The effect of design parameters of the closed type regenerative pump the energy characteristics

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Abstract. The article deals with the application area and features of the regenerative pumps. The main disadvantages of the existing methods for the design of the vortex pump flow part are given. Based on the nature of the flow in the original flow part, calculated according to the classical method, a number of design parameters have been selected and a parametrized 3-D model of a closed type vortex pump has been developed on the base of it, which allows changing parameters together and separately in a wide range. Method for determining the influence of a single constructive parameter on energy characteristics using computational fluid dynamics methods is described in the article. A complex optimization of the flow section of the closed type regenerative pump using the LP-τ search method in the STAR-CCM+ software package was carried out.

1. Introduction

The application area of regenerative pumps is wide enough. They are used in the chemical industry for supplying reagents, for pumping volatile liquids at gas-dispensing stations, in public utility pumping facilities, instead of liquid-ring pumps as vacuum pumps and low-pressure compressors, on vessels for supplying industrial and drinking water, and in some other cases. For regenerative pumps, the following parameters are usually characteristic: pump flow up to 720 l/min, pump head up to 250 m, and overall efficiency does not exceed 35-38% [1]. A regenerative pump in comparison with centrifugal pump with the same dimensions and shaft rotation frequency has the head 4-9 times larger, and its design is technologically simpler and cheaper. The regenerative pump is not able to pump highly viscous liquids due to a sharp drop in pressure. It is also impossible to pump liquids with solid inclusions, since the rapid wear of the walls of the end and radial gaps leads to an increase in overflows and, accordingly, to a drop in pressure and efficiency [2].

The principle scheme of the closed type regenerative pump is shown at figure 1 The impeller 1 usually has radial or inclining blades. The impeller is in a cylindrical housing with a concentric channel starting from the suction chamber and ending in the pressure chamber. The channel is shut down by a closing dike 2, which performs the function of sealing between the suction and pressure chambers [3-6].

Nowadays there are several classic methods for regenerative pumps analysis, the main representatives of which are: O.V. Baibakov, B.I. Nahodkin, K.P. Fleuderer, V.V. Shaumyan and a number of other scientists [1-4]. All of these techniques either require a large amount of calculations, or are applicable to narrow classes of pumps, or are based on statistics from the regenerative pumps existing at that time. In addition, now there is no comprehensive method for optimizing the flow part of regenerative pumps based on hydrodynamic modeling methods. Since the increased requirements for energy efficiency have been recently made against pumps [7–10], the question of increasing the efficiency of regenerative pumps has become particularly acute.

Therefore companies set BMSTU department "Hydromechanics, Hydromachines and Hydro-Pneumoautomatics" a task of developing a parametrized model of a closed type regenerative pump and determining which design parameters affect the energy characteristics of the regenerative pump mostly using modern hydrodynamic modeling methods.

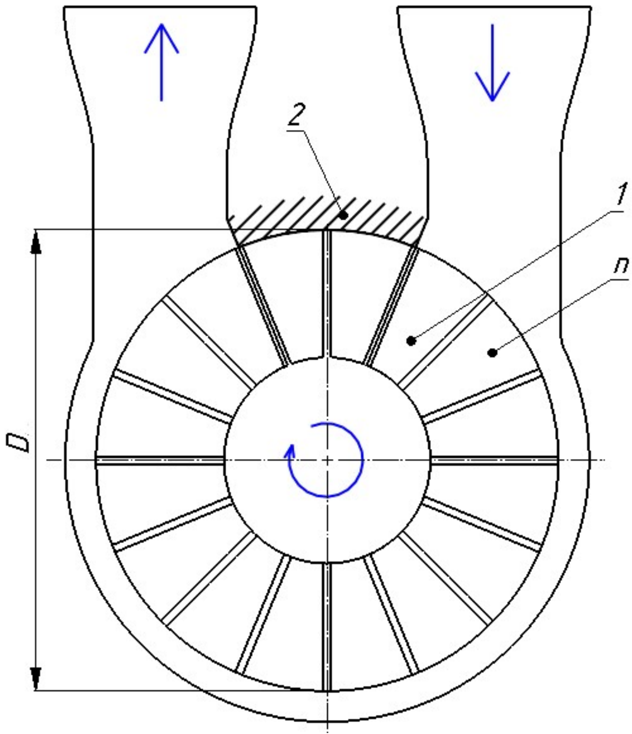

Figure 1. The principle scheme of the closed type regenerative pump.

