UDC 62-82:631.659
DOI: 10.37128/2520-6168-2025-1-10

DEVELOPMENT OF A DYNAMIC MODEL OF A HYDRAULIC DRIVE OF A POSITIONING MECHANISM OF A WIDE-SCALE AGRICULTURAL UNIT

Serhiy SHARGORODSKIY, Candidate of Technical Sciences, Associate Professor
Dmutro KONDRATUK, Recipient of the Third Educational and Scientific Level
Vinnytsia National Agrarian University
Vita SHARHORODSKA, Teacher English and German languages
Separate structural subdivision of technological and industrial vocational college
Vinnytsia National Agrarian University

ШАРГОРОДСТКИЙ Сергій Анатолійович, к.т.н., доцент
КОНДРАТЮК Дмитро Анатолійович, здобувач третього освітньо-наукового рівня
Вінницький національний аграрний університет
ШАРГОРОДСЬКА Віта Миколаївна, викладач
Відокремлений структурний підрозділ «Технологічно-промисловий фахой коледж» Вінницького національного аграрного університету

The article is devoted to the most pressing problem of developing a dynamic mathematical model of a hydraulic drive for folding and unfolding a wide-cut agricultural unit and research. To achieve this goal, a system analysis of the processes involved in the hydraulic drive, which indicates the transfer of the unit from the transport position in operation, was carried out. Monitored by hydraulic systems. An examination of known mathematical models and their methods was carried out. An analysis was carried out and trends in the development of agricultural units and equipment for soil treatment. Their advantages and shortcomings are analyzed. It is noted that one of the direct developments of current agricultural machinery is an increase in working width. This trend is justified from the point of view of the economy of burning, a reduction in the number of passes of the unit and a change in the strengthening of the soil, which positively flows into the soil and promotes its fuelability. To a few of these machines, the complexity of the maneuvers should be brought into account. To make it easier to maneuver around the turning method, it is better to equip these units with a system for transferring from the transport position to the operator. This system can respond to current situations, and itself: high speed, but the flexibility of the folding section, compactness. To complete this task, the hydraulic drive must be turned on completely. This drive has numerous advantages, such as compactness, high strength, ability to operate at high speeds and dampen vibrations. A hydraulic drive circuit has been designed to transfer the unit from the transport position to the operating position due to the incoming hydraulic motors. A mathematical model of the drive has been developed, which includes the balance of the drive and the forces that act on the working bodies of the hydraulic motors. The extracted results allow you to analyze in detail the processes in the drive and determine the optimal parameters to ensure efficient operation of the system.

Key words: *wide-cut unit of agricultural significance, hydraulic drive, hydraulic cylinder, three-position distributor, mathematical model of the drive, structure diagram, parameter, follow-up.*

Eq. 70. Fig. 3. Ref. 19.

1. Problem formulation

As we know, the rural dominion is a priority sector of the economy of our state. Therefore, the production of these galusa is intensively developing, and will require modern economical and handy units and machines. Having analyzed the current trends in the development of machine design, it can be seen that hydraulic drives are increasingly used. This is due to the significant advantages of this type of drives, whether mechanical or electrical. And the compactness, tightness, freedom of arrangement. With the remaining time of stagnation of wide-reach soil-cultivating units, a significant increase in width has arisen due to a decrease in the number of passes during field cultivation, as a result of a change in the hardening of the soil, and savings in soil. With a wide-grip unit, the worker is positioned to fold the turns and collapse along a folding path. Therefore, in order to achieve this task, wide-cut units have hydraulic systems that allow the unit to be quickly

DEVELOPMENT OF A DYNAMIC MODEL OF A HYDRAULIC DRIVE OF A POSITIONING MECHANISM OF A WIDE-SCALE AGRICULTURAL UNIT © 2025 by Serhiy SHARGORODSKIY, Dmutro KONDRATUK, Vita SHARHORODSKA is licensed under CC BY 4.0

moved from the transport position to the working position and to the back. Hydraulic systems of this type offer advantages in terms of fluidity, smoothness and accuracy of movement of working parts. To secure the data, it was necessary to carry out a complex investigation of this system in order to identify specific design characteristics. This problem can be solved with the help of mathematical modeling. Therefore, the investigation of these processes using the method of avoiding them is an urgent task.

2. Analysis of recent research and publications

The issues of research, design and implementation of the hydraulic drive were dealt with by many scientists both in our country and abroad.

Such scientists as T.M. Bashta, I.A. Nemyrovsky were involved in the development and research of hydraulic drives for agricultural machines. and many others.

The publication [1] is dedicated to the study of the operation of the hydraulic drive of the cutting device. This work is devoted to the research questions of the investigation of the processes occurring in the hydraulic drive of working bodies in the event of shock loads and taking into account the oscillatory processes associated with both the occurrence of significant fluctuations in pressure and the flow of the working fluid, as well as the consequence of the technical imperfection of the circuit design solution of the hydraulic drive. It is noted that oscillating processes have a negative effect on the resource of the hydraulic drive, reduce the utilization ratio of the set power of the drive and productivity, reduce reliability, and increase the energy consumption of the drive and the machine as a whole.

A fundamentally new structural and technological scheme of a cutting mechanism for unloading silage from trench storages with hydraulic drive of working bodies is proposed. The structure of the hydraulic drive of the cutting mechanism has been developed, adapted to the change in the technological load on the working bodies, which will allow to coordinate the operating modes of the cutting mechanism and correspond to modern, innovative approaches to the creation of resource-saving and energy-efficient equipment.

The proposed approaches to the modeling of these systems make it possible to apply them to a wide range of machines.

