Доклади на Българската академия на науките
Comptes rendus de l'Académie bulgare des Sciences

Tome 77, No 6, 2024

ENGINEERING SCIENCES

Mechanics

RESEARCH ON IMPROVING SEALS TO SUPPRESS VIBRATION OF ROTARY MACHINES

Lu Congxiang[1], Serhii Shevchenko[2], Volodymyr Geichuk[3], Mykola Korchak[4], Andrii Topalov[5]✉

Received on March 1, 2024
Presented by Ch. Roumenin, Member of BAS, on April 23, 2024

Abstract

There is a constant demand for higher equipment parameters, such as pressure of a sealing medium and shaft rotation speed. However, as the parameters rise it becomes more difficult to ensure hermetization efficiency. Moreover, sealing systems affect the overall operational safety of the equipment, especially vibratory. Non-contact seals are considered hydrostatodynamic supports that can effectively damp rotor oscillations. Models of an impulse and a groove seals, models of rotor-seals system and rotor-auto-unloading system, models of a shaftless pump are studied to evaluate the effect of these sealing systems on oscillatory characteristics of rotor. Analytical dependencies for computation the dynamic characteristics of impulse seals, hydromechanical systems rotor-seals and rotor-auto-unloading, as well as shaftless pumps are obtained. These dependencies describe the radial-angular vibrations of a centrifugal machine rotor in seals-supports. Equations for computation the amplitude-frequency characteristics are given. The directions of improving the environmental safety of critical pumping equipment by purposefully increasing the rigidity of non-contact seals that leads to higher rotor vibration stability have been determined.

Key words: impulse seals, groove seals, auto unloading device, sealssupports, mathematical model, radial-angular vibrations, frequency characteristics

---

This study was supported by the National Foreign Expert Program (Grant No. G2022014114L).

`https://doi.org/10.7546/CRABS.2024.06.11`

Introduction. Centrifugal pumps and compressors are widely used throughout all industries. There is a steady tendency towards an increase in operating parameters of pumps: capacity, pressures and speeds, i.e. to the higher concentration of parameters in a single unit. The strengthened rotor is one of the most complex and critical points that determines the reliability of the entire unit. This is due to the strict working conditions of seals combined with high requirements for tightness in all operating modes. For contact seals, the development of the chemical industry is a special issue [1,2], since the chemical composition of contact seals is taken into account depending on their design [3].

Thermal and nuclear power stations constantly require an increase in parameters of feed, main circulation and other pumps. This need stimulated a detailed study of hydrodynamic processes in non-contact seals and their influence on the vibration state of centrifugal machines rotors [4]. Designing turbopump units for high-power rocket engines for reusable spacecraft have further raised the researchers' interest in the dynamic characteristics of non-contact seals and rotor vibrations in seals [5]. The effect of the pumping medium is especially significant with large gradients of velocities and pressures. These conditions are typical for small gaps of non-contact seals, on which large pressure is throttled, and one of the surfaces belongs to a rotor that rotates and vibrates. Therefore, when modelling groove seals, besides their designated purpose – to reduce volume losses, we need to ensure the required vibration characteristics of rotor, which is also a very important function.

Noncontact seals, on which a huge pressure is throttled, can play the role of static, and with the right design approaches, dynamic supports. This must be considered when designing critical power equipment in order to improve its environmental safety.

Current approaches for refinement of mathematical models of oscillatory systems according to experimental data are presented in [6, 7]. The monograph [6] evaluates coefficients of mathematical models for oscillatory systems, which includes rotary systems for multistage centrifugal machines. Pavlenko et al. [7] address the phenomena of rotor rotation stability loss at rolling bearing.

Modern approaches in the field of linear and non-linear rotor dynamics and their practical applications are presented in [8–12]. Yashchenko et al. [8] provide an assessment of segment bearing stiffness with the balancing procedure for flexible rotors of turbocharge units in the accelerating-balancing stand. Current methods for determination of active magnetic bearings stiffness and damping identification from frequency characteristics of control systems were introduced in [9]. Application of the finite element analysis for stiffness and critical speed calculation of a magnetic bearing-rotor system for electrical machines was described in [10]. A phenomenon of subharmonic resonance of a symmetric ball bearing-rotor system is investigated in [11]. Pavlenko et al. [12] propose models for investigation of critical frequencies of the centrifugal compressor rotor that considers non-linear stiffness characteristics of bearings and seals.

