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Sealing systems and their influence on rotor dynamics

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In liquid rocket engines, the fuel supply system is implemented by turbopump units. It is very important to ensure the sealing of the unit cavities with different pressures, as well as to prevent the leakage of aggressive liquids to the outside. Non-contact seals, in addition to sealing, perform an equally important function - they improve the vibration state of the centrifugal machine. Dynamic characteristics are especially important for seals in high-speed rotary machines. A targeted selection of seal design parameters allows improving the vibration state of the rotor. An initially dynamically "flexible" rotor in combination with correctly designed seals can become dynamically "rigid".

1 Introduction

Multistage high-pressure centrifugal pumps, widely used in various industries, are characterized by a steady trend toward increasing operating parameters—flow rates, pressures, and rotor speeds—that is, concentrating higher capacities in single units [1]. The pressure developed by centrifugal machines is proportional to the square of the rotor speed, so increasing the speed is the most efficient way to achieve higher pressures. As a result, high-pressure centrifugal machines are typically high-speed [2]. For such machines, rotor dynamics issues are particularly pressing [3].

When designing centrifugal pumps for any parameters, the primary objectives are vibration reduction and the development of reliable and sufficiently tight seals [4]. A unique feature of centrifugal machines is that these challenges are interrelated and, in most cases, can be satisfactorily addressed through the targeted development of non-contact seals designs [5]. Design measures aimed at increasing the hydraulic resistance of seals typically increase their hydrostatic stiffness and damping, thereby improving their dynamic performance [6].

The rotor seals of high-speed centrifugal machines are developing into complex systems that determine the reliability of the units [7]. When creating sealing systems for non-standard operating conditions, it is necessary to take into account their influence on the vibration characteristics of equipment [8].

Energy conversion in non-contact seals must be considered to determine radial stiffness that effectively reduces rotor vibration. In this case, non-contact seals play the role of additional dynamic supports of the rotor [9].

The rotor and non-contact seals represent a closed hydromechanical system. When solving the problem of calculating the rotor dynamics of a centrifugal ma-chine, it is necessary to take into account the influence of non-contact seals, which can change the critical frequencies of the rotor and influence the amplitude of its forced oscillations [10].

Clearance seals act as hydrostatic dynamic bearings, the radial stiffness of which is proportional to the throttled pressure drop. Typically, the stiffness of seals is either comparable to or superior to that of plain bearings. Due to this, seals act as additional intermediate supports [11].

The geometric and design parameters of seals determine their power characteristics. The analysis of the influence of clearance seals on the dynamics of the rotor makes it possible to select their design in such a way that the level of rotor vibration does not exceed acceptable limits throughout the entire operating range [12].

In liquid rocket engines, the fuel supply system is implemented by turbopump units, which include pumps and a pump drive - a gas turbine. Turbopump units supply highly aggressive and toxic fuel components to the combustion chamber, which can enter into a chemical reaction when interacting. The physical properties of the working fluids sup-plied by the pumps and the working fluid in the gas tur-bine tract differ significantly.

Therefore, it is very important to ensure the sealing of the unit cavities with different pressures, as well as to prevent the leakage of aggressive liquids to the out-side.

Automatic compensation systems for axial forces acting on the rotor of a multi-stage centrifugal pump simultaneously perform the functions of a self-adjusting non-contact mechanical seal and a highly loaded radial-axial hydrostatic bearing [13]. Such systems largely determine the vibrational state of the rotor. The leakage energy can not only provide the necessary bearing capacity but, most importantly, reduce rotor vibration to an acceptable level even in the presence of significant imbalance [14].

2 Models of hydromechanical systems

The rotor and non-contact seals should be considered a closed hydromechanical system, in which the flow of the pumped medium in the throttling channels, along with the inertial and hydrodynamic characteristics of the flow, act as the connecting link [15].

The influence of the medium is particularly significant in the presence of large velocity and pressure gradients. Such conditions are typical for the small gaps of non-contact seals, where large pressure differences are throttled, and one of the walls belongs to the rotating and vibrating rotor.

The model of the hydromechanical system “rotor-slotted seals”, presented in the author’s paper [16], is shown in Figure 1.

