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THE CHARACTER OF THE DYNAMIC PROCESSES IN THE PROPORTIONAL SERVODRIVING SYSTEMS OF THE HYDRAULIC PUMPS CAPACITY

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ABSTRACT

The proportional servodriving systems of hydraulics pump capacity are used in many more applications in tehnological equipments driving. Thus, the construction of specialized pumping hydraulics units, like the A10VG and A4VG Bosch-Rexroth pumps, with HD hydraulic proportional command, HW angular proportional mechanical command, EP electrical proportional command, it offer a wide posibilities for driving the tehnological equipments, at which the first condition required to the system is the proportinality beetween the command signal and the momentan value of pump capacity. Thus, it is obtained the proportinality beetween the command and the liniar or angular speed, for liniar or rotative hydraulic motors which drive the working body of the equipment. In the paper the authors elaborates the mathematical model for a look like servodriving system, deduces the mathematical expresions for dynamic parameters of servodriving control system, the transfer functions for the components and for the ensemble.

1. Introduction

The systems for capacity proportional servo-driving of a volumic pump with variable capacity are wore and more used for technological equipments acting. Through making the hidraulic pumping units, with incorporated servodriving system, like the Bosch – Rexroth pump units – A10VG and A4VG, with HD proportional hydraulic command; HW proportional angular mechanical command; EP proportional electrical command, offer the large opportunities for acting the technological equipments at which the first condition imposed to the system is: the proportionality between the command signal and the momentary value of pump capacity, respectively the flow value of the hydraulic agent send by the pump in the acting system.
Thus, it obtain the proportionality between the maded command and the angular or liniar speed of the hydraulic motor which drive the working tool of the equipment.
The principial diagram of the proposed acting system is presented in Figure 1, where the working tool of the equipment is represented by the OL symbol, and it means the equipment which need an angular speed chara cteristic that must be proportional whith the operator command.

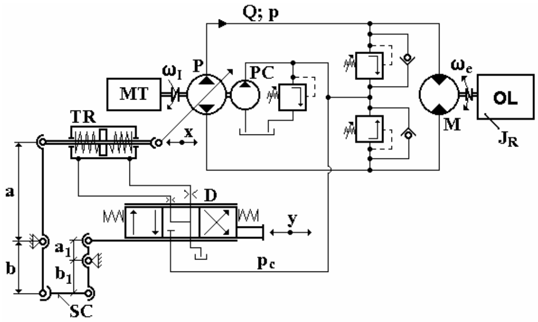

Figure 1. Servo-driving system diagram

In the Figure 1 the symbols are: P - variable capacity pump; M - hydraulic motor; PC - command system pump (with fixed capacity); TR - pump regulator; D - proportional distributor for the pump capacity command; SC - mechanical bvalance system, composed by the levers system which realize the servo-tracing system; x - regulator rod deplacement; y - command deplacemenent; J_R – the moment of inertia, reduced to the hydraulic motor shaft, for the equipment components in movement.

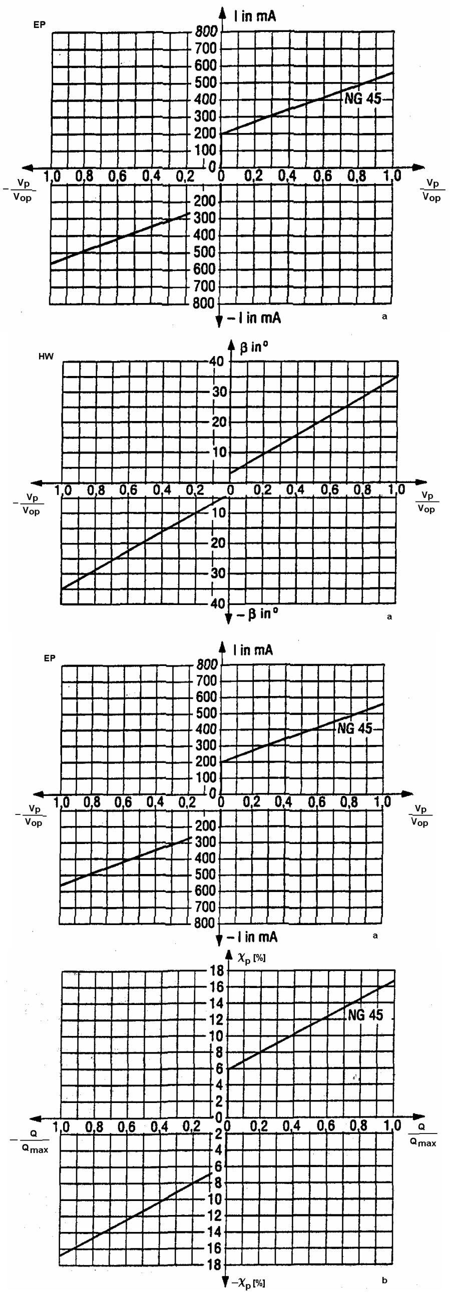

Figure 2. The command and the executive of the command characteristics

a – the command characteristic: HD – hydraulic type; HW – angular type; EP – electrical type; b – the execution characteristic of the command.

