Determination of angular stiffness coefficient of the annular seal by experiment-calculation

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Abstract. Hydrodynamic forces in annular seals of centrifugal pumps create a significant effect on the vibrational activity of the rotor as a whole. Theoretical and experimental studies of various authors made it possible to establish the structure of hydrodynamic forces and determine the magnitude of the coefficients of radial forces. The authors of this work obtained quantitative load characteristics of the rotor in an annular seal at specially created laboratory experimental stands at various pressure drops and significant angles of misalignment of the rotor axis relative to the seal axis. Static experiments were also conducted with annular seals of various lengths with a constant differential pressure of the liquid and a fixed angle of misalignment of the rotor axis relative to the axis of the annular seal. The given misalignment angle was obtained as a result of the action of various external forces on the rotor. In addition to static experiments, dynamic rotor tests were carried out. Experimentally obtained amplitude-frequency characteristics of the forced radial-angular oscillations of the rotor, which is dynamically mounted in two symmetrical annular seals. From the experimental characteristics, the critical frequency of the rotor angular oscillations in the seals was determined. According to theoretical formulas, taking into account the value of the coefficient of angular stiffness of the annular seal, the frequency of the natural angular oscillations of the rotor in the annular seals was calculated. A comparison of theoretical and experimental values indicates their good quantitative agreement.

1. Introduction

Centrifugal pumps are widely used in various sectors of the economy, from nuclear energy to agricultural production and utilities. According to statistics, the hydromechanical system “rotor-annular seal” accounts for over 70% of all centrifugal pump crashes (rotor breakdown, grazing on stator bushings of annular seals, damage to seal bushings, wear of the latter, etc.) [1, 2]. Hydrodynamic forces in the clearances of non-contact seals can cause loss of dynamic stability and destructive for rotor self-oscillation pumps. And these same forces can stabilize the rotor, significantly reducing its vibrational activity. It is the hydromechanical system “rotor-annular seal” that has a decisive influence on the performance of the entire centrifugal pump [3, 4, 5].

2. Analysis of the main achievements and publications

Analysis of recent achievements and publications. The fundamental reflection of the methodology for calculating the hydrodynamic parameters of non-contact seals was received in the works of V.A. Martsinkovsky [3, 4, 6, 7]. Analytical linearized expressions of the coefficients of radial forces in an annular seal were obtained for a laminar and turbulent flow of a viscous incompressible fluid in a short
annular clearance \((l < r\), i.e., the length of the clearance is less than its radius) taking into account the taper of the clearance and the misalignment of the axes of the rotor and stator bushings of the seal. Due to the skewness of the axes, the total eccentricity of the rotor changes, which is taken into account by additives to radial forces through correction factors. It was also pointed out that the action of the circulating force can lead to a loss of dynamic stability by the rotor, accompanied by self-oscillations with a significant amplitude.

The theoretical and experimental results of studies of radial forces in the case of turbulent fluid flow in short annular seals are reflected in the works of A.N. Guliy [8], I.N. Beda [9]. The authors of these works indicate a high agreement between the experimental data and the calculations based on the given analytical expressions for the radial force coefficients. The expressions obtained by these scientists for the coefficients of radial hydrodynamic forces in annular seals give results very close to calculations based on theoretical dependences, which were obtained by one of the authors of this study in [10]. Radial hydrodynamic forces have received attention in a number of works by authors [11-16]. In this case, the direct influence of the skew of the stator and rotor walls of non-contact seals on the magnitude of hydrodynamic forces has not been adequately considered.

3. **Formulation of the goals**

Rotors of centrifugal pumps perform radial-angular vibrations in annular seals. The natural frequencies of these vibrations are determined by the radial and angular hydrostatic stiffness of the contactless seals. In the works of various authors presented in section 2, there are reliable data on the coefficient of radial hydrostatic stiffness, but there are no data on the angular hydrostatic stiffness of annular seals. In this work, the task is set to obtain by experiment-calculation quantitative values of the coefficient of angular stiffness of the annular seal for different values of the pressure drop across the seal. Also get the dependence of the coefficient of angular stiffness on the length of the seal. Compare the calculation results from the theoretically obtained expressions for the coefficient of angular stiffness with the results of measurements at laboratory facilities in the static and dynamic versions of the experiment.

