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THE INFLUENCE OF WAVE PROCESSES OF HYDRAULIC OILS ON THE OPERATION OF A HYDRAULIC DRIVE

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ABSTRACT

The paper reviews the state of research of wave processes in hydraulic systems of machines, and their impact on the quality and stability of hydraulic drives. As noted in the works of previous researchers, these phenomena occur in hydraulic systems and adversely affect the quality and stability of their work, significantly reducing reliability. The method of construction of mathematical models is offered. A mathematical model of the hydraulic system with two series-connected hydraulic motors is built, taking into account transients. The Runge-Kutta-Feldberg method with automatic change of the integration step was used to solve this model. The application of this method makes it possible to estimate the amplitude and frequency of the pressure wave in real time for each part of the pipeline. As a result of the analysis of the obtained transients it was concluded that at the length of the pressure line in a group hydraulic drive with 2 series-connected hydraulic motors up to 1.5 m, wave processes do not significantly affect the system and in the mathematical model they can be ignored. With the length of the pressure line from 1.5 m to 9 m, the wave processes in the cavity do not affect the stability of the system, although significantly impair the quality of its work. Hydraulic systems with a pressure line length of more than 9 m are not recommended for implementation, because the wave processes in the cavity lead to vibrations and noise in the hydraulic system and require additional measures to eliminate the impact of this phenomenon.

Introduction

Due to the development of technologies for growing crops, agricultural machinery manufacturers are faced with the issue of providing new technologies with appropriate machines and units with improved technical characteristics (Karaiev et al., 2021; Havrylenko et al., 2021a; Havrylenko et al., 2021b; Khasawneh et al., 2021; Mannheim and Simenfalvi, 2020).

One of the effective means of improving the technical performance of agricultural machinery is the hydrography of the drive of active working bodies by replacing the existing mechanical with a group hydraulic drive with a series connection of hydraulic motors (Khasawneh et al., 2021; Mannheim and Simenfalvi 2020).

One of the features of group hydraulic drives of executive bodies of agricultural machines is that the mains connecting hydraulic units can be of considerable Khasawneh length. This is due to the fact that the fluid is supplied to the hydraulic system from the pump, which is driven by the motor power take-off shaft, and the active working bodies (conveyors, diggers, fans) are on the periphery of the machine, i.e. at a considerable distance from the pump (Maxit et al., 2021; Panchenko et al., 2018a; Chang et al., 2006).

Therefore, during the operation of technological machines with hydraulic motors connected in series, in a long pressure line there may be fluctuations in pressure and speed of movement of the working fluid due to wave processes in the pipeline (Choi et al., 2012; Stryczek et al., 2014a; Gamez-Montero et al., 2012; Furustig et al., 2015; Stryczek et al., 2014b) .

In (Gunko, 1998; Panchenko et al., 2018b) the authors presented mathematical models of a group hydraulic drive with 2 series-connected hydraulic motors and investigated their stability without taking into account the wave processes in the pipeline. The publication (Zhang, 2019) defined the ranges of parameters (characteristic volumes of hydraulic motors, volumes of switching lines) at which the system operates stably. One of the conclusions made in this paper is the conclusion about the lengths of the switching lines, which are desirable when designing a group hydraulic drive. According to these data, the volume of the pressure line should not exceed $3000x10^{-6}$ m^3, which corresponds to the possible length of the line up to 10 m, and the volume of the line connecting the hydraulic motors should not exceed $600x10^{-6}$ m^3, respectively, the length of the switching line is up to 2 m.

With the recommended length of the pressure line, the occurrence of wave processes is possible, which significantly affect the operation of the group hydraulic drive. In this regard, the task was to develop a mathematical model of group hydraulic drive with 2 series-connected hydraulic motors taking into account wave processes in a long pressure cavity and to investigate the influence of wave processes on the system at different values of pressure cavity volume.

The question of studying the influence of wave processes on the stability and quality of hydraulic drive systems today is quite relevant. Thus, in Zhao et al. (2017) and Zhang et al. (1999) the information on possible wave loads in two-dimensional space and methods of their damping is given.

