

Assessment of the level of influence of various geometrical parameters of the centrifugal pump guide vane on its energy characteristics

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Abstract: The article is devoted to the research of the influence of geometrical parameters of the inlet section of the multistage pump radial channel guide device on its hydraulic efficiency. The obtained values of hydraulic efficiency, considered depending on the geometrical parameters of the guide device, are given. The efficiency parameters were obtained via hydrodynamic modeling. Recommendations on the optimal geometric parameters are offered and compared with already known semi-empirical formulas.

Introduction

One of the most important steps in the design of a centrifugal pump is the design of a diversion device. Radial channel guide devices are mainly used in multistage pumps with a single stage specific speed of $n_s < 120$, as well as in some single-stage pumps with $n_s = 200 - 500$. The guide device geometry is one of the most important factors determining the operating parameters and efficiency of the pump. [1]

Thus, it is necessary to take into account the influence of the basic geometric parameters on the overall efficiency of the pump flow part.

The proposed research can be carried out as experimentally, obtaining regularities by empirical methods, as well as by methods of hydrodynamic modeling [2, 3]. In this paper an attempt is made to optimize the parameters of the guide device by using the methods of hydrodynamic modeling, and a comparison with the available empirical data is made [4]–[8].

The object under consideration is the impeller of the centrifugal multistage pump and radial channel guide device.

Figure 1 represents the 3D-model of the impeller with the guide device mounted on a shaft.

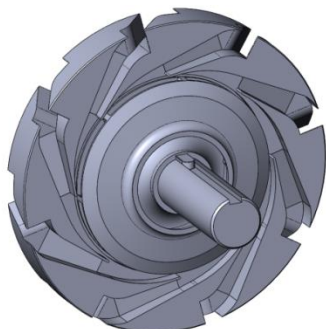


Figure 1. The guide device with the impeller mounted on a shaft

Methods

The guide device consists of 4 main sections: spiral section, diffuser, transition channel and return channel. In the calculation given in [1], the idea of spiral capacity was used to determine the parameters of the spiral section:

$$\int_{r_3}^r \frac{b}{r} dr = k_{sp} \cdot \frac{Q}{K_3} = A_s,$$

where b is the width of the spiral; k_{sp} is the coefficient that depends on the specific speed; r_3 is the radius of the initial section of the spiral; K_3 is the moment of flow velocity at the inlet of the spiral; A_s is the expected throughput capacity.

The objective of this study was to analyze the influence of the volute throughput capacity and tongue angle on the operating parameters and efficiency of the pump. Also of special interest is the comparison of the obtained optimal parameters with the parameters of the guide device, designed on the basis of known empirical dependencies.

The spiral section is based on the principle that the throughput capacity of the section in it increases linearly with the increase of the angle. As the cross section of the spiral channel in the guide device considered has a rectangular shape, it was mathematically obtained that the shape of the curve providing the required increase in throughput capacity is a logarithmic spiral.

$$b = \text{const};$$

$$A_s = b \cdot \int_{r_3}^r \frac{1}{r} dr = b \cdot \ln(r) + c$$

$$r = a \cdot e^{\frac{A_s}{b}}$$

This spiral should provide a steady increase in throughput capacity, but its shape does not take into account the influence of blade thickness in the target cross-section of the spiral. In order to provide the calculated throughput capacity in the target cross-section, the upper edge of the spiral moves upwards from the initial target cross-section to which the spiral is built.

The parameterized model of the initial section of the guide device was designed for the study.

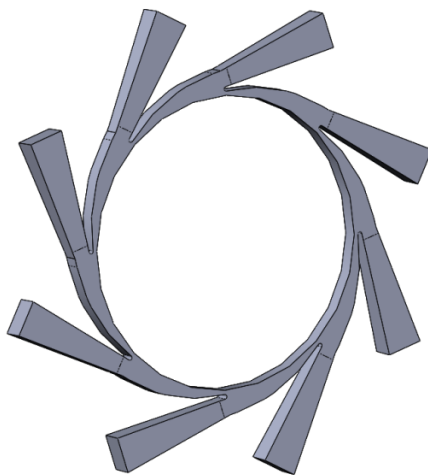


Figure 2. The parameterized model of the initial section of the guide device

The parameterized model is designed using the following method: mathematically precise logarithmic spiral is drawn on the basis of the defined radius r_3 , the angle step $t = \frac{2\pi}{z}$, where the

z — number of channels of the guide device and the spiral endpoint that can be varied in height during the study.