This study had been conducted in the framework of the development of a double-stage centrifugal-regenerative pump for the following parameters:

Pressure H = 54 m;

Feed Q = 32 m^3/h;

Shaft speed n=2900 rpm.

2. Methods

The method of numerical modelling is based on solving discrete analogs of the basic hydrodynamic equations [11-14]. In the case of an incompressible fluid model (ρ = const), this is:

Mass conservation equation (continuity equation)

$$\frac{\partial \bar{u}_j}{\partial x_j} = 0$$

where

$\bar{u}_j$ – the averaged value of the fluid velocity in the projection on the j-th axis (j = 1, 2, 3);

The equation of momentum conservation (Reynolds averaging):

$$\rho\left[\frac{\partial U_i}{\partial t}+U_i\frac{\partial U_i}{\partial x_j}\right]=-\frac{\partial P}{\partial x_i}+\frac{\partial}{\partial x_i}\left[\tilde{T}_{ij}^{(v)}-\rho\langle u_i u_j\rangle\right],$$

where
U, P – averaged speed and pressure;
$\tilde{T}_{ij}^{(v)}=2\mu\tilde{S}_{ij}$ – viscous stress tensor for incompressible fluid;
$\tilde{S}_{ij}=\frac{1}{2}\left[\frac{\partial\tilde{u}_i}{\partial x_j}+\frac{\partial\tilde{u}_j}{\partial x_i}\right]$ – instant strain rate tensor;
$\rho\langle u_i u_j\rangle$ – Reynolds stresses.

The introduction of the Navier—Stokes equation, Reynolds-averaged, makes the system of equations non-closed, since additional unknown Reynolds stresses appear. A semiempirical k—ω SST model of turbulence has been used for solving the system in this task, which introduces the necessary additional equations: the transfer equations for the kinetic energy of turbulence and the relative dissipation rate of the energy:

$$\frac{\partial k}{\partial t}+U_j\frac{\partial k}{\partial x_j}=P_k-\beta k\omega+\frac{\partial}{\partial x_j}\left[\left(\upsilon+\sigma_k\upsilon_T\right)\cdot\frac{\partial k}{\partial x_j}\right]$$

$$\frac{\partial\omega}{\partial t}+U_j\frac{\partial\omega}{\partial x_j}=\alpha\cdot S^2-\beta\cdot\omega^2+\frac{\partial}{\partial x_j}\left[\left(\upsilon+\sigma_\omega\upsilon_T\right)\cdot\frac{\partial\omega}{\partial x_j}\right]+2\cdot(1-F_l)\cdot\sigma_{\omega2}\cdot\frac{1}{\omega}\cdot\frac{\partial k}{\partial x_j}\cdot\frac{\partial\omega}{\partial x_j}$$

The flow part is divided into a set of finite cells, for each of which discrete analogues of continuous equations are composed. The combination of all discrete analogs has formed a closed system of algebraic equations.

To obtain a parametric model, as a first approximation the flow part calculated by the method of B.I. Nahodkin, was used. The modelling results are shown at figures 2, 3. Head is H = 46.2 m, hydraulic efficiency η = 39.6%.

