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Limitation of the dynamic load in the hoist mechanism of the shovel based on the additional drive

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Abstract. In the article, the results of studies of the synthesis of the optimal structure and parameters of the electromechanical system of the hoist mechanism of the rope shovel EKG-8I, which provides the limitation of dynamic loads, are presented. The procedure of structural-parametric synthesis is described, which is based on the determination of the required control actions, independent from the structure of the control system of the hoist mechanism. Control actions were found by solving the inverse dynamic problems for a given type of oscillatory motions of the dipper in the start-up and load change modes. Based on the expression of the obtained time dependencies through the phase coordinates of the system and its combinations, the structure and parameters of the control system were found. The found structure of the control system of the standard electric drive of the hoist mechanism and the additional drive ensures the implementation of a given law of motion of the dipper. The numerical simulation of the electromechanical system of the hoist mechanism with an additional drive confirmed the effectiveness of using the proposed approach of reducing the dynamic loads with maintaining the productivity of the excavator.

1. Introduction

The main reason of the decline in the efficiency and reliability of the rope shovel is significant dynamic loads that occur in the mode of digging and locking the dipper and lead to oscillations of the drive torque in the transmission and actuator [1,2]. The presence of significant load peaks during digging requires limiting the drive torque during start-up and overloads and the formation of the mechanical characteristic of the excavator drive. However, limiting the drive torque during locking does not solve the problem of reducing dynamic loads in the digging mechanism. The kinetic energy stored in the mechanical part of the excavator turns into potential deformation energy, which causes the appearance of mechanical oscillations leading to overloads in mechanical elements, vibrations and overheating. Thus, shortening the excavation cycle makes it necessary to increase the starting and braking drive torque, but to reduce of dynamic loads arising from this, it is necessary to limit the rate of change of the driving and braking forces. The solution of this controversial task is complicated by the presence of a one-sided elastic connection in the form of the cable with weak internal damping. The large length of the cable and actual openness of the kinematic chain connecting the drive of the hoist mechanism with the dipper contribute to increased oscillation. The magnitude of oscillation also depends on the variability of the digging conditions and loads and the large inertia of the drive and gearbox compared to the inertia of the dipper. The significant inertia of the electric drive limits the possibilities of using the standard electric drive control system for limiting dynamic loads. Since the control system does not provide the necessary acceleration values due to the presence of the power-up sensor and limiting the drive torque a form of mechanical characteristic. For example, the use of feedback on the elastic torque in the cable reduces the load in the dipper when the mechanism is locked [3]. But it also reduces the productivity when the mechanism operates on a falling section of the mechanical characteristic with significant fluctuations of the load on the dipper and negatively affects reliability of the electric motor work [4]. The limitation of dynamic loads due to the choice of a special form of mechanical characteristics, ensuring the constancy of power consumption of drive, proposed in [5,6], reduces the productivity of the excavator due to the decrease of the speed of the hoist mechanism under load. The spring-damping devices studied in [7,8] are effective reducing dynamic loads only under constant digging conditions, a change which can lead, in some cases, even to an increase in oscillatory motions and loads. In addition, the use of these devices is often limited by design considerations and overall dimensions of the excavator.

With that in mind, the method proposed in [9] for limiting elastic oscillations and dynamic loads in the traction mechanism of a dragline based on the use of additional drive installed in the runner of the traction cable is interested. When applying this method, the problem of rational distribution of functions between a standard control system and an additional drive arises. For solving this problem it seems appropriate to use the concept of inverse dynamic problems, which allows performing the procedure of structural-parametric synthesis of a motion control system [10].

In this article, using the example of the EKG-8I rope shovel, the synthesis of the optimal structure and parameters of the electromechanical system of the hoist mechanism with an additional drive is performed. The used method of solving the inverse dynamic problems for a given type of oscillatory motions allowed limiting the dynamic loads with maintaining productivity.

