Research Article

Research and Application of Vibration Isolation Platform Based on Nonlinear Vibration Isolation System

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In order to weaken the ground vibration caused by mechanical equipment, a vibration isolation platform of nonlinear vibration isolation system is proposed. The influence of flexible foundation and nonlinear isolator parameters on the performance of nonlinear vibration isolation platform is studied. By optimizing the nonlinear parameters of the vibration isolation platform, the vibration response of equipment vibration is effectively blocked. The research results show that the structural parameters such as stiffness and damping of the nonlinear vibration isolation system affect the vibration mode of the vibration isolation platform, thus affecting the vibration response of the whole system. Applying the nonlinear vibration isolation system to the project, the impact of the vibration of the refiner on the vibration of the floor is greatly reduced. It shows that this method can be used to reduce the foundation vibration caused by the equipment, improve the operation status of the equipment, and increase the safety factor of the building.

1. Introduction

1.1. Research Background. Controlling structural vibration is a long-standing problem and a key design requirement in the several applications, as witnessed by the huge effort made in the last decades in developing innovative devices, and associated modeling methodology, specifically suited for this purpose. Moreover, the range of engineering branches facing this problem is definitely wide as all systems involving moving parts lead to vibrations [1].

It is, for instance, the case of the refiner in a factory production workshop, it will produce vibration when it works. The installation did not take appropriate fixing methods and vibration isolation measures, and its operating status changed greatly, which also brought great safety hazards to the building structure. In order to solve the vibration problem of the installation floor of the refiner in this factory, a vibration isolation platform of a nonlinear vibration isolation system is proposed in this paper to block the transmission of vibration. Effectively improve the operating status of equipment and improve building safety.

Over the years, many achievements have been made in the field of vibration control, and remarkable results have been achieved in the practical applications. However, due to the constraints of the actual working environment, many engineering problems have caused a series of technical difficulties. In this paper, on the basis of considering the non-rigidity of the vibration isolation platform and the nonlinearity of the vibration isolator, the parameter optimization analysis of the nonlinear vibration isolation system is carried out.

1.2. Classification of Vibration Control. Vibration source, transmission path, and controlled object must be considered in a vibration control. The corresponding measures include: control of vibration source, control of transmission path, and control of the controlled object. From the different points of view, a variety of control classification methods have been formed. However, the vibration control methods that have received attention and are widely used are shown in Figure 1.
[2–4]: (1) controlling the vibration of the vibration source; (2) vibration isolation; (3) dynamic vibration absorption; (4) damping vibration reduction; and (5) modify the structure.

Among the above vibration control methods, vibration isolation is the most widely used. According to the purpose of a vibration isolation, vibration isolation is usually divided into two categories. One is to take vibration isolation measures for the mechanical equipment as the vibration source to prevent the vibration generated by the vibration source from spreading outward, which is called active vibration isolation. The other is to take vibration isolation measures for equipment that is afraid of vibration interference to weaken or eliminate the adverse effects of external vibration on the equipment, which is called negative vibration isolation. Of course, active systems are usually able to provide higher performance in vibration control; however, these are often costly while offering relatively low reliability due to their intrinsic higher complexity. Furthermore, their applicability to large scale problems is usually limited by the availability of appropriate actuators. Passive systems are therefore preferred when dealing with sufficiently large systems. It is the case, for instance, of ensuring high reliability to primary building such as schools, museums power plants, as well as structural control in bridges, with strong social, political, and economic implications. In this view, nonlinear passive isolation techniques are probably the most promising, as the nonlinearity can lead to better isolation performance in a much wider input spectrum. For these reasons, several possible source of nonlinearity have been explored in the last few decades.

2. Research Status of Nonlinear Vibration Isolation Theory

2.1. Research Status of Vibration Isolation Theory. Vibration isolation is to connect a vibration isolator in series between the vibration source and the vibration-isolated equipment. The near-rigid connection between the vibration source and the foundation or connection structure are changed to an elastic connection to reduce the transmission of vibration energy, and finally achieve the purpose of reducing vibration and noise. At present, the main vibration isolators can be divided into three categories according to the different materials: metal springs, rubber, and air springs [5].

