

## THEORETICAL AND EXPERIMENTAL INVESTIGATIONS REGARDING THE DYNAMIC PERFORMANCES OF THE SERVO-SOLENOID DIRECTIONAL VALVE

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*Performanțele slabe ale echipamentelor electrohidraulice proporționale pot afecta performanțele unei întregi instalații. În prezent, tehnica de control electrohidraulic proporțional mai are nevoie de studiu, deoarece sunt încă probleme teoretice și practice de rezolvat. Cercetările cu privire la orice fel de tehnici de control sau îmbunătățire a echipamentelor proporționale vor promova și vor spori dezvoltarea tehnicii electrohidraulice. Pentru distribuitorii electrohidraulici pot fi observate o mulțime de inovații variind de la idei noi pentru realizarea convertorului electromecanic la modele mecanice complet noi. În articol se prezintă elemente de modelare a echipamentului și rezultate experimentale privind performanțele dinamice rezultate în urma unor diferite condiții de testare.*

*The nonperforming poportional electro hydraulic equipment may affect the performance of an entire installation. Nowadays the technique of proportional electrohydraulic control requires thorough study, because there are still unsolved theoretical and practical problems. The research regarding any kind of control techniques or improvement of the proportional equipment will promote and accelerate the development of the electrohydraulic technique. In what refers to the electrohydraulic servo valves a multitude of innovations should be mentioned varying from new ideas for creating the electromechanic converter to brand new mechanical models. In the article are presented elements of modelling the equipment and the experimental results regarding the dynamic performances obtained in various testing conditions.*

**Key words:** servo-solenoid, proportional, dynamics, electrohydraulic, control

### 1. Introduction

Applications with proportional electrohydraulic servo-solenoid directional valves are found in the industrial field and that of mobile installations. These equipment satisfy the special requirements of the industrial and mobile installations. They have known an increasing development lately and the use of electronics and microprocessors contributed to the improvement of their dynamic performances. The progress registered in the domain of proportional electrohydraulic directional valves enhanced their performances which approached those of the servo valves. The performant proportional solenoid-valves are used in application of control in close loop with a low price and low

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maintenance needs. At the proportional solenoid valves the command force generates the displacement of a spool which modifies the area of certain apertures that control flow and pressure in the system. The change of the area is designed to be approximately linear with the displacement of the spool. The perturbations in the motion of the spool are undesirable and may be caused by the friction forces between spool and body, by the static pressure forces caused by the geometry of the spool or by the inertia forces generated by the flow.

Advanced strategies of control may be used for obtaining satisfactory dynamic characteristics of the slide which should reduce the influence of the nonlinear conditions of operation of the servo-solenoid valve.

This type of servo-solenoid valves will operate optimally if provided with a controller dedicated with position loop with LVDT sensor [1], [2], [3]. The controller minimizes the effects of the nonlinear perturbations at the displacement of the slide and improves the answer in frequency.

In the present paper are presented elements regarding the modeling of the dynamics of the servo-solenoid directional valves and experiments performed on a model, which may be used for validating a simulation model. For the modeling friction forces as a function of speed, are taken into account according to Stribeck diagram.

## 2. The modelling of the servo-solenoid directional valve

The servo-solenoid directional valve comprises a hydraulic body, two poportional electromagnets, the distributing spool and two centering springs of the spool. In fig.1 may be noticed the symbolic representation of such a valve.

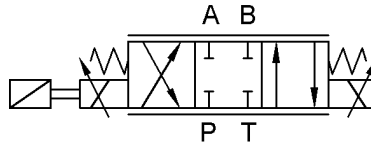


Fig. 1. The symbol of the proportional electrohydraulic servo-solenoid directional valve

