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ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

GRADUATE PROGRAM IN RAILWAY ENGINEERING

THERMAL AND MECHANICAL ANALYSIS OF A.A. LRT BRAKE DISC FROM COMPOSITE MATERIAL BY FEM

A thesis submitted to the

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By

Fitsum Getaneh

Adviser: Dr. Ing. Demis Alemu

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Approval sheet

Approved by:

1. Dr.Ing Demiss Aleme	____________________	____________________
Advisor	**Signature**	**Date**
2. Ato Tsegaye Feleke (M.Sc.)	____________________	____________________
Internal examiner	**Signature**	**Date**
3. Ato Habtamu (M.Sc)	____________________	____________________
External examiner	**Signature**	**Date**
4. Dr. Birhanu Beshah	____________________	____________________
Railway center head	**Signature**	**Date**

Dedication

To my beloved Father for his immeasurable believes encouragement and patience

Acknowledgement

The pursuit of a MSc. dissertation is often thought of as an individual project. In retrospective, it is clear to me that this is true only to a certain extent. The process and the eventual outcome depend largely on the surrounding environment and in this respect I have been very fortunate. Over the past year I have enjoyed a daily work that has been intellectually challenging and rewarding. This would not have been the case without the contributions by people in my vicinity next to my **Almighty God.**

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Abstract

The modeling of thermal and mechanical effects during braking has become increasingly important in product design in different transport means, road vehicles, airplanes, railway vehicles, and so forth. The thermal and mechanical analysis is a very important stage in the study of braking systems, especially of railway vehicles where it is necessary to brake huge masses because the thermal load of a braked railway disc prevails compared to other types of loads. In the braking phase, kinetic energy transforms into thermal energy resulting in intense heating and high temperature states of railway discs. Thus induced thermal loads determine thermo-mechanical behavior of the structure of railway discs. In cases of thermal overloads, which mainly occur as a result of long term braking on down grade railroads and braking to standstill on a flat railroad, the generation of stresses and deformations occurs whose consequences are the appearance of cracks on the discs, brake judder, thermo-elastic instability and finally brake disc failure. The importance to precisely determine the temperature distribution caused by the transfer process of the heat generated during braking due to the friction on contact surfaces of the braking system makes it a challenging research task.

Therefore, analytical and numerical modeling of thermal and mechanical analysis during long term braking for maintaining a constant speed and braking to standstill afterwards on a down grade railroad and braking to a standstill on a flat track is processed in detail in this research. Even though brake disc is commonly made from cast iron, an attempt has been made to investigate the suitable composite material which is lighter than cast iron and has good young's modulus, yield strength and density properties. Aluminum base metal matrix composite has a promising friction and wear behavior when used as a disc brake. Brake disc made from SiC/6061Al alloy composite material is used and an investigation is made on thermal and mechanical loads analysis for Addis Ababa LRT EW route brake applications. Consequently the result revealed that SiC/6061Al alloy composite material can effectively enhance the thermal and mechanical performance of the brake disc.

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Notations

M	Mass
a	Acceleration or deceleration
v	Final velocity
v_o	Initial velocity
S	Distance
t	Time
l_b	Braking distance
g	Acceleration due to gravity
μ	Coefficient of friction
α	Gradient
q_x	Conduction heat fluxes in x direction
q_y	Conduction heat fluxes in y direction
q_z	Conduction heat fluxes in z direction
c_p	Specific heat
ρ	Specific mass
Q	Internal heat generation
T	Temperature
Q_s	Surface heat flux
h	Convective heat transfer coefficient
T	Unknown surface temperature
T∞	Convective exchange temperature
ED	Evation difference
HD	Horizontal difference
ω	Angular velocity
FT	Tractive force
FR	Rolling resistance
FD	Drag force
FG	Normal force component in the direction of the inclined plane
N	Normal force

Tm	Motor torque
T wheel	Torque on the wheel
P	Power of the motor
CD	Drag coefficient
ρ	Density of fluid
V	Velocity of passenger train
D_{oD}	Disc outer diameter
η_o	Efficiency of brake transmission
z	Rotational coefficient
i	Brake lever ratio
D_{oW}	Wheel diameter
EW	East west route
NS	North south route

CHAPTER ONE

INTRODUCTION

1.1Background

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

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

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.

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

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

At first, the rather low locomotives power and traction force allowed braking using quite simple handbrakes that equipped locomotives and eventually other vehicles of the train.

As the development of rail transport and according to increasing traffic speeds, tonnages and length of trains, it was found that braking has to be centralized 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 that were developed, following the studies and especially the results of numerous tests, the indirect compressed air brake system proved to have the most important advantages. Therefore, it was generalized and remains even nowadays the basic and compulsory system for rail vehicles.

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

Brakes have been retuned and improved ever since their invention. The increases in travelling speeds as well as the growing weights of cars have made these improvements essential. The faster a rail car goes and the heavier it is and the harder it is to stop. An effective braking system is needed to accomplish this task with challenging term where material need to be lighter than before and performance of the brakes must be improved [5].

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

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

The railway vehicles are the most popular and widely available way of transportation. Therefore the train must be suitable, reliable and safe. The most important part of the vehicle is the braking system. Thermal and mechanical analysis is involved in almost every kind of physical processes and it can be the limiting factor for many processes. Therefore, its study is of vital importance and the need for powerful thermal and mechanical analysis tools is virtually universal. Furthermore, thermal and mechanical effects often appear together with or as a result of other physical phenomena [3].

The modeling of thermal effects has become increasingly important in product design including areas such as electronics, automotive, aerospace, railway (e.g. wheel and rail rolling contact, braking systems and so on) and medical industries etc. Computer simulation has allowed engineers and researchers to optimize process efficiency and explore new designs while at the same time reduce costly experimental trials [6].

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

Precise prediction of the maximum temperature is needed for the design of many systems as well as braking systems especially for both discs and linings. How to handle the high spinning speed of disc is the point of the thermal/mechanical coupled analyses.

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

In such cases, the temperatures from a transient thermal analysis are used as inputs to a mechanical analysis for thermal stress evaluations. Thermal analysis is the primary stage in the study of braking systems because the temperature determines the thermo-mechanical behavior of the structure. In the braking phase, kinetic energy transforms into thermal energy resulting in intense heating of the railway brake discs. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks on brake discs and eventually fractures of the whole brake disc [1].

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 down grade railroads on which trains run. Having concluded their researches, the European Rail Research Institute (ERRI) proposed a valuation procedure with the ultimate goal of making a decision on taking special measures to prevent the occurrence of thermal overload. This implies the application of the regulation and alternating brake or other measures related to regime change and drive train formation. However, the proposed procedure which involves calculating the coefficient for the particular estimation does not provide an efficient and reliable use in all potential cases particularly those relating to the long term braking of the train at a very long fall with the changing slope. Therefore, the research for the improvement of the thermal overload assessment process continues [4]. Many research results have confirmed the dominant influence is thermal loads in regard to mechanical loads and residual stresses induced by high thermal loads in brake discs. Therefore, it is important to determine with high precision the temperature field of the braking system as well as to emphasize that high thermal loads, in other words, overloads of disc very often occur as a result of long term braking on down grade railroads or unwanted locking of wheels and braking to standstill.

Those are the main goals of this research which presents the results of analysis of a braking system of railway vehicles using analytical and numerical modeling of thermal and mechanical analysis during long term braking for maintaining a constant speed on a down grade railroad and braking to standstill in order to analyze the performance of brake discs on Addis Ababa LRT train. Even though brake disc is made from mainly grey cast iron in this research another alternative is introduced (composite material) and its thermal and mechanical analysis is done for comparison. Since many composites also exhibit great resistance to high temperature, corrosion, oxidation and wear.

These unique characteristics provide the mechanical engineer with design opportunities not possible with conventional monolithic (unreinforced) materials. Composites are important materials that are now used widely not only in the aerospace industry but also in a large and increasing number of commercial mechanical engineering applications such as internal combustion engines, machine components, thermal control and electronic packaging, automobile, trains, aircraft structures and mechanical components such as brakes, drive shafts, flywheels, tanks and pressure vessels, dimensionally stable components and process industries equipment requiring resistance to high temperature corrosion, oxidation and wear, offshore and onshore oil exploration and production, marine structures, sports and leisure equipment and biomedical devices [3,6]. Therefore brake disc prepared with SiC network ceramic frame reinforced 6061 aluminum alloy composite (SiC/Al) composite is investigated in detail.

1.2 Statement of the Problem

During long term braking for maintaining a constant speed on a down grade railroad and braking to standstill a heavy mass vehicle like trains, a great deal of energy is required to stop or decelerate the spinning discs artificially. This process gave birth to the following basic braking phenomena:-

- Stresses and deformations of brake disc material resulted from high temperature.
- Stresses and deformations of brake disc 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 due to 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 braking a heavy railway vehicle is the great input of heat flux into the disc in a very short time. Because of the high heat flux (temperature difference) the brake disc material is exposed to high stress and deformation [3].

Thermal analysis is the primary stage in the study of braking systems because the temperature determines the thermo-mechanical behavior of the structures of the contacting materials. In the braking phase, kinetic energy transforms into thermal energy resulting in intense heating of the railway brake discs. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks on brake discs and eventually fractures of the whole brake discs [6]. This research is designed to come up with solution by investigating the thermal and mechanical analysis of continuous SiC/6061 Al alloy composite material made brake disc for Addis Ababa LRT under centrifugal and thermal load during long term braking for maintaining a constant speed on a down grade railroad and braking to standstill considering airflow cooling.

1.3 Objective:

1.3.1 General Objective

The main objective of the present research is to conduct thermal and mechanical analysis of continuous SiC/6061 Al alloy composite material made brake disc for Addis Ababa LRT and investigation of its braking performance during long term braking for maintaining a constant speed and braking to standstill afterwards on a downgrade railroad and braking to standstill on a straight railroad.

1.3.2 Specific Objective

The specific objectives are:

- To develop brake disc model and simulation using ANSYS software package.
- To perform the brake disc fluid flow analysis.
- To analyze the thermal and mechanical load of a brake disc for railway vehicles using the finite element method (FEM).
- To diminish the weight of unsuspended mass of the vehicle, brake disc made from continuous SiC/6061 Al alloy composite is introduced and analyzed.
- To consider Addis Ababa LRT track for locating the potential areas of brake application to maintain constant speed and braking to standstill.
- To conduct transient thermal to investigate the temperature variation across the disc using axis-symmetric elements.
- To investigate the effect of air ventilation on vented brake disc.

1.4 Significance of the Study

Many research results have confirmed that the dominant influence is the thermal loads in regard to mechanical loads and residual stresses induced by high thermal loads in brake discs. Therefore, it is important to determine with high precision the temperature field of the braking system as well as to emphasize that high thermal loads, in other words, overloads of disc very often occur as a result of long term braking on down grade railroads or unwanted locking of wheels and braking to standstill.

The Ethiopian railway corporation significantly will be benefited from this work for the reason that knowing the temperature distribution of the brake disc and the friction pad at different potential location of the changing slope of the Addis Ababa LRT track is of great importance. Therefore this research is established to address and help designers understand the condition that the Addis Ababa LRT brake discs could encounter during braking operation at potential areas of sloppy LRT track. In addition to this it could be used as a design tool and a starting point for improvements.

1.5 Scope

In the present work of the research, the continuous SiC/6061 Al alloy composite material which is lighter than grey cast iron and has good young’s modulus, yield strength and density will be investigated. Thermal load is analyzed. A transient thermal analysis had also been carried out to investigate the temperature variation across the disc surface using axis-symmetric elements. Further mechanical analysis had been carried out by coupling thermal analysis. The effect of ventilation on the brake disc is also investigated.

1.6 Limitation

In this research, the mathematical model which represents the physical phenomenon of brake disc has been built considering limited boundary conditions like the convection heat transfer, heat flux, clamping force and the centrifugal load. Since they are the dominant and extremely influential. However, conduction and radiation heat transfer, humidity, shearing force, residual stress, cyclic loads during brake disc life time and more complicated brake disc geometry together with its component and brake disc interfaces to fully represent the real physical phenomenon is not considered. And it is left for future work.

1.7 Methodology

Thermal and mechanical analysis of disc brakes under specific loads (moving downhill and braking to standstill) will be calculated. The FEM (finite element method) together with ANSYS software packages are used to carry out the analysis. The analysis deals with centrifugal load for two cases of braking is investigated, braking to a standstill on a flat surface and braking on downhill track for maintaining constant speed and braking to standstill afterwards. The main boundary condition in both cases was the entered heat flux on the braking surface of the disc and the force of the brake clamps.

Evaluating the resources that are available for the desired study is the first thing to do. Going through different literatures, researches, books and any other resources related to the topic. After knowing what to do, how to do is the next step that follows. The procedure implemented here shows the methodology this research would accomplish it step by step throughout work.

1.7.1Literature Review and Data Collection

Under this category all the required information and data would be gathered. In such a case, here is basic process under taken under data collection category:

- Evaluate the resources that are available
- Identifying the functional requirement
- Evaluating mechanisms and structures to perform the analysis
- Material with proper thermal and mechanical property selection
- Analytical calculation of brake disc

1.7.2 Data Analysis

All the data gathered in the previous step will be analyzed to represent the real physical phenomena. Under such category, the following processes are performed:

- Building the 3D model using ANSYS CFX for flow analysis
- Building the 3D model using ANSYS workbench for thermal and mechanical analysis
- Determination of the physical model
- Load determination
- Model simulation
- Transient thermal analysis
- Transient structural analysis

1.7.3Result Presentation

Result presentation focuses on the solution part of the research. It can be explained under the following topics:

a. Result and Discussion

- Temperature distribution of disc brake
- Variation of temperature of brake disc wrt time
- Deformation of disc brake
- Von Misses stress of disc brake
- Effect of centrifugal load during braking
- Effect of air cooling system of the disc brake

b. Conclusion

- Explanation of the result obtained
- Conclusion and recommendation

1.8 Organization of the Research

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

Chapter two discusses Literature review of the disc brake system of rail vehicle. Similar past studies had been reviewed under this chapter. Also Definition, key points, models and analytical approaches will be 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.

The physical model of passenger train disc brake, load determination, thermal analysis and mechanical analysis as well as the material and boundary conditions are discussed in detail under chapter four.

Chapter five deal with ANSYS simulation, model preparation, characterizing the physical model and loading of the model.

Chapter six verify the result of the simulation and discussion of the result obtained from simulation of the model.

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

CHAPTER TWO

LITERATURE REVIEW

The production of safe vehicle is one of the major problem facing automobile and locomotive manufacturing industries today. Since effective braking system is one of the determining factor in vehicle safety. Enhancement of the brake system's efficiency is a crucial scientific problem facing manufacturers. The most important units in the vehicles braking system are the brake discs because these components transform the control force from a brake calipers into a brake torque which is applied to a wheel. Disc brake is one of the most complicated vehicle units with respect to a mathematical description of the processes that take place in them during braking [16].

The braking performance of the train is one of the most important factors that affect the traffic safety and the brake is the key components related to the running safety of the railway vehicle. It was found that the 50% of the observed accident is caused by braking performance of the vehicle [5]. It is mostly due to the thermal and mechanical failure of the brake equipment. The numerical simulation of the temperature field is the main parameter that causes thermal recession and thermal fatigue. The biggest difficulty lies on the analysis of the transient temperature and stress field during the braking process and the establishment of a simplified model of the thermal /mechanical coupling together with better thermal and mechanical property of the material used.

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

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 down grade railroads on which trains run. Having concluded their researches, the European Rail Research Institute ERRI proposed a valuation procedure, with the ultimate goal of making a decision on taking special measures to prevent the occurrence of thermal overload. This implies the application of the regulation and alternating brake or other measures related to regime change and drive train formation.

However, the proposed procedure which involves calculating the coefficient for the particular estimation does not provide an efficient and reliable use in all potential cases particularly those relating to the long term braking of the train at a very long fall with the changing slope. Therefore, the research for the improvement of the thermal overload assessment process continues [4].

Recent theoretical and experimental works [39, 57] have been developed to characterize the thermal parameters which govern the heat transfer at the vicinity of a sliding interface. In certain industrial applications, the solids are provided with a surface coating. A recent study has been carried out to analyze the effect of surface coating on the thermal behavior of a solid subjected to the friction process [12]. Increased thermal efficiency and the integrity of materials in high temperature environments is an essential requirement in modern engineering structures in automotive, railway, aerospace, nuclear, offshore, environmental and other industries. Nowadays, the finite element method is used regularly to obtain numerical solutions for heat transfer problems. The most common choice when using finite elements is a standard Galerkin formulation [13].

Here are some preliminary studies previously undertaken regarding disc brake analysis and disc brake performance.

Daniel Das.A: Thermal and stress analysis of disc brake which is made from grey cast iron under specific loads (driving downhill and braking to standstill) was calculated. The FEM (Finite Element Method) was used to carry out the analysis. The analysis dealt with centrifugal load for two cases of braking, braking to a standstill on a flat surface and braking on a downhill track for maintaining constant speed and afterwards braking to a standstill. The main boundary condition in both cases was the entered heat flux on the braking surface of the disc and the force of the brake clamps. Two different discs were used, one brand new (unused) and one with permitted wearing.

The purpose of the model and analysis was to determine the effect of thermal loads on the temperature field and the effect of centrifugal load in a specific brake disc. To determine the boundary conditions parts of car brake disc calculation were used. This assumption can be applied because the working physical principal is the same.

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

They observed and conclude from their study that, due to the brake heating up during braking it is obvious that the effect of the heat flux is outstanding. Since the highest values of temperature fields occurred in the worn disc during braking downhill with later stopping on flat surface. They reached 598 °C within the time of simulated braking 136 s. The highest allowable temperatures in the brake pad and consequently, in the disc were 350 °C (long standing). The results of stress analyses show that by considering the centrifugal loads the stress increased by 10 to 20 MPa. The highest stress value was 201 MPa and was also smaller than the permissible stress for the material of the disc. In all cases the effect of centrifugal loads on the disc was small in comparison to thermal loads. The highest values of stresses of concerned stress fields occurred during braking downhill with stopping on a flat surface. This is due to the fact that in this case the temperature difference between the warmest and coldest part is much higher than in case of braking on a flat surface only.

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

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

Whoever orders the research will compare the results of numerical analyses also with experimental results, which is at the same time a recommendation for their further work [12].

Thundil karuppa raj.R, Ramsai.R and Mathew.J,Soniya.G: Modeling of the thermo-mechanical behavior of the dry contact between the discs and brake pads at the time of braking phase was conducted based on the software ANSYS 11. This last is elaborate mainly for the resolution of the complex physical problems. The numerical simulation of the coupled transient thermal field and stress field is carried out by sequentially thermal-mechanical coupled method based on ANSYS. The goal was to measure current vane air flow and to improve this vane flow to increase brake disc cooling.

Raised temperature can strongly influence the properties of the surface of materials in slip, support physic-chemical and micro-structure transformations and modify the tribology of the interface elements present in the contact. Recent numerical models presented to deal with rolling processes have shown that the thermal gradients can attain important levels which depend on the heat dissipated by friction, the rolling speed and the heat convection coefficient.

Many other works dealt with the evaluation of temperature in solids subjected to frictional heating. In their publication, they presented the analysis of thermo-mechanical behavior of the dry contact between the brake disc and pads during the braking process; the modeling is based on the ANSYS 11.0. They have shown that the ventilation system plays an important role in cooling discs and provides a good high temperature résistance. The analysis result showed that, temperature field and stress field in the process of braking phase were fully coupled. The temperature, Von Mises stress and the total deformations of the disc and contact pressures of the pads increases as the thermal stresses are additional to mechanical stress which causes the crack propagation and fracture of the bowl and wear of the disc and pads. Regarding the calculation results, it was concluded that the results were satisfactory compared to commonly found in the literature investigations. It would be interesting to solve the problem in thermo-mechanical disc brakes with an experimental study to validate the numerical results [39].

Jiangsu,Zhenjiang: Determination of the temperature distribution and comparison of simulation results with experimental results in the disc by 2D thermal analysis using axis-symmetric model was conducted. The disc brake used in the automobile is divided into two parts; a rotating axis-symmetrical disc and the stationary pads. The friction heat which is generated on the interface of the disc and pads can cause high temperature during the braking process. The influence of initial velocity and deceleration on cooling of the brake disc was also investigated. The thermal simulation is used to characterize the temperature field of the disc with appropriate boundary conditions. A Finite element method was developed for determining the critical sliding speed for thermo elastic instability of an axis-symmetric clutch or brake. Linear perturbations on the constant speed solution were sought that they vary sinusoidal in the circumferential direction and grow exponentially in time. These factors cancel in the governing thermo elastic and heat-conduction equations leading to a linear Eigen value problem on the two-dimensional cross-sectional domain for the exponential growth rate for each Fourier wave number.

The imaginary part of this growth rate corresponds to a migration of the perturbation in the circumferential direction. The algorithm was tested against an analytical solution for a layer sliding between two half-planes and provided excellent agreement for both the critical speed and the migration speed. Criteria were developed to determine the mesh refinement required to give an adequate discrete description of the thermal boundary layer adjacent to the sliding interface.
The method was then used to determine the unstable mode and critical speed in geometries approximating current multi-disc clutch practice [17].

Ćirović V. and Aleksendrić D.: They conduct an investigation on the hot spotting in automotive friction system. When sliding occurs with significant frictional heating, thermo-elastic deformation may lead to a transition from smoothly distributed asperity contact to a condition where the surfaces are supported by a few thermal asperities. This circumstance may be associated with a transition to a condition of severe wear because of the elevated contact pressure and temperature and also because of production of tensile stresses. This second stress component may lead to heat checking where upon the rough checked surface acts to abrade the mating material [29].

Ali Belhocine and Mustafa Bouchetara: The motive of undertaking this project of “Coupled Structural / Thermal Analysis of Disc Brake” is to study and evaluate the performance under severe braking conditions and there by assist in disc rotor design and analysis. Their study is of disc brake used for cars. ANSYS package is a dedicated finite element package used for determining the temperature distribution, variation of stresses and deformation across the disc brake profile. An attempt has been made to investigate the effect of stiffness, strength and variations in disc brake rotor design on the predicted stress and temperature distributions. By identifying the true design features, the extended service life and long term stability is assured. A transient thermal analysis has been carried out to investigate the temperature variation across the disc using axis-symmetric elements. Further structural analysis is also carried out by coupling thermal analysis**.** An attempt is also made to suggest a best combination of material and flange width used for disc brake rotor which yields a low temperature variation across the rotor, less deformation and minimum Von Mises stress possible.

The following conclusions are drawn from the present work.

a) An axis-symmetric analysis of disc brake has been carried out using ANSYS 14 software.
b) Transient thermal analysis is carried out using the direct time integration technique for the application of braking force due to friction for time duration of 4, 5 and 6 seconds.
c) The maximum temperature obtained in the disc is at the contact surface.
d) Static structural analysis is carried out by coupling the thermal solution to the structural analysis and the maximum Von Mises stress is observed.
e) The brake disc design is safe based on the strength and rigidity criteria.
f) To arrive at a best combination of parameters of the disc brake like flange width and material, transient thermal and structural analysis for three different combinations in each of the three different analyses are carried out separately and the results were compared.
g) Comparing the different results obtained from the analysis, it is concluded that disc brake with 10 mm flange width of 6.5 mm wall thickness and of material cast iron is the best possible combination for the present application.

In their investigation of thermal analysis of disc brake, a simplified model of the disc brake without any vents with only ambient air cooling is analyzed by FEM package ANSYS. As a future work, a complicated model of ventilated disc brake can be taken and there by forced convection is to be considered in the analysis. The analysis still becomes complicated by considering variable thermal conductivity, variable specific heat and non uniform deceleration of the vehicle. This can be considered for the future work [27].

Adam Adamowicz, Piotr Grzes: The project is conducted to design, model a disc brake. The disc brake is a device for slowing or stopping the rotation of a wheel. A brake disc (rotor), usually made of cast iron or ceramic composites (including carbon, kevlar and silica), 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 friction to heat but if the brakes get too hot, they will cease to work because they cannot dissipate enough heat. This condition of failure is known as brake fade.

Disc brakes are exposed to large thermal stresses during routine braking and extraordinary thermal stresses during hard braking. The aim of the project is to design, model a disc brake. Modeling is done using Pro/Engineer. Structural and thermal analysis is to be done on the disc brakes using two materials stainless steel and carbon steel. Structural analysis is done on the disc brake to validate the strength of the disc brake and thermal analysis was done to analyze the thermal properties. Comparison can be done for displacement, stresses, thermal gradient etc. for the two materials to check which material is best. They are also providing manufacturing process for making disc brake and also preparing prototype. Manufacturing process is done using Pro/Engineer. Pro/Engineer is a 3d modeling software widely used in the design process. 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. Using the software ANSYS with the FEM could bring best solution. It is suggested that carbon steel is best material for brake disc application [28].

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

When this process leads to an accelerated change of contact pressure distribution, the unexpected hot roughness of thermal distortion may grow unstably under some conditions resulting in local hot spots and leaving thermal cracks on the disc. This is known as thermo-elastic instability (TEI).The thermo elastic instability phenomenon occurs more easily as the rotating speed of the disc increases. This region where the contact load is concentrated reaches very high temperatures which cause deterioration in braking performance. Moreover, in the course of their presence on the disc the passage of thermally distorted hot spots moving under the brake pads causes low frequency brake vibration. Their investigation is aimed to study the given disc brake rotor of its stability and rigidity (for the thermal analysis and coupled structural analysis) was carried out on a given disc brake rotor and to investigate best combination of parameters of disc brake rotor like flange width, wall thickness and material there by a best combination was suggested [34].

