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Analysis And Reduction Of Squeal Noise In A Disc Brake Of An Automotive

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ABSTRACT

Disc brake squeal is regarded as one of the difficult problem to solve in NVH since its inception. It occurs between ranges 0 to 16 KHz of frequency. It creates customer dissatisfaction which increases the warranty costs. Disc brake squeal is considered to be the effect of friction induced vibrations between the disc and the pad interface. This paper deals with solving the nonlinear problem arising due to frictional contact between rotor and the pad of the disc brake system of an automotive. This frictional contact generates unsymmetrical stiffness matrix and thus making the system unstable. A finite element approach was used to study the nonlinear behaviour of the disc brake system. An experimental modal analysis was carried out to validate the components of disc brake. A complex modal analysis was performed in ANSYS which gives the complex frequency wherein the positive real part shows the unstable mode. It was found that the instability so generated is due to the coupling of modes of rotor and the pad assembly. A squeal test was also performed to check the squealing frequency and the noise generated in the system. Structural modifications in rotor and the pad are also included in order to reduce the instability as well as to reduce the squeal noise

Keywords- **Disc brake system, friction induced vibrations, finite element analysis, complex modal analysis, mode coupling, squealing frequencies.**

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I. INTRODUCTION

From many years brake squeal is regarded as one of the major problems in automobile industry due fact that it causes customer dissatisfaction by creating the noise and also it leads to high warranty cost as squealing brakes are regarded as faulty brakes by customer. It still proves to be a major problem in automotive industry. The reason for this is that a brake has to operate without squeal under very different conditions and that it is very hard to predict whether a brake will be quiet under all of those. It is also difficult to develop accurate models of the frictional contact due to the complex structure of the pad material. Squeal is noise generally occurring above 1 kHz of frequency. Low-frequency squeal is defined as noise which occurs between 1 and 7 kHz, usually below the first rotor-in-plane mode. High frequency squeal which is above 5 kHz of frequency occurs usually at rotor-out-plane. This brake noise, with frequencies ranging from1 to 10 kHz is quite annoying since levels of 110 dB may be reached. The schematic diagram of disc brake and its working is shown in Fig.1.

Kinkaid et. al.[6] presented a wide literature over the various models and mechanism utilized for prediction and elimination of disc brake squeal. The major focus was on the vibration of disc brake assemblies. A linear vibration analysis model was introduced form the canonical form of equation of motion. This model deliberately was chosen so as to get roots of the equation formed which states the stability of system. As the central feature of disc brake model was to exploit the friction between the pads and the rotor to dissipate kinetic energy, so a friction model was introduced later with a simple example of slider on a horizontal plane. Effect of wear and temperature on coefficient was also included in it. A wide overview of experimental procedure utilized was included based on theories of vibration and tribology. To determine the mode shapes of the squealing brake a holographic interferometry was used. A few methods to eliminate the disc brake squeal were suggested. It includes use of an anti-squeal product, application of grease, use of vibration shims between backing pads and calipers, chamfering or slotting the pad structure, sanding the rotor surfaces and lubrication of pins that connects the caliper to its mounting brackets. It was also mentioned that squeal greatly occurred when the natural frequencies of components were close to each other. Many models based on certain theories were further discussed which includes decrease in coefficient of friction μ_k with increase in velocity v_s, sprag slip effect, decrease in coefficient of friction μ_k with increase in velocity v_s combined with sprag slip effect, self excited vibration with constant μ_k, splitting doublet modes and hammering.

Gottfried Spelsberg-Korspeter [1] stated relation between rotor symmetry and multiple eigen values. The excitation mechanism was based on rotating Kirchhoff's plate. With a simple example of four equal parts connected by spring with each particle having two degree of freedom, he determined the multiple non distinct eigen values which causes instability in a symmetric design. A topology optimization was performed based on splitting of eigen values so as to stabilize the design for which problem was to maximize the minimum value of eigen values. Symmetry was broken as a result of optimized rotor design. This modified rotor was then scanned under laser vibrometer with and without pads. It was found that the solid disc started squealing immediately as compared to modified rotor design.

