Simulation of performance characteristics of centrifugal pumps by the electro-hydrodynamic analogy method

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Abstract

The paper specifies the model of oil pipeline centrifugal pump, made on the basis of electro-hydrodynamic analogy method and designed to calculate performance characteristics of the hydraulic machine according to its catalogue data depending on the change of the flow rate and the rotational speed of the pump impeller considering the physical properties of the fluid (its density and kinematic viscosity). The article offers a presentation of mechanical losses as a nonlinear active resistance, the value of which is a quadratic function of the flow rate of a centrifugal pump. This enabled the authors to obtain the formulae for calculating pressure characteristics, power consumed by the driving motor shaft and the machine efficiency depending on the change of flow rate. There is made an investigation of the established operation modes of main pipeline pumps of NM series by constructing a set of performance characteristics with different values of impeller rotational speed. Performance characteristics of the main pump NM-7000-210 have been calculated and the pump has shown a high convergence of calculated and characteristics obtained experimentally. The offered model of a centrifugal pump is easily adaptable to modern interactive computer-based tools (SIMULINK, 20SIM, PSPICE, etc.) assigned for simulation of operation modes of mechatronic engineering systems, which include pumping stations of main oil pipelines. This method of simulation contributes to the theoretical analysis and optimization of the oil pipeline centrifugal pump efficiency.

Key words: *centrifugal pump, electro-hydrodynamic analogy, mechanical losses, performance characteristic, simulation.*

The problem of efficiency of oil and gas equipment is particularly acute in today's rapid growth in energy source prices. One of the main electricity consumers are motor drive pumping units of oil pipelines, equipped with centrifugal pumps (CP). We know that they often operate in suboptimal, usually underloaded modes associated with significant energy consumption. In this regard, there is a practical need to determine the best modes of CPs by constructing their performance characteristics, which set the dependence of head H_R, power consumption N_C and efficiency η on flow rate Q_R based on catalogue and design parameters for pumps with regard to the change in physical properties of the working fluid. The question of calculating of these characteristics with regard to the change in rotational frequency *n* of impellers is of particular relevance due to the widespread introduction of electric drive based on thyristor frequency converters. It will also help simplify the process of analysis and operational control both of fixed and transitional operation modes of the CP. Thus, the purpose of the article is a precise study of the pipeline CP model made on the basis of the electro-hydrodynamic analogy method to calculate its performance characteristics by the catalogue data of a pump depending on the change of the flow rate and the impeller`s rotational frequencies considering the physical properties of the fluid.

The electro-hydrodynamic analogy method is based on the isomorphism of formulae, which describe physical processes in electrical and hydraulic systems. There are widely used the following analogies: electric voltage – pressure (or head) and electric current – mass (or volumetric) flow. The parameter R_H, which characterizes heat losses caused by the friction force of the working fluid, and parameters L_H and C_H, which represent inertial properties and fluid compressibility are hydraulic analogues of passive elements of the alternating current circuit, resistance R_E, inductance L_E and capacitance C_E respectively [1]. This approach enabled us to synthesize "electric" equivalent circuits of hydraulic systems, which can be analyzed using a well-developed electrical apparatus.

Among the models with lumped parameters for analysis of hydraulic process in pipelines there should be mentioned the scientific work [2], which studied these models for transient processes in pipelines, considering the elastic and inertial effects. A detailed definition of such components as resistance, capacitance and inductance for modeling of fluid movement can also be found in [3] where there is made a comparative analysis of models with lumped and distributed parameters based on the study of a transfer function. There is analyzed the one-dimensional differential

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waterhammer equation in the pipeline system and a structural equation of the pipe by the Galerkin method in [4]. This enabled us to express the system of equations through the matrix of mass coefficients, damping and stiffness. A good example of using lumped parameters to study the dynamics of low viscosity fluids in the systems with the piezohydraulic drive is given in [5]. Here hydrotract is divided into *n* unified sections with the length l_i and cross section S_i, where $i=1,n$ is a serial number of the area.

The equivalent circuit of the pipeline for the analysis of compressible fluid movement is shown in Figure 1(a) and it consists of *n* equivalent circuits of individual sections.

