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Modeling and Analysis of Disc Brake mounted on axle of the Train

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Abstract

Axle mounted disc brakes are generally employed in high speed trains owing to the lesser weight and sustainable thermal properties. Most widely used material for manufacturing of the disc brake is Gray Cast Iron, because of its availability. However, it being a brittle material has certain limitations. Hence, an Aluminium Composite Material (AlSiC) is considered for analysis under different loading conditions. Two models of brake discs are developed on which both Structural and Coupled Analysis were done and results are compared.

Keywords:—*Disc Brake, Finite Element Modeling, Stress Analysis.*

I. Introduction

Initially, the braking system in railway bogies consisted of wheel-thread braking, in which, the brake shoe applies friction force to the wheel tread, creating a sliding effect. High-speed trains cannot use this type of brake as it causes damage to the wheel-thread. The newly adopted railway coaches like the LHB coaches employed Axle-Mounted Disc Brakes to encounter the existing problem in high-speed trains. The braking system of a train involves a very complex process and of great importance by the essential contribution on the safety of the traffic. This complexity results from the fact that during braking, numerous phenomena occur - mechanical, thermal, pneumatic, electrical, etc. The actions of these processes take place in various points of the vehicles and act on different parts of the train with varying intensities. Overheating and thermal deformation of disc brake systems during emergency braking are becoming critical. Because of the amount of heat transferred to the discs may depend on their design and the train speed, it is difficult to determine beforehand and analyse discs and linings separately. In this paper, the effects of thermal and structural loads coupled together on the brake disc are analysed with different materials.

II. Literature Review

Marko Reibenschuh et al [1]: Two brake discs (new and old) were used for the both structural and thermal analysis. Material used for both the discs was Spheroidal Graphite Cast Iron. They found that the stresses developed in the new disc were more than worn disc but stress values for both the discs were within the limits.

M.A. Maleque et al [2]: The aim of this paper was to develop the material selection method and select the optimum material for the application of brake disc system emphasizing on the substitution of this cast iron by any other lightweight material. Material performance requirements were analysed and alternative solutions were evaluated among cast iron, aluminium alloy, titanium alloy, ceramics and composites. The analysis led to aluminium metal matrix composite as the most appropriate material for brake disc system.

Y. Yildiz and M. Duzgun et al [3]: had studied stress analysis of ventilated brake discs using the Finite Element Method. In this study, three different ventilated brake discs, the cross drilled disc, the cross-slotted disc, and the cross-slotted with a side groove disc, were manufactured, and their braking force performances were investigated experimentally together with a solid disc. However, these comparisons indicate that the application of varying force distributions along brake pads reduces the stresses on ventilated discs by 8.8% to 19.1%.

Sung Pil Jung et al [4]: had studied Thermal Characteristic Analysis and Shape Optimization of a Ventilated Disc. In this study, an analysis technique that can estimate the temperature rise and thermal deformation of the ventilated disc considering vehicle information, braking condition and properties of the disc and pad was developed. The analytical process of the braking power generation during braking was mathematically derived.

III. Problem Definition

3.1 Aim of the Project

The main objective of this work is to find the stresses and deformations induced in brake disc model made up of GCI and AlSiC under Structural and Coupled loads and suggest suitable material geometry for the brake disc. The model chosen is similar to the existing one and is of bolted type. A case of emergency braking from 160 kmph to stop on horizontal track is considered. The input loads Pressure, Rotational velocity, Heat flux were calculated and applied on the brake disc and the output in terms of maximum values Von Mises, Total deformation and Temperature induced in the brake disc were recorded and the results were analysed to suggest the suitable model for brake disc.

