Optimal design of inlet device of multistage centrifugal pump

A Petrov\textsuperscript{1,2}, A Lysenko\textsuperscript{1}, T Valiev\textsuperscript{1} and N Isaev\textsuperscript{1}

Bauman Moscow State Technical University, 5 Second Baumanskaya Street, Moscow, 105005, Russian Federation

\textsuperscript{2}E-mail: alex_i_petrov@mail.ru

Abstract: In the article the authors present a new type of inlet device for multistage centrifugal pumps. An overview of the existing types of inlet devices is given, their advantages and disadvantages are observed. The form of the proposed type of inlet device is described, which makes it possible to combine good cavitation properties of the pump with low losses in the inlet device. The results of hydrodynamic modeling of the proposed type of inlet device and its comparison with the existing ones are given.

1. Introduction

An important factor within the process of creation of modern multistage pumps is the designing of inlet chambers. The geometry and nature of fluid flow in these chambers directly affect pump characteristics such as NPSHR \cite{1-3}, the efficiency rate, the steepness of the pressure characteristics in the area of low flow rates \cite{4-9}. Moreover, steeply decreasing characteristic is especially important for the pumps used in thermal power engineering and nuclear power engineering \cite{10}.

Now the following types of inlet devices are used for these pumps:
- annular chambers;
- annular chambers with a separating edge;
- semi-spiral chambers;
- channel inlets (deflectors) with annular chamber.

All these devices have their advantages and disadvantages. Thus, the pumps with an annular chamber have a higher efficiency rate due to its relatively small losses in the inlet devices. But when operating in the low feed rate area, the pump head decreases due to the reverse currents in the inlet devices, which leads to the characteristic's falling-off.

The integration of a separating edge of sufficient size to the inlet device allows to minimize this phenomenon, but also reduces the efficiency rate of the pump, as well as its cavitation properties \cite{11–13}.

The appliance of semi-spiral inlet devices, in turn, allows to improve the cavitation property of the pump, which is especially important for the multistage pumps designed for large feeds, but at the same time reduces the pump head (due to appearance of the moment of speed at the inlet to the first stage) and its efficiency rate. Channel inlet devices are used, on the contrary, for the pumps with low feed rates, but the losses in them are large compared to the other types of inlet devices. Figure 1 shows the lateral semi-spiral and annular inlet devices.
The purpose of this study was to create such a geometry of the inlet device of a multistage pump in order to ensure the smallest losses at the inlet to the first pump stage.

2. Methods

To determine the losses in the flow part of the inlet device, the 3D models of the original intake system and the optimized one were built. For these models, hydrodynamic modeling of fluid flow was carried out [14]. As the initial model an annular inlet of centrifugal pump type CNS 300-120 was taken.

Considered inlet systems are shown in figure 2. The distinguishing feature of the suggested inlet device lies in the fact that in addition to the reduced volume, it does not have thin-walled edges, and a massive streamlined separator with a small curvature is used instead. It is assumed that this solution will reduce the volume of stagnant zones and improve the shaft streamlining as a whole.

The method of numerical simulation is based on solving discrete analogs of the basic hydrodynamic equations [15-17]. In case of an incompressible fluid model ($\rho = \text{const}$) this is:

Mass conservation equation (continuity equation)

$$\frac{\partial u_j}{\partial x_j} = 0,$$
where
\( \bar{u}_j \) – the average value of the fluid velocity in the projection on the \( j \)-th axis \((j = 1, 2, 3)\).

The equation of conservation of momentum (Reynolds averaging):
\[
\rho \left[ \frac{\partial U_i}{\partial t} + U_j \frac{\partial U_i}{\partial x_j} \right] = - \frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ \bar{T}_{ij}^{(v)} - \rho \langle u_i u_j \rangle \right]
\]

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

The introduction of the Navier—Stokes equation, averaged according to Reynolds, makes the system of equations not closed, since the additional unknown Reynolds stresses appear. To solve this system, in this problem a semiempirical model of \( k-\omega \) SST turbulence was used, which introduced the additional equations needed: the equations of transfer of the kinetic energy of turbulence and the relative dissipation rate of this energy:
\[
\frac{\partial k}{\partial t} + U_j \frac{\partial k}{\partial x_j} = \nu_k \left[ \frac{\partial}{\partial x_j} \left( \nu + \sigma_x u_T \right) \right] \frac{\partial k}{\partial x_j} - \beta k \omega - \frac{\partial}{\partial x_j} \left[ \left( 1 - F_f \right) \cdot \sigma_{o2} \cdot \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \right] + \frac{\nu}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}.
\]

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

As the boundary conditions, the velocity values at the inlet device and the pressure at the outlet from it were taken. The speed was set in such a way as to ensure the flow through the intake system equal to 300 m³/h.

The computational grid for each model consisted of about 200 thousand cells. Figure 3 shows the computational grids for the original model and the optimized one.

Figure 3. Computational grids for the original model and the optimized one.
3. Results

Based on the above modeling methods, a comparative numerical study of the flow in the original and proposed inlet devices was performed. The comparison criteria were the magnitude of the losses at the pump inlet, the even distribution of the absolute velocity at the inlet to the pump impeller.

Figure 4 shows the vector fields of the absolute velocity distribution in the meridional section of the original inlet device and the optimized one.

