Brake Squeal Analysis Of Disc Brake Assembly

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Abstract— A brake is a mechanical device to be used to decelerate a moving body in a controlled manner. Because of its frictional mechanism, Brake squeal is a noise problem caused by vibrations induced by friction forces that can induce a dynamic instability. During the braking operation, the friction between the pad and the disc can induce a dynamic instability in the system. Usually brake squeal occurs in the frequency range between 1 and 20 kHz

This paper summarizes the application of complex eigenvalue analysis in a finite element model of a commercial brake system. The effect of the operational parameters (friction coefficient and braking pressure) and wear on the dynamic stability of the brake system is examined. After identifying unstable frequencies and the behavior of the brake system under different conditions, the performance of some control methods are tested. Changes in parameter like density of Rotor, Young's Modulus, Poison ratio and Friction coefficient and the application of brake noise insulators are presented and their effects discussed.

The complicated geometry of disc brake and the complex force applied by calipers make their analysis difficult. But optimized meshing and accurate simulation of boundary conditions along with ability to apply force, provided by various FEM packages have helped the designer to carry structural and modal vibration analysis with the investigation of critical stresses. In our dissertation work we have analyzed a simplified Disc Brake Assembly for squeal with the help of advance FEA software package. CAD data is correlated with current physical model. For Correlation FFT analyzer is used to extract natural frequency of real brake disc. We used Lancoz's method to extract the natural frequencies by FEA. Complex eigen value extraction is used to identify the Squeal frequencies. Finally squeal frequency will be eliminated with optimized rotor design.

Keywords—*Brake squeal; Complex eigen value; young's modulus; Poison ratio; Friction coefficient; Density; system parameter;*

I. INTRODUCTION

Brake squeal is noise problem caused by vibrations induced by friction forces that can induce a dynamic instability. During the braking operation, the friction between the pad and the disc can induce a dynamic instability in the system. Usually brake squeal occurs in the frequency range between 1 and 20 kHz. Squeal is complex phenomenon, partly because of its strongly dependence on many parameters and partly because of the mechanical interaction in the brake system. The mechanical interaction are considered to be very complicated because of nonlinear contact effect at the friction interface.

A car disc brake system consists of a rotating disc and stationary (non- rotating) pads, carrier bracket, calliper and mounting pins. The pads are loosely housed in the calliper and located by the carrier bracket. The calliper itself is allowed to slide fairly freely along the two mounting pins in a floating caliper design. The Disc is mounted to a car wheel and thus rotates at the same speed as the wheel. When the disc brake is applied the two pads are brought into contact with the disc surfaces. Most of the kinetic energy of the travelling car is converted to heat through friction. But a small part of it converts into sound energy and generates noise. A squealing brake is difficult and expensive to cure. Preferably the noise issue should be resolved at the design stage.

There are several categories of brake noise that are classified according to the frequency of noise occurrence.

Basically, there are three general categories of brake noise: low frequency noise, low frequency squeal and high frequency squeal. Low frequency disc brake noise is a problem that typically occurs in the frequency range between 100 and 1000 Hz. Typical examples of noise problems from this category are groan and moan noise. The generation mechanism of this kind of problem is the friction excitation at the rotor and lining material, which provides energy to the system. This energy is transmitted as a vibratory response through the brake assembly and couples with components of the suspension and chassis.

Although the low frequency noise is an important problem for certain types of brake systems, the most common and annoying problem is squeal noise. Squeal is defined as a noise whose frequency content is 1000 Hz or higher that occurs when a system experiences very high amplitude mechanical vibrations. There are two theories that try to explain how this phenomenon occurs. The first one is called "stick-slip". According to this theory, squeal is a self-excited vibration of the brake system caused as a result of two factors: the static friction coefficient is greater than the sliding friction coefficient; the relationship between sliding friction coefficient f and relative sliding velocity Vr is $\delta f/\delta Vr < 0$. However, this theory cannot explain why the tendency of squeal is different when the same friction couple pair (rotor and pads) is used in different brake systems. Therefore, a second theory, called "sprag-slip", was developed. It demonstrates that the self-excited vibration of the brake system and the high levels of vibration result from an improper selection of geometric parameters of the brake system. In this case, two system modes that are geometrically matched move closer in frequency as the friction coefficient increases. These two modes eventually couple at the same frequency and matching mode shapes, becoming unstable (Dihua and Dongying, 1998). Both theories attribute the brake system vibration and consequent noise to variable friction forces at the pad-rotor interface. These variable friction forces introduce energy into the system. During the squeal event, the system is not able to dissipate part of this energy and the result is the high level in the amplitude of vibration.

