Journal of Theoretical and Applied Vibration and Acoustics 9(1)14-28 (2023)

I S A V

Journal of Theoretical and Applied Vibration and Acoustics

journal homepage: http://tava.isav.ir

# Experimental investigation of natural frequency uncertainties in disc brakes in contact with pads

**Fahimeh Noohi[a]*, Amin Nikoobin[b], Mohammad Reza Ashory[c]**

[a]*Ph.D. Candidate, School of Mechanical Engineering, Semnan University,Semnan, Iran.*

[b]*Assistant Professor, Mechanical Engineering Department, Semnan University, Semnan, Iran.*

[c]*Associate Professor, School of Mechanical Engineering, Semnan University, Semnan, Iran.*

**ARTICLE INFO**

*Article history:*
Received 06 October 2023

Received in revised form
11 December 2023

Accepted 30 December 2023

Available online 5 January 2024

*Keywords:*
Brake disc,

Uncertainty,

Brake squeal,

Dynamic analysis.

**ABSTRACT**

The brake system is a critical component of any vehicle, ensuring safety and control while driving. Disc brake stability, understanding and optimizing the contact area between the disc and the pad is crucial for improving braking performance and preventing issues like brake squeal. Several sources of uncertainty in a braking system can make it difficult to completely eliminate problems like squealing. This paper attempts to study the uncertainty of natural frequencies by changing the contact area between the disk and the pads with an experimental investigation. To this end, three stages of testing have been performed. An experimental modal study under free disk conditions has been conducted to identify material properties, followed by a modal examination of a clamped brake system without pads to determine boundary conditions. Then, modal tests have been performed to investigate the response uncertainties by changing the contact surface between the disc and pads. This research approach can provide valuable insights into the factors that influence the performance and noise characteristics of the braking system, particularly in relation to brake squeal.

*© 2023 Iranian Society of Acoustics and Vibration, All rights reserved.*

## 1. Introduction

The development of the automobile industry has seen significant advancements in various components, with the brake system being a critical one. The brake system plays a pivotal role in ensuring the safety and functionality of a vehicle, and it has evolved significantly since the first

* Corresponding author:
*E-mail address:* fahime_noohi@semnan.ac.ir (F. Noohi)

http://dx.doi.org/10.22064/tava.2024.2013013.1229

cars were built. Indeed, brake noise is a concern for both drivers and passengers. It can be annoying and distracting; in some cases, it may even indicate a potential problem with the brake system. A wide array of brake noise exists in the literature and can be categorized into low-frequency and high-frequency noise. The low-frequency noise includes groan, judder, and moan. Squeal is a high-frequency noise and probably the most annoying to vehicle passengers and the environment [1,2]. Brake squeal can occur in various situations and is often attributed to factors related to the brake pad material, components of the brake system, and driving conditions. Indeed, brake squeal can be a common reason car owners visit car dealerships or repair shops frequently. While brake squeal is often a nuisance rather than a safety concern, it can be bothersome enough to prompt drivers to seek solutions.

Brake noise, including issues like brake squeal, is a global concern in the automotive industry, affecting automobile manufacturers and component companies worldwide. It's a widespread issue that has led to significant costs and efforts to address it. Research in the field of brake noise has led to high-level projects at prestigious universities and the development of advanced testing and measurement devices. However, it's crucial to emphasize that not every instance of brake noise indicates a fault or safety concern.

Brake squeal is a complex and often unpredictable phenomenon influenced by a variety of factors, including temperature, pressure, humidity, friction, and rotor speed. Complex eigenvalue and transient analysis are two deterministic numerical methods and typically rely on certain input parameters and conditions. Performing thousands of brake analyses in different working conditions can be impractical due to the time and computational resources required for each analysis. Analyzing brake systems under a wide range of conditions can be a time-consuming and resource-intensive process. Brake system design and analysis often involve deterministic procedures that rely on specific input parameters, such as material properties, component geometry, and assembly conditions. However, in reality, these factors can vary from one brake system to another, and investigating all these factors with deterministic methods is impossible. Incorporating the effects of various sources of uncertainty in a braking system is challenging but essential for effectively addressing issues like brake squeal. Uncertainty propagation methods are valuable tools in this regard. Brake squeal remains a complex and persistent problem, and uncertainty is indeed a key element in understanding why it hasn't been completely solved.

The focus on uncertainty in brake squeal research is indeed a critical development in addressing this complex issue. Researchers often grapple with the challenge of balancing the need for accurate and comprehensive brake noise models with the practical constraints of cost and computation time. This has led to the utilization of simplified low-order models in the study of brake squeal. Butlin and Woodhouse [3] focused on the effects of uncertainty on the instability of a simple brake system model. Their approach, which used a simplified pin-on-disk model with point contact for frictional interaction, aimed to understand the sensitivity and uncertainty associated with predictions of brake system behavior. In another work [4], they used the same analytical model and laboratory settings to calculate a large volume of brake screeching events, followed by the analysis of the dynamic properties of the system under the influence of uncertainties. They experimentally proved that the friction coefficient alone is insufficient to start the squealing. Gauger et al. [5] adopted a non-probabilistic method for complex eigenvalue analysis of a simplified brake model, including rotor, pads, and caliper. The effects of two uncertain parameters, friction coefficient, and contact parameter, have been investigated in this study. Culla and Massi [6] performed an uncertainty

analysis of contact instabilities leading to noise in brake systems. Squeal noise is highly unpredictable due to uncertainties in system parameters. They presented a simplified finite element model consisting of a disc, a pad, and a support. Research conducted by Oberst and Lai [7] presented a statistical experimental method for investigating brake squeal. Their study evaluated the effects of various geometric features, specifically pad grooves, on a noise index associated with brake squeal. The use of non-linear statistical analysis tools to analyze disc brake squeal represents an important advancement in brake squeal research.

