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Modeling of Hydraulic Load of Electric Drive in Electrical Complex of Pumping Station

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Abstract

The paper analyses the contemporary state of the electric drive hydraulic load modelling in the pumping stations' electrotechnical complex applications. It was found that in the vast majority of cases, mathematical models do not allow taking into account the specificities of fluid pumping and its consumption at the same time with a balanced degree of detail. The studies conducted provide sufficient ground for making a conclusion that when modelling the electric drive operation, the centrifugal hydraulic load cannot be presented in a general case as the resistant torque with the fan mechanical characteristic. It was shown that to present such hydraulic load of the electric drive correctly, one need to use the mathematical models that simultaneously account for the effect of the pump impeller rotation speed, fluid viscosity and hydraulic network's spatial structure on both the fluid's pumping modes and the modes of its consumption. A complete mathematical model of the hydraulic load of the pumping station's electric drive in steady-state modes was proposed, which takes into account both the internal parameters of the centrifugal pump and the spatial distribution of the pipeline.

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

1. Definition of the research problem

Modelling of the hydraulic subsystem and its components as the electric drive load for pumping station's electrotechnical complex applications is a separate research area, in the scope of which a lot of studies have been carried out. The structure of the model, its degree of detail, modelling method and equations used depend on the subsystem structure and tasks that the model is required to solve. For the analysis of the modes of pumping station's electrotechnical complexes, the hydraulic subsystem needs to be presented in a way that is, on the one hand, easy to use and, on the other, correctly reflects its main properties. Therefore, it is considered viable to rely, first of all, on those available models that are easily adaptable to the electric circuit theory applications. Such a choice of models will make it possible to take into consideration the effect of their internal parameters and to apply a formalized approach.

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

The hydraulic network, which conventionally consists of pipelines, isolation valves, head and flow rate regulation devices, consumers, etc., is a complex subsystem with spatially distributed parameters, and its mathematical modelling is a separate scientific problem. Selection of this or that mathematical model relies on the type and end use of the hydraulic network or its model. A special attention is paid to the study and modelling of pipelines and their networks with a certain degree of detail [1][4]. In these and many other studies, the pipeline is considered separately from the hydraulic fluid source. A formal source of the hydraulic fluid in most pipelines, which is centrifugal pumps, is also modelled in isolation from them, specifically in [5][8]. Much fewer studies (for instance, [9]–[10]), deal with centrifugal pump modelling in inseparable connection with the pipeline, and only some of the researches [11] are based on the integral composition of the electrical and hydraulic subsystems.

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The analysis of the above-mentioned and many other solutions suggests that they can be classified into three groups:

1) phenomenological, regression, empirical, simulation, transfer function models, which are completely or partially devoid of physical sense;

2) models with a detailed description of the physical characteristics of the motor, but with a simplified presentation of the turbo generator units not taking into account their design (the significant majority of the considered papers);

3) specialized mathematical models of the frequency-controlled electric drive units with a detailed mathematical description of the physics of the electromechanical subsystem and superficial description of the hydraulic subsystem.

Therefore, the solid majority of the mathematical models do not enable taking into consideration the specific features of fluid pumping and consumption processes concurrently with a balanced degree of detail.

3. Aim of the research

The study aims at substantiating and building a model of the hydraulic load of the pumping station electric drive adapted for the analysis of steady-state modes of the electrotechnical complex.

4. Results and their discussion

To solve the above-set tasks, let us analyse the mathematical model [10] of the pump taking into consideration the spatial distribution of the pipeline and fluid consumers which was formed on the basis of the diagram presented in Fig. 1.