II. METHODOLOGY

a) To validate the FEA model or CAD data with existing model.
b) To determine the natural frequency of the system by FFT.
c) To determine the squeal frequencies by Complex Eigen Value analysis.
d) Optimize the parameters to stable the system.

III. CAD MODEL GENERATION

To generate the 3D cad data we use reverse engineering technique. With the help of measuring instruments we took the dimensions to create the 3D cad model. Fig. 1 shows a 3-D plot of the model used.

Fig.1 Brake Disc Assembly

The brake model used in this study is a simplified model consisting of the two main components contributing to squeal: the disc and the pad (Fig.2)

A simplified model was used in this study for the following reasons:

1. For brake squeal analysis, the most important source of nonlinearity is the frictional sliding contact between the disc and the pads.
2. The simulation includes geometry simplifications to reduce CPU time, allowing far more configurations to be computed.

Fig.2 Simplified Disc & Pad Assembly

IV. MODAL ANALYSIS BY FEA

A. *FEA Mesh of Disc & Brae Pad*

The FE mesh is generated using 19,000 solid elements. The friction contact interactions are defined between both sides of the disc and the friction material of the pads. A constant friction coefficient and a constant angular velocity of the disc are used for simulation purposes.

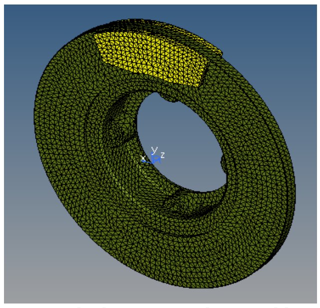


Fig.3 Meshed Disc & Pad Assembly

B. *Boundary condition & Loading*

Boundary conditions for our problem are:

For Disc: All DOF's for disc are locked except rotary motion about z -axis.

For Brake Pad: All DOF's for brake pad are locked except translatory motion of brake pad in z- axis.

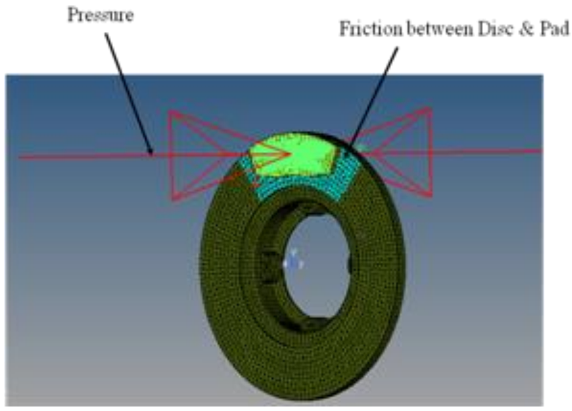


Fig.4 Loading and B.C for Disc & Pad Assembly

For our analysis, we apply line pressure of 10 bar on both pads.

So, Pressure = 10 bar = 1 N/mm2

The pressure is applied as concentrated load on a single node by connecting all nodes on brake pad with a single node.

We know,

Pressure = Force/Area i.e. P= F/A--------------------- Eq.1

Area of brake pad = 550 mm2 (Calculated directly from 3D data)

