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Dynamic Modelling of Centrifugal Hydraulic Load of Pumping Station Electric Drive

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Abstract

The paper analyzes modern trends in modelling the centrifugal hydraulic load of the pumping station electric drive. It was found that a very simplified approach to the representation of one of the inextricably linked subsystems (hydraulic or electromechanical) is mostly used, which significantly reduces the possibility of a complex analysis of the processes in its individual elements. Models based on this approach can be effectively applied to solve highly specialized tasks and do not always give a clear idea of the state and modes of all the elements of the subsystems. This problem was partially solved while carrying out the simulation of the steady state modes of the pumping stations. The paper offers an advanced dynamic model of the centrifugal hydraulic load of the pumping station electric drive which enables studying the processes in the main elements of the centrifugal pump, taking into account its internal parameters and dependences of these parameters on the physical parameters of the fluid.

Keywords: electric drive; load; pumping station; centrifugal pump; hydraulic network.

1. Definition of the research problem

The electrotechnical complexes of the pumping stations (PS), which ensure the movement of fluids through pipelines, are big consumers of electricity. Their operating modes can be both quasi-stationary (with the fluid consumption change rate being below 0.1 % per second [1], except for emergency modes and for starting or stopping of assemblies in powerful trunk pipeline systems) and varying (in a large number of less powerful distribution systems). According to [2], the electricity overconsumption stemming from the non-optimality of the modes and number of simultaneously operating units occurring during transient processes is quite significant and can reach 14 % of the total PS energy consumption. In the former case, it is due to the large capacity of separate objects and in the latter one, it stems from the large number of less powerful objects. According to the same data, as many as 16 switchovers of pumping units can occur at the main oil PS within one day only. It should be noted that electricity overconsumption at the pumping station also leads to a significant overconsumption in the elements of electrical networks [3]. Electric-driven centrifugal pumps (hereinafter referred to as EDCPs) are devices with a low electrical efficiency, especially in the case of deep submersible EDCPs, for which power losses can reach 65.8 % of the total PS power consumption [3]. Besides, the power losses are significantly affected by such factors as the quality of electricity and availability of uninterrupted power supply, and conducting full-scale experiments on the operating PSs is expensive and often unallowable [4] due to the interruptions in their work caused by the experiments. Therefore, simulation of emergency and operational processes occurring in such objects is in most cases the only possible means of their safe study and of the prediction and implementation of trouble-free energy-saving modes and measures.

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2. Analysis of the recent studies and publications on the problem

In the overwhelming majority of cases, the translational motion of the fluid using pipeline transport is ensured by pumping stations equipped with EDCP, which is usually actuated by an asynchronous (less frequently synchronous) electric drive [3], [4], inextricably linked to the EDCP by a common rigid shaft. Often, the functions of the induction motor (IM) and the centrifugal pump (CP) are performed by one machine – an induction motor with a squirrel-cage rotor, which is also the CP impeller [5]. Therefore, the electromechanical and hydraulic processes in the respective electric elements of PS are indirectly interrelated and interdependent. Due to this, the PS hydraulic load and its power supply system should be considered as a single complex. The analysis of the modes of this complex was set as an objective and was partially solved in [6], whereas the control systems of the complex were developed in [7], [8]. A more detailed consideration of the main internal hydraulic parameters of EDCP and the pipeline was implemented in the mathematical model [9] of the hydraulic load of the PS electric drive. In these studies, the representation of each subsystem of the PS electric complex provide sufficient detail, but they deal with steady state modes only.

The paper [10] focuses on the simulation of static and dynamic modes of the systems for water pumping into the oil reservoirs in oil fields, while the process of sludge transportation and dosing is considered in [11]. In these works, CPs are presented in the form of approximation expressions with one generalized fictitious hydraulic resistance and the power proportional to the cubed rotation frequency of the CP shaft. The simulation of the dynamic modes of frequency controlled asynchronous EDCPs with independent multi-stage water CPs is described in [12], and those with parallel operating CPs are dealt with in [13]. However, these studies used the standard model of the CP from the MATLAB / SIMULINK library, fictitious parametrization of which is carried out based on the CP passport head and flow parameters or those obtained experimentally.

