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CONTENTS

PRELIMINARY TENSION OF CLAMP DEVICE OF MECHANISM MO-10 INFLUENCE ON ITS OSCILLATIONS

Eremyants V.E., Arstanbek A.

Keywords: impact mechanism, tool, clamp device, amplitude of oscillations, impact velocity.
Abstract. The mathematical model of the impact mechanism oscillations is developed taking into account pre-tension of the spring device of its clamping to the tool. Formulas are obtained for determining the displacement and speed of the body as a function of time, preliminary tension of the clamping device and parameters of the forcing load.

ВЛИЯНИЕ ПРЕДВАРИТЕЛЬНОГО НАТЯГА УСТРОЙСТВА ПРИЖИМА МЕХАНИЗМА МО-10 НА ЕГО КОЛЕБАНИЯ

Еремьянц В.Э., Арстанбек А.

Ключевые слова: ударный механизм, инструмент, устройство прижима, амплитуда колебаний, скорость удара.
Аннотация. Разработана математическая модель колебания корпуса ударного механизма, учитывающая предварительный натяг пружинного устройства его прижима к инструменту. Получены формулы для определения перемещения и скорости корпуса в зависимости от времени, предварительного натяга устройства прижима и параметров вынуждающей нагрузки.

The MO-10 crank-rocker impact mechanism is used in the MB-10 machine for vibro-impact cleaning of thermoelectric plant coal bunkers and systems for transporting coal fines to furnaces. Its construction and principle of operation are described in papers [1, 2]. These machines are mounted on the side surfaces of an object being treated and are periodically turned on to prevent from coal fines sticking on its internal surfaces.

During the machine operation under the action of reactive loads, the body of the impact mechanism oscillates relative to the working tool and strikes the instrument. It is shown in [2] that the oscillations' amplitude of the impact mechanism body and the speed of its collision with the tool are the main criteria for effective operation of the machine determining its productivity and durability. Therefore, certain limitations are imposed on the amplitude of the oscillations and the impact velocity related to the reduction of impact loads on the mechanism body and the tool.

Denoting the movement of the mechanism body relative to the tool y_0 these limitations can be written as:

$$y_0 \le h; \quad \dot{y}_0 \le [V_0],$$

where h is the value of free path of the body relative to the tool (Figure 1); $[V_0]$ - the speed of impact of the body with the tool, allowed by the condition of the strength of elements.

To meet these limitations, it is necessary to know the interrelationships of the characteristics of the mechanism body oscillations with the parameters of the impact mechanism and the mechanism of its clamping to the tool, as well as the law of the change in the forcing forces acting on the body of the impact mechanism. To identify these relationships, it is necessary to develop and analyze an appropriate dynamic model. The solution of this problem was begun in previous papers [1-3].

It was established in [1], that the forcing loads are determined mainly by the reactions occurring in supports of the crank-rocker impact mechanism between impacts. In paper [2] the characteristic of the forcing loads is presented, which is represented in the form of a sinus function, acting on a certain length of time before the impact striker strikes the instrument. In [3] the vibrations of the body of the impact mechanism under the action of the forcing load were investigated without taking into account the possible pre-tension of the springs of the clamping mechanism.

The tasks of this work included the investigation of the pre-tension influence of the springs on the body dynamics of the MO-10 mechanism.

The calculation scheme of the system under study is shown in Figure 1.

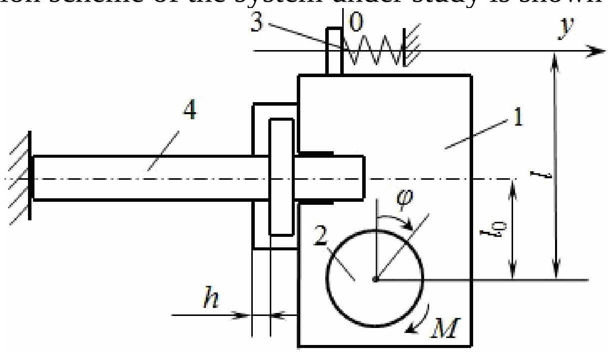

Fig. 1. Scheme of the system under study

It consists of the mechanism body 1, which can rotate relative to the fixed support 2 under the action of the driving torque *M*(*t*). The body is pressed by the spring mechanism 3. Under the influence of these forces, the body oscillates, striking the shoulder of the tool 4, which rests on the outer surface of the bunker. Under the action of impacts, the inner surface of the bunker is cleaned of the adhering coal mass.

