DESIGN AND RESEARCH INTO THE NONLINEAR MAIN VIBRATION SPRING IN DOUBLE-MASS HIGH ENERGY VIBRATION MILLING

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Abstract:

Due to the shortcomings of one - mass vibration mill such as inefficiency, high energy consumption and big noise, a double - mass high energy vibration mill, in which transient high vibration intensity is produced, is investigated by applying the non - linear vibration theory. The nonlinear hard - feature variable-pitch spring i0s used in the main vibration system which has the characteristic of the stiffness that can be varied along with the dynamic load. In this way, the goals of operation stabilization and energy saving will be achieved. Results from the field test show that the efficiency is obviously improved, i.e. a 28% increase in the vibration intensity, 10% decrease in energy consumption and 4% decrease in noise. That verifies the correctness of the main vibration system construction. This system can be used by others as a reference design for this field.

1 Introduction

Vibration mill, also known as a reactor, is the typical mass - energy conversion equipment which can convey vibration energy into particulate matter. This one piece equipment can carry out everything from particulate matter refining and reuniting to modifying. Therefore, it has broad application prospects in the industry [1-2]. Currently, the most commonly adopted main vibration spring design in a double - mass structure is employing an equal - pitch cylindrical helix spring which is of a linear type. The reason for this is due to the easiness of linear spring in design and production. The linear spring stiffness cannot be changed along with the dynamic load, so linear spring cannot save more energy than that with nonlinear spring. Based on the existing problems, the experience from actual production and the new nonlinear technology theory and the construction of nonlinear vibration dryer and nonlinear vibration screener, we come up with adopting the hard - feature variable - pitch cylindrical helical spring in the vibration mill design to take a full advantage of the nonlinear vibration theory [3-4].

2 The design of the nonlinear main vibration system in double - mass high energy vibration mill

The structure of the double-mass high energy vibration mill is shown in Fig. 1. The upper mass is

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driven by the centrifugal force, which is produced by the rotating eccentric block in the vibration motor. The power is passed on to the grinding cylinder which contains abrasives and stainless steel balls with the diameter of 4, 6, 8 and 10 mm respectively. The strong longitudinal vibration is produced by the cylinder which will create impact and friction between material and grinding medium like steel balls. In this way, the goal of material smashing and fine grinding is achieved. The granularity and the productivity are controlled by adjusting the offset of the eccentric block and granular composition of the grind medium [5-7].

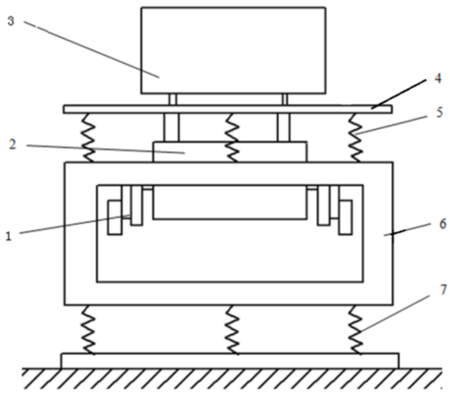

Figure 1. Structural representation of the vibration mill (1. eccentric block, 2. vibration motor, 3. cylinder, 4. mounting plate, 5. nonlinear spring, 6. lower mass, 7. vibration-isolation spring).

Comparing with linear spring, nonlinear spring has the following features [8-9]:

- The load and deformation that change in nonlinear can fit into the linear relation in terms of load and stiffness. That is to say, when the load is small, the stiffness of the system is small too, and when the load becomes larger, the stiffness of the system becomes larger accordingly. Therefore, the power drawn from the vibration source is reduced and the energy is therefore saved. It is especially suitable for variable-load system. The vibration mill is just a variable load system because of the material thrown up and down constantly.

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- According to reference [8], in the variable - load system, the natural frequency is set up as $\omega = \sqrt{k/m}$. When mass m changes and stiffness k remains unchanged, the ω will vary with m. If the k changes with m, then ω maintains in a constant which will make the system have a more stable vibration amplitude during operation.
- By adjusting preload of the nonlinear spring, the operating point of the system can be set easily to ensure steady operation near the point of resonance. The exciting force will be reduced which improves the system efficiency by consuming less power and having more powerful drive.

a. The choice of the nonlinear spring

The nonlinear spring has many types and the common ones are listed below [8, 10-11].

