Investigation of operating regimes of locking seat hydrovalve with air-powered drive

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Abstract
Construction of locking seat hydrovalve with air-powered drive and hydraulic braking action, which excludes disadvantages of known constructions of locking hydrovalves, is considered in the article. There suggested refined mathematical model for investigation of operating regimes of suggested hydrovalve construction taking into account the change of temperature in the cage of air-powered drive while opening of hydrovalve. In result of investigation of operating regimes of pressure and bleeding hydrovalve it was stated, that suggested construction of hydrovalve has high speed of operation of functioning. Considered construction of locking seat hydrovalve allows fast action timing during exploitation with the help of throttle, entering into the hydrovalve.
Key words: LOCKING VALVE, AIR-POWERED DRIVE, NEEDLE VALVE, INVESTIGATION, SPEED, MOTION, ACTION TIME.

At the present time in metallurgical equipment hydraulic systems, which work on industrial water or emulsion, are widely used. Typical representatives of such hydraulic systems are the following systems: control of presses of pipe and wheel-mill production; control of supply mechanism and supply device of pilger tube rolling installations; hydraulic descaling; thermal hardening of rolled stock; cooling of working rolls of rolling mills; inter-stand cooling and others. In this hydraulic systems of control of technological process of hydraulic fluid consumption is fulfilled with the help of standard series-produced stop valving – valved distributors, cut-off valves or shutoff gate valve both with manual and electromechanical drive.

Exploitation experience of stop valving in the mentioned above hydraulic systems showed that valved distributors, cut-off valves and shutoff gate valves with electromechanical drive have one common disadvantage - slow response, app. 3-5 sec. Low speed of this reinforcing rod does not allow to create and build fast hydraulic system.

Recently in these hydraulic systems there frequently used bilinear two-stage seated hydraulic cutouts. Advantages of these valves are hermiticity, high speed of operation and reduction of wear rate because of seated performance, small losses of pressure both in hydraulic valves and in link channels, ability of hydraulic valve to fulfill several functions etc. Considered hydraulic valves are fixed both on the pressure and draining hydraulic lines.

From the patent and scientific and technical literature review there known bilinear two-stage seated hydraulic cutouts with air-powered drive, which are considered in depth in [1, 2]. However, stated hydraulic valves are noted to be of great complexity, require fitting with complex hydraulic equipment and the means for oil clarification and that is why they are reliable.

To simplify braking action of the ram of cutout seated hydraulic valve, before its fitting on the seating and so increase of its reliability, at Iron and Steel Institute of National Academy of Sciences of Ukraine there developed the construction of cutout seated hydraulic valve with braking liquid chamber placed in the body cavity of air-powered drive, which is considered in [3]. Considered hydraulic valve of ISI construction provides necessary braking action of the ram only while its placing on the seating. When the valve is being opened, ram braking action is not fulfilled, which affects the reliability of work and is the only main disadvantage of considered hydraulic valve.

There are cutout seated hydraulic valves with air-powered drive and hydraulic braking action [4], where there provided hydraulic valve ram braking while its opening and closing. However the experience of exploitation of hydraulic valves in hydraulic systems of metallurgical equipment, using industrial water as power fluid, showed that in the course of time there appear healing and blocking of orifice hole of brake assembly of hydraulic valve, which leads to increase of action time for hydraulic valve and for all hydraulic fluid power.

There also known the construction of cutout seated hydraulic valves with air-powered drive and hydraulic braking action, which is detailed considered in the work [5], eliminates disadvantages of known constructions of hydraulic valves. Suggested construction of cutout seated hydraulic valves with air-powered drive and hydraulic braking action provides necessary braking action of the ram in the end of its course, both while opening and closing. Herein there achieved increase in working reliability of the device due to significant simplification of means for ram braking during its seating and presence of rubber bumpers.

Operation of control cutout hydraulic valves influences greatly transition processes in hydraulic systems of metallurgical equipment. If choose wrong constructional parameters of hydraulic valve, there may appear hydraulic impacts in the hydraulic system. That is why during dynamic calculations of equipment hydraulic systems, the last should be considered together with dynamic model of controlling valves. Let us consider calculation scheme (fig. 1) of cutout hydraulic valve with air-powered drive and hydraulic braking action [5].

