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Theory of the piston air spring in the boom machine working equipment

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Theory of the piston air spring in the boom machine working equipment

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Abstract. The efficiency coefficient of the boom machine working equipment is around 50%. When payload lifting in the bucket, gravity force of the working equipment performs useless operation. The piston air spring balances the working equipment gravity force and eliminates the indicated defect. The selection problem of the piston air spring parameters has been solved. The theory of the piston air spring and hydraulic power cylinders is supplemented by a new equation connecting the design parameters of kinematic triangles with parameters of hydraulic mechanisms of the boom machine working equipment.

1. Introduction

Technological boom machines have a cyclic nature of the working processes of loading materials and loose cargoes into transport vehicles. The total mass of such machines can be equal to 250–300 tons, and the engine power reaches 1–2 thousand kW. The specific feature of the bucket boom machines consists in the cantilever mounting of the working equipment relative to the machine, at the same time the working equipment mass, reduced to the center of the bucket masses is commensurable in size with the payload mass in the bucket. This means that when lifting the working equipment half of the engine's energy is wasted overcoming the gravity force of the bucket, boom and other elements of the working equipment.

The paper [1] is known, where the authors Yogeshkumar R. Chheta, Rajesh M. Joshi, Krishan Gotewal, M. ManoahStephen pay attention to the negative influence of the gravity force of the manipulator elements on the power characteristics. Passive methods of the gravity force compensation are considered to increase the manipulator efficiency.

In paper [2] the authors Z. Miaofen, S. Shaohui, G. Youping, Z. Dada establish the relationship of mechanical and hydraulic models controlling the working process of a loader. The multiparametric optimization of the engine operation mode have been performed by the numerical integration methods.

In paper [3] the authors Z. Zhihong, W. Yunxin and M. Changxun consider the dynamic modeling of a boom in horizontal and other positions. Mechanical parameters reducing dynamic loads have been established.

In paper [4] the authors S. Kang, J. Park, S. Kim, B. Lee, Y. Kim, P. Kim, H.J. Kim consider the excavator working equipment control using the controller, implementing efficient control processes of the boom, bucket, stick, modeling the experienced operator actions.

In paper [5] the authors H. Xie, G. Zhang determined that in the truck crane upper boom position the hydraulic system pressure increases and vibration increases due to the reduced mass increase and force arm decrease.

In papers [6, 7] the authors V.N. Tarasov, I.V. Boyarkina proposed a piston-type pneumatic energy-saving drive consisting of a balance cylinder which head end is connected with a gas cylinder charged with compressed air.
The performed review shows that the balancing issue of the gravity force of the working equipment bodies of boom machines is of high priority.

2. Task definition

The article has solved the task of the theory creation relative to the piston air spring mounted in the working equipment of the boom technological machine for balancing the gravity force of the working equipment.
Figure 1 shows the design diagram of the gravity force of excavator working equipment bodies, balanced by the piston air spring *AB*, which is shown in three boom positions: bottom, horizontal and upper. The hydraulic power cylinders are not shown for clarity at this diagram.

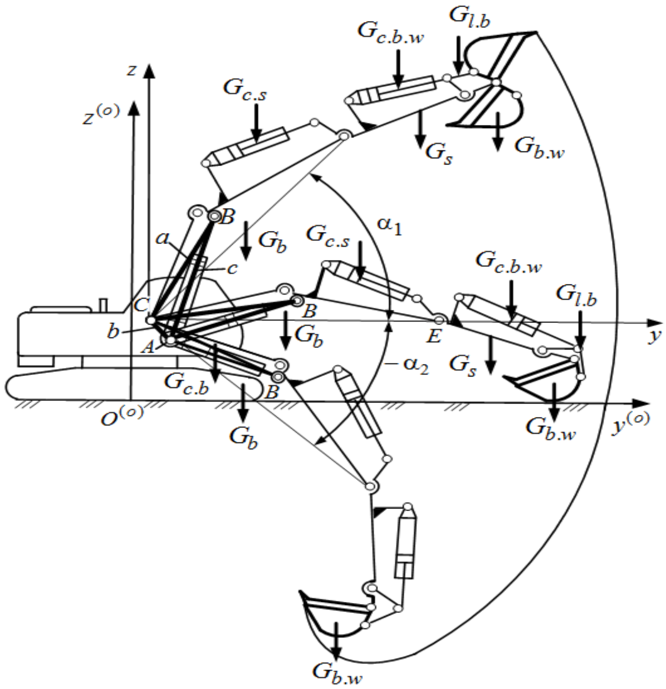

Figure 1. Design diagram of the boom machine working equipment gravity force balancing by the piston air spring

