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DEVELOPMENT OF THE MATHEMATICAL MODEL OF THE VOLUME HYDROSTATIC TRANSMISSION HST-112 IN THE BRAKING MODE

Based on the analysis of modern designs of domestic and foreign agricultural machines, it was established that one of the most responsible systems that affect the operation and reliability of this equipment is the volumetric hydraulic drive. The hydraulic drive is also the most complex system of modern self-propelled machines, the use of hydrostatic transmission is a particular difficulty for operators HST-112, which is increasingly used on self-propelled machines and tractors. The article describes the structure and principle of operation of the volumetric hydrostatic transmission HST-112, and also presents its mathematical model in the braking mode of a self-propelled agricultural machine. The developed mathematical model makes it possible to understand and predict the operation of volumetric hydrostatic transmission under different conditions of load and pressure of the working fluid. The mathematical model takes into account the variable technological load acting on the working bodies of the volumetric hydrostatic transmission under different operating conditions of the self-propelled agricultural machine. The mathematical model of the HST-112 volumetric hydrostatic transmission as a part of a self-propelled agricultural machine was studied in the Mathcad software, which made it possible to understand what factors affect the efficiency of the transmission and how they can be optimized. As a result of the research of the mathematical model, the characteristic features of the work of the volumetric hydrostatic transmission HST-112 and its individual components in the presence of a significant inertial load were revealed. The mathematical model was developed on the basis of fundamental laws of hydraulics, hydromechanics, theoretical mechanics and real characteristics of drives of mobile agricultural machines. Based on the theoretical analysis of the processes that determine the characteristics of the hydrostatic transmission, it was established that with significant inertial loads on the executive hydraulic motor, processes occur that can lead to a decrease in its working life, wear resistance, and deterioration in reliability. The developed mathematical model calls for further experimental research in order to establish rational parameters of volumetric hydrostatic transmission.

***Key words**: mathematical model, hydraulic drive, hydrostatic transmission HST-112, research, self-propelled agricultural machine, fluid consumption, hydraulic motor, technological load, simulation results.*

Introduction. Today, in the conditions of the food crisis, the issue of the development of agricultural production is one of the priorities, which predetermines the constant improvement of agricultural technology in order to achieve positive dynamics in the development of this area.

One of the important issues in the improvement of self-propelled agricultural machines is the automation of their control. Automation of the control of self-propelled agricultural machines can be carried out using continuously variable transmissions, which include hydrostatic transmission [1, 2]. Modern agricultural machinery, both domestic and imported, mainly has hydrostatic transmissions: HST-90; HST-112; 90R100; 90M100; 6423-618; 6433-113; BMW 70R; BMF75; HPV105; HMF105; AA4VG90; A2FM90. Possibility to select any machine speed at a set number of engine revolutions (using full engine power at maximum efficiency), as well as maintaining traction with speed changes, especially at low drive speeds, ease of reverse and distribution of power flow, reduction wear of tires (as a result of preventing wheel slippage) and elements of brake systems (due to hydraulic deceleration during braking), simplicity and ease of use stimulate an increasing expansion of the field of use of this type of gear in self-propelled agricultural machines [3]. Therefore, the development of a mathematical model for the operation of a volumetric hydrostatic transmission HST-112in the braking mode is of national economic importance and is relevant.

Analysis of recent research and publications. Leading manufacturers of grain and forage harvesters such as: CLAAS, John Deere, New Holland and others in most cases use hydrostatic transmissions on their machines, and also pay great attention to their improvement, since this type of transmission significantly affects the technical characteristics of the machine , its ergonomics, fuel efficiency and economic performance [4].

The vast majority of combine harvesters of domestic and foreign production, such as: KZS-9-1 "Slavutich", PCM-142 "ACROS 530", "Yenisei 960-01"
are equipped with a hydrostatic transmission HST - 112. The absence of a mechanism for breaking the power flow during movement, braking by working friction brakes must necessarily be performed with the synchronous transfer of the HST to the braking mode [2, 4]. Therefore, the issue of studying the features of the operation of hydrostatic transmission as part of self-propelled agricultural machines in the braking mode using mathematical modeling methods is relevant today.