Despite the diversity of designs, automatic balancing devices are built on a common principle: negative feedback is created between the balancing force and the axial position of the rotor, ensuring only small deviations in the axial position of the rotor from a given value [17].

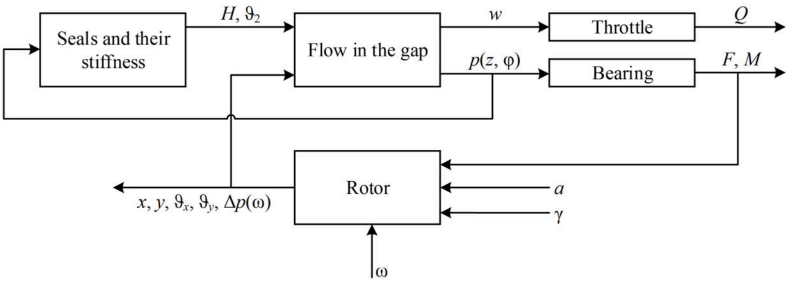


Figure 1: The model of the hydromechanical system rotor-slotted seals: x, y — radial vibrations of the rotor; ϑ_x, ϑ_y — rotor angular oscillations; F, M — hydrodynamic forces and moments arising in sealing gaps; H, ϑ_2 — mean radial clearance and taper; p (z, φ) — gap pressure; w — gap fluid flow rate; Q — seal leakage; Δp — sealing pressure; a — eccentricity of mass center; ω — rotor speed; γ — the angle of deviation of the inertia axis.

The model of the hydromechanical system "rotor-automatic balancing device", presented in the author's monograph [18], is shown in Figure 2.

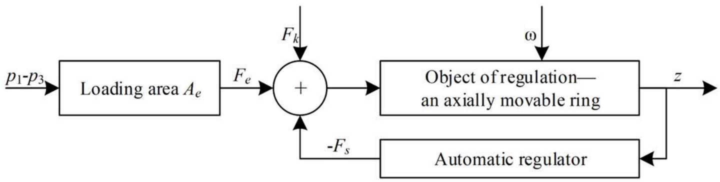


Figure 2. Model of the balancing device z — end gap (adjustable); Δp — sealing pressure; T — axial force; F_z — pressure force in the end gap; k, Δ — compression ratio and deformation of the pressing device.

3 Results

Research has shown that all sealing assemblies with throttling gaps or sealing paths filled with a high-pressure medium to be sealed should be considered dynamic systems.

3.1 Using non-contact seals as dynamic rotor supports

The sealing medium, acting on the walls of the sealing paths, affects the dynamic state of the rotor. The basic idea of this technology [19] (Figure 3) is to utilize the energy lost when the sealing medium overcomes the throttling channels of the seals to dampen the kinetic energy of the moving elements of centrifugal machines, thereby improving the vibration characteristics of the equipment. In particular, radial and axial non-contact seals, which throttle enormous pressure differences, can act as static supports, and with the right design, as dynamic supports. This should be taken into account when de-signing critical power equipment.

