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Impact Analysis and Optimization of Material Parameters of Insulator on Brake Squeal

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ABSTRACT In this paper, factors that account for the disc brake noise were explored with a bench test to fix the methods of reducing the brake squeal noise. The influence of the stiffness and damping of insulator on the noise of the braking system was analyzed by establishing a mathematical model for the 6-DOF braking system with internal and external insulators. It is proposed that the insulator with a multilayered structure of different materials has a better restraining effect on brake squeal. Four different insulator schemes were designed, the test results of which were compared with those of the original scheme. The comparative analysis shows that the stiffness and damping of insulator have an important influence on the braking system noise, which can be reduced by improving the material parameters of the structure layer of the insulator. The test results show that the insulator with one layer of rubber and one layer of steel has a good effect on restraining the low-frequency noise, while the insulator with two layers of rubber and one layer of steel has a good inhibition on the high-frequency noise around 7∼8 kHz. The final conclusion shows that a rubber-steel-rubber structure insulator made of a 0.5-mm-thickness steel sheet can reduce the overall incidence of the brake system noise from 19.27% to 1.27%. An appropriate selection of material structure parameters of the insulator can minimize the occurrence of brake squeal.

INDEX TERMS Disc brake, brake squeal, insulator, multilayered parts, parameter optimization.

I. INTRODUCTION

As a key safety component of vehicle chassis, the brake system has strict requirements for its braking efficiency, such as friction coefficient and the braking efficiency in a high temperature environment. To meet the ever higher demand of passengers for comfort, manufacturers have also paid much attention to the NVH characteristics of brake system. The cost of solving brake NVH problems in North America adds up to nearly $1 billion a year [1]–[3], and friction material suppliers spend more than 50% of their money on brake vibration and noise annually [4], [5]. Therefore, the study of the problem of brake system noise is of great significance.

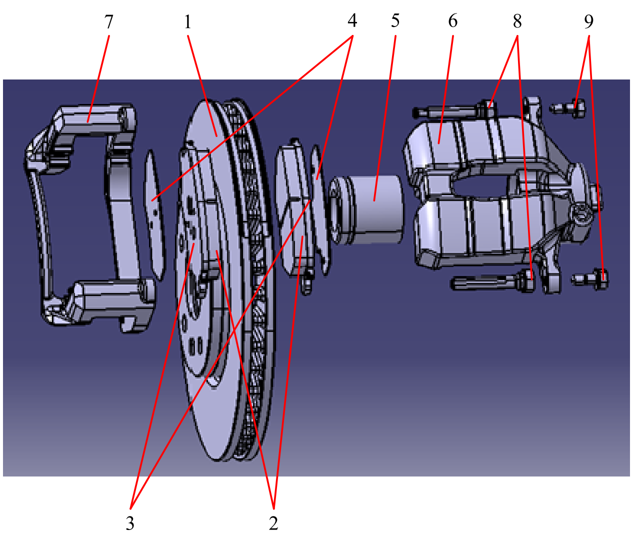
According to its frequency, noise can be divided into low-frequency vibration noise, such as Moan, Groan, Judder and Roughness, and high-frequency vibration noise, such as Squeal and Squeak [6]–[8]. The noise above 1kHz is mainly squeal, which can be further divided according to its frequency into low frequency squeal (1∼3kHz) and high frequency squeal (5∼20kHz) [9], [10].

Kim and Zhou [11] modified the chamfer of brake pad and found that its suppression effect on squeal is remarkable. Sujatha *et al.* [12] modified the geometry of brake disc and made a finite element analysis that revealed that the brake squeal could be reduced by increasing the thickness of brake disc backplate and friction material. Most scholars focus on the shape of friction disc, chamfer, wrap angle size parameters and the optimization of brake disc structure. However, in practical application, the development cost of friction disc and brake disc turns out high. Particularly, since priority should be given to the brake system to ensure its stable safety performance, in the process of friction disc development, it is difficult to achieve an ideal balance between braking ability and noise performance in choosing friction material. Therefore, to suppress the occurrence of noise, it is an ideal choice using damping liner to absorb and suppress the system vibration. Chen [13] of Tongji University studied the bonding position of insulator and the thickness of rubber layer and steel plate on the contact pressure distribution, and, by analyzing various modes, demonstrated the influence of the design parameters of insulator on brake squeal. Festjens [14] *et al.* only theoretically analyzed the possibility of restraining brake squeal by using viscoelastic materials for insulators, but their studies lacked test support.

