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MECHATRONIC SYSTEM USED FOR FLOW CONTROLLING OF HYDRAULIC PUMPS WITH AXIAL PISTONS

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Lucrarea isi propune sa scoata in evidenta performantele tehnice importante ale sistemului mecatronic proiectat in scopul reglarii debitului la pompele hidraulice cu pistonase axiale. Metoda de reglare este continua si se refera in principal la un sistem de pozitionare in circuit inchis, de precizie ridicata. Prin rezolvarea modelului matematic al intregului sistem, in conditii dinamice, am obtinut o deplasare teoretica a pistonului, care se incadreaza in gama de valori acceptata. Aceasta poate fi o metoda de imbunatatire a randamentului pompei hidraulice cu pistonase axiale si a sistemului actionat in acelasi timp.

The paper aims to emphasize the main technical performances of the mechatronic system designed in order to control the flow of hydraulic pump with axial pistons. The control method is a continuous one and refers to a very accurate positioning system with mechanical feed-back. By solving the mathematical model of the entire system with dynamic conditions, we have concluded that the theoretical piston displacement is inside an acceptable range. This could be a method for improving the efficiency of the hydraulic pump and of the actuated system too.

Keywords: Control system design, Feed-back systems, Electro-hydraulic systems

1. Introduction

Since the development of control technique was pointed out, more applications for hydraulic systems were found, due to the possibilities of adding to them the accurate positioning. In such a way, the values of increasing actuation forces were considered nearby the main features regarding the kinematical parameters. Even more, they may provide the continue control of these parameters by using the adequate electronics and software.

Taking into account the theoretical equations for the hydraulic pump/motor flow, we assume that if the rotational speed of the electric motor is constant, a variable geometric volume will provide a variable flow. Variable flow

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of the pump is required by the hydraulic equipments of the system, such as motors and other actuated elements. The control of flow, pressure and hydraulic power of volumetric pumps and motors is necessary whether some reasons are imposed: the speed has to be attended with minimum energy loss; the mechanical power of the electric motor should be usually constant and limited to certain values; the flow delivered by the pump should be decreased meantime an imposed value of pressure has been achieved, in order to avoid the thrown of fluid to the tank [1], [2].

Because of the reversibility between hydraulic pumps and motors, we may control the flow as well as the speed of the system. Moreover, hydraulic equipments could be added to the system, if we needed both movement directions.

The existing technical solution is based on the mechanism whose main feature is to tilt the piston block of the pump following an imposed mechanical law. Such a flow control unit may vary the piston block angle between [-25^o, $+25^o$] by using a special shape of a steel part that allow a single point contact with the connecting rod of the pump. This steel part, which has an oscillatory movement around a grounded point, is fully fastened to the piston of the hydraulic cylinder that is a component of the control system. Consequently, the force equilibrium is the only way for feed-back position of the piston and the control process is a discrete one, which could be a disadvantage [5].

In order to control the hydraulic flow of the pump with axial pistons, we have designed a genuine mechatronic system including a hydraulic cylinder, a feed-back unit comprising a screw-nut mechanism fastened into the piston of this cylinder, a hydraulic distributor and an electric stepping motor. It is the simplest way to take the advantages of hydraulic system, characterized by high values for mechanical force and hydraulic power and the electric system as well, with the possibility of signal controlling and adaptability for feed-back.

The important characteristic of such a system is the continuous control of hydraulic parameters using electric input, which may be electric tension. Finally, this control process may influence in the same way the values of eccentricity, referring to the pumps with radial pistons, or the slope angle of the block with axial pistons as component of such a hydraulic pump. Consequently, the proposed technical solution could be applied for both types of hydraulic pumps.

Meantime, we have designed the feed-back electro-hydraulic system for controlling the variable pump flow, without modifying the shape of the existing steel part mentioned before, because of adaptability required by the previous technical solution for the hydraulic pump assembly.

Generally speaking, it is a feed-back positioning system with continuous linear or angular displacement control providing great accuracy based on electro-hydraulic components.

