У роботі визначені основні геометричні розміри, які впливають на характеристики вільновихрових насосів. Запропоновані зміни дозволили знизити гідравлічні втрати у міжлопатевих каналах робочого колеса. У результаті проведеного факторного експерименту досягнуто підвищення к.к.д. насоса на 4–5 %. Робота виконана з використанням методу чисельного дослідження. Результати підтверджені шляхом проведення фізичного експерименту

Ключові слова: вільновихровий насос, Turo, робоче колесо, чисельне дослідження, Ansys CFX, факторний експеримент

В работе определены основные геометрические размеры, которые влияют на характеристики свободновихревого насоса. Предложенные изменения позволили снизить гидравлические потери в межлопастных каналах рабочего колеса. В результате проведенного факторного эксперимента достигнуто повышение к.п.д. насоса на 4–5 %. Работа выполнена с использованием метода численного исследования. Результаты подтверждены путем проведения физического эксперимента

Ключевые слова: свободновихревой насос, Turo, рабочее колесо, численное исследование, Ansys CFX, факторный эксперимент

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INVESTIGATION OF THE IMPACT OF THE GEOMETRIC DIMENSIONS OF THE IMPELLER ON THE TORQUE FLOW PUMP CHARACTERISTICS

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1. Introduction

Torque flow pumps (TFP) are applied:

– in housing and communal services for the transportation of ground and waste water, fecal masses, sewage sludge;

– in the food industry for pumping the syrups, juices, suspensions and other liquids with elevated viscosity;

– in metallurgy and coal industries for the transportation of coal, ores, ashes, sludge;

– in the oil industry for pumping the waste of petroleum-chemical production;

– in agriculture to transport organic fertilizers, potato, vegetables;

– for pumping other liquids, which have abrasive, fibrous and easily damaged particles.

Improving energy efficiency of pumping equipment is an important task today given high energy prices.

Torque flow pumps of the «Turo» type are equipped with impellers with radial blades or with the blade outlet angle $\beta_2<90°$ (Fig. 1).

Such arrangement of the impeller blades ensures low energy efficiency indicators related to a mismatch between the blade inlet angle β_1 and the blade outlet angle β_2 from the impeller and the angle of flow arriving to the blade.

Improvement of the design of impeller without changing its overal dimensions looks promising. This makes it possible to improve existing pumps by replacing the impeller. In this case, the remaining elements of the flowing part do not need to be replaced. Thus, the improvement of energy efficiency of torque flow pumps is achieved at minimal investment cost.

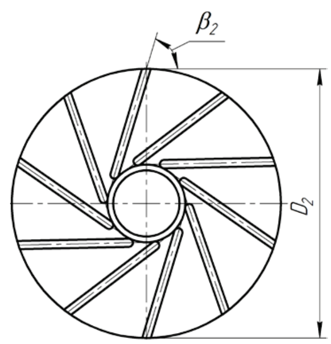


Fig. 1. Working impeller of a torque flow pump

The relevance of present work is in improving the efficiency of existing torque flow pumps at minimal investment funding.

2. Literature review and problem statement

Life cycle cost of torque flow pumps (TFP) is significantly lower than that of other types of pumps [1]. This is linked to bringing down the investment costs, related to a significant increase in the operating resource of elements of the flowing part of the torque flow pumps, in comparison with centrifugal pumps.

Authors of article [2], using a high-speed video camera and a thread of various length, showed a flow pattern in the flowing part of a torque flow pump when transporting a liquid with fibrous inclusions. As a result, a method was developed for assessing pump clogging with fibrous inclusions that are contained in the liquid.

Investigation [3] determined that an increase in the blade inlet angle β_1 at impeller input leads to lower head and pump efficiency while in the blade outlet angle β_2 – to an increase in the head and efficiency.

Paper [4] addressed the ways of improving efficiency by changing the structural elements of the impeller of torque flow pump of the «Seka» type: impeller diameter D_2, the impeller width b_2 the blade inlet angle β_1 the blade outlet angle β_2 of the impeller, as well as the number of blades z. In contrast to the article reviewed, the paper assessed mutual influence of factors, as well as the degree of influence of each factor on the pump parameters. In addition, an increase in the impeller width b_2 leads to the movement of the impeller to the pump free chamber. However, that results in a decrease in the operation resource, as well as the damage is possible to the transported products that contain easily damaged particles.

Improved parameters of TFP are attained by applying the winglets on the blades of the impeller [5]. Thus, in the workflow of the pump, the centrifugal component increases while the vortex component decreases. However, this results in a significant decrease in the operation resource of elements in the flowing pump part and possible deterioration in the quality of transported product as a result of hitting the blade of the impeller.

