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# Investigation of Natural Frequency and Modal Analysis of Brake Rotor Using Fea and Ema

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**ABSTRACT:** Brake squeal is largely accepted by Scientists and technologists as a major problem which is induced by friction induced vibrations. It is well accepting to be frequently occurring at a frequency above 1 KHz. This report is interested in the finite element analysis (FEA) and experimental modal analysis (EMA) of a commercial disc of the brake system. The modal analysis has been used in this work to determine the natural frequency and mode shape pattern. The destination is to predict the squeal at the former phase of invention using a more realistic model. Firstly, the FE models of the disc brake parts were produced using 3D solid element. The experimental modal testing technique known as an impact hammer test has been held away to obtain modal parameters of the disc brake structure. The FE model is validated by comparing experimental results of the brake components. It is ground that good correlation is achieved in the dynamic properties of the brake parts.

**KEYWORDS:** Squeal, natural frequency, mode shapes, frequency response function, Finite element analysis.

## I. INTRODUCTION

Squeal noise that generated in the disc brakes of the automobile has been handled as a one of the major businesses in the automotive industry due to the persistent complaints that reduces customer satisfaction with their vehicle. Most of the scientist and engineers have agreed that squeal noise in the disk brake is initiated by instability due to friction forces, contributing to self-excited vibrations [1]. Therefore, the elimination of brake squeal noise is very significant as the problem causes discomfort towards the vehicle occupants as well as walkers.

There have been many related studies carried out by the different researchers on the behavior of the disc brake through modal analysis with certain successes. Finite element method has been used by the researcher to several conclusions. The most usual technique applied is to compute M and K matrices to determine the natural frequencies and mode shape of the brake system components [2, 3]. In recent years, the finite element (FE) has reached wide acceptance for modeling brake vibration and noise problem.

In 1975, G.M.L. Gladwell and D. K. Vijay [4], develop a finite element modeling technique to recover out the natural frequencies of a hollow cylinder. Bae and Wickert [5], concentrated on the top-hat structure type of disc brake. Finite element model of the brake rotor disc was developed to study the influence of the top hat structure on the modalities of the brake rotor disc. The result proves how the natural frequencies of the structure related to disc thickness and hat structure. Further research by Tuchinda et al. [6], considered the same top hat structure type disc brake for their finite element analysis and resolved that the frequency of squealing are influenced by natural frequencies and modes of stationary rotors. Ibrahim [7], in his literature reviewed many theorems and mechanism such as stick-slip, sprag- slip, negative friction velocity slop and mode coupling of structures. Theoretically, it was shown that the mechanism such as stick-slip, sprag- slip, and negative friction velocity slop can cause chaos and

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instability of the system. One of the earliest researches of Liles [8], reported in 1989, developed a finite element model of each brake component using solid elements and validated their accuracy using an experimental modal analysis. He showed that squeal propensity increases at higher friction coefficients. Although successful case studies and research (GML. Gladwell, 1975; Liles, 1989; Ripin, 1995; Lee et al., 1998; Blaschke et al., 2000; Bajer et al., 2003; AbuBakar et al., 2006; Liu et al., 2007; Mario et al., 2008; Nouby et al., 2009) with certain success have been conducted in the last few years but no real reliable brake noise prevention tool truly exists yet. In fact, research done on a special case of brake or on a fussy type of vehicle is not assignable to other types of brakes or vehicle.

## II. MATERIALS AND METHODS

A detailed three dimensional FE model of a commercial disc was developed considering the realistic dimensions for better correlation of the consequences with the experimental. Disc of the brake assembly was examined considering the free-free boundary conditions as it allows the structure to vibrate freely without interference of the other parts. It also facilitates better visualization of mode shapes associated with each natural frequency. For validation of FE model one must recognize the dynamic properties as modulus of elasticity, density and Poisson's ratio of the components. Only linear behavior is viewed in a modal analysis while Non-linear properties are neglected.

## III. FE MODAL ANALYSIS OF BRAKE ROTOR

Modal analysis of brake rotor has been carried out for the materials Gray cast iron (GCI) as it is one of the essential parts of the brake system which contributes more in the generation of noise. First ten modes are extracted for the rotor for predicting the natural frequencies. Solid 45-3D structural solid element is utilized for the three-dimensional modeling of structure. The element is defined by eight nodes having three degrees of freedom at each node. Role of element shape and order of interpolation decides the element selection.

## IV. EXPERIMENTAL MODAL ANALYSIS (EMA) PROCEDURE

The disc of a brake assembly was tested through the EMA with free – free boundary conditions. The observational approach to investigate the way patterns and natural frequencies of the structure through impact hammer test consists of the next steps.

1. Generation of model.
2. Model test setting
3. Divide the structure inadequate number of points with the appropriate special distribution.
4. Shake up the structure with impact hammer.
5. Taking the measurements
6. Analysis of measured output data.
7. Establishment with the FEM data.