The lumped parameters of resistance (damping) R_i, inertia (mass) L_i, and capacitance (stiffness) C_i, are calculated by the design parameters of a pipeline and physical properties of the working fluid and they are shown in this Figure. For the laminar flow regime the equations are the following

$$R_i = \frac{8\pi\rho\nu\, l_i}{S_i^2}, \tag{1}$$

$$L_i = \frac{\rho l_i}{S_i}, \tag{2}$$

$$C_i = \frac{S l_i}{B}, \tag{3}$$

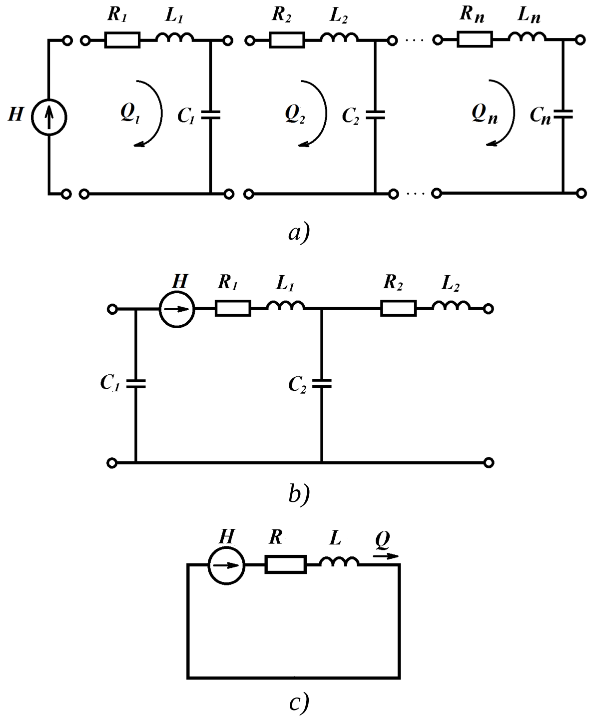
where ν, B, ρ are kinematic viscosity, compressibility and density of the working fluid (for the incompressible fluid $B=\infty$) respectively.

Although models with lumped *R*, *L*, *C* parameters are commonly used for modeling fluid flow in piping systems, we can state a fact of their limited application for study of operation modes of hydraulic machines. This situation is caused by a distinct historical development of classical hydraulics and engineering fluid mechanics.

The theoretical studies on the use of electro-hydrodynamic analogy to analyze modes of performance of hydraulic turbomachinery are of particular importance. There are many scientific works, where the problem is only mentioned without any comprehensive solution, e.g. [6], which indicates the possibility of building and even shows the equivalent circuit of a CP (Figure 1 (b)) without any justification and estimation of its parameters *H, R, L, C).*

We should make a detailed overview of the scientific work [7], which addressed the application of the generalized theory of circuits with lumped parameters for the research of fixed and dynamic modes of a hydraulic turbine (Fransis turbine). A simplified equivalent circuit of the machine is offered in this scientific work (Figure 1 (c)).

However, the proposed model has significant disadvantages as it does not enable us to consider the geometry change of the hydromachine flow, the number of impeller blades and does not provide the solution of the dependence of turbine performance characteristics on the fluid viscosity. Furthermore, the author did not consider physical causes of the change in mode parameters by the harmonic law.



a – pipeline areas in case of analysis of compressible fluid movement;
b – centrifugal pump; c – hydraulic turbine

Figure – 1 Equivalent circuits

The most detailed is a scientific work [8], which offers the model of a CP, which allows calculating head-flow characteristics $H_R = f(Q_R)$ at the full range of actual flow change at a CP output Q_R from zero (idle mode) to maximum (the mode of conditional pipeline breakage) by the catalogue and design parameters of the machine on the basis of electro-hydrodynamic analogy. This model is based on the application of a complex equivalent circuit of a CP (Figure 2) obtained in the rotating coordinate system *d, q,* which is strictly connected with the CP impeller.

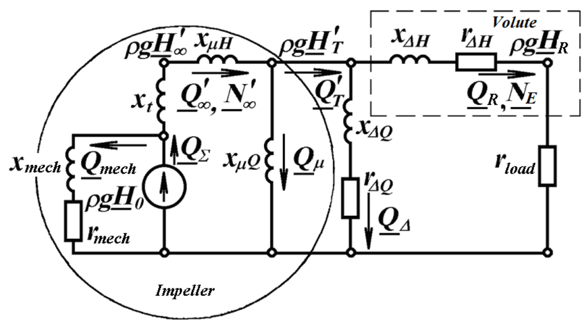


Figure 2 – A complex equivalent circuit of the CP

The elements of the equivalent circuit of a CP are complex parameters – a source $\rho g \underline{H}_0$ with the internal reactance x_t made by the pressure (force) at the outlet of the impeller (an analogue of the electromotive force in electrical circuit) and a set of active resistances *r* and inertial reactances *x*, which symbolize the losses and energy transformations in the unit (the working fluid is considered as incompressible). Here ρ is the density of fluid, g is a gravitational acceleration, H_0 is the head of an idealized CP (lossless and with an infinite number of very thin blades) in an idle mode.