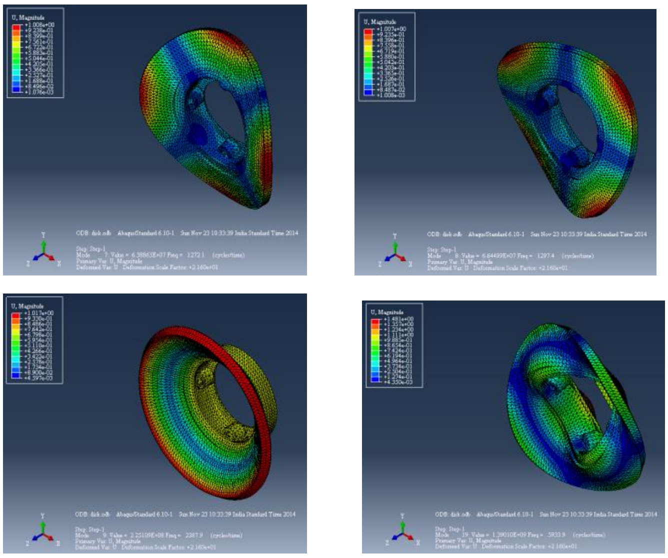
From 1 , 1 = F/ 550

F = 1 x 550 = 550N

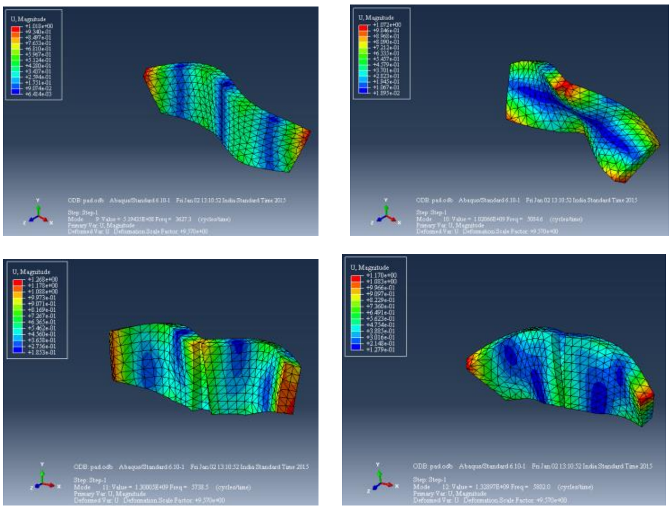
Concentrated force , F = 550N

We assume Rotor speed = v = 5 rad/sec

C. *Mode Shapes for Disc at free-free condition (Rotor)*



D. *Mode shapes for brake pad free-free condition*



V. MODAL ANALYSIS BY FFT

Experimental modal analysis is done and first seven natural frequencies and corresponding mode shapes are obtained using FFT analyzer setup. The CAD model of disk brake and brake pad is created using PRO-E wildfire-2.0 required for further numerical analysis using Abaqus.

The rotor and brake pad modal analysis is done using FFT testing. The result are summarized below.

TABLE I
NATURAL FREQUENCIES FOR CURRENT ROTOR & BRAKE PAD BY FFT TESTING

Mode	Frequency (Hertz) (Rotor)	Frequency (Hertz) Brake Pad
1	1224	3664
2	1248	3672
3	1440	3680
4	1448	5400
5	1464	5408
6	2440	5448
7	6240	5456

A. *Variation of natural Frequency*

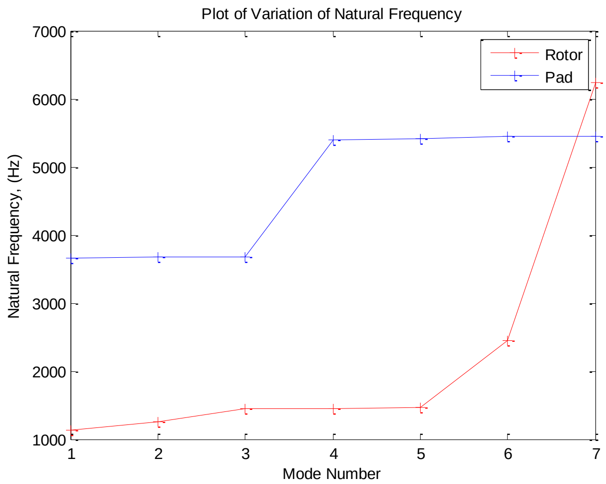


Fig. 5 Variation of natural Frequency

B. *Correlation Analysis*

To validate numerical model, a correlation analysis between numerical and experimental data is conducted for healthy disc & brake pad. In general, correlation analysis is a technique to examine quantitatively and qualitatively the correspondence and difference between analytically and experimentally obtained modal parameters.

The frequencies measured on the disc and calculated by the simulated model for modes with free-free boundary conditions are shown in Table 5.2. It can be observed that the measured and simulated frequencies are in good agreement.

In a similar way, the parameters for the pads are estimated based on the measured data indicated in Table I. The measured and simulated frequencies are in good agreement.