### 2.1 The dynamic model

The model comprises the spool with mass  $m_s$ , the centering springs of the slide, the two armatures of the electromagnets with the masses  $m_{A_1}$  and  $m_{A_2}$  and the armature (core) of the device with the mass  $m_{A_T}$ . The schematization of the subsystem may be seen in fig. 2 where  $x$  is the position of the spool  $K_1$  and  $K_2$  are the non linearities generated by the centering springs and  $F_1$  and  $F_2$  the forces generated by the electromagnets. The linear component of viscous friction  $d\dot{x}$  and the hydrodynamic force  $F_h$  are taken into account. The dynamics of the subsystem has the following form:

$$m \frac{d^2 x}{dt^2} + F_f(\dot{x}) + d \frac{dx}{dt} + cx = F_1(I) - F_2(I) - F_K(x) + F_h(x, \Delta p) \quad (7)$$

The equivalent mass of the subsystem is:

$$m = m_{A_1} + m_{A_T} + m_{A_2} + m_S \quad (8)$$

and

$$K_1 = K_2 \quad (9)$$

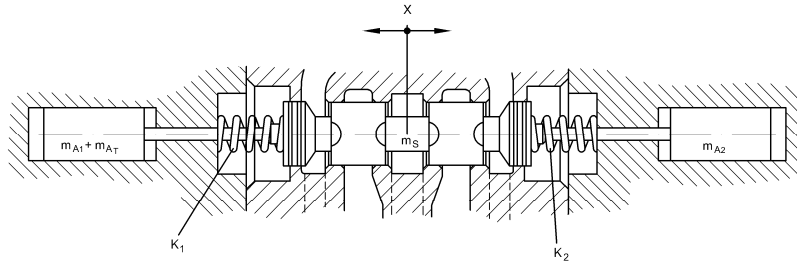


Fig. 2. The mobile equipment of the solenoid

A standard method of modelling the friction force in the electrohydraulic servosystems is the expression of friction as a function of speed, through the Stribeck friction curve (fig. 3).

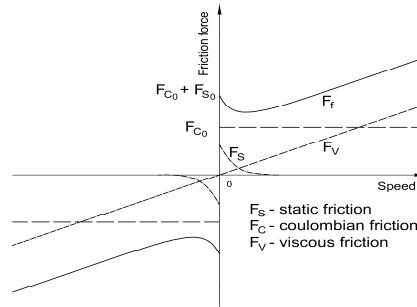


Fig. 3. Stribeck diagram

For low speeds the hydraulic fluid acts as a superfluous layer and the shear forces determine the friction force. For high speeds and low pressures a permanent layer of fluid is formed and friction is in relation with the hydrodynamic effects. Friction in this case depends on the viscosity of the hydraulic fluid and the distribution of the speeds in the fluid layers.

$$F_f(\dot{x}) = F_V(\dot{x}) + F_C(\dot{x}) + F_S(\dot{x}) = \alpha \dot{x} + \text{sign}(\dot{x}) \left[ F_{C_0} + F_{S_0} \exp\left(-\frac{|\dot{x}|}{v_S}\right) \right] \quad (10)$$

The three characteristic parts of the curve are: viscous friction  $F_V$ , static friction  $F_S$ , and Coulombian friction  $F_C$ .

$\sigma$  is the parameter for viscous friction,  $F_{C0}$  is the parameter for Coulombian friction, and  $F_{S0}$  and  $v_S$  (named Stribeck speed) are the parameters for the static friction.

The hydrodynamic force of stationary mode  $F_h$  is given by the relation:

$$F_h = 2n \cdot c_d A(x) \Delta p \cdot \cos \theta \quad (11)$$

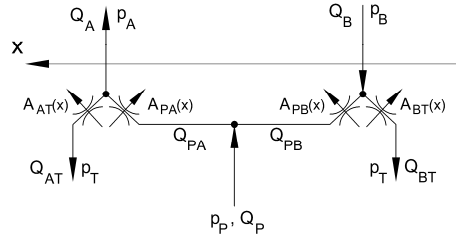


Fig. 4. The diagram with half bridges of the servo solenoid valve

To calculate the hydrodynamic force, which is dependent on opening  $x$ , the diagram with half bridges of the servo solenoid valve from fig. 4 is considered.