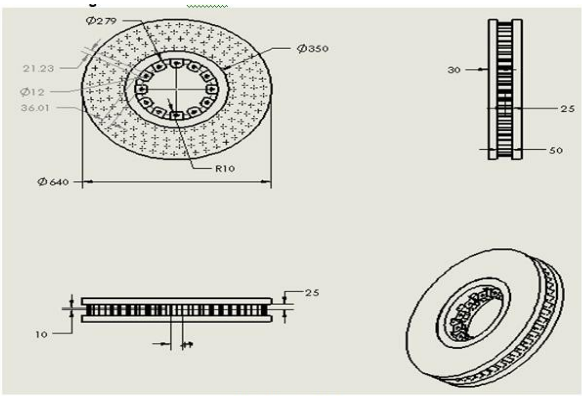

Figure 2: CAD Model of disc brake (Model 2)

Details of model, materials, loading, geometry and software used are given below: From Railway specification No. RDSO/2015/CG-03 'Brake disc for LHB type coaches'

Table 1: Dimensions of brake disc

SNo	Description	Dimension
1	Dimensions	640x110 mm
2	Outer Diameter of friction ring	640 +0/-1 mm
3	Inner Diameter of friction ring	350mm
4	Width of friction ring	110+0/-0.3 mm
5	Inner bore	199 mm (H6)
6	Width of hub	150mm

Table 2. Material Properties

Properties	GCI	AlSiC
Ultimate Tensile Strength (MPa)	240	340
Max. Stresses developed (MPa)	212.76	265.13
Max. Deformation (mm)	0.238	0.16
Thermal Stress (MPa)	0.0011	0.00087
Weight (kg)	133.51	54.702

Table 3: Disc brake materials and their properties

SN	Parameter	Unit	Grey Cast Iron	Aluminum Composite*
1	Density (□)	(kg/m3)	7200	2950
2	Yield strength	(MPa)	---	240
3	Ultimate Tensile Strength	(MPa)	240	340
4	Young modulus	(GPa)	110	230
5	Poison ratio	---	0.28	0.24
6	Shear modulus	(GPa)	43	93
7	Bulk modulus	(GPa)	83	147
8	Thermal conductivity (□)	W/m-C	52	197
9	Coefficient of Thermal Expansion	*10-6	0.11	6.5
10	Specific Heat (□□)	J/Kg-C	447	808

Table 4: Composition of materials % by weight

Al 6063	Si	Mn	Mg	Cu	Fe	Ti	Al
	0.44	0.07	0.6	0.018	0.2	0.008	98.664
GCI (SA EJ431)	C	Mn	P	Si	S		Fe
	3.0-3.3	0.6-0.9	0.08	1.8-2.2	0.15		94

*Aluminium Composite 80% Al 6063 + 20% SiC by weight

Boundary conditions: For static structural analysis; Pressure on disc: 1.5 MPa, Rotational velocity: 97.12 rad/sec. For coupled analysis; Temperature as derived from Transient thermal analysis for different materials and above mentioned structural load.

Software used :

Modeling – SOLID WORKS 2016

Analysis – ANSYS WORKBENCH R16

IV. RESEARCH METHODOLOGY

Introduction: As the loading of brake disc is complex in nature, it is safe to consider **Von Mises** developed to examine the failure of the model. The Von Mises yield criterion (also known as the Maximum Distortion Energy Theory of Failure) suggests that yielding of a ductile material begins when the second deviatoric stress invariant reaches a critical value. It is part of plasticity theory. It works well for most cases, especially when the material is ductile in nature. Prior to yield, material response can be assumed to be of a nonlinear elastic, viscoelastic or linear elastic behaviour. **Distortion energy theory:** The concept of Von Mises Stress arises from the distortion energy failure theory. Distortion energy failure theory is comparison between 2 kinds of energies, 1) Distortion energy in the actual case 2) Distortion energy in a simple tension case at the time of failure. According to this theory, failure occurs when the distortion energy in actual case is more than the distortion energy in a simple tension case at the time of failure. **Expression for Von Mises Stress:** The failure criterion according to the Distortion Energy theory is given by:

$$\left[\frac{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}{2}\right]^{\frac{1}{2}} = \frac{S_{yt}}{(fs)} \qquad (1)$$

