Proceedings of the 10th World Congress on Mechanical, Chemical, and Material Engineering (MCM'24)
Barcelona, Spain - August 22-24, 2024
Paper No. ICMIE 151
DOI: 10.11159/icmie24.151

Disc Brake Rotor Thermal Analysis for a Formula SAE Race Car

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Abstract - A major component in designing a one-seated race vehicle for the Formula Society of Automotive Engineers (FSAE) student competition is the brake system. In order to prevent failure from brake fade, thermal stress, and thermo-mechanical fatigue, car brakes must not only be robust enough to withstand extreme mechanical loads but also disperse the heat generated in order to maintain the temperature below Maximum Design Temperature (MDT) limit. Furthermore, the weight of the brake discs plays a crucial role in the overall weight and performance of the vehicle. The methodological analysis of a disc brake rotor's temperature profile under sporadic loads during consecutive braking and acceleration/cruise events is presented in this work. “OptimumLap” is used to determine a time-dependent vehicle speed and braking intensity profile for the 2019 Formula SAE competition Michigan endurance track and default Formula SAE vehicle configuration in “OptimumLap”. The temperature distribution of the brake disc during a sequence of successive braking and acceleration/cruise events across several laps is simulated using "SolidWorks" thermal analysis. This study examines a practical approach for Formula SAE racing car disc brake temperature profile analysis. For the future work, geometry optimization will be done for the disc brake rotor in order to get appropriate maximum temperature.

Keywords: Thermal analysis, Formula student, SolidWorks, OptimumLap, Disc Rotor

1. Introduction

Formula SAE is a well-known engineering competition that began in 1981 at the University of Texas in Austin and has since spread around the world [1]. College students are tasked with designing, building, and competing in formula-style one-seated open wheel race cars. Teams need to develop high-performance vehicles under stringent budgetary, safety, and design limitations. These vehicles must then be evaluated in fast-paced dynamic events like autocross and endurance races to evaluate their durability, speed, and agility [2]. Simultaneously, static exercises like design presentations assess technical ideas and economic viability. These activities provide students excellent chances to refine their problem-solving, collaborative, and project management abilities while advancing automotive technology and developing a community of driven engineers committed to innovation in motorsports. Formula SAE cars are designed with a braking system that provides superior stopping force and control. To lower weight and enhance handling, it usually has lightweight brake parts like aluminium callipers and high strength metal brake disc rotor components.