The command characteristic and the execution characteristic of the command are presented in Figure 2.

This paper are proposed to elaborate the global function for the previous preseted automatic system. The concrete cases concerning the system stability will be analyzed separated for each application.

2. The Mathematical Model for the Automatical Regulator System

The diagram of the proposed system, like a automatical regulating system, is presented in Figure 3.

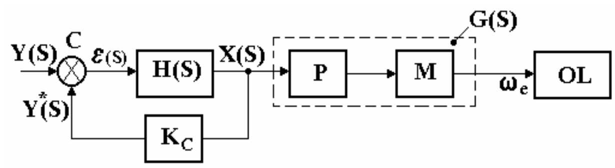

Figure 3. The system diagram

G(s) – the transfer function for the hydraulic pump-motor circuit (execution branch); H(s) – the trensfer function for the command circuit, composed by the distribuitor D and the pump regulator TR (command branch); C-comparator block, composed by the levers system SC, which realize the link between the commanded value y(s) and the maded value $y^*(s)$.

3. The Calculus of the Execution Branch Transfer Function – G(S)

For the writing of the system model is necesary to write the model for the execution branch, composed by the variable capacity pump P, hydraulic motor M and the acting mechanical system. The model is composed by the flow equation between the pump and the mootr, and the dynamic equipment equation at the hydraulic motor shaft.

Flow equation is given by the next expression

$$x\cdot\frac{V_{0P}}{2\pi\cdot x_{\max}}\cdot\omega_i=\frac{V_{0M}}{2\pi}\cdot\omega_e+\alpha_{PM}\cdot p+\frac{V}{E_r}\cdot\frac{dp}{dt}\qquad(1)$$

The dynamic equilibrium equation at the hidraulic motor shaft is given by the

$$J_R\cdot\frac{d\omega_e}{dt}+\left(\gamma_M+\frac{V_{0M}}{2\pi}\cdot\delta_M\right)\cdot\omega_e=\frac{V_{0M}}{2\pi}\cdot p\qquad(2)$$

In the equations (1) and (2) was neched the terms which have the minimal influences about the system behavior.

From equation (2) we can obtain the momentary pressure value and the time variation of the pressure:

$$p = \frac{2\pi \cdot J_R}{V_{0M}} \cdot \frac{d\omega_e}{dt} + \frac{2\pi \cdot (\gamma_M + \frac{V_{0M}}{2\pi} \cdot \delta_M)}{V_{0M}} \cdot \omega_e \quad (3)$$

$$\frac{dp}{dt} = \frac{2\pi \cdot J_R}{V_{0M}} \cdot \frac{d^2\omega_e}{dt^2} + \frac{2\pi \cdot (\gamma_M + \frac{V_{0M}}{2\pi} \cdot \delta_M)}{V_{0M}} \cdot \frac{d\omega_e}{dt} \quad (4)$$

Taking into account the previous equations (3) and (4), and the equation (1), result the next expression

$$x \cdot \frac{V_{0P}}{2\iota \cdot x_{\max}} = A_2^* \cdot \frac{d^2\omega_e}{dt^2} + A_1^* \cdot \frac{d\omega_e}{dt} + A_0^* \cdot \omega_e \quad (5)$$

We apply the Laplace transform to equation (5), and after the terms sorting, result the transfer function of the execution branch

$$G(S) = \frac{\omega_e(S)}{X(S)} = \frac{\frac{V_{0P}}{2\pi \cdot x_{\max}} \cdot \omega_i}{A_2^* \cdot S^2 + A_1^* \cdot S + A_0^*} \quad (6)$$

or

$$G(S) = \frac{\frac{V_{0P}}{2\pi \cdot x_{\max} \cdot A_0^*} \cdot \omega_i}{\frac{1}{\omega_{HS}^2} \cdot S^2 + \frac{2\xi_H}{\omega_{HS}} \cdot S + 1} \quad (7)$$

where ω_{HS}-eigen throb of the executio branch and ξ_H-damping coeficient of the execution branch.

4. The Calculus of the Command Branch Transfer Function – H(S)

The command branch is composed by the proportional distribuitor D, as a part of a command regulator, and the levers system SC for servo-tracing.

