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Study of unsteady operating conditions of a three-port spool flow control valve

The paper focuses on static and dynamic models of operation of a three-port spool flow control valve. The study takes into account dynamics of moving elements and tests of an improved three-port spool flow control valve. The static performance characteristics and spool geometry selection guidelines are provided. Finally, the areas of further research are specified.

Keywords: hydraulics, spool valve, hydraulic control, characteristics of spool devices, flow control valve, steadiness, work simulation.

Introduction

Three-port spool flow control valves (SFCV) are widely used in throttle-controlled hydraulic drives to limit the volume flow rate supplied to the actuators in a wide range of loads. Generally speaking, SFCV are subject to stability and fast response requirements, while the SFCV of hoisting machines are not subject to special fast response requirements, however, the stability of their operation must be ensured.

The control valve under consideration is used in a hydraulic drive of high lifting capacity, which does not have to comply with stringent requirements regarding the response time. During operation of the drive with the installed SFCV, in some cases abnormal noise was observed, caused by oscillations of the SFCV movable elements. The acoustic noise level and pressure fluctuations observed in this case suggest resonance phenomena in the hydraulic system and SFCV.

SFCV is a hydraulic device designed to maintain constant volume flow rate Q_2, supplied to the working member regardless of the load pressure p_2 applied to the member. Structurally, the flow control valve consists of a throttle A_2 and a spool and sleeve installed in parallel. The sensing element of the control valve section is an adjustable-section throttle A_2. The adjustment of its resistance defines the volume flow rate Q_2, supplied through SFCV to the load pressure p_2. The actuating element is a spool, regulating the resistance of the discharge line, through which the excessive volume flow Q_1 is discharged. For the purposes of possible oscillation damping, a hydraulic linear throttle B_2 is foreseen in the SFCV design. The damping properties of the damper B_2 are evidently insufficient to ensure stable SFCV operation. The given paper presents the results of SFCV characteristics analysis and diagnostics of the causes of its unstable operation.

Unstable operation of SFCV is known to be probably caused by the resonance phenomena resulting from:

- erroneously selected geometrical and physical characteristics of SFCV elements (spring stiffness, closure conditions, hydraulic damping, etc.);
- fluctuations of volume flow rate and discharge line pressure caused by pump operation;
- load fluctuations acting upon the hydraulic drive actuator;
- internal disturbing forces (flow turbulence, cavitation, friction conditions in the spool and sleeve, etc.) acting upon the actuating element (spool).

Static performance characteristics of SFCV

Based on the analysis of SFCV design, it was assumed that the possible cause of its unstable operation is unsuccessful choice of geometric parameters of the throttling orifice closure in the spool and sleeve. For the purposes of defining the performance characteristics of SFCV, a static model of its operation has been developed. The analytical diagram of the model is provided in Fig. 1.

In the course of mathematical model development describing the static condition of SFCV, the following assumptions have been accepted:

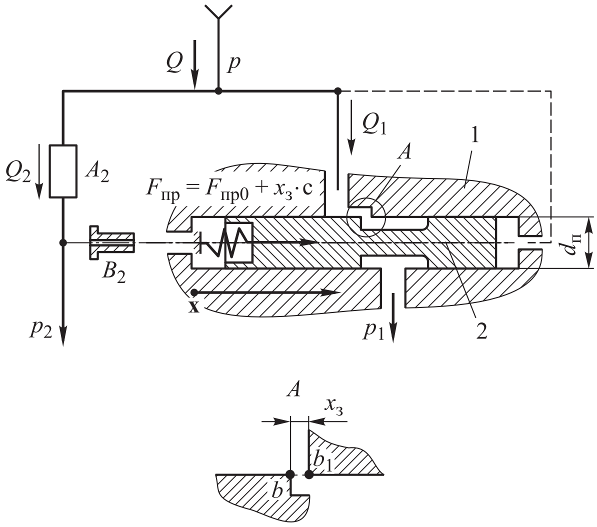

Fig. 1. Analytical diagram of the three-port SFCV static model: 1 – sleeve; 2 – spool; *A* – detail; **x** – coordinate vector

• the operating fluid is incompressible;

• the discharge coefficient of the flow ξ coming through the throttling orifice in the spool and sleeve does not depend on the spool position, $\xi = 2$;

• the friction forces between the spool and sleeve are negligibly small;

• the hydrodynamic force affecting the spool is negligibly small;

• when coming through a throttling orifice, the flow movement is parallel to line $b - b_1$.