4. **Calculated and experimental studies**

Taking into account the general approach of V. Martsinkovsky, the flow of the condensed fluid in non-contact annular seals of centrifugal pumps can be considered as an unsteady turbulent flow in the self-similar region of motion of a viscous incompressible fluid in an annular channel (Fig. 1):
To describe the axial and circumferential motion of a viscous incompressible fluid in the annular channel of an annular seal, we use the Reynolds equations:

\[
\begin{align*}
\frac{\partial P}{\partial z} + \frac{\rho}{h} \frac{d\omega}{dt} dy &= - \frac{K_\omega}{h^2} W, \\
\frac{\partial P}{\partial x} + \frac{\rho}{h} \frac{dU}{dt} dy &= - \frac{K_u}{h^2} U,
\end{align*}
\]

where is the fluid pressure in the annular seal; \(P\) - local radial clearance; \(\mu\) – the dynamic viscosity of the fluid; \(\rho\) – the fluid density; \(W\) – the average axial velocity of the fluid in the annular seal; \(U\) – the average velocity of the circumferential fluid flow in the annular seal; \(x, y, z\) – the circumferential, radial and axial coordinates in the system \((x, y, z)\) are mutually perpendicular fixed axes in the geometric center \(O\) of the annular seal sleeve, see Fig. 1. The coefficients \(K_\omega\) and \(K_u\) enter into (1) as turbulent functions of the Reynolds number.

Equations (1) are solved together with the continuity equation:

\[
\frac{\partial U}{\partial x_i} + \frac{\partial V}{\partial y_i} + \frac{\partial W}{\partial z_i} = 0.
\]

The boundary conditions for solving equations (1) and (2) are: the pressure at the inlet to the annular seal \(P_1\) and the pressure at the outlet \(P_2\) as well as the condition for the absence of sliding of liquid molecules both on the outer \(U=V=W=0\) and inner \(U=U_t, V=V_t, W=0\), walls of the annular channel Fig. 1.

By jointly solving equations (1) and (2) using the basic principles of the V.A. Martsinkovsky method, one of the authors of this study obtained expressions for the projections of the hydrodynamic force on the axis of the fixed coordinate system \((x, y, z)\) Fig. 1, as eccentricity of the rotor and its angular displacements. The expressions for the projections of forces are presented in a compact matrix form [10]:

\[
\begin{pmatrix}
F_x \\
F_y \\
F_z
\end{pmatrix} =
\begin{pmatrix}
K_x & oq & x \\
oq & K_y & y \\
oq & 0 & m_z
\end{pmatrix}
\begin{pmatrix}
x \\
y \\
m_z
\end{pmatrix} +
\begin{pmatrix}
b & oq & \dot{x} \\
oq & b & \dot{y} \\
oq & 0 & \dot{m}_z
\end{pmatrix}
\begin{pmatrix}
x \\
y \\
m_z
\end{pmatrix} +
\begin{pmatrix}
\bar{K}_x & oq & \dot{x} \\
oq & \bar{K}_y & \dot{y} \\
oq & 0 & \dot{m}_z
\end{pmatrix}
\begin{pmatrix}
x \\
y \\
m_z
\end{pmatrix},
\]

where the radial and angular (with a dash on top) force factors:

\[
K_x = (2h_b)^{-1} \pi rl \Delta P(\alpha_1 + \alpha_2 - \alpha_0^2 + \alpha_0^2),
\]

\[
\bar{K}_x = (4h_b)^{-1} \pi rl \Delta P\left[1 + \alpha_1 - \alpha_2 - 2(\alpha_0^2 + \alpha_0^2) - 4\alpha_1 \alpha_2 + (\alpha_1 + \alpha_2)4K(\xi_0)^{-1}\right],
\]

\[
b, m_x, \bar{b}, m_z, q = m_y, \bar{q} = m_y, q = 0.5b, \bar{q} = 0.5b - \text{the coefficients are given in work [10]},
\]

\[
\xi_0 = \xi_1 - \xi_2 + \lambda(2h_b) / 2, \quad \alpha_1 = \xi_1 / \xi_0, \quad \alpha_2 = \xi_2 / \xi_0.
\]

The coefficients \(K_x, b\) and \(q\) characterize the elastic, damping and circulation force in the seal. The coefficient \(K\) in the expressions of forces allows one to take into account \((K=1)\) or not to take into account \((K=0)\) the inertial effects of the liquid in the annular seal.

