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DYNAMIC MODEL OF CAGE HOIST

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ABSTRACT
The article treats a dynamic model for investigation of the force load in the rope system of cage hoists. For this purpose, a bi-mass mathematical model of a dissipative system has been used, proceeding from the structure specifics and the technological process, obtaining solutions satisfactory for the investigation task.

Control tests and investigations of mine hoists are inseparable part of their operation. These activities are directly orientated to the operating security of mine hoists.

In the last years, non-destructive control methods have gained increasingly wider practical application which, no doubt, involves theoretical site investigation by modeling of the dynamic processes.

For complex multi-mass vibrating systems, where the vibrations of some masses are described by linear coordinates, and others by angular coordinates, the most convenient method is Lagrange's method. For a non-conservative system (with attenuation, damping), Lagrange's equation:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right) - \frac{\partial T}{\partial \dot{q}_i} + \frac{\partial D}{\partial q_i} + \frac{\partial \Pi}{\partial q_i} = Q \qquad (1)$$

where:

T and *Π* are the kinetik and potential energy of the system:;

D – energy of dissipation (distance);

q_i – generalized coordinates;

Q – generalized forces;

t – time.

Knowing the structure of a given type of mine shaft system, a number of dynamic models can be composed in connection with the investigation task preliminarily set.

Thus, for example, of particular importance for mine hoist is the force load in the rope system. According to [2], when the object of investigation is the force load in the rope system of a hoist-conveyor mechanism, the dynamic model is reduced to a single bi-mass vibrating system, which is described with the mass inertia moment of all rotating parts, reduced to the drum shaft, and with the equivalent hardness of the whole mechanism.

In the most general case of a cage hoist (Fig.1), the mathematical description of the dynamic force load in the rope system according to Lagrange's method, will be:

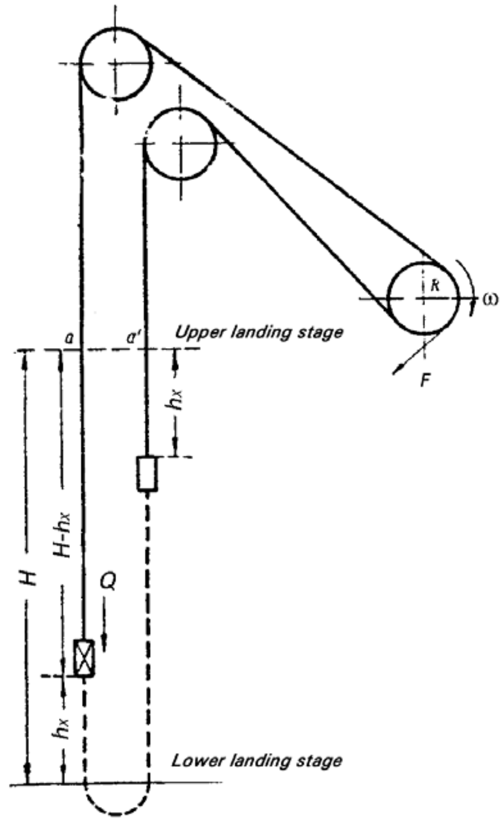

Figure 1.

$$\left| \begin{aligned} & m\ddot{x} + d\dot{x} + kx - d\frac{R}{i}\dot{\varphi} - k\frac{R}{i}\varphi = Q \\ & J\ddot{\varphi} + d\left(\frac{R}{i}\right)^2 \dot{\varphi} + k\left(\frac{R}{i}\right)^2 \varphi - d\frac{R}{i}\dot{x} - k\frac{R}{i}x = M \end{aligned} \right. \qquad (2)$$

where:

x, φ - are the generalized coordinates;

m – load mass;

J – mass inertia moment of all rotating parts reduced to the drum shaft;

κ – equivalent hardness of the system;

Q – generalized force of the weight (+Q for lowering, -Q for hoisting);

M – moment of the engine (brakes) reduced to the drum axle;

R – drum radius;

d – coefficient of damping, resp. for progressively moving and rotating masses [2].

