

Investigation of the influence of wear in impeller seals on the axial force in double suction pumps

N Isaev^{1,2}, G Budaev¹, D Danilov¹ and K Dobrokhodov¹

¹Bauman Moscow State Technical University

²E-mail: isaev.nikita@bmstu.ru

Abstract. This article is devoted to the investigation of axial force on the rotor of the double suction centrifugal pump at unequal wear of impeller seals. The annular seal of the double suction centrifugal pump is considered as an object of the study. The distribution of the fields of velocities and pressure in the seals of the pump are obtained by numerical modeling. The leakage rate resulting from the simulation is compared to the existing formulas.

1. Introduction.

Double suction pumps [1,2,3] are horizontal, centrifugal, single-stage pumps with double suction impeller. The characteristics of these pumps are characterized by high efficiency and high permissible suction height (low value of allowable NPSH).

The pump housing of this type has an axial connector. The pressure and suction pipes of this type of the pumps are located at the bottom of the housing. This construction allows the D-type pumps to be assembled and disassembled without disconnecting the piping and the motor. The rotor bearings are antifriction bearings.

The pumped liquid can be water and other liquids with similar chemical parameters, the temperature of the liquid should not exceed 85 ° C, and the viscosity should not exceed 36 cSt.

The investigation was carried out for a pump with the following parameters [4](Table 1).

Table 1.

Parameter	Head, m	Flow rate, m ³ /h	Rotational speed, rpm	Power, kW	Gap of the impeller seal mm
Value	50	315	2900	56	0,2

Double suction pumps (Fig. 1) are designed to have no axial force during normal working conditions, but the impeller seals may wear out unequally during operation [5,6]. As a result of wear of the gap, the leakage through one of the seals changes, which in its turn leads to asymmetry of the pressure distribution on different sides of the impeller and the appearance of unpredictable axial force, which leads to increased loads on the bearings and the reduction of service life.

Therefore, it is necessary to investigate the effects of wear on the annular seal in order to be able to predict the axial forces to prevent problems with the unit.



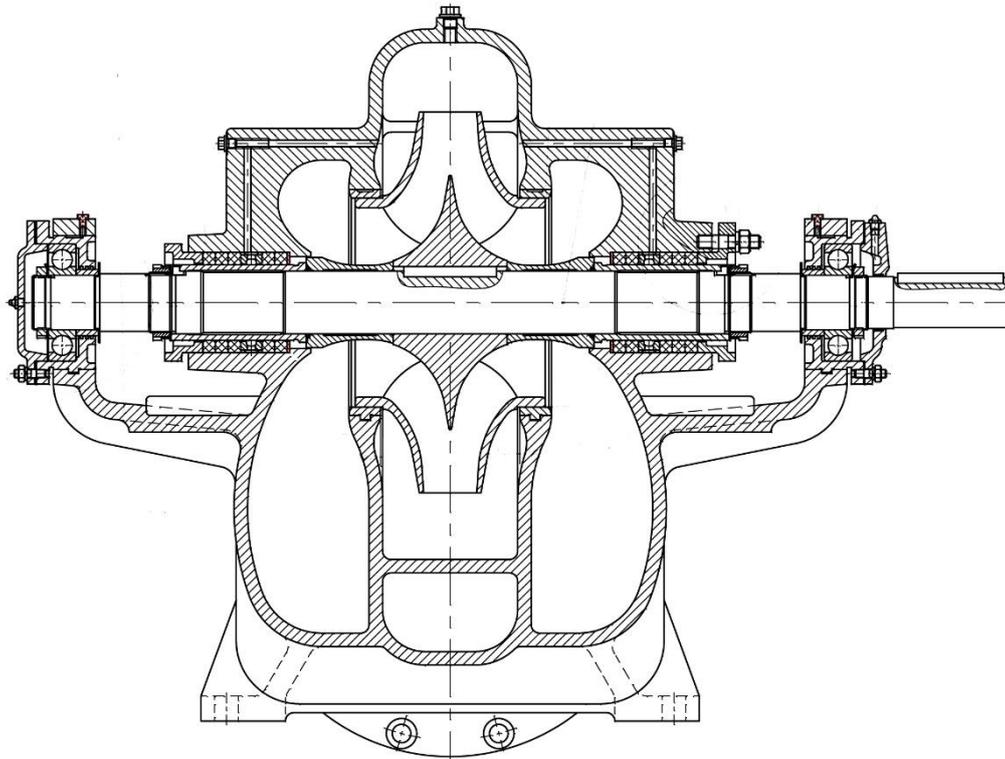


Figure1. Double suction pump.

