Chapter 1
Brake Squeal Analysis and Optimization Using the Finite Element Method and Taguchi Techniques

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Abstract It is well-known that automobile brakes generate several kinds of noises such as the squeal, groan, chatter, judder, moan, hum, and squeak. Squeal is the most prevalent, annoying, and can be reduced by variations in geometry, such as coefficient of friction and stiffness of a material. The brake squeal generally occurs in the range of 1–16 kHz. Furthermore, the development of methods to predict noise occurrence during the design of a brake system has been the target of many researchers in recent years. In order to withstand these vibrations and have a smooth ride, automotive engineers face the challenge of designing mechanisms. The design of the brake disc rotor is built in the CAD SolidWorks and was analyzed and simulated by using the finite element analyses.

Keywords Brake squeal · Brake disc rotor · SolidWorks · Finite elements analysis (FEA)

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1.1 Introduction

A disc brake is a mechanical device that is used to decelerate a moving vehicle. The mechanism of braking works as one of the most basic safety–critical elements of industrial technology cars for tourists. The heat produced during the braking force can also be quickly dissipated as disc brakes are exposed to the environment, offering a much better output than the drum. Brake noise has been among the top vehicle quality warranty concerns in recent years. The most frequent of all brake sounds is brake squealing. This kind of noise emerges when a braking device displays elevated vibration levels, resulting in highly elevated sound waves. Most scientists and engineers accepted that squealing noise in the disc brake would cause instability due to friction forces, which would lead to self-excited vibrations. The disc, the pads, and the caliper in which they are mounted will vibrate under the right conditions.

In the last few decades, a considerable number of studies has been conducted by several experts around the world on the prospect of understanding the mechanisms of brake squeal noise in order to enhance the performance of car users' comfort and lower the required amount of ambient noise. Brake squeal investigations can be traced all the way back to the 1930s. To now, however, there is no technology and system available to fully reduce the brake squeal. Understanding the processes of generation of squeal is important for the creation of quiet brakes and the handling of noisy brakes. Key mechanisms would be set in order to clarify the phenomena of brake squeal—for example, stick–slip, sprag-slip, modal coupling, and hammering of the excitation mechanism.

The first mechanism explains that the brake squeal is the product of a stick–slip mechanism. As Kinkaid et al. [1] put it in his review article, Mills [2] believed that the brake squeal emerged because the dynamic friction coefficient decreased with increasing slippery velocity and contributed to steady-state sliding becoming unstable and caused friction-induced vibrations. A multi-degree finite element model was used to simulate a braking device and to analyze friction-velocity negative squealing effects [3]. It came to the conclusion that the squeal inclination improved as the friction level increased because there was no negative friction-speed curve. When an increase in friction level is present, it does not have any effect on squeal tendency and, in some brake systems, negative friction-speed inclination may have a superior effect on the tendency of the brake squeal. The Single Degree of Freedom (SDOF) system consists of a brake pad as a mass (m) which rests on a rotating disc at constant speed (v) and attaches to a linear spring at a fixed end, as shown in Fig. 1.1a. Mathematically, when the coefficient of friction μ between the pad and the disc in Fig. 1.1b is considered to decrease linearly with the sliding velocity, for example, $\mu = \mu s - \alpha v$, the motion equation of the pad will be as follows:

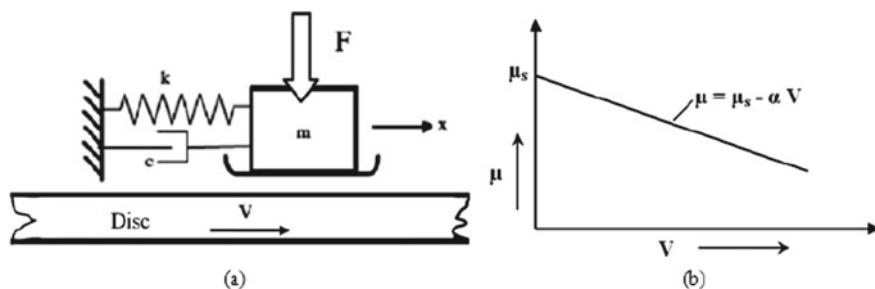$$M\ddot{X} + C\dot{X} + KX = \mu F = F(\mu s - \alpha(v - \dot{X})) \tag{1.1}$$

or

Fig. 1.1 Schematic diagram for stick–slip mechanism

$$M\ddot{X} + (c - \alpha F)\dot{X} + KX = F(\mu s - \alpha v) \tag{1.2}$$

where K is the spring stiffness, m is the mass, and F is the friction force. The coordinate x is a measure of the displacement of the mass from the direction of the balance. The damping coefficient $(c - \alpha F)$ can then be negative if $(\alpha F > c)$ results in an oscillation of the pad with a greater amplitude which may lead to squealing rather than decreasing oscillations consistent with positive damping.