Analysis and systematization of a number of works by such researchers as: Towers T. M., Balykova M. M., Velichka S. A., Volkova V. M., Didura V. A., Khaimovich E. M., Gorbatov V. V., Lebedeva M. S., Finkelstein Z. L., Morsina V. M., Novikov A. M., Prokofiev V. M., Ivanov M. I., Sreda L. P., engineers and designers of plants-manufacturers dedicated to increasing the durability of machine units, which made it possible to determine the scientific problem and directions for further research [4-6].

Today, the design departments of manufacturing plants, industry research and scientific subsections of higher educational institutions are actively involved in improving the reliability of equipment. The analysis performed showed that there is no common idea about the mechanisms and causes of failures of volumetric hydraulic drives, different approaches are used to increase their durability [2, 4, 5]. In world practice, this problem is solved by improving the design of the working elements of machines; for their manufacture, new materials are created that have high strength properties, composite materials are used, coatings with new functional properties are applied to the working surfaces of parts. It has been established that in order to ensure 90-100% of the overhaul life of the units, it is necessary to reduce the wear rate of the working surfaces by at least 1.5-2 times [7]. To create such coatings, it is proposed to use sources of concentrated energy. For example, during the manufacture of hydraulic transmission distributors, two-layer materials with a soft and hard side are used. However, the proposed approaches, for economic reasons, are not implemented in service enterprises and do not allow to radically solve the problem of increasing the durability of volumetric hydraulic drives. The main recommendations of the manufacturer to service enterprises are to replace worn parts with new ones, however, the average overhaul life of repaired GTS-112 remains low, no more than 80% of the pre-repair one. There is an urgent need to develop new, cost-effective technologies for repairing and simulating the operation of hydraulic transmissions of self-propelled agricultural machines under different conditions and operating modes.

Purpose and objectives of the study. The aim of the research is to develop a mathematical model of the operation of the hydrostatic transmission HST-112 in order to assess the effectiveness of using the HST for braking self-propelled agricultural machines.

Presentation of the main material. Volumetric hydrostatic transmission HST-112, which is shown in fig. 1, this is a set of devices designed to transfer the mechanical energy of circulation from the engine drive shaft to the drive wheels by stepless regulation of the magnitude and direction of the flow of the working fluid. The hydrostatic transmission consists of a regulated axial-plunger pump 15 with a feed pump 16, an unregulated axial-plunger hydraulic motor 13 with a valve box, a tank 2, filters 3 and 5, a radiator 17, control equipment and a piping system [2].

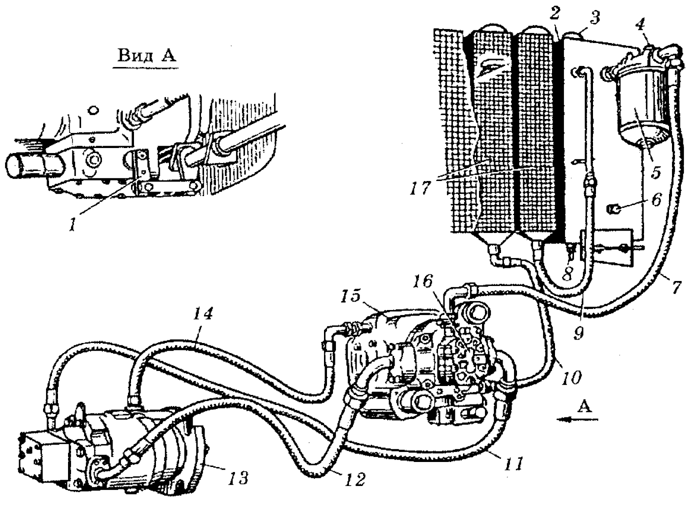


Fig. 1. General view of hydrostatic transmission HST-112
1 - spool control lever, 2 - tank, 3 - coarse filter, 4 - vacuum gauge, 5 - fine filter, 6 - temperature sensor, 7, 9 and 10 - low pressure lines, 8 - drain plug, 11, 12 - lines high pressure, 13 - hydraulic motor, 14 - drainage line, 15 - axial plunger pump, 16 - charge pump, 17 - radiator section

The input shaft of the axial plunger pump is connected to the drive motor, and the hydraulic motor shaft is connected to the mechanism that is driven.

The structural scheme of the GST-112 volumetric hydrostatic transmission is shown in fig. 2.