Numerous research studies have examined various aspects in thermal analysis of disc brake design in Formula SAE vehicles including material choices, heat dissipation techniques, and thermal control tactics to improve the effectiveness and dependability of these braking systems. Chengal Reddy (2013) used “Pro-E” and “Ansys” to model and analyze solid and vented disc brakes [3]. “Pro-E” is used to produce finite element (FE) models of the brake disc, and “Ansys”, which is based on the finite element method (FEM), is used to analyze the FSAE car disc brake. The optimum material for a disc brake is determined as maraging steel, according to the study results. It is concluded that ventilated type disc brake is the best possible profile for the present application. Eshaan Gupta (2022) made a study that optimizes the FSAE race car's braking system for weight reduction and performance enhancement compared to OEMs [4]. Employing “Matlab” for calculations and innovative design tools like “Altair Inspire”, the researcher iteratively designed and simulated components with “SolidWorks” and “Ansys” for safety and effectiveness. This study has found the following results: Shear force of 4500 N, max stress of 3.10×10^8 Pa, heat flux of 18000 W/m^2 and maximum temperature of 194.04 °C. Manthan Vidiya and Balbir Singh (2017) conducted thermal analysis of brakes, including energy conversion methods, convection coefficients, and temperature rise [5]. They validated findings through simulations and real-world car testing. This paper goes beyond previous studies by analyzing temperatures across laps, crucial for optimizing brakes in formula student cars. Using “Ansys” Transient Thermal profile, they plotted temperatures on the disc, with a maximum of 61.3°C. A temperature-time graph showed increase with each braking, aiding comparison with real-world testing. Actual car testing revealed a maximum disc temperature of 62.3°C, showing a similar trend. Comparing simulation and experimental results, they found nearly identical temperature variations, with a maximum difference of around 5°C. Pragya Mahajan (2021) conducted a research with the goal of creating an effective braking system with an emphasis on thermal stability, weight reduction, and compactness [6]. For optimum performance and safety compliance, he entails calculating braking torque, choosing components, CAD modelling, simulations, and real-world testing. The study's conclusions show that even under severe braking situations, the planned brake components, especially the disc, remain thermally and structurally robust. For manufacturing, laser cutting is used. Determining the boundary conditions for the thermal analysis depends heavily on the airflow surrounding the disc. In a later study, using “CFX”, Belhocine and Omar conducted a computational fluid dynamics (CFD) investigation to ascertain the airflow around the disc rotor and the heat transfer coefficients (HTCs) in each surface [7]. The authors were able to model the temperature distribution for the grey cast iron material disc using the HTCs. Karnik created a coupled model with Matlab and Ansys to calculate torque to be used in thermal analysis [8]. The researchers examined three distinct discs: Rotors without slots, with slots, and with slots and dimples. The use of dimples increases the rate of convective cooling. Deepak Hugar and Kadabadi (2017) investigated various slot designs with the goal of enhancing thermal conductivity and reducing weight through thermal analysis on an actual disc brake rotor model [9]. The author's objective is to increase disc brake efficiency in order to lower daily accident rates. The researchers used “Catia” for modelling and “Ansys” for static and transient thermal analysis. When compared to the regular Bajaj Pulsar 2-wheeler disc brake and other new designs, the study reveals that a modified disc brake design, known as "new disc 5," improves brake performance. Its safety is confirmed by structural study, and its temperature distribution is improved. In a similar study, Kumar, Thriveni, Reddy, and Gawd (2014) discussed how to optimize automotive brake discs and examine the steady-state thermal behavior of the contact between the disc and pads during braking [10]. The authors performed this by using “Ansys”, “SolidWorks”, and “HyperMesh” for design, analysis, and optimization, paying close attention to thermal-structural details. The findings indicated that while the curved vented disc has a larger heat flux than the straight vented disc, substituting straight vents with curved vents in the brake disc lowers von Mises stresses, displacement vector sum, and disc weight.

The aim of this study is to design a disc brake rotor component that will be employed in a formula SAE student single-seater open-wheel race car, allowing it to meet the SAE constraints and driver requirements. The disc brake is designed to be strong so that it withstands the braking forces while having adequate heat dissipation which reduces maximum operating temperatures. The disc brake should be light weight so that it satisfies the brake weight requirement.

2. Methodology

There are several phases in the project's methodology that must be followed in order. First, a market benchmarking study of disc brakes already available at the market was carried out to gather information on common designs. The disc brake's geometric configuration was then the focus of attention, which was performed by using “SolidWorks” to create a 3D model. Afterwards, available track and vehicle files (Michigan 2019 endurance track and FSAE aero vehicle) are employed in “OptimumLap” in order to determine the FSAE race endurance event brake profile. Later on, the thermal simulations were performed for consecutive braking/accelerating events using “SolidWorks” to obtain the brake rotor’s maximum temperature profile. The flowchart of the method is represented in Figure 1.

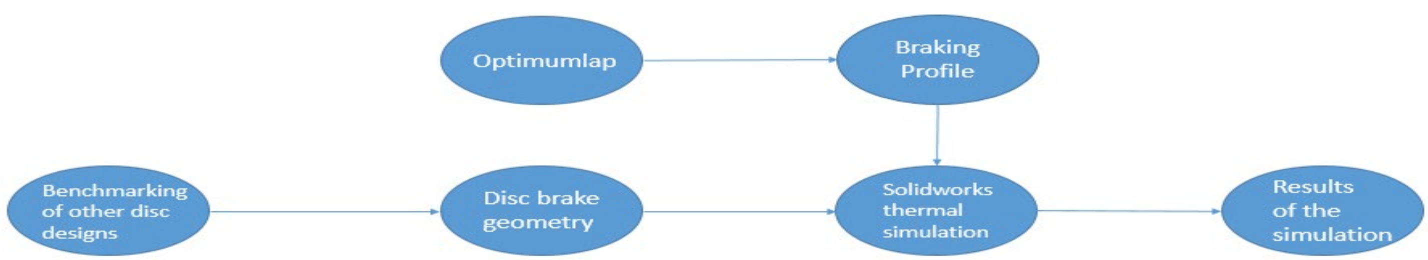

Fig. 1: Flowchart of the methodology of the study.

