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INVESTIGATION OF OPERATION OF THE HYDROVOLUMIC MANUAL CONTROL SYSTEM ON THE BASIS OF A DOSING PUMP WITH AXIAL SPOILER WITH ACCIDENTAL LOAD

One of the factors ensuring the stable development of agriculture is high-quality and reliable equipment. Today, the development of hydraulic drives has largely led to its widespread use in agricultural machinery and tractors. One of the systems fully hydrogenated in heavy agricultural machinery is the steering system. Hydrovolume steering system has the following advantages - compactness, freedom in the layout of the machine, high efficiency, accuracy, the ability to transfer significant effort to the steered wheels with minimal effort of the operator. Disadvantages include certain restrictions when the associated load on the steered wheels when turning. This parameter is one of the indicators of the quality of hydrovolume steering systems and the minimum effort at which the process of assembling the wheels begins, is regulated by international standards.

The article performs mathematical modeling of the hydrovolume steering system. The mathematical model consists of the equations of continuity of flows of the working fluid, the equations of moments and the Lagrange equation of the 2nd kind, which describes the operation of the planetary gearbox. The search for solutions of this model was carried out using the original software, built on the basis of the Runge-Kutta-Feldberg method with an automatic integration step. As a result of the analysis of the received transient processes areas of steady work were constructed that allowed to receive a ratio of parameters of system at which its steady work is provided. The proposed design solution allowed to increase the limit value of the associated load in excess of which there is a spontaneous movement of the steered wheels. The analysis of the workload at the associated load had a significant impact on the values of the overlap of the working edges on any work of the system under the action of the associated load, and the correct selection of these floors can achieve significant quality improvement.

Keywords: *hydrovolume steering system, hydraulic drive, mathematical model, transient process, area of stable operation, associated load*

Introduction. Hydrostatic steering systems are widely used in technological machines for agricultural purposes due to their reliability, ease of maintenance, advantageous layout, and compactness. But in addition to the above advantages, systems of this type have a number of disadvantages, one of which is the possibility of uncontrolled movement of the rod of the executive hydraulic cylinder under the action of a passing load due to a discrepancy between the characteristics of the system. Therefore, research in this direction is an urgent task.

Analysis of recent research and publications. Hydraulic systems are used in mobile machines to perform many operations, from moving cargo and hydraulic control of equipment to moving and driving vehicles. Due to the growing demand for fossil fuels, many studies have been conducted to find new and alternative solutions for hydraulic systems in the vehicle, as shown in [1].

In [2] the main attention is paid to the operation of hydraulic transmissions and the control system. The results of research published in [1], [2] show that mathematical modeling is the basis for a better understanding of the principles of hydraulic system, as well as the impact of design and operational parameters on the performance and quality of hydraulic system.

Tracking systems not only play a role in improving the efficiency of agricultural machinery, but can also influence the dynamics of the machine, determining its efficiency in performing technological operations. The tracking system affects the durability of the car, increasing the safety and comfort of the driver.

Hydrostatic control is an effective way to drive trucks that reduce driver effort, especially at low speeds. This aspect is especially important in agricultural tractors. The feature of these machines is a high loading axle, large tires, off-road driving [4]. The hydrostatic steering system must ensure the comfort and safety of the driver, ensuring the correct ratio of the steering angle to the turn of the steered wheels.

In [5], studies of the proposed algorithm for tracking the path of an all-wheel drive tractor based on inverse kinematic modeling were performed. However, nothing is said about the effect of system design parameters or the amount of torque on the steering wheel during the maneuver. The work is devoted to vibrations on the steering wheel [6]. The purpose of the work is to study the comfort of the driver with reference to fluctuations during the operation. A number of experiments have been performed on various agricultural tractors and a number of solutions have been proposed to reduce oscillations.

The purpose of [7] is to analyze the effect of steering torque feedback in agricultural tractors; This problem arises because electronic and electro-hydraulic control systems remove the mechanical connection to the tires, but can actually supply any desired "artificial" torque to the steering wheel. The work explores the topic through experimental measurements on the vehicle and on the simulator: it concludes that if adjustment is the only effort required of the driver, the torque feedback moment can be removed without affecting the work; but if the driver also performs any other control task, the torque feedback helps to improve performance and effort, avoiding unnecessary steering adjustments made by the driver when no "reference feedback" is available. This publication notes the importance of estimating steering torque in a traditional hydrostatic system because of its importance as a feature of the control system design.

Publications [8-10] consider a detailed physical model of the hydrostatic control system of a tractor when driving on a hard road. A similar approach to modeling hydrostatic steering systems is presented in publications [12,13]. In particular, in [12] we can find a similar approach to physical modeling, but we consider a three-axle heavy truck. The modeling of the vehicle is carried out taking into account the full load of the car. An analysis of some parts of the crane-type distributor was carried out to assess their impact on the controllability of the vehicle.

Therefore, it should be noted that the topic of the publication is relevant and requires further consideration.

Research goal there is a search for such a design of the working edge of the spool valve, which would fully ensure, in accordance with the current standards, the quality of the hydrostatic steering system under the action of a passing load.

To achieve the goal, it is necessary to solve the following tasks:

- to develop a mathematical model of the hydrostatic steering system, which allows to fully determine the influence of the shape of the working edges on the accuracy and speed of processing the input signal;

carry out theoretical studies of the obtained mathematical model and determine the permissible limits of the model parameters under which the control process takes place.

Materials and methods. The theoretical study is based on a promising program analysis based on the Runge-Kutti-Feldberg method.

The constructive scheme of the steering hydromechanism showed in Fig.1. The steering wheel (not shown in the diagram) transmits rotational motion to the input shaft 1 of the steering hydromechanism, which is detached from the sleeve 2 by a mechanism consisting of a pin 3, pressing into the shaft 1, and a groove in the sleeve 2 made at an angle of 45 . As a result, an angular installation of the input shaft was formed, leading to movement in the axial stresses of the sleeve and together with it a spool 4, in the opening of which two locking washers are installed in contact with the ends of the sleeve. Moving the control spool to the shift without critical workers performed spool pair, as formed by the corresponding grooves at the spool and the housing 5.

The movement of the spool thus leads to a change in the conditions of switching the hydraulic line of this hydromechanism, which are determined by the values of the opening of the corresponding working windows of the spool pair.

The working fluid under pressure p0 is supplied to two sockets in the case - to the third evil and the first area (Fig. 1). The first of these sockets, together with the edges on the gold spools, opens with such a "revelation" - if the spool is in the center of the position, all the fluid flow generated by the pump station through the working windows 5 and 6 opens as significant, L5 for significance, resulting in the pressure p0 in the pressure cavity approaches the drain pressure. The pressure in the evil magic, which is associated with the corresponding bores in the body, equal to p7. The opening of these edges depends on the cut-off edges 9 and 10, the opening of which in the neutral position is L9 and L10.

The discharge line also supplies the working fluid to the end of the borehole in the housing, which is connected to the cavities of the dosing mechanism 6 by the respective working windows. The spool collar, the working edges of which 3 and 4 are made with positive overlaps L3 and L4, together with the edges of the corresponding bore in the housing create working windows of the spool pair connecting the right cavity of the actuating hydraulic cylinder 7 under pressure p3 with the dosing mechanism cavity. , pressure to some extent p2, or evil (pressure p7)

The extreme left side of the spool, the edges of which 7 and 8 are made with positive overlaps L7, L8 boring edges, rectilinearly left cavity of the actuator with the cavity of the dosing mechanism, which is conducted by intermediate pressure p1, and when moving zlotys in gold).

We assume that the movement of the spool occurs in the positive direction shown in Fig. 1 as x. At the same time working windows 1,3,7 having opening L1, L3, L7 accordingly open (fig. 1). The working fluid from the pressure channel enters the dosing unit. Further from the dosing case, the working fluid through the working window 3 enters the right cavity of the hydraulic cylinder. The piston of the hydraulic cylinder moves, pulling the fluid from the left cavity to the left. The shaft of the dosing mechanism 6 by means of a planetary gear 8 and a cardan 9 is extended with a spool sleeve.

