Regularities of the working equipment elements mass reduction to the hydraulic power cylinder piston for the bucket boom machines size standard

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Abstract. The working equipment mass reduction is performed proceeding from the condition of the kinetic energy equality of the reduced mass and kinetic energy of the working equipment. Analytic connections of the boom hydraulic cylinder piston kinematic characteristics with the kinematic characteristics of the working equipment rotation are determined for the establishing of this dependence. Dependences of the working equipment reduced masses are obtained for different boom positions: transport, horizontal and maximum raised position of the working equipment. As opposed to similar researches in mechanical engineering the results of this research consist in the establishing of reduced masses dependences on the machine weight, its lifting capacity, boom positions for the machines size standard. Relative characteristics of the reduced mass depending on the lifting capacity and weight of the machine are obtained.

1. Introduction

Modern process boom hydraulic excavators, loaders and other machines at the world market are represented by samples of the standard series, power of which ranges from several kW to 2000-3000 kW. The total weight of super heavy samples of machines reaches 250-300 tons and this is not the limit.

Machines of such class are in demand at the market and have heavy unbalanced boom working equipment, which inertial and stiffness properties are understudied.

In paper [1] the authors Z. Miaofen, S. Shaohui, G. Youping, Z. Dada establish connections of mechanical and hydraulic control models of the loader working process. Multiparametric optimization of the engine operation mode is performed by the methods of numerical integration.

In paper [2] the authors Z. Zhihong, W. Yunxin, M. Changxun consider the dynamic simulation of the boom in horizontal and other positions. Mechanical parameters, reducing dynamic loads are established.

In paper [3] the authors S. Kang, J. Park, S. Kim, B. Lee, Y. Kim, P. Kim, H. J. Kim consider the excavator working equipment control by means of the controller, implementing efficient control processes of a boom, bucket, arm, simulating actions of an experienced operator.

In paper [4] the authors H. Xie, G. Zhang established that when the automobile crane boom was in the upper position then the pressure in the hydraulic system was increased and vibration was increased at the expense of the reduced mass increasing and power arm decreasing.

The reduced masses of the attached working equipment of heavy hydraulic machines have large values and they are significantly changed in different working positions. That's why it's expedient to establish the reduced masses influence on the quality of transient processes of acceleration, piston speed and pressure changing in hydraulic cylinders of the working equipment [5, 6, 7].

For bucket machines the main parameters of the loading equipment are nominal lifting capacity Q, nominal bucket capacity V_Γ, geometric and mass characteristics of the working equipment [7].

Paper [5] examines the task of the scraper bucket mass reduction to the bucket digging edge. The formula of the reduced mass m_Π represents the function from the elements mass of the working equipment. Paper [5] gives the graph of the reduced mass dependence at the cutting edge of the bulldozer DET-250 on the hydraulic cylinder rod movement S_{III}. It's concluded that the reduced mass m_Π is changed insignificantly during the rod movement of such machines as scrapers, bulldozers.

Paper [8] examines the tasks of the mass reduction on impact of two objects, rotating on two parallel axes. The reduced masses of each rotating object to the impact point are performed according to the formulas:

$$m_{1\Pi} = J_1 \big/ r_1^2 \; ; \; m_{2\Pi} = J_2 \big/ r_2^2 \; ,$$

where J_1, J_2 – inertia moments of rotating objects; r_1, r_2 - distances from rotation axes to the objects impact point.

Paper [8] examines actuation devices control systems. The inertia moment of moving masses at small steering angles is determined by the formula

$$J_\Pi = ml^2 + J_p , \tag{1}$$

where m – moving mass; l – distance from the mass gravity center to the rotation axis; J_p – inertia moment of the reduced mass relative to the own gravity center .

Essentially, formula (1) is the theorem on the object inertia moment for parallel axes.

This paper examines boom machines, which boom swing angle changes for more than 90°, these machines include cranes, excavators, loaders. In scientific and technical literature there are no data about the mass reduction methods for this class of machines. This is a constraint of the dynamic characteristics improvement of such machines due to the absence of methods and numerical data relative to the boom machine inertia moments.

2. Task definition

The article sets the task to obtain the general rated function of the boom bucket machine working equipment elements mass reduction to the hydraulic power cylinder piston. The problem peculiarity consists in a significant change of the reduced masses values depending on a boom swing angle.

