DETERMINATION OF THE ACTUATION FORCE OF SPOOL VALVES WITH ZERO COVERAGE

(1st Report, Theoretical Analysis)

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Abstract:
The adjustment elements of hydraulic parameters flow and pressure are complex closed loop systems, which involve phenomena associated with fluid flow, electromechanical phenomena, as well as phenomena specific to automatic adjustment. Due to complexity of such phenomena, determination of optimal solutions to their design and implementation is iterative. A key problem to be solved in order to design / develop / optimize a hydraulic directional control valve is to determine the force necessary for displacement of slide valve. Hydrodynamic forces that arise due to the flow of the working fluid through the section delimited by the shoulders of the spool valve and the edges of the sleeve tend to stop distribution. Meeting the performance requirements involves the use of modeling processes and numerical simulation based on realistic models, obtained by calibration of equations systems based on experimental data.

Keywords: proportional directional valve, axial flow forces, solenoid actuator

1. Introduction

Directional control valves are hydraulic components used to make different connections and adjust the flow along the circuits created between joints. The most widespread directional control valves consist of a cylindrical slide with translational motion and a fixed body provided with toroidal inner channels. Between the shoulders of the slide valve and the toroidal chambers several variable throttles are placed simultaneously which regulate the flow. The less complicated directional control valve has two positions and two joints, actually being a throttle used to control flow or for interruption of hydraulic circuits. The most widespread directional control valves have four joints and three positions. In terms of the scheme of connections, directional control valves can have closed, open or partially open centre.

Since the change of a mechanical parameter causes variation of a hydraulic parameter, directional control valves can be considered mechano- hydraulic converters. Using an electromechanical conversion stage (proportional magnet) with a mechano-hydraulic converter (directional control valve) allows creation of a complex subsystem (electrohydraulic amplifier) which performs conversion of signal from electrical signal into a hydraulic one [1]. These are referred to as amplifiers because the ratio of hydraulic controlled power and electrical control power is greater than 1 (usually 103... 106). Direct command of slide valves can be performed by means of proportional force or position electromagnets.
Proportional force electromagnets (Fig. 1) ensure proportionality between the coil current and the force axially developed inside plunger. Force provided by the electromagnet is proportional to intensity of control current and is affected by hysteresis.

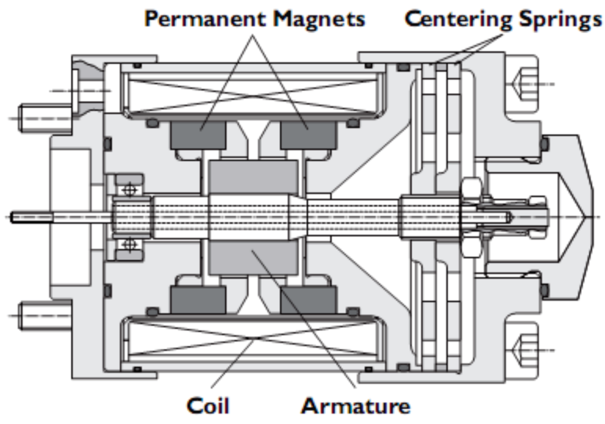

Fig.1. Moog linear force motor [4]

For small strokes force inside the rod of plunger is independent of its position (Fig. 2). Bearings of plunger are usually made of sintered and raylon bronze and are immersed in oil.

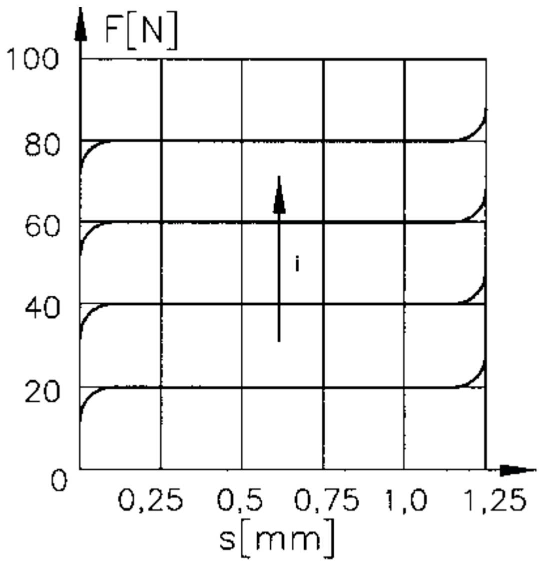

Fig.2. Stationary characteristic of a proportional force electromagnet

Proportional stroke electromagnets (Fig. 3) are made of a block comprising a proportional force electromagnet, an inductive position transducer and a servo controller.