2. Technical Approach

The main components of the technical solution of the proposed system are described in Fig. 1 and it could be applied for axial pistons hydraulic pumps. It includes the electric stepping motor 1, the hydraulic distributor 2 and the subassembly for mechanical feed-back 3 joined together with the piston of the hydraulic cylinder 4. The piston is fastened with the steel part 5, whose linear movement outside the cylinder brings about the oscillatory movement of the block with axial pistons of the pump, which is not shown in the picture.

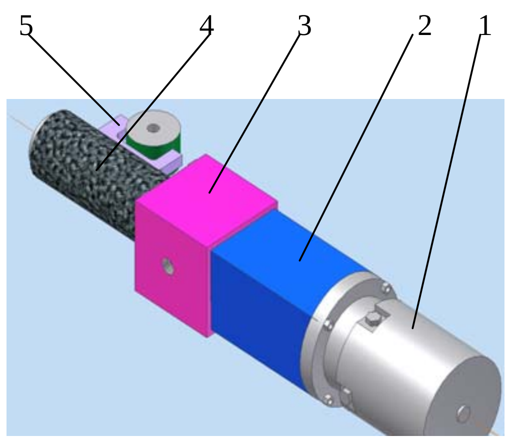

Fig. 1. The main components of the technical solution.

The cylinder piston CH (Fig. 2.) has an active surface on its left side and a very well known pressure value is acting on permanently.

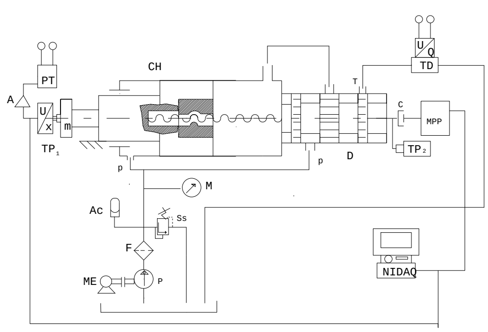

Fig. 2. The schematic of the proposed mechatronic system

This pressure force will cause the piston movement on the right direction, whose value depends on the controlled parameters as it is explained below.

The mechanical feed-back comprising the screw-nut mechanism is fitted with the piston and has the task of controlling the movement of the distributor slider D (Fig. 2.). Theoretically speaking, the screw position has to be attended by imposition of the known step number ordered to the electrical stepper motor MPP (Fig. 2.). By making the comparison between the effective piston displacement and the theoretically one, the screw will provide a linear movement of the distributor slider. Consequently, one hydraulic circuitry will be opened and it could afford the pressure value acting on the right side of the piston. Because the two active surfaces of the piston are in a 0.5 range, the pressure forces acting over them will cause the left direction displacement of the screw fitted with the slider and its 'zero' position will be attended.

The movement of the slider following the right direction will be done just in case the real displacement of the piston will be lower than the theoretical one. Finally, the slider movement provides the supplier pressure acting on the right side of the piston and the hydraulic circuit will be closed.

The main idea of this electro-hydraulic mechatronic system is based on its 'zero' position attended after each controlled movement. It means that the theoretical displacement will be equal to the real one, after each ordered number of step for the electric motor. The speed is controlled by the input impulse frequency to the electrical stepping motor and the flow inside the hydraulic system, so that the speed nearby zero point will be very low.

In case of inertia forces increasing, the piston displacement will be greater than the theoretical one, so the screw-nut feed-back mechanism will continue the left direction displacement of the slider.

3. The Mathematical Model of the Mechatronic System

The dynamic mathematical model of this proposed mechatronic system implies the computation of the main resistant pressure force acting inside the axial piston block of the pump. It was the reason why we have studied the maximum pressure variation along the period when one piston is passing through the damp area at the beginning of the supplier phase, for instance. The same functional parameters appear during the end of this phase.

The hydraulic pump with axial pistons provides the working pressure of the hydraulic system by rotational movement of the sloping block around its axis in front of the pump distributor. During this process, the axial pistons are connected alternatively to the supply pressure of the system as well as to the tank pressure. Almost all functional instances are working with positive covered distributor of the pump, so we may conclude that some transitory working phases could appear during the piston going by from the aspiration to the rejected oil zone.

