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DYNAMICS OF THE SYSTEM FOR FEED KINEMATICAL CHAIN HYDROSTATIC DRIVING

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ABSTRACT

Transitory phenomena affect the machine tools with hydrostatic driving functioning, during their exploitation. Finding out the laws that give the dependence between the driving system hydraulic parameters variation and the machine tool operational parameters values is the target of the studies concerning these transitory phenomena. The longitudinal feed kinematical chain of the plain grinding machine was analyzed for this purpose. The machine tool structural scheme was considered in relation to the transfer functions specific to each component. The longitudinal feed slay dynamical driving system characteristics were also determined. This way, the premises for establishing the interdependence between the hydrostatic system functional parameters were created, aiming to reduce the transitory phenomena negative influence onto both machined surface quality and machine tool productivity.

KEYWORDS: hydrostatic driving, dynamical system, transitory phenomenon, transfer function

1. Introduction

The hydrostatic driving had as main purpose, ever since its first utilizations in machine building, to develop the force needed by a machine tool execution element, at the moment required by the technological cycle. This type of driving fundamental characteristic is defined, functionally speaking, by the absence of cause – effect interdependence. This process can be considered as determined only if the operational characteristics in stationary regime are known.

The hydrostatic driving systems with automatical regulation are widely spreading in the recent period. In their case, it is necessary to know the operating characteristics in both stationary and dynamic regime: motion stability, positioning precision, transitory regime quality etc. The hydrostatic automatically driving systems are structures which enable a functional dependence between the output value of a controlled parameter (displacement speed, working force, transitory regime quality etc.) and its input value.

The equipment dynamical regime takes place during the evolution time interval between two successive periods of stationary regime and it is determined by input value modification or by perturbation. The investigation of a hydrostatical driving system dynamical behaviour has as purpose the transitory dynamics properties study and the process parameters establishment. The dynamical regime analysis is based on introducing in system typified inputs followed by the output signal values analysis. This depends on the driving system structure and on the applied signal type.

The researches concerning the transitory phenomena from hydrostatical driving systems aim to find the laws that are governing the dependence between the system hydraulic parameters variation and the output functional parameters of the driven machine element. The research also tries to diminish the transitory process negative influence on machined surfaces quality and on productivity.

2. Driving system structure

The hydrostatical feed system of a plain grinding machine (Fig. 1) is characterized, from functional point of view, by several energetic transforms.

The electric motor, 8, transforms the electrical power into mechanical power. At this stage we have the first conversion (C_1). The hydrostatical generator (pump 7) transforms, during the next stage (C_2 conversion), the mechanical power into pressure potential energy which is then received by the regulation & control devices, transformed according to the equipment functioning cycle and finally transmitted to the hydraulic motor (4). The last one realizes the hydrostatic energy conversion into mechanical power (C_3), by driving the plain grinding machine slay feed motion.

