Design and Analysis of Disc Brake for Low Brake Squeal

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Abstract: **Vibration induced due to friction in disc brake is a theme of major interest and related to the automotive industry. Squeal noise generated during braking action is an indication of a complicated dynamic problem which automobile industries have faced for decades. For the current study, disc brake of 150 cc is considered. Vibration and sound level for different speed are measured. Finite element and experimentation for modal analysis of different element of disc brake and assembly are carried out. In order to check that precision of the finite element with those of experimentation, two stages are used both component level and assembly level. Mesh sensitivity of the disc brake component is considered. FE updating is utilized to reduce the relative errors between the two measurements by tuning the material. Different viscoelastic materials are selected and constrained layer damping is designed. Constrained layer damping applied on the back side of friction pads and compared vibration and sound level of disc brake assembly without constrained layer damping with disc brake assembly having constrained layer. It was observed that there were reduction in vibration and sound level. Nitrile rubber is most effective material for constrained layer damping.**

Keywords: **Friction-induced vibration, Brake Squeal, Viscoelastic material, Constrained Layer damping**

I. INTRODUCTION

The most important safety feature of a car is the braking system, which must be slowed down quickly and reliably in different conditions. There are many types of braking systems that have been used since the beginning of a motor vehicle, but in principle they are all similar. The main function of the brake system is to slow down the vehicle by converting the kinetic energy of the vehicle to friction by heat, which must be efficiently and efficiently distributed around the brake components. The principle of disc braking was patented by Frederick Lanchester in 1902 at the Birmingham factory. However, in 1957, Jaguar was not as popular until it was introduced to the public in a way that the advantages of racing cars could be seen. Since the early 1960s, disc braking systems have become more common on most passenger vehicles, but some passenger cars use drum brake on the rear wheels, which are used to simplify parking brake conditions as well as reduce costs and weight. This can be a logical compromise, as front brakes do most of the braking effort. The current research has attempted to investigate the noises caused in passenger cars, so the disc brake will be the focus of this work. It is an important economic and technical issue in the automobile industry that a squeal noise is generated during braking. Reevaluation of customer requirements brings high comfort to the list of the most important design elements of the vehicle in order to offer the public a competitive and attractive product. Akay (2002) stated that warranty claims arising from noise, vibration and harshness (NVH) problems, such as brake sounds in North America, are only up to one billion dollars a year. The disc brake noise loudness is mainly due to friction-induced vibration, which causes noise at a sound frequency of 1 kHz to 16 kHz due to the dynamic imbalance of the brake system. Various theories and methods have been proposed to explain and predict the phenomenon of brake whining. Nevertheless, it is clear that no one can explain all the events related to noise. It has presented challenging problems for researchers and engineers due to the complexities of multidisciplinary structures such as theories about glazes, nonlinear dynamics, contact mechanics and tribology. Over the last few decades, a significant amount of research has been done by many researchers on the possibility of lifting the brakes to improve vehicle comfort, comfort, and overall environmental noise. Very good progress has been made and a number of solutions have been proposed, such as the addition of damping shims and shifting of modal connections. Despite this effort, squeal in the audible frequency range is still common. For this reason, in addition to eliminating brake leaks, a detailed and detailed study is required for forecasting. Sujay Hegde and B S Suresh [1] concentrated the wavering between the brake pad and disc utilizing a basic 1-DOF display. In this review, stick-slip phenomenon is recreated and considered by utilizing complex non-direct rubbing laws and contacts between a mass and moving belt. Additionally, a parametric review is directed to concentrate the impact of mass, spring and grinding parameters on the stick-slip impact. N.M. Kinkaid et al [2]. This paper provides a comprehensive review and bibliography of works on disc brake squeal. In an effort to make this review accessible to a large audience, background sections on vibrations, contact and disc brake systems are also included. Choe-Yung Teoh et al.