Structural Analysis and Thermal Analysis of Automotive Ventilated Brake Disc

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Abstract—A brake disc rotor forms a part of foundation brake and rotates with the wheel hub assembly. The main function of a foundation brake is to generate a retarding torque by converting mechanical energy to thermal energy by virtue of the frictional work done in relative sliding at the rotor pad interface. In practice, most of the brake discs are made from cast iron and sometimes subjected to high structural and thermal stresses, which can lead to permanent plastic deformation and occasionally rotor cracking. The aim of this paper is to analysis the brake disc under calculated brake torque, temperature and heat flux. A commercial vehicle of 9.6 T is considered for calculation. It is necessary to carry out finite element approach in order to evaluate the exact stress distribution and make sure that the stress values are well below the allowable limits.

I. Introduction

Disc type brake development and its use began in England in the 1890s. Disc brakes were patented by Frederick William Lanchester in 1902 but the commercial use of these brakes started in the early 1950s [1]. A brake disc rotor is the rotating part of a disc brake assembly normally located on the front axle. It consists of a rubbing surface, a top-hat and a neck section. The rubbing surface is where a tangential friction force between the rotor and the stationary pad is generated that gives rise to the brake force in the tire-ground plane which retards the vehicle. The top-hat section is mounted to the hub of the wheel. The connection between the rubbing surface and the hat is known as the neck. Brake discs are favored by most manufacturers of vehicles as the standard foundation brake at the front wheel.

I. Requirements of Brakes

Brake disc is a device which is used to bring the vehicle stop or slow down. Safe operation of the vehicle is dependable on requirements of the brakes. It is observed that during stopping the vehicle, the kinetic energy of the moving parts are converted into heat energy. And this heat energy is absorbed by the brakes and the same is dissipated to the surrounding atmosphere to stop the vehicle. So the brake system must have the following requirements [2]:-

(1) The brakes must be strong enough to stop the vehicle from its maximum test speed.
(2) It should also act as an emergency brake system.
(3) The vehicle must be stopped within minimum stopping distance.
(4) The driver must have proper control over the vehicle during braking.
(5) During braking the vehicle must not skid.
(6) The brake should have well anti fade characteristics even operating at high temperature range.
(7) It should have well anti wear characteristics.

II. Brake Disc Assembly

The brake disc has been more widely used in cars, commercial vehicles and racing cars. Braking power is obtained when the brake pads are pushed against the rotor braking surface which rotates along with the wheel. Mostly, disc brake assembly are installed on the front axle of the vehicle. In some vehicles both the axles are equipped with disc brake assembly.

The brake calliper is bolted on some stationary part which is called as module. A typical brake disc assembly is shown in the fig.1.

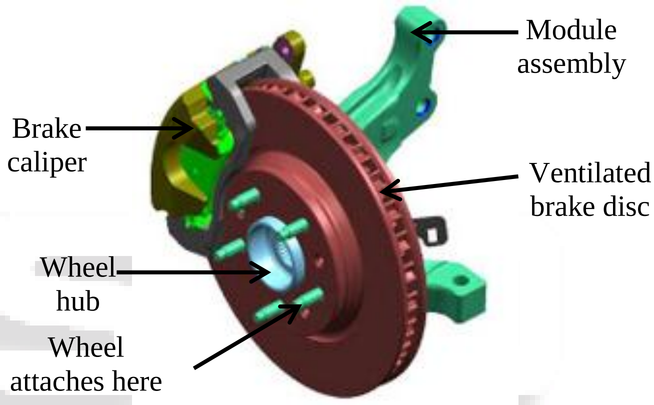

Fig. 1. Brake disc assembly on module.

III. RQUIREMENTS OF BRAKE DISC

The important requirements of the brake disc are as follows [2]:-

(1) It should provide a surface having well anti wear qualities.
(2) It should allow the optimum rate of heat transfer.
(3) Heat is generated at every cycle of brake application and must be dissipated to the atmosphere..
(4) Sufficient strength and minimum weight.
(5) It must be accommodate within the minimum space available.

