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RESEARCH ON OPTIMAL DESIGN OF VIBRATION REDUCTION OF CENTRIFUGAL AIR CONDITIONING CHILLER BASED ON PARTICLE DAMPING

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Considering vibration of centrifugal air-conditioning chillers at working conditions of 300 Hz and 386 Hz, a vibration reduction method for centrifugal air-conditioning chillers based on particle damping is proposed. Firstly, the vibration transfer path of the chillers is determined based on dynamic characteristics analysis of the chillers. Secondly, the vibration sensitive area of the compressor is determined by finite element analysis. Then the energy dissipation is calculated by the discrete element method (DEM) to determine the optimal installation scheme of the damper. Finally, the vibration reduction effect of the chillers after arranging the damper is verified by experiments.

Keywords: centrifugal air-conditioning chiller, particle damping, DEM, optimal scheme, vibration reduction

1. Introduction

As a new type of high-efficiency air-conditioning technology, centrifugal air-conditioning chillers have attracted a lot of attention in recent years due to their high energy-saving efficiency and excellent load performance (Hassan *et al.*, 2020; Deng and Yang, 2020). As the core equipment of air-conditioning, chillers are widely used in all walks of life. During operation of centrifugal air-conditioning chillers, vibrations will directly affect their safety. After the vibration intensity of the chillers exceeds a certain value, it will affect the precision of related equipment, extremely easy to cause deterioration of structural stress and thus affecting the service life of the chillers. Moreover, because structural vibration of the chillers is transmitted to the installation foundation, it will cause vibration damage to the installation and reduce comfort of the working environment (Geng *et al.*, 2021; Jiang, 2015). Therefore, it is particularly necessary to conduct vibration reduction research on centrifugal air-conditioning chillers.

At present, the commonly used method to reduce vibration for chillers is to arrange rubber shock isolation pads, spring vibration isolators and additional vibration damping supports between the chillers support and the foundation, or to change the resonance frequency of the chillers through structural design (Mo and Ding, 2019). Wang (2004) distributed rubber damping pads and low-frequency large-load damping spring shock absorbers evenly between the chillers support and the foundation, and used multi-layer damping pads in series to improve the damping effect. The vibration reduction measures adopted for the chillers are to install vibration damping rubber pads at the foundation, and to use expansion joints as a flexible connection between the compressor and the pipeline to absorb the vibration displacement of the chillers (Xing, 2016).

According to the test results of the electric motor balance, Fang *et al.* (2019) modified the motor by adding counterweight to adjust the electric motor balance, which solved the problem of excessive vibration of the chillers. However, whether it is to improve dynamic balance of the compressor or to install rubber and spring shock absorbers on the chillers, there are defects. Dynamic balancing will change the original structure of chillers, making it difficult to ensure balance accuracy, making the assembly process more complicated and bringing difficulties to the design process of the chillers. In the installation of rubber and spring shock absorbers, due to defects of rubber and springs such as oil resistance and deformation, the chillers require frequent replacement of the installed materials during use. Over time, vibration isolation performance will be significantly reduced.

Particle damping is a type of passive control technology (Lu *et al.*, 2018). Based on theory of energy dissipation mechanism, the technology fills particles inside a structure or a specific cavity container. It consumes vibration energy of the system through the damping effect formed by friction and collision between particles and the container wall as well as between particles, so as to achieve the purpose of suppressing vibration of the system. (Xiao and Xu, 2021; Nallusamy *et al.*, 2020; Zhang *et al.*, 2020). Particle dampers are arranged flexibly and insensitive to temperature and environmental changes (Lei *et al.*, 2018). They are suitable for use in harsh environments with high reliability (Wang and Li, 2011). At the same time, particle dampers introduce minor changes to the original structure and are suitable for use in narrow spaces such as chillers (Severson *et al.*, 2008; Cheng and Yang, 2014; Romdhane *et al.*, 2013). Due to the above advantages, particle dampers are widely used in aerospace, mechanical vibration control, civil engineering and other fields (Lu *et al.*, 2020).

In this paper, considering advantages and disadvantages of the traditional air conditioning damping method and design requirements of the centrifugal chiller, particle dampers is introduced into the centrifugal air-conditioning chillers. Dynamic characteristics of the chiller are analyzed and the vibration transfer path is obtained. Through modal analysis of the compressor, the vibration sensitive region of the compressor is found, and the installation position of dampers is preliminarily determined and designed. The effects of different particle sizes and excitation on energy dissipation of particles of the chiller and the optimal installation scheme of dampers on the chiller were studied by DEM. Finally, experimental results show that particle dampers can effectively reduce vibration of the chiller.

2. Dynamic characteristics of a centrifugal chiller

A centrifugal air-conditioning chiller is excited by special equipment to cause vibration during the actual working process. The compressor is the main vibration source in the chiller. In order to better design a particle damper, it is necessary to analyze vibration characteristics of the compressor to find the root cause of the vibration and optimize and improve the structure according to the principle of particle damping.

2.1. Analysis of vibration characteristics of the centrifugal chiller

A centrifugal chiller is mainly composed of a compressor, pipeline, condenser, evaporator, support frame and raft frame. Regular frequency multiplication in the dynamic response curve of the compressor indicates that the main source of vibration is the fixed frequency excitation of the compressor rotor. When the compressor is running at a frequency of 386 Hz, acceleration sensors are arranged at a total of 6 measuring points from the compressor to the raft frame. According to the vibration test standard, the sensors are all arranged vertically. The positions of measuring points 1-6 are shown in Fig. 1. The root mean square of vibration acceleration is measured, and the vibration transmission path is analyzed and identified.