In turn, the work Zhang (2019) and Gao et al. (2021) is devoted to the emergence, research and development of ways to protect hydraulic systems of aircraft from wave processes that occur during the operation of these systems. Analysis of the factors which influencing the occurrence of this phenomenon in the hydraulic system of the aircraft is carried out. Vibrations in the hydraulic piping system of aircraft, due to excitation in several sources of fluid pressure fluctuations and severe vibration of the glider, can cause resonant phenomena, which in turn lead to the destruction of piping, shut-off valves and overloading the system as a whole. As noted in the publication, the control of wave processes in the hydraulic system is a complex but necessary task to ensure the operation of this type of machine. It is shown that the generally accepted technologies of control of wave processes are effective in typical constructions, such as aerospace constructions, shipbuilding constructions, sea constructions, motor constructions, etc. However, research in the field of wave processes, their control and mitigation have been conducted on a small scale and do not fully disclose the state of the issue. Given the current trends in the development of hydraulic systems and increasing requirements for them, this publication considers modern technologies for controlling wave processes in the system of long pipelines. An overview of general approaches to solving this problem, which correspond to different control technologies - passive and active. These control technologies are based on the principle of optimal technology of pipeline layout and fastening, methods of damping transients, active damping of wave processes. As a result of the conducted researches several offers for the decision of a question of control of wave processes in hydraulic systems are offered. The results presented in (Gao et al., 2021) are the result of physical modeling of processes with rather limited ranges of initial parameters, which does not allow to fully assess the impact of wave processes on the hydraulic system and select the optimal parameters.

In contrast to Gao et al. (2021) Zhang et al. (1999) is devoted to the study of wave processes that occur during the operation of hydraulic systems by the methods of automatic control theory. In this paper, a simplified mathematical model of this process is built. To study it, a discrete Fourier transform was performed and amplitude-phase frequency characteristics were constructed. The analysis of the system operation was performed in the frequency domain. The advantage of this method is the ability to determine the related characteristics of the hydraulic system at which the system will operate in the required mode. For comparison, the following Laplace transform of this model was performed. It is emphasized that when using both methods, the system has only one pair of equations for each mode of wave propagation. The proposed research methods used in the publication highlight the general features between the analysis in the frequency domain and the analysis by the method of amplitude-phase-frequency characteristics (AFCH) in the time domain. The method is confirmed by comparison with an alternative exact analytical solution, the results obtained by discrete Fourier transform based on AFCH analysis, and comparison with the measured data of the laboratory apparatus. But it should be noted that the proposed methods are quite approximate, as they allow the analysis of static models.

In Zhang (2019) the results of researches of influence of wave processes on axial vibration of hydraulic highways are resulted. The mathematical model of this process consists of four equations. Of which two equations describe the process of fluid motion, and the other two describe the deformation of the pipe wall. This mathematical model was solved taking into account the assumption that the distributed friction between the layers of the working fluid is absent. The finite element method was used to find the solutions of the mathematical model. The disadvantage of this method is that it takes a long time to find a solution. This paper Zhang (2019) proposes an improved method based on this exact solution that uses time line interpolation rather than a recursive algorithm to speed up the calculation. Since numerical diffusion and oscillations strongly affect accuracy, different interpolation methods are investigated and compared. The mean absolute error and the index of structural similarity were used to assess the accuracy of the proposed methods, and the latter was originally developed to assess image quality. The proposed hybrid interpolation scheme, which combines cubic spline and linear interpolation, and achieves the highest accuracy, and quadratic-linear interpolation is a good choice, given both accuracy and efficiency. The solution method developed in article Zhang (2019) significantly improves efficiency while maintaining an acceptable level of accuracy, especially suitable for long-term events. The proposed method is quite interesting, but to obtain comprehensive information about the research process it is necessary to perform a significant number of calculations with variable initial conditions, which in turn significantly increases the time spent on research.

In Zecchin et al. (2014), a complex model for predicting the dynamic response of the pipeline, taking into account the influence of mechanical transmissions of the drive, Coriolis forces and centrifugal forces on the example of the hydraulic system of the aircraft. It is noted that pressure fluctuations caused by the hydraulic pump can cause severe vibrations in the aero-hydraulic piping system, which poses a serious threat to the safety of the aircraft. Therefore, an effective method for predicting fluctuations in fluid pressure and dynamic response of the pipeline is proposed. To solve the equations of the hydraulic pipeline, a numerical code is presented. In this code, the hydraulic equations are solved by the characteristic method, and the dynamic equations of the pipeline are solved by the finite element method in combination with the Newmark algorithm. The numerical code is verified by comparing the dynamic response of a typical hydraulic line with an experiment. The obtained results indicate that the modern combined method of characteristics and the finite element method approach can predict the dynamic response of the hydraulic pipeline with sufficient accuracy, which can serve as an effective tool in the design and maintenance of the aerohydraulic pipeline.

