The ways to increase the efficiency of the stage of low specific speed

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Abstract. The paper raises the question of the possibility of reducing the energy loss of the stage of the centrifugal pump on the disk friction, the relative value of which for the stages of low and ultralow speed can be a significant proportion (up to 40%) of all losses. This is seen as one of the effective ways to increase the efficiency of such stages of centrifugal pumps.

1. Introduction

High efficiency (η) is one of the main indicators that determine the choice of pump type, method of calculation and method of manufacture. Nowadays, the normal value of total efficiency is 80 - 95%, depending on their type and size. The achievement of such indicators became possible due to the development of the theory of losses and ideas about their physical essence on the basis of viscous fluid mechanics. As you know, pump losses can be divided into three main categories: hydraulic, mechanical and volumetric. When the pumping equipment are developed, the greatest attention is paid to reducing hydraulic losses. However, especially in centrifugal stages with a low speed factor, the internal mechanical energy losses have the greatest weight. At the same time, the share of volume losses remains virtually unchanged, and the relative share of hydraulic constantly decreases with decreasing coefficient n₃.

Figure 1. Impeller side room flow.
In turn, there are three main groups of mechanical losses: losses due to disk friction; friction in bearings; friction in the seals. Mechanical losses can also include losses of hydraulic braking. But when the pump is running in the designed mode, they are absent. The least researched in that the most interesting are losses on disk friction. Therefore, they will be further considered in this paper.

Disk friction losses are understood as the power of friction of the outer surface of the wheels against the liquid. It consists of the friction power of the side surfaces and the friction power of the cylindrical part of the rim. When the disk rotates in a confined space (Fig. 1), the liquid between the disk and the wall of the housing rotates at an angular velocity equal to half the angular velocity of the disk. In this case, the moment of friction between the fluid and the disk is balanced by the moment of inhibition due to the friction of the fluid against the walls of the housing. The liquid particles directly adjacent to the surface of the disk rotate at an angular velocity equal to the speed of the disk. The centrifugal forces acting on them are not balanced by the pressures in the main flow. These particles are thrown from the center to the periphery of the disk. Due to the continuity of the flow along the walls of the housing, a reverse flow to the center is formed. Thus, a secondary flow in the form of two annular vortices is superimposed on the main motion [1].

In a simplified form, the amount of losses due to disk friction is derived from the Reynolds number. But in the works of Pantell and studies conducted at the Department of Applied Hydro- and Aeromechanics Department SSU (in particular I.O. Kovalyov and M.M. Olada), it is proved that the losses depend on the following parameters: flow regime (Reynolds number $R_e$), the relative width of the lateral sinuses $\frac{s}{r_2}$, relative surface roughness $\frac{\Delta}{r_2}$, direction and velocity of flow through the lateral sinus $Q$, sinus geometry.

2. Experimental studies of disc friction losses

Experimental studies are aimed at measuring the effect of both surface roughness and axial distance between the moving disk and the housing on the loss of friction power of the disk in centrifugal pumps.

Figure 2. Schematic drawing of test rig.

Most experiments in this area are based on the work of Pantell [2]. The test rig consists of one or two bearings 1 mounted on a shaft to which the lens or cylinder 2 to be tested is mounted. The outer part is a housing 3 consisting of one or more cylindrical parts centered relative to each other. The side windows are pressed against the ends of the cylindrical part with anchor bolts. Since the moment of friction of the fluid is measured in the housing, the moving parts must be able to rotate around the axis as smoothly as possible without contact with the stator parts.

According to Fottinger's instructions, this is achieved by attaching stuffing box seals in a special housing. The housing 3 is attached to the frame 4 by means of anchor bolts 5. The inlet allows you to supply fluid close to the axis. At the highest point of the case there is a ventilation valve with a funnel, at the lowest point - a drain valve. The test rig is a closed loop. The pump inlet is connected to the tank via a flexible connection. The pump and the test disk are connected to the motor 6 through the clutch 7 torque meter 8, which facilitates the measurement of transmitted torque, and a special speed sensor for measuring the speed 9 [3].
The aim of the research is to determine the influence of surface roughness and axial clearance on disk losses in the pump.

The surface roughness of the rotating disk is changed by machining or by gluing sandpaper with different surface roughness. The axial clearance is changed by reducing the thickness of the disk by machining. It is proposed to use a Taylor-Hobson device with a diamond pin to measure the surface roughness of the impeller, hard disk and sandpaper used. Disk friction power losses are defined as the difference between the measurement results when the housing is filled with water and the power measured when the housing was empty. So energy losses in the bearings will be eliminated from the net power loss.

