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## Vibration resistance of HST 90 hydrostatic transmission

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## Vibration resistance of HST 90 hydrostatic transmission

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**Abstract.** Hydrostatic transmissions, which are usually used in the drives of mobile machines, have recently been actively used in drives of other working bodies of machines. For example, in drives of mixers of concrete carriers, drives of working bodies of agricultural machines - straw cutters, etc. This area of use of hydrostatic transmissions is actively developing. The successful solution of this problem is prevented by the occurrence of vibrational operating modes while reducing inertial and technological loads, and in some cases leads to significant disturbances in the operation of machines. Using computer simulation of the hydrostatic transmission, the influence of the inertial and technological loads, as well as the parameters of the hydraulic system, the conditions for the occurrence of vibrational modes of its operation are analyzed. Based on the results obtained, measures have been developed to improve the design of measures and shunt valves, which allow to exclude vibrational modes of operation with a significant expansion of the range of variation of inertial and technological loads.

The results of a bench simulation of hydrostatic transmissions with advanced valves, conducted in the laboratories of PJSC Gidrosila APM (Kropyvnytskyi), confirmed the effectiveness of the proposed measures. The enterprise has begun mass production of hydrostatic transmissions of an improved design.

### 1. Introduction

The use of hydrostatic drives on self-propelled machines has been a global trend in agricultural, construction and road engineering for a long time. One of the main directions of increasing their technical level of agricultural machines is the widespread use of hydraulic drives. The volume hydraulic drive is actively developing and the hydraulic system parameters are increasing every year: nominal pressure, energy saving and reliability [1]. Typical representatives of hydrostatic drives are hydrostatic transmissions (HST 90), which need research and improvement to meet modern requirements, which are determined by the development of agricultural engineering, construction, road and other areas.

The expansion of the use of hydrostatic transmissions for various occasions, which have been widely practiced recently, can lead to unstable operating modes during which significant vibrations occur, which in some cases lead to significant disruptions in the operation of machines. Hydrostatic transmissions are manufactured and delivered as a continuous hydraulic unit, which, in addition to a volumetric pump and a hydraulic motor, includes hydraulic devices - safety, overflow valves and distributors, air conditioners and others. Solving the problem of ensuring the stable operation of a hydrostatic transmission, which eliminates the occurrence of vibrational modes of its operation,

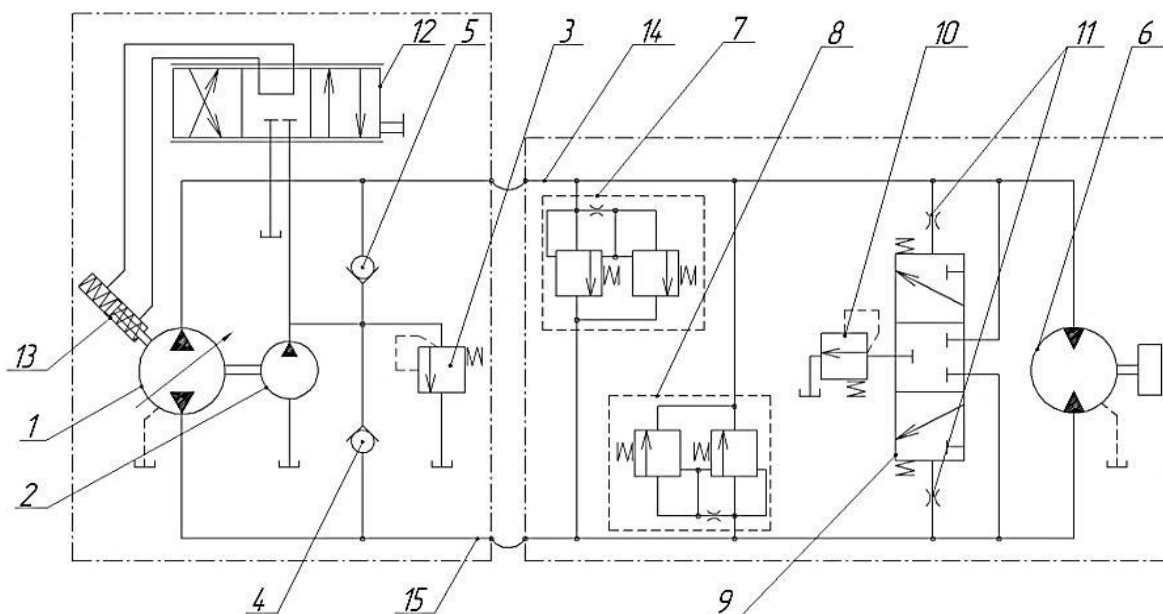


requires a comprehensive review of the features of the hydrostatic transmission, taking into account the expansion of the range of inertial and technological loads.

