FORCE CONTROL DETERMINATION OF A SPOOL VALVE WITH CRITICAL CENTER

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Abstract: *The control elements of the hydraulic parameters, flow and pressure, are complex systems in closed loop, characterized by fluid flow phenomena and automatic control processes. Due to the complexity of these phenomena, the optimal solutions for designing and realization are determined iteratively. An important problem to be solved in order to design/realize/optimize a hydraulic valve is to determine the force required slide valve movement. Hydrodynamic forces that arise due to the nature of the working fluid flowing through the shoulder section realized between the drawer and bush edges tend to close distribution. Achieving the required performances invole the use of mathematical modeling procedures and numerical simulatiion based on realistic models obtained by calibration of the systems of equations on the basis of experimental data.*

INTRODUCTION

Directional control valves (DCV) are one of the most fundamental parts in hydraulic machinery as well and pneumatic machinery. They allow fluid flow into different paths from one or more sources. They usually consist of a spool inside a cylinder which is mechanically or electrically controlled. The movement of the spool restricts or permits the flow, thus it controls the fluid flow.
The spool (sliding type) consists of lands and grooves.The lands block oil flow through the valve body. The grooves allow oil or gas to flow around the spool and through the valve body. There are two fundamental positions of directional control valve namely normal position where valve returns on removal of actuating force and other is working position which is position of a valve when actuating force is applied. There is another class of valves with 3 or more position that can be spring centered with 2 working position and a normal position.
Solenoid operated DCV are widely used in the hydraulics industry. These valves make use of electromechanical solenoids for sliding of the spool. Because simple application of electrical power provides control, these valves are used extensively. However, electrical solenoids cannot generate large forces unless supplied with large amounts of electrical power. Heat generation poses a threat to extended use of these valves when energized over time. Many have a limited duty cycle. This makes their direct acting use commonly limited to low actuating forces. Often a low power solenoid valve is used to operate a small hydraulic valve (called the pilot) that starts a flow of fluid that drives a larger hydraulic valve that requires more force. A bi-stable pneumatic valve is typically a pilot valve that is a 3 ported 2 position detented valve. The valve retains its position during loss of power, hence the bi-stable name. Bi-stability can be accomplished with a mechanical detent and 2 opposing solenoids or a "magna-latch" magnetic latch with a polarity sensitive coil. Positive opens and negative closes or vice -versa. The coil is held in position magnetically when actuated. Spool is of two types namely sliding and rotary. Sliding spool is cylindrical in cross section, and the lands and grooves are also cylindrical.
Directional control valves are generally specified using the number of ports and the number of switching positions. It can be represented in general form as n_p/n_s, where n_p is the number of ports connected to the direction control valve and n_s the number of switching positions. In addition, the method of actuation and the return method can also be specified. A hypothetical valve could be specified as 4-way, 3-position direction control valve or 4/3 DCV since there are four ports and three

switching positions for the valve. In this example, one port is called the pressure port which is connected to the pump; one port is the tank port and is connected to the tank (or reservoir); and the two remaining ports are called working ports and are connected to the actuator. Apart from characteristics of valve the fluid suitable for valve,working temperature and viscosity also thought upon before selecting a particular type of valve. The most expensive valve available is one in which the groove width andthe land width are matched, so leakage flow and deadband at the centered position are minimized. Such valves are called critical center valves. Because critical center valves are expensive, they are only used for specialized applications such a servo control of position or velocity. Such an application would be control of flight control surfaces in an airplane.

Because by changing a mechanical parameter can be obtained the variation of hydraulic parameter, DCV can be considered as mecano-hydraulics converters.

The use of an electro-mechanical converter stage (proportional electromagnet) together with a mecano hydraulic converter (DCV) allows a complex subsystem (electro-hydraulic amplifier) that performs processing of the signal of electric nature into a hydraulic nature [1]. These systems are called amplifiers because the ratio between hydraulic power and electric power is much higher than 1. The direct command of the spool can be done with proportional force or position electromagnets.