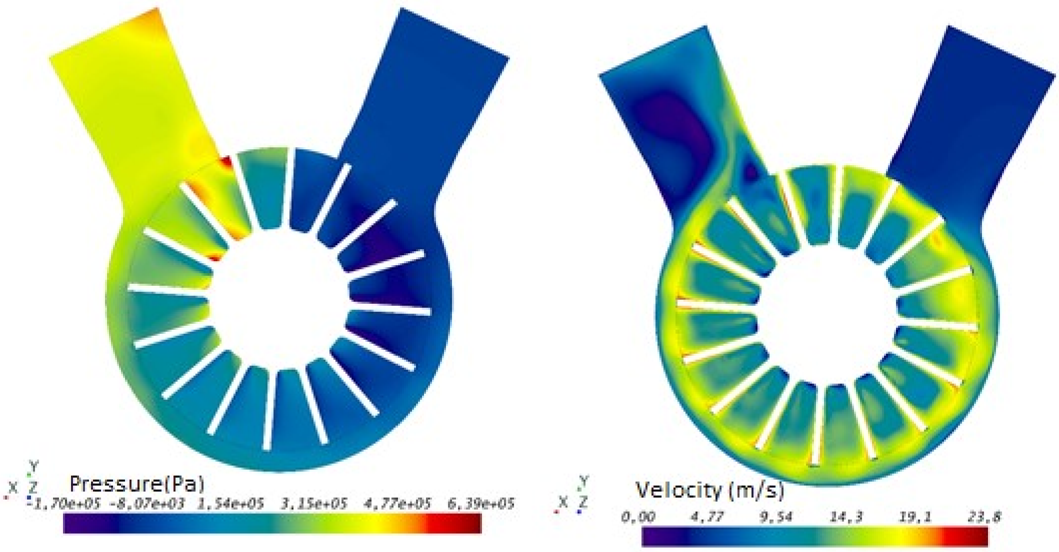

Figure 2. The fields of pressure and velocity distribution in the section, perpendicular to pump rotation axis.

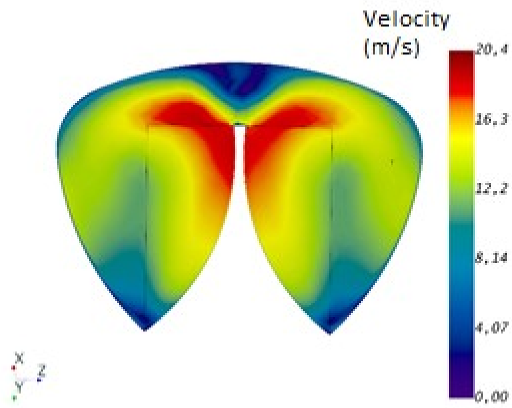

Figure 3. The field of velocity distribution in the between blades space.

After analyzing the fluid flow in the flow part of the regenerative stage, it was decided to consider the effect on the energy characteristics of the following parameters (figure 1, figure 4):
D – the outer diameter of the impeller;
b – vortex wheel width;
n – the number of vortex wheel blades (on the one side);
h – radial gap between the impeller and the casing;
Kf1 = *A*/*B* the coefficient of "roundness accuracy" of between blades space;
Kf2 = *L*2/*L*1 the symmetry coefficient of between blades space.

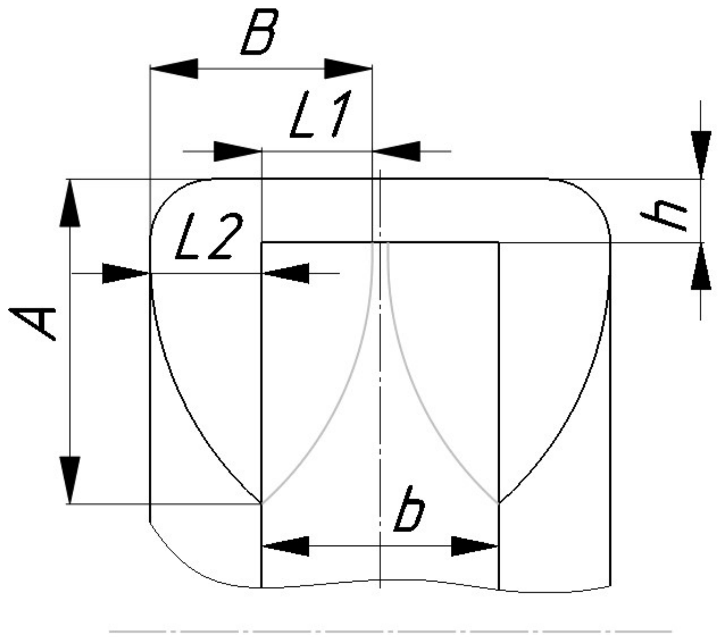

Figure 4. The parameters that have influences on the energetic characteristics of closed type regenerative pumps.

