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Study and Analysis of Disc Brake to Reduce Disc Brake Squeal

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Abstract: Brake squeal noise has been under investigation by automotive manufactures for many years due to consistent customer complaints and high warranty costs. Disc brake squeal remains a complex problem in the automotive industry, since the early 20th century, many researchers have examined the problem with experimental, analytical and computational techniques. As it is complex phenomenon, still complete solution is not found. Although brake squeal do not affect braking performance, still it is not acceptable. So brake noise issues have led vehicle manufacturers, brake and friction material suppliers to investigate various ways of improving their processes in order to reduce vehicle noise and increasing passenger comfort. In this project various parameters influencing disc brake squeal are studied. Various parameters are braking pressure, rotational velocity, coefficient of friction, damping, modifications in disc, pad. During braking operation braking pressure, rotational velocity are not in control. Decrease in coefficient of friction reduces the brake squeal, but it is not applicable because it lowers the braking performance. Damping shims to reduce squeal increases the cost of damping material. So best way to reduce disc brake squeal is structural modification in disc brake assembly. In this project asymmetry is introduced to the disc to reduce squeal. It is found that introduction of cyclic asymmetry to the disc increases the difference between natural frequencies of the adjacent modes.
Keywords: Disc brake, EMA, FEA, Natural frequency, Squeal.

I. INTRODUCTION

Brake squeal is one of the most frequent field claims in automotive industry. Experts and scientists have been conducted researches over the decades in academics and industry. Friction induced vibrations are very common problems in engineering. Squealing of vehicle brakes is complex phenomenon and thus much effort is spent to overcome this problem. So its investigation requires expertise from different disciplines (e.g. vibrations, tribology, acoustics, etc.) as evidenced by the number of approaches to the problem and the variety of explanations on how brake squeal originates that have been proposed so far. However, there is no complete understanding of the problem and not a generalized theory of squeal mechanism. The high complexity of a brake system is the main difficulty encountered in studying brake squeal. Commonly, investigations are done by combining theory and experiments by most of research groups working on this subject. Many such studies started by performing experiments on simplified test rigs, trying to correlate the experimental results with theoretical models. The aim of studies is to get clear understanding of the squeal mechanisms and then to extend such knowledge to build complex models and, finally, to control the squeal occurrence in commercial brakes. Brake squeal is a high-pitched noise in the frequency range between 1 kHz and 16 kHz originating from self-excited vibrations which are caused by the frictional contact between brake pads and brake disc.

A. Components of Disc Brake

There are several major components of a disc brake: the rotor, calliper, brake pad assemblies and a hydraulic actuation system. The rotor (or disc) is rigidly mounted on the axle hub. So it rotates with the automobile's wheel. The pair of brake pad assemblies consists of friction material, backing plates and other components. Brake pads are pressed against the disc which generates a frictional torque to slow the disc (and wheel's) rotation. The calliper houses the hydraulic piston(s) which actuate the pad assemblies. When a driver depresses the brake pedal, there is increase in hydraulic pressure in the pistons housed inside the calliper. The device which converts the brake pedal's motion to hydraulic pressure is known as the master cylinder. It is connected by brake lines and hoses to the disc brake's calliper.

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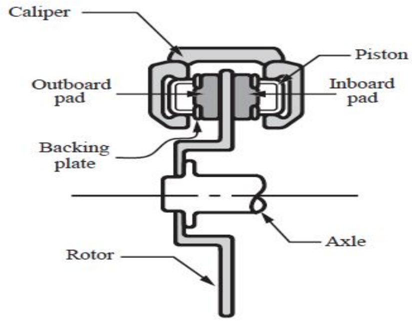

Fig.1 Components of Disc Brake

B. Classification of Disc Brake Squeal

Brake squeal has certain frequency. Brake squeal is also indicated by amplitudes of the vibrations of the brake disc in the micrometer range and the created sound pressure level (SPL). Depending on the various frequencies disc brake noise is classified in following categories

