MODELING OF A THREE WAY ROTATABLE FLUID DISTRIBUTOR USED TO COMMAND AND CONTROL A HYDRAULIC ROCK DRILL

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Abstract:
A hydraulic command system comprising a combination of two hydraulic half bridge elements of type A and E is analyzed. The system can be used to command and control an impact mechanism for hydraulic rock drills. The A+E half bridges combination reflects the presence of the active and passive hydraulic resistances. Under the form of a three-way valve with angular command displacement, the A+E circuit is used to control a hydraulic differential motor. Unlike a linear valve, for this rotary valve, shock waves are reduced. Moreover, its positioning in a structure, as it is proposed, leads to minimized parasitic hydraulic capacities. Two methods of calculus the circular section area through which the flow crosses the rotary valve are developed. The characteristics of the three way rotary valve are expressed mathematically and graphically. The hydraulic rotatable distributor – linear hydraulic motor subassembly is mathematical analyzed.

Key words: *hydraulic rotary valve, three ways valve, pressure-flow curves, half bridge, hydraulic resistance, steady-state characteristics, valve coefficients, area*

1. Introduction

A hydraulic impact mechanism presented in [1], [2] and [12], is analyzed. The percussion piston of the impact mechanism is controlled by use of an innovative command and control structure which is a rotatable distributor. The system formed by the rotatable distributor and the hydraulic motor of the impact piston minimizes the parasitic hydraulic capacities, the weight, and improves the dynamic behavior of the piston.
The present paper is a completion of the mathematical modeling from [3].

2. The hydraulic rotating distributor

A rotatable spool which is to be patented is presented and analyzed in figure 1. The rotatable spool presents axially slots, notated with P and T, made circumferentially and positioned alternatively and reversely. These slots operate as input and return flow distribution slots.

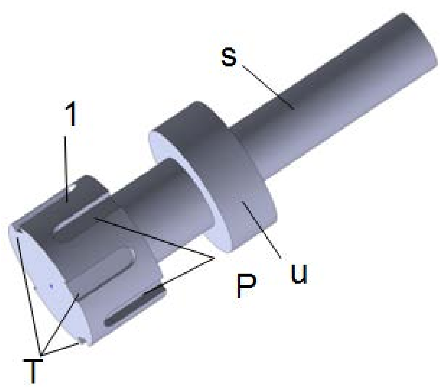

Fig.1. A rotating spool 1 with: T and P – slots; u – shoulder; s – shaft.

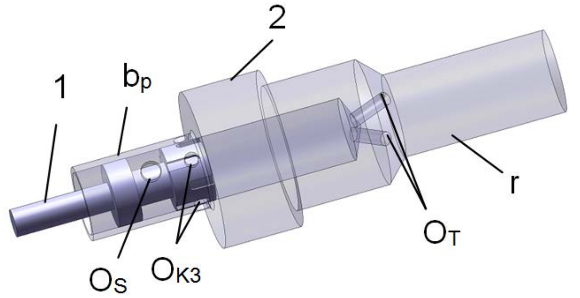

Fig.2. The rotating spool positioned in the body of a motor piston, where: 2 – the impact piston; r – piston rod; b_p – piston prominence; O_{K3}, O_S, O_T – orifices.

The sleeve of the rotatable spool is constituted by the prominence b_p of the piston 2, as suggested in figure 2. Thus, the rotatable spool 1 is contained within the impact piston 2 and the shaft s imparts a rotation movement to the spool. The rotation motion of the rotatable spool can be provided by means of a hydraulic motor, an electric motor or a hydraulic turbine, through the shaft s.

The slots P and T communicate with the working chamber C_2 of a hydraulic linear motor MHL, as in figure 2 and figure 3, by means of a number of orifices O_{K3} realized through the piston prominence b_p. The circumferentially slots are brought alternatively in communication through four orifices O_{K3} with the chamber C_2. Thus, the working chamber C_2 is fed with hydraulic fluid through the orifices O_{K3} and the evacuation of the fluid from the chamber C_2 is realized through the same orifices O_{K3}. Until the chamber C_2 is connected to the return line through the slots T and the orifices O_T, a shoulder u confines the fluid fed through the orifices O_S, the annular chamber delimitated by the shoulder u and the spool head acts as an accumulator.

