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STUDY OF THE TRAJECTORY OF MOVEMENT OF THE REDUCED PLUNGER PRESSURE FORCE ON THE AXIAL PISTON PUMP CRADLE

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Today, the hydraulic drive is increasingly used in agricultural machinery as motion drives, drives of working bodies, and drives of control systems. The power source of this type of drives is pumping stations based on piston-type pumps. This type of pump units has a number of advantages and disadvantages compared to gear pumps. The advantages include power, reliability, higher efficiency, the ability to smoothly regulate the flow of working fluid, and the disadvantages include increased requirements for the degree of purification of working fluids.

The article reviews known hydraulic schemes of agricultural machines, analyzes them, and identifies their advantages and disadvantages. One of the elements of these hydraulic systems is the pump supply control system depending on the load on the working bodies of the hydraulic system elements. The correct choice of the parameters of the elements of this system has a significant impact on the quality of its work as a whole. In order to study the influence of system parameters, the structure and principle of operation of the system for regulating the supply of the working liquid of the axial-piston pump are described, the coordinates of the movement of the center of the reduced force from the action of the plunger pairs on the cradle of the control system for the supply of the working liquid are calculated. The trajectory of its movement along the plane of the feeding adjustment cradle is determined. The proposed dependencies make it possible to determine the magnitude of the moment acting on the cradle depending on the magnitude of the eccentricity. it was found that the point of application of the combined force to the end of the cradle moves along a loop-like trajectory, symmetrical to the axis, parallel to the axis of rotation of the cradle. At the same time, the magnitude of the moment of the combined force acting on the cradle has a cyclic nature, the periodicity of which is determined by the angle of rotation of the cylinder block by a value equal to the angular distance between adjacent cylinders. The frequency of fluctuations of the moment of the combined force when rotating the cylinder block of the PVC 1.63 pump with a frequency of 1500 rpm is 450 Hz, which requires the development of measures to eliminate their negative impact on the operation of the pump supply regulator.

The moment of combined forces acting on the cradle from the side of the cylinder block depends on the pressure in the pump discharge line and the angle of rotation of the cylinder block. At a nominal pressure of 25 MPa, the range of fluctuations of the combined force moment acting on the PVC 1.63 pump cradle reaches 290 Nm. If the axes of the cylinder block and the trunnion of the cradle are in the same plane, the moment of combined forces acting on the cradle will change symmetrically relative to the axis of rotation of the cradle and periodically cause it to deviate in opposite directions. This nature of the load of the cradle interferes with the process of adjusting the angle of its inclination and, accordingly, the working volume of the pump.

In order to reduce the negative impact of the moment of the combined force on the quality of the regulation of the supply of the working fluid of the PVC 1.63 pump, it is recommended to introduce a shift of the axis of rotation of the cradle relative to the axis of the cylinder block by 5 mm.

Key words: *hydraulic drive, working fluid, axial-piston pump, research, self-propelled agricultural machine, fluid consumption, hydraulic motor, technological load.*

F. 14. Fig. 7. Ref. 12.

1. Problem formulation

The growth of competition in the world market of agricultural machinery is due to the introduction of new agricultural technologies, rapid development of machinery and technologies, increased requirements for the quality of machines, ensuring traffic safety both during the use of the machine for its technological purpose and during transportation. In market conditions, manufacturers are trying to create the most safe, functional and high-quality car. Therefore, the use of a hydraulic drive, thanks to a significant number of advantages, has become widely used in agricultural machines.

The use of a hydraulic drive is impossible without a high-quality pumping station. In fact, a pumping station based on an axial-piston hydraulic pump is the only pumping station in case of the need to regulate the speed of the working organs of an agricultural machine. Therefore, the issue of designing and improving the design of pumps of this type is urgent.

2. Analysis of recent research and publications

The most widespread in mobile self-propelled machines are rotary-piston volumetric hydraulic drives, their share is up to 30% of the total number [1, 2]. An example of such a hydraulic drive is the hydrostatic transmission of the HGST-90 type, which belongs to axial-piston hydraulic machines.

A feature of axial hydraulic machines is the arrangement of cylinders and pistons (plungers) in them, located in the rotor (cylinder block) parallel to the axis of its rotation.

The maximum angle of inclination for the cradle of the axial-piston hydraulic pump is no more than 450.

To date, the manufacturers of volumetric hydraulic drives are: in Ukraine, PrJSC «Hydrosila» (Kropivnytskyi), since 2009 - separated into a separate enterprise PrJSC «Hydrosila APM», «Sauer-Danfoss», «PSM-Hydroulics», Rexroth - Bosch Group (Germany), Kawasaki Precision Machinery (Japan), HIDRAULICA UMPLOPENI (Romania), EUROPARTS (Slovakia), etc. [1-4].

The study of special literature [5-10] shows that HGST-90 and its analogues are used on various types of domestic agricultural and construction and road machinery: ACROS harvesters of the 530, 535, 560, 580 model series, KZS-9-1 «Slavutich», «VECTOR-410, 420», SK-5M-1 (with "Kherson Combines" driving bridge); asphalt pavers SD-404, DS-504/506.

