DESIGN AND THERMO-STRUCTURAL ANALYSIS OF DISC BRAKE

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ABSTRACT

The disc brake is a device for slowing or stopping the rotation of a wheel. Repetitive braking of the vehicle leads to heat generation during each braking event. Transient Thermal and Structural Analysis of the Rotor Disc of Disk Brake is aimed at evaluating the performance of disc brake rotor of a car under severe braking conditions and there by assist in disc rotor design and analysis. Disc brake model and analysis is done using ANSYS workbench 14.5. The main purpose of this study is to analysis the thermo mechanical behavior of the dry contact of the brake disc during the braking phase. The coupled thermal-structural analysis is used to determine the deformation and the Von Mises stress established in the disc for the both solid and ventilated disc with two different materials to enhance performance of the rotor disc. A comparison between analytical and results obtained from est suitable design, material and rotor disc is suggested based on the performance, strength and rig FEM is done and all the values obtained from the analysis are less than their allowable values. Hence bidity criteria.

Keyword: Fea/ansys/disc brake.

INTRODUCTION

In today's growing automotive market the competition for better performance vehicle is growing enormously. The racing fans involved will surely know the importance of a good brake system not only for safety but also for staying competitive. The disc brake is a device for slowing or stopping the rotation of a wheel. A brake disc usually made of cast iron or ceramic composites includes carbon, Kevlar and silica, is connected to the wheel and the axle, to stop the wheel. A friction material in the form of brake pads is forced mechanically, hydraulically, pneumatically or electromagnetically against both sides of the disc. This friction causes the disc and attached wheel to slow or stop. Generally, the methodologies like regenerative braking and friction braking system are used in a vehicle . A friction brake generates frictional forces as two or more surfaces rub against each other, to reduce movement. Based on the design configurations, vehicle friction brakes can be grouped into drum and disc brakes. If brake disc are in solid body the heat transfer rate is low. Time taken for cooling the disc is low. If brake disc are in solid body, the area of contact between disc and pads are more. In disc brake system a ventilated disc is widely used in automobile braking system for improved cooling during braking in which the area of contact between disc and pads remains same. Brake assembly which is commonly used in a car.

PROBLEM STATEMENT

This project concerns of the temperature distribution and constraint of the disc brake rotor. Most of the vehicle today have disc brake rotors that are made of cast iron and stainless steel. Both are chosen for its relatively high thermal conductivity, high thermal diffusivity and low cost. In this project, analyze on the thermal issues of normal vehicle disc brake rotor to be done, and to determine the temperature behaviour of the disc brake rotor due to severe braking of the disc brake rotor by using Finite Element Analysis (FEA). Braking performance of a vehicle can be significantly affected by the temperature rise in the brake components. High temperature during braking will caused to: Brake fade, Premature wear, Brake fluid vaporization, Bearing failure, Thermal cracks, Thermally-excited vibration. Therefore, it is important to study and predict the temperature rise of a given brake component and assess its thermal performance in the early design stage. Finite element analysis (FEA) has been preferred and chosen method to analyse. Some of the above concerns such as disc brake rotor temperature rise and thermal cracks .

METHODOLOGY OF THE PROJECT

Begin with a literature review, a lot of paper and journal has been read up and a part of it has been considered in this project. Meanwhile, the precise dimensions have been used to translate in 2D and 3D drawing by using proe. In the second stage, load analysis has been done where the heat flux and convectional heat transfer coefficients has been calculated. Load analysis calculated based on full load vehicle. Later, value of load analysis has been applied on finite element analysis.

Next, the 3D model of disc brake rotor has been transfer to finite element software which is ANSYS. Thermal analysis has been done on both stainless steel and cast iron. Assigning material properties, load and meshing of the model has been done in this stages. Finally an expected result from the thermal analysis has been obtained.

LITERITURE REVIEW

Thundil Karuppa Raj Rajagopal, Ramsai Ramachandran, Mathew James, and Soniya Chandrakant (2014) “numerical investigation of fluid flow and heat transfer characteristics on the aerodynamics of ventilated disc brake rotor using cfd” In the present work, an attempt is made to study the effect of vane-shape on the flow-field and heat transfer characteristics for different configurations of vanes and at different speeds numerically.

S. Sarip,(2013) “design development of lightweight disc brake for regenerative braking – finite element analysis” International Journal of Applied Physics and Mathematics, Vol. 3, No. 1, January 2013. Regenerative braking would extend the working range of an EV or HEV provided that any extra energy consumption.