A hydraulic scheme of the subassembly rotary valve – linear hydraulic motor is illustrated in figure 3. The subassembly is proposed by [1] to be used in rock drilling, with a proper rotary mechanism, hydraulic accumulators and other hydro-mechanical elements needed to perform the drilling process.

By use of the four inlet slots P and four return slots T of the rotatable distributor DHR (figure 3), the working chamber C_2 is supplied intermittently with hydraulic fluid from a hydraulic pressure source PV and consecutively returned to the tank T. By supplying fluid under pressure and rotating the spool 1, the piston executes a linear oscillatory motion. A safety valve SS is mounted in the high pressure circuit.

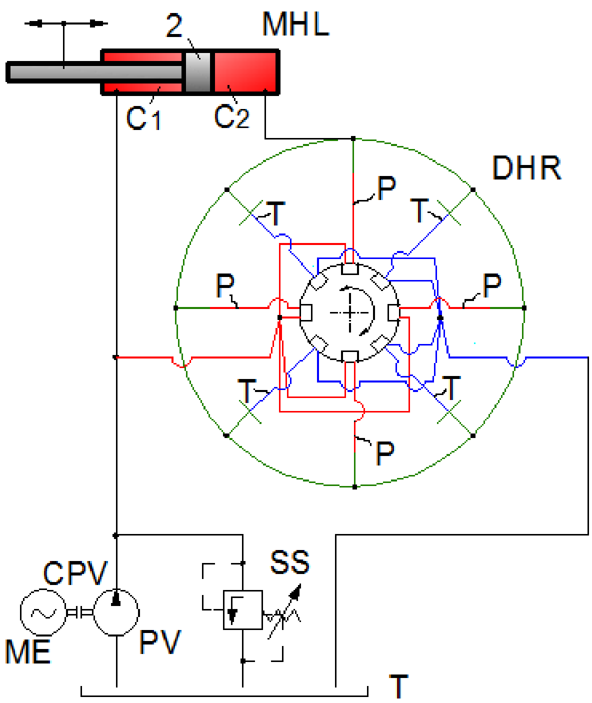

Fig.3. The hydraulic scheme of the system comprising: ME – electrical motor; CPV – pump coupling; PV – volumetric pump; SS – safety valve; T_1, T_2 – tank; DHR – hydraulic rotary distributor; MHL – linear hydraulic motor; C_2, C_1 are the motor chambers.

In figure 4, a hydraulic scheme of the subassembly rotary valve – linear hydraulic motor outlines the command and control structure of the motor MHL. The hydraulic resistances R_i and R_e of the command half bridge have variable flow cross areas but there can't be made displacement commands as for a linear distributor. When a stepper motor is used, working positions as illustrated in

figure 5 can be obtained. Using other type of motor the opening and the closing of the orifices O_{K3} are realized continuously.
The command half bridges is of type A+E. The input resistances R_i are formed by the slots P and the piston prominence b_p. In a similar manner, the discharge slots T form the output resistance R_e when the motor chamber C_2 is connected to the tank.

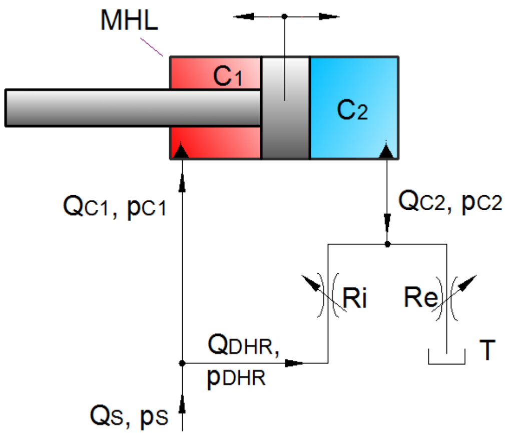

Fig.4. The A+E half bridge and the hydraulic linear motor MHL, where: Q_{C1}, Q_{C2} – supply/exhaust flow; Ri, Re – hydraulic resistances; Q_{DHR} – the distributor input flow; Q_S, p_S – supply flow, respectively, supply pressure.

