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Thermo-Mechanical Analysis of Automotive Disc Brake Composite Rotor

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Abstract: *The heat generated due to friction during braking operation causes several important negative effects on the brake system. It is then important to determine the temperature field of the brake disc. In the present work, a transient thermo-mechanical finite element analysis (FEA) is performed to determine the braking efficiency of a Carbon Ceramic disk brake and compared to a Glass Fiber (S-2) brakes. The aim of the work is to investigate the rise of the temperature of the disk brake under severe braking conditions and the stresses generated from it. The investigation will be done using ANSYS software. ANSYS is a powerful FE package used to determine stress, strain and heat transfer in complicated problems.*
Keywords: *Ansys, composite material, disc brake, thermo-mechanical analysis.*

Nomenclature

q_0	heat flux entering the disc, W	T^*	temperature specified on the surface,°C
m	mass of the vehicle, kg	S_Q	surface heat flux, m^2, is, T is the temperature°C, is is the
g	the acceleration due to gravity, 9.81 m/s^2	Q	the heat generated during friction , J
v	initial speed of the vehicle, m/s	n	unit vector normal to the surface
z	braking effectiveness	Q^*	heat flux specified on the surface, W
$\in_p$	the factor of load distribution on the disc surface	S_c	surface in convection, m^2
A_d	disc surface swept by a brake pad, m^2	h	convective heat transfer coefficient in $Wm^{-2}k^{-}$
S_T	surface on which temperature is specified, m^2	T_p	temperature imposed, °C
T	temperature, °C	T_f	fluid temperature, °C

I. Introduction

During operation the vehicles brake system components require brake discs that constantly undergo thermal and mechanical deformations relatively high, which in time can lead to intense wear, formal changes or deterioration of this. Also, the braking torque is influenced by a number of random factors, such as weather conditions and type of surface which the vehicle runs [1].

Thermal judder occurs as a result of non-uniform contact cycles between the pad and the disk brake rotor, which is primarily an effect of the localized Thermo-Elastic Instabilities at the disk brake rotor surface [2]. Thermoelastic instability occurs when a pressure perturbation in the system causes more energy to be induced at one point in the rotor. Since more energy enters the rotor at one point, this spot becomes hotter than the adjacent material noting that these pressure perturbations are almost always present. Thermal growth causes it to become thicker than other regions of the rotor and that results in even higher torque variation. Eventually, this process continues until the brake is released or equilibrium is reached between the uneven energy into the rotor surface, the conduction within the rotor between the hot and cool regions, and friction behavior [3].

The wear mechanisms present in braking systems have studied by several researchers over the years. These are quite unpredictable because of the contact variations that exist in these specific and complex systems [4]. In the last decades, great attention has been given to improve brake discs performance concerning its behavior when there is friction between the brake pads. This great effort led to materials development, such as non-ferrous copper alloys, aluminum matrix composites and, nowadays, carbon composites [5]. The temperature distribution of solid rotors has been investigated by Limpert using Duhamel's Theorem [6]. The derivation of the temperature equation is accomplished with the assumption of a constant heat flux during constant-speed downhill braking and a constant heat flux.

Newcomb [7] investigated the thermal properties of rotors with different materials involving cast iron, steel, aluminum bronze and duralumin. The steel rotors have the lowest temperature among these four materials and the duralumin rotors have the highest temperature. Choi and Lee [8] performed a transient thermoelastic analysis of disc brakes in repeated braking applications, using a finite element method with frictional heat generation.

Thilak et al. [9] evaluated the performance of disc brake rotor of a car under severe braking conditions and with different materials involving cast iron, ALMMC, E-glass, and S2-glass. The suitable material for the braking operation is S2 glass fiber and all the values obtained from the analysis are less than their allowable values. To investigate elastoplastic thermal stresses in a thermoplastic composite disc that is reinforced by steel fibers, curvilinear. Finite element method (FEM) was used to calculate the thermal stress distribution in the model of composite disc by Faruk et.al [10]. You et al. [11] improved a numerical method for the analysis of deformations and stresses in the elastic-plastic rotating discs with arbitrary cross-sections of continuously varying thicknesses and arbitrary variable density made of nonlinear strain-hardening materials.

