Grenze International Journal of Engineering and Technology, Special Issue

Design and NVH Analysis of Disc Brake System

Harikumar Andem[1], Sri Sai Bhargav Pulugurtha[2] and Ramesh Siripuram[3]
[1]Osmania University/Mechanical Engineering Department, Hyderabad, India
Email: ahk.me@kitsw.org
[2-3]Kakatiya Institute of Technology and Science/Mechanical Engineering Department, Warangal, India
Email: {pssbhargav, sramesh.kits}@gmail.com

***Abstract*—Noise, vibration and Harshness (NVH) also known as noise and vibration is the study and modification of the noise and vibration characteristics of automobile vehicles particularly cars and trucks. Automobile noise is a problem which is related to the characteristics of geometric design. Squeals and other high frequency noises in various names have been a major concern in the design of the automobiles. In this Analysis, the Disc Brake Rotor is modeled using Catia and Modal analysis of Disc Brake rotor with some rotational velocity and Harmonic response of the Disc Brake rotor with some force acting and finally thermal analysis of the disc brake rotor with some temperature applied by using the silicon carbide as the material.**

***Index Terms*— Squeal, Disc Brake Rotor, NVH Analysis, Modal Analysis, Harmonic Response.**

I. INTRODUCTION

Noise, vibration, and harshness (NVH) in structures and have received increased attention in recent years. The emphasis on improving customer satisfaction and brand image has affected such industries as automotive and aviation. Customer awareness and sensitivity to noise and vibration have been raised through increasing television advertisement, in which the vehicle noise and vibration performance is used as the main market differentiation.

Technologies that have become an essential part of the design process are Computer Aided Design (CAD) and Finite Element Analysis (FEA). Using these tools, one can make design changes and evaluate their effects with considerably less effort. One potential problem with FEA models is the need for limitless resources as the structure complexities increase. For instance, for vehicles, the sheer number of elements required to model the structure can become so large that a full-scale model cannot be run without significant computational capability. Alternatively, sub-assemblies are often modeled and individually run to estimate the behaviour of the complete structure. Correctly estimating the boundary conditions between different elements, however, can be difficult to model and also can result in modeling errors. Therefore, alternative modeling methods that can provide relatively accurate predictions without the need for large finite element models are desired. This study investigates a method that combines experimental and analytical modeling for predicting structural vibrations. The method is applied to a complex structure such as a locomotive cab, and the accuracy of the results is evaluated.

Grenze ID: 01.GIJET.6.2.3_1

II. CHARACTERISTICS OF NOISE, VIBRATION, HARSHNESS

A. Noise

It is well known that sound is a result of mechanical vibrations which act in the elastic medium. Sound source produces a certain amount of sound energy per time, which means that the sound source is determined by sound power, which does not depend on the characteristic of the environment. On the other hand, sound intensity which is measured on a certain spot depends not only from the power of sound source and distance from it, but also from the amount of energy absorbed by the environment. This explains the significance of sound insulation in qualitative assessment of cabin noise, especially to aerodynamic, road and powertrain noise. Related shows the percentage contribution of different noise sources, originating from powertrain, into total noise According their technicians for the purposes of diagnostics the most common NVH problems, different types of noise which can be felt in the cabin include:

- Droning
- Beat
- Road noise
- Brake squeal

B. Vibration

Vibrations are result of acting of compelling force upon an object. Because of a complex structure and many rotating components, there are many sources and types of vibrations on a vehicle. Because of the "telegraphing effect", transmission of vibration to other components, determination of the main source of vibrations is sometimes very difficult. Inside the cabin, vibrations can be classified as:

- Shake
- Shimmy
- Shudder

C. Harshness

The causes of harshness may be deterioration of vehicle components, damage, or modification of the original equipment. In most cases, harshness is related to chassis components. The impact force from the road surface causes the tires to vibrate. The tires transmit the vibration through the suspension system to the car body. Harshness has frequency of 30 to 60 Hz.

