Розглянута по фазах математична модель роботи гідроприводу двоциліндрового діафрагмового розчинонасоса в процесі його роботи, а саме: рух гідроциліндрів, золотників керування та основного золотника. Дані, які отримані в результаті дослідження, дозволять ефективніше проектувати обладнання для оздоблювальних робіт, орієнтуючись на необхідні показники (тиск, потужність). В результаті проектування конструкції розчинонасоса згідно розробленої математичної моделі буде забезпечена рівномірність зміни тиску подачі розчину, збільшення всмоктувальної спроможності та рівня об'ємного ККД. Можливий підбір силової установки

Ключові слова: розчинонасос, гідропривод, золотник керування, ККД, будівельний розчин, тиск подачі, рівномірність пульсації, оштукатурювання, будівництво

Рассмотрена по фазам математическая модель работы гидропривода двухцилиндрового диафрагменного растворонасоса в процессе его работы, а именно: движение гидроцилиндров, золотников управления и основного золотника. Данные, полученные в результате исследования, позволят эффективнее проектировать оборудование для отделочных работ, ориентируясь на необходимые показатели (давление, мощность). В результате проектирования конструкции растворонасоса по разработанной математической модели будет обеспечена равномерность изменения давления подачи раствора, увеличение всасывающей способности и уровня объемного КПД. Возможен подбор силовой установки

Ключевые слова: растворонасос, гидропривод, золотник управления, КПД растворонасоса, строительный раствор, давление подачи, равномерность пульсации, оштукатуривание, строительство

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STUDY OF THE OPERATING ELEMENT MOTION LAW FOR A HYDRAULIC-DRIVEN DIAPHRAGM MORTAR PUMP

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1. Introduction

At present stage of development of Ukrainian building industry, plastering as the part of construction process cycle continues to be characterized by considerable complexity, duration and labor input. Share of these works exceeds 25 % in total manhours and 15 % in costs. That is why improvement of productivity and quality of finishing works as well as smaller proportion of manual labor in modern construction can be achieved through the introduction of new mechanized job practices with a reduced number of operations. This can be realized by means of elimination of rather significant process breaks connected with manual layered application of plaster on the treated surfaces of building structures.

In some cases, when mortar pumps of existing designs are used for applying plaster mixtures [1, 2] on construction structures, residual pulsations of supply pressure take place at the main line exit. This leads to a loss of plastering mortar which no longer has necessary spraying properties after its bouncing from the plastered surfaces and falling on the floor. In this regard, in order to ensure uniform spraying of the mortar during plastering in existing practice, the use of very energy-intensive compressor installations is necessary. These units supply compressed air that disperses particles of the plaster mixture coming from the compressor nozzle. However, in these cases, because of use of compressors, electric power consumption by the technological complex grows significantly. Compressed air forms a mist-like mixture of sand and cement particles which can endanger plastering operators with silicosis. The noise produced by compressed air exceeds the level permissible by sanitary code and adversely affects workers' hearing.

To determine productivity of the mortar pumping process using mortar pumps, it is necessary to know transportation distance, pressure in the hydraulic system of the mortar pump and rheological properties of the mortar. These parameters influence velocity of the moving mortar pump parts, so a problem arises to determine mathematical dependence of velocity of the hydraulic cylinder piston on time and pressure in individual sections of the mortar pump.

2. Literature review and problem statement

Currently, mortar pump drives are quite widespread with a crank-and-rod mechanism [3] which convert rotary motion into reciprocating motion. This design provides smoothest dynamic characteristics of the moving components of the pump which has a beneficial effect on performance of the drive and the pump as a whole. Along with this, the piston velocity in the drive with a crank mechanism varies in accordance with a law, close to the sinusoidal law. This fact most of all contributes to the rise of pressure pulsation in the mortar fed to the injection chamber. Such type of pumps includes one-piston pumps with a direct piston action on the transported medium. The single-acting mortar pump performs injection in the first half-cycle and suction of the pumped medium to the cylinder occurs in the second half-cycle, therefore the strongest pulsation occurs just in the single-acting mortar pumps.

