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Computational method for mechanical seal as a dynamic system

A functional block diagram of a mechanical seal as a dynamic system giving an idea of interaction of its basic elements has been developed. A method for computing loads in a friction pair of the mechanical seal, sealed liquid leakage, friction power losses in the end pair, as well as deformations of sealing rings is proposed. This allows the operational factors that affect seal performance to be taken into account during the design phase.

Key words: mechanical seal, functional block diagram, force and temperature deformations, friction power, leakage.

Importance of the issue of unit rotary shaft sealing derives from the fact that leakages via rotor seals of industrial pumps and compressor result in huge losses of energy, valuable raw materials, water and other resources. According to some reports, about 60% of air emissions are caused by uncontrolled leakage via seals [1].

Extensive knowledge is required to design, manufacture and operate seals and sealing systems. Processes in the sealing space are determined by liquid and gas properties, physical phenomena of the processes of heat transmission and heat transfer, constitutional changes, wear and corrosion processes, balance of effective forces and torques, vibration and other factors.

It is very important to choose seals as up to 70% of forced shutdown during operation is caused by seal outage [2]. That's why selection of the seal type is determined by the equipment requirement and working conditions of maintenance staff as well as economic feasibility [3]. At the present time the mechanical seals having such important qualities as leak resistance and durability find expanding applications.

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Fig. 1. Diagram of mechanical seal

Physical principles of mechanical seal performance. Simple mechanical seal (Fig.1) has an offset 2 and rotating axially-movable 3 sealing rings made of wear-resistant material, which are caged 1 and 5. Preliminary contact pressure between the rings is achieved through spring force 6, and then enhanced due to pressure force of the sealed liquid. The clearance between shaft and axially-movable ring 3 is sealed by secondary seal 4. The torque required to overcome friction on the end contact surfaces is transferred from shaft to rotating ring via a driving device (pin 7 and ring skirt with a longitudinal groove).

Sealing is completed as a result of compression of the end surfaces of offset 2 and rotating 3 rings. With increase of contact pressure, sealing is increased, but along with this friction power losses increase, thus increasing the wear of friction surfaces, their heating and temperature deformations. Consequently, the performance efficiency of seal is determined by contact pressure and physical process on sensing and rotating away from each other end surfaces. Despite seeming external simplicity of mechanical seal assembly, the processes proceeding at the joint of mating end surfaces of two parts (one of which rotates with rotor) are extremely complicated. This is due to simultaneous affect and communication of the processes of friction, hydrodynamics, heat processes as well as change of the form of mating surfaces in the sealing joint when load parameters of the seal are changed.

According to the experimental data available [4], seal performance can be more simply presented as follows. When the whole is filled with a liquid, leakage is unavoidable, resulting in the heat to be tapped from friction pair and under normal conditions a heat balance is established. With increased friction power losses (for example, due to increase in peripheral velocity), temperature in the layer rises and the moment might come, when liquid in this layer begins to boil. As a rule, it happens in the region adjacent to external low-pressure chamber, where temperature is maximal, as liquid temperature increases as it

flows in the clearance gap from high-pressure to low pressure region. Liquid and vapour phases are formed and border line between them can shift radially. The region of liquid phase decreases as friction power losses increase. On the partition surface vapour is intensively formed due to frictional energy. That's why temperature in the clearance gap is regulated as steam generation reduces viscous friction force. Under unfavourable conditions a liquid phase can reduce so much so that liquid layer in the clearance gap will lose its integrity. This will lead to a sharp rise in temperature and proper seal performance is disturbed. Therefore, if a visible leakage is permitted, it is necessary to ensure a good heat rejection to prevent vapour generation.

In order seals to perform without visible leakage, it is necessary to stabilize the position of phase interface. In connection therewith, by small peripheral velocity and viscous liquids even heat rejection may be appropriate. With high speed it is necessary to ensure a reliable cooling of friction pairs. The main difficulties of calculating mechanical seals and performance prediction are conditioned by the complexity of friction and wear processes. In this case practice is considerably in advance of theory: immediacy of the problem of rotor sealing often forces to blindly search for and find the proper constructive and process design solutions for various parameters of the sealed liquids, their pressures, peripheral velocity, temperature, requirements for reliability, durability and sealing at economically sound cost.

When designing mechanical seals, it is reasonable to take a durability indicator as an optimality criterion, for example, an average resource, i.e. mathematical expectation of the operating time before limit state, when further operation becomes technically impossible or unreasonable. Other important requirements — integrity, safety margin, minimum friction power losses, admissible dimensions and cost — should be considered as limitations in the optimization problem.

The durability of mechanical seals is determined by the wear resistance of materials of friction pairs, operating conditions and contact pressure, which, in turn, depends on a range of design and operational factors. The design factors, except for coefficient of load and direction of radial flow, include geometric shape of radial section of the sealing rings, their diameter and manner of fastening. The main operational factors are: rotor speed, physical properties, pressure and temperature of the sealed medium, content of abrasive contaminant in it, value and nature of rotor vibrations.

Let's introduce the following definitions:

I — heat rejection efficiency; t — temperature; k — coefficient of load; P_1 — sealed pressure; N — friction power loss; P_c — contact pressure; P_s — hydrostatic pressure in the clearance gap; φ — temperature deformations; h — end clearance; U — wear; N — friction power loss; P_c — contact pressure;

The mechanical seal is a dynamic system [2], a simplified structural diagram of which (Fig. 2) gives an idea of interaction of its main components, nature of their transfer functions, external actions and outputs of the system and allows the computational problems of transfer functions of components to be determined and the system reaction to external influences to be estimated. Complexity of the processes of thermal hydroelasticity, friction and wear does not allow giving an accurate definition of most transfer functions of the system components. Therefore, the computation has to be limited to approximate estimates and a comparison of the newly designed seals with proven-in-use analogues.

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The main factor that determines wear of the friction pair of a mechanical seal is contact pressure, which, in turn, depends not only on the design and operating parameters, but also on a range of internal processes accompanying the seal performance. In particular, force and temperature deformations, which are determined by geometric shape of the sealing rings, friction power losses and the conditions of heat