The test equipment used for the experimentation is the Fast Fourier Transform (FFT) with sixteen channels along with data acquisition system made of Scadas Front End. The structure was excited using impact hammer (Dytran Make 5800B3) at all predefined locations as indicated in fig. 1 and the response was collected using tri-axial accelerometer (PCB T356A02 & 356B21**)** at an identified driving point transfer function (DPTF) location. The type of EMA is known as the Frequency Response Function (FRF) method which evaluates the input excitation and output response simultaneously. The essence of all frequency response functions (FRF's) was resolved to extract natural frequencies and mode shapes. Fig.2 shows the experimental modal analysis set up for rotor.

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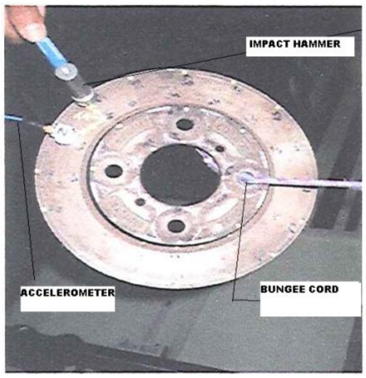


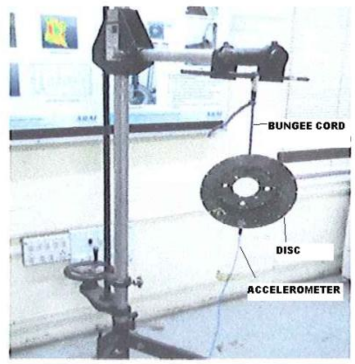


Fig.1 Accelerometer localization    Fig.2 Free-Free condition for Disc EMA

### V. RESULTS AND DISCUSSION

Natural frequencies obtained for disc of gray cast iron material using FEM for first ten modes are presented in the table 1. While table 2 shows the values of deformation i.e. displacement of disc at particular mode. From table 1, it is observed that the gray cast iron has acceptable lower natural frequency for first ten modes with the help of FEA. In fact, as per the definition of squeal range prescribed by the various researchers and experts found in the literature, GCI does not much contribute to the generation of brake squeal. Finally, it is noted that gray cast iron is considered as the best material composition for the disc rotor.

Table 1. Natural frequencies for first 10 modes of GCI materials of rotor.

| Mode No. | Frequency. (Hz) | Mode No | Frequency (Hz) |
|---|---|---|---|
| 1 | 1557.6 | 6 | 3395.4 |
| 2 | 1562.3 | 7 | 3433.9 |
| 3 | 2930.6 | 8 | 3729.2 |
| 4 | 3261.3 | 9 | 3762.1 |
| 5 | 3263.5 | 10 | 4649.3 |

Confirmation has also pulled in for the effects of natural frequencies of GCI materials of the same model (Mode1) obtained in the FEA in terms of damping ratio coefficients which is the ratio of modulus of elasticity and compactness. For Gary C.I, modulus of elasticity ($E_b$) = 125*Gpa* and density ($\rho_b$) = 7100 $Kg/m^{3.}$Thus the ratio of natural frequencies and damping ratio coefficient of the materials is equal to 0.7and are calculated by using the relation$f_a/f_b$and$\sqrt{E_a \times \rho_b}/\sqrt{E_b \times \rho_a}$respectively.

As the ratio between natural frequencies of GCI is equal to the damping ratio coefficient, which is the ratio of modulus of elasticity and density, therefore the modes of frequencies obtained through Finite element modeling are correct.

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Table2. Minimum and Maximum Deformation of rotor for First 10 Modes in mm

| Sr. No | Gray cast iron | |
|---|---|---|
| | Min. | Max |
| 1 | 9.35 | 1230 |
| 2 | 11.09 | 1232 |
| 3 | 59.37 | 985.05 |
| 4 | 3.42 | 1437 |
| 5 | 2.65 | 1417.7 |
| 6 | 58.56 | 1012.4 |
| 7 | 52.23 | 1043.6 |
| 8 | 31.15 | 1203.6 |
| 9 | 57.03 | 1230.1 |
| 10 | 1.50 | 3483.5 |

As talked about in the first place, a commercial disc was tested with the aid of experimental modal analysis. As a consequence, the natural frequencies and mode shapes obtained are really much nearer to the gray cast iron materials. Therefore, it is reasoned that the rotor material examined in EMA is the gray cast iron-2.

Table 3. Comparison of FEM and EMA Results for a Disc material Gray C. I

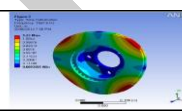
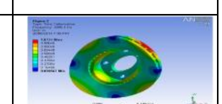
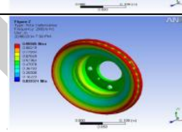
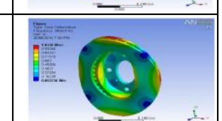
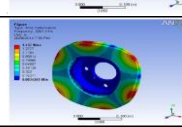
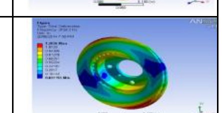
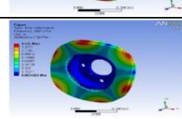
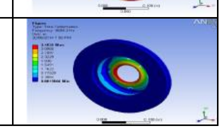
| Mode No | Natural Frequency (Hz) | | Mode Shapes | Mode No | Natural Frequency (Hz) | | Mode shapes |
|---|---|---|---|---|---|---|---|
| | FEM | EXPT. | | | FEM | EXPT | |
| 1 | 1557.6 | 1526 |  | 5 | 3395.4 | 3379 |  |
| 2 | 2930.6 | 3092.7 |  | 6 | 3433.9 | 3392.8 |  |
| 3 | 3261.3 | 3233.2 |  | 7 | 3729.2 | 3811.6 |  |
| 4 | 3263.5 | 3266.2 |  | 8 | 4649.3 | 4659.2 |  |

