TECHNICAL SOLUTIONS FOR HYDRAULIC SUPPLY SYSTEM OF THE DRILLING EQUIPMENT

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Abstract: The hydraulic supply system of the drilling installations is provided with a pressure control valve. Since the operating principle for currently produced driven pressure valves does not allow the efficient operation of the hydraulic supply system for drilling installations, there was studied, conceived and designed a new constructive solution for disconnecting valves. The proposed solution leads to the enhancement of the hysteresis, with the possibility to be adjusted together with the adjustment of the pressure value for idle hydraulic pump operation and of the maximum operating pressure for the hydraulic circuit.

Key-words: hydraulic supply system, disconnection valve, drilling equipment

1. INTRODUCTION

The energy feeding of hydraulic consumers (rotary and linear hydraulic engines) part of the mining drilling installations is done by an electric-hydraulic system. This is usually composed of an electric-hydraulic group, a pressure valve block and a hydraulic battery. When the hydraulic consumers are not fed (while drilling), the hydraulic pump will operate at maximum pressure or close to it, while the safety valve is connected intermittently to the tank with high frequency. This leads to an additional consumption of electrical energy, the hydraulic agent gets overheat and worsens the value of its parameters (viscosity), while the hydraulic pump and the safety valve is overused. The removal of these disadvantages is possible when the hydraulic energy supply system allows the operation of the electric-hydraulic group at low charge while the hydraulic consumers of the drilling installation are not fed.

2. THE HYDRAULIC SUPPLY SYSTEM

The hydraulic supply system of the drilling installation, Romanian design type

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IMP-2, switches the operation of the hydraulic pump to low charge while idle (when all the hydraulic feeders are neutral).

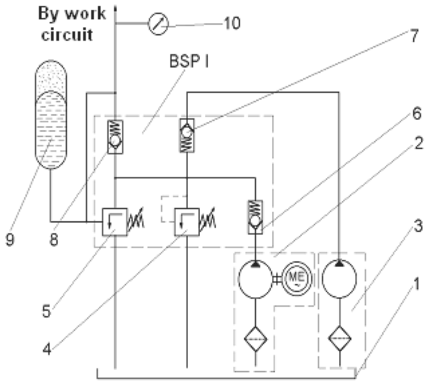

Fig. 1. The principle scheme of the hydraulic supply system.

Figure 1 presents the general hydraulic scheme of the hydraulic supply system for the drilling installation IMP-2. The scheme contains the following elements: 1 - the tank; 2 – the axial piston hydraulic pump engaged by the electric engine ME; 3 – manual hydraulic pump; 4 – safety valve; 5 – driven disconnection valve; 6, 7, 8 – one way valves; 9 – nitrogen hydraulic accumulator; 10 – manometer.

If the hydraulic consumers of the drilling installation are not fed (when all the hydraulic feeders of the command circuit are neutral), the hydraulic pump 2 will operate towards reaching the circuit maximum pressure, while the hydraulic battery 9 will be charged at this pressure. This will lead to sending a hydraulic command signal from the battery to the normally closed driven pressure valve, which will open. As a result the hydraulic oil flow from pump 2 will be driven towards the tank.

If the hydraulic consumers of the drilling installation are fed (through the opening of one or more hydraulic feeders) or due to hydraulic losses caused by the command hydraulic elements not being sealed properly, the pressure towards the working circuit is reduced, respectively the pressure of the hydraulic accumulator 9 is reduced.

This leads to the decrease of the command pressure at the driven valve 5 which will lead to its closure. Therefore the connection with the tank is interrupted and the hydraulic pump will supply with under pressure oil the work circuit and the hydraulic battery 9 through the one way valve 8.

Since the driven pressure valves currently manufactured cannot be used by the hydraulic supply system circuit, considering the operation principles there was studied, conceived and designed a new constructive solution for disconnecting valves. The suggested solution leads to the enhancement of the hysteresis, with the possibility to be adjusted together with the adjustment of the pressure value for idle hydraulic pump operation and of the maximum operating pressure for the hydraulic circuit.

3. THE SUGGESTED SOLUTION

The suggested solution leads to the enhancement of the hysteresis (the difference between the opening and the closure pressures), with the possibility to be adjusted together with the adjustment of the pressure value for idle hydraulic pump operation (when it spills into the tank) and of the maximum operating pressure for the hydraulic circuit [7].