Incomplete suction in the working chamber of the mortar pump is one more significant cause of pressure pulsation. This phenomenon is caused by expansion of mortar [4] under the action of depression arising from intake, presence of air bubbles in the mortar and discontinuation of the jet of mortar having reduced mobility.

Obviously, occurrence of incomplete suction is due to the properties and composition of mortars [5]. Large amounts of fine air bubbles enter the mortar during its mixing and intense agitation. At a reduced mortar mobility, small amount of bubble air in an unsaturated state remains in it for a long time. Fall of external pressure leads to an increase in bubble volume according to the Boyle-Mariotte law.

The occurrence of pressure pulsation is also due to such an adverse factor as an inverse mortar loss in work of the valves.

It is assumed that the main source of inverse losses of the pumped mortar is the presence of a “dead” subvalve space the lateral surface of which is limited by a straight circular cylinder with a diameter equal to the diameter of the hole in the valve saddle. If the return leakages were determined only by the size of the “dead” subvalve space, then pulsation of the pressure of mortar supply would depend little on the mortar mobility. But, as experimental data show, the lower mobility of the mortar, the higher supply pressure pulsation. Appearance of the mortar pressure pulsation is mainly caused by the inverse mortar losses because of the valves. These losses are equal to the sum of losses because of the suction and discharge valves.

Thus, pulsation of the delivery pressure in the mortar pumps is caused both by the structural features of their drives [6] and internal (rheological [7, 8]) properties of the building mortars. Moreover, at the stage of designing new mortar pumps, it is necessary to develop such a design of the drive and entire mortar pump which would provide minimal pulsations of the delivery pressure [9].

The existing calculation technique is only valid for the upward and downward piston movements under a control of a sliding valve, but it does not consider mortar pumping with the use of rubber diaphragms [10].

Thus, no research of the processes occurring in the hydraulic section of the hydraulically driven mortar pump was found. Probably, this is due to the fact that previous studies concerned mostly the processes occuring in the working chamber of the mortar pump.

3. The research objective and tasks

The work objective was to develop a mathematical model describing the processes occurring during automatic reciprocating movement of a piston in a hydraulic cylinder under the action of high oil pressure.

To achieve the goal, the following tasks were set:

– analyze suction capacity and the degree of uniformity of delivery pressure;

– improve existing methods of calculation and evaluation of process-dependent parameters that accompany mechanized plastering of surfaces;

– determine the nature of interaction of the delivery process indicators.

4. Materials and methods of studying operation process of a hydraulic diaphragm mortar pump

As a material for the study of operation of a hydraulic diaphragm mortar pump RNGD-3,8, a hydraulic drive system for the mortar pump consisting of a hydraulic cylinder, control sliding valve and a main hydraulic cylinder were used. The mortar pump was fitted with two diaphragms and a device for switching direction of mortar movement through the pipeline. Fraction size of the pumped mortar was up to 12 mm, and the pumping productivity was 3.8 m^3/h.

5. Results obtained in the studies of the process of operation of the hydraulic diaphragm mortar pump

Consider the diagram (Fig. 1) of the hydraulic drive system of the mortar pump which was used in description of the mathematical model.

The state of the system will be described by the following parameters:

– $x(t)$: axial coordinate of the cylinder 1 piston relative to time t;

– $x_{dr}(t)$: axial coordinate of the diaphragm relative to time t;

– $\upsilon(t)$: velocity of the hydraulic cylinder 1 piston;

– $\upsilon_2(t)$: velocity of the hydraulic cylinder 2 piston;

– $\upsilon_m(t)$: velocity of the main sliding valve;

– $y(t)$: is the axial coordinate of the hydraulic cylinder 2 piston relative to time t.

The above geometric and kinematic parameters describe movement of working bodies in the hydraulic system. The state of the working medium (oil) is described by parameter $P_i(t)$, the pressure in various system cavities.

Consider equations of three types that connect above parameters of the hydraulic system. Write the Newtonian differential equations of motion for the working bodies (solids):

$$m \cdot \frac{d\upsilon}{dt} = \sum F, \tag{1}$$

where m is the working body mass; $\sum F$ is the geometric sum of forces acting on the working body.