IV. STRUCTURAL ANALYSIS

Structural analysis of brake disc is probably the most common application of finite element method (FEM). Because the brake disc is subjected to high stresses during braking. ANSYS is useful software for design engineers. This software uses the finite element method to simulate the working conditions of any component and predicts the results about their behaviour. FEM requires the solution of large system of equations. It is powered by various types of solvers. ANSYS make it possible to designers to check and verify the results for their design.

The procedure for developing the prototype brake disc includes the following steps:-

(1) Build the model in any CAD software. Even designers can model the brake disc in ANSYS.
(2) Import the model in ANSYS workbench.
(3) Decide the suitable element size and mesh the model.
(4) Apply boundary conditions.
(5) Evaluate the result.
(6) Modify the design as per result.

V. THERMAL ANALYSIS

Thermal analysis is used to determine the temperature distribution, thermal gradient, rate of heat flow and heat flux in an object that occurred due to thermal loads. Such load includes the following parameters [2]:-

(1) Convection
(2) Radiation
(3) Heat flow rates
(4) Heat flow per unit is i.e. Heat Flux.
(5) Heat flow per unit volume i.e. Heat Generation Rate.
(6) Constant temperature boundaries.

The thermal analysis may be either linear or nonlinear. Linear includes with material properties and nonlinear includes material properties that depend on temperature. The thermal properties of most material vary with nonlinear.

VI. TRANSIENT THERMAL ANALYSIS

Transient thermal analysis is the combined effect that occurs due to structural and thermal loads. This analysis determines the temperature and other thermal quantities. Designers commonly use the temperature that transient thermal analysis calculates as input to structural analysis for thermal stress evaluation. Many types of applications such as engine blocks, hot piping systems and exhaust systems are analysed in transient analysis. The main difference is that most applied loads in a transient analysis are the functions of time [4]. To specify the time dependent loads, designers can use the function tool to define an equation or function describing the curve or then apply the function as boundary condition.

VII. PROBLEM DEFINITION

The original brake disc on front axle that has been used previously is modified for some functional requirements. The modification is done for medium capacity of commercial vehicle having gross vehicle weight as 9.6T. Due to the modification the dimension and the profile of the brake disc has changed. And material optimization is the main concern for this commercial vehicle application. As the brake disc is modified, it is necessary to check that its function properly under given load conditions.

VIII. VEHICLE SPECIFICATIONS

For calculating the braking torque, heat flux and single stop temperature rise, we need vehicle specifications. The vehicle specifications include various parameters which are mentioned in below table I.

Table I

Vehicle parameters		
Sr. No.	Description	values
1	Gross vehicle weight (M)	9.6 T
2	Wheel base (WB)	5.22 m
3	Maximum vehicle speed (V_1)	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)	2.77m
7	Tire rolling radius (R_t)	0.445 m
8	Stopping distance (S_d)	55 m

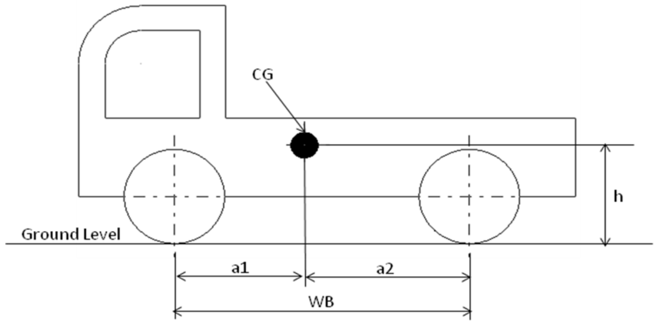

Fig. 2 : Vehicle dimensions

IX. 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.

a. *Static load distribution (R_f)*: - 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 [7].

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

Calculated static load distributions on front axle is **5094 kg**.

b. *Mean fully developed deceleration (MFDD):-* MFDD can be calculated by considering the maximum test speed (V_1) 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, V_b is the 80% of V_1 in km/h, V_e is final speed of the vehicle after braking in km/h as 10% of V_1, S_e is the total distance travelled by the vehicle in meters during braking and S_b is the final distance travelled in meters i.e. zero.

The calculated MFDD or $\dot{\upsilon}$ is **4.41 m/s²** i.e. **0.45g units**.

c. *Dynamic load distribution after braking (F_{zf}): -* 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 [5]

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

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

d. *Total load on front axle due to braking (T_f): -* 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 [5]

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

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

e. *Brake force on front axle (B_f): -* 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, μ_r 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 μ_r as **0.4**.

