

Investigation of dynamics of excavator digging mechanism with additional drive

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Abstract. In the article, an active method for reducing dynamic loads in the digging mechanism of the ESC 20.90 excavator is presented on the basis of additional drives built into the mechanical system. A kinematic scheme for realizing this method is described, a mathematical model of the digging mechanism equipped with additional drives is presented, as well as control algorithms are obtained by solving the inverse problem of dynamics according to the desired law of motion of the actuating mechanism. The method of mathematical modeling shows that the use of additional drives makes it possible to obtain monotonous transient processes of the speed of the actuating mechanism and the torque in the elastic element in the start-up mode, in the load-up and load-down modes.

1. Introduction

The use of a modern electric drive in the mechanisms of excavators allows one to maintain with a high accuracy the referenced speed of movement in transient operating modes and to form the required rate of change of torque [1-5]. At the same time, the presence of cables in the mechanical system of excavators makes it necessary to take into account their elastic compliance when creating electric drive control systems in order to reduce dynamic loads and limit the oscillation movements of the actuating mechanism (bucket). The use of feedbacks on the load in the elastic element proposed in works [6, 7] makes it possible to limit the amplitudes of the torque and speed oscillations. But the degree of damping of the oscillations decreases in comparison with the standard control system of the electric drive, which increases the transient time and reduces the productivity of the digging mechanism. In addition, the considerable inertia of the electric motor with the gearbox and the alternating nature of its loading, connected with the control of oscillations, can lead to undesirable phenomena in the operation of the electric drive and to a decrease in reliability.

In this connection, special elastic-damping devices described in [8, 9] that can be built into the mechanical system of the excavator and allow one to change the stiffness and damping of the elastic element can be of some interest. The disadvantage of this oscillation damping system is the complication of the construction of the digging mechanism due to the transition from the two-drum version of the winch to the one-drum type and the need for an equalization unit located in the digging working area, which can lead to its damage and premature wear of the cable. In order to get rid of this drawback, an elastic-damping device with constant stiffness and damping coefficients, installed under the cable string between the guide block and the drum, was proposed in [10]. At the same time, the proposed damping device can effectively reduce dynamic loads only for specific digging conditions.



In this paper, we propose an active method of changing the stiffness of the elastic element between the drive and the bucket of the traction mechanism of the walking excavator ES 20.90 on the basis of additional drives. The introduction of active elements will allow controlling the stiffness of the elastic element, and, consequently, the level of oscillation movements and dynamic loads under various operating conditions.

2. Object and methods of investigation

The kinematic scheme of the traction mechanism of the walking excavator ES 20.90 with additional drives is shown in Figure 1. Additional drives 4 in the form of hydraulic cylinders with reciprocating motion or linear electric AC or DC drives allow changing the position of the additional blocks 3 installed in the running part of the traction cable between the drums 2 and the guide blocks 5 and the stiffness of the elastic connection.

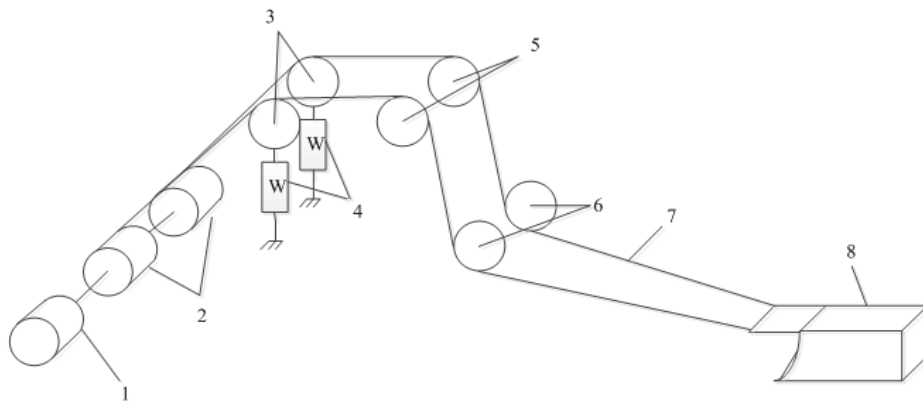


Figure 1. Kinematic scheme of the traction mechanism with additional drives: 1 - electric motor with a gearbox; 2 - drums of the traction winch; 3 - additional blocks; 4 - additional drives; 5 - guide blocks; 6 - guidance blocks, 7 - traction cable; 8 - bucket

