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A STUDY OF MECHANICAL SEAL RINGS THERMAL AND FORCE DEFORMATIONS IN ENERGY ROTOR MACHINES

Weidong Hui[1], Serhii Shevchenko[2], Mykola Korchak[3], Andrei Lesko[4], Andrii Monakhov[5], Yurii Halynkin[5], Oleksandr Gerasin[6]✉

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Abstract

The operating experience of mechanical seals shows that as a result of angular deformations of the rings, wear of the contact surfaces along radius is uneven. In the area of hydrodynamic load support in the case of a confusor joint or a converging film, the slope of the pressure diagram is such that a decrease in the film thickness increases the hydrodynamic support. This enables stable non-contact sealing operation. When designing mechanical seals, it is necessary to calculate the deformations of the sealing rings caused by both the pressure of the medium being sealed and thermal effects in order to predict the size and direction of the radial cone and avoid unstable operation of the seal. The execution of the rubbing surfaces of the interface of the end pair should ensure the formation of the calculated confusor shape of the sealing gap in all modes of operation.

Key words: vibration, mechanical seal, thermal deformations, vibration characteristics, confusor, rotor machines

Introduction. The higher the load level of energy facilities and their power systems, the greater the thermal and mechanical loads that cause deformation of

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the sealing units of rotary machines [1, 2], which creates problems in improving the theoretical basis of their design.

Due to the increase in energy consumption and other parameters of centrifugal machines, their reliability, and therefore safety, is determined by shaft sealing systems [3], and the development of the chemical industry [4, 5] affects the manufacture of new seals.

The main difficulties in creating equipment with high parameters are associated with the calculation and design of components of centrifugal machines [6–8] and payments in particular [9]. These problems are complicated by the fact that the rotor, together with the shaft seals, represents a single closed hydromechanical system, the performance of which is determined by the properties and interaction of all its elements [10].

Despite the apparent simplicity of assembling the mechanical seal, the processes that occur when connecting the mating end surfaces of two parts (one of which rotates with the rotor) are extremely complex [11]. This is due to the simultaneous influence and connection of friction processes, hydrodynamics and thermal processes, as well as changes in the shape of the mating surfaces in the sealing joint when the load parameters of the seal change [12].

When designing a conventional mechanical seal, the designer needs relationships to estimate the expected seal leakage, the magnitude of the frictional force, and the likely durability of the assembly. These calculated dependencies and estimates can be obtained on the basis of a closed working model containing a mathematical description of the processes occurring in the sealing gap and a given criterion for their optimality [13].

The most significant mechanism for the occurrence of the supporting force in the end connection of the seal is the influence of the deviation of the sliding surfaces from parallelism. The reason for such deviations, according to researchers [14, 15], may be thermal and mechanical deformations of the sealing surfaces that arise during the operation of the mechanical seal, as well as the initial errors of these surfaces, obtained during operation, finishing operations during the manufacture of rings, or assembly of the sealing unit.

Methods. ***Thermal deformations of the friction pair ring.*** The operating experience of mechanical seals shows that as a result of angular deformations of the rings, wear of the contact surfaces along radius is uneven. Deformations in the first approximation can be considered a rotary movement of the ring crosssection without changing its shape and not taking into consideration interaction between the ring fibres, i.e., considering the stress state as uniaxial, which makes it relatively easy to calculate rotational angle of the ring [16]:

$$\phi = \frac{M_t + M_p}{El_y} y_c, \tag{1}$$

where y_c is the radius of centroid of section; I_y is the second area moment towards the axis Oy, passing through the centroid and perpendicular to the ring axis; E is the elasticity modulus of the ring material; M_t, M_p are force moments towards the axis Oy, conditioned by unevenness of the temperature and pressure fields.

The moment conditioned by temperature change over the ring length is determined by integral [17]. $M_t = \int_{(s)} E\beta\theta x dx$, i.e., a decrease in the temperature moment can be achieved using combined rings: a slip ring made of an antifriction material with a low elasticity modulus and linear expansion coefficient is fixed in the steel retaining ring.

For the ring shown schematically in Fig. 1, rotation of the section as a result of temperature deformations occurs counter clockwise, therefore, the temperature moment is positive $M_t > 0$. If the ring section is close to a rectangular shape, then $I_y = bl_3/12$, $dS = bdx$, $y_c = 0.5(r_1 + r_2)$. With a constant by section elasticity modulus and linear expansion coefficient, temperature component of slope of the elastic curve is calculated by the formula

$$\phi_t = \frac{6\beta(r_1 + r_2)}{l_3} \int_0^l \theta(x) x dx, \tag{2}$$

and taking into account the temperature distribution along the length of the ring [18]

$$\phi_t = \frac{12\beta\theta_0(r_1 + r_2)}{m^2 l^3} \frac{\mathrm{sh}^2(ml/2)}{\mathrm{ch}(ml)}. \tag{3}$$

Formula (3) may be used for rough estimation of temperature deformation.

