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Development of the mathematical model of the hydraulic drive of the lift mechanism of the working equipment of the front loader

***Abstract**. This article presents the development of a mathematical model for the hydraulic drive of the lift mechanism of the working equipment of a front loader. The aim of the study is to optimize the performance of the lift mechanism by accurately modeling its hydraulic system. The model is developed using the principles of fluid mechanics and the governing equations are derived for the hydraulic system. The model is then validated using experimental data obtained from the lift mechanism of an actual front loader.*

Streszczenie.** W artykule przedstawiono opracowanie modelu matematycznego napędu hydraulicznego mechanizmu podnoszenia osprzętu roboczego ładowacza czołowego. Celem badań jest optymalizacja pracy mechanizmu podnoszącego poprzez dokładne modelowanie jego układu hydraulicznego. Model został opracowany z wykorzystaniem zasad mechaniki płynów, a równania rządzące układem hydraulicznym zostały wyprowadzone. Model jest następnie weryfikowany przy użyciu danych eksperymentalnych uzyskanych z mechanizmu podnoszenia rzeczywistej ładowarki czołowej.* (Opracowanie modelu matematycznego napędu hydraulicznego mechanizmu podnoszenia osprzętu roboczego ładowacza czołowego***)

Keywords: front loader, lifting mechanism, hydraulic drive, mathematical model, technological load, hydraulic cylinder.
Słowa kluczowe: ładowacz czołowy, mechanizm podnoszenia, napęd hydrauliczny, model matematyczny, obciążenie technologiczne, siłownik hydrauliczny.

Introduction

Front-end loaders are widely used in various sectors of the national economy, have high mobility, a wide range of technological operations due to variable working equipment, are convenient when performing small volumes of work on unprepared loading platforms, and are maximally hydraulic. At the same time, a large number of technological operations with various types of cargo determine the increased dynamic load capacity of the equipment, which depends on both technical and operational parameters and dynamic characteristics designs [1].

Increasing the speed of work, expanding the range of technological operations require new approaches in the process of designing and manufacturing wheel loaders. The main direction of optimal design of the structure and its manufacture is a reliable assessment of loading modes. The traditional calculation methods used in design and construction organizations, which do not take into account real loads in operating conditions, do not allow obtaining reliable and highly efficient hydrofication equipment for agricultural enterprises [2, 3].

In this regard, the task of developing a mathematical model of the hydraulic drive of the lifting mechanism of the working equipment of the front-end loader, which would take into account the variable technological load in the attached equipment of the loaders at the design stage, became urgent.

Analyzing the ways to solve the problem

Modern energy-rich and high-performance machinery of domestic and foreign production is equipped with hydraulic, electric, pneumatic, electro-hydraulic systems of automatic or mechanical control. These systems are used to change the vertical or horizontal position and mode of operation of individual working bodies and units, turn on the drive, facilitate maneuvering during operation, and increase the efficiency of the use of equipment in various technological operations [4, 5, 6].

Analysis of the design of modern equipment of domestic and foreign production showed that one of the most responsible systems affecting the reliability and performance of the equipment is the hydraulic system [4, 7, 8]. Hydraulic systems are used to control and transfer energy to various units and units. The failure of the hydraulic system during operation leads to the loss of efficiency of the entire equipment as a whole, as a result of which there are costs for carrying out repair work and losses due to its downtime [2, 9, 10].

Currently, modern domestic and foreign agricultural machinery (grain and forage harvesters, front loaders, tractors, self-propelled mowers, etc.) includes various hydraulic systems in its design [11].

The hydraulic system of the machine most often consists of a large number of elements, but to describe the dynamics of the machine, it is important to create a model of the hydraulic system as a whole [12, 13]. In the hydraulic mechanisms of machines, dynamic processes occur either as a result of a change in the technological regime, when the loads acting on the working body of the hydraulic mechanism change according to a certain law, or as a result of the influence on the hydraulic system of control equipment, that is, distributors, various types of valves. The course of dynamic processes depends, first of all, on the system parameters, that is, on the size and mass distribution of individual system elements, fluid elasticity, hydraulic lines and solid links, dissipative resistances of system elements, as well as external and internal resistances.

