TEKA Kom. Mot. Energ. Roln. – OL PAN, 2009, 9, 267–276

SELECTION OF DISCRETELY ADJUSTABLE PUMP PARAMETRES FOR HYDRAULIC DRIVES OF MOBILE EQUIPMENT

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Summary. The structural-functional scheme of rotary hydrodrives with discrete regulating of pump displacement on the basis of the simplified mathematical model which allows dynamic processes calculating and parameters optimizing at various operational modes is constructed.

Key words: hydrodrive, hydraulic pump, regulating, control variables, mathematical model.

INTRODUCTION

Upgrading of mobile equipment is realized by development, production and mastering of energetically saturated, economically effective, easy in control, comfortable, convenient in usage repair and service methods. A majority of modern mobile equipment: road-building, handling, agricultural etc. is equipped with hydrodrives which are used both in transmissions and in working equipment drives. It is due to the fact that power units of hydraulic drives (HD) have small overall dimensions, weight and capability of parameters regulating at high values of efficiency and reliability.

The technological level of modern HD is determined by power units - pumps and hydraulic motors parameters and features. In mobile equipment HD axial piston hydromachines (APH), both pumps and hydromotors with reducing transmissions or reduction gearboxes are of the most frequent use. Hydromachines (HM) of such a type have small overall dimensions and weight, admitting operation at high pressures (up to 60 … 70 MPa) and rotation rate (5000 rpm and higher), have high efficiency and long life cycle and also the capability of parameters regulating in a large range of operational modes changes - rotation rate and (or) the moment on the output shaft. For the last decades there was an obvious tendency of mobile equipment leading manufacturers refusal from application of non-regulated power units in HD. Thus two variants of adjustable HM use [4] are possible: an adjustable pump and a non-regulated hydraulic motor or an adjustable pump and an adjustable motor. The best variant is the second scheme which is actively introduced the last years. In spite of cost increase, application of adjustable APH instead of non-regulated one ensures the regulating range extension, reduction of pump displacement and increase of its rotation rate up to an actuating motor shaft rotation rate without intermediate mechanical transmissions, due to which overall dimensions and drive weight are reduced and its reliability is increased in the whole.

From an analysis of operational modes of HD mobile equipment it has been established that for many of them there is no acute necessity of parameters permanent regulating [5]. In most cases for HD it is enough to provide two rates of the output shaft: operating and transport (low and high rate of movement).Therefore for an alteration of HD operational parameters application of power hydraulic units with discrete regulating of displacement is possible. It should be noted that at discrete regulating of power units extension of operational parameters range in comparison with non-regulated aggregates is also ensured. In comparison with HM permanent regulating efficiency, an increase is ensured due to transient processes time periods reduction at parameters regulating and weight, overall dimensions and aggregates cost decrease as well, at the whole system's reliability increases.

The discrete change of power units operating capacity leads to instant increase or reduction of hydraulic energy quantity which is transformed into mechanical energy at the hydraulic drive output. HD operational changes lead to dynamic processes which are expressed in pressure fluctuations in the pressure hydraulic line, moment and rotation rate of the output shaft.

The analysis of works which are devoted to HD dynamics has proved that problems of power units operational parameters discrete regulating are not revealed enough. Therefore the actual task is the development of HD mathematical model with power units parameters discrete regulating and its dynamics investigation. Thus selection of pumps and HM regulating parameters is of significant practical importance and investigation and a definition of HD optimal operational parameters is an important scientific task.

Creation of modern hydraulic mobile equipment and its upgrading should be based on a technological level of foreign engineering, on capabilities of Ukrainian manufacturers of hydraulic equipment and use of modern simulation methods and designing at development of such systems as well[6].

This article is devoted to an investigation of HD dynamics at discrete regulating of pump displacement.

TASK DEFINITION

HD discrete regulating can be executed by step-by-step change of pump displacement V_0^p or (and) the motor capacity V_0^m from initial value to values of displacement volumes V_1^p, V_1^m accordingly. An increase (decrease) of pump (motor) displacement leads to HD working element rate enhancement, and their reverse change - to rate reduction. Change of motor and pump displacement occurs in time periods – t_0^p and t_0^m, accordingly. The pump and motor displacement, time periods of their change are considered as control variables of the task.