Current trends in the development of agricultural machinery require the development of fundamentally new and improvement of existing hydraulic drive schemes and designs of hydraulic machines, as well as new approaches to solving the problem of ensuring the reliability and quality of agricultural machinery. An effective method for solving these problems is to simulate the operation of hydraulic systems, which allows you to analyze the processes that occur during their work as part of the machine, taking into account the real conditions of their work. Measures based on the simulation results, are developed, that allow solving the tasks.

## 2. Methods

Today, hydrostatic transmissions of the HST 90 type are manufactured by the Hydrosila APM Prat (Kropyvnytskyi) under a license from Sauer-Sundstrand, the hydraulic circuit of which is shown in Figure 1.



**Figure 1.** Hydraulic diagram of a hydrostatic transmission

1 - pump; 2 - feed pump; 3, 7, 8 - safety valves; 4, 5 - check valves; 6 - hydro motor; 9 - shunt distributor; 10 - overflow valve; 11 - a throttle; 12 - the distributor; 13 - a hydraulic cylinder; 14 - discharge hydro line; 15 - suction hydro line.

The main components in the design of the HST 90 are an adjustable axial rotary piston pump 1, connected by hydraulic lines 14, 15 to an unregulated axial piston hydraulic motor 6. A gear pump 2 is installed on the same shaft as pump 1, which serves to pump fluid through check valves 4 and 5 of the working fluid in the main hydraulic lines 14 and 15. To prevent exceeding the maximum allowable pressure in the hydraulic make-up line, a safety valve 3 is installed.

In the cover of the hydraulic motor 6, two safety valves of indirect action 7, 8 are mounted. They are designed to limit peak pressures in the main hydraulic lines 14 and 15 and to supply the working fluid from the charge pump to the suction line of the main pump 1. In order to supply the working fluid for cooling and purification of wear products, a shunt distributor 9 is installed in the valve box of the hydraulic motor 6, which connects the line with lower pressure (drain line) to the drain line into the hydraulic motor housing through the overflow valve 10. Overflow valve 10 - is designed to maintain pressure in the drain hydraulic line. On this hydraulic line, the working fluid at a pressure of 10-12.7 bar enters the heat exchanger (radiator) and further into the tank [2-4].

To study the operation of a hydrostatic transmission of the HST90 type by mathematical modeling, a design diagram is developed, which is shown in Figure 2. The diagram shows the generalized coordinates of the elements of the system - for mechanical links it is linear or angular displacement, and for a hydraulic system - pressure and flow rate of the working fluid in typical areas. Also, the calculation diagram indicates the parameters of physical processes that were considered during mathematical modeling, namely: the volume of cavities of characteristic sections, leakage and overflow coefficients, viscous friction coefficients, masses and moments of inertia of moving parts, and stiffness of elastic elements.

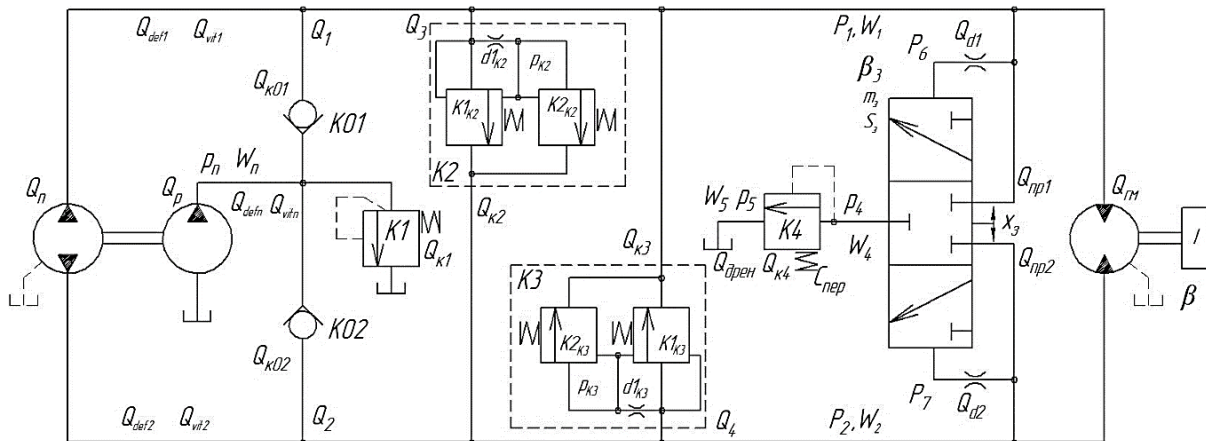


Figure 2. Calculation scheme of hydrostatic transmission type HST90.

The mathematical model of hydrostatic transmission includes equations that describe the change in each generalizing coordinate. The physical laws that determine the features of processes during hydrostatic transmission are the equations of continuity of the flow of the working fluid in the characteristic sections of the hydraulic system, as well as the equation of forces and moments that determine the laws of motion of the mechanical parts of the system.

