ГАЛУЗЕВЕ МАШИНОБУДУВАННЯ

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INVESTIGATION OF DYNAMIC LOADING OF TRACTORS WITH VOLUMETRIC HYDRAULIC DRIVE

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Summary

The processes that occur in the transmissions of tractor units and self-propelled agricultural machines under various modes of movement and in the process of regulation are characterized by complex dependencies that are studied analytically or experimentally. Various methods are known for obtaining mathematical models. One of them is the classical direct description method. Another is the use of passive and active methods of regression analysis. It is rational to use both methods, the combination of which makes it possible to obtain the necessary mathematical model.

Volumetric hydraulic drive (OGP) is increasingly used in transmissions of modern tractors and self-propelled agricultural machines. The presented article discusses the mathematical description of axial piston hydraulic machines.

A study of transient processes was carried out and their assessment was carried out to determine the loads arising in the transmission of the machine with a stepwise change in the load, the gear ratio of the OGP and constant fuel supply when the unit is accelerated from a standstill. The unit acceleration mode was studied while driving on plowing and during transport operations for the following parameters and such initial conditions: rotation speed of the hydraulic motor shaft and engine shaft; torque on the motor shaft; hook load; pressure in the OGP pressure line. The dynamic characteristics of hydraulic machines, fluid leaks and its elastic properties, as well as variable values of the hydraulic drive efficiency are taken into account. The simulation results are compared with experimental studies. The objects of research were: a model of a caterpillar tractor T-150E with independent full-flow OGP of the left and right sides; mock-up of a wheeled root harvester with independent OGP

sides of the rear driving wheels. Depending on the operating modes with unsteady motion, the following control options are possible, providing high performance at a certain level of dynamic loads, or minimum dynamic loads when the time factor is not prevalent. Optimal control is also promising when additional parameters are included in the goal function.

Key words: *mathematic model of a volumetric hydraulic drive in the transmissions of tractors and root-harvesters.*

ДОСЛІДЖЕННЯ ДИНАМІЧНОГО НАВАНТАЖЕННЯ ТРАКТОРІВ З ОБ'ЄМНИМ ГІДРАВЛІЧНИМ ПРИВОДОМ

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Анотація

Процеси, що виникають у трансмісіях тракторних агрегатів та самохідних сільськогосподарських машин за різних режимів руху і в процесі регулювання, характеризуються складними залежностями, які вивчаються аналітично або експериментально. Відомі різні способи отримання математичних моделей. Одним із них є класичний метод прямого опису. Іншим – використання пасивних і активних методів регресійного аналізу. Раціональним є використання обох методів, поєднання яких дає можливість отримати необхідну математичну модель.

Об'ємний гідропривід (ОГП) все більше знаходить застосування в трансмісіях сучасних тракторів і самохідних сільськогосподарських машин. У наведеній статті розглядається математичний опис аксіально-поршневих гідромашин.

Проведено дослідження перехідних процесів, їх оцінка проводилися для визначення навантажень, що виникають у трансмісії машини за ступінчастої зміни навантаження, передавального числа ОГП і постійної подачі палива під час розгону агрегату з місця. Режим розгону агрегату вивчався під час руху на оранці і на транспортних роботах для таких параметрів і таких початкових умов: швидкість обертання валу гідромотора і валу двигуна; крутний момент на валу двигуна; Крюкова навантаження; тиск у напірній магістралі ОГП. Враховано динамічні характеристики гідромашин, витоку рідини і її пружні властивості, а також змінні значення ККД гідроприводу. Результати моделювання зіставлені з експериментальними дослідженнями. Як об'єкти дослідження використовувалися: макет гусеничного трактора Т-150Е з незалежними повнопотоковий ОГП лівого і правого бортів; макет колісного коренезбирального комбайна з незалежними ОГП бортів задніх ведучих коліс. Залежно від режимів роботи за несталого руху можливі такі варіанти управління, що забезпечують високу швидкодію за *деякого рівня динамічних навантажень або мінімальні динамічні навантаження, коли часовий чинник не є превалюючим. Перспективним є й оптимальне управління, коли у функцію мети включені додаткові параметри.*

Ключові слова: *математична модель об'ємного гідроприводу в трансмісіях тракторів, коренезбиральні комбайни.*

Introduction. The processes, arising in the transmissions of tractor units and self-propelled agricultural machines at various motion modes and in the process of adjustment, are characterized by complex dependencies that are studied theoretically or experimentally. Different ways of obtaining mathematic models are known. One of them is the classical method of direct description. Another one is the use of passive and active methods of regressive analysis. It is expedient to use both methods, the combination of which enables obtaining a needed mathematic model. The main elements of a tractor assembly in their totality determine the character of transition processes and dynamic loads within the transmission in various operation modes. This enables considering them as some modules, from which it is possible to compose a mathematic model needed for a research.

