Finite Element Modelling and Analysis of Brake Squeal

Rushikesh D. Savant[1], Shekhar Y. Gajjal[2]

[1]*NBN Sinhgad School of Engineering, Pune, India.*

[2]*Head, P. G. Mechanical Design Engineering, NBN Sinhgad School of Engineering, Pune, India.*

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.**

This work presents 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 eigenfrequencies of unstable modes are obtained. Analysis is performed by varying the coefficient of friction and outer diameter of disc-pad assembly. Increasing friction coefficient has no desirable effect on squeal frequency while squeal propensity decreases as the outer diameter of disc is increased.

Keywords - Brake Squeal, Stick-slip phenomenon, Mode coupling, FEM, Perturbed modal analysis.

I. INTRODUCTION

During the application of the brake 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 nose is categorised as Low frequency Squeal (1000-5000 Hz) and High frequency squeal (5000-16000 Hz).

A. Brake noise generation mechanisms

There are two theories that attempt to explain why this phenomenon occurs.

1) Mode coupling theory

If two vibration modes are close to each other in the frequency range and have similar characteristic, they may merge if the coefficient of friction between the pad and the disc increases. When these modes merge at the same frequency called couple frequency, one of them become unstable producing noise. This noise is called as squeal. The variable friction forces caused by variable normal forces are sources for squeal.

2) Stick-slip mechanism

Stick-slip vibration is self-excited oscillation induced by dry friction. The resistance against the beginning of the motion from the state of the rest is called a stick mode while resistance against of an existence motion is called a slip mode. Stick-slip mode can be introduced by the difference between the coefficient of static and kinetic friction. A variable friction coefficient with respect to sliding velocity between pads and rotor provides the energy source for the brake squeal.

II. FINITE ELEMENT MODELLING OF THE DISC-PAD ASSEMBLY

A. Solid model of disc pad assembly

The solid model of disc pad assembly is modelled using CATIA V5 R20 software tool. The disc has a thickness of 10 mm and the brake pads have a thickness of 15 mm. The inner diameter of the disc is 250 mm and outer diameter is of 350 mm.

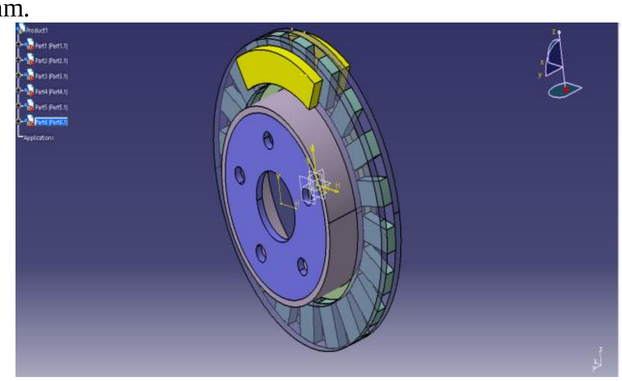

Fig. 1 Disc-pad assembly solid model

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B. Material properties and boundary conditions

Linear elastic isotropic materials are assigned to all the components of the disc-pad assembly.

Table I
Material Properties

Young's Modulus	2.0 E+11 Pa
Density	7850 Kg/m^3
Poisson's Ratio	0.3

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.

C. FE mesh generation

The sweep method is used to generate a hexahedral dominant mesh of the brake system assembly. Brake discs and pads are meshed with 20-node structural solid element SOLID186 and 10-node structural solid element SOLID187 with uniform reduced-integration element technology. CONTA174 (3-D, 8-node surface to surface contact) elements are used to define the contact surface and TARGE170 (3-D target segment) elements are used to define the target surface. The target elements are defined on the disc surface and the contact elements are defined on the pad surface. The brake disc-pad assembly is meshed with total 0f 60351 nodes and 11473 elements.