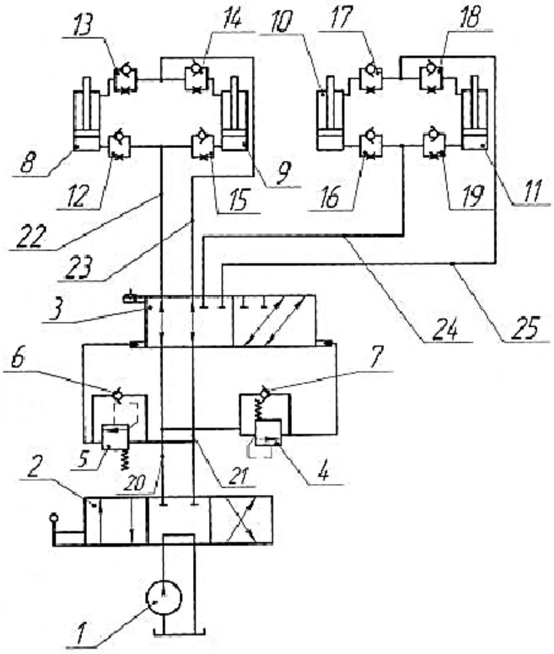

Fig. 1. Hydraulic drive for transferring a wide-grip agricultural unit from the transport position to the working position.
Source: grouped by author based on [1]

There is a well-known scheme of the hydraulic drive for transferring a wide-grip transport machine from the transport position to the working position and vice versa [2], which is presented in fig. 1.

Presented in fig. 1 scheme works in the following sequence. In the event that the distributor is turned on in the neutral position, the working fluid is not supplied to the corresponding hydraulic lines, but is drained into the tank through the safety valve of the pumping station.

In the case when the switching of the right cavity occurs, the working fluid is supplied to the input of the logical valve "or" through the hydraulic line 20. The spool of the valve "Or" is currently fixed in the extreme left position.

Then, from the outlet of the "Or" valve, through the hydraulic line 22 and the throttle with the block of check valves 12 and 15, the working fluid enters the piston cavities of the hydraulic cylinders 8 and 9.

From the drain cavities of the hydraulic cylinders 8, 9, throttles and non-return valves 13 and 14, the working fluid along line 23 is diverted to the logic valve "or". After that, through the hydraulic spool distributor 2, the working fluid enters the drain hydraulic line. In this case, pistons with rods move up. This movement occurs until the sections of the cultivator stop, after which this movement stops. After that, the pressure in the cavities of the hydraulic cylinders 8 and 9 begins to increase, and the pressure in the hydraulic line 20, to which the pressure valve 4 is connected, also increases. line, the pressure increases in the hydraulic line 20, to which the inlet of the pressure valve 4 is connected. This valve opens to discharge the working hydraulic fluid when the pressure increases to the setting pressure value. If this valve is opened, the working fluid gets under the end of the spool of logic valve 3 and throws it to the right position with subsequent fixation. As a result, the inlet of logic valve 3, throttles with check valves 16 and 19 and the lower piston cavities of hydraulic cylinders 10 and 11 are under pressure, from which the working fluid enters the throttles with check valves 17 and 18, through hydraulic line 25, logic valve 3, hydraulic distributor 2 and from it to the drain line. As a result, the pistons of the hydraulic cylinder 10 and the hydraulic cylinder 11 move up and accordingly open the extreme sections until the stop.

The disadvantages of this scheme include its complexity and the rather significant cost of completing it.

In the work [4], the authors covered the issue of the occurrence and propagation of vibrations in the structures of agricultural tractors.

Thus, solving the problem of justifying the parameters of the hydraulic drive for transferring the wide-grip agricultural machine from the transport position to the working position and vice versa is an urgent scientific and practical task. In the case of a positive solution to this problem, domestic enterprises will be able to modernize existing wide-ranging agricultural units and create new ones at a higher quality level, which in turn will contribute to the process of entering foreign markets and improve their position on the domestic market.

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

3. The purpose of the article

The aim of the research is to develop a mathematical model of the hydraulic drive of the positioning mechanism of a wide-grip agricultural unit with a tracking system.

4. Results and discussion

As it follows from the analysis of literary sources, the most effective way of regulating the speed of movement of working bodies for use in hydraulic systems for positioning parts of agricultural machinery units is a hydraulic lock.

The hydraulic scheme of the proposed drive is shown in Fig. 2, where 1, 3, 8, 17, 21, 23 – supply and discharge mains of the working fluid; 2 – spreader; 4, 7, 18, 20 – inlet and outlet channels of the working fluid in the hydraulic lock housing; 5 – the closing element of the non-return valve; 6 – check valve springs; 9, 11, 16, 19 – working cavities of the hydraulic lock; 10 – check valve seats; 12 – plunger; 13, 14 – hydraulic cylinder cavity; 15 – hydraulic cylinder; 22 – hydraulic lock assembly, 24 – safety valve.

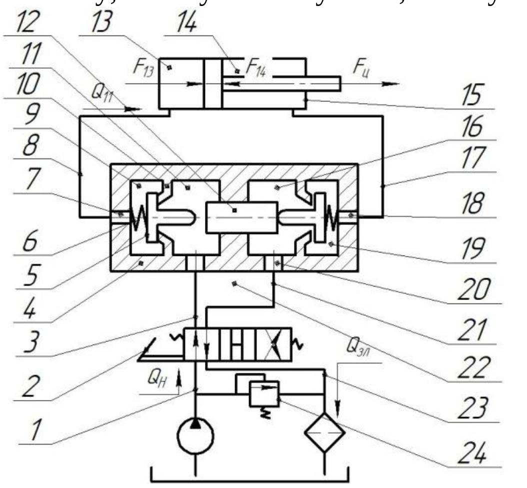

Fig. 2. Scheme of the hydraulic drive for the deployment of the section of the wide-grip soil tillage tool

To build a mathematical model, we will use the following assumptions:

• pump supply is unchanged during drive operation;

• the hydraulic system works at a fixed temperature regime, so we consider the viscosity of the working fluid to be a constant value;

• pipelines and hydraulic mains are considered absolutely rigid;

• to take into account the compressibility of the working fluid due to the gases dissolved in it, the reduced modulus of elasticity of the system is introduced;

• we believe that there is no delay when the hydraulic equipment is activated.

We build our mathematical model based on the equations of the continuity of the working fluid and Newton's second law.

The cost balance equation between cavities 13 and 14 will look like this:

$$Q_{13} = kQ_{14}, \quad (1)$$

where k – the multiplication factor.

The force balance equation of plunger 12 will have the following form:

$$F_{p11} - F_{p16} - F_{кл} - F_{pl}^{a} - F_{pl}^{fr} = 0 \quad (2)$$

where F_{p11} – pressure force of the working fluid from the side of the cavity 11 on the plunger 12, N; F_{p16} – pressure force of the working fluid from the side of the cavity 16 on the plunger 12, N; $F_{кл}$ – the force of action of the closing element of the valve 5 on the plunger 12, N; F_{pl}^{a} – force of inertia of the plunger, N; F_{pl}^{fr} – the force of liquid friction that occurs during the movement of the plunger in the housing groove, N.