Let us present a mathematical model of SFCV operation:

$$\begin{cases} p - p_2 = A_2 Q_2^2; \\ Q = Q_1 + Q_2; \\ p - p_1 = \dfrac{\rho \xi Q_1^2}{2(2\pi \cdot 0{,}5 d_п x_3)^2}; \\ (p - p_2) f_п = F_{пр0} + \mathbf{x}c, \end{cases} \quad (1)$$

where p – SFCV inlet pressure;

p_2 – load pressure;

A_2 – local drag coefficient of the throttle;

Q_2 – volume flow through the throttle;

Q – volume flow pumped to the control valve;

Q_1 – volume flow coming through the throttling orifice in the spool and sleeve (excessive flow);

p_1 – pressure downstream of spool (discharge pressure) at $p = \text{const}$;

ρ – fluid density equal to 860 kg/m^3;

x_3 – spool travel;

$d_п$ – spool piston diameter;

$F_{пр0}$ – preliminary spring contraction at $x_3 = 0$;

c – spring stiffness;

$f_п$ – working area of the spool piston.

The results of the mathematical simulation are given in Fig. 2.

As Fig. 2 shows, the dependence of the spool x_3 travel on the load pressure p_2 is very

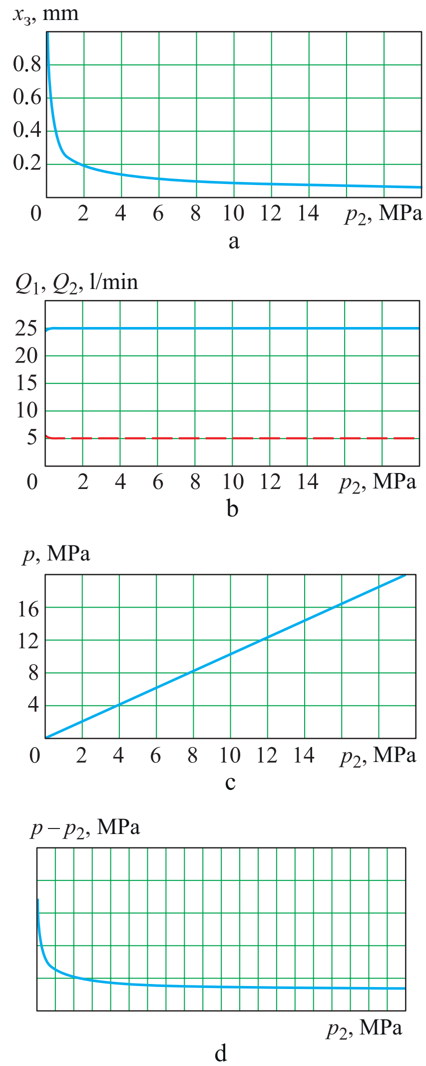

Fig. 2. Static performance characteristics of SFCV: a – dependence of spool position x_3 on load pressure p_2; b – dependence of volume flow rates Q_1 and Q_2 supplied through SFCV on load pressure; c – dependence of SFCV inlet pressure p on load pressure p_2; d – dependence of SFCV pressure differential on load pressure

rigid. For example, if the load pressure varies within the range of $p_2 = 60 \ldots 200$ kgf/cm^2, the operating position of the spool changes within the range of $x_3 = 0.05 \ldots 0.10$ mm. The detected feature initiated the search for other geometrical solutions for the spool operating edge to provide less rigid characteristics of $x_3(p_2)$.