A typical example of the use of this type of hydrostatic transmission as a self-propelled machine drive is the KZS-9-1 «Slavutych» grain harvesters, which have been produced by the Kherson Machine-Building Plant since 1995 and formed the basis of the latest grain-harvesting equipment.

The hydraulic drive of the grain harvester KZS-9-1 «Slavutich» (fig. 1) includes an adjustable axial-piston pump NA NP-112 [5], which is actuated through the cardan shaft by the crankshaft of the diesel engine, also mounted on the pump shaft gear-type NP feed pump with PC1 safety valve. Also, the pump includes non-return valves OK1, OK2 and a supply control system, which consists of a distributor P3, which controls the angle of inclination of the pump cradle and throttles J1, J2, J3.

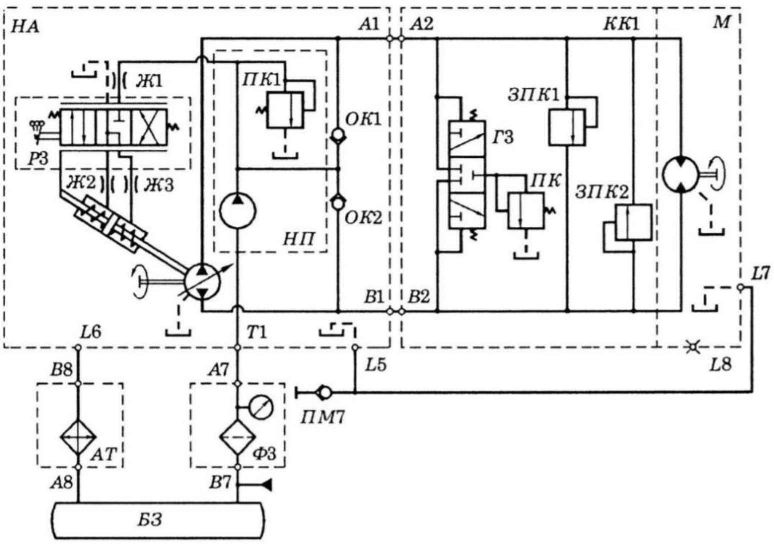


Fig. 1. Schematic diagram of the hydraulic drive of the drive wheels of the KZS-9-1 «Slavutych» grain harvester

The pump is powered by an unregulated axial-piston hydraulic motor M MP-112, which drives the drive wheels of the combine. The valve box KK1 is attached to the hydraulic motor, in which the high-pressure safety valves ZPK1 and ZPK2, the shunting hydraulic distributor Г3 and the overflow valve PC are installed. To fill the hydraulic drive with working fluid, a PM7 half-coupling is provided. Also, the hydraulic drive of the stroke includes a FZ filter of the working fluid with a vacuum meter, a heat exchange device AT and a common tank B3 for the entire hydraulic drive of the machine.

Similar in design and principle of operation to the above-mentioned hydraulic drive is the hydrostatic transmission of the grain harvester drive of the «CLAAS» model «MEDION», the schematic diagram of which is shown in figure 2. It consists of an axial-piston adjustable pump 1 and an unregulated reversible hydraulic motor 10. The new thing in this hydraulic drive, unlike the previous ones, is the use of a cold start safety valve 8, high-pressure safety valves 13 and 14 are combined in one body with non-return valves, and a non-return valve on filter 2 is also installed.

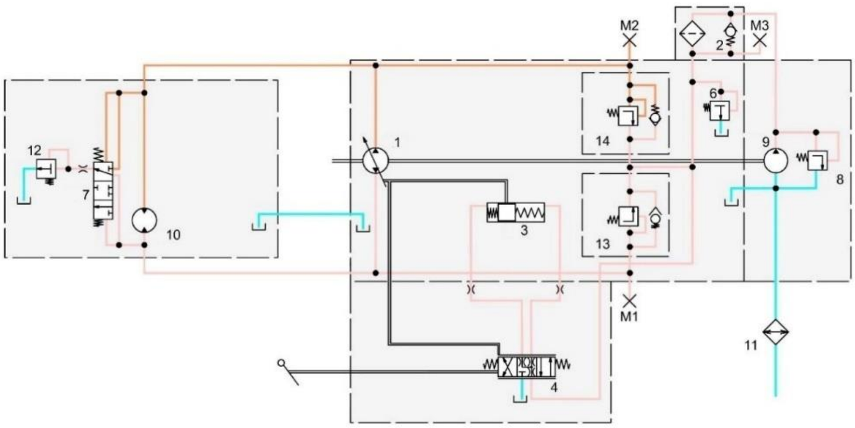


Fig. 2. Schematic diagram of the hydrostatic transmission of the grain harvester drive of the «CLAAS» model «MEDION»:

1 – adjustable pump; 2 – filter with non-return valve; 3 – pump control hydraulic cylinder; 4 – pump control distributor; 6 – safety valve of the refueling pump; 7 – shunt distributor; 8 – cold start safety valve; 9 – feed pump; 10 – non-regulated hydraulic motor; 11 – radiator of working fluid; 12 – overflow valve; 13 – reverse safety valve; 14 – forward safety valve

The use of a hydraulic drive on self-propelled machines for drive and as a drive of working bodies has long been a global trend in agricultural, construction and road engineering. Today, the volumetric hydraulic drive is actively developing, and every year the parameters of the hydraulic system, pressure and flow requirements, reduction of vibrations and cavitation processes are increasing, which necessitates their research and improvement to meet new requirements.

