Design and Stress Analysis of Disc Brake

Hanamant Yaragudri[1] & Rajesh A[2*]

[1]Assistant Professor, Department of Mechanical Engineering
New Horizon College of Engineering, Bangalore, India

[1]E-mail: *hanamant0011@gmail.com*

[2]Assistant professor, Department of Mechanical Engineering
New Horizon College of Engineering, Bangalore, India

[2]E-mail: *rajesh.inspiring@gmail.com**

Abstract: Contact analysis forms part of the whole procedure in the complex eigen value method. The essence of such a method lies in the asymmetric stiffness matrix derived from the contact stiffness and the friction coefficient at the disc-pads interfaces. This leads to complicated mathematical equations and it is very time consuming process. The main aim of this paper is towards the analysis of the contact stress distributions at the disc-pad interfaces using a detailed 3-dimensional finite element model of a real car disc brake. A general-purpose commercial software package (Hypermesh 9.0 andAnsys 12.0) has been utilized and assessed. An investigation is made for different modifications on the geometry and materials of disc brake components were performed to search for a more uniform contact stress distribution. It is believed that a uniform contact stress distribution could prevent excessive tapered wear on the pads and subsequently could prolong the life of pads. In this work, the model is created using Catia V5. The solid model is taken to hypermesh software for meshing. Vibration analysis has been carried out using Ansys software from which nature frequency and maximum displacement of the component have been determined and the results of natural frequency is compared with experimental data provided for the existing disc. Analysis has been done for finding the stress values for different material sets and selecting set of materials which gives minimum stress value.

Keywords: Disc Brake, Contact Analysis, Natural Frequency, Disc Materials

I. INTRODUCTION

In general, there are three main functions of a brake system, i.e., to maintain a vehicle's speed when driving downhill, to reduce a vehicle's speed when necessary and to hold a vehicle when in parking. Today, most passenger vehicles are fitted with disc brake systems. A disc brake of floating caliper design typically consists of two pads, caliper, disc, piston, carrier bracket and two guide pins. One of the major requirements of the caliper is to press pads against the disc and it should ideally achieve uniform interface pressure as possible. But it has been found that more wear appears on the leading side than the trailing side. This agrees with the result that higher pressure occurred on the leading side when the disc starts to slide. Limpert[1] stated that uniform pad wear and brake temperature, and more even friction coefficient could only be achieved when stress distributions between the pads and disc are uniform. In addition, unevenness of the pressure distribution causes uneven wear and consequently shortens the life of pads. This might lead to dissatisfaction to the customers who need to visit their garage more frequently in order to replace tapered wear pads. Sincelast decade advent of powerful finite element analysis (FEA) packages have proven good tool to accurately analyze them. The complicated geometry of disc brake and the complex force applied by calipers make their analysis difficult. But optimized meshing and accurate simulation of boundary conditions alongwith ability to apply force, provided by various FEM packages have helped the designer to carry structural and modal vibration analysis with the investigation of critical stresses. FEM enables to find critical locations and quantitative analysis of the stress distribution and deformed shapes under loads. However detailed modeling and specialized knowledge of FEM theory are indispensable to perform these analyses with high accuracy. They also require complicated meshing strategies.

A. Methodology

The following are the main objectives of the project.

- Building a 3-D Solid parametric model of disc brake and calliper in CATIA V5.
- Meshing the model by Tetrahedral Solid 45, MPC 184 elements in Hypermesh.
- Specifying frictional contact at pad-disc interface and slider-taper bolt interface in Hypermesh.
- Rigid body modes and Normal modes calculated in free vibration analysis for disc brake in Ansys for existing model.
- Comparing the experimental data for existing model with Ansys results for validification of meshing.
- Specifying boundary conditions and loads in Ansys.
- Performing analysis for finding the stress values for different material sets and select the set of material which gives minimum stress value.
- Analyzing of the contact stress distributions at the disc and pad interfaces for selected material set.Investigating different modifications on the geometry and materials of disc brake components for finding a more uniform contact stress distribution without changing the original shape of all components.

B. Disc Brake systems

Many of the vehicles of the 1950s and 1960s used four wheel-drum brakes. These systems used single- or dual-master cylinders. In most cars of that time, all four wheels held plain drums. Performance cars used finned drums.

Drum brakes, however, presented several different problems. First, they retained water, which caused large amounts of brake fade during rainstorms or after going through a puddle. Secondly, they didn't easily get rid of heat and would also fade going down long hills or after repeated hard stops. Finally, their braking distances were much longer than disc brakes. In order to remove these problems with drum braking systems, disc brakes were developed.

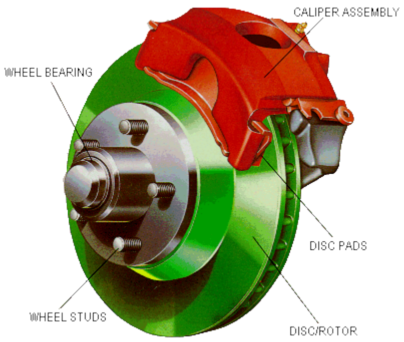

Figure 1. Parts of a disc brake [10]

C. Construction and working of disc brakes

A disc brake consists of a cast iron disc bolted to the wheel hub and a stationary housing called caliper. The caliper is connected to some stationary part of the vehicle, like the axle casing or the stub axle and is cast in two parts, each containing a piston. In between each piston and the disc, there is a friction pad is held in position by retaining pins, spring plates etc. Passages are drilled in the caliper for the fluid to enter or leave each housing. These passages are also connected to another one for bleeding. Each cylinder contains a rubber sealing ring between the cylinder and the piston.

