Single Stop Transient Thermal Coupled with Structural Analysis and Repeated Braking Analysis with Convective Cooling for Ventilated Rotor

Tirth Sodagar[1], Avadhoot Ahire[2]

[1]*Tirth Sodagar, Student, School Of Mechanical Engineering, Vellore Institute of Technology, Vellore, India*
[2]*Avadhoot Ahire, Student, School Of Mechanical Engineering, Vellore Institute of Technology, Vellore, India*

Abstract – *The motive of undertaking this project is to first use quantitative comparison methods to determine the best suitable material for carrying out the analysis of the ventilated disc brake. Numerical analysis is carried out in excel sheet to determine the type of flow associated with the rotor, calculating the mass flow rate of the air through the rotor and ultimately determining the heat transfer convection coefficient of the brake rotor. The reason behind determining the convection coefficient was because it will serve as an input for carrying out single stop transient thermal analysis coupled with static structural analysis of a ventilated rotor which will be undergoing convective cooling. Repeated thermal braking analysis with convective cooling has been done to determine the temperature distribution of the brake rotor after a particular number of brake applications. The rotor was modeled in Solidworks and Ansys as well as excel spreadsheet was used to carry out the analysis. At the time of repeated braking and single stop braking the brake rotor undergoes severe thermal and mechanical stress which could result in warping of the brake rotor. Hence it is highly essential to determine the structural deformation, stress concentration and the thermal gradient of the disc brake. In Ansys thermal, First the thermal analysis is run to determine the maximum temperature and thermal distribution during single stop braking and then static structural analysis is performed to see the effect of the thermal load as well as mechanical braking forces that are generated by the brake calipers at the time of braking.*

Key Words: convective cooling, repeated braking, ventilated rotor, transient analysis, structural analysis, Ansys

1.INTRODUCTION

There are basically two types of brake disc –

- Solid disc
- Ventilated brake disc

In the earlier days solid disc brakes were used but with the enhancement in the automotive technology new type of brake rotors have been developed to replace the conventional solid brake rotors which could provide better cooling. In ventilated disc brake there are two braking surfaces known as the inboard and the outboard surface which are joined together by vanes.

The working of the ventilated disc brake could be compared with a centrifugal impeller. Air flows in radial directions through the vane passages because of the action of the centrifugal force.

Advantages of the ventilated rotor over the solid rotor is that it has better heat dissipation from the rotor because of the flow of air in between the vanes and since the ventilated rotors provide an increase in surface area available for heat transfer as compared to solid rotor. The ventilated disc brake also exhibits higher convection coefficient than solid disc brake approximately twice the value for the solid disc brake.

1.1 Brake System Design Considerations

Here we list out different parameters which are very essential while designing the brake system and that have been kept in mind for carrying out the design, calculations, and analysis

- Heat transfer convection coefficient of the disc brake
- Braking temperatures attained during single stop and at the time of repeated braking
- The amount of thermal stresses developed in the rotor to avoid cracking or warping of the disc
- Heat flux into the rotor
- Clamping force and the frictional braking force produced by the brake calipers and the resultant mechanical stresses

When the caliper pads come in contact with the disc, the kinetic energy of the vehicle is converted to thermal energy and the heat produced is dissipated to the surrounding. There will be a rise in temperature due to the generation of frictional heat between the rotor and caliper pads. If the temperature exceeds the critical value for a given material, the following could take place.

- Brake fail
- Premature wear
- Vaporization of fluid
- Thermal cracking
- Buckling or coning of the disc

The calculations and analysis was performed considering the data from the official Formula Hybrid Student car of VIT, Vellore.