YU Liang, JIANG Yan-Li, LU Sen-kai, LUO Kun, RU Hong-qiang.: Thermal and structural analysis of vented and normal disc brake rotors, steady state and transient responses have been conducted through the heat transfer analysis to predict the worst case scenario and temperature behaviors of disc brake rotor. In their study, finite element analysis approached has been conducted in order to identify the temperature distributions and behaviors of disc brake rotor in steady state and transient responses. ANSYS has been used as finite elements software to perform the thermal analysis on both responses. Both results have been compared for better justification. Thus both results provide better understanding on the thermal characteristic of disc brake rotor and assist the automotive industry in developing optimum and effective disc brake rotor. The heat generated on the surfaces of disc brake rotor when brake is applied. Materials of disc brake rotor usually are made from cast iron, spheroidal graphite cast iron or cast steel. It is chosen as a rotor material due to low cost of material and performs high thermal resistance. This type of material normally suit to normal passenger vehicle but not for high performance car. Once brake pads contacts to rotating rotor, there will be huge amount of heat generated to stop or slow down the vehicle. The rotor temperature can exceed 350 for normal cars and 1500 for race cars. Disc brake rotor is a crucial part in the brake system where the main role of the rotor is to reduce the heat generated by dissipating all of the heat. In that case, ventilated disc brake rotor is much better than solid rotor where more airflow from the surrounding area to dissipate produced heat.

The internal vanes allow air to circulate between two friction surfaces of the rotors. Their study can provide a useful design tool and improve the brake performance of disc brake system. Hence the brake disc design is safe based on the strength and rigidity criteria. Comparing the different results obtained from analysis. It is concluded that ventilated type disc brake was the best possible for the present application [51].

Khong Keng Leng, AM34.: Transient thermal and structural analysis of the rotor disc of disc brake is aimed at evaluating the performance of disc brake rotor of a LRT under severe braking conditions and there by assist in disc rotor design and analysis. An investigation into usage of new materials was required which improve braking efficiency and provide greater stability to vehicle. This investigation can be done using ANSYS software. ANSYS 11.0 is a dedicated finite element package used for determining the temperature distribution, variation of the stresses and deformation across the disc brake profile. In the their work, an attempt has been made to investigate the suitable hybrid composite material which is lighter than cast iron and has good young's modulus, yield strength and density properties. High strength glass fiber composites have a promising friction and wear behavior as a disc brake rotor. The transient thermo-elastic analysis of disc brakes in repeated brake applications has been performed and the results were compared. The suitable material for the braking operation is S2 glass fiber and all the values obtained from the analysis are less than their allowable values. Hence the brake disc design is safe based on the strength and rigidity criteria. By identifying the true design features, the extended service life and long term stability is assured.

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

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

To obtain the simulation of thermo-elastic behavior appearing in disc brakes, the coupled heat conduction and elastic equations 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. It is observed that the orthotropic disc brakes can provide better brake performance than the isotropic ones because of uniform and mild pressure distributions.

Their study can provide a useful design tool and improve the brake performance of disc brake system. It can be said that S2 glass fiber is the suitable material for the braking operation and all the values obtained from the analysis are less than their allowable values. Hence the brake disc design is safe based on the strength and rigidity criteria [41].

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

By observing the structural analysis and thermal analysis results using aluminum alloy and carbon reinforced polymer the stress values are within the permissible stress value. So using aluminum alloy and carbon reinforced polymer is safe for disc brake. By observing the frequency analysis, the vibrations are less for aluminum alloy than other two materials since its natural frequency is less. And also weight of the aluminum alloy reduces almost 3 times when compared with alloy steel and cast iron since its density is very less. Thereby mechanical efficiency will be increased. But the strength of carbon reinforced material is more than aluminum alloy. Since the thermal analysis also carbon reinforced is also permissible. By observing analysis results, carbon reinforced polymer is best material for disc brake [36].

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

The temperature distribution due to friction process necessitates a good knowledge of the contact parameters. In fact, the interface is always imperfect due to the roughness from a mechanical and thermal point of view. Recent theoretical and experimental works have been developed to characterize the thermal parameters which govern the heat transfer at the vicinity of a sliding interface. Increased thermal efficiency and the integrity of materials in high temperature environments is an essential requirement in modern engineering structures in, automotive, aero space, nuclear, offshore, environmental, and other industries. Nowadays, the finite element method is used regularly to obtain numerical solutions for heat transfer problems. The most common choice when using finite elements is standard Galerkin formulation.

At conclusion part of their publication, it had been presented the thermo-mechanical behavior analysis of the dry contact between the disc and brake pads during the braking process; the modeling is based on the ANSYS software 11.0. They have shown that the ventilation system plays an important role in cooling discs and provides a good high temperature resistance.

The analysis results showed that, a temperature and stress fields in the braking process phase of were fully coupled. The temperature, the von Mises stress and the total deformations of the disc and the contact pressure distribution in the pads increases as the thermal stresses are additional to mechanical stress which causes the crack propagation and fracture of the bowl and wear of the disc and brake pads. Regarding the calculation results, it can be said that they are satisfactory compared to commonly found in the literature investigations. It would be interesting to solve the thermo-mechanical problem of disc brakes with an experimental study to validate the numerical results [33].

Milo {evi}, M. S., et al.: The objective of the study is to investigate the temperature and thermal stress in the ventilated disc/pad brake during single brake. The brake disc is decelerated at the initial speed with constant acceleration until the disc comes to a stop. The ventilated brake pad and disc brake assembly is built by a 3D model with a thermo-mechanical coupling boundary condition and multi body model technique. To verify the simulation results, an experimental investigation is carried out. The objective of the study is to investigate the temperature and thermal stress in the ventilated brake disc and brake pad during single brake based on multi body technique and 3D thermo-mechanical coupling model.

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

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

This thermo-mechanical coupling behavior due to a relative high sliding speed that exceeds the critical sliding speed can be unstable, leading to localized high temperature contact regions called “hot spots” on the sliding interface. This shows the temperature field with the distortional hot spot in the ventilated brake disc in the different stages of braking operation. The temperature distribution is not uniform field, since the frictional heat generated in the contact region and the rest region dissipates the heat by the convection and heat conduction of the disc. The maximum temperature is occurred around the middle circle of the contact surface at the exit of contact region, since the middle circle region is where the maximum contact pressure region is generated Temperature in the surfaces of vanes is lower than the temperature in the work surface, because of the conductive behavior in the disc and higher convection in the vanes. The temperature field in the pad presents no uniformity characteristics, and presents approximately axis-symmetric at the end of braking, similar with to the temperature field in the disc. Temperature field affects the thermal expansion and leads to variation of contact pressure distribution.

Due to the heat generation ratio decrease, conduction in the disc and convection on the surface of brake disc temperature field presents approximately axis-symmetric in the end stage of the braking operation.
Lower temperature in the vanes is due to the effect of disc conductivity and higher convection in the vanes. The node temperature on the work surface in the disc is presenting fluctuation. The thermal strain and thermal stress distributions are in accordance with the temperature distribution [4].

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

Many research results have confirmed, the dominant influence is the thermal loads in regard to mechanical loads and residual stresses induced by high thermal loads in brake discs. Therefore, it is important to determine with high precision the temperature field of the braking system, as well as to emphasize that high thermal loads, in other words, overloads, of disc very often occur as a result of long term braking on down grade railroads or unwanted locking of wheels and braking to standstill. Since the Ethiopian railway corporation significantly will be benefited from this work for the reason that knowing the temperature distribution of the brake disc and the friction pad at different potential location of the changing slope of the Addis Ababa LRT track is of great importance. Therefore this research is established to address and help designers understand the condition that the Addis Ababa LRT brake discs could encounter during braking operation at potential areas of sloppy LRT track. This research is designed to come up with solution by investigating the temperature, mechanical and thermal analysis of the continuous SiC/6061 Al alloy composite material (SiC/6061 Al) made brake disc for Addis Ababa LRT under centrifugal and thermal load during long term braking for maintaining a constant speed on a downgrade railroad and braking to standstill considering airflow cooling.

CHAPTER THEE

MODEL PREPARATION OF DISC BRAKE

3.1 Disc Brake

The brake disc is a brake which slows rotation of the wheel by the friction caused by pushing brake pads against a brake disc with a set of calipers. The disc brake is usually made of grey cast iron and structural steel but may in some cases it can be made from composition such as reinforced carbon-carbon or ceramic matrix composition. This is associated to the wheel or the axle. Brakes convert motion to heat and if the brakes get too hot, they become less effective and this phenomenon is known as brake fade [21]. The friction surface is invariably displayed to the air checking good heat dissipation minimizing brake fade. It also allows for self cleaning as dust and water is thrown off for diluting friction difference. Unlike drum brakes, disc brakes have limited self energizing action making it necessary to apply greater pneumatic pressure to find enough braking force. This is achieved by increasing the size of the caliper clamp force. The simple design helps easy maintenance and pad replacement [22]. Disc brakes could be solid or ventilated according to the required performance and condition of the braking. But in this research paper ventilated type of disc brake is investigated.

3.1.1 Ventilated Brake Disc

The ventilated disc is lighter than the solid disc and additional convective heat transfer occurs on the surface of the vent hall. Thus the ventilated disc can control its temperature rise and minimize the effects of thermal problems such as the variation of the pad friction coefficient, brake fade and wheel lock [43, 44].

The ventilated disc however, may increase judder problems by inducing an uneven temperature field around the disc. Also the thermal capacity of the ventilated disc is less than that of the solid disc and the temperature of the ventilated disc can rise relatively faster than that of the solid disc during repetitive braking [45]. Therefore, thermal capacity and thermal deformation should be carefully considered when modifying the shape of the ventilated disc.

3.1.2 Disc Rotor

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

3.1.3 Brake Pads

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

3.2 Materials for Brake Disc and Brake Pad

3.2.1 Rotor Disc

The braking system is a vital safety component of ground based transportation systems. Hence the structural materials used in brakes should possess some combination of properties such as good compressive strength, higher friction coefficient, wear resistant, light weight, good thermal capacity and economically viable [11].

3.2.2. Candidate Material for Brake Disc

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

Cast Iron: Metallic iron containing more than 2% dissolved carbon within its matrix (as opposed to steel which contains less than 2%) but less than 4.5% is referred to as grey cast iron because of its characteristic color. Considering its cost, relative ease of manufacture and thermal stability this cast iron (particularly grey cast iron) is actually a more specialized material for brake applications particularly the material of choice for almost all automotive brake discs. To work correctly, the parts must be produced at the foundry with tightly monitored chemistry and cooling cycles, to control the shape, distribution and form of the precipitation of the excess carbon. This is done to minimize distortion in machining, provide good wear characteristics, damped vibration and resist cracking in subsequent use [40].

Titanium alloys: Titanium alloys and their composites have the potential to reduce weight of the brake rotor disc component which is about 37% less than a conventional cast iron with the same dimensions and offering good high temperature strength and better resistance to corrosion [23].

Aluminum-Metal Matrix Composite (AMC): Aluminum alloy based metal matrix composites (MMCs) with ceramic particulate reinforcement have shown great promise for brake rotor applications. These materials having a lower density and higher thermal conductivity as compared to the conventionally used grey cast irons are expected to result in weight reduction of up to 50-60% in brake systems. The repeated braking of the AMC brake rotor lowered the friction coefficient μ and caused significant wear of the brake pad. The friction properties of the AMC brake disc are thus remarkable poorer than those of conventional brake disc. After increasing hard particles content the result showed that the repeated braking operations did not lower the friction coefficient. Wilson et. al. [52] studied the abrasive wear resistance of the AA6061 with 20 vol % SiC reinforced composite in short sliding distance testing (about 20m). Adding 20 vol. % SiC particulate greatly enhanced the wear resistance, raised room-temperature strength and stiffness, and improved high temperature strength [53]. Three major problems exist with this aluminum composite rotor. First, because of the density difference between aluminum and SiC, segregation or inhomogeneous distribution of SiC particles during solidification cannot be avoided. Also, adding SiC particles in an aluminum matrix dramatically reduces the ductility of the material, resulting in low product liability. The third problem is a lack of a solid lubricant, such as graphite. The lack of graphite in the system results in low braking efficiency, adhesive wear, and galling. In a cast iron rotor, graphite is always present in the iron. As the break wears, the graphite is freed from the iron matrix to be used as a solid lubricant on the wear surface. An aluminum SiC and graphite composite brake rotor was developed and reported as having better wear resistance than a cast iron rotor [55].

This material contains 10 vol % SiC particulate and 5 vol % nickel-coated graphite particulate reinforcement in an aluminum-silicon alloy matrix. These types of rotors were produced by casting. Although incorporating graphite particulate improved the wear resistance. The wear resistance and frictional performance of Al-Cu alloys reinforced with SiC particles are superior to those of cast iron brake rotors [53]. In addition, the lower density of aluminum MMCs gives them an economic advantage over cast iron with respect to efficient use of fuel and fabrication expenses.

Based on the properties, potential candidate materials for brake disc were selected as:

- Grey cast iron (GCI)
- Ti-alloy (Ti-6Al-4V)
- 7.5 wt% WC and 7.5 wt% TiC reinforced Ti-composite (TMC)
- 20% SiC reinforced Al-composite (AMC 1)
- 20% SiC reinforced Al-Cu alloy (AMC 2)

Table 3.1 Properties of candidate materials for brake disc [53, 55]

Material	**Properties**				
Name	Comprehensive strength(MPa)	Friction coefficient(μ)	Wear rate ($10^{-6}mm^3$ / N/m)	Specific heat $C_p(KJ/Kg.K)$	Specific gravity(Mg/m^3)
GCI	1293	0.41	2.36	0.46	7.2
Ti-6Al-4V	1070	0.34	246.3	0.58	4.42
TMC	1300	0.31	8.19	0.51	4.68
AMC 1	906	0.42	2.25	0.98	2.7
AMC 2	761	0.44	2.40	0.92	2.8

In this research SiC/6061aluminium alloy composite material is selected based on the crucial requirement of disc brake. Wilson Et. Al. studied the abrasive wear resistance of the AA6061 with 20 vol % SiC reinforced composite in short sliding distance testing (about 20m). Adding 20 vol. % SiC particulate greatly enhanced the wear resistance, raised room temperature strength and stiffness and improved high temperature strength [53].

Since composites tend to have the following characteristics: high strength, high modulus, low density, excellent resistance to fatigue, creep, creep rupture, corrosion, and wear and low coefficient of thermal expansion (CTE) [48, 52].

The SiC/6061 Al composite was obtained by infiltration of molten 6061 Al alloy (0.40%−0.80% Si, 0.70% Fe, 0.15%−0.40% Cu, 0.15% Mn, 0.80%−1.20% Mg, 0.04%−0.35% S Cr, 0.25% Zn, 0.15% Ti, balance Al) into SiC network ceramic by vacuum-pressure casting process [51]. The SiC continuous network structure ceramic preforms were prepared by the porous polyurethane coated with large amounts of aqueous reactant containing α-SiC power (purity>99%, d50=1.5 μm). The microstructure of SiC/6061 Al is shown in Fig. 1. The bright phase is the 6061 Al alloy matrix, and the dark phase is the SiC network. The models of the SiC/6061 Al brake disk for LTR train are shown in Fig. 2. The white part shown in Fig. 2(a) is the 6061 Al alloy matrix for brake disk. The black parts shown in Figure 2(b) are SiC/6061 Al alloy composites which are embedded in the disk. In this analysis the brake disk without wear model was considered.

The brake disc has 150 cooling vanes equally spaced, which enables the usage of symmetry with a basic angle of 15°.And also weight of the SiC/6061 Al composite material reduces almost 2 times when compared with grey cast iron since its density is very less.

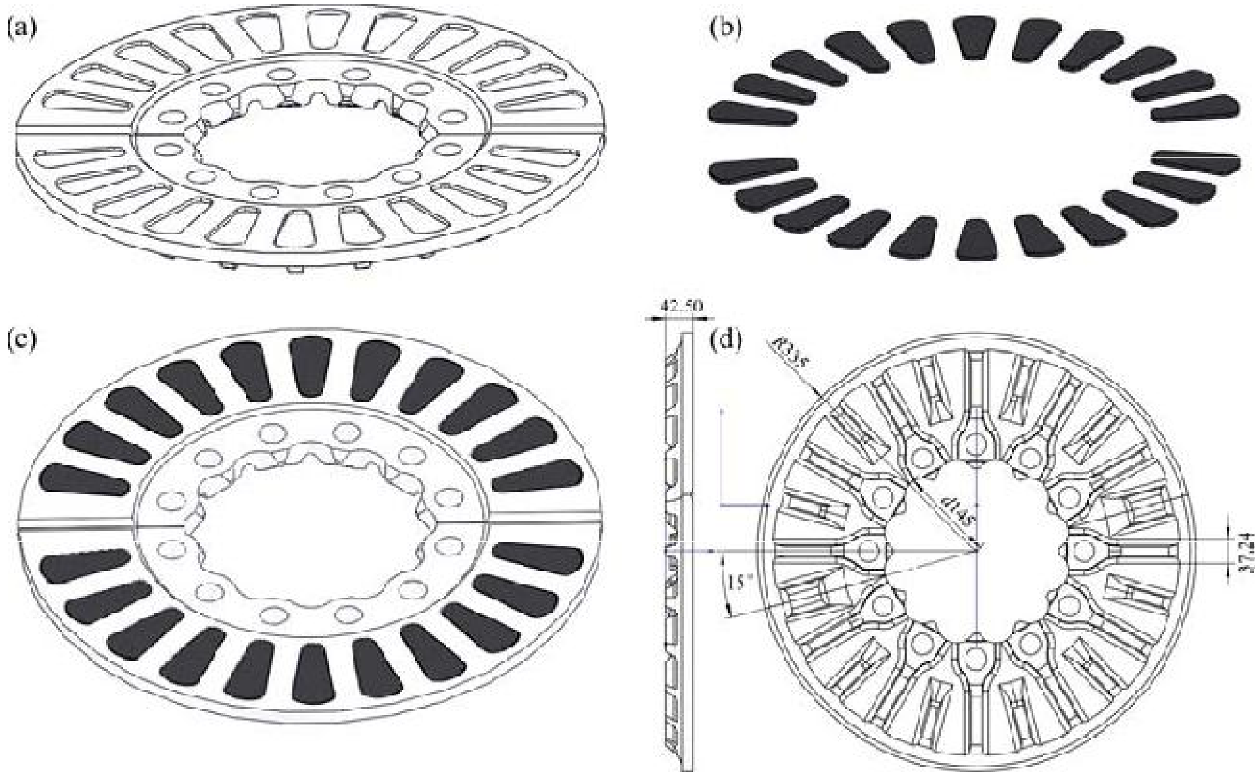


Figure3.1 SiC/6061 Al embedded in brake disc: (a) 6061 Al alloy matrix for brake disc;
(b) SiC/6061 alloy composites array; (c) SiC/6061 Al alloy brake disc; (d) Basic geometry of section of brake disc.

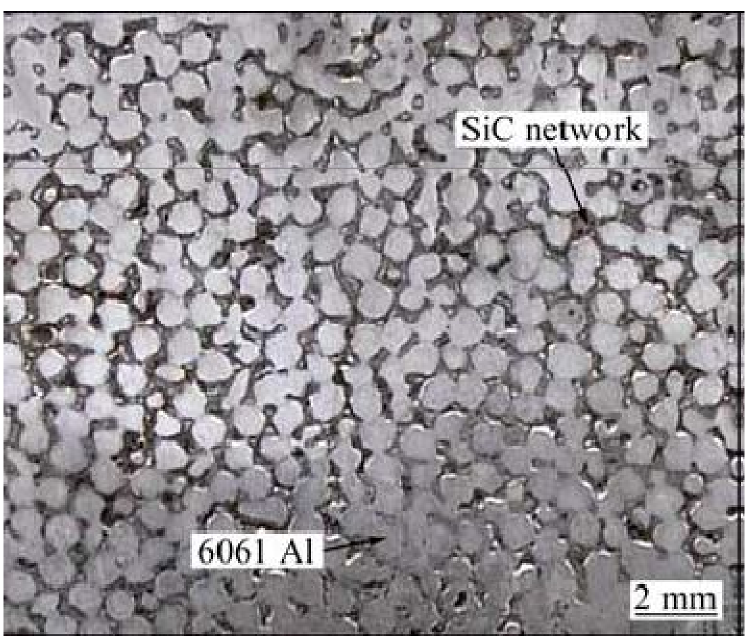


Figure3.2 Microstructure of SiC/6061 Al alloy composites

3.2.3 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 and decrease the weight of brake system and raise the service life. Therefore, many advanced and commercial vehicles have employed C/C composite brake pads. Carbon/carbon (C/C) composites are prepared using rapid direct diffused chemical vapor infiltration (RDDCVI) processes. CVI technology is the most common method to prepare C/C composite brake pad and the discs made are more uniform and contain fewer defects in microstructure. However, the traditional CVI technology would often take over a thousand hours of furnace heat time to densify samples. A new rapid direction diffused chemical vapor infiltration (RDD CVI) method is developed in rapid densification of C/C composites [9] but like other categories of C/C composites, brake pad characteristics of RDD CVI C/C composites are also much influenced by the processing and operating conditions. Therefore, research of the effect of these factors on brake features of C/C composites prepared using RDD CVI processes is significant in their fabrication and proper application as well as in the friction theory.
The material use for the brake lining should have the following characteristics.

- It should have high coefficient of friction with minimum fading. In other words, the coefficient of friction should remain constant with change in temperature.
- It should have low wear rate
- It should have high heat resistance
- It should have high heat dissipation capacity
- It should have adequate mechanical strength and
- It should not be affected by moisture and oil [31].

3.5 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 [15] they do not have constant chemical and 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 [kg/m3], the thermal conductivity [W/m.K] and the specific heat [J/kg.K] of the disc's (index *d*) and braking pad's materials respectively (index *p*). The brake disc assumes the most part of the heat usually more than 90% [17] 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 (Fig.3.3). The initial heat flux q entering the disc is calculated by the following formula [1]:

$$\boldsymbol{q_o = \frac{1-\varphi}{2}\frac{mgZv_o}{2A_c\varepsilon_p}} \qquad \textbf{3.1}$$

Where $z = a/g$: Braking effectiveness, a : Deceleration of the vehicle [m/s-2], φ : Rate distribution of the braking forces between the front and rear axle, A_c : Disc surface swept by a brake pad [m2], v_o : Initial speed of the vehicle [m/s-1], ε_p : Factor load distribution of the on the surface of the disc., *m* : Mass of the vehicle [kg] , g : Acceleration of gravity (9.81) [$^{m}/_{s^2}$].

The disc rotor and its vane passage chosen for the numerical analysis are shown in Fig. 3.3 the dimension for the baseline study with an outer diameter of 460 mm and inner diameter of 220mm and 150 vanes. The geometric model was created using CATIA V5 software.

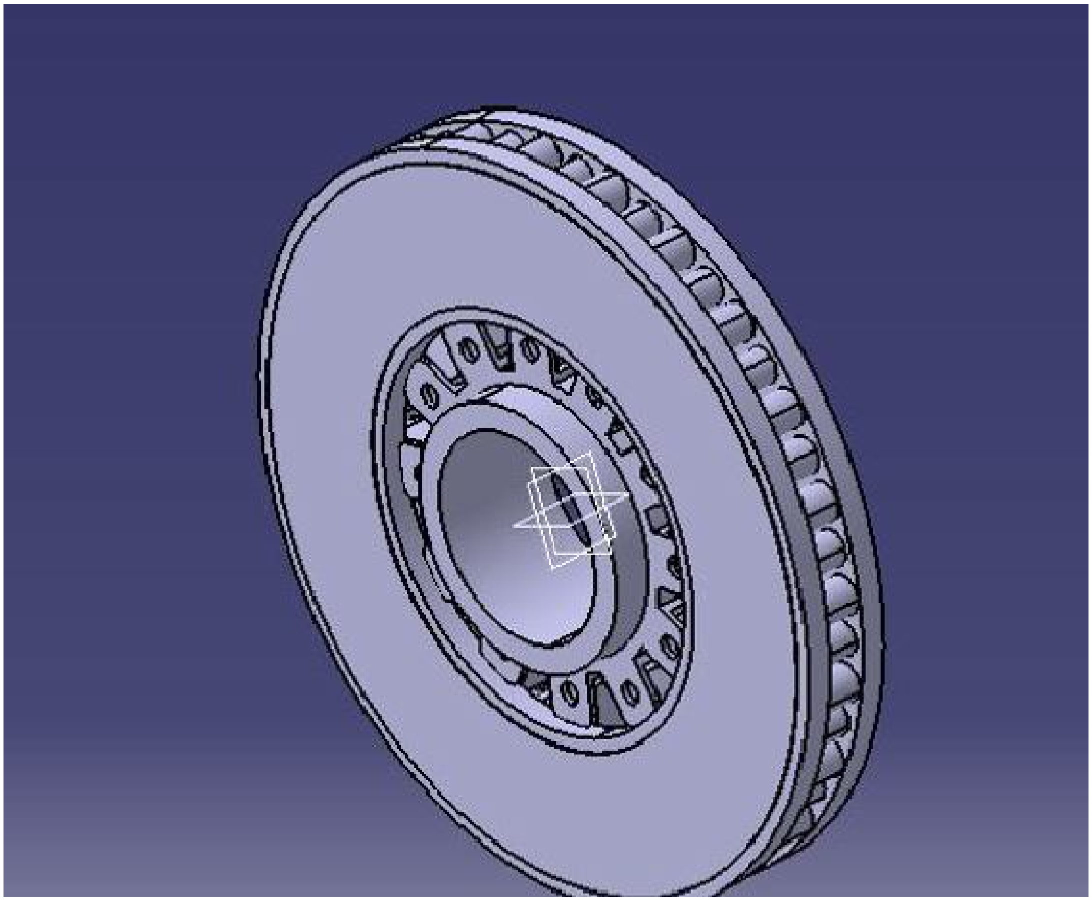


Figure3.3Geometric characteristics of ventilated disc brake of full geometry of brake disc used in ANSYS CFX analysis.

3.6 Building the 3D Model Using ANSYS CFX for Flow Analysis

Brake disc convective cooling has been historically studied by means of experimental and theoretical methods [3,4] and the optimization were only boosted with the advent of modern computational resources in the late eighties [56]. Currently, although of not simple usage and requiring previous understanding of the basics of fluid mechanics and heat transfer coupled with knowledge of flows numerical modeling CFD has significantly gained preference in the automotive industry design process as tool for predicting complex flow and heat transfer behavior in regions where otherwise very laborious and time consuming by a combination of node to surface and surface to surface contact elements [57].

