Development and research of elastic-damping device with variable stiffness

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Abstract. The article is devoted to the reduction of dynamic loads of actuating mechanisms of mining machines based on the use of elastic-damping devices with variable parameters built into the construction of these machines. It is shown that the determination of the parameters of these devices can be represented as a solution of the inverse dynamic problem based on a differential equation with variable coefficients. Using the example of a damping device proposed for reducing the dynamic loads of the digging mechanism of dragline ES 20.90, we describe the procedure for constructing the desired differential equation for synthesizing the device parameters and present the results of a numerical study of its effectiveness.

1. Introduction
An effective way to reduce dynamic loads of mining machines is the use of elastic damping devices (EDD) in the form of springs, air springs, hydraulic and pneumatic dampers, which are built into the kinematic motion transmission circuits and allow to move the natural frequencies from the frequencies of disturbing influences and increase the intensity consumption of oscillation energy [1-3]. Known methods for installing the EDD in the crowd mechanisms of the cable [4] and rack [5,6] type of shovel, in the hoist [7] and the boom suspension mechanism [8] of a shovel, in the traction mechanism of dragline [9,10], in the drive of the bucket-wheel excavators [1,11], in the mechanisms of mine hoist [12].

As the analysis shows, EDD with fixed parameters does not allow ensuring their effective operation in mining machines, because the parameters of the actuating mechanisms change during operation and this type of mining machines include the shovel, dragline, and mine hoist. In this regard, it seems appropriate to use the EDD, the parameters of which can also change during operation, as was done, for example, in [13], where the use of proportional stiffness regulation according to a special law is proposed.

From control theory, it is known that the law of motion control, optimal in speed, requires an abrupt change in the control action. In this regard, there is every reason to believe that a similar effect can be achieved by instantly changing the parameters of the EDD in a function, for example, time or an adjustable coordinate to form the desired law of motion of the actuating mechanism of the machine [14]. Using this approach, EDD can be represented in the form of an elastic mechanical system with a variable structure, and the determination of its parameters can be represented as a solution of the inverse dynamic problem based on the differential equation with variable coefficients.

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In the present article, by the example of EDD proposed to reduce the dynamic loads of the traction mechanism of dragline ESh 20.90 in [10], the procedure for synthesizing its parameters based on the concept of inverse dynamic problems is described and the results of a numerical study of its effectiveness are presented.

2. Object and method of investigation

The kinematic scheme of the traction mechanism of a dragline equipped with EDD is shown in Figure 1a. The mathematical model of this mechanism, taking into account the properties of a variable speed drive is described by the following system of differential equations:

\[ J_1 \ddot{\phi}_1 s = M_{dv} - \frac{c_{12} \Delta \dot{\phi}_{13}}{s} - b_{12} \Delta \dot{\phi}_{13} \]  
\[ J_2 \ddot{\phi}_2 s = \frac{c_{12} \Delta \dot{\phi}_{13}}{s} + b_{12} \Delta \dot{\phi}_{13} - M_C \]  
\[ J_3 \ddot{\phi}_3 s = M_{12} - \frac{c_3 \Delta \dot{\phi}_3}{s} - b_3 \Delta \dot{\phi}_3 \]  
\[ \Delta \dot{\phi}_{13} = \phi_1 - \phi_2 - \phi_3 \]  
\[ M_{dv} = C_e I_a \]  
\[ E_{dv} = C_e \phi_1 \]  
\[ I_a = \frac{E_p - E_{dv} k_a}{T_e s + 1} \]  
\[ \frac{(T_a s + 1) k_p}{T_{CR} s (T_p s + 1)} (U_{CR} - I_a k_{CS}) = E_p \]  
\[ (U_{ref} - k_{SS} \phi_1) k_{SR} = U_{CR} \]

Where \( J_1 \) is the inertia of the shaft two motors, and equivalent moment of inertia gearbox and drums; \( J_2 \) is the inertia of the bucket filled with rock; \( J_3 \) is the inertia of damper; \( c_d \) is the stiffness of damper; \( b_d \) is viscous friction of damper; \( c_{12} \) is the stiffness of cable; \( b_{12} \) is viscous friction of cable; \( M_{dv} \) is the drive torque; \( M_C \) is the load torque; \( \dot{\phi}_1, \dot{\phi}_2 \) and \( \dot{\phi}_3 \) are angular velocities of the masses; \( \Delta \dot{\phi}_{13} = \dot{\phi}_1 - \dot{\phi}_2 - \dot{\phi}_3 \) is the velocities of the elastic deformation; \( C_e \) is the voltage constant; \( E_{dv} \) is the DC-machine voltage; \( E_p \) is the converter voltage; \( k_a \) is the gain of the armature circuit; \( T_a \) is the time constant of the armature circuit; \( k_p \) is the converter gain; \( T_p \) is the converter time constant; \( k_{CS} \) is the gain of the current sensor; \( T_{CR} \) is the time constant of the current controller; \( U_{CR} \) is the voltage at the input of the current controller; \( k_{SS} \) is the gain of the speed sensor; \( k_{SR} \) is the coefficient of the speed controller; \( U_{ref} \) is the reference voltage; \( s = \frac{d}{dt} \) is a Laplace operator.