Equation of motion of moving elements of considered hydraulic valve in respect with viscous friction force and Coulomb friction is as follows [6, 7, 8]:

$$m_{val.}\frac{d^2y}{dt^2}-G_{val.}-p_{c.a..}F^{s.}_{a.p.}+C\left(p_{un.val.}-p_{ab.\,val.}\right)f^{s.}_{val.}+$$
$$+T_{fr.}\mathrm{sign}\left(\frac{dy}{dt}\right)+p_{br.ass.}f^{s.}_{b.d.}+h\left(\frac{dy}{dt}\right)+p_{un.val.}\left(f^{r.}_{br.ass.}-f^{s.}_{val.}\right)=0, \quad (1)$$

where m_{val} and G_{val} – mass and weight of moving elements of hydraulic valve; h - viscous friction coefficient; T_{fr} – total friction force in the hydraulic valve compaction; v and y – velocity and motion of ram of hydraulic valve; C – coefficient of hydrodynamic drag of fluid stream on the ram of hydraulic valve, which is determined by experiment. $F^{s}_{a.p.}$ –surface of air-powered drive piston of hydraulic valve; f^{s}_{val} – surface of hydraulic valve ram; $f^{s}_{b.d.}$ – surface of head end of orifice hole; $f^{r.}_{br.ass.}$ – surface of rod end of braking assembly of hydraulic valve; $f^{or.}_{h.}$ –surface of open flow area of orifice hole; $p_{c.a.}$ - stored air pressure in the cavity of air-powered drive of hydraulic valve; $p_{un.val.}$ – pressure of power fluid under the ram of hydraulic valve; $p_{ab.val.}$ - pressure of power fluid above the ram of hydraulic valve; $p_{br.ass.}$ - pressure of power fluid in the brake assembly of hydraulic valve, which is determined from the expression:

$$\begin{cases} \text{during opening}: \dfrac{dp_{br.ass.}}{dt} = \dfrac{E_l}{\left(l^0_{т.у.} + y\right)\cdot f^s_{b.d.}}\left(q^c_{n.val.} - f^s_{b.d.}\dfrac{dy}{dt}\right); \\ \text{during closing}: \dfrac{dp_{br.ass.}}{dt} = \dfrac{E_l}{\left(l^0_{br.ass.} + h_{val.} + y\right)\cdot f^s_{b.d.}}\left(f^s_{b.d.}\dfrac{dy}{dt} - q^{c.}_{n.val.}\right). \end{cases} \quad (2)$$

Here E_l – liquid elasticity modulus; $l^{o}_{br.ass.}$ – length of initial volume of working cavity of braking assembly; $h_{val.}$ – hydraulic valve ram displacement; $q^{c.}_{n.val.}$ - hydraulic fluid consumption through the needle valve.