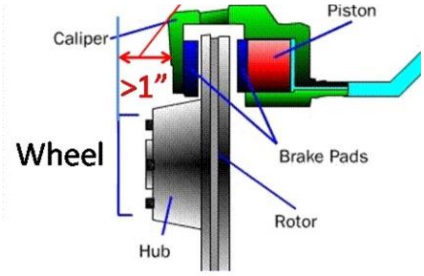

Figure 2. A side view of a typical disc brake system. [11]

When the brakes are applied, hydraulically actuated pistons move the friction pads into contact with the disc, applying equal and opposite forces on the later. On releasing the brakes, rubber sealing rings act as return springs and retract the pistons and the friction pads away from the disc.

Special types of disc brakes include the swinging caliper type and the sliding caliper type. In the swinging caliper type, the caliper is hinged about a fulcrum pin and one of the friction pads is fixed to the caliper. The fluid under pressure presses the other pad against the disc to apply the brake. The reaction on the caliper causes it to move the fixed pad inward slightly applying equal pressure to the other side of the disc. The caliper automatically adjusts its position by swinging about the pin. In the sliding caliper type there are two pistons between which the fluid under the pressure is sent which presses one friction pad directly onto the disc, where as the other pad is pressed indirectly via the caliper. Both these types are self adjusting and have resulted in simpler and lighter construction.

II. SOLID MODELING OF DISC BRAKE ASSEMBLY

A. Modeling Details

For generation of a 3-D model, 2-D orthographic views are required. The 2-D drawings of every individual part and their assembly are prepared as shown in figure 3, figure 4 and figure 5. Using 2-D drawings one can prepare isometric views of a component and using that solid model is generated. A feature based modeling technique is used for every individual part. These parts are assembled to get complete disc brake assembly. After the assembly, fine fillets and chamfer details at shaft ends, transition sections, joint between web and shaft etc. are created by surface generation techniques. Finally Boolean operation is performed to extract the required geometry.

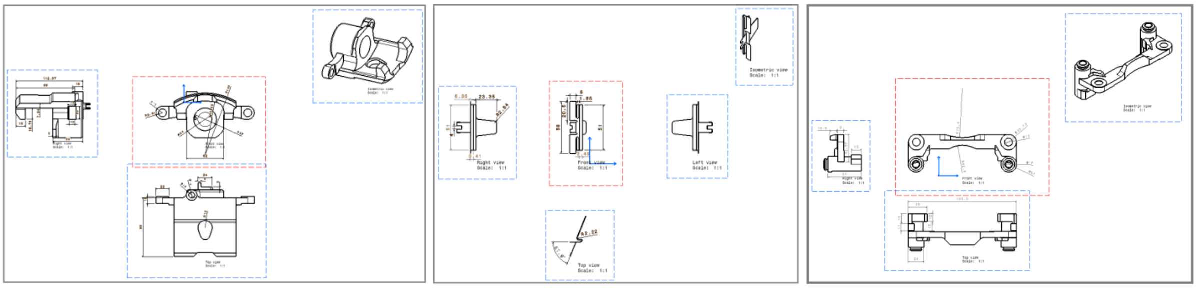

Figure 3. 2D and isometric view of caliper, clip and slider.

