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Investigation of control system of traction electric drive with feedbacks on load

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Abstract. In the article, by the example of a walking excavator, the results of a study of a control system of traction electric drive with a rigid and flexible feedback on the load are mentioned. Based on the analysis of known works, the calculation scheme has been chosen; the equations of motion of the electromechanical system have been obtained, taking into account the elasticity of the rope and feedbacks on the load in the elastic element. A simulation model of this system has been developed and mathematical modeling of the transient processes to evaluate the influence of feedback on the dynamic characteristics of the mechanism and its efficiency of work was carried out. It is shown that the use of rigid and flexible feedbacks makes it possible to reduce dynamic loads in the traction mechanism and to limit the elastic oscillation of the executive mechanism in transient operating modes in comparison with the standard control system; however, there is some decrease in productivity. It has been also established that the sign-variable of the loading of the electric drive, connected with the opening of the backlashes in the gearbox due to the action of feedbacks on the load in the elastic element, under certain conditions, can lead to undesirable phenomena in the operation of the drive and a decrease in the reliability of its operation.

1. Introduction

The electric drive is increasingly used in modern machine building to realize the movement of any complexity, including controlled transients associated with acceleration, braking and reversing the executive mechanisms of technological and transport vehicles. The increase in the speeds and loads of these machines, as well as the toughening of the requirements for the accuracy and reliability of their operation, make it necessary to take into account when creating electric drive control systems the elastic compliance of the actuators in order to reduce dynamic loads and limit oscillation that disrupt the accuracy of work, reduce the strength of the basic elements and reliability. The solution of this problem is complicated by the fact that the executive mechanisms of the controlled machines are both sources of elastic oscillations and objects of protection against them, which requires consideration when creating control systems for the motion of close interaction of drives with control objects [1-3].

This problem is especially important when creating machines characterized by large masses, sizes, power, the presence of clearly expressed elastic elements and operating under severe dynamic conditions (walking and rotary excavators, mine hoisting machines, elevators, cranes, rolling mills, etc.) [4-7].

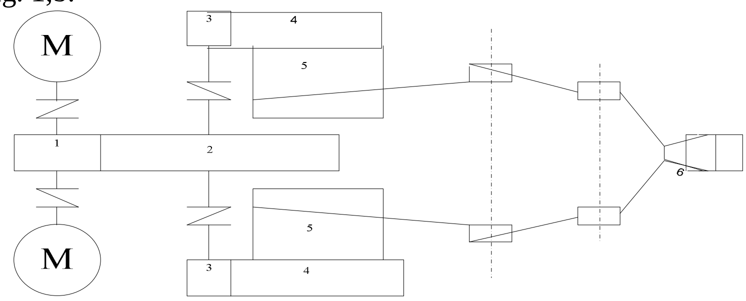
To reduce dynamic loads and to limit elastic vibrations, special vibration protection systems are used in the form of spring and pneumatic damping devices built into the structure of controlled machines [8-11]. The disadvantages of vibration protection systems are the complexity of the design of machines and a narrow range of frequencies of effective operation. At present, an active method of reducing dynamic loads and damping of elastic oscillations, based on the use of feedbacks on the load in the elastic element introduced into the electric drive control system, is gaining increasing use [1,3,6,8,12-14]. The modern high-speed electric drive allows one to create with a high accuracy the specified character of motion on the output shaft of the motor and control the torque in a wide range of disturbance frequencies.

In the present article, using the example of the walking excavator ESh 20.90, the results of studies on the efficiency of the use of rigid and flexible feedbacks on the load in the elastic element for reducing dynamic loads in the traction mechanism and limiting the oscillation movements of the executive mechanism in transient operation modes are given.

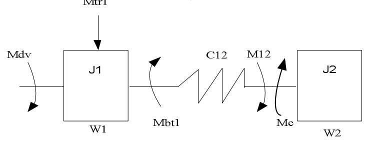
2. Object and methods of investigation

The kinematic scheme of the traction mechanism of the excavator ESC 20.90 is shown in fig. 1,a. In this scheme the following designations are accepted: M - electric motors; 1 - a motor shaft-gear wheel; 2 - chevron wheel; 3 - intermediate shaft-gears; 4 - gears of the second gear stage; 5 - drum; 6 - bucket. Movement from the drums to the bucket is transmitted by means of a cable, guiding and guide blocks. Motion control is carried out using an electric drive with a standard two-loop subordinate control system: thyristor converter - motor (TP-D). The electric drive generates an excavating mechanical characteristic, which ensures the limitation acceleration at start-up and the motor torque.