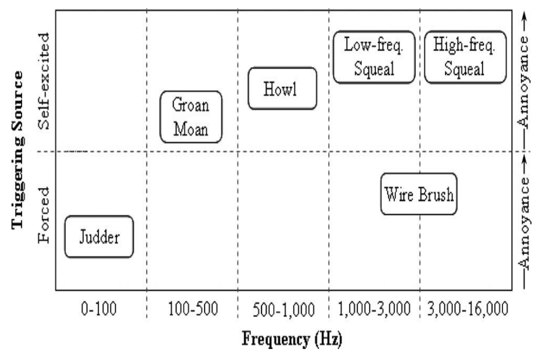

Fig.2 Classification of squeal

1) Low-frequency noise having range from 100Hz to 1,000 Hz
2) Low-frequency squeals having range from 1,000 Hz to 7,000 Hz
3) High-frequency squeals having range from 8,000 Hz to 16,000 Hz

The main function of brake is to reduce the speed of vehicle. Brake squeal does not affect the main function of brakes. Although it does not affect the braking performance, it is not acceptable because of high frequency noise emissions. Brake squeal can occur in all types of friction-based brake systems, for disc brakes used in the aircraft, automotive, transport and motorcycle industry and drum brakes used in the automotive and motorcycle industry, block brakes of trains. There are different types of disc brakes, still, they apply the basic working principle: With an application of brake pressure to the brake piston, the brake pads makes frictional contact with the brake disc rotating at an angular velocity mounted rigidly on the vehicle axle.

II. PARAMETERS STUDIED

A. Braking Pressure

Increase in braking pressure leads to a linear increase in the main unstable frequency. In some cases, the braking pressure leads to some other unstable frequencies, such as a pressure of 75 psi. This result concluded that for a high braking pressure the frequency where the noise occurs tends to be higher [1].It is found that with an increase in pressure, the value of the damping ratio is increased, so the squeal probability is increased. This is because larger hydraulic pressure increases more friction between the pads and the disc [2].

B. Various Modifications in Disc and Pad

Higher coefficient of friction increases squeal propensity. Shorter lining reduces the occurrence of squeal. Various contact angles of pin-disc are most reasoning to squeal. Higher damping in the brake system tends to reduce squeal [3]. Groove-textured surfaces with a specific dimensional parameter showed good properties in reducing and suppressing squeal. This happens because impact behavior between the ball and the edge of the groove-textured surface reduces the self-excited vibration of the friction system, and lowers the probability of generation of high frequency components of the acceleration [4].

C. Velocity

It is found that low sliding velocity is responsible for occurrence of squeal. Because, low velocity is able to excite limit cycle of large amplitude vibration due to the alternately varying direction of friction forces. There is decrease in the critical value of sliding velocity as coefficient of friction increases. The increase in the braking load increases the critical value of sliding velocity and affects the limit cycle of the vibration [5].As the angular velocity increases, the value of the damping ratio gradually decreases. This shows the value of the damping ratio varies with variation of the rotational velocity of the disc [2].

D. Coefficient of Friction

Disc brake squeal is caused mainly by friction-induced dynamic instability. The value of the damping ratio is decreases with a decrease of the coefficient of friction. It is also observed that with an increase in the friction coefficient, there is an increase in the instability of the system, thus an increase in the damping ratios. This shows that the squeal can be reduced or eliminated by reducing coefficient of friction between the pads and the disc. But, no doubt this reduces braking performance and is not a preferable method to apply [2].

E. Damping

Damping is applied to reduce vibrations. Increase in damping can be applied to reduce the amplitude of the limit cycle of the brake squeal [5].Multi-layered viscoelastic parts, called as shims, used to prevent squeal noises of automotive brake systems [6].

III.INTRODUCTION OF ASYMMETRY

It is found that rotor with cyclic symmetry with an angle less than 180° comes in friction contact has at least one double eigen frequency, which makes it susceptible to self-excited vibrations. This happens because adjacent modes have closer value of natural frequencies. By introduction of symmetry it is possible to increase difference of natural frequency between two adjacent modes. So in this paper we are going to break symmetry of disc by drilling of hole with varying radial distance and diameters.

