DYNAMICS ANALYSIS, SELECTION AND CALCULATION ON THE PARAMETERS OF A ROTARY VIBRATING SCREEN

Jing Jiang, Shuying Liu and Bangchun Wen

School of Mechanical Engineering and Automation, Northeastern University, Shenyang, P.R. China

E-mail: jjiang@mail.neu.edu.cn; shuyliu45@163.com; bcwen1930@mail.neu.edu.cn

IMETI 2015 Q1038_SCI

No. 16-CSME-13, E.I.C. Accession Number 3899

ABSTRACT

This paper introduces the structural characteristics and screening principle of rotary vibrating screen, and establishes nonlinear equations of rotary vibrating screen in view of the material effects. An approximate solution, material combination coefficient, and material drag coefficient for the nonlinear equations are solved by equivalent linearization method based on nonlinear theory. The selection and calculation on main parameters and a practical calculation example are carried out. It is of guiding significance and reference value for the design of this kind of machinery.

Keywords: rotary vibrating screen; nonlinear force; nonlinear equation; equivalent linearization method; dynamics analysis.

ANALYSE DYNAMIQUE, SÉLECTION ET CALCUL DES PARAMÈTRES D'UN CRIBLE ROTATIF VIBRANT

RÉSUMÉ

Dans cet article on présente les caractéristiques de la structure et les principes de tamisage d'un crible rotatif vibrant, et on a établi les équations nonlinéaires d'un crible rotatif vibrant d'un point de vue des effets sur les matériaux. Une solution approximative, le coefficient de combinaison de matériau, le coefficient de granulat sont résolus pour les équations nonlinéaires par la méthode de linéarisation équivalente basé sur la théorie de nonlinéarité. La sélection et le calcul sur les principaux paramètres et un exemple de calcul pratique sont effectués. Ce sont des principes significatifs et des valeurs de référence pour la conception de ce type de machinerie.

Mots-clés : crible rotatif vibrant; force nonlinéaire; équation nonlinéaire; méthode de linéarisation équivalente; analyse dynamique.

1. INTRODUCTION

The rotary vibrating screen is widely used in mining, coal, petroleum, chemical, electric power, food, environmental protection, and medical departments, and so on [1–3]. In ore-dressing plant, the rotary vibrating screen, instead of the double spiral classifier, is applied for the classification according to the grinding granularity, to improve the concentrate grade. The rotary vibrating screen is also used to classify the tailings of selected scenes to enhance the recovery rate of concentrate. In coal preparation plant, the rotary vibrating screen has solved the problem of plugging holes for the difficultly screened fine coal with water content 7 to 14% in the process of wet sieving and increased the screening efficiency. In the field of salt manufacturing and food processing, etc., the screening operation is usually used for purification treatment of materials. In chemical engineering department, the rotary vibrating screen is applied for the screening of chemical raw materials and products, and for the classification of chemical fertilizer and compound fertilizer as well. In food sector, the rotary vibrating screen can be applied in the purification treatment of food, and so on. From the above, the rotary vibrating screen is the key equipment in various fields [4–8]. Therefore, the design and study on rotary vibrating screen is remarkably important.

The screens are generally regarded as linear vibration system in the process of design on similar vibrating screens [9]. But as a matter of fact, the materials perform throwing motion on the screen surface during the working process of vibrating screen. In a periodic vibration, the materials sometimes contact with screen surface. During this time, the materials and the screen surface have the same acceleration. When the acceleration of screen surface is greater than a certain value, the materials will leave machine body to show throwing movement; after a certain time, the materials fall back on the machine body and impact on it, and then move with machine body together. In this process, the materials produce all kinds of nonlinear forces acting on the working machine body, such as interval inertial force, impact force and interval friction force (interval refers to a certain time interval in a cycle), etc. [10–12]. In order to make accurate analysis on vibration system, the influence of various nonlinear forces of materials on the vibration of machine should be considered. This paper introduces the structural characteristics and screening principle of rotary vibrating screen, and establishes the nonlinear equations for rotary vibrating screen in view of the material effects. An approximate solution, material combination coefficient, and material drag coefficient for the vibration equations are solved by equivalent linearization method based on nonlinear theory. The selection and calculation for main parameters and a practical calculation example are carried out. It is of guiding significance and reference value for the design of this kind of machinery.

2. STRUCTURAL CHARACTERISTICS AND SCREENING PRINCIPLE

The structure of rotary vibrating screen is shown in Fig. 1. It is made up of screen box cover 1, round screen mesh 2, rubber ball 3, ball tray 4, screen box 5, vibration isolation spring 6, motor base 7, vibration motor 8, screen frame 9, etc.