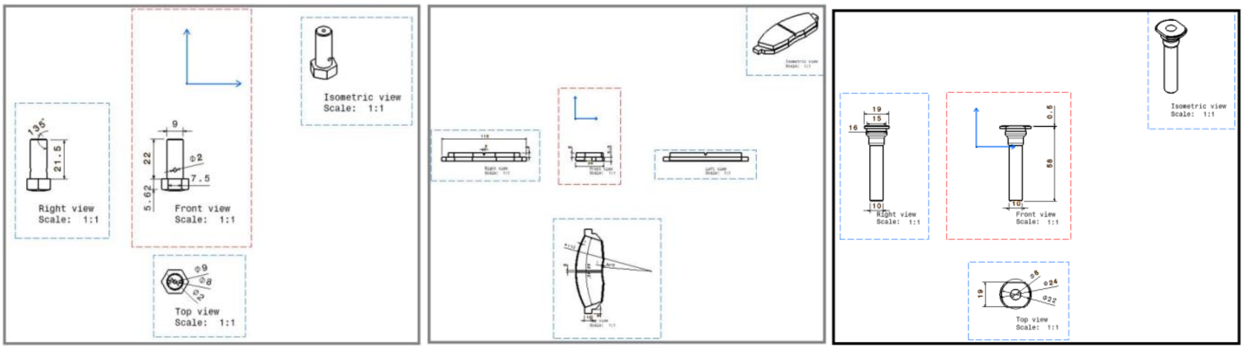

Figure 4. 2D and isometric view of cylinder bolt, pad and tapper bolt

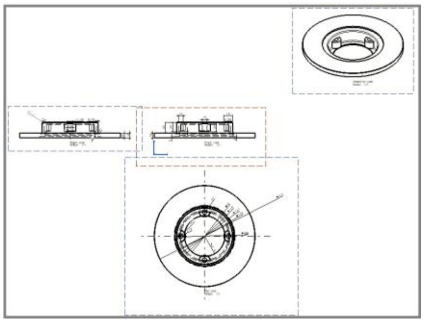

Figure 5. 2D and isometric view of disc

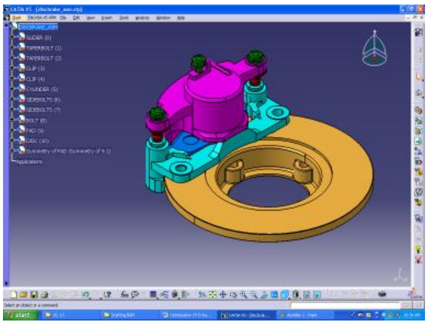

Figure 6. 3D model of disc brake

*B. **Meshing of disc brake assembly:***

Fig.7 shows fine mesh model of disc brake assembly using hypermesh. It ensures finer mesh around fillets. These are the areas of concern because the load gets applied over them. Finer mesh model caused hardware disc space problem for final dynamic response analysis. It required more than 4 GB space on disc. To solve this problem, a coarse mesh was generated.After first solve with finer mesh it was observed that only fillet area is critical. So, the finer mesh in other areas is replaced by coarse mesh. For this the original fillet area, which was fine mesh, is replaced with coarse part by applying 'displacement constraints' at interface between finer mesh and coarse mesh. These constraints will ensure that at interface both nodes will have same rotationalcomponent of displacement.

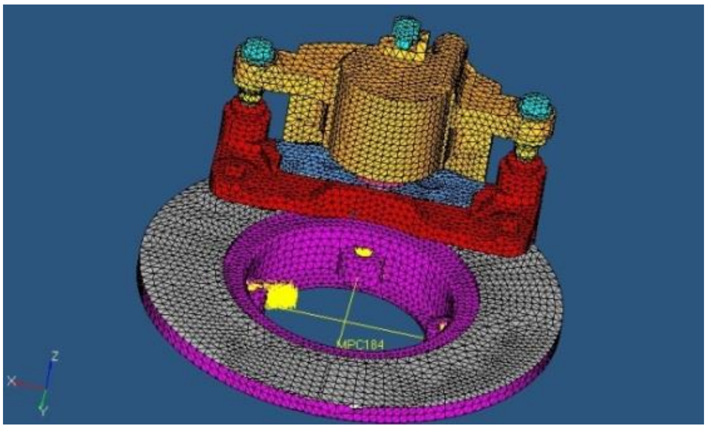

Figure 7. Disc brake assembly.

III. CALCULATIONS

Table 1. Mass Moment of Inertia, Centre of Gravity Calculations for Disc.

Value of mass = 2.27 kgs .	*Mass moment of inertia about origin*	*Mass moment of inertia about C.G.*
Centre of gravity-	I_{xx} = 44.84 mm^4	I_{xx} = 8.689 mm^4
X_c = 36.230 mm	I_{yy} = 17.31 mm^4	I_{yy} = 8.643 mm^4
Y_c = 108.46 mm	I_{zz}= 51.35 mm^4	I_{zz} = 16.96 mm^4
Z_c = -44.536 mm	I_{xy} = -10.33 mm^4	I_{xy} = -0.4447E-03 mm^4
	I_{yz} = 12.70 mm^4	I_{yx} = -0.9523E-03 mm
	I_{zx} = 4.243 mm^4	I_{zy} = -0.1284E-03 mm^4

IV. VIBRATION ANALYSIS FOR VARIOUS MODES

Table 2. Ansys Result for Specific Disc Brake.

Mode	*Ansys*	*Type Of Mode*	*Mode*	*Ansys*	*Type Of Mode*
1	0.0000	Rigid body mode	11	3578.4	
2	0.0000		12	4153.5	
3	0.0000		13	4191.0	
4	1.99795E-03		14	4381.8	
5	5.80192E-03		15	4419.5	
6	6.21154E-02		16	58441.1	Combined mode
7	1582.5	First bending mode	17	5852.1	
8	1695.7		18	7444.3	
9	3058.7		19	7534.2	
10	3569.2		20	8599.7	

A. *Displacement plot of various modes of disc.*

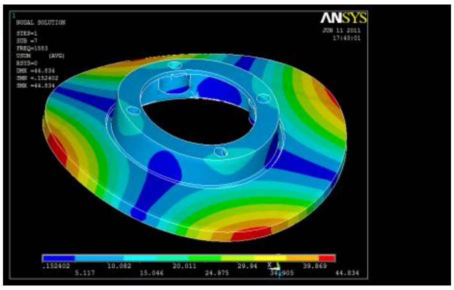

Figure 8. Displacement plots of mode no 7

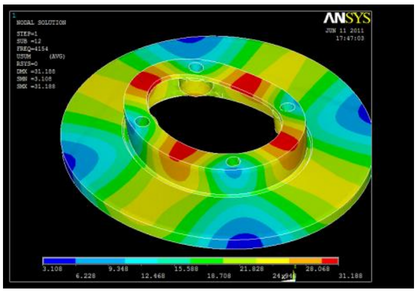

Figure 9. Displacement plots of mode no 12

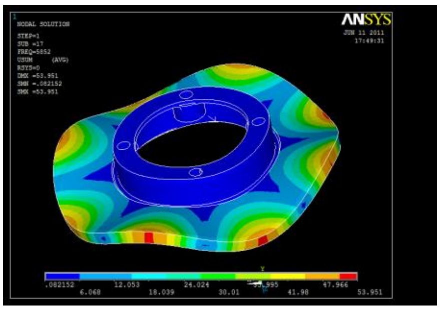

Figure 10. Displacement plots of mode no 17

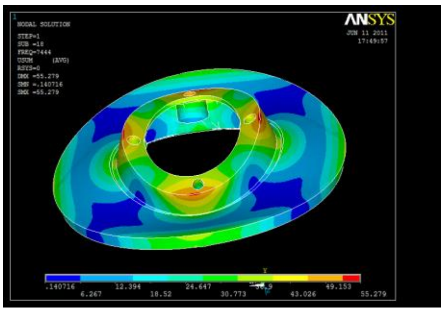

Figure 11. Displacement plots of mode no 18

Table 3. Displacement Result for Various Modes of Disc.

