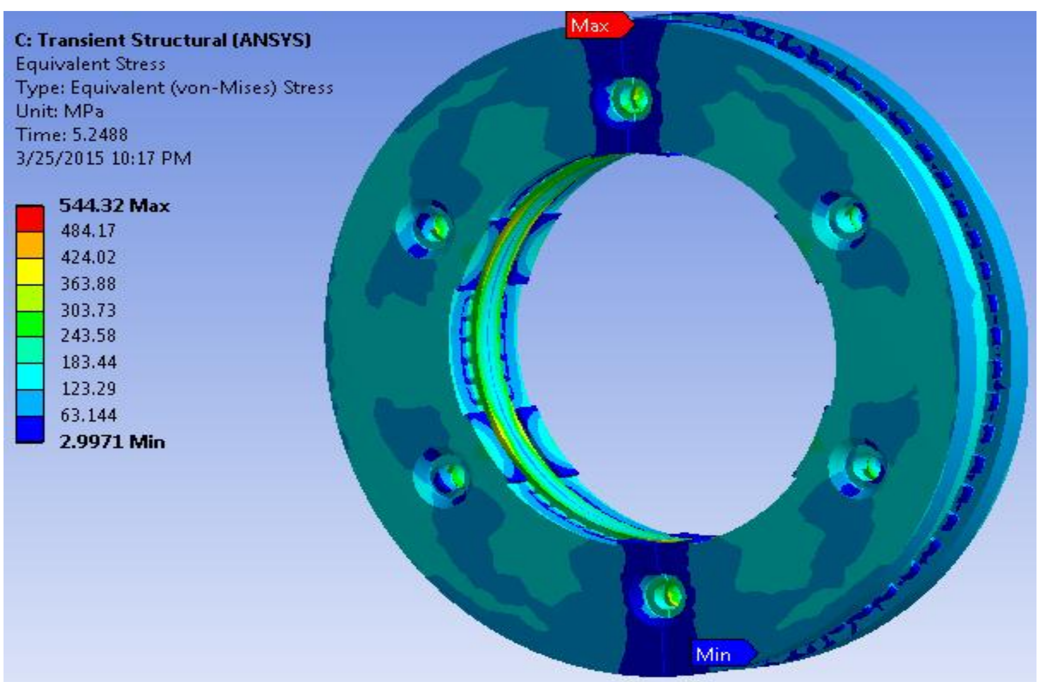



**Fig 4.24 Von mises stress of ventilated disc at vane number 70**

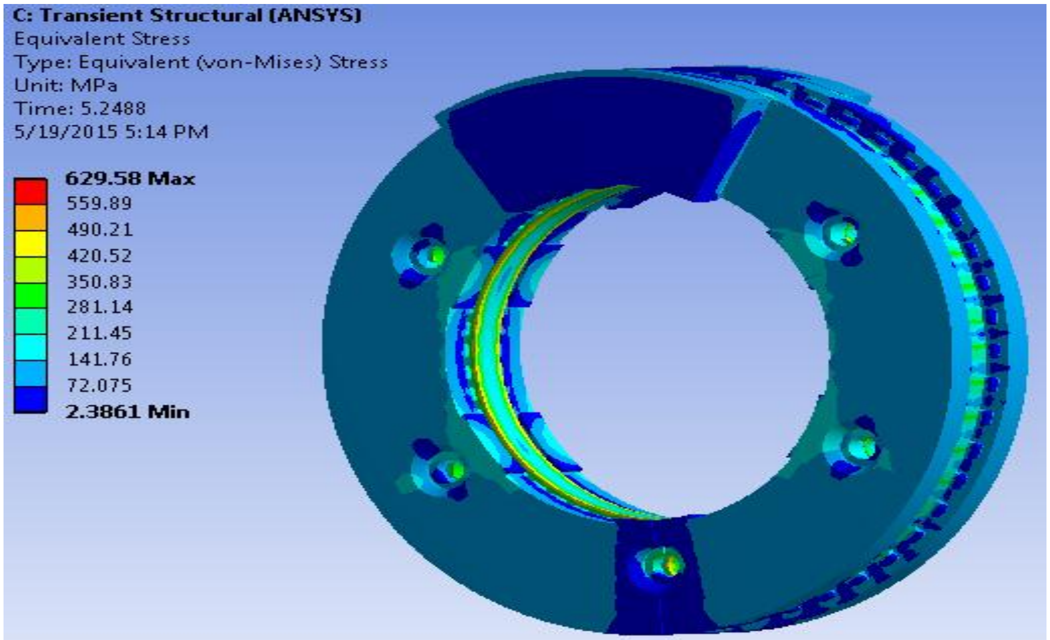



**Fig 4.25Von mises stress of ventilated disc at vane number 50 at emergency brake-2**

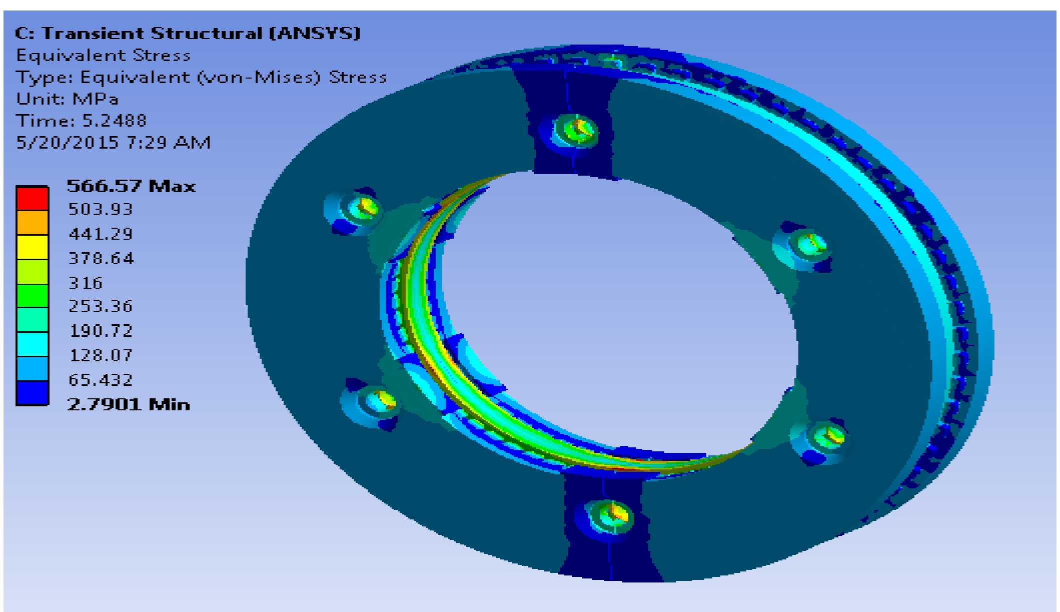



**Fig 4.26 Von mises stress of ventilated disc at vane number 70 at emergency brake-2**

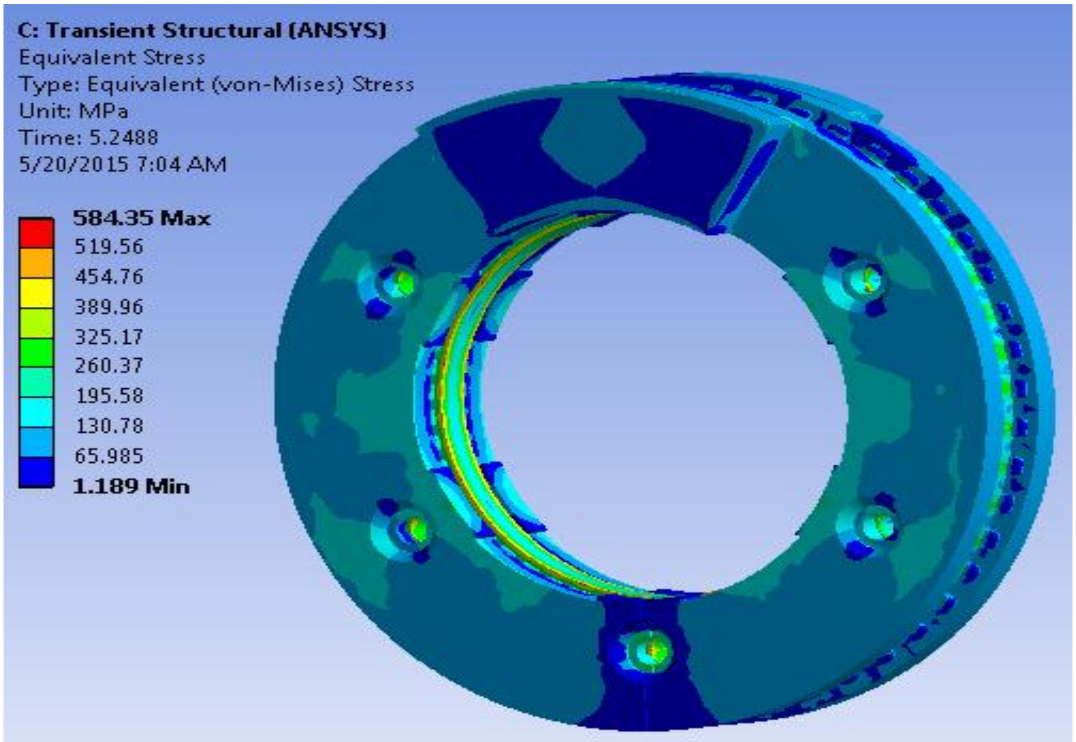



**Fig 4.27 Von mises stress of ventilated disc vane number 50 at vane angle 5$^o$ at emergency brake-2**

## 4.5 Results and discussions

### 4.5.1ANSYS results and discussions based vane numbers

In this study it covers the effect vane numbers without changing the shape and dimension of the current AALRT dimensions like its length 66mm, width 10mm and thickness 5 mm. But only vary the current vane number from 50 to 60, 70, and 80, to observe the effect of vane numbers in emergency braking with  the braking time 9.72[s] , the maximum results are tabulated in table 4.2, at 5.2488 [s]

**Table 4.2 Temperature distribution at vane numbers 50, 60, 70 and 80 and disc weights**

| Vane number | Maximum temperature distribution[°C] | Weight of the disc [Kg] |
|---|---|---|
| 50 | 311.76 | 18.03 |
| 60 | 307.11 | 18.542 |
| 70 | 302.64 | 19.060 |
| 80 | 308.94 | 19.530 |