Compressibility of the working medium (oil) will be described by the following differential equation:

$$\frac{dP}{dV} = \frac{E}{V}, \tag{2}$$

where P is the oil pressure; V is the oil volume; E is the oil compression modulus.

Diaphragm elasticity is described by the following differential equation:

$$x'_{dr}(t) = \upsilon(t), \tag{3}$$

where $x'_{dr}(t)$ is the axial coordinate of the diaphragm.

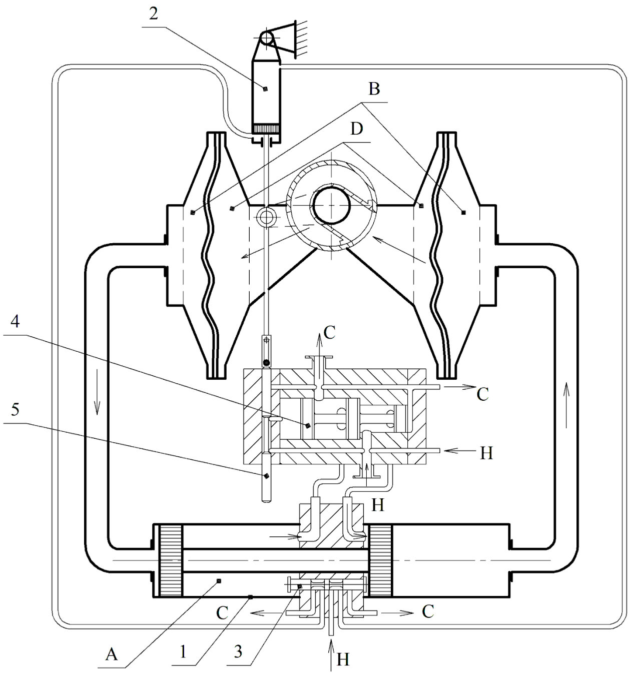

Fig. 1. Diagram of the hydraulic drive system of the mortar pump: hydraulic cylinder (1); hydraulic cylinder (2); control sliding valves (3); main sliding valve (4); control sliding valves (5); rod cavity of the hydraulic cylinder 1 (A); chambers forming the diaphragms (B, D); discharge line (C); delivery line (H)

Make widely used assumptions concerning both generation of differential equations and the choice of parameters of the hydraulic system state:

1. The working medium (oil) having distributed parameters is replaced by a model having concentrated parameters. This means that pressure $P_i(t)$ in each closed cavity of the hydraulic system is considered to be the same and does not depend on location in the system interior.
2. The instantaneous value of oil flow through constant sections in the hydraulic system is assumed to be equal to the average value Q in a steady state, that is when the pressure line opens to some cavity, the pressure will alter according to some law $P(t)$ and oil consumption in the cavity from the side of the pressure line will immediately make Q.
3. With a short length of hydraulic lines during the change in directions of the oil movement velocities, there are no hydraulic impacts.

Taking into account the above assumptions, describe motion of the hydraulic system at its various phases: acceleration, steady motion and slowdown of working bodies (pistons of the hydraulic cylinders 1 and 2, the control sliding valves and the main sliding valve).

Phase 1. Rightward acceleration of the hydraulic cylinder 1 piston.

The state of the system is as follows. The hydraulic cylinder 1 piston is in the leftmost position and the hydraulic cylinder 2 piston is in the extreme lower position. The control sliding valve 3 of the hydraulic cylinder 2 is in the extreme left position. The main sliding valve 4 is in the extreme right position. The sliding valve 5 of the main sliding valve control is in the extreme lower position (Fig. 1).

The acceleration phase will be considered complete when the velocity of the hydraulic cylinder 1 piston reaches the following value:

$$\upsilon = \frac{Q_o}{S_1 - S_1'}, \tag{4}$$

where Q_o is the oil supply from the oil pump; S_1 is the area of the hydraulic cylinder 1 piston; S_1' is the cross-sectional area of the hydraulic cylinder 1 rod.

