

STUDY OF THE SEALS THROTTLING GAPS
PARAMETERS INFLUENCE ON THE VIBRATION
CHARACTERISTICS OF ROTARY MACHINES

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Abstract

For modern high-speed high-pressure power pumps, the issues of rotor dynamics are particularly relevant, in particular for modern aviation power plants, high-speed high-pressure fuel pumps, and other industrial facilities. Non-contact seals, in addition to their main purpose – limiting the flow between cavities with different pressures, can be used as dynamic rotor supports. The nature of radial forces in the throttling channels of non-contact seals is considered and the influence of the shape of the annular channel on the forces acting on the rotor is determined. The amplitude-frequency characteristics of the rotor in slot seals are obtained. The influence of the taper of the gaps on the vibrations of the rotor in the seals is determined. The results of experimental studies of the frequency characteristics show that the calculation errors do not exceed 5%. The studies allow us to determine the directions for increasing the vibration reliability of power pumping equipment by purposefully increasing the rigidity of the slot seals.

Key words: gap seals, taper, gap forces, frequency characteristics, seal rigidity

Introduction. Multistage high-pressure centrifugal pumps, widely used in power engineering, are characterized by a steady tendency to increase operating

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parameters: feed rates, pressures and rotor speeds, that is, to concentrate ever higher capacities in individual units [1]. The pressure developed by centrifugal machines is proportional to the square of the rotor speed, so increasing the speed is the most rational way to achieve high pressures [2]. As a result, high-pressure centrifugal machines are usually high-speed. And for such machines, the problems of rotor dynamics are especially relevant [3]. The rotor of a modern centrifugal machine, rotating in seals, together with the automatic unloading system of axial forces, represents a special closed hydromechanical system [4]. The peculiarity of the system is due to the fact that hydrodynamic forces and moments of different nature act on the rotor from the side of the seals [5]. These forces affect the rotor oscillations in different ways and at the same time themselves depend on the nature of these oscillations [6]. The conducted studies [7] showed that the influence of the sealed medium is especially significant in the presence of large velocity and pressure gradients. Such conditions are typical for small gaps of non-contact seals, where large pressure differences are throttled, and one of the walls belongs to the rotating and vibrating rotor [8]. It is practically impossible to ensure complete tightness of modern equipment, even using complex sealing systems, especially in machines with high parameters [9]. A lot of energy is spent on limiting the leakage of the pumped medium along the rotating shaft of the pump, which can be used to improve the dynamic characteristics of the equipment [10]. Thermal and force deformations of seals in rotary machines significantly affect the tightness and stability of toxic gas transportation. Their changes can disrupt the flow and change the sorption conditions. Therefore, algorithms for predicting the intensity of sorption must take into account the thermomechanical state of the seals. Combining such models with deformation analysis increases the accuracy of leak risk assessment and contributes to more reliable design of safety systems [11, 12]. In this case, the leakage energy can not only provide the necessary bearing capacity of the supports, but also, most importantly, reduce rotor vibration to an acceptable level even in the presence of significant imbalance [13]. This must be taken into account when designing critical power equipment.

Material and methods. Let us consider the nature of the occurrence of radial forces in the throttling channels of contactless seals. These forces are perpendicular to the direction of the flow that generates them. The effects of the annular channel taper and the skew of the axes of the shaft and the housing of the unit are of particular interest. The effect of the annular channel shape on the force characteristics is shown in Fig. 1.

In an annular channel with parallel walls (Fig. 1, *a*) the diagrams in the upper and lower halves are the same, their areas are equal $A_1 = A_2$.

For confusor channels (Fig. 1, *b*) in the upper section the ratio of the gap at the inlet to the gap at the outlet differs little from unity in the lower half $h_1/h_2 \gg 1$, therefore the convexity of the pressure curve is greater than in the upper half and the areas of the diagrams are different: $A_1 > A_2$. In other sections

