

# Damping elastic oscillations of digging mechanism

**N K Kuznetsov, D E Makhno, I A Iov**

Irkutsk National Research Technical University,  
83 Lermontov St., Irkutsk, Russia

E-mail: [knik@istu.edu](mailto:knik@istu.edu)

**Abstract.** The article studies methods for reducing dynamic loading and elastic oscillations of excavator buckets using dampers. The authors suggest a structural scheme for damping bucket oscillations using a damping device installed in a running gear of the traction cable. The results of numerical efficiency simulation are presented. The article shows that the system helps to reduce intensity of elastic oscillations and a transition period in acceleration and deceleration modes.

## 1. Introduction

Openness of the kinematic structure of excavator buckets, cantilever nature of external forces, and the presence of elements with elastic properties (cables) cause high dynamic loading, elastic deformations of links and moment and velocity oscillations in a bucket in transient modes. They reduce accuracy of motions, productivity and reliability [1–6]. The method suggested in [7–9], aimed to reduce dynamic loading and oscillations, using flexible feedback by loading in an elastic element, added at the input of a speed/current regulator, is not always effective and acceptable due to high moments of inertia of an engine and a gearbox compared to the ones of an excavator bucket which are up to 80–90% of the total moment of inertia of the mechanism, and low rigidity of the cable. By increasing constants of control system time and transition periods, the authors damp all oscillations and reduce excavator performance. Accordingly, one should pay attention to the use of passive and active oscillation protection systems which are developed and used in different technological machines [10–16]. The use of a damper in a reactive branch of the traction cable of a digging mechanism was studied in [17]. The disadvantage of the suggested oscillations damping system is a more complicated digging mechanism structure due to substitution of a two-drum winch for a one-drum one and the need for adding an equalization unit in a digging workspace, which can damage rock elements, cause premature wear of the cable and decrease excavator reliability.

The article presents the results of efficiency analysis for a damper installed in a running gear of the traction cable as exemplified by ESh 20/90.

## 2. Research object and methods

The research object is a walking damper-based dragline ESh 20/90. Figure 1 shows the basic scheme of cable connections. Damper 4 installed in a running gear of the traction cable between drum 2 and leading unit 5 is suggested to use for reducing elastic oscillations of the bucket.



Let us construct a dynamic model of the electric mechanical system for controlling motions of an excavator using an electric drive with a standard two-loop subordinate control system equipped with regulators and flexible feedbacks by loading in the elastic element for velocity and current.

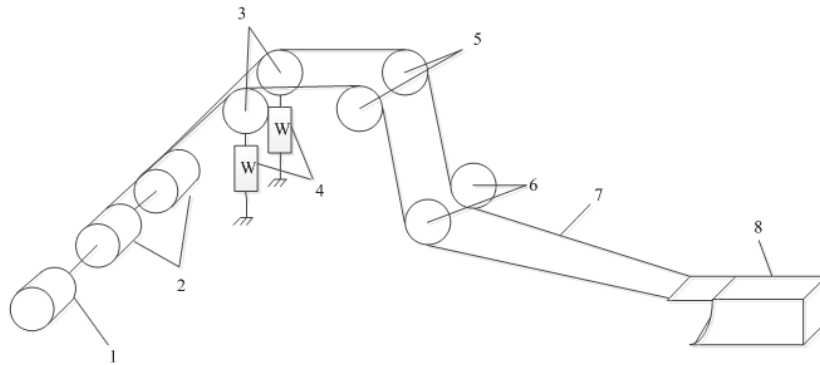


**Figure 1.** Bucket oscillation damping scheme for ESh 20/90: 1 is an electric motor with gear reducers; 2 are drums of the traction winch; 3 are working units of the damper; 4 is an elastic damping device; 5 are leading units; 6 are laying units; 7 is a traction cable; 8 is a bucket

