ANALYSIS OF FLOW ALONG SMOOTH SIDE OF DISC IMPELLER IN A PUMP

E M Kraeva

Department of Engines of Flight Vehicles, Siberian State Aerospace University named after academician M.F. Reshetnev,
31 "KrasnoyarskiyRabochiy" prospect, Krasnoyarsk, 660037, Russia.

E-mail: EM-Kraeva@yandex.ru

Abstract. We consider fluid flow in an axial gap around the smooth disc rotating in a closed cavity. According to the results of experiments on different working fluids in a wide range of operating modes for smooth discs, the formula is presented to determine the coefficient of fluid flow spin subject to a rate fluid flow from the periphery to the center. On the basis of the experimental data the empirical formula is obtained for the friction torque with respect to a rate fluid flow.

Modern pumps have very compact construction and low mass. They have an electric motor or a gas turbine as a rotor drive. Given the functional purpose, in addition to flowing parts, pumps contain many rotating cavities and channels with fluid or gas. These are varied hydraulic components: tip clearances and side cavities between disks of impellers and the body; axial gaps and cavities in contactless and hydrodynamic seals of the rotor; cavies between the rotor and the pump body [1, 2, 3, 4].

Achieving high performance and efficiency in a centrifugal pump, in addition to effective energy conversion in its flowing part, substantially depends on perfection of the hydraulic path with a complex internal interaction of fluid flows and its mutual influence on the flowing part of the centrifugal pump [1, 2, 4, 5, 6].

Let us consider the flow in the axial gap near a smooth disk rotating in a closed cavity. Imagine a smooth disc with a cylindrical surface of radius R_H, rotating in a closed chamber with an angular velocity ω in the absence of an expenditure fluids flow (figure 1). As a result of viscous forces impact, a circular movement arises in the chamber: near the disc a fluid moves from the center to the periphery, and near the wall it moves from the periphery to the center. Flow continuity is ensured by axial movement of the fluid: it moves from disk at the periphery in the boundary layer of thickness δ, and it moves to the disc at the central part.

In systems of such type, the flow characteristics (angular velocity of the fluid and disk friction loss) significantly depend on a change of the Reynolds number $\mathrm{Re} = \omega R_H^2 / \nu$, relative axial gap L/R_H, and roughness.

Experimental researches of a fully wetted disc have shown that there are four flow regimes in the gap between the chamber wall and the rotating disk (figure 2):
-   regime I - laminar fluid flow and merged boundary layers due to the small axial gap *L*. Circumferential velocity in the axial gap varies linearly;
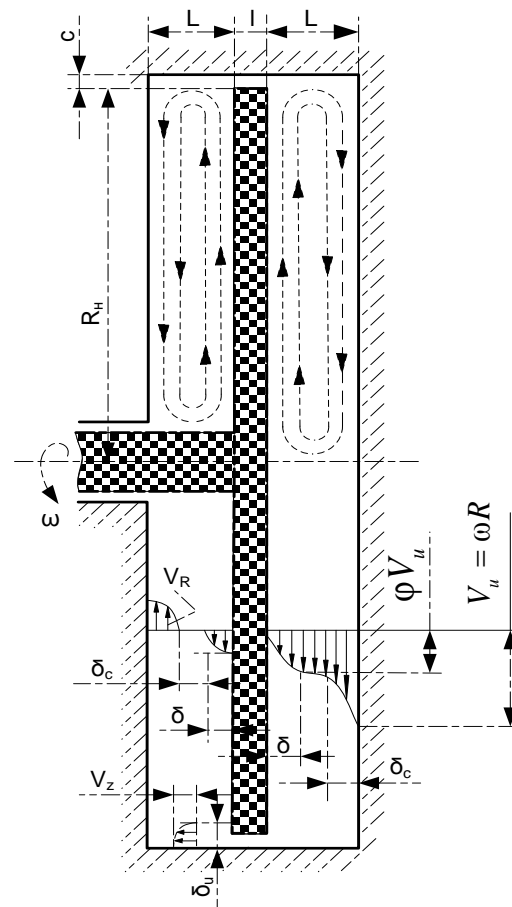
Figure 1. Disk rotating in a closed cavity.