The metal spring type vibration isolator is not afraid of oil pollution and high-temperature resistance. It is widely used. Many researchers have done a lot of research on it. Jiang Hongyuan and others established a bilinear hysteresis vibration model with nonlinear viscous damping for multilayer steel plate vibration isolators. Through experimental research, it is shown that the stiffness and energy consumption coefficient of the multilayer steel plate vibration isolator change nonlinearly with the change of the vibration amplitude, and the damping coefficient is affected by the number of steel plate layers and the installation preload [6]. Tong and Qinglin [7] established the linear model of the system under different states, and proved that it is feasible to use a linear model to describe the nonlinear characteristics within a certain experimental range. Yong and Yingyun [8] and others studied the shock resistance characteristics of steel wire rope vibration isolators under no-load and preload conditions by a hammering method. Liu et al. [9] proposed a geometrically nonlinear vibration isolator with low stiffness and high damping, which can flexibly adjust the design parameters of the vibration isolation system to meet different vibration isolation requirements.

At the same time, many researchers have done research on the dynamic characteristics of rubber vibration isolators. The research of Debao and Xigeng [10] showed that the stiffness and damping of the rubber vibration isolator have a curved surface relationship with the amplitude and frequency. Xiaoyong et al. [11] and others established a nonlinear model of the rubber isolators dynamic characteristics based on the hyperelasticity, fractional derivatives, and friction models, which is used to describe the correlation between the dynamic characteristics of rubber isolators and preload, frequency, and amplitude.

The structural parameters of the traditional vibration isolators are optimally set for a specific environment. Once the parameters are set, they cannot be changed, which lacks the flexibility of control. In order to meet the needs of flexible control, a large number of new technologies and materials have emerged in recent years. Foreign experts or the institutions have developed new types of vibration isolators such as spherical rubber vibration isolators, high-performance reinforced polyurethane vibration isolators, and airbag type vibration isolators. Some researchers have applied technologies such as electrorheology, magnetorheology, and quasi-zero stiffness to the field of vibration isolation, realizing the adjustable performance parameters of the vibration isolator. It can adapt to the requirements of different working conditions, and has made breakthroughs in low-frequency vibration isolation. Domestic researcher Mi [12] established a double-layer active and passive vibration isolation system model with the additional springs for the multivibration transmission path system, and analyzed the force transmission rate and energy transmission characteristics of the rigid and elastic foundation systems. Bo et al. [13] adopted the principle of positive and negative stiffness parallel to construct a tensile quasi-zero stiffness vibration isolator for low-frequency vibration isolation. Yuanhan et al. [14] proposed a
connecting-rod tilt spring negative stiffness mechanism to solve the problem of low-frequency vibration isolation performance of traditional passive vibration isolators, and formed a new quasi-zero stiffness vibration isolator in parallel with linear spring. Yubin et al. [15] designed and manufactured a composite bidirectional magnetorheological elastomer vibration isolator, used finite element software to conduct electromagnetic simulation analysis of the isolator, and built an experimental platform to test the performance of the isolator. Liang et al. [16] and others used MATLAB to simulate and analyze the response characteristics of the positive and negative stiffness parallel vibration isolation system under sweep frequency excitation and random excitation, and then built a force transfer rate test platform to verify the correctness and rationality of the established dynamic model.

2.2. Research Status of Nonlinear Vibration Isolation System. The modeling of vibration isolation systems is mostly limited to linear theory. However, due to the impact of the non-ground floor structure, the nonrigidity of the foundation in vibration isolation is very important. Hard nonlinear and damping properties of commonly used vibration isolation materials. The traditional linear shock isolation theory can no longer meet the requirements [17]. Therefore, nonlinear vibration has received more and more attention.

In the 17th century, C. Huygens noticed the deviation of the isochronism of the large-amplitude vibration of the single pendulum and the synchronization of two frequencies close to the clock, which was the earliest record of the nonlinear vibration phenomenon. In the late 19th century, people began to study rigorous nonlinear vibration theory. Poincaré laid the theoretical foundation, and he proposed a new direction of qualitative theory [18].

The nonlinear factors brought about by the nonrigidity of the foundation is an important aspect of nonlinear research. Foreign countries began to study the theory of flexible vibration isolation from the late 1950s to the early 1960s. Snowton [19] studied a fully symmetrical one-dimensional vibration isolation system, considering the influence of foundation flexibility. Den Hartog [20] discussed the effect of foundation impedance on vibration isolation. Hamme [21] proved the mutual coupling of machine and foundation with experiments. Ravindra and Mallik [22] studied the stability of nonlinear systems with symmetric and asymmetric restoring forces and the influence of various parameters on the system. Shekhar et al. [23] studied nonlinear shock isolators.