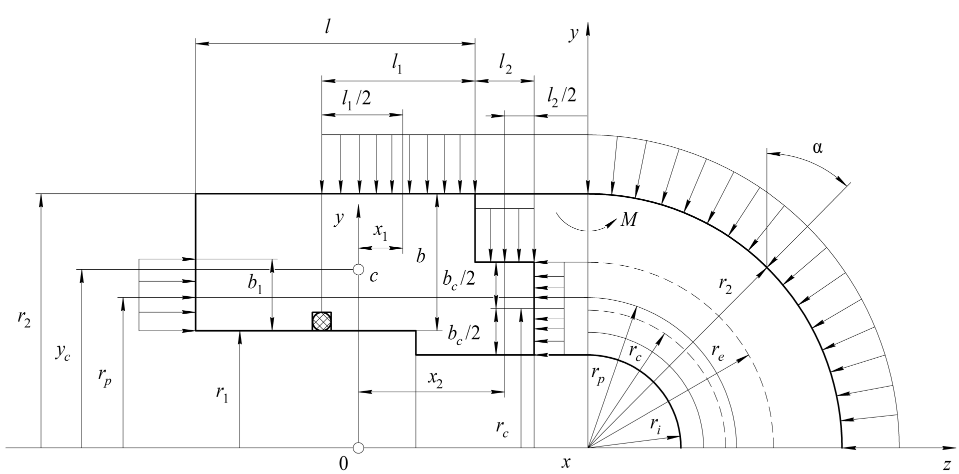

Fig. 1. Diagram of sealing ring deformation

Total moment over two cylindrical surfaces is

$$M_r = \int_0^{\pi/2} dM_r = -p_1 \left(l_1 r_2 x_1 + l_1 r_e x_2\right).$$

Taking into consideration that $p_c = kp_1$, $k = S/S_c = b_1 r_p / b_c r_c$, it should be written

$$M_a = p_1 \left(b_1 r_p^2 - k b_c r_c^2\right) = p_1 b_1 r_p^2 \left(1 - \frac{r_c}{r_p}\right). \tag{4}$$

To reduce the moment of radial forces, it is necessary to select the ring so that the displacement x_1 of the centre of radial load towards the centroid of section is minimum.

Total slope of the elastic curve (1) is determined by algebraic expression of moments or algebraic expression of the corresponding components of slope of the elastic curve ($\phi = \phi_t + \phi_p$). Herewith, the possibilities for reducing total ring deformations expand: temperature deformations can be compensated for by force deformations. Displacement of the outer points of the contact surface towards the inner ones can be determined on the basis of total slope of the elastic curve: $\delta = \phi b_c$. Based on experience of many years in the development and operation of mechanical seals in various conditions [19], admissible limit value δ is determined from the ratio $\delta / r_e \leq 1.2 \cdot 10^{-4}$. Positive displacements correspond to opening of the end clearance from the side of larger radius r_e of the contact surface. Based on the results of experimental and computational studies [20], it is possible to obtain a solution to the temperature problem that is quite acceptable for calculating the thermal deformations of the mechanical seal rings, limited to estimating heat generation in the sealing gap under conditions of liquid friction.

Method for determining thermal deformations in mechanical seal rings. We present a detailed algorithm for calculating the axial temperature difference and temperature deformations in the mechanical seal rings in the form of a calculation procedure.

The design of the assembly, the geometric dimensions of the parts are selected:

– r_1, r_2 are the outer and inner radii of the sealing band of the rings;

– materials of friction pair rings (including thermal conductivity λ and linear expansion α coefficients);

– the area F_K of the sealing contact of the surfaces of a pair of rings; load factor k; slot shape parameter C.

The operating parameters of the seal are assigned: differential pressure Δp across the seal, shaft angular frequency ω, allowable leakage through the seal Q, temperature t and dynamic viscosity μ of the medium to be sealed in operating mode.

A draft design of the mechanical seal unit is being developed. In the subsequent calculation of the sealing unit, the shape of the sealing joint obtained as a result of force and temperature deformations is considered to correspond to such a shape in which the specific value of the bearing force $\bar{F}$ in the sealing joint corresponds in magnitude to the load factor k.

Using dependence $Q = -\dfrac{\pi \Delta p h_0^3}{6\mu \ln \frac{r_1}{r_2}}$ for leakage through a mechanical seal with a flat slot, the size of the minimum gap h_0 in the sealing slot is calculated taking into account the correction factor C_Q. The size of the leak Q is specified by correcting for the shape of the gap according to the selected load factor k of the seal assembly.

