Structural simulation analysis of a certain vehicle type disc brake based on ANSYS Workbench

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ABSTRACT—Taking the disc brake as the research object, the three-dimensional model established by Solidworks is imported into ANSYS Workbench, and its strength and modal simulation analysis are carried out. Taking the disc brake as the research object, the 3D model established by SolidWorks is imported into ANSYS Workbench, and the strength and modal simulation analysis are carried out then. Using the maximum stress point and maximum deformation position under the emergency braking state, combined with the results of modal analysis cloud chart, the resonance range is verified by comparing with the natural frequency, which provides reference for the actual tests and structural improvement.

1. Introduction
Automobile safety systems are mainly divided into passive safety and active safety systems. Active safety systems are the most important safety guarantee. In a car, the braking performance of a car is one of the main performances of a car, which is directly related to safety and comfort. Its main evaluation indicators are braking efficiency, directional stability during braking, and resistance to decay. In addition, it also has a certain impact on the noise of the car during driving. The structure of the brake has a major influence on the above performance indicators [1].

According to different structures, automobile brakes are mainly divided into drum brakes and disc brakes. Compared with drum brakes, disc brakes have better thermal stability and anti-decay performance. They have the advantages of compact size and light weight in terms of structural size and quality, and are also easy to maintain in practical applications. The working principle of the disc brake is mainly to transfer the hydraulic pressure of the hydraulic pipeline to the friction block, and the friction between the friction block and the end surface of the brake disc produces a reaction torque to achieve the braking effect [2]. At present, most cars have applications on four wheels, and only a few cars are only used as front wheel brakes in conjunction with rear wheel drum brakes to ensure higher directional stability during braking. In some commercial vehicles, disc brakes are gradually adopted in new and high-end models [3]. But at the same time, disc brakes also have some drawbacks. For example, disc brakes are prone to howling when braking. Wang Dengfeng of Jilin University and others have analyzed and experimented on the howling phenomenon of disc brakes and found that the main cause of howling is The reason lies in the contact friction between the brake disc and the friction plate [4]; Wei Jian and others from the University of Shanghai for Science and Technology have combined experiments with the theory of complex characteristics and found that the increase in the stiffness of the brake disc will increase the probability of howling [5]; Peng Long of Hunan
University and others used interval probability mixed model to describe the uncertainty of related parameters, and carried out a robust optimization design to improve the stability of the brake system [6]. University of Shanghai for Science and Technology Tian Yudong and others combined the finite element modal analysis theory and DASP test results to optimize the brake and reduce a certain weight [8].

2. Establishment of finite element model

2.1. Geometric model

In this paper, the front wheel brake disc of a special vehicle is used as the research object. The brake disc used has a ventilating rib between the two working surfaces as ventilation holes, which can increase the strength and reduce part of the weight. The above model uses a circular plate, The material is HT250 and the yield strength is 250MPa. The following figure (Figure 1) shows the three-dimensional model of the brake disc.

![Figure 1 Three-dimensional model of brake disc](image_url)

2.2. Meshing

Due to the special structure of the disc brake, there are many grooves and chamfers, and there are multiple heat dissipation spokes between the double-layer discs, and the internal structure is more complicated. In order to ensure the quality of meshing and calculation accuracy, some simplifications are made when importing the 3D model. According to the actual size of the model, the selected grid cell size is 5mm. The statics analysis module is established on the ANSYS Workbench software platform, and then the above model is exported from Solidworks to the .x_t format and imported into ANSYS Workbench. The unit type used for meshing is solid 186, and the unit is selected (mm, t, N, s, mv, mA). The final result is shown in the figure below (Figure 2). As a result of the division, the number of nodes is 57124, and the number of units is 31436.
3. **Brake disc strength analysis**

Disc brakes use the inner and outer caliper bodies as the main application objects during the braking process. The brake fluid pressure transmitted from the brake pads and pistons, and the force from the bolts fixed between the caliper body and the axle and the brake pads; the magnitude of the brake fluid pressure is mainly related to the external force applied by the brake pedal and the vehicle speed during braking. Related to [4]. In the simulation analysis, take the car traveling at 120km/h as an example, combined with the calculation method of the contact friction force (torque) between the friction surface of the friction pad in the disc brake and the brake disc:

\[ N = P_1 S_1 = P_1 \pi d_1^2 \times 0.25 \]  
\[ T_f = 2fNR \]  

**3.1. Load and restraint determination**

Adding load, the disc brake mainly acts on the brake disc through the friction block when the load is applied, and the load is applied to it as shown in the figure below (Figure 3).

Defining constraints, apply full constraints on the center hole surface of the brake disc, apply X constraints on the inner and outer end surfaces of the disc, and apply Y and Z constraints on the friction surface of the brake disc and the friction lining. Its structure is shown in the figure below (Figure 4):
3.2. Simulation results

The simulation calculation and analysis of the structural strength of the brake disc, and the results of stress solution and modal analysis deformation cloud diagram are shown in the following figure (Figure 5-6):
From the strength calculation and cloud diagram of the brake disc structure, it can be seen that the maximum stress of the brake disc during the braking process is 52MPa, and the yield strength of the brake disc is 250MPa, so the brake disc fully meets the strength requirements. It can be seen from the deformation cloud diagram that the maximum deformation of the brake disc during braking is 0.034mm, and the closer the center is the smaller the deformation.