As indicated in [13], energy of volumetric losses can be converted into net energy, if the groove seals are used simultaneously as hydrostatic bearings, that are able to have not only high radial rigidity but also to effectively damp the rotor fluctuations to the acceptable level even if there is a significant imbalance. This effect is especially considerable if there are existing steep velocity and pressure gradients, which are peculiar to close gaps of the groove seals, on which high pressure differentials are choked and one of the surfaces belongs to rotor that both rotates and vibrates [14]. The dynamic characteristics of groove seals as intermediate supports have been studied in [15]. However, the problems of rotor dynamics in groove seals are slightly neglected as to solve them it is necessary to account for the hydrodynamic characteristics of groove seals. And this is a separate problem in the hydrodynamics of three-dimensional unsteady viscous fluid flows in annular channels, whereof surfaces rotate and simultaneously perform radial-angular oscillations [16]. Since the problems of the rotor dynamics without groove seals have been mainly solved, this paper focuses more on the analysis of oscillatory processes caused by the hydrodynamic characteristics of seals.

In monograph [4], a model of a slot seal was considered, which is an annular throttle formed by an inner cylinder (shaft) with a small cone angle ϑ_A and an outer cylinder (bushing) with a cone angle ϑ_B; total channel taper angle $\vartheta_0 = \vartheta_B - \vartheta_A$. Channel taper parameter $\theta_0 = \vartheta_0 l/2H$, $|\theta_0| \leq 1$. Shaft and bushing rotate around their own axes with the frequencies of their own rotation ω_1, ω_2. The axes themselves rotate around the fixed centre O with precession frequencies Ω_1, Ω_2, and also perform radial and angular oscillations. Thus, when developing groove seals, it is necessary to consider not only their direct purpose to reduce volumetric losses, but also their equally important function, which is to provide the necessary vibration characteristics of the rotor.

Martsinkovsky [16] has provided an assessment of the force characteristics for laminar and turbulent flow regimes taking into consideration local resistances and in view of flow swirl at the gap inlet.

Material and methods. ***Impulse seal model.*** Impulse seals with self-adjusting clearance have several undeniable advantages over conventional mechanical seals [2]. In impulse seals, as rotational speed rises, the face clearance increases, meaning that increase in energy losses due to friction is negligent. Therefore, impulse seals are especially effective in high-speed machines.

There is negative feedback between the face clearance z (adjustable value) and the force F_s (control response), which ensures self-regulation of the face clearance (Fig. 1).

The operation of impulse compaction is accompanied by complex unsteady hydrodynamic processes, the mathematical description of which is complicated. Success in the analytical solution of the dynamic computation problem depends on the correct choice of a simplified computational model of compaction.

The simplifications can only be justified by the results of experimental studies [17].

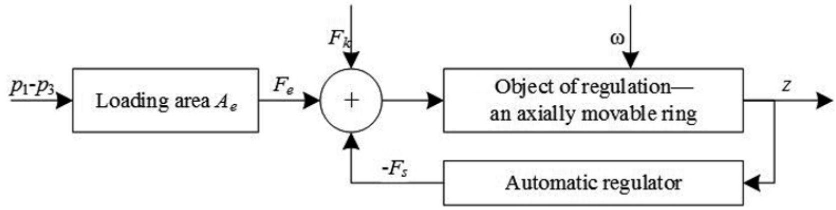

Fig. 1. The model of impulse seal as the automatic control system

Groove seals. A simplified model of the rotor-groove seals system is shown in Fig. 2. Radial (x, y) and angular (ϑ_x, ϑ_y) oscillations of the rotor are largely determined by hydrodynamic forces (F) and moments (M) arising in the sealing gaps (in annular throttles), and the very forces and moments depend on the nature of rotor movement.

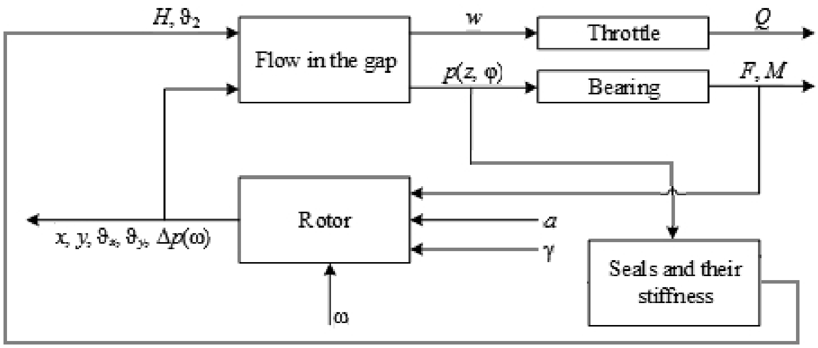

Fig. 2. Model of the hydromechanical rotor-groove seal system

There is another feedback between geometric shape of the gap (average radial gap H and taper ϑ_2) and pressure in the gap $p(z, \phi)$: deformations of the sealing rings are determined by pressure distribution, and the latter is very sensitive to changes in size and shape of the gap.