Taking into account the particular elliptical shape of the working area as well as the damp area at starting and final point, we have computed the pressure variation during an entire displacement of one piston in front of the distributor slit. We have studied the geometrical parameters of the distributor in front of the piston block of the pump, in order to compute the functional areas as there are shown in Fig. 2.

Considering the notations of Fig. 2, we have computed the damp area $A_{r1}(t)$ and the elliptical working area $A_{r2}(t)$ [3], [4]:

$$A_{r1}(t) = 0.5 \cdot \begin{bmatrix} 1 & R \cdot \cos(\rho_{23}(t)) & R \cdot \sin(\rho_{23}(t)) \\ 1 & R_m \cdot \cos(\rho_{24}(t)) & R_m \cdot \sin(\rho_{24}(t)) \\ 1 & R_M \cdot \cos(\rho_{24}(t)) & R_M \cdot \sin(\rho_{24}(t)) \end{bmatrix} \quad (1)$$

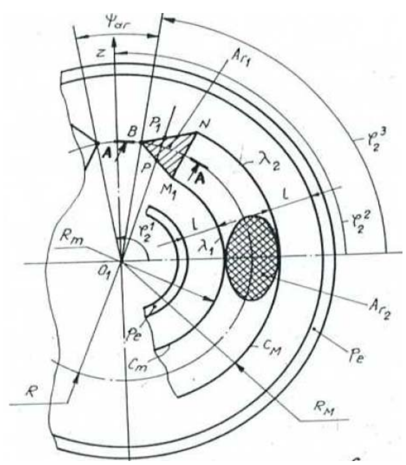

Fig. 3. The main areas for computing.

where $\rho_{24}(t)$ is the positioning angle for the line PP_1 computed for the time value t. The final position is when the line PP_1 is coincident with the line MN.

$$A_{r2}(t) = \pi \cdot \left(\frac{d_g}{2}\right)^2 \cdot \cos(\alpha) \quad (2)$$

where d_g is the diameter of the sloped hole through the supply pressure circuit and α the slope value of this hole.

In order to compute the pressure variation from the aspiration phase to repression we may use the following mathematical equation:

$$\frac{dp}{dt} = E \cdot \frac{v(t) \cdot A_p - \alpha_D \cdot A_r \cdot \sqrt{\frac{2}{\varphi} \cdot \Delta p} - Q_p}{V_0 + s(t) \cdot A_p} \tag{3}$$

where: E [N/m^2]-the elasticity modulus of the hydraulic oil; v(t) [m/s] and s(t) [m] are the speed and displacement of the piston as time function; A_p [m^2] the piston area; φ [kg/m^3] the oil density; Δp [N/m^2] the pressure variation; Q_p [l/s] the flow loss; V_o [m^3] is the initial volume inside the hydraulic cylinder of the block. This pressure variation will cause the pressure force as the main resistant force of the system.

The dynamic working process of the system is described by a mathematical model comprising seven first order differential equations grouped in a mathematical system, which may be solved with Runge-Kutta numerical method.

This mathematical model [3] is written below taking into account the mathematical model of the electrical stepping motor, the dynamic movement equation for the mobile subassembly (comprising the rod of the electric motor, the coupling subassembly and the screw as part of the feed-back system) and the equation describing the flow continuity (Spanu, 1999):

$$\begin{bmatrix} \frac{di_a}{dt} \\ \frac{di_b}{dt} \end{bmatrix} = \begin{bmatrix} L_{aa} & L_{ab} \\ L_{ab} & L_{bb} \end{bmatrix}^{-1} \cdot \begin{bmatrix} U_a - R \cdot I_a \\ U_b - R \cdot I_b \end{bmatrix} - \begin{bmatrix} L_{aa} & L_{ab} \\ L_{ab} & L_{bb} \end{bmatrix}^{-1} \cdot \begin{bmatrix} \frac{\partial L_{aa}}{\partial \theta_m} & \frac{\partial L_{ab}}{\partial \theta_m} \\ \frac{\partial L_{ab}}{\partial \theta_m} & \frac{\partial L_{bb}}{\partial \theta_m} \end{bmatrix} \cdot \begin{bmatrix} i_a \\ i_b \end{bmatrix} \cdot \frac{\partial \theta_m}{\partial t} \cdot \tag{4}$$