A type hydraulic bridge is represented by two hydraulic resistances both with variable sectional cross areas.
E type hydraulic bridge is always used with hydraulic linear differential motors [13].
This combination of half bridges A+E can be studied with the rest of the combinations (matrix of hydraulic bridges) in [4], [5] where they are symbolically and constructively presented and analyzed.
In figure 5, a radial view of the spool-piston assembly is presented correspondingly to the two working positions of the rotating distributor.

a) the feeding position; b) the exhaust position.

Fig.5. The working positions of the rotatable distributor with their corresponding sectional views through the rotating spool and its sleeve.

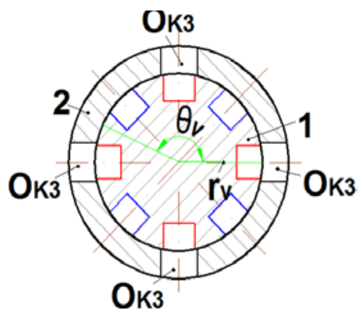

Fig.6. Radial view through the rotatable distributor DHR,where:
θ_v – the central angle of the rotary spool; r_v – the spool radius.

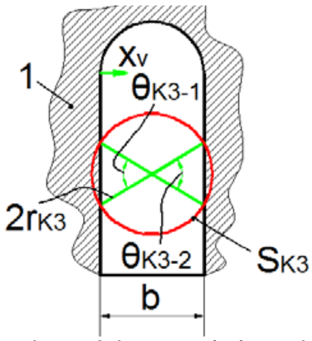

Fig.7. Top view of the spool slot, where: S_{K3} – the section of the orifice O_{K3}; θ_{K3-1}, θ_{K32} – central angles.

3. Equations and mathematics

The flow equations of the hydraulic rotating distributor are:

$$Q_{C2} = \begin{cases} C_d \cdot A_{K3}(x_v) \cdot \sqrt{\frac{2}{\rho} \cdot (p_{DHR} - p_{C2})} & 0 \le x_v \le 2r_{K3} + b \\ C_d \cdot A_{K3}(x_v) \cdot \sqrt{\frac{2}{\rho} \cdot p_{C2}} & 0 \ge x_v \ge -2r_{K3} - b \end{cases} \quad (1)$$

Where: Q_{C2} – the flow sent to the motor through the distributor DHR; C_d – the discharge coefficient; A_{K3} – the flow passing area through an orifice O_{K3}; x_v – the spool displacement; α is the piston area ratio; p_{DHR} – the pressure of the fluid delivered to the distributor DHR; p_{C2} – the pressure of the fluid sent to the motor MHL; ρ – the fluid density; b – the slot wide; r_{K3} – the radius of the orifice O_{K3}.
The notations that appear in the flow expressions (1) are presented also in figures 4, 6 and 7.
In figure 6 and figure 4 it is illustrated a radial view through the impact piston – rotating spool subassembly, revealing four orifices O_{K3} displayed circularly and eight grooves. From the top view of one spool groove, figure 7, the parameters needed in cross area computation are marked.

3.1 The calculus steps of the flow passing area

The spool displacement is an instant variable that can be calculated regarding the central angle of the rotary spool, θ_v, using the expression (2).

$$x_v = \frac{\pi \cdot r_v \cdot \theta_v}{180^\circ} \quad (2)$$

The parameter x_v is used also to determinate the flow passing area A_{K3}.
For x_v varying between 0 and $2r_{K3}$+b, the effective passing area to motor chamber can be calculated with the expressions (3), where central angles θ_{K3-1}, θ_{K32} are in radian.
The variation of the dimensionless flow passing area is presented in figure 8. This variation begins with a full obstructed orifice O_{K3} and by keeping rotating the spool in the same direction the area function will have a peak when the orifice O_{K3} is full opened.