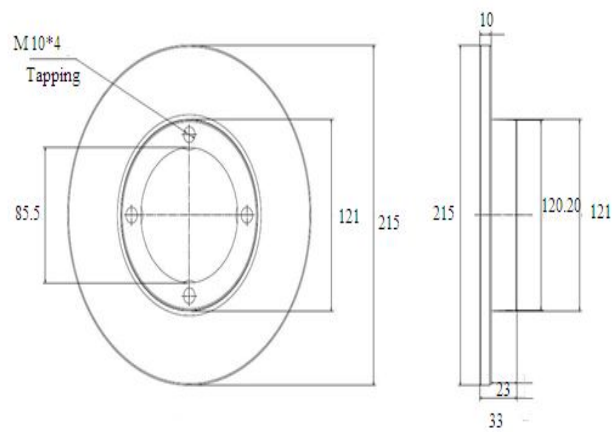

Fig.3 Details of disc rotor

For the study disc brake of Maruti-800 is selected. In most of the four wheel vehicles front wheel brakes are disc brakes. Details of disc are as above. All dimensions are in mm. Material used for the disc is Grey Cast Iron (GCI).

A. Modelling of Disc Brakes

Modelling of discs is done in CATIA-V5. For this realistic dimensions of disc are taken. There is modelling of four discs. One is original disc and other three are provided cyclic asymmetry. Asymmetry is introduced by drilling holes at 45° with varying radial distance. Various radial distances are 95 mm, 90 mm, 85 mm, 80 mm from centre of disc. Also three different diameter holes are drilled to three discs to introduce asymmetry. These diameters are selected from standard drill size 10 mm, 12 mm, 15 mm.

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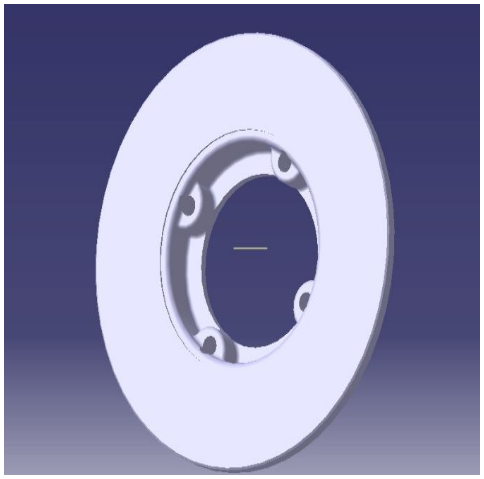

Fig.4 Original disc

While introducing cyclic asymmetry, diametrical opposite holes are kept at equal radial distance from the center of the disc. This is done to avoid dynamic unbalancing.

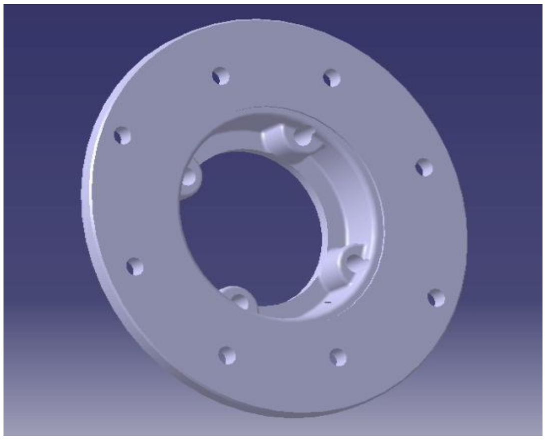

Fig. 5 Disc with holes

B. FEA of Disc Brake

Finite element analysis is done in ANSYS. Model is imported in ANSYS and modal analysis is done to find natural frequencies of the discs.

For original disc

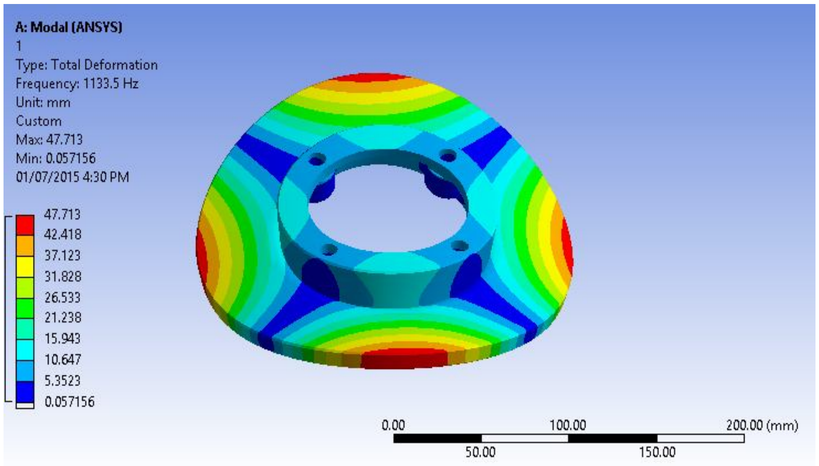

Fig.6 Original disc

TABLE I
Natural Frequencies For First 6 Modes

Mode no.	Natural Frequency(Hz)
1	1133.5
2	2179.1
3	2355.8
4	3064.1
5	3686.0
6	3689.0

