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Розглянуто гідравлічний привід з двома гідравлічними циліндрами, синхронізація швидкості руху штоків яких здійснюється дільником потоку робочої рідини. На основі розробленої математичної моделі проведено розрахунок роботи синхронізованих гідроциліндрів в неусталеному режимі при раптовій зміні навантаження на одному з гідроциліндрів. Визначено швидкості руху штоків гідроциліндрів і тиск в міждросельних камерах дільника потоку. Встановлено, що при перехідних режимах роботи приводу, обумовлених раптовою зміною навантаження гідроциліндрів, виникають коливання тиску в міждросельних камерах дільника потоку і, в результаті цього, похибка синхронізації швидкості руху штоків гідроциліндрів на початковому етапі. Відносний перепад тисків в міждросельних камерах досягає 1, а відносна різниця швидкостей руху – до 0,43. Для підвищення точності синхронізації руху гідравлічних двигунів запропонований дільник потоку, в якому додано додатковий зворотний зв'язок по перепаду тиску в міждросельних камерах дільника. Додатковий зворотний зв'язок реалізований за рахунок застосування двохщілинного дроселюючого розподільника золотникового типу. Виходячи з умов забезпечення мінімальної похибки синхронізації, визначена необхідна залежність зміни площі робочої щілини регульованих дроселів та наведено рекомендації щодо профілізації робочих щілин золотникового дроселя.

Встановлено розрахунковим шляхом і підтверджено в експерименті, що застосування регульованих дроселів зменшує похибку синхронізації швидкості руху штоків гідроциліндрів до 0,27, а перепад тисків в міждросельних камерах дільника потоку – до 0,53.

В перехідному процесі для швидкості і тиску виникли гармоніки вищого порядку, зумовлені рухом золотника двохщілинного розподільника.

Наявність гармоніки вищого порядку в коливаннях тиску і швидкості несуттєво впливає на роботу гідравлічних двигунів, оскільки амплітуда коливань незначна

Зменшення похибки синхронізації швидкості зумовлено одночасною зміною площі дроселя, який стабілізує перепад тиску та площі регульованого дроселя

Ключові слова: дільник потоку, гідравлічний двигун, золотник, дроселюючий розподільник, синхронізація, перехідний процес

ANALYZING AN ERROR IN THE SYNCHRONIZATION OF HYDRAULIC MOTOR SPEED UNDER TRANSIENT OPERATING CONDITIONS

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1. Introduction

In the modern production equipment being a complicated automated complex with a large number of operating devices, a necessity of synchronous operating device movement often arises. Proportionality of displacements, speeds and accelerations of mechanisms is the prerequisite of synchronization. Conditions for synchronization of two operating devices are as follows:

$$\begin{cases} y_1 = ky_1, \\ \dfrac{\mathrm{d}y_1}{\mathrm{d}t} = k\dfrac{\mathrm{d}y_1}{\mathrm{d}t}, \\ \dfrac{\mathrm{d}^2 y_2}{\mathrm{d}t_2} = k\dfrac{\mathrm{d}^2 y_1}{\mathrm{d}t^2}, \end{cases} \tag{1}$$

where y_1, y_2 are displacements (linear or angular) of operating devices whose movements must be synchronized; k is proportionality factor.

In the mechanisms for which hydraulic drives are used, synchronization of the operating device movement consists in ensuring synchronization of hydraulic motors. In actual mechanisms, drive usually realizes one of the equations (1). This may be proportionality of displacements y_i, speeds $\mathrm{d}y_i/\mathrm{d}t$ or accelerations $\mathrm{d}^2 y_i/\mathrm{d}t^2$ of the outlet link of the hydraulic motor.

There are many parameters affecting the error of synchronization of hydraulic motors [1]:

– magnitude and nature of loads;
– speed and acceleration of hydraulic motors;
– time (value) of displacement;
– rigidity of the drive components and the mechanism as a whole;
– deviation from nominal values of the size of operating devices and motors caused by manufacture errors and changes occurred during operation;
– leaks of working fluid in gaps of hydraulic motors and apparatus.

The task of synchronization consists in ensuring tolerances in the coordinated movements of the operating devices.

Additional problems associated with non-stationary operating conditions arise in operation of hydraulic drives with synchronized motors. Such operating conditions are determined by:

– switching on/off hydraulic motors of the group drive results in a pressure change at the flow distributor inlet since the overflow valve responds with a delay to the change of flow rate;

– a sudden change of load on one of the hydraulic motors.

