Досліджено робочий процес коліс двостороннього входу відцентрових насосів та виявлено залежність крутизни напірної характеристики від відносної ширини колеса. При числовому моделюванні змінювалася лише ширина робочого колеса на виході, а інші геометричні розміри колеса лишалися без зміни. Результати досліджень важливі при проектуванні робочих коліс двостороннього входу з наперед визначеною крутизною напірної характеристики

Ключові слова: напірна характеристика, насосна станція, робоче колесо, ширина виходу, відцентровий насос

Исследован рабочий процесс колес двустороннего входа центробежных насосов и обнаружена зависимость крутизны напорной характеристики от относительной ширины колеса. При численном исследовании менялась только ширина рабочего колеса на выходе, а другие геометрические размеры колеса оставались неизменными. Результаты исследований важны при проектировании рабочих колес двустороннего входа с предопределенной крутизной напорной характеристики

Ключевые слова: напорная характеристика, насосная станция, рабочее колесо, ширина выхода, центробежный насос

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ANALYSIS OF THE IMPACT OF IMPELLER OUTLET WIDTH ON THE STEEPNESS OF PRESSURE CHARACTERISTIC

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1. Introduction

In the design of pumping stations of industrial and drinking water supply, pumps are chosen based on the maximal water consumption considering its gradual increase in the future. The most common of the pumping units that work on such pumping stations are the pumps with double-entry impellers with estimated supply from 1250 m^3/h to 6300 m^3/h. At modern rates of slowing down production and, as a result, water consumption, when using inefficient methods of regulation, according to [1] about 60 % of the pumping stations work with an efficiency (Eff) of only 10–40 %. It should be noted that 75 % of the pumping systems have overestimated values of pressure higher than by 20 % [2].

In the efforts to increase the efficiency of the pumping stations (PS) they resort to the following methods of regulating the pumps performance in the system: using a variable-frequency drive (VFD), the use of the “on-off” method of the pumps working in parallel and their combinations.

For the majority of cases, functioning of the pump systems at variable operating modes of VFD is one of the most efficient ways to regulate the pump unit. However, the changes in the parameters of the pump can be obtained by replacing the impeller, designed for the new parameters of the functioning of the network and in this case it is necessary to take into account the change in the steepness of pressure characteristics of the pump. Also, when applying a particular method of regulating the supply system, there are some requirements for pressure characteristics of the pump. Thus, at VFD Q-H characteristic should have the largest possible steepness, and during the work of pumps in parallel, their characteristics should be maximally sloping [3].

In the well-known scientific literary sources [4, 5] it is stated that the steepness of the pressure characteristic is one of the most important parameters of the pump, which must be taken into account when designing the impellers of pumps or PS control method.

The most important task is to increase efficiency of the PS performance, to reduce their energy consumption, provided they meet the requirements of consumers. Since, for the most part, this is due to a mismatch of the characteristics of a pump and the chosen way of regulating, the total efficiency of PS remains quite low, and actual issue is to define interrelations between geometric dimensions of the impeller of the centrifugal pumps and the parameters of their pressure characteristics.

2. Analysis of scientific literature and the problem statement

The impeller outlet width is one of the geometric parameters that defines the structure of the liquid flow in impeller, the magnitude of hydraulic losses, in particular, and the characteristics of impeller (H=f(Q), Eff=f(Q), P=f(Q)) as a whole.

According to the papers [6, 7], a steepness is measured through $tg\varphi^H$, the tangent of the angle to the pressure characteristic, which is determined by the following formula:

$$tg\varphi^H = \frac{q^p}{1-q^p}, \qquad (1)$$

where q^p is the consumable option in the reference point:

$$q^p = \frac{v_{m2}}{u_2 \cdot tg\beta_2}, \quad (2)$$

where u_2 is the circular velocity:

$$u_2 = \frac{\pi \cdot n \cdot D_2}{60}, \quad (3)$$

where v_{m2} is the meridian component of the absolute velocity:

$$v_{m2} = \frac{Q_{рк}}{\pi \cdot D_2 \cdot b_2}, \quad (4)$$

where $Q_{рк}$ is the impeller supply.

Converting the equation (2) with (3) and (4) we have:

$$q^p = \frac{60Q}{\pi^2 \cdot D_2^2 \cdot n \cdot b_2 \cdot tg\beta_2}. \quad (5)$$

As can be seen from (5) and (1), a consumable parameter, and hence the steepness of the pressure characteristic in the reference point, depends only on the external diameter of impeller D_2, the impeller outlet width b_2, the angle of the blade at the output of impeller β_2 at the constant frequency of rotations of the rotor n. Moreover, the largest impact is produced exactly by β_2, b_2. The parameters q^p and β_2, b_2 are inversely dependent: with increasing values of b_2 and β_2, the value q^p decreases and vice versa. That is, given the equation (1), at increasing β_2 or b_2, the characteristic grows more sloping.