III. FINITE ELEMENT ANALYSIS OF DISC-PAD ASSEMBLY

A. Modal analysis

Modal analysis is used to determine vibration characteristics such as natural frequencies and mode shapes of a structure or a machine component. The frequencies obtained from the modal solution have real and imaginary parts due to 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.

B. Brake squeal analysis

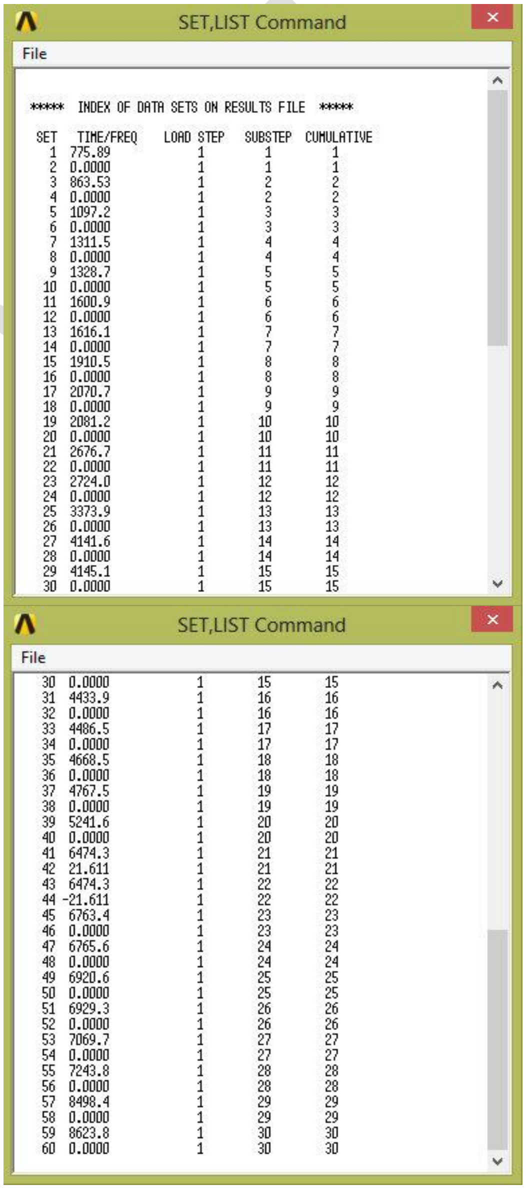
Disc brake squeal is an occurrence of dynamic instability that manifests itself at one or more of the natural frequencies of a disc brake system. This analysis is concerned with the prediction of the natural frequencies at which brake squeal occurs.

C. Linear non-prestressed modal analysis

A linear non-prestressed modal analysis is effective when large deflection or stress stiffening effects are not critical and prestress effect is not included. This method involves a single linear QR damped or unsymmetric eigen solver. This method is less time consuming, as the Newton-Raphson iterations are not required.

Results

Total 30 modes are extracted by using unsymmetric eigensolver. The imaginary frequency reflects the damped frequency and the real frequency indicates whether the mode is stable or not. Linear non-prestressed modal analysis predicts unstable modes 21 and 22 at 6474.25 Hz.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***** INDEX OF DATA SETS ON RESULTS FILE *****

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2	0.0000	1	1	1
3	863.53	1	2	2
4	0.0000	1	2	2
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8	0.0000	1	4	4
9	1328.7	1	5	5
10	0.0000	1	5	5
11	1600.9	1	6	6
12	0.0000	1	6	6
13	1616.1	1	7	7
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16	0.0000	1	8	8
17	2070.7	1	9	9
18	0.0000	1	9	9
19	2081.2	1	10	10
20	0.0000	1	10	10
21	2676.7	1	11	11
22	0.0000	1	11	11
23	2724.0	1	12	12
24	0.0000	1	12	12
25	3373.9	1	13	13
26	0.0000	1	13	13
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29	4145.1	1	15	15
30	0.0000	1	15	15