Structural Load calculations: In this case of a railway vehicle travelling at a speed of 160 kmph on a horizontal track stops due to application of emergency brake was considered. Time of travel before stopping, deceleration, weight of the vehicle, clamping force on the brake disc, brake pad area, coefficient of friction etc. for calculating the loads are taken from the railway specification. Mass of the Railway vehicle – M=64000kg, No. of axles per vehicle=4, Maximum load per axle=16000kg, no. of brake discs per axle=2, Load on each wheel=8000kg, Start speed

v_0=44.4m/s, Deceleration a=1.2 m/s2, Braking time t_a=36sec, Effective radius of the brake disc r_{disc}=0.247m, Radius of the wheel r_{wheel}=0.458m, Mean coefficient of friction brake pad μ=0.35, Clamping force F_c=42.1kN, Surface area

of brake pads A_c=400cm2, Maximum temperature under sun=70oC, Maximum temperature under shade=45oC, Factor of Safety=1.5

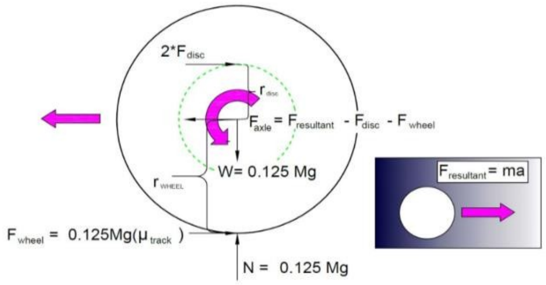

Figure 3: Representation of forces acting on wheel and disc brake

Stopping Distance:

$$S = v_0 t_s - \frac{1}{2} a t_s^2 = 822.24m \qquad (2)$$

Determination of pressure on disc:

Pressure acting on the brake disc (p)

$$= \frac{F_c}{A_c \times \mu} = \frac{42.1 \times 1000}{800 \times (10)^{-4} \times 0.35} = 1.5 \text{ MPa on each side} \qquad (3)$$

Where F_c= Clamping Force (i.e. 42kN)

A_c= Contact area of Brake Pad on each side (i.e. 400 cm^2)

μ= Coefficient of Friction (i.e. 0.35)

Angular Velocity:

$v_0 = r_{wheel} * \omega$

$$\omega = \frac{\text{Velocity}}{\text{radius}} = \frac{v_0}{r_{wheel}} = \frac{44.44}{0.458} = 97.12 \text{ rad/s} \quad (4)$$

Thermal Load:

The kinetic energy for one wheel (disc brake) is equivalent to the energy balance

$$0.125*\frac{1}{2}*M*v_0^2 = \int_0^{t_s} P(t)dt \quad (5)$$

(Mass of rail car is 64000 kg, which is distributed over 8 wheels hence the factor

$$= 2*F_{disc}\int_0^{t_s} v_{disc}\,(t)dt \quad (6)$$

The energy change at the moment is equal to the heat flux on the surface of the disc. The Eq. (6) is valid in the case of constant braking deceleration. The braking force on the disc is equal to Eq. (7)

$$F_{disc} = \frac{0.125*\frac{1}{2}*M*v_0^2}{2*\frac{r_{disc}}{r_{wheel}}(v_0*t_s - \frac{1}{2}*a*t_s^2)} = 8940\text{N} \quad (7)$$

The heat flux at the moment, which affects one half of the disc, is calculated according to the Eq.

$$\text{Area of friction surface} = \frac{\pi}{4}(0.64^2 - 0.35^2) * 2 = 0.45 \text{ m}^2$$

$$Q(t) = \frac{214261 - 5786*t}{0.45} \text{W/m}^2 \quad (8)$$

For the case of emergency braking on horizontal track from 160 kmph to stop, the analysis was carried out in 36 steps, each step being 1s long.