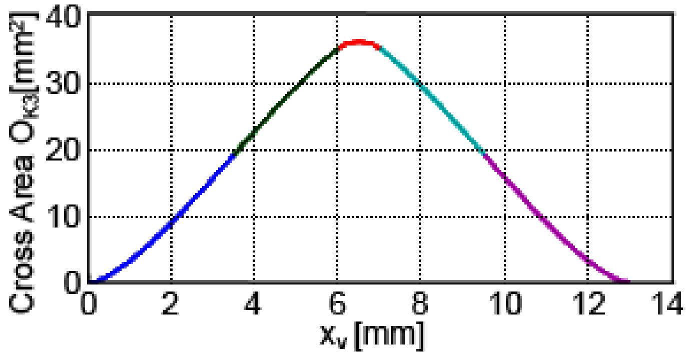

Fig. 8. The variation of the passing area corresponding to one spool slot.

$$A_{K3}(x_V)=$$
$$\begin{cases} r_{K3}^2\cdot\left(\left(2\arccos\frac{r_{K3}-x_V}{r_{K3}}\right)\cdot\frac{1}{2}-\frac{1}{2}\cdot\sin\left(2\arccos\frac{r_{K3}-x_V}{r_{K3}}\right)\right), & 0\le x_V\le r_{K3}\\ r_{K3}^2\cdot\left(\pi-\frac{\left[\left(2\arccos\frac{x_V-r_{K3}}{r_{K3}}\right)-\sin\left(2\arccos\frac{x_V-r_{K3}}{r_{K3}}\right)\right]}{2}\right), & r_{K3}\le x_V\le b\\ {}_{K3}^{2}\cdot\left(\pi-\frac{\left(\left(2\arccos\frac{r_{K3}-x_V+b}{r_{K3}}\right)-\sin\left(2\arccos\frac{r_{K3}-x_V+b}{r_{K3}}\right)\right)}{2}-\frac{\left(\left(2\arccos\frac{x_V-r_{K3}}{r_{K3}}\right)-\sin\left(2\arccos\frac{x_V-r_{K3}}{r_{K3}}\right)\right)}{2}\right), & b\le x_V\le 2r_{K3}\\ {}_{K3}^{2}\cdot\left(\pi-\frac{\left(2\arccos\frac{r_{K3}-x_V+b}{r_{K3}}\right)-\sin\left(2\arccos\frac{r_{K3}-x_V+b}{r_{K3}}\right)}{2}\right), & 2r_{K3}\le x_V\le r_{K3}+b\\ {}_{K3}^{2}\cdot\left(\left(2\arccos\frac{x_V-r_{K3}-b}{r_{K3}}\right)\cdot\frac{1}{2}-\frac{1}{2}\cdot\sin\left(2\arccos\frac{x_V-r_{K3}-b}{r_{K3}}\right)\right), & r_{K3}+b\le x_V\le 2r_{K3}+b \end{cases} \quad (3)$$

By substituting in figure 7 the central angles θ_{K3-1}, θ_{K32} with $2\theta_{K3-1}$, respectively θ_{K32}, the expression (3) can be rewritten as:

$$A_{K3}(x_V)=\begin{cases} r_{K3}^2\cdot(\theta_{K3-1}-\sin\theta_{K3-1}\cdot\cos\theta_{K3-1}), & x_V<r_{K3}\\ r_{K3}^2\cdot(\pi-\theta_{K3-2}+\sin\theta_{K3-2}\cdot\cos\theta_{K3-2}), & r_{K3}<x_V<2\cdot r_{K3} \end{cases} \quad (4)$$

The functions sine and cosine from the expression (4), for $0<x_V<r_{K3}$ are developed below:

$$\cos\theta_{K3-1}=\frac{r_{K3}-x_V}{r_{K3}}=1-\frac{2\cdot x_V}{d_{K3}} \quad (5)$$

$$\sin\theta_{K3-1}=\frac{\sqrt{r_{K3}^2-(r_{K3}-x_V)^2}}{r_{K3}}=\sqrt{\frac{4\cdot x_V}{d_{K3}}\cdot\left(1-\frac{x_V}{d_{K3}}\right)} \quad (6)$$

