Design and Analysis of Braking System for Formula Student Car

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Abstract:- This paper discusses the components used for brake system and analysis of brake rotor. A brake is used for slowing or stopping a moving vehicle, typically by applying pressure to wheels. The material used for rotor is mostly stainless steel and sometimes grey cast iron, carbon-carbon composite in high-end cars or bikes. In addition, it is necessary to study stress distribution, heat generation for improve braking efficiency. The rotor is created in Solid Works and imported into ANSYS for analysis. Trident-racing team implemented this braking system for Supra 19 event organized by SAE India.

Keywords:- *SAE, Braking System, Solid Works, Ansys, Braking Efficiency, Analysis.*

I. INTRODUCTION

The Society of Automotive Engineers (FSAE) is an interdisciplinary competition organized by the Society of Automotive Engineers (SAE). It challenges university students (graduate and undergraduate students) from all over the world and provides them with their own car design, construction and competition. High performance, durability and reliability must be the attributes of this car. The car is not only tested under dynamic racing conditions, but also based on its design, functionality, marketability and cost. The design aims to ensure an effective and efficient braking system. A brake is a device that is applied to a moving machine part through artificial frictional resistance to stop the machine's movement, thereby stopping the machine's movement. In the process of performing this function, the brake absorbs the kinetic energy of moving parts or the potential energy released by descending objects such as elevators and elevators. The energy absorbed by the brake is dissipated in the form of heat. This heat will dissipate into the surrounding atmosphere to stop the vehicle. Therefore, the braking system should have the following requirements:

The brakes must be strong enough to stop the vehicle at a minimum distance in an emergency.

a) The driver must correctly control the vehicle when braking, and the vehicle must not skid.
b) The brake must have good anti-fading performance, that is, its effectiveness should not be reduced under continuous long-term use.
c) The brake should have good anti-wear performance.

If there is no braking system, the vehicle will place passengers in an unsafe position. Therefore, all vehicles must have proper braking.

II. DESIGN OBJECTIVE AND OVERVIEW

The main purpose of the braking system is to convert the kinetic energy of the car into heat energy, thereby reducing the car's performance. The brake system is designed as a hydraulic system, consisting of two cylinders, one that makes the first two tire brakes, and the other that has two rear tire brakes. Each master cylinder has two adjustable straps, one caliper per wheel, four system calipers, and four rotors or brake discs. The process of the braking system is as follows: the driver applies power to the brake pedal, and the brake station operates forcefully on the main cylinder, thus delivering the brake fluid to the master cylinder. After that, the extracted liquid is pressurized on each caliper, causing the caliper piston to apply gripping force to the Rotor. Therefore, the input of the system is the foot force applied by the driver, and the output is the grip strength applied to the Rotor by wire. Depending on the rules and regulations of the competition, the design of the brake system must be able to completely close all four-car tires in the event of an emergency brake. Therefore, in this type of compression design, the main purpose is to design a system so that the driver can apply a lot of energy but not excessively to the brake structure to completely lock the car tires. The system is designed for foot capacity of approximately 2500N.

III. DEFINITION OF PROBLEM

The goal / specificity of the current task is to design and analyze the disc rotor made with SS410. SS materials are used to design the disc rotor. After that, the Rotor will be upgraded according to certain parameters to get the best design. The Rotor was created in Solid Works and submitted to ANSYS for analysis.

IV. SELECTION OF COMPONENTS

A. Brake rotor and Calipers:

The important requirement of the brake disc is,

- It should provide a surface with good anti-wear properties.
- It should allow the best heat transfer rate.

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- Each cycle of brake application generates heat, which must be dissipated to the atmosphere.
- Sufficient strength and minimal weight.
- It must be contained in the smallest space available.

It is helpful to choose a rotor that is as wide as possible. This is because the distribution of the same torque, as the size increases, the proper power decreases. Therefore, a Rotor with a diameter of 160 mm is preferred for a 10-inch rim. Rotor is attached to promote heat transfer. The size of the disc is 3mm.