The solution methodology to determine the better material and better geometry for railway brake disc is as follows Dimensions as given in table 1 are taken from Railway specification No. RDSO/2015/CG-03 'Brake disc for LHB type coaches' [5] for modeling of brake discs. These are modelled using Solid Works 2016 software and saved as an IGES file. The 3D model of brake discs is imported to ANSYS WORKBENCH 16. Static Analysis of brake is performed by applying the boundary conditions as given Table 3. Static structural analysis is done by applying the pressure on each side of the disc surface and rotational velocity to investigate the Von Mises and total deformation developed in the models. Coupled analysis is carried out at all-time points with respective temperatures by applying structural loads to obtain the stresses, deformation and strain under the coupled load of thermal and structural. The obtained results are tabulated and compared. The material and geometry with least values is considered as better material and better geometry.

V. Results and Discussion

5.1 Static Structural Analysis (SS):

The model developed in SOLIDWORKS 2016 is converted into IGES file and imported into ANSYS WORKBENCH R16 for analysis. Boundary conditions as calculated in section 3.5 is applied on the brake disc model. The constant pressure of 1.5 MPa is applied on both sides normal to the frictional surface of the disc. Rotational velocity of 97.12 rad/sec (ramped) is given at the axis of the disc. Fixed support is provided in the eye of fixing bolt.