When the liquid passes through the dosing unit, its shaft rotates, which is transmitted through the gearbox and the cardan to the spool sleeve and move it in the opposite direction, turning the spool to its original position. The water supply to the power circuit is stopped. Naturally, all the working fluid from the pressure channel through the working window, as it forms "from the critical center", passes through the sinister channel.

The mathematical model of the hydrovolume manual mechanism includes the level of continuity of series flows, acting forces and moments, as well as equations that determine the special leaf of the work of this hydraulic system.

When storing a mathematical model taking into account the results of previous assumptions [2], the same assumptions were made:

- viscosity, viscosity and coefficient of neutralization of working processes, not freezing from temperature delays of robots on the steering hydromechanism in a tired temperature mode;
- the loss of tic at the directed steering hydromechanism and on external valves as usually having insignificant size will not be considered;
- the coefficient of pliability of the liquid did not heal from the carcass and the content of the gas composition, as in the tedious mode of operation of the steering hydromechanism, its number varies slightly;
- pressure of support on evil insignificant and healthy invariable;
- leakage factor and fluid flow in the units of hydraulic units is constant and has not healed from the size and shape of the gaps;
- the separation between the phenomena of the hydromechanism is insignificant, which allows the identification of its system with zoo-average parameters and does not allow the inclusion of wave processes;
- pulsation of the pump supply taking into account its significant frequency allows not to take into account its influence on the operation of the steering hydromechanism;
- in the process of testing the control signal, which is a force action on the spool sleeve, the movement of the hydraulic system elements occurs without stopping, which allows not to take into account the dependence of dry friction forces on speed.

The considered system can be conditionally divided into two subsystems - hydraulic and mechanical. The operation of the hydraulic subsystem is described by the equations of continuity of the working fluid flows, and the operation of the mechanical subsystem is described by the equations of moments and forces; in addition, the operation of the planetary reducer is described using the Lagrange equation of the second kind.

The equation of the balance of the flow of working fluid in the pressure line connecting the pump with the corresponding channels in the spool housing (Fig. 1) has the following form:

$$Q_H = Q_1 + Q_2 + Q_5 + Q_6 + Q_{УТ.H} + Q_{ДЕФ.H}\ , \tag{1}$$

where Q_H - the flow rate of the fluid supplied from the pump; Q_1 - flow through the first working window of the spool; Q_2 - flow through the second working window of the spool; Q_5 and Q_6 - fluid flow through the working windows 5 and 6 of the spool pair, forming an open center; $Q_{вит.н}$ - is the flow rate of fluid leakage from the pressure line due to its leak; $Q_{деф.н.}$ - flow rate aimed at compensating for the deformation of the pressure line filled with the working fluid under the action of pressure p_0.

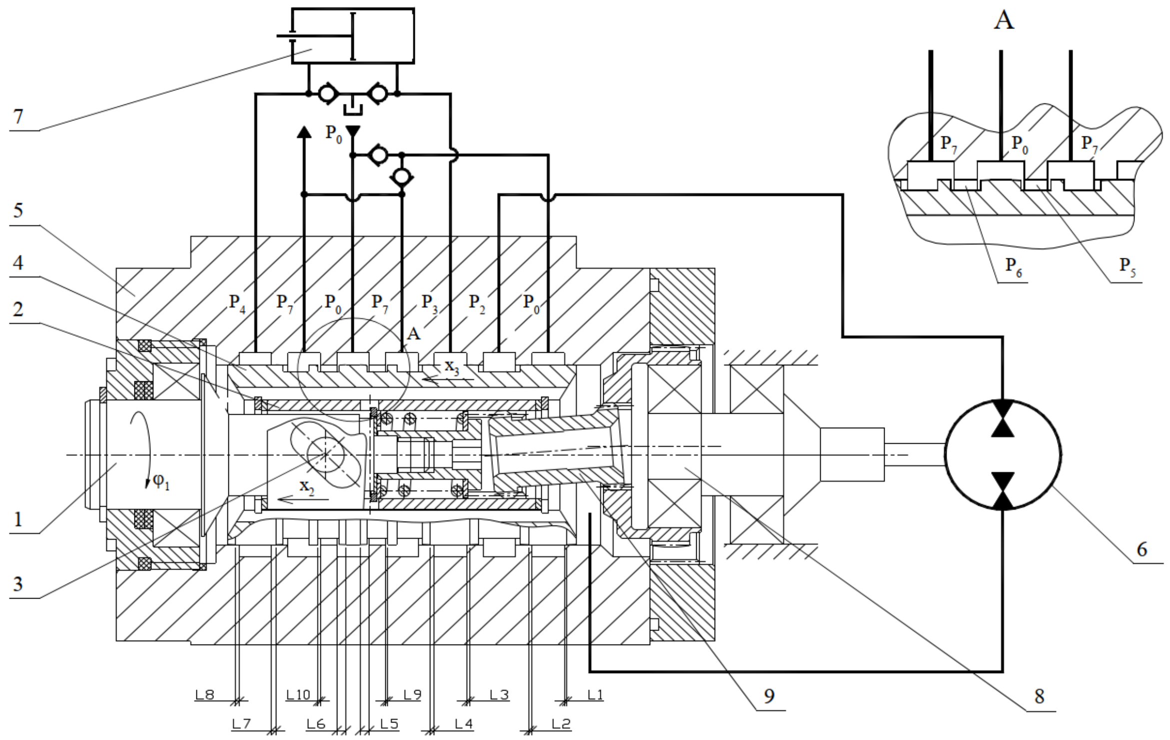

Fig. 1. Constructive scheme of the hydrovolume steering system on the basis of the pump-dispenser MRG.01.

The flow equation in the cavity connecting the first working window with the dosing mechanism cavity and the eighth working window (with overlap L_8) and under pressure p_1, has the form

$$Q_1 = Q_Д + Q_8 + Q_{ВИТ.Д1} + Q_{ПЕР.Д} + Q_{ДЕФ1}\,, \quad (2)$$

where $Q_Д$ - the consumption of fluid consumed by the dosing mechanism during rotation; Q_8 - fluid flow through the eighth working window of the spool pair; $Q_{ВИТ.Д.1}$ - the cost of leakage of fluid from the cavity of the dosing mechanism under pressure p_1; $Q_{ПЕР.Д}$ - the flow rate of the liquid through the gaps between the cavities of the dosing mechanism, which are under pressure p_1 and p_2. $Q_{def\ 1}$. - fluid flow rate caused by deformation of the cavity and the working fluid under pressure p_1.

The balance of fluid flow in the cavity, which is connected to the second cavity of the dosing mechanism, with the pressure line through the second working window and the right cavity of the cylinder through the third working window can be determined by equation

$$Q_Д + Q_{ПЕР.Д} = Q_3 + Q_2 + Q_{ДЕФ.2} + Q_{УТ.Д2}\,, \quad (3)$$

where Q_3 – fluid flow through the third working window; $Q_{ДЕФ.2}$ - fluid consumption to compensate for the deformation of the cavity under pressure p_2. $Q_{УТ.Д2}$ - the cost of fluid leakage from the second cavity of the dosing mechanism due to leaks.

The supply of fluid through the third window of the spool pair into the cavity of the hydraulic cylinder is the reason for the movement of the piston at the appropriate speed. Thus a part of liquid through the fourth working window (with overlapping L_4) enters the housing bore associated with the drain. Thus, the cost equation for this cavity will be as follows

$$Q_3 = Q_Ц + Q_4 + Q_{УТ.Ц1} + Q_{ПЕР.Ц} + Q_{ДЕФ.3} - Q_{В3}\,, \quad (4)$$

where $Q_Ц$ – fluid consumption, providing the movement of the piston at a given speed; Q_4 – fluid flow through the fourth working window of the spool pair; $Q_{УТ.Ц1}$ - the cost of fluid leakage from the right cavity of the cylinder due to the pressure drop p_3; $Q_{ПЕР.Ц}$ - the cost of flowing fluid from the cavity of the cylinder under the action of pressure drop p_3 and p_4; $Q_{ДЕФ.3}$ - fluid consumption to compensate for the deformation of the cavity under pressure p_3; $Q_{В3}$ – anti-vacuum valve in case of pressure p_3 becomes lower than atmospheric.