Therefore, this paper chooses the structural material parameters of insulator to probe into the brake scream problem, trying to suppress the incidence of scream noise by optimizing the material parameters of insulator. First, the important influence of the material parameters of insulator on the system stability is analyzed by the six-DOF kinematics model of brake disc-brake pad-insulator. Then, the corresponding test scheme is designed, whose effectiveness is verified through bench test. Finally, the optimum structure material parameters are obtained from the comparison of test results. It turns out the structure insulator reduces the overall incidence of system noise from 19.30% to 1.27%. This paper contrives a method of great engineering value that can reduce the screaming noise of disc brake system..

II. SIX-DOF KINEMATICS MODEL OF BRAKE DISC-BRAKE PAD-INSULATOR

Festjens *et al.* [14], by setting up a brake coupling model to analyze the structure of brake pad steel back damping layer, concluded that the size of the damping has an important impact on brake noise, so it can be inferred that the damping of anechoic material will also have an important impact on the brake noise.



Note: 1-Disc, 2-Lining material, 3-Brake backplate, 4-Insulator, 5-Piston, 6-Caliper, 7-Bracket, 8-Guide pin, 9-Screw

FIGURE 1. **Schematic diagram of the structure of disc brake.**

The brake selected in this paper is disc brake for a passenger car. The system components mainly include: disc, inner and outer friction linings, inner and outer friction lining backplates, inner and outer insulators, pistons, cliper, brackets, guide pins and fixing screws, etc., as shown in Fig.1. When car brakes, under the action of hydraulic pressure, the piston(5) pushes the side brake pad to press the disc. At this time, under the action of reverse force, the caliper(6) slides along the guide pin(8), and the brake pad fixed is pressed against the disc until the brake pads on both sides are balanced by force.

It can be seen that the disc, brake pads and insulators are the most important parts of the brake system. Based on this conjecture, this paper will establish a six-DOF kinematics model of brake disc-brake pad-insulator to analyze the influence of insulator on the stability of the system.

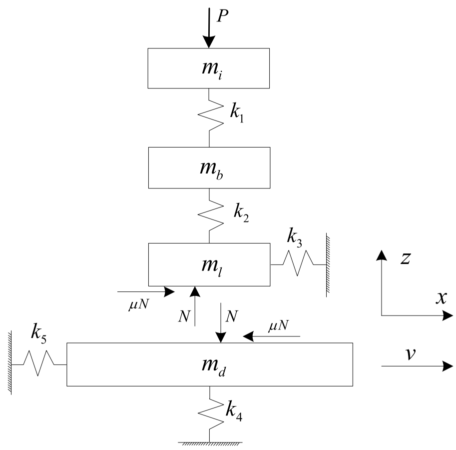


FIGURE 2. **Six-DOF kinematics model of brake disc-brake pad-insulator.**

Since the damping of the system is small (as shown in Fig.2), the influence of damping is ignored in kinematics analysis. Among the parameters, insulator m_i has a z-direction degree of freedom z_1, brake pad m_b has a z-direction degree of freedom z_2, friction material m_b has a degree of freedom z_2, friction material m_l has a z-direction degree of freedom z_3 and an x-direction degree of freedom x_3, and brake disc x_d has two degrees of freedom, respectively x-direction of x_d and z-direction of z_d. Moreover, μ, P and N are namely friction coefficient, force of piston acting on the brake pad, and positive force acting on the brake disc.