Fig 1. Drilled Rotor

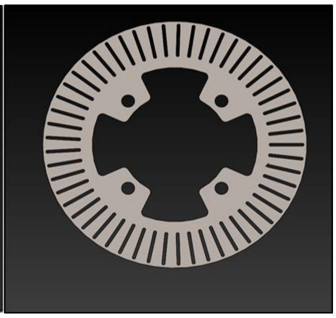
Fig 2. Slotted Rotor

TABLE I . SPECIFICATION OF ROTOR

Material	**Stainless Steel 410**
Diameter	160 mm
PCD	80 mm
Thickness	3 mm

B. Master Cylinder:

Two master cylinders have been selected, to obtain two independent circuits for compression, and can be obtained with a single control of the brake base with a balance bar. It contains DOT3 as brake fluid. The circuit consists of solid pipes followed by flexible brake tubes connected to the drivers by Benjo bolts.

C. Brake Pedal:

The base of the brake is mechanically made from 6061 aluminum plate with two cylinders. Its design can withstand the force of 2000N on a ladder. The ratio of the pedal is set to 6: 1.

V. CALCULATIONS

A. Stopping Distance:

All calculations take into account an initial speed of 40 km/h and a final speed of 0 km/h. During braking, the weight will shift due to the increased load on the front axle.

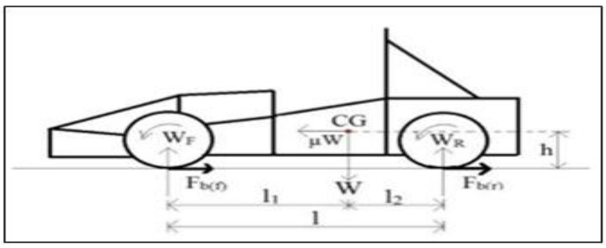

Fig 3. FSAE Car Diagram [5]

By considering the moments of the rear and front wheels, we obtain the following respective dynamic load transfer equations,

$W_{Front} = W\frac{l_2}{l} + \frac{h}{l}F_b$; $W_{Rear} = W\frac{l_1}{l} - \frac{h}{l}F_b$;

The maximum braking force provided between the road and the tire is given by the following formula,

$$F_{bmax} = \mu W = W\left(\frac{d}{g}\right)$$

$$F_{b(f)} = W_f\left(\frac{d}{g}\right) \; ; \quad F_{b(r)} = W_r\left(\frac{d}{g}\right)$$

Where,
W= Weight of the vehicle.
F_b = Braking Force (total).

The Equations are for ideal condition where braking efficiency is 100%
Braking Efficiency:

$$\eta = \left(\frac{d}{g}\right) \times \left(\frac{1}{\mu}\right)$$

Now, consider braking efficiency = 85%

$$\left(\frac{d}{g}\right) = 0.85\mu$$

Hence, braking forces at axles are:

$$F_{b(f)} = W_f\left(\frac{d}{g}\right) = 0.85\mu \times \left[W\frac{l_2}{l} + \frac{h}{l}0.85\mu W\right]$$

$$F_{b(r)} = W_r\left(\frac{d}{g}\right) = 0.85\mu \times \left[W\frac{l_1}{l} - \frac{h}{l}0.85\mu W\right]$$

Considering the values,
l= 1.565m;
l_1= 0.890m;
l_2= 0.675m;
h= 0.300m;
W= 270×9.81 = 2648.7 N

Coefficient of friction (μ) depends on several factors.
Let us assume its value =μ= 0.67.

$$F_{b(f)} = 0.85\mu \times \left[W\frac{l_2}{l} + \frac{h}{l}0.85\mu W\right]$$

$$F_{b(r)} = 0.85\mu \times \left[W\frac{l_1}{l} - \frac{h}{l}0.85\mu W\right]$$

$F_{b(f)}$ = 815.278 N
$F_{b(r)}$ = 693.157 N
F_b = 0.85μW = 1508.435 N

Percentage biases on front and rear wheel respectively are:

$Kb(f) = \frac{(F_{bf})}{F_b} = 0.540$; $Kb(r) = \frac{(F_{br})}{F_b} = 0.460$

Braking torques at front and rear wheels respectively are:

$$T_F = \frac{F_{bf}}{2} \times R_{wheel} = 407.638 \times 0.165 = 67.260 \text{ Nm}$$

$$T_R = \frac{F_{br}}{2} \times R_{wheel} = 346.579 \times 0.165 = 57.186 \text{ Nm}$$

Where,
$R_{wheel} = 0.165$ m

a) Calculation for master cylinder:

Considering that, the driver applies pedal force (F_p)250 N on a brake pedal having advantage 6:1
Force on the push rod of master cylinder $= 6 \times 250 = 1500$ N

The diameter of master cylinder is 19.08mm.

