Addis Ababa University

Addis Ababa Institute of Technology

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Thermal Analysis of Different Train Brake Disc Shapes by Finite Element Method: Case Study on Addis Ababa Light Railway Transit

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Abstract

Disc brake is used to decelerate and stop a train. During the process, kinetic energy is changed into heat energy due to the friction between the disc surface and disc pad. The main problem with braking and stopping a vehicle is the huge quantity of heat flux that enters the disc in a short period of time, causing a rapid temperature rise which leads to brake fade, gas build-up, and stress. Therefore, it is important to come up with a way of reducing the temperature- rise by investigating the thermal behavior of the train brake disc. In this paper the thermal analysis and comparison of four different brake disc models is done using ANSYS 19.2 software. The analyzed brake discs are modelled using the design specifications of Addis Ababa light railway transit by SOLIDWORKS13. The disc models include one model with straight cooling fins and without a surface drill and 3 models with a curved cooling fins and equally spaced surface drills. The three drilled models also include one model with holes of 5 mm diameter, one model with holes of 7.5 mm diameter, and other model with holes of 10 mm diameter to investigate the effect of varying the holes' diameter on the temperature-rise. The analysis of each model is done at emergency braking conditions of Addis Ababa light railway transit (at 9.72s braking time, 2 m/s^2 deceleration and at a speed of 19.44m/s) by considering air cooling and maximum gradient of Addis Ababa light railway transit (3.15^0). Sic/6061Al alloy is used as a comparison material thereby to show its effectiveness at emergency braking and at the highest gradient. ANSYS-FLUENT is used to calculate the average heat transfer coefficient of each model so as to use it as a boundary condition for the thermal analysis. In addition to this, the structure of each model is investigated so as to show the maximum von-mises stresses and deformations. It has been shown that adding a surface drill and increasing the diameter of the drilled holes has a significant effect on enhancing the heat transfer coefficient value thereby reducing the maximum temperature-rise. The disc model with hole diameter of 10 mm is the best (348.33℃), the model with hole diameter of 7.5 is the second best (377.5℃), and the model with hole diameter of 5 mm takes the third rank (382.09℃) while the non-drilled model is the worst (399.81℃). The model having holes of 10 mm diameter is recommended to be utilized by Addis Ababa light railway transit by using the Sic/6061Al alloy brake disc material.

Key words: ANSYS-FLUENT, ANSYS 19.2, CFD, SOLIDWORKS13, von-mises stress, transient thermal, transient structural

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Acronyms and Abbreviations

a = vehicle decelaration

A_d= the brake disc area

Fdisc = force applied on the disc

h= rate of heat transfer by convection ($W/m^2 K$)

I = moment of inertia of rotating parts

ke = vehicle intial kinetic energy

M = vehicle mass

Pe=potential energy due to track gradient

$Q(t)$ = rate of heat generation in watt

rw = wheel radias

rd = effective radius of the disc

t = time in seconds

tb= braking time

vo = vehicle initial velocity

wo = intial angular velocity of vehicle wheel

γ = proportion of heat energy entering the disc(0.9)

α =angle of inclination

μ=the coefficient of friction between disc and pad

AALRT= Addis Ababa light railway transit

CAD= computer aided design

CFD= computational fluid dynamics

FEA= finite element analysis

FEM= finite element method

HTC= heat transfer coefficient

CHAPTER ONE

INTRODUCTION

1.1 Background

The term "railway" refers to a system of land mass transportation. Trains use steel wheels on a designated steel guideway defined by two parallel rails to travel on their own (diesel traction) or with remotely transmitted power (electrical traction). Passengers and freight are transported by train. Its range of capability allows it to cross any distance in any situation (urban, suburban, peri urban, regional and interurban). Its range for passenger transportation is normally over 1,500 kilometers, while lengths for freight can be significantly longer [1]. It is an environmentally beneficial mode of transportation for the twenty-first century and acts as a link between cities. Because of the economic benefits, there has recently been a surge in interest in increasing the speed of railway cars[2]. Many of the world's large cities grew alongside railways, and they can no longer rely solely on road cars for mobility. With growing concern about global environmental challenges, public transportation is becoming a more essential means of expanding and revitalizing large cities while using less energy and other resources [3].

In modern railway systems, increasing maximum running speed and reducing weight are significant development problems. While increasing speeds and increasing economy, it is also necessary to maintain safe operating conditions. The integrity of the brake system, as a crucial component, is the guarantee of priority in the train speed design process[4].The brakes on railway vehicles are used to permit for deceleration, manipulate acceleration (downhill), and maintain their status even as parked. While the primary idea is much like that of an avenue car, the utilization and operational factors are extra complicated because of the need to function several connected carriages and achieve success on cars without a primary mover. the brake must be powerful enough to bring the car to a complete stop in the shortest period of time and the motive force have to hold properly and manipulate the vehicle in an emergency. Even if the motive force isn't always there, the brake also has to maintain the car motionless [5]. In recent years, braking systems are being forced to meet competing requirements for stable braking performance and quality, in addition to improved power output and faster speeds obtained by cars [6].The brakes utilized in railway cars may be labeled into the subsequent categories primarily based on how they are activated as

Pneumatic Brake, Mechanical Brake and Electromagnetic Brake. The Pneumatic(air) brake system is categorized as automatic air brake system and straight air brake system. In the automatic air brake system air compressors are situated every two to four carriages and supply compressed air to the air brakes. The compressed air is pumped beneath the coach flooring to major air reservoirs, where it is compressed to an approximate density of 8 kg per square centimeter. Whereas, in straight air brake system each coach does not have a control valve or an auxiliary air reservoir. When the brake valve is opened, pressurized air from a straight air pipe is forced into the brake cylinders, which activates the fundamental braking system. In electromagnetic braking system a braking force is generated by magnetic repulsion caused by eddy currents developed on the top surface of the rails. Wheel tread brakes, axle-mounted disc brakes, and wheel-mounted disc brakes are also the three fundamental braking devices employed by mechanical braking systems. In these brake devices, a brake shoe applies friction force to the disc. The applied pressure is adjusted to control the braking force. The brake shoe generates a sliding action by supplying friction force to the wheel tread in a wheel-tread brake. It is hooked up to the wheel and/or the axle. To prevent the wheel, friction fabric inside the shape of brake pads, established on a tool referred to as a brake caliper, is pressurized mechanically, hydraulically, pneumatically, or electromagnetically in opposition to each aspect of the disc [5]. The brake disc is generally made from cast iron; however**,** it is able to additionally be made from composites along with strengthened carbon-carbon or ceramic matrix composites [7].

Many systems, including braking systems, require precise predictions of maximum temperatures, particularly for both discs and pads. The goal of thermal and structural analysis is to figure out how to deal with the disc's high spinning speed. For the fact that the temperature determines the thermal behavior of the brake disc, thermal analysis is the first step in the study of braking systems [8]. During braking process by disc brake, kinetic energy is changed into heat energy due to the friction between the disc surface and disc pad, resulting in the disc surface temperature to rise in a short period of time. This causes thermal stress leading to brake fade and crack which then leads to brake failure and eventually the fracture of the entire brake disc [9,10,8].Train brake performance is highly affected during emergency braking conditions due to minimum stopping distance and delay time [11]. Maximum temperature is observed in the middle of braking process instead of braking end time [12]. The linear momentum of the vehicle (vehicle mass and velocity), as well as the deceleration rate and friction coefficient, govern the temperature rise of a brake disc during braking. Almost all of the generated heat energy is absorbed by the friction material and the disc rotor during short braking and relatively moderate deceleration [13]. To dissipate the absorbed heat and improve the life time of the brake disc either the shape of the brake disc or the material used to manufacture the brake disc should have a very good heat dissipation characteristic. Many approaches have been proposed to minimize the maximum temperature rise, based on the type of the operating system, by enhancing the disc structure and utilizing different brake disc materials. One way is to use a ventilating brake disc that increases the surface area of brake rotor and enhances air flow around the surface by increasing the convective heat transfer coefficient. It has faster rate of heat dissipation to surrounding air compared to full disc brake and has critical effect in reduction of thermal stress and weigh[14, 7]. In this research work a further temperature-rise reduction ways are presented by utilizing and comparing different four train brake disc models. The models consist of one model, the current AALRT brake disc, that has straight cooling fins without any surface drill (except bolting holes) and other three models that have a curved fins and equally spaced surface drills with different hole diameter. The effect of varying drill diameter on the enhancement of the average heat transfer coefficient and on minimizing the maximum temperature rise has been studied in order to compare the models and come up with a disc model with minimum temperature-rise for AALRT. The structure of each model has also been analyzed by determining the von-mises stress and deformations. In addition to this, selecting a suitable brake disc materials in terms of maximum temperature rise and mechanical design point of view has a vital importance. Even though the effectiveness of the Sic/6061Al alloy composite material is shown at straight and downhill braking conditions of AALRT [3], one should also investigate, to fully recommend its utilization, the effectiveness of the material at the highest gradient and emergency braking conditions of AALRT. For this reason (and based on the material selection on chapter three), the Sic/6061Al is selected as a comparison material of the different brake disc shapes thereby to show its effectiveness at the mentioned braking conditions of AALRT.

1.2 Problem statement

During emergency braking, the braking distance is short. The shorter the braking distance, the higher the braking energy. The quantity of heat flux entering the brake disc increases as the braking energy increases [11]. The main problem with braking and stopping a vehicle is the huge quantity of heat flux that enters the disc in a short amount of time, causing a rapid temperature rise. The disc material will be subjected to severe thermal stress and distortion as a result of the high temperature-rise. Overheating of the friction surface can lead to a decrease in the friction coefficient and a significant increase in the wear of both discs and pads [15]. The heat distribution in the brake disc is uneven, resulting in thermal stress that causes brake fade and cracking [10]. The main cause of disc brake failure is hot areas and thermal stress produced by temperature- rise [4]. To address the problem and to come up with a brake disc design with better temperature- rise, not only an appropriate material has critical effect. The internal shape of the brake disc also plays a great role in minimizing the temperature rise by allowing air cooling. This will allow to design a brake disc that will be able to minimize overheating and loss of braking capacity from brake disc material with good heat dissipation characteristics and appropriate disc shapes. For this reason, this research has come with the idea of investigating the thermal behavior of different enhanced ventilated brake disc shapes by introducing holes to the current AALRT model and varying the hole diameter to investigate the effect of the diameter variation on the maximum temperature rise considering a ventilated Sic/6061Al composite as a comparison material at emergency braking, at highest gradient of AALRT and by considering air cooling so as to compare the shapes and come up with a brake disc design providing a relatively minimum temperature- rise for AALRT case study.

1.3 Objectives

1.3.1 General objective

The general objective of this paper is to investigate the thermal behavior of different train brake disc models using Sic/6061Al composite as comparison material during emergency braking and high gradient of AALRT by considering air cooling.

1.3.2 Specific objectives

- To draft and analyze different brake disc models using SOLIDWORKS13 software and conduct the thermal analysis using, the finite element analysis software, ANSYS.
- To conduct computational fluid dynamics (CFD) analysis of each brake disc model for determining their wall heat transfer coefficient using ANSYS- FLUENT.
- To conduct transient thermal and transient structural analysis of each brake disc model so as to determine their total heat flux, maximum temperature rise, deformation and von-Mises stress.

1.4 Significance of the research

The research work will help to determine the thermal loads of different brake disc shapes by considering Sic/6061Al composite material. As far as thermo- structural behavior of train brake disc is concerned, this research will be very crucial for the Ethiopian railway corporation to enhance the design of the brake system of AALRT. This study will further provide an important framework for future advanced researches that are aimed on improving the design of train brake discs for Addis Ababa LRT and other high-speed trains.

1.5 Scope of the study

In this research work, the thermal and structural behavior of four different brake disc shapes will be investigated and compared by using Sic/6061Al composite brake disc material.3D model of the brake discs will be constructed using SOLIDWORKS13 software. The fluid flow, transient thermal and transient structural simulation of each brake disc model will be done by the finite element method using ANSYS software so as to show and compare the total heat flux, the maximum temperature rise, deformation and von-Mises stress of the disc models by considering air cooling and emergency braking at the highest gradient of AALRT.

1.6 Limitations

The effects of fatigue load, wear, vibration, humidity and tribology are not taken into consideration throughout the analysis. The only heat transfer mechanism taken into account in this paper is convection. The effect of heat transfer by conduction and radiation are neglected due to their relatively little effect.

1.7 Organization of the thesis

Step 1: Literature review and data collection: All necessary data, formulas and vehicle specifications are collected by browsing different published articles, journals and scientific researches. The modeling dimensions of the brake disc are also collected by visiting AALRT main office at ‘Kaliti’ Addis Ababa

Step 2: Analytical calculation

The braking force and braking energy of the vehicle are calculated to determine the amount of heat flux entering the disc which in turn is used as a boundary condition for the thermal analysis. The pressure applied on the disc and rotational velocity are also determined to be used as a boundary condition for the structural analysis. All the necessary input parameters are calculated by considering emergency braking conditions and higher gradient of AALRT.

Step 3: CAD modelling

The four different brake disc models are designed using a CAD software called SOLIDWORKS13 and imported to ANSYS workbench as IGS files.

Step 4: CFD analysis

The CFD simulation of each disc model is conducted in ANSYS- FLUENT database so as to determine the average convective heat transfer coefficient of each model and to be used as a boundary condition in the thermal analysis.

Step 5: Thermal and structural simulation

First, the thermal analysis of each disc models is conducted to determine the maximum temperature rise and total heat flux using transient thermal database in ANSYS workbench by applying the average convective heat transfer coefficient (from step 4) and heat flux entering the disc (from step 2) as boundary conditions. Then, the structural analysis of each disc model is simulated in transient structural database by applying pressure, rotational velocity, deceleration and fixed supports as boundary conditions so as to determine the von-mises stress and deformation of each disc model. Sic/6061Al material has been used for the thermal as well as the structural analysis of each brake disc model.

Step 6: Result and discussion: The results of each disc model are discussed and compared with each other to recommend the best model for AALRT.

Step 7: Conclusion, recommendation and future work

Based on the obtained results, conclusions and recommendations are explained and possible future works are suggested.

CHAPTER TWO

LITERATURE REVIEW

Thermal and structural analysis of train brake disc is very crucial to predict the thermal and structural behavior of the disc.

Fitsum Getaneh [3] investigated thermal and structural behavior of Sic/6061Al during long term braking for keeping a constant speed and braking to a standstill afterwards for AALRT. He looked into a suitable composite material that was lighter than cast iron and had good young's modulus, yield strength, and density qualities. He used a single ventilated brake model for his investigation. After 16 seconds and 38 seconds, the highest temperature rise for a horizontal track and a downhill track was determined to be 349.48°C and 431.31°C, respectively. His analysis' maximum stress was similarly found to be 3.704MPa. He came to the conclusion that the Sic/6061Al alloy composite material may effectively improve the brake disc's thermal and mechanical performance. His research, however, did not take into account emergency braking situations and the high AALRT inclination. Furthermore, no mesh independence test had been carried out for the CFD analysis.

Nong et al. [15] proposed and analyzed a ventilated brake disc made of SiC3D/Al alloy co-continuous composites (SiC3D/Al, 3D denotes three-dimensional network structure).The researchers investigated the numerical thermal and thermal stress of the brake disc during emergency braking at a speed of 350 km/h utilizing a high temperature and low wear Sic foam material filled with a metal alloy using CFD and finite element methods. At 30s of braking phase, maximum thermal stress was found to be 186.44 MPa (lower than the disc material's permissible stress). However, the hottest temperature recorded was 471.08°C. The results revealed that the temperature was distributed equally over the brake disc, which is due to the disc material's strong thermal conductivity and great cooling ability. the brake disc can match the requirements of high-speed trains running at 350 km/h, according to the authors.

Wang et al.[4] investigated the temperature variation of the railway vehicle ventilated brake disc at high-speed retardation conditions by using tests, numerical simulation, and finite element analysis with forged steel. The maximum temperature variation was predicted using a one-dimensional heat conduction equation. The maximum temperature evolution and temperature distribution of the disc were simulated using the finite element method at a braking time of 45 seconds. The simulation's maximum temperature and effective von-Mises stress were 400°C and 485MPa, respectively. According to the findings, the temperature distribution on the friction surface first experienced the formation of a hot ring, followed by the hot ring's expansion and endurance. The disc's maximum temperature rose significantly at first, then remained steady before gradually decreasing. The disc's immediate frictional coefficient and thermal stress distribution fluctuated as a result of the temperature change, resulting in thermal damage.

Fisher, J. P. [16] conducted CFD analysis of Class 8 semi-trailers to calculate the average heat transfer coefficients for the wheel-end components and to represent the airflow under the semi-trailer with and without side skirts. In his simulation, transient FEAs were built to model the temperature distribution vs time in a disc brake wheel-end and The FEA was manually integrated with the CFD analysis. In an iterative process, heat transfer coefficients from CFD were sent to FEA, and temperature from FEA was transferred to CFD. His findings show that the methodology can accurately forecast wheel-end temperatures based on convection heat transfer from the airflow surrounding the brake and the maximum rotor temperature value was found to be 729.1°C. Convection heat transmission from the wheel-end assembly is mostly influenced by the wheels around the wheel-end assembly. This literature has massive importance on understanding the CFD analysis concept.

Jiang et al.[17] analyzed a conceptual perspective of three-dimensional transient temperature distribution for Al alloy brake discs with Al2O3-SiC (3D)/Al alloy by developing and establishing a wear-resistant surface layer. The authors suggested transient temperature analysis by finite element (FE) and CFD models of ventilated brake discs in 3D dimensions. They also looked at the effects of initial braking velocity (IBV) and the thickness of the composite material on the maximum temperature rise under various emergency braking conditions, as well as the brake discs' internal temperature gradient and transient temperature variation regularities. The reduced temperature and stress were evenly dispersed across the wear-resisting region, which was influenced by the Al alloy brake disc's superior conductivity and cooling characteristics, according to their findings. The highest temperature rise was unaffected by the wear-resistant layer's thickness. As the IBV climbed, the maximum temperature rise and stress of the brake discs rose, reaching about 517°C and 192 MPa at an IBV of 97 m/s, respectively. In this research work the only used form of energy was the kinetic energy. effect of potential energy at high gradients has not been shown.

Lan et al. [18] evaluated the thermal and stress evaluations of a Sic/Al ventilated brake disk during emergency braking at a speed of 300 km/h while considering airflow cooling by using CFD and finite element methods. The results indicated that using airflow cooling to produce larger convection coefficients reduces not only the maximum temperature in the braking system, but also the thermal gradients, because heat is removed faster from hotter portions of the disk. The greatest temperature after emergency braking in this investigation was 461°C without airflow cooling and 359°C with airflow cooling, respectively. Without and with airflow cooling, the corresponding stress was 269MPa and 164MPa, respectively. The largest temperature rise measured in the experiment was 532°C.however, the effect of potential energy has not been considered.

Kishore & Vineesh [19] investigated the temperature rise of a ventilated brake disc during the braking phase of 'Indian linke hofmann' coach by considering the kinetic energy dissipation to evaluate heat dissipation. Temperature rise was estimated using a FEA model by assuming uniform and axis-symmetric heat distribution. According to their findings, a maximum temperature- rise of 214 °C was recorded at a speed of 160 kmph and 38s for a disc model having 46 vanes. Their analysis revealed that maximum temperature appears to increase linearly with braking speed at various braking speeds.

2.1 Literature summary

Almost all the related literatures have given a reasonable insight of the utilization of CFD and finite element analysis for a train brake disc. Some of the related literatures have analized the combination of a ventilated geometry and Sic/6061Al alloy composite to reduce the maximum temperature-rise of a brake disc. However, most of them didn't take into account the effects of high gradient and potential energy on the rise of heat flux and temperature. Additionally, the Sic/6061Al alloy composite is suggested to be utilized as a brake disc material for AALRT without analyzing its behavior at highest gradient and emergency braking conditions of AALRT. The intention of this research work is to come up with a way of reducing the maximum temperature-rise of the AALRT brake disc by analyzing and comparing different enhanced ventilated brake disc models(by introducing a surface drill to the current AALRT disc model and by varying the diameter of the holes) that allow more ventilation and by, additionally, using the Sic/6061Al alloy composite as a comparison material thereby to analyze its effects at highest gradient and emergency braking conditions of AALRT. The most related literatures (in terms of CFD, thermal, finite element analysis, and the brake disc material used) are summarized in table 2.1.

Table 2. 1: summary of related literatures

No.	Title	Qualitative output	Quantitative out put		Reference number
			Maximum temperature	Maximum von-mises stress	
			(℃)	MPa	
1	Thermo-Mechanical Coupling Analyses for Al Alloy Brake Discs with Al2O3-SiC(3D)/Al Alloy Composite Wear-Resisting Surface Layer for High-Speed Trains.	Showed that maximum temperature could be reduced by reducing the initial braking velocity.	517	192	[17]
2	Numerical analysis of novel SiC3D/Al alloy co-continuous composites ventilated brake disc.	The Sic/Al composite material can meet the requirement of high- speed trains at 350 km/h.	471.08	186.44	[15]

3	Thermal analysis for brake disks of Sic/6061 Al alloy co-continuous composite for CRH3 during emergency braking considering airflow cooling.	Showed that the Sic/6061 Al composite decreases maximum temperature- rise and naturally minimizes thermal gradients.	359	164	[18]
4	Thermal and mechanical analysis of AALRT brake disc from composite material	Showed that the Sic/6061 Al could be used as brake disc material for AALRT.	431.31	3.704	[3]
5	Temperature evolution in disc brakes during braking of train using finite element analysis	Peak temperature varies linearly with braking speed	214	Not analyzed	[19]

This paper has conducted thermal analysis of four different brake disc models by introducing a surface drill to the current AALRT disc model and by varying the diameter of the holes. The analysis is conducted by considering air cooling,emergency braking conditions and highest gradient of AALRT. Additionaly, the structure of the models is also investigated and the Sic/6061Al is used as a comparison material.

CHAPTER THREE

METHODOLOGY

The methodology of the research work is summarized in the figure below.

