

# Suppression of Brake Squeal by Design and Analysis of Disc Brake Used In Two Wheeler

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**ABSTRACT:** FE models of the disc brake components an assembly are developed using FE software (ANSYS 15.0). In order to ensure that accuracy of the FE model agree with those of the physical components, two validation stages are used through experimental measurements at both individual component and at assembly levels. First, FE modal analysis at the component level is carried out and then, the mesh sensitivity of the each disc brake components is considered. In order to correct the predicted frequencies with the experimental results, a FE updating is used to reduce relative errors between the two sets of results by tuning the material. Finally, the integrated brake assembly model is corrected with measured data using proper contact interaction between brake components.

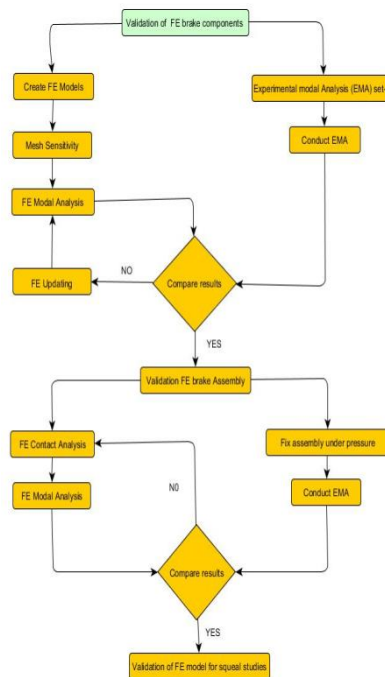
**KEYWORDS:** FE Modal analysis, Mesh sensitivity, FE updating.

## I. INTRODUCTION

Finite element analysis (FEA) is widely used to model the dynamic response of a structure and has the advantage that complex geometries can be accurately modeled. But accuracy of the FEA can be questionable and the reliability of the FE model must be validated by comparing the predicted results of natural frequencies and mode shapes of the FE model with the experimental results.

Experimental modal analysis (EMA) is one of the most useful areas of structural dynamics testing. It is a technique which has been widely used in structural engineering for finding the structure's dynamic characteristics under real mechanical conditions by determining modal parameters, such as natural frequencies, damping factors and mode shapes of a structure through experiments, then using them to formulate a mathematical model for its dynamic behavior. The formulated mathematical model is referred to as the modal model of the system and the information on the characteristics is known as its modal data.

In the last three decades, there have been numerous applications of modal analysis reported in literature covering wide areas of engineering, science and technology. One common reason for experimental modal analysis is the correction of the results of numerical methods. In practice, the accuracy of FE models is often limited by uncertainties about the actual geometry and material properties. For disc brake components, it is not possible to specify their exact material properties or geometry. Uncertainties in material properties or structural dimensions can be due to manufacturing and assembly imperfections, or lack of knowledge of material properties and coupling parameters between subsystems. Hence, experimental modal analysis is necessary to correlate the measured vibration behavior of disc brake components with that predicted by FEA.



**Fig.1 Methodology adopted in validation of FE model**

## II. EXPERIMENTAL MODAL ANALYSIS

It is a technique which has been widely used in structural engineering for finding the structure's dynamic characteristics under real mechanical conditions by determining modal parameters, such as natural frequencies, damping factors and mode shapes of a structure through experiments, then using them to formulate a mathematical model for its dynamic behavior.

### 2.1 Equipment for Experimental modal Analysis

In this work, Impact hammer, accelerometer and Fast Fourier Transform (FFT) are used to conduct experimental modal analysis, as shown in Fig.2. Frequency response functions (FRFs) are obtained by exciting the individual disc brake components and assembly using impulse hammer. The response for the given impulse is captured using an accelerometer. The natural frequencies and mode shapes are obtained through FFT analyzer. In the following subsection, equipment used for experiments will be discussed.



**Fig.2 Experimental modal analysis set-up**

#### 2.1.1 Impact hammer

For the excitation of the disc brake components and assembly, an impact hammer (Bruel & Kjaertype 8206) is used to make sure that enough energy is put into the structural. The impact hammer consists of impact tip, force sensor, balancing mass and handle. Impact tip is usually made of different materials (steel, plastic, various rubbers) to satisfy the requirement of frequencies of interest of the structure under test. The hard tip is selected, as the solid metal structure like disc brake as all modes have higher frequency above 200 Hz, by providing sufficient energy for high frequencies.