2.1 Braking Profile Determination

Standard FSAE Aero vehicle and endurance track used in FSAE Michigan 2019 competition is used for braking profile determination. All specifications of the track and vehicle used in the study are listed in Tables 1 and 2.

Table 1: Specifications of the “FSAE Aero” vehicle used in “Optimumlap”.

Parameter	**Value**
Total Mass	230 kg
Max Torque	75.5 N.m at 7862 rpm
Type of Fuel	Gasoline
Type of Transmission	Sequential Gearbox
Max Power	93.8 hp at 9264 rpm
Power Mass Ratio	0.41 hp/kg
Downforce @ 100 km/h	624.36 N
Drag @ 100 km/h	502.66 N

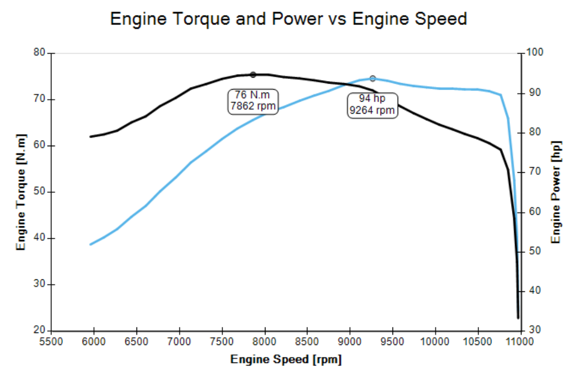

Table 2: Specifications of the “Michigan 2019” endurance track.

Parameter	**Value**
Type of Track	Temporary Circuit
City	Michigan
Country	United States
Track Direction	Forward Direction
Total Track Length	2168.21
Percent Left Corners	33.96%
Percent Right Corners	32.46%
Percent Straights	33.58%
Average Corner Radius	25.4 m
Minimum Corner Radius	1.39 m
Longest Straight	45.95 m

In particular, track simulation is run for successive laps, which means that the starting vehicle speed is not equal to 0 kph. The car finishes one lap in 142 seconds. Figure 2 show the outcomes of the simulation. The vehicle was simulated to have a peak speed of 110 kph and a minimum speed of 17.5 kph. There are 70 braking and accelerating events in total. The events that has a duration of less than 0.2 seconds are neglected, and previous and later events are combined together. During braking and accelerating, the maximum longitudinal acceleration is simulated to be 6.8 m/s^2 and the maximum longitudinal deceleration to be -19 m/s^2 respectively. It can be noticed that the high deceleration values are due to additional downforce with aero components of the vehicle such as like and difusers.