It can be said that the theory of studying the hydraulic drive has already been formed. Her works are dedicated to V.M. Prokofieva, T.M. towers, M.S. Gaminina, E.M. Khaymovicha, V.P. Bocharova, I.A. Nemyrovsky, G.Y. Zayonchkovsky, R.D. Iskovich-Lototskyi, B.L. Korobochkina, E. Ivanova, L.P. Wednesday, Z.Ya. Lurie, Z.L. Finkelshtein, O.M. Yahna, V.B. Strutynskyi and others [14, 15]. In their works, the fundamental foundations of the construction of hydraulic devices are considered, which are based on their complete mathematical models, and allow obtaining hydraulic aggregates with specified static and dynamic characteristics. According to this theory, to describe the dynamic processes taking place in the hydraulic system of the machine, it is necessary to compile a system of equations describing the behavior of each hydraulic element, and also, if necessary, take into account the properties of the liquid as a material [16, 17]. When considering fairly complex hydraulic circuits, this leads to the creation in the general case of a system of nonlinear differential equations that simultaneously describes fast and slow processes [18, 19, 20]. If these difficulties can be solved in one way or another to describe an already existing system, then at the stage of designing such machines, when different options are considered, they may turn out to be insurmountable. Evaluation of the dynamics of the machine generally requires the creation of simple adequate dynamic models of the hydraulic system that describe only the processes of interest to the researcher [10, 13, 21]. All of the above determines the relevance of this study.

The purpose and tasks of the research

The purpose of the work is to increase the efficiency of front-end loaders by developing a mathematical model of the hydraulic drive of the lifting mechanism of the front-end loader's working equipment, which would take into account the variable technological load in the attached equipment of the loaders at the design stage.

Research objectives:

1) Construction of the calculation scheme of the hydraulic drive of the lifting mechanism of the working equipment of the front-end loader;

2) Develop a mathematical model of the hydraulic drive of the lifting mechanism of the working equipment of the front-end loader to study its dynamic properties in the functional mode.

Materials and methods

The experimental part of the work was performed at the laboratory stands of the "Machines and Equipment of Agricultural Production" department of the Vinnytsia National Agrarian University. The purpose of experimental research was to prove the effectiveness of mathematical modeling in the creation of new and modern agricultural machinery.

Theoretical methods of research of the hydraulic drive of the lifting mechanism of the working equipment of the front-end loader are based on analytical methods of kinematic and dynamic analysis of machine units, methods of dynamic research of mechanisms taking into account the elasticity of links, classical theory of oscillations, computational mathematics using fundamental laws of hydraulics, hydromechanics and theoretical mechanics using differential calculus. The MathCAD, Delphi application program package was used to process research results and develop calculation methods.

Research results

One of the main drives of intermittent loaders is the loader boom lift drive, the schematic diagram of which is shown in fig. 1. The drive works as follows. The working fluid from the pump enters the hydraulic distributor with manual control. If the spool is in the extreme right position, the working fluid enters the left piston cavity of the hydraulic cylinder, the boom is raised. In the neutral position of the spool, the working fluid enters the tank. In the extreme right position of the spool, the working fluid enters the right cavity of the hydraulic cylinder, the boom is lowered.

The following assumptions were made during the development of the mathematical model:

– the density, viscosity and flow rate of the working fluid do not depend on the temperature due to the system in a stable temperature regime [22];

– pressure losses in internal channels and on external valves are not taken into account, as they usually have an insignificant value;

- the coefficient of compliance of the liquid does not depend on the pressure and the content of the gas component, since in the established mode of operation its value changes slightly;

– the pressure of the support on the drain is insignificant and practically unchanged;

– the coefficient of liquid leaks and flows in the components of hydraulic aggregates is constant and does not depend on the size and shape of the gaps;

- the distance between the elements of the hydraulic system is insignificant, which allows it to be considered as a system with concentrated parameters and not to consider the influence of wave processes;

- pulsation of the pump supply, taking into account its significant frequency, does not cause pressure fluctuations in the hydraulic system.