For disc with 10 mm holes

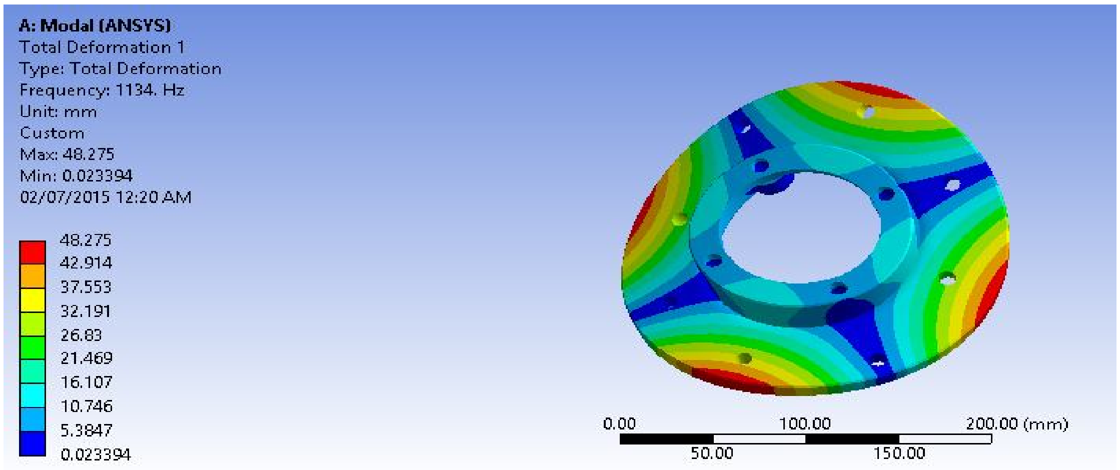

Fig.7 Disc with 10mm hole

TABLE III
Natural Frequencies For First 6 Modes

Mode no.	Natural Frequency(Hz)
1	1134
2	2162.4
3	2341.6
4	3053.1
5	3611.8
6	3681.9

For disc with 12 mm holes

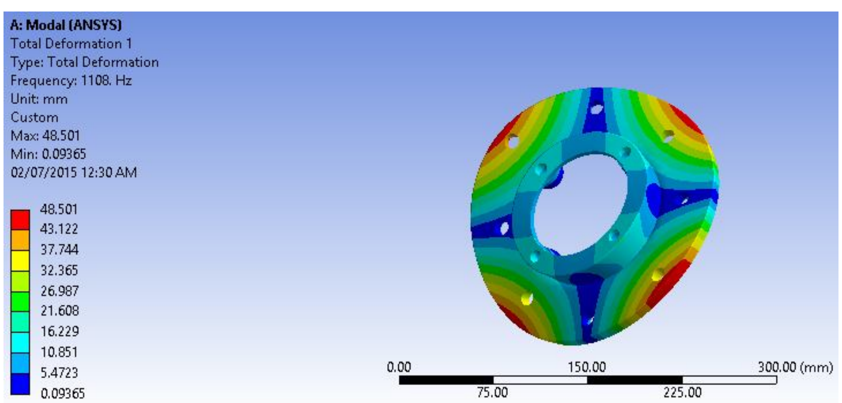

Fig. 8 Disc with 12 mm hole

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TABLE IV
Natural Frequencies For First 6 Modes

Mode no.	Natural Frequency(Hz)
1	1108
2	2154.7
3	2330.2
4	3015.9
5	3580.5
6	3682.8

