Study of multi-stage centrifugal pump guide vanes in a package of hydrodynamic simulating STAR CCM+

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Abstract
For multistage centrifugal pumps, one of the main components is the guide vane. A rather important task for the guide vane is compliance with the requirements of the technical specification. It is equally important technologically correct to produce a guide vane. The manufacture of such parts is a time consuming process. To avoid a number of difficulties, this article explores the two most frequently used guide vanes of hydrodynamic simulating tools.

Introduction
Multistage pumps are one of the most popular among those used in industry. The main advantage of this type of pump is the ability to change the head. This energy characteristic is directly related to the number of stages in the pump. The pump head is equal to the sum of the head generated by each stage. The pump flow rate remains constant at a given rotor speed.

In multistage pumps, the flow of the pumped liquid moves sequentially between several impellers mounted on the same shaft. Fluid is delivered from one stage to another through a guide vane. One of the main tasks of the guide vane is to supply the liquid to the next stage with the least hydraulic losses. Therefore, to perform this task requires high manufacturability of the design of the guide vane. [1, 2]

There are many kinds of guide vanes. In this article, the following types will be considered: a radial channel translational guide vane and a guide vane with a non-blade translation channel.

In channel-type guide vane of a multistage pump, conditionally there are three sections: spiral, transfer blade and diffuser (in which the main conversion of kinetic energy takes place). The transfer channel and the blade inlet area provide a uniform liquid supply with a given circulation into the suction area of the next-stage impeller.

A guide vane with a no-blade transfer channel is much easier to manufacture than the channel type. This guide vane consists of a diffuser blade channel, a transfer blade channel and a blade return (supply) channel to the next stage of the centrifugal pump. The circular no-blade channel begins and ends at the same diameter, while the flow rate in the circular space does not change.

This article provides further research and comparison of the above described guide vanes.

The initial data for the calculation of the guide vane are:
Q – flow rate through the vane without counting leakage in the seals of the impeller;
H – pump head;
n – rotation speed.
The calculation of the guide vanes is made for a multistage centrifugal pump, which is shown in Figure 1 with the following initial parameters:

\[ Q = 150 \text{ m}^3/\text{h}; \]
\[ H_p = 1000 \text{ m}; \]
\[ n = 2900 \text{ r/min}. \]

By design, it was determined that the number of stages is equal to 9. The head per each stage of the pump is 111.1 m.

Outlet diameter of impeller \( D_2 = 300 \text{ mm}. \)

The study of guide vanes is carried out in the STAR CCM + hydrodynamic simulation package. To do this, it is necessary to create a solid-state three-dimensional model of each guide vane. [3, 4]

3D-models of the channel-type guide vane and the guide vane with no-blade transfer channel are shown in Figures 2 and 3, respectively.

The criteria for comparing the work of guide vanes are the energy characteristics such as the head and hydraulic efficiency.

The study is conducted under the same conditions for both guide vanes.
The mathematical model of the processes, on the basis of which the hydrodynamic modeling is carried out, is presented below.

**Mathematical model**

Pump stage head:

\[ H = \frac{p_2 - p_1}{\rho g} + \frac{V_2^2 - V_1^2}{2g}, \]

where
- \( p_2 \) is the pressure at the outlet of the pump stage;
- \( p_1 \) is the pressure at the inlet of the pump stage;
- \( V_2 \) is the average speed at the outlet from the pump stage;
- \( V_1 \) is the average speed at the inlet from the pump stage.

Rotor speed and pump flow rate are parameters which are set in the numerical model, except for some special cases.

In the overwhelming majority of cases, the flow of fluid in the flow path of the pump is turbulent. In a general form, the equations of dynamics of an incompressible viscous medium for a turbulent flow mode are as follows:

- **Fluid continuity equation:**
  \[ \frac{\partial \bar{u}_x}{\partial x} + \frac{\partial \bar{u}_y}{\partial y} + \frac{\partial \bar{u}_z}{\partial z} = 0, \]

where \( \bar{u}_i \) is the time-averaged projections of fluid velocities on the corresponding axes.

