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FINITE ELEMENT MODELLING AND ANALYSIS OF BRAKE SQUEAL

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ABSTRACT

It is well-known fact that automobile brakes generate several kinds of noises like squeal, groan, chatter, judder, moan, hum and squeak. Squeal is the most prevalent, annoying and can be reduced by variations in geometry, parameters such as coefficient of friction, stiffness of material. The brake squeal generally occurs in the range of 1-16 kHz. Basically, two methods are available to study the disc brake squeal, namely complex eigenvalue analysis and dynamic transient analysis. Complex eigenvalue analysis is the standard method used for squeal analysis. Analytically it is very difficult to solve because of complex brake mechanisms. Experimental and numerical techniques have been developed by various researchers in order to study brake squeal. Experimental techniques are unable to predict brake squeal at the early stages of design process and also very costly due to associated design iterations. Therefore, finite element analysis has emerged as a viable approach for brake squeal analysis.

In this work Finite Element modelling and modal analysis of disc-pad assembly using high end software tools. Linear non-prestressed modal analysis and full nonlinear perturbed modal analysis is applied to predict frequency at which squeal occurs. Real and imaginary eigen frequencies of unstable modes are obtained.

Keywords: *Brake-squeal, frequency, FEA, PRO-E and ANSYS software etc.*

INTRODUCTION

The problem of brake noise is in general related to comfort and refinement rather than to safety or performance. Increased refinement in the other parts of a vehicle such as suspension, passenger compartment acoustics and transmission has turned attention to noise emanating from the brake. Legislation relating to noise level however is limited to continuous noise sources and therefore does not cover the intermittent nature of brake noise. Nevertheless the ideal solution in the form of a silent brake would help to bring about a better environment and reduce noise level particularly in places where stopping frequently occurs the ubiquitous noise from bus brakes for example.

Brake is a device by means of which artificial frictional resistance is applied to moving machine member, in order to stop motion of machine. During this the undesirable noise is produced called as brake squeal. Physically, squeal noise occurs when the friction coupling between the rotor and pad creates a dynamic instability. This leads to vibration of structure, which radiates a high frequency noise in the range of 1-16 kHz. Brake noise may be sub-divided into Low frequency squeal (1000-7000Hz) and High frequency squeal (8000- 16000Hz). Brake squeal has been one of the most difficult concerns associated with vehicle brake system. 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 complexity of the mechanisms that cause brake squeal. Experimental, numerical and analytical techniques have been developed in order better understand, predict and prevent the occurrence of brake squeal.

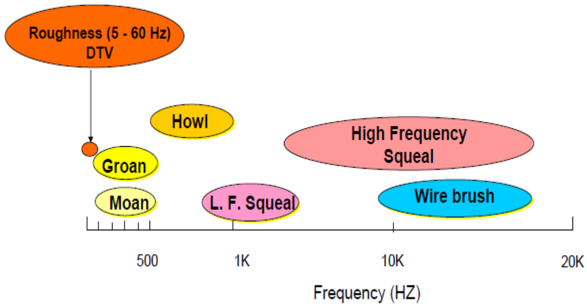

Fig 1 :-Frequency Range for Brake Noise

Disc Brake Components

A disc brake assembly, such as the one shown in Figure generally consists of four main components: (i) a brake disc (also called the rotor), (ii) a calliper, (iii) two brake pads, and (iv) mounting components.

The calliper, which contains a cylinder with a piston (multiple pistons may be used in some types of heavy-duty brakes), holds the two brake pads on either side of the rotor. The movement of the piston is controlled by a hydraulic system. When hydraulic pressure is applied, the piston is pushed forward to press the inner pad against the rotor while the housing is pushed in the opposite direction to press the outer pad against the rotor, hence, generating a braking torque. Brake pads are normally made of complex resin-based, short fibre Reinforced composites containing various friction modifiers (some types of brake pad also contain metallic material).

