E3S Web of Conferences **252**, 02030 (2021)
PGSGE 2021 https://doi.org/10.1051/e3sconf/202125202030

Influence and Improvement of Hydraulic Power on Spool Valve Reversing

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Abstract. This paper analysed the effect of steady-state flow force and transient flow force to sliding direction valve, and two examples were given to illustrate adverse consequences caused by excessive fluid power, put forward the compensation measures. The effect of flow force should be considered when designing the hydraulic system in order to make the hydraulic system work more stable.

1 Introduction

Among the three types of directional valves, i.e. slide valve, rotary valve and ball valve, slide valve is widely used in hydraulic system. It uses the slide of the valve core in the valve body to change the size of the opening of the slide valve, so as to control the pressure and flow of the liquid in the hydraulic system. [1]

When the liquid flows through the valve port of the slide valve, the direction and velocity will change. According to the momentum theorem, the change of velocity causes the change of momentum, and the change of liquid momentum makes the valve core subject to reaction force, that is, hydrodynamic force. The hydrodynamic force is divided into steady-state hydrodynamic force and transient hydrodynamic force. [2] If the steady-state hydrodynamic force is too large, it may lead to difficult or impossible reversing of the slide valve. If the transient hydrodynamic force is too high, it will produce hydraulic impact, damage the equipment and affect the normal production. This paper analyzes the influence of steady and transient hydrodynamic force on the slide valve, and puts forward the compensation measures.

2 Influence of steady state hydrodynamic force on spool valve reversing

When the opening of the valve port remains unchanged, the force acting on the valve core due to the change of momentum of the liquid flowing through the valve port is called steady-state hydrodynamic force. The magnitude of steady-state hydrodynamic force can be analyzed and calculated according to the momentum theorem in liquid dynamics.

Figure 1 shows two cases of liquid flowing out and flowing into the valve port. The radial component of steady-state hydrodynamic force is usually offset by the symmetrical design of valve chamber, and its axial component is expressed by formula (1), and the direction is to make the valve port close.

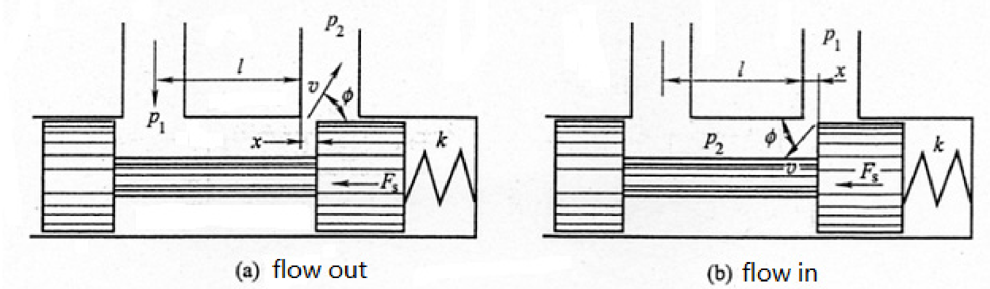

Fig. 1 steady state hydrodynamic force of slide valve

$$F_S = \rho q v \cos\phi = (2C_d C_v W \cos\phi) x \Delta p = K_s x \Delta p \qquad (1)$$

Where, C_d ——Liquid flow coefficient;

C_v ——Liquid velocity coefficient;

W ——Flow area gradient of spool valve port;

ϕ ——Jet angle of liquid flow;

x ——Opening of slide valve port;

Δp ——Pressure difference before and after the flow

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through the valve port, $\Delta p = p_1 - p_2$;

K_s ——Hydrodynamic coefficient, $K_s = 2C_d C_v W \cos\phi$.

It can be seen from equation (1) that when the pressure difference Δp between the front and rear of the valve port is constant, the steady-state hydrodynamic force F_s is directly proportional to the opening x of the valve port. When the opening x of the valve port remains unchanged, the steady-state hydrodynamic force F_s is directly proportional to the valve port pressure difference Δp. The direction of steady-state hydraulic force is consistent with the direction of restoring force, which makes the valve port tend to close. At this time, the steady-state hydrodynamic force is beneficial to the working performance of the slide valve. The disadvantageous side is: the steady-state hydraulic power increases the force required to push the slide valve when the system is reversing. In the high-pressure and large flow hydraulic system, the force will be very large, and the hydraulic force will be greater than the control force, which will cause the valve core action failure.