The authors in [14] proposed a mathematical model of the dynamic operating modes of the Iranian irrigation systems. Therein, CP is represented as an approximation polynomial, taking into account the rotation frequency of the impeller, which includes coefficients determined by the CP passport head and flow parameters or those obtained experimentally. A similar approach to CP modelling is also used by the authors of the mathematical model of frequency controlled water supply systems with asynchronous EDCP in [1] for studying the hydraulic modes at the pipeline inlet and outlet and electromagnetic processes in the asynchronous electric drive. A number of papers focus on a separate issue of synthesis and analysis of neural networks for EDCP control. What is common in them is that the mathematical modelling of the electromagnetic subsystem is detailed and that of the hydraulic subsystem is extremely simplified. For example, in [15] , as in the previous cases, the CP is also represented as an approximation polynomial.

In [16], the authors, being the closest to solving the research problem raised in this study, presented a refined CP mathematical model built by applying the principle of electrohydraulic analogy to combine models of its main structural elements. This model makes it possible to obtain the operational characteristics of the CP with a sufficient accuracy, taking into account the impact on them of the rotational speed of the impeller, as well as of the main physical parameters of the working fluid. The application of the principle of electrohydraulic analogy creates preconditions for applying the provisions of the theory of electric circuits for the analysis of hydraulic circles. However, the paper does not mention the possibility of studying transients using the proposed mathematical model.

Thus, the analysis of the researches dealing with the mathematical modelling of the centrifugal hydraulic load of the PS electric drive reveals the lack of a dynamic EDCP model with an appropriate detailing of the electromagnetic and hydraulic subsystems, which therefore does not allow studying the interrelation and interdependence of their parameters.

3. Aim of the research

The research aims at constructing an improved dynamic model of the hydraulic subsystem of EDCP, i.e. the centrifugal hydraulic load of the electric drive of the pump station. This model will enable studying the processes in the main elements of the centrifugal pump, taking into account its internal parameters and the dependence of these parameters on the mode coordinates and physical parameters of the working fluid.

4. Results and their discussion

To solve the set tasks, we will base on the CP mathematical model presented in [16]. This model makes it possible to study the steady state modes both of CP as a whole and of the main elements of its internal structure in sufficient detail, taking into account the effect of the rotational speed of the impeller, as well as of the density and kinematic viscosity of the working fluid. The last parameter allows considering the influence of the working fluid temperature on the parameters of the CP elements and pumping modes, since the change in temperature of the fluid by tens of degrees can lead to a change in the kinematic fluid viscosity by orders of magnitude more. An important distinctive feature of the model presented in [16] from the previous solutions offered by the same authors is that the dissipative mechanical losses in the bearings, gaskets and disk friction of the CP fluid are presented by the non-linear active resistance dependent on the structural parameters of CP and the hydraulic coordinates of its mode.

Applying the mesh-current method, we will form the equation of the CP mathematical model based on its complex equivalent circuit presented in [16]. To do this, we transform the said circuit into a form suitable for writing the equations:

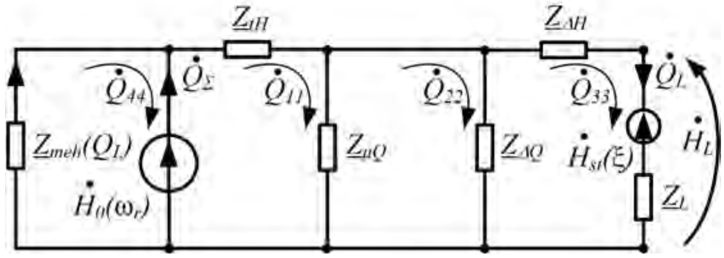

Fig. 1. Transformed equivalent circuit of the centrifugal pump.