The brake force acting on front axle is the function of total load acting and coefficient of adhesion (μr). Below table II shows the different values of coefficient of adhesion as per different road conditions [6].

TABLE II

Surface	Values
Asphalt and concrete (dry)	08 - 0.9
Asphalt (wet)	0.5 - 0.6
Concrete road	0.75
Earth road (dry)	0.68
earth road (wet)	0.55
Gravel	0.6
Tar road	0.4
Ice surface	0.1
Snow (hard packed)	0.2

f. *Brake force acting on per wheel of front axle (B_{fw}): -* 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.**

g. *Brake torque acting on per wheel of front axle (T_{bw}):-* The brake torque acting on per wheel of front axle is the function of brake force and tire rolling radius (R_t) 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 14.5 workbench, the value of braking torque **2000 kg-m** is taken further.

X. Heat Flux And Temperature Calculation

Frictional heat is generated at the rubbing surface due to the interactions between the pad and disc. The energy must be quickly dissipated to the surrounding air. Radiation also helps to dissipate the heat energy stored within the rotor when the temperature is high. The non-uniform heat flux input to the disc brake is calculated from the non-uniform pressure distribution, friction coefficient and sliding velocity along the disc-pad interface. The amount of heat flux that flow into each component depends on the disc and pad material.

Following steps are followed to calculate heat flux and single stop temperature rise.

a. *Kinetic energy (KE)*:- Following equation is used to calculate the kinetic energy of the vehicle travelling with 100 km/h, .i.e. **27.77 m/s**.

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

Whereas, v is the speed of vehicle in m/s. The calculated kinetic energy is **3703704 J.**

b. *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**.

c. *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 **3814815 J**.

d. *Disc usable area (A_d):-* 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**.

e. *Time required to stop the vehicle (t_s):-* 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**.

f. *Braking power (P_b) and highest braking power ($P_{b(0)}$):-* Braking power is the ratio of total energy to time required to stop the vehicle. It can be expressed in below equation [6]

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

The calculated braking power is **607455 Watt**.
And highest brake power ($P_{b(0)}$) produced at the onset of braking can be calculated with the below equation [6] and can be also shown in fig.4.

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

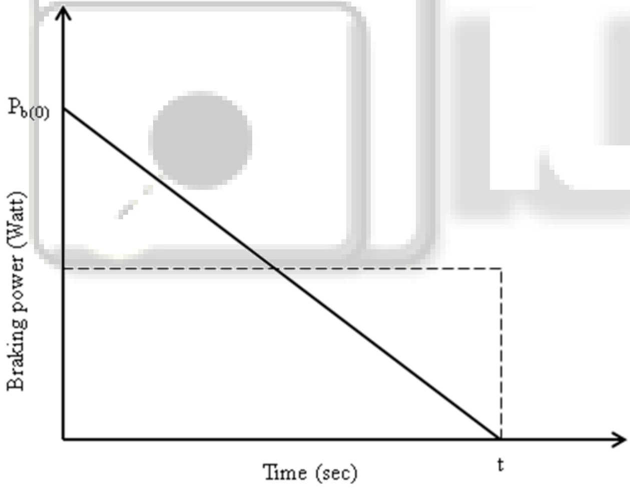

Fig.3. Plot for brake power vs. Time [6].

The calculated onset of braking power is **1214910 Watt**.

g. *Power distributed per disc (P_d):-* 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 [6].

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

The value of power distributed per disc is **303728 Watt**.

h. *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 [6].

$$q = P_d / 2A_d$$

The calculated heat flux will be **2664277 Watt/m²**.

i. *Single stop temperature rise (T_s):-* 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 [6].

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

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 (T_{amb}) can be calculated with help of below equation

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

Following materials were selected for the modified brake disc. The calculated final temperature for single stop is tabulated in the table III based on their respective material properties

TABLE III

Material	ρ kg/m3	Cp J/kg-k	K W/m-k	T °K
GG26Cr	7100	490	35	**644**
FG220MoCr	7800	480	49	**603**

Further these respective temperatures are applied on the respective disc's material in steady state thermal analysis using ANSYS 14.5 workbench.

