Numerical study of the centrifugal contra rotating blade system

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Abstract. The manuscript presents the results of a numerical study of a centrifugal contra rotating blade system based on a low-speed impeller (ns = 65). They were carried out to assess the possibility of using a contra rotating blade mesh operating in conjunction with a standard centrifugal impeller as a method of intensifying the transmission of energy to a fluid. The designs of the systems under consideration and their pressure and energy characteristics are presented.

1. Introduction
It is known that one of the main problems in the industry is the significant energy consumption of pumping equipment. In various industries, pumps consume 25-60% of the total energy expended [1]. “The Directive on the requirements for the design of energy-using products” adopted by the European Parliament in 2005 calls for reduction of energy consumption in Europe by 2020 by 20% [2-4].

Based on this Directive, it is necessary to search for new ways of developing systems that transfer energy to the fluid. Taking into account the modern level of development of the pump construction, a promising way of implementing such plans consists in increasing the intensity of energy transfer from the pump to the flow of fluid. It should be noted that this will inevitably lead to a more complex workflow [5]. However, the proposed approach is an important and urgent task, since it basically has the potential to create new groups of pumping systems possessing unique qualities.

2. Analysis of the literature and statement of the problem
A theoretical description of the operation of a centrifugal contra rotating blade system is given in the article [6], and it is shown that it can receive significantly higher head pressures compared to a single impeller. The size of the contra rotating part II (Fig. 1) can be much smaller than the diameter of the impeller (rotor) I. In this case, naturally, the rotor and the contra rotor rotate in opposite directions.

Attractiveness of the contra rotating blade system consists in the fact that the first rotor, in accordance with the basic equation of hydraulic machines, creates a head HI and a significant circulation at the output (+ G₂) in the direction of its rotation. This circulation is negative at the inlet for the second rotor, which rotates in the opposite direction (G₃ = - G₂). Thus, hydrodynamic conditions are created for the force interaction of the flow with a rigid surface of the blade, which moves towards the fluid. That is, for the blades of the second rotor, conditions for a substantially greater force interaction are created, which for the theoretical head of the second rotor can be determined as H₀II = H₁I (+ω₀/2πg) [G₄ - (− G₂)].
It should be noted that today there is experience in developing the contra rotating blade systems in the aviation and shipbuilding industry, in the creation of axial pumps for turbines and fans, in the field of alternative energy [7-9]. In this case, there are actually no attempts to use counter-resistance in centrifugal systems. A natural explanation for this is the significant complication of the pump design, the inevitable growth of volumetric losses associated with a significant increase in pressure.

However, a naturally large number of constructive difficulties arising in the development of the concept of a new class of technical systems cannot be an obstacle to further progress. European organization of pumping equipment manufacturers EuroPump ascertains that the potential solution of the energy problem of pumping equipment at hierarchical levels “pump” and “pumping unit” has almost exhausted itself [10]. In this case, we can talk about the actual lack of opportunities for the development of many types of pumps of customary designs according to the laws of the development of technical systems [11]. This also confirms the need to search for new opportunities for the development of the pumping industry.

Thus, taking into account the above, we are faced with an urgent problem of searching for and developing methods for intensifying the process of energy transfer to a fluid. As such a method, it is proposed to study the centrifugal contra rotating blade system.

3. Purpose and objectives of the study
The purpose of this work is to conduct numerical studies of the centrifugal contra rotating blade system and to obtain their pressure and energy characteristics. To achieve this goal, it is necessary to solve the following tasks:
- to propose the design of a blade system of a counter rotor and to develop 3D models of the fluid in the centrifugal contra rotating blade system;
- to perform numerical studies of contra rotating structures, to obtain their pressure and energy characteristics.

It should be noted that in this article the task of developing constructive recommendations is not posed. At this stage of the study of centrifugal contra rotating blade systems, the authors consider it important to quantify the range of their parameters.

4. Materials and methods of research
The university version of software product ANSYS CFX was used, based on the method of numerical solution of the fundamental hydromechanics laws [12, 13]. It should be noted that ANSYS CFX was repeatedly tested in solving the problems of pump construction, and the discrepancy between the results of numerical and physical modeling does not exceed 5%; therefore, this software product is suitable for solving the stated research problem.
Modeling of the turbulent flow of a fluid flow was performed using the Reynolds equations (i, j = 1-3):
\[
\frac{\partial}{\partial t} (\rho u_i) + \frac{\partial}{\partial x_j} (\rho u_i u_j) + \frac{\partial}{\partial x_j} (\rho u_i u_j) = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left[ \mu \frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_j} \right] + f_i,
\]
where \( \bar{u}_i, \bar{u}_2, \bar{u}_3 \) — averaged for time values of speeds; \( \bar{u}'_i, \bar{u}'_2, \bar{u}'_3 \) — pulsating velocity components.