2. Mathematical model and methods.

This paper uses the model of multiphase flow of incompressible liquid ($\rho = \text{const}$). Numerical modeling is based on solving discrete analogues of basic equations of hydrodynamics [7,8,9,10,11,12]:

Mass conservation equation (continuity equation)

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

where \tilde{u}_j — average value of liquid velocity in the projection on the j-axis ($j=1,2,3$);

Equation of motion conservation (Reynolds averaging):

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

where U, P — averaged speed and pressure;

$\tilde{T}_{ij}^{(v)} = 2\mu\tilde{s}_{ij}$ — viscous stress tensor for incompressible liquid;

$\tilde{s}_{ij} = \frac{1}{2} \left[\frac{\partial \tilde{u}_i}{\partial x_j} + \frac{\partial \tilde{u}_j}{\partial x_i} \right]$ — viscous stress tensor for incompressible liquid;

$\rho u_i u_j$ — Reynolds stresses.

The Reynolds stresses are modeled via the $k-\omega$ SST turbulence model, which proved its validity in the calculation of dynamic pumps

The secondary flows of the pump unit was simulated on a grid consisting of ~6 million cells. Cells are multi-faceted at the core of the flow and prismatic near solid walls. Volume mesh for the gaps is shown in Fig. 2.

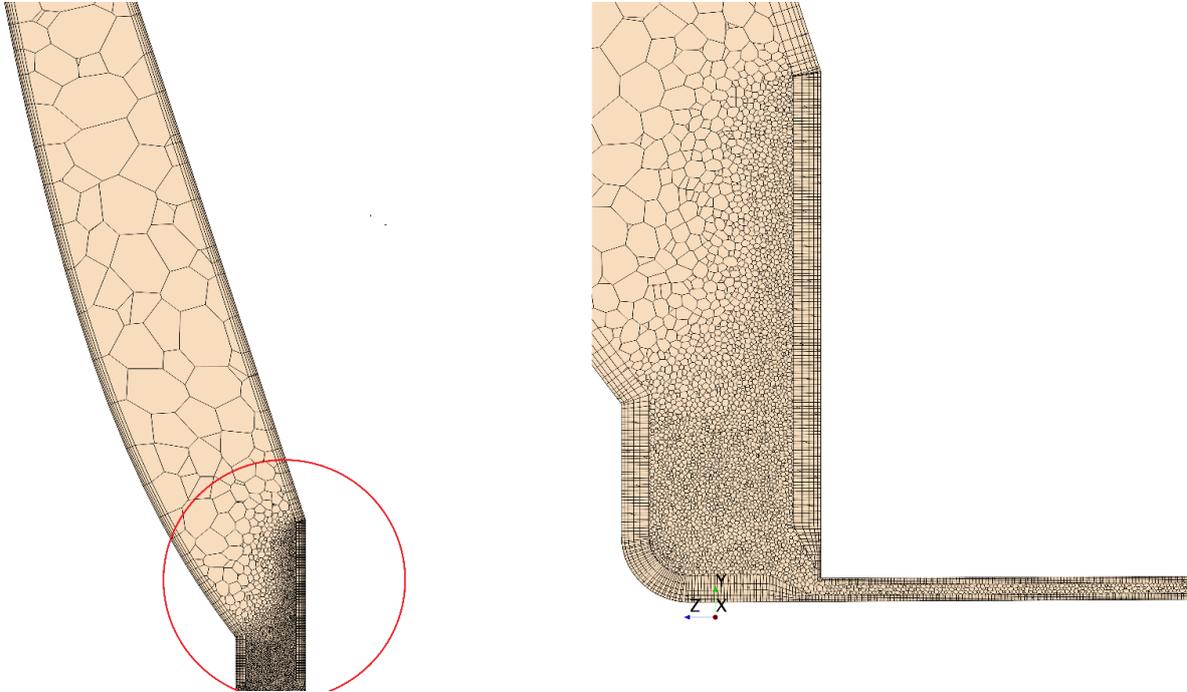


Figure 2. Volume mesh (The research area is enlarged).

