Research on Braking Vibration Characteristics of Grooved Brake Interface of Disc Brake

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Abstract. With the good performance and wide application of surface texture in various aspects, more and more scholars are devoted to investigating the effect of texture interface on tribological properties. Disc brake friction pair is a typical friction brake interface. In order to consider the influence of grooved brake interface on vibration characteristics during braking, a simplified finite element model of brake interface is established in this paper. By processing parameterized grooves with different lengths and angles on the surface of the brake disc, the effects of “blind groove”, “through groove” and “angle” on the vibration characteristics of the brake disc during braking process are simulated and studied, which can provide reference for the structural design and optimization of the brake disc texture.

1. Introduction

The brake has the function of decelerating, stopping or maintaining the stop state of the moving machinery, and it is an important part to ensure the safety of the device[1]. At present, surface texture technology shows good application potential in tribology. It is of great significance to apply texture technology to the surface of brake pairs to study the influence of texture parameters on braking performance during braking process. Andersson et al. studied the effect of texture on surface lubrication behavior on steel surface. The results showed that texture greatly reduced surface friction and wear[2]. Saw et al. studied the influence of groove arrangement on brake system instability. The results show that when the groove inclines at a certain angle, the degree of unstable vibration of the brake system decreases[3,4]. Wang et al. proposed that the impact of the grinding ball on the groove would produce low-frequency friction fluctuation. As a discontinuous excitation, it can effectively suppress the formation of high-frequency components of interface friction and achieve the effect of reducing friction noise[5]. Based on the common operating conditions of high-power wind turbine disc brake, this paper studies the influence of brake interface groove parameters on the vibration characteristics of brake disc by establishing parametric grooves on the surface of brake disc, which can provide reference for the structural design and optimization of brake disc.

2. Establishment of brake rigid body model

2.1. Establishment of three-dimensional brake model

The brake mainly includes upper and lower calipers, compensation mechanism and friction parts components. This paper focuses on the influence of groove on braking process, without considering secondary braking, wear and other conditions. Therefore, the compensating mechanism and brake
calipers can be subtracted, but the inertia provided by them should be equivalent attached to the central axis connected with the brake disc. The upper and lower friction pairs in the brake are symmetrical, so the center plane of the brake disc is used as the symmetry plane, and the upper half of the disc is combined with the upper brake to model. The model is shown in Figure 1.

Figure 1. Brake rigid body three-dimensional model.

2.2. Groove type brake interface parameter establishment

The model in Figure 1 is based on the material and size of friction pairs components in wind turbine disc brakes. The brake disc is of arc structure, and the material is copper-based powder metallurgy. Its size is 520mm in Inner diameter, 760mm in outer diameter, 26mm in thickness, and 36° in center angle. The material of the brake disc is Q345b, with an outer diameter of 800mm, an inner diameter of 200mm and a thickness of 20mm. The material parameters are shown in Table 1.

| Brake component | Material                      | $E$ (GPa) | $\mu$ | $\rho$ (kg/m$^3$) |
|-----------------|-------------------------------|-----------|-------|-------------------|
| Brake disc      | Q345b                         | 210       | 0.31  | 7850              |
| Brake pad       | Copper-based powder metallurgy| 5.2       | 0.3   | 5250              |

On the upper surface of the brake disc, four uniformly distributed rectangular grooves are formed according to the angle, and the length is $l$, the cross-sectional width is $a=2$mm, and the height is $b=1$mm. The groove angle of the brake disc is defined as shown in Figure 2. Taking the brake center line of the matching position as the boundary, the angle between the direction of the brake disc rotation speed $v$ and the groove edge of the brake disc periphery is defined as $\alpha$ [6] ($0^\circ < \alpha < 90^\circ$). For the convenience of the following description, it is called a small angle, and its complementary groove angle $\beta=180^\circ-\alpha$ ($\alpha > 90^\circ$), which is called a large angle.

In Figure 3, a simulation angle groove is set up every 15° between 0° and 180°. In Figure 3(a), the groove length $l$ of the surface of the brake disc is limited according to the radius $R1$, $R2$ and a certain angle, and at least one side of the length direction of the groove is outside the radius of two curvatures of the brake pad. In this case, the groove on the brake disc surface is called “through groove”. In Figure 3(b), the groove length $l$ of the surface of the brake disc is a fixed value of 120mm, and the length thereof is included in the two curvature radius of the brake pad. In this case, the groove on the surface of the brake disc is called “blind groove”.

Figure 2. Definition of groove angle.