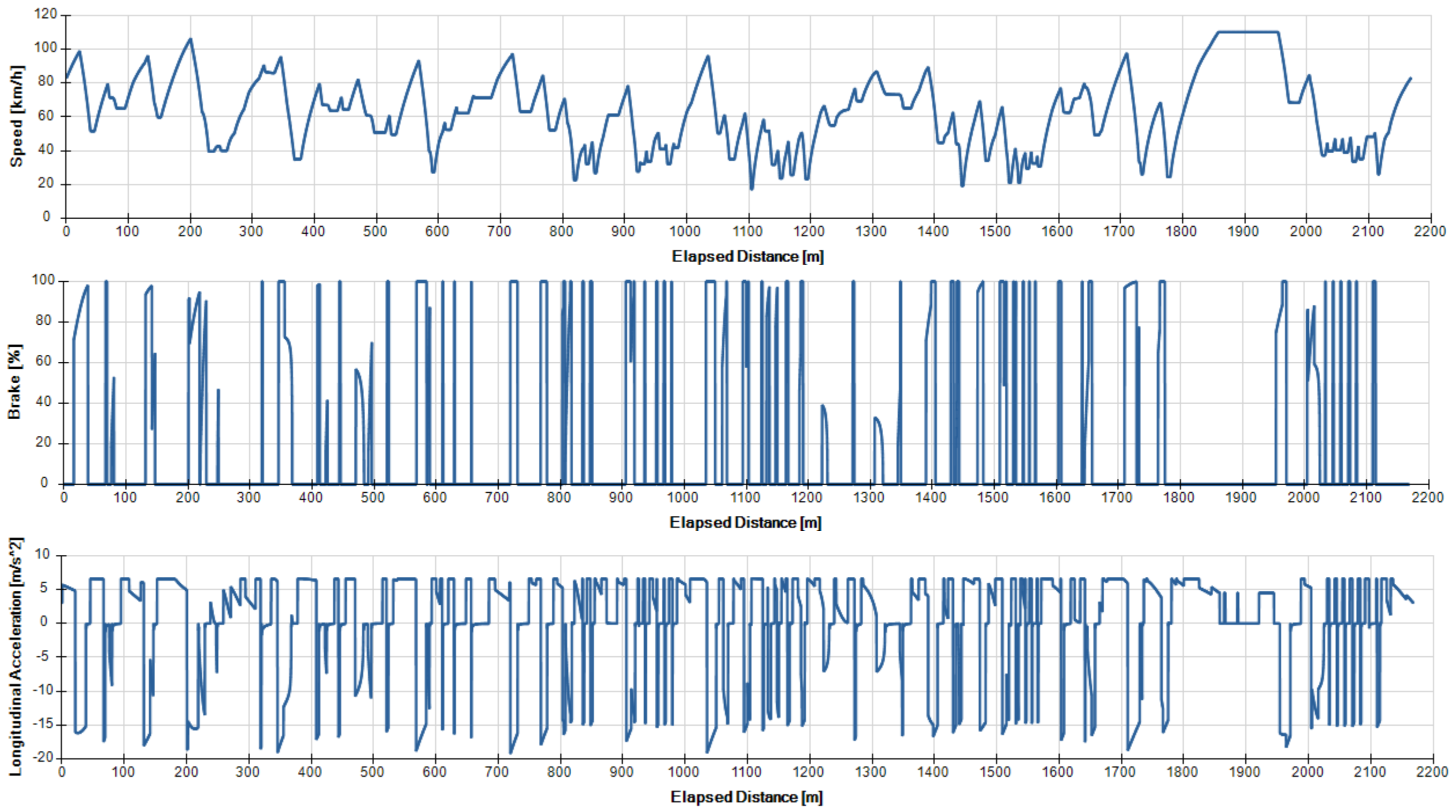

Fig. 2: "OptimumLap" simulation results (speed profile-brake profile-acceleration profile).

2.2 Thermal Simulation

The brake rotor used for the study is shown in Figure 3. The disc brake rotor is TEKTRO brand 6-bolt brake rotor TR-8 with outer diameter of 161 mm and thickness of 2 mm. For the thermal analysis, the inner part of brake rotor is excluded from the thermal analysis as shown in Figure 4 since the heat power generated by the calipers is only applied on the outer part of the disc brake rotor. As a result, more accurate temperature profiles are obtained in the simulation. Heat power applied to the disc circularly through the brake pad projection on both sides during the brake event is calculated using Equations 1-2. [11]

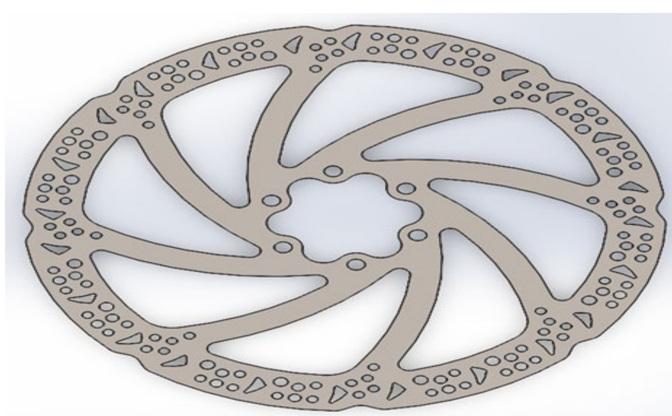
Figure 3: Brake rotor designed in "SolidWorks".