Bektas et al. [12] studied an elastic–plastic stress analysis of a thin aluminum metal matrix composite disk under internal pressure. An analytical solution was performed for satisfying the elastic–plastic stress–strain relations and boundary conditions for small plastic deformations. Mackin et al. [13] studied thermal stresses as a result of cracking in the disc brakes.

The main goal of this study is to improve the design of the rotor disc of a disc brake, using the concept of finite element method utilizing the ANSYS finite element package. The material of disc brake will be changed to two composite materials, Carbon ceramic, and S-2 glass fiber. The finite element analysis will be carried out using the ANSYS package. A 3D transient Thermo-mechanical analysis will be conducted to determine the temperature changes, stresses generated in the brake due to braking.

II. Finite Element Modeling

The finite element method is numerical analysis technique for obtaining approximate solutions to a wide variety of engineering problems. Thermal analysis of two different materials is carried out and structural analysis is also performed for analyzing the stability of the structure.

2.1 Modeling and Analysis

It is very difficult to exactly model the brake disk, in which there are still researches are going on to find out transient thermo elastic behavior of disk brake during braking applications. There is always a need of some assumptions to model any complex geometry. These assumptions are made, keeping in mind the difficulties involved in the theoretical calculation and the importance of the parameters that are taken and those which are ignored.

The assumptions which are made while modeling the process are given below:

- The disk material is considered as homogeneous and isotropic .
- The domain is considered as axis-symmetric .
- Inertia and body force effects are negligible during the analysis .
- The disk is stress free before the application of brake .
- Brakes are applied on the entire four wheels .
- The analysis is based on pure thermal loading and vibration and thus only stress level due to the above said is done. The analysis does not determine the life of the disk brake .
- Only ambient air-cooling is taken into account and no forced convection is taken .
- The kinetic energy of the vehicle is lost through the brake disks i.e. no heat loss between the tire and the road surface and deceleration is uniform .
- The disk brake model used is of solid type and not ventilated one .
- The thermal conductivity of the material used for the analysis is uniform throughout .

The specific heat of the material used is constant throughout and does not change with temperature.

2.1.1 Dimensions

The dimensions of brake disk used for transient thermal and static structural analysis are shown in Table 1 and the actual disc brake rotor is given in fig 2.1.

Description	Values
Inner disc diameter, mm	116
Outer disc diameter, mm	262
Disc thickness, mm	12
Disc height, mm	50

Table 1 Dimensions of Disk break rotor

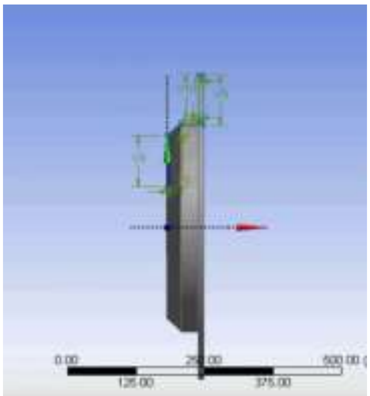

Figure 2.1: Actual disc rotor model

2.1.2 Finite element mesh

The element has 6475 nodes with single degree of freedom and temperature at each node. The 19538 elements have compatible temperature shape and are well suited to model curved boundaries. The 19538 thermal elements are applicable to a 3-D, steady state or transient thermal analysis.

The modeling in Ansys CFX stage consists in preparing the mesh of the fluid field. In our case, one used element with 206156 nodes and 1022400 elements. The mesh model is shown in fig 2.2.