Chi, Z., (2008) “Thermal Performance Analysis and Geometrical Optimization of Automotive Brake Rotors”, M. Sc. thesis, University of Ontario Institute of Technology, Oshawa, Canada, 2008] has performed the thermal characteristics of brake discs using analytical approaches on the assumption of laminar flow and simplified geometry. The air flow through the passage of vane rotors is a complex turbulent flow.

Manohar Reddy, S., (2008) “Flow and Heat Transfer Analysis of a Ventilated Disc Brake Rotor Using CFD”, Journal of SAE, 0822 (2008), 1, pp. 1-12 Manohar Reddy has validated the flow field velocities of a simple radial vane with the experimental data measured using PIV and modified the vane shape for maximum heat transfer. Structural analysis of ventilated rotor brake discs using CAE.

Amol, A., Ravi, R., (2008) “ FE Prediction of Thermal Performance and Stresses in a Disc Brake System “, Journal of SAE , 2572 (2008), 1, pp. 5-8 finite element analysis (FEA) was reported. In their study, they reported the procedure for prediction of thermo-mechanical performance like temperature distribution and heat transfer coefficient estimation.

Manohar Reddy, S.(2007), " Flow and Heat Transfer Analysis of a Ventilated Disc Brake Rotor Using CFD", M.Sc. thesis, Indian Institute of Technology, Madras, Chennai, India, 2007 . The predicted results are validated with the numerical results of circular pillared vane.

David,A., (2003) "Analysis of the Flow through a Vented Automotive Brake Rotor", Journal of Fluids Engineering, 125 (2003), 1, pp. 979-986 have carried out the analysis for both flow surrounding the brake disc and inside the radial rotor passages using a two-component particle image velocimetry (PIV) system. Most of the previous studies have measured the air flow velocity profile at the exit of the passage of disc brake rotor using pressure probes. In some cases, they have used hot wire anemometry (HWA) to examine the unsteady flow-field at rotor exit.

Cho, M. H.; Kim, S. J.; Baschk R. H.; Fashk J. W.;Jang, H. R. A. Burton et al (2003) "Tribological study of gray cast iron with automotive brake linings: the effect of rotor microstructure".Tribology International, v. 36, p. 537-545, (2003). showed the thermal deformation in frictionally heated contact wheel-mounted on disc brakes were exposed to severe non-symmetrical mechanical and thermal loads. Degarmo, E. Paul; Black, J T.; Kohser, Ronald A (2003). "Materials and Processes in Manufacturing (9th ed.), Wiley, ISBN 0-471-65653-4," showed the thermal deformation in frictionally heated contact wheel-mounted on disc brakes were exposed to severe non-symmetrical mechanical and thermal loads.

Halderman, J. D.; Mitchell Jr., C. D. (2000) " Automotive brake systems", Upper Saddle River, NJ: Prentice-Hall, (2000). Shows thermo elastic instability in an automotive disk brake system was investigated experimentally under drag braking conditions. Ultimately a design method for lightweight brakes suitable for use on any car-sized hybrid vehicle was used from analysis.

CALCULATION TO INPUT PARAMETER

S.NO	DESCRIPTION	VALUES
1.	Gross weight(M)	9600N
2.	Wheel base (WB)	5.22m
3.	Maximum vehicle speed (V1)	100 (km/h)

4.	CG height (h)	1.2 m
5.	Distance of CG from front axle (a1)	2.45 m
6.	Distance of CG from rear axle (a2)	2.77 m
7.	Tire rolling radius (Rt)	0.445 m
8.	Stopping distance (Sd)	55 m

Table 2.1: Vehicle parameter

BRAKE TORQUE CALCULATION

The gross vehicle weight is 9.6 T and as per Indian Braking Standard IS: 11852, it falls under N2 category. For brake torque calculation following methodology is implemented.

Static load distribution (Rf): - Static load distribution describes the weight distribution according to horizontal position of centre of gravity. Static load distribution on front axle can be calculated with the following equation

$$R_f = M\left(\frac{a2}{a1 + a2}\right)$$

A calculated static load distribution on front axle is 5094 kg.

Mean fully developed deceleration (MFDD):- MFDD can be calculated by considering the maximum test speed (V1) of the vehicle in km/h. below equation [5] is used to calculate MFDD.

$$MFDD \ or \ \dot{\upsilon} = \frac{V_b^2 - V_e^2}{25.92\,(S_e - S_b)} \quad m/s^2$$

Whereas, Vb is the 80% of V1 in km/h, Ve is final speed of the vehicle after braking in km/h as 10% of V1, Se is the total distance travelled by the vehicle in meters during braking and Sb is the final distance travelled in meters i.e. zero. The calculated MFDD or υ̇ is 4.41 m/s2 i.e. 0.45g units.