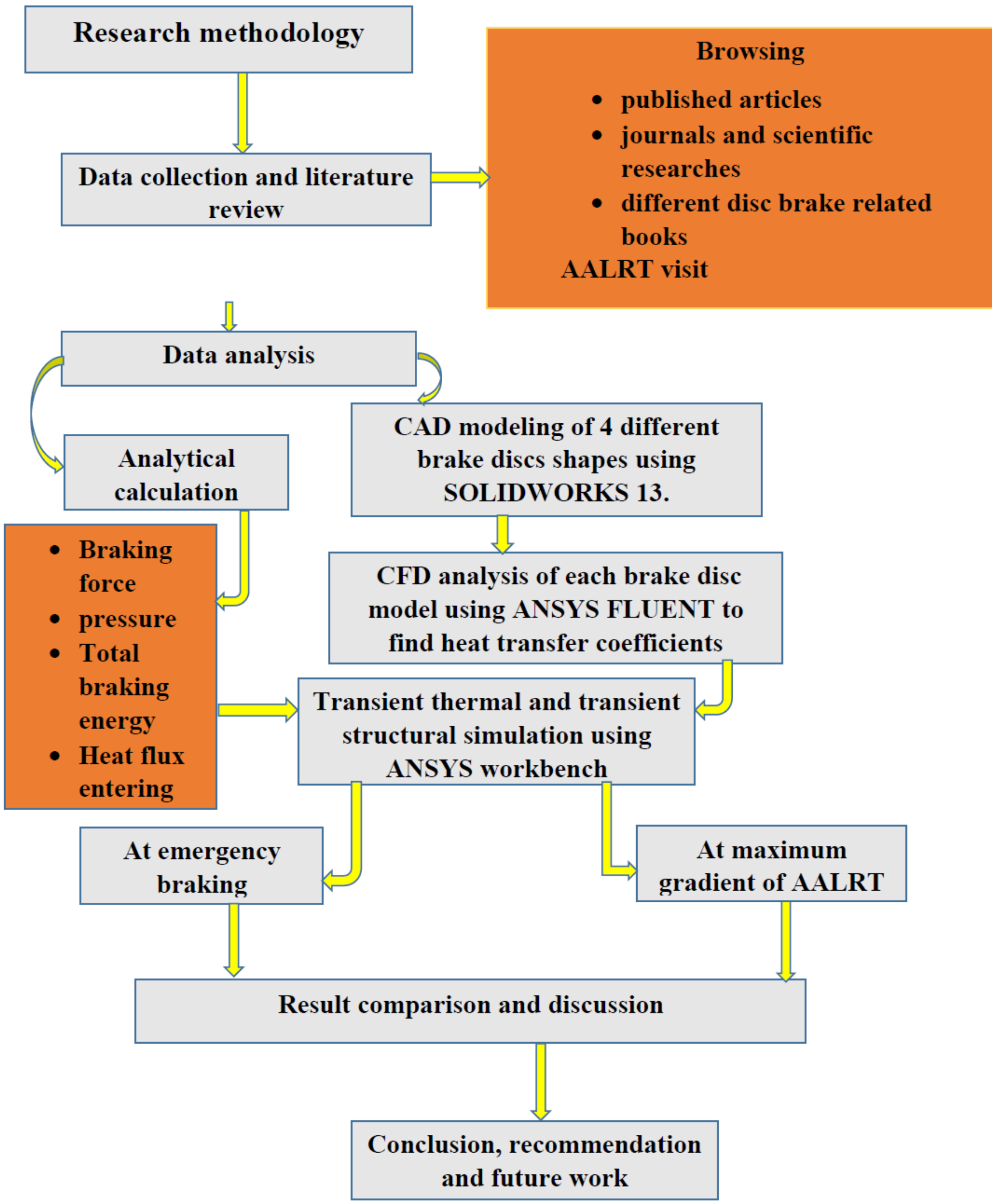


Figure 3. 1: Research methodology flow chart

3.1 Analytical calculation

3.1.1 Vehicle braking energy

The kinetic and potential energy of vehicles is turned into thermal energy during braking by frictional heating between the friction pair's surfaces. Heat due to friction is generated between the brake disc and brake pad interfaces [20].

The available energy of the vehicle that must be converted into frictional heat must be correctly determined during the braking operation. This research addresses the rail vehicle's kinetic as well as potential (when running on gradients) energy that should be dissipated by the braking operation. Rail vehicles have significant rotating masses. As a result, the contribution of rotating kinetic energy is considered.

The initial kinetic energy of the vehicle is determined by the sum of total translational and rotational energy of the vehicle.

$$k_e = \frac{1}{2} M {v_o}^2 + \frac{1}{2} I {w_o}^2 \tag{3.1}$$

$$w_o = \frac{v_o}{r_w} \tag{3.2}$$

$$k_e = \frac{1}{2} M {v_o}^2 \left(1 + \frac{I}{M {r_w}^2}\right) \tag{3.3}$$

the term $\frac{I}{M{r_w}^2}$,with a value of 0.1, is the rotational masses involved of the wheel set and discs and contributes to the 10% of the axle load of the vehicle [9].

Hence

$$k_e = \frac{1}{2} M {v_o}^2 (1 + 0.1) \tag{3.4}$$

$$k_e = \frac{1}{2} M {v_o}^2 (1.1) \tag{3.5}$$

Where:

$ke =$ vehicle intial kinetic energy

$\mathrm{I} =$ moment of inertia of rotating parts

$\mathrm{M} =$ vehicle mass

$\mathrm{vo} = \text{vehicle initial velocity}$

$\mathrm{wo} = \text{intial angular velocity of vehicle wheel}$

$\mathrm{rw} = \text{wheel radias}$

The Potential energy of the vehicle is also given by:

$$P_e = Mgsin\alpha \tag{3.6}$$

Where:

Pe=potential energy due to track gradient

M= vehicle mass

g=gravitational acceleration (9.8 m/s^2)

α =angle of inclination

In this paper the maximum gradient of AALRT, with a maximum angle of inclination 3.15^o is considered for the analysis.

The total vehicle braking energy is therefore determined by summing up the kinetic energy (including the rotational masses involved) and the potential energy of the vehicle. i.e.,

$$E_B = k_e + P_e \tag{3.7}$$

$$E_B = \frac{1}{2} M{v_o}^2 (1.1) + Mgsin\alpha \tag{3.8}$$

As long as the rail vehicle totally has 8 brake discs, out of which 4 are located at the motor car (one disc per axle) and the other 4 discs are located at the middle car (2 disc per axle) the final expression of the braking energy should be divided by 8.

$$E_B = \frac{\frac{1}{2} M{v_o}^2 (1.1) + Mgsin\alpha}{8} \tag{3.9}$$

3.1.2 Heat dissipation by convection

In a braking action, the temperature rise of a brake disc is determined by the linear momentum of the vehicle (vehicle mass and velocity), as well as the deceleration rate and fiction coefficient. For short braking and relatively medium deceleration, almost all the generated heat energy is absorbed by the friction material and the disc rotor [13]. The rotor geometry and vehicle velocity influence the airflow, which is exceedingly unstable and turbulent. Because of the pattern of the flow in and around the brake rotor, only one Reynolds number is insufficient to accurately represent the flow in this area. On the rotor's face, the flow may be laminar, but turbulent within and exiting the fins. There has to be enough amount of heat dissipation to achieve continues and reliable braking. Heat dissipation is the rate at which the heat accumulated at the surface of the disc is frequently removed and cooled. The physical phenomenon of heat transfer by moving fluids that convey heat energy from one location to another is known as convection (fluid can be air or liquid). Out of the three modes of heat transfer (convection, radiation and conduction) for disc brake cooling, convection contributes the most to overall heat loss[21].

The rate of heat dissipation by convection can be expressed by:

$$Q = Ah(T_s - T_{\infty}) \tag{3.10}$$

Where:

A=surface area of the disc

h= rate of heat transfer by convection ($W/m^2\,K$)

T_s =disc surface temperature(K)

T_∞ =ambient temperature(K)

Q=heat transfer rate(watts)

The rate of heat dissipation can be increased by increasing the convective heat transfer coefficient and the surface area of the disc or by minimizing the surface temperature. Both the heat transfer coefficient and the surface area are design parameters which can be increased by different methods. Whereas, the surface temperature is a transient parameter and should be reduced as much as possible.

3.1.3 Heat flux entering the disc

The thermal study of a railway vehicle's braking system necessitates determining the amount of heat created by friction, as well as how this energy is distributed between the disc and the braking pads. It is considered that the braking energy is changed into heat and transferred to the disc and pad approximately 90% and 10% respectively. Therefore, the proportion of the amount of heat energy entering the disc is 90% percent of the total. Due to the complexity of the analysis, the effect of the pads is represented by the amount of entering heat flux[9]. Therefore, the rate at which disc heat energy is generated is given by the following formula:

$$Q(t) = \gamma(F_{disc})V_{disc} \tag{3.11}$$

$$Q(t) = \gamma(F_{disc})\frac{r_d}{r_w}(vo - at) \tag{3.12}$$

The force applied on the disc can also be calculated by the following formula:

$$F_{disc} = \frac{E_B}{2\frac{r_d}{r_w}\left(vot_b - \frac{1}{2}at_b{}^2\right)} \tag{3.13}$$

Where:

$tb = braking\ time$, which is given by:

$$tb = \frac{vo}{a} \tag{3.14}$$

$Q(t)$ = rate of heat generation in watt

vo = vehicle initial velocity

γ = proportion of heat energy entering the disc(0.9)

rd = effective radius of the disc

rw = wheel radius

a = vehicle decelaration in m/s^2

t = time in seconds

Fdisc = force applied on the disc

The maximum heat flux entering the disc is given by the ratio of the rate of heat transfer and surface area of the disc.

$$heat\ flux = \frac{Q(t)}{A_d} = \frac{(0.9)(F_{disc})\frac{r_d}{r_w}(vo - at)}{A_d} \qquad (3.15)$$

where A_d= the brake disc area

3.1.3 Pressure of the disc

The pressure applied to the brake disc can be calculated using the following expression.

$$p = \frac{F_{disc}}{2\mu A_d} \qquad (3.16)$$

Where μ=the coefficient of friction between disc and pad.

A_d= brake disc area which is given by:

$$A_d = 2\pi({r_o}^2 - {r_i}^2) \qquad (3.17)$$

Fdisc = force applied on the disc

Table 3. 1: AALRT analytical parameters [9,21]

Parameter	value
Rail Vehicle mass(M)	63020kg
Effective disc radius (average of inner and outer radius) (r_d)	0.13725m
Emergency braking Deceleration (a)	2 m/s^2
Emergency braking time (t_b)	9.72 s
Outer disc radius (r_o)	180 mm
Inner disc radius (r_i)	91.5 mm
Initial velocity (vo)	19.44 m/s
Disc/pad friction coefficient (μ)	0.32
Wheel radius (r_w)	0.33 m
Area of the disc (A_d)	0.151m^2
angle of inclination (α)	3.15^o

Using table 3.1, all required parameters that will be used for the next CFD, thermal and stress analysis are calculated and listed in table 3.2.

Table 3. 2: List of values of calculated parameters

Analyzed parameter	Calculated value	Equation number used
Angular velocity	58.9 rad/s	(3.2)
Total braking energy	1,641,589.4375 J	(3.9)
Emergency braking time	9.72 s	(3.14)
Force of the disc	20,888 N	(3.13)
Pressure of the disc	0.216 MPA	(3.16)

The amount of heat flux entering the disc varies with time. The variation, starting from zero second (no brake application) up to the maximum braking time (9.72 s), is given below so that it can be used as a boundary condition for the thermal analysis.

$$heat\ flux = \frac{Q(t)}{A_d} = \frac{(151{,}996.659 - 15{,}637.516t)w}{0.151m^2}$$

Table 3. 3: Variation of heat flux with time

Time (seconds)	Heat flux ($\frac{w}{m^2}$)
0	1,006,600.4
1	903040.68
2	799480.97
3	695921.26
4	592361.55
5	488801.85
6	385242.14
7	281682.43
8	178122.72
9	74563.01
9.72	0.023

3.2 Modelling

Disc brakes have the benefits of a simple design and reliability over alternative braking technologies, which plays an essential role in the safety performance of high-speed trains a rotor, brake pads, piston, hydraulic unit, caliper, and clamping unit make up the brake system. The rotor revolves at the same speed as the train wheel. On either surface of the brake rotor, which is held in a caliper, brake pads are attached. The brake pad is pushed onto the rotor by the translation motion of a piston inside a cylinder. Hydraulic fluid runs the entire braking mechanism [22].

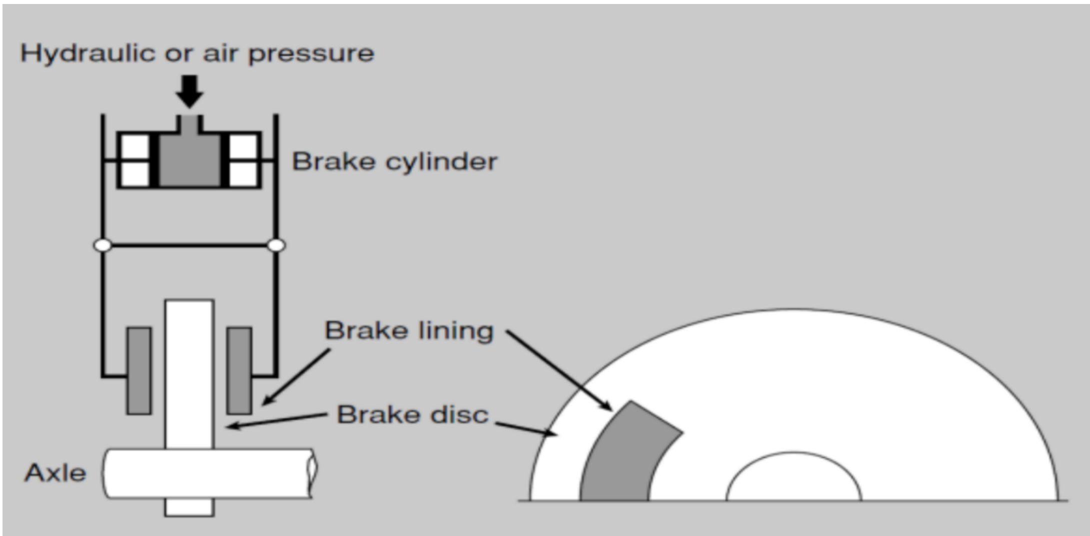


Figure 3. 2: Axle mounted disc brake assembly[5]

Solid rotors and vented rotors are the two most common types of rotors. The term “solid rotor” refers to a rotor that has no gaps between machined surfaces. As a result, air passes over the rotor’s exterior surface, cooling it. Solid rotors are compact in size and contribute to the system’s weight reduction.Internal ribs between the two contact surface are present in vented rotors. Due to impeller effect, the fins suck air into the center of the rotor and discharge it from the ends as the rotor spins [23].

Figure 3. 3: Types of brake discs [21]

Four different 3D brake disc models have been prepared for analysis using a CAD modeling software named as SOLIDWORKS13. The first model is the current AALRT brake disc that consists of straight cooling fins, 180 mm outer diameter and 91.5 mm inner diameter. The other three models include curved cooling fins, having the same outer and inner disc diameters as that of the first model, and equally spaced drilled holes. The hole diameters are 5 mm, 7.5 mm and 10 mm for the second, third and fourth model respectively. CFD and thermo-structural analysis have been conducted for each model so as to compare those four models in terms of average heat transfer coefficient, maximum temperature -rise and von-mises stress. The effect of introducing a surface drill and varying its hole diameter at different arbitrary values has been investigated so as to come up with a best model having a minimum temperature distribution when compared to the other models. The candidate models were designed according to the design dimensions and specifications of AALRT [24]. The disc model with straight fins and without a surface drill, and the disc model with curved fins and a 5 mm surface drill are shown in the figures below.

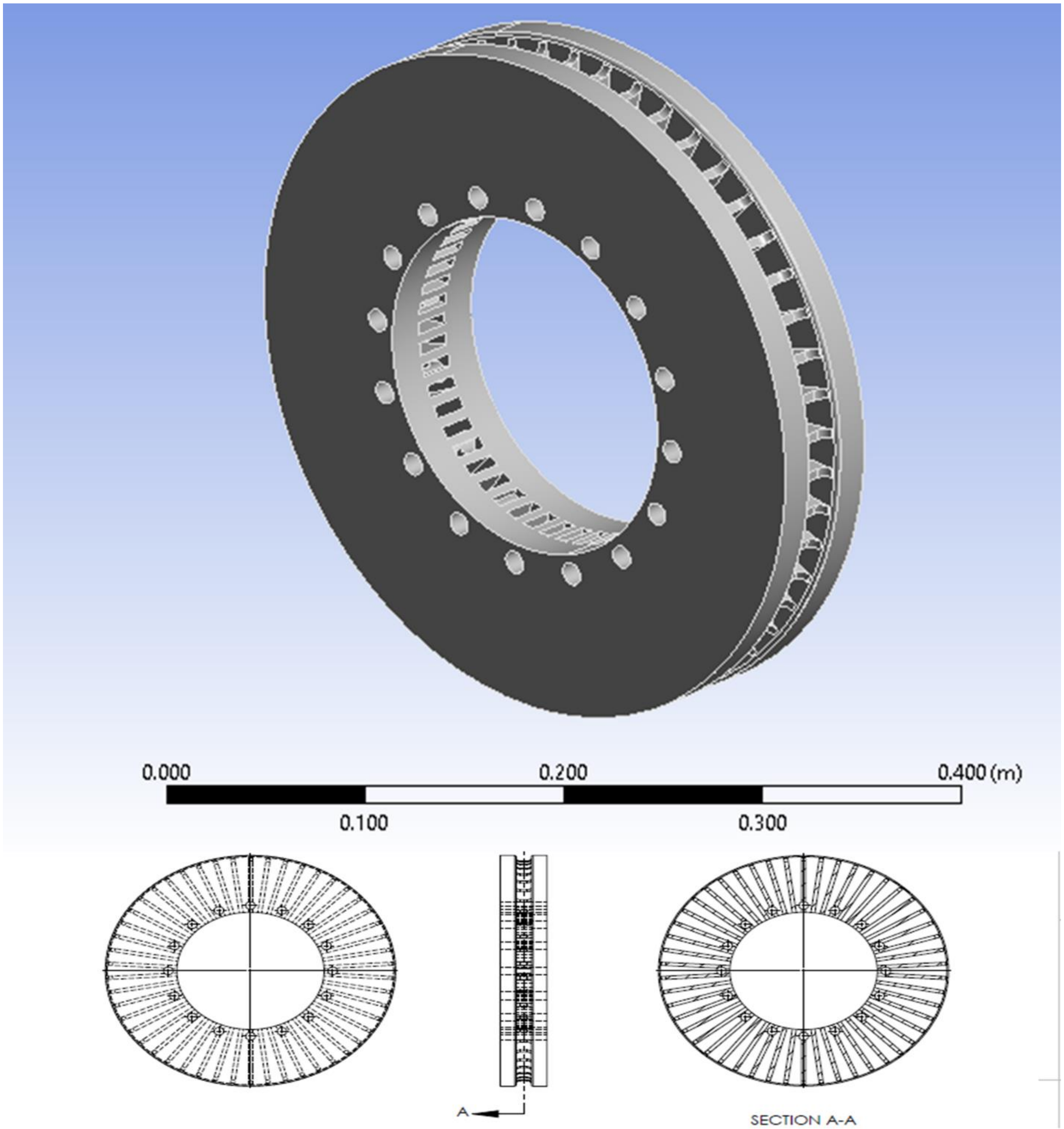


Figure 3. 4: Disc model with straight fins and without a surface drill

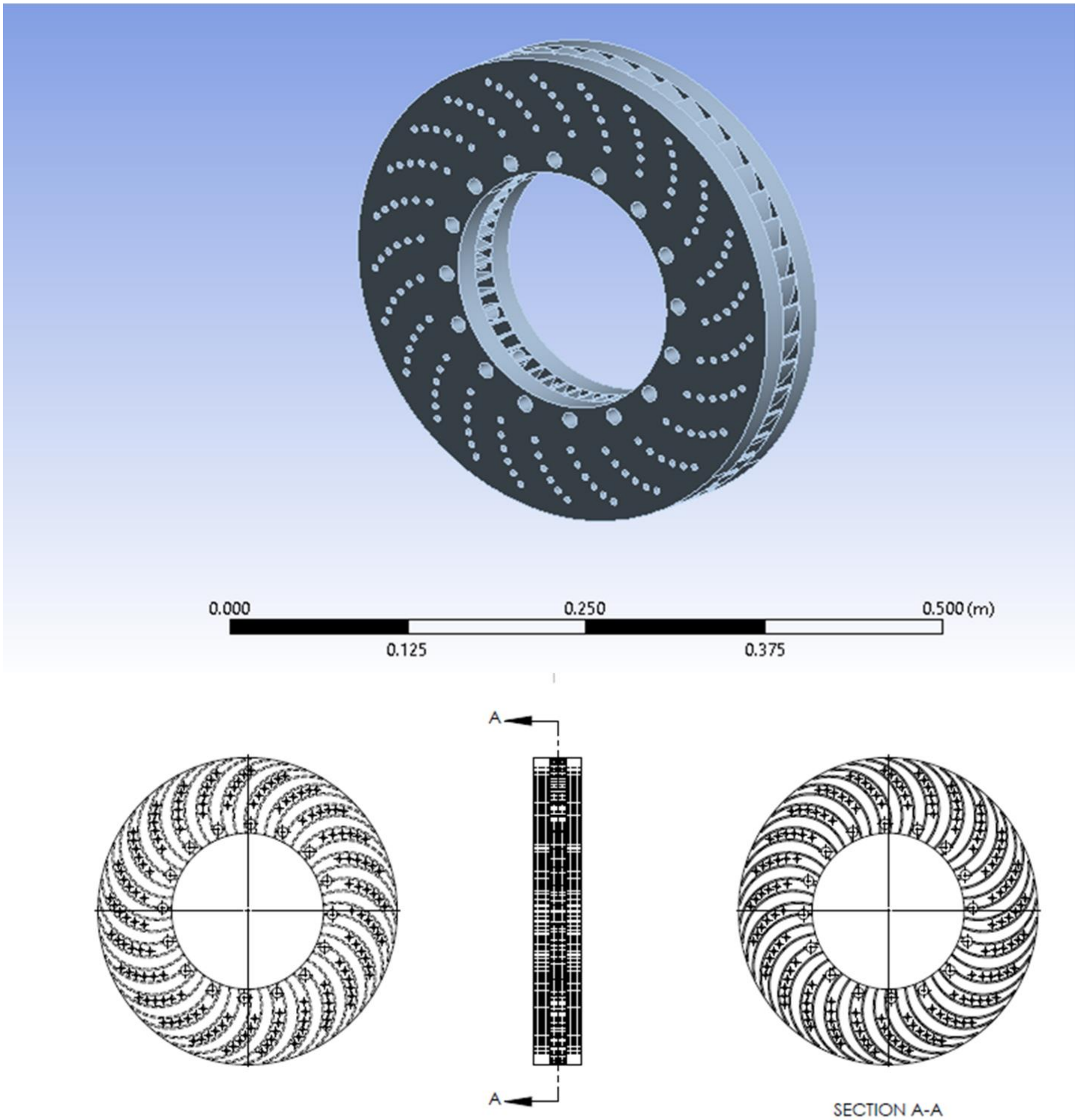


Figure 3. 5: Disc model with curved fins and a surface drill

All the four models are taken from SOLIDWORKS13 and imported into the finite element analysis software, ANSYS, in IGS format.