In drawing up the dynamic model of the mechanical part of the traction mechanism, let us take the following assumptions: the masses of the main elements of the traction mechanism will be assumed to be lumped; the stiffness of the connecting shafts of the gearbox is many times higher than the stiffness of the cable; the cable is represented in the form of a weightless elastic system with constant stiffness and damping coefficients; the authors will not take into account the backlashes in the gearbox; the inertia moment of the bucket is assumed to be constant; the authors take into account the internal friction of the rotating elements of the gearbox. Under the assumed assumptions, the design scheme of the traction mechanism of the excavator can be represented in the form of a two-mass mechanical system, shown in fig. 1,b.



a)



b)

Figure 1. A kinematic and calculation scheme of the traction mechanism of the excavator

In the design scheme, the following designations are used: J_1 – the inertia moment of the two DC machine, the pinion shaft 1, the chevron wheel 2 and the gear shafts 3, the wheels 4 and the drums 5, recakculated to the motor shaft; J_2 – the inertia moment of the bucket half filled with rock, M_{dv} –

torque of the both DC machine; M_{tr1} – the torque of internal friction in the gearbox; M_{bt1} – the torque of viscous friction in the elastic element; M_{12} – the torque transmitted through the elastic element; M_c – the torque created by the load on the bucket; c_{12} – stiffness of the cable; ω_1, ω_2– angular velocities.

The electric drive of the mechanism is constructed according to a standard two-loop subordinate control system with feedbacks on current and speed. When constructing the differential equations of motion, let us take into account the limitations imposed on the electric drive: the mechanical characteristic formed by the DC machines; the presence of a rigid feedback on the current that participates in the formation of static characteristics and flexible feedback, providing the required change in the rate of current rise in transient modes and limiting the maximum acceleration at start-up. Taking into account the adopted assumptions, the system of differential equations of motion of the two-mass electromechanical system of the traction mechanism will take the form:

$$\begin{cases} U_{SR} = (U_{ref} - K_{SS}\cdot\omega_1)\cdot K_{SR}; U_{SR} \le U_{\lim} = (K_1 - K_{SS}\cdot\omega_1)\cdot(\cos\alpha/\sin\alpha) + K_2; \\ U_{CR} = \dfrac{(U_{SR} - (K_{CS}\cdot I_a))(T_a s + 1)}{T_{CR}s}; E_p = \dfrac{U_{CR}\cdot K_b}{T_b s + 1}; E_{dv} = C_e\cdot\omega_1; \\ I_a = \dfrac{E_p - E_{dv}}{T_a s + 1}K_a; M_{dv} = C_e\cdot I_a; J_1 s\omega_1 = M_{dv} - M_{12} - M_{bt1} - M_{tr1}; \\ J_2 s\omega_2 = M_{12} - M_c + M_{bt1}; M_{12} = \dfrac{c_{12}}{s}(\omega_1 - \omega_2); M_{bt1} = b_1(\omega_1 - \omega_2); M_{tr1} = a_1\cdot\omega_1, \end{cases} \quad (1)$$

where U_{SR}– the voltage at the output of the speed controller; U_{ref}– reference voltage; $U_{\lim}$– constraint voltage to form a falling section of the excavator characteristic; U_{CR}– voltage at the output of the current controller; E_p– converter voltage; E_{dv}– DC-machine voltage; I_a– the armature current; K_1 – tuning coefficient of mechanical characteristics; K_2 – limitation of the falling part of mechanical characteristic; K_{SS}– gain of the speed sensor; K_{SR}– coefficient of the speed controller; K_{CS}– gain of the current sensor; K_a – the gain of the armature circuit; K_b – converter gain; α – angle of inclination of the falling part of the mechanical characteristic; T_a– time constant of the armature circuit; T_{CR}– time constant of the current controller; T_b– converter time constant; C_e– voltage constant; c_{12}– stiffness of the elastic element; a_1– coefficient of proportionality; b_1– coefficient of viscous friction; ω_1, ω_2– angular velocities of the masses; s – Laplace operator.