We perform the synthesis of control actions from the side of additional drives, representing the latter in the form of inertia-free elements. With the assumptions made in [10], the design scheme of the digging mechanism can be represented as a two-mass mechanical system, which is described by the equations:

$$\begin{cases} J_1 \ddot{\varphi}_1 + c_{12} \Delta \varphi + b_1 \Delta \dot{\varphi} = M_{dv}; \\ J_2 \ddot{\varphi}_2 - c_{12} \Delta \varphi - b_1 \Delta \dot{\varphi} = -M_c; \\ \Delta \varphi = \varphi_1 - \varphi_2, \end{cases} \quad (1)$$

where φ_1 - the coordinate of the displacement of the first mass (engine); φ_2 - the coordinate of the movement of the actuating mechanism; J_1 - the moment of inertia of the motor and gearbox; J_2 - the moment of inertia of actuating mechanism; M_{dv} and M_c - the driving torque and the torque of the load; c_{12} - coefficient of stiffness of the elastic element; b_1 - coefficient of viscous friction.

Taking into account the fact that the moment of inertia of the motor and the gearbox is almost an order of magnitude greater than the moment of inertia of the actuating mechanism [10] and the elastic oscillation has practically no effect on the motion of the first mass, we determine the control actions of the additional drives using the second equation of system (1).

To find these control action, we use the concept of inverse dynamics problems [11, 12] by specifying the exponential law of change of the coordinate:

$$\varphi_2 = C_1 e^{\lambda_1 t} + C_2 e^{\lambda_2 t}, \quad (2)$$

where C_1 and C_2 are integration constants; λ_1 and λ_2 are the real or complex roots of the equation, such that $\text{Re}\lambda_i < 0$.

Solving the second equation of system (1) is responsive φ_2 and substituting the coordinate (2) and its first derivative into it, we obtain the law of motion of the first mass under the initial conditions $\varphi_2(0) = \varphi_2$ and $\dot{\varphi}_2(0) = \dot{\varphi}_2$.

$$(\lambda_1^2 + \frac{b_1}{J_2}\lambda_1 + \omega_{02}^2)C_1e^{\lambda_1 t} + (\lambda_2^2 + \frac{b_1}{J_2}\lambda_2 + \omega_{02}^2)C_2e^{\lambda_2 t} + \frac{M_c}{J_2} - \frac{b_1}{J_2}\dot{\varphi}_1 = \omega_{02}^2\varphi_1, \quad (3)$$

where ω_{02} is the frequency of oscillation of the second mass.

Expressing in (3) the function of time $C_1e^{\lambda_1 t}$ and $C_2e^{\lambda_2 t}$ through the coordinate and its derivative, we obtain the following dependence for finding the control actions:

$$(1 - \frac{\lambda_1\lambda_2}{\omega_{02}^2})\varphi_2 + (\frac{\lambda_1 + \lambda_2}{\omega_{02}^2} + \frac{b_1}{c_{12}})\dot{\varphi}_2 - \frac{b_1}{c_{12}}\dot{\varphi}_1 + \frac{M_c}{c_{12}} = \varphi_1. \quad (4)$$

The coefficients of the coordinate φ_2 and its derivative in expression (4) can be interpreted as the feedback coefficients: $K_A = 1 - \frac{\lambda_1\lambda_2}{\omega_{02}^2}$ - in the coordinate; $K_V = \frac{\lambda_1 + \lambda_2}{\omega_{02}^2} + \frac{b_1}{c_{12}}$ - on the speed of

movement of the actuating mechanism (see Figure 2 on which ω_1 and ω_2 are the angular velocities of the masses). In accordance with the stability conditions, these coefficients should be negative.

