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*Research Article*

# Comparison of Disc Brake Squeal between Ordinary and Drilled Brake Discs Using Finite Element Method

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**Abstract:**
*During the braking of the automobile, the brake squeal is a direct concern for the driver. Sometimes, the automaker loses revenue from the customer's warranty claims. The automaker needs to solve or alleviate the squealing noise during the brake action. The study aimed to compare the disc brake squeal of general and drilled brake discs. The disc brake squeal is an instability of the disc brake system. One popular theory used in the analysis is Complex Eigenvalue Analysis and simulation using Finite Element Technique. In this work, three models of brake discs were compared: an ordinary brake disc and two drilled brake discs. For drilled brake discs, the first and second models had 30 and 60 holes on the disc, respectively. The study revealed that a more stiffness structure generates higher frequencies than a flexible structure. Each model generated a different amount of unstable frequency modes.*

**Keywords:** *Brake squeal, Drill brake disc, Complex Eigenvalue Analysis, Finite Element*

## 1. Introduction

Nowadays, automobiles are the main means of transportation. And what all automobiles have in common is the braking system. The braking system stops or reduces speed as soon as the driver uses it. During braking, the brake pads on both sides begin to splice the brake disc. The friction between the brake disc and the brake pads causes the automobile to stop or slow down. Sometimes, there is a squealing noise due to the vibration of the brake disc. The driver can hear this sound even in the passenger cabin. The brake noise is a direct problem for drivers who may not be confident, or for pedestrians who feel uncomfortable or are concerned about their vehicle's braking performance.

Automaker needs to fix the problem of braking noise. This may cause the automaker to lose revenue from the customer warranty claims. The automaker needs to solve or alleviate the squealing noise during the brake action. With the increasing demand for driving comfort, a greater need to address Noise, Vibration, and Harshness (NVH) [1]. Therefore, the squealing noise or brake squeal is one of the noise problems caused by disc brakes. It is a sound that can be heard in the frequency range of 1000-16,000 Hz, while humans can hear it in the frequency range of 20 – 20,000 Hz.

Brake squealing is a current problem that cannot be completely solved. This is despite the fact that it has been widely studied for more than 30 years. There are many factors involved, such as elastic modulus [2], friction [3], contact area [4], brake pad pressure [5], brake pad design [6], temperature on the brake disc and pad [7], etc. There was studies about the relationship of temperature changes on the brake pads [8] and disc [9]. It was found that the friction

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coefficient decreases with increasing the temperature and is called "thermal fade". During braking, the brake pads on both sides begin to splice the brake disc, and the heat is produced from their frictional traction. To improve the performance of the brake system, it must be cooled quickly. Brake discs are often drilled to increase cooling area by the convection heat transfer of air passing through the drilled holes. To analyze and avoid the squealing noise of the brake system, this paper will study an ordinary brake disc compared to two drill brake discs using Finite Element Method and Complex eigenvalue analysis [10]. To use as a guideline for adjusting the sound frequency of the brake discs that occur. This may reduce the driver's annoyance and concern about the vehicle's braking performance.

## 2. Analysis of Brake Noise through Finite Element Method

For a general equation of vibration system, the equation of vibration can be expressed as follows:

$$M\ddot{x} + C\dot{x} + Kx = F \tag{1}$$

where $M$, $C$, and $K$ are the mass, damping, and stiffness matrices of the vibration system, respectively, and $x$ is the displacement matrix. For a brake system, the force matrix ($F$) is the variable of friction force at the lining-disk interface, which can be expressed as follows [11]

$$F = \mu K_f x \tag{2}$$

where $\mu$ is the friction coefficient between the brake lining and brake disc and $K_f$ is the friction stiffness matrix. Combining Eq. (1) and (2) yields the following homogeneous Equation [12]:

$$M\ddot{x} + C\dot{x} + (K - \mu K_f)x = 0 \tag{3}$$

When solving the resulting equation, the solution to the equation takes the form of an Eigenvalue, which is a complex number consisting of real part and imaginary part.

$$\lambda = \alpha \pm \omega i \tag{4}$$

Where $\lambda$ is the eigenvalue, $\alpha$ is the real part of the eigenvalue and represents real frequency, and $\omega$ is the imaginary part of the eigenvalue and represents the damped frequency. The real part has two values, a positive value and a negative value. The positive value indicates the instability. When the $\alpha$ value is positive, the system becomes unstable and makes the brake squeal. While the system is stable when the $\alpha$ value is negative [13].