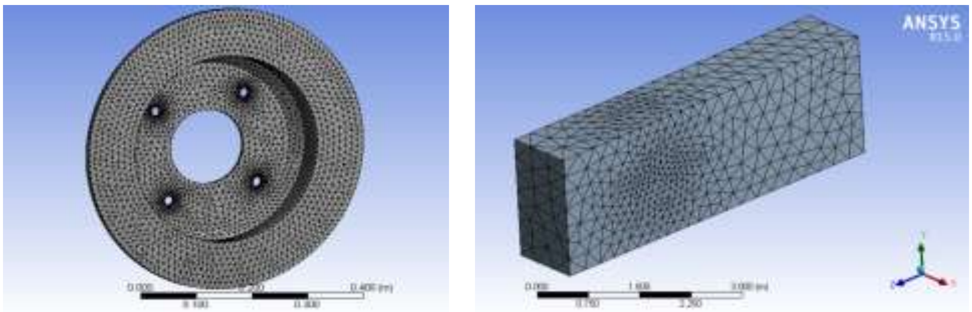

Fig 2.2: Mesh model of Disc break rotor and Fluid field

2.1.3 Modeling in ANSYS CFX

The first stage is to create CFD model which contains the fields to be studied in Ansys Workbench. In our case, we took only complete disc then we defined the field of the air surrounding this disc. ANSYS ICEM CFD will prepare various surfaces for the two fields in order to facilitate the mesh on which that one will export the results toward CFX using the command "output to cfx". After obtaining the model on CFX Pre and specifying the boundary conditions, we must define the physical values come into play on CFX to start calculation. The disc is related to four adiabatic surfaces and two surfaces of symmetry in the fluid domain whose ambient temperature of the air is taken equal to 25 ^{o}C.

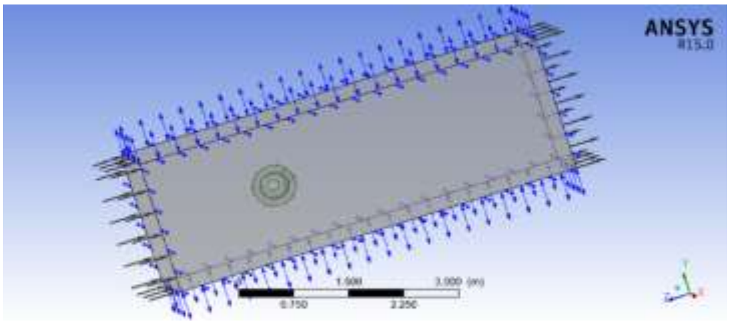

Fig 2.3: Fluid model and disc brake rotor with pads

2.1.3Heat Flux entering the disc

In a braking system, the mechanical energy is transformed into a calorific energy. This energy is characterized by total heating of the disc and pads during the braking phase. Generally, the thermal conductivity of material of the brake pads is smaller than that of the disc. We consider the heat produced will be completely absorbed by the brake disc. The heat flux emitted by this surface is equal o the energy generated by friction. The initial heat flux q_0 entering the disc is calculated by

$$q_0 = \frac{1-\emptyset}{2}\frac{mgvz}{2A_d\epsilon_p} \qquad (1)$$

2.1.4 Boundary conditions

The boundary conditions in this model are embedded configuration. The bolt holes of the disc are rigidly constrained in all the direction and are allowed to have rotational movement. The pad is fixed in all degrees of freedom except in the normal direction to allow the pads to move up and down and in contact with the disc surface.

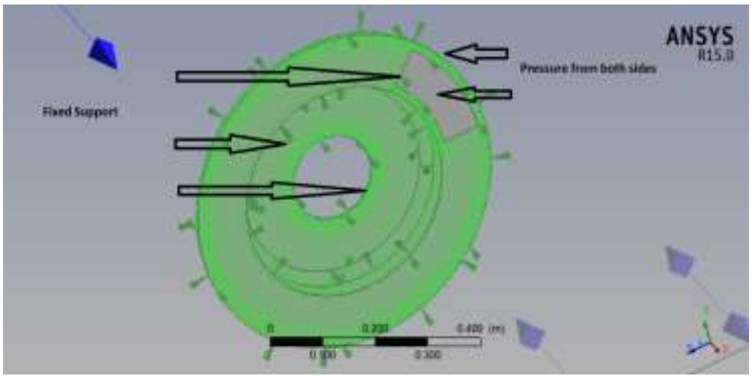

Fig 2.4: Boundary conditions and loading imposed on the disc–pads

In thermal and structural analysis of disk brake, we have to apply the following boundary conditions on 3-D disk model of disk brake.