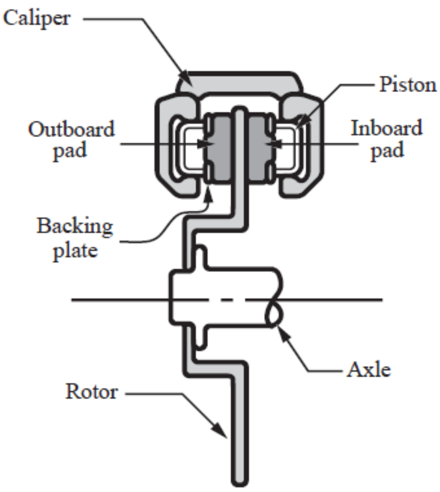


Fig.1 Disc brake system [6]

II. NUMERICAL ANALYSIS

Numerical Modal Analysis:

The numerical analysis was performed consisted of modelling disc and the brake pad under free-free conditions to obtain the natural frequency of the rotor and to validate it with the natural frequency obtained from experimental analysis. Modelling was done in Ansys workbench with the material properties as shown in table I.

TABLE-I

Material properties of disc brake system.

	Material Properties		
	Disc	**Backplate**	**Pad**
Young's Modulus (MPa)	195000	210000	87000
Poisson's Ratio	0.3	0.3	0.3
Density (Kg/m^3)	7850	8333	2520

A. Free-Free condition of Rotor

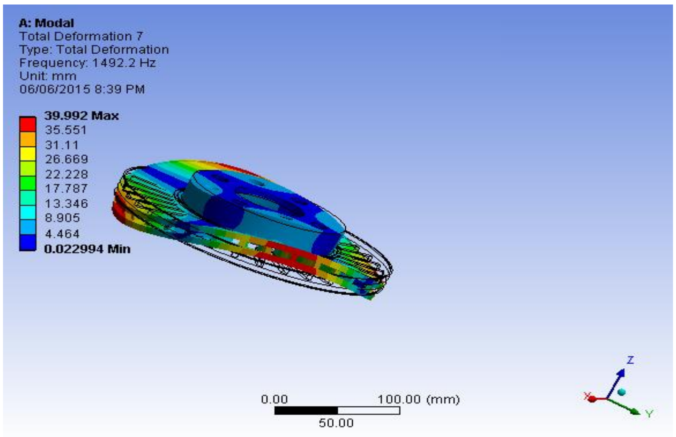


Fig.2 First natural frequency of rotor

The first natural frequency for the disc brake rotor was found to be 1492.2 Hz under free-free boundary conditions.

B. Free-Free condition of Pad

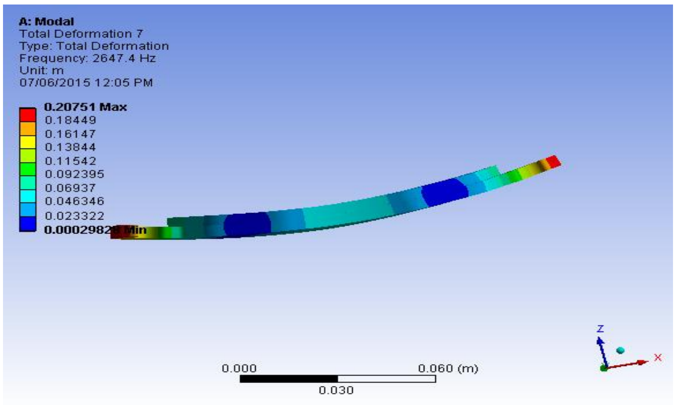


Fig.3 First natural frequency of brake pad

The first natural frequency of the brake pad was found to be around 2647.4 Hz under free-free conditions.