Figure 1 shows the working equipment of a front loader in two positions: transport position with the bucket at the bottom over the bearing surface and horizontal position of the boom when the overturning moment created by the gravity forces of the load and bucket is minimum one.
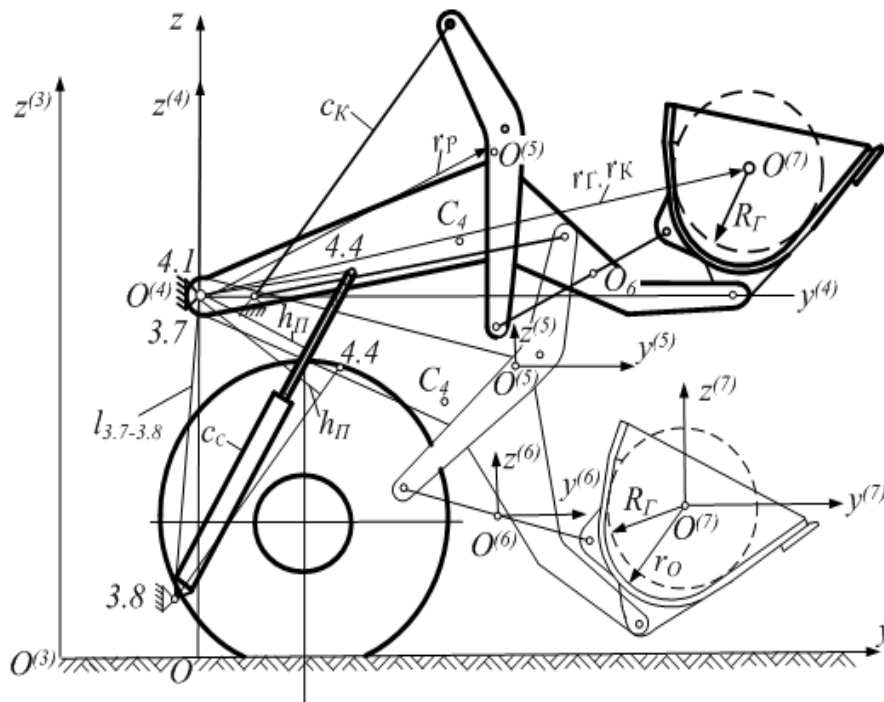
Figure 1. Design diagram of the boom bucket machine working equipment.

Figure 1 shows h_{Π} – arm of the hydraulic power cylinder; r_K, r_{Γ}, r_P – accordingly radius vectors of the elements mass centers of the working equipment; c_c – length of the boom hydraulic cylinder.

For the determination of the bucket mass inertia moment (of the bottom and walls) we can use the thin-walled cylinder inertia moment formula, having taken the cylinder radius equal to the radius of the bucket bottom r_o (see Figure 1). Then the bucket inertia moment relative to the own mass center can be calculated according to formula [7]

$$J_{K.Xc} \approx m_K r_o^2 , \qquad (2)$$

where m_K – mass of the bucket.

For the calculation of the load inertia moment in the bucket $J_{\Gamma.Xc}$ relative to the own mass center we'll perform the conventional replacement of the cargo in the bucket by an equivalent ground cylinder, which radius is calculated according to the formula:

$$R_{\Gamma} = \sqrt{\frac{V_{\Gamma}}{\pi B_o}} , \qquad (3)$$

where V_{Γ} – nominal capacity of the bucket; B_o – linear dimension of the ground cylinder, equal to the internal dimension of the bucket width.

For small and medium-size machines the bucket bottom radius r_o differs from the ground cylinder radius R_{Γ} by 2…3%, for super heavy machines $R_{\Gamma} > r_o$ by 12 … 20%.

The lever inertia moment $J_{P.Xc}$ relative to the own gravity center can be determined by mechanics formula [7]

$$J_{P.Xc} = \frac{m_p L_p^2}{12} \quad , \tag{4}$$

where m_p – lever mass; L_p – lever arm.

3. Theory

The mass reduction of the working equipment to the hydraulic power cylinder piston is performed proceeding from the equality condition of the working equipment kinetic energy and kinetic energy of the mass reduced to the piston.