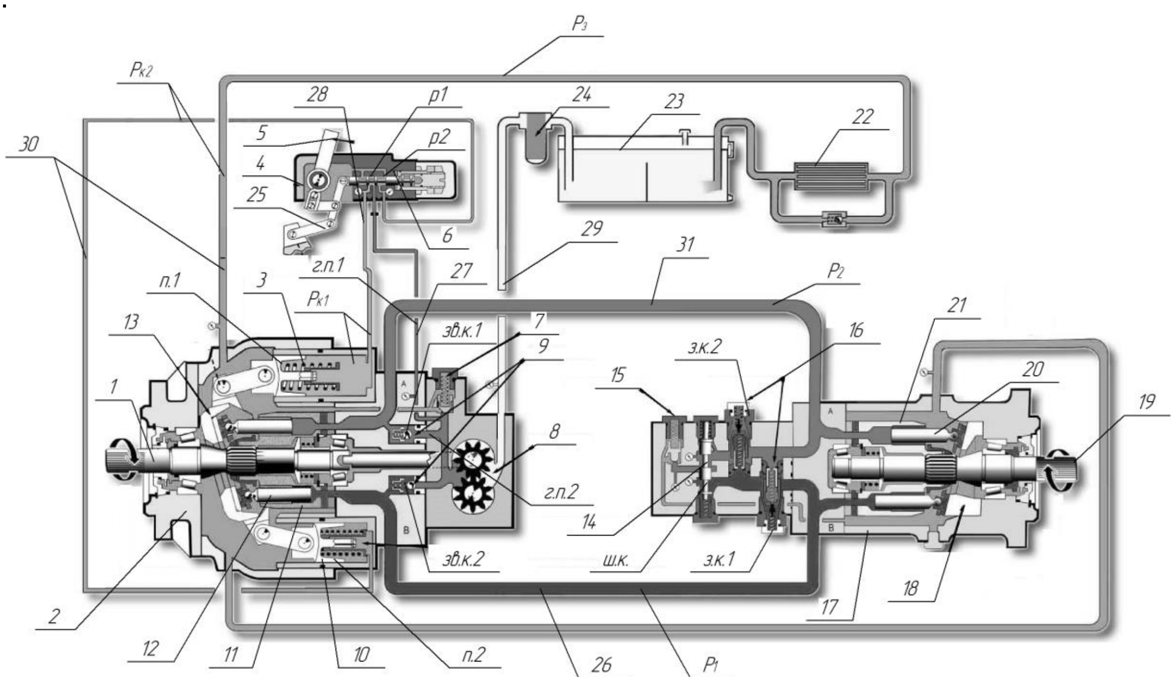


Fig. 2. Structural scheme of hydrostatic transmission HST-112
1 - pump input shaft, 2 - axial plunger pump, 3 - hydraulic booster spring, 4 - control unit, 5 - spool control lever, 6 - control spool, 7 - safety valve of the pump feed system, 8 - charge pump, 9 - make-up system check valves, 10 - hydraulic booster of the pump washer rotation mechanism, 11 - pump cylinder block, 12 - pump plunger, 13 - pump swashplate, 14 - shunt valve, 15 - overflow valve, 16 - high pressure safety valves, 17 - hydraulic motor, 18 - hydraulic motor swash plate, 19 - hydraulic motor input shaft, 20 - hydraulic motor plunger, 21 - hydraulic motor cylinder block, 22 - radiator, 23 - tank, 24 - filter, 25 - feedback link, 26 - high pressure line, 27 and 31 low pressure lines, 28 control line, 29 suction line, 30 drain line

The engine of the machine drives the input shaft 1 of the reversible adjustable pump 2, with which the associated cylinder block 11 and the boost pump 8. The boost pump sucks the working fluid from the tank 23 through the filter 24 and delivers it to the low pressure hydraulic line 27, and through the check valve 9 into the low-pressure hydraulic line 31, which is connected to the suction cavity of the reversible regulated pump and the inlet cavity of the unregulated hydraulic motor 17. The pressure in the hydraulic lines 27 and 31 is determined by setting the overflow valve 15. To protect the low-pressure hydraulic line from overload, a safety valve 7 is used.

