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Brake Noise Analysis of High-Speed EMU Based on Complex Mode Analysis Method

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Abstract

This paper analyses the braking noise by establishing the shaft disc brake finite element model using ANSYS Workbench platform. Through a comprehensive investigation of influencing factors such as the friction coefficient of the brake disc, brake pressure, shape of the brake friction block and their arrangement, it is found that the friction coefficient has a great influence on the brake noise. Besides, the simulation results also uncover that the influence of brake pressure on brake noise depends on whether the brake pressure reaches a specific critical value. Lastly, through extensive simulations, it shows that the braking noise level is distinct for different shapes and arrangements of the brake pads.

Keywords: complex mode method, braking noise, finite element simulation, mode coupling, shaft disc brake, high-speed train

1 Introduction

The high-speed rail provides convenience for people's travel while also bringing certain inconveniences, one of which is the noise pollution caused by braking. Braking noise is generally generated by the friction between the brake disc and the brake pad, and one or more resonance frequencies are generated during the braking process[1]. During braking, the brake disc acts as a loudspeaker. The main frequency of braking noise is usually relatively single, often accompanied by a harmonic component of low amplitude[2][3]. The frequency of braking noise is generally distributed between a few tens of Hz to 20,000 Hz, which is usually divided into low frequency noise and high frequency noise according to the different frequency range. The braking noise studied in this paper is generally high-frequency noise with vibration frequency ranging from 1 kHz to 16 kHz and above[4]. This kind of high-frequency noise is significantly more harmful to human body than low-frequency noise, which is also the focus of most researchers in related fields. This kind of braking noise will not only cause some interference to passengers and residents along the line, but also the self-excited vibration which causes this kind of braking noise is likely to cause fatigue of the braking device, and then endanger the safety of the train and passengers.

The mechanism explanation of braking noise can be roughly divided into two categories: self-excited vibration and tribology theory[2]. For this kind of problem, researchers generally started from the perspective of self-excited vibration. In the early study of braking noise (before the 1970 s), the problem of friction side itself was first considered, which leads to two simpler models, the Stick-Slip motion[5] and the negative slope model of friction vs relative sliding speed[6]. After the 1980s, scholars began to consider that the noise problem may be caused by structural problems, so they put forward the theory of Sprag-Slip[7], which believed that vibration and friction noise are caused by the structural instability of the whole friction system caused by self-locking between friction surfaces[8]. Since the 1990s, numerous theories have emerged to explain the braking noise. For example, scholars have tried to use the Hot Spot theory to explain the braking noise problem[9], but there is no obvious progress at present. Since the beginning of the 21st century, it has been widely believed that the mode coupling mechanism developed based on the two-mode separation theory[10] is the main cause of friction noise[11], that is, friction noise is a self-excited vibration phenomenon caused by mode coupling induced by friction. It is obvious that the above models have their own shortcomings. The Stick-Slip motion and the negative slope model of friction vs relative sliding speed only involve the friction characteristics of the friction side, which cannot be explained well in many cases, and it is difficult to solve the problem of vibration and noise[12]. After considering the merits and demerits of each model and the advanced application of complex modal analysis method based on modal coupling theory in finite element simulation**Hiba! A hivatkozási forrás nem található.**[13], this paper has decided to adopt modal coupling theory as the theoretical foundation.

This paper adopts ANSYS Workbench platform to establish the shaft disc brake finite element model, then uses the complex mode analysis method as the simulation method, and lastly, analyzes the brake noise through the control variable method. Investigation focused on factors such as the friction coefficient of the brake disc, brake pressure, shape of the brake friction block, and their arrangement, culminating in the development of an optimization strategy.

2 Methods

To conduct the stability analysis of braking noise, it is necessary to discretize the model, then define the contact relationship between the model components, and establish the corresponding motion equation[14][15][16]. The dynamics of the braking system can be expressed as:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{0\} \tag{1}$$

where [M] is the mass matrix, [C] is the damping matrix, [K] is the stiffness matrix, and {x} is the displacement vector.

Because of the friction in the braking system, the stiffness matrix [K] can be expressed in the following form:

$$[K] = [K_s] + \mu[K_f] \tag{2}$$

Where $[K_s]$ is the structural stiffness matrix, $[K_f]$ is the asymmetric stiffness matrix caused by the friction force, and μ is the friction coefficient. Note that the stiffness matrix here is an asymmetric matrix, which directly leads to the failure to solve the equation with the conventional method, and also leads to the solved eigenvalues of complex numbers, which will be mentioned below.