The discrete change of displacement occurs against the background of external drag torque M_c on HD output shaft. It's supposed that the external moment can either be increased or decreased or remains constant. For optimization of HD operation it is necessary to consider various combinations of control variables states (discrete regulating with increase, decrease or permanent displacement volume) and external moment. It is necessary to develop proposals on optimal values of control variables by criterion of HD stable operational basic parameters (pressure in a pressure pipe, HM rotation rate) during a transient period.

As a proof of HD main parameters stabilization possibility at discrete regulating in this article the case of abrupt increase of pump displacement is considered at progressive drag torque on HM shaft. In Fig. 1 the circuit diagram of HD is shown. At mathematical description of this drive the following assumptions are admitted:

- pressure in HM drain line and in pump suction line are equal to zero as they are essentially lower than pressure in the pressure hydraulic line at the pump outlet;
- operational liquid characteristics: volumetric module of elasticity, density, viscosity are constant and equal to average values;
- wave processes in pipelines are not considered, as their length is not significant;
- effect on transient processes of regulating and control equipment is not considered;
- pressure fluctuations and flow rate during one revolt of HM shaft are not considered.

In the article transient period at pump constant rotation rate and closed safety-valve $p_k > p_H\,(t)$ is considered.

CALCULATIONS

For mathematical simulation of transient processes in HD (Fig. 1), at an assumption of wave processes lack, the following equations [1,3] are used.

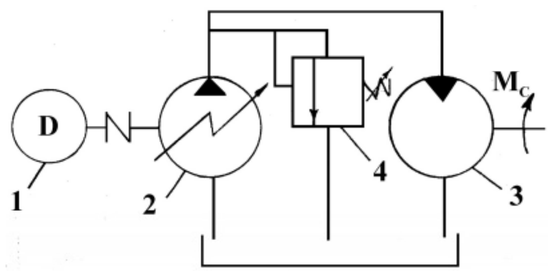

Fig. 1. Mathematical simulation of transient processes in HD

Pump flow rate q_H is expressed as:

$$q_{p1} = q_{pg} - q_{pl} - q_{pd}, \tag{1}$$

where $q_{pg} = \frac{V_0^p}{2\pi}\cdot w_p$ - pump geometrical head with displacement V_0^p, which operates at rate of w_p, leaks are not considered; $q_{pl} = Ñ_{pl}\cdot\left(p_p - p_s\right)$ - overflows in the pump; $q_{pl} = \frac{V_{p1}}{E}\cdot\frac{dp_p}{dt}$ - deformational flow rate in the pump outlet cavity - $V_{p1} = \frac{1}{2}\cdot\left(V_0^p + V_{0m}^p\right)$, connected with liquid compressibility effect at pressure in pressure line - p_p. $Ñ_{pl}$ - pump overflow coefficient; p_s - pressure in the drain line; V_{0m}^p- motor "dead" displacement volume.

HM flow rate q_{m1} is set by the formula:

$$q_{m1} = q_{mg} + q_{ml} + q_{md}, \tag{2}$$

where $q_{mg} = \frac{V_0^m}{2\pi}\cdot w_m$ - motor geometrical flow rate by displacement volume V_0^m at rotation rate w_m; $q_{ml} = C_{ml}\cdot\left(p_p - p_2\right)$ - motor overflows; $q_{md} = \frac{V_{m1}}{E}\cdot\frac{dp_p}{dt}$ - deformational flow rate in the motor working cavity with displacement volume $V_{m1} = \frac{1}{2} \cdot \left(V_0^m + V_{0m}^m \right)$. C_{ml} - motor overflow coefficient; - motor "dead" V_{0m}^m displacement volume.

Equation of continuity:

$$q_{p1} = q_{p1}. \tag{3}$$

Equation of moments:

$$M_{dc} - M_{lf} - M \; = J \frac{dw_m}{dt}, \tag{4}$$

where: $M_d = \frac{V_0^m}{2\pi} \cdot \left(p_p - p_2 \right) \cdot \eta_m \cdot \eta_g$ - HM torque with mechanical efficiency η_m and hydrodynamic efficiency η_g; M_c - external drag torque on HM shaft; $M_{lf} = \beta \cdot w_m$ - liquid friction moment with coefficient β; J - total moment of inertia reduced to the shaft.