The structural diagram of a mechanical system based on equations (1) - (3) is presented in Figure 1b. In Figure 1 the following notations are adopted: 1 is a reduced moment of inertia of the motor and gearbox; 2 and 3 are guide blocks; 4 is a cable; 5 is a bucket; 6 is an EDD.

We define the parameters of the EDD by setting a differential equation with variable coefficients [14]. Assuming in equations (1) and (2) \( b_{12} = 0, \) \( \phi_3 = 0, \) we solve them with respect to the elastic force \( M_{12} = \frac{c_{12} + b_{12} s}{s} \Delta \dot{\phi}_{13} \)
\[ M_{12}s^2 + \omega_{12}^2(t)M_{12} = 0, \]  

\( \text{where } \omega_{12}(t) = \sqrt{\frac{c_{12}(t)(J_1 + J_2)}{J_1J_2}}. \)

The function \( \omega_{12}(t) \) is conveniently interpreted as a variable natural frequency. Moreover, if \( \omega_{12}(t) \) is a periodic function, then (10) will be called the Hill equation, and if it is harmonic, it will be called the Mathieu equation [15].

We construct equation (10) by determining the law of variation of the natural frequency to ensure the desired character of the damping of the elastic force. Accepting the initial conditions in the form \( M_{12}(0) = M_0 \) and \( M_{12}(0) = 0 \), and assuming that at the initial instant of time the natural frequency of the system is \( \omega_0 = \omega_{12} \) and later it changes stepwise to \( \omega_1 \), we find a solution to equation (10). Using the operator method for the initial period of oscillations, we obtain

\[ M_{12} = M_0 \cos \omega_0 t. \]  

\( \text{Figure 1. Kinematic (a) and structural (b) scheme of the traction mechanism.} \)

The zero value of the elastic force achieved after the end of the transition process will be observed for the first time at a point \( t = \pi / 2\omega_0 \). Changing the natural frequency from \( \omega_0 \) to \( \omega_1 \), and solving (10) we find

\[ M_{12} = -\frac{\omega_0}{\omega_1}M_0 \sin \omega_1 t. \]

The maximum value of the amplitude of the elastic force will be observed at time \( t = \pi / 2\omega_1 \). Switching the structure again from frequency \( \omega_1 \) to \( \omega_0 \), we obtain
$M_{12} = -\frac{\omega_0^2}{\omega_1^2} M_0 \cos \omega_0 t$.

Changing the parameter of the natural frequency at the moment of transition of the elastic force through zero, we will have

$M_{12} = \frac{\omega_0^2}{\omega_1^2} M_0 \sin \omega_0 t$.  \hspace{1cm} (12)

A comparison of the amplitudes of the elastic force obtained based on expressions (11) and (12) shows that a change in the natural frequency of oscillations of a mechanical system allows reducing oscillations by $\frac{\omega_0^2}{\omega_1^2}$ after the third switch, for example, a change in frequency by half leads to a decrease in amplitude by 75%. Consequently, an effective decrease of the amplitude of oscillations is possible with a four-time change of the natural frequency per period. It should be noted that a similar effect can be achieved by instantly changing the coefficient of viscous friction.

The fundamental possibility of changing the natural frequency of the traction mechanism of the excavator is to turn on and off the EDD at the right time. When you turn on the EDD the equivalent stiffness of the elastic mechanical system will vary from $c_{12}$ to $c = \frac{c_{12}c_d}{c_{12} + c_d}$. In accordance with the condition obtained earlier, it is necessary to turn on and off the EDD four times during the oscillation period, therefore, it is necessary to develop a control signal according to the coordinates of the mechanical system. Since the purpose of switching the mechanical structure is to control elastic forces, it is advisable to use a signal of the rate of change of the elastic force $\dot{M}_{12}$. It is known, as the load change on the bucket leads to a decrease in its speed until a force is formed in the cable that exceeds the load torque. It follows that at the initial moment the mechanical system must possess maximum stiffness (equal to the stiffness of the cable). At the time point when the bucket speed begins to increase, the stiffness of the system should be minimized, and damping should be maximized to prevent an increase of the elastic force in the cable. This corresponds to a control based on the switching of the structure at the time when $\dot{M}_{12} = \max$. The construction of such a control method requires a special device for controlling the peak points or another differentiator, which complicates its technical implementation. At the same time, a signal analogous to the switching condition can be generated by measuring the acceleration and speed of the bucket, for which it is enough to use one differentiating link. By controlling the sign of the acceleration of the bucket, it is possible to give signals for turn on (when acceleration is more than zero) and for turn off (when acceleration is less than zero) EDD in the traction mechanism of the excavator.