Let the solution of the equation be:

$$\{x\} = \{\varphi\}e^{\lambda t} \tag{3}$$

The equation obtained by reverting formula (3) to formula (1) is as follows:

$$(\lambda^2[M] + \lambda[C] + [K])\{\varphi\} = [D(\lambda)]\{\varphi\} = \{0\} \tag{4}$$

Where [D(λ)] is the eigenmatrix of the vibration system, which shows that the equation involves the problem of "quadratic eigenvalues". It is clear that this equation is a 2n algebraic equation of λ with 2n eigenvalues, whose eigevectors are:

$$\lambda_i = \alpha_i \pm i\omega_i \ , \ (i = 1,2,3,\ldots,2n) \tag{5}$$

All of these eigevectors are complex numbers, and their imaginary part ω_i represents the natural frequency of the system, while their real part α_i represents the damping coefficient. αi occurs in positive and negative conjugate pairs, that is, at the same natural frequency, two real parts will occur and the absolute values of the two real parts are equal. The vector solution of the displacement of the system can be expressed as follows:

$$x = Ae^{\lambda_i t} = e^{\alpha_i t}(A_1\cos\omega_i t + A_2\sin\omega_i t) \tag{6}$$

As can be seen from the formula, when the real part value is positive, the system will lose its stability as the amplitude index increases with time, which is easy to produce noise, and the larger the value of the real part, the greater the growth rate of the amplitude. A larger absolute value of the real part doesn’t mean the greater the noise, because the noise generated by the same amplitude at different vibrational frequencies is not comparable. However, a large number of experiments show that the large positive real value mode will produce braking noise in most cases, and the frequency most prone to noise is consistent with the maximum real value mode. Therefore, we can describe the likelihood of noise occurrence by the real value of the mode, that is, the absolute value of the real value of the mode at this natural frequency is positively correlated with the likelihood of noise occurrence.

In addition, only considering the material friction without considering material damping may lead to excessive extracted mode, which is the reason why there are many unstable modes does not occur braking noise. According to the reference, the unstable modes that are predicted to be too conservative can be screened according to the size of the mode damping ratio (the real ratio of the complex eigenvalue to the vibration frequency), so the unstable modes with the mode damping ratio less than -0.001 are extracted here.

In this paper, the static analysis and mode analysis module of ANSYS Workbench platform are used for brake disc noise, and the brake disc of CRH series EMU is selected as the research object[17]. In order to simplify the calculation and improve the calculation speed, this simulation eliminates the brake clamp and other mechanisms, and only builds the geometric model of the brake disc and the brake plate, as shown in Figure 1 below:

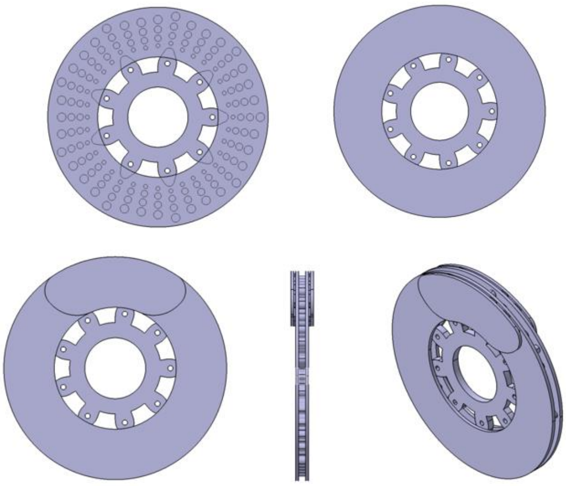


Fig.1 Geometric model of the brake disc and the brake plate

As shown in Figure 2, a total of 5 different shapes of friction blocks are designed, which are rotundity, regular hexagon, sector, triangle (chamfered), and ellipse, and there are 18 symmetrical distribution friction blocks on each brake plate[18].

