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Vertical Oscillations of the Working Body Installed On an Elastic Bearing Support

N.N. Juraev,A. Djuraev,S. Khudoykulov

Navoi State Mining Institute, Navoi, Uzbekistan
Navoi branch of the Academy of Sciences of the Republic of Uzbekistan
Tashkent Institute of Textile and Light Industry, Tashkent, Uzbekistan

ABSTRACT: The article presents the results of solving problems on vertical oscillations. Based on the numerical solution, the regularities of oscillations of the shaft of the working body in the vertical direction are obtained. Analyzes of the constructed graphs determined the recommended parameter values.

KEY WORDS: Machine, working body, support, bearing, elastic element, deformation, oscillation, speed, acceleration, load, energy, stiffness, dissipation, dependence.

I.INTRODUCTION

The recommended working body has elastic bearings [1,2]. In the process, due to the deformation of the elastic support, the working body is actually minimized and bent. But, while the working body will make small vertical vibrations. These fluctuations to some extent provides the necessary process. Exceeding the oscillation amplitudes of the working body can lead to disruption of the technological process, as well as to a decrease in the life of the machine. The study of the nature of vertical vibrations of the working body is an important task. We conditionally accept that the loads on the elastic supports will be the same, and therefore the vibrations of the working body will be one-dimensional in the vertical direction.

II. SIGNIFICANCE OF THE SYSTEM

To compile a mathematical model of vertical vibrations, the working body on tapered rubber bearing bearings slipped out by the Lagrange II equation of the kind according to [3, 4]:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}}\right)-\frac{\partial T}{\partial q}+\frac{\partial P}{\partial q}+\frac{\partial F}{\partial \dot{q}}=Q(q) \qquad (1)$$

where, q,$\dot{q}$ is the generalized coordinate and its time derivative; T, P - kinetic and potential energy of the oscillatory system; Rayleigh f-dissipative function [3,5]; Q (q) is the generalized force, depending on the generalized coordinate.

III. LITERATURE SURVEY

According to the fig.1 of the design scheme, the potential and kinetic energy during vertical oscillations of the working body, taking into account the generalized coordinate z, the axis axis displacements, the working body will be:

$$P=\frac{1}{2}c_n z^2,\ c_n=c_1+kc_2, T=\frac{1}{2}m_c\dot{z}^2 \qquad (2)$$

where, c_n −n-reduced stiffness coefficient of the elastic support, c_1 −linear component of the stiffness coefficient, kc_2 −nonlinear composed stiffness coefficient, m_c −mass of the cylinder.
Dissipative Rayleigh function of an oscillatory system:

$$F=\frac{1}{2}b_n\dot{z}^2 \qquad (3)$$

where, b_n −the dissipation coefficient of the elastic bearing support of the working body.
Define the terms of the Lagrange equation (1):

$$\frac{\partial T}{\partial z}=\frac{\partial}{\partial z}\left(\frac{1}{2}m_c\dot{z}^2\right)=m_c\dot{z};\ \frac{d}{dt}\left(\frac{\partial T}{\partial \dot{z}}\right)=m_c\ddot{z};$$

$$\frac{\partial P}{\partial z}=\frac{\partial}{z}\left(\frac{1}{2}c_n z^2\right)=(c_1+kc_2)z;\ \frac{\partial \Phi}{\partial \dot{z}}=b_n\dot{z};\ Q(z)=F_0+\delta F_0 (4)$$

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where, $F_0, \delta F_0$ −the average value and the random component of the perturbing force of the forced vibrations of the working body.

IV. METHODOLOGY

In this case, substituting the obtained expression (4) in (1), we obtain the differential equation of random vertical vibrations of the working body, taking into account the elastic bearing bearings of the genie, in the following form:

$$m_c \frac{d^2t}{dt^2} + b_n \frac{dt}{dt} + (c_1 + kc_2)z = F_0 + \delta F_0 \qquad (5)$$

The numerical solution of differential equation (5) is carried out numerically by a PC using standard programs.

