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Research Article

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Studying of movement kinematics of dynamically active sieve

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Abstract: The article presents the movement kinematics of the modular bar element of a dynamically active polymer sieve of a vibrating screen. On the basis of analytical methods, the mathematical model was obtained, which makes it possible to determine the law and trajectory of the modular bar element movement depending on its geometric characteristics, physico-mechanical properties of the polymer material, the regime and technological parameters of the vibrating screen. The results of this research show that in the working frequency range of the vibrating screen grate, modular bar element of the dynamically active sieve moves along the trajectories, the envelope of which is represented by Cassini's ovals, which along with the generation of the amplitude component in the horizontal and vertical directions allows one to obtain the self-cleaning effect of the sowing surface.

Keywords: Active sieve, polymer, movement kinematics, vibrating screen

1 Introduction

Sorting of materials is one of the most important technological operations of the mining and metallurgical industry, which is carried out mainly on vibrating screens [1–3].

One of the most important elements of screens that affect an effective separation of the material by size is the sowing surface. At present, a large number of various metal, polymer and reinforced polymeric and other sieves have been developed [4, 5] that differ from each other by design, hole shape, clear opening, dynamic activity, wear resistance and overall durability. Obviously, one of the main criteria of the effective working capacity of screens is its ability to self-clean from the stuck material and to prevent blinding during the entire lifetime. To solve this problem, the self-cleaning effect is most fully manifested in screens with relative oscillations of a dynamically active sowing surface with its kinematic excitation through the screen body.

On the basis of the analysis carried out, the study of the operational experience of different types of vibratory screens for charge and from practical information on the actual performance of their sowing surfaces, it was established that the most promising direction for further intensification of the screening process is the development and application of special sowing surfaces from wear-resistant polymer materials characterized by increased dynamics activity, the ability to self-cleaning and heterogeneous influence on the elaborated medium. As a result, the design of a dynamically active modular bar self-cleaning sowing surface (Figure 1) was developed consisting of detachable elastic elements fixed on longitudinal beams of a screen having a curved work surface of variable thickness and the cantilevered shelves, which form the dynamically active surface of the screen [6]. Since the aperture of the sowing surface is formed by active cantilevered shelves, it becomes necessary to study the trajectories of their motion, taking into account the structural parameters of the screen itself, as well as forced oscillations of the screen. Despite numerous studies related to the development of such sowing surfaces, the questions of their kinematics, depending on the forced vibrations of the box, have not been studied at the present sufficiently.

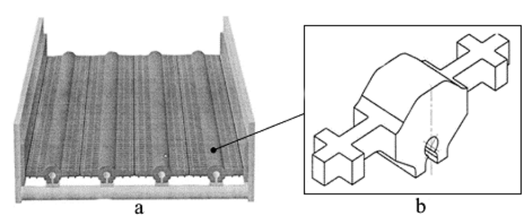

Figure 1: Dynamically active modular bar sowing surface: a - general view; b - removable elastic element

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The purpose of this research was to study the kinematics of the movement of the modular bar element of a dynamically active polymer sieve, depending on the circular and elliptical vibrations of the screen.

2 Modeling of Movement of Dynamically Active Sieve

Let's consider a modular bar element of a dynamically active sieve in the form of a cantilever beam (Figure 2) with a rectangular cross-section. Let the inertial force P be applied to the element from the side of the working member at the contact point C in a plane parallel to xOy, as shown in Figure 2, and changing in time according to the law:

$$\begin{cases} P_x(t) = mA_x\omega^2\cos(\omega\ t) \\ P_t(t) = mA_y\omega^2\sin(\omega\ t), \end{cases} \quad (1)$$

where m is the mass of the working element at the point of application of the inertial force; A_x, A_y are the amplitudes of the displacement of the contact point of the actuator with the reference element being considered (determined by the law of motion of the actuator); ω is the circular frequency of oscillation of the actuator.

Let's consider the form of the oscillation of the point C of the elastic element as a function of time, assuming that, due to the relatively small body length, inertia can be neglected.

The equation of deflection of the elastic line end point for a cantilever beam of rectangular cross-section under the action of a static transverse bending force in accordance with [7] has the form of:

$$\begin{cases} f_{Cx} = -P_{x\ \max}\frac{L^3}{3EI_y} = -P_{x\ \max}\frac{4L^3}{EHh^3} \\ f_{Cy} = -P_{y\ \max}\frac{L^3}{3EI_x} = -P_{y\ \max}\frac{4L^3}{EH^3h}, \end{cases} \quad (2)$$

where E is the Young's modulus for the material of the elastic element; $P_{x\ \max}$ and $P_{y\ \max}$ are the amplitude value of the perturbing force; I_x, I_y are the moments of inertia of the beam section relative to the corresponding axes.