2. MATERIAL SELECTION

A quantitative comparison method for material selection was carried out to determine which material was to be used for the brake disc.
The materials that we have used for investigation are

- Grey cast iron(GCI)
- Titanium alloy
- Stainless steel 420
- Alloy steel AISI 4140

A decision matrix has been proposed to choose appropriate material. The following properties were considered

Table -1:

Weighing factor for different properties	
Properties	**Weighing factor**
strength	0.9
Friction coefficient	0.8
Wear rate	0.7
Specific gravity	1
Specific heat	1
Thermal conductivity	0.9

Each property was scaled out of 100, relative to the best value of the property in the list.

Table -2:

Materials scaled properties						
Materials	**1**	**2**	**3**	**4**	**5**	**6**
GCI	97	100	100	63	79	100
Titanium alloy	80	83	1	100	100	28
SS 420	100	98	98	58	79	52
AISI 4140	70	98	98	58	83	93

Each of these scaled properties was multiplied by their weighting factor and the total points were calculated

Table -3:

Performance index							
Material	**1**	**2**	**3**	**4**	**5**	**6**	**Total**
GCI	87	80	70	63	79	90	469
Titanium alloy	72	66	1	100	100	25	365
SS 420	90	78	69	58	79	47	421

The performance index of each of the materials was calculated and is shown in the following graph.

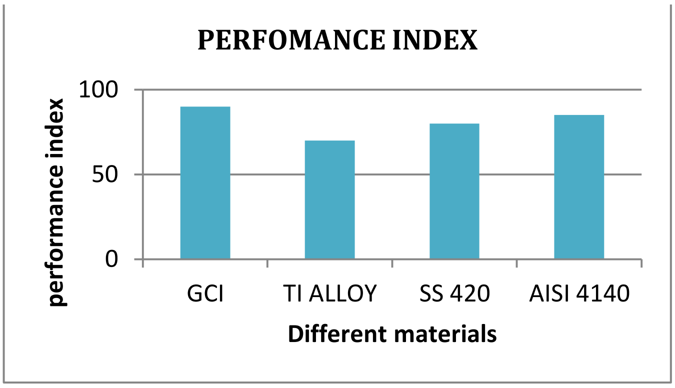

Chart -1: Performance index of different materials

Inference
From the graph, we can say that Grey cast iron has the best performance index followed by AISI 4140. Hence with the help of quantitative analysis done we have reached a conclusion that grey cast iron is the best suitable material for the ventilated brake rotor.

Table -4:

Properties of Grey cast iron		
	Value	**Abbreviation**
Density	7288 kg/m3	ρ_R
Specific heat	419 Nm/kg K	c_R
Melting temperature	1473 K	T_m

3. MODELING OF THE VENTILATED DISC

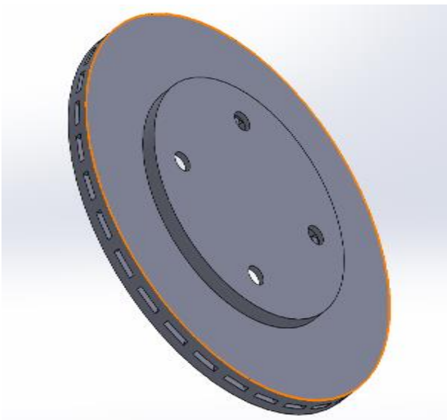

Fig -1: Model of ventilated disc

Table -5:

Specifications of the disc		
Specification	**Value**	**Abbreviation**
Outer diameter	200 mm	D_0
Inner diameter	120 mm	D_i
Effective diameter	160 mm	D_e
Length of cooling vanes	40 mm	l
Surface area	5100 mm²	A
Volume	260.8 mm3	v_R
Mass	2.0 kg	m
Fin thickness	5 mm	T_f
Flange thickness	5 mm	t_f
Rotor thickness	13 mm	T_r
Vane inlet area	25.176 mm²	A_{in}
Vane outlet area	41.992 mm²	A_{out}

Determining the total number of vanes for the ventilated brake rotor

- Total number of vanes

$$n_v = \frac{4\pi D_0}{D_0 - D_i} = 31.88$$

Since the value has to be a whole number we take it as 32. Hence the total number of vanes in the ventilated brake rotor is 32.