When the rotor is skewed and angularly vibrated in annular seal, angular components of hydrodynamic force arise, which are characterized by the coefficients: angular stiffness \(\bar{K}_x\), angular damping \(\bar{b}\), angular circulation force \(\bar{q}\). Their origin is associated with the deformation of the pressure diagram in the seal due to the angular displacements of the shaft in the annular seal. Such a separation of the hydrodynamic force into angular and eccentricity-dependent components is possible.
because they are functions of independent parameters - eccentricity and skew angles of the rotor in the seal.

Since the pressure plots along the axis of the annular seal for all components of the hydrodynamic forces vary either linearly or according to parabolic laws \([6, 8]\), the transfer of the resultants to the mid-section of the annular seal generates moments of forces with respect to the x and y axes passing through the center of this section. In the work of one of the authors [17], expressions were obtained for the moments of elastic, damping and circulation forces generated by both the eccentricity and the skewness of the rotor axis in the annular seal. Moments relative to the x and y axes are presented in a compact matrix form:

\[
\begin{bmatrix}
M_x \\
M_y
\end{bmatrix} = \begin{bmatrix}
\frac{\omega}{2} \alpha_x & \alpha_y \\
\alpha_x & -\frac{\omega}{2} \alpha_y
\end{bmatrix}
\begin{bmatrix}
x \\
y
\end{bmatrix} + \begin{bmatrix}
0 & \alpha_v \\
\alpha_v & 0
\end{bmatrix}
\begin{bmatrix}
x \\
y
\end{bmatrix} + \begin{bmatrix}
\beta_v & \frac{\omega}{2} \beta_{vy} \\
\frac{\omega}{2} \beta_{vx} & \beta_v
\end{bmatrix}
\begin{bmatrix}
v_x \\
v_y
\end{bmatrix} + \begin{bmatrix}
-\beta_{vy} & 0 \\
0 & -\beta_{vx}
\end{bmatrix}
\begin{bmatrix}
v_x \\
v_y
\end{bmatrix}
\]

where \(\alpha_x, \alpha_y, \beta_v, \beta_{vy}\) are the coefficients and are given in [11].

Of the obtained analytical expressions (3) and (4), the coefficient of angular stiffness \(K_x\) is of particular interest. It is the angular stiffness of the annular seal that sets the natural frequency of the angular oscillations of the rotor in the seals.

For the practical determination of the coefficient of angular stiffness of an annular seal in laboratory conditions, an experimental stand was created. The basis of the experimental bench was a test head, which made it possible to carry out both static and dynamic tests of the rotor in annular seals.

An attempt to directly measure the angular stiffness of the seal, made using the structural scheme shown in Fig. 2, confirmed only the qualitative side of the effect of the emergence of force factors when the shaft is skewed in the annular seal.

**Figure 2.** The scheme of the shaft on a spherical support. 1 - shaft - sleeve, 2 - spherical support, 3 - hydraulic cylinder, 4 - displacement sensor.

**Figure 3.** Hydraulic head diagram for static tests of a annular seal. 1 - annular seal, 2 - shaft, 3 - hydraulic housing, 4 - thrust bearing shaft, 5 - annular shaft support, 6 - bypass holes, 7 - block, 8 - cable, 9 - load, 10 - displacement sensor.

When applying the pressure of the working medium to the annular seal between the wall of the hydraulic cylinder 3 and the shaft-sleeve 1, the latter spontaneously turned on the spherical joint 2, which was recorded by the displacement sensor 4, and the skew was limited only by the walls of the annular seal of the hydraulic cylinder 3. This experiment confirmed the theoretically obtained...
conclusion about the fact that the hydrostatic angular momentum $M_{x|v} = \beta \cdot v_s \cdot [v_y]$ increases the skew angle of the shaft axis in the annular seal.

Further static tests of the shaft in the annular seal were carried out with the option of the hydraulic head fig. 3.

On an experimental setup with a hydraulic head option, fig. 3, indirect measurements of the coefficient of angular stiffness of the annular seal were made in accordance with the power scheme of the shaft loading (Fig. 4).

\[ -F \cdot L + F_v \cdot l_v + F_e \cdot l_e = 0, \]  
\hspace{1cm} (5)

where $F$ is the force of external loading;
$F_v$ – the resultant radial stiffness force of the annular seal;
$F_e$ – the resultant force of the angular stiffness of the annular seal;
\( l_v \) – shoulder radial strength $F_v$ relative to point O;
\( l_e \) – shoulder angular force $F_e$ relative to point O.