In the steady state modes, the equations of the CP mathematical model (Fig. 1) will have the following form:

$$\left(\underline{Z}_{tH}+\underline{Z}_{\mu Q}\right)\dot{Q}_{11}-\underline{Z}_{\mu Q}\,\dot{Q}_{22}-\rho g\,\dot{H}_0\left(\omega_{\mathrm{r}}\right)=0\,; \tag{1}$$

$$\left(\underline{Z}_{\mu Q}+\underline{Z}_{\Delta Q}\right)\dot{Q}_{22}-\underline{Z}_{\mu Q}\,\dot{Q}_{11}-\underline{Z}_{\Delta Q}\,\dot{Q}_{33}=0\,; \tag{2}$$

$$\underline{Z}_{mech}\left(\sqrt{\left(\mathrm{Re}\left(\dot{Q}_{33}\right)\right)^2+\left(\mathrm{Im}\left(\dot{Q}_{33}\right)\right)^2}\right)\dot{Q}_{44}-\rho g\,\dot{H}_0=0\,; \tag{3}$$

$$\underline{Z}_{\Delta Q}\,\dot{Q}_{33}-\underline{Z}_{\Delta Q}\,\dot{Q}_{22}+\rho g\,\dot{H}_L=0\,; \tag{4}$$

$$\mathrm{Re}\left(\dot{H}_L\right)\mathrm{Im}\left(\dot{Q}_{33}\right)-\mathrm{Im}\left(\dot{H}_L\right)\mathrm{Re}\left(\dot{Q}_{33}\right)=0\,; \tag{5}$$

$$Z_L\sqrt{\left(\mathrm{Re}\left(\dot{Q}_{33}\right)\right)^2+\left(\mathrm{Im}\left(\dot{Q}_{33}\right)\right)^2}+\rho g H_{st}(\xi)-\rho g\sqrt{\left(\mathrm{Re}\left(\dot{H}_L\right)\right)^2+\left(\mathrm{Im}\left(\dot{H}_L\right)\right)^2}=0\,, \tag{6}$$

where ω_r and $\omega_{r.nom}$ are the actual and nominal value of the rotation frequency of the CP impeller, respectively; $\dot{H}_0\left(\omega_{\mathrm{r}}\right)=H_{0.\mathrm{nom}}\left(\dfrac{\omega_{\mathrm{r}}}{\omega_{\mathrm{r.nom}}}\right)^2\left(\cos\left(\omega_{\mathrm{r.nom}}t+\Psi_{0CP}\right)+j\sin\left(\omega_{\mathrm{r.nom}}t+\Psi_{0CP}\right)\right)$ is the complex fictitious pressure of the idealized CP [16] (in the steady state modes, the moment of time t and the initial value Y_{0CP} of the rotation angle of the CP impeller, which are part of the equations, are arbitrary); $\dot{Q}_{\Sigma}=\dot{Q}_{11}-\dot{Q}_{44}$ is the complex volumetric flow rate of the idealized CP ([16] and Fig. 1); $H_{st}(\xi)$ is the static anti-pressure of the CP hydraulic load (ξ is any parameter on which it can depend); $\dot{Q}_{22}$ is the complex fictitious flow rate of the CP equivalent circuit; $H_L=\sqrt{\left(\mathrm{Re}\left(\dot{H}_L\right)\right)^2+\left(\mathrm{Im}\left(\dot{H}_L\right)\right)^2}$ is the real pressure head at the CP outlet; $Q_L=Q_{33}=\sqrt{\left(\mathrm{Re}\left(\dot{Q}_{33}\right)\right)^2+\left(\mathrm{Im}\left(\dot{Q}_{33}\right)\right)^2}$ is the real volumetric flow rate at the CP outlet; $\dot{Q}_{mech}=\dot{Q}_{44}$ is the complex fictitious flow rate of the mechanical losses of CP ([16] and Fig. 1) ; $Z_L = R_L$ is the equivalent hydraulic resistance of the pipeline; Y_{0CP} is the initial value of the rotation angle of the CP impeller; $\underline{Z}_{tH} = j\dfrac{\omega_{\mathrm{r}}}{\omega_{\mathrm{r.nom}}}\left(L_{t.nom} + L_{\mu H.nom}\right)$, $\underline{Z}_{\Delta Q} = \dfrac{k_v}{k_{v.nom}} r_{\Delta Q.nom} + j\dfrac{\omega_{\mathrm{r}}}{\omega_{\mathrm{r.nom}}} L_{\Delta Q.nom}$, $\underline{Z}_{mech} = \dfrac{k_v}{k_{v.nom}} r_{meh.nom} + j\dfrac{\omega_{\mathrm{r}}}{\omega_{\mathrm{r.nom}}} L_{meh.nom}$, $\underline{Z}_{\mu Q} = j\dfrac{\omega_{\mathrm{r}}}{\omega_{\mathrm{r.nom}}} L_{\mu Q.nom}$ are complex hydraulic resistances of CP; k_v and $k_{v.nom}$ are the actual and nominal values of the kinematic viscosity of the working fluid, respectively; t is time.