Domestic research on flexible vibration isolation systems began in the early 1980s. Yuguo et al. [24] studied the power flow transfer characteristics of a multisupport vibration isolation system on a flexible board. Kongjie [25] studied the influence of foundation elasticity on the natural frequency of the vibration isolation system through perturbation theory. Yongbin and Shuhui [26] used the incremental harmonic balance method to solve the motion response of the nonlinear vibration isolation system. Xiansheng et al. [27] established the hysteretic restoring force model of the wire rope isolator. On the basis of considering the nonlinear stiffness and nonlinear damping effect of the nonlinear vibration isolator, Li et al. [17] established the Newton–Euler equation of the double-layer vibration isolation system. From the viewpoint of mechanical impedance, Jianbo et al. [28] deduced the calculation formula of transmissibility based on flexibility. The research results show that the fundamental frequency and quality of the flexible foundation are the most important factors affecting the vibration isolation effect [28]. Xiao et al. [29] established the geometric nonlinear vibration equation of the inclined spring damping system under the excitation of the foundation displacement, studied the resonance problem of the inclined spring damping system under the excitation of the foundation displacement, and obtained the oblique spring damping system by using the KBM method. The magnitude–frequency response equations of resonance [29].

Many important achievements have been made in the research of nonlinear vibration, and many problems in theory and engineering application have been solved. Due to the rapid development of computer technology, many nonlinear vibration problems are solved by means of numerical simulation and numerical calculation. This makes the solution of nonlinear vibration problems a big step forward. Nonlinear problems have a certain complexity, and it is still difficult to completely solve many nonlinear problems in mathematics and mechanics.

At present, many nonlinear vibration problems in engineering are still analyzed and calculated by using some approximate methods or ignoring some nonlinear factors. In most cases, there is still a certain amount of error between the obtained results and the actual results. Aiming at this problem, this paper establishes a refiner model. Considering the nonlinear vibration isolator and the elastic foundation, the influence of each parameter on the vibration transfer characteristics of the system is studied through dynamic analysis.

3. Establishment of the Vibration Isolation System Model of the Refiner

3.1. Mathematical Model of Nonlinear Vibration Isolation System of Vibration Isolation Platform

3.1.1. Research Object. The mechanical model of the nonlinear vibration isolation system selected in this paper is shown in Figure 2. The middle layer of the vibration isolation system is the vibration isolation platform. Above the vibration isolation platform is the unit equipment, its mass is m1. The lower layer of the vibration isolation system is a cement foundation. There are several vertical nonlinear vibration isolators on the upper and lower floors. The numbering of the vibration isolators is from right to left, and from front to back is No. 1–8.
3.1.2. Model Establishment (1) Derivation of the elastic force and damping force of the vibration isolator [30]

It is assumed that the vibration isolators are modeled with nonlinear springs and dampers. For this vibration isolation system, the vibration isolators are connected between the floor foundation and the steel structure platform.

The deformation of the vibration isolator connected to the foundation is shown in Figure 3. The subscripts \( i \) and \( j \) indicate the number of connection points at both ends of the vibration isolator with number \( i \).

Let the height vector of the vibration isolator before deformation be

\[
\vec{r}_{j0} = \vec{r}_j - \vec{r}_0 = \vec{r}_{cm0} + B_m\vec{r}_{ci} - \vec{r}_{j0}. \tag{1}
\]

The height vector of the vibration isolator after deformation is

\[
\vec{r}_j = \vec{r}_i - \vec{r}_j = \vec{r}_{cm} + B_m\vec{r}_{ci} - \vec{r}_j. \tag{2}
\]

The above two equations are subtracted to obtain the deformation of the vibration isolator:

\[
\Delta \vec{r}_i = \vec{r}_{cm} + B_m\vec{r}_{ci} - \vec{r}_j - \vec{r}_{cm0} - B_m\vec{r}_{ci0} + \vec{r}_{j0} = (\vec{r}_{cm} - \vec{r}_{cm0}) + (B_m\vec{r}_{ci} - B_m\vec{r}_{ci0}) - (\vec{r}_j - \vec{r}_{j0}) = \Delta \vec{r}_m + (B_m\vec{r}_{ci} - B_m\vec{r}_{ci0}) - (\vec{r}_j - \vec{r}_{j0}). \tag{3}
\]

In the formula:

\( \Delta \vec{r}_i \) is the coordinate change of vibration isolator \( i \) in the inertial coordinate system;

\( \Delta \vec{r}_m \) is the coordinate change of the center of mass of rigid body \( B_m \) in the inertial coordinate system;

At time \( t = 0 \), \( B_m\vec{r}_{ci0} \) is the coordinates of the \( i \)-end of the isolator \( i \) on the rigid body \( m \) relative to the connected body base.