$$F_h = 2nc_d A_{PA}(p_P - p_A) \cos \theta - 2nc_d A_{AT}(p_A - p_T) \cos \theta \quad \text{for } -x_0 < x < x_0 \quad (12)$$

$$F_h = -2nc_d A_{PB}(p_P - p_B) \cos \theta + 2nc_d A_{BT}(p_B - p_T) \cos \theta \quad \text{for } x \geq x_0 \quad (13)$$

$$F_h = -2nc_d A_{PB}(p_P - p_B) \cos \theta + 2nc_d A_{AT}(p_A - p_T) \cos \theta \quad \text{for } x \leq -x_0 \quad (14)$$

where:  $n$  – number of hydraulic circuits,  $n = 4$ ;

$c_d$  – drosseling constant,  $c_d = 0,7$ ;

$A_i(x)$  – flow sections;

$x_0$  – coverage;

$\theta$  – the direction of the resultant of the hydrodynamic force,  $\theta \leq 69^\circ$ .

In fig. 5 the flow through the aperture at an opening of the spool  $x$  and the angle of the hydrodynamic force  $\theta$  are shown.

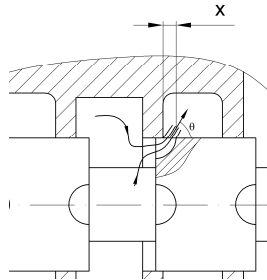


Fig. 5 Diagram of the flow at the opening of  $x$  and the angle of the flowing jet  $\theta$

For circular apertures of the spool, the area depending on the aperture  $x$  is given by the relation:

$$A(x) = NR^2 \arccos \frac{R-x}{R} - (R-x)\sqrt{2Rx-x^2} \quad (15)$$

where  $N$  is the number of apertures from the lamination edge and  $R$  the radius of the aperture.

### 2.3 The valve flow

The flow depends upon the pressure from the taps A, B, P and T. The studied solenoid valve is with close center, meaning that if the command solenoid is in a neutral position, all its four apertures are closed and without connection between themselves. In fig. 6 are shown the flow directions through the valve at an aperture of the slide towards the right side field.

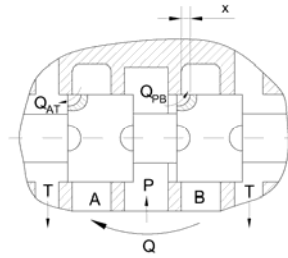


Fig. 6. Flow directions for the field  $P \rightarrow B$  and  $A \rightarrow T$

The flows for the aperture shown in fig. 6 are:

$$Q_{PB} = c_d A(x) \sqrt{\frac{2}{\rho} (P_P - P_B)} \quad (16)$$

$$Q_{AT} = c_d A(x) \sqrt{\frac{2}{\rho} (P_A - P_T)} \quad (17)$$

### 3. Experimental model and testing stand

For finding the dynamic performances a servo solenoid valve manufactured by BoschRexroth type 4WRE was used as an experimental model.

The results may be used for finding the values of certain parameters in order to use them in a simulation model, for validation or synthesis with another apparatus.

The dynamic characteristics are usually specified through the frequency answer which shows the relation between the outlet parameter (flow) and the inlet parameter (current) when the last one has a sinusoidal variation of constant amplitude in a certain frequency range. This relation is expressed through attenuation representing the relation between the flow at any frequency and the one obtained at a reference frequency and through the phase angle between the outlet and the inlet parameters.



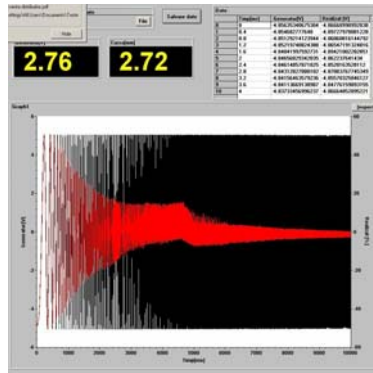


Fig. 8 TESTPOINT application panel for testing frequency response

For the measurement a sweep signal with a sinusoidal frequency from 1 Hz to 200 Hz and duration of 10 seconds was generated. First test was conducted with an signal amplitude of 10 Volts and second test with an amplitude of  $\pm 2.5$  V. Data for the signal generated, servo-solenoid directional valve response and time were recorded by three channels and stored in a data file with an application made in the TESTPOINT environment (Fig. 8).