The piston air spring balances the gravity force: $G_{b.w}$ – bucket weight; $G_{l.b}$ – weigh of bucket levers; G_s – stick weight; $G_{c.b.w}$ – bucket cylinder weight; $G_{c.s}$ – stick cylinder weight; G_b – boom weight; $G_{c.b}$ – boom cylinder weight.
Figure 1 selects a kinematic triangle with *CAB* vertices, formed by the sides *a*, *b*, *c*, which are connected respectively with the boom, portal and piston air spring. The parameter -*c* of the piston air spring is a variable length link that varies within the limits $c_{\min} \le c \le c_{\max}$.
Figure 2 shows the design scheme for the mechanism research of the hydraulic power cylinders $A_b B_b$ for the boom travel. The kinematic triangle CA_bB_b is selected which is formed by the sides a_b, b_b, c_b [8, 9].

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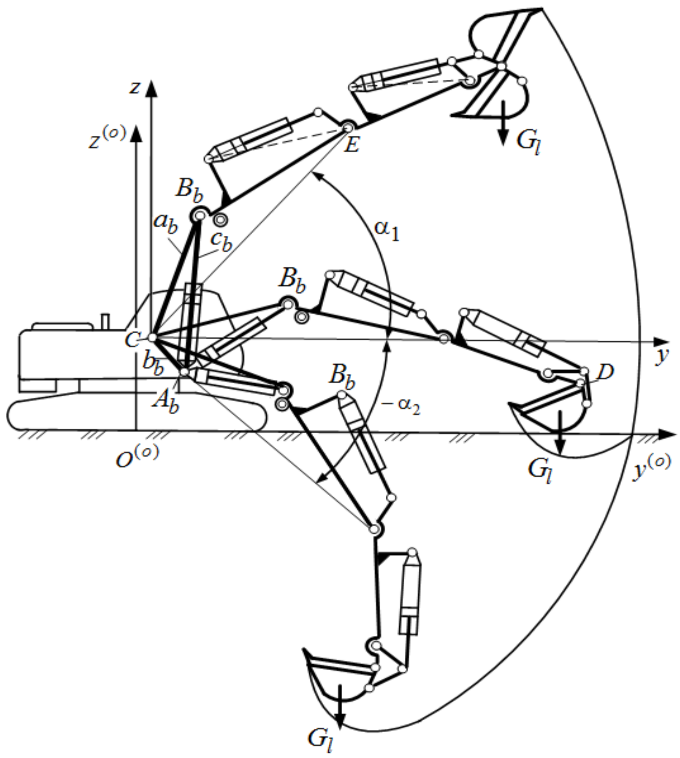

Figure 2. Design diagram of the vertical travel of the gravity force G_l of the payload in the bucket by hydraulic power cylinders

In Figure 2 the hydraulic power cylinders take up only the payload G_l, as all gravity forces are balanced by the piston air spring (see Figure 1).

The idea of the air spring creation is based on the concept of two balanced forces in a mechanical system, one of which is the main vector of the gravity force of the working equipment bodies, reduced to the rod of the air spring, the other is the compressed air pressure force on the piston of the air spring. These balanced forces do not perform operation when moving the boom machine working equipment. The piston air spring ensures the complete balancing of the working equipment gravity force. The piston air spring theory and mechanics of the boom hydraulic power cylinders are based on supporting facts and laws of mechanics [9].

3. Theory

When changing the working equipment position by the boom hydraulic power cylinders (see Figure 2), accordingly the air spring length -*c* is changed. The travel *S* of the air spring piston and travel S_b of the hydraulic power cylinder piston are determined as the length change of the links *AB* (Figure 1) and A_bB_b (Figure 2) for different positions of the working equipment. In Figure 1 and Figure 2 the horizontal position with the angle $\varphi_b = 0$ is taken as the initial boom position; upper position has a positive angle $\varphi_b = \alpha_1$ and lower position is characterized by a negative angle $\varphi_b = -\alpha_2$.

In Figure 3,*a* the dependence $S = f(\varphi_b)$ has been receive of the air spring piston travel, and in Figure 3,*b* – dependence $S_b = f(\varphi_b)$ of the piston travel of the hydraulic power cylinders form the boom angle φ_b.

