Research of the flat end gap’s clearance size influence on mechanical power losses in a centrifugal pump by hydrodynamic modeling methods

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Abstract. This article highlights the problems associated with calculating disk losses in pumping units. The paper addresses the problems of a theoretical description of the process of fluid flow in narrow end clearances. By varying several parameters of the system, the authors made attempts to identify the dependence of mechanical losses on the flow rate, rotor rotation frequency, the value of the clearance and the direction of the flow. The results of the research are given in a dimensionless form, based on which the corresponding conclusions are drawn.

Introduction

At present, the theory of calculation [7] and optimization [12] of the flow path of impellers is thoroughly developed, due to which it is always possible to obtain the maximum possible value of hydraulic efficiency. Thus, due to the small effect on the overall value of efficiency, in most pumps, the calculation of disk losses provides approximate results [5]. However, in the case of small pumps, pumps operating on viscous liquids, the value of disk losses is significant and can greatly reduce the energy characteristics of the unit.

It is also worth noting that theoretical formulas for the calculation [6] often do not take into account the size of the gap and the flow rate flowing through the gap, which causes considerable errors and is not suitable to cases where high accuracy is required.

Modern methods of solving various problems allow for new approaches to their implementation. At present, in hydraulics, when calculating the fluid flow, numerical methods are used based on the solution of discrete analogues of basic equations of hydrodynamics. This approach makes it possible to obtain a picture of the distribution of physical parameters without carrying out physical experiments and their corresponding costs.

This research aims to create a correct model for further hydrodynamic modeling [13], to determine ways to reduce losses due to disc friction in various parts of hydraulic units, for example, on the cover disks of impellers, on impeller seals and on other end clearances with rotating and stationary walls.
Methods
The power of disk friction for one side of a rotating disk can, for example, be determined by the formula [3]

\[ N = C_f \cdot \rho \cdot \omega^3 \cdot R^5, \]

where \( C_f \) – empirical total friction coefficient equals to [1]

\[ C_f = C_{f0} + \Delta C_f, \]

where \( C_{f0} \) – friction coefficient in the absence of flow equals to

\[ C_{f0} = \frac{0.0277}{Re^{0.2} \cdot (\delta^*)^{0.2}}, \]

\( \Delta C_f \) – correction for the specified flow. For the flow from the centre to the periphery

\[ \Delta C_f = 0.42 \cdot 10^{-2} \cdot (\delta^*)^{0.75} \cdot Ga^{0.3} / K^{0.8}, \]

for the flow from the periphery to the centre

\[ \Delta C_f = 3.25 \cdot 10^{-2} \cdot (\delta^*)^{0.3} / (Ga^{0.05} \cdot K^{0.4}), \]

where \( Ga \) – Galilei number,

\[ Ga = \frac{g \cdot R^3}{\nu^2}, \]

\( K \) – coefficient characterizing the ratio of the circumferential speed of the disk periphery to the average radial speed of the fluid in the clearance,

\[ K = \frac{\omega \cdot R}{u_{cp}} = \frac{\pi^2 \cdot R^2 \cdot \delta \cdot n}{15 \cdot q}, \]

where \( q \) – volumetric flow rate through the clearance.

In addition to theoretical calculations, numerical method calculations were carried out in this paper as well. The numerical simulation method is based on solving discrete analogues of basic equations of hydrodynamics. In the case of the chosen model of an incompressible fluid (\( \rho = \text{const} \)), these are:

The equation of continuity of a liquid medium [4]:

\[ \frac{\partial \bar{u}_x}{\partial x} + \frac{\partial \bar{u}_y}{\partial y} + \frac{\partial \bar{u}_z}{\partial z} = 0, \]

where \( \bar{u}_i \) – time-averaged projections of fluid speed on the corresponding axes;

Equation of the change in the amount of motion averaged across time:

\[ \rho \left( \frac{\partial \bar{u}_i}{\partial t} + \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} \right) = - \frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \tau_{ij}^{(v)} - \rho \langle u_i u_j \rangle \right), \]
where $\bar{p}$ – average pressure;

$$
T_{ij}^{(v)} = 2 \mu s_{ij} – \text{viscous stress tensor for an incompressible fluid} ;
$$

$$
s_{ij} = \frac{1}{2} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) – \text{deformation speed tensor};
$$

$$
\rho (u_i u_j) – \text{Reynolds stresses}.
$$

The system of Reynolds equations [4] is not closed due to the presence of unknown Reynolds stresses. The system is closed using the k-ω SST turbulence model [9]. This model combines the k-ω and k-ε models [2], [8]: the first is used in the near-wall area, the second in the central part of the flow.