$P_{area} = \left(\frac{\pi}{4}\right) \times 19.08^2 = 285.9 \text{ mm}^2 = 2.859 \times 10^{-4} \text{ m}^2$

Pressure in master cylinder $= \frac{\text{Force on the push rod of master cylinder}}{\text{Area of piston } (P_{area})}$

$$= 5.246589 \times 10^6 \text{ Pa}$$
$$= 5.246 \text{ MPa}$$

b) Calculation for Caliper:

The diameter of piston = 32 mm

Force at calliper = Pressure in master cylinder× Area of piston Calliper × No. of pistons

$$F_{caliper} = 5.246 \times 804.248 \times 2$$
$$F_{caliper} = 8438.170\ N$$

Clamping Force = 2 × force on caliper
=16876.340 N

c) Calculation for disc:

Let the co-efficient of friction between the disc and pad (μp) in the brake calliper be 0.3

$F_{(friction)} = 2 \times \mu_p \times F_{caliper}.$
$= 2 \times 0.3 \times 8438.170$
$= 5062.902\ N$

Also,

$$r_{(effective)} = \frac{r_1 + r_2}{2} = \frac{80 + 60}{2} = 70\ mm = 0.070\ m$$

Where,
r_1 = Radius of circle formed by inner edge of friction pad.
r_2 = Radius of circle formed by outer edge of friction pad.

Now,
$Torque = F_{(friction)} \times r_{(effective)}$
$= 5062.902 \times 0.070$
$= 354.403\ Nm$

Therefore, the pedal power of 250N can provide enough torque of the brakes on all lock wheels.

The statistical reduction used in the calculation is a stable condition called MFDD (low rate of total reduction). It determines whether the car breaks down or not. In practice, it takes a while for system pressure to increase and a conflict to form. This is not the driver's response time, but the system's response time. If the calculation requires a distance stop or a reduction between stops, this delay should be considered. According to the calculation, a delay of 0.2 seconds.

$$a_{ave} = \frac{v}{\left[\left(\frac{v}{a}\right) + 0.2g\right]}$$
$$= \frac{11.11}{\left[\left(\frac{11.11}{1}\right) + (0.2 \times 9.81)\right]}$$
$$= 0.85$$

Where,
a_{ave} = Average deceleration for the whole stop (g unit).
v = Test speed (m/sec).
a = Deceleration (MFDD) (g units).
g = Acceleration due to gravity (m/sec^2).

Stopping distance,

$$s = \frac{v^2}{(2g \times a_{ave})} = \frac{(11.11)^2}{(2 \times 9.81) \times (0.85)} = 7.4\ m$$

Where,
s = Stopping distance (m).
a_{ave} = Average deceleration for the whole stop (g unit).
v = Test speed (m/sec).
g = Acceleration due to gravity (m/sec^2).

B. Thermal Considerations:

Kinetic Energy is given by,

$$KE = \frac{1}{2} \times M \times v^2 = \frac{270 \times (11.11)^2}{2} = 16663.33 \text{ J}$$

Where,
KE = Kinetic energy (J).
M = Total vehicle mass (Kg).
v = Test speed (m/sec).

Now,
Brake on time,

$$t = \frac{v}{a \times g} = \frac{11.11}{1 \times 9.81} = 1.133 \text{ sec}$$

Where,
t = Brake on time (secs).
v = Test speed (m/sec).
a = Deceleration (g units).
g = Acceleration due to gravity (m/sec^2).

Now,
Braking power is given by,

$$BP = \frac{KE}{t} = \frac{16663.33}{1.133} = 14746.309\ W$$

Then,

$$Area = \frac{\pi(D^2 - d^2)}{4} = 0.002199\ m^2$$

Where,
D = Disc useable outside diameter (m).
d = Disc useable inside diameter (m).