3.2.3 Material selection

Nowadays, different composite materials are being introduced to be used as train brake disc materials due to their higher weight reduction (when compared to gray cast iron) without losing the required thermal and mechanical properties [15] and [23]

A, Ti-6Al-4V, often known as TC4, Ti64, or ASTM Grade 5, is an alpha-beta titanium alloy with an excellent strength-to-weight ratio and robust corrosion resistance.

B, Titanium matrix composites (TMCs) are stiffer and have a higher specific strength than steel and nickel-based materials. When compared to monolithic super alloys that adjust for strength and stiffness, high temperature TMCs can offer up to a 50% weight reduction.

C, Copper metal matrix composite (AMC2): Adding other elements to aluminum metals to harden them is a frequent way; one of these approaches is to add copper to the metal to considerably enhance the material's hardness; currently, the most often used composition is 6.7 percent copper [25].

D, silicon carbide reinforced Aluminum (AMC 1): reinforced with silicon carbide have a lot of value as an engineering material, because they have a plenty of specific strength and stiffness, which is important in situations where the material is expected to last for a long time without needing to be replaced and saving weight is a priority.

Table 3. 4: Material properties of candidate composite materials [25]

Material	Specific heat capacity (kj/kg. k)	Friction coefficient	Wear rate $(\frac{X10^{-6}mm^3}{N}/m)$	Specific gravity (kg/m^3)	Comprehensive strength (MPa)
Ti-6Al-4V	0.58	0.34	246.6	4.42	1070
TMCs	0.51	0.31	8.19	4.68	1300
AMC1	0.98	0.35	3.25	2.7	406
AMC2	0.92	0.44	2.91	2.8	761

Even though the candidate materials have similar properties, the aluminum alloy composite is chosen as a comparison material because the usage of aluminum silicon carbide material can improve friction and wear qualities while also transferring heat generated on the disc friction surface. It has a good cooling ability, which accelerates air movement through the brake disc, effectively dissipates heat, and provides uniform temperature distribution. High specific stiffness, high plastic flow strength, creep resistance, and superior oxidation and corrosion resistance are among the mechanical features of Sic/Al composites [10]. In addition to lowering the maximum temperature, it also has a natural tendency to minimize thermal gradients [14]. Furthermore, using the AMC 1 as a brake disc material also provides a significant weight reduction (up to 60%) when compared to the current AALRT brake disc material (gray cast iron) [9].

Table 3. 5: Properties of Sic/6061Al alloy continuous composite material [18]

Material Properties	**Disc**
Thermal conductivity, k (W/m k)	5
Density, ρ (kg/m3)	3700
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

3.3 Analysis

Before starting the analysis, the following assumptions are made for the purpose of simplicity.

- The heat flux distribution is uniform throughout the disc surface.
- The type of flow turbulence is considered to be k-epsilon turbulence.
- The velocity of air at the inlet is assumed to be 19.44 m/s [9].
- Due to their little contribution, the Heat transfer due to radiation and conduction is neglected [13,21,3].
- The Sic/6061Al material is assumed to be isotropic and homogenous.
- The ambient temperature is considered to be 25 degrees Celsius.
- The pressure distribution on the surface of the disc is assumed to be uniform.

- No stress is available on the disc before brake application.
- The effects of wear and other loads are also neglected.
- The thermal conductivity and specific heat of the material are assumed to be constant.

3.3.1 Finite element method (FEM)

The Finite Element Method is a numerical approach for solving problems in continuum mechanics with an acceptable level of accuracy for engineers. It is a mathematical modeling technique that involves discretizing a continuous domain using building-block entities called finite elements that are connected by nodes for once and moment transmission. It is a computational approach for solving ordinary differential equations in equilibrium. Complex simulations incorporating extremely non-linear fluid flow and dynamic events can be successfully studied with ANSYS, ranging from simple linear static stress and heat transfer analysis. ANSYS is a software suite for doing finite element analysis (FEA). Finite Element Analysis (FEA) is a numerical approach for breaking down a large system into small (user-defined) parts called elements. The software creates a thorough explanation of how the system functions as a whole by implementing equations that regulate the behavior of various elements and solving them altogether. The results can then be displayed in tabular or graphical format. This form of analysis is usually used to build and optimize a system that is far too complicated to evaluate by hand. The software consists of three basic steps[9].

Step-1 the pre-processing: Converts the solution domain to finite elements and discretize it. This entails partitioning the domain into 'elements,' or sub-domains, and picking nodes on the inter-element boundaries or within the elements' interiors. Assumes the element's behavior is represented by a function. This function, known as the "shape Function," is approximate and continuous. It formulates an element's equations, Assembles the pieces to form a complete picture of the issue and applies the boundary, starting, and loading conditions.

Step-2 analysis: Depending on the type of problem, solves a collection of linear or nonlinear algebraic equations at the same time to produce nodal outcomes, such as displacement or temperature values.

Step-3 post- processing: This stage entails analyzing the nodal data to obtain additional information such as fundamental stress values, heat flows, temperature distribution, and so on.

3.3.3.2 ANSYS's General Analysis Procedure

The steps for conducting analysis with ANSYS software are outlined below.

A, Definition of geometry: Key points, lines, areas, and volumes are the four distinct geometric entities in the preprocessor. The geometric representation of the structure can be obtained by using these entities. Every entity is self-contained and has its own set of identification labels.

A, Modeling: In this section, we should define the model's geometric boundaries, set size and shape controls for the elements, and then tell the ANSYS program to construct all nodes and elements immediately.

B, Material definition: The user must define the elements' material attributes. Elasticity and Poisson's ratio are required for linear static analysis. Coefficients of thermal expansion, densities, and other factors are needed for heat transfer analysis. The material property set can assign them to the elements.

C, Mesh generation: The basic concept of finite element analysis is to examine a structure, which is made up of discrete parts called elements that are joined at a finite number of places called nodes. These elements and nodes are then subjected to loading boundary conditions. Mesh refers to a network of these pieces.

D, Finite element generation: Generating elements and nodal data takes the most time in a finite element analysis. The user can utilize the preprocessor to automatically construct nodes and elements at the same time, giving them more control over the size and number of elements.

E, Loading and boundary conditions: After the finite element model has been completed, the model must be constraint and loaded. Constraints and loads can be defined in a variety of ways by the user. All loads and limitations have been assigned. This makes it easier for the user to keep track of the load cases.

F, Solution: The problem is solved according to the problem definitions in the solution phase. The computer does all of the arduous work of forming and building matrices, and then outputs displacements and stress values. Linear static analysis, non-linear static analysis, transient dynamic analysis, and other features of ANSYS are only a few**.**

G, Post processor: It's a strong post-processing tool with dynamic color visuals that's easy to use. It comes with a number of plotting options for displaying the results of the finite element analysis. A single visual representation of the study results can often show in seconds what an engineer would take an hour to assess from a numerical output, such as a tabular form. Important characteristics of the results that can be overlooked in numerical data are highlighted.

3.3.2 CFD analysis

The utilization of CFD simulations and structural simulations utilizing the finite element model is an essential tool in product development [26]. CFD is a sub-field of fluid dynamics that uses numerical solutions of the governing equations, notably the Navier-Stokes' equations, to simulate fluid flow and heat transport. The governing partial differential equations are replaced by systems of algebraic equations, which are significantly easier to solve with computers. The turbulent closure problem of the Reynolds-averaged Navier-Stokes' equations is still an active topic of research, as is the development of simplified forms of these equations (RANS). Validation and exploration of the limitations of alternative approximations to the governing equations have been aided by experimental approaches [21]. In CFD analysis the air flow patterns surrounding the braking components are extremely complicated, and they can change dramatically depending on the undercarriage structure and component shapes. Instead of employing empirical equations, which are widely used in thermal analysis, an iterative technique is utilized to derive the average heat transfer coefficients from the measured cooling coefficients[20]. The quality of the mesh and the user's ability to use the correct modeling techniques within the solver determine the accuracy of CFD simulation results. As a result, validating the outcomes of the CFD study is critical in order to make meaningful forecasts[27].

3.3.2.1 ANSYS FLUENT-database

ANSYS FLUENT is one part of CFD analysis which is very essential to calculate different thermal related parameters like convective heat transfer coefficient. Analysis could be started by opening the fluid flow(fluent) option on the toolbox of ANSYS workbench. Once its opened, the geometry is imported or modelled in ANSYS geometry modeler. The loaded geometry should then be meshed by double clicking the cell indicating mesh. In finite element concept, meshing is the breaking down of a given geometry into finite thousands of shapes in order to precisely define the physical model of the given geometry. A named selection of the geometry boundaries should also be done at this stage by naming the inlet, outlet and walls of the model so as to simplify the analysis later in the fluent data base. After finishing the mesh, one can see the statistics and details of the mesh. Then by closing the mesh one can proceed to the ANSYS fluent to set up the CFD simulation. After fluent is launched, we can set the convenient units and improve our mesh quality.in cell called ‘setting up physics’ the material properties, physical models, flow turbulence and zone conditions of our simulation are specified. The boundary conditions at each named selection should be specified before solving the model. The boundary conditions could be velocity inlet and outlet, pressure inlet and outlet, wall heat flux or any other parameter that we want to set in our analysis. We also need to choose an initialization method to initialize the solution before solving. There are two ways of initializing our project. They are called hybrid and standard initialization. The hybrid initialization should be kept during the analysis so that the simulation will keep into account initial reference values of the fluid medium chosen. Then, the calculation could be launched by choosing an appropriate iteration. During the calculation, the progress of the analysis will be displayed using plots as it can be seen in the figure below.

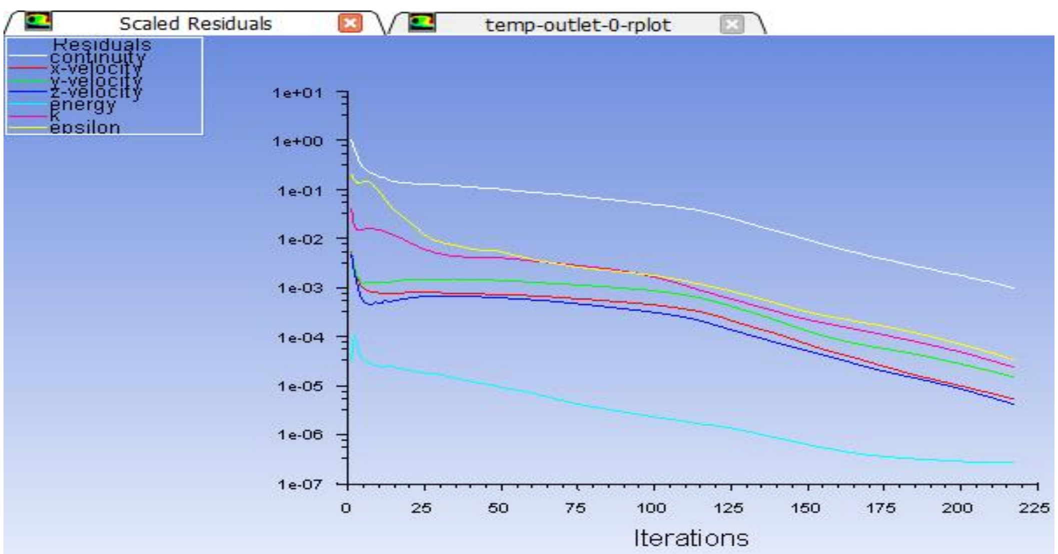


Figure 3. 6: ANSYS-FLUENT solution progress display[28].

Finally, the required results are displayed in the post processing cell. Like Velocity, pressure and convective heat transfer coefficient contours are displayed here.it is also possible to calculate the maximum, average and minimum value of the parameters.

3.3.2.2 K-epsilon turbulence

Turbulence is a condition of fluid motion that involves random and extremely disorganized three-dimensional vorticity. As a numerical simulation of the average flow characteristics in the turbulent flow domain is concerned, the turbulence model (k- €) is the most extensively used model in the field of CFD analysis. It consists of two equation models that utilizes two partial differential equations (PDEs) termed the transport equation to give a broad description of turbulence, one for turbulent kinetic energy (k) and the other for dissipation (€). For current models, the model provides a good agreement for consistency and accuracy. The conditions affecting the turbulent kinetic energy are the main focus of the model and it is numerically robust [29].

In this paper ANSYS-FLUENT analysis is made in order to determine the average convective heat transfer coefficients (HTC) of each brake disc model so that the determined HTC could be used as the one of the main boundary conditions for the thermal analysis of each brake disc model. During the analysis, the 3D models of each brake disc are inserted in to the ANSYS FLUENT database and a named selection of each model is made. It is assumed that the medium of flow is air and its inlet velocity to be 19.44 m/s .The air flow inlet, airflow outlet and walls are assigned (see figure below) for each model so as to indicate the necessary boundary conditions of the analysis.

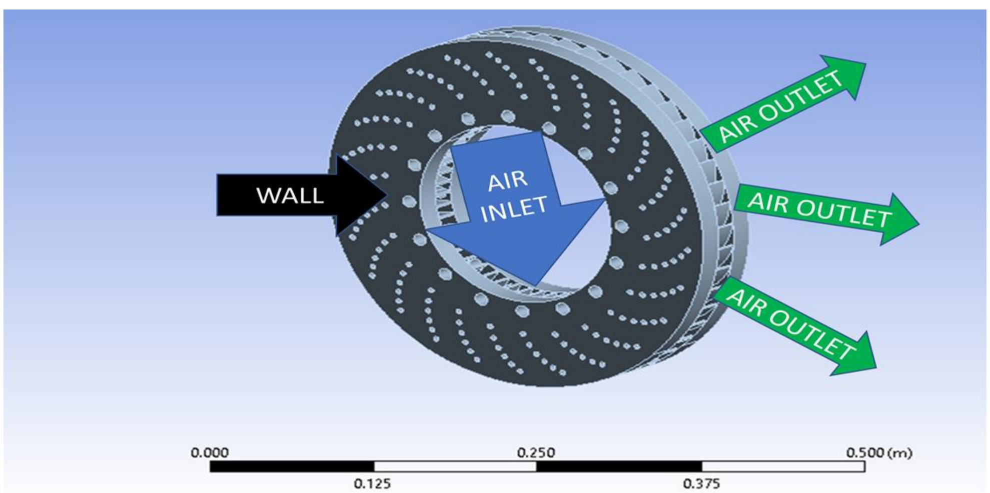


Figure 3. 7: Named selections

3.3.2.3 Meshing

The fundamental concept of finite element analysis is to look at a structure that is made up of discrete parts called elements that are connected at a finite number of nodes. The mesh size of the model is one key in producing correct conclusions, as these elements and nodes are subsequently exposed to loading boundary conditions.

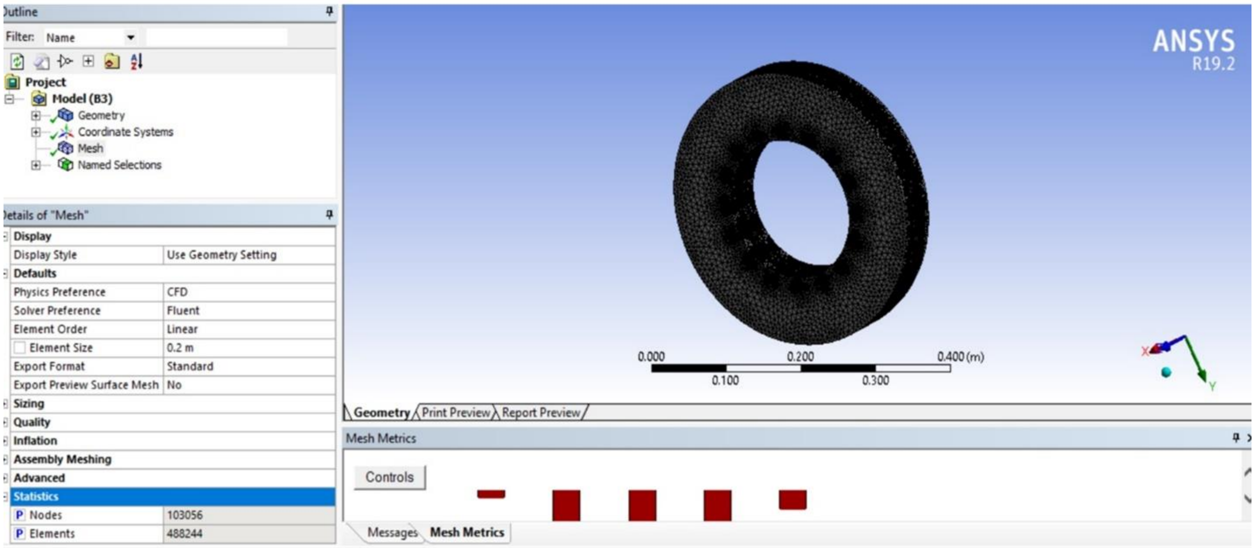


Figure 3. 8: ANSYS-FLUENT meshing

3.3.2.4 Mesh independence test

An appropriate meshing is done for each disc models in the ANSYS fluent meshing database by selecting the best mesh size using mesh independence test. Mesh independence test is used to determine a suitable mesh size of a given CFD analysis so as to show that the acquired results are due to the design parameters instead of the element mesh size. To achieve this, the value of average heat transfer coefficient (HTC) is determined for a course mesh, medium mesh and fine mesh with 200mm,150mm and 90mm of mesh size respectively. At each mesh size, different numbers of elements and nodes was obtained. The number of nodes and element at the different mesh sizes are shown in the following figures.