**Figure 3:** Height vector of ground vibration isolator.

**Figure 4:** The main axis coordinate system of the vibration isolator.

At time \( t = 0 \), \( \vec{r}_{j0} \) is the coordinate of end \( j \) of vibration isolator \( i \) in the inertial coordinate system;

At time \( t = 0 \), \( \vec{r}_{cm0} \) is the coordinates of the center of mass of the rigid body \( B_m \) in the inertial coordinate system;

The main axis coordinate system of the vibration isolator is shown in Figure 4. Suppose the included angles between the longitudinal axis \( e'_1 \) of the vibration isolator and the three bases \( e_1 \), \( e_2 \), and \( e_3 \) of the inertial coordinate system are \( \theta \), \( \varphi \), and \( \psi \), respectively, and the transverse axis \( e'_2 \) is located in the plane formed by \( e_2 \), \( e_3 \). For vibration isolators arranged arbitrarily in Figure 3, the coordinate transformation is as follows:

\[
e'_3 = e_1\cos \theta + e_2\cos \varphi + e_3\cos \psi. \tag{4}
\]

For

\[
e'_2 \times e_1 = (e_1\cos \theta + e_2\cos \varphi + e_3\cos \psi) \times e_1 = e_2\cos \varphi - e_3\cos \psi \tag{5}
\]

So

\[
e'_2 = \frac{e_2\cos \varphi - e_3\cos \psi}{\sqrt{\cos^2 \varphi + \cos^2 \psi}}. \tag{6}
\]

Let \( W = \sqrt{\cos^2 \varphi + \cos^2 \psi} \), then

\[
e'_1 = e'_2 \times e'_3 = \frac{e_2\cos \varphi - e_3\cos \psi}{W} \times (e_1\cos \theta + e_2\cos \varphi + e_3\cos \psi) = \frac{1}{W} (-e_2\cos \theta \cos \varphi - e_2\cos \theta \cos \psi + e_1\cos^2 \varphi + e_1\cos^2 \psi) = \frac{1}{W} [e_1(\cos^2 \varphi + \cos^2 \psi) - e_2\cos \theta \cos \psi - e_3\cos \theta \cos \varphi]. \tag{7}
\]
For the vibration isolator \( i \), the deformation of the vibration isolator in each axis direction is:

\[
l_{ix} = \Delta r_i \cdot e'_i = \Delta r_{ix} W_i - \frac{1}{W_i} \left( \Delta r_{i\theta} \cos \theta_i \cos \phi_i + \Delta r_{i\phi} \cos \theta_i \cos \phi_i \right)
\]

(8)

\[
l_{iy} = \Delta r_i \cdot e'_y = \frac{1}{W_i} \left( \Delta r_{i\theta} \cos \phi_i - \Delta r_{i\phi} \cos \phi_i \right)
\]

(9)

\[
l_{iz} = \Delta r_i \cdot e'_z = \Delta r_{iz} \cos \theta_i + \Delta r_{i\phi} \cos \phi_i + \Delta r_{i\theta} \cos \phi_i
\]

(10)

Let the nonlinear elastic force in the direction of the main axis of the vibration isolator be as follows:

\[
F_{ix} = \kappa (\dot{\ell}_{ix}) \quad F_{iy} = \lambda (\dot{\ell}_{iy}) \quad F_{iz} = \eta (\dot{\ell}_{iz})
\]

(11)

According to the above results, the elastic force can be transformed into the form of the following inertial system, namely

\[
F_i = F_{ix} \cdot e_1 + F_{iy} \cdot e_2 + F_{iz} \cdot e_3,
\]

(12)

among

\[
\begin{align*}
F_{ix} &= F_{ix} W_i + F_{iz} \cos \theta_i \\
F_{iy} &= \frac{1}{W_i} \left( -F_{ix} \cos \theta_i \cos \phi_i + F_{iz} \cos \phi_i \right) + F_{iz} \cos \phi_i \\
F_{iz} &= -\frac{1}{W_i} \left( F_{ix} \cos \theta_i \cos \phi_i - F_{iz} \cos \phi_i \right) + F_{iz} \cos \phi_i
\end{align*}
\]

(13)

Similarly, the damping force described in the inertial coordinate system is:

\[
\begin{align*}
R_{ix} &= R_{ix} W_i + R_{iz} \cos \theta_i \\
R_{iy} &= \frac{1}{W_i} \left( -R_{ix} \cos \theta_i \cos \phi_i + R_{iz} \cos \phi_i \right) + R_{iz} \cos \phi_i \\
R_{iz} &= -\frac{1}{W_i} \left( R_{ix} \cos \theta_i \cos \phi_i - R_{iz} \cos \phi_i \right) + R_{iz} \cos \phi_i
\end{align*}
\]