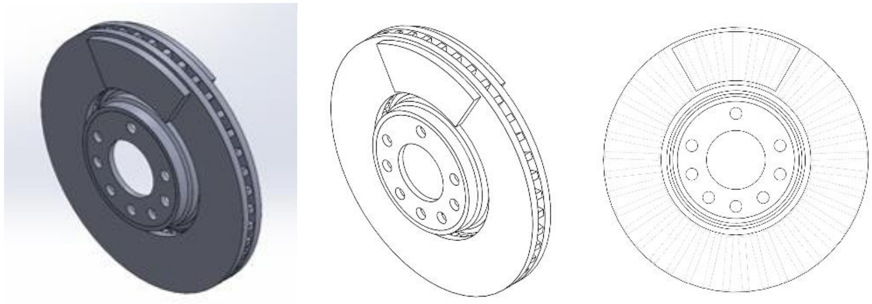
## 3. Modelling the Brake Disc and Pads

In this work, there were two steps. In the first step: three models of brake discs: ordinary brake disc and 2 drilled brake discs were studied for this work. The ordinary model was based on a commercial brake model [14]. Both models of drilled brake discs are based on the model of ordinary brake disc. The difference between both drilled brake discs was the number of drilled holes: 60 and 30 holes. In the second step, a model of brake disc was created and compared with Savant's brake model [4] to verify the study. In this Savant's work, a comparison was made between the analytical results and the experimental results. All models were analyzed in ANSYS Program.

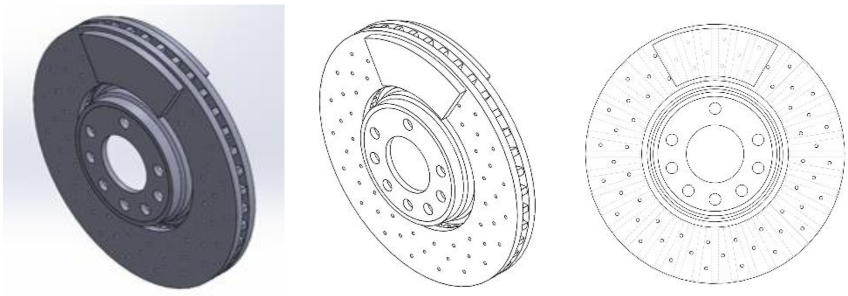
### *3.1 Modelling*

*(a) Ordinary Brake Disc*
The brake disc had a thickness of 9 mm, and the brake pads had a thickness of 10 mm. The inner diameter of the disc was 178 mm and the outer diameter was 314 mm. The inside of each brake disc had 40 cooling Fins.

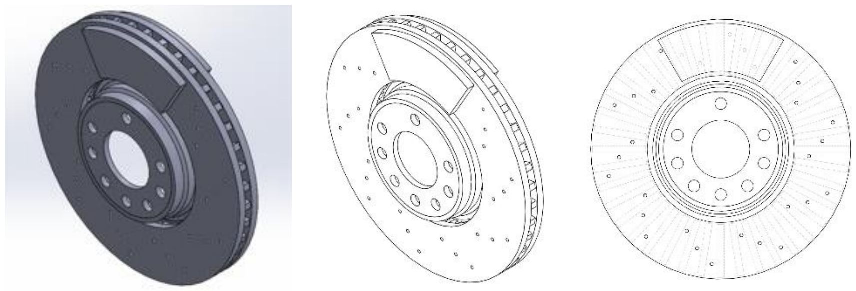
*(b) Type of two drilled Brake Discs: Type I and Type II*
There were 2 models of drilled brake discs: Type I and Type II. The ordinary brake disc was made with 60 and 30 holes as Type I and Type II, respectively. For drilling conditions, the diameter of drilled holes was 4 mm and all holes were not drilled through the cooling fins as shown in Fig. 1.