The kinematic equation can be obtained from the kinematic model of Fig.2 :

$$[M]\{\ddot{u}\} + [K]\{u\} = \{F\} \tag{1}$$

In this formula, $u = (z_1, z_2, z_3, x_3, z_d, x_d)^T$ and $(\cdot)$ is the differentiation with respect to time. Mass matrix $[M] = diag(m_i, m_b, m_l, m_l, m_d, m_d)$, motivation $\{F\} = (P, 0, N, \mu N, -N, -\mu N)^T$, and $[K]$ is the stiffness matrix:

$$[K] = \begin{bmatrix} k_1 & -k_1 & 0 & 0 & 0 & 0 \\ -k_1 & k_1+k_2 & -k_2 & 0 & 0 & 0 \\ 0 & -k_2 & k_2 & 0 & 0 & 0 \\ 0 & 0 & 0 & k_3 & 0 & 0 \\ 0 & 0 & 0 & 0 & k_4 & 0 \\ 0 & 0 & 0 & 0 & 0 & k_5 \end{bmatrix} \tag{2}$$

At the initial stage, the system is in a steady state, the brake disc rotates at a constant speed, and the system does not produce vibration, so the steady-state equation can be obtained. Assuming that the brake disc and brake pad are not separated during the process of vibration, the constraint condition is $z_d = z_3$, and then we know

$$N = m_l\ddot{z}_3 - (k_1 + k_2)z_1 + k_2z_2 \tag{3}$$

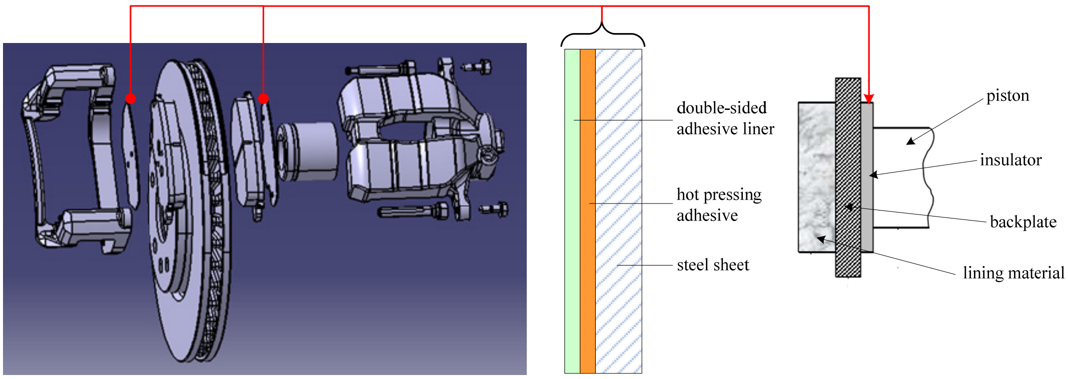


FIGURE 3. Structure and paste position of insulator.

Taking (3) into the steady-state equation, we obtain

$$\begin{bmatrix} m_i & 0 & 0 & 0 & 0 \\ 0 & m_b & 0 & 0 & 0 \\ 0 & 0 & m_l+m_d & 0 & 0 \\ 0 & 0 & -\mu m_l & m_l & 0 \\ 0 & 0 & \mu m_l & 0 & m_d \end{bmatrix} \begin{Bmatrix} \ddot{z}_1 \\ \ddot{z}_2 \\ \ddot{z}_3 \\ \ddot{x}_3 \\ \ddot{x}_d \end{Bmatrix} + \begin{bmatrix} k_1 & -k_1 & 0 & 0 & 0 \\ -k_1 & k_1+k_2 & -k_2 & 0 & 0 \\ 0 & -k_2 & k_2+k_4 & 0 & 0 \\ \mu(k_1+k_2) & -\mu k_2 & 0 & k_3 & 0 \\ -\mu(k_1+k_2) & \mu k_2 & 0 & 0 & k_5 \end{bmatrix} \begin{Bmatrix} z_1 \\ z_2 \\ z_3 \\ x_3 \\ x_d \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{Bmatrix} \tag{4}$$