Recent research and publications analysis. At present, volumetric hydraulic drives (VHD) find an ever wide-spread application in tractors' and self-propelled agricultural machines' transmissions. The development and research of full-threaded and double-threaded transmissions with a VHD are viewed in detail and from various points in works by Gorodetskiy K.I., Petrov V.A., Babayev O.M., Samorodov V.B., et al.

Nevertheless, the main issues accentuated in these works were exploitation efficiency, technical-economic indicators improvement ensured by infinitely variable change in the transmission' gear ratio. The influence by a VHD on dynamic load decrease in tractors' and self-propelled machines have been viewed incompletely.

The proposed methodology of compiling a mathematical model for viewing dynamic load of a VHD transmission is topical.

The goal of the research and problem setting. The goal of the research is to show on examples the methodology of mathematical models compilation for different tractors and self-propelled machines with VHD in their transmissions. To this effect, it is necessary to substantiate and formulate mathematical models with different positions of full-band VHD in transmissions of the viewed research objects to determine dynamic loads.

Researched objects. As objects for the research, the following were used: a model of the T-150E caterpillar tractor with independent full-threaded VHD on the left and right sides; KS-6B root-harvesting machine with a VHD on the rear drive axle; KS-6V root-harvesting machine with the independent VHD on the rear and front drive axles; a model of a combine root-harvester with the independent right and left sides' VHD on the rear drive wheels.

A volumetric hydraulic drive's mathematical model. There are several ways of compiling such a model. In accordance with one of them, the value of the rotation moment on a hydro-machine's shaft is obtained by using the values of kinematic and power efficiency indices [1]. In the general case, the efficiency ratio index is a function of the torque and the rotation speed of a hydro-machine's shaft as well as a hydro-drive's gear ratio.

In the work [2], the mathematical model is compiled with taking into consideration the hydro-drive's dynamics. The dynamic model for the "regulated pump – regulated hydro-drive motor" system is represented in (Figure 1).

On the basis of the D'Alambert principle, the description of the dynamic model represented in Fig.1 is the following:

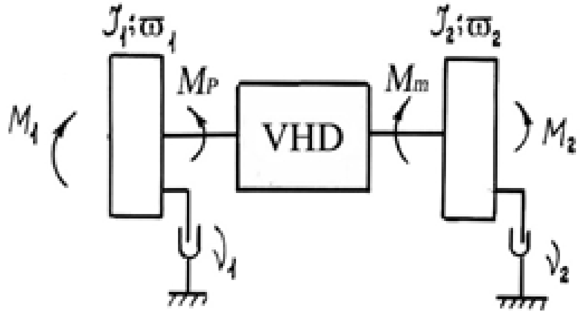

Fig. 1. Dynamic model of the "regulated pump – regulated hydro-drive motor" system

$$I_1 \frac{d\omega_1}{dt} + \nu_1\omega_1 = M_1 - M_p, \tag{1}$$

$$I_2 \frac{d\omega_2}{dt} + \nu_2\omega_2 = M_m - M_2. \tag{2}$$

where I_1, I_2 are inertia masses of the pump and the hydro-drive; ν_1, ν_2 are the factors of links imitating non-elastic resistance forces; ω_1, ω_2 are rotation speeds of the pump's and the hydro-drive's inertia masses; M_1, M_2 are force moment and the resistance forces' moment; M_p, M_m are the torque values on the shafts of the pump and the hydro-drive.

It is known [3, 4] that the moments on hydro-machines' shafts are the function of the operation pressure p gradient at the input and the output of the machine and its mechanic efficiency factor η_m, that is $M_p, M_m \propto (p, \eta_m)$. Nevertheless, the methodology to determine η_m, according the authors' opinion, is connected with certain difficulties.