The mathematical model of hydrostatic transmission includes such equations of continuity of working fluid flows in characteristic sections of the hydraulic system.

$$\begin{cases}
 Q_P = -Q_{k01} + Q_{k2} - Q_{k3e} + Q_{d1} + Q_{sp1} + Q_{hm} + Q_{leak1} + Q_{def1}, \\
 Q_{hm} = Q_{sp2} + Q_{d2} - Q_{k2e} + Q_{k3} - Q_{k02} + Q_p + Q_{leak2} + Q_{def2}, \\
 Q_n = Q_{k01} + Q_{k02} + Q_{k1} + Q_{leakn} + Q_{defn}, \\
 Q_{sp} = Q_{k4} + Q_{vk4} + Q_{leak3} + Q_{def3}, \\
 Q_{d1k2} = Q_{d2k2} + Q_{ut1} + Q_{def4}, \\
 Q_{d1k3} = Q_{d2k3} + Q_{ut2} + Q_{def5}, \\
 Q_{d1} = Q_{vsp} + Q_{ut3} + Q_{def6}, \\
 Q_{d2} = Q_{vsp} + Q_{ut4} + Q_{def7},
 \end{cases} \tag{1}$$

where  $Q_P$  - pump flow,  $Q_{k01}$ ,  $Q_{k02}$  – working fluid flow rates through check valves  $K01$  and  $K02$ ,  $Q_{k2}$ ,  $Q_{k3}$  – working fluid flow rates through safety valves  $K2$  and  $K3$ ,  $Q_{k2e}$ ,  $Q_{k3e}$  – the flow rate of the working fluid at the outlet of the valves  $K2$  and  $K3$ ,  $Q_{d1}$ ,  $Q_{d2}$  – the flow rate of the working fluid through the throttle  $d1$  and  $d2$  in the cavity under the ends of the shunt distributor,  $Q_{sp1}$ ,  $Q_{sp2}$  – the flow rate of the working fluid through the working windows of the shunt distributor,  $Q_{hm}$  – the flow rate of the working fluid that the hydraulic motor consumes,  $Q_n$  – feed pump feed,  $Q_{k1}$  – working fluid flow

through safety valve  $K1$ ,  $Q_{np}$  – working fluid flow through a shunt distributor,  $Q_{\kappa 4}$  – fluid flow through overflow valve  $K4$ ,  $Q_{v\kappa 4}$  – fluid flow caused by the movement of the overflow valve  $K4$  from speed  $V$ ,  $Q_{d1k2}$ ,  $Q_{d1k3}$  – the flow rate of the working fluid through the throttle of the main valves  $K2$  and  $K3$ ,  $Q_{d2k2}$ ,  $Q_{d2k3}$  – the flow rate of the working fluid through the throttle auxiliary valves  $K2$  and  $K3$ ,  $Q_{vsp}$  – is the flow rate of the working fluid caused by the movement of the shunt distributor with speed  $V$ ,  $Q_{leak1}$ ,  $Q_{leak2}$ ,  $Q_{leak3}$ ,  $Q_{leakn}$ ,  $Q_{ut1}$ ,  $Q_{ut2}$ ,  $Q_{ut3}$  – costs of leaks and overflows of the working fluid from the corresponding hydraulic lines,  $Q_{def1}$ ,  $Q_{def2}$ ,  $Q_{def3}$ ,  $Q_{def4}$ ,  $Q_{def5}$ ,  $Q_{def6}$ ,  $Q_{def7}$ ,  $Q_{defn}$  – the flow rate of the working fluid caused by the deformation of the voids filled with the working fluid under the action of pressures  $p_1$ ,  $p_2$ ,  $p_4$ ,  $p_{k2}$ ,  $p_{k3}$ ,  $p_6$ ,  $p_7$ ,  $p_n$ .

The mathematical model of hydrostatic transmission includes such equations of force and motion of mechanical links.