Fluid from the left cavity of the hydraulic cylinder under pressure p_4 enters the borehole in the spool housing and through the seventh working window with an overlap L_7 merges into the tank. Fluid is also supplied to this cavity through the eighth working window. Given the above, the equation of cost balance for a given cavity will look like

$$Q_Ц + Q_8 + Q_{ПЕР.Ц} + Q_{В4} - Q_{ДЕФ.4} - Q_{УТ.Ц2} = Q_7\,, \quad (5)$$

where $Q_{В4}$ - fluid flow through the appropriate overflow valve into the line connecting the left cavity of the hydraulic cylinder with the channel in the housing, in the case when the pressure p_4 becomes lower than atmospheric; $Q_{ДЕФ.4}$

- fluid consumption to compensate for the deformation of the fluid-filled cavity of the hydraulic system under pressure p4; $Q_{УТ.Ц2}$ - the flow rate of fluid leakage from the cavity due to its leaks.

To determine the pressure values of p0 at all positions of the spool, it is also necessary to take into account the flow of fluid in the cavities formed by boring in the housing and the edges of the "open center" (L_5, L_6), and clipping (L_9, L_{10}), through which the unloading of the pumping station in the absence of a control signal on the input shaft. These equations will look like this

$$Q_6 = Q_{10} + Q_{ДЕФ6}\,, \tag{6}$$

$$Q_5 = Q_9 + Q_{ДЕФ5}\,, \tag{7}$$

where Q_9, Q_{10} – drain fluid flow through the ninth (L_9) and the tenth (L_{10}) clipping edges; $Q_{деф5}$, $Q_{деф6}$ – fluid consumption to compensate for the deformation of the fluid volume concentrated in the grooves between the clipping edges and the edges of the "open center".

Components of equations (1) – (7) are determined by the following dependencies.

Fluid flow through the first spool pair window (L_1)

$$Q_1 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_0 - p_1)}{12 \cdot \nu \cdot \rho \cdot (-L_1 + x_3)} & \to x_3 < L_1 \\ \mu \cdot f_1(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_0 - p_1|} \cdot sign(p_0 - p_1) & \to x_3 \geq L_1 \end{vmatrix}, \tag{8}$$

where d_3 – spool diameter; δ_9 – gap in the pair of spool - housing; ν - kinematic viscosity of the working fluid; ρ - the density of the working fluid; μ - the coefficient of consumption of the working fluid; x_3 – spool position.

Fluid flow through the second working window of the spool pair (L_2)

$$Q_2 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_0 - p_2)}{12 \cdot \nu \cdot \rho \cdot (L_2 + x_3)} & \to x_3 < -L_2 \\ \mu \cdot f_2(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_0 - p_2|} \cdot sign(p_0 - p_2) & \to x_3 \geq -L_2 \end{vmatrix}, \tag{9}$$

Fluid flow through the third working window of the spool pair (L_3)

$$Q_3 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_2 - p_3)}{12 \cdot \nu \cdot \rho \cdot (-L_3 + x_3)} & \to x_3 < L_3 \\ \mu \cdot f_3(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_2 - p_3|} \cdot sign(p_2 - p_3) & \to x_3 \geq L_3 \end{vmatrix}, \tag{10}$$

Fluid flow through the fourth working window of the spool pair (L_4)

$$Q_4 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_3 - p_7)}{12 \cdot \nu \cdot \rho \cdot (L_4 + x_3)} & \to x_3 < -L_4 \\ \mu \cdot f_4(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_3 - p_7|} \cdot sign(p_3 - p_7) & \to x_3 \geq -L_4 \end{vmatrix}, \tag{11}$$

Fluid flow through the fifth working window of the spool pair (L_5)

$$Q_5 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_0 - p_5)}{12 \cdot \nu \cdot \rho \cdot (L_5 - x_3)} & \to -x_3 < L_5 \\ \mu \cdot f_5(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_0 - p_5|} \cdot sign(p_0 - p_5) & \to -x_3 \geq L_3 \end{vmatrix}, \tag{12}$$

Fluid flow through the sixth working window of the spool pair (L_6)

$$Q_6 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_0 - p_6)}{12 \cdot \nu \cdot \rho \cdot (L_6 + x_3)} & \to x_3 < L_6 \\ \mu \cdot f_6(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_0 - p_6|} \cdot sign(p_0 - p_6) & \to x_3 \geq L_6 \end{vmatrix}, \tag{13}$$

Fluid flow through the seventh working window of the spool pair (L_7)

$$Q_7 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_4 - p_7)}{12 \cdot \nu \cdot \rho \cdot (-L_7 + x_3)} & \to x_3 < L_7 \\ \mu \cdot f_7(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_4 - p_7|} \cdot sign(p_4 - p_7) & \to x_3 \geq L_7 \end{vmatrix}, \tag{14}$$

Fluid flow through the eighth working window of the spool pair (L_8)

$$Q_8 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_4 - p_1)}{12 \cdot \nu \cdot \rho \cdot (L_8 + x_3)} & \to x_3 < -L_8 \\ \mu \cdot f_8(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_4 - p_1|} \cdot sign(p_4 - p_1) & \to x_3 \geq -L_8 \end{vmatrix}, \tag{15}$$

In the equations (8) – (15) the actual change in the size of the opening area of the working window of the spool pair is taken into account, provided that the working edge of the spool is made conical (Fig. 1). In this case, the dependence of the opening area of the working window on its movement x will look like this

$$f(x_3) = \begin{vmatrix} \pi \cdot (d_3 + \delta_9) \cdot (\delta \cdot \cos(\gamma) + (x - L) \cdot \sin(\gamma)) & \to x_3 \leq b \\ \pi \cdot (d_3 + \delta_9) \cdot \sqrt{(x_3 - b - L)^2 + (\delta_9 + b \cdot tg(\gamma))^2} & \to x_3 > b \end{vmatrix} \tag{16}$$

Fluid flow through the ninth and tenth edges (with openings L9 and L10), taking into account the fact that the edges are formed by the straight end of the spools, is determined by the dependences:

$$Q_9 = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_5 - p_7)}{12 \cdot \nu \cdot \rho \cdot (L_9 - x_3)} & \to x_3 > L_9 \\ \mu \cdot f_9(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_5 - p_7|} \cdot sign(p_5 - p_7) & \to x_3 \leq L_3 \end{vmatrix}, \tag{17}$$

$$Q_{10} = \begin{vmatrix} \dfrac{\pi \cdot d_3 \cdot \delta_9^3 \cdot (p_6 - p_7)}{12 \cdot \nu \cdot \rho \cdot (L_{10} + x_3)} & \to -x_3 > L_{10} \\ \mu \cdot f_{10}(x_3) \cdot \sqrt{\dfrac{2}{\rho}} \cdot \sqrt{|p_6 - p_7|} \cdot sign(p_6 - p_7) & \to -x_3 \leq L_{10} \end{vmatrix}. \tag{18}$$

The value of the area of the working window of the spool pair, formed by the clipping edge, is determined by the expression:

$$f(x_3) = \begin{vmatrix} \pi \cdot (d_3 + \delta_9) \cdot \sqrt{(-x_3 + L)^2 + (\delta_9)^2} & \to x_3 < L \\ 0.25 \cdot \pi \cdot \sqrt{(d_3 + \delta_9)^2 - (d_3)^2} & \to x_3 = L \end{vmatrix}, \tag{19}$$

Leakage losses through gaps in joints are calculated as fluid flow through a flat slit under accepted assumptions:

- the shape of the surfaces forming the flow channel is perfect;

- surface roughness is not taken into account;

- the gap is symmetrical.

In this case, the flow of fluid through the cross section of the leakage gap will be determined by the following dependencies

$Q_{ут.н} = \sigma_0 \cdot p_0$,

$Q_{ут.д1} = \sigma_1 \cdot p_1$,

$Q_{ут.д2} = \sigma_2 \cdot p_2$,

$Q_{ут.ц.1} = \sigma_3 \cdot p_3$,

$Q_{ут.ц.2} = \sigma_4 \cdot p_4$.