In the following figures the recording charts obtained as a result of experiments are shown.

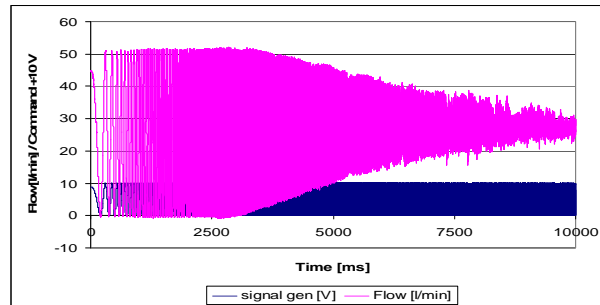


Fig. 9 Chart for 100 % command measured for  $t_{oil} = 40^{\circ}\text{C}$ ,  $p_s = 20$  bar, 1÷200 Hz

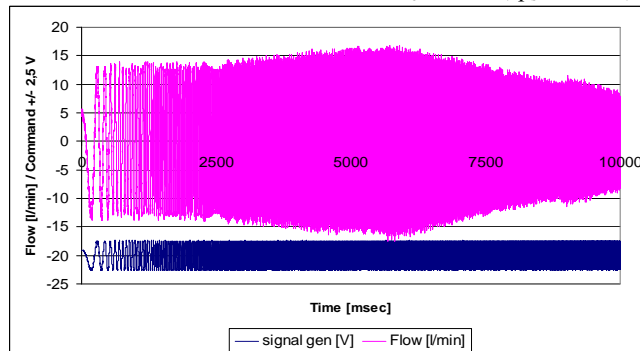


Fig. 10 Chart for 25 % command measured with  $t_{oil} = 40^{\circ}\text{C}$ ,  $p_s = 20$  bar, 1÷200 Hz

After data processing the Bode diagrams were drawn according to the following figures.

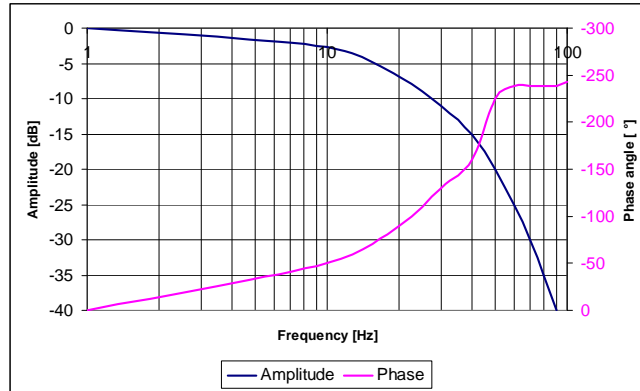


Fig. 11 Bode diagrams for 100 % command signal

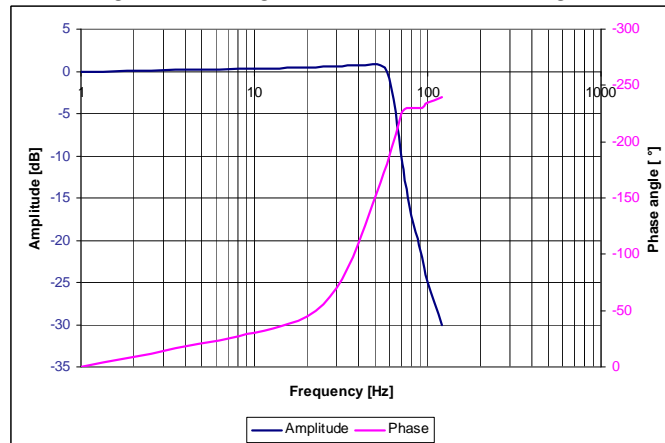


Fig. 12 Bode diagrams for 25 % command signal

In the following figure is presented the response to a -5 V to +5V step signal.