It should be noted that the equation of the CP model, except for the equation (6) describing the pipeline, is formed in the rotary system of d-q coordinates, rigidly connected with the CP impellers [16]. In this coordinate system, the internal parameters of CP do not depend on the impeller rotation angle, and the pressure heads and flow rates are its harmonic (approximately sinusoidal) functions. The equation (5) specifies, in accordance with [16], the collinearity of the representational vectors of the actual pressure head and flow rate at the CP output. The physical content and data on the calculation of the values of the dissipative hydraulic resistance and inductive hydraulic resistance, which make up the above-mentioned total resistances Z_{tH}, $Z_{\mu Q}$, Z_{DQ} and Z_{meh} and are the parameters of the internal structural elements of the CP, are presented in [16]. The dissipative hydraulic resistance depends on the density and kinematic viscosity of the working fluid, while the inductive hydraulic resistance depends on the rotation frequency of the impeller. In dynamic modes, the CP state will be described by the following equations:

$$r_{\mu Q}\dot{q}_{11} + \left(L_t + L_{\mu H} + L_{\mu Q}\right)\frac{d\dot{q}_{11}}{dt} - L_{\mu Q}\frac{d\dot{q}_{22}}{dt} - \dot{h}_0 = 0; \tag{6}$$

$$-r_{\mu Q}\dot{q}_{11} - L_{\mu Q}\frac{d\dot{q}_{11}}{dt} + r_{\Delta Q}\dot{q}_{22} + \left(L_{\mu Q} + L_{\Delta Q}\right)\frac{d\dot{q}_{22}}{dt} - r_{\Delta Q}\dot{q}_{33} - L_{\Delta Q}\frac{d\dot{q}_{33}}{dt} = 0; \tag{7}$$

$$-r_{\Delta Q}\dot{q}_{22} - L_{\Delta Q}\frac{d\dot{q}_{22}}{dt} + \left(r_{\Delta Q} + r_{\Delta H}\right)\dot{q}_{33} + \left(L_{\Delta Q} + L_{\Delta H}\right)\frac{d\dot{q}_{33}}{dt} + \rho g\dot{h}_L = 0; \tag{8}$$

$$r_{meh}\left(q_L\right)\dot{q}_{44} + L_{meh}\frac{d\dot{q}_{44}}{dt} - \rho g\dot{h}_0 = 0; \tag{9}$$

$$h_{Ld}q_{33q} - h_{Lq}q_{33d} = 0; \tag{10}$$

$$r_L q_L + L_L\frac{dq_L}{dt} + \rho g h_{st} - \rho g h_L = 0; \tag{11}$$

$$\omega_r M_{CP} - \rho g h_0 q_\Sigma = 0; \tag{12}$$

$$J_\Sigma\frac{d\omega_r}{dt} = M_D - M_{CP}, \tag{13}$$

where $\dot{q}_{11} = q_{11d}(t) + jq_{11q}(t)$; $\dot{q}_{22} = q_{22d}(t) + jq_{22q}(t)$; $\dot{q}_{33} = q_{33d}(t) + jq_{33q}(t)$; $\dot{q}_{44} = q_{44d}(t) + jq_{44q}(t)$; $\dot{h}_0 = H_{0.\mathrm{nom}}\left(\omega_r(t)/\omega_{\mathrm{r.nom}}\right)^2\left(\cos\left(\omega_{\mathrm{r.nom}}t + \Psi_{0CP}\right) + j\sin\left(\omega_{\mathrm{r.nom}}t + \Psi_{0CP}\right)\right)$; $\dot{q}_\Sigma = \dot{q}_{11} + \dot{q}_{44}$ are the complex fictitious pressure head and volumetric flow rate of the idealized CP, respectively; $h_L = \sqrt{\left(h_{Ld}(t)\right)^2 + \left(h_{Lq}(t)\right)^2}$, $q_L = \sqrt{\left(q_{33d}(t)\right)^2 + \left(q_{33q}(t)\right)^2}$, $h_{st} = h_{st}(t)$ are the actual values of the pressure head and volumetric flow rate at the pipeline outlet, as well as of the static pipeline anti-pressure, respectively; $M_D = M_D(t)$, $M_{CP} = M_{CP}(t)$ are the torque of the electric drive and the braking torque of the pump on a common shaft, respectively; J_Σ is the total moment of inertia of the electric drive and CP.