SET,LIST Command
File

30	0.0000	1	15	15
31	4433.9	1	16	16
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35	4668.5	1	18	18
36	0.0000	1	18	18
37	4767.5	1	19	19
38	0.0000	1	19	19
39	5241.6	1	20	20
40	0.0000	1	20	20
41	6474.3	1	21	21
42	21.611	1	21	21
43	6474.3	1	22	22
44	-21.611	1	22	22
45	6763.4	1	23	23
46	0.0000	1	23	23
47	6765.6	1	24	24
48	0.0000	1	24	24
49	6920.6	1	25	25
50	0.0000	1	25	25
51	6929.3	1	26	26
52	0.0000	1	26	26
53	7069.7	1	27	27
54	0.0000	1	27	27
55	7243.8	1	28	28
56	0.0000	1	28	28
57	8498.4	1	29	29
58	0.0000	1	29	29
59	8623.8	1	30	30
60	0.0000	1	30	30

Fig. 2 The mode shape plots for unstable modes

The mode shape plots for the unstable mode suggest that the bending mode of the pads and the disc have similar characteristics. These bending modes couple due to friction and produce a squealing noise. When linear non-prestressed modal analysis is performed the obtained mode shape plots for unstable modes are shown below.

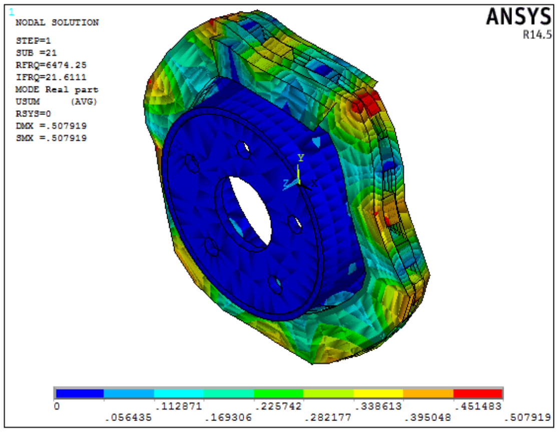


Fig. 3 Mode shape for unstable mode 21

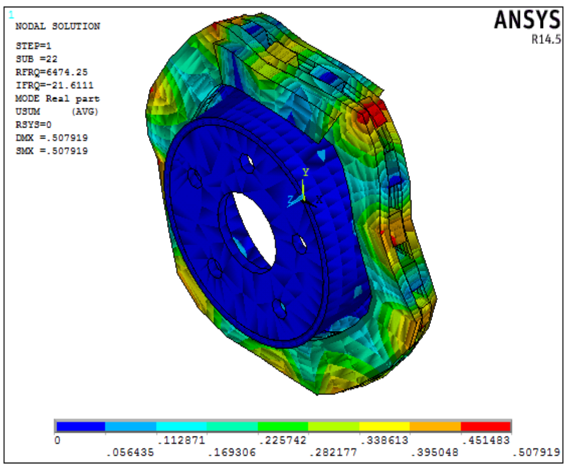


Fig. 4 Mode shape for unstable mode 22

D. Full non-linear perturbed modal analysis

The squeal analysis method used previously is less accurate and includes no prestress. A full nonlinear perturbed modal analysis is the most accurate method to solve brake squeal problem than linear non-prestressed modal analysis. This method uses nonlinear static solutions to both establish the initial contact and compute the sliding contact. This method includes prestress effects. The displacement on the brake pad surfaces where the pressure loading is applied is constrained in all directions except axial one i.e. along Z axis.

1) Parametric study with increasing the outer diameter of the disc

A parametric study is performed on the disc-pad model using a full nonlinear perturbed modal solution by increasing the outer diameter of the disc and simultaneously the dimensions of the pad are increased in the range of 4% (14 mm) upto 120%. With increasing outer diameter of disc, the dimensions of the pad also varied accordingly.

Mode shapes obtained for the full non-linear perturbed modal analysis when the outer diameter of the disc is increased are shown below.