Substituting (1) and (2) in (3) and considering (4) we will get the system which consists of two differential equations of the first order.

$$\begin{cases} \frac{dp_p}{dt} = a_0 + a_1 p_p + a_2 w_m \\ \frac{dw_m}{dt} = b_0 + b_1 p_p + b_2 w_m + b_3 M_c \end{cases}, \tag{5}$$

where:

$a_0 = \frac{E}{V_{m1} + V_{p1}} \left(\frac{1}{2\pi} V_0^p w_m + C_{pl} p_s + C_{ml} p_2 - Q_k \right)$; $a_1 = -\frac{E}{V_{m1} + V_{p1}} \left(C_{pl} + C_{ml} \right)$;

$a_2 = -\frac{E \cdot V_0^m}{\left(V_{m1} + V_{p1} \right) \cdot 2\pi}$;

$b_0 = -\frac{V_0^m}{2\pi} \frac{p_2 \cdot \eta_m \cdot \eta_g}{J}$; $b_1 = \frac{V_0^m}{2\pi} \cdot \frac{\eta_m \cdot \eta_g}{J}$; $b_2 = -\frac{\beta}{J}$; $b_3 = -\frac{1}{J}$.

Transient simulation in HD was implemented at following parameters values:

$E = 1500\, MPa$; $V_0^p = 80\, cm^3$; $V_0^m = 3500\, cm^3$; $V_{0m}^p = 8\, cm^3$; $V_{0m}^m = 300\, cm^3$; $w_p = 157\, s^{-1}$;

$C_{pl} = 10 \frac{cm^3}{s \cdot MPa}$; $C_{ml} = 20 \frac{cm^3}{s \cdot MPa}$; $p_2 = 5\, MPa$; $\beta = 100\, N \cdot m \cdot s$.

Initial values of integration variables - $p_p(0) = 0.1 MPa$; $w_m(0) = 0 s^{-1}$.

It was supposed that external drag torque M_c on HM shaft, measured in [$N \cdot m$] had been changing in due course by the following law:

$$M_c = \begin{cases} 10000 \cdot t, & t \le 0.1 s \\ 1000, & t > 0.1 s \end{cases}. \tag{6}$$

Hydraulic drive operation step-by-step regulating was supposed to be executed by means of abrupt change of pump initial displacement V_0^p up to value V_1^p = 80cm^3 in time period - t_0^p. Values of control variables V_0^p and t_0^p were supposed to be defined from considerations of pressure fluctuations p_p minimization in the system pressure line at transient process.

Pressure-time chart $p_p(t)$ at fixed pump displacement - (V_0^p = 80cm^3; t_0^p = 0) is shown in Fig. 2 (the dashed curve). For construction of objective function dependence $p_p(t)$ was smoothed by the method of moving average (Fig. 2. - the curve from discrete points). The values of smoothed function were calculated as arithmetic mean of near points:

$$\overline{p}_p = \frac{1}{2n+1}\sum_{i=k-n}^{k+n} p_p, \tag{7}$$

Where n - parameter of averaging which was selected as equal to n = 41.

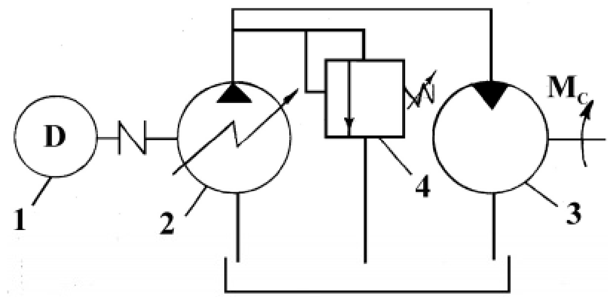

Fig. 2. The block curve

Dependence $\overline{p}_p(t)$ was approximated by polynomial of the sixth degree (Fig. 2. – the block curve):

$\overline{p}_p(t) = -1.19\cdot 10^7\cdot t^6 + 7.642\cdot 10^6\cdot t^5 - 1.898\cdot 10^6\cdot t^4 + 2.291\cdot 10^5\cdot t^3 - 1.39\cdot 10^4\cdot t^2 + 404.5\cdot t + 2.087$

The difference was considered as objective function F, for minimization of pressure fluctuations $p_p(t)$:

$$F(V_0^p, t_0^p) = \sum_{t=0.002}^{0.095}\left|p_p(t) - \overline{p}_p(t)\right| \to \min. \tag{8}$$

At constraints: $20 \le V_0^p \le 80; \quad 0 < t_0^p < 0.1$.

