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Suppression of squeal frequency in disc brake assembly

#1S.B. Indurkar ,#2Prof.S.V. Bhaskar
1sachinindurkar28@gmail.com
2santoshbhaskar12002@yahoo.co.in

#12M.E. Mechanical Design Engineering, Savitribai Phule Pune University
S.R.E.S. College of Engineering Kopargaon, Ahamednagar

ABSTRACT

Noise and vibration is an increasingly important consideration in the design and study of disc brakes. Certain vibrations may only result in minor annoying squeals, while others may be severe enough to result in structural damage or failure. In either case, it is desirable to predict the conditions under which these vibrations arise, so that they may be controlled, or eliminated. Brake squeal, which usually occurs in the frequency range between 1 and 16 kHz, has been one of the most difficult concerns associated with vehicle brake systems. It causes customer dissatisfaction and increases warranty costs. Although substantial research has been conducted into predicting and eliminating brake squeal, it is still difficult to predict its occurrence due to the complexity of the mechanisms that cause brake squeal. 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 along with 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. In our dissertation work we have analyzed a simplified Disc Brake Assembly for squeal with the help of advance FEA software package. CAD data is correlated with current physical model. For Correlation FFT analyzer is used to extract natural frequency of real brake disc. We used Lancoz's method to extract the natural frequencies by FEA. Complex eigen value extraction is used to identify the Squeal frequencies. Finally squeal frequency will be suppressed with optimized rotor design

Keywords— **Brake disc, Brake squeal, Complex Eigen Value, FFT analyzer, Natural frequency**

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I. INTRODUCTION

Brake squeal, which usually occurs in the frequency range between 1 and 16 kHz, has been one of the most difficult concerns associated with vehicle brake systems. It causes customer dissatisfaction and increases warranty costs. Although substantial research has been conducted into predicting and eliminating brake squeal, it is still difficult to predict its occurrence due to the complexity of the mechanisms that cause brake squeal. The complicated geometry of disc brake and the complex force applied by callipers make their analysis difficult. But optimized meshing and accurate simulation of boundary conditions along with 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.

A car disc brake system consists of a rotating disc and stationary (non- rotating) pads, carrier bracket, calliper and mounting pins. The pads are loosely housed in the calliper and located by the carrier bracket. The calliper itself is allowed to slide fairly freely along the two mounting pins in a floating caliper design. The Disc is mounted to a car wheel and thus rotates at the same speed as the wheel. When the disc brake is applied the two pads are brought into contact

with the disc surfaces. Most of the kinetic energy of the travelling car is converted to heat through friction. But a small part of it converts into sound energy and generates noise. A squealing
brake is difficult and expensive to cure. Preferably the noise issue should be resolved at the design stage.

There are several categories of brake noise that are classified according to the frequency of noise occurrence.
Basically, there are three general categories of brake noise: low frequency noise, low frequency squeal and high frequency squeal. Low frequency disc brake noise is a problem that typically occurs in the frequency range between 100 and 1000 Hz. Typical examples of noise problems from this category are groan and moan noise. The generation mechanism of this kind of problem is the friction excitation at the rotor and lining material, which provides energy to the system. This energy is transmitted as a vibratory response through the brake assembly and couples with components of the suspension and chassis.

Although the low frequency noise is an important problem for certain types of brake systems, the most common and annoying problem is squeal noise. Squeal is defined as a noise whose frequency content is 1000 Hz or higher that occurs when a system experiences very high amplitude mechanical vibrations. There are two theories that try to explain how this phenomenon occurs. The first one is called "stick-slip". According to this theory, squeal is a self-excited vibration of the brake system caused as a result of two factors: the static friction coefficient is greater than the sliding friction coefficient; the relationship between sliding friction coefficient f and relative sliding velocity Vr is $\delta f/\delta Vr < 0$. However, this theory cannot explain why the tendency of squeal is different when the same friction couple pair (rotor and pads) is used in different brake systems. Therefore, a second theory, called "sprag-slip", was developed. It demonstrates that the self-excited vibration of the brake system and the high levels of vibration result from an improper selection of geometric parameters of the brake system. In this case, two system modes that are geometrically matched move closer in frequency as the friction coefficient increases. These two modes eventually couple at the same frequency and matching mode shapes, becoming unstable (Dihua and Dongying, 1998). Both theories attribute the brake system vibration and consequent noise to variable friction forces at the pad-rotor interface. These variable friction forces introduce energy into the system. During the squeal event, the system is not able to dissipate part of this energy and the result is the high level in the amplitude of vibration.

