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Research Article

Assessment of sealing systems impact on the vibration and environmental safety of rotary machines

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Abstract: Energy saving control algorithms of centrifugal fans/pumps are based on the use of the frequency-controlled induction motor drives and pressure or flow rate sensors, the costs of which are comparable to the cost of the fans/pumps for low-power applications. The paper develops a new and simple estimation approach of the pressure and flow rate, utilising the measured Root Mean Square (RMS) value of the stator current, estimated motor's input active power, reference stator voltage frequency and feed-forward backpropagation artificial neural network. The error percentage for both flow rate and pressure in experimental and estimated data is within the range of ±5%, which conforms to the ISO 13348 standard. A test rig for the rapid control prototyping of the fan is designed, and necessary design and test procedures are developed. The estimation approach is verified experimentally and demonstrates better estimation accuracy compared to the existing and possible similar simple approaches. The developed algorithm can be easily embedded into the industrial variable frequency drives without any hardware changes.

Keywords: *Balancing, Bearing seal, Environmental safety, Non-contact, Vibration*

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Symbols and Units

a	Eccentricity of mass center	m
a_{ij}	Parameters of the gap seals	-
H	Mean radial clearance of gap seal	m
M	Hydrodynamic moments arising in sealing gap	Nm
$p(z, \varphi)$	Gap pressure	Pa, N/m^2
p_n	Nominal discharge pressure	Pa, N/m^2
Q	Seal leakage	m^3/h
$u, \tilde{u}$	Radial component of amplitude and complex amplitude of forced vibrations	
U_i, Vi	Real and imaginary parts of differential operators	
w	Gap fluid flow rate	m/s
x, y	Radial vibrations of the rotor	m
α_i	Coefficients of hydrodynamic forces depending on the angular oscillations of the rotor	
β_i	Coefficients of hydrodynamic moments depending on the radial oscillations of the rotor	
$\theta, \tilde{\theta}$	Angular component of the amplitude and complex amplitude of forced oscillations	
θ_0, θ_{0*}	Taper parameter of the annular channel and its critical value	
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

1. INTRODUCTION

The rotary machines include centrifugal pumps and compressors as the most widespread type of machinery. They occupy practically all the branches in machine building as well as stationery and transport energetics: Railway, in particular for feeding exhaust boilers aboard ship, where the issues of vibration and tightness are of great priority [1,2]. The rotary machines are desired for transporting liquids as heated [3] and cooled water [4] at thermal power plants [5] and compressed gaseous media, for instance in gas transportation industry driven by gas turbines (GT) [6,7] equipped by cooling towers and other heat exchangers [8] or to increase the pressure of working fluids [9] for jet devices [10] such as ejectors [11] or injectors [12,13] in refrigeration and air conditioning [14]. Their application is quite efficient in extended two or three intermediate charged air-cooling contours of marine combustion engines [15], while cooling sucked air in internal combustion engines (ICE) and gas turbines (GT) [16]. The centrifugal pumps are widely applied in integrated energy systems for combined power, heat and cooling (CPHC) [17], or trigeneration, including so called in-cycle trigeneration with using the refrigeration for cooling engine cyclic air [18], so as combined energy production is much more efficient compared to its distributed generation due to extended heat utilization with exhaust boilers utilizing the heat of exhaust gas and charged air from combustion engines. The heat of hot water, circulated in cogenerative circuit of combustion engines by centrifugal pumps, are converted into refrigeration by waste heat recovery chillers [19].