SS-Model 1 - GCI

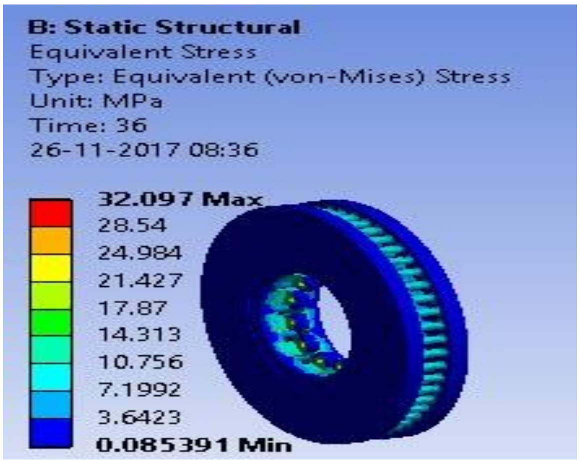

Figure 4:SS_GCI_Profile of Von Mises Stress

SS-Model 1 - AlSiC

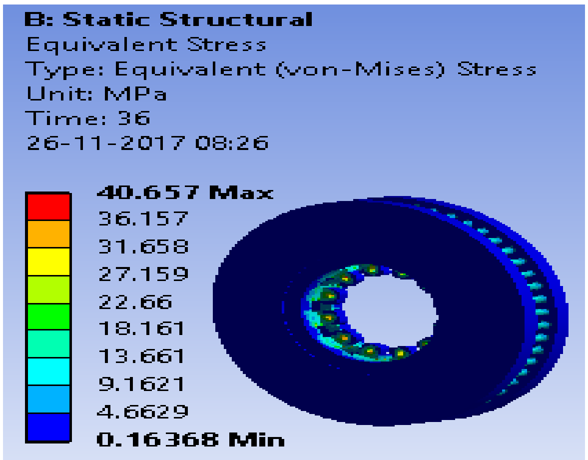

Figure 5: SS_AlSiC_Profile of Von Mises Stress

SS-Model 2 - GCI

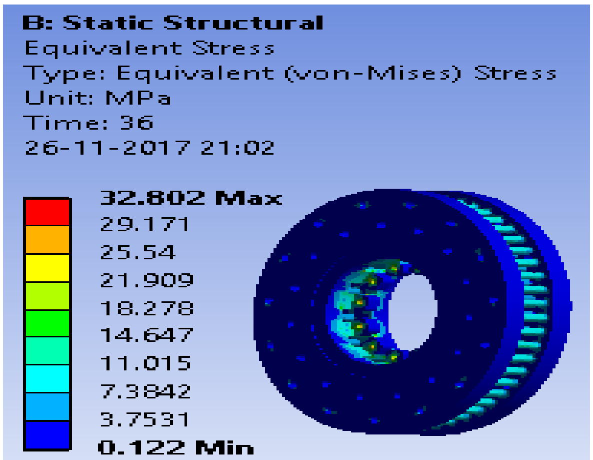

Figure 6: SS_GCI_Profile of Von Mises Stress

SS-Model 2 - AlSiC

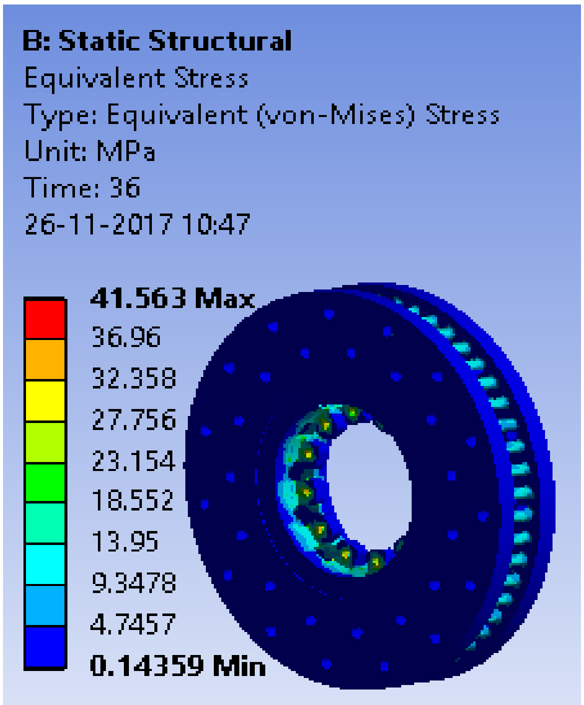

Figure 7: SS_AlSiC_Profile of Von Mises Stress

Both disc temperature and surrounding environment temperature is taken as 450C. Analysis was run for 36 secs i.e. the time taken by the vehicle to stop due to the application of emergency brake. The output results Von Mises stress and total deformation developed in the model were recorded.

5.2 Coupled Analysis (CA)

In coupled analysis, thermal load is coupled with structural load to find out the combined effect on brake disc model. Temperature induced at various time points are imported into Static Structural and then structural loads are applied. Analysis was run for 36 secs i.e. the time taken by the vehicle to stop due to the application of emergency brake. The output results Von Mises stress and total deformation developed in the model were recorded.