Analytical calculation of the impeller seal flow rate is used to check the accuracy of the numerical simulation results, since the axial force depends on the leakage rate:

$$Q_L = \mu \pi D_S \delta \sqrt{2gH_S};$$

where:

δ – gap width;

D_S – impeller seal diameter;

μ – discharge coefficient;

H_S – pressure drop on the seal;

$$\mu = \frac{1}{\sqrt{\zeta_{in} + \lambda_S \frac{c}{2\delta} + \zeta_{out}}};$$

where:

ζ_{in}, ζ_{out} – seal inlet and outlet resistance coefficients;

c – gap length;

λ_S – gap friction coefficient;

$$H_S = H_T - \frac{V_{2U}^2}{2g} - \frac{U_2^2}{8g} \left(1 - \frac{D_S^2}{D_2^2} \right);$$

where:

H_T – theoretical head;

V_{2U} – circumferential component of the absolute flow rate
at the impeller outlet;

U_2 – circumferential velocity at the impeller outlet;

D_2 – outlet diameter;

3. Results and analysis.

For the purpose of numerical modeling, models of flow gaps were created together with impeller seals. The segment model was used to reduce the calculation time (Fig. 3).

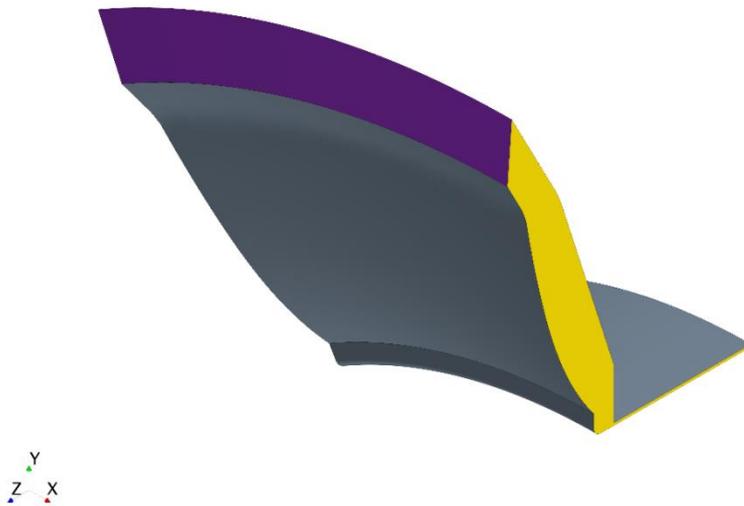


Figure 3. Impeller cavities segment.

When simulating, the gap of the impeller seal varied from 0.1 mm to 0.4 mm with a increment of 0.05 mm. The following boundary conditions are specified for the simulation: total inlet pressure — 0.55 MPa and outlet pressure — 0 MPa. To visualize the flow pattern of the gap appropriate scenes were created, Fig. 4, Fig. 5 show the scalar pressure distribution scene and velocity distribution vector field for the annular seal width $\delta = 0,2\text{mm}$. Leakage flow rate and force generated by the fluid flow in the gap are obtained.