The geometric dimensions of the annular seal (Fig. 3):

**Figure 4.** Power scheme of shaft loading: 1 - cargo of mass M, 2 - shaft, 3 - displacement sensor.
seal radius: \( r = 30 \text{ mm} = 3 \times 10^{-2} \text{ m} \);
sealing length: \( l = 20 \text{ mm} = 2 \times 10^{-2} \text{ m} \);
radial clearance: \( h_0 = 0.2 \text{ mm} = 2 \times 10^{-4} \text{ m} \).  

Hydraulic coefficients from formulas (3) according to [8] have the following values:

\[
O_O = r + \frac{1}{2} l, \quad l_e = r + \frac{1}{3} l, \quad l_v = r + \frac{5}{12} l.
\]

Shoulder L of the external loading force \( F \) (Fig. 4):

\[
L = r + l + 15 \text{ mm} = 6.5 \times 10^{-2} \text{ m}.
\]

The theoretical calculations of [6] allow us to accept the points of application of forces \( F_e \) and \( F_v \) at such distances from the input edge of the annular seal with a length \( l \) (Fig. 4):

\[
F_e \text{ in the distance } \frac{1}{3} l, \quad F_v \text{ in the distance } \frac{5}{12} l.
\]

Taking into account formula (9), Fig. 3 and Fig. 4 we have expressions for \( O_O, l_e, l_v \):

\[
O_O = r + \frac{1}{2} l, \quad l_e = r + \frac{1}{3} l, \quad l_v = r + \frac{5}{12} l.
\]

Taking into account expressions (6), we obtain numerical values for \( O_O, l_e, l_v \):

\[
O_O = 4 \times 10^{-2} \text{ m}, \quad l_e = 3.7 \times 10^{-2} \text{ m}, \quad l_v = 3.8 \times 10^{-2} \text{ m}.
\]

With a small rotation around the point O, the upper end of the shaft moves to \( \Delta h \) with the shoulder of rotation \( L \), and the center of the annular seal \( O \), with the shoulder of rotation of \( O_O \) moves by the value of the radial eccentricity of \( e \). Moreover, we have an expression for \( e \):

\[
e = \frac{\Delta h}{L} \cdot O_O.
\]

The rotation angle of the shaft axis is determined by the expression:

\[
\nu = \frac{\Delta h}{L}.
\]

The magnitudes of the forces \( F_e \) and \( F_v \) depend on the eccentricity and angle of rotation of the shaft axis, as well as on the radial and angular stiffness coefficients of the annular seal from formulas (3):

\[
F_e = K_s \cdot e, \quad F_v = \overline{K_s} \cdot \nu.
\]

From equation (5), taking into account expressions (12), (13), (14), we obtain the expression for \( \overline{K_s} \):

\[
\overline{K_s} = F \cdot L^2 \cdot (\Delta h \cdot l_e)^{-1} - K_s \cdot O_O \cdot l_e \cdot l_v^{-1}.
\]

It is convenient to present the expression for \( \overline{K_s} \) from formulas (3) taking into account geometric and hydraulic data (6), (7) in the form:

\[
K_s = 0.857 \cdot \Delta P.
\]

where \( \Delta P \) the pressure drop across the annular seal in [MPa] = \( 10^6 \) Pa.
\[ K_s = (11.1 \cdot \frac{F}{\Delta h} - 3.4 \cdot \Delta P) \cdot 10^4, \]  
(17)

where \( F \) is the force of external loading in [N];
\( \Delta h \) – displacement of the shaft end in [\( \mu \text{m} \)] = \( 10^{-6} \) m;
\( \Delta P \) – pressure drop across the annular seal in [MPa] = \( 10^6 \) Pa.

The ratio \( F/\Delta h \) is easily determined from Fig. 5 for each experimental loading line at a given pressure drop across the annular seal.

**Figure 5.** Shaft loading characteristics.

It is convenient to imagine enmity \( K_s \) for from formulas (3) taking into account formulas (6), (7) in the form:

\[ K_s = 6.06 \cdot 10^{-2} \cdot \Delta P, \]  
(18)

where is the pressure drop across the annular seal in [MPa] = \( 10^6 \) Pa.

Calculated by the formula (18) and experimentally calculated values of the coefficient of angular stiffness \( K_s \) of the annular seal by the formula (17) for various pressure drops \( \Delta P \) are summarized in table 1.

