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RESEARCHES REGARDING FUNCTIONAL STABILIZATION OF ELECTRO-HYDRAULIC SERVO-SYSTEMS

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Abstract: *The objective of this paper is to present a comprehensive application of using the electro-hydraulic servo-systems in acting advance kinematics-chains of the tool-machines. The work presents the construction, functionality and dynamic behaviour of the advance-system by using the transfer function. It is determined the kinematics system's stability when the variable is the inertial mass. This comparative research based on three methods is based on the Bode diagrams, Nyguist criteria, and Hurwitz criteria.*

Key Words*: servo-system, electro-hydraulic, modelling, transfer, functions.*

1. GENERAL PROBLEMS OF DYNAMICS AND COMMAND

In general, electro-hydraulic servo-systems are made by placing the electric directly on the hydraulic motor, thus resulting modules having small volumes in the rooms of the motor and in the coupling pipes, being characterized by high static and dynamic performances.

The structure of the hydraulic servo-systems must consider the requirements of the tool-machine, regarding the dynamics and of the basic characteristics of automatic systems: to be stable, rapid and precise.

An essential influence on the qualities in dynamic regime of the whole system is the own frequency of the hydro-motor, because in every servo-system this assembly has the lowest own frequency, thus imposing the minimum reaction time of the whole system, and implicitly its precision level [1].

1.1 Electro-hydraulic servo-systems with integrated command

There are four constructive alternatives of electro-hydraulic servo-actuation, of which two have a high importance: the twisting moment electro-hydraulic amplifier and the linear electro-hydraulic amplifier. The latter will be referred as follows. Its functioning principle is transforming the revolving input movement in a translation output movement and amplifying the forces in the same time. These give the linear amplifier some advantages compared with the other alternatives.

The remembered servo-actuations are powered by small electric motors. Due to their force, rapidity, simplicity and precision these actuations are recommended especially where an increase of the capacity can’t be made with conventional actuations.

The small electric input powers will be transformed with the hydraulic energy in final big output powers, assuring between the input and output quantities well determined cinematic, static or dynamic rapports. This is obtained through one or more inverse intern or extern reaction couples [1].

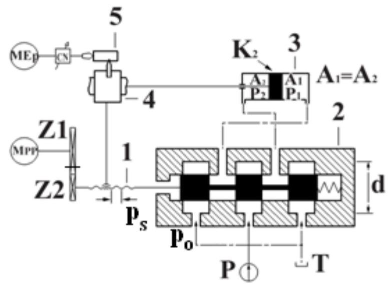

Fig. 1.The Considered Servo-system.

2. THE CONSTRUCTION AND FUNCTIONING OF THE CONSIDERED SERVO-SYSTEM

Figure 1 represents the structural scheme of a servo-system's movement for a lathing machine. The functioning of the servo-system is based on the rotating movement of the command screw 1 (having a pitch p_s) that will induce an axial movement of the spur gearing 2 that will cause a shifting of the valve from the neutral position and will open the runways toward the hydraulic motor 3. This will move the lever 4, processing the semi-product 5. The lever's movement (which bounds the lever 4 with the command screw 1) will turn the valve in the initial neutral position afterwards. Z_1/ Z_2 is the gear which carries the movement from the step-by-step motor M_{pp} to the command screw 1.

Through this mechanism it is realized a good amplification of power and a precise positioning of the tool, combining the precision of the step by step motor with the power provided by the hydraulic motor – distributor assembly.

In the following part this assembly will be presented from the point of view of automation, analyzing various aspects that can occur and can influence its functioning.

3. FUNCTIONAL ANALYSIS OF THE SERVO-SYSTEM

The general problem of the study is to determine the way in which the output argument varies in relation with

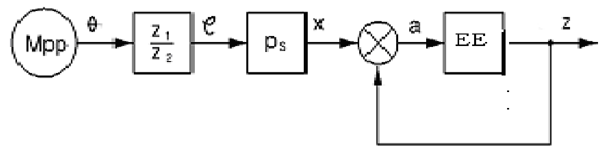

Fig. 2 The block scheme of the system from Fig. 1.

the exterior influences (command and perturbation arguments) and the characteristics of the functional elements of which the system is composed. This, of course, can be made experimentally on the system or on its functional model, or with theoretical means.

The linear models are described by transfer functions, and utilizing them allows the researcher to determine the answer to the frequency, which is a relatively complete test on the behaviour of the system, without being necessary to solve the differential equation.

The block scheme of the system from Fig. 1 is presented in Fig. 2, where *EE* is the execution element (the motor-distribution group), θ is the input value, Z_1/Z_2 gear, p_s pitch of the command screw, *Z* output measure, a action measure.