Now,
We know that, Heat flux is given by,

$$q = \frac{BP}{Area} = \frac{14746.309}{113.682} = 67{,}05{,}915.8708\ W/m^2$$

Film Co-efficient,

$$P_r = \frac{C_p \times \mu_v}{k} = \frac{1007 \times 1.983 \times 10^{-5}}{0.024} = 0.832034$$

Where,
C_p = Specific heat of air at constant pressure.
μ_v = Dynamic viscosity of air.
k = Thermal Conductivity of air.

Now,

$$Re = \frac{V \times x}{v} = \frac{11.11 \times 2 \times \pi \times 0.080}{2 \times 10^{-5}} = 296676.302.$$

Where,
V = Velocity of air = 40kmph = 11.11 m/s
x = Distance travelled by air = $2\pi r = 2\pi 0.080$
r = radius of disc = 80mm = 0.080m
ν = kinematic viscosity = 2×10^{-5} m^2/s

Film heat transfer coefficient,

$$h = 0.04 \times \frac{K_a}{D_o} \times Re^{0.8} = 148581.8089\ W/m^2K$$

Where,
Re = Reynolds number.
K_a = Thermal conductivity of material.
D_o = Outer diameter of disc.

VI. FINITE ELEMENT ANALYSIS

A. Static structural analysis:

In this project, structural and thermal analysis of two disc rotors, i.e., drilled disc rotor and a slotted disc rotor are performed to compare the results and derive inferences and conclusions. However, the same methodology followed for the analysis of both the rotor.Steady state thermal analysis coupled with static structural analysis was performed in Ansys 19. The CAD model of rotor was created in Solidworks 2019 then was exported for analysis.

