Hydraulic drive boom lifting mechanism

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Abstract. The paper presents the calculation of the pressure of the hydraulic cylinder of the boom at the inlet to the throttle, which regulates the speed of lowering the boom with a load, which provides a non-cavitation operation mode of the hydraulic drive. The pressure calculation takes into account the change in the angular acceleration of the boom with a load and the deformation of the working fluid and the walls of the hydraulic cylinder.

When designing a hydraulic drive for lifting machines and mechanisms, it is necessary to take into account the possibility of a break in the flow of working fluid in the rod end of the hydraulic cylinder when the boom is lowered with a load. The rapid lowering of the working body leads to the release of dissolved gases from the working fluid, the occurrence of cavitation. The study of the bulk strength of mineral oils showed that lowering the pressure to –0.06 ... –0.015 MPa at an oil temperature of 45 ° C… 50 ° C leads to a rupture of the flow of working fluid with the release of vapours and gases dissolved in it [1]. When the flow breaks, a sharp change in the volume of the fluid occurs, its elasticity changes dramatically, which leads to a violation of the steady operation mode of the hydraulic drive.

When lowering the boom, it is necessary to ensure the continuity of the fluid flow in the rod end of the hydraulic cylinder (figure 1). This is achieved by installing a throttle at the outlet of the piston cavity of the hydraulic cylinder. With a constant flow of pump Q_p, the piston speed of the hydraulic cylinder v_p must be constant, and under this condition the continuity of flow is ensured [2].

Piston speed, m/s:

$$v_p = \frac{Q_p}{S_r} \tag{1}$$

where S_r – the area of the rod end of the hydraulic cylinder, m^2; Q_p – pump feed, m^3/s.

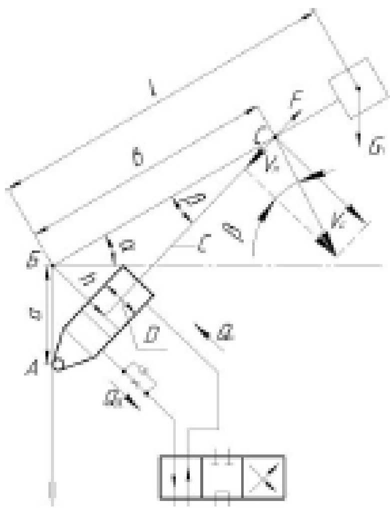

Figure 1. The design scheme of lowering the boom lifting mechanism.

At this speed, the flow rate of the working fluid from the piston cavity of the cylinder through Q_d throttle will be determined [2,5], m^3/s:

$$Q_d = Q_p \cdot \psi \tag{2}$$

where ψ − the ratio of the areas of the hydraulic cylinder, un.

$$\psi = \frac{S_p}{S_{mr}} \tag{3}$$

where S_p − area of the piston cavity of the hydraulic cylinder, m^2.

The flow rate of the working fluid through the throttle is determined [4, 10, 14] from the expression, m^3/s:

$$Q_t = \mu f \sqrt{\Delta P \frac{2}{\rho}} \tag{4}$$

where ΔP − pressure difference at the throttle inlet and outlet, Pa; ρ − fluid density, m^3/kg; μ − the flow coefficient, depending on the design of the throttle, the Reynolds number, the shape and size of the hole; f − throttle hole area, m^2.

Expression (4) in view of (2) is as follows:

$$Q_p \cdot \psi = \mu f \sqrt{\Delta P \frac{2}{\rho}} \tag{5}$$

Due to the relatively small value of the outlet pressure, in further calculations it can be ignored, then:

$$Q_p \cdot \psi = \mu f \sqrt{P \frac{2}{\rho}} \tag{6}$$

where P − pressure at the inlet to the throttle, the same pressure in the piston cavity of the hydraulic cylinder, since the throttle is installed at the outlet of the hydraulic cylinder, Pa.

To find the orifice of the throttle, it is necessary to determine the pressure P. To do this, we make the equation of dynamic equilibrium [6, 7, 8, 15]:

$$G_n \cdot l \cdot \cos\alpha - F \cdot h = J_n \cdot \frac{d\omega}{dt} + \frac{\omega^2}{2} \cdot \frac{dJ_n}{d\alpha} \tag{7}$$

where G_n – reduced weight of the process equipment to the center of mass of the drop cargo, H; J_n – reduced moment of inertia to the axis of rotation of the boom, kg m^2; $\frac{d\omega}{dt}$ – angular acceleration, s^{-2}; F – hydraulic cylinder force, H; h – shoulder effort F, m.

In view of the small value of the second term in the right-hand side of equation (7), we can write it as follows:

$$G_n \cdot l \cdot \cos\alpha - F \cdot h = J_n \cdot \frac{d\omega}{dt} \tag{8}$$

The angle α is found from the kinematic scheme of the mechanism for the maximum and minimum lengths of the hydraulic cylinder and several intermediate positions. For the same positions from the ABS triangle (figure 1) we shall find the angle βon three sides.

The magnitude of the shoulder h is determined from the expression, m:

$$h = b \cdot \sin\beta \tag{9}$$

where b - distance from articulated boom joints with base and piston rod, m.