For disc with 15 mm holes

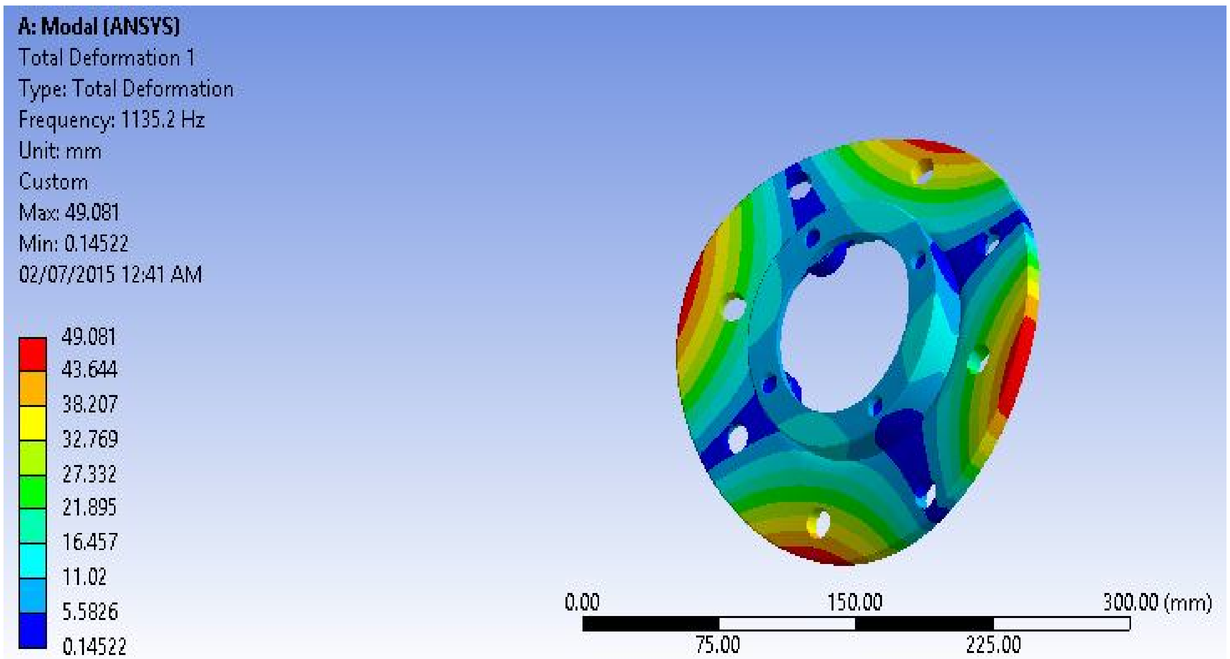

Fig.9 Disc with 15 mm hole

TABLE V
Natural Frequencies For First 6 Modes

Mode No.	Natural Frequency(Hz)
1	1135.2
2	2137.8
3	2316.3
4	3004.0
5	3017.9
6	3523.7

C. Drilling Operation on Discs

Drilling operation is done on three discs. Drills of standard sizes,10 mm, 12 mm, 15 mm is used as done in modeling. Drilling is done on varying radial distances from the center of the disc and angular distance between two drill holes is 45°.These varying radial distances are 95 mm, 90 mm,85 mm, and 80 mm. Drilling operation is carried on vertical axis drilling machine.

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Fig. 10 Vertical axis drilling m/c

Fig. 11 Drilling operation

D. Experimental Modal Analysis

Experimental modal analysis consists of exciter; analyzer data storage, display etc. The important part of the test system is the controller, or computer. It can be configured with various levels of memory, displays and data storage. The analyzer provides data acquisition and signal processing operations. It can be configured with several input channels, for force and response measurements, and with one or more excitation sources for exciters and driving shakers. Measurement functions such as windowing, averaging and Fast Fourier Transforms (FFT) computation are usually processed within the analyzer. Here impact hammer is used as exciter. Experimental modal analysis is done at Rajarambapu Institute of Technology, Sakharale. Specifications of components of FFT analyzer are as follows:

1) FFT Type 3050 – B - 040 4 channel input module 50 KHz make- Bruel & Kjaer.
2) Accelerometer - type 4514 frequency range 1Hz to 10 KHz.
3) Impact hammer – type 8206.

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Fig. 12 FFT Analyzer

For finding natural frequencies of discs were hanged with cord on supportive fixture. Then excitation is given by impact hammer. Accelerometer is localized at behind the position of excitation.

Fig. 13 Accelerometer location

Fig. 14 Free-free condition modal analysis

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Fig. 15 FFT analyzer setup

Natural frequency obtained from experimental modal analysis compared with natural frequency obtained from ANSYS.