According to [4], the values of the coefficients of leakage σ_i of the liquid through the gaps can be determined by the following formula

$$\sigma_i = \beta_i \cdot \delta_i^3 / 1_i \cdot \mu(p_0, t^0C) \cdot l_0 \pm v_i \cdot \beta_i \cdot \delta_i, \quad i = 0 \ldots 4, \qquad (21)$$

where β_i - the width of the gaps of fluid flow in the direction perpendicular to the direction of fluid flow in the i-th section of the hydraulic system; δ_i - the depth of the fluid leakage gaps in the i-th section of the hydraulic system; l_i - the length of the fluid leakage gaps in the direction of fluid flow in the i-th section of the hydraulic system; $\mu(p,t^0C)$ - dynamic viscosity of the fluid, which depends on the pressure and temperature of the working fluid; V_i – the relative speed of movement of one of the walls, which sets the boundary of the gap. The sign "+" refers to the case when the wall moves in the direction of the zone of reduced pressure, "-" - in the opposite direction.

Viscosity change $\mu(p,t^0C)$ depending on the pressure p and temperature t^0C is determined according to the dependence [4]:

$$\mu(p,t^0C) = \mu_{50,p0} \cdot \exp(b_p \cdot p_i) \cdot (50/t^0C)^{n_B}, \quad i=0..4, \qquad (22)$$

where b_p – coefficient that depends on the properties of the working fluid, b_p = 0.0023 … 0.0033; n_B - an indicator of the degree whose value depends on the initial viscosity of the liquid.

Flow in hydraulic units from the high pressure zone to the low pressure zone due to incomplete tightness of the hydraulic units is determined by the dependences: - in the case of gear hydraulic machines [4]:

$$Q_{пер.д} = \sigma_д \cdot (p_1 - p_2). \qquad (23)$$

- in the case of hydraulic cylinders [5]:

$$Q_{пер.ц} = \sigma_ц \cdot (p_3 - p_4), \qquad (24)$$

where $\sigma_д$, $\sigma_ц$ - the coefficients of fluid flow between the cavities of the dosing mechanism from the high pressure chamber to the low pressure chamber. The values of these coefficients can also be determined by dependencies (21), (22).

The costs incurred in the deformation of the volumes of the hydraulic cavities filled with fluid due to changes in pressure in these cavities are determined by the dependences [5]: (20)

$$Q_{деф.i} = K_i(p_i) \cdot W_i \cdot dp_i/dt, \quad \text{where } i = 0,..,6, \qquad (25)$$

where $K_i(p_i)$ – coefficients of pliability of the corresponding highways and cavities of this hydraulic system; W_0 - the volume of the line from the pumping station to the inlet of the metering pump; W_1 - the volume of the cavity connecting the first working window of the spool with the corresponding cavity of the dosing mechanism; W_2 - the volume of the cavity connecting the second working window with another cavity of the dosing mechanism; W_3 - the volume of the cavity of the hydraulic line connecting the spool distributor with the right cavity of the hydraulic cylinder; W_4 - the volume of the cavity of the hydraulic line connecting the left cavity of the hydraulic cylinder to the spool distributor; W_5, W_6 - the volumes of cavities between the sides of the spool that connect the windows of the "open center" with the clipping edges.

Study of the dependence of the coefficient of pliability on pressure $K_i(p_i)$ engaged in many researchers both experimentally and theoretically. As a result of their research [4], obtained a formula for calculating the coefficient of ductility, which takes into account the total deformation of the working fluid, as a gas-liquid mixture under isothermal compression, and the pipeline (cavity):

$$K_i(p_i) = E_ж^{-1} \cdot (1 - 2 \cdot p_{i-1}/\alpha \cdot (p_{i-1} + p_{i+1}) - 2 \cdot p_{i-1} \cdot E_ж/\alpha \cdot (p_{i+1}^2 - p_i^2) x (1 - p_{i-1}/\alpha \cdot E_ж) \cdot \ln(p_{i-1} + +\alpha \cdot p_{i+1})/(p_{i-1} + \alpha \cdot p_i)), \quad i = 0,..,6. \qquad (26)$$

The second group of equations that are part of the mathematical model of the system of hydrovolume steering is the equation of motion of individual elements. The calculation scheme for determining the current forces and moments is shown in Fig. 2.

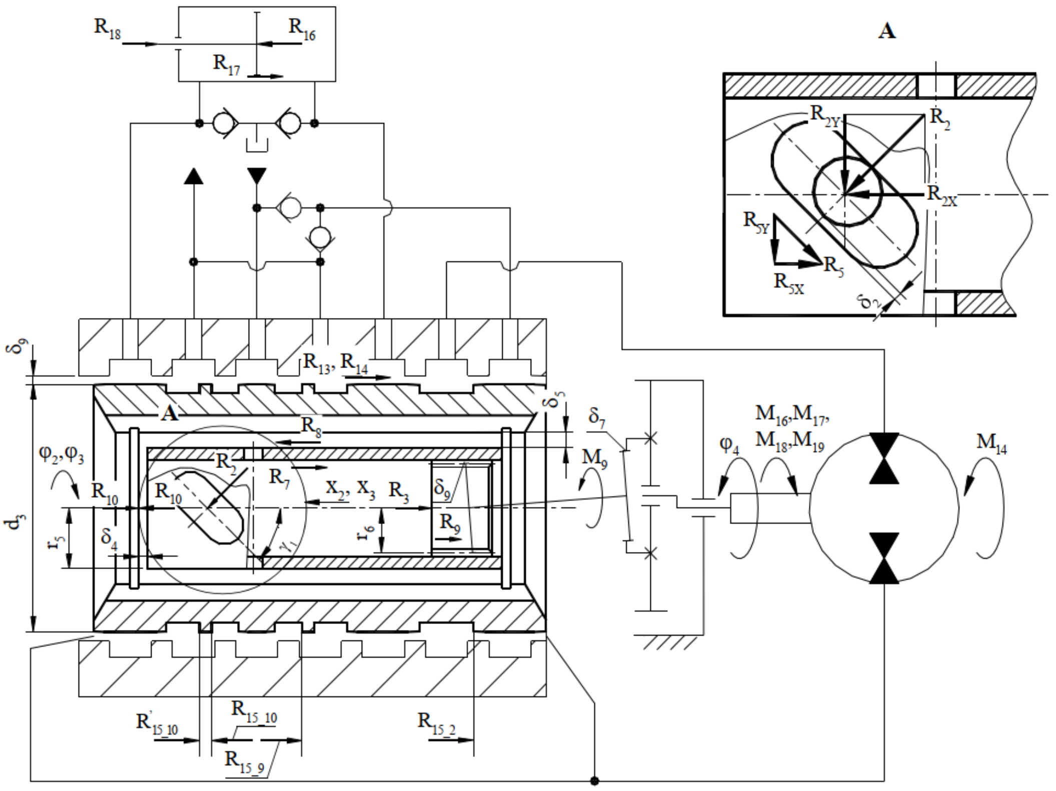

Fig. 2. Estimated scheme of forces and moments acting on the elements hydrovolume steering system.

Equation of moments of forces acting on the sleeve:

$$M_2 = M_3 + M_8 + M_{9.} + \\ + M_{10} + M_5 + M_6 + M_{11} + M_{26}, \tag{27}$$

where M_2 – the moment that occurs when the pin on the input shaft on the surface of the groove in the sleeve; M_3 – torque to overcome dry friction between the end face of the sleeve 2 and the lock washer; M_5 – torque to overcome dry friction between the pin and the groove in the sleeve; M_6 – the moment to overcome the viscous fluid friction between the shaft 1 and the sleeve 2; M_8 – moment to overcome the liquid friction between the sleeve and the spool; M_9 – the moment of resistance from a cardan; M_{10} – the moment of dry friction forces acting between the end face of the sleeve and the spool washer; M_{11} – the moment of inertia of the sleeve; M_{26} – the moment to overcome the dry friction between the end of the spring zero and the end of the sleeve.