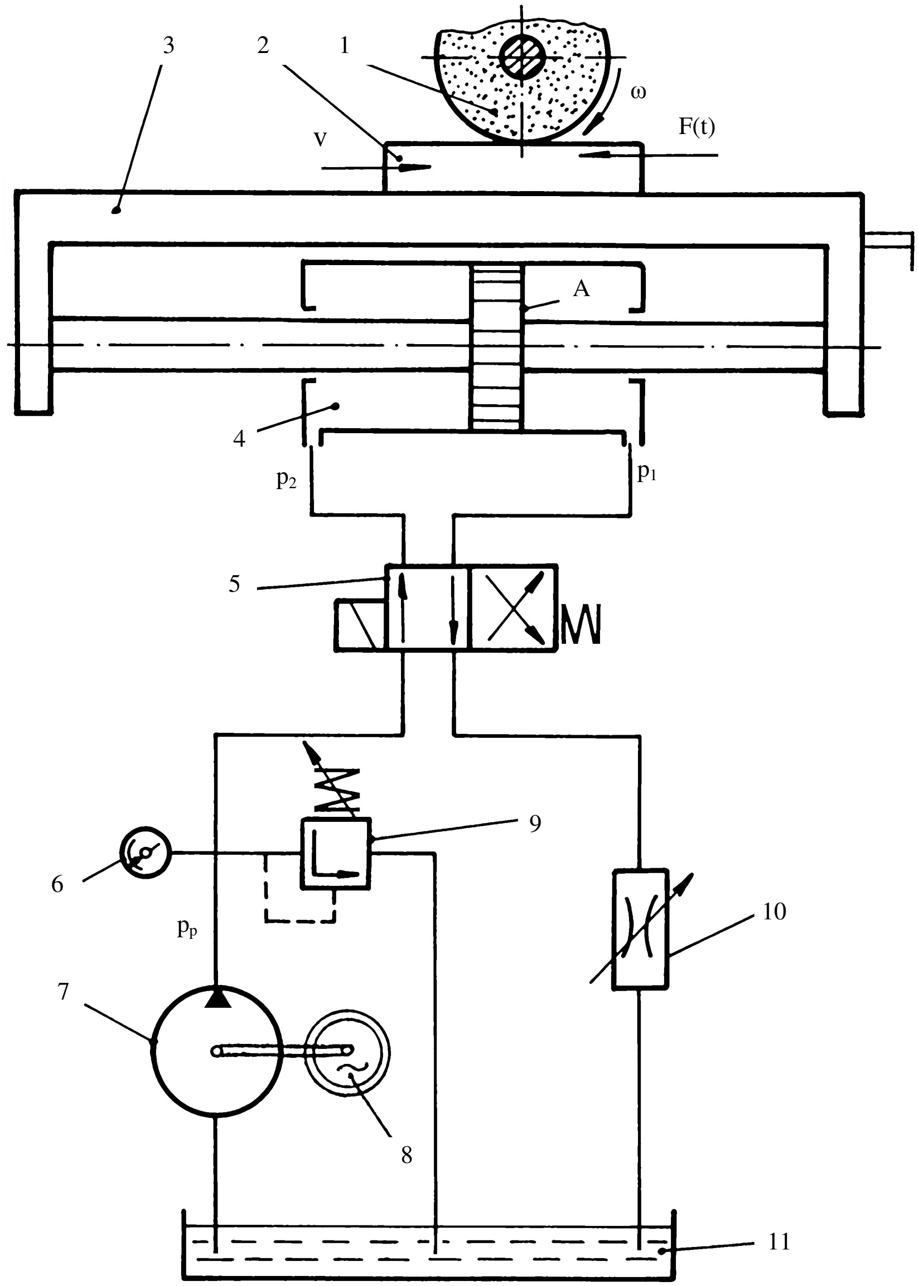

Fig. 1. *Plain grinding machine hydrostatical feed system structure*:
1 – grinding wheel; 2 – worked piece; 3 –longitudinal feed slay; 4 – hydraulic motor; 5 – distributor; 6 – manometer; 7 – pump; 8 –electric motor; 9 – valve; 10 – flow regulator; 11 – tank..

Both systems, mechanical and hydraulic, which the plain grinding machine feed hydrostatical system consists of, are presented in Fig. 2. The mechanical system (detailed in Fig. 2b) is influenced by the cutting regime parameters, by the characteristics of the friction processes between the machine tool surfaces having a relative motion and by worked piece – machining system – tool system elastic deformations. According to Fig. 2c, the hydrostatical system is influenced by the processes taking place in the pump, in the regulation & control devices, in the hydraulic motor and in the hydraulic environment.

By knowing the equipment flowchart and its transfer function, its dynamical system analysis consists of applying the stability criteria from the automatic regulation theory. The system component elements transfer functions have to be determined for this purpose.