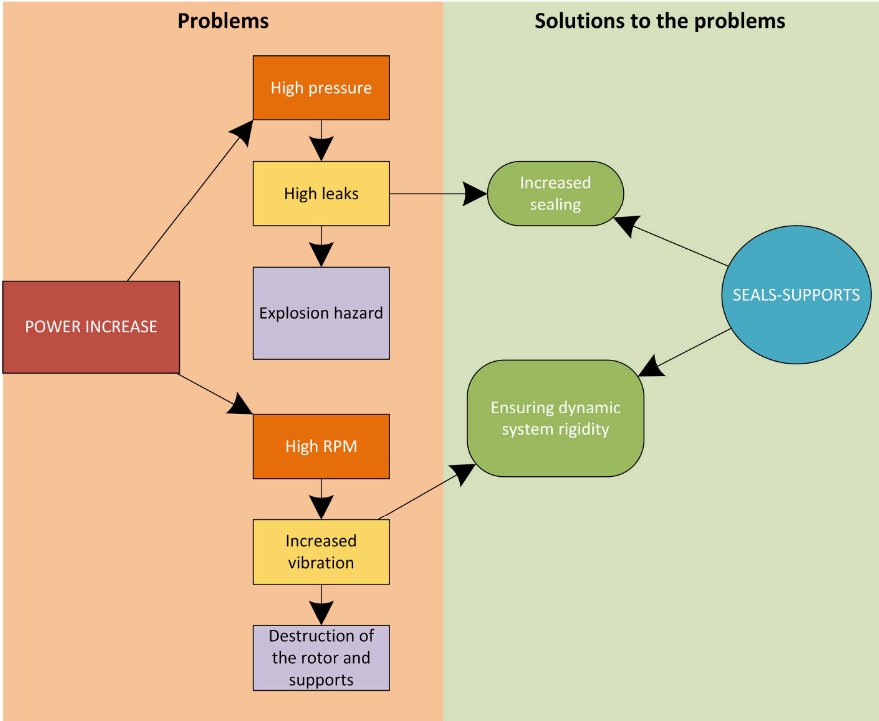


Figure 3. Using non-contact seals as dynamic rotor supports

The rotor of a modern centrifugal machine, rotating within the seals, together with the automatic axial force relief system, represents a unique closed-loop hydromechanical system. The system's unique feature stems from the fact that the rotor is subjected to hydrodynamic forces (and moments) of various natures from the seals: inertial, dissipative, gyroscopic, potential, and circulatory. All these forces influence rotor vibrations in different ways and, at the same time, are themselves dependent on the nature of these vibrations. It is these feedback loops that allow rotor vibration suppression, but only if they are properly directed. Otherwise, a reverse reaction can occur—instead of vibration suppression, an excellent vibration exciter can result.

The proposed method allows the creation of pumps without outriggers, using gap seals on impellers as rotor supports, operating due to the pressure differential of the pumped fluid.

Because the seals are closely spaced along the rotor, their action stiffens the high-speed flexible rotor.

In aerospace engineering, where, in addition to high sealing pressures and rotor speeds, there are very high restrictions on the weight and dimensions of equipment, the use of seals as dynamic bearings is especially important.

3.2 Sealing systems for rocket engines

Work on the development of turbopump assemblies for high-power rocket engines for spacecraft has raised researchers' interest in the dynamic characteristics of gap seals and rotor vibrations within the seals [20].

Turbine pump seals for liquid rocket engines (LRE) operate under extreme conditions, for which vibration reliability and leak-tightness are critical requirements [21]. Requirements to limit the weight and dimensions of equipment lead to the development of units with dynamically flexible rotors, which can exhibit significant vibration levels during transien/t conditions.

The rotational speed of rocket engine turbopump rotors reaches 200,000 rpm and is limited only by the strength of the impellers (the maximum permissible peripheral speed of the impeller rim is 450 m/s). This is the rotational speed of the fuel (liquid hydrogen) turbopump assemblies (TPA) manufactured by “Aerojet Liquid Rocket” (USA). Turbomachinery units produce hundreds of megawatts of power, while maintaining extremely low weight and dimensions. Among all types of blades used in propulsion and power plants, TPA have the highest specific power of 50-150 kW/kg [22].

The most important requirements for rocket and aircraft components are vibration reliability and leak tightness. One TPA schematic is shown in Figure 4.