The machine is equipped with a variety of screen surfaces with different mesh sizes so that it can sieve dry, light and thin materials, can screen wet, heavy and thin materials, can also carry out dehydration and other solid-liquid separation operations. The screened materials enter the central part of screen surface from the material inlet of screen box cover. The materials can be quickly spread onto the whole screen surface to cause the material layer thinning and the increased opportunities for the contact between material particles and screen surface. At the same time, the materials make circulation movement on the screen surface to prolong their movement distance on screen surface, which is equivalent to lengthen the length of screen surface, increase the screening time, and thus improve the screening efficiency. The rotary pendulum vibration of machine can speed up the materials to become loose, tumbling and stratification to increase screening efficiency. Because the materials move in a helical direction towards periphery, the material outlet can be arbitrarily arranged along the circle tangent. The smooth discharge can improve the processing ability

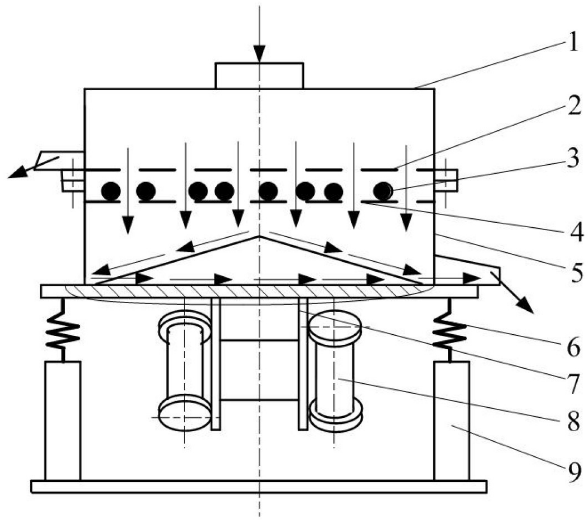

Fig. 1. Structure of rotary vibrating screen.

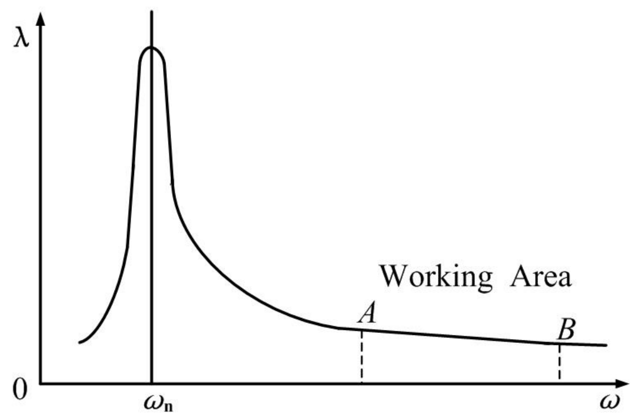

Fig. 2. Amplitude-frequency response curve of rotary vibrating screen.

of the machine. The machine has the features of easy manufacture, convenient installation and disassembly, and whole machine closure without polluting environment. The machine adopts the hole cleaning rubber ball colliding the screen surface constantly to clean the mesh, avoiding the blocking during the working process and improving the penetrating ability of materials and screening efficiency. The machine adopts the composite spring with the advantages of metal spiral spring and rubber spring, having good vibration isolation performance and small noise to meet the requirements of environmental protection. The machine works in the state far beyond resonance. As shown in Fig. 2, the working amplitude is stable in AB area in amplitude-frequency response curve.

Due to the axis of two vibration motors and the vertical axis of machine installed at an angle of cross, so when the two vibration motors execute synchronous reverse rotation, the machine present the rotating pendulum vibration under the combined action of the exciting force in the vertical direction and the reciprocating rotary vibration torque along circle tangent. The rubber balls bounce constantly to clear the screen surface so as to ensure the normal work for screening or dehydration. Materials are added in from the center of circular screen surface. The rotary pendulum vibration of screen surface makes the materials move

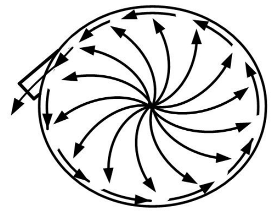

Fig. 3. Movement locus of materials on screen surface.