This model includes two equations for the transfer of turbulence parameters [11]:

1. Turbulence kinetic energy transfer equation

$$
\frac{\partial k}{\partial t} + \bar{u}_j \frac{\partial k}{\partial x_j} = P_k - \beta \cdot k \omega + \frac{\partial}{\partial x_i} \left[ (\nu + \sigma_k \nu_T) \frac{\partial k}{\partial x_i} \right],
$$

where

$$
k = \frac{1}{2} (u_x^2 + u_y^2 + u_z^2) – \text{kinetic energy of turbulence};
$$

$$
\bar{u}_i – \text{speed ripple};
$$

$$
P_k – \text{turbulence energy generation term};
$$

$$
\omega – \text{relative speed of turbulence dissipation};
$$

$$
\nu_T – \text{turbulent viscosity}.
$$

2. The transfer equation for the relative dissipation rate of turbulence energy

$$
\frac{\partial \omega}{\partial t} + \bar{u}_j \frac{\partial \omega}{\partial x_j} = a S^2 - \beta \omega^2 + \frac{\partial}{\partial x_i} \left[ (\nu + \sigma_\omega \nu_T) \frac{\partial \omega}{\partial x_i} \right] + 2 (1 - F_j) \sigma_{\omega 2} \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}
$$

The Reynolds stresses in the equations of dynamics are found on the basis of the Boussinesq hypothesis:

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

where $\delta_{ij}$ – Kronecker symbol.

Using the empirical closure coefficients of these equations [10], it is possible to obtain a numerical solution of the turbulent fluid flow in the computational area.

The investigated geometry of the fluid flow in the end clearance is a disk with rotating (rotor) and static (stator) walls. The geometric parameters of the models have the following meanings:

$$
R_1 = 20 \text{ mm} – \text{inner radius of the rotor},
$$

$$
R_2 = 40 \text{ mm} – \text{outer radius of the rotor},
$$

$$
r_1 = 15 \text{ mm} – \text{inner radius of the flow model},
$$

$$
r_2 = 45 \text{ mm} – \text{outer radius of the flow model},
$$

$$
\delta = 0.1 \ldots 2.0 \text{ mm} – \text{clearance width}.
$$

To reduce the influence of reverse currents, which significantly reduces the calculation accuracy near the boundaries, rings with fixed 5.0 mm wide walls were created on the inner and outer parts of the geometry, which also contributes to the establishment of the flow at the moment of entering the research area. An image of the examined end clearance is shown in Picture 1. It shows the boundaries of the geometry of the simulated flow (3, 6), the boundaries of the investigated flow area (4, 5), which are limited by the geometric parameters of the rotor, and the clearance size - $\delta$. 

When simulating a fluid flow, it is necessary to break the geometry of the flow into cells for which the equations given above [9] will be applied. A model that generates polyhedral cells was chosen for this, as well as a model for constructing prismatic cells and thin layers, since in this case there is a plane flow. The base size was taken as 1 mm. The maximum and minimum cell sizes were specified as a percentage of the base size - 100% and 50%, respectively. Prismatic layer thickness: 0.05 mm on the stator and 0.13 mm on the rotor. The stretching of the prismatic layer occurs with a factor of 1.2. A total of 8 layers are created on the stator and 25 on the rotor. The number of thin layers in the flow core is 10. This number of layers on the rotor wall allow one to accurately determine mechanical losses (Figure 2) at a relatively high calculation speed, an increase in this parameter will not lead to a strong improvement in accuracy, but will require significantly more computing time and resources. An example of the generated mesh is shown in Figure 3.
Thus, to obtain correct results and experience calculation errors of less than 5%, it is necessary to use at least 25 prismatic layers when constructing.

In this study, the following parameters were changed: rotor speed - 1450, 2900 and 5000 rpm; clearance flow rate - 10 l/h, from 100 to 1000 l/h with a step of 150 l/h; the value of the end clearance - from 0.1 to 2.0 mm with a certain pitch, the rest of the values were selected to find the local minimum of mechanical losses; flow direction - from the centre to the periphery and from the periphery to the centre.

To model the flow, it is necessary to specify a physical model of the flow, in addition to creating a volumetric mesh. This calculation was carried out as a time-stationary problem for a liquid with constant density. It was also specified that the flow is turbulent and has constant density.