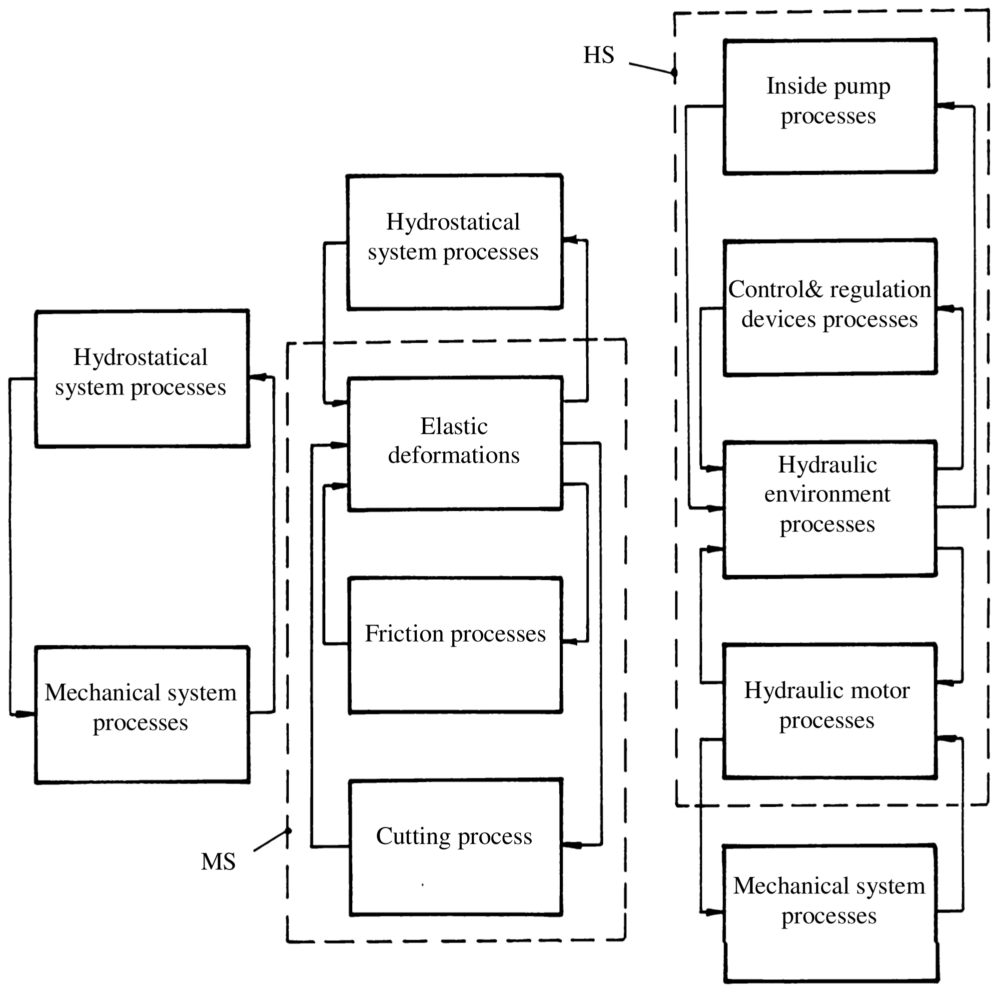

Fig. 2. *Plain grinding machine hydrostatic feed system flowchart*

3. System transfer functions

The linear hydraulic motors dynamical characteristic is determined by considering the two compartments (active and counter-pressure) as being connected in parallel. In this case, the transfer function expression is [1]

$$W = -\frac{\sigma}{\mu} = -\left(\frac{1}{Rd} + \frac{d}{T_1 s + W_1} + \frac{R}{T_2 s - W_2}\right), \qquad (1)$$

where: $\sigma = F/F_0$ means the driving force relative variation;

$\mu = v/v_0$, the speed relative variation;

d – the piston active surfaces ratio;

$R = p_{20}/p_{10}$, the ratio of the pressures from the two hydraulic motor compartments;

$$T_1 = \frac{L_1 \cdot p_{10}}{E \cdot v_0}, \quad T_2 = \frac{L_2 \cdot p_{20}}{E \cdot v_0},$$

the fluid compressibility time constants, corresponding to the two hydraulic motor compartments;

E – the hydraulic agent elasticity module;

L_1, L_2 – the length of the two liquid columns from the motor;

W_1, W_2 – the transfer functions of the devices from the motor compartments circuits, respectively.