In works [5, 6], it is supposed to consider for VHD in general the pump as having no losses at operation, while uniting all the volume and mechanical losses to the total efficiency factor of the hydro-drive η_h, and relate them further to the hydro-motor. In this case,

$$M_p = f(p)M_p = f(p) \text{ while } M_m = f(p, \eta_h).$$

To determine the pressure within a VHD, let's write the conditions of consumptions equality, taking into consideration that the pump supply Q_p is spent on hydro-motor's geometric supply Q_m, on leakages consumption Q_l, and pressure consumption Q_c [5, 6], i.e.,

$$Q_p = Q_m + Q_l + Q_c. \tag{3}$$

The values of working liquid consumptions for the "regulated pump – regulated hydro-motor" system explicated in Fig. 1 are represented as follows [2]:

$$Q_p = W_p \omega_1 e_p, \tag{4}$$

$$Q_m = W_m \omega_2 e_m. \quad (5)$$

where W_p, W_m – characteristic volumes of the pump and the hydro-motor; e_p, e_m – regulation parameter for the pump and the hydro-motor, which is the ratio between the current value of the hydro-machine's washer inclination angle to its maximum value.

For mathematic description of liquid's leakages through clearances, various methods are used. Constant losses in packages can be a product of some fiction pressure and the largest consumption [7], and the system dampering depends considerably on the losses in the pump and the hydro-motor proportional to the pressure [8]. Usually, leakage losses in a system are accounted for by the expression:

$$Q_l = L \cdot p, \quad (6)$$

where L – proportionality factor, $L = f(W)$.

Similarly, compression consumption is found, with taking into consideration hydro-machines' elasticity-dynamic characteristics [9]:

$$Q_c = \frac{V}{\chi}\frac{dp}{dt}, \quad (7)$$

where V – the volume of working liquid in the working space, χ – the volume elasticity module of working liquid.

It is known that the studying of transition processes in a VHD regardless of elasticity-dynamic characteristics of all the system's elements leads to considerable errors in amplitude and in phase. The volume elasticity module χ and the volume of working liquid in the pressure mains V are non-linear functions depending on liquid's pressure and temperature and, notably, at small pressures the module of liquid's volume elasticity changes considerably as applied to a VHD [9; 10; 11].

Suppose that the working liquid's temperature at long operation of hydro-transmission V stabilizes and χ at changing loads remains constant further on. This assumption enables to consider the values of V and χ nonlinear, depending only on pressure p of liquid.

Figures 2 and 3represent the mean dependencies of the elasticity module χ and the volume of mineral oils V used in the studied VHD in the function of pressure p [10].

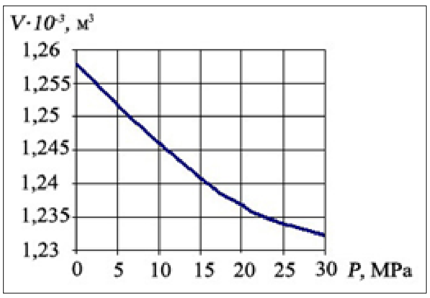

Fig. 2. Dependency of the volume of mineral oils V in the function of pressure p

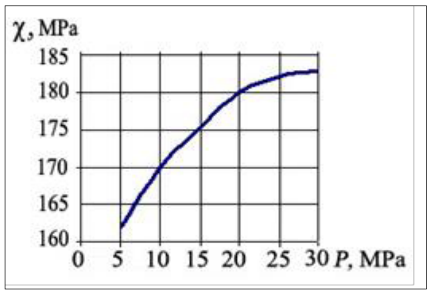

Fig. 3. Dependency of the elasticity module χ in the function of pressure p

Confining to considering the pressures arising in VHD at loads close to nominal ones makes it possible to describe the approximation of the corresponding curves as parabolas:

$$V = V_0 + k_{v1}p + k_{v2}p^2, \qquad (8)$$

$$\chi = \chi_0 + k_{\chi 1}p + k_{\chi 2}p^2. \qquad (9)$$

Here, V_0, χ_0 – the volume of working liquid and volume elasticity module corresponding to the minimal values of accepted pressure change zone; $k_{v1}, k_{v2}, k_{\chi 1}, k_{\chi 2}$ – approximation factors.