Automatic balancing devices as sealing systems. Automatic balancing devices have many different designs [18], however, there is a general principle: negative feedback is created between the balancing force and the axial position of the rotor, providing only small deviations of the axial position of the rotor from some predetermined position (Fig. 3). The groove seals of the rotors have clearances of the same order as journal bearings. Therefore, the seal is a full-grip bearing, the bearing capacity of which is provided not only by the rotation of the eccentrically located shaft, but also by the significant pressure drop throttled on the seal. The hydrostatic component of the bearing capacity is predominant since it usually exceeds the bending stiffness of the shaft and the stiffness of radial journal bearings. Face groove seals of auto-unloading systems simultaneously serve as thrust hydrostatic bearings.

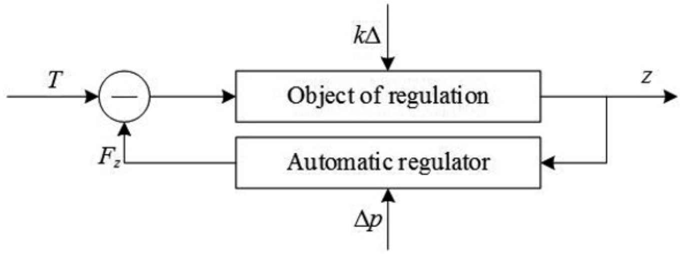

Fig. 3. Model of the balancing device

High sealing pressure results in significant losses of the sealed medium. To reduce them, the gaps are made as small as possible. Therefore, the balancing device additionally functions as a face seal with self-regulating leaks.

Shaftless pumps with seals-bearings. In high-pressure pumps, groove seals, in addition to their main purpose – to limit the crossflows between cavities with different pressures, can be used as rotor supports. Implicitly, these sealing functions have always been implemented in centrifugal pumps. There are recent designs where groove seals are also functioning as only supports [19].

The advantages of shaftless pumps are possible only if there is no contact between the rotating impeller and the stationary casing. In turn, the non-contact mode of operation is determined by the hydrodynamic characteristics of groove seals and axial force auto-unloading system.

Effect of seals on the dynamic characteristics of the rotor systems frequency response. Oscillations of the rotor under the action of gyroscopic forces and moments arising in the seals are described by the equations [20]:

$$a_1\ddot{u} + a_2\dot{u} + a_3u - i(a_4\dot{u}+a_5u)\omega - (\alpha_2\dot{\theta}+\alpha_3\theta)\omega - i(\alpha_4\dot{\theta} + \alpha_5\theta - \alpha_0\theta) = \omega^2\mathrm{A}e^{i\omega t},$$
$$b_1\ddot{\theta} + b_2\dot{\theta} + b_3\theta - i(b_4\dot{\theta} + b_5\theta)\omega + (\beta_2\dot{u} - \beta_3u)\omega - i(\beta_4\dot{u} + \beta_5u + \beta_0u) = \omega^2\Gamma e^{i\omega t}.$$

The notations can be found in [20]. General view of the solution to these equations is

$$u = u_a e^{i(\omega t+\varphi_u)} = \tilde{u}e^{i\omega t}, \quad \theta = \theta_a e^{i(\omega t+\varphi_\vartheta)} = \tilde{\theta}e^{i\omega t},$$

which after substitution leads to a system of algebraic equations for complex amplitudes

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

Using standard programs one can immediately find a numerical solution to these equations. However, the traditional approach produces analytical expressions of amplitudes and phases, which allow to see how different forces and moments affect them.