For ease of use in the systems of computer mathematics (in particular, in MATLAB / SIMULINK), we solve the system of equations (6)–(13) of the model to define the first derivatives. As a result, we finally get:

$$\frac{d\dot{q}_{11}}{dt} = a_{11}\dot{q}_{11} + a_{12}\dot{q}_{22} + a_{13}\dot{q}_{33} + b_1\dot{h}_0; \tag{14}$$

$$\frac{d\dot{q}_{22}}{dt} = a_{21}\dot{q}_{11} + a_{22}\dot{q}_{22} + a_{23}\dot{q}_{33} + b_2\dot{h}_0\,; \tag{15}$$

$$\frac{d\dot{q}_{33}}{dt} = a_{31}\dot{q}_{11} + a_{32}\dot{q}_{22} + a_{33}\dot{q}_{33} + b_3\dot{h}_0\,; \tag{16}$$

$$\frac{d\dot{q}_{44}}{dt} = a_{44}\left(q_L\right)\dot{q}_{44} + b_4\dot{h}_0\,; \tag{17}$$

$$h_{Ld}q_{33q} - h_{Lq}q_{33d} = 0\,; \tag{18}$$

$$\frac{dq_L}{dt} = c_1 q_L + c_2\sqrt{\left(h_{Ld}\left(t\right)\right)^2 + \left(h_{Lq}\left(t\right)\right)^2} + c_3 h_{st}\,; \tag{19}$$

$$\frac{d\omega_r}{dt} = \frac{1}{J_\Sigma}\left(M_D - \rho g H_{0.\text{nom}}\left(\frac{\omega_r\left(t\right)}{\omega_{\text{r.nom}}}\right)^2\sqrt{\left(q_{11d} + q_{44d}\right)^2 + \left(q_{11q} + q_{44q}\right)^2}\right), \tag{20}$$

where $a_{11} = \frac{r_{11}}{L'_{11}} - \frac{r_{21}}{L'_{12}}$; $a_{12} = \frac{r_{22}}{L'_{12}} - \frac{r_{32}}{L'_{13}}$; $a_{13} = \frac{r_{33}}{L'_{13}} - \frac{r_{23}}{L'_{12}}$; $a_{21} = \frac{r_{11}}{L'_{21}} - \frac{r_{21}}{L'_{22}}$; $a_{22} = \frac{r_{22}}{L'_{22}} - \frac{r_{32}}{L'_{23}}$; $a_{23} = \frac{r_{33}}{L'_{23}} - \frac{r_{23}}{L'_{22}}$; $a_{31} = \frac{r_{11}}{L'_{31}} - \frac{r_{21}}{L'_{32}}$; $a_{32} = \frac{r_{22}}{L'_{32}} - \frac{r_{32}}{L'_{33}}$; $a_{33} = \frac{r_{33}}{L'_{33}} - \frac{r_{23}}{L'_{32}}$; $a_{44}\left(t\right) = -\frac{r_{me\tilde{n}h}\left(q_L\right)}{L_{me\tilde{n}h}}$; $c_1 = -\frac{r_L}{L_L}$; $c_2 = \frac{\rho g}{L_L}$; $c_3 = -\frac{\rho g}{L_L}$; $b_1 = -\frac{\rho g}{L'_{11}}$; $b_2 = -\frac{\rho g}{L'_{21}}$; $b_3 = -\frac{\rho g}{L'_{31}}$; $b_4 = \frac{\rho g}{L_{mech}}$, where $r_{11}=r_{12}=R_{\mu Q.nom}$; $r_{22}=r_{23}=R_{\Delta Q.nom}$; $r_{33}=R_{\Delta Q.nom}+R_{\Delta H.nom}$; $L_{11}=L_{t.nom}+L_{\mu H.nom}+L_{\mu Q.nom}$; $L_{12}=L_{21}=L_{\mu Q.nom}$; $L_{23}=L_{32}=L_{\Delta Q.nom}$; $L_{33}= L_{\Delta Q.nom}+L_{\Delta H.nom}$; $L_{22}=L_{\mu Q.nom}+L_{\Delta Q.nom}$, where $L'_{11} = \left(\frac{L_{12}L_{21}}{L_{11}^2 L''} - \frac{1}{L_{11}}\right)^{-1}$; $L'_{22} = L''$; $L'_{33} = \left(\frac{L_{32}L_{23}}{L_{33}^2 L''} - \frac{1}{L_{33}}\right)^{-1}$ $L'_{12} = L'_{21} = \frac{L_{11}L''}{L_{12}}$; $L'_{13} = L'_{31} = \frac{L_{11}L_{33}L''}{L_{12}L_{23}}$; $L'_{23} = L'_{32} = \frac{L_{33}L''}{L_{23}}$, where $L'' = \frac{L_{12}L_{21}}{L_{11}} - L_{22} + \frac{L_{23}L_{32}}{L_{33}}$.