Dynamic load distribution after braking (Fzf): - The vehicle is considered as one rigid body which moves along an ideally even and horizontal road. At each axle the force in the wheel are combined actions of normal and longitudinal force. In dynamic load distribution of the vehicle after braking, centre of gravity height (h in meters) plays an important role and influence dynamic part of the axle loads. Dynamic load distribution for front axle can be calculated by below equation

$$F_{zf} = M\, g \left(\frac{h}{a1 + a2} \right) \left(\frac{\dot{\upsilon}}{g} \right)$$

Whereas, g is acceleration due to gravity 9.81 m/s2. The calculated dynamic load distribution for front axle is 9733 kg.

Total load on front axle due to braking (Tf): - While decelerating, the total load is the sum of static load and dynamic load. And front axle is subjected to maximum dynamic load during braking. So calculation is done only for front axle. It can be calculated with help of below equation

$$T_f = Rf + F_{zf}$$

Tf is the total load acting on front axle. The calculated values for front and rear axle are 14827 kg.

Brake force on front axle (Bf): - After deciding the static loading, dynamic loading and total loading, the brake force acting on front axle can be calculated with the help of below equation

$$B_f = T_f\, \mu_r$$

Whereas, μr is the coefficient of road adhesion and varies according to road conditions. The calculated value for front axle is 5932 kg considering as tar road μr as 0.4. The brake force acting on front axle is the function of total load acting and coefficient of adhesion (μr). Below table 2.2 shows the different values of coefficient of adhesion as per different road conditions.

Brake force acting on per wheel of front axle (Bfw): - Brake force on each wheel of front axle is calculated with following equation

$$B_{fw} = \frac{B_f}{N_w}$$

Whereas, Nw is the number of wheels on front axle. Considering number of wheels on front axle as 2, then brake force on per front wheel is 2966 kg.

Brake torque acting on per wheel of front axle (Tbw):-The brake torque acting on per wheel of front axle is the function of brake force and tire rolling radius (Rt) can be calculated with the help of following equation

$$T_{bw} = B_{fw} R_t$$

The brake torque on per front wheel is 1320 kg-m. The brake will need to overcome this load before it can start to slow down the vehicle. But, if the load is at rest, the static brake torque will prevent the load from moving. In practice, a safety factor should be used in the case where the brakes is called upon only to hold this load and is only infrequently used in a dynamic manner. In this case a safety factor of 1.5 is used to calculate final brake torque on the disc. By considering the safety factor, the final braking torque will be 2000 kg-m. For structural analysis in ANSYS 14 workbench, the value of braking torque 2000 kg-m is taken further.

HEAT FLUX CALCULATION

Kinetic energy (KE):- Following equation is used to calculate the kinetic energy of the vehicle travelling with 100 km/h,

$$KE = \left(\frac{1}{2}\right) M v^2$$

Whereas, v is the speed of vehicle in m/s. The kinetic energy is 3703704J. Rotational energy (RE):- Rotational energy is the energy needed to slow down the rotating parts. It varies for different vehicles and which gear is selected. However taking 3% of kinetic energy is a reasonable assumption. The calculated rotational energy for the vehicle considered is 111111 J. Total energy (TE):- Total energy is the sum of kinetic energy and rotational energy. It can be expressed with below equation

$$TE = KE + RE$$

The calculated total energy is 3814815J.

Disc usable area (Ad):- Size of the brake disc considered for 9.6 T vehicle is, outer diameter (OD) as 0.323 m and inner diameter (ID) as 0.18 m. The usable area of the brake disc can be calculated with the help of below equation

$$A_d = \left(\frac{\pi}{4}\right) \left[(OD)^2 - (ID)^2\right]$$

The disc usable area will be 0.057 m2.

Time required to stop the vehicle (ts):- Time required to stop the vehicle is calculated with the help of following equation

$$t = \frac{v}{\dot{\upsilon}}$$

The time required to stop the vehicle is 6.28sec.

Braking power (Pb) and highest braking power (Pb(0)*):-* Braking power is the ratio of total energy to time required to stop the vehicle. It can be expressed in below equation

$$P_b = \frac{TE}{t}$$

$$P_{b(0)} = 2P_b$$

The calculated onset of braking power is 1214910 Watt.