In the initial position, the inclined disk is perpendicular to the axis of rotation of the shaft 1, so the pump performance is zero. It is regulated by the control system: when the control lever 5 is moved, the position of the control spool 6 changes, as a result of which the working fluid from the hydraulic line 27 enters the control line 28, and from it to the hydraulic amplifier 10 of the swash plate rotation mechanism. Under the action of the pressure of the working fluid from the control system, the inclined disk moves, which provides an increase in pump performance. With the help of the feedback link 25, the spool 6 is set to a position in which the required angle of the inclined disk 13, set by the control lever 5, is achieved and constantly maintained. Rotating, the cylinder block 11 moves the plungers along the inclined disk, which pump the working fluid into the high-pressure hydraulic line 26. The working fluid from the hydraulic line 26 enters the cylinder block 21 of the hydraulic motor 17 and, moving the plungers along the fixed inclined disk 18, rotates the cylinder block 21 and the shaft 19; at the same time, through the hydraulic line 31, the working fluid returns to the suction line.

When the hydraulic transmission is operating in steady state, the charge pump continuously supplies hydraulic fluid to the low pressure hydraulic line. Excess working fluid through the overflow valve 15 is constantly discharged into the hydraulic motor housing. Leaks of the working fluid, which were formed as a result of leaks in the system, accumulate in the body of the hydraulic motor and are combined with the liquid, which is discharged by the overflow valve, and are fed through the drainage hydraulic line 30 to the pump housing, where they are connected to the pump leaks and through the radiator 22 are drained into the tank 23 providing the required temperature regime of the system. Safety valves 16 serve to protect the system from overloads. The distribution of high and low pressure hydraulic lines and the connection of the low pressure line with overflow valve 15 is carried out by a shunt valve 14 [4, 8].

Let's make mathematical equations for the characteristic areas that determine the operation of the hydraulic transmission.

Flow rate equation in the make-up circuit [2,4]

$$Q_{fp} = Q_{p.1} + Q_{cv1} + Q_{svms} + Q_{def.1} + Q_{out.1} + Q_{svms2}, \quad (1)$$

where Q_{fp} – fluid flow in the feed pump, $Q_{p.1}$ – fluid flow in the first channel of the spool, Q_{cv1} – fluid flow in the first check valve of the axial piston pump, Q_{svms} – liquid flow in the safety valve of the make-up system, $Q_{def.1}$ – liquid consumption to compensate for the deformation of the emptiness of the feeding circuit, $Q_{out.1}$ –fluid outflow rate, Q_{svms2} –fluid flow in the second hydraulic make-up line.

Liquid flow in the pump supply control circuit

$$Q_{p.1} = Q_{hb1} + Q_{\partial ef.2} + Q_{out.2}, \quad (2)$$

where $Q_{p.1}$ – fluid flow in the first channel of the spool, Q_{hb1} – fluid flow in the first hydraulic booster of the pump, $Q_{\partial ef.2}$ – fluid flow to compensate for deformation of the emptiness of the control loop, $Q_{out.2}$ –flow rate of fluid outflow from the void of the circuit control of the pump supply.

Liquid flow rate at the outlet of the control circuit

$$Q_{hb2} = Q_{p.2} + Q_{out3} + Q_{\partial ef.3}, \quad (3)$$

where Q_{hb2} – fluid flow in the second hydraulic booster of the pump, $Q_{p.2}$ – liquid flow in the second channel of the spool, Q_{out3} –flow rate for liquid leakage from the cavity at the outlet of the control loop, Q_{def3} – liquid outflow rate for void deformation compensation at the outlet of the control circuit.

Fluid flow rate for pressurized hydraulic line

$$Q_{app} = Q_{hm} + Q_{rv1} + Q_{out4} + Q_{def.4} + Q_{shv} - Q_{rv.2} - Q_{cv.2}, \quad (4)$$

where Q_{app} – fluid flow in axial piston pump, Q_{hm} – fluid flow in the hydraulic motor, Q_{rv1} – fluid flow in the first hydraulic motor relief valve, Q_{out4} – flow rate of fluid outflow from the cavity of the discharge circuit, $Q_{def.4}$ – liquid outflow rate to compensate for the deformation of the emptiness of the discharge circuit, Q_{shv} – fluid flow in the shunt valve of the hydraulic motor, $Q_{rv.2}$ –fluid flow in the second hydraulic motor relief valve, $Q_{cv.2}$ – fluid flow at the second check valve of the axial piston pump.