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 and power plant as the whole. This is due to the harsh operating conditions of the seals, combined with high tightness requirements in all operating modes [20]. The demand for centrifugal machines with high parameters for thermal and nuclear power is constantly growing. Therefore, ensuring their tightness and vibration reliability has become a real technical problem [21]. The creation of highly loaded equipment with sealing systems for the non-standard operating conditions is impossible without taking into account the effect of seals on the vibration characteristics of the equipment [22]. Pumps for the power industry need an increased resource of sealing units. To satisfy this, it is necessary to use sealing systems with the guaranteed controlled leakage in order to provide lubrication and cooling, and, consequently, the required service life [23,24]. The seals of LRE 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 [25]. 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 [26]. The creation of highly loaded equipment with sealing systems for non-standard operating conditions is impossible without taking into account the effect of seals on the vibration characteristics of the equipment [27]. When designing sealing systems, it is necessary to take into account the amount of leakage of the sealing medium due to the inevitable gaps between the rotating, vibrating shaft and the stationary body, friction losses and vibrations that may occur. In order to achieve an acceptable balance between fluid losses through the seals, frictional force in the seals and vibrations, multi-link hydromechanical systems with a several sealing stages should be used. For their creation, it is important to develop a methodological base and models of various types of seals. On this basis the characteristics of medium losses, friction forces and vibration characteristics can be obtained. Moreover, it is necessary to consider the peculiarities of their combination as part of complex sealing systems in order to achieve balance between sealing and vibration reliability, taking into account the hydrodynamic characteristics of seals [20,21,22,23].

When designing the non-contact seals on which high pressure is throttled, in addition to sealing, a very important function is to provide vibration characteristics [24,25,26]. Therefore, while designing critical centrifugal machines in order to improve their vibration and environmental safety, it is necessary to design non-contact seals that also act as dynamic bearings [27,28]. 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 are throttled and one of the surfaces belongs to the rotor, which simultaneously rotates and vibrates. In Refs. [29,30,31], the dynamic characteristics of slotted seals as intermediate supports were studied. The hydrodynamic characteristics of slotted 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 Ref. [32]. 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 [33,34]. As shown by the literature review, there was an acute need to determine ways to improve existing methods for designing and calculating sealing systems by developing common approaches for creating models of non-contact seals and complex sealing systems.

The present paper is devoted to the creation of a general algorithm for constructing the dynamic characteristics of non-contact seals, due to their hydrodynamic properties. Therefore, objectives are:

1. The creation of a unified methodology for assessing the impact of the design characteristics of seals on their dynamic characteristics,

2. The determination of ways for improving the vibration characteristics of the rotor of a centrifugal machine due to a purposeful change in the parameters of sealing units.

2. RESEARCH METHODOLOGY

2.1. Gap Seals

A rotor-gap seals system [28] is shown in Fig. 1.

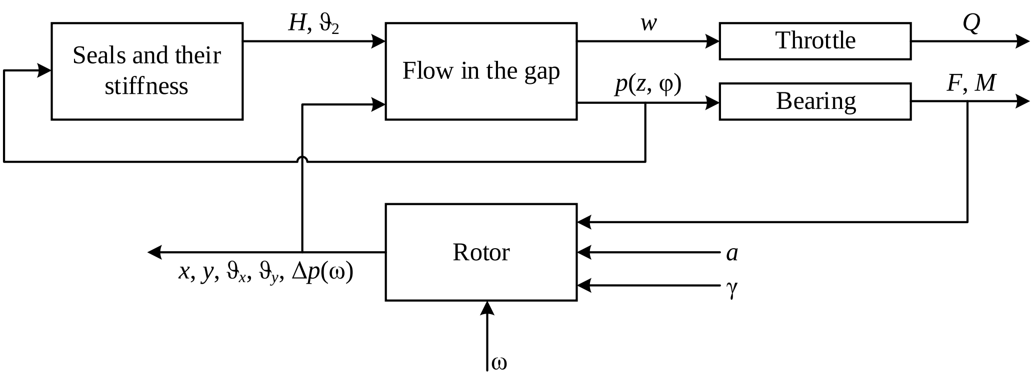


Figure 1. Hydromechanical rotor-gap seal system.

Hydrodynamic forces (F) and moments (M) act from the side of the sealed liquid on the rotor and affect its radial (x, y) and angular ($\vartheta\mathbf{x}$, ϑy) vibrations. In turn, the radial-angular movements of the rotor change forces as well as moments taking place inside sealing gaps. Also, deformations of the sealing rings lead to the shape change of the gap and redistributing the pressure $p(z,\varphi)$. The latter, in turn, changes its geometric shapes (average radial gap H and taper ϑ_2).

As can be seen from the constructed model, the design of non-contact seals has a significant impact on the vibration characteristics of the rotor.

2.2. Impulse Seals

The impulse seals (Fig. 2) are mechanical seals with a self-adjusting gap, so they especially effective for high-speed pump applications [24].