4. CALCULATING THE CONVECTION COEFFICIENT OF THE VENTILATED DISC BRAKE

For temperature analysis of the disc brake it is foremost important to calculate the convection heat transfer coefficient, which varies with the vehicle speed. The assumptions while calculating the convection coefficient are –

- heat transfer convection coefficient is evaluated at the mean speed.
- the equations will be applied to the disc that is not obstructed by the tire and rim or disc caliper
- the effect of radiation is neglected

The following set of equations have been used to obtain the heat transfer coefficient inside the vanes of the brake rotor –

Firstly, the warping temperature of the material is obtained. The warping temperature is taken to be 70% percent of the melting temperature of the material

- melting temperature of cast iron = 1473 K
- warping temperature = 0.7 * 1473 = 1031.1 K

The ambient temperature of the surrounding is taken as 300K
The properties of air were taken at film temperature [1]

$$\textit{Film temperature} = \frac{\textit{Warping temperature} + \textit{Ambient temperature}}{2}$$

= 665.55 K

Table -6:

Properties of air at 665.55 K		
	Value	**Abbreviation**
Thermal conductivity	0.05572 kW/m K	k_a
Density of air	0.4565 kg/m³	ρ_a
Prandtl's number	0.698	Pr
Dynamic viscosity	3.56*10⁻⁵ kg/m s	ν

- Calculation of hydraulic diameter (d_h)[2]

$$d_h = 4\frac{\text{wetted area}}{\text{wetted perimeter}}$$

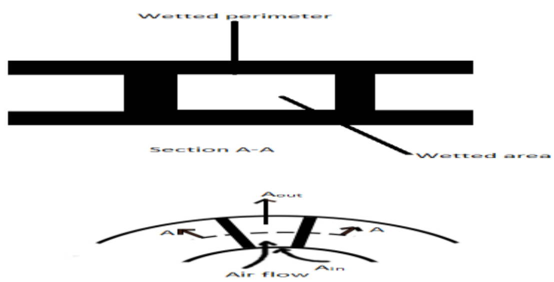

Fig -2: Wetted perimeter and wetted area

The hydraulic diameter is calculated at the effective rotor radius.

- Vane width (w_v)

$$w_v = T_r - 2t_f$$
$$w_v = 3\ mm$$

- Vane circumference (c_v)

$$c_v = \frac{\pi\,\mathrm{De}}{n_v} - T_f$$
$$c_v = 10.7\ mm$$

- Wetted perimeter (P_w)
  P_w = 2 (vane width) + 2 (vane circumference)

$$P_w = 2w_v + 2c_v = 27.4\ mm$$

- Wetted area (A_w)
  A_w = Vane width * Vane circumference

$$A_w = w_v * c_v$$
$$A_w = 10.7\ mm^2$$

∴Hydraulic diameter = 1.562 mm

- Velocity through the inlet and outlet vanes

The velocity through the inlet and outlet vanes is not the same as the velocity of the vehicle
The inlet and outlet velocity through the vane is determined by [2]

$$V_{in} = 0.0158 n_T (D_0^2 - D_i^2)^{\frac{1}{2}}$$
$$V_{in} = 49.92\ m/s$$

$$V_{out} = V_{in}({}^{A_{in}}/_{A_{out}})$$

$$V_{out} = 29.93\ m/s$$

The average velocity through the cooling vanes

$$V_{avg} = \frac{V_{in} + V_{out}}{2}$$

$$V_{avg} = 39.925\ m/s$$

- Mass flow rate(m_a) [2]

$$m_a = 0.00147 n_T \left((D_o^2 - D_i^2) A_{in}\right)^{\frac{1}{2}}$$

$$m_a = 0.076172\ m^3/s$$

Where,
n_T = revolutions per min of disc = 6000 rpm

Calculation of Reynolds number
If the Reynolds number is greater than 10^4, the flow is turbulent and if the Reynolds number is less than 10^4 the flow is laminar