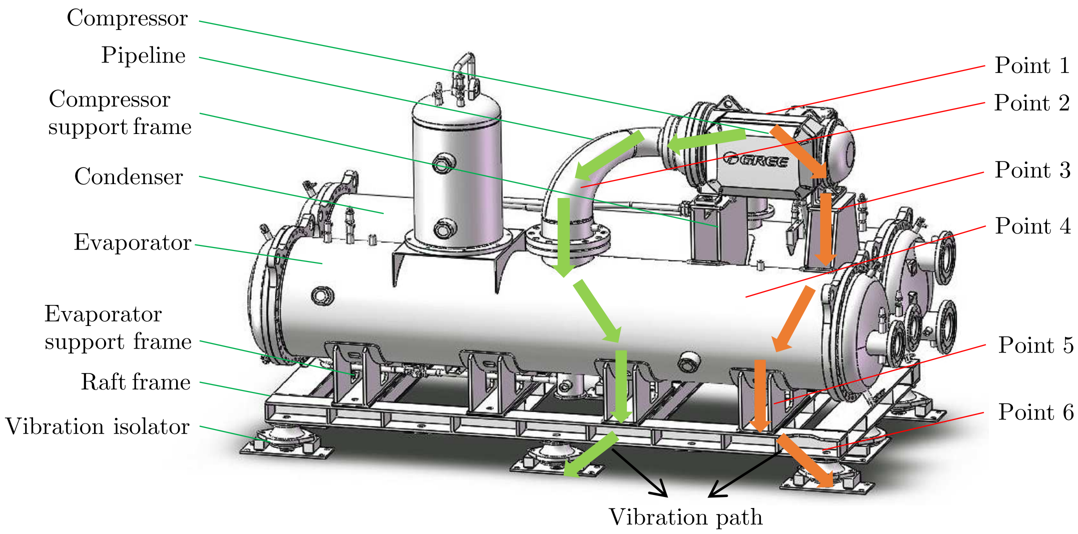

Fig. 1. Composition and location of the measurement point of the chiller

There are two main paths for vibration transmission in the chiller. One is compressor → compressor support frame → condenser → condenser support frame → raft frame → vibration isolator → platform. The second is compressor → pipeline → condenser → condenser support frame → raft frame → vibration isolator → platform. According to the actual test, the dynamic response of the whole chiller decreases gradually from top to bottom. The response of the compressor top and pipeline is the largest, followed by the response of the compressor foot, and the bottom of the raft has the smallest response.

2.2. Finite element analysis of the chiller

Modal parameters are the basis for the optimal design of structural dynamic characteristics of the chiller. Through modal analysis, the vibration mode of the chiller is obtained when it is subjected to external excitation, especially the vibration mode with greater vibration, which lays a foundation for design of particle dampers and optimization of the chiller in the next step.

As a continuous structure, the chiller can be discretized into a system with n degrees of freedom during finite element analysis. Since finite element analysis is only applicable to linear steady systems, assuming that the chiller has linear dynamic characteristics at a certain stage and does not change with time, a multi-degree of freedom dynamic differential equation belonging to the linear system is established considering damping characteristics of the structure, as follows

$$\mathbf{M}\ddot{\mathbf{u}}(t) + \mathbf{C}\dot{\mathbf{u}}(t) + \mathbf{K}\mathbf{u}(t) = \mathbf{F}(t) \tag{2.1}$$

Among them, $\mathbf{M}$, $\mathbf{C}$, $\mathbf{K}$ are the mass, damping and stiffness matrices of the chiller system. $\ddot{\mathbf{u}}(t)$, $\dot{\mathbf{u}}(t)$ and $\mathbf{u}(t)$ are the acceleration, speed and displacement response vectors of the system. $\mathbf{F}(t)$ are the external stable excitations that the chiller bears.

The characteristic equation of the system can be obtained as

$$(\mathbf{K} - \omega^2\mathbf{M})\boldsymbol{\varphi} = \mathbf{0} \tag{2.2}$$

where ω^2 is the eigenvalue corresponding to the characteristic equation, and $\boldsymbol{\varphi}$ is the eigenvector. In order to make the equation easy to solve, Eq. (2.2) needs to be decoupled. Introduce the coordinate transformation

$$\mathbf{u}(t) = \boldsymbol{\varphi}\mathbf{q}(\omega)\mathrm{e}^{\mathrm{j}\omega t} \tag{2.3}$$

Decouple Equation (2.3)

$$-\omega^2 \mathbf{M}\boldsymbol{\varphi}\mathbf{q}(\omega) + \mathbf{K}\boldsymbol{\varphi}\mathbf{q}(\omega) = \mathbf{f}(\omega) \tag{2.4}$$

Solving the above formula, the i-th natural frequency ω_i and the i-th displacement u_i, that is, the modal vibration shapes of the chiller system can be obtained.