Tison et al. [8] aimed to improve the predictability of brake squeal simulations by introducing the concept of uncertainty. They studied a brake model including disc, pads, caliper and suspension links. They represented a significant step forward in improving the predictability of brake squeal simulations by considering the uncertainty associated with surface topography. This article can determine the probability of unstable vibrations leading to brake squealing due to changes in different parameters. The research conducted by Iroz et al. [9] focused on transient simulation and uncertainty analysis of braking systems using the multi-member system approach with fuzzy parameters. This research combines finite element modeling with complex eigenvalue analysis to study the dynamic behavior of an industrial braking system. The brake system includes a rotor, two pads with a support plate, a fixed caliper, and four hydraulic pistons. Zhang et al. [10] considered the uncertainty to predict the tendency of disc brake squeal. Their previous work [11,12] focused on predicting the tendency for brake squeal in a complete brake system using finite element modeling, uncertainty analysis, and the investigation of uncertainties in various parameters. Renat et al. [13] conducted an experimental investigation to quantify the uncertainty in brake squeal analysis. This study aims to improve the correlation between experimental and numerical predictions of unstable frequencies for car braking systems considering uncertainty. Nobari et al. [14] represented an approach to quantify the uncertainty associated with brake squeal instability using alternative modeling and statistical methods. The study aims to provide valuable insights into the sources and magnitudes of uncertainty, ultimately contributing to improved brake system designs and reduced noise in automotive braking systems.

The research described in the paper investigates uncertainties in the dynamic responses of a braking system through experimental studies. Other conditions, such as braking pressure, pad material, and boundary conditions, are held constant while the contact surface between the disc and the pads changes. At first, two experimental stages were considered: The experimental modal study of a free disk is used to identify material properties, and the modal examination of a clamped brake system without pads is used to identify boundary conditions. By understanding the effects of these cases, the focus of the study narrows down to the uncertainty in the responses of the structure resulting from variations in the contact section between the brake disc and pads and the braking pressure. Then, the statistical distributions obtained from the experimental studies are presented. At the end, the discussion and conclusions are discussed.

## 2. Experimental model

In this section, the experimental model has been examined in three main stages. Figure 1 presents a schematic diagram of the actuation loop.

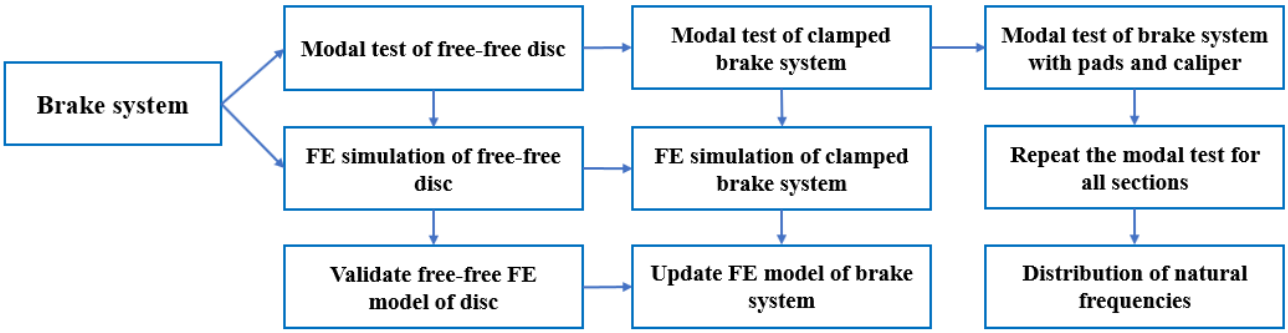


**Fig. 1.** Schematic diagram of the actuation loop

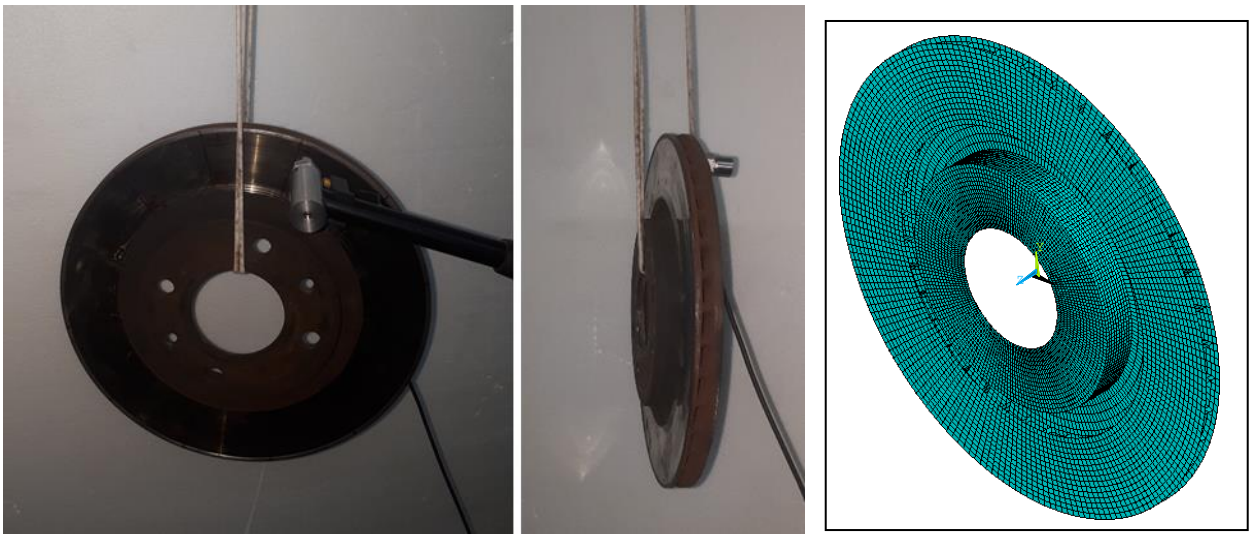

**Fig. 2**. Left) Experimental modal test, Right) FE simulation for car brake disc with free-free boundary condition.

## *2.1. Free-free disk*

When considering the uncertainty of natural frequencies in a structure like a brake disc in contact with pads, it's essential to have a profound understanding of the material properties and boundary conditions. By applying the correct parameter values of the material properties and boundary conditions in the modeling, the uncertainties in these two parts can be minimized, and the uncertainties in the different contact sections of the brake disc with pads can be addressed. Therefore, the brake disc in the free-free boundary condition, shown in the left panel of Figure 2, has been experimentally studied to obtain the mechanical properties of the brake disc. The equipment used in the modal test is a modal hammer (8202BK) to apply the excitation force, an 8200BK force transducer with a 2647A amplifier to calculate the applied force, and a DJB A120V accelerometer to calculate the response of the structure to the input excitation force and a

BK3560D analyzer. The frequency range is 0 to 3200 Hz. The test was performed, and the force and acceleration signals were measured. The frequency response functions (FRF) were calculated using the analyzer and Pulse8 software, and the results are shown in Figure 3. After the test, the received frequency response functions have been sent to Icats software and analyzed. As a result of this analysis, the natural frequencies and mode shapes of the disk have been obtained. The obtained natural frequencies are reported in Table 1.