The model provides the ability to vary the height of h_3 from h_{3init} to h_{3calc}' , where:

h_{3init} — initial calculated height of the channel cross-section without taking into account the existence of blade;

h_{3calc}' — calculated height of the channel cross-section taking into account the existence of blade ($d_b = 3 \text{ mm}$)

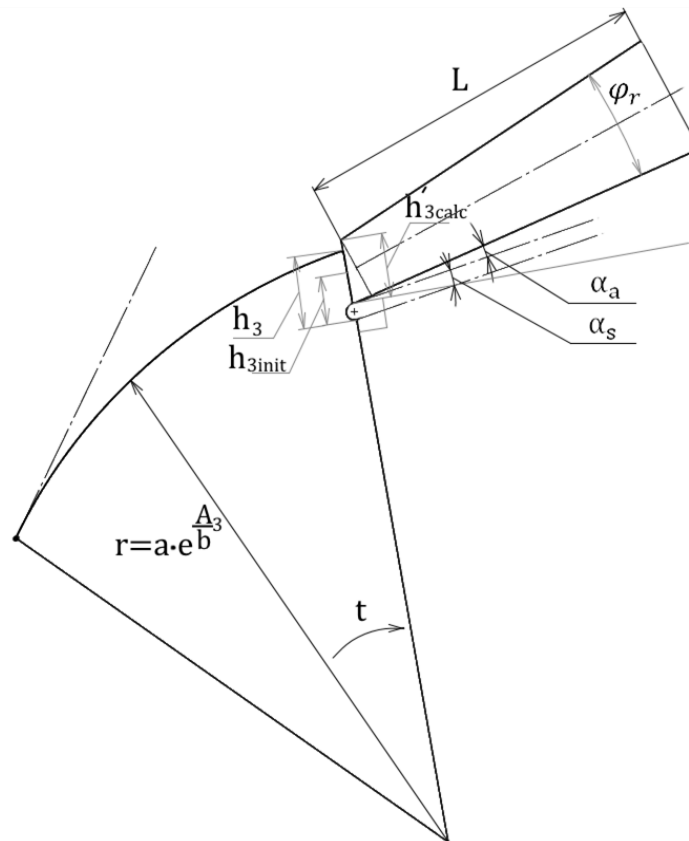


Figure 3. The method of designing the parameterized model

This is done by introducing the k_{sp} coefficient, which allows to vary the throughput capacity of the spiral section.

The diffuser is based on the strictly calculated input height and has remained constant in this study. Diffuser parameters were chosen to be optimal based on empirically developed relations, the angles of diffuser in radial and axial direction: $\varphi_r = 9^\circ$, $\varphi_b = 4^\circ$, diffuser coefficient 3,5.

Due to the difference in the helix final height and the initial diffuser height, the parameterized model automatically performs splicing of the helix based on the incircle tangent to the splicing lines.

It is possible to define the spiral throughput coefficient in the model, which allows to vary the difference between the initial and calculated cross-sections and the actual spiral endpoint.

$$A_{s1} = k_{sp} \cdot A_s,$$

where k is a variable coefficient; A_{s1} — is the resulting helix throughput capacity.

Another investigated parameter of the model is the initial angle of the blade, the minimum value of which is determined by the angle of tangent to the helix at its starting point.