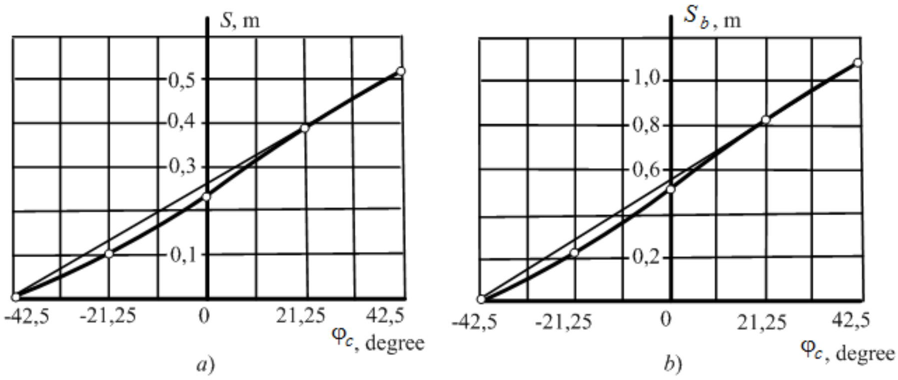

Figure 3. Dependences of the piston travel from the angle of the boom position φ_b : *a*) air spring piston $S = f(\varphi_b)$; *b*) piston of hydraulic power cylinders $S_b = f(\varphi_b)$

The indicated dependences are characterized by polynomial regression equations of third order with high correlation coefficients $R^2 = 0.999$.
For the air spring

$$S = -0.0728\left(\frac{\varphi_b \pi}{180}\right)^3 + 0.0419\left(\frac{\varphi_b \pi}{180}\right)^2 + 0.3958\left(\frac{\varphi_b \pi}{180}\right) + 0.2399. \tag{1}$$

For the boom hydraulic power cylinders

$$S_b = -0.1555\left(\frac{\varphi_b \pi}{180}\right)^3 + 0.0444\left(\frac{\varphi_b \pi}{180}\right)^2 + 0.8303\left(\frac{\varphi_b \pi}{180}\right) + 0.5251. \tag{2}$$

From the theory of mechanics it's known that the derivative of the function $S_b = f(\varphi_b)$ of the hydraulic mechanism piston travel along the boom angle φ_b is the arm of force of the hydraulic cylinder [10, 11].
That's why for the air spring hydraulic mechanism and hydraulic power cylinders the expressions of the derivatives are true

$$\frac{dS}{d\varphi_b} = h; \qquad \frac{dS_b}{d\varphi_b} = h_b, \tag{3}$$

where h, h_b – accordingly, arm of force of the air spring and arm of the hydraulic power cylinders of the boom.
Consequently, it's possible to determine the air spring arm by the numerical differentiation of dependences (Figure 3). We set discrete intervals of the boom angle φ_b and take up the pitch $\Delta\varphi_b = 1^{\circ}$ of the derivative calculation. According to formula (1) for the given angle φ_b we calculate two current values S_1, S_2, differed by 1° and determine $\Delta S = S_2 - S_1$. We calculate the air spring arm according to the formula $h = \Delta S / \Delta\varphi_b$.
The numerical differentiation operation of the functions $S(\varphi_b)$ and $S_b(\varphi_b)$, shown in Figure 3,*a*,*b*, allows to obtain the dependences of the arm *h* of the air spring and arm h_b of the hydraulic power cylinders from the boom angle φ_b, which are shown in Figure 4,*a*,*b*.

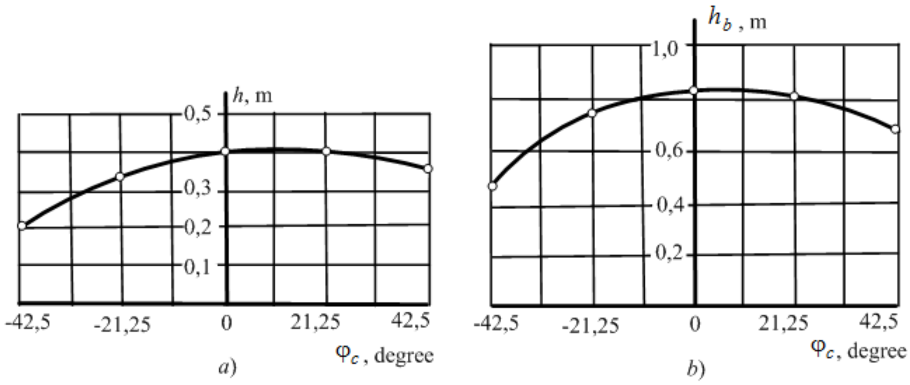

Figure 4. Dependence of the piston air spring arm $h = f(\varphi_b)$ and hydraulic power cylinders $h_b = f(\varphi_b)$ on the boom angle φ_b

The indicated parameters of the air spring and hydraulic power cylinders depend on the working equipment geometric parameters.