2.3. Modal analysis of the chiller compressor

There are many parts in the actual compressor model. In order to improve calculation efficiency, the compressor model is simplified under the premise of satisfying the real performance. The material of the compressor is steel. Its material parameters are elastic modulus of 200 GPa, Poisson's ratio of 0.3, and density of 7850 kg/m^3. The boundary condition is that the lower end of the support frame connected to the compressor is set to a fixed constraint, and the compressor foot and the upper end of the support frame are constrained by bolts. Too large mesh size can lead to insufficient accuracy, and mesh size that is too small consumes computer resources. To ensure precision and accuracy of meshing and to avoid excessive calculations, the mesh element size is set to 5 mm to divide the model into tetrahedral elements. Extracting the first 6 modes of the compressor, the frequency distribution is shown in Table 1, and the mode shape cloud diagram is shown in Fig. 2.

Table 1. Compressor mode

Mode order	1	2	3	4	5	6
Frequency [Hz]	51.35	173.87	334.36	479.18	509.54	525.34

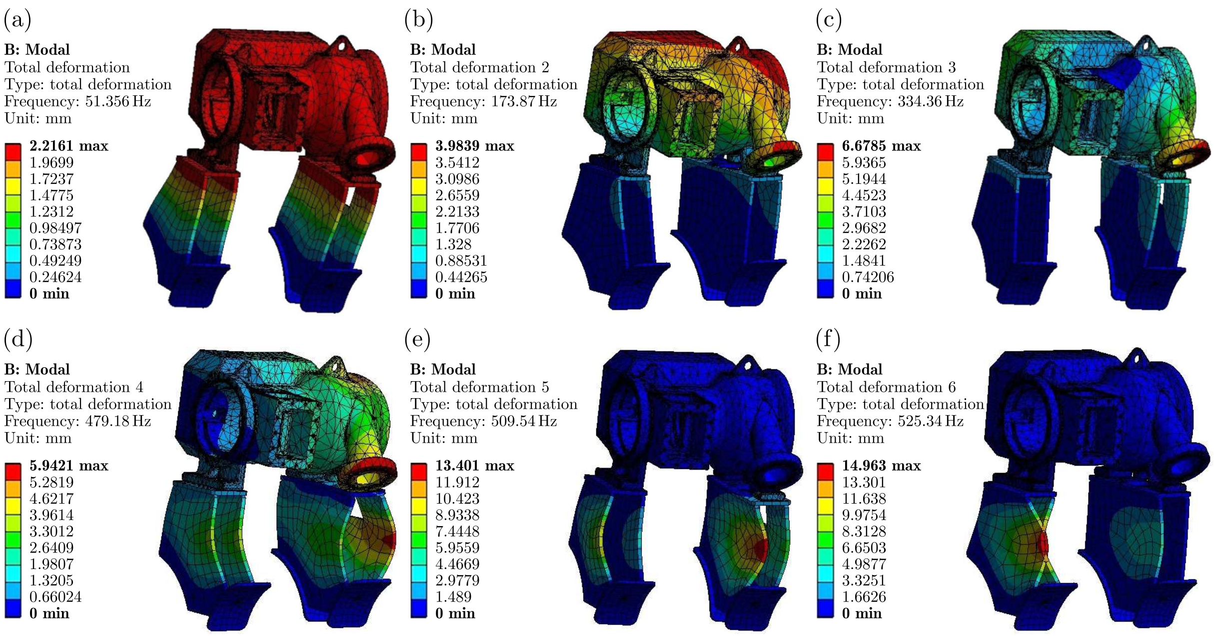

Fig. 2. Compressor modal vibration shape diagram: (a) first-order mode, (b) second-order mode, (c) third-order mode, (d) fourth-order mode, (e) fifth-order mode, (f) sixth-order mode

According to Table 1 and Fig. 2, it can be seen that the first and second modes are mainly bending and deformation of the upper part of the support frame and the compressor. The third and fourth modes are bending and deformation of the compressor foot and the support frame. The fifth and sixth modes are bending and deformation of the support frame.

The modal analysis shows that the top, middle and the support frame of the compressor are vibration sensitive areas, and the installation area of the particle damper is preliminarily determined. For the compressor, the particle damper is mounted on the top of the compressor and installed in parallel at the position of the supporting plate and rigidly installed in series between the compressor and the supporting plate.

3. Discrete element model of the particle damper

3.1. Design of the particle damper for the compressor

In order to better suppress generation and propagation of vibration from the source, considering actual characteristics of a relatively uneven structure of the compressor, a particle damper was rigidly installed between the compressor and the support frame in this paper. The installation scheme of the particle dampers is shown in Fig. 3. J1 and J2 are two types of particle dampers glued to the compressor top. J3 and J4 are two types of particle dampers bolted to the compressor foot. J5 is a type of particle damper bolted to the compressor support frame. The damper housing is made of steel.

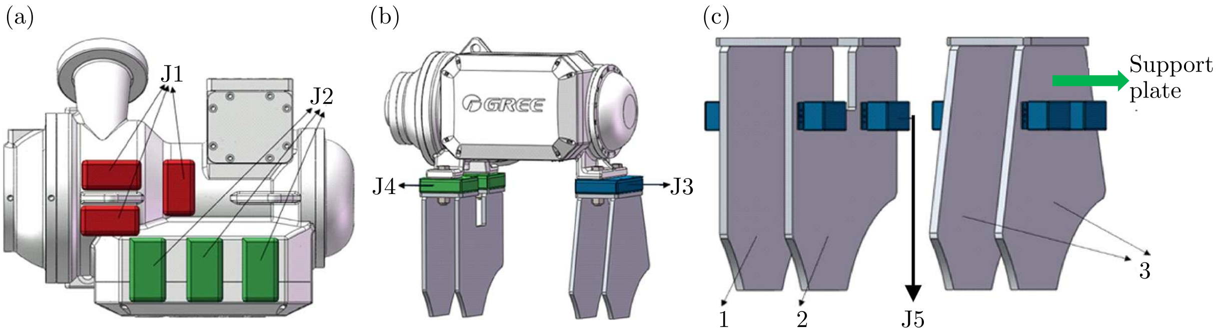

Fig. 3. The installation scheme of particle dampers: (a) on top, (b) on foot, (c) on support frame