In this study, modeling of the thermal behavior of the dry contact between the discs and brake pads at the time of braking phase. The strategy of calculation is based on the software ANSYS 14. This last is comprehensive mainly for the resolution of the complex physical problems. As a current study of this problem ANSYS simulations with less assumption and less program restrictions have been performed for the thermo-mechanical case. Temperature distribution obtained by the transient thermal analysis is used in the calculations of the stresses on disc surface.

It is very difficult to exactly 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, we always ignore the things that are of less importance and have little impact on the analysis. The assumptions are always made depending upon the details and accuracy required in modeling.

Due to the application of brakes on the A.A. LRT disc brake rotor, heat generation takes place due to friction and this thermal flux has to be conducted and dispersed across the disc rotor cross section. The condition of braking is very much severe and thus the thermal analysis has to be carried out. The thermal loading as well as structure is axis-symmetric. Hence axis-symmetric analysis can be performed but in this study I have performed a 3D analysis which is an exact representation for this thermal analysis. Thermal analysis is carried out and with the above load mechanical analysis is also performed for analyzing the stability of the material.

The loading corresponds to the heat flux on the disc surface, the dimensions and the parameters used in the thermal calculation are recapitulated 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*	170
Rate distribution of the braking forces ϕ, %	12
Track inclination δ, $^{0}/_{00}$	11
Surface disc swept by the pad A_c, mm^2,	128890
Wheel diameter, *mm*	660
Deceleration a, m/s^2	1.4
Initial speed *V*o, *km/h*	80
Vehicle mass m, *kg*	64000

The material property of SiC/6061Al alloy composite of brake disc with good thermo-physical characteristics and thermo-elastic characteristics of which adopted in this simulation of the rotor disc are recapitulated in Table (3.2).

Table3.3Material properties of brake disc

Material Properties	**Disc**
Thermal conductivity, k (*W/m k)*	5
Density, ρ (*kg/m3)*	3700
Mass of the brake disc, *(Kg)*	33.038
Specific heat, *C (J/Kg k)*	650
Poisson’s ratio, v	0.13
Thermal expansion, α *(10-6 / K)*	0.15
Elastic modulus, E *(GPa)*	254
Allowed temperature, *(*℃ *)*	600
Tensile yield strength, *(Mpa)*	314
Ultimate tensile strength, *(Mpa)*	410

Table3.4 Material properties of brake pad

Material Properties	**Pad**
Thermal conductivity, *k (W/m °C)*	26
Density, *ρ (kg/m3)*	1800
Specific heat, *C (J/Kg °C)*	880
Poisson's ratio, v	0.3
Thermal expansion, *α (10-6 / K)*	0.3
Elastic modulus, *E (GPa)*	50.2

3.7 Assumptions Used for the Development of the Physical Model of Brake Disc

To simplify the analysis, several assumptions have also been made and it is recapitulated as follows [27]:

- All kinetic energy at disc brake rotor surface is converted into frictional heat or heat flux. In other words, a high amount of kinetic energy is converted into heat energy at interfaces according to the first law of thermodynamics during the slipping period and the heat generated between contact surfaces will be dissipated by convection to the environment.
- The heat transfer involved for this analysis is only convection process. The heat transfer due to radiation can be neglected in this analysis because of its small amount and contribution which is 5 % to 10 % [31]. Indeed, the heat radiation only plays an important role at high temperature and low speeds.
- The disc material is considered as homogeneous and isotropic because young's modulus, Poisson's ratio and the thermal expansion coefficient are assumed to be constant for isotropic material.
- The domain is considered as axis-symmetric. The temperature field is symmetry with respect to the central plane of the brake disc.
- Inertia and body force effects are negligible during the analysis. Here a transient analysis calculates the effects of thermal loading on a structure while ignoring inertia and damping effects.

- The disc is stress free before the application of brake. Due to the application of brakes of the rotor, heat generation takes place due to friction and this temperature so generated has to be conducted across the disc across section.
- In this analysis, the ambient temperature and initial temperature has been set to 38 °C.
- All other possible disc brake loads are neglected for example, the effects of wear or other atmospheric factors.
- Only certain parts of disc brake rotor will apply with convection heat transfer such as cooling vanes area, outer ring diameter area and disc brake surface on which, each surface of the rotor was subjected to different values of convection heat transfer coefficient obtained from this calculations.
- Uniform pressure distribution by the brake pad onto the disc brake surface. Uniform pressure distribution in the contact region is often valid when the pad is new.

3.8 Preparation of the Geometry and FEM Discretization

3.8.1 ANSYS ICEM CFD Modeling and Mesh Preparation

Here are the procedures for developing brake disc model using ANSYS ICEM CFD [30].

a) Create/import the Geometry

The geometry will be created in units of millimeters and then scaled to meters at the output step. Begin this phase by selecting “Geometry” from the option tabs. And select import geometry from the menu having a file type of IGS, since the geometry was developed using CATIA SOFTWARE.

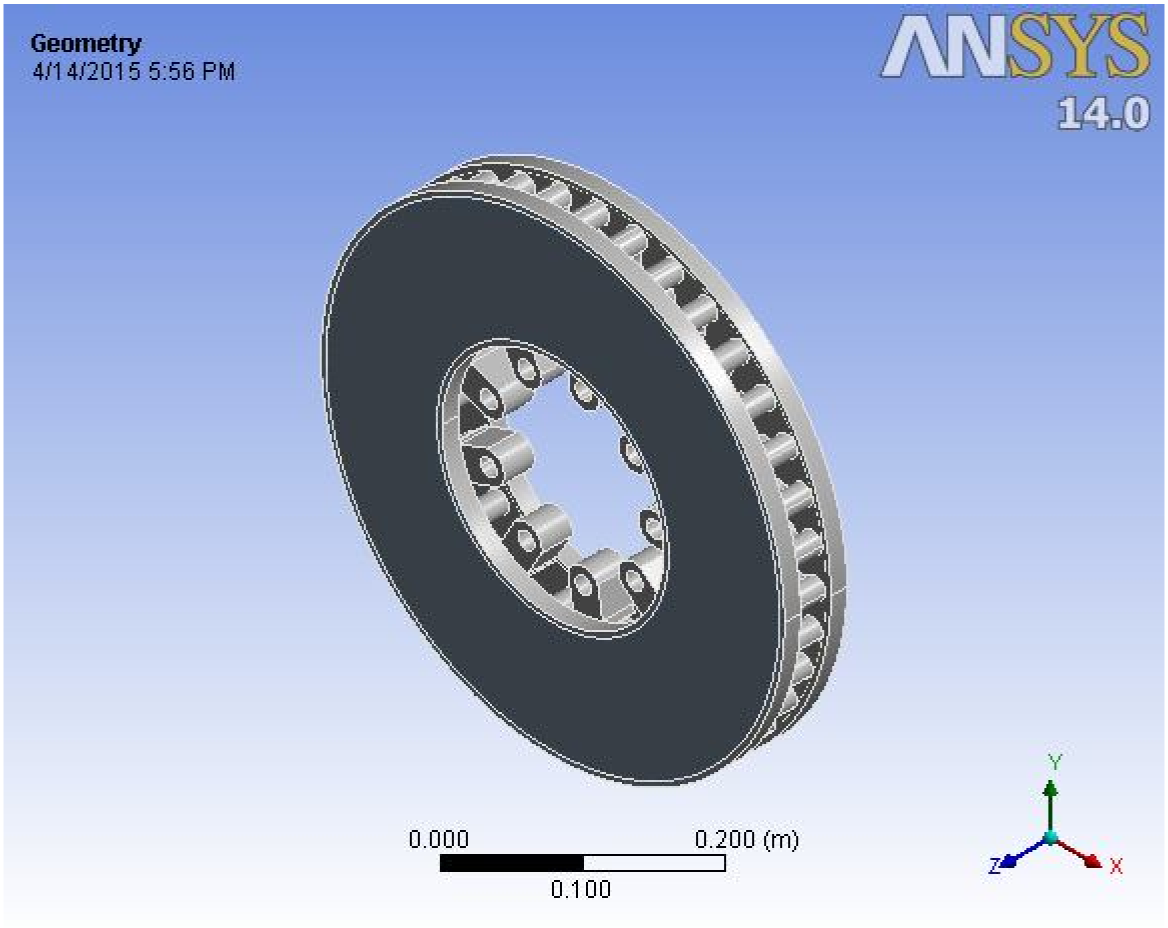


Figure3.4 Imported geometry of the brake disc

b) 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 ‘duct’. 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.

- Press the 'dismiss' button in the sidebar. Note that the way we defined this body it is in the centre of the domain and that any two, opposite vertices could have been chosen with the same final result. Also, the 'by topology' option in the 'create body' sidebar could have been chosen and either the entire model or all of the surfaces used to define the body in the centre.
- You may now turn on the surfaces again in the model tree.

c) 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 we 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.
- Within the sidebar, enter the name TOP in the 'Part' field and click the 'Create Part by Selection' button.
- Within the sidebar, click the 'Select entities' button.
- In the isometric view, select the top surface of the geometry in the view window:
- Press the middle mouse button, and then select 'Apply' in the sidebar. The new part TOP will now appear in the geometry under 'Parts'. Uncheck TOP, so the surface is no longer displayed. Removing top from the view window will make it easier to select the other surfaces.
- For the remaining five surfaces, follow the procedure above. Remember to uncheck each part as it is created. This will also help to know which surfaces are remaining.
- Once all of the parts are created, press 'dismiss' in the sidebar
- Check points, curves, surfaces, bodies, and all the new part names in the model tree

d) Creating a Block

Select 'blocking' from the option tabs.

- Select 'create block' from the option buttons. The new sidebar 'create block' will be opened.
- From within the sidebar select the 'initialize blocks' button and set the 'part' field to ‘duct’. Leave the 'type' field as its default '3D bounding box'.
- Reduce the size of the geometry in the view window (by scrolling the mouse scroll button or using the right mouse button). The image must not be near the edges of the view window because you are going to select all of it next.
- Click the “select geometry” button.
- In the view window, select all of the geometry by clicking, holding, and dragging a box around the entire geometry. Press the middle mouse button and then press 'apply' in the sidebar. Press 'Dismiss' in the sidebar. In the model tree, you should now have a new branch named 'blocking'. Also in the model tree, under 'parts', you should have something named 'VORFN'.
- In the model tree, under 'geometry', turn off 'points', 'curves', and 'surfaces' so that only 'bodies' has a check mark next to it. You should see the following after pressing 'fit window' and choosing 'isometric' from 'view' in the main menu: You are seeing the edges of the block. They are colored white until they are associated with a curve in your geometry.

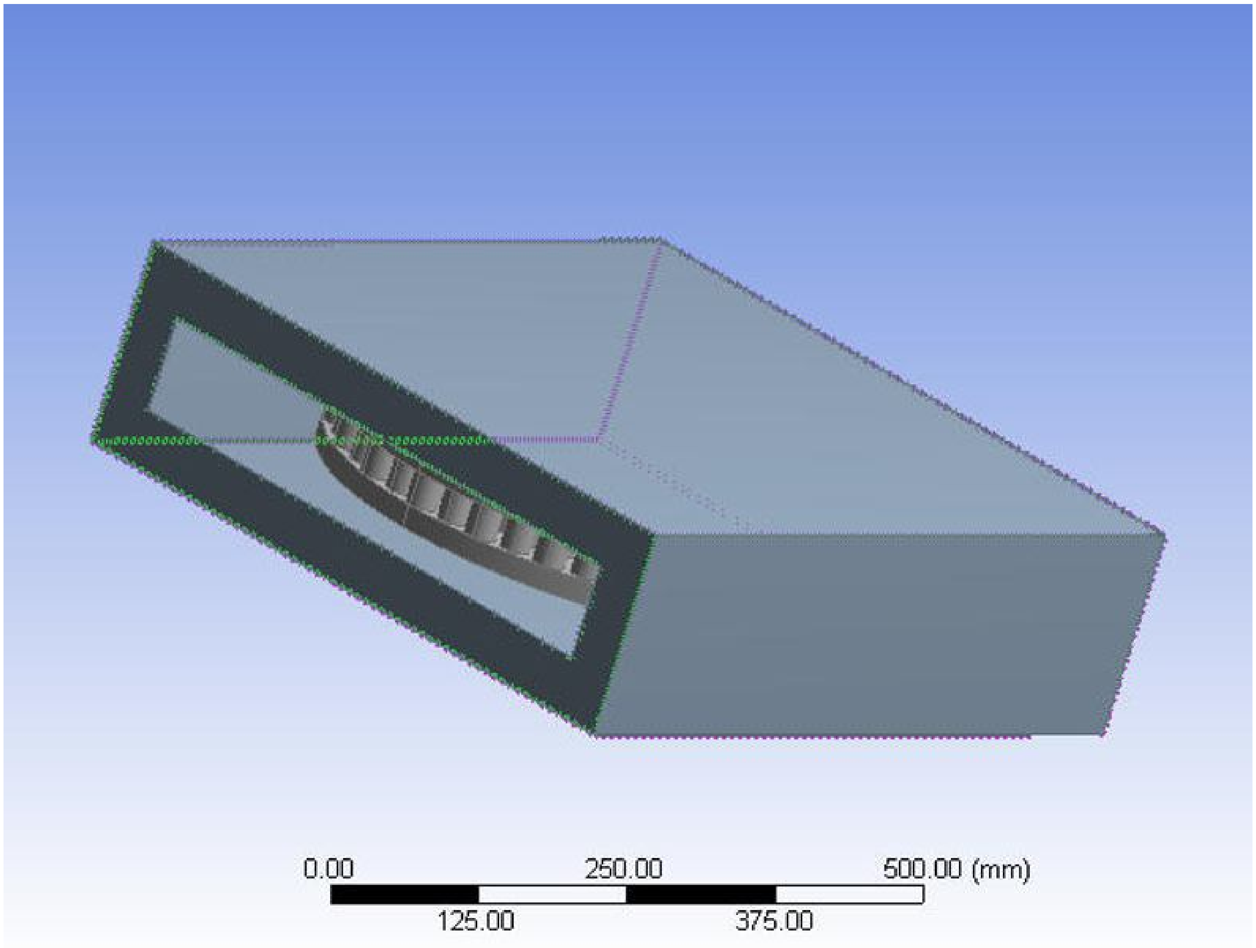


Figure3.5 Imported geometry of the brake disc after fluid boundary blocking

e) Doing the Edge Associations

Associate the created geometry to the block

- Select 'associate' from the option buttons. The new sidebar 'blocking associations' will be opened. From within the sidebar select 'auto association'.

Leave the default 'snap project vertices' and then press 'apply' and 'dismiss' in the sidebar. You should see the white lines turn green:

Note that auto-association can only be used because this is a very simple geometry. For more complicated geometries, manual association must be performed.

f) Nodal Distributions

Allocate nodal distribution along the block edges

- Select 'pre-mesh Params' from the option buttons. The new sidebar 'pre-mesh params' will be opened.
- From within the sidebar select 'edge params'.
- Click 'Select edge(s)
- In the sidebar again, enter 50 in the 'nodes' field and set the 'mesh law' field to 'big geometric' (note that you may have to reselect the curve and enter the number of nodes a second time).
- Scroll down the sidebar and select 'copy parameters' and then set the 'method' field to 'to all parallel edges'.
- Press 'Apply' and you should now see small red ticks along all the lines in the z-direction. There is also the number of nodes printed in red.

g) Meshing the Block

- Select 'file' from the main menu, then select '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.

h) 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'.
- Click 'Apply' and 'dismiss'.
- Select 'write input' from the option buttons.

- You will be prompted to save the current project first, select 'yes'
- A new window with the title 'CFX5' will appear. Highlight 'yes' in the 'scaling' field and set the x-, y- , and z-scaling factors to 0.001. Note the scaling is necessary because we input dimensions 1000 times greater than those dictated by the geometry.
- Click done.

This stage consists in preparing the mesh of the fluid field. In our case, one used a linear tetrahedral element with 67298 nodes and 34448 elements (Fig. 3.6).

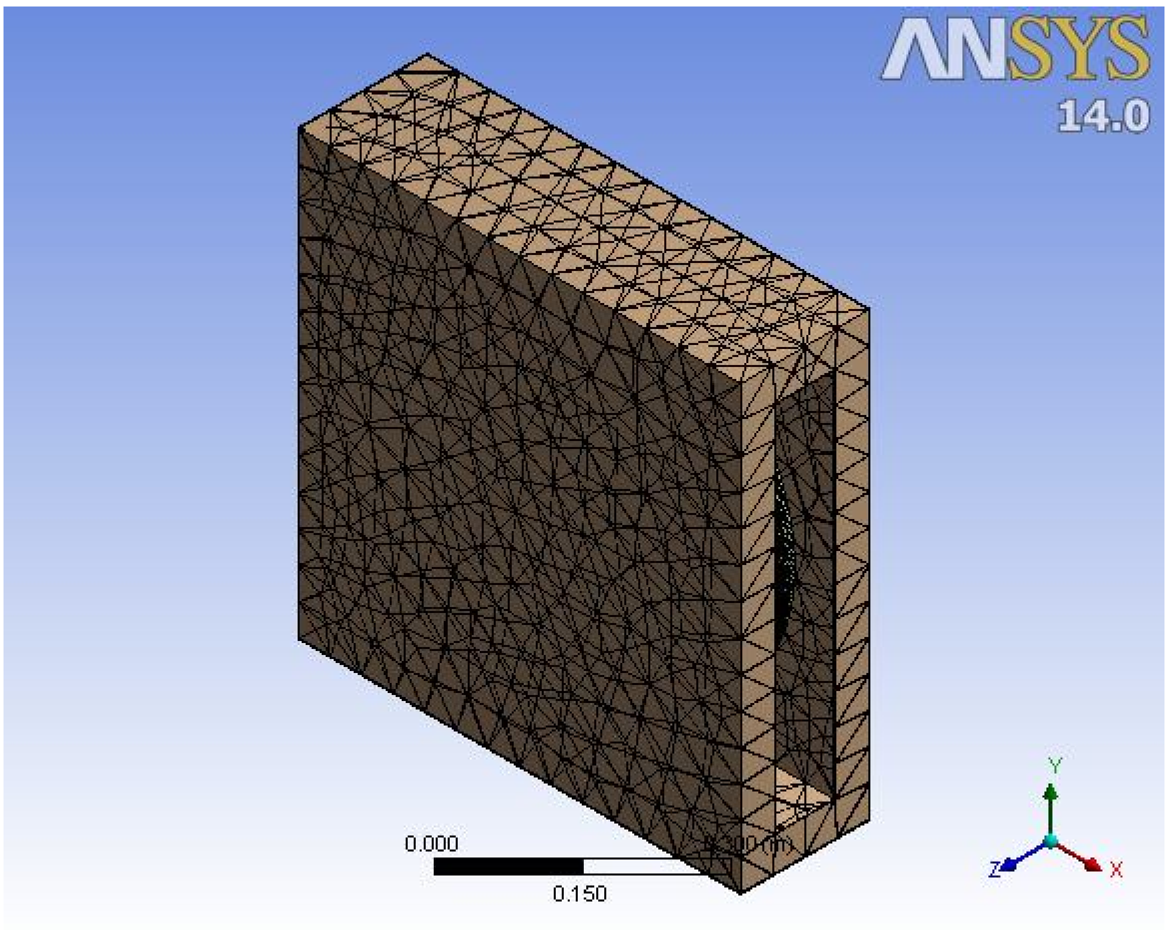


Figure3.6 Brake disc rotor after meshing

At the time of the braking process, a part of the frictional heat escapes into the ambient air by convection and radiation. Consequently, the determination of the heat transfer coefficients is essential.

Their exact calculation is rather difficult because these coefficients depend on the location and the construction of the braking system, the speed of the vehicle and consequently on the air circulation. Since the process of heat transfer by radiation is not too significant, we will determine using ANSYS CFX 14 code only the convection coefficient (*h*) of the disc. This parameter will be exploited to determine the three dimensional distribution of the temperature of the disc.
For reasons of symmetry of the disc, one can took only the quarter of the geometry in the case of ventilated disc and kept the tetrahedral form to generate the mesh of the discs. But in this paper the full ventilated and solid disc is used (Figs. 3.9 and 3.10).

3.9 Modeling in ANSYS CFX

For the preparation of the mesh of CFD model, one defines initially various surfaces of the disc in integrated computer engineering and manufacturing (ICEM) CFD as shown in Fig.3.6; what is used is a linear tetrahedral element with 67298 nodes and 34448 elements. In order not to weigh down calculation, an irregular mesh is used in which the mesh is broader where the gradients are weaker (non uniform mesh),

3.9.1 Mesh Report

Table3.5 Mesh information for CFX

Domain	Nodes	Elements
Brake disc zone	18741	89376
Fluid zone	1627	4823
All Domains	67298	34448

Table3.6 Mesh Information for Fluid Flow

Domain	Nodes	Elements
Brake disc zone	18741	89376
Fluid zone	1627	4823
All Domains	67298	34448

3.9.1Boundary conditions

Table3.7 Models BC

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	773k

Here are the procedure followed while modeling in **ANSYS CFX**

a) Physical model

In this step, one declares 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.

Table3.8 Domain definition

Domain - brake disc zone	
Type	Solid
Location	Brake disc
Settings	
Domain Motion	Rotating
Angular Velocity	6.7300e+01 [radian s^-1]
Axis Definition	Coordinate Axis
Rotation Axis	Z
Domain - fluid zone	
Type	Fluid
Location	fluid zone
Materials	
Air at 38 °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 materials

We introduce into the computer code the physical properties of used material. In this study we selected SiC/6061Al alloy composite material.

d) 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 disc.

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

Table3.9 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	2.3699e+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	2.9300e+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	7.7300e+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	4.2300e+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

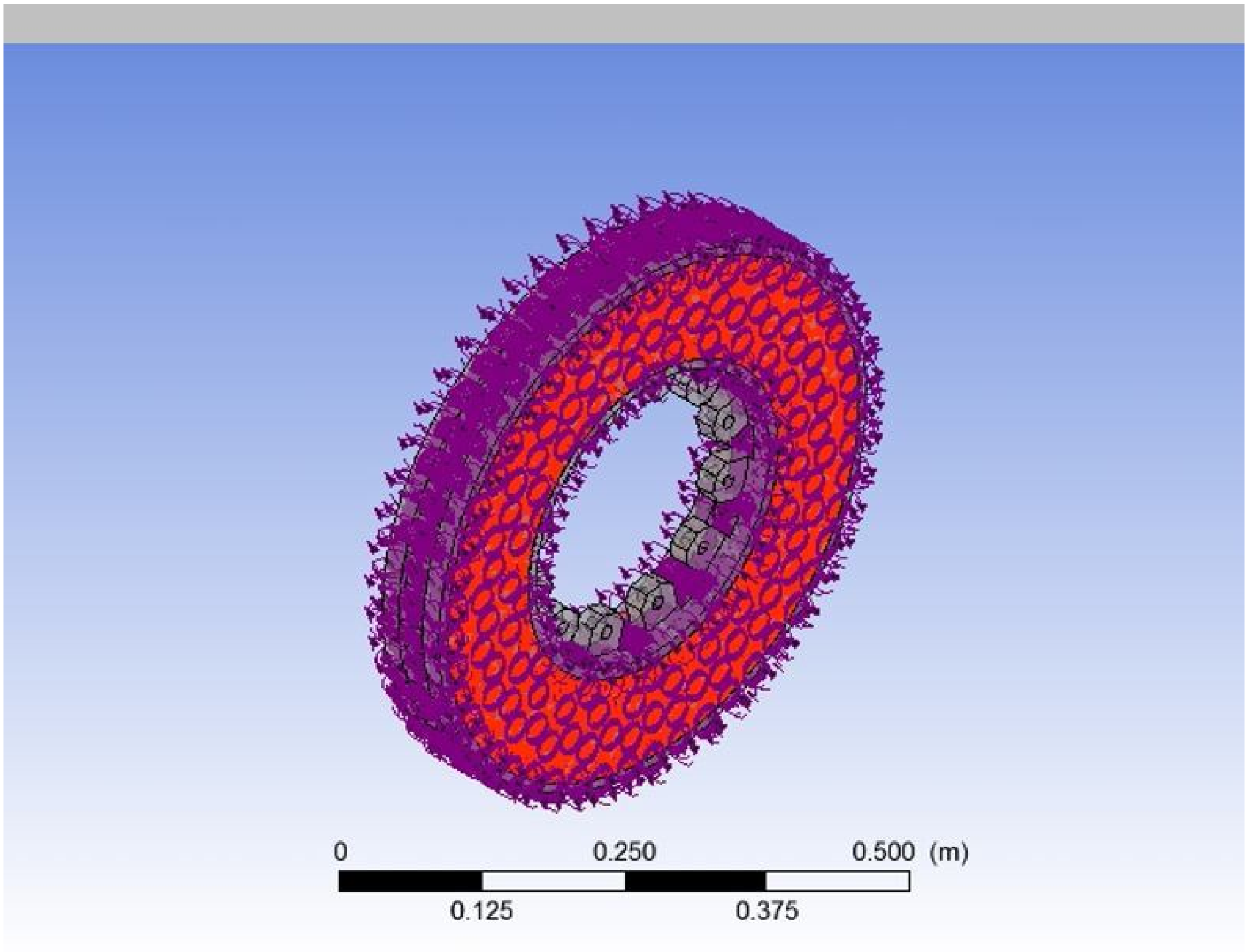


Figure3.7 Imported geometry of the brake disc after the application of the load

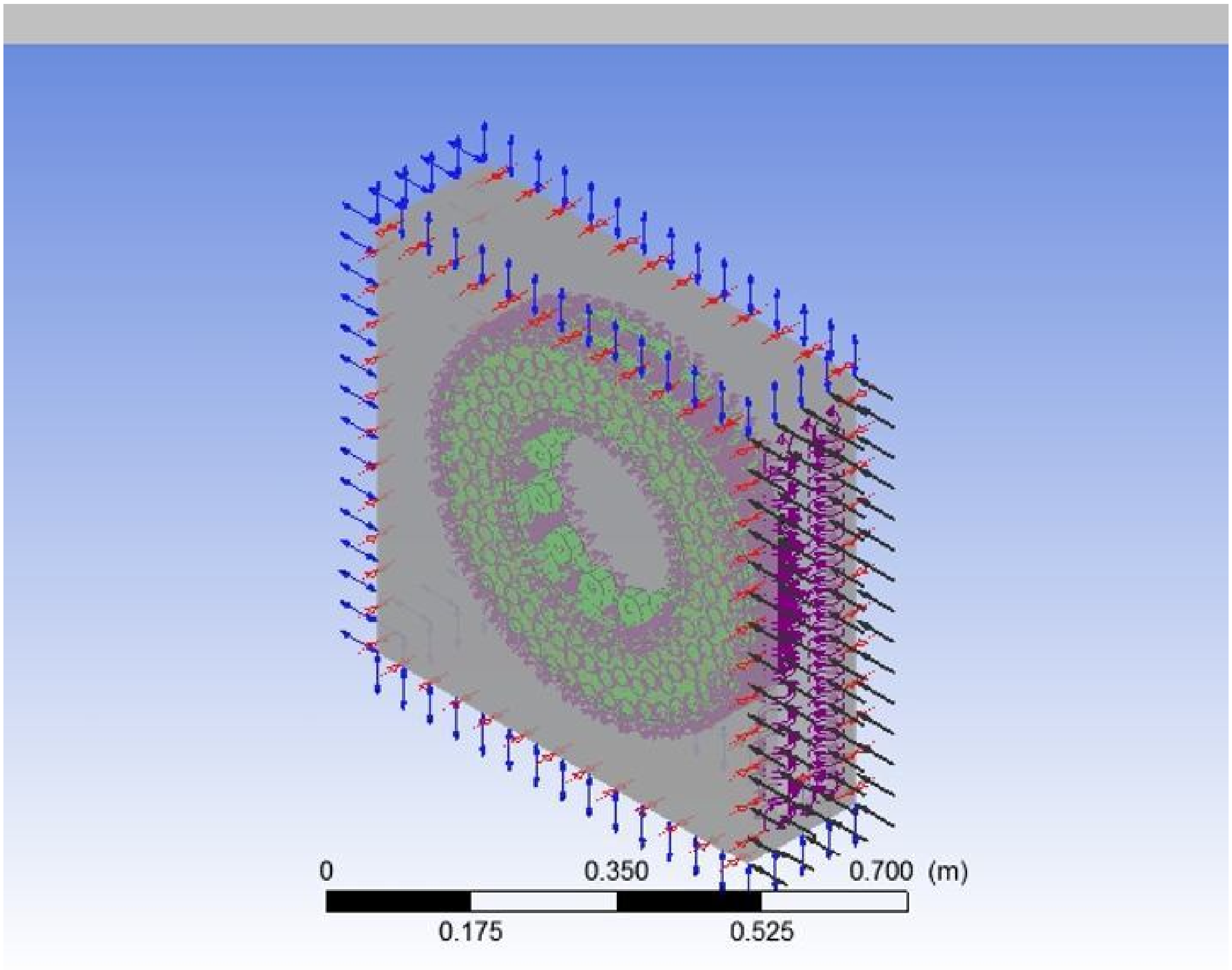


Figure3.8 Brake disc model after all the domain & BCs loaded

e) 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.

