Design Optimization of Disc Brake for Reducing Squeal Noise by Frequency Sensitivity Analysis

Bawoo Cha, Taeksu Jung, Yunhwa Hong, Sungsu Park, Cheongmin Kim, Younghoon Hong, and Chongdu Cho

Abstract—As widely noted by researched in last decades, an automotive brake system works by transforming kinetic energy of vehicle into thermal energy, and dissipating the energy through heat conduction. The typical concerns in this kind of system are the noise problems in high frequency which can be figured the vibration characteristics out though modal analysis of the system. As noted in recent research, the mentioned problem is mainly related with the specific modes of components where potentially can be superposed. The separation of these modes should be handled as a crucial design consideration at the initial stage of development. The rotor is a significant contributor of noise and vibration, hence the modal separation is to be considered in designing detail shapes of the component, since the shape of the object generally affects the mode frequencies. In this study, frequency sensitivity analysis through finite element method by design of experimental process has conducted to find the dominant shape factors among the multiple designable figures of the rotor body. Several shape factors are selected as design variables for the implementation of the modal separation process. The contribution of the shape factor to the mode frequency transition has calculated to achieve the optimal design of given component.

Index Terms—Disc brake, squeal noise, taquchi method, modal separation.

I. INTRODUCTION

A. Brake Noise

Research of automotive brakes has been practiced by automotive manufacturers for decades due to the importance of the brake system as a significant contributor to the safe operation of vehicles as well as the safety of the vehicle's occupants. Recently, vehicle issues such as quietness and passenger comfort have become a major concern. As a result, researchers for the automotive industry face the challenging problem of managing the noise, vibration and hardness (NVH) issues of brake systems [1], [2]. In previous reviews, Millner *et al.* [3] listed a wide classification of brake noise and vibration phenomena such as judder and squeal. For the case of squeal noise, it is well known that the noise is due to friction induced vibration or self-excited vibration of the rotating disc during the late stage of the braking sequence [4], [5].

It is assumed from former studies that the squeal noise is induced by dynamic instability, circumstances during the braking sequence, and the constitution of the braking system itself. However, identifying all the mentioned factors required for designing a squeal preventive brake starting from an initial drawing is not a simple task. Therefore, several research efforts have been published regarding the discovery of applicable methods in an attempt to understand, predict, and prevent squeal occurrence through experimental, analytical, and numerical methods [6]. Moreover, the use of complex eigenvalue analysis for brake squeal has been widely accepted in industry [7], [8]. Mode frequency based research has been used to predict brake squeal occurrence since the early 2000's. It was determined from numerical simulation and experimental results for disc brake systems that unstable modes were generated at the frequency of the particular mode superposition [9].

The mode shape of the brake rotor can be classified as in-plane mode or out-of-plane mode by the prior direction of the motion as shown in Figs. 1 and 2. In-plane modes are further categorized as tangential or radial modes. The tangential modes have two prior directions of motion, i.e., compression direction or shear direction. In addition, the tangential modes are a significant contributor to the dynamic instability of the brake rotor. The tangential and radial modes, which experience different prior motion directions, were previously studied using analytical and experimental methods by other researchers. Subsequently, it was determined that the compression directional tangential in-plane mode played a dominant role in the generation of dynamic instability [10]. Out-of-plane modes were further classified as diametrical bending or circumferential bending modes. Chen et al. [11] assumed the coupling of two adjacent modes (compression directional tangential in-plane mode and diametrical bending mode) as the dominant cause of the squeal noise.
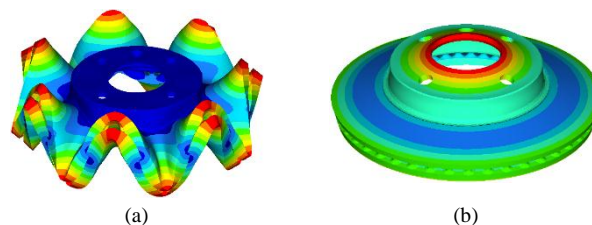

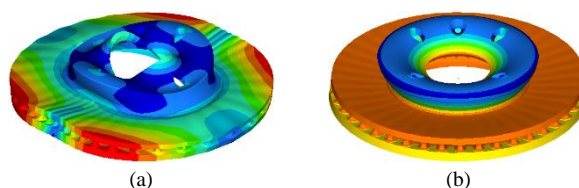Fig. 1. Shape of out-of-plane mode (a) Diametrical bending, (b) Circumferential bending.