Investigation [6] found that in order to improve pump parameters when using an impeller blades with winglets, a gap between the impeller and the casing is required. This gap can be of conical shape with a cosine angle of 15°, or of cylindrical shape with size not less than 10 mm.

With the aim of increasing head in the pump based on a torque flow pump, a spiral pump was designed [7]. Improved head is achieved by increasing the centrifugal component in the pump workflow. A wide gap between the casing and the impeller prevents blockage of the pump.

Paper [8] proposed a design of the TFP diffuser, which enables improving efficiency of the pump by up to 4–5 %. This is achieved through the harmonization of geometrical parameters of free chamber of the pump casing with the direction of the vortex flow.

Study [9] reported a significant increase in the centrifugal pump head for flow rate modes $Q<0.5Q_{BEP}$ with the introduction of a vortex stage in the composition. Eliminating a lagging zone of Q-H characteristic for flow rate modes $Q<0.5Q_{BEP}$ made it possible to improve vibration characteristics of the pump, as well as to avoid surge.

Articles [10–12] compared pump parameters derived from numerical analysis using the Ansys CFX programming package with data obtained as a result of physical experiment. The difference in results does not exceed 5 % for the operating range, $Q=0.7–1.2Q_{BEP}$, as well as 7 % for supply modes $<0.7Q_{BEP}$, and $Q>1.2Q_{BEP}$. This does not exceed an error of experimental research methods. Thus, a numerical investigation can be applied to improve the torque flow pumps.

The need for further research is predetermined by the following. Existing studies do not make it possible to determine geometric dimensions of the impeller, which demonstrates maximal efficiency. For example, in order to improve energy efficiency of a torque flow impeller, it is recommended to increase the blade outlet angle β_2. For the same purpose, it is recommended to reduce the blade inlet angle β_1. The simultaneous use of these two recommendations leads to a significant change in the blade installation angle in the direction of fluid motion in the inter-vane channels. As a result, losses in the impeller will grow while its energy efficiency falls.

3. Research goal and objectives

The goal of present research is to identify the influence of geometrical dimensions of the impeller on the efficiency of a torque flow pump of the «Turo» type. This will make it possible to reduce hydraulic losses in the nter-vane channels.

To accomplish the set goal, the following tasks have been set:

– to derive regression equations in the natural form for the pump head and efficiency. To identify statistically significant factors influencing the pump head and efficiency. To determine the degree of impact of each factor on the pump head and efficiency in accordance with regression equations;

– to determine the effect of change in the examined structural elements of the impeller on the head and efficiency of a torque flow pump using the method of numerical investigation;

– to design an improved impeller of a torque flow pumps with elevated efficiency;

– to carry out experimental investigations in order to confirm adequacy of the undertaken series of numerical investigations.

4. Examining the impact of design of the impeller on pump characteristics

4. 1. Procedure of conducting the research

In order to improve efficiency of a torque flow pump taking into account the minimal life-cycle cost of the pump [13], it is proposed to improve design of the impeller. In this case, other design elements of the pump, in particular, casing, remain unchanged.

To conduct the investigation we employed a planning method of factorial experiment. Similar research procedure was applied in papers [14, 15].

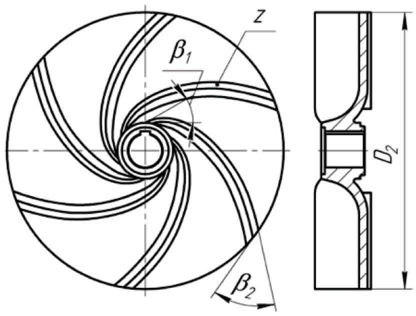
The 2^3 first order plan was accepted (Table 1). It allows us to estimate the impact of each factor on the optimization parameters. The head H and the pump efficiency η served as the optimization parameters.

Table 1

Levels and intervals of factor variation

Factor	Denotation	Interval	Factor levels		
			Primary 0	Upper +1	Lower –1
z	x_1	2	6	8	4
β_2, °	x_2	20	50	70	30
D_2, mm	x_3	5	320	325	315

Based on the review of scientific literature, we chose as the main influence factors (Fig. 2) on the optimization parameters a number of blades z, impeller diameter D_2, as well as the blade outlet angle β_2 of the impeller. Impeller diameter changed in narrow limits to enable the possibility of using the existing casing.



a *b*

Fig. 2. Influence factors on the optimization parameters: *a* — impeller in the plane; *b* — meridional section of the impeller

Adequacy check was implemented using the Fisher's *F*-criteria of adequacy.