- **Mass conservation equation:**
  \[ \frac{\partial \bar{u}_x}{\partial t} + \frac{\partial \bar{u}_y}{\partial t} + \frac{\partial \bar{u}_z}{\partial t} = 0, \]

where \( \bar{u}_i \) is the time-averaged projections of fluid velocities on the corresponding axes.

- **Equation of the change in the amount of motion averaged over time:**
  \[
  \rho \left[ \frac{\partial \bar{u}_x}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} \right] = -\frac{\partial \rho \bar{u}_i}{\partial x_j} + \frac{\partial}{\partial x_i} \left[ \bar{f}_j^{(v)} - \rho \langle u_i u_j \rangle \right],
  \]

where \( \bar{u}_i \bar{p} \) is the averaged velocity and pressure;
- \( \bar{f}_j^{(v)} = 2\mu \bar{\dot{s}}_j \) is the viscous stress tensor for an incompressible fluid;
- \( \bar{\dot{s}}_j = \frac{1}{2} \left[ \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right] \) is the strain rate tensor;
- \( \rho \langle u_i u_j \rangle \) are Reynolds stresses.

The Reynolds system of equations is open due to the presence of unknown Reynolds stresses. The system is closed by using the k-ω SST turbulence model. This model combines the advantages of both the k-ω and k-ε models: in the near-wall region the k-ω model is used, and in the core of the flow is the k-ε model. [5 – 9]

This model includes two equations of turbulence parameters transfer:

- **Turbulence kinetic energy transfer equation:**
Reynolds stresses in the equations of dynamics are based on the Boussinesq hypothesis:

\[
\frac{\partial k}{\partial t} + u_j \frac{\partial k}{\partial x_j} = P_k - \beta \cdot k \omega + \left( \nu + \sigma \nu_T \right) \frac{\partial k}{\partial x_j},
\]

where \( k = \frac{u''_x + u''_y + u''_z}{2} \) — kinetic energy of turbulence;

\( u'_i \) — velocity pulsations;

\( P_k \) — turbulence power generation member;

\( \omega \) — relative rate of turbulence dissipation;

\( \nu_T \) — turbulent viscosity.

- transfer equation for the relative rate of energy dissipation of turbulence:

\[
\frac{\partial \omega}{\partial t} + u_j \frac{\partial \omega}{\partial x_j} = \alpha S^2 - \beta \omega^2 + \left( \nu + \sigma \nu_T \right) \frac{\partial \omega}{\partial x_j} + 2 \left(1 - F_j \right) \sigma \omega \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}.
\]

Reynolds stresses in the equations of dynamics are based on the Boussinesq hypothesis:

\[
\rho \left( u_i u_j \right) = 2 \mu_T \left[ \frac{1}{2} \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{1}{3} \frac{\partial u_k}{\partial x_k} \delta_{ij} \right] - \frac{2}{3} \rho k \delta_{ij},
\]

where \( \delta_{ij} \) — Kronecker symbol.

The calculation can be carried out both in steady mode and in non-steady.[10 – 13]

When conducting a stationary calculation in the hydrodynamic equations, there are no terms with time differentiation. This calculation allows you to quite accurately calculate the parameters of the pump at the optimum operating point and near it, and also allows you to significantly reduce the calculation time. However, far from the nominal point, the error of the stationary calculation significantly affects the calculation due to the intensive vortex formation, since the vortex formation processes are non-stationary processes in their essence.[14 – 21]

Also, with a stationary calculation, it is impossible to obtain by calculation the oscillations of hydrodynamic parameters caused, for example, by rotation of the impeller, therefore, if necessary, take into account all the abovementioned factors, as well as evaluate the pump behavior not only at the optimum point, but throughout the entire flow rate range. The calculation is carried out in non-stationary mode.