The proportional electromagnets (fig. 1) ensure proportionality between the control current and force developed axial in plunjer. The force provided by the electromagnet is proportional to the intensity of the control current and is affected by hysteresis. For small displacement the force from the plunjer rod is independent of its position.
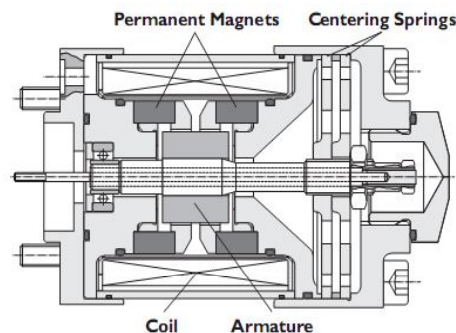

Figure 1. Moog linear force motor

The proportional displacement electromagnet is composed from an proportional current electromagnet, an inductive position transducer and an servocontroler. Axial force of plunjer depends on its position.

Reducing overall dimensions, electric power consumed and thus the costs of the production and use of proportional electromagnets need to optimize them according to the axial forces that need to be developed for the sliding spools.

THE EQUILIBRIUM EQUATIONS OF DYNAMIC REGIME

The axial forces which act upon the sliding spool of DCV are: inertial forces; friction forces; elastic forces of the compression springs and hydrodynamic forces due to flow of working fluid through the holes of variable area.

The most important component of this forces is the hydrodynamic force (figure 2)

$$F_j = \rho \cdot V \frac{Q_2^2}{A_2 \cdot V} = \frac{\rho \cdot Q_2^2}{A_2} = \frac{\rho \cdot Q_2^2}{C_c \cdot A_0} \tag{1}$$

It decomposes in the axial and radial components

$$F_1 = -F_j \cdot \cos\theta \tag{2}$$

$$F_2 = -F_j \cdot \sin\theta \tag{3}$$

Neglecting the fluid compressibility in DCV and taking into account the conservation of flow can be write

$$Q_1 = Q_2 = C_d \cdot A_0 \sqrt{\frac{2}{\rho}(p_1 - p_2)} \tag{4}$$

where C_d is orifice discharge coefficient.

Thus the axial force required to change the momentum of the fluid passing through the valve orifice for steady state regime is

$$F_1 = 2 \cdot C_d \cdot C_v \cdot A_0 \cdot (p_1 - p_2)\cos\theta \tag{5}$$

The open area of the orifice A_0 is proportional to displacement so

$$F_1 = 2 \cdot C_d \cdot C_v \cdot w \cdot (\Delta p)\sqrt{x_v^2 + C_r^2} \cos\theta \tag{6}$$

where w is called the area gradient, x_v is valve opening and C_v is orifice speed coefficient.
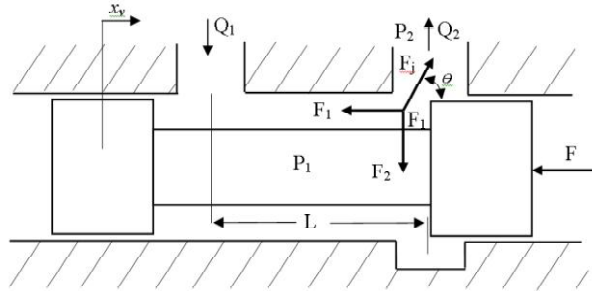

Figure 2. Flow forces on a spool valve

For non-steady state regime, the inertial force of the fluid mass M between input and output port is

$$F_3 = M \cdot a = \rho \cdot L \cdot A_v \frac{d\left(\dfrac{Q_1}{A_v}\right)}{dt} = \rho \cdot L \frac{dQ_1}{dt} \tag{7}$$

or

$$F_3 = L \cdot C_d \cdot w\sqrt{2 \cdot \rho \cdot (p_1 - p_2)}\frac{dx_v}{dt} + \frac{L \cdot C_d \cdot w \cdot x_v}{\sqrt{\left(\dfrac{2}{\rho}\right)(p_1 - p_2)}}\frac{d(p_1 - p_2)}{dt} \tag{8}$$