The sleeve is also under the action of forces directed along the axis, the balance equation of which has the following form

$$R''_2 = R_3 + R_5 + R_7 + R_8 + R_9 + R_{10} + R_{17}, \tag{28}$$

where R''_2 – axial component of the pin pressure force on the groove; R_3 – the force of resistance of the spring of the zero installer; R_5 – axial component of the dry friction force between the pin and the groove of the sleeve; R_7 – force of liquid friction at relative movement of a shaft and the plug; R_8 – the force of fluid friction during the relative movement of the sleeve and the spool in the axial direction; R_9 – the force of dry friction in the splined pair of the bushing - cardan; R_{10}– the force of resistance from the spool when pressing the sleeve on the snap ring; R_{17} – the force of inertia of the sleeve.

The equation of forces acting on the spool has the form:

$$R_8 + R_{10} = R_{12.} + R_{13} + R_{14} + R_{15}, \tag{29}$$

where R_{12} – the force of inertia of the spool; R_{13} – the force of liquid friction between the spool and the housing; R_{14} – the force of dry friction between the spool and the housing; R_{15} – hydrodynamic force.

Equation of forces applied to the rod of the hydraulic cylinder:

$$R_{16} = R_{18} + R_{19} + R_{20}, \tag{30}$$

where R_{16} – the force developed by the hydraulic cylinder in the presence of a pressure drop in its cavities; R_{18} – external technological load on the hydraulic cylinder rod; R_{19} – the force of inertial load on the rod of the cylinder; R_{20} – friction losses when moving the piston of the hydraulic cylinder.

Equation of moments applied to the dosing mechanism shaft:

$$M_{14} = M_{15} + M_{16} + M_{17} + M_{18}, \tag{31}$$

where M_{14} – the moment that develops the dosing mechanism; M_{15} – the moment of inertia of the moving parts of the dosing mechanism, reduced to its shaft; M_{16} – moment from the forces of liquid friction in the dosing mechanism; M_{17} – moments of external loading of the dosing mechanism from a reducer; M_{18} –

loss of torque, which develops a dosing mechanism to overcome the forces of dry friction.

Equations (27) - (31) include the values of moments and forces acting on the constituent elements of the hydraulic system, determined in accordance with the conditions of their operation as part of the hydraulic steering mechanism.

External control torque on the input shaft M_2, which actuates the spool sleeve is determined by the amount of elastic deformation of the pin, which presses on the side surface of the groove, taking into account the gap between the finger and the groove:

$$M_2 = \begin{vmatrix} 0 & \to |\varphi_1 - \varphi_2| < \dfrac{\delta_2}{r_2 \cdot \cos(\gamma_1)} \\ C_2 \cdot \left(\varphi_1 - \varphi_2 - \dfrac{\delta_2}{r_2 \cdot \cos\gamma_1}\right) & \to |\varphi_1 - \varphi_2| < \dfrac{\delta_2}{r_2 \cdot \cos(\gamma_1)} \end{vmatrix}, \quad (32)$$

where φ_1 – the angle of rotation of the input shaft; φ_2 – the angle of rotation of the sleeve; C_2 – bending stiffness of the pin; δ_2 – gap in a pair of pin grooves; γ_1 – the angle of the helical groove in the sleeve; r_2 – the radius of the sleeve.

The moment of dry friction M_3 arises due to pressing by the spring established with initial tension, the plug to the end face of a lock ring:

$$M_{3.} = f_1 \cdot F_3 \cdot r_3, \quad (33)$$

where f_1 – coefficient of friction between the specified surfaces; r_3 – the value of the radius at which the friction actually occurs; F_3 – the force of elasticity of the spring, which in this case is equal to

$$F_3 = C_1 \cdot (\Delta x + x_2), \quad (34)$$

where C_1 – the coefficient of elasticity of the spring; $\Delta x + x_2$ – complete deformation of the spring, which is determined by the previous deformation Δx and moving the sleeve x_2, which occurs during the operation of the mechanism.

The moment of dry friction of a pin on a groove M_5 occurs when the condition of selecting the gap δ_2 and proportional to the normal force of pressing the pin to the side surface of the groove, which in turn is determined by the amount of torque M_2.

$$M_5 = \begin{vmatrix} 0 & \to |\phi_1 - \phi_2| < \dfrac{\delta_2}{r_2 \cdot \cos(\gamma_1)} \\ f_2 \cdot C_2 \cdot \left(\phi_1 - \phi_2 - \dfrac{\delta_2}{r_2 \cdot \cos\gamma_1}\right) \cdot tg(\gamma_1) & \to |\phi_1 - \phi_2| \ge \dfrac{\delta_2}{r_2 \cdot \cos(\gamma_1)} \end{vmatrix}, \quad (35)$$

Mechanical losses to overcome friction during the relative movement of the surfaces of the parts separated by a layer of working fluid depends on the speed of the relative movement of the parts [5].

The moment of liquid friction between the shaft and the sleeve

$$M_6 = \beta \cdot r_1 \cdot \left(\frac{d\phi_1}{dt} - \frac{d\phi_2}{dt}\right), \quad (36)$$

where β – coefficient of liquid friction (active resistance) during the rotational movement of parts; r1 - radius of friction in the gap shaft 1 - sleeve 2

The moment of liquid friction in the pair of bushings - spool;

$$M_8 = \beta \cdot r_3 \cdot \left(\frac{d\phi_2}{dt} - \frac{d\phi_3}{dt}\right), \quad (37)$$

where ϕ_3 – the angle of rotation of the spool; r1 - radius of friction in the pair of vulka - spool.

The moment of liquid friction in a pair of spools - the case:

$$M_{13} = \beta \cdot \frac{d_3}{2} \cdot \frac{d\phi_3}{dt}, \quad (38)$$

where d_3 – spool diameter.

The moment of liquid friction in the pump - dispenser

$$M_{16} = \beta_Д \cdot \frac{d\phi_4}{dt}, \quad (39)$$

where ϕ_4 - the angle of rotation of the shaft of the dosing mechanism; $\beta_Д$ - the coefficient of mechanical losses due to liquid friction in the dosing mechanism, which takes into account the design features and size of this mechanism.

Moment M_9 occurs when turning the propeller shaft and is determined by the magnitude of the elastic deformation of the pair of propeller - sleeve and is calculated depending:

$$M_9 = \begin{vmatrix} 0 & \to |\phi_5 - \phi_2| < |\delta_6 + \delta_7| \\ C_4 \cdot (\phi_5 - \phi_2) & \to |\phi_5 - \phi_2| \ge |\delta_6 + \delta_7| \end{vmatrix}, \quad (40)$$

where C_4 – torsional stiffness of the propeller shaft; φ_2 – the angle of rotation of the sleeve; φ_5 – the angle of rotation of the propeller shaft; δ_6 – gap in the connection cardan shaft - bushing; δ_7 – gap at the driveshaft drive shaft - gearbox satellite.

The moment that develops as a dosing mechanism depends on the fallow

$$M_{14} = q \cdot (p_1 - p_2). \quad (41)$$

Inertial advancement of elements in hydromechanism, as they are wrapped, signified by advancing fallows.

Moment of forces of inertial tension, applications to the bushing

$$M_{11} = I_2 \cdot \frac{d^2\phi_2}{dt^2}, \quad (42)$$

where I_2 - moment of inertia of the bushing.

Moment of inertial load applied to the propeller shaft:

$$M_{15} = I_5 \cdot \frac{d^2\phi_4}{dt^2}, \quad (43)$$

where I_5 - moment of inertia of the propeller shaft.

Moment of inertial loading forces of the dosing mechanism:

$$M_{23} = I_7 \cdot \frac{d^2\phi_5}{dt^2}, \quad (44)$$

where I_7 - the moment of inertia of the moving parts of the dosing mechanism, reduced to its shaft.

The components of the equations of acting forces are determined by the following dependencies.

The forces of liquid friction are equal:

$$R_7 = \beta_Л \cdot \frac{dx_2}{dt}, \quad (45)$$

$$R_8 = \beta_Л \cdot \left(\frac{dx_2}{dt} - \frac{dx_3}{dt}\right), \quad (46)$$

$$R_{13} = \beta_Л \cdot \frac{dx_3}{dt}, \quad (47)$$

$$R_{17} = \beta_Л \cdot \frac{dx_4}{dt}, \quad (48)$$

where $\beta_Л$ - the coefficient of proportionality of the force of liquid friction of the speed of relative linear movement of the parts of the hydromechanism.