The presence of rigid feedback on the coordinate of the actuating mechanism with the gain K_A assumes constant movement of additional drives, however, the drives work efficiently only until the cable string is fully straightened and have a limited travel path; therefore, the constant use of rigid feedback is inexpedient and it must be connected only in transient operation modes.

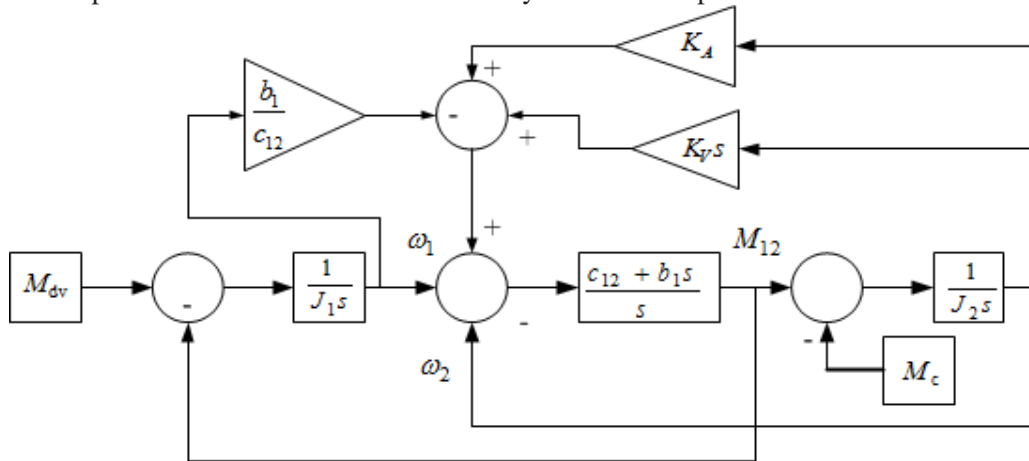


Figure 2. Structural shreme of control system

Control action, formed by a flexible feedback on the speed of movement of the actuating mechanism with the gain K_V , for a digging mechanism with monotonic transient processes has a negligible value (acceleration and deceleration occurs according to inclined characteristics), is effective only in transient modes of operation and its realization is quite possible. The coefficient K_V depends on the parameters λ_1 and λ_2 - the given exponential law of motion. To find K_V , we obtain the transfer function $W(p) = \frac{\omega_2}{\omega_1}$. After rolling the feedback loops of the structural scheme, the transfer function will have the form:

$$W(p) = \frac{\omega_2}{\omega_1} = \frac{(1 - p \frac{b_1}{c_{12}})(c_{12} + b_1 p)}{(J_2 + b_1 K_V)p^2 + (c_{12} K_V + b_1) + c_{12}}. \quad (5)$$

Equating the denominator in (5) to zero for a given oscillation damping coefficient, we find the value of the flexible feedback gain:

$$K_V = \frac{2\xi\sqrt{c_{12}J_2}}{c_{12}}. \quad (6)$$

Assuming equality $\lambda_1 = \lambda_2$, we obtain expression:

$$\lambda_1 = \lambda_2 = \frac{-2\xi\sqrt{c_{12}J_2} - b_1}{2J_\gamma}. \quad (7)$$

For the traction mechanism with the parameters given in [10] ($J_1 = 575$ and $J_2 = 60 \text{ kg} \cdot \text{m}^2$; $c_{12} = 7500 \frac{\text{N} \cdot \text{m}}{\text{rad}}$; $b_1 = 150 \frac{\text{N} \cdot \text{m}}{\text{sec}}$), the calculated values of the roots of the differential equation and the feedback coefficient were $\lambda_1 = \lambda_2 = -10,4$ and $K_v = 0,1264$, respectively. The feedbacks obtained allow changing the response of the mechanical system to the control and disturbing actions, the process of acceleration and the change in the speed of the actuating mechanism in the case of a load is of an aperiodic nature, and there are no oscillations.

