Analysis of noise during unbalanced braking of disc brake friction plates

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Abstract. The non-equilibrium contact phenomenon caused by the time difference between the friction plates on both sides and the brake disc during the braking process of the disc brake was analyzed. The brake noise problem in this non-simultaneous clamping condition was studied. Then a three-dimensional assembly model was established and the finite element method was used to find the complex mode and its instability coefficient under different operating conditions when the brake was in unbalanced contact. The results show that during non-equilibrium braking, the inherent frequency and low-frequency stability of the braking system decreases, the instability modes increase significantly, and the shorter time difference leads to a strong tendency of whistling.

1. Introduction

Disc brake is a widely used car braking device, its structure mainly includes brake discs, friction plates, calipers, hydraulic cylinders and pistons, etc. The brake disc is directly connected to the wheel hub of the car and rotates synchronously with the wheel. When the brake disc is installed on both sides of the frame of the friction plates at the same time clamping, will be subject to the role of friction in the opposite direction and deceleration, thereby reducing the car driving speed. The brake is an essential part of the car to control the speed and ensure the safety of driving.

However, in the process of braking, vibration and noise often occur, and these noises are both low-frequency noise and high-frequency whistling. This will seriously affect the ride comfort of the passengers, and make the driver prone to fatigue, so that it may misjudge the vehicle and road conditions and cause dangerous driving.

For the problem of automotive brake noise, scholars have done a lot of work. Some scholars put forward the hammering excitation theory [1], using the principle of mechanical impact to explain the mechanism of brake noise generation. The brake will be deformed during the braking process due to frictional heating, resulting in the impact phenomenon between the brake disc and the friction plate in contact, i.e., the hammering effect, which causes brake noise.

Some scholars use modal coupling theory to explain brake noise. When the brake is working, if the contacting parts have similar modal characteristics, noise will be generated due to coupling.

As disc brakes have a certain time difference between the contact of friction plates and discs on both sides during braking, the existence of this time difference will lead to a non-equilibrium state of pressure...
on both sides of the disc, and some scholars point out that if the sliding speed between the two friction surfaces is higher than a certain critical value, elastic instability will occur [2]. However, there is no research in the existing literature on the brake noise of disc brakes under non-equilibrium contact conditions, and this paper will analyze the brake noise under such non-equilibrium contact conditions.

2. Vibration analysis theory about brakes

2.1. Contact stiffness theory

When two objects come into contact with each other, the contact area and pressure distribution will change to a certain extent due to the action of external forces. Disc brakes in the braking, its material properties will be affected by the temperature and change, and because both sides of the friction plate and the brake disc contact time is not exactly the same, resulting in the braking process will appear twice the external excitation and sudden changes in friction, and in the actual working condition, the driver to the brake pedal load process has a strong non-linear characteristics, the brake disc and the friction plate contact conditions are not constant, is Typical nonlinear contact process. According to Herzi contact mechanics theory, two surfaces in contact with each other, if there are nonlinear factors, the initial contact of a point or a surface, will be affected by external forces and evolve into a finite contact area.

The normal variables of the contact surface in the nonlinear state of the brake [3], can be expressed as follows.

\[ \delta = 2G^{(D-1)}(\ln \gamma)^{\frac{1}{2}}(2r')^{(3-D)} \]  

Where: \( G \) is the fractal roughness parameter; \( \gamma \) is the contour spectral density parameter, generally taken as 1.5; \( D(1<D<2) \) is the contour fractal dimension; \( r' \) is the truncated micro-convex body radius.

\[ F_n(\delta) = \frac{4}{3} E^* R^{\frac{3}{2}} \delta^{\frac{3}{2}} \]  

Where: \( E^* \) is the equivalent modulus of elasticity; \( R \) is the radius of curvature.

\[ E^* = \frac{(1-\mu_1^2) E_1 + (1-\mu_2^2) E_2}{E_1} \]  

\[ R = \frac{2^{(5-D)} \pi^{-D/2} G^{(D-2)} (\ln \gamma)^{\frac{1}{2}}}{(a')^{D-3/2}} \]  

Substitute equations (1) and (4) into (2) to obtain:

\[ F_n(\delta) = \frac{2}{3} \frac{E^* G^{2(D-2)}(\ln \gamma)^{\frac{1}{2}}(a')^{D-3/2}(2r')^{9-D/2}}{\pi^{D/4}} \]  

For the total normal elastic force \( P^* \) and the distance \( d \) from the rigid surface to the datum, the normalized expression can be shown in the following equation. \([6]\)

\[ P^* = \frac{F_n R^{\frac{3}{2}}}{N E^* \sigma_i^{\frac{3}{2}}} \]  

\[ d^* = \frac{d - Z}{\sigma_i} \]  

Where: \( \sigma_i \) is the standard deviation; \( N \) is the number of vertices in contact; \( Z \) is the height of the vertex to the reference.