The law of the change in the driving moment has the form, shown in Figure 2. According to the data of [2], it is described by a function:

$$0 < t < t_1, \qquad M = 0;$$

$$t_1 < t < T, \qquad M(t) = H \sin \mu t,$$

where t_1 – beginning time of the driving moment action; T – one cycle time of the impact mechanism movement:

$$T = 1/f = \omega / 2\pi,$$

f – frequency of impacts; ω – angular velocity of the impact mechanism crank; H, μ – the amplitude and frequency of the change in the driving torque respectively:

$$\mu = \pi / \left(T - t_1\right).$$

If there is a preliminary deformation of springs of value a, then the movement of the impact mechanism body will begin at time t_0 (Figure 2), when the driving torque will reach the value due to the pre-deformation of the springs. The equality of these moments is determined by the relation:

$$cal = H \sin \mu \left(t_0 - t_1\right),$$

where c – coefficient of spring stiffness; l – shoulder of elastic force (distance from the axis of rotation of the mechanism body to the longitudinal axis of the springs).

From this relation follows:

$$t_0 = t_1 + \left(1/\mu\right) \arcsin \nu, \qquad \nu = cal / H,$$

where v – is the dimensionless parameter equal to the ratio of the torque of the preliminary tightening of springs to the amplitude of the driving torque.

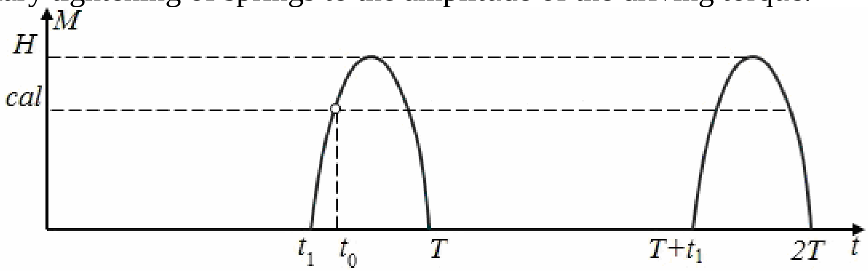

Fig. 2. Dependence of the driving torque on time

From the beginning of the movement of the impact mechanism the equations of the system motion under consideration at different time intervals have the following form:

$$t_0 < t < T, \qquad \ddot{\varphi} + k^2\varphi = H\sin\mu(t - t_0) - k^2\varphi_0; \tag{1}$$

$$T < t < T + t_1, \qquad \ddot{\varphi} + k^2\varphi = -k^2\varphi_0, \tag{2}$$

where φ – is the angular displacement of the mechanism body relative to the initial position; k – is the natural oscillation frequency of the body; φ_0 – is the angle corresponding to the preliminary deformation of springs,

$$k = l\sqrt{c / J}; \quad \varphi_0 = a / l,$$

J – moment of inertia of the impact mechanism relative to the axis of its rotation.

Considering the relationship between the angle of rotation of the mechanism body and its movement along the arc relative to the shoulder of the tool:

$$y_0(t) = l_0\varphi(t),$$

where l_0 – is the distance from the axis of rotation of the body to the longitudinal axis of the tool, we can rewrite the equation of motion in new coordinates:

$$t_0 < t < T, \qquad \ddot{y}_0 + k^2 y_0 = Hl_0\sin\mu(t - t_0) - k^2 a\lambda; \tag{3}$$

$$T < t < T + t_1, \qquad \ddot{y}_0 + k^2 y_0 = -k^2 a\lambda, \tag{4}$$

where $\lambda = l_0/l$.

The solution of the equation of motion (3) under initial conditions

$$y_{01} = y_0(t_0) = 0; \ \dot{y}_{01} = \dot{y}_0(t_0) = 0,$$

has the form:

$$y_0(t) = A_1\sin\mu s(t - t_0) + B_1\cos\mu s(t - t_0) + C\sin\mu(t - t_1) - a\lambda_0, \tag{5}$$

where $C = \dfrac{Hl_0}{J\mu^2\left(s^2 - 1\right)}$; $A_1 = -C\cos\mu(t_0 - t_1)/s$; $B_1 = a\lambda - Cv$; $s = k/\mu$.

In this case, the speed of the body of the mechanism relative to the impact end of the tool is determined as:

$$\dot{y}_0(t) = \mu s\left[A_1\cos\mu s(t - t_0) - B_1\sin\mu s(t - t_0) + \left(C / s\right)\cos\mu(t - t_1)\right] \tag{6}$$

Depending on the parameter s, the impact on the instrument can occur at different time intervals.

In the case of the first interval $t_0 < t < T$, the body does not strike the instrument, then it is necessary to turn to the equation of motion (4). The initial conditions for solving it are:

$$y_{02} = y_0(T); \;\; \dot{y}_{02} = \dot{y}_0(T),$$

and the solution is described by a function:

$$y_0(t) = \frac{\dot{y}_{02}}{\mu s} \sin \mu s(t-T) + (y_{02} + a\lambda) \cos \mu s(t-T) - a\lambda ; \qquad (7)$$

$$\dot{y}_0(t) = \dot{y}_{02} \cos \mu s(t-T) - \mu s(y_{02} + a\lambda) \sin \mu s(t-T) . \qquad (8)$$

In the case of the considered interval of the motion of the mechanism body impact with the tool did not occur, then the next interval should be considered, etc. In this case, the initial conditions for the next interval are the values of the displacements and velocities of the impact mechanism body at the previous interval.

As a result, for different intervals of motion, we have

$$T + t_1 < t < 2T$$

$$y_0(t) = A_3 \sin \mu s(t-T-t_1) + B_3 \cos \mu s(t-T-t_1) + C \sin \mu(t-T-t_1) - a\lambda_0 ;$$

$$2T < t < 2T + t_1$$

$$y_0(t) = \frac{\dot{y}_{04}}{\mu s} \sin \mu s(t-2T) + (y_{04} + a\lambda) \cos \mu s(t-2T) - a\lambda ;$$

$$2T + t_1 < t < 3T$$

$$y_0(t) = A_5 \sin \mu s(t-2T-t_1) + B_5 \cos \mu s(t-2T-t_1) + C \sin \mu(t-2T-t_1) - a\lambda_0 .$$

In these solutions for the i- th interval of motion, we have

$$A_i = \left(\dot{y}_{0i} - C\mu\right) / \mu s; \quad B_i = y_{0i} + a\lambda .$$

To find the amplitude of the body oscillations at different intervals of motion, it is necessary to equate the body speed in a given interval to zero, find the time corresponding to this condition, and substitute it in the displacement function.

To find the impact speed of the body on the tool shoulder, it is necessary to equate the movement in this interval to zero, find the time corresponding to this movement, and substitute it in the formula for determining the speed. If the time values found are outside the range considered, then it is necessary to proceed to the next time interval.

As an example, Figure 3 shows diagrams of the amplitude of the oscillations of the hull (a) and the speed of its collision with the tool (b) from the parameter s when the springs are pre-tensioned by 4.5 mm.

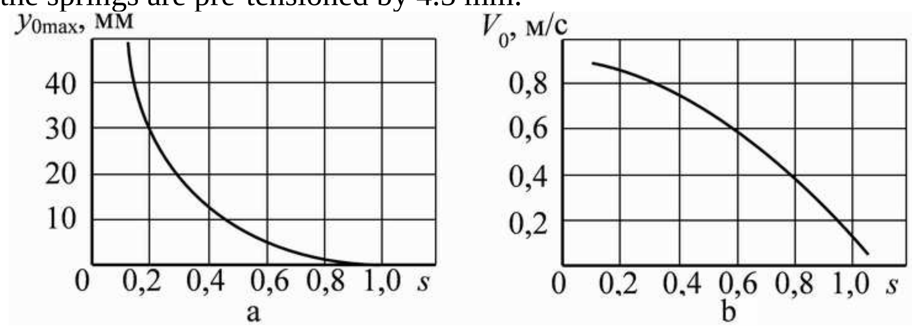

Fig. 3. Dependence of the vibration amplitude of the body (a) and the velocity of its collision with the tool (b) on the parameter s

From these dependences it follows that, for example, when the vibration amplitude is limited to 5 mm, the ratio *s* must be greater than 0.6. This is ensured by choosing the appropriate spring stiffness.

The formulas obtained make it possible to analyze the influence of the parameters of the impact mechanism and the clamping mechanism on the amplitude of the vibrations of the impact mechanism body and the speed of its collision with the tool. Based on the established dependencies, it is possible to develop recommendations for choosing rational parameters of the clamping mechanism that satisfy the specified limitations.

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