The force of friction at the point of contact of the pin and the groove of the sleeve is in the nature of dry friction. When determining its value, the presence of a gap is taken into account δ_2 in the connection of the specified parts. Then

$$R_5 = \begin{vmatrix} 0 & \to |\phi_1 - \phi_2| < \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \\ f_2 \cdot \frac{M_2}{r_2} & \to |\phi_1 - \phi_2| \ge \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \end{vmatrix} = \begin{vmatrix} 0 & \to |\phi_1 - \phi_2| < \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \\ f_2 \cdot \frac{C_2 \cdot (\phi_1 - \phi_2 - \delta_2)}{r_2} & \to |\phi_1 - \phi_2| \ge \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \end{vmatrix} \quad (49)$$

Accordingly, the force of dry friction during the longitudinal movement of the slotted hole in the sleeve relative to the toothed gimbal, taking into account the gap in their connection is determined by this dependence.

$$R_9 = \begin{vmatrix} 0 & \to |\phi_5 - \phi_2| < |\delta_6 + \delta_7| \\ f_4 \cdot \frac{M_9}{r_6} & \to |\phi_5 - \phi_2| \ge |\delta_6 + \delta_7| \end{vmatrix} = \begin{vmatrix} 0 & \to |\phi_5 - \phi_2| < |\delta_6 + \delta_7| \\ f_4 \cdot \frac{C4 \cdot (\phi_5 - \phi_2 - \delta_6 - \delta_7)}{r_6} & \to |\phi_5 - \phi_2| \ge |\delta_6 + \delta_7| \end{vmatrix}, \quad (50)$$

where f_4 – the coefficient of friction in the pair of cardan - slotted hole.

Strength of resistance R_2, acting on the sleeve from the input shaft is determined by the amount of torque M_2, attached to the steering wheel. Given the backlash (gap) in the connection pin - groove, this relationship takes the form

$$R_2^{"} = \begin{vmatrix} 0 & \to |\phi_1 - \phi_2| < \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \\ \frac{M_2}{r_2} \cdot tg(\gamma_1) & \to |\phi_1 - \phi_2| \ge \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \end{vmatrix} = \begin{vmatrix} 0 & \to |\phi_1 - \phi_2| < \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \\ \frac{C_2 \cdot (\phi_1 - \phi_2 - \frac{\delta_2}{r_2 \cdot \cos\gamma_1})}{r_2} \cdot tg(\gamma_1) & \to |\phi_1 - \phi_2| \ge \frac{\delta_2}{r_2 \cdot \cos\gamma_1} \end{vmatrix}. \quad (51)$$

The force developed by the spring of the zero installer, within the actual displacements of the sleeve, is determined by the linear relationship:

$$R_{3.} = C_1 \cdot x_2, \quad (52)$$

where C_1 – spring stiffness; x_2 – deformation of the spring, which is equal to the movement of the sleeve.

Strength of resistance R_{10}, which occurs due to the elastic deformation of the snap ring under the action of linear movement of the sleeve, taking into account the axial gap δ_4 in the combination of these parts is determined by the dependence

$$R_{10} = \begin{vmatrix} 0 & \to x_2 < \delta_4 \\ C_3 \cdot (x_2 - x_3 - \delta_4) & \to x_2 \ge \delta_4 \end{vmatrix}, \quad (53)$$

where C_3 – stiffness of the retaining ring.

The forces of inertial load, which occur when moving with the acceleration of the bushing, spool and brought to the rod of the hydraulic cylinder mass of the moving parts of the steering mechanism, are determined by the following dependencies:

$$R_{17} = m_2 \cdot \frac{d^2x_2}{dt^2}; \quad (54)$$

$$R_{12} = m_3 \cdot \frac{d^2x_3}{dt^2}; \quad (55)$$

$$R_{19} = m_4 \cdot \frac{d^2x_4}{dt^2}, \quad (56)$$

where m_2 – bushing mass; m_3 – spool weight; m_4 – the mass of moving parts of the steering mechanism is reduced to the rod of the hydraulic cylinder.

In determining the hydrodynamic force is taken into account the fact that in this case the total value of the hydrodynamic force is determined by the simultaneous action of individual hydrodynamic forces at several edges.

In the zero position of the spool, the pressure in the cavities of the distributor acts evenly on all surfaces of the open center, hydrodynamic forces $P_{ГД9}$, $P_{ГД10}$ act on the edges 9 and 10, and they are directed in the opposite direction, and therefore the resultant of these forces is zero. In the case where the spool shifts in a positive direction, the hydrodynamic forces $P_{ГД9}$, $P_{ГД10}$, acting on the edges 9, 10 of the open center, have different sizes, in addition, as you open the spool opens the first edge, begins to rotate the shaft of the pump - dispenser and the liquid enters the cavity between edges 2 - 3, and through edge 3 passes into the cavity of the cylinder, during this time, the hydrodynamic force acts on the edge 2 $P_{ГД2}$. At the exit of the hydraulic cylinder, the liquid enters the cavity between the edges 7 - 10 and passes to the drain, in which case there is a hydrodynamic force $P'_{ГД10}$ on the surface of the edge 10, but on the other hand, a force acts $P_{ГД10}$. Therefore, the resulting hydrodynamic force acting on the spool with its positive displacement is:

$$R'_{15} = P'_{ГД10} - P_{ГД10} + P_{ГД9} + P_{ГД2}. \quad (57)$$

When the spool is negatively moved, the forces act in the same way on the edges of the open center 9, 10. But not the first, but the second edge opens, so the hydrodynamic force acts on the spool $P_{ГД3}$ and accordingly in the drain cavity - hydrodynamic force $P'_{ГД9}$. The resulting hydrodynamic force in the negative movement of the spool is:

$$R''_{15} = -P'_{ГД9} + P_{ГД10} - P_{ГД9} - P_{ГД3} \quad (58)$$

According to the results of a number of studies [5], the magnitude of the hydrodynamic force acting on the individual edges of the spool can be correctly determined by the formula

$$P_{ГД} = -\rho \cdot Q \cdot V \cdot \cos\theta, \quad (59)$$

where ρ - the density of the working fluid; Q - fluid flow through the gap; θ - the angle of direction of fluid flow through the gap between the edges of the spool and the housing, for the spool pair of this design, the angle of the working edge is equal to 3°, that will determine the direction of flow of the working fluid through the slit.

The force developed by the hydraulic cylinder is proportional to the pressure drop in its cavities

$$R_{16} = F_5 \cdot (p_4 - p_3) \quad (60)$$

To simulate the operation of a planetary gearbox, which is used as a transmission mechanism between the dosing mechanism and the feedback cardan, the most appropriate was the use of the Lagrange equation of the second kind.

For our case, the Lagrange equation will take the form:

$$\frac{d}{dt}\left(\frac{\partial}{\partial\phi}(T_1+T_2)\right) = \frac{M_{17}\cdot\delta\phi_4 + M_9\cdot\delta\phi_6}{\delta\phi_4}, \quad (61)$$

where T_1 – kinetic energy led, T_2 – kinetic energy of the gear, φ_4 – the angle of rotation of the input shaft of the gearbox, φ_6 – the angle of rotation of the output shaft of the gearbox.

We accept as the generalized coordinate φ_4, then

$$\phi_6 = \frac{2\cdot e}{r_8}\cdot\phi_4, \quad (62)$$

where e – gear eccentricity, r_8 – the radius of the dividing circle of the gear. Considering (62), equation (61) takes the form:

$$\frac{d}{dt}\left(\frac{\partial}{\partial\phi}(T_1+T_2)\right) = M_{17} + \frac{2\cdot e}{r_8}\cdot M_9 \quad (63)$$

The kinetic energy of the carrier is determined by the dependence:

$$T_1 = \frac{1}{2}\cdot I_8\cdot\left(\frac{d\phi_4}{dt}\right)^2, \quad (64)$$

where I_8 - moment of inertia led; φ_4 - the angle of rotation of the planetary gearbox.