The parametrized model of the flow part of the closed type regenerative pump is shown at figure 5. It has opportunity to change the chosen parameters simultaneously or separately in a wide range of values', which makes it convenient for obtaining the optimal flow part by the LP-τ search method [15].

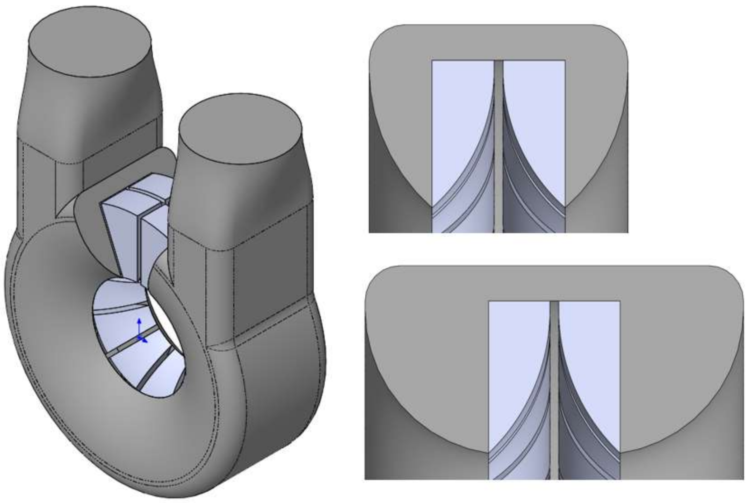

Figure 5. The parametrized model of the flow part of the closed type regenerative pump.

3. Results

In order to evaluate the effect of a single parameter on the energy characteristics of the regenerative pump with the other committed parameters, for each parameter a range of values was selected, which is divided into 15 equal intervals.

The range of parameter changing:
D = 110…130 mm;
b = 26…40 mm;
n = 13…24;
h = 5…15 mm;
Kf1 = 0.8…1.2;
Kf2 = 1.0…2.0.

The boundary conditions were the total value of the outlet pressure and the fluid velocity amplitude at the outlet of the flow part [16–18].

The number of nodes in each model varied from 380 thousand to 420 thousand nodes.

Figures 6–11 show the dependences of head and efficiency on variable parameters.

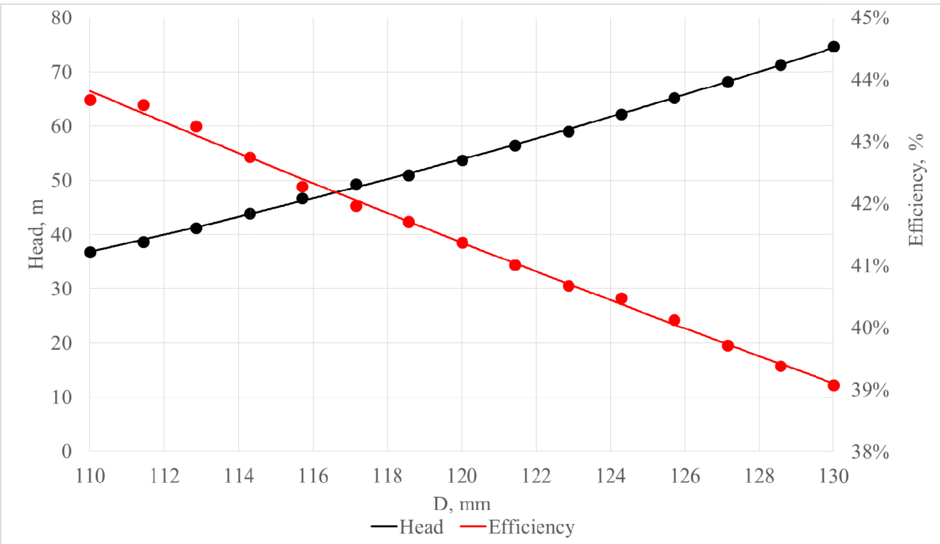

Figure 6. The dependencies of efficiency and pressure from D.