III. BRAKING SYSTEM

A brake is a mechanical device by means of which frictional resistance is applied to moving vehicle element, in order to stop the motion of a vehicle. The main idea behind the process of performing this function, the brakes absorb kinetic energy of the moving element. The energy absorbed by brakes is dissipated in the form of heat. This heat is dissipated in the surrounding atmosphere to stop the vehicle. During operation of brake system, a pressure is applied towards the disc rotor with the caliper and hydraulic assembly.

A. Factors Governing Braking

Four basic factors determine the brake power of the system. The first three factors govern the generation of friction: pressure, coefficient of friction, and frictional contact surface. The fourth factor is a result of friction. It is heat or, more precisely, heat dissipation.

Pressure:
The amount of friction generated between moving surfaces in contact with each other depends in part on the pressure exerted on the surfaces. In a brake system, pressure applied with the help of hydraulic system.

Coefficient of Friction:
The amount of friction generated between the two surfaces is expressed as coefficient of friction. The required COF depend on the vehicle and the other factors are carefully chosen by manufacturers to ensure safe and reliable braking.

Frictional Contact Surface:
The third factor is amount of surface area that is in contact. For most part, the vehicle weight ad potential speed determines the size of frictional surface area.

B. Disc Brake

A disc brake consists of a disc component 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 as is cast 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. passages are drilled in the caliper for hydraulic fluid to enter or leave each housing. The passages are also connected to another one for bleeding. Each cylinder contains rubber-sealing ring between the cylinder and piston.

C. Parts of A Disc Brake

- Disc Calipers
- Brake Pads
- Disc Brake Rotor
- Master Cylinder

D. Working Principle

The working principle applied on disc brake is that the applied force (pressure) acts on the brake pads, which comes into contact with the rotating disc. At this point of time due to friction the relative motion is constrained. When the brakes are applied, hydraulically actuated pistons move the friction pads in to contact with the disc, applying equal and opposite forces on the later. On releasing the brakes, the rubber-sealing ring acts as return spring and retracts the pistons and the friction pads away from the disc, as shown in the fig. 1.

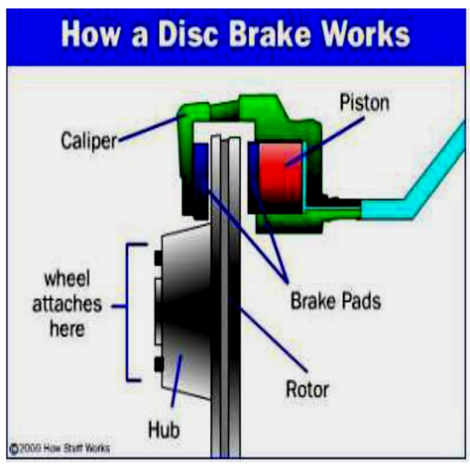

Figure 1. Working Principle of Disc Brake

E. Classification of Disc Brake Squeal

Squeal has certain frequency. Brake squeal is also indicated by amplitudes of the vibrations of the brake disc in the micrometer range and the created sound pressure level (SPL). Depending on the various frequencies disc brake noise is classified in following categories. Brake squeal is a high-pitched noise in the frequency range between 1 kHz and 16 kHz originating from self-excited vibrations which are caused by the frictional contact between brake pads and brake disc.

- Low-frequency noise whose range is 100Hz to 1,000 Hz
- Low-frequency squeals whose range is 1,000 Hz to 7,000 Hz
- High-frequency squeals whose range is 8,000 Hz to 16,000 Hz

IV. Pre-stressed Loading Condition

This is the special condition of analysis that in the Ansys workbench that the results of one analysis is taken as the analysis settings of another analysis this condition is known as pre-stressed loading condition. For example, if we want to find the deformation of any object due to thermal conditions then first we will conduct the Thermal analysis and this result are taken setup for steady state so that we can find the deformation due to thermal this condition is known as the pre-stressed loading condition.

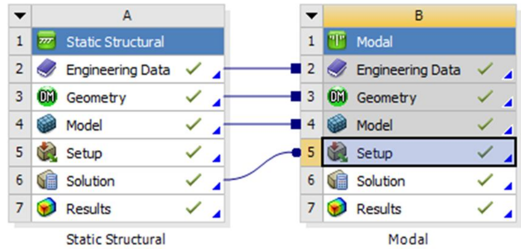

Figure 2. Pre-Stressed Loading Condition

V. TYPES OF ANALYSIS FOR DISC BRAKE ROTOR

The Disc brake rotor is designed in the Catia as per the design considerations and the different types of analysis conducted to the disc brake rotor in the Ansys workbench is as follows.