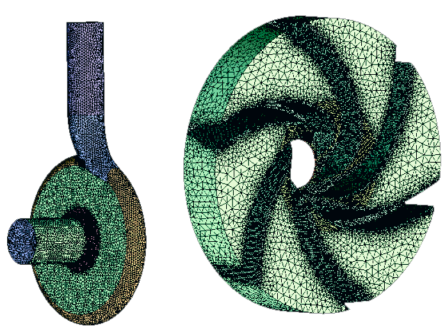
Evaluation of significance of the results obtained was performed using the Student significance criterion. In this case, we employed a 5 % significance level. This is typical for engineering calculations.

4. 2. Numerical study of the flow in a flowing part of the pump

In order to accomplish the set goals, we conducted a series of experiments based on the method of numerical investigation. The estimated model was created in the Ansys CFX programming environment.

We used water at a temperature of 25 °C as a working medium. Operating mode is turbulent. For closing the Reynolds equations we used a standard k-ε turbulence model.

The estimated region consists of two elements: stator – a flowing part of the pump, and rotor – the impeller. For each element of the workspace we constructed unstructured computational grid (Fig. 3). To simulate the flow in the boundary layer with sufficient accuracy, we isolated near solid walls a layer consisting of prismatic cells. We built a tetrahedral grid in the region of flow core for a free chamber and the impeller. The total number of elements in the computational grid is 1 million 500 thousand cells. The simulation of the flow was carried out in the stationary setting.



a *b*

Fig. 3. Computational grid: *a* — stator element; *b* — rotor element

We assigned as a boundary condition at the input of the estimated area the mass flow rate through the flowing part of the pump for a range of flow rates $Q=0.7-1.2Q_{BEP}$.

The pressure of 1 MPa served as a boundary condition at the output of the estimated region.

Given the existence of reverse flows, at the output of the estimated region of stator element we assigned «opening» as a boundary condition.

Rotation frequency for all the experiments in a series is $n=1500$ rpm.

In order to achieve convergence of the results, numerical investigation was conducted with assigning the interface of interaction between estimated regions «Frozen rotor». To refine the data, results were employed as the initial approximations when carrying out numerical investigation, setting the interface of interaction between estimated regions «Stage».

5. Results of investigation of the influence of geometrical dimensions of the impeller on the torque flow pump characteristics

In accordance with the set objectives, we propose a design of the impeller, which implies the use of profiled blades (Fig. 2). In line with recommendations [3], in order to improve energy efficiency of the pump, the blade inlet angle β_1 is chosen to be less than the blade outlet angle β_2. We propose to select the values of the above angles as the ratio of $\beta_2/\beta_1=1.5$. This will make it possible to minimize the flow tear-off regions in the impeller inter-vane channels.

The dispersion of the head parameter optimization was equal to $S^2_{yH}=4.33\cdot10^{-2}$. The dispersion of the energy efficiency parameter optimization was equal to $S^2_{y\eta}=2.33\cdot10^{-6}$. The root-mean-square deviation by head equals $S_H\{b_i\}=7.36\cdot10^{-2}$, by efficiency $S_\eta\{b_i\}=5.4\cdot10^{-6}$. Regression coefficient confidence integral by head is equal to $\Delta b_{iH}=\pm3.17\cdot10^{-1}$, by efficiency $\Delta b_{i\eta}=\pm2.32\cdot10^{-4}$.

As a result of conducting a factorial experiment, we derived regression equations by head and efficiency of a torque flow pump. Following a transition from the encoded values, regression equations are received in the natural form:

$$H=29.48+0.37z+0.96\beta_2+0.34D_2, \quad (1)$$

$$\eta=0.417-0.006\beta_2+0.012D_2-0.0025z\beta_2. \quad (2)$$

According to the model obtained, over the specified intervals of change in the impact factors (Table 1), the head of a torque flow pump grows with an increase in all the three selected factors. In this case, the greatest impact on the head growth is exerted by an increase in the blade outlet angle β_2.

The influence of change in the impeller diameter on the pump head is negligible. This is due to the narrow range over which the given factor was selected. It is obvious that at an increase in the impeller diameter the head of a torque flow pump increases. However, under operating conditions, it is required to modernize the pump with minimal investment cost.

Therefore, when replacing the impeller, it is required to ensure that the casing of a torque flow pump remains unchanged.

Increasing the number of blades leads to increased head. This is explained by an increase in energy gain, which is transferred from the impeller to the flow. This also increases losses in the impeller inter-vane channels.