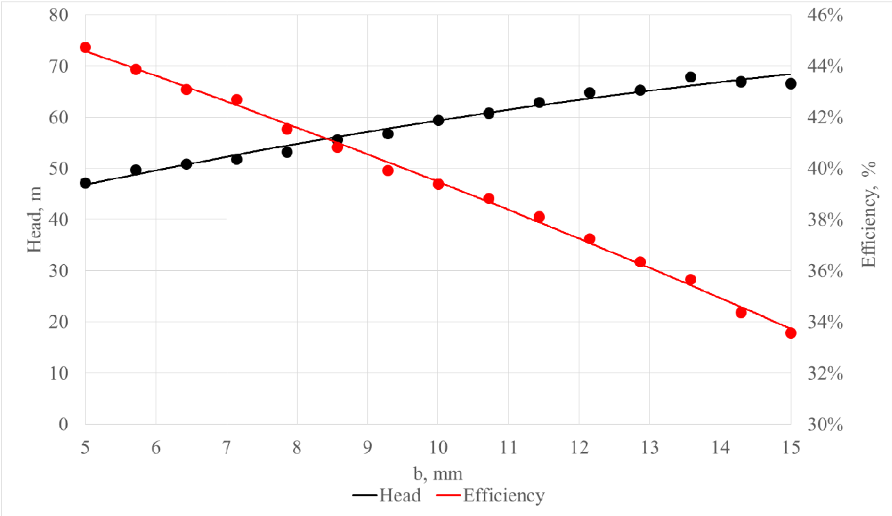

Figure 7. The dependencies of efficiency and pressure from *b*.

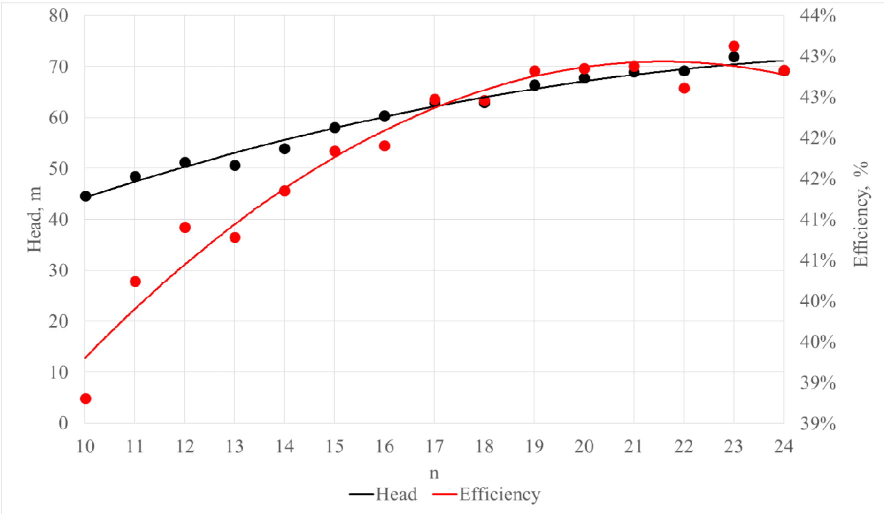

Figure 8. The dependencies of efficiency and pressure from *n*.