$$\left\{ \begin{array}{l} F_{\kappa 1k2} = C_{1zk} \times \Delta x_{\kappa 1k2} + m_p \times \frac{d^2 x_{\kappa 1k2}}{dt^2} + \beta_{zk} \times \frac{dx_{\kappa 1k2}}{dt}, \\ F_{\kappa 2k2} = C_{2zk} \times \Delta x_{\kappa 2k2} + m_z \times \frac{d^2 x_{\kappa 2k2}}{dt^2} + \beta_{zk} \times \frac{dx_{\kappa 2k2}}{dt}, \\ F_{\kappa 1k3} = C_{1zk} \times \Delta x_{\kappa 1k3} + m_p \times \frac{d^2 x_{\kappa 1k3}}{dt^2} + \beta_{zk} \times \frac{dx_{\kappa 1k3}}{dt}, \\ F_{\kappa 2k3} = C_{2zk} \times \Delta x_{\kappa 2k3} + m_z \times \frac{d^2 x_{\kappa 2k3}}{dt^2} + \beta_{zk} \times \frac{dx_{\kappa 2k3}}{dt}, \\ F_{\kappa 4} = C_{\kappa 4} \times \Delta x_{\kappa 4} + m_{\kappa 4} \times \frac{d^2 x_{\kappa 4}}{dt^2} + \beta_{\kappa 4} \times \frac{dx_{\kappa 4}}{dt}, \\ F_3 = C_3 \times \Delta x_3 + m_3 \times \frac{d^2 x_3}{dt^2} + \beta_3 \times \frac{dx_3}{dt}, \\ M_{zM} = I \times \frac{d^2 j}{dt^2} + \beta_{zM} \times \frac{dj}{dt} + M_{mex}, \end{array} \right. \quad (2)$$

where  $F_{\kappa 1k2}$ ,  $F_{\kappa 1k3}$  are the hydrostatic pressure forces that act on the spools of the safety valves  $K2$  and  $K3$ ,  $C_{1zk}$  is the stiffness of the springs of the safety valves  $K2$ ,  $K3$ ,  $C_{2zk}$  is the stiffness of the needle springs of the safety valves  $K2$  and  $K3$ ,  $\Delta x_{\kappa 1k2}$ ,  $\Delta x_{\kappa 1k3}$  is the change in the tension of the springs depending on the movement of the piston of the safety valves  $K2$  and  $K3$ ,  $\Delta x_{\kappa 2k2}$ ,  $\Delta x_{\kappa 2k3}$  – a change in the tension of the needle springs depending on the movement of the needle of the safety valves  $K2$  and  $K3$ ,  $\Delta x_{\kappa 4}$  – the change in the tension of the spring of the warning valve  $K4$  pan, depending on the movement of the locking and regulating element,  $\Delta x_3$  – change in the spring tension of the spool of the shunt valve depending on the movement of the spool,  $m_p$  – mass of the spools of the safety valves  $K2$ ,  $K3$ ,  $m_{\kappa 4}$  – mass of the locking and regulating element of the valve  $K4$ ,  $m_z$  – mass of the spool of the shunt valve,  $\beta_{zk}$ , is the coefficient of knitting friction that occurs when the locking and regulating elements of the safety valves  $K2$ ,  $K3$ ,  $\beta_{\kappa 4}$  are the coefficient of astringent friction that occurs when the locking and regulating movement of the valve element  $K4$ ,  $\beta_z$  is the coefficient of knitting friction that occurs when the shunt valve spool moves,  $M_{gm}$  is the torque created by the pressure difference in the voids of the hydraulic motor,  $I$  – is the reduced moment of inertia on the hydraulic motor shaft,  $\varphi$  – is the rotational speed of the hydraulic motor shaft,  $\beta_{gm}$  – coefficient of knitting friction that occurs during operation of the cylinder block of the hydraulic motor,  $M_{tehn}$  – the moment of the technological load on the shaft of the hydraulic motor.

The study of the mathematical model is based on solving a system of nonlinear differential equations. At present, numerical methods for studying nonlinear systems are widely used, which make

it possible to obtain information about the system camp for any time by calculating the transient processes that occur during the operation of the system.

The presented mathematical model of the pump includes algebraic and differential equations of the 22nd order. Some composite systems are described by nonlinear dependencies, which complicates the solution of equations. Therefore, to study the mathematical model, a numerical method for solving equations was chosen using a program created in the programming environment Borland Delphi. The developed program calculates the differential equation systems of the mathematical model behind the Runge-Kutta-Feldberg numerical solution method.

