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THE OPTIMAL CONTROL OF DYNAMIC LOADS IN A PUMP COMPLEX WITH ADJUSTABLE PIPELINE VALVES

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ОПТИМАЛЬНЕ КЕРУВАННЯ ДИНАМІЧНИМИ НАВАНТАЖЕННЯМИ В НАСОСНОМУ КОМПЛЕКСІ З РЕГУЛЬОВАНОЮ ТРУБОПРОВІДНОЮ АРМАТУРОЮ

Purpose. Determination of the optimum law of control for locking and regulating valve electric drive, providing change in pressure in hydronetwork within the admissible limits and minimum time of pipeline valve control in emergencies at forced outage of the power supply system.

Methodology. A control method for locking-and-regulating pipeline valve electric drive has been substantiated taking into account the nonlinear dependence of the stopcock hydraulic resistance coefficient on a relative degree of the stopcock opening. Using the telegraph equations apparatus and finite element method, a mathematical model of a pump complex has been worked out taking into consideration wave processes in the hydronetwork, which occur during control of pipeline valves with electric drive. Applying the dynamic programming method, the optimum law of control of the stopcock electric drive for the pump complex emergency operation modes has been determined.

Findings. It has been proved that generation of irregular control of pipeline valves with electric drive is an efficient method for reduction of dynamic loads in the pump complex. A quality criterion for a closed electromechanical system of dynamic load reduction in a pump complex has been offered. Weight coefficients at quality criterion components in its emergency and operating modes have been substantiated. An optimum law of control of the valve frequency-controlled electric drive has been determined. This law provides change in pressure in the hydrosystem within admissible limits and the most rapid control of pipeline valves in emergencies.

Originality. Expediency of developing an irregular law of locking and regulating pipeline valve electric drive has been theoretically proved which takes into account nonlinear dependence of the stopcock hydraulic resistance coefficient on a relative degree of opening at the whole interval of its actuator movement. For the first time a mathematical model of a pump complex has been proposed allowing research of the influence of various laws of pipeline valve control on the value of pipeline network dynamic loads caused by occurrence of surges in its emergency and operating modes.

Practical Value. A structure of an electromechanical system of reducing dynamic loads in a pipeline network on the basis of a locking and regulating valve frequency-controlled electric drive with a standby power supply has been substantiated. It will allow eliminating inadmissible pressure growths in the hydrosystem, extending manufacturing equipment life time, improving reliability and efficiency of pump complex functioning in emergencies.

Keywords: *pump complex, dynamic loads, forced outage of power supply, pipeline valves, frequency-controlled electric drive, optimum control law*

Introduction. Processes accompanied by surges, pressure pulsations, cavitation self-oscillations, equipment vibration, etc. inevitably occur in a hydrosystem during operation of pumping plants (PP), and variation of their operating conditions. The mentioned phenomena can be caused by a number of factors: failure of operating pumping units (PU) electrical supply; check valve response, rapid shutting or opening of

safety or locking and regulating valves; periodic stops, repeated starts as well as pumps commutation switching; disconnection or connection of a heavy consumer, etc.

Analysis showed that 60% of pipeline destruction is due to surges, pressure drops, equipment vibration caused by jumps and pulsating character of pressure variation in the pipeline. Breaks of water supply networks take place particularly often when PP renews water supply after emergency disconnections in power

networks where pressure oscillation is 5–7 times as large as admissible values. Such conditions result in reduction of pipeline valve lifetime, failure of pump outfit, pipeline network breaks and considerable financial expenditure for elimination of the consequences of the breakage.

The existing state of affairs is conditioned by the lack of reliable and efficient means of protection against surges and pressure pulsations. The conventional hydro-protective valve system based on safety and check valves, dampers without dissipative elements, liquid dump valves, etc. does not meet the requirements of technological reliability; it responds when the breakage has occurred and results in premature wear of equipment.

Sufficient attention has been paid to the problem of research of surges, the analysis of wave processes in a pressure pipeline system [1–5]. Papers [2–5] contain a comparative characteristic of counter-surge valves used in pipeline networks. The main drawbacks of such control means include:

- bulkiness of the design and impossibility of considerable decrease of pressure oscillation amplitude when air chambers, receivers, liquid dump valves, etc. are installed;
- lack of the possibility of smooth variation of the flow rate, taking into consideration nonlinear dependence of the valve hydraulic resistance coefficient on the relative rate of its opening;
- uncontrollable valves at sudden power interruption at the pump plant.

In this connection, it is expedient to develop pipeline valve control systems providing decrease in dynamic loads in pump complexes and improved reliability of technological equipment operation in emergency.

Presentation of the main research and explanation of scientific results. Pipeline valves make the most significant element of pump complex technological equipment and function as protective, safety and regulating means. The control of pump complex valves influences the character of transient processes in the hydrosystem.

To exclude increased dynamic loads under the mentioned conditions, an electromechanical system of decreasing dynamic loads (ESDDL) has been developed on the basis of variable-frequency electric drive (ED) of locking and regulating stopcock with a standby power supply (Fig. 1). Such a system can be used for valve control under both usual (operating) conditions, when head or capacity is regulated according to the current water consumption, and under emergency conditions connected with sudden electric power outage, and occurrence of surges.