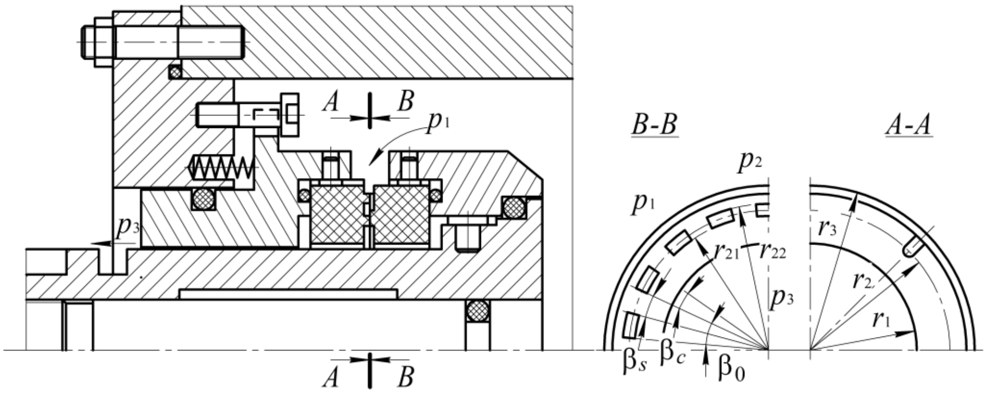


Figure 2. The impulse seal.

The self-adjustment of the end gap is provided by negative feedback between the force F_s and the end gap z (Fig. 3).

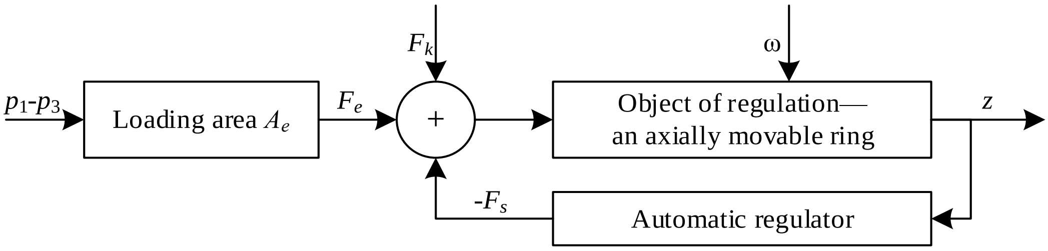


Figure 3. The model of the seal impulse type.

Optimum end clearance and friction torque on the sealing contact surfaces can be achieved by targeted selection of the main sealing geometries.

2.3. Balancing Face as a Sealing Device

Automatic balancing devices are widely used to balance the axial forces that occur in large multistage high-pressure pumps [29]. The rotor of a centrifugal pump with an auto-unloading system has freedom of axial movement within the end gap of the hydraulic heel. In the simplest case, the rotor, as an absolutely rigid body, performs one-dimensional axial oscillations, the characteristics of which are determined by the parameters of the balancing system.

Automatic balancing devices provide small movements of the rotor from a given position by creating negative feedback between the device parameters (Fig. 4) [30].

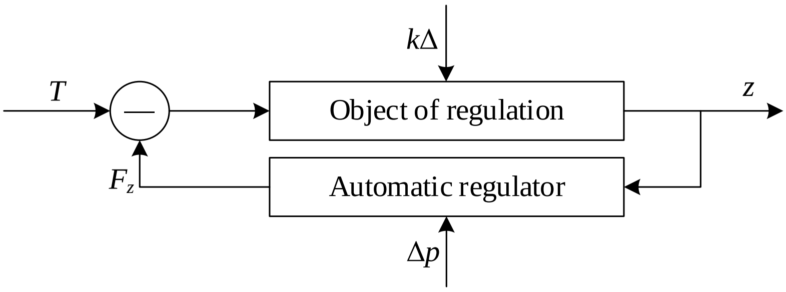


Figure 4. The diagram of the balancing face.

To reduce the leakage of the sealed liquid, the gaps of the gap seals are made as small as possible. Thus, the mechanical seal of the balancing face additionally performs the functions of a seal with self-adjusting leaks.