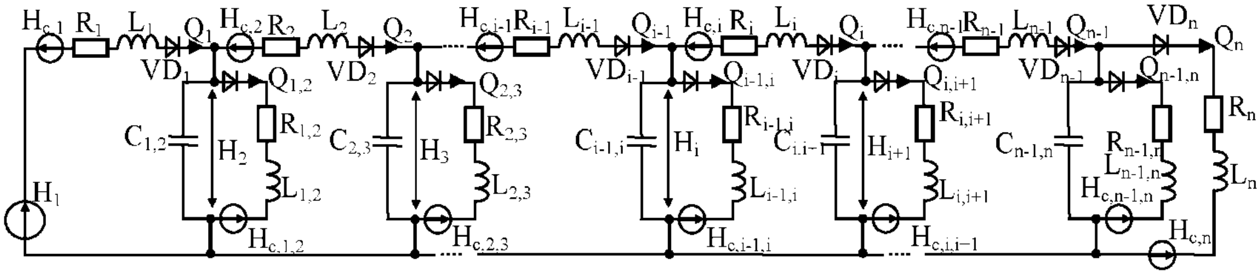

Fig. 1. Equivalent circuit [10] of the pump with the spatial distribution of the pipeline and fluid consumers taken into consideration

According to [10], generally at each pipeline section there occur head losses at the internal dissipative hydraulic resistances ***Ri*** connected in series in the equivalent circuit, which are proportional to the squared volume fluid flow rate. The static counter heads H_{Ci} of the sections characterize the geodetic differential heights of their start and end. A gradual reduction of the flow rate along the pipeline due to the losses or consumption of the fluid is accounted for by introducing parallel-connected dissipative hydraulic resistances $R_{i,i+1}$ into the equivalent circuit. The static counter heads $H_{Ci,i+1}$ are descriptive of the geodetic differential heights of the pipeline braches and fluid consumption points. In addition, the equivalent circuit comprises hydraulic inductances taking into account the impossibility of an instant change of the fluid velocity, hydraulic capacitances accounting for the impossibility of an instant change of the head and hydraulic diodes that provide for adequate modelling of the hydraulic fluid flow direction. The major drawback of this model is the utterly simplified representation of its pumping component in the form of one hydraulic head H_1, which significantly lowers the accuracy of modelling the modes, especially in case of the variable rotation speed of the centrifugal pump. Let us distinguish this problem and consider it later. In the meantime, the focus will be on the pipeline and consumers.

The flow rate of high-capacity pumping stations fairly slowly changes over time, except for the cases of starting or shutting down the equipment and in emergencies. For instance, according to the data in the water consumption curve for the water supply network [12], the highest speed of increase or drop in consumption does not exceed 0.1%/s. Due to this, such modes can be regarded as quasi steady-state ones. It is in these modes that the major volumes of electrical power are consumed and its major losses are sustained, and it is for them that the application of energy-saving structural and circuit design solutions will produce the most significant effect.

In the steady-state modes, all the hydraulic inductances in the equivalent circuit become elements with zero resistance, hydraulic diodes turn into elements with zero or infinite resistance, depending on the direction of the fluid flow, and hydraulic capacities become elements with infinite resistance. Therefore, part of the equivalent circuit that corresponds to the pipeline and consumers is transformed into the circuit presented below:

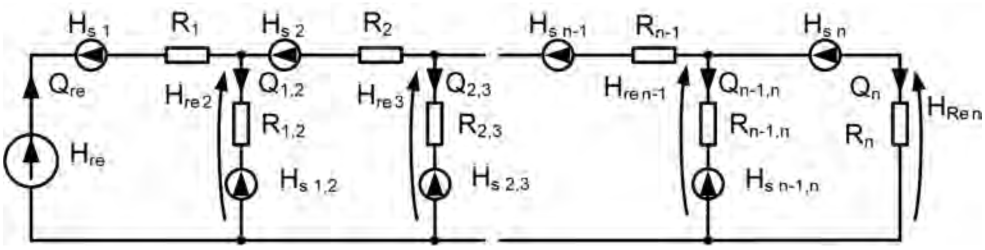

Fig. 2. Equivalent circuit of the pipeline and consumers in steady-state modes

In this figure, the hydraulic resistance R_i denotes the fluid consumer at the end of the pipeline, $H_{s\,i}$ are static counter heads of the pipeline sections; the hydraulic resistances $R_{i,i+1}$ are denote losses or equivalent hydraulic resistances of the pipeline branches, $H_{s\,i,i+1}$ are static counter heads of the pipeline branches and consumption points, $H_{re\,i}$, $Q_{re\,i}$, Q_i are other resistances and losses. It should be notes that if the $i,(i+1)^{th}$ branch in the scheme simulates the fluid losses, there is zero static counter head in this branch.

