Finite Element Analysis on the Influence of the Change of Characteristic Parameters of Brake Pad on Structural Modal

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Abstract. The modal coupling of disc brake disc and brake pad is one of the main causes of brake noise, so the structural modal of the disc and brake pad is very important for the research of brake noise. In this paper, finite element theory is used to analyze the influence of the characteristic parameters of the brake pad on the structural modal. We use modal analysis to calculate the modal frequency and modal shape of the brake pad that has the different thickness, the depth and width of the groove, and the chamfer. According to the simulation results, the thickness of the brake pad, the groove depth of the brake pad, the groove depth of the brake pad and the size of the brake pad chamfer all have influence on the modal characteristics of the brake pad structure, especially, the thickness of the brake pad has the greatest influence. Therefore, changing the thickness of the brake pad and the depth of the brake pad groove can be a potential way to reduce the brake noise of disc brakes.

1. Introduction

Disc brakes have replaced drum brakes as the mainstream braking device due to their excellent performance, but there are still some defects that have not been overcome, among which brake vibration and noise are one of the technical problems[1]. The frequency of Disc brakes' braking noise is mainly distributed in 10~16000 Hz[2], which can be divided into low-frequency vibration noise (10~1000Hz), high-frequency vibration noise (>1000 Hz) which can be further divided into Low-far Squeal (1000~3000 Hz) and High-far Squeal( >3000 Hz). Research on disc brakes' braking noise mainly focuses on high-frequency vibration noise (>1000 Hz)[3,4]. In recent years, the study of the mechanism of brake noise and how to reduce the brake noise has been the focus of many scholars and auto enterprises. Modal coupling theory is currently considered to be the most likely mechanism for the generation of braking squeal. It holds that when two or more parts with contact relationship have similar modal shapes, the phenomenon of modal coupling leading to system instability and resulting in braking noise will occur[5,6]. Guan et al.'s research on brake noise proved that the modal coupling between brake pad and brake disc was the main cause of brake noise[7-10]. Therefore, studying the modal characteristics of brake disc and brake pad is an important way for research and control of brake noise.

Many researches show that changing the modal characteristics of brake disc and brake pad is an effective way to prevent and reduce brake noise. The structural modal of the brake disc can be changed...
by changing damping, constraint state of the brake, material characteristic parameters of the brake disc, etc[11-15]. The current researches mainly focuses on controlling the modal characteristics of the brake pad by changing the material characteristic parameters, but less researches on the characteristic parameters of the brake pad structure[16,17]. This paper uses finite element method to study the influence of structural parameter changes on the structural modal of brake pad, which provides a theoretical basis and experimental reference for the design of structural modal characteristics of brake pad and the reduction of braking noise.

2. Methodology and finite element model

2.1 Modal analysis theory

The modal analysis theory transforms the physical coordinates with linear time-invariant properties into modal coordinates in the system of differential equations of vibration, and decouples the equations to solve a set of independent equations, which are composed of modal coordinates and modalities. The matrix obtained by modal coordinate transformation is the system modal shape matrix, whose column vectors are the corresponding mode modes of the system, and this process is defined as modal analysis [18]. The modal analysis of the structure can obtain the modal parameters of the system, then the relevant vibration problems of the system can be predicted and avoided in advance, which can also avoid unnecessary damage and failure caused by vibration. At the same time, it can provide strong theoretical data support for the optimization of structural force characteristics. The system modal analysis processes are as follows.

System vibration differential equation:

\[ M \dddot{x} + C \dot{x} + K x = F(t) \]  \hspace{1cm} (1)

\( M \), \( C \), and \( K \) represent mass, damping and stiffness matrix respectively; \( F(t) \) represents the load function changed with time; \( \dddot{x} \) represents the node acceleration vector; \( \dot{x} \) represents the node velocity vector; and \( x \) represents the node displacement.

In view of the fact that free vibration usually does not take into account the influence of load related factors such as damping, the free vibration equation under undamped condition can be simplified into the following form:

\[ M \dddot{x} + K x = 0 \]  \hspace{1cm} (2)

Mode superposition method is usually adopted more often. The key to using this method is to transform the general vibration equation of natural frequency and complete coupling of main modes of various parts into independent coupling equation. When mass matrix \( M \) and stiffness matrix \( K \) are regarded as constants, free vibration is simply harmonic vibration.

\[ \{x\} = \{\varphi_i\} \cos \omega_i t \]  \hspace{1cm} (3)

The basic eigenvalue algorithm is used to transform the vibration equation without considering damping.

\[ [K]\{\varphi_i\} = \omega_i^2 [M]\{\varphi_i\} \]  \hspace{1cm} (4)

The characteristic equation of the structure can be obtained as follows:

\[ [K] - \omega_i^2 [M] = 0 \]  \hspace{1cm} (5)

The natural vibration frequency \( \omega_i \) of the \( i \) order and the eigenvector \( \{\varphi_i\} \) corresponding to the \( i \) modal shape of the system can be calculated through the above equation.