To carry out the calculation in STAR CCM+, it is necessary to import into the hydrodynamic package a solid-state three-dimensional model of the flow part of the pump stage.
**Figure 4.** 3D model of the flow part of the pump stage with a channel-type guide vane

**Figure 5.** 3D model of the flow part of the pump stage with a guide vane with a no-blade transfer channel

**Calculated mesh**

In the study, in order to set the parameters of the computational grid, the surface partitioning and the generator of polyhedral cells are used together with the generator of the wall layers. For clarity, the calculated grids in sections perpendicular to the impeller rotation are presented below. The calculated grid for a model with a channel-type guide vane is shown in Figure 6, for a model with a guide vane with a no-blade transfer channel in Figure 7.

![Figure 6](image1.png)

![Figure 7](image2.png)

After generating the volume mesh and further the computational mesh, it is necessary to set a physical model. The following models are selected:
- unsteady task;
- working medium - liquid;
- equation of state - constant density;
- solver - divided;
- flow mode - turbulent;
- k-omega SST model of turbulence.

The results of hydrodynamic calculations are scalar scenes of the distribution of velocity and pressure in the flow part of the pump stage. The energy characteristics of the pump stage were calculated as criteria for comparison: pressure and hydraulic efficiency. In the hydrodynamic calculation were not incorporated volumetric and mechanical losses.

Scalar scenes of speed and pressure distribution for a stage with a channel-type guide vane are shown in Figures 8 and 9.

![Figure 8. Scalar scenes of speed distribution](image1)

![Figure 9. Scalar scenes of pressure distribution](image2)

For a stage with a guide vane with a no-blade transfer channel, scenes of speed and pressure distribution are also similarly represented in Figures 10 and 11.
The calculation of stages for both types of guide vanes was carried out under the same conditions, physical models, and reference values of the grid builder. The boundary conditions remained unchanged.

The calculation was carried out at 2100 iterations. Such a number of iterations is sufficient for the full rotation of the impeller.

Figure 10. Scalar scenes of speed distribution

Figure 11. Scalar scenes of pressure distribution

Figure 12. Head graph
Energy characteristics are obtained in the form of iterative graphs. To calculate the final value of head and efficiency, it is enough to consider the last 25% of iterations. The pressure and hydraulic efficiency graphs for a stage with a channel-type guide vane are shown in Figures 12 and 13, respectively.

The average value of the hydraulic efficiency is 93%. The average head is 120 meters. The obtained energy characteristics of the stage correspond to the required values of the hydraulic efficiency - 88% and a head -111.1 m., obtained by mathematical calculations.

![Hydraulic efficiency graph](image1)

**Figure 13.** Hydraulic efficiency graph

![Head graph](image2)

**Figure 14.** Head graph
Similarly, the graphs of head (Figure 14) and hydraulic efficiency (Figure 15) were calculated for a stage with a guide vane with a no-blade transfer channel.

The average value of hydraulic efficiency - 91%, head - 112 meters. The characteristics obtained in this case also satisfy the required. However, stabilization of processes in the second case took longer.

**Conclusion**

For a visual demonstration of the processes occurring in the stages of a multistage centrifugal pump, scalar scenes of the distribution of speed and pressure were presented.

The calculation was carried out in a non-steady mode, despite the fact that the hydrodynamic modeling was carried out near the optimal point.

Thus, although the research took place under equal conditions and the results met the requirements, they differ. The model of a stage with a channel-type guide vane proved to be better from an energetic point of view. In spite of the fact that the guide vane with the no-blade transfer channel is easier to manufacture, its test model turned out to be less technologically than the model of the channel-type guide vane, what could have caused a number of hydraulic losses. In the future, it is necessary to consider in more detail the processes in the transfer channel, since the rotation of the fluid from the forward channel to the transfer channel and from transfer to the return channel (leading to the next stage) turned out to be quite steep, which could have been the cause of unnecessary losses.

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