After switching to dimensionless frequencies and introducing some notation, equations take the form:

$$(U_{11}+iV_{11})\tilde{u}+(U_{12}+iV_{12})\tilde{\theta}=\mathrm{A}\overline{\omega}^2,$$
$$(U_{21}+iV_{21})\tilde{u}+(U_{22}+iV_{22})\tilde{\theta}=\Gamma\overline{\omega}^2.$$

Here $U_{11}+iV_{11}$, $U_{22}+iV_{22}$ are proper operators of the independent radial and angular oscillations, respectively. Cross operators $U_{12}+iV_{12}$, $U_{21}+iV_{21}$ characterize the effect of angular oscillations on radial ones and the reverse effect, i.e., interconnection of these oscillations.

Substituting the values of the determinants, we obtain the amplitudes and phases expressed in terms of external disturbances:

$$u_a=\overline{\omega}^2\sqrt{\frac{(\mathrm{A}U_{22}-\Gamma U_{12})^2+(\mathrm{A}V_{22}-\Gamma V_{12})^2}{U_0^2+V_0^2}},$$
$$\theta_a=\overline{\omega}^2\sqrt{\frac{(\Gamma U_{11}-\mathrm{A}U_{21})^2+(\Gamma V_{11}-\mathrm{A}V_{21})^2}{U_0^2+V_0^2}},$$
$$\varphi_u=-\operatorname{arctg}\frac{(\mathrm{A}U_{22}-\Gamma U_{12})V_0-(\mathrm{A}V_{22}-\Gamma V_{12})U_0}{(\mathrm{A}U_{22}-\Gamma U_{12})U_0+(\mathrm{A}V_{22}-\Gamma V_{12})V_0},$$
$$\varphi_\vartheta=-\operatorname{arctg}\frac{(\Gamma U_{11}-\mathrm{A}U_{21})V_0-(\Gamma V_{11}-\mathrm{A}V_{21})U_0}{(\Gamma U_{11}-\mathrm{A}U_{21})U_0+(\Gamma V_{11}-\mathrm{A}V_{21})V_0}.$$

For impulse seals, the amplitudes and phases are expressed by the formulas [5]:

$$A(\omega)=\frac{u}{\Psi}=\sqrt{\frac{b_1^2+\omega^2b_0^2}{U^2+\omega^2V^2}},\quad \varphi=-\operatorname{arctg}\omega\frac{b_0U-b_1V}{b_1U+\omega^2b_0V}.$$

The corresponding notations can be found in [5]. The amplitude frequency characteristics can be used to estimate the dimensional values of the amplitudes of the forced axial oscillations of the ring at any rotation frequency.

Amplitude and phase frequency characteristics of the rotor-gap seals system for the corresponding external influences:

$$A_\tau(\omega)=\frac{u_{za\tau}}{\tau_a}=\frac{z_{a\tau}\mathrm{A}_0p_n}{H_2T_a}=\sqrt{U_\tau^2+\omega^2V_\tau^2}=\sqrt{\frac{U_p^2+\omega^2V_p^2}{U^2+\omega^2V^2}},$$
$$\varphi_\tau(\omega)=\operatorname{arctg}\omega\frac{V_\tau}{U_\tau}=\operatorname{arctg}\omega\frac{UV_p-VU_p}{UU_p+\omega^2VV_p},$$
$$A_\Psi=\frac{u_\Psi}{\Psi}=\frac{z_{a\Psi}p_n}{H_2p_a}=\sqrt{\frac{U_n^2+\omega^2V_n^2}{U^2+\omega^2V^2}},\quad \varphi_\Psi=\operatorname{arctg}\omega\frac{(UV_n-VU_n)}{UU_n+\omega^2VV_n}.$$

The frequency characteristics of the rotor-balancing device system are calculated in a similar way:

$$A_{r\tau}(\omega) = \frac{u_{ra\tau}}{\tau_a} = K\kappa_s\beta\sqrt{\frac{U_{r\tau}^2+\omega^2V_{r\tau}^2}{U^2+\omega^2V^2}}, \qquad \varphi_{r\tau}(\omega) = \operatorname{arctg}\omega\frac{UV_{r\tau}-VU_{r\tau}}{UU_{r\tau}+\omega^2VV_{r\tau}},$$
$$A_{ra}(\omega) = \frac{r_{aa}}{a} = \omega^2\sqrt{\frac{U_{ra}^2+\omega^2V_{ra}^2}{U^2+\omega^2V^2}}, \qquad \varphi_{ra}(\omega) = \operatorname{arctg}\omega\frac{UV_{ra}-VU_{ra}}{UU_{ra}+\omega^2VV_{ra}};$$