The equation of motion of the piston of the hydraulic cylinder 15, taking into account the inertial load, will have the following form:

$$F_{p13} - F_{p14} + F_{hc} - F_{hc}^{a} - F_{hc}^{fr} = 0\,, \tag{3}$$

where F_{p13} – force of liquid pressure on the piston of the hydraulic cylinder from the side of the piston cavity, N; F_{p14} – force of liquid pressure on the piston of the hydraulic cylinder from the side of the rod cavity, N; F_{hc} – the force acting on the hydraulic cylinder rod from the side of the external load, N; F_{hc}^{a} – reduced inertial force of moving parts of the system moving together with the piston of the hydraulic cylinder, N; F_{hc}^{fr} – the total force of liquid friction that occurs when the piston moves in the hydraulic cylinder housing, N.

For equation (1), we determine the flow rate of the working fluid through cavity 13, taking into account the compressibility of the fluid, respectively:

$$Q_{13} = Q_{11} - Q_{8}\,, \tag{4}$$

where Q_{11} – consumption through cavity 11, m^3/s; Q_8 – flow rate of the working fluid, taking into account its compressibility, m^3/s.

We define the multiplication factor *k* as the ratio of the areas of the through sections:

$$k = \frac{f_{13}}{f_{14}}\,, \tag{5}$$

where f_{13}, f_{14} – the area of the piston of the hydraulic cylinder, respectively, from the side of the piston and rod cavities, m^2.

The terms of equation (2) can be described as follows:

$$F_{p11} = p_{11} f_{pl}\,, \tag{6}$$

where p_{11} - the pressure in the cavity 11, the sought value, is one of the main investigated parameters, N/m^2; f_{pl} - area of the end surface of the plunger, m^2.

$$F_{p16} = p_{16} f_{pl}\,, \tag{7}$$

where p_{16} – pressure in the cavity 16.

The equation of the balance of forces acting on the plunger of the hydraulic lock has the following form according to Fig. 3:

$$F_{v} = F_{s} + F_{p19}^{v} - F_{pз}^{v} - F_{p16}^{v} + F_{dyn}\,, \tag{8}$$

where F_s – the force of action on the valve spring 6, *N*; F_{p19}^{v} – the force perceived by the valve from the action on its surface of pressure p_{10}, N; $F_{pз}^{v}$ – the force perceived by the valve from the action on its surface of the pressure p_{sr} in the gap S_4, N; F_{p16}^{v} – the force perceived by the valve from the pressure acting on its surface p_{16}, N; F_{dyn} – hydrodynamic force acting on closing the valve, N.

$$F_{пл}^{a} = m_{пл}^{/} \frac{d^2 S_{пл}}{dt^2}\,, \tag{9}$$

where $m_{пл}^{/}$ - total weight of plunger and valve, *kg*; $S_{пл}$ – displacement of the plunger during time *t*, corresponding to the displacement of the valve S_4 m.

Figure 3 shows the following designations – S_1, S_4 – valve clearances; $S_2 = 2S_0 - S_3 + S_1 + S_4$ – the gap between the valve pusher and the plunger in cavity 11; $2S_0$ – the sum of the clearances S_2 and S_3 at the neutral position of the plunger; S_3 – the gap between the valve pusher and the plunger in cavity 16; d_1 – diameter of the plunger; d_2 – diameter of valve pushers; d_3 – inner valve seat diameter; d_4 – outer valve seat diameter; L – the length of the valve channel; $X_1 = X_0 + S_1$ – amount of deformation of the check valve spring; X_0 is the value of the pretension of the spring.

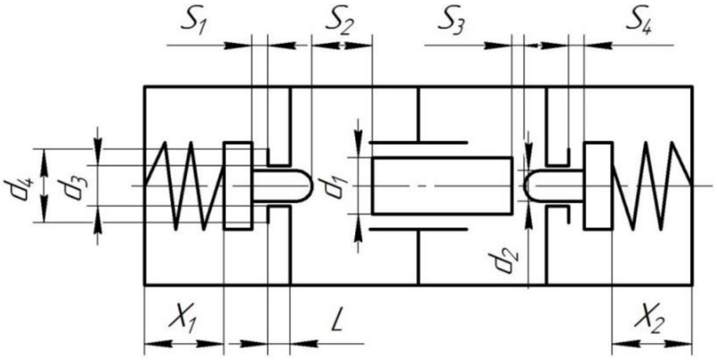

Fig. 3. Hydrokinematic scheme of the hydraulic lock

We believe that friction changes according to a linear law and is proportional to the speed of movement of the element.

$$F_{pl}^{fr} = k_{ll} \frac{dS_{pl}}{dt} \qquad (10)$$

where k_{ll} – a complex coefficient that takes into account fluid friction in the annular gap between the plunger and the hydraulic lock body.

The power factors included in equation (3) are described by the following calculation dependencies:

$$F_{p13} = p_{13} f_{13}, \qquad (11)$$

where p_{13} – pressure in the pipeline 13, N/m^2;

$$F_{p14} = p_{14} f_{14}, \qquad (12)$$

where p_{14} – pressure in the pipeline 14, N/ m^2;

$$F_{hc}^{a} = m_{hc} \frac{d^2 S_{hc}}{dt^2}, \qquad (13)$$

where m_{hc} – the weight of the moving parts of the unit section reduced to the piston, *kg*; S_{hc} – displacement of the piston of the hydraulic cylinder during time t, m.

$$F_{hc}^{fr} = k_{hc} \frac{dS_{hc}}{dt}, \qquad (14)$$

where k_{hc} – complex coefficient of friction, which takes into account the friction conditions in the annular gap between the piston and the hydraulic cylinder body.

We calculate the flow rate of the working fluid in cavity 11 according to the following relationship:

$$Q_{11} = Q_p - Q(p_1) - Q_{y1} + f_{pl} \frac{dS_{pl}}{dt}, \qquad (15)$$

where Q_P – pump performance 1, m^3/s; $Q(p_1)$ – consumption of the working fluid caused by the operation of valve 24, in the event of an increase in the pressure in the hydraulic drive to the level of the beginning of its opening, *m*^{3}*/s*; Q_{y1} – increase in flow due to the compression of the volume of liquid V_1, which is at this moment in the area from the pump to cavity 16 inclusive, m^3/c; $f_{pl} \frac{dS_{pl}}{dt}$ – increase in flow due to piston movements over time t, m^3/s.

$$Q_{y8} = \frac{V_8}{E} \frac{d(p_{13})}{dt}, \qquad (16)$$

where V_8 – the volume of liquid that is at this moment in the area from gap S_1 to cavity 13 inclusive, *m*^{3}.

