Research on Structure Design of The Dual Steel Disc in Brake Based on Rigid-Flexible Coupling Simulation Analysis

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Abstract. A research method is proposed to improve the structure of the dual steel disc in wet multi-disc brake based on virtual prototype simulation analysis. The finite element flexible body model of the dual steel disc was established by CAE software, and the accuracy of the flexible body model was verified by the method of identifying natural frequency by near-field acoustic radiation. Based on the dependable finite element flexible body model, the rigid-flexible coupling virtual prototype model of the wet multi-disc brake was established, and the dynamic simulation analysis of brake during the braking period was processed. According to the results of brake dynamics simulation, the time-frequency characteristics of friction pair contact force and the change characteristics of piston spring force during the braking process are analysed. Based on credible dynamic simulation analysis, the mises stress distribution and deformation changes of the steel disc at critical moments are acquired. Depending on the virtual prototype analysis results, it is powerful to guide the structural design and improvement of the dual disc, and some phenomena and problems which are difficult to find through actual measurement can be researched effectively.

1. Introduction
Wet multi-disc brake has been used widely in construction machinery equipment due to its excellent braking performance, stability and reliability, and strong resistance to thermal degradation. In recent years, the structural dynamic performance of wet multi-disc brake has attracted more and more attention. With regard to wet multi-disc brake, most scholars [1]-[2] focused on the thermal characteristics of key components of the brake and the influence of surface material Properties on friction and wear. However, there are few researches on the vibration and dynamic performance of the dual steel disc in brake. Furthermore, the design of the dual steel disc is mainly based on experience design.

Recently, Yang et al. [3] studied the chatter problem of wet multi-disc clutch during dynamic contact. Yang et al. [4] studied the influence of operating parameters on engagement characteristics of wet clutches through numerical model and experimental setup. Indira Roy et al. [5] deployed Multi-objective optimization through response surface methodology for improving the structural performance of ventilated brake discs. Yin et al. [6] used high-speed photography and tooth-root stress coupling to evaluate the impact damage of brake friction linings. Chen et al. [7] established a thermo-structural coupling finite element model of disc brake to explore the causes of partial wear of pad in disc brake and the general wear process of pad.

In this work, a rigid-flexible coupling virtual prototype model of the wet multi-disc brake is established, and the braking characteristic curve of the dual steel disc is calculated through kinematics...
simulation, and the dynamic characteristic change of the dual steel disc in braking is analysed. According to the mises stress distribution characteristics of the dual steel disc, the weak part of the dual steel disc structure is analysed. This method can provide reliable support for the dynamic analysis of the wet multi-disc brake system, provide effective reference and guidance for the design and selection of the dual steel disc in the wet multi-disc brake, and point out the direction for the optimization of the mating steel structure.

2. Structure and working characteristics of wet-multi disc brake

The friction braking system of wet multi-disc brake is mainly composed of the components shown in figure 1. In the wet multi-disc brake, the external spline of the friction disc is connected with the inner hub of the brake cylinder block, the internal spline of the dual steel disc is connected with the spline of the connecting shaft, the friction disc contacts with the dual steel disc to form a friction pair, the dual steel disc rotates with the internal connecting shaft, and the friction disc does not rotate with the connecting shaft. The surface material of friction disc is sintered by metal powder particles. The basal body of friction disc is steel. The metal friction layer is evenly distributed in the friction disc in the circumferential direction. The two ends of the pressure spring are piston block and brake cylinder block respectively. The brake cylinder block is fixed with the construction machinery.

According to the above analysis, when the connecting shaft starts to rotate, the hydraulic system pressurizes, so that the piston pressure block moves to the right, and the pressure spring compresses. At this time, there is no pressure between the friction disc and the dual steel disc, and the contact is separated. When the marine crane starts to brake, the hydraulic system begins to release pressure. The combined spring directly acts on the piston pressure block to move the pressure block to the left and compress the friction disc and dual steel disc. After multiple friction pairs are pressed, the friction torque increases gradually until the connecting shaft stops rotating, and the braking process is completed.
3. Modal analysis and experiment of dual steel disc of wet multi-disc brake

3.1. Modal analysis of dual steel disc of wet multi-disc brake

The finite element method for modal analysis is the basis of studying the dynamic performance of parts. For the undamped dynamic system with finite degrees of freedom, the differential equation of motion is as follows:

\[ m \ddot{x} + kx = 0 \]  

(1)

Where \( m \) is the mass matrix, \( k \) is the stiffness matrix and \( x \) is the generalized coordinate.

The generalized coordinate \( x \) can be expressed as:

\[ x = u \cos(\omega t + \phi) \]  

(2)

Where \( u \) is the modal vector and \( \omega \) is the natural frequency of the system.