From equation (6), the contact stiffness is obtained as follows:

\[ K_c = \frac{N E^* \sigma_i^{\frac{3}{2}}}{R} \left[ \frac{d^*}{dP^*} \right]^{\frac{1}{2}} \]

\[ \]
2.2. Modal Analysis Theory

Vibration mode is the main factor of disc brake noise. Therefore, by decomposing the mode of the braking system, the natural frequency and vibration mode of the system can be solved, and then the self-excited vibration and noise of the system can be revealed. Its kinetic equation is:

\[ M\ddot{u} + C\dot{u} + Ku = F(t) \] (9)

Where, \( M \) is the mass matrix of the object; \( C \) is the object damping matrix; \( K \) is the system stiffness matrix; \( \ddot{u} \) is the node acceleration vector; \( \dot{u} \) is node velocity vector; \( u \) is the node displacement vector; \( F(t) \) is the time-varying load function applied to the system.

In order to simplify the calculation, the damping matrix \( C \) is ignored and the load function \( F(t)=0 \), then the vibration equation of the system in free vibration state can be obtained as follows:

\[ M\ddot{u} + Ku = 0 \] (10)

In a linear vibration system where \( M \) matrix and \( K \) matrix are constant, the form of free vibration is simple harmonic motion:

\[ u = \phi_i \cos \omega_i t \] (11)

Under the condition of ignoring the damping coefficient, the vibration equation is solved by the basic eigenvalue algorithm:

\[ K\phi_i = \omega_i^2 M\phi_i \] (12)

where: \( \phi_i \) is the eigenvector, representing the vibration mode of the \( i \)-th order of free vibration; \( \omega_i \) is the \( i \)-th order angular frequency of self-oscillation, representing the fixed frequency of the \( i \)-th order mode, that is, the eigenvalue.

Free vibration frequency \( f_i=\omega_i/2\pi \) (13)

When the brake is braking, due to the appearance of friction, the whole braking system produces coupling effect. At this time, the motion equation of the system is as follows [7]:

\[ M\ddot{x} + C\dot{x} + (K + \mu K_f)x = 0 \] (14)

Where: \( \mu \) is the friction factor of brake system; \( K \) is the frictional contact stiffness matrix.

From the analysis of equation (14), it is clear that the coupling effect of friction makes the stiffness matrix asymmetric, which leads to the asymmetry of the eigenmatrix. From the mathematical point of view, the eigenvalues of the asymmetric matrix take the form of complex numbers under certain conditions.

The eigenvalue of the system is the modal frequency. From control theory, we know that if the complex eigenvalue of the system has a negative real part, the system is stable. Conversely, when the real part is positive, the system is unstable. Therefore, the results of the solution for the brake system are unstable if the real part of the complex eigenvalue of the mode is positive, and noise may occur in engineering [8]. Therefore, the prediction of noise can be achieved by solving for the complex mode of the brake system.

The characteristic equation of Eq. (14) is given by:

\[ \det(s^2M + sC + K + \mu K_f) = 0 \] (15)

\[ s = \sigma + j\omega \] (16)

Where, \( s \) is the complex eigenvalue of the system; \( \sigma \) is the real part of the characteristic value of the system, that is, the damping coefficient; \( \omega \) is the imaginary part of the system eigenvalue, that is, the natural frequency.

To evaluate the instability state of the system for a certain order of modes, the damping ratio \( \zeta \) of the complex mode is defined as [9].

\[ \zeta = -\frac{\sigma}{\pi\omega} \] (17)

The negative sign in Eq. (17) represents negative damping, and the modal instability coefficient \( \gamma \) is written as:

\[ \gamma = \frac{\sigma}{\pi\omega} \] (18)

If the value of \( \zeta \) of the system is negative, it means that in this mode, the damping not only does not consume energy, but also feeds new energy into the system, thus causing self-excited vibration, so it is
an unstable mode. Combined with the application in real engineering, usually the unstable mode is distinguished by the value of its coefficient $\gamma$. When its value is greater than 0.01, it is considered as unstable mode, and conversely when the value is less than 0.01, it is considered as stable mode. [6]~[12]

3. Simulation analysis of the braking process

3.1. Modeling of disc brakes

Taking the disc brake of a certain mass-produced passenger car as the research object.