By presenting the equation describing the throttling orifice resistance in the form

$$p - p_1 = \frac{\rho \xi Q_1^2}{2(2\pi R x_3)^2} =$$

$$= \frac{\rho \xi Q_1^2}{2\left(f(x_3)\right)^2} = A_1(x_3) Q_1^2, \qquad (2)$$

SFCV performance characteristics of varying dependence between the throttling orifice area and spool travel $f(x_3)$ can be developed.

Fig. 3 shows some of the considered spool geometry options, and Fig. 4 demonstrates their performance characteristics.

In terms of the performance characteristic rigidity, SFCV with spool No. 3 is the best among the considered options. However, despite the fact that the given results indirectly confirm the assumption on unsuccessful spool geometry acting as a cause of unstable SFCV operation, those do not suffice to unambiguously define the cause of unstable SFCV operation.

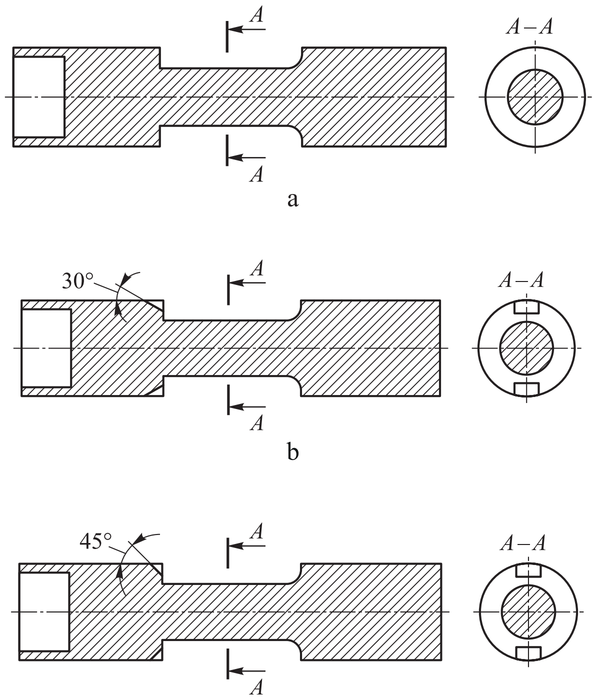

Fig. 3. Several spool geometry options:
a – spool with sharp edge (No. 1); b – spool with two 45° grooves (No. 2); c – spool with two 30° grooves (No. 3)

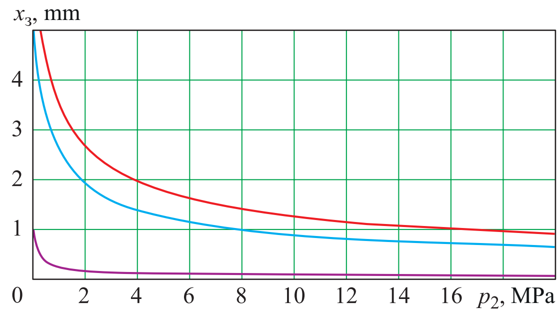

Fig. 4. Dependence of spool x_3 positions on load pressure for:
— – sharp-edged spool; — – spool with two 45° grooves; — – spool with two 30° grooves

Dynamic model of SFCV operation

In order to determine the conditions of unstable SFCV operation, a dynamic mathematical model of a three-port SFCV was developed as shown in Fig. 5.

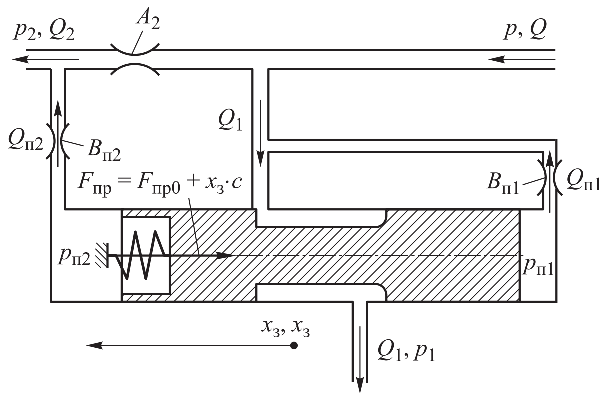

Fig. 5. Analytical diagram of the three-port SFCV dynamic model

The following factors have been considered in developing the dynamic mathematical model:

• compressibility of the operating fluid in spool piston underside cavities, loading line and discharge line;

• movement of the operating fluid from piston underside cavities, caused by spool displacement.