When developing the dynamic model of the mechanical part of a digging mechanism, let us assume that rigidity of a gear reducer and connecting shafts manifold exceeds stiffness of the cable; the cable is a weightless elastic system with constants of stiffness  $c_{12}$  and viscous friction coefficients  $b_1$ ; the damper has a moment of inertia  $J_3$ , constant stiffness  $c$  and damping factors  $b$ . Under these assumptions, the mechanical part of the digging mechanism can be represented as a three-mass mechanical system where  $J_1$  expresses the moment of inertia of an electric engine and a gear reducer;  $J_2$  – the moment of inertia of the bucket,  $J_3$  – the moment of inertia of the damper. Taking into account these assumptions, the system of differential equations for motions of a three-mass electromechanical system of the traction mechanism can be of the form:

$$\begin{cases} U_{SR} = (U_{ref} - K_{SS} \cdot \omega_1) \cdot K_{SR}; U_{CR} = \frac{(U_{SR} - (K_{CS} \cdot I_a + 0.15sK_{CS} \cdot I_a)(T_a s + 1))}{T_{CR}s}; \\ E_p = \frac{U_{CR} \cdot K_b}{T_b s + 1}; E_{dv} = C_e \cdot \omega_1; I_a = \frac{E_p - E_{dv}}{T_a s + 1} K_a; M_{dv} = C_e \cdot I_a; J_1 s \omega_1 = M_{dv} - M_{12} - M_{b1}; \\ J_2 s \omega_2 = M_{12} - M_c + M_{b1}; J_3 s \omega_3 = K_L M_{12} - \frac{c}{s} \omega_3 - b \omega_3; M_{12} = \frac{c_{12}}{s} (\omega_1 - \omega_2 - \omega_3); \\ M_{b1} = b_1 (\omega_1 - \omega_2 - \omega_3). \end{cases}$$

where  $U_{ref}$  is the control voltage;  $U_{SR}$  is the speed controller output voltage;  $U_{CR}$  is the current controller output voltage;  $K_{SS}$  is the speed sensor coefficient;  $K_{SR}$  is the speed controller coefficient;  $K_{CS}$  is the current sensor coefficient;  $T_{CR}$  is the time constant of a current controller;  $K_b$  is the converter gain coefficient;  $T_b$  is the time constant of a converter;  $E_p$  is the converter voltage;  $E_{dv}$  is the engine voltage;  $C_e$  is the structure constant of an engine;  $K_a$  is the armature circuit gain coefficient;  $T_a$  is the armature circuit time constant;  $J_1, J_2$  and  $J_3$  are inertia moments for the first, second and third masses;  $c_{12}$  and  $c$  are cable and damper stiffness coefficients;  $b_1$  and

$b$  are viscous friction coefficients for a cable and a damper;  $K_L$  is a coefficient of the load on a damper;  $M_{dv}$ ,  $M_{12}$ ,  $M_c$  are a torque of the engine, a torque on the bucket and a load of the mechanism;  $M_{b1}$  is the moment of viscous friction in the cable;  $\omega_1, \omega_2, \omega_3$  are angular velocities of the first, second masses and a damper.

The load value can be represented as deviation angles of entering  $\angle b$  and leaving  $\angle a$  branches of the cable from the string between a drum and a leading unit (see Figure 2).  $K_L$  is determined by formula

$$K_L = \frac{(\sin \angle a + \sin \angle b)}{2}.$$

By selecting angles  $a$  and  $b$ , one can change the force, acting on the unit of a damper, i.e. one can change force characteristics.

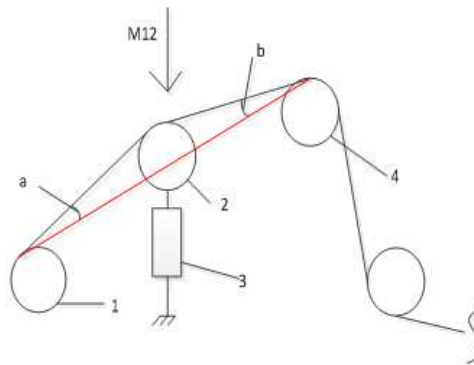


**Figure 2.** The calculation scheme for loading on a damper: 1 is a drum; 2 is a damper unit; 3 is a damper; 4 is a leading unit