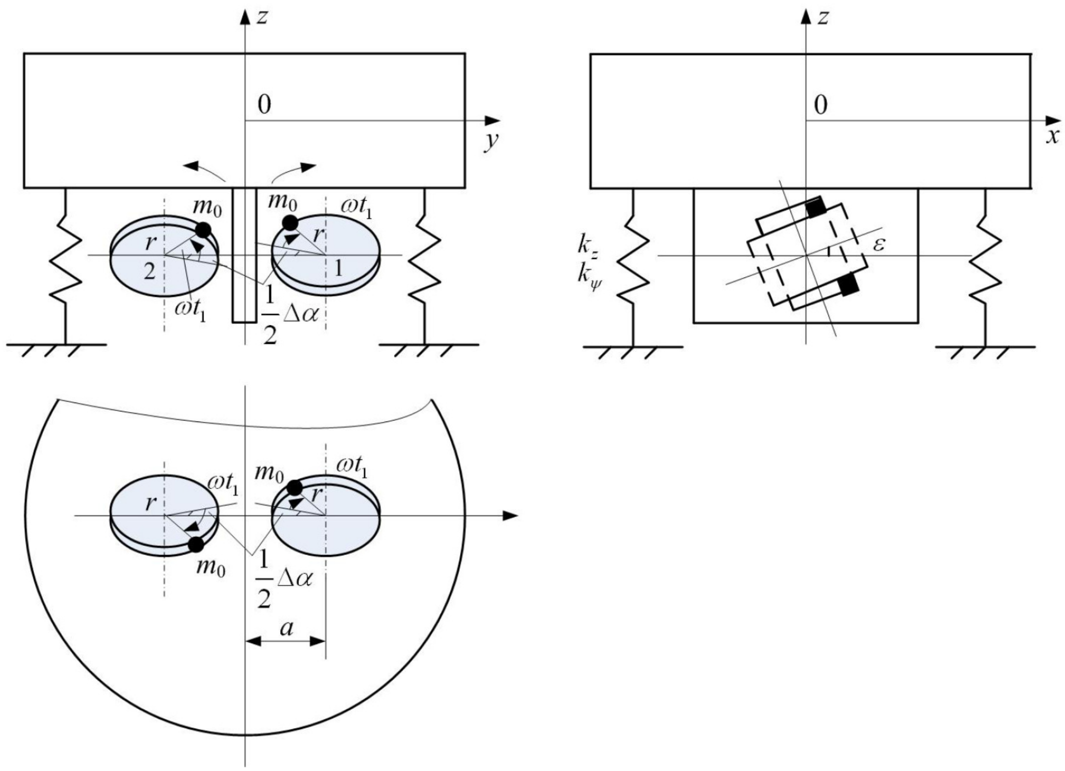

Fig. 4. Mechanical model of rotary vibrating screen.

towards the periphery in a spiral direction, as shown in Fig. 3. At the same time, the screened materials are thrown up to become loose and layered. When the materials fall back onto screen surface, the materials with the size less than sieve pore will go freely through the screen mesh, fall onto the cone bottom, move towards the periphery, and discharge from the fine material outlet (as shown in Fig. 1). The materials on screen surface move towards the periphery in a spiral direction and discharge from the coarse material outlet so as to realize the classification or solid-liquid separation.

3. DYNAMICS ANALYSIS

3.1. Establishment of Nonlinear Equations

The mechanical model of rotary vibrating screen is shown in Fig. 4. When the phase difference angle between the two eccentricblocks of two vibration motors is $\Delta\alpha = 0$, the machine can be simplified into a vibration system with two degrees of freedom under the condition of meeting synchronization, namely the vertical vibration along Z axis and the torsional vibration around Z axis.

The exciting force along Z axis direction is

$$F_z(t_1) = 2m_0 r\omega^2 \cos\varepsilon \sin\omega t_1 \quad (1)$$

The excitation torque around Z axis direction is

$$M_z(t_1) = 2m_0 r\omega^2 a \sin\varepsilon \sin\omega t_1 \quad (2)$$

The nonlinear equations of rotary vibrating screen are

$$m_j\ddot{z} + c_z\dot{z} + F_m(\ddot{z},\dot{z},z,t) + k_z z = 2m_0 r\omega^2 \cos\varepsilon \sin\omega t_1 \quad (3)$$

$$J_j\ddot{\psi} + c_\psi\dot{\psi} + M_m(\ddot{\psi},\dot{\psi},\psi,t) + k_\psi\psi = 2m_0 r\omega^2 a \sin\varepsilon \sin\omega t_1 \quad (4)$$

where m_j is the mass of vibration body (including eccentric block); J_j is the rotary inertia of vibration body (including eccentric block) to Z axis; c_z, c_ψ are the equivalent drag coefficients of movement and rotation of vibration body; k_z, k_ψ are the spring stiffness along Z axis and around Z axis; $z, \dot{z}, \ddot{z}$ are respectively the displacement, the velocity and the acceleration of machine body along Z axis; $\psi, \dot{\psi}, \ddot{\psi}$ are respectively the angular displacement, the angular velocity and the angular acceleration of machine body around Z axis; m_0 is the mass of eccentric block on exciter shaft; r is the distance from centroid of eccentric block to rotation axis; ε is the angle between axis of exciter and horizontal plane; a is the half distance between the axes of two exciters; ω is the rotating angular velocity of exciter; $F_m(\ddot{z},\dot{z},z,t)$ is the nonlinear force of materials in Z direction; $M_m(\ddot{\psi},\dot{\psi},\psi,t)$ is the nonlinear torque of materials around Z axis.