The equation (2) is introduced into equation (1):

\[ ku - \omega^2 mu = 0 \]  

(3)

If equation (3) has a solution, it can be obtained that:

\[ \Delta = |k - \omega^2 m| = 0 \]  

(4)

Equation (4) can be expanded into a polynomial equation of degree \( n \) of \( \omega^2 \). The solution of the equation is \( n \) values of \( \omega^2 \). When \( k \) and \( m \) are positive definite matrices, \( n \) positive real roots can be obtained, and then \( n \) natural frequencies of the system can be obtained \( \omega_1 \leq \omega_2 \leq \cdots \leq \omega_n \), \( \omega_i \) is the \( i \)-th natural frequency. By introducing \( \omega_i \) into equation (3), the \( i \)-th mode array can be obtained.

The accuracy of the finite element model directly determines the reliability of the calculation results in the finite element simulation analysis of the dual steel disc. In order to ensure the efficiency and speed of simulation, the chamfering of the original structure of dual steel disc is removed. As shown in figure 2, the finite element model is divided into 18512 nodes and 12834 elements. The element parameter settings of the finite element model are shown in table 1.

![Figure 2. Finite element model of dual steel disc](image)

| Table 1. Material parameters of dual steel disc |
|-----------------------------------------------|
| Elastic Modulus (GPa) | Density (kg · m⁻³) | Poisson's Ratio |
|-----------------------|---------------------|-----------------|
| 210                   | 7850                | 0.3             |

The Optistruct module in HyperMesh is used for analysis and calculation. The first six order modal frequency values are shown in table 2, and the corresponding first six order modal array is shown in figure 3.
Table 2. First six modal frequencies of dual steel disc

| Modal Order Number | 1   | 2   | 3   | 4   | 5   | 6   |
|--------------------|-----|-----|-----|-----|-----|-----|
| Natural Frequency (Hz) | 331 | 917 | 1275| 1744| 2043| 2805|

Figure 3. Finite element model of dual steel disc. (a) The first mode of dual steel disc, (b) The second mode of dual steel disc, (c) The third mode of dual steel disc, (d) The fourth mode of dual steel disc, (e) The fifth mode of dual steel disc, (f) The sixth mode of dual steel disc.

The modal of the dual steel disc is shown in figure 3. As matters stand, the external excitation of brake friction pair mainly comes from brake piston vibration and brake load fluctuation. In the process of braking, as the friction coefficient presents a negative slope changes with the relative sliding, the friction surface will stick-slip. The friction self-excited vibration may occur in the system, resulting in the vibration of the dual steel disc. Hence, the rigid flexible coupling analysis of the friction pair of the brake is carried out to analyse the dynamic performance of the dual steel disc in the braking process.

3.2. Modal experiment of dual steel disc of wet multi-disc brake
The dual steel disc of brake is a thin plate component with small volume and light weight. How to test the modal of such components accurately and quickly is a vital technical problem. Up to now, the contact sensors for modal testing include acceleration sensor, resistance strain gauge, piezoelectric thin film and others. While non-contact measurement includes laser vibration meter, high-speed camera, near-field sound pressure measurement and others. The sensor of contact measurement will lead to the additional mass of the original structure, which will affect the modal parameter identification of the structure. For components such as dual steel disc, it is impossible to accurately measure and identify its modal by using acceleration sensor. Due to the symmetry and simplicity of the dual steel disc structure, the modal configuration of the finite element analysis is reliable. Hence, this paper tests the
natural frequency of the dual steel disc by near-field acoustic radiation, and compares the results with the results of finite element analysis to verify the accuracy of the finite element model.

The structure is forced to vibrate under the action of an instantaneous force, and then vibrates freely with the natural frequency $\omega$. The vibration wave generated propagates in the air and causes the pressure change at a certain point in the air medium. This process can be regarded as the unidirectional action of the structure on the air.

The normal velocity and acceleration of structural vibration are as follows:

$$v(t) = Ae^{i\omega t}$$  \hspace{1cm} (5)

$$a(t) = v'(t) = i\omega Ae^{i\omega t}$$  \hspace{1cm} (6)

Where $A$ is the amplitude of normal velocity of structure vibration and $\omega$ is vibration frequency.

The near-field sound pressure of the structure is expressed as:

$$p(d, t) = i\omega \rho \frac{V}{4\pi d} e^{i(\omega t - nd)}$$  \hspace{1cm} (7)

Where $V$ is the amplitude of the volume change velocity, $\rho$ is the air density, and $n$ is the wave number.

$$V = 2AS$$  \hspace{1cm} (8)

Where $S$ is the surface area of vibrating structure. The equation (5), equation (6) and equation (8) are brought into equation (7), the following results can be obtained:

$$p(d, t) = i\omega A\rho \frac{S}{2\pi d} e^{i(\omega t - nd)} = a(t)D$$  \hspace{1cm} (9)

In equation (9), $D$ is a constant. According to equation (9), the sound pressure frequency radiated by the structure in the near field is the same as the vibration frequency of the structure, and the natural frequency of the structure can be measured according to the sound pressure frequency.