TABLE III
MODAL RESULTS OF THE ROTOR AT FREE-FREE BOUNDARY CONDITIONS

Sr. no.	Experimental Frequency (Hz)	FEA Frequency (Hz)	Difference (%)
1	1224	1272.1	3.8
2	1248	1297.4	3.8
3	2440	2387.9	-2.2
4	6240	5933.9	-5.2

TABLE IIIII
MODAL RESULTS OF THE PAD AT FREE-FREE BOUNDARY CONDITIONS

Sr. no.	Experimental Frequency (Hz)	FEA Frequency (Hz)	Difference (%)
1	3680	3627.3	-1.5
2	5802	5520	-5.1

VI. BRAKE SQUEAL ANALYSIS USING ABAQUS

From our brake squeal analysis we found the squeal at frequency 7972.9 Hz i.e. 8kz, which is having positive real part value. We can define this low frequency squeal as it is below 10 kz.

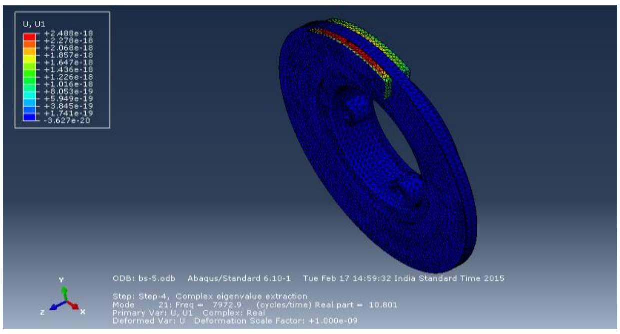


Fig.5 Squeal frequency of disc brake assembly

The effect of the stiffness of the disc on the disc brake squeal is studied by changing density of disc. In below Table evaluated density of system at stable.

TABLE IVV
INSTABILITY FOR DIFFERENT DENSITIY VALUE

Sr. No	Density	Real part of complex Eigen Value	Result
1	7.1	+ve	Instability
2	7.12	+ve	Instability
3	7.15	+ve	Instability
4	7.2	-ve	Stability
5	7.25	+ve	Instability

This chapter is devoted to the study of brake squeal frequencies and instability of the system. Below image shows graph of frequency vs damping ratio. The values we got from brake squeal analysis done with the help of Abaqus.

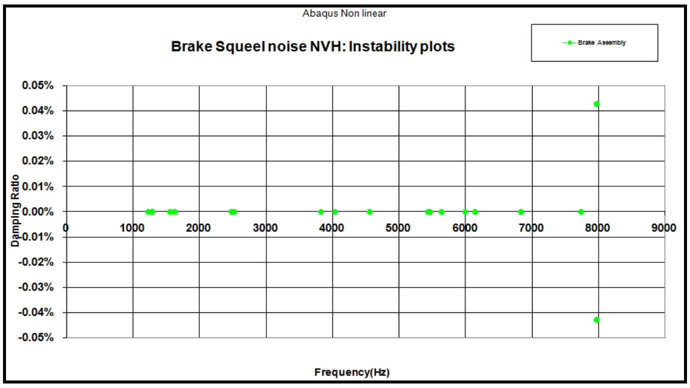


Fig.6 Brake Squeal Noise: Instability plots

We found only four squeal frequency in our analysis, which may produce instability in the system. From our comparison between results of modal analysis using FFT (experimental analysis) and Abaqus, we confirm the density of rotor as (7.1, 7.12, 7.15, 7.25) kg/mm3. The density value shows the grey cast iron which is used to manufacture brake rotor is SAE G1800 grade. As we know change in density affects the stiffness. So we tried with higher grade i.e. G3000 grade grey cast iron which have the density of 7.2 kg /mm3and we found successful removal of squeal frequency. We repeated the complex eigen value extraction analysis with changed density for rotor. Below image shows the same mode shape but with the real part value as zero. So we can assure that there is no instability present in the brake system by using higher grade (G3000) material for rotor.