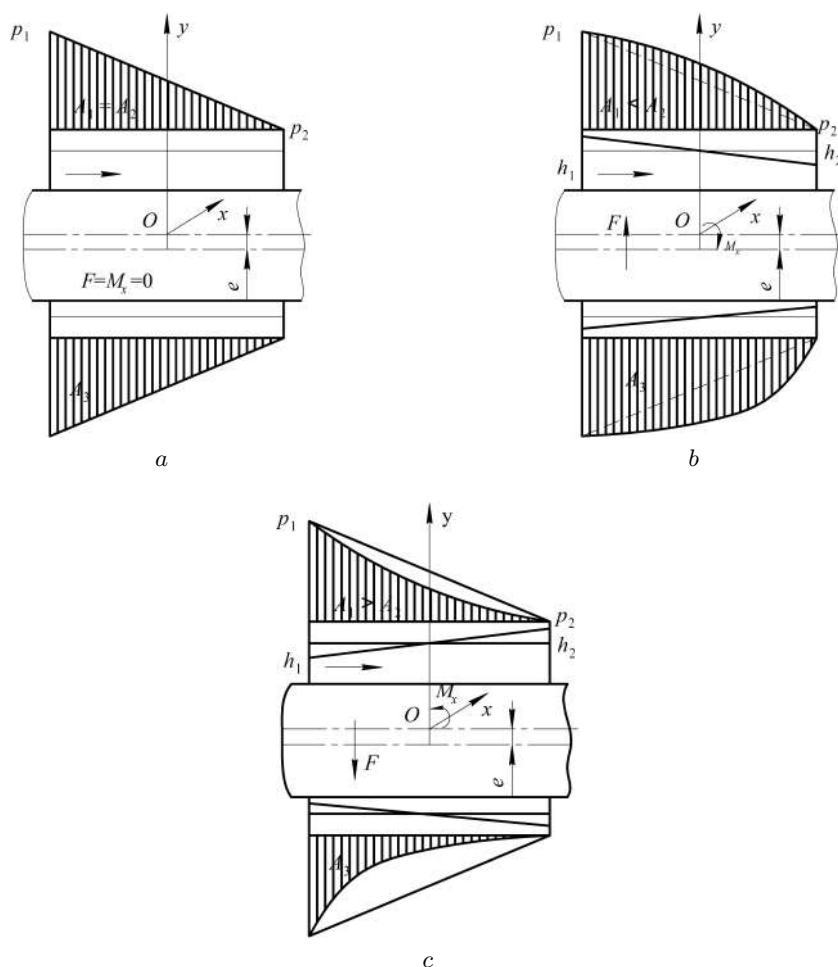


Fig. 1. Influence of the annular channel shape on the power characteristics
a – smooth slot, b – confusor channel, c – diffuser channel

the difference decreases, and in the section perpendicular to the line of centres it disappears. In this case, the axial symmetry of the pressure is disturbed. Only the symmetry relative to the line and centres (axis Oy) is preserved. As a result, a radial centering force arises, which tends to reduce the eccentricity, and a positive moment relative to the axis Ox .

In diffuser channels (Fig. 1, c) the situation is the other way around: $A_2 < A_1$. The radial force and the eccentricity vector are directed the same way, tending to increase the eccentricity. In contrast to the confusor, the force in the diffuser channel is decentering. It increases the shaft deflection, thereby reducing its bending rigidity and critical rotation frequency.

Let us introduce the concept of the channel taper parameter

$$\theta_0 = \vartheta_0 l / 2H, \quad |\theta_0| \leq 1,$$

where ϑ_0 is the taper angle of the annular channel of the gap seal, l is the length of the annular gap of the gap seal, and H is the average radial gap in the gap seal.

The effect of the misalignment of the shaft and housing axes is evident from Fig. 2, which shows a special case of intersecting shaft and housing axes located in the plane of the section.

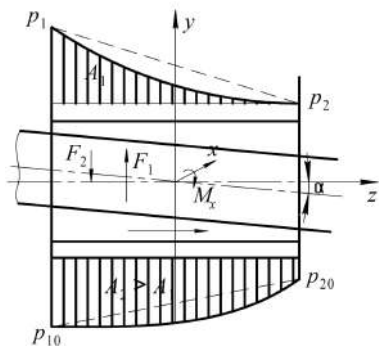


Fig. 2. Effect of shaft misalignment on pressure change

The upper half of the channel (above the Ox axis) is a diffuser, the lower is a confuser. In the upper part of the section, the pressure diagram is concave, and in the lower part, it is convex. As a result, $A_2 > A_1$, i.e., a positive radial force is created, directed along the centre line against the eccentricity, and a positive moment of this force relative to the Ox axis. Such a moment, if the shaft is insufficiently rigid, can cause a loss of stability of the equilibrium state of the shaft: the slightest deviation from the equilibrium position is enough for the shaft to be pressed against the housing, i.e., the divergence phenomenon occurs.