Fluid flow rate for low pressure hydraulic line

$$Q_{hm} - Q_{shv} - Q_{rv.2} - Q_{def.5} - Q_{out.5} + Q_{rv.1} + Q_{cv.1} = Q_{app} \quad (5)$$

where $Q_{h.m}$ – fluid flow in the hydraulic motor, $Q_{sh.v}$ – fluid flow in the shunt valve of the hydraulic motor, $Q_{rv.2}$ – fluid flow at the second hydraulic motor relief valve, $Q_{def.5}$ – liquid outflow rate to compensate for the deformation of the cavity of the low-pressure hydraulic line, $Q_{out.4}$ – flow rate of fluid outflow from the cavity of the low-pressure hydraulic line, $Q_{rv.1}$ – fluid flow at the first hydraulic motor relief valve, $Q_{cv.1}$ – fluid flow at the first check valve of the axial piston pump, Q_{app} – liquid flow rate on the axial piston pump.

The actual flow rate of the liquid that is supplied from the boost pump and the axial piston pump is determined according to the following expressions

$$Q_{f.p.} = \frac{V_{0f.p.}}{2\pi} \cdot \omega_{f.p.}, \quad (6)$$

$$Q_{app} = \frac{V_{0p}}{2\pi} \cdot \omega_p, \quad (7)$$

where $V_{0f.p.}$, V_{0p} – working volume of the feed pump and axial piston pump, respectively, $\omega_{f.p.}$, ω_p – the angular velocity of the feed pump shaft and the shaft of the axial piston pump, respectively [11].

The flow rates of the working fluid that occur when the fluid flows through the first and second channels of the spool:

$$Q_{p.1} = \mu \cdot f_{p.1} \sqrt{\frac{2}{\rho}} \sqrt{P_n - P_{c.1}}, \quad (8)$$

$$Q_{p.2} = \mu \cdot f_{p.2} \sqrt{\frac{2}{\rho}} \sqrt{P_{c.2} - P_{ld}} \quad (9)$$

where μ – flow rate, $f_{p.1}$, $f_{p.2}$ – the area of the 1st and 2nd working window of the distributor, respectively, ρ –working fluid density, P_n – make-up pressure, $P_{c.1}$, $P_{c.2}$ – pressure at the outlet of the distributor in the first and second control lines, respectively, P_{ld} – pressure in the hydraulic line drain.

The fluid costs that occur when fluid flows through the charge pump valve, axial piston pump check valves, shunt valve and hydraulic motor safety valves are determined by the expressions.

$$Q_{cvp} = \left| \begin{array}{l} \mu \cdot f_{cvp} \sqrt{\frac{2}{\rho}} \sqrt{P_n - P_3} \Rightarrow P_3 < P_n \\ 0 \Rightarrow P_3 > P_n \end{array} \right., \quad (10)$$

$$Q_{cv.1} = \left| \begin{array}{l} \mu \cdot f_{cv.1} \sqrt{\frac{2}{\rho}} \sqrt{P_n - P_2} \Rightarrow P_2 < P_n \\ 0 \Rightarrow P_2 > P_n \end{array} \right., \quad (11)$$

$$Q_{cv.2} = \left| \begin{array}{l} \mu \cdot f_{cv.2} \sqrt{\frac{2}{\rho}} \sqrt{P_n - P_1} \Rightarrow P_1 < P_n \\ 0 \Rightarrow P_1 > P_n \end{array} \right., \quad (12)$$

$$Q_{shv} = \left| \begin{array}{l} \mu \cdot f_{shv} \sqrt{\frac{2}{\rho}} \sqrt{P_1 - P_{ш.к}} \Rightarrow P_1 < P_{shv} \\ 0 \Rightarrow P_1 > P_{shv} \end{array} \right. \quad (13)$$

$$Q_{sv.1} = \left| \begin{array}{l} \mu \cdot f_{sv.1} \sqrt{\frac{2}{\rho}} \sqrt{P_1 - P_{sv.1}} \Rightarrow P_1 > P_{sv.1} \\ 0 \Rightarrow P_1 < P_{sv.1} \end{array} \right. \quad (14)$$