As suggested by Bennett and Worster [4], Worster [5], and Varley [6], some of the energy expended on disk friction eventually returns to the impeller and helps combat bulk losses. To facilitate experimental research, the fraction of disk friction power added to the impeller power can be considered equal to the volume loss [7].

Against the general background, the work of Hergt and Prager stands out [8]. They measured the friction losses of the disk in real pumping conditions. An experimental installation with a bifurcated disk was taken. Thus, the front and rear walls of the cover were driven by two different engines running at the same speed. Thus, the power of the disk friction absorbed by the rear wall could be measured separately from the power of the impeller. The conditions of supply to the lateral sinus were determined by the speed of exit from the impeller [9].

This approach is necessary because the fluid flow conditions are completely different from those shown in the tests. The main flow at the outlet of the channels of the impeller has a large circumferential velocity and any flow into the lateral sinus has a momentum. It seeks to accelerate lateral flow and reduce disk friction.

3. Estimated determination of disk losses

3.1. Influence of current mode on disk losses

The impellers of turbomachines usually rotate in narrow cylindrical chambers. In this case, the hydrodynamics of the flow differs significantly from the flow around the disk in unlimited space. The circular velocity of the medium rotating with the disk is no longer zero at the outer boundary of the boundary layer of the disk. If near the disk the radial flow is directed from the center to the periphery, then near the walls of the body the radial velocity is directed to the center [10].

With a very small gap width \( S \) between the disk and the housing, when the individual boundary layers on the walls disappear, the distribution of circular velocities has a character similar to the distribution of velocities in the flow of Couette-Taylor. With Reynolds numbers \( R_e = \frac{r^2 \omega}{v} \) there is a laminar flow of Couette-Taylor with a linear velocity profile. The coefficient of friction will be equal to:

\[
c_f = 2\pi \frac{R_e}{S} \frac{1}{R_e}
\]  

At large Reynolds numbers, the flow in the gap becomes turbulent. If at the same time separate boundary layers on walls of a disk and the case do not appear yet, for definition of friction forces it is possible to apply the corresponding relations for a turbulent current of Cuette. The coefficient of friction will be determined by the formula:

\[
c_f = 0.0277 R_e^{-0.2} \left( \frac{S}{R_e^2} \right)^{-0.2}
\]  

The obtained ratio is consistent with the experimental data described in the work of Pantell [2]. It can be applied only to a certain range of Reynolds numbers \( 10^4 \leq R_e \leq 10^6 \). When there are no separate boundary layers on the walls of the disk and the body.

The studies of Daily and Nece [11] allowed to obtain equations to cover different flow regimes from laminar flow with connected boundary layers to turbulent flow with separated boundary layers.
Table 1. Coefficient of friction at different flow modes.

| №  | Flow mode                                             | Reynolds number | Coefficient of friction          |
|----|-------------------------------------------------------|-----------------|----------------------------------|
| 1  | Laminar flow with connected boundary layers           | \( R_e \leq \frac{8.7}{S^{1.87}} \) | \( k = \frac{\pi}{2} \cdot R_e \cdot \frac{S}{2h} \) (3) |
| 2  | Laminar flow with separate boundary layers            | \( \frac{8.7}{S^{1.87}} < R_e \leq 10^5 \) | \( k = \frac{0.925}{R_e^{0.5}} \cdot S^{0.1} \) (4) |
| 3  | Turbulent flow with connected boundary layers         | \( 10^5 < R_e \leq 10^6 \) | \( k = \frac{0.02}{R_e^{0.25}} \cdot S^{1/6} \) (5) |
| 4  | Turbulent flow with separate boundary layers          | \( R_e > 2 \cdot 10^5 \) | \( k = \frac{0.0255}{R_e^{0.2}} \cdot S^{0.1} \) (6) |

3.2. Influence of surface roughness of a disk

The roughness of the surface of the rotating disk is manifested in a marked increase in friction. However, as in the case of flow in the tubes and along the flat wall, if the height of the roughness is much less than the thickness of the laminar sublayer \( \delta_l \), the friction does not increase. This is due to the fact that the tubercles flow around in this case without separations and vortices. This is a mode without the manifestation of roughness.