Since in the hydrodynamic lubrication regime the load factor k is equal to the dimensionless bearing force $\bar{F}$ you can set the value of the slit shape parameter C.

From the expression $C = \dfrac{y}{h_0}$, the height of the wedge at the junction of the surfaces forming the sealing gap is determined, which occurs under the hydrodynamic friction regime as a result of force and temperature deformations of the rings.

For a mechanical seal with a known minimum sealing gap, the friction power losses are determined taking into account the correction factor C_N:

$$N = \frac{\pi \mu \omega^2 (r_2^4 - r_1^4)}{2 C_N h_0}.$$

The specific heat flux into the rings from heat generation in the sealing gap is determined: $q = \dfrac{N}{F_K}$.

The relative distribution of heat flows into the mechanical seal rings of the friction pair is determined: $k_\lambda = \dfrac{\lambda_1}{\lambda_2}$ or $\dfrac{q_1}{q_2} = \dfrac{\lambda_1}{\lambda_2}$, where $q_1 + q_2 = q$.

The dissipations of the heat flux in the ring's coefficients are determined k_1 and k_2. In this case, the following simplifying assumptions are used:

– the heat flow, brought to the sealing surface of the ring, is distributed over the entire volume of the ring;

– the temperature of the non-working rear end of the ring is assumed to be equal to the temperature of the medium being sealed in the seal chamber;

– $k_1 = \dfrac{r_{21} - r_{11}}{r_2 - r_1}$ for ring 1; $k_2 = \dfrac{r_{22} - r_{12}}{r_2 - r_1}$ – for ring 2.

The general axial temperature differences in the rings of the friction pair are determined: $t_1 - t_{cp} = \dfrac{q_1 H_1}{\lambda_1 k_1}$, $t_2 - t_{cp} = \dfrac{q_2 H_2}{\lambda_2 k_2}$.

The temperature deformations of the sealing surfaces of each of the rings are determined due to the rotation of the section from the axial temperature gradient: $\phi_1 = \dfrac{\alpha_1 (t_1 - t_{cp})(r_2^2 - r_1^2)}{2 l_a}$, $\phi_2 = \dfrac{\alpha_2 (t_2 - t_{cp})(r_2^2 - r_1^2)}{2 l_b}$.

The total temperature deformation of the rings in the seal is determined $\phi = \phi_1 + \phi_2$.

An analysis of the factors affecting the deformation of rings from heat releases shows that the deformation of each of them is expressed by the functional dependence $\phi = f(\alpha_t, \mu, v^2, b, B, l^{-1}, \lambda^{-1})$, where α_t is the coefficient α of linear expansion of the material; μ is the viscosity of the medium to be sealed; b is the width of the sealing band of the friction pair; B is the width of the ring along the radial coordinate; H is the ring thickness; λ is the coefficient of thermal conductivity of the ring material; v is the relative slip velocity in the friction pair.

The greatest influence on the deformation is exerted by the relative sliding speed of the rings in the sealing pair.

Results and discussions.. Consider the options for the mutual arrangement of the sealing surfaces of the mechanical seal rings (Fig. 2).

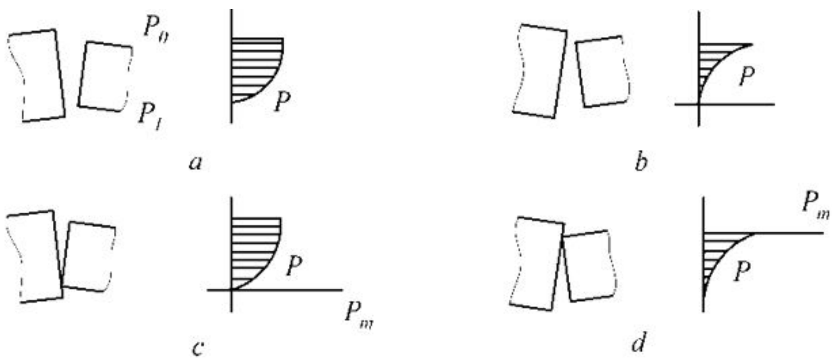

Fig. 2. The location of the sealing surfaces of the mechanical seal rings and the diagram of the pressure of the sealed liquid: a – confusor open joint, b – diffuser open joint, c – confusor joint with contact, d – diffuser joint with contact

In the first case (Fig. 2a) there is no contact between the sealing surfaces, so the force that opens the joint is created due to fluid pressure in the gap. As the fluid pressure plot for this case shows, the average fluid pressure is somewhat greater than the average of the internal and external pressures. For the case of a diffuser open joint (Fig. 2b), the pressure in the film is less than the average value of the internal and external pressures. This case is not stable and will result in contact between the rings as shown in Fig. 2d. Fluid pressure distribution diagrams in cases of contact between the rings, shown in Fig. 2c and d will take extreme forms as the minimum film thickness approaches zero. Thus, the total possible change in the magnitude of the load created by the pressure of the liquid, when moving from a fully divergent (confusor) joint to a fully convergent (diffuser) joint, ranges from zero to a pressure equal to the pressure being sealed. Since in practical case the film thickness cannot be zero due to roughness, these limits cannot be reached.