The scheme constructed in *Matlab Simulink* based on the above-cited equation system for ESh 20/90 is shown in Figure 3. Engine velocity in  $U_{ref}$  was set between 0 and 10; the upper value was equal to the nominal velocity of the mechanism. The load set by  $M_c$  for a digging mechanism changed between  $0 - 2M_{nom}$  ( $M_{nom} = 30000N \cdot m$ ); the upper value was equal to the stopping moment of the electric drive. The stopping moment value was specified in Limit, the maximum acceleration limit was specified in Acceler. System parameters were as follows:  $Limit = -10 - +10$ ;  $Acceler = -10 - 1,3$ ;  $U_{ref} = 0...10$ ;  $K_{ss} = 0.151$ ;  $K_{SR} = 8$ ;  $K_{CS} = 0.00313$ ;  $T_{CR} = 0.864$ ;  $K_b = 120$ ;  $T_b = 0.01$  sec;  $C_e = 17.37$ ;  $K_a = 33$ ;  $T_a = 0,082$  сек;  $J_1 = 575, J_2 = 60$  и  $J_3 = 0,395$   $kg \cdot m^2$ ;  $c_{12} = 7500 \frac{N \cdot m}{rad}$ ;  $b_1 = 150 \frac{N \cdot m}{sec}$ . Stiffness and viscous friction coefficient values for a damper were

selected based on maximum damping of oscillations in a bucket which were  $c = 1300 \frac{N \cdot m}{rad}$  and

$b = 380 \frac{N \cdot m}{sec}$  for pre-set parameters of the system  $K_L = 0.34$  ( $\angle a = \angle b = 20^\circ$ ).

### 3. Research and discussion

Numerical simulation of electrical and mechanical system dynamics in *Matlab Simulink* was used to test the efficiency of an elastic oscillations damping system. The tests were carried out under following conditions: starting a mechanism at a nominal rate ( $U_{ref} = 10$ ) with loading  $M_c = 0.5 \cdot M_{nom}$ , loading rise up to  $M_c = 1.5 \cdot M_{nom}$ , which is similar to the operation of a

mechanism in a steep area of the mechanical characteristics, stopping a mechanism with loading  $M_c = 3 \cdot M_{nom}$ .

Tests showed that the use of a damper can reduce oscillatory amplitudes of the engine torque  $M_{dv}$  and torque on the bucket  $M_{12}$  during loading rise, compared to a standard control system by 10 %, and fit the form of a transient process  $M_{12}$  to the aperiodic one. It reduces time of a transient process  $M_{12}$  and velocity  $\omega_2$  and increases decrement of oscillations in the cable approximately 2.5-fold. Despite some decrease in velocity  $\omega_2$  compared to a standard control system caused by decrease in total stiffness of the mechanical system, the steady value after loading rise was attained without overshooting.

In the stopping mode, the damper ensures complete damping of oscillations of the engine torque  $M_{dv}$ , its value does not exceed a stopping value and the transient process is a monotonic function of time. The amplitude loading value in the cable  $M_{12}$  decreases by 15% compared to a standard control system,  $M_{12}$  and  $\omega_2$  do not change at all.