We believe that the distributor is in the extreme right position. In this case, the working fluid is fed into the piston cavity of the hydraulic cylinder. The flow equation for the pressure cavity of the hydraulic cylinder will be as follows:

1. $$Q_H = Q_{ц1} + Q_{ут.ц.1} + Q_{пер.ц} + Q_{деф.1}$$

where Q_H –flow of liquid supplied from the pumping station to the first cavity of the hydraulic cylinder; $Q_{ц1}$ – fluid flow in the piston cavity, which ensures the movement of the piston at a given speed; $Q_{ут.ц.1}$ –flow rates of liquid from the piston cavity of the hydraulic cylinder; $Q_{пер.ц.}$ – costs for the flow of liquid from the cavity of the hydraulic cylinder under the influence of the pressure difference p1 and p2; $Q_{деф.1}$ –fluid consumption to compensate for the deformation of the cavity under pressure p1.

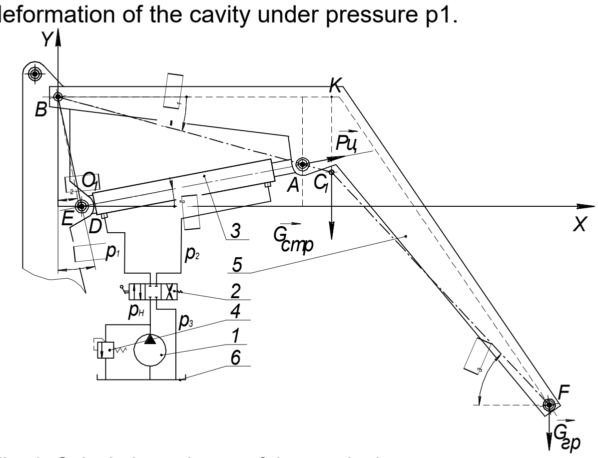

Fig. 1. Calculation scheme of the manipulator

The piston of the hydraulic cylinder, moving from left to right, pushes the liquid out of the rod cavity. The cost balance equation for this cavity has the following form:

2. $$Q_{ц2} + Q_{пер.ц.} = Q_{деф.2} + Q_{ут.ц.2} + Q_{б}$$

where $Q_{ц.2}$ – fluid flow in the rod cavity of the hydraulic cylinder, which ensures the movement of the piston at a given speed; $Q_{деф.2}$ – fluid consumption to compensate for the deformation of the liquid-filled cavity of the hydraulic system, which is under pressure p4; $Q_{ут.ц.2}$–flow of liquid from this cavity due to its leakage; $Q_{б}$–flow of liquid entering the hydraulic system tank.

The actual flow of liquid supplied from an unregulated pump is determined according to the expression [23]:

3. $$Q_{н} = q_{н} \cdot n_{н} \cdot \eta_{об}$$

where $q_{н}$–working volume of the pump; $n_{н}$–pump shaft rotation frequency; $\eta_{об}$–volume tric efficiency.

The consumption of liquid entering the drain hydraulic line is calculated according to the dependence

4. $$Q_{\acute{a}} = \mu \cdot f \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_2}$$

where μ–cost factor; f–cross-sectional area of the working window; ρ– густина робочої рідини.density of the working fluid.

Costs consumed by the hydraulic cylinder for the piston and rod cavities, respectively:

5. $$Q_{ц1} = S_1 \cdot \frac{dx}{dt}$$

6. $$Q_{ц2} = S_2 \cdot \frac{dx}{dt}$$

where S_1–the area of the piston cavity of the hydraulic cylinder; S_2–the area of the rod cavity of the hydraulic cylinder;

Losses due to leakage of liquid through the gaps in the connections of parts of hydraulic equipment and hydromechanisms are calculated as the flow of liquid through a flat gap under the accepted assumptions:

– the shape of the surfaces forming the outflow channel is perfect;

– the roughness of the surfaces is not taken into account;

– the gap is symmetrical.

In this case, the flow rate of the liquid flowing through the passage section of the gap will be determined by the following dependencies

7. $$Q_{ут.ц1} = \sigma_1 \cdot p_1$$

8. $$Q_{ут.ц2} = \sigma_2 \cdot p_2$$

where σ_1,· σ_2– the coefficients of liquid leaks from cavities under pressure p_1,· p_2, respectively.

Flows in hydraulic aggregates from the high-pressure chamber to the low-pressure chamber through the incomplete tightness of the chambers of the hydraulic aggregates are determined in the case of hydraulic cylinders according to the following:

9. $$Q_{пер.ц} = \sigma_ц \cdot (p_1 - p_2)\ ,$$

where $\sigma_ц$·–the coefficient of liquid flow between the cavities of the hydraulic cylinder, respectively, from the high-pressure chamber to the low-pressure chamber.

