

Research Article

# Complex Eigen Value Analysis of Automotive Disc Brake for Squeal Reduction

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## Abstract

The main aim of present work lies in the fact that squeal of disc brake depends on many parameters. Evaluating the brake system using traditional studies that is by changing one factor at a time is not sufficient. In order to improve design of brake components there is a necessity of a new approach to investigate the effects of combination of several parameters. In this work Finite Element Technique which is widely used in engineering fields is preferred for analysis on brake squeal. This work deals with improvising 3 dimensional model of finite element model of brake system which includes assembly of disc brake and steering knuckle. Complex Eigen Value Analysis (CAE) is used to predict unstable frequencies ranging from 1 kilo hertz to 20 kilo hertz. Different materials were implemented for individual components of brake system to reduce unstable frequencies and to shift squeal frequencies.

**Keywords:** Complex Eigen Value Analysis, stability analysis.

## 1. Introduction

The most significant safety aspect of an automobile is its brake system, which must slow the vehicle quickly and reliably under varying conditions. There are many types of brake systems that have been used since the inception of the motor car, but the working principle is same. The function of system is to decelerate the vehicle by transforming the kinetic energy of the vehicle into heat by friction, and this heat must be effectively and efficiently dissipating to the surroundings by the brake components.

There are two types of disc brakes, fixed caliper and floating caliper. The fixed caliper usually contains two or more pistons that act directly on both the inner and outer pads. In order to increase the braking power two pistons or more can be used. The pistons attempt to stop the rotating disc by exerting an equal amount of pressure against each side of the rotor as the each piston moves with equal pressure. The main limitation of the fixed caliper is requiring a rotor position which is not compatible with the geometric needs of the most modern front suspensions.

Nowadays, the floating caliper is commonly used on the majority of automotive brake systems. It consists of a rotor that is rigidly mounted on the axle hub and therefore rotates with the automobile's wheel through drive axles, a floating caliper-piston assembly where

the piston slides inside the caliper, which is attached to an anchor bracket on the vehicle suspension system, and a pair of brake pads. The caliper is free to float laterally on its mounting pins. When hydraulic line pressure is applied, the piston is moved frontwards to press the inner pad against the disc and simultaneously the outer pad is pressed by the caliper against the disc. The hydraulic pressure is converting into an applied force that presses the pad against the rotor. Due to this friction is induced for applying brakes.

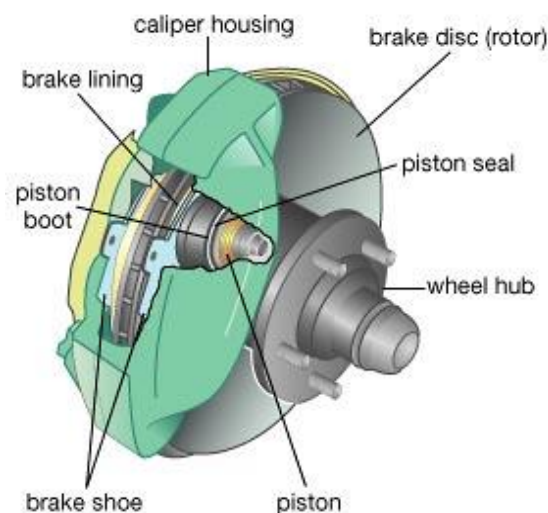


Fig.1 Disc Brake Assembly

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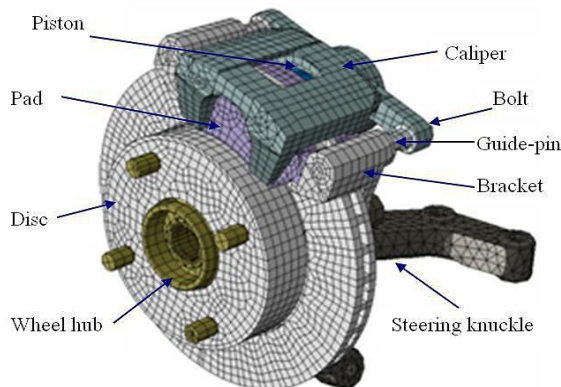
**Table 1** Material properties of disc brake component