Considering importance of creating high-precision production equipment, studies aimed at reducing the error of synchronization of hydraulic motor speeds in non-stationary operating conditions are relevant.

2. Literature review and problem statement

A great deal of scientific and design studies are devoted to solving the problem of synchronization of movement of machine operating devices. The methods and design solutions that ensure synchronous movement of mechanisms depend on the mechanism operating conditions and the tasks set before it.

For the mechanisms driven by hydraulic motors, proportionality of speed of the operating devices means proportionality of the working fluid flow rate in hydraulic motors:

$$Q_1 = kQ_2, \tag{2}$$

where Q_1, Q_2 are the working fluid flow rates in hydraulic motors.

Methods of synchronization of movement of hydraulic motors and the potential field of their application were analyzed in detail in [2–26].

For example, a vibration mechanism with four unbalanced masses driven by four synchronized hydraulic motors was considered in [2]. Rotational speed of shafts of both motors was synchronized by means of four identical pumps individually feeding each motor. The authors believed that such scheme will provide synchronous rotation of the shafts of both motors and considered just dynamics of the vibration mechanism itself without taking into account the errors of speed synchronization of hydraulic motor shafts and the effect of load on synchronization accuracy.

Criteria of synchronization of two hydraulic motors in an eccentric rotary vibrating machine were considered in [3]. The authors believed that the working fluid flow rate in the hydraulic motors is the same and took into account only the effect of fluid leaks in the gaps between pistons and apertures in the cylinder block of the axial piston hydraulic motor. The issue of ensuring equal flow rates of the fluid supplied to the hydraulic motors has remained unresolved.

Simulation of synchronization of hydraulic cylinders in the main drive of the heavy forging system was analyzed in [4]. The authors considered influence of loading asymmetry on operation of the control system in steady-state operating conditions.

Time of stabilization of the system of synchronous action of two high-speed hydraulic motors during operation of the hydraulic drive was studied in [6]. However, the issues of synchronization accuracy in transient operating conditions remained unresolved.

An overview of synchronous hydraulic drives for lifting heavy building structures was performed in [7] and criteria for their selection based on hoisting capacity and displacement were proposed without taking into account errors in the movement speed synchronization.

Procedures of designing hydraulic drives to provide greater power efficiency were presented in [5, 9–11, 13, 14, 17].

The results of study of operation of a drive with two hydraulic motors working with a total load with pressure synchronization in the motor cavities were presented in [12]. Mutual effect of hydraulic motors on consistency of displacement of output links of the motors was considered. At the same time, the results of analysis of the effect of load asymmetry on operation of motors in transient operating conditions were not presented.

A system of power regeneration in parallel operation of hydraulic excavator motors was considered in [16].

Results of theoretical and experimental studies of movement synchronization of hydraulic motors in hydraulic drives using a throttling flow divider were given in [18]. The authors considered effect of leakage of the working fluid on accuracy of movement speed synchronization of hydraulic motors.

Recommendations on controlling synchronization of a crane with two motors working in steady-state operating conditions were provided in [19].

Control of motion of two hydraulic cylinders used for variation of a heavy mold of a continuous casting machine by means of a fuzzy logic controller was proposed in [21]. A drive was presented in which separate control loops combined by the controller into one system were used to control each hydraulic cylinder.

Transient operating conditions of hydraulic motor operation were mentioned only in [8, 15, 20–22] but detailed analysis of such operating conditions was not made.

For example, results of simulation of pressure in cavities of synchronized hydraulic cylinders of a brake fed from a pump with controlled flow rate were presented in [20].

However, the issue of the effect of load asymmetry on accuracy of synchronization of movement of hydraulic cylinders remained unresolved. Influence of parameters of a control system on operation of synchronized hydraulic cylinders with a harmonic input signal was analyzed in [22].

The considered published sources usually pay attention to operation of hydraulic motors in steady-state operating conditions when loads on their output links and movement speed are constant. Dependence of error in movement speed synchronization on loading asymmetry $\Delta v = f(\Delta F)$ was established and methods to minimize this error were proposed [8, 14–19].