Figure 4: Extruded brake rotor used for the thermal analysis.

$$P = \frac{KE}{\Delta T} \tag{1}$$

$$KE = \frac{m({V_{max}}^2 - {V_{min}}^2)}{2} \tag{2}$$

where:

• P is the power
• KE is the kinetic energy that will be converted to heat during the brake event,
• ΔT is the duration of the brake event,
• m is the ¼ of the mass of the vehicle and the driver,
• V_{max} and V_{min} are the starting and end longitudinal velocities of the vehicle.

The heat convention coefficient between the rotor side surfaces and air is calculated based on Equations 3-4 for each acceleration and brake event. [11]

$$hc = 3.974 \frac{Ka}{D} Re^{0.55} \tag{3}$$

$$Re = \frac{\frac{V_{avg}}{R_{tyre}} {R_{disc}}^2}{v} \tag{4}$$

where:

• Ka is air heat transfer coefficient,
• D is the rotor diameter,
• Re is the Reynolds number,
• V_{avg} is average speed of the vehicle for the event,
• R_{tyre} is the radius of the tyre,
• R_{disc} is the radius of the disc rotor
• v is air kinematic viscosity.

Values used in the calculations are listed in Table 3.

Table 3: Thermal analysis parameters.

Parameter	Value
m	95 kg
Ka	0.0294 W/(mK)
D	0.161 m
R_{tyre}	0.2 m
R_{disc}	0.0805 m
v	$1.93\ 10^{-5}\ kg/m.s$

3. Results and Discussion

The outcomes are divided into two sections: The “OptimumLap” vehicle simulation and “SolidWorks” thermal analysis. Table 4 presents a summary of the heat power for braking events and heat transfer coefficients for brake and acceleration events using vehicle speed results from “OptimumLap” outcomes for a single lap. Only the first four and last two braking and acceleration events are displayed on the table due to space restriction.

Table 4: Heat power and convection heat transfer coefficient results obtained from “OptimumLab” vehicle speed profile.

	Start time (s)	Start speed (m/s)	End time (s)	End speed (m/s)	Duration (s)	Average speed (m/s)	KE (J)	P (W)	Re	hc (W/m2.K)
brake 1	0.63	26.49	1.66	14.62	1.0260	22.96	23170.02	22582.87	38548.07	241.57
acceleration 1	1.68	14.40	3.78	19.80	2.1010	17.51	NA	NA	29393.36	208.11
brake 2	3.79	19.76	4.01	18.27	0.2250	19.11	2695.03	11977.90	32088.80	218.40
acceleration 2	4.02	18.16	6.41	26.69	2.3910	21.67	NA	NA	36384.25	234.02
brake 3	6.42	26.73	7.15	16.82	0.7270	21.43	20506.81	28207.44	35973.87	232.56
acceleration 3	7.16	16.68	9.56	29.57	2.4030	23.17	NA	NA	38894.09	242.76
brake 4	9.57	29.49	11.05	11.28	1.4820	24.00	35273.57	23801.33	40291.24	247.52
⋮ .	⋮ .	⋮ .	⋮ .	⋮ .	⋮ .	⋮ .	⋮ .	⋮ .	⋮ .	⋮ .
brake 34	128.50	23.45	129.77	10.44	1.2660	17.13	20952.82	16550.37	28757.28	205.62
acceleration 34	129.79	10.39	137.36	14.07	7.5683	11.43	NA	NA	19186.13	164.59
brake 35	137.37	13.93	137.81	7.71	0.4344	11.08	6392.32	14715.25	18596.21	161.78
acceleration 35	137.84	7.29	141.99	23.11	4.1555	16.86	NA	NA	28312.39	203.86

Using “SolidWorks”, a total of 70 braking and accelerating events were simulated inputting parameters such as power, heat transfer coefficient, duration and increment for each event using Table 4 values. Also, an ambient temperature of 298.15 K has been set for the first braking event in the first lap. Afterwards, for all the events after the first braking event, the temperature of the previous event was used as an input temperature for the current event. Simulations are continued until the temperature difference between two consecutive laps is less than 10 K (Figure 5). Figures 6 & 7 show the temperature distribution profile for the first braking event in the first lap and the first braking event in the second lap. Figure 8 shows the temperature profile for the maximum temperature the brake rotor reached (Lap2, Brake 15). It is noticed that the maximum temperature that the brake rotor can reach is 1,713 K which is higher than the MDT of 873 K for the brake rotor to operate properly. As a result, a modified design of the brake rotor has been made with bigger thickness value as shown in Figure 9 obtaining a maximum temperature of 847.7 K which is less than the MDT of 873 K.