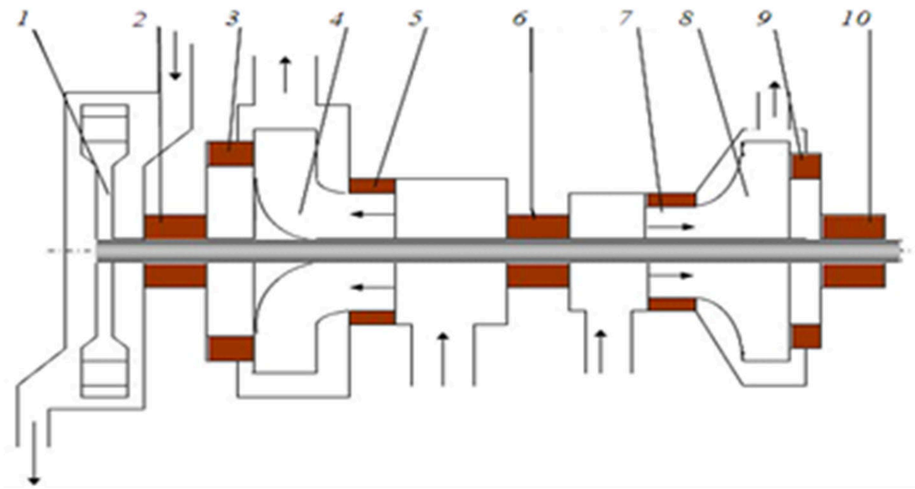


Figure 4. Schematic diagram of the turbopump unit and the location of the main elements of the rotor seals

The diagram shows the arrangement of the seals: 5, 7 — front gap seals of the oxidizer 4 and fuel 8 pumps; 3, 9 — rear impeller seals; 10 — fuel pump end seal; 6 — intermediate seal, preventing mixing of dissimilar, often self-igniting liquid working media of the oxidizer and fuel pumps; 2 — seal, limiting the ingress of the oxidizer, which can be a cryogenic liquid, into the gas turbine 1 cavity (the gas temperature can reach one and a half thousand degrees Kelvin).

In turbopump units, seals are the most important components determining their reliability. According to statistics, approximately 60% of failures of modern liquid propellant rocket engine units are associated exclusively with malfunction of the sealing systems. Thus, the operability, service life, reliability, and efficiency of turboprop units largely depend on the perfection of their sealing devices. High quality, preventing fuel component mixing, and separation of adjacent pump and turbine cavities are essential for the successful operation of turbopump units. In turbopump shaft seal design, additional structural elements, such as drains and redundant seals, are used to reliably separate the cavities in the seal assembly blocks. As a result, the seal assembly becomes one of the most complex components of the turbopump design, and its design has a decisive impact on the overall layout of the turbopump assembly.

Currently, the most effective method for balancing axial forces on large multistage high-pressure pumps is the use of automatic balancing devices. In this case, the automatic unloading system simultaneously functions as a high-capacity radial thrust bearing operating on the pumped liquid and as an end seal with a self-adjusting end clearance. Thus, the correct design and key parameters of the system largely determine the magnitude of volumetric losses and, most importantly, the reliability of the entire machine.

In turbopump units, the impellers 7 of the pump and the drive gas turbine 6 are mounted on the same shaft, so the automatic unloading system must balance the total axial force acting on the pump and turbine stages (Figure 5). Unlike a pump, the axial force on a turbine is directed from the inlet to the outlet, since the gas pressure at the inlet is greater than at the outlet.

The complexity of axial rotor balancing means that balancing devices are self-regulating feedback systems. Under certain conditions, self-exciting oscillations can occur in them, exerting a decisive influence on the vibrational state of the entire unit. Moreover, the radial hydrodynamic forces and moments in the gap seals, which determine radial oscillations, depend on the pressure in the hydraulic bearing chamber, which, in turn, depends on the axial oscillations of the rotor. As a result, axial oscillations can excite radial oscillations, and vice versa.

During pump operation, the rotor is subjected to near-harmonic disturbances in the form of discharge pressure pulsations and axial force. The pulsation frequency is equal to or a multiple of the rotational speed. Under the influence of these disturbances, the rotor undergoes forced oscillations, the amplitude of which depends on the distance between the rotational speed and the natural frequencies of the rotor-seals-balancer system. Therefore, to tune out possible resonant oscillations, it is necessary to know the natural frequencies of the system. To estimate the amplitudes of forced oscillations in operating modes, it is necessary to construct amplitude-frequency characteristics.