XI. ASSUMPTIONS

Following assumptions [4] were made while calculating the heat flux and temperature rise of the brake disc:-

1. Brakes are applied on all the four wheels.
2. The analysis is based on pure thermal loading .The analysis does not determine the life of the disc brake.
3. Only ambient air-cooling is taken in to account and no forced convection is taken.
4. The kinetic energy of the vehicle is lost through the brake discs i.e. no heat loss between the tyres and the road surface and the deceleration is uniform.
5. The thermal conductivity of the material used for the analysis is uniform throughout.
6. The specific heat of the material used is constant throughout and does not change with the temperature.

XII. RESULTS OF STRUCTURAL ANALYSIS

A. *Results for GG26Cr*

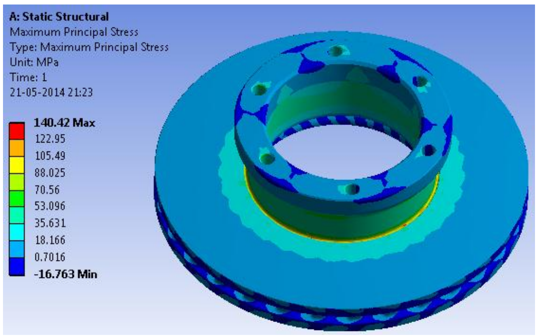

Fig. 4: Maximum principle stress

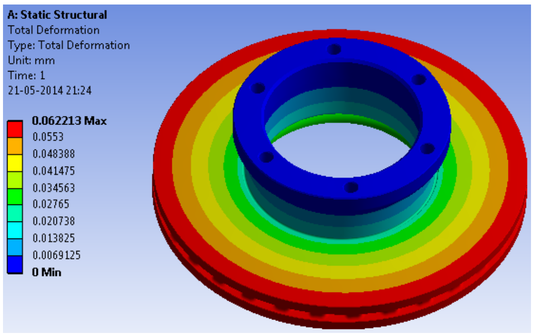

Fig. 5 : Total deformation

B. *Results for FG220MoCr*

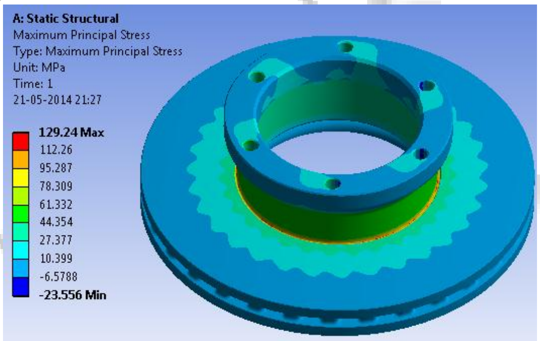

Fig. 6 : Maximum principle stress

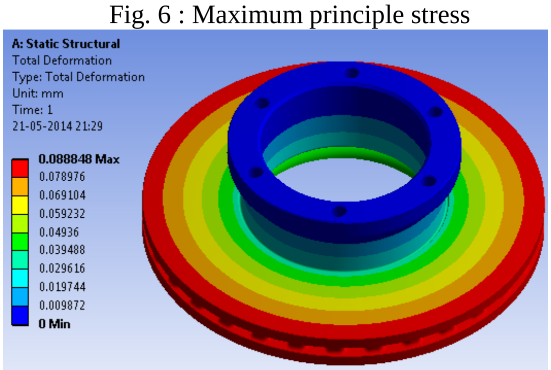

Fig. 7 : Total deformation

Summary of structural analysis is tabulated in table IV.

TABLE IV

Material	Max. principle stress (Mpa)	Total deformation (mm)
GG26Cr	**140.42**	**0.0622**
FG220MoCr	**129.24**	**0.0888**