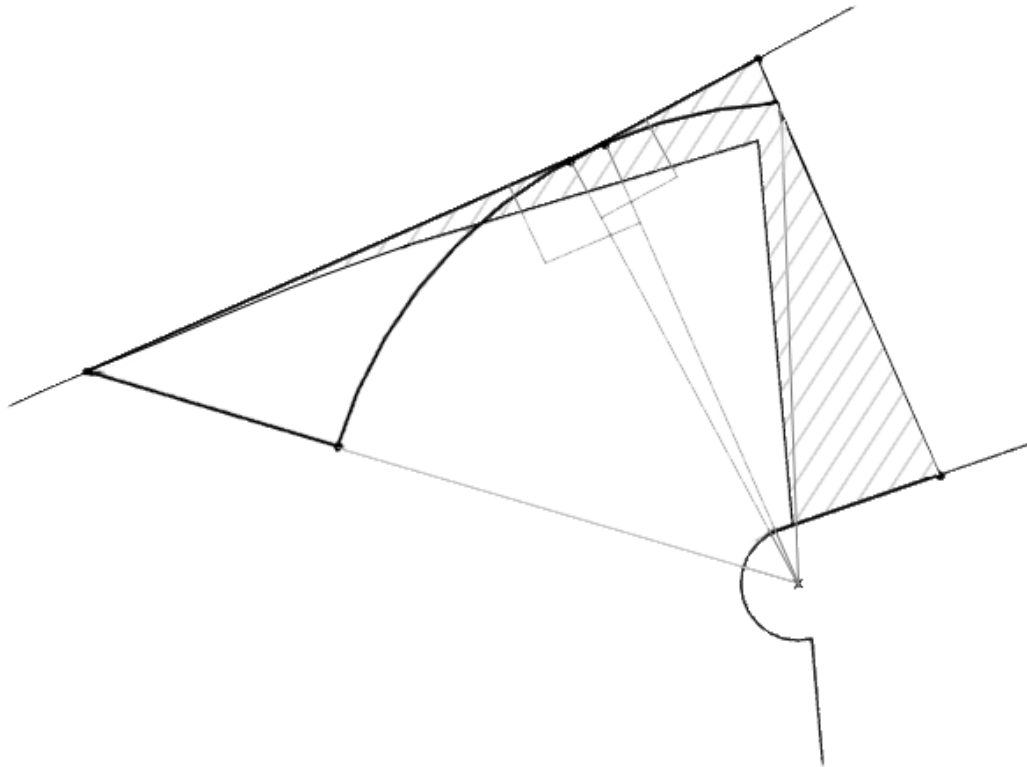


Figure 4. The method of splicing of the helix

Thus, this parameterized model was used to study the influence of the described parameters of the geometry of the input part of the guide device on its energy characteristics, especially the hydraulic efficiency. In order to capture the most comprehensive combination of possible combinations of parameters, 81 models were built, where the installation angle varied from 7 degrees to 25,6 degrees, and the capacity factor varied within (0.83..1.17).

Also the multistage pump impeller model was used for the study, the parameters of which have not been changed.

$$D_2 = 186 \text{ mm}, b_2 = 9 \text{ mm}, \beta_{2b} = 24^\circ$$

Figure 5. The multistage pump impeller



The opportunities of modern hydrodynamic modeling software were used to measure hydraulic efficiency of the different guide devices. The numerical modeling method is based on solving discrete analogues of basic equations of hydrodynamics [9,10]. For incompressible fluid ($\rho = \text{const}$) these are as follows:

Mass conservation equation (continuity equation)

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

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

The equation of conservation of momentum (Reynolds averaging):

$$\rho \left[\frac{\partial U_i}{\partial t} + U_i \frac{\partial U_i}{\partial x_j} \right] = - \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left[\overline{T_{ij}^{(v)}} - \rho u_i u_j \right], \text{ where}$$

U, P — averaged velocity and pressure;

$\overline{T_{ij}^{(v)}} = 2\mu \overline{S_{ij}}$ — viscous stress tensor for incompressible fluid;

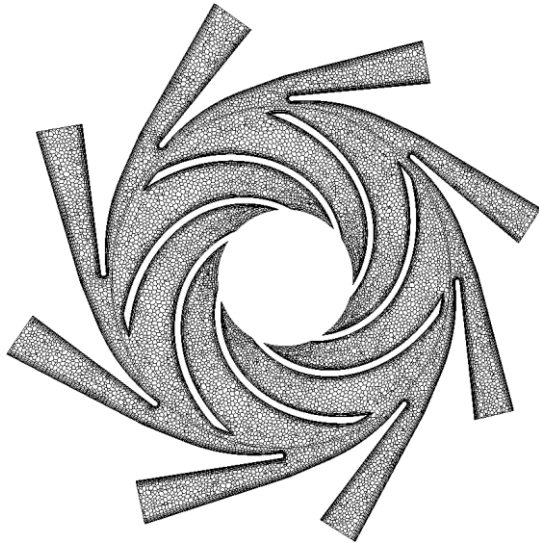
$\overline{S_{ij}} = \frac{1}{2} \left[\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right]$ — instantaneous strain rate tensor;

$\rho u_i u_j$ — Reynolds stresses.