II. METHODOLOGY

The analysis is to be carried out by using following approaches:

A. Modal Analysis

The model is created in Pro-E Wildfire and meshed using Hypermesh. The natural frequencies of model in free-free conditions are calculated using Abaqus 6.10.

B. Validation of Model

The model of Disc & Pad is created and analyzed for natural frequency using Abaqus and the results are compared with Experimental modal analysis using FFT for validation of the model. For this purpose the two parameters are checked weight comparison and natural frequency comparison.

C. Squeal Analysis (Complex Eigen Value Analysis)

A new functionality of ABAQUS/Standard, which allows for a nonlinear analysis prior to a complex eigenvalue extraction in order to study the stability of brake systems, is used to analyse disc brake squeal.

For brake squeal analysis, the most important source of nonlinearity is the frictional sliding contact between the disc and the pads. ABAQUS allows for a convenient, but general definition of contact interfaces by specifying the contact surface and the properties of the interfaces. ABAQUS version 6.10 has developed a new approach of complex eigenvalue analysis to simulate the disc brake squeal. Starting from preloading the brake, rotating the disc, and then extracting natural frequencies and complex eigenvalue, this new approach combines all steps in one seamless run . The complex eigen problem is solved using the subspace projection method, thus a natural frequency extraction must be performed first in order to determine the projection subspace.

The governing equation of the system is

$$Mx'' + Cx' + Kx = 0,$$

Where M is the mass matrix, C is the damping matrix, which includes friction-induced contributions, and K is the stiffness matrix, which is asymmetric due to friction. The governing equation can be rewritten as

$$(\mu 2\, M + \mu C + K)\, \phi = 0,$$

Where μ is the eigenvalue and ϕ is the corresponding eigenvector. Both eigenvalues and eigenvectors may be complex. In order to solve the complex eigen problem, this system is summarized by ignoring the damping matrix C and the asymmetric contributions to the stiffness matrix K. Then this symmetric eigenvalue problem is solved to find the projection subspace. The N eigenvectors obtained from the symmetric eigenvalue problem are expressed in a matrix as $[\phi 1, \phi 2 \ . \ . \ . \ ,\phi N]$. Next, the original matrices are projected onto the subspace of N eigenvectors.

$$M^* = [\phi 1,\phi 2\ldots\ldots,\phi N]T\ \ M[\phi 1,\phi 2,\ldots\ldots,\phi\ N],$$

$$C^* = [\phi 1,\phi 2\ldots\ldots,\phi N]T\ \ C[\phi 1,\phi 2\ldots\ldots,\phi N],$$

And

$$K^* = [\phi 1,\phi 2\ldots\ldots,\phi N]T\ \ M[\phi 1,\phi 2\ldots\ldots,\phi N],$$

Then the projected complex eigen problem becomes

$$(\mu 2\, M^* + \mu C^* + K^*)\, \phi^* = 0,$$

Finally, the complex eigenvectors of the original system can be obtained by

$$\phi k = [\phi 1,\phi 2\ldots\ldots., \phi N\ ]\phi^* k$$

A more detailed description of the algorithm may be found in . The complex eigenvalue

l, can be expressed as $\mu = \alpha \pm i\omega$ where α is the real part of μ, $Re(\mu)$, indicating

the stability of the system, and ω is the imaginary part of μ, $Im(\mu)$, indicating the mode

frequency. The generalized displacement of the disc system, x, can then be expressed as

$$x = A e^{\mu t} = e^{\alpha t} (A1 \cos\omega t + A2 \sin\omega t).$$

This analysis determines the stability of the system. When the system is unstable, a becomes positive and squeal noise occurs. An extra term, damping ratio, is defined as

$-\alpha/(\pi|\omega|)$. If the damping ratio is negative, the system becomes unstable, and vice versa. The main aim of this analysis is to reduce the damping ratio of the dominant unstable modes.

