Investigation of Disc Brake Squeal using FE Simulation and Experimentation

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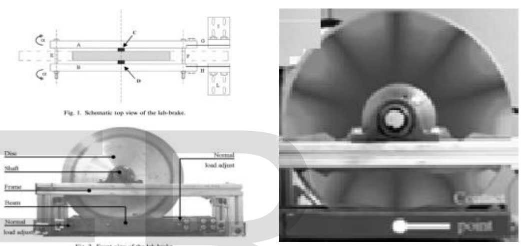
Abstract - **Brake squeal has been a continuing problem in the automotive industry and there has been several attempts to study and analyses the problem with the intention to solve it. Brake squeal occurs due to the instability during braking action between rotor and pad. There are various parameters which affect the brake squeal. In this paper, the results for the effect of speed and angle of attack are presented. Simulation and experimentation for the disc brake squeal are carried out. The static modal analysis of disc brake is done to find modal frequencies of mode shapes which developed during squealing. A Giannini's Laboratory brake setup is designed and fabricated to simulate squeal. Using Minitab, the design of the experiment is done by the factorial method to minimize the experiments. Squealing frequencies are compared with static modal frequencies. Using these frequencies in Minitab software the dominant parameter and regression equation is obtained. The angle of attack is found to be a more dominant parameter than speed for generation of squeal.**

Keywords- disc brake, squeal, speed, angle of attack

I. INTRODUCTION

Since the dawn of the automobile industry, squeal has been a phenomenon which occurs in disc brakes which is a serious problem. Squeal in disc brakes causes discomfort to the humans both on and off the vehicle and also decreases the life of the braking assembly making the vehicle vulnerable to accidents. The squeal is a high-pitched noise observed in disc brakes caused due to a dynamic instability of the brakes when the brakes are applied. Most squeal occurs due to vibration of the brake components, especially the disc rotor. Brake squeal is sub-divided into different types based on the range of squeal frequency: Low-frequency squeal- (1000Hz-7000Hz) and High-frequency squeal- (8000Hz-16000Hz). This problem is caused due to various factors which include breaking pressure, speed, friction, material variation, the angle of attack, etc. which individually or collectively contributes to the squeal [1]. The effects of speed and angle of attack are checked by simulation and experimentation in this study.

O. Giannini et al. [1] did the experimental modal analysis to study disc brake squeal and proposed laboratory brake setup as shown in figure 1(a) to simulate the squeal noise in experimental environment. Using high speed camera deformed shapes during squeal are observed are shown in figure 1(b).



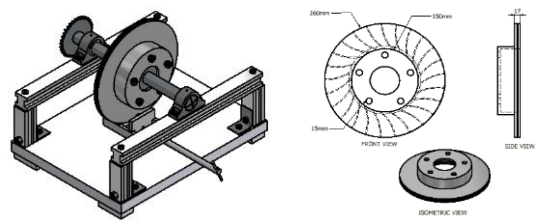
(a) (b)

Figure 1: (a) Laboratory brake setup V3.2 (b) The deformed shapes [1].

Masaaki Nishiwaki et al. [2] studied disc brake squeal experimentally by using dual pulse holographic interferometer (DPHI) technique. It was observed that the rotating disc during brake squeal generation vibrated in the bending mode with diametral nodes which seemed to be static on the coordinate of ground i.e. the brake squeal occurred in the vicinity of natural frequencies of the disc. The laboratory brake setup was found as a bridge between actual squeal generating environments in vehicles to the laboratory environment for the squeal. The disc brake squeal study is carried out in a simulation environment and on laboratory brake setup experimentally. The effect of speed and angle of attack are studied and presented.

II. MODELLING AND SIMULATION

The Experimental model is based on the model proposed by O. Giannini. Figure 2(a) and (b) shows the model of disc brake set up and disc brake rotor respectively. The main components of the model include disc brake rotor, caliper beam and brake pad.