In the reversing circuit of the three hydraulic cylinders shown in Fig. 2 (a), the quantitative pump supplies oil, and the pressure oil supplies oil to the three hydraulic cylinders through three position reversing valves. Sometimes, the reversing valve fails to work in actual operation. After testing, all parts of the electromagnetic directional valve work normally, and the regulating pressure of the relief valve is lower than the allowable working pressure of the electromagnetic directional valve. Sometimes two or three hydraulic cylinders act at the same time, sometimes only one action. Since the output flow of the quantitative pump needs to meet the simultaneous action of three cylinders, the flow rate is relatively large. When only one hydraulic cylinder acts, the output flow of the quantitative pump passes through a solenoid valve, and the flow through the solenoid valve greatly exceeds the rated flow of the solenoid valve.

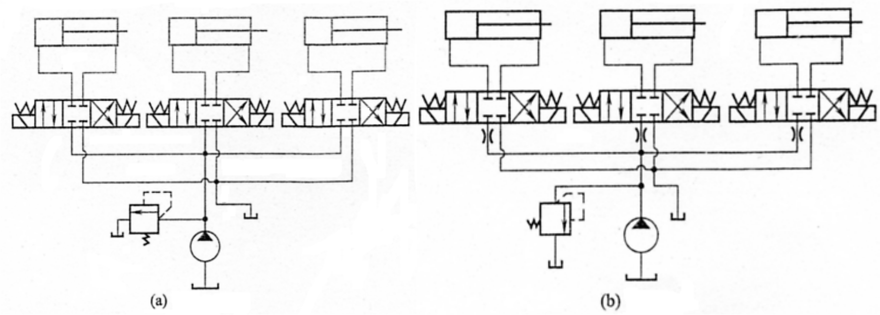

Fig. 2 three cylinder reversing circuit

According to the formula $q = C_d A \sqrt{\frac{2\Delta p}{\rho}}$ of valve port flow, when the valve port area is constant, the increase of flow leads to the increase of pressure difference. According to formula (1), the steady-state hydrodynamic force increases with the increase of pressure difference when the opening of valve port is unchanged. Therefore, in the circuit shown in Fig. 2 (a), when all the flow flows through a directional valve, the steady-state hydrodynamic force of the directional valve is greatly increased, and the electromagnet thrust cannot push the valve core to change direction, resulting in commutation failure. The improved circuit is shown in Fig. 2 (b). When only one hydraulic cylinder is working, part of the pump output flow is regulated by the throttle valve into the hydraulic cylinder, and the other part is returned to the oil tank through the overflow valve. In this way, the flow through the solenoid valve is controlled, thus avoiding the reversing failure.

In order to reduce the adverse effect of steady-state hydrodynamic force on spool valve commutation, the following measures are often taken to compensate the steady-state hydrodynamic force.[3]

(1) In view of the large flow through the slide valve, two-stage or multi-stage control mode is often used. For example, the electro-hydraulic slide valve is used to replace the electromagnetic slide valve; the manual pressure reducing valve type pilot valve is used to replace the manual slide valve for pilot control, and the oil source is controlled to push the spool valve to change direction.

(2) Valve sleeve movement method. In this method, the spool does not move and the valve sleeve moves relative to the spool to realize the reversing of the slide valve. In this way, part of the hydraulic force on the valve core is transferred to the valve sleeve, and the movement resistance of the valve core is reduced correspondingly.