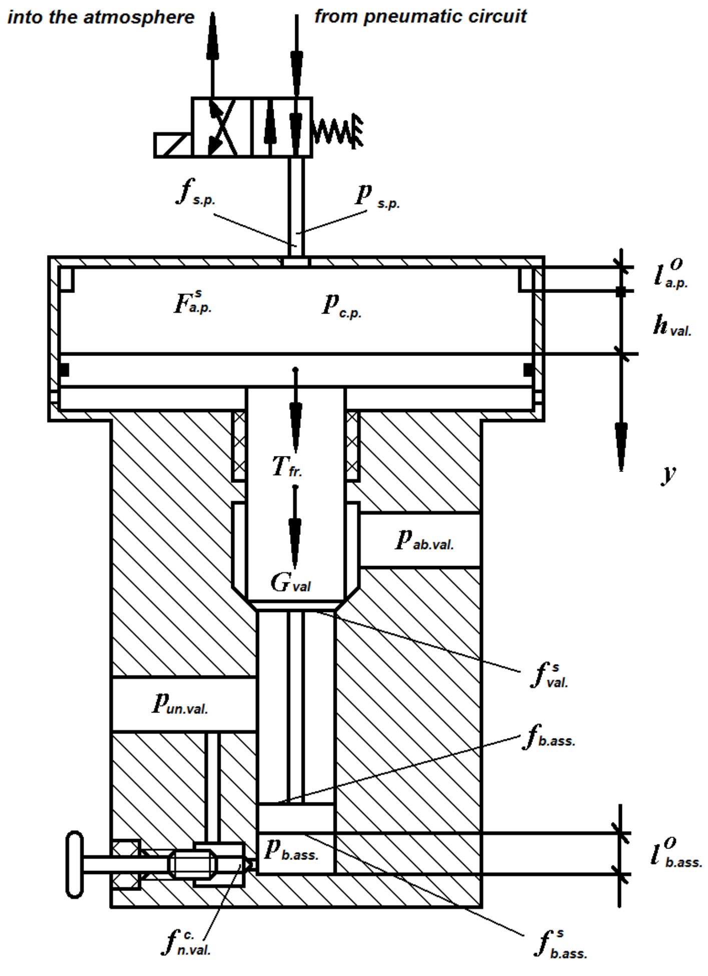

Figure 1. Design scheme locking saddle hydraulic valve with air-powered drive and hydraulic braking

Hydraulic fluid consumption through needle valve during filling or drawoff of the working cavity of hydraulic braking assembly may be determined from the expression:

$$q^{c}_{n.val.} = \mu^{c}_{n.val.} f^{c}_{n.val.} \sqrt{\frac{2}{\rho}\left|p_{un.val.} - p_{br.ass.}\right|}\, sign\left(p_{un.val.} - p_{br.ass.}\right), \quad (3)$$

where $\mu^{c}_{n.val.}$ – discharge coefficient through needle valve; $f^{c}_{n.val.}$ - passage area of needle valve.

Equation (1) of motion of hydraulic valve ram should be considered together with equation of change of compressed air pressure in the cavity of air-powered drive, which are considered in a detailed way in the works [6, 9, 10].

Equation of change of compressed air pressure in the cavity of air-powered drive during hydrovalve opening looks as follows:

$$\frac{dp_{c.a.}}{dt} = -\frac{k\mu_{s.p.} f_{s.p.} Kp_{c.a..}\sqrt{RT_{c.d.}}}{F^{s.}_{a.p.}\left(h_{val.} + l^{0}_{val.c..} + y\right)}\varphi(\sigma_1) + \frac{kp_{c.a.}}{h_{val.} + l^{0}_{val.c..} + y}\cdot\frac{dy}{dt}, \quad (4)$$

where k - ratio of specific heats; $\mu_{s.p.}$ – discharge coefficient through pneumatic pipeline during gas flow from the cavity of air-powered drive; $f_{s.p.}$ - passage area of pneumatic pipeline; R - gas constant; $T_{c.d.}$ – temperature of compressed air within the cavity of air-powered drive; $h_{val.}$ – hydraulic valve ram travel; $l^{o}_{val.c.}$ - length of initial volume of hydraulic valve cavity; $K = \sqrt{2k/(k-1)}$ - coefficient; $\varphi(\sigma_1)$ - consumable function determined from the expressions:

$$\varphi(\sigma_1) = \sqrt{\sigma_1^{\frac{2}{k}} - \sigma_1^{\frac{(k+1)}{k}}} \quad \text{at } 0{,}528 < \sigma_1 < 1; \quad (5)$$
$$\varphi(\sigma_1) = 0{,}2588 \quad \text{at } 0 < \sigma_1 \le 0{,}528.$$

Here $\sigma_1 = \left(p_{s.p.}/p_{c.a.}\right)$ is the function of pressure ratio; $p_{s.p.}$ – pressure of compressed air with the hydraulic pipeline.

In the methodology considered in [5], temperature of compressed air $T_{c.d.}$ in the equation (4) was taken as constant and equal to the temperature of compressed air in the hydraulic pipeline. According to the laws of thermodynamics it is known that during change of pressure in the cavity of air-powered drive, temperature of compressed air also changes. In this account the temperature of compressed air $T_{c.d.}$ in the equation (4) may be expressed through the pressure $p_{c.a.}$ on the base of adiabatic equation:

$$p_{c.a.}/p_{s.p..} = \left(T_{c.a..}/T_{s.p..}\right)^{k/(k-1)}. \quad (6)$$

Then we will obtain the following equation of change of compressed air pressure in the cavity of hydraulic drive during opening of hydraulic valve, which will be as follows [9, 10]:

$$\frac{dp_{c.a.}}{dt} = -\frac{k\mu_{s.p..} f_{п.м.} Kp^{(3k-1)/2k}_{c.a..}\sqrt{RT_{s.p.}}}{F^{s.}_{a.p..}\left(h_{val.} + l^{0}_{c.a.} + y\right)p^{(k-1)/2k}_{s.p.}}\varphi(\sigma_1) + \frac{kp_{c.a.}}{h_{val} + l^{0}_{c.a.} + y}\cdot\frac{dy}{dt}. \quad (7)$$

Equation of change of compressed air pressure in the cavity of hydraulic drive during closing of hydraulic valve will be as follows [9, 10]:

$$\frac{dp_{c.a.}}{dt} = \frac{k\mu_{s.p.} f_{s.p.} Kp_{s.p.}\sqrt{RT_{s.p.}}}{F^{s.}_{a.p..}\left(l^{0}_{c.a.} + y\right)}\varphi(\sigma_2) - \frac{kp_{c.a..}}{l^{0}_{c.a.} + y}\cdot\frac{dy}{dt}, \quad (8)$$

Where $T_{s.p.}$ – temperature of compressed air in the hydraulic pipeline; $\varphi(\sigma_2)$ - consumable function determined from the expressions:

$$\varphi(\sigma_2) = \sqrt{\sigma_2^{\frac{2}{k}} - \sigma_2^{\frac{(k+1)}{k}}} \quad \text{at } 0{,}528 < \sigma_2 < 1; \quad (9)$$
$$\varphi(\sigma_2) = 0{,}2588 \quad \text{at } 0 < \sigma_2 \le 0{,}528.$$

Here $\sigma_2 = \left(p_{c.a..}/p_{s.p.}\right)$ – function of pressure ratio

As the example let us investigate the regimes of opening/closing of pressure and draining hydraulic valves, being a part of hydromechanical system of the press with force 20 MN of wheel-mill line of Interpipe NTZ at the following parameters: $F^{s.}_{a.p.}$ = 0.152 m^2; $f^{s}_{b.d.} = 0.0023\,m^2$; $f^{r}_{br.ass.} = 0.002123\,m^2$; $f^{s}_{val.} = 0{,}002123\,m^2$; $m_{val.}$ = 23kg; $h_{val.}$ = 0.013m;

$h = 2{,}8 \cdot 10^{-4}; T_{fr} = 150\ N; C = 0.001;\ k = 1.4;$
$E_1 = 2 \cdot 10^9 Pa; \rho = 1000\ kg/m^3\ p_{s.p.} = 0.1 MPa;$
$\mu_{s.p.} = 0.4; l^0_{br.ass.} = l^0_{c.a.} = 0.05\ m;$
$f_{s.p.} = 3.14 \cdot 10^{-4}\ m^2; T_{s.p.} = 288\ K;$
$R = 287\ J/(kg \cdot K); \mu^c_{n.val.} = 0.2;$
$f^c_{n.val.} = 7.85 \cdot 10^5\ m^2; p_{ab.val.} = 0.75 \cdot 10^6\ Pa.$

Pressure under the ram of delivery valve $p_{un.val.}$ was taken at its opening and closing as $p_{un.val.} = 30 \cdot 10^6$ Pa, basing on the presence of constant high pressure in the hydraulic pipeline. Pressure under the ram of draining hydraulic valve $p_{un.val.}$ was taken at its opening as $p_{un.val.} = 21 \cdot 10^6$ Pa, basing on the presence of high pressure in hydraulic pipeline and at its closing - $p_{un.val.} = 0.75 \cdot 10^6$ Pa, basing on the presence of low pressure in the hydraulic pipeline.

Herein varying parameter during research of working regimes of cutout seating hydraulic valve with air-powered drive and hydraulic braking action there was chosen a scale of passage area of needle valve $f^{c.}_{n.val.}$.

Research results of working regimes of cutout seating hydraulic valve during opening and closing are presented in the form of velocity graphs $v_{val.}$ and motion $y_{val.}$ of rams of pressure (fig.2) and draining (fig.3) hydraulic valves depending on the scale $f^{c.}_{n.val.}$.