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 in Fig. 3.9 and Fig. 3.10.

f) Results of the simulation of the heat transfer coefficient (h)

The heat transfer coefficient is a parameter related to velocity of air and the shape of brake disc and many other factors. In different velocity of air, the heat transfer coefficient in different parts of brake disc changes with time [53]. Heat transfer coefficient will depend on air flow in the region of brake rotor and vehicle speed, but it does not depend on material. In this simulation, it is determined the average value of the coefficient h "Wall heat Transfer Coefficient", variable with time (Figs. 3.9&3.10).

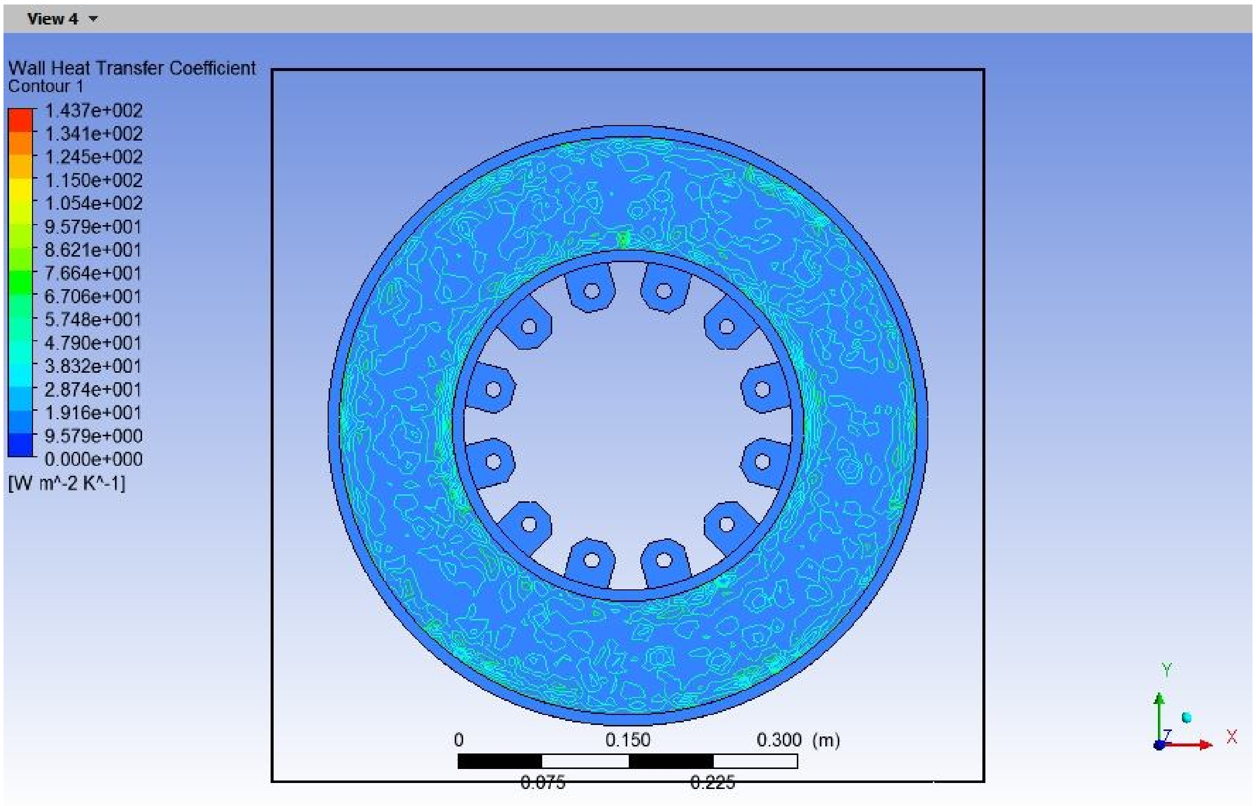


Figure3.9 Wall heat transfer coefficient of solid brake disc

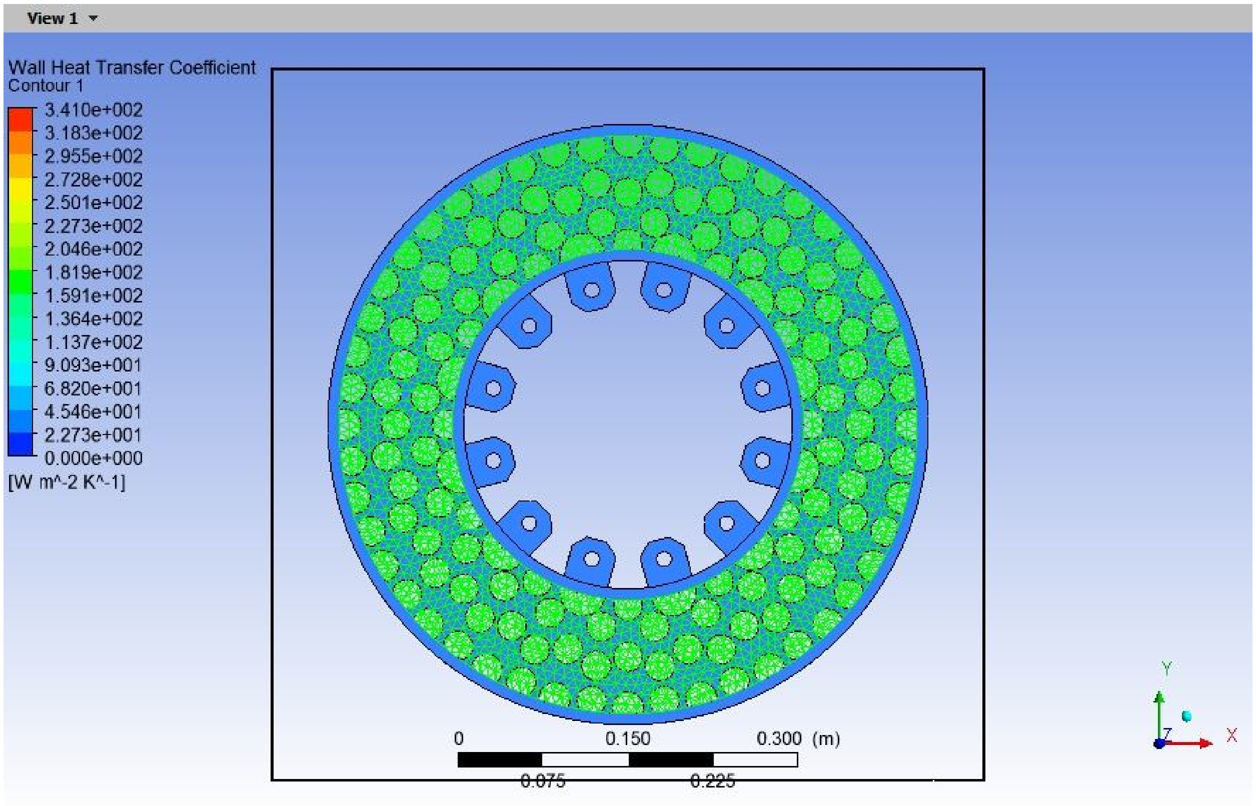


Figure3.10 Wall heat transfer coefficient of ventilated brake disc

The comparison between figs. 3.9 and 3.10 concerning the variation of heat transfer coefficient in the non stationary mode for the two types of design of the full and ventilated disc, it is noted that the introduction of the ventilation system influences directly the value of the convective heat transfer coefficient for same surface and is logically significant because this mode of ventilation intervenes in the reduction of wall-fluid temperature.

The airflow through and around the brake disc 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 they could be used as a boundary condition in thermal analysis [30].

CHAPTER FOUR

MODELING THERMO-MECHANICAL EFFECTS IN RAILWAY DISC BRAKE

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

The three dimensional axis-symmetrical model of transient heat transfer has been chosen in this research paper. It should be noticed that the geometry of a disc brake rotor is axis-symmetrical. In this case the assumption of axis-symmetric heat flux on the friction surface means only time averaged boundary conditions over a rotor revolution is applied.

4.1 Theoretical Approach to Railway Disc Braking Condition

In the process of braking of railway vehicles, it is necessary to define the model for thermal analysis that describes the heat transfer of the heat generation by friction at surfaces which are in contact between a railway disc and braking pads through the disc and pads as well as heat outflow of the whole braking system due to cooling effect 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 research [14].

4.1.1 Braking Time and Angular Velocity

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 [36].

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

$$v(t) = v_o - at \tag{4.1}$$

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

$$\boldsymbol{\omega(t) = \frac{v(t)}{r_w}} \tag{4.2}$$

$$\boldsymbol{\omega_o = \frac{v_o}{r_w}} \tag{4.3}$$

Where,

v = velocity of the locomotive

v_o = Initial running velocity of the locomotive

ω_o = Initial angular velocity the wheel

ω = angular velocity of the wheel at any time

r_w = the radius of locomotive 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 A.A.LRT as follows.

$$\boldsymbol{v_{disc} = \frac{r_{disc}}{r_w} v(t)} \tag{4.4}$$

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

$$\boldsymbol{S_b = v_o t_b - \frac{1}{2} a t_b^2} \tag{4.5}$$

$$\boldsymbol{t_b = \frac{V_0}{a}} \tag{4.6}$$

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

$$\boldsymbol{\omega = \omega_0 \left(1 - \frac{t}{t_b}\right)} \tag{4.7}$$

Where,

t_b = brake time to stop the locomotive

S_b = total brake distance

4.1.2 Braking Energy

During the process of braking, available energy of the A.A. LRT 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 A.A. LRT 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 [27].

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

$$\boldsymbol{E_k = \frac{1}{2}Mv_0^2 + \frac{1}{2}I\omega_0^2} \qquad \textbf{(4.8)}$$

Where,

E_k = Initial Kinetic Energy of the A.A. LRT just before braking starts

M = Mass of the A.A. LRT

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.

$$\boldsymbol{I = \sum_{i=1}^{n} m_i\, r_i^2} \qquad \textbf{(4.9)}$$

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 A.A. LRT. It is the parallel to the axis of the wheel axle. Hence, the rotating radius is equal to the radius of the brake disc ($\boldsymbol{r_d}$).Hence,

$$\boldsymbol{E_k = \frac{1}{2}Mv_0^2 + \frac{1}{2}I\omega_0^2 = \frac{1}{2}Mv_0^2\left[1 + \frac{I}{M{r_d}^2}\right] = \frac{1}{2}(1.1)Mv_0^2} \qquad \textbf{(4.10)}$$

In Some literatures the contribution of the rotating masses of the wheel set and discs is taken to be 10% the tare weight of the axle load of the vehicle [38]. The term $\left[\frac{I}{Mr_d{}^2}\right]$ accounts to the rotational masses involved and its value equals 0.1 and it is adopted here.

$$\boldsymbol{E_k = \frac{1}{2}[0.1](1.1)Mv_0^2} \qquad \textbf{(4.11)}$$

The potential energy of the A.A. LRT depends on the track gradient δ [mm/m] and on the travelled distance S_b [m]. The exact definition of gradient is $i=tan\ \alpha$ where α is angle of inclination. According to EN 14531- 6 [38] for calculation of external forces that result from gradients in railway applications, the Simplification $sin\ \alpha = tan\ \alpha$ is commonly used. Therefore the potential energy is expressed as:

$$\boldsymbol{E_p = M.\mathrm{g}.S_b\frac{\delta}{1000}} \qquad \textbf{(4.12)}$$

Where, g = 9.81m/s^2

Therefore, the total available braking energy of the A.A. LRT is determined from the equation below:

$$\boldsymbol{E_b = \frac{1}{2}MV_0^2[1.1] + M.\mathrm{g}.S_b\frac{\delta}{1000}} \qquad \textbf{(4.13)}$$

4.1.3 Heat Energy

In braking system, the mechanical energy is transformed into heat energy. This energy is characterized by a total heating of the disc and the pads during the braking phase. The energy dissipated in the form of heat can generate rise in temperature ranging from 250^{o}C to 800^{o}C [26]. The heat quantity in contact area is the result of plastic micro deformation generated by the friction forces between disc rotor and brake pad.

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

$$\boldsymbol{Q_{gen} = E_b} \qquad \textbf{(4.14)}$$

4.1.4 Work of Friction Force

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

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

The total heat flux is equal to:

$$\textbf{Total heat flux} = \frac{\dot{Q}_{gen}}{2A_b} = \frac{E_b}{2A_b} \qquad \textbf{(4.15)}$$

Where,

A_b = *is* the area swept by the brake pad at one face of the disc

S_{disc} = distance traveled by F_{disc} during the whole braking time

4.1.5 Load Application

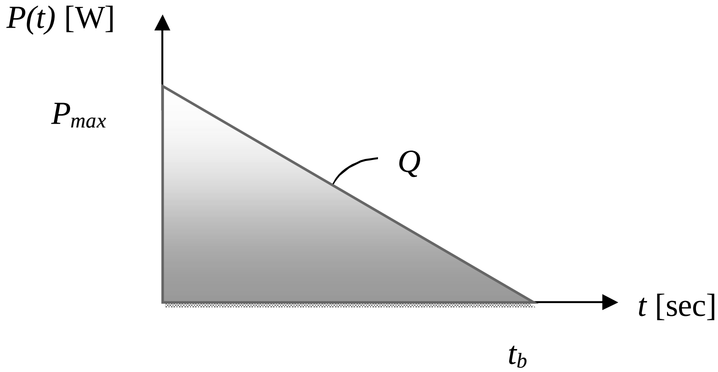
a) Centrifugal Load at Constant Temperature of the Surrounding Air

The figure below represents the application of the centrifugal load on the A.A. LRT vehicle when travelling on a straight track (ROAD AUTHORITY to MEXICO SQUARE, EW18 to EW19, and 554m of distance).

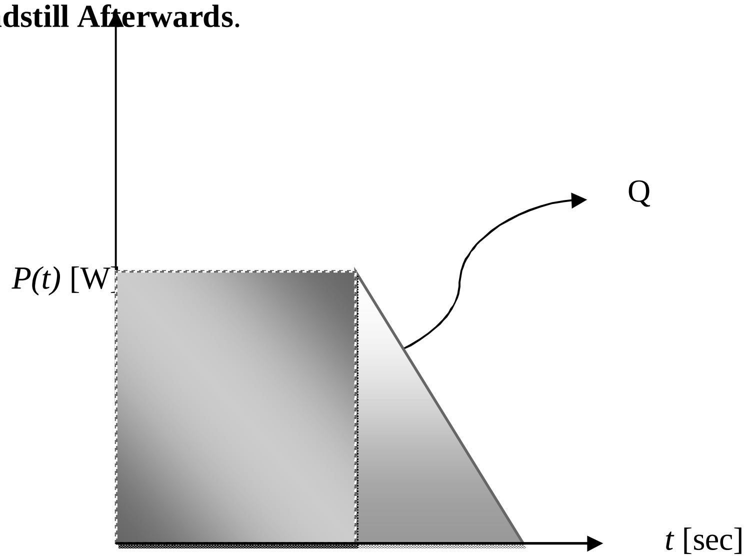
a)

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

The figure below represents the application of the braking force on the A.A. LRT vehicle from maximum velocity $(80km/hr)$ to standstill when travelling on a straight track (ROAD AUTHORITY to MEXICO SQUARE, EW18 to EW19 and 554m of distance). After the application of the brake, the vehicle comes to stop after 176meters and takes 16 seconds.



c) **Braking on a Downhill Track for Maintaining Constant Velocity and Braking to Standstill Afterwards**.



c)

The figure above represents the application of the braking force on the A.A. LRT vehicle to maintain a constant velocity and it covers 484metrs for 22 seconds and braking from maximum velocity to standstill for 16 seconds when travelling on a downhill track from LIDETA TO COCA COLA MAZORIA, EW20 to EW21 and 848m of distance). After the application of the brake, the vehicle comes to stop after 660meters and takes 38 seconds.

Figure4. 1 a) Centrifugal load, b) Heat input during braking to standstill from max velocity and c) Heat input to maintain a constant velocity and braking to standstill afterward

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

$$\boldsymbol{Q_{gen} = E_b = \int_0^{t_b} P(t)dt} \qquad \textbf{(4.16)}$$

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

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

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

The friction force that work on the disc to retard the A.A. LRT is:

$$\boldsymbol{F_{disc} = \frac{\frac{1}{2}MV_0^2[1.1]+M.g.S_b\frac{\delta}{1000}}{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b-\frac{1}{2}at_b^2\right]}} \qquad \textbf{(4.17)}$$

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

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

$$\boldsymbol{\dot{Q}(t) = (2F_{disc})v_{disc} = 2F_{disc}\frac{r_{disc}}{r_{wheel}}(v_o - at)} \qquad \textbf{(4.18)}$$

4.1.6 Heat Flux Entering the Disc

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.

Generally, the thermal conductivity of material of the brake pads is smaller than that of disc ($k_p < k_d$). We consider the energy from brake application is converted into heat and transferred to the disc and pad approximately 95% and 5% respectively [25]. This ratio normally is called the proportion of heat transferred to disc ($\gamma = 0.95$). The rate of heat generation is:

$$\dot{Q}_d(t) = \gamma(2F_{disc})v_{disc} = (0.95)(2F_d)\frac{r_d}{r_w}(v_o - at) \quad \textbf{(4.19)}$$

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

$$\rho C_p \dot{T} = \frac{1}{r}\frac{\partial}{\partial r}\left(rk_r\frac{\partial T}{\partial r}\right) + \frac{\partial}{\partial z}\left(k_z\frac{\partial T}{\partial Z}\right) \quad \textbf{(4.20)}$$

Where,

ρ -the density for material of the disc

K-represents the thermal conductivity,

C_p- the specific heat capacity,

At the contact surface between the railway disc and the braking pads, the braking process produces heat according to expression (4.18).The heat dissipation from the free surface of the disc to the surrounding air is described by both convection and radiation:

$$Q_{diss} = -h(T - T_{ref}) - \varepsilon\sigma(T^4 - T_{ref}^4) \quad \textbf{(4.21)}$$

In this equation,

h -equals the convective film coefficient,

ε -the material's emissivity,

σ- Stefan Boltzmann constant and

T_{ref}- 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 [39].

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

The value of convective heat transfer coefficient (h) had already been analyzed and simulated in chapter three and the result of the simulation for heat transfer coefficient is used as an input for the thermal analysis.

4.2 Calculation of Railway Disc Brake

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

4.2.1 Determination of the Load Conditions

The specific loading condition under this analysis is braking from max velocity of 80km/hr to standstill on a flat track (ROAD AUTHORITY to MEXICO SQUARE, EW18 to EW19, 554m of distance) and braking on a downhill track to maintain a constant velocity of 80km/hr and braking to standstill afterwards (from LIDETA TO COCA COLA MAZORIA, EW20 to EW21, 848m of distance) with the temperature of the surrounding air is 38°C. The assumed temperature is average environmental temperature of the operating route of the A.A.LRT in the region.

In case of braking on a horizontal track to a standstill, the analysis was carried out in 16 steps, each step being 1 s long.

In case of braking on a downhill track, a physical model was used to determine the heat flux in dependence of the changing potential energy. The vehicle descended on a downhill track and maintained a constant speed of 80km/h. Consequently, the heat flux entering into the disc was constant. In case of braking on a downhill track and maintaining a constant speed, the analysis was carried out in 22 steps, each step being 1 s long.

Additional 16 steps were added to simulate braking to standstill after maintaining a constant velocity on track. Altogether this case included 38 steps, each step being 1 s long.

In this loading condition, the brake cycle dealt is with no recent brake application or previous warmed up of the disc had already been fully dissipated to the ambient.

Hence, the initial temperature of the disc and the surrounding is the same (38°C).The goal is to find out how the temperature and stress are distributed on the whole construction of the disc by braking on an inclined track to a standstill and braking on a straight track. The deceleration factor is 1.4 m/s^2. The heat flux uses the convection with a heat transfer coefficient of the previous result simulated in chapter three.

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

Table4.1 Addis Ababa LRT E-W rout stations

No	Stations	Gradient	Sin δ
1	EW1	0.435	0.0076
2	EW2	0.894	0.0156
3	EW3	-0.286	-0.005
4	EW4	-1.861	-0.0325
5	EW5	-0.6588	-0.0115
6	EW6	-0.0974	-0.0017
7	EW7	0	0
8	EW8	-0.0859	-0.0015
9	EW9	-0.802	-0.014
10	EW10	-0.544	-0.0095
11	EW11	-0.0593	-0.001
12	EW12	1.528	0.0265
13	EW13	0.573	0.01
14	EW14	1.088	0.019
15	EW15	0.802	0.014
16	EW16	-0.172	-0.003
17	EW17	-0.0859	-0.0015
18	EW18	0.0859	0.0015
19	EW19	1.547	0.027
20	EW20	-1.547	-0.047
21	EW21	1.747	0.0325
22	EW22	0.17	0.003

Table4.2 Addis Ababa LRT E-W distance between consecutive stations

No	Addis Ababa LRT E-W rout Station name	Addis Ababa LRT E-W rout stations symbol	Station distance
1	Ayat 2-------------- Ayat 1	EW1--------------------EW2	1.30km
2	Ayat 1--------------Meri/CMC 2	EW2--------------------EW3	1.034km
3	Meri/CMC 2-------CMC 1	EW3--------------------EW4	0.896km
4	CMC 1------------- St.Michael Church	EW4------------------- EW5	0.865km
5	St.Michael Church---- Civil Service College	EW5------------------- EW6	0.785km
6	Civil Service College—Sahlite Mhiret Church	EW6--------------------EW7	0.906km
7	Sahlite Mhiret Church-----Gurd Sholla	EW7--------------------EW8	0.922km
8	Gurd Sholla---------Megenagna/Adwa square	EW8--------------------EW9	1.004km
9	Megenagna/Adwa square--Lem Hotel	EW9--------------------EW10	1.035km
10	Lem Hotel--------Mazoria/Traffic Police HQ.	EW10------------------EW11	0.838km
11	Mazoria/Traffic Police HQ.------- Chemical Corporation	EW11------------------ EW12	0.694km
12	Chemical Corporation---Urael church	EW12-------------------EW13	0.875km
13	Urael Church------ Yordanose Hotel	EW13------------------ EW14	0.570km
14	Yordanose Hotel---Estifanose/Meskel Square2	EW14------------------ EW15	0.569km
15	Estifanose/Meskel Square2---- Stadium/Meskel Square 1	EW15------------------ EW16	0.700km
16	Stadium/Meskel Square 1----La Gare	EW16------------------ EW17	0.865km
17	La Gare----------------Road Authority	EW17-------------------EW18	0.555km
18	Road Authority--------Mexico Square	EW18-------------------EW19	0.554km
19	Mexico Square-------------------Lideta	EW19-------------------EW20	0.936km
20	20 Lideta-------------------------Coca Cola	EW20-------------------EW21	0.848km
21	21 Coca Cola------------------Tor Hailoch	EW21-------------------EW22	1.149km

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

This shows that the heat flux entering the wheel when the train is moving up the gradient is less that of moving on a straight track and down the gradient. The operating line has different value of gradient so that at each and every station the value of heat energy is different. The different station designation and its gradient are given in the table above.