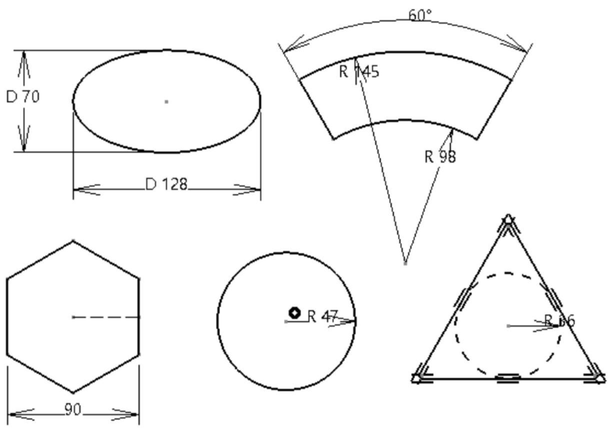


Fig.2 Geometric dimensions of friction blocks on brake pads

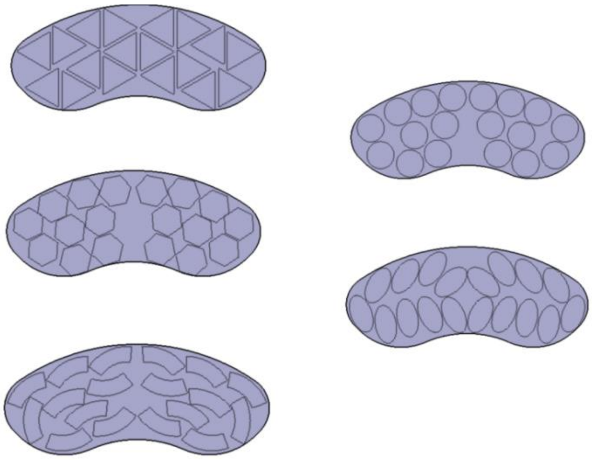


Fig.3 Arrangement of friction blocks on brake pads

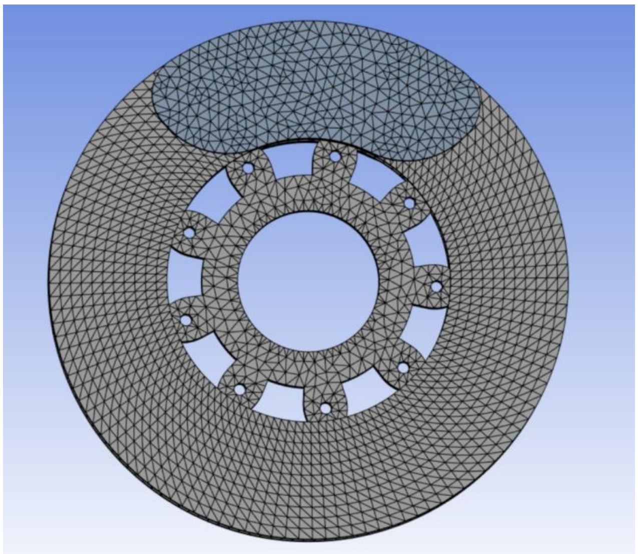


Fig.4 Finite element model of axle disc braking device

In this paper, the commonly used 25Cr2MoV alloy material is used as the material of the brake disc, and the metallurgical material is used as the material of the brake pads[17]. The material properties of both are shown in the following Table 1.

Table 1 Material properties of brake discs and pads

Parameter (Unit)	Brake discs (25Cr2MoV)	Pads (Metallurgical materials)
Density (kg/m^3)	7620	5250
Young's modulus (GPa)	211	200
Poisson's ratio	0.25	0.28

Due to the complete nonlinear method adopted in this paper, after mesh division, a static analysis should be carried out to determine the contact state, and then the perturbation mode analysis should be carried out[20][21]. After extracting all the obtained unstable modes, the positive real part of the complex eigenvalue is taken as the basis for the possibility of braking noiseHiba! A hivatkozási forrás nem található..

3 Results

First, three different friction coefficients (μ =0.25, μ =0.35, μ =0.30, μ = 0.35) were selected to compare the brake noise instability modes under each friction coefficient (hexagonal friction block, brake disc speed, 6 rad/s and 1.5MPa), and the following results were obtained:

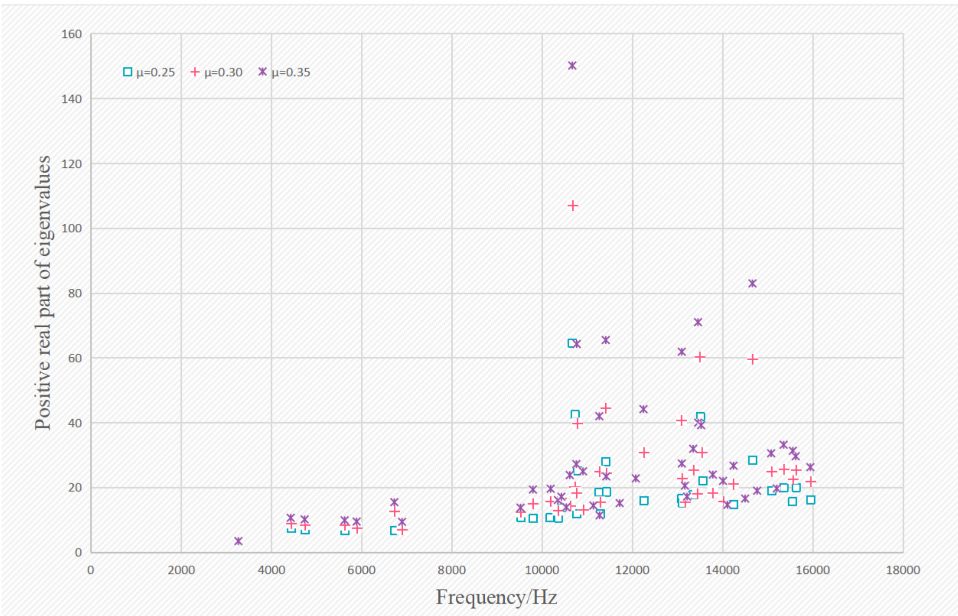


Fig.5 The distribution of the positive real parts of eigenvalues under different friction coefficients

It can be seen from Figure 5 that the friction coefficient has a greater influence on braking noise, and the greater the friction coefficient, the higher the probability of braking noise induction. Especially around 10670 Hz, the positive real part of the complex eigenvalues of the three is quite different (Fig.6). At the same time, it is also easy to find that the points generating unstable modes under different friction coefficients are at almost the same brake noise frequency. For instance, the three points with the highest positive real part of eigenvalues in all three brake noise systems are consistently found at 10670 Hz, irrespective of changes in the friction coefficient.

Therefore, it can be concluded that the friction coefficient does not change the frequency of brake scream.

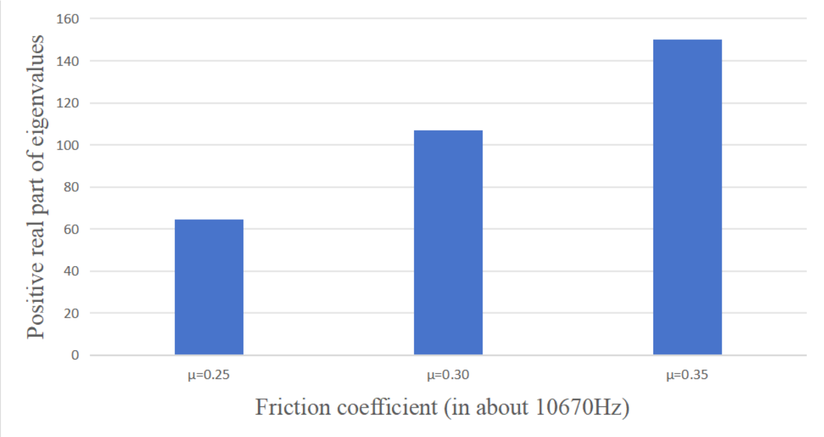


Fig.6 The positive real parts of eigenvalues under different friction coefficients in 10670Hz

Next, four different brake pressures (P=0.5MPa, P=1.0MPa, P=1.5MPa, P=2.0MPa) are selected, and the brake noise instability modes extracted under the same working condition (hexagonal friction block, brake disc speed is 6 rad/s, friction coefficient is μ =0.30) are drawn in the same figure, and the following results are obtained:

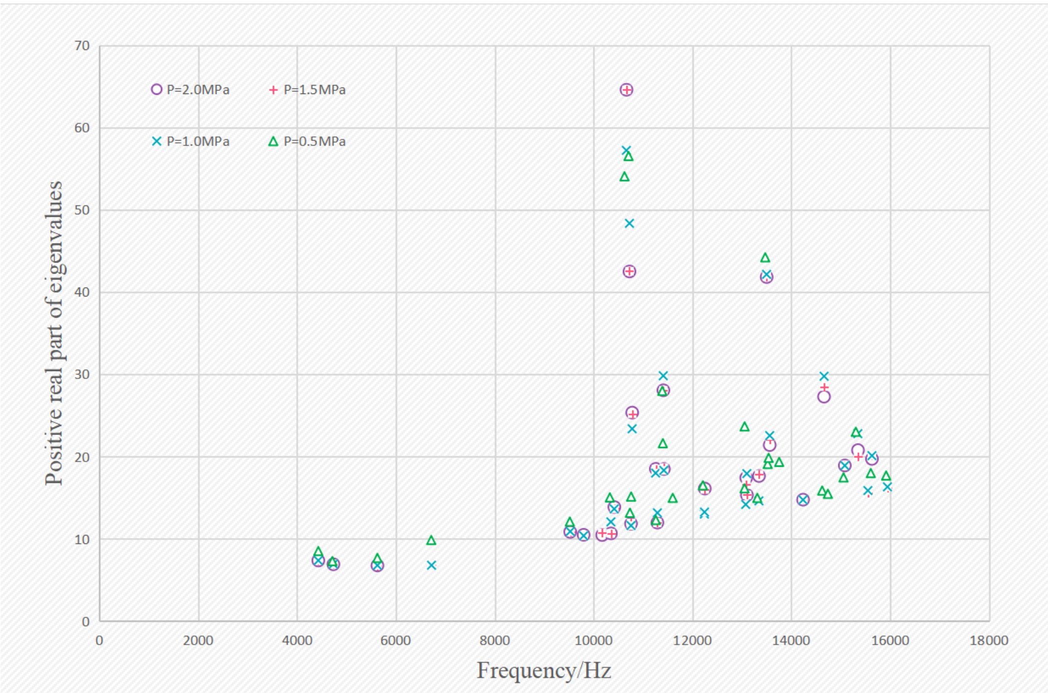


Fig.7 The distribution of the positive real part of eigenvalues under different braking pressures

It can be found that when the braking pressure reaches a certain level, the effect of the braking pressure on the braking noise is almost constant. For example, the braking pressure of the system at 10665 Hz is almost equal to the positive real value generated by the braking pressure of 1.5 MPa. Only when the brake pressure is low, the brake pressure will affect the generation of braking noise, that is, the smaller the brake pressure, the lower the probability of braking noise generation. However, considering that too small braking force may lead to other braking problems, it can be considered that braking pressure has little impact on braking noise.

Then, the brake pieces of five different shapes shown in Fig. 2 and Fig. 3 are selected, and the unstable modes of brake noise extracted under the same working condition (friction coefficient μ =0.25, brake disc speed is 6 rad/s, brake pressure is 1.5MPa) are drawn in the same picture for comparison, and the following results are obtained:

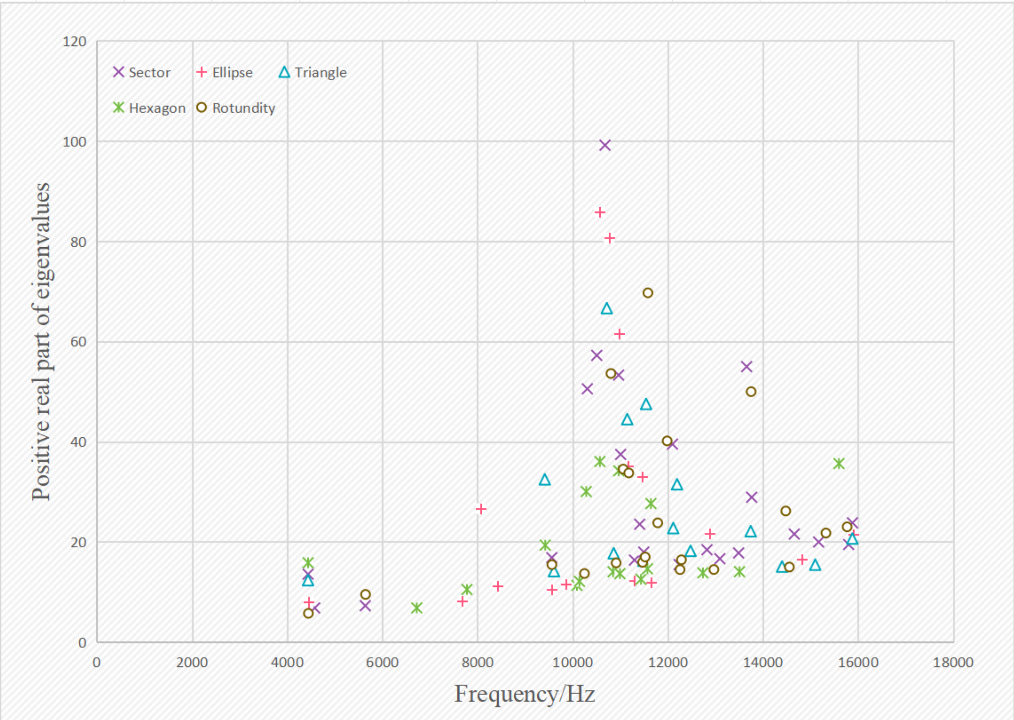
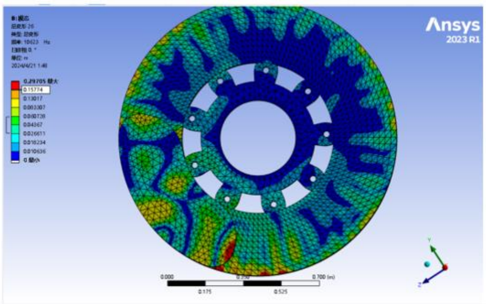


