Modeling operation of sliding valve distribution in hydraulic percussion device with adjustable blow energy

LV Gorodilov and AI Pershin*
Chinakal Institute of Mining, Siberian Branch, Russian Academy of Sciences, Novosibirsk, Russia
E-mail: *pershin.aleksey.ivit22@gmail.com

Abstract. The authors discuss the simulation model of a sliding valve distributor enabling adjustment of capabilities of hydraulic percussion devices. The paper presents the theoretical oscillograms of the valve characteristics and the plots of change in the valve mode versus the valve parameters.

1. Introduction
A promising flow chart of fluid flow control in hydraulic percussion devices is the sliding valve distribution with delayed action piston [1, 2].

In a modification of sliding valve distribution protected by the patent [3] and adjusted later on [4], the piston action is delayed in the beginning of the backward stroke owing pressure difference between the pressure line and the distribution control chamber. The value of the pressure distribution is governed by the parameters of throttle coupling between the pressure and drain lines. Adjustment of orifice sizes helps change the system pressure to reposition the valve and, consequently, to switch to the working cycle of the hydraulic percussion device. Efficiency of the proposed distribution design was experimentally proved on a physical model of a hydraulic percussion device.

This study uses SimulationX [5] to analyze the effect exerted by parameters of the throttles on the switchover time of sliding valve in one (backward) direction. Ratios of the parameters are optimized to ensure the highest-speed response of the valve.

Figure 1. Fluid distribution layout at home position: P—fixed-capacity pump; Ac—hydropneumatic accumulator; SV—sliding valve; S—spring; T1—adjustable throttle; T2—unadjustable throttle; CC—control chamber.
2. Structural layout and mode of sliding valve distribution

Figure 1 shows the structural layout of the sliding valve distribution under analysis. Initially (position I), the control chamber CC is connected with the drain line and the spring S holds the sliding valve SV in the ultimate right position (as per the figure). Power fluid is fed by the pump to the pressure line, and some fluid enters, through the throttle T1, the control chamber CC. The pressure margin in the control chamber is given by:

\[ P_{CC} = \frac{c \cdot x_{preload}}{S_{SV}}, \]

where \( c \) is the spring stiffness; \( x_{preload} \) is the spring preload, \( S_{SV} \) is the valve surface area on the side of the control chamber CC.

When the pressure in the control chamber exceeds the pressure margin above, such that the fluid force on the sliding valve exceeds the spring force, the valve is displaced leftward from position I. The pressure in the control chamber is less than the pressure in the pressure line. As the sliding valve passes a distance \( x_2 \), the control chamber CC directly connects with the pressure line and the throttle T2 is choked. The pressure in the control chamber becomes equal to the pressure in the pressure line, which ensures accelerated displacement of the sliding valve to the ultimate left position (position II in Figure 1) and positive hold of SV in this position in spite of any pressure fluctuations in the system. Homing of the sliding valve occurs after removal of the power fluid pressure and under the action of the spring S.

3. Influence of throttle parameters on switchover time of sliding valve

The sliding valve operation is analyzed using a simulation model constructed in SimulationX intended for the component modeling of complex dynamic systems [6].

The calculation series was carried out at the parameters of the distribution system given in Table 1, for different pairs of throttles T1 and T2 which maintained the delay pressure of 4–10 MPa. Diameter of the throttle T2 was varied from 0.2 to 0.5 mm, and the diameter of the throttle T1 was changed so that to maintain the constant delay pressure \( p_{[d]} \).

| Table 1. Parameters of sliding valve distribution system |
|---------------------------------------------------------|
| Sliding valve                                           |
| Weight, kg                                             | 0.3 |
| Total travel \( x_1 \), mm                            | 6   |
| Travel until direct connection of control chamber and pressure line, \( x_2 \), mm | 0.5 |
| Surface area on the side of control chamber, cm\(^2\)   | 2, 3, 4 |
| Spring                                                 |
| Stiffness, kN/m                                        | 26  |
| Preload, mm                                            | 15  |
| Accumulator                                            |
| Initial gas pressure, MPa                              | 3   |
| Volume, dm\(^3\)                                       | 0.08 |

Figure 2 presents the oscillograms for the sliding valve travels, and pressure in the pressure line and control chamber at the delay pressure \( p_{[d]} \) of 6 MPa.

From Figure 2a, the total travel time \( \Delta t \) of the sliding valve in the backward direction has two intervals \( \Delta t_1 \) and \( \Delta t_2 \). The first interval is the control chamber–pressure line connection via the throttle T1, the second interval is the direct connection between the control chamber and the pressure line. The interval \( \Delta t_2 \) is much shorter than \( \Delta t_1 \); therefore, this travel time of the sliding valve in the first interval is assumed to govern the total changeover time of the sliding valve from position I to position II.
Figure 2. Theoretical oscillograms of (a) valve travels and (b) pressures in pressure line (dashed line) and in control chamber (solid line): $\Delta t$—total switchover time of valve; $p_{[d]}$—delay pressure.