The components of equations (15) and (16) are defined as:

$$f_{pl} = \frac{\pi d_1^2}{4}, \text{ m}^2 \qquad (17)$$

where d – plunger diameter (Fig. 2), m.

$$p_{16} = \Delta p_{16} = \left(\sum_{i=1}^{n} \xi_{16i} + \lambda_{16} \frac{\sum l_{16}}{d_{16}} \right) \cdot \frac{\gamma Q_{16}^2}{2gf_{16}^2}, \; N/\text{m}^2, \tag{18}$$

where Δp_{16} – pressure losses in local and linear supports along the length of the pipeline from cavity 16 to the tank, N/m^2; ξ_{16i} – loss coefficient of the *i-th* local resistance; *i*, *n* – ordinal index and number of local resistances on the pipeline section from cavity 16 to the tank; λ_{16} – pipeline resistance coefficient; l_{16} – pipeline length from cavity 16 to the tank, m; f_{16} – live cross-sectional area of the pipeline 21, m^2; Q_{16} – flow through the cavity 16, m^3/s.

The elastic force of the spring included in equation (8) is determined as follows:

$$F_c = c \cdot \left(X_0 + S_4 \right), \; N, \tag{19}$$

where c – spring stiffness 6, N/m; X_0 – precompression of the spring, m; S_4 – the amount of additional compression, which corresponds to the gap S_4, m.

$$F_{p19}^{v} = f_4 \, p_{19}, \tag{20}$$

where f_4 – the area of influence on the sealing element of the pressure in the cavity 19, m^2; p_{19} – cavity pressure 19, N/m^2.

$$F_{p3}^{кл} = f_{4/3} p_{cp}, \tag{21}$$

where $f_{4/3}$ – the area of influence on the closing element of the pressure in the gap S_4, m; p_a – average pressure in the gap S_4, N/m^2.

$$F_{p16}^{v} = f_3 \, p_{16}, \tag{22}$$

where f_3 – the area of influence on the closing element of the pressure in the cavity 16, m^2.

The hydrodynamic force can be determined based on the dependence presented in the work [5]:

$$F_{dyn} = \frac{Q\gamma}{g} \left(\upsilon_1 - \upsilon_2 \cdot \cos\left(\frac{\alpha}{2} \right) \right), \tag{23}$$

where

$$\upsilon_1 = \frac{4Q}{\pi d^2} \; \text{ i } \; \upsilon_2 = \frac{Q}{f_k} \tag{24}$$

where a – valve cone angle in a flat valve $\alpha = 180^\circ, \cos(\alpha) = 0$, therefore, for a locking element of this type, the dynamic force can be determined as follows:

$$F_{dyn} = \frac{4Q_{19}^2 \gamma}{g\pi d_3^2}, \tag{25}$$

where Q_{19} – flow through the cavity 19.

The components in equation (26) are calculated based on the conditions:

$$m_{pl}^{/} = m_{pl} + m_v, \tag{26}$$

where m_{pl} – weight of the plunger, kg; m_v - valve mass, kg;

$$S_{пл} = S_0 + S_4 \tag{27}$$

where S_0 – clearance in neutral position, we think that $S_0 = 0$.

The components of pressure equations (2.57) and (2.58) in cavities 13 and 14 can be expressed as

$$p_{13} = p_{11} - \Delta p_{11з} - \left(\sum_{i=1}^{n} \xi_{11i} + \lambda_{11} \frac{\sum l_{11}}{d_{11}} \right) \cdot \frac{\gamma Q_{13}^2}{2gf_{11}^2} \tag{28}$$

where $\Delta p_{11з}$ – pressure loss in the gap S_1, N/m^2; ξ_{11i} – coefficient of the i-th local resistance; *i*, *n* – ordinal inwherex and the number of local resistances on the pipeline section from gap S1 to cavity 13;

λ_{11} – pipeline resistance coefficient 8; 1_{11} – pipeline length 8, m; f_{11} – live cross-sectional area of the pipeline 8, m^2.

$$p_{14} = \left(\sum_{i=1}^{n} \xi_{14i} + \lambda_{14} \frac{\sum l_{14}}{d_{14}} \right) \cdot \frac{\gamma Q_{13}^2}{2gk^2 f_{17}^2} + p_{16} + \Delta p_{16з}, \tag{29}$$

where $\Delta p_{16з}$ – pressure loss in the gap S_4, N/m^2, i, n – ordinal index and the number of local resistances on the pipeline section from cavity 14 to the gap S_4; l_{14} – pipeline length from cavity 14 to cavity 19, *m*; f_{17} – live cross-sectional area of local or linear resistance, m^2.

The current stroke of the piston can be determined from the dependence:

$$S_{гц} = \int_{t_0}^{t} \frac{Q_{13}(t)}{f_{17}} dt, \tag{30}$$

where t_0 – the time of the start of the movement, s; t – current time, s.

The consumption of the working fluid in the 16th cavity is described by the following dependence:

$$Q_{16} = Q_{19} - f_{пл} \frac{dS_{пл}}{dt} + Q_{yS_4}, \text{ m}^3/\text{c}, \tag{31}$$

where Q_{yS_4} – increase in flow rate due to pressure drop across the throttle element.

The cross-sectional area 4 of the throttle window is calculated according to the formula:

$$f_4 = \frac{\pi d_4^2}{4}, \tag{32}$$

where d_4 – the outer diameter of the seat of the locking element, m;

$$f_{4/3} = \frac{\pi \cdot \left(d_4^2 - d_3^2\right)}{4}, \tag{33}$$

where d_3 - valve seat inner diameter, m;

$$f_3 = \frac{\pi d_3^2}{4}, \tag{34}$$

$$p_{19} = p_{16} + \Delta p_{16}, \tag{35}$$

where Δp_{16} – pressure loss in the gap S_4, N/m^2.