(14)

(2) Determination of the torque of the vibration isolator [30]

Let the moment of the vibration isolator \( i \) on the unit \( m \) be \( M_i \), and for the vibration isolator connected to the foundation:

\[
M_i = b_n r_{zi} \times (F_i + R_i).
\]

(15)

The moment of the vibration isolator to the underlying foundation:

\[
b_n M_i = b_n r_{zi} \times (F_i + R_i)
\]

(16)

(3) Vibration of elastic foundation platform [31]

The flexible foundation can be simplified as a thin plate simply supported at four ends. The thin plate is a plate whose thickness is much smaller than the planar area. If the position of each point on the plate is defined by the rectangular coordinate \((x, y)\) in the neutral plane, the displacement \(u_z\) of the \(z\)-axis perpendicular to the plate surface is used to represent the transverse bending deformation of each point. Then \(u_z\) is a function of the two-dimensional coordinate \((x, y)\) and time \(t\), namely \(u_z(x, y, t)\). Plates in transverse bending vibration can be divided into two cases. The first ignores the effect of shear deformation, assuming that the line perpendicular to the neutral plane of the plate before deformation is still perpendicular to the plane during deformation. Such plates are called Kirchhoff plates. The second type considers shear deformation, and this kind of plate is called Mindlin plate [32].

As shown in Figure 5, \(xoy\) is the Cartesian coordinate system in the middle plane of the thin plate, and \(w\) is the deflection in the \(z\) direction. According to the Kirchhoff assumption, there is a relationship between the internal moment and the deflection.

\[
\begin{align*}
M_x &= -D \left( \frac{\partial^2 w}{\partial x^2} + \mu \frac{\partial^2 w}{\partial y^2} \right) \\
M_y &= -D \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} \right) \\
M_{xy} &= -D(1 - \mu) \frac{\partial^2 w}{\partial x \partial y}
\end{align*}
\]

(17)

In Equation (17): \(M_x\) is the bending moment acting on the midplane of unit length perpendicular to the \(x\) direction; \(M_y\) is the bending moment acting on the midplane of unit length perpendicular to the \(y\) direction; \(M_{xy}\) is the bending moment acting on the midplane per unit length perpendicular to the \(x\) and \(y\) directions; \(D = Eh^3/12(1 - \mu^2)\) is the bending stiffness of the sheet and \(\mu\) is Poisson’s ratio.
For the Kirchoff plate, the shear force $Q_x$ and $Q_y$ on the midplane per unit length perpendicular to the $x$-direction or $y$-direction can be obtained according to the moment balance condition of the microelement body:

$$Q_x = \frac{\partial M_x}{\partial y} + \frac{\partial M_{xy}}{\partial y},$$

$$Q_y = \frac{\partial M_y}{\partial x} + \frac{\partial M_{xy}}{\partial x}. \quad (18)$$

Applying D’Alembert’s principle in the $z$ direction, the dynamic balance equation of the plate is obtained:

$$\frac{\partial^2 M_x}{\partial x^2} + 2 \frac{\partial^2 M_{xy}}{\partial x \partial y} + \frac{\partial^2 M_y}{\partial y^2} + \rho h \frac{\partial^2 w}{\partial t^2} = 0. \quad (19)$$

In Equation (19), $\rho h$ is the mass per unit area. The Equation (17) is substituted into the Equation (19), and the vibration equation is obtained:

$$\frac{\partial^2}{\partial x^2} \left( \frac{\partial^2 w}{\partial x^2} + \mu \frac{\partial^2 w}{\partial y^2} \right) + \frac{\partial^2}{\partial y^2} \left( \frac{\partial^2 w}{\partial x^2} + \mu \frac{\partial^2 w}{\partial y^2} \right) + \rho h \frac{\partial^2 w}{\partial t^2} = 0. \quad (20)$$

Using the separation of variables method to solve Equation (20), set

$$w(x, y, t) = \varphi(x, y) \tau(t). \quad (21)$$

Substituting Equation (21) into (20), we get

$$\frac{\partial^4 \varphi}{\partial x^4} + 2 \frac{\partial^4 \varphi}{\partial x^2 \partial y^2} + \frac{\partial^4 \varphi}{\partial y^4} + \frac{\rho h}{D} \frac{\partial^2 \varphi}{\partial t^2} = 0. \quad (22)$$

In Equation (22), $\tau + w^2 \tau = 0$. \quad (23)

In Equation (23), $\omega$ is the natural angular frequency of the transverse bending vibration of the thin plate.