The system have a pump for a command circuit, with low flow value and with the maximumm pressure value at 3,5 MPa. The same pump realize both the compensation funtion of a volumic losses an the execution branch. In Figure 4 is presented the command circuit part.

Like the previous case, for wring the mathematical model the authors use the flow equations and the movement equation of the regulator piston.

The flow equation is given by the next expresision

$$Q_D = Q_{reg} \quad (8)$$

where: Q_D - the distributor flow; Q_{reg} - the agent flow from working room of the regulator.

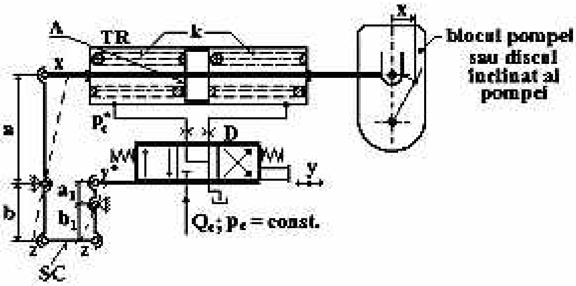

Figure 4. Servo-driving system

Writing the separate expression for Q_D and Q_{reg} and using equation (8), result.

$$p_C^* = p_C + \frac{\Delta_1}{\Delta_2} \cdot (y - y^*) - \frac{A}{\Delta_2} \cdot \frac{dx}{dt} \quad (9)$$

The movement equation of the regulator piston is

$$m_{TR} \cdot \frac{d^2x}{dt^2} + \gamma_R \cdot \frac{dx}{dt} + k \cdot x = A \cdot p_C^* - F_{SM} + F_0 \quad (10)$$

where: m_{TR} - the regulator piston mass, included the reduce mass at the piston rod for the pump block, spings and SC command levers system; k - the spring rigidity; A - the aria of working surface of the regulator piston; F_{SM} - resistent force of servo-distribuitor movement; F_0 - the initial compression force in the springs.

With equations (9) and (10), and with expression of F_{SM} force we can write

$$\begin{aligned}&(m_{TR} + m_C) \cdot \frac{d^2x}{dt^2} + (\gamma_R + \gamma_C + \frac{A}{\Delta_2}) \cdot \frac{dx}{dt} + k \cdot x = \\ &A \cdot p_C - F_0 + A \cdot \frac{\Delta_1}{\Delta_2} \cdot (y - y^*)\end{aligned} \quad (11)$$

or, in stationary state

$$\begin{aligned}&(m_{TR} + m_C) \cdot \frac{d^2x}{dt^2} + (\gamma_R + \gamma_C + \frac{A}{\Delta_2}) \cdot \frac{dx}{dt} + \\ &(k + K_C \cdot A \cdot \frac{\Delta_1}{\Delta_2}) \cdot x = A \cdot \frac{\Delta_1}{\Delta_2} \cdot y\end{aligned} \quad (12)$$

If we apply the Laplace transform for the equation (12), result the transfer function of a command branch

$$H(S) = \frac{\frac{A}{C_0^*} \cdot \frac{\Delta_1}{\Delta_2}}{\frac{1}{\omega_C^2} \cdot S^2 + \frac{2\xi_C}{\omega_C} \cdot S + 1} \qquad (13)$$

5. The Global Transfer Function for S.H.R.A

Taking into account the equation (7) and (13) it can be obtain the global transfer function for S.H.R.A., by the form

$$R(S) = H(S) \cdot G(S) = \frac{X(S)}{Y(S)} \cdot \frac{\omega_e(S)}{X(S)} = \frac{\omega_e(S)}{Y(S)} \qquad (14)$$

or, after replaces and processing the expresion

$$R(S) = \frac{\omega_e(S)}{Y(S)} = \frac{K_{SC} \cdot \omega_C^2}{S^2 + 2\xi_C \cdot \omega_C \cdot S + 1} \cdot \frac{K_{SE} \cdot \omega_{HS}^2}{S^2 + 2\xi_H \cdot \omega_{HS} \cdot S + 1} \qquad (15)$$

6. Conclusions

The mathematical model elaborated in this paper affer the apportunity to determine the expresions which describe the dynamic behaviour of the command branch and the execution branch of the system. In this work is given the analitial expressions for the eighen values and for the damping coefficient for automatic system components and the transfer function both for the components and for the entire system.

The analitial expressions offer a way for a formal analysis about the behaviour of the system components and about the way in which theses influences the dynamic process of the system.

For a concrete case of a hydraulic driving system, when the specific coefficients could be determinate exactly, it could be realize the analysis of the influences factors about the dynamic stability, through the root place method.

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