The pressure p_{sr} can be determined as an arithmetic mean, since the Reynolds number Re for the operating range of the gap and flow rate is less than the critical one for a flat valve. And in this case, according to the work of M.I. Levytskyi and E.I. Tsukhanova "Calculations of control devices for braking hydraulic drives", the change in pressure in the gap occurs according to a linear law:

$$p_{ср} = \frac{p_{19} + p_{16}}{2}, \tag{36}$$

and taking into account (35);

$$p_{ср} = p_{16} + \frac{\Delta p_{16з}}{2}. \tag{37}$$

Consumption of working fluid Q_{19} we determine taking into account (15), (16) i (18) as:

$$Q_{19} = Q_{14} - Q_{y17} = \frac{1}{k}\left(Q_P - Q(p_1) - Q_{y1} + f_{pl} \frac{dS_{pl}}{dt} - Q_{y8} \right) - Q_{y17} \tag{38}$$

where Q_{y17} – increase in flow due to compression of the volume of liquid that is at this moment between cavities 14 and 19 inclusive, m^3/s.

The amount of pressure loss Δp for clearance S_1, we calculate according to the following dependence:

$$\Delta p = \frac{\gamma \xi Q^2}{2 f_з^2 g}, \tag{39}$$

where Q – fluid flow through this throttling element, m^3/c; Δp – pressure drop across the throttle, N/m^2; ξ – coefficient of local resistance; g - acceleration of free fall; γ – specific gravity of the working fluid, N/m^3.

Coefficient of local resistance ξ for a flat valve, provided the ratios are met, can be determined from the following relationship:

$$\xi = \alpha_0 + \beta_0 = 0.55 + 4\left(\frac{b_T}{D_0} - 0.1\right) + \frac{0.155D_0^2}{h^2} \quad (40)$$

For clearance S_1

$$\xi_{11з} = 2\frac{d_4}{d_3} + \frac{0.155d_з^2}{S_4^2} - 1.852. \quad (41)$$

For clearance S_1 given that the area of the liquid jet f_3:

$$f_3 = f_{11з} = \pi d_3 S_1. \quad (42)$$

pressure losses are determined by the following expression:

$$\Delta p_{11з} = \frac{\gamma \xi_{11з} Q_{11}^2}{2g\left(\pi d_3 S_4\right)^2} = \frac{\gamma\left(2\frac{d_4}{d_3} + \frac{0{,}155d_4^2}{S_1^2} - 1{,}85\right)Q_{11}^2}{2g\left(\pi d_3 S_4\right)^2} = \frac{\gamma\left(2S_1^2\frac{d_4}{d_3} + 0{,}155d_4^2 - 1{,}85S_1^2\right)Q_{11}^2}{2g\left(\pi d_3 S_4\right)^2 S_1^2} \quad (43)$$

We determine pressure losses $\Delta p_{16з}$ in the gap S_4:

$$\Delta p = \frac{\gamma Q^2}{\mu^2 f^2 2g}, \quad (44)$$

where μ – cost factor.

Cost factor μ is a function of the ratio of clearance and diameters – $\mu_з = f\left(d_3, d_4, S_4\right)$, and for our case:

$$\mu_з = K_\psi \cdot \frac{S_4}{d_3}. \quad (45)$$

Since the cross-sectional area of the gap can be defined in our case as:

$$f_{S4} = \pi d_3 S_4, \quad (46)$$

then the pressure losses in the gap S_4 can be expressed as a dependence:

$$\Delta p_{16з} = \frac{\gamma Q_{19}^2}{K_\psi^2 \pi^2 S_4^4 2g}. \quad (47)$$

Then the cost increase Q_{yS_4}, is defined by dependence:

$$Q_{yS_4} = \frac{V_{21}}{E} \cdot \frac{d\left(\Delta p_{16з}\right)}{dt}, \quad (48)$$

where V_{21} – the volume of liquid located in the section of the pipeline from the gap S_4 to the tank, m^3.

Cost growth Q_{y17}:

$$Q_{y17} = \frac{V_{17}}{E} \cdot \frac{dp_{14}}{dt}, \quad (49)$$

where V_{17} – volume of working fluid, which is in this period of time in the area from cavity 14 to cavity 19 inclusive, m^3.

To determine the gap S_1 in equation (50) buwhere, an additional equation is required, which can be made without proceeding from the equilibrium condition of the closing element (valve opening moment)

$$F_{P9} + F_C^{/} - F_{P11}^{kl} + F_{din}^{/} - F_{P3}^{kl/} + F_{kl}^{a} = 0, \quad (50)$$

where F_{P9} – force of liquid pressure p_9 from the side of cavity 9 on the closing element, *N*; $F_{P11}^{кл}$ – force of liquid pressure p_{11} from the side of cavity 11 on the closing element, *N*; $F_C^{/}$ – the force with which the spring acts on the closing element of the valve 5, *N*; $F_{P3}^{кл/}$ – the force perceived by the valve from acting on its pressure surface $p_{ср}^{/}$ in the gap S_1, *N*; $F_{кл}^{a}$ - valve inertia force, *N*.

Power $F_{P9}^{кл}$ is determined in the following way:

$$F_{P9}^{кл} = f_4 p_9, \quad (51)$$

where f_4 – the area of influence on the closing element of the pressure in the cavity 9, m^2; p_9 – cavity pressure 9, *N*/*m*^{2};

$$F_C^{/} = c\left(X_0 + S_1\right), \quad (52)$$

where c – spring stiffness 6, *N*/*m*; X_0 – spring preload, *m*; S_1 – additional spring tension, according to the gap S_1, m;

$$F_{P11}^{кл} = f_3 \cdot p_{11}, \quad (53)$$

where f_3 – the area of influence on the closing element of the pressure in the cavity 11, *m*^{2}; p_{11} – cavity pressure 11, *N*/*m*^{2}.

The dynamic force is accordingly described by the following relationship:

$$F_{дин}^{/} = \frac{4Q_{11}^2\gamma}{g\pi d_3^2}, \quad (54)$$

$$F_{P3}^{кл/} = f_{4/3} p_{ср}^{/}, \quad (55)$$

where $f_{4/3}$ – the area of influence on the closing element of the pressure in the gap S_1, m^2; $p_{ср}^{/}$ – average pressure in the gap S_4, *N*/m^2.

$$F_{кл}^{а} = m_{кл}\frac{d^2 S_1}{dt^2}, \quad (56)$$

where $m_{кл}$ – weight of the valve, kg.