4.2.2 Weight Transfer during Braking

When A.A. LRT is standing still on straight track, its weight is uniformly distributed on all axles. 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. On the front bogie there are five discs located on the two axle and one disc could able to take only 12% (0.6/5*100%) of the whole weight of the vehicle.

4.2.3 Determination of the Physical Model

LRT train stop braking on track derives from the physical model for determination of the heat transfer in dependency from the braking time.

The weight distribution of the LRT train considered is distributed between the front and rear bogies. Each bogie consists of two wheel sets.

Hence, only 12 % of the whole brake force is applied to one disc from the forward part of the carriage.

The braking energy for one wheel considering constant deceleration is:

$$\mathbf{0.12}\left[\frac{1}{2}Mv_0^2[1.1] + M.g.S_b\frac{\delta}{1000}\right] = (2F_{disc})\int_0^{t_b} v_{disc}\,dt \qquad \textbf{(4.22)}$$

This change of energy is equal to the heat flux on the surface of the disc. This ratio is used to calculate the thermal load on the brake disc. Other operational data used for the analysis are listed in table 3.1. The material property of SiC/6061Al alloy composite of brake disc with good thermo-physical and thermo-elastic characteristics of which adopted in this simulation of the rotor are recapitulated in Table4.3

Table4.3 Material properties of brake disc

Material Properties	**disc**
Thermal conductivity, *k (W/m k)*	5
Density, *ρ (kg/m3)*	3700
Mass of the brake disc, *(Kg)*	33.038
Specific heat, *C (J/Kg k)*	650
Poisson's ratio, v	0.13
Thermal expansion, *α (10-6 / K)*	0.15
Elastic modulus, *E (GPa)*	254
Allowed temperature, *(℃)*	600
Tensile yield strength, *(Mpa)*	314
Ultimate tensile strength, *(Mpa)*	410

Table4.4 Material properties of brake pad

Material Properties	**pad**
Thermal conductivity, *k (W/m °C)*	5
Density, *ρ (kg/m3)*	1800
Specific heat, *C (J/Kg °C)*	880
Poisson's ratio, v	0.3
Thermal expansion, *α (10-6 / K)*	0.3
Elastic modulus, *E (GPa)*	50.2

The coefficient of friction has significant effects on the form of the thermal field as well as on temperature distribution. The proper definition of the coefficient of friction is very important for perception of temperature field qualities for analyzing thermal effects in braking systems at railway discs. The value of the coefficient of friction depends on a number of tribological parameters such as speed of relative movement of parts in contact, the contact surface roughness, quality of materials of a tribo-pair, temperature of contact parts and so on [47].

However, this research did not consider tribological parameters of braking processes. According to DIN EN ISO1183 the value μ = 0.32 of the friction coefficient for materials of the disc and the pad for A.A. LRT is taken as the recommended value.

4.2.4 Determination of Friction Surface of Disc

According to UIC 541-3 Standard, a Carbon/carbon (C/C) composites brake pad and sic/6061Al alloy composite of brake disc with dimension tabulated in table 4.5 is used to calculate friction area swept on the disc by the brake pad.

Table4.5 Input parameters

Description	**Value**
Inner disc diameter ,*mm*	220
Outer disc diameter ,*mm*	460
Disc thickness ,*mm*	60
Effective disc radius, *mm*	170
Rate distribution of the braking forces ϕ, %	10
Factor of charge distribution of the disc ε_p	0.5
Track inclination δ, $^0/_{00}$	11
Surface disc swept by the pad A_c, mm^2,	128890
Wheel diameter, *mm*	660
Deceleration a, *m/s2*	1.4
Initial speed *V*o, *km/h*	80
Vehicle mass *m*, *kg*	64000

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

4.2.5 Determination of Heat Flux

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

4.2.5.1Braking on a Flat Track

$$F_{disc} = \frac{0.12 * \frac{1}{2} M V_0^2 [1.1]}{2 \frac{r_{disc}}{r_{wheel}} \left[v_o t_b - \frac{1}{2} a t_b^2 \right]} = 6611.17\, N$$

The instant heat flux entering brake disc:

$$\frac{\dot{Q}(t)}{2A_b} = \frac{\gamma(2F_{disc}) v_{disc}}{2A_b} = \frac{(0.95)(2F_d)\frac{r_d}{r_w}(v_o - at)}{2A_b} = 484896 - 42428.4t\ [{}^{W}/_{m^2}]$$

4.2.5.2Braking on a Downhill Track

a) Braking on downhill track for maintaining constant velocity

$$Q_{disc} = \frac{0.95 * M.\mathrm{g}.\mathrm{v_{osin\delta}}}{12} = 26389.894W$$

$$F_{disc} = \frac{Q(t).\mu.\mathrm{r_d}}{2V_{O}.r_w} = 118.0466N$$

The heat flux entering brake disc:

$$\frac{\dot{Q}(t)}{2A_b} = \frac{\gamma Q(t)}{2A_b} = 97368.32[W/m^2]$$

b) Braking to stand still after maintaining constant velocity

$$\boldsymbol{F_{disc}} = \frac{\mathbf{0.12\{\frac{1}{2}MV_0^2[1.1] + M.g.v_{osin\delta}\}}}{\mathbf{2\frac{r_{disc}}{r_{wheel}}\left[v_o t_b - \frac{1}{2}at_b^2\right]}} = \mathbf{6731.76[N]}$$

The Instant heat flux entering brake disc:

$$\frac{\dot{\boldsymbol{Q}}(\boldsymbol{t})}{\boldsymbol{2A_b}} = \frac{\boldsymbol{\gamma(2F_{disc})v_{disc}}}{\boldsymbol{2A_b}} = \frac{\boldsymbol{(0.95)(2F_d)\frac{r_d}{r_w}(v_o - at)}}{\boldsymbol{2A_b}} = \boldsymbol{493740.3 - 43202.278t\ [W/m^2]}$$

4.2.6 Force 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 calculated for each braking conditions.

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

$$p = \frac{F_{disc}}{\mu A_p} = \frac{9520.085}{0.32 * 0.12874} = \mathbf{160477.7[\boldsymbol{Pa}]}$$

b) Braking on a downhill track for maintaining constant velocity

$$p = \frac{F_{disc}}{\mu A_p} = \frac{118.0466}{0.32 * 0.12874} = \mathbf{2865[\boldsymbol{Pa}]}$$

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

$$p = \frac{F_{disc}}{\mu A_p} = \frac{9693.7}{0.32 * 0.12874} = \mathbf{163405[\boldsymbol{Pa}]}$$

4.2.7 Centrifugal Load Determination

$$\boldsymbol{F_c = m_d\, \omega^2\, R}$$

Where; F_c – centrifugal force

m-mass of the brake disc

ω - Angular velocity

R-radius of the brake disc

The angular velocity at any time t is:

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

$$\omega = (67.33 - 4.208t)^2$$

$$m = 33.038kg$$

$$R = 110mm \ or \ 0.11m$$

Hence, $\boldsymbol{F_c = 3.6342(67.33 - 4.208t)^2}$

$$\boldsymbol{F_{c\ at\ t=0} = 16475N}$$

All the above calculated values are used throughout the finite element analysis.

CHAPTER FIVE

FEA ANALYSIS OF A.A.LRT DISC BRAKE USING ANSYS

Most numerical approaches on disc brakes performs a brake time or deceleration rate, in practice the caliper press the pad onto the disc by pneumatic cylinders and the constant deceleration rate can be maintained.

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

This study will use the FEA software ANSYS14 version to carry out transient thermal analysis and the coupled thermo-mechanical stresses of a railway disc under different braking conditions.

5.1 Introduction to Finite Element Analysis

Finite Element Method is a numerical procedure for solving continuum mechanics of problem with accuracy acceptable to engineers. Finite element method is a mathematical modeling tool involving discretization of a continuous domain using building block entities called finite elements connected to each other by nodes for once and moment transfer. This process includes finite element modeling and finite element analysis [5].

In displacement based FEM, stiffness of the entire structure (part or assembly) is assembled from stiffness of individual elements. Loads and boundary conditions are applied at the nodes and the resulting sets of the simultaneous equations are solved using matrix methods and numerical techniques. In short, FEM is a numerical method to solve ordinary differential equations of equilibrium. Starting with simple linear static stress and heat transfer analysis to complex simulations involving highly non-linear, fluid flow and dynamic events can be successfully analyzed on a personal computer using a host of popular software like ANSYS. In practice a finite element analysis usually consists of three principal steps.

Preprocessing: Create and discretize the solution domain into finite elements. This involves dividing the domain into sub-domains, called 'elements', and selecting points called nodes on the inter-element boundaries or in the interior of the elements. Assume a function to represent the behavior of the element, this function is approximate and continuous and is called the "shape function". Develop equations for an element. Assemble the elements to represent the complete problem. Apply boundary conditions, initial conditions and the loading.

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

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

5.2 Introduction to 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 and 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 [17].

5.3 General Analysis Procedure of ANSYS

Similar to solving problems analytically, it needs to define: the solution domain, the physical model, the boundary conditions and the physical properties of the material used for the model. Then solving the problem and present the results. In numerical methods, the main difference is an extra step called mesh generation is performed. This is the step that 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 thermal and mechanical analysis of brake disc model using ANSYS software.

a) Geometrical Definition:

There are four different geometric entities in pre processor 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.

b) 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 to establish controls over the size and desired shape of the elements and then instruct ANSYS program to generate all the nodes and elements automatically.
On the contrary, the direct generation method determines 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 book keeping 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.

c) 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.

d) Generate Mesh

In the finite element analysis the basic concept is to analyze the structure which is an assembly 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.

e) Finite element generation

The maximum amount of time in a finite element analysis is spent on generating elements and nodal data. Pre processor 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 pre processor 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 pre processor 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. Also one can plot, delete or search nodes.

f) Boundary conditions and loading

After completion of the finite element model, it has to be constrained 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 Ids. This helps the user to keep track of load cases.

g) 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.

h) **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 art 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.

5.4 Analysis of A.A.LRT Disc Brake using ANSYS 14

The analysis is carried out for three dimensional ventilated brake disc (Fig.5. 1) (the disc is 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 Eq. (4.17).Transient thermal and mechanical analysis is carried out both for braking from maximum velocity to standstill on a straight track and braking in a downhill track to maintain a constant velocity and braking to standstill afterwards. The numerical modeling of thermal effects was performed according to the following assumptions in ANSYS 14 workbench module.

5.4.1Assumptions

- 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.

5.4.2 Geometrical Model

In the three dimensional model of ventilated disc single surface of its symmetry in axial direction is insulated owing nature of considered phenomenon of heating. The disc is screwed on to the hub. It has twelve holes to fix the disc with the hub with sixteen diameter bolt. The assembled geometry of the disc is illustrated in Fig 5.1 below.

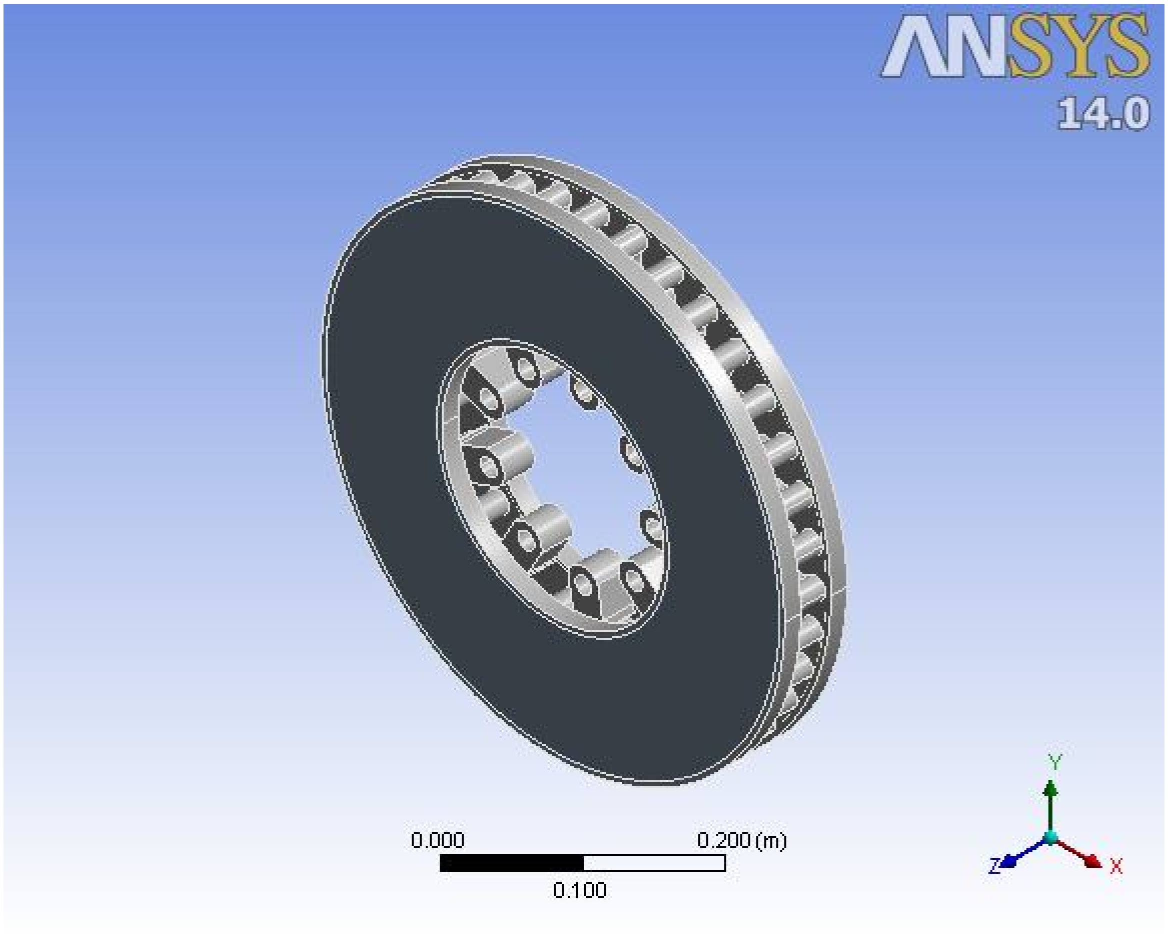


figure5.1 Geometric models of the disc

5.4.3 Definition of Material

The material property of SiC/6061Al alloy composite disc material 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 5.1below.

Table5.1 Material properties of SiC/6061Al alloy composite disc

Material Properties	**SiC/6061Al alloy composite Disc**
Thermal conductivity - *k [W/m k]*	5
Density – ρ *[kg/m^3]*	3700
Specific heat - *c [J/Kg k]*	650
Poisson's ratio, *v*	0.13
Elastic modulus - *E [G Pa]*	2.54×10^{11}

5.4.4 Meshing

In the finite element analysis, the basic concept is to analyze the structure which is an assembly 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, tetrahedron patch conforming mesh generating capabilities were used in ANSYS rather than defining the nodes individually (fig. 5.2). The finite elements used for the meshing were triangular. The number of generated elements in the mesh is 47007 with 90918 nodes.

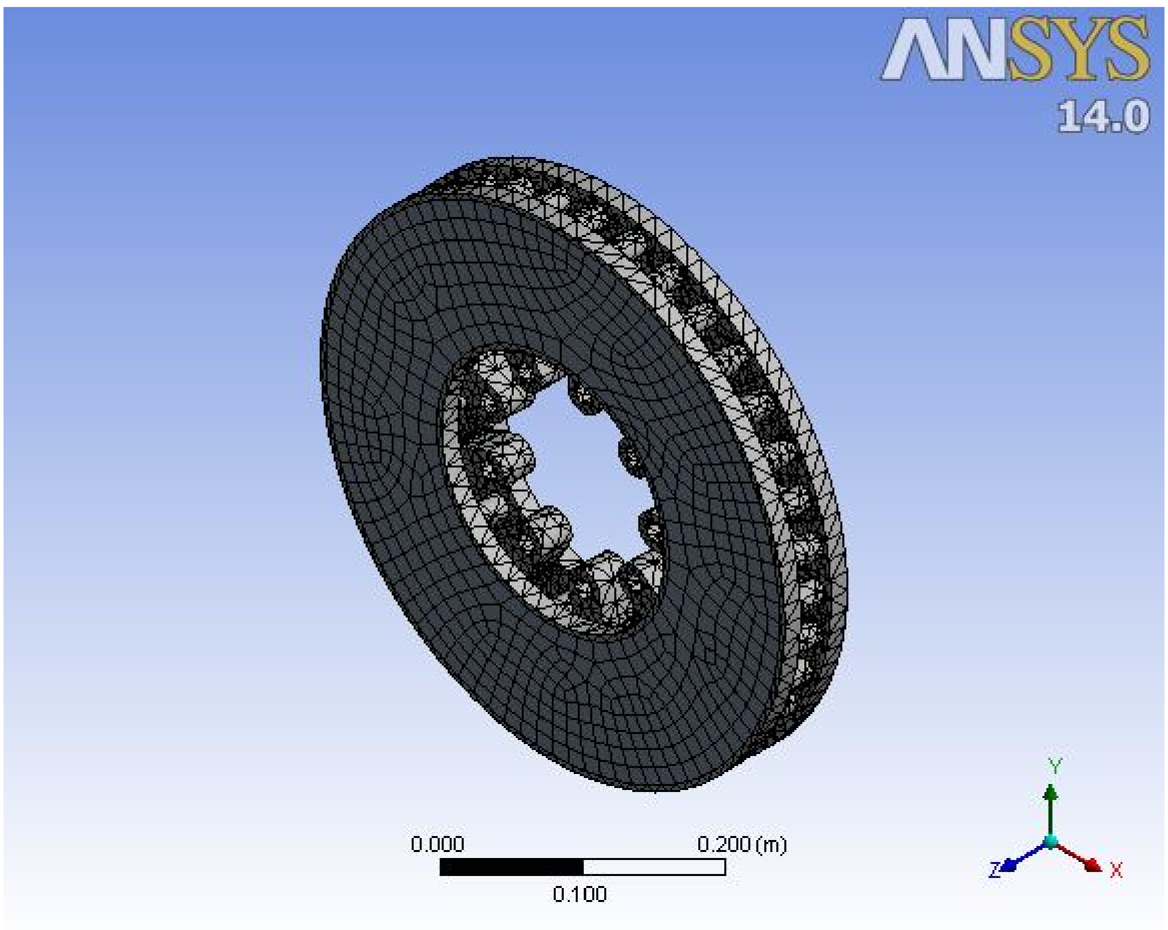


figure5.2 Meshed Model of Disc

5.4.5 Definition of the Loads and Boundary Conditions

After completion of the finite element model, it is necessary to apply constraints and loads to the model. The A.A. LRT analyzed railway disc brake is subjected to the following loads.

- The initial temperature of the disc and the pads is 38 °C.
- The surface convection condition is applied at all surfaces of the disc except the friction contact surface area of the disc and the pad with the values of the convection coefficient (h) obtained from ANSYS CFX simulation result obtained in chapter three.
- The heat flux into the brake disc during braking can be calculated before by the formula described in equation 4.18.
  a) on a straight track is: ***484896 – 42428.4t***(W/m^2)
  b) on a downhill track for maintaining constant velocity is: ***97368.32***(W/m^2)
  c) on a downhill track braking to standstill after maintaining constant velocity is: ***493740.3 – 43202.27t***(W/m^2)
- The pressure on the rubbing contact surfaces of the disc and brake pad is applied for three cases of brake applications.
  a) Pressure application for braking on a straight track has a value of ***160477.75Pa.***
  b) Pressure application for braking on a downhill track for maintaining a constant velocity has a value of ***12865Pa.***
  c) Pressure application for braking on a downhill track to a standstill after maintaining a constant velocity has a value of ***163405pa.***
- Angular velocity of the disc rotor is given ***67.33***rad/sec.
- Translational acceleration of the rotor disc is ***1.4***m/s^2.
- Fixed support is applied at the holes where the bolts are located and cylindrical support at the inner surface of the hub.

5.4.6 Thermal Load

a) **Thermal load application for braking from maximum velocity to standstill on a straight track.**

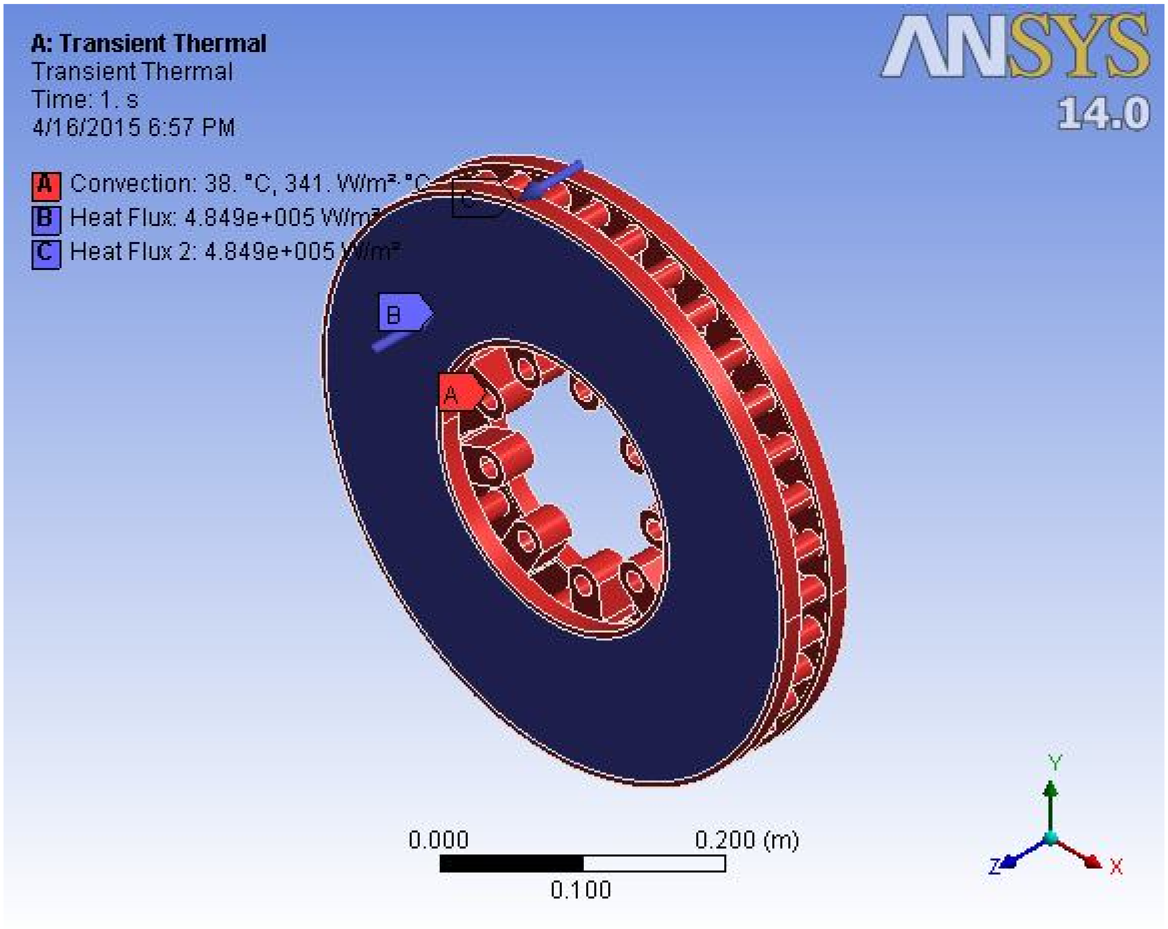


Figure5.3a Thermal loading of flux and convection coefficient for straight track

b) Thermal load application for braking on a downhill track to maintain a constant velocity.