The force sensor is a piezoelectric transducer that is built into the hammer head to capture the impact force. This transducer generates a charge which is converted to a voltage which is used in the FFT Analyzer.

### 2.1.2 Accelerometer

For the present experiments, accelerometer (Deltron type 4534-B) with sensitivity of 9.507 mV/g is used. The mass of accelerometer is small to reduce the inertia effect of the accelerometer. This accelerometer is used to measured acceleration of disc brake components, in the form of voltage which is fed directly to the FFT Analyzer.

### 2.1.3 FFT Analyzer

The measured signals are analyzed by the FFT, where the excitation and response signals from the hammer and the accelerometer are acquired through a four-channel analyzer (Bruel & Kjaer Type 3050-B-040). The FFT also performs the transformations to convert the measured time domain signals into FRFs

## 2.2 Experimental Procedure

Generally, the frequency domain of EMA can be classified into three different types based on the number of FRF's which are to be included in the analysis. The simplest of the three methods is referred to as SISO (Single Input, Single Output) which involves measuring a single FRF for a single input given. In this experimental method, SISO method is adopted to perform EMA. The roving accelerometer technique is adopted in this research to perform modal test, this means that the Impact Hammer location is stationary and the Accelerometer location is changed.

### 2.3 Performing Impact Hammer Test

Before starting the test, two issues need to be considered. The first one is to identify a suitable location for mounting the accelerometer and to identify various locations on the structure to excite it using the impact hammer. The location should be at that point of the structure where a high response of the structure is expected. The second issue in the modal testing is to determine the tip of the impact hammer which is used for exciting the structure. The head tip to be used for exciting the structure should activate as many natural frequencies as possible.

### 2.4 Extraction of Modal Parameters

In order to extract modal parameters (natural frequency, damping factor and mode shape) the geometry of the disc brake components is created in commercial software (Solid Works 2013) to define the various points on which excitation is given in the experimental and the points at which response is measured. Once the geometry is created, the measured FRFs are given as input at the respective points at which they are measured.

### 2.5 Results of Experimental Modal Analysis

There are two types of experimental measurements which are conducted. The first is component level measurement using the free-free boundary condition which allows the structure to vibrate without interference from other parts, making easier visualization of mode shapes associated with each natural frequency and validation of corresponding FE model. This is implemented by hanging the components with the help of elastic cord during the testing. The other is assembly level test, using the actual boundary conditions. This is carried out by exciting the brake assembly under applied pressure.

#### 2.5.1 Experimental modal analysis of brake components

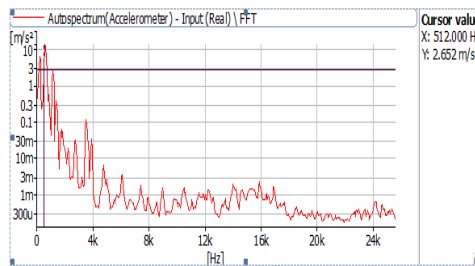
Experimental modal analysis of the rotor, pad, caliper and piston is conducted and analyzed individually with free-free boundary.

##### (i) Rotor

Measurements are taken at the response at each point in z (axial) direction, while the excitation is applied with the impact hammer in the z directions at the other points. As a result, direct measurement of only out-of plane is possible. The vibration responses at all the nodes are measured with accelerometers.



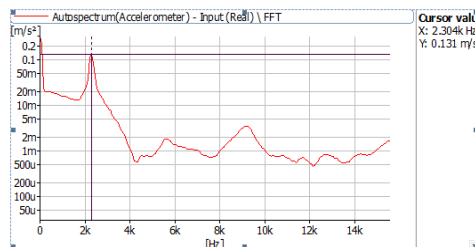
Fig. 3 Experimental modal analysis of Rotor



**Graph.1 FRF results of the rotor**

**(ii) Brake pad**

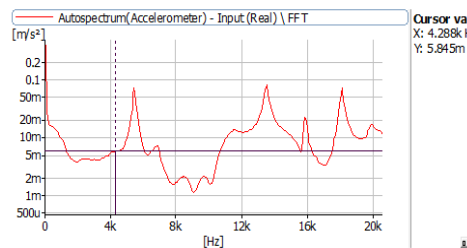
The modal analysis of the pad is carried out on the backing plate of the pad. The accelerometer fixed at middle point and the excitation in out-of-plane direction is applied to the rest of points. Figure shows FRF of brake pad.