The vibration noise in the braking process mainly comes from the contact between the brake disc and the friction plate. In order to improve the calculation efficiency and reduce the difficulty of mesh division, and to ensure the accuracy of the simulation, the model of each component was simplified under the condition that the core structure of the brake disc and friction plate would not be changed, and the rounded corners and chamfers in the three-dimensional structure were deleted, and the main structural dimensions are shown in Table 1.

| Brake disc structure                  | numeric value |
|--------------------------------------|---------------|
| Brake disc outer diameter /mm        | 330           |
| Brake disc inner diameter /mm        | 150           |
| Brake disc thickness /mm             | 24            |
| Ventilation hole thickness /mm       | 8             |
| Ventilation hole wrap corner /˚      | 10            |
| Reinforced rib wrap angle /˚         | 8             |
| Mounting tab diameter /mm            | 140           |
| Positioning hole diameter /mm        | 16            |
| Mounting tab thickness /mm           | 8             |
| Mounting tab height /mm              | 50            |
| Friction plate outer diameter /mm    | 308           |
| Friction plate inner diameter /mm    | 220           |
| Friction plate thickness /mm         | 14            |
| Friction plate wrap angle /˚         | 60            |

The simplified assembly model was imported into ANSYS Workbench, and the materials used for this simulation were selected. The material of the friction plate is resin matrix composite, and the brake disc and the rest of the components are gray cast iron HT250. The added materials were given to each component, and the model was meshed. Judging by the parameters of cell mass, cell aspect ratio and Jacobi ratio, the meshing achieved a good result. The finite element meshing of the disc brake assembly model is shown in Fig. 1, including 102,157 cells and 169,059 nodes.
3.2. Load variation during unbalanced contact

3.2.1 Working Principle of Disc Brake Braking Process
As shown in Fig. 2, for disc brakes, the caliper is generally designed to move axially with respect to the disc in structural form. The inner side of the brake disc is equipped with a hydraulic cylinder, and the outer side of the fixed friction plate is attached to the caliper body. When braking, brake fluid is pressed into the cylinder, the piston moves to the left under the action of hydraulic pressure, pushing the movable friction plate also moves to the left and pressed against the brake disc, so the brake disc to the piston a reaction force to the right, so that the piston together with the caliper as a whole along the guide pin to the right, until the brake disc on the left side of the fixed friction plate also pressed against the brake disc. At this time, both sides of the friction plate are pressed on the brake disc, the friction plate clamps the brake disc, generating a friction torque to prevent the wheel from rotating to achieve braking. From the working process of the disc brake can be seen, the two friction plates are not clamping the brake disc at the same time, therefore, the brake disc in the process of braking will be subject to unbalanced load.

3.3. Simulation of unbalanced braking process
According to the test data, the brake pressure under emergency braking was set to 5 MPa and the braking time was set to 2 s. The friction plates on both sides had a 1 mm gap with the brake disc, and ideally the friction plates on both sides were simultaneously extruded to the middle and contacted with the brake disc at the same time, and the displacement-time diagram is shown in Fig. 3. In order to simulate the actual working condition of the brake, the moment of contact between the brake disc and the friction plate on the side with the piston is set to the zero moment. There is a certain time delay between the contact moment of the brake disc and the friction plate on the opposite side of the piston and the zero moment. The braking state under ideal condition is set as operating condition 1. In order to study the influence of the contact time difference between the friction plate and the brake disc on the stability of the brake system under the non-equilibrium contact condition, the simulation conditions were set as operating conditions 2-4 with a time difference of 0.1 s, 0.2 s and 0.3 s, respectively. The friction plate displacement-time relationships under different contact time differences are shown in Fig. 4.

![Fig. 3 Displacement time diagram of friction plate in ideal state](image)

![Fig. 4 Displacement time diagram of friction plate under nonlinear contact state](image)
4. Calculation results and analysis
The relevant settings were made in ANSYS Workbench, and the first 30 orders of inherent frequencies and vibration patterns of the brake were obtained for each operating condition. The three operating conditions in the unbalanced contact condition have approximately equal inherent frequency values, so the inherent frequency values of the brake in the ideal condition and the unbalanced contact condition are organized as shown in Fig. 5.

