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GENERAL APPROACH TO THE CONSTRUCTION OF NON-CONTACT SEALS DYNAMIC CHARACTERISTICS

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Abstract. Higher parameters of centrifugal machines are constantly required, such as the pressure of the medium to be sealed and the speed of rotation of the shaft. However, as the parameters increase, it becomes more and more difficult to ensure the effectiveness of sealing. In addition, sealing systems affect the overall safety of equipment operation, especially vibration. Non-contact seals are considered as hydrostatic-dynamic bearings that can effectively dampen rotor oscillations. Models of “rotor-gap seals” system and impulse seals have been studied to assess the effect of these seal systems on the oscillatory characteristics of the rotor. Analytical dependencies are obtained for calculating the dynamic characteristics and stability limits of seals as hydromechanical systems. The proposed general technique makes it possible to purposefully select the design parameters of seals to adjust sealing systems to work in vibration-safe modes due to a targeted increase in the rigidity of non-contact seals are determined. It leads to an increase in the vibration resistance of the centrifugal machine’s rotor. Comparison of the results of calculations of frequency characteristics according to the obtained expressions with the data of experimental studies showed that the calculation errors do not exceed 5%, which allows using for practice the obtained dependencies in the calculation and design of centrifugal machines sealing systems with sufficient accuracy.

***Keywords**:* seals-bearings, impulse seal, automatic control system, dynamic characteristics, stability, experimental data.

1. INTRODUCTION

The rotor sealing system of a high-speed centrifugal machine is one of the most complex and critical units that determine the reliability of the entire unit. This is due to the harsh operating conditions of the seals, combined with high tightness requirements in all operating modes.

Non-contact seals, in addition to the function of sealing, perform no less important – they improve the vibration state of the centrifugal machine. Structural measures aimed at increasing the hydraulic resistance of seals, as a rule, increase their hydrostatic stiffness and damping, and thereby improve their dynamic qualities. Dynamic performance is especially important for seals in high-speed rotary machines.

Increased vibrations are accompanied by work near critical frequencies, the calculation of which, due to the lack of reliable data on the rigidity of supports and due to many random factors, cannot guarantee reliable detuning from resonant modes [1].

Another important indicator is the resource of the assembly, which is determined by the wear of the sealing surfaces. Therefore, the demand for centrifugal machines with high parameters is constantly growing [2, 3]. When designing sealing systems, it is necessary to coordinate their tightness and reliability, on the one hand, and resource indicators, on the other. In this case, depending on the design of contact seals, their chemical composition is also taken into account [4–7].

The creation of highly loaded equipment with sealing systems for non-standard operating conditions is impossible without taking into account the above factors. Examples of such non-standard products are pumping units for nuclear power plants [8] and turbopump units for rocket engines [9].

Pumps for the power industry need an increased resource of sealing units. To do this, it is necessary to use sealing systems with guaranteed controlled leakage in order to provide lubrication and cooling, and, consequently, the required service life.

The seals of rocket engines turbopump units operate under extreme conditions, for which vibration reliability and tightness are the most important requirements. The requirement for a low mass of equipment leads to the creation of units with flexible rotors, which can have significant deflections during transient conditions.

In connection with the transition of aerospace technology to reusable systems, their engines are designed for repeated activation, so the required resources reach tens of hours. All these new increased requirements for engine operation complicate the already difficult work of sealing assemblies.

Modern approaches to the creation of mathematical models of oscillatory systems based on experimental data are presented in [10–12]. Monograph [13] evaluates the coefficients of mathematical models of oscillatory systems, including rotor systems for multistage centrifugal machines. In work [14] the phenomena of loss of stability of rotation of the rotor in rolling bearings are considered.

Modern approaches in the field of linear and nonlinear dynamics of rotors and their practical applications are presented in the article [15]. The work [16] gives an estimate of the rigidity of segment bearings during balancing of flexible rotors of turbine units on an accelerating-balancing stand. Modern methods for determining the stiffness of active magnetic bearings and identifying damping from the frequency responses of control systems are presented in [17]. The application of the finite element method for calculating the stiffness and critical speed of the magnetic bearing-rotor system for electrical machines is described in the article [18]. The article [19] presents an analysis of the stability and vibration of a complex flexible rotor-bearing system. In [20], the phenomenon of subharmonic resonance of a symmetric ball-bearing-rotor system was studied. In [21], models are proposed for studying the critical frequencies of the rotor of a centrifugal compressor, taking into account the nonlinear stiffness characteristics of bearings and seals.