[3] Transient analysis is carried out to determine the brake drum response under braking condition and the model produces squeal mode at 2026 Hz comparable to the measured squeal frequency of 1950 Hz. There are limited combinations of the location of Centre of pressure of the shoes that cause squeal. The amplitude of the limit cycle of the disc brake squeal can be reduced by increasing damping, mode frequency separation and reducing the contact stiffness. C.W. Park et al [4] The effect to brake disc corrosion on friction-induced stick-slip was studied to find the possible causes of friction instability In humid conditions. The friction and wear characteristics of gray iron discs and a commercial friction material were examined using a 1/5 scale dynamometer. Use of the corroded discs result in higher friction coefficient, oscillation amplitude of the brake torque. Disc corrosion increased the critical velocity showing transition from steady sliding to stick-slip. This suggests a rapid increase of the initial static friction coefficient as a function of dwell time in a humid condition, which is supported by the increased hydrophilicity of the friction films. P. Liu et al [5] 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 analyze disc brake squeal. An attempt is made to investigate the effects of system parameters, such as the hydraulic pressure, the rotational velocity of the disc, the friction coefficient of the contact interactions between the pads and the disc, the stiffness of the disc, and the stiffness of the back plates of the pads, on the disc squeal. G. Lou [6] Disk brake squeal noise is mainly due to unstable friction-induced vibration. A typical disk brake system includes two pads, a rotor, a caliper and a piston. In order to predict if a disk brake system will generate squeal, the finite element method (FEM) is used to simulate the system. At the contact interfaces between the pads and the rotor, the normal displacement is continuous and Coulomb's friction law is applied. Thus, the resulting FEM matrices of the dynamic system become unsymmetric, which will yield complex eigenvalues. Any complex eigenvalue with a positive real part indicates an unstable mode, which may result in squeal. A recently developed iterative method named ABLE is used in this paper to search for any unstable modes within a certain user-specified frequency range. The complex eigenvalue solver ABLE is based on an adaptive block Lanczos method for sparse unsymmetric matrices. Numerical examples are presented to demonstrate the formulation and the eigenvalues are compared to the results from the component modal synthesis (CMS) M. Triches Jr, et al. [7] The addition of a constrained layer material to brake pads is commonly utilized as a means of introducing additional damping to the brake system. Additional damping is one way to reduce vibration at resonance, and hence, squeal noise. This work demonstrates the use of modal analysis techniques to select brake dampers for reducing braking squeal. The proposed methodology reduces significantly the insulator selection time and allows an optimized use of the brake dynamometer to validate selected insulators. B. Ryzhik [8] The mechanism of excitation of friction-induced vibrations in a system comprising a flexible annular disk and two rigid surfaces is studied analytically. The surfaces are pressed together, and the rotating disk slides between them. It is shown that the sliding friction in the contact between the disk and the surfaces, together with the transverse contraction in the disk material, set up a feedback between the orthogonal eigen modes of the disk corresponding to the same eigen frequency, thus initializing instability. The instability mechanism is illustrated by simple analytical considerations. The obtained results are confirmed by finite-element analysis. S. Oberst et al. [9] Brake squeal has become of increasing concern to the automotive industry but guidelines on how to confidently predict squeal propensity are yet to be established. While it is standard practice to use the complex eigenvalue analysis to predict unstable vibration modes, there have been few attempts to calculate their acoustic radiation. Here guidelines are developed for numerical vibration and acoustic analysis of brake squeal using models of simplified brake systems with friction contact by considering the selection of appropriate elements, contact and mesh; the extraction of surface velocities via forced response; and the calculation of the acoustic response itself. Results indicate that quadratic tetrahedral elements offer the best option for meshing more realistic geometry.. Valery Pilipchuk et al. [10] Non-stationary effects in the friction-induced dynamics of a two-degree-of-freedom brake model are examined in this paper. The belt–spring–block model is designed to take into account variations of the normal load during the braking process. It is shown that due to the adiabatically slowing down velocity of the belt, the system response experiences specific qualitative transitions that can be viewed as simple mechanical indicators for the onset of squeal phenomenon .In particular, the creep-slip leading to a significant widening of the spectrum of the dynamics is observed at the final phase of the process. M. Graf and G.-P. Ostermeyer [11] We show how the stability of an oscillator sliding on a belt will change, if a dynamic frictional with inner variable is considered instead of a velocity-dependent coefficient of friction. Unstable vibration can even be found in the case of a positive velocity-dependency of the coefficient of friction. J.-J. Sinou et al[12] Friction-induced vibrations are a major concern in a wide variety of mechanical systems. This is especially the case in aircraft braking systems where the problem of unstable vibrations in disk brakes has been studied by a number of researchers. Solving potential vibration problems requires experimental and theoretical approaches. A non-linear model for the analysis of mode aircraft brake whirl is presented and developed based on experimental observations. The non-linear contact between the rotors and the stators, and mechanisms between components of the brake system are considered. Stability is analyzed by determining the eigenvalues of the