Equation (4) is the kinematic equation of the brake disc-brake pad-insulator system. The equation shows that the mass matrix and the stiffness matrix of the kinematic equation of system are both asymmetric due to the existence of friction, so the eigenvalues of the system may be complex numbers. The imaginary part of the complex eigenvalue represents the modal frequency, but the real part represents the instability tendency. The larger the real part is, the more likely the system will be unstable. Therefore, it is of great significance to improve the quality matrix and stiffness matrix of the system. It can be seen that the friction coefficient, the mass of friction material ml and the stiffness of insulator and the brake backs k_1, k_2 determine that the mass matrix and the stiffness matrix are asymmetric matrices. Since the actual friction coefficient change is irregular, there's no way to change or control artificially. The change of friction material quality will affect the life of brake pad, and the change of backplate stiffness will affect the safety of brake system. The screaming noise characteristic of brake system can be explored by changing the stiffness of insulator without affecting other performances of brake system. Therefore, this paper focuses on the structure of and the materials for insulators.

III. MATERIAL LAYER STRUCTURE DESIGN OF INSULATOR

A. STRUCTURE AND WORKING PRINCIPLE OF INSULATOR

The brake used in this study is a ventilating disc brake for the front axle of a passenger car. The system components used in bench test mainly include: brake disc, caliper body, bracket, piston, guide pin, internal and external friction linings, internal and external brake backplates, internal and external insulators and fixed screws, etc. The insulator is a thin sheet attached to the surface of an internal or external brake backplate, which contacts the caliper piston and the caliper finger for damping vibration. The installation position and structural characteristics of insulator can be seen clearly from the assembly explosion diagram of the disc brake, as shown in Fig.3. The total thickness of the insulator is generally between 0.8mm and 1.2mm. After tearing off the double-sided adhesive lining paper, the insulator is pasted on the back of the friction plate steel after holding the pressure at 50bar for 10 seconds at 50-70 °C.

B. MATERIAL PARAMETER DESIGN OF INSULATOR'S STRUCTURE LAYER

The principal difference between different types of insulators lies with changing the thickness, quantity and layout of steel sheet and rubber layer. This paper mainly studies the influence of the thickness and quantity of rubber layer on the brake squeal noise. Therefore, the insulator is chosen made of one steel sheet with different thicknesses of rubber layer and different structures of steel sheet. The material structure of the selected insulators is shown in Table 1.

The relationship in frequency, temperature and damping coefficient between four insulators is shown in Fig.4 (a), (b), (c), and (d), respectively. Damping characteristic is a key factor in the reduction of insulator noises. The frequency range of each type of insulator can be seen in Fig.4. Type A insulator is mainly used to solve the noise at temperatures between 80∼130 °C and at a frequency of 1.5kHz. Type B insulator is mainly used to solve the noise at low temperatures between 10∼50 °C and at a frequency of 1.5kHz. Type C insulator is mainly used to solve the noise at low temperatures between 10∼50 °C with a wide frequency range or the noise above the temperature of 110 °Cat a high frequency about 11.5 kHz. Type D insulator, the most widely used type, is mainly for a low frequency within 4 kHz at temperatures between 10∼50 °C, or a medium or high frequency between 8∼10 kHz at a high temperature above 110 °C. The actual

TABLE 1. Material parameters of insulator.

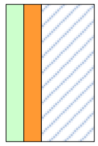
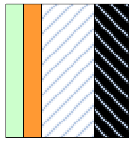
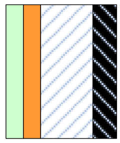
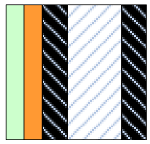
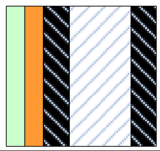
Design of insulator	Material layout sequence and thickness	Schematic diagram of structure
Single layer	Hot pressing adhesive 0.12mm Steel sheet 0.4mm	
A-two layers	Hot pressing adhesive 0.12mm Steel sheet 0.4mm Rubber 0.22mm	
B-two layers	Hot pressing adhesive 0.12mm Steel sheet 0.4mm Rubber 0.14mm	
C-three layers	Hot pressing adhesive 0.12mm Rubber 0.14mm Steel sheet 0.4mm Rubber 0.14mm	
D -three layers	Hot pressing adhesive 0.12mm Rubber 0.14mm Steel sheet 0.5mm Rubber 0.14mm	

TABLE 2. Brake benchtest parameters.

