RANDOM VIBRATION ANALYSIS OF VENTED AND NON VENTED DISC BRAKE

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Abstract

The current study essentially deals with the modeling and analyzing the vented and Non vented disc brake by CATIA and ANSYS. Finite element models of the brake-disc are shaped with CATIA and simulated using ANSYS which is based on the finite element methods. The Discrete and Finite element Dynamic models are constructed and they are validated by modal tests for the frequency range of the interest. A simple test is constructed and these properties are found by using a single degree of freedom model. The impacts of vibration on the system performance are investigated under random vibration load conditions by utilizing the software developed process. It is concluded that the analytical model suggested can effectively be utilized in preliminary design stage of a simple disc brake behave rigidly in the frequency range of interest. The main aim of this paper is to resist the shock loading in four wheel environmental conditions for an different dynamic loadings and to minimize the Total deformation, Directional deformation, Equivalent stress and strain with best suited Stainless steel Material Analysis is done on both Vented and Non vented disc brake. The Geometry of the models is carried out within the CATIA V5 R20 Software and is designed in Mechanical Design. The analysis part is done by utilizing the ANSYS R14.5 Software.

Keywords*—vented, Non vented, Dynamic, Random, Vibration, Catia and Ansys.*

I. INTRODUCTION

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A vehicle requires a brake system to stop or adjust its speed with changing street and activity conditions. The fundamental rule used in braking systems is to convert the kinetic energy of a vehicle into some other form of energy. For example, in friction braking it is converted into heat, and in regenerative braking it is changed over into electricity or compressed air etc. During a braking operation not all the kinetic energy is converted into the desired form, e.g. in friction braking some energy may be disseminated within the shape of vibrations.

So to get an optimum performance in demanding applications, ventilation is presented within the brake discs which increases the cooling rate. Brake discs may be divided in two categories:

1. Ventilated disk brake
2. Non Ventilated or Solid disk brake

II. PRINCIPLE OF BRAKING SYSTEM

A brake is a device by means of which artificial frictional resistance is connected to moving machine part, in order to stop the motion of a machine. Break play major part in moving auto motive vehicles.

A disk brake consists of a cast iron disc bolted to the wheel hub and a stationary hosing called caliper the caliper is associated to some stationary part of the vehicle like the axle casing or stub axle as in two parts each part containing a piston. In between each piston and the disc there is a friction pad held in position by retaining pins, spring plates etc. Sections are penetrated within the caliper for the fluid enter or leave each housing the passages area lso connected to another one for bleeding .Each cylinder and piston.

III. DESIGN OF DISC BRAKE

Designing of vented and non vented disc brake rotor utilizing the ISO standard measurements of BMW car in CATIAV5.

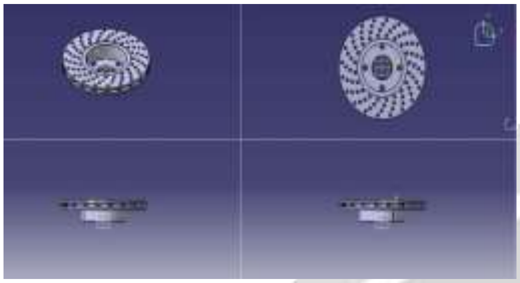

Fig 1: Three Dimensional Vented Disc brake

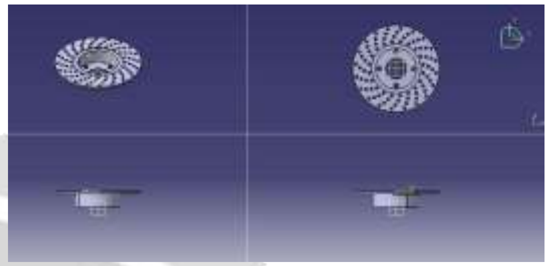

FIG 2: Three Dimensional Non Vented Disc brake

IV. MATERIAL SELECTION

It plays an important role in product design and manufacturing. This can be guided by the way the material behaves in the real time conditions according to its use. The material response to the environment stimulus is called property. Based on the application and utilization of material, its properties become significant.

Properties	Stainless Steel
Density	7.75e-006 kg mm^-3
Young's Modulus	1.93e+005 MPa
Poisson's Ratio	0.31
Coefficient of Thermal Expansion	1.7e-005 C^-1
Specific Heat	4.8e+005 mJ kg^-1 C^-1
Thermal Conductivity	1.51e-002 W mm^-1 C^-1
Resistivity	7.7e-004 ohm mm

V. RANDOM VIBRATION ANALYSIS IN WORKBENCH

Perform modal analysis to find natural frequency of the system. Perform a random vibration analysis using the modal analysis as the beginning condition environment, with the PSD Base Excitation connected within the desired direction. Assess desired stresses and deformations at 1 sigma values.