Table 3.Shows the comparison of FEA and EMA results for a gray cast iron material for the first eight modes. The FEM analysis deviation is considered acceptable as it fits within the scope. (Most of the deviation is zero) Small deviation is noted due to component production variability. (Geometric and material variations). For mode 1 and

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mode 2, the higher deflection region occurs on both sides of the rubbing surface with two nodal diameters. This is because the bottom sides are furthest away from the restraints. Both are having $90^0$ repetitions of rotational over the rubbing surface at one axis of symmetry that is 2 nodal diameters of normal modes of vibration. A circumferential mode for mode 3 can be described as a compression wave in the disc circumference similar to longitudinal mode of solid bar. On a mode 3, the higher deflection is restrained at outer region of rubbing surfaces throughout the circular disc. Modes 4 and 5 having a rotating pattern of $45^0$ with two axis of symmetric on the rubbing surface, which are 3 nodal diameter modes. Higher deflection is restraint at 3 different locations over the surface area of rubbing surfaces.

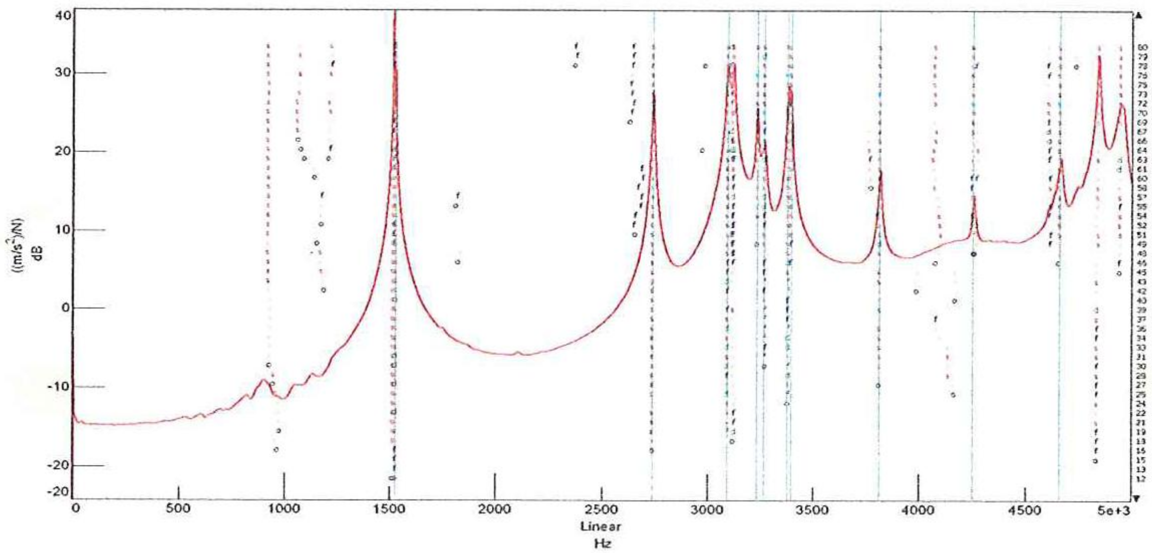


Fig. 4 Stabilization diagram for brake disc

The characteristics of a system that describe its response to excitation as a purpose of frequency is the frequency response function. Frequency of response for every excitation is measured as shown in fig 4. The phase of response in general case will be dissimilar than that of the excitation. The phase deviation between the response and the excitation will vary with frequency. Fig. 4 shows the resonant frequency for rotor which is chosen based on phase separation at that frequency. The peaks of amplitude correspond to the natural frequency of every mode. In fact, they correspond to the natural frequencies of the principal modes of the rotor that were excited, or participated in vibration of the rotor due to impacts.

## VI.CONCLUSION

The material properties of brake disc play an important role towards the vibration mode patterns. From the finite element analysis, the result shows that the ratio between Young's modulus over density increases the natural frequencies of the disc with respect to the number of modes and disc stiffness. When the values vary from 66 GPa (Gray Cast Iron 1) to 210 GPa (Steel), the two distinct modes at different frequency changes as the instability reaches its maximum level of frequencies when $E_{disc}$ = 210 GPa. From the values obtained it is concluded that Materials having lower value of the modulus of elasticity and higher value of density have lower values of natural frequencies.

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