In Figure 1 the geometric parameters of mechanisms are represented by the parameters *a*, *b*, *c*, ensuring the air spring reference to the boom machine working equipment, and in Figure 2 – by the parameter a_b, b_b, c_b, ensuring the reference of the hydraulic power cylinders to the boom machine working equipment.

In the article an excavator is considered as an example of the boom machine, which weight is $m_{w.w} = 26000$ kg, working equipment mass is $m_{w.e} = 4051$ kg.

Table 1 lists the working equipment geometric parameters.

Table 1. Geometric parameters of the piston air spring and hydraulic power cylinders of the boom machine

Piston air spring	a, mm	b, mm	$c_{\min}$, mm	$c_{\max}$, mm
	2497	400	2140	2655
Hydraulic power cylinders	a_b, mm	b_b, mm	$c_{b.\min}$, mm	$c_{b.\max}$, mm
	2477	837	1770	2890

In Figure 1 the parameters *a*, *b*, *c* form the arm of force *h* of the piston air spring. The functional connection [10, 11] of these parameters is determined by the theorem of the height of the triangle vertices according to the equation from paper [10, 11]

$$h^2 = \left(\frac{a \cdot b}{c}\right)^2 - \left(\frac{a^2 + b^2 - c^2}{2c}\right)^2 . \tag{4}$$

Similarly for Figure 2 by means of the parameters a_b, b_b, c_b according to expression (4) the arm h_b of the boom hydraulic power cylinders is determined.

The comparison of the values h and h_b, calculated by theorem (4), shows their near-complete coincidence with experimental data for h and h_b, received by the numerical differential method using Figure 3.

The dependences of the arm h of the air spring $h = f(\varphi_b)$ and arm h_b of the hydraulic power cylinders $h_b = f(\varphi_b)$ represent function by which it's possible to determine the piston travel with the final boom turn at an angle of φ_b. Elementary travels of the air spring dS and hydraulic power cylinders dS_b are determined according to the expressions

$$dS = h d\varphi_b ; \qquad dS_b = h_b d\varphi_b . \tag{5}$$

That's why the final piston travels of the air spring and hydraulic power cylinders are determined by the integration of expressions (5)

$$S=\int_{-\varphi_b}^{\varphi_b} h d\varphi_b\,; \qquad S_b=\int_{-\varphi_b}^{\varphi_b} h_b d\varphi_b\,. \tag{6}$$

Thus, in Figures 4,*a*,*b* the area, located under the curves $h=f(\varphi_b)$ and $h_b=f(\varphi_b)$, numerically characterize the final travels *S* of the air spring piston and travels S_b of the hydraulic power cylinders.
The considered reference positions consist of the basis of the theory of the air spring and hydraulic power cylinders of the boom machine working equipment.
For the determination of the balancing force *T*, acting at the air spring rod, we apply the principle of virtual displacements. Specify the virtual displacement ds to the piston air spring rod, in the result the boom with the working equipment (see Figure 1) starts virtual angle displacement $d\varphi_b$.
For the system of forces *T* and $\sum G_j$ there is the balance equation in the form of amount elementary works

$$\sum \delta A_i^e = 0\,; \qquad T\delta S - \sum G_j y_{j.2}\delta\varphi_b = 0\,, \tag{7}$$

where δS, $\delta\varphi_b$ – accordingly the virtual linear and virtual angular displacement of the boom machine working equipment.
The piston air spring force can be defined by equation (7)

$$T=\frac{\sum G_j y_{j.2}\delta\varphi_b}{\delta S}=\frac{\sum G_j y_{j.2}}{h}\,. \tag{8}$$

where h – arm of the piston air spring.
Thus, obtained expression (8) may be considered as the proof of the provision that the derivative of the function *S* of the piston movement of the air spring along the boom angle φ_b is considered as the arm of force $h=\dfrac{dS}{d\varphi_b}$.
Figure 5 shows the dependence of the force *T* on the piston air spring rod $T=f(S)$ on the piston movement *S*.