Brake disc size	321×28mm	Rolling radius	337mm
Inertia	79.96kg·m^2	Effective radius	133mm
Piston surface area	32.17cm^2	Caliper efficiency	95%

effect of noise reduction will be verified by the following bench test.

IV. NOISE BENCH TEST VERIFICATION

The experiment was carried out on the Model 3900 inertial test bench of LINK Company. The bench can reproduce the screaming noise of actual working conditions. The equipment separates the test parts from the drive and support of the dynamometer, which can isolate external noise from vibration. The NVH test adopted the Disc and Drum Brake Dynamometer Squeal Noise Matrix published by the Society of Automotive Engineers (SAE J2521 [15]). The frequency range of noise falls between 0.9~17kHz. The microphone fixed position and the real object are shown in Fig.5 and 6, respectively. The brake system adopts the front disc brake of a passenger car, and the vehicle parameters are shown in Table 2.

By installing the corresponding physical friction piece in the test system, the corresponding SAE J2521 test was completed.

A. BENCH TEST OF SINGLE-LAYER INSULATOR

The noise test results of a friction plate with a rigid insulator without slotting or chamfering (hereinafter referred to as bare plate) are shown in Fig.7. It can be seen from the figure that the brake noises are mainly concentrated at about 2kHz, 3kHz, 4.5kHz, 6kHz, 7kHz, 8.5kHz, 12kHz and 15kH. It contains a wider noise frequency range, so that the test result can be used as a matching benchmark. A total of 458 braking noises were collected, with the noise incidence of 19.27%. The most dense noises were below 4 kHz and around 7kHz, with 165 and 162 times, respectively. The noise of the highest sound pressure level occurs at 6981Hz, reaching 104dB.

B. BENCH TEST OF A-DOUBLE LAYER INSULATOR

For follow-up tests, all friction discs are processed with inclined chamfer and middle groove. The A-double layer insulator is used, whose rubber thickness is the largest. The test results are shown in Fig.8. The incidence of noise is 16.67%. The relatively scattered frequency noise within 6kHz was concentrated between 1.6-1.8kHz, but the braking frequency of noise increased. The incidence of cold relative noise was as high as 21.3%, and 114.3 dB noise was collected at 1669Hz.

C. BENCH TEST OF B-DOUBLE LAYER INSULATOR

The B-double layer insulator is adopted here, with the steel thickness unchanged, while the rubber layer thickness is reduced to 0.14mm. The test results are shown in Fig.9. The noise incidence was 6.89%. The screaming noise at 1.8kHz