### 4.5.2 ANSYS results and discussions based vane angles

Currently AALRT uses straight vane type disc brake in addition to electromagnetic type train brake**.** In this study it deals the effect of vane angle changes without changes the dimension of the current AALRT dimensions like length 66mm, width 10mm and its thickness 5 mm. But only vary the current vane angle from 0 to 5, 10 and 20 to observe the effect of the vane angle in emergency braking time of the 9.72[s], braking time with importing wall heat coefficient. The maximum results at time 5.2488[s] for vane number 50 and 60 and for vane number 70 at time 5.0868 [s] are tabulated as follows.

**Table 4.3 Temperature distribution at vane angles 5 and 20 at vane number 50, 60, 70 and 80 and disc weights**

| angles | Vane angle at vane number 50 | | Vane angle at vane number 60 | | Vane angle at vane number 70 | |
|---|---|---|---|---|---|---|
| | Temperature Maximum[°C] | Weight [Kg] | Temperature Maximum[°C] | Weight [Kg] | Temperature Maximum[°C] | Weight [Kg] |
| 5 | 302.9 | 18.04 | 310.44 | 18.554 | 309.86 | 19.070 |
| 20 | 302.24 | 18.134 | 310.19 | 18.668 | 309.72 | 19.198 |

### 4.5.3 Results and discussions based on vane curve radius

This study try to indicate the effect of the shape of the curve vane at vane radius 60, 80,100 and 120 is simulated at 9.72 [s],emergency braking time; its main purpose is to decrease temperature that generates during emergency braking at the high gradient of AALRT truck at the maximum operation speed 70 km/h. The result or the distribution of the temperature during the emergency braking is tabulated at time 5.2488[s].

**Table 4.4 Temperature distributions at vane curve radius 60, 80, 100 and 120; disc weights**

| Vane curve radius | Maximum-temperature distribution[°C] | Weight of the disc [Kg] |
|---|---|---|
| 60 | 309.83 | 18.204 |
| 80 | 311.55 | 18.114 |
| 100 | 312.02 | 18.082 |
| 120 | 312.03 | 18.066 |

From above results and discussions based on vane numbers, vane angles and vane curve radius of the ventilated disc, their better results found on vane number 70 and vane number 50 with vane angle $5^0$, when comparing with vane number 50 currently used AALRT of ventilated type disc brake, for further observed the effect of tentative emergency brake at second station with applying initial temperature investigated at the first emergency brake and the results are simulated at 9.72 [s] emergency braking time but the maximum value achieved at 5.2488[s] is 382.26, 361.99 and 362.55[°C] for vane number 50, 70 and vane number 50 with vane angle $5^0$ disc brake respectively.