The works [6, 8–11] propose to determine the width of impeller using the dependencies obtained both theoretically and statistically.

Thus, taking the equation of continuity as the basis, the following dependence was obtained:

$$b_2 = \frac{Q_{рк}}{\pi \cdot D_2 \cdot v_{m2}}. \quad (6)$$

By supplementing dependence (6) with the results of statistical analysis, the aforementioned authors proposed to determine the impeller outlet width by the formula:

$$b_2 = \frac{Q_{рк}}{K \cdot \pi \cdot D_2 \cdot \sqrt{2gH}}, \quad (7)$$

where K is the coefficient obtained as a result of statistical analysis, H is the pressure.

The following dependencies were obtained entirely by statistical analysis of the data in the works [9–11], respectively:

$$b_2 = k_v \sqrt[3]{\frac{Q}{n}}, \quad (8)$$

where k_v is the coefficient obtained as a result of statistical analysis,

$$b_2 = k_{v2} \cdot \frac{\sqrt{H}}{n}, \quad (9)$$

where k_{v2} is the coefficient obtained as a result of statistical analysis,

$$b_2^* = \frac{b_2}{d_{2a}} = 0{,}017 + 0{,}262\frac{n_q}{n_{qRef}} - -0{,}08\left(\frac{n_q}{n_{qRef}}\right)^2 + 0{,}0093\left(\frac{n_q}{n_{qRef}}\right)^3, \quad (10)$$

where

$$n_q = n\frac{\sqrt{\dfrac{Q_{opt}}{f_q}}}{H_{opt}^{0,75}}$$

is the specific speed, n_{qRef}=100.

These authors offer to focus on the pressure H, the supply Q, even the speed of rotation of the rotor n. However, neither method of calculation mentioned the steepness of the pressure characteristic and the influence of the impeller outlet width.

Since 2000, following the development of numerical methods for solving hydrodynamic problems, the studies have been conducted to determine the impact of the change in the impeller outlet width on the steepness of the pressure characteristic.

Thus, the authors in the work [12] argue that the effect of changing the impeller outlet width on the characteristic of the pump is a very efficient method of regulating the supply of a pump unit.

The paper [13] demonstrated that the increase in an impeller outlet width leads to a decrease in losses to friction, which leads to increasing pressure and hydraulic efficiency of a centrifugal pump.

The results presented in the article [14] also demonstrated the increase in the efficiency of the pump by increasing the width of the impeller.

The paper [15] presented a study of six centrifugal pumps that have a specific speed n_s=45...260 at variable values of the width of impeller. According to the results, the change in the impeller outlet width significantly influences the parameters of the working point, the nature of the flow and the steepness of a pressure characteristic under condition of the immutability of geometric dimensions of other elements of the flowing part of the pump.

The results of the studies of the impact of the change in the impeller outlet width on the characteristics of a centrifugal pump, described in [16], show that the change in its value significantly affects the efficiency of the working process of the pump at the supplies larger than the nominal. In this case the curve of dependence Q-η increases its curvature, indicating a decrease in the efficiency of the working process of a pumping unit.

According to [17], a pump pressure decreases with decreasing impeller outlet width in the pumps of high rapidity. However, in the pumps with low rapidity, the pressure hardly depends on the width of the impeller.

The authors in [18] presented results of a numerical study of the pump with four different values of the width, according to which the pumps with a wider impeller have the best energy efficiency. However, they noted that a too large value of the impeller outlet width leads to a decrease in efficiency of the pump. Similar results were obtained in [19].

All of the above-mentioned researchers in their papers agree on that the impeller outlet width is one of the main geometrical parameters, which significantly influences the characteristics of centrifugal pumps in general and its steepness in particular. However, they did not present a clear correlation between the value of the impeller outlet width and the steepness of its pressure characteristic. Conclusions about broad impellers providing a higher efficiency value of the pump operation while too wide impellers, on the contrary, leading to its decrease, cannot be used in the design of the pumps because they do not allow selecting the optimal value of the impeller outlet width. Therefore, it is important to conduct additional research to identify a clear mathematical dependence between the width of the output impeller and the steepness of its pressure characteristic.