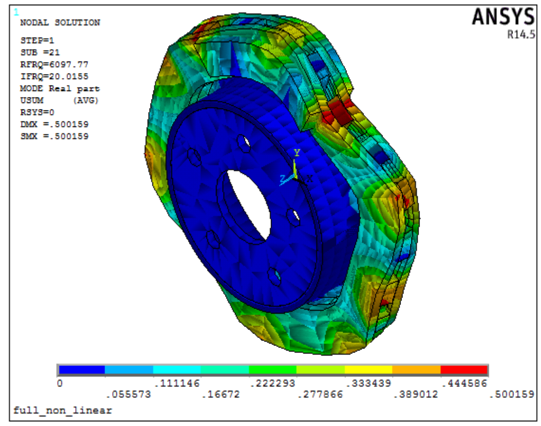


Fig. 5 Mode shape for unstable mode 21 when D_o=364

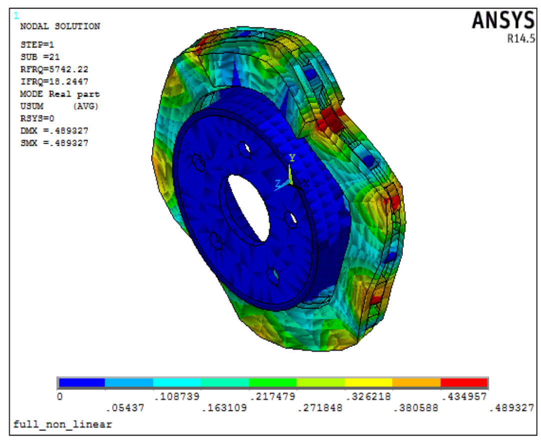


Fig. 6 Mode shape for unstable mode 21 when D_o=378

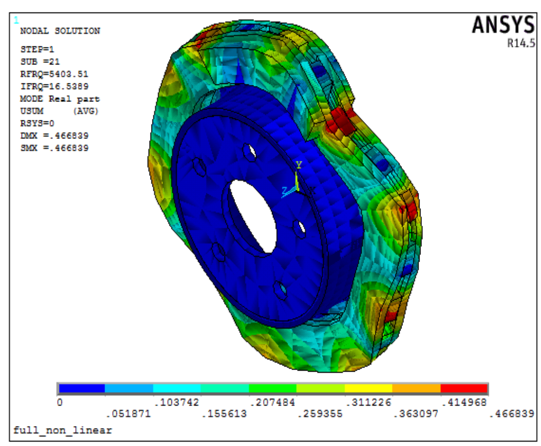


Fig. 7 Mode shape for unstable mode 21 when D_o=392

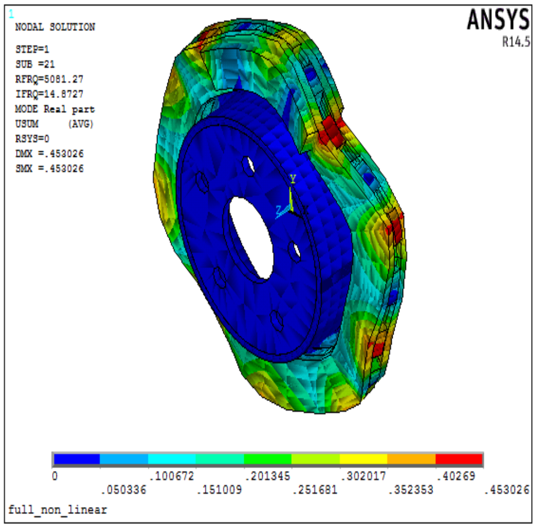


Fig. 8 Mode shape for unstable mode 21 when D_o=406

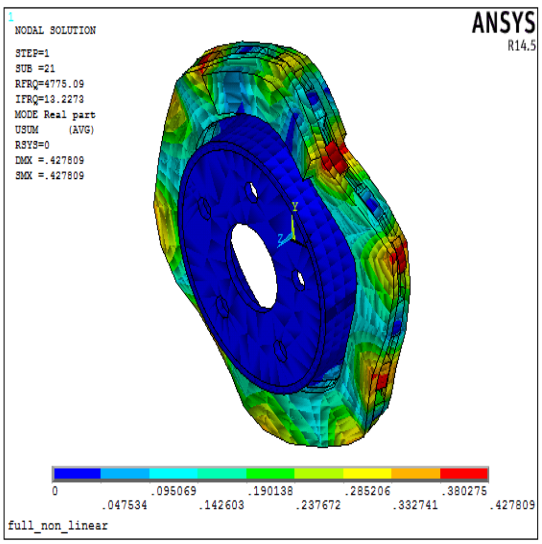


Fig. 9 Mode shape for unstable mode 21 when D_o=420

Results

The eigenfrequencies obtained by increasing the outer diameter of disc are listed below.

Table II
Eigen frequencies for mode 21 and 22 with increasing outer diameter of disc

D_o	Mode 21		Mode 22	
	RFRQ (Hz)	IFRQ (Hz)	RFRQ (Hz)	IFRQ (Hz)
364	6097.77	20.0155	6097.77	-20.0155