1) Unequal-pitch cylindrical spiral spring

When the spring is compressed to the point when neighboring coils begin to contact with each other, the characteristic line becomes nonlinear. The stiffness and the natural vibration frequency all come to variable which helps eliminate or mitigate the impact of the resonance. It is used to support high-speed variable load structure.

2) Taper spiral spring

When the spring is compressed to the point when neighboring coils begin to contact with each other, the characteristic line becomes nonlinear. The natural vibration frequency is variable. In comparison with the variable-pitch compress spring, the taper spiral spring ability of anti-resonance is stronger. Because of its compact structure and good stability, it is usually used to support large load and absorb shock.

3) Belleville spring

The Belleville spring was invented by J. Belleville, a French engineer, 100 years ago. The sharp of the Belleville spring is a conical disk. Compared with traditional spring, it has special features like large load, short stroke, little space, convenient use, easiness to install and maintain, and it is less expensive and safer. It is widely used in precision heavy machinery which has small space and large load. Its maximum compress stroke spans from 10 to 75%.

Acting as the spring to support the upper mass whose vibration is large, the variable-pitch cylindrical helical spring offers better features and is more suitable for being applicable as the main vibration spring of the vibration mill than other springs.

b. Design of the nonlinear main vibration spring

The overall performance of the vibration mill is decided by the non-linear design of the main vibration spring. The stiffness of the spring should be increased along with an increase in the load. Meanwhile, the stiffness of the spring should be identified as direct linear variation because it has a linear relationship to stabilize the system. The pitch of the variable-pitch cylindrical helical spring can be arranged in the order of small to large or can be placed large in the middle and small in ends. The latter is used in this design.

2.2.1 Condition and assumption of the field processing

According to the condition in the field, when the system is of no-load type, the total mass m including the upper mass, vibration exciter and cylinder is 120 kg. The mass m_1 of the material is about 20 kg. So, the total of upper mass is 140 kg.

In the field there are 6 equal - pitch springs whose original length is 130 mm. The length of the spring will be compressed to 116 mm due to the gravity of the upper mass (the deformation of the length is 14 mm). The expected vibration amplitude of the spring will be 16 and 3 mm. The maximum deformation of the spring is (14 + 16) mm when the spring is forced by maximum dynamic load (the maximum vibration displacement of the system). The deformation of the spring is 17 mm when the system is forced by the maximum static load, and the deformation of the spring is 14 mm when it is a not-load type.

Supposing that the non-linear characteristics line equation of the main vibration spring is:

$$F = af^2 + bf + c\,, \tag{1}$$

where, F is the load, N; f is the deformation, mm.

2.2.2 The nonlinear characteristic equation

Considering the load condition and the above-mentioned mass, when the system is of no - load type, the load on each spring is as follows:

$$F_0 = \frac{\mathrm{m}g}{6} = 196\ \mathrm{N}\,. \tag{2}$$

When the system is of full - load type, the load on each spring is expressed as:

$$F_1 = \frac{(m+m_1)g}{6} = 228.7\ \mathrm{N}\,. \tag{3}$$

Considering the additional load caused by the non - uniform load on each spring, taking non - uniform coefficient of load on each spring is 1.4, accordingly, the maximum static load on each spring is:

$$F_{\max 1} = 1.4F_1 = 320.1\ \mathrm{N}\,. \tag{4}$$

Taking 5 times of maximum static load as the maximum dynamic load:

$$F_{\max} = 5F_{\max 1} = 1600.5\ \mathrm{N}\,. \tag{5}$$

One can figure out that the 3 points of the non - linear curve is C_1 (14,196), C_2 (17,320.1) and C_3 (30, 1600.5). After inserting those into Equation (1), load deformation equation of the nonlinear hard spring will be:

$$\begin{cases} 196 = 196a + 14b + c \\ 320.1 = 289a + 17b + c \\ 1600.5 = 900a + 30b + c \end{cases}. \tag{6}$$

$$\text{Solution out} \begin{cases} a = 3.6 \\ b = -69.3 \\ c = 466.6 \end{cases}.$$

So, the load deformation equations of non - linear hard characteristics spring can be obtained as follows:

$$F = 3.6 f^2 - 69.3 f + 466.6. \qquad (7)$$

The fitted curve is shown in Fig. 2 based on Matlab simulation [12].