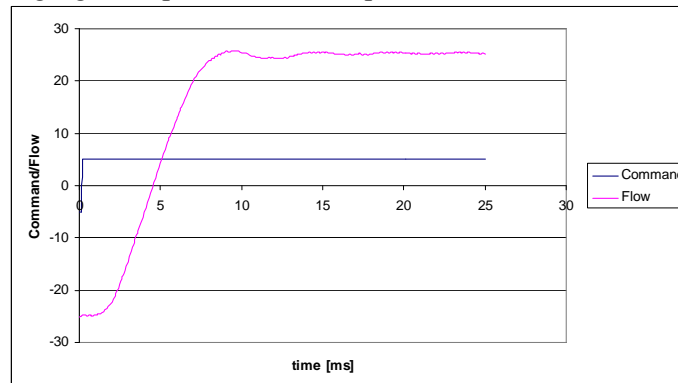


Fig. 13 Step response for -5 V to 5V command signal



In the case of proportional solenoid valve with integrated electronics and output for feedback signal from distribution spool (case available for the device used in experiments) the transfer function of this electrohydraulic device can be determined, using an FFT analyzer. Was used type SR 780 analyzer manufactured by Stanford Research Systems (Fig. 14).

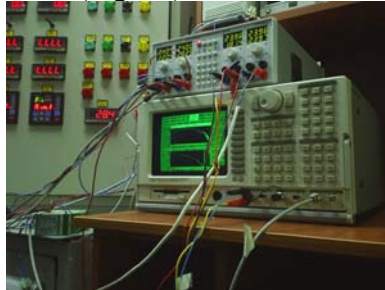


Fig. 14 Determining the transfer function using a Network Signal Analyzer type SR780 produced by the Stanford Research Systems

The input signal is digitized at a high sampling rate. Nyquist's theorem says that as long as the sampling rate is greater than twice the highest frequency component of the signal, then the sampled data will accurately represent the input signal (in the frequency domain). In the SR780, sampling occurs at 262 kHz. To make sure that Nyquist's theorem is satisfied, the input signal passes through an analog anti-aliasing filter that removes all frequency components above 102.4 kHz. The resulting digital time record is then mathematically transformed into a frequency spectrum using an algorithm known as the Fast Fourier Transform or FFT. The resulting spectrum shows the frequency components of the input signal. The signal generated by the internal source of the analyzer was applied to the voltage input of the servo-solenoid directional valve and to one of the entries in the analyzer. Response signal from the spool stroke transducer was introduced in the second input of the analyzer resulting the diagram in Fig. 15.

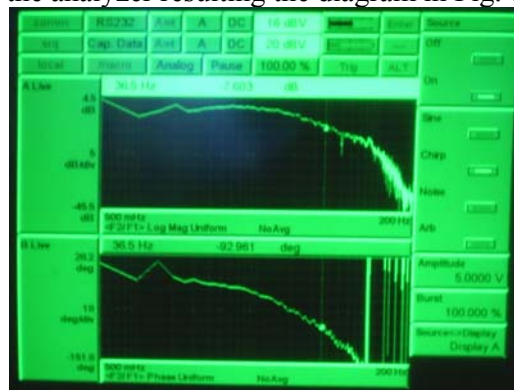


Fig. 15 Sine wave input signal, 5 V amplitude,  $p_0 = 100$  bar

## 6. Conclusions

It may be noticed a delay in the displacement of the cylinder caused by the friction forces. This means that the time required for the proportional elements to surpass friction forces and forces generated by the centering springs of the spool increases.

The experimental results may be used for:

- The check up of the theoretical model
- The validation of a simulation model realized on the basis of the theoretical model and the synthesis of another proportional apparatus through simulations
- The calculation of certain parameters which cannot be measured directly.

In the perspective, based on the developed model, simulations for optimizing performances of this equipment can be performed.

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