Let us consider the qualitative side of the obtained dependencies. Figure 3 shows the dependencies of the bearing force $\bar{F}$ and rigidity of the lubricating layer

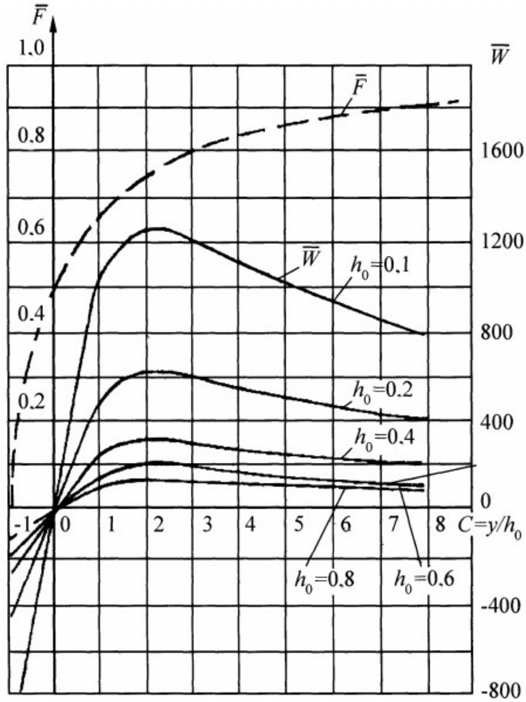

Fig. 3. Dependence of the bearing axial force $\bar{F}$ and axial rigidity $\bar{W}$ of the layer of the compacted medium on the parameter C of the slot shape

$\bar{W}$ in the gap for a number of values of approach of the sealing surfaces h_0 on the gap shape parameter $C = y/h_0$ (y is the taper parameter).

It follows from them that positive values of the axial rigidity of the liquid layer in the gap, as well as the regime of liquid friction in the end pair, are ensured only with a confusor-shaped gap, i.e., at $C > 0$.

A seal with a gap formed by parallel surfaces ($C = 0$) has zero hydrostatic stiffness. The friction mode in the end pair is in unstable equilibrium and can turn either into liquid or mixed friction mode.

A seal with a diffuser slot shape has a negative axial rigidity of the liquid layer. The friction mode in the end pair can be either mixed (at $-1 < C < 0$) or dry (at $C < -1$).

As follows from Fig. 3, the maximum axial rigidity of the liquid layer for a wedge-shaped slot exists when the slot shape parameter is $C = 2$. In this case, the value of the hydrostatic load-bearing force in the slot is $\bar{F} = 0.75$.

A parameter for optimizing a seal with a wedge-shaped gap is the axial rigidity $\bar{W}$ of the load-bearing layer of liquid at the junction of the mechanical seal rings end surfaces. The controllable seal parameters are the load factor k and the value of the taper parameter y on the high-pressure side. These parameters determine the magnitude of the closest approach between the working surfaces of the seal h_0. From the condition of the optimum value $\bar{W}$, the load factor k of the seal should be taken close to the value $k = 0.75$.

Conclusions. Highly loaded centrifugal machines are mostly high-speed units, so an important factor in their operation is high heat generation in the end gaps of the sealing units, which causes significant thermal deformations of the sealing surfaces of the rings of the end pairs. It is necessary that the level and type of deformations of the rings due to the pressure of the medium and temperature factors provide the necessary calculated optimal shape of the sealing gap in the end sealing joint of the pair of rings. The execution of the rubbing mating surfaces should ensure the formation of a confusor form of the sealing gap in all modes of operation and axisymmetric pressure fields during the rotation of the rotor.

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[1]*School of Intelligent Manufacturing, Yancheng Polytechnic College,*
224005 Yancheng, China
e-mail: `2015170412@yctei.edu.cn`

[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]*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`

[4]*Scientific Department, National University of Civil Protection of Ukraine,*
18034 Cherkasy, Ukraine
`https://orcid.org/0009-0003-2053-2362`, e-mail: `8482244@gmail.com`

[5]*Department of Engineering Mechanics and Engineering Technology,*
Admiral Makarov National University of Shipbuilding, 54007 Mykolaiv, Ukraine
e-mails: monachow88@gmail.com, yurii.galynkin@nuos.edu.ua

[6]*Department of Computerized Control Systems,*
Admiral Makarov National University of Shipbuilding, 54007 Mykolaiv, Ukraine
e-mail: oleksandr.gerasin@nuos.edu.ua