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INCREASING THE LEVEL OF ENVIRONMENTAL SAFETY OF PUMPING UNITS BY ENERGY SAVING DURING REGULATION

С. Сурков, О. Бутенко, А. Карамушко. **Підвищення рівня екологічної безпеки насосних установок шляхом енергозаощадження при регулюванні.** Обгрунтовано актуальність вдосконалення методів розрахунків насосних установок гідравлічних систем задля енергозбереження і відповідного підвищення їх рівня екологічної безпеки. Для прискорення гідравлічних розрахунків та підвищення їх точності запропоновано апроксимувати наявні характеристики відцентрових насосів, що зазвичай подаються у паспорті обладнання у графічній формі, квадратичною параболою. Отримано загальні рівняння для аналітичного визначення параметрів робочої точки, побудови напірної характеристики насоса при новій частоті обертання та формулу геометричного місця вершин напірних характеристик для різних частот обертання робочого колеса. Доведено, що при частотному регулюванні характеристика насоса зміщується еквідистантно вздовж параболи пропорційності. Аналогічним чином на характеристики насоса впливає обточка робочого колеса. Отримано рівняння, яке дозволяє, не звертаючись до графоаналітичного методу, розрахувати зміну характеристик насоса при пропорційній зміні усіх його розмірів і постійній частоті обертання та рівняння кривої подібних режимів. Цю криву запропоновано називати кривою подібності. Показано, що вздовж кривої подібності коефіцієнт швидкохідності є постійною величиною. На підставі отриманих результатів запропоновано метод порівняльного аналізу енергоефективності регулювання насосної установки шляхом зміни частоти обертання та пропорційної зміни розмірів робочого колеса. Застосування методу продемонстровано на конкретних прикладах. Досягнення найвищого ККД насосної установки забезпечує найменші енергозатрати, отже мінімізує викиди шкідливих речовин в атмосферу.
Ключові слова: екологічна безпека, ефективність, відцентрові насоси, регулювання подачі, гідравлічні опори, зміна частоти обертання, закони подібності, закони пропорційності

S. Surkov, O. Butenko, A. Karamushko. **Increasing the level of environmental safety of pumping units by energy saving during regulation.** The relevance of improving calculation methods for pumping units of hydraulic systems in order to save energy and correspondingly increase their level of environmental safety is substantiated. To speed up hydraulic calculations and increase their accuracy, it is proposed to approximate the existing characteristics of centrifugal pumps, which are usually presented in the equipment passport in graphic form, with a quadratic parabola. The general equations for the analytical determination of the working point parameters, the construction of the pressure characteristics of the pump at a new rotation frequency and the formula for the geometric location of the tops of the pressure characteristics for different rotation frequencies of the impeller were obtained. It has been proven that with frequency control, the pump characteristic shifts equidistantly along the parabola of proportionality. In a similar way, the characteristics of the pump are affected by the casing of the impeller. An equation has been obtained that allows, without resorting to the grapho-analytical method, to calculate the change in the characteristics of the pump with a proportional change in all its dimensions and a constant frequency of rotation and the equation of the curve of similar modes. This curve is proposed to be called the similarity curve. It is shown that the velocity coefficient is a constant value along the polynomial curve. On the basis of the obtained results, a method of comparative analysis of the energy efficiency of the regulation of the pumping unit by changing the rotation frequency and proportionally changing all dimensions of the impeller is proposed. The application of the method is demonstrated on concrete examples. Achieving the highest efficiency of the pumping unit ensures the lowest energy consumption, thus minimizing emissions of harmful substances into the atmosphere.
Keywords: environmental safety, efficiency, centrifugal pumps, flow regulation, hydraulic resistances, change of rotation frequency, laws of similarity, laws of proportionality

Introduction

Energy saving when adjusting the parameters of pumping systems is an important means of increasing the level of environmental safety. Throughout the entire time of technological development of human society, there is a clear trend of increasing energy consumption, both overall and per person. Energy consumption grew especially rapidly during the last century, when electrical energy became the main form of energy. Thus, before the beginning of the Second World War (in 1938), 460 billion kWh were produced; in 1950 – already 970 billion kWh, in 1970...5050 billion kWh, in 2000...15600 billion kWh, and in 2022...29100 billion kWh. Despite the development of nuclear power, hydropower and alternative sources, thermal power remains the main producer of electricity in the world (in general, about 60% of global production). It is clear that the use of the appropriate amount of fossil fuels at TPPs creates incredibly high pressure on the natural environment and is the main driver of climate change. Therefore, the issue of ensuring a high level of energy efficiency in both the industrial and

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household spheres is an important factor in ensuring global environmental security.