(a) Ordinary Brake Disc




(b) Drilled Brake Disc with 60 holes as Type I




(c) Drilled Brake Disc with 30 holes as Type II

**Fig. 1.** Three models of brake disc.

*3.2 Meshing on Models*

All three models as shown in Fig. 2 were meshed in ANSYS program. However, the mesh construction can affect the analyzed results [15]. The mesh metrics tool was used to check mesh quality. The skewness value of mesh metrics was in the range of 0-0.25 and the orthogonal quality of mesh metrics was in the range of 0.95-1 [16]. The model of ordinary brake disc had 49,824 elements and 243,454 nodes. The average value of skewness mesh metrics of 0.120 was excellent. The average value of orthogonal quality mesh metrics of 0.967 was excellent.

The model of Type I drilled brake disc had 72,315 elements and 244,005 nodes. The average value of skewness mesh metrics of 0.089 was excellent. The average value of orthogonal quality mesh metrics of 0.911 was excellent. The model of Type II drilled brake disc had 78,471 elements and 255,671 nodes. The average value of skewness mesh metrics of 0.088 was excellent. The average value of orthogonal quality mesh metrics of 0.912 was excellent.

*3.3 Meshing*

Table 1 shows the material properties of the ordinary brake disc, Type I and Type II of drilled brake discs. The material properties of brake pad are shown in Table 2.

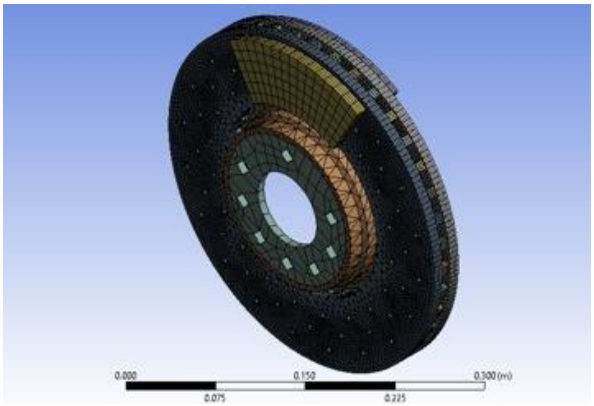
**Table 1:** Material of brake disc.

| | |
|---|---|
| Young's Modulus | $1.01 \times 10^{11}$ Pa |
| Density | 7293 kg/m$^3$ |
| Poisson's Ratio | 0.3 |

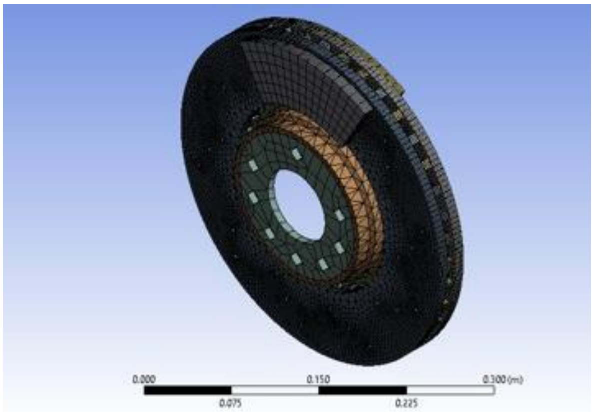
**Table 2:** Material of brake pad.

| | |
|---|---|
| Young's Modulus | $2.8 \times 10^{10}$ Pa |
| Density | 2700 kg/m$^3$ |
| Poisson's Ratio | 0.3 |


(a)




(b)




(c)

**Fig. 2.** Finite element Mesh on three brake disc models (a) Ordinary Brake Disc, (b) Drilled Brake Disc with 60 holes as Type I, (c) Drilled Brake Disc with 30 holes as Type II.

## 4. Analysis and Solution Controls

This section discusses the simulation of the brake model using Finite element method through ANSYS program to analyze squealing noise. In the first step, the models of analyzed simulation had 3 models: Ordinary brake model, Type I drilled brake model and Type II drilled brake model. In the second step, Savant's brake model was used to verify the study. For Savant's brake disc model [4], there were 11,473 elements and 60351 nodes in the FE model while there were 11,168 elements and 61,890 nodes for this work. The average value of skewness mesh metrics of 0.0.007 was excellent. The average value of orthogonal quality mesh metrics of 0.985 was excellent.