2.2 Description of the modal shape of the brake pad

The study found that the brake noise of the disc brake is mainly due to the modal coupling between the bending mode shown in figure 1, 2 (or torsion mode shown in figure 3, 4) of the brake pad and the out-
of-plane mode of the disk[19]. Therefore, we mainly study the influence of the change of characteristic structural parameters on the bending modal and torsion modal of the brake pad. The brake noise frequency of the disc brake is in the range of 10 to 16000 Hz, so the main bending and torsion modal shapes of the brake pad in this range are extracted as the research object.

Figure 1. The first-order bending modal shape  
Figure 2. The second-order bending modal shape

Figure 3. The first-order torsion modal shape  
Figure 4. The second-order torsion modal shape

2.3 Finite element model
By using the finite element technique to analyze the modal characteristics of the brake pad, firstly, the three-dimensional model of the brake pad should be established, and the establishment of an accurate finite element model is the basis of the finite element analysis. The brake pad model studied in this paper is in accordance with the 1:1 ratio, drawing with 3D drawing software, as shown in figure 5. Secondly, the 3D model is imported into the finite element software for meshing to prepare for modal analysis. We need to mesh the brake pads with different feature size parameters multiple times, and the brake pad is mainly plate-shaped structure, in order to ensure the reliability of the analysis results, so Hex Dominant meshing method is selected for finite element model of all brake pads, and the cell size is set to 3mm. Figure 6 shows the finite element model after meshing.

Figure 5. Brake pad 3D model  
Figure 6. Brake pad finite element meshing results

2.4 Modal analysis settings
After the meshing is completed, the natural frequency and modal shape of the brake pad are obtained through modal analysis. Before the analysis, it is necessary to define the brake pad material properties, such as mainly density, Young's modulus and Poisson's ratio (as shown in table 1). The modal extraction method we choose is the BlockLanczos algorithm, and during the process of solving the brake pad modal, the modal extraction order is set to 50, and the frequency extraction range is set to 0-16000 Hz to meet the requirements of the braking noise frequency range. We solve the free modal and constraint modal of the brake pad. The first, second and third order bending modal and the first,
second and third order torsion modal of the brake pad are selected as the analysis objects. Further, the influence of the change of characteristic parameters on the modal parameters of the brake pad is analyzed through the change of the above six order modal. When the constraint modal solution is solved, combined with the actual working state of the brake pad, the three translational degrees of freedom of the brake pad are restrained, and translational freedom of the brake pad of the x-direction and y-direction are fixed, and the z-direction (axial direction) is retained.

Table 1. Material properties of Brake pad.

| Name                      | Description                   |
|---------------------------|-------------------------------|
| Types of materials        | resin matrix composite        |
| Material property         | anisotropic                   |
| Density (Kg m\(^{-3}\))   | 1550                          |
| Young’s modulus           | 2.2E9                         |
| Poisson’s ratio           | 0.25                          |

3. Results and discussion

3.1 Characteristic parameters of brake pad

The brake pad structure is composed of a brake back plate and a brake pad body. After excluding the detailed features of the part, the main characteristic parameters are thickness of brake pad, groove depth of brake pad, groove width of brake pad, both sides chamfer and thickness of brake pad back plate thickness. The selected four characteristic parameters are thickness of brake pad T, groove depth of brake pad D, groove width of brake pad W, right-angle side l of chamfer, as shown in figure 7.

Figure 7. The characteristic parameter of brake pad

The change of the modal characteristic of the brake pad can be perceived by comparing the frequency difference between the modal frequency of the brake pad characteristic parameter and the modal frequency of the original size. When the frequency difference is negative, it indicates that the modal frequency after the change of the characteristic parameter is reduced compared to when it is unchanged, and the frequency difference is increased when the frequency difference is positive. The larger the absolute value of the frequency difference is, the larger the frequency change after the change of the characteristic parameter is, namely, the greater the change in modal characteristics is, indicating that the change of the characteristic parameter has a greater influence on the modal characteristic of the brake pad.