C. Modal analysis of the assembly

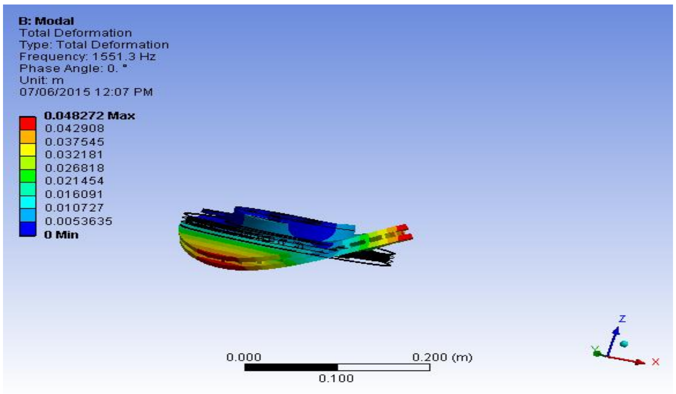


Fig.4 Modal analysis of disc brake assembly

The first natural frequency of the brake assembly was found out to be 1551.2 Hz.

III. EXPERIMENTAL ANALYSIS

Experimental modal analysis is done to validate the numerical model. The first natural frequency of numerical and experimental modal analysis was compared and percentage error was found. For experimental modal analysis accelerometer, an impact hammer, and Bruel Kjaer Analyser compatible with RT Photon software were used to detect the responses of the component and assembly. Fast Fourier responses of acceleration were noted to determine the natural frequency of the component and assembly.

A. Free-Free modal analysis of rotor:

The free-free condition for rotor was obtained by freely suspending the rotor with a harness string as shown in fig.5. A plastic tip was used for impact hammer to excite the modes of disc rotor. Around 3 trial runs were taken and the most prominent natural frequency was recorded.

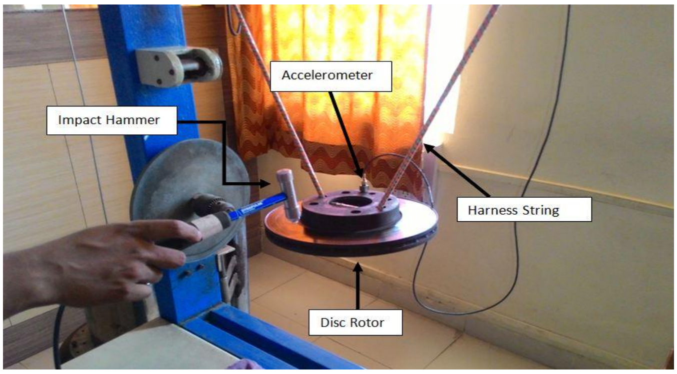


Fig.5 Free-Free boundary condition disc rotor

B. Free-Free modal analysis of brake pad:

The Free-Free boundary condition on the brake pad was implemented according to SAE J2598-2006 standard procedure. The brake pad was left resting on the foam bed as shown in fig.6. At one end accelerometer was kept and impact was given on the other end. The procedure was repeated by repositioning the accelerometer by interchanging the end positions meant for accelerometer and the hammer impact.

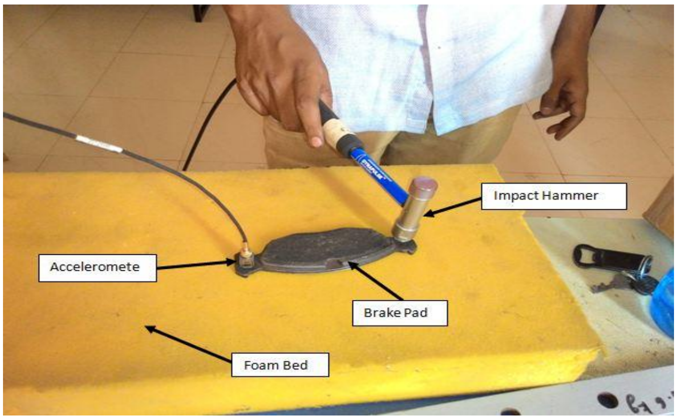


Fig.6 Free-Free boundary condition for pad

C. Modal analysis of assembly:

To obtain the natural frequency of the brake pad system, the disc brake was bolted to a chuck as shown in fig.7. A master cylinder was used to actuate the piston of the disc brake assembly. The pressure was recorded and the hand lever actuating the master cylinder was locked at 4 bar of oil pressure. Accelerometer was mounted on the hat of the disc brake rotor and impact was given at the opposite end of rotor surface.