Such scientists as T. Bashta, I. Nemyrovsky, L. Sereda, N. Ivanov, O. Yakhno, Z. Lur' made a significant contribution to the research and development of the hydraulic volumetric drive of domestic agricultural machines is Z. Lur', H. Avrunin etc.

The works of N. Ivanov are aimed at research and improvement of existing hydraulic drive systems of agricultural machines. The paper [6] describes the simulation study of wave processes in long hydraulic lines of hydraulic systems of agricultural machines. The article provides calculation schemes, a mathematical model of a hydraulic drive for the study of wave processes, and the results of studies that characterize the parameters of the hydraulic system depending on the length of the hydraulic lines. In [7], the tilt control system of the axial rotary-piston pump PVC-1.63, which is used to drive the active working bodies of agricultural machines, was studied. As a result, the pulsating nature of the combined force of the pistons of the cylinder block on the pump cradle was revealed, which causes a torque directed to the rotation of the cradle relative to the trunnion. Article [8] is devoted to modeling the operation of the hydraulic transmission of a self-propelled machine in braking mode. A description of the hydrostatic transmission of a self-propelled agricultural machine was carried out, a mathematical model was developed that takes into account the peculiarities of the technological load under different working conditions.

In work [9] H. Avrunin conducted an analysis of changes in power losses in volumetric hydraulic machines. The results of the study make it possible to clarify the share of mechanical friction losses in the overall balance of losses and establish the direction of further work to reduce friction losses, wear of friction nodes and increase the durability of hydraulic motors.

In work [10], scientists presented a dynamic nonlinear mathematical model of a volume hydraulic drive with machine regulation. As a result, a mathematical model of the dynamics of a volumetric hydraulic drive was developed, which includes a regulated pump and an unregulated hydraulic motor, which makes it possible to study the dynamics of a volumetric hydraulic drive in the acceleration mode and to detect high-frequency fluctuations of the pressure of the working fluid and the frequency of rotation of the hydraulic motor, and to establish ways of further scientific research search in order to reduce fluctuations.

Research by foreign scientists is also of great importance for the development of hydraulic drives. In work [11], mathematical modeling of the dynamics of transient processes in the hydrostatic transmission was carried out to substantiate the diagnostic parameters of the technical condition assessment. It was established that the losses of the working fluid in the pump and the hydraulic motor have the greatest effect on the time of the transition process, the amplitude of pressure fluctuations in the injection line, the frequency of rotation of the shaft of the hydraulic motor, and the value of the maximum pressure in the hydraulic system.

Also, great attention is paid to improving the design of hydrostatic transmissions. Modeling and research of a hydrostatic transmission with installed hydraulic accumulators was carried out in order to achieve greater energy efficiency. The results showed that the low efficiency of traditional HSTs under partial load conditions can be improved by using a configuration with hydraulic accumulators [12]. Scientists also modeled the hydrostatic transmission in order to improve the transmission control system. Feedback and how much the system is affected by a change in one parameter was investigated [10].

In [12], the monitoring of the state of the hydraulic transmission with an adjustable axial-piston pump and an unregulated hydraulic motor was investigated. Mathematical modeling of the transmission was carried out in order to identify leaks in the hydraulic system as the main cause of malfunctions. As a result, the influence of the gaps between the pistons and the cylinder block of the axial-piston pump was investigated, and the dependence of the output power of the pump on the gap, which causes liquid leaks, was revealed.

Having analyzed the research of hydraulic drives of agricultural machines and especially hydrostatic transmissions carried out by domestic and foreign scientists, we can conclude that this type of drives is quite relevant for use. Scientists conducted a detailed study of hydraulic drives as part of various machines and mechanisms in different operating modes and with different system parameters. Although a large number of studies of hydrostatic transmissions have already been carried out, the nature of the operation of hydrostatic transmissions as drives of working bodies of agricultural machines has not yet been fully investigated. Therefore, it is relevant to carry out further research on hydrostatic transmissions of agricultural machines in order to improve their technical characteristics.

3. Aim of the researches

The purpose of the work is to study the reduced load on the feed control unit of the axial-piston pump of the hydrostatic transmission of the HGST-90 type.

4. Results of the researches

Structurally, the PVC 1 pump, which is part of the HGST-90, is an axial rotary-piston pump with an inclined disk (cradle), so the change in working volume is ensured by changing the angle of inclination of the cradle.

The schematic diagram of the PVC type pump 1 is shown in figure 3. Nine pistons 2 are installed to the cylinder block 1, which contact the surface of the cradle 3 with the left end through the heel. The rotation of the cradle in order to regulate the pump supply is performed by a hydraulic cylinder 4, the rod of which presses on the right end of the cradle. From the opposite side, spring 5 acts on the cradle, designed to rotate the cradle to the position of maximum supply of working fluid.