Components	Density	Young's modulus(GPa)	Poisson's ratio
Rotor	7154	125	0.23
Friction material	2040	2.6	0.34
Back plate	7850	210	0.3
Caliper	7000	171	0.27
Anchor bracket	7050	166	0.27
Steering knuckle	7625	167	0.28
Wheel hub	7400	168	0.28
Piston	8018	192	0.27
Gudgeon pin	2850	72	0.3
Bolt	7860	210	0.3

**2. Research Methodology**

1. Development of an improved FE model of the whole disc brake corner, as an extension to earlier FE disc brake models.
2. Applying the boundary conditions.
3. Extracting the natural frequencies of individual brake components.
4. Suppression of the squeal noise using several material modifications of the disc brake components.
5. Validating the results.

**3. Finite Element Analysis of Brake Model**



**Fig.2** Meshed Model of brake system

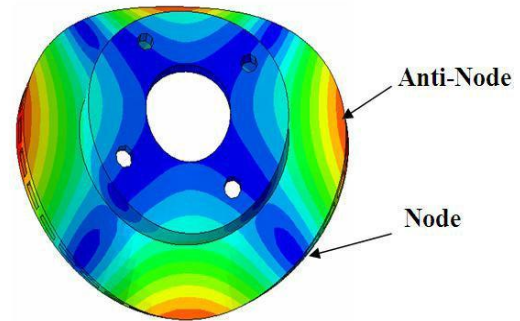
The disc brake system consists of many individual components.

**Table 2** Type and No. of elements for individual components

Disc brake components	Types of elements	No. of elements
Disc	C3D8	2428
Friction material	C3D8	340
Back plate	C3D8	252
Caliper	C3D8	2586
Anchor bracket	C3D8	968
Steering knuckle	C3D8	1720
Wheel hub	C3D8	422
Piston	C3D8	556
Gudgeon pin	C3D8	79
Bolt	C3D8	66

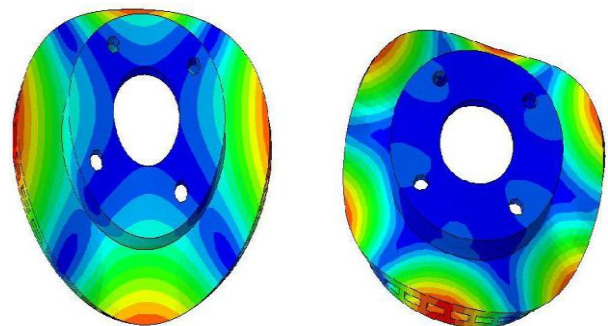
**3.1 Modal Analysis of individual brake components**

Finite element modal analysis is performed on the individual disc brake components for natural frequencies and mode shape. The results are presented in the form of displacement contour to indicate maximum anti-node and minimum node amplitudes.

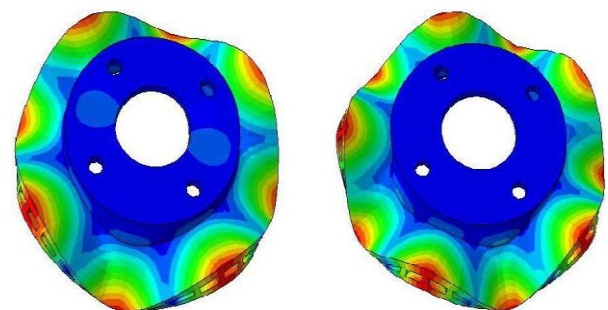


**Fig.3** Vibratory mode of brake component

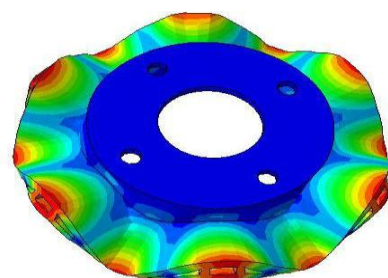
**3.1.1 Natural frequencies and mode shape of Rotor**