(a) Temperature specified on the surface

$$S_T : T = T^* \qquad (2)$$

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(b) Heat flux specified on the surface

$$S_Q: \{Q\}^T \{n\} = -Q^* \qquad (3)$$

(c) Convection specified on the surface

$$S_c : \{Q\}^T \{n\} = h\,(T_p - T_f) \qquad (4)$$

In the aspect of the car accident prevention, the braking performance of vehicles has been a critical issue. The rotor model heat flux is calculated for the car moving with different velocities as 60, 90 and 120 kmph and the following is the calculation procedure.

Description	Value
Vehicle mass, kg	2500
Initial speed *u*, kmph	60, 90 and 120
Vehicle speed at the end of brake application *v*, m/s	0
Acceleration due to gravity *g*, m/s^2	9.8
Brake rotor diameter, m	0.262
Axle weight distribution, γ	0.3
Tire radius, mm	406
Percentage of kinetic energy that disc absorbs, k	0.9
Coefficient of friction for dry pavement, μ	0.7

Table 2: Input Parameter

(d) Analytical temperature rise

The contact area between the pads and disc of brake components, heat is generated due to friction. For calculation of heat generation at the interface of these two sliding bodies, two methods are suggested on the basis of law of conservation of energy which states that the kinetic energy of the vehicle during motion is equal to the dissipated heat after vehicle stop. The material properties and parameters adopted in the calculations are as shown in table3.

PROPERTIES	Carbon Ceramic	S-2 Glass Fiber
Density, Kg/m^3	2450	2460
Thermal conductivity, w/m-K	40	1.45
Young's Modulus, N/m^2	30×10^9	87×10^9
Poisson's Ratio	0.1	0.28
Specific Heat, J/Kg-K	800	737
Coefficient of Linear Expansion, /K	2.6×10^{-6}	0.9×10^{-6}

Table 3 Material properties

Single stop temperature rise T_{max} is the temperature rise due to single braking condition.

$$T_{max} = \frac{0.527 \times q \times \sqrt{t}}{\sqrt{(\rho.c.k)}} + T_{amb} \qquad (5)$$

The relative brake temperature after the n^{th} brake application can be calculated using relation,

$$T_{roa} - T_i = \frac{\left[1 - e^{\left(-\frac{n \square \Delta t_c}{\rho c v}\right)}\right][\Delta t]}{1 - e^{\left(-\frac{\square \Delta t_c}{\rho c v}\right)}} \qquad (6)$$

The compressive stresses $'\sigma'$ developed in the surface of a disc from sudden temperature increases is

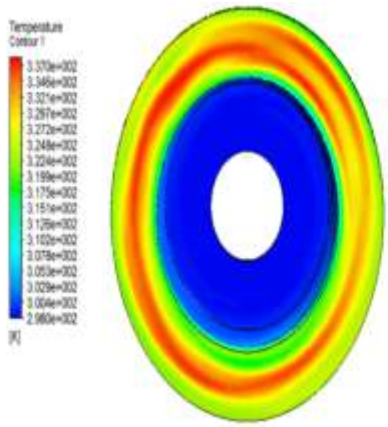
$$\sigma = \frac{E}{1 - v} \times \alpha \times \Delta \qquad (7)$$

III. Results

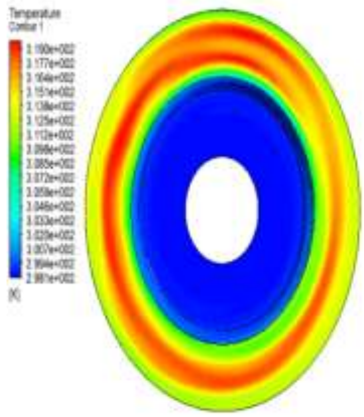
Transient thermo-mechanical analysis behavior of disk brakes with properties of S-2 Glass Fiber and Carbon Ceramic the ANSYS simulation is obtained on braking at 3 different speeds after applying the thermal and structural boundary conditions.