2. Object and methods of investigation

As an object of study, we use the hoist mechanism of a rope shovel EKG-8I equipped with a dual circuit control system [6,11]. The dynamic properties of hoist can be described with a sufficient degree of accuracy by a two-mass design scheme [12], in which the first mass is the inertia of drive and gearbox with a drum and the second mass is the reduced mass of the dipper filled with rock. Under the assumptions and notation adopted in [8], a dynamic model describing the behavior of an electromechanical system with regard to a control system can be represented as the following system of differential equations:

$$\begin{cases} U_{SR} = (U_{ref} - \dfrac{K_{SS}\cdot\omega_1}{T_{fs}s+1})\cdot K_{SR}; U_{SR} \le U_{\lim} = (K_1 - \dfrac{K_{SS}\cdot\omega_1}{T_{fs}s+1})\cdot(\cos a/\sin a) + K_2; U_{\lim} \le U_{ST}; \\ U_{CR} = \dfrac{\left[U_{SR} - (\dfrac{K_{CS}}{T_{fc}s+1} + \dfrac{0.3K_{CS}s}{T_{fc}s+1})I_a\right]\cdot(T_a s+1)}{T_{CR}}; E_p = \dfrac{U_{CR}\cdot K_b}{T_b s+1}; E_{dv} = C_e\cdot\omega_1; \\ I_a = \dfrac{E_p - E_{dv}}{T_a s+1}\cdot K_a; M_{dv} = C_e\cdot I_a. \end{cases} \tag{1}$$

$$\begin{cases} J_1 s^2\varphi_1 = M_{dv} - M_{12} - M_b - M_t; J_2 s^2\varphi_2 = M_{12} + M_b - M_C \\ M_{12} = c_{12}(\varphi_1 - \varphi_2); M_b = bs(\varphi_1 - \varphi_2); \\ M_t = as\varphi_1 \end{cases} . \tag{2}$$

where φ_1 is the coordinate motion of first mass; φ_2 is the coordinate motion of second mass; J_1 and J_2 is the moment of inertia of first mass and second masses; c_{12} is the stiffness of the elastic element; a is the coefficient of proportionality; b is the coefficient of viscous friction in elastic element; M_{dv}

is the drive torque; M_C is the load of mechanism. Notation of differential equations (1) is taken from [8].

To find the control action that provides the minimum load of the hoist mechanism based on the concept of inverse dynamic problems, we define the elastic displacement of the reduced masses as an exponential equation:

$$\Delta\varphi_{12} = C_1 e^{\lambda_1 t} + C_2 e^{\lambda_2 t}, \tag{3}$$

where $\Delta\varphi_{12}$ is the elastic displacement of the reduced masses; λ_1 and λ_2 are real or complex conjugate roots of the equation, the choice of which is carried out on the basis of the following conditions: $\operatorname{Re}\lambda_i < 0$; $\lambda_1\lambda_2 > \omega_{12}^2$ and $\lambda_1 \neq \lambda_2$; C_1 and C_2 are constant coefficients determined by the initial conditions.

Let us take into account the large inertia of the drive motor and gearbox, the definition of the control action will be carried out without considering the system of equations (1) describing the dynamic properties of the electrical part of the system. Solving equations (2) with respect to elastic displacements $\Delta\varphi_{12}$ under the condition $M_t = 0$ and substituting the first and second derivatives of expression (3) into it, we obtain the required control action taking into account the load torque M_C [8]. Considering that the load torque, in the general case, is a non-stationary random process, which is compensated by the electric drive control system in accordance with the mechanical characteristic, without loss of generality, can be taken equal $M_C = 0$. After a series of mathematical transformations, we obtain an expression that determines the required control torque of the drive for the formation of a given law (3) of oscillatory motion:

$$J_1 K_1 C_1 e^{\lambda_1 t} + J_1 K_2 C_2 e^{\lambda_2 t} = M_m. \tag{4}$$

Here $K_1 = \lambda_1^2 + \omega_{12}^2 + \lambda_1 b_{12} J_{12}$ and $K_2 = \lambda_2^2 + \omega_{12}^2 + \lambda_2 b_{12} J_{12}$; $J_{12} = \dfrac{J_1 + J_2}{J_1 J_2}$; $\omega_{12}^2 = \dfrac{c_{12}(J_1 + J_2)}{J_1 J_2}$ is a frequency of oscillations of a two-mass system.