As indicated in [22], the energy of volume losses can be converted into useful energy if non-contact seals are used simultaneously as hydrostatic supports, capable of not only having high radial rigidity, but also effectively damping rotor vibrations to acceptable values, even in the presence of a significant imbalance. This effect is especially significant in the presence of steep velocity and pressure gradients inherent in the small gaps of slotted seals, on which high pressure drops are throttled and one of the surfaces belongs to the rotor, which simultaneously rotates and vibrates [23]. In [24], the dynamic characteristics of slotted seals as intermediate supports were studied.

The hydrodynamic characteristics of non-contact seals, taking into account the flow of the sealed liquid in the annular channels, the surfaces of which rotate and simultaneously perform radial-angular oscillations, are considered in [25]. The results of the conducted studies show that when creating sealing systems of modern centrifugal machines, it is necessary to take into account the effect of non-contact seals on the dynamic characteristics of the rotor [26].

Gap seals operate as hydrostatic dynamic bearings, which radial stiffness is proportional to the throttled differential pressure. As a rule, the rigidity of the seals is either comparable to or greater than that of plain bearings. This makes the seals act as additional intermediate supports [27].

The power characteristics are determined by the geometric and regime parameters of the seals: the initial taper and radial clearance, the length and average radius of the channel, the throttled pressure drop, the rotor speed, the flow swirl at the gap inlet, and the physical properties of the liquid. Analysis of the impact of gap seals on the dynamics of the rotor makes it possible to choose their design so that the vibration level of the rotor does not go beyond the permissible limits over the entire operating range [28–30].

2. MATERIALS AND METHODS

2.1. Model of the hydromechanical system "rotor-gap seals"

For an analytical description of the processes occurring in sealing units, we will consider them as automatic control systems. The model of the hydromechanical system "rotor-gap seals", presented in the paper [1], is shown in Fig. 1.

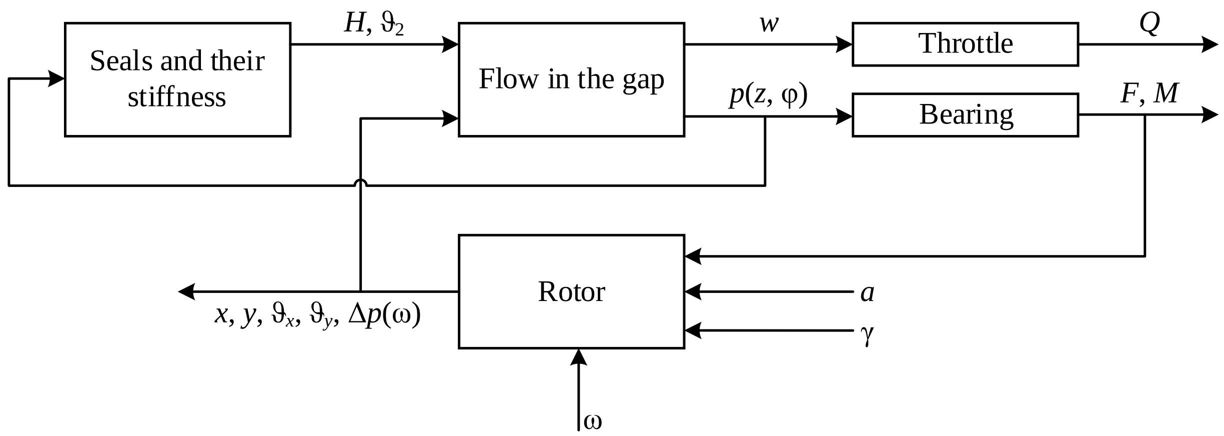

Fig. 1 – Hydromechanical system rotor-gap seals.

Hydrodynamic forces and moments arise in the sealing gaps, which depend on the nature of the rotor movement. At the same time, these forces and moments affect the radial and angular oscillations of the rotor [8].

On the other hand, the pressure distribution in the gap depends on the shape of the gap, and the deformations of the gap walls are determined by the pressure distribution in this gap. Thus, due to the occurrence of feedbacks, a hydromechanical system is formed.

2.2. Model of the impulse mechanical seal

The model of a non-contact mechanical seal, as a system for automatic control of the mechanical clearance and leakage, on the example of impulse compaction [9] is shown in Fig. 2.