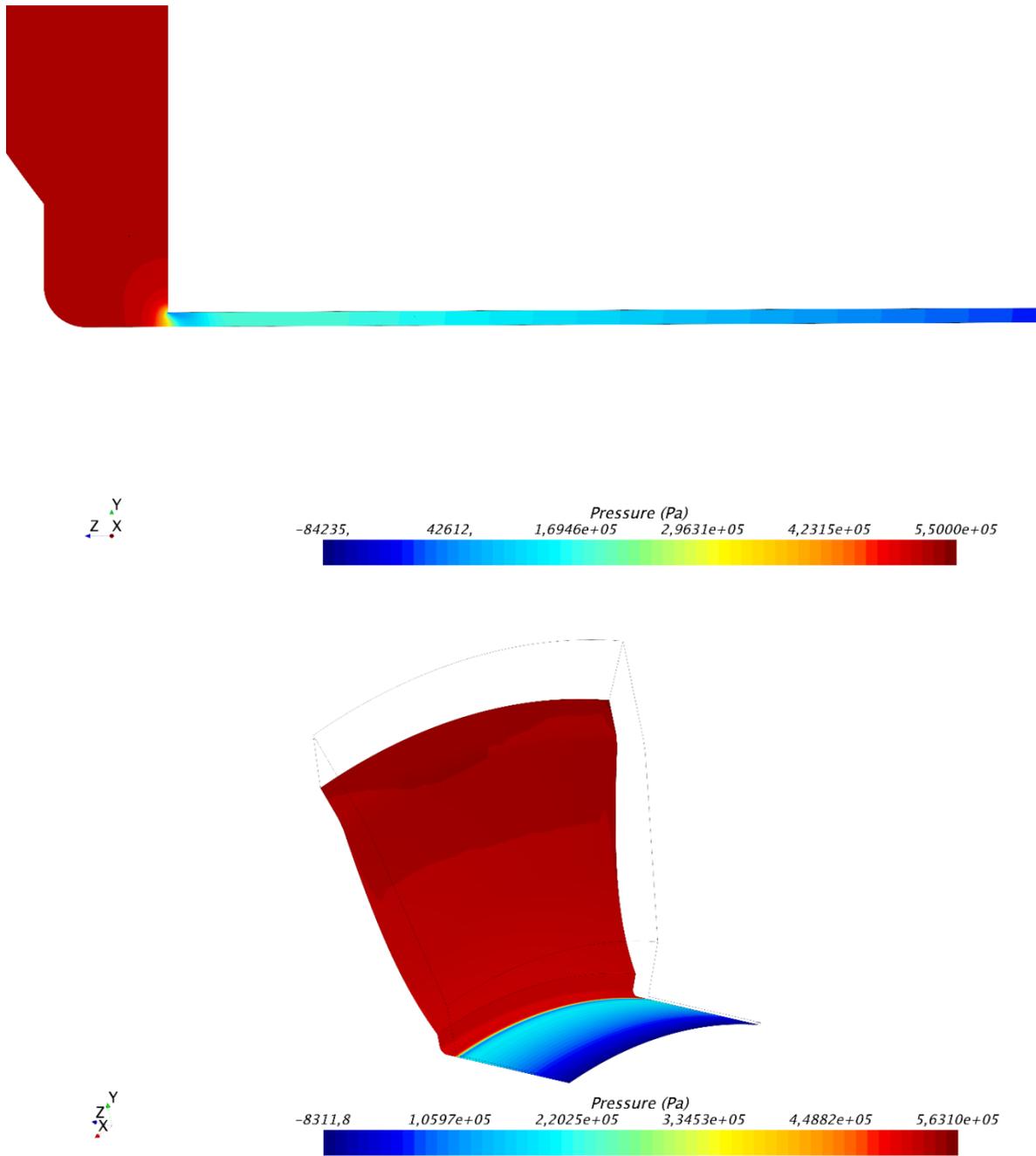


Figure 4.Scalar pressure distribution scene.