![Fig. 5 Distribution of brake natural frequency](image)

4.1. The inherent frequencies and unstable modes under ideal conditions
As can be seen from Fig. 5, ideally, the first 30 orders of the brakes are inherently frequency distributed between 237.7Hz and 2524.8Hz. There exists an unstable mode, that is, the 25th order mode, whose frequency value is 1956Hz, and its instability coefficient is 0.007, which is less than 0.01, so it can be regarded as a stable mode, and its vibration diagram is shown in Fig. 6. As can be seen in Fig. 6, each component of the brake is deformed to some degree, with the largest deformation occurring at the point where the friction plate is attached to the bracket. The brake disc has six deformation areas evenly distributed along its circumference with its axis as the center, and the vibration in this order mode is the main inducing factor for brake noise.

![Fig. 6 Mode diagram of unstable mode](image)

4.2. The influence of unbalanced braking factors on brake noise
As can be seen from Fig. 5, the brake's inherent frequency value of each order in the unbalanced contact condition decreases compared with the ideal condition and is distributed between 199.25 Hz and 2202.1 Hz. Therefore, the stability of the brake in the unbalanced contact condition is lower than the ideal condition under the action of low frequency excitation. Fig. 7 shows the unstable modes in the simulation results of Operating condition 2 - Operating condition 4.
As can be seen from Fig. 7, three unstable modes appear in the simulation results of operating condition 2 with the frequency values of 310 Hz, 678 Hz and 1124 Hz, respectively. With the increase of the frequency value, the instability coefficient shows an obvious rising trend, in which the instability coefficient at 1124 Hz reaches 0.088, which is close to 0.1, with a strong tendency of noise. There are four unstable modes in operating condition 3, and their frequency distribution is more dispersed, from 310 Hz at the low order frequency to 1794.3 Hz at the high order frequency, and the instability coefficients also range from 0.01-0.05. In operating condition 4, there are three unstable modes with a concentrated frequency distribution of 678 Hz, 915 Hz and 1311 Hz, and the instability coefficients are 0.038-0.06. In the three operating conditions, the brake has more than two unstable modes at 310 Hz and 678 Hz, so the brake is prone to instability under unbalanced contact conditions at these two orders of frequencies.

By comparing the simulation results under the four operating conditions, it can be seen that in the non-equilibrium contact state, the brake has significantly more unstable modes than the ideal state, and the unstable modes are mainly found in the lower order modes. In combination with Fig. 5, the inherent frequency of the brakes in all orders is lower than the ideal state in the non-equilibrium contact state, that is, the non-equilibrium contact between the friction plate and the brake disc can make the low frequency stability of the brakes significantly reduced, and the trend of noise increases significantly. In condition 2, when the contact time difference between the friction plates on both sides and the brake disc is 0.1s, an unstable mode with an unstable coefficient close to 0.1 appears, with strong noise tendency. The most orders of unstable mode in condition 3, the most dispersed distribution, the most wide range of frequency to produce braking noise. As the contact time difference between friction plate and brake disc expands further, the noise tendency in operating condition 4 is less than that in the first two working conditions. This is because in the non-equilibrium contact state, when the brake disc and the piston side friction plate begin to contact, the shape of the brake is different from that when the two friction plates contact the brake disc at the same time, resulting in the reduction of its natural frequency. In the case of short time difference, the impact of uneven load on the brake disc is greater, so there is a strong tendency of noise. With the increase of the time difference, the speed of the brake disc has decreased to a certain extent under the action of the friction resistance of one side of the friction plate, so the stability of the brake disc also shows a trend of improvement under the additional excitation, but compared with the ideal state, its noise tendency is still significantly improved.

5. Conclusion
(1) The unbalanced contact state of disc brakes will have some effect on its own inherent frequency, resulting in a decrease in the inherent frequency of the system and a decrease in low frequency stability.

(2) Unbalanced contact of disc brakes will cause unstable mode, and unstable mode mainly appears in low frequency areas, that is, the tendency of brakes to produce low frequency noise is greatly increased.
(3) When the non-equilibrium contact time difference is short, there is a strong tendency of low frequency whistling, with the extension of the time difference, noise tendency appears to rise and then decline.

Acknowledgments
This paper is supported by the Key scientific research projects of colleges and universities in Henan Province, Project Title: "Research on NVH Influencing Factors and Optimization of Automotive Disc Brake", Project No.: 21B460023

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