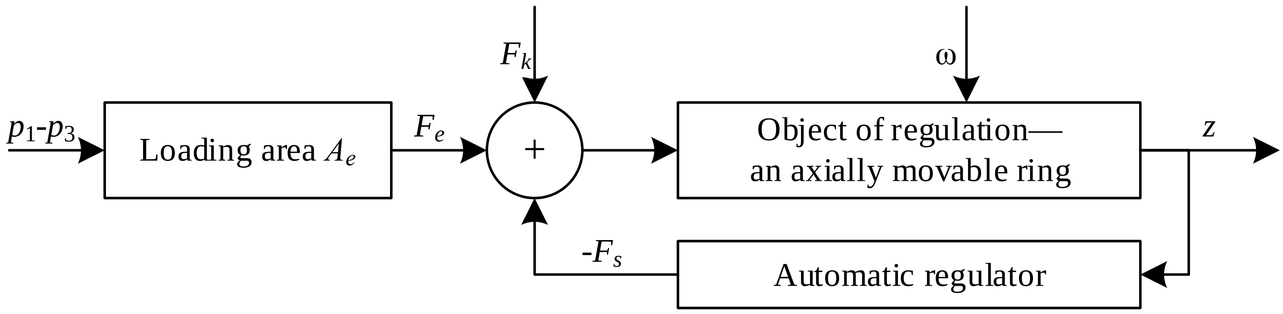

Fig. 2 – Impulse seal as the automatic control system.

3. RESULTS

3.1. Oscillation equations and dynamic stability assessment

Forced joint radial-angular oscillations of the rotor at a constant pressure drop across slotted seals are described by the equations [25]

$$\begin{gathered} a_1\ddot{u} + a_2\dot{u} + a_3 u \mp \mathrm{i}(a_4'\dot{u} + a_5' u)\omega - \left(\alpha_2'\dot{\theta} + \alpha_3'\theta\right)\omega \mp \\ \mp\mathrm{i}\left(\alpha_4\dot{\theta} + \alpha_5\theta - \alpha_0\theta\right) = \omega^2 a^* = \omega^2|a^*|\mathrm{e}^{\pm\mathrm{i}\omega t} \\ b_1\ddot{\theta} + b_2\dot{\theta} + b_3\theta \mp \mathrm{i}\left(b_4'\dot{\theta} + b_5'\theta\right)\omega + (\beta_2'\dot{u} - \beta_3' u)\omega \mp \\ \mp\mathrm{i}(\beta_4\dot{u} + \beta_5 u + \beta_0 \mathrm{u}) = \left(1 - j_0\right)\omega^2\gamma^* = \left(1 - \mathrm{j}_0\right)\omega^2|\gamma^*|\mathrm{e}^{\pm\mathrm{i}\omega t}; \end{gathered} \tag{1}$$

Substituting the solution of equations (1) in the form

$$u = u_a e^{\mathrm{i}(\omega t + \phi_u)} = \tilde{u} e^{\mathrm{i}\omega t}, \qquad \theta = \theta_a e^{\mathrm{i}(\omega t + \phi_\vartheta)} = \tilde{\theta} e^{\mathrm{i}\omega t}$$

we obtain a system of algebraic equations for the complex amplitudes A and Γ:

$$\begin{aligned} &[-a_1\omega^2 + a_3 + a_4\omega^2 + \mathrm{i}(a_2 - a_5)\omega]\tilde{u} - [(\alpha_3 - \alpha_4)\omega + \mathrm{i}(\alpha_2\omega^2 + \alpha_5 - \alpha_0)]\tilde{\theta} = A\omega^2 \\ &[-(\beta_3 - \beta_4)\omega + \mathrm{i}(\beta_2\omega^2 - \beta_5 - \beta_0)]\tilde{u} + [-b_1\omega^2 + b_3 + b_4\omega^2 + \mathrm{i}(b_2 - b_5)\omega]\tilde{\theta} = \Gamma\omega^2. \end{aligned} \tag{2}$$

From the system of inhomogeneous algebraic equations (2), after a series of transformations, we obtain the amplitudes and phases expressed in terms of external perturbations:

$$\begin{aligned} u_a &= \bar{\omega}^2 \sqrt{\frac{(AU_{22} - \Gamma U_{12})^2 + (AV_{22} - \Gamma V_{12})^2}{U_0^2 + V_0^2}}, \\ \theta_a &= \bar{\omega}^2 \sqrt{\frac{(\Gamma U_{11} - AU_{21})^2 + (\Gamma V_{11} - AV_{21})^2}{U_0^2 + V_0^2}}, \\ \phi_u &= -\mathrm{arctg}\frac{(AU_{22} - \Gamma U_{12})V_0 - (AV_{22} - \Gamma V_{12})U_0}{(AU_{22} - \Gamma U_{12})U_0 + (AV_{22} - \Gamma V_{12})V_0}, \\ \phi_\vartheta &= -\mathrm{arctg}\frac{(\Gamma U_{11} - AU_{21})V_0 - (\Gamma V_{11} - AV_{21})U_0}{(\Gamma U_{11} - AU_{21})U_0 + (\Gamma V_{11} - AV_{21})V_0}. \end{aligned} \tag{3}$$

where $\varpi = \omega/\Omega_{u0}$ – dimensionless frequency.