Table 5: "SolidWorks" simulation results.

	Max Temp Lap 1 (K)	Max Temp Lap 2 (K)	Lap 2 and 1 Temp Difference (K)
brake 1	663.60	1211	547.40
acceleration 1	607.90	1073	465.10
brake 2	649.70	1104	454.30
acceleration 2	583.90	954	370.10
brake 3	897.80	1244	346.20
acceleration 3	781.20	1060	278.80
brake 4	1256.00	1499	243.00
⋮	⋮	⋮	⋮
brake 34	1446.00		
acceleration 34	1030.00		
brake 35	1118.00		
acceleration 35	899.90		

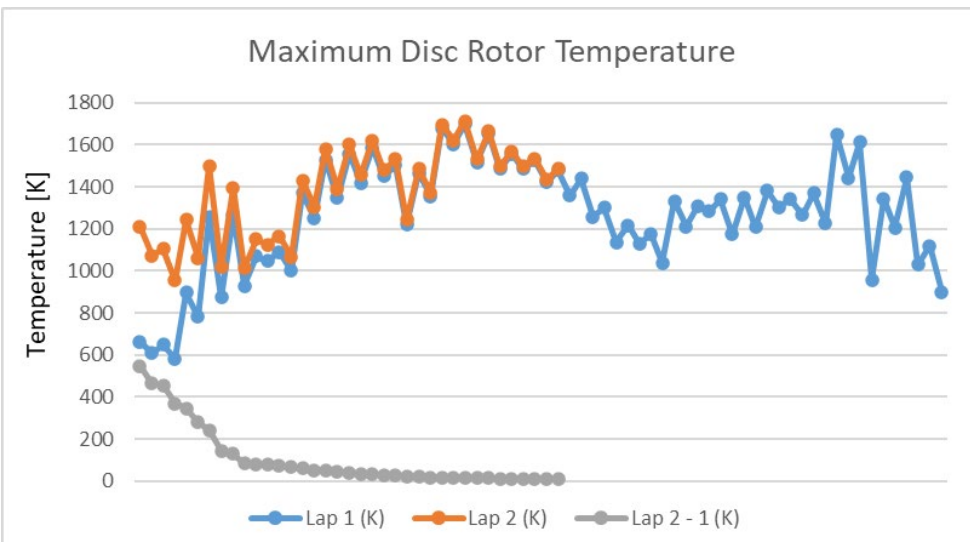

Figure 5: Maximum brake rotor temperature difference between consecutive laps.