- Modal Analysis of Disc Brake Rotor with the given conditions
- Modal analysis of Disc Brake Rotor with some rotational velocity
- Harmonic response at the particular frequency
- Steady state thermal analysis of Disc brake Rotor

This analysis is done by taking the different material and by the results we can know that how we can reduce the brake squeal. The following analysis is done by taking Silicon carbide as materials.

A. Modal Analysis of Disc Brake Rotor

The modal analysis deals with the dynamics behaviour of mechanical structures under the dynamics excitation. The modal analysis is used to determine the dynamic characteristics of a system such as natural frequency, mode shapes etc.

Applying Analysis Conditions:

In the modal analysis we can get the six mode shape results and frequency of the disc brake rotor by taking these six mode frequencies we can get the mode frequency deformation as follows Modal analysis of the disc was conducted by using QR damping method in order to find the disc free-free mode shapes by considering the Frictionless support for this analysis as settings.

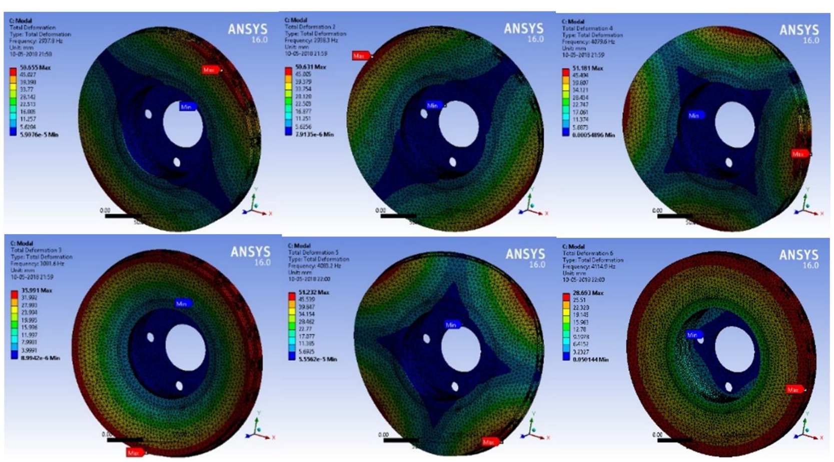

Figure 3. Shows the Mode Frequency of the Disc Brake Rotor

B. Modal Analysis of Disc Brake Rotor with Rotational Velocity

In this analysis we can find the frequency of the Disc Brake Rotor when it is moving at the particular Rotational Velocity we can the deformation of the mode shape results of the disc brake rotor at that particular frequency.

Applying Analysis Conditions:
In this analysis there are two parts i.e. first we have to give the rotational velocity to the disc brake rotor in the static structural analysis and the results of these analysis is consider as the setup to the modal analysis for the rotational velocity. we cannot give the rotational velocity directly in the modal analysis settings so first we have given the rotational velocity in the static structural analysis and these results are taken consider as the setup in modal analysis for the frequency of disc brake rotor moving with some rotational velocity. and this conditional analysis is called as the pre-stressed condition.
In this analysis first, we have to consider the geometry as the disc brake rotor made of silicon carbide as the material and make the fine meshing for the accurate results as mentioned above and in the analysis settings we take rotational velocity as 7.46 rpm according to the design considerations and we also take the frictionless support to the joints corners of the disc brake rotor as shown in the fig.4 below.

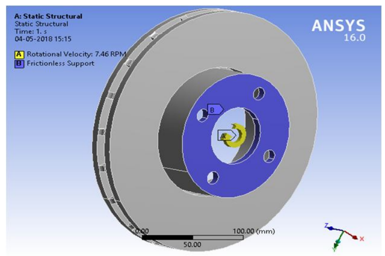

Figure 4. Rotational Velocity of Rotor

Results
In this modal analysis, we can get the mode frequencies and deformation of the mode shape at every particular frequency as shown in fig.5.