The issues of analyzing accuracy of synchronization of movement of hydraulic cylinders in transient operating conditions remained unresolved. The effect of a sudden change of hydraulic motor loading on accuracy of synchronization of speed of hydraulic motors has not been analyzed. Influence of parameters of the hydraulic flow divider on the error of synchronization of movement in transient operating conditions and the time of motor speed stabilization at a sudden load change were not considered. Therefore, there is reason to consider that the lack of analysis of accuracy of synchronization of hydraulic motor speed in transient operating conditions necessitates the need for studies in this direction. The next step is searching for the ways to reduction of the synchronization error and analysis of effectiveness of the results obtained.

3. The aim and objectives of the study

The study objective was to reduce the error of synchronization of speed of hydraulic motors under transient conditions of hydraulic unit operation.

To achieve this objective, the following tasks were set:

– determine error of synchronization of speed of movement of piston rods of hydraulic cylinders in transient operation conditions by means of a flow divider;

– substantiate the possibility of reducing the error of synchronization when using a double-slotted throttling distributor in the flow divider and determine parameters of the controlled throttle;

– confirm experimentally reduction of the error of synchronization of hydraulic motor speed in transient operating conditions by applying a flow divider with additional feedback.

4. The materials and methods used in studying efficiency of the flow divider with additional feedback

4. 1. Determining the error of synchronization of speed of movement of the hydraulic cylinder piston rods in transient operating conditions

Design diagram of the hydraulic unit used in the study of movement of synchronized hydraulic motors with a throttling flow divider is shown in Fig. 1.

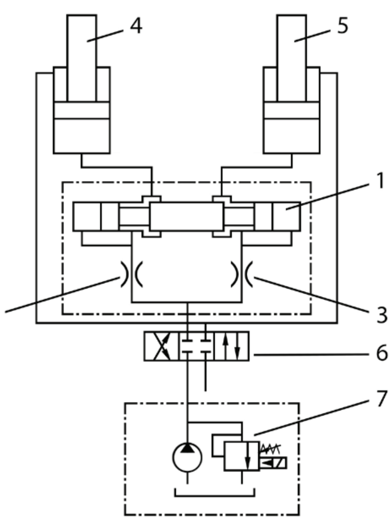

Fig. 1. Design diagram of a hydraulic drive for synchronous displacements: spool valve (1); non-controlled throttles (2, 3); hydraulic cylinders (4, 5); distributor (6); pumping station (7)

The drive consists of two hydraulic cylinders 4 and 5 whose movement should be synchronized, a flow divider of a throttle type, a distributor 6 and a pumping station 7. Fluid from the pumping station 7 enters the flow divider through the distributor 6. The liquid is divided in the divider into two parts which then enter chambers of the hydraulic motors 4, 5 through noncontrolled throttles 2 and 3 and the working slots of a spool-valve distributor 1. Independent of the load acting on the cylinder rods, the same pressure differential is maintained on the non-controlled throttles. Due to this point, flow rates through them will be proportional to the area of the throttles. However, this statement is valid only for steady operation of the hydraulic drive when loads do not change. In transient operating conditions, the spool valve does not have time enough to provide the same pressure differentials on the non-controlled throttles because of its sluggishness, so a deviation in speed of the cylinder rods occurs.

A mathematical model was developed to analyze movement unevenness. It describes the processes occurring in the elements of the hydraulic unit in transient operating conditions.

Equations of dynamics of movement of hydraulic motors were derived on the assumptions usually applied in analytical studies of dynamic characteristics of hydraulic systems [15].

Operation of the hydraulic unit is described by a system of equations including:

– an equation of movement of the hydraulic cylinder rods and regulating elements of hydraulic devices;

– an equation of fluid flow rates through regulating elements of hydraulic devices;

– an equation of balance of the working fluid flow rates in cavities of hydraulic devices taking into account displacement of the regulating elements and compressibility of the working fluid;

– impact of contact and viscous friction on operation of hydraulic motors and regulating elements.

$$m_i \frac{\mathrm{d}^2 y_i}{\mathrm{d}t^2} = \sum F_i, \tag{3}$$

where m_i is reduced to the rod mass of movable parts of the piston and the mechanism; y_i is the piston displacement; ΣF_i is the sum of forces applied to the rod.

$$\sum F_i = F_{pi} - F_{mi} - F_{fi}, \tag{4}$$

where F_{pi} is the force depending on pressure of the working fluid on the piston; F_{mi} is the reduced to the rod force acting on the mechanism operating device; F_{fi} is the force arising from friction.