3. The purpose and objectives of the study

The aim of this work is to determine a mathematical dependence between the width of impeller of double-entry impellers of centrifugal pumps with the specific speed n_s=80...210 and the steepness of a pressure characteristic, for the design of them with a predetermined steepness of pressure characteristic adjusted for necessary changed parameters of water supply network.

To achieve this goal, the following problems were identified:

– to perform numeric simulation of the working process of double-entry impellers with different values of the impeller outlet width;

– to obtain a mathematical dependence between the width of the output impeller and the steepness of its pressure characteristic.

4. Materials and methods of numerical research into the working process of double-entry impellers of centrifugal pumps

We studied impeller of the following pumps: D 3200-75, D 6300-80-2, SE 2500-180-8, SE 2500-180-8 UHL 4E, D 6300-27-2. We modeled their working process at the modes from $0{,}5Q_{ref}$ to $1{,}2Q_{ref}$, where Q_{ref} is the supply in the reference point. By the method of conducting research, at unchanging angles of installed blades at the input β_1 and the output β_2 and geometric sizes of the impeller D_2, D_0, d_{hub}, only the width of impeller b_2 at the output changed, as shown in Fig. 1.

Thus, the values of the width of the impellers were selected in increments of 10 % of the value b_2 of the original impeller of the pump. The selected values of the impeller outlet width are displayed in Table 1.

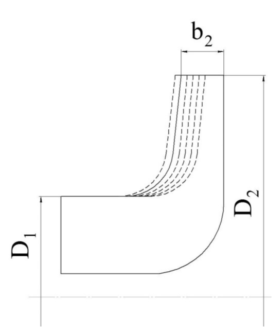

Fig. 1. Diagram of the meridian section of the impeller of centrifugal pump

Table 1

Variants of the impeller outlet width b_2 of the studied pumps impellers

Pump type	Impeller outlet width, b_2, mm
D 3200-75	30,7; 37,9; 45,2; 52,4; **59,6**; 66,9
D 6300-80-2	40,8; 51; 61; 71; **82,2**; 92
SE 2500-180-8	18; 22,5; 27; 31,5; **36**; 39,5
SE 2500-180-8 UHL 4E	25,7; 29,6; 33,4; 37,3; **41,4**; 44,1
D 6300-27-2	65; 80; 95; 110; **125**; 140

Note: bold denotes the value b_2 of the original impeller

For conducting numerical study, by using SolidWorks software package, we created 3D models of impellers with different values of the impeller outlet width. In this the following assumptions were adopted:

– internal flow is symmetric;

– internal flow at the input to the estimated area is axisymmetric;

– leaks through the impeller's seal are neglected.

In connection with the above-mentioned assumption, the estimated area is one channel of the half of the impeller without seals. The input and output borders of the estimated area are at such a distance from control sections that is sufficient for the flow to be stable.

To test the influence of the number of cells in the grid on the results of simulation, in the grid generator CFD we built unstructured grids of the estimated areas with the number of cells in a range from 140 000 to 5 million. Results of the analysis are shown in Table 2. An example of a constructed grid for the estimated area is presented in Fig. 2.

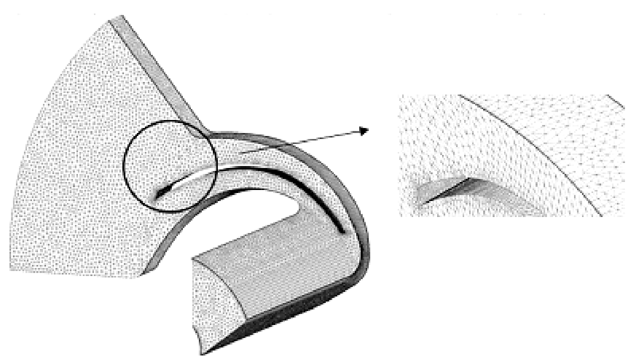

Fig. 2. Example of the constructed unstructured grid for the estimated area

Numeric simulation was performed in the Ansys CFX software package.

At the input of the estimated area and at its output we set the components of velocity and static pressure, accordingly. For numerical simulation we selected standard k–ε – turbulence model and scalable functions of the wall. The roughness of the surface of the walls is 6.3 μm.

The flow in the estimated area was adopted as three-dimensional, viscous, turbulent. As the working fluid we chose water at the temperature of 25 °C, and the criterion of convergence was 10^{-4}. The Reynolds number was 10^7.

All calculations were performed in a stationary state. The input and output segments were fixed while the impeller was able to rotate. In addition, periodic borders were defined for the estimated area.