Jacobian matrix of the linearized system at the equilibrium point. Linear stability theory is applied in order to determine the effect of system parameters on stability. Choe-Yung Teoh et al[13]Transient analysis is carried out to determine the brake drum response under braking condition and the model produces squeal mode at 2026 Hz comparable to the measured squeal frequency of 1950 Hz. There are limited combinations of the location of centre of pressure of the shoes that cause squeal. The amplitude of the limit cycle of the drum brake squeal can be reduced by increasing damping, mode frequency separation and reducing the contact stiffness. Rob Redfield [14] Previous research incorporating bike frame structural dynamics and brake friction modeling has shown that stick-slip friction is likely the cause of much of this vibration and noise. Bicycle design parameters such as brake friction behavior and bike component structural properties are central in producing and/or sustaining these vibrations. The predicted dynamics of these models has correlated reasonably well with the testing of braking systems. This research extends the modelling of previous efforts to improve correspondence with brake noise/vibration testing and gain further understanding into the contributors and possible cures of this unwanted vibration. Specifically, the extended model incorporates torsional wheel dynamics (including rotor/hub, rim, and tire inertias, and spoke, rotor, and tire stiffnesses) into previous models. This new model allows the dynamics of the bike frame and wheel to couple through braking application. To support and validate the modelling, motion/vibration measurements are recorded during noisy braking with a non-contact laser vibrometer in the laboratory and with an accelerometer in field tests. Vibration measurements are studied along with model predictions toward the goal of connecting unwanted noise/vibration with specific design parameters of the bicycle brake-frame-wheel system. Eskil Lindberg et al [15] An experimental study of the friction-induced noise generated by the disc brake system of a passenger car is presented. In particular, the brake noise usually referred to as wire brush or roughness noise is studied. This is, in terms of frequency spectral content a broadband phenomenon, resulting from the interaction of multiple asperities in the tribological contact. A new experimental method for measurements of disc brake roughness noise is proposed, and is used in a lab environment where the vehicle speed and the brake pressure are accurately controlled. Mario Triches Junior, et al [16] Brake squeal noise has been under investigation by automotive manufacturers for decades due to consistent customer complaints and high warranty costs. J.D. Power surveys consistently show brake noise as being one of the most critical vehicle quality measurements. Furthermore, the development of methods to predict noise occurrence during the design of a brake system has been the target of many researchers in recent years. This paper summarizes the application of complex eigenvalue analysis in a finite element model of a commercial brake system. The effect of the operational parameters (friction coefficient, braking pressure and brake temperature) and wear on the dynamic stability of the brake system is examined. After identifying unstable frequencies and the behavior of the brake system under different conditions, the performance of some control methods are tested. The results show that the effect of brake temperature changes the coupling mechanisms between rotor and pad, which in some cases can be useful in order to reduce the instabilities and generated noise.. K. Soobbarayen, [17] This work proposes a complete characterization of brake squeal from the calculation of the non-linear vibration to the calculation of the associated sound pressure. A simplified finite elements brake system model composed of a disc and a pad is investigated. The contact is modelled by introducing several local contact elements at the friction interface and a cubic contact law is used to describe the contact force. The classical Coulomb law is applied to model friction and the friction coefficient is assumed to be constant. The stability analysis of this system provides two classical cases of instabilities which are single and multi-instabilities. For one and two unstable modes, non-linear time integrations and spectrum analysis are performed to detect all the harmonic components in the velocity spectrum. In this paper, the decomposition by harmonic components of the velocity is used to calculate the acoustic radiation by applying the boundary element method for each contributions. The sound pressure radiated is calculated for the two cases under study and a comparison in terms of levels and directivity is provided. It can be noted that the two unstable modes case presents significantly higher levels of acoustic pressure. In near field, directivity patterns for both cases are composed of four main lobes with different orientations. S. Oberst et al [18] In recent years, research has been focused on the prediction of unstable vibration modes by the complex eigenvalue analysis (CEA) for the mode-coupling type of instability. There has been very limited consideration given to the calculation of the acoustic radiation properties due to friction contact between a pad and a rotor. Recent analyses using a forced response analysis with harmonic contact pressure excitation indicates negative dissipated energy at some pad eigen frequencies predicted to be stable by the CEA. A transient nonlinear time domain analysis with no external excitation indicates that squeal could develop at these eigen frequencies. Here, the acoustic radiation characteristics of those pad modes are determined by analysing the acoustic power levels and radiation efficiencies of simplified brake models in the form of a pad rubbing on a plate or on a disc using the acoustic boundary element method based on velocities extracted from the forced response analysis. Results show that unstable pad modes trigger unstable disc vibrations resulting in instantaneous mode squeal similar to those observed experimentally. Changes in the radiation efficiency with pressure variations are smaller than those with friction coefficient variations and are caused by the phase difference of the velocities out-of-plane vibration between the pad and the disc. Jaeyoung Kang et al [19] The mathematical formulation for determining the dynamic instability due to transverse doublet modes in the self-excited vibration of a thin annular plate is presented in this paper. An analytical approach is developed to obtain the stability results from the eigenvalue problem of a stationary disc with a finite contact area. The approach uses the eigen functions of transverse doublet modes in classical plate theory and establishes the formulation of modal instability due to the modal-interaction of a doublet mode pair. The one-doublet mode model of a disc and a discrete model equivalent to the one-doublet mode model are proposed for providing a more fundamental understanding of the onset of squeal. The analytical models are validated through a comparison of results from a modal expansion model obtained from finite element component models. Throughout the analytical investigation, the pad arc length is found to be a critical design parameter in controlling squeal propensity. Daniel Hochlenert et al [20] Brake squeal is mostly considered as a comfort problem only but there are cases in which self-excited vibrations of the brake system not only cause an audible noise but also result in safety-relevant failures of the system. In particular this can occur if lightweight design rims having very low damping are used. Considering the special conditions of lightweight design rims, a minimal model for safety-relevant self-excited vibrations of brake systems is presented. It is shown that most of the knowledge emanated from investigations of the comfort problem can be used to understand and avoid safety-relevant failures of the brake system.