Sl. No	*Mode Number*	*Frequency Hz*	*Displacement, mm*	
			Maximum	*Minimum*
1	7th	1583	44. 834	0.152
2	12th	4154.2	31.188	3.108
3	17th	5852.1	53.951	0.082152
4	18th	7444.3	55.279	0.140716

Table 4. Comparison of Experimental and Ansys Results.

Sl. NO	*Natural frequency for ANSYS in Hz*	*Experimental natural freq. in Hz*	*Percentage deviation (error in %)*
1.	1582.5	1497.2	5.39
2	1595.7	1523.5	4.53
3	3058.7	2893.3	5.05
4	3578.4	3450.0	3.59
5	4153.5	4028.3	3.021
6	4419.5	4487.4	1.54
7	58441.1	5775.0	1.18
8	5852.1	5775.0	1.13
9	7444.3	7360.1	1.131

V. CONTACT ANALYSIS

A. *Introduction*

Contact conditions are formed where bodies meet. You can transfer structural loads and heat flows across the contact boundaries and connect the various bodies. There are no limits placed on the number of bodies that comprise an assembly. Depending on the type of contact, the analysis can be linear or nonlinear. A nonlinear analysis can increase runtime significantly, as the solver will internally run iterations to arrive at a converged solution.

B. *Contact Elements:*

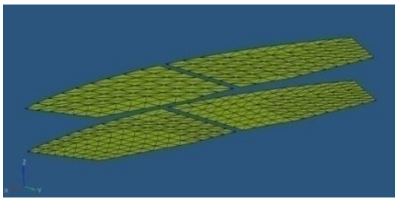

Figure 12. Pad as Contact Elements (with μ=0.43)

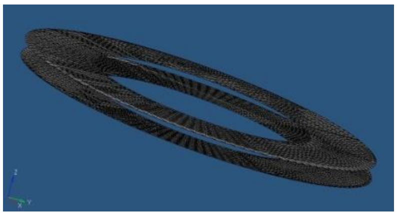

Figure 13.Disc as Target Elements

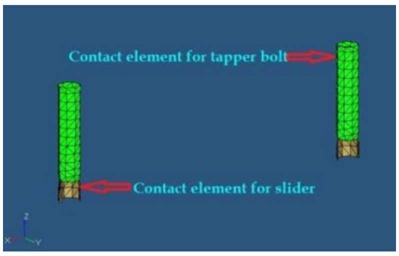

Figure 14. Contact for Slider and Tapper Bolt

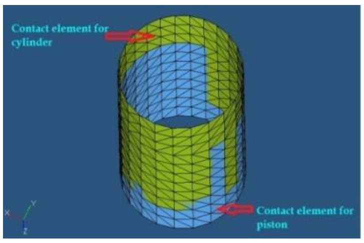

Figure 15. Contact for piston and cylinder

C. Boundary Conditions

Here various constraints are applied for different parts of the disc brake assembly.

- *Constraints for Slider:* Slider is constrained in all DOF's at bolting holes through which caliper is bolted to axel as shown in figure 16.
- *Constraints for Disc:* Disk is constrained in Z direction. As it bolted to wheel hub, but it free to rotate in XY plane (for rotation of disc) as shown in figure 17.
- *Constraints for MPC's:* MPC's are constrained in X, Y,& Z direction to avoid motion of disc in Plane (i.e.in radial direction) and axially direction but it free for rotational as shown in figure 18.
- *Constraints for Pad:* The various constraint for pad are applied at Face_1 and Face_2 in Y direction and Face_3 and Face_4 in X-direction as shown in figure 19.

Constraints for Disc:

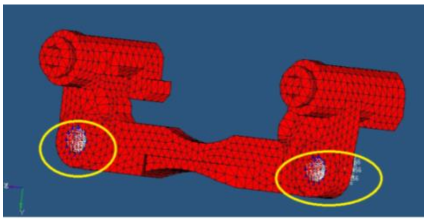

Figure 16. Constrained Slider.

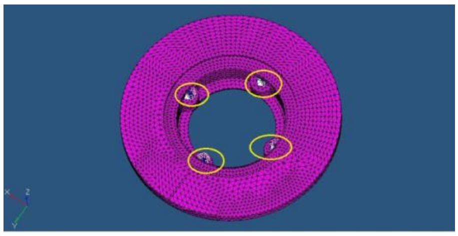

Figure 17. Disk is constrained in Z direction.

Constraints for MPC's:

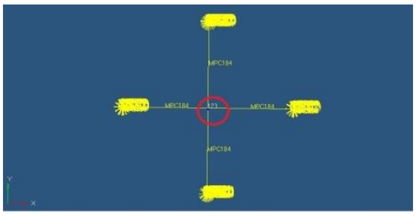

Figure 18. MPC's are constrained in X, Y& Z directions.