C. *Results of Thermal Analysis*

1) *Temperature distribution for GG26Cr.*

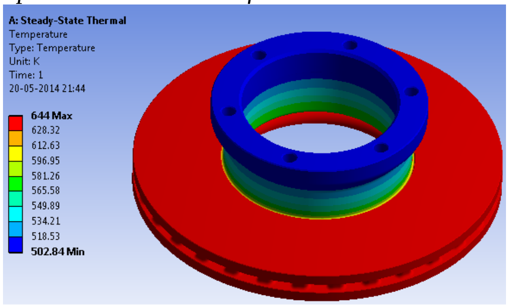

Fig. 8 : Temperature distribution

2) *Total heat flux analysis for GG26Cr.*

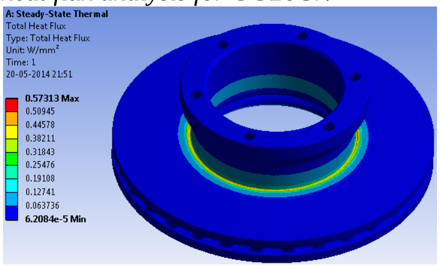

Fig. 9 : Total heat flux analysis

3) *Temperature distribution for FG220MoCr.*

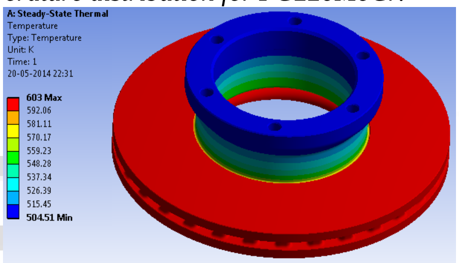

Fig. 10 :Temperature distribution

4) *Total heat flux analysis for FG220MoCr*

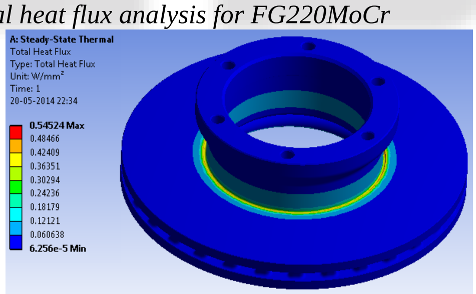

Fig. 11 :Total heat flux analysis.

XIII. CONCLUSIONS

The following conclusions are drawn from the current work.

(1) The above braking parameters are calculated for 9.6T commercial vehicle. Braking standard IS: 11852 is referred to decide the vehicle class and MFDD.
(2) Braking torque, heat flux and single stop temperature rise are calculated with the help of vehicle specifications.
(3) Calculated final braking torque is taken for structural analysis.
(4) Maximum heat flux and single stop temperature rise are calculated for two different materials, i.e. GG26Cr and FG220MoCr. Further temperature distribution and heat flux are analysed in ANSYS 14.5 workbench.

(5) Structural analysis is carried out by fixing the disc at mounting holes and applying the calculated braking torque.
(6) Maximum principle stress for the material GG26Cr and FG220MoCr are 140.42 Mpa and 129.24 Mpa respectively.
(7) Total deformation observed during analysis in ANSYS for GG26Cr and FG220MoCr are 0.0622mm and 0.0888mm respectively.
(8) Comparing the results of two different materials, it is concluded that the material GG26Cr is suitable for the 9.6T commercial vehicle brake disc.

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REFERENCES

[1] Daniel Das. A, C. Reegan Raj. A, Preethy. S and R. Bharani. G, “Structural and Thermal Analysis of Disc Brake in Automobiles”, International Journal of Latest Trends in Engineering and Technology, vol. 2, Issue 3, pp 18-25, May 2013.

[2] Anup Kumar and R. Sabarish, “Structural and Thermal Analysis of Brake Drum”, Middle East Journal of Scientific Research 20(8):1012-1016, 2014, ISSN 1990-9233

[3] N. Reddy, K. Reddy and N. Ganesh “Design, Structural and Thermal Analysis of Brake”, International Journal of Futuristic Science Engineering and Technology, vol. 1, Issue 3, pp 167-171, March 2013.

[4] G. Nathi, T. Charyulu, K Gowthan and P. Reddy, “Coupled Stuctural / Thermal Analysis of Disc Brake”, International Journal for Research in Engineering and Technology, vol. 1, Issue 4, pp 539-553, December 2012.

[5] IS: 11852, Indian Standard for Automotive Vehicles– Brakes and Braking Systems, Part 3, Performance Requirements and Evaluation.

[6] “Brake Design and Safety” by Rudolf Limpert, SAE Publication 2009.

[7] “Vehicle Dynamics ” by Prof. Dr. Georg Rill, Hochschule Regensburg University of Applied Science.