II. EXPERIMENTAL ANALYSIS OF DISC BRAKE ASSEMBLY

For the analysis of the disc brake group, it consists of the test setup, DC motor, two bearings and shaft. The disc is mounted on the center shaft and the shaft is mounted on two bearing at both ends. The caliper is fixed to the base frame in contact with the disc. The shaft is rotated by means of the PMDC motor and the shaft is rotated in the other bearing bed. All adjustments must be horizontally corrected to remove the effect of alignment. The entire installation is mounted on the rigid base frame with the aid of nuts and bolts. The proposed experimental setup is shown below in Figure 1;

Figure.1 Experimental set-up

For the brake squeal study of the disc brake system, it is necessary to measure the vibration and sound level of the disc brake system without constrained layer damping. For the current case study, a 150 cc disc brake is taken and the disc brake at different speed measures the vibration and sound level. DC motor is used for speed change. The rectifier is used to convert AC voltage to DC voltage. The input DC voltage varies with the help of the dimmerstat and the speed is measured with a tachometer. The accelerometer for measurement is mounted on the caliper in the radial direction and microphone is used to measure a sound level held at 1 meter from the mount. For measurement, the braking pressure is held constant and the speed changes from 300 to 1500 in five steps.

Table 1: Displacement and sound level of disc brake assembly without constrained layer damping under different speed at frequency of 0.250 Hz

Sr. No	Speed (RPM)	Displacement (mm)	Sound level (dBA)
1	300	4.278	75.317
2	600	4.424	76.332
3	900	4.454	76.633
4	1200	5.453	79.688
5	1500	6.245	80.203

2.1 Experimental modal analysis

It is a method which has been mostly 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.