The inverse direction is unitary and it is made from the exit (the translation movement of the piston's core bar) to the end "a" of the lever 4. The transfer function of the execution element is presented in following.

4. THE APPOINTING OF THE LEVEL OF STABILITY OF THE SERVO-SYSTEM.

It is well known that to determine the transfer function of a servo-system, should be considered the flowing equation thru distribuitor:

$$Q = C_d \cdot S_C \cdot \sqrt{\frac{2}{p}(p_0 - p_1}\ , \qquad (1)$$

where: $S_c = \pi \cdot d \cdot a$,

or liniarizing: $Q = A_Q \cdot a$,

where: $A_Q = C_d \cdot \pi \cdot d \cdot \sqrt{\frac{2}{\rho}} \cdot p_0 -$ flow amplification,

$$Q = A\frac{dz}{dt} + K_2 \cdot (p_1 - p_2) + \frac{V_M}{2E} \cdot \frac{d(p_1 - p_2)}{d_t}\ ,$$

$$Q = A \cdot z + K_2 \cdot (p_1 - p_2) + \frac{V_M}{2E} \cdot \frac{d(p_1 - p_2)}{d_t}\ , \qquad (2)$$

motor flow equation, where: $\Delta p = p_1 - p_2$ and the medium volume is $V_M / 2$.

(Capacity of pump should assure the speed of throwing of piston, to prevent and support the losses and compressibility of the volume of liquid in motor)

- Condition of equilibrium of forces in motor. The work hypothesis $A_1 = A_2 = A$.

$$A \cdot \Delta p = M\frac{d^2 z}{dt^2} + K_1\frac{dz}{dt} + F_R = M \cdot \ddot{z} + K_1 \cdot \dot{z} + F_R$$

$$\Delta p = \frac{M}{A} \cdot \ddot{z} + \frac{K_1}{A} \cdot \dot{z} + \frac{F_R}{A} \qquad (3)$$

derivating obtain:

$$\Delta\dot{p} = \frac{M}{A} \cdot \dddot{z} + A \cdot \ddot{z} \qquad (4)$$

after substituting of Δp and $\Delta\dot{p}$ we obtain (5):

$$A_Q \cdot a = A \cdot z + K_2 \cdot \left(\frac{M}{A} \cdot z + \frac{K_1}{A} \cdot z + \frac{F_R}{A}\right) + \frac{V_M}{2 \cdot E} \cdot \left(\frac{M}{A} \cdot z + \frac{K_1}{A} \cdot z\right)$$

$$A_Q \cdot a = \frac{V_M}{2 \cdot E} \cdot \frac{M}{A} \cdot \dddot{z} + \left(K_2 \cdot \frac{M}{A} + \frac{K_1}{A} \cdot \frac{V_M}{2 \cdot E}\right) \cdot \ddot{z} + \left(A + \frac{K_1}{A} \cdot K_2\right) \cdot \dot{z}$$

- After Laplace transformation:

The transfer function of the direct circuit is:

$$T_d(s) = \frac{z(s)}{a(s)} \qquad (6)$$

The transfer function of the feedback circuit is:

$$T_{r(s)} = \frac{r(s)}{z(s)} = 1 \qquad (7)$$

and the transfer function of the closed circuit is:

$$T_0(s) = T_d(s)/[1 + T_d(s) * T_r(s)] \qquad (8)$$

- The transfer function of the execution element:

$$T_{EE}(s) \equiv T_d(s) \qquad (10)$$

$$\frac{z(s)}{a(s)} = \frac{A_Q}{\frac{V_M}{2 \cdot E} \cdot \frac{M}{A} \cdot s^3 + \left(K_2 \cdot \frac{M}{A} + \frac{K_1}{A} \cdot \frac{V_M}{2 \cdot E}\right) \cdot s^2 + \left(A + \frac{K_1}{A} \cdot K_2\right) \cdot s}$$

where:

$$B_1 = \frac{K_1 \cdot K_2}{A} + A \qquad B_2 = K_2 \cdot \frac{M}{A} + \frac{K_1}{A} \cdot \frac{V_M}{2 \cdot E}$$

$$B_3 = \frac{V_M}{2 \cdot E} \cdot \frac{M}{A}$$

Results:

$$T_{d(s)} \equiv T_{EE(s)} = \frac{A_Q}{B_3 \cdot s^3 + B_2 \cdot s^2 + B_1 \cdot s}\ . \qquad (11)$$