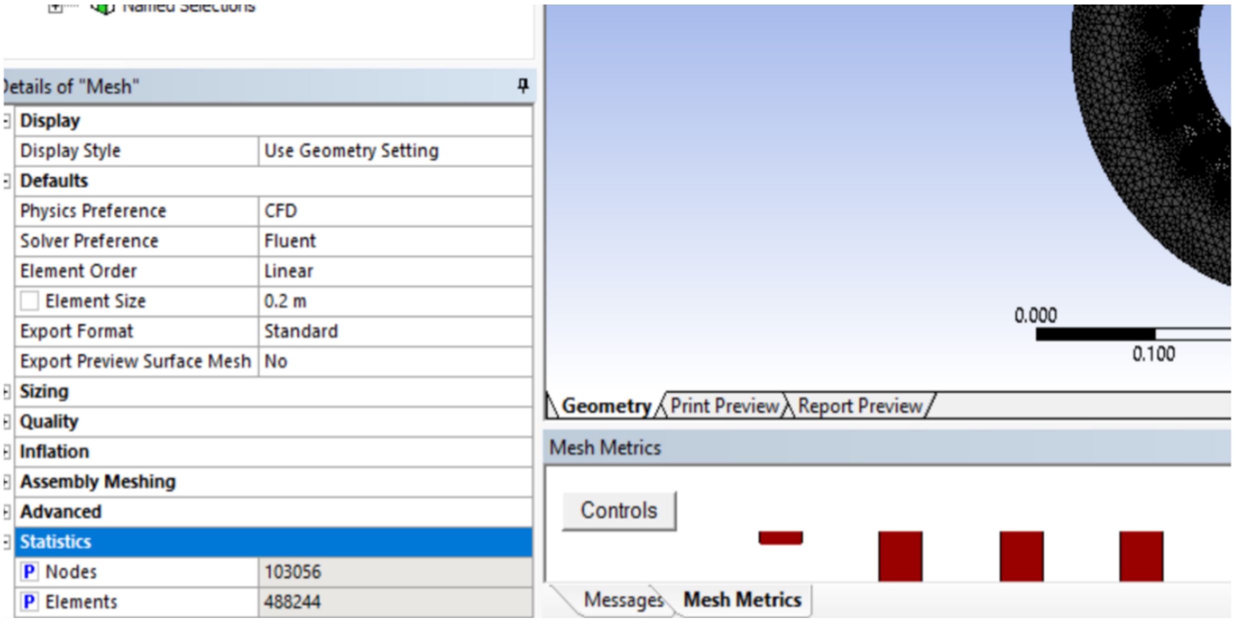


Figure 3. 9: Nodes and elements at 200mm element size

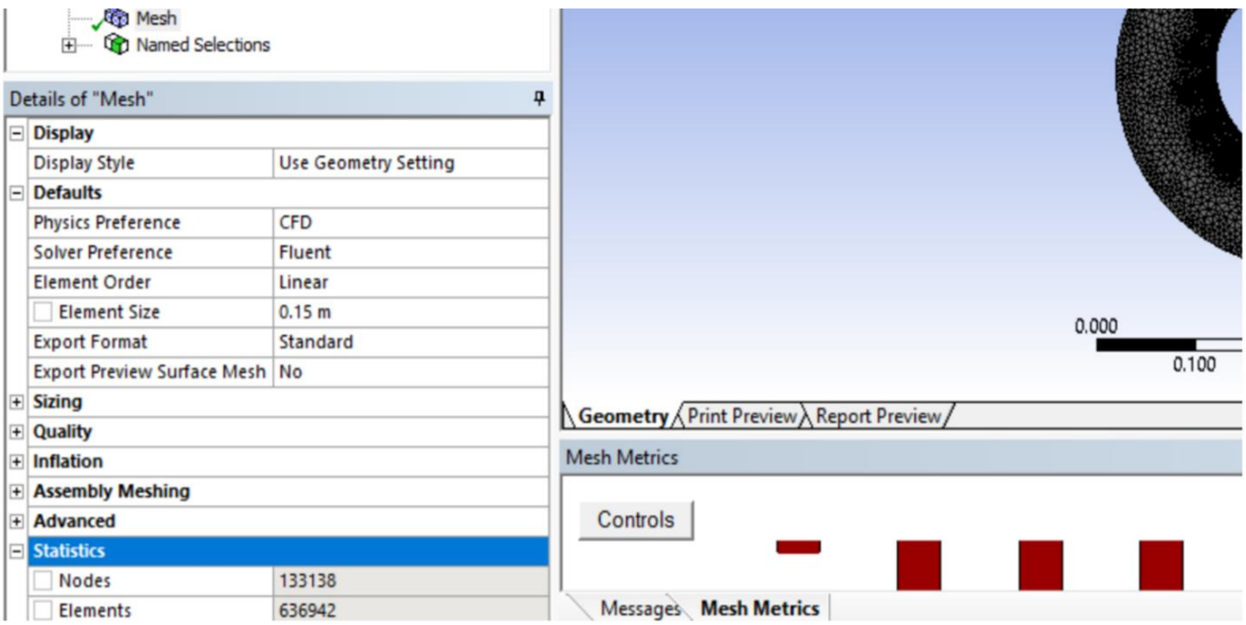


Figure 3. 10: Nodes and elements at 150 mm element size

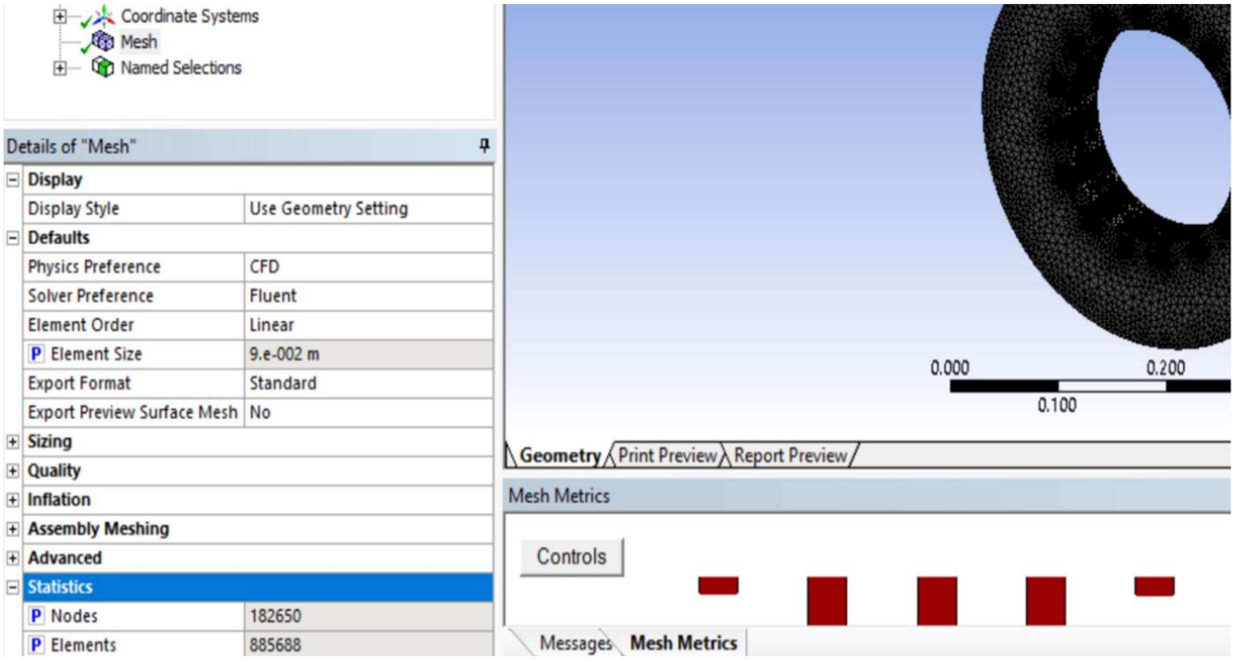


Figure 3. 11: Nodes and elements at 90mm element size

The average heat transfer coefficient values at the selected different element sizes are shown in the following figures.

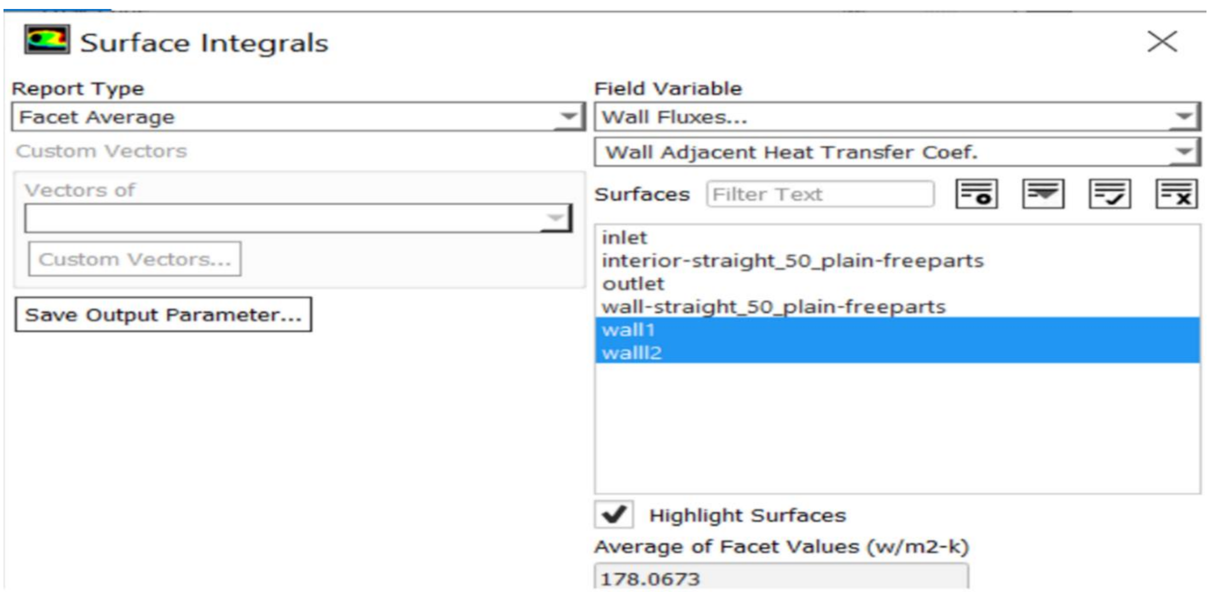


Figure 3. 12: HTC output at 200mm mesh size

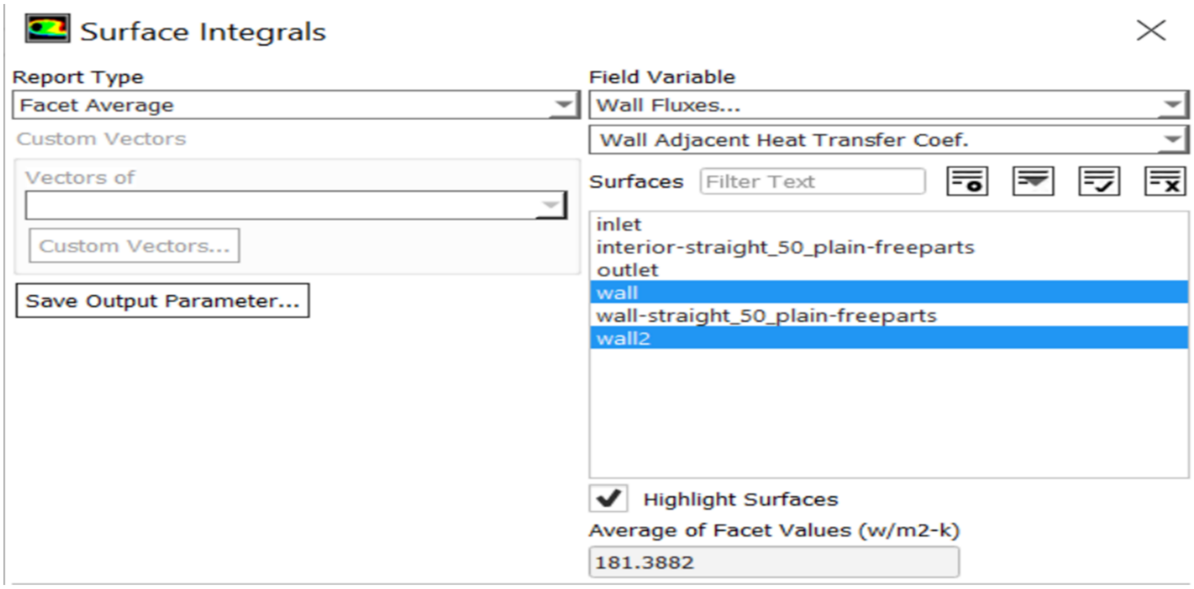


Figure 3. 13: HTC output at 150mm mesh size

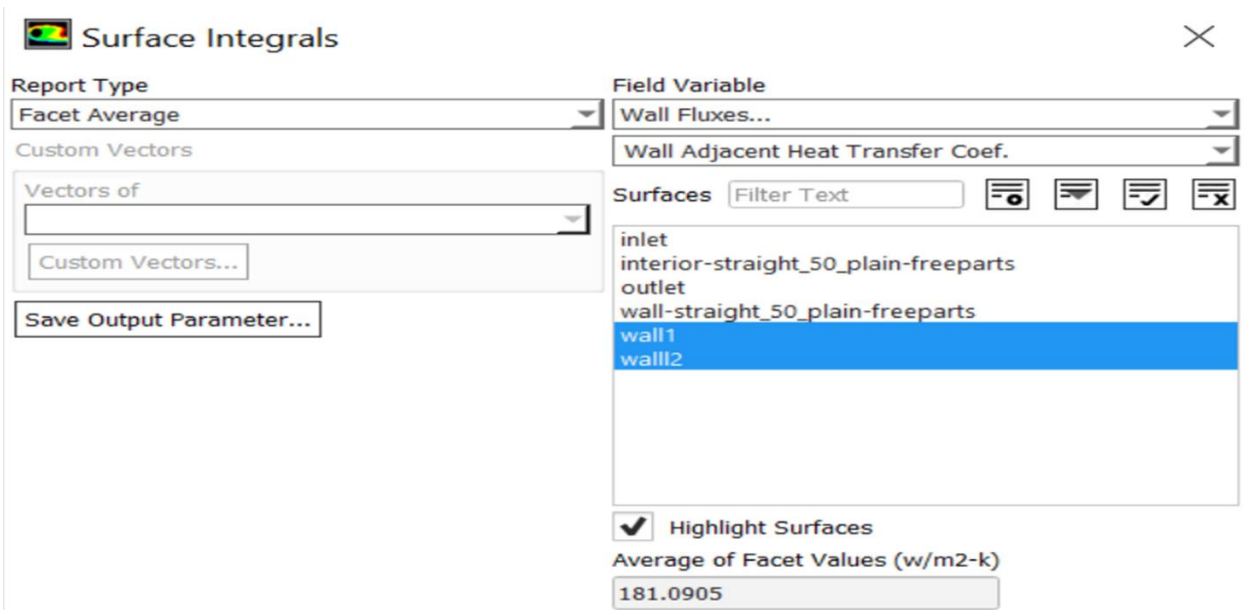


Figure 3. 14: HTC output at 90mm mesh size

The element number, node number and their respective average heat transfer coefficient values of each selected mesh element size are listed in the following table and are used to conduct the mesh impendence test of the analysis.

Table 3. 6: Mesh independence testing values

Mesh type	Mesh size(mm)	Element number	Node number	HTC ($\frac{w}{m^2\,K}$)
coarse	200	488244	103056	178.06
medium	150	636942	133138	181.38
Fine	90	885688	182650	181.09

The test was conducted according to Richardson extrapolation theory which states that for a given CFD analysis to be said mesh independent, the refinement ratios (ratio of fine mesh to medium mesh and ratio of medium mesh to course mesh) must be greater than 1.3 [30].

Refinement ratio 1=fine mesh/medium mesh=885690/636942=1.3905

Refinement ratio 2=medium mesh/coarse mesh=636942/488244=1.3045

Table 3.6 shows that the obtained convective heat transfer coefficients at different mesh sizes are quite similar and the refinement ratios are under the Richardson extrapolation theory limit. For the above reasons, the CFD analysis carried out is mesh independent.

In addition to this, the heat transfer coefficient variation with respect to the mesh element size could be observed in the figure below. As the element size varies from 50mm to 200mm no significant change is observed in the average heat transfer coefficient value.

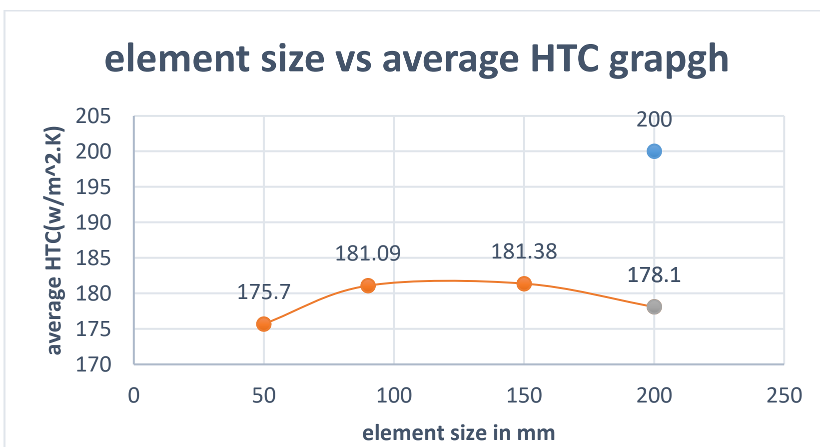


figure 3. 15: Element size vs average heat transfer coefficient graph

3.3.2.5 Boundary conditions assignment

Before proceeding into the calculation, the energy button is turned on and a k-epsilon flow turbulence is assigned. In ANSYS FLUENT, it is possible to add any material as long as its required properties are available. In addition to this, it is also possible to change the fluid medium of the simulation as per the requirement of analysis being conducted and air is chosen as the fluid flow medium. the properties of the Sic/6061Al alloy composite brake disc material are also assigned. The fluid medium and material assignment are shown in the following figures.

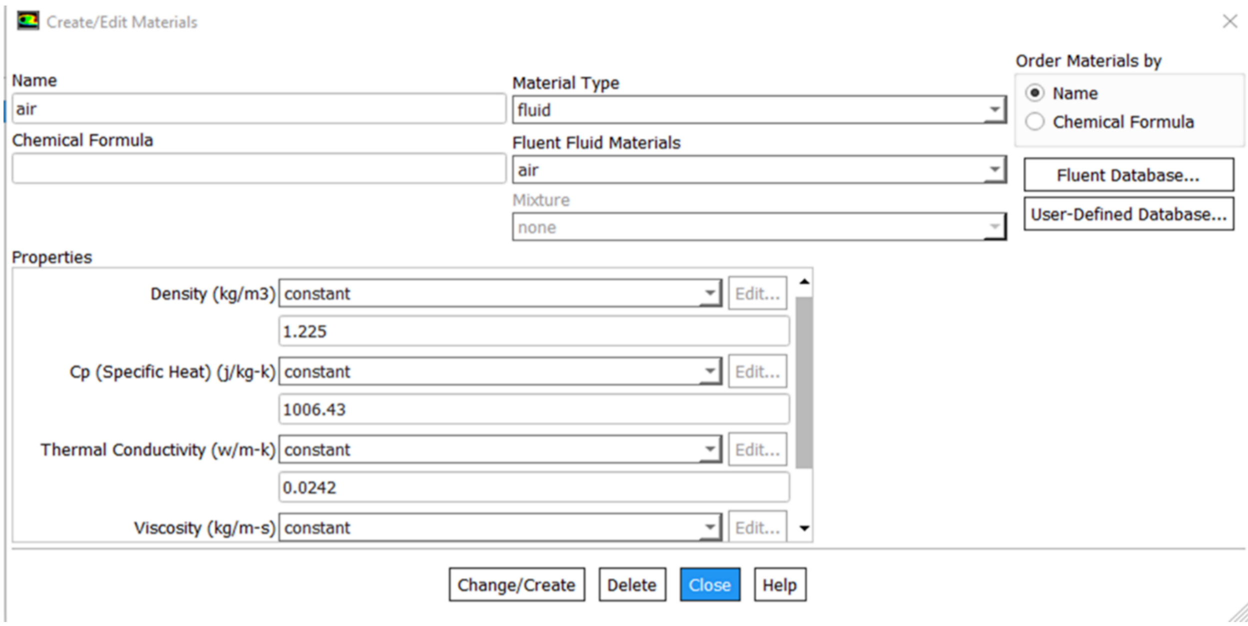


Figure 3. 16: ANSYS-FLUENT fluid medium assignment

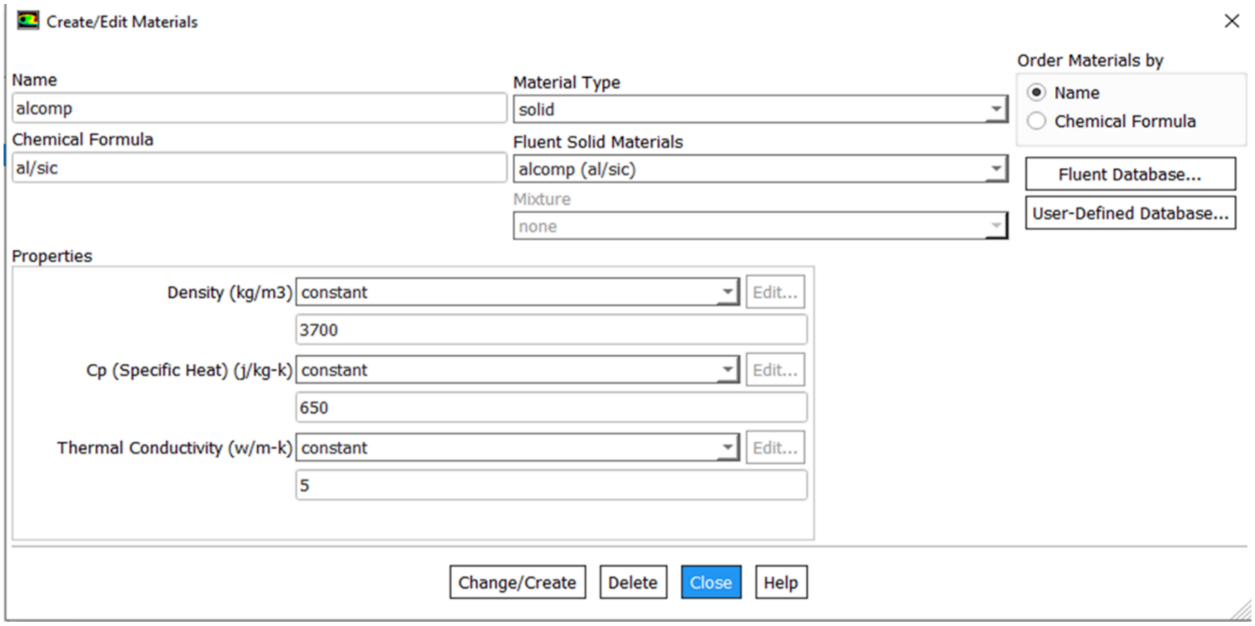


Figure 3. 17: ANSYS-FLUENT fluid material assignment

The air inlet velocity, the brake disc rotational velocity and heat flux are also assigned for each brake disc model. The initial conditions of 1 atm pressure and 25 degrees Celsius are assigned at the reference values window. A hybrid initialization is selected so that the analysis considers the initial conditions and it was finally solved by choosing the required parameter to be calculated (the wall convective heat transfer coefficient).

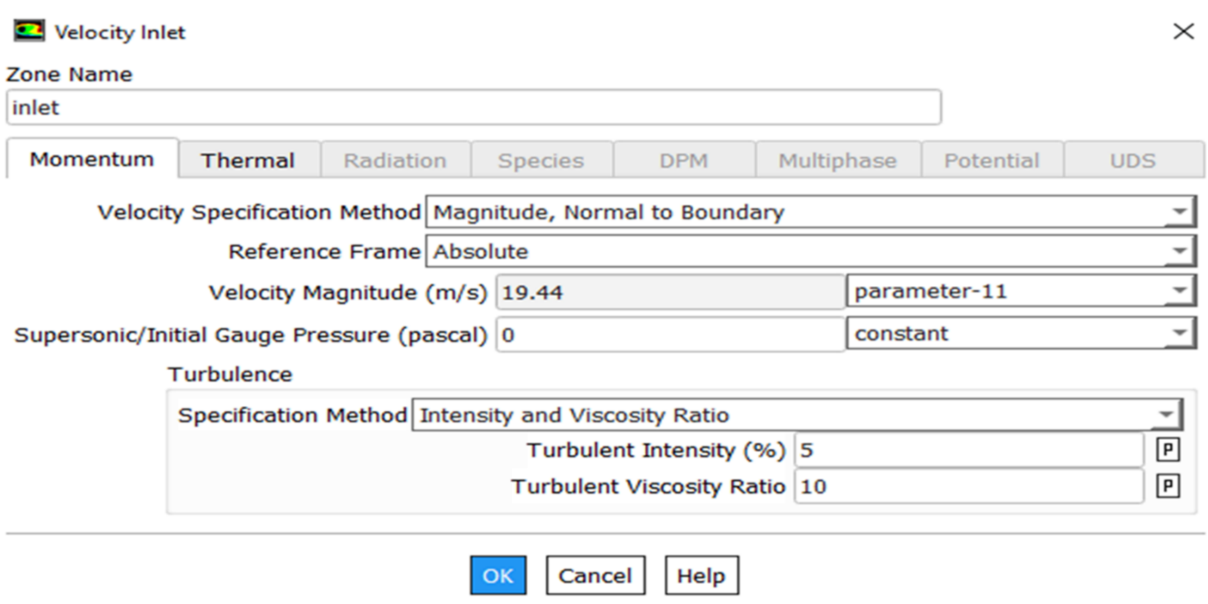
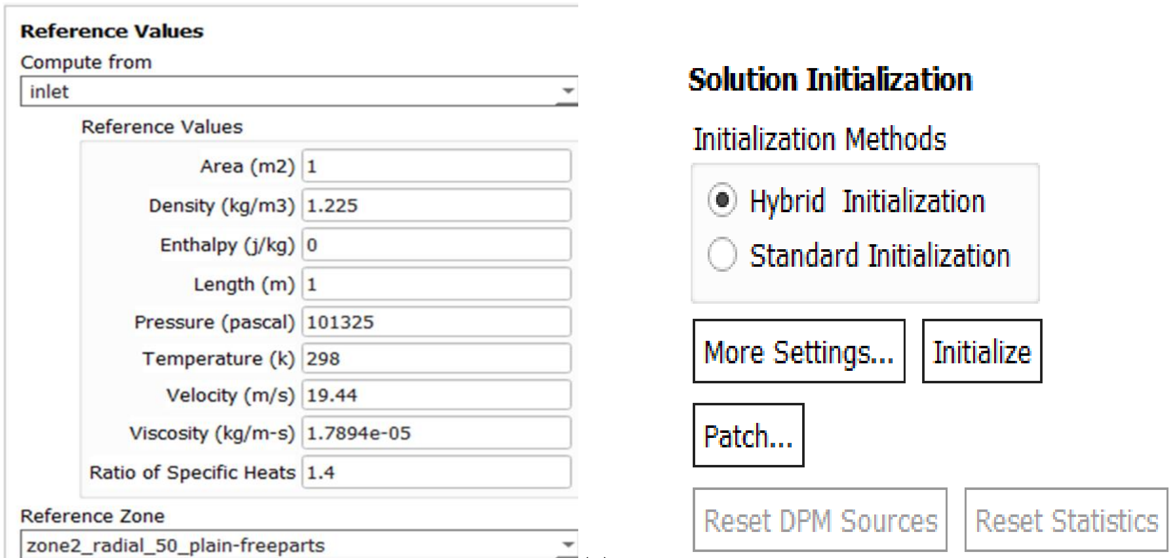


Figure 3. 18: Inlet air velocity assignment



(a) (b)

Figure 3. 19: Inlet reference values(a) and solution initialization methods(b)

After the calculation is done for each model, the following plots of results are displayed at the "results" interface of the ANSYS-FLUENT.