2.1.1 Experimental Procedure

Generally, the frequency domain of EMA can be divided into three different types depends upon the number of FRF’s which are to be included into the analysis. The easiest of the three methods is referred to as SISO (Single Input, Single Output) which includes measuring a single FRF for a single input given. In this experiment, SISO method is considered to perform EMA. The roving accelerometer method is adopted for this research to perform the modal analysis tests. This means that the Impulse Hammer location is stationary, and the Accelerometer location is varying.

2.1.2 Performing Impact Hammer Test

Before performing the test, two problems need to be considered. The first one is to identify a suitable position for mounting the accelerometer and to search various locations within the structure to excite it using the impact hammer. The position of impact should be at that point on the structure where a large response to the structure is expected. The second problem in the modal testing is to determine the tip end with the impact hammer which is used for exciting the components. The head tip used for exciting the structure should activate as many natural frequencies as possible.

2.1.3 Extraction of Modal Parameters

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

2.1.4 Results of Experimental Modal Analysis

There are two types of experimental measurements, which are measured. The first is part level measurement using the free-free boundary condition which helps the structure to vibrate without disturbance from other parts, making easier visualization of mode shapes with each natural frequency and validation of corresponding Finite element model. This is implemented by hanging the components with the assistance of elastic string during the testing. The other is an assembly level test, using the actual boundary conditions. This is carried out by exciting the brake assembly under applied pressure with the help of hydraulic assembly.

Table 2: Mode and Experimental natural frequency

Components	Mode	Natural frequency (Hz)
	1	267
	2	275
Rotor	3	340
	4	530
	5	898
	1	2223
	2	4930
Brake pad	3	7456
	4	8697
	5	13472
	1	67
	2	77
Caliper	3	103
	4	465
	5	499
	1	5786
	2	5787
Piston	3	7256
	4	7257
	5	12098
	1	384
Disc brake Assembly	2	640
	3	896
	4	1216
	5	1536

III. FINITE ELEMENT ANALYSIS OF DISC BRAKE ASSEMBLY

Finite element analysis (FEA) is, for the most part, used to display the dynamic reaction of a structure and has the benefit that complex geometries can be precisely demonstrated. However, exactness 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 result. Experimental modal analysis (EMA) is one of the most useful areas of structural dynamics testing. It is a procedure which has been widely utilized in structural dynamic testing for finding the structure's dynamic attributes under genuine mechanical conditions by deciding modular parameters, for example, common frequencies, damping components and mode states of a structure through tests, then by using them to formulate a mathematical model for its dynamic behavior. The formulated mathematical model is referred to as the average model of the system and the information about the characteristics is known as its average information.

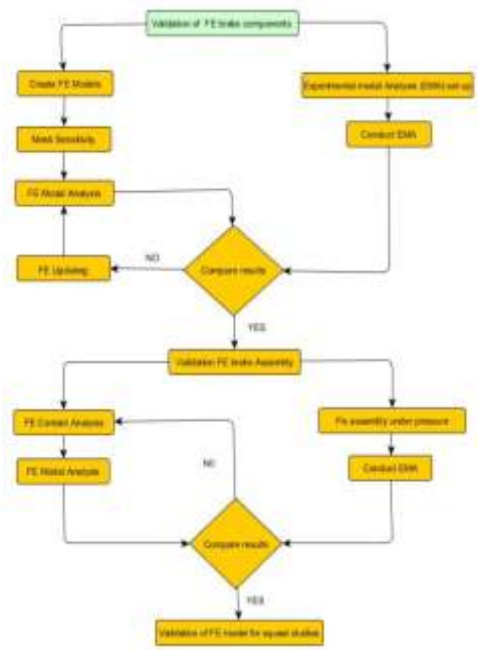

Figure 2: Methodology adopted in validation of FE model

3.1 Modal analysis using finite element technique

The way for the success of the prediction of squeal is the correlation between physical experiment and virtual FE modeling, at both part and assembly level. A three-dimensional FE model of a ventilated disc brake corner is produced and approved for identifying and fixing brake squeal issue in the earlier design stages. Figure 3 shows the 3-dimensional FE model of the disc brake corner. For this study, the development of finite-element meshes for each brake component using the software is briefly described