III. CAD MODEL GENERATION

To generate the 3D cad data we use reverse engineering technique. With the help of measuring instruments we took the dimensions to create the 3D cad model. Fig. 1 shows a 3-D plot of the model used.

Fig.1 Brake Disc Assembly

The brake model used in this study is a simplified model consisting of the two main components contributing to squeal: the disc and the pad (Fig.2)

A simplified model was used in this study for the following reasons:

1. For brake squeal analysis, the most important source of nonlinearity is the frictional sliding contact between the disc and the pads.
2. The simulation includes geometry simplifications to reduce CPU time, allowing far more configurations to be computed.

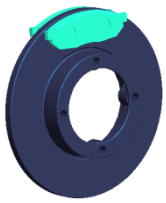


Fig.2 Simplified Disc & Pad Assembly

IV. MODAL ANALYSIS BY FEA AND FFT

A. FEA Mesh of Disc & Brae Pad

The FE mesh is generated using 19,000 solid elements. The friction contact interactions are defined between both sides of the disc and the friction material of the pads. A constant friction coefficient and a constant angular velocity of the disc are used for simulation purposes.

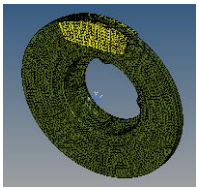


Fig.3 Meshed Disc & Pad Assembly

B. Boundary condition & Loading

Boundary conditions for our problem are:

For Disc: All DOF's for disc are locked except rotary motion about z -axis.

For Brake Pad: All DOF's for brake pad are locked except translatory motion of brake pad in z- axis.

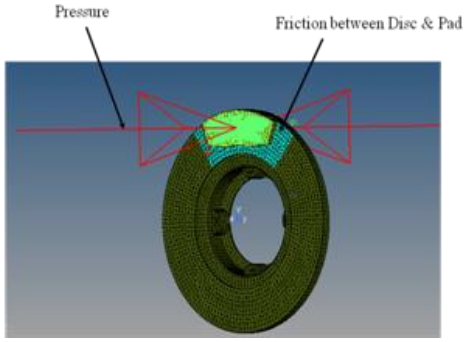


Fig.4 Loading and B.C.for Disc & Pad Assembly

For our analysis, we apply line pressure of 10 bar on both pads.

So, Pressure = 10 bar = 1 N/mm2

The pressure is applied as concentrated load on a single node by connecting all nodes on brake pad with a single node.

We know,

Pressure = Force/Area i.e. P= F/A---------- Eq.1

Area of brake pad = 550 mm2 (Calculated directly from 3D data)