E. Graphs obtained from FFT

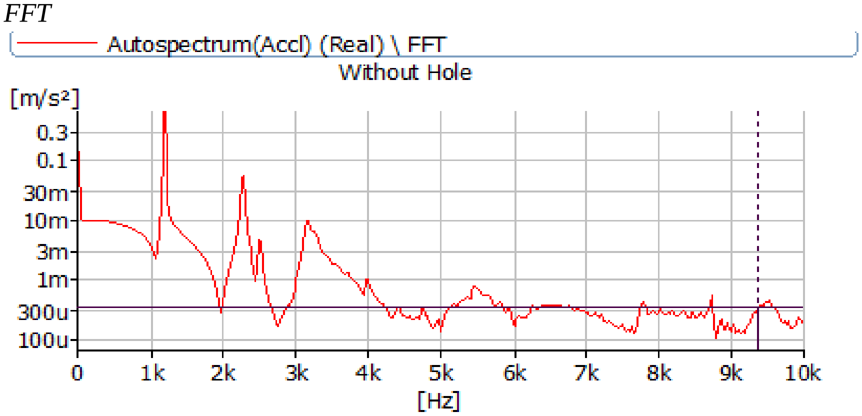

Fig. 16 Acceleration v/s Natural frequency for original disc

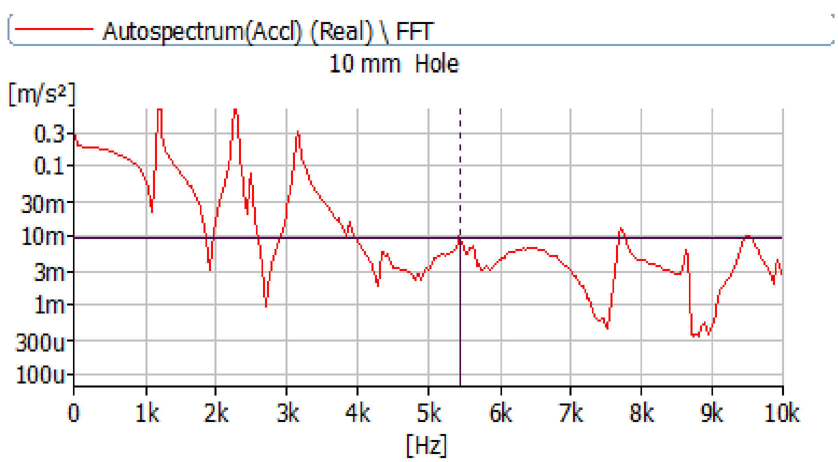

Fig. 17 Acceleration v/s Natural frequency for 10 mm holes disc

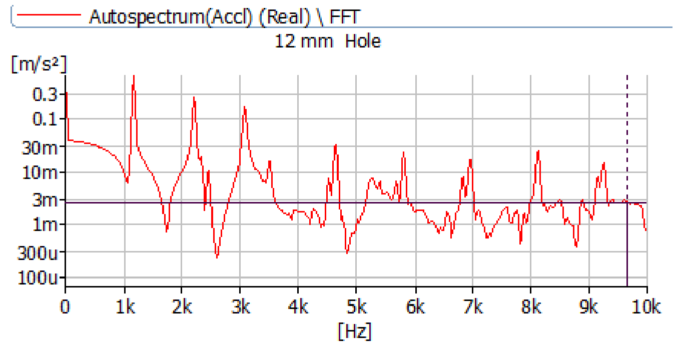

Fig. 18 Acceleration v/s Natural frequency for 12 mm holes disc

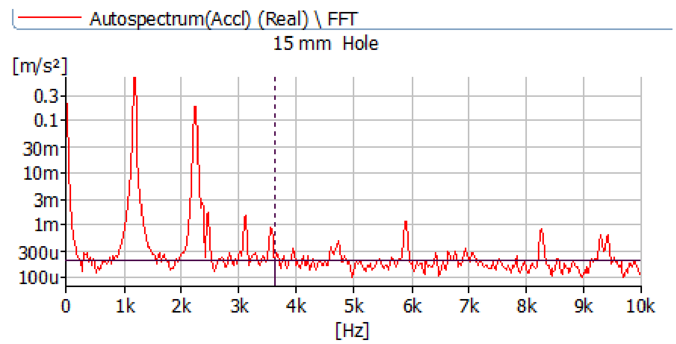

Fig. 19 Acceleration v/s Natural frequency for 15 mm holes disc

F. Comparison of Experimental and ANSYS Results

TABLE V

COMPARISON OF ORIGINAL DISC

Mode No.	Natural frequency(ANSYS) Hz	Natural frequency(Expt.) Hz	% Deviation
1	1133.5	1175	3.66
2	2179.1	2275	4.4
3	2355.8	2500	6.12
4	3064.1	3150	2.8
5	3686	3700	0.37
6	3689	3975	7.75

TABLE VI

COMPARISON OF 10 MM HOLES DISC

Mode No.	Natural frequency(ANSYS) Hz	Natural frequency(Expt.) Hz	% Deviation
1	1134	1200	5.82
2	2162.4	2275	5.2
3	2341.6	2500	6.76
4	3053.1	3150	3.17
5	3611.8	3750	3.82
6	3681.9	3900	5.92

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Table VII
Comparison Of 12 Mm Holes Disc

Mode No.	Natural frequency(ANSYS) Hz	Natural frequency(Expt.) Hz	% Deviation
1	1108	1150	3.79
2	2154.7	2200	2.10
3	2330.2	2325	0.22
4	3015.9	3075	1.95
5	3580.5	3500	2.24
6	3682.8	3850	4.54

TABLE VIII
Comparison Of 15 Mm Holes Disc

Mode No.	Natural frequency(ANSYS) Hz	Natural frequency(Expt.) Hz	% Deviation
1	1135.2	1175	3.50
2	2137.8	2225	4.07
3	2316.3	2350	1.45
4	3004.0	3125	4.02
5	3017.9	3275	8.51
6	3523.7	3400	3.51

It is observed from the comparison of the results obtained from the ANSYS and experimentation, there is less percentage of deviation. After comparing FEA and experimental results for natural frequency ,average percentage deviation for original disc is 4.18%, for 10 mm holes disc is 5.11%, for 12 mm holes disc is 2.47%, for 15 mm holes disc is 5.02%.

IV.RESULT ANALYSIS

When values of natural frequency observed following results are obtained.

TABLE IX
Difference Between Adjacent Modes Of Original Disc