**Graph.2 FRF results of brake pad**

**(iii) Caliper**

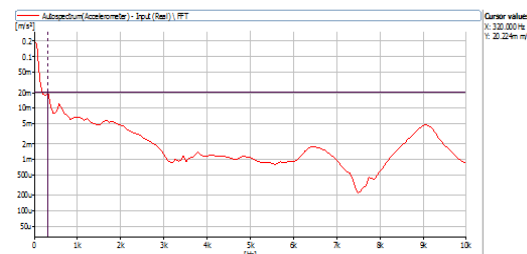
The accelerometer is fixed at the middle surface of the caliper housing and excitation is conducted on all points to capture as much of the vibration characteristics as possible.



**Graph.3 FRF results of caliper**

**(iv) Piston**

The following graph shows the FRF of piston

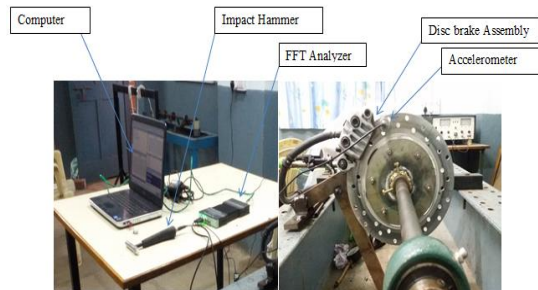


**Graph.4 FRF results of piston**

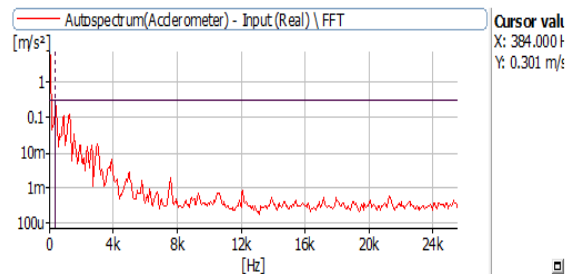
**(v) Disc brake assembly**

When the brake system works under applied pressure, the dynamics of the brake components are changed significantly. In this section, the individual components are fixed on a brake test rig under applied pressure using hydraulic pump as shown in Fig.4. Measurements are confined to the rotor surface. The experimental set-up otherwise is the same as for the rotor. The excitation is applied with the impact hammer in the normal direction. Graph.5 FRF results

of the brake assembly.



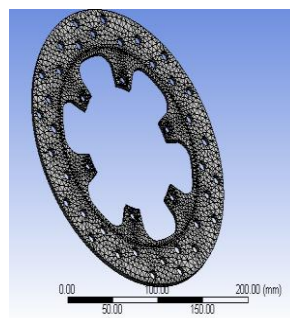
**Fig.4.Experimental modal analysis of disc brake assembly**



**Graph.5. FRF results of the disc brake assembly**

### III. MODAL ANALYSIS USING FINITE ELEMENT TECHNIQUE

The key to the success of the prediction of squeal is the correlation between physical tests and virtual FE modeling, at both component and assembly level. A three-dimensional FE model of a ventilated disc brake corner is developed and validated for identifying and fixing brake squeal problem in the earlier design stages. Figure 3.11 shows the 3-dimensinal FE model of the disc brake corner. For the purpose of this study, the development of finite element meshes for each brake components using the software (Solid Works 2013) is brieflydescribed. Detailed descriptions about the dynamic characteristics of FE models validated through experimental modal analysis are also made.



**Fig.5 Details of the FE model of Disc**

#### 3.2 FE Model Updating

The aim of FE updating is to reduce maximum relative errors between the predicted and experimental results. In some cases, it is not possible to know exact material properties of many dynamic structures due to a number of factors such as variation in material properties, geometry dimensions or changes in the excitation over time. Uncertainties in material properties or structural dimensions can be due to manufacturing and assembly imperfections, or imprecise knowledge of material properties and coupling parameters between subsystems.