On the base of analysis of working regime of pressure hydraulic valve during opening and closing, obtained during different $f^{c.}_{n.val.}$ values, one may conclude, that the character of change $v_{val.}$ of traversing speed of the ram and y_{val} of ram movement on the set value are not similar and differ significantly. Herein with the increase of $f^{c.}_{n.val.}$ of passage area the velocity v_{val} of the ram of hydraulic valve increases, which affects operation speed of the valve. This points to the need of setting of passage area of the throttle $f^{c.}_{n.val}$, due to the necessary actuating time and need of calculating the parameters of hydromechanical system, excluding hydraulic impacts.

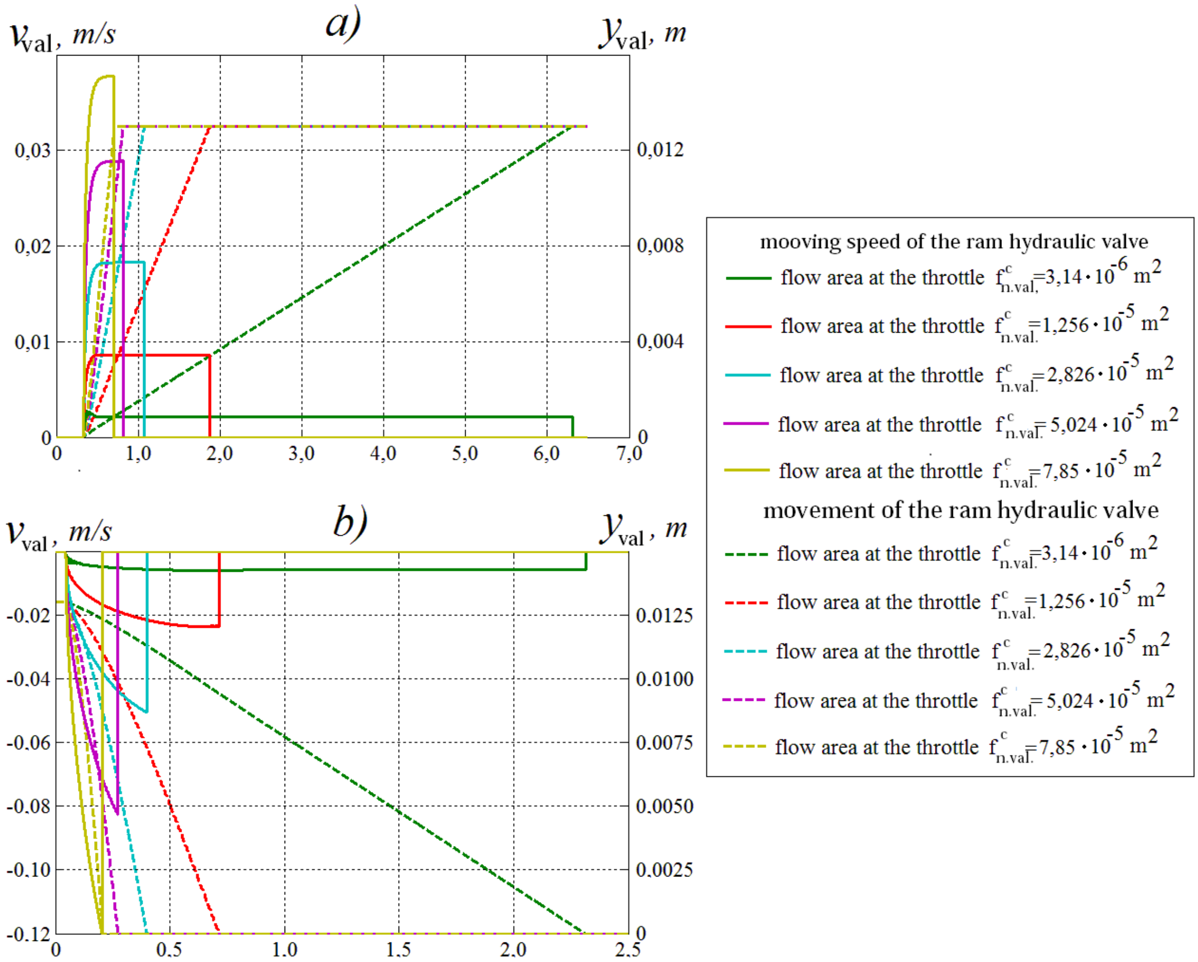

Figure 2. Operating modes of pressurized hydraulic valve at the opening (a) and closing (b) depending on the throttle flow area $f^{c.}_{n.val.}$