![Figure 4. Vector fields of the absolute velocity distribution in the meridional section of the compared devices.](image)

Then, after processing the results of the numerical experiment, the hydraulic losses of both devices were determined. With fluid flowing through the original intake system, the losses, expressed in terms of the differential pressure, were about 5000 Pa. The optimized model showed a result of about 3000 Pa.

The coefficient of evenness of the velocity distribution (Coriolis coefficient) allows to quantify the real distribution of velocities over the flow cross section:

$$\alpha = \frac{\int V^2 \, dS}{V_{av}^2 \, S},$$

where

- $V_{av} = \frac{Q}{S}$ – average flow velocity;
- $S$ – sectional calculated area;
- $V$ – real velocity within limits of $dS$.

For the annular inlet device $\alpha = 1.2814$, for the optimized one $\alpha = 1.1137$.

Figure 5 shows the fields of the velocities at the cross-sections, where the Coriolis coefficient is determined.

From the data analysis it can be seen that the proposed configuration of the inlet system allows a 30-40% reduction in losses at the pump inlet compared to an annular chamber with a separating edge. And also it allows to reduce the Coriolis coefficient at the inlet to the first stage of sectional pump with $\alpha = 1.2814 \Leftrightarrow \alpha = 1.1137$, which improves the conditions of flow impingement on the impeller blades and improves the cavitation property of the pump.
As the numerical study has shown, the proposed design of the inlet system can significantly improve the flow structure at the inlet to the first stage of a multistage pump and increase its efficiency rate;

2. Currently, work is underway to manufacture an experimental pump with the geometry of the inlet device given, for carrying out its comprehensive tests and determining the prospects for the proposed design use.

Published under licence in Materials Science and Engineering by IOP Publishing Ltd. Content from this work may be used under the terms of the Creative Commons Attribution 3.0 licence. Any further distribution of this work must maintain attribution to the author(s) and the title of the work, journal citation and DOI.

References
[1] Lomakin V O, Chaburko P S and Kuleshova M S 2017 Multi-criteria Optimization of the Flow of a Centrifugal Pump on Energy and Vibroacoustic Characteristics Procedia Engineering176 pp 476-482
[2] Lomakin V O, Kuleshova M S and Bozh’eva S M 2016 Numerical Modeling of Liquid Flow in a Pump Station Power Technology and Engineering5 pp 324-327
[3] Lomakin V O, Kuleshova M S and Kraeva E A 2015 Fluid Flow in the Throttle Channel in the Presence of Cavitation Procedia Engineering106 pp 27-35
[4] Kraposhin M V, Banholzer M, Pfitzner M and Marchevskiy I K 2018 A hybrid pressure-based solver for non-ideal single-phase fluid flows at all speeds International Journal for Numerical Methods in Fluids88 (2) pp 79-99
[5] Min’kov L L, Krokhina A V, Duecke J 2011 Hydrodynamic mechanisms of the influence on the classification characteristics of a hydrocycloneJournal of Engineering Physics and Thermophysics84 (4) pp 807-819
[6] Ionaitis R R and Chain P L 2009 Principles for engineering active-passive hydrodynamic control-and-safety systems for nuclear reactors Atomic Energy106 (3) pp 175-184
[7] Krapivtsev V G 2017 Model Studies of Coolant Flow Hydrodynamics in VVER-1000 In-Reactor Pressure Channel Atomic Energy122 (5) pp 304-310
[8] Marchevskiy I K and Puzikova V V 2016 Numerical simulation of the flow around two fixed circular airfoils positioned in tandem using the LS-STAG method Journal of Machinery Manufacture and Reliability45 (2) pp 130-136
[9] Morch K A 1976 Dynamics of Cavitation bubbles and Cavitating liquids *Treatise on Materials Science and Technology* **16** pp 309-335

[10] Light K H 2005 *Development of cavitation erosion resistant advanced material system* (Orono, Maine USA: The University of Maine) 76 p

[11] Zharkovsky A A and Pospelov A Yu 2014 The use of 3D methods for the flow calculation, characteristics prediction and optimization of the shape of flow parts of hydraulic turbines *Hydraulic engineering* (11) pp 104-109

[12] Lomakin V O and Petrov A I 2012 Verification of the calculation results in the package of hydrodynamic modeling STAR-CCM+ of the flow part of the centrifugal pump AX 50-32-200 *Higher Educational Institutions Bulletin. Sociology. Economics. Politics* p 6

[13] Lomakin V O and Bibik O Yu 2017 Influence of the empirical coefficients in the Rayleigh-Plesset model on the calculated cavitation characteristics of a centrifugal pump *Hydraulics* (3)

[14] Patankar S 1984 *Numerical methods for solving problems of heat transfer and fluid dynamics* (Moscow: Energoatomizdat Publ.) 152 p

[15] Chaburko P S, Lomakin V O, Kuleshova M S and Baulin M N 2016 Comprehensive optimization of the flow part of the hermetic pump using the LP τ method *Pumps. Turbins. Systems* (1) pp 55-61

[16] Lomakin V O and Chaburko P S 2015 Influence of flow twist on the hydraulic efficiency rate of pump *BMSTU Engineering Bulletin. E-Magazine* (10)

[17] Karelin V Ya 1976 Cavitation phenomena in centrifugal and axial pumps (Moscow: Mashinostroenie Publ.) p 325