- Reynolds number (Re)

$$Re = \frac{\rho_a d_h V_{avg}}{\nu}$$

$$Re = 2299.48$$

Since the Reynolds number calculated is less than 10^4, the flow of air through the disc for the car is laminar.
For laminar flow, the convection heat transfer coefficient is given by [3]

- Convective heat transfers co-efficient (h_R)

$$h_R = 1.86 (Re\ \mathrm{Pr})^{\frac{1}{3}} \left(\frac{d_h}{l}\right)^{0.33} \left(\frac{k_a}{d_h}\right)$$

$$h_R = 131.372\ W/m^2K$$

5. TEMPERATURE CALCULATION FOR REPEATED BRAKING OF THE VEHICLE

In the case of repeated braking, the vehicle is taken to the maximum speed of 120 km/hr. and then decelerated to a zero speed, again accelerated to the test speed after which the next braking cycle is carried out
Assumptions taken while computing temperature rise after nth application of brakes:

- Braking power remains unchanged during the brake test
- cooling intervals remain unchanged during the brake test
- braking times remain unchanged during the brake test
- the rotor is treated as a lumped system
- heat transfer coefficient and thermal properties are constant

The average temperature increase per stop (ΔT)

$$\Delta T = \frac{q_0 t_s}{\rho_R c_R v_R}$$

The relative brake temperature after the nth application is [4]

$$[T(t) - T_\infty] = \frac{\left[1 - e^{\frac{-n_a h_R A_R t_c}{\rho_R c_R v_R}}\right][\Delta T]}{1 - e^{\frac{-h_R A_R t_c}{\rho_R c_R v_R}}}$$

T_c= Cooling time cycle time for the vehicle and is assumed to be 55 seconds.
The above equation was written in excel sheet and temperature rise after 20 brake applications was calculated as well as graph was plotted for number of brake applications vs. relative temperature rise after each brake application

Table -7:

Final temperatures after nth brake application		
Number of brake applications	**Relative brake temperature rise(K)**	**Final temperature(K)**
1	46.53	346.53
2	71.95	371.95
3	85.83	385.83
4	93.41	393.41
5	97.55	397.55
6	99.82	399.82
7	101.05	401.05
8	101.73	401.73
9	102.3	402.3

10	102.41	402.41
11	102.47	402.47
12	102.5	402.5
13	102.52	402.52
14	102.53	402.53
15	103	403
16	103	403
17	103	403
18	103	403

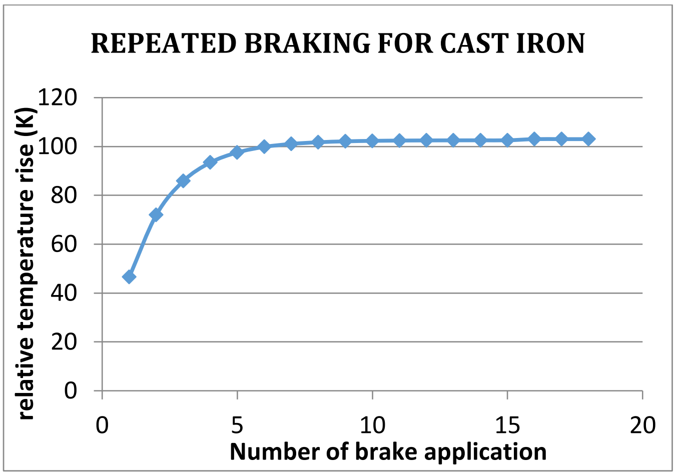

Chart 2 : Relative temperature vs number of brake application

Results from the graph reveals that after the 15th application of the brake the temperature does not increase and hence we can conclude from the results that final brake temperature is reached after the 15th brake application for the vehicle.