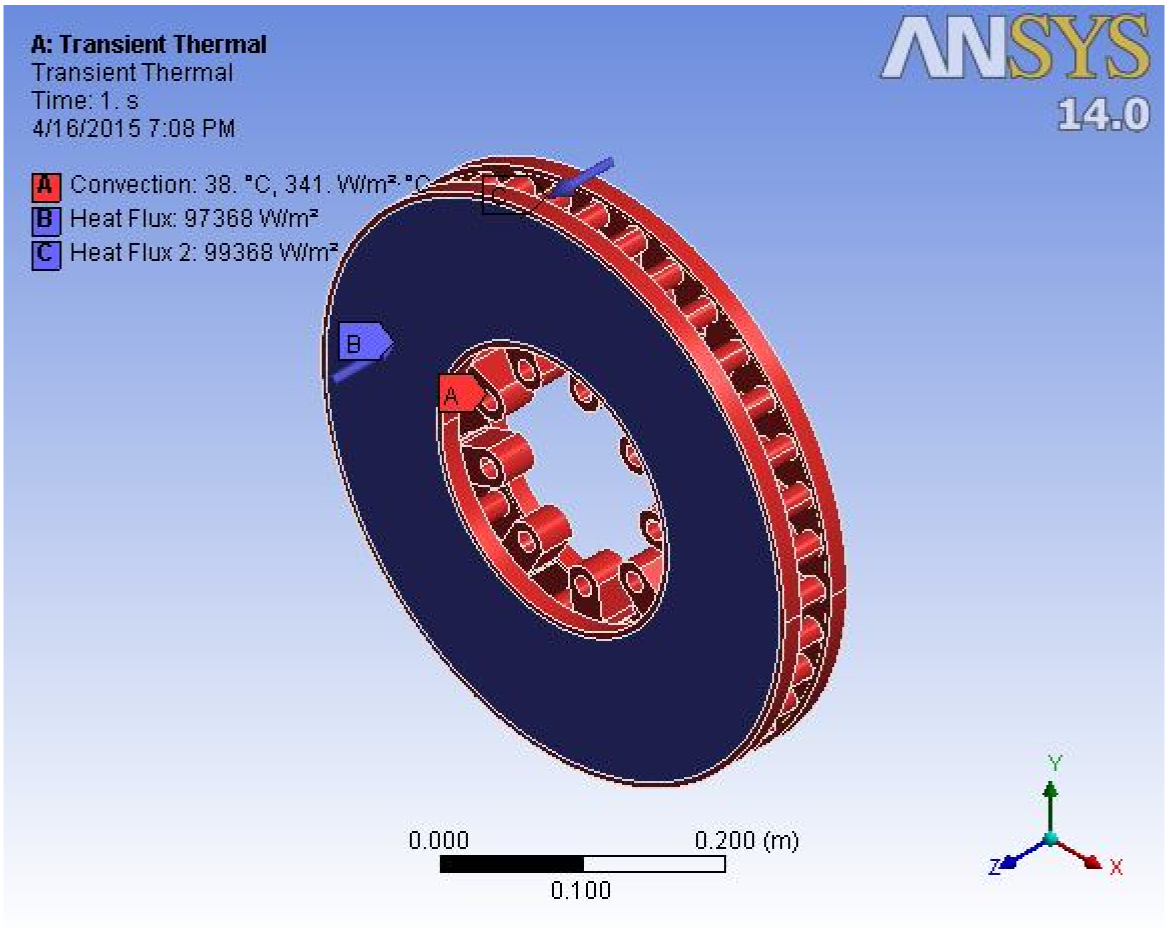


Figure5.3b Thermal loading of flux and convection coefficient for downhill track

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

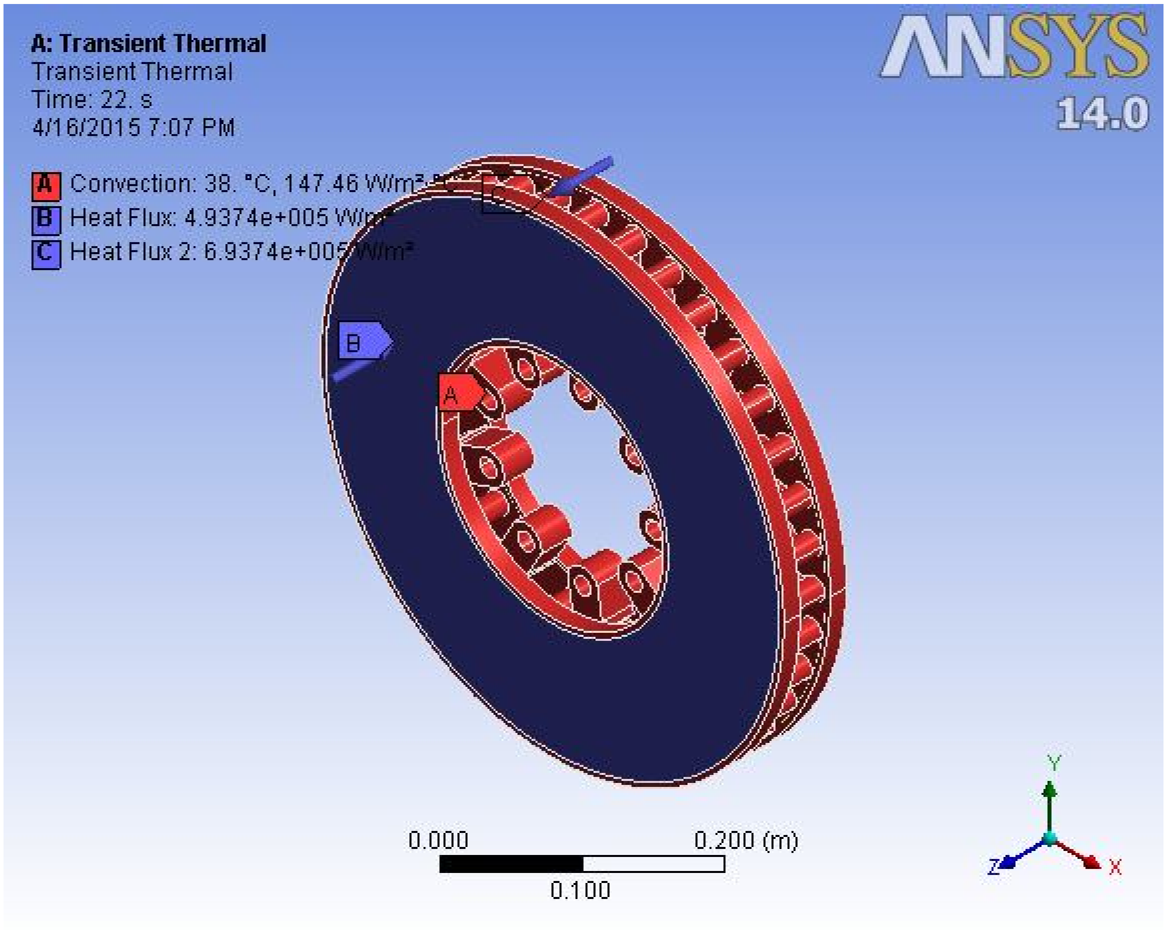


Figure5.3c Thermal loading of flux and convection coefficient for downhill track

5.4.7 Mechanical Load

a) **Pressure application for braking on a straight track has a value of 160477.75pa.**

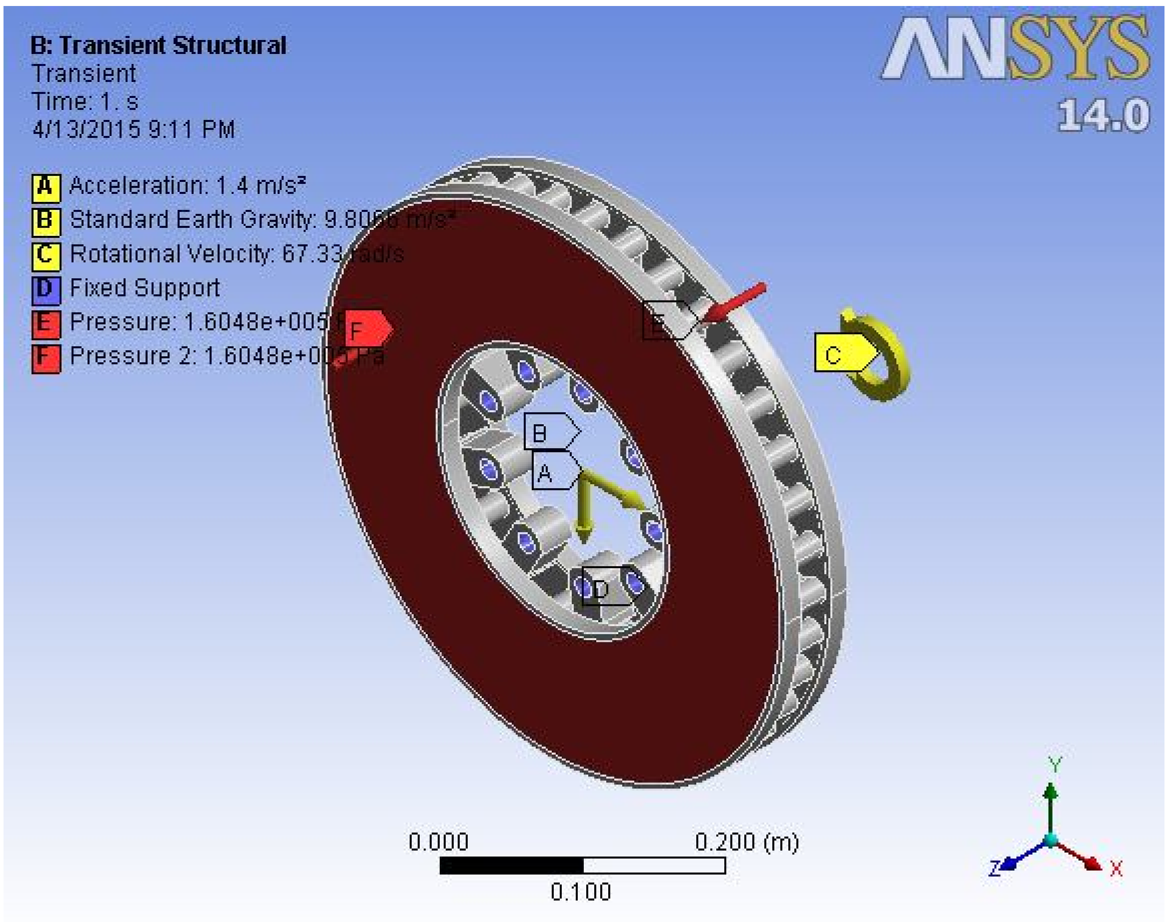


Figure5.4a Pressures acting on the rubbing surface of the disc for straight track

b) Pressure application for braking on a downhill track for maintaining a constant velocity has a value of 12865pa.

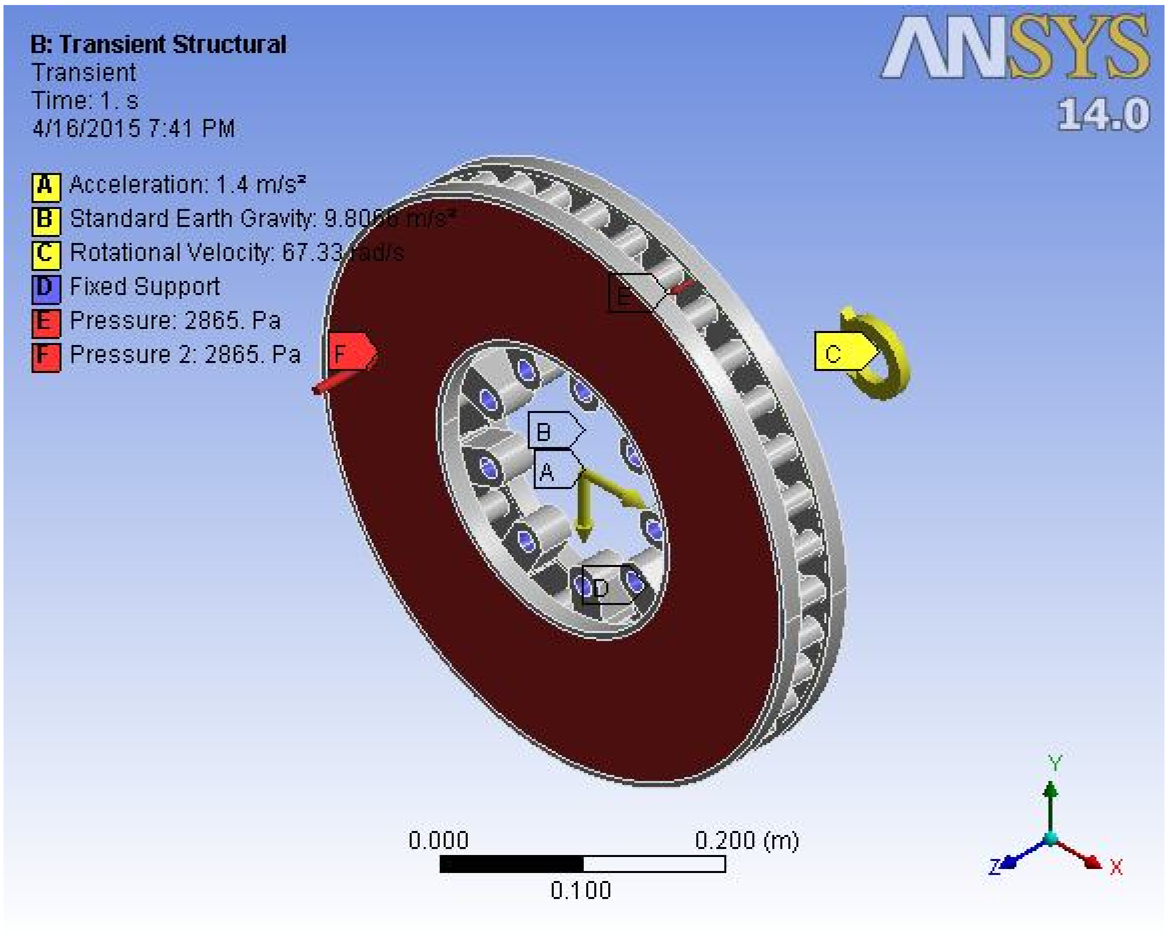


Figure5.4b Pressures acting on the rubbing surface of the disc for downhill track

c) Pressure application for braking on a downhill track to a standstill after maintaining a constant velocity has a value of 163405pa.

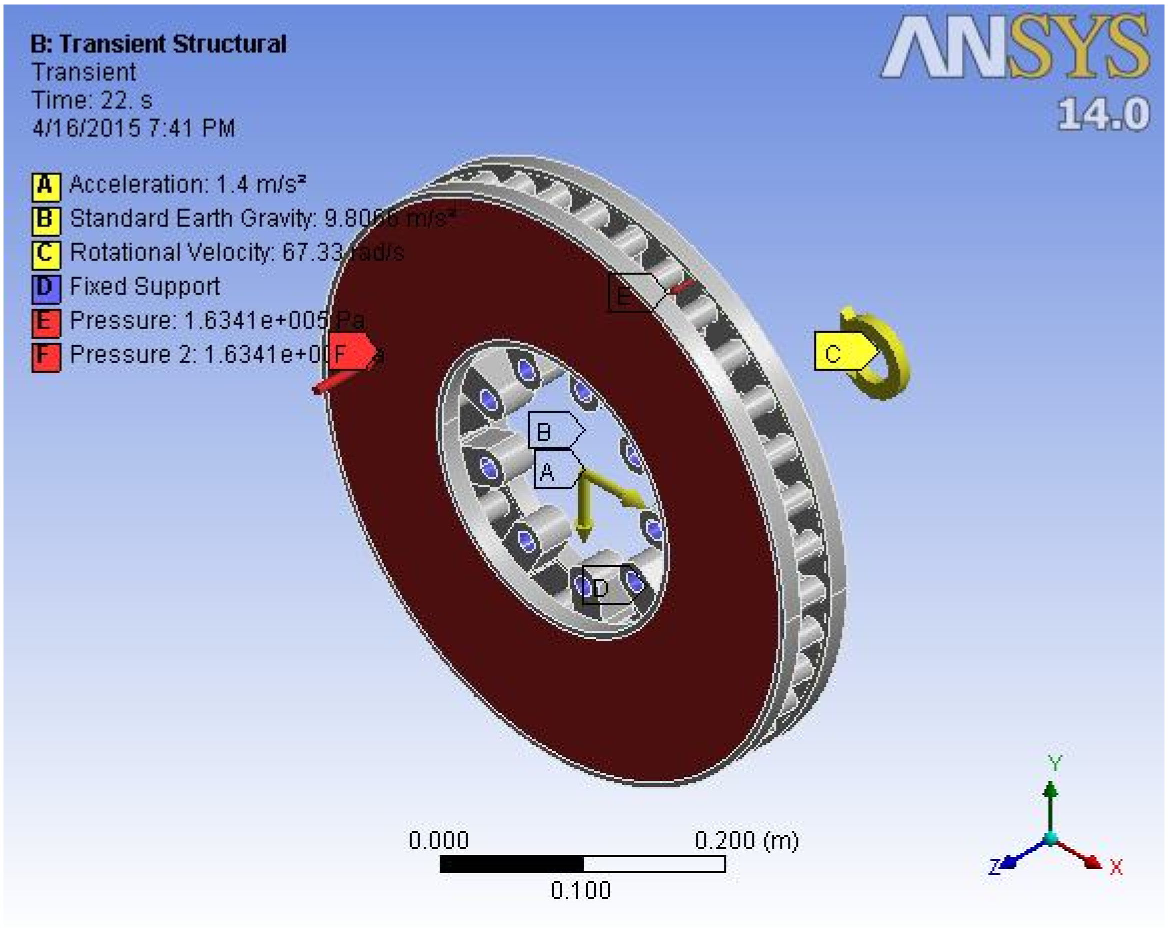


Figure5.4c Pressures acting on the rubbing surface of the disc for downhill track

d) Imported body temperature resulted from thermal analysis of braking from maximum velocity to standstill on a straight track and the imported temperature becomes an input to the structural analysis.

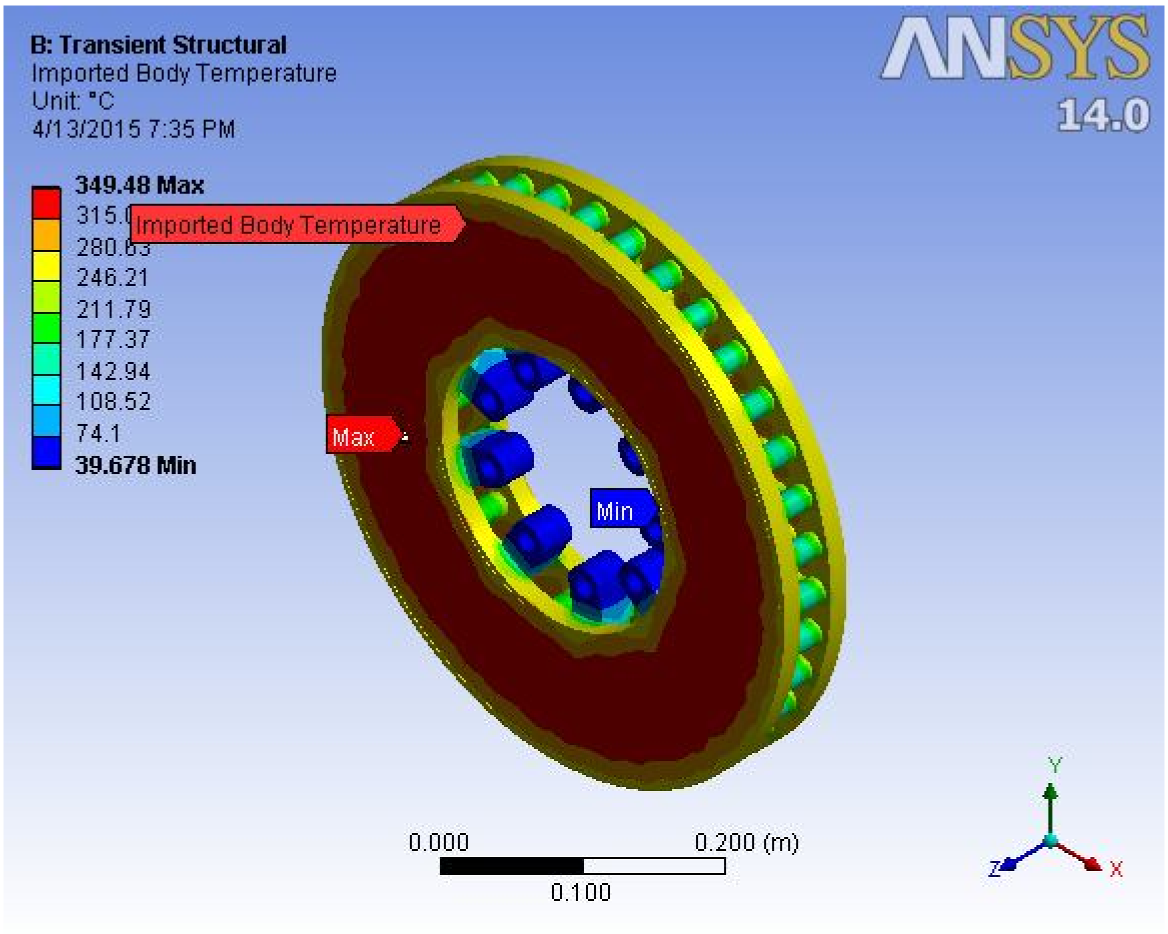


Figure5.4d Imported body temperature of straight track

e) **Imported body temperature resulted from thermal analysis of braking in a downhill track for maintaining a constant velocity and braking to standstill afterwards and the imported temperature becomes an input to the structural analysis.**

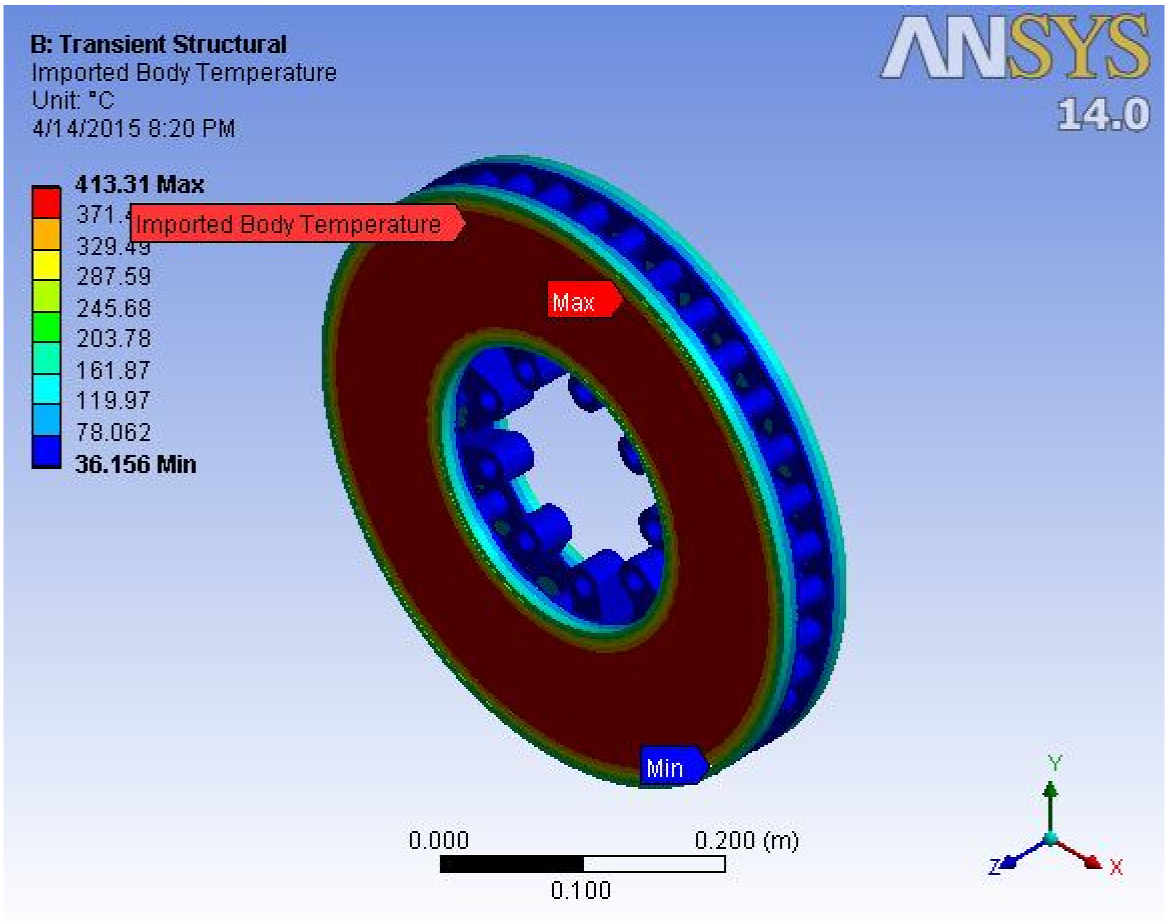


Figure5.4e Imported body temperature of downhill track

CHAPTER SIX

RESULT AND CONCLUSION

Due to the application of brake on the disc brake rotor, heat generation takes place due to friction at the contact area and this temperature has to be dissipated to the ambient and dispersed across the disc brake cross section. The condition of braking is very severe and thus thermal analysis is carried out and with the above load coupled thermal/mechanical analysis is also performed for analyzing the stability of the structure.

After verification of the model, boundary conditions and input parameter applications, the calculation is performed for two cases of braking. Namely:

I. Braking on a straight track (ROAD AUTHORITY to MEXCO SQUARE, EW18 to EW19). For the time of braking t_b = 16 sec in order to stop the A.A.LRT train.

II. Braking on a downhill track from LIDETA to COCA COLA MAZORIA (EW20 to EW21) to maintain a constant velocity and braking to standstill afterwards. For the time of braking t_b = 38 sec in order to maintain a constant velocity and braking to standstill the A.A.LRT train.
   a) Braking on a downhill track from LIDETA TO COCA COLA MAZORIA (EW20 TO EW21) to maintain a constant velocity for 22 seconds.
   b) Braking on a downhill track at COCA COLA MAZORIA (EW21) to standstill after maintaining a constant velocity for 16 seconds.

6.1 RESULTS

The thermal and mechanical results obtained from ANSYS software package are; temperature gradient, thermal flux, deformation and Von Mises stress. The values are summarized and presented in the following tables and figures.

6.1.1 Thermal Result

6.1.1.1 Braking from Maximum Velocity to Standstill on a Straight Track

Table6.1 Analysis results of SiC/6061Al alloy composite

Description	Minimum	Maximum	Time(second)
Temperature [^{o}C]	39.678	349.48	16
Temperature [^{o}C]	50.56	430.87	8.94
Total Heat Flux [W/m^2]	78.359	1.6179e^6	16
Total Heat Flux [W/m^2]	4.7223	2.2376e^6	2.05

Thermal Gradient - A temperature gradient is a physical quantity that describes in which direction and at what rate the temperature changes most rapidly around a particular location.

The results show that in case of emergency braking on the flat track, the maximum temperature reached 283.17°C at standstill. The highest temperature in the friction interface of the SiC/6061Al disc reaches up to 359.9 °C considering airflow cooling and the highest temperature regions are in the friction ring as shown in Fig. 6.1(a). The highest allowed temperature of the disc is 600 °C under the long term braking condition. In Fig. 6.1(a), it is noted that the temperature decreases in the direction of median plane of the disc to reach its minimal value. The maximal temperature is reached after a time period of 7.14 s. The SiC/6061Al brake disc has the best thermal behavior as can be seen from the result for this type of braking conduction.

a) The maximum temperature obtained at the end of simulation time.