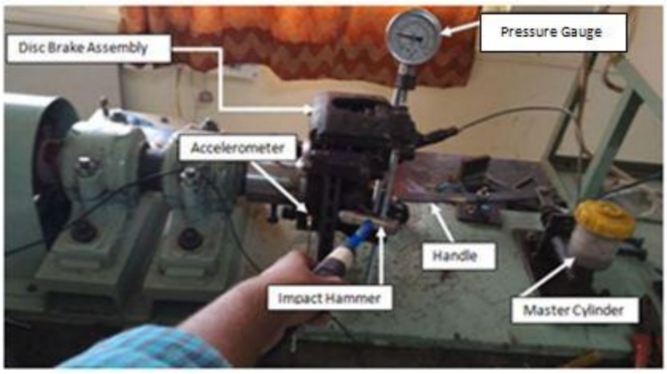


Fig.7 Modal analysis of disc brake assembly

D. Experimental vs Numerical:

From the natural frequencies obtained from the numerical and experimental analysis, it was found that numerical results are satisfactory and hence can be used for further analysis. The least error was obtained at the assembly level as shown in table II.

TABLE II
Numerical Vs Experimental Modal Analysis.

	NUMERICAL	EXPERIMENTAL	%AGE ERROR
ROTOR	**1492**	**1324**	**11.2**
BRAKE PAD	**2647**	**2288**	**13.56**
ASSEMBLY	**1551**	**1500**	**3.28**

IV.BRAKE SQUEAL ANALYSIS

A. Complex Modal Analysis

To study the instability in the disc brake model complex modal analysis is performed in Ansys work bench. Complex modal analysis is performed on validated model of disc brake system. Frictional contacts are given between the disc rotor surface and the brake pad surface. Pressure on pad surface was kept at 0.5 MPa. Disc rotor was given a rotational velocity of 8.33 rad/sec. Backplate surface was constrained in x and y direction and z direction was kept free to translate. The rotor motion was restricted using the cylindrical support and was free to rotate in radial direction and tangential and axial direction motion was restricted.

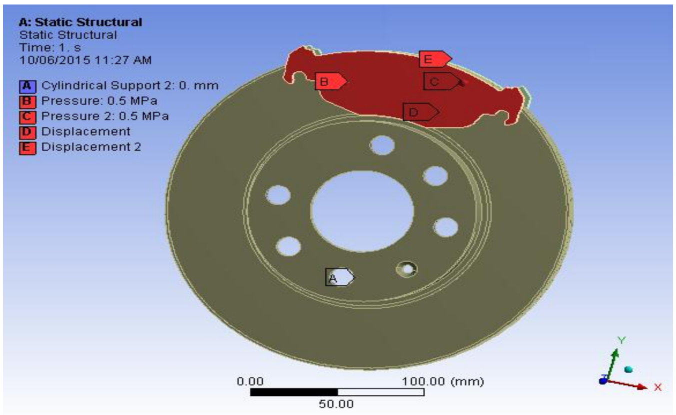


Fig.8 Boundary conditions for the assembly

The procedure so performed is known as full nonlinear perturbed modal analysis. In this procedure first a static structural analysis is performed so as to calculate the contact relation between the rotor surface and the pad surface. This contact is fed to modal analysis and the instabilities were calculated as shown in table III.

Table III
Complex modal frequencies

Mode Shape and number.	**Complex Modal Analysis**	
	Damped frequency	**Real Part**
16	**6712**	**16.04**
17	**6712**	**-16.04**

B. Mode Coupling Phenomenon:

When two or more component in contact having modal frequencies close to each other, the system tries to get couple and start to vibrate together at that frequency. So the wavelength coincides with each other leading to a coupled system. Due to coupling of modes the system damping gets reduced and the whole system tries to act like a loudspeaker. From fig.9 and fig.10 it can be seen that the 5th nodal diameter of disc and 2nd twisting mode has frequency close to each other. These modes try to couple together which generates the instability in the system.