The nonlinear force of materials in Z direction is $F_m(\ddot{z},\dot{z},z,t)$. It equals to zero when the materials are thrown off; it turns into $m_m(\ddot{z}+g)$ when the materials move with machine body together. The instantaneous impact force is $F_m(\ddot{z},\dot{z},z,t)$ when the materials fall back. Therefore, the nonlinear force of materials in Z direction $F_m(\ddot{z},\dot{z},z,t)$ can be expressed in Eq. (5)

$$F_m(\ddot{z},\dot{z},z,t) = \begin{cases} 0 & \varphi_d < \varphi < \varphi_z \\ m_m(\ddot{z}+g) & \varphi_z - 2\pi + \Delta\varphi \le \varphi \le \varphi_d \\ m_m(\dot{z}_m - \dot{z}_z)/\Delta t & \varphi_z \le \varphi \le \varphi_z + \Delta\varphi \end{cases} \quad (5)$$

where φ_d, φ_z are the throw starting angle and the throw ending angle; $\dot{z}_m, \dot{z}_z$ are the vertical velocity of materials at drop point and the instantaneous vertical velocity of machine body at the same moment; $\dot{z}_m = \lambda_z\omega\cos\varphi_d - g(\varphi_z - \varphi_d)/\omega$; Δt is the fall and impact time of materials; $\Delta\varphi = \omega\Delta t$, compared with the vibration period, the time is very short, $\Delta\varphi \to 0; \varphi$ is the vibration phase angle.

The nonlinear torque of materials around Z axis is $M_m(\ddot{\psi},\dot{\psi},\psi,t)$. It equals to zero when the materials are thrown off; it turns into $J_m\ddot{\psi}$ when the materials move with machine body together. The torque of instantaneous impact force to Z axis is $J_m(\dot{\psi}_m - \dot{\psi}_z)/\Delta t$ when the materials fall back. The nonlinear torque of materials around Z axis can be expressed in Eq. (6)

$$M_m(\ddot{\psi},\dot{\psi},\psi,t) = \begin{cases} 0 & \varphi_d < \varphi < \varphi_z \\ J_m\ddot{\psi} & \varphi_z - 2\pi + \Delta\varphi \le \varphi \le \varphi_d \\ J_m(\dot{\psi}_m - \dot{\psi}_z)/\Delta t & \varphi_z \le \varphi \le \varphi_z + \Delta\varphi \end{cases} \quad (6)$$

where $\dot{\psi}_m, \dot{\psi}_z$ are respectively the angular velocity of materials and the angular velocity of machine body in the touch moment between materials and machine.

3.2. An Approximate Solution of Nonlinear Equations

Suppose that an approximate solution of the equation is

$$
\begin{aligned}
z &= \lambda_z \sin(\omega t_1 - \alpha_z) = \lambda_z \sin\varphi_v \quad \varphi_v = \omega t_1 - \alpha_z \\
\psi &= \theta_\psi \sin(\omega t_1 - \alpha_\psi) = \theta_\psi \sin\varphi_u \quad \varphi_u = \omega t_1 - \alpha_\psi
\end{aligned} \tag{7}
$$

The nonlinear force is expanded by the Fourier series expansion, then

$$
\begin{aligned}
F_m(\ddot{z}, \dot{z}, z, t) &= \frac{a_{0z}}{2} + \sum_{n=1}^{\infty} (a_{nz} \cos n\varphi_v + b_{nz} \sin n\varphi_v) \\
M_m(\ddot{\psi}, \dot{\psi}, \psi, t) &= \frac{a_{0\psi}}{2} + \sum_{n=1}^{\infty} (a_{n\psi} \cos n\varphi_u + b_{n\psi} \sin n\varphi_u)
\end{aligned} \tag{8}
$$

For the general nonlinear vibration system, a harmonic force according to the Fourier series expansion is much larger than two or more times of the harmonic force, so the harmonic force of far more than two times can be omitted. Due to the study on the dynamic state of machine body, the constants only affecting static state will not be considered, the nonlinear force can be approximatively expressed as

$$
\begin{aligned}
F_z(\ddot{z}, \dot{z}, z, t) &\approx a_{1z} \cos\varphi_v + b_{1z} \sin\varphi_v \\
M_\psi(\ddot{\psi}, \dot{\psi}, \psi, t) &\approx a_{1\psi} \cos\varphi_u + b_{1\psi} \sin\varphi_u
\end{aligned} \tag{9}
$$