From this, the forced vibration equation of a simply supported rectangular thin plate with four sides can be obtained:

$$\frac{E h^3}{12(1-\mu^2)} \nabla^4 w(x, y) + \rho h \frac{\partial^2 w}{\partial t^2} = f(x, y). \quad (24)$$

Boundary conditions: $w|_{x=0,a} = w|_{y=0,b} = 0$ and $\partial^2 w/\partial x^2|_{x=0,a} = \partial^2 w/\partial y^2|_{y=0,b} = 0$

Among them, $a$, $b$, $h$, $\rho$ are the length, width, thickness, and density of the rectangular thin plate, respectively; $\mu$, $E$ are Poisson’s ratio and Young’s modulus, respectively; $w(x, y)$ is the displacement response of point $(x, y)$ on the thin plate; $\delta(\cdot)$ is the $\delta$ function; $\sigma_c$ is the installation point of the vibration isolation support on the thin plate.

Assumptions:

$$\varphi(x, y) = \sum_{i=1}^{\infty} \sum_{j=1}^{\infty} B_{ij} \sin \frac{i\pi x}{a} \sin \frac{j\pi y}{b}. \quad (25)$$

Among them: $B_{ij}$ is the amplitude constant determined by the initial conditions.

Substitute Equation (25) into (22) to get

$$\sum_{i=1}^{\infty} \sum_{j=1}^{\infty} B_{ij} \left[ \left( \frac{i^2}{a^2} + \frac{j^2}{b^2} \right) \pi^4 - \frac{\rho h}{D} \pi^2 \right] \sin \frac{i\pi x}{a} \sin \frac{j\pi y}{b} = 0. \quad (26)$$

Then, multiply $\sin i\pi x/a \sin j\pi y/b$ by Equation (26) and perform area integration, using the orthogonality of trigonometric functions, we can get

$$\left( \frac{i^2}{a^2} + \frac{j^2}{b^2} \right) \pi^4 - \frac{\rho h}{D} \pi^2 = 0 \quad \text{for} \quad i, j = 1, 2, \ldots. \quad (27)$$

So the natural frequency is

$$\sigma_{ij} = \left( \frac{i^2}{a^2} + \frac{j^2}{b^2} \right) \pi^2 \sqrt{\frac{\rho h}{D}} \quad \text{for} \quad i, j = 1, 2, \ldots. \quad (28)$$

The modal function of the response is

$$\varphi_{ij}(x, y) = \sin \frac{i\pi x}{a} \sin \frac{j\pi y}{b}. \quad (29)$$
3.2. Influence of Platform Flexibility on Vibration Isolation System. A steel structure vibration isolation platform is established in SOLIDWORKS, and its size is 3,600 long * 1,200 wide * 240 high (mm). Import the geometric model into ANSYS, and perform grid division according to the calculation requirements, as shown in Figure 6. Table 1 is the first 20 natural frequencies of the vibration isolation platform. Figures 7–12 are part of the vibration mode diagrams of the free modal analysis of the platform.

<table>
<thead>
<tr>
<th>Order</th>
<th>Frequency (Hz)</th>
<th>Order</th>
<th>Frequency (Hz)</th>
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<th>Frequency (Hz)</th>
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<td>12</td>
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</tbody>
</table>

Table 1: The first 20 order natural frequencies of the vibration isolation platform.
Export the mnf file of the platform from ANSYS, import the mnf file into ADAMS, replace the original rigid platform, and form a flexible foundation platform. The constraints of the flexible foundation platform are simplified as four-sided fixation, and a linear vibration isolation system on the flexible foundation platform is obtained. The simulation calculation and analysis of the linear vibration isolation system with flexible foundation are carried out.

It can be seen from Figure 13 that the characteristic parameters of the flexible system tend to be stable after 0.05 s, while the rigid system takes about 0.3 s. This shows that the equipment in the flexible system can reach stability faster than the rigid system.

3.3. Influence of Nonlinearity on Vibration Isolation System

At present, nonlinear vibration isolators are generally divided into two kinds of vibration isolators with hard characteristics and soft characteristics. The force of the vibration isolation support can be expressed as:

$$F(x, \dot{x}) = -kx|x|^{a-1} - c\dot{x}|x|^{p-1} (p, q > 0).$$  \hspace{1cm} (30)

Among them, $k$ and $c$ are the stiffness and damping coefficients of the vibration isolator, respectively; $x$ is the deformation of the vibration isolator; $p$ and $q$ are the nonlinear characteristic parameters of damping and stiffness, respectively.