The size of J1 is 120 mm×60 mm×50 mm and the size of J2 is 110 mm×60 mm×50 mm. The sizes of J3, J4 and J5 are 250 mm×150 mm×50 mm,165 mm×150 mm×50 mm and 127 mm×50 mm×40 mm, respectively. Two types of particle dampers J1 and J2 are glued on top of the compressor. The damperd raised the compressor vertically by 50 mm. Several through-holes with a diameter of 6 mm were opened on the support frame, and particle damper J5 was bolted to the support frame.

3.2. Mechanical model of the compressor particle dampers

When the compressor runs at the excitation frequency, it sets the particle dampers in motion, causing collision and friction between particles in the dampers and between the particles and the damper wall. In this process, a force between the particles must be generated. Based on the Hertz-Mindlin contact theory, the mechanical contact model of particle damping is shown in Fig. 4.

When the compressor is running under rated conditions, the equation of motion of the particles at a certain moment is

$$m_i\frac{d^2X_i}{dt^2} - m_ig = \sum_{j=1}^{S_i}(F_{nij} + F_{tij}) + F_{ij}(t) \qquad I_i\frac{d^2\varphi_i}{dt^2} = \sum_{j=1}^{S_i}T_{ij} \tag{3.1}$$

where F_{nij} is the normal contact force between the particle i and j, F_{tij} is the tangential contact force between the particle i and j, $F_{ij}(t)$ is the external excitation of the particle system. T_{ij} is

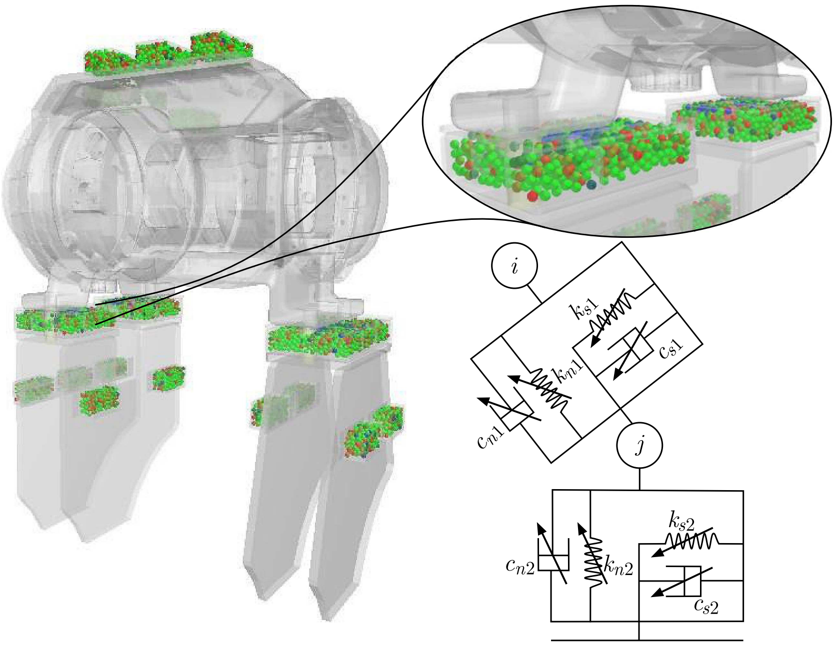

Fig. 4. Particle contact model of the compressor

the torque generated by the tangential contact force. m_i is mass of the particle i. I_i is the moment of inertia of the particle i. g is the acceleration of gravity. X_i is the displacement vector of the particle i. φ_i is the angular displacement vector of the particle i. s_i is the number of particles in contact with the particle i at a given moment.

The normal contact force between the particles is

$$F_{nij} = F_{kn} + F_{cn} \tag{3.2}$$

and

$$F_{kn} = \frac{4}{3}E^*\sqrt{R^*\alpha^3} \qquad F_{cn} = -2\sqrt{\frac{5}{6}}\beta\sqrt{S_n m^*}v_n^{rel} \tag{3.3}$$

where E^* is the equivalent elastic modulus. R^* is the equivalent particle radius. α is the amount of normal overlap. m^* is the equivalent mass. v_n^{rel} is the value of the normal component of the relative velocity. S_n is normal stiffness. $\beta = \ln e/\sqrt{\ln^2 e + \pi^2}$. e is the coefficient of restitution.

The tangential contact force between the particles is

$$F_{tij} = F_{kt} + F_{ct} \tag{3.4}$$

and

$$F_{kn} = \frac{4}{3}E^*\sqrt{R^*\alpha^3} \qquad F_{cn} = -2\sqrt{\frac{5}{6}}\beta\sqrt{S_n m^*}v_n^{rel} \tag{3.5}$$

where S_t is tangential stiffness, δ is tangential overlap, v_t^{rel} is tangential relative velocity.