The general formula of the kinetic energy of the reduced mass and kinetic energy of the working equipment elements is as follows:

$$\frac{m_{\Pi} V_1^2}{2} = \frac{(J_{K.Xc} + J_{\Gamma.Xc})\omega_C^2}{2} + \frac{J_{P.Xc}\omega_C^2}{2} + \frac{(m_K + m_\Gamma)(r_K \omega_C)^2}{2} + \frac{m_P (r_P \omega_C)^2}{2} + \frac{(m_C L_C^2 / 3)\omega_C^2}{2} , \tag{5}$$

where m_{Π} – working equipment mass reduced to the piston; V_1 – piston speed; m_C, L_C – boom mass and length accordingly; ω_C – boom angular speed; r_K, r_P – radius vectors, drawn from the boom rotation axis to the gravity centers of the bucket and lever.

The first two components of expression [5] determine the working equipment kinetic energy during relative rotation around the own gravity centers. Three other components characterize the kinetic energy of the working equipment mass movable rotation together with the boom.

Taking into consideration that the boom angular speed is determined by operation formula [7]

$$\omega_C = \frac{V_1}{h_{\Pi}} ,$$

we'll receive the final form of the formula for the working equipment mass reduced to the piston

$$m_{\Pi} = \frac{1}{h_{\Pi}^2}(J_{K.X.c} + J_{\Gamma.X.c} + J_{P.X.c} + (m_K + m_\Gamma)r_K^2 + (m_K + m_\Gamma)r_K^2 + m_P r_P^2 + m_C L_C^2 / 3). \tag{6}$$

The important value in the reduced mass formula is the arm h_{Π} of the hydraulic power cylinder, which can be determined in any position of the boom according to the formula for the determination of the triangle altitude h_{Π} with vertices *3.7*, *3.8*, *4.4* by known lengths of the sides of this triangle l_1, l_2, c_c, where $l_1 = l_{3.7-3.8}$; $l_2 = l_{3.7-4.4}$; $c_c = l_{3.8-4.4}$.

The theorem of the triangle vertex heights was obtained for the first time by the authors of this article in papers [9, 10]

$$h_4^2 = \left(\frac{l_1 \cdot l_2}{c_c}\right)^2 - \left(\frac{l_1^2 + l_2^2 - c_c^2}{2c_c}\right)^2 . \tag{7}$$

4. Research results

The general formula is obtained for the determination of the reduced mass m_{Π} of boom machines for different boom positions: lower, horizontal and higher.

Table 1 represents the calculation results of the reduced masses m_{Π} for the modern size standard of front loaders.

Table 1. Results of the working equipment mass reduction to the hydraulic power cylinder piston for the loaders size standard

Lifting capacity Q, t	Reduced mass m_Π, kg		
	Transport boom position with cargo / without cargo	Horizontal boom position with cargo / without cargo	Upper boom position with cargo / without cargo
2.2	$\dfrac{76409}{18552}$	$\dfrac{76584}{17633}$	$\dfrac{382022}{82971}$
3.3	$\dfrac{78432}{18554}$	$\dfrac{121669}{27536}$	$\dfrac{771725}{161711}$
3.8	$\dfrac{209514}{84297}$	$\dfrac{176201}{70554}$	$\dfrac{692063}{277310}$
6.6	$\dfrac{183093}{73944}$	$\dfrac{261122}{103685}$	$\dfrac{1433033}{550669}$
7.3	$\dfrac{210524}{92251}$	$\dfrac{318438}{142322}$	$\dfrac{1341941}{571617}$
15.0	$\dfrac{405808}{170782}$	$\dfrac{499917}{209762}$	$\dfrac{1679183}{697663}$
30.0	$\dfrac{572611}{213574}$	$\dfrac{933802}{375873}$	$\dfrac{3891260}{1378189}$
75.0	$\dfrac{1313186}{489803}$	$\dfrac{2427603}{890688}$	$\dfrac{10971278}{4115359}$

The reduced mass values are shown for three positions of the working equipment with cargo in the bucket and without cargo.

For the transport position of the working equipment Table I shows the reduced mass for a super heavy loader with the lifting capacity Q=75 tons, which value is equal to m_Π =1313186 kg.