Squeal produced by the disc brake of the car was considered to be one of the major problems in the automobile industry due to repeated complaints that decrease customer satisfaction. Abendroth and Wernitz [4] noted that many friction material manufacturers have to spend up to 50% of their engineering budget on NVH problems. A variety of similar experiments have been performed on the action of disc brakes by modal analysis. Finite element model of the brake rotor disc was developed by the researcher to analyze the effect of the top hat configuration on the modality of the brake rotor disc. The result shows how the natural frequency of the configuration relates to the thickness of the disc and the structure of the hat. In 2009, Nouby et al. [5] found in their study a complex study of the value of the brake system, along with the configuration of the experiments, in order to determine the ideal nature of the brake system. They mentioned that by increasing Young's back plate modulus and changing the form of the friction content by inserting chamfer and slots, the inclination of the brake squeal can be significantly reduced [6].

1.2 Methodology

The key concept behind this project is to create a 3D model of the vehicle's brake disc to achieve vibration and deformation accuracy. In order to do this, the 3D model was constructed using SOLIDWORKS. The research was performed using the finite element method software. In the finite element method, the various properties of the materials would be used in the term to derive the result. The primary aim of the

approach was to ensure that the project was properly planned in order to be effective in the study of the project.

1.2.1 3D Design if the Brake Disc Model

1.2.1.1 3D CAD Modeling

SOLIDWORKS is a modular parametric architecture that is simple to use and enables the user to edit design at any point. The brake disc rotor and caliper was modeled in the SOLIDWORKS 2019 software. The material of the brake disc rotor is gray cast iron. Figures 1.2 and 1.3 are 3D CAD model for the brake disc rotor design.
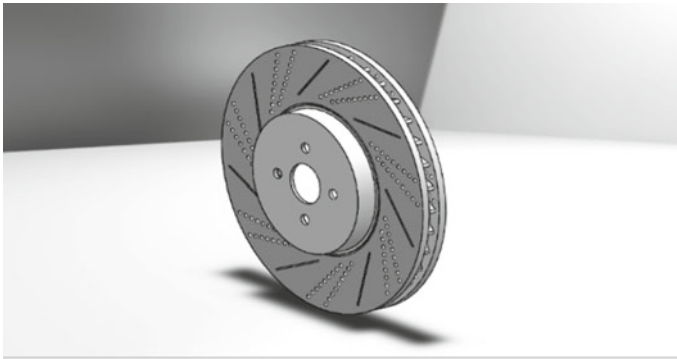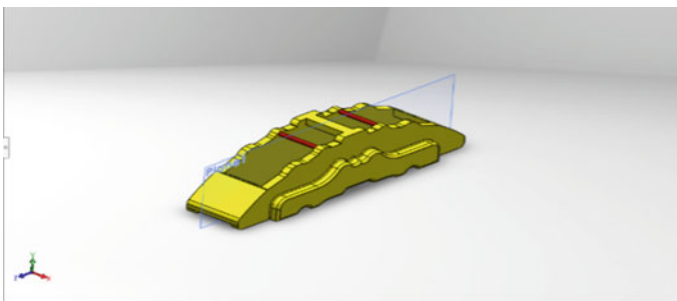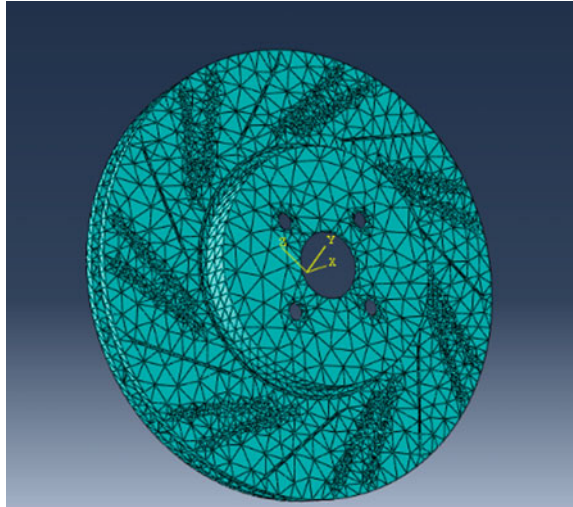