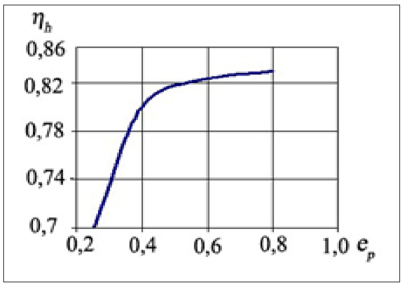

Fig. 3. Dependency of the elasticity module χ in the function of pressure p

It is known that the total efficiency factor of hydro-transmission η_h is a non-linear function depending on the regulating parameter e_p, rotation speed of the pump shaft ω_1, and pressure p. In Figure 4, there are demonstrated experimental dependencies of the “Dowty” hydro-transmission’s efficiency factor η_h on the regulating parameter e_p obtained for pressure p corresponding to nominal load on the unit [2]. Similar data, presented in the work [1], were obtained when analysing the all-modes characteristics of the S-21 type VHD by “Sauer” company.

In general, the dependency $\eta_h = f(\omega_1, e_p)$ can be presented as a function invariant in relation to the rotation speed of the pump shaft, which is approximated by the expression,

$$\eta_h = \eta_{h0}[1 - \exp(-\frac{e_p}{k_\eta})], \qquad (10)$$

where η_{h0}, k_η – approximation factors.

In work [12], the value of the GST-90 hydro-drive’s efficiency factor is presented as a polynomial obtained through the use of the regression method considering the pressure, angle velocity, and the regulating parameter.

Taking into consideration the equations (2–6), the equation describing the change in pressure in a VHD will be the following:

$$\frac{V_0 + k_{v1}p + k_{v2}p^2}{\chi_0 + k_{\chi 1}p + k_{\chi 2}p^2}\frac{dp}{dt} + Lp = W_p\omega_1 e_p - W_m\omega_2 e_m, \qquad (11)$$

For the drive in question, $W_p = W_m = W, e_m = 1$ (not regulated hydro-motor). Considering the presented data, the equation (8) is the following:

$$\frac{dp}{dt} + \frac{(\chi_0 + k_{\chi 1}p + k_{\chi 2}p^2)}{(V_0 + k_{v1}p + k_{v2}p^2)}Lp = \frac{(\chi_0 + k_{\chi 1}p + k_{\chi 2}p^2)}{(V_0 + k_{v1}p + k_{v2}p^2)}W(\omega_1 e_p - \omega_2) \qquad (12)$$

The estimated values of torques on hydro-machines’ shafts M_p, M_m on condition that all losses in the VHD appear only in the hydro-motor [6], are as follows:

$$M_p = e_p Wp, M_m = e_m Wp\eta_h, \tag{13}$$

Considering the equations (7), (9), (10), the mathematical model of the “regulated pump – not regulated hydro-motor” system is described as:

$$I_1 \frac{d\omega_1}{dt} + \nu_1\omega_1 = M_1 - e_p Wp; \tag{14}$$

$$I_2 \frac{d\omega_2}{dt} + \nu_2\omega_2 = Wp\eta_{h0}[1 - \exp(-\frac{e_p}{k_\eta})] - M_2; \tag{15}$$

$$\frac{dp}{dt} + \frac{(\chi_0 + k_{\chi 1}p + k_{\chi 2}p^2)}{(V_0 + k_{v1}p + k_{v2}p^2)} Lp = \frac{(\chi_0 + k_{\chi 1}p + k_{\chi 2}p^2)}{(V_0 + k_{v1}p + k_{v2}p^2)} W(\omega_1 e_p - \omega_2). \tag{16}$$

The momentums describing the engine’s work (a diesel with an efferent all-mode regulator enabling work at full supply of fuel and at partial modes) and the resistance momentum with taking into account the machine’s slipping on the soil, of the translationally moving masses of the load and the resistance forces of technologic equipment or the hook load are shown in general and given in [11].

The mathematic model (11) is a system of non-linear differential equations, which are solved through applying numerical methods in the MatLab program packet, Simulink section.