Using a similar algorithm, it is possible to obtain expressions for amplitudes and phases for other types of non-contact seals. For impulse seals [9]:

$$A(\omega) = \sqrt{\frac{b_1^2 + \omega^2 b_0^2}{U^2 + \omega^2 V^2}}, \qquad \phi = -\,\mathrm{arctg}\,\omega\,\frac{b_0 U - b_1 V}{b_1 U + \omega^2 b_0 V}. \tag{4}$$

As can be seen, expressions (3, 4) for the amplitudes and phases of various non-contact sealing systems have a similar form.

The stability is determined using the Routh-Hurwitz criterion for a system of 4th order

$$a_2(a_2 a_3 + a_4 a_5) - a_1 a_5^2 > 0,$$

which reduces to the form [24]:

$$\omega_u^2 < \frac{a_{21}^2 \Omega_{\mathrm{u0}}^2}{a_1 a_5^2 - a_{21}^2 a_{31} - a_{21} a_4 a_5} \tag{5}$$

It can be seen from inequality (5) that the main destabilizing factor is the circulation force characterized by the coefficient a_5. Damping a_{21}, gyroscopic force a_4 and shaft flexural stiffness Ω_{u0} stabilize the rotor in seals.

For impulse seals, the stability criterion is reduced to the inequality [8]:

$$V_0 < \frac{A_s E z_0 g_{s0}}{3(1 + n_i)(k_1 g_{30} - k_3 g_{10})(p_{10} - p_{30})} \tag{6}$$

from which it is possible to determine the stability-admissible volume of impulse seal chambers V_0, where

$$g_{s0} = g_i + g_{10} + g_{30} = g_{sn} z_0^3, \qquad k_1 = \frac{g_i' + g_{10}}{g_{s0}}, \quad k_3 = \frac{g_{30}}{g_{s0}}.$$

It can be seen from inequality (6) that the stability region of the seal expands due to a decrease in the volume of the chambers and in the coefficient of hydrostatic stiffness.

As can be seen from inequalities (5, 6), the dynamic stability conditions for various types of non-contact seals have a similar form. By adjusting the design parameters of the seals, which are included in these inequalities, it is possible to expand the boundaries of stable operation.

3.2. Frequency characteristics

Figure 3 shows numerical calculations the frequency responses built for three values of the taper parameter $\theta_0 = -0.3; 0; 0.3$. A positive phase characteristic corresponds to a negative total resistance.

Comparison of the results of calculations of frequency characteristics according to the obtained expressions with the data of experimental studies (Fig. 4) showed that the calculation errors do not exceed 5%, which suggests the possibility of using the obtained formulas in the calculation and design of centrifugal machines sealing systems with sufficient accuracy.

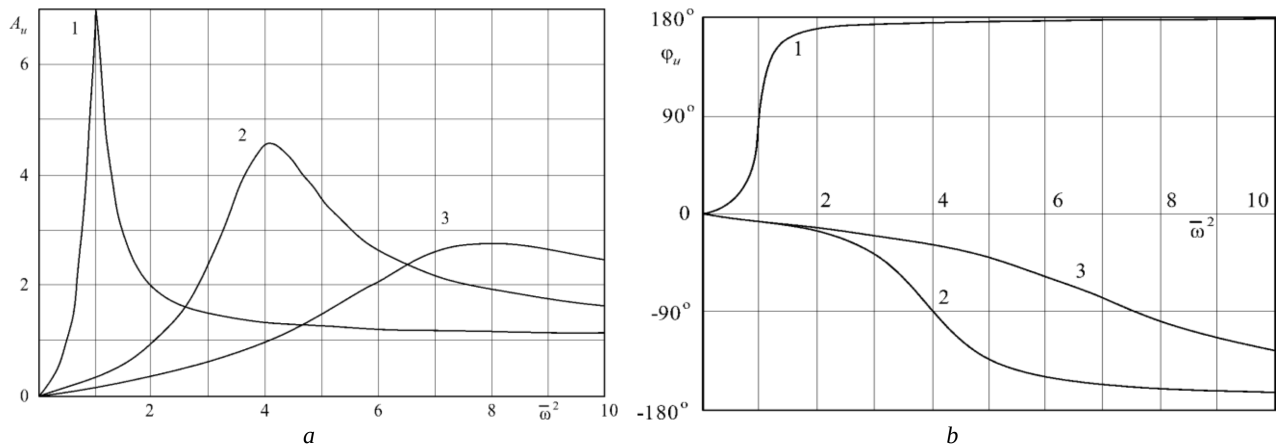

Fig. 3 – Frequency characteristics at a constant pressure drop across the seals:
a – amplitude, *b* – phase, $1 - \theta_0 = -0.3$; $2 - \theta_0 = 0$; $3 - \theta_0 = 0.3$.