A block diagram constructed on the basis of the system of equations (1) in the visual simulation environment of *Matlab Simulink* is shown in fig. 2. The structural scheme takes into account the form of the mechanical characteristic of the electric drive – the Mech_character unit. In the block, the value of the voltage limitation at the current controller input is calculated in accordance with the second equation in the system (1). Block U_{scor} limits the mismatch between the set voltage value at the current controller input and the feedback circuit, forming the required acceleration of the drive in transient modes.

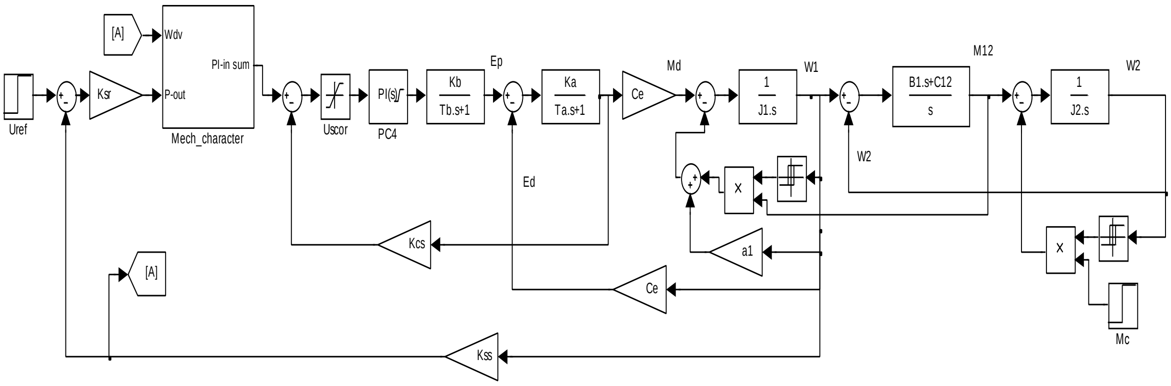


Figure 2. A structural diagram of the electromechanical traction drive system

The values of the parameters of the electromechanical system were determined on the basis of the characteristics of the ESh 20.90 excavator in accordance with standard methods, described in detail in [1, 6]. In this case, the model parameters were recalculated to the motor and were: U_{scor} = – 10 - 1,3; U_{ref} =0…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 sec; C_e =17.37; K_a =33; T_a =0,082 sec; J_1 =572, J_2 =60 $kg \cdot m^2$; c_{12} = 7500 $\frac{N \cdot m}{rad}$; b_1 =150 $\frac{N \cdot m}{\sec}$; $a_1 = 20$.

The choice of feedback structure by load was made on the basis of frequency analysis of transfer functions by obtaining an amplitude-frequency response with a given coefficient of damping oscillation according to the recommendations of work [1] - for flexible feedback and works [6,12] for rigid and flexible feedbacks. Feedbacks on load were introduced into the speed controller, thereby performing a parallel correction on the load in the speed loop and preserving the principle of current limitation of the electric drive.

Synthesis of flexible feedback on load. The obtaining of a given coefficient of damping of oscillations in a two-mass system is achieved by using a negative flexible feedback on the load. In connection with the specifics of the mechanism, the load in the cable was determined by the difference in motor speeds and the speed of the bucket. The feedback coefficient was determined by equation:

$$K_{OS} = (\gamma_0 - \gamma)/\gamma, \tag{2}$$

where $\gamma = \frac{J_1 + J_2}{J_1}$ – the ratio of the inertia moment of the first and the second mass.

If $\gamma_0 = 5{,}8$ the second mass move will be close to optimal.

The coefficient of the speed controller was determined with help of the equation:

$$K_{SR} = T_m / (T_u \gamma_0^{3/4}), \tag{3}$$

where $T_m = \frac{\omega_{nom}(J_1 + J_2)}{M_{nom}}$ – total mechanical constant of the system; $T_u = \left(\frac{c_{12}(J_1 + J_2)}{J_1 J_2} \right)^{-1}$ – time constant of a two-mass system.

The advantage of this method of organizing feedback is the exclusion of the differentiation operation in obtaining the corrective signal, and the disadvantage is the need to measure the

velocity of the bucket. The values of the parameters found with the help of expressions (2) and (3) were as follows: $K_{SR} = 4{,}2$ and $K_{OS} = 4{,}2$ where $T_u = 0{,}0847\sec$ and $T_m = 1{,}374\sec$.
The block diagram of the system with parallel correction in the speed loop is shown in fig. 3.