378	5742.22	18.2447	5742.22	-18.2447
392	5403.51	16.5389	5403.51	-16.5389
406	5081.27	14.8727	5081.27	-14.8727
420	4775.09	13.2273	4775.09	-13.2273

As the size of the disc and pad is increased, the real eigenfrequency decreases linearly given by equation y= -23.616x + 14678, where, y = real eigenfrequency and x = coefficient of friction.

For mode 21 imaginary frequency decreases, while for mode 22 imaginary frequency increases linearly. The relationship is given by equation y = ±0.1211 ±64.035.

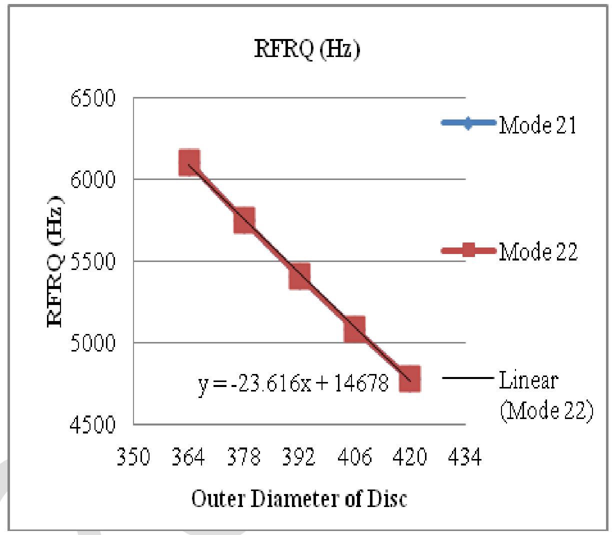


Fig. 10 Effect of increase of outer diameter of disc on real eigenfrequency

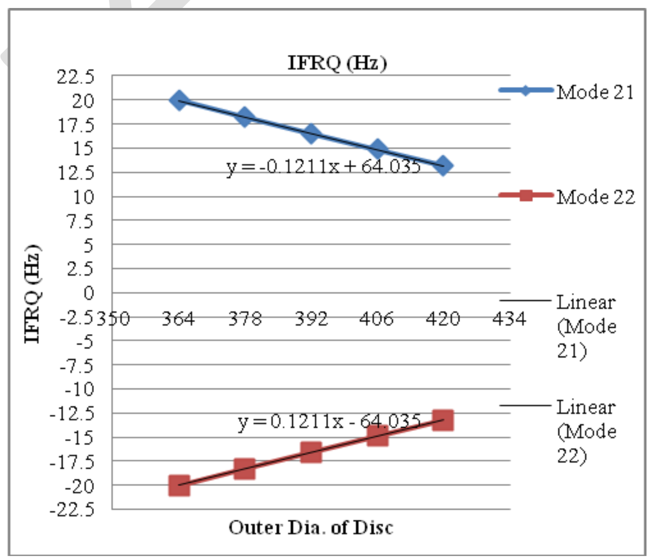


Fig. 11 Effect of increase in outer diamater of disc on imaginary eigenfrequency.