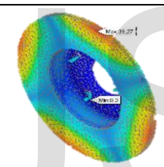
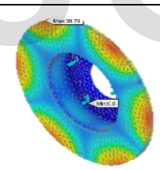
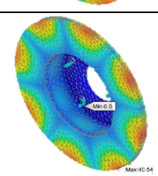
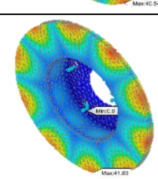
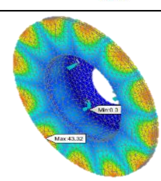
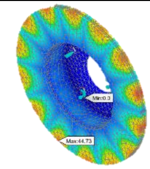
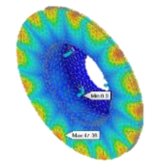


(a) (b)

Figure 2: (a) Model of Disc Brake Setup (b) Disc brake rotor

Nishiwaki et al [2] concluded that frequencies to be close to natural frequencies. Hence static modal analysis is done of disc brake rotor till 25 modes and those mode shapes were selected which were in the form of nodal diameters along with the modal frequencies. Selected mode with frequency value and mode shapes obtained from the simulation study are shown in table 1.

Table 1: Nodal frequency and mode shape for selected mode.

Nodal Diameter (mode)	Frequency	Mode shapes
2	4th (1640.149Hz) and 5th mode (1641.324Hz)	
3	6th (2246.345Hz) and 7th mode(2247.377Hz)	
4	8th (3103.385Hz) and 9th mode(3105.559Hz)	
5	13th (4106.852Hz) and 14th mode(4109.179Hz)	
6	17th (5194.163Hz) and 18th mode(5194.973Hz)	
7	19th (6333.676Hz) and 20st (6339.955Hz)	
8	24th (7521.046 Hz)	

III. EXPERIMENTATION AND ANALYSIS

Disc brake rotor selected for the experimentation is Tata sumo ventilated disc brake rotor. Structural components include a main robust bed, bearings, motor, load cell, speed reducer mechanism, and chain drive. The caliper beams which are originally present in O. Giannini's model were replaced by Double jaw vice for easiness in the application of brake with a single rotation of handle from both the sides of the rotor. The angle of attack and speed are selected as parameters. The pressure is applied by rotating the handle of double jaw vice acting as caliper between pad and rotor disc. The angle of attack is given to the pad by adding a washer at one of the two screws through which brake pads calipers are clamped to a jaw of the vice. The speed changes through the use of VARIAC connected to 1hp-1500rpm AC single phase motor and measured using a Laser tachometer. During experimentation, it was observed that deformed shapes during squeal generation were in the form of nodal diameters. Laboratory brake has three main components; Disc brake rotor, caliper jaws of double jaw vice and brake pad. Actual caliper pads were machined to the small size to contact surface area of $1cm^2$ [1]. The reason was given for the small size of a pad and low-pressure application between disc and brake so that their coupled behavior can be explained as some of those of separate components i.e. as if the components were vibrating on their own. Hence the commercial pad was machined with the help of abrasive cutter to give a contact area of $1cm^2$.

Laboratory brake setup consists of beams as caliper to hold the pads. The pressure application was done by tightening the screws provided on the caliper. But to increase robustness a slight modification was made in the form of double jaw bench wise clamp (figure 3). This clamp is such that both the clamp pads move equal distances upon rotation of handle present on one side of the clamp.

Figure 3: Double bench wise clamp

The angle of attack is one of the parameters being studied. To facilitate an angle between the surface of the pad and rotor, a washer was inserted between caliper and jaw in one of the two screws used to attach them together. The stuffing in the form of the washer at one side of caliper tilted to give an angle of 4.2 degrees as shown in figure 4. Maximum and minimum angles taken for study are +4.2 degrees and -4.2 degrees.

Figure 4: Setting for angle of attack

Speed is another parameter which is chosen for study. An electrical device called VARIAC is chosen to change the speed of the motor. The principle behind its working is it changes voltage about the motor to change its speed. The continuous speed change can be obtained from 0 rpm to 1495 rpm. One major limitation behind its use is that change in voltage means the change in torque and hence when the voltage is decreased speed also decrease and torque also decreases which jeopardizes the integrity of the study since torque must be sufficiently large to simulate stopping of the vehicle. Hence, the maximum and minimum speeds have taken for study are 1495 rpm and 1400 rpm respectively. Microphone attachment of audio technical headphone was used for greater accuracy for data acquisition of sound frequency of brake squeal and sound analyzer application in a smartphone to capture and analyses the frequency. Figure 5 shows the entire experimental setup for the squeal measurement.