2.4. Bearing Seals

In high-pressure pumps, non-contact seals have always implicitly performed the role of rotor supports, in addition to their main purpose – sealing high-pressure cavities [31]. Recently, designs of so-called shaftless pumps have appeared (Fig. 5), in which seals replace supports [32].

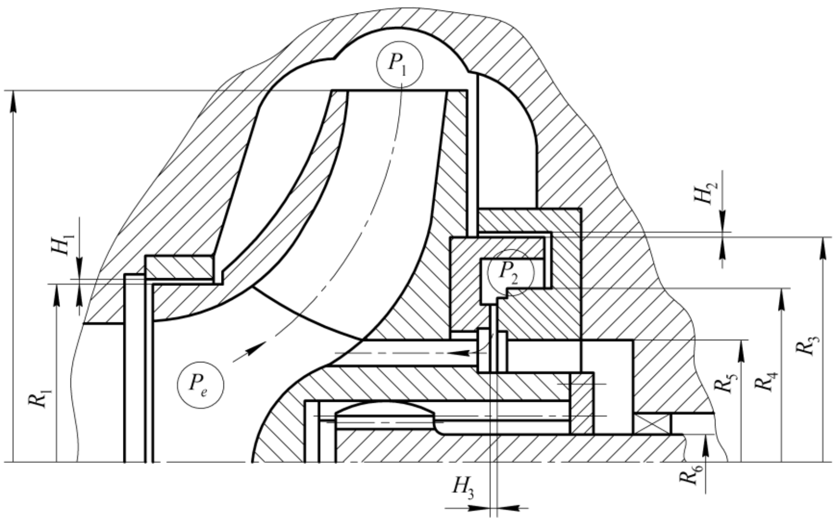


Figure 5. The scheme of a shaftless pump.

The non-contact operational mode of a shaftless pump is ensured by the hydrodynamic characteristics of the system for automatic unloading of axial forces and slotted seals. Therefore, it is necessary to take into account the dynamics of its impeller, that is freely floating in the slotted seals. To describe these processes, the above models can be used together.

The impeller or rotor of a shaftless pump performs joint radial, angular and axial vibrations. The relationship between radial and angular vibrations is due to hydrodynamic moments that occur in gap seals. Radial and axial oscillations are connected by the dependence of the conductivities of the annular chokes on the eccentricity, i.e., from the radial displacement of the rotor axis relative to the housing axis.

3. RESULTS AND DISCUSSIONS

We write down the equations for calculating the frequency characteristics obtained by converting the equations of the rotor oscillations under the action of force factors acting from the side of the seals according to Ref. [33]:

$$A_\tau(\omega) = \frac{u_{za\tau}}{\tau_a} = \frac{z_{a\tau} A_0 p_n}{H_2 T_a} == \sqrt{\frac{U_p^2 + \omega^2 V_p^2}{U^2 + \omega^2 V^2}},$$

$$\phi_\tau(\omega) = \operatorname{arctg} \omega \frac{V_\tau}{U_\tau} = \operatorname{arctg} \omega \frac{UV_p - VU_p}{UU_p + \omega^2 VV_p}.$$

For impulse seals:

$$A\,(\omega)=\frac{u}{\psi}=\sqrt{U_1^2+\omega^2V_1^2},\phi=-\mathrm{arctg}\,\omega\frac{b_0U-b_1V}{b_1U+\omega^2b_0V},$$

The frequency characteristics of the balancing face:

$$A_{ra}(\omega)=\frac{r_{aa}}{a}=\omega^2\sqrt{\frac{U_{ra}^2+\omega^2V_{ra}^2}{U^2+\omega^2V^2}},$$

$$\phi_{ra}(\omega)=\mathrm{arctg}\,\omega\frac{UV_{ra}-VU_{ra}}{UU_{ra}+\omega^2VV_{ra}};$$

and of the bearing seals:

$$A_e(\omega)=\frac{u_{ea}}{A_e\psi_{ea}}=\sqrt{\frac{U_e^2+\omega^2V_e^2}{U_0^2+\omega^2V_0^2}},$$

$$\phi_e(\omega)=\mathrm{arctg}\,\omega\frac{U_0V_e-U_eV_0}{U_0U_e+\omega^2V_0V_e}.$$