The functions sine and cosine from the expression (4), for $r_{K3}<x_V<2r_{K3}$ are developed below:

$$\cos\theta_{K3-2}=\frac{2\cdot x_V}{d_{K3}}-1 \quad (7)$$

$$\sin\theta_{K3-2}=\sqrt{\frac{2\cdot x_V}{r_{K3}}\cdot\left(1-\frac{x_V}{2\cdot r_{K3}}\right)}=2\cdot\sqrt{\left(1-\left(1-\frac{x_V}{d_{K3}}\right)\right)\cdot\left(1-\frac{x_V}{d_{K3}}\right)} \quad (8)$$

Where: d_{K3} – the diameter of the orifice O_{K3}.

To bring the area function A_{K3} into a simpler and compact form, it can be used the steps from [6], where the fluid passing area through a circular plan surface is theoretically computed. In this manner, the function sine, cosine and arcos are developed using binomial and Maclaurin (Taylor) theorems from [7].

By expressing the functions (5), (6), (7), (8) as Binomial and Taylor series, the obtained expressions depend on the number of binomial and maclaurin terms selected (usually the first three or four terms, omitting the higher powers).

For a flow cross area variation from zero to full opening, the calculus area expression is:

$$A_{K3}(xv) = d_{K3}^2 \cdot \frac{4}{3} \cdot \left(\frac{x_V}{d_{K3}}\right)^{\frac{3}{2}} \cdot \left(1 - \frac{3}{10} \cdot \left(\frac{x_V}{d_{K3}}\right)\right) \quad (9)$$

For a flow cross area variation from the full opening to a complete obstruction, the calculus area expression is:

$$A_{K3}(x_V) = d_{K3}^2 \cdot \left(\frac{\pi}{4} - \frac{4}{3} \cdot \left(1 - \frac{x_V}{d_{K3}}\right)^{\frac{3}{2}} \cdot \left(1 - \frac{3}{8} \cdot \left(1 - \frac{x_V}{d_{K3}}\right)\right)\right) \quad (10)$$

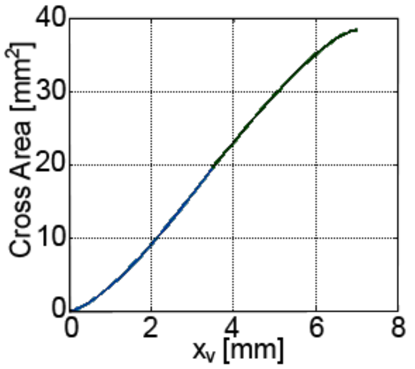

Fig.9. The cross area variation in the rotating hydraulic distributor, obtained with binomial theorems, from zero to a full opening.

By applying binomial and Maclaurin theorem approximations are made. To obtain a better accuracy the error of the approximations can be adjusted by modifying the number of series coefficients.

In figure 9, the resulting cross area variation from the closed position to the full opening of a resistance is illustrated. The diameter is set of 80 mm.

3.2 The null operating point of the rotatable distributor

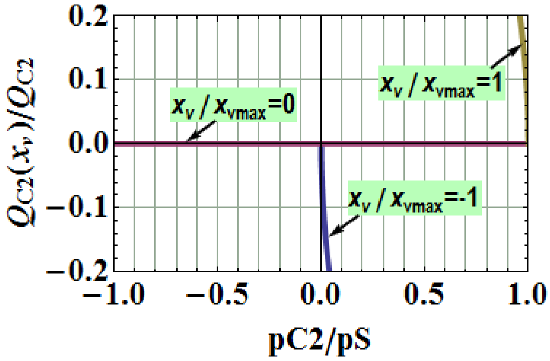

Fig.10. The pressure-flow curves for three different command displacements of the rotatable spool: x_v/x_{vmax} =-1; x_v/x_{vmax} = 0; x_v/x_{vmax} =1.