Constraints for Pad:

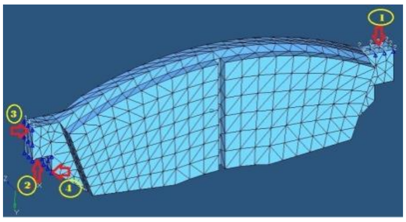

Figure 19. The various constraints for pad.

D. ***Loading conditions****:*

- Pressure of 15.00 MPa is applied at top of the piston.
- Pressure of 15.00 MPa is applied in cylinder of caliper.
- Moment or Torque of 710 × 10^3 N-mm at center of Disc.

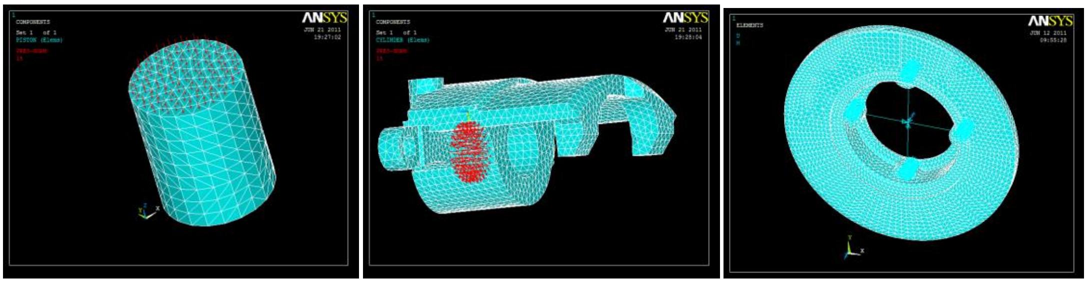

Figure 20. Pressure acting on piston, cylinder of caliperand Moment acting on disc.

VI. MATERIAL SETS

Table 5. Disc Materials and Pad Materials

Disc Materials	***E (N/mm²)***	***v***	***ρ (tones/mm³)***	***Pad Materials***	***E (N/mm²)***	***v***	***ρ (tones/mm³)***
Structural Steel	2e5	0.26	7.89e-9	SAE 1055 Steel	1.9e5	0.30	7.89e-9
SAE J431 G3000	1.7e5	0.26	7.23e-9	SAE J661	2.8e5	0.26	9.8e-9
BS1452 Grade 220	1.2e5	0.29	7.62e-9	SAE 1055 Steel	1.9e5	0.30	7.89e-9
BS1452 Grade 260	1.015e5	0.26	7.21e-9	SAE 1055 Steel	1.9e5	0.30	7.89e-9
BS1452 Grade 250	1.056e5	0.27	7.53e-9	SAE J2430	2.5e5	0.27	8.7e-9

A. ***Stress and displacement plots are given below for 1st set***

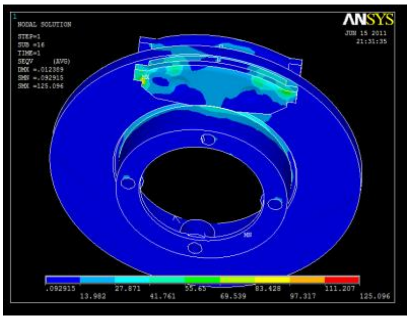

Figure21. Stress plot for 1st set of material.

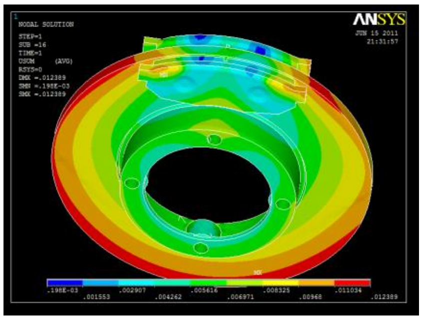

Figure 22.Displacement plot for 1st set of material

B. ***Stress and displacement plots are given below for 2nd set***

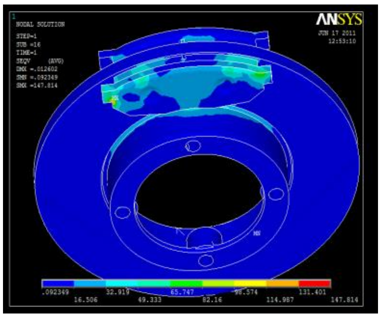

Figure 23. Stress plot for 2nd set of material.

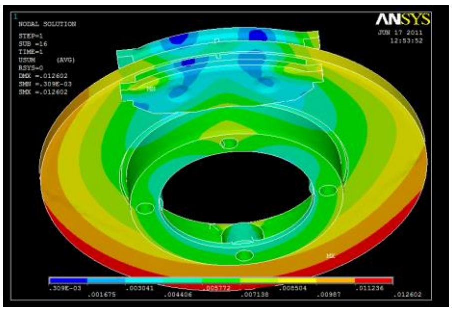

Figure 24. Displacement plot for 2nd set of material

C. ***Stress and displacement plots are given below for 3rd set***

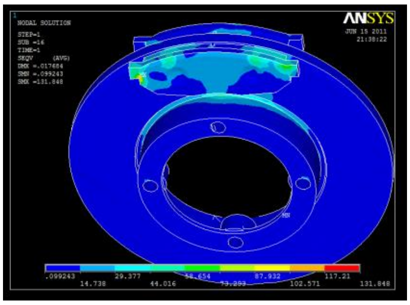

Figure 25. Stress plot for 3rd set of material.