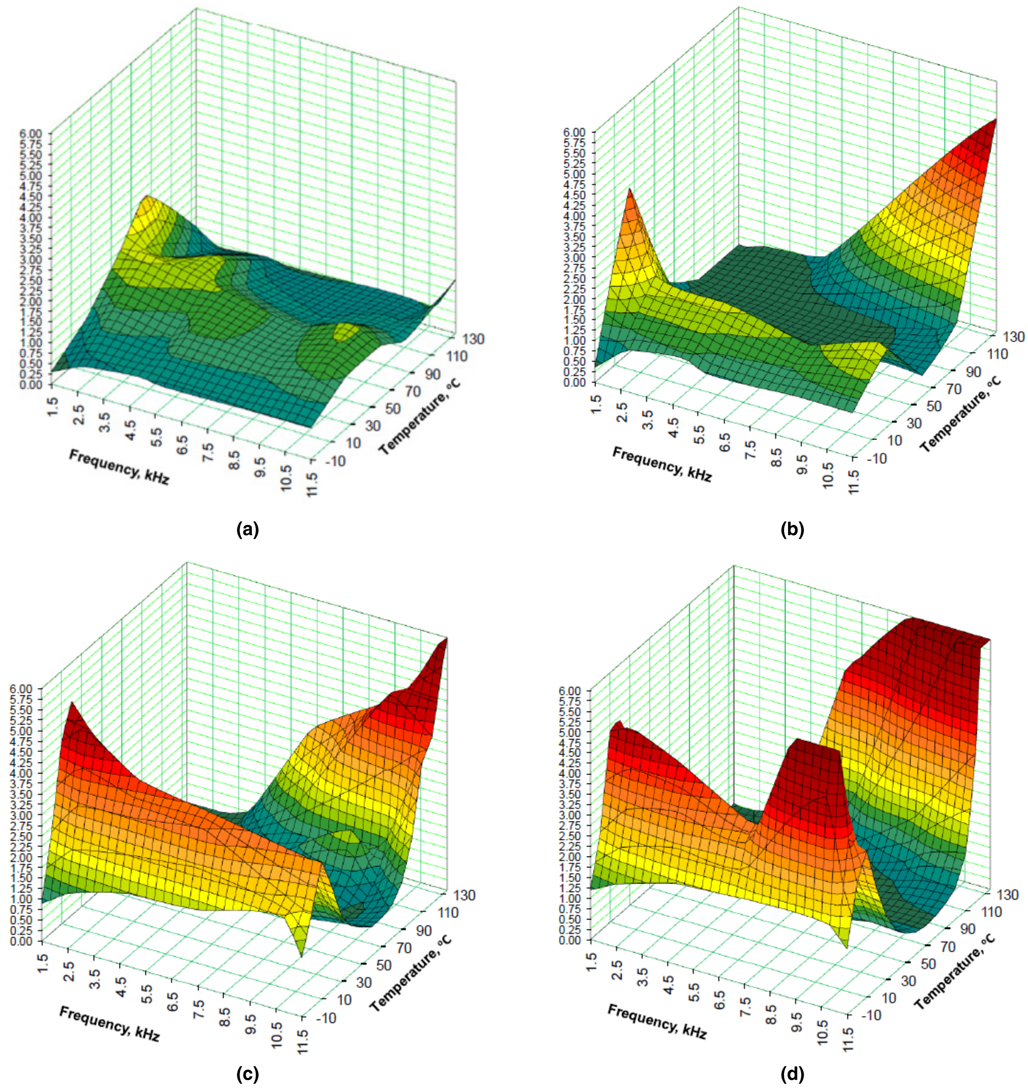


FIGURE 4. **Damping characteristics of four insulators. (a) A-double layer insulator. (b) B-double layers insulator. (c) C -three layers insulator. (d) D -three layers insulator.**

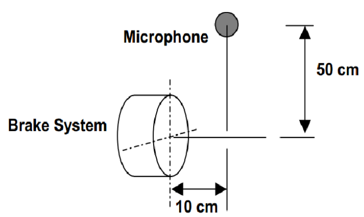


FIGURE 5. **Microphone fixed position.**

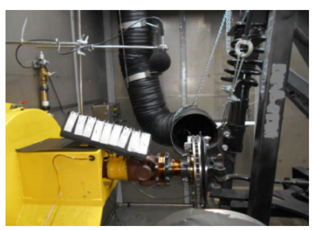


FIGURE 6. **Bench fixed position of microphone.**

was significantly reduced and the sound pressure level was also reduced when using B-type insulator with a thinner rubber layer. The high frequency cold noises at 11kHz and 15kHz were also mitigated compared with those of the previous scheme. Although the problem of low-frequency scream concentration can be improved by using a thinner rubber layer structure, the noise at 7kHz still exists, which is basically a thermal noise whose relative proportion is 3.43%, with the highest sound pressure level at 98.1dB. The incidences of cold noise at 1.8kHz and 11kHz are still relatively high.

D. BENCH TEST OF C-THREE LAYER INSULATOR

The C-three layer insulator retains the thickness, to which one layer of rubber is added. The bench test results are shown in Fig.10. The noise incidence was 2.52%, which was further reduced compared with that of B-double layer insulator.

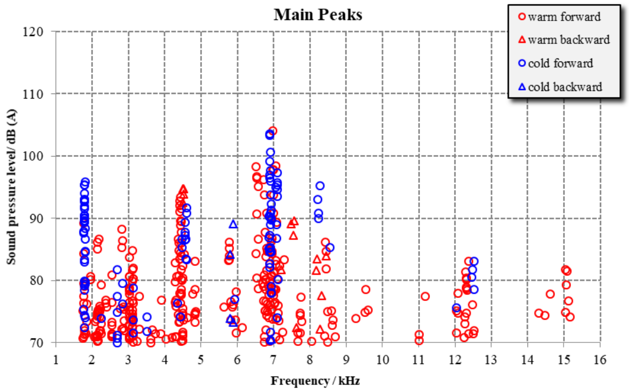


FIGURE 7. Bench test results of single layer insulator.