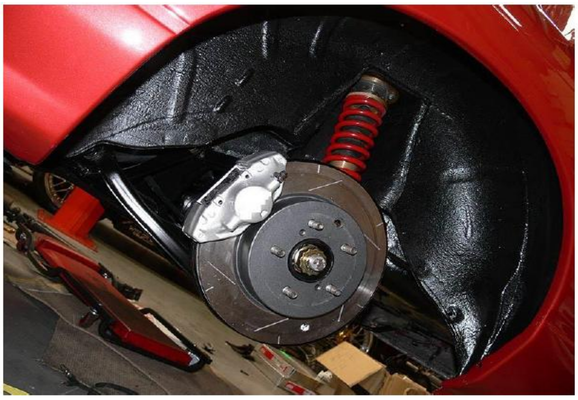

Fig 2:- Actual image of Disc Brake

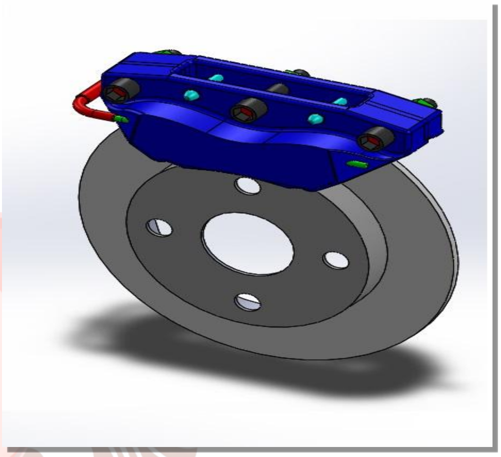

Fig 3 :- Assembly of Disc and Calliper

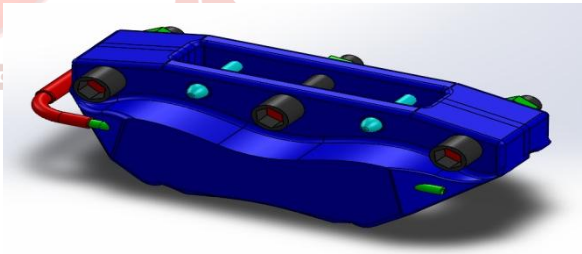

Figure 4:- Caliper

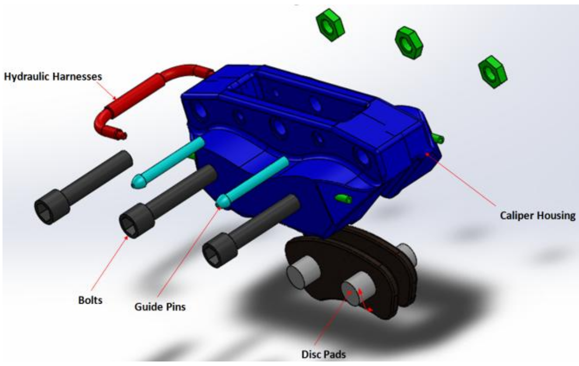

Fig 5:-Components Of Caliper

Solid Modelling of Disc-Pad Assembly

The brake disc assembly is modeled using CREO2.0 software. Dimensions of the disc-pad assembly are as follows: Inner diameter of disc: 225 mm ,Outer diameter of disc: 325mm ,Stud hole diameter: 20mm, Centre pad angle: 60^0 , Width of pad: 37.5 mm ,Outer diameter of pad: 167.5 mm, Inner diameter of pad: 130 mm, ventilation gap: 15 mm , Disc thickness: 10 mm ,Brake pad thickness: 15 mm

Following is the perspective view of model which gives clear idea about the disc-pad assembly.

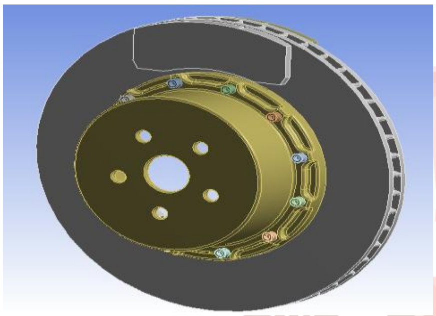

Figure 6:- Modelling of Disc Pad Assembly (first view)

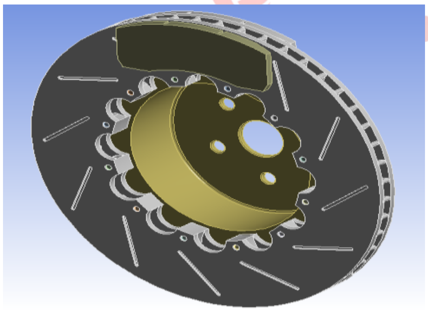

Figure 7:-Modelling of Disc Pad Assembly (second view)

Material properties and boundary conditions

Linear elastic isotropic materials are assigned to all the components of disc-pad assembly. Material properties assigned to disc-pad assembly are listed in table below.

Material Properties	
Young's Modulus (N/m^2)	2.0 E+11 Pa
Density	7850 Kg/m^3
Poisson's Ratio	0.2

Table:1 material properties of disc pad assembly

The inner diameter of the cylinder hub and bolt holes is constrained in all directions. Small pressure loading is applied on both ends of the pad to establish contact with the brake disc and to include prestress effects. The displacement on the brake pad surfaces where the pressure loading is applied is constrained in all directions except axial one (along Z).