This system includes: locking and regulating valves LV installed at the centrifugal pump CP output with a drive induction motor IM1; a frequency converter FC connected to valve drive induction motor IM2 whose shaft is connected to valve spindle through reducer R; head sensors HS and flow sensors FS installed in the pump flow tube; position sensor PS of pipeline valve locking device LV; mains voltage sensor MVS; unin-

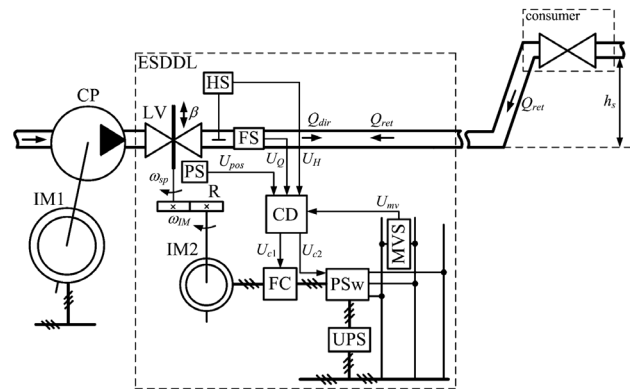


Fig. 1. Functional diagram of electromechanical system of decreasing dynamic loads

interruptible power supply UPS; power switch PSw, control device CD.

To reduce dynamic loads in pump complex at sudden power supply outage the system switches the power of the stopcock variable-frequency electric drive to a standby power supply.

The dependence of hydraulic resistance coefficient ξ_v on the relative rate β of valve opening is the main characteristic of the pipeline valves influencing the character of transient processes in the hydrosystem

$$\xi_v(\beta) = A((1/\beta) - 1)^C + B((1/\beta) - 1)^D + \xi_0. \quad (1)$$

It follows from analysis (1) that the variation $\xi_v(\beta)$ is of nonlinear character: a considerable growth of hydraulic resistance coefficient ξ_v occurs at the section at $\beta \leq \beta_{cr}$ where β_{cr} is threshold value of relative rate of opening depending on the type of pipeline valves.

Formation of irregular stopcock opening (shutting) is an efficient way of reducing dynamic loads in pump complex

$$\beta(t) = 1 - (t/t_{sh})^{1/n}. \quad (2)$$

At section $\beta > \beta_{cr}$ stopcock actuator moves with spindle rotation frequency equal to ω_1 ; at section $\beta \leq \beta_{cr}$ it moves with the rotation frequency of $\omega_2 < \omega_1$.

In (1) and (2): β is a relative rate of pipeline valves opening, A, B, C, D are approximation coefficients depending on the pipeline valve type; ξ_0 is the hydraulic resistance coefficient when valves are completely open ($\beta = 1$); n is the intensity coefficient of pipeline valve control ($n \geq 1$); t, t_{sh} are current time and time of valves complete shutting, respectively, s.

Fig. 2 contains dependences of head relative increase $\Delta h = \Delta H / \Delta H_{dir}$ on the control intensity coefficient n , where $\Delta H, \Delta H_{dir}$ are head increase at the assigned value n and direct surge, respectively, m. In this case time t_{sh} of valves complete shutting is multiple of the period of pressure oscillation distribution along the pipeline network, equal to surge phase T_{ph} .

The analysis of the obtained curves demonstrated that at $n = 1 \div 7$ considerable dynamic loads occur in the hydrosystem. These loads are characterized by

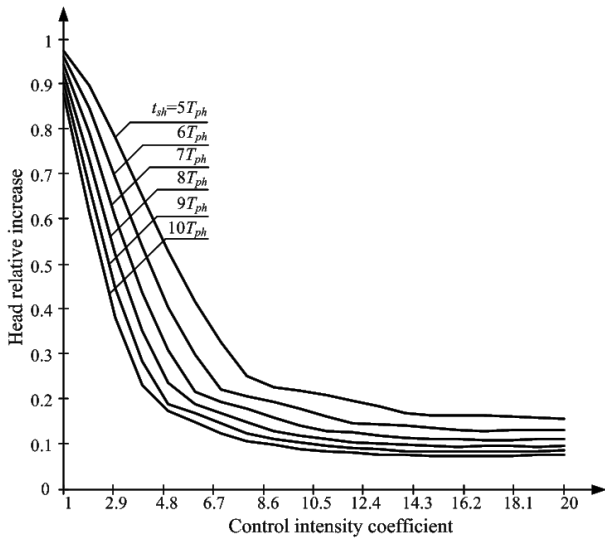


Fig. 2. Dependence of head relative increase Δh on coefficient n of the valves control intensity

sharp change of pressure in the pipeline network, often exceeding the nominal value by 60 ÷ 90 %; at $n = 7 \div 20$ pressure in communications network varies within admissible limits up to 20 ÷ 25 % of the nominal value.

To research dynamic processes in pump complex hydronetwork with different control of pipeline valves, a mathematical model of a pump complex with a stopcock adjustable induction ED was created. The model block diagram is shown in Fig. 3.

The pump is presented by an equation of the type

$$H_p = H_0 v_p^2 - R_p Q^2, \quad (3)$$

where H_0 , H_p are head at zero pump capacity and at the pump impeller output, respectively, m; R_p is pump internal resistance, s^2/m^5 ; $v_p = \omega_p/\omega_{pn}$ is relative fre-

quency of pump impeller rotation; ω_p , ω_{pn} are current and nominal pump rotation frequencies, respectively, s^{-1} .

Pipeline network is described by telegraph equations known from [6]

$$\begin{cases} \frac{\partial H}{\partial x} - \frac{1}{gS} \frac{\partial Q}{\partial t} - \frac{1}{dS^2} \frac{Q|Q|}{2g} = 0 \\ \frac{\partial H}{\partial t} + \frac{c^2}{gS} \frac{\partial Q}{\partial x} = 0 \end{cases}, \quad (4)$$

where $H = p/(\rho g)$, Q is piezometric head and flow of liquid in the current section of the pipeline, respectively, m, m^3/s ; p is pressure, Pa; $S = \pi d^2/4$ is cross-sectional area, m^2 ; d is pipeline diameter, m; λ is the dimensionless coefficient of the pipeline resistance; c is speed of surge wave or sound propagation in the operating environment, m/s; ρ is operating environment density, kg/m^3 ; $g = 9.81$ is gravitational acceleration, m/s^2 .