According to the model received, the greatest influence on the efficiency of a torque flow pump is exerted by a change in the diameter of the impeller. At an increase in the impeller diameter, a gap between it and the casing bore reduces. This leads to a decrease in the losses.

An increase in the blade outlet angle β_2 leads to a decrease in the efficiency of a torque flow pump. The difference with the results obtained in [3, 4] is explained by the following. At a change in the blade outlet angle β_2, the blade installation angle in the blade inlet angle β_1 does not change. In the present work we proposed the dependence $\beta_2/\beta_1=1.5$. Therefore, an increase in the blade outlet angle β_2 results in the increase in the blade inlet angle β_1. This reduces the mismatch between the given angle and the angle of a fluid inleakage on the blade.

At the same time, while increasing the number of blades z and the blade outlet angle β_2, there is a certain fall in the efficiency of a torque flow pump. This is explained by increasing losses at the input and in the impeller inter-vane channels. Thus, these losses exceed the gain of energy, which is transferred from the impeller to the flow.

Adequacy check of the received model was conducted using the Fisher's criterion:

$$F_{pH} = 2.02 < F_{TH} = 6.2, \qquad (3)$$

$$F_{p\eta} = 2.02 < F_{T\eta} = 6.2, \qquad (4)$$

where F_{pH}, $F_{p\eta}$ are the estimated values of the Fisher's criterion by head and efficiency; F_{TH}, $F_{T\eta}$ are the theoretical values of the Fisher's criterion by head and efficiency.

The estimated values of the Fisher's criterion for head and efficiency are lower than those theoretical. Accordingly, the resulting model is adequate.

In order to refine the extent of influence of the structural elements of impeller on the characteristics of a torque flow pump, we conducted an additional series of studies with one variable impact factor by the method of numerical analysis.

We examined the influence of the blade installation angle in the blade outlet angle β_2 on the pump operating parameters (Fig. 4): head H (Fig. 4, *a*), efficiency η (Fig. 4, *b*), flow rate Q (Fig. 4, *b*). The given parameter varied ranging from 30° to 50°. To minimize hydraulic losses due to the increased difference between the installation angle in the blade inlet angle β_1 and the blade outlet angle β_2, the value of the given angles was chosen as a ratio of $\beta_2/\beta_1=1.5$.

The head of a torque flow pump increases with an increase in the blade outlet angle β_2 in a non-linear way. Parabolic curve of head dependence on the value of this angle is explained by the following. An increase in the blade outlet angle β_2 also increases the blade outlet angle β_1. In this case, the losses at the input and in the inter-vane impeller channels increase faster than the gain in energy that is transferred from the impeller to the fluid.

The highest value of the torque flow pump efficiency was achieved for the impeller with the blade outlet angle $\beta_2=50°$. In this case, the blade inlet angle $\beta_1=33°$.

A separate series of investigations implied identifying a degree of influence of the change in the number of blades on the pump operating parameters (Fig. 5): head H (Fig. 5, *a*), efficiency η (Fig. 5, *b*), flow rate Q (Fig. 5, *c*).

An increase in the number of blades z leads to increased head of a torque flow pump. This occurs as a result of an increase in energy, which is transferred from the impeller to the fluid flow. The pump head increases with an increase in the number of blades in a nonlinear way. Parabolic curve and reduced head gain at an increase in the number of blades is explained by a decline in energy that is transmitted from the impeller to the flow. In this case, the losses at the input of the impeller and in the inter-vane channels grow.

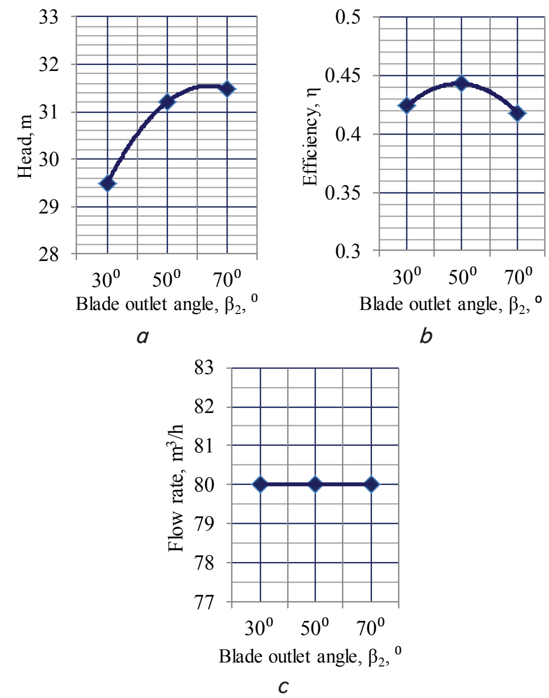


Fig. 4. Effect of the blade outlet angle β_2 on the torque flow pump operating parameters (z=6; D_2=325 mm): *a* – on head H; *b* – on efficiency η; *c* – on flow rate Q