To close the Reynolds equations, we used k-ε turbulence models. Using this model, the system of fluid motion equations is supplemented by two differential equations describing the transport of the kinetic energy of turbulence \( k \) and the dissipation rate \( \varepsilon \) (2), (3).
\[
\frac{\partial}{\partial t} (\rho k) + \frac{\partial}{\partial x_j} (\rho u_j k) = \frac{\partial}{\partial x_j} \left( \Gamma_k \frac{\partial k}{\partial x_j} \right) + P_k - \rho \varepsilon,
\]
\[
\frac{\partial}{\partial t} (\rho \varepsilon) + \frac{\partial}{\partial x_j} (\rho u_j \varepsilon) = \frac{\partial}{\partial x_j} \left( \Gamma_\varepsilon \frac{\partial \varepsilon}{\partial x_j} \right) + \frac{\varepsilon}{k} (C_{\mu} P_k - \rho C_{\varepsilon} \varepsilon),
\]
where \( P_k = -\rho u_i' u_i' \frac{\partial u_i}{\partial x_j} \) — energy-exponential term k.
\[
\Gamma_k = \mu + \frac{\mu_t}{\sigma_k}, \quad \Gamma_\varepsilon = \mu + \frac{\mu_t}{\sigma_\varepsilon}.
\]

The parameters \( \varepsilon \) and \( \mu_t \) are determined as follows:
\[
\varepsilon = \frac{\rho}{\mu} \left( \frac{\partial u_i'}{\partial x_j} \right)^2, \quad \mu_t = \rho C_{\mu} \frac{k^2}{\varepsilon}.
\]

For numerical investigation using the ICEM CFD mesh generator, unstructured meshes of a 3D fluid model in impellers were constructed. To describe the boundary layer near the solid walls, eight layers of prismatic cells were created. A preliminary study was carried out for mesh independence.

After generating the mesh, the initial data was set in the preprocessor. As a boundary condition, a mass flow was set at the entrance to the calculation region, and the static pressure at the outlet. For all solids, the condition that the velocity be zero is given. The roughness of the walls was adopted as Ra 3.2.

The value of the variable \( y^+ \), which characterizes the mesh thickening near the walls, was within 20 < \( y^+ < 100 \) units, which corresponds to the recommendations given in the user manual [13].

5. Results of the numerical study
We will take the parameters of the impeller of the intermediate stage of the pump CNS 180-1900 as the reference standard for comparison of the results of the study, and denote it as PK No.1. The 3D fluid model of the impeller is shown in Figure 2. a. As a basis for the design of the counter rotor, the blade system of the directing apparatus of the intermediate stage of the pump CNS 180-1900 (Fig. 2 b) is used as a well-proven blade system that senses the flow at the outlet from the above impeller.

Figure 3 shows the 3D models of fluid in counter rotors, which were investigated in conjunction with the reference impeller in accordance with the scheme of Figure 1. These structures were
designated as PK No.2 - PK No.4 (Fig. 3a-c). Peculiarities of the choice of designs of counter rotors of schemes PK No.3 and PK No.4 will be discussed below. Two designs of the blade system of J- and C-shaped counter rotors with $Z_{\text{counter rotor}} = 25$ were investigated for comparison (schemes PK No.5 and No.6, respectively – Fig. 3 d-e).

Thus, numerical studies were carried based on the impeller ($ns = 65$) and the guiding apparatus of the intermediate stage of the pump CNS 180-1900 with the following parameters:

$Z_{\text{rotor}} = 7$, $Z_{\text{counter rotor}} = 6$, $Q = 180 \text{ m}^3/\text{h}$, $n = 3000 \text{ rev/min}$ at $Q = 0.5; 0.7; 0.9; 1; 1.1; 1.2$.