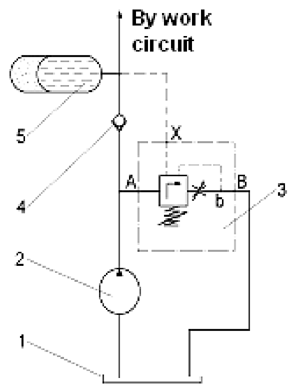

Fig. 2. The simplified hydraulic scheme.

Figure 2 presents the simplified hydraulic scheme, composed of the suggested disconnection driven valve with the role of switching the hydraulic pump operation to low charge when the installation hydraulic consumers are not fed. This figure shows: 1 – tank; 2 – hydraulic pump engaged mechanically by an electric engine; 3 – suggested pressure valve; 4 – one-way valve; 5 – nitrogen hydraulic battery. The design valve 3 has also the role of a safety valve for the hydraulic pump circuit.

Figure 3 presents the constructive solution for the driven disconnection valve, designed to operate in the hydraulic supply system circuit for the drilling installation IMP-2. This solution was conceived to ease the construction with less moving elements, to achieve a higher operation safety.

From the construction point of view the disconnection valve is composed of the steel body 1; screw cap 2; sealing rubber rings 3, 8, 11, 15 and 16; guiding muff 4; piston 5; muff 6; valve piston 7; helical spring 9; spring guiding muff 10; adjusting screw 12; blocking nut 13; adjusting throttle 14. The figure notes: *A* – pressure hole; *B* – back hole; *X* – command hole; I, II, III, IV, V, VI – work rooms; S_1 – adjusting valve; *b* – throttled hole. The operation of the disconnecting valve is based on the pressure adjustment in the back circuit, on the flow section throttling towards *B* using the adjusting throttle 14. The pressure difference created in the chambers II and IV participates to applying an additional force on the conical surface of the valve piston 7, keeping the valve S_1 open for a longer time (increasing the valve hysteresis), even if the pressure within the command circuit decreases.

This is possible due to the difference between the surfaces where pressure is applied, respectively the difference between the conical surface and the cylindrical surface of the valve piston 7. Figure 4 presents the disconnection valve prototype.

4. DIMENSIONAREA SUPAPEI DE DECONACTARE

The sizing calculation for the disconnection valve consists in determining the active passing section “*S*”, the valve hole “*h*” and the sizing of the elastic spring [1, 3].

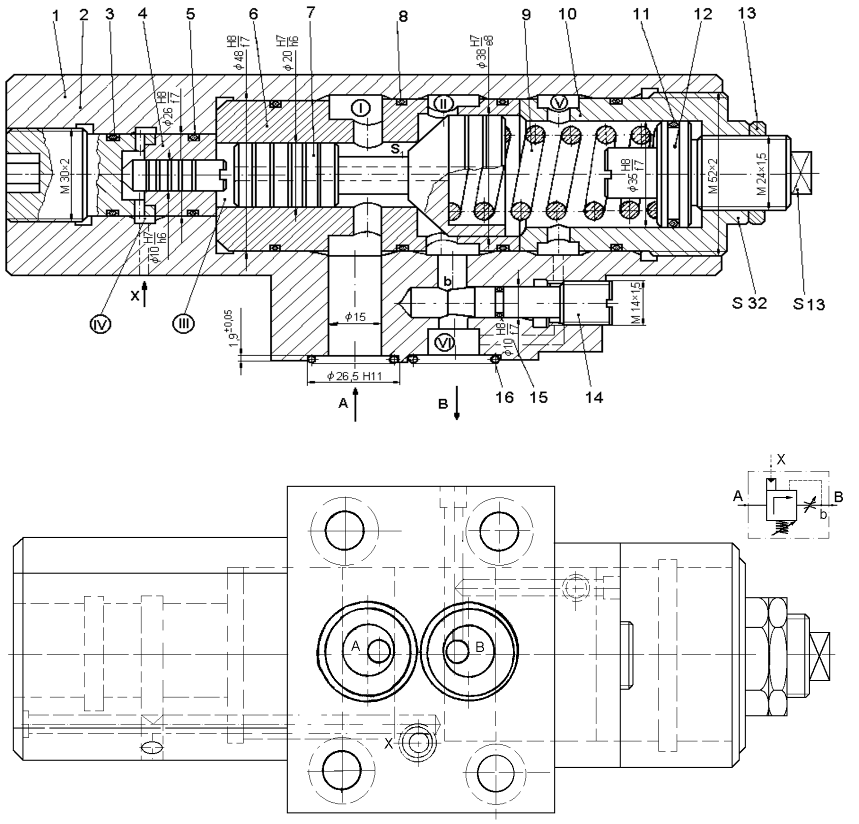

Fig. 3. The disconnection valve suggested proposed.