This pump supply regulator uses as a control signal the value of the pressure of the working fluid at the output of the adjustable throttle 7, which corresponds to the load on the hydraulic motor of a specific load sensing hydraulic drive connected to the common main pumping line of the pump 1. Accordingly, a signal is formed at the output of the load sensing distributor 6, which changes the pressure in the piston cavity of the hydraulic cylinder 4. A force is created on the rod of the hydraulic cylinder aimed at turning the cradle 3 in the direction of decreasing the angle of inclination (the minimum value of the angle of inclination of the cradle of the PVC 1.63 pump is (p_{min} = 8°). When the angle of inclination of the cradle is reduced, the working volume decreases of the pump and the supply of working fluid to the injection line. At the same time, in this design, the pressure of the pistons on the surface of the cradle creates a torque on the cradle, aimed at turning the cradle in the direction of increasing the angle of inclination of the cradle. The maximum inclination of the cradle is p_{max} = 18°.

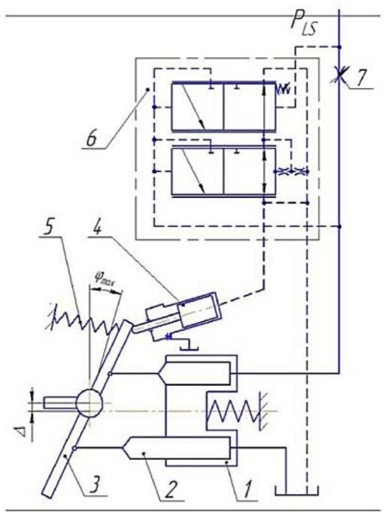


Fig. 3. Schematic diagram of the PVC 1.63 pump supply regulator

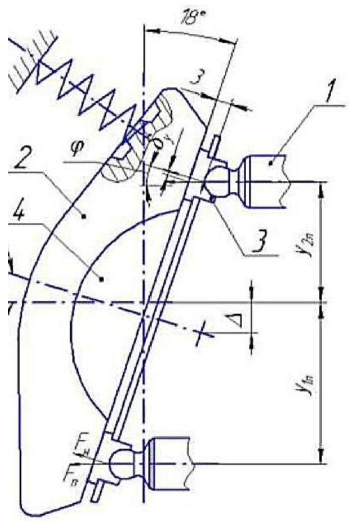


Fig. 4. Location scheme of the PVC 1.63 pump cradle relative to the axis of the cylinder block: 1 – piston, 2 – inclined disc (cradle), 3 – heel, 4 – trunnion

One of the design features of the PVC 1.63 pump cradle is the displacement of the axis of rotation of the cradle (pin axis) relative to the axis of rotation of the cylinder block by 4 mm (fig. 4).

This measure is used by the world's leading manufacturers of adjustable axial rotary-piston hydraulic machines. As a result of this, the pistons of the cylinder block create a torque on the cradle, which is aimed at increasing the angle of inclination of the cradle and supplying fluid to the hydraulic system, sufficient to turn the cradle, overcoming the moment of resistance created by the pistons of the cylinder block.

The value of the indicated torque of the cradle affects the operation of the feed control system of the axial rotary-piston pump, and also determines the operating conditions of the hydrostatic bearing, which is under the action of pressing forces from the pistons of the cylinder block. In this regard, there is a need to determine the specified turning moment of the cradle in order to further take into account when analyzing the impact of the force action of the pistons on the cradle and on the quality of the PVC 1.63 pump supply regulator, as well as determining the possibility of improving the quality of the pump supply control process.

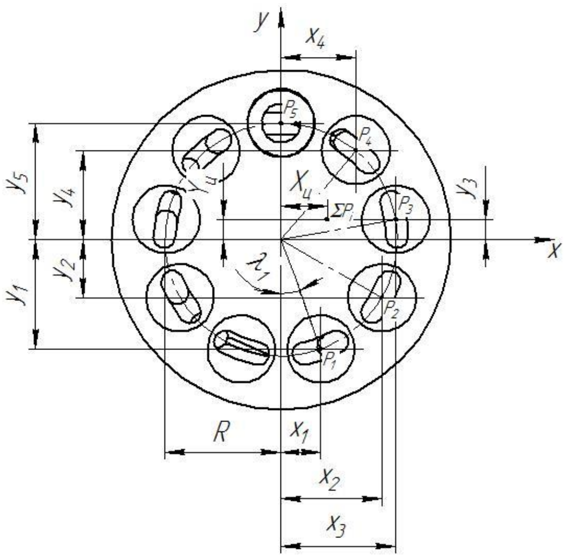


Fig. 5. Calculation scheme for determining the coordinates of the center of application of the reduced pressure force from the plunger pairs in the injection zone

The design of the PVC 1.63 pump incorporates the best design solutions used by global manufacturers of similar hydraulic machines. At the same time, a number of problems arose in the process of introducing this design into production. Including problems related to pump supply regulation. In the process of adjusting the supply of the working fluid, the angle of inclination of the inclined disk (cradle) changes. At the same time, the quality of adjustment significantly depends on the operation of the bearing assembly. In this case, hydrostatic bearings are used, on which the pivots of the cradle rest. The choice of rational parameters of the bearing unit requires a detailed analysis of its operating conditions and an assessment of the influence of design parameters.