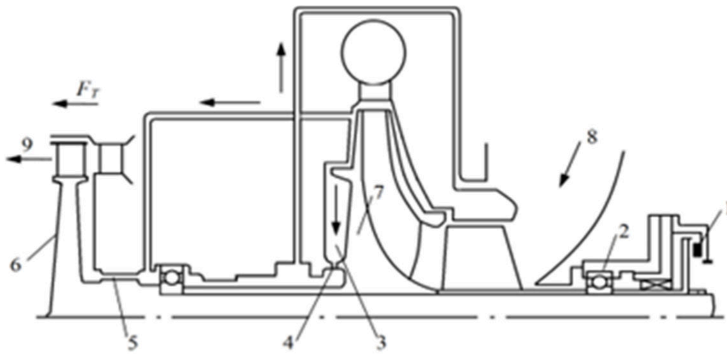


Figure 5. Typical diagram of the automatic unloading device of the liquid rocket engine turbopump

The axial and radial hydrodynamic forces arising in the throttling gaps of the balancing device are interrelated. As a result, the rotor-automatic unloading system, under the influence of inevitable radial static imbalance, discharge pressure pulsations, and harmonic changes in the axial force acting on the rotor, experiences interrelated forced radial-axial oscillations. At rotational speeds coinciding with any natural frequency, the resonant amplitudes of these oscillations can exceed permissible limits, so determining resonant rotational frequencies and adjusting for them is of great practical importance.

The inertia of the fluid during its unsteady flow in the throttling channels of automatic balancing devices has a damping effect on the axial oscillations of the rotor: it reduces the amplitudes of the resonant oscillations and decreases the critical rotational speeds.

The effect of inertia increases with decreasing discharge pressure. Therefore, taking inertia into account when determining critical frequencies is especially important for pumps with high-speed factors and for pumps operating over a wide range of discharge pressures.

Due to small end clearances and high pressures, the balancing system has high axial hydrodynamic stiffness, which determines comparatively high critical frequencies. However, for high-speed pumps, there is a real risk of resonant axial vibrations, which worsen the overall vibration performance of the machine. Also dangerous are alternating stresses that occur in heavily loaded components of balancing devices during forced axial vibrations of the rotor.

3.3 The influence of the seals - supports design parameters on rotor vibrations

Targeted selection of seal design parameters improves the rotor's vibration behavior. In this case, a high-speed, dynamically flexible rotor, combined with properly designed seals, becomes "rigid." The proposed method allows exploiting this effect depending on the seal design characteristics and, by modifying them at the design stage, adjusting the "seal-rotor-automatic unloading" system to eliminate resonant operating modes. Figure 6, a show a non-contact seal with a confusor gap.

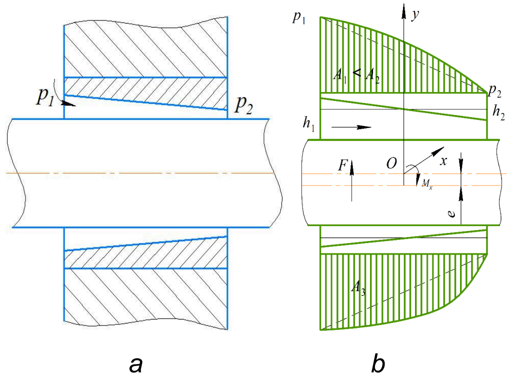


Figure 6. Seal with a confusor gap (a). Pressure distribution diagram in the confusor gap (b).

As can be seen from Figure 6 b, the pressure in the confusor gap is distributed in such a way that when the shaft is displaced relative to the housing (misalignment occurs), a radial force is generated that is directed against this misalignment, which leads to self-centering of the rotor in the housing.

We introduce the concept of the channel taper parameter

$$\theta_0 = \vartheta_0 l / 2H,$$

where ϑ_0 is total taper angle of the channel; l is channel length; H is average channel clearance.

In the author's work [16], expressions were obtained for calculating the amplitudes and phases of rotor oscillations in seals, expressed through external disturbances:

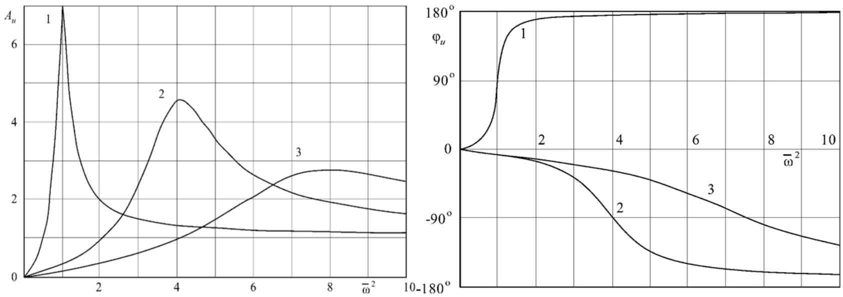
$$u_a = \bar{\omega}^2 \sqrt{\frac{(A U_{22} - \Gamma U_{12})^2 + (A V_{22} - \Gamma V_{12})^2}{U_0^2 + V_0^2}},$$