In Equation (29), in the case of small amplitude ($x < 1$), when $q > 1$, the vibration isolator behaves as soft characteristics; when $q < 1$, the vibration isolator behaves as hard characteristics. As shown in Figure 14.

![Figure 13: Comparing diagram of vibration characteristics of flexible linear system and rigid linear. (a) Vibration displacement of refiner. (b) Vibration acceleration of refiner.](image1)

![Figure 14: Hard and soft characteristic curve.](image2)

![Figure 15: Force–displacement diagram of the stiffness curve in the $x$-direction and $y$-direction of the vibration isolator.](image3)
The vibration isolator with soft characteristics has a larger deformation. The smaller the stiffness of the vibration isolator, the larger the reaction force can be provided at the initial stage of deformation, and the large-amplitude vibration can be suppressed. In the later stage of the response of the vibration isolator, the stored energy is gradually released, so that the vibration is effectively controlled. For a vibration isolator with rigid characteristics, as the amplitude increases, the reaction force provided also increases.

According to the actual situation of the refiner, use the cubic polynomial 30 to obtain the stiffness curves of the vibration isolator in the $x$-direction and $y$-direction. Force and displacement function and relationship diagram (Figure 15) are as follows:

\[
F_x = 5.61 \times 10^4 r^3 + 1.88 \times 10^6 r. \tag{31}
\]

Likewise, a cubic polynomial 31 is used to obtain the isolator $z$-direction stiffness curve. The relationship diagram of force and displacement function (Figure 16) is as follows:

\[
F_z = 9.76 \times 10^4 z^3 + 7.67 \times 10^4 z^2 + 3.76 \times 10^6 z. \tag{32}
\]

The damping of the vibration isolator is parameterized, and the stiffness curves of the vibration isolator in the three directions of $x$, $y$, and $z$ are used to represent the stiffness values in each direction. The calculation result is shown in Figure 17.
TABLE 2: Test equipment and instruments.

<table>
<thead>
<tr>
<th>Name</th>
<th>Size</th>
<th>Manufacturer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accelerometer</td>
<td>PCB356A16</td>
<td>PCB, USA</td>
</tr>
<tr>
<td>Data acquisition instrument</td>
<td>SCM205</td>
<td>SIEMENS</td>
</tr>
<tr>
<td>Data processing software</td>
<td>Simcenter Testlab 2021</td>
<td>SIEMENS</td>
</tr>
</tbody>
</table>

FIGURE 19: The position of the measuring point on the left side of the refiner (viewed from the mill to the motor).

FIGURE 20: The location of measuring points on the right side of the refiner (viewed from the mill to the motor).

TABLE 3: Main parameters of equipment.

<table>
<thead>
<tr>
<th>Device name</th>
<th>Name</th>
<th>Parameter</th>
<th>Name</th>
<th>Parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motor</td>
<td>Size</td>
<td>YX3-355M2-6</td>
<td>Rated current (A)</td>
<td>341</td>
</tr>
<tr>
<td></td>
<td>Weight (kg)</td>
<td>1,715</td>
<td>Power (kW)</td>
<td>185</td>
</tr>
<tr>
<td></td>
<td>Rated voltage (V)</td>
<td>380</td>
<td>Frequency (Hz)</td>
<td>50</td>
</tr>
<tr>
<td></td>
<td>Rated speed (rpm)</td>
<td>990</td>
<td>Efficiency</td>
<td>95.0%</td>
</tr>
<tr>
<td></td>
<td>Insulation class</td>
<td>F</td>
<td>Connection method</td>
<td>Δ</td>
</tr>
<tr>
<td>Mill</td>
<td>Maximum allowable motor power (kW)</td>
<td>400</td>
<td>Maximum pulping pressure (MPa)</td>
<td>0.5</td>
</tr>
<tr>
<td></td>
<td>Maximum allowable speed (rpm)</td>
<td>1,500</td>
<td>Flow (m³/hr)</td>
<td>23–67</td>
</tr>
<tr>
<td></td>
<td>No-load shaft power (kW)</td>
<td>60</td>
<td>Maximum working temperature/°C</td>
<td>60</td>
</tr>
<tr>
<td></td>
<td>Inlet pressure (MPa)</td>
<td>0.08–0.3</td>
<td>Seal water flow (L/min)</td>
<td>0.5–5</td>
</tr>
<tr>
<td></td>
<td>Water seal water pressure is higher than slurry inlet pressure (MPa)</td>
<td>0.05–0.1</td>
<td>Seal water temperature (°C)</td>
<td>&lt;20</td>
</tr>
</tbody>
</table>
FIGURE 21: Test-site.