If the height of the bumps \( \Delta \) has the same order of magnitude as the thickness of the laminar layer, then the effect of roughness begins to appear - a transitional or intermediate mode.

Finally, the roughness is significantly manifested in the case when the roughness bumps extend significantly beyond the laminar sublayer. The detachable flow around the tubercles leads to the fact that the structure of the flow and friction become independent of the Reynolds number and depend only on the relative roughness \( \frac{\Delta}{R_2} \). This is a mode of developed roughness [10].

Experimental studies of the friction of a rough disk rotating in a large volume of fluid are given in the work of Kanaev [12]. Steel disk radius \( R_2 = 0.135m \) and thickness \( b = 0.01m \) was covered with artificial sand roughness \( \Delta = 0.45 \text{ mm} \). The disk rotated in a large tank filled with water. The maximum achieved value of the coefficient of friction of the disk and rim is equal to \( c_m = 0.027 \). If we exclude the friction of the rim, based on the formula \( c_m = c_m \left( 1 + 2.5 \frac{b}{R_2} \right) \), then for the coefficient of friction of the two sides of the disk we obtain the value \( c_m = 0.0228 \) at \( \frac{R_2}{\Delta} = 300 \). Theoretical calculations give a very close value \( c_m = 0.0229 \). It should be noted that the data of Kanaev's experiments on the friction of a rough steel disk in mercury do not agree with theoretical calculations.

This is entirely due to the horizontal location of the disk adopted in these experiments. In this case, the rough disk begins to noticeably affect the volumetric forces due to the force of gravity of the deep layer of heavy fluid located above the disk. Friction resistance increases significantly and becomes proportional to the roughness.

3.3. Influence of relative width of lateral sinuses

At a very small distance between the disk of the impeller and the body \( S < \delta \) (\( \delta \) - the thickness of the boundary layer) will be observed Couette-Taylor flow. That is, the velocity distribution in the axial gap of the side wall is linear. The average speed will be about half the circular speed of the disk. As the axial clearance of the side wall increases, the friction losses are reduced to a minimum (which is of no practical significance, as it is not really possible to do so) before it increases again. As already mentioned, the effect of the axial gap of the side wall is minimal in turbulent flow, while it is significant in laminar flow. The more the surfaces of the sidewall gap cover get wet, the slower the liquid rotates Pantell's experiments showed that the relative effect of the axial gap \( \frac{\Delta}{R_2} \) in the presence of a casing for a rough disk is the same as for a smooth disk. It is schematically shown in Figure 3.
As you can see, the minimum value of the coefficient of friction of the disk is observed at point 2 at the width of the axial gap $S = 2\delta$.

With a further decrease in the axial gap $S < 2\delta$ (sections 1–2), a significant increase in the coefficient of friction is observed. This can be explained by the fact that when applying the boundary layers, the velocity gradient in the fluid flow increases sharply. In sections 2–3 we can see a gradual increase in the magnitude of the friction moment. Which is caused by the cost of energy to disperse the fluid inside the flow core.

With the further growth of the axial gap (sections 3–4), the flow can be considered as for the rotation of the disk in free space. The value of the coefficient of friction will be almost unchanged.

In this case, according to numerous experimental data, the ratio of the minimum value $(c_M)_{\text{min}}$ to the value for the "free" disk $(c_M)_{fr}$ is equal to:

$$\frac{(c_M)_{\text{min}}}{(c_M)_{fr}} \approx 0.474$$

(7)

According to the theoretical calculations of Karman and Okay - Hasegawa we get a little more value:

$$\frac{(c_M)_{\text{min}}}{(c_M)_{fr}} = \frac{0.0714}{0.146} = 0.474$$

(8)

After mathematical calculations, the following formula was obtained for the coefficient of resistance of the rough disk rotating in the casing for the mode of developed roughness:

$$c_M \approx 0.051 \left( \frac{\Delta}{R^2} \right)^{0.272}$$

(9)

The figure shows a comparison with this formula of experimental data obtained by measurements on the hydraulic brakes of the disk structure. The experiments were performed in the laboratory of turbomachines of the Nevsky Machine-Building Plant [13].

Power was determined by the angular velocity and torque transmitted by water from the rotating disk to the body by the brakes.