Mode No.	Natural frequency(Expt) Hz	Difference in two adjacent modes Hz
1	1175	1100
2	2275	225
3	2500	650
4	3150	550
5	3700	275
6	3975	

TABLE X
Difference Between Adjacent Modes Of 10 Mm Holes Disc

Mode No.	Natural frequency(Expt) Hz	Difference in two adjacent modes Hz
1	1200	1075
2	2275	225
3	2500	650
4	3150	600
5	3750	150
6	3900	

TABLE XI
Difference Between Adjacent Modes Of 12 Mm Holes Disc

Mode No.	Natural frequency(Expt) Hz	Difference in two adjacent modes Hz
1	1150	1050
2	2200	125
3	2325	750
4	3075	425
5	3500	350
6	3850	

TABLE XII
Difference Between Adjacent Modes Of 15 Mm Holes Disc

Mode no.	Natural frequency(Expt) Hz	Difference in two adjacent modes Hz
1	1175	1050
2	2225	125
3	2350	775
4	3125	150
5	3275	125
6	3400	

It is observed that, there is increase in difference in natural frequency at higher modes for 10 mm and 12 mm holes disc. Also from the graphs it is observed that, for 10 mm holes disc curve is without fluctuations at higher modes. For 15 mm holes disc there are much lower values of acceleration are observed. That shows with 15 mm holes disc, there is avoidance of higher accelerations of disc, which is also useful for avoidance of squeal. So from result analysis, it is possible to avoid high frequency squeal at higher modes by introduction of asymmetry.

V. CONCLUSION

For elimination or reduction of disc brake squeal most important parameter is structural modifications in brake disc. Modifications are done in the disc rotor by introduction of asymmetry. There is increase in difference in natural frequencies of adjacent modes. Specially, these differences are observed at higher modes. So it is possible to reduce high frequency squeal at higher modes by introduction of asymmetry. This can be implemented further at commercial level. So by reducing squeal there will be increase in comfort level to passengers and surrounding environment from squeal. Also there will be reduction in high warranty costs occurring due to the brake squeal.