ANALYSIS OF DISC-PAD ASSEMBLY

A] Modal analysis

It is commonly accepted that brake squeal is initiated by instability due to the friction forces, leading to self-excited vibrations. To predict the onset of instability, a modal analysis of the prestressed model is performed. Modal analysis is used to determine vibration characteristics, natural frequencies and mode shapes of a structure or a machine component while it is being designed. The frequencies obtained from the modal solution have real and imaginary parts due the presence of an unsymmetric stiffness matrix. The imaginary frequency reflects the damped frequency, and the real frequency indicates whether the mode is stable or not. A real eigen frequency with a positive value indicates an unstable mode. Modal analysis also can be a starting point for another, more detailed, dynamic analysis, such as a transient dynamic analysis, a harmonic response analysis, or a spectrum analysis.

B]Results of modal analysis of disc-pad assembly

The results of modal analysis of disc-pad assembly are shown Figure.11 below. modes of vibration are extracted by using Block Lanczos mode extraction method. Block Lanczos method is the default mode extraction method for modal analysis in Ansys. It is used for extraction of large number of modes of large models.

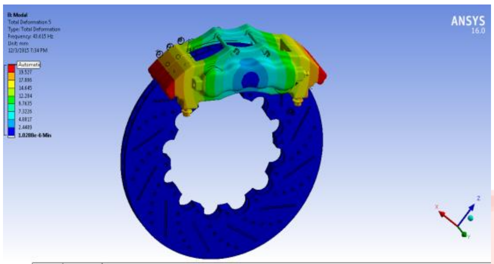

Figure 8:-Mode Shape 1: 43.61 Hz

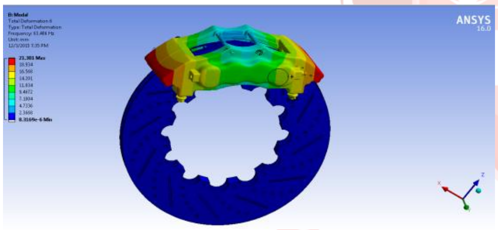

Figure 9:-Mode Shape 2: 63.40 Hz

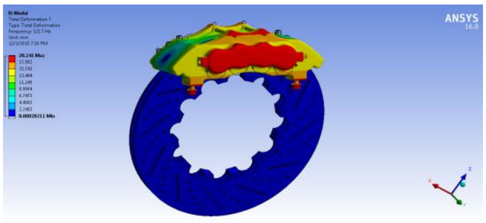

Figure 10 Mode Shape 3: 122.7 Hz

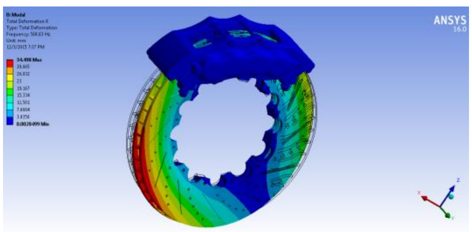

Figure 11:-Mode Shape 4: 568.03 Hz

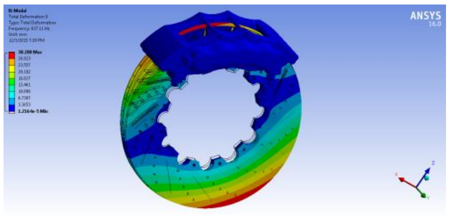

Figure 12:-Mode Shape 5: 637.11 Hz

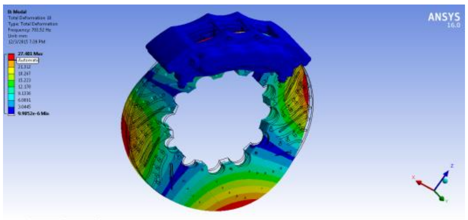

Figure 13 :-Mode Shape 6: 703.52 Hz

Results

The frequencies obtained from this modal solution (for μ=0) have real and imaginary parts due to presence of an unsymmetric stiffness matrix. Unsymmetric eigen solver is used to verify the eigen frequencies and mode shapes. modes are extracted as shown in table.

mode shape	Frequency (hz)
1	43.615
2	63.406
3	122.7
4	568.03
5	637.11
6	703.52

TABLE:-3

Parametric study with increasing the outer diameter of disc

A parametric study is performed on the disc-pad model using a full nonlinear perturbed modal solution with an increasing the outer diameter of disc in the range of 5% upto 10%. With increasing outer diameter of disc, the dimensions of pad also varied accordingly. As the size of disc and

Outer dia of disc (mm)	Frequency(hz)
325	6668.4
341.25	6373.5
358.3	6120.5

pad is varied changes in the frequencies and mode shapes are observed.