For most of the vibrating machines, $a_z \approx \alpha_\psi$ so $\varphi_v \approx \varphi_u = \varphi$, then there is

$$
\begin{aligned}
F_z(\ddot{z}, \dot{z}, z, t) &\approx a_{1z} \cos\varphi + b_{1z} \sin\varphi \\
M_\psi(\ddot{\psi}, \dot{\psi}, \psi, t) &\approx a_{1\psi} \cos\varphi + b_{1\psi} \sin\varphi
\end{aligned} \tag{10}
$$

According to Euler–Fourier formula, the coefficients a_{1z}, b_{1z} and $a_{1\psi}, b_{1\psi}$ can be calculated as follows:

$$
\begin{aligned}
a_{1z} &= \frac{1}{\pi} \int_{-\pi}^{\pi} F_m(\ddot{z}, \dot{z}, z, t) \cos\varphi \mathrm{d}\varphi \\
&= \frac{m_m \lambda_z \omega^2}{\pi} \left[\int_{\varphi_z - 2\pi}^{\varphi_d} \left(\frac{g}{A_z \omega^2} - \sin\varphi \right) \cos\varphi \mathrm{d}\varphi + \int_{\varphi_z}^{\varphi_z + \Delta\varphi} (\theta_d \sin\varphi_d - \cos\varphi_d + \cos\varphi_z) \frac{1}{\Delta\varphi} \cos\varphi \mathrm{d}\varphi \right] \\
&= \frac{m_m \lambda_z \omega^2}{\pi} \left[\left(\frac{g}{\lambda_z \omega^2} \sin\varphi - \frac{1}{2} \sin^2\varphi \right) \Big|_{\varphi_z - 2\pi}^{\varphi_d} + (\theta_d \sin\varphi_d - \cos\varphi_d + \cos\varphi_z) \cos\varphi_z \right]
\end{aligned} \tag{11}
$$

$$
\begin{aligned}
b_{1z} &= \frac{1}{\pi} \int_{-\pi}^{\pi} F_m(\ddot{z}, \dot{z}, z, t) \sin\varphi \mathrm{d}\varphi \\
&= \frac{m_m \lambda_z \omega^2}{\pi} \left[\int_{\varphi_z - 2\pi}^{\varphi_d} \left(\frac{g}{\lambda_z \omega^2} - \sin\varphi \right) \sin\varphi d\varphi + \int_{\varphi_z}^{\varphi_z + \Delta\varphi} (\theta_d \sin\varphi_d - \cos\varphi_d + \cos\varphi_z) \frac{1}{\Delta\varphi} \sin\varphi \mathrm{d}\varphi \right] \\
&= \frac{m_m \lambda_z \omega^2}{\pi} \left\{ \left[-\frac{g}{\lambda_z \omega^2} \cos\varphi - \frac{1}{2} \left(\varphi - \frac{1}{2} \sin 2\varphi \right) \right] \Big|_{\varphi_z - 2\pi}^{\varphi_d} + (\theta_d \sin\varphi_d - \cos\varphi_d + \cos\varphi_z) \sin\varphi_z \right\}
\end{aligned} \tag{12}
$$

Therein, $\theta_d = \varphi_z - \varphi_d$.

$$
\begin{aligned}
a_{1\varphi} &= \frac{1}{\pi}\int_{-\pi}^{\pi} M_m(\ddot{\psi},\dot{\psi},\psi,t)\cos\varphi \mathrm{d}\varphi \\
&= \frac{J_m\theta_\psi\omega^2}{\pi}\left[\int_{\varphi_z-2\pi}^{\varphi_d} -\sin\varphi\cos\varphi \mathrm{d}\varphi + \int_{\varphi_z}^{\varphi_z+\Delta\varphi}(-\cos\varphi_d+\cos\varphi_z)\frac{1}{\Delta\varphi}\cos\varphi d\varphi\right] \\
&= \frac{J_m\theta_\psi\omega^2}{\pi}\left[\left(-\frac{1}{2}\sin^2\varphi\right)\Big|_{\varphi_z-2\pi}^{\varphi_d} + (-\cos\varphi_d+\cos\varphi_z)\sin\varphi_z\right]
\end{aligned}
\tag{13}
$$

$$
\begin{aligned}
b_{1\varphi} &= \frac{1}{\pi}\int_{-\pi}^{\pi} M_m(\ddot{\psi},\dot{\psi},\psi,t)\sin\varphi \mathrm{d}\varphi \\
&= \frac{J_m\theta_\psi\omega^2}{\pi}\left[\int_{\varphi_z-2\pi}^{\varphi_d} -\sin^2\varphi \mathrm{d}\varphi + \int_{\varphi_z}^{\varphi_z+\Delta\varphi}(-\cos\varphi_d+\cos\varphi_z)\frac{1}{\Delta\varphi}\sin\varphi \mathrm{d}\varphi\right] \\
&= \frac{J_m\theta_\psi\omega^2}{\pi}\left[-\frac{1}{2}\left(\varphi-\frac{1}{2}\sin 2\varphi\right)\Big|_{\varphi_z-2\pi}^{\varphi_d} + (-\cos\varphi_d+\cos\varphi_z)\sin\varphi_z\right]
\end{aligned}
\tag{14}
$$