The objective function chart at various values of control variables is shown in Fig. 3. The objective function (in the indicated limits) is the absolute minimum in point $V_0^p = 27cm^3; \quad t_0^p = 0.016s.$ Therefore, the discrete change of pump displacement from V_0^p = 27cm^3 to maximum V_1^p = 80cm^3 at time period t_0^p = 0,016s allows pressure fluctuations minimizing p_p.

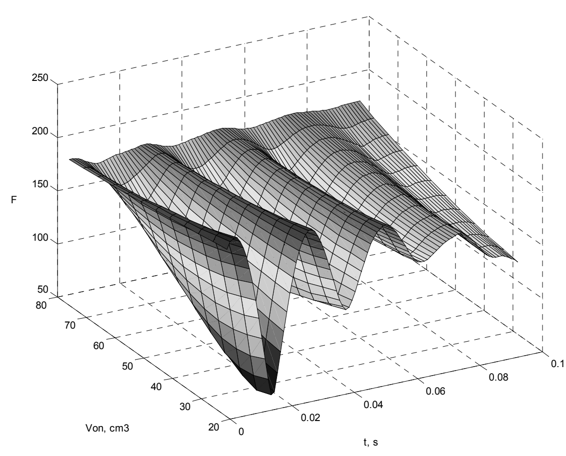

Fig. 3. The objective function chart at various values of control variables

For verification of results obtained by program Simulink (appendix of MatLab) [2], on the basis of differential equations system (5), HP simulated model was created (Fig. 4). The model allows defining of pressure amplitude-time parameters in the pressure line, motor rotation rate, liquid flow rate etc. at various values of control variables and types of external loads.

The scheme of simulated model consists of separate blocks in which values of hydrodrive constant parameters are set and values of derivative variables $\frac{dp_p}{dt}$ and $\frac{dw_m}{dt}$ at each time step of simulation process are calculated. Values of derivatives are input in the block of integrator which calculates values of variable parameters p_p and w_m for the given step. The determined values of variables are used for the finding of derivatives $\frac{dp_p}{dt}$ and $\frac{dw_m}{dt}$ during the next step.

Numerical integration of system (5) was conducted by means of MatLab ode113 built in solver, based on Adams - Basehor - Milton method [7]. The variable step of integration with average length - Δt = 0,0015s was used. The integration time interval - T = 0,2s.