3.1 Thermal analysis

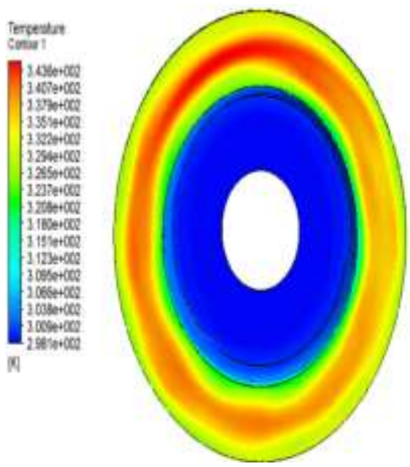
The temperature changes at different speeds are shown in the fig 3.1. The initial temperature of the disc rotor and pad is kept at 298 K. It can be noted that the maximum temperature reached in S-2 Glass Fiber at 120kmph is 418 K where as in Carbon ceramic at 120 kmph the temperature reaches maximum at 363.8 K which is lower than S-2 Glass Fiber.



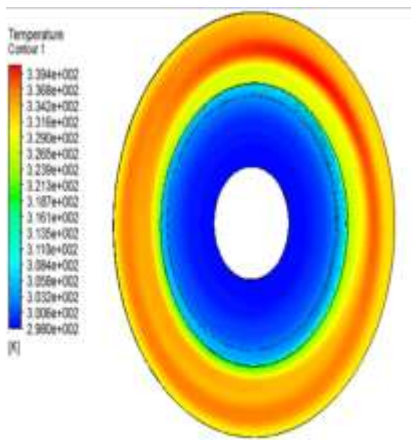
(a) S-2 Glass Fiber at 60 kmph



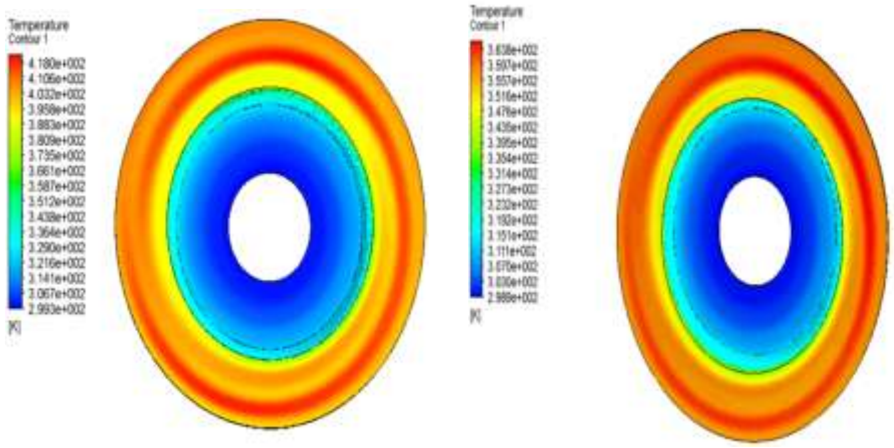
(b) Carbon Ceramic at 60 kmph



(c) S-2 Glass Fiber at 90 kmph



(d) Carbon Ceramic at 90 kmph



(e) S-2 Glass Fiber at 120 kmph (f) Carbon Ceramic at 120 kmph

Fig 3.1 Temperature distributions at different speeds

Also from the result one notices that the brake disc of S-2 Glass Fiber has fast temperature rise under the same loads as compared to the brake disc of Carbon Ceramic.

The table 2 shows complete comparison of temperature changes at different speeds between S-2 Glass Fiber and Carbon Ceramic. It is evident from the table the temperature rise in Carbon Ceramic disc rotor is less when loads are applied. Therefore, disc brake rotor made up to Carbon Ceramic material is preferred

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Material	Temperature (K) At		
	60 kmph	90 kmph	120 kmph
Glass Fiber	337	343.6	418
Carbon Ceramic	319	339.4	363.8