6. BRAKE CALCULATIONS

Table -8:

Vehicle data		
	Value	**Abbreviation**
Mass of the car	250 kg	m
Static front load (%)	40 %	n_f
Static rear load(%)	60 %	n_r
Front weight	980 N	w_f
Rear weight	1470 N	w_r
Wheelbase	1.524 m	l_w
Center of gravity height	0.28 m	h_{CG}
Maximum deceleration	1.5g	a
Tire rolling radius	0.2286 m	r
Coefficient of tire friction	1.5	μ
Maximum vehicle speed	120 km/hr	v
Number of revolutions	600 rpm	n_v
Caliper height	25.4 mm	h_c

6.1 Calculation of Heat Power and Heat Flux

The heat flow is obtained from the kinetic energy that is possessed by the vehicle at its maximum speed. Assuming, the kinetic energy of the vehicle is completely lost through the brake discs i.e. no heat loss between the tires and the road surface and the deceleration is uniform. The maximum deceleration of the vehicle is equal to the friction co-efficient between the road and tire times the acceleration due to gravity. Since the slick tires of friction co-efficient (=1.5), as found from the tire data, is used, maximum deceleration = 1.5g

- Kinetic energy $= \frac{1}{2}mv^2 = 138888.9\ J$
- Total kinetic energy = the heat generated = Q
- Stopping time (t_s)

$$t_s = \frac{v}{a}$$

$$t_s = 2.2\ sec$$

- Swept area (A_s)

$$A_s = \frac{\pi(D_0^2 - (D_0 - h_c)^2)}{4}$$

$$A_s = 0.0139\ m^2$$

- Total weight(w)= Front weight + Rear weight

$$w = w_f + w_r$$

$$w = 2450\ N$$

- During braking weight transfer from the rear to the front tires takes place. Therefore, it is necessary to calculate weight transfer which is based on the average coefficient of friction between tires and road, the weight of the vehicle, wheelbase and the center of gravity height of the vehicle.

  Weight transfer at maximum deceleration (w_T) [5]

$$w_T = \frac{w\mu h_{CG}}{l_w}$$
$$w_T = 675.19\ N$$

- Dynamic front weight ($w_{f''}$)

$$w_{f''} = w_f + w_T$$
$$w_{f''} = 1655.19$$

- Dynamic front weight ratio ($n_{f''}$)

$$n_{f''} = \frac{w_{f''}}{w}$$
$$n_{f''} = 0.675$$

- Total power generated (P)

$$P = \frac{Q}{t_s}$$
$$P = 61250\ W$$

- Power generated in one front disc (P_f)

$$P_f = \frac{n_{f''}P}{2}$$
$$P_f = 20689.96\ Watt$$

- $Heat\ flux\ for\ one\ front\ disc\ (q_f) = Power\ generated\ in\ front\ / 2 * Swept\ area$

$$q_f = \frac{P_f}{2A_s}$$
$$q_f = 1488.41\ kW/m^2$$

6.2 Calculation of Mechanical Forces

- Determination of dynamic forces acting on the tires at the time of maximum deceleration.

  Braking force on one front tire (F_f)

$$F_f = \frac{w_{f''}\mu}{2}$$

$$F_f = 1241.398\ N$$

- Front braking torque (τ_f)

$$\tau_f = (Braking\ force\ on\ one\ front\ tyre) * (Rolling\ radius)$$

$$\tau_f = F_f r$$
$$\tau_f = 283.78\ Nm$$

- The maximum braking torque obtained at the tires is equal to the maximum braking torque at the brake rotor as the torque transmitted by the wheels through the axle to the rotor remains the same.