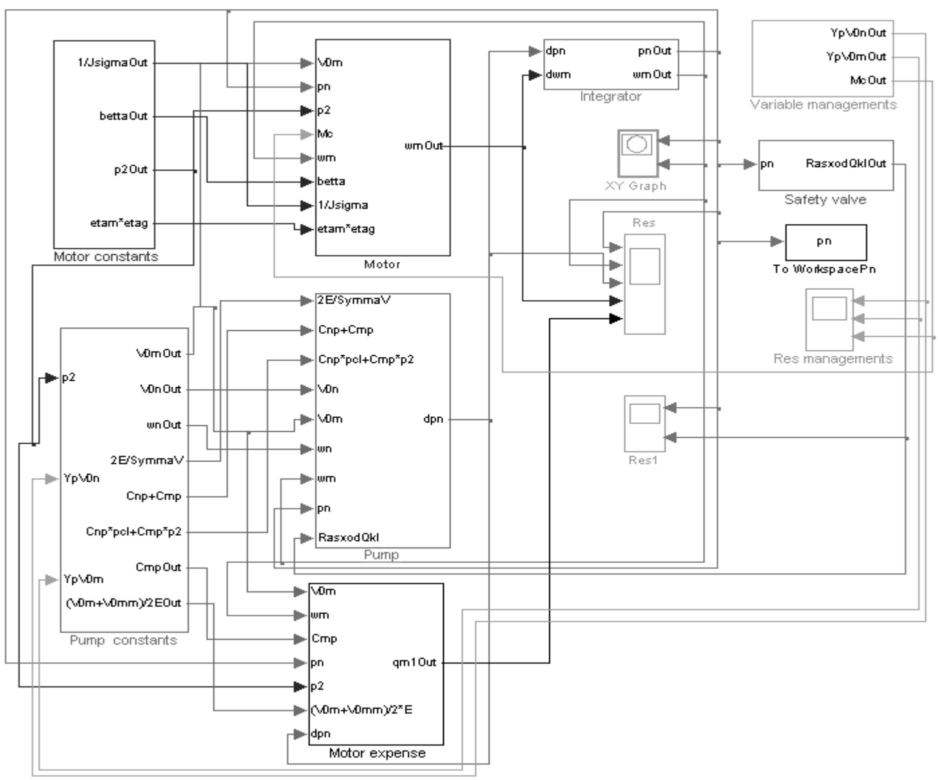

Fig. 4. The internal structure of blocks

In the chart of Fig. 4 “Motor constants” and “Motor” blocks simulate the second equation of system (5). At the “Motor” block output we get the derivative value $\frac{dw_m}{dt}$ of HM rotation rate. The internal structure of blocks is shown in Fig. 5.

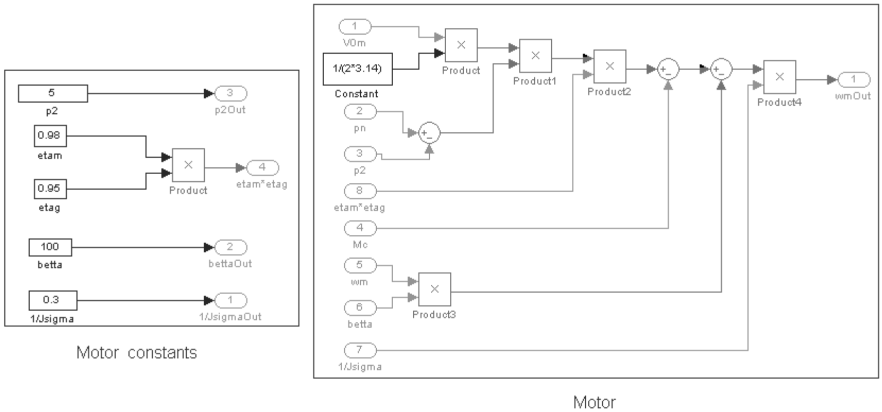

Fig. 5. The internal structure of blocks

The first equation of system (5) simulates «Pump constants » and «Pump» (Fig. 6). At «Pump» block output we get the value of pressure derivative - $\frac{dp_p}{dt}$. Derivative of HM pressure and rotation rate are input into «Integrator» block and then we get values of pressure p_p and rotation speed w_m.

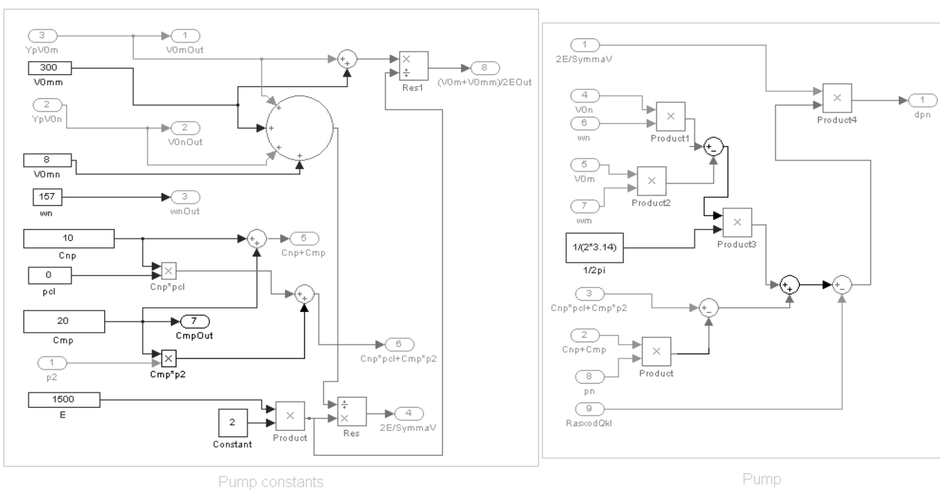

Fig. 6. “Pump constants” and “Pump”

«Variable managements» block, the internal structure of which is shown in (Fig. 7.), sets control parameters V^p, V^m, t_0^p dynamics and external moment M_c in graphic form as well.