The value of the average reduced pressure in the gap:

$$p_{ср}^{/} = p_{11} - \frac{\Delta p_{11з}}{2}, \quad (57)$$

Substituting into equation (50), dependencies (51 - 57) get possessed:

$$f_4 p_9 + c\left(X_0 + S_1\right) - f_3 p_{11} + \frac{4Q_{11}^2\gamma}{g\pi d_3^2} - f_{4/3} p_{ср}^{/} + m_{кл}\frac{d^2 S_1}{dt^2} = 0. \quad (58)$$

Let's rewrite the equation (58) in a more visual form:

$$c\left(X_0 + S_1\right) - f_3 p_{11} + \frac{4Q_{11}^2\gamma}{g\pi d_3^2} - \left(f_4 + f_3\right)\frac{\left(2S_1^2\frac{d_4}{d_3} + 0.1555 d_4^2 - 1.85 S_1^2\right)Q_{11}^2\gamma}{4g\pi^2 d_3^2 S_1^4} + m_{кл}\frac{d^2 S_1}{dt^2} = 0. \quad (59)$$

For convenience when writing mowherel (reducing bulkiness), we introduce coefficients that include constant values (constants) that depend on specific design parameters.

The equation for determining the pressure p_{16} let's convert to the form:

$$p_{16} = K_{16} Q_{16}^2. \quad (60)$$

Accordingly, the coefficient K_{16};

$$K_{16}=\left(\sum_{i=1}^{n}\xi_{11}+\lambda_{11}\frac{\sum l_{11}}{d}\right)\frac{\gamma}{2gk^2 f_{11}^2}, \tag{61}$$

and the coefficient K_S;

$$K_S=\frac{\gamma\left(2S\frac{d_4}{d_3}+0.155d_4^2-1.85S_1^4\right)}{2g\pi^2 d_3^2 S_1^4}. \tag{62}$$

Taking into account the above, the equation for p_{13} will take the following form:

$$p_{13}=p_{11}-K_S Q_{11}^2-K_{11}Q_{13}^2. \tag{63}$$

At the same time, it should be taken into account that the coefficient K_S is a function of the gap S_1. Since the gap S_1 – theoretically, the value is variable during the process, so is the coefficient K_S is a variable quantity.

Accordingly, the coefficient K_μ.

$$K_\mu=\frac{\gamma}{2gK_\psi^2\pi^2k^2}. \tag{64}$$

Dependency for pressure calculation:

$$p_{14}=K_{14}Q_{13}^2+\frac{\mu}{S_4^4}Q_{19}^2+K_{16}Q_{16}^2. \tag{65}$$

As a result of transformations, we get the following equation:

$$\begin{gathered}p_{11}f_{пл}-p_{16}f_{пл}-\left(c\left(X_0+S_1\right)+\left(p_{16}+\Delta p_{16}\right)f_4-\right.\\ \left.\left(p_{16}+\frac{\Delta p_{16}}{2}\right)f_{4/3}-p_{16}f_3+\frac{4Q_{19}^2\gamma}{g\pi k^2 d_3^2}\right)-k_{пл}\frac{dS_4}{dt}=m_{пл}\frac{d^2S_4}{dt^2}\end{gathered}. \tag{66}$$

After simplification, equation (66) will have the following form:

$$\begin{gathered}p_{11}f_{пл}-p_{16}f_{пл}-c\left(X_0+S_1\right)-\frac{K_\mu Q_{19}^2}{2S_4^2}\left(f_4+f_3\right)-\\ -K_д Q_{19}^2-k_{пл}\frac{dS_4}{dt}=m_{пл}\frac{d^2S_4}{dt^2}\end{gathered}. \tag{67}$$

The dependence for determining the speed of movement of the hydraulic cylinder rod can be written in the form:

$$\begin{gathered}p_{11}-K_S Q_{11}^2-K_{11}Q_{13}^2-K_{14}\frac{Q_{13}^2}{k}-\frac{K_\mu}{S_4^2}\frac{Q_{19}^2}{k}-\\ -K_{16}\frac{Q_{16}^2}{k}+\frac{F_ц}{f_{13}}-\frac{k_{гц}}{f_{13}}\frac{dS_{гц}}{dt}=\frac{m_{гц}}{f_{13}}\frac{d^2S_{гц}}{dt^2}\end{gathered}. \tag{68}$$

Since the increase in the stroke of the piston over time $dS_{гц}/dt$ is its speed, and if the condition of uninterrupted flow is observed, it is numerically equal to the speed of the fluid in the hydraulic cylinder, then we can write:

$$\frac{dS_{гц}}{dt}=\frac{Q_{13}}{f_{13}}, \tag{69}$$

Then:

$$\begin{gathered}p_{11}-K_S Q_{11}^2-K_{11}Q_{13}^2-K_{14}\frac{Q_{13}^2}{k}-\frac{K_\mu}{S_4^2}\frac{Q_{19}^2}{k}-\\ -K_{16}\frac{Q_{16}^2}{k}+\frac{F_ц}{f_{13}}-\frac{k_{гц}}{f_{13}^2}Q_{13}=\frac{m_{гц}}{f_{13}^2}\frac{dQ_{13}}{dt}\end{gathered}. \tag{70}$$

Thus, a system of equations consisting of one linear and three differential equations with four variables is proposed $F_ц, S_1, Q_{13}, p_{11}$, the solution of which allows to reveal the dynamic characteristics of the hydraulic positioning mechanism, equipped with decelerating equipment in the form of a typical hydraulic lock.

The resulting mathematical problem was solved using the Runge–Kutta–Feldberg numerical method with automatic change of the differentiation step. As a result of numerical modeling, the transient processes of the change of the gap S_4 and the current pressure p_{11} in the cavity 11 of the considered hydraulic mechanism were obtained, depending on the magnitude of the external load on the output link Fc of the executive hydraulic motor of the positioning mechanism. In fig. 3 presents the transient processes that were obtained when using the initial data, according to the hydraulic drive of the Verdis 3.5 cultivator, aggregated with the John Deere 8010 tractor.

Graphs of changes in the gap on the regulating element and pulsation of the pressure rc in the cavity 11 of the decelerating equipment with a constant accompanying load on the rotary mechanism.