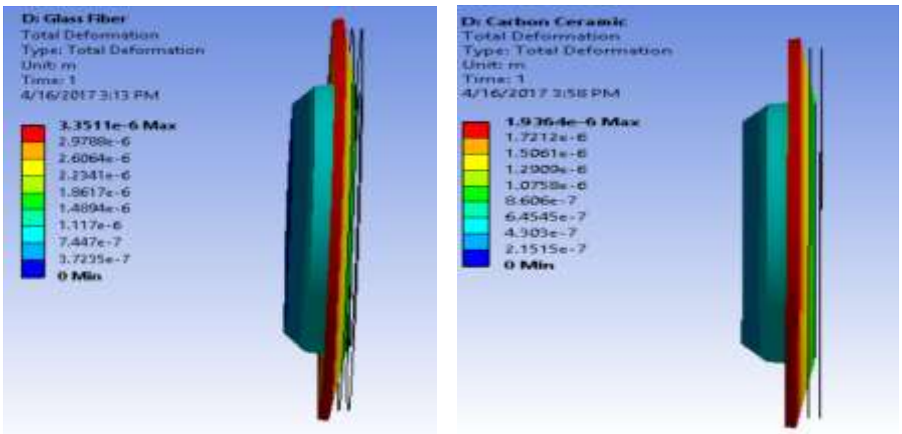
Table 4 Temperature distribution

3.2 Structural analysis

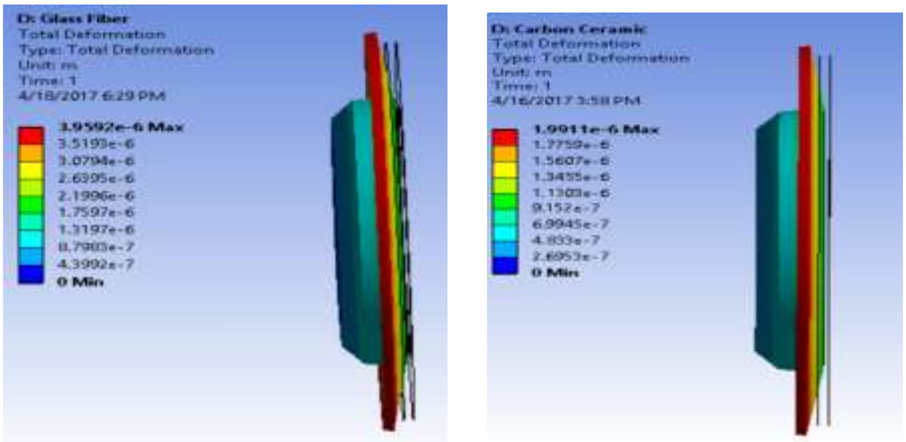
The aim of this thermo mechanical analysis is to get comparison and better understanding of the deformation and stress generated or equivalent stress (Von Mises) of S-2 Glass Fiber and Carbon Ceramic at different speeds when it is not only subjected to loads from the pads but also the expansion induced by the temperature effect.

3.2.1 Deformation

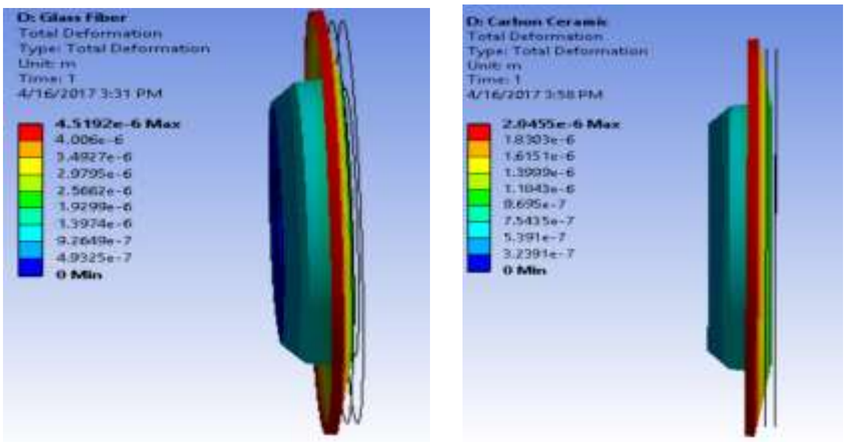
The deformations at different speeds are shown in the fig 3.2. It can be noted that the deformation always found at the outer surface of the disc i.e. the area in the contact with the disc and pad. Due to breaking at higher speed it can be seen that the deformation occurs more. It is also observed that the disc deformation increases linearly as a function of the disc radius. The maximum deformation occurs in S-2 Glass Fiber at 120 kmph is 4.5192e^{-6} and the lowest in Carbon ceramic at 120kmph is 2.0455e^{-6}, which is approximately over 45% lower than the S-2 Glass Fiber disc rotor. This is due to the structural stiffness in the material properties of Carbon ceramic. It also clearly indicates that the temperature has a strong influence in the thermo-mechanical response of the disc break.