Analysis of recent research and publications

A necessary element of the vast majority of modern technological processes and productions are hydraulic systems with pump supply. Of the total amount of energy produced in the world, pumping systems consume about 20% [1, 2]. Such systems are basic for energy, chemical, metallurgical industries, housing and communal services, irrigation systems, etc. Almost all of these systems use centrifugal pumps of various designs, the common feature of which is the ability to adjust the supply and pressure in a sufficiently wide range. The main methods of regulation are throttle (by means of adjustable local resistance) and frequency (by changing the frequency of rotation of the impeller). The bypass method is used less often. Taking into account the fact that even at the design stage, pumping equipment is always selected with a certain head and supply margin, the task of adjustment is very common [3]. In addition, there are technological processes that, by their very nature, require frequent changes in parameters. For example, the hydraulic tract of feed water on units of thermal power plants with a variable daily mode of operation or the domestic water supply system of settlements.

The modern development of pumping technology no longer allows to significantly increase the efficiency of the process of transferring drive energy to the liquid in the pump itself. Therefore, a reserve for increasing energy efficiency is the optimization of the process of adjusting the parameters of the pump, given that it is an element of the pump installation (hydraulic system), that is, to take into account not only how the adjustment process affects the efficiency of the pump itself, but also to analyze the impact of this process on losses energy in the hydraulic system. The energy saving potential that can be achieved by adjusting the pumping system is about 30% [2, 4].

The issue of quantitative comprehensive assessment of the efficiency of the hydraulic system with pump supply was considered in [5, 6]. Based on the introduction of the concept of the efficiency of the system element, the authors proposed the following formula:

$$\frac{1}{\eta_s^2} = \frac{1}{\eta_p^2} + \sum_{n=1}^{k} \left(\frac{1}{\eta_i^2} - 1 \right) \frac{\omega_j^2}{\omega_i^2}, \qquad (1)$$

where η_s, η_p, η_i – efficiency of the hydraulic system as a whole, the pump and the *i*-th element of the system, respectively;

ω_i and ω_j are cross-sectional areas of individual sections.

Formula (1) makes it possible to quantitatively estimate the energy efficiency of a hydraulic system taking into account the efficiency of each of its elements, but it is not convenient for real networks, where the number of such elements can be significant. In addition, for example, when the valve is fully closed, its efficiency is zero, which leads to the need to divide by zero.

In work [7], for the overall energy efficiency of the hydraulic network, the coefficient of energy perfection of the network is proposed:

$$\varepsilon = \eta_p \left(1 - \frac{S Q_p^2}{H_p} \right), \qquad (2)$$

which is the ratio of the flow power at the output of the hydraulic network to the supplied power, which is the power on the pump shaft (here Q_p and H_p – supply (m^3/s) and pump head (m), S is the resistance of the hydraulic network, s^2/m^5).

In [8], formula (2) is transformed for the case of a complex hydraulic network. The head losses in all sections of the network are summarized here:

$$\eta = \eta_p \left(1 - \frac{\sum S_i Q_i^2}{H_p} \right). \qquad (3)$$

In [9], new efficiency calculation methods were used to compare two options for the modernization of the aerodynamic system – when replacing the electric motor to increase the frequency of rotation of the fan impeller, and when replacing the fan without replacing the electric motor.

But the problem of energy saving with different methods of flow control in hydraulic systems has not been fully investigated.

The purpose of the work is to improve calculation methods for increasing the level of environmental safety of pumping units by ensuring energy savings during their regulation.

Presentation of the main material

The full head of the pump H_p, which must be applied to the liquid in a simple pipeline, can be represented as the sum of the static head H_{st}, the change in the speed head and head losses $\Delta h = SQ^2$:

$$H_p = H_{st} + \frac{v_2^2 - v_1^2}{2g} + SQ^2 .$$

The useful power of the pump is usually represented in the form:

$$N_u = \rho g H_p Q = \rho g Q\left(H_{st} + \frac{v_2^2 - v_1^2}{2g} + SQ^2 \right).$$

But in reality, the term SQ^2 is not useful, because this part of the power cannot be used by the consumer (this is taken into account in [7, 8] when introducing the concept of the coefficient of energy perfection of the network).

The term $\frac{v_2^2 - v_1^2}{2g}$ takes into account the change in the kinetic energy of the flow. Since liquid is taken into the pump, as a rule, it is carried out from a tank or reservoir of a large size, then almost always $v_1 = 0$. If the output of the liquid from the pipeline is carried out into a large tank, then it can be assumed that $v_2 = 0$. But in the general case, the change in dynamic head $\frac{v_2^2 - v_1^2}{2g}$ should be included in the right-hand side of the equation for useful power.