Using the Routh-Hurwitz criterion, we write down the inequality that determines the dynamic stability condition:

$$\omega_u^2<\frac{a_{21}^2\Omega_{u0}^2}{a_1a_5^2-a_{21}^2a_{31}-a_{21}a_4a_5} \tag{1}$$

The statement in Eq. (1) shows that the circulation force, characterized by a coefficient a_5, destabilizes the rotor. Damping a_{21}, gyroscopic force a_4 as well as shaft bending stiffness Ω_{u0} influence upon dynamic stability of the rotor. Permissible chamber volume for stable operation of impulse seal is:

$$V_0<\frac{A_sEz_0g_{s0}}{3(1+n_i)(k_1g_{30}-k_3g_{10})(p_{10}-p_{30})}. \tag{2}$$

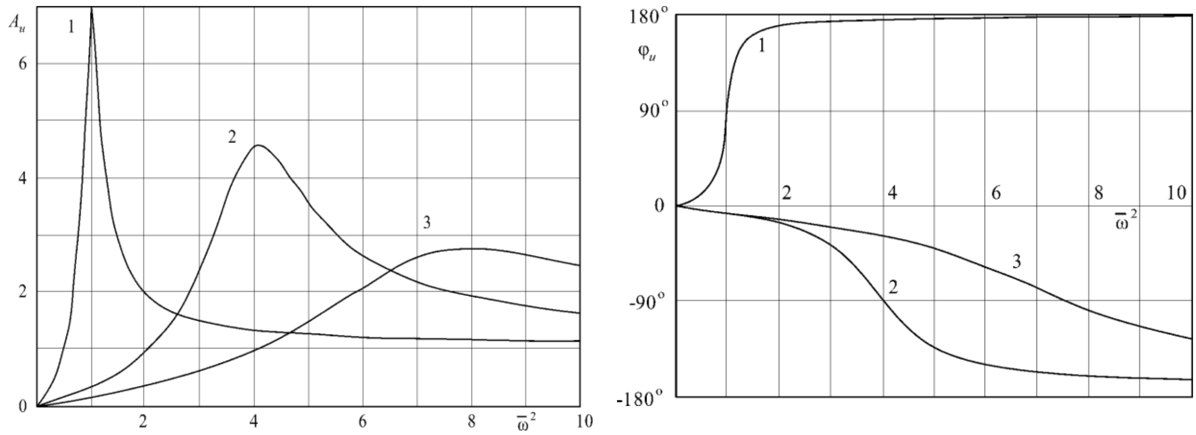
A decrease in the volume of the chamber and hydrostatic stiffness coefficient makes it possible to expand the stability region of the impulse type seal. For the balancing face,

$$\left(\frac{\mathrm{V}}{\mathrm{A_e}H_2}\right)<\frac{Eg_{sm}^2u_{z0}}{3Q_{0m}^2} \tag{3}$$

Eq. (3) shows the influence of the geometrical parameters of the hydro-heel on the stability of the rotor oscillations. Similarly, we write the inequality describing the condition of axial stability of a shaftless pump,

$$H<\frac{Ez_0}{3p_n}\frac{\Delta\psi_{s0}}{\Delta\psi_{20}\Delta\psi_{c0}} \tag{4}$$

Using the condition in Eq. (4), it is possible to ensure the stability of a shaftless pump with bearing seals by changing the depth of the chamber. Examples of calculations of the rotor frequency characteristics in gap seals are shown in Figs. 6(a,b) and 7(a,b). The calculations have been performed for three parameters of the taper of gaps in seals.