Fig. 2. Shape of tangential in-plane mode (a) Motion in compression direction, (b) Motion in shear direction.

B. Design of Experiment — Taguchi Method

The Taguchi method is an approach for planning and analyzing experiments that allows for optimization of functional properties at the initial design stage and is widely adopted program used in various fields of research. A well-planned set of experiments, in which all parameters of interest varied over a specified range, will obtain systematic data for presenting the effect of the control factors to the functional properties [12].

A product generally has functional properties which have desirable goals and are identified as the ideal values. These ideal values can be divided into several categories in accordance with the Taguchi method including the smaller-is-better case, the larger-is-better case or, finally, the nominal-is-better-case. Researchers have made efforts to determine the ideal cases; however, there are obvious obstacles such as usage, environment, product deterioration and production imperfections. These obstacles are termed as noise factors that cause functional properties to separate from the ideal cases. The Taguchi method adopts a loss function to quantify the separation as

$$E\{L(y)\} = E\left\{\frac{k}{y^2}\right\} \approx \frac{k}{\mu^2}\left\{\frac{3\sigma^2}{\mu^2} + 1\right\} \qquad (1)$$

where y represents the functional properties, E is the expected value, L is the expected loss, σ^2 is the variance of y, and μ is the average of y.

In addition, the Taguchi method is based on orthogonal array experiments that provide a significantly reduced variance based on the optimum settings of the control parameters. The orthogonal arrays provide a set of well-balanced experiments and signal-to-noise ratios (S/N) that assists in data analysis and prediction of optimum results [13].

C. Subject of the Research

The effective design method for squeal noise reduction by separation of specific coupled modes will be introduced in following chapters. This type of design method, known as modal separation, presents discrepancies regarding the output of the application for single part or an assembled system. For this aspect, Lee *et al.* [14] calculated the level of contribution for the modes that were related to the squeal noise for each component that constitute the brake system. The vector contribution method and kinetic energy contribution method were adopted for this calculation. In addition, it was revealed that the rotor was the dominant contributor, representing 51.3 % of the squeal noise among 13 different parts including the rotor, knuckle and hub. For this connection, a rotor that has a significant relationship with the occurrence of squeal noises should be controlled to avoid mode coupling from the initial design stage.

In this research, the design variables, which are related to squeal noise, have been selected and arrayed by the Taguchi method using a commercially available rotor. Additionally, the rotor will be analyzed by a finite element method to understand the frequency sensitivity. Through this analysis, the dominant factors among the multiple designable shapes on the rotor body are prioritized for practical application on the rotor design, and analyzed for understanding their effect on the mode frequency transition.

II. DESIGN OF EXPERIMENT

A. Overview

This paper increases the understanding of the relationship between mode frequency transition and shape modification, which is practiced in accordance with the procedure described in Fig. 3. The control factor in this study is focused on the rotor, as a single part, given that it was a dominant contributor to the squeal noise. The experiment and analysis plan based on the Taguchi method was designed at the initial stage of this study. It was aimed to translocate the frequency of the squeal-related modes such as the compression directional tangential in-plane mode and diametrical bending mode. The ideal value was set as the larger-the-better case, since a large distance between squeal-related modes created a dynamically stable condition for the rotor.
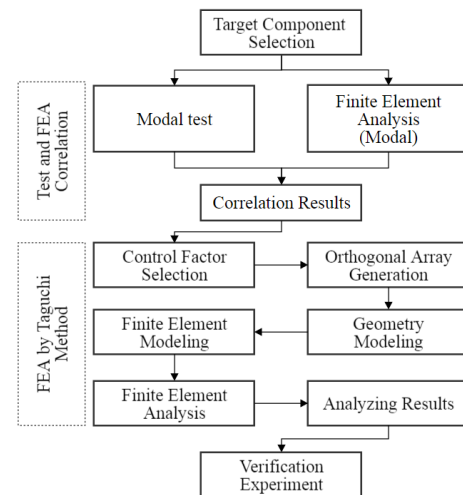

Fig. 3. Flow chart of the research.

The reliability of the finite element analysis model was achieved by the correlation between the modal test result and the modal analysis result based on the original shaped rotor. According to the orthogonal array, the shape-modified geometry and finite element model were made based on the reliable analysis model where the modal analysis was practiced for these models. Finally, the results were evaluated by the S/N ratio where it was discovered that the particular shape modification for the rotor could ensure the modal transition was an outcome.