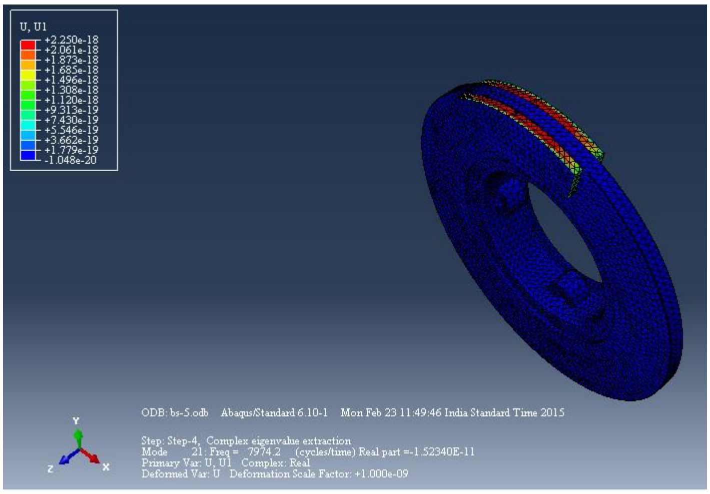


Fig.7 Removed Squeal frequency of disc brake assembly

VII. NUMERICAL ANALYSIS

A. *Parameter to be considered*

For the optimum solution we have considered five parameter these are mentioned in below table.

TABLE V
DIFFERENT PARAMETER

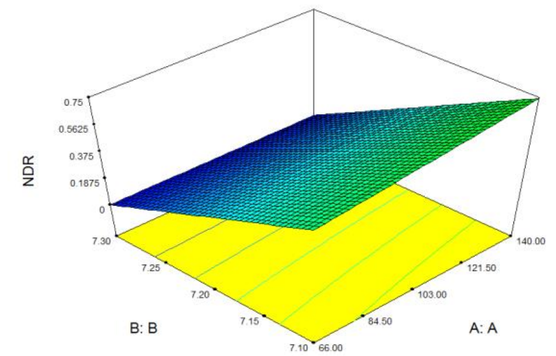
Property	Low	High	Units
Density of Rotor (A)	7.1	7.3	kg/mm2
Young's Modulus for Rotor	66	140	Gpa
Poison's ratio for rotor	0.2	0.3	
Friction after applying the brakes	0.5	0.8	
Piston pressure	5	10	Bar

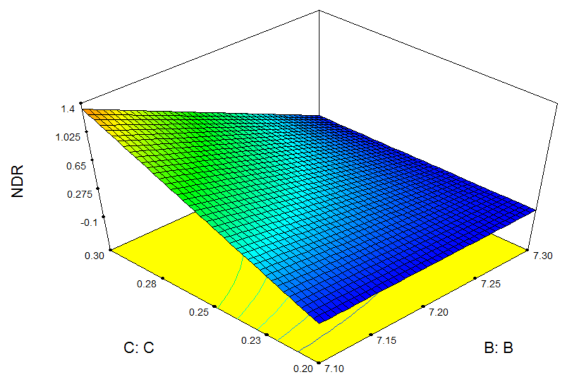
B. *Parameter with Negative Damping ratio*

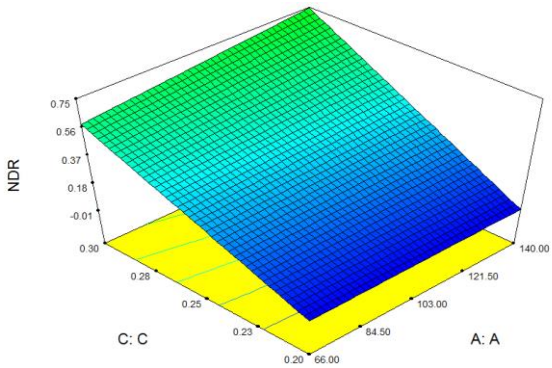
TABLE VI
PARAMETER WITH NEGATIVE DAMPING RATIO