and of shaftless pumps:

$$A_e(\omega) = \frac{u_{ea}}{\overline{A}_e\psi_{ea}} = \sqrt{\frac{U_e^2+\omega^2V_e^2}{U_0^2+\omega^2V_0^2}}, \qquad \varphi_e(\omega) = \operatorname{arctg}\omega\frac{U_0V_e-U_eV_0}{U_0U_e+\omega^2V_0V_e};$$
$$A_\varepsilon(\omega) = \frac{u_{\varepsilon a}}{\overline{A}_2\varepsilon_a} = \frac{k_3}{\sqrt{U_0^2+\omega^2V_0^2}}, \qquad \varphi_\varepsilon(\omega) = -\operatorname{arctg}\omega\frac{V_0}{U_0}.$$

Stability criteria. The stability is determined using the Routh–Hurwitz criterion for a system of 4th order

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

which for the rotor-groove seals system is reduced to the form:

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

Therefore, the main destabilizing factor is the circulating force, characterized by the coefficient a_5. Damping a_{21}, gyroscopic force a_4 and shaft bending stiffness Ω_{u0} stabilize the rotor.

For impulse seal it is possible to determine the chamber volume admissible in stability:

$$V_0 < \frac{A_sEz_0g_{s0}}{3(1+n_i)(k_1g_{30}-k_3g_{10})(p_{10}-p_{30})}.$$

Therefore, stability region of the seal expands due to a decrease in the volume of the chambers and in the coefficient of hydrostatic stiffness. For balancing device

$$\left(\frac{V}{A_eH_2}\right) < \frac{Eg_{sm}^2u_{z0}}{3Q_{0m}^2}.$$

The last inequality limits the volume of the hydraulic prop chamber, at which the stability of independent axial oscillations of the rotor is maintained. For a shaftless pump, the axial stability condition is reduced to the inequality

$$H < \frac{Ez_0}{3p_n}\cdot\frac{\Delta\psi_{s0}}{\Delta\psi_{20}\Delta\psi_{c0}}.$$

The design of the chamber can be easily changed since its depth is an independent parameter. Therefore, the last condition can be used for pump design to ensure its stability.

Results and discussions. Examples of calculations of the rotor frequency characteristics in gap seals are shown in Fig. 4.