For the estimation of the parameters, it is convenient to use the ratio of the dimensionless pressure factors ($\psi$), power ($\mu$), feed ($\varphi$) and efficiency ($\eta$) of the systems under consideration to the dimensionless coefficients of these parameters at the point of maximum efficiency ($\psi_0$, $\mu_0$, $\varphi_0$, $\eta_0$):

$$\psi = \frac{2gH}{u^2}; \quad \varphi = \frac{4Q}{\pi D^2 u_2}; \quad \mu = \frac{\varphi \psi}{\eta} = \frac{8N}{\rho \pi D^2 u_2^3 \eta}.$$
Distribution of the relative velocities for the schemes PK No.2 – PK No.6 is shown in Figure 4 a-e, respectively. The results of the calculations are shown in Figure 5, changes in pressure parameters, power and efficiency relative to the reference impeller at nominal feed ($Q = 180 \text{ m}^3/\text{h}$) are presented in Table 1.

Figure 4. Relative velocity distribution.
Figure 5. Pressure (a), power (b) and performance characteristics (c) of the schemes under study.
| PK scheme | ΔH, % | ΔN, % | Δη, % |
|-----------|-------|-------|-------|
| No.2      | +142  | +237  | -28   |
| No.3      | +182  | +274  | -24.4 |
| No.4      | +175  | +264  | -24.4 |
| No.5      | +163  | +488  | -54.9 |
| No.6      | +253  | +744  | -58.5 |

6. Discussion of results

The integral characteristics of the investigated centrifugal contra rotating blade systems, obtained as a result of the numerical study, confirmed the assumption that such systems can significantly intensify the process of energy transfer from working parts to the flow of fluid. Increase in pressure in the system under consideration is many times greater than the results of all other possible methods [14]. Let's pay attention to the fact that all results are received in comparison with parameters of one impeller. To compare the parameters of the stages, it is natural to conduct a study of the centrifugal contra rotating blade system in combination with the outlet device.

At the same time, it is impossible not to note the negative trend of a significant decrease in the efficiency of the system. As can be seen from Figures 4a and 4b (schemes PK No.2 and No.3), a vortex zone is observed on the rear side of the contra rotating blades. An attempt to change the design of this blade mesh, introduced by solving the inverse problem [15], was designed to match the position of the blade with the direction of flow in the impeller after the flow turning zone, eliminating the vortex region (PK No.4). This measure did not give a positive result due to the increase in hydraulic losses due to the lengthening of the blade.

Thus, we can say that the responsibility for the decrease in efficiency is a different process, most likely directly related to the increase in the pressure of the system. It can be assumed that in the area of transition from the rotor to the counter rotor there is an intense vortex responsible for a significant increase in the pressure and, at the same time, a noticeable decrease in efficiency. It should be noted that similar vortexes cause the working process of a number of pumps - free-vortex, rotor-vortex, labyrinth-vortex [1,16,17].

This fact can be explained by the regularity of the development of technical systems, which consists in a more complex working process of the machine, its lower efficiency [5]. At the same time, the counter-current systems, based on the nature of their work, naturally require increasing the drive capacity.

Confirmation of these assumptions is the results of the study of schemes PK No.5 and especially PK No.6. When using a counter rotor with C-shaped blades, it was possible to achieve the highest degree of intensification of energy transmission (Tab. 1). However, the vortex structures in this case filled almost all of its flowing part (Fig. 4e) reducing the efficiency of the system by more than 50% compared to the reference impeller.

However, the results obtained make it possible to talk about the prospectivity of the study of the centrifugal contra rotating blade systems in terms of improving the blade system of the counter rover, as well as taking into account the volume losses and in combination with the discharge devices. In particular, the study of the design of a pump with a possible rotating guide device.
7. Conclusions

Studies of the centrifugal contra rotating blade system were carried out based on the impeller and the guiding apparatus of the pump CNS 180-1900 using the university version of ANSYS CFX software. In total, 5 designs of the counter rotor working with an unchanged rotor—a reference impeller—were studied.

The numerical investigation provided the pressure and energy characteristics of these systems. The results were compared with the parameters of the reference impeller.

For systems with a blade mesh of the counter-rotor, based on the design of the guiding apparatus of the pump CNS 180-1900, it was possible to achieve an increase in pressure by 142-182% with the decrease in efficiency by 24.4-28%.

When using a J-shaped blade system, the head pressure was 163%, which is comparable to the result of the previous schemes; however, the efficiency decrease was 55%.

The use of the centrifugal contra rotating blade system with a C-shaped blade mesh of the counter rotor showed the maximum increase in the pressure of the system - 253% with the decrease in efficiency by 58.8%.

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