Figure 5: Experimental Setup for squeal measurement

Verification of the data acquisition system was done by producing a sound frequency of 8 kHz and measuring that frequency with the help of sound analyzer application in the smartphone as shown in figure 6.

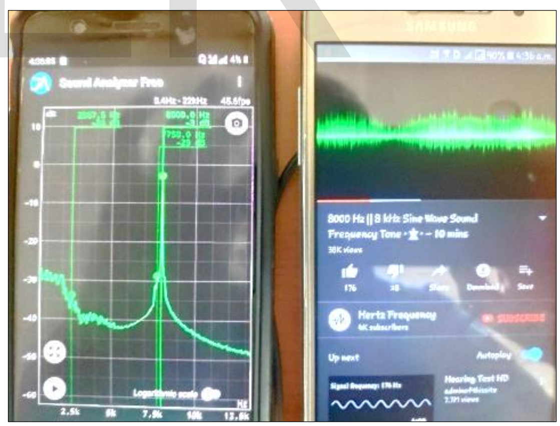


Figure 6: Sound Analyser Application Verification

During experimentation, a lot of noise due to the operation of motor and vibration due to minute misalignment existing between the rotor shaft and motor shaft was observed. Noise is so much that the actual intensity of squeal occurrence is difficult to distinguish. To find out that squeal frequency in all that noise first general graph of frequencies was obtained when brakes were not engaged as shown in figure 7. After brakes were engaged that frequency was noted whose decibel increase was found a maximum. The result of each combination of levels of parameters are obtained and are presented in figure 8 to figure 11.