3. Research and discussion

The effectiveness of the proposed method of controlling the EDD parameters was verified by numerically simulating equations (1) - (9) in the Matlab Simulink environment with the following parameters of the electromechanical system of the dragline traction mechanism similar to real values:

- $J_1 = 572 \text{kg} \cdot \text{m}^2$, $J_2 = 60 \text{kg} \cdot \text{m}^2$, $c_{12} = 7500 \text{N} \cdot \text{m} / \text{rad}$, $b_{12} = 150 \text{N} \cdot \text{m} \cdot \text{s} / \text{rad}$, $C_e = 17.37 \text{V} \cdot \text{s} / \text{rad}$,
- $k_a = 33 \Omega^{-1}$, $T_a = 0.082 \text{sec}$, $k_p = 120$, $T_p = 0.01 \text{sec}$, $k_{SS} = 0.15 \text{V} \cdot \text{s} / \text{rad}$, $k_{CS} = 0.00313 \text{V} / \text{A}$, $T_{CR} = 0.25 \text{sec}$, $k_{SR} = 8$. In the mathematical model, the form of the mechanical characteristic and the limitation of the drive acceleration were taken into account. The choice of parameters EDD was carried out based on the dependence $b_d = 0.115c_d + 472$. [10]. It should be noted that an abrupt change in the structure of a mechanical system leads to significant nonlinearity and can cause self-sustained oscillation. Preliminary calculations showed that the viscous friction parameter $b_d$ should be by 1.5-2 times greater than the calculated value for a given stiffness of the EDD, this condition eliminates the self-sustained oscillation in the mechanical system. In the final version, the acceptable
parameters of the EDD were $J_3 = 1 \text{kg} \cdot \text{m}^2$, $c_d = 7500 \text{N} \cdot \text{m} / \text{rad}$ and $b_d = 2001 \text{N} \cdot \text{m} \cdot \text{s} / \text{rad}$, while the viscous friction $b_d$ was increased by 1.5 times from the calculated value.

The studies were conducted for the start-up mode of the traction mechanism at the rated speed and following load changes on the bucket. Oscillograms of the elastic force in the cable $M_{12}$ and the drive torque $M_{dv}$, the speed of the drive $\omega_1$ and the bucket $\omega_2$ for these modes are shown in Figure 2, where the graphs of transient processes occurring in the mechanism equipped with the usual EDD [10] are indicated by the number 1, in the mechanism equipped with the EDD with variable parameters are indicated by the number 2.

**Figure 2. Oscillograms of transient processes**

Based on the obtained oscillograms, the following conclusions can be drawn:

1. Changing the parameters of the EDD allows us to create an almost straightforward nature of the movement of the bucket during start-up and to reduce the amplitude of the oscillation in the cable. The amplitude of the oscillation elastic force compared with the traction mechanism equipped with the usual EDD reduce by 12% in the load change mode, with no repeated oscillations. This system provides a lower reduction of the bucket speed in the load change mode since at this moment the stiffness of the mechanical system is maximum (when the influence of the EDD is excluded). This circumstance will lead to an increase in the performance of the traction mechanism due to an increase in the path traveled by the bucket during load fluctuations.

2. In a traction mechanism equipped an EDD with variable parameters, the change in drive torque is smoother with a lower oscillating component. In this case, the change in load leads to less overshoot, which does not exceed 7% and is lower than in a mechanism with usual EDD.

It should be noted that more effective indicators of the quality of transient processes of force in the cable can be obtained when using the EDD mechanism with variable parameters, possess higher values of the stiffness and viscous friction coefficients compared with similar parameters obtained in [10]. It was possible to provide lower amplitude values of the elastic force in the cable due to an increase in stiffness and damping coefficient of the EDD by more than two times. A comparison of the displacement of the elastic deformation of the traction mechanism equipped with the usual type EDD with its displacement when using the EDD with variable parameters shows that when the load changes, the elastic deformation of the traction mechanism decreases from 0.53 m to 0.21 m. This circumstance has fundamental importance since lower deformation values reduce the requirements for the dimensions of the EDD and facilitate the search for new technical solutions.
4. Conclusion

The results of the studies have shown high efficiency in the application of the EDD with variable parameters in the traction mechanism of the excavator and the efficiency of the proposed procedure of their synthesis as a solution to the inverse dynamics problem based on differential equation with variable coefficients. The proposed technical solution of the EDD made it possible to obtain higher quality indicators of transient processes at acceptable deformation values of the device. Developed EDD control algorithm based on four-time switching of its stiffness during the oscillation period allowed to decrease the amplitude of the oscillation of the force in the cable and to get a more optimal movement of the bucket with fewer fluctuations of its speed, which will increase its productivity and reliability of the excavator.

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