Figure 2b illustrates the change in the pressure in the pressure line and in the control chamber. Initially, the pressure in the control chamber grows much slower than in the pressure line. In both cases, the change in the pressure increment rate is connected with the beginning of fill of the accumulator. In the next interval, the pressure in the control chamber becomes equal to the pressure in the pressure line, and a small drop of pressure is observed in the pressure line at the beginning.

Apparently, reduction in the time $\Delta t_1$ can decrease malfunction probability in operation of the sliding valve, which can improve the valve reliability. For this reason, we focused on this objective.

\[ p_{[d]} = \begin{align*} 
4 \text{ MPa} \\
6 \text{ MPa} \\
8 \text{ MPa} \\
10 \text{ MPa}
\end{align*} \]

Figure 3. Time interval $\Delta t_1$ and throttle T1 diameter versus throttle T2 diameter at different delay pressures and the different sliding valve surface areas on the side of control chamber: (a), (d) 2 cm$^2$; (b), (e) 3 cm$^2$; (c), (f) 4 cm$^2$.

According to Figures 3a–3c, regarding minimization of the time interval $\Delta t_1$, the optimal diameter of the throttle T2 ranges from 0.35 to 0.4 mm, which agrees with the optimal value of T1 diameter from 0.25 to 0.35 mm.
In Figures 3d–3f, the diameter of the throttle T1 linearly changes relative to the diameter of the throttle T2 at the same delay pressure.

The results for the delay pressure of 4 MPa differ from the other data. In this case, the sliding valve starts moving before the pressure in the systems equalizes with the pressure in the accumulator.

Figure 4 presents the time interval $\Delta t_1$ versus the sliding valve surface area on the side of the control chamber at different diameters of the throttles.

![Figure 4](image)

**Figure 4.** Time interval $\Delta t_1$ versus different sliding valve surface areas on the side of control chamber at T2 diameters of (a) 0.2, (b) 0.3, (c) 0.4 and (d) 0.5 mm.

It is seen in Figure 4 that an increase in the surface area of the sliding valve on the side of the control chamber results in the linear increase in the time interval $\Delta t_1$, which can be explained by a decrease in the control chamber pressure $p_{CC}$ to make the valve move, and by the large volume of power fluid required to displace the valve.

At the same time, it is only possible to reduce the valve area to a certain limit as at small areas the spring stiffness is also lower and, accordingly, the forward reposition time of the valve (from position II to position I) enlarges, which weakens efficiency of the valve.

It follows from Figure 4d that at insufficient flow friction of T2 at the drain line, the switchover time of the valve is independent of the valve surface area.

4. Conclusions

1. The simulation modeling has revealed some features in the backward travel of the valve, namely, two intervals of different travel times due to different types of connection between the control chamber and the pressure line: the throttle coupling in the first interval and the direct connection in the second interval.

2. Variation of the throttle diameters has allowed optimizing their values in terms of minimization of switchover time of sliding valve in its backward travel.

3. If the surface area of the sliding valve on the side of the control chamber is decreased, the travel time of the valve linearly decrease. However, in this case, it is necessary to reduce the spring preload, which leads to longer time of the forward travel of the valve and to potential reduction in the valve reliability. Thus, optimization of parameters of a sliding valve should consider its full duty cycle.

Acknowledgments

The study was carried out in the framework of the Basic Research Program, Project Registration No. AAAA-A17-117122090003-2.

References

[1] Arkhipenko AP andd Fedulov AI 1991 *Hydraulic Percussion Machines* Novosibirsk: IGD SO
AN SSSR (in Russian)

[2] Gorodilov LV 2012 Analysis of the dynamics of two-way hydropercussion systems. part i: basic properties *Journal of Mining Science* Vol 48 No 3 pp 487–496

[3] Gorodilov LV, Pashina OA, Tkachuk AK and Kudryavtsev VG RF Patent No 2321777 Distributor for Hydraulic Percussion Machines (Modifications) *Byull. Izobret*. 2008 No 10

[4] Gorodilov LV RF Patent No 2674289 Distributor for Hydraulic Percussion Machine *Byull. Izobret*. 2018 No 34

[5] *Modeling and Simulation of Technical Systems* Available at: http://www.simulationx.com/system-simulation.html (last visited 05.06.2019)

[6] Kolesov YuB and Senichenkov YuB 2012 *Component Technologies of Mathematical Modeling: Educational Aid* Saint Petersburg: Polytechnic University Press (in Russian)