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

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

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

Figure 7 shows the frequency responses built for three values of the taper parameter θ_0. A positive phase characteristic corresponds to a total negative resistance.



a *b*

Figure 7. Frequency characteristics at a constant pressure drop across the seals: a - amplitude, b - phase, $\mathbf{1-\theta_0=-0,3}$; $\mathbf{2-\theta_0=0}$; $\mathbf{3-\theta_0=0,3}$

4 Discussion

Let us consider the case when the pressure difference throttled at the seals is proportional to the square of the rotor speed $\Delta p_0 = B\omega^2$, then for laminar flow and a self-similar region of turbulent flow we can write:

$$U = U_l = U_t = \Omega_{u0}^2\left[1-\left(a_1-a_{31}^{'}-a_4^{'}\right)\overline{\omega}^2\right]$$

$$V_l = \Omega_{u0}^2\bar{\omega}\left[\left(\frac{a_{20}+2k_d}{\Omega_{u0}}-\frac{a_5^{'}}{\Omega_{u0}}\right)+2k_g\theta_0K_i^{'}\Omega_{u0}\overline{\omega}^2\right],$$

$$V_t = \Omega_{u0}^2\bar{\omega}\left[\frac{a_{20}}{\Omega_{u0}}+\left(a_{21}^{'}-a_5^{'}\right)\bar{\omega}\right]$$

In dimensionless form

$$U = \Omega_{u0}^2a_1\left[\frac{1}{a_1}-\left(1-\frac{a_{31}^{'}}{a_1}-2\eta_u^{'}\right)\overline{\omega}^2\right],$$

$$V_l = \Omega_{u0}^2a_1\bar{\omega}\left[2(\xi_{u0}+2\xi_{u1d})-\varsigma_u^{'}+\xi_{u1g}^{'}\overline{\omega}^2\right],$$

$$V_t = \Omega_{u0}^2a_12\bar{\omega}\left[\xi_{u0}+\left(\xi_{u1}^{'}-\varsigma_u^{'}\right)\bar{\omega}\right],$$

where $\xi_{u1d} = \frac{k_d}{2a_1\Omega_{u0}}$, $\xi_{u1g}^{'} = 2k_g\theta_0\frac{K_i^{'}\Omega_{u0}}{a_1}$.

Using these expressions, we find the amplitude and phase frequency characteristics. In this case, they are different for the laminar and self-similar regions of turbulent flows:

$$A_u^{(l)}(\bar{\omega}) = \frac{\frac{\bar{\omega}^2}{a_1}}{\sqrt{\left[\frac{1}{a_1} - \left(1 - \frac{a_{31}'}{a_1} - 2\eta_u'\right)\bar{\omega}^2\right]^2 + \bar{\omega}^2\left\{\left[2(\xi_{u0} + 2\xi_{u1d}) - \varsigma_u'\right] + \xi_{u1g}'\bar{\omega}^2\right\}^2}}, \quad (1)$$

$$\phi_u^{(l)}(\bar{\omega}) = -arc\bar{t}g\,\omega \frac{\left[2(\xi_{u0} + 2\xi_{u1d}) - \varsigma_u'\right] + \xi_{u1g}'\bar{\omega}^2}{\frac{1}{a_1} - \left(1 - \frac{a_{31}'}{a_1} - 2\eta_u'\right)\bar{\omega}^2}$$

$$A_u^{(t)}(\bar{\omega}) = \frac{\frac{\bar{\omega}^2}{a_1}}{\sqrt{\left[\frac{1}{a_1} - \left(1 - \frac{a_{31}'}{a_1} - 2\eta_u'\right)\bar{\omega}^2\right]^2 + 4\bar{\omega}^2\left[\xi_{u0} + (\xi_{u1}' - \varsigma_u')\bar{\omega}\right]^2}} \quad (2)$$