The branches with complex impedances $Z_{\Delta H} = r_{\Delta H} + jx_{\Delta H}$; $Z_{\Delta Q} = r_{\Delta Q} + jx_{\Delta Q}$ and $Z_{mech} = r_{mech} + jx_{mech}$ reflect hydraulic, volumetric and mechanical losses of the CP respectively, while the branches with pure reactances $x_{\mu H}$ and $x_{\mu Q}$ reflect reduction of the pressure and flow rates of the pump caused by a finite number of blades. Here the active resistances $r_{\Delta Q}$ and $r_{\Delta H}$ simulate the irreversible losses (dissipation) of energy into the medium as heat due to viscous friction between the fluid layers, while r_{mech} represents heat losses caused by the disk friction of impellers, bearings and stuffing boxes, and $r_{load} = \frac{\rho g H_R}{Q_R}$ is a hydraulic resistance of a pipeline. In general, the active resistance reflects the viscosity and the density of the working fluid

$$r = 2\rho\nu l \frac{\chi^2}{S^3}, \qquad (4)$$

where χ, S are the wetted perimeter and the cross-sectional area of hydraulic tract of a CP with the length l.

The inertia reactances $x_{\Delta H}$ and $x_{\Delta Q}$ are caused by the inertia forces that counteract the change in CP flows and model vortex processes of internal energy transformations (kinetic energy into potential and vice versa). In general, the inertia reactance depends on the rotational frequency n of the CP`s impeller

$$x = \frac{\pi n}{30} \frac{\rho l}{S}. \qquad (5)$$

Calculation of the equivalent circuit elements and operation modes of a CP is made in the per-unit system, where nominal parameters of the machine are usually chosen as basic parameters. It is enough to set up only two basic parameters – the pressure H_{bas} and flow rate Q_{bas}, which equal nominal values of the discharge head and flow rate of a CP respectively.

$$\left.\begin{aligned} H_{bas} &= H_R^{nom}, \\ Q_{bas} &= Q_R^{nom}. \end{aligned}\right\} \qquad (6)$$

In this case basic power N_{bas} and basic impedance Z_{bas} are the following:

$$\left.\begin{aligned} N_{bas} &= \rho g H_{bas} Q_{bas}, \\ Z_{bas} &= \frac{\rho g H_{bas}}{Q_{bas}}; \end{aligned}\right\} \qquad (7)$$

$$\left.\begin{aligned} H_* &= \frac{H}{H_{bas}},\ Q_* = \frac{Q}{Q_{bas}}, \\ N_* &= \frac{N}{N_{bas}},\ Z_* = \frac{Z}{Z_{bas}}. \end{aligned}\right\} \qquad (8)$$

In particular, dimensionless values of pressure P_* and head H_* are equal in the per-unit system (for the incompressible fluid)

$$P_* = \frac{\rho g H}{\rho g H_{bas}} = H_*.$$

The system of equations of the pressures and flow rates balance in a CP fits the complex equivalent circuit

$$\left.\begin{aligned} &\underline{Q}_\Sigma - \underline{Q}_{mech} - \underline{Q}'_\infty = 0, \\ &\underline{Q}'_\infty - \underline{Q}_\mu - \underline{Q}'_T = 0, \\ &\underline{Q}'_T - \underline{Q}_\Delta - \underline{Q}_R = 0, \\ &\underline{Q}_{mech}(r_{mech} + jx_{mech}) = \rho g \underline{H}_0, \\ &\underline{Q}'_\infty j(x_t + x_{\mu H}) + \underline{Q}_\mu jx_{\mu Q} = \rho g \underline{H}_0, \\ &\underline{Q}_\Delta (r_{\Delta Q} + jx_{\Delta Q}) - \underline{Q}_\mu jx_{\mu Q} = 0, \\ &\underline{Q}_\Delta (r_{\Delta Q} + jx_{\Delta Q}) - \underline{Q}_R (r_{\Delta H} + jx_{\Delta H}) = \rho g \underline{H}_R. \end{aligned}\right\} \qquad (9)$$

Here $\underline{Q}$ are complex parameters of: $\underline{Q}'_\infty$ is flow rate of an idealized CP, $\underline{Q}'_T$ is the flow at the output of the impeller before volute, $\underline{Q}_\Delta$ is the volumetric leakage, $\underline{Q}_\mu$ is the volumetric losses caused by a finite number of blades, $\underline{Q}_{mech}$ is the mechanical losses, and $\underline{Q}_\Sigma$ is the total flow rate of a CP, respectively.