4. Modal analysis of brake disc

The brake disc is an elastic system that vibrates under the action of an external time-varying excitation, and the resonance is related to the natural frequency of the system. When the external excitation frequency is close to the fixed frequency (i.e., resonance), it is not only unfavorable to braking safety, but also has a certain impact on the life of the brake disc and is prone to noise. Through modal analysis, the natural frequency of the system can be analyzed, which can avoid the interval of resonance frequency and reduce the influence caused by resonance [9].

4.1. Theory of Modal Analysis

Modal analysis is a way of calculating equations by means of coordinate transformation, which mainly includes four kinds of dynamic characteristic analysis. The definition is to use the physical coordinates in the vibration differential equation of the linear constant system to be converted into modal coordinates to solve the equations and obtain a set of independent equations described by modal coordinates and modal parameters. It is a common method in today's research on the dynamic characteristics of structures. The analysis process and results are obtained by the method of finite element calculation, which is called computational modal analysis. However, the modal parameters are obtained through parameter identification of the collected system input and output signals through experiments, which is called a modal analysis experiment [10].

According to the analysis of the structural characteristics of the disc brake, it can be seen that the disc brake is a typical linear steady system. As a continuous n-degree-of-freedom small damping continuous elastic body, it can be expressed by the following vibration differential equation:

$$M \ddot{x}(t) + C \dot{x}(t) + Kx(t) = F(t)$$  \hspace{1cm} (3)

Where: M—mass matrix; K—stiffness matrix; C—damping matrix; F—sum of external excitation forces; (x(t))'; (x(t))'; x(t)—respectively represents acceleration vector, velocity vector, and displacement vector.

The main influencing factors for the modal shape and natural frequency of the structure are mass and stiffness, and the influence of external damping on the mode shape of the structure can be ignored. When there is no external force, the external load F=0, that is, the damping C and the external excitation force F described in equation (1) can be ignored, and the above equation (1) can be simplified as:
\[ M \ddot{x} + K x = 0 \]  \hspace{1cm} (4)

The general form of solving equation (2) is:
\[ x = X e^{i \omega t} \]  \hspace{1cm} (5)

The necessary and sufficient condition for a non-zero solution of the above equation (2) is that the determinant of the coefficient matrix is zero, and the characteristic equation of equation (2) can be obtained by solving:
\[ (K - w^2 M) x = 0 \]  \hspace{1cm} (6)

For an n-degree-of-freedom system, the system displacement x cannot be zero at the same time. The above formula has a solution, so the above formula (4) is valid only when the coefficient matrix is 0, namely:
\[ |K - w^2 M| = 0 \]  \hspace{1cm} (7)

The natural frequency of the system can be obtained by using formula (5):
\[ f_n = \frac{\omega_n}{2 \pi} \]  \hspace{1cm} (8)

After solving the natural frequency, substituting the above equation (3) to obtain the target solution. Therefore, for a differential equation expressed by an n-degree-of-freedom system, there are also n characteristic equations and n characteristic solutions, which correspond to n groups of modal shape vectors and n-order natural frequencies [11].

4.2 Modal analysis process and results

The modal analysis is executed in the modal analysis module of ANSYS Workbench software, and then the geometric model of the brake disc is imported, the material properties are set, the brake disc is meshed, and the modal analysis result of the brake disc is finally obtained. When analyzing the modal results, select the first ten vibration frequency value and mode description of the brake disc as shown in the following figure (Figure 6-15):

![Figure 7 First-order mode](image)

Figure 7 First-order mode
Figure 8 Second mode.

Figure 9 Third mode.

Figure 10 Fourth-order mode.
Figure 11 Fifth-order mode

Figure 12 Sixth mode

Figure 13 Seventh mode
Figure 14 Eighth mode

Figure 15 Ninth order

Figure 16 Tenth order

Table 2 Frequency description of brake disc

| Order | Frequency Hz | Mode description |
|-------|--------------|------------------|
| 1     | 634.33       | Bending around X direction and deforming along Z direction |
| 2     | 660.77       | Bending around Y direction and deforming along Z direction |
| 3     | 1138.7       | Bending around the X direction, the maximum deformation position is at the symmetrical position of the friction block |
4 1226.4 Around the center, it deforms like petals
5 1230.2 Symmetrically deformed on both sides along the Y axis
6 1806.9 Outer ring deformation
7 2542.3 Wavy deformation
8 2601 Symmetrically deformed along the friction block
9 2706.1 Curved toward the center in a petal shape
10 2708.5 Bending around the Z axis and deforming along the Y axis

From the above tenth order frequency distribution, it can be seen that the lowest order frequency and the first order frequency of the brake disc are about 630Hz. Due to different brake structures, the squeal frequency of this disc brake is about 2100Hz. The modal analysis result shows 6 The first and seventh-order modal frequencies are closer to the squeal frequency, and there is a greater possibility of screaming.

5. Conclusion
This paper uses the finite element method to perform static and modal simulation analysis on the front brake disc of a special vehicle, and analyzes and compares the results. It is concluded that the rigidity requirements are met in the static analysis, but the brake squeal is prone to phenomenon. In the follow-up analysis, the occurrence of brake squeal can be reduced from the aspect of structural optimization, which provides a basis for the simplification of the brake disc structure, the reduction of costs, and the reduction of the occurrence of brake squeal.

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