Fig. 1.2 3D model of brake rotor

Fig. 1.3 3D model of caliper

Fig. 1.4 Mesh for the rotor

1.2.2 Finite Element Analysis

Simulia Abaqus CAE is the program used to do the finite element analysis (FEA) for the brake disc. Structural and machinery final element analysis for solid (3D) modeling or surface (2D) development, object meshing, analysis setup, and post-processing solvers in a variety of other software applications including ANSYS, Autodesk simulation, and Marc. The benefit of using this app is an easier use as it allows the user to create/import geometry, to mesh the surface, to evaluate the input data, and to achieve animation and meaning in principle with any mode shape and frequency.

The physical model of the entire brake disc was modeled using the SOLID-WORKS simulation tools. The primary purpose of this research is to study and evaluate the disc brake disc rotor. The entire SOLIDWORKS assembly is stored as an initial graphics exchange specification (IGES) file that can be imported into the FEA program. This section has to be translated to an IGES file to convert a CAD model to a FEM model, so that it can run for review. Step manager is created to define the simulation in terms of natural frequency in linear perturbation. The next step is to create a finite element mesh for the solid using the tetrahedral mesh generator and to select all the disc rotor structure (Fig. 1.4).

1.2.3 Modal Analysis

The modal analysis was carried out in order to achieve the performance results using the impact test. The dynamic analysis of the brake disc rotor was studied using modal

Fig. 1.5 Experiment setup

testing. The software used to conduct the analysis to extract modal properties is the LMS testing lab software. The disc rotor that implemented this ideal effect would get the right impulse that causes a limited duration and determines the constant amplitude of the frequency domain. A specific hammer is used to conduct this test which cell load was put in its tip to obtain the force of impact. This technique is also effective in terms of transducers and measuring methods, it is least efficient in terms of time because it requires a lot of time to shift the accelerometer. This approach is widely used in cases where space constraints make it possible to repair accelerometers when there is not enough space to use the impact hammer. In this project, the accelerometer used has a reasonable accuracy capability, with a sensitivity of 50.81 mV/g, the round accelerometer will deliver a frequency from 0.5 to 5000 Hz at a certain time. The accelerometer, laptop, NI Acoustic and Vibration Data Recorder, effect hammer, and frame structure are the equipment that was used (Fig. 1.5).

The drawing of the brake rotor structure has been created in LMS Test Lab software using the drawing tool according to the coordinate of the actual engine mounting bracket that has been measured. After finishing the drawing, there is the need to label the node points at the drawing (Figs. 1.6 and 1.7).

1.3 Result and Discussion

The results of the studies that have been performed will be discussed in this section. All variations in the outcomes will be explained in relation to the problem, whether during the experiment or the simulation procedures.
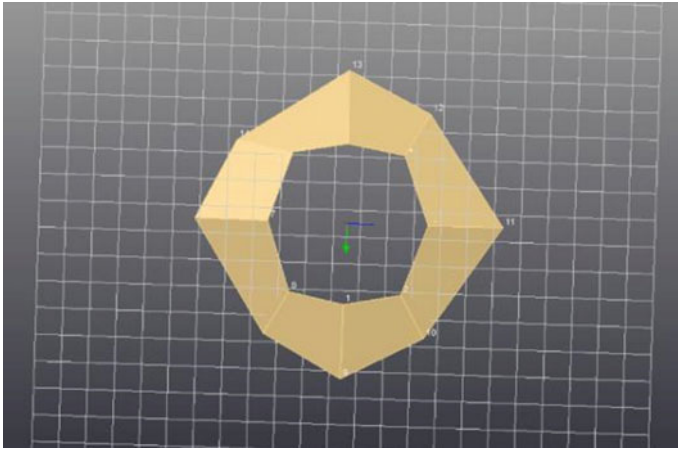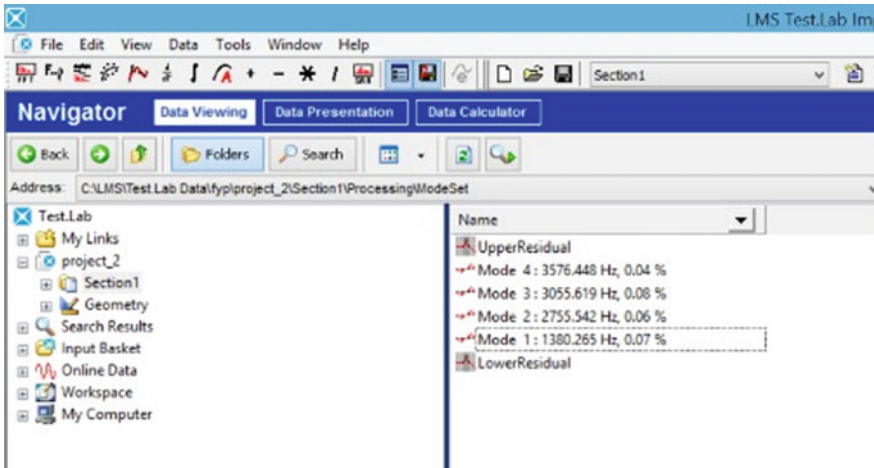
Fig. 1.6 The drawing of disc rotor in LMS software