The values of the hydraulic inductances, which are included in the equations (6)–(13) and (14)–(20), are obtained from the corresponding hydraulic inductive resistances at the nominal rotational speed of the CP impeller.

Verification of the model is carried out in the Mathcad system. For the test simulation, 14NDs-N CP was chosen, the nominal parameters of which are given in Table 1. The table presents the relative values of the parameters of the elements of the CP mathematical model obtained using the method described in [16]. The total moment of inertia of the electric drive and centrifugal pump J_Σ=10.5 kg·m^2. The nonlinear dissipative resistance of the mechanical energy losses in CP is often represented in the form of a simplified expression $r_{mech}\left(q_L\right) = \left(H_{CP.nom}/Q_{CP.nom}\right)R_{me\tilde{n}h.nom*}^{\left|q_L\right|/Q_{CP.nom}}$, which can be applied on condition that the operating modes of the centrifugal pump predominantly fall into the range of the operating fluid consumption from 60 to 110 % of the nominal value. Therefore, it is calculated in this case according to the refined algorithm described in detail in [16].

Table 1. Nominal parameters of the 14NDs-N CP and the parameters of the elements of its mathematical model.

H_{nom}, m	Q_{nom}, m^3/h	η_{nom},	n_{nom}, rpm	$P_{hydr\cdot nom}$, kW	$H_{0.nom*}$	$R_{\Delta Q*}$	$L_{\Delta Q*}$	$R_{\Delta H*}$	$L_{\Delta H*}$	L_{t*}	$L_{\mu H*}$	$L_{\mu Q*}$	R_{mech*}	L_{mech*}
45	1260	0.809	980	154	1.302	29.47	9.49	$6.627\cdot10^{-4}$	0.4144	0.00876	0.0352	0.2375	7.180	0.02287

Figures 2–7 show the simulation of the time dependencies of the main real and fictitious coordinates of the operating mode of CP, the power losses in the CP elements and its family of head and flow characteristics at different values of the rotational speed of the impeller.

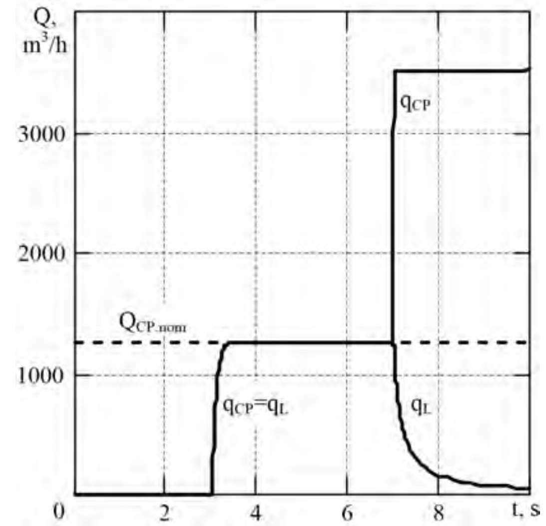

Fig. 2. Real actual and nominal volumetric flow rates of the working fluid at the CP outlet.