The node of the axial-piston hydraulic pump, which is responsible for the amount and accuracy of the supply of the working fluid to the hydraulic system, is a cradle, depending on the angle of inclination of which the amount of supply of the working fluid changes. A significant number of power factors act on this node, the exact consideration of which will ensure the necessary quality and reliability of its design and the pump as a whole. One of the forces acting on this node is the pressure force from the plunger pairs in the injection zone. The number of plungers is constantly changing, so the point of application of the reduced force also constantly moves along a certain trajectory.

In fig. 5 shows the calculation scheme for determining the coordinates of the center of application of the reduced force of pressure of the plunger pairs located in the pumping zone of the pump.

To determine the coordinates of the point of force application, we write the equation of moments relative to the center of action of the reduced force.

Equation of moments relative to the center of the drive in projections on the *X* axis:

$$\begin{gathered} P_1 \cdot \left(X_Ц - X_1\right) - P_2 \cdot \left(X_2 - X_Ц\right) - P_3 \cdot \left(X_3 - X_Ц\right) - \\ -P_4 \cdot \left(X_4 - X_Ц\right) + P_5 \cdot \left(X_Ц - X_5\right) = 0 \end{gathered} \tag{1}$$

Equation of moments relative to the center of the drive in projections on the Y axis:

$$\begin{gathered} P_1 \cdot \left(Y_Ц + Y_1\right) + P_2 \cdot \left(Y_2 + Y_Ц\right) - P_3 \cdot \left(Y_3 - Y_Ц\right) - \\ -P_4 \cdot \left(Y_4 - Y_Ц\right) - P_5 \cdot \left(Y_5 - Y_Ц\right) = 0 \end{gathered} \tag{2}$$

Solving the system of equations in polar coordinates R, λ_1, relatively $X_Ц$, $Y_Ц$ we get

$$X_Ц = \frac{1}{5} \cdot R \cdot \sum_{i=1}^{5} \sin(\lambda_i) \tag{3}$$

$$Y_Ц = -\frac{R}{5} \cdot \left(\sum_{i=1}^{5} \cos(\lambda_i) \right) \tag{4}$$

Given that the number of plungers in the block, depending on the design and volume of the pump, can vary from 7 to 13, the number of plungers in the high pressure zone will change accordingly depending on the angle of rotation of the plunger block and can be calculated according to the following relationship:

$$k = \left\lfloor \frac{\pi - \varphi_n}{\chi} \right\rfloor, \tag{5}$$

where φ_n – the current angle of entry of the plunger into the high pressure zone; χ – the angle between a pair of adjacent plungers, which can be calculated by the following expression:

$$\chi = \frac{2 \cdot \pi}{n} \tag{6}$$

The angle of entry of the plunger into the high pressure zone varies according to the following dependence:

$$\varphi_n = \varphi_1 - \chi \cdot \left\lfloor \frac{\varphi_1}{\chi} \right\rfloor, \tag{7}$$

where φ_1 – angular coordinate of the first plunger.

$$\varphi_1 = \varphi - 2 \cdot \pi \cdot \left\lfloor \frac{\varphi}{2 \cdot \pi} \right\rfloor, \tag{8}$$

where φ – angular coordinate of the block of plungers.

Taking into account dependencies (5) – (8), the dependence for determining the angular coordinate of the *i – th* plunger has the following form,

$$\lambda_i = \lambda_1 + \chi \cdot i, \ i \in 0..k. \tag{9}$$

So, taking into account dependencies (9) – (13), dependencies (8) – (9) for the case *n-oi* number of plungers have the following form:

$$X_Ц = \frac{1}{k+1} \cdot R \cdot \sum_{i=0}^{k} \sin(\lambda_1 + \chi \cdot i) \tag{10}$$

$$Y_Ц = -\frac{R}{k+1} \cdot \left(\sum_{i=0}^{k} \cos(\lambda_1 + \chi \cdot i) \right) \tag{11}$$

Using MathCad, we get the trajectory of the center of combined force along the plane of the inclined cradle of the axial-piston pump:

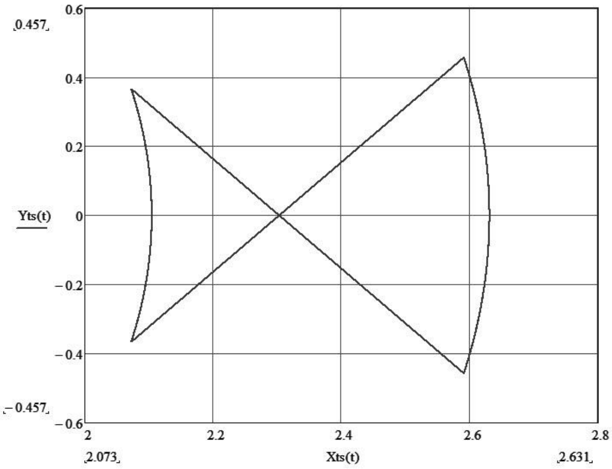


Fig. 6. Trajectory of movement of the center of application of the reduced force from the plunger pairs on the surface of the cradle of the axial-piston pump