**Fig.4** Mode shape at 1453 Hz and 3225 Hz

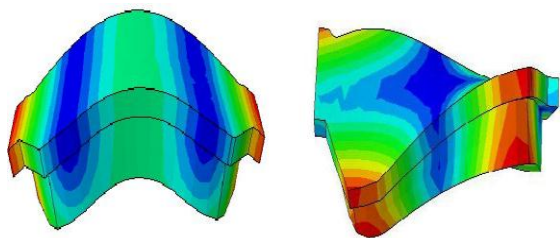


**Fig.5** Mode shape at 5062 Hz and 7067 Hz

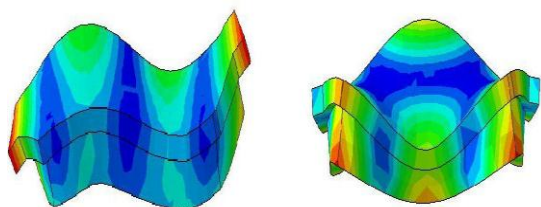


**Fig.6** Mode shape at 9170 Hz

3.1.2 Natural frequencies and mode shape of Brake pad

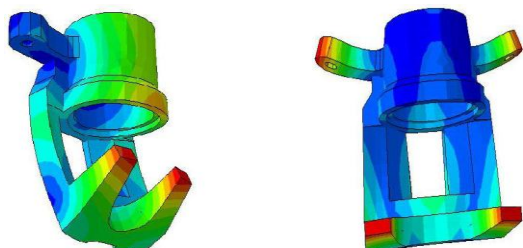


**Fig.7** First bending mode at 2889 Hz and twisting mode at 4460 Hz

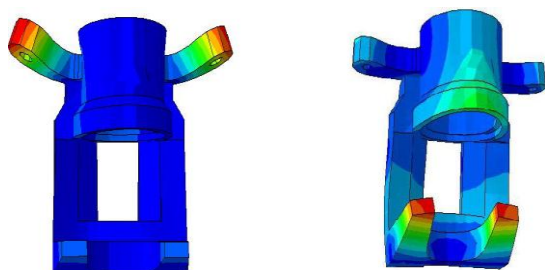


**Fig.8** Second bending mode at 6735 Hz and twisting mode at 8976 Hz

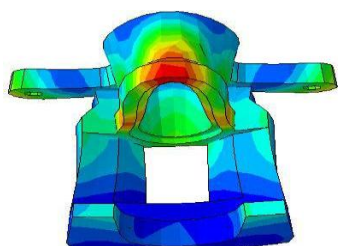
3.1.3 Natural frequencies and mode shape of Brake caliper



**Fig.9** Mode shape at 2293 Hz and 3964 Hz

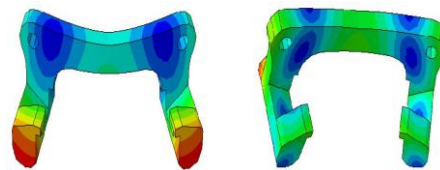


**Fig.10** Mode shape at 5667 Hz and 6587 Hz

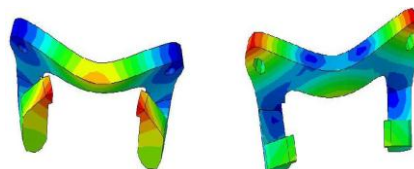


**Fig.11** Mode shape at 8221 Hz

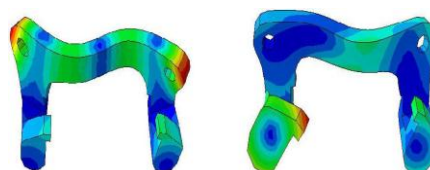
3.1.4 Natural frequencies and mode shape of Anchor bracket