(a) At 60 kmph



(b) At 90 kmph



(c) At 120kmph

Fig 3.2 Deformation at different speeds

The deformation results are given in fig 3.3 for the given materials. Disc with material properties of Carbon ceramic shows less deformation at higher speed is preferred.

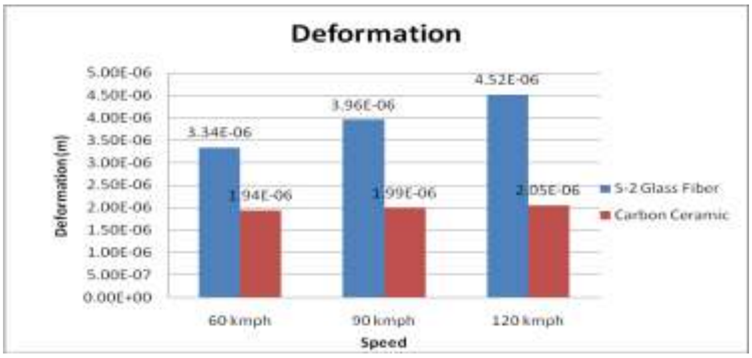
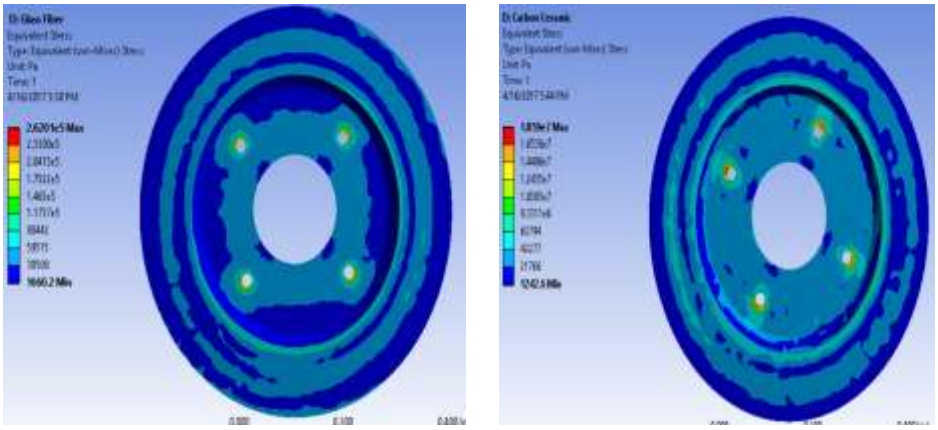


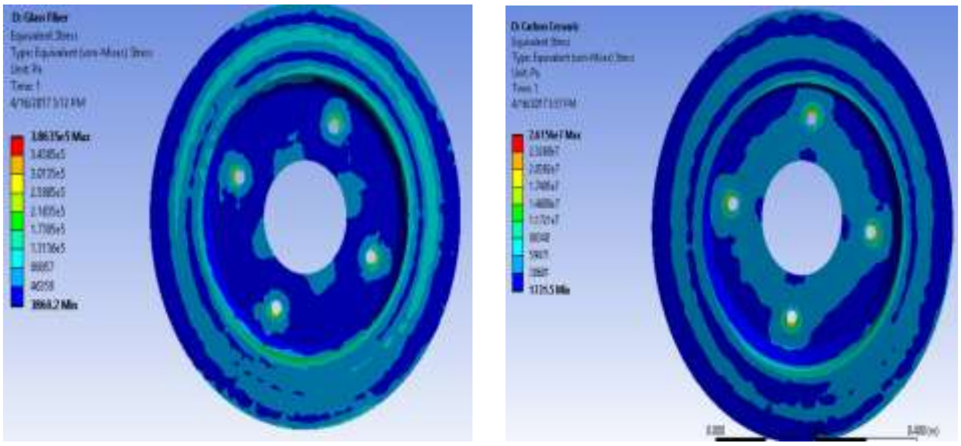
Fig 3.3: Deformation at different speeds