The module «Signal Builder» used in the block allows investigating of HD response to arbitrary laws of control variables and external load changes. In Fig. 7 the situation of motor displacement discrete regulating at linear increment of external moment on HM shaft is simulated.

In Fig. 8 and 9 results of numerical simulation of dynamics $\overline{p}_p(t)$ are given at values of control parameters: V_0^p = 80cm^3; t_0^p = 0s (Fig. 8.) and V_0^p = 27cm^3; t_0^p = 0,016s (Fig. 9.).

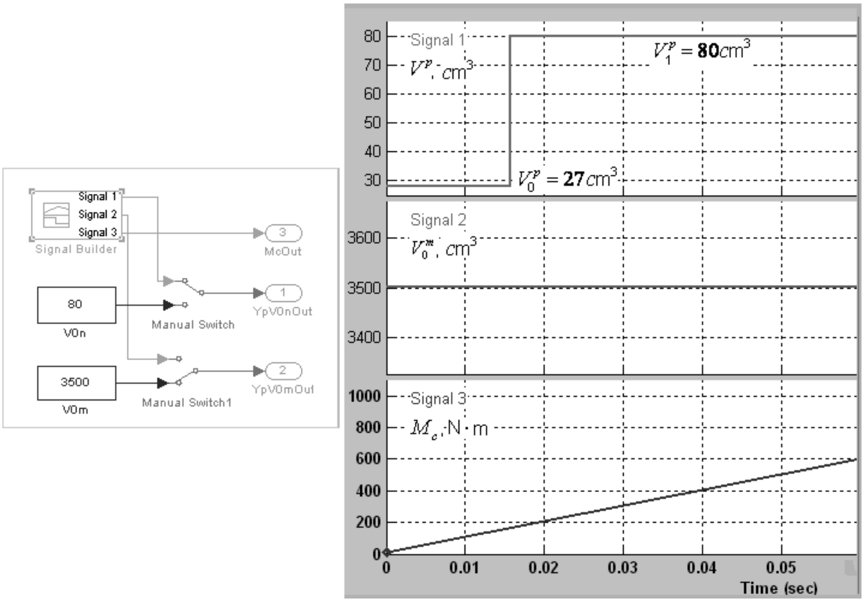

Fig. 7. The internal structure of “variable managements” block

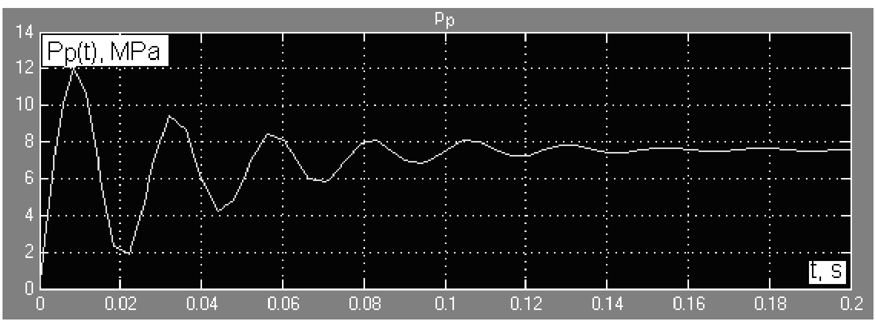

Fig. 8. Results of numerical simulation of dynamics $\overline{p}_p(t)$: V_0^p = 80cm^3; t_0^p = 0s