The introduction of the Reynolds averaged Navier — Stokes equation leaves the system of equations unclosed, since additional unknown Reynolds stresses appear. To solve this system, a semi-empirical $k-\omega$ SST turbulence model was used. It introduces the necessary additional equations: the equations for the transfer of the turbulence kinetic energy and the relative rate of dissipation of this energy:

$$\begin{aligned} \frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} &= P_k - \beta k \omega + \frac{\partial}{\partial x_j} \left[(v + \sigma_k v_T) \cdot \frac{\partial k}{\partial x_j} \right] \\ \frac{\partial \omega}{\partial t} + U_j \frac{\partial \omega}{\partial x_j} &= \alpha \cdot S^2 - \beta \cdot \omega^2 + \frac{\partial}{\partial x_j} \left[(v + \sigma_\omega v_T) \cdot \frac{\partial \omega}{\partial x_j} \right] + 2 \cdot (1 - F_1) \cdot \sigma_{\omega 2} \cdot \frac{1}{\omega} \cdot \frac{\partial k}{\partial x_j} \cdot \frac{\partial \omega}{\partial x_j} \end{aligned}$$

Figure 6. Mesh



Each design flow part was simulated on a mesh of approximately 380,000 cells. The cells were built multi-faceted in the flow core and prismatic near the solid walls. The design mesh for one of the designs is represented in Figure 6.

Results

The field of distribution of pressures and velocities in the flow part for the best in efficiency model and for the model designed on the basis of known empirical dependencies is shown in Figures 7–10. The resulting hydraulic efficiency values depending on the investigated parameters of the guide device geometry are shown in Table 1.

Table 1. Hydraulic efficiency values depending on variable parameters

Capacity factor K_{sp}	Spiral angle $\alpha_s, ^\circ$	Additional angle $\alpha_a, ^\circ$								
		0.1	2	4	6	8	10	12	14	16
0.83	6.9	83.92	84.45	84.82	84.59	84.74	84.68	84.98	84.70	84.47
0.8725	7.2	84.36	84.58	84.84	84.76	84.72	84.81	84.90	84.71	84.46
0.915	7.5	84.52	84.66	84.73	84.71	84.80	84.96	84.71	84.58	84.51
0.9575	7.9	84.67	84.53	84.89	84.89	84.99	84.69	84.75	84.60	84.51
1	8.2	84.65	84.86	84.72	84.96	85.09	85.04	85.16	84.57	84.14
1.0425	8.6	84.58	84.80	84.86	84.97	84.84	85.15	84.70	84.63	84.32
1.085	8.9	84.61	84.80	84.66	84.75	84.65	84.98	84.37	84.57	84.17
1.1275	9.3	84.42	84.66	84.42	84.50	84.32	84.53	84.26	83.74	83.61
1.17	9.6	84.33	84.44	84.37	84.38	84.31	84.19	84.06	83.60	83.44

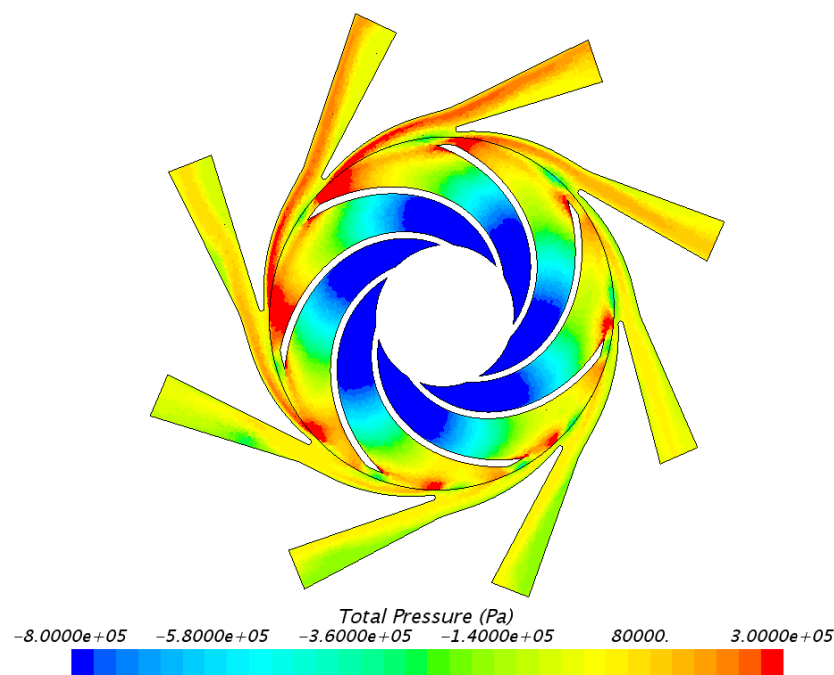


Figure 7. The field of distribution of pressure in the flow part for the best in efficiency model