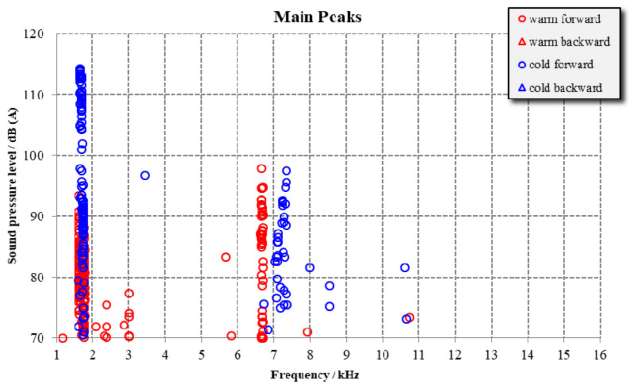


FIGURE 8. Bench test results of A-double layer insulator.

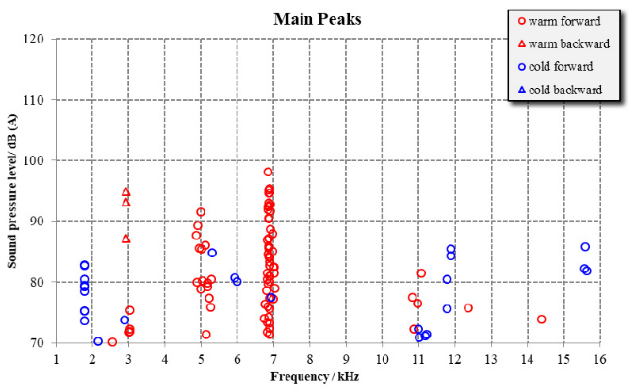


FIGURE 9. Bench test results of B-double layer insulator.

The number of noises at 7kHz was partially reduced, and the low-frequency noise below 2 kHz and the high-frequency noise above 7kHz all disappeared. The number of cold noises was also greatly reduced. The C-three layer insulator verifies that the lower thickness of the rubber layer can well suppress the low frequency squeal below 2kHz, and the rubber-steel-rubber structure can greatly suppress the stubborn noise of 7kHz, but the disadvantage is that it can easily produce noise near 3 kHz again.

E. BENCH TEST OF D-THREE LAYER INSULATOR

Compared with that of C-three layer insulator, the composition of the material and rubber thickness of D-type

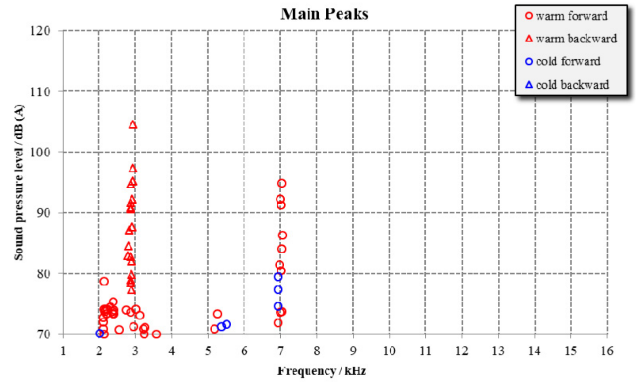


FIGURE 10. Bench test results of C-three layer insulator.