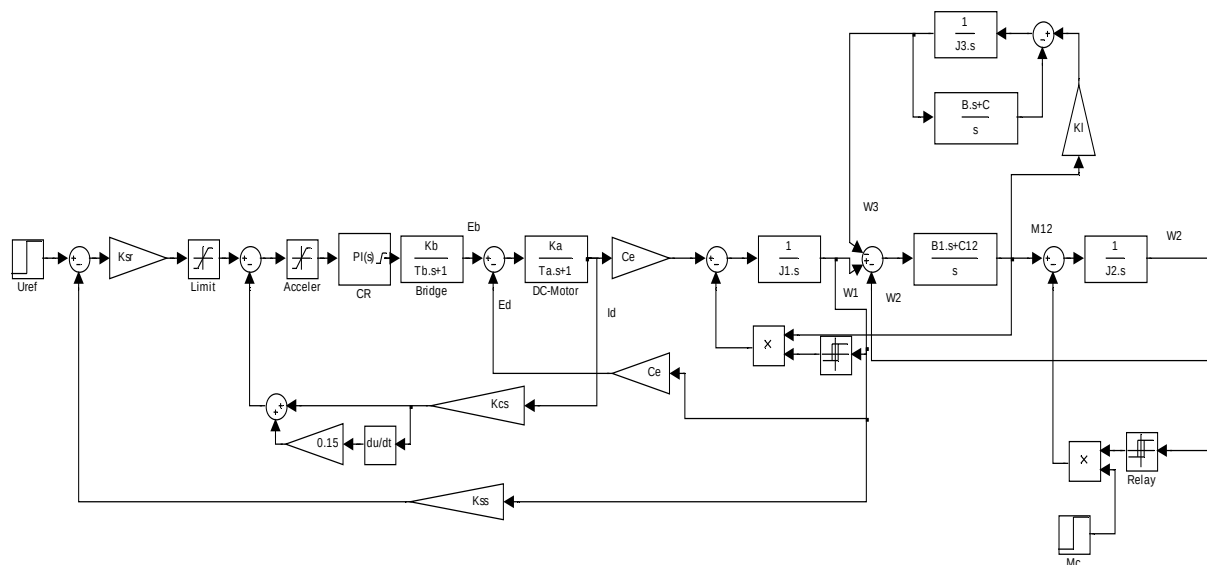


**Figure 3.** The structural diagram of the electric and mechanical traction mechanism system with an elastic damper in *Matlab Simulink*

Figure 4 shows oscillograms of transient processes of engine torque  $M_{dv}$ ,  $M_{12}$  and velocity  $\omega_2$  of the bucket at starting, loading rise and stopping. The decrement of a loading moment in the cable  $M_{12}$  increased from  $\chi = 1.375$  for a standard control system to  $\chi = 3.4$  for a damper-based system; oscillations amplitude of  $M_{12}$  at stopping decreased from 3.8 to 3.28, and transient process time at loading rise  $\omega_2$  changed from 2.0 to 1.0 sec. The use of a damper helped to decrease the frequency of system oscillations two-fold (from 11 rad/sec to 5.5 rad/sec) and oscillations amplitude  $\frac{M_{12}}{M_c}$  on a resonance frequency - 3.5-fold. The damper lacks drawbacks of the device, described in [17], and ensures efficient damping of elastic oscillations of a bucket.

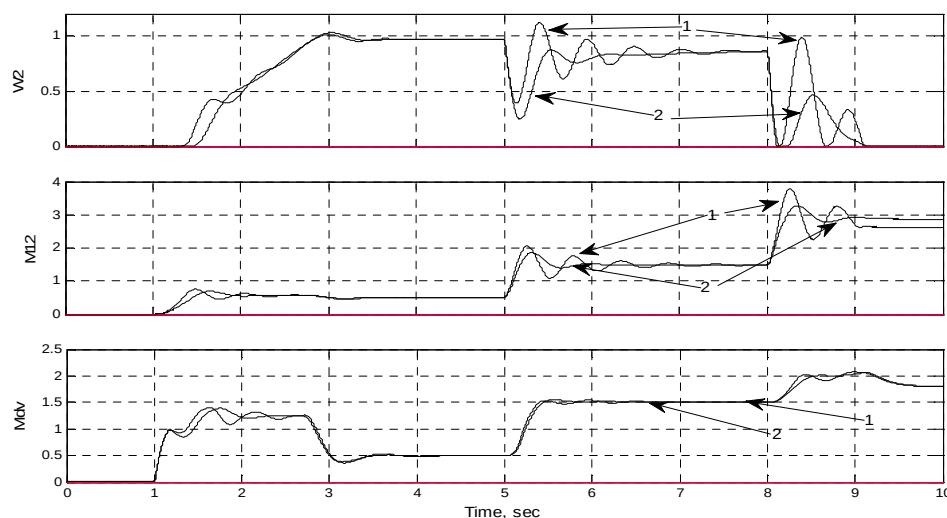


**Figure 4.** Oscillograms of the torque of loading in the cable  $M_{12}$ , engine torque  $M_{dv}$  and bucket velocity  $\omega_2$ , at starting, loading rise and stopping for a standard electric drive (1) and a damper-based electric drive (2)