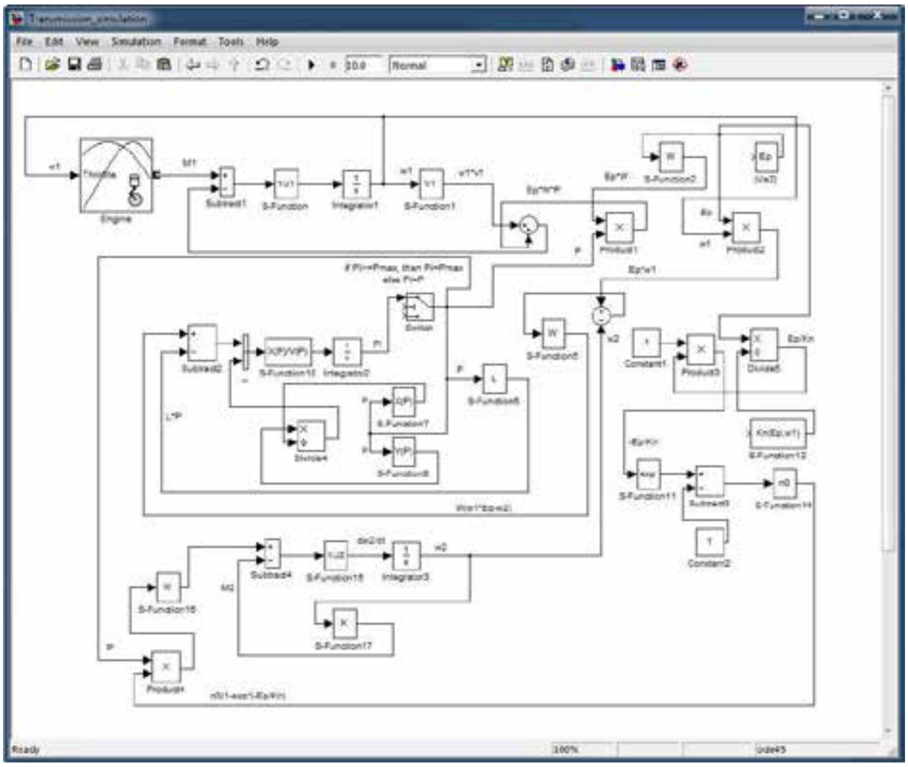

Fig. 5. A section of the mathematical model

In Fig. 5, there is presented a mathematical model, which with accepted allowances describes the operation of the "engine – VHD transmission system – the load" system.

The study of the transition processes and their estimation were performed to determine the loads appearing in a machine's transmission at scale-like change in load, the VHD ratio and constant fuel supply at the machine's acceleration from standstill.

The machine's acceleration mode was studied when on the move on plowed land and at transportation work for the following parameters and the initial conditions that follow: the rotating speed of the hydro-motor's shaft and that of the engine's shaft; the engine shaft's torque; the hook load; the pressure in the VHD's pressure line: ($\omega_{hm} = 0$; $\omega_e = 210$rad/sec; $M_e = 70$Nm; $P_{\text{hk}} = 0$; $P_{VHD} = 0$). The regulating parameter was changed step-like for the value of e_p ranging (0,4…0,7) following which it remained constant.

For the accepted initial conditions, the integration process corresponds to the machine's acceleration from standstill with the deepened plow or a loaded hind carriage at spasmodic increase in load from zero to the predetermined at calculation value. The position of the fuel pump regulator's handle x_r is set in the range ensuring fuel supply from $x_{r1} = 100$ %, $x_{r2} = 90$ %, $x_{r3} = 80$ %, till $x_{r4} = 70$ %.

On the modelling results (Fig. 6), the transition processes of the studied parameters of the system were obtained in a continuum.