In figure 10, it can be observed that for a null command displacement (x_V), this is when the spool is symmetrical positioned in the distributor body, the flow sent to the motor is zero. Thus, the null operating point of the rotatable distributor is characterized by:

$$Q_{C2}(x_V, p_{C2}) = x_V = 0 \quad (11)$$

Based on the figure 10 the statement written referring to the expression (11) is valid for any control pressure, the following condition must be fulfilled:

$$p_{C2-PSF} = \alpha \cdot p_S \quad (12)$$

The condition (12) has to be disposed because in a steady state analyze the pressures in the linear hydraulic motor must set the piston in equilibrium.

3.3 The pressure-flow curves of the rotatable distributor

In figure 11 it is illustrated the pressure-flow curves for different command displacements of the rotary spool of the hydraulic rotatable distributor DHR which is used to command and control the linear hydraulic motor MHR (figure 3). The pressure-flow curves of a linear three way control valve are presented in figure 12 and are identical with those of the rotary valve.

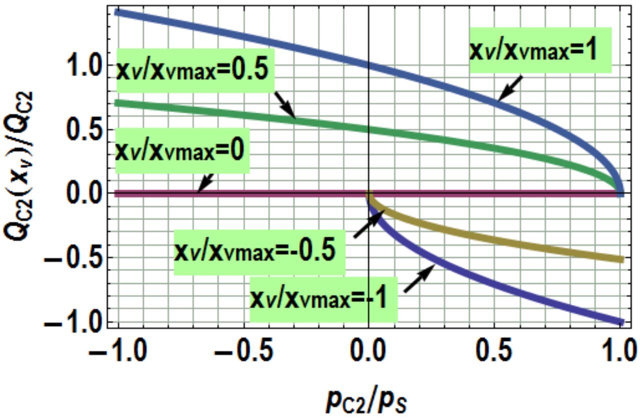

Fig.11. The pressure – flow curves of the rotatable hydraulic distributor for different spool command displacements.

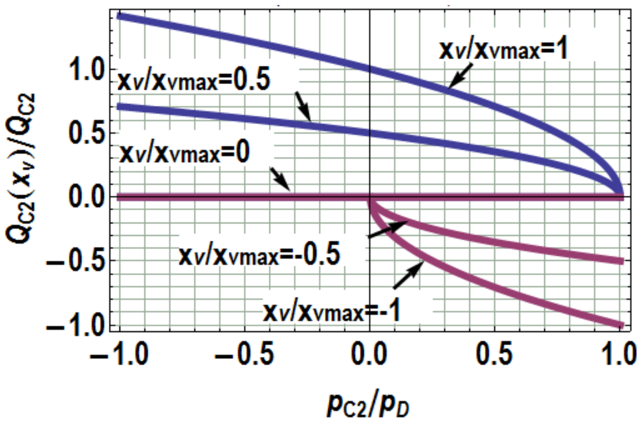

Fig.12. The pressure – flow curves of a linear hydraulic distributor for different spool command displacements.

Similar pressure-flow curve plots are revealed and analyzed in many research labors as [8], [9], [10].

3.4 The rotatable distributor coefficients and the linearized flow equation

Usually, the nonlinear algebraic equation (1) which describes the pressure-flow curves illustrated in figure 10 can be seen as a Taylor series and expressed regarding to the valve null operating point. The linear equation which is obtained by developing the equation (1) as a Taylor polynomial of degree 1 is easier to analyze.
Similarly with the steps used in [8], the partial derivatives which appear in the Taylor polynomial are obtained by differentiation or graphically from the pressure-flow curve plots (figure 9 or 10). The rotatable distributor coefficients are defined by these partial derivatives and the computed values are:

$$K_{Q-xv} = 0 \tag{13}$$

$$K_{Q-p} = 0 \tag{14}$$

$$K_p = \infty \tag{15}$$

The flow gain or the partial derivative of the load flow with respect to rotary spool displacement, as is defined a valve flow-displacement coefficient, is the most important parameter for a valve [5-9] but in the same time questionable [11]. The valve pressure drop is always changing yielding to a variable flow gain [11]. As the pressure drop on rotatable distributor changes, the system flow gain will not be constant.
Flow-displacement coefficient of a valve is proportional with system flow gain.
Based on the results (13) and (14), the general linearized flow equation is:

$$\Delta Q_{C2}(x_v, p_{C2}) = 0 \tag{16}$$

The flow differentiation is determined for an infinitesimal variation, in the vicinity of the null operating point. In consequence, a zero result (16) will show that for infinitesimal variation of the rotating spool displacement and for infinitesimal variation of the control pressure in steady state regime, theoretically, there will not be changes in the load flow.