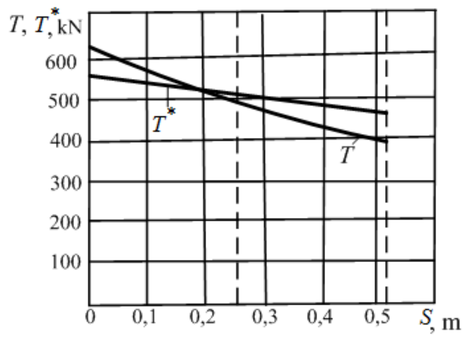

Figure 5. Dependence of the reduced force *T* on the air spring rod on the working equipment gravity force and dependence of the force T^* – compressed air pressure on the air spring piston on the piston movement *S* of the air spring

In Figure 5 the dependence of the force *T*, which represents the reduced force of all gravity forces of the working equipment, it's necessary to balance by the compressed air pressure on the air spring piston. For this purpose it's necessary to assign the maximum force value $T^*_{\max}$ on the piston for the boom in the lower position (see Figure 5).
The piston air spring shown in Figure 6 represents a cylindrical case which has the maximum length equal to $c_{\min}$ =2140 mm and piston movement in the cylinder *S*=515 mm.

Inside the spring case there is the rod with piston and cylinder. The piston diameter of the air spring $d = 0.16$ m, internal diameter of the gas cavity filled with compressed air $D = 0.25$ m.
The start charging pressure of the air spring for the working equipment in the lower boom position is determined by the formula

$$p_{wH} = p_{w\max} = \frac{4T^*_{\max}}{\pi d^2}. \tag{9}$$

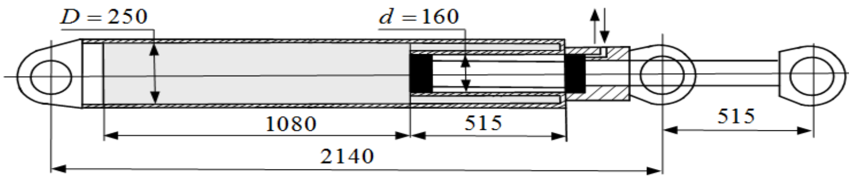

Figure 6. Design of the piston air spring for the gravity force balancing of the boom machine working equipment

The air spring internal cavity volume has the maximum value $V_{\max}$ with the extended rod. While lifting the working equipment the gas chamber volume is increased. The volume change of the air spring internal operating chamber is equal to the volume displaceable while moving the air spring by the operating piston $V = \frac{\pi d^2}{4} S_{\max}$, where $S_{\max}$ – air spring piston movement.
While expanding the compressed air volume in the air spring the pressure is determined by the polytropic process formula [12]

$$p_{w.\min} = p_{w.\max} \left(\frac{V_{\min}}{V_{\max}} \right)^{1,15}. \tag{10}$$

The force T^* of the air spring in the upper boom position is determined by the formula $T^*_{\min} = \frac{\pi d^2}{4} p_{w\min}$.
In Figure 5 the dependence of force is constructed on the air spring piston, created by the air flow for the boom machine $T^* = f(S)$. The obtained dependences allow to determine the work of these forces $T = f(S)$ and $T^* = f(S)$.
The gravity force work of the excavator working equipment is equal to the area, located under the curve $T = f(S)$ (see Figure 5), which is determined by the formula $A(T) = \int_0^S T dS = 261620$ J. The work performed the air spring compressed air is equal to the curve $T^* = f(S)$, which is calculated by the formula $A(T^*) = \int_0^S T^* dS = 261858$ J.
The working process of the hydraulic power cylinders is performed by the similar way. The force in the hydraulic power cylinder rods, implementing the payload lifting G_l (see Figure 2) with the complete balancing of the gravity force is determined by the expression

$$T_b = \frac{G_l y_l \delta\varphi_b}{\delta S} = \frac{G_l y_l}{h_b}, \tag{11}$$

where y_l – bucket gravity center coordinate; h_b – arms of the hydraulic power cylinders.
The dependence of force in the rods of the hydraulic power cylinders on the piston movement $T_b = f(S_b)$ according to formula (11) is given in Figure 7.

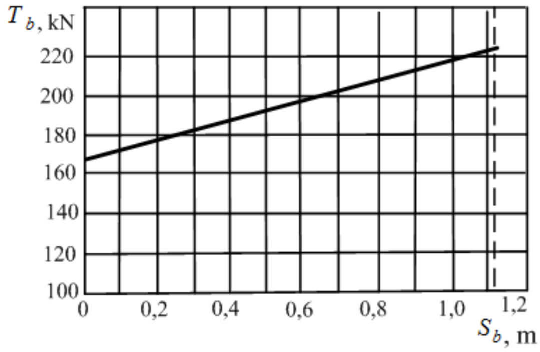

Figure 7. Dependence of force in the rods of the hydraulic power cylinders on the piston movement $T_b = f(S_b)$

The work of the indicated forces of the boom hydraulic cylinders while complete balancing the gravity force is equal to $A(T_b) = 219359$ J. The received values of work of the hydraulic power cylinders $A(T_b^*)$ and piston air spring $A(T^*)$ of the boom machine while complete balancing allow to determine the excavator engine power during the working equipment lifting with cargo in the bucket.