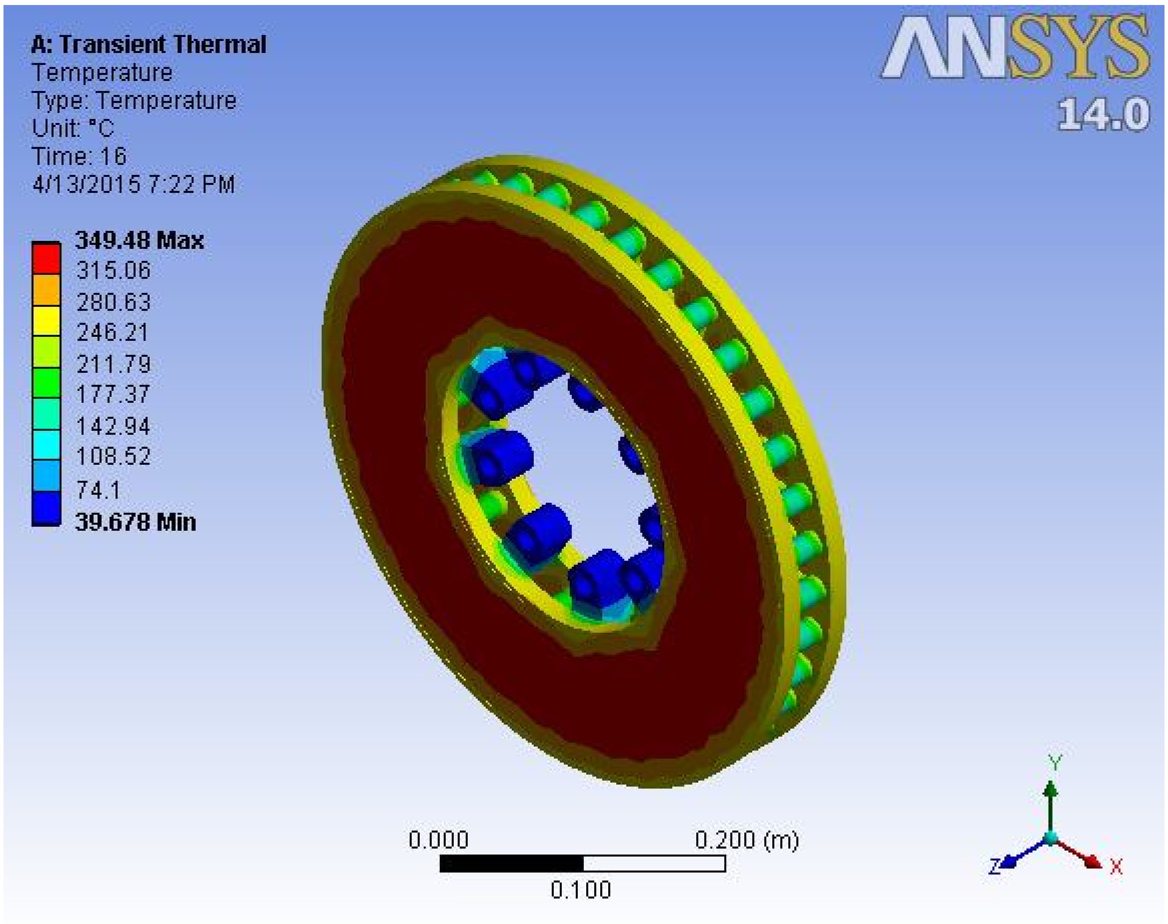


Figure6.1a Temperature gradient

Thermal flux - Heat flux or thermal flux is the rate of heat energy transfer through a given surface.

b) The maximum heat flux obtained at the end of simulation time.

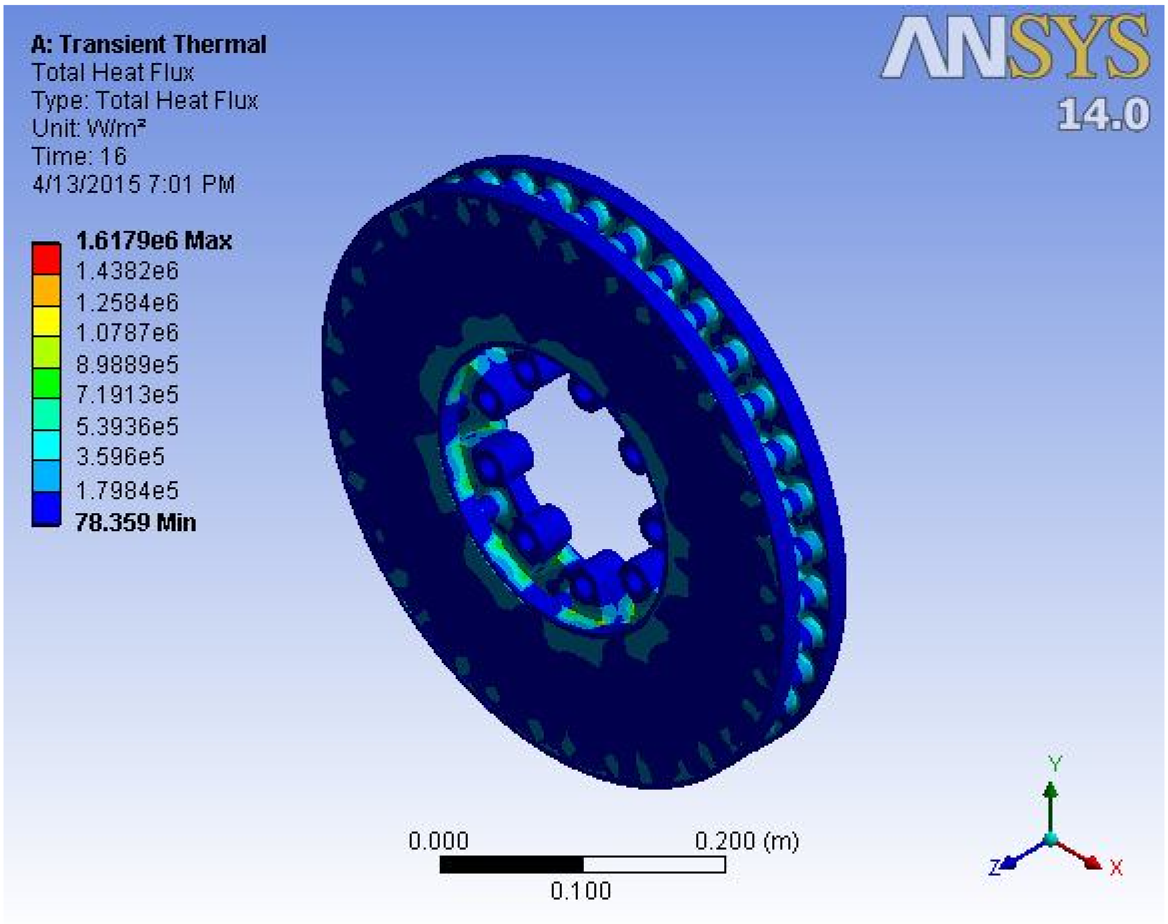


Figure6.1b Heat flux at stop

6.1.1.2 Braking on a Downhill Track to Maintain a Constant Velocity and Braking to Standstill Afterwards

I. Braking on a downhill track to maintain a constant velocity

Table6.2 Analysis Results of SiC/6061Al alloy composite for downhill track

Description	Minimum	Maximum	Time(second)
Temperature [^{o}C]	15.394	253.08	22
Total Heat Flux [W/m^2]	13256	1.8926e^5	21.94

Thermal Gradient - A temperature gradient is a physical quantity that describes in which direction and at what rate the temperature changes most rapidly around a particular location.

a) The maximum temperature obtained at the end of simulation time.

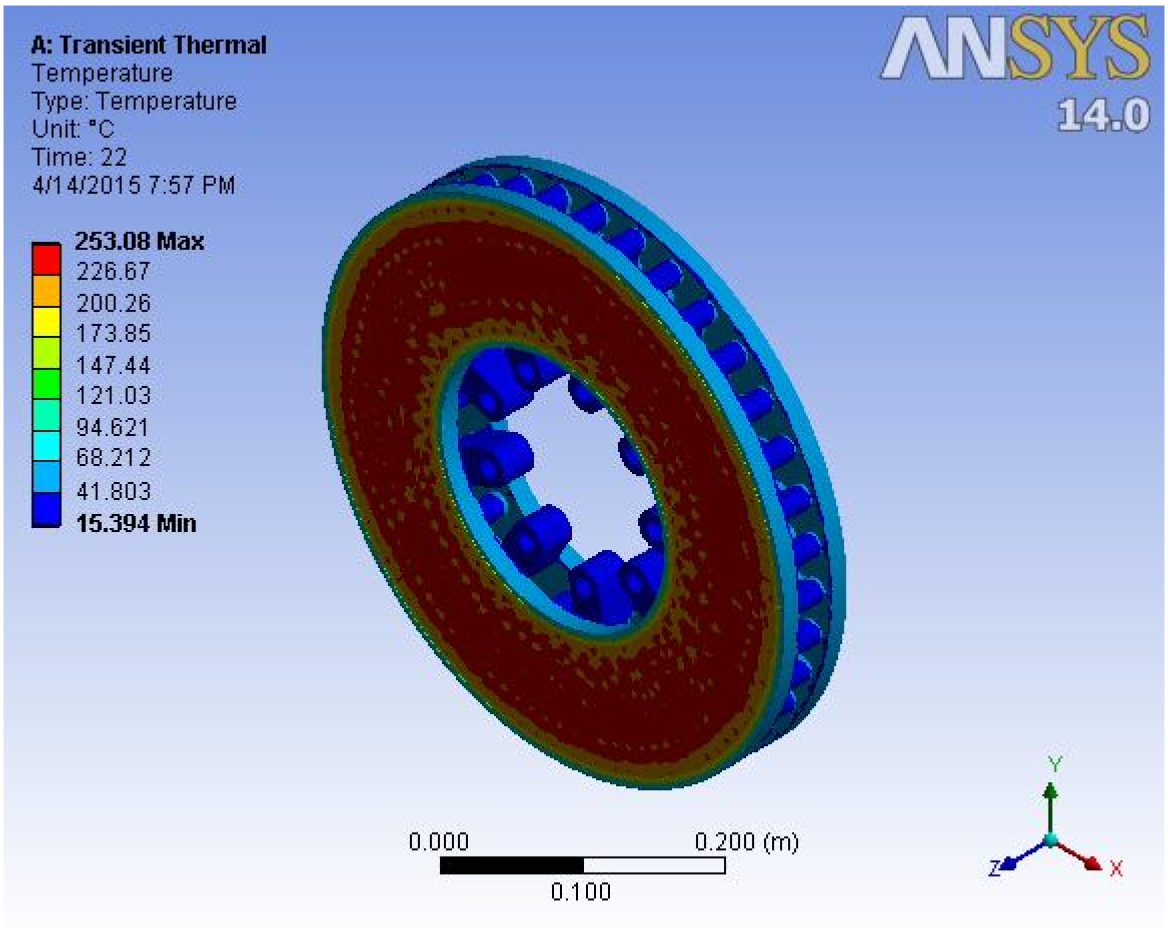


Figure6.2a Temperature gradient

b) The maximum heat flux obtained at the end of simulation time.

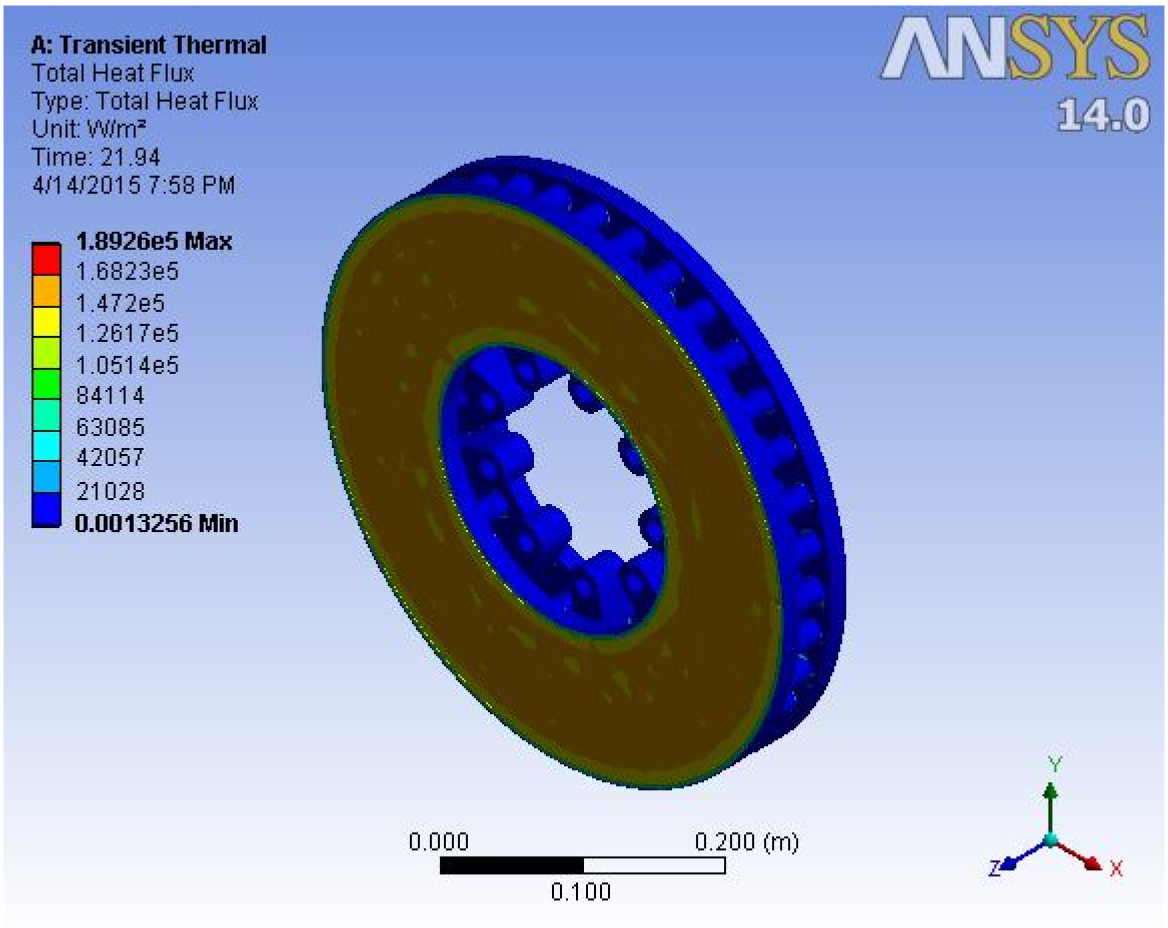


Figure6.2b Heat flux

II. Braking on a downhill track to standstill after maintaining a constant velocity

Table6.3 Analysis Results of SiC/6061Al alloy composite when braking to standstill on a downhill track

Description	**Minimum**	**Maximum**	**Time(second)**
Temperature [^oC]	36.156	413.31	38
Temperature [^oC]	170.45	530.57	29.64
Total Heat Flux [W/m^2]	0.01542	$1.4447e^6$	38
Total Heat Flux [W/m^2]	0.3208	$3.1733e^6$	26.54

Table6.4 Result summery

Object Name	**Temperature**	**Total Heat Flux**	**Directional Heat Flux**
State	Solved		
Scope			
Geometry	All Bodies		
Definition			
Type	**Temperature**	**Total Heat Flux**	**Directional Heat Flux**
Display Time	Last		
Orientation			X Axis
Coordinate System			Global Coordinate System
Results			
Minimum	36.156 °C	1.5423e003 W/m²	1.0078e+005 W/m²
Maximum	413.31 °C	1.4447e+006 W/m²	95300 W/m²
Minimum Value Over Time			
Minimum	36.156 °C	1.9479e-005 W/m²	1.9101e+005 W/m²
Maximum	170.54 °C	2.5441e-002 W/m²	214.24 W/m²
Maximum Value Over Time			
Minimum	170.45°C	1052. W/m²	193.1 W/m²
Maximum	530.57 °C	3.1733e+005 W/m²	1.867e+005 W/m²
Information			
Time	38. s		
Load Step	38		
Sub step	13		
Iteration Number	494		

Figure 6.3(a) shows that in case of emergency braking after maintaining a constant velocity on the downhill track, the highest temperature in the friction interface of the SiC/6061Al disc reaches up to 460.62 °C with considering the airflow cooling. The highest temperature gradient of the friction layer decreases more smoothly along the *z*-direction.

The numerical simulation shows that radial ventilation plays a very significant role in cooling of the disc 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 disc are bolted to the hub, since heat will be removed faster from hotter parts of the disc. So the airflow cooling should be very effective to reduce the risk of MHS formation.

The thermal expansion coefficient and the elastic modulus of the SiC/6061Al composite have an important effect on the properties of friction surfaces of the brake disc [30, 50, and 51].

It can be concluded that the continuous microstructure of the SiC/6061Al composite has significant effect on the thermo-mechanical response of the discs. In an emergency stop, when most of the kinetic energy of the vehicle is transmitted to the discs as heat, the disc may reach a temperature of 460.62 °C. One of the most dangerous conditions in frictional brakes is the formation of MHSs. In this case, we note that the SiC/6061 composite will not only reduce the maximum temperature but also naturally tend to reduce thermal gradients.

And SiC/6061Al composite might remove heat from the brake disc much more rapidly than the solid Al disc, which may thus provide a means of alleviating some of the thermal problems encountered in practical situations.

a) **The maximum temperature obtained at the end of simulation time.**

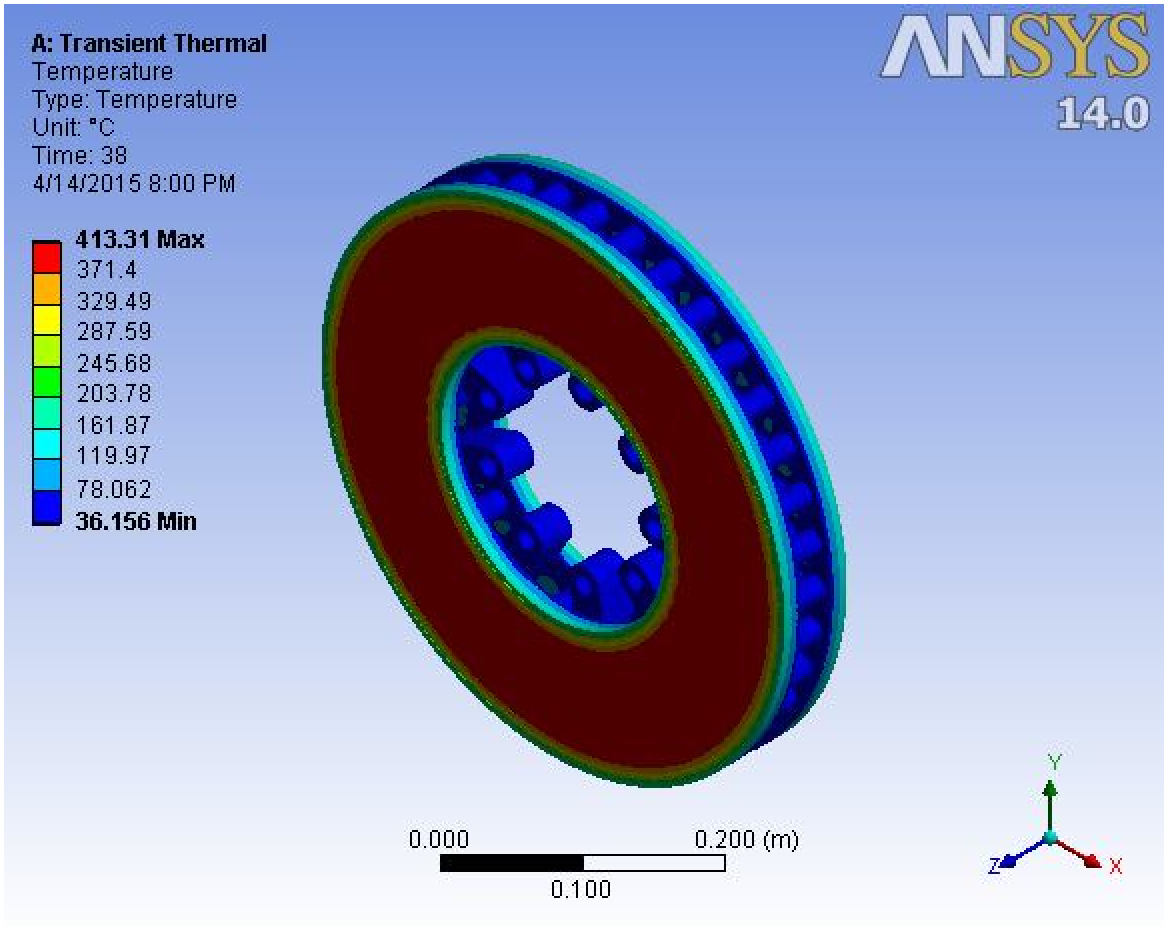


Figure6.3a Temperature gradient

b) The maximum heat flux obtained at the end of simulation time.

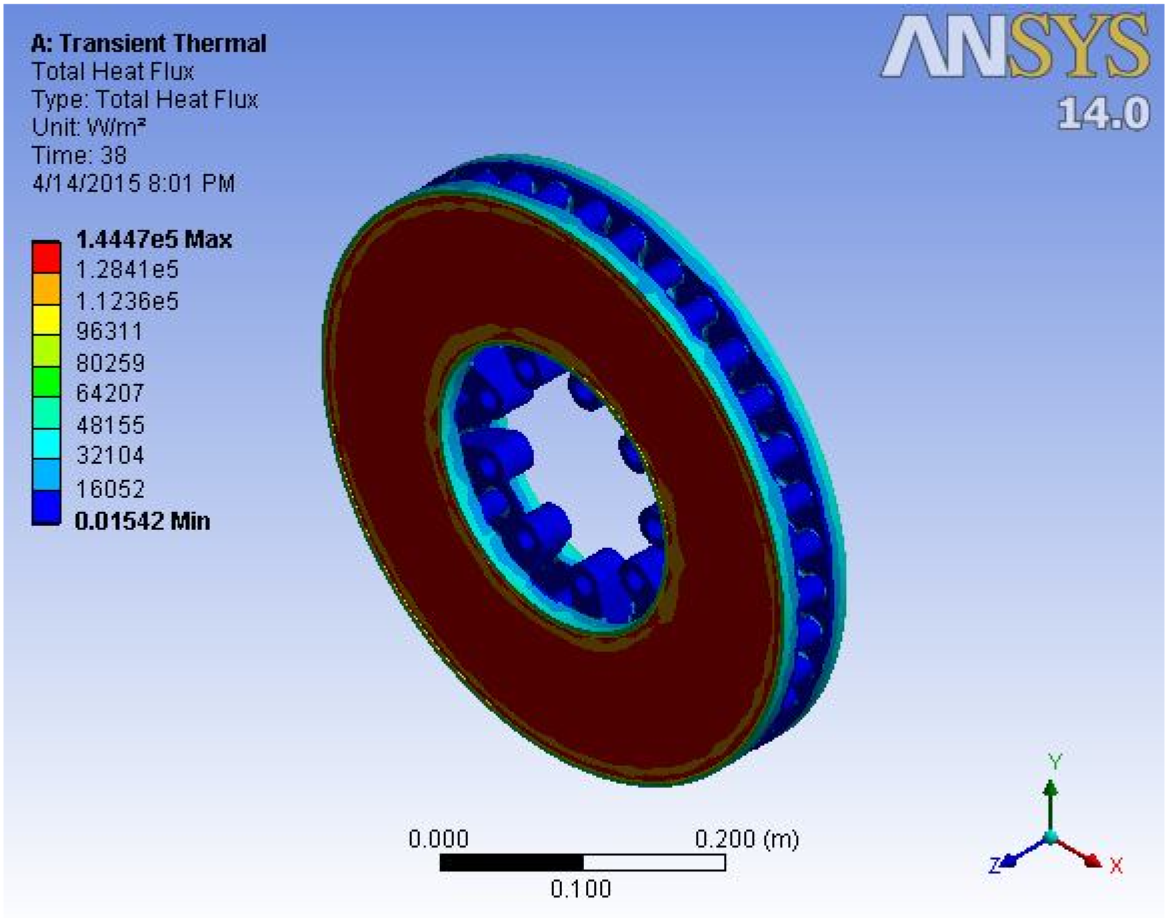


Figure6.3b Heat flux at stop

6.1.2 Mechanical Result

6.1.2.1 Braking from Maximum Velocity to Standstill on a Straight Track

Table6.5 Structural results of SiC/6061Al alloy composite for straight track

Description	Minimum	Maximum	Time(second)
Total Deformation [m]	0	$3.7943e^{-7}$	16
Von Misses Stress [Pa]	9552.8	$3.704e^{6}$	16

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.

a. Total deformation

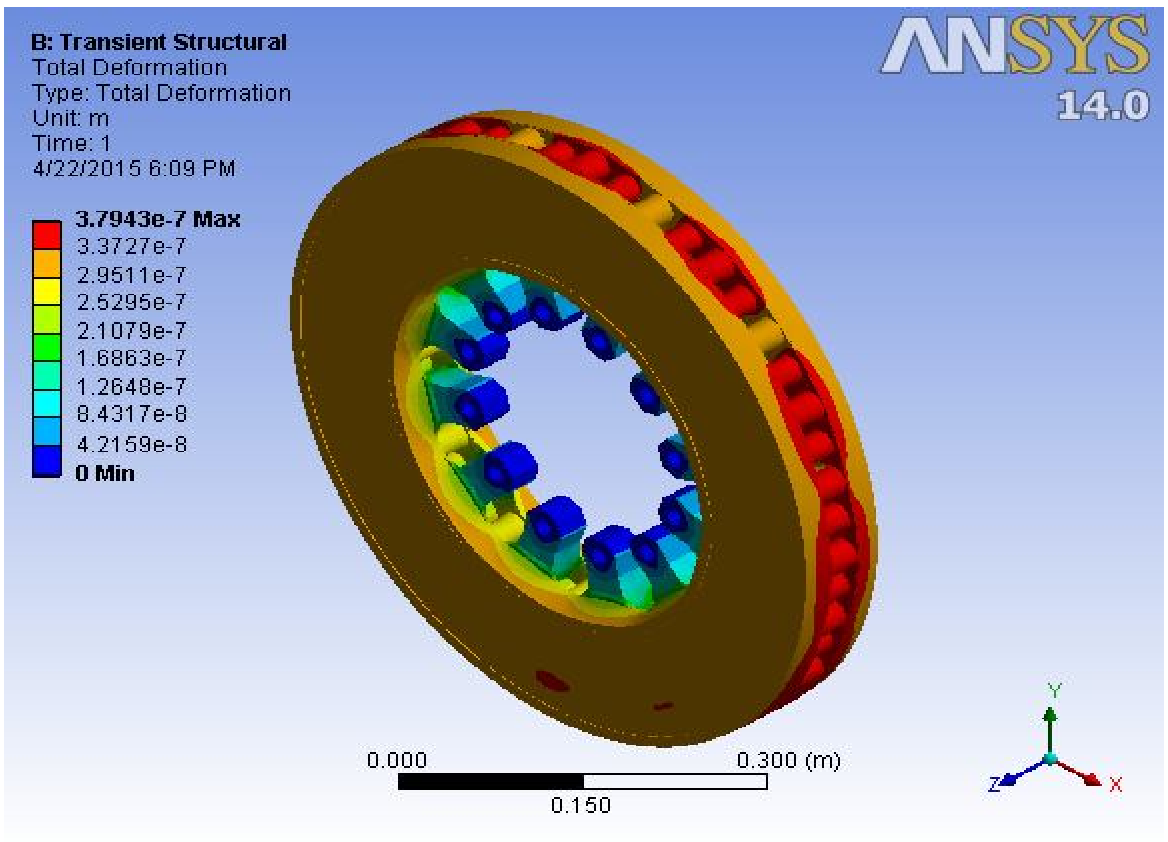


Figure6.4a Total deformation for straight track

b. Von Mises equivalent stress

Von Mises 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. Figure 6.4(b) shows that cooling ribs play a very significant role in cooling of the disc in the braking stage considering airflow cooling. The variation of non dimensional radial thermal stress is due to thermo-mechanical load of the mounted SiC/6061 brake disc with different values of the temperature gradient. This phenomenon can be explained by the presence of interactive effects between thermo-mechanical load and contact area. It is seen that the maximum thermal radial stress is 3.704 MPa, which is lower than 314 MPa as shown in Figure 6.4(b). Thermal stress distribution of friction interface is not Centro-symmetry and the thermal stress of course has a curved elliptical shape as a result of airflow cooling.