The active section of passing through the conical valve is given by the side surface of the conic trunk with the base diameters d and d_1 and the generator AB, which complicates the calculus (fig. 5).

To simplify the calculus, we may estimate the side surface of the conic trunk with one of an average diameter cylinder:

$$d_m = \frac{d_1 + d_2}{2} \tag{1}$$

and the height „a" equal to the generator AB:

$$a = h \cdot sin\alpha \tag{2}$$

The valve hole „h" is calculated if the entire flow Q passing through the valve to be drained through its section with the speed v_2 equal to the draining liquid speed through the entry hole v_1.

Fig. 4. The disconnection valve prototype.

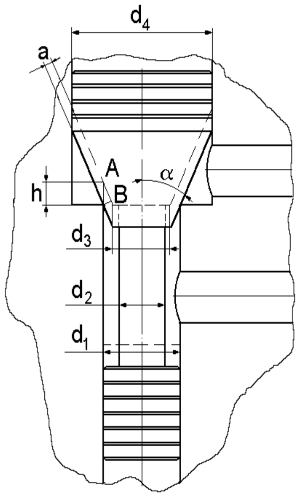

Fig. 5. The valve calculation scheme.

According to the continuity equation [4]:

$$v_1 \cdot S_1 = \pi \cdot d_m \cdot a \cdot v_2 \tag{3}$$

considering $v_1 = v_2$ we get:

$$\frac{\pi(d_1 - d_3)}{4} = \frac{\pi(d_1 + d_2)}{2} h \cdot sin\alpha \tag{4}$$

It results that for the designed valve with the characteristics: d_1=20 mm; d_3=12 mm; α=45°, we get for the hole „*h*" the value h≈5mm.

The surface of the valve transition to the hole „*h*" is determined by the relation:

$$S_h = \pi d_1 \cdot h \sin\alpha \left(1 - \frac{h}{2d_1} \sin 2\alpha\right) \tag{5}$$

For the constructive characteristics of the designed valve we get the maximum transition section $S_{h\,max} \approx 178$ mm^2. The drive valve flow „*h*" of the work element (conic valve) is determined by the relation:

$$Q = C_d \cdot S_h \sqrt{\frac{2\Delta p_1}{\rho}} \tag{6}$$

where: C_d is the valve flow coefficient, C_d=0,63...0,75; ρ – mineral oil density, ρ=900 kg/m^3.

The pressure valve has to open at a command pressure of the drive (piston 5), considered maximum pressure of the hydraulic circuit with the value p_1=p_{max}=20 MPa.

For this there was designed a helical spring of elastic constant of 187,5 N/mm, with the following characteristics: wire diameter: d=6 mm; spring average diameter D_m=24 mm; number of whirls: n=5; propeller step: t=10 mm. The pre-compressed spring is mounted with an arrow of 8,4 mm, corresponding to a force of 1570,8 N, developed by the pressure p_1=20 MPa, considered maximum by the circuit.

Right after the valve opens, due to the liquid flow, it appears a pressure loss Δp_2 (total pressure loss, sum of the pressure losses Δp_1 from the work element of the valve and Δp_{dr} from the adjusting throttle 14) which acts in addition on the conical surface of the valve piston 7, moving it with the drive „*h*", increasing the transition surface until the complete opening of the valve. The drive „*h*" is limited mechanically by the muff cap 10.

The pressure loss Δp_2 may be done and adjusted by the variable throttle 14, which is used to progressively close the transition way of the working liquid towards the tank. The hydraulic pump will flow into the tank as long as the pressure and flow losses due to the improper sealing of the hydraulic command elements do not reach a maximum value, for which the pressure valve is closed. The pressure valve closure is done when the command pressure provided by the hydraulic battery decreases to a determined value (p_2=8,07 MPa).