Estimation of the efficiency of additional drives will be performed on the basis of the refined electromechanical model of the traction mechanism developed and described in [13]. The structural scheme of the electromechanical model of the traction mechanism with synthesized feedbacks is shown in Figure 3.

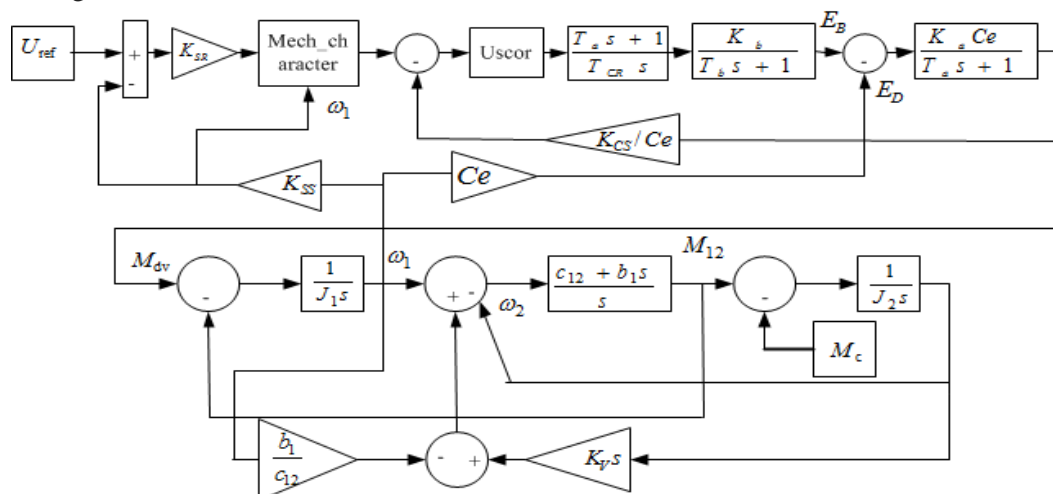


Figure 3. Structural scheme of the electromechanical model of the traction mechanism with feedbacks

The calculated values of the model parameters were: $U_{scor} = -10 - 1,3; U_{ref} = 0 \dots 10; K_{SS} = 0.151; K_1 = 10; K_2 = 8; K_{SR} = 8; K_{CS} = 0.00313; T_{CR} = 0.864; K_b = 120; T_b = 0.01 \text{ sec}; C_e = 17.37; K_a = 33; T_a = 0,082 \text{ sec}; J_1 = 572, J_2 = 60 \text{ kg} \cdot \text{m}^2; c_{12} = 7500 \frac{\text{N} \cdot \text{m}}{\text{rad}}; b_1 = 150 \frac{\text{N} \cdot \text{m}}{\text{sec}}.$

3. Research and discussion

To test the efficiency of using additional drives, a numerical simulation of the electromechanical system shown in Fig. 3, for start-up modes, load-up and load-down modes. At the same time, the