System (4) was solved by means of the finite element method enabling one to pass from partial derivative equations to difference analog and present a pipeline network as a finite number of sections with equal parameters [7]. Then equations of head and flow for the i -th section are

$$\begin{cases} \frac{H_i - H_{i-1}}{l} + l_0 \frac{dQ_i}{dt} + r_0 Q_i |Q_i| = 0 \\ \frac{dH_i}{dt} + c_0 \frac{Q_i - Q_{i-1}}{l} = 0 \end{cases}, \quad (5)$$

where $l = L/N$ is pipeline section length, m; L is pipeline network length, m; N is the number of sections into which the pipeline is divided; $r_0 = 1/(2gS^2d)$, $l_0 = 1/(gS)$, $c_0 = c^2/(gS)$ are specific parameters of the pipeline section, s^2/m^6 , s^2/m^3 , m^{-1} ; $H_i = h_i + h_s$, Q_i , $H_{i-1} = h_{i-1} + h_{s,i-1}$, Q_{i-1} are head and flow at the output

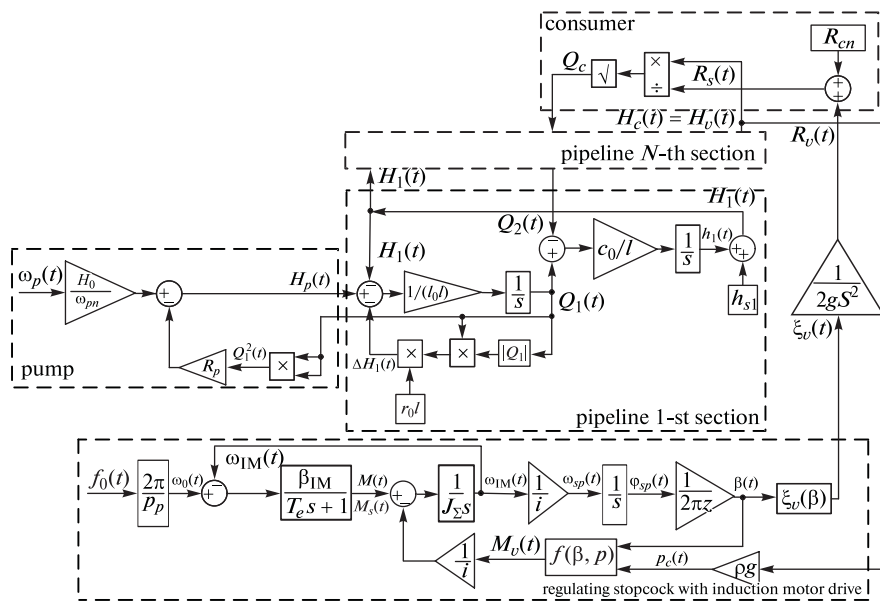


Fig. 3. Block diagram of a model of pump complex with adjustable valves

and input of the pipeline i -th section, respectively, m, m^3/s ; $\lambda = 0.11(k/d)$ is the pipeline resistance coefficient; k is pipe roughness, m; $h_s, h_{s_{i-1}}$ stand for the height of the beginning and the end of the i -th section in relation to the pump axis, respectively, m; $h_i = p_i/(\rho g)$, $h_{i-1} = p_{i-1}/(\rho g)$ stand for absolute head at the output and input of the pipeline i -th section, respectively; p_i, p_{i-1} are absolute pressure at the output and input of the pipeline i -th section, respectively, Pa.

Regulating stopcock is described by expression (1) wherein a relative rate of the stopcock opening is equal

to $\beta = \varphi_{sp}/(2\pi z)$, where $\varphi_{sp} = \int_0^{t_{sh}} \omega_{sp}(t) dt$ is a current

angle of the valve spindle rotation, rad; $\omega_{sp} = \omega_{IM}/i$ is current rotational frequency of the valve spindle, s^{-1} ; ω_{IM} is current rotational frequency of the drive motor shaft, s^{-1} ; i is an angular speed ratio of the stopcock ED reducer; z is the number of turns necessary for stopcock complete shutting.

Pipeline valves in the model (Fig. 3) are installed at the last section of the hydronetwork to provide the necessary values of the head H_c and flow Q_c at the consumer's. The head at the stopcock output is

$$H_v(t) = (R_v(t) + R_{cn}) Q_{cn}^2(t), \quad (6)$$

where $R_{cn} = (H_{cn} - h_s)/Q_{cn}^2$ is the consumer's nominal hydraulic resistance, s^2/m^5 ; H_{cn}, Q_{cn} are respectively, nominal head and flow in the consumer's network, m, m^3/s^3 ; $R_v(t) = \xi_v(\beta(t))/(2gS^2)$ is stopcock hydraulic resistance, s^2/m^5 .

The consumer is described by a hydraulic characteristic of the type

$$H_c - h_s = R_{cn} Q_c^2, \quad (7)$$

where H_c, Q_c are head and flow in the consumer's network, respectively, m, m^3/s ; h_s is static head in the pipeline system, m.

A pump complex with the following parameters was taken for modelling: pump nominal head $H_{pn} = 100$ m, operating environment flow rate $v = 1.4$ m/s, a consumer's hydraulic resistance $R_{cn} = 52$ s^2/m^5 ; the length $L = 5000$ m and the diameter $d = 1.2$ m of the pipeline; the number of pipeline sections $N = 20$, surge velocity $c = 1000$ m/s, the number of turns of regulating stopcock spindle for its complete shutting $z = 28$. Specific parameters of each pipeline section were: $r_0 = 0.000162$ s^2/m^6 , $l_0 = 0.0901$ s^2/m^3 , $c_0 = 90$ 132 m^{-1} . An induction motor AOC2-41-4U3 with nominal power of $P_n = 5.2$ kW and reducer with gear ratio of $i = 250$ was chosen as a regulating stopcock ED.