The following forces act on the cylinder 1 piston during acceleration.

The force of inertia $F=m{\cdot}a$ acting from the right to the left and the force of diaphragm resistance F_d, determined by the following equation:

$$F_d = x_d \cdot K, \tag{5}$$

where K is the coefficient of diaphragm elasticity; x_d is the axial coordinate of the diaphragm; m is the mass of the moving parts (pistons, rods, sliding valves); a is the speedup acceleration,

$$a = \upsilon'(t) = \frac{d\upsilon}{dt}.$$

The diaphragm coordinate is determined by the formula:

$$x_d = x(t) \cdot \frac{S_{p.s.}}{\tilde{s}_B}, \tag{6}$$

where $S_{p.s.}$ is the area of the pressure pipe passage section; $\tilde{s}_B$ is the chamber B area.

Note that, $x(t) > x_{d,}$ that is the velocity of the hydraulic cylinder 1 piston will be greater than the diaphragm velocity.

Then the equation of the diaphragm resistance force F_d will take the following form:

$$F_d = x(t) \cdot \frac{S_{p.s.}}{\tilde{s}_B} \cdot K + \tau \cdot S_D, \tag{7}$$

where τ is the mortar displacement stress; S_D is the area of chamber D.

In accordance with the Shvedov-Bingham law [10]:

$$\tau = \tau_o + \mu \cdot \frac{dV}{dr}, \tag{8}$$

where τ_o is the limit stress of the mortar displacement; dV/dr is the velocity gradient taken equal to the average velocity υ_{av} [11].

Since the mortar pump has two diaphragms forming two chambers B and two chambers D, then the right chambers B is not filled with mortar at the start of operation. Therefore, one of the forces F_d has $\tau=0$.

The resistance force of oil going to the discharge:

$$F_M = P_{dis} \cdot (S_1 - S_1'), \tag{9}$$

where P_{dis} is the pressure of oil going to discharge.

The only force F_{dr} is acting from the left to the right due to oil pressure acting on the piston from the side of the rod cavity. It has the following form:

$$F_{dr} = P(t)\cdot(S_1 - S_1'). \quad (10)$$

Taking into account all forces acting on the hydraulic cylinder 1 piston, one can write the following differential equation:

$$m\cdot\frac{\mathrm{d}\upsilon}{\mathrm{d}t} = P(t)\cdot(S_1 - S_1') - F_d - F_M. \quad (11)$$

The following differential equation is valid for the oil pressure variation in time $P_i(t)$. The modulus of oil volume elasticity [12]:

$$E = \frac{\mathrm{d}P}{\mathrm{d}V}\cdot V \text{ or } \frac{V}{E}\cdot\mathrm{d}P = \mathrm{d}V.$$

Divide this expression by dt and obtain:

$$\frac{V}{E}\cdot\frac{\mathrm{d}P}{\mathrm{d}t} = \frac{\mathrm{d}V}{\mathrm{d}t}, \quad (12)$$

where dV/dt is the balance of oil consumption in the rod cavity of the hydraulic cylinder 1.

That is

$$\frac{\mathrm{d}V}{\mathrm{d}t} = Q_o - \upsilon(t)\cdot(S_1 - S_1'). \quad (13)$$

Thus the following is obtained:

$$\frac{V}{t}\cdot\frac{\mathrm{d}P}{\mathrm{d}t} = Q_o - \upsilon(t)\cdot(S_1 - S_1'), \quad (14)$$

where V is the volume of the rod cavity in the hydraulic cylinder 1 and the pipeline from the oil pump to the hydraulic cylinder 1; $V/E=\beta$ is the coefficient of volumetric deformation of oil.