The circular frequency of its own flexural vibrations with a static action of the given forces relative to the x and y axes, respectively, will be

$$\begin{cases} \omega_{0x} = \sqrt{\frac{g}{f_{Cx}}} = \sqrt{\frac{gh^3H\cdot E}{4L^3P_{x\ \max}}} \\ \omega_{0y} = \sqrt{\frac{g}{f_{Cy}}} = \sqrt{\frac{ghH^3\cdot E}{4L^3P_{y\ \max}}}, \end{cases} \quad (3)$$

The differential equation of the forced oscillations of a cantilever elastic element has the form

$$\begin{cases} \ddot{x} + \frac{\mu}{m}\dot{x} + \omega_{0x}^2\cdot x = \frac{P_x}{m} \\ \ddot{y} + \frac{\mu}{m}\dot{y} + \omega_{0y}^2\cdot y = \frac{P_y}{m}, \end{cases} \quad (4)$$

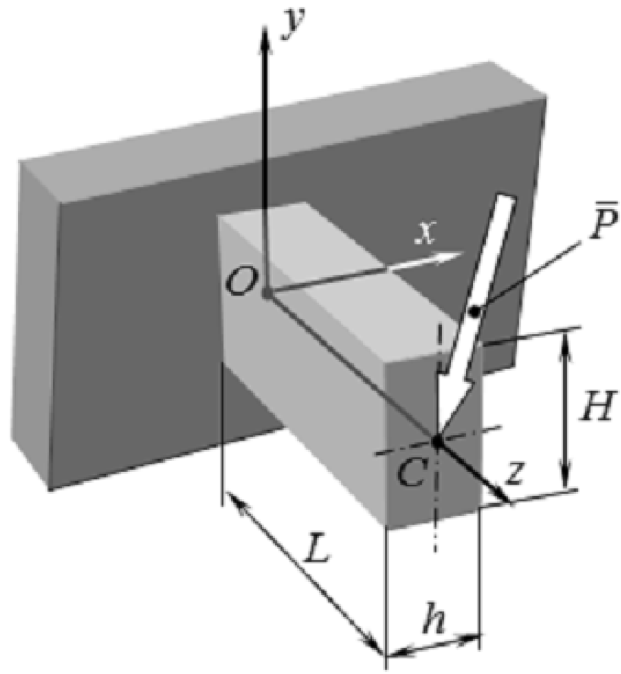

Figure 2: Modular bar element of dynamically active sieve

where m is the mass of the actuator at the point of application of the inertial force; ω_{0x} and ω_{0y} are, respectively, the circular frequency of natural oscillations of the working element in the characteristic directions; μ is the coefficient of resistance to movement of the point under consideration, associated with the energy losses due to internal friction in the material of the curved elastic element, which can be determined in accordance with [8] in the following way:

$$\mu = \frac{30\cdot c}{\pi\cdot n}\cdot\sqrt{\frac{P^2}{A^2\cdot c^2} - \left(1 - \frac{(\pi\cdot n)^2\cdot m}{900\cdot c}\right)^2} \quad (5)$$

where n is the rotational speed of the vibrator shaft, rot/min; c is the stiffness of the elastic element in the oscillation direction, N/m; P is the modulus of the disturbing force in the direction of oscillations, N; A is the amplitude of oscillations, m.

After substituting (1) to (3) into (4), we obtain an expanded expression in the form of:

$$\begin{cases} \ddot{x} + \frac{\mu}{m}\dot{x} + \frac{gh^3H\cdot E}{4L^3mA_x\omega^2}x = A_x\omega^2\cos(\omega\ t) \\ \ddot{y} + \frac{\mu}{m}\dot{y} + \frac{ghH^3\cdot E}{4L^3mA_y\omega^2}y = A_y\omega^2\sin(\omega\ t) \end{cases} \quad (6)$$

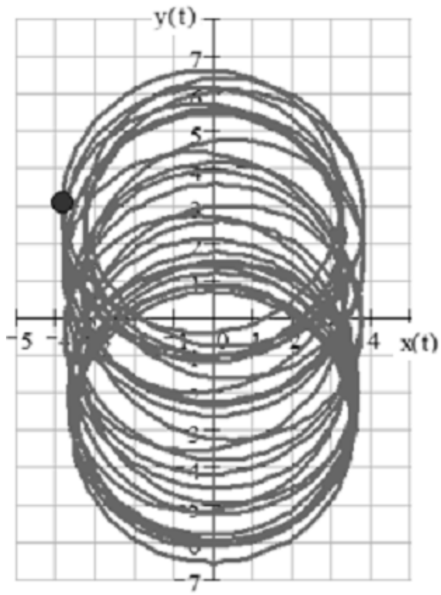

Figure 3: Trajectories of the sieve point movement during the screen oscillations circular frequency 12.5 Hz