The torque generated by the tangential contact force is

$$T_{ij} = -\mu_t F_{nij} R_i \omega_i \tag{3.6}$$

where μ_t is the rolling friction coefficient, R_i is the distance between the center of mass and the point of contact, ω_i is the unit angular velocity of the object at the contact point.

3.3. Energy consumption mechanism in the compressor particle dampers

Under an external excitation, the movement of the particle damping system mainly includes collision and friction between particles themselves and between particles and the wall of the damper. Therefore, damping particles mainly consume energy by collision and friction. According to the law of momentum conservation, when the particles i and j collide, the total energy consumption is

$$\Delta E = \sum \Delta E_{collision} + \sum \Delta E_{friction} \tag{3.7}$$

and

$$\Delta E_{collision} = \frac{m_i m_j (1-e^2)}{2(m_i + m_j)} |\Delta v|^2 \qquad \Delta E_{friction} = \mu F_{nij} \Delta S \tag{3.8}$$

where $E_{collision}$ is collision energy consumption, and $E_{friction}$ is friction energy consumption. e is the particle coefficient of restitution. Δv is the relative velocity before the collision. μ is the friction coefficient between the two particles. ΔS is the relative tangential displacement of the particles i and j.

4. Influence of different parameters on the energy consumption of dampers

4.1. Particle size

In order to make particle dampers achieve the purpose of energy dissipation and vibration reduction, it is necessary to ensure that the particles have geometrical characteristics of the rigid body and the flow state of a fluid, and the particle size is the main influencing factor.

The optimal particle size for different dampers is simulated as shown in Fig. 5. Considering the application environment of damping particles in the chiller, iron-based particles are used as the material. Material density is 7800 kg/m^3, elastic modulus is 210 GPa, Poisson's ratio is 0.3, particle recovery coefficient is 0.63, and the damper filling rate is 80%. The model is imported into DEM software, the excitation frequency is 386 Hz and the amplitude is 1 mm.

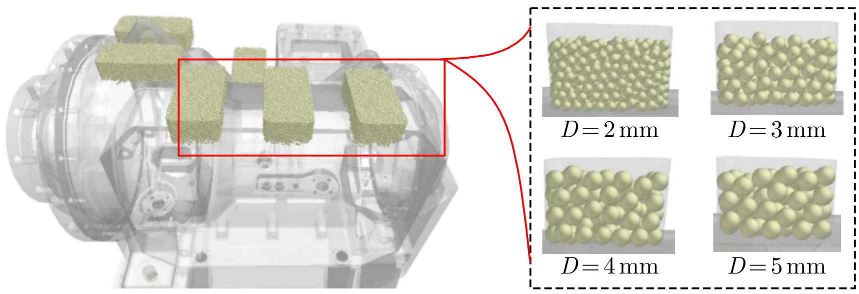

Fig. 5. Different particle sizes

The energy dissipation of each damper at different particle sizes and different times is shown in Fig. 6, and the energy dissipation trend of each damper particle is shown in Fig. 7. As can be seen from Figs. 6 and 7, when the particle size increases from 1 mm to 5 mm, the optimal particle size of different dampers varies. The optimal particle size of the top dampers of the compressor is 2 mm, for the foot dampers it is 5 mm, and for the support frame is 3 mm.

4.2. Excitation frequency and amplitude

According to the energy dissipation mechanism of particle damping, the more energy dissipated by particles, the better the effect of vibration reduction. In this paper, the influence

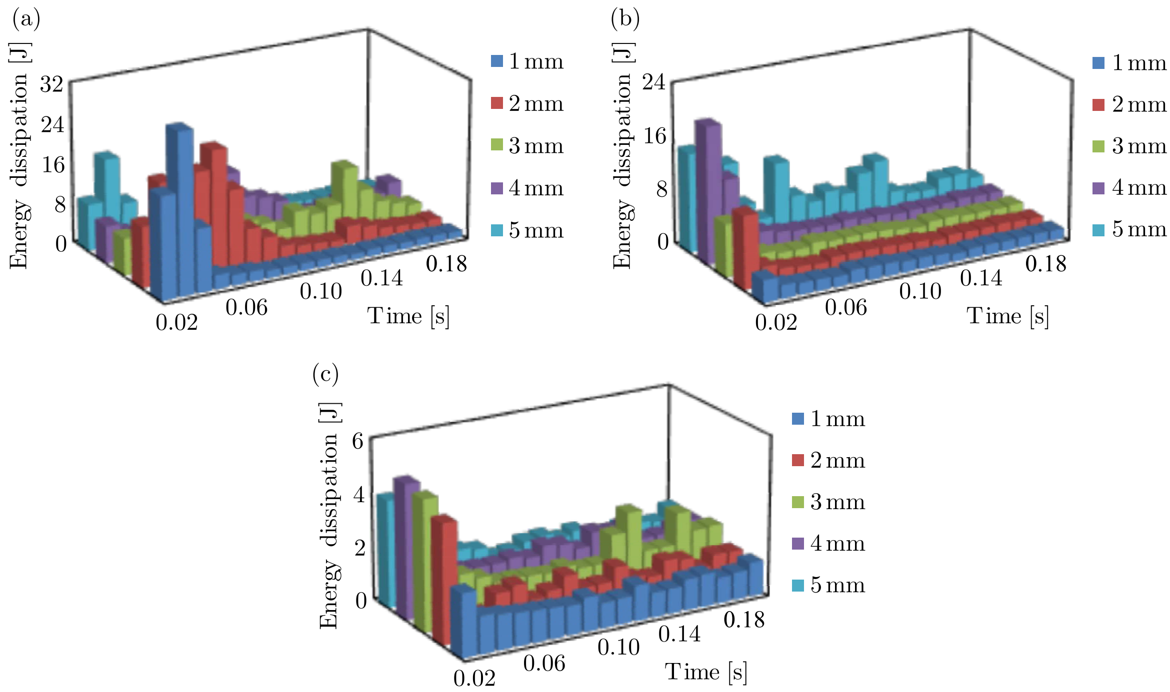

Fig. 6. Energy dissipation of dampers at different times: (a) compressor top, (b) compressor foot, (c) compressor support frame