Costs that occur during the deformation of the volumes of the hydraulic drive cavities filled with liquid due to the change in pressure in these cavities are determined by the following dependencies:

10. $$Q_{деф.i} = K_i(p_i) \cdot W_i \cdot dp_i/dt, \quad де\ i = 1,2,$$

where $K_i(p_i)$ – compliance coefficients of the corresponding mains and cavities of this hydraulic system; W_1·–the volume of the cavity that connects the pumping station with the piston cavity of the hydraulic cylinder; W_2 –the volume of the cavity that connects the rod cavity of the hydraulic cylinder with the tank of the hydraulic system.

The mechanical part of this system is described by the equation of moments relative to point B:

11. $$\vec{M}_{ін} - \vec{M}_{Gстр} - \vec{M}_{Gгр} + \vec{M}_{Pц} - \vec{M}_{TP} = 0$$

where $\vec{M}_{ін}$ –moment of inertia of the mechanism relative to point B; $\vec{M}_G$ – moment from the weight of the load; $\vec{M}_{Pц}$ – the moment that occurs under the action of the force of the hydraulic cylinder; $\vec{M}_{TP}$ – moment from the force of friction.

The moment of inertia of the mechanism is determined by the dependence [24, 25]:

12. $$M_{ін} = I \cdot \frac{d^2\varphi_1}{dt^2}$$

The moment of the force of the weight of the arrow is defined as the product of the force of the weight of the arrow on the arm of the action by dependence:

13. $$M_{Gcmp} = m_{cmp} \cdot g \cdot BK$$

As can be seen from Figure 1, the BC arm can be calculated based on the dependence $BK = BC_1 \cdot \cos(-\varphi_1)$

14. $$M_{Gcmp} = BC_1 \cdot m_{cmp} \cdot g \cdot \cos(-\varphi_1)$$

The moment of the force of the weight of the load:

15. $$M_{Gгр} = m_{гр} \cdot g \cdot O_1N$$

The value of the O1N arm of the force action is determined by dependence:

16. $$O_1N = BC_1 \cdot \cos(-\varphi_1) + C_1F \cdot \cos(\varphi_3)$$

Taking into account dependence (16), dependence (15) has the form:

17. $$M_{Gгр} = m_{гр} \cdot g \cdot \left(BC_1 \cdot \cos(-\varphi_1) + C_1F \cdot \cos(\varphi_3)\right)$$

The moment of force developed by the hydraulic cylinder is determined by the dependence:

18. $$M_{Pц} = \left(S_1 \cdot p_1 - S_2 \cdot p_2\right) \cdot BE$$

where the arm BE of force action is determined by the following expression

19. $$BE = BD \cdot \cos\left(\varphi_2 - \varphi\right)$$

Taking into account dependence (19), expression (18) will have the form:

20. $$M_{Pц} = \left(S_1 \cdot p_1 - S_2 \cdot p_2\right) \cdot BD \cdot \cos\left(\varphi_2 - \varphi\right)$$

The moment of the friction force is calculated as the product of the coefficient of friction by the speed of rotation of the mechanism relative to point C:

21. $$M_{TP} = \beta_M \cdot \frac{d\varphi}{dt}$$

Angles φ_1 and φ_2 are interdependent. To determine the angle's dependence on:

22. $$AD = \sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}$$

The projection of the ABD contour on the Y axis has the following form:

23. $$AD \cdot \sin\left(\varphi_2\right) = O_1B - AB \cdot \sin\left(-\varphi_1\right)$$

From equation (23), taking into account equation (22), we calculate the value of the angle using the following expression:

24. $$\varphi_2 = arcsin\left(\frac{O_1B - AB \cdot sin(-\varphi_1)}{\sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}}\right)$$

Combining equations (1) – (24), we get the following system of equations:

25. $$\left|\begin{array}{l} q_н \cdot n_н \cdot \eta_{об} = S_1 \cdot \frac{dx}{dt} + \sigma_1 \cdot p_1 + \sigma_ц \cdot (p_1 - p_2) + K_1 \cdot W_1 \cdot \frac{dp_1}{dt} \\ S_2 \cdot \frac{dx}{dt} + \sigma_ц \cdot (p_1 - p_2) = K_2 \cdot W_2 \cdot \frac{dp_2}{dt} + \sigma_2 \cdot p_2 + \mu \cdot f \cdot \sqrt{\frac{2}{\rho}} \cdot \sqrt{p_2} \\ -m \cdot \frac{d^2x}{dt^2} + S_1 \cdot p_1 - S_2 \cdot p_2 - \beta_{TP} \cdot \frac{dx}{dt} - \\ -\left(m_{cmp} + m_{гр}\right) \cdot g \cdot \frac{O_1B - AB \cdot sin(-\varphi_1)}{\sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}} = 0 \end{array}\right.$$

$$\left|\begin{array}{l} I \cdot \frac{d^2\varphi}{dt^2} - m_{cmp} \cdot g \cdot BC_1 \cdot cos(-\varphi_1) - m_{гр} \cdot g \cdot BL \cdot cos(-\varphi_1) - \beta_M \cdot \frac{d\varphi}{dt} + \\ + (S_1 \cdot p_1 - S_2 \cdot p_2) \cdot AB \cdot sin\left(-\varphi_1 + arcsin\left(\frac{O_1B - AB \cdot sin(-\varphi_1)}{\sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}}\right.\right. \end{array}\right.$$

Taking into account the above assumptions, the system of equations (25) takes the following form:

$$
\left\{
\begin{aligned}
&q_i \cdot n_i \cdot \eta_{\text{íá}} = S_1 \cdot \frac{dx}{dt} + \sigma_1 \cdot p_1 + \sigma_{\text{ö}} \cdot p_1 + K_1 \cdot W_1 \cdot \frac{dp_1}{dt},\\
&-\left(m_{\text{ñòð}} + m_{\text{ãð}}\right) \cdot \frac{d^2x}{dt^2} + S_1 \cdot p_1 - \beta_{TP} \cdot \frac{dx}{dt} -\\
&-\left(m_{\text{ñòð}} + m_{\text{ãð}}\right) \cdot g \cdot \frac{O_1B - AB \cdot sin(-\varphi_1)}{\sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}} = 0,
\end{aligned}
\right. \tag{26}
$$

$$
\left|
\begin{aligned}
&I \cdot \frac{d^2\varphi}{dt^2} - m_{\text{ñòð}} \cdot g \cdot BC_1 \cdot cos(-\varphi_1) - m_{\text{ãð}} \cdot g \cdot BL \cdot cos(-\varphi_1) - \beta_M \cdot \frac{d\varphi}{dt} + (S_1 \cdot p_1 -\\
&- S_2 \cdot p_2) \cdot AB \cdot sin\left(-\varphi_1 + arcsin\left(\frac{O_1B - AB \cdot sin(-\varphi_1)}{\sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}}\right)\right) = 0.
\end{aligned}
\right.
$$

The MathCad package for mathematical calculations was used to solve this mathematical model. In the Cauchy form, the system of equations (26) has the following form:

$$
\left\{
\begin{aligned}
&\frac{dp_1}{dt} = \frac{1}{K_1 \cdot W_1} \cdot \left(q_i \cdot n_i \cdot \eta_{\text{íá}} - S_1 \cdot \frac{dx}{dt} - \sigma_1 \cdot p_1 - \sigma_{\text{ö}} \cdot p_1\right),\\
&\frac{d^2x}{dt^2} = \frac{1}{m_{\text{ñòð}} + m_{\text{ãð}}} \cdot \Bigg(S_1 \cdot p_1 - \beta_{TP} \cdot \frac{dx}{dt} -\\
&-\left(m_{\text{ñòð}} + m_{\text{ãð}}\right) \cdot g \cdot \frac{O_1B - AB \cdot sin(-\varphi_1)}{\sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}}\Bigg),\\
&\frac{d^2\varphi}{dt^2} = \left(AB \cdot sin\left(-\varphi_1 + arcsin\left(\frac{O_1B - AB \cdot sin(-\varphi_1)}{\sqrt{AB^2 - BD^2 - 2 \cdot AB \cdot BD \cdot cos\left(\frac{\pi}{2} + \varphi_1 - \varphi\right)}}\right)\right)\right) \times\\
&\times \frac{(S_1 \cdot p_1 - S_2 \cdot p_2)}{I} - \frac{m_{\text{ñòð}} \cdot g \cdot BC_1 \cdot cos(-\varphi_1)}{I} - \frac{m_{\text{ãð}} \cdot g \cdot BL \cdot cos(-\varphi_1)}{I} - \frac{\beta_M}{I} \cdot \frac{d\varphi}{dt}.
\end{aligned}
\right. \tag{27}
$$