Having taken into consideration the effect of the hydraulic fluid viscosity k_v on the dissipative hydraulic resistance in accordance with [5] and having assumed the rated head and centrifugal pump flow rate to be the base values, we receive the equation of the mathematical model of the pipeline and consumers in steady-state modes, except for the viscosity k_{v*}, in which case we will omit the asterisk * in relatives units to keep the records concise:

$$H_{re} - H_{s\ 1} - H_{s\ 1,2} - k_{v*}\left(R_1 Q_{\ddot{A}}^2 - R_{1,2}\left(Q_{re} - Q_2\right)^2\right) = 0\,, \tag{1}$$

$$H_{s\ 1,2} - H_{s\ 2} - H_{s\ 2,3} - k_{v*}\left[R_{1,2}\left(Q_{re} - Q_2\right)^2 - R_2 Q_2^2 - R_{2,3}\left(Q_2 - Q_3\right)^2\right] = 0\,; \tag{2}$$

$$\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots; \qquad (\ldots.)$$

$$H_{s\ n-2,n-1} - H_{s\ n-1} - H_{s\ n-1,n} - k_{v*}\left[R_{n-2,n-1}\left(Q_{n-2} - Q_{n-1}\right)^2 - R_{n-1}Q_{n-1}^2 - R_{n-1,n}\left(Q_{n-1} - Q_n\right)^2\right] = 0\,; \tag{3}$$

$$H_{s\ n-1,n} - H_{s\ n} - k_{v*}\left[R_{n-1,n}\left(Q_{n-1} - Q_n\right)^2 - R_n Q_n^2\right] = 0\,; \tag{4}$$

Instead of the utterly simplified representation of the pumping component (H_{re}, Fig. 2), we will assume the solutions presented in [5] and [8]. After reducing the number of equations of the model [5] and taking into account the throttle and bypass changing the internal parameters of the pump regardless of its external hydraulic load, we obtain the improved equivalent circuit of the centrifugal pump (Fig. 3).

On the basis of this circuit, the equations of the centrifugal pump mathematical model are written as follows:

$$\left(\underline{Z}_{0e} + \underline{Z}_{0e}\right)\dot{Q}_{\Sigma} - \underline{Z}_{0e}\dot{Q}_{\Sigma} - \underline{Z}_{0e}\dot{Q}_{cp} - \dot{H}_0 = 0\,, \tag{5}$$

$$-\underline{Z}_{0e}\dot{Q}_{\Sigma} + \left(\underline{Z}_{0e} + \underline{Z}_{2e}\right)\dot{Q}_{cp} + \dot{H}_{cp} = 0\,; \tag{6}$$

$$H_0 - H_{0nom}\omega_{\mathrm{r}*}^2 = 0\,; \tag{7}$$

$$Q_{cpd} - Q_{cp}\cos(\Psi_{0QHcp}) = 0\,; \tag{8}$$

$$Q_{cpd}H_{cpq} - Q_{cpq}H_{cpd} = 0\,; \tag{9}$$

$$H_{cp} - R_{dr}Q_{re} - H_{re} = 0\,; \tag{10}$$

$$Q_{cp} - H_{cp} / R_{bp} - Q_{re} = 0\,, \tag{11}$$

where ω_{r*} is the relative rotation speed of the centrifugal pump impeller;

Q_{re}, H_{re} are the actual volume flow rate of the fluid and actual head with the throttle and bypass of the real centrifugal pump taken into consideration, respectively;

R_{dr}, R_{bp} are equivalent dissipative hydraulic resistances of the throttle and bypass, which are the functions of the isolation valve rotation angle and, in the end, depend on the actual volume flow rate and head;

$\sqrt{Q_{cpd}^2 + Q_{cpq}^2} = Q_{cp}$ is the actual volume flow rate of the fluid of the centrifugal pump without the throttle and bypass;

$\sqrt{H_{cpd}^2 + H_{cpq}^2} = H_{cp}$ is the actual head of the centrifugal pump without the throttle and bypass;