Table.2 Material properties of disc brake components

Component s	Density (kg/m <sup>3</sup> )	Young's modulus (GPa)	Poisson's ratio
Rotor	7200	130	0.260
Friction Material	2800	2.6	0.340
Back plate	7850	200	0.300
Caliper	7500	175	0.260
Piston	8018	193	0.270

3.3.1 FE modal analysis of brake components

(i) Rotor

After conducting mesh sensitivity calculation the final FE model of the rotor modal analysis is carried on the rotor. There are number of natural frequencies and mode shapes exhibited in the FE results. However, only nodal diameter (ND) type mode shapes are considered to compare with experimental results, as illustrated in Fig.6

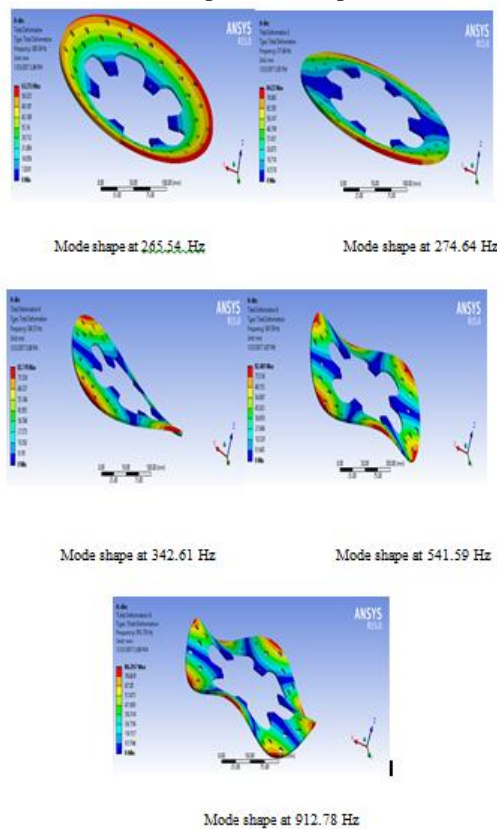
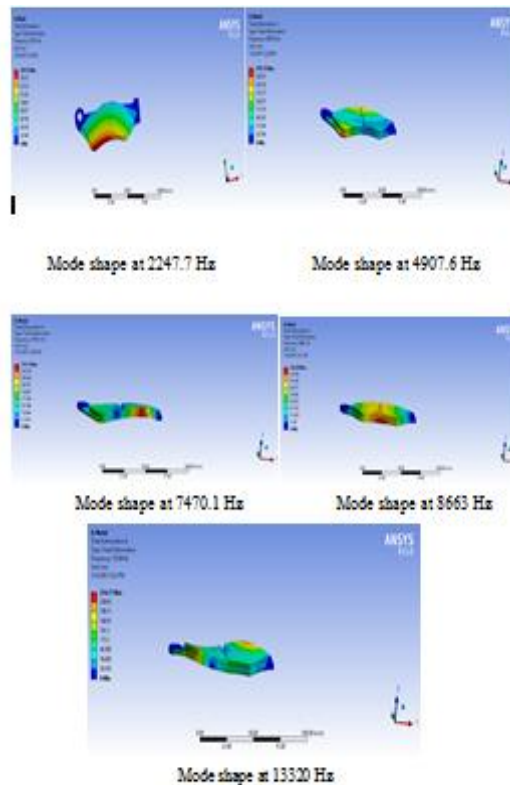


Fig.6 Rotor natural frequencies and mode shapes

(ii) Brake pad

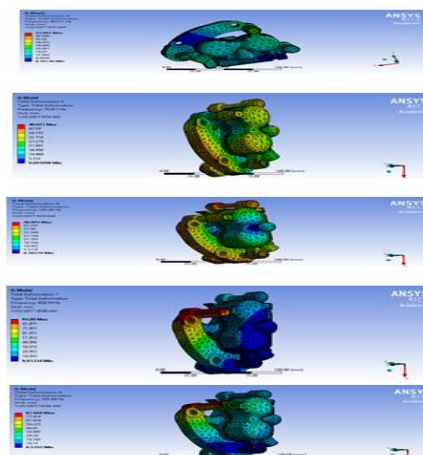
In the FE model, the friction material are assumed to be linear isotropic, for validation, the standard values of steel properties are used for back plate, and tuning material is examined for friction material. The natural frequencies and corresponding mode shapes obtained from the FE model are shown in Fig.7