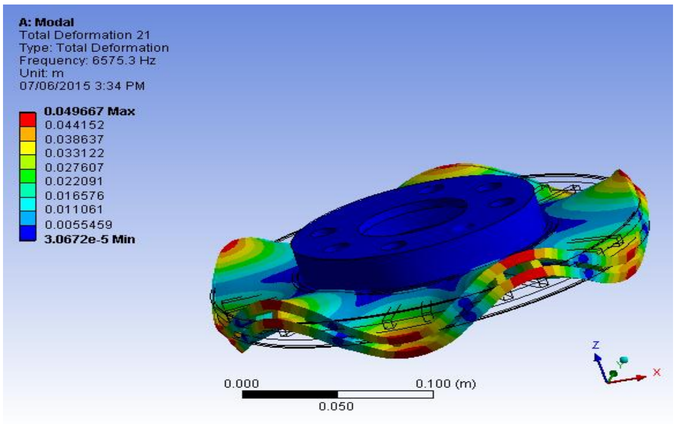


Fig.9 Fifth nodal diameter of disc

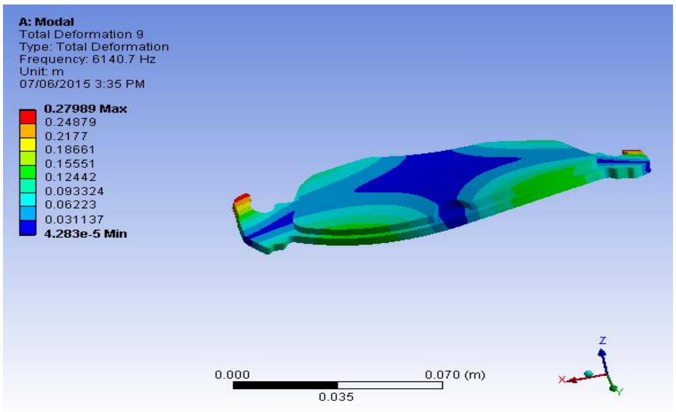


Fig.10 Second twisting mode of brake pad

3. Experimental Determination of Noise:

The schematics of experimentation are shown in fig.11. The disc was mounted on a shaft run by the belt drive. The belt drive was driven by an A.C. motor. The calliper of the disc brake was driven by the master cylinder. To note the applied pressure a pressure gauge was used. A micro phone attached to the analyser was used to detect the squealing noise in the disc brake system. The microphone was placed around 500 mm apart from the disc brake system. It was found that the squeal occur between the range 5500 Hz to 7000 Hz of frequency with sound pressure level of around 92 db value.