B. Constitution of Finite Element Model

The geometry of the selected target component is shown in Fig. 4. The component was a 13-inch commercial rotor for a lightweight vehicle, which has 236 mm outer diameter and a height of 41 mm.
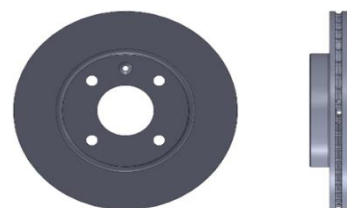

Fig. 4. Model geometry of disc brake rotor.

The material properties of the rotor used for the analysis was adopted from a general library. The brake rotors were made with gray case iron, which has material properties mentioned in Table I [15].

TABLE I: MATERIAL PROPERTIES OF GRAY CAST IRON

Material	Gray Cast Iron
Density	7070 kg/m³
Poisson's Ratio	0.29
Modulus of Elasticity	121 GPa

The finite elements of the selected rotor geometry were constituted as shown in Fig. 5. The model was organized using 8-node hexahedron elements and 6-node triangular prism elements. The total number of elements and nodes used were 153,783 and 183,757, respectively. To ensure the performance of a reliable simulation the nodes and elements were constrained for the other modified models in the orthogonal array.
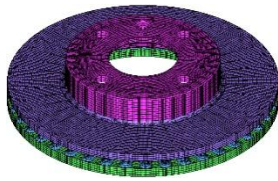

Fig. 5. Finite element model of 13-inch disc brake.

The boundary conditions for the analysis were applied as free-free conditions to allow the structure to vibrate freely without any interference from other objects [16]. For the frequency range of interest, the region from 1 Hz to 21,000 Hz was selected for simulating the desired compression directional tangential in-plane mode, which generally occurred three times within the chosen frequency range.

C. Correlation with Experimental Result

For the purpose of validating the test data with the analysis results, the modal test result was measured based on the original brake rotor using the equipment shown in Fig. 6.
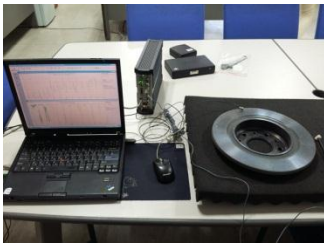

Fig. 6. Experiment equipment for frequency response function.

Frequency response functions were measured at the free-free boundary conditions by exciting the component with an impact hammer in two different directions, as shown in Fig. 7. The modal frequencies related to the out-of-plane mode resulted from the perpendicular directional excitation as shown in Fig. 7(a). Alternately, the frequencies for the in-plane mode resulted from the transverse directional excitation as shown in Fig. 7(b).

(a)   (b)

Fig. 7. Excitation direction for modal test (a) out-of- plane direction, (b) in-plane direction.

Each excitation for the two different directions was carried out five times. The outcomes were calculated as the average frequency of the peak occurrence. The modal test results from the original rotor was plotted as shown in Fig. 8.

The correlation of the frequency response function between the actual part and the analysis model was accomplished by modifying the material properties of the model to be within the nominal range of the gray cast iron.

Table Ⅱ shown below, provides the correlation results after finite element model modification. The average of the difference between the analysis result and the measured data was calculated as 0.47 %, which is considered to be in good agreement.
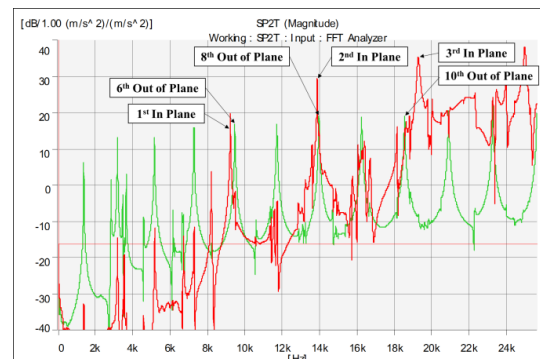

Fig. 8. Frequency response function as measured.