Expressions for hydrodynamic forces and moments arising in slot seals with a cone and acting on the rotor of a centrifugal machine are given in the works of SHEVCHENKO [14] and YU et al.[15].

Results and discussions. From the initial equations previously obtained by the authors, describing the rotor oscillations under the action of gyroscopic forces and moments arising in the seals [16], the amplitudes and phases of oscillations were obtained, expressed through external disturbances [17, 18].

$$u = \bar{\omega}^2 \sqrt{\frac{(AU_{22} - \Gamma U_{12})^2 + (AV_{22} - \Gamma V_{12})^2}{U_0^2 + V_0^2}},$$

$$\theta_a = \bar{\omega}^2 \sqrt{\frac{(\Gamma U_{11} - AU_{21})^2 + (\Gamma V_{11} - AV_{21})^2}{U_0^2 + V_0^2}},$$

$$\varphi_u = -\arctg \frac{(AU_{22} - \Gamma U_{12}) V_0 - (AV_{22} - \Gamma V_{12}) U_0}{(AU_{22} - \Gamma U_{12}) U_0 + (AV_{22} - \Gamma V_{12}) V_0},$$

$$\varphi_\vartheta = -\arctg \frac{(\Gamma U_{11} - AU_{21}) V_0 - (\Gamma V_{11} - AV_{21}) U_0}{(\Gamma U_{11} - AU_{21}) U_0 + (\Gamma V_{11} - AV_{21}) V_0}.$$

Here

$$U_{11} = (-a_1 + a'_4) \bar{\omega}^2 + 1 + \frac{a_{31}}{\Omega_{u0}^2}, \quad V_{11} = \frac{a_{20} + a_{21} - a'_5}{\Omega_{u0}} \bar{\omega},$$

$$U_{12} = \frac{-\alpha'_3 + \alpha_4}{\Omega_{u0}} \bar{\omega}, \quad V_{12} = -\alpha'_2 \bar{\omega}^2 - \frac{\alpha_5 - \alpha_0}{\Omega_{u0}^2},$$

$$U_{21} = \frac{-\beta'_3 + \beta_4}{\Omega_{u0}} \bar{\omega}, \quad V_{21} = \beta'_2 \bar{\omega}^2 - \frac{\beta_5 + \beta_0}{\Omega_{u0}^2},$$

$$U_{22} = (-b_1 + b'_4) \bar{\omega}^2 + \frac{\Omega_{\vartheta 0}^2}{\Omega_{u0}^2} + \frac{b_{31}}{\Omega_{u0}^2}, \quad V_{22} = \frac{b_{20} + b_{21} - b'_5}{\Omega_{u0}} \bar{\omega}.$$

Expressions for calculating the coefficients of hydrodynamic forces and moments in the gaps a_{ij} , b_{ij} , α_i , β_i are given in [15]; Ω_{u0} , $\Omega_{\vartheta 0}$ are the initial radial and angular rigidity of the shaft.

Figure 3 shows the calculated dependences of the frequency characteristics of the rotor in gap seals, constructed for three values of the conicity parameter (1 – diffuser, 2 – smooth gap, and 3 – confuser). A positive phase characteristic corresponds to a total negative resistance.

As can be seen from the obtained dependencies for the diffuser channel, the total resistance is negative. For cylindrical and confuser channels, the stability condition is satisfied at all speeds.

The given examples show that the geometry of the seal has a significant effect on the amplitude characteristics. For diffuser seals, the critical frequency is significantly lower and the oscillation amplitude is significantly higher than for

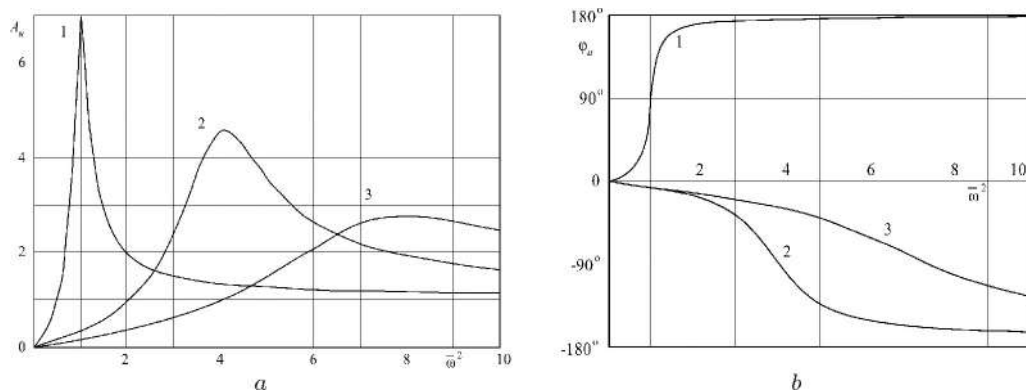


Fig. 3. Frequency characteristics: a – amplitude, b – phase, 1 – $\theta_0 = -0.3$; 2 – $\theta_0 = 0$; 3 – $\theta_0 = 0.3$