The trajectory has the form of a loop, a cyclic nature with a periodicity of repetition due to the rotation of the cylinder block by an angle, in this case a multiple of 400, which corresponds to the angular distance between adjacent cylinders. At the same time, the coordinate $X_{Ц}$ varies from 20,73 mm to 26,31 mm, and the coordinate $Y_{Ц}$ varies within the limits of +4,57 mm to - 4,57 mm. The moment that turns the cradle relative to the trunnion axis in the direction of increasing the working volume is created due to the displacement of the center of the combined force of the cylinders along the axis Y. The moment of the combined force of the cylinders relative to the axis X will be determined by dependence

$$M_x = -\frac{R}{k+1} \cdot \left(\sum_{i=0}^{k} P_i \cos\left(\lambda_i + \chi \cdot i\right) \right), \tag{12}$$

The specified moment of the combined force also acts on the working end of the cradle and causes its rotation relative to the pivot axis. But dependence (12) is valid in the case when the axis of rotation of the cylinder block is in the same plane as the axis of the cradle trunnion. As mentioned earlier, the design of this pump provides for the offset of the trunnion axis by Δ so that the axis of the cylinder block is higher than the trunnion axis. The presence of this displacement leads to the fact that the center of the combined force of the cylinders also shifts in this direction. The displacement of the center of the combined force leads to the occurrence of a moment aimed at increasing the angle of inclination of the cradle.

Thus, the dependence (12) must be refined taking into account the presence of a displacement of the axis of the trunnion, as well as the presence of an inclination of the cradle by an angle φ, which leads to the need to take into account when determining the turning moment of the cradle the normal component of the forces acting on the end of the cradle from the side of the cylinder block. At the same time, it is also necessary to take into account that when turning the cradle at an angle φ, the normal component is directed in such a way that at the same time there is an additional displacement of the forces in the direction of a decrease in the actual displacement of the pivot axis Δ by size δy:

$$\Delta y = l \cdot tg\left(\varphi\right), \tag{13}$$

where l – the distance from the contact point of the piston head to the surface of the working end of the cradle (fig. 3). For the pump PVC 1.63 l =3 *mm.*

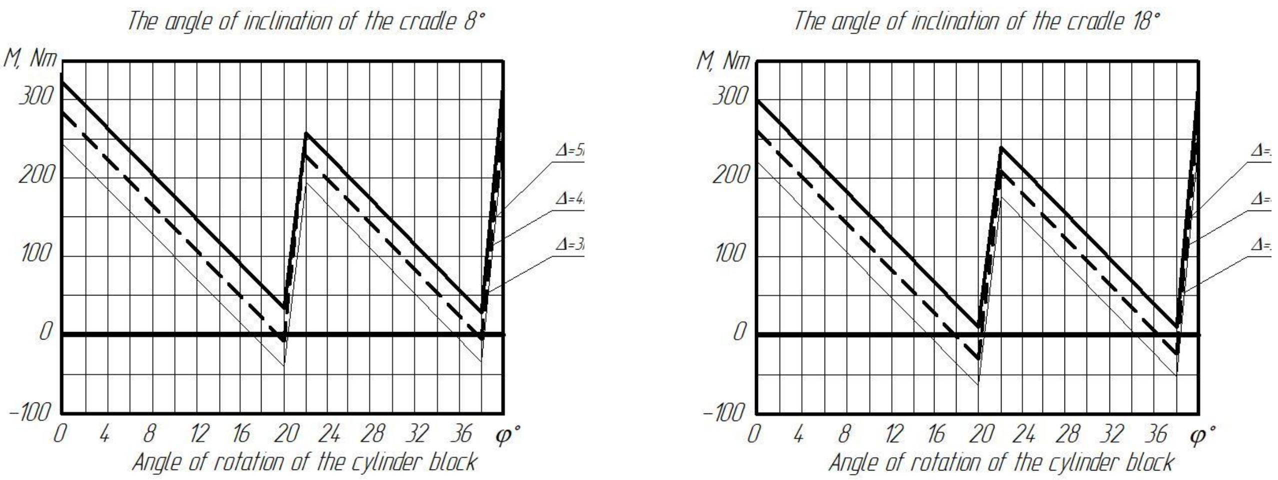
Then the dependence (12) of determining the moment of the combined force, taking into account the above, takes the following form

$$M_x = \left[-\frac{R}{k+1} \cdot \left(\sum_{i=0}^{k} \cos\left(\lambda_i + \chi \cdot i\right) - \Delta + \delta y \right) \right] \sum_{i=0}^{k} P_i \tag{14}$$

Thus, the obtained dependence (14) allows us to estimate the effect on the magnitude of the moment of the combined force of the cylinders, which ensures the rotation of the cradle in the direction of increasing the angle of inclination of the cradle, the number of cylinders, the magnitude of the angle of inclination of the cradle, the pressure in the pump discharge line, the amount of displacement of the pivot axis of the cradle relative to the axis rotation of the cylinder block.

Figure 3 shows the calculated graphs of the dependence of the moment of the combined force of the cylinders, which acts on the cradle, on the angular position of the cylinder block, which changes during its rotation.