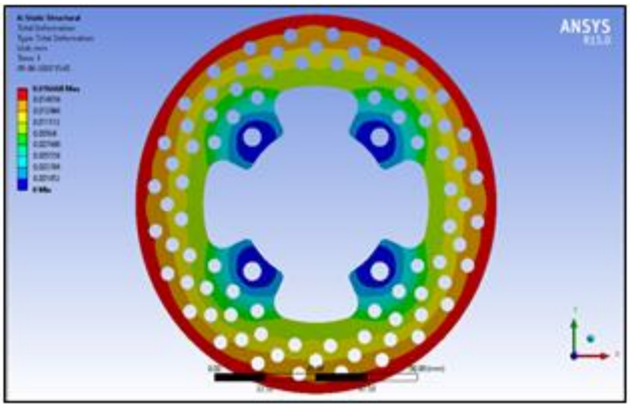

Fig 4. Total Deformation in Drilled Disc Rotor

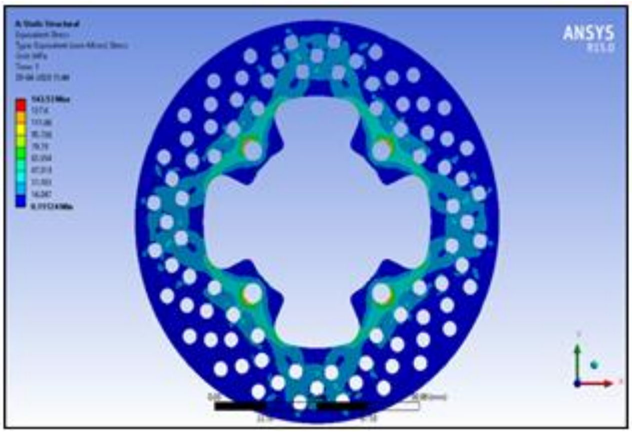

Fig 5. Equivalent Stress in Drilled Disc Rotor

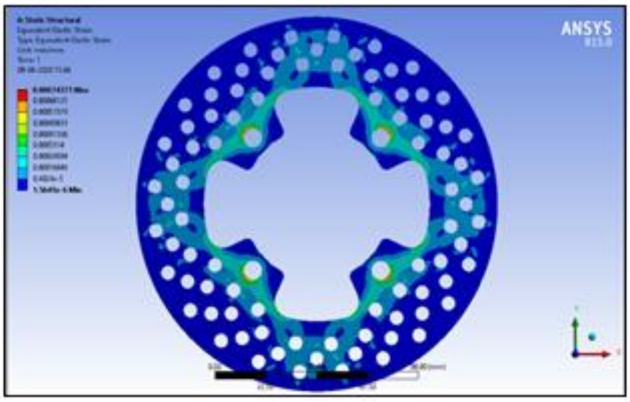

Fig 6. Equivalent Strain in Drilled Disc Rotor

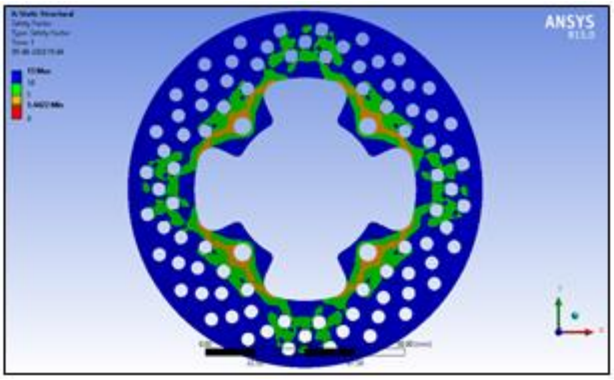

Fig 7. Safety Factor in Drilled Disc Rotor

Object Name	Total Deformation	*Equivalent Stress*	*Equivalent Elastic Strain*	*Safety Factor*
Minimum	0. mm	0.11124 MPa	1.5641e-006 mm/mm	1.4422
Maximum	1.6668e-002 mm	143.53 MPa	7.4371e-004 mm/mm	