$$Q_{sv.2} = \left| \begin{array}{l} \mu \cdot f_{sv.2} \sqrt{\frac{2}{\rho}} \sqrt{P_1 - P_{sv.2}} \Rightarrow P_2 > P_{sv.2} \\ 0 \Rightarrow P_2 < P_{sv.2} \end{array} \right., \quad (15)$$

where μ – flow rate, f_{cvp} – area of the working window of the charge pump valve, $f_{cv.1}$, $f_{cv.2}$ –the area of the working window of the first and second check valves of the axial piston pump, f_{shv} – the area of the working window of the shunt valve, $f_{sv.1}$, $f_{sv.2}$ – the area of the working window of the 1st and 2nd hydraulic motor safety valve, ρ – fluid density, P_n – boost pressure, P_3 – pressure in the hydraulic line of the drain (drainage), P_1 –pressure in the high pressure hydraulic line, P_2 –pressure in the low pressure hydraulic line, $P_{ш.к}$ – shunt valve actuation pressure, $P_{sv.1}$, $P_{sv.2}$ –opening pressure of the first and second hydraulic motor relief valves, respectively.

Losses due to the outflow of liquid through gaps in the joints of parts of hydraulic equipment and hydraulic mechanisms are calculated as the flow rate of liquid through a flat slot under the assumptions made:

- the shape of the surfaces that form the outflow channel is perfect;
- surface roughness is not taken into account;
- gap is symmetrical.

In this case, the cost of fluid for the outflow through the cross section of the gap will be determined by the following dependencies:

$$Q_{out.1} = \sigma_{out.} \cdot P_n, \quad (16)$$

$$Q_{out.2} = \sigma_{out.} \cdot P_{\kappa.1}, \quad (17)$$

$$Q_{out.3} = \sigma_{out.} \cdot P_{\kappa.2}, \quad (18)$$

$$Q_{out.4} = \sigma_{out.} \cdot P_1, \quad (19)$$

$$Q_{out.5} = \sigma_{out.} \cdot P_2, \quad (20)$$

where σ_{out} – leakage rate, P_n – make-up pressure, $P_{\kappa.1}, P_{\kappa.2}$ – pressure in the first and second control lines, respectively, P_1, P_2 – pressure in the hydraulic line of low and high pressure.

The costs that arise during the deformation of the volumes of the cavities of the hydraulic drive filled with liquid, due to a change in pressure in these cavities, are determined by such dependencies

$$Q_{def.1} = K_1 \cdot W_n \frac{dP_n}{dt}, \quad (21)$$

$$Q_{def.2} = K_2 \cdot W_{\kappa.1} \frac{dP_{\kappa.1}}{dt}, \quad (22)$$

$$Q_{def.3} = K_3 \cdot W_{\kappa.2} \frac{dP_{\kappa.2}}{dt}, \quad (23)$$

$$Q_{def.4} = K_4 \cdot W_1 \frac{dP_1}{dt}, \quad (24)$$

$$Q_{def.5} = K_5 \cdot W_2 \frac{dP_2}{dt}, \quad (25)$$

where K_1, K_2, K_3, K_4, K_5 – compliance coefficients of the corresponding line and cavity of the given hydraulic system, W_n –make-up cavity volume, $W_{\kappa.1}$ – volume of the first control cavity, $W_{\kappa.2}$ – volume of the second control cavity, W_1 – the volume of the discharge line cavity, W_2 – low pressure line cavity volume, dP_n/dt, $dP_{\kappa.1}/dt$, $dP_{\kappa.2}/dt$, dP_1/dt, dP_2/dt – rate of pressure change in cavities.

The flow rates that occur during the flow of fluid in the hydraulic boosters of the axial piston pump are determined by the following expressions

$$Q_{n.1} = S_{n.1} \cdot \frac{dx_{n.1}}{dt}, \quad (26)$$

$$Q_{n.2} = S_{n.2} \cdot \frac{dx_{n.2}}{dt}, \quad (27)$$

where $S_{n.1}$, $S_{n.2}$ – the area of the pistons of the first and second hydraulic boosters of the axial piston pump, $dx_{n.1}/dt$, $dx_{n.2}/dt$ – displacement speed of the pistons of the first and second hydraulic boosters of the axial piston pump.

The actual fluid flow rate of the hydraulic motor is determined according to the expression

$$Q_{hm} = \frac{V_{0hm}}{2\pi} \cdot \omega_{hm}, \quad (28)$$

where V_{hm} – displacement of hydraulic motor, ω_{hm} –angular velocity of the hydraulic motor shaft.