(3) Special valve chamber method. This method changes the flow state of liquid in the valve chamber by changing the design of valve chamber. That is to change the angle of outflow and inflow into the valve chamber to make them nearly equal, so as to produce a force opposite to the steady-state hydrodynamic force. This method is used in FMV servo valve developed by Hitachi. The disadvantage of this method is that the shape of valve core and valve sleeve is complex and the machining is difficult. The valve chamber structure is shown in Fig. 3.[4]

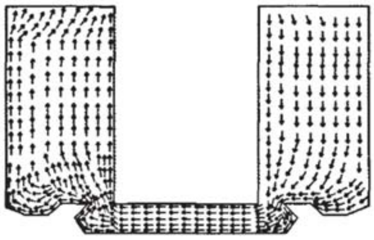

Fig. 3 slide valve with special chamber shape

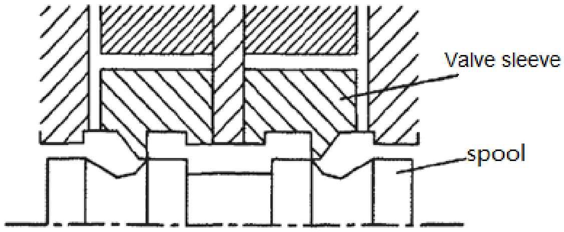

Fig. 4 improvement of flow passage in valve

(4) The method of channel transformation. In this method, the flow channel in the valve is reformed reasonably to change the flow state of the fluid in the valve and reduce the steady-state hydrodynamic force. The structure of flow passage in the valve is shown in Fig. 4.

(5) "U" notch. Different from the conventional full circle opening slide valve, it processes several "U" shaped grooves on the valve core. The oil flows in the "U" groove first, and then enters the valve chamber. The effect of the notch reduces the injection angle of the oil, thus reducing the hydrodynamic force. This method is applied to servo valve produced by Hitachi motor company.

3 Influence of transient hydrodynamic force on spool valve reversing

The transient hydrodynamic force is the force acting on the valve core by accelerating or decelerating the liquid flow in the valve chamber when the valve opening changes during the movement of the slide valve. Figure 5 shows two cases of liquid flowing out and flowing into the valve port. The transient hydrodynamic force is expressed by equation (2).

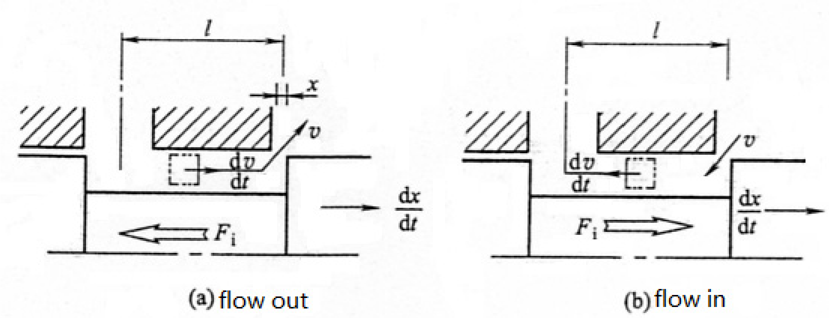

Fig. 5 transient hydrodynamic force of spool valve

$$F_i = -\rho l \frac{dq}{dt} = -\rho l C_d W \sqrt{\frac{2}{\rho}\Delta p}\frac{dx}{dt} = -C_d W l\sqrt{2\rho\Delta p}\frac{dx}{dt} = K_1\frac{dx}{dt} \quad (2)$$

Where, l ——Length of valve chamber;

K_1 ——Damping coefficient, $K_1 = -C_d W l\sqrt{2\rho\Delta p}$;

It can be seen from equation (2) that the transient hydrodynamic force F_i is directly proportional to the moving speed ((dx/dt)) of the slide valve, which acts as a viscous damping force. The damping coefficient K_1 is related to the length l of the valve chamber and the pressure difference between the front and the back of the valve port. When the liquid flows out of the valve port, as shown in Fig. 5 (a), the transient hydrodynamic force direction is opposite to the moving direction of the valve core, which prevents the valve core from moving and increases the resistance of the valve core movement. When the liquid flows in from the valve port, as shown in Fig. 5 (b), the direction of transient hydrodynamic force is the same as the moving direction of the spool. This force promotes the movement of the spool, which is one of the reasons for the instability of the slide valve.