Figure 3. Groove parameters of the brake disc surface in the mating view.
3. Establishment of brake finite element model

The modeling problem of contact collision has always been a very important and difficult problem to solve in multi-body dynamics systems. So far there is no universal model. The models of such problems are mostly based on experience and related experiments[6]. If the deformation of the object is not considered, the ideal solution result can be obtained by the multi-rigid method and the rigid body contact theory[7]. However, during the braking process, the brake disc and the brake pad in the brake belong to the easily deformable component, and the research focuses on the instantaneous deformation of the component in the contact collision and the result of the deformation reaction on the contact collision. Therefore, object deformation cannot be ignored, it cannot be limited to rigid body and classical collision model assumptions[8]. In this paper, the deformable brake disc and brake pad are flexibly processed in the HyperMesh software, then import both into the ADAMS software to replace the rigid body parts. Finally, the simulation is carried out after the constraints of each component and the initial conditions are set. The constraints are shown in Table 2.

| Constraint component                  | Constraint type     | Constraint component                  | Constraint type     |
|---------------------------------------|---------------------|---------------------------------------|---------------------|
| Brake disc and central shaft          | Fixed pair          | Central shaft and ground              | Rotating pair       |
| Upper brake pad and ground            | Mobile pair         | Upper brake pad and brake disc        | Contact pair        |

The simulation conditions adopt the common working conditions of the wind turbine brake, that is, the initial speed of the brake disc is \( n=1000\text{r/min} \), the braking force applied to the brake pad is \( F=17000\text{N} \), and the friction coefficient between the brake disc and the brake pad is \( \mu=0.3 \). The completed dynamic model diagram is shown in Figure 4.

![Figure 4. Brake rigid-flexible coupling model.](image)

4. Vibration characteristics analysis of grooved brake disc

This paper does not consider wear and tear, and investigates the influence of the groove interface on the braking performance based on braking time and braking stability. Braking time can be expressed by the angular velocity or kinetic energy curve of the brake disc. This paper uses the kinetic energy curve. Because the square of the velocity or angular velocity is included in the kinetic energy calculation formula, the braking rate can be more intuitively distinguished.

4.1. Simulation results and analysis of through-groove brake disc

As mentioned above, \( \alpha \) smaller than 90° is called a small angle, and \( \beta \) larger than 90° is called a large angle, where \( \beta=180-\alpha \) is a complementary angle of \( \alpha \). In Figure 5(a), the vibration amplitude curves of the through groove brake discs of 15°-30° and their corresponding complementary angle sections are almost all in a straight line. That is to say, when \( \alpha \) is small or \( \beta \) is large, the change of angle has no obvious influence on the vibration of brake disc during braking. In the 30°-150° angle range, the vibration amplitude curve rises sharply first, and the upward trend is moderated at 60°. The amplitude reaches the maximum at 75°, and then the curve drops sharply and the downward trend becomes slower at 105°. At the limit of 90°, the complementary angle is \( \alpha-\beta \), and the vibration amplitude of the small angle in the through groove brake disc is larger than the large angle. In Figure 5(b), the strain energy amplitude curve of the brake disc has the same regularity as the vibration amplitude curve in Figure 5(a), and it is considered that the vibration and strain of the brake disc mutually promote each
other. That is, the larger the axial impact, the greater the amount of strain change, and the better the strain recovery performance of the impact, which in turn exacerbates the vibration amplitude. From Figure 5(a) and (b), the two figures are normal distributions with 90° as the center line and offset to the left. One reason for the offset is that when the through groove angle is close to 90°, the span of the groove length becomes smaller, and the shock oscillation time is shortened when the brake pad passes the groove, and the vibration and strain do not form a chain interaction effect. Another reason is that the vibration and impact of the small-angle grooved brake disc during braking are greater than or equal to the large angle.

![Figure 5. Through-groove brake disc amplitude curve.](image)