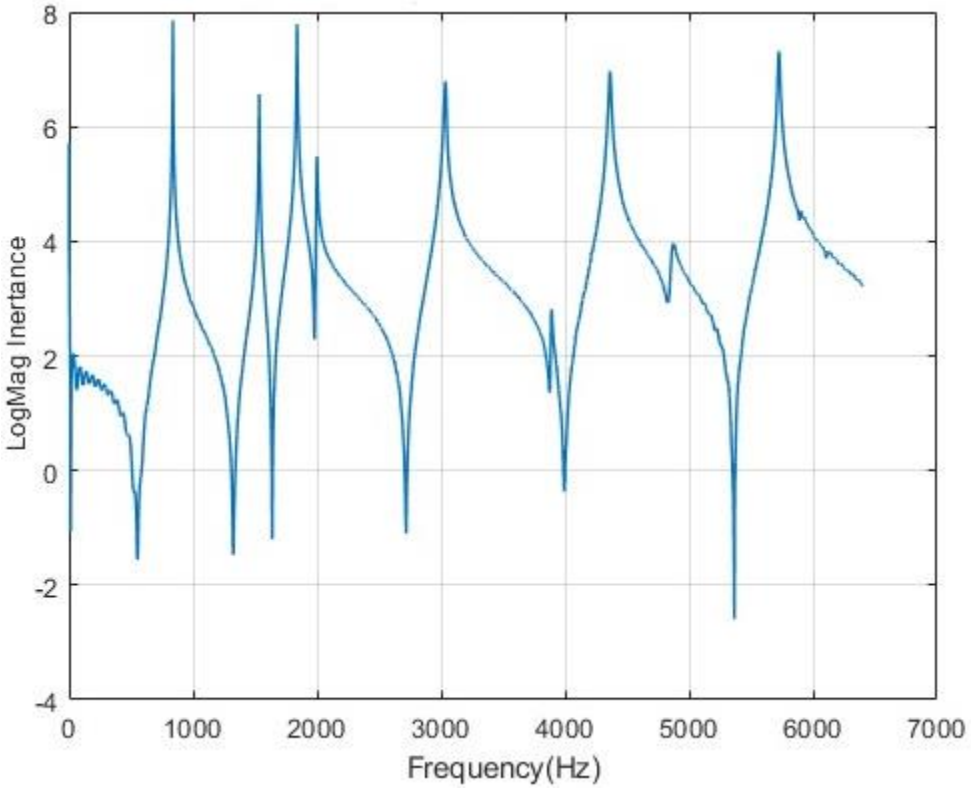


**Fig. 3.** Frequency response functions for car brake disc with free-free boundary condition.

Next, by measuring the dimensions of the disc and considering the approximate material properties, the simulation of the disc is performed in ANSYS APDL finite element software. The appropriate mesh size is selected for the elements by performing mesh convergence analysis, as shown in Figure 2(right). The responses of the structure are obtained by choosing the modal solution. An updating process is performed with the help of MATLAB software, and material properties, including density and elastic modulus, are updated as $\rho$=7793 Kg/m$^3$ and E=194 Gpa. Updated finite element model responses are presented in Table 1. The result of comparing the modal responses of the finite element model with the experimental test is also presented in this table. Low errors indicate that the updating process was successful. The first nine mode shapes of the updated structure are shown in Figure 4. The mode shapes resulting from finite element modeling are also consistent with the test. The natural frequencies of the FE model of the disc with initial parameters, updated parameters, and their differences with experimental values are listed in Table 1. The updating process decreased the frequency of errors, which indicates an increase in the accuracy of the model.

In the following, the convergence trend of the updated parameters and the objective function is presented in Figure 5. The objective function is defined as the following equation:

$$Objective\ Function = \log\big(norm(Er)\big) \tag{1}$$

$$Er = (w_a^2 - w_e^2)/w_e^2 \tag{2}$$

As can be seen from the graphs, the updating process has progressed well, and the objective function has reached its lowest value. The result of the updating process is a correct and accurate model of the brake disc that can be used in the following steps.

**Table 1.** Natural frequencies of experimental modal test and FEM of brake disc with free boundary condition

| Mode No | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
|---|---|---|---|---|---|---|---|---|---|
| TEST | 837.51 | 844.77 | 1533.2 | 1837.9 | 1857.81 | 2001.2 | 2019.1 | 3030.3 | 3040.3 |
| Initial FEM | 764.6 | 764.6 | 1416.6 | 1660.1 | 1660.1 | 1840.0 | 1840.0 | 2722.5 | 2792.2 |
| Err (%) | 10.9 | 11.6 | 7.61 | 9.67 | 10.6 | 8.06 | 8.87 | 10.2 | 8.16 |
| Updated FEM | 824.79 | 824.79 | 1565.1 | 1834.0 | 1834.0 | 2032.9 | 2039.9 | 3007.9 | 3084.8 |
| Err (%) | 1.52 | 2.36 | 2.08 | 0.21 | 1.28 | 1.58 | 0.68 | 0.74 | 1.46 |