Work Ivanov et al. (2019) is devoted to the issues of mathematical modeling of the hydraulic system of a front loader equipped with a mechanism for separating compressed feed from the total volume. The shown mathematical model was built on the basis of the equations of continuity of the working fluid and the equations of forces and moments acting on the working bodies of the actuators of the hydraulic system.

As noted Ivanov et al. (2019), the accuracy and adequacy of solving the problem by the finite element method depends on the number, nodes, shape of the element under consideration and the performance of computer technology. The solution of the problem obtained using this method does not allow us to trace the change in the parameter of interest to us for some time, therefore, the application of this method to the study of wave processes is limited. More informative are the solutions of mathematical models using the Runge – Kutta – Feldberg method with an automatic change in the calculation step. The graphical dependencies obtained with the help of this mathematical apparatus make it possible to study the change in the amplitude and frequency of the pressure change in the pipelines of the hydraulic system at the moment of starting, as well as at the moment of switching on the technological load with different ratios of the parameters of the hydraulic system. Analysis of the information received allows us to formulate recommendations for the design of systems of this type.

Materials and Methods

In the hydraulic drive of self-propelled agricultural machines, the most widespread are petroleum-based mineral oils, which contain antioxidant, antifoam, anti-corrosion and anti-wear additives. Mineral oils are low cost, readily available in large quantities, good lubricity and long service life at high pressures. In industrial hydraulic systems, transformer, spindle AU, industrial, turbine, cylinder, VMGZ, MG-30, GM-50I and other oils are widely used. Some oils are designed for a narrow temperature range. For a wide range of temperatures, ATM, MVP, GM-50I oils are used, but their cost is high. Mineral oils have several disadvantages that limit their use. At elevated temperatures, the lubricity deteriorates, and at low temperatures, the viscosity rises sharply. In addition, at elevated temperatures, resinous sediments fall out of mineral oils, which clog the filter cells and settle in the internal passages of hydraulic equipment. In hydraulic drives and hydraulic systems operating at a temperature of about 450 K, synthetic silicone and silicon organic fluids are used. They have high chemical resistance, mix well with mineral oils, and have no corrosive activity.

For research in this hydraulic drive, HM-46 hydraulic oil was used. This hydraulic oil has improved anti-wear performance, good thermal stability, good hydrolytic stability. A feature of this oil is antiwear, anti-corrosion properties, with stable antifoam characteristics.

The calculated scheme of the group hydraulic drive, which takes into account the wave processes, is presented in Fig. 1.

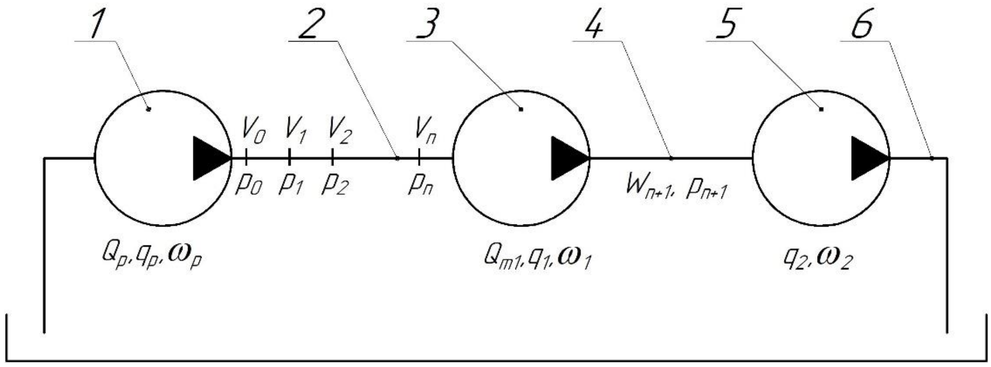


Figure 1. Scheme of a group hydraulic drive with 2 series-connected hydraulic motors. V_0 - the initial velocity of the pipeline fluid, P_0 - initial speed of liquid supply from the power supply

The liquid supplied from the pump of constant capacity 1 Q_p enters the long pressure pipe 2, which is divided into n sections, the output of which is connected to the first hydraulic motor 3. The main line 4 connects it in series with the hydraulic motor 5, the output of which is connected to the drain highway 6.