The variables with “0” index correspond to the stable functioning regime.

The hydraulic agent continuity equations are:

$$Q_{M1} - Q_1 = Q_{\Delta 2}, \tag{2}$$

$$Q_2 - Q_{M2} = Q_{\Delta 1}, \tag{3}$$

where: Q_1 and Q_2 are the hydraulic motor incoming / out coming debits;

$Q_M = Av$, the debit of the fluid handled by the motor.

Q_Δ, the volumetric deformation of the fluid from the motor, due to its compressibility, can be calculated by using Jukovski energetic equation of deformation:

$$\Delta\dot{v} = \frac{du}{d(\Delta p)} = \frac{d\left(\frac{A \cdot L}{2E} \cdot \Delta p^2\right)}{d(\Delta p)} = \frac{A \cdot L}{E} \cdot \Delta p \tag{4}$$

and by using Castilliano relation:

$$Q_\Delta = \frac{dv}{dt} = \frac{A \cdot L}{E} \cdot \dot{p}. \tag{5}$$

After developing in Taylor series and by keeping only the first degree terms, relations (2) and (3) become:

$$A \cdot \Delta v - \left(\frac{\partial Q_1}{\partial p_1}\right)_0 \cdot \Delta p_1 = \frac{A \cdot L_1}{E} \cdot \Delta\dot{p}_1, \tag{6}$$

$$\left(\frac{\partial Q_2}{\partial p_2}\right)_0 \cdot \Delta p_2 - A \cdot \Delta v = \frac{A \cdot L_2}{E} \cdot \Delta\dot{p}_2. \tag{7}$$

If the following notations are used:

$$\mu = \frac{\Delta v}{v_0},\ \tau_1 = \frac{\Delta p_1}{p_{10}},\ \tau_2 = \frac{\Delta p_2}{p_{20}},\ \eta_1 = \frac{\Delta Q_1}{Q_{10}},$$
$$\eta_2 = \frac{\Delta Q_2}{Q_{20}},$$

The (6) and (7) equations system will look like

$$\begin{cases} A \cdot v_0 \cdot \mu - Q_{10} \cdot \eta_1 = \dfrac{A \cdot L_1}{E} \cdot p_{10} \cdot \dot{\tau}_1 \\ Q_{20} \cdot \eta_2 - A \cdot v_0 \cdot \mu = \dfrac{A \cdot L_2}{E} p_{20} \cdot \dot{\tau}_2. \end{cases} \tag{8}$$

Because $Q_{10} = Q_{20} = Av_0$, the system (8) becomes

$$\begin{cases} \mu - \eta_1 = \dfrac{L_1 \cdot p_{10}}{E \cdot v_0} \cdot \dot{\tau}_1, \\ \eta_2 - \mu = \dfrac{L_2 \cdot p_{20}}{E \cdot v_0} \cdot \dot{\tau}_2, \end{cases} \tag{9}$$

or, by using the Laplace transform,

$$\begin{cases} \mu - \tau_1 \cdot W_1(s) = T_1 \cdot s \cdot \tau_1, \\ \tau_2 \cdot W_2(s) - \mu = T_2 \cdot s \cdot \tau_2. \end{cases} \tag{10}$$