It is easy to illustrate these ideas on the example of a simple pipeline in which the supply is regulated by means of a throttle. The characteristics of such a pump installation are shown in Fig. 1. There are head losses:

$$\Delta h = S_{pl} Q^2 + S_{thr} Q^2 ,$$

where S_{pl} – pipeline resistance, s²/m⁵;

S_{thr} – throttle resistance, s²/m⁵.

Since the product ρg is constant for the hydraulic system under consideration, the ratio of the areas of the colored rectangles in Fig. 1 is equal to the ratio of the corresponding powers. In particular, the product $H_p Q$ is proportional to the total area of rectangles *1 – 4* (Fig. 1).

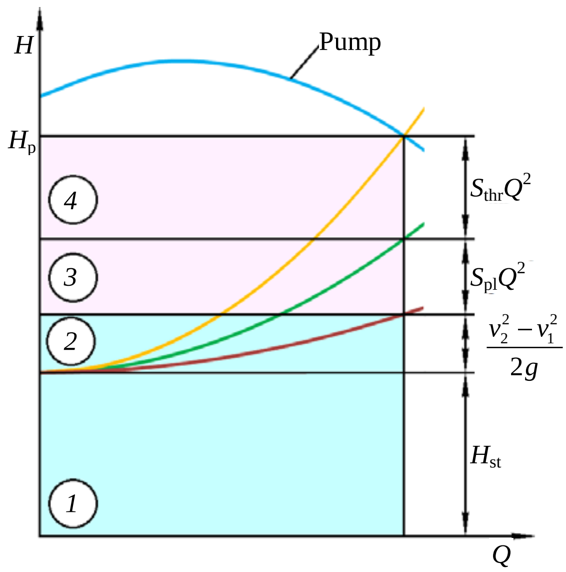

Fig. 1. Useful power and power losses when calculating the efficiency of the hydraulic system

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In Fig. 1, rectangles *1* and *2* reflect useful power components $-\rho g Q\left(H_{\mathrm{st}}+\frac{v_2^2-v_1^2}{2g}\right)$. Rectangles *3* and *4* reflect power losses in the pipeline and in the throttle – $\rho g(S_{\mathrm{pl}}+S_{\mathrm{thr}})Q^3$. In this case, the total efficiency of the system, according to (3), is equal to:

$$\eta=\eta_{\mathrm{p}}\left(1-\frac{(S_{\mathrm{pl}}+S_{\mathrm{thr}})Q^2}{H_{\mathrm{p}}}\right).$$

Since the pump characteristics are usually presented graphically, the problem of calculating the control parameters is also solved by the graph analytical method. For the vast majority of engineering personnel, this way is familiar, but at the same time it is inconvenient and does not take into account the modern possibilities of computer technology.

Modern software allows you to largely automate such calculations, that is, speed them up and increase accuracy.

To a large extent, the approximation of the pump characteristic by a quadratic parabola contributes to the achievement of this goal. At the same time, the characteristic of the pump is described by the equation:

$$H_{\mathrm{p}}(Q)=a_{\mathrm{p}}Q^2+b_{\mathrm{p}}Q+c_{\mathrm{p}}, \tag{4}$$

where H_{p} – pump head, m;

Q – volumetric supply, i.e. volumetric flow of liquid at the pump outlet, m^3/s;

a_{p}, b_{p}, c_{p} – regression coefficients.

Coefficients a_{p}, b_{p} and c_{p} are found from the passport characteristics of the pump by approximation by the method of least squares.

It can be assumed that the pump manufacturers, who experimentally determined the passport characteristics, often used the quadratic approximation of the experimental data. For example, the characteristics of the *K* 90/55 pump [10] are approximated by a parabola (Fig. 2). The coefficient of determination R^2 exceeds 0.999, which indicates a very high level of agreement. It can be shown that the coefficient R^2 is close to unity for other centrifugal pumps as well.

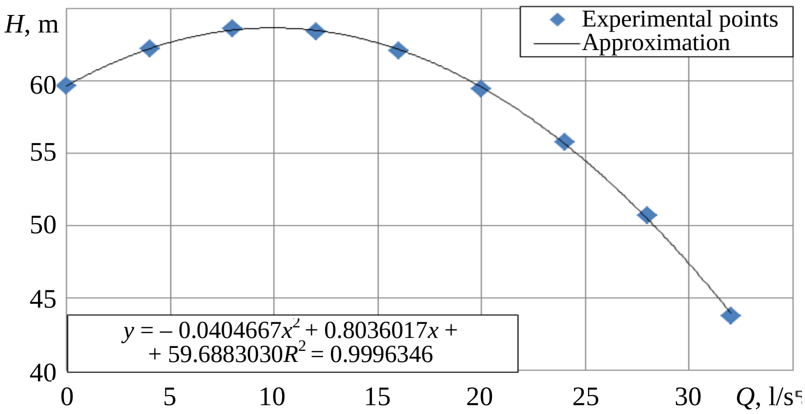

Fig. 2. Approximation of the pressure characteristic of a centrifugal pump by a quadratic parabola and regression equation

Most of the typical problems of hydraulics involve finding the working point of the pump-pipeline system. If the coefficients of local losses and losses along the length of the pipe lie in the region of self-similarity, the characteristic of a simple pipeline is described by the equation:

$$H_{\mathrm{pl}}(Q)=SQ^2+H_{\mathrm{st}}, \tag{5}$$

where H_{st} – static head, m;

S – pipeline resistance, s^2/m^5.