3.2.2 Von Mises

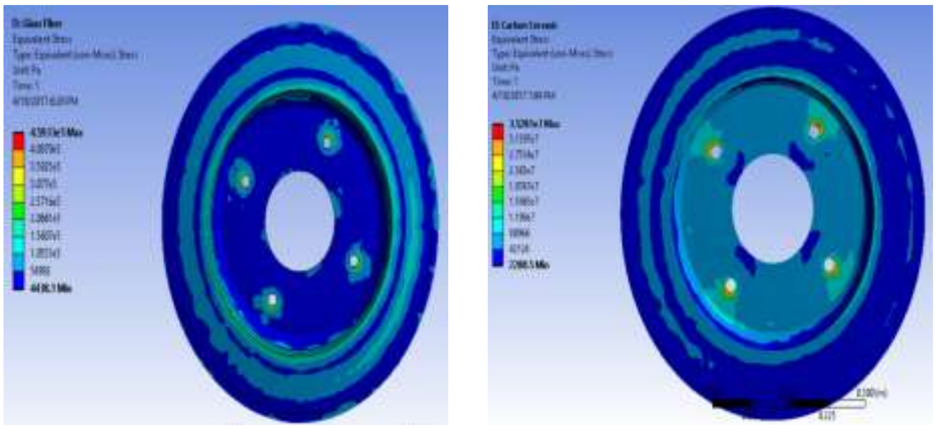
The stress generated or equivalent stresses (Von Mises) at different speeds are shown in the fig 3.4. It can be noted that maximum stress generated in S-2 Glass Fiber at 120kmph is $4.59e^5$ where as in Carbon ceramic at 120 kmph the equivalent stress reaches maximum to $3.52e^7$ which is lower than S-2 Glass Fiber.



(a) At 60 kmph



(b) At 90 kmph



(c) At 120 kmph

Fig 3.4: Equivalent stress (Von Mises) at different speeds

The fig 3.5 shows complete comparison of stress generated or equivalent stress (Von Mises) at different speeds between S-2 Glass Fiber and Carbon Ceramic. It is evident from the graph that the stress generated in Carbon Ceramic disc rotor is less when loads are applied. Therefore, disc brake rotor made up to Carbon Ceramic material is preferred.

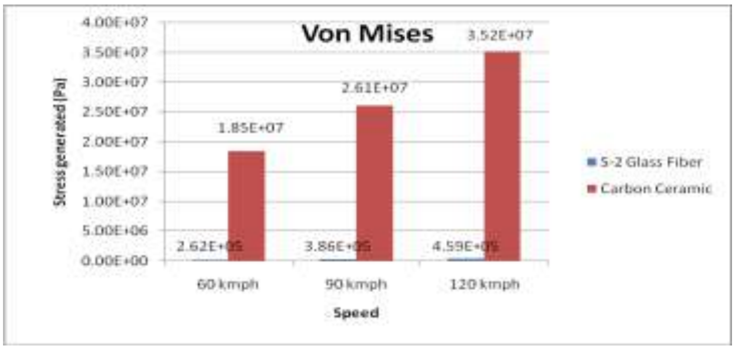


Fig 3.5: Stress generated (Von Mises) at different speeds

IV. CONCLUSIONS AND RECOMMENDATION

4.1 Conclusion

The thermo mechanical analysis of disc break rotor with two different material properties has been performed. In addiditon to this the analysis is performed at three different speeds to get complete understanding of temperature disribution, stress generated and deformation of disc break rotor. In order to improve the braking efficincy and to provide greater stability to the vehicle by comparing from the result of the analysis it is concluded that even at higher speed Carbon Ceramic shows less temperature deviation and less deformation as compared to S-2 Glass Fiber. Hence composite matrial like Carbon Ceramic is better material than S-2 Glass Fiber for disk brake rotor under different breaking condition.

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