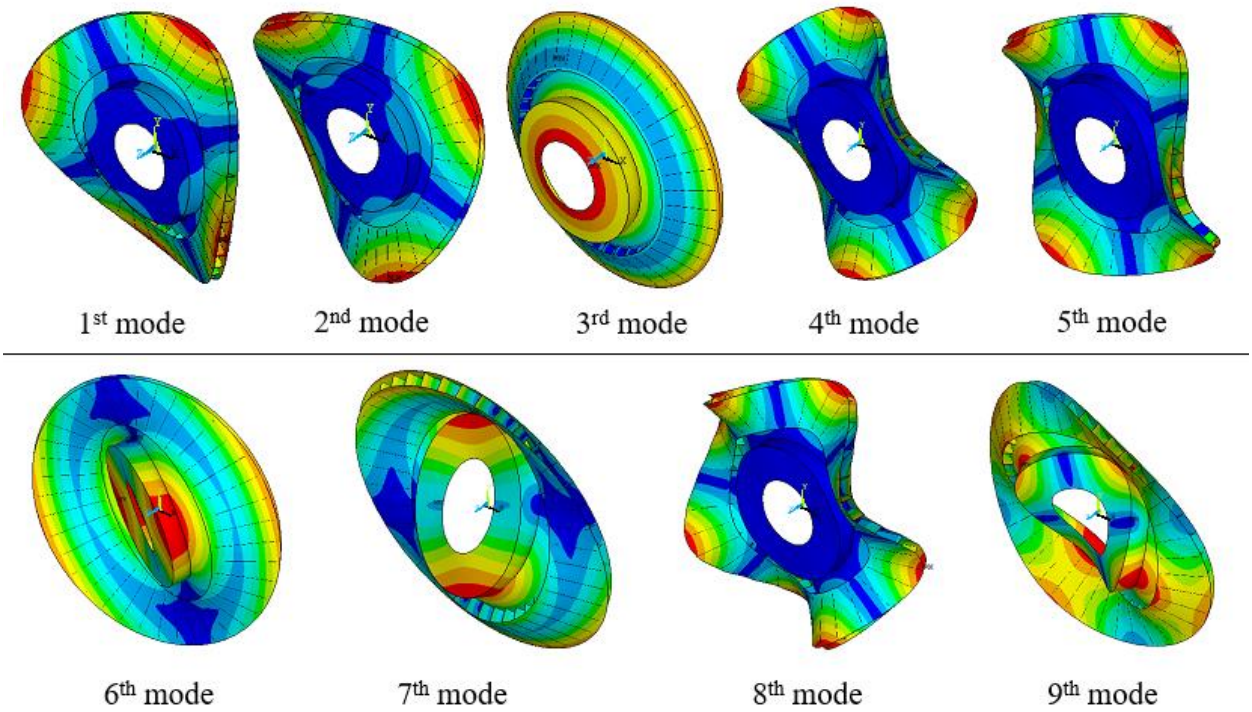


**Fig.4**. The first nine mode shapes of the updated structure in ANSYS

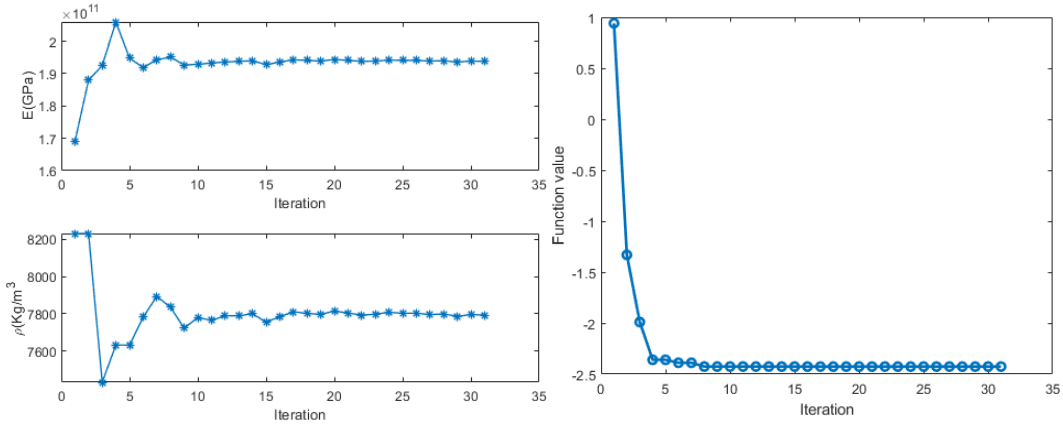


**Fig. 5**. Convergence diagram of left) updated parameters and right) objective function

## *2.2. Brake system*

The experimental setup includes an assembled disc with a pulley and knuckle, as shown in Figure 6. The star screws and the wheel bolts are tightened. The brake system components are shown separately in Figure 7. The knuckle is clamped from one part in a bench vise. In this case, the experimental modal test has been investigated to identify the dynamic behavior of the support and its effect on modal responses. More precisely, this test aims to calculate the contact parameters between the disk and the pulley. The experimental modal test of this model was performed with the help of the equipment described in the previous section. The frequency response functions resulting from the test can be seen in Figure 8. By analyzing the test results in the Icats software, the natural frequencies of the structure are obtained and reported in Table 2.