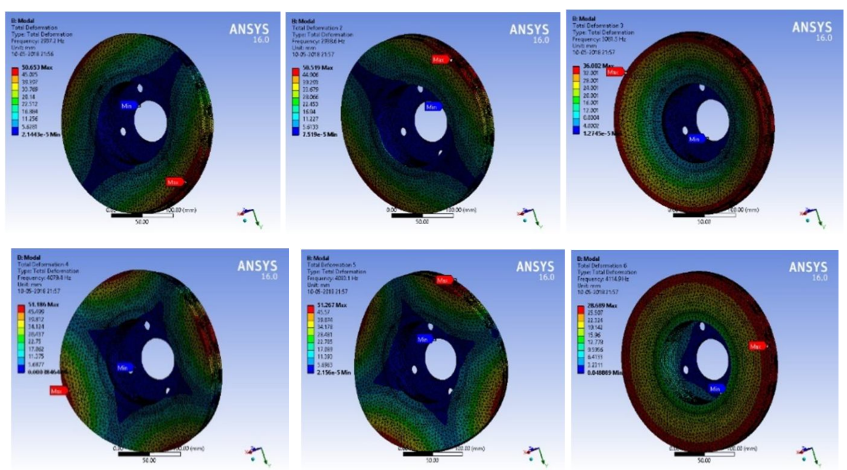

Figure 5. Shows the Mode Shape Results

C. Harmonic Response Analysis

Harmonic response is the response of the system under a sustained cyclic load. In this analysis we can find the deformation of the disc brake rotor at the highest range of frequency. In this analysis, we can find the deformation of the disc brake rotor at high range of frequency when the force is applied by the caliper.
As per the design considerations of the disc brake rotor, considering the frictionless support, the force applied by the caliper is taken as 28493N, which is acting on the disc brake rotor; and considering the frequency value as 1390 Hz, we can find the stress and deformation on the rotor at the high range of frequency.

Result
In this analysis results we can get the results of deformation of the disc brake rotor at the high range of frequency of 1390HZ as per the material and design considerations when the force is applied by the caliper.

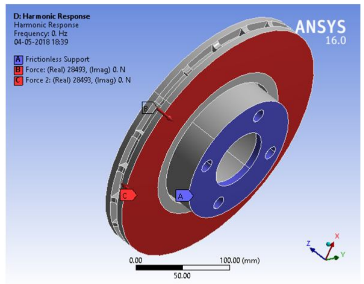

Figure 6. Shows the Setup for the Analysis

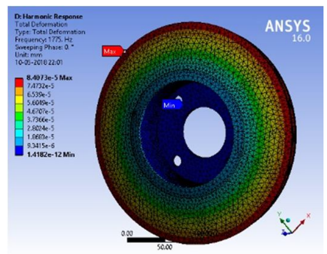

Figure 7. Deformation

In the figure shows the deformation of the disc rotor when it is made of silicon carbide the maximum and minimum deformation in the fig that the force is acting high at the max and low at the min so that the deformation will be spread and starts at the high max point and slowly propagate to the min position as we consider that the when disc rotor is made of grey cast iron the maximum deformation value is about 0.0001602 mm i.e. the minimum deformation is occur when the disc is made of silicon carbide.

D. Thermal Analysis

In this analysis we can get the rate of heat transfer for the material that the heat flux value can be find out in this analysis. As per design considerations of the disc brake rotor the Heat generated in the rotor is about 600 degrees centigrade (600` c). In this analysis we give the initial and the final temperature of the rotor that is generated and then the Initial temperature as the room temperature and the final temperature as the 600`c. As per the results the heat is distributed in the disc brake rotor and the heat flux is given below in this analysis we can get the rate of heat transfer for the material that the heat flux value can be find out in this analysis. As per the results the heat is distributed in the disc brake rotor and the heat flux is given below.