The operating point shows what the actual flow rate and the actual pressure will be set in the system during the operation of this pump on this pipeline. At the working point:

$$H_{\mathrm{p}} - H_{\mathrm{pl}} = 0 . \tag{6}$$

Thus, the search for the operating point of the "pump-pipeline" system consists in the analytical solution of the system of equations (4) – (6) or in the graphical search for the point of intersection of the pump and pipeline characteristics.

In the case of a simple pipeline, the system of equations (4) – (6) reduces to the equation:

$$(a_{\mathrm{p}} - S)Q^2 + b_{\mathrm{p}}Q + c_{\mathrm{p}} - H_{\mathrm{st}} = 0 .$$

So, in order to find the point of intersection of two characteristics, it is necessary to solve a quadratic equation:

$$aQ^2 + bQ + c = 0 , \tag{7}$$

in which:

$$a = a_{\mathrm{p}} - S ,$$

$$b = b_{\mathrm{p}} ,$$

$$c = c_{\mathrm{p}} - H_{\mathrm{st}} .$$

The pump supply at the operating point is found by the formula:

$$Q_{\mathrm{op}} = \frac{-b - \sqrt{b^2 - 4ac}}{2a} .$$

The "minus" sign in front of the square root ensures a positive value of Q_{op}, considering that the parameter a_{p} in the pump characteristics almost always is negative.

Fig. 3 shows the operating point *A*, which is located at the intersection of the pump characteristic $H_{\mathrm{p}}(Q)$ and the pipeline characteristic $H_{\mathrm{pl}}(Q)$. As a result of solving equation (7), the flowrate Q_A was found.

Throttle regulation, despite its energy-consuming nature, is nowadays the most common method of regulating pump supply due to the simplicity and cheapness of the equipment used.

There are head losses in the throttle, which can be calculated using the formula:

$$\Delta h_{\mathrm{thr}} = H_{\mathrm{p}} - H_{\mathrm{pl}} = a_{\mathrm{p}}Q^2 + b_{\mathrm{p}}Q + c_{\mathrm{p}} - SQ^2 - H_{\mathrm{st}} .$$

The graph (Fig. 3) shows that with throttle adjustment, the operating point moves along the pump characteristic from point *A* to point *B*. In this case, head losses Δh_{thr} occur, which are displayed by the vertical segment BC.

The throttle resistance required to obtain the supply Q_B is calculated by the formula:

$$S_{\mathrm{thr}} = \frac{\Delta h_{\mathrm{thr}}}{Q_B^2} .$$

The physical meaning of the throttling process is that part of the flow's mechanical energy is converted into thermal energy, that is, the usable energy of the flow (or its exergy) is lost, and this, according to (3), reduces the efficiency of the system.

The main alternative to throttle regulation is regulation by changing the frequency of rotation of the pump impeller. As a rule, the calculation part of the task of such adjustment consists in determining the new frequency n_2, which will provide a new value of flow rate .. in the hydraulic net-

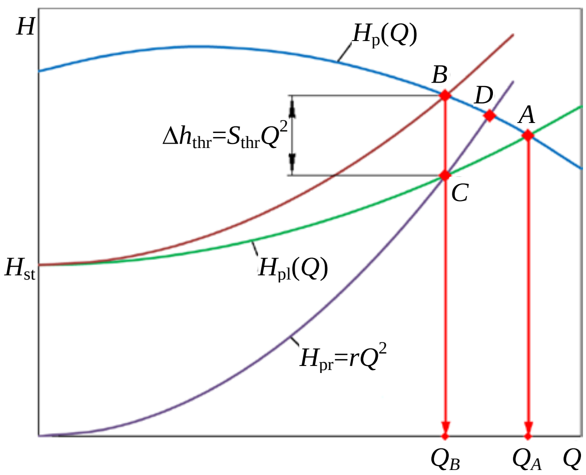

Fig. 3. Finding the operating point and ways to adjust the feed

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work. At the same time, the pressure characteristic of the centrifugal pump at the rotation frequency n_1 is given by equation (4).