The kinetic energy of the gear is determined by the dependence:

$$T_2 = \frac{1}{2}\cdot m_5\cdot V_{O1}^2 + \frac{1}{2}\cdot I_9\cdot\left(\frac{d\phi_6}{dt}\right)^2, \quad (65)$$

where m_5 - gear weight; V_{O1} - linear speed of rotation of a point on the rim of the gear, defined as the product of the angular velocity of rotation of the carrier by the value of the eccentricity and is calculated depending on $V_{O1} = e\cdot\frac{d\phi_4}{dt}$; I_9 - the moment of inertia of the gear; ϕ_6 - gear rotation angle.

Then addiction (65) takes the form

$$T_2 = \frac{1}{2}\cdot m_5\cdot\left(e\cdot\frac{d\phi_4}{dt}\right)^2 + \frac{1}{2}\cdot I_9\cdot\left(\frac{d\phi_6}{dt}\right)^2 \quad (66)$$

Based on equations (63), (64) and (65), as well as the corresponding kinematic relations for the planetary gear we get the following equation for the planetary gear

$$\frac{1}{2}\cdot\left(\frac{d\phi_4}{dt}\right)^2\cdot\left(I_8 + e^2\cdot\left(m_5 + 4\cdot\frac{I_9}{r_8^2}\right)\right) = M_{17} - \frac{2\cdot e}{r_8}\cdot M_9 \quad (67)$$

Research results and their discussion.

The obtained system of equations (1) - (26), (27) - (60), (67) unambiguously determines the state of the parameters that describe the behavior of the hydraulic system of volumetric steering. In compiling these equations, as well as in determining the dependences that determine the values of the parameters, we used the results of experimental studies of steering mechanisms such as CBA.01, as well as similar ones manufactured by other companies, which were conducted for many years Vinnytsia National Technical University. The performed computational experiments allowed to obtain results that closely coincided with the experimental ones.

The study of this mathematical model has some difficulties due to the high degree of nonlinearity and high order of the system. The mathematical model was studied by conducting a numerical experiment using an original package of applications developed on the basis of the Runge - Kutt - Feldberg method. The developed model and the developed technique of its research allow to analyze in detail the nature and features of dynamic processes during the operation of the hydraulic system and to develop measures to improve the quality of the characteristics of steering hydromechanisms such as MRG.01.

Transients obtained as a result of the program under the action of the associated load - in Fig. 3.

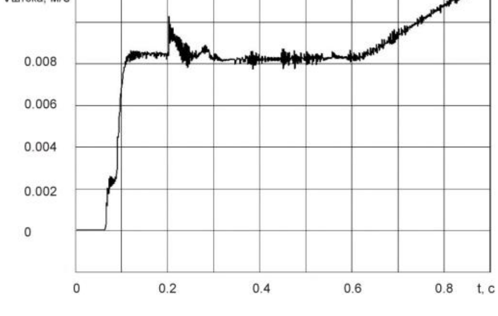

Fig. 3. Transient process of the speed of the executive hydraulic cylinder of the system when using a spool distributor with rectangular grooves on the working edges

When studying the dependence of the stability region in the case of a co-load on the rod of the steering cylinder, it was found that the system responds to this perturbation in a slightly different way than the action of the counter load. Under the action of the associated load, the system is almost stable, but when the load reaches a certain value, the cylinder rod breaks, the fluid breaks, and the machine loses control, so when considering the associated load, it is advisable to consider the failure area of the steering cylinder rod. The transition process illustrating the process of failure of the rod is shown in Fig. 4.

In fig. 4 shows the transient process of the speed of the rod of the executive cylinder when using a spool distributor with conical edges with the following system parameters: q= 80 cm^3/rad, w_1=20 rpm, γ =3°, Fz = 30 cm^2, l_1=0.6 mm, l_2=0.6 mm, l_3=0.6 mm, l_4=0.9 mm, l_5=-2 mm, l_6 =-2 mm, l_7 = 0.9 mm, l_8 = 0.6 mm, l_9= - 0.5 mm, l_{10} = - 0.5 mm, b_1 = 0.185 mm, b_2 = 0.195 mm, b_3 = 0.195 mm, b_4 = 0.165 mm, b_5 = 0.155 mm, b_6 = 0.155 mm, b_7 = 0.165 mm, b_8 = 0.185 mm.

It should be noted that when obtaining the transients, which are presented in Fig. 3 and fig. 4 values of the width of the groove on the edge (parameter a) and the number of grooves (parameter n) were selected so that the cross-sectional area of the modernized spool distributor is equal to the cross-sectional area of the spool distributor window of the basic structure.

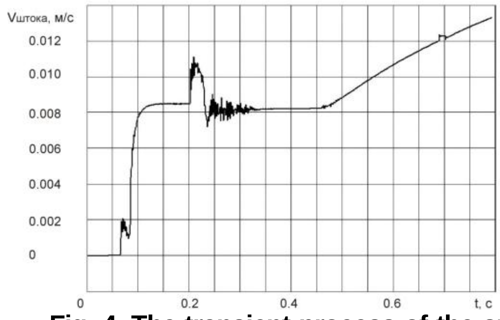

Fig. 4. The transient process of the speed of the executive hydraulic cylinder of the system when using a spool distributor with conical working edges

Comparative analysis of the processes shown in Fig. 2 and 3, shows that the transition to the design of working windows in the form of rectangular grooves, when all other conditions of operation of the hydraulic steering mechanism significantly increases the limit value of the associated load. So the mode of uncontrolled operation of the steering hydromechanism at which speed of movement of a rod of the executive hydraulic cylinder starts to exceed that set by giving of the pump, at the rectangular form of edges comes at the accompanying loading $p_{поп}$ = 45.0 кN, while in the conical shape of the edges, this mode occurs with the associated load $p_{поп}$ = 24.0 кN at the given parameters of system of hydrovolume steering.

When studying the system, it was found that the most informative about the state of the system under the action of the associated load is the dependence of fluid flow through the anti-vacuum valve on the pressure drop across the actuator (Fig. 4).

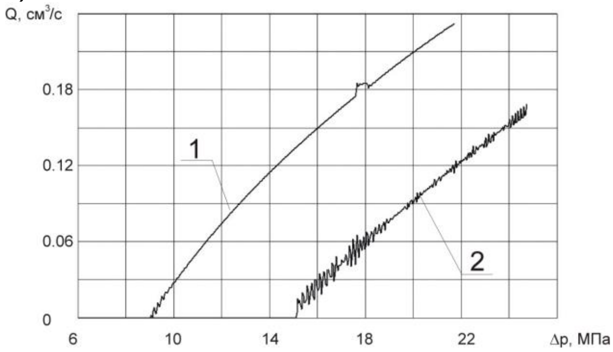

Fig. 5. Dependence of the flow of working fluid through the anti-vacuum valve at: 1 - conical shape of the spool distributor windows; 2 - modernized design of the spool distributor windows.

In case of uncontrolled movement of the hydraulic cylinder rod, the pressure in the drain cavity of the hydraulic cylinder increases rapidly, while in the pressure cavity the pressure drops and when the pressure reaches its minimum allowable value - anti-vacuum valves are triggered. thereby rupture of the working fluid. In fig. 4. the expense characteristic of the anti-vacuum valve for two constructive variants of execution of a spool of the distributor is shown. The first curve is obtained when modeling a spool with conical throttle edges. In the case of using a spool of this design, the phenomenon of uncontrolled movement of the hydraulic cylinder rod occurs when the pressure drop on the actuator of the hydrovolume steering system, which is 8 MPa. While the use of a spool with grooves on the throttle edges (curve 2) allows you to work with a pressure drop in the cavities of the cylinder up to 15 MPa. Also, compared with the results of the study of the basic model, the modernized model when changing the angular velocity of the steering wheel works under the action of the associated load on the cylinder rod better than the base model. As the angular velocity of the steering wheel increases, the pressure drop at which the uncontrolled movement of the hydraulic cylinder stem of the hydraulic steering system begins decreases in both the basic and modernized models, but in the modernized model, the pressure drop on the hydraulic cylinder is 1.8 times greater than in the basic models. Under the action of counterload, the upgraded system works equally efficiently compared to the base, so the proposed change in the shape of the throttle edge of the spool increases the ability of the dosing unit of the hydraulic steering system under the associated load by approximately 1.8 times. spool and does not affect the stability of the system.