Simultaneously measured the water temperature at the inlet and outlet. The friction moment of the bearings, which is also transmitted to the housing, is very small. Therefore, they can be neglected.
In Figure 4 shows the following values $c_m$ for different hydraulic brakes:

**Table 2.** Geometric dimensions of the studied hydraulic brakes (line number corresponds to the number of the line in Figure 4).

| № | $R_2$, m | Roughness, mm | The relative width of the rim $\frac{b}{R}$ | Relative axial clearance $\frac{s}{R_2}$ | Relative radial gap |
|---|----------|---------------|---------------------------------|---------------------------------|-------------------|
| 1 | 0.25     | ≈ 0.03        | 0.04                            | 0.06                            | 0.068             |
| 2 | 0.25     | ≈ 0.03        | 0.04                            | 0.06                            | 0.068             |
| 3 | 0.175    | ≈ 0.0082      | 0.05                            | 0.1                             | 0.2               |
| 4 | Results Pantell for a rough disk | | | | |
| 5 | Results of Schultz-Grunov, Tsumbush for a smooth disk | | | | |

Note that in the region of Reynolds numbers, where the regime of developed roughness has not yet arrived, all four curves have the same character of dependence on $Re$, which is observed for technically rough pipes.

The obtained experimental values (Fig. 5) are located near the theoretical line. Slightly smaller in comparison with the calculated curve, the value for the hydraulic brake № 2 is explained by the fact that the casing had a roughness less than the disk. As is known from Pantell’s experiments, this should reduce the amount of friction moment. At the same time, the somewhat inflated value for the hydraulic brakes № 3 is entirely caused by the increased value of the radial gap [10].

3.4. **Influence of fluid flow through the lateral sinuses**

Fluid flow through the lateral sinus of the impeller: radially inward flow causes the appearance of the angular momentum $\rho Q c_{2u} R_2$ in the gap of the side wall, which reduces the friction of the disk, if $\frac{c_{2u}}{u_2} > 0.5$. Conversely, the flow radially directed to the periphery slows down the rotation of the fluid in the gap of the side wall of the impeller (if it has no or has a small pre-rotation). Due to this, the friction of the disk increases. The effect of leakage can be estimated by the correction factor, which was obtained from the calculations. This is roughly consistent with the measurements of Lobanov and Ross [14]. In practice, the effect of fluid flow through the lateral sinuses (reduction of disc friction) and roughness (increased disc friction) compensates for some expansion, so that the frequency can be equated for simplification [15].
3.5. Influence of pump speed
The accuracy of determining the coefficient of friction and power loss on disk friction largely depends on how reliably assessed the compliance of real friction conditions with the model, according to which these or those recommendations are given [16]. With proper design of the flow transition from the impeller to the outlet channel, which involves open articulation of the impeller and the outlet, it is possible to partially use the disk friction power in the form of kinetic flow energy flowing from the outer walls of the housing. This phenomenon is called the "pumping effect" and was described in the work of Zakharov [17]. This allows you to somewhat mitigate the impact of disk losses in the overall balance of losses of pumps with low speed. Due to the restoration of part of the kinetic energy of the flow in the axle between the wheel and the wall of the housing, the actual value of power loss per disk friction is reduced and can be determined by the following relationship:

\[ N_d' = (1 - \eta_{pe})N_d \]  

(10)

The relative losses on disk friction depend on the speed factor and can be estimated by the formula. Graphical dependence of relative disk losses on the speed coefficient is presented in Figure 6 [16]. As the author himself notes, the values of \( \varepsilon_{df} \) at \( n_s < 80 \) are slightly inflated, but nevertheless the share of disk losses in steps with a low speed coefficient is significant.

\[ \varepsilon_{df} \approx \frac{6.5}{\left( \frac{n_s}{100} \right)^2} \]  

(11)

Figure 6. Dependence of relative disk losses on specific speed.

List of symbols

- \( \eta \) – efficiency.
- \( \eta_{pe} \) – conditional efficiency of the "pump effect".
- \( \delta \) – thickness of the liquid boundary layer.
- \( n_s \) – speed factor.
- \( N_{fr} \) – friction disc losses.
- \( N_b \) – friction losses in bearings.
- \( N_s \) – friction losses in the seals.
- \( R_e \) – Reynolds number.
- \( \frac{s}{R_2} \) – relative width of the lateral sinuses.
- \( \frac{h}{R_2} \) – relative surface roughness.
- \( Q \) – flow through the lateral sinus.
- \( c_m \) – moment of friction of the disk.
- \( (c_m)_{min} \) – minimum moment of friction of a disk.
- \( (c_m)_{fr} \) – moment of friction of the "free" disk.
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