Differential equation takes the following finite form:

$$\beta\cdot\frac{\mathrm{d}P}{\mathrm{d}t} = Q_o - \upsilon(t)\cdot(S_1 - S_1'). \quad (15)$$

Thus, the system of differential equations describing speedup of the cylinder 1 piston from the extreme left point will take the following form:

$$\begin{cases} m\cdot\dfrac{\mathrm{d}\upsilon}{\mathrm{d}t} = P(t)\cdot(S_1 - S_1') - F_d - F_M; \\ \beta\cdot\dfrac{\mathrm{d}P}{\mathrm{d}t} = Q_o - \upsilon(t)\cdot(S_1 - S_1'); \\ x'_{dr}(t) = \upsilon(t) \end{cases} \quad (16)$$

at initial conditions:

$$\begin{cases} \upsilon(0) = 0; \\ P(0) = P_{\min}; \\ x(0) = 0. \end{cases}$$

Phase 2. Upward movement of the cylinder 2 piston after the control sliding valve 3 of the hydraulic cylinder 2 has been moved to the right under pressure of the hydraulic cylinder 1 piston. Since the force required to ensure operation of this control sliding valve is insignificant, it can be neglected.

The hydraulic cylinder 2 tail piece is connected by a movable joint 2 (Fig. 2) with the tail piece of the control sliding valve 5, so the phase of upward movement of the cylinder 2 piston is divided into two parts: a and b.

a) upward movement of the piston until response of the control sliding valve 5. The hydraulic cylinder 2 piston will start its movement when the hydraulic cylinder 1 piston goes to its extreme right-hand position: $x(t)=x_o$.

Write the equation of motion of the hydraulic cylinder 2 piston:

$$m_2\cdot\upsilon_2'(t) = P\cdot(S_2 - S_2') - K_{rad}\cdot\mu\cdot S(t) - m_2\cdot g - F_{2M}, \quad (17)$$

where K_{rad} is coefficient of mean radial velocity; $\upsilon_2'(t)$ is acceleration of the hydraulic cylinder 2 piston; F_{2M} is the force of resistance of the oil in the hydraulic cylinder 2 going to discharge, $F_{2M}=P_{dis}\cdot S_2$.

The axial coordinate of movement of the hydraulic cylinder 2 piston corresponds to its velocity: $y_2'(t) = \upsilon_2(t)$.

The angle α of flag 1 turning (Fig. 2) is determined by the same formula:

$$tg\alpha = \frac{y}{l},$$

from which

$$\alpha = \frac{y}{l}.$$

The system of differential equations takes the following form:

$$\begin{cases} m_2\cdot\upsilon_2(t) = P\cdot(S_2 - S_2') - K_{rad}\cdot\mu\dfrac{y_2}{l}\cdot K_1 - m_2\cdot g - F_{2M}; \\ y_2'(t) = \upsilon_2(t), \end{cases} \quad (18)$$

where K_1 is the coefficient of slot parameter; μ is the dynamic viscosity.

b) since the movable joint 2 (Fig. 2) is a finger moving along the slot in the tail piece of the control sliding valve, then after the rod passes through the gap l_1, the control sliding valve starts to respond. Therefore, the forces acting on this sliding valve, namely the friction force F_{fr} and its own weight m_5 are added:

$$\begin{cases} m_2\cdot\upsilon_2(t) = P\cdot(S_2 - S_2') - \\ -K_{rad}\cdot\mu\dfrac{y_2}{l}\cdot K_1 - m_2\cdot g - F_{2M} - F_{fr} - m_5\cdot g; \\ y_2'(t) = \upsilon_2(t), \end{cases} \quad (19)$$

where F_{fr} is the friction force of control sliding valve 5, $F_{fr}=K_{fr}\cdot S_{lat.sl.}$; K_{fr} is the coefficient of friction of control sliding valve 5; $S_{lat.sl.}$ is the lateral area of control sliding valve 5; m_5 is the weight of control sliding valve 5.

Phase 3. Upward movement of control sliding valve 5.