The solution to this problem is based on the laws of proportionality, which relate the parameters of the same pump at different rotation frequencies:

$$\frac{Q_2}{Q_1}=\frac{n_2}{n_1}, \qquad \frac{H_2}{H_1}=\frac{n_2^2}{n_1^2}.$$

The laws of proportionality are valid for points that lie on the same parabola of proportionality, that is, on the curve given by the equation:

$$H_{\mathrm{pr}}=rQ^2. \tag{8}$$

The peculiarity of this problem is that the coordinates of the new working point Q_C and H_C are given. This allows you to draw a parabola of proportionality through it.

The algorithm for finding a new operating point is illustrated by a graph (Fig. 3). To draw the parabola of proportionality through point *C*, the parameter *r* is calculated as:

$$r=\frac{H_{\mathrm{C}}}{Q_C^2}.$$

Now it is necessary to find the coordinates of point *D*, that is, the point of intersection of the old characteristic of the pump with the proportionality curve. Searching for a feed at point *D* – Q_D – again comes down to solving the quadratic equation (7). This time:

$$a=a_{\mathrm{p}}-r,$$

$$b=b_{\mathrm{p}},$$

$$c=c_{\mathrm{p}}.$$

When Q_D is found, we find the new rotation frequency as: $n_2=\dfrac{n_1 Q_C}{Q_D}$.

When changing the rotation frequency, there are no pressure losses on the regulating device. But in this case, it will be necessary to spend money on an adjustable pump drive.

The problem of frequency control is more complicated, when the pressure characteristic at the initial rotation frequency n_1, described by equation (4), the new rotation frequency n_2 and the required flowrate Q_2 are given, and the pump head is to be determined.

The peculiarity of this problem is that it does not specify two coordinates of any point that lies on the required proportionality curve. Therefore, it is necessary to derive the general equation of the pump characteristics at the new rotation frequency n_2.

For the solution, we again use the laws of proportionality and enter the value *k* as the ratio of rotation frequencies:

$$k=\frac{n_2}{n_1}.$$

Then:

$$Q=\frac{Q_2}{k}, \qquad H=\frac{H_2}{k^2}.$$

Substitute these expressions into the pump characteristic equation:

$$\frac{H_2}{k^2}=\frac{a_{\mathrm{p}}Q_2^2}{k^2}+\frac{b_{\mathrm{p}}Q_2}{k}+c_{\mathrm{p}},$$

and we will get the characteristic of the pump at the new rotation frequency:

$$H_2=a_{\mathrm{p}}Q_2^2+b_{\mathrm{p}}kQ_2+c_{\mathrm{p}}k^2. \tag{9}$$

This equation allows you to find the head of the pump H_2 at a given supply Q_2 and rotation frequency n_2. The cases $k > 1$ correspond to an increase in the rotation frequency, and $k < 1$ to its decrease.

The graph (Fig. 4) shows several characteristics of the pump at different values of k.

The question of how the shape and location on the graph of the pump characteristics changes when the rotation frequency changes is of interest.

If the pump characteristic is described by equation (4), then the coordinates of the top of the parabola can be found:

$$Q_0 = \frac{-b_p}{2a_p}, \qquad H_0 = \frac{4a_p c_p - b_p^2}{4a_p}.$$

With such designations, the pump characteristic equation (4) takes the form:

$$H = H_0 + a_p (Q - Q_0)^2.$$

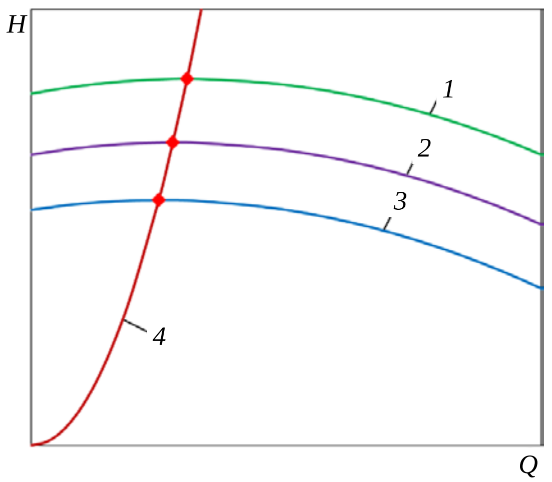

Fig. 4. Characteristics of the pump at different frequencies of rotation of the impeller. *1* – $k = 1.1$; *2* – $k = 1$; *3* – $k = 0.9$; *4* – geometric locus of points of maximum head (vertices of parabolas)

Please note that the b_p and c_p parameters affect the location of the top of the parabola. But it is immediately clear that the shape of the parabola does not change, because it is determined only by the coefficient a_p.