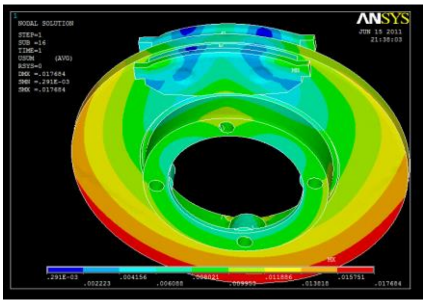

Figure 26. Displacement plot for 3rd set of material

D. ***Stress and displacement plots are given below for 4th set***

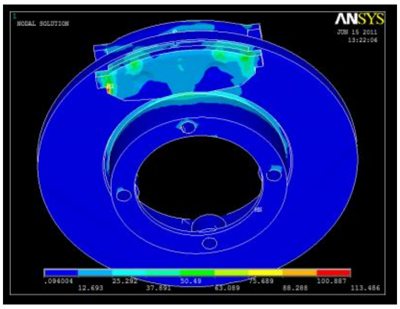

Figure27. Stress plot for 4th set of material.

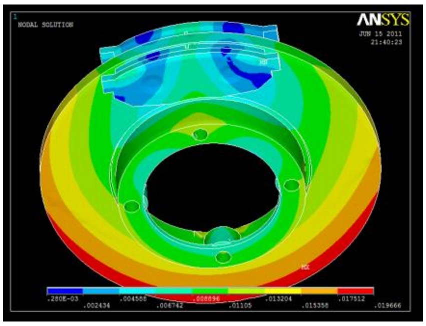

Figure 28. Displacement plot for 4th set of material

E. ***Stress and displacement plots are given below for 5th set***

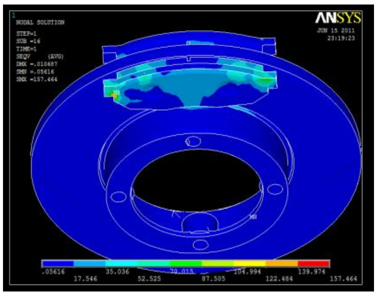

Figure 29. Stress plot for 5th set of material.

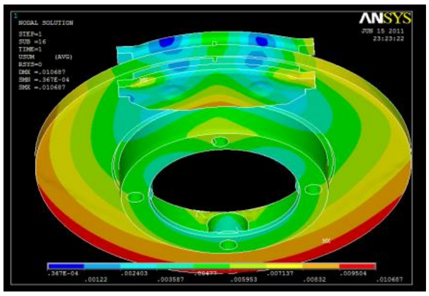

Figure 30. Displacement plot for 5th set of material

Table 6. Maximum stress displacement values for given material sets obtained by Ansys.

Material Sets	*Max.Stress Values(MPa)*	*Max. DisplacementValues (mm.)*
1st Set	125.096	0.012389
2nd Set	147.814	0.012602
3rd Set	131.848	0.010329
4th Set	113.486	0.01966
5th Set	157.464	0. 010687

Conclusion: 4th set of material gives minimum value of stress compared to other sets of materials.Disc brake system with 4th material set will used for further analysis.

VII. CONTACT ANALYSIS FOR STRESS DISTRIBUTION OVER PAD CONTACT SURFACE

A. *Stress distribution for both pad*

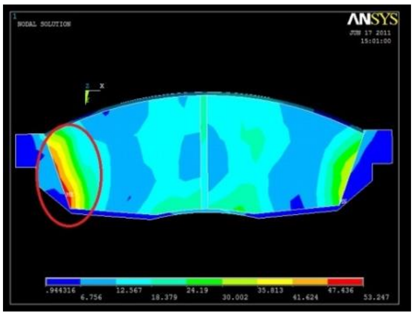

Figure 31. Stress distribution for Pad 1

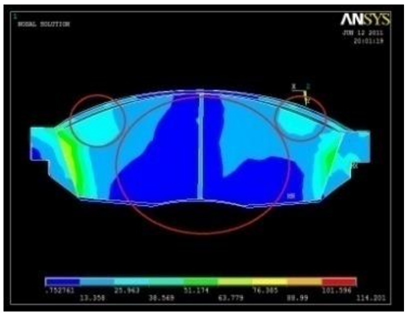

Figure 32. Stress distribution for Pad 2

- For pad_1 result showshigh stress at leading edge (53.247 MPa) compared to trailing edge (41.624MPa).Figure 31 and figure 32 showing the finger side pad upper portion is more stressed and middle surface is unaffectedand variation of stresses on face is 13.358 MPa to 101.596 MPa. Piston side pad shows more stress at leading edge. This is mainly because of higher pressure occurred on the leading side when the disc starts to slide as shown in figure 33.

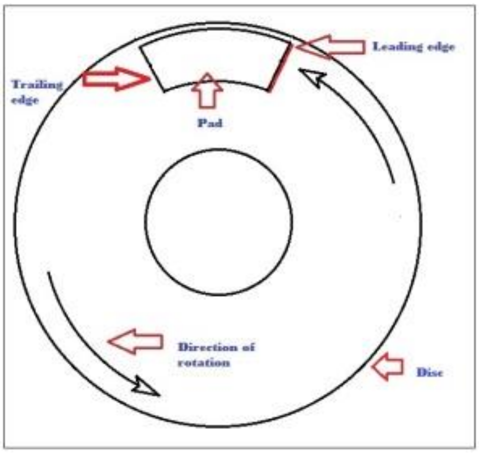

Figure 33. Action of disc rotation on leading and trailing edge of a pad

- Caliper bends while transferring the load and it has been found in the contourplot that the bending occurs since the fingers presses the upper portion of pad against the disc, this is mainly because the caliper get bend by taking support of upper portion of the pad, due to this pad_2 shows more stresses at the corresponding regions.