The system of differential equations will be the same as the second part b of the phase 2 because control sliding valve 5 is connected by a movable joint with the rod of the cylinder 2:

$$\begin{cases} m_2 \cdot \upsilon_2(t) = P \cdot (S_2 - S_2') - \\ -K_{rad} \cdot \mu \dfrac{y_2}{l} \cdot K_1 - m_2 \cdot g - F_{2M} - F_{fr} - m_5 \cdot g; \\ y_2'(t) = \upsilon_2(t). \end{cases} \quad (20)$$

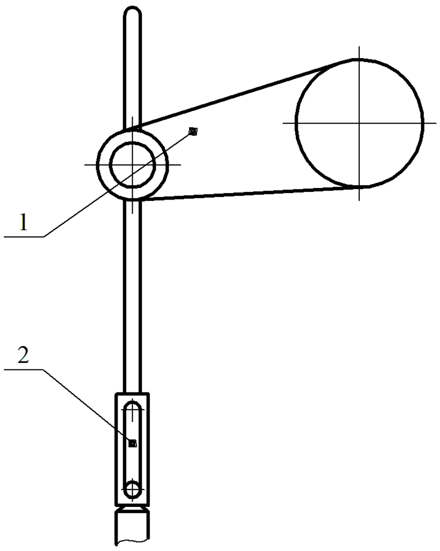

Fig. 2. Diagram of joint of the hydraulic cylinder rod with the tail piece of the control sliding valve and with the flag of the feeding pipeline: flag (1); movable joint (2)

Phase 4. Leftward movement of main sliding valve 4. The movement will begin when α=30°, y=30°·l (Fig. 1).

We have a Newtonian differential equation of the sliding valve movement in which there is only one force $F_{r.sl.}$ resulted from the oil pressure on the sliding valve due to the difference in forces from oil pressure on the shoulders of larger and smaller diameters:

$$m \cdot \upsilon_m'(t) = F_{\text{r.sl.}}, \quad (21)$$

where $\upsilon_m'(t)$ is the acceleration of main sliding valve 4.

Taking into account the fact that there is a change in oil pressure in this sliding valve, the differential equation can be written as:

$$\beta \cdot \frac{dP}{dt} = Q_o - \upsilon_m \cdot (S_{c.m.} - \tilde{s}_{\text{c.m.}}), \quad (22)$$

where υ_m is the velocity of main sliding valve 4; $S_{c.m.}$ is the area of the larger shoulder of main sliding valve 4; $s_{c.m.}$ is the area of the smaller shoulder of main sliding valve 4.

The final form of the system of differential equations will take the following form:

$$\begin{cases} m \cdot \upsilon_m'(t) = P(t) \cdot (S_{c.m.} - \tilde{s}_{c.m.}); \\ \beta \cdot \dfrac{dP}{dt} = Q_o - \upsilon_m(t) \cdot (S_{c.m.} - \tilde{s}_{c.m.}). \end{cases} \quad (23)$$

Phase 5. Leftward movement of cylinder 1 piston (for a steady operation).

The system of differential equations will be the same as for the first phase, however, in the steady operation of the mortar pump, right chamber D will be filled with mortar due to underpressure in the right chamber B:

$$m \cdot \frac{d\upsilon}{dt} = P(t) \cdot (S_1 - S_1') - F_{\text{d.st.}} - F_M, \quad (24)$$

where $F_{d.st.}$ is the resistance of diaphragm at a steady motion.

The mortar displacement stress τ in chambers D will be different:

$$F_{d.st.} = F_{d.r.} + F_{d.l.}, \quad (25)$$

where $F_{d.r.}$, $F_{d.l.}$ is the resistance force of the right and the left diapragms.

The following is obtained:

$$\begin{cases} m \cdot \dfrac{d\upsilon}{dt} = P(t) \cdot (S_1 - S_1') - F_{d.st.} - F_M; \\ \beta \cdot \dfrac{dP}{dt} = Q_o - \upsilon(t) \cdot (S_1 - S_1'); \\ x_{dr}'(t) = \upsilon(t). \end{cases} \quad (26)$$

Phase 6. Downward movement of cylinder 2 piston.

The cylinder 2 rod will move when x_a=x_o, i. e. the hydraulic cylinder 1 is in the leftmost position x_o.