There are usually mounted four NM series double-suction CP (one of which is a reserve) in the pumping station of oil pipelines in Ukraine. The catalogue nominal parameters of these machines are shown in Table 1 [8].

Table 2 shows parameter values of an equivalent circuit of a NM-7000-210 pump in the per-unit system for the working fluid (oil ρ = 800 kg/m^3, ν = 10^{-6} m^2/s) calculated by the method [8].

It is well known that power losses in a CP are conventionally divided into hydraulic, mechanical and volumetric [9]. The scientific work [8] offers the simulator of a CP on the basis of the method of electro-hydrodynamic analogy. The simulator allows calculating its head-flow characteristics $H_R = f(Q_R)$ based on the catalogue and design parameters of the machine. The high accuracy of results (relative error of calculation is typically less than 5% at the operating range of the CP) proves the adequacy of simulation of the pump's volumetric and hydraulic losses using a complex equivalent circuit with constant parameters (Figure 2). This is presented in Figure 3 that shows a good coincidence of calculated and obtained experimentally head-flow characteristics for the NM-7000-210 pump [8]. However, an attempt to include the branch with the permanent hydraulic impedance $Z_{mech} = r_{mech} + jx_{mech}$ into the circuit for simulating mechanical losses was unsuccessful (here $j = \sqrt{-1}$ is the imaginary unit). The errors of consumed

Table 1 – Catalogue nominal parameters of NM series CP

Pump	H_R^{nom}, m	Q_R^{nom}, m^3/h	N_C^{nom}, kW	n^{nom}, min^{-1}	η^{nom}	η_{mech}^{nom}	n_S
NM-1250-260	260	1250	1107	3000	0.80	0.912	70
NM-2500-230	230	2500	1822	3000	0.86	0.964	109
NM-3600-230	230	3600	2593	3000	0.87	0.968	131
NM-5000-210	210	5000	3327	3000	0.86	0.950	165
NM-7000-210	210	7000	4604	3000	0.87	0.956	195
NM-10000-210	210	10000	6430	3000	0.89	0.973	233

Table 2 – Parameters of the equivalent circuit of the NM-7000-210 pump

Pump	H_{0*}	x_{t*}	$x_{\mu H*}$	$x_{\mu Q*}$	$x_{\Delta Q*}$	$r_{\Delta Q*}$	$x_{\Delta H*}$	$r_{\Delta H*}$ *10^{-3}
NM-1250-260	1.387	0.059	0.003	0.256	11.60	24.95	0.440	5.07
NM-2500-230	1.504	0.124	0.055	0.583	0.29	0.04	0.424	2.14
NM-3600-230	1.661	0.274	0.120	1.138	0.44	0.04	0.415	1.32
NM-5000-210	1.759	0.375	0.230	1.899	0.61	0.06	0.407	0.81
NM-7000-210	1.909	0.539	0.254	2.306	0.80	0.05	0.398	0.53
NM-10000-210	2.195	0.786	0.327	2.793	1.00	0.03	0.390	0.31

Table 3 – Calculated parameter values of the main pumps of NM series

Pump	γ_c^{nom}	N_{mech*}^{ID}	N_{mech*}^{nom}	$N_{T*}^{'ID}$	H_{eq*}	x_{eq*}	r_{eq*}
NM-1250-260	0.803	0.23	0.10	0.048	1.118	0.490	0.0052
NM-2500-230	0.899	0.29	0.04	0.037	1.149	0.560	0.0026
NM-3600-230	1.085	0.44	0.04	0.041	1.227	0.707	0.0029
NM-5000-210	1.260	0.61	0.06	0.042	1.323	0.862	0.0040
NM-7000-210	1.380	0.80	0.05	0.045	1.405	0.982	0.0048
NM-10000-210	1.546	1.00	0.03	0.064	1.546	1.173	0.0067

power and efficiency characteristics increased dramatically in low flow modes although they did not exceed 10-15% in the working range of flow rates.

This model has gained widespread use despite this shortcoming. In particular, there should be noted the scientific works [10 – 11] because their authors have successfully investigated the steady operation modes of a CP. It is necessary to mention the works [12 – 14] that have analyzed dynamic modes of pumping stations using the BOND GRAPH models of a CP (made on the basis on the above mentioned model).