**Table 1.** Calculated by the formula and experimentally calculated values of the coefficient of angular stiffness \( K_s \) of the annular seal.

| \( \Delta P \times 10^{-6} \), Pa | \( \frac{F}{\Delta h} \), \( \mu \text{m} \) | \( K_s \times 10^{-4}, \text{N} \) calculation (17) | \( K_s \times 10^{-4}, \text{N} \) theory (18) |
|-------------------------------|--------------------------|--------------------------------|--------------------------|
| 0.1                           | 0.105                    | 0.83                          | 0.606                    |
| 0.2                           | 0.174                    | 1.25                          | 1.21                     |
| 0.3                           | 0.25                     | 1.8                           | 1.82                     |
| 0.4                           | 0.3                      | 2.0                           | 2.42                     |
| 0.5                           | 0.43                     | 3.07                          | 3.03                     |
| 0.6                           | 0.53                     | 3.84                          | 3.64                     |
| 0.7                           | 0.64                     | 4.7                           | 4.24                     |
| 0.8                           | 0.74                     | 6.2                           | 4.85                     |
| 1.0                           | 1.0                      | 7.7                           | 6.06                     |
| 1.25                          | 1.27                     | 9.85                          | 7.58                     |
According to table 1, we have the dependences of the coefficient of angular stiffness on the pressure drop across the annular seal (Fig. 6).

![Graph showing the dependence of the coefficient of angular stiffness on the pressure drop across the annular seal.](image)

**Figure 6.** The dependence of the coefficient of angular stiffness from seal pressure drop:

- + + + - experimental - calculated points according to the formula (17),
- ----------- - theoretical calculation line by the formula (18).

Type of constructions according to Fig. 6 demonstrates a satisfactory agreement between the experimental - calculated points and the theoretical line.

In order to check the effect of the length of a smooth annular seal on the value of the coefficient of angular stiffness of a seal, experimental points of force loading of the shaft 2 were obtained (Fig. 4) with a fixed value of the pressure drop across the annular seal: \( \Delta P = 0.5 \times 10^6 \text{ Pa} \). The constant misalignment of the shaft sleeve 2 in the annular seal was set by offset \( \Delta h = 100 \mu\text{m} = 10^{-4} \text{ m} \). (Fig. 3), which was ensured by the selection of the required value of the external loading force F due to the different mass \( M \) of load 1. The length \( l \) of the annular seal changed stepwise through 5 mm.

To calculate the value of the coefficient of angular stiffness \( K_s \), with the known experimental value of the force \( F \) and the length of the annular seal \( l \), we used formula (15), which included the values for \( OO_c, l_e \) and \( l_v \) from formulas (10) and \( K_f \) from formulas (3).

The obtained theoretical-calculated, experimental and experimental-calculated data are summarized in table 2.

**Table 2.** The obtained theoretical-calculated, experimental and experimental-calculated data.

| \( l \times 10^3 \) | \( OO_c \times 10^1 \) | \( l_e \times 10^3 \) | \( l_v \times 10^3 \) | \( K_f \times 10^1 \) | \( F \), N | \( \bar{K}_s \times 10^{-4} \), N calculation (17) | \( \bar{K}_f \times 10^{-4} \), N theory (18) |
|---|---|---|---|---|---|---|---|
| 10 | 35 | 33.3 | 34.2 | 2.91 | 14 | 0.74 | 0.94 |
| 15 | 37.5 | 35 | 36.3 | 3.72 | 25 | 1.63 | 1.86 |
| 20 | 40 | 36.6 | 38.3 | 4.25 | 45 | 3.3 | 3.03 |
| 25 | 42.5 | 38.3 | 40.4 | 4.63 | 66 | 5.0 | 4.6 |
| 30 | 45 | 40 | 42.5 | 4.92 | 88 | 6.8 | 6.4 |

According to table 2, we have the dependence of the coefficient of angular stiffness on the length of the annular seal with a constant pressure drop across the seal (Fig. 7).
Figure 7. The dependence of the coefficient of angular stiffness on the length of the annular seal:

+ + + + - experimental - calculated points according to the formula (15),
----------- - calculated curve by the formulas (3).

Type of constructions according to Fig. 7 shows a satisfactory agreement between the experimental - calculated points and the theoretical curve.

Dynamic tests of a self-aligning rotor in slotted seals were carried out on an experimental bench (Fig. 8).

Figure 8. Scheme of the bench for dynamic tests rotor in annular seals: 1 - rotor, 2 - electric motor, 3 - elastic flexible coupling, 4 - shaft, 5 - cardan joint, 6 - upper annular seal, 7 - lower annular seal, 8 - bushings with radial holes.