Define the structure and parameters of the controller that implements the found control action (4). As the inputs of this regulator can be selected any phase coordinates or its combinations. To determine the dependence the velocity of the first mass with a given type of oscillatory motion (3), we substitute the expression (4) into the equation of motion of the first mass J_1 in (2), integrating it:

$$\dot{\varphi}_1 = \frac{B_1}{\lambda_1} C_1 e^{\lambda_1 t} + \frac{B_2}{\lambda_2} C_2 e^{\lambda_2 t} + A_1. \tag{5}$$

Here $B_1 = \dfrac{K_1 J_1 - c_{12} - b_{12}\lambda_1}{J_1}$ and $B_2 = \dfrac{K_2 J_1 - c_{12} - b_{12}\lambda_2}{J_1}$; A_1 are an integration constant determined by the initial conditions of motion.

The relationship of the control action with the acceleration of oscillatory motion is determined by time differentiation expression (3):

$$\Delta\ddot{\varphi}_{12} = \lambda_1^2 C_1 e^{\lambda_1 t} + \lambda_2^2 C_2 e^{\lambda_2 t}. \tag{6}$$

Expressing from (5) and (6) the time functions through the coordinates of a two-mass system, we find:

$$C_1 e^{\lambda_1 t} = \frac{\lambda_1 B_2 \Delta\ddot{\varphi}_{12} - \lambda_1 \lambda_2^3 \dot{\varphi}_1 + \lambda_1 \lambda_2^3 A_1}{B_2 \lambda_1^3 - B_1 \lambda_2^3} \text{ and } C_2 e^{\lambda_2 t} = \frac{\lambda_1^3 \lambda_2 \dot{\varphi}_1 - \lambda_2 B_1 \Delta\ddot{\varphi}_{12} - A_1 \lambda_1^3 \lambda_2}{B_2 \lambda_1^3 - B_1 \lambda_2^3}. \tag{7}$$

Substituting (7) into (4), we determine the structure and parameters of the controller:

$$M_{dv} = K_C \Delta\ddot{\varphi}_{12} + K_V \dot{\varphi}_1 + M_0, \tag{8}$$

where $K_C = \dfrac{J_1(K_1 B_2 \lambda_1 - K_2 \lambda_2 B_1)}{B_2 \lambda_1^3 - B_1 \lambda_2^3}$ and $K_V = \dfrac{J_1(K_2 \lambda_2 \lambda_1^3 - K_1 \lambda_1 \lambda_2^3)}{B_2 \lambda_1^3 - B_1 \lambda_2^3}$ are the coefficients of proportionality of the controller; $M_0 = \dfrac{J_1(A_1 K_1 \lambda_1 \lambda_2^3 - K_2 A_1 \lambda_2 \lambda_1^3)}{B_2 \lambda_1^3 - B_1 \lambda_2^3}$ – the drive torque determining the magnitude of the steady-state value of the angular velocity of the first mass.

The found control action (8) can be implemented as a closed-loop control system with feedbacks on the angular velocity of the first mass and acceleration of the elastic motion of the second mass with the gain K_V and K_C, respectively (Figure 1a). The form of feedback on the speed of the first mass, present in the standard control system already, provides limiting its reference value. Some additional feedback is on the elastic torque (Figure 1b), the transfer function of which can be obtained on the basis of structural transformation [13]. Transferring the output signal of the feedback circuit shown in Figure 1a through the integrating link and its input through the elastic link, we obtained:

$$W(s) = \frac{\varphi_3}{M_{12}} = \frac{K_C}{J_1 b_{12} s + J_1 c_{12}} \quad . \tag{9}$$