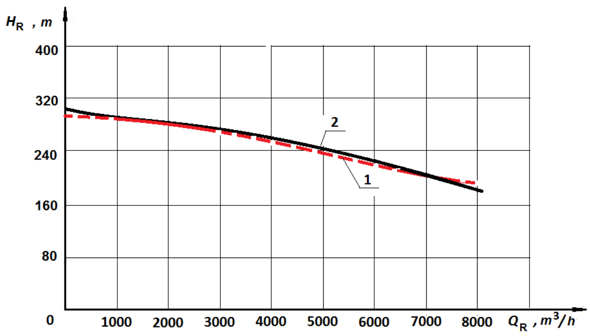


Figure 3 – The head-flow curves of the NM 7000-210 pump, calculated by the model with constant parameters (curve 1) and obtained in experimental way (curve 2) [8]

Obviously, the irreversible mechanical losses are caused by the friction effect and they should be emitted in the form of heat at an active resistance r_{mech}. Therefore a reactive component of impedance is $x_{mech} = 0$. In addition, studies have shown that mechanical losses depend on the flow rate Q_R and their minimal value N_{mech}^{nom} is in nominal operating mode of a CP.

One of the first attempts to introduce mechanical losses as a dissipative element – an active resistance with the variable parameter $r_{mech} = (r_{mech}^{nom})^{Q_{R*}}$ of the equivalent circuit of a CP – has been made in scientific works [15, 16], where r_{mech}^{nom} is the value of mechanical resistance in a nominal mode. However, this presentation of mechanical losses provides with adequate results only in the range (0.6–1.1) of the nominal flow rate Q_R^{nom} and is fundamentally erroneous for low flow modes of a CP.

A complex equivalent circuit (Figure 2) can be simplified to the form shown in Figure 4 by equivalentiation [8]. Calculated dimensionless parameter values of the scheme H_{eq*}, x_{eq*} and r_{eq*} for the NM series of a CP are shown in Table 3.

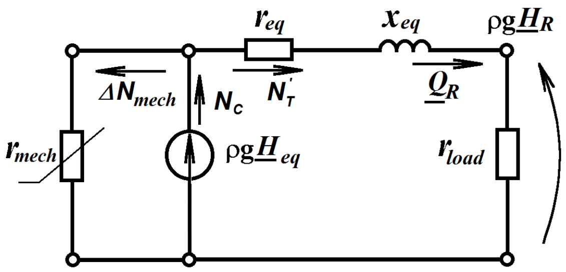


Figure 4 – Equivalent scheme of a CP

The non-linear active resistance of mechanical losses simulation is

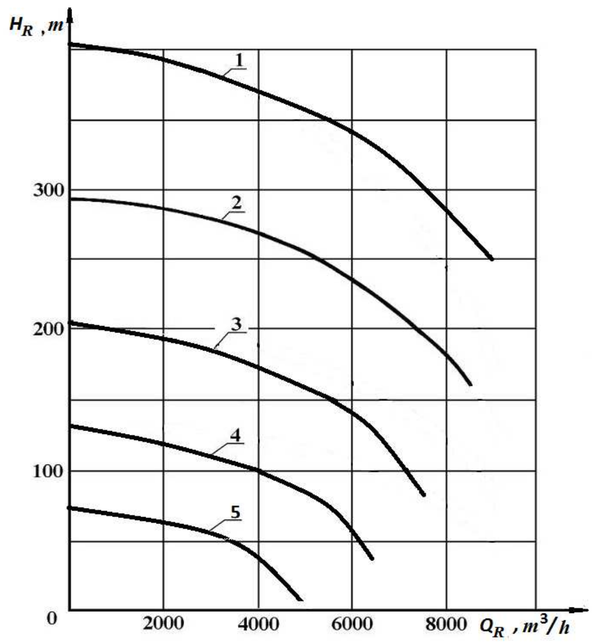
$$r_{mech*} = \frac{H_{eq*}^2}{\Delta N_{mech*}} . \tag{10}$$

Based on the equivalent circuit (Figure 4) with regard to the similarity theory (affinity laws) of a CP [9] we can obtain an expression to re-calculate head-flow characteristic of a CP from one rotation frequency to another with regard to fluid viscosity

$$H_{R*} = k_n^2 \sqrt{(H_{eq*})^2 - (Q_{R*} x_{eq*} / k_n)^2} - Q_{R*} r_{eq*} , \tag{11}$$

where $k_n = \dfrac{n}{n_{nom}}$ is the coefficient of the CP rotational frequency change.

Figure 5 shows a series of head-flow characteristics for different rotational frequencies of the main pump NM-7000-210 calculated on the basis of equation (11).