*4.1 Contact Surface*

The contact configuration of the models included a friction contact and a bone contact. Friction contact was defined as the area where the brake pads and brake disc came into contact. As shown in Fig. 3, the coefficient of friction was set to 0.3. The bond contacts were defined to the rest or all parts of the brake disc. Pinball region designation improved the accuracy of fixing that point if the contact was enabled for the pinball region. If the contact distance was less than the size of the pinball radius then the contacts were immediately bonded together. The pinball was also used in friction contacts if the two elements of the contact were closed to each other in pinball distance, resulting in more contact validation and greater accuracy in the program.

| Details of "Frictional - pad brembo To Part1[2]" | |
|---|---|
| Scoping Method | Geometry Selection |
| Contact | 1 Face |
| Target | 1 Face |
| Contact Bodies | pad brembo |
| Target Bodies | disc_front |
| Protected | No |
| **Definition** | |
| Type | Frictional |
| Friction Coefficient | 0.3 |
| Scope Mode | Manual |
| Behavior | Auto Asymmetric |
| Trim Contact | Program Controlled |
| Suppressed | No |
| **Advanced** | |
| Formulation | Augmented Lagrange |
| Small Sliding | Program Controlled |
| Detection Method | Program Controlled |
| Penetration Tolerance | Program Controlled |
| Elastic Slip Tolerance | Program Controlled |
| Normal Stiffness | Program Controlled |
| Update Stiffness | Program Controlled |
| Stabilization Damping Factor | 0. |
| Pinball Region | Radius |
| Pinball Radius | 2.e-003 m |
| Time Step Controls | None |
| **Geometric Modification** | |
| Interface Treatment | Adjust to Touch |
| Contact Geometry Correction | None |

| Details of "Multiple Selection" | |
|---|---|
| **Scope** | |
| Scoping Method | Geometry Selection |
| Contact | |
| Target | |
| Contact Bodies | |
| Target Bodies | |
| Protected | No |
| **Definition** | |
| Type | Bonded |
| Scope Mode | Automatic |
| Behavior | Symmetric |
| Trim Contact | Program Controlled |
| Trim Tolerance | 1.1187e-003 m |
| Suppressed | No |
| **Advanced** | |
| Formulation | MPC |
| Detection Method | Program Controlled |
| Constraint Type | Program Controlled |
| Pinball Region | Radius |
| Pinball Radius | 2.e-003 m |
| **Geometric Modification** | |
| Contact Geometry Correction | None |
| Target Geometry Correction | None |

**Fig. 3.** Details of Frictional and Boned contacts.

*4.2 Boundary Conditions and Loading*

As shown in Fig. 4, the letter A was the displacement or movement of the outboard or inboard pad onto the brake disc. The letter B and C were the hydraulic pressure applied on the front and back surfaces of the brake pads to squeeze these pads on the brake disc. The applied pressure was 10 MPa on the front and back surfaces of the brake pads. Finally, the letter D was fixed supports on the shaft mounted holes and bolt holes. The rotational speed of brake disc was set to 2 rad/s. The unsymmetric stiffness matrix was generated by using "NROPT, unsym" command.