**Fig.12** Mode shape at 880 Hz and 1755 Hz

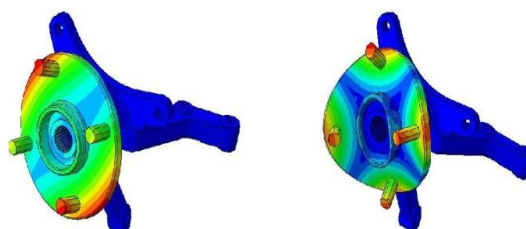


**Fig.13** Mode shape at 3164 Hz and 4680 Hz

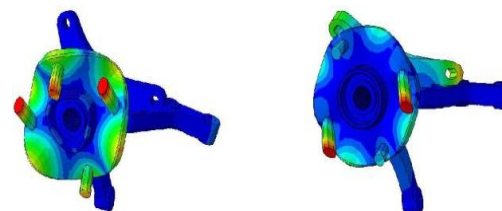


**Fig.14** Mode shape at 7533 Hz and 9262 Hz

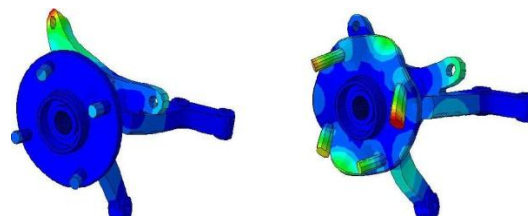
3.1.5 Natural frequencies and mode shape of Knuckle assembly



**Fig.15** Mode shape at 1211 Hz and 2242 Hz



**Fig.16** Mode shape at 4421 Hz and 6389 Hz



**Fig.17** Mode shape at 7992 Hz and 8665 Hz

3.1.6 Natural frequency and mode shape of piston

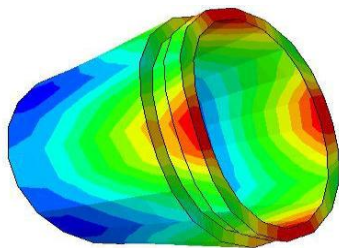


Fig.18 Mode shape at 7392 Hz

3.1.7 Natural frequency and mode shape of guide pin and bolt

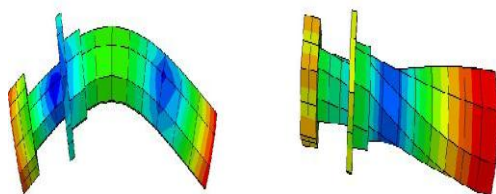


Fig.19 Mode shape at 4709 Hz and 9960 Hz

Table 3 modes and frequency range for individual components of braking system

Components	Mode	Frequency (Hz)
Rotor	1	1462
	2	3234
	3	5052
	4	7076
	5	9168
Anchor bracket	1	876
	2	1760
	3	3168
	4	4678
	5	7536
	6	9259
Caliper	1	2290
	2	3959
	3	5670
	4	6589
	5	8219
Brake pad	1	2890
	2	6732
Piston	1	7389
Steering knuckle and wheel hub	1	1209
	2	2239
	3	4419
	4	6390
	5	7993
	6	8662

3.2 Modal Analysis of brake assembly

In this stage of analysis, all the brake components models are integrated together to form an assembly model and all boundary conditions and component interfaces are considered. In the FE assembly model, disc brake components are generally assembled by

friction springs through a number of imaginary linear spring elements.

The software used for this study ABAQUS provides algorithms to represent interface between contact pairs. Direct contact interaction between disc brake components is represented by a combination of node-to-surface and surface-to-surface contact elements. The pad is consequently selected as the slave surface. FE modal analysis then is conducted on the assembly model by considering all boundary conditions and interactions between all components.

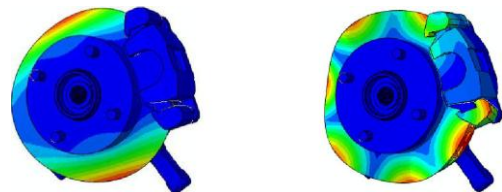


Fig.20 Mode shape at 1562 Hz and 3174 Hz

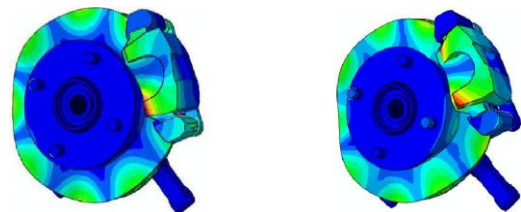


Fig.21 Mode shape at 5184 Hz and 6597 Hz

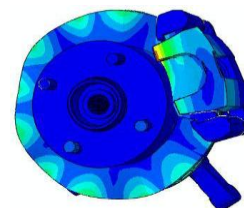


Fig.22 Mode shape at 9452 Hz

3.3 Squeal prediction using CEA

Complex Eigen value analysis is performed to predict squeal occurrence.