FIGURE 22: Spectrum diagram of vibration velocity and vibration displacement at test point 1.

FIGURE 23: Spectrum diagram of vibration velocity and vibration displacement at test point 2.

FIGURE 24: Spectrum diagram of vibration velocity and vibration displacement at test point 3.
It can be seen from Figure 17 that the nonlinearity of the vibration isolator has a very obvious impact on the vibration isolation system. After being acted by the external force, the time required for the nonlinear vibration isolation system with flexible foundation to reach stability is shorter.

4. Test Conditions

4.1. Test Equipment. Table 2 is the name, size, and manufacturer of the equipment and instruments used in this test. The test equipment and instruments used this time are all calibrated and valid.
4.2. Sensor Arrangement. According to the design plan, the motor and the refiner are placed on the steel structure vibration isolation platform, as shown in Figure 18. At the same time, Figure 18 shows the installation direction of the triaxial Accelerometer. The direction of the arrow is the positive direction. The red dot in the figure indicates that the positive direction points out of the paper. The mounting orientation of the sensors is the same.

4.3. Parameters of the Refiner. The motor of the tested refiner is a three-phase asynchronous motor produced by Xi’an Tai-fuxima Motor Co., Ltd. The mill is SC380 cylindrical refiner produced by Shanghai Shangding Machinery Technology Co., Ltd. Their main parameters are shown in Table 3 below.

4.4. Measurement Conditions. During the test, the refiner was at the rated speed and was in a stable condition of feeding.
5. Measurement Position

This test is carried out in strict accordance with the relevant standards. A total of 12 measuring points are set up during the test, and the measurement is divided into two times. Measure one side of the device at a time. At the same time, six triaxial acceleration sensors are installed on the steel structure platform and the corresponding position of the vibration isolator on the ground. As shown in Figures 19 and 20. The figure 21 shows the test-site.

6. Measurement Results

6.1. Vibration of Refiner and Floor. Conduct vibration tests on the refiner and the ground, and obtain the vibration

![Figure 31: Spectrum diagram of vibration velocity and vibration displacement at test point 10.](image1)

![Figure 32: Spectrum diagram of vibration velocity and vibration displacement at test point 11.](image2)

![Figure 33: Spectrum diagram of vibration velocity and vibration displacement at test point 12.](image3)
velocity and vibration displacement of each test point at 10–1,000 Hz, as shown in Figures 22–33.

The test points 1, 3, 5, 7, 9, and 11 on the vibration isolation platform and the test points 2, 4, 6, 8, 10, and 12 on the floor will be compared, respectively. It can be seen that the vibration of the vibration isolation platform is obviously larger than that of the floor. This shows that the nonlinear vibration isolation system has a good vibration isolation effect.

The floor is the base on which the refiner is installed. The vibration of the floor is evaluated according to the permissible vibration standard of construction engineering. From the
vibration displacement diagram of the floor in the time domain in Figure 34, it can be seen that the peak value of the vertical vibration displacement is 0.01 mm, which is within the standard allowable range. Compared with the peak value of vertical vibration displacement of 0.22 mm before the transformation, the drop rate reaches 95.45%. According to relevant standards, the peak value of vertical vibration shall not exceed 0.1 mm.

During the test, other equipment on the factory floor was also running. It can be seen from the vibration coherence between the platform and the ground in the Z direction in Figures 35–40 that not all the vibration of the floor is caused by the transformation equipment. Therefore, the nonlinear vibration isolation platform plays a very significant role in the vibration reduction of the equipment.

7. Conclusion

(1) After being subjected to external force, the free vibration of the rigid linear vibration isolation system attenuates slowly, and the stabilization process takes a long time.

(2) By adjusting the stiffness and damping of the vibration isolation system, the nonlinear vibration isolation platform can effectively isolate the vibration of the equipment and block the transmission of vibration. Therefore, the nonlinear vibration isolation platform has good vibration isolation effect and broad application prospects.

(3) After using the nonlinear vibration isolation platform, the peak value of the vertical vibration of the floor is 0.01 mm, which meets the requirements of relevant standards. Compared with the nonlinear vibration isolation platform, the peak vertical vibration of the floor is reduced by 95.45%.

Meantime, due to limitations in experimental conditions, there are still some shortcomings in the research. This research only considered the impact of nonlinear stiffness on the system and did not analyze nonlinear damping. Further research will be conducted in the future.

Data Availability

The data used to support the findings of this study are included within the article.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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