As shown in Figure 6, the kinetic energy curve of the 45°-90° through groove brake disc is clearly separated from other angle curves, and the kinetic energy declining trend is \( E_{75°} > E_{60°} > E_{45°} > E_{90°} > \) other. Referring to Figure 5, when the groove angle \( \alpha \) is small, the amplitude of the vibration/strain energy of the brake disc is close to the corresponding complementary angle \( \beta \) amplitude. This phenomenon causes the two curves on the kinetic energy curve to coincide. When the groove angle \( \alpha \) is gradually increased and gradually approaches 90°, the vibration/strain energy amplitude of the brake disc is significantly different from the amplitude of the corresponding complementary angle. At the same time, the kinetic energy curves of the complementary angle \( \alpha - \beta \) of the through-groove brake discs is obviously separated, and the kinetic energy of the through groove brake disc with larger amplitude of vibration/strain energy decreases faster. It is concluded that the magnitude of the vibration/strain energy of the brake disc has a one-to-one correspondence with the speed of the brake. That is, the greater the axial vibration/strain energy amplitude of the brake disc, the faster the kinetic energy of the corresponding angle through-groove brake disc drops. Moreover, with 90° as the boundary, in the complementary angle pair \( \alpha - \beta \), the small angle groove brake is not less than the large angle groove.

![Figure 6. Kinetic energy drop curves of through-groove brake discs.](image)

The above analysis reflects the influence of the through-groove brake disc on the braking performance, and the change in the groove angle has a significant effect on the braking process. In short, the change in the through-groove angle significantly affects the fluctuation amplitude of the vibration/strain energy, which in turn affects the braking kinetic energy drop curve. The amplitude of
the through grooved brake disc varies greatly, the vibration is more severe, and the energy loss of the system is also faster, which is the main factor affecting the braking rate. Therefore, the through-groove brake disc is suitable for working conditions where the system vibration requirements are not critical but require rapid braking.

4.2. Simulation results and analysis of blind groove brake disc

In Figure 7(a), the vibration amplitude curve first oscillates increase with the angle change; at 90°, the curve rises sharply and the amplitude reaches the maximum at 105°; then the curve decreases to a large extent. The strain energy amplitude curve in Figure 7(b) is consistent with the change trend of Figure 7(a), where they also have a mutually reinforcing relationship. However, with 90° as the boundary, the vibration amplitude of the large-angle blind slot brake disc is larger than the small angle. Because under the premise that the blind groove brake disk has a small degree of fluctuation, when the groove is turned to the brake pad with the brake disc, in the complementary angle pair \( \alpha - \beta \), the large angle \( \beta \) blind groove is in contact with the brake pad first on the side away from the central axis, and the small angle \( \alpha \) blind groove is opposite, as shown in Figure 8. However, the closer to the outside along the radial direction of the brake disc, the greater the linear velocity, and the shock vibration caused by the brake pad passing through the groove is relatively large.

Figure 7. Blind groove brake disc amplitude curve.

Figure 8. Contact condition of the brake pad when passing through the blind slot.

In Figure 9, the kinetic energy curves of the blind grooved discs at different angles overlap. Because the blind groove is completely contained within the two curvature radii of the brake pad, and the arc surface of the two radii of the brake pad does not directly participate in the impact when passing through the groove. Therefore, the braking process is gentler than the through groove, and the vibration/strain energy amplitude is also much smaller than the through groove, which is not enough to cause the kinetic energy to drop significantly. In this case, the change in the angle of the blind groove has little effect on the braking process.
The above analysis reflects the effect of the blind groove brake disc on the braking performance during braking. The change of the blind groove angle has no obvious influence on the kinetic energy drop of the brake disc. The influence on the braking performance is mainly reflected in the vibration/strain energy amplitude fluctuation (small compared with the through-groove amplitude fluctuation). This paper focuses on the influence of the groove parameters on the braking process, and does not consider the advantages of heat dissipation and chip resistance of the groove on the surface of the brake disc under actual working conditions. Therefore, the blind groove brake disc can be used in a working condition that is dedicated to heat dissipation, chipping, and requires less system vibration, but has no obvious requirements on braking time.

5. Conclusions
In this paper, by establishing parameterized grooves on the surface of the brake disc, the related laws of vibration and kinetic energy curves of the series of grooves in braking are obtained, which provides reference for the design and optimization of the texture structure of the brake disc.

- The change of the through groove angle of the brake disc affects the vibration/strain energy amplitude change of the brake disc, further affecting the braking rate; the vibration/strain energy amplitude of the small-angle groove is greater than or equal to the large-angle groove, and the larger the vibration/strain energy amplitude of the brake disc is, the faster the brake is.

- The change of the blind groove angle of the brake disc has no obvious influence on the braking rate. The influence on the braking performance is mainly reflected in the fluctuation of the vibration/strain energy amplitude of the brake disc; the large-angle blind groove amplitude fluctuation is larger than the small-angle blind groove, but much smaller than the through groove.

- The strain energy amplitude and vibration amplitude curve of the grooved brake disc have a good consistency, and the two have mutually reinforcing relationship.

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