At the same time, the following factors were not considered due to their negligibility:

- the hydrodynamic forces acting on the spool;
- the attached masses of the operating fluid in spool piston underside cavities;
- the friction forces between the spool and sleeve.

The dynamic mathematical model of SFCV is a system of differential algebraic equations:

$$\begin{aligned}
&m\ddot{x} = -F_{\text{пр}0} - xc - p_{\text{п}2} f_3 + p_{\text{п}1} f_3;\\
&\dot{p}_{\text{п}1} = E\frac{-\dot{x}_3 f_3 - Q_{\text{п}1}}{V_{0\text{п}1} + x_3 f_3};\\
&\dot{p}_{\text{п}2} = E\frac{\dot{x}_3 f_3 - Q_{\text{п}2}}{V_{0\text{п}2} - x_3 f_3};\\
&\dot{p} = E\frac{Q - Q_2 - Q_1 + Q_{\text{п}1}}{V};\\
&\dot{p}_2 = E\frac{Q_{\text{п}2}}{V_2};\\
&p = f(t);\\
&Q = f(t);\\
&Q_2 = \sqrt{\frac{p - p_2}{A_2}};\\
&Q_1 = \sqrt{\frac{p - p_1}{A_1(x_3)}};\\
&Q_{\text{п}2} = \frac{p_{\text{п}2} - p_2}{B_{\text{п}2}};\\
&Q_{\text{п}1} = \frac{p_{\text{п}1} - p}{B_{\text{п}1}},
\end{aligned} \tag{3}$$

where m, $\ddot{x}$, $\dot{x}_3$ – mass, acceleration and speed of the spool, respectively;

$p_{\text{п}1}$, $p_{\text{п}2}$ – pressure in spool piston underside cavities;

$\dot{p}_{\text{п}1}$, $\dot{p}_{\text{п}2}$ – rate of piston underside pressure change;

E – volume modulus of operating fluid elasticity;

$Q_{\text{п}1}$, $Q_{\text{п}2}$ – volume flow from piston underside cavities caused by spool motion;

$V_{0\text{п}1}$, $V_{0\text{п}2}$ – design volume of piston underside cavities;

V – design volume of discharge line;

V_2 – design volume of lines and actuating members;

$A_1(x_3)$ – throttling orifice resistance in the spool and sleeve as defined by spool travel x_3 (see equation (2));

$B_{\text{п}1}$, $B_{\text{п}2}$ – drag coefficients of linear damping throttles. In the original SFCV design, the linear damping throttle $B_{\text{п}2}$ is not foreseen; however, it was introduced into the mathematical model in order to evaluate the impact of potential installation thereof.

The system of differential algebraic equations (3) was solved by sequential solution of their differential part by the Runge – Kutta method of the fourth order and by direct solution of the algebraic part.

In the course of testing, it was indirectly determined that the spool oscillation frequency in SFCV is $\omega_Q = 40\ldots60$ Hz. The oscillations at these frequencies can be related to the fluctuating incoming volume flow Q caused by hydraulic pump operation.

According to the simulation results, it was established that when the system is affected by a flow rate of $Q = 30$ l/min at the frequency ω_Q, equal to 40, 50 and 60 Hz and at an amplitude of $A_Q = 5$ l/min, SFCV does not take on the resonance mode and no spool oscillation is observed. Simulation results for the fluctuating incoming flow Q at a frequency of $\omega_Q = 50$ Hz are given in Fig. 6.

Thus, the simulation of SFCV operation using the developed dynamic model did not indicate unstable operation at the fluctuating incoming flow with the parameters determined in the course of test performance. When using the proposed approach, the search for the values of incoming impact leading to unstable SFCV operation is reduced to enumeration or random search, therefore, the direct simulation of SFCV operation for the purposes of finding the areas of its stability had to be rejected.