The adjustment of the valve opening pressure is done through the adjusting throttle screw 12 (M 24×1,5).

Considering the initial position of the screw 12 where the elastic spring 9 is not pre-compressed and the screw step p=1,5 mm, we may determine the value of the valve pressure opening depending on the pre-compressing arrow „*f*" or on the number of rotations „*n*" applied on the adjusting screw.

$$p_1 \cdot S_1 = C_a \cdot f = C_a \cdot p \cdot n \tag{7}$$

$$p_1 = \frac{C_a \cdot f}{S_1}, \quad \text{or} \quad p_1 = \frac{C_a \cdot p \cdot n}{S_1} \tag{8}$$

where n is the number of complete rotations of the adjusting screw.

The adjustment of the valve closing pressure p_1 is done by the adjusting throttle 14, mounted on the tank transition way of the hydraulic pump repression circuit.

The stationary throttle pressure difference depends on the opening „S_{dr}" of its adjusting element:

$$\Delta p_{dr} = \frac{1}{C_d^2} \cdot \frac{\rho}{2} \cdot \frac{Q_{max}^2}{S_{dr}^2} \tag{23}$$

where: $C_d=k\cdot\alpha$ is the hole flow coefficient; k=0,885 for mineral oils; α=0,6...0,7; Q_{max}=18,8 l/min – maximum flow of the hydraulic pump.

The throttle transition section S_{dr} depends on the „n” number of adjusting screw rotations and is determined by:

$$S_{dr}(n)=\frac{\pi d^2}{4}\left[\pi-\arccos\left(\frac{d-2pn}{d}\right)+\frac{2\sqrt{pn(d-p\cdot n)}}{d^2}(d-2pn)\right],\ \text{ptr.}\ \ t<\frac{d}{2} \quad (24)$$

$$S_{dr}(n)=\frac{\pi d^2}{4}\left[\pi-\arccos\left(\frac{2pn-d}{d}\right)+\frac{2\sqrt{pn(d-pn)}}{d^2}(d-2pn)\right],\ \text{ptr.}\ \frac{d}{2}<t<d \quad (25)$$

where: p is the step of the adjusting screw 12, p=1,5 mm; n – number of adjusting screw rotations needed for achieving a certain transition section; d=9 mm is the throttle transition hole diameter.

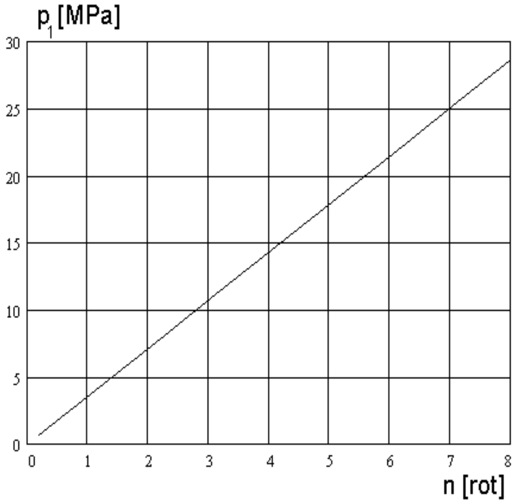

Fig. 6. The caracteristic of the disconect valve.

Figure 6 shows the adjusting pressure for valve opening characteristics, $p=f(n)$. We notice that for obtaining the pressure p_1=20 MPa there is needed a number of n=5,6 rotations.

5. CONCLUSIONS

In conclusion, while the hydraulic consumers of the drilling installation are not fed and the pressure within the hydraulic pump circuit is kept on feeders, the hydraulic pump operates in charge and loads the hydraulic battery up to the maximum pressure (20 MPa). At the moment, the hydraulic battery commands the pressure valve opening which will remain that way and will connect the pump flow into the tank. Meanwhile the pump will operate at pressure of 1,143 MPa (equal to the valve pressure decrease). When the pressure in the valve command circuit decreases to the pressure p_2=8,07, due to the command elements not being sealed properly, the valve will close and the hydraulic pump will flow again into the battery circuit which will be reloaded, the connection – disconnection process repeating cyclically.

Therefore, we avoid an additional electric energy consumption, the overheating of the hydraulic agent worsening its parameters (viscosity), and the hydraulic pump and the safety valve being overused.

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