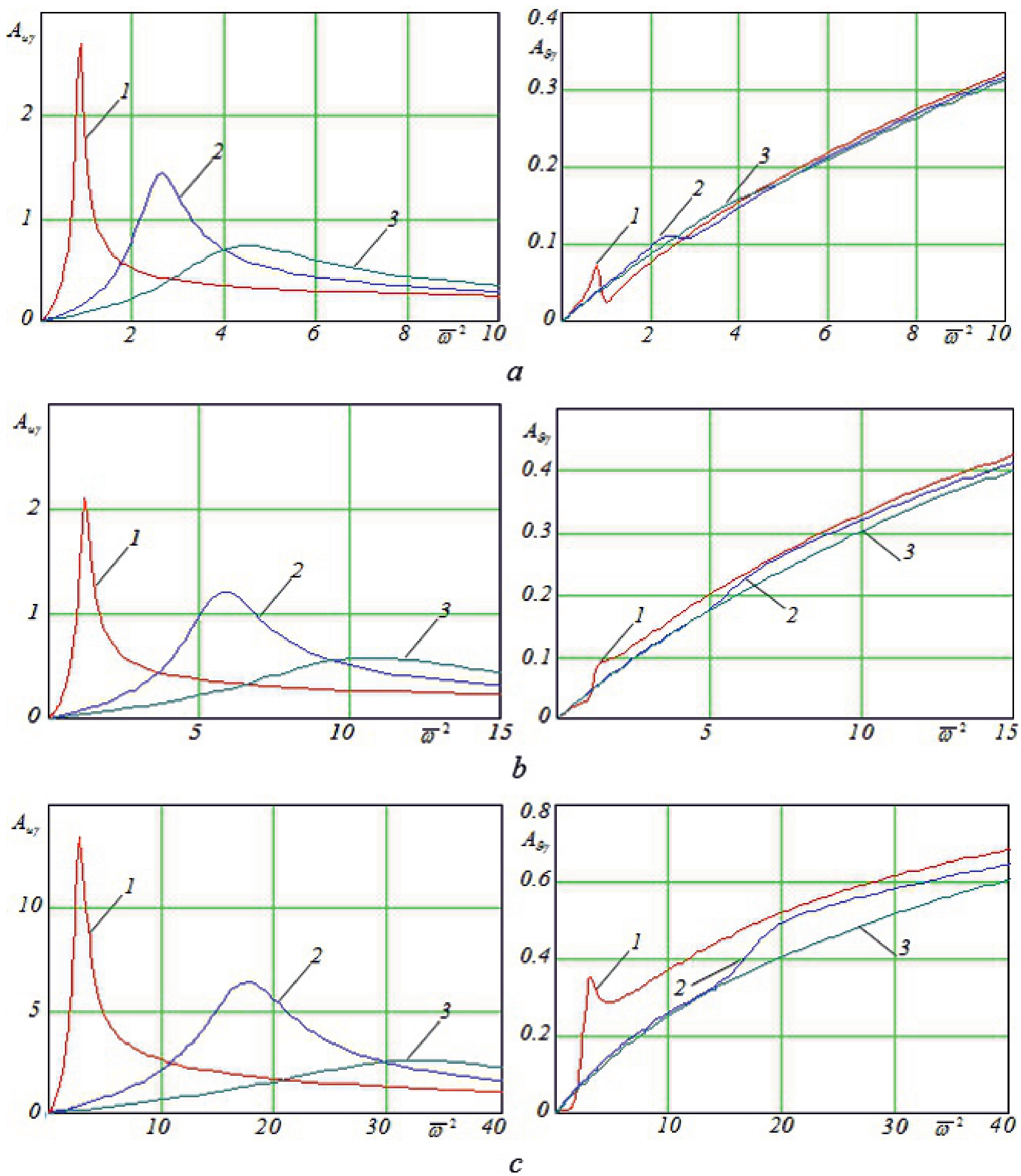

Fig. 4. Amplitude frequency characteristics as a response to dynamic unbalance:
a – $\Delta p_0 = 1.5$ MPa = const, b – $\Delta p_0 = 4$ MPa = const, c – $\Delta p_0 = 13.3$ MPa = const

The gap seals with three taper parameters were considered: $\theta_0 = -0.3; 0; 0.3$. Diagrams for these parameters in Fig. 4 are designated by numbers 1, 2, and 3, respectively. For a diffuser channel ($\theta_0 = -0.3$), the total resistance is negative, and the imaginary part is positive. As a result, the phase characteristic is positive. For cylindrical ($\theta_0 = 0$) and confusor ($\theta_0 = 0.3$) channels, condition is satisfied. In this case, the real part is positive at all rotation frequencies (does not pass through zero), and the phase characteristic does not go beyond the fourth quadrant: $-\pi/2 < \varphi_u \leq 0$. 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.

Conclusions. Based on the study of models of non-contact seals and models of rotor-groove seals-auto unloading systems, analytical dependencies are obtained that describe the effect of non-contact seals on the dynamics of a centrifugal machine. It is shown that a purposeful choice of the design parameters of the seals can influence the vibration state of the rotor. In this case, the initially "flexible" in the dynamic sense rotor combined with correctly designed seals becomes "rigid". This is especially important for machines with high parameters. The studies carried out make it possible to determine the directions of increasing the vibration stability of critical power centrifugal machines and, as a consequence, increasing environmental safety.

REFERENCES

[1] Korchak M., O. Bragin, O. Petrova et al. (2022) Development of transesterification model for safe technology of chemical modification of oxidized fats, Eastern-European Journal of Enterprise Technologies, **6**(120), 8–13, https://doi.org/10.15587/1729-4061.2022.266931.

[2] Korchak M., O. Bliznjuk, S. Nekrasov et al. (2022) Development of rational technology for sodium glyceroxide obtaining, Eastern-European Journal of Enterprise Technologies, **6**(119), 15–21, https://doi.org/10.15587/1729-4061.2022.265087.

[3] Whalen J. K., J. Allen, J. D. Cardell, J. R. Dugas (2013) Polymer seal use in centrifugal compressors – two users' experiences over 15 years. In: Proc. Second Middle East Turbomachinery Symposium 1720 March 2013, Doha, Qatar.

[4] Martsynkovsky V. A., S. S. Shevchenko (2018) Pumps of Nuclear Power Plants: Calculation, Design, Operation, Sumy, Ukraine, University Book Publ. House, (in Ukrainian).