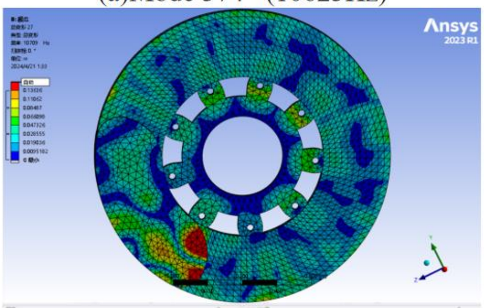
Fig.8 Distribution of Positive Real Parts of Eigenvalues under Different Friction Block Shapes

It can be seen from Fig. 8 that the brake noise is partly related to the friction block shape of the brake pads. In the case of the random arrangement of friction block shapes, the probability of noise in different brake pieces can be obtained according to the area surrounded by the point and the coordinate axis in the figure. For example, in the figure, the probability of brake noise occurring in the brake pads with hexagonal friction block is significantly lower than that in other shape friction blocks. Conversely, brake pads equipped with sector-shaped friction blocks demonstrate a significantly higher likelihood of brake noise occurrence when contrasted with other shapes of friction blocks.

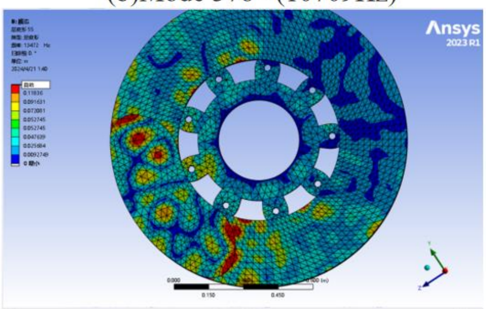
Finally, this article will explain the impact of the arrangement of friction blocks on brake noise. To elucidate how the arrangement of the friction blocks of the brake pads influences brake noise, the following conditions are described: a friction coefficient of μ = 0.25, hexagonal friction block shape, brake disc speed of 6 rad/s, and brake pressure of 1.5 MPa.



(a)Mode 374th (10623Hz)



(b)Mode 378th (10709Hz)



(c)Mode 566th (13472Hz)

Fig.9 Three unstable modal vibration cloud maps

Choosing the unstable modes with the largest real part of three complex eigenvalues and observing the vibration cloud diagram of these three unstable modes, we can find that Mode 374th meets the characteristics of circumferential modes, while Mode 378th and Mode 566th meet the characteristics of different internal and external circles, and these two modes are the most common two vibrating modes in this simulation. The common point of the three unstable modes is the maximum amplitude at the contact between the brake pads and the brake disc leads to poor contact during vibration, which is an important cause of screaming. It can be concluded that the amplitude of the brake pads at the edge can be reduced to a certain extent by arranging the friction blocks at the edge. Hence, in the case of a brake disc featuring elliptical friction blocks, the arrangement of the friction blocks will be adjusted according to the principle of placing them towards the periphery of the disc to minimize edge vibrations. Subsequently, a simulation will be carried out once more to see if this action can reduce the probability of braking noise.