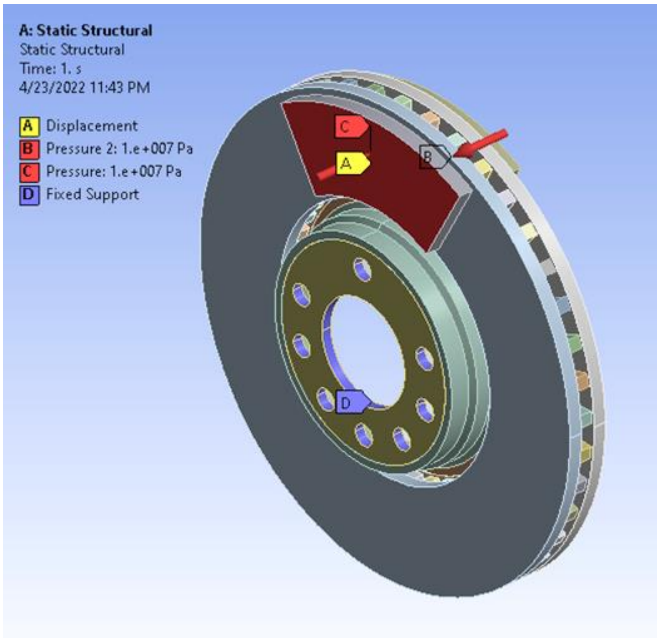



**Fig. 4.** Boundary condition applied on each model of brake disc.

*4.3 Solver Setting*

In the modal analysis, the unsymmetric solver type was determined and the frequency range of 1000 – 16000 Hz was considered.

## 5. Result and Discussion

The results of the analysis were divided into two parts: imaginary and real part. This was due to the metric asymmetry caused by the brake pads and brake discs. The imaginary part was the damped frequency and the real part was the real frequency. The real part indicated the stability of the system. According to Complex eigenvalue analysis method, when the real part is positive, it indicates that the system is unstable mode.

*5.1 Result*

In the second step, the brake disc model in this work was compared and the results were verified with Savant's research [4]. The frequency range of 1,000-16,000 Hz was considered. In this work, it was found that the squealing noise occurred for the 1$^{st}$ mode shape with a frequency of 6442.6 Hz as shown in Fig. 5(a), and the experiment result [4] was 6440 Hz as shown in Fig. 5(c) or an error of 0.040%. Meanwhile, the 1$^{st}$ mode shape of Savant's work using the friction coefficient of 0.3 was 6470.24 Hz as shown in Fig. 5(b), or an error of 0.4696% compared with the experiment result as shown in Fig. 5(c).

In the first step, the model of ordinary brake disc, Type I and Type II drilled brake discs were analyzed to find the squealing noise in the frequency range of 1000-16000 Hz. As shown in Fig. 6(a), the model of ordinary brake disc was found that the squealing noise occurred in 4 mode shapes at the frequencies of 4841.6, 9134.8, 12678 and 15386 Hz, respectively. Fig. 6(b) shows the result of the squealing noise of the Type I drilled brake disc model occurring in 4 mode shapes at the frequencies of 4712.7, 7892.5, 9041, and 14963 Hz, respectively. While the Type II drilled brake disc model was found that the squealing noise occurring in 6 mode shapes at the frequencies of 4729, 6161, 9093.1, 11313, 13365 and 15016 Hz, respectively, as shown in Fig. 6(c).

| | Mode | Frequency [Hz] | Stability [Hz] |
|---|---|---|---|
| 15 | 15. | 3483.8 | 0. |
| 16 | 16. | 4054.3 | 0. |
| 17 | 17. | 4056.4 | 0. |
| 18 | 18. | 4406.1 | 0. |
| 19 | 19. | 4511.7 | 0. |
| 20 | 20. | 5196.1 | 0. |
| 21 | 21. | 5996.7 | 0. |
| 22 | 22. | 6107.6 | 0. |
| 23 | 23. | 6442.6 | -21.626 |
| 24 | 24. | 6442.6 | 21.626 |
| 25 | 25. | 6653.6 | 0. |
| 26 | 26. | 6654.1 | 0. |

(a)

| $\mu$ | Mode 21 | | Mode 22 | |
|---|---|---|---|---|
| | **RFRQ (Hz)** | **IFRQ (Hz)** | **RFRQ (Hz)** | **IFRQ (Hz)** |
| 0 | 6469.72 | 10.5827 | 6469.72 | -10.5827 |
| 0.3 | 6470.24 | 21.9003 | 6470.24 | -21.9003 |

(b)

| Experimental Result | FEA Result | %Error |
|---|---|---|
| 6440 Hz | 6470.24 Hz | 0.4696 |