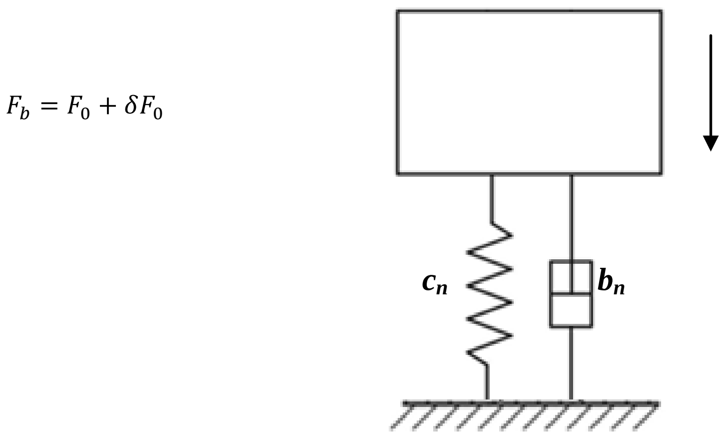
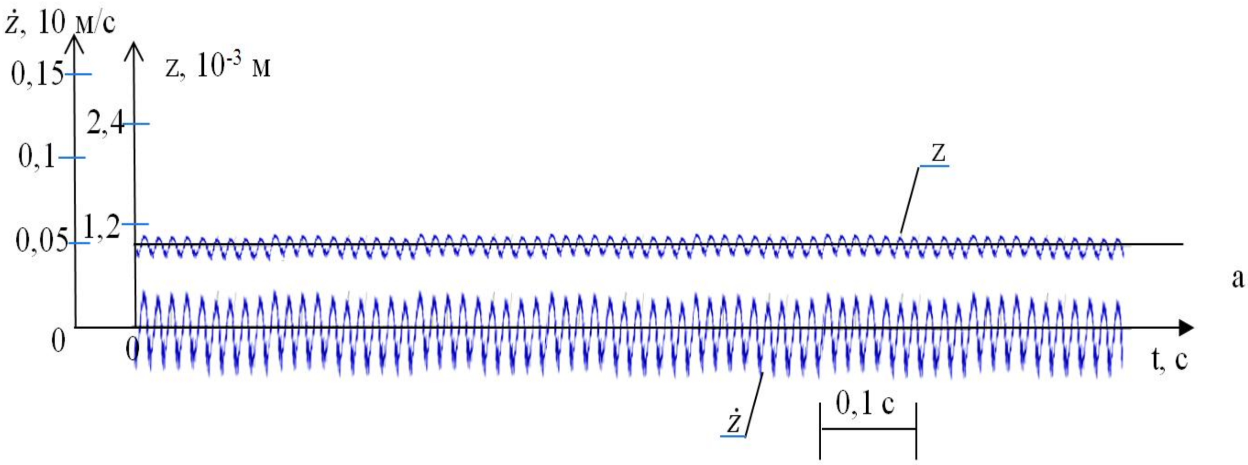


Fig.1 Design scheme of oscillations of the working body



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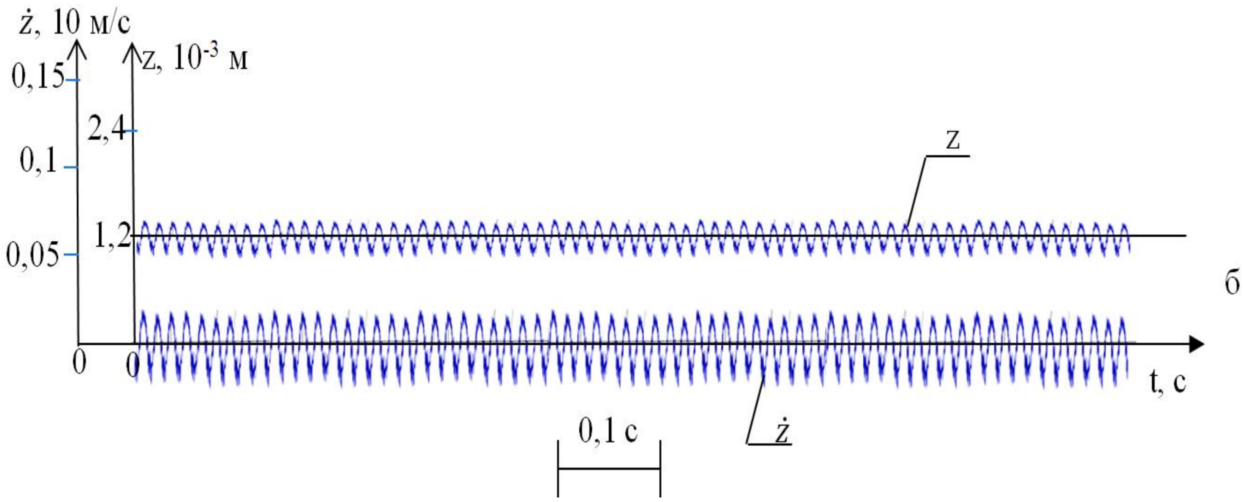