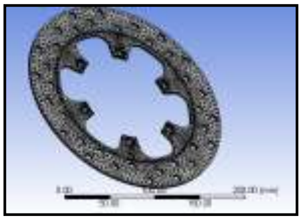

Figure 3: Details of the FE model of disc

3.1.1 Mesh Sensitivity

The accuracy for the results of FEM is very much dependent upon the mesh size of models. A mesh sensitivity calculation is utilized to choose the optimum number of elements within the FE model. Two different mesh sizes are connected to the same structure, where two modal data sets can be obtained. The natural frequency differences between the same modes of these two sets can then be used to determine the convergence frequency range

Table 3: Natural frequency difference with different mesh densities

Mode No.	Coarse mesh 1256 element	Medium mesh 2559 element	Fine mesh 5137 element	Difference ratio
	Natural frequency (Hz)			Medium-fine
1	269.12	265.54	255.15	3.9%
2	289.30	274.64	267.89	2.4%
3	356.89	342.61	337.54	1.4%
4	535.22	541.59	534.74	1.2%
5	905.51	912.78	869.25	4.7%

3.1.2 FE Model Updating

The main goal of FE updating is to decrease maximum relative errors between the anticipated and experimental results. Sometimes, it is impractical to know exact material properties of numerous dynamic structures due to a number of factors. For example,variation in material properties, geometry dimensions or changes in the excitation over time. Uncertainties in material properties or structural dimensions can be because of manufacturing and assembly imperfections, or imprecise knowledge of material properties and coupling parameters between subsystems.

Table 4: Material properties of disc brake components

Components	Density (kg/m^3)	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

3.1.3 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 a number of natural frequencies and mode shapes displayed in the FE results. However, only nodal diameter (ND) type mode shapes are displayed to compare with experimental results, as shown in Figure 4.

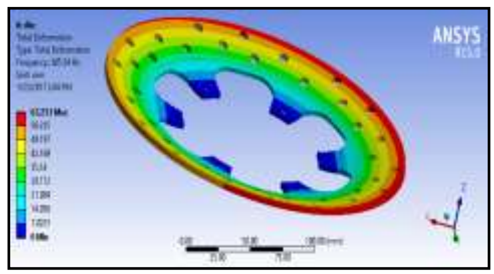

Figure 4: Rotor natural frequencies and mode shapes

(ii) Brake pad

In the FE model, the friction materials are assumed to be linear, isotropic, for approval, the standard values of steel properties are used for back plate, and tuning material is inspected for friction material. The characteristic frequencies and relating mode shapes acquired from the FE model are appeared in Figure 5.

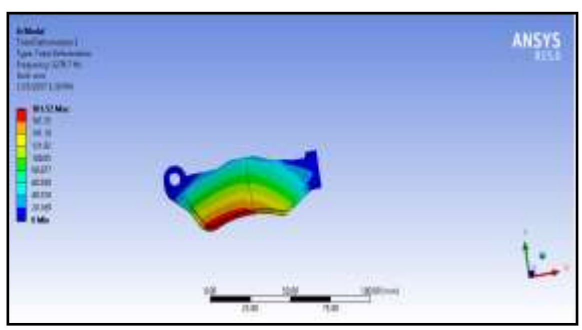

Figure 5: Brake Pad natural frequencies and mode shapes

(iii) Caliper

Material properties are changed in accordance with match modular frequencies of FEA and EMA comes about by utilizing similar strides utilized for approval of the rotor. The results of the predicted results show good agreement with the measured data.