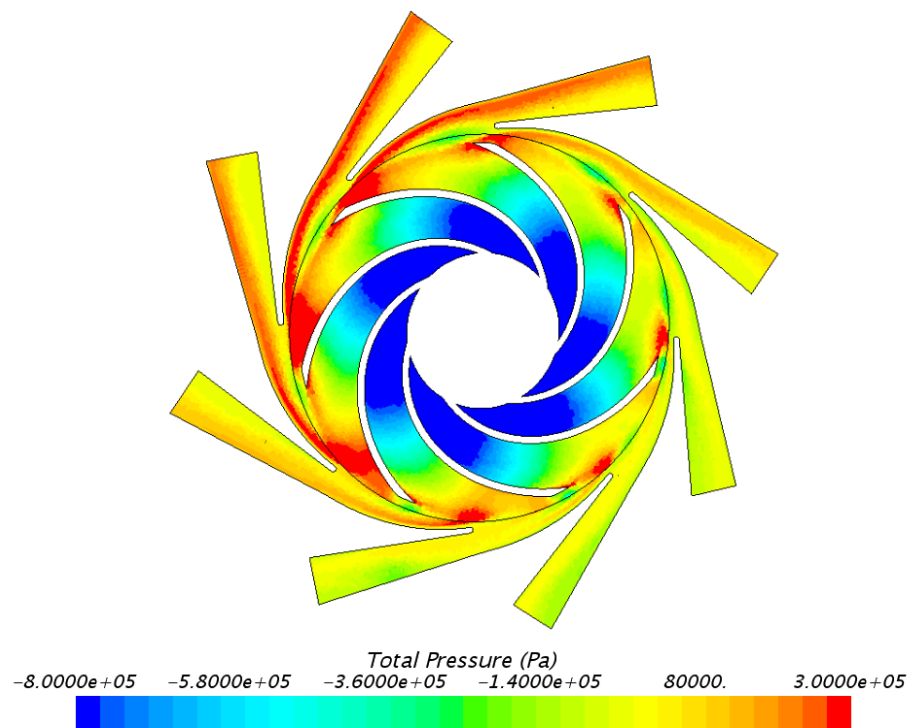


Figure 8. The field of distribution of pressure in the flow part for the model designed on the basis of known empirical dependencies.

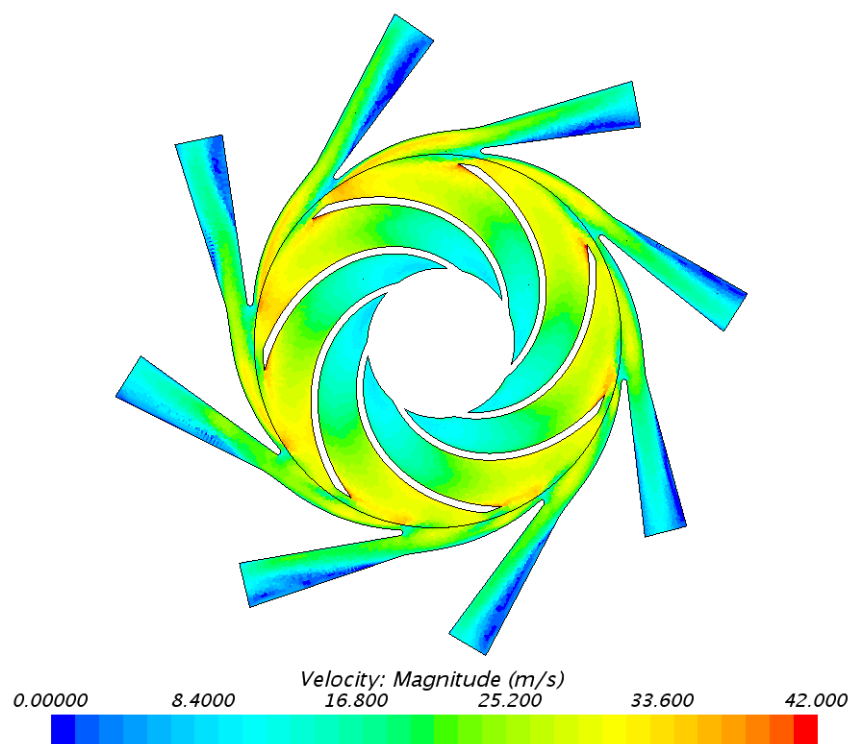


Figure 9. The field of distribution of velocity in the flow part for the best in efficiency model