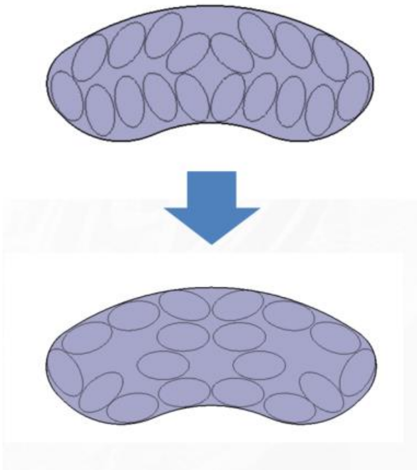


Fig.10 Comparison of the arrangement of friction blocks on brake pads before and after changes

The rearranged brake pads will undergo simulation analysis to extract the unstable mode of brake noise under the same working conditions as the previous experiment, and the results are shown in the figure:

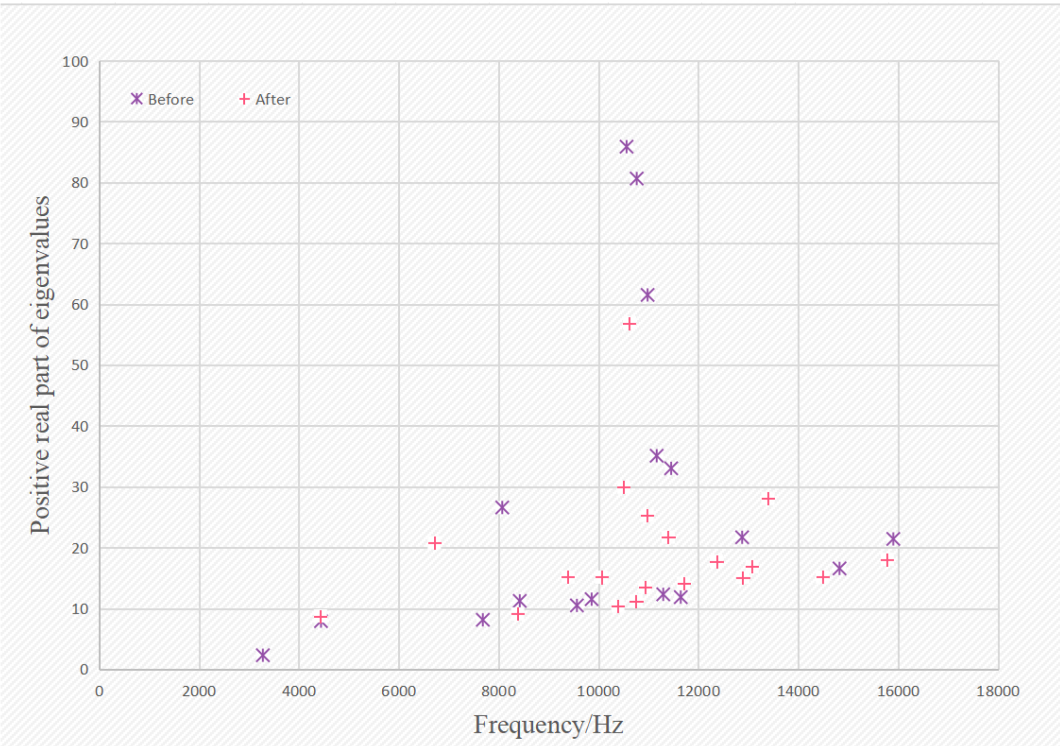


Fig.11 The Distribution of Positive Real Parts of Eigenvalues under Different Layout Patterns

Obviously, the occurrence probability of brake noise in the changed elliptical friction block is greatly reduced, indicating that the above inference of friction block layout change is correct. That is, the probability of friction noise can be reduced by changing the shape of friction blocks and the arrangement of friction blocks.

4 Conclusions and Contributions

(1) The friction coefficient has a great influence on the brake noise. The larger the friction coefficient, the greater the probability of brake noise, while the change of friction coefficient itself does not change the frequency of brake noise.

(2) The influence of brake pressure on brake noise depends on whether the brake pressure reaches a specific critical value. When the brake pressure reaches a certain level, the impact of the brake pressure on the brake noise is almost constant. Only when the brake pressure is small, the brake pressure will affect the generation of the brake noise.

(3) The brake noise has a certain relationship with the shape of the friction block of the brake pads. The brake noise generated by the hexagon is the least among the five different shapes.

(4) Brake noise is also related to the friction block arrangement of the brake pads. After rearrangement according to the marginalization of the brake sheet, the occurrence probability of the brake noise of the elliptical friction block is greatly reduced, that is, the probability of friction noise can be reduced by changing the shape of the friction block and the arrangement of the friction block at the same time.

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