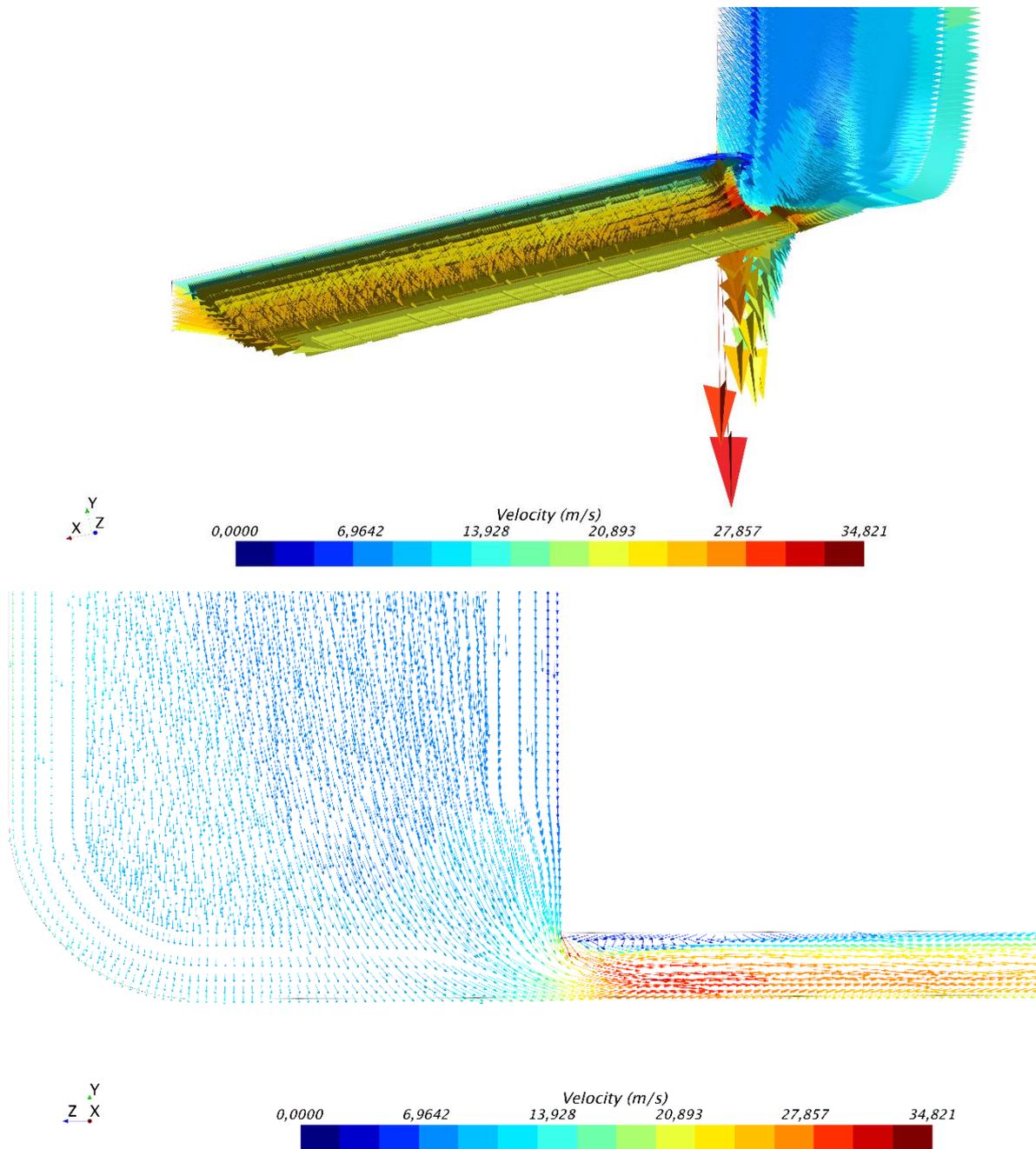


Figure 5. Vector velocity distribution field.

The data obtained during the analytical and numerical calculations are presented in Fig. 6. As a result of comparison [13] it may be concluded that the values of hydrodynamic modelling, are appropriate, the difference in 15% a between the results is explained by incorrect behavior of a current theory when the gap width is considerably more than normal values. Additional experimental research is required to refine the theory.