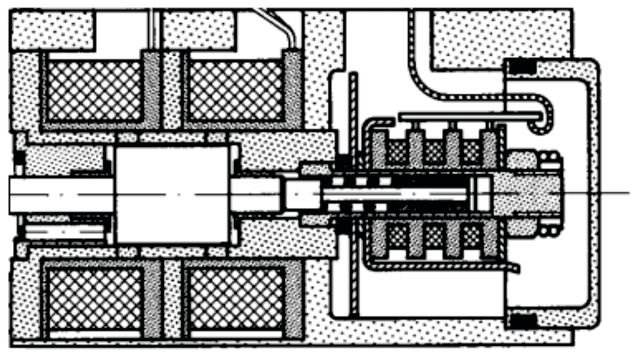

Fig.3. Bosch Rexroth control solenoid with position transducer [5]

Electromechanical converters [2] of such type contain a closed feedback loop in servo controller supported by position information received from the inductive transducer. Axial force inside plunger (Fig.4) depends on its position.

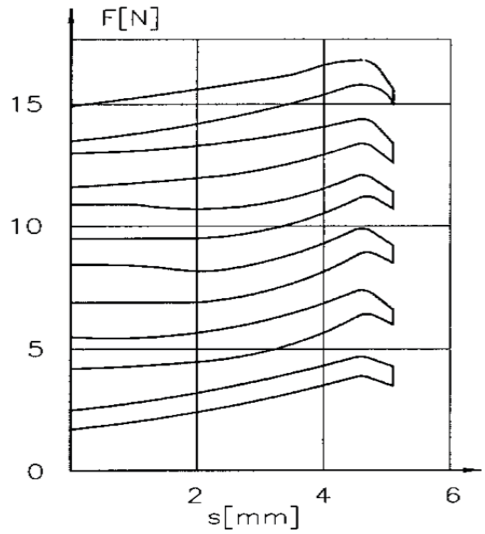

Fig.4. Stationary characteristic of a proportional stroke electromagnet

Design of proportional electromagnets used to control the slide valves of hydraulic directional control valves raises the issue: reducing electromagnetic gauge when increasing oil flows decanted through the directional control valve. Decrease of gauge, electric power consumption and consequently of production and use costs of proportional electromagnets requires their optimization depending on the axial forces that need to be developed for displacement of slide valves.

2. Mathematical modeling of axial forces needed for displacement of the slide valve

Axial forces needed for displacement of slide valves of hydraulic directional control valves consist of inertial forces, frictional forces, elastic forces of centering springs and hydrodynamic forces [3] due to the flow of working fluid through holes of variable area.

Q – volumetric flow
A0 – area of flow section
Cc – coefficient of contraction
Cd – coefficient of flow
ρ – density of working fluid
θ – jet angle
F – force
P – pressure
w – area gradient

2.1. Determination of hydrodynamic forces due to flow

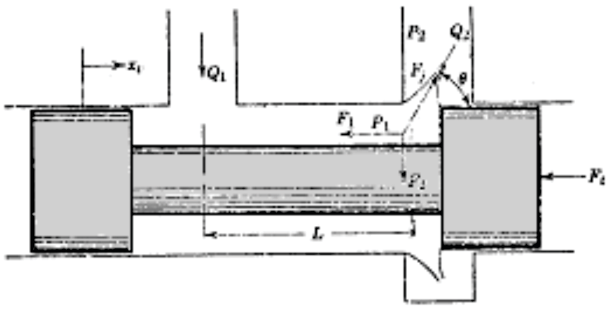

Fig.5. Flow forces due to flow

The most important component of the actuation force is of hydrodynamic type. Force developed due to flow through the section formed between the slide valve edges and sleeve:

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

Components (axial and lateral) of the equal and opposite force caused by Fj:

$$F_1 = -F_j \cdot \cos\theta \qquad (2)$$

$$F_2 = -F_j \cdot \sin\theta \qquad (3)$$

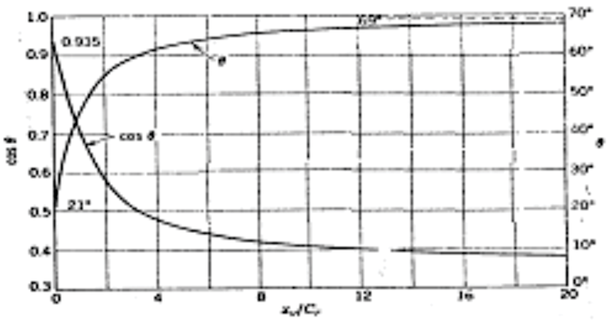

Fig.6.Effect of radial clearance on the jet angle

Neglecting the fluid compressibility inside the directional control valve and considering flow conservation:

$$Q_1 = Q_2 = C_d \cdot A_0 \sqrt{\frac{2}{\rho}(P_1 - P_2)} = C_c \cdot C_v \cdot A_0 \sqrt{\frac{2}{\rho}(P_1 - P_2)} \qquad (4)$$

It results:

$$F_1 = 2 \cdot C_d \cdot C_v \cdot A_0 \cdot (P_1 - P_2)\cos\theta \qquad (5)$$

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

In the transitory regime:

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

$$F_3 = L \cdot C_d \cdot w \sqrt{2 \cdot \rho \cdot (P_1 - P_2)} \frac{dx_v}{dt} + \frac{L \cdot C_d \cdot w \cdot x_v}{\sqrt{\left(\frac{2}{\rho}\right)(P_1 - P_2)}} \frac{d(P_1 - P_2)}{dt} \qquad (8)$$

In the dynamic regime force due to flow depends on the displacement speed of the slide valve and variation of pressure drop. Usually the influence of pressure drop is neglected, considering that it has no significant impact on system dynamics [3]. Component determined by the displacement speed of the slide valve is considered more important as it represents a factor of attenuation.

2.2. Determination of axial forces needed for displacement of the slide valve

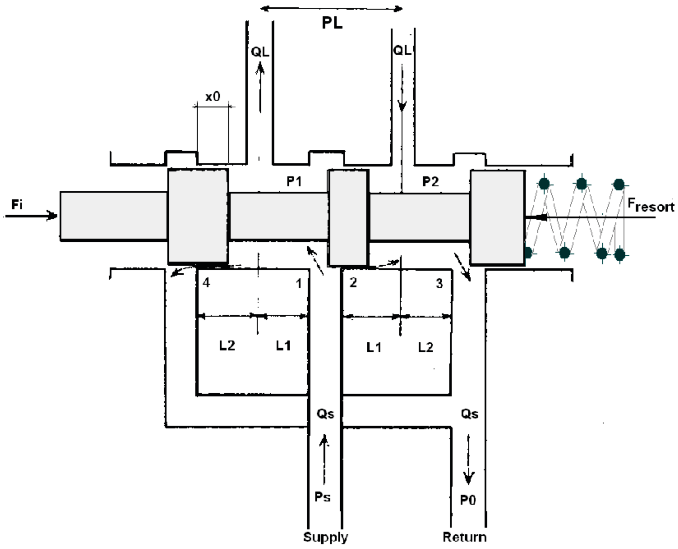

Fig.7. Three-joint four-way spool valve

Axial force needed for displacement of the slide

$$F_R = 2 \cdot C_d \cdot C_v \cdot (\cos\theta) \cdot w \cdot x_v \cdot (P_s - P_1) -$$
$$- L_1 \cdot \rho \frac{dQ_1}{dt} + 2 \cdot C_d \cdot C_v \cdot (\cos\theta) \cdot w \cdot x_v \cdot P_2 +$$
$$+ L_2 \cdot \rho \frac{dQ_3}{dt} + F_{resort} \qquad (9)$$

$$P_1 = \frac{P_s + P_L}{2} \qquad (10)$$

$$P_2 = \frac{P_s - P_L}{2} \qquad (11)$$

$$Q_1 = C_d \cdot A_1 \cdot \sqrt{\frac{2}{\rho}(P_s - P_1)} \qquad (12)$$

$$Q_3 = C_d \cdot A_3 \cdot \sqrt{\frac{2}{\rho} P_2} \qquad (13)$$

$$F_R = 2 \cdot C_d \cdot C_v \cdot w \cdot (\cos\theta) \cdot (P_s - P_1) \cdot x_v +$$
$$+ (L_2 - L_1) \cdot C_d \cdot w \cdot \sqrt{\rho \cdot (P_s - P_L)} \cdot \frac{dx_v}{dt} + F_{resort} \qquad (14)$$