The dependence between the pressure relative variation (τ) and the flow one (η),

$$W(s) = \frac{\eta(s)}{\tau(s)}, \tag{11}$$

respectively,

$$\begin{cases} \eta_1(s) = \tau_1(s) \cdot W_1(s), \\ \eta_2(s) = \tau_2(s) \cdot W_2(s), \end{cases} \tag{12}$$

is influenced by the control & regulation devices (included in the hydraulic motor compartments circuits) dynamical characteristics. As consequence, the system (10) will take the form

$$\begin{cases} \mu - \tau_1 \cdot W_1(s) = T_1 \cdot s \cdot \tau_1, \\ \tau_2 \cdot W_2(s) - \mu = T_2 \cdot s \cdot \tau_2. \end{cases} \tag{13}$$

The driving system proposed to be analyzed has, on the outgoing pipe, a flow regulation device, without speed stabilisation (10). If we assume that the exterior pressure, at flow regulation device exit, is equal to the atmospheric one, than its transfer function can be found starting from the hydraulic agent equation of flowing through the equipment and has the form

$$W_1(s) = \frac{\eta_1(s)}{\tau_1(s)} = n_1, \tag{14}$$

where n_1 means the flow equation exponent in the case of the flow regulation device ($0{,}5 \le n_1 \le 1$).

An overflow valve (9) is attached to the pipe connecting the pump to the hydraulic motor. The valve transfer function can be obtained starting from the valve plunger dynamics equation and from the fluid which passes through continuity equation, after using a Taylor series development:

$$W_2(s) = \frac{\eta_2(s)}{\tau_2(s)} = = -\frac{\left[n_2\left(T_2'^2 \cdot s^2 + T_2''s + 1\right) + K_1\right] \cdot K_2}{T_2''^2 s^2 + T_2''s + 1}, \tag{15}$$

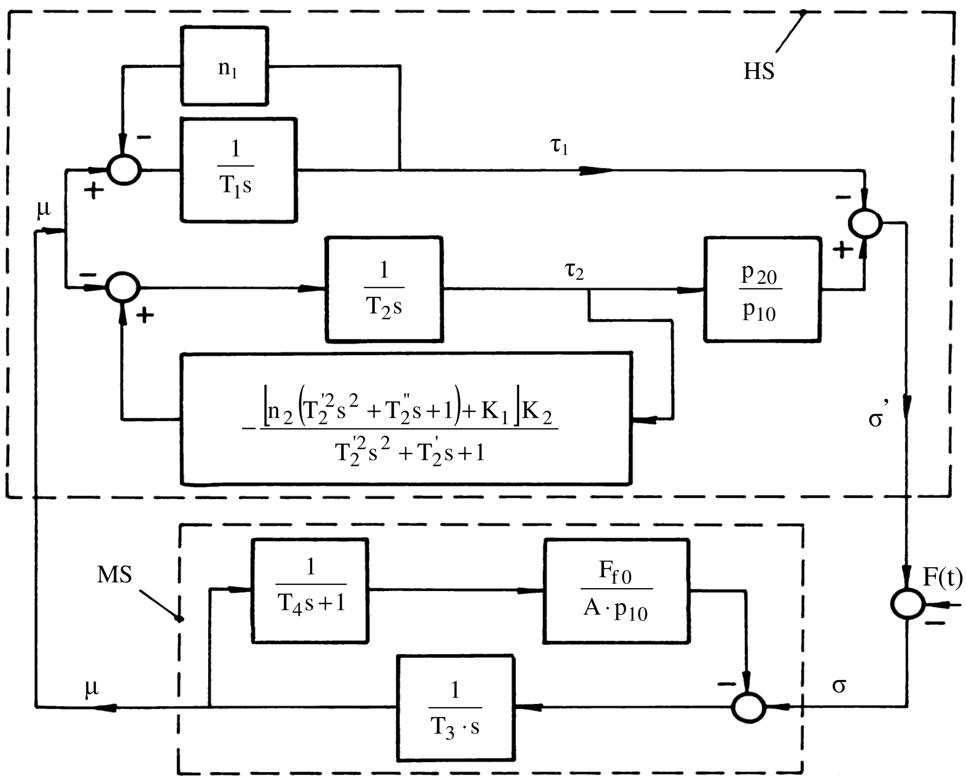

Fig. 3. *Feed slay – hydrostatical driving dynamical system flowchart*

where: $n_2 = 0{,}5$ means the flow equation exponent in the case of the valve;