The equation for the operation of the mechanism for controlling the supply of an axial piston pump will be considered from the analysis of the balance of forces:

$$S_{p.1} \cdot P_{\kappa.1} + F_{spr.1} + F_{fr.1} = S_{p.2} \cdot P_{\kappa.2} + F_{spr.2} + F_{mp.2}, \quad (29)$$

where $S_{p.1}$, $S_{p.2}$ – the area of the pistons of the first and second hydraulic boosters of the axial piston pump, respectively, $P_{\kappa.1}$, $P_{\kappa.2}$ – pressure in the first and second control lines, $F_{spr.1}$, $F_{spr.1}$ – the force of the spring of the pistons of the first and second hydraulic boosters of the axial piston pump, $F_{mp.1}$, $F_{mp.2}$ – friction force of the pistons of the first and second hydraulic boosters of the axial piston pump.

The equation of motion of the hydraulic motor shaft is considered from the analysis of the balance of moments:

$$\frac{V_{0hm}}{2\pi} \cdot \left(P_1 - P_2\right) = M_{in.hm} + M_{fr.hm} + M_{tex.}, \quad (30)$$

where $M_{in.hm}$ – moment of inertia of the hydraulic motor, $M_{mp.гм}$ – moment of friction in the hydraulic motor, $M_{mex.}$ – moment from the technological load.

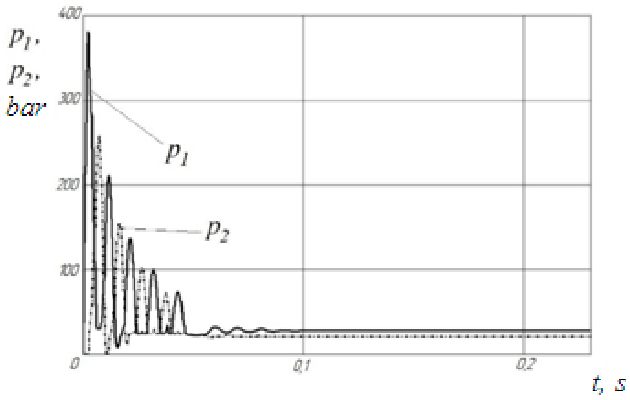
$$\frac{V_{0h.m}}{2\pi} \cdot \left(P_1 - P_2\right) = I \frac{d\omega_{hm}}{dt} + \alpha \cdot \omega_{hm} + M_{tex.} \quad (31)$$

where I – reduced moment of inertia of the hydraulic motor, $\frac{d\omega_{hm}}{dt}$ – angular speed of rotation of the shaft of the hydraulic motor, α – viscous friction coefficient.

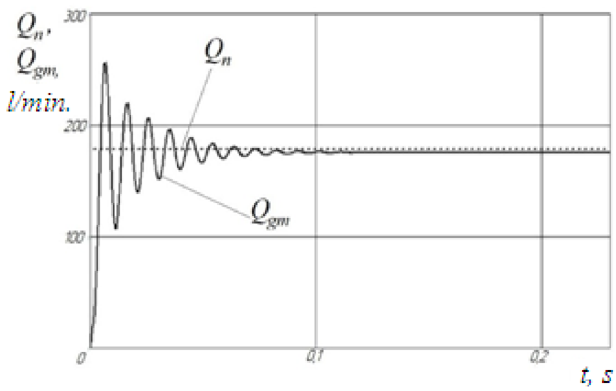
The results of the research of the mathematical model of the HST-112 hydrostatic transmission make it possible to analyze the nature of the change in pressure over time in the hydraulic lines and cavities of the hydraulic system, the flow rate, the movement of the shut-off and regulating elements of valves and distributors. At the first stage of the computer simulation of the operation of the hydrostatic transmission, 137 series of calculations were carried out.

Since many years of successful experience in using hydrostatic transmissions as part of drives of various types of machines are known, the operation of the model was first checked under the condition of no load on the shaft of the executive hydraulic motor, i.e. when idling.