Disc Brake system was tested.

- Three geometries were utilized for the initial testing.
- Advance testing was performed on eight geometries with a varying output bar width.
- One conclusion of the Disc brake has a fixed boundary condition placed on four faces of circles. (Zero Displacement in X, Y, and Z).
- The Stainless Steel material was utilized for Disc brakes.
- Q was held constant at 5 by changing the steady damping ratio for the random vibration analysis to 1.e-002.
- The PSD base excitation was applied in the +Z direction for the random vibration analysis.

DETAILS OF TESTING

- A modal analysis was run to find the natural frequencies of the model
- These natural frequencies were utilized to perform the random vibration analysis in Workbench
  – Random Vibration analysis was performed utilizing the same PSD level chart as within the previous experiments.
- The natural frequency of the bracket was utilized within the Miles Equation analysis
  – The same PSD levels were utilized for the Miles Equation as the previous experiments.

– Miles Equation analysis included a static structural analysis with an acceleration applied to the device.

VI. MODAL ANALYSIS

• The modal analysis indicated the natural frequency of the bracket is 1082.4 Hz.
• 10 Faces on the loading parts were fixed (Zero displacement in X, Y, and Z).
• The modal solution was limited to frequencies between 10 Hz and 1100 Hz in order to reduce the time for solving the analysis

The source frequency range lies between 0 to 2000 Hz. Random levels obtain W0 is 0.0020 g2 /Hz and W1 is 0.020 g2 /Hz for materiel location drive system elements.

The Loss factor is found then the Structural damping and it can be calculated by utilizing frequency response function of material. In frequency response work, structural damping can be obtained by using half power points ratio. Moreover it is known that, structural damping is twice of the modal damping ratio. In random vibration analysis viscous modal damping ratio is used.

$$\eta = \frac{\omega_2 - \omega_1}{\omega_n}$$

$$\zeta = \frac{\omega_2 - \omega_1}{2\omega_n}$$

VII. RESULTS

COMPARISON OF RANDOM VIBRATION ANSYS ON VENTED & NON VENTED DISC BRAKE RESULTS

MATERIAL	MODEL	TOTAL DEFORMATION (mm)	DIRECTIONAL DEFORMATION (mm)	EQUIVALENT STRESS (MPa)
STAINLESS STEEL	VENTED	24.892	9115.7	7.717e6
	NON VENTED	52.918	22469	1.0261e7

Tab 1: Comparison results for random vibration ansys

Mode	Vented Frequency [Hz]	Vented Displacement [(mm²)/Hz]	Non Vented Frequency [Hz]	Non Vented Displacement [(mm²)/Hz]
1.	1087.5	619.61	906.1	2800.3
2.	1089.9	622.45	918.9	2887.9
3.	1231.4	363.93	937.48	2147.8
4.	1510	683.77	1050.9	2569.4
5.	1620.1	709	1082.4	2562.7

Tab 2: Comparison results of frequency range and displacement

1. RANDOM VIBRATION ANALYSIS OF VENTED DISC BRAKE STAINLESS STEEL

Mass= 9.0751 kg, Nodes= 34213, Elements= 18347.