$T_2^{'} = \frac{m}{k}$ is the valve plunger time constant;

m – the plunger mass;

k – the valve spring elasticity constant;

$K_1 = \frac{p_{p0} \cdot a}{k \cdot h_0}$ defines the valve amplifying coefficient;

a – the valve plunger effective working surface;

h_0 – the plunger stroke corresponding to the fluid flow $Q_{s0} = Q_{p0} - Q_{20} = Q_{p0} - Av_0$ going through the valve;

$K_2 = \frac{Q_{s0}}{Q_{20}}$, where the flow Q_{s0} represents the difference between the flow delivered by the pump and the flow received by the hydraulic motor.

If the Dynamics fundamental equation is applied to the longitudinal feed slay (3) motion, by involving the parameters: relative speed (μ) and relative driving force (σ), the next relation results:

$$\left[\frac{1}{W_3(s)} + K_3 W_4(s)\right] \cdot \mu = \sigma, \tag{16}$$

where the following notations were made:

$$W_3(s) = \frac{1}{T_3 \cdot s} = \frac{A \cdot p_{10}}{s \cdot M \cdot v_0} \tag{17}$$

means the slay transfer function, from inertial reasons (in the absence of friction forces);

$$W_4(s) = \frac{F_{f0}}{v} = \frac{1}{T_4 \cdot s + 1}, \tag{18}$$

the slay transfer function, if friction forces are taken into account;

$$\sigma = \tau_2 \cdot K_4 - \tau_1 - F(t); \tag{19}$$

$$K_3 = \frac{F_{f0}}{A \cdot p_{10}}; \tag{20}$$

$$K_4 = \frac{p_{20}}{p_{10}}. \tag{21}$$

By F_{f0} was denoted the friction force at the slay slides level, in stationary regime.

After calculus developing, the transfer functions of the hydrostatical driving system and of the mechanical system result as:

$$W_{MS}(s) = \frac{W_3(s)}{1 + K_3 \cdot W_3(s) \cdot W_4(s)} \tag{22}$$

respectively

$$W_{HS}(s) = \frac{\frac{1}{T_1 \cdot s}}{1 + \frac{W_1}{T_1 s}} + \frac{\frac{1}{T_2 s}}{1 - \frac{W_2}{T_2 s}} K_4 . \tag{23}$$

The longitudinal feed slay – hydrostatical driving dynamical system flowchart, if the volumetric losses are neglected, found by using the relations (13), (17), (18), (20), (21), (22), (23), is depicted in Fig. 3.

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Sistemul dinamic de acţionare hidrostatică al unui lanţ cinematic de avans

—Rezumat—

În timpul exploatării, funcţionarea maşinilor-unelte acţionate hidrostatic este influenţată de fenomene tranzitorii. Studiind fenomenele tranzitorii specifice sistemelor de acţionare se urmăreşte determinarea legilor după care variaţia parametrilor hidraulici ai instalaţiei influenţează valoarea parametrilor funcţionali ai maşinii-unelte. În acest scop, s-a analizat lanţul cinematic de avans longitudinal al maşinii de rectificat plan având în vedere atât schema structurală cât şi funcţiile de transfer specifice fiecărei componente. De asemenea, s-au determinat caracteristicile sistemului dinamic de acţionare a săniei de avans longitudinal. Astfel, s-au creat premizele stabilirii unei interdependenţe între parametrii funcţionali ai sistemului hidrostatic în scopul atenuării influenţei negative a fenomenelor tranzitorii asupra calităţii suprafeţelor prelucrate şi a productivităţii maşinii-unelte.