In fig. 3 shows the graphs of the transient processes of pressure changes in pressure and suction hydraulic lines (Fig. 3, a) and the flow rate of the working fluid (Fig. 3, b).



a)



b)

Fig 3.Oscillograms of pressure changes in pressure and suction hydraulic lines and the flow rate of the working fluid when starting the hydrostatic transmission without load on the hydraulic motor shaft: a) pressure change in pressure p_1 and suction p_2 hydraulic lines, b) change in pump supply Q_n and flow rate Q_{gm} consumed by the hydraulic motor

In the computer simulation, the least loaded mode of starting the hydrostatic transmission was selected - an instantaneous increase in the pump supply. At the same time, there are fluctuations in the speed of rotation of the hydraulic motor shaft, which is determined by fluctuations in the flow rate of the working fluid Q_{gm}. Also, the process of pressure change p_1 (pressure hydraulic line) and p_2 (suction hydraulic line) has an oscillatory character. The process shows the stable nature of the hydrotransmission operation - oscillations subside and the hydrostatic transmission enters a stable mode of operation - pressure values and flow rates stabilize. Moreover, this happens quickly enough - in 0.08 seconds, which indicates high speed.

In fig. 3 b it is clearly visible that in the steady state the flow rate of the pump Q_n exceeds the consumption of the working fluid by the hydraulic motor Q_{gm}. The difference in the specified flow rates exactly corresponds to the volume flow rates of liquid from cavities under high pressure. This fact emphasizes the adequate reproduction of hydrostatic transmission work processes during computer simulation.

Conclusions. The developed mathematical model of the hydrostatic transmission HST-112 takes into account the main characteristics of the volumetric hydraulic drive and the features of the technological load under different transmission operating conditions. This will allow us to study the influence of the design parameters of the transmission on its operation under different braking modes of self-propelled agricultural machines.

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РОЗРОБКА МАТЕМАТИЧНОЇ МОДЕЛІ РОБОТИ ОБ'ЄМНОЇ ГІДРОСТАТИЧНОЇ ТРАНСМІСІЇ ГСТ-112 В РЕЖИМІ ГАЛЬМУВАННЯ

На основі аналізу сучасних конструкцій вітчизняних та закордонних сільськогосподарських машин встановлено, що одною з найбільш відповідальних систем, які впливають на роботу і надійність даної техніки є об'ємний гідропривід. Гідравлічний привод є також найбільш складною системою сучасних самохідних машин, особливу складність для експлуататорів представляє використання гідростатичної трансмісії ГСТ-112, яка знаходить все ширше застосування на самохідних машинах та тракторах.

В статті описано будову та принцип роботи об'ємної гідростатичної трансмісії ГСТ-112, а також представлено її математичну модель в режимі гальмування самохідної сільськогосподарської машини. Розроблена математична модель дозволяє зрозуміти і передбачити роботу об'ємної гідростатичної трансмісії при різних умовах навантаження та тиску робочої рідини. Математична модель враховує змінне технологічне навантаження, що діє на робочі органи об'ємної гідростатичної трансмісії при різних умовах роботи самохідної сільськогосподарської машини.

Досліджено математичну модель роботи об'ємної гідростатичної трансмісії ГСТ-112 в складі самохідної сільськогосподарської машини в програмному продукті Mathcad, що дозволило зрозуміти, які фактори впливають на ефективність роботи трансмісії та як їх можна оптимізувати.

В результаті дослідження математичної моделі виявлені характерні особливості роботи об'ємної гідростатичної трансмісії ГСТ-112 і її окремих складових при наявності значного інерційного *навантаження. Математична модель розроблена на основі фундаментальних закономірностей гідравліки, гідромеханіки, теоретичної механіки та реальних характеристик приводів мобільних сільськогосподарських машин. На основі теоретичного аналізу процесів, що визначають характеристики гідростатичної трансмісії встановлено, що при значних інерційних навантаженнях на виконавчому гідромоторі виникають процеси, що можуть призводити до зменшення ресурсу її напрацювання, зносостійкості та погіршення надійності в роботі.*

Розроблена математична модель викликає необхідність подальших експериментальних досліджень з метою встановлення раціональних параметрів об'ємної гідростатичної трансмісії.

Ключові слова: *математична модель, гідравлічний привод, гідростатична трансмісія ГСТ-112, дослідження, самохідна сільськогосподарська машина, витрата рідини, гідромотор, технологічне навантаження, результати моделювання.*

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