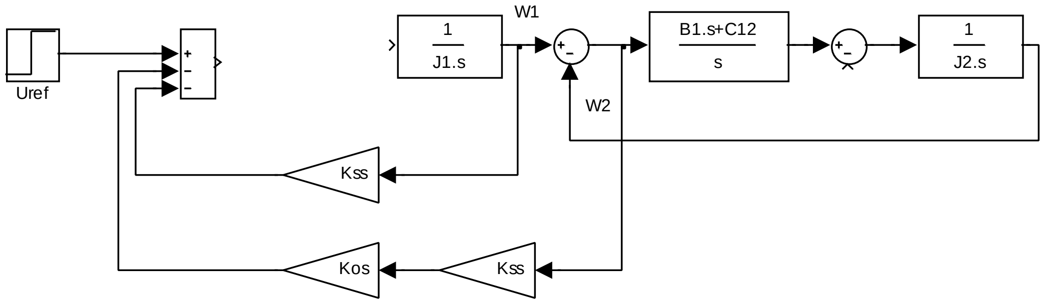


Figure 3. Parallel correction based on flexible feedback

Synthesis of rigid and flexible feedback on load. Let us check the possibility of the existence of rigid and flexible feedbacks in the electromechanical system of the traction mechanism. For this, let us define the following parameters:

$$\gamma = \frac{J_1 + J_2}{J_1}; \varepsilon_m = \frac{T_m}{T_\mu}; \varepsilon_{12} = \frac{1}{\omega_{12} T_\mu}; K_m = K_e \cdot Ce; T_m = \frac{J_1 + J_2}{\beta}; \beta = M_S / \omega_{10}.$$

Here: ε_m – relative electromechanical time constant of the system; ε_{12} – the relative time constant of the conservative link of the two-mass system; T_μ – small uncompensated time constant; $K_e = K_a$ – the gain of the armature circuit; $\omega_{12} = \sqrt{\frac{c_{12}(J_1 + J_2)}{J_1 J_2}}$ – oscillation frequency of a two-mass system; $M_S = Ce \cdot I_{S.C.}$ – the starting (short-circuit) torque; $I_{S.C.} = U_{nom} / R_a$ – short-circuit current of the DC motor; $\omega_{10} = U_{nom} / Ce$ – speed of ideal idling.
The criterion for the existence of rigid and flexible feedbacks is the fulfillment of the following inequalities:

$$4{,}88 \le \varepsilon_{12} \le 9{,}54 \text{ и } 1{,}0 \le \gamma \le 2{,}76.$$

The calculated values of the parameters for the traction mechanism with the electric drive (thyristor converter-DC motor) were: $I_{S.C.} = 1200 / 0{,}0303 = 39603A$; $M_S = 17{,}37 \cdot 39603 = 687920 N \cdot m$; $\omega_{10} = 1200/17{,}37 = 69{,}08 rad/\sec$; $K_m = 33 \cdot 17{,}37 = 573{,}21$; $\beta = 687920 / 69{,}08 = 9958$; $T_m = (572 + 60) / 9958 = 0{,}0634 \sec$; $T_\mu = 0{,}01\sec$. Obtained values $\gamma = 1{,}1$ and $\varepsilon_{12} = 8{,}51$ satisfy the conditions of inequalities, therefore, in the traction electric drive it is allowed to use rigid and flexible feedbacks on the load.
The parameters of rigid and flexible feedbacks on the load in the elastic element are found using the correspondences:

$$\lambda_3 = \frac{\lambda_1 \gamma \left[(9 + 4\sqrt{2})\varepsilon^4{}_{12} - (188 + 138\sqrt{2})\varepsilon^2{}_{12} + (401 + 298\sqrt{2})\right]}{(\gamma - 1)\varepsilon_м \varepsilon^4{}_{12}};$$

$$\varepsilon_н = \frac{\lambda_1 \gamma \left[(52 + 34\sqrt{2})\varepsilon^2{}_{12} - \gamma\varepsilon^4{}_{12} - (401 + 298\sqrt{2})\right]}{(\gamma - 1)\varepsilon_м \varepsilon^2{}_{12}}. \quad (4)$$

Parameter λ_1 in (4) is determined by the equation:

$$K_{.SS} = \lambda_1 \cdot Ce / K_b \,, \tag{5}$$

where K_{SS} – gain of speed sensor; K_b – gain of thyristor converter.