**Fig. 6**. Experimental modal test of car brake disc assembly with clamped boundary condition.

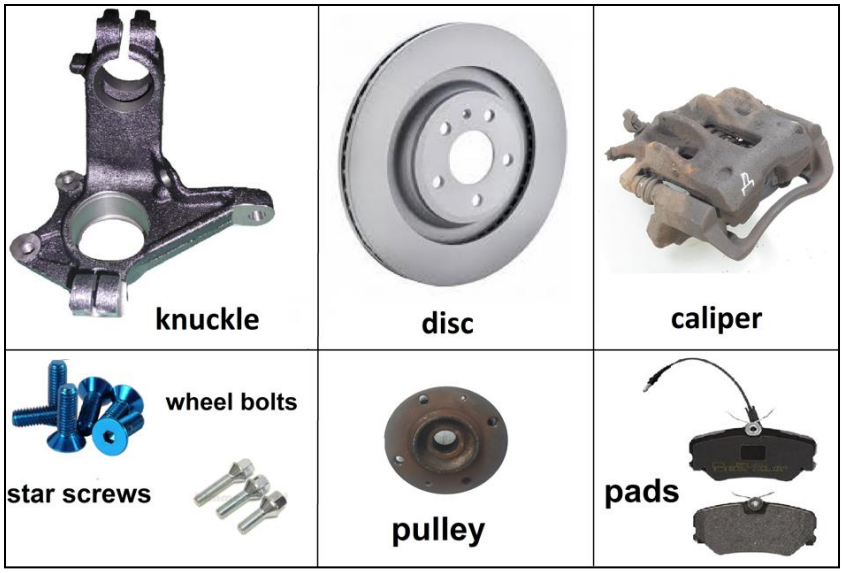


**Fig.7**. Brake system components

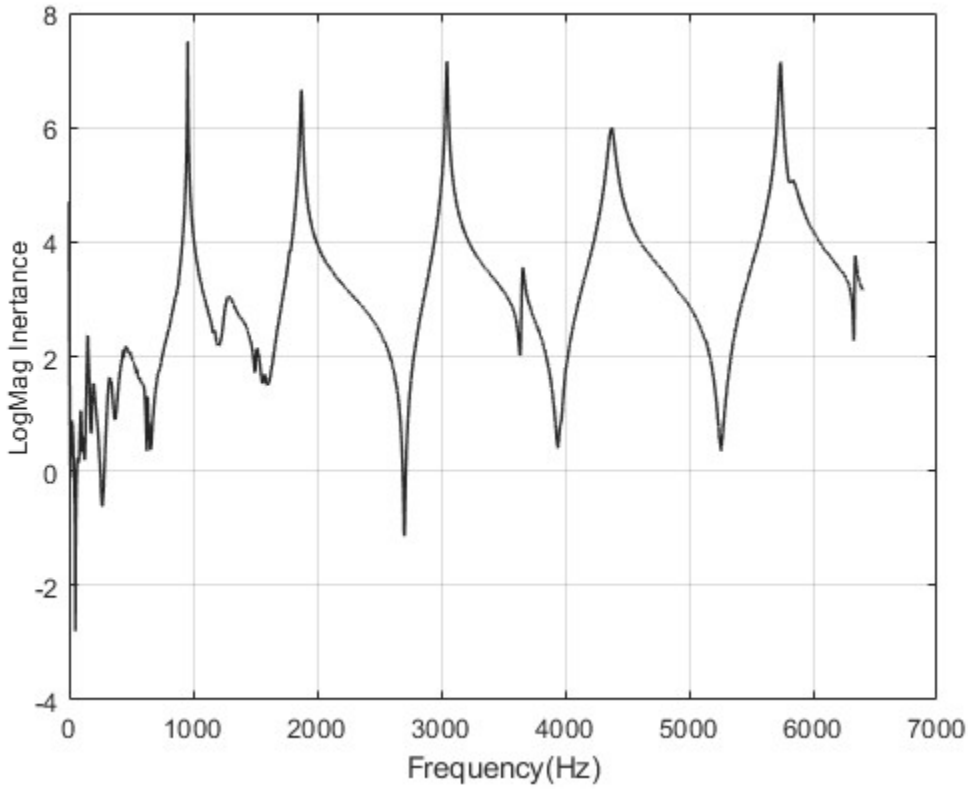


**Fig. 8.** Frequency response functions for car brake disc assembly in clutch mode.

A simplified model of the knuckle is added to the updated FEM model so that its mass and moment of inertia are equal to the real knuckle. The goal of modeling a simplified knuckle is to eliminate possible modes related to knuckle deformation. The contact between the pulley and the inner surface of the brake disc is modeled by defining linear springs in three coordinate directions.

Among various methods presented for modeling the connection between structures, the method presented by [15] is used in this paper. The contact surface between the simplified model of the knuckle and the inner surface of the disk with a large number of spring elements in all three directions is defined in such a way that each element is defined between both nodes of the disk and the knuckle. This modeling can be seen in Figure 9. The responses of the structure are obtained by applying boundary conditions similar to real conditions and performing modal analysis. Similar to the previous part, an updating process is performed, and this time, the connection parameters between the disk and knuckle are identified. In other words, the stiffness of linear springs defined in three coordinate directions is identified and presented in Table 3, and errors are presented in Table 2. Natural frequencies resulting from clamped FE modeling are consistent with the clamped test, and the error between the frequency vectors is deficient. In the following, the convergence trend of the updated parameters and the objective function is presented in Figure 10. The objective function is defined in Equation 1. As can be seen from the graphs, the updating process has progressed well, and the objective function has reached its lowest value. This led to a correct and accurate structure model that is available to integrate the pads into the structure.

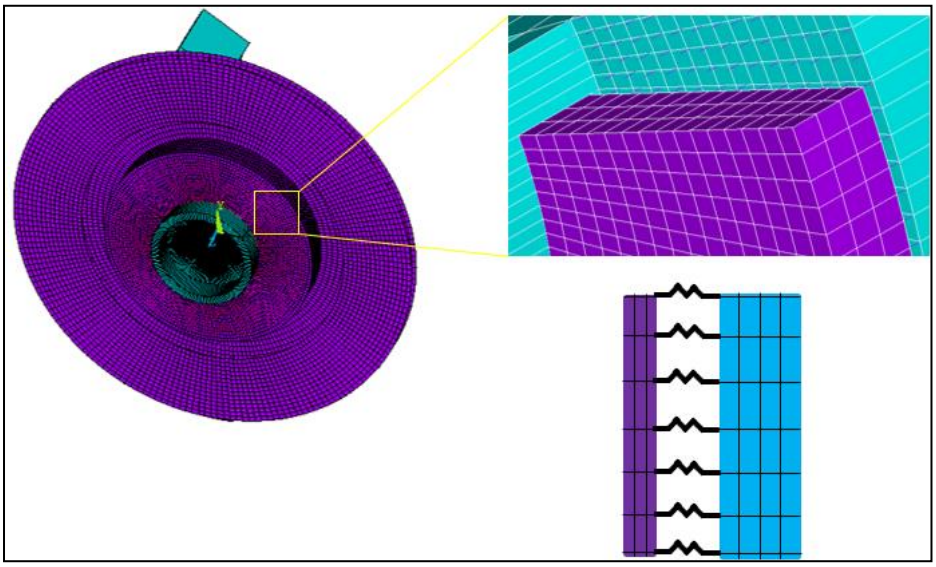

**Fig. 9.** Modeling of contact area between disc and knuckle

**Table 2**. Natural frequencies of modal test and FEM of brake system with clamped boundary condition

| Mode No | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| TEST | 637.54 | 955.89 | 1274.9 | 1505.3 | 1874.5 | 2986.8 |
| Updated FEM | 653.0 | 947.8 | 1270.0 | 1542.0 | 1842.5 | 2971.1 |
| Err (%) | 1.99 | 0.85 | 0.39 | 2.44 | 1.72 | 0.53 |

**Table 3.** Identified contact parameters between the brake disc and pulley

| Parameter | Identified values |
|---|---|
| $K_z$(Normal) | $2.503\times10^7$ N/m |
| $K_y$(Tangential) | $1.671\times10^7$ N/m |
| $K_x$(Tangential) | $1.829\times10^7$ N/m |