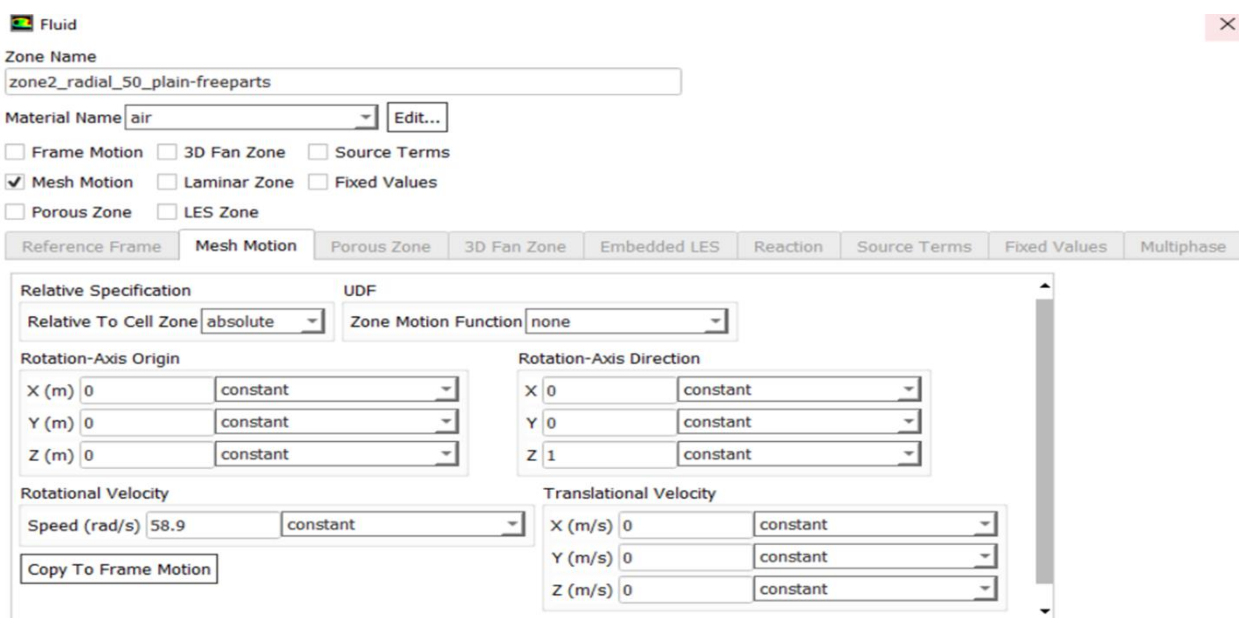


Figure 3. 20: Rotational velocity assignment

The results of average heat transfer coefficient obtained during the ANSYS-FLUENT analysis are collected and used as boundary conditions for the thermal analysis. The details of the CFD analysis results are tabulated in chapter four.

3.3.3 Thermal analysis

3.3.3.1 Meshing: a tetrahedron meshing has been selected for the analysis as shown below.

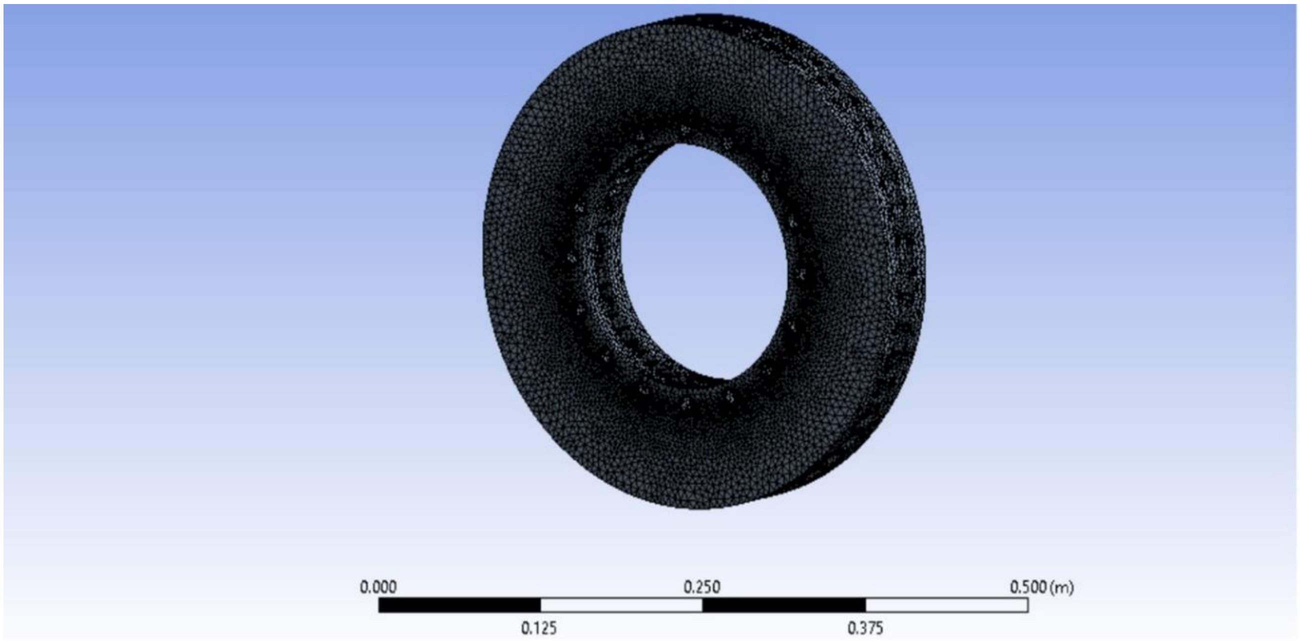


Figure 3. 21: Transient thermal tetrahedron meshing

For the purpose of comparison, a fine tetrahedron mesh can be used to examine the brake disc's thermal qualities for the fact that very fine mesh takes a long time to conduct the analysis there by increasing the engineer’s final cost[29] .

3.3.3.2 Heat flux and HTC assignment

The temperature and total heat flux distribution of the brake disc are calculated using transient thermal analysis. This type of analysis is conducted under circumstances that fluctuate with time. For this case, the heat flux applied to the brake disc is highest at the start of braking and drops to zero when the vehicle comes to a complete stop. Thus, this time variable heat flux and the convective heat transfer coefficients are used as the main boundary conditions for the temperature and total heat flux calculations.

Table 3.7: Convective heat transfer coefficient assignment

Object Name	*Convection*	*Heat Flux*
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	2 Faces	
Definition		
Type	Convection	Heat Flux
Film Coefficient	189.36 W/m²·°C (step applied)	
Ambient Temperature	25. °C (step applied)	
Convection Matrix	Program Controlled	

The variation of the heat flux input with time (starting from zero second up to the maximum emergency braking time) could be observed in the figure below. At the beginning of the braking process the amount of heat flux entering the surface of the brake disc reaches its maximum and starts to decrease and finally becomes zero when the train comes into stop.

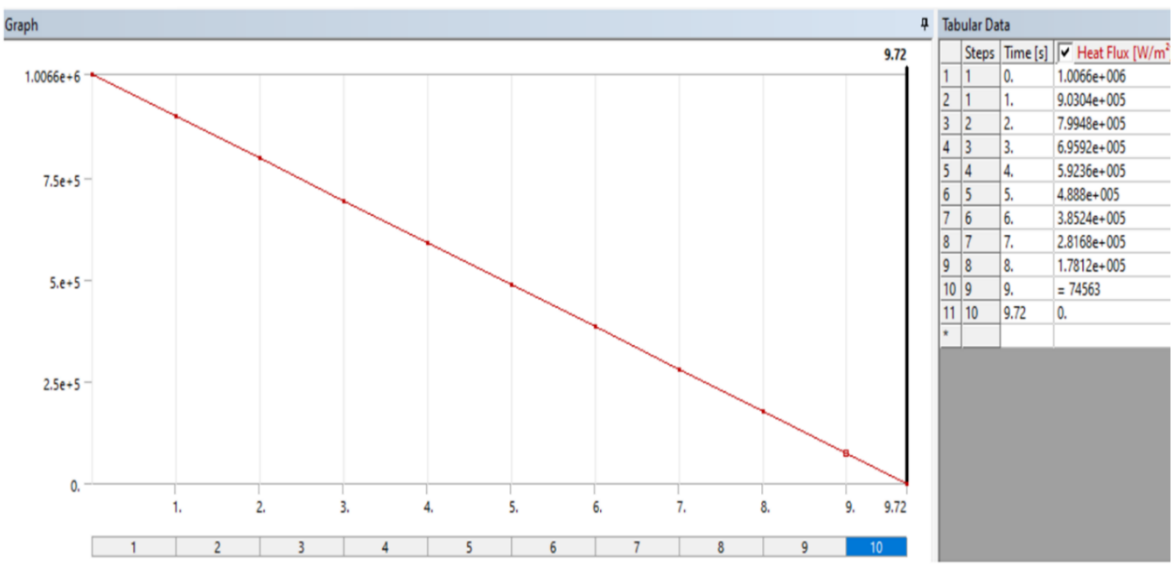


Figure 3. 22: Heat flux distribution

3.3.3.3 Material assignment

While conducting any analysis in ANSYS workbench, many different types of materials are available in the ANSYS material library. One can select the required material and apply to the type of analysis being conducted. However, some types of materials are not available in the ANSYS material library. It is possible to assign any material in ANSYS by importing the thermal and mechanical properties of the material so that it will be added to the library and could be used for further analysis.

Details of "al/sic6061 Assignment"

Nonlinear Effects	Yes
Thermal Strain Effects	Yes
Reference Temperature	By Environment
Suppressed	No
Common Material Properties	
Density	3700 kg/m³
Young's Modulus	2.54e+11 Pa
Thermal Conductivity	5 W/m·°C
Specific Heat	650 J/kg·°C
Tensile Yield Strength	3.14e+08 Pa
Tensile Ultimate Strength	4.1e+08 Pa
Nonlinear Behavior	False
Full Details	Click To View Full Details

Figure 3. 23: Material assignment

3.3.3.4 Analysis settings

During transient thermal analysis, the end of the simulation time should be set because the maximum temperature-rise highly depends on the time at which it is being analysied.at the end of the simulation, ANSYS displays the time at which the simulation is conducted together with the temperature contour. Since the goal is to determine the maximum temperature-rise at the end of the train braking time, The braking time of 9.72 seconds was set in the analysis setting as the comparison time for the thermal parameters of the brake discs.

Model (B4) > Transient Thermal (B5) > Analysis Settings

Object Name	*Analysis Settings*
State	Fully Defined
Step Controls	
Number Of Steps	10.
Current Step Number	10.
Step End Time	9.72 s
Auto Time Stepping	Program Controlled
Initial Time Step	7.2e-003 s
Minimum Time Step	7.2e-004 s
Maximum Time Step	7.2e-002 s
Time Integration	On

Figure 3. 24: Analysis settings

3.3.4 Stress analysis

The rotational velocity, rail vehicle deceleration, pressure and gravitational acceleration are used as boundary conditions during the transient structural analysis in addition to the imported thermal result through the transient thermal and transient structural coupling.

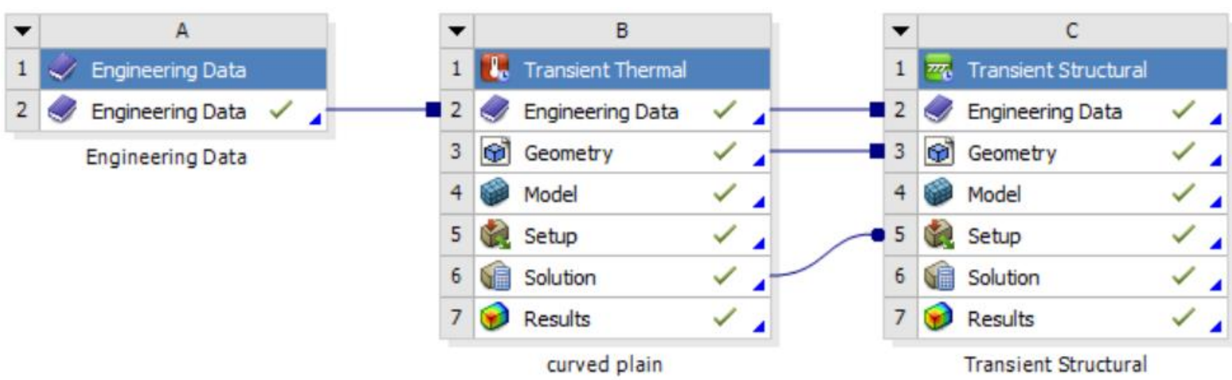


Figure 3. 25: Transient thermal and transient structural coupling

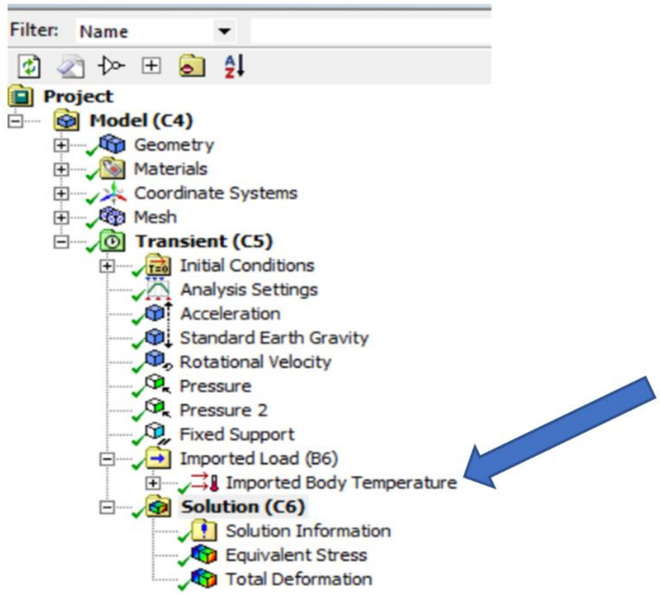


Figure 3. 26: An imported temperature from transient thermal to transient structural

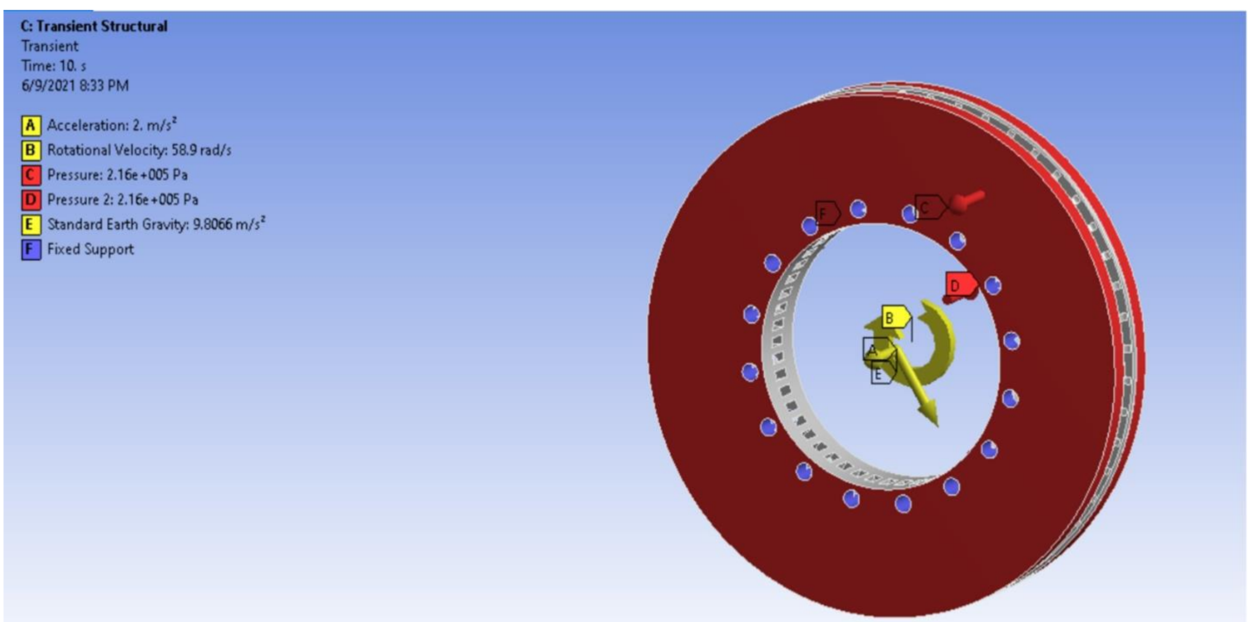


Figure 3. 27: Transient structural boundary conditions

CHAPTER FOUR

RESULTS AND DISCCUSION

4.1 Results of CFD analysis

At the end of the CFD analysis, the contour of heat transfer coefficient is displayed at the post-processing part of the ANSYS -FLUENT. In addition to this, it automatically calculates the average heat transfer coefficient values at the surfaces of the brake disc. The average heat transfer coefficients of the four different models are shown in the following figures.

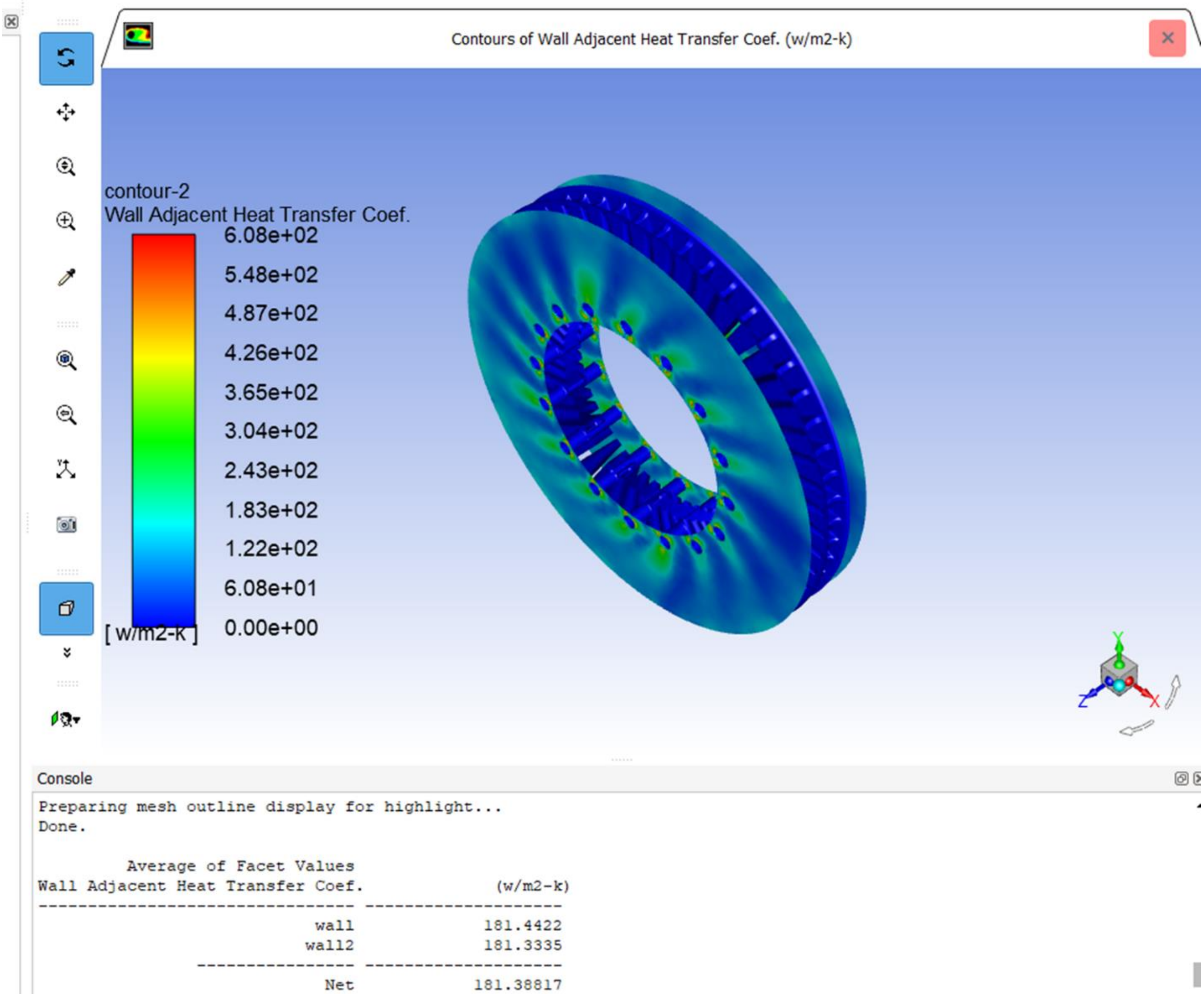


Figure 4. 1: Contour of wall adjacent heat transfer coefficient for ventilated straight (non-drilled)