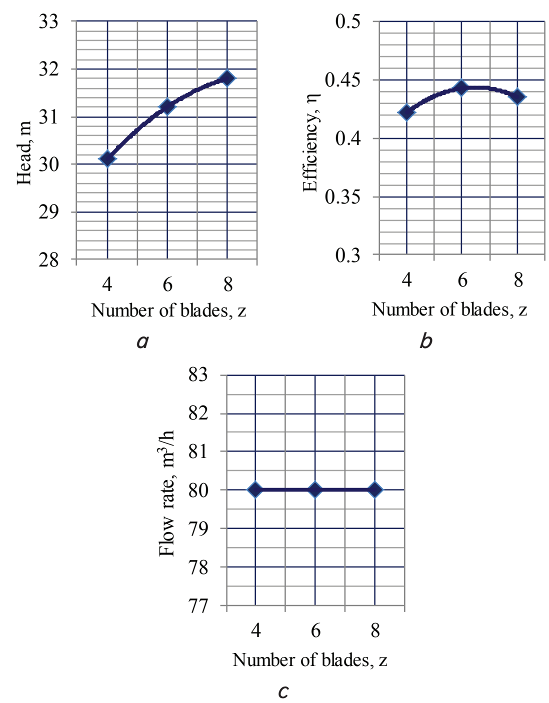


Fig. 5. Effect of the number of blades z on the torque flow pump operating parameters (β_1=33°; β_2=50°; D_2=325 mm): *a* – on head H; *b* – on efficiency η; *c* – on flow rate Q

As Fig. 4, *c*, 5, *c* show, the flow rate of a torque flow pump is not affected by geometrical dimensions of the impeller. Thus, the best efficiency point of the *Q-H* characteristic does not depend on the design of the impeller. Position of the best efficiency point is determined by the geometrical dimensions of the pump free chamber.

As a result of investigation conducted, we chose a torque flow pump impeller with the highest energy efficiency indicators. It has the following values of the selected geometrical dimensions: D_2=325 mm; z=6; β_2=50°. In this case, the blade installation angle in the plane at impeller output is β_1=33°.

The existing impeller employs direct blades with the blade outlet angle β_2=80° (Fig. 1). The number of blades in the existing impeller z=10.

The torque flow pump characteristics using the existing and the new impeller are shown in Fig. 6. As a result of the research undertaken, we achieved a 4–5 % increase in the efficiency.

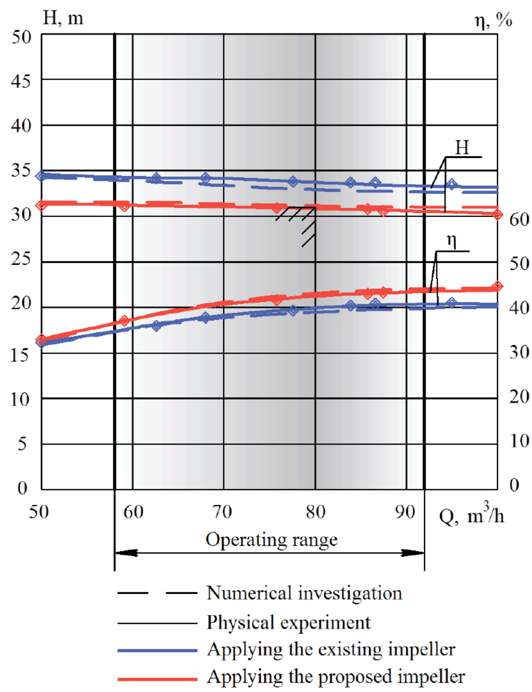


Fig. 6. Characteristics of the torque flow pump

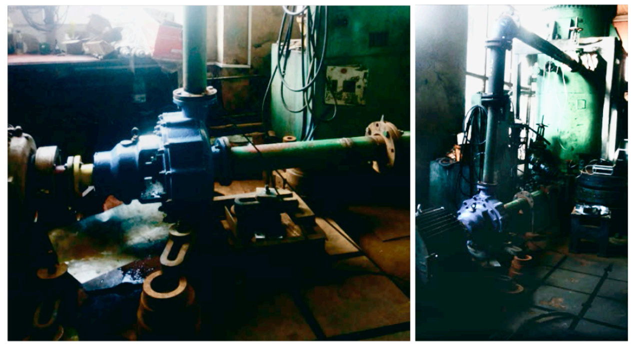
The results obtained were confirmed in practice. We conducted physical experiment using a test stand.