1 – n=3500 rpm; 2 – n=3000 rpm; 3 – n=2500 rpm; 4 – n=2000 rpm; 5 – n=1500 rpm

Figure 5 – Calculated head-flow characteristics of the pump NM-7000-210 in case of changing rotational frequency *n* of a CP

The balance of pump powers can be represented in the following way based on the complex equivalent circuit of a CP (Figure 2)

$$N_C = N_E + \Delta N_H + \Delta N_Q + \Delta N_{mech}, \tag{12}$$

where N_C is the pump power consumed by the motor drive shaft; N_E is an efficient hydraulic pump power at the output of a CP [8], which is transferred with the working fluid to the pressure pipeline; $\Delta N_H, \Delta N_Q, \Delta N_{mech}$ are the powers of hydraulic, volumetric and mechanical losses in a CP respectively, emitted at impedances of an equivalent circuit $r_{\Delta H} + jr_{\Delta H}$, $r_{\Delta Q} + jr_{\Delta Q}$ and $r_{mech} + jr_{mech}$.

Let's write the equation of internal power N_T' as follows

$$N_T' = N_E + \Delta N_H + \Delta N_Q . \tag{13}$$

On the other hand, according to the equivalent circuit of a CP (Figure 2) this power is calculated as the following product

$$N_T' = \rho g H_T' Q_T' . \tag{14}$$

This approach enables to obtain formulae for calculating the efficiencies of a CP

$$\eta_H = \frac{H_R}{H_T'}, \eta_Q = \frac{Q_R}{Q_T'}, \eta_{mech} = \frac{N_T'}{H_C}. \tag{15}$$

where $\eta_H, \eta_Q, \eta_{mech}$ are hydraulic, volumetric and mechanical efficiencies, respectively. Their product determines the overall efficiency of the pump

$$\eta = \frac{N_E}{N_C} = \eta_H \eta_Q \eta_{mech} . \tag{16}$$

In addition, we obtain an expression of consumed power as the sum of internal power and mechanical losses, based on the expressions (12) and (13), illustrated by the equivalent circuit of a CP (Figure 4)

$$N_C = N_T' + \Delta N_{mech} . \tag{17}$$

The researches have enabled us to introduce the model of mechanical power losses in a CP in the following form (Figure 6)

$$\Delta N_{mech} = \Delta N_{mech}^{const} + \Delta N_{mech}^{var} . \tag{18}$$

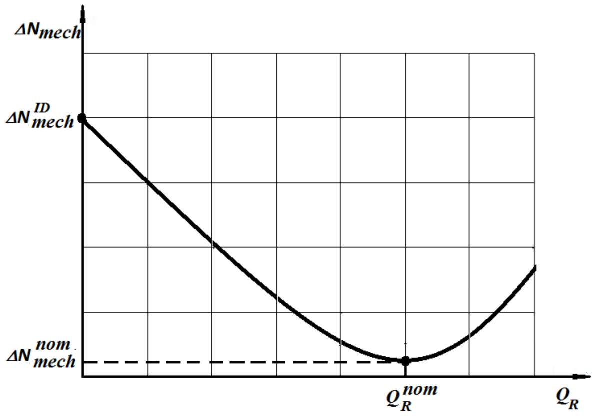


Figure 6 – Dependence of mechanical power losses ΔN_{mech} on the flow rate Q_R

The permanent (at a constant rotational speed of the impeller) component of these losses ΔN_{mech}^{const} shows irreversible dissipative power losses of the disk friction, friction in bearings and friction of a drive shaft seal, independent from the flow regime. This component is equal to the value of mechanical power losses in nominal operation mode of a CP N_{mech}^{nom}, which is typically less than 5–7% of the total consumed power of a CP and is defined in a per-unit system considering the affinity laws for the CP [2] as:

$$\Delta N_{mech*}^{const} = \Delta N_{mech*}^{nom} = \frac{k_n^3\left(1-\eta_{mech}^{nom}\right)}{\eta^{nom}} . \quad (19)$$

Here η^{nom} and η_{mech}^{nom} are the values of the overall and mechanical efficiencies of a CP in a nominal operation mode.

In addition there are variable power losses $\Delta N_{mech}^{\mathrm{var}}$ caused by deviation of the operation mode from the nominal one, accompanied by fluid hammer with the working surface of the blade. These losses can also be related to mechanical losses because they illustrate the heat dissipation in a CP. They are represented as a quadratic function

$$\Delta N_{mech*}^{\mathrm{var}} = \left(\Delta N_{mech*}^{ID} - \Delta N_{mech*}^{nom}\right)\left(\frac{Q_{R*}}{k_n}-1\right)^2 . \quad (20)$$