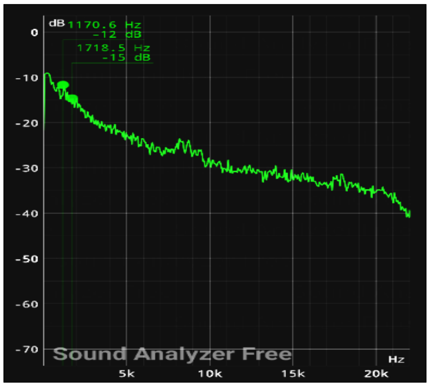


Figure 7: General Frequency Graph

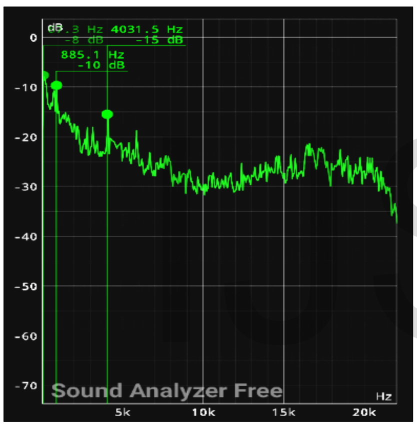


Figure 8: Frequency graph for Angle of Attack= -1(-4.2°) and Speed=1400 rpm

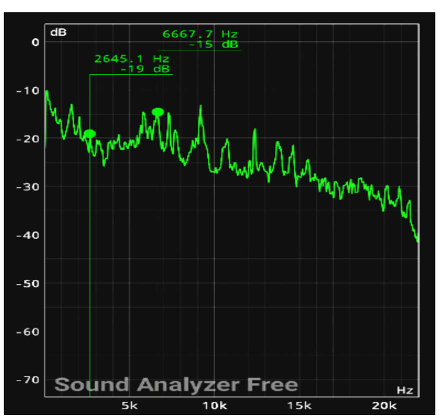


Figure 9: Frequency graph for Angle of Attack= +1(+4.2°) and Speed=1400 rpm

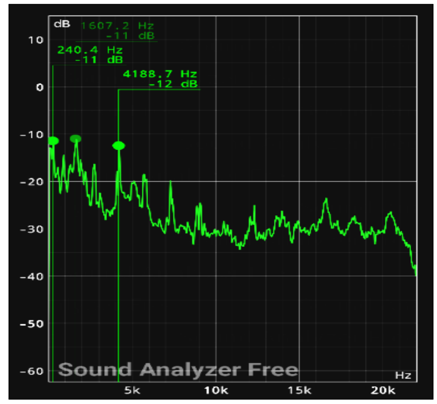


Figure 10: Frequency graph for Angle of Attack= -1(-4.2°) and Speed=1495 rpm

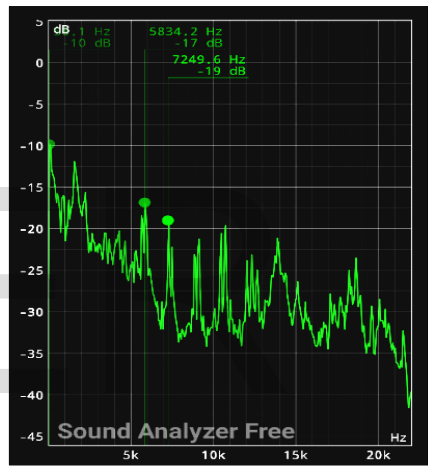


Figure 11: Frequency graph for Angle of Attack= +1(+4.2°) and Speed=1495 rpm

Values obtained from experimentation is substituted into the Minitab work sheet that was formed and analysis is done to get the regression equation and Pareto chart displaying effects of a parameter in comparison with each other on output (frequency). While filling out values of factors for speed actual values were given but for the angle of attack +1 was given for +4.2 degrees and -1 was given for -4.2 degrees. This was done for simplicity of computation. Pareto effect chart is shown in figure 11.
The regression equation obtained as

Frequency = -96.4 - 1811 Angle of Attack + 3.890 Speed +2.235 Angle of Attack * Speed

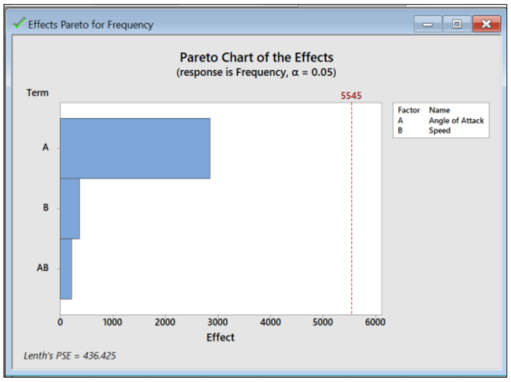


Figure 12. Pareto effect chart for frequency

From Pareto effect chart it is found that Angle of attack is more dominant parameter than speed which affects the frequency.

IV. RESULT AND CONCLUSION

A comparison between squeal frequency obtained from experimentation and modal frequencies obtained from FEM analysis is as given in table 2.

Table 2: Comparison of squeal frequencies obtained from experiment and FE simulation

Sr. No.	Squeal Frequency from experimentation (Hz)	Closest Modal frequency from FEM analysis (Hz)	Nodal Diameter mode
1	4031.5	4109.179	5
2	4188.7	4109.179	5
3	6667.7	6339.955	7
4	7249.6	7521.046	8

From the result shown in table 2, it is observed that squeal frequencies are close to the static modal frequency of the disc brake rotor. Upon observing frequencies, it can be seen that at lower squeal frequency it conformed to a modal frequency more closely than at higher squeal frequency. Hence, it reaffirms the observation made by Nishiwaki et al [2] that squeal frequencies occur in the vicinity of natural frequency disc brake rotor. Hence one solution appears to be to increase the overall natural frequency of the material by choosing a stiffer material so that increased Young's modulus would mean more stiffness and hence would increase natural frequency. This should be such that natural frequencies for nodal diameters 5, 6, 7, 8, 9 appear quite near or above 20 kHz so that it goes beyond the human range of hearing. But this would mean more capital investment and hence would lead to loss to the manufacturer.

V. CONCLUSION

From the experimentation, it is observed that the angle of attack proved to be a more dominant parameter for generation of a squeal than speed. Since this parameter can be controlled in designing of a brake system, It has to be set in such way that angle of attack is minimum i.e. tend to zero at all times of operation.

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