To simplify the calculations, we assume that the hydraulic system operates at a steady temperature.

In this case, the mathematical model, in contrast to the model presented in Panchenko et al. (2018b), will include a model of a pipeline with distributed parameters, described by two first-order differential equations in partial derivatives and boundary conditions describing fluid flow in units connected to the pipeline.

$$\frac{\partial p_n}{\partial x} = -\rho \cdot \left(\frac{\partial v_n}{\partial t}\right) - \xi_n \cdot v_n; \tag{1}$$

$$\frac{\partial v_n}{\partial x} = -\frac{1}{E_n} \cdot \frac{\partial p_n}{\partial t}; \tag{2}$$

$$v_1 = v_p - \frac{V_p}{f_1 \cdot E_p} \cdot \frac{dp_p}{dt}; \tag{3}$$

$$Q_{m1} = v_n \cdot f_2, \tag{4}$$

where:

$\frac{\partial p_n}{\partial x}$ – change of pressure on the coordinate of section of the pipeline along its axis;
$\frac{\partial v_n}{\partial t}$ – change in the average fluid velocity over time;
v_1 – average velocity of the liquid in the inlet section of the pipeline;
v_p – average the speed of liquid supply from the power supply (pump);
V_p – volume of fluid in the cavities of the power supply;
f_1 – cross-sectional area of the pipeline at the pump outlet;
f_2 – the cross-sectional area of the pipeline at the inlet of the first hydraulic motor;
E_n – is the average modulus of volume of the power supply;
p_p – fluid pressure at the outlet of the power supply;
p_n – fluid pressure at the outlet of the n-th section of the pipeline;
v_n – fluid flow rate at the outlet of the n-th section of the pipeline;
ρ – density of the working fluid;
$\xi_n \zeta_n$ – specific hydraulic resistance of the pipeline;
E_p – average modulus of volume elasticity of the liquid in the pipeline;
Q_{m1} – fluid flow at the inlet of the first hydraulic motor.

For the numerical solution of such a system of equations, a difference method of transforming equations into partial derivatives into a system of ordinary differential equations is used.

Then the mathematical model of the group hydraulic drive with two series-connected hydraulic motors taking into account wave processes in the pressure line has the form:

$$\frac{dv_1}{dt} = \frac{2}{h \cdot \rho} \cdot (p_H - p_1) - \frac{\xi_1 \cdot v_1}{\rho}; \tag{5}$$

$$\frac{dv_2}{dt} = \frac{1}{h \cdot \rho} \cdot (p_1 - p_2) - \frac{\xi_2 \cdot v_2}{\rho}; \tag{6}$$

$$\frac{dv_n}{dt} = \frac{2}{h \cdot \rho} \cdot (p_{n-1} - p_n) - \frac{\xi_n \cdot v_n}{\rho}; \tag{7}$$

$$\frac{dp_p}{dt} = \left(v_p - v_1\right) \cdot \frac{f_1 \cdot E_p}{V_p}; \tag{8}$$

$$\frac{dp_1}{dt} = \frac{E_1}{h} \cdot (v_1 - v_2); \tag{9}$$

$$\frac{dp_2}{dt} = \frac{E_2}{h} \cdot (v_2 - v_3); \tag{10}$$

$$\frac{dp_{n-1}}{dt} = \frac{E_{n-1}}{h} \cdot (v_{n-1} - v_n); \tag{11}$$

$$\frac{dp_n}{dt} = \frac{v_n \cdot f}{2 \cdot \pi \cdot q_1 \cdot K_1} - \frac{q_1 \cdot \omega_1}{2 \cdot \pi \cdot q_1 \cdot K_1} - \frac{(\sigma_1 + \sigma_{1,2}) \cdot p_{n-1}}{2 \cdot \pi \cdot q_1 \cdot K_1} + \frac{\sigma_{1,2} \cdot p_n}{2 \cdot \pi \cdot q_1 \cdot K_1}; \tag{12}$$

$$\frac{dp_{n+1}}{dt} = \frac{q_1 \cdot \omega_1}{W_2 \cdot K_2} + \frac{\sigma_{1,2} \cdot p_n}{W_2 \cdot K_2} - \frac{q_2 \cdot \omega_2}{W_2 \cdot K_2} - \frac{(\sigma_{1,2} + \sigma_2 + \sigma_{2,3}) \cdot p_{n+1}}{W_2 \cdot K_2}; \tag{13}$$