Here ΔN_{mech*}^{ID} is a relative value of mechanical power losses in an idle mode and can be calculated by the scalar model of a CP [8]

$$\Delta N_{mech*}^{ID} = \frac{k_n^3\left(1-\gamma_c^{nom} ctg\gamma_c^{nom}\right)}{\eta^{ном}} - N_{T*}^{'ID} , \quad (21)$$

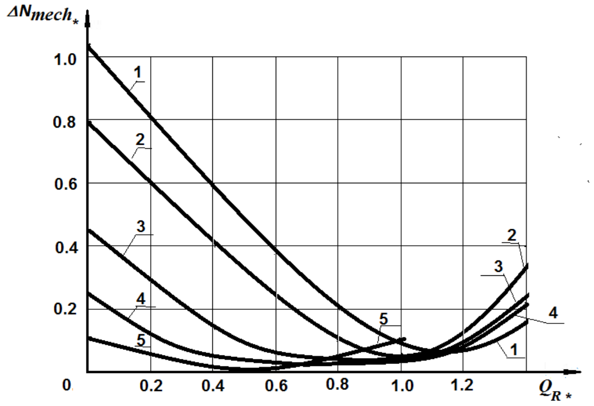
where γ_c^{nom} is a nominal load angle of the pump, associated with the specific speed of a CP n_s in the first approximation by the expression [8]

$$\gamma_c^{nom} \approx 0.475\left(1+\frac{n_s}{100}\right) ; \quad (22)$$

$N_T^{'ID}$ is the value of internal power of a CP in an idle mode, determined by the solution of equations (9) for a given rotational frequency *n* of a CP in case of $Q_R = 0$.

Obviously, the shock-free entering of the fluid in the impeller of a CP $N_{mech}^{\mathrm{var}} = 0$ occurs only in the nominal mode of operation. This approach is caused by the fact that the change of pressure losses of a CP in case of deviation from the shockless mode increases almost linearly [9], determining the corresponding change of power losses according the parabolic law.

The calculated values of parameters γ_c^{nom}, ΔN_{mech*}^{ID} and $\Delta N_{T*}^{'ID}$ in the per-unit system are shown in Table 3 where nominal parameters are adopted for pipeline pumps of NM series. Figure 7 shows the dependence of mechanical power losses on the flow rate for NM-7000-210 pump by changing the rotational frequency of a CP.



1 – n=3500 rpm; 2 – n=3000 rpm; 3 – n=2500 rpm; 4 – n=2000 rpm; 5 – n=1500 rpm.

Figure 7 – The dependence of mechanical power losses ΔN_{mech} on the flow Q_R for NM-7000-210 in case of changing rotational frequency *n* of a CP

The study of expressions (16) – (21) enabled us to calculate the performance characteristics of the consumed power and the efficiency of a CP. Figures 8 and 9 in particular show good coincidence of the calculated and experimentally obtained above-mentioned characteristics for the main pump NM-7000-210. In addition, these expressions allowed analytical calculation of these characteristics in case of changing the rotational frequency of a CP impeller. Their results are shown in Figures 10 and 11.

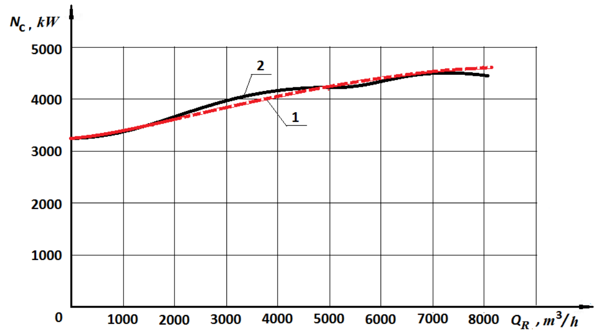


Figure 8 – The dependence of characteristics of consumed power N_C on the flow Q_R of the main pump NM 7000-210 calculated by the presented model (curve 1) and experimentally obtained (curve 2) [17]