Figure 34. Bending of caliper.

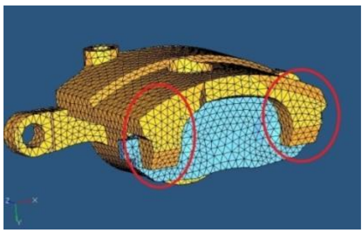

Figure 35. Contact between pad and disc.

B. ***Modifications:*** Based on the reasonsthe following threemodifications have been done

Table 7.Modifications

Modifications	***Descriptions***	***Changes***
1	Partial connectionfor piston	Figure 35
2	Stiffer caliper	E = 700 GPa
3	Partial connectionfor finger pad	Figure 38

- Partial connection between piston head and pad: Piston side pad shows more stress at leading edge. This is mainly because of higher pressure occurred on the leading side when the disc starts to slide. For reducing pressure on pad a partial contact has been madebetween the piston and pad at leading edge side. Nodes show elimination of contact (figure 36).

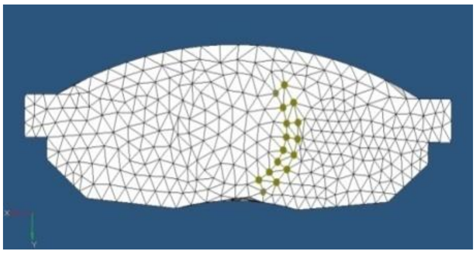

Figure 36. Partial connections in the axial direction for piston pad (the dot represents removal of connection)

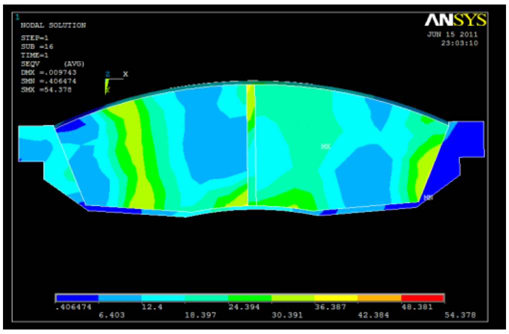

Figure 37. Modification showing reduced stresses at leading edge.

- Increasing the stiffness of caliper**:**The stiffness of caliper is increased to avoid bending. After number of iterations it has been found that caliper should have E=700Gpa. Following figure 38 shows the result for caliper with E=700GPa for both pads.

Due to increase in the stiffness of caliper it minimizes the loss of force transformation due to bending of caliper, because of this modification caliper transfers maximum forces and consequently cause to uniformly pressing of pad_2 against disc.

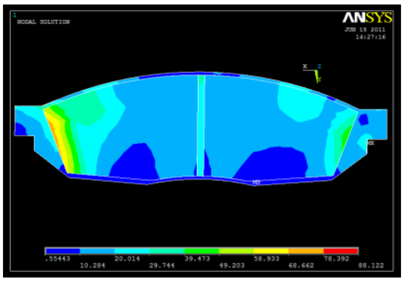

Figure 38. Stress distribution for Pad 2

For pad 2 this modification not only changes the stress pattern but also reduces the value of stress from 114.201MPa to 88.122MPa and variation of stresses on face is 10.284 MPa to 78.392 MPa.

- Partial connection between finger and pad: Finger side pad showing more stress at upper edge and middle portion is unaffected. For reducing stresses at upper side of pad_2 a partial contact has been established between caliper and pad. Figure 39 shows, nodes toeliminate the contact.

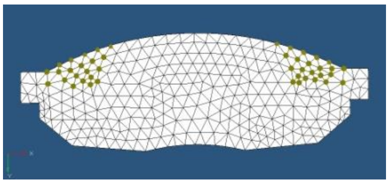

Figure 39. Partial connections for finger pad (the dot represents removal of connection)

Result: Due to the elimination of contact between pad 2 and fingers as shown in figure 37,this modification eliminate the support which uses the caliper for bending and helps the pad to press against disc at its middle portion only, which helps to have better stress distribution.

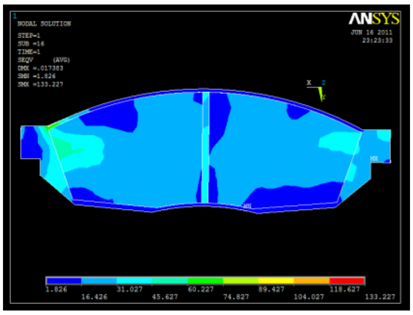

Figure 40. Stress distribution for Pad 2

This modification helps to reduce the stresses at upper side of pad and improve the stress pattern. Variation of stresses on face is 16.426 MPa to 89.427 MPa.