Table 2

Parameters of the grids of the studied impellers

Variant of impeller geometry	Number of nodes in the grid, thousand
D6300-80-2	
Original geometry	270
Impeller № 1	530
Impeller № 2	360
Impeller № 3	900
Impeller № 4	610
Impeller № 5	700
D3200-75-2	
Original geometry	450
Impeller № 1	290
Impeller № 2	280
Impeller № 3	560
Impeller № 4	560
Impeller № 5	680
SE 2500-180	
Original geometry	540
Impeller № 1	530
Impeller № 2	470
Impeller № 3	690
Impeller № 4	470
Impeller № 5	490
SE 2500-180-8 UHL 4E	
Original geometry	410
Impeller № 1	530
Impeller № 2	420
Impeller № 3	420
Impeller № 4	430
Impeller № 5	410
D6300-27-2	
Original geometry	1000
Impeller № 1	1000
Impeller № 2	680
Impeller № 3	370
Impeller № 4	610
Impeller № 5	590

5. Results of the research into the influence of the impeller outlet width on its pressure characteristic

As a result of the conducted numerical research we obtained integrated pressure characteristics for impellers. Based on the obtained data about the pressure H, we built dependences of the steepness of the pressure characteristic K_H on the relative width of impeller b_2/D_2 for each group of impellers (Fig. 3).

The value of the steepness of a pressure characteristic is calculated by the standard formula:

$$K_H = \frac{H_{max} - H_{ref}}{H_{ref}}, \tag{11}$$

where H_{max} is the pressure at 0,5 Q_{ref}; H_{ref} is the pressure at Q_{ref}.

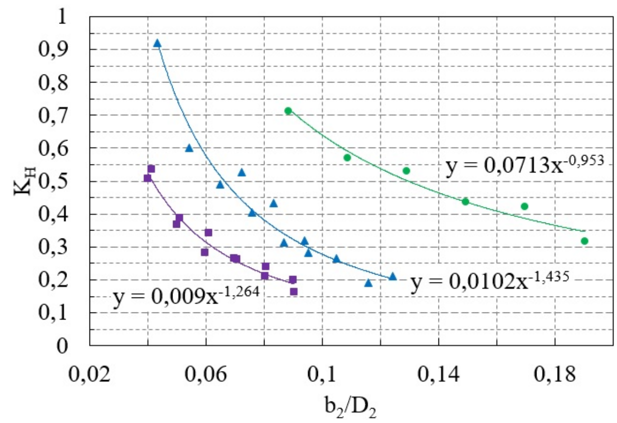

Fig. 3. Chart of dependence of the steepness K_H of a pressure characteristic on the relative width of impeller b_2/D_2, $K_H=f(b_2/D_2)$ for the pumps impellers with ■ – n_s=80...110, ▲ – n_s=110...150, ● – n_s=150...210

Fig. 3 shows the points obtained in the result of the numerical simulation of the working process of impeller pumps with different values n_s and the equation of their approximating curves. The equation that describes the dependence of the steepness of a pressure characteristic on the relative width of impeller, generally takes the form:

$$K_H = a\left(\frac{b_2}{D_2}\right)^{-k}, \tag{12}$$

where a and k are the obtained coefficients that have the following numerical values:

a=0,009, k=1,264 – for impellers of pumps with n_s=80...110;

a=0,0102, k=1,435 – for impellers of pumps with n_s=110...150;

a=0,0713, k=0,953 – for impellers of pumps with n_s=150...210.

Fig. 3 also displays the dependence, which shows that with the increase of the specific speed n_s, the value of steepness K_H increases as well. At decreasing b_2, the graph $K_H=f(b_2/D_2)$ is more sloping at n_s=150...210 than at n_s=80...150.

Using a dependence (2) and the data that were obtained as a result of a numerical experiment, we built the graphs of dependence (Fig. 4) of consumable parameter q^p on the relative width of the impeller b_2/D_2 for each group of impellers at D_2=const, n=const, β_2=const.

As we can see from Fig. 3 and Fig. 4, the nature of the dependences $K_H=f(b_2/D_2)$ and $q^p=f(b_2/D_2)$ is similar that allows making a conclusion about the identity of the influence of the width of the output impeller on q^p and K_H.

Fig. 5 shows the graph of dependence of the steepness of a pressure characteristic on a consumable parameter in the reference point $K_H=f(q^p)$ for the groups of impellers.