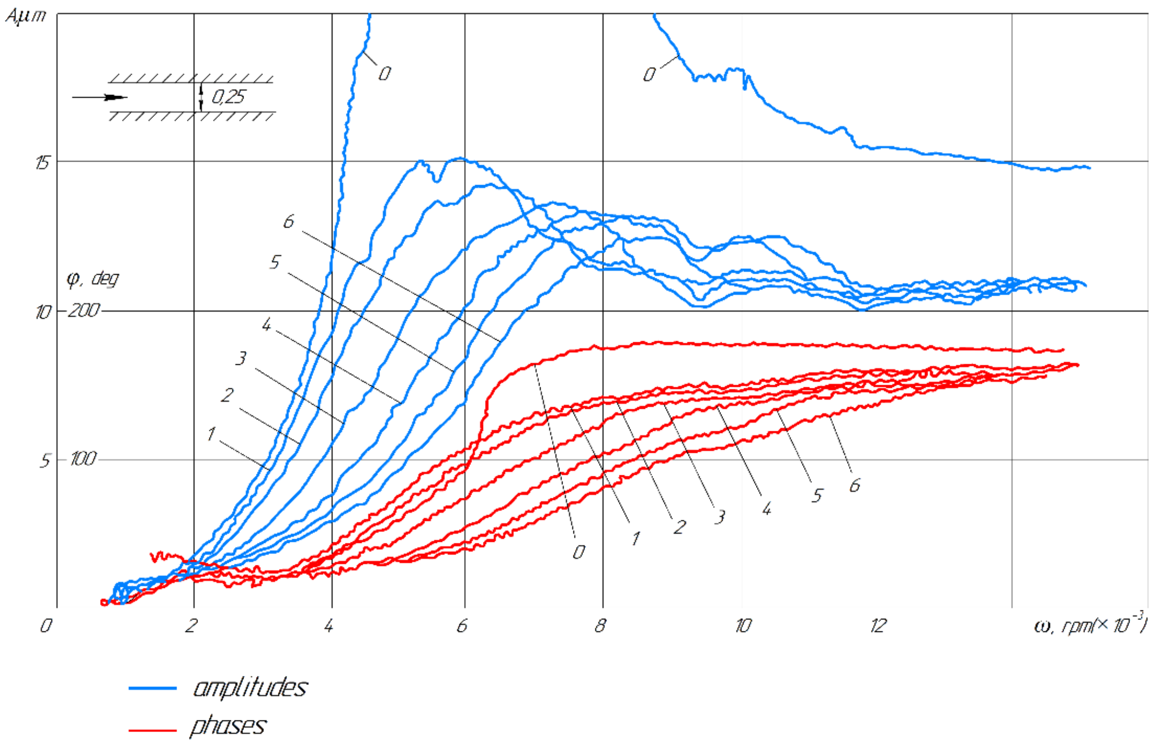

Fig. 4 – Amplitude and phase-frequency characteristics of the rotor in gap seals (experimental data)
Sealing pressure in MPa: 0 – 0; 1 – 0.18; 2 – 0.2; 3 – 0.4; 4 – 0.6; 5 – 0.8; 6 – 1.0.

For impulse seal practical interest are, first of all, the amplitude frequency characteristics. The results of their calculation for various values of the nominal gap are shown in Fig. 5.