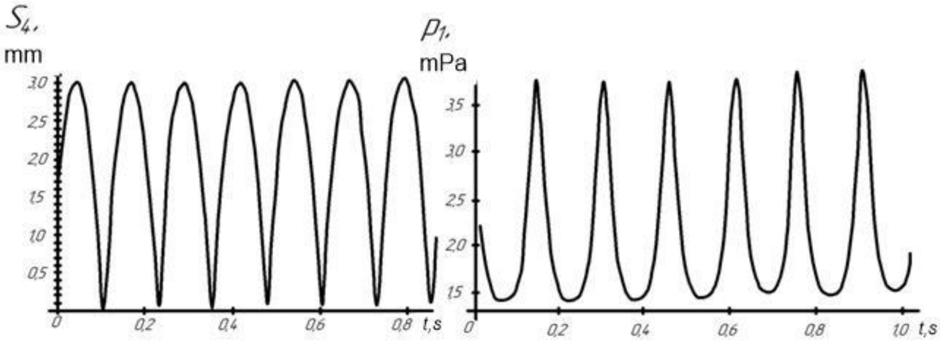

Fig. 4. Transient processes of gap and pressure changes

From a technical point of view, it is quite difficult to track the fluctuations of the regulating element, so it is with the help of mathematical modeling that we can investigate the processes that take place directly in the hydraulic unit.

As can be seen from fig.4, the closing element of the valve performs an oscillating movement with an amplitude of 2 MPa and a period of 0.2 s, and the amplitude of the oscillations increases noticeably. Therefore, the system works in an unstable mode. The presence of unstable operating modes negatively affects the operation of hydraulic systems and mechanical systems in general. The consequences of this phenomenon are premature failure of the locking and regulating equipment, increased wear of hinges and friction pairs.

5. Conclusion

In order to identify and eliminate the cause of the unstable operation of the rotary mechanism, the parameters of the system that have the greatest influence on the control elements should be determined.

The analysis of dynamic characteristics and mathematical description of the positioning mechanism of the section of the wide-grip tillage tool showed that the greatest influence on the working characteristics of the positioning mechanism is revealed by the influence of the regulated pressure drop on the regulating element 6 (Fig. 2). Since the value of the regulating gap S_4 stands at a given total area in the fourth degree in the denominator, and its insignificant change is where there is a significant change in pressure in the system, as a result of which the force on the regulating element increases, causing it to move with significant acceleration. As a result, there is a significant overregulation of pressure, as a result of which the regulating element makes oscillating movements, thereby causing unstable operation of the hydraulic system. It is possible to reduce the influence of the regulated pressure drop on the regulating element in this design both by reducing the diameters of the outlet holes and by reducing the pressure drop in the system.

The pressure drop in this design is determined by the resistance of the hydraulic system and depends on the external load on the hydraulic cylinder rod, that is, on the direct technological purpose of the rotary mechanism. Therefore, reducing the influence of the regulated pressure on the regulating element due to the reduction of the total pressure drop in the system is unacceptable.

A decrease in the diameters of the output holes d_3 will have a negative effect on the functioning of the rotary mechanism in the active load mode, since in this case the reduced cross-section causes additional non-productive energy losses due to throttling.

References

1. Rutkevich, V, Ostapenko, O, Zalogin, R. (2024). Increasing the efficiency of the loader with a hydraulic drive cutting mechanism for unloading silage from trench storage. *Herald of Khmelnytskyi National University. Technical Sciences, 337(3),* 109–114. https://doi.org/10.31891/2307-5732-2024-337-3-15. [in Ukrainian].
2. Veselovska, N., Shargorodsky, S., Rutkevych, V., Kupchuk, I., Burlaka, S. (2023). Development of the mathematical model of the hydraulic drive of the lift mechanism of the working equipment of the front loader. *Przeglad Elektrotechniczny, 99 (10),* 34–38. [in English].
3. 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].
4. Triet Hung HO, Kyoung Kwan AHN. (2010). Modeling and simulation of hydrostatic transmission system with energy regeneration using hydraulic accumulator. *Journal of Mechanical Science and Technology, 24(5),* 1163–1175. [in English].
5. Likaj, R., Shala, A. (2017). Design of automatic system with hydrostatic transmissio. *ANNALS of Faculty Engineering Hunedoara – International Journal of Engineering, XV(2),* 159–162. [in English].
6. Šulc, B., Jan, J.A. (2002). Non Linear Modelling and Control of Hydraulic Actuators. *Acta Polytechnica, 42(3),* 41–47. [in English].
7. Kumar, N., Kumar, R., Kumar, B., Maity, S. (2020). Condition monitoring of hydraulic transmission system with variable displacement axial piston pump and fixed displacement motor. *Materials Today: Proceedings, 46(3),* 1–8. [in English].
8. Biswaranjan Mohanty, Kim A Stelson. (2018). High fidelity dynamic modeling and control of power regenerative hydrostatic wind turbine test platform. *Fluid Power and Motion Control.* Proceedings of *2018 ASME/Bath Symposium, Bath, United Kingdom – Septemebr 12-14.* 1–7. [in English].
9. Deldar, M., Izadian, A., Anwar, S. (2018). Analysis of a hydrostatic drive wind turbine for improved annual energy production. *AIMS Energy, 6(6),* 908–925. [in English].
10. Ivanov, M.I., Hrechko, R.O. (2019). Doslidzhennia vplyvu ob`iemiv hidrolinii na robotu hidrostatychnoi transmisii typu HST90. *Kompleksne zabezpechennia yakosti tekhnolohichnykh protsesiv ta system*: *materialy IX Mizhnarodnoi naukovo-praktychnoi konferentsii, 14-16 travnia 2019 roku*. Chernihiv, 1, 232–233. [in Ukrainian].
11. HMG 4000. URL: https://www.hydac.com/shop/en/GP_1000189407. [in English].
12. Kaletnik, G., Honcharuk, I., Okhota, Y. (2020). The Waste-free production development for the energy autonomy formation of Ukrainian agricultural enterprises. *Journal of Environmental Management and Tourism, 11(3),* 513–522. DOI: 10.14505//jemt.v11.3(43).02. [in English].
13. Honcharuk, I., Kupchuk, I., Solona, O., Tokarchuk, O., Telekalo, N. (2021). Experimental research of oscillation parameters of vibrating-rotor crusher. *Przeglad Elektrotechniczny, 97(3),* 97–100. https://doi.org/10.15199/48.2021.03.19. [in English].
14. Yehorov, B.V. (2018). *Kontrol' iakosti ta bezpeka produktsii v haluzi (kombikormova haluz').* Oldi+. [in Ukrainian].
15. Kupchuk, I., Telekalo, N. (2021). *Substantiation of the constructive and mode parameters of vibratory disc crusher for the farm animals compound feeds production.* Development of scientific, technological and innovation space in Ukraine and EU countries. Sollective monograph. Riga, Latvia: “Baltija Publishing”. [in English].
16. Yanovych, V., Honcharuk, T., Honcharuk, I., Kovalova, K. (2017). Design of the system to control a vibratory machine for mixing loose materials. *Eastern-European Journal of Enterprise Technologies, 6,* 4–13. https://doi.org/10.15587/1729-4061.2017.117635. [in English].
17. Mazur, V.A., Honcharuk, I.V., Didur, I.M., Pantsyreva, H.V., Telekalo, N.V., Kupchuk, I.M. (2021). *Innovatsijni aspekty tekhnolohij vyroschuvannia, zberihannia i pererobky zernobobovykh kul'tur: monohrafiia.* Vinnytsia: VNAU. [in Ukrainian].
18. Poberezhets, Ju., Chudak, R., Kupchuk, I., Yaropud, V., Rutkevych, V. (2021). Effect of probiotic supplement on nutrient digestibility and production traits on broiler chicken. *Agraarteadus, 32(2),* 296–302. https://doi.org/10.15159/jas.21.28. [in English].
19. Petrova, Z., Paziuk, V., Tokarchuk, O., Polievoda, Y. (2021). Special aspects of soybean drying with high seedling vigor. *UPB Scientific bulletin, Series D: Mechanical Engineering, 83(2),* 327–336. [in English].