Maximum friction force acting on the front disc (f_f)

$$f_f = {Braking\ torque}/{Effective\ radius\ of\ disc}$$

$$f_f = \frac{\tau_f}{(D_e/2)}$$

$$f_f = 3914.255\ N$$

7. FEA ANALYSIS

7.1 Transient thermal analysis

Transient thermal analysis is performed when we want to evaluate temperature and other thermal quantities that are varying with time. We have considered transient analysis for the study because we want to find out the heat distribution during braking when the vehicle decelerates from the maximum speed to zero.

Meshing of the model

The elements that we have used for meshing the disc are 3 dimensional tetrahedral elements with iso -parametric nodes. In the disc- pad region the temperature varies significantly and hence the meshing had to be very refined in that particular zone and we were particularly interested in evaluating the stress at sharp corners hence significant mesh requirement was needed.

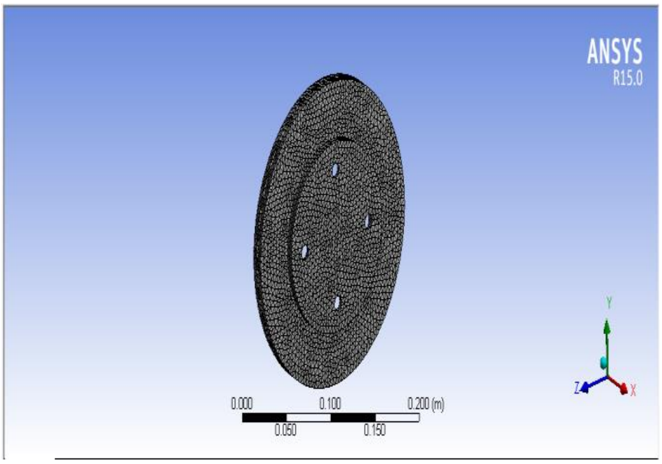

Fig -3: Meshing of the ventilated brake disc

Transient Thermal analysis boundary conditions

- The calculated heat power entering into the rotor is applied on the surfaces which only come in contact with the caliper pads
- Since effect of convection is considered in the analysis of the brake disc, the next boundary condition is applying the heat transfer convection coefficient to the entire disc. the value convection coefficient is taken from the numerical analysis we have done above
- Initial temperature is also given as input to the brake rotor

Table 9:–

Boundary conditions for thermal analysis	
Total heat power into the rotor	20689.96 W
Heat transfer convection coefficient	131.372 W/m²K
Initial Temperature	300 K

Adjust study properties

- Braking time = 2.2 [s]
- Increment time = 0.01 [s]
- Initial time = 0 [s]

Result of transient thermal FEA

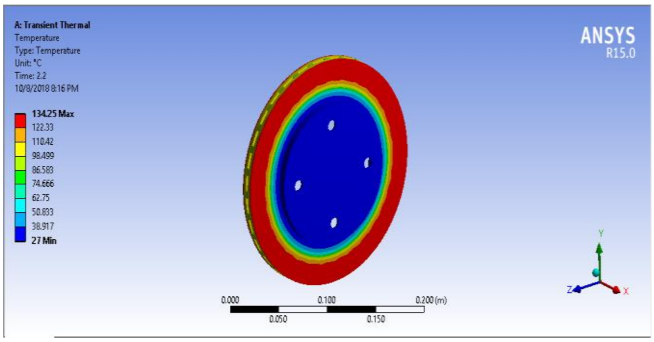

Fig -4: Temperature distribution of ventilated brake disc

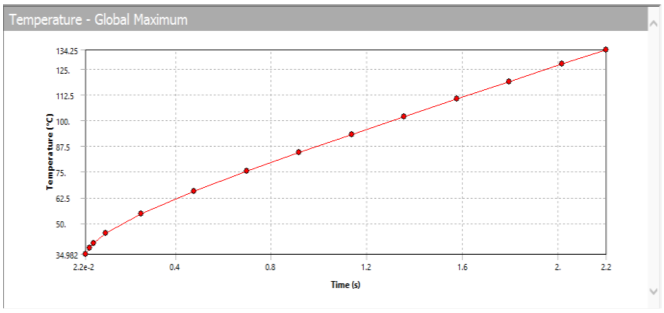

Chart -3: Time vs final temperature

Table -10:

Tabular Data

	Time [s]	Minimum [°C]	Maximum [°C]
1	2.2e-002	25.665	34.982
2	3.8674e-002	25.694	38.081
3	5.5348e-002	26.162	40.325
4	0.10537	26.853	45.099
5	0.25543	27.	54.71
6	0.47543	27.	65.562
7	0.69543	27.	75.212
8	0.91543	27.	84.333
9	1.1354	27.	93.193
10	1.3554	27.	101.88
11	1.5754	27.	110.44
12	1.7954	27.	118.9
13	2.0154	27.	127.27
14	2.2	27.	134.25

7.2 Coupling of Structural analysis with transient thermal analysis (static study)

The results of the transient thermal analysis obtained are now fed as input for carrying out the structural analysis. This is known as the thermal analysis coupled with structural analysis of the brake rotor. After carrying out the transient thermal analysis we now know the temperature distribution of the car for single stop braking from maximum speed to zero. The temperature distribution obtained will now serve as an input loading condition and in addition, we will now apply the friction braking force at the pad- disc region as well as the rotational velocity of the rotor. The simulation will be allowed to run to check whether the rotor will warp or not under extreme braking conditions.

Fixtures

We have applied two fixtures to the brake rotor

- Fixed geometry

Fixed geometry is applied to the four bolting points of the brake disc

- Roller/slider fixture

We selected one split face on one side of the rotor where the fixed brake pad comes in contact with the rotor. The reason for selecting this fixture is because the calipers are designed in such a way that load is applied by one pad to push the rotor into the other pad since the vehicle is using single piston floating calipers and not fixed calipers. Hence with this constraint with the assumption that the pad will not warp under loading condition we simulate the stationary brake pad

Boundary conditions
We have applied three loads to carry out the structural analysis

- Frictional braking force acting on the disc pad region in the circumferential direction

- Rotational velocity of the brake rotor

- Thermal loading obtained from the transient thermal analysis of the disc. Addition of thermal loading will cause additional thermal and displacement stress to be developed inside the structure. The temperature distribution at the end of braking will give us the maximum amount of thermal loading attained during braking.

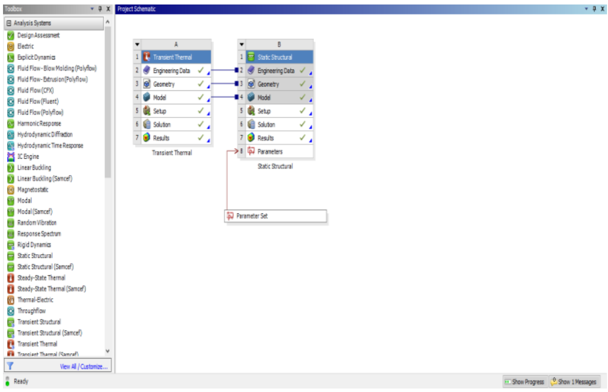

Fig -5: Transient thermal analysis coupled with structural analysis

Table -11:

Boundary conditions for static analysis	
Frictional Braking force	3914.255 N
Rotational speed of the rotor	2093.33 rad/s
Thermal load	From the transient thermal analysis