Fig. 1.7 Result obtained from EMA

1.3.1   Natural Frequency Using Mild Steel, Gray Cast Iron, and Carbon Ceramic

See Tables 1.1, 1.2 and 1.3.

Table 1.1 Value of natural frequency for mild steel

	Natural frequency (Hz)		
Mode	Design 1	Design 2	Design 3
1	2039.0	1973.3	1827.8
2	2040.7	1974.3	1827.9
3	2648.8	2647.1	2640.9
4	4403.8	4398.1	4251.2
5	4408.9	4402.2	4521.6
6	4696.4	4546.7	4359.2
7	4698.3	4549.6	4379.2
8	4912.1	4734.9	4383.3

Table 1.2 Value of natural frequency for gray cast iron

	Natural frequency (Hz)		
Mode	Design 1	Design 2	Design 3
1	1653.1	1599.8	1477.9
2	1654.5	1600.6	1478.6
3	2147.5	2146.1	2140.3
4	3570.3	3565.6	3434.9
5	3574.4	3569.0	3438.9
6	3807.5	3686.1	3526.8
7	3809.0	3688.5	3548.5
8	3982.3	3838.7	3550.4
9	4501.8	4412.8	4239.3
10	4513.1	4425.8	4252.6
11	4856.1	4714.5	4438.3
12	4860.8	4720.0	4447.4

1.3.2   Comparison Natural Frequency Between All Three Design and Materials

Table 1.4 compares design 1 simulation to mild steel, gray cast iron, and carbon ceramic. According to the results, carbon ceramic provides the best value of natural frequency. The carbon ceramic has an 8-mode frequency range of 1378.6–3321.2 Hz.

Table 1.5 compares it to mild steel, gray cast iron, and carbon ceramic for design 2 simulation. According to the results, carbon ceramic provides the best value of natural frequency. The carbon ceramic has an 8-mode frequency range of 1334.2–3201.4 Hz.

Table 1.6 compares design 3 simulation to mild steel, gray cast iron, and carbon ceramic. According to the results, carbon ceramic provides the best value of natural frequency.

Table 1.3 Value of natural frequency for carbon ceramic

Mode	Natural frequency (Hz)		
	Design 1	Design 2	Design 3
1	1378.6	1334.2	1235.8
2	1379.8	1334.9	1235.9
3	1790.9	1789.8	1785.6
4	2977.5	2973.6	2874.4
5	2980.9	2976.4	2874.6
6	3175.3	3074.1	2947.4
7	3176.6	3076.1	2960.9
8	3321.2	3201.4	2963.6
9	3754.4	3680.2	3542.4
10	3763.8	3691.0	3552.3
11	4049.8	3931.8	3709.3
12	4053.8	3936.4	3720.0
13	4947.6	4939.2	4922.3
14	4956.1	4947.1	4927.8

Table 1.4 Comparison natural frequency of design 1 with various material

Mode	Design: 1		
	Natural frequency (Hz)		
	Mild steel	Gray cast iron	Carbon ceramic
1	2039.0	1653.1	1378.6
2	2040.7	1654.5	1379.8
3	2648.8	2147.5	1790.9
4	4403.8	3570.3	2977.5
5	4408.9	3574.4	2980.9
6	4696.4	3807.5	3175.3
7	4698.3	3809.0	3176.6
8	4912.1	3982.3	3321.2

In comparison to all designs and materials, the value for natural frequency is smaller, and carbon ceramic with design 3 also delivers a smaller value than carbon ceramic with designs 1 and 2. The simulation results for all three designs and materials demonstrate that carbon ceramic has the lowest value when compared to other materials. Carbon ceramic has extraordinarily high and consistent coefficients of friction when compared with mild steel and gray cast iron, despite of surface moisture or temperature.