3. Test system

Device for determining the actuation force of directional control valves with zero coverage (Fig. 10) was designed and developed in the Laboratory of General Hydraulics of INOE 2000 – IHP. In order to measure the actuation force for moving the slide valve, this device was equipped with a precise manual drive device, namely a micrometric screw.

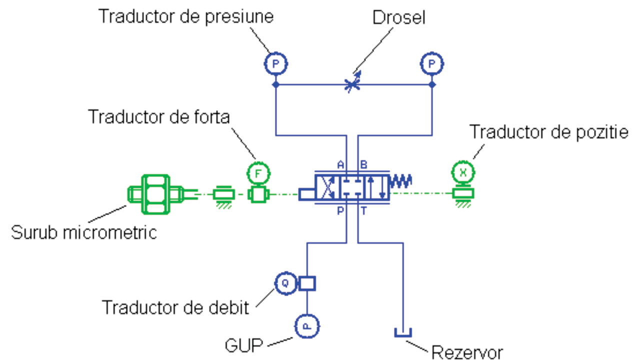

Fig.8. Basic schematic diagram of the device for measurement of axial forces

The device (Fig. 8) drives the control rod by means of a force transducer. Rod position is measured with an inductive position transducer. Return of the slide valve to initial position is made with a spring (Fig. 9) located on the side of the slide valve which is opposite to the one operated by the micrometric screw.

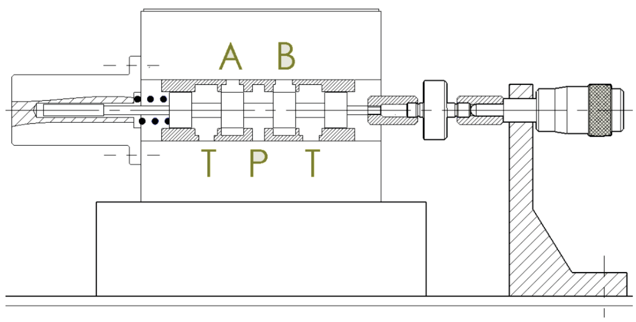

Fig.9. Actuation device used for positioning of the slide valve

Hydraulic parameters are measured by manometers placed on the joints A and B of the directional control valve. Hydraulic load can be achieved by changing the flow section of the throttle valve located between the joints A and B of the directional control valve. The device is supplied with pressurized oil from a generator block. Oil flow is measured using the flow meter located on the joint P. All transducers are connected via an acquisition board to a PXI NI system.

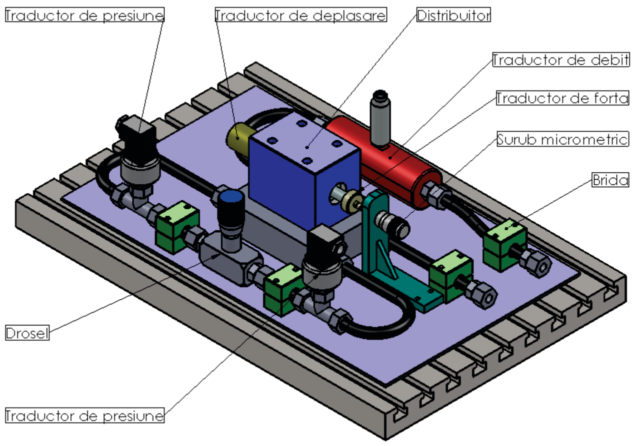

Fig.10. Device for measurement of axial forces

4. Conclusions

Complexity of natural phenomena usually doesn't allow systemic development of "perfect" mathematical models. Verification and tuning of mathematical models based on experimental data acquired from the processes of interest is mandatory in any study.
In the second part of the paper, "Determination of the actuation force of directional control valves with zero coverage (2nd Report, Experimental Results)", will be presented in full procedures for determining the axial forces, the experimental results and the procedure for adjusting parameters of mathematical model based on experimental data acquired.

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