A]When outer diameter 325mm

When outer diameter is 325mm then the obtained mode shapes are shown in figure below.

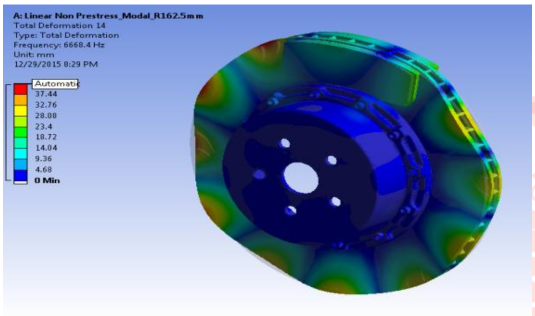

Fig 14:- Modal Analysis – Non Prestress

B] When outer diameter is increased by 5% is 341.25 mm

When outer diameter of disc is 341.25mm then the obtainedmode shapes are shown in figure below.

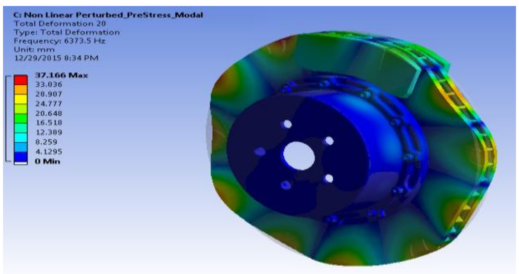

Fig15:-Prestress Modal Analysis – Non Linear Perturbed

C]When outer diameter is increased by 10% is 358.3mm

When outer diameter of disc is 358.3 mm then the obtained mode shapes are shown in figure below.

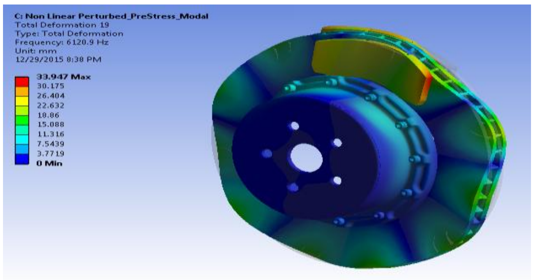

Fig 16:- Prestress Modal Analysis – Non Linear Perturbed

TABLE:-4

As above table show the increasing the outer diameter of disc the frequency get decreases

Test Set-up

Fig 17:-Experimental Setup of Laser Scanning Vibrometer

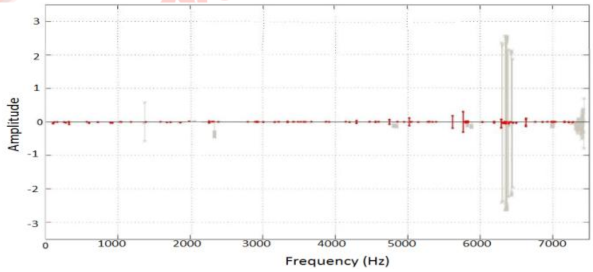

Fig:-18 Graph of frequency

RESULTS AND DISCUSSION

1. The first stage is to obtain dynamic characteristics of the individual disc brake components with free-free boundary conditions. The second stage is to perform dynamic characteristics of the complete assembly. The results showed that good agreement between the FE model using solid elements and measured natural frequencies.

2. we see that, when outer diameter is 325mm then the frequency is 6668.4hz. when outer diameter increase by 5% is 341.25mm then the frequency is 6373.5hz.when outer diameter is increase by 10% is 358.3 mm then the frequency is 6120.5hz. As the outer diameter of disc is increased the frequency of disc is decrease.
3. The frequency is obtain by FEA (ANSYS) method is 6373.5 Hz and frequency obtain from experimental method is 6320 Hz. Finite Element Analysis result error is 0.883% which is within the acceptable limit of 1%.

Table:-5 comparison of experimental result and FEA result

Experimental Result	FEA (ANSYS) Result	% Error
6320 Hz	6373.5 Hz	0.833

The above table compares the results of these two methods experimental and FEA analysis.As the full nonlinear perturbed modal analysis is accurate than linear non-prestressed modal analysis, it is used for squeal analysis.

CONCLUSION

In this paper work a methodology to analyze the squeal problem in brake using finite element method is presented. During this work following conclusions are found, Finite Element Analysis result error is 0.883% which is within the acceptable limit of 1%..Hence we successfully determine squeal frequency and unstable squealing modes of the disc-pad assembly. Hence we successfully Analysis of effect of increased outer diameter of disc on the modes and squealing.

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