This implies that carbon ceramic brakes, on the other hand, counteract the so-called fading effect. Fading is the unwanted decline in braking impact that particularly because to longer braking and often successive braking movements, brakes

Table 1.5 Comparison natural frequency of design 2 with various material

| | Design: 2 | | |
| | Natural frequency (Hz) | | |
Mode	Mild steel	Gray cast iron	Carbon ceramic
1	1973.3	1599.8	1334.2
2	1974.3	1600.6	1334.9
3	2647.1	2146.1	1789.8
4	4398.1	3565.6	2973.6
5	4402.2	3569.0	2976.4
6	4546.7	3686.1	3074.1
7	4549.6	3688.5	3076.1
8	4734.9	3838.7	3201.4

Table 1.6 Comparison natural frequency of design 3 with various material

| | Design: 3 | | |
| | Natural frequency (Hz) | | |
Mode	Mild steel	Gray cast iron	Carbon ceramic
1	1827.8	1477.9	1235.8
2	1827.9	1478.6	1235.9
3	2640.9	2140.3	1785.6
4	4251.2	3434.9	2874.4
5	4521.6	3438.9	2874.6
6	4359.2	3526.8	2947.4
7	4379.2	3548.5	2960.9
8	4383.3	3550.4	2963.6

can get hot and the braking effect can be substantially diminished as a result of overheating. Carbon ceramic brake materials offer significant properties that help to minimize this so-called squeal. Carbon ceramic has a better toughness, lower density, extraordinarily high thermal shock resistance and breaking strength, and excellent conductivity. Indeed, design 3 will be the finest among the others in terms of design. It is because that the modification on design 3 is on the brake rotor which is the hole diameter. The hole is more compared to other designs while the hole also has a different diameter. It can be demonstrated that the value for 8 modes for design 3 with carbon ceramic is less than 3000 Hz.

Fig. 1.8 Design summary of
Taguchi method

Design Summary

Taguchi Array L9(3^3)
Factors: 3
Runs: 9

Columns of L9(3^4) array: 1 2 3

1.3.3 Taguchi Method Using Minitab

The orthogonal arrays columns represent the factor while the rows of the orthogonal array represent the runs. A Taguchi design box appeared with 3-Level Design and Number of Factors = 3 (Fig. 1.8).

For the 3-Level Design, there are 3 different designs of brake disc had been selected and also there are 3 factors were chosen. The factors that were chosen are rotor diameter (mm), material density (kg/m^3), and hole diameter (mm) (Tables 1.7; Figs. 1.9 and 1.10).

As a result form Table 1.8 and 1.9, the density of materials ranks first in terms of means. The density of material is likewise in the first rank, as is signal noise to ratios. Meanwhile, the rotor diameter and hole diameter came in the third spot, respectively. The phrase "smaller is better" was used to discover the optimal value of natural frequency. According to the graph, the optimal value for rotor diameter is 180 mm, density of material is 7840 kg/mm^3, and hole diameter is 2.5 mm. Meanwhile, the optimal value for signal noise to ratios is 185 mm for rotor diameter, 2450 kg/mm^3 for material density, and 5.0 mm for hole diameter.

Table 1.7 Taguchi orthogonal array design with complete data

Rotor diameter (mm)	Material (density kg/m^3)	Hole diameter (mm)	Frequency (Hz)
175	2450	2.5	1378.6
175	7100	4	1523.4
175	7840	5	1572.3
180	2450	4	1290.7
180	7100	5	1505.6
180	7840	2.5	1953.7
185	2450	5	1235.8
185	7100	2.5	1354.9
185	7840	4	1736.6