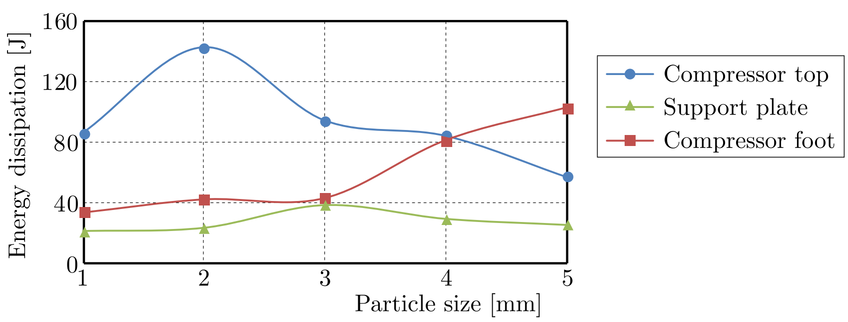

Fig. 7. Optimal particle sizes of different dampers

of excitation frequency and excitation amplitude on particle energy consumption is studied according to vibration reduction requirements of different frequency bands in practical operation. Based on DEM, the energy dissipation of dampers under different excitation frequencies and different excitation amplitudes is simulated, and the results are shown in Fig. 8.

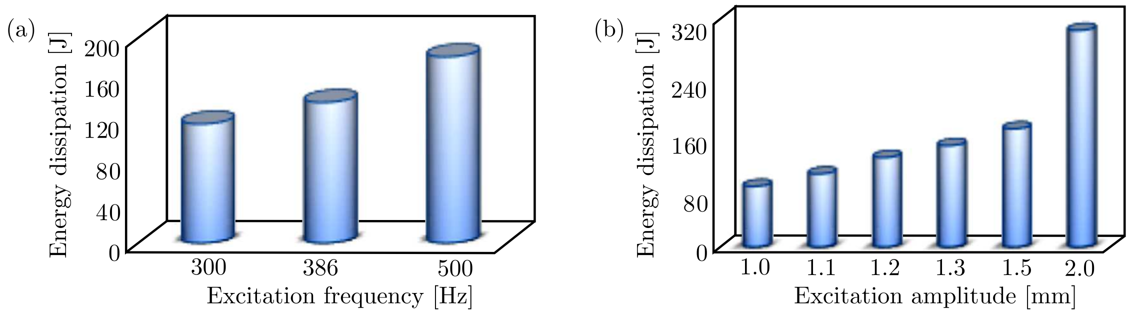

Fig. 8. Effects of different excitation frequencies and amplitudes on particle energy consumption: (a) different excitation frequencies, (b) different excitation amplitudes

It can be seen from Fig. 8 that the energy consumption of particles increases with an increase of frequency assumming that the excitation amplitude remains unchanged. Under the premise of constant excitation frequency, the energy consumption per unit time increases with growth of the amplitude, indicating that the energy dissipation effect of particles is superior for large amplitudes.

4.3. Optimal arrangement of particle dampers

According to the modal results, four installation schemes are designed, and the corresponding damper arrangement is shown in Fig. 9. The corresponding damper models and energy consumption at different excitation frequencies are shown in Table 2. The energy dissipation of particles with different damper schemes at different excitation frequencies is shown in Fig. 10.

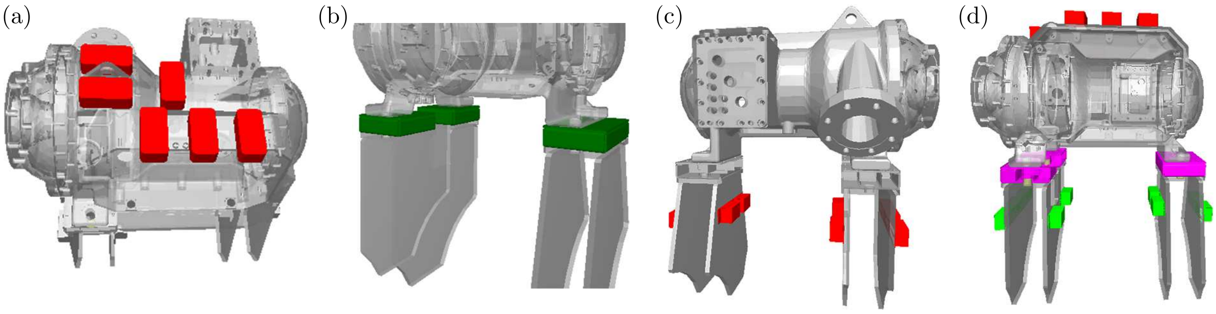

Fig. 9. Damper layout scheme: (a) scheme 1, (b) scheme 2, (c) scheme 3, (d) scheme 4