TABLE II. RESULTS FOR STATIC STRUCTURAL ANALYSIS FOR DRILLED DISC ROTOR

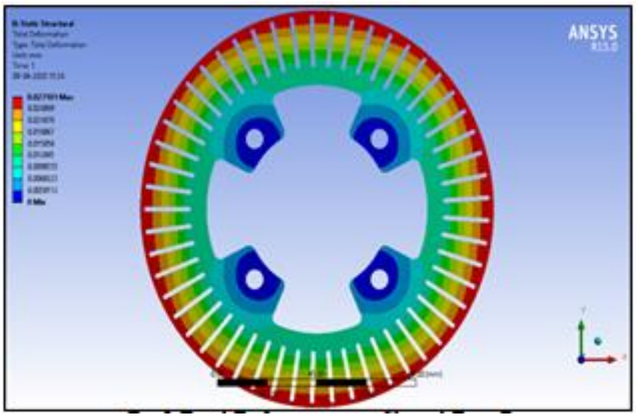

Fig 8.Total Deformation in Slotted Disc Rotor

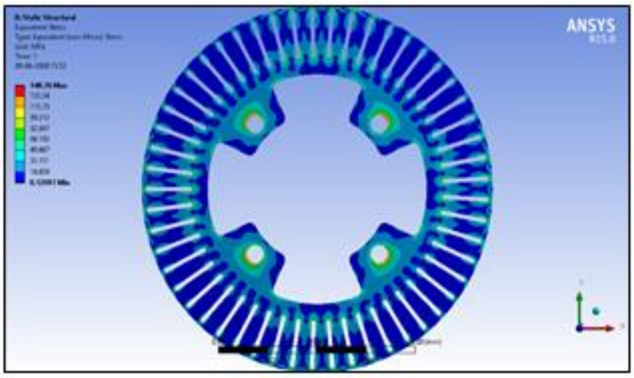

Fig 9. Equivalent Stress in Slotted Disc Rotor

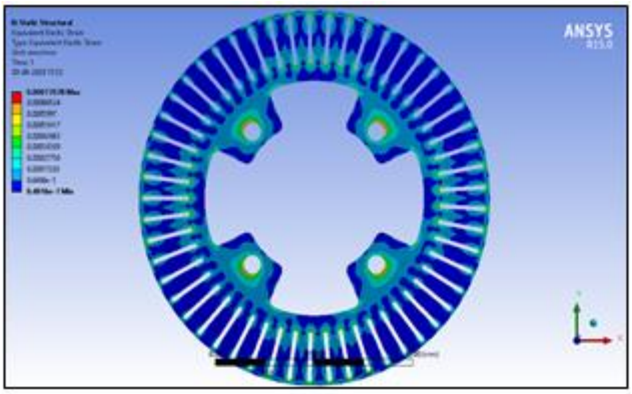

Fig 10. Equivalent Strain in Slotted Disc Rotor

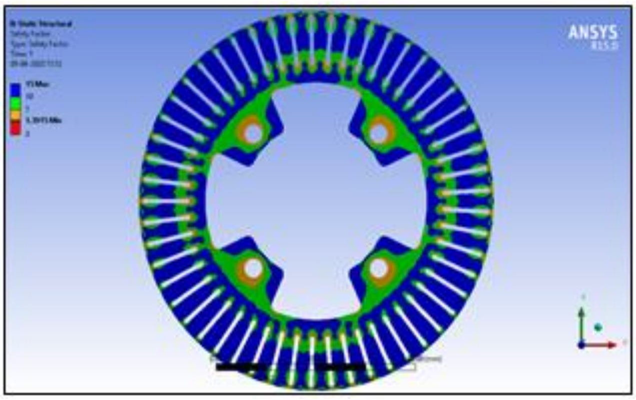

Fig 11. Safety Factor in Slotted Disc Rotor

Object Name	Total Deformation	*Equivalent Stress*	*Equivalent Elastic Strain*	*Safety Factor*
Minimum	0. mm	0.12097 MPa	9.4916e-007 mm/mm	1.3915
Maximum	2.7101e-002 mm	148.76 MPa	7.7078e-004 mm/mm	

TABLE III. RESULTS FOR STATIC STRUCTURAL ANALYSIS FOR DRILLED DISC ROTOR

B. Transient Thermal Analysis:

Temperature changes in rotor geometry are calculated and analyzed here. As the brakes are applied to the disc brake rotor of the car, the friction will produce heat, so the temperature generated must be driven and spread across the cross section of the disc rotor. Braking conditions are very difficult, so a hot analysis should be done. Temporary thermal analysis determines the distribution of temperature under conditions of high load.

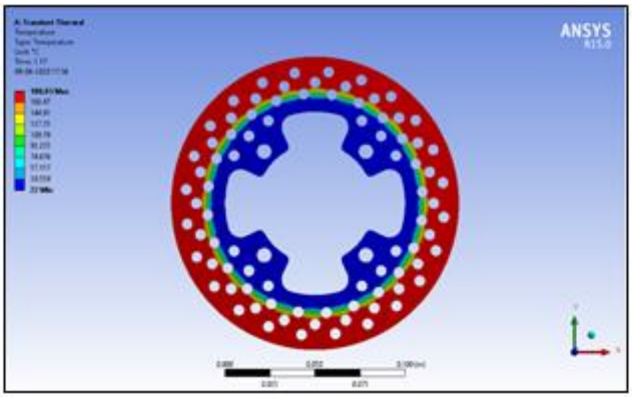

Fig 12. Temperature distribution in Drilled Disc Rotor

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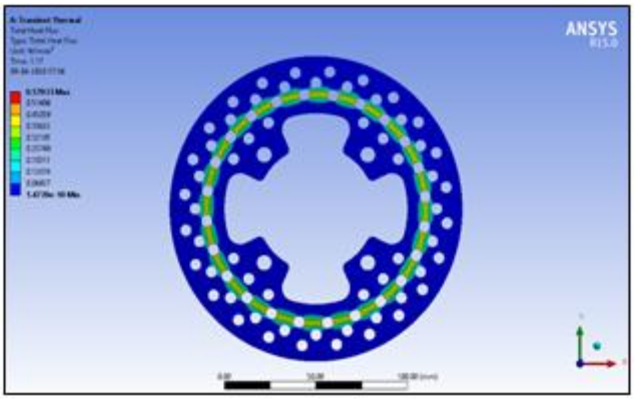

Fig 13. Total heat flux in Drilled Disc Rotor

Object Name	Temperature distribution	Total heat flux
Minimum	22. °C	1.4739e-010 W/mm²
Maximum	180.03 °C	0.57933 W/mm²