4. Mathematical modeling of the percussion mechanism

In figure 13 it is created a hydraulic scheme for the percussion mechanism where the parameters used in the mathematical modeling of the impact system are presented.

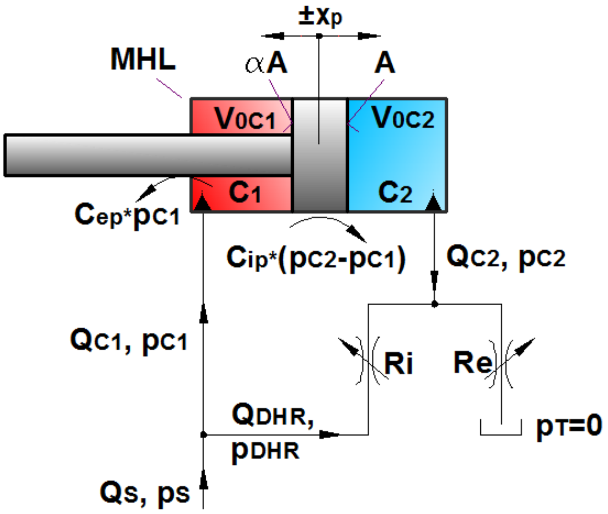

Fig.13. The symbolical hydraulic scheme of the subsystem, where: A, αA – the piston areas with α the ratio between them; C_{ep} is the external leakage coefficient; C_{ip} is the internal leakage coefficient; p_{C1} and p_{C2} are the pressures in the supply, respectively the discharge lines; x_p is the impact piston displacement.

4.1 The continuity equations for steady one-dimensional flow

The continuity equations of the liquid fluid in the linear motor chambers are:

$$Q_{C1} = \frac{V_{0C1}}{E} \cdot \frac{\Delta p_{C1}}{\Delta t} + \frac{\Delta V_{C1}}{\Delta t} + C_{ep} \cdot p_{C1} + C_{ip} \cdot (p_{C1} - p_{C2}) \quad (17)$$

$$Q_{C2} = \frac{V_{0C2}}{E} \cdot \frac{\Delta p_{C2}}{\Delta t} + \frac{\Delta V_{C2}}{\Delta t} + C_{ip} \cdot (p_{C2} - p_{C1}) \quad (18)$$

By summing the continuity equations (17) and (18), and by using the notations (20), (21), (22) and (23) it gives:

$$Q_{Load} = \frac{1}{2} \cdot \left(\frac{A \cdot x_p}{E} \cdot C_{pLoad\_1} \cdot \dot{p}_{Load} + (\alpha + 1) \cdot A \cdot \dot{x}_p + C_{pLoad\_2} \cdot p_{Load} \right) \quad (19)$$

Where:

$$Q_{Load} = \frac{Q_{C2} + Q_{C1}}{2} \quad (20)$$

$$p_{Load} = p_{C2} - p_{C1} \quad (21)$$

$$C_{pLoad\_1} \cdot p_{Load} = p_{C2} + \alpha \cdot p_{C1} \quad (22)$$

$$C_{pLoad\_2} \cdot p_{Load} = C_{ep} \cdot p_{C1} \quad (23)$$

Where: E – the liquid bulk modulus of elasticity; m_p – the total inertial mass reduced to impact piston mass; p_{Load} – the pressure drop on the hydraulic linear motor; Q_{Load} – the average volumetric flow rate supplied to the motor chambers; x_{pmax} – the maximum stroke of the impact piston.