We shall determine the angular velocity ω of the boom, s^{-1}:

$$\omega = \frac{v_c}{b} \tag{10}$$

where v_c – boom linear velocity at C point, which is equal to, m/s:

$$v_c = \frac{v_n}{\sin\beta} \tag{11}$$

where v_n – hydraulic cylinder piston speed, m/s, from which:

$$\omega = \frac{v_n}{b \cdot \sin\beta} \tag{12}$$

Force of F hydraulic cylinder is equal to [2, 3, 5], H:

$$F = P \frac{\pi D^2}{4} \tag{13}$$

where P - pressure in the piston cavity of the hydraulic cylinder, equal to the pressure at the entrance to the throttle, Pa; D – hydraulic cylinder diameter, m.

Under the action of pressure in the piston cavity of the hydraulic cylinder, the working fluid and cylinder liner are deformed, therefore the initial volume Q decreases by ΔQ [3, 16]. The change in volume ΔQ over dt time is determined from the expression [11, 12], m^3 / s:

$$\frac{d(\Delta Q)}{dt} = \frac{Q}{E_z} \cdot \frac{dP}{dt} \tag{14}$$

where E_z – elastic modulus, Pa.

In expression (14) we shall replace:

$$\frac{Q}{E_z} = K^v \tag{15}$$

where K^v – coefficient of proportionality, taking into account the dependence of the change in the volume of working fluid from pressure, then:

$$\frac{d(\Delta Q)}{dt} = K^v \frac{dP}{dt} \tag{16}$$

Angular speed is as follows, taking into account the deformation of the working fluid and cylinder liner, s[-1]:

$$\omega = \frac{v_n + \dfrac{4 \cdot K^v \cdot \dfrac{dp}{dt}}{\pi D^2}}{b \cdot \sin\beta} \tag{17}$$

We shall replace:

$$c = \frac{4 \cdot K^v}{\pi \cdot D^2} \tag{18}$$

then:

$$\omega = \frac{v_n}{b \cdot \sin\beta} + c \cdot \frac{1}{b \cdot \sin\beta} \cdot \frac{dp}{dt} \tag{19}$$

From equation (8) taking into account the value of F (13) we shall find the pressure P, Pa:

$$P = \frac{4 \cdot G_n \cdot l \cdot \cos\alpha}{\pi D^2 \cdot h} - \frac{4 \cdot J_n}{\pi D^2 \cdot h} \cdot \frac{d\omega}{dt} \tag{20}$$

In equation (20) we shall make the replacement of constant values:

$$A = \frac{4 \cdot G_n \cdot l}{\pi D^2} \tag{21}$$

$$B = \frac{4 \cdot J_n}{\pi D^2} \tag{22}$$

then:

$$P = A \cdot \frac{\cos\alpha}{h} - B \frac{1}{h} \frac{d\omega}{dt} \tag{23}$$

We shall put (9) into (23):

$$P = \frac{1}{b \cdot \sin\beta} \left(A \cdot \cos\alpha - B \frac{d\omega}{dt} \right) \tag{24}$$

The process of changing P pressure, when lowering the boom with a load, is described by a system of equations:

$$\begin{cases} P=\dfrac{1}{b\cdot \sin\beta}\left(A\cdot\cos\alpha\text{-}B\dfrac{d\omega}{dt}\right) \\ \omega=\dfrac{v_n}{b\cdot\sin\beta}+c\dfrac{1}{b\cdot\sin\beta}\dfrac{dP}{dt} \end{cases} \tag{25}$$

Having solved the system of equations (25), we can find the required pressure and, accordingly, the throttle hole for the given parameters of the boom, hydraulic cylinder and the speed of lowering the load, taking into account the deformation of the working fluid and hydraulic cylinder, which will allow lowering the load without "jerks".

Let us consider ABS triangle, sides of AB and BS of constant length, side AC - of variable length, depending on t, i.e. moving point C, m:

$$AC=\int V_n dt=\int \frac{dS}{dt}dt=S(t) \tag{26}$$

where $S(t)$ – displacement of the hydraulic cylinder rod in time, m.

Angle $\beta=\beta(t)$. By the cosine theorem:

$$\cos\beta(t)=\frac{b^2+S^2(t)\text{-}a^2}{b\cdot S(t)} \tag{27}$$

$$\sin\beta(t)=\sqrt{1\text{-}\cos^2\beta} \tag{28}$$

$$\beta(t)=\arccos\left(\frac{b^2+S^2(t)\text{-}a^2}{b\cdot S(t)}\right) \tag{29}$$