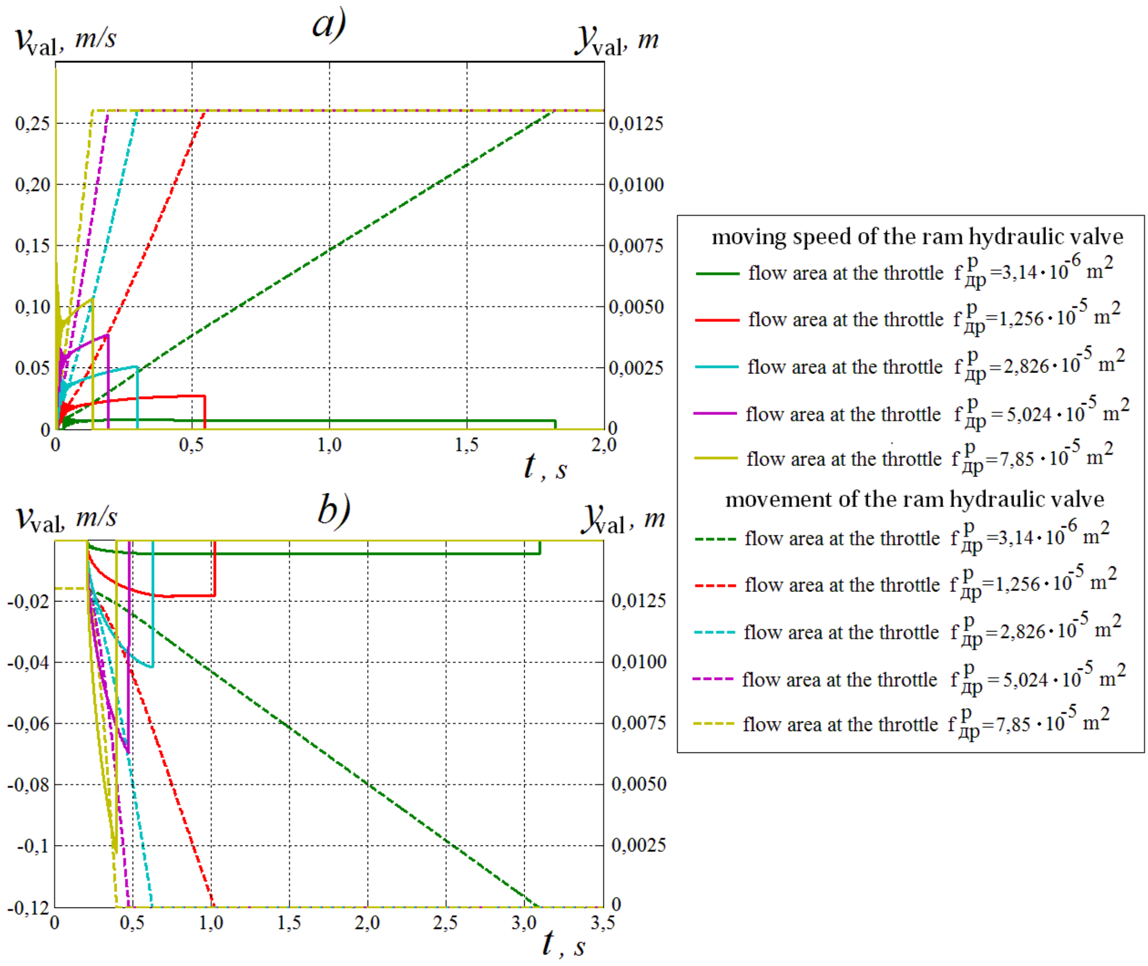

Figure 3. Operating modes of drain hydraulic valve at the closing (a) and opening (b) depending on the throttle flow area $f^{c.}_{n.val.}$.

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