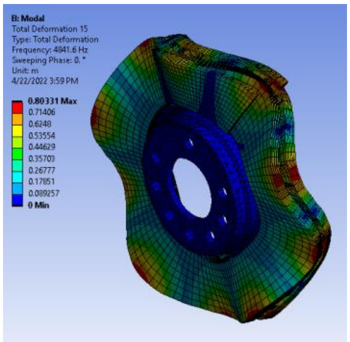
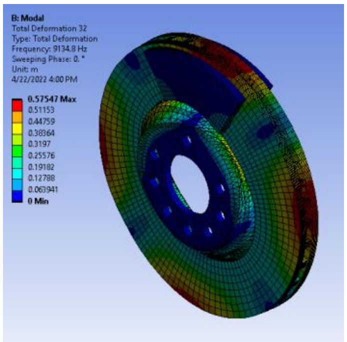
(c)

**Fig. 5.** Results of compared model: (a) FEA result in this work: Eigen frequencies for mode 23 and 24, (b) Eigen frequencies for mode 21 and 22 with increasing friction coefficient of 0 and 0.3 in Ref. [4], (c) Experimental and FEA results using friction coefficient of 0.3 in Ref. [4].

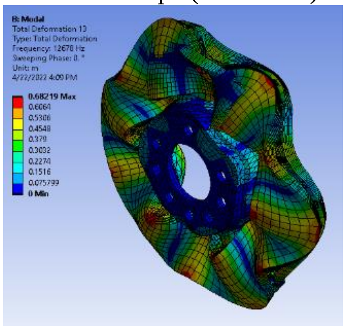
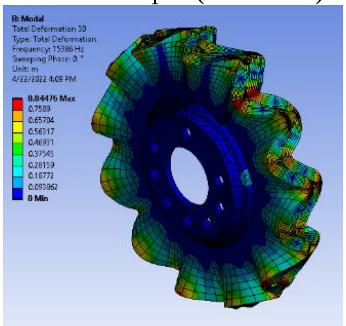
*5.2 Discussion*

This study confirmed the accuracy of the analysis by comparing the result with Savant's study. The result of his study for the squealing noise was occurred at a frequency of 6470.24 Hz or an error of 0.4696% while the frequency of this work was 6442.6 Hz or an error of only 0.04%. In addition, it was found that the mesh structure and material properties of the model directly affected the accuracy of the results. It was not just the number of nodes and elements that affected accuracy, but also the shape and appearance of the mesh. The equilateral mesh gave the most detailed results, such as square and equilateral triangles.

Of the three models, the one with the strongest structure was the model of the ordinary brake disc. Type II drilled brake discs (30 holes) was stronger than Type I drilled brake disc (60 holes). The simulations of three brake disc models revealed that three similar instability mode shapes occurred as shown in Fig. 7. As shown in Table 3 and 4, the ordinary brake disc model became unstable at a frequency of 4841.6 Hz in case of the 1$^{st}$ mode shape. The frequency of squeal noise of Type I drilled brake disc was reduced to 4712.7 Hz or a difference of 2.677 % compared with the ordinary brake disc model. The frequency of squeal noise of Type II drilled brake disc was reduced to 4729 Hz or a difference of 2.326 %. For the 2$^{nd}$ mode shape, the ordinary brake disc model became unstable at a frequency of 9134.8 Hz. The frequency of squeal noise of Type I drilled brake disc was reduced to 9041 Hz, or a difference of 1.027 %. The frequency of squeal noise of Type II drilled brake disc was reduced to 9093.1 Hz or a difference of 0.456 %.

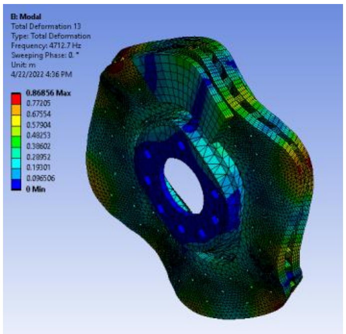
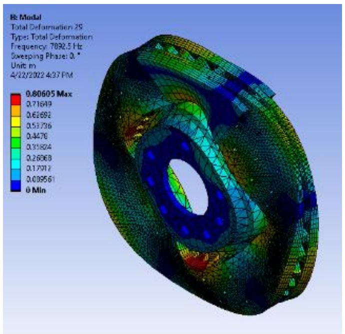
 

1$^{st}$ mode shape (4841.6 Hz) 2$^{nd}$ mode shpae (9134.8 Hz)