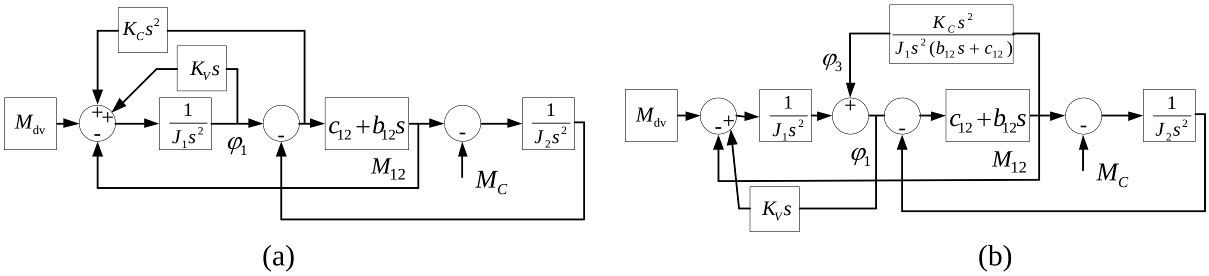


Figure 1. Structural scheme of control systems based on feedback.

Due to the fact that a given speed is determined by the signal U_{ref} entering in the control system, the drive torque M_0 in further calculations can be ignored. Additional feedback on the elastic torque can be implemented as a mechanical device built into the design of a hoist mechanism. The mechanical device consists of a weightless beam m reciprocating with an additional drive M (Figure 2a), which can be installed on the boom 3 or on the swinging platform, between the hoist drum 1 and the heading block 2 under the string of the cable hoisting the bucket 4 (Figure 2b).

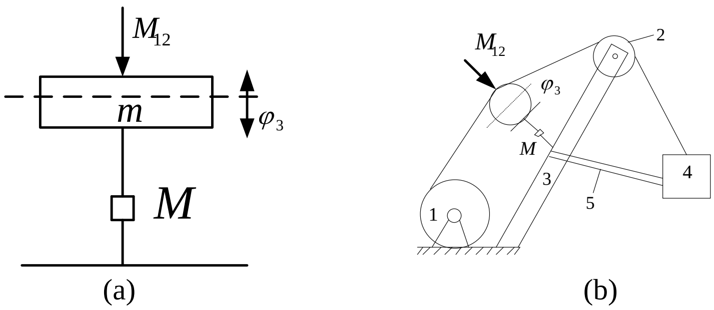


Figure 2. Kinematic scheme (a) and location (b) of an additional drive

In Figure 3 the structural scheme of the electromechanical system of the hoist mechanism with the additional drive is shown. A structural scheme obtained on the basis of the systems of equations (1), (2) and dependencies (8) and (9).

To test the effectiveness of the proposed system for reducing dynamic loads, we will perform the calculation of the main parameters of the structural scheme shown in Figure 3. The parameters of the current loop are determined by the standard method. Finding gain speed feedback can be performed as follows.