When the rotation frequency changes, the coordinates of the top of the parabola become dependent on the parameter k:

$$Q_0(k) = kQ_0 = \frac{-kb_p}{2a_p},$$

$$H_0(k) = k^2 H_0 = k^2 \frac{4a_p c_p - b_p^2}{4a_p}.$$

Thus, we obtained the equation of the curve, given parametrically. By removing k from this system, we get the equation of the geometric locus of the parabola vertices:

$$H_0(k) = a_p \left(\frac{4a_p c_p}{b_p^2} - 1 \right) Q_0^2.$$

That is, the geometric locus of the peaks of the pressure characteristics is a quadratic parabola that passes through the maximum point at the initial rotation frequency n_1 and through the origin of the coordinates. In fig. 4 characteristics of the pump at different rotation frequencies are shown by curves 1, 2 and 3, and the geometric locus of the parabola vertices is shown by curve 4.

In the special case $b_p = 0$, the top of the parabola “slides” along the vertical axis, without changing its shape, that is, it moves equidistantly. Then a new characteristic of the pump:

$$H_2 = a_p Q_2^2 + c_p k^2.$$

Another alternative to throttle control is to consider changing the range of the pump by turning the impeller. The peculiarity of the calculation is that the change in the diameter of the impeller during turning affects the proportionality equation similarly to the change in the rotation frequency. Therefore, for calculations, you can use the equation of the curve (8), which this time is called the “turning parabola”, and the equation of the pump characteristic (9). But in this case:

$$k = \frac{D_{turn}}{D_{out}}.$$

where D_{turn} – diameter of the impeller after turning;

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D_{out} – the outer diameter of the impeller.

The coefficient *k* in the case of turning must always be less than unity.

The graph (Fig. 3) can also be used to calculate the required diameter of the turning, at the same time:

$$D_{\text{turn}} = \frac{D_{\text{out}} Q_{\text{C}}}{Q_{\text{D}}}.$$

It is obvious that, compared to throttling, the regulation of the pump parameters by the frequency of rotation and the turning of the impeller will ensure higher energy efficiency, and therefore the level of environmental safety of the pump installation.

Another type of hydraulic problem is obtaining the characteristics of a pump geometrically similar to this one at a constant rotation frequency. Given the characteristic of the existing pump (4) and the required supply. It is necessary to determine the head of the new pump.

Solving such a problem arises at the stage of designing a new pump, when it is necessary to calculate the expected characteristics of the new pump. Also, such a calculation can be useful when choosing a new pump from the catalogs of manufacturers.

Since two coordinates of the new operating point are not specified in the problem, it is necessary to obtain the equation of the entire characteristic of the pump.

Laws of similarity for centrifugal pumps:

$$\frac{Q_{\text{n}}}{Q_{\text{m}}} = \frac{n_{\text{n}} D_{\text{n}}^3}{n_{\text{m}} D_{\text{m}}^3}, \qquad \frac{H_{\text{n}}}{H_{\text{m}}} = \frac{n_{\text{n}}^2 D_{\text{n}}^2}{n_{\text{m}}^2 D_{\text{m}}^2},$$

where Q_{n} and Q_{m} – supplies of full-scale and model pumps;

H_{n} and H_{m} – head of natural and model pumps;

D_{n} and D_{m} – outer diameters of impellers of full-scale and model pumps;

n_{n} and n_{m} – rotation frequency of working wheels of full-scale and model pumps.

If you enter a linear modeling scale:

$$l = \frac{D_{\text{n}}}{D_{\text{m}}},$$

and keep in mind that under the condition of the problem $n_{\text{n}} = n_{\text{m}}$, then:

$$\frac{Q_{\text{n}}}{Q_{\text{m}}} = l^3, \qquad \frac{H_{\text{n}}}{H_{\text{m}}} = l^2.$$

By removing the parameter *l*, we will obtain the equation of the curve of similar modes with a proportional change of all dimensions of the impeller and at a constant rotation frequency:

$$\frac{H_{\text{n}}}{H_{\text{m}}} = \left(\frac{Q_{\text{n}}}{Q_{\text{m}}}\right)^{2/3}. \qquad (10)$$

We suggest calling such a curve a similarity curve, or a curve of similar regimes.

We see that the lines of similar regimes at a constant rotation frequency have the form of a power dependence with a degree of 2/3.

We have:

$$Q_{\text{n}} = l^3 Q_{\text{m}}, \qquad H_{\text{n}} = l^2 H_{\text{m}}.$$

Substitute H_{n} and Q_{n} in the pressure characteristic equation of the natural pump (4). For simplification, we omit the "m" index:

$$l^2 H = a l^6 Q^2 + b l^3 Q + c.$$

Dividing all terms by l^2, we get the characteristic equation of the model pump:

$$H = a l^4 Q^2 + b l Q + \frac{c}{l^2}. \qquad (11)$$

It can be seen from equation (11) that the coefficient before Q^2 depends on the scale of modeling *l*. Therefore, when changing the geometric dimensions of the pump, the shape of its characteristics changes.