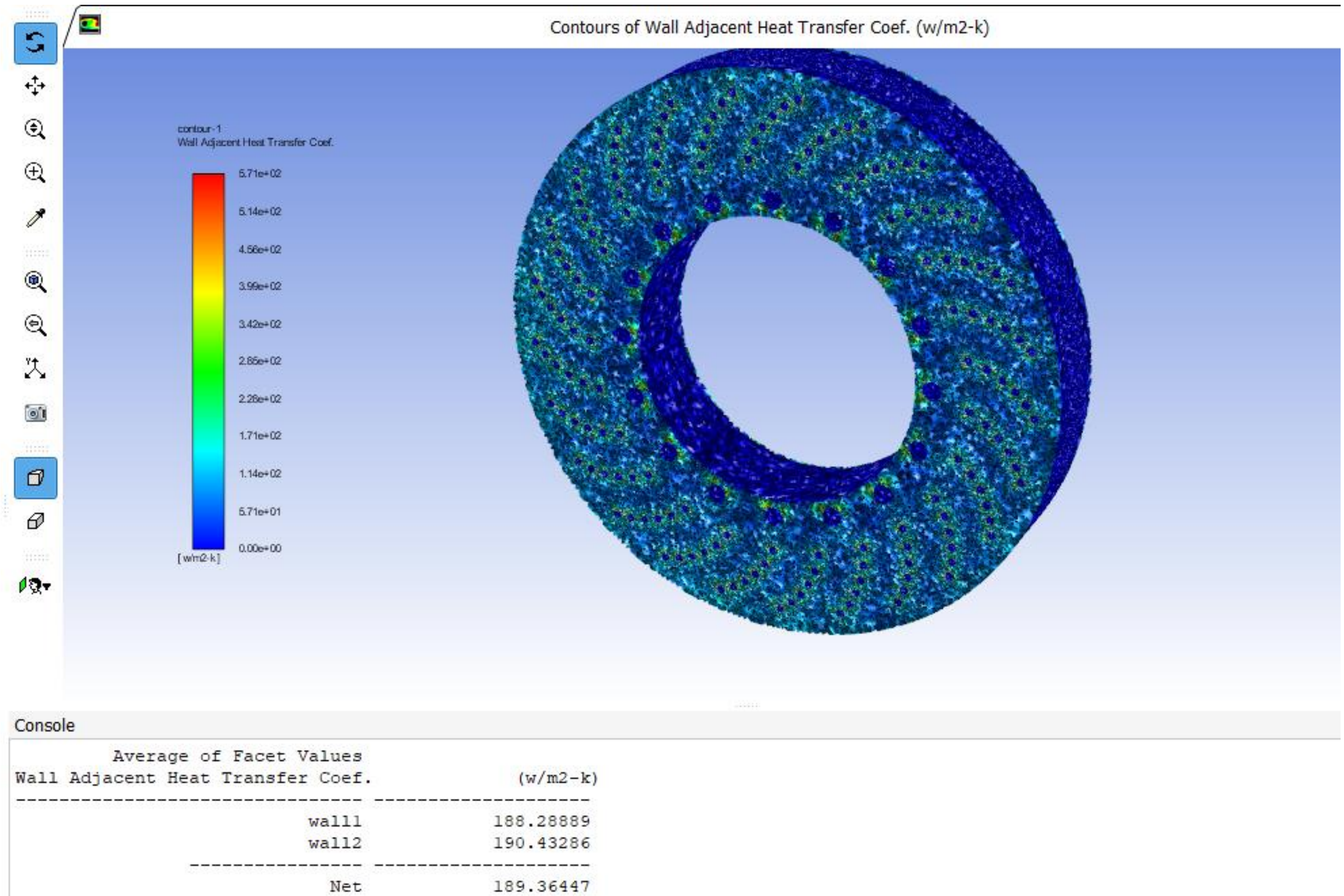


Figure 4. 2: Contour of wall adjacent heat transfer coefficient for ventilated with curved fins and a 5 mm hole diameter

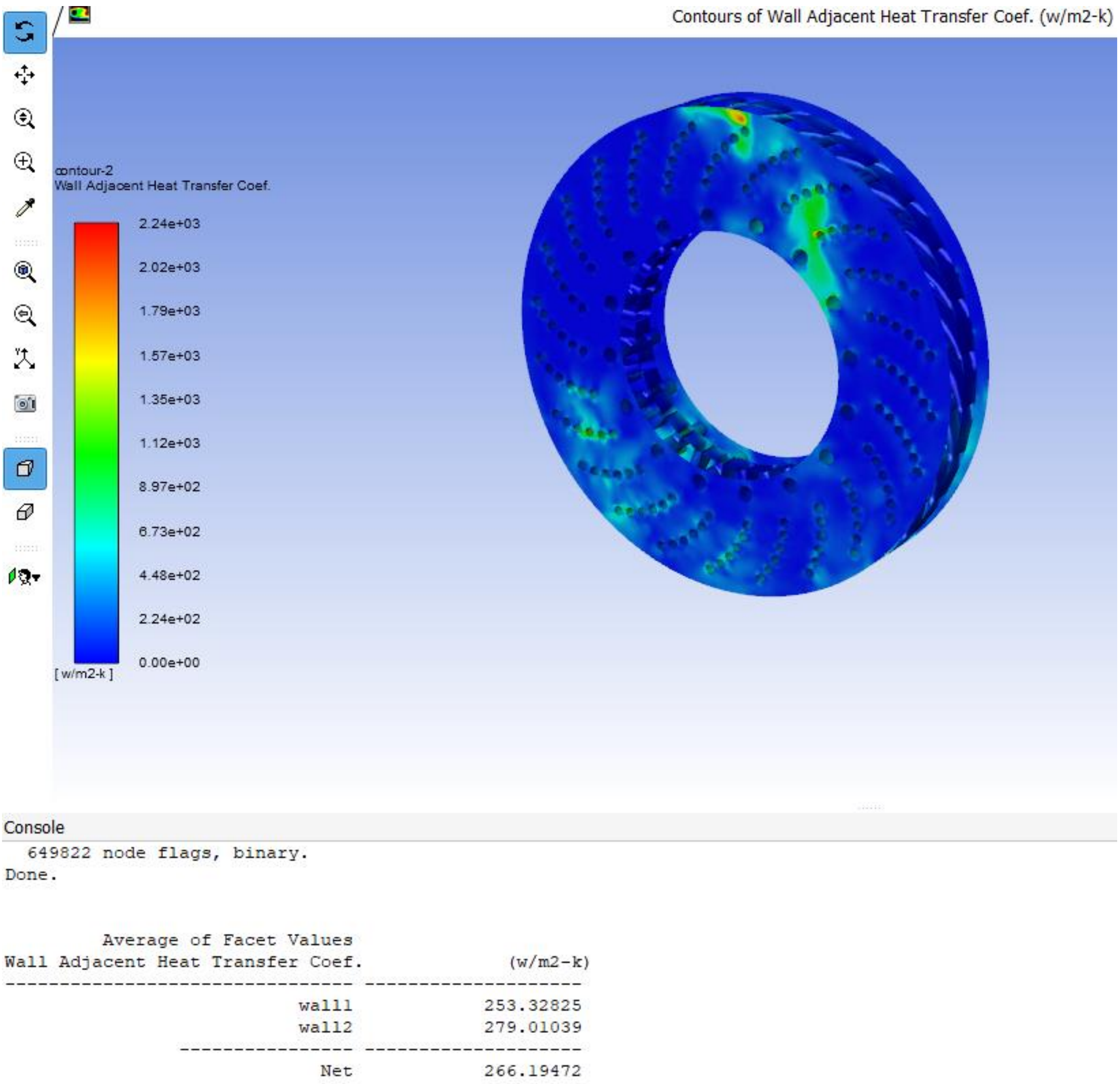


Figure 4. 3: Contour of wall adjacent heat transfer coefficient for ventilated with curved fins and a 7.5 mm hole diameter

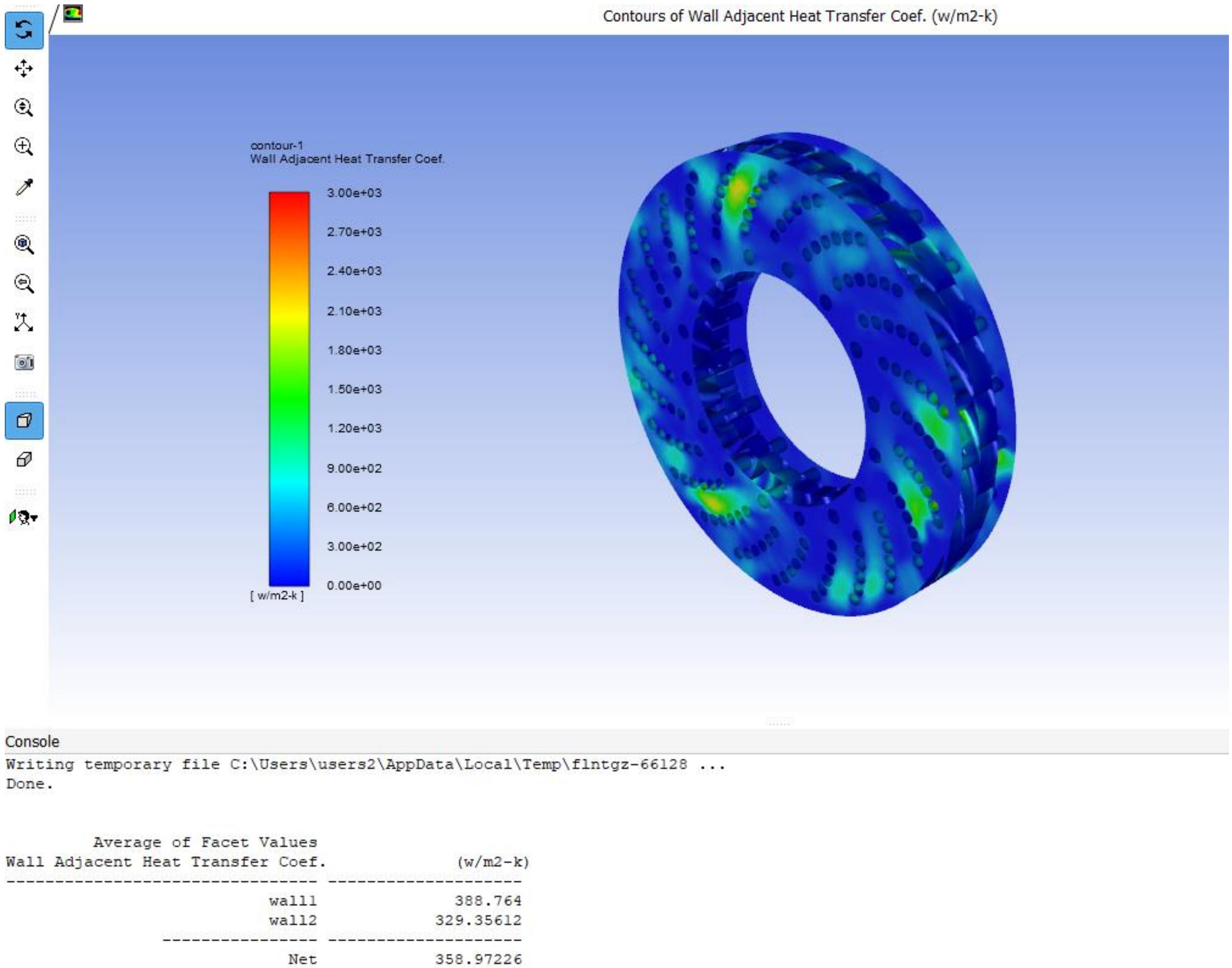


Figure 4. 4: Contour of wall adjacent heat transfer coefficient for ventilated with curved fins and a 10 mm hole diameter

For each disc model the maximum and minimum convective heat transfer coefficient values are tabulated with different colors. Whereas, the average heat transfer coefficient value is automatically calculated by the ANSYS-FLUENT database and has been displayed under each contour plots. It can be observed that the net value of average heat transfer coefficient for the model having holes of 10mm diameter has the highest value when it is compared to the other three models. Hence, the maximum temperature value of this model is expected to be the minimum one.

4.2 Thermal results

4.2.1 Temperature results

At the end of the transient thermal simulation, the maximum temperature-rise and total heat flux contours of the different brake disc models has been presented. Each model has its own maximum and minimum temperature values as well as maximum and minimum total heat flux distribution values at the disc surface. The thermal results of each model are shown in the following figures.

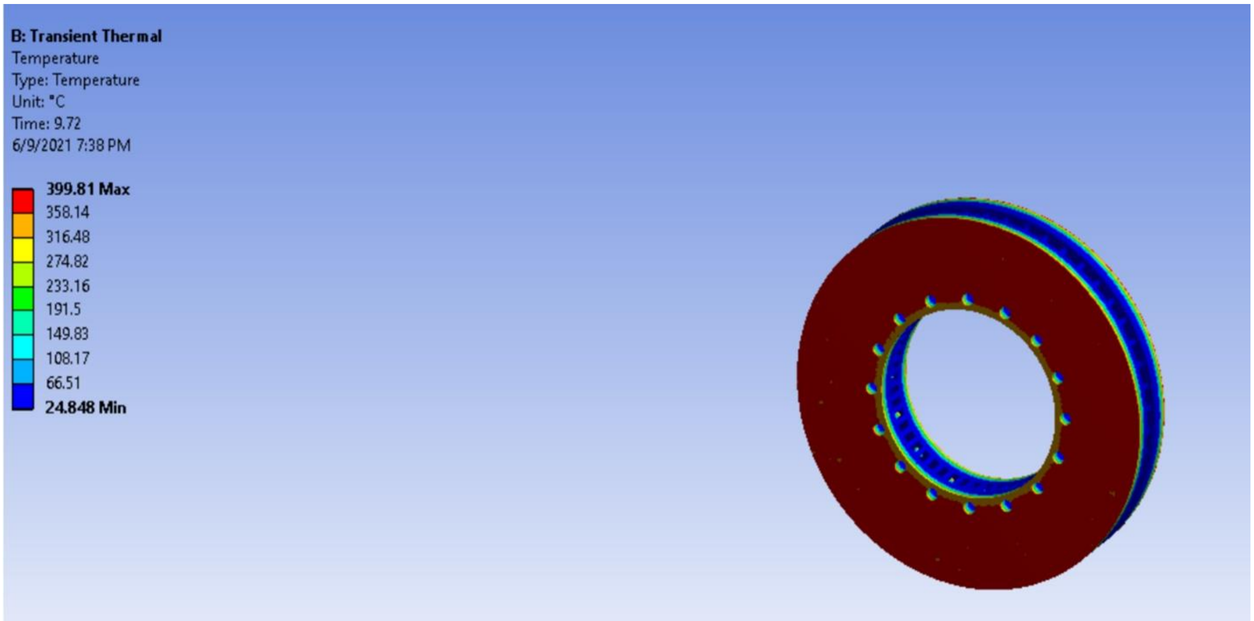


Figure 4. 5: Temperature variation of a ventilated disc with straight fins model(non-drilled)

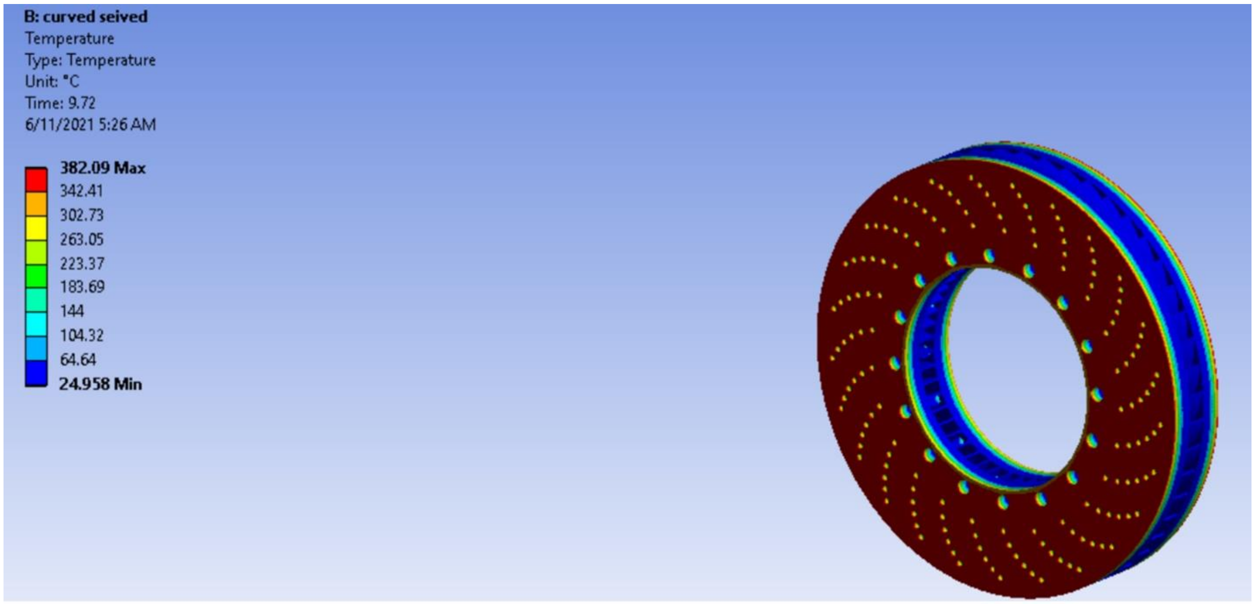


Figure 4. 6: temperature variation of a ventilated disc with curved fins model and a 5mm drill hole diameter.

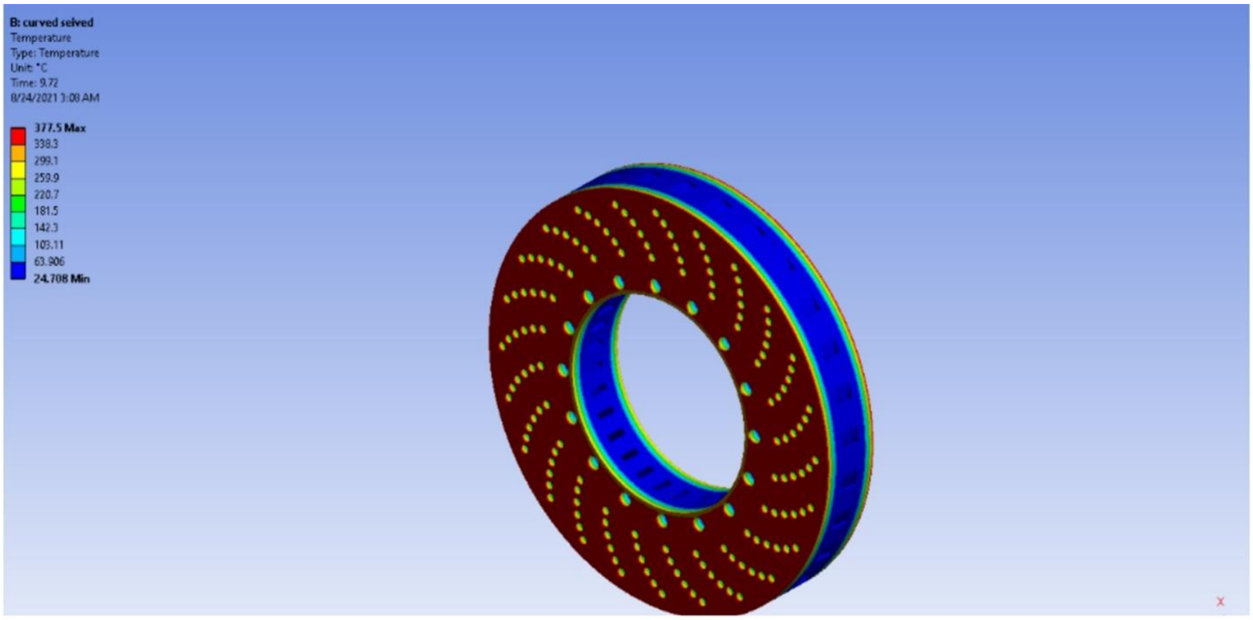


Figure 4. 7: Temperature variation of a ventilated disc with curved fins model and a 7.5mm drill hole diameter.

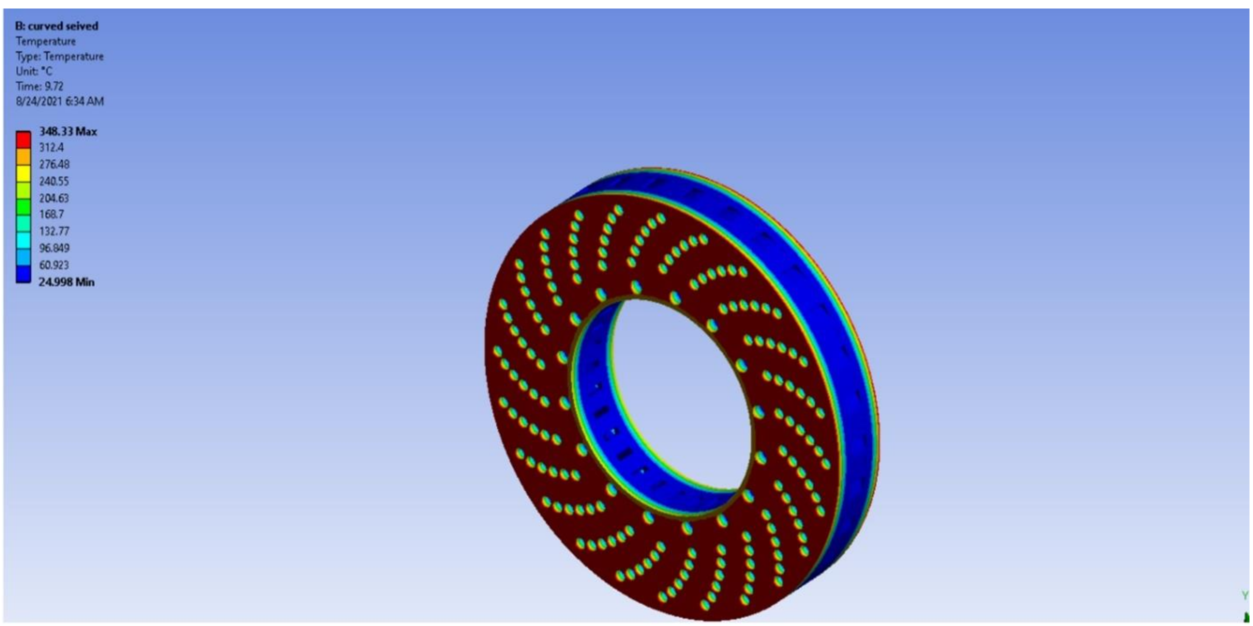


Figure 4. 8: Temperature variation of a ventilated disc with curved fins model and a 10mm drill hole diameter.

4.2.2 Total heat flux results

The contours of total heat fluxes of each model are also shown in the following figures.