Table 2. Total energy consumption of the system in different schemes

Scheme	J1	J2	J3	J4	J5	300 Hz	386 Hz	500 Hz
1	√	√	×	×	×	43.36182	51.40913	61.16093
2	×	×	√	√	×	87.1398	105.8247	149.9172
3	×	×	×	×	√	15.74722	18.88648	21.72246
4	√	√	√	√	√	114.9274	148.2866	195.4099

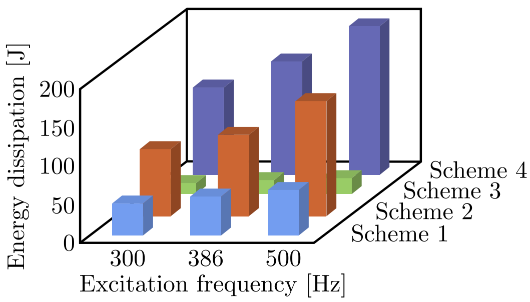

Fig. 10. Damping effect of different damper schemes

As can be seen from Table 2 and Fig. 10, particle dampers installed at different positions of the compressor have different energy dissipation levels under different excitation frequencies. The sequence from high to low is: scheme 4 > scheme 2 > scheme 1 > scheme 3. The combined installation is the optimal scheme.

5. Experimental verification of the vibration reduction effect of the centrifugal chiller

In order to verify the actual vibration reduction effect of the chiller in the optimal arrangement an experimental platform is built based on the centrifugal air-conditioning chiller, and experimental tests are carried out. The actual installation of the damper is shown in Fig. 11.

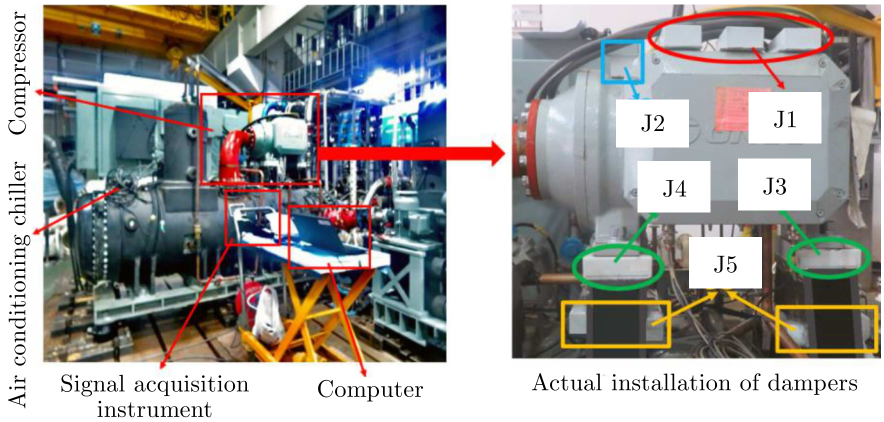

Fig. 11. Field experimentation

Acceleration sensors are arranged on the foot of the chiller compressor in the vertical direction. The particle dampers are instaled according to the optimal solution and the test system is calibrated, including the acceleration sensor and signal processing system. After setting the specified working conditions of the compressor (conversion frequency 386 Hz and 300 Hz), the data are collected at the vibration signal acquisition end, and the signal is transmitted to the computer end after being calculated and processed by the signal acquisition instrument. The waveform is displayed in the corresponding Date Acquisition & Signal Processing (DASP) software, and the signal data are recorded. The experimental data are shown in Table 3.

Table 3. Experimental data

Centrifugal air conditioner chiller (unit: dB)				
Measuring point	Excitation frequency	Without particle damper	With particle damper	Damping
Compressor foot	300 Hz	105.81	96.57	9.24
	386 Hz	107.25	96.63	10.62
	10-315 Hz (at 386 Hz)	97.74	90.81	6.93
	10-10k Hz (at 386 Hz)	125.3	118.88	6.42
	10-315 Hz (at 300 Hz)	107.63	97.16	10.47
	10-10k Hz (at 300 Hz)	117.08	110.04	7.04

The frequency spectrum of compressor acceleration in the target vibration reduction frequency band before and after installing the dampers is shown in Fig. 12. It can be seen from Table 3 and Fig. 12 that the particle dampers installed on the compressor have a significant damping effect, and the total vibration level of the compressor is reduced by 10.62 dB at 300 Hz. At 386 Hz, the total vibration level is reduced by 9.42 dB. When the compressor is running at 386 Hz, the total vibration stage value is reduced by 6.93 dB in the frequency range of 10 Hz--315 Hz. At 10 Hz-10 kHz, the total vibration stage value decreases by 6.42 dB. When the compressor runs at 300 Hz, the total vibration stage value decreases by 10.47 dB in the frequency range of 10 Hz-315 Hz. At 10 Hz-10 kHz, the total vibration stage value decreases by 7.04 dB.