For non-steady state regime, force due to fluid flow depends on the speed of sliding spool and the variation of pressure loss.
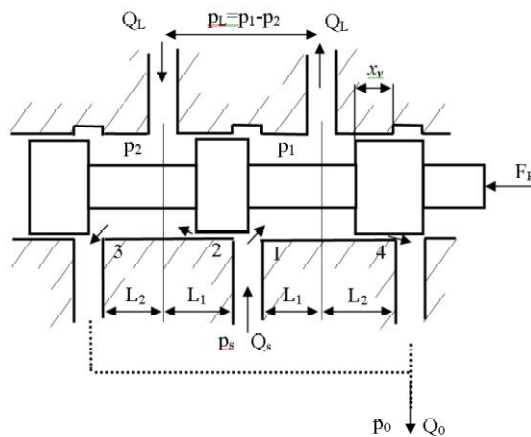

Figure 3. Three-land-four-way spool valve

For a three-land-four-way spool valve (figure 3), the axial force necessary to move the spool is

$$F_R = 2 \cdot C_d \cdot C_v \cdot (\cos\theta) \cdot w \cdot x_v \cdot (p_s - p_1) - L_1 \cdot \rho\frac{dQ_1}{dt} + 2 \cdot C_d \cdot C_v \cdot (\cos\theta) \cdot w \cdot x_v \cdot p_2 + L_2 \cdot \rho\frac{dQ_3}{dt} \tag{9}$$

$$p_1 = \frac{p_s + p_L}{2} \tag{10}$$

$$p_2 = \frac{p_s - p_L}{2} \tag{11}$$

where p_s is pressure at the supply, Q_s is flow to the valve from the supply and p_L is pressure at the load.

$$Q_1 = C_d \cdot A_1 \cdot \sqrt{\frac{2}{\rho}(p_s - p_1)} \tag{12}$$

$$Q_3 = C_d \cdot A_3 \cdot \sqrt{\frac{2}{\rho} p_2} \tag{13}$$

$$F_R = 2 \cdot C_d \cdot C_v \cdot w \cdot (\cos\theta) \cdot (p_s - p_1) \cdot x_v + (L_2 - L_1) \cdot C_d \cdot w \cdot \sqrt{\rho \cdot (p_s - p_L)} \cdot \frac{dx_v}{dt} \tag{14}$$

TEST SYSTEM

The device for determining the command force of DCV is shown in figure 4. To measure the command force, the device was equipped with a manually operated system precisely, micrometric screw. The micrometric device move the command rod through a force transducer. The position of the rod is measured with an inductive position transducer. Hydraulic load was simulated by changing the flow of the variable hydraulic restrictor located between connections A and B of the DCV. The device is supplied with oil under pressure from a group generator. Oil flow is measured using flow meter located on the P connection. Hydraulic pressures are measured with the aid of gauges located on the connections A and B of the valve. All sensors are connected via an data acquisition board to a PXI NI system.
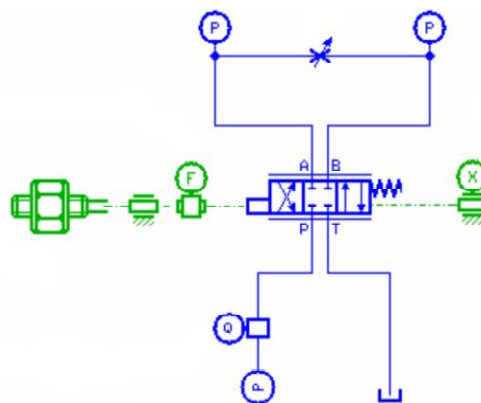

Figure 4. The operating principle scheme of the device for measuring axial forces

CONCLUSIONS

The complexity of the physical phenomena not usually allows the development of "perfect" mathematical models. Due to this aspect, verification and validation of mathematical models based on experimental data is necessary in any study. Achieving the required performances invole the use of mathematical modeling procedures and numerical simulatiion based on realistic models obtained by calibration of the systems of equations on the basis of experimental data.

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