ANNALS of the ORADEA UNIVERSITY.
Fascicle of Management and Technological Engineering, Volume VI (XVI), 2007

SOME ASPECTS REGARDING ELECTROHYDRAULIC POSITIONING SYSTEMS

Irina TIȚA
Technical University „Gh. Asachi”, Iaşi

Keywords electrohydraulic system, adjusting controller, servocylinder

Abstract For an electrohydraulic system with cylinder we must consider nonlinearities due to the variable load and find a way to compensate them. In order to diminish the influence of variable load we propose an association including the control valve and a controller adjustable with load. This assembly acts like a control valve with a constant gain. Using this structure the nonlinearities are compensated “on-line” and the accuracy of the system does not change when the load varies. The proposed structure was applied, and some results are presented in this paper.

1. INTRODUCTION

We consider the system in figure 1, which includes the association proportional valve - double acting cylinder.

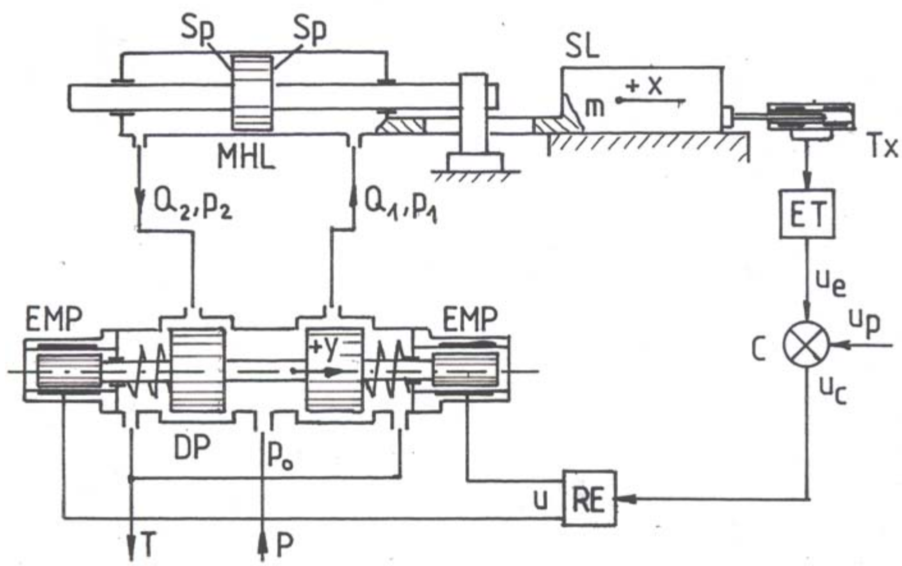

Fig.1. The structure of the system

The system has a control valve with four edges and an actuator double acting double rod.

Variable load of the system changes the pressure drop on the control valve and its gain. These changes have as consequence the diminishing of the accuracy of the system

2. THE STRUCTURE OF THE CONTROLLER

In order to diminish the influence of variable load we propose an association including the control valve and a controller adjustable with load (figure 2). This assembly acts like a control valve with a constant gain.

BC2 computes the variable gain k_R for the adjustable controller. In some cases the controller can be a digital one. The source of information is the pressure transducer connected to the chamber of the cylinder. If we put on the forward path a controller having the gain $k_R^* k_R^{**}$ adjustable, it is possible that the gain of this combination stays invariable at the value k_R. The relation to determine k_R^* results from the condition that k_R^* compensates the flow force spring rate.

$$k_R^{\ *} = 1 + \frac{k_{hd}}{k_r} \tag{1}$$

where k_{hd} is the flow force spring rate and k_r is the spring rate

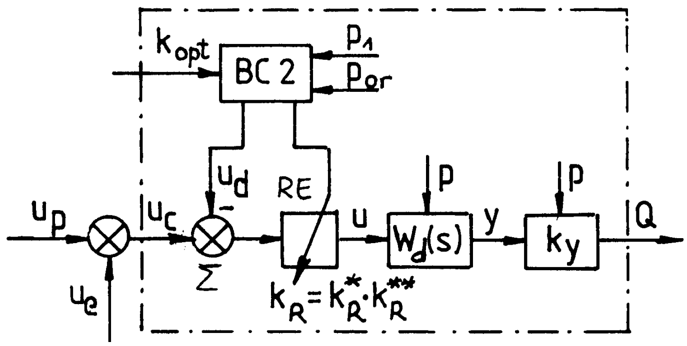

Fig.2. The structure of the controller

In figure 2 we can see that the working pressure p acts like a perturbation on the control valve having the transfer function W_d.