Fig. 4 shows plots of time variation of the head $H_p(t)$ at the pump output and $H_v(t)$ at the stopcock at: uniform shutting (curves 1 and 3, Fig. 4) with invariable frequency of induction ED supply voltage equal to 50 and 20 Hz, respectively;

irregular shutting with voltage frequency of induction ED supply equal to 50 Hz at section $0.1 \leq \beta \leq 1$ and 20 Hz at $\beta \leq 0.1$, respectively (curve 2, Fig. 4).

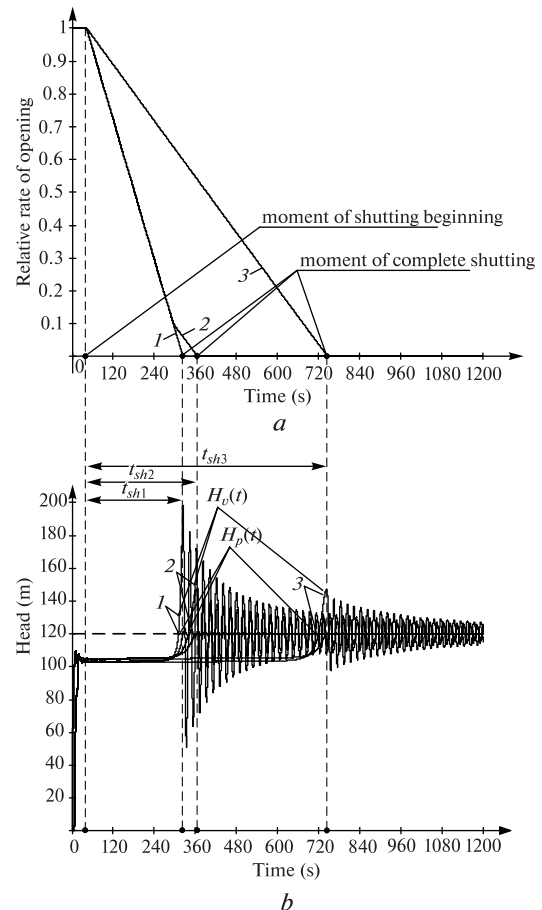


Fig. 4. Pump complex transient processes diagrams obtained on the basis of a mathematical model at even closing of the valve with the voltage frequency of asynchronous electric drive constant supply equal to 50 (curve 1) and 20 Hz (curve 3), respectively, and at uneven closing with asynchronous electric drive supply voltage frequency equal to 50 Hz at section $0.1 \leq \beta \leq 1$ and 20 Hz at $\beta \leq 0.1$ (curve 2):
a – stopcock opening degree variation curves; b – pipeline network head variation curves

It can be seen from the analysis of the obtained curves that stopcock uniform control (curve 1, Fig. 4) is accompanied by rapid growth of pressure at the stopcock; its value is twice as large as the pump nominal head. Decrease in supply voltage frequency at $\beta = 0.1$ results in reduction of the head down to the value of $1.4H_{pn}$ and increase in valve control time up to maximum value of t_{sh3} (curve 3, Fig. 4).

During the irregular control of the valves (curve 2, Fig. 4), valve shutting time t_{sh2} reduces by several times, which is especially important in emergency conditions connected with sudden electric power outage in pump complex and occurrence of liquid backflow. In this case the pressure in the pipeline network does not exceed the value of $1H_{pn}$.

Pipeline valves ED control results in pressure reduction at an irregular rate of shutting at the section when $\beta \leq 0.1$ and in a considerable decrease in resist-

ance moment, which is provided by reduction of frequency of voltage supplied to the stopcock induction ED.

In [8] methods of determining the electric motor power and gear ratio of the reducer of variable frequency electric drive of pipeline valves are proposed. These methods are based on:

- taking into account nonlinear dependences of hydraulic $\xi_v(\beta)$ and mechanical $M_v(\beta, t)$ characteristics of regulated pipeline valves;
- meeting the condition of eliminating the moment overload and inadmissible overheating of the stopcock induction motor;
- providing minimum time values of pipeline valve control and head increase in the hydronetwork when surge in the system is impossible.

The above stated made it possible to decrease the electric motor power and gear ratio of the reducer contained in the stopcock variable-frequency ED, providing pressure change in the pipeline within admissible limits. Later on induction motor 4AC90L4U3 with nominal power of $P_n = 2.4$ kW and reducer with gear ratio of $i = 91$ were considered for research.

When closed ESDDL was created and optimal trajectory of pipeline valve control was determined, a quality criterion was offered. It provides minimization of dynamic loads in hydrosystem and operative control of pipeline valves in emergency (contingency) situations

$$J = \int_0^{t_{sh}} (v_v(t)\Delta h(t) + k\Delta t) dt \rightarrow \min, \quad (8)$$

where $v_v(t) = \omega(t)/\omega_0(t)$ is stopcock induction ED relative speed; $\omega(t)$, $\omega_0(t)$ are current and synchronous speeds of stopcock ED shaft, respectively, s^{-1} ; $\Delta h(t) = (H(t) - H_{v_0}) / \Delta H_{dir}$ is relative increase of head at the stopcock; H_{v_0} is head in the pipeline in the steady state before stopcock shutting, m; $H(t)$ is current head at the stopcock, m; ΔH_{dir} stands for head increase at a direct surge (maximum head increase at the stopcock shutting), m; $\Delta t = (t - t_0) / t_{max}$ is relative time of stopcock shutting; t , t_0 are the current time and moment of the beginning of stopcock shutting, s; t_{max} is maximum time of shutting at minimum value of stopcock electric drive supply voltage frequency, s; $k = 0...1$ is the coefficient taking into account the significance of the quality criterion component Δt .

Search for an optimum law of pipeline valve control corresponds to such ED supply voltage frequency f_0 dependence on relative rate β of its opening at which minimum value of the quality criterion is provided. To find this law a dynamic programming method was used. This method enables control optimization at each stage of measuring relative rate β of opening pipeline valves thus eliminating complete analysis of all possible variants of control.