$$\frac{d\omega_1}{dt} = \frac{(q_1 - \varepsilon_1) \cdot p_n}{I_1} - \frac{(q_1 + \varepsilon_1) \cdot p_{n+1}}{I_1} - \frac{M_{T1}}{I_1} - \frac{b_1 \cdot \omega_1}{I_1}; \tag{14}$$

$$\frac{d\omega_2}{dt} = \frac{(q_2 - \varepsilon_2) \cdot p_{n+1}}{I_2} - \frac{M_{T2}}{I_2} - \frac{b_2 \cdot \omega_2}{I_2}, \tag{15}$$

where:

σ_1, σ_2 – coefficients of leakage of working fluid from the pressure lines of hydraulic motors;

$\sigma_{1,2}$, $\sigma_{2.3}$ – coefficients of fluid flow between the cavities of the first and second hydraulic motor;

K_1 – the coefficient of pliability of the cavities filled with liquid between the first hydraulic motor and the pump is given;

K_2 – the coefficient of pliability of the liquid-filled cavity connecting the output of the first hydraulic motor with the input of the second hydraulic motor

W_2 – the volume of the second pipeline;

I_1, I_2 – the moment of inertia of the moving parts of the working bodies, brought to the shafts of hydraulic motors;

b_1, b_2 – coefficients of active resistance that characterize the loss of viscous friction;

ε_1, ε_2 – coefficients of mechanical losses in hydraulic motors characterizing losses on dry friction;

h – sampling step, calculated by the formula $h = L \cdot n^{-1}$;

L – length of the pressure pipeline;

n – the number of sections into which the pressure pipeline is divided along its length.

When performing transformations, one of the important issues is the rational choice of the sampling step by the coordinate of the pipeline length, i.e. the choice of the optimal number of sections into which the pipeline should be divided in order to obtain sufficient accuracy of the transition process. According to [20], the optimal number of sites is determined by the condition:

$$n \geq \frac{W_{pipe}}{W_{pump}} \tag{16}$$

where:

W_{pipe} – pipeline volume;

W_{pump} – pump displacement.

The mathematical model was studied by conducting a numerical experiment. To do this, a Delphi program for PC was compiled, in which by solving a system of $2n + 4$ first-order differential equations by the Runge-Kutta-Feldberg method, transient functions of changing the angular velocities of hydraulic motor shafts and working fluid pressure at each hydraulic motor inlet were obtained.

The program works as follows:

1. The parameters of the system are introduced (the volume of the cavity cavity - is entered in steps of $50 \cdot 10^{-6}$ m^3; the characteristic volumes of hydraulic motors are entered after the entire range of cavity volumes will be investigated), the accuracy of calculating the solution of the system of differential equations is $1 \cdot 10^{-5}$ .1e^{-5}.
2. The mathematical model (5) - (15) is solved by the Runge-Kutt-Feldberg method with a variable differentiation step.
3. As a result, we obtain transient processes of pressure and velocity changes for each section of the pipeline.
4. Based on the results of the calculation transient functions are built, which can be written to a file if desired.
5. End of the program.

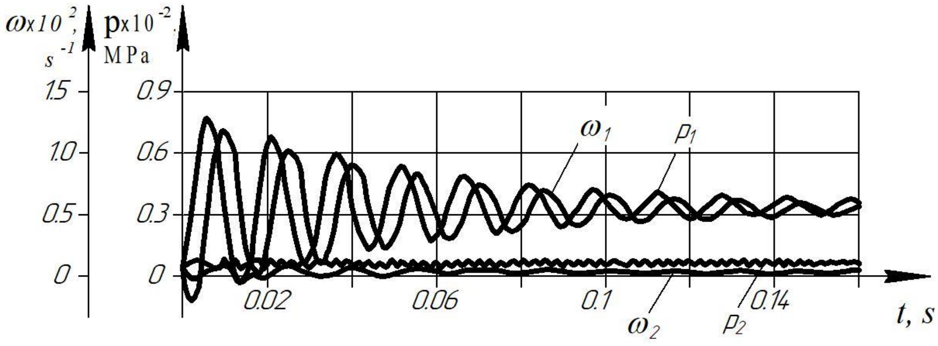
As a result of the program, the calculated curves of pressure and fluid velocity changes were obtained, as well as the numerical values of these parameters at each of the conditional sections of the first pipeline.