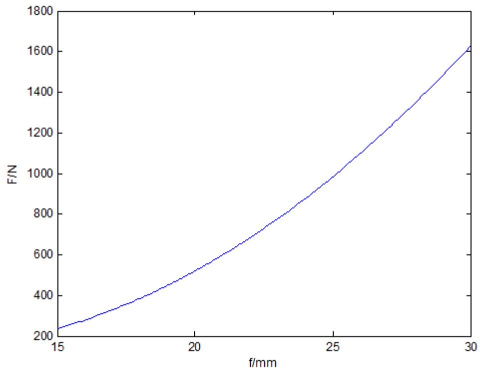

Figure 2. Load - deformation diagram of nonlinear hard characteristics spring.

2.2.3 Calculation of the nonlinear spring design

1) The shear stress of the spring

According to the known condition, alloy steel wire which has been oil-quenched and tempered is selected as the material for the spring. The spring parameters are as follows:

Shear modulus $G = 78800$ MPa;

Allowable shear stress;

$$[\tau] = (0.34 \sim 0.40)\sigma_b = (0.35 \sim 0.40) \times 1618$$
$$= 566.3 \sim 647.2 \text{ MPa},$$

The middle diameter of the spring $D = 45$ mm;

The diameter of the wire: $d = 8$ mm;

The curvature coefficient: $C = 1.15$.

Then,

$$\tau = C\frac{8D}{\pi d^3}F = 1.15 \times \frac{8 \times 45}{\pi \times 8^3} \times 1600.5$$
$$= 412 < [\tau] = 647.2 \text{ MPa}. \qquad (8)$$

2) The stiffness p' of the spring without load

Alterable-pitch cylindrical screw pressure spring has the same function of the inline combination of springs with different pitch; its stiffness p' is [8]:

$$p' = nF' = \frac{Gd^4}{8D^3} = \frac{78800 \times 8^4}{8 \times 45^3} \approx 442.8 \text{ N/mm}, \qquad (9)$$

where,

G is the shear modulus of the spring; n is the number of active coil of the spring; F' is the stiffness without load on spring. According to Equation (7), the relationship between the stiffness F' of the spring without load and the load F can be derived as:

$$F' = \frac{dF}{df} = 7.2 f - 69.3 = 7.2\sqrt{\frac{F - 133.1}{3.6}}. \qquad (10)$$

According to the reference [8]:

$$F' = \frac{F_0}{f_0} = \frac{140 \times 9.8}{3 \times 6} = 76 \text{ N/mm}, \qquad (11)$$

we can get,

$$n = \frac{P'}{F'} = \frac{442.8}{76} = 5.6 \approx 6\,. \qquad (12)$$

Taking a coil on each side of the spring as the support spring, there are a total of 8 coils.

3) The stiffness and the bearing load of the spring when the first coil was compressed near the second coil

The pitch of the spring is in the order that the large is in the middle and small at ends.

When the i the coil or the（$6-i$）the coil of the spring is compressed, neighboring coils will contact with each other by load F_i. The stiffness of the rest spring is:

$$\frac{1}{F'_i}=\frac{1}{F'}-\frac{i}{P'}. \tag{13}$$

So it can be obtained that:

$$\frac{1}{F_1'}=\frac{1}{F'}-\frac{1}{P'}=\frac{1}{76}-\frac{1}{442.8}, \tag{14}$$

$$F_1'=91.8\approx 92\ \mathrm{N/mm}. \tag{15}$$

According to Equation (10), we can conclude that:

$$F_1=737\ \mathrm{N/mm}. \tag{16}$$

The space between the first coil and the second coil can be obtained as follows:

$$\begin{aligned}\delta_1&=\frac{1}{2P'}(F_0+F_1)\\&=\frac{1}{2\times 442.8}(196+737)=1.05\ \mathrm{mm.}\end{aligned} \tag{17}$$

The pitch between the first coil and the second coil can be gotten as:

$$t_1=\delta_1+d=1.05+8=9.05\ \mathrm{mm}. \tag{18}$$

And so on, the spaces and pitches of result coils are listed in Table 1.

From Table 1, due to the maximum load $F_{\max}$ =1600.5 N<3325.85 N, the spring will have no contact with each other. So the system will not produce noise.