As to the problem considered in the paper, a search procedure for the optimum law of regulating stopcock ED control (Fig. 5), where at every stage of measuring

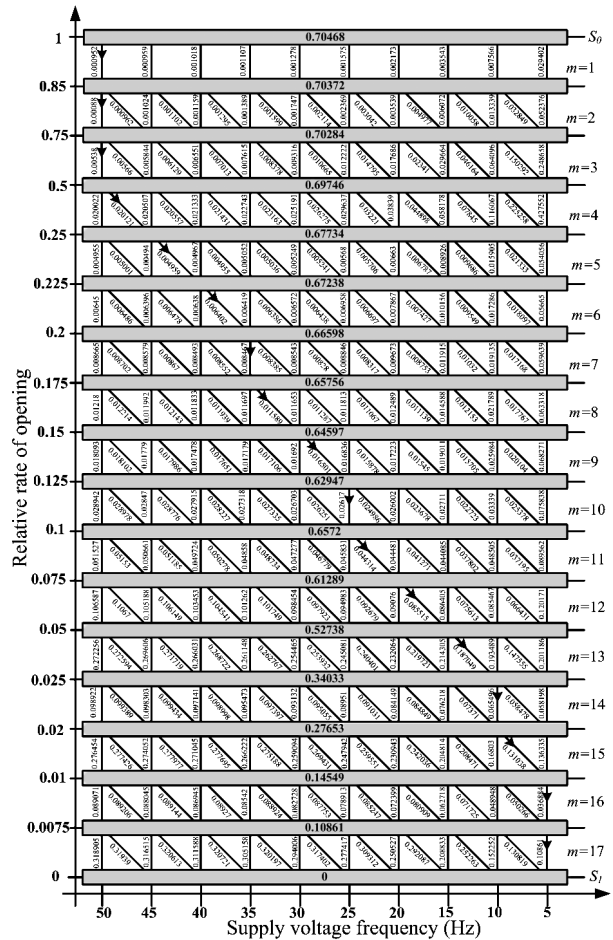


Fig. 5. Search procedure for optimum law of stopcock electric drive control

β there occurs either decrease of supply voltage frequency (diagonal sections) or it remains unchanged (vertical sections), has been proposed. Quality criterion increase is shown for each section

$$\Delta J = J_m - J_{m-1}, \quad (9)$$

where J_m , J_{m-1} are values of quality criterion at the following m and the previous ($m - 1$) stages, respectively.

Search for the optimum control law is performed in reverse time direction – from point S_1 at complete shutting of the stopcock ($\beta = 0$) to point S_0 corresponding to its complete opening ($\beta = 1$). According to the dynamic programming principle, minimum values of the quality criterion obtained at an optimum control are stated at each stage at a certain value of β , starting from the current point and finishing with final S_1 . In Fig. 5 bold arrows show the optimum law of stopcock ED control for a hydrosystem where maximum allowed head increase is $\Delta H_{max} = 40$ m.

Fig. 6 contains time variable curves of the relative rate of stopcock opening (a) and pressure in the pipeline network (b) in an open system when nonadjustable ED is used (curve 1) and a closed system with an adjustable ED at optimum (curve 2) and intermediate (different from the optimum one, curve 3) control laws, respectively.

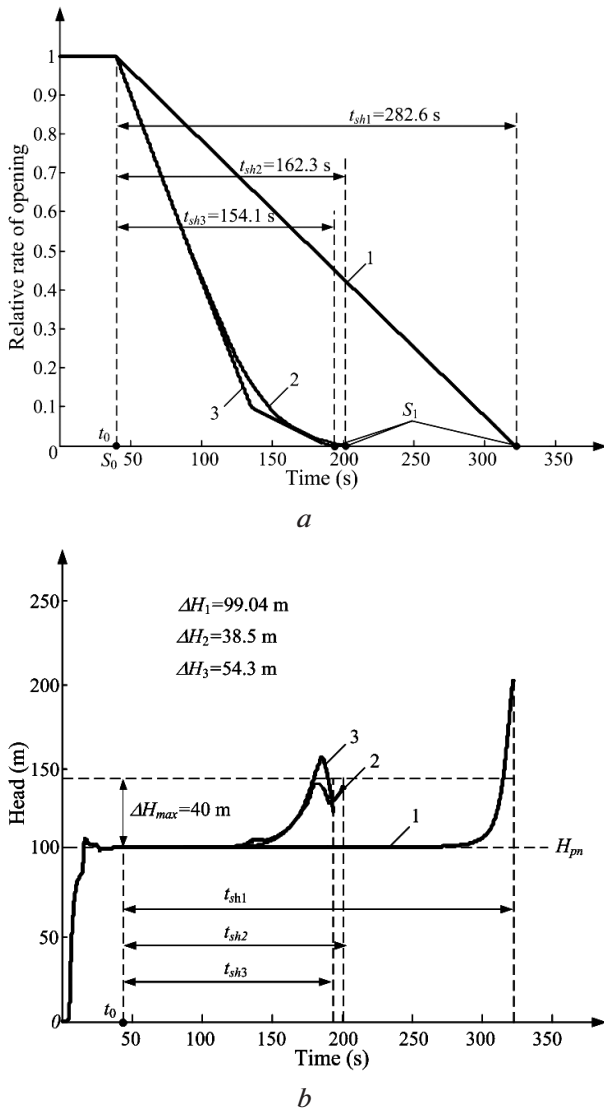


Fig. 6. Time variable curves of the relative rate of stopcock opening (a) and pressure in the pipeline network (b) according to different laws of stopcock electric drive control

Use of nonadjustable ED of pipeline valves (curve 1, Fig. 6) is accompanied by the highest excess of the head $\Delta H_1 = 99.04$ m in the hydrosystem with maximum time of valves shutting $t_{sh1} = 320$ s. At the optimum law of stopcock ED control (curve 2, Fig. 6) the lowest increase in pressure $\Delta H_2 = 38.5$ m is observed with quicker shutting of the valves $t_{sh2} = 190$ s. In case of intermediate control law, different from the optimum one (curve 3, Fig. 6) there occurs increase in ΔH by 35.7 % compared with ΔH_2 ; time of stopcock control practically does not change.