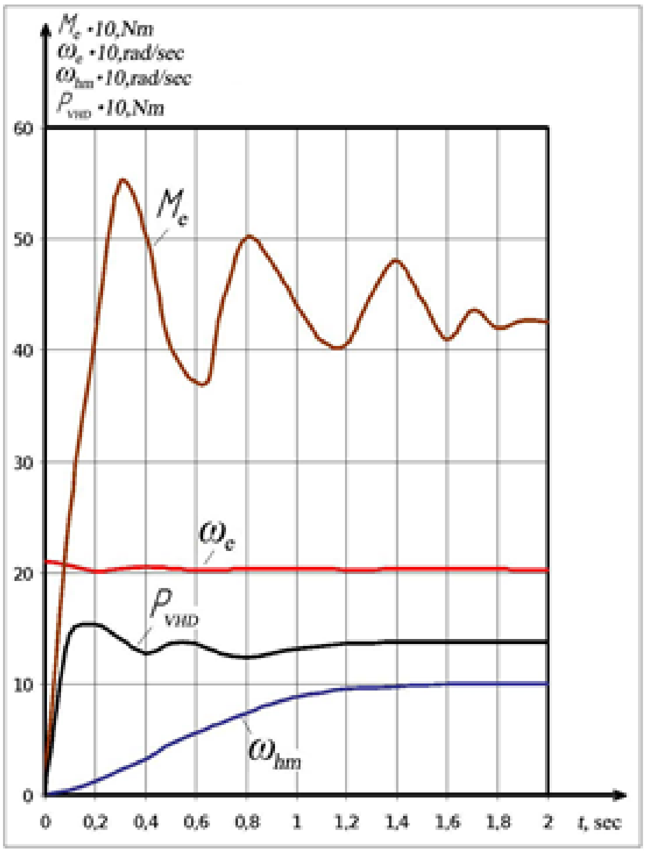

Fig. 6. Changing of the system's parameters with time

The loads in the transmission at transient motion are estimated through the dynamic coefficient:

$$\delta_e = \frac{M_{e\max}}{M_{eest}}, \tag{17}$$

where $M_{e\max}$ and M_{eest} are the maximum and the settled meaning of the engine shaft's torque.

In Figure 7, there are presented the dynamic coefficient's dependencies on the regulation parameter for different positions of the fuel supply handle in the process of acceleration.

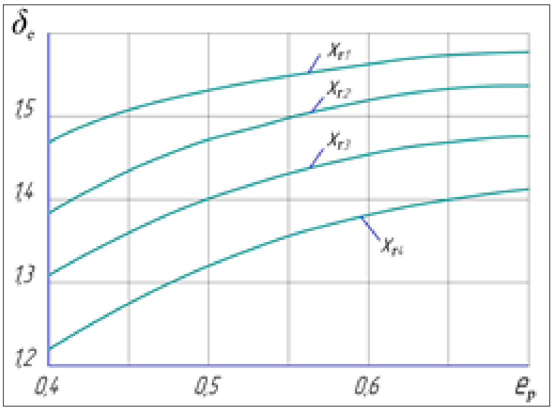

Fig. 7. Dependency of the dynamic coefficient on the regulation parameter for different positions of the fuel pump regulator

With the increase in the value of the regulation parameter in the range between 0,4 and 0,7 at the constant fuel supply, the value of the dynamic coefficient increases by 10 % on average. This is connected with the fact that the momentum of resistance to movement consists of a constant value and a variable component, which is the function of the motion speed, which at the zero initial conditions equals zero. This is why the $M_{e\max}$ values for this position of the fuel supply handle is a function depending on the value of the transmission's ratio. Nevertheless, as e_p grows, there increases both the average speed for the supposed movement conditions and the mean settled value of the resistance momentum, and, correspondently M_{eest}. This is why the dependency δ_e on e_p at the permanent fuel supply changes according to a non-linear law.

With a decrease in fuel supply for this ratio of hydro-volumetric transmission there occurs a more intensive decrease in $M_{e\max}$ and M_{eest}. This leads to a decrease in dynamic coefficient by 12 % on average.

One of important tasks at modelling is the process of identification of a mathematical model for the obtained results to be adequate to the experimental data with certain precision. The solving of the set task was made in laboratory conditions. On the test bench, there were determined dynamic loads in transmission elements, which appear at ratio changing in a hydro-volumetric drive (VHD), at fuel supply or with load application.

When studying transition processes, there were measured torques on the primary shaft of the transfer gear M_{pr} and on the load generator's shaft M_G, the rotation speeds of the engine ω_e and generator ω_G shafts along with their rotation numbers n_e and n_G, the pressure in the pressure line P_{VHD}, the position of the servo stock S_{st} that changes the adjustment value of the pump regulating e_p, the position of the fuel pump adjustment handle x_r.