Power distributed per disc (Pd):- The commercial vehicle of 9.6 T is two axle vehicle and as stated earlier front and rear axle is equipped with disc brake assembly. The braking power distributed per brake is calculated with the help of below equation

$$P_d = \frac{P_{b(0)}}{4}$$

The value of power distributed per disc is 303728 Watt.

Heat flux calculation (q):- Heat flux or thermal flux is the rate of heat energy transfer through a given surface. The brake calliper applies the brake force on effective radius on both the surface of brake disc .The heat flux for the brake disc can be calculated with the help of below equation

$$q = P_d / 2A_d$$

The calculated heat flux will be 2664277 Watt/m2.

Single stop temperature rise (Ts):- Single stop temperature rise depends on the time required (t) to stop the vehicle, heat flux and material properties.

Finally, single stop temperature rise can be calculated based on the material properties. This temperature is calculated for the required deceleration to stop at the distance of 50m. Following equation is used the calculate the temperature rise in °K

$$T_s = \left(\frac{0.527\, q\, \sqrt{t}}{\sqrt{(\rho\, Cp\, k)}} \right)$$

Whereas, ρ is density of material in kg/m3, Cp is specific heat in J/kg-k and k is thermal conductivity of in W/m-k. Final temperature (T) of the brake disc, considering ambient temperature as 50°C i.e. 323°K (Tamb) can be calculated with help of below equation

$$T = T_s + T_{amb}$$

Material properties	Stainless steels	Cast iron
Thermal conductivity(w/m k)	36	50
Density, ρ(kg/m3)	7100	6600
Specific heat , c (J/Kg)	320	380
Thermal expansion ,(10-6 / *k*)	0.12	0.15
Elastic modulus, *E* (GPa)	210	110
Coefficient of friction, µ	0.5	0.5
heat transfer coefficient h(w/km2)	150	120
Hydraulic pressure, *P* (M pa)	1	1
Applied braking torque	2000kg-m	2000kg-m
Applied braking power	303728 Watt.	303728 Watt.
Applied heat flux	2664277 Watt/m2	2664277 Watt/m2
Stopping time	6.28sec	6.28sec

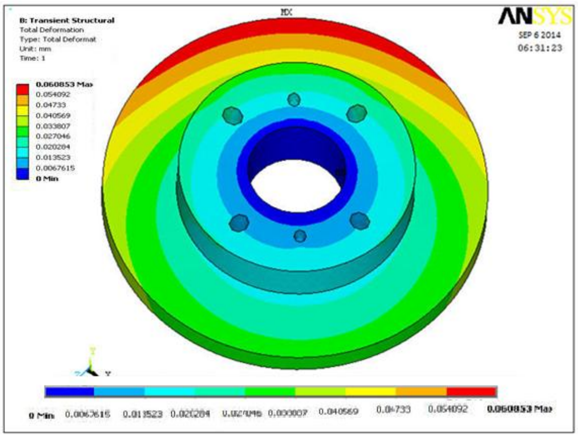
Table 2.3 : Material properties

S.NO	DESCRIPTION	VALUES (mm)
1.	Inner disc diameter, mm	30
2.	Outer disc diameter, mm	130
3.	Disk thickness, mm	24
4.	Disk height, mm	72
5.	Vehicle weight	9600N

Table 2.4: Disk dimensions

RESULTS

STRUCTURAL ANALYSIS OF SOLID DISC



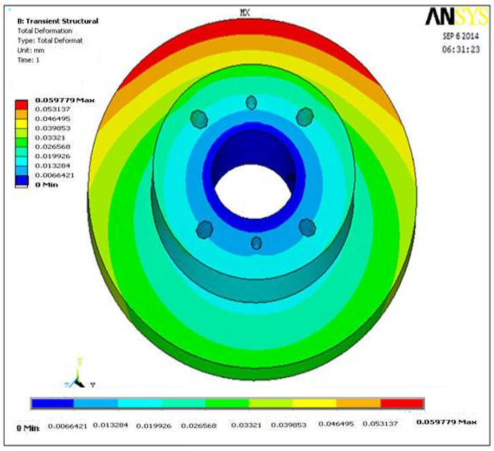
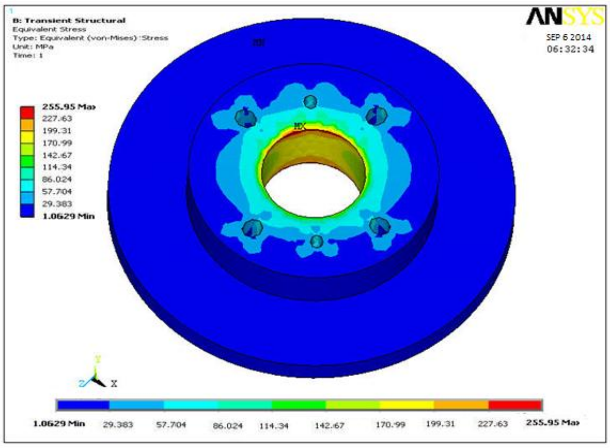