The graph (Fig. 5) shows examples of the characteristics of geometrically similar pumps and constant rotation frequency.

Curve 5 in Fig. 5 is one of the similarity curves defined by equation (10). For the drawing, such a similarity curve is chosen, which is the geometric locus of the peaks of the pressure characteristics of the pumps.

The similarity curve equation (10) can be rewritten in the form:

$$\frac{Q^{2/3}}{H} = \text{const}\,,$$

or, raising the numerator and denominator to a power 3/4:

$$\frac{Q^{1/2}}{H^{3/4}} = \text{const}\,.$$

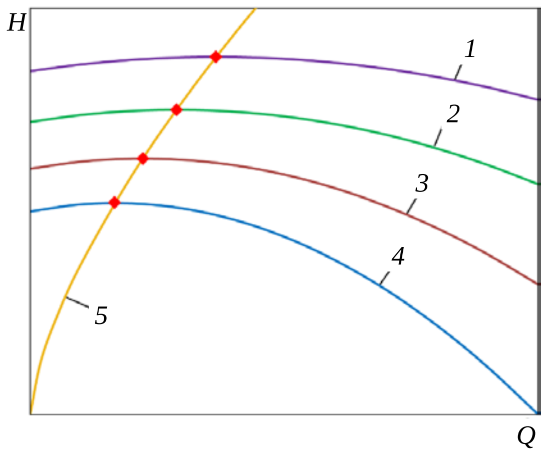

Fig. 5. Characteristics of geometrically similar pumps at constant rotation frequency and different linear modeling scales: *1* – $l = 1.3$; *2* – $l = 1.2$; *3* – $l = 1.1$; *4* – $l = 1$; *5* – geometric locus of points of maximum head (vertices of parabolas)

This constant can be multiplied by another constant $3.65 \cdot n = \text{const}$, since in our case $n = \text{const}$. Then along the similarity curve:

$$3.65\frac{n\sqrt{Q}}{H^{3/4}} = n_s = \text{const}\,.$$

That is, the similarity lines obtained are lines with a constant speed coefficient n_s. If you plot a large number of lines of similarity, you can see that at each point of the pump characteristic, the speed coefficient is different. But in the passport characteristics of the pump, the speed coefficient calculated for the point with maximum efficiency is indicated.

By analogy with [9], we will find out with specific examples which means of increasing the pressure will ensure a higher efficiency of the hydraulic system: changing the electric motor to ensure a higher rotation frequency, or changing the pump to a geometrically similar one without changing the rotation frequency.

We take as a basis the characteristics of the *K* 90/55 pump at the passport speed of rotation $n_1 = 2900$ rpm. Let's imagine that the existing hydraulic system needs to be modernized to ensure a head $H_A = 78$ m at a flow rate $Q_A = 28$ l/s. Pump characteristics and the operating point *A* are shown in graph (Fig. 6).

We will consider two ways to solve this technical problem.

The first way is to increase the rotation frequency of the existing impeller. We draw a parabola of proportionality through the given operating point *A* and find point *B* (Fig. 6), where it intersects with the pump characteristic. Solving the quadratic equation (7) and taking into account the proportionality curve equation (8), we get $Q_B = 23.72$ l/s at the point of intersection.

Then the rotation frequency is required:

$$n_2 = \frac{n_1 Q_A}{Q_B} = \frac{2900 \cdot 28}{23.72} = 3423 \text{ rpm.}$$

The second method is the use of a pump, geometrically similar to this one, at a constant rotation frequency.

We draw a similarity curve described by equation (10) through the given operating point *A*. Unfortunately, the system of equations (4) and (10) cannot be solved analytically. Numerical methods give for point *C*, in which these characteristics intersect, $Q_C = 19.04$ l/s. According to this method, all dimensions of the impeller must be increased by *l* times, where:

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$$l = \sqrt[3]{\frac{Q_A}{Q_C}} = \sqrt[3]{\frac{28}{19.04}} = 1.137\,.$$