speed ω_1 and torque M_{dv} of the motor, the speed of the actuating mechanism ω_2 and the load in the elastic element M_{12} were monitored. The oscillograms of the transient processes are shown in Figure 4, which adopted the following designations: 1 - system with an additional drive; 2 - system with an elastic-damping mechanism (UDM), described in [10]. It is established that in a system with additional drives, the transient processes of the speed of the actuating mechanism ω_2 and the load in elastic element M_{12} are of aperiodic nature, a rapid restoration of the new steady-state value of the speed of the bucket after the application of the load is ensured. The amplitude of the oscillations is reduced by 12% in the start-up mode and by 13% - in load-up mode, in comparison with the system equipped with UDM. The change in the speed of the actuating mechanism during start-up is practically linear, the deviation from the set value in the load-up mode is 13%, in the mechanical system with UDM, this deviation reaches 58%. At the same time, for both systems, the transition processes are approaching to a monotonous form, which minimizes the dynamic loading of the equipment elements and increases the service life of the main parts of the digging mechanism. The transient processes of the motor torque M_{dv} and speed ω_1 for the two systems in the start-up and load-up modes differ insignificantly, which is due to the localization of oscillations in the elastic system and minimization of their influence on the first mass.

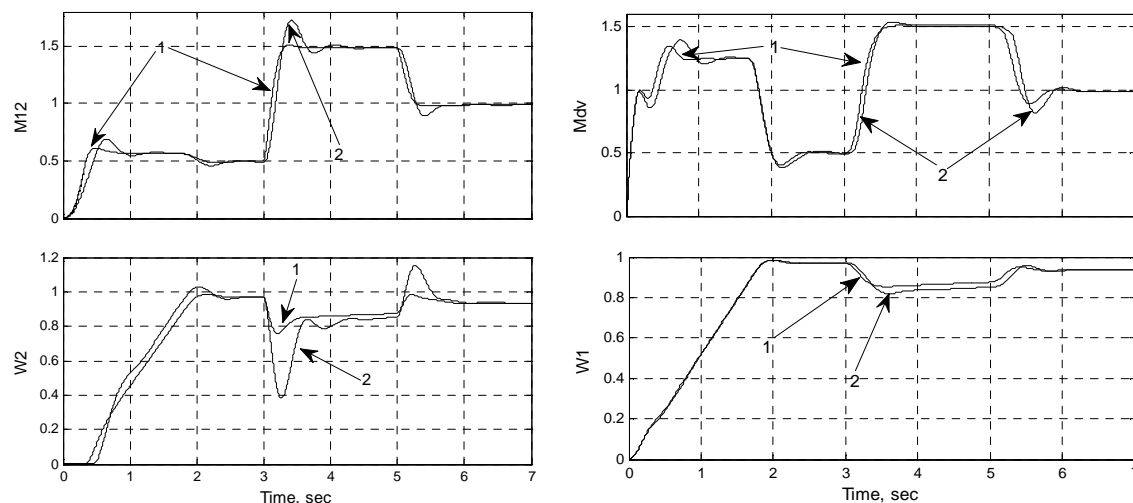


Figure 4. Oscillograms of transient processes of speed ω_1 and torque M_{dv} of the motor, bucket speed ω_2 and load in the elastic element M_{12} .

Comparison of the traction mechanism with additional drives and with UDM shows that additional drives provide an effective damping of the oscillations of the elastic torque and the speed of the actuating mechanism in the modes of start-up and load-up, maximally approaching the speed and torque curves to aperiodic form. Advantage of the proposed system is the formation of corrective actions directly at the place of occurrence of elastic oscillation by changing the coordinate of one tower of the towing system supports, and also the possibility of its exclusion from the mechanism by fixing the movable part of additional drives in case of failure.

4. Conclusion

The performed studies showed high efficiency of use with additional drives to limit elastic oscillation and dynamic loads in the excavator digging mechanism. Reduction of the oscillation time and the approach of transient processes of torque and speed to aperiodic form allows one not only to reduce the dynamic loads in actuating mechanism, but also to increase the resource and reliability of operation. As additional drives are built into the existing winch design, there is always the possibility

of excluding them from the kinematic scheme by fixing the moving part and moving to the standard control scheme. This circumstance is especially important for cyclic machines built into the continuous technological chain, where the complication of the mechanical system of the digging mechanism should not affect the operability, reliability and productivity of the excavator.

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