It is evident from Fig. 5 that in a general case the dependence $K_H=f(q^p)$ takes the form:

$$K_H = E \cdot q^p + F. \tag{13}$$

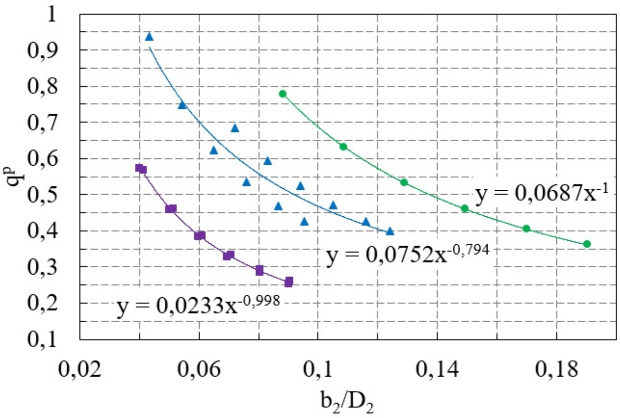

Fig. 4. Graph of dependence of consumable parameter q^p on the relative width of impeller b_2/D_2 for the pumps impellers with ■ – n_s=80...110, ▲ – n_s=110...150, ● – n_s=150...210

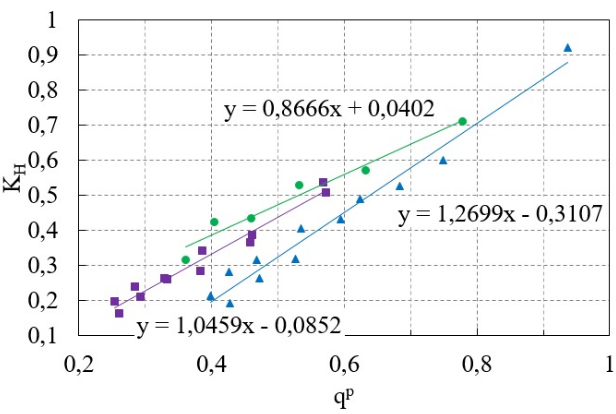

Fig. 5. Graph of dependence of the steepness of a pressure characteristic on a consumable parameter in the reference point K_H=f(q^p) for the pumps impellers with ■ – n_s=80...110, ▲ – n_s=110...150, ● – n_s=150...210

By using the equation (5), we received:

$$K_H = E \cdot \frac{60Q}{\pi^2 \cdot D_2^2 \cdot n \cdot b_2 \cdot tg\beta_2} + F, \quad (14)$$

where E and F are the obtained coefficients that have the following numerical values:

E=1,0459, F=–0,0852 – for impellers of pumps with n_s=80...110;

E=1,2699, F=–0,3107 – for impellers of pumps with n_s=110...150;

E=0,8666, F=0,0402 – for impellers of pumps with n_s=150...210.

6. Discussion of the results of the research into dependence of impeller outlet width and the steepness of a pressure characteristic

As a result of the research into the working process of impeller with n_s=80...210 of the pumps D 3200-75, D 6300-80-2, SE 2500-180-8, SE 2500-180-8 UHL 4E, D 6300-27-2, mathematical dependences between impeller outlet width and the steepness of a pressure characteristic were established for the first time.

The coefficients a, k, E and F from the equations (12) and (14) are different for impellers with different rapidity and depend on their design features.

Due to the fact that the studies were conducted only for the impellers of aforementioned pumps, to refer established dependences to other brands of pumps would not be justified. Therefore, the purpose of further research is to define the main geometric parameters of impeller, which essentially influence the coefficients a, k, E and F, to obtain universal dependences for centrifugal pumps with double-entry impellers for all types.

Received dependences are expedient to use when designing the adjustable impeller of the pumps D 3200-75, D 6300-80-2, SE 2500-180-8, SE 2500-180-8 UHL 4E, D 6300-27-2 during their modernization for new parameters of the supply Q and the pressure H with a predetermined steepness of a pressure characteristic.

7. Conclusions

1. Due to the use of numerical simulation in the software package ANSYS CFX, we received a possibility to research into the working process of 30 impellers with changing only one geometric parameter – the impeller outlet width b_2, while keeping other geometric parameters unchanged, excluding their cumulative impact on the results.

2. For the double-entry impellers with the specific speed n_s=80...210 of the pumps D 3200-75, D 6300-80-2, SE 2500-180-8, SE 2500-180-8 UHL 4E, D 6300-27-2, we established dependences between the impeller outlet width b_2 and the steepness of the pressure characteristic K_H, and also the consumable parameter q^p in the form of mathematical formulas ($K_H = a\left(\frac{b_2}{D_2}\right)^{-k}$, $K_H = E \cdot q^p + F$) and the constructed curves. The identity of the curves of these dependences indicate the similarity of the impact of the change in the impeller outlet width on K_H and q^p provided no other geometric dimensions change. The coefficients in the equations are variable and depend on the design features of the impeller.

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