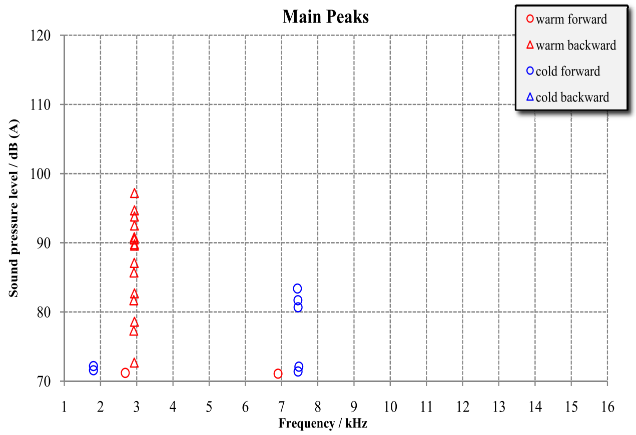


FIGURE 11. Bench test results of D-three layer insulator.

remained unchanged, and the thickness of steel sheet was increased from 0.4mm to 0.5mm. The test results are shown in Fig.11. Compared with that of C-type insulator, the noise within 2kHz and around 7kHz basically disappeared when the thickness of the thin steel sheet was increased. Though there is still a 0.85% thermal noise at 3kHz, the result is acceptable because the condition occurs only when the initial brake disc temperature is above 150 °C. The increased thickness of steel sheet, combined with the adopted structure of rubber-steel sheet-rubber, can effectively restrain the recurrence of noise near 3 kHz and further restrain the noise near 7 kHz.

V. RESULTS ANALYSIS AND DISCUSSION

When the insulator adopts a brake backplate only pasted with a single-layer steel plate, the stiffness coefficient is so large that the vibration ability cannot be attenuated, which will easily affect the stability of the system. The use of a double layer insulator (steel sheet + rubber) may greatly improve the system noise, almost eliminating the noise of a specific frequency and greatly reducing the incidence of noise. Here with the use of a rubber layer for the system to increase a certain damping, the system vibration energy has been effectively attenuated. Meanwhile, it can also be found that the noise pressure level and noise incidence near 2kHz increase a lot due to the resonance phenomenon between brake pad and insulator. For this reason, we optimized the thickness of rubber layer. The test results show that the noise near 2kHz is effectively suppressed and the resonance phenomenon is eliminated, but there are still many noise points near 5 kHz and 7 kHz. In order to further reduce the incidence of noise, a three-layer structure of insulator is adopted, where a layer of rubber is added to form a rubber-steel-rubber structure. The test results show that the noise near 5kHz is suppressed and the noise point near 7kHz is greatly improved, but the incidence of noise near 3kHz obviously increases. This paper tries to improve the noises near 3kHz and 7kHz by improving the thickness of steel sheet. The experimental results show that the improved three-layer structure insulator can effectively suppress the noise near 7kHz, and greatly improve the noise near 3kHz. Finally, the noise incidence of the brake system is reduced from 19.27% to 1.27%. This method provides great guidance for the brake noise reduction engineering.

VI. CONCLUSION

(1) When using a free damping structure insulator with one layer of rubber and one layer of steel sheet, the choice of a thinner rubber can well suppress the low-frequency screaming at 1.8 kHz. The B-double layer insulator can more obviously solve the noise within 2 kHz. However, the noise reduction effect of the rubber-steel sheet insulator is smaller than that of the single rubber layer.

(2) The overall noise reduction effect of the free damping structure insulator with two layers of rubber and one layer of steel is better than that of the free damping structure insulator with one layer of rubber and one layer of steel. Due to its tangential and radial in-plane modes, the brake disc is apt to produce screams at middle and high frequencies. Particularly, 7-8 kHz is the first tangential in-plane mode of the general brake disc. The addition of a layer of inner rubber can, to a great extent, absorb the tangential stress and restrain the vibration caused by the in-plane mode of the brake disc.

(3) The cold and warm state noises near 3 kHz can be effectively suppressed by using a free damping structure insulator with two layers of rubber and one layer of steel sheet. The stubborn noise at 7kHz can be eliminated by using steel sheet with the thickness of 0.5mm instead of 0.4mm. Although there is still some thermal noise at 3 kHz in the final scheme, the overall noise incidence is reduced to 1.27%, which achieves the expected results.

In summary, the study shows that the selection of the material parameters of insulator has an important impact on the brake squeal noise, and, by changing the material parameters of insulator, the purpose of reducing the incidence of the brake system squeal noise is finally achieved.

However, this paper contains more experiments and fewer theories, the mechanism of noise reduction will be further studied in next research.

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