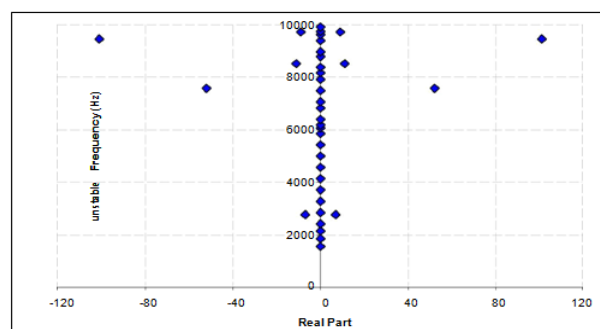


Fig.23 Eigen values extracted from disc brake model

It is performed at rotational speed of 5 rad/s and applied pressure of 0.7 MPa, constant friction

coefficient. In order to predict the squeal occurrence of the disc brake. The CEA is performed between 1 kHz and 10 kHz. The CEA result is plotted on the complex plane. In this case, there are five unstable (squeal) frequencies predicted at 2762 Hz, 7568 Hz, 8528 Hz, 9449 Hz and 9719 Hz.

### 3.3.1 Influence of friction material

Brake squeal is generally defined as a friction induced unstable condition. Since friction is the main cause of instability, a Complex Eigen value analysis has been undertaken to assess the brake stability with the variation of friction coefficient. In the past, the friction coefficient of the brake pad in real vehicle was considered to be typically 0.35. However, brake system today possess friction coefficient that is 0.5 or higher, which increases the propensity of squeal.

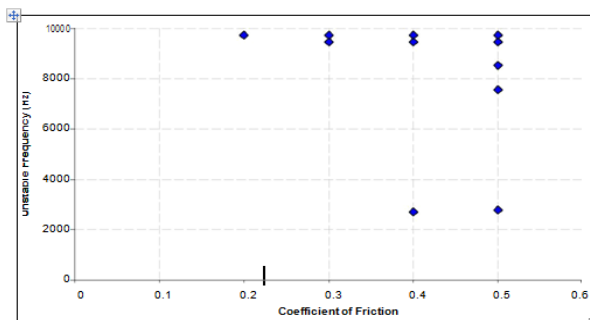


Fig.24 Eigen values extracted from disc brake model

### 3.3.2 Predicting unstable frequencies for varying friction coefficient.

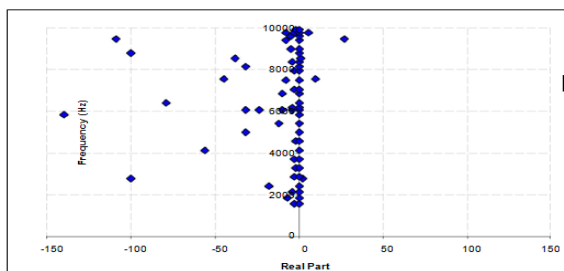


Fig.25 Effect of positive damping on predicted results

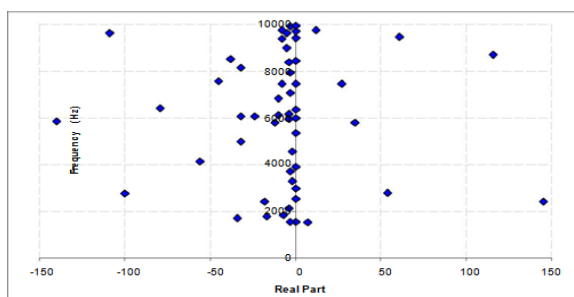


Fig.26 Effect of positive and negative damping on predicted results

### 3.3.3 Influence of real pad

Complex Eigen Value analysis is conducted after replacing perfect pad surface with actual pad surface. From simulation results, due to surface roughness static contact pressure distribution between the brake pad and the rotor is changed. It is also found that contact pressures are distributed symmetrically and the red colour indicates the highest contact pressure.