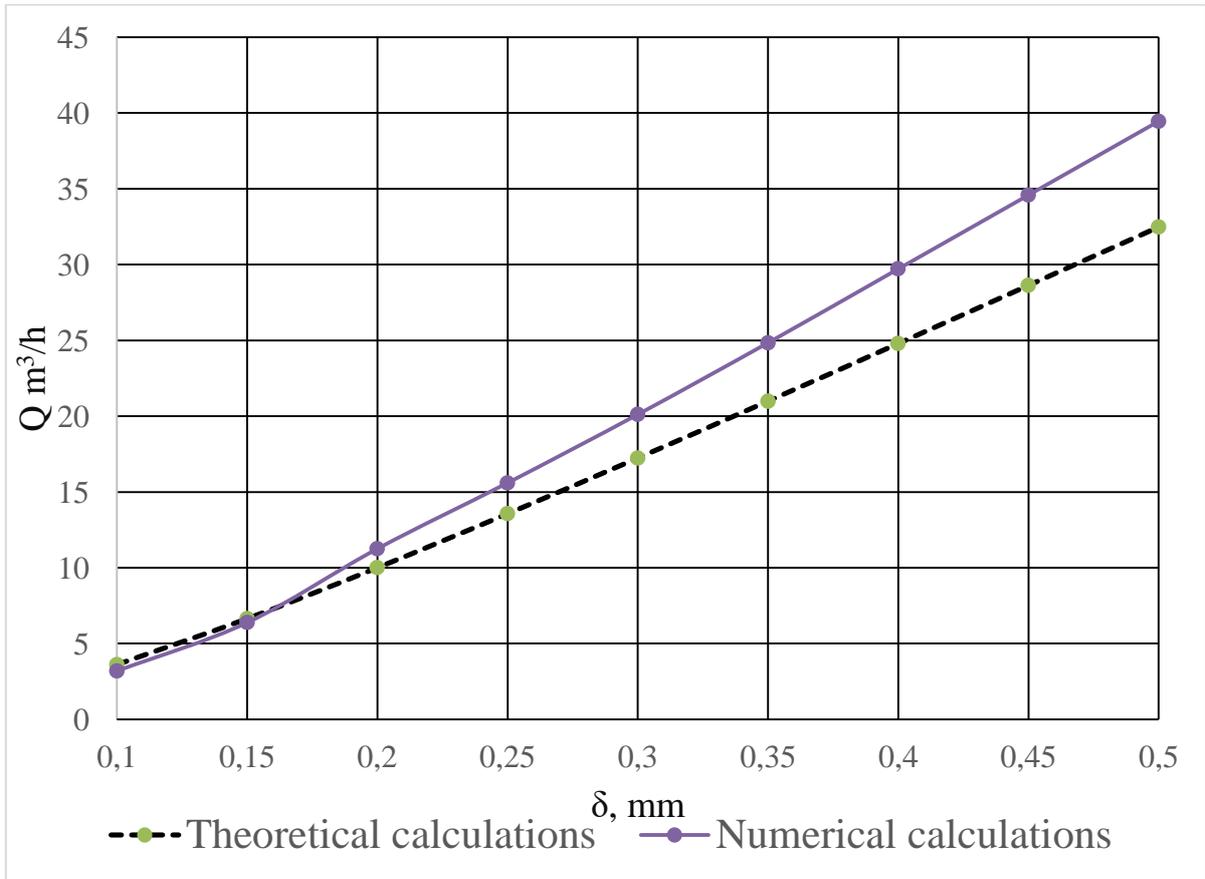


Figure 6. Leakage flow rate.