- The transfer function of the opened circuit

$$T_D(s) = T_d(s) \cdot T_r(s) = T_{EE}(s)\ ,\ T_r(s) = 1. \qquad (12)$$

- The transfer function of the closed circuit:

$$T_0(s) = \frac{z(s)}{\theta(s)} = \frac{A_Q \frac{Z_1}{Z_2} \cdot p_s}{B_3 \cdot s^3 + B_2 \cdot s^2 + B_1 \cdot s + A_Q}\ , \qquad (13)$$

$$\frac{Z_1}{Z_2} = 1 \text{ and } p_s = 1 \ \ cm\ ,$$

C_d = 0.6 the constant of the distribuitor;

D = 0.8 cm, the diameter of the valve;

p_0 =15 daN/cm^2 the pressure of the agent provided by the pump;

p_s = 1cm the pitch of the screw;

ρ = 0.86 × 10^{-6} daN s^3/cm, density of the hydraulic agent;

A = 32 cm^2, piston's surface;

K_1 = 1.2 daN·s/cm, viscous friction quotient;

K = 1.4 cm^3/daN·s^2, volume losses quotient;

M = 0.02 … 0.1 daN·s^2/cm, inertial mass to be moved;

V_M = A*curse (20 … 80 cm), linear hydraulic motor volume;

E = 1500 daN/cm^2, elasticity module of the hydraulic agent.

In the following part, various criteria are applied to verify the stability of the studied system:

5. DETERMINING THE STABILITY OF THE SERVO-SYSTEM CONSIDERING THE INERTIAL MASS AS VARIABLE.

Using the MathCAD program, we calculate the coefficients of the transfer function.

The five values of the inertial mass vector are:

$$M = \begin{pmatrix} 0.02 \\ 0.04 \\ 0.06 \\ 0.08 \\ 0.1 \end{pmatrix}.$$

With the above mention value the transfer functions coefficients are (B_0, B_1, B_2, B_3):

$$B_3 = \begin{pmatrix} 3.3333\times10^{-4} \\ 6.6667\times10^{-4} \\ 1\times10^{-3} \\ 1.3333\times10^{-3} \\ 1.6667\times10^{-3} \end{pmatrix} = \begin{pmatrix} 0{,}000333333 \\ 0.000666667 \\ 0.001000000 \\ 0.001333333 \\ 0.001666667 \end{pmatrix},$$

$$B_2 = K_2 \cdot \frac{M}{A} + \frac{K_1 \cdot V_M}{A \cdot 2 \cdot E}, \qquad (14)$$

$$B_2 = \begin{pmatrix} 2.0875\times10^{-2} \\ 2.175\times10^{-2} \\ 2.2625\times10^{-2} \\ 2.35\times10^{-2} \\ 2.4375\times10^{-2} \end{pmatrix} = \begin{pmatrix} 0.020875 \\ 0.021750 \\ 0.022625 \\ 0.023500 \\ 0.024375 \end{pmatrix},$$

$$B_1 = A + K_1 \cdot \frac{K_2}{A},\ B_1 = 32.0525\ ,$$

$$A_q = C_d \cdot \pi \cdot 0.8 \sqrt{\frac{2 \cdot p_0}{\rho}}\ , \qquad (15)$$

$$A_q = 1001.9707\ .$$

5.1.1. The Hurwitz Criteria.

For a closed system (MathCAD):

$$T_{0(s)} = \frac{A_q \dfrac{Z_1}{Z_2} \cdot p_s}{B_3 \cdot s^3 + B_2 \cdot s^2 + B_1 \cdot s + A_q}. \qquad (16)$$

$$A_q = B_0 = 1001.9707\ .$$

The stability condition according to the Hurwitz criteria is that all the determinants until $D_{n-1} > 0$, $n = 3$

$D_1 = B_2 > 0$

$$D_1 = \begin{pmatrix} 0.0209 \\ 0.0217 \\ 0.0226 \\ 0.0235 \\ 0.0244 \end{pmatrix} \text{positive,}$$

$$D_2 = B_2 \cdot B_1 - B_3 \cdot B_0 \text{ positive, stabile,}$$

$$D_2 = \begin{pmatrix} 0.3351 \\ 0.0292 \\ -0.2768 \\ -0.5827 \\ -0.8887 \end{pmatrix} \text{negative, instable.}$$

First two values are positive, stable and the last three negative so that the system is instable.

5.1.2. BODE characteristics.

They are:

a) amplitude-frequency characteristic

b) phase-frequency characteristic

The BODE characteristics give us information about the relative stability of the system through: module relay and phase relay.

One can notice that starting with the value 0.06 of the inertial mass, the module relay becomes positive, meaning that the system is instable [3].

5.1.3. Nyquist Criteria.

It analyses the transfer locus of the opened system TD ($j\omega$). According to this criteria, a linear automatic system is stabile if the transfer locus of the opened circuit TD ($j\omega$) routed from $\omega = -\infty$ to $\omega = +\infty$ wouldn't surround (hour sense), the critical point (–1, 0).