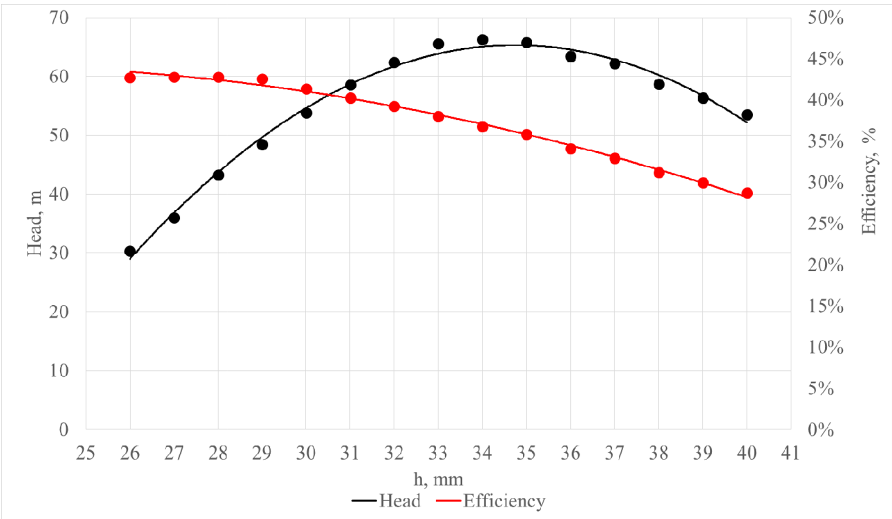

Figure 9. The dependencies of efficiency and pressure from *h*.

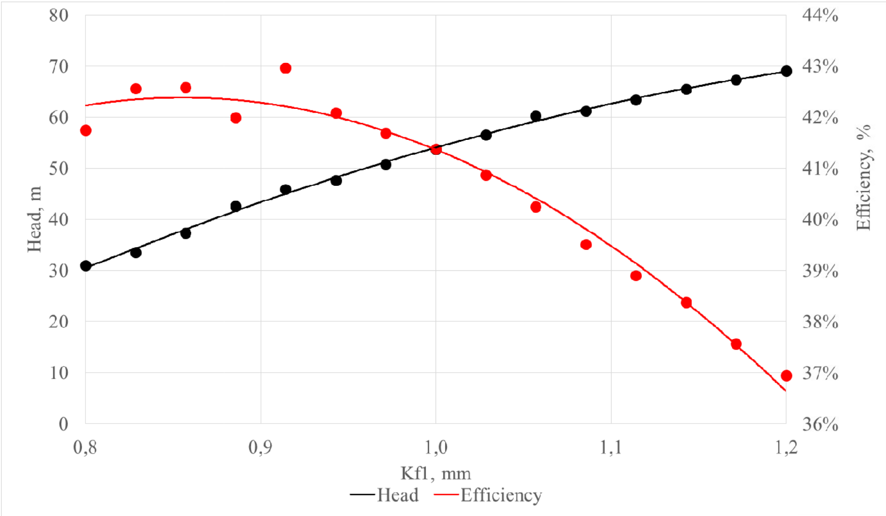

Figure 10. The dependencies of efficiency and pressure from Kf1.

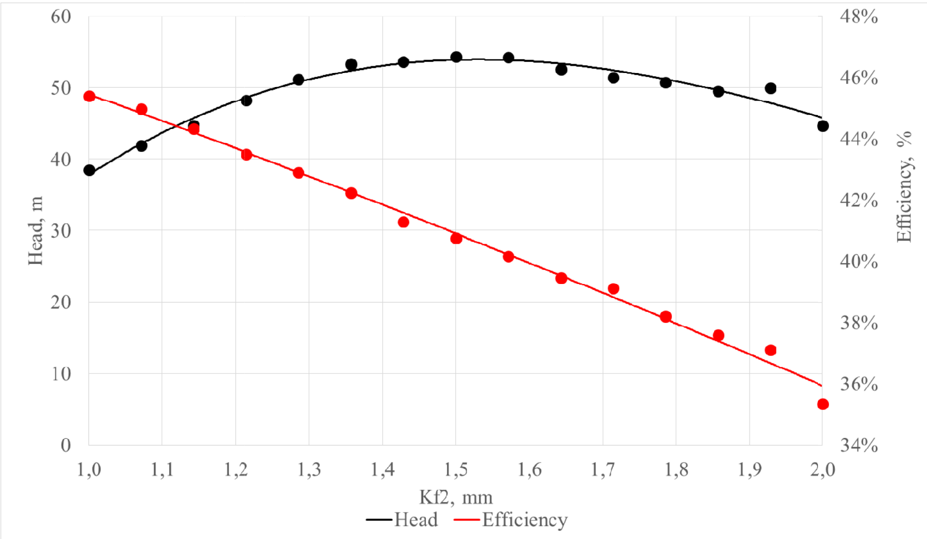

Figure 11. The dependencies of efficiency and pressure from Kf2.