$$F_{pi} = s_1 p_{1i} - s_2 p_2, \tag{5}$$

where s_1, s_2 are the areas of the piston and the rod cavities in the hydraulic cylinder; p_{1i}, p_2 are pressures in the hydraulic cylinder cavities.

$$\frac{\mathrm{d}p_{1i}}{\mathrm{d}t} = \frac{1}{\beta_{1i}}\left(Q_{2i} - s_1 \frac{\mathrm{d}y_i}{\mathrm{d}t}\right), \tag{6}$$

$$\frac{\mathrm{d}p_2}{\mathrm{d}t} = \frac{1}{\beta_2}\left[s_2\left(\frac{\mathrm{d}y_1}{\mathrm{d}t} + \frac{\mathrm{d}y_2}{\mathrm{d}t}\right) - Q_{2v}\right], \tag{7}$$

where β_{1i}, β_2 are reduced factors of volumetric deformation of the working fluid and cavities; Q_{2i} is fluid flow rate through the working slots of the flow divider; Q_{2v} is fluid flow rate through the working slot of the distributor.

$$\beta_{1i} = \frac{V_{l.i}}{E_l} + \frac{V_{e.i}}{E_{e.i}} + \frac{V_{g.i}}{np_i}, \tag{8}$$

where $V_{l.i}$ is volume of the working fluid; $V_{e.i}$ is volume of a cavity with elastic walls; $V_{g.i}$ is volume of undissolved gases in the working cavity; E_i is modulus of bulk elasticity of the fluid; $E_{e.i}$ is modulus of bulk elasticity of the cavity walls; n is polytropic coefficient.

$$V_{l.1i} = V_{c.1i} + s_1 y_i, \tag{9}$$

$$V_{l.2} = V_{c.2} + s_2 (2l - y_1 - y_2), \tag{10}$$

where $V_{l.1i}$, $V_{l.2}$ is volume of the working fluid contained in the hydraulic lines; l is the rod stroke.

$$Q_{11} = \mu_1 b_1 (x_0 + x_1) \sqrt{\frac{2|p_{t.1} - p_{11}|}{\rho}} \operatorname{sgn}(p_{t.1} - p_{11}), \tag{11}$$

$$Q_{12} = \mu_1 b_1 (x_0 - x_1) \sqrt{\frac{2|p_{t.2} - p_{12}|}{\rho}} \operatorname{sgn}(p_{t.2} - p_{12}), \tag{12}$$

$$Q_{ti} = \mu_t s_t \sqrt{\frac{2|p_{v.1} - p_{t.i}|}{\rho}} \operatorname{sgn}(p_{v.1} - p_{t.i}), \tag{13}$$

$$Q_{ti} = Q_{1i}, \tag{14}$$

$$Q_{v.1} = \mu_v s_v \sqrt{\frac{2|p_p - p_{v.1}|}{\rho}} \operatorname{sgn}(p_p - p_{v.1}), \tag{15}$$

$$Q_{v.2} = \mu_v s_v \sqrt{\frac{2|p_2 - p_d|}{\rho}} \operatorname{sgn}(p_2 - p_d), \tag{16}$$

$$Q_{v.1} = Q_{t.1} + Q_{t.2}, \tag{17}$$

$$Q_p = Q_{v.1} + Q_{vo} + Q_{hm}, \tag{18}$$

$$Q_{vo} = \begin{cases} 0 & \text{at } p_p < p_n, \\ k_v \dfrac{p_p - p_n}{\Delta p_{v.n}} & \text{at } p_p > p_n \text{ and } Q_{vo} < Q_p, \\ Q_p & \text{at } p_p > p_n \text{ and } Q_{vo} > Q_p, \end{cases} \tag{19}$$

where μ_1 is coefficient of flow rates through the working slots of the flow divider; b_1 is width of the working slot of the flow divider; x_0 is initial gap between the spool valve and the flow divider sleeve; x_1 is displacement of the spool valve of the flow divider from neutral position; μ_v is coefficient of flow rates of the distributor working slots; s_v is area of the distributor working slots; p_2 is pressure at the inlet of the throttling element; p_d is pressure at the outlet of the throttling element; Q_p is pump capacity; Q_{w} is flow rate through the overflow valve; Q_{1m} is flow rate of fluid in the pump pressure line caused by operation of other hydraulic motors.