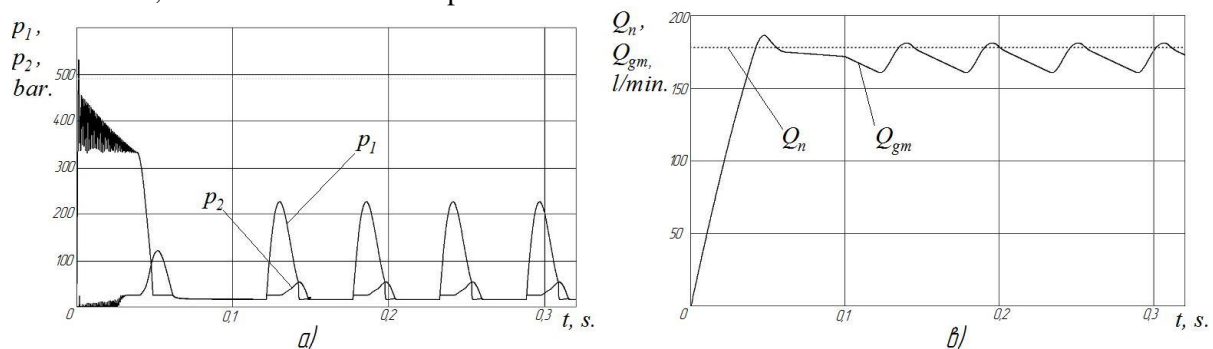
The simulation of the hydrostatic transmission was carried out on the basis of the following parameters of the HST 90 hydrostatic transmission: pump volumetric capacity  $V_n = 89 \text{ cm}^3$ , make-up pump volumetric capacity  $V_{nr} = 18.1 \text{ cm}^3$ , volumetric capacity of the hydraulic motor  $V_{gm} = 89 \text{ cm}^3$ , pump shaft speed  $n = 2000 \text{ rpm}$ ., the pressure relief valve K1 of the pump is adjusted to a crush of 25 Bar, the coefficient of astringent friction when the shut-off and regulating elements of the safety valves move  $\beta_{zk} = 0.1 \text{ kg}\cdot\text{cm}/\text{c}^2$ , the coefficient of astringent friction when the shut-off and regulating element moves over of the drain valve  $\beta_{k4} = 0.6 \text{ kg}\cdot\text{cm}/\text{c}^2$ , the coefficient of knitting friction during movement of the spool of the shunt distributor  $\beta_3 = 0.6 \text{ kg}\cdot\text{cm}/\text{c}^2$ , the coefficient of knitting friction that occurs when the cylinder block of the hydraulic motor is  $\beta_{gm} = 0.44 \text{ kg}\cdot\text{cm}/\text{c}^2$ , the safety valves K2 and K3 of the discharge and suction hydraulic lines are configured the operating pressure of 320 Bar, the spring stiffness of the safety valves K2 and K3 is equal to  $C_{1zk} = 4.2 \text{ N}/\text{mm}$ ,  $C_{2zk} = 4.9 \text{ N}/\text{mm}$ , the spring stiffness of the overflow valve K4  $S_{k4} = 5.7 \text{ N}/\text{mm}$ , the inertial load on the shaft hydraulic motor  $I = 1000 \text{ kg}\cdot\text{cm}^2$ , reduced torque of those technological load on the shaft of the hydraulic motor  $M_{ex} = 10 \text{ H m}$ .

### 3. Results and discussion

As a result of computer simulation of a hydrostatic transmission, an oscillogram of the processes of time-varying pressure in the hydraulic system, flow rates of the working fluid and movement of the locking and regulating elements of the hydrostatic transmission of the GST 90 type was obtained, which made it possible to identify the features of the hydrostatic transmission when the inertial and technological loads change

#### 3.1 Influence of inertial load on vibration resistance of hydrostatic transmission

Hydrostatic transmissions are mainly used as a propulsion drive for self-propelled machines, however, today the range of applications for such transmissions has expanded significantly, as manufacturers of not only self-propelled machines, but also stationary equipment, use hydrostatic transmissions as occasions for the working bodies of machines for different functional purposes. So hydrostatic transmissions of the HST 90 type are used as a drive for the feed rollers of straw cutters, concrete mixer trucks, and a number of other special machines.



**Figure 3.** Oscillograms of transients of hydrostatic transmission type HST90; a) a change in pressure in the pressure  $p_1$  and suction  $p_2$  hydraulic lines, b) graphs of the change in the pump supply time  $Q_n$  and the flow rate  $Q_{gm}$ , which the hydraulic motor consumes.

In the process of studying the mathematical model, the existence of compounds of hydrostatic transmission parameters was revealed, in which unstable operating modes arise. Such modes occur under conditions when a hydrostatic transmission is operating when the inertial load on the motor shaft changes (decreases). This leads to a significant amplitude of pressure fluctuations in the discharge and suction hydraulic lines (Figure 3, a), and the operation of the hydraulic motor in a galloping mode (Figure 3, b). The hydraulic motor starts to work periodically as a pump, the pressure in the hydraulic suction line rises, while the pressure in the discharge line drops. There is a pressure differential on the shunt distributor, which changes the switching of the hydraulic lines, which ultimately causes a pulse increase in pressure and the corresponding fluctuations. As a result of this, a galloping mode of movement of the initial links of the working bodies occurs, which causes vibrations and oscillations of the entire machine.

The problem is solved by making changes to the design of the hydrostatic transmission by adding new elements [5]. Due to the installation of permanent chokes 11 (Figure 1) at the inlet to the cavities under the ends of the spool of the shunt distributor 9, the speed of its spool decreases when the hydraulic line switching signal is worked out to direct part of the working fluid from the main hydraulic line, which is under low pressure, to the cooling tank. Due to this, while reducing the technological and inertial loads on the shaft of the hydraulic motor, there is no excitation of oscillations of the shunt distributor and, accordingly, fluctuations in the speed of the hydraulic motor and pressure.