Considering the facts shown above, one can notice that starting with the value 0.06 of the inertial mass, the system becomes instable. This means that the spot where the transfer takes place surrounds the critical point (–1, 0) for the values of the inertial mass of 0.06; 0.08 and 0.1 daN·s^2/cm. [3]

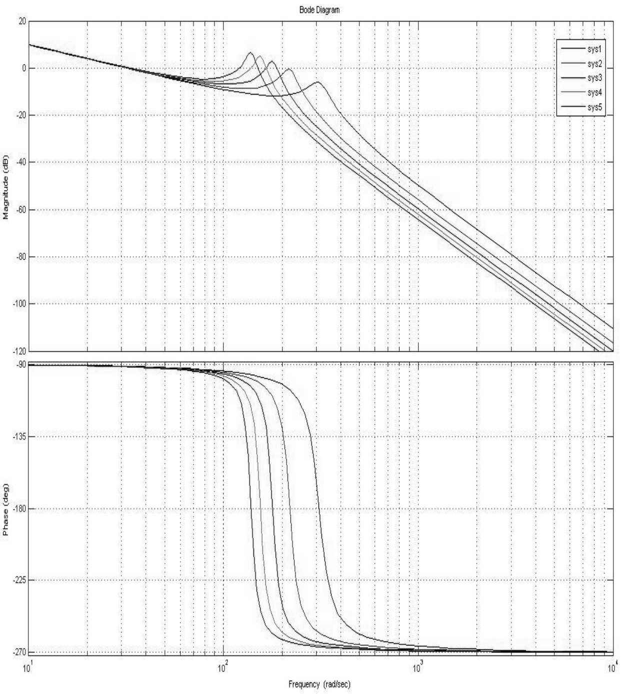

Fig. 3a and b. Bode characteristics.

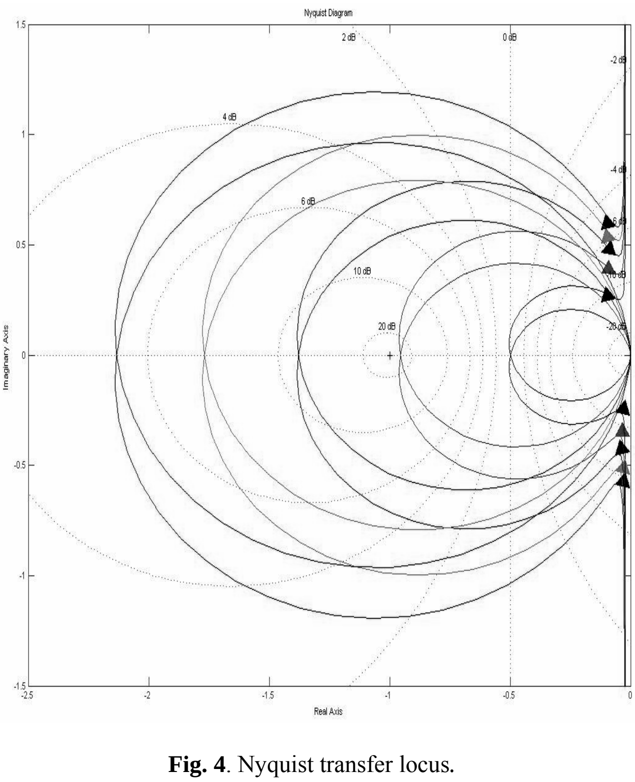

Fig. 4. Nyquist transfer locus.

6. CONCLUSIONS

1. The paper researched the behaviour of an electro-hydraulic servo-system in actuation feede kinematic chains of machine tools. The behaviour has been analysed from the point of view of the stability, through the presented methods.

The following steps were followed:

- drawing the block scheme according to the constructive sketch of the system;
- liniarizing and obtaining the transfer function by applying Laplace transformation;
- analyzing the stability of the servo-system by modifying a parameter in the presented situation, the inertial mass;
- using the algebraic Hurwitz criteria and frequency methods - Nyquist and Bode diagrams.

From the analysis of the stability of the presented servo-system, the same stability condition was concluded, respectively the inertial mass must be less than 0.04. The present analysis refers strictly to this parameter.

2. For the inertial mass value over 0.04 the system becomes instable.

3. The comparative dynamic analysis of the kinematic feed chain through the three presented methods leads us to the same conclusion mentioned at the first point.

4. These analysis methods are beneficial in the design phase. In this case it is possible to establish the variation field of the inertial mass in order to achieve a stable functioning of this system.

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