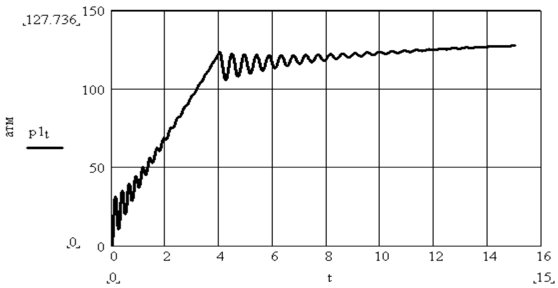

Fig. 2. Transient process of pressure change in the pressure line.

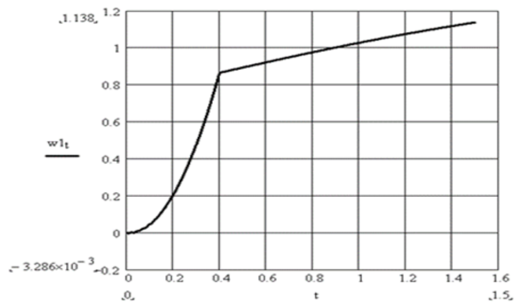

Fig. 3. Changing the angular velocity of the boom.

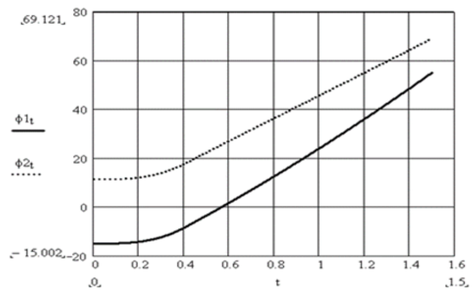

Fig. 4. Changing the angle of inclination of the boom (φ_1) and hydraulic cylinder (φ_2).

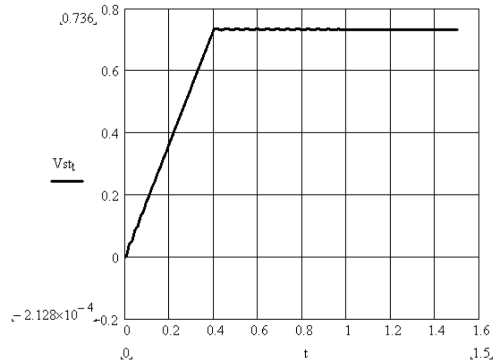

Fig. 5. Changing the speed of movement of the hydraulic cylinder rod.

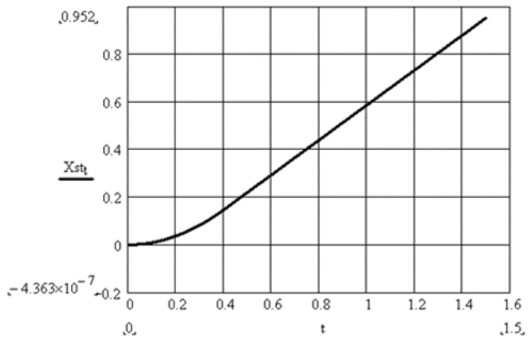

Fig. 6. Changing the movement of the hydraulic cylinder rod.

To find a solution to this system of equations, we use the Runge-Kutt-Feldberg method with an automatic change of the integration step [26, 27]. As a result of solving this system of equations, we obtain the following transient processes of changing system parameters (Fig. 2 - Fig. 6).

The load on the system was set according to the following algorithm:

The pump turned on.

After some time, when the system entered the nominal mode of operation, the hydraulic distributor was turned on, and the working fluid was supplied to the corresponding cavity of the hydraulic cylinder.

At a certain point in time, a payload was applied to the rod of the hydraulic cylinder (Fig. 3).

Conclusions

As evidenced by the transition process shown in Figure 3, the average value of the pressure in the pressure line is 12.8 MPa, which is acceptable for the hydraulic system of the motor vehicle. The fluctuation of the transition process is insignificant.

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