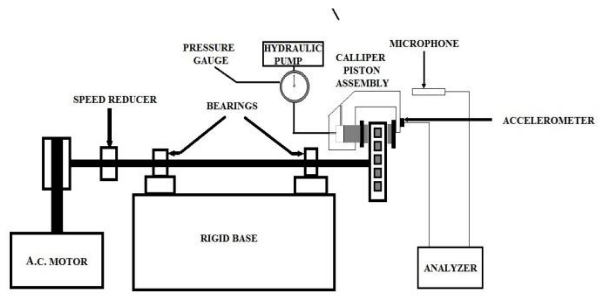
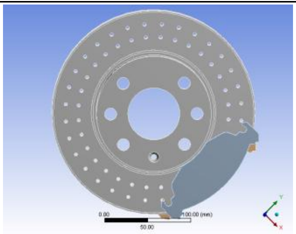
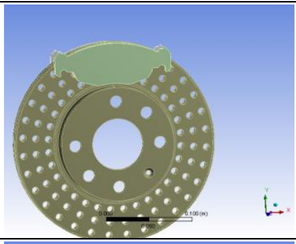
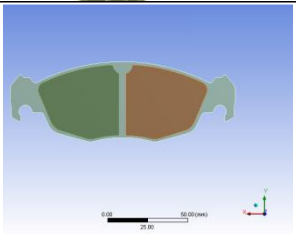
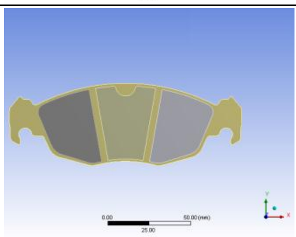
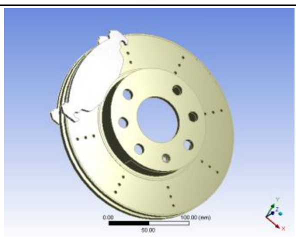
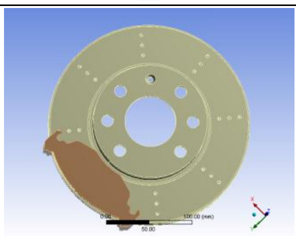
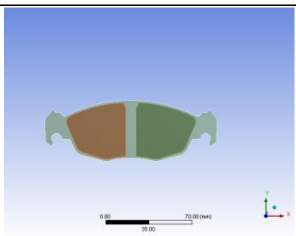
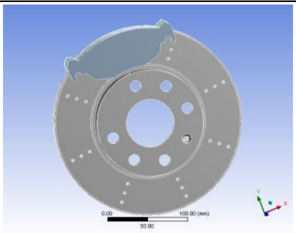
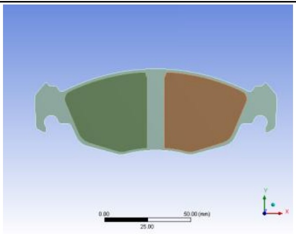


Fig.11 Schematics for noise determination

C. Structural Modifications in Disc Brake System

Various modifications were done in the disc brake system. The modifications and their results are shown in tabular form in table IV.

TABLE IV
Structural Modifications

Sr. No.	Configuration	Instability
1.		**14.032 at frequency of 3342 Hz.**
2.		**No instabilities.**
3.		**12.789 at frequency of 8666.5 Hz**
4.		**24.46 at frequency of 6377.4 Hz.**
5.		**15.366 at frequency of 6736 Hz.**
6.		**5.139 at frequency of 9356.2 Hz**
7.		**6.3419 at frequency of 8620.3 Hz.**
8.		**7.5711 at frequency of 9353.8 Hz.**
9.		**No instabilities.**

The first configuration has two holes of 4 mm diameter at an angle of 11.25˚. Second configuration has 3 hole of 4 mm diameter at an angle of 11.25˚. In third configuration modifications has been done a 4 mm slot has been given on the friction material. In fourth configuration two slots of 4 mm has been given. Fifth configuration is combination of disc having 4 mm holes at an angle of 45˚ and pad with double slot as shown in configuration number four.. Sixth configuration has same disc configuration as that in previous with only difference of pad configuration. The pad had a single slot of 4 mm as shown in third configuration. In seventh configuration the pad slot has been increased to 8 mm. The eighth configuration is combination of disc with 4 mm hole at an angle of 45˚ with pad slot of 8 mm. The last configuration has a 10 mm slot at the centre of the friction material.

V. CONCLUSION

The present work discuss about instabilities generated in a disc brake of an automotive. The main source of instability was found to be coupling of modes of rotor and the brake pad. The system tends to be unstable when the modes get coupled resulting in generation of squeal noise. The experimental investigation also proposes that the noise being generated is due to the coupling mode.

The structural modification studied shows the two configuration are suitable for reducing the squeal noise in a disc brake system. The one wherein the disc is modified with 3 hole of 4 mm diameter can't be a good solution to the problem as a large amount of surface interaction would be reduced resulting poor brake performance. Thus pad with 10 mm slot at centre is good solution to this problem as compared to that of drilled disc.

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