- regime II − laminar fluid flow in the axial gap at separate boundary layers on the disc and the stationary wall of the chamber. Between the boundary layers, there is a flow core within which the circumferential velocity is constant;

- regime III − turbulent fluid flow with merged boundary layers due to the small axial gap *L*. Flow velocity of the fluid in the gap varies linearly;

- regime IV − turbulent fluid flow in the axial gap at separate boundary layers. Between the boundary layers, there is a flow core within which the circumferential component of the absolute velocity V_u is constant.

Each flow regime characterized by its own relations that allow us to estimate the velocity of fluid flow in the gap and the moment resistance of the rotating disk.

For the Reynolds number $Re > 3 \cdot 10^5$ the flow stops being laminar. And turbulent behavior of the flow is characterized at the pump elements in the real range of the viscosity of the working fluid for any value of the axial gap. Moreover, considering the large axial gaps in the high-speed centrifugal pump, i.e. $L \gg (\delta_c + \delta)$, we come to the conclusion that the analysis is required only for the regime IV of the fluids flow.
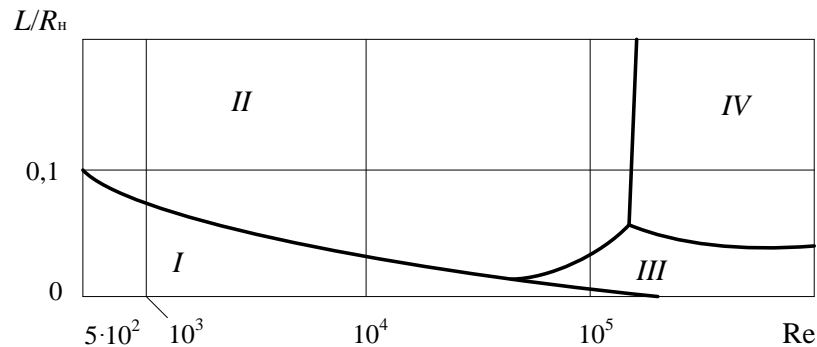
Figure 2. The area of existence of different flow regimes in the gap between the chamber wall and the rotating disc.

Such flow near a smooth disc has been studied in [7]. In this paper experimental data are presented, which show that the main part of fluid in the axial gap, called a flow core, rotates in the circumferential direction according to the law of solid body with constant angular velocity ω. The transition from the circumferential velocity of the fluid in the core to the speed at the disk and to the zero velocity at the wall occurs in a thin boundary layer between the core and the walls.

By the condition of equality of the moments on the wall and on the smooth disc, the value of the coefficient of the resistance moment for the smooth disk can be found by the formula [7]:

$$C_{\text{м}} = \frac{0.2335 \left[(R_H + C)/R_H \right]^{23/5} \varphi^{9/5}}{4 \, \text{Re}^{1/5}} .$$

Where the coefficient φ is determined for small values $n = 100 \dfrac{R_H + C}{R_H} - 1$ from the expression $1/\varphi = 2.06 \, (1 + 0.014 \, 3n + 0.000 \, 141 \, n^2 + \ldots)$.

For the axial gap limited by the ratio $L/R_H \le 0.44$, what is typical for the construction of a centrifugal pump unit, with sufficient accuracy we can use the empirical formula for determining the coefficient of the resistance moment for the smooth side of the disk [7]:

$$C_{\text{м}} = \frac{0.151}{(L/R_H)^2 \, \text{Re}^{1.2}} + \frac{1.02 + L/R_H}{12(6 + L/R_H) \, \text{Re}^{0.182}} . \tag{1}$$

In a pump unit the flow of fluid is forced and its flow rate is usually determined by the leak of a centrifugal pump or by conditions of heat removal from the seal cavities.