For solving (1) we must know the spring rate k_r and flow force spring rate k_{hd}. It represents the gradient of flow force with spool displacement *y.*

Flow through the valve is:

$$Q = \alpha \pi d_s (y_0 + y)\sqrt{\frac{2}{\rho}(p_0 - p_1)} - \alpha \pi d_s (y_0 - y)\sqrt{\frac{2}{\rho}(p_1 - p_T)} \tag{2}$$

Considering the adimensionalized equation:

$$\overline{Q} = (1 + \overline{y})\sqrt{1 - \overline{p}} - (1 - \overline{y})\sqrt{\overline{p}} \tag{3}$$

In figure 3a one can see the dependence $\overline{Q} = f(\overline{y})$ for different values for $\overline{p}$.

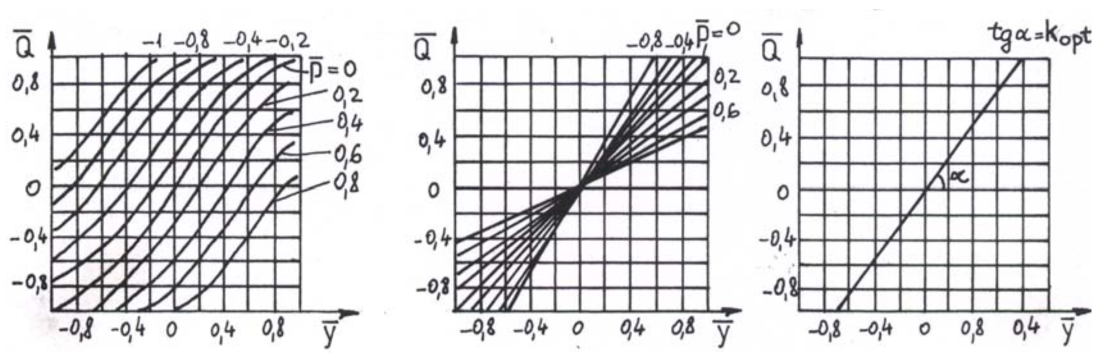

Fig.3. Compensation of static gain

Considering an optimum value for the gain k_o between command tension and the flow rate, we can calculate the suplimentary command tension u_d:

$$u_d = \frac{Q}{k_o} \tag{4}$$

In this way the characteristics of the valve are passing through origin, having different gradients as one can see in figure 3b.

$$u_d = \frac{1}{k_o}\alpha\pi d_s y_0\sqrt{\frac{2}{\rho}}\left(\sqrt{p_0 - p_1} - \sqrt{p_1}\right) \quad (5)$$

The gradient of the flow trough the valve is:

$$k_y = \frac{\partial Q}{\partial y} = \alpha\pi d_s\sqrt{\frac{2}{\rho}}\left(\sqrt{p_0 - p_1} + \sqrt{p_1}\right) \quad (6)$$

We can make a system resulting from the association valve with variable gradient and controller with variable gradient k_R^{**} compensating the variaton of k_y such as the global gain k_y^{**} is invariable and his value to be the optimum value.

$$k_y^{**} = k_R^{**}k_y = k_R^{**}\alpha\pi d_s\sqrt{\frac{2}{\rho}}\left(\sqrt{p_0 - p_1} + \sqrt{p_1}\right) = \frac{k_o k_r}{k_F} \quad (7)$$

The variation with the load of the second term is compensated with k_R^{**} which can make the global gain have the optimum value k_o , value that do not change when the load is changing.

$$k_R^{**} = \frac{k_o k_r}{k_F\alpha\pi d_s\sqrt{\frac{2}{\rho}}\left(\sqrt{p_0 - p_1} + \sqrt{p_1}\right)} \quad (8)$$

Using the proposed structure for the controller of an electrohydraulic positioning system we can improve accuracy and rapidity of such systems.

3. THE EQUIPMENT

The electrohydraulic positioning system that we have conceived (figure 5) was used at the noncircular cutting of a cam. This system has a loop for the adaptive adjustment of the controller

The force part is built with a constant pressure source, single rod actuator, and a control valve with two edges. Such a control valve has different gains depending on the displacement of the spool [6].

The command part, figure 4, has a pulse generator with rotation GIR coupled on the spindle. It can send 1000 pulses on each rotation. It is possible, in these conditions, to have 1000 values for the radius of the piece and we can obtain different shapes for the cross section. BUP sends to the system the reference input up. The information regarding the effective displacement of the slide is given by the transducer Tx. The gain of the controller has two parts. One of them k_{Rn}* is numerically adjustable and the other k_{Rn}** is invariable. The values for k_{Rn}* are calculated for different values of the pressure drop Δp on active edges and are put in two tables in the computer. The block BCA must search in the tables the value for k_{Rn}* at different measured pressure drops Δp. The values for k_{Rn}* included in the tables can be calculated or experimental determined.