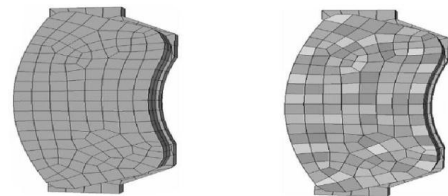


Fig.27 FE model of Perfect pad and real pad surface

### 3.3.4 Static contact pressure distribution (V=0)

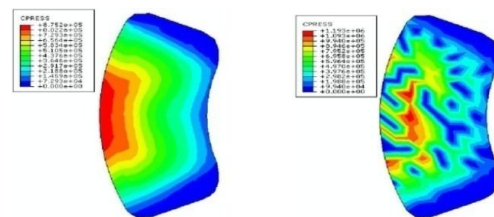


Fig.28 Perfect pad surface and real pad surface

### 3.3.5 Dynamic contact pressure distribution (V=5)

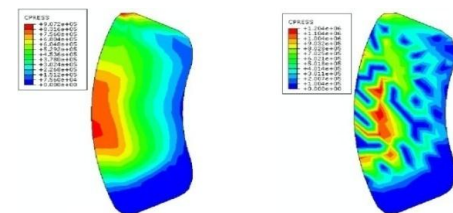


Fig.29 Perfect pad surface and real pad surface

Thus, the CEA is performed by considering the friction positive and negative damping and real pad surface.

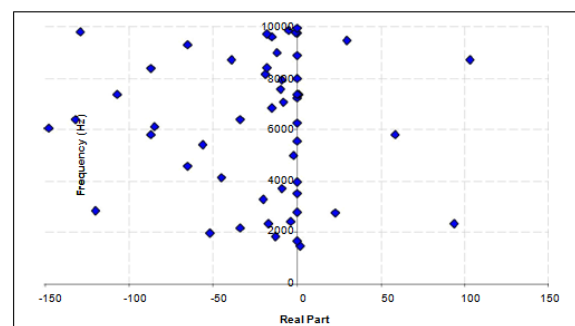


Fig. 30 Effect of real pad surface for both positive and negative damping on predicted results

From the result it is found that there are seven unstable (squeal) frequencies at 1468, 2341, 2775, 5812, 7379, 8702 and 9469 Hz.

### 3.4 Squeal reduction technique

The influence on structural modifications in order to improve squeal performance of the automotive disc brake system under evaluation, by modifying mass and stiffness analysis has been carried out. Moreover, the role of damping added to the system using pad shims has been evaluated. Theoretically, this is accomplished when the positive real parts of complex eigen values of the baseline model are reduced or eliminated. It is convenient to divide structural modifications into three general categories: material modifications, geometric modifications and adding damping shim.

For the material modifications, a traditional evaluation tool has been proposed by a number of researchers to investigate the effect of material stiffness by varying Young's modulus of the disc brake components. In this research a non-traditional evaluation is introduced to examine the effects of different types of material as found in practice, which are used in manufacturing disc brake components for commonly used vehicles or special heavy duty performance vehicles and racing cars.

Material properties of the disc brake components usually have a significant effect on brake squeal generation. Researchers have reported the links between the squeal occurrence and brake components material properties. It is necessary to design the brake pads, rotor, caliper and anchor bracket such that their natural frequencies in the audible range are isolated to avoid mode coupling between them to reduce the squeal occurrence.

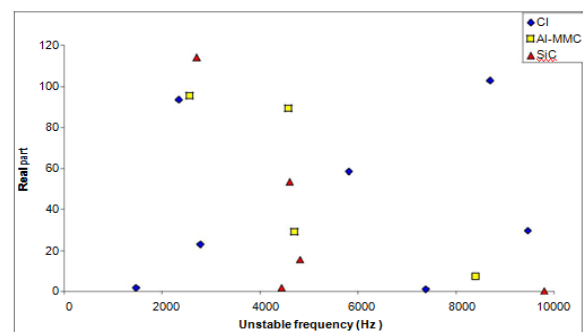
**Table 4** Material properties for different components

Component	Material	Youngs modulus (GPa)	Density (Kg/m <sup>3</sup> )
Pads	Friction material (soft)	0.5	2045
	Friction material (base line)	2.6	2045
	Friction material (stiff)	4	2045
Rotor	C.I (base line)	125	7155
	Al-MMC	70	2800
	C/C-SiC	50	2100
Caliper	C.I (base line)	171	7005
	Steel	210	7850
	Aluminium	71	2800
Bracket	C.I	166	7050
	Steel	210	7850
	Aluminium	71	2800