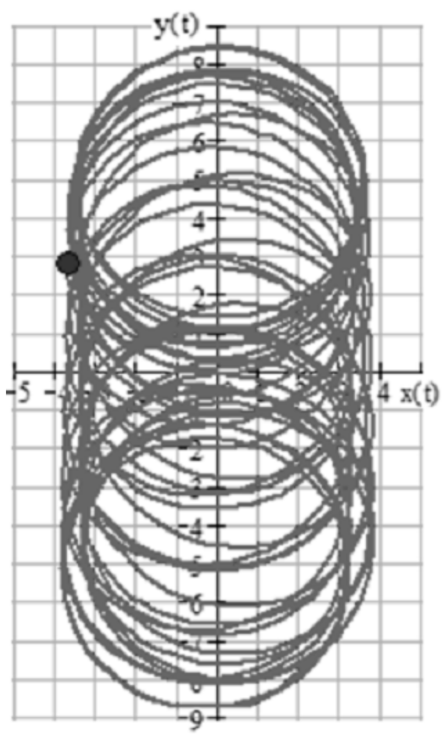

Figure 4: Trajectories of the sieve point movement during the screen oscillations circular frequency 16 Hz

The solution of the resulting system of differential equations makes it possible to determine the law and the trajectory of the motion of point *C* for the studied elastic element.

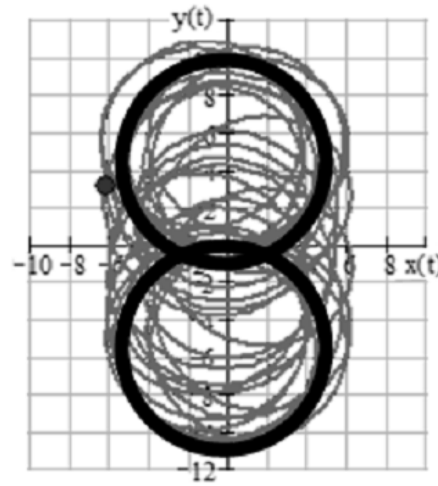

Figure 5: Characteristic trajectory of the movement of the point *C* of the screen during the circular oscillations

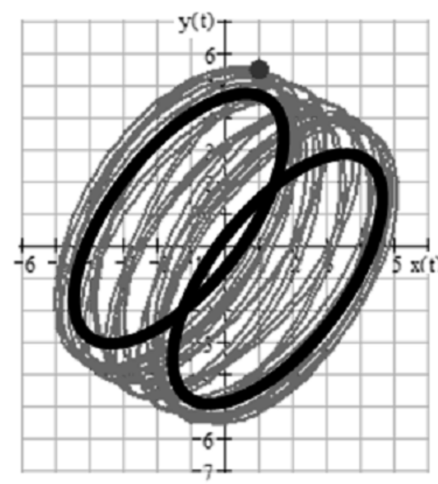

Figure 6: Characteristic trajectory of the movement of point *C* of the screen during the elliptical oscillations

3 Results and Discussion

In Figures 3 and 4 are shown the calculated trajectories of the point *C* for a cantilever rubber support element with a modulus of elasticity $E = 4 \cdot 10^{10}$Pa, the connected mass of which is 15 kg, excited by circular oscillations with an amplitude of 5 mm and a frequency of 12.5 and 16 Hz with a coefficient of resistance of 40 kg/s. As we can see from the figures, the trajectory of point S of the screen in both the first and second cases has a characteristic circular shape of the eight figure with additional generation of oscillations in the horizontal direction, and the envelope of these trajectories represents Cassini's ovals (Figure 5).

Performing similar calculations for elliptical oscillations of the machine box, it is possible to see that the finite elastic elements of the screen move along the ellipse path (Figure 6). Moreover, each part of the figure eight repeats the shape of the trajectory of the sieve contact points. It is also seen, in the first and in the second case, that there is an additional generation of oscillations in the horizontal direction, simultaneously with an increase in the amplitude component in the vertical direction.

4 Conclusions

The obtained results have shown us that in the working frequency range of the screen from 12.5 to 16 Hz, the elastic element of the dynamically active sieve moves along trajectories with time-varying focal parameters and angles of location of the axes so that the envelopes of these trajectories represent Cassini's ovals, which alongside additional generation of the amplitude component in the horizontal and vertical direction allows self-cleaning of the sowing surface.

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