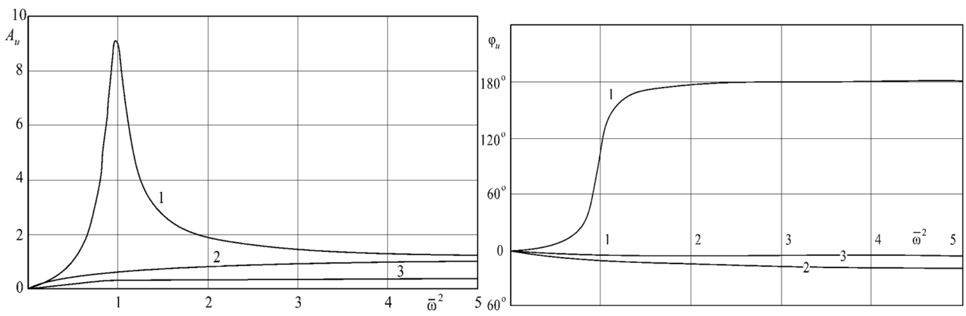
$$\phi_u^{(t)}(\bar{\omega}) = -arc\bar{t}g\,\omega \frac{2\left[\xi_{u0} + (\xi_{u1}' - \varsigma_u')\bar{\omega}\right]}{\frac{1}{a_1} - \left(1 - \frac{a_{31}'}{a_1} - 2\eta_u'\right)\bar{\omega}^2}$$

In both cases, the amplitude characteristics do not have resonant peaks under the condition

$$1 - \frac{a_{31}'}{a_1} - 2\eta_и' < 0 \quad (3)$$

the denominators in formulas (1) and (2) increase monotonically with increasing rotational speed.

Figure 8 shows the frequency characteristics of the rotor test model, built according to formulas (2) for three values of the taper parameter.



a *b*

Figure 8. Amplitude (a) and phase (b) frequency characteristics, taking into account the dependence of the pressure drop across the seals on the rotor speed: $\mathbf{1 - \theta_0 = -0,3}$; $\mathbf{2 - \theta_0 = 0}$; $\mathbf{3 - \theta_0 = 0,3}$

For a diffuser channel ($\theta_0 = -0{,}3$), the total resistance is negative, and the imaginary part V_1 is positive. As a result, the phase characteristic is positive. For cylindrical ($\theta_0 = 0$) and confusor ($\theta_0 = 0{,}3$)channels, condition (3) is satisfied. In this case the phase characteristic does not go beyond the fourth quadrant: $-\frac{\pi}{2} < \phi_{и} \leq 0$.

A comparison of the results of the calculations of frequency characteristics according to the obtained expressions with the data of experimental studies [23] (Figure 9) shows that the calculation errors do not exceed 5%, which suggests the possibility of using the obtained formulas.