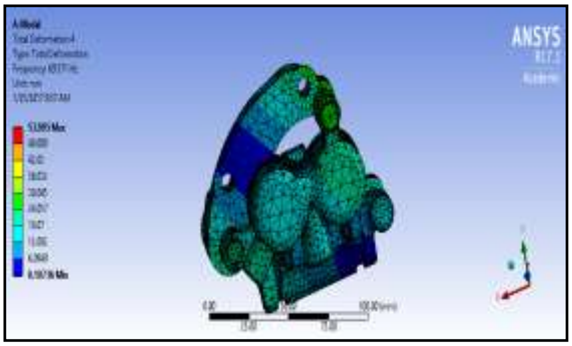

Figure 6: Caliper natural frequencies and mode shapes

(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

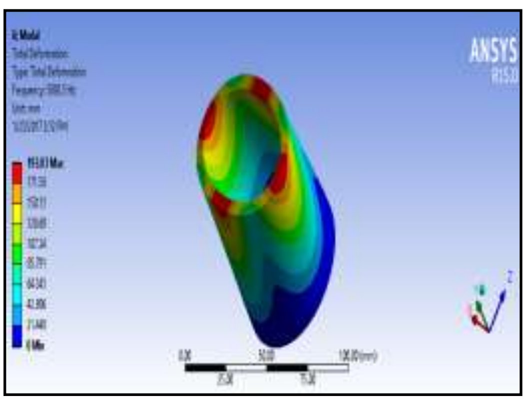

Figure 7: Piston natural frequencies and mode shapes

(iv) Disc brake assembly

In this phase of investigation, all the brake parts models are incorporated together to shape a get together model, and all limit conditions and segment interfaces are considered. In the FE gathering model, disc brake segments are, for the most part, collected by contact springs through various nonexistent straight spring components. A uniform pressure of 1 MPa is applied on the pads back plates. FE modal analysis then is led on the gathering model by considering all limit conditions communications between all parts. The regular frequencies and relating mode shapes acquired from the FE model are appeared is found between the anticipated outcomes and the deliberate ones

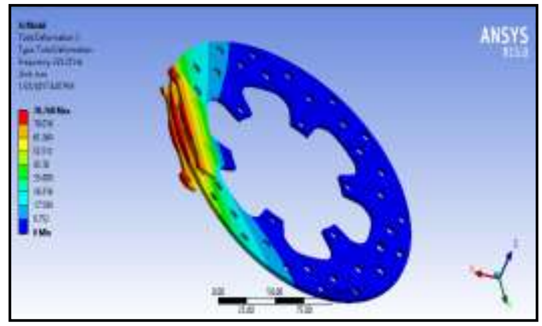

Figure 8: Disc brake assembly natural frequencies and mode shapes

IV. ANALYSIS OF DISC BRAKE ASSEMBLY WITH CONSTRAINED LAYER DAMPING

A most effective alternative to unconstrained layer damping in this condition is constrained layer damping (CLD). CLD works by adding a constraining layer of relatively stiff material to the outside of viscoelastic damping layer so that damping losses are generated by the shear strain in the damping layer. The following figure shows the CLD configuration. The main benefit of this compared with unconstrained layer damping is that the High loss factor of the composite system is no longer dependent solely on increases the thickness of damping layer. Very high composite loss factor can be created using very small quantities of damping material and the optimum configuration exists, which is a function of the damping material modulus, loss factor, thickness and wavelength of structural vibration in the parent and constraining plates.

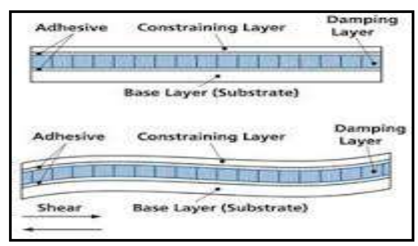

Figure 9: Constrained layer damping configuration

Brake noise insulators consist of a sandwich of two steel plates isolated by a viscoelastic or an elastic core, as seen in Figure.8, and its application to control brake noise has become one of the most efficient solutions. The insulator is very thin, and it is bonded or mechanically connected to the pad back plate. When the pad vibrates in the bending modes, a constrained layer material with a viscoelastic core bonded onto the pad back plate is submitted to mechanical distortion, converting part to the energy into heat by shear damping, reducing the vibration amplitude of the component.

There is different viscoelastic material are available in the market. For the convience five distinctive materials are selected based upon their availability, cost, thickness. These are natural rubber, Nitrile rubber, Neoprene (Chloroprene) rubbers, ETHYLENE PROPYLENE DIENE MONOMER (EPDM) rubber. Canvas rubbers are selected. All the materials are available with the minimum thickness of 1 mm. So based upon the loss factor of viscoelastic material, thickness of base plate and viscoelastic material, the thickness of constraining plate is determined for maximum performance.