# CHAPTER FIVE

# CONCLUSIONS, RECOMMENDATION AND FUTURE WORK

## 5.1 Conclusion

In this study the analysis of temperature generating on the ventilated disc rotor during the emergency braking was done. The analysis was simulated with CFD-cfx package software and the modeling is based on ANSYS-12.From CFD-CFX package software simulation more accurate solution to ventilated disc rotors was achieved. From the analysis various values of temperature distribution on the disc with vane number of 50, 60, 70, and 80; also with of vane angle 5, 10 and 20 disc with vane numbers 50, with vane curve radius 120, 100, 80 and 60 mm was also investigated.

The results showed that at different vane numbers, vane angles and vane radius are different. At vane number 70 with straight vane the maximum temperature which is 302.67 [°C] achieved and at vane number 50 with angle 5 vanes the temperature which is 302.9 [°C] achieved. These achieved values were lowest to the other maximum values.

Finally the conclusion of this study based on the simulating with the braking time 9.72[s] as the vane number increase from 50 to 70; their generating temperature is decrease from 311.76 to 302.67 at time 5.2488 [s]. But at vane number 80 its value increase to 308.94 °C due to number vanes increases more than beyond the gab among vanes became narrow nearly solid. Similarly vane angle at the vane numbers of 50, 60 and 70; the better one is vane number 50 at vane angle 5 and vane number 70. As considering weight of the disc among vane number, vane angle and vane curve radius their variation is small. So that in this thesis the disc with vane number 50 at vane angle 5 achieved temperature is 302.9 [°C], at time 5.2488[s] is best one. Since the mass of the disc vane number 70 is miner as comparing with disc with vane number 50 with vane angle 5, the disc with vane number 70 is achieved 302.67 [°C] at highest gradient of AALRT rout as being emergency brake-1 and tentatively take emergency brake -2, the stimulated temperature on the vane number 70 and vane number 50 with vane angle $5^0$ are increased to 362.55 and 361.99 [°C] respectively are the best from the others including the current one which is uses AALRT vane number 50 disc brake, its simulated temperature at emergency brake-2 is 383.26 [°C].

## 5.2 Recommendation

In this paper made an effort to analyze ventilated disc brake of railway locomotive using analytical and numerical modeling of thermal effects during emergency braking on a down-grade railroad for stopping purpose.

The intention was to make the contribution in analyzing stresses in the vented disc brakes caused by the thermal and mechanical loading.

The presented approach is the basis for the following 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 the currently used AALRT ventilated disk brake. From the result of this study we can recommended for Ethiopian rail way corporation and other automotive sectors and manufacturers instead straight vane type it is better inclined in vane angles range from 5 to 10 with given dimension outer radius 185mm, inner radius 110 mm and disc thickness 60 mm and vane number 50, disc material grey cast iron. It also selected with curve vane than straight with curve radius 60m with change the other dimensions and with 70 vane number instead 50 without change the other parameters that currently AALRT used, finally vane number 50 with vane angle $5^o$ and vane number 70 are preferable including their weights.

## 5.3 Future work

Regarding the outlook, there are recommendations for the expansion of future work related to disc brake that can be done to further understand the effects of thermal analysis of the vented and optimization the shape of the vented curves, number of vanes and van angles of the disc rotor, as recommendations are as follows:

1. Experimental study to verify the accuracy of the numerical model developed.
2. Study with considering tribological and vibratory study of the contact disc – pads;
3. Study using different materials to compare the effects of vane number, shape and vane angle

# REFERENCES

[1] William L.Garrison. Historical transportation development, transportation engineering and planning-volume-I

[2] Fling, Harry M. Railroads of the United States, Their History and Statistics. Philadelphia: John. E. Potter and Co. (1868).pp.12- 13.

[3]Georgius Agricola (Trans Hoover), De remetalica (1913), p. 156

[4] The miners called the wagons Hund -"dog" - from the noise they made on the tracks. Lee,Charles E (1943).The Evolution of Railways(2Ed.). London: Railway Gazette. p.16. OCLC 1591369

[5] Warren Allison, Samuel Murphy, and Richard Smith, 'An Early Railway in the German Mines of Caldbeck' in G. Boyes (ed.), Early Railways 4: Papers from the 4th International Early Railways Conference 2008 (Six Martlets, Sudbury, 2010), pp.52–69.