$$\frac{d\omega}{dt} = \frac{1}{2 \cdot J_r} \cdot \left(i_a{}^2 \cdot \frac{\partial L_{aa}}{\partial \theta_m} + i_b{}^2 \cdot \frac{\partial L_{bb}}{\partial \theta_m}\right) + i_a \cdot i_b \cdot \frac{\partial L_{ab}}{\partial \theta_m} \cdot \frac{1}{J_r} - - \frac{M_r}{J_r} - \frac{D_r}{J_r} \cdot \frac{d\theta_m}{dt} \tag{5}$$

$$\frac{d\theta_m}{dt} = \omega \tag{6}$$

$$\frac{dy}{dt} = v_p \tag{7}$$

$$\frac{d^2y}{dt^2} = \frac{1}{m_p} \cdot (F_b - D_{ro} \cdot \frac{dy}{dt} + A_2 \cdot p_o - A_1 \cdot p_1) \qquad (8)$$

$$\frac{dp_1}{dt} = (C_d \cdot \pi \cdot D_s \cdot c \cdot \sqrt{\frac{2}{\varphi} \cdot p_o} - A_1 \cdot \frac{dy}{dt} - a_m \cdot (p_1 - p_o)) / (V^1{}_0 + y \cdot A_1) / E) \qquad (9)$$

There were used the following notations: i_a, i_b [A]-the current for the two electrical phases of stepping motor; L_{aa}, L_{bb}, L_{ab} [H]–own and mutual inductance of electrical phases; U_a, U_b [V]–voltage of both supplied phases; R_a, R_b [Ω]-electric resistance of both supplied phases; θ_m [deg]-mechanical angle of the stepping motor rod; ω [rad/s]-rotational speed of the electric motor rod; $J_r[Kg \cdot m^2]$- equivalent inertia constant of the mechanical system; M_r [Nm]-the resistant torque acting in the system; D_r, D_{ro} [Kg.m/s]-the coefficients of viscous friction; y [m]-the piston displacement; v_p [m/s]-the piston speed; m_p [Kg]-the mass of the piston; F_b [N]-the pressure force developed by the block with axial piston of the hydraulic pump; A_1, A_2 [m^2]-the arrays of active surfaces of the hydraulic piston; C_d [-]-the flow coefficient; D_s [m]-the diameter of the piston; c [m]-the displacement of the distributor slider; p_0, p_1 [N/m^2]-the pressure on both sides of the piston; φ[Kg/m^3]-the oil density; the coefficient a_m is given considering: $a_m = 2.5 \cdot \frac{\pi \cdot D_p \cdot j^3}{12 \cdot \eta \cdot l}$; D_p [m]-the piston diameter; j [m]-the distance between the grounded part and the slider; η [Ns/m^2]-dynamic viscosity of the hydraulic oil; E [N/m^2]-the elasticity modulus of the hydraulic oil.

4. Numerical Setup

The electric stepping motor provides a mechanical step of 1.8^o and a torque of 0.2 [Nm]. A number of 600 steps are enough for the hydraulic piston displacement, because of the working frequency limitation. The screw has a coarse pitch of 6 [mm], which means 18 [mm] the maximum displacement of the piston. The screw-nut mechanism is placed inside the hydraulic cylinder piston and has to provide a very accurate position by means of the hydraulic flow control of the distributor slider as we mentioned above.

Considering the technical characteristics of the axial piston pump with seven axial piston and the rotational speed of the electric motor about 2700 [rot/min], by using the mathematical equations described above, we may compute the variations of pressure along the main specific working areas, as direct dependency on time and initial pressure values at the beginning of the movement in front of the pump block distributor (Fig. 4). The speed of one pump piston in front of the block distributor slit is decreasing during the pressure variation as it is shown in Fig. 5.

Analyzing these figures, we may infer the very short period of time - 6.5 [ms] for pressure supplier provided by this hydraulic pump. Moreover, these pressure values will be the main cause of continuous variation of the resistant force required for sloping the axial piston block of the pump. This resistant force will be taken into account for the mathematical model of the electro-hydraulic controller.