Results of static FEA

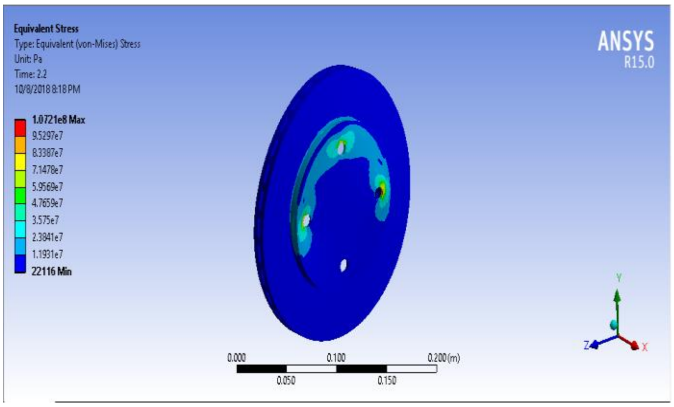

Fig -6: Equivalent (Von misis) stress distribution

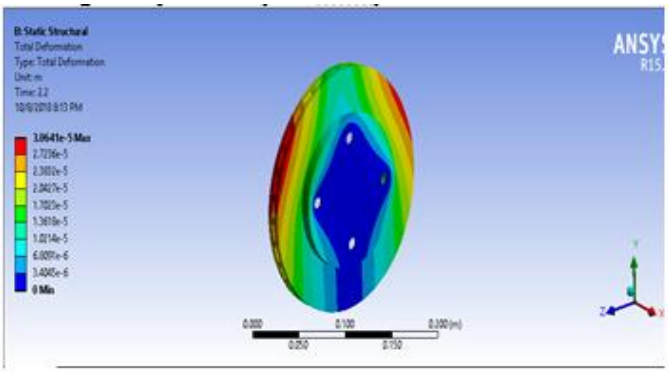

Fig -7: Total deformation

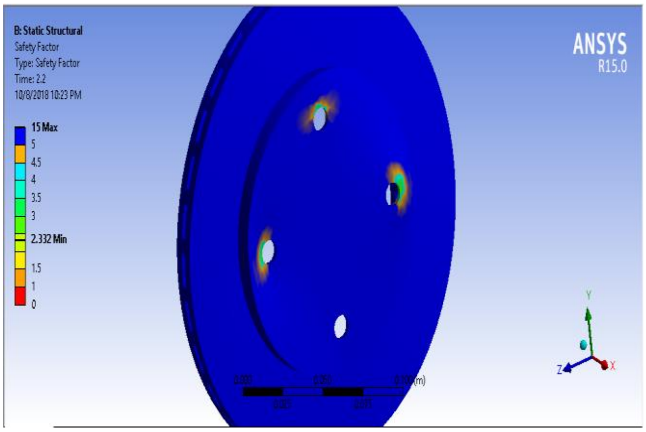

Fig -8: Factor of Safety

8. RESULT AND CONCLUSIONS

- The maximum and minimum value for final temperature, stress, strain, and factor of safety after carrying out the transient thermal analysis coupled with static structural analysis are noted. We can conclude that for a single stop braking of the vehicle that is from 120 km/hr to zero the maximum temperature attained by the ventilated disc lies within the safe limit as well the stresses and displacement induced due to the friction braking force and thermal loads also lie within the safe limit for a grey cast iron disc.

Table -12:

Results		
Parameter	**maximum**	**minimum**
Final temperature	134.25 ^{0}C	27 ^{0}C
Equivalent stress	1.07218e8 Pa	22116 Pa
Strain	0.00053781	1.8264e-7
Deformation	3.0641e-5 m	0 m
Factor of safety(FOS)	15	2.332

- Analysis of repeated braking of the vehicle reveals that the maximum temperature is reached after few application of brakes. The ventilated disc reached its maximum temperature after the 15th application of the brake and the temperature does not change or the relative temperature remains constant for the further braking cycle. We also see that the maximum temperature for repeated braking also lies within the safe limit and hence the disc will not warp under the conditions of repeated braking.

9. FUTURE SCOPE

- While performing the single stop transient thermal analysis coupled with the structural analysis we considered the heat flux to be constant. For further studies, one can consider more realisitic scenario where the heat flux decreases with time .

- We have used quantitative analysis method to calculate the heat transfer convection coefficient , one can use software based approach to calcultate the same.

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