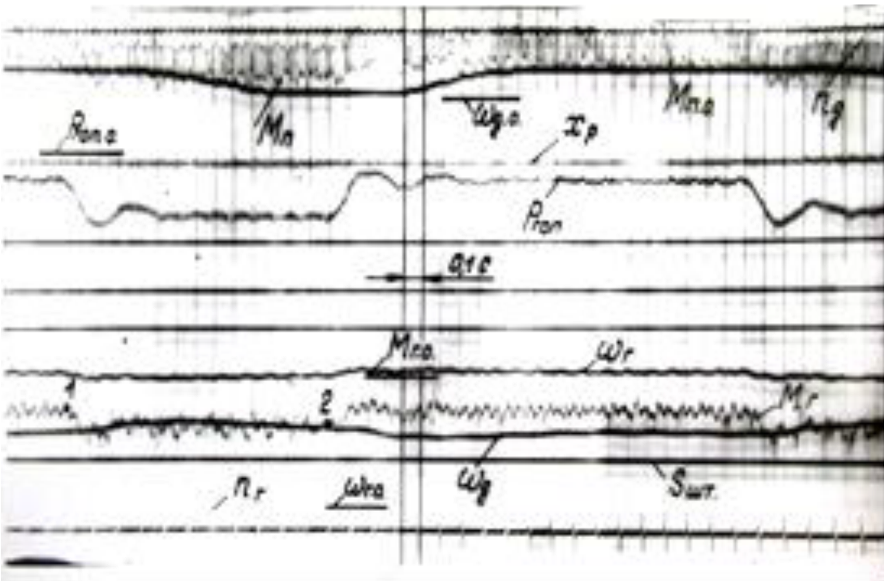

Fig. 8. Transition processes in the "engine-VHD-load" system at spasmodic change in load

In Figure 8, transition processes in the studied parameters of the system at applying and removing of a load (points 1 and 2) are shown.

At the initial position, the system's parameters were as follows: M_{pr} = 150 Nm; ω_e = 165 rad/sec; M_G = 300 Nm; ω_G = 57rad/sec; P_{VHD} = 3,5 MPa.

In point 1, when the contactor was switched, the load on the generator's shaft increases and reaches the value of $M_{G\max}$ = 580Nm. The pressure P_{VHD} raises and reaches the value of $P_{VHD\max}$ = 9MPa. This causes an increase in resistance of the hydro-pump's shaft. The rotation speed of the engine's shaft decreases by 8 % compared with the initial speed. The torque M_{pr} increases and reaches the value of $M_{pr\max}$ = 267 Nm. In point 2, there is the removal of load through switching off of a part of resistance from the excitement coil of the load generator. The system's parameters return to the initial position.

The analysis of the dependencies presented in Fig. 8 demonstrates a stability of the processes occurring within the engine – VHD – load" system at a step-like disturbance. The transition processes within transmission are of aperiodic or oscillatory character.

A comparison of the pre-calculated and the experimental dependencies demonstrates a good correlation. The maximum error for the torque is 8 %, for the engine shaft rotation speed is 4 %, for the hydro-motor shaft rotation speed is 7 %, for the pressure in the pressure line is 11 %.

In the course of the theoretic research, it was established that one of the reasons for the occurrence of increased dynamic loads in the power transmission elements is a high speed of the servo stock S_{st} movement that changes the value of the pump regulation parameter e_p .

In Fig. 9, there is demonstrated the dependency of the dynamic coefficient's value δ_e on the speed of the servo stock movement V_{st} .

Notably, at this condition the system's speed-performance decreases. This is why the selection of the efficient speed of the control organ's movement parameter that regulates the VHD pump e_p should be made with taking into consideration the dynamic loads occurring in the process, as well as the time of the transition processes within the system. For this system, these requirements are observed at the speed range V_{st} of 10…16 mm/sec, the dynamic coefficient δ_e of 1.4…1.6. The system's acceleration time at these conditions is 1,4…1,6 sec.

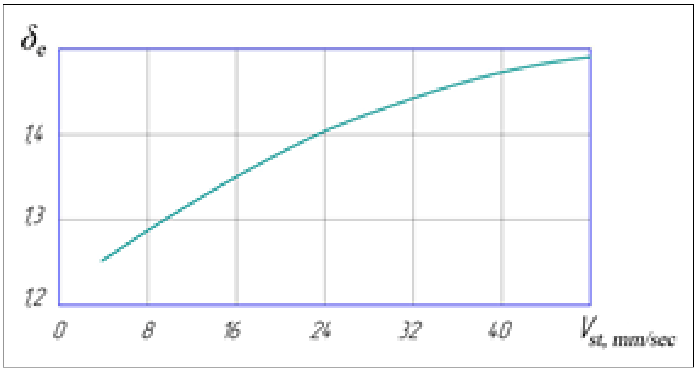

Fig. 9. The dependency of the dynamic coefficient's value δ_e on the speed of the servo stock movement V_{st}

Conclusions. Depending on the modes of operation when the motion has not been established, the operation options that ensure high speed performance at some levels of dynamic loads, or the minimum dynamic loads, when the time factor is not prevailing, are possible. Also, there is a perspective optimum control wherein the target function includes additional parameters.

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