$$F_{f.i} = F_{fv.i} + F_{fk.i}, \tag{20}$$

$$F_{f.v.i} = k_{v.i} \frac{dy_i}{dt}, \tag{21}$$

$$F_{f.k} = \begin{cases} |F_{fd}| \operatorname{sgn}(v) & \text{at } v \neq 0 \\ F_a & \text{at } v = 0 \text{ and } |F_{fps}| > |F_a|, \\ |F_{fps}| \operatorname{sgn}(F_a) & \text{at } v = 0 \text{ and } |F_{fps}| < |F_a| \end{cases} \tag{22}$$

where F_{fvi} is force of viscous friction; F_{fki} is force of contact friction; k_{vi} is coefficient of viscous friction; F_{fd} is friction force in the mechanism movement; F_{fps} is friction force at the mechanism stoppage; v is speed of the rod movement; F_a is sum of forces acting on the rod.

Active forces include forces from the working fluid pressure on the hydraulic cylinder piston and a force resulting from the mechanism operation.

The results obtained in calculating the speed of movement of the hydraulic cylinder rods at a sudden increase in load at the hydraulic cylinder 5 by $\Delta F_m = 0.2 F_{m.0}$ are shown in Fig. 2. For convenience, the results are given in a dimensionless form. The speed of movement of the rods in steady-state operating conditions under symmetrical load, v_0, was taken as standard for reduction to the dimensionless form:

$$\bar{v}_1 = \frac{v_1}{v_0}, \quad \bar{v}_2 = \frac{v_2}{v_0}, \quad \delta v = \frac{v_1 - v_2}{v_0},$$

where v_1, v_2 are speeds of the rods.

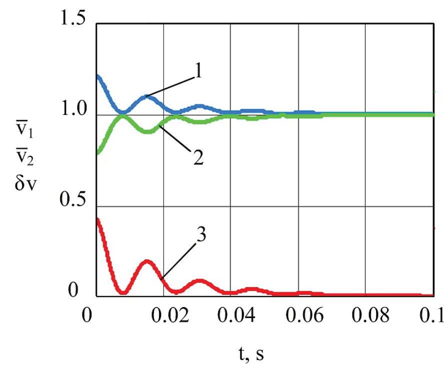

Fig. 2. Speeds of the hydraulic cylinder rods at a sudden change of load: $\bar{v}_1$ (1); $\bar{v}_2$ (2); $\bar{v}_2$ (3)

Deviation of speed from the nominal value is caused by the fact that the spool valve of the flow divider cannot instantly change area of the controlled throttle. It will take some time for the pressure differential on non-controlled throttles to attain same values (Fig. 3).

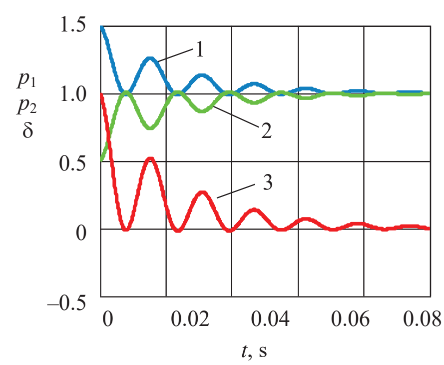

Fig. 3. Pressures in the inter-throttle chambers of the flow divider at a sudden load change: $\bar{p}_{t1}$ (1); $\bar{p}_{t2}$ (2); δp (3)

Pressures were reduced to the dimensionless form according to the following dependences:

$$\bar{p}_{t1} = \frac{p_{t1}}{p_{t0}}, \quad \bar{p}_{t2} = \frac{p_{t2}}{p_{t0}}, \quad \delta p_t = \frac{p_{t2} - p_{t1}}{p_{t0}}.$$

It is seen from Fig. 2, 3 that speed of the hydraulic cylinder rods and pressure in the inter-throttle chambers of the flow divider vary according to similar dependences.

This result is explained by the fact that speed depends on the flow rates through the throttles of the flow divider and the flow rates, in turn, depend on the pressure differential on the non-controlled throttle.

4. 2. Determining parameters of the controlled throttle of the double-slot throttling distributor

Since the flow through the throttle is determined by dependence (13), a solution suggests itself to install a double-slot throttling distributor instead of the non-controlled throttles (Fig. 4). Position of the spool valve 4 in the distributor depends on the pressure differential in the inter-throttle chambers and stiffness of the springs 2, 3.