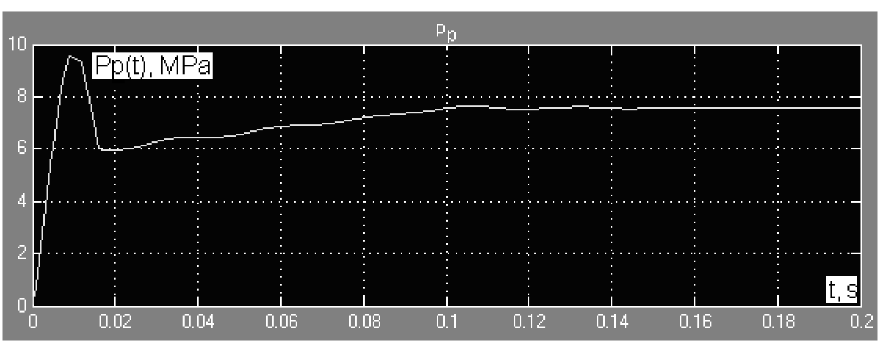

Fig. 9. Results of numerical simulation of dynamics $\overline{p}_p(t)$: V_0^p = 27cm^3; t_0^p = 0,016s

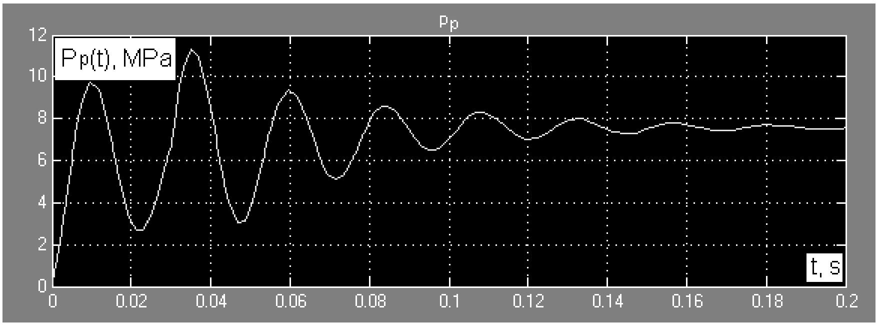

Fig. 10. Results of dynamics calculation $\overline{p}_p(t)$ are given at values of control variables - V_0^p = 80cm^3 and t_0^p = 0,03s

Comparison of pressure increase dynamics in HD pressure line in Figs. 7 and 8 allows for a conclusion about possibility of transient processes stabilizations in the system under consideration by means of pump displacement discrete regulating.

At an attempt to change control variables values arbitrarily, the stability of transient process abruptly becomes worse. For comparison, in Fig. 10, results of dynamics calculation $\overline{p}_p(t)$ are given at values of control variables - V_0^p = 80cm^3 and t_0^p = 0,03s.

CONCLUSION

1. Structural- functional scheme of rotary hydraulic drive with discrete regulating of pump displacement on the basis of a simplified mathematical model which allows calculating of dynamic processes at various operational modes is constructed.

2. It is established that value of displacement change coefficient at discrete regulating essentially affects the dynamic processes in HD with discrete regulating.

3. The algorithm of transient process optimization at discrete regulating is offered. As the objective function the sum of absolute values of pressure parameters deviations from pressure average value in system pressure line is considered.

4. The example of HD dynamic process calculation with discrete regulating at the mode of pump displacement increase at linearly incremental load on HM output shaft is considered. Displacement increase coefficient $\frac{V_1^p}{V_0^p} = 2.96$ and time of switching t_0^p = 0,016s values are analytically defined, at which the reduction of pressure fluctuations amplitude of the first peak in 1.3 times and practically complete damping of the subsequent peaks is provided. Non-observance of optimal switching conditions can lead to increase of pressure fluctuation amplitude by time.

5. The results of calculations are verified by numerical modeling at HD simulation model developed in the MATLAB environment.

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WYBÓR DYSKRETNIE REGULOWANYCH PARAMETRÓW DLA POMP W NAPĘDACH HYDROSTATYCZNYCH SPRZĘTU RUCHOMEGO

Streszczenie. Przedstawiono strukturalno-funkcjonalny schemat obrotowych napędów hydrostatycznych z regulacją przemieszczenia pompy na podstawie uproszczonego modelu matematycznego. Pozwala on na optymalną kalkulację parametrów procesów dynamicznych w zróżnicowanych warunkach operacyjnych.

Słowa kluczowe: napęd hydrostatyczny, pompa hydrauliczna, regulacja, zmienne kontrolne, model matematyczny.