The physical experiment was performed for a torque flow pump with flow rate Q=80 m^3/h with the use of the current (Fig. 7, *a*) and the new (Fig. 7, *b*) impeller.

a *b*

Fig. 7. Impeller: *a* – existing; *b* – new

When conducting a physical experiment, we employed a torque flow pump (Fig. 8, *a*), as well as a test stand, from the Department of Applied Hydro- and aeromechanics of Sumy State University, Sumy, Ukraine (Fig. 8, *b*).



a *b*

Fig. 8. Pumping unit design:
a – tested torque flow pump; *b* – test stand

As a result of physical experiment, we obtained *Q-H* characteristics using the existing and the new impeller (Fig. 6).

The difference in results between numerical investigation and physical experiment does not exceed 5 % for the flow rate of the operating range. This value is within the limits of error for experimental research methods.

6. Discussion of results of the study of influence of geometrical dimensions of the impeller on the characteristics of a torque flow pump

The improvement in the torque flow pump efficiency was achieved as a result of alignment between the blade installation angles and the angles of a fluid inleakage. In addition, losses in the inter-vane impeller channels decreased through a reduction in the flow tear-off zones.

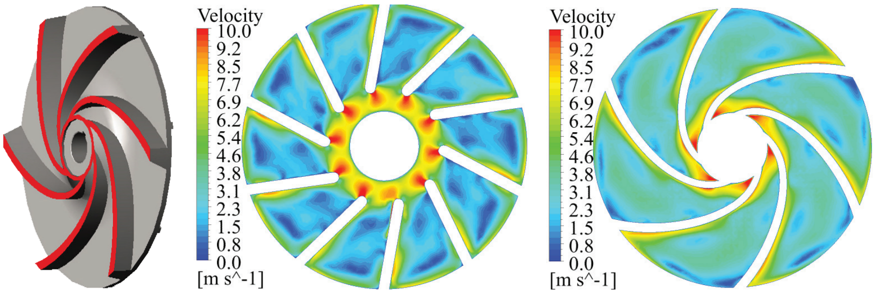
Fig. 9 shows velocity distribution in the inter-vane channels near its rim (Fig. 9, *a*) for the torque flow pumps that employ the existing (Fig. 9, *b*), and the new (Fig. 9, *c*), impeller.

It is obvious that in the new impeller the angle of fluid inleakage is aligned with the blade inlet angle β_1. This is not observed when using the existing impeller. Thus, the proposed design of the impeller makes it possible to reduce losses at its input by reducing the resistance of the input blade edge.

Fig. 10 shows velocity distribution in the inter-vane impeller channels in the middle of the width of its blades (Fig. 10, *a*) for the torque flow pumps using the existing (Fig. 10, *b*), and the new (Fig. 10, *c*), impeller.

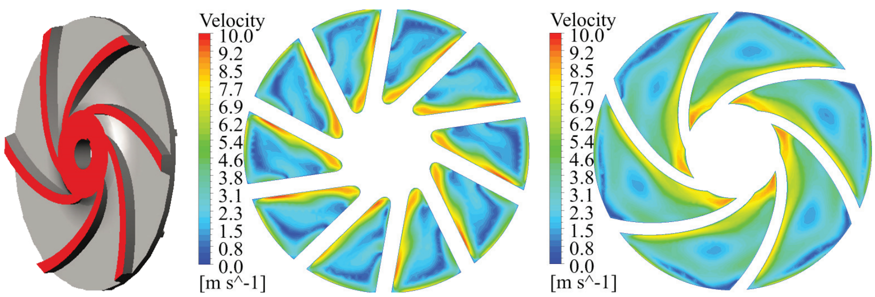
Fig. 11 shows velocity distribution in the impeller inter-vane channels near the disc (Fig. 11, *a*) for the torque flow pumps using the existing (Fig. 11, *b*), and the new (Fig. 11, *c*), impeller.

Fig. 10, 11 show that in the inter-vane channels of the new impeller velocity distribution is more uniform than in the existing impeller. There are no flow tear-off regions. This makes it possible to considerably reduce hydraulic losses in the inter-blade channels of the impeller. The improvement in the efficiency of a torque flow pump is thus achieved.