2) Parametric study with increasing the friction coefficient

The role of friction is to couple the different vibrating modes of sliding components. The effect of friction coefficient on the squeal was analysed by applying a range of friction coefficient values from 0.0 to 0.3 with an increment of 0.05.

Mode shapes obtained for full perturbed modal analysis when coefficient of friction is increased are shown below.

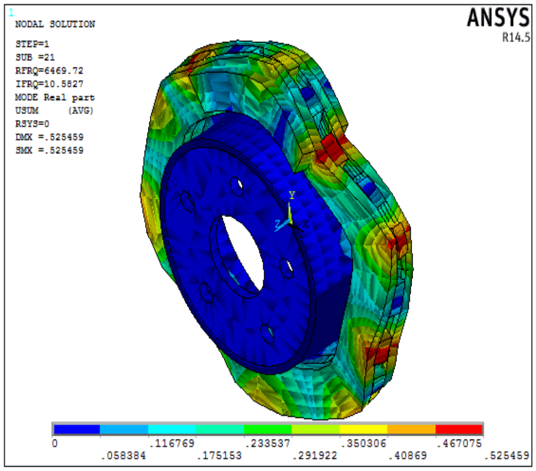


Fig. 12 Mode shape for unstable mode 21 when μ=0

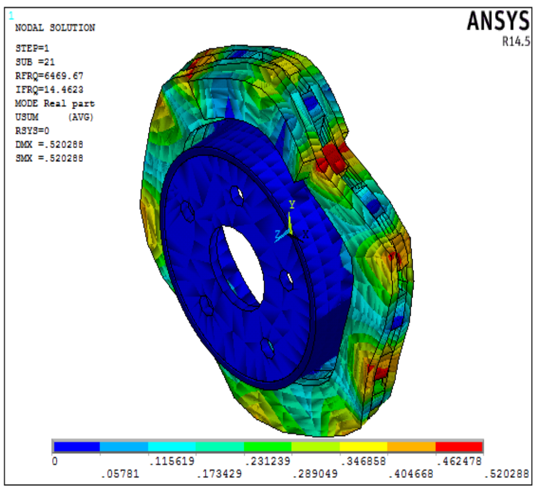


Fig. 13 Mode shape for unstable mode 21 when μ=0.1

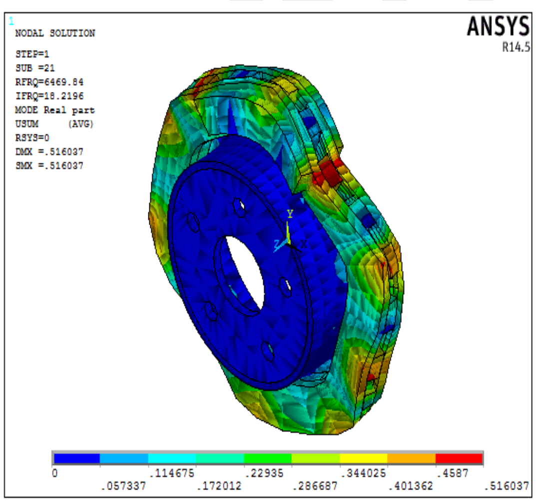


Fig. 14 Mode shape for unstable mode 21 when μ=0.2

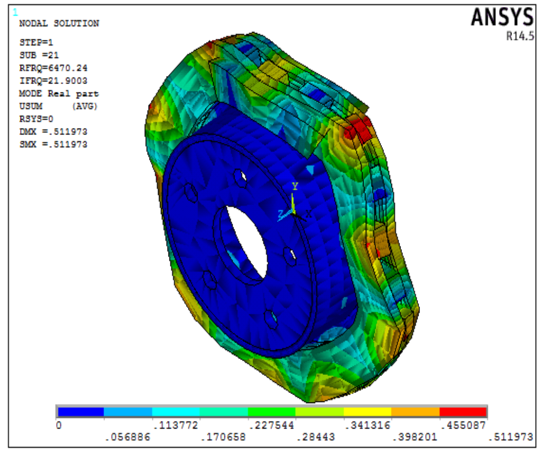


Fig. 15 Mode shape for unstable mode 21 when μ=0.3

Results