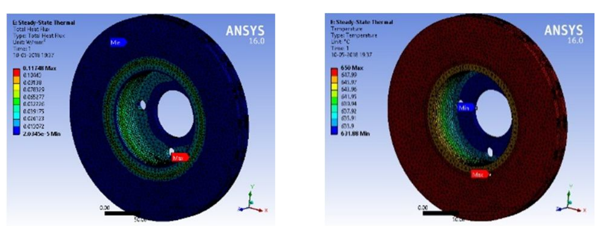

Figure 8. Shows the Temperature Distribution and Heat Flux

From the results obtained, the deformation due to the thermal analysis by the above condition will be zero because the maximum temperature sustained by the silicon carbide disc brake rotor is about 1600^0 c.

E. Advantages of Silicon Carbide

The major advantages of AMCs compared too unreinforced materials are as follows,

- Greater strength
- Improved stiffness
- Reduced density (weight)
- Improved high temperature properties

- Controlled thermal expansion coefficient
- Thermal/ heat management
- Enhanced and tailored electrical performance
- Improved abrasion and wear resistance
- Control of mass (especially in reciprocating applications)
- Improved damping capabilities.

VI. CONCLUSIONS

From the results obtained above, we can conclude the following:

- Practical use of Silicon Carbide composite material produces much effective braking compared to steel disc brakes.
- Deformation in steel is much higher than composite, which implies the deformation resistance of the composite structure than the steel material.
- Stress accumulated on the composite is much less, which proves the wear resistance, rigid and stable braking during high speeds.
- The maximum operating temperature of silicon carbide Disc Brake Rotor is about 1650^0c, the maximum operating temperature in the sports cars is about 600^oc – 700^oc. Hence silicon carbide maximum heat bearing capacity.
- The corrosion and material wear rate due to the heat generation and deformation will be very low compared to the other materials, so that the noise causes during the braking operation will be also low in silicon carbide disc brake rotor.
- But due to the economic considerations the maximum operating temperature in the sports cars is about 600^oc but for the normal cars it is about 300^oc so that we can use the aluminum alloy in case where the operating temperature rate is comparatively low.

REFERENCES

[1] Ammar A. Yousif Mohammed, Inzarulfaisham Abd Rahim (2013), Disc-Plate Squeal Investigation Using Finite Element Software: Study and Compare, *International Journal of Scientific & Technology Research,* Volume 2, Issue 1, ISSN 2277-8616, pp.143-154.

[2] Matsuzaki M. and Izumihara T. (1993), Brake noise by longitudinal vibration of the disc rotor, 930804, *SAE.*

[3] McDaniel et al, (1999) Acoustic radiation models of brake systems from stationary LDV measurement, *Proceedings of IMEC 99, International Mechanical Engineering Congress and Exposition*.

[4] Lee AC, (1992), Study of disc brake noise using multi-body mechanism with friction interface, *In: Ibrahim RA, Soom A, editors, Friction-induced vibration, chatter, squeal, and chaos, New York. ASME, DE*-Vol. 49: pp. 99- 105.

[5] R. Nicole, "Title of paper with only first word capitalized", *J. Name Stand. Abbrev.* in press.

[6] Nishiwaki et al, (1989), Study on disc brake squeal, *SAE, Technical Report 890864*.

[7] McDaniel, J.G. et al, (1999), Acoustic radiation models of brake systems from stationary LDV measurements, *In: Proceedings of IMEC' 99, American Society of Mechanical Engineers.*

[8] Fieldhous, J.D. and Newcom, T.P., (1993), The Application of Holographic Interferometry to the Study of Disc Brake Noise, *SAE, Technical Report 930805, Warrendale*.

[9] Sherif, H.A., (1993), Geometric induced instability in Drum Brakes, *SAE, Military Technical College*, 933072.

[10] Okamura, H. and Nishiwaki, M., (1988), Study on brake noise, (first report, on drum brake squeal*). Trans. Japan Soc. Mech. Eng., Series C, Vol. 54, No. 497*, pp. 166-174.

[11] Kung, S. et al, (2000), Complex Eigen value Analysis for Reducing Low Frequency Brake Squeal, *SAE, World Congress Detroit*, *Michigan.*

[12] Feildhouse, J.D. and Newcomb, T.P., (1991), An investigation into Disc brake squeal using Holographic interfermotery, *3rd int 1 EAEC conference on Vehicle Dynamics and power train engineering-EAEC, paper No. 91084.*