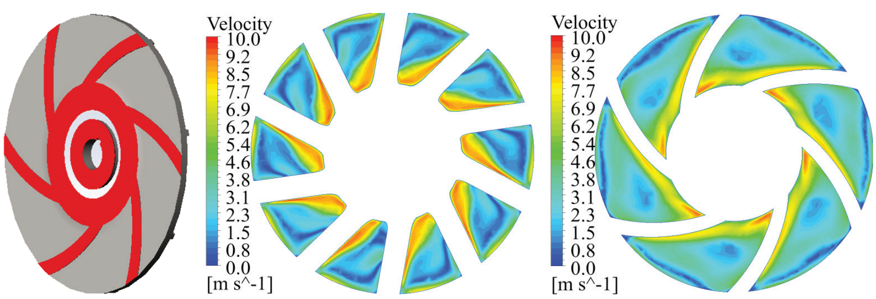
a *b* *c*

Fig. 9. Velocity distribution at the rim of the blade (Q= 80 m^3/h): *a* – location of the cutting plane; *b* – velocity distribution in the inter-vane channels of the existing impeller; *c* – velocity distribution in the inter-vane channels of the new impeller



a *b* *c*

Fig. 10. Velocity distribution in the middle of the inter-blade channels (Q= 80 m^3/h): *a* – location of the cutting plane; *b* – velocity distribution in the inter-vane channels of the existing impeller; *c* – velocity distribution in the inter-vane channels of the new impeller



a *b* *c*

Fig. 11. Velocity distribution near the disc (Q= 80 m^3/h): *a* – location of the cutting plane; *b* – velocity distribution in the inter-vane channels of the existing impeller; *c* – velocity distribution in the inter-vane channels of the new impeller

7. Conclusions

1. Regression equations for the head and efficiency revealed significant impact of the blade design on the operating parameters of a torque flow pump of the «Turo» type. In terms of head, statistically significant factors include: the number of blades z, the blade outlet angle β_2, as well as the outer impeller diameter D_2. In all cases, at an increase in the factors, the pump head increases. In terms of efficiency, statistically significant factors are: the blade outlet angle β_2 and the impeller outer diameter D_2. Statistically significant is also the mutual influence of the number of blades z and the blade outlet angle β_2. With an increase in the impeller diameter D_2, the efficiency improves.

2. As a result of a series of numerical investigations conducted, we revealed the influence of change in the examined structural elements of the impeller on the torque flow pump characteristics. An increase in the blade outlet angle β_2 to 50° leads to improved efficiency of the pump. The same result is obtained at a simultaneous increase in this angle and the number of blades z. A further increase in the blade outlet angle β_2 to 70° leads to a decrease in the pump efficiency. Such difference between the results of previous studies is explained by the following. In the present work, the blade inlet angle β_1 changes by dependence β_2/β_1=1.5. This contributes to reducing the flow tear-off regions in the impeller inter-vane channels. An increase in the blade inlet angle β_1 leads to an increased mismatch between the given angle and the angle of a fluid inleakage to the blade. As a result, hydraulic losses at the input to the impeller increase. A special feature of the present work is the examination of mutual influence of the the blade inlet angle β_1 and the blade outlet angle β_2 on the torque flow pump characteristics.

3. We determined values of the examined structural elements of the impeller, at which the largest efficiency of the pump is reached: impeller diameter D_2=325 mm, number of blades z=6, the blade outlet angle β_2=50°. In this case, the blade inlet angle β_1=33°. By changing the impeller design, we achieved a 4–5 % increase in the efficiency as a result of reducing hydraulic losses in the impeller inter-vane channels.

4. The physical experiment confirmed adequacy of conducting a numerical investigation. The difference in results between the numerical investigation and physical experiment study does not exceed 5 % for the flow rate of the operating range, which does not exceed the error limit in experimental research methods.

References

1. Kotenko, A. Rationalisation of Ukrainian industrial enterprises in a context of using torque flow pumps on the basis of valuation of the life cycle of pumping equipment [Text] / A. Kotenko, V. Herman, A. Kotenko // Nauka i Studia. – 2014. – Issue 16 (126). – P. 83–91. – Available at: http://essuir.sumdu.edu.ua/bitstream/123456789/38769/3/kotenko_poland1.PDF