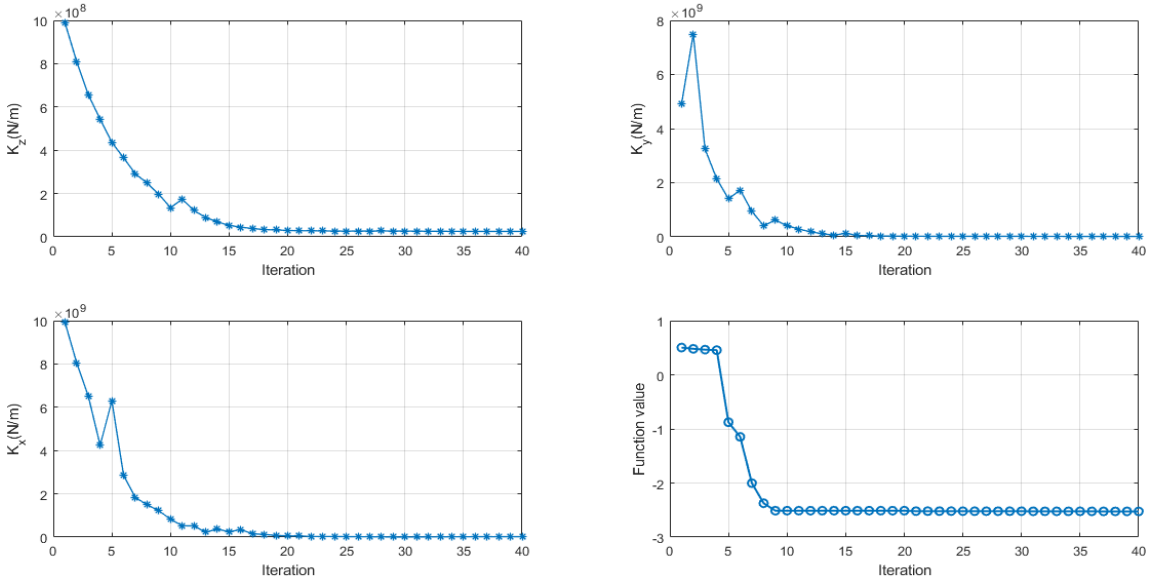


**Fig. 10.** Convergence diagram of updated parameters and objective function

## *2.3. Brake system with caliper and pads*

The experimental test of the entire structure, including the brake system along with the caliper and pads, is examined. The purpose of performing the previous two steps is to create a correct and accurate model for the brake system and eliminate other sources of uncertainty. The caliper is mounted on the structure, and the pads are replaced. A hydraulic circuit is then provided to apply pressure to the back of the pads and braking. This hydraulic circuit consists of hydraulic oil, a rotary pump, a direction control valve, a pressure relief valve, an accumulator, a pressure gauge, and a manifold. The picture of the structure and the hydraulic circuit can be seen in Figure 11. Braking is applied at three different pressures. Starting with a pressure of 6 bar, a full modal test is performed with the help of the mentioned modal equipment. The frequency response functions obtained from the test can be seen in Figure 12. The natural frequencies are obtained by analyzing the test results in Icats software. These results can be seen in Table 4. The same process is repeated for the braking pressure equal to 9 and 12 bars. The natural frequencies of these two pressures can also be seen in Table 3. It should be noted that the results in this table are related to a fixed sector (sector 2) of the disk in contact with the pads.

**Fig.11.** Experimental modal test of brake disc assembly in contact with pads

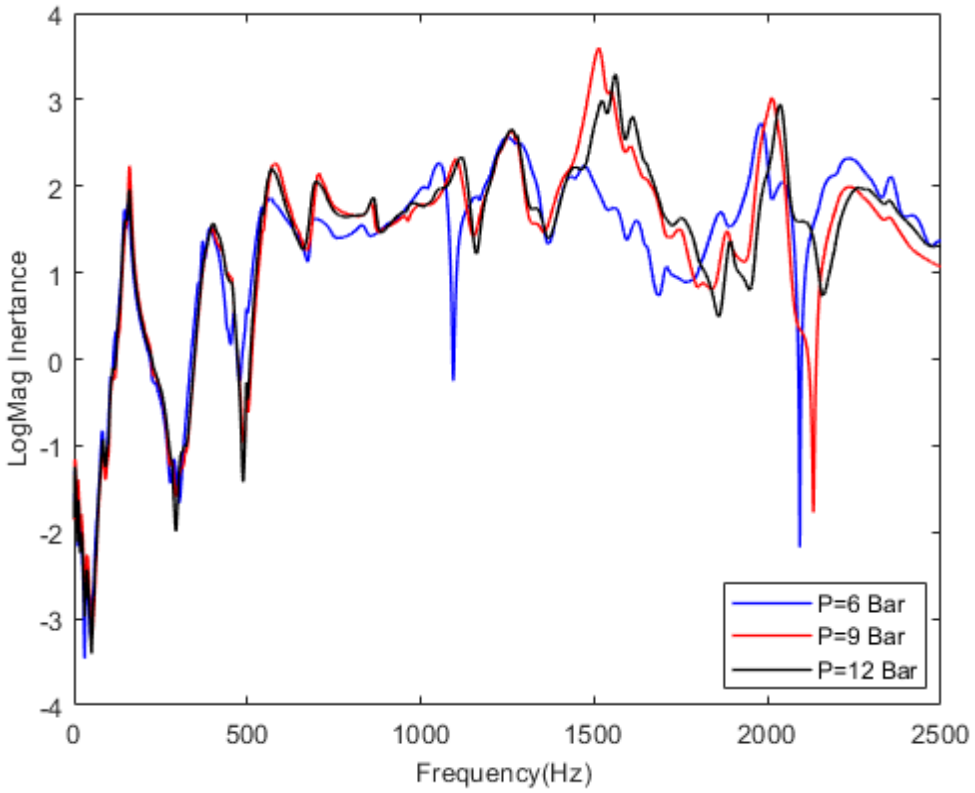


**Fig. 12.** FRF of brake disc assembly in contact with pads at three braking pressures

The outer surface of the brake disc, which is in contact with the pads, is divided into twelve parts, and once rotated, one segment of the disc is in contact with the pads at one time. The schematic of this division on disk is shown in Figure 13. The modal test is performed for all disk sectors shown in this figure. In this way, every time the disc is rotated about 30 degrees around the axis perpendicular to the disc, a new section is in contact with the pads. After applying pressure with the help of the hydraulic circuit, the modal test is performed. For each section, the test was repeated

several times, and the hydraulic circuit turned off and on each time. This is repeated for all three braking pressures, and the results are presented in the next section.