The eigenfrequencies obtained by varying coefficient of friction between the pad and the disc are listed in the table III.

Table III
Eigen frequencies for mode 21 and 22 with increasing friction coefficient

μ	Mode 21		Mode 22	
	RFRQ (Hz)	**IFRQ (Hz)**	**RFRQ (Hz)**	**IFRQ (Hz)**
0	6469.72	10.5827	6469.72	-10.5827
0.05	6469.67	12.5436	6469.67	-12.5436
0.1	6469.67	14.4623	6469.67	-14.4623
0.15	6469.73	16.3521	6469.73	-16.3521
0.2	6469.83	18.2196	6469.83	-18.2196
0.25	6470.01	20.0685	6470.01	-20.0685
0.3	6470.24	21.9003	6470.24	-21.9003

The real frequency first decreases when friction coefficient is 0.1 and then goes on increasing as friction coefficient increases. The relationship is given by the equation $y = 11.333x^2 - 1.6857x + 6469.7$, where y = real eigenfrequency and x = imaginary eigenfrequency.

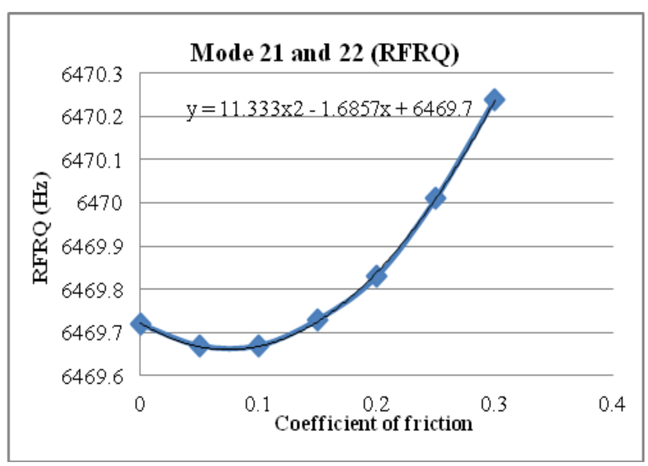
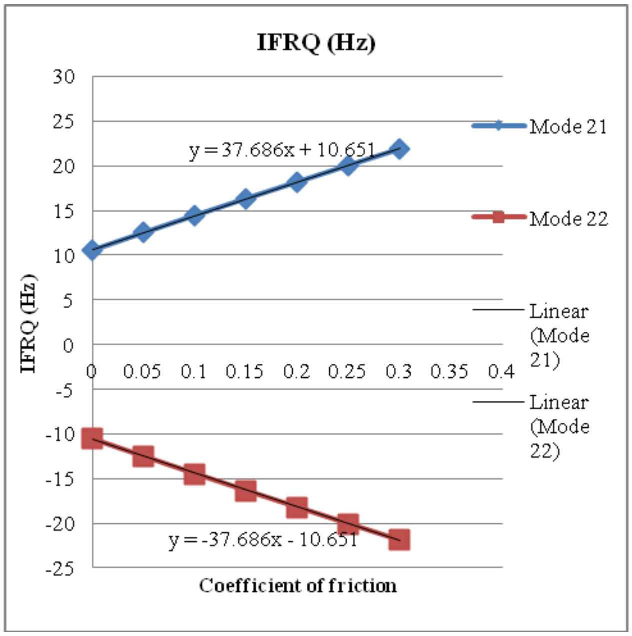