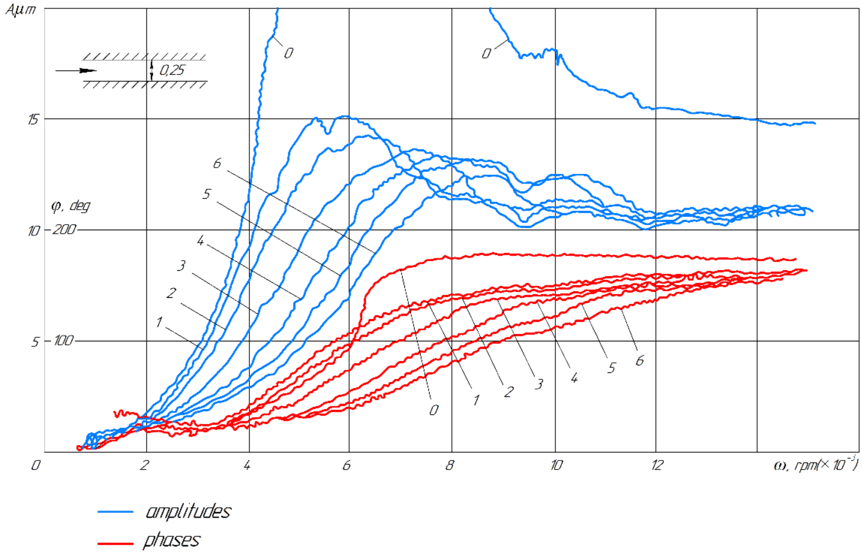


Figure 9. Amplitude (A) and phase (φ) frequency characteristics of the rotor in gap seals (experimental data): ω is a rotor rotation frequency; compaction pressure in MPa: 0—0; 1—0.18; 2—0.2; 3—0.4; 4—0.6; 5—0.8; 6—1.0.

An analysis of the gap seals' dynamic characteristics showed that the force coefficients of gap seals are determined by geometric (clearance, radius, length, taper, shape of the leading edges) and operational (pressure drop, operating speed range, physical properties of the pumped medium) parameters. With a purposeful choice of these parameters, it is possible to influence the vibrational state of the rotor and the machine itself.

An important feature of centrifugal machines is that the pressure drops throttled at the gap seals are proportional to the rotor speed. This is due to the self-tightening effect of the rotor, which leads to a positive shift of the critical frequencies.

The stability is determined using the Routh-Hurwitz criterion for a system of 4th order [24]

$$a_2(a_2a_3 + a_4a_5) - a_1a_5^2 > 0$$

which reduces to the form:

$$\omega_u^2 < \frac{a_{21}^2 \Omega_{u0}^2}{a_1a_5^2 - a_{21}^2a_{31} - a_{21}a_4a_5}$$

where: $\Omega_{u0}$is the bending stiffness of a shaft;

a_1, a_2, a_3, a_4, a_5, a_{21}, a_{31} are the coefficients of the equations of motion, which depend on the parameters of the seals.

It can be seen from inequality (4) 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.

Thus, the proposed technology enables the creation of centrifugal machines with seals- bearings that enhance their vibration resistance. Traditional rotor bearings (rolling or sliding) are static supports that support static loads and do not dampen rotor vibrations. Non-contact seals can function as dynamic supports by throttling large pressure differences across them with careful selection of design parameters.

5 Summary and Conclusion

Research has shown that convergent seals ensure rotor stability at all studied rotational speeds and increase its critical frequency by 4-5 times compared to a rotor sealed with smooth gaps. The proposed seal design, which creates convergent gaps, suppresses vibration amplitude and eliminates the critical frequency of the centrifugal rotor, which positively impacts its vibrational state.

The inertia of the fluid during its unsteady flow in throttling channels has a damping effect on rotor vibrations: it reduces the amplitude of resonant oscillations and distances critical rotational speeds. Therefore, taking inertia into account when determining critical speeds is especially important for pumps with a high-speed factor and for pumps operating over a wide range of discharge pressures.

6 Nomenclature

LRE	Liquid Rocket Engine
TPA	Turbo pump assemblies

Variable	***Description***	***Unit***
a	eccentricity of mass center	m
F	hydrodynamic forces arising in sealing gap	N
H	mean radial clearance of gap seal	m

LRE	Liquid Rocket Engine
TPA	Turbo pump assemblies

Variable	***Description***	***Unit***
k	compression ratio of the pressing device	-
M	hydrodynamic moments arising in sealing gap	Nm
$p(z, \varphi)$	gap pressure	Pa, N/m^2
p_1	sealing pressure	Pa, N/m^2
p_2	outside pressure	Pa, N/m^2
Q	seal leakage	m^3/h
T	axial force	N
w	gap fluid flow rate	m/s
x, y	radial vibrations of the rotor	m
z	end gap (adjustable)	m
z_0	clearance in steady state of the unloading device	m
γ	angle of deviation of the inertia axis	rad
Δ	deformation of the pressing device	m
θ_0	channel taper parameter	-
ϑ_2	mean radial taper	rad
ϑ_x, ϑ_y	rotor angular oscillations	rad
ω	rotor speed	s^{-1}
Ω_{u0}	the bending stiffness of a shaft	N/m

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