The transition from parameters for rigid λ_3 and flexible $\varepsilon_и$ feedbacks to their absolute values can be performed on the basis of the equations:

$$K_{on} = \frac{\lambda_3}{K_b K_m} \text{ и } K_{on} T_{on} = \frac{\varepsilon_m T_\mu}{K_b K_m}. \tag{6}$$

The structural scheme with rigid and flexible feedbacks in the speed loop is shown in fig. 4.

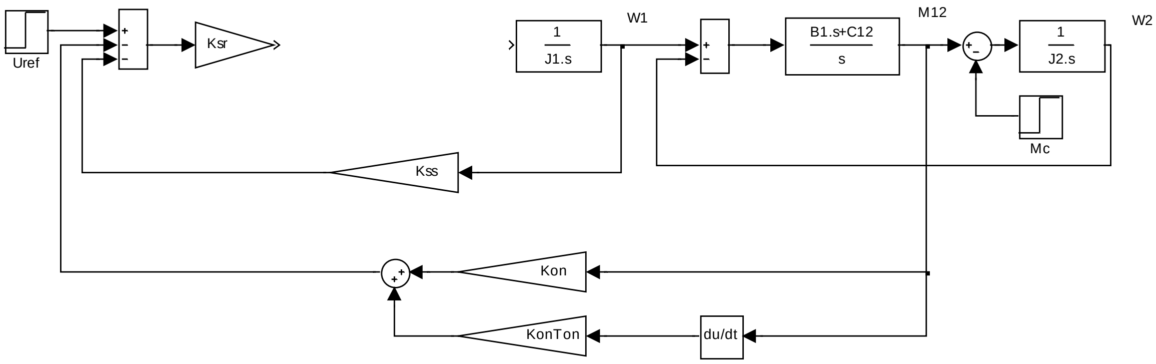


Figure 4. Parallel correction with rigid and flexible feedbacks

The values of the feedback parameters found with the help of expressions (4)–(6) were as follows: $\lambda_1 = \frac{0{,}151 \cdot 120}{17{,}37} = 1{,}043$; $\lambda_3 = 17{,}23$; $\varepsilon_m = 16{,}11$; $K_{on} = 0{,}00025$; $K_{on} T_{on} = 0{,}000002342$.

3. Research and discussion

To evaluate the influence of feedbacks on dynamic loads and elastic oscillations of the traction mechanism in transient operation modes, numerical modeling of the electromechanical system based on equations (1)–(6) in the *MatlabSimulink* environment was carried out. During the simulation of the electromechanical system, the effectiveness of three variants of the control system implementation was compared: standard (ST); with flexible feedback (FLX) and flexible and rigid feedbacks (F&H) on the load. To assess the effectiveness of these control systems, the following quality criteria for transient processes were used: maximum motor torque $\max M_{dv}$ and on the bucket $\max M_{12}$; the average value of the load $\int M_{dv}$ and $\int M_{12}$; time of the transient process of the speed of the bucket $t_{\omega 2}$; the steady-state value of the speed of the bucket $\omega_{2уст}$; the path traversed by the bucket L_{bucket}; logarithmic decrement of load oscillations in the elastic element $\delta = \ln(A_i / A_{i+1})$; the rate of change in armature current in transient modes dI/dt. Values were presented in relative units, referred to the nominal values of torque (current) and motor speed.

Since the main mode of operation of the traction mechanism is digging and the most interesting moment is the load imbalance, let us simulate the behavior of the system at start-up at the nominal speed at the moment of the resistance forces, equal $M_c = 0{,}5 M_{ном}$, with a further load to the value: $M_c = 1{,}5 M_{ном}$. As an illustration of the effectiveness of the feedbacks on the load in fig. 5 shows the oscillograms of the speed of the motor ω_1 and the bucket ω_2, the torque of the motors

M_{dv} and the elastic element M_{12} under the specified mode, without any limitations on the part of the electric drive. It can be seen from the oscillograms that the F&H control system demonstrates the best results when the load is jumped. The transient process of the velocity of the bucket is practically aperiodic with a damping factor $\xi = 0{,}707$, confirming the reliability of the calculation of feedback parameters. It should be noted that the use of FLX also allows one to significantly reduce the oscillation in the system, but noticeably inferior to F&H in productivity. The presence of limitations on the part of the electric drive inevitably leads to a qualitative change in the transient processes, namely, to a decrease in the productivity and damping parameters.