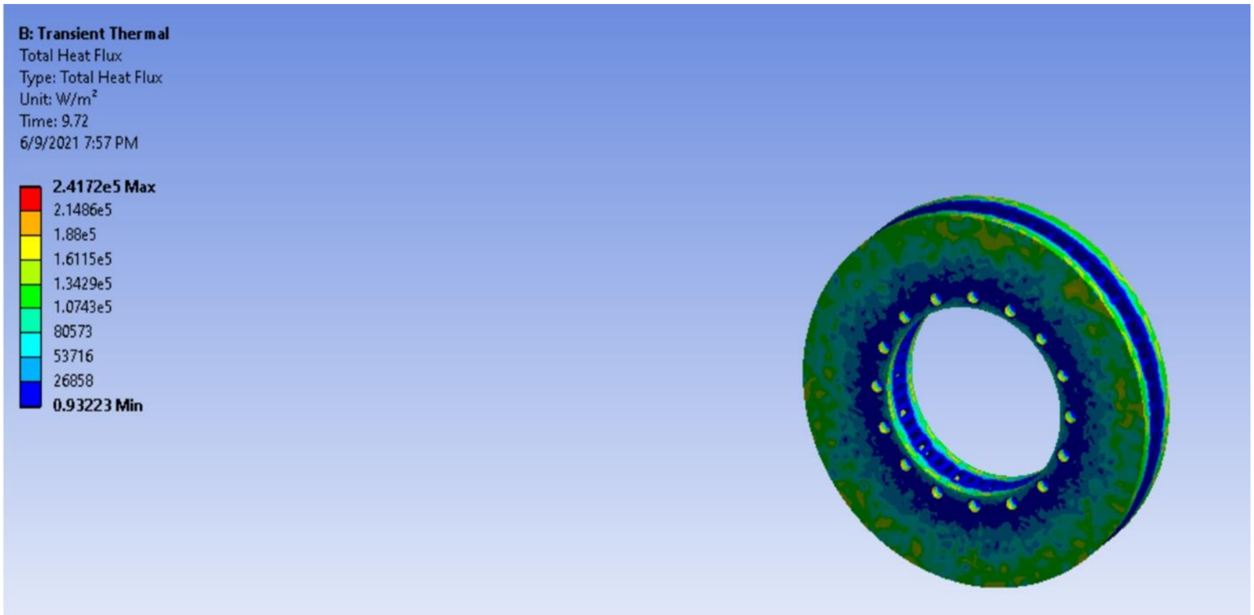


Figure 4. 9: Total heat flux variation of a ventilated disc with straight fins (non- drilled)

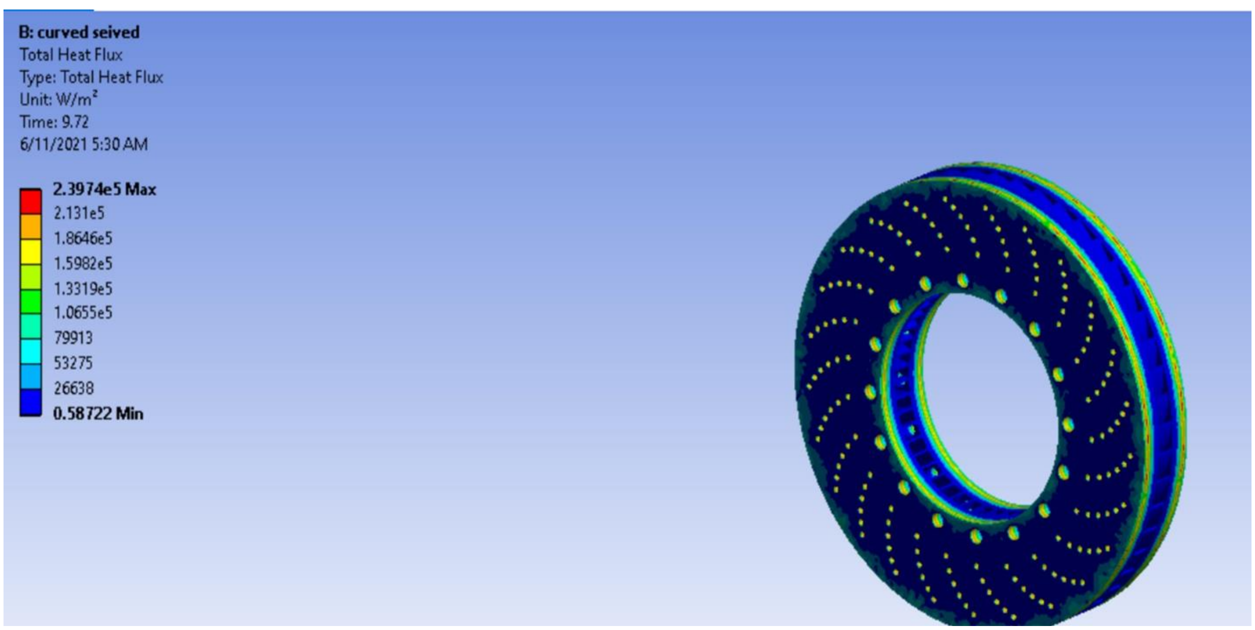


Figure 4. 10: Total heat flux variation of a ventilated disc with curved fins and a 5 mm drill diameter

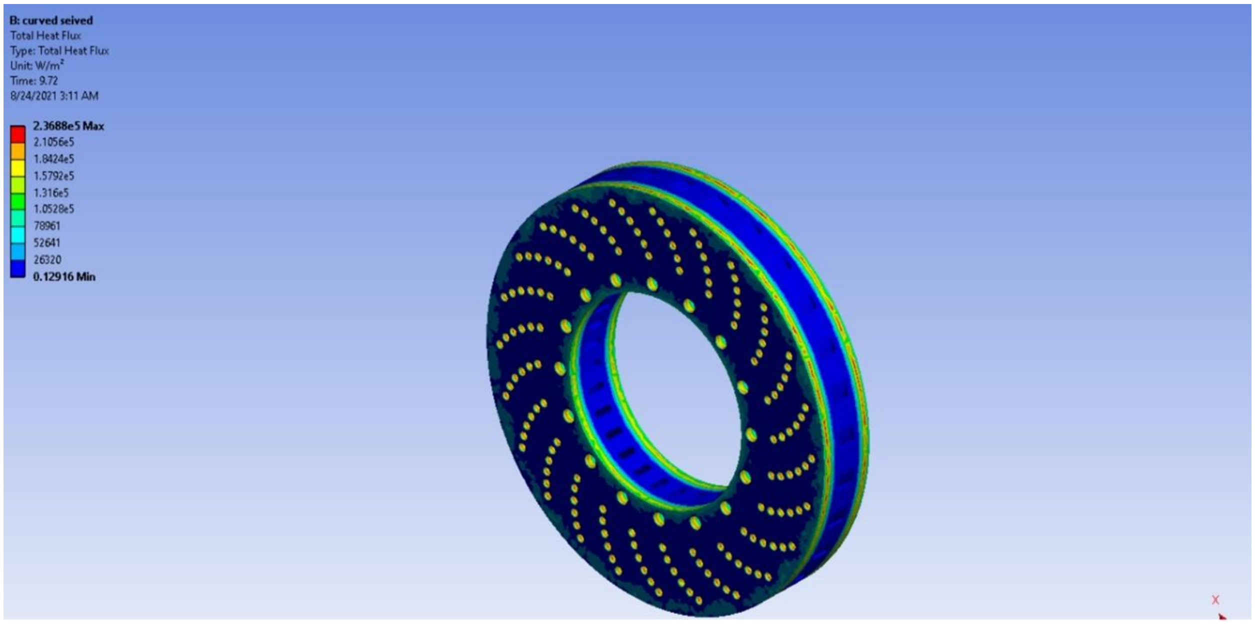


Figure 4. 11: Total heat flux variation of a ventilated disc with curved fins and a 7.5 mm drill diameter

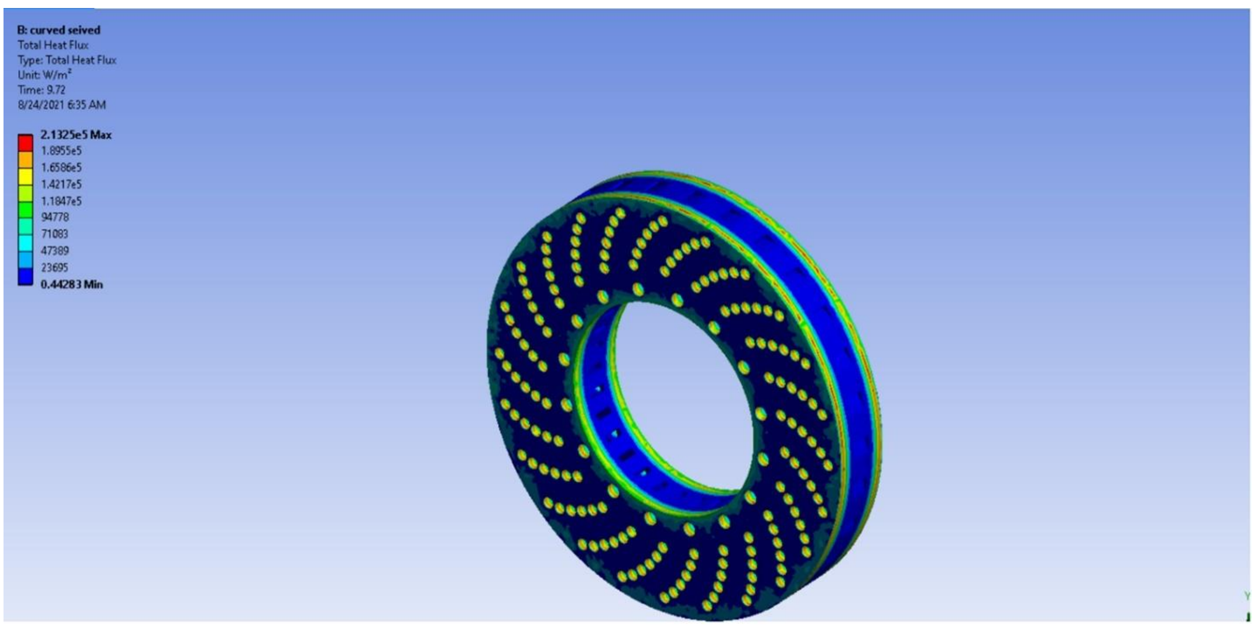


Figure 4. 12: Total heat flux variation of a ventilated disc with curved fins and a 10 mm drill diameter

It can be observed that the maximum value of total heat flux is minimum for the model having holes of 10 mm diameter.

4.3 Structural results

4.3.1 Equivalent (von-mises) stress results

the simulated von-mises stresses of each brake disc model are shown in the following figures. Maximum stress values are observed at the fixed support of the models and decreases towards the end of the disc.

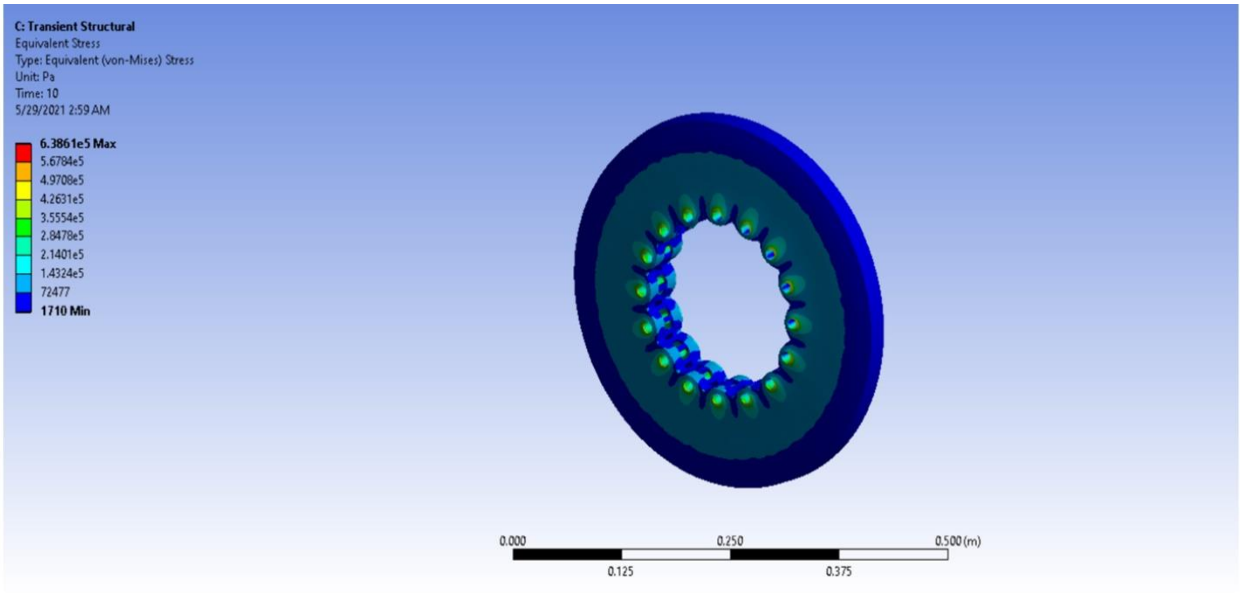


Figure 4. 13: Stress variation of ventilated disc with straight fins(non-drilled)

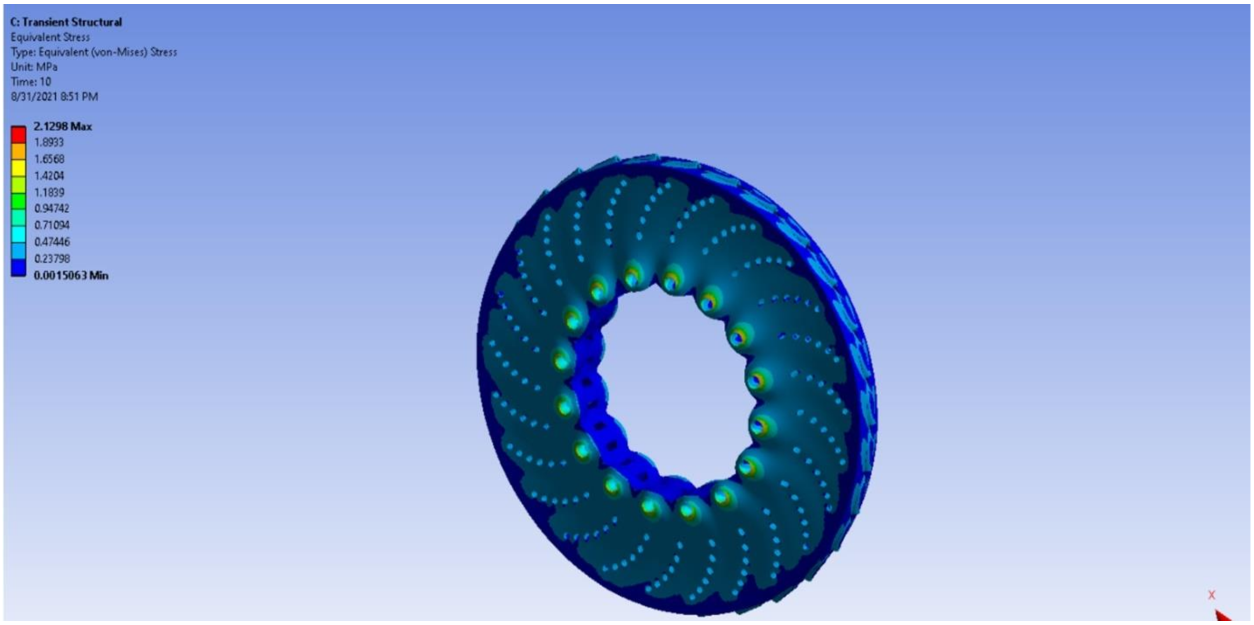


Figure 4. 14: Stress variation of ventilated disc with curved fins and a 5 mm drilled holes

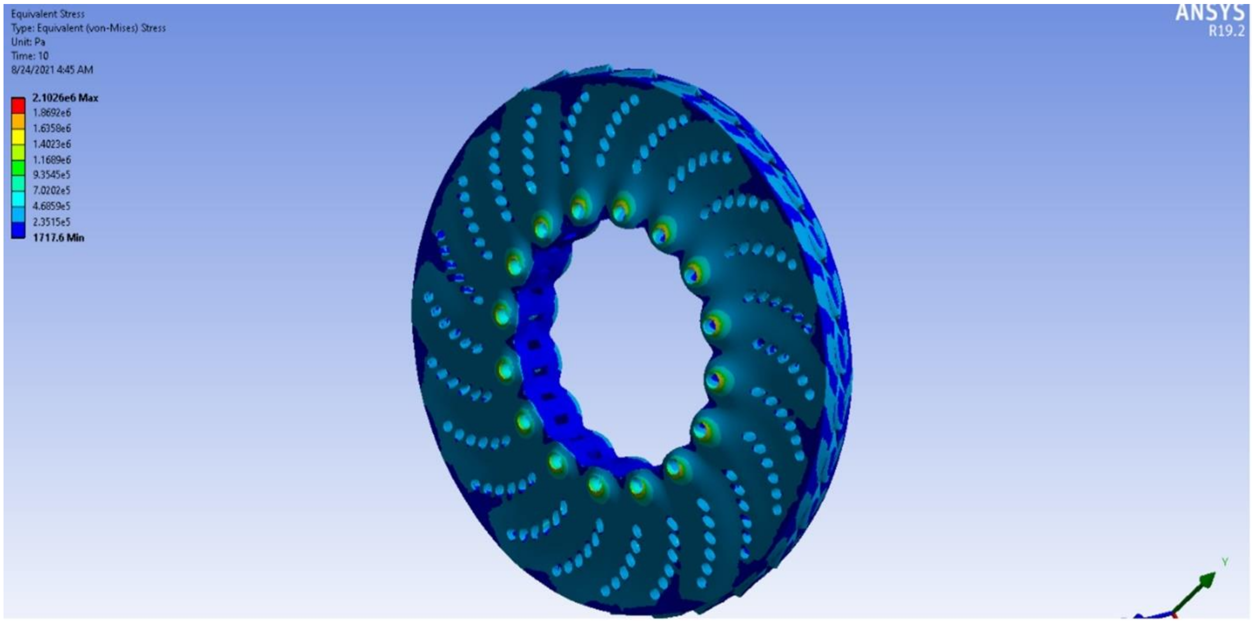


Figure 4. 15: Stress variation of ventilated disc with curved fins and a 7.5 mm drilled holes

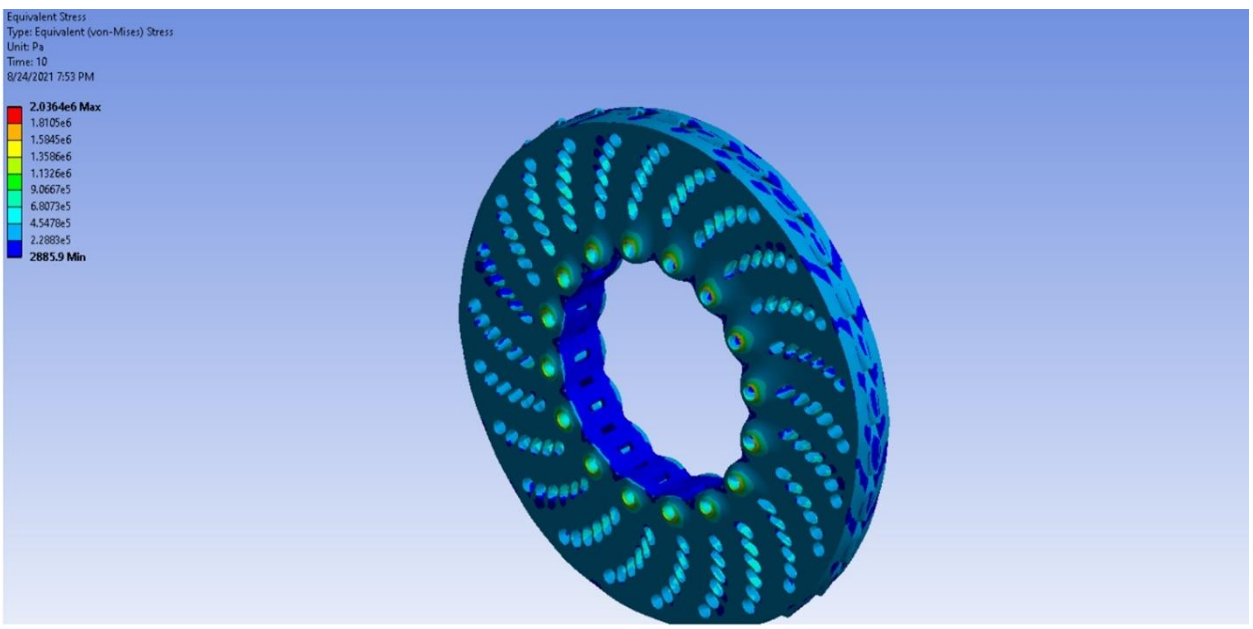


Figure 4. 16: Stress variation of ventilated disc with curved fins and a 10 mm drilled holes

4.3.2 Deformation results

The minimum deformation values for each model are observed at the fixed support of the disc while the maximum deformation values are observed at the tip of the disc surface. The deformation values of each brake disc model are also shown in the figures below.