### 3.4.1 Influence of rotor material

**Table 5** mode shapes for different materials of rotor

Base line	Al-MMC	C/C SiC	Mode shape
1453	1737	1697	
3225	3858	3765	
5062	6055	5909	
7067	8454	8250	
9170	10970	10705	

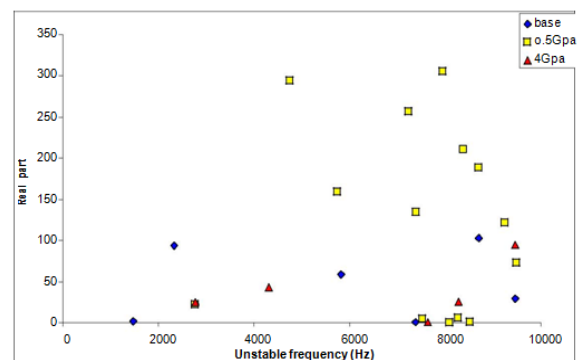


**Fig.31** Effect of rotor material modification on brake squeals.

### 3.4.2 Influence of pad material

**Table 6** Mode shapes for different materials of Pad

Base line	Al-MMC	C/C SiC	Mode shape
2889	2358	3133	
4460	3849	4748	
6735	5681	7217	
8976	7475	9543	

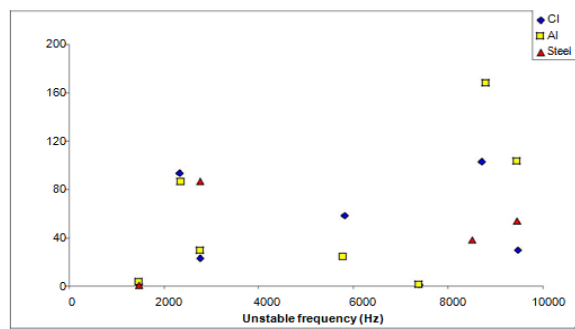


**Fig.32** Effect of pad material modification on brake squeals

3.4.3 Influence on caliper material

**Table 7** Mode shapes for different materials of Caliper

Base line	Aluminium	Steel	Mode shape
2889	2358	3133	
4460	3849	4748	
6735	5681	7217	
8976	7475	9543	

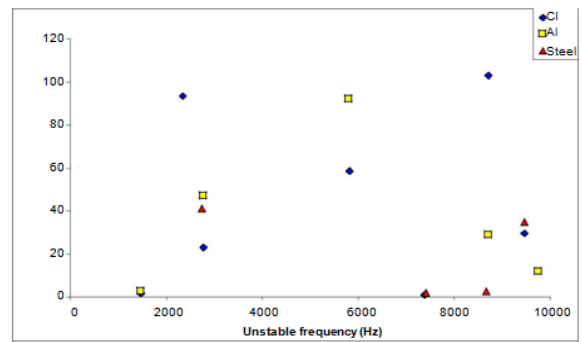


**Fig.33** Effect of caliper material modification on brake squeals

3.4.4 Influence of bracket material

**Table 8** Mode shapes for different materials of Bracket

Base line	Aluminium	Steel	Mode shape
880	916	941	
1755	1827	1876	
3164	3292	3382	
4680	4871	5003	
7533	7842	8054	
9262	9641	9902	



**Fig.34** Effect of bracket material modifications on brake squeal

**Conclusions**

Parametric studies are carried out to examine the effect of changing materials of the brake components, geometric modifications of the pad and rotor and insertion of pad shims into the brake assembly on the propensity of squeal occurrence. Based on the results obtained from parametric study considered in the present study, the following are concluded.

- 1) From the predicted results, it is concluded that using C/C-SiC or Al MMC rotor increases the stability of the brake system and shifting squeal frequencies.
- 2) Increasing pad modulus reduces the overall number of unstable frequencies and improves the overall system stability.
- 3) It is observed that the steel caliper generates less squeal propensity than either cast iron or aluminum caliper.
- 4) It is observed that the material of the anchor bracket also significantly influences the stability of the system. Increasing bracket stiffness reduces the overall brake squeal propensity of the system.

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