The total value of damper's movements was 0.5 m. at loading rise, 0.9 m at stopping. In view of winch design with two branches of the traction cable, the value of damper's movements should not exceed 0.45 m under each branch. By installing a damper between a leading unit and a drum, one can remove it from a kinematic scheme in the case of damage of elements (for example, by fixing a unit of the damper). It is a crucial advantage, increasing reliability of the digging mechanism.

#### 4. Conclusion

The studies showed high efficiency and confirmed the possibility of using a damper installed in the running branch of the traction cable for damping bucket oscillations. It reduces dynamic loading and elastic oscillations of the bucket, improves quality of transient processes and reliability of operation. The location scheme for a damper ensures parameters selection and adaptive control under varying operating conditions.

#### References

- [1] Stepanov A.G and Korniyakov M V 2014 *Machine Dynamics* (Irkutsk: Irkutsk State Technical University) p 412
- [2] Lipsett M G 2009 *Methods for Assessing Dynamic Performance of Shovels* (Department of Mechanical Engineering, University of Alberta, Edmonton) p 9
- [3] Kong W K, Liu J X and Ma M H 2012 Research on dynamic characteristics of implement mechanism of mining excavator *Applied Mechanics and Materials* **229-231** 512-516
- [4] Geu Flores F, Kecskemethy A and Pottker A 2007 *Workspace analysis and maximal force calculation of a face-shovel excavator using kinematical transformers*. 12th IFToMM World Congress, Besancon, June 18 – 21, p 6
- [5] Li Ying and Liu Wenyuan 2014 *Comparsion of advanced dragline dynamics models for efficient engineering analysis* *Int. Journal of Advanced Manufacturing Technology* **72 (5-8)** 757
- [6] Samuel Frimpong and Nuray Demirel 2009 Case Study: Planar Kinematics of Dragline for Efficient Machine Control *Journal of Aerospace Engineering* **22 (2)**
- [7] Rebenkov E S and Babokin G I 1995 Synthesis of structures and parameters identification for an automatic control system with a variable link stiffness *The Electricity* **6** 48-54
- [8] Lyakhomskiy A V and Fashilenko V N 2004 *Control of electromechanical mining machine*

systems (Moscow: Moscow State Mining University)

[9] Bortsov U A 1979 *Thyristor systems of the electric drive with elastic links* (Leningrad: Energy)

[10] Mead D J 1999 *Passive vibration control* (Chichester, England: Wiley)

[11] Karnovsky I A and Lebed O 2001 *Formulas for structural dynamics. Tables, graphs and solutions* (New York: McGraw Hill)

[12] Upyr' R, Logunov A, Nasnikov D and Eliseev S 2007 *The Decision of Generalized Tasks Vibroprotection on the Basis of Structural Methods of Mathematical Modeling* (Annals of DAAAM for 2007 & Proc. of the 18th Int. DAAAM Symp., Editor B. Katalinie, Published by DAAAM Int.), Vienna, Austria pp 429-435

[13] Eliseev S V, Reznik U N, Khomenko A P and Zasyadko A A 2008 *Dynamic synthesis in generalized problems of vibration protection and vibration isolation of technical objects* (Irkutsk: Irkutsk State Transport University) p 523

Karnovsky I A and Lebed O 2010 *Advanced methods of structural analysis*. (New York, Springer) p 674

[14] Kuznetsov N K 2014 *Dynamic synthesis of controlled machines* (Saarbrücken: Verlag Palmarium Academic Publishing) p 357

[15] Eliseev S V and Khomenko A P 2015 *Dynamic oscillations damping: feedback theory and mathematical simulation methods* (Novosibirsk: Science)

[16] Solov'ev S V and Kuziev D A 2014 Dependence of dynamics of the working process of a career dragline on elastic-damping parameters of its traction mechanism *Coal* **2** 60-62