The investigation of the pressure distribution in the axial gap between the disk and the chamber wall in the presence of a radial flow was first held by A. A. Lomakin. The analysis of the results of measurement in the flow between the rotating disc and the chamber wall, carried out in [7], only generally confirmed the results of A. A. Lomakin.

The results of experiments on different working fluids [8] in a wide range of modes in operating of smooth discs allowed us to obtain the dependence to determine furling coefficient for the liquid flow at expendables fluid flows from the periphery to the center:

$$\varphi = 0.578 - 11 V_R / (\omega R_H) - 0.0658 \overline{L}, \qquad (2)$$

where $\overline{L} = L / R$.

For research of influence of expendables flows along the smooth side of the impeller disc on the pressure distribution over its radius, a series of experiments was held with smooth disks, which have the same geometric parameters with the tested impellers [9]. Experiments have shown that the experimental points obtained in testing a smooth disk with all the working fluids at the expendables flows in the axial gap from the center to the periphery for this range of parameters are well described by relation (2).

For the flow from the periphery to the center the expression is obtained

$$\varphi = 0.3865 + 95.3 V_R / \omega R. \qquad (3)$$

For real values of clearance between the impeller and the chamber wall $L = 0.5 \ldots 4.0$ mm in the range of expendable parameter $V_R/\omega R = 0.005 \ldots 0.015$.

Under presence the flow rate, changes of the circumferential velocity in the gap influence on change in tension of friction and on the moment of resistance of the rotating disk [7]. This can be expressed by increasing the friction moment on the elementary circular area through width ΔR:

$$\Delta M = 2\pi R^2 \Delta R \rho C_M (\omega\text{-}RV_u)^2/2. \qquad (4)$$

The comparison of the friction moments calculated by this equation for the conditions $V_u = \omega R/2$ (no flow rate) and $V_u \to 0$ (high flow rate) shows that for the second condition the friction moment on the disk is increased fourfold. On the basis of the experimental data the empirical formula was obtained for the friction moment taking into account an expendable flow:

$$M = \left(\frac{0.0187}{\sqrt[5]{\text{Re}}} + \frac{0.08}{\sqrt{\omega R_H / V_{Rcc}}} \frac{L}{R_H} \right) \rho \, \omega^2 R_H^5 . \qquad (5)$$

The difference of the expressions (5) and (1) is that the second term in the bracket of equation (5) is the friction coefficient, due to the flow rate

$$C_{M\dot{V}} = \frac{0.08 L}{R_H} \sqrt{\frac{V_{Rcc}}{\omega R_H}} . \qquad (6)$$

Accurate expression (6) for the expendable flow of water and air takes the form

$$C_{M\dot{V}} = 0.42 \cdot 10^3 \frac{(L R_H)^{0.75} Ga^{0.3}}{(\omega R_H / \nu_{Rcc})^{0.8}} , \qquad (7)$$

where *Ga* is the number of Galilee;

$$\omega R_H / V_{Rcc} = 2\pi R_H^2 L \omega \frac{1}{V} .$$ (8)

In testing bladed disks of impellers for the modes, which are the same as the test modes for smooth disks, the value φ obtained that is larger than the value determined by the formula (2).