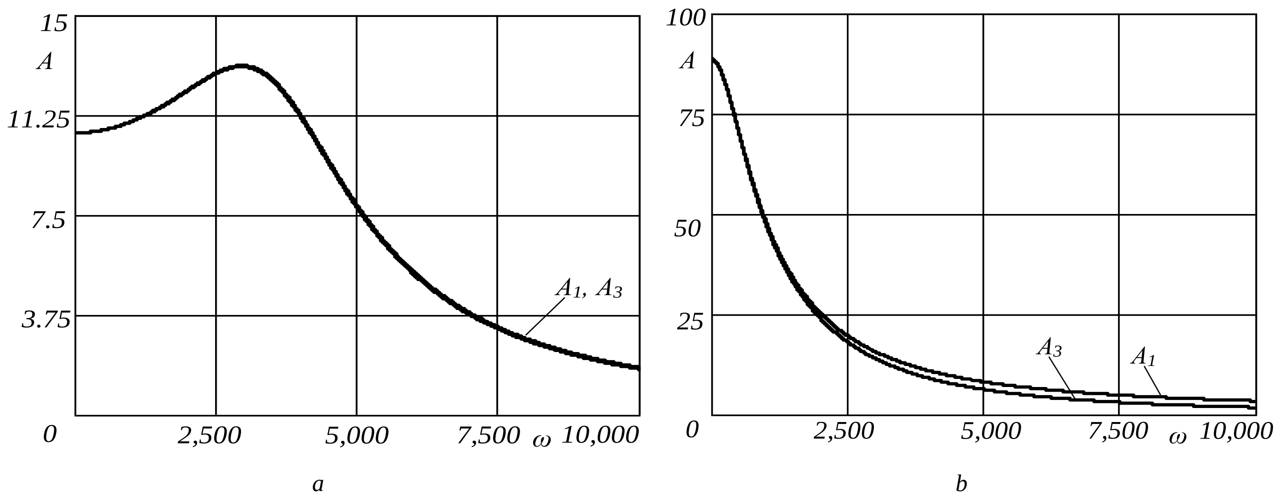

Fig. 5 – Amplitude-frequency characteristics for various nominal gaps z_n: *a* – z_n=3 μm; *b* – z_n=8 μm.

Also, the amplitude frequency characteristics make it possible to identify dangerous regions of rotational frequencies and select the sealing parameters so that the amplitudes of the forced axial oscillations of the ring do not go beyond the permissible limits.

4. DISCUSSION

The stability of the rotor can be judged by the sign of the real parts of the characteristic equation [22]: the presence of roots with a positive real part indicates the instability of the movement of the rotor in gap seals. To assess the impact on the stability of the taper and the pressure drop throttled on the seals in Fig. 6, graphs $\bar{n}_i(\bar{\omega})$ are shown along with the corresponding frequency diagrams $\bar{s}_i(\bar{\omega})$.

As can be seen from Fig. 6 in seals with a diffuser gap ($\theta_0 = -0{,}3$), the rotor is unstable even in the absence of rotation. The critical diffuser is $\theta_{0*} = -0.27$. In a cylindrical gap at $\Delta p_0 = 1.5$ MPa, the limiting stability dimensionless speed is $\bar{\omega}_* \approx 6.5$. With an increase in the pressure drop to 4 MPa, an instability region appears in the range $\bar{\omega} \approx 3 - 4$. Confusor seals ensure the stability of the rotor at all investigated speeds.

In the cylindrical $\theta_0 = 0$ and confusor $\theta_0 = 0.3$ gaps, there is an intensive increase in natural frequencies (except for the first) with increasing rotation frequency. Because of this, three critical frequencies, which are determined by the points of intersection of frequency diagrams with straight lines $\bar{s} = \bar{\omega}$, exist only for the rotor in slotted seals with a diffuser shape of the annular gap. In seals with cylindrical and confusor gaps, only the first critical rotation speed occurs.

In Fig. 6 also shows graphs of the real parts of the roots of the characteristic equation for the generalized constant B. From the graphs of the real parts one can judge stability: if among the roots there are roots with a positive real part, then the rotor is unstable at the corresponding rotation frequencies. In particular, the rotor in seals with a diffuser gap $\theta_0 = -0.3$ is unstable at all frequencies. The rotor in confusor seals remains stable throughout the entire range of rotation speeds under consideration. Thus, the destabilizing effect of slotted seals with a diffuser gap and the pronounced stabilizing effect of confusor channels are confirmed.