confusor seals. The confusor shape of the seal suppresses the oscillation amplitude and increases the critical frequency.

Subsequent studies have shown that in seals with a diffuser gap, the rotor is unstable even without rotation. The critical value of the diffuser channel taper parameter is 0.27. With a greater taper of the gaps, feeding liquid under pressure into the sealed cavity even without shaft rotation, we obtain an excellent vibration exciter.

The confusor shape of the gap ensures rotor stability at all studied speeds and increases the critical frequency of the rotor by 4–5 times compared to the rotor in smooth gaps.

This is a fairly clear example of how simple design differences can affect the vibration state of the rotor of a centrifugal machine.

A comparison of the results of frequency characteristic calculations using the obtained expressions with the data of experimental studies presented in [19] shows that the calculation errors do not exceed 5%, which indicates the possibility of using the obtained formulas in engineering calculations.

Thus, the analysis of dynamic characteristics showed that with a targeted selection of geometric parameters of non-contact seals, it is possible to change the vibration characteristics of the rotor of a centrifugal machine.

An example of the use of gap seals in the design of a high-pressure energy pump. NPP emergency pumps (Fig. 4) are designed to supply condensate from spare tanks to steam generators [20].

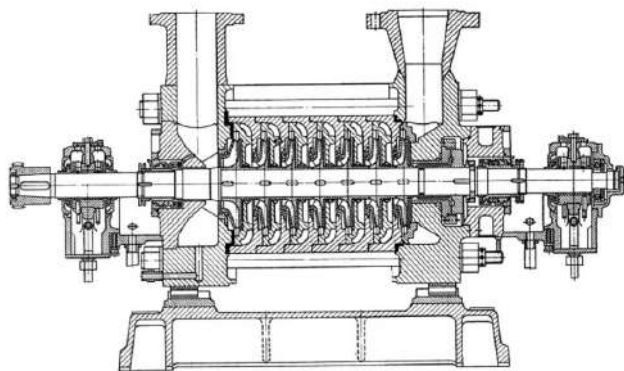


Fig. 4. Emergency pump

The pump is sectional, seven-stage. The first stage wheel has a high suction capacity. End seals – mechanical seals with body cooling. The axial force on the rotor is balanced by the hydraulic heel, and the rotor shift is limited by a ball bearing. Plain bearings are lubricated by rings placed in oil baths.

By the example of the considered pump design, it is clearly seen that the slotted seals are located quite tightly along the entire length of the rotor. Therefore, they significantly affect the vibration state of the rotor and the pump as a

whole. Moreover, each stage of the pump develops a pressure of 4 to 6 MPa, and a pressure of more than 20 MPa is throttled on the hydraulic heel.

Conclusions. The differences in the patterns of rotor oscillations in gap seals from its oscillations in air are caused by the action of hydrodynamic forces arising in gap seals.

The increase in natural frequencies depends on the taper of annular throttling gap. The pressure differential of 1.5 MPa across the converging seals provides almost threefold increase in natural frequency of the rotor. This confirms the possibility and effectiveness of simultaneous use of groove seals as hydrostatic bearings.

The hydrodynamic force depends on the channel taper and can change sign in the diffuser channel turning into a force of negative resistance, i.e., is a destabilizing factor. Unlike the rotating force due to proper rotation of the rotor, the dissipative force does not depend on rotary speed and can become negative in the absence of rotation. Thus, in seals with diffuser channels, even a non-rotating rotor can go beyond oscillatory stability.

Force factors of the gap seals are determined by the geometric (gap, radius, length, taper, shape of the input edges) and operational (pressure differential, operating speed range, physical properties of the pumped medium) parameters. A purposeful choice of these parameters can influence the vibratory condition of the rotor and the machine as a whole.

The studies make it possible to determine the directions of increasing the vibration reliability of energy pumping equipment by purposefully increasing the rigidity of gap seals.

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