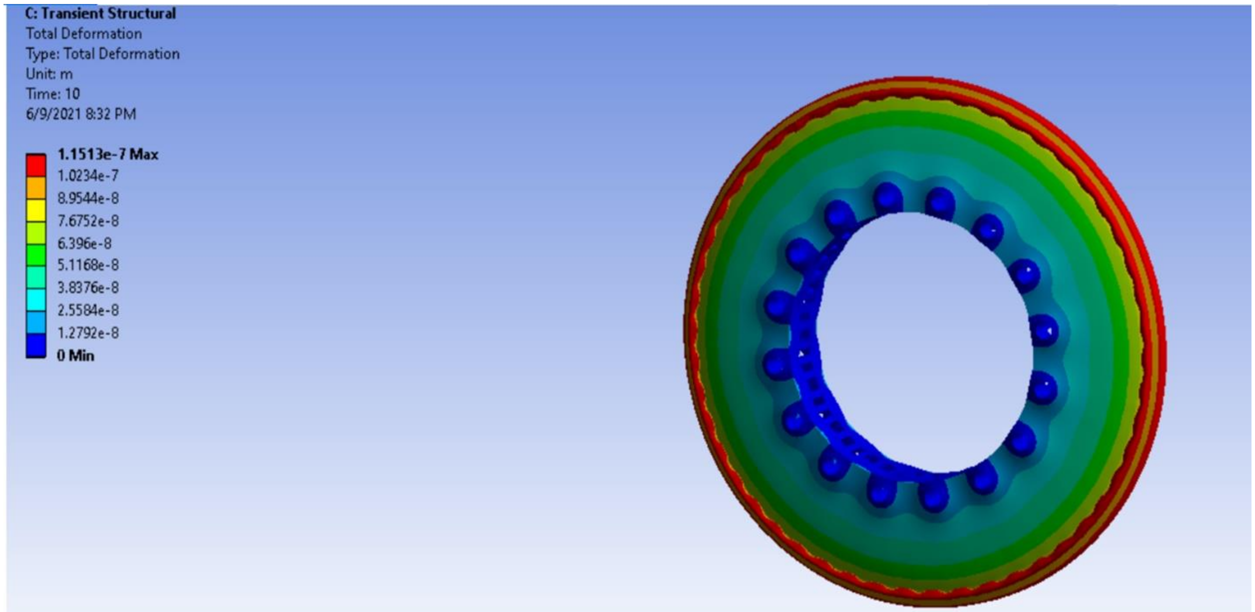


Figure 4. 17 : Deformation variation of ventilated disc with straight fins(non-drilled)

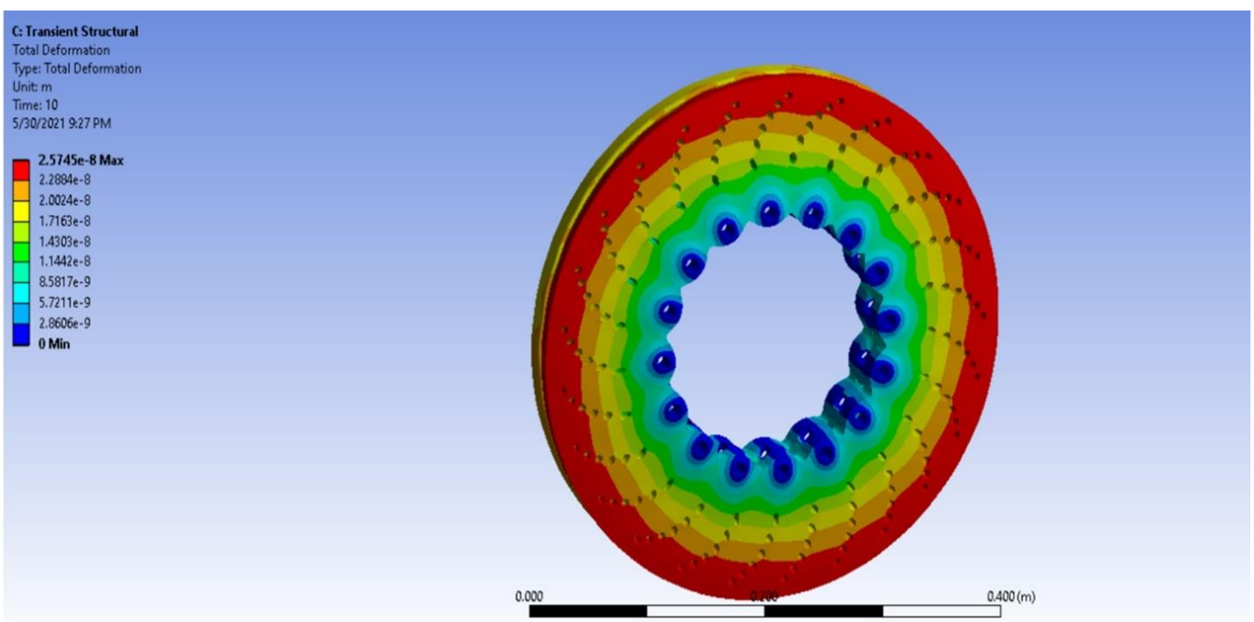


Figure 4. 18: deformation variation of ventilated disc with curved fins and a 5mm drilled holes (surface drilled)

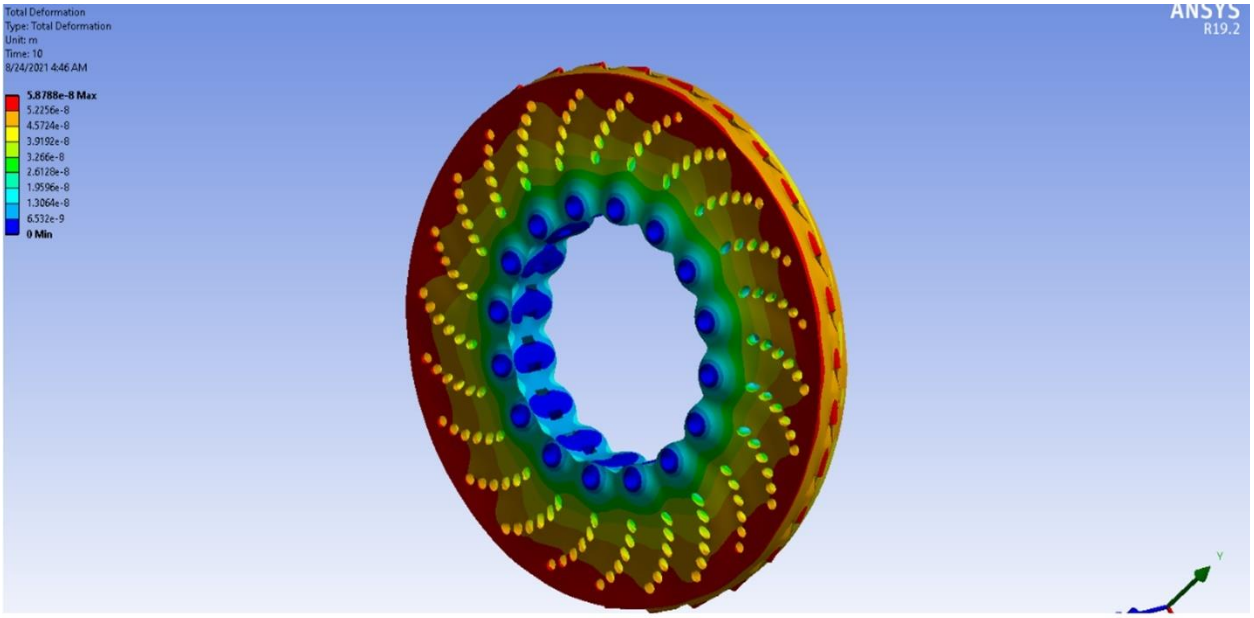


Figure 4. 19: Deformation variation of ventilated disc with curved fins and a 7.5mm drilled holes (surface drilled)

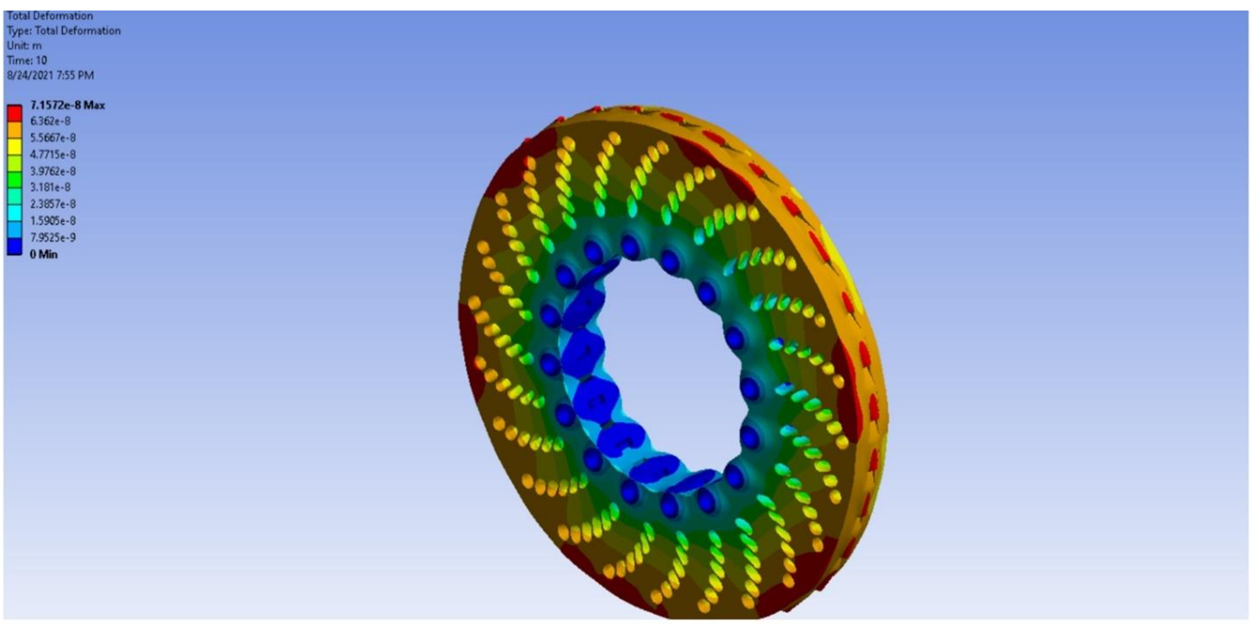


Figure 4. 20: Deformation variation of ventilated disc with curved fins and a 10 mm drilled holes (surface drilled)

All the ANSYS simulation results for each of the analyzed brake disc model are tabulated in table 4.1.

Table 4. 1: simulation results of all disc models

Parameters	Model with straight fins and no surface drill	Curved fin Model with 5mm surface drill	Curved fin Model with 7.5mm surface drill	Curved fin Model with 10 mm surface drill
Average convective heat transfer coefficient ($\frac{w}{m^2\,K}$)	181.38	189.36	266	358.97
Maximum temperature (°C)	399.81	382.09	377.5	348.33
Mass(kg)	10.39	9.97	9.49	8.79
Total heat flux ($\frac{w}{m^2}$)	241720	239740	236880	213250
Maximum von-mises stress (MPa)	0.64	2.12	2.1	2.03
Maximum Deformation (m)	$0.115e^{-8}$	$2.57e^{-8}$	$5.87e^{-8}$	$7.15e^{-8}$

The model with holes of 10 mm diameter has the highest HTC of 358.97 $\frac{w}{m^2\,K}$ and the lowest temperature value of 348.33°C while the one with holes of 7.5 mm diameter has HTC value of 266 $\frac{w}{m^2\,K}$ and temperature value of 377.5°C. whereas, the model with holes of 5 mm diameter has HTC value of 189.36 $\frac{w}{m^2\,K}$ and a temperature value of 382.09°C while the model having no any surface drill (except bolting drills) has HTC value of 181.38 $\frac{w}{m^2\,K}$ and temperature value of 399.81°C. It can be observed that adding a surface drill and increasing the diameter of the drill hole has a significant effect in enhancing the average HTC value thereby reducing the maximum temperature-rise of the brake disc. The von-mises stress of each model is an equivalent value due to the temperature and the structure and could be compensated by the effect of both parameters. For this reason, the variation might not be very high. It can also be observed the von-mises stresses of all of the four models are also below the tensile yield strength value of the selected material(314MPa).

In addition to this, it can be observed that the models are not very much stressed in comparison to the tensile yield strength of the material. For the fact that the maximum temperature could affect the material property and thereby the right functionality of the whole system, The most important comparison factor should be the max temperature. The maximum temperatures of all the four-disc models are also under the temperature limit of the selected material (600℃) for the maximum emergency braking time(9.72sec). however, as it can be observed in the figure below, the maximum temperature rise could go as high as 518.78℃ up to the braking time of 5 seconds and drops to 399.81℃ at 9.72 second while considering the model with the worst temperature value.

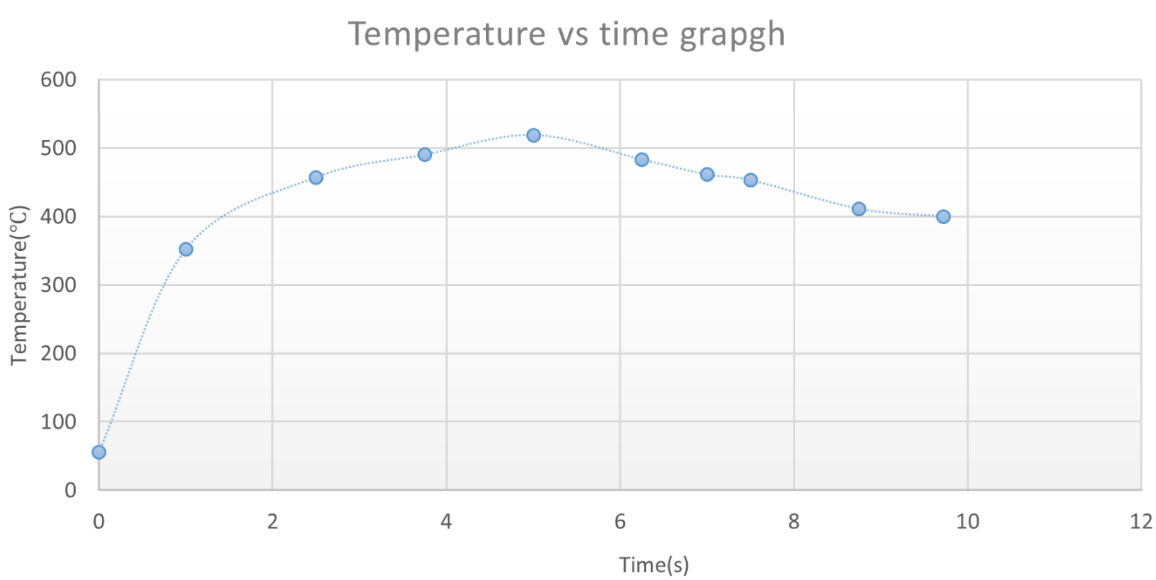


Figure 4. 21: Temperature vs time graph of the disc model with straight fins(non-drilled)

The finite element model of this paper is validated by re-simulating the results obtained by **Kishore & Vineesh(2020)** [19] and their simulation result was 214 ℃. The validation is conducted at 160km/hr and 38 seconds of simulation time using cast iron material and the maximum temperature of the simulation was obtained to be 225.41 ℃ with 5.3% error. The input parameters of the literature used for the simulation are listed in the following table.

Table 4. 2: Brake disc input parameters of the re-simulated literature

Parameters	value
Outer diameter	640m
Inner diameter	240m
speed	160km/hr
Vane number	46
Deceleration	$0.7m/s\hat{}2$
Disc surface area	$0.553m^2$
Heat transfer coefficient	$12\,\frac{w}{m^2\,K}$
Braked mass	8000 Kg
Wheel diameter	458m
Initial temperature	30℃
Braking time	38s

The simulated temperature variation of the related literature could be observed on the following figure.

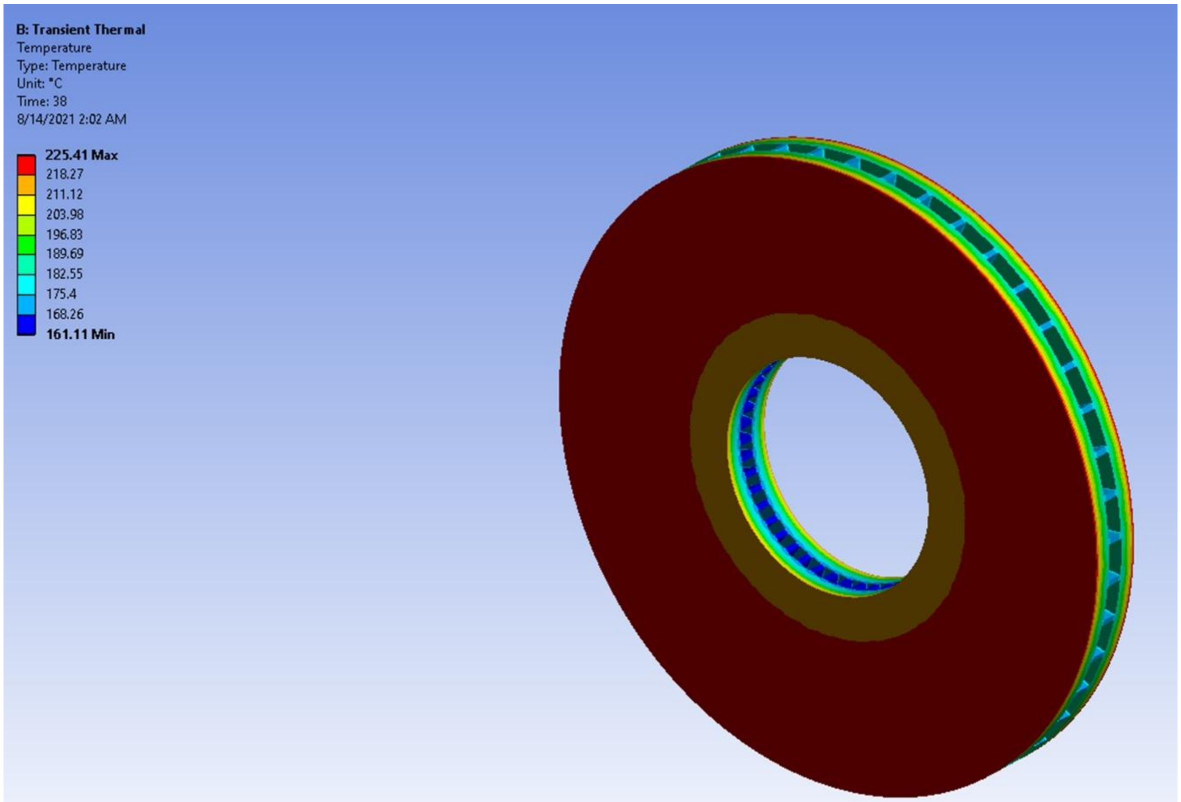


Figure 4. 22: Temperature variation of the re-simulated literature

The variation between the obtained result and **Kishore & Vineesh(2020)** [19] results is shown in the figure below.

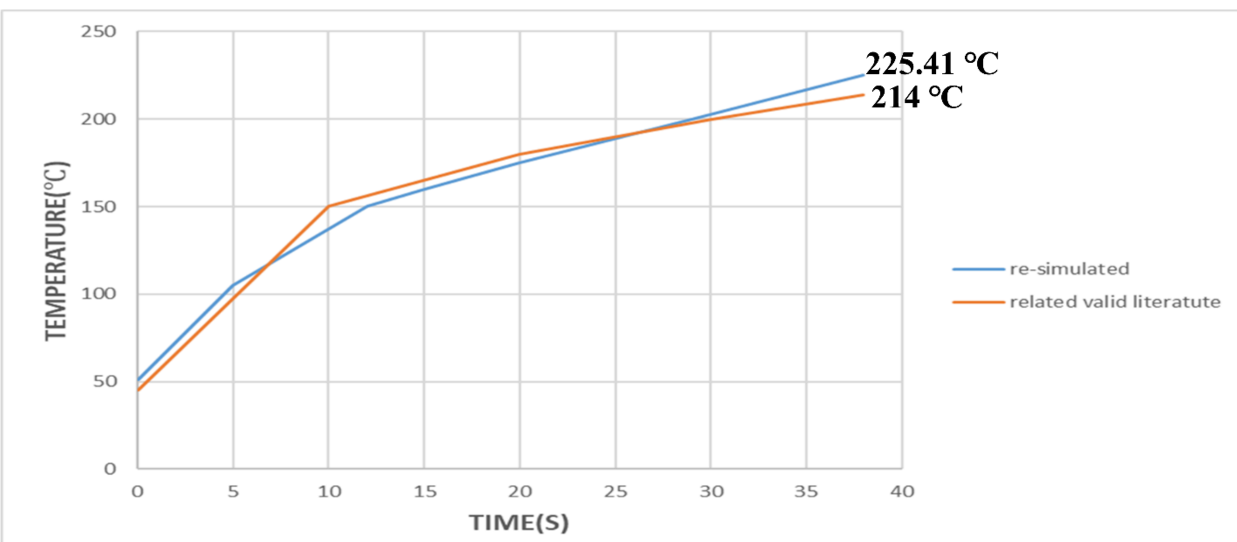


Figure 4. 23: Temperature vs time graph of related literature and its re-simulation

CHAPTER FIVE

CONCLUSION, RECOMMENDATION AND FUTURE WORK

5.1 Conclusion

In this paper the thermal comparison of four different brake disc models is done using ANSYS 19 software. The analyzed brake discs are modelled using the design specifications of AALRT by SOLIDWORKS 13. The disc models include one model with straight cooling fins and without a surface drill (current AALRT model) and 3 models with a curved cooling fins and equally spaced surface drills. The three drilled models also include one model with holes of 5 mm diameter, one model with holes of 7.5 mm diameter, and other model with holes of 10 mm diameter so as to investigate the effect of varying the holes' diameter on the enhancement of average HTC and reduction of temperature value. The analysis of each brake disc model is done at emergency braking conditions of AALRT (at 9.72 braking time, $2\, m/s\text{^}2$ deceleration and at a speed of 19.44m/s) by considering air cooling, maximum gradient of AALRT (3.15^{0}) and Sic/6061Al material. ANSYS FLUENT database is used to calculate the average HTC of each brake disc model so as to use them as a boundary condition for the thermal analysis together with the time variant heat flux. In addition to this, the structure of each disc model is investigated on the transient structural database so as to show the maximum von-mises stress and deformation. Based on analysis it can be concluded that:

- Adding a surface drill and increasing the diameter of the drill hole has a significant effect on enhancing the average HTC value thereby reducing the maximum temperature-rise of the brake disc.
- As far as maximum temperature rise is concerned, the disc model with hole diameter of 10 mm is the best (348.33℃), the model with hole diameter of 7.5 is the second best (377.5℃), and the model with hole diameter of 5 mm takes the third rank (382.09℃) while the non-drilled model is the worst (399.81℃).
- As far as maximum temperature- rise and von-mises stress at the highest gradient and emergency braking conditions of AALRT are concerned, The Sic/6061Al material could be used as a brake disc material for AALRT.

5.2 Recommendation

Concerning maximum temperature- rise, it has been shown that the disc model with curved fins and a 10 mm surface drill is much better than that of the current AALRT brake disc model. Therefore, it is highly recommended to replace the current AALRT disc model by the curved and drilled disc model (with a surface drill diameter of 10 mm). In addition to this, since it is known that the Sic/6061Al alloy material reduces the mass of the disc by about 60% when compared to cast iron, and for the fact that the analysis conducted shows the effectiveness of the material at the emergency braking and high gradient of AALRT conditions (for the fact that the maximum temperature rise observed are below the temperature limit of the material) it is also recommended to utilize the material for the AALRT brake disc.

5.3 Future work

Study of the brake disc models that take in to account the effects of Wear, Vibration, Tribology and Fatigue load are recommended for future work. Additionally, the use of programming language to determine the exact value of convective heat transfer coefficient at each and every point of the surface of the disc is also recommended as a future work.

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