Also of great importance for the stability of the system is the amount of fluid supplied. So with increasing size of the dosing unit, and constant supply of fluid to the system, the area of stable operation at the associated load decreases, until there is nowhere to switch to a new size of the pumping station.

The research allows us to draw the following conclusions:

1. The compiled mathematical model accurately reflects the processes and dynamics of the behavior of the hydrovolume steering system.
2. Based on the analysis of the model, the influence of the main parameters of the hydraulic system on the stability of its operation is revealed.
3. Identified dependencies allow to develop recommendations for improving the quality or stability of work by synthesizing the parameters of the hydraulic system.
4. The need for a more differentiated allocation of pumping station costs is identified.
5. In order to increase the manufacturability of the spool pair, it is necessary to increase the angle of the working edge of the spool distributor, but since this parameter adversely affects the stability of the system, it is necessary to change the design of this element.

The analysis of work at the associated load revealed a significant impact of the values of the overlap of the working edges on some work of the system under the action of the associated load, and the correct selection of these floors, you can achieve a significant increase. The obtained results allow recommendations for the choice of standard size of work surfaces.

References

1. Casoli, P.; Gamba A.; Pompini, N.; Ricco, L. (2016) Hybridization methodology based on DP algorithm for hydraulic mobile machinery—Application to a middle size excavator. *Autom. Constr. 61,* 42-57. [in English].
2. Rossetti, A.; Macor, A.; Scamperle, M. (2017) Optimization of components and layouts of hydromechanical transmissions. *Int. J. Fluid Power*,*18,*123-134. [in English].
3. Zardin, Barbara & Borghi, M. & Gherardini, Francesco & Zanasi, Nicholas. (2018). Modelling and Simulation of a Hydrostatic Steering System for Agricultural Tractors. Energies. 11. 230. 10.3390/en11010230.
4. Daher, N.; Ivantysynova, M. (2014) Energy analysis of an original steering technology that saves fuel and boosts efficiency. *Energy Convers. Manag. 86,*1059-1068. [in English].
5. Gultekin, I.Y.; Comert, S.; Erkal, G.; Balkan, T.; Unlusoy, Y.S. (2016) Modeling and simulation of power steering system for agricultural tractors. In Proceedings of the International Conference on Advances in Automotive Technologies, Istanbul, Turkey, 11-14 October 2016; pp. 1-6. [in English].
6. Oksanen, T.; Backman, J. (2013) Guidance system for agricultural tractor with four wheel steering. In Proceedings of the IFAC Bio-Robotics Conference, Sakai, Japan, 27-29 March 2013. [in English].
7. Sakthivel, A.; Sriraman, S.; Verma, R. (2012) Study of Vibration from Steering Wheel of an Agricultural Tractor. *SAE Int. J. Commer. Veh.* 2012, 5, 441-454. [in English].
8. Ammon, D.; Borner, M.; Rauh, J. (2006) Simulation of the perceptible feed-forward and feed-back properties of hydraulic power-steering systems on the vehicle's handling behavior using simple physical models. *Veh. Syst. Dyn.* 2006, 44,158-170. [in English].
9. Panetta, G.; Mancarella, F.; Borghi, M.; Zardin, B.; Pintore, F. (2015) Dynamic Modelling of an Off-Road Vehicle for the Design of a Semi-Active, Hydropneumatic Spring-Damper System. In Proceedings of the ASME International Mechanical Engineering Congress and Exposition, Volume 4B: Dynamics, Vibration, and Control, Houston, TX, USA, 13-19 November 2015. [in English].
10. Sun, Y.; He, P.; Zhang, Y.; Chen, L. (2011) Modeling and Co-simulation of Hydraulic Power Steering System. In Proceedings of the Third International Conference on Measuring Technology and Mechatronics Automation, Shangshai, China, 6-7 January 2011. [in English].
11. Kaletnik H., Sevostianov I., Bulgakov V., Holovach I., Melnik V., Ihnatiev Ye, Olt J. (2020) Development and examination of high-performance fluidisedbed vibration drier for processing food production waste. *Agronomy Research.* 18(4), P. 2391-2409. [in Ukrainian]
12. Bulgakov V., Sevostianov I., Kaletnik G., Babyn I., Ivanovs S., Holovach I., Ihnatiev Y. (2020) Theoretical Studies of the Vibration Process of the Dryer for Waste of Food. *Rural sustainability research* 44(339), [in English].
13. Iskovych-Lototskyi (2006) Osnovy teorii rozrahunku processive ta obladnannya dlya vibroudarnoho pressuvannya. Monografiya [Foudations of calculation theory and elaboration of processes and equipment for vibro-blowing pressing. Monograph]. Vinnytsia: UNIVERSUM-Vinnytsia [in Ukrainian].
14. Ivanov, N., Sharhorodskyi, S., Rutkevych V. (2013) Matematicheskaia model hidroprivoda blochno-portsionoho otdelitelia konservirovannykh kormov. *MOTROL*. –. – № 5. – S. 83–91. [in Ukrainian]
15. Ivanov, M.I., & Sharhorodskyi, S.A., & Rutkevych, V.S. (2013). Pidvyshchennia ekspluatatsiinoi efektyvnosti blochno-portsiinoho

vyvantazhuvacha konservovanych kormiv shliakhom hidrofikatsii pryvoda robochikh orhaniv [Improving the operational efficiency of the block-batch unloader of canned fodder by hydration of the drive of working bodies.]. *Promyslova hidravlika i pnevmatyka – Industrial hydraulics and pneumatics.* №1(39). 91-96 [in Ukrainian].

ДОСЛІДЖЕННЯ РОБОТИ СИСТЕМИ ГІДРООБ'ЄМНОГО РУЛЬОВОГО КЕРУВАННЯ НА БАЗІ НАСОСА-ДОЗАТОРА ІЗ ОСЬОВИМ ЗОЛОТНИКОМ ПРИ ДІЇ ПОПУТНОГО НАВАНТАЖЕННЯ

Одним із факторів забезпечення стабільного розвитку сільського господарства є якісна та надійна техніка. На сьогодні розвиток гідравлічних приводів у значній мірі призвів до всебічного його застосування у сільськогосподарських машинах та тракторах. Однією із систем, яка є повністю гідрофікованою у важких сільськогосподарських машинах – є система рульового керування. Гідрооб'ємна система рульового керування має наступні переваги – компактність, вільність у компоновці машини, високий ККД, точність, можливість передачі значних зусиль на керовані колеса при мінімальних зусиллях механізатора. До недоліків слід віднести певні обмеження при дії попутного навантаження на керовані колеса під час повороту. Даний параметр є одним із показників якості гідрооб'ємних систем рульового керування і мінімальне зусилля при якому починається процес складання коліс регламентується міжнародними стандартами.

У статті виконано математичне моделювання гідрооб'ємної системи рульового керування. Математична модель складається із рівнянь нерозривності потоків робочої рідини, рівнянь моментів та рівняння Лагранжа 2 роду, за допомогою якого описується робота планетарного редуктора. Пошук розв'язків даної моделі здійснювався за допомогою оригінального програмного забезпечення побудованого на основі методу Рунге-Кутти –Фельдберга із автоматичним кроком інтегрування. У результаті аналізу отриманих перехідних процесів були побудовані області стійкої роботи, що дало змогу отримати співвідношення параметрів системи при яких забезпечується її стійка робота. Запропоноване конструктивне рішення дозволило підвищити граничне значення попутного навантаження при перевищенні якого відбувається самовільне переміщення керованих коліс. Проведений аналіз роботи при попутному навантаженні виявив суттєвий вплив значень перекриття робочих кромок на якість роботи системи при дії попутного навантаження, причому правильним підбором цих перекрить, можна досягти значного підвищення якості.

Ключові слова: *гідрооб'ємна система рульового керування, гідравлічний привод, математична модель, перехідний процес, область стійкої роботи, попутне навантаження*

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