D. Design Parameters

A brake system represents an assembly of several individual parts such as pads, caliper and a hub. Therefore, the assembled condition of the rotor and connected parts act as a factor restricting the designed shapes. In this study, the goal was to analyze the shapes that are not related to the connected parts of the brake system. Therefore, the shape of the vane on the rotor was the primary focus of our research. The design parameters reviewed in this paper were selected

TABLE II: CORRELATION BETWEEN EXPERIMENT AND ANALYSIS

Index	Out-of-plane modes										In-plane modes		
	2nd	3rd	4th	5th	6th	7th	8th	9th	10th	11th	1st	2nd	3rd
Analysis [Hz]	1347	3154	5143	7281	9497	11750	14020	16310	18610	20930	9166	13750	19090
Experiment [Hz]	1355	3163	5139	7253	9448	11678	13959	16231	18538	20881	9204	13860	19262
Comparison [%]	0.59	0.29	0.08	0.38	0.52	0.61	0.44	0.48	0.39	0.23	0.41	0.8	0.9

as shown in Fig. 9. The control factors were located on the vane. Additionally, the control factors did not have any effect on the adjacent parts.
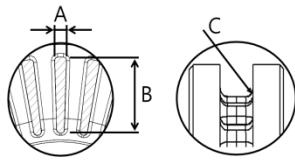

Fig. 9. Design parameters of the vane on the brake rotor.

The design of the experiment was planned based on the three factors shown on Fig. 9. Additionally, each of the factors were controlled in three dimensions as shown in Table III.

TABLE III: DESIGN PARAMETERS

	A (Width)	B (Length)	C (Fillet)
1	4.2 mm	41.25 mm	2 mm
2	4.7 mm	41.75 mm	2.5 mm
3	5.2 mm	42.25 mm	3 mm

Based on the three dimensions and the three levels, the orthogonal array as organized as shown in Table IV. Nine cases in total were developed for understanding relationship of the frequency transition through shape modification. These cases were modeled for following steps.

TABLE IV: ORTHOGONAL ARRAY WITH DESIGN PARAMETER ON VANE

Model	A (Width)	B (Length)	e	C (Fillet)
Case 1	1	1	1	1
Case 2	1	2	2	2
Case 3	1	3	3	3
Case 4	2	1	2	3
Case 5	2	2	3	1
Case 6	2	3	1	2
Case 7	3	1	3	2
Case 8	3	2	2	3
Case 9	3	3	1	1

III. FINITE ELEMENT ANALYSIS

A. Finite Element Analysis Result

Table V shows the results of the modal analysis based on the vane modified cases in accordance with the orthogonal array. The results include eleven out-of-plane modes and three in-plane modes within the region of interest from 1 Hz to 21,000 Hz. For understanding the trend of the modal separation, the position of the in-plane mode frequency was analyzed for all cases. Cases 1, 2 and 9 had a 1st in-plane mode frequency between the 6th and 7th out-of-plane mode frequency. Otherwise, the other cases 1st in-plane mode was located between the 5th and 6th out-of-plane mode frequency. Also, for the 2nd in-plane mode frequency, Cases 4, 7 and 8 experienced at the next 7th out-of-plane mode frequency. However, the 2nd in-plane mode for the other cases were located at the next 8th out-of-plane mode frequency.

This means that the transition tendency of the in-plane mode and out-of-plane mode frequencies indicated separated behavior for each case. For this connection, the S/N ratios for the in-plane mode and out-of-plane mode frequencies were calculated separately.

B. Calculation for Contributions

Table VI shows the average of the levels and the amount of contributions for the out-of-plane mode frequencies, where u is average value for overall levels. Based on the mentioned table, the S/N ratio was calculated and plotted as shown in Fig. 10.

The calculated S/N ratios show that the contribution from factors A and C with a level 3 value provided the dominant portion of the out-of-plane mode frequency transition.

TABLE VI: THE AVERAGE LEVELS AND AMOUNT OF CONTRIBUTION FOR OUT-OF-PLANE MODES

Level	Average		Amount of contribution	
			u	10356.6
A (Width)	A1	10218.9	a1	-137.5
	A2	10342.2	a2	-14.4
	A3	10508.6	a3	152
B (Length)	B1	10443	b1	86.5
	B2	10329.6	b2	-27
	B3	10297	b3	-59.5
C (Fillet)	C1	10248.6	c1	-108
	C2	10310.8	c2	-45.9
	C3	10510.8	c3	153.9