3.2 Effect of the thickness of the brake pad

The original size of the brake pad thickness T is 10mm, and the dimension change gradient is set to 1mm, namely, the thickness T is 8mm, 9mm, 10mm, 11mm, and 12mm respectively to study the effect of the change of the brake pad thickness T on the modal characteristics of brake pad. Figure 8 shows the effect of the change of the thickness T of the brake pad on its free modal and constraint modal. It can be seen from figure 8 that as the thickness of the brake pad increases, the frequency of each order in the free modal and the constraint modal also increases. With the change of T, the third-order free torsion mode changes maximally, and the second-order torsion mode in the constraint mode was affected most; the change of the thickness T remarkably affects the frequencies of the free modal and
the constraint modal. It can be seen that the variation of the thickness \( T \) of the brake pad has a great influence on the structural modal characteristics of the brake pad.

Figure 8. The influence of the variation of the thickness \( T \) of the brake pad on the free and constraint modals

### 3.3 Effect of the groove depth of the brake pad

The influence of the change of the groove depth \( D \) of the brake pad on the modal characteristics of the brake pad is studied according to the five dimensions of 2 mm, 3 mm, 4 mm, 5 mm and 6 mm, wherein the original depth of the brake pad groove is 4 mm. Figure 9 shows the effect of the change of the groove depth \( D \) of the brake pad on the free modal and the constraint modal of the brake pad. It can be seen from figure 9 that the free modal frequency is reduced with the increase of the groove depth \( D \) of the brake pad; the second-order bending and torsion modals in the constraint modal are little affected by the variation of the groove depth \( D \), and the modal frequency is basically unchanged. The other order frequencies decrease with the increase of the groove depth of the brake pad; in the free modal, the third-order torsion mode is most affected by the change of \( D \), and the frequency difference between the groove depth of 2 mm and 6 mm is about 280 Hz. With the increase of \( D \), the frequency of the first-order bending mode in the constraint modal decreases by about 420 Hz. The change of the depth \( D \) of the brake pad is negatively correlated with the change of the free and constraint modal frequencies. Basically, the larger \( D \) is, the smaller modal frequency is.

Figure 9. The influence of the variation of the groove depth \( D \) of the brake pad on the free and constraint modals
3.4 Effect of the groove width of the brake pad
The original dimension of the groove width \( W \) of the brake pad is 3 mm, and the dimension change gradient is 1 mm, namely, the groove width \( W \) is 1 mm, 2 mm, 3 mm, 4 mm and 5 mm respectively, to study the influence of the change of the groove width \( W \) of the brake pad on the modal characteristics of brake pad. Figure 10 shows the effect of the variation of the groove width \( W \) of the brake pad on the free modal and the constraint modal of the brake pad. It can be seen from figure 10 that the frequency of each order in the free and constraint modal decrease gradually when the groove width \( W \) of the brake pad grows, but the magnitude of the change of frequency is relatively less affected by the variation of \( W \); the first-order bending mode under free and constraint modals is most affected by the of change of \( W \). The variation tendency of the second-order torsion modal under the two modals are opposite to those of other modes, which increases with the increase of the groove width \( W \) of the brake pad.

![Figure 10](image_url)

Figure 10. The influence of the variation of the groove width \( W \) of the brake pad on the free and constraint modals

3.5 Effect of the chamber of the brake pad
In this paper, the chamfer is changed by changing the length of two right-angle sides \( l \) and \( h \) of the chamfer of the brake pad. The chamfer right-angle side \( l \) is set by the five lengths of 3mm, 4mm, 5mm, 6mm and 7mm to calculate the free and constraint modal of the brake pad. At the same time, other right-angled edge \( h \) changes along with the ratio of \( l:h \), so that the chamfer of the brake pad is changed in the same proportion. Figure 11 shows the influence of the change of the chamfer right-angle side \( l \) of the brake pad on the free modal and the constraint modal of the brake pad. It can be seen from figure 11 that the free and constraint modal of the brake pad are increased with increase of the chamfer right angle side \( l \), but the frequency increase is small. The absolute value of the frequency difference in the free modal is no more than 8 Hz, and the absolute value of the frequency difference in the constraint modal is within 20 Hz. It can be seen that the change of chamfer has little effect on the free modal and constraint modal of each order.
4. Conclusions
In this paper, finite element modal analysis method is used to study the influence of four characteristic parameters of brake pad thickness $T$, groovedepth $D$ of brake pad, groove width $W$ of brake pad and chamfer on the modal characteristics of the brake pad. According to the research results, the change of the thickness $T$ and chamber of the brake pad is positively correlated with the change of the modal characteristics of the brake pad. The change trend of the depth $D$ and width $W$ of the groove of the brake pad is negatively correlated with the change of the modal characteristics of the brake pad. The impact of Brake pad thickness $T$ on modal characteristics of brake pad changes is most obvious, followed by the groove depth of brake pad $D$, the changes of the groove width of brake pad $W$ affect the characteristics of the brake pad modal are smaller compared with the groove depth of brake pad $D$. The change of chamfer size on the brake pad has a smaller effect on the changes of modal characteristics of the brake pad, especially for the free modal characteristics almost had no effect. The change of the thickness $T$ and the depth $D$ of the groove of the brake pad can obviously change the modal characteristics of the brake pad, which provides experimental reference and theoretical basis for future research on how to reduce the brake noise.