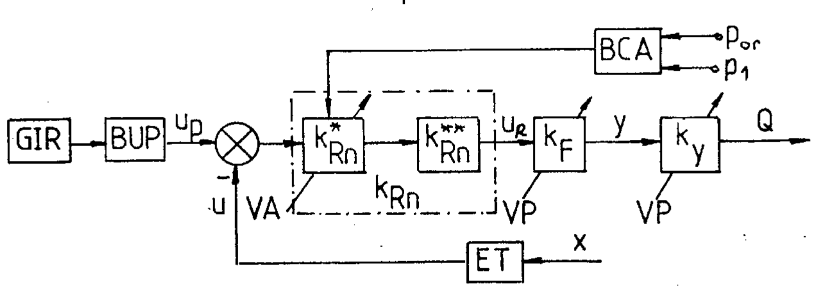

Fig.4. The structure of adaptive loop

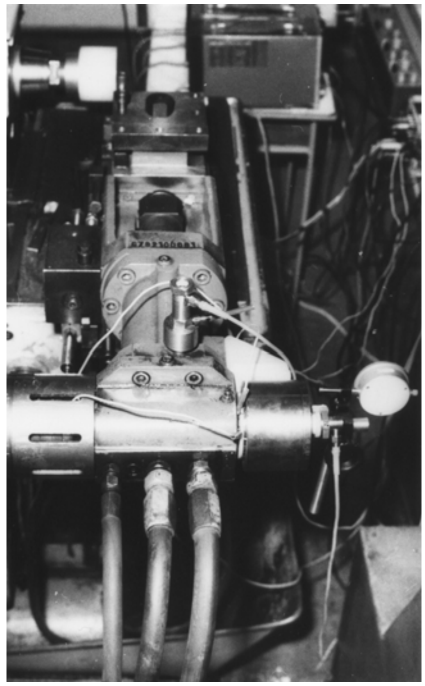

Fig.5. The electrohydraulic positioning system

4. EXPERIMENTAL RESULTS

The system was tested at the cutting of a cam with and without the adaptive loop, at the different values for the pressures p_0 at the source.

In figure 6 we can see the domain for the gains k_F and k_y regarding the control valve. In figure 6a we can see the values for positive displacements of the valve's spool and in figure 6b for negative ones These variations had serious consequences regarding the accuracy of the system.

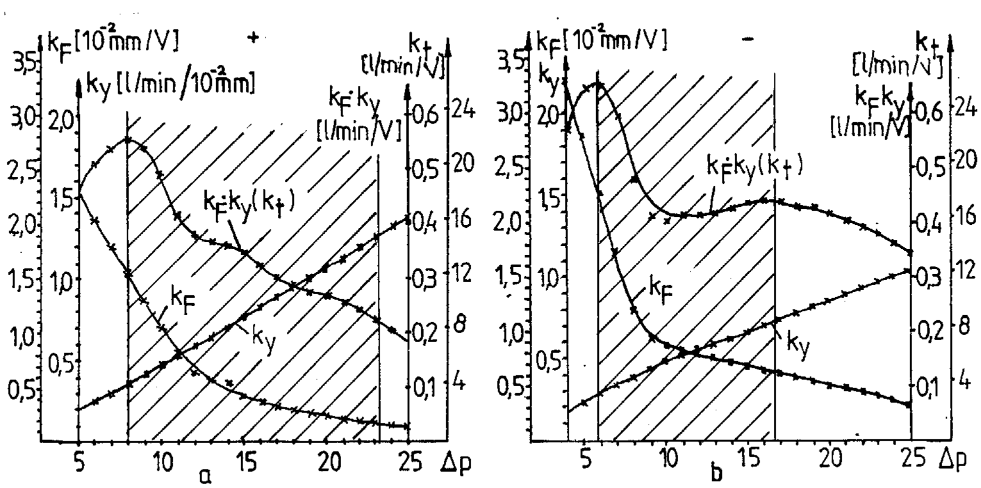

Fig.6 The domains for the gain of the control valve without adaptive loop

The input x_p is the eccentricity of the cam. Figure 7 shows the registered profile of the cam obtained for the system without the adaptive loop at different pressures of the source.