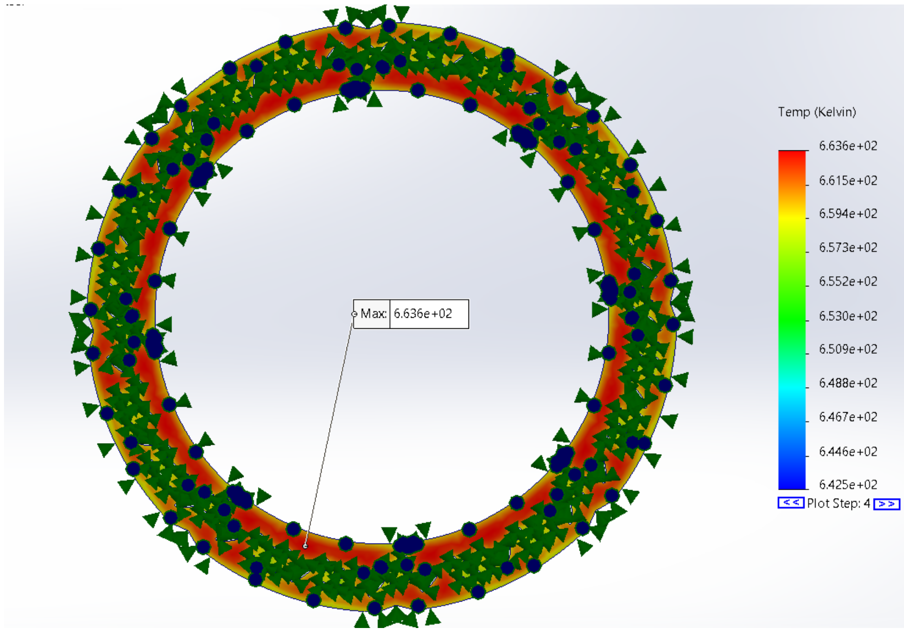

Figure 6: Temperature distribution of Lap 1 brake.

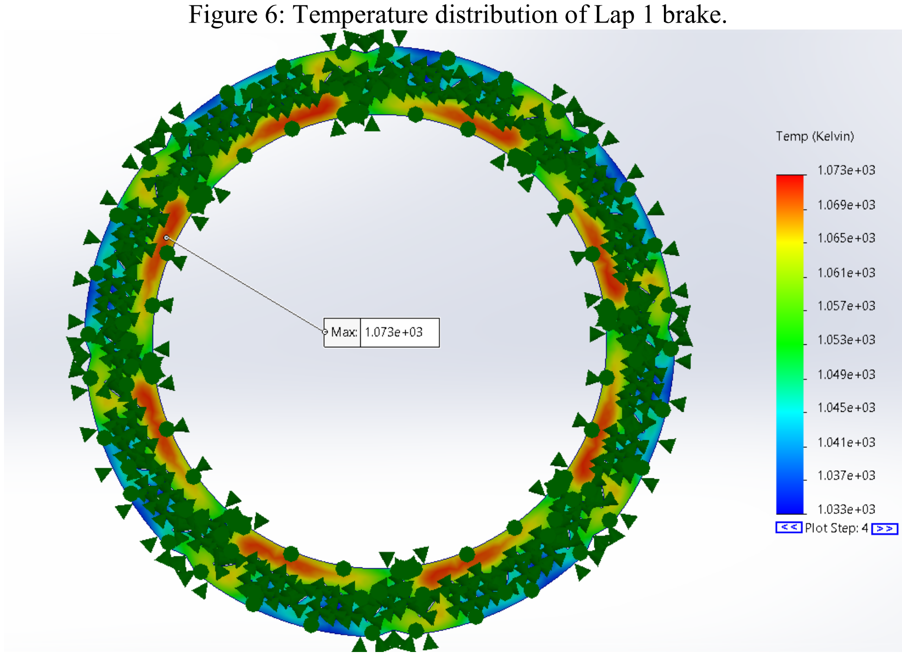

Figure 7: Temperature distribution of Lap 2 Brake 1.