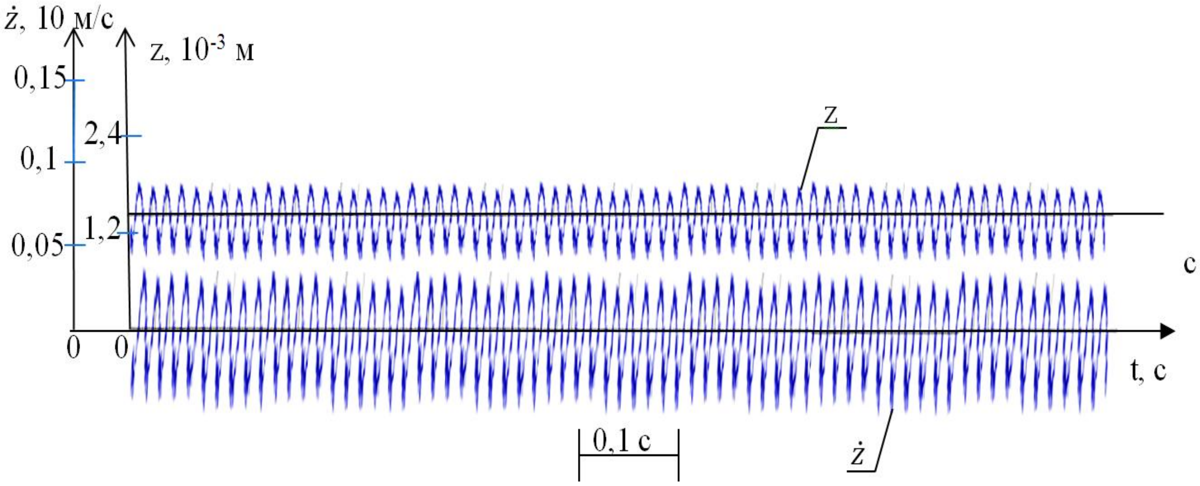

where, a - at F_0=1,2 N±(0,05÷0,12) N; б - at F_0=1,5 N±(0,08÷0,15) N; в - at F_0=1,8 N±(0,1÷0,18) N;

Fig. 2. Patterns of change in vertical vibrations of the working body with different ginning performance.

Random values of the disturbing force were produced taking into account the random number generator. The initial values of the system parameters were adopted:

$$\omega_c = 76{,}4\ \text{c}^{-1}; n = 730\frac{turn}{min};\ D_c = 2\ R_c = 0{,}32\ \text{м};$$
$$m_c = (3{,}75 \div 5{,}0)\cdot 10^2 kg;\ m_{\text{хв}} = (0{,}15 \div 0{,}5)kg;$$
$$F_0 = (1{,}2 \div 1.8)H \pm (0{,}05 \div 0{,}15)H;\ c_1 = (1{,}5 \div 6{,}5)\cdot 10^4\frac{H}{m};$$
$$\text{c}_2 = (0{,}08 \div 0{,}12)\cdot 10^4\frac{H}{m}; b_n = (4{,}2 \div 12{,}5)\,{}^{Hc}/_{m}; k = 0{,}2 \div 0{,}6$$

Based on the numerical solution of (5), regularities were obtained for changes in the vertical mixed and speeds of the working body mounted on elastic bearing bearings, which are presented in Fig.2. Analysis of the patterns presented in Fig.2 shows that an increase in the load leads to an increase in the amplitude of velocity fluctuations and vertical displacements of the working body (see Fig.2, a, b, c). In addition, the greater the external load, the greater the static displacement of the axis of the working body due to its weight. This movement under load 1,2 N$\pm$(0,05÷0,12) N reaches $1.2\cdot 10^{-3}$ m, and under load 1,8 N$\pm$(0,1÷0,18)N, reaches to $1{,}5\cdot 10^{-3}$ m.

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With an increase in load from 0.25 N to 1.8 N with a cylinder mass of $5.0 \cdot 10^{-2}$ kg, the amplitude of its oscillations of the working body increases from $0.3 \cdot 10^{-4}$ m to $0.252 \cdot 10^{-3}$ m, and with a decrease in the mass of the working body up to 3.75·102kg the amplitude of its movement oscillations increases from $0.4 \cdot 10^{-4}$m to $0.71 \cdot 10^{-3}$m. Therefore, to ensure the amplitude of the working body oscillations is not more than $0.3 \cdot 10^{-3}$m, it is considered advisable to choose $m_c \leq (3{,}75 \div 4{,}25) \cdot 10^2 kg и F_0 \leq (1{,}0 \div 1{,}2)N$.

Findings. The method of reducing the bending of the shaft of the working body of the technological machine is recommended through the use of elastic bearing bearings. The vertical vibrations of the working body are studied theoretically.

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