TABLE IV. RESULTS FOR TRANSIENT THERMAL ANALYSIS FOR DRILLED DISC ROTOR

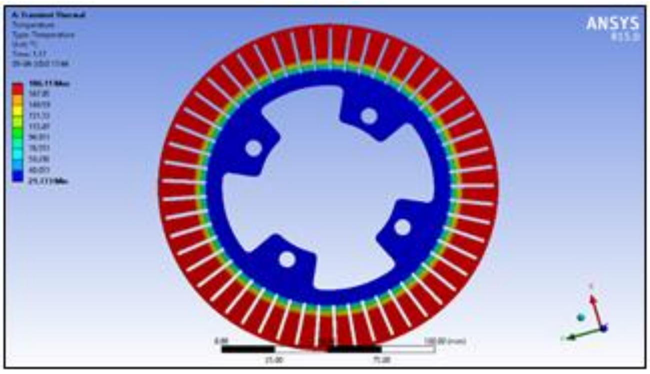

Fig 14. Temperature distribution in Slotted Disc Rotor

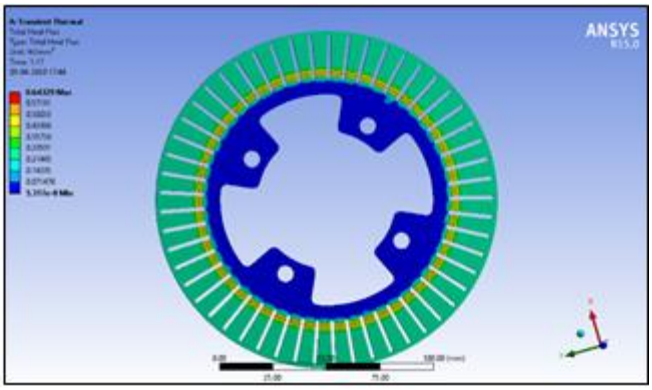

Fig 15. Total heat flux in Slotted Disc Rotor

	Drilled disc rotor		Slotted disc rotor	
	Minimum	Maximum	Minimum	Maximum
Total Deformation .(mm)	0	1.6668e-002	0	2.7101e-002
Equivalent Stress (MPa)	0.11124	143.53	0.12097	148.76
Equivalent Elastic Strain(mm/mm)	1.5641e-006	7.4371e-004	9.4916e-007	7.7078e-004
Temperature (^0C)	22	180.03	22	186.11
Total Heat Flux (W/mm²)	1.4739e-010	0.57933	5.317e-008	0.64329

TABLE V. RESULTS FOR TRANSIENT THERMAL ANALYSIS FOR DRILLED DISC ROTOR

VII. CONCLUSION

A rotated and molded disk rotor with a diameter of 160 mm and a diameter of 3 mm is made using computer-assisted software Solidworks and analyzed in ANSYS analysis software. Table VI summarizes the observations of this calculated analysis. Based on research and results, it can be seen that the rotor used and the molded rotor are safe for construction, as the total pressure is within the range of the SS 410 material. Used with a rotating rotor are 0.027101 mm and 0.016668 mm, respectively. The temperature change and temperature of the two rotors are almost the same, but with the molded rotor, the total temperature and total ambient temperature is very small due to the increase in the thermal conductivity during braking due to drilling.

Object Name	Temperature distribution	Total heat flux
Minimum	22. °C	5.317e-008 W/mm²
Maximum	186.11 °C	0.64329 W/mm²

TABLE VI. RESULT SUMMARY

As a result and inclination, it is not bad that a drilled rotor has better performance than a slotted rotor due to reduced pressure, stiffness, total twisting and thermal stability. Therefore, drilled rotors are often used in more efficient vehicles such as sports cars, ATVs and UTVs. However, drilled rotors are usually weak and sometimes difficult to produce. Therefore, standard streetcars prefer solid rotor.

VIII. FUTURE SCOPE

After construction and analysis are completed, as indicated in this report, there is always room for improvement and improvement of the project based on existing results. The design can be further developed and refined to reduce total pressure, modification and temperature changes. Because of continued design efficiency, excellent design can be achieved depending on the size change and actual use.

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