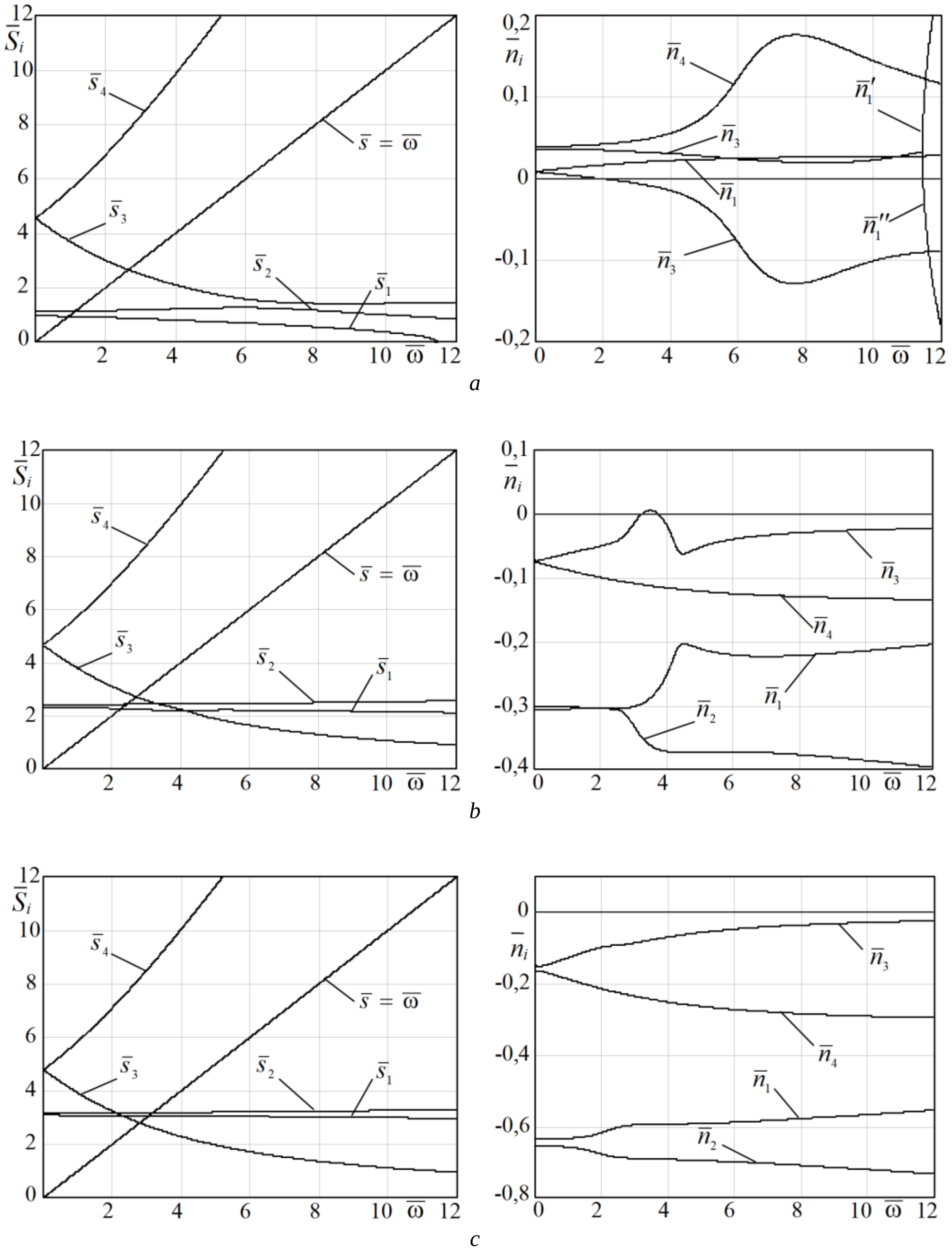

Fig. 6 – Frequency diagram and damping coefficients at $\Delta p_0 = 4$ MPa; taper index: *a* – $\theta_0 = -0.3$; *b* – $\theta_0 = 0$; *c* – $\theta_0 = 0.3$.

5. EXAMPLES OF BUILDING A COMPLEX SEALING SYSTEM

Figure 7 shows the sealing system of the Nuclear Power Plant (NPP) main circulation pump. The operating pressure and water temperature in the primary circuit are 12.5 MPa and 270°C, respectively. The seal operates on blocking water, which is taken from the primary circuit, cooled to 40°C and cleaned by passing through the cooler and ion-exchange filter. Automatic regulators maintain a predetermined (0.5–0.6 MPa) excess of the blocking water pressure over the pressure in the pump cavity, as a result of which about 50% of the input water (0.3–0.5 m^3/h) enters the pump, excluding the exit from it of a hot radioactive coolant.

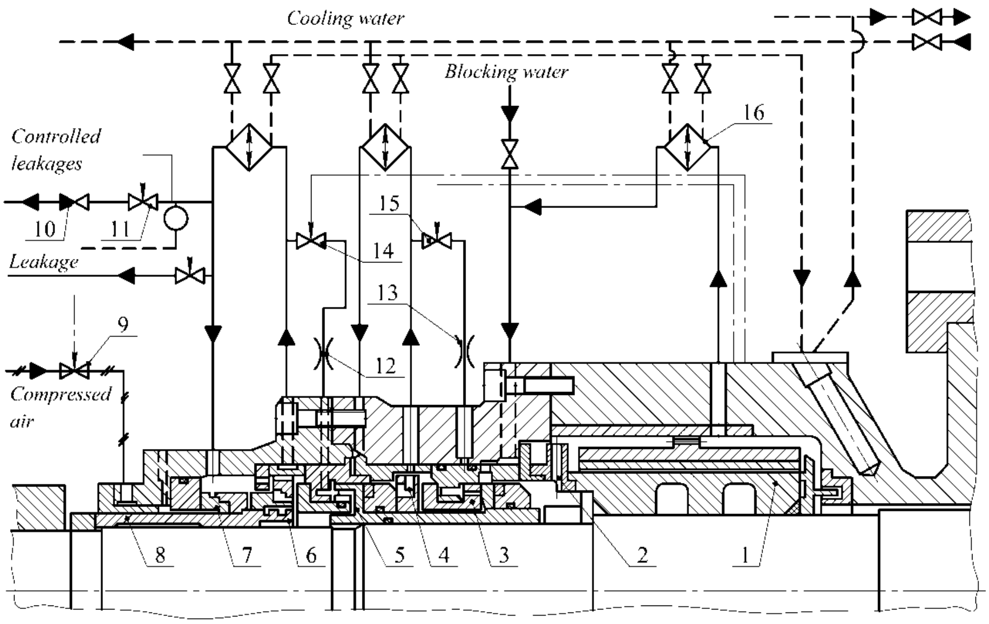

Fig. 7 – Diagram of RCP shaft seal.