C. ***Results for Partial connection for piston and increase the stiffeness of the caliper:***

Stress distribution for Pad 1:

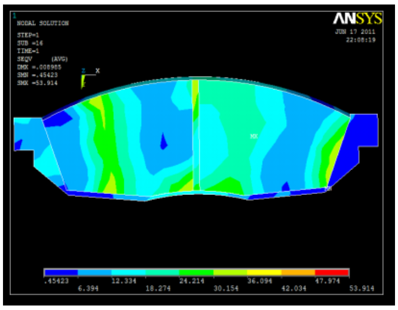

Figure 41. Stress distributions for Pad 1

Stress distribution for Pad 2:

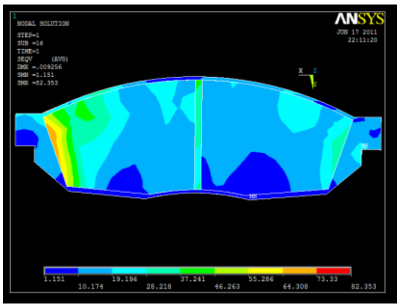

Figure 42. Stress distributions for Pad 2

D. ***Results for Partial connection for piston and Partial connection for finger pad:***

Stress distribution for Pad 1:

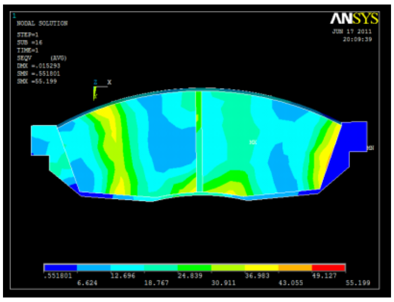

Figure 43. Stress distributions for Pad 1

Stress distribution for Pad 2:

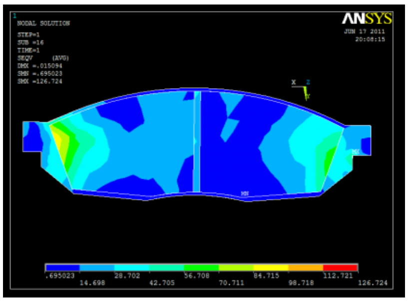

Figure 44. Stress distributions for Pad 2

E. ***Discussion over these modifications***:

Increase in stiffness of caliper gives good results but from in the manufacturing point of view it is very difficult to machine or manufacture the caliper as the material is too stiff i.e., E=700GPa. The significant advantage of modification for partial contact between piston head and pad_1, alsocaliper fingers and pad_2 over stiffer caliper is that it only requires inserting another component (the adapter) between the piston and the pad back plate and hence, this does not affect modal behavior of other individual disc brake components. However, the disadvantages of this modification are that it needs more assembly processes to fit the adapters and should need extra costs to design and fabricate those adapters.

VIII. CONCLUSIONS

This paper studies the contact stress distribution of a disc brake.Before modifications for contact stress distribution, an investigation has been done on five different disc and pad material sets for stress distribution and chosen the set which gives minimum stress value. The modification of the partial connections between the piston head and the pad back plate, gives reduced stress at leading edge of piston pad. The modification of stiffening the caliper gives better stress distribution by minimizing the loss of force transformation due to bending of caliper. Last modification of the partial connections between the fingers and the pad back plate eliminate the support which uses the caliper for bending and helps the pad to press against disc at its middle portion only, which helps to have better stress distribution. The advantages and disadvantages of both modifications in terms of manufacturing issues are also discussed briefly. This work could help to obtain a more uniform stress distribution and subsequently satisfy customer's needs by making pad life longer.

Conflict of interest: The authors declare that they have no conflict of interest.

Ethical statement: The authors declare that they have followed ethical responsibilities

REFERENCES

[1] Limpert, R. 1999. Brake Design and Safety. 2nd Edition. Warrendale, PA: Society of Automotive Engineers.
[2] Tirovic, M., and A. J. Day. 1991. Disc Brake Interface Pressure Distributions. Proc. IMechE Part D. 205:137-146.
[3] Ripin, Z. B. M. 1995. Analysis of Disc Brake Squeal Using the Finite Element Method. PhD Thesis. TheUniversity of Leeds.
[4] Lee, Y. S., P. C. Brooks, D. C. Barton, and D. A. Crolla. 1998. A Study of Disc Brake Squeal Propensity Using Parametric Finite Element Model. In IMechE Conf. Trans. European Conf. on Noise and Vibration: 191-201.
[5] Tamari, J., K. Doi, and T. Tamasho. 2000. Prediction of Contact Pressure of Disc Brake Pad. SAE Review21: 133-141.
[6] Tumbrink, H. J. 1989. Measurement of Load Distribution on Disc Brake Pads and Optimization of Disc Brakes Using the Ball Pressure Method. SAE Technical paper 890863.
[7] Fieldhouse, J. 2000. A Study of the Interface Pressure Distribution Between Pad and Rotor, the Coefficient of Friction and Caliper Mounting Geometry with Regard to Brake Noise. Proc. of the International Conference on Brakes 2000 Automotive Braking. 3-18.
[8] Pardeep Kumar & S.R.Chauhan, (2016), “An Investigation on Cutting Forces and Surface Roughness during Hard Turning of AISI H13 Die Tool Steel with CBN Inserts using RSM”, “International Journal of Advanced Engineering Research and Applications” (IJA-ERA) -2377 Vol. –1, Issue –9, January–2016.
[9] Available at: Ansys 10.0 Release notes.
[10] Available at: Ansys 10.0 Workbench & Release notes.
[11] Available at: https:// corymmoore.wordpress.com
[12] Available at: http://auto.howstuffworks.com/auto-parts/brakes/brake-types/disc-brake1.html