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References
[1] Nacivet, S., Sinou, J. J. (2017) Modal amplitude stability analysis and its application to brake squeal. Appl. Acoust., 116: 127–138.
[2] Cao, Q., Ouyang, H., Friswell, M.I. (2004) Linear eigenvalue analysis of the disc-brake squeal problem. Int. J.Nuwer. Meth. Eng., 61: 1546–1563.
[3] Guan, D.H., Su, X.D. (2004) An overview on brake vibrations and noise. Eng. Plast. Appl., 21: 150–155.
[4] Zhang, Z., Oberst, S., Lai, J.C.S. (2017) Uncertainty analysis for the prediction of disc brake squeal propensity. In: Inter-Noise and Noise-Con Congress and Conference Proceedings. Hong Kong. pp. 649–658.
[5] Fritz, G., Sinou, J.J., Duffal,J.M.(2007) Investigation of the relationship between damping and mode-coupling patterns in case of brake squeal. J. Sound. Vib., 307: 591–609.
[6] Jiang, D.Y., Guan, D.H. (1998) Study on disc brake squeal using closed-loop coupling model. J. Tinghua. Univ: Nat. Sci. Ed., 38: 88–91.
[7] Zhang, L.J., Liao, W.J., Yu, Z.P. (2008) Experimental investigation into friction-vibration coupling characteristics of vehicle disc brake. Die. Mould. Ind., 5: 480–484.

[8] Lyu, H., Walsh, S. J., Chen, G. (2017) Analysis of friction-induced vibration leading to brake squeal using a three degree-of-freedom model. Tribol. Lett., 65: 105.

[9] Monteil, M., Besset, S., Sinou, J.J. (2016) A double modal synthesis approach for brake squeal prediction. Mech. Syst. Signal. Pr., 70-71: 1073–1084.

[10] Brunetti, J., Massi, F., W, D’Ambrogio. (2016) A new instability index for unstable mode selection in squeal prediction by complex eigenvalue analysis. J. Sound. Vib., 3771: 106–122.

[11] Guan, D.H., Du, Y.C., Wang, X.F., Li, Q. (2014) Analysis of a disc brake high frequency squeal and reduction. Eng. Plast. Appl., 31: 217–222.

[12] Mário, T.J., Gerges, S.N.Y., Jordan, R. (2008) Analysis of brake squeal noise using the finite element method: A parametric study. Appl. Acoust., 69: 147–162.

[13] Hui, L., Yu, D. (2016) Optimization design of a disc brakes system with hybrid uncertainties. Adv. Eng. Softw., 98: 112–122.

[14] Lars, H., Jacobson, S. (2006) Surface modification of brake discs to reduce squeal problems. Wear., 261: 53–57.

[15] Bergman, F., Eriksson, M., Jacobson, S. (2000) Effect of reduced contact area on the occurrence of disc brake squeals for an automotive brake bad. P. I. Mech. Eng. D-J. Aut., 214: 561–568.

[16] Zhang, J.M., Yang, K. (2011) Vibration characteristics of elastic disc brake pad device. Appl. Mech. Mater., 101-102: 29–32.

[17] Kang, J. (2012) Finite element modelling for the investigation of in-plane modes and damping shims in disc brake squeal. J. Sound. Vib., 331: 2190–2202.

[18] Du, Y., Wang, Y. (2017) Squeal analysis of a modal-parameter-based rotating disc brake model. Int. T. J. Mech. SCI., 131: 1049–1060.

[19] Liu, P., Zheng, H., Cai, C. (2007) Analysis of disc brake squeal using the complex eigenvalue method. Appl. Acoust., 68: 603–615.