Figure 8 shows a design diagram of the separation unit of the TPU seals between the oxygen pump and the TPU turbine of the Space Shuttle engine, driven by hot gas with excess hydrogen (“sweet” gas). With an “acidic” fuel environment in the pump and “sweet” gas in the turbine adjacent to the pump, leakage from the pump to the turbine and mixing of these leaks with turbine gas is unacceptable, since this mixture is explosive. The separating block of the sealing system in this case is a very important element of the pump and consists of several sealing units: a bellows mechanical seal with a friction pair 1, 2 on the side of the cavity 3 with liquid oxygen, a block of floating rings 4, 5 and a block of floating rings 6, 7 with sides of the cavity 8 with gaseous hydrogen, between which there is a chamber for supplying 9 helium barrier gas. Spiral grooves are made on the contact surface of the rotating ring 1. Drain-age channels 10 serve to drain the helium mixture and oxygen leaks, and channels 11, 12 – to drain the helium mixture and the hydrogen-enriched gas mixture over-board the engine.

Compressed helium on board the ship is stored in a cylinder, and its consumption can determine the number of engines starts and the total duration of the ship's flight.

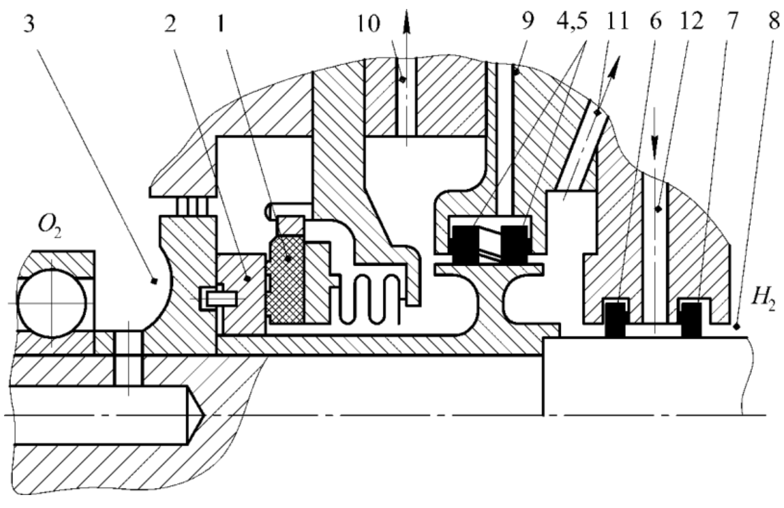

Fig. 8 – Sealing system between oxygen pump and turbine with reducing media:
1, 2 – rotating and non-rotating rings of a friction pair; 3 – cavity of liquid oxygen; 4, 5 – block of floating rings from the side of the oxygen cavity; 6, 7 – a block of floating rings from the side of the cavity with gaseous hydrogen; 8 ~ cavity with gaseous hydrogen; 9 – chamber for supplying helium barrier gas; 10 – drainage channels for removing helium mixture and oxygen leaks; 11, 12 – drainage channels for removing a mixture of helium and gas enriched with hydrogen.

The operating parameters of such a mechanical parking seal TPU LRE: sealing pressure drop 3.1 MPa, sliding speed 180 m/s; the required resource is 10 h with the number of inclusions of TPU on the order of 300.

5. CONCLUSIONS

The studies carried out have shown that all sealing units with throttling gaps or sealing paths filled with a high-pressure medium to be sealed should be considered as dynamic systems. The medium to be sealed, acting on the walls of the sealing paths, affects the dynamic state of the rotor.

It is shown that the purposeful choice of the design parameters of the seals makes it possible to improve the vibrational state of the rotor. In this case, the initially “flexible” dynamically rotor, in combination with properly designed seals, becomes “rigid”. By purposeful selection of seal parameters, it is possible to reduce the amplitude of forced oscillations of the rotor by 3-4 times.

Studies have shown that when develop-ing non-contact seals, it is necessary to take into account not only their direct purpose – reducing volume losses, but also their no less important function, which is to provide the necessary vibration characteristics.

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