Graphs of dependences of the moment of the combined force acting on the cradle are constructed for three different values of displacement of the axis of rotation of the cradle relative to the axis of rotation of the cylinder block: l = 3, 4, 5 mm. At the same time, the pressure in the pumping line of the pump is 25 MPa, the rotation frequency of the cylinder block corresponds to the nominal one and is n = 1500 rpm. The tilt angle j of the cradle in this case is taken as j = 80 (fig. 7, a), which corresponds to the minimum working volume of the pump, and φ = 180, at which the working volume of the pump is maximum (fig. 7, b).



a ***b***

Fig. 7. Dependence of the moment of the combined forces of the cylinder pistons acting on the cradle on the angular position of the cylinder block

According to the graphs presented in fig. 4, the moment dependences of the forces acting on the cradle have a pulsating character with a significant range of oscillations. The oscillation period corresponds to the term for which the block of cylinders turns to an angle corresponding to the angular distance between adjacent cylinders. At the same time, the graph has two peak values - when the cylinder enters the high pressure zone (fig. 4) in the lower part and when the cylinder leaves the high pressure zone at the upper point. This leads to the occurrence of a high-frequency force on the cradle - the frequency of oscillations at a rotation frequency of 1500 rpm is 450 Hz. It is obvious that when the pump cradle is made sufficiently massive, it can perform the functions of a high-frequency filter and thus avoid oscillations caused by the pulsation of the combined force of the cylinders.

The initial angular position of the cylinder block is when the cylinder in the lower part of the cylinder block (fig. 5) passes from the suction zone to the discharge zone. It is in this position of the cylinder block that the moment of the combined force of the cylinders reaches its maximum value (fig. 7).

The maximum values of the moment of the combined force acting on the cradle (fig. 7, a), depending on the amount of displacement of the axis of rotation of the cradle relative to the axis of the cylinder block, have different values. Thus, with a shift of l=3 mm, the maximum moment of the combined force acting on the cradle is 244 Nm, with a shift of the rotation axis of the cradle by 1= 4 mm, the maximum moment of the combined force increases to 283 Nm. When the displacement is further increased to l = 5 mm, the moment of the combined force acting on the cradle increases to 323 N.m.

When rotating the cylinder block in the following angular positions, the moment of the combined force acting on the cradle in all three cases decreases almost linearly. The moment of forces decreases before turning the cylinder block to an angle of 20°, during which the cylinder that creates force P moves in the upper half of the cylinder block (fig. 5) from the high-pressure zone to the low-pressure (suction) zone.

At the same time, the moment of the combined force acting on the cradle decreases to 36 N.m when the cradle axis shifts by 5 mm, to 3 N. m when the cradle axis shifts by 4 mm, to 42 N. m when the cradle axis shifts by 3 mm Nm. The negative value of the moment of forces in the last two cases means that there is a moment of combined force aimed at turning the cradle in the direction of decreasing the angle φ and, accordingly, reducing the working volume of the pump, which can interfere with the normal course of the process of regulating the supply of the PVC pump 1.63.

With further rotation of the cylinder block (rotation angle equal to 240), the moment of the combined force acting on the cradle increases again to the second maximum value, which is slightly smaller than the previous one. So, when shifting the axis of rotation of the cradle by l = 5 mm, the value of the moment of the combined force increases to 25 8 Nm, when shifting the axis of the cradle by l = 4 mm, the moment of the combined force is equal to 227 Nm, and when shifting the axis of rotation of the cradle by the amount of 1 = 3 mm moment of the combined force equals 196 Nm. During further rotation of the cylinder block, the moment of the combined force decreases to a minimum value when the cylinder block is rotated to an angle of 400. At the same time, when the axis of rotation of the cradle is shifted by the amount of l = 5 mm, the moment of the combined force remains positive, and when the axis of rotation of the cradle is shifted by the amount of 1 = 3 mm and 4 mm, the moment of the combined force acting on the cradle becomes negative.

The range of oscillations of the moment of the combined force can reach 290 Nm regardless of the amount of shift of the axis of rotation of the cradle at the angular position of the cylinder block within 00 - 200. At the angular position of the cylinder block within the range of 200 - 400, the range of oscillations decreases to the value of 218 Nm regardless of the amount of shift of the axis of rotation cradles.

When the angle of inclination of the cradle increases to the value φ = 180, the nature of the dependence of the moment of the combined force on the angular position of the cylinder block (fig. 7, b) remains unchanged.

The analysis of the dependencies shown in figure 7, b, shows that the moment of the combined force acting on the cradle decreases slightly with an increase in the angle of inclination of the cradle. Thus, the value of the moment of the combined force acting on the cradle at the 1st (the cylinder passes the lower point) angular position of the cylinder block (the displacement of the axis of rotation of the cradle in this case is 1 = 5 mm) decreased to the value of M = 301 Nm. At an angle of rotation of the cradle by 80, this moment was 323 N.m. Similarly, the maximum values of the moment of the combined force change at other values of the shift of the axis of rotation of the cradle. At the same time, it should be noted that the range of fluctuations of the moment of the combined forces remains unchanged, therefore, at the rotation angle of the cylinder block 200, the negative values of the moment of the combined forces when shifting the rotation axis of the cradle increases by 3 mm and 4 mm. In this case, when the cradle axis shifts l = 3 mm, the moment of combined forces MX = -70 Nm, and when the cradle axis shifts l = 4 mm, the moment MX = -30 Nm, which worsens the conditions of normal operation of the PVC 1.63 pump supply regulator.