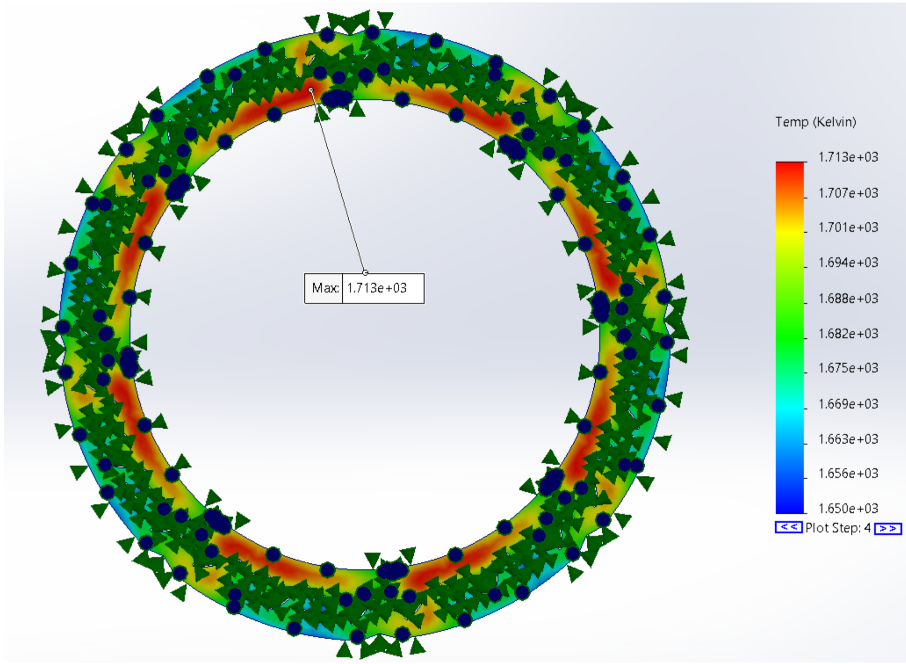

Figure 8: Temperature profile of thermal simulation of the maximum temperature (Lap 2, Brake 15)

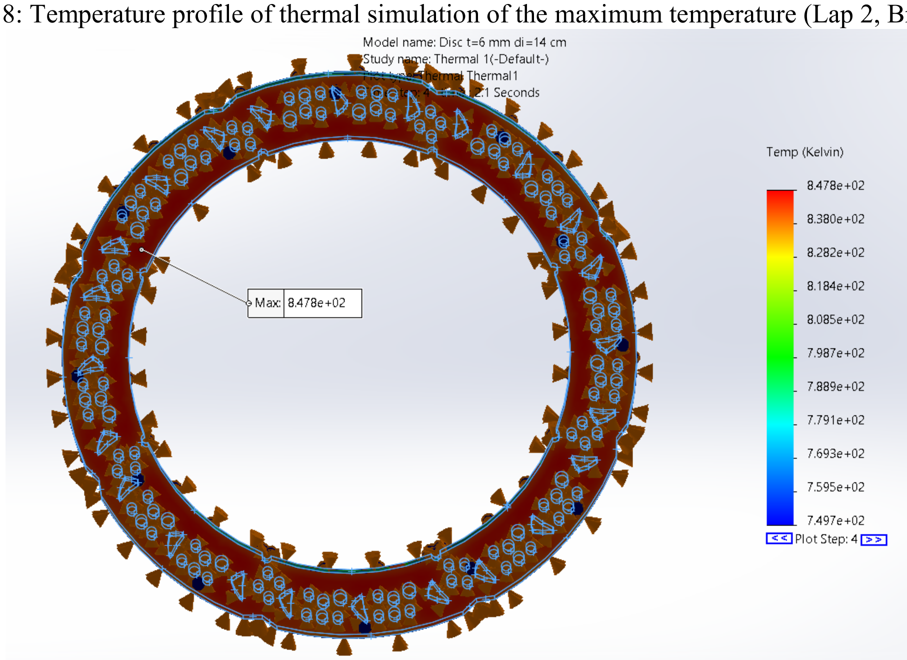

Figure 9: Temperature distribution profile of thermal simulation for the modified brake rotor design (Thickness = 6mm and inner diameter= 14 cm)

4. Conclusion and Future Work

One of the main goals for the disc brake design in the FSAE student contests is rotor dimension optimisation. To outperform its competitors, the majority of colleges develop and produce their own rotors. Better results will undoubtedly come from optimising the rotor particularly for the event that is being attended, as every race has different circumstances. When thermal analysis is considered, the worst-case scenario - sequential braking during dynamic events (such endurance and autocross) when the disc rotors reach their maximum temperatures - is used to establish the boundary conditions. Most of the prior research in the literature has relied on a single braking event, which is unrealistic given the worst-case situation in which acceleration and braking events occur consecutively. In order to optimise the design of disc rotors and account for subsequent braking and acceleration events in a racing scenario, this paper outlines a methodology. The vehicle and track pair are used to create the driving cycle for the race using "OptimumLap". Following that, the thermal simulation's input parameters are determined using the acceleration and deceleration events. Heat power is computed using the kinetic energy change that occurs during the braking event, and the heat convection coefficient is computed for both occurrences taking average velocities into account. Thermal simulations showed that the maximum temperature achieved is higher than the MDT. Thickness of the rotor is increased in order to reduce the temperature profile and prevent the MDT breach. For the future work, geometry optimization will be done for the disc brake rotor changing parameters such as inner diameter, thickness, and material type to obtain a maximum temperature less than the MDT of 873 K with weight optimization.

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