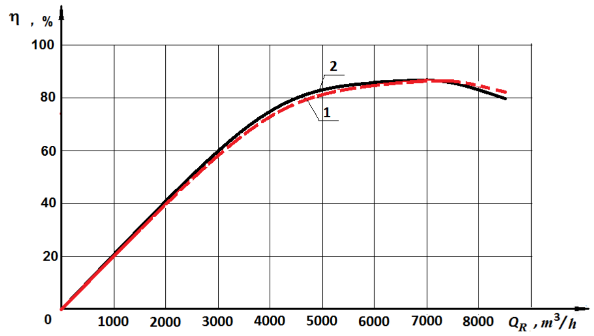
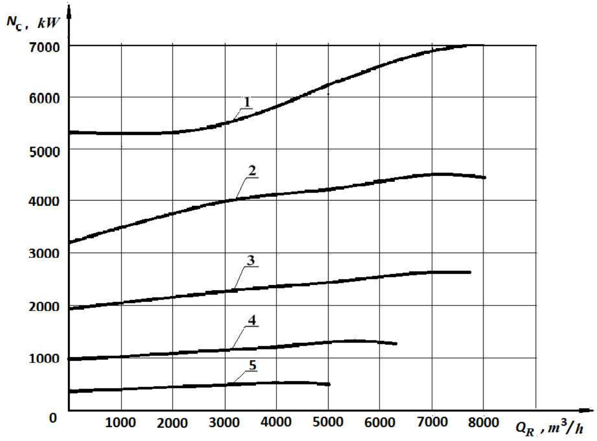
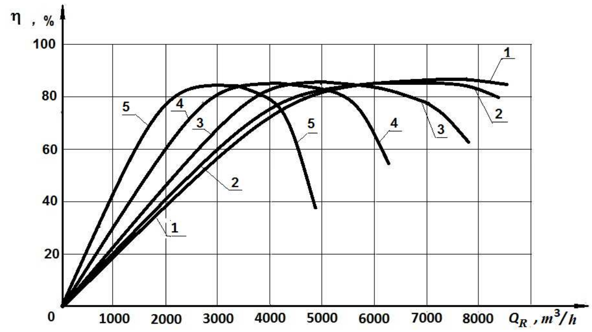


Figure 9 – The dependence of characteristics of the overall efficiency η on the flow Q_R of the main pump NM 7000-210 calculated by the presented model (curve 1) and results obtained experimentally (curve 2) [17]



1 – n=3500 rpm; 2 – n=3000 rpm; 3 – n=2500 rpm; 4 – n=2000 rpm; 5 – n=1500 rpm.

Figure 10 – The calculated power characteristics of the NM-7000-210 pump for impeller rotational frequencies *n*



1 – n=3500 rpm; 2 – n=3000 rpm; 3 – n=2500 rpm; 4 – n=2000 rpm; 5 – n=1500 rpm

Figure 11 – Calculated characteristics of the NM-7000-210 pump overall efficiency for impeller rotational frequencies *n*

Conclusions

There is offered to use the method of electro-hydrodynamic analogy for presentation mechanical losses as a nonlinear active resistance, the value of which is a quadratic function of a CP required flow rate. This approach enabled us to obtain analytical formulae for calculating characteristics of head, power and efficiency of a CP from the flow rate by its catalogue data taking into account changes in rotational frequency *n* and the properties of the working fluid contributing to the increase of the pump efficiency. The high convergence of calculated and obtained experimentally performance characteristics of the main pump NM-7000-210 is shown in the paper. The integrated model of a CP is easily adaptable to modern computer-based interactive tools (SIMULINK, 20SIM, PSPICE, etc.), assigned for simulation of operation modes of mechatronic engineering systems, which include pumping stations of main oil pipelines.

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Моделювання характеристик відцентрових насосів за методом електрогідродинамічних аналогій

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Уточнено модель відцентрового насоса магістральних нафтопроводів, яка створена на основі методу електрогідродинамічної аналогії та призначена для розрахунку робочих характеристик гідравлічної машини за її каталоговими даними залежно від зміни витратного навантаження і частоти обертання робочого колеса насоса з урахуванням фізичних властивостей рідини (густина, кінематична в’язкість).

Запропоновано представлення механічних втрат у вигляді нелінійного активного опору, величина якого є квадратичною функцією витратного навантаження відцентрового насоса. Це дало змогу отримати формули для розрахунку характеристик напору, споживаної з валу приводного двигуна потужності та коефіцієнта корисної дії машини залежно від зміни витрати.

Проведено дослідження усталених режимів роботи магістральних насосів серії НМ шляхом побудови сімейства робочих характеристик за різних значень частоти обертання робочого колеса насоса. Виконано розрахунок робочих характеристик магістрального насоса НМ-7000-210, який показав високу збіжність розрахункових та експериментальних характеристик.

Запропонована модель відцентрового насосу легко адаптується до сучасних комп’ютерно-орієнтованих інтерактивних інструментів (SIMULINK, 20SIM, PSPICE тощо), призначених для моделювання режимів роботи мехатронних технічних систем, до яких відносяться насосні станції магістральних нафтопроводів. Такий спосіб моделювання відкриває шлях до теоретичного аналізу та оптимізації ефективності роботи відцентрового насосу магістральних нафтопроводів.

Ключові слова: *відцентровий насос, електрогідродинамічна аналогія, механічні втрати, моделювання, робоча характеристика*.