5. Conclusions

Therefore, it can be concluded that the center of application of the reduced force moves along a rather complex trajectory, which leads to the occurrence of a torque that varies in magnitude and coordinate of application, depending on the angle of rotation of the plunger block.

In the process of researching the mechanism of the formation of the moment of the combined force acting on the cradle of the PVC 1.63 pump from the side of the pistons of the cylinder block, it was found that the point of application of the combined force to the end of the cradle moves along a loop-like trajectory, symmetrical to the axis, parallel to the axis of rotation of the cradle. At the same time, the magnitude of the moment of the combined force acting on the cradle has a cyclic nature, the periodicity of which is determined by the angle of rotation of the cylinder block by a value equal to the angular distance between adjacent cylinders. The frequency of fluctuations of the magnitude of the moment of the combined force when rotating the cylinder block of the PVC 1.63 pump with a frequency of 1500 rpm is 450Hz, which requires the development of measures to eliminate their negative impact on the operation of the pump supply regulator.

The moment of combined forces acting on the cradle from the side of the cylinder block depends on the pressure in the pump discharge line and the angle of rotation of the cylinder block. At the same time, the range of fluctuations of the moment of the combined force reaches significant values. Thus, at a nominal pressure of 25 MPa, the range of oscillations of the combined force moment acting on the PVC 1.63 pump cradle reaches 290 Nm. If the axes of the cylinder block and the trunnion of the cradle are in the same plane, the moment of combined forces acting on the cradle will change symmetrically relative to the axis of rotation of the cradle and periodically cause it to deviate in opposite directions. This nature of the load of the cradle interferes with the process of adjusting the angle of its inclination and, accordingly, the working volume of the pump.

In order to reduce the negative impact of the moment of the combined force on the quality of the regulation of the supply of the working fluid of the PVC 1.63 pump, it is recommended to introduce a shift of the axis of rotation of the cradle relative to the axis of the cylinder block by 5 mm.

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ДОСЛІДЖЕННЯ ТРАЄКТОРІЇ РУХУ ПРИВЕДЕНОЇ СИЛИ ТИСКУ ПЛУНЖЕРІВ НА ЛЮЛЬКУ АКСИАЛЬНО-ПОРШНЬОВОГО НАСОСА

Гідравлічний привод на сьогодні все більше застосовується у сільськогосподарській техніці у якості приводів руху, приводів робочих органів, приводів систем керування. Джерелом живлення даного виду приводів є насосні станції на основі насосів поршневого типу. Даний тип насосних агрегатів має ряд переваг та недоліків у порівнянні із шестеренними насосам. До переваг слід віднести потужність, надійність, більш високий коефіцієнт корисної дії, можливість плавного регулювання витрати робочої рідини, до недоліків слід віднести підвищені вимоги до ступеня очищення робочих рідин.

В статті проведено огляд відомих гідравлічних схем сільськогосподарських машин, виконано їх аналіз визначено їх переваги та недоліки. Одним із елементів даних гідравлічних систем є система *керування подачею насоса у залежності від навантаження на робочі органи елементів гідравлічної системи. Правильний вибір параметрів елементів даної системи у значній мірі впливає на якість її роботи в цілому. З метою проведення дослідження впливу параметрів системи описано будову та принцип роботи системи регулювання подачі робочої рідини аксіально-поршневого насоса, проведені розрахунки координати руху центра приведеної сили від дії плунжерних пар на люльку системи керування подачею робочої рідини. Визначено траєкторію її руху по площині люльки регулювання подачі. Запропоновані залежності дозволяють визначити величину моменту що діє на люльку у залежності від величини ексцентриситету. Виявлено, що точка прикладання зведеної сили до торця люльки переміщується по петлеподібній траєкторії, симетричній до осі, паралельній осі повороту люльки. При цьому величина моменту зведеної сили, діючої на люльку, має циклічний характер, періодичність якої визначається кутом повороту блока циліндрів на величину, що дорівнює кутовій відстані між сусідніми циліндрами. Частота коливань величини моменту зведеної сили при обертанні блока циліндрів насоса PVC 1.63 з частотою 1500 об/хв становить 450 Гц, що вимагає розроблення заходів по усуненню їх негативного впливу на роботу регулятора подачі насоса.*

Момент зведених сил, який діє на люльку з боку блока циліндрів, залежить від тиску в лінії нагнітання насоса та кута повороту блока циліндрів. При номінальному тиску 25 МПа розмах коливань моменту зведеної сили, діючого на люльку насоса PVC 1.63, досягає 290 Нм. Якщо осі блока циліндрів та цапф люльки знаходяться в одній площині, момент зведених сил, що діє на люльку, змінюватиметься симетрично відносно осі повороту люльки і періодично викликатиме її відхилення в протилежних напрямах. Такий характер навантаження люльки заважає процесу регулювання кута її нахилу і, відповідно, робочого об'єму насоса.

З метою зменшення негативного впливу моменту зведеної сили на якість регулювання подачі робочої рідини насоса PVC 1.63 рекомендовано запровадити зсув осі повороту люльки відносно осі блока циліндрів на 5 мм.

Ключові слова: *гідравлічний привод, робоча рідина, аксіально-поршневий насос, дослідження, самохідна сільськогосподарська машина, витрата рідини, гідромотор, технологічне навантаження.*

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