The boundary conditions were set: mass flow rate at the inlet, converted from l/h to kg/s, and the outlet pressure. The research was carried out for a liquid with a density $\rho = 778 \text{ kg/m}^3$ and dynamic viscosity $\mu = 1.04 \cdot 10^{-3} \text{ Pa} \cdot \text{s}$.

All initial parameters and obtained data were reduced to a dimensionless form:
characteristic rotor speed $n' = 2900$ rpm;
characteristic linear dimension $L' = R_2$;
Reynolds number $Re = \frac{\omega \cdot L'^2}{\nu} = \frac{\pi \cdot n \cdot R^2}{30 \cdot \nu}$;
reduced rotor speed $n^* = n/n'$;
reduced consumption...
\[ Q^* = \frac{Q}{\frac{\omega \cdot L^3}{\pi \cdot n \cdot R_2^3}}; \]

reduced clearance thickness

\[ \delta^* = \frac{\delta}{L'} = \frac{\delta}{R_2}. \]

characteristic power \( N' \) – without correction for the presence of flow through the gap

\[ N' = C_f \cdot \rho \cdot \omega^3 \cdot R_2^5, \]

characteristic power \( N'' \) – corrected for the presence of flow in the gap

\[ N'' = C_f \cdot \rho \cdot \omega^3 \cdot R_2^5. \]

dimensionless power

\[ N^* = \frac{N}{N''}. \]

**Results**

In this study, 10-15 numerical calculations were made for the size of the gap at different flow rates, rotation frequencies and flow directions, which in total adds up to more than 500 models.

![Picture 4 - dependence of disk losses on the gap size at different flow rates, \( n = 5000 \) rpm, fluid flow from the centre to the periphery.](image_url)
The data obtained was converted to a dimensionless form. The dependences of the dimensionless losses $N^*$ on the relative gap are shown in Pictures 6 for the flow from the centre to the periphery and 7 for the flow from the periphery to the centre.

![Picture 6](image)

**Picture 6 - dependence of dimensionless mechanical losses on the value of the relative gap at different flow rates. The direction of the flow from the centre to the periphery, $n^* = 1.00$.**
Figures 8-11 show the dependences of losses calculated by theoretical formulas without taking into account the flow rate in the slot N' as well as with the correction for including the flow rate N' and using numerical methods N. These graphs show that theoretical formulas give overestimated results for lower flow rates and underestimated results for larger ones, so, when using these formulas, an accurate result is obtained only with certain system parameters. It is also worth noting that an increase in mechanical losses occurs at large gaps, which is not taken into account in theoretical formulas.

Picture 7 - dependence of dimensionless mechanical losses on the size of the relative gap at different flow rates. The direction of the flow from the periphery to the centre, \( n^* = 1.00 \).

Picture 8 – dependence of mechanical losses during flow from the centre to the periphery, rotation frequency \( n^* = 1.72 \), flow rate \( Q^* = 0.83 \cdot 10^{-3} \).
9 – dependence of mechanical losses during flow from the centre to the periphery, rotation frequency $n^* = 1.72$, flow rate $Q^* = 7.05 \cdot 10^{-3}$.

Picture 10 – dependence of mechanical losses during flow from the periphery to the centre, rotation frequency $n^* = 1.72$, flow rate $Q^* = 0.83 \cdot 10^{-3}$.

Picture 11 – dependence of mechanical losses during flow from the periphery to the centre, rotation frequency $n^* = 1.72$, flow rate $Q^* = 7.05 \cdot 10^{-3}$.

The resulting graphs require field tests to determine the accuracy of the performed calculations.
Conclusion
For a small range of variable values, there is a match of the results obtained by theoretical formulas and calculated using numerical methods: when flowing from the centre to the periphery at a flow rate $Q^* = 0.003 \ldots 0.006$ and $\delta^* < 0.025$, when flowing from the periphery to the centre at a flow rate $Q^* = 0.006 \ldots 0.009$ and $\delta^* < 0.050$. For most dependencies, the error is 20-50%. Therefore, this technique is not relevant for small pumps.

An important feature is an increase in disk losses at larger gaps, for example, when flowing from the centre to the periphery at $\delta^* = 0.050$, the value of the relative power loss reaches the value $N^* \approx 1.4$, which is not taken into account in theoretical formulas.

The disadvantages of theoretical calculations described above create the need for a number of experiments to verify the calculations and create an improved method for determining mechanical losses.

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