VI. ACKNOWLEDGEMENT

It gives me great pleasure to present this dissertation paper on "Study and Analysis of Disc Brake to Reduce Disc Brake Squeal". I am very thankful to Prof. Shelge S.V. for his valuable guidance and encouragement. His inspiration and guidance constantly helped me at various stages of dissertation work.

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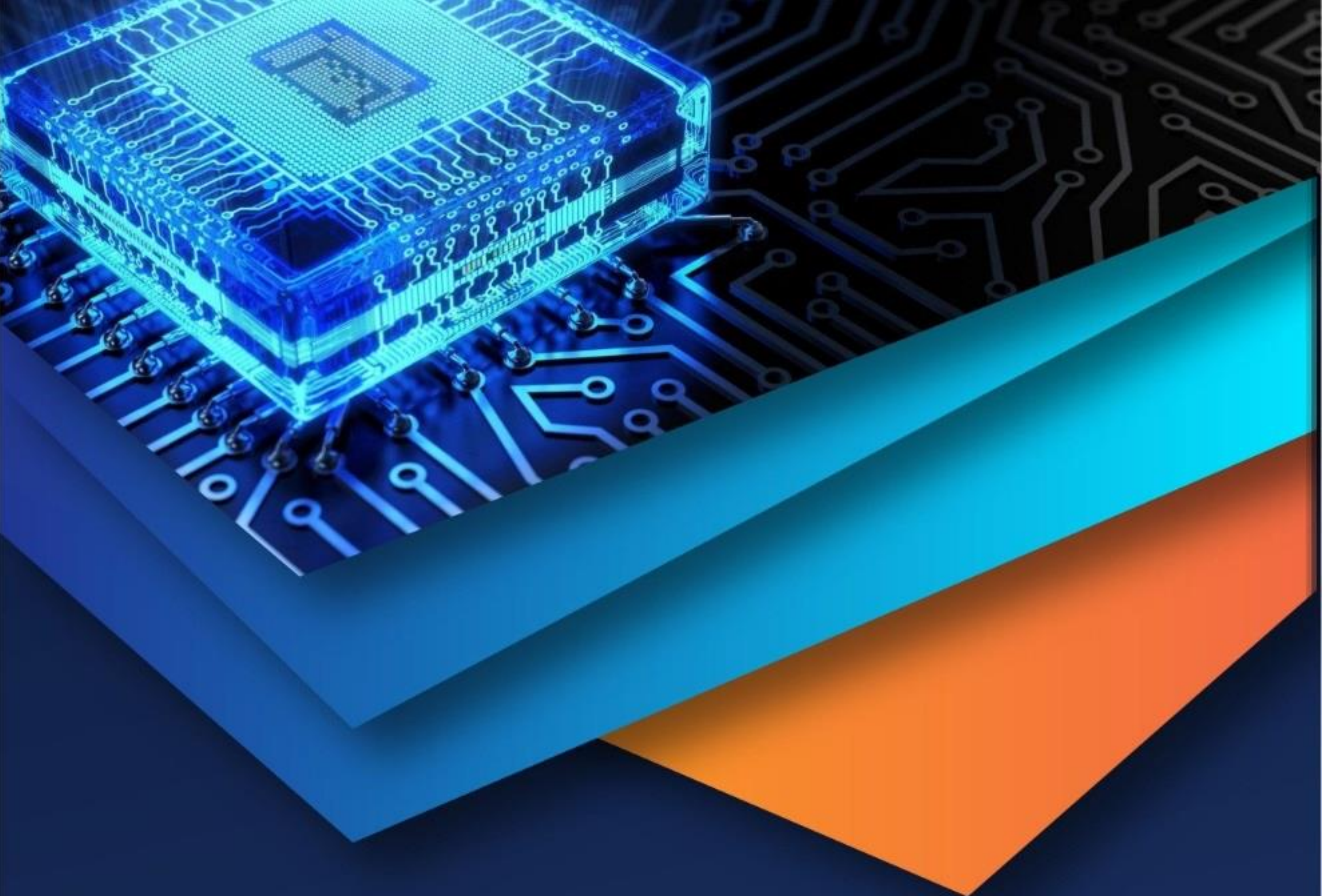

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