Eq. (9) is substituted into Eqs. (3) and (4) to gain

$$m_j\ddot{z} + c_z\dot{z} + a_{1z}\cos\varphi_v + b_{1z}\sin\varphi_v + k_z z = 2m_0 r\omega^2\cos\varepsilon\sin\omega t_1 \tag{15}$$

$$J_j\ddot{\psi} + c_\psi\dot{\psi} + a_{1\psi}\cos\varphi_u + b_{1\psi}\sin\varphi_u + k_\psi\psi = 2m_0 r\omega^2 a\sin\varepsilon\sin\omega t_1 \tag{16}$$

Substituting

$$\varphi_v = -\frac{\ddot{z}}{\lambda_z\omega^2} \quad \text{and}$$

$$\cos\varphi_v = \frac{\dot{z}}{\lambda_z\omega}$$

into Eq. (15), and substituting

$$\sin\varphi_u = -\frac{\ddot{\psi}}{\theta_\psi\omega^2} \quad \text{and} \quad \cos\varphi_u = \frac{\dot{\psi}}{\theta_\psi\omega}$$

and into Eq. (16), the following equivalent linear equations can be gotten by simplifying sorting

$$\left(m_j - \frac{b_{1z}}{A_z\omega^2}\right)\ddot{z} + \left(c_z + \frac{a_{1z}}{A_z\omega}\right)\dot{z} + k_z z = 2m_0 r\omega^2\cos\varepsilon\sin\omega t_1 \tag{17}$$

$$\left(J_j - \frac{b_{1\psi}}{\theta_\psi\omega^2}\right)\ddot{\psi} + \left(c_\psi + \frac{a_{1\psi}}{\theta_\psi\omega}\right)\dot{\psi} + k_\psi\psi = 2m_0 r\omega^2 a\sin\varepsilon\sin\omega t_1 \tag{18}$$

Eq. (7) is substituted into Eqs. (17) and (18) after second derivative to get the equivalent linearization amplitude λ_z, the equivalent linear vibrating angle θ_ψ and the equivalent linear phase difference angles α_z and α_ψ are as follows:

$$\lambda_z = \frac{2m_0 r\omega^2\cos\varepsilon\cos\alpha_z}{k_z - \left(m_j - \frac{b_{1z}}{m_m\lambda_z\omega^2}m_m\right)\omega^2}; \quad \alpha_z = \arctan\frac{\left(c_z + \frac{a_{1z}}{\lambda_z\omega}\right)\omega}{k_z - \left(m_j - \frac{b_{1z}}{m_m\lambda_z\omega^2}m_m\right)\omega^2} \tag{19}$$

$$\theta_\psi = \frac{2m_0 r\omega^2 a\sin\varepsilon\cos\alpha_\psi}{k_\psi - \left(J_j - \frac{b_{1\psi}}{J_m\theta_\psi\omega^2}J_m\right)\omega^2}; \quad \alpha_\psi = \arctan\frac{\left(c_\psi + \frac{a_{1\psi}}{\theta_z\omega}\right)\omega}{k_\psi - \left(J_j - \frac{b_{1\psi}}{J_m\theta_\psi\omega^2}J_m\right)\omega^2} \tag{20}$$

Arc length amplitude λ_ψ at screen surface radius R_c is given by

$$\lambda_\psi = R_c\theta_\psi \tag{21}$$

The resultant amplitude λ_c and vibrating direction angle β_c at screen surface radius R_c are respectively given by

$$\lambda_c = \sqrt{\lambda_z^2 + \lambda_\psi^2}, \quad \beta_c = \arctan\frac{\lambda_z}{R_c\theta_\psi} \tag{22}$$

So the combination coefficient and the drag coefficient of materials are respectively

$$K_{mz} = \frac{-b_{1z}}{m_m\lambda_z\omega^2}, \quad c_{mz} = \frac{a_{1z}}{\lambda_z\omega}, \quad K_{m\psi} = \frac{-b_{1\psi}}{J_m\theta_\psi\omega^2}, \quad c_{m\psi} = \frac{a_{1\psi}}{\theta_\psi\omega}$$

where the combination coefficient of materials $K_{mz}, k_{m\psi}$ and the drag coefficient of materials $c_{mz}, c_{m\psi}$ are two important parameters on dynamics computation of vibration system, which have close relationship with dynamic state and lifetime of designed vibrating machine.