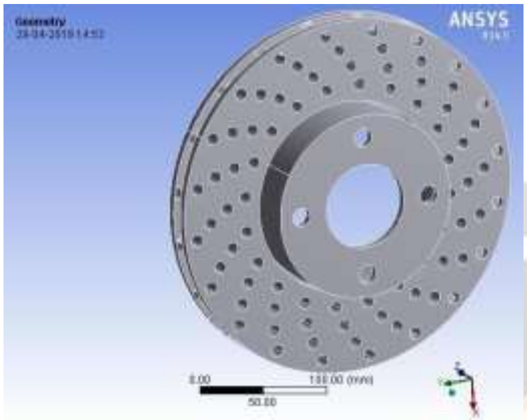

Fig 3: Geometry Diagram

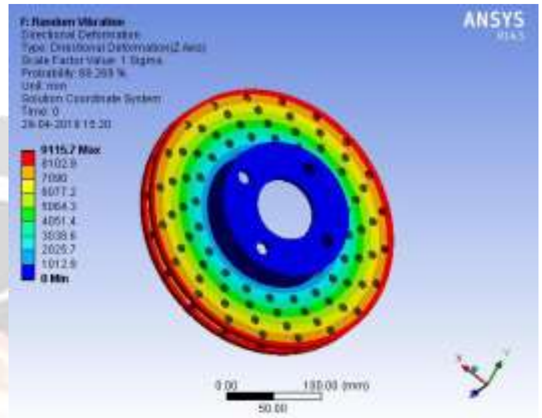

Fig 6: Equivalent stress

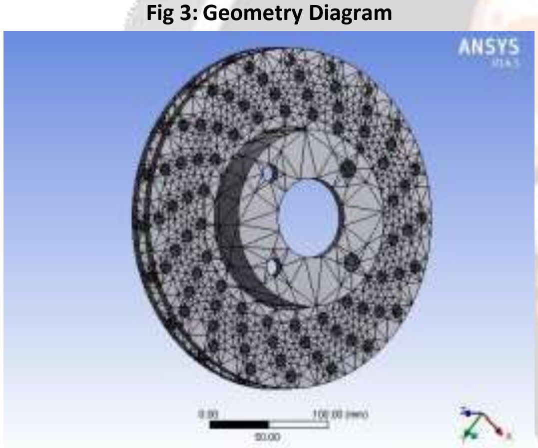

Fig 4: Mesh diagram

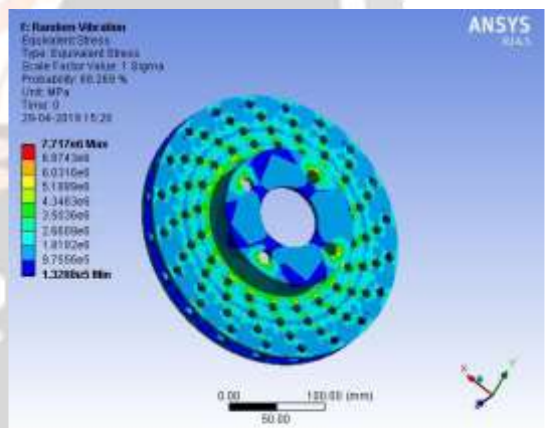

Fig 7: Equivalent Elastic strain

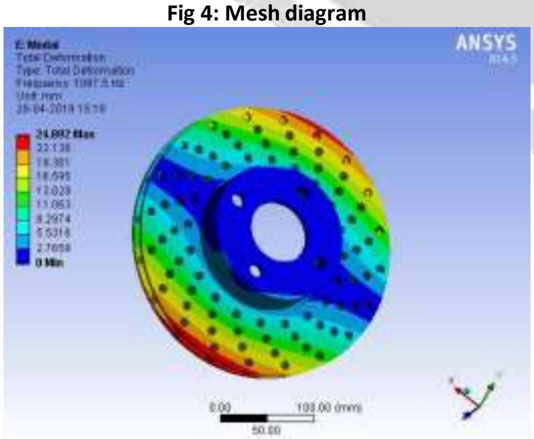

Fig 5: Total deformation

2. RANDOM VIBRATION ANALYSIS OF NON VENTED DISC BRAKE STAINLESS STEEL Mass= 3.9363kg, Nodes= 16537, Elements= 7967.

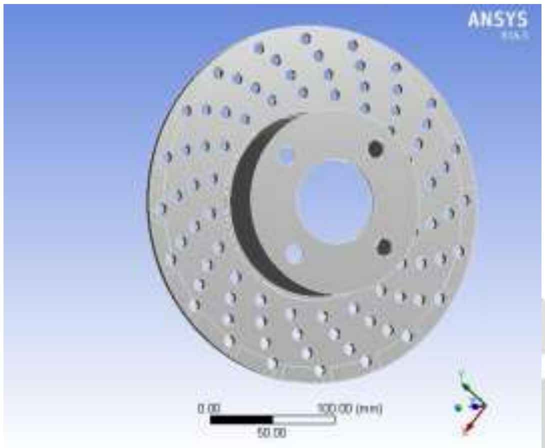

Fig 8: Geometry Diagram

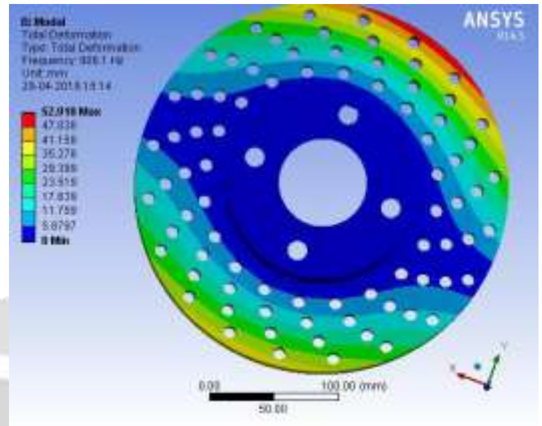

Fig 10: Total deformation

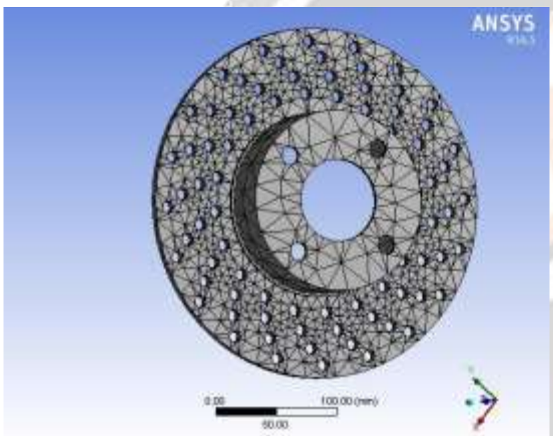

Fig 9: Mesh diagram

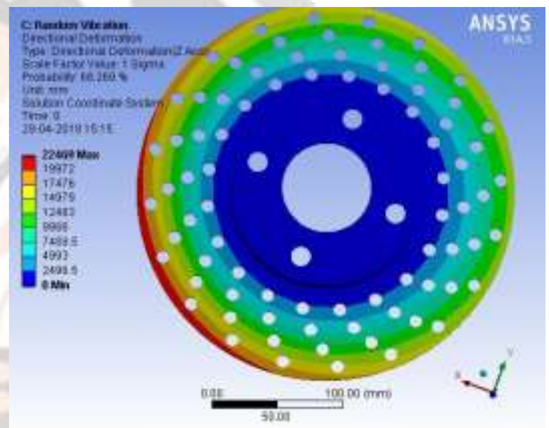

Fig 11: Equivalent stress

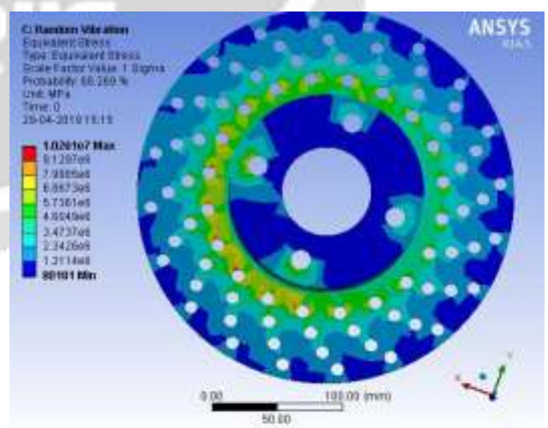

Fig 12: Equivalent Elastic strain

After completing the modal analysis, random vibration analysis is carried out. In random vibration analysis, directional accelerations can be obtained by utilizing known random input. Frequency range of interest is 10-1100 Hz and random vibration level W_0 is 0.0020 g2 /Hz and W_1 is 0.020 g2 /Hz.

This random vibration input is applied to the system in each axis Z, than output (directional acceleration) is obtained for Z axis. Moreover, damping value should be defined in ANSYS Workbench. The loss factor η = 0.0119 for Non vented disc brake and η =0.0015 for Vented disc brake. It is explained that Damping ratio ζ is half of the loss factor η. Therefore, ζ will be 0.00595 for Non Vented & 0.00075 for Vented disc brake.

VIII. CONCLUSION

This project has analyzed random vibration of the Disc brake by utilizing the finite element software ANSYS, obtained the random response of the structure and the position of equivalent stress. Based on the results of random vibration analysis and gives a reference for structure application in engineering practice.

It is observed that the Vented type disc brakes can provide superior Random Vibration analysis than the Non Vented disc Brake. It provides a useful design tools and improvement of the brake performance in disk brake system. We can say that from all the values obtained from the analysis i.e. the Total Deformation, Equivalent Stress, Equivalent strain, Frequency, Damping ratio, Loss factor and Directional deformation exhibit that the vented disc is best suited design. It is concluded that disk brake with vents and of material Stainless steel is observed best possible combination for present application.

The finite element results show that natural frequencies of the disc brake used in this study are out of the frequency range of interest, however, in general, natural frequencies of the disc brake may be in the frequency range of interest because of geometry and material properties, Moreover, the frequency range of interest may be high enough to cover the natural frequencies of the disc brake (For instance, frequency range may be 10- 1100 Hz). Since the analytical model developed is valid only for Disc brake. Only random vibration analysis input is considered in getting dynamic output. Disc brake is exposed to shock loadings in four wheeler environmental conditions. Therefore, analytical model may be improved by considering different dynamic loadings.

IX. REFERENCES

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