STUDY REGARDING ELECTRO-HYDRAULIC LOAD-SENSING SYSTEMS

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Abstract: The paper describes the structure and the mathematical model for a load-sensing system having in view the effects of a simplified analysis (neglecting the flow dynamic effects). The proposed model helps us to see the way to improve the energy balance of the system and how to introduce a load-sensing loop.

Keywords: electro-hydraulic system, load-sensing, energy saving

1. INTRODUCTION

For the positioning systems the most important problems are point to point control and path following. In the first case it is important to move the object from one point to another and the transient path it is not important. In the second case the object must be moved precisely along the desired trajectory [1], [2].

In this paper is studied an electro-hydraulic system used for the path following case, having in view a variable load.

2. THE STRUCTURE OF THE SYSTEM

From the point of view of energy balance the load-sensing systems are to be preferred. A simple load-sensing loop can be made out using a fixed displacement pump and a relief valve with a load-sensing connection which changes continuously the reference pressure [3]. This is a load-sensing which improves the energy balance and it is less complicated.

The structure of the system is presented in figure 1. We propose a loop for calculation and adjustment of pressure of the source, on the basis of the estimation of the pressure during a working cycle. After that, the loop disconnects (we assume that the load is not changing) until a new command for a working cycle with different parameters. This loop has as source of information the pressure transducer T_p that measures the load pressure. At the beginning of each working cycle the pressure of the source is adjusted at maximum value. The calculus block BC1, which receives the information regarding the load pressure during a rotation of the part does this adjustment. BC1 processes the information considering the minimum pressure drop Δp_{min} on the active edge of the control valve and sends the rational value for the pressure of the source p_{0r} to the relief valve SLP.

The system studied in this paper has a constant pressure source made out of a pump with constant flow and a relief valve. The flow control and distribution is done with a proportional valve with two active edges (Fig.1).

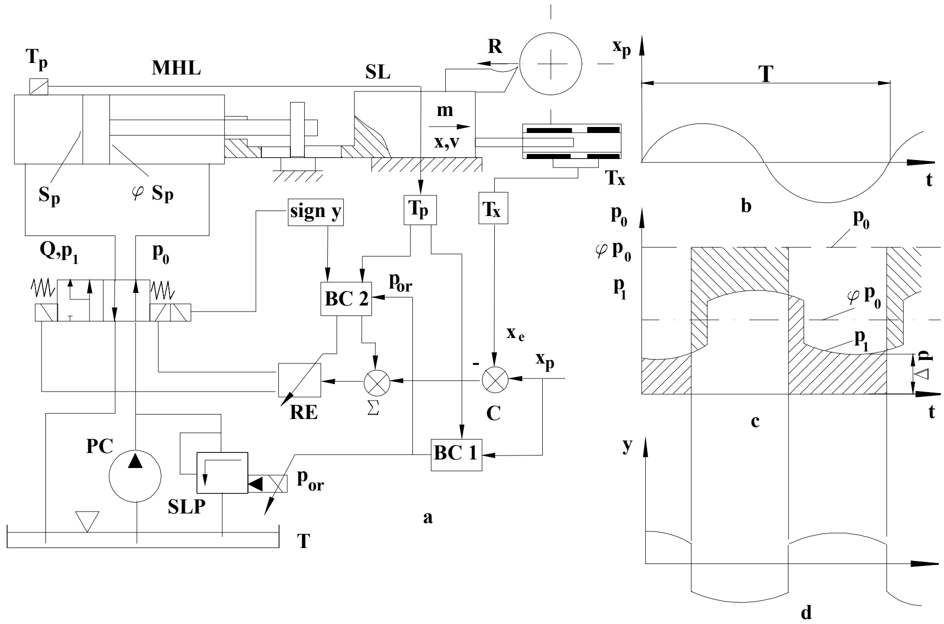

Fig. 1 The electro-hydraulic system with load-sensing

3. THE LOAD-SENSING LOOP

We propose an association including the calculus block BC1 and the relief valve SLP with continuously variable set value of the pressure.
At the beginning of each working cycle the pressure p_0 of the source is adjusted at the maximum value. The calculus block BC1 receives the evolution of the load pressure during a rotation of the part.

The equilibrium at the actuator is:

$$m \cdot \ddot{x} + h \cdot \dot{x} = \varphi \cdot S_p \cdot p_0 - p_1 \cdot S_p - R \tag{1}$$

Where: m is the mass of the assembly, h is the rate of viscous friction, Sp is the area of the piston and R is external load.

It results the load pressure:

$$p_1 = \varphi \cdot p_0 - \frac{1}{S_p}\left(m \cdot \ddot{x} + h \cdot \dot{x} + R\right) \tag{2}$$

Figure 1b depicts a possible law for motion. The corresponding evolution of the pressure p_1 is presented in figure 1c. The pressure drop on the active edge of the valve is $\Delta p = p_0\text{-}p$, for negative displacements y of the spool of the valve, and $\Delta p = p_1$ for positive values of y.

We consider that the condition for the maximum efficiency is the work with pressure drops on the active edges greater than a preset value Δp_a. It must be evaluated the rational pressure of the source, for the working cases with y positive, assuming that $p_{1\,min}$ appears in these situations (figure 1c, d), with:

$$p_{0r} = p_0 - \frac{1}{\varphi} \cdot p_{1\max} + \frac{1}{\varphi} \Delta p_a \tag{3}$$

This relation is the necessary condition for obtaining a minimum set pressure drop Δp_a on the active edge. For negative values for y:

$$p_{0r} = \frac{1}{1-\varphi} \cdot p_{1\max} - \frac{\varphi}{1-\varphi} \cdot p_0 + \frac{1}{1-\varphi} \cdot \Delta p_a \tag{4}$$

The equations (3) and (4) may be used only if the load pressure does not change essentially with the pressure of the source.

Finally we have the value for p_{0r} as the maximum between the two aroused from (3) and (4), which is sent to the relief valve as set value of pressure.

4. THE LOOP FOR THE ADJUSTMENT OF THE CONTROLLER

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. BC2 computes the variable gain k_R for the adjustable controller. In some cases the controller can be a digital one [1]. The source of information is the pressure transducer T_p. If we put on the forward path a controller (Fig. 2) 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^* is the same with that for the four edges valve [4] and results from the condition that k_R^* compensates the flow force spring rate.

$$k_R^* = 1 + \frac{k_{hd}}{k_r} \tag{5}$$

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

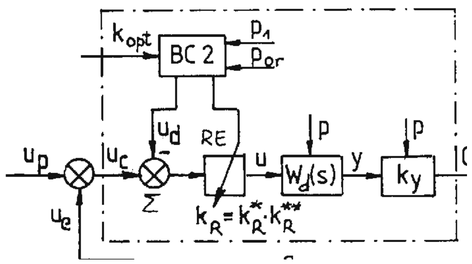

Fig. 2 The structure of the control valve with constant gain.

The gain of the control valve changes with the load also because of the flow to the actuator.

4. CONCLUSIONS

We can state that the algorithm for BC1 is: after initial set value for p_{0r} at maximum and the first turn of the part, from the registered values of p_1 are determined $p_{1\ min}$ and $p_{1\ max}$ and then, in concordance with sign y, is calculated the value for the rational pressure p_{0r}, which is sent to the relief valve and will be the pressure of the source for all the working cycles with similar conditions

Using the proposed structure of the controller of a electro-hydraulic system we can improve accuracy and rapidity of such systems.

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