In this work, the sound pressure is measured in a semi anechoic chamber with a sound pressure measurement and analysis system. The distance between the acoustic sensor and the dual steel disc is 20mm. The test experiment is shown in figure 4. The acoustic pressure signal of the dual steel disc can be measured by hammering the dual steel disc to cause free vibration of the dual steel disc. The frequency spectrum of the sound pressure signal can be obtained by FFT processing of the sound pressure signal, as shown in figure 5.

Figure 4. Sound pressure test experiment
Figure 5. Sound pressure spectrum

Table 3. Comparison of natural frequency between FEA and Sound Pressure Experiment

| Modal Order Number | Natural Frequency by FEA (Hz) | Natural Frequency by Sound Pressure Experiment (Hz) | Relative Error |
|--------------------|-------------------------------|----------------------------------------------------|----------------|
| 1                  | 331                           | 336                                                | 1.5%           |
| 2                  | 917                           | 912                                                | 0.5%           |
| 3                  | 1275                          | 1200                                               | 6.2%           |
| 4                  | 1744                          | 1728                                               | 0.9%           |
| 5                  | 2043                          | 1936                                               | 5.5%           |
| 6                  | 2805                          | 2776                                               | 1.0%           |

According to figure 3, the first, second, fourth and sixth order modes of the dual steel disc are mainly the deformation and overturning at the outer edge of the dual steel disc, and the third and fifth order modes of the dual steel disc are mainly the deformation and overturning of the gear ring in the dual steel disc. According to the relative error analysis in table 3, it can be seen that the first, second, fourth and sixth order natural frequency errors of the dual steel disc finite element model are all less than 2%, and the third and fifth order natural frequency errors are 6%. The errors are mainly caused by the simplification of the internal gear ring model. The main deformation of dual steel disc in practical work is the surface deformation of dual steel disc, so the finite element model is credible.

4. Rigid-flexible coupling dynamic analysis of wet multi-disc brake

4.1. Establishment of rigid-flexible coupling dynamic model of brake

In the virtual prototype model of wet multi-disc brake, each component is relatively independent, and the motion state is determined by the motion pairs and contact settings. The traditional Craig-Bampton method is suitable for linear dynamic problems with small motion range. In HyperMesh, the Craig-Bampton method is used to calculate the modal matrix and modal coordinates of the dual steel disc, which makes it suitable for multi-body system dynamics analysis. Based on the finite element theory and modal synthesis method, the mnf file of dual steel disc is generated by HyperMesh. After the mnf
file is imported into ADAMS to replace the original rigid component, the rigid flexible coupling analysis is implemented in ADAMS.

For the wet multi-disc brake, the motion constraints of each component are set as follows: the connecting shaft is set as the rotating pair, the piston is set as the moving pair, the friction disc is set as the moving pair, and the flexible body dual steel disc is not set with the kinematic pair. The dual steel disc and the connecting shaft of the flexible body are set as rigid-flexible body contact, and the dual steel disc and friction disc of the flexible body are set as rigid-flexible body contact. The freedom of the flexible body is limited by setting rigid-flexible coupling contact with the steel disc. The Coulomb friction effect with static friction effect is set between the flexible body and the friction disc, the dynamic friction coefficient is set to 0.1, and the static friction coefficient is set to 0.14.

In ADAMS, through the rigid body dynamics analysis, it can be seen that due to the effect of spring piston, the components closest to the piston are the most affected by the impact force on the dual steel disc and friction disc. Hence, the first dual steel disc closest to the piston is selected to replace the flexible body for rigid-flexible coupling analysis. When the brake is in emergency braking condition, the torque at the load end of the connecting shaft is 55000 N·mm, the stiffness of the piston spring is 344 N·mm⁻¹, the spring preload is 5877 N, and the initial speed of the connecting shaft is 173 rad·s⁻¹. The piston is affected by hydraulic pressure at the same time, and the direction of hydraulic pressure is opposite to that of spring force. After the piston hydraulic pressure is released, the brake presses the friction pair by spring force to realise braking.

4.2. Dynamic analysis of friction engagement process

The virtual prototype simulation analysis of the wet multi-disc brake is completed by ADAMS. The flexible body file generated by the finite element modal calculation is used to replace the rigid body component in the original brake for rigid-flexible coupling analysis. Through the rigid-flexible coupling multi-body dynamic analysis, this work explores the impact and friction jointing state of the dual steel disc in the braking process of the wet multi-disc brake, and researches the dynamic characteristics and stress-strain state of the dual steel disc.