The torque from the engine 2 was transmitted to the rotor 1 by means of an elastic flexible coupling 3 and a universal joint 5, providing the rotor 1 with freedom of radial-angular movements with holding in the axial direction. The vertical layout of the rotor system made it possible to obtain a picture of the
radial - angular oscillations of the rotor 1 under the action of force factors only from the side of the annular seals 6, 7.

The geometric dimensions of the annular seals of the stand (Fig. 8):

- seal radius: \( r = 30 \text{ mm} = 3 \times 10^{-2} \text{ m} \);
- sealing length: \( l = 20 \text{ mm} = 2 \times 10^{-2} \text{ m} \);
- radial clearance: \( h_0 = 0.25 \text{ mm} = 2.5 \times 10^{-4} \text{ m} \);
- distance along the axis from the center of the rotor to the center of the seal: \( L = 3.5 \times 10^{-2} \text{ m} \).

Characteristics of rotor 1 (Fig. 8): mass \( m = 2 \text{ kg} \), axial and equatorial moments of inertia:

\[ J_o = 21.1 \times 10^{-4} \text{ kg} \cdot \text{m}^2; \quad J_e = 20.0 \times 10^{-4} \text{ kg} \cdot \text{m}^2. \]

Experimental amplitude-frequency characteristics of forced radial-angular rotor oscillations were obtained at two pressure drop values: \( \Delta P = 0.2 \times 10^6 \text{ Pa} \) and \( \Delta P = 0.3 \times 10^6 \text{ Pa} \) on annular seals (Fig. 10).

![Figure 9. Amplitude-frequency characteristics rotor in annular seals.](image)

An analysis of the experimental amplitude-frequency characteristics (Fig. 9) made it possible to determine the critical frequencies of the forced radial and angular oscillations of the rotor in annular seals.

In the works [18, 19] of one of the authors of this study, a calculation formula is given for determining the natural frequency of angular oscillations:

\[
\omega_o = \left( 2 \cdot (-\alpha_2 L - \beta_2 + K_s L^2 + \bar{K}_s) \cdot (2m_s L^2 + 2\bar{m}_s L + l)^{-1} \right)^{1/2}
\]

(19)

where \( \alpha_2, \beta_2, K_s, \bar{K}_s, m_s, \bar{m}_s \) - the expressions are given in formulas (3) and (4).

Formula (19) made it possible to calculate the natural frequency of rotor angular vibrations in annular seals of an experimental setup using theoretically obtained coefficients of radial \( K_s \) and angular stiffness.

The obtained calculated and experimental data are summarized in table 3.

| \( \Delta P \times 10^6 \text{ Pa} \) | \( K_s \times 10^4 \) | \( \bar{K}_s \times 10^2 \) | \( \alpha_2 \times 10^{-1} \) | \( \beta_2 \times 10^{-1} \) | \( m_s \times 10^{-4} \) kg | \( \bar{m}_s \times 10^{-4} \) kg | \( \omega_o \times 10^4 \text{ s}^{-1} \) | \( \omega_{exp} \times 10^4 \text{ s}^{-1} \) |
|---|---|---|---|---|---|---|---|---|
| 0.2 | 1.5 | 1.02 | 4.68 | 1.03 | 1.14 | 3.4 | 675 | 690 |
| 0.3 | 2.3 | 1.53 | 7.0 | 1.545 | 1.14 | 3.4 | 830 | 870 |
A good coincidence of the eigenfrequency and critical frequency of the forced rotor angular oscillations in annular seals indicates a high reliability of the calculation of the angular stiffness coefficient according to formulas (3) of this work.

5. Conclusions
As a result of processing the data obtained by experimentally calculating the determination of the coefficient of angular stiffness of the annular seal, a satisfactory correspondence was found between the theoretical and experimentally calculated dependences for a given parameter of the annular seal as a function of the pressure drop across the seal. A satisfactory agreement was also obtained for the theoretical and experimental - calculated dependences of the coefficient of angular stiffness of the annular seal on the length of the seal. Using the experimentally obtained amplitude-frequency characteristics of the forced radial-angular oscillations of a self-aligning rotor, the critical angular oscillation frequencies were determined for two values of the pressure drop across the seals. Good agreement between the experimental and calculated values of the critical frequencies of angular oscillations allows the use of theoretical dependences of the coefficient of angular stiffness of the annular seal in engineering calculations.

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