2. Kudo, H. Study on clogging mechanism of fibrous materials in a pump by experimental and computational approaches [Text] / H. Kudo, T. Kawahara, H. Kanai, K. Miyagawa, S. Saito, M. Isono et. al. // IOP Conference Series: Earth and Environmental Science. – 2014. – Vol. 22, Issue 1. – P. 012011. doi: 10.1088/1755-1315/22/1/012011
3. Gerlach, A. Design Parameters of Vortex Pumps: A Meta-Analysis of Experimental Studies [Text] / A. Gerlach, P. Thamsen, S. Wulff, C. Jacobsen // Energies. – 2017. – Vol. 10, Issue 1. – P. 58. doi: 10.3390/en10010058
4. Gao, X. Optimization design and test of vortex pump based on CFD orthogonal test [Text] / X. Gao, W. Shi, D. Zhang et. al. // Nongye Jixie Xuebao/Transactions of the Chinese Society for Agricultural Machinery. – 2014. – Vol. 45, Issue 5. – P. 101–106.
5. Gerlach, A. Experimental Investigation on the Performance of a Vortex Pump using Winglets [Text] / A. Gerlach, P. Thamsen, F. Lykholt-Ustrup // ISROMAC 2016. International Symposium on Transport Phenomena and Dynamics of Rotating Machinery. – 2016. – Available at: http://isromac-isimet.univ-lille1.fr/upload_dir/finalpaper/181.finalpaper.pdf
6. Chervinka, M. Computational Study of Sludge Pump Design with Vortex Impeller [Text] / M. Chervinka // Engineering Mechanics. – 2012. – P. 191–201. – Available at: http://www.engmech.cz/2012/proceedings/pdf/087_Cervinka_M-FT.pdf
7. Ishii, K. Hydrodynamic characteristics of the helical flow pump [Text] / K. Ishii, K. Hosoda, M. Nishida, T. Isoyama, I. Saito, K. Ariyoshi et. al. // Journal of Artificial Organs. – 2015. – Vol. 18, Issue 3. – P. 206–212. doi: 10.1007/s10047-015-0828-y
8. Krishtop, I. Creating the flowing part of the high energy-efficiency torque flow pump [Text] / I. Krishtop // Eastern-European Journal of Enterprise Technologies. – 2015. – Vol. 2, Issue 7 (74). – P. 31–37. doi: 10.15587/1729-4061.2015.39934
9. Mihalic, T. Improving centrifugal pump by adding vortex rotor [Text] / T. Mihalic, S. Medic, Z. Kondic // Tehnicki vjesnik. – 2013. – Vol. 20, Issue 2. – P. 305–309. – Available at: http://bib.irb.hr/datoteka/628405.tv_20_2013_2_305_309.pdf
10. Krishtop, I. Numerical Approach for Simulation of Fluid Flow in Torque Flow Pumps [Text] / I. Krishtop, V. German, A. Gusak, S. Lugova, A. Kochevsky // Applied Mechanics and Materials. – 2014. – Vol. 630. – P. 43–51. doi: 10.4028/www.scientific.net/amm.630.43
11. Kotenko, A. The computational model of the emergence and development of cavitation in torque flow pump [Text] / A. Kotenko, L. Nikolaenko, S. Lugova // 4th International Meeting on Cavitation and Dynamic Problems in Hydraulic Machinery and Systems. – 2011. – P. 87–94. – Available at: http://essuir.sumdu.edu.ua/bitstream/123456789/20799/1/1003.pdf
12. Steinmann, A. Numerical and experimental investigations of the unsteady cavitating flow in a vortex pump [Text] / A. Steinmann, H. Wurm, A. Otto // Journal of Hydrodynamics, Ser. B. – 2010. – Vol. 22, Issue 5. – P. 324–329. doi: 10.1016/s1001-6058(09)60213-4
13. Nault, J. Lifecycle Assessment of a Water Distribution System Pump [Text] / J. Nault, F. Papa // Journal of Water Resources Planning and Management. – 2015. – Vol. 141, Issue 12. – P. A4015004. doi: 10.1061/(asce)wr.1943-5452.0000546
14. Kim, J.-H. Design technique to improve the energy efficiency of a counter-rotating type pump-turbine [Text] / J.-H. Kim, B.-M. Cho, S. Kim, J.-W. Kim, J.-W. Suh, Y.-S. Choi et. al. // Renewable Energy. – 2017. – Vol. 101. – P. 647–659. doi: 10.1016/j.renene.2016.09.026
15. Kim, S. High performance hydraulic design techniques of mixed-flow pump impeller and diffuser [Text] / S. Kim, K.-Y. Lee, J.-H. Kim, J.-H. Kim, U.-H. Jung, Y.-S. Choi // Journal of Mechanical Science and Technology. – 2015. – Vol. 29, Issue 1. – P. 227–240. doi: 10.1007/s12206-014-1229-5