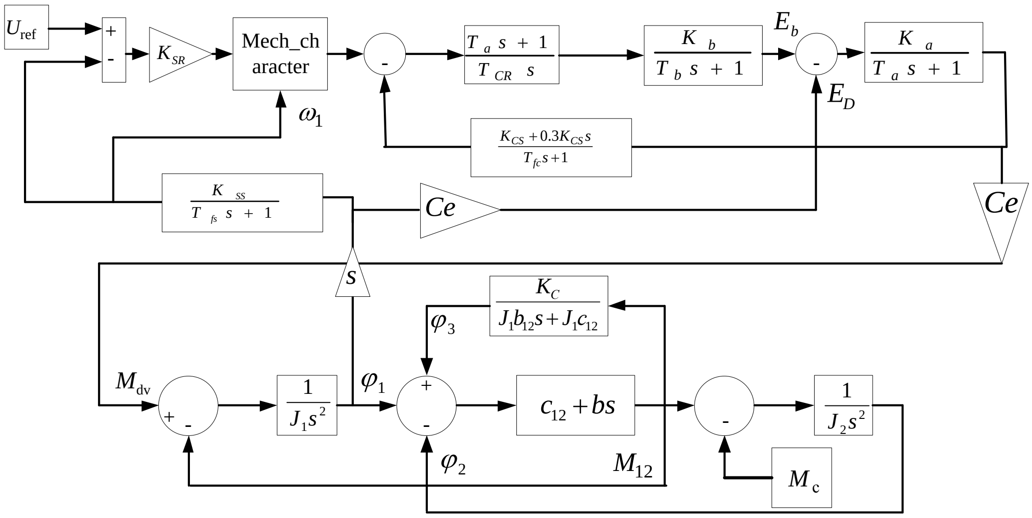


Figure 3. Structural scheme of the electromechanical system of the hoist mechanism.

At the rated speed of the hoist mechanism, equal to $\omega_{rate} = 78.5 rad/\sec$, the value of the signal in the feedback circuit will be $M = \omega_{rate} \cdot K_V$, and the value of the correction coefficient will be determined using the expression $K = U_{ref} / (\omega_{rate} \cdot K_V)$. Then the value of the gain feedback speed can be found from the product $K_{ss} = K_V \cdot K$. The remaining parameters of the electromechanical system of the hoist mechanism of the EKG-8I excavator were taken: $U_{ref} = 10V$, $K_{SR} = 9.67$, $K_1 = 10$, $K_2 = 6$, $\cos a = \sin a = 1.05 rad$, $U_{ST} = 10V$, $T_{fs} = 0.005\sec$, $K_{CS} = 0.00657 V/A$, $K_{SS} = 0.127 V/rad$, $T_a = 0.0635\sec$, $T_{CR} = 2T_b K_a K_b K_{CS} = 0.2\sec$, $K_b = 76.8$, $K_a = 19.68$, $T_b = 0.01\sec$, $T_{fc} = 0.005\sec$, $C_e = 6.838 V \cdot \sec$, $J_1 = 51.1 kg \cdot m^2$, $J_2 = 5.4 kg \cdot m^2$, $c_{12} = 160 N \cdot m / rad$, $b_{12} = 8 N \cdot m \cdot s / rad$, $\lambda_1 = -6$, $\lambda_2 = -7$, $K_V = -162$, $K_C = 41$. In the process of modeling the operation of the power-up sensor, the saturation of the controllers and the limitation on the rate of change of the armature current, the form of the mechanical characteristic of the drive were taken into account; the additional drive was represented as a gain inertia-free link.