Fig 1.1: Deflection of stainless steel solid disc Fig 1.2: Deflection of cast iron solid disc



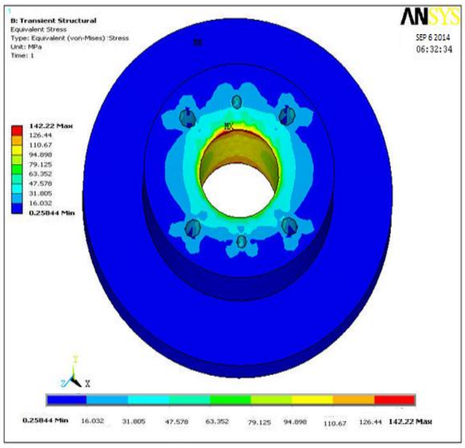
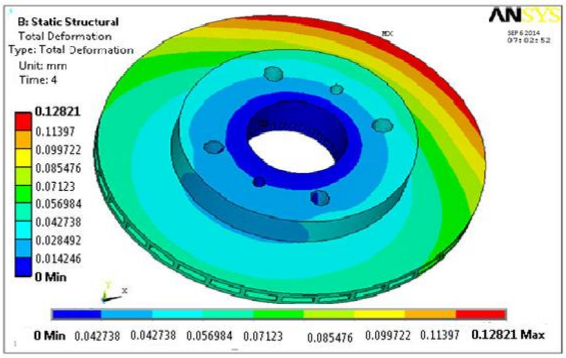


Fig 1.3: Von Mises stress stainless steel solid disc Fig 1.4: Von Mises stress cast iron solid

STRUCTURAL ANALYSIS OF VENTED DISC



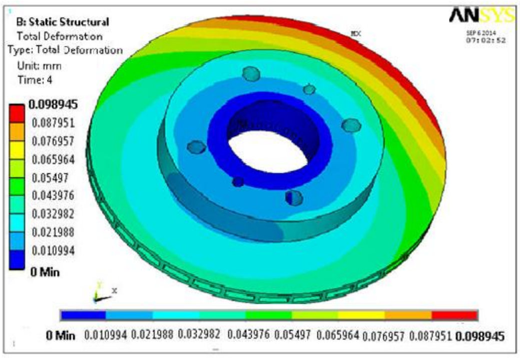
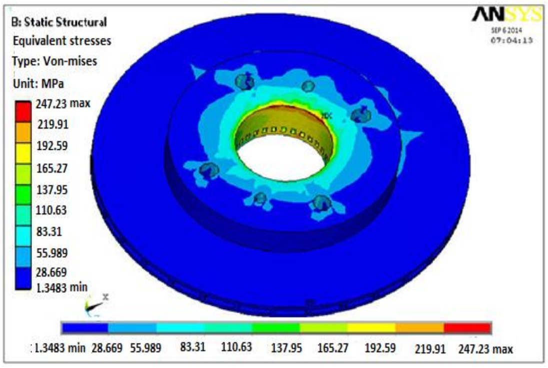


Fig 1.5: Deflection on stainless steel Vent disc Fig 1.6: Deflection on cast iron Vent disc