$\underline{Z}_{0e}$, $\underline{Z}_{1e}$, $\underline{Z}_{2e}$ are the equivalent complex hydraulic resistances of the centrifugal pump defined by the hydraulic fluid viscosity and impeller rotation speed, namely:

$$\underline{Z}_{0e} = \omega_{r*} x_{\mu Qnom} \left(k_{v*} r_{i\,nom} + j\omega_{r*} x_{i\,nom}\right)(k_{v*} r_{\Delta Qnom} + j\omega_{r*} x_{\Delta Qnom}) / ((k_{v*} r_{\Delta Qnom} + j\omega_{r*} \left(x_{\mu Qnom} + x_{\Delta Qnom}\right))(x_{i\,nom} + x_{tnom} +$$
$$+ x_{\mu Hnom} + x_{\mu Qnom} (k_{v*} r_{\Delta Qnom} + j\omega_{r*} x_{\Delta Qnom}) / (k_{v*} r_{\Delta Qnom} + j\omega_{r*} \left(x_{\Delta Qnom} + x_{\mu Qnom}\right))) - jk_{v*} r_{i\,nom});$$

$$\underline{Z}_{1e} = \left(\omega_{r*} \left(x_{tnom} + x_{\mu Hnom}\right)\left(k_{v*} r_{i\,nom} + j\omega_{r*} x_{i\,nom}\right)\right) \Big/ (\omega_{r*} (x_{i\,nom} + x_{tnom} + x_{\mu Hnom} + x_{\mu Qnom} \left(k_{v*} r_{\Delta Qnom} + j\omega_{r*} x_{\Delta Qnom}\right) /$$
$$/ k_{v*} r_{\Delta Qnom} + j\omega_{r*} \left(x_{\Delta Qnom} + x_{\mu Qnom}\right)) - jk_{v*} r_{i\,nom});$$

$$\underline{Z}_{2e} = k_{v*} r_{\Delta Hnom} + j\omega_{r*} x_{\Delta Hnom} + \omega_{r*}^2 x_{\mu Qnom} \left(x_{tnom} + x_{\mu Hnom}\right)(\omega_{r*} x_{\Delta Qnom} - jk_{v*} r_{\Delta Qnom}) / ((\omega_{r*} \left(x_{\Delta Qnom} + x_{\mu Qnom}\right) - jk_{v*} \times$$
$$\times r_{\Delta Qnom})\omega_{r*} (x_{tnom} + x_{\mu Hnom} + x_{\mu Qnom} \left(k_{v*} r_{\Delta Qnom} + j\omega_{r*} x_{\Delta Qnom}\right) / (k_{v*} r_{\Delta Qnom} + j\omega_{r*} \left(x_{\mu Qnom} + x_{\Delta Qnom}\right))) - jk_{v*} r_{i\,nom}).$$