On the basis of the obtained results, it was possible to narrow the range of parameters changes and carry out a comprehensive optimization of the flow section of a closed-type regenerative pump using computational fluid dynamics methods with the LP-τ sequence. The number of design models appeared to be 128 [19-20].

According to the optimization results, the best model has the following characteristics: head $H = 44$ m, hydraulic efficiency $\eta = 51\%$, which is 11.4% more than the original model has. The fields of pressure and velocity distribution in the flow part are presented in figures 12, 13.

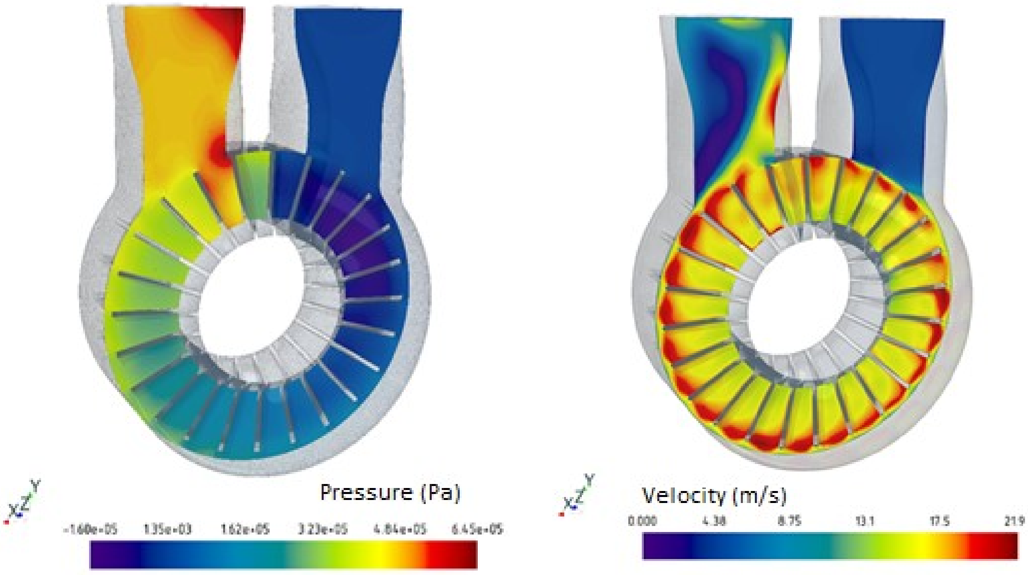

Figure 12. Fields of pressure and velocity distribution in the section perpendicular to the rotation axis of the pump.

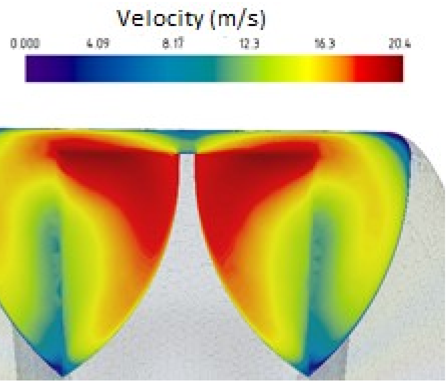

Figure 13. Velocity distribution field in between blades space.

4. Discussion

Based on the results of the work, the following conclusions may be drawn:

1. A parametrized model of the flow part of the closed type regenerative pump has been developed, which allows the design parameters changing in a wide range;

2. As it was shown by hydrodynamic modeling, the hydraulic efficiency decreases while parameters D, b, h, Kf1 are increasing. As the number of blades increases, hydraulic efficiency increases greatly. The highest value is obtained at n = 24. The coefficient of symmetry is advisable to choose from the range of Kf2 = 0.8 ... 1.

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