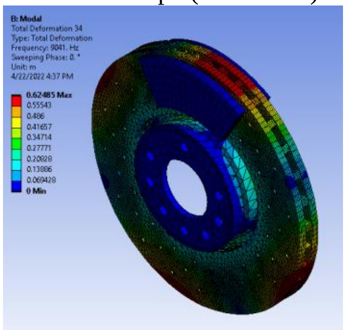
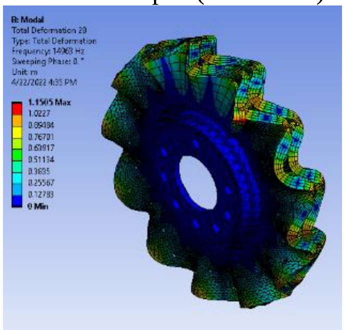
 

3rd mode shape (12678 Hz) 4th mode shpae (15386 Hz)

(a) Ordinary brake disc model

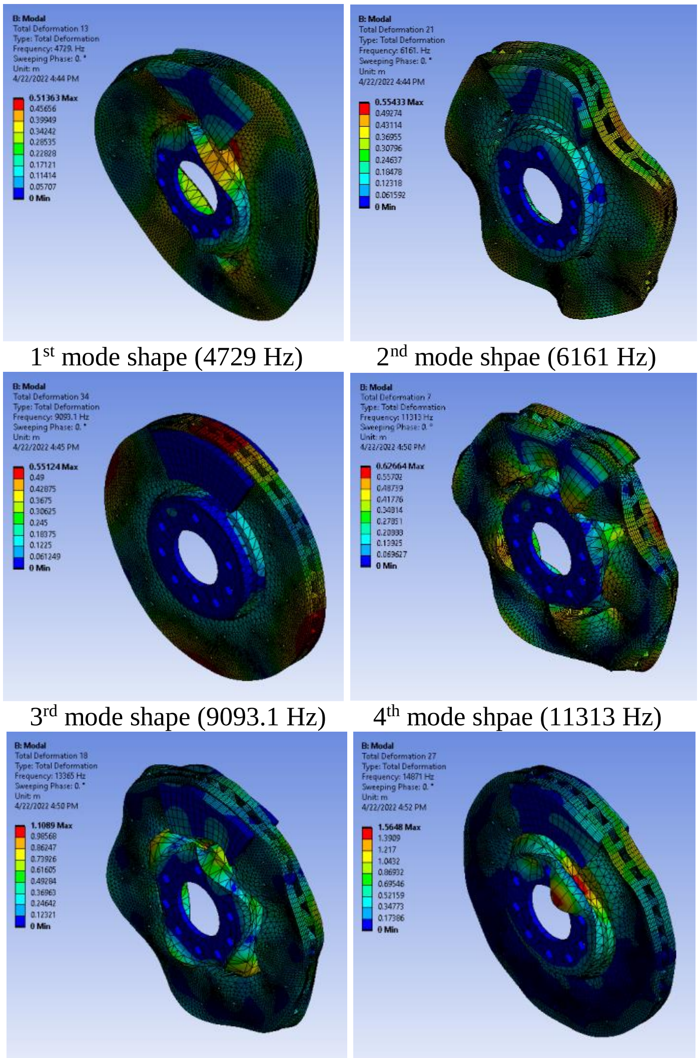
1$^{st}$ mode shape (4712.7 Hz) 2$^{nd}$ mode shpae (7892.5 Hz)

3$^{rd}$ mode shape (9041 Hz) 4$^{th}$ mode shpae (14963 Hz)

(b) Type I drilled brake disc model

**Fig. 6.** Mode shape for squealing noise on (a) Ordinary brake disc model, (b) Type I drilled brake disc model, and (c) Type II drilled brake disc model.



1st mode shape (4729 Hz) 2nd mode shpae (6161 Hz)

3rd mode shape (9093.1 Hz) 4th mode shpae (11313 Hz)

5th mode shape (13365 Hz) 6th mode shpae (14871 Hz)

(c) Type II drilled brake disc model

**Fig. 6. (continued)** Mode shape for squealing noise on (a) Ordinary brake disc model, (b) Type I drilled brake disc model, and (c) Type II drilled brake disc model.