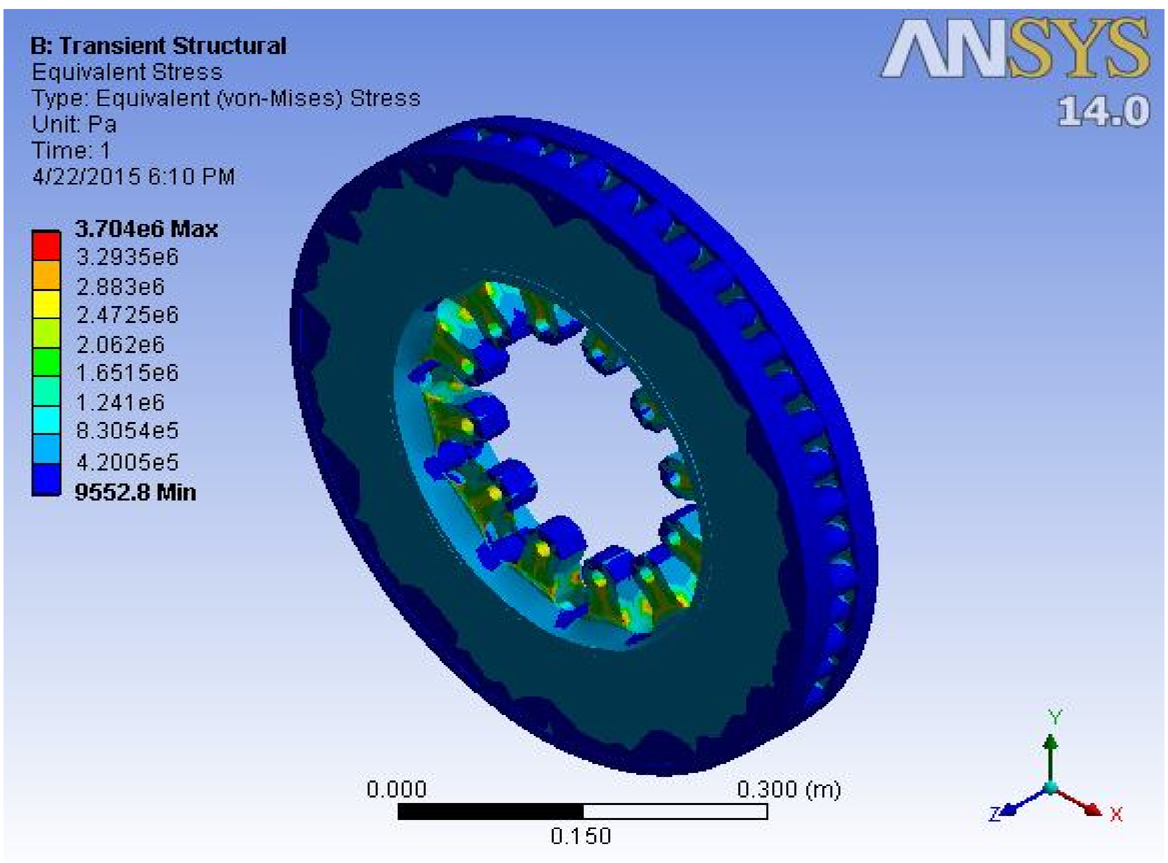


Figure6.4b Equivalent stress for straight track

6.1.2.2 Braking on a Downhill Track to Maintain a Constant Velocity

Table6.6 Structural results of SiC/6061Al alloy composite for downhill track

Description	Minimum	Maximum	Time(second)
Total Deformation [m]	0	$2.9969e^{-7}$	22.125
Von Misses Stress [Pa]	8993	$3.7039e^{6}$	22.125

a. Total deformation

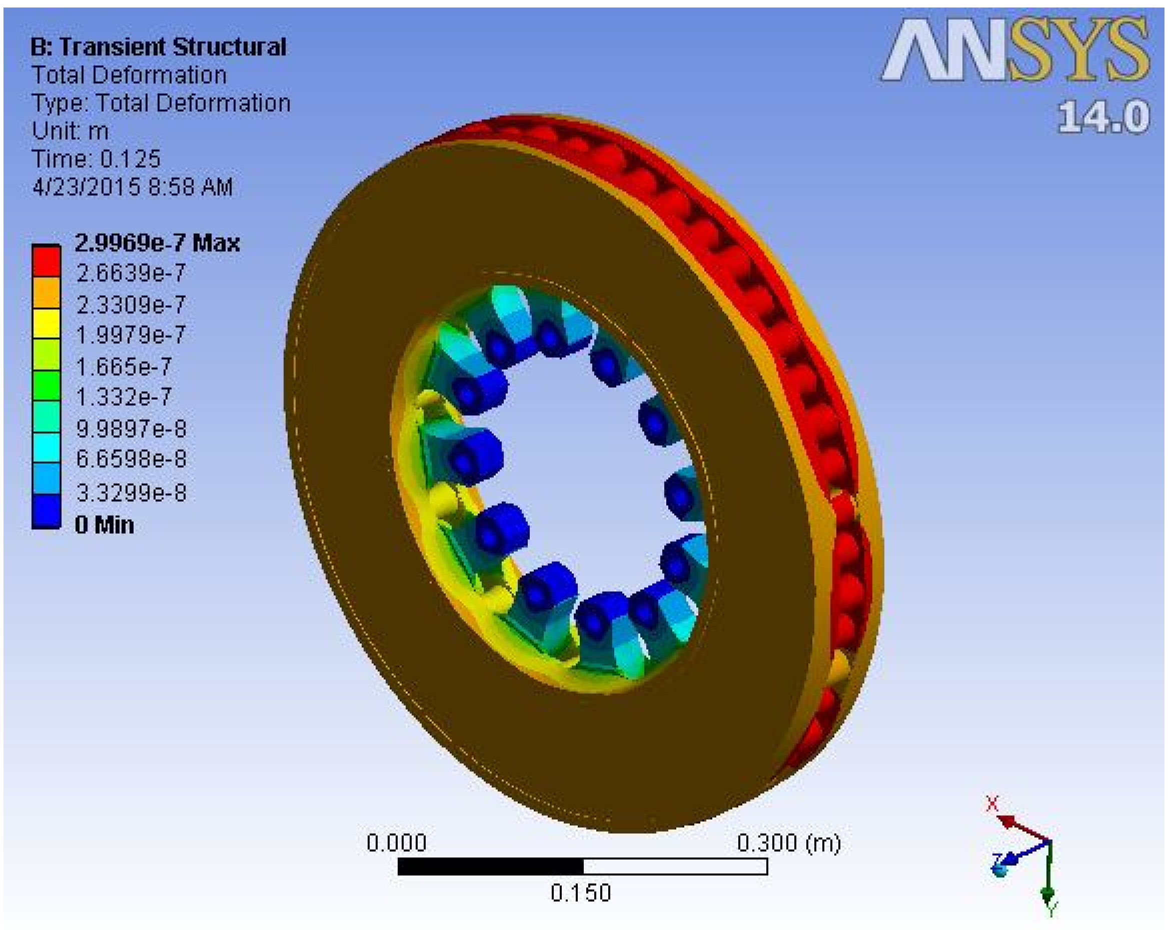


Figure6.5a Total deformation for downhill track to maintain constant velocity

b. Von Mises equivalent stress

Figure 6.5(b), presents the distribution of the constraint equivalent of Von Mises stress to various moments of simulation, the scale of values varies from 0.009 MPa to 3.7039 MPa.and the maximum stress is seen at the fixing zone.

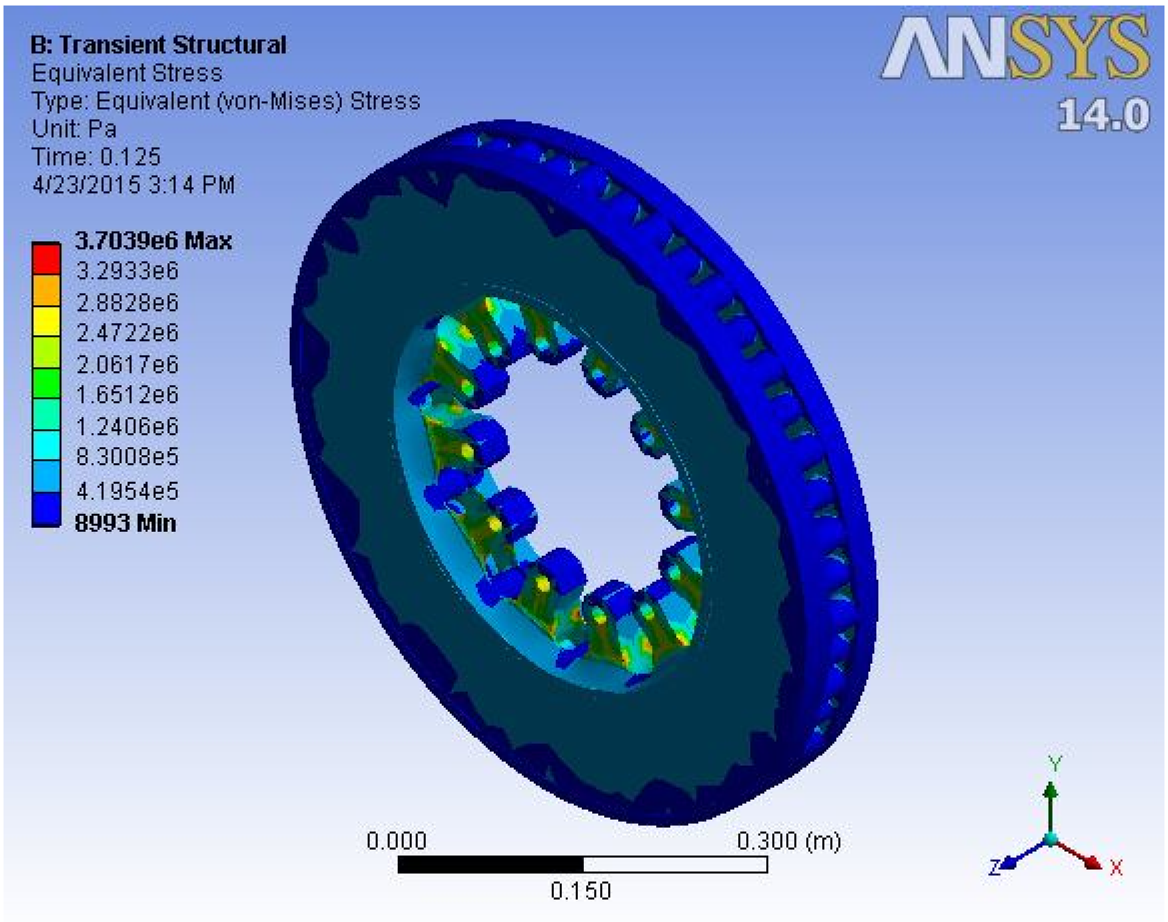


Figure6.5b Equivalent stress for downhill track to maintain constant velocity

6.1.2.3 Braking on a Downhill Track to Standstill after Maintaining a Constant Velocity

Table6.7 Structural results of SiC/6061Al alloy composite for downhill track when braking to standstill

Description	Minimum	Maximum	Time(second)
Total Deformation [m]	0	$2.5425e^{-6}$	38
Von Misses Stress [Pa]	8993	$1.9794e^{7}$	38

Thermal stresses appear due to the rise of the disc working temperature. However, in addition to thermal loads themselves also the effect of centrifugal loads on the disc brake and the effect of mechanical pressure of brake clamps were considered. The stated stresses are combined stresses according to Mises. It is known that the thermal stresses in the disc appear because of the temperature rising. The goal of this analysis is to determine the influence of the centrifugal load and the thermal load. The stress is given by von Mises stress. The stress is 19.8 MPa between the disc and pad at the friction interface as shown in Fig. 6.6(b).

a) Total deformation

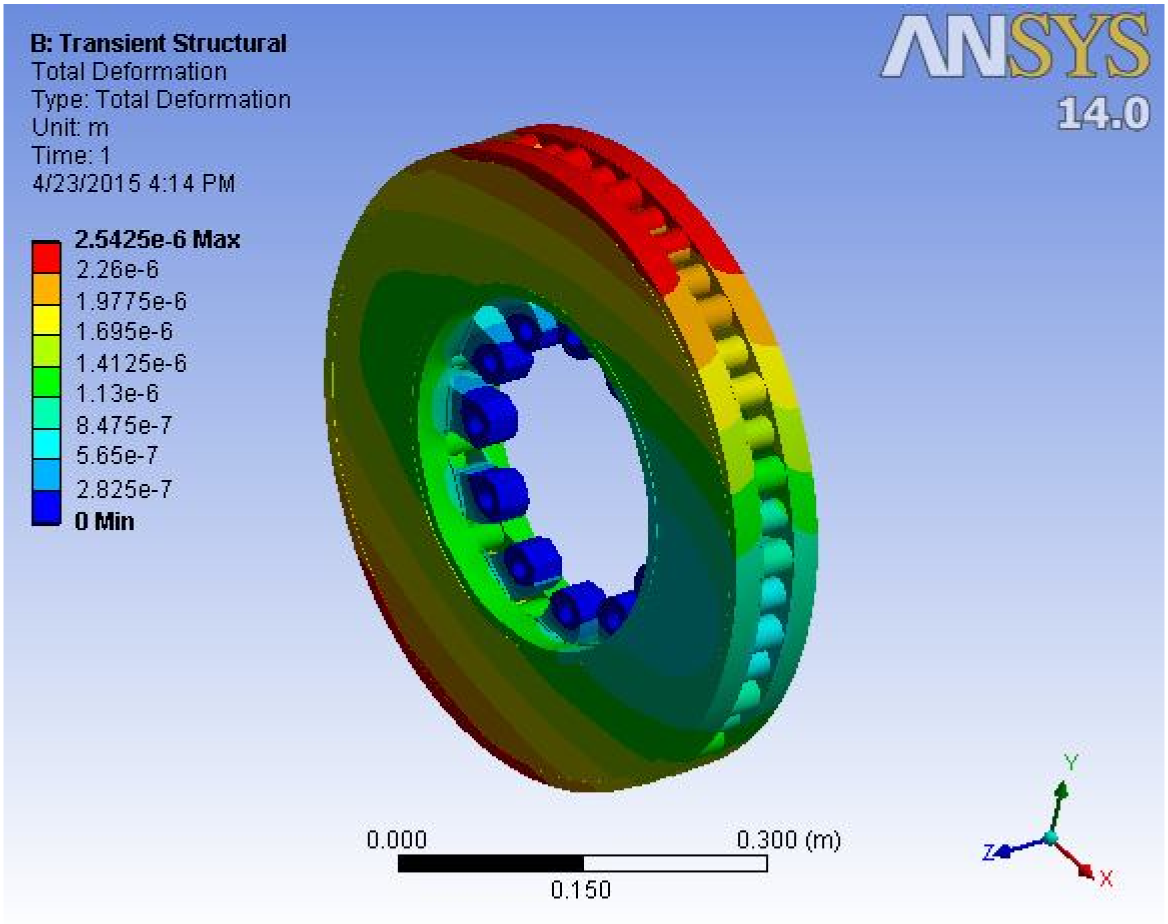


Figure6.6a Total deformation on a downhill track when braking to standstill

b) Equivalent stress

In case of braking on downhill track, the highest stresses amounted to *19.8 MPa*, namely beside the point of transition of fixing teeth. At the points with the highest heat load the stresses were equal to *19.8 MPa*. Therefore as can be seen clearly the prolonged braking increases the stress from ***3.704MPA*** to ***19.8MPA.***

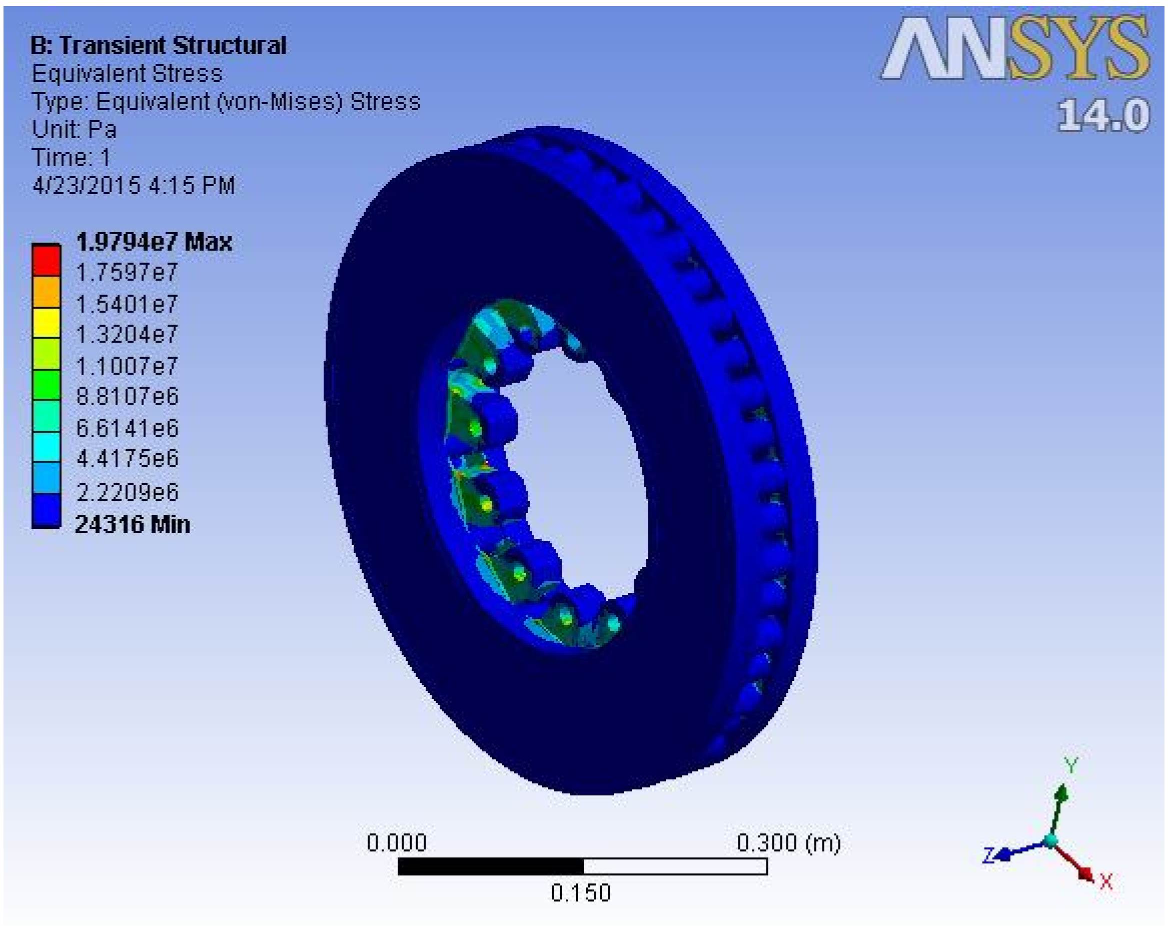


Figure6.6b Equivalent stress on a downhill track when braking to standstill

6.2 Discussion

This analysis of the brake disc was implemented for two braking conditions; braking on a horizontal track to a standstill and braking on a downhill track for maintaining constant velocity and braking to a standstill afterwards. Following temperatures were taken into account; the initial temperature of the disc, disc fixture and the ambient temperature, all equal to 38 °C.

In case of braking on the horizontal track, the contact surface temperature of the disc after 8.94 s is 430.87 °C. Given the lesser and lesser input of the heat flux, the temperature of the disc decreased (lower speed) to a final temperature of 349.48 °C (after 16 s).

It was taken into account that the temperature of the cooling ribs and channels did not notably rise during the period of one braking. This fact comes from the limited heat conduction speed of the material. Mechanical analysis shows that the deformation and the stress at the end of the simulation are $3.7943e^{-7}$and $3.704e^{6}$ respectively.

In case of braking downhill the highest temperature on the disc after expiration of 22 s was equal to 253.08 °C, namely on the disc/brake pad contact surface (Fig. 6.2a). During further braking to a standstill on the track the brake disc warmed up to the highest temperature of 530.57 °C within 29.64 s (Fig.6.3a). Given the lesser and lesser input of the heat flux, the temperature of the disc decreased (lower speed) to a final temperature of 431.31 °C after 38 s (Fig.6.3a).

The deformation and the stress when braked to maintain a constant velocity is found to be $2.9969e^{-7}$and $3.7039e^{6}$ Fig.6.5a & 6.5b) respectively and when braked to standstill at 38s, the deformation and the stress are $3.7791e^{-7}$and $1.9794e^{7}$ (Fig.6.6a & 6.6b) respectively.

6.3 Conclusions

The thermal and stress analyses of a SiC/6061Al composite brake disc for A.A. LRT railway vehicles during emergency braking at a speed of *80 km/h* with the airflow cooling were studied using FE and CFD methods. The results revealed that SiC/6061 composite can effectively enhance the thermal performance of the brake disc; the highest temperature after emergency braking was *530.57 °C* and *430.87 °C* braking on downhill track after maintaining constant velocity and braking on a straight track, which occurred at about *29.64 s* and *8.94 s* respectively. The maximum stress can reach *19.8 MPa* and *3.704 MPa* braking on downhill track after maintaining constant velocity and braking on a straight track for emergency braking respectively. The thermal expansion coefficient and the elastic modulus of the SiC/6061Al composite have an important effect on the properties of friction surfaces of the brake disc.

It can be concluded that the continuous microstructure of the SiC/6061Al composite has significant effect on the thermo-mechanical response of the discs. In an emergency stop, when most of the kinetic energy of the vehicle is transmitted to the discs as heat, the disc may reach a temperature of *530.57 °C*. One of the most dangerous conditions in frictional brakes is the formation of MHSs. In this case, it is noted that the SiC/6061 composite will not only reduce the maximum temperature but also naturally tend to reduce thermal gradients.

And SiC/6061Al composite material might remove heat from the brake disc much more rapidly than the solid Al disc which may thus provide a means of alleviating some of the thermal problems encountered in practical situations. To help the disc cooling down the heat exchange surface was increased for the ventilated disc. Similar to compressors the air is sucked through the vanes by the rotational motion. This leads to a much higher heat exchange compared to the single solid disk and therefore lower surface temperatures. It shows that airflow cooling can remove heat from the brake disc rapidly. The present work shows that the ventilation system plays an important role in cooling the brake disc and provides a good high temperature resistance.

The SiC/6061Al alloy composite is a lightweight material. Using it for disc brake is an effective choice. The new trend of equipping railway vehicles is orientated towards replacing the grey cast iron brake discs by brake discs made from composite materials. Reduced sound barriers (wear) during braking and higher resistance to temperatures are other major advantages of composite materials other than their densities.

The investigation of usage of new materials is required which improves the braking efficiency and provide greater stability to vehicle. The suitable composite material which is lighter than cast iron and has good young's modulus, yield strength and density properties. The low weight, the hardness, the stable characteristics also in case of high pressure and temperature, the resistance to thermal shock and the ductility provide long life time of the brake disc and avoid problems resulting of loading which are typical for the classic grey cast iron brake discs. The most suitable SiC/6061Al composite material is appropriate.

In the present research, the mass of the brake disc made from SiC/6061Al composite is *33.038Kg* and the mass of the brake disc made from grey cast iron for A.A LRT brake disc equals *62.332Kg*. Hence *29.294Kg* of A.A LRT brake disc mass will be reduced from a single brake disc rotor. The lower density of SiC/6061Al composite made them an economic advantage over grey cast iron with respect to efficient use of fuel and fabrication expenses.

The present study can provide a useful design tool and improve the brake performance of the disc brake system and also reduce the weight of the brake disc. From table3.2, it can be concluded that all the values obtained from the analysis are less than their allowable values. Hence, it can be concluded that, the brake disc is safe based on the strength and rigidity criteria.

6.4 Recommendation

The analysis has not considered the effects of conduction and radiation heat transfer, humidity, shearing force, material tribology, wear analysis, manufacturing and production cost, residual stresses and the effect of cyclic loads during brake disc life time and more complicated brake disc geometry together with its component and brake disc interfaces to fully represent the real physical phenomenon is not considered. And it is left for future work. It would be interesting to solve the thermo-mechanical problem of disc brakes with an experimental study to validate the numerical results. And it is also left for future work.

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