4.2 The equation of movement

The equation of impact piston movement is:

$$m_p \cdot \ddot{x}_p + \mu \cdot \dot{x}_p + F_{pC1} = F_{pC2} \tag{24}$$

Where: μ – the friction coefficient; F_{pC1}, F_{pC2} – the pressure forces acting on the impact piston active surfaces.
The following notation is made:

$$C_{pLoad\_3} \cdot p_{Load} = p_{C2} - \alpha \cdot p_{C1} \tag{25}$$

Using the notation (25), the equation (24) can be rewritten as:

$$p_{Load} = \frac{m_p}{A \cdot C_{pLoad\_3}} \cdot \ddot{x}_p + \frac{\mu}{A \cdot C_{pLoad\_3}} \cdot \dot{x}_p \tag{26}$$

4.3 The system characteristic equation

The system characteristic equation is:

$$\frac{2}{s} \cdot Q_{Load}(s) = X_p(s) \cdot \left[\left(\frac{\frac{A \cdot x_{p\max}}{E} \cdot C_{pLoad\_1} \cdot m_p}{A \cdot C_{pLoad\_3}} \cdot s^2 + \frac{\frac{A \cdot x_{p\max}}{E} \cdot C_{pLoad\_1} \cdot \mu}{A \cdot C_{pLoad\_3}} \cdot s \right) + \left(\frac{C_{ep} \cdot C_{pLoad\_2} \cdot m_p}{A \cdot C_{pLoad\_3}} \cdot s + \frac{C_{ep} \cdot C_{pLoad\_2} \cdot \mu}{A \cdot C_{pLoad\_3}} \right) + A \cdot (1 + \alpha) \right] \tag{27}$$

The system characteristic equation gives a natural pulsation ω_n in the presence of damping and leakages defined as:

$$\omega_n = \sqrt{\frac{E \cdot \left(A^2 \cdot (1+\alpha) \cdot C_{pLoad\_3} + C_{ep} \cdot C_{pLoad\_2} \cdot \mu \right)}{A \cdot x_{p\max} \cdot C_{pLoad\_1} \cdot m_p}} \tag{28}$$

And a damping ratio δ defined as:

$$\delta = \frac{1}{2} \cdot \left(\frac{\mu}{m_p} + \frac{E \cdot C_{ep}}{A \cdot x_{p\max}} \cdot \frac{C_{pLoad\_2}}{C_{pLoad\_1}} \right) \cdot \sqrt{\frac{A \cdot x_{p\max} \cdot C_{pLoad\_1} \cdot m_p}{E \cdot \left(A^2 \cdot (1+\alpha) \cdot C_{pLoad\_3} + C_{ep} \cdot C_{pLoad\_2} \cdot \mu \right)}} \tag{29}$$

These two characteristic parameters describe the dynamic behavior of the analyzed subassembly through the expressions (28) and (29).

5. Discussions and conclusions

An innovating command structure for a hydraulic impact mechanism used in industrial applications such as rock, concrete or asphalt penetration is presented, described and a mathematical modeling is developed.
The command structure is essentially a rotary spool with circular grooves and mounted in a piston prominence. Cylindrical orifices are radial drilled in the piston prominence to allow the fluid flow from the rotary spool grooves into a working motor chamber. The rotary movement of the hydraulic rotary spool and the hydraulic feeding of the motor assure a linear oscillatory movement for the impact piston.
Two methods of calculus the flow cross area through the hydraulic resistances of the hydraulic rotating distributor are revealed.

The pressure – flow curves of the hydraulic rotating distributor are computed.
The hydro mechanical impact mechanism is assumed to be a dynamic system with constant parameters unlike the real situation according to which the coefficients are determined by the impact mechanism position. Moreover, the liquid bulk modulus is assumed to be constant in the present mathematical modeling.
The obtained linearized flow equation of the hydraulic rotating distributor equals zero. A mathematical interpretation for this result is that for small deviation of the rotary spool from its steady symmetrical position in the piston prominence, the hydraulic linear motor – hydraulic rotary distributor subassembly behaves as a hydraulic linear motor.

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