Fig. 16 Comparison of friction coefficient with real eigenfrequency

For mode 21, imaginary frequency increases and for mode 22, imaginary frequency decreases linearly as the coefficient of friction increases, given by equation y = ±37.686 ± 10.561.



ig. 17 Comparison of friction coefficient with imaginary eigenfrequency

IV. EXPERIMENTAL VALIDATION

The experimental validation for full nonlinear perturbed modal analysis of brake squeal prediction is carried out by NVH brake testing machine. NVH test standard SAE J2521 is used for experimental testing. The SAE J2521 procedure is applicable to high frequency squeal noise occurrences for on road cars and passenger trucks.

The test setup consists of single ended inertia dynamometer, a semi-anechoic chamber, environmental controlled cooling and auto spectrum mic.

Fig. 18 NVH brake testing bench

Result

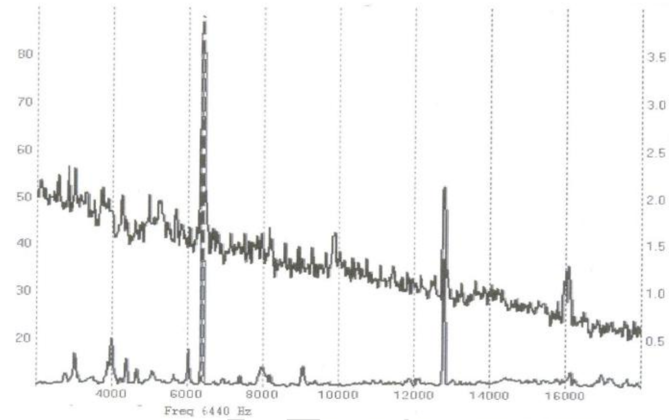


Fig. 19 NVH brake testing machine output

Not every up and down on the plot can be considered as a squeal event. From the spectrum shown in fig. obtained after testing, it is observed that at frequency 6440 Hz, there is distinctive peak i.e. squeal occurs. The unstable mode exists at frequency 6440 Hz which causes disc-pad assembly to squeal.

V. RESULTS AND DISCUSSION

Full non-linear perturbed modal analysis of the disc-pad assembly is performed by varying the outer diameter of disc and the friction coefficient between disc and pad. It is found that modes 21 and 22 are unstable and causes squeal.

Following table shows comparison of experimental and FEA results.

Table IV

Comparison of Experimental and FEA results

Experimental Result	**FEA (ANSYS) Result**	**% Error**
6440 Hz	6470.24 Hz	0.4674

The FEA (ANSYS) result gives squeal frequency at 6470.24 Hz while results experimentally show that the squeal occurs at the frequency 6440 Hz. It is quite near to FEA results. Hence, error of FEA results is within 1% i.e. 0.4674%. This shows that the squeal problems can be solved by ANSYS 14.5 with less error.

VI. CONCLUSIONS

1. As the outer diameter of disc is increased, real eigenfrequency decreases linearly for both modes 21 and 22.
2. For mode 21, imaginary eigenfrequency decreases and for mode 22, imaginary eigenfrequency increases as the outer diameter of the disc is increased.
3. When the coefficient of friction is increased from 0 to 0.1, the real eigenfrequency decreases, further increase in

coefficient of friction real eigenfrequency increases again for both modes 21 and 22.

4. For mode 21, imaginary eigenfrequency increases and for mode 22, imaginary eigenfrequency decreases linearly as the friction coefficient increased.
5. Finite Element Analysis result error is 0.4674% which is within the acceptable limit of 1%.

FUTURE SCOPE

1. Further brake squeal analysis can be carried out by variations in structural design.
2. Squeal analysis can be performed by varying parameters such as brake pressure, brake temperature, wear etc.
3. The materials of the assembly can be optimized by composite materials.

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