Main Effects Plot for Means
Data Means

Fig. 1.9 Main effect plot for means

Main Effects Plot for SN ratios
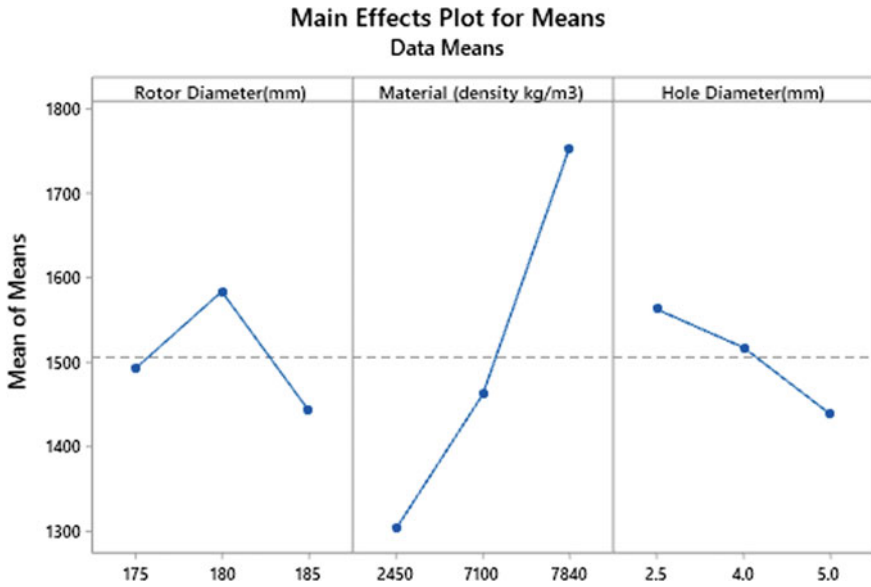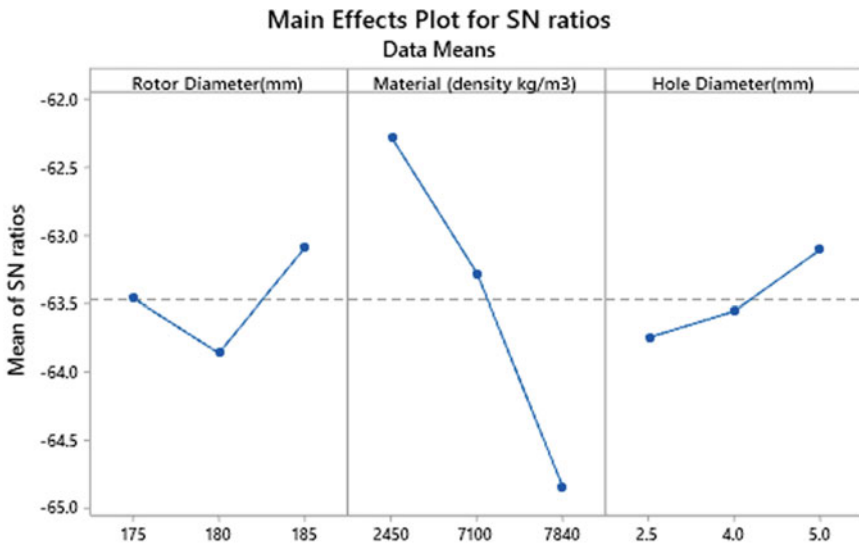Data Means

Signal-to-noise: Smaller is better

Fig. 1.10 Main effect plot for signal noise to ratios

Table 1.8 Response table for means

Level	Rotor diameter (mm)	Material (density kg/m^3)	Hole diameter (mm)
1	1419	1302	1562
2	1583	1461	1517
3	1442	1754	1438
Delta	141	453	125
Rank	2	1	3

Table 1.9 Response for signal noise to ratios

Smaller is better			
Level	Rotor diameter (mm)	Material (density kg/m^3)	Hole diameter(mm)
1	−63.46	−62.28	−63.75
2	−63.86	−63.28	−63.56
3	−63.09	−64.85	−63.11
Delta	0.77	2.57	0.64
Rank	2	1	3

1.4   Conclusion

The goal of this chapter is to verify the relation between the results of both FEA and EMA, as well as in the change of the brake disc. Furthermore, Minitab software was used to fulfill the objectives of Taguchi methods optimization. To maximize the optimal factor, the response and main effect for means and signal noise to ratios were determined. The simulation of 3D model of brake disc on three different designs was simulated by using FEA. It also analyzes the natural frequency on the brake disc from simulation in order to reduce the noisy squeal of the brake disc. It may be said that the lower the value of natural frequency, the less noise there will be during braking. By altering the design of the brake disc, it is able to attain a satisfactory result in terms of natural frequency in terms of vibration.

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