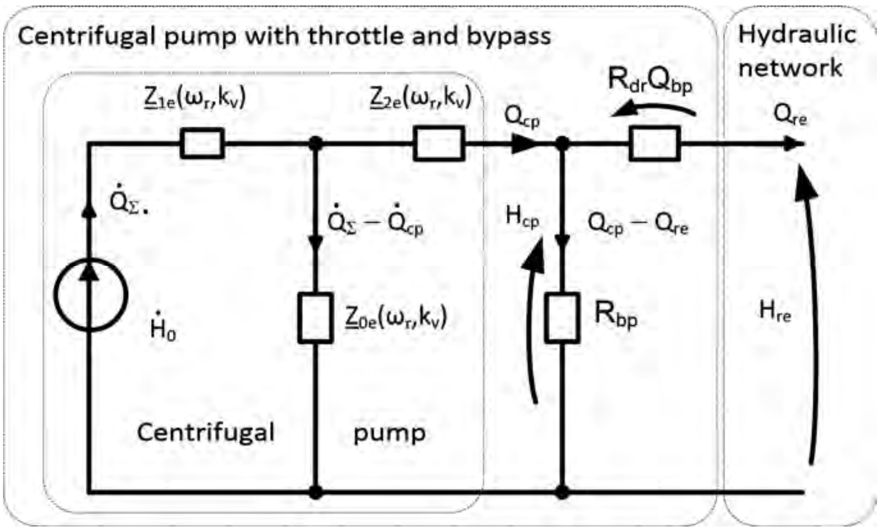

Fig. 3. Equivalent circuit of the centrifugal pump in steady-state modes

The active and inductive hydraulic resistances in the rated duty included in the above-adduced expressions for the complex hydraulic resistances are calculated using the technique presented in detail in [5]. The authors also cite the values of these resistances calculated for a variety of centrifugal pumps that operate at the pumping stations of the oil trunk pipelines.

Fig. 4 presents the set of head/flow curves of the centrifugal pump 14 NDsN for various rotation frequency of the impeller. Fig. 5 shows three variants of the head/flow curve of this same pump: 1 – for the constant hydraulic resistance of the external hydraulic network, and two limit variants: 2 – for the constant rated volume flow rate at the pump outlet, 3 – for the constant rated head at the pump outlet. The calculations were based on the condition that R_{dr} = 0 and R_{bp} = ∞. The rated values of the centrifugal pump 14 NDsN and parameters of its equivalent circuit's elements (active and inductive hydraulic resistances) are presented in [5].

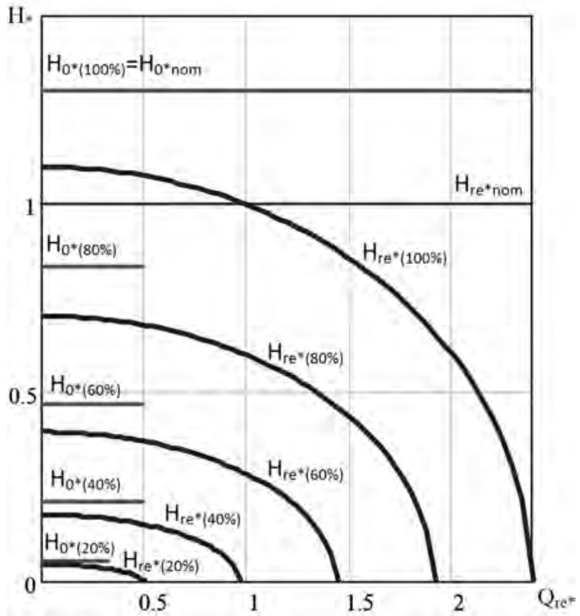

Fig. 4. Set of static head/flow curves of the centrifugal pump 14 NDsN for various (%) impeller rotation frequency

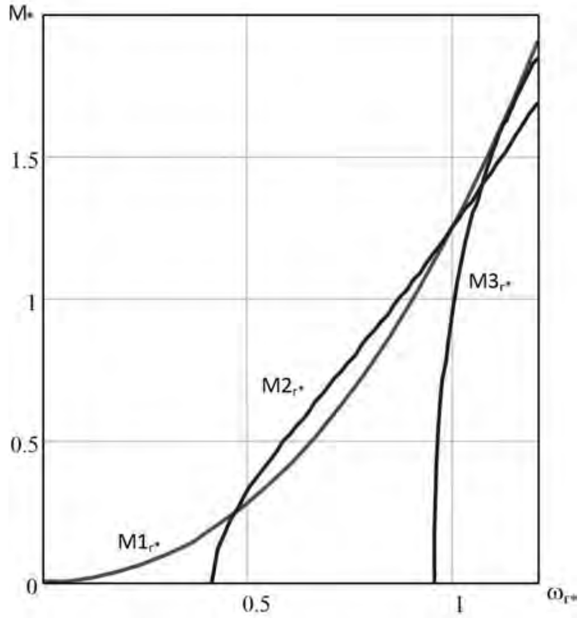

Fig. 5. Static speed/load curves of the centrifugal pump 14 NDsN for various external hydraulic loads

Fig. 5 shows that the fan mechanical characteristic of the centrifugal pump denoted as $M1_{r*}$, which is typically used for most cases of modelling its electric drive operation, is only a partial (subject to the constant hydraulic resistance of the external hydraulic network) case out of many possible ones. The composition of the models of the centrifugal pump (Fig. 3) and hydraulic network (Fig. 2) results in a complete mathematical model of the hydraulic load of the pumping station's electric drive in the steady-state modes taking into account both the internal parameters of the centrifugal pump and the spatial distribution of the pipeline. The model consists of the equations (1)–(4) and (5)–(11). It should be noted that the effect of the fluid's temperature conditions on its pumping modes can also be taken into consideration, as the viscosity factor of the fluid depends on its temperature. The equivalent circuit corresponding to this model is shown in Fig. 6.