The system of differential equations is similar to that for the second phase, however, it differs by the resistance force F_{2M} acting on hydraulic cylinder 2 piston from the side of the rod cavity applied by the oil going to discharge and force F_{dr} due to the oil pressure on the piston from the side of the piston cavity:

$$F_{dr} = P \cdot S_2, \quad (27)$$

$$F_{2M} = P_{dis} \cdot (S_2 - S_2'), \quad (28)$$

where S_2 is the area of hydraulic cylinder 2 piston; S_2' is the cross section of the hydraulic cylinder 2 rod.

a) The downward movement of the piston until control sliding valve 5 responds. The hydraulic cylinder 2 piston will start to move when the hydraulic cylinder 1 piston takes its leftmost position:

$$\begin{cases} m_2 \cdot \upsilon_2'(t) = P \cdot S_2 - K_{rad} \cdot \mu \cdot \dfrac{y_2}{l} \cdot K_1 + m_2 \cdot g - F_{2M}; \\ y_2'(t) = \upsilon_2(t). \end{cases} \quad (29)$$

b) Due to the movable joint 2 (Fig. 2) described in phase 2, the following is obtained:

$$\begin{cases} m_2 \cdot \upsilon_2'(t) = P \cdot S_2 - K_{rad} \cdot \mu \cdot \dfrac{y_2}{l} \cdot K_1 + \\ +m_2 \cdot g - F_{2M} - F_{fr} + m_5 \cdot g; \\ y_2'(t) = \upsilon_2(t). \end{cases} \quad (30)$$

This phase is over when α=–30° (Fig. 2), namely when the hydraulic cylinder 2 rod takes its lowermost position.

Phase 7. Downward movement of control sliding valve 5.

The system of differential equations will repeat the second part b of phase 6 (Phase 3):

$$\begin{cases} m_2 \cdot \upsilon_2'(t) = P \cdot S_2 - K_{rad} \cdot \mu \cdot \dfrac{y_2}{l} \cdot K_1 + \\ + m_2 \cdot g - F_{2M} - F_{fr} + m_5 \cdot g; \\ y_2'(t) = \upsilon_2(t). \end{cases} \qquad (31)$$

Phase 8. Rightward movement of the main sliding valve 4. After control sliding valve 5 has connected the pressure line with the left chamber of imain sliding valve 4, then due to the difference in oil pressures acting on the shoulders of larger and smaller diameters, the following is obtained:

$$\begin{cases} m \cdot \upsilon_m' = P(t) \cdot (S_{c.m.}) - P(t) \cdot (S_{c.m.} - \tilde{s}_{c.m.}); \\ \beta \cdot \dfrac{\mathrm{d}P}{\mathrm{d}t} = Q_o - \upsilon_m \cdot (S_{c.m.} - (S_{c.m.} - \tilde{s}_{c.m.})). \end{cases} \qquad (32)$$

The above eight phases fully describe the cycle of the mortar pump operation and repeat in the abovementioned order.

Application of this mathematical model with the following initial data makes it possible to determine dependence of the oil pressure level P in the pressure pipeline on time t which in turn will enable determining its performance.

Q_0: oil supply from the oil pump, $6.96 \cdot 10^{-4}$ m^3/h;

m: mass of the moving parts (pistons, rods, connection yoke), 22 kg;

P_{dis}: pressure of oil going to discharge, 0.2 MPa;

V: volume of oil in cavities between the oil pump and the hydraulic cylinder rod cavity, $0.3 \cdot 10^{-3}$ m^3;

E: modulus of oil volume elasticity, $1.5 \cdot 10^9$ Pa;

S_2: area of the hydraulic cylinder piston, $38.5 \cdot 10^{-4}$ m^2;

S_2': cross-sectional area of the hydraulic cylinder rod, $19.6 \cdot 10^{-4}$ m^2;

P (0): initial pressure of the piston speedup, $2 \cdot 10^6$ Pa;

g: acceleration of Earth gravity, 9,81 m/s^2;

β: coefficient of volumetric deformation of oil $1 \cdot 10^{-12}$;

y_1: stroke of the control sliding valve before opening of the slot, 6 mm;

μ: dynamic viscosity, 5 Pa·s;

ρ: oil density, 900 kg/m^3.