In case of the 3rd mode shape, the ordinary brake disc model become unstable at a frequency of 15386 Hz. The frequency of squeal noise of Type I drilled brake disc was reduced to 14963 Hz, a difference of 2.749 %. The frequency of squeal noise of Type II drilled brake disc was reduced to 15016 Hz or a difference of 2.405 % as shown in Table 3 and 4.

As shown in Fig. 7, Table 3 and 4, the comparison of three unstable mode shapes, it was found that Type I drilled brake disc model produced lowest frequencies. For Type II drilled brake disc model, the frequency was higher than the Type I drilled brake disc model, but still had a lower frequency than the ordinary brake disc model. The ordinary brake disc had the highest stiffness corresponding to the high frequency. Type II had more stiffness than Type I then the frequency of Type II was higher than the frequency of Type I. Therefore, these results lead to the conclusion that the more stiffness a structure is, the higher the frequency.

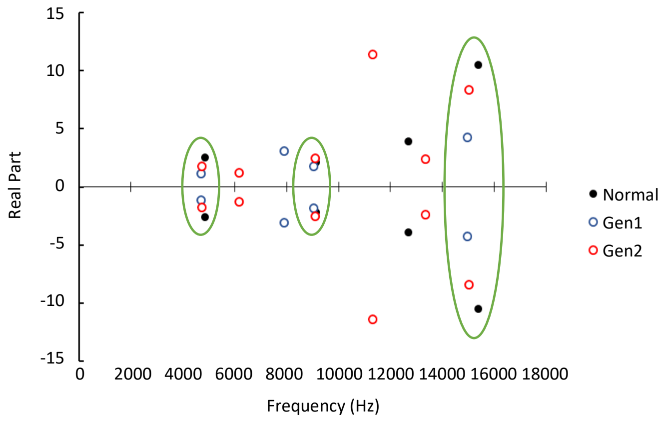



**Fig. 7.** Comparison of results of three difference brake models

**Table 3:** Mode shape that produces similar frequencies

| Brake Model | Mode shape that produces similar frequencies | | |
|---|---|---|---|
| | **Frequency of 1st mode shape** | **Frequency of 2nd mode shape** | **Frequency of 3rd mode shape** |
| Ordinary brake disc | 4841.6 Hz | 9134.8 Hz | 15386 Hz |
| Type I Drilled brake disc | 4712.7 Hz | 9041 Hz | 14963 Hz |
| Type II Drilled brake disc | 4729 Hz | 9093.1 Hz | 15016 Hz |

**Table 4:** The percentage difference in frequency compared to normal disc brake models.

| Brake Model | The percentage difference in frequency compared to normal disc brake models | | |
|---|---|---|---|
| | **Frequency of 1st mode shape** | **Frequency of 2nd mode shape** | **Frequency of 3rd mode shape** |
| Type I Drilled brake disc | 2.677 % | 1.027 % | 2.749 % |
| Type II Drilled brake disc | 2.326 % | 0.456 % | 2.405 % |

## 6. Conclusion

From the problem of braking systems that create noise while braking. The noise output can be divided into several levels according to the principle of sound. e.g., groans, squeals, and chatter, etc. The squealing noise is the most annoying and avoidable sound. The noise is a direct problem to drivers and causes them to worry about the performance of their vehicle's brake. Therefore, it is necessary to deal with the problem of squealing noise. This study was brake disc simulation to compare the squealing noise produced by an ordinary brake disc and two drilled brake discs. The simulation results reveal the occurrence of squealing noises on brake discs with different structures and characteristics. The results can be used as a guideline for designing brake discs to further reduce the squealing noise generated by braking.

## Acknowledgement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## Nomenclature

| | |
|---|---|
| C | damping matrix |
| $F$ | force matrix |
| $i$ | $\sqrt{-1}$ |
| $K$ | stiffness matrix |
| $K_f$ | friction stiffness matrix |

| | |
|---|---|
| M | mass matrix |
| $x$ | displacement matrix |
| $\alpha$ | real part of the eigenvalue |
| $\omega$ | imaginary part of the eigenvalue |
| $\mu$ | fiction coefficient between brake lining and disk |

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