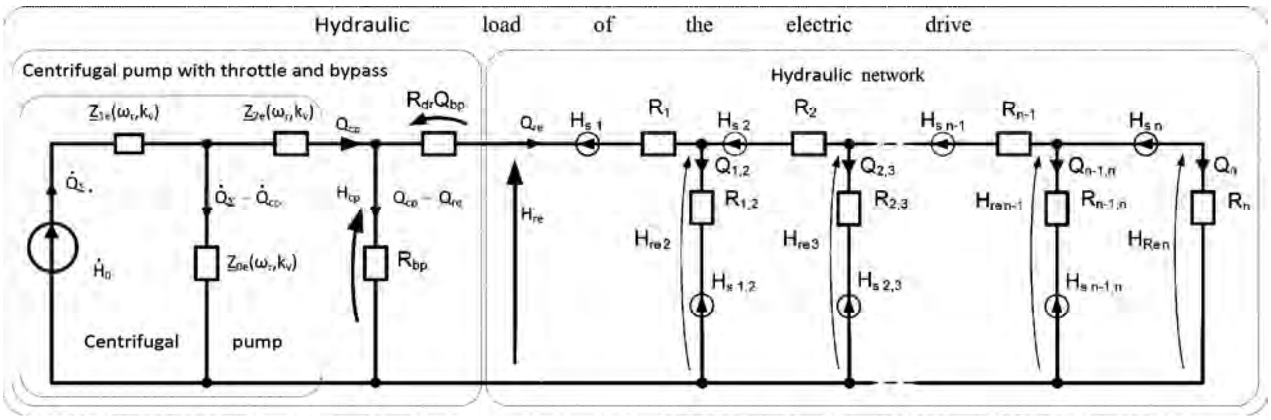

Fig. 6. Equivalent circuit of the hydraulic load of the pumping station's electric drive in steady-state modes

5. Conclusion

The results of the study provide ground for making a conclusion that, when modelling the electric drive operation, the centrifugal hydraulic load cannot be presented in a general case as the resistant torque with the fan mechanical characteristic. In order to present the hydraulic load of the electric drive within the pumping station's electrotechnical complex correctly, it is necessary to use the mathematical models that simultaneously account for the effect of the pump impeller rotation speed, fluid viscosity and hydraulic network's spatial structure on both the fluid pumping modes and the modes of its consumption.

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Моделювання гідравлічного навантаження електроприводу у складі електротехнічного комплексу помпової станції

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Анотація

Проаналізований сучасний стан моделювання гідравлічного навантаження електроприводу у складі електротехнічного комплексу помпових станцій. Виявлено, що більшість математичних моделей не дає можливості зі збалансованим ступенем деталізації враховувати особливості процесів помпування та споживання рідини одночасно. Проведені дослідження дають підстави зробити висновок про неможливість представлення у загальному випаду відцентрового гідравлічного навантаження у вигляді моменту опору з "вентиляторною" механічною характеристикою під час моделювання роботи електроприводу. Показано, що для коректного відображення такого гідравлічного навантаження електроприводу необхідно застосовувати такі математичні моделі, які одночасно враховують вплив кутової швидкості обертання робочого колеса помпи, в'язкості рідини та просторової будови гідравлічної мережі як на режими помпування рідини, так і на режими її споживання. Запропоновано повну математичну модель гідравлічного навантаження електроприводу помпової станції в усталених режимах з урахуванням як внутрішніх параметрів відцентрової помпи, так і просторового розподілу трубопроводу.

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