Moreover, the deformation and spacing of each coil has the following relationship:

$$f_i=\sum_{i=1}^{n}\delta_i+(n-i)\delta_i. \tag{19}$$

Inserting data from Table 1 into Equations (1) and (19), the revised equation has now a new form as follows:

$$F=7.7f^2-99.1f+1022.1. \tag{20}$$

Three quarter coil of both ends of the spring was grounded into a plane as a support coil. The revised total coils of the spring is 1.5+6=7.5≈8. Free height is 95 *mm*, and it is in accordance with the ideal height.

The diameter of the spring wire is 8 mm. All parameters of the nonlinear spring have also been determined. The nonlinear spring CAD model is shown in Fig. 3.

Table.1 Main parameters for spring design and calculation

No.	Each coil stiffness	Stiffness of the doubling up spring	Loads of the doubling up spring	Space	Pitch
	P' (N/mm)	F_i' (N/mm)	F_i (N)	δ_i (mm)	t_i (mm)
0	442.8	76	196		
1	442.8	93	737	1.05	9.05
2	442.8	159.2	3325.85	4.56	12.56
3	442.8	567	43880.2	5.5	13.5
4	442.8	567	43880.2	5.5	13.5
5	442.8	159.2	3325.85	4.56	12.56
6	442.8	93	737	1.05	9.05
7	442.8	76	196		

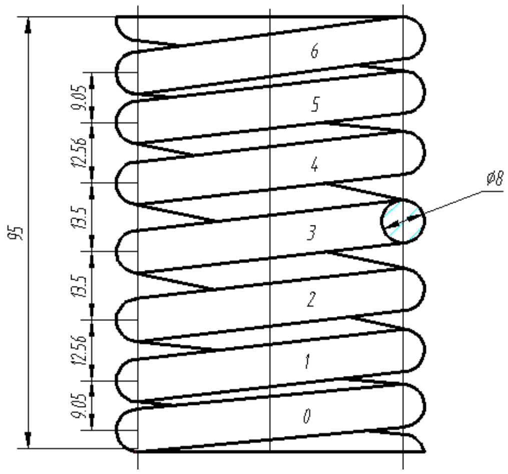

Figure 3. The nonlinear spring CAD model.

According to Table 1, when the load is normal, the system deformation is increased along with an increase in the system load. Even when the load suddenly reaches a certain level due to two or more spring coils contacting with each other, the system exhibits hardly any more deformation as a result of the load changing. It shows that the spring carries clearly non-linear hard-feature characteristics, which is good for improving the system amplitude stability.

According to Equation (20), load-stiffness diagram can be drawn as Fig.4, and we see that the load-stiffness diagram is nearly a straight line, which means that the system stiffness is linear along with the load. The system is energy saving and stable.

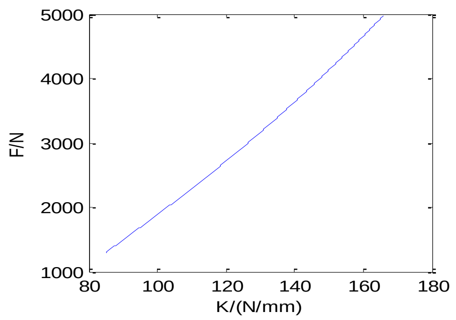

Figure 4. Load - stiffness diagram of nonlinear hard characteristics spring.

From the analysis of deformation energy of the spring, the rate of spring deformation energy can be gotten [8].

$$\frac{U}{U_F} = \frac{3890}{960} = 4.05. \tag{21}$$

where, U is deformation energy of linear equal pitch spring; U_F is nonlinear unequal spring deformation energy.

According to the above calculation, the pre-tightening force of the linear spring is 4.05 times larger than the force of the nonlinear spring. In this case, the pre-tightening force of the nonlinear spring is small so that it can be easily used to adjust spring preloading amount in order to change the frequency of the machine. Thus, the goal of stabilizing the systems and avoiding resonance has been achieved.

3 Conclusions

The variable pitch nonlinear spring is used in the main vibration system of the vibration mill whose stiffness can change along with the load to meet the requirements of dynamic load. A vibration system which avoids resonance, works stable and saves energy can be obtained.

Operations in our vibration mill prove that the vibration intensity was increased by 28%. Meanwhile, the noise and energy consumption was reduced by 4% and 10% respectively, which verifies the correctness of the main vibration system design. This design acts as a good reference source for other researchers in this field.

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