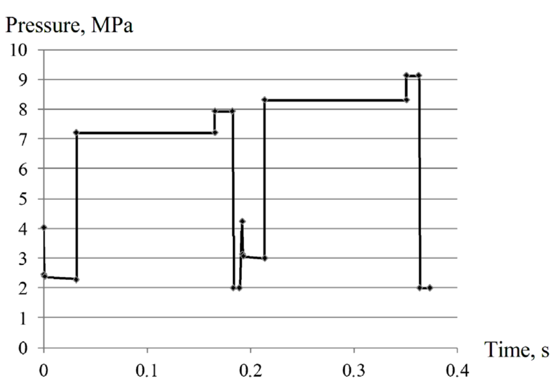

Fig. 3. Chart of the oil pressure variation during working cycle of the hydraulic mortar pump drive cylinder

Due to the inertial properties of the electric motor and the drive parts, deformation of the rubberized fabric pressure hoses of the pipeline, finite time of switching sliding valves, the transient processes are substantially smoothed out and the real diagram of pressure variation in the working cycle of the hydraulically driven pump is as shown in Fig. 4.

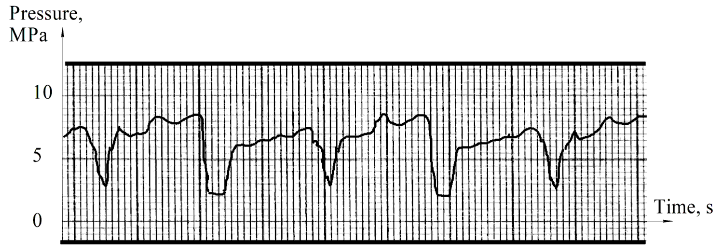

Fig. 4. Experimental curves of oil pressure variation in the pressure pipeline

Indeed, there is a temporary increase in oil pressures in the areas near the two dead points due to the start of movement of the control sliding valve. Upon braking and stopping of the piston, the oil pressure drops sharply. At this time, switching of the main sliding valve takes place. One can not but notice that the area of reduced oil pressure when the piston passes the upper dead point is much wider than at the lower dead point.

6. Discussion of results obtained in the study of the hydraulic diaphragm mortar pump operation

The developed mathematical model of operation of the hydraulic diaphragm mortar pump has allowed the authors to divide the complete cycle of the mortar pump into separate phases. For each phase, it was possible to determine duration, nature of the piston velocity variation and the oil pressure level in the pipeline.

It should be noted that this mathematical model does not take into account elasticity of the pumped medium which can significantly change dynamics of the hydraulic diaphragm mortar pump. Air content in oil can also vary within a wide range. According to [13], the relative amount of air in oil can be from 0.02 % after a long break in operation to 2...3 % during work. Also, it should be considered that oil elasticity decreases with pressure growth. This phenomenon can be explained as follows: almost all free air dissolves in oil at a higher pressure and ceases to compress under the law of Boyle-Mariotte. At atmospheric pressure, air in the mortar is in a form of fine bubbles.

Thus, development of the mathematical model of the hydraulic drive pump has provided a rich material for a further design-related improvement of the hydraulic system of the hydraulic cylinder automatically realizing reciprocating motion.

7. Conclusions

1. The proposed mathematical model of operation of the hydraulic diaphragm mortar pump makes it possible to determine dependence of the hydraulic cylinder velocity on time t in various sections. It is also possible to determine pressure P in the mortar pump chambers. This, in turn, allows us to determine the machine performance.

2. The rapid transition of the piston from the braking and acceleration process in the «dead» points to the working velocity causes motion of the piston at a constant velocity during the most cycle part. This positively affects uniformity of supply of the pumped mortar and reduces pressure pulsation.

3. Comparison of theoretical results of the hydraulic pump operation with the experimental curve of the oil pressure variation in the pressure pipeline allowed us to assert that this mathematical model makes possible the following:

1) analyze the suction capacity;

2) determine degree of uniformity of the supply pressure;

3) if necessary, develop special measures to improve performance of the mortar pump.

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