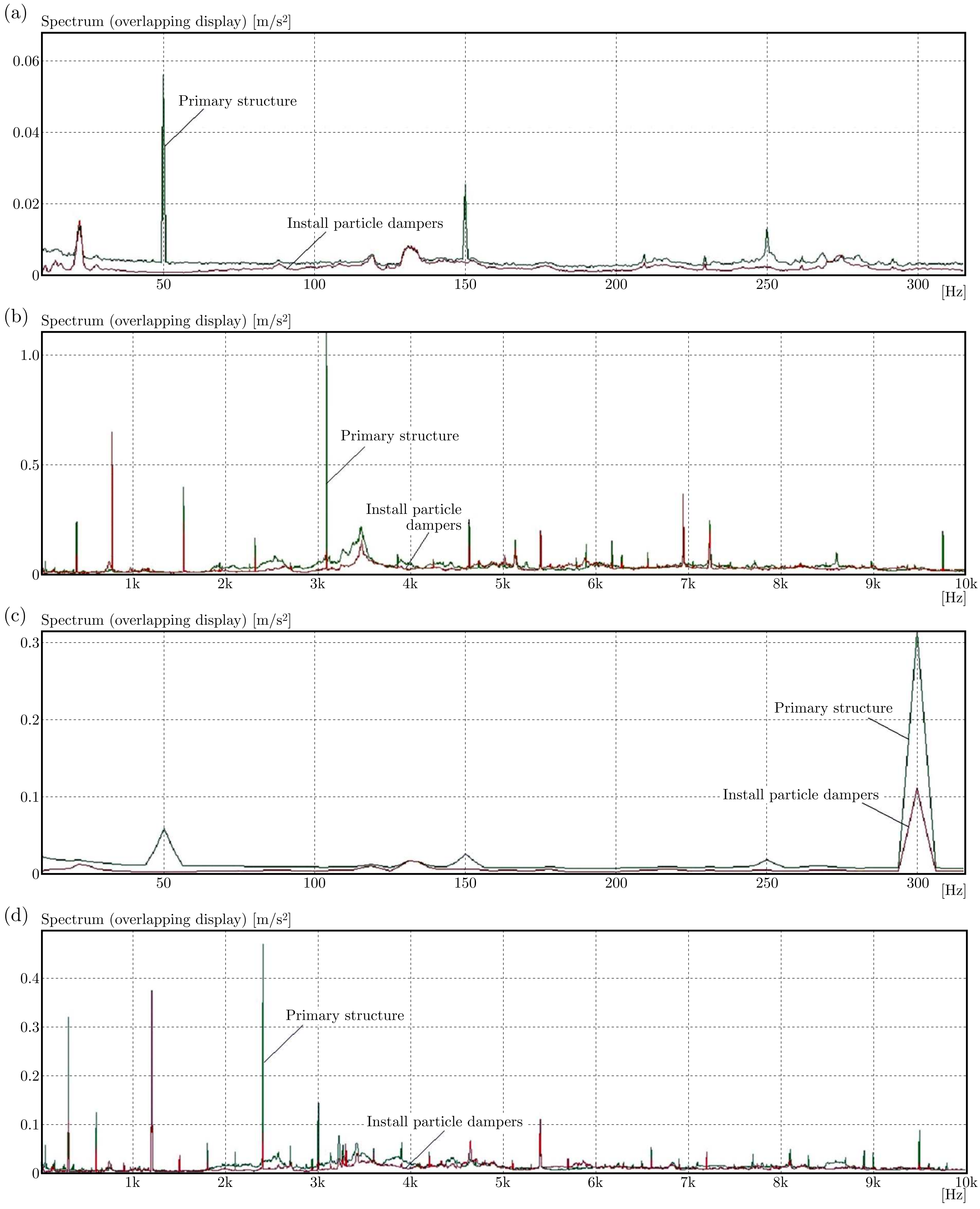

Fig. 12. Experiment effect. Vibration reduction effect diagram of: (a) 10 Hz-315 Hz unit (at 386 Hz), (b) 10 Hz-10 kHz unit (at 386 Hz), (c) of 10 Hz-315 Hz unit (at 300 Hz), (d) 10 Hz-10 kHz unit (at 300 Hz)

6. Conclusion

- Based on the dynamic model of a centrifugal air-conditioning chiller, dynamic characteristics of the chiller were analyzed, and the vibration transmission path of the chiller was obtained. Based on the finite element analysis, the vibration sensitive area of the chiller was obtained, and the installation area of the dampers was determined. The particle dampers were installed on the top of the compressor, and installed in parallel at the support frame position and mounted rigidly in series between the compressor and the support frame.

- Based on the particle damping and DEM, the optimal particle size and the optimal installation scheme of each damper of the unit were studied. According to simulation results, the optimal particle size of the top damper was 2 mm, the optimal particle size of the foot was 5 mm, and the optimal particle size of the support frame was 3 mm. Among the four installation methods of particle dampers, the combined installation occurred to be the best one.
- The damping effect of the compressor in the optimal configuration of dampers was verified by experiments. The experimental results showed that under the optimal configuration, the total vibration level was reduced by 10.62 dB at 300 Hz and was reduced by 9.42 dB at 386 Hz. When the compressor was running at 386 Hz, the total vibration stage value was reduced by 6.93 dB in the frequency range of 10 Hz-315 Hz and was reduced by 6.42 dB in the frequency range of 10 Hz-10 kHz. When the compressor ran at 300 Hz, the total vibration stage value decreased by 10.47 dB in the frequency range of 10 Hz-315 Hz and decreased by 7.04 dB in the frequency range of 10 Hz-10 kHz.
- A vibration reduction method for centrifugal chiller based on particle damping was proposed. Based on dynamic analysis and DEM, particle dampers were introduced to reduce vibration of the chiller. Simulation calculations and experimental results verified feasibility of the particle discrete element model, and the a method was proposed for vibration control of a centrifugal air-conditioning chiller.

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