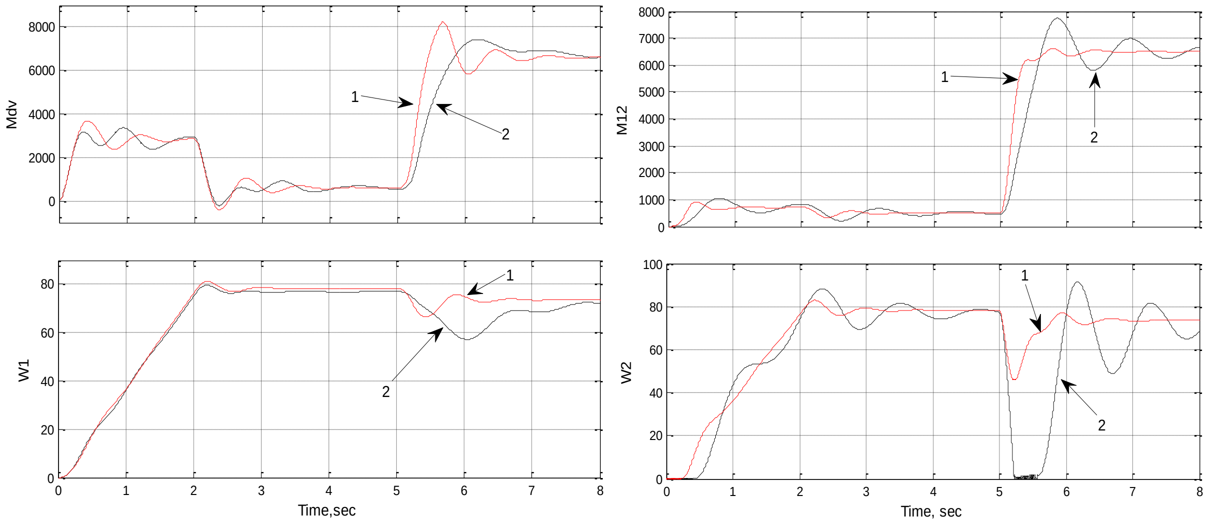
3. Research and discussion

The test of the effectiveness of the synthesized system for reducing dynamic loads compared to the standard control system of the hoist mechanism of a rope shovel was carried out in the Matlab Simulink package for start-up mode ($M_C = 0.1 M_{rate}$) with a further step load $M_C = 1.2 M_{rate}$. At the same time, oscillograms of changes torque M_{dv} and speed ω_1 of the drive (Figure 4a), cable load M_{12} and bucket speed ω_2 (Figure 4b) were recorded. On the oscillograms curve (1) corresponding to the application of the proposed load reduction system based on an additional drive, and curve (2) is the standard control system.

The performed numerical simulation leads to the following conclusions:

1. The proposed system for reducing dynamic loads on the basis of an additional drive in the start-up mode provides a smoother increase the speed of movement of the bucket during acceleration to the rating value. In this case, the decrement of oscillations decreased twice as compared with the standard control system. The duration of the transient process was reduced to two times until the steady-state speed was reached. The value of the overshoot of the speed bucket was reduced by 6%. There was also

a reduction in the transient time for the elastic torque, the steady-state value of which was achieved almost twice as fast as the standard control system. At the same time, an increase in overshoot for the driving torque by 14% compared with the standard control system was recorded.



(a) (b)

Figure 4. Oscillograms of changing the moments and speeds.

2. In the load change mode, a significant increase in the efficiency of the proposed system for reducing dynamic loads was observed compared to the standard control system. In this case, the movement of the bucket occurs without stopping with a deviation of speed from the steady-state value by 38%, the curve of the speed ω_2 change approaches an aperiodic law, and the value of the overshoot decreases by 15%. Change of the load with the use of a standard control system led to the stopping of the bucket. The change of the elastic torque M_{12} as close as possible to the aperiodic law, the value of the overshoot torque was reduced by 15%. However, oscillations of the drive torque with the proposed system for reducing loads increased by 10%, which can be explained by the action of an additional drive providing faster transients in the elastic system. The increase in oscillations M_{dv} in the proposed system is insignificant compared with [4], where the limitation of loads is associated with a significant increase in amplitude and a change in the sign of oscillations of the drive torque.

3. The total displacement of the additional drive in the mode of changing the load was 0.4 m (taking into account 2 hoisting cables when moving each additional drive is 0.2 m), and the maximum speed of the drive extension reached 1.4 m/s. It should be noted that the value of the displacement of an additional drive from the action of a constant load can reach significant values, which must be taken into account in the practical implementation of this method of limiting dynamic loads.

4. Conclusion

The performed studies showed high efficiency of using an additional drive to reduce the dynamic loads of the hoisting mechanism. Capabilities of the proposed method of structural-parametric synthesis based on inverse dynamic problems to solve the problem of rational distribution of functions between the standard control system and mechanical elements were demonstrated. The proposed method allowed obtaining the desired law of motion of a controlled object in a complex electromechanical system. The proposed system of the hoist mechanism with an additional drive allowed reducing the dynamic loads in the cable and improving the accuracy of maintaining the reference speed of the bucket. The proposed control system improved the reliability and durability of the actuator and sustained the productivity of the mining excavator.

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