Conclusion

The results of defining the static performance characteristics and simulation of SFCV operation

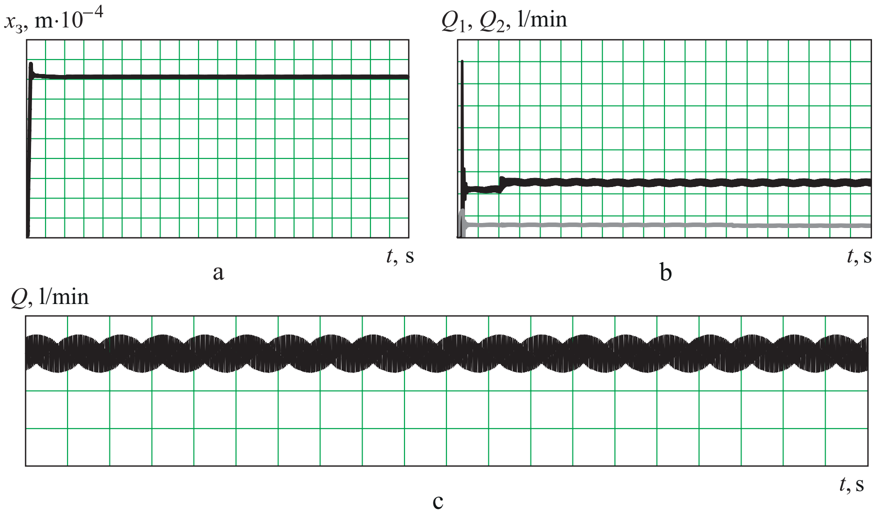

Fig. 6. Simulation results at incoming flow fluctuations: $Q = 30$ l/min at a frequency of $\omega_Q = 50$ Hz and an amplitude of $A_Q = 5$ l/min: a – dependence of spool travel on time *t*; b – dependence of flow rates through the throttling orifice of the spool and sleeve Q_1 and through throttle Q_2 on time; c – dependence of flow rate *Q*, supplied to SFCV, on time *t*

at the fluctuating incoming flow suggest the following statements.

1. The available spool geometry with the cylindrical shape of the working end face ensuring a throttling orifice in the spool and sleeve leads to very rigid performance characteristic with the spool working position ranging between 0.05...0.10 mm.

2. The spool with two grooves executed at an angle of 45° should have better characteristics as compared with the existing ones, thus ensuring the spool working position within the range of 0.70...1.25 mm.

3. The developed dynamic model does not indicate the presence of unstable SFCV operation at the fluctuating incoming flow within the frequency range of 40...50 Hz.

Based on the conducted research, SFCV design has been changed, in particular, two grooves have been introduced at the end face of the spool at an angle of 45°, the damping properties of the damper $B_{п2}$ have been reinforced and the damper $B_{п1}$ has been introduced. The experiment proved that all these modifications, taken together, ensured stable SFCV operation. However, the effect of changes in load pressure p_2 on the regulated volume flow rate Q_2 has noticeably increased, which may indicate excessive damping.

The provided simulation results do not allow us to reliably determine the cause of unstable SFCV operation. There are two methods suggested for troubleshooting the causes and researching the factors impacting the stability of SFCV operation and the ranges of its stability:

1) improving the dynamic model of SFCV operation by introducing a model of hydrodynamic forces acting on the spool and arising due to the operating fluid discharge through the throttling orifice in the spool and sleeve;

2) linear approximation, development of the amplitude-frequency and phase-frequency characteristics and search for zones of stability on the proposed dynamic model of SFCV.

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Исследование условий неустойчивой работы трехлинейного золотникового регулятора расхода

Рассмотрены статическая и динамическая модели работы трехлинейного золотникового регулятора расхода с учетом динамики подвижных элементов и испытаний усовершенствованного трехлинейного золотникового регулятора расхода. Приведены статические рабочие характеристики и рекомендации по выбору геометрии золотника. Сформулированы направления для дальнейшего исследования.

Ключевые слова: гидравлика, золотниковые регуляторы, гидравлические регуляторы, характеристики золотниковых устройств, регулятор расхода, устойчивость, моделирование работы.

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