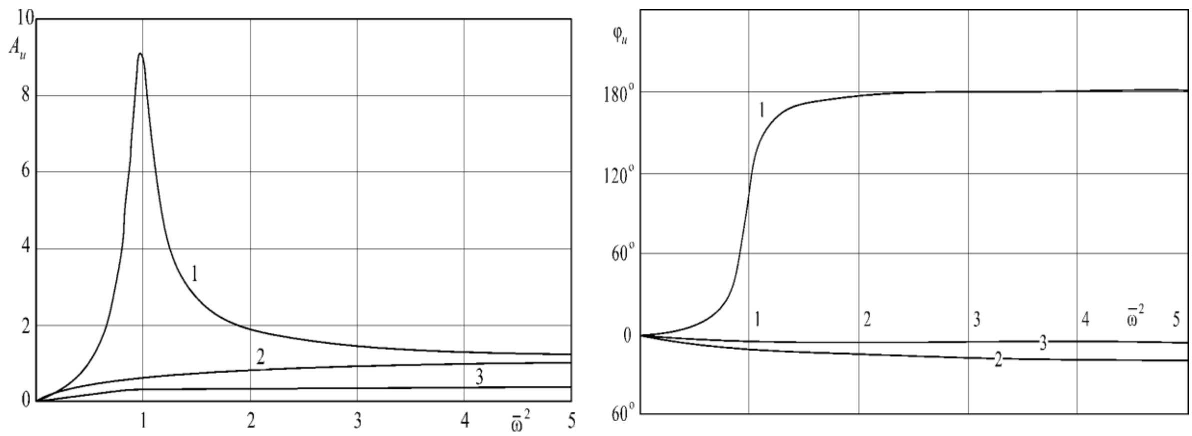


(a) (b)

Figure 6. Frequency characteristics for (a) Amplitude, and (b) phase for the tapers 1 – θ_0=-0.3; 2 – θ_0=0; 3 – θ_0=0.3.

For the case when the sealing pressure $\Delta p_0 \sim \omega^2$, the frequency characteristics have the form (Fig. 7(a,b)). Comparing the results of frequency calculations according to the expressions obtained with the data of experimental studies [20,26,33,34] showed that the calculations do not exceed 5%.

The conducted studies have shown that changing the design parameters of gap seals makes it possible to influence the vibration parameters of a rotary machine. It is important to note that the pressure drops throttled at the slotted seals are proportional to the rotor speed.



(a) (b)

Figure 7. (a) Amplitude and (b) phase frequency characteristics for the tapers 1 – θ_0=-0.3; 2 – θ_0=0; 3 – θ_0=0.3.

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. In seals with cylindrical and confusor gaps, only the first critical rotation speed occurs.

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.

Thus, when studying non-contact seals, they should be considered as automatic control systems that affect the vibration characteristics of the centrifugal machine rotor. It is possible to improve the vibrational characteristics of the rotor by increasing the hydraulic resistance in the sealing gaps. This not only reduces the leakage of the sealed liquid through the seals, but also increases the dynamic rigidity of the rotor.

Fig. 8 shows the sealing system of the Nuclear Power Plant (NPP) main circulation pump [21]. 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 MPa – 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.