The system was investigated at the following parameter values:

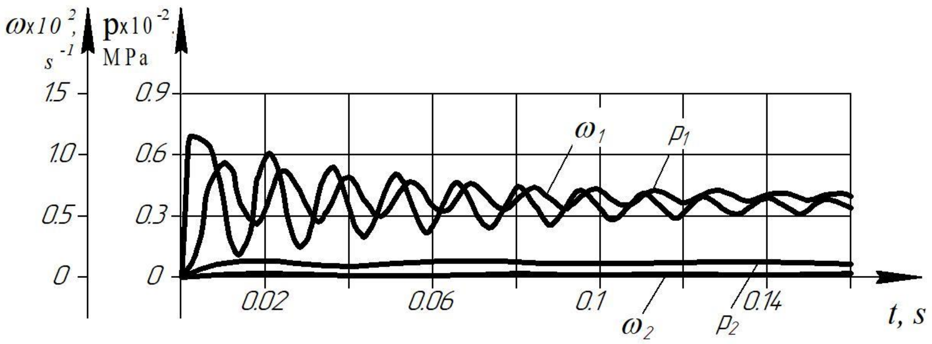
$$f_1 = 2.01 \cdot 10^{-4} m^2;$$
$$K_1 = 6.118 \cdot 10^{-8} Pa^{-1};$$
$$b_1 = b_2 = 0.06 \cdot \frac{kg \cdot m^2}{sec}$$
$$\Delta p = 8 \cdot 10^6 Pa;$$
$$E_n = 1.5 \cdot 10^9 Pa;$$
$$\rho = 912 \frac{kg}{m^3};$$
$$q_{m1} = 5 \cdot 10^{-6} \frac{m^3}{rad}$$
$$q_{m2} = 3 \cdot 10^{-5} \frac{m^3}{rad};$$
$$W_2 = 1 \cdot 10^{-4} m^3.$$

Results and Discussion

Figure 2 a, b shows the transient functions of changing the pressure of the working fluid HM-46 used in this type of drive, at the inlet of each of the hydraulic motors and the angular speeds of rotation of the shafts, obtained for a group hydraulic drive with a pressure cavity with a volume of W_1=$2 \cdot 10^{-4}$ m^3 without taking into account wave processes (Fig. 2a) and taking into account wave processes (Fig. 2b). As can be seen in the figures, the wave processes in the pressure cavity for the cavity length l_{w1}<1,5 m do not have a significant effect on the operation of the group hydraulic drive, the frequency of the first harmonic of the polyharmonic process coincides with the frequency of the monoharmonic process.



a)

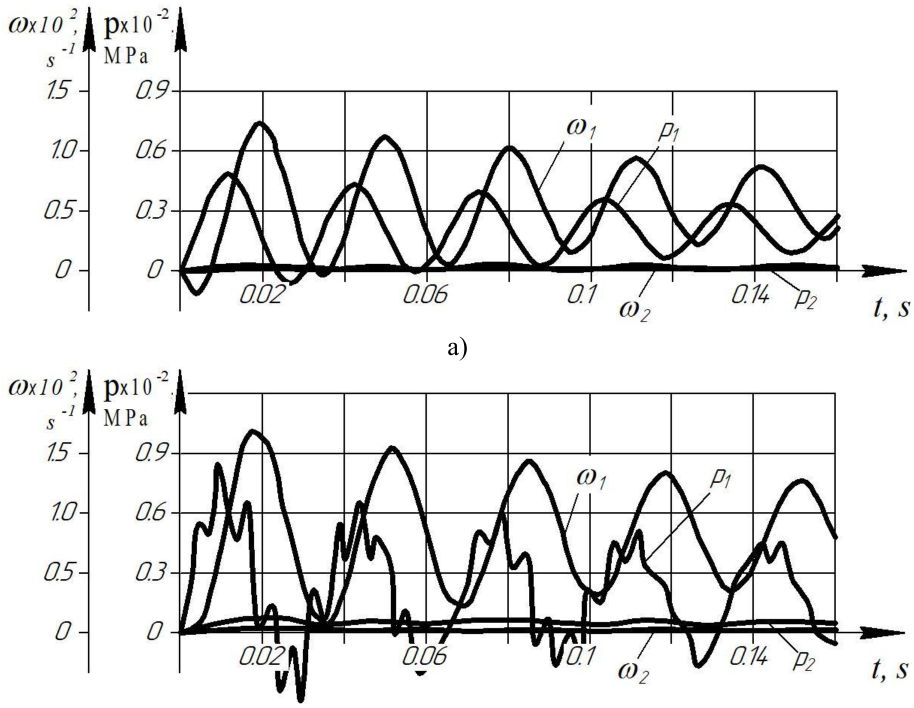


b)