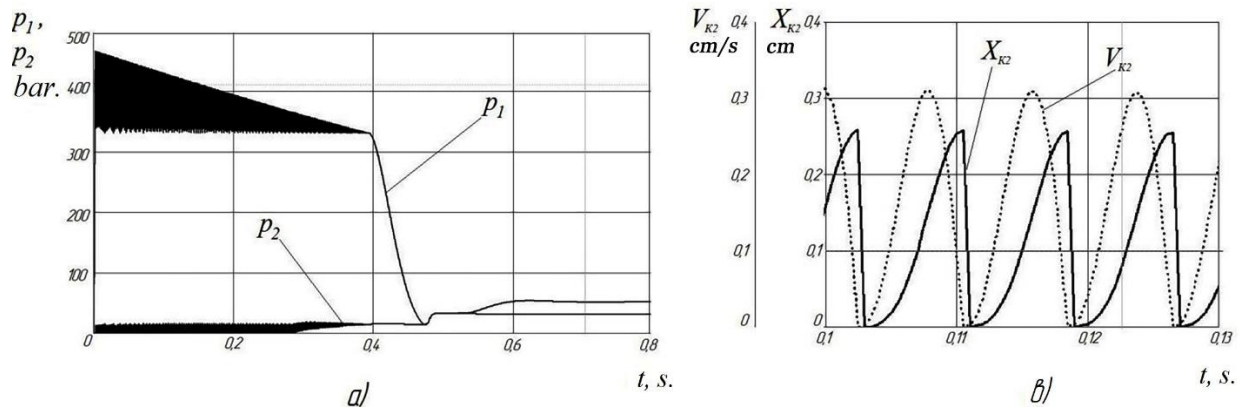
To confirm the results of computer simulations in the laboratories of "Hydrosila APM" (Kropyvnytskyi), bench simulations of hydrostatic transmissions with fixed constant chokes at the entrance to the cavities under the ends of the spool of the shunt distributor were carried out. The results of the bench simulation note the significant effectiveness of the proposed measures. The conducted experimental studies have shown high stability of the rotation frequency of the hydraulic motor shaft with significant reductions in technological and inertial loads. During the test, there was no missing load - there was no reduced moment of inertia on the hydraulic motor shaft. Thus, the actual inertial load was determined by the moment of inertia of the rotor of the hydraulic motor. The pressure drop in the main hydrostatic transmission hydraulic lines did not exceed 5 bar, which corresponds to the level of mechanical losses in the hydraulic motor.

### *3.2 The effect of safety valves on the vibration resistance of a hydrostatic transmission*

At present, manufacturers of hydrostatic transmissions pay considerable attention to the design of safety valves, since the quality of their characteristics significantly affects the vibration resistance of the hydrostatic transmission and the operability of the drive as a whole. In the design of hydraulic actuators of past years, direct-acting safety valves are widely used. However, such valves have a number of drawbacks, they must have a spring designed for significant compression force, there is also a significant increase in pressure with an increase in fluid flow through the valve and instability in operation, especially at high pressure. The absence of damping elements in the design of these valves makes them very sensitive to pressure fluctuations.

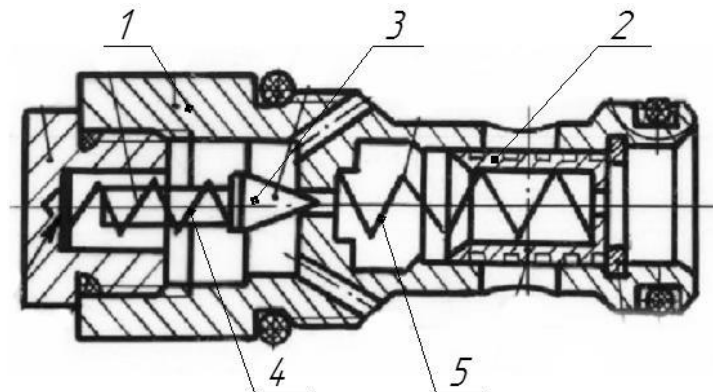
These shortcomings of direct-acting safety valves were identified during the simulation of the hydrostatic transmission using these valves. In Fig. 4, a, the changes in time shown by the oscillograms of the pressure in the hydraulic lines of the HST 90 during the start of the hydrostatic transmission. Significant in amplitude pressure fluctuations  $p_1$  arise in the discharge line, the peak values of which at the time of starting the pump reach 475 bar with an oscillation frequency of 625 Hz. The reason for these fluctuations is the oscillatory mode of operation of the direct-acting warning valve. As can be seen from the oscillograms in Fig. 4, the movement of the valve spool  $X_{k2}$  corresponds to the oscillatory mode, and the movement  $X_{k2}$  and the valve speed  $V_{k2}$  periodically drops to zero, which corresponds to contact with the seat. At the same time, the slide valve landing on the saddle occurs with significant acceleration, which leads to a shock action on the saddle, which in this case is made in the form of a sharp edge.