[5] Shevchenko S., A. Chernov (2020) Development of pulse mechanical seal calculation methods on the basis of its physical model construction, Eastern-European Journal of Enterprise Technologies, **2**(105), 58–69.
[6] Simonovskiy V. I. (2015) Evaluation of coefficients of mathematical models for oscillatory systems, LAP LAMBERT Academic Publishing, Saarbrücken.
[7] Pavlenko I., V. I. Simonovskiy, J. Pitel et al. (2018) Dynamic analysis of centrifugal machines rotors with combined using 3D and 2D finite element models, RAM-Verlag, Lüdenscheid, ISBN: 978-3-942303-64-4.
[8] Yashchenko A. S., A. A. Rudenko, V. I. Simonovskiy, O. M. Kozlov (2017) Effect of bearing housings on centrifugal pump rotor dynamics. In: IOP Conf. Ser.: Materials Science and Engineering, IOP Publishing, https://doi.org/10.1088/1757-899X/233/1/012054.
[9] Jin C., Y. Xu, J. Zhou et al. (2016) Active Magnetic Bearings Stiffness and Damping Identification from Frequency Characteristics of Control Systems, Hindawi Publ. Corp., Cairo, Published 2015.
[10] Zhang K., Z. Yang (2017) Identification of load categories in rotor system based on vibration analysis, Sensors, **17**(7), 1–16.
[11] Pozovnyi O., A. Zahorulko, J. Krmela et al. (2020) Calculation of the characteristics of the multi-gap seal of the centrifugal pump, in dependence on the chambers' sizes, Manufacturing Technology, **20**(3), 361–367.
[12] Pavlenko I. V., V. I. Simonovskiy, J. Pitel et al. (2017) Investigation of critical frequencies of the centrifugal compressor rotor with taking into account stiffness of bearings and seals, J. Engin. Sci., **4**(1), 1–6.
[13] Martsynovskyi V. A. (2005) Groove Seals: Theory and Practice, Sumy, Sumy State University Publ. House, (in Ukrainian).
[14] Kundera Cz., W. Marcinkowski (2010) The effect of the annular seal parameters on the dynamics of the rotor system, J. Appl. Mech. Engin., **15**(3), 719–730.
[15] Djaidir B., A. Hafaifa, A. Kouzou (2017) Faults detection in gas turbine rotor using vibration analysis under varying conditions, J. Theor. Appl. Mech., **21**(3), 393–403.
[16] Martsinkovsky V. A. (2012) Dynamics of Rotors of Centrifugal Machines, Sumy, SSU Publ. House, (in Ukrainian).
[17] Martsinkovsky V., A. Zahorulko, S. Gudkov, S. Mischenko (2012) Analysis of buffer impulse seal, Procedia Engineering, **39**, 43–50.
[18] Martsinkovsky V., A. Zhulyov, Cz. Kundera (2014) Static and dynamic analysis of a pump impeller with a balancing device. Part 2: Dynamic analysis, J. Appl. Mech. Engin., **19**(3), 621–631.
[19] Gorovoy S. A. (2019) Experimental studies of the pump with selfaligning impeller, J. Chem. Oil Gas Mech. Engin., **2**, 36–40.
[20] Shevchenko S., O. Shevchenko, S. Vynnychuk (2021) Mathematical modelling of dynamic system rotor-groove seals for the purposes of increasing the vibration reliability of NPP pumps, Nuclear and Radiation Safety, **1**(89), 80–87.

[1]*School of Intelligent Manufacturing, Yancheng Polytechnic College,*
224005 Yancheng, China
e-mail: lucongxiang@126.com

[2]*Department of Simulation of Energy Processes and Systems,
Pukhov Institute for Modelling in Energy Engineering,
03164 Kyiv, Ukraine*
e-mail: shevchenkoss@nas.gov.ua

[3]*Science Department, National Technical University of Ukraine
"Kyiv Polytechnic Institute", 03056 Kyiv, Ukraine*
e-mail: poroshinv@gmail.com

[4]*Department of Agricultural Engineering and Systems Engineering Names Mykhaila
Samokisha, Higher Educational Institution "Podillia State University",
32316 Kamianets-Podilskyi, Ukraine*
e-mail: nikolaykorchak@gmail.com

[5]*Department of Computerized Control Systems,
Admiral Makarov National University of Shipbuilding,
54025 Mykolaiv, Ukraine*
e-mail: topalov_ua@ukr.net