For the horizontal boom position the reduced mass of this loader m_Π =2427603 kg, i.e. it has increased by 1.85 times. And finally, for the upper boom position the working equipment reduced mass for this loader with the lifting capacity Q=75 tons is equal to m_Π =10971278 kg, i.e. it has increased by 8.35 times in comparison with the transport mode.

For the size standard of frontal loaders Figure 2 shows dependencies of the reduced weight m_Π on the lifting capacity Q for three positions of the working equipment with cargo in the bucket. The regularity is established which lies in the fact that the reduced mass m_Π for this type of machines exceeds many times the operating mass $m_\Im$ of the whole machine.

Dependencies shown in Figure 2 have regression equations: for transport mode $m_\Pi = 18,28Q$; for horizontal boom position $m_\Pi = 32,4Q$; for upper boom position $m_\Pi = 143,96Q$.

These results prove the proper use of huge reduced masses for analytical calculations in differential equations of the machine working equipment dynamics.
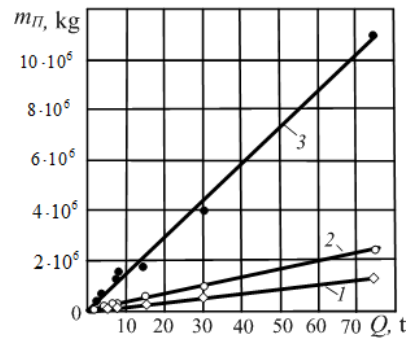
Figure 2. Dependence of the reduced mass $m_{\varPi}$ on the lifting capacity for the front loaders size standard in different boom positions: 1 – boom is in transport position; 2 – boom is in horizontal position; 3 – boom is in upper end position.

Table 2 represents relative values of the reduced mass and operating mass $m_{\varPi}/m_{\ni}$, as well as relative value of the reduced mass and lifting capacity $m_{\varPi}/Q$, where $m_{\ni}$ – operating mass without a cargo in the bucket.

Table 2. Relative parameters of reduced masses of loaders size standard

Lifting capacity Q, tons	Operating mass $m_{\ni}$, tons	Ratio $m_{\varPi}/m_{\ni}$ for three boom positions			Ratio $m_{\varPi}/Q$ for three boom positions		
		Transport position	Horizontal position	Upper position	Transport position	Horizontal position	Upper position
2.2	7.5	10.187	10.21	50.936	34.731	34.810	173.646
3.3	11.1	7.0559	10.96	69.520	23.767	36.869	233.856
7.5	26.0	8.097	12.20	51.595	28.069	42.456	178.865
15.0	74.0	5.4839	6.76	22.69	27.053	33.327	111.945
30.0	139.0	4.1195	6.761	27.990	19.087	31.126	129.708
75.0	245.0	5.3599	9.91	44.780	17.509	32.368	146.283

5. Discussion of results

Dynamic processes of the working equipment are described by differential equations of hydraulic power cylinders.

Figure 3 shows the transient processes of acceleration and piston speed of the hydraulic power cylinder of the super heavy loader Le Torneau L-2350 with different reduced masses, obtained in the result of working equipment differential equation solution [5, 7]

$$\ddot{V} + 2n\dot{V} + \omega^2 V = \frac{K_{\varGamma}}{m_{\varPi}} x, \qquad (8)$$