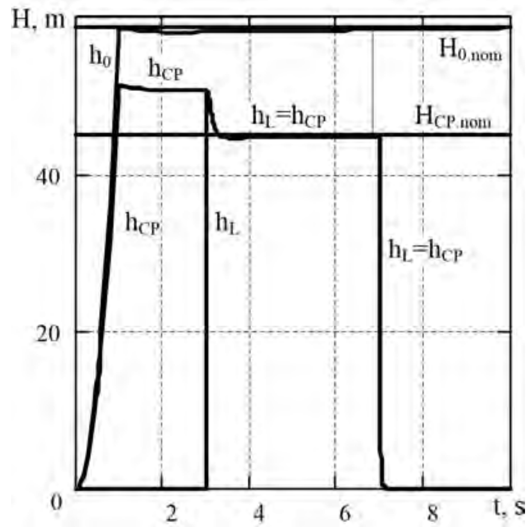

Fig. 3. Real actual pressure heads at the CP outlet and at the pipeline inlet, idealized CP pressure head and the corresponding nominal pressure heads.

The simulation was carried out under the following conditions. The torque of the electric drive M_D=0 for t <0 provided a linear increase in the rotation frequency of the impeller from zero to the nominal value for a time from t = 0 to t = 0.1 s, after which it continued to provide a constant nominal value of the impeller rotational speed. The starting of CP occurred in the idle mode (the pipeline is closed; the actual flow rate of the fluid is equal to zero). At a time t = 3 s, the pipeline valve started to open (the opening time of the valve is 2 seconds). At t = 7 s, a complete pipeline burst was simulated (duration of its formation is 1 s). Figures 2–7 show the simulation of the time dependencies of the main real and fictitious CP mode coordinates and power losses in the elements of the CP, as well as its head and flow characteristics family at different values of the impeller rotational speed.

A series of mathematical experiments, the results of one of which are presented in Figures 2–7, and comparison of their outcomes with the operational characteristics of the corresponding CP confirm the validity of the proposed mathematical model. For the nominal rotational speed of the impeller, the discrepancy between the head and flow characteristics and energy conversion efficiencies obtained on this model and the model of the steady state modes [9] does not exceed 15 % in the flow rate range of 13.3–249 % of the nominal value.

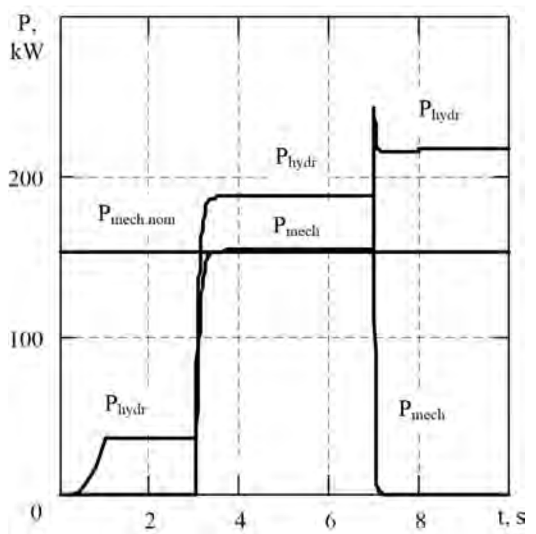

Fig. 4. Useful hydraulic power at the outlet of CP and mechanical power consumed by it on the drive shaft.

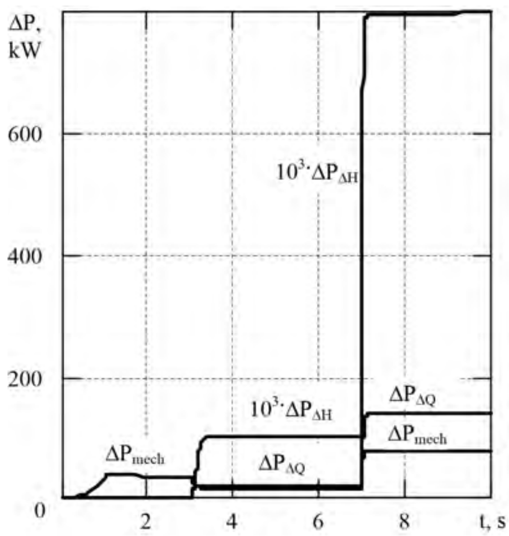

Fig. 5. Dissipative power losses in CP elements: mechanical, in the impeller seals and in the volute diffuser.