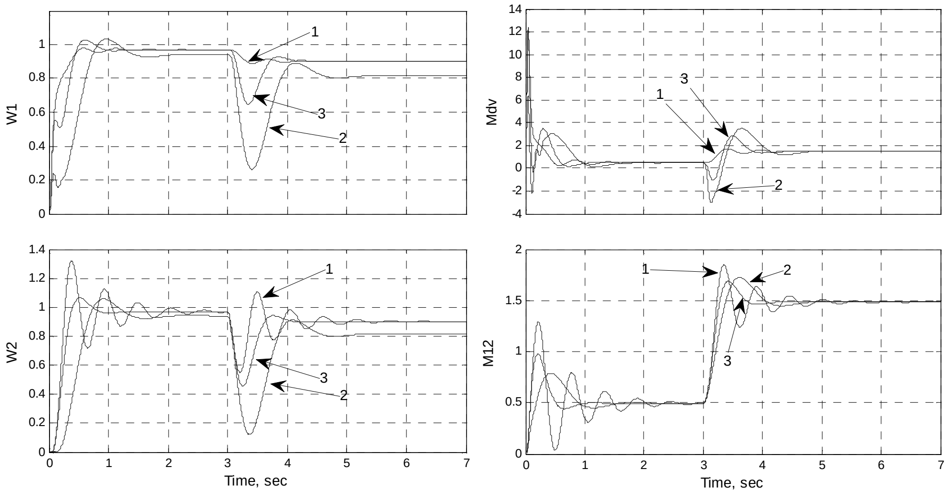


Figure 5. Transient processes in case of a load build-up in the system without any limitations on the part of the electric drive: 1 - standard system (ST); 2 - with flexible feedback (FLX); 3 - with flexible and rigid feedbacks (F&H)

Transient processes in an electromechanical system with limitations on the side of the electric drive are shown in fig. 6. From the above oscillograms it follows that the use of F&H and FLX allows one to reduce the amplitude of the oscillations; however, there is a significant decrease productivity of the system for restoring the speed of the bucket.

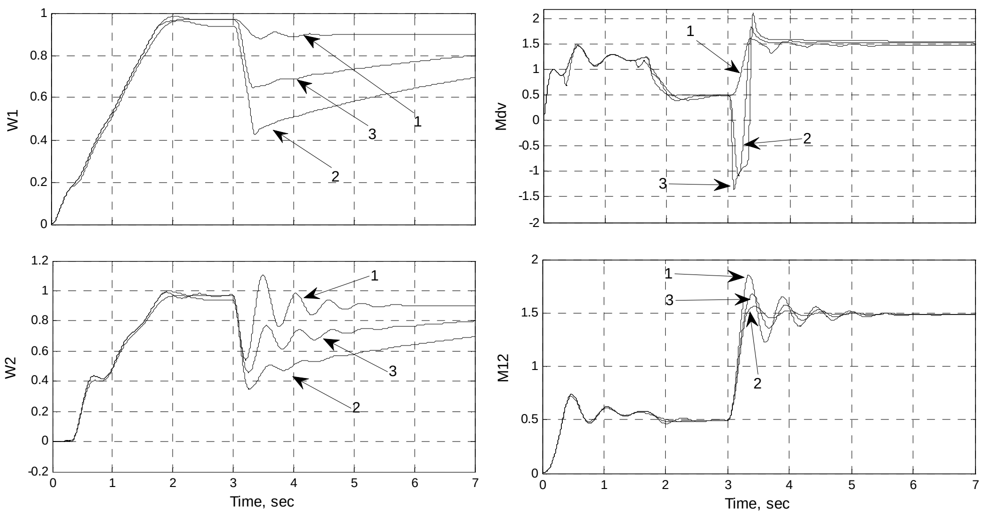


Figure 6. Transient processes when load is loaded in a system with constraints:
1 - standard system (ST); 2 - with flexible feedback (FLX); 3 - with flexible and rigid feedbacks (F&H)

In table 1, the indicators of the quality of transient processes are presented for various versions of the electric drive control system.