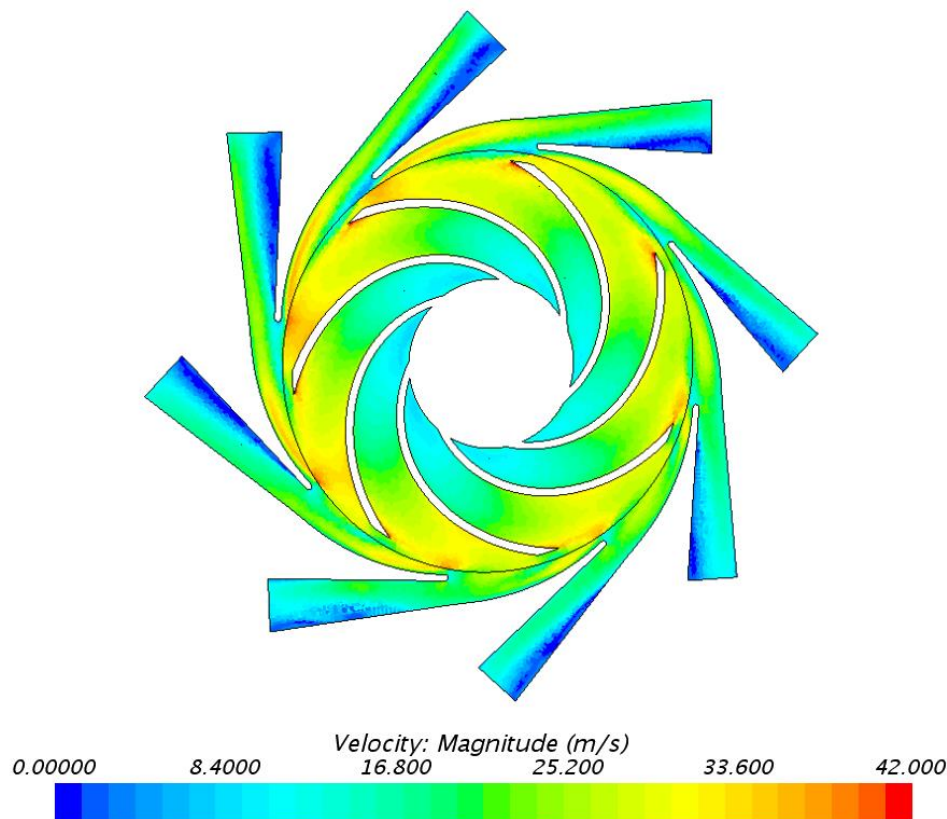


Figure 10. The field of distribution of pressure in the flow part for the model designed on the basis of known empirical dependencies.

Discussion

The analysis of the results of the hydrodynamic modeling has shown that in the case of the study of the influence of the initial angle of the blade, the model of the guide device with the angle deviating from the angle of shockless inflow $\alpha_{a-sh} = 10,2^\circ$ has the best efficiency (calculated angle of shockless inflow $\alpha_{sh} = 10^\circ$; the angle deviating from the angle of shockless inflow α_{a-sh} ; the absolute angle is $\alpha = \alpha_a + \alpha_s = \alpha_{sh} + \alpha_{a-sh}$). This result deviates from the recommended by the empirical method, where $\alpha_{a-sh} = 2..3^\circ$. [11]

In the case of the study of helix capacity influence, it was proved that the exactly calculated throughput capacity is the most optimal option for the design of guiding devices.

Conclusion

1. The determining factor in terms of losses in the guide apparatus is the flow in the diffuser, which is not necessarily associated with the shockless inflow condition.
2. The optimization of the guide apparatus must be carried out in a comprehensive way, taking into account both the parameters of the spiral part and the diffuser.
3. Due to the optimization it has been achieved to increase the hydraulic efficiency of the guide device from 84.33% to 85.16%. It is expected that a more complete optimization of the guide device will result in an even better result.

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