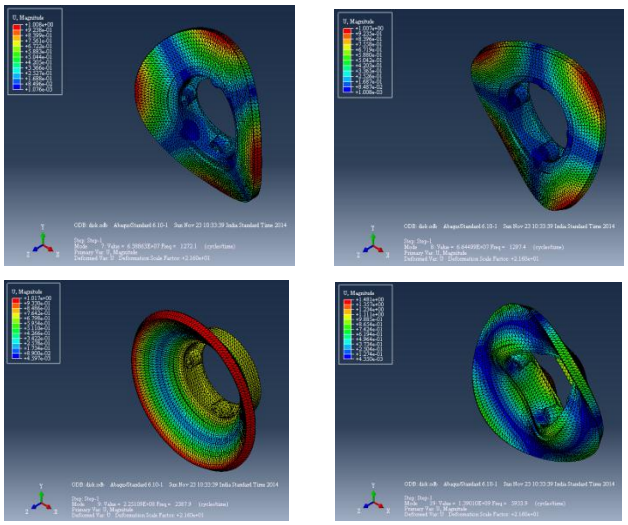
From 1 , 1 = F/ 550

F = 1 x 550 = 550N

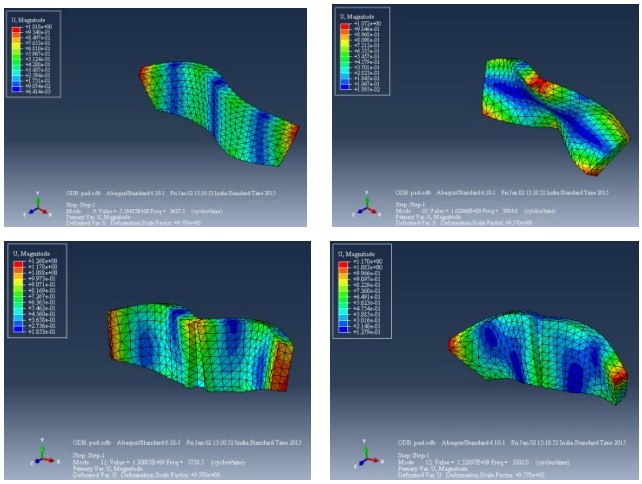
Concentrated force , F = 550N

We assume Rotor speed = v = 5 rad/sec

C. Mode Shapes for Disc at free-free condition (Rotor)



D. Mode shapes for brake pad free-free condition



E. Correlation Analysis

To validate numerical model, a correlation analysis between numerical and experimental data is conducted for healthy disc & brake pad. In general, correlation analysis is a technique to examine quantitatively and qualitatively the correspondence and difference between analytically and experimentally obtained modal parameters.

The frequencies measured on the disc and calculated by the simulated model for modes with free-free boundary conditions are shown in Table 5.2. It can be observed that the measured and simulated frequencies are in good agreement.

In a similar way, the parameters for the pads are estimated based on the measured data indicated in Table I. The measured and simulated frequencies are in good agreement.

TABLE I
MODAL RESULTS OF THE ROTOR AT FREE-FREE BOUNDARY CONDITIONS

Sr. no.	Experimental Frequency (Hz)	FEA Frequency (Hz)	Difference (%)
1	1224	1272.1	3.8
2	1248	1297.4	3.8
3	2440	2387.9	-2.2
4	6240	5933.9	-5.2

TABLE III
MODAL RESULTS OF THE PAD AT FREE-FREE BOUNDARY CONDITIONS

Sr. no.	Experimental Frequency (Hz)	FEA Frequency (Hz)	Difference (%)
1	3680	3627.3	-1.5
2	5802	5520	-5.1

F. Brake Squeal Analysis using abaqus

From our brake squeal analysis we found the squeal at frequency 7972.9 Hz i.e. 8kz, which is having positive real part value. We can define this low frequency squeal as it is below 10 kz.

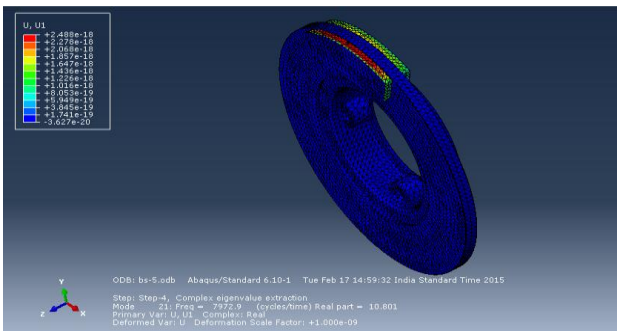


Fig.5 Squeal frequency of disc brake assembly

V. RESULT AND DISCUSSION

The effect of the stiffness of the disc on the disc brake squeal is studied by changing density of disc. In below Table evaluated density of system at stable.