TABLE V: MODAL ANALYSIS RESULTS

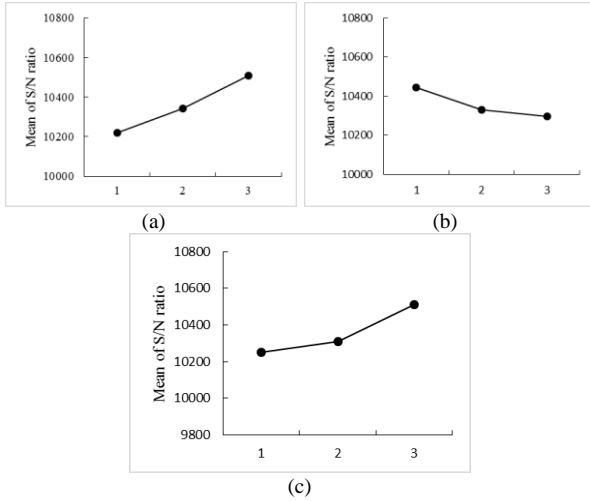| Model	Out-of-plane modes										In-plane modes		
	2nd	3rd	4th	5th	6th	7th	8th	9th	10th	11th	1st	2nd	3rd
Case 1	1337	3092	4989	6995	9058	11150	13280	15440	17630	19850	9295	13880	19350
Case 2	1300	3015	4874	6849	8883	10950	13040	15160	17310	19490	8988	13450	18710
Case 3	1309	3047	4943	6967	9055	11180	13320	15490	17680	19890	8971	13440	18670
Case 4	1309	3066	5005	7088	9247	11440	13660	15890	18130	20400	8957	13410	18650
Case 5	1302	3027	4906	6905	8967	11060	13180	15320	17480	19670	8952	13410	18640
Case 6	1309	3050	4951	6983	9080	11220	13370	15550	17740	19960	8960	13430	18660
Case 7	1308	3064	5004	7090	9250	11450	13660	15890	18130	20390	8926	13370	18600
Case 8	1315	3090	5060	7190	9405	11660	13930	16220	18510	20820	8936	13400	18600
Case 9	1337	3092	4989	6995	9058	11150	13280	15440	17630	19850	9295	13880	19350

Fig. 10. S/N ratio of out-of-plane modes. (a) factor A, (b) factor B, (c) factor C.
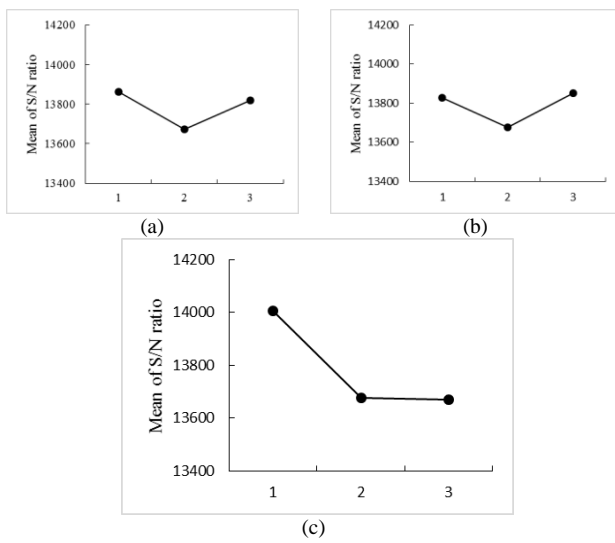

Fig. 11. S/N ratio of in-plane modes. (a) factor A, (b) factor B, (c) factor C.

The average levels and the amount of contributions for in-plane mode frequencies are described in Table VII. The S/N ratio was plotted in Fig. 11 by calculating the results from the data mentioned in Table VIII. It is observed from the S/N ratio that the contribution for in-plane mode frequency transition was dominantly made by factor C at level 1. The amount of contribution indicated considerable difference with the other outcomes of factors A and B.

Therefore, it is understood that the fillet radius (C) of the vane shape was the dominant frequency sensitive control factor, and the amount of contribution exhibited a significant difference when compared with the other factors. Additionally, the tendency was where the design factors for the rotor vane were principally related with in-plane mode frequency transition.

TABLE VIII: THE AVERAGE LEVELS AND AMOUNT OF CONTRIBUTION FOR IN-PLANE MODES

Level		Average	Amount of contribution	
			u	13784.4
A (Width)	A1	13861.7	a1	77.1
	A2	13674.3	a2	-110.1
	A3	13817.3	a3	33
B (Length)	B1	13826.3	b1	42
	B2	13676.3	b2	-108.2
	B3	13850.3	b3	66.2
C (Fillet)	C1	14005.7	c1	221.3
	C2	13677.3	c2	-107.3
	C3	13670.7	c3	-114

IV. VERIFICATION BY EXPERIMENT

A. Experiment for Modified Part

The dominant design factor of the vane's fillet radius was discussed in the previous sections. This result was verified for this research by experimental methods. From the factors revealed, the fillet radius was modified from the original part for verification in accordance with the analyzed model dimensions. Table VIII shows the dimensional comparison of the control factors between the original part and modified part. Additionally, as shown in the table, only the fillet radius was modified from 2.5 mm to 3 mm.