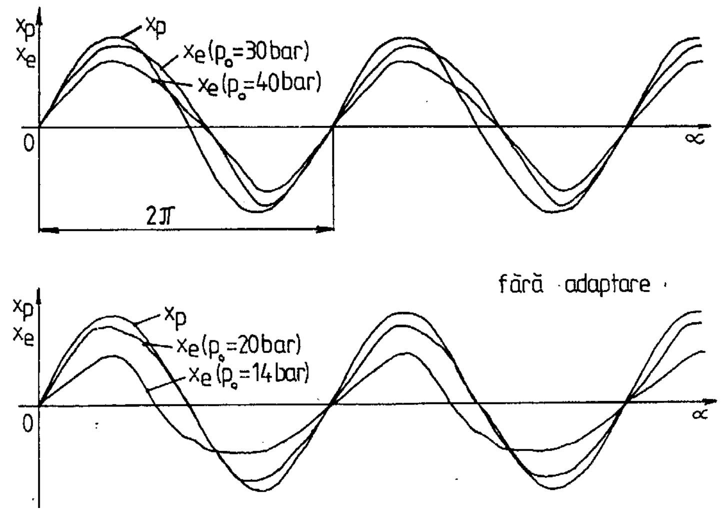

Fig.7 The registered shape of the cam cutted when the system has not adaptive loop

We can see that the most important effect of the diminishing of the global gain is the error of shape illustrated in figure 7.

Using an adaptive loop the domain for the valve's coefficients is changing and the global gain k_t has a constant value.

In figure 8 we can see the registered shapes of the cam obtained when the adaptive loop of the system is working. The improvement is illustrated both in amplitude and shape of output value.

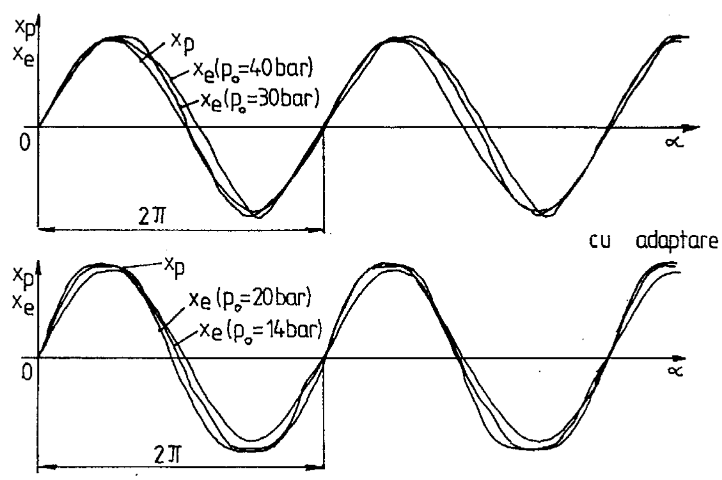

Fig.8 The registered shape of the cam cutted when the system has adaptive loop

5. CONCLUSIONS

Comparing the two cases with and without the adaptive loop it is obvious that using an adaptive loop we improve both the amplitude and the shape of output signal. The structure of the proposed loop was successfully verified and we can use the structure and the proposed methodology for a lot of electrohydraulic positioning systems working in different conditions.

BIBLIOGRAPHY

[1] Backé, W., Systematik der hydraulischen Widerstandsschaltungen in Ventilen und Regelkreisen, Krausskopf-Verlag, Mainz, 1974.
[2] Manring, N., Hydraulic Control Systems, John Wiley & Sons Inc, 2005.
[3] Merritt, H., Hydraulic Control Systems, John Wiley & Sons Inc., New York, 1967.
[4] Prodan, D., Masini-unelte - modelarea si simularea elementelor si sistemelor hidrostatice, Ed. Printech, Bucuresti, 2006.
[5] Tita, I., Contributions at the design of an adaptiv controller for a hydromechanical positioning system, Buletinul I. P. Iaşi, tom XLVI, Supliment Mecanica Fluidelor, Iasi, 2000.
[6] Tita, I., Drutu, C., Noncircular cutting using a hydromechanical positioning system with an adaptive loop, , Buletinul I.P. Iaşi, tom XLVI, Supliment Mecanica Fluidelor, Iasi, 2000.
[7] Tita, I., Researches regarding the design of a controller for an electrohydraulic system. TMCR Chisinau, 2007.
[8] Tomizuka, M., Chen, M.S., Renn, S., Tsao, T.C., Tool Positioning for Noncircular Cutting with Lathe, Transactions of the ASME, Journal of Dynamic Systems, Measurement and Control, 109, March, 1987.