The results of pump complex ESDDL operation modelling are confirmed by experimental research based on a physical model of a pumping facility including [9]:

- two centrifugal pumping units WILLO which can be connected both in series and in parallel;
- devices for regulation of pump induction electric motor speed on the basis of LENZE and MITSUBISHI frequency converters, respectively;

- manifold pipeline network containing horizontal and vertical pipeline sections;
- locking and regulating valves in the form of manually operated ball cocks and a check valve, respectively;
- receiving tanks for water;
- ZETKAMA regulating stopcock with a rubber-coated wedge, installed in the pump complex head line and assigned for change of the complex technological parameters by alteration of the rate of its opening;
- measuring equipment for registration and control of technological (OWEN pressure sensors and ERGOMERA flow sensors), power (ALLEGRO MICROSYSTEMS current sensors; AVAGO TECHNOLOGIES voltage sensors based on galvanically uncoupled amplifiers; wattmeters) and mechanical (rotational frequency sensors) parameters of pump complex operation;
- control and data gathering module in the form of an A-to-D and D-to-A card with a USB interface L-CARD.

The experimental facility makes it possible to solve a whole complex of scientific-research and scientific-applied problems [9], one of which consists in the research of influence of pipeline valve ED control on the values of dynamic loads in a pump complex. With this purpose in view, regulating stopcock is equipped with ESDDL (Fig. 7).

ESDDL includes:

- regulating stopcock itself;
- worm gear motor with a DAVID BROWN cylindrical pass, equipped with a three-phase induction motor;
- MITSUBISHI three-phase frequency converter enabling one to control the rate and trajectory of stopcock shutting by changing supply voltage frequency;
- OWEN control device assigned for indication and determination of the current rate of stopcock opening according to the pulse number and repetition interval, obtained from the sensor situated on the stopcock shaft;
- actively inductive sensor of the stopcock actuator position CARLO GAVAZZI providing the possibility

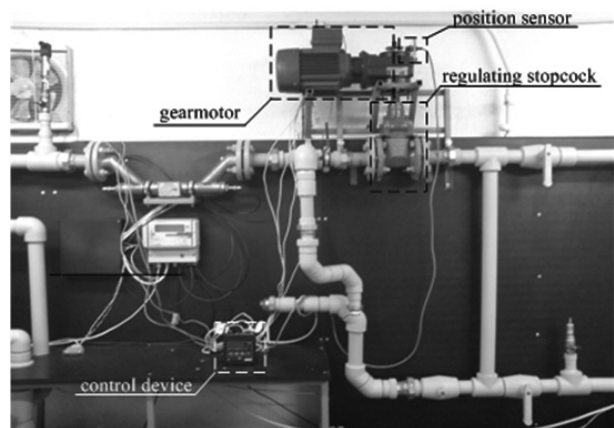


Fig. 7. Regulating stopcock with induction motor drive

to form rectangular pulses whose number is proportional to the rate of its opening;

- automated interface converter assigned for reading the values of the current rate of stopcock opening from the control device by means of a computer.

To analyse dynamic loads in a pump complex at a sudden outage of pumping unit power supply and shutting of regulating stopcock the following cases are considered:

- shutting of the stopcock at uniform control with invariable frequency f_0 of supply voltage ($f_0 = 40; 20; 10; 5$ Hz) along the whole trajectory of shutting;

- shutting of the stopcock at irregular control with supply voltage frequency f_0 equal to 40 Hz at section $0.2 \leq \beta \leq 1$ and variable frequency (5, 10, 20 Hz) at section $\beta \leq 0.2$.

Fig. 8 shows pressure $H_v(t)$ and current $I_{ph}(t)$ curves of IM stator phase for the case of a sudden pumping unit hitch and stopcock shutting in the head pipeline under the uniform (a) and irregular (b) control. In the former case the supply voltage frequency $f_0(t)$ is 40 Hz at the relative rate of stopcock opening $0 \leq \beta \leq 1$; when irregular pace is used frequency is 40 Hz at section $0.2 \leq \beta \leq 1$, whereas at section $0.2 \leq \beta \leq 1$ it is 5 Hz. In Figs. 8, 9 the following designations are used: t_1 is the moment of sudden disconnection of the pumping unit from the electric mains; t_2 is the moment of connecting IM to the frequency converter and start of regulating stopcock shutting, corresponding to relative rate of its opening $\beta = 1$; t_3 is the moment of variation of IM supply voltage frequency f , corresponding to relative rate of stopcock opening $\beta = 0.2$; t_4 is the mo-