Let us represent the dependence φ = *f* (*V*) calculated according to the formula (2) and the experimental points obtained in a test of a smooth disk (figure 3). It should be noted that a smooth disk is characterized by satisfactory convergence of the calculated dependences and the experimental points. However, for a pump impeller there is a significant difference between these data. This indicates that there is essential influence of vortex impulses on the hydrodynamics of the flow along the smooth side of the impeller. These impulses come from the blade side. At the same time the value φ for smooth side of the impeller depends on the flow rate through the cavity of the pump unit.
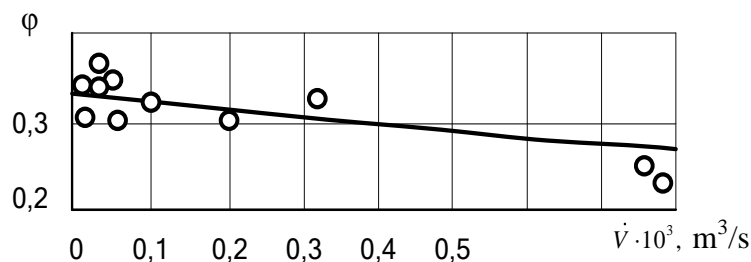
Figure 3. A graphical representation of φ on the flow rate in the axial gap
for a smooth disk with $D_H = 80$ mm [4].

For estimation of the vortex impulse exchange in the cavity of the pump unit, in the paper [6] a series of experiments is shown for smooth disks with different diameters (from 46 to 80 mm) with open impellers and hydrodynamic seals with a visor. Experiments were performed at the same axial and radial clearances with flow rate $\dot{V} = 0.05 \cdot 10^{-3}$ m³/s, which was caused by the necessity of heat transfer from the cavity of the pump unit.

Analysis of the obtained expressions and the calculation of φ for the different disks showed that these values for impellers with a visor and open impellers are different and at the same time similar quantities are more for smooth disks. Taking into account existing dependencies for smooth discs at zero and expendable flows, the value φ_s for impellers can be determined by the expression

$$\varphi_s = k\varphi ,$$ (9)

where *k* is a coefficient determined empirically, it takes into account additional furling liquid in the smooth side of the impeller. As a result of processing of data obtained during the testing impellers and smooth disks with different sizes and different liquids at ω = 400 ... 2500 rad / s

and flow rate $V = (0.05 \ldots 1) \cdot 10^{-3}$ m^3 / s, values $k = 2 \pm 0.1$ were obtained for an open impeller and values $k = 1.2 \pm 0.06$ were obtained for a hydrodynamic seal with visor.

Thereby, the results of researches conducted for the basic hydrodynamic parameters of the rotation cavities of pumps units allowed us to develop and experimentally prove the basic calculation expressions for calculating the flow along the smooth side of the disk impeller. The obtained analytical expressions describe the accumulated experimental data for the rational design of the flow part for cavities of rotation and can be used in the development of design solutions to provide the required level of energy excellence. All this contributes to efficiency and reliability of modern high-speed pumps units.

References

[1] Kraeva E M and Kraev V M 2009 Hydrodynamic analysis and visualization of spatial flow in centrifugal force field *Preprint* 2009.2 (Krasnoyarsk: Inst. of Computational Modeling SD RAS) p 34

[2] Baybikov A S and Karahatyan V K 1982 *Hydrodynamics auxiliary paths bladed machines* (Moscow: Mashinostroyeniye) p 112

[3] Kraev M V, Ovchinnikov B V and Shapiro A S 1976 *Hydrodynamic seal high-speed shafts* (Moscow: Mashinostroyeniye) p 104

[4] Kraev M V and Taldykin Y A 1999 *Hydrodynamics cavities of rotation in rotating machines* (Krasnoyarsk: SAA Press) p 134

[5] Idelchik I E 1975 *Handbook of hydraulic resistance* (Moscow: Mashinostroyeniye) p 559

[6] Kraev M V 1983 *Theory and calculation of hydraulic pumps paths* (Krasnoyarsk: Krasnoyarsk Polytechnic Institute Press) p 100

[7] Dorfman L A 1960 *Pressure drop and heat transfer of rotating bodies* (Moscow: Fizmatgiz) p 260

[8] Kraeva E M 2009 On calculation of operating parameters of a low specific speed centrifugal pumps *Vestnik SibSAU* **2(23)** 168-170

[9] Kraeva E M 2011 *High-speed centrifugal pumps* (Krasnoyarsk: SibSAU Press) p 212