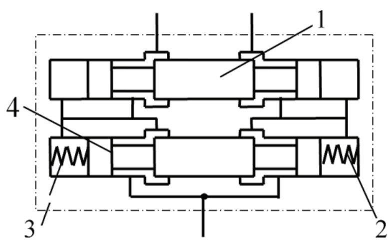

Fig. 4. Flow divider with a double-slot throttling distributor: spool valve (1, 4); centering springs (2, 3)

The working fluid enters the flow divider, separates in the working slots of the spool distributor 4 and then enters the end chambers of the spool valves 1 and 4. If loads on the hydraulic motors are equal, pressures at the flow divider outlet will be the same and the spool valves 1 and 4 will be in the neutral position. When loads on the hydraulic motors change, pressures in the hydraulic lines change as well resulting in a pressure change in the inter-throttle chambers of the flow divider. In this case, the spool valves 1 and 4 are displayed in such a way that the working slots get smaller in the less loaded line and bigger in the more loaded line. When pressures in the inter-throttle chambers equalize, the spool valve 1 will remain in the new position and the spool valve 4 will return to the neutral position under action of the centering springs 2, 3. Displacement of the spool valve 1 provides equal resistances of the hydrolines and the same pressure differential in the working slots of the spool valve 4. Since area of the working slots of the spool valve 4 is the same in the neutral position, flow rates in the hydrolines will be the same as well which provides movement synchronization of the hydraulic motors.

Given that $Q_{ti} = v_0 s_1$, dependence of the area of the working slots in the spool valve 4 on pressures in the inter-throttle chambers is obtained from (13) based on the condition that $Q_{ti} = \text{const}$:

$$s_{t.i} = \frac{v_0 s_1}{\mu_t \sqrt{\dfrac{2\left|p_{v.1} - p_{t.i}\right|}{\rho}}}. \qquad (23)$$

Dependence of areas of the working slots of the spool valve 4 reduced to the areas of the working slots in the neutral position on pressure differential in the inter-throttle chambers is shown in Fig. 5.

It is clear from the design diagram of the flow divider (Fig. 4) that displacement of the spool valve 4 is proportional to the pressure differential in the inter-throttle chambers. Therefore, the dependence in Fig. 5 enables profiling of the working slots of the spool valve 4.

The results obtained in the study of movement dynamics of synchronized hydraulic cylinders at the same perturbation as for the flow divider with non-controlled throttles are shown in Fig. 6, 7.

It is seen from Fig. 6, 7 that pressures in the inter-throttle chambers and speeds of movement of the hydraulic motors deviate from the stabilized values at a change of loading.

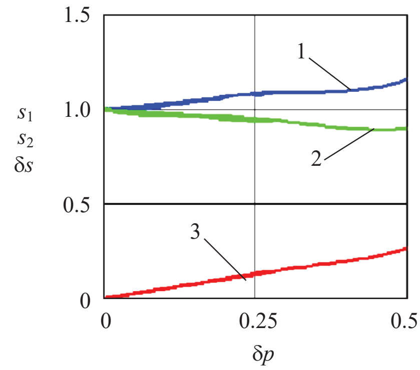

Fig. 5. Dependence of the throttling slot area of the flow divider on pressure differential in the inter-throttle chambers: $\bar{s}_1$ (1); $\bar{s}_2$ (2); δs (3)

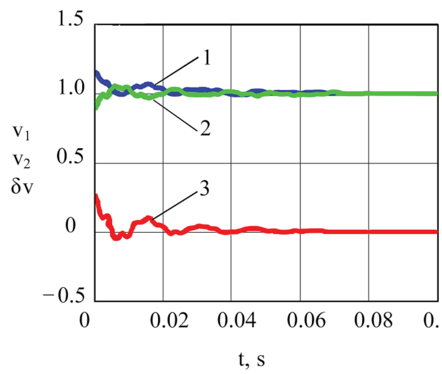

Fig. 6. Speed of movement of the hydraulic cylinder rods at a sudden increase in load on the flow divider with controlled throttles: $\bar{v}_1$ (1); $\bar{v}_2$ (2); δv (3)