In the design and application of the directional valve, only the influence of the moving speed (dx/dt) of the slide valve is often considered. Because the speed of the spool is far less than the flow acceleration, the transient hydrodynamic force is small and often ignored. The influence of pressure difference on transient hydrodynamic force is not considered. In fact, in the high-pressure and large flow hydraulic system with electromagnetic directional valve, the pressure difference of directional valve is large when it works, and the influence on transient hydrodynamic force cannot be ignored.

In the hydraulic system of the machine tool shown in Fig. 6 (a), the load is unloaded by the three position four-way electromagnetic directional valve, and the middle function of the directional valve is M-type. The circuit is a high pressure and large flow system. When the pump outlet pressure is switched from high pressure to zero pressure or from zero pressure to high pressure, the pressure difference between the front and back of the valve port of the reversing valve increases rapidly, resulting in the rapid rise of transient hydraulic power. At the same time, due to the rapid switching of solenoid valve, the value of speed (dx/dt) is also large, and the hydraulic impact is intensified. Even small hydraulic impact will reduce the processing quality of parts, so hydraulic impact is not allowed in the hydraulic system of machine tool.

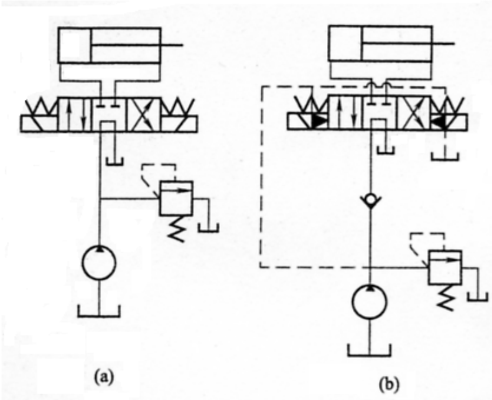

Fig. 6 unloading circuit of three position four way solenoid valve

As shown in Fig. 6 (b), the obvious pressure impact can be avoided by using electro-hydraulic directional valve instead of electromagnetic directional valve. This is because the reversing time of the hydraulic valve in the electro-hydraulic directional valve is adjustable. By extending the reversing time, the change process of the pump outlet pressure rising or falling can be slowed down, thus reducing the transient hydraulic power and improving the reversing stability.[5]

The following methods can be adopted to reduce the pressure impact and improve the stability of the spool valve reversing.[6]

(1) In the hydraulic circuit, the diameter of pipe can be increased, the velocity of liquid flow can be limited, and the rubber hose with smaller elastic coefficient can be used.

(2) Using DC solenoid valve instead of AC solenoid valve can reduce hydraulic impact. If the solenoid valve is replaced by hydraulic directional valve or electro-hydraulic directional valve, the reversing time of valve core will be prolonged, and the impact will be obviously reduced or eliminated.

(3) The structure of the control side of the oil inlet and return chambers of the valve core is improved, so that the movement state of the liquid flow is gradually changed during the reversing of the directional valve. The control edge of the valve core can be made into a cone angle. The cone angle is generally taken. The cone length depends on the length of the sealing edge, or an axial triangular groove is opened on the valve core.

(3) By improving the structure of the control side of the oil inlet and return chambers of the valve core, the movement state of the liquid flow can be changed gradually when the valve is reversing. The control edge of the valve core can be made into a cone angle. The cone length depends on the length of the sealing edge, or an axial triangular groove is opened on the valve core.

(4) An accumulator is set in front of the directional valve to absorb the shock pressure, which can shorten the propagation distance of the pressure shock wave.

(5) Sensitive small relief valves are set at the inlet and outlet of the hydraulic cylinder to limit the pressure impact during reversing.

4 Conclusions

Through the above analysis, the existence of steady-state hydraulic power increases the difficulty of spool valve reversing. The fast reversing of slide valve and the rapid change of system pressure produce transient hydrodynamic force, which causes hydraulic impact and affects the normal operation of hydraulic reversing system. Therefore, in the design and operation of the hydraulic reversing system, the influence of steady-state and transient hydraulic power should be comprehensively considered according to the field conditions, and appropriate measures should be taken to eliminate the adverse effects of hydraulic power.

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