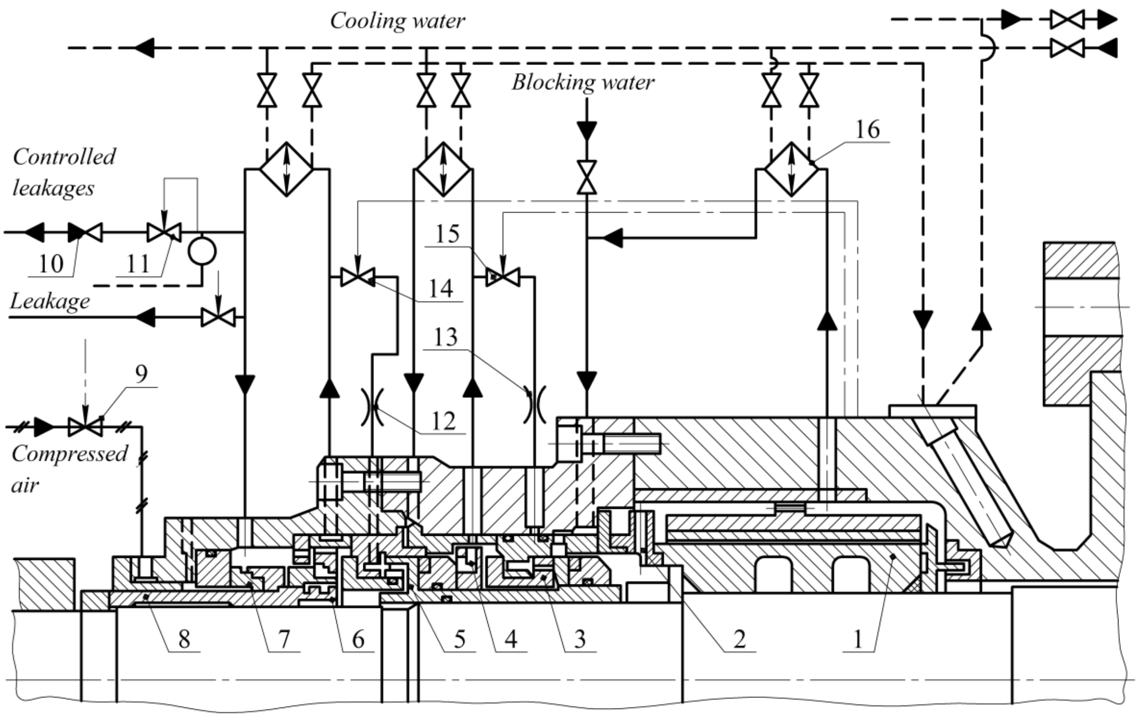


Figure 8. Diagram of RCP shaft seal after Ref. [21].

The seal block, together with the plain bearing 1, running on water, is separated from the pump housing by a special thermal barrier - a water-cooled neck. The impeller 2 pumps the shut-off water through the bearing chamber and cooler 16 to prevent local boiling. The same function is performed by impellers 4 and 6, located in the first 3 and second 5 stages of the hydrostatic seal. A pressure of 0.42–0.45 MPa is maintained in front of the closing mechanical seal 7 by the bypass valve 10. External leaks through seal 7 are about 300 cm^3/h, organized leaks are 0.3 m^3/h. Leaks through hydrostatic seals 3 and 5, and hence the end clearance, are kept constant by changing the conductivity of external throttles 13 and 12 by control valves 15 and 14.

If the supply of blocking water stops or its leakage through the damaged seal increases, the temperature in the bearing chamber rises, and when 65°C is reached, the magnetic valve 15 begins to close, reducing the end gap in the seal 3. With a further increase in temperature to 70°C, the seal 3 closes completely and works as a contact mechanical seal with minimal leakage. The blocking water is brought into the cavity before the seal from an automatically activated back-up system and reduces the temperature in the bearing chamber. If the desired effect is not achieved, then when the temperature rises to 80°C, the second stage of sealing 5 closes.

In the event of a failure of both seal stages 3 and 5, the pressure in front of the closing mechanical seal 7 and valve 10 becomes more than permissible and leads to the closing of valve 1 and the opening of valve 9, through which compressed air is supplied to the labyrinth seal chamber 8. In this case, seal 7 should briefly hold the full differential pressure before the pump stops. If it also fails, the role of the seal is performed by the labyrinth sleeve 8 with compressed air supplied to it under a pressure of 0.7 MPa. All the above measures should prevent the leakage of radioactive water during the 3–10 minutes that are necessary for the normal shutdown of the unit.

4. CONCLUSION

The influence of the non-contact type seals upon the rotor dynamics in centrifugal machines has been studied. The conducted studies have shown that all sealing units with throttling gaps or sealing paths filled with a high-pressure sealed medium should be considered as dynamic systems. The sealed medium affects the dynamic state of the rotor by acting on the walls of the sealing paths. The greater the hydraulic resistance we create in the sealing gaps (for example, due to convergency), the more energy is spent to overcome this resistance by the sealed medium, the more the sealing units toughen the rotor in a dynamic sense, improving its vibrational characteristics.

The studies have shown that by purposeful selection of seal parameters, it is possible to reduce the amplitude of forced oscillations of the rotor by 3-4 times. Thus, 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.

Using the proposed general approach to the analysis of non-contact seals as automatic control systems, and the algorithm for constructing their dynamic characteristics at a design stage, it is possible, by changing the geometric parameters of the seals, to ensure their vibrational resistance margin. Formulas are obtained for calculating the frequency characteristics, as well as the stability limits for slotted seals, impulse seals, balancing face and bearing seals. Thus, when developing non-contact seals, it is necessary to take into account not only their direct purpose as reducing medium losses, but also another important function, which is to provide the necessary vibration characteristics.

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