**Table 4.** Natural frequencies of experimental test of disc and pads at different braking pressures (Hz)

| Mode No | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|
| P =6 Bar | 385.11 | 839.97 | 1079.6 | 1277.0 | 1470.8 | 1859.4 | 1965.9 |
| P =9 Bar | 388.34 | 857.55 | 1105.0 | 1307.3 | 1500.8 | 1874.0 | 2006.4 |
| P =12 Bar | 391.78 | 865.21 | 1125.4 | 1326.8 | 1540.5 | 1881.1 | 2037.1 |

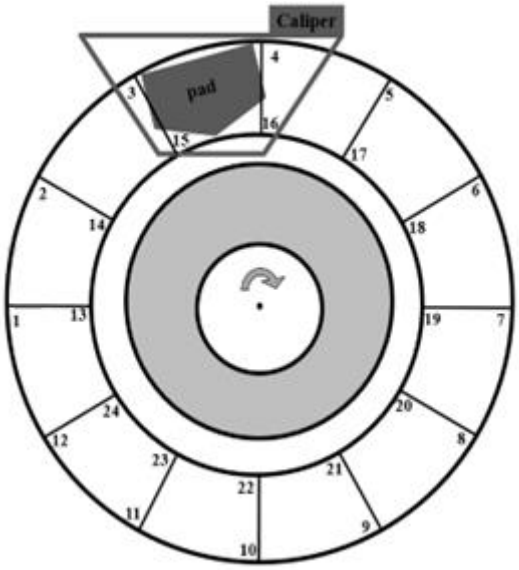


**Fig. 13.** Schematic of disc surface segmentation in contact with the pad

This experimental study has been done in static conditions of the brake system. Because the modal experimental investigation of this system in dynamic conditions is difficult, researchers in this field conduct experimental studies on the braking system in static conditions [16-18].

## 3. Results

The modal test was performed in three stages, each repeated twelve times. In this way, in each pressure, a large number of frequency data is created, and with the help of existing methods like peak picking, a frequency vector consisting of the first seven natural frequencies is obtained from each frequency data. A frequency matrix will be formed by repeating this process for all the test data. A distribution is obtained for each natural frequency, which can be drawn and analyzed using statistical analysis tools of MATLAB software. Figure 14 shows the statistical distribution obtained from the experimental modal test for the first seven natural frequencies of the disc at three different braking pressures.

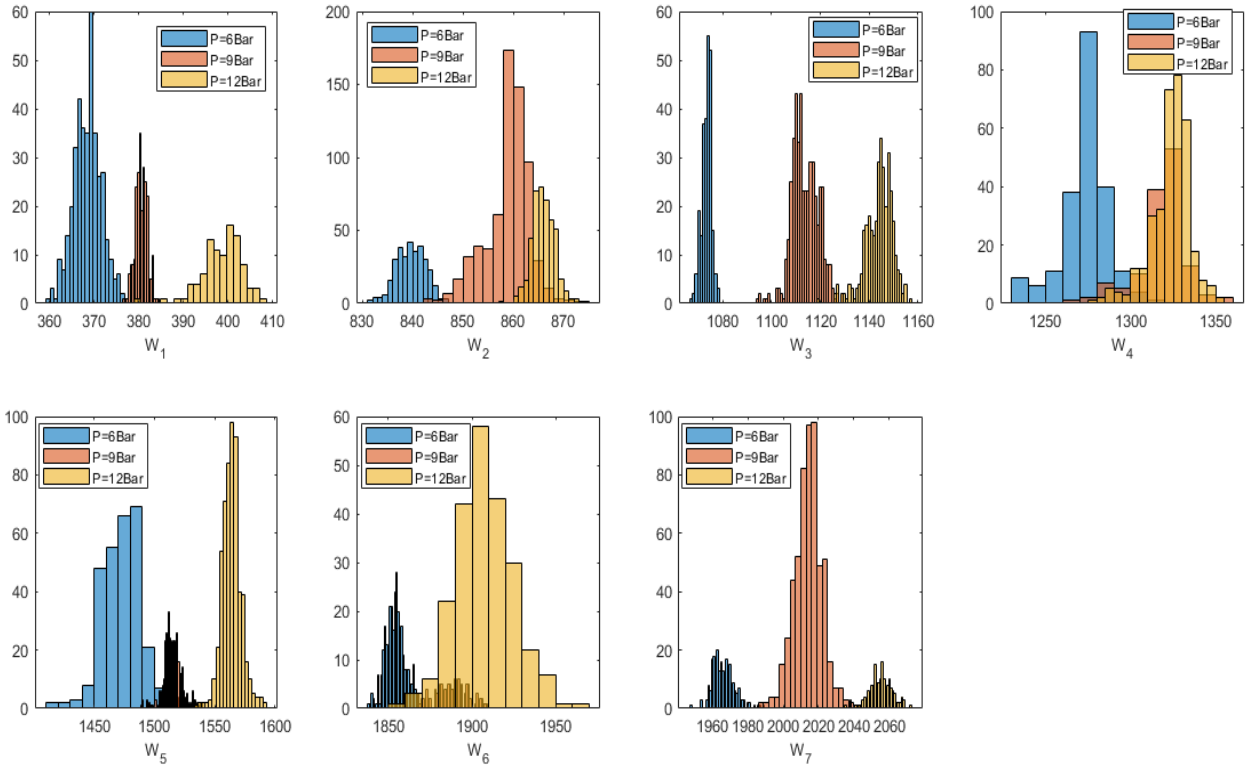


**Fig. 14:** Statistical distribution of the first seven natural frequencies of the disc at different braking pressures obtained from the experimental test

As can be seen from the graphs in Figure 14, the natural frequencies are altered by changing the contact area of the disc with the pads. By statistical consideration of each frequency range, a distribution function can be fitted, and the constants related to the fitted function can be calculated. These results are presented in Table 5.