TABLE IIIII
INSTABILITY FOR DIFFERENT DENSITIY VALUE

Sr. No	Density	Real part of complex Eigen Value	Result
1	7.1	+ve	Instability
2	7.12	+ve	Instability
3	7.15	+ve	Instability
4	7.2	-ve	Stability
5	7.25	+ve	Instability

This chapter is devoted to the study of brake squeal frequencies and instability of the system. Below image shows graph of frequency vs damping ratio. The values we got from brake squeal analysis done with the help of Abaqus.

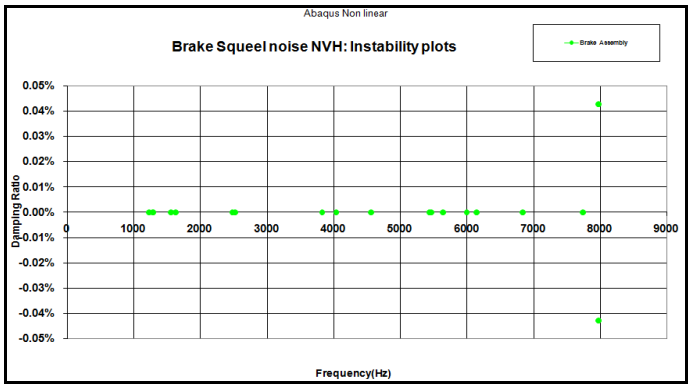


Fig.6 Brake Squeal Noise: Instability plots

We found only four squeal frequency in our analysis, which may produce instability in the system. From our comparison between results of modal analysis using FFT (experimental analysis) and Abaqus, we confirm the density of rotor as (7.1, 7.12, 7.15, 7.25) kg/mm3. The density value shows the grey cast iron which is used to manufacture brake rotor is SAE G1800 grade. As we know change in density affects the stiffness. So we tried with higher grade i.e. G3000 grade grey cast iron which have the density of 7.2 kg /mm3and we found successful removal of squeal frequency. We repeated the complex eigen value extraction analysis with changed density for rotor. Below image shows the same mode shape but with the real part value as zero. So we

can assure that there is no instability present in the brake system by using higher grade (G3000) material for rotor.

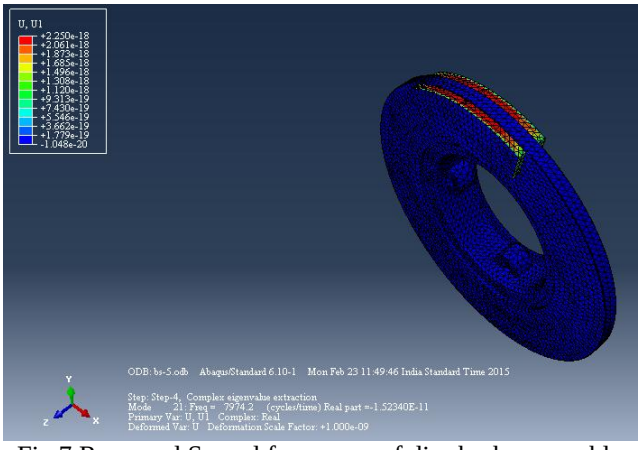


Fig.7 Removed Squeal frequency of disc brake assembly

VI. CONCLUSIONS

A method of complex eigen value analysis to identify the squeal frequency was presented. The method was then applied to disc brake assembly.

The proposed method was effective for obtaining the exact squeal frequency which realizes greater instability in the system. In the disc brake assembly example, the squeal found at the interaction between disc and brake pad. The model of disc brake assembly is created and analysed for natural frequency using Abaqus 6.10 and the results are compared with Experimental modal analysis using FFT for validation of model. Then squeal frequency of disc brake assembly with constraints is obtained using Abaqus 6.10. The proposed change in material is safe from squeal point of view. Grey Cast iron grade G3000 is suitable to removed the squeal frequency.

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