VIBRATION ANALYSIS OF DYNAMIC PARAMETERS OF SCREW CONVEYORS USED IN THE WOOD PROCESSING.

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Abstract

Screw Conveyors are used to transport tasks: to feed shredder in the manufacture of wood chip boards to discharge dust and sawdust and chips to power their transport by thermal treatment facilities. Correlation function with the constructive dynamic parameters, it is necessary to generate a model able to reflect as accurately determined theoretical results with the experimental ones.

Key words: vibration, amplitude, pulsation, transporters.

INTRODUCTION

On helical transporters, the effect of timber material feeding trough is due to the generation of two simultaneous movements namely: one vertical and one horizontal. The two movements are maintained because of a vertical harmonic force and a moment, both are out of phase. The dynamic model considered in this paper is outlined as a system with two degrees of freedom (Fletcher, H, J, Glazman I. M.,_Liubici Iu. I ,Manescu T.): z and φ. The both vibrating movements are coupled. Taking into account certain structural conditions imposed in compiling machine, so the vibration modes are decoupled by two coordinates z, φ and can be treated separately each motion in part.

MATERIAL AND METHOD

The design of the machine consists in a central tube which is attached to the outside of a water transport screw. Angle of gutter coil is between 2 and 10 degrees.Vertical movement of the timber is determined by solving the differential equation (Barsan G, Bratu, P, Petrila T,) of the form:

$$\ddot{z} + 2n\dot{z} + p^2 z = \frac{F_0}{m}\sin\omega t \tag{1}$$

Solution of differential equation is of the form:

$$z=\frac{F_0}{mp^2}\frac{1}{\sqrt{\left(1-\frac{\omega^2}{p^2}\right)^2+\frac{4\omega^2n^2}{p^4}}}\sin(\omega t-\theta) \tag{2}$$

Vertical amplitude is given by:

$$A_z=\frac{F_0}{mp^2}\frac{1}{\sqrt{\left(1-\frac{\omega^2}{p^2}\right)^2+\frac{4\omega^2n^2}{p^4}}} \tag{3}$$

Where:

$z-$ coordinates of vertical movement of wood;

$p-$ natural pulsation of machine;

$\omega-$pulsation of harmonic force;

$\theta-$ force phase difference;

F_0-force module;

$m-$mass system;

For the vibratory movement of the horizontal plan, wood disruptive forces and moments are determined using the relationship:

$$F_z=2m_0r\omega^2\cos\gamma\sin\omega t \tag{4}$$

$$M_{0z}=2m_0r\omega^2a\cos\gamma\sin\omega t \tag{5}$$

It is considered to be negligible damping system. Differential equation of horizontal vibration is:

$$J\ddot{\varphi}=M_{0y} \tag{6}$$

Solution of differential equation (Manescu T, Fetea M) is of the form:

$$\varphi=\frac{2m_0ra}{J}\sin\gamma\sin\omega t$$

Where:

$J-$moment of inertia about a vertical axis machine symmetry;

$\varphi-$horizontal angular displacement;

m_0r-total static moments of mass located eccentric of equipment;

$\gamma-$angle with the horizontal axis vibrators;

$2a-$distances between axes vibrators.

Horizontal displacement of a point on the circle of radius R of gutter coil is:

$$x = R\varphi = \frac{2m_0 raR}{J}\sin\gamma \sin\omega t \qquad (7)$$

Roll angle of the corresponding timber radius R of the trough is given by:

$$\beta = \frac{z}{x}, \qquad (8)$$

where:

$$z = \frac{2m_0 r}{m}\cos\gamma \sin ax, \qquad (9)$$

for the general situation in the post resonance machine operation

$$\omega > p.$$

Amplitudes corresponding for this two movements are:

$$A_z = \frac{2m_0 r}{m}\cos\gamma \qquad A_\varphi = \frac{2m_0 ra}{J}\sin\gamma \qquad (10)$$

The validity of these relations is satisfied for conditions:

$$\omega = (2 \div 3)p_z \qquad (11)$$

$$\omega = (2 \div 9)p_\varphi \qquad (13)$$

RESULTS AND DISCUSSION

In this paper the authors consider the following initial values:

$$F_0 = 10[daN]$$
$$\omega = 5[rad/\sec]$$
$$\theta = 0$$
$$m = 200[Kg]$$
$$m_0 = 70[Kg]$$
$$p_z = 2,5[rad/\sec]$$
$$r = 0,1[m]$$
$$\gamma = 30^\circ$$
$$a = 0,3[m]$$

Intervals at which the calculation is performed are, $t_1 = 2[\sec]$, $t_2 = 4[\sec]$. Following calculation algorithm presented, results the dynamic parameters:

1. for vertical movement at $t_1 = 2[\sec], t_2 = 4[\sec]$ solutions of differential equation are:

$$z_1 = 0,0014[m],.$$
$$z_2 = 0,00237[m].$$

Amplitudes corresponding for this movement are

$$A_{z1} = A_{z2} = 0{,}026[m].$$

2. for horizontal movement at $t_1 = 2[\sec], t_2 = 4[\sec]$:

$$F_{z1} = 29{,}370[daN]$$

$$M_{0z1} = 56{,}43[daN \cdot m]$$

$$F_{z2} = 49{,}280[daN]$$

$$M_{0z2} = 94{,}79[daN \cdot m]$$

$$\varphi_1 = 0{,}117[rad]$$

$$\varphi_2 = 0{,}197[rad]$$

In the post resonance, decoupled vibration amplitude values given by perturbing pulse are:

$$A_z = 0{,}1079$$

$$A_\varphi = 0{,}216$$

CONCLUSIONS

Given the fact that vibration is one of the most dangerous phenomena involved in the operation of machinery, the paper is a model for reducing dynamic parameter represented by the amplitude of vibration.

The data obtained by the authors from the calculation shows the following:

- adjust the dynamic parameters of operation of the machine.
- correct operation under resonance position.
- adjusting the dynamic parameters depending of the constructive parameters of equipment.

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