[6] Peter King, ‘The First Shropshire Railways’ in G. Boyes (ed.), Early Railways 4: Papers from the 4th International Early Railways Conference 2008 (Six Martlets, Sudbury, 2010), P 70–84.

[7]Wells,David A.Recent Economic Changes and Their Effect on Production and Distribution of Wealth and Well-Being of Society. New York: D. Appleton and Co. ISBN 0-543-72474-3. (1891).

[8] Grübler, Arnulf (1990). The Rise and fall of Infrastrucres: Dynamics of Evolution and Technological in Transport. Heidelberg and New York: Physica-Verlag.

[9] Fogel, Robert W. (1964). Railroads and American Economic Growth: Essays in Econometric History. Baltimore and London: The Johns Hopkins Press.ISBN 0-8018-1148-1.

[10]Rosenberg, Nathan (1982). Inside the Black Box: Technology and Economics. Cambridge, New York: Cambridge University Press. p.60. ISBN 0-521-27367-6.

[11] "Ilg, Alfred" in the *Encyclopedia Ethiopia*, ff. Accessed 17 Mar 2014. Vol. 3, pp. 120

[12] "Abyssinia" in the Encyclopedia *Britannica* 11th Ed.

[13]"Zaila" in the *Encyclopedia Britannica* 11th ed. 1911.

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

[15] C. Owen, *Automotive Brake Systems: Classroom Manual*, 3rd Ed. Clifton Park, New Jersey: Thomson Delmar Learning, 2004, pp. 164-175.

[16] H. Jacobsson. "Aspects of Disc Brake Judder", *Proc. Instn Mech. Engrs, Part D: J. Automobile Engineering*, Vol. 217, 2003, pp. 419-430.

[17]R. Limpert, *Brake Design and Safety*. 2nd Edition, Warrendale, Pennsylvania: Society of Automotive Engineering Inc., 1999, pp. 137-144.

[18] Sung-Soo Kang, and Seong-KeunCho,"Thermal deformation and stress analysis of disk brakes by finite element method",Journal of Mechanical Sciences and Technology, (2012) Vol.26 (7), pp.2133-2137

[19]MesutDuzgun," Investigation of thermo-structural behaviors of different ventilation applications on brake discs", Journal of Mechanical Science and Technology, Vol.26 (1) (2012) pp.235-240

[20] Ali Belhocine, MostefaBouchetara," Thermal analysis of a solid brake disc", Journal of Applied Thermal Engineering, (2012),Vol.32 pp.59-67

[21] *Belhocine et al., Tribology in Industry Vol. 36, No. 1 (2014) 49 ‐ 66*

[22] Marko Reibenschuh, Grega Oder, FranciČuš, IztokPotrč: Modeling and Analysis of Thermal and Stress Loads in Train Disc Brakes – Braking from 250 km/h to Standstill, University of Maribor, Faculty of Mechanical Engineering, Slovenia

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

[24]KhongKengLeng, AM34: Simulation of Temperature Distribution in Brake Discs, National University of Singapore 2007/2008.

[25] Mackin,TJ: Thermal cracking in disc brakes ,Engineering Failure Analysis (2002), no 9,str 63-76.

[26]Fiche U.I.C. 541-3: FREIN –Frein à disquesetgarnitures de frein à disques, 4e édition, 01.07.1993.

[27]Saumweber,E.,Temperaturberechnung in Bremsscheiben füreinbeliebiges Fahrprogramm, Leichtbau der Verkehrsfahrzeuge, Heft 3 / 1969, Augsburg.

[28] Cruceanu, C., Frannepentru vehiculeferovaire(brakes for railway vehicles). Ed MATRIXROM, Bucuresti, 2007

[29]M.Noubyand K. Srinivasan.Parametric Studies ofDisc Brake Squeal using.Finite. Element-Approach"AnnaUniversity Chennai-600025 *India*.*Jurnal Mekanikal*. December 2009, No.29, 52-66.

[30] Mackin, T.J.: *Thermal cracking in disc brakes*. *Engineering Failure Analysis* (2002), no. 9, str.63-76

[31].H.K.Versteeg and W.Malalasekera–An Introduction to Computational Fluid Mechanics, The Finite Volume Method.

[32]Ansys v.11 user' Manual guide, ANSYS, Inc., Houston, USA (1996).

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

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

[35] Cruceanu, C, *Brakes for Railway Vehicles* (in Romanian), 3rd edition, Ed. MATRIXROM, Bucharest, (2009)-ISBN 978-973-755-200-6