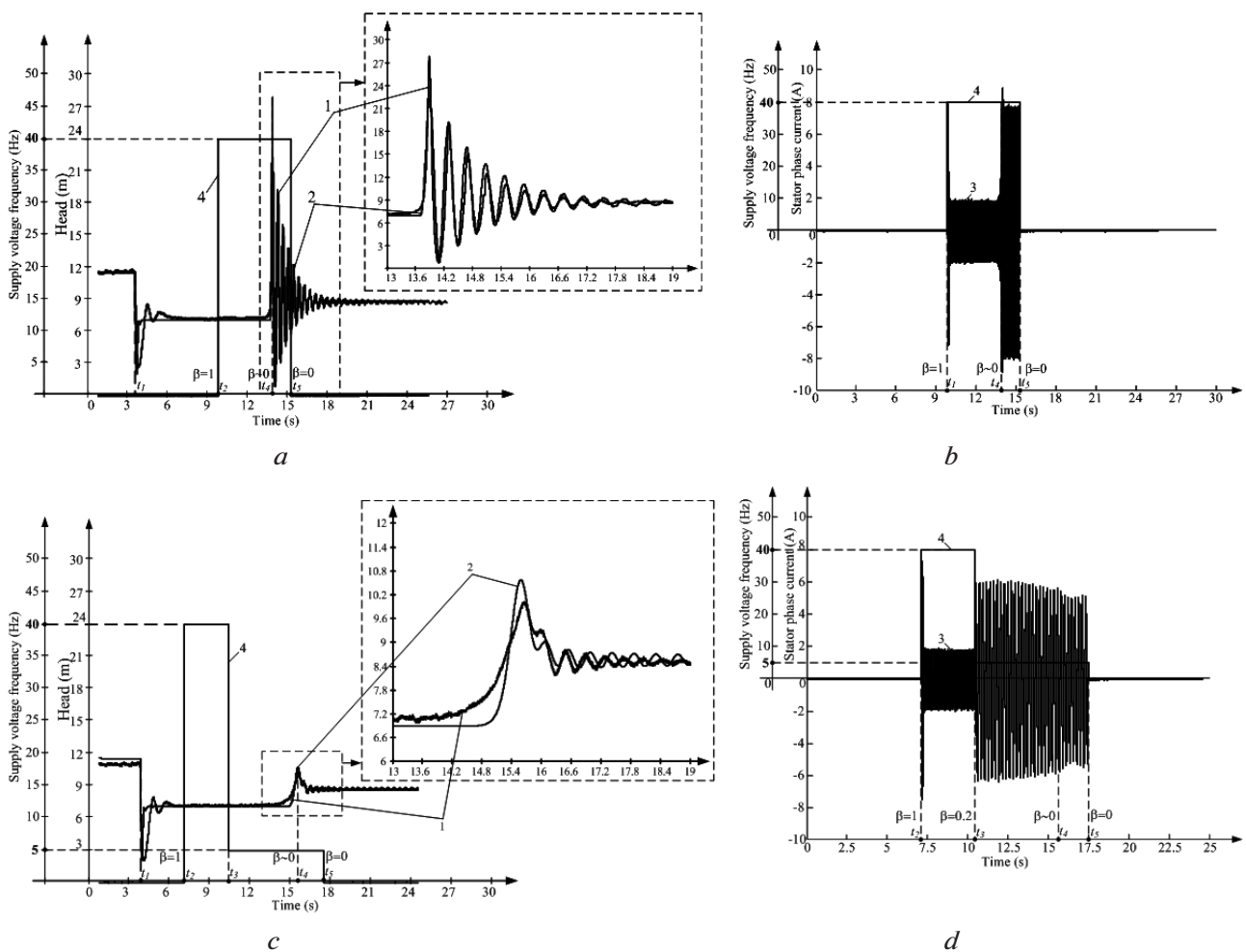


Fig. 8. Diagrams of pump complex transient processes under different laws of control of the stopcock electric drive:

a – head variation curves $H_v(t)$ for pipeline network at even closing of the stopcock with supply voltage frequency $f_0(t)$ equal to 40 Hz, where 1, 2 are stopcock head variation curves obtained by experiment and by modelling, respectively; b is for variation curves for the stator phase current $I_{ph}(t)$ (3) and supply voltage frequency $f_0(t)$ (4) of the stopcock IM at its even closing with supply voltage frequency $f_0(t)$ equal to 40 Hz, obtained by experiment; c is for pipeline network head variation curves $H_v(t)$ at uneven closing of the stopcock with supply voltage frequency $f_0(t)$ equal to 40 Hz at section $0.2 \leq \beta \leq 1$ and 5 Hz at section $\beta \leq 0.2$, where 1, 2 are stopcock head variation curves obtained by experiment and by modelling, respectively; d shows variation curves for stator phase current $I_{ph}(t)$ (3) and supply voltage frequency $f_0(t)$ (4) of the stopcock IM at uneven closing of the stopcock with supply voltage frequency $f_0(t)$ equal to 40 Hz at section $0.2 \leq \beta \leq 1$ and 5 Hz at section $\beta \leq 0.2$ obtained by experiment

ment of the highest increase of pressure ΔH in the pipeline network, corresponding to practically complete shutting ($\beta \sim 0$) of the stopcock; t_5 is the moment of IM disconnection from the frequency converter and finish of regulating stopcock shutting, corresponding to relative rate of its opening $\beta = 0$.

Fig. 8 shows pressure curves $H_v(t)$ obtained by experiment using a pumping unit physical model (curve 1) and by modelling (curve 2).

Indices of transient processes at different stopcock control are given in Table.

Table

Indices of transient processes in the pump complex under different stopcock control

Control method	Supply voltage frequency f_0 , Hz	Shutting time t_{sh} , s	Increase ΔH in the head, m
uniform	40	5.6	19.62
	20	10	8.3
	10	17.4	3.9
	5	34	1.6
irregular	40; 20	6.2	8.3
	40; 10	8	3.9
	40; 5	9.7	1.6

The analysis of the obtained results demonstrated that at uniform shutting of the stopcock, when supply voltage frequency is 40 Hz, pipeline network pressure is 2.5 times as high as the pump nominal head. When pipeline valves are controlled at supply voltage frequency of 5 Hz, pressure is 12-fold higher, but shutting time is six fold larger. Irregular control is optimum. In this case supply voltage frequency is 40 Hz at section $0.2 \leq \beta \leq 1$ and 5 Hz at section $\beta \leq 0.2$. Pressure increase is inconsiderable $\Delta H = 1.6$ m, and valve shutting time decreases by 3.5 times compared with the last analysed case.