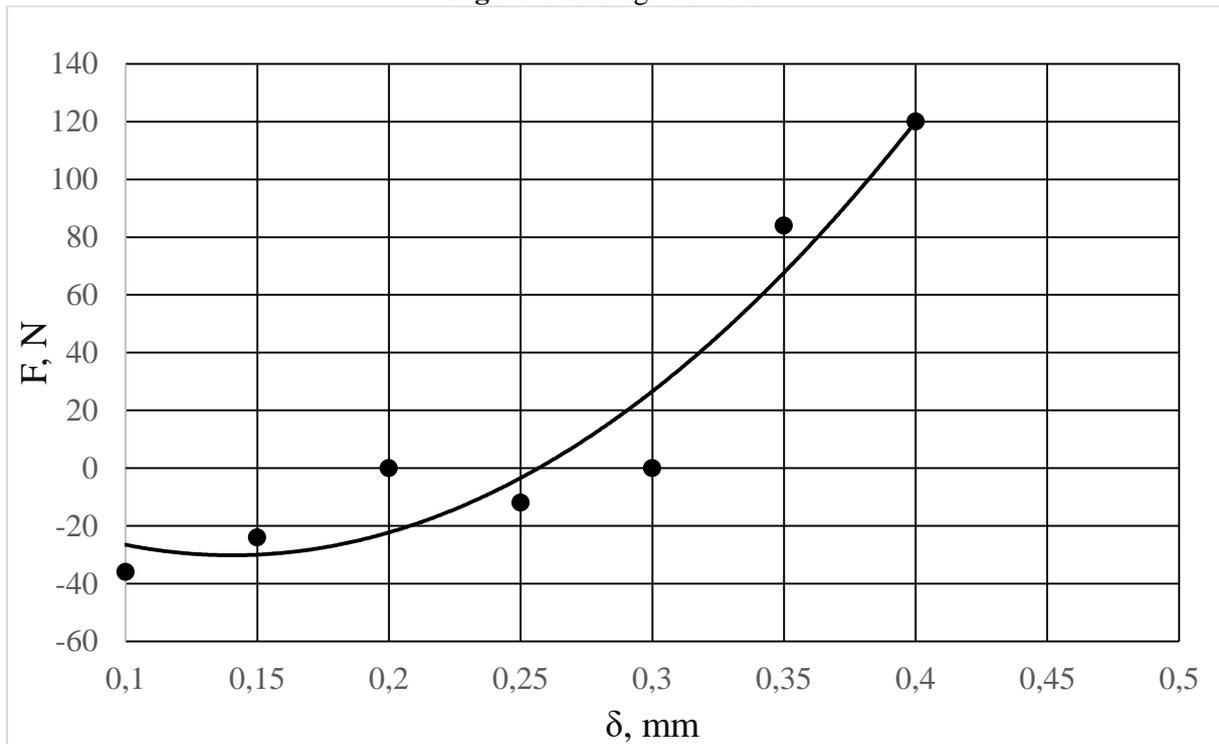


Figure 7. Axial force

An analysis of the dependence of the axial force [14], acting on the impeller on the width of the gap (Fig.7) was carried out. Axial force is the difference between two forces acting on the rotor from different sides. It occurs as a result of pressure difference on the outer covering discs of the impeller. Therefore, we calculate it as a difference of force at a normal gap ($\delta = 0,2\text{ mm}$) and the force obtained in the calculation of the model with the current thickness of the gap.

$$F_a = F(\delta_i) - F(\delta = 0,2\text{ mm})$$

$$\delta_i = 0,1 + 0,05i; i = 0 \dots 6$$

4. Conclusions.

According to the results of the performed investigation, the dependence of leakage flow rate and axial force on the gap of the seal was obtained, which makes it possible to make a conclusion about the change of forces acting on the impeller depending on the wear of the seal.

As the gap increases, the flow rate increases and the value of the force acting on the cavity side of the pump rotor increases.

With an increase in the gap difference of the impeller seals, the asymmetry of the loads on the impeller driven discs appears and leads to the appearance, and subsequently to an increase in the axial force.

References.

- [1] Hatano, S., Kang, D., Kagawa, S., Nohmi, M., & Yokota, K. (2014). Study of cavitation instabilities in double-suction centrifugal pump. *International Journal of Fluid Machinery and Systems*, 7(3), 94–100.
- [2] Song, Y., & Luo, Y. (2019). Influence of impeller vane arrangement on efficiency performance and pressure fluctuations of a double-suction centrifugal pump. Paper presented at the IOP Conference Series: Earth and Environmental Science, , 240(3)
- [3] Maeda, T., Tsuru, W., Kang, D., Tsuneda, T., Kagawa, S., An, B., . . . Yokota, K. (2019). The suppression of cavitation surge in a double-suction centrifugal pump by using branch-type and slit-type accumulations installed at a pump outlet pipe. Paper presented at the IOP Conference Series: Earth and Environmental Science, , 240(3)
- [4] A Protopopov and C Jakovich 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012034
- [5] Wu, T., & Andres, L. S. (2019). Pump grooved seals: A computational fluid dynamics approach to improve bulk-flow model predictions. *Journal of Engineering for Gas Turbines and Power*, 141(10)
- [6] Untaroiu, A., Morgan, N., Hayrapetian, V., & Schiavello, B. (2017). Dynamic response analysis of balance drum labyrinth seal groove geometries optimized for minimum leakage. *Journal of Vibration and Acoustics, Transactions of the ASME*, 139(2)
- [7] A Protopopov and V Vigovskij 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012003
- [8] Majidi, K. (2005). Numerical study of unsteady flow in a centrifugal pump. *Journal of Turbomachinery*, 127(2), 363–371.
- [9] P Chaburko and Z Kossova 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012011
- [10] Lee, G. -, & Bae, J. -. (2019). Comparative study for the prediction of cavitating flow inside a square-edged orifice using different commercial CFD software. Paper presented at the E3S Web of Conferences, , 128
- [11] T Valiev and APetrov 2019 IOP Conf. Ser.: Mater. Sci. Eng. 492 012038
- [12] Patil, A., Sundar, S., Delgado, A., & Gamboa, J. (2019). CFD based evaluation of conventional electrical submersible pump for high-speed application. *Journal of Petroleum Science and Engineering*, 182
- [13] Zhang, W., Jiang, Q., Bois, G., Li, H., Liu, X., Yuan, S., & Heng, Y. (2019). Experimental and numerical analysis on flow characteristics in a double helix screw pump. *Energies*, 12(18)
- [14] A Protopopov et al 2019 IOP Conf. Ser.: Mater. Sci. Eng. 589 012006