**Figure 4.** Oscillograms of transients of hydrostatic transmission type HST 90: a) a change in pressure in the pressure  $p_1$  and suction  $p_2$  hydraulic lines; b) graphs of changes in the movement of  $X_{k2}$  and speed  $V_{k2}$  of the spool of the direct-acting warning valve.

As a result of this valve operating mode, the hydrostatic transmission perceives vibration loads, which can cause premature exit of the warning valve [6].

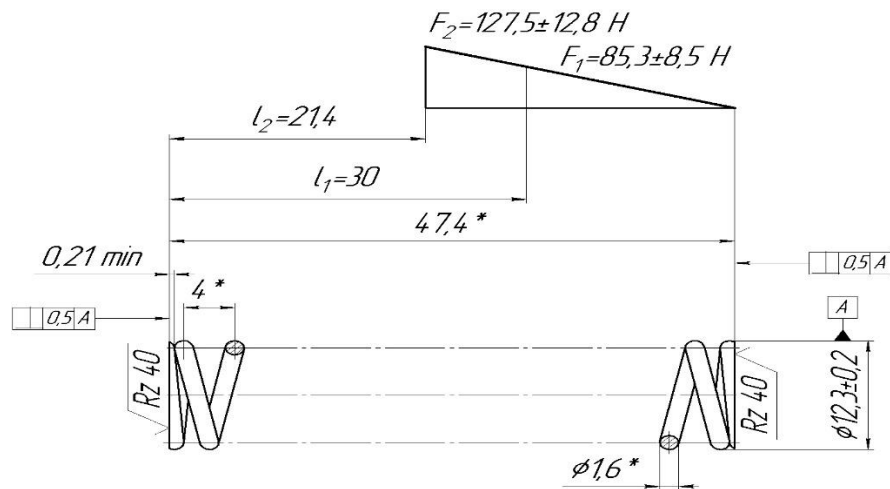


**Figure 5.** Safety valve of indirect action 1 - body; 2 - spool; 3 - needle; 4 - spring of the needle; 5 - piston spring.

To ensure reliable operation, the latest hydrostatic transmissions (Figure 1), which operate at a peak pressure of 400-500 Bar, use safety relief valves of indirect action (Figure 5), in which the movement of the main locking element - spool 2 depends on the movement of the auxiliary locking-regulating element - needles 3. Spool 2 is pressed against the valve seat by the force of the fluid pressure and the strength of a relatively weak spring 5. When the pressure rises to the set value, the needle 3 rises, the crush after the spool 2 falls and moving it recirculate liquid to drain. As a result of the damping effect of the hole on the end face of the spool 2, there are no significant fluctuations in the latter and, accordingly, pressure fluctuations in the pressure head hydraulic line of the hydrostatic transmission.

As a result of a computer simulation of the operation of a hydrostatic transmission with indirect warning valves, the influence of the spring stiffness of the warning valve needle on the vibration resistance and the nature of the HTS 90 operation was revealed. In domestic hydrostatic transmissions, the setting pressure of the safety valves is 320 Bar. To control the valve operating pressure, washers are used that place needles under the spring, thereby changing the spring pre-compression.





**Figure 6.** Spring needle indirect safety valve.

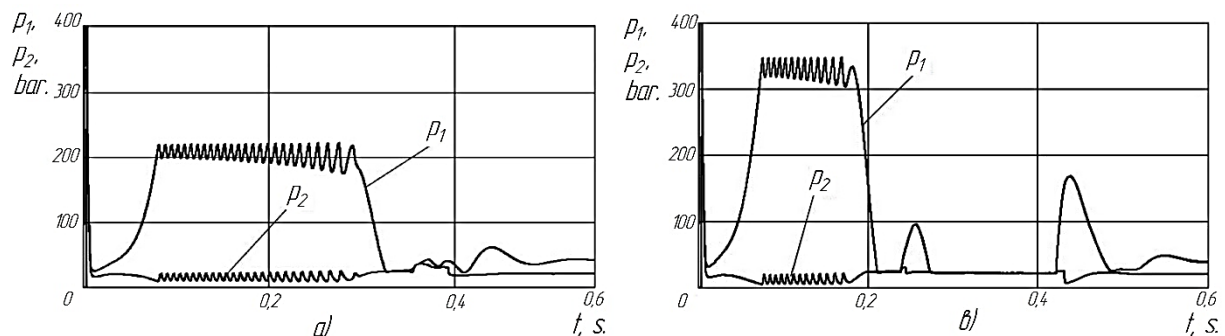
Figure 6 shows the drawings of the needle spring of an indirect pressure relief valve, from which the necessary parameters for mathematical modeling were determined:

- initial compression of the spring

$$l = l_0 - l_1 = 47,4 - 30 = 17,4 \text{ mm} \quad (3)$$

- spring stiffness

$$C = F_1 / l = 8,5 / 17,4 = 0,49 \text{ N/mm} \quad (4)$$



**Figure 7.** Oscillograms of pressure changes in hydraulic lines of hydrostatic transmission of HST 90 type a) with the spring of the needle warning valve initial stiffness; b) with a spring of a needle of the warning valve of the increased rigidity.

The first experiments were carried out using the parameters of the needle spring according to the drawings. The obtained oscillograms (Figure 7, a) display the pressures  $p_1$  in the discharge and  $p_2$  in the hydrostatic transmission suction hydraulic lines during the start-up process. To achieve a valve setting pressure of 320 Bar, the needle spring must be replaced with a stiffer one. Using mathematical modeling, it was determined that the required spring pressure of the safety valve needle should be 0.82 N/mm.

The oscillograms (Figure 7, b) show the pressure changes  $p_1$  and  $p_2$  of a hydrostatic transmission of the HST90 type with indirect warning valves with a needle spring of increased stiffness. As can be seen from the oscillograms, the response pressure of the warning valve corresponds to a value of 320 Bar. At the same time, the valve operating time in the oscillatory mode will be significantly reduced, which significantly improves the operational performance of the hydrostatic transmission as a whole.

Safety valves are important components of a hydrostatic transmission of the HST 90 type, therefore, there is a need for further research in order to improve performance and improve their designs.

#### 4. Conclusions

Hydrostatic transmissions are complex mechanisms that have a number of components that can cause vibrations and oscillations in the hydraulic system. These components reduce the quality, reliability and durability of these machines. In this regard, studies of the hydrostatic transmission and the search for solutions to the identified drawbacks are necessary.

The study was conducted by mathematical modeling and computer simulation. As a result, the causes of oscillations and vibrations in the hydraulic system, which may be the cause of the failure of the entire drive, are identified. The use of hydrostatic transmissions in drives with a change in the inertial load on the motor shaft in a wide range can lead to a pumping mode of operation at a low inertial load on the shaft. This mode leads to self-oscillations in the hydraulic system and the machine as a whole, which excludes the possibility of operating such machines. To solve this problem, it was proposed to establish constant throttle at the entrance to the cavities under the ends of the spool of the shunt distributor. Laboratory studies confirm the effectiveness of this proposal. This made it possible to establish serial production of hydrostatic transmissions of an improved design.

Factors are found due to which the safety valves can cause oscillations in the hydrostatic transmission. The use of safety valves of archaic designs, such as direct-acting safety valves, is not rational at present. In addition to the known drawbacks, computer simulation showed that such valves are the cause of fluctuations in the hydraulic system, which negatively affect the reliability of the hydrostatic transmission. Therefore, it is advisable to use indirect pressure relief valves that do not cause pressure fluctuations in the hydraulic system. At the same time, it is obvious that with current trends in increasing nominal pressure, there is a need for significant changes in the design of safety valves.

#### References

- [1] Panchenko A I, Voloshin A A, Obernikhin Yu P 2013 The main directions of hydration of mobile equipment *Proc. of the Tavriya State Agrotechnological University* **13** vol 6 pp 3-19
- [2] Gevko B M, Bilyk S G, Linik A Yu, Flonts A V 2016 *Hydraulic actuator and hydroautomatics of agricultural machinery* (Ternopil: TNTU im. I. Pulyuia) p 384
- [3] Pogorelets O M, Volyansky M S, Voytyuk V D 2004 *Hydraulic actuator of agricultural machinery* (Kyiv: Vyshcha schola) p 368
- [4] Lovkis Z V 1990 *Hydraulic actuators of agricultural machinery: design and calculation* (Moscow: Agropromizdat) p 239
- [5] Shamshur O Z, Ivanov M I, Mazurenko V F, Pereyaslavsky O M Patent Golovko S M Hrechko R O 2019 *Hydrostatic transmission* (Patent UA 136934 U of Ukraine, IPC8 B60K7/10, F15B15/00 № u201905741, app. 27/05/2019; publ. 09/10/2019 Bull. № 17 from 10.09. 2019)
- [6] Ivanov M I, Pereyaslavsky O M, Kovaleva I M, Hrechko R O 2018 Simulation of hydrostatic transmission type HST 90 *Industrial hydraulics and pneumatics* **62** vol 4 pp 51-62