The obtained experimental characteristics of the pump complex under different methods of locking-regulating pipeline valve control confirmed the expediency of stopcock electric drive irregular control enabling elimination of pipeline head pulsations causing surges.

Conclusions. A method of reducing the dynamic loads in a pump complex has been offered under both operating and emergency conditions by means of forming an irregular rate of locking and regulating valves control.

A structure of electromechanical system of dynamic loads decrease in a pipeline network on the basis of locking and regulating stopcock variable-frequency electric drive with the use of uninterruptible power supply has been grounded. The system makes it possible to exclude inadmissible pressure increase in the hydraulic network when the pipeline valves are closed at the quickest rate.

An optimum law of stopcock variable-frequency electric drive control has been determined. This law provides minimum dynamic loads in the hydronetwork and the quickest pipeline valve control in emergency (contingency) situations.

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Мета. Визначення оптимального закону керування електроприводом запірно-регулюючої засувки, що забезпечує зміну тиску в гідромережі у припустимих межах і мінімальний час керування арматурою в аварійних ситуаціях за раптового відключення енергопостачання системи.

Методика. Обґрунтовано метод керування електроприводом запірно-регулюючої трубопроводної арматури з урахуванням нелінійної залежності коефіцієнта гідравлічного опору засувки від відносного ступеня її відкриття. З використанням апарату телеграфних рівнянь і методу кінцевих елементів розроблена математична модель насосного

комплексу, що враховує хвильові процеси в гідромережі при керуванні трубопровідною арматурою з електроприводом. Шляхом використання методу динамічного програмування визначено оптимальний закон керування електроприводом засувки в аварійних режимах роботи насосного комплексу.

Результати. Доведено, що ефективним способом зниження динамічних навантажень у насосному комплексі є формування нерівномірного керування електроприводом трубопровідної арматури. Запропоновано критерій якості замкненої електромеханічної системи зниження динамічних навантажень у насосному комплексі. Обґрунтовані вагові коефіцієнти при складових критерію якості в аварійних і експлуатаційних режимах його роботи. Визначено оптимальний закон керування частотно-регульованим електроприводом засувки, що забезпечує зміну тиску в гідросистемі у припустимих межах і найбільш швидко керування трубопровідною арматурою в аварійних ситуаціях.

Наукова новизна. Теоретично доведено доцільність формування нерівномірного закону керування електроприводом запірно-регулюючої трубопровідної арматури, що враховує нелінійну залежність коефіцієнта гідравлічного опору від відносного ступеню відкриття на всьому інтервалі переміщення її робочого органу. Уперше запропонована математична модель насосного комплексу, що дозволяє досліджувати вплив різних законів керування трубопровідною арматурою на величину динамічних навантажень у трубопровідній мережі, які викликані виникненням гідравлічних ударів в аварійних і експлуатаційних режимах його роботи.

Практична значимість. Обґрунтована структура електромеханічної системи зниження динамічних навантажень у трубопровідній мережі на базі частотно-регульованого електроприводу запірно-регулюючої засувки з резервним джерелом електропостачання, що дозволить виключити неприпустиме підвищення тиску в гідромережі, продовжити строк служби технологічного обладнання, підвищити надійність і ефективність функціонування насосного комплексу в аварійних режимах роботи.

Ключові слова: насосний комплекс, динамічні навантаження, раптове відключення енергопостачання, трубопровідна арматура, частотно-регульований електропривод, оптимальний закон керування

Цель. Определение оптимального закона управления электроприводом запорно-регулирующей задвижки, обеспечивающего изменение давления в гидросети в допустимых пределах и минимальное время управления арматурой в аварийных ситуациях при внезапном отключении энергоснабжения системы.

Методика. Обоснован метод управления электроприводом запорно-регулирующей трубопроводной арматуры с учетом нелинейной зависимо-

сти коэффициента гидравлического сопротивления задвижки от относительной степени ее открытия. С использованием аппарата телеграфных уравнений и метода конечных элементов разработана математическая модель насосного комплекса, учитывающая волновые процессы в гидросети при управлении трубопроводной арматурой с электроприводом. Путем использования метода динамического программирования определен оптимальный закон управления электроприводом задвижки в аварийных режимах работы насосного комплекса.

Результаты. Доказано, что эффективным способом снижения динамических нагрузок в насосном комплексе является формирование неравномерного управления электроприводом трубопроводной арматуры. Предложен критерий качества замкнутой электромеханической системы снижения динамических нагрузок в насосном комплексе. Обоснованы весовые коэффициенты при составляющих критерия качества в аварийных и эксплуатационных режимах его работы. Определен оптимальный закон управления частотно-регулируемым электроприводом задвижки, обеспечивающий изменение давления в гидросистеме в допустимых пределах и наиболее быстрое управление трубопроводной арматурой в аварийных ситуациях.

Научная новизна. Теоретически доказана целесообразность формирования неравномерного закона управления электроприводом запорно-регулирующей трубопроводной арматуры, который учитывает нелинейную зависимость коэффициента гидравлического сопротивления от относительной степени открытия на всем интервале перемещения ее рабочего органа. Впервые предложена математическая модель насосного комплекса, которая позволяет исследовать влияние различных законов управления трубопроводной арматурой на величину динамических нагрузок в трубопроводной сети, вызванных возникновением гидравлических ударов в аварийных и эксплуатационных режимах его работы.

Практическая значимость. Обоснована структура электромеханической системы снижения динамических нагрузок в трубопроводной сети на базе частотно-регулируемого электропривода запорно-регулирующей задвижки с резервным источником электроснабжения, которая позволит исключить недопустимые повышения давления в гидросети, продлит срок службы технологического оборудования, повысит надежность и эффективность функционирования насосного комплекса в аварийных режимах работы.

Ключевые слова: насосный комплекс, динамические нагрузки, внезапное отключение энергоснабжения, трубопроводная арматура, частотно-регулируемый электропривод, оптимальный закон управления

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