Figure 2. Transient functions of group hydraulic drive operation with two series-connected hydraulic motors with parameters: L = 1.5 m, ($W_1 = 2 \cdot 10^{-4}$ m^3), in which HM-46 hydraulic oil was used: a) without taking into account wave processes; b) taking into account wave processes.

Therefore, in further studies, it is possible to use the mathematical model represented by equations (5) - (15). However, a further increase in the volume of the pressure main to $8 \cdot 10^{-4}$ m^3 (Fig. 3b) leads to the appearance of an explicit high-frequency component in the transient functions of pressure change at the inlet of the first hydraulic motor, which is the result of the influence of wave processes in the pipeline, while the amplitude of oscillations pressure and angular velocity of rotation of its shaft increases, although the period of oscillations remains unchanged and the system operates stably.

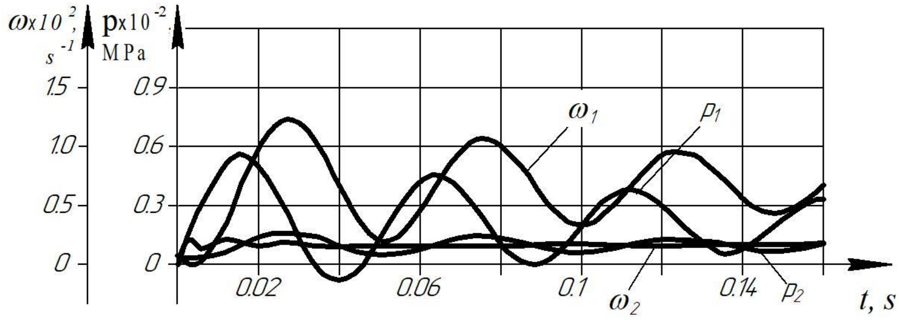
An increase in the length of the pressure line to l_{w1} = 9.7 m ($W_1 = 19.5 \cdot 10^{-4}$ m^3) leads to a significant difference in the results of modeling a group hydraulic drive according to a mathematical model without taking into account wave processes (Fig. 4a) and according to a mathematical model with taking into account wave processes (Fig. 4b). As can be seen from the graphs obtained, the frequency of fluid pressure fluctuations at the inlet of the first hydraulic motor and the angular velocity of rotation of the shaft increases significantly, which can cause vibration and noise. Using the already known means to reduce the influence of wave processes in the pressure lines of hydraulic systems with hydraulic motors (increasing the characteristic volume of the hydraulic motor and introducing the accumulator into the cavity), the expediency of their use was proved for a group hydraulic drive with a series connection of hydraulic motors.



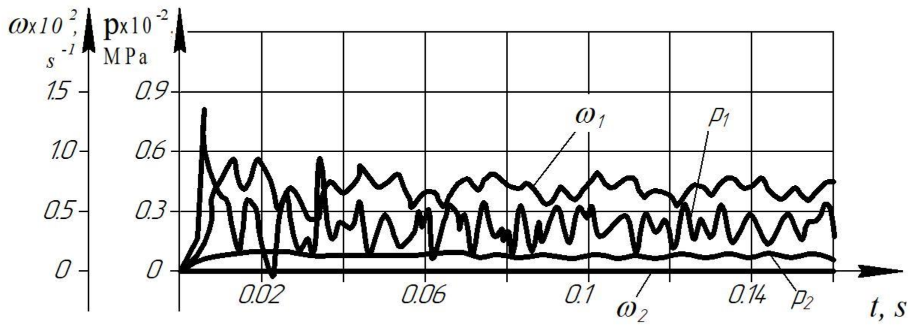
b)

Figure 3. Transient functions of group hydraulic drive operation with two series-connected hydraulic motors with parameters: L = 4.5 m, (W_1 = 8·10^{-4} m^3), in which HM-46 hydraulic oil was used: a) without taking into account wave processes; b) taking into account wave processes.