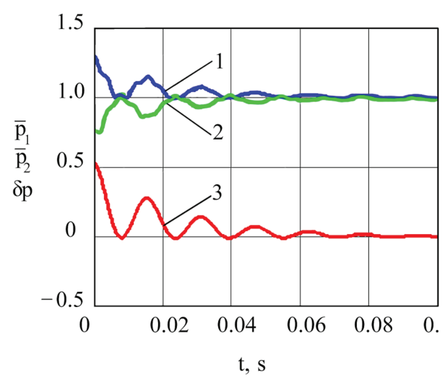

Fig. 7. Pressures in inter-throttle chambers at a sudden increase in load on the flow divider with controlled throttles: $\bar{p}_1$ (1); $\bar{p}_2$ (2); δp (3)

4. 3. Experimental bench and the results obtained in the studies of the error of synchronization of speed of movement of hydraulic motors

To test operation of a hydraulic unit with speed synchronization of hydraulic motors by means of a flow divider with a double-slot throttling distributor installed instead of standard non-controlled throttles, an experimental bench was designed. Its design diagram is shown in Fig. 8.

The stand consists of a pumping station 11, two vertical hydraulic cylinders 2 and 3, a flow divider 1 with controlled spool-valve throttles. To control pressure, pressure sensors 5 and 6 are installed in the inter-throttle chamber of the flow divider. Displacement of the rods is controlled by linear displacement sensors 7 and 8.

Force on the hydraulic cylinder rods is regulated by a load. An extra load is used on a special platform to instantly change force of one of the cylinders.

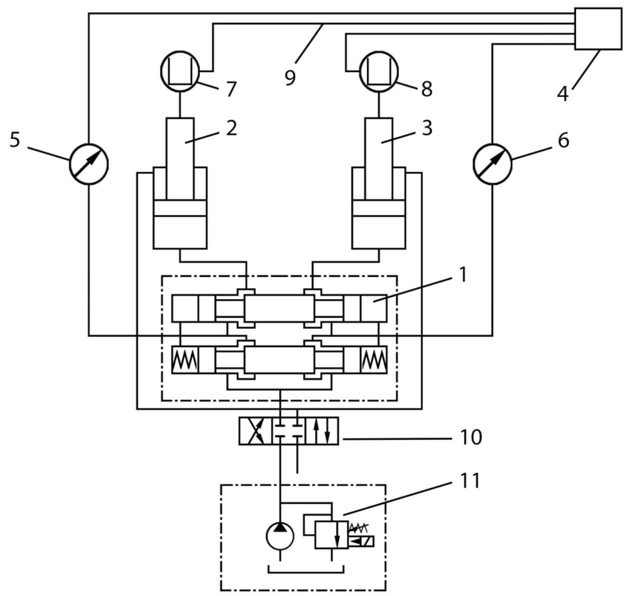

Fig. 8. Experimental bench: flow divider (1); hydraulic cylinders (2, 3); computer (4); pressure sensors (5, 6); linear displacement sensors (7, 8); cables (9); distributor (10); pumping station (11)

Information about pressures in the inter-throttle chambers of the flow divider and displacements of the hydraulic cylinder rods are transmitted from the sensors to the computer 4 via cables 9.

5. The results obtained in theoretical and experimental studies of transient operating conditions of movement of the hydraulic cylinder rods

According to the experimental data (Fig. 9), speeds of movement of the hydraulic cylinder rods were obtained by numerical differentiation of displacement of the rods.

Pressures in the inter-throttle chambers are shown in Fig. 10.

The graphs (Fig. 9, 10) also show calculated dependences of pressures and movement speeds.

The graphs (Fig. 9, 10) show that the pressure and speed variations were of a higher order than those for the flow divider with non-controlled throttles. However, these variations had a smaller amplitude and therefore affect the operation of hydraulic motors less.

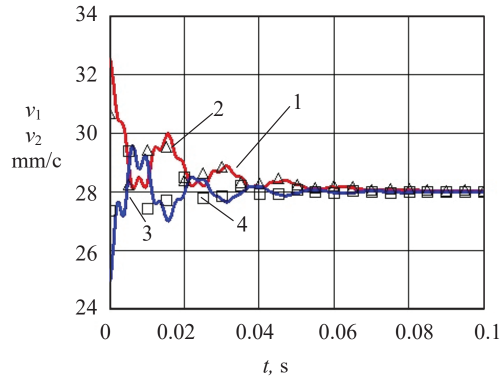

Fig. 9. Speeds of movement of the hydraulic cylinder rods at a sudden increase in load for the flow divider with controlled throttles: theoretical v_1 (1); experimental v_1 (2); theoretical v_2 (3), experimental v_2 (4)