The speed change curve of the connecting shaft is shown in Figure 6. The speed of the connecting shaft decreases uniformly and the deceleration is completed within 0.28s.

![Figure 6. The speed change curve of the connecting shaft](image)

The normal contact force of friction pair fluctuates around 5000N during the braking process of 0–0.28s, and the maximum normal contact force can reach 17000N. After 0.28s, the normal contact force of friction pair is equal to the spring pressure, and the contact force does not change. It can be seen from the time-domain variation curve of normal contact force that the amplitude of contact force changes greatly before 0.075s, and the fluctuation of contact force is obvious. At the beginning of
contact, the speed of the connecting shaft is always above 120 rad·s⁻¹. After 0.075s, with the decrease of the speed of the connecting shaft, the pressure piston is compressed gradually, and the contact force fluctuation of the friction pair decreases gradually.

![Figure 7. Time-frequency domain characteristics of normal contact force of friction pair](image)

From the spectrum of normal contact force of friction pair, it can be seen that the frequency component of contact force is mainly distributed in the high frequency region from 10000Hz to 25000Hz, and the amplitude of low frequency area with frequency below 5000Hz is small. According to the spectrum of Figure 7, it can be seen that the normal contact force of high frequency plays a leading role in the engagement and collision of brake friction pairs in the engagement process. In the braking process, the dual steel disc may produce vibration under high frequency excitation, which affects the braking characteristics.

![Figure 8. Change curve of piston spring force](image)

According to the change curve of piston spring force in Figure 8, the spring force suddenly increases from 5845N to 5848N at about 0.027s. This phenomenon indicates that collision occurs during the pressing process of friction pair at the initial stage of braking, and the piston is ejected and then compressed. In the later engagement process, the spring force still fluctuates slightly around
5845.5N. During the whole braking process, the spring force also changes abruptly at 0.06s and 0.25s respectively. After 0.28s of braking, the speed of connecting shaft is reduced to 0, and the spring force is no longer fluctuating.

4.3. Structural performance analysis of brake dual steel disc

Through the rigid flexible coupling dynamic analysis of Adams, the stress contour at different time in the whole simulation process can be viewed in the Adams post-processing module. Figure 9 shows the mises stress contour of dual steel disc at 0.027s, 0.066s and 0.25s. It can be seen from the mises stress contour of the dual steel disc that the mises stress at the tooth root of the internal tooth of the dual steel disc is the maximum, and the maximum mises stress of the dual steel disc is 20MPa. The stress is small at the top of the inner ring tooth and the outer edge of the dual steel disc. On the surface of the dual disc, the stress decreases gradually along the radial direction.

From the mises stress contour, it can be seen that the circumferential stress distribution of the dual steel disc surface is inhomogeneous, which is mainly caused by the collision during the friction jointing process. According to the stress contour of the dual steel disc at different times, it can be seen that the maximum mises stress of the dual steel disc in the initial contact stage of the braking process is greater than that in the later contact stage of the braking process, which is consistent with the change of the spring force and the normal contact force of the friction pair.
Figure 10 shows the deformation contour of dual steel disc at 0.027s, 0.066s and 0.25s. It can be seen from the deformation contour of the dual steel disc at different time points that at 0.027s and 0.25s, the whole surface of the dual steel disc presents wavy deformation, and the deformation of the dual steel disc is symmetrical at 0.066s. The deformation of the dual steel disc surface is distributed in 0~10 μm. The results show that the deformation of the dual steel disc surface is consistent with the distribution of the mises stress on the surface of the dual steel disc.

According to Figure 9 and Figure 10, the analysis can provide suggestions and guidance for the structural design of the dual steel disc. In order to reduce the deformation of the dual steel disc in the jointing process, the structure can be changed to improve the stiffness of the dual steel disc. At the same time, the chamfering at the tooth root of the gear ring in the dual steel disc can reduce the maximum stress of the dual steel disc in braking, thus improving the fatigue life of the dual steel disc and increasing reliability of the dual steel disc.

5. Conclusion

- The finite element flexible body model of brake dual steel disc is established, and the accuracy of the finite element model is verified by experimental test.
- The virtual prototype simulation experiment of the brake based on rigid-flexible coupling analysis is completed. The bending vibration of the dual steel disc in the braking process will affect the braking performance.
- By increasing the stiffness and damping of the dual steel disc, the vibration of the dual steel disc can be reduced and the brake performance can be improved.
- Through virtual prototype simulation analysis, the weak part of dual steel disc structure is found, and the direction of dual steel disc structure design is pointed out.
- The dynamic simulation analysis based on rigid-flexible coupling can research some phenomena and problems which are difficult to be found in actual measurement, and provide guidance and suggestions for the structural design of components.

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