Std	Run	Block	A Young's Modulus for rotor (Gpa) (66-140)	B Density of Rotor kg/mm^2 (7.1 to 7.3)	C Poisons ratio (0.2 - 0.3)	D Friction coeifficient at spinning (0.5 - 0.8)	E Piston pressure in Bar (5 - 10)	NDR (Negative damping ratio)	NDR X 1000
19	1	Block 1	66	7.3	0.2	0.5	10	0.00000	0
30	2	Block 1	140	7.1	0.3	0.8	10	0.00144	1.44
9	3	Block 1	66	7.1	0.2	0.8	5	0.00000	0
31	4	Block 1	66	7.3	0.3	0.8	10	0.00000	0
3	5	Block 1	66	7.3	0.2	0.5	5	0.00000	0
25	6	Block 1	66	7.1	0.2	0.8	10	0.00000	0
22	7	Block 1	140	7.1	0.3	0.5	10	0.00153	1.53
13	8	Block 1	66	7.1	0.3	0.8	5	0.00129	1.29
4	9	Block 1	140	7.3	0.2	0.5	5	0.00000	0
6	10	Block 1	140	7.1	0.3	0.5	5	0.00153	1.53
21	11	Block 1	66	7.1	0.3	0.5	10	0.00107	1.07
7	12	Block 1	66	7.3	0.3	0.5	5	0.00000	0
8	13	Block 1	140	7.3	0.3	0.5	5	0.00000	0
24	14	Block 1	140	7.3	0.3	0.5	10	0.00000	0
28	15	Block 1	140	7.3	0.2	0.8	10	0.00000	0
16	16	Block 1	140	7.3	0.3	0.8	5	0.00000	0
23	17	Block 1	66	7.3	0.3	0.5	10	0.00000	0
18	18	Block 1	140	7.1	0.2	0.5	10	0.00000	0
15	19	Block 1	66	7.3	0.3	0.8	5	0.00000	0
12	20	Block 1	140	7.3	0.2	0.8	5	0.00000	0
20	21	Block 1	140	7.3	0.2	0.5	10	0.00000	0
14	22	Block 1	140	7.1	0.3	0.8	5	0.00144	1.44
11	23	Block 1	66	7.3	0.2	0.8	5	0.00000	0
10	24	Block 1	140	7.1	0.2	0.8	5	0.00000	0
27	25	Block 1	66	7.3	0.2	0.8	10	0.00000	0
26	26	Block 1	140	7.1	0.2	0.8	10	0.00000	0
17	27	Block 1	66	7.1	0.2	0.5	10	0.00000	0
2	28	Block 1	140	7.1	0.2	0.5	5	0.00000	0
32	29	Block 1	140	7.3	0.3	0.8	10	0.00000	0
5	30	Block 1	66	7.1	0.3	0.5	5	0.00107	1.07
29	31	Block 1	66	7.1	0.3	0.8	10	0.00129	1.29
1	32	Block 1	66	7.1	0.2	0.5	5	0.00000	0

C. *Interaction factors*







VIII. RESULT AND DISCUSSION

From that we can optimized parameter 7.3x 10-6 kg/mm2 for density, 103.54 GPa for Young's Modulus, 0.25 for Poisons' Ratio, 0.68 for coefficient of friction after applying the brakes and 8.3 bar for Piston Pressure.

TABLE VI
OPTIMUM PARAMETER

Sr. No.	Young's Modulus for Rotor (Gpa)	Density For rotor (kg/mm2)	Poison's ratio	Dynamic friction coefficient Between rotor & pad	Piston pressure	Negative Damping Ratio
1	103.54	7.3	0.25	0.68	8.3	0.000000089
2	128.13	7.19	0.2	0.76	5.55	0.000000140
3	120.18	7.25	0.2	0.56	9.01	0.000000173
4	69.17	7.3	0.23	0.5	8.9	0.000000234
5	129.25	7.22	0.2	0.61	6.18	0.000000295
6	78.96	7.3	0.27	0.66	7.19	0.000000446
7	113.64	7.2	0.2	0.75	7.25	0.000000893
8	125.61	7.3	0.23	0.58	5.36	0.000001471
9	122.83	7.3	0.24	0.76	7.98	0.000001970
10	78.06	7.3	0.28	0.76	9.69	0.000002136

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IX. CONCLUSION

A method of complex eigen value analysis to identify the squeal frequency was presented. The method was then applied to disc brake assembly.

The proposed method was effective for obtaining the exact squeal frequency which realizes greater instability in the system. In the disc brake assembly example, the squeal found at the interaction between disc and brake pad. The model of disc brake assembly is created and analyzed for natural frequency using Abaqus 6.10 and the results are compared with Experimental modal analysis using FFT for validation of model. Then squeal frequency of disc brake assembly with constraints is obtained using Abaqus 6.10. The proposed change in considered five parameters i.e. Density, Poisons' ratio, Pressure, Young's modulus and Friction after applying the brakes to removed the squeal frequency.

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