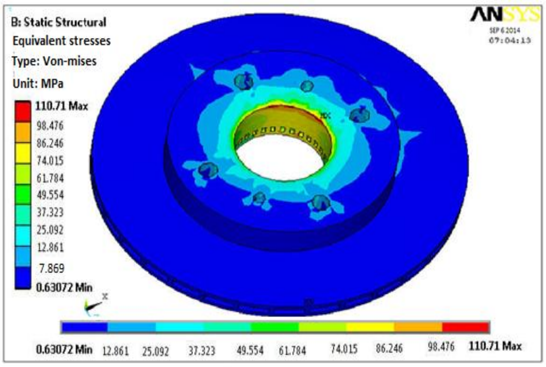
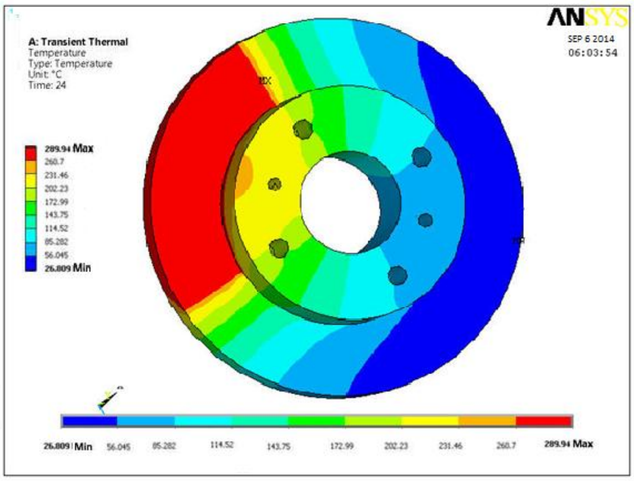


Fig 1.7: Von Mises stress stain steel Vent disc Fig 1.8: Von Mises stress cast iron Vent disc

THERMAL ANALYSIS OF SOLID DISC



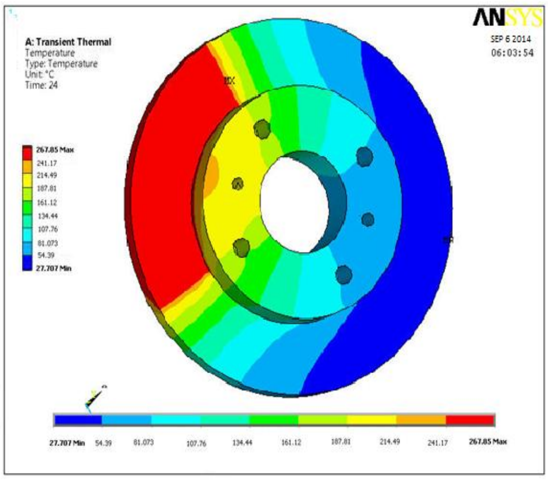
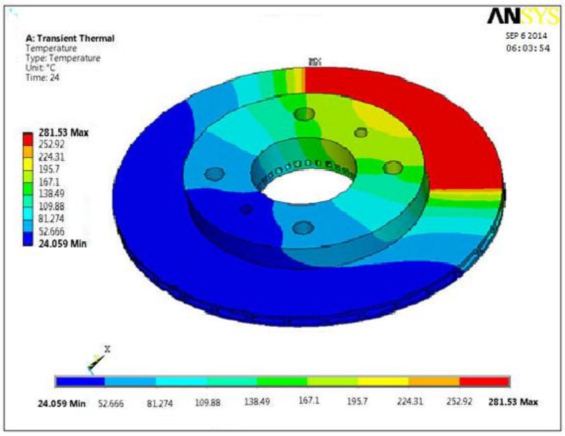


Fig 1.9: Temp.distribution solid stainless steel disc Fig1.10:Temp.distribution on solid cast iron

THERMAL ANALYSIS OF VENTED DISC



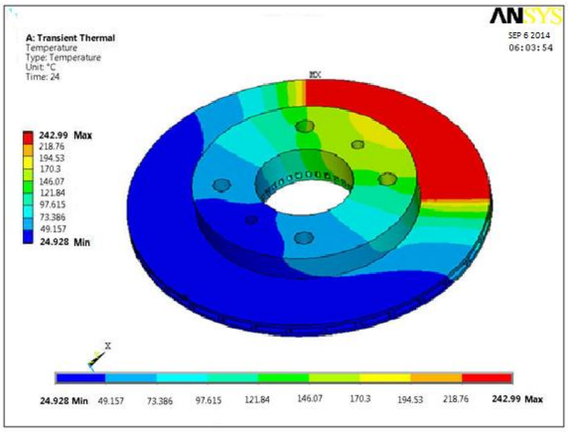


Fig 1.11: Temp.distribution stainless steel disc Fig 3.17: Temp.distribution Vented cast iron

COMPARISON THE RESULTS OF SOLID AND VENTED DISC

S.No	Flange width (mm)	Max.temp (o c)	Deflection in mm	Von misses stress in MPa
Solid Brake				
1.	SS 24	446	0.0608	256
2.	CI 24	413	0.0590	142
Ventilated Disc Brake				
1.	SS 24	321	0.105	247
2.	CI 24	**283**	0.**098**	**110**

Table 1.3: Comparison results of Solid and Ventilated Disc

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