4. DYNAMICS ANALYSIS

4.1. Selection of the Movement State of Materials

According to the process requirements and the material properties treated by rotary vibrating screen, material throwing motion state is selected. Then the throwing index D is given by Eq. (23)

$$D = \frac{n^2\pi^2\lambda_c\sin\beta_c}{900g} \tag{23}$$

where n is the vibration frequency of machine (r/min), β_c is the vibrating direction angle at screen surface radius $R_c(^\circ)$, g is the acceleration of gravity (9.81 m/s^2), other symbols can refer to the previous content.

4.2. Calculation of the Average Velocity of Material Movement

The average velocity of material movement can be calculated by Eq. (24)

$$v = \pi\omega\lambda_c i_D^2 C_h C_m C_w \cos\beta_c / D \tag{24}$$

where v is the actual average speed of material movement (m/s), i_D is the throw-away coefficient $(i_D = \theta_d/2\pi), \theta_d$, is throw-away angle; C_h is the material thickness influence coefficient, $C_h = 0.5 \sim 0.7$; C_m is the material properties influence coefficient, $C_m = 0.6 \sim 0.9$, powder materials take small value and granular materials take large value; C_w is sliding movement influence coefficient (refer to literature [2]), other symbols can refer to the previous content.

4.3. Calculation of the Vibrating Mass

The vibrating mass is composed by the vibrating mass of machine m_j (kg) and the vibrating mass of materials $K_m m_m$ (kg)

$$m = m_j + K_m m_m \tag{25}$$

where K_m is the combination coefficient of materials, $K_m = 0.25 \sim 0.4$.

4.4. Calculation of the Stiffness of Vibration Isolation Spring

The total vibration isolation spring stiffness $\sum k$ is given by

$$\sum k = \frac{m\omega^2}{z^2} \tag{26}$$

The vibration isolation spring stiffness for each is:

$$k = \sum k/i \tag{27}$$

where z is the frequency ratio ($z = \omega/\omega_n = 3 \sim 6$); ω is the exciting frequency; ω_n is the natural frequencies of system; i is the number of vibration isolation spring.

4.5. Selection of the Vibration Motor

The required exciting force F is represented as follow when the machine works normally.

$$F = m\omega^2\lambda_c \tag{28}$$

The rotary vibrating screen adopts two vibration motors with same specifications and features, so the half of calculated exciting force will be consulted to choose the two vibration motors with same specifications and features.

5. CALCULATION EXAMPLE

The diameter of rotary vibrating screen is supposed to be 1000 mm, with two vibration motors crossing installed, vibrating direction angle $\beta = 50°$, amplitude $\lambda_c \approx 0.314$ cm, exciting frequency $\omega = 100.5$ rad/s, and spring stiffness $k_z \ll m_j\omega^2, k_\psi \ll J_j\omega^2$, the combination coefficient and the equivalent drag coefficient of materials are tried to be calculated out.

According to the known conditions, the value of throw starting angle φ_d and θ_d can be calculated as Eq. (29)

$$\varphi_d = \arcsin\frac{g}{\lambda_c\omega^2\sin\beta_c} = \arcsin\frac{981}{0.314\times(100.5)^2\sin 50} = 23.82° \tag{29}$$

The throwing index is $D = 1/\sin\varphi_d = 1/\sin 23.82° = 2.48$. According to the throwing index value D, the throw-away coefficient can be looked up to be $i_D = 0.86$. So the throw-away angle φ_d can be obtained as Eq. (30)

$$\theta_d = 2\pi i_D = 360° \times 0.86 = 309.6° \tag{30}$$

The throw ending angle φ_z can be gotten from the throw starting angle φ_d and the throw-away angle θ_d as follow

$$\varphi_z = \varphi_d + \theta_d = 23.82 + 309.6 = 333.42° \tag{31}$$

Substituting the values of φ_d, φ_z and θ_d into Eqs. (12) and (14), b_{1z} and $b_{1\psi}$ can be calculated out as given by

$$b_{1z} = -0.328 m_m\lambda_z\omega^2, \quad b_{1\psi} = -0.0146 J_m\theta_\psi\omega^2 \tag{32}$$

The combination coefficients K_{mz} and $K_{m\psi}$ are respectively 0.328 and 0.0146. The composite combination coefficient K_m is

$$\begin{aligned} K_m &= \frac{K_{mz}m_m\lambda_z\omega^2\sin\beta_c + K_{m\psi}J_m\theta_\psi\omega^2\cos\beta_c/R_c}{m_m\lambda_c\omega^2} = K_{mz}\sin^2\beta_c + K_{m\psi}\cos^2\beta_c \\ &= 0.328\sin^2 50° + 0.0146\cos^2 50° = 0.198 \end{aligned} \tag{33}$$

where $J_m = m_m R_c^2, \lambda_\psi = R_c \theta_\psi, R_c$ is the equivalent radius of working surface, λ_ψ is the tangential amplitude at equivalent radius R_c.