4.1 Design of constrained layer damping

The performance characteristics of CLD applications are described with combined loss factor (η) for the base plate, viscoelastic material interlayer and the constrained plate being described by [21],

$$\eta = \frac{\beta YX}{1 + (2 + Y)X + (1 + Y)(1 + \beta^2)X^2}$$

Where shear parameter (X) and the structural parameter (Y) are given by,

$$X = \frac{G_2}{P^2 H_2} S$$

and $\frac{1}{Y} = \frac{E_1 \ H_1^3 + E_3 \ H_3^3}{12 H_{31}^2} S$

$$S = \frac{1}{E_1 H_1} + \frac{1}{E_3 \ H_3}$$

Where,

β=Loss factor of viscoelastic material

G_2 =Shear modulus of viscoelastic material

E_i =Elastic modulus of i^{th} layer

H_i =Thickness of i^{th} layer

H_{13}=Distance between neutral axis of the base plate and constraining plate

P= Wave number

It can be shown that the combined loss factor (η) of the CLD system will be Higher when the Shear parameter(X) is at an optimum (X_{opt})

$$X_{opt} = \frac{1}{\sqrt{(1 + Y)(1 + \beta^2)}}$$

For different viscoelastic materials, the steel plates at the thickness calculated by calculating the thickness of the different layers of the restricted layer damping are cut in the form of brake pad. With the help of the adhesive material, each viscoelastic material is applied to the back of the friction plate and applied to this constrained plate of specific thickness and the vibration and sound level of each damper is measured by means of the FFT Analyzer.

V. RESULT AND DISCUSSION

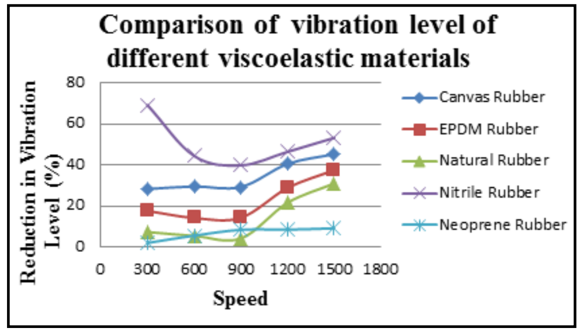

Figure 10: Comparison of vibration level of different viscoelastic materials

From the Figure 10, It is observed that up to speed of 900 RPM, there is a decrease in percentage reduction in vibration level for all viscoelastic materials but then there is increasing in percentage reduction in vibration level as speed goes beyond the 900 RPM.

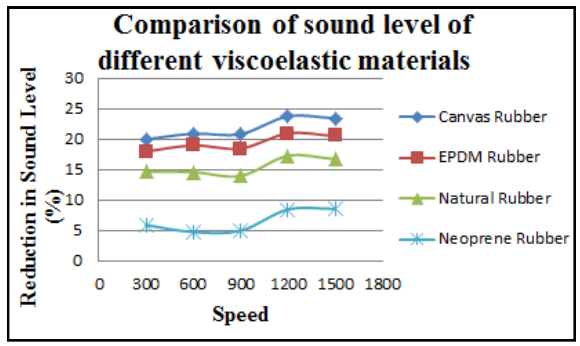

Figure 11: Comparison of sound level of different viscoelastic materials

From the Figure 11, It is observed that as speed goes on increasing, percentage reduction in sound level goes on increasing for all viscoelastic materials.

VI. CONCLUSION

The results obtained from the above experimental work are as follows:

1. From the measurement, it is observed that the vibration and sound level of the disc brake with constrained layer damping increases as the speed increases.
2. It has been observed that the natural frequencies and mode shapes extracted from the experimental modal analysis of the parts are largely the same as those found from finite element analysis.
3. From the modal analysis, it is observed that there is no mode coupling between the rotor and the brake pad, and thus can be used for more dynamic simulation studies to investigate brake squeal problems.

4. From Table 5.1 it is shown that the maximum loss factor of the composite system is independent of increasing the thickness of the viscoelastic layer.
5. By comparing the vibration and sound level of disc brake assembly without constrained layer damping and with constrained layer damping, it is found that there is reduction in vibration and sound level with viscoelastic material.
6. Nitrile Rubber material is often an effective material for reducing vibration and sound level.
7. It is observed that with nitrile rubber there is a 69% reduction in vibration level at 300 rpm and a 27% decrease in sound level at 1200 rpm.

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