TABLE VIII: DIMENSION COMPARISON OF CONTROL FACTORS BETWEEN ORIGINAL PART AND MODIFIED PART

Model	A (Width)	B (Length)	C (Fillet)
Original	5.2 mm	42.25 mm	2.5 mm
Modified	5.2 mm	42.25 mm	3 mm

The modified fillet radius part was tested in accordance with the experimental conditions previously explained in Section 2.3 above. The excitations for the two different directions were carried out five times. The outcome was calculated as the average frequency of the peak occurrence.

B. Result of Verification

The correlation between the modified part and the original part was accomplished in accordance with the frequency response functions as measured for each part. The outcomes from each test are shown in Table Ⅸ. The frequency response function from the modified rotor was plotted in Fig. 12. Plots on the figure indicate the frequency response function from in-plane mode and out-of-plane mode. Additionally, the peaked points on the plot were considered as critical frequencies.
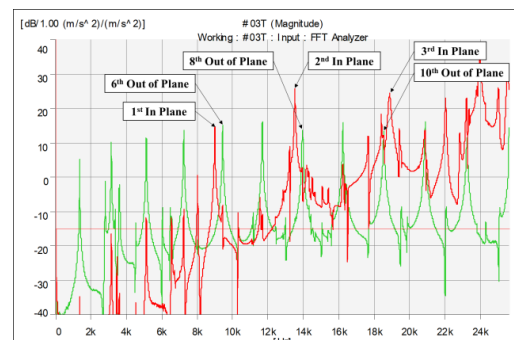

Fig. 12. Frequency response function measured from the modified part.

To predict squeal noise occurrence, the squeal noises were compared with the frequency difference between the compression directional tangential in-plane mode and the diametrical bending mode. The compared data for modal

separation between the original part and the modified part are shown in Table X.

TABLE IX: COMPARISON BETWEEN ORIGINAL AND MODIFIED PARTS

Model	Out-of-plane										In-plane		
	2nd	3rd	4th	5th	6th	7th	8th	9th	10th	11th	1st	2nd	3rd
Original [Hz]	1355	3163	5139	7253	9448	11678	13959	16231	18538	20881	9204	13860	19262
Modified [Hz]	1327	3108	5075	7181	9375	11615	13875	16165	18458	20798	8919	13441	18721

TABLE X: MODAL SEPARATION OF ORIGINAL AND MODIFIED PART

Model	Modal Separation					
	1st IP		2nd IP		3rd IP	
	Front OP	Back OP	Front OP	Back OP	Front OP	Back OP
Original [Hz]	1951	244	2183	99	724	1620
Modified [Hz]	1738	455	1826	433	263	2077
Compare [%]	-10.9	86.4	-16.3	337.3	-63.6	28.2

The results shown in Table X indicated that the distance from the IP (in-plane mode) to the OP (out-of-plane mode) was dispersed at the 1st and 2nd in-plane modes. For the 3rd in-plane mode, the frequency difference decreased creating dynamic instability. However, the modified part had a more robust design for the squeal occurrence even for the 3rd in-plane mode case around 19,000 Hz since the prior frequency range for squeal occurrence was from 1,000 Hz to 16,000 Hz.

V. CONCLUSION

This study was based on prior research which revealed the relationship between brake squeal noise and modal superposition. The study then attempted to classify the design factors of the brake rotor to determine the dominant factors affecting modal separation. As a result of the analysis, the fillet radius of the vane was identified as having considerable influence over the modal separation. The calculated S/N ratio indicated the distinct tendency of the in-plane and out-of-plane mode to undergo a factor value change. This resulted in a relatively small modal transition of the out-of-plane mode, while for the in-plane mode a larger value for the factors brought about a larger mode movement.

These results indicate that the selected controllable factor was suitable to achieve the modal separation. Additionally, the modal separation was necessary to divide the compression directional tangential in-plane mode and the diametrical bending mode. This division occurred because the in-plane mode can only be transferred in accordance with an increase of the controllable factors since the out-of-plane mode did not move significantly.

This outcome, as a measure of squeal reduction, could be applied at the initial design stage by brake system manufacturers. However, it should be noted that for this application, decreased structural stiffness may occur because of the proposed vane modification. Moreover, the results will be more valuable based on an understanding of the quality control of the vane fabrication since it was revealed that the vane had a significant relationship with the occurrence of brake squeal.

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