**Table 5**: Statistical investigation results of natural frequencies obtained from experimental testing

| Freq. No | P =6 Bar | | | P =9 Bar | | | P =12 Bar | | |
|---|---|---|---|---|---|---|---|---|---|
| | Dist. name | Mean | Variance | Dist. name | Mean | Variance | Dist. name | Mean | Variance |
| 1 | Burr | 368.47 | 9.85 | Normal | 380.57 | 1.87 | Logistic | 384.97 | 7.06 |
| 2 | Normal | 839.55 | 8.24 | t-Location -Scale | 859.41 | 22.56 | Normal | 865.94 | 5.09 |
| 3 | Burr | 1073.32 | 4.19 | Normal | 1113.76 | 29.98 | Burr | 1144.08 | 31.6 |
| 4 | t-Location -Scale | 1274.75 | 450.01 | t-Location -Scale | 1328.0 | 78.24 | t-Location -Scale | 1324.25 | 168.36 |
| 5 | Normal | 1472.6 | 258.82 | t-Location -Scale | 1514.89 | 62.04 | Normal | 1563.55 | 62.76 |
| 6 | Burr | 1854.08 | 35.18 | Weibull | 1886.34 | 158.49 | Burr | 1907.38 | 308.7 |
| 7 | Epsilon Skew Normal | 1966.37 | 39.03 | Burr | 2013.77 | 61.45 | t-Location -Scale | 2055.21 | 40.62 |

## 4. Conclusion

The aim of this paper was to experimentally investigate the uncertainties in dynamic responses of the brake system while braking, like natural frequencies. First, the experimental modal test of the disk was performed with free boundary condition as well as the brake system without pads. By using these two test results, the material properties of the disk and its contact area with the pulley were investigated and identified. With this procedure, the uncertainties in the parameters of these two parts have been minimized, and the existing uncertainty in dynamic responses of the structure is only due to the changing of the contact section and braking pressure. Then, statistical distributions for natural frequencies from experimental studies were obtained. Investigations showed that changing the contact area between the disc and pads is the most important source of uncertainty in the dynamic responses of the brake system. It will definitely have a significant effect on the issue of disc instability and brake squeal. Therefore, the authors are trying to complete the modeling of this system by creating a suitable model for the contact between the disc and the pads. In the future, with the help of the existing model, the effect of these uncertainties on the instability will be studied. Moreover, the uncertainties in other parts of the brake system will also be considered. A better understanding of the brake squeal phenomenon can help reduce this annoying noise in the future.

## References


1. Papinniemi, Antti, Joseph CS Lai, Jiye Zhao, and Lyndon Loader. "Brake squeal: a literature review." Applied acoustics 63, no. 4 (2002): 391-400.
2. Kinkaid, N. M., Olivier M. O'Reilly, and Panayiotis Papadopoulos. "Automotive disc brake squeal." Journal of sound and vibration 267, no. 1 (2003): 105-166.
3. Butlin, T., and J. Woodhouse. "Friction-induced vibration: Quantifying sensitivity and uncertainty." Journal of Sound and Vibration 329, no. 5 (2010): 509-526.
4. Butlin, T., and J. Woodhouse. "A systematic experimental study of squeal initiation." Journal of Sound and Vibration 330, no. 21 (2011): 5077-5095.
5. Gauger, U., M. Hanss, and L. Gaul. "On the inclusion of uncertain parameters in brake squeal analysis." In IMAC-XXIV: Conference & Exposition on Structural Dynamics, St. Louis, MO, January. 2006.
6. Culla, Antonio, and Francesco Massi. "Uncertainty model for contact instability prediction." The Journal of the Acoustical Society of America 126, no. 3 (2009): 1111-1119.
7. Oberst, Sebastian, and JC Ss Lai. "Statistical analysis of brake squeal noise." Journal of Sound and Vibration 330, no. 12 (2011): 2978-2994.
8. Tison, Thierry, Arnaud Heussaff, Franck Massa, Isabelle Turpin, and R. F. Nunes. "Improvement in the predictivity of squeal simulations: Uncertainty and robustness." Journal of Sound and Vibration 333, no. 15 (2014): 3394-3412.
9. Iroz, Igor, Sergio Carvajal, Michael Hanss, and Peter Eberhard. "Transient simulation and uncertainty analysis of brake systems using a fuzzy-parameterized multibody system approach." Mathematics and Mechanics of Solids 24, no. 1 (2019): 40-51.
10. Zhang, Zhi, Sebastian Oberst, and Joseph Lai. "Uncertainty analysis for the prediction of disc brake squeal propensity." In Inter-Noise and Noise-Con Congress and Conference Proceedings, vol. 255, no. 7, pp. 649-658. Institute of Noise Control Engineering, 2017.
11. Zhang, Zhi, Sebastian Oberst, and Joseph CS Lai. "Instability analysis of friction oscillators with uncertainty in the friction law distribution." Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science 230, no. 6 (2016): 948-958.

12. Zhang, Zhi, Sebastian Oberst, and Joseph CS Lai. "On the potential of uncertainty analysis for prediction of brake squeal propensity." Journal of Sound and Vibration 377 (2016): 123-132.
13. Renault, Alexandre, Franck Massa, Bertrand Lallemand, and Thierry Tison. "Experimental investigations for uncertainty quantification in brake squeal analysis." Journal of Sound and Vibration 367 (2016): 37-55.
14. Nobari, Amir, Huajiang Ouyang, and Paul Bannister. "Uncertainty quantification of squeal instability via surrogate modelling." Mechanical Systems and Signal Processing 60 (2015): 887-908.
15. Jalali, Hassan, Hamed Haddad Khodaparast, and Michael I. Friswell. "The effect of preload and surface roughness quality on linear joint model parameters." Journal of Sound and Vibration 447 (2019): 186-204.
16. Abdullah, Yaqoub, and Amr Baz. "Brake squeal: A control strategy using shunted piezoelectric pads." Journal of Vibration and Acoustics 143, no. 3 (2021): 031005.
17. Oberst, Sebastian, and JC Ss Lai. "Statistical analysis of brake squeal noise." Journal of Sound and Vibration 330, no. 12 (2011): 2978-2994.
18. Renault, Alexandre, Franck Massa, Bertrand Lallemand, and Thierry Tison. "Experimental investigations for uncertainty quantification in brake squeal analysis." Journal of Sound and Vibration 367 (2016): 37-55.