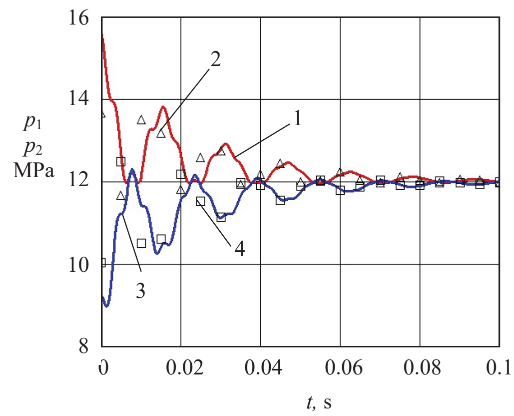

Fig. 10. Pressures in the inter-throttle chambers of the flow divider with controlled throttles at a sudden load increase: theoretical p_1 (1); experimental p_1 (2); theoretical p_2 (3); experimental p_2 (4)

6. Discussion of results obtained in the study of the error of speed synchronization of hydraulic motors in transient operating conditions

Comparison of the results obtained for the flow divider with a double-slot throttling distributor with the results for the flow divider with non-controlled throttles has shown the following:

1. The use of a double-slot throttling distributor controlled by the pressure differential in the inter-throttle chamber has made it possible to reduce the error in speed synchronization from 0.43 to 0.27, that is 1.6 times and the relative pressure differential in the inter-throttle chamber was reduced from 1 to 0.53, that is 1.9 times.

2. In the transient process, there were harmonics of higher order for speed and pressure. They were caused by movement of the spool valve of the double-slot distributor.

3. Presence of harmonics of higher order in variations of pressure and speed had no significant effect on performance of hydraulic motors since amplitude of these variations was negligible.

Reduction of the speed synchronization error was caused by simultaneous change of area of the throttle which stabilizes pressure differential and the area of the controlled throttle.

Theoretical graphs (Fig. 9, 10) had a series of deviations from the experimental ones. This can be explained by the fact that the actual hydraulic unit failed to instantly change load on the hydraulic cylinder rod. Despite the theoretically instantaneous change in the force acting on the hydraulic cylinder rod when lifting the load from the platform to create an additional load, the force changed gradually. Therefore, the change of pressure in the inter-throttle chambers of the flow divider was smoother which also led to a smaller amplitude of speed variations. However, it is clear from the graphs in Fig. 9, 10 that the characters of change in pressure and speed obtained in experimental studies coincide with the results of theoretical studies. Therefore, there are grounds to conclude on adequacy of the proposed model of operation of a hydraulic unit with simultaneous displacement of two hydraulic motors.

Despite this deviation of the experimental data from the calculated ones, it can be concluded that the proposed method of synchronization of hydraulic motors is in principle effective.

The results obtained may be considered feasible from a practical point of view since they make it possible to reasonably approach to the choice of the flow divider design. From a theoretical point of view, they suggest the possibility of reducing the speed synchronization error due to the use in the flow divider of the throttling distributor which is the advantage of this study. However, it should be noted that the study results (Fig. 6, 7) indicate an ambiguous influence of the throttling distributor on nature of the transient process. This is manifested, first of all, by the presence of higher-order harmonics in the obtained graph of the transient process (Fig. 6, 7).

The presence of variations of higher order in the transient process imposes some restrictions on the use of the results obtained which may be interpreted as a disadvantage of this study. The inability to perform a detailed analysis of the factors influencing amplitude and frequency of higher-order variations within the frames of this study gives rise to a potentially interesting direction of further studies. In particular, studies can be focused on analysis of influence of design parameters of the throttling distributor (diameter of the spool valve, rigidity of the centering springs) on nature of the transient process. Such analysis will make it possible to optimize design of the throttling distributor in time and amplitude of variations in non-stationary operating conditions of drive operation.

7. Conclusions

1. It was established for transient operating conditions of operation of synchronized hydraulic motors that there is a possibility of occurrence of an error in synchronization of speed of movement which reaches 43 % of the established speed. This error is caused by inertia of the spool valve of the flow divider. It manifests itself in a delayed compensation of the effect of load change on pressure differential in non-controlled throttles.

2. It was established that when applying a double-slot throttling distributor in the flow divider, the error of speed synchronization is reduced to 26 % of the established speed.

3. Theoretical conclusions concerning reduction of the speed synchronization error when using a flow divider with a double-slot throttling distributor were experimentally confirmed.

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