The angle alpha by the sine theorem is:

$$\alpha := \arccos\left(\frac{S\cdot\sin\beta}{a}\right) \tag{30}$$

From system (25) we shall express $P\ (t)$, for this we shall differentiate the second equation of the system with respect to t and substitute it into the first:

$$\frac{d\omega}{dt}=\frac{V_n\cdot\cos\beta}{b\cdot\sin^2\beta}\cdot\frac{d\beta}{dt}\text{-}\frac{C\cdot\cos\beta}{b\cdot\sin^2\beta}\cdot\frac{d\beta}{dt}\cdot\frac{dP}{dt}+\frac{C}{b\cdot\sin\beta}\cdot\frac{d^2P}{dt} \tag{31}$$

we shall denote:

$$\frac{C}{b\cdot\sin\beta}=M(t) \tag{32}$$

$$\frac{C\cdot\cos\beta}{b\cdot\sin^2\beta}\cdot\frac{d\beta}{dt}=N(t) \tag{33}$$

$$\frac{V_n\cdot\cos\beta}{b\cdot\sin^2\beta}\cdot\frac{d\beta}{dt}=R(t) \tag{34}$$

we shall obtain:

$$\frac{d\omega}{dt}=M(t)\cdot\frac{d^2P}{dt^2}+N(t)\cdot\frac{dP}{dt}+R(t) \tag{35}$$

The first equation of system (25) is transformed into the form:

$$P=\frac{A\cdot\cos\alpha}{b\cdot\sin\beta}-\frac{B}{b\cdot\sin\beta}\cdot\frac{d\omega}{dt} \tag{36}$$

we shall denote:

$$\frac{A\cdot\cos\alpha}{b\cdot\sin\beta}=K(t) \tag{37}$$

$$-\frac{B}{b\cdot\sin\beta}=L(t) \tag{38}$$

we shall obtain:

$$P=L(t)\cdot\frac{d\omega}{dt}+K(t) \tag{39}$$

We shall substitute equation (35) into equation (39):

$$P=L(t)\cdot\left(M(t)\frac{d^2P}{dt^2}+N(t)\cdot\frac{dP}{dt}+R(t)\right)+K(t) \tag{40}$$

we shall open brackets:

$$P=L(t)\cdot M(t)\frac{d^2P}{dt^2}+L(t)\cdot N(t)\frac{dP}{dt}+L(t)\cdot R(t)+K(t) \tag{41}$$

we shall denote:

$$-L(t)\cdot M(t)=A_*(t) \tag{42}$$

$$-L(t)\cdot N(t)=B_*(t) \tag{43}$$

$$L(t\cdot R(t)+K(t)=C_*(t) \tag{44}$$

We obtain a second-order inhomogeneous linear differential equation:

$$A_*(t)\cdot\frac{d^2P}{dt^2}+B_*(t)\cdot\frac{dP}{dt}+P=C_*(t) \tag{45}$$

When solving the resulting equation, conditions (initial, boundary) are required, based on the design parameters of the calculated mechanism.

Thus, the presented algorithm allows to ensure the continuity condition of the working fluid flow in the hydraulic drive of the lifting mechanism when it is lowered at the design stage. Calculation of the non-cavitation mode of the hydraulic drive is made taking into account the angular acceleration of the boom and the deformation of the working fluid and the wall of the hydraulic cylinder liner.

References

[1] Khokhlov V 1966 *Hydraulic power amplifiers* (Moscow: Nauka)

[2] Bashta T 1968 *Volumetric hydraulic drives* (Moscow: Machinebuilding)

[3] Gavrilenko B, Minin V and Rozhdestvensky S 1969 *Hydraulic drive* (Moscow: Machinebuilding)

[4] Lepeshkin A, Mikhailin A and Shepak A 2003 *Hydraulics and hydraulic drive* Ed A A Sheipak (Moscow: MSCHU)

[5] Nikitin O 2012 *Hydraulics and hydraulic drive* Ed O F Nikitin (Moscow: Bauman MSTU)

[6] Gursky A and Zhurbin V 2014 Mathematical model of the working equipment lifting mechanism with parallelogram linkage *Reshetnev State Siberian aerospace university: Reshetnevs Readings* 233-4

[7] Zhuravlev V 2001 *Fundamentals of technical mechanics* (Moscow: Publishing house of physical and mathematical literature)

[8] Khakhalin M and Zhurbin V 2014 Dynamic analysis of hydromechanical drive *Reshetnev State Siberian aerospace university: Reshetnevs Readings* 317-8

[9] Popov D, Panaiotti S and Ryabinin M 2014 *Hydraulics* Ed D N Popov (Moscow: Bauman MSTU)

[10] Belevich M 2006 *Hydromechanics. Fundamentals of classical theory* (SPb: RSGMU)

[11] Boltyansky V 2012 *Mathematical methods of optimal control* (Moscow: Book on Demand)

[12] Mustafina D, Rebro I, Kuzmin Sand Korotkova N 2009 Differentiation of the function of one and several variables with applications (Volgograd: VolSTU)

[13] Minin V 2010 Modeling of operational parameters of small loaders *Bulletin of Tomsk polytechnic university* **316(2)**

[14] Zubrilov G and Melnikov V 2015 Throttle control of the speed of lowering the boom of the hydraulic lift mechanism *Construction and road machines* 32-4

[15] Artbolevsky I 1988 *Theory of Mechanisms and Machines* (Moscow: Nauka)

[16] Prokofiev V 1973 The effect of fluid deformation on the dynamic response of the hydraulic drive (Moscow: Machinebuilding)