The power of the hydraulic power cylinders reduced to the internal combustion engine shaft is determined by formula [12]

$$N_{b.e} = \frac{A(T_b^*)}{t_b \eta_t}. \tag{12}$$

where t_b – working equipment lifting time; η_t – total efficiency of the excavator working equipment drive.

The total efficiency η_t is determined in the following way $\eta_t = \eta_e \eta_m \eta_{m.w.e}$=0.34, where η_m – mechanical engine efficiency; $\eta_{m.w.e}$ – mechanical working equipment efficiency; η_e – effective efficiency of an internal combustion engine [12].

The power of the piston air spring reduced to the internal combustion engine shaft is determined by the formula

$$N = \frac{A(T^*)}{t_b \eta_t}. \tag{13}$$

In Table 2 for the discrete values of the working equipment lifting time t_b the power values of the hydraulic power cylinders $N_{b.e}$, air spring power N_e and full power N of the working equipment lifting are given reduced to the engine shaft.

The total capacity of the working equipment while lifting is made up of the power N_{ce} of the hydraulic power cylinders and power N_e of the piston air spring reduced to engine shaft

$$N = N_{be} + N_e. \tag{14}$$

Table 2. Dependence of the hydraulic power cylinder capacity reduced to the engine shaft; power of the piston air spring reduced to the engine shaft on the boom lifting time

Boom lifting time t_b, s.	5	7	9	11	13	15
Average speed of piston movement of hydraulic power cylinders V_b, m/s	0.224	0.16	0.124	0.102	0.086	0.075
Average speed of air spring piston V, m/s	0.103	0.0736	0.0572	0.0468	0.0396	0.0343
Capacity of hydraulic power cylinders	129035	92168	71686	58652	49629	43012

reduced to engine shaft N_{be}, W						
Power of air spring reduced to engine N_e, W	154034	110244	85575	70016	59244	51345
Total capacity of working equipment reduced to engine *N*, W	283069	202412	157261	128660	108872	94357

4. Research results

The boom lifting time $t_b \approx 15$ s. indicated in the first line of Table 2 corresponds to the installed engine power $N_e = 95.7$ kW indicated in the lower line of Table 2. Such result testifies that there is an insufficient engine power value for this excavator. However, if this boom machine is equipped with the piston air spring, then the boom lifting time will be equal to less than $t_b \approx 7$ s., i.e. working equipment lifting time is reduced by 8 s.

The power value reduction of the hydraulic power cylinders which is equal to the power developed by the engine while the boom lifting, by means of the piston air spring is the important result. The piston air spring balances all the gravity forces of the working equipment by the compressed air pressure in the air spring. The compressed air application as an operating fluid in the piston air spring instead of an expensive working liquid reduces the cost and simplifies the design of the boom machine working equipment.

5. Discussion of results

In this article the results have been received on the basis of power without considering dynamic processes in hydraulic mechanisms. Taking into consideration that the dynamic transient processes in the hydraulic mechanisms are rapidly damping and short-term [13], so their consideration and registration do not exert influence on the excavator boom lifting time (with an error of no more than 3-5%).

That's why the offered method of calculation and theory of the piston air spring are characterized by high accuracy. Errors in the calculation of forces and energy does not exceed 3-5%, as the considered processes of moving huge masses possess low speed of movements (see Table 2, lines 2 and 3).

6. Summary and conclusion

The gravity force balancing of the working equipment bodies by the piston air spring leads to the reduction of the boom lifting time and of the boom machine general operation cycle time. In the considered definite example of the piston air spring application in the context of the excavator with the operating weight of $m_{w.w} = 26$ tons and installed engine power of $N_e = 95.7$ kW, the complete balancing of all the gravity forces appeared to be excessive one, i.e. lifting time of large masses of the working equipment $t_b = 7$ s. is a little bit underestimated. In similar cases it is expedient to fulfill a partial (half) balancing of the gravity forces by the reduction of the maximum charging pressure of the piston air spring by a factor of 2.

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