РОЗРОБКА ДИНАМІЧНОЇ МОДЕЛІ ГІДРАВЛІЧНОГО ПРИВОДУ МЕХАНІЗМУ ПОЗИЦІОНУВАННЯ ШИРОКОЗАХВАТНОГО СІЛЬСЬКОГОСПОДАРСЬКОГО АГРЕГАТУ

Стаття присвячена вирішенню актуальної проблеми розробки динамічної математичної моделі гідравлічного приводу для складання та розкладання широкозахватного сільськогосподарського агрегату та її дослідження. Для досягнення цієї мети було проведено системний аналіз процесів, що відбуваються у гідравлічному приводі, який відповідає за переведення агрегату з транспортного положення в робоче. Досліджено відомі гідравлічні системи. Проведено огляд відомих математичних моделей та способів їх побудови. Проведено аналіз та тенденції розвитку сільськогосподарських агрегатів та знарядь для обробітку ґрунту. Проаналізовано їх переваги та недоліки. Зазначено, що одним із напрямків розвитку сучасних сільськогосподарських агрегатів є збільшення ширини захвату. Дана тенденція є виправданою з точки зору економії палива, скорочення кількості проходів агрегату та зменшення ущільнення ґрунту, що позитивно впливає на його стан ґрунту та підвищує його родючість. До недоліків даних машин слід віднести складність виконання маневрів. Для спрощення маневрування з метою розвороту бажано оснастити ці агрегати системою переведення з транспортного положення у робоче. Дана система має відповідати наступним вимогам, а саме: висока швидкодія, однакова швидкість складання секцій, компактність. Для вирішення даної задачі доцільно використати гідравлічний привод. Цей привод має численні переваги, такі як компактність, висока потужність, здатність працювати на високих швидкостях та гасити вібрації. Запропоновано схему гідравлічного приводу для переведення агрегату з транспортного в робоче положення із поступовим вмиканням гідродвигунів. Розроблено математичну модель приводу, яка включає рівняння балансу рідини, сили та моменти, що діють на робочі органи гідродвигунів. Отримані результати дозволяють ретельно аналізувати процеси в приводі та визначити оптимальні параметри для забезпечення ефективної роботи системи.

Ключові слова: *широкорізний агрегат сільськогосподарського призначення, гідропривод, гідроциліндр, трипозиційний розподільник, математична модель приводу, структурна схема, параметр, супроводження.*

Ф. 70. Рис. 3. Літ. 19.

INFORMATION ABOUT THE AUTHORS

Serhiy SHARGORODSKIY – Candidate of Technical Sciences, Associate Professor of the Department of Machines and Equipment of Agricultural Production of the Vinnytsia National Agrarian University (3, Solnychna str., Vinnytsia, Ukraine, 21008, e-mail: sergey20@vsau.vin.ua, https://orcid.org/0000-0003-2125-773X).

Dmutro KONDRATUK – Recipient of the Third Educational and Scientific Level, Vinnytsia National Agrarian University (3, Solnychna str., Vinnytsia, Ukraine, 21008, e-mail: 1at15b.kondratiuk@gmail.com)

Vita SHARHORODSKA – teacher of English and German. Separate structural subdivision of Technological and Industrial Vocational College of Vinnytsia National Agrarian University (Yunosty Avenue, 8, Vinnytsia, 21021, Ukraine, e-mail vitakomar@yahoo.com, https://orcid.org/0000-0003-1992-1355).

ШАРГОРОДСКИЙ Сергей Анатолоійович – кандидат технических наук, доцент кафедры машин и оборудования сельскохозяйственного производства, Винницкого национального аграрного университета (ул. Солнечная, 3, г. Винница, 21008, Украина, e-mail: sergey20@vsau.vin.ua, https://orcid.org/0000-0003-2125-773X).

КОНДРАТЮК Дмитро Анатолійович – здобувач третього освітньо-наукового рівня, Вінницький національний аграрний університет (вул. Сонячна, 3, м. Вінниця, 21008, Україна, e-mail: 1at15b.kondratiuk@gmail.com)

ШАРГОРОДСЬКА Віта Миколаївна – викладач англійської та німецької мови Відокремлений структурний підрозділ технологічно-промисловий фаховий коледж вінницького національного аграрного університету (проспект Юності, 8, м. Вінниця, 21021, Україна, e-mail: vitakomar@yahoo.com, https://orcid.org/0000-0003-1992-1355)