**Fig.7.Brake Pad natural frequencies and mode shapes**

**(iii) Caliper**

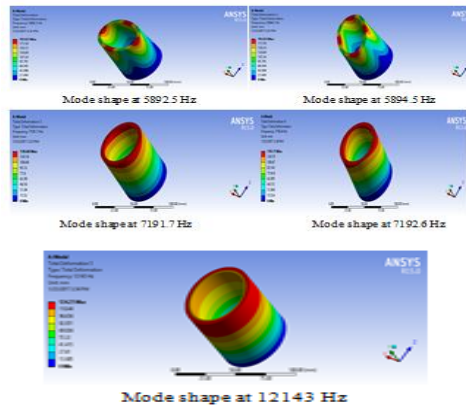
Material properties are adjusted to match modal frequencies of FEA and EMA results by using the same steps used for validation of the rotor. The results of the predicted results show good agreement with the measured data as shown in Table.3



**Fig.8.Caliper natural frequencies and mode shapes**

**(iv) Piston**

The predicted result from FEA is compared with experimental modal analysis and it is found that a good agreement with the predicted and measured data

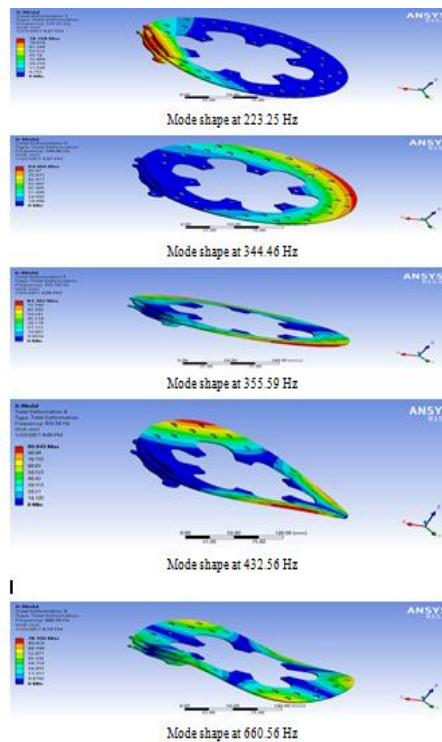


**Fig.9. Piston natural frequencies and mode shapes**

### 3.3.2 FE modal analysis of brake assembly

In this stage of analysis, all the brake components models are integrated together to form an assembly model and all boundary conditions and component interfaces are considered. In the FE assembly model, disc brake components are generally assembled by friction springs through a number of imaginary linear spring elements

A uniform pressure of 1 MPa is applied on the pads back plates. FE modal analysis then is conducted on the assembly model by considering all boundary conditions the interactions between all components. The natural frequencies and corresponding mode shapes obtained from the FE model are shown in as shown in Table.3 a good agreement is found between the predicted results and the measured ones.



**Fig.10 Disc brake assembly natural frequencies and mode shapes**





Table .3 Comparison between experimental and FE results of brakeComponents

Components	Mode	Exp. (Hz)	FE (Hz)	Error (%)
Rotor	1	267	265.5	-0.5
	2	275	274.6	-0.2
	3	340	342.6	0.7
	4	530	541.5	2.1
	5	898	912.7	1.6
Brake pad	1	2223	2274	0.2
	2	4930	4907	-0.4
	3	7456	7470.	0.1
	4	8697	8663	- 4
	5	13472	13320	-1.1
Caliper	1	67	69.57	2.8
	2	77	75.81	-2.5
	3	103	105.0	1.9
	4	465	450.3	-3.2
	5	499	503.0	0.7
Piston	1	5786	5892	1.8
	2	5787	5894	1.8
	3	7256	7191	-0.8
	4	7257	7192	-0.8
	5	12098	12143	-0.3
Disc brake Assembly	1	213	223.2	4.5
	2	334	344.4	3.0
	3	360	355.5	-1.3
	4	424	432.5	1.8
	5	667	660.5	-1.0

IV. CONCLUSION

In this work a comprehensive method for conducting modal analysis on a disc brake by both FE and experimental methods is described. The natural frequencies and mode shapes extracted from the FE model of the components are found to be in good agreement with those found from the experimental method and hence could be used for further dynamic simulation studies to investigate brake squeal problems.

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