Table 1. Indicators of the quality of transient processes for various versions of the electric drive control system

Parameter	ST	FLX	F&H
$\max M_{dv} / \min M_{dv}$	$\frac{1{,}7/-}{1{,}62/-}$	$\frac{3{,}5/-3{,}0}{2{,}12/-1{,}35}$	$\frac{2{,}85/-1{,}0}{1{,}85/-1{,}1}$
$\max M_{12}$	$\frac{1{,}85}{1{,}86}$	$\frac{1{,}72}{1{,}55}$	$\frac{1{,}7}{1{,}68}$
$\int M_{dv}$	$\frac{2{,}77}{2{,}77}$	$\frac{2{,}69}{2{,}39}$	$\frac{2{,}77}{2{,}55}$
$\int M_{12}$	$\frac{2{,}86}{2{,}86}$	$\frac{2{,}85}{2{,}82}$	$\frac{2{,}86}{2{,}83}$
$t_{\omega 2}, \sec$	$\frac{1{,}8}{1{,}8}$	$\frac{1{,}6}{7{,}0}$	$\frac{1{,}0}{9{,}0}$
$\omega_{2st-state}$	$\frac{0{,}9}{0{,}9}$	$\frac{0{,}82}{0{,}82}$	$\frac{0{,}9}{0{,}9}$
$L_{bucket}, meter$	$\frac{9{,}366}{9{,}343}$	$\frac{7{,}844}{6{,}13}$	$\frac{9{,}148}{7{,}6}$
$\delta_{M12} = \ln(A_i / A_{i+1})$	$\frac{0{,}12}{0{,}11}$	$\frac{0{,}13}{0{,}03}$	$\frac{0{,}11}{0{,}06}$
dI / dt	5	35-40	15

In table 1 above the line, the value of the corresponding indicator for the system is displayed without taking into account the limitations on the part of the electric drive, and below the bar - with limitations. The analysis of the simulation results of the electric drive control systems of the traction mechanism of the excavator allows us to draw the following conclusions.

1. For the system without the limitations of the electric drive, the use of F&H makes it possible to obtain practically aperiodic transient processes with a damping factor $\xi = 0{,}707$, as a result of which it is possible to completely eliminate the influence of the elastic element on the dynamic characteristics of the two-mass system without reducing the productivity of the mechanism (the difference in L_{bucket} is not more than 2%).

2. The introduction of limitations on the drive side reduces the oscillation decrement for FLX and F&H and the average load on the motor, limiting the amplitude of the motor torque oscillations to the shape of the mechanical characteristic. At the same time, the smallest amplitudes of the oscillations of the motor and the bucket are obtained with FLX, and F&H has a greater productivity.

3. For FLX, there is a lower speed of the bucket after the load is applied, which is associated with a decrease in the gain of the speed controller. The recovery time of the bucket speed to the steady-state value increases 4 times for F&H and 5 times - for FLX compared to the standard electric drive. The productivity of the mechanism (estimated by the path traveled by the bucket), compared to the standard system, for FLX is reduced by 35% and for F&H - by 19%.

4. The sign-variable of the drive torque in transient modes with amplitudes in the direction of negative values, the magnitude of which is higher than M_{nom}, is typical for both feedbacks on the load, which can lead to the opening of backlashes in the gearbox.

5. The rate of change of the motor current in the transient modes reaches the order of magnitude $40I_{nom}/\sec$ for FLX and $15I_{nom}/\sec$ for F&H, while the recommended value of this parameter for the DC drive is less than $5I_{nom}/\sec$. This phenomenon can lead to a deterioration of the switching conditions of the brush-collector unit and a decrease in the reliability of the operation of the electric drive.

4. Conclusion

The conducted researches of the control system of the electric drive of the traction mechanism of dragline with feedbacks on load in the elastic element showed that, without taking into account the limitations on the side of the electric drive, both ways of introducing feedbacks on the load, practically not reducing the machine's efficiency, are more effective than the standard control system in terms of the level of decrease in dynamic loads and the limitations of oscillational motions. Taking into account the limitations on the side of the electric drive shows that feedbacks on the load also allow one to limit the amplitudes of elastic oscillations, however, the duration of the oscillations and the degree of their damping are reduced in comparison with the electric drive control system without limitation. At the same time, there was a significant decrease in productivity of the system for restoring the speed of the bucket and the productivity of the traction mechanism by increasing the time of the transient processes. It is also established that the sign-variable of the electric drive loading associated with the opening of the backlashes in the gearbox due to the effect of feedbacks on the load in the elastic element, under certain conditions, can lead to undesirable phenomena in the operation of the drive and a decrease in the reliability of its operation.

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