CA-Model 1 - GCI

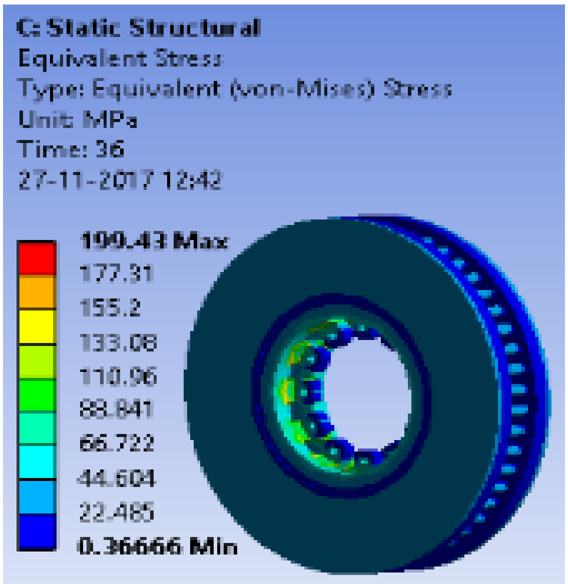

Figure 8: CA_GCI_Profile of Von Mises Stress

CA-Model 1 - AlSiC

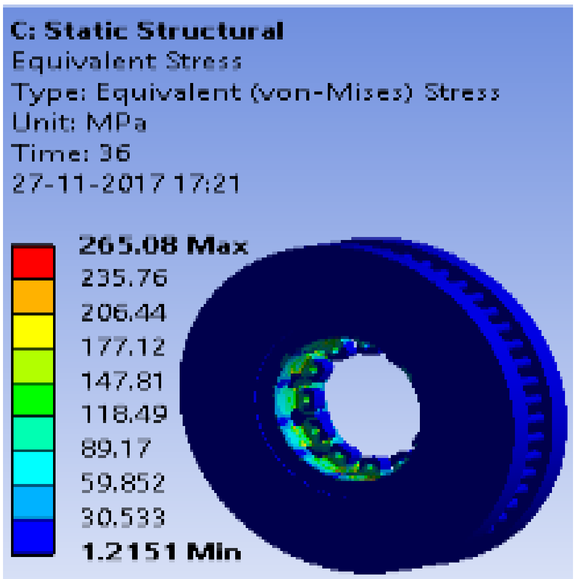

Figure 9:CA_AlSic_Profile of Von Mises Stress

CA-Model 2 - GCI

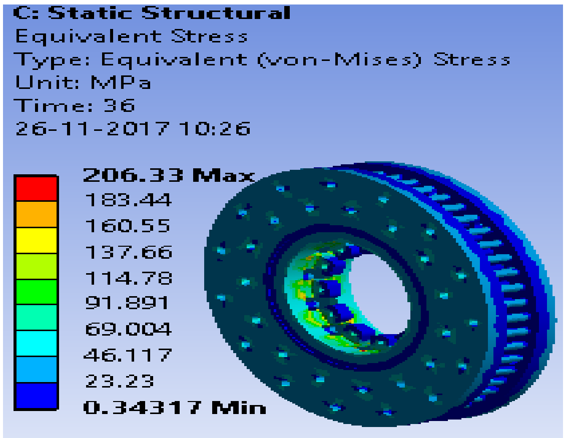

Figure 10: CA_GCI_Profile of Von Mises Stress

CA-MODEL 2 - ALSIC

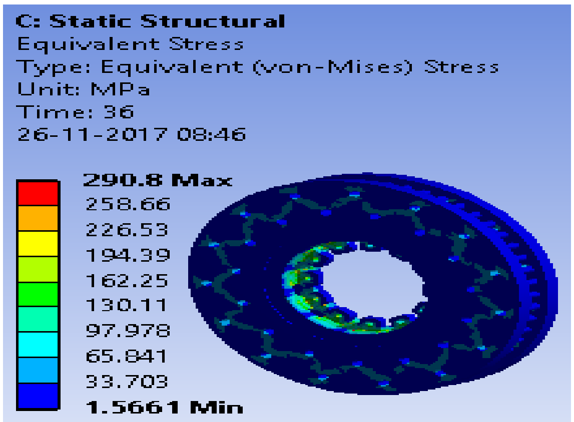

Figure 11:CA_AlSic_Profile of Von Mises Stress

VI. CONCLUSIONS

Some important points that can be drawn from the analysis are:

1. The stresses developed in the models made of AlSiC are lesser than the Ultimate Tensile Strength and when

Table 4: Comparison of Results obtained in both the models

	Static Structural Analysis (Von Mises Stress(MPa))		Coupled Analysis (Von Mises Stress(MPa))	
	Grey cast iron	Aluminum silicon carbide	Grey cast iron	Aluminum silicon carbide
Model 1	32.097	40.657	199.43	265.08
Model 2	32.802	41.563	206.33	290.8

compared to the Factor of Safety of the GCI model (equal to 1.128), the AlSiC models offers higher Factor of Safety (equal to 1.282).

2. For the same applied load, the AlSiC models have lesser deformation and lower thermal stresses than the GCI models and moreover, the weight of AlSiC models is lesser when compared to the GCI models.

3. Even though the stresses developed in the first model are lesser than the second one, the weight of the second model (52kg) is lesser than the weight of the first model (54kg).

4. Depending on the availability and machinability, the first and second models made of AlSiC material can be utilized.

VII. Future Scope

- Suggested model of brake disc is of solid design which need to be pressed on to the axle before pressing of wheels. For replacement of brake disc, wheels are necessarily to be pressed out which is not required if the models are of split type. Hence further experiments may be done with split type brake disc models.
- Carbon Matrix Composites may be used to reduce the sound barrier and higher resistance to temperatures.
- Brake pad analysis with various materials suitable to the disc material may be carried out to optimize pad material.

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