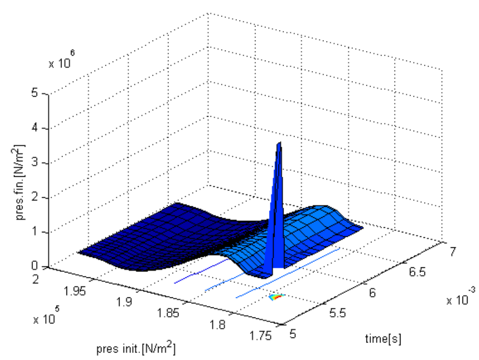

Fig. 4. The pressure variation along the area as function of initial pressure values and displacement time

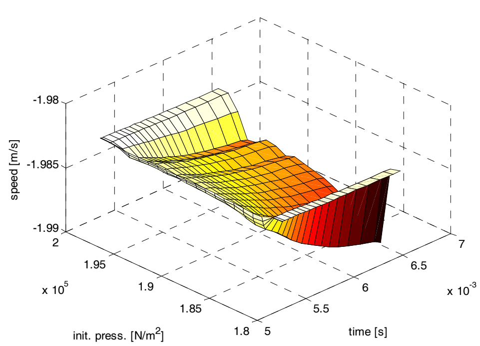

Fig. 5. The speed variation as function of time and initial pressure values.

In Fig. 6 it is presented the pressure variation as time dependency for the period of completing the process of pressure supplier.

Analyzing the Fig. 7, we conclude that the piston displacement is positive, meaning from right side to the left, at the beginning of the period. After the distributor opens the hydraulic circuit, the force equilibrium will be modified, so the piston will have an opposite side displacement.

The mechatronic control system designed with mechanical feed-back has achieved the imposed accuracy of positioning.

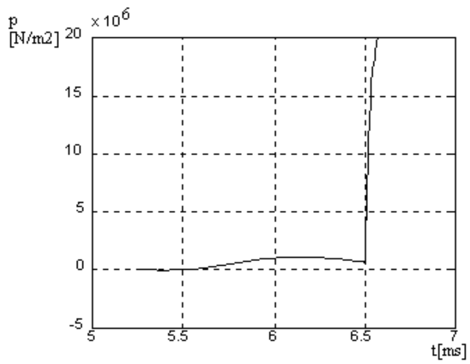

Fig. 6. The pressure variation along the supply working area and the tank pressure circuit.

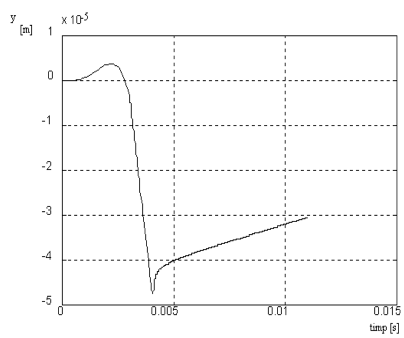

Fig. 7. The piston movement y[m] during the period of control process.

which is a continuous one. The technical solution is very simply and could afford some dimensional changes in order to adapt it in a broad sense.

5. Conclusions

This type of mechatronic system as a hydraulic flow controller one might be directly used to the hydraulic pump with axial pistons, without any other dimensional changes. The system described above could be used to the hydraulic motors and for other positioning systems with high accuracy too. It is a mechatronic solution for controlling the efficiency of hydraulic pumps, motors and systems.

In order to compute the dynamic mathematical model of this controller, the resistant pressure force is the main perturbation, which has to be very well completed. We have concluded its variation during the piston pump passing in front of the block distributor slit.

The dynamic mathematical model implying the electrical stepper motor, the distributor and the cylinder with its screw-nut mechanism reveals the technical parameters for the movement of the cylinder piston. Due to the very short period of time required for this displacement, the dynamic stability of the entire system has to be study as future work.

As future improvements, the brushless electric motor is the best way for accomplishing the functional requirements, because of its higher torque and command frequency, but the effective cost of the components is higher.

R E F E R E N C E S

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