An increase in the length of the pressure chamber leads to an increase in the frequency of pressure fluctuations and rotational speed, which can cause vibration, noise and is unacceptable during the operation of the drive. This testifies to the inexpediency of using pipelines in pressure voids with a length exceeding 9 m.



a)



b)

Figure 4. Transient functions of group hydraulic drive operation with two series-connected hydraulic motors with parameters: L = 9 m, ($W_1 = 19.5 \cdot 10^{-4}$ m^3), in which HM-46 hydraulic oil was used: a) without taking into account wave processes; b) taking into account wave processes.

Conclusions

As can be seen from the studies carried out, the proposed mathematical model makes it possible to take into account wave processes that arise in long lines of hydraulic drives. This mathematical model is nonlinear, and an increase in the accuracy of accounting for this phenomenon leads to a rapid increase in the order of the system of differential equations. This creates certain obstacles to its widespread use. On the other hand, this phenomenon has a significant impact on the quality and durability of hydraulic drives, therefore, its consideration and assessment is mandatory in the study and design of hydraulic drives. The use of this mathematical model made it possible to determine the permissible limits of the geometric parameters of the drive, to estimate the amplitude and frequency of pressure fluctuations in pipelines in different sections and to provide for a number of constructive measures to ensure the necessary operating modes of the hydraulic drive.

The conducted researches allow to draw the following conclusions:

1. When the length of the pressure line in a group hydraulic drive with 2 series-connected hydraulic motors up to 1.5 m ($W_1 = 2\cdot10^{-4}$ m^3), wave processes do not significantly affect the system and in the mathematical model they can be ignored.
2. When the length of the pressure line is more than 1.5 m, the use of the mathematical model given in Gunko (1998) is unacceptable, as it does not reflect the real process in the system.
3. With the length of the pressure line from 1.5 m to 9 m, the wave processes in the cavity do not affect the stability of the system, although they significantly worsen the quality of its work.
4. Hydraulic systems with a pressure line length of more than 9 m are not recommended for implementation, because the wave processes in the cavity lead to vibrations and noise in the hydraulic system.
5. The research procedure carried out, turned out to be correct, valid for the design of hydraulic drive systems. It can be successfully used in the industrial design methodology of this type of installation. It allows for the verification and proper selection of consumables in the form of oils, hydraulic fluids, appropriate for the configuration of the installation.

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WPŁYW PROCESÓW FALOWYCH WYSTĘPUJĄCYCH W OLEJACH HYDRAULICZNYCH NA DZIAŁANIE NAPĘDU HYDRAULICZNEGO

Streszczenie. Niniejszy artykuł stanowi przegląd badań nad procesami falowymi w układach hydraulicznych maszyn, ich wpływ na jakość i stabilność układów hydraulicznych. Jak zauważono w pracach poprzednich badaczy, te zjawiska występują w układach hydraulicznych i negatywnie wpływają na jakość i stabilność ich pracy znacząco zmniejszając ich niezawodność. Zaproponowano metodę budowy modeli matematycznych. Skonstruowano model matematyczny układu hydraulicznego z dwoma szeregowo połączonymi silnikami hydraulicznymi z uwzględnieniem napięć przejściowych. Do rozwiązania tego modelu zastosowano metodę Runge-Kutta-Feldberga z automatyczną zmianą kroku całkowania. Zastosowanie tej metody pozwala na oszacowanie amplitudy i częstotliwości fali ciśnienia w czasie rzeczywistym dla każdego odcinka przewodu. W wyniku przeprowadzonej analizy stwierdzono, że przy długości przewodów ciśnieniowych do 1,5 m w grupowym napędzie hydraulicznym z dwoma szeregowo połączonymi silnikami hydraulicznymi, procesy falowe nie wpływają znacząco na układ, a w modelu matematycznym można je ignorować. Przy długości przewodu ciśnieniowego w zakresie 1,5 m do 9 m, procesy falowe we wnętrzu nie wpływają na stabilność układu, ale znacząco wpływają na jakość jego pracy. Układy hydrauliczne o długości przewodów ciśnieniowych większej niż 9 m nie są zalecane, ponieważ procesy falowe we wnętrzu prowadzą do wibracji i hałasu w systemie hydraulicznym i wymagają dodatkowych działań w celu wyeliminowania wpływu tego zjawiska.

Słowa kluczowe: napędy hydrauliczne, modelowanie matematyczne, metoda Runge-Kutta-Feldberga, procesy falowe