where V – hydraulic power cylinder piston speed; n – resistance coefficient, $n = \dfrac{\vartheta_{\varGamma}}{2m_{\varPi}}$; $\vartheta_{\varGamma}$ – viscous drag coefficient; ω – rotational frequency of natural oscillations, $\omega = \sqrt{C_{\varGamma}/m_{\varPi}}$; $C_{\varGamma}$ – stiffness coefficient of hydraulic power cylinders; $K_{\varGamma}$ – working fluid flow coefficient of hydraulic power cylinders, $K_{\varGamma} = \dfrac{C_{\varGamma} V_{con}}{x}$; V_{con} – steady-state value of the piston speed after the electric spool opening.
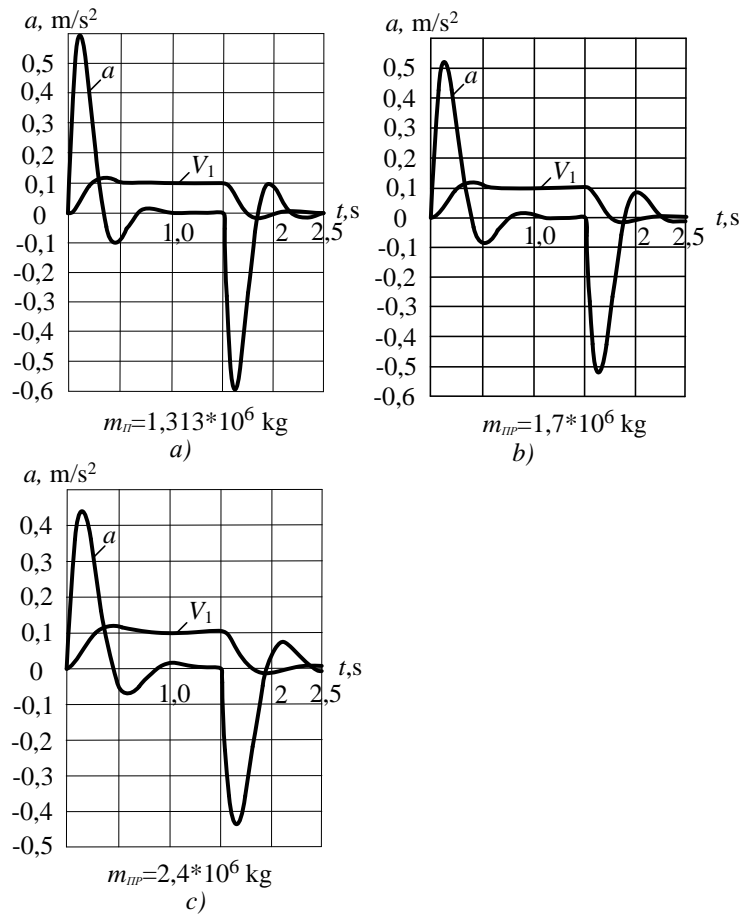
Figure 3. Transient processes of the hydraulic power cylinder piston movement for different reduced masses of the loader Le Torneau L-2350.

The reduced masses varied within the real variation range. With the nominal lifting capacity Q=75 tons for the loader Le Torneau L-2350 with a cargo in the bucket in the transport mode the reduced mass is equal to m_{Π} =1313186 kg.

Transient processes shown in Figure 3, performed with different reduced masses within the indicated range, allow to make important conclusions. The graphs show that the first acceleration amplitude $a = a_{max}$ is decreased with the mass increase practically in inverse proportion to the reduced mass changing. At the same time the transient process of the piston speed changing depends little on the reduced mass changing within the given range.

Obtained results allow to determine dynamic pressures in hydraulic power cylinders excited by inertia forces.

For the loader Le Torneau L-2350 the dynamic inertia force on the piston for a maximum mass is determined by the formula $\Phi_{\partial} = m_{\Pi} a_{max}$.

The dynamic pressure in hydraulic power cylinders, corresponding to this force

$$p_{\partial} = \frac{\Phi_{\partial}}{2(\pi D_{\text{Ц}}^2 / 4)} \text{.}$$

The following parameters: acceleration a =0.42 m/s², dynamic inertia force on the piston of hydraulic power cylinder Φ_{∂} =4607820 N; pressure in hydraulic power cylinders p_{∂} =18.33 MPa correspond to

the reduced mass $m_{\varPi}$ =10971000 kg and cylinder diameter $D_{\varUpsilon}$ =0.4 m.

Peak maximum pressure acts during the time interval t=0.3 s (see Figure 3,a).

6. Summary and conclusion

1. The proper use in engineering of the reduced masses of the boom bucket machine working equipment is proved, which have order of values, for example, $m_{\varPi} = 1{,}1 \cdot 10^{7}$ kg with the operating mass of the machine itself $m_{\vartheta} = 2{,}4 \cdot 10^{5}$ kg.

2. The reduced mass of the bucket boom machine working equipment is the main parameter in the differential equation which forms oscillation frequency, period of oscillations, dynamic inertia forces and pressure in the hydraulic system.

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