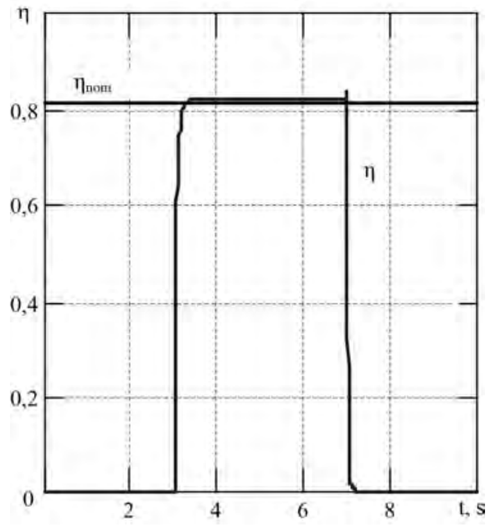

Fig. 6. Nominal and current CP energy conversion efficiency.

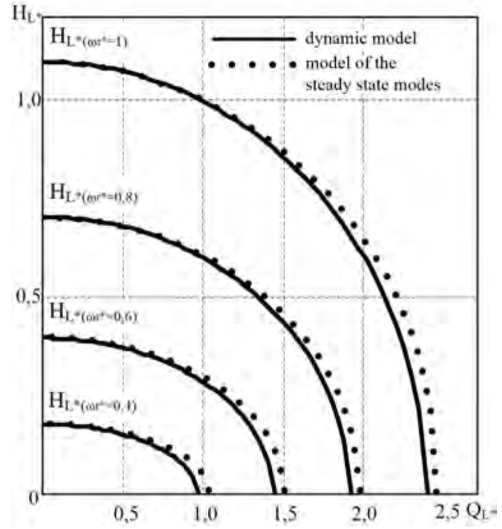

Fig. 7. Family of CP head and flow characteristics at different values of the impeller rotational speed (per-unit).

5. Conclusion

An improved dynamic model of the centrifugal pump unit is proposed, taking into account the influence of the pipeline. The use of the principle of electro-hydrodynamic analogy for the formation of model equations makes it possible to apply the basic provisions of the theory of electric circuits for the analysis of hydraulic circles. This allows considering the interrelated hydraulic and electromechanical subsystems of the pump station as a single complex in sufficient detail. Testing the developed model of the centrifugal hydraulic load of the electric drive of the pumping station by simulating a number of its dynamic modes verified its efficiency. The model can be effectively applied for studying the behaviour of both centrifugal pumps and the processes occurring in their main elements, taking into account its internal parameters and the dependence of these parameters on the fluid's physical parameters and the hydraulic coordinates of the mode. The degree of detail of the model can extend its application also to troubleshoot individual main elements of the pump. The study, the results of which are presented herein, can be continued to refine the presentation of individual elements of the simulated electric centrifugal pump unit, as well as to improve the existing and create new mathematical models of the dynamic modes of electrotechnical complexes of multi-unit pumping stations, taking into account the influence of hydraulic networks.

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Динамічне моделювання відцентрового гідравлічного навантаження електроприводу помпової станції

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Анотація

Проаналізовано сучасні тенденції моделювання відцентрового гідравлічного навантаження електроприводу помпових станцій. Виявлено, що переважно застосовується гранично спрощений підхід до представлення однієї з нерозривно пов'язаних підсистем (гідравлічної чи електромеханічної), що значно звужує можливості комплексного аналізу процесів, які відбуваються в окремих її елементах. Моделі, побудовані на таких засадах, можна ефективно застосовувати для вирішення вузькоспеціалізованих задач, і вони не завжди дають достатнє уявлення про стан і режими усіх елементів підсистем. Частково цю проблему вирішено під час моделювання усталених режимів помпових станцій. Запропоновано вдосконалену динамічну модель відцентрового гідравлічного навантаження електроприводу помпової станції, яка дає змогу досліджувати процеси, що відбуваються в основних елементах відцентрової помпи з урахуванням її внутрішніх параметрів та залежності цих параметрів від фізичних параметрів рідини.

Ключові слова: електропривод; навантаження; помпова станція; відцентрова помпа; гідравлічна мережа.