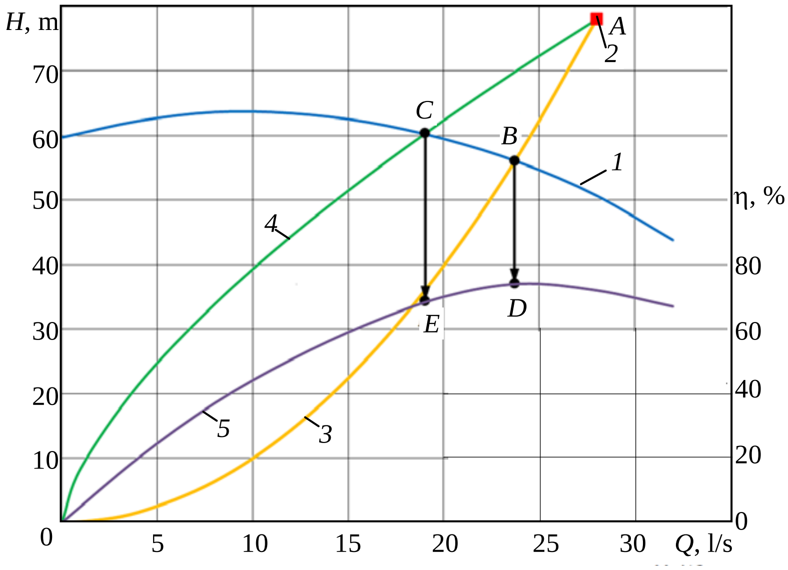

Fig. 6. Comparison of pump installation modernization options at $H_A = 78$ m and $Q_A = 28$ l/s:
1 – Pump head; *2* – Operating point; *3* – Proportionality curve;
4 – Similarity curve; *5* – Pump efficiency

The useful power of the pumping unit is the same in both cases, because it is determined by the operating point parameters. Therefore, according to (3), the efficiency of the system will depend only on the efficiency of the pump.

Dropping the perpendiculars *BD* and *CE* from the operating points to the pump efficiency curve, we see that with the first method, the pump efficiency is about 74% (point *D*), and with the second – 68.5% (point *E*). So, in this particular case, the first method provides less electricity consumption, and therefore less anthropogenic load on the environment.

Let's repeat the same calculations for the case when the parameters of the new operating point $H_A = 65$ m and $Q_A = 32$ l/s (Fig. 7).

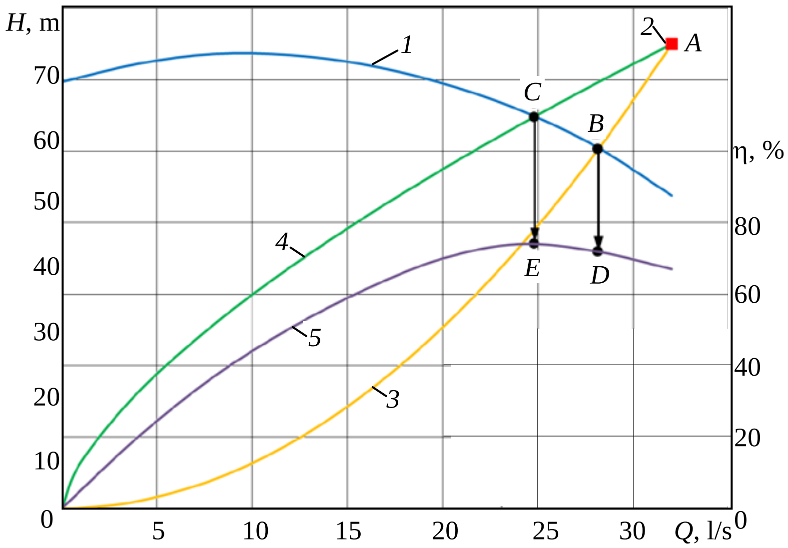

Fig. 7. Comparison of pump installation modernization options at
$H_A = 65$ m and $Q_A = 32$ l/s

From Fig. 7, it can be seen that this time point *E* is higher than point *D*. When the rotation frequency increases, the efficiency of the pump is about 71.8%, and when the pump size increases proportionally, it is 74%, that is, the second method will be optimal.

Conclusions

1. Through the analysis of previous studies, the relevance of improving the calculation method for regulating pumping units of hydraulic systems is substantiated.

2. To automate hydraulic calculations and increase their accuracy, it is proposed to approximate the passport characteristics of centrifugal pumps with a quadratic parabola. It is shown that such a simple method provides a very high coefficient of determination and is universal.

3. General equations for analytical determination are obtained:

– operating point parameters,

– pressure characteristic of the pump at a new rotation frequency (it has been proven that with frequency control the pressure characteristic of the pump shifts equidistantly),

– the geometric locus of the peaks of the pressure characteristics at different frequencies of rotation of the impeller,

– characteristics of the pump with a proportional change of all its dimensions and a constant frequency of rotation and a curve of similar modes.

4. A method of comparative analysis of the energy efficiency of the regulation of the pumping unit by changing the rotation frequency and proportional change of all the pump dimensions is proposed. It is shown that even for the same pump, the optimal method can vary depending on specific control parameters. This confirms the relevance and importance of analytical calculation methods from the point of view of ensuring energy efficiency and, accordingly, environmental safety.

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