Substituting the values of φ_d, φ_z and θ_d into Eqs. (11) and (13), a_{1z} and $a_{1\psi}$ can be calculated out to be $a_{1z} = 0.72 m_m A_z \omega^2, a_{1\psi} \approx 0$. The drag coefficient c_{mz} and $c_{m\psi}$ of materials are respectively $0.72 m_m \omega$ and 0, the equivalent drag coefficient c_m induced to composite vibration direction is

$$c_m = c_{mz} \sin^2 \beta_c + c_{m\psi} \cos^2 \beta_c = 0.74 m_m \omega \sin^2 50^\circ + 0 = 0.42 m_m \omega \tag{34}$$

It is concluded from the calculating results that about 20% of materials are involved in vibration. The nonlinear forces of these materials will have a certain influence on the strength of parts of vibrating screen, which must be considered in the calculation of dynamic parameters.

6. CONCLUSIONS

1. The rotary vibrating screen is a kind of high efficient equipment for the screening classification of dry and wet fine materials. It has the features of simple and compact structure, convenient installation and disassembly, stable work operation, no sieve pore blocking, high screening efficiency, and whole machine closure without polluting the environment.

2. The nonlinear forces of materials, including interval inertial force and impact force, will have certain impact on the dynamic state of inertial rotary vibrating screen and the lifetime of parts. Therefore, the vibrating machine considering the nonlinear forces of materials on design will obtain the best dynamic state and the service life.

3. The values of $b_{1\psi}$ and $a_{1\psi}$ are close to zero by calculation and can be omitted compared with b_{1z} and a_{1z}. Therefore, for the influence of combination coefficient and drag coefficient of materials on the movement of screen, only Z direction can be considered.

4. The equivalent linearization method is an effective way to deal with two degree of freedom nonlinear system under the effect of periodic excitation.

ACKNOWLEDGEMENTS

The study is supported by "the Fundamental Research Funds for the Central Universities" (N120403001) and "Scientific Research General Projects from Department of Education of Liaoning Province" (L2012075).

REFERENCES

1. Wen, B.C., Zhang, H., Liu, S.Y., He, Q. and Zhao, C.Y., *Theory and Techniques of Vibrating Machinery and Their Applications*, China Science Press, 2010.
2. Wen, B.C., Liu, S.Y. and He, Q., *Theory and Dynamic Design Method on Vibrating Machinery*, China Machine Press, 2001.
3. Wen, B.C. and Liu, S.Y., *Modern Vibration Screening Technology and Equipment Design*, Metallurgical Industry Press, 2013.
4. Wen, B.C., Liu, S.Y. and Chen, Z.B., *Theory and Application of Mechanical Vibration*, Higher Education Press, 2009.
5. Liu, S.Y., Yuan, Y. and Wen, B.C., "Dynamic analysis on vibration conveyor with large inclination considering material forces", *Journal of Northeastern University*, Vol. 3, pp. 315–318, 1996.
6. Wen, B.C., *Modern mechanical designer manual (Volume two)*, China Machine Press, 2012.
7. Wen, B.C., Liu, S.Y. and Zhang C.Y., *Mechanical Vibration Theory*, Metallurgical Industry Press, 2011.
8. Chen, Y.S., *Nonlinear Vibration*, Tianjin Science and Technology Press, 1983.
9. Wen, B.C. and Liu F.Q., *Theory and Application on Vibrating Machinery*, China Machine Press, 1982.

10. Jiang, J., Zhu, T., Xie,Y.H., Han, J. and Wang, L., “Research and application on vibrating style drier with conduction and heat transfer”, *Applied Mechanics and Materials*, Vol. 16–19, pp. 224–228, 2009.
11. Liu, S.Y., Yuan, Y. and Wen, B.C., “Dynamic analysis of the vibrating conveyer with large inclination when considering material forces”, in *Proceedings of Asia-Pacific Vibration Conference*, Kuala Lumper, Malaysia, pp. 337–342, November 26–30, 1995.
12. Jiang, J., Xie, Y.H., Han, J. and Wang, L., “Research on vibrating spin-drier of two-way and half-spiral”, in *Proceedings International Conference on Vacuum Metallurgy and Surface Engineering*, Shenyang, China, pp. 69–73, August 24–27, 2009.