In the bi-mass model, the generalized force Q is assumed as equal to the maximal one for dimensioning of the hoist force, determined in [1], and the el.motor moment M as proportional to the angular speed of the rotor.

The system of linear differential equations (2) is solved for different initial conditions whereby the specific peculiarities of the physical process are introduced in the mathematical model. These peculiarities in the cage hoists must reflect the characteristic periods of the technological cycle, and namely:

First period t_1 – equi-accellerating movement with speed increase from *0* up to V_{max} by linear law, with constant accelleration a;

Second period t_2 – movement with constant speed V_{max} for accelleration $a = 0$;

Third period t_3 – equi-retarding movement with speed decrease from V_{max} down to *0* for constant accelleration a.

During the period of movement of hoist vessels with constant speed V_{max} = const., only static forces act in the hoist system, determined by the weight of the hoisted loads, the weight of the rope with the engaging units and the friction forces. In the periods of equi-accellerating and equi-retarding movement, besides the static forces, there also act inertia forces of the moving masses. Consequently, in this specific case, of interest are the dynamic loads upon acceleration and stopping of the load Q. If in the theoretical investigations it is assumed that the starting torque of the engine and the brakes torque are constant, and also that the washer block is undeformable, the solution of system (2) is of type [2]:

$$\left|\begin{array}{l} x = \dfrac{A_1}{2}t^2 + B_1 t + C_1 + D_1 e^{-\lambda t}\left(\cos\omega t + \dfrac{E_1 - D_1\lambda}{\omega D_1}\sin\omega t\right) \\ \varphi = \dfrac{A_2}{2}t^2 + B_2 t + C_2 + D_2 e^{-\lambda t}\left(\cos\omega t + \dfrac{E_2 - D_2\lambda}{\omega D_2}\sin\omega t\right) \end{array}\right. \quad (3)$$

where:

ω is the frequency of vibrations, which depends on the parameters of the bi-mass model and is determined by:

$$\omega = \sqrt{k\left(\frac{1}{m} + \frac{R^2}{Ji^2}\right) - \frac{d^2}{4}\left(\frac{1}{m} + \frac{R^2}{Ji^2}\right)^2}$$

λ is coefficient of attenuation, determined by:

$$\lambda = \frac{d}{2}\left(\frac{1}{m} + \frac{R^2}{Ji^2}\right)$$

$A_{12}, B_{12}, C_{12}, D_{12}, E_{12}$ are coefficients depending on the system parameters and the initial conditions;

t – time.

The velocities $\dot{x}$ and $\dot{\varphi}$ can be determined through differentiation of system (3).

To solve the problem, it is necessary to determine the constants $A_{12}, B_{12}, C_{12}, D_{12}, E_{12}$, in accordance with [2], for the respective initial conditions, which for the period of load acceleration, are: $t = 0$, $x_0 = 0$, $\dot{x}_0 = 0$, $\varphi_0 = 0$, $\dot{\varphi}_0 = 0$

$$\left|\begin{array}{l} A_1 = \left(\dfrac{M}{m} \pm g\dfrac{R}{i}\right)\dfrac{k}{J}.\dfrac{R}{i} / \left(\lambda^2 + \omega^2\right) \\ B_1 = \left[\left(\dfrac{M}{J}.\dfrac{d}{m} \pm g\dfrac{d}{J}\dfrac{R}{i}\right)\dfrac{R}{i} - 2A_1\lambda\right] / \left(\lambda^2 + \omega^2\right) \\ C_1 = \left(\pm g - A_1 - 2B_1\lambda\right) / \left(\lambda^2 + \omega^2\right) \\ E_1 = -B_1 - 2C_1\lambda \\ D_1 = -C_1 \end{array}\right. \quad (4)$$

$$\left|\begin{array}{l} A_2 = \left[\dfrac{k}{J}\left(\dfrac{M}{m} \pm g\dfrac{R}{i}\right)\right] / \left(\lambda^2 + \omega^2\right) \\ B_2 = \left[\dfrac{M}{J}\dfrac{d}{m} \pm g\dfrac{d}{J}\dfrac{R}{i} - 2A_2\lambda\right] / \left(\lambda^2 + \omega^2\right) \\ C_2 = \left(\dfrac{M}{J} - A_2 - 2B_2\lambda\right) / \left(\lambda^2 + \omega^2\right) \\ E_2 = -B_2 - 2C_2\lambda \\ D_2 = -C_2 \end{array}\right. \quad (5)$$

For the period of load stopping for initial conditions $t_0 = T - t_c$, (t_c – time of the braking process), $x_0 = H - h_x$ (from Fig.1), $\dot{x}_0 = V_{\max} = const$, $\varphi_0 = \dfrac{2\pi R}{i}$ и $\dot{\varphi}_0 = \dot{\varphi}_{\max}$, the constants $A_{12}, B_{12}, C_{12}, D_{12}, E_{12}$, are determined by systems of equations (6) and (7):

$$\left|\begin{array}{l} A_1\left(\lambda^2 + \omega^2\right) = \left(\dfrac{M}{m} \pm g\dfrac{R}{i}\right)\dfrac{k}{J}.\dfrac{R}{i} \\ 2A_1\lambda + B_1\left(\lambda^2 + \omega^2\right) = \left(\dot{x}_0\dfrac{k}{J}\dfrac{R}{i} + \varphi_0\dfrac{k}{m} + \dfrac{M}{J}\dfrac{d}{m} \pm g\dfrac{d}{J}\dfrac{R}{i}\right)\dfrac{R}{i} \\ A_1 + 2B_1\lambda + C_1\left(\lambda^2 + \omega^2\right) = \dot{x}_0\dfrac{d}{J}\dfrac{R^2}{i^2} + x_0\dfrac{k}{J}\dfrac{R^2}{i^2} + \dot{\varphi}_0\dfrac{d}{m}\dfrac{R}{i} + \varphi_0\dfrac{k}{m}\dfrac{R}{i} \pm g \\ B_1 + 2C_1\lambda + E_1 = \dot{x}_0 + x_0 d\left(\dfrac{1}{m} + \dfrac{1}{J}\dfrac{R^2}{i^2}\right) \\ C_1 + D_1 = x_0 \end{array}\right. \quad (6)$$

$$\left|\begin{array}{l} A_2\left(\lambda^2 + \omega^2\right) = \dfrac{k}{J}\left(\dfrac{M}{m} \pm g\dfrac{R}{i}\right) \\ 2A_2\lambda + B_2\left(\lambda^2 + \omega^2\right) = \dfrac{M}{J}\dfrac{d}{m} + \dot{\varphi}_0\dfrac{k}{m} + \dfrac{k}{J}\dfrac{R}{i}x_0 \pm g\dfrac{d}{J}\dfrac{R}{i} \\ A_2 + 2B_2\lambda + C_2\left(\lambda^2 + \omega^2\right) = \dfrac{M}{J} + \dfrac{d}{m}\dot{\varphi}_0 + \dfrac{k}{m}\varphi_0 + \dfrac{d}{J}\dfrac{R}{i}\dot{x}_0 + \dfrac{k}{J}\dfrac{R}{i}x_0 \\ B_2 + 2C_2\lambda + E_2 = \dot{\varphi} + \varphi_0 d\left(\dfrac{1}{m} + \dfrac{1}{J}\dfrac{R^2}{i^2}\right) \\ C_2 + D_2 = \varphi_0 \end{array}\right. \quad (7)$$

The sign (-) refers to a case of IQ load lifting, and the sign (+) for a case of its lowering.

Thus, for known height of the hoist *H* and duration of the movement of the hoist vessel *T*, the force in the rope *F* can be determined as a difference between the coordinates of the moving masses, multiplied by the hardness *к*, and namely:

$$F = \left[\frac{R}{i} \varphi(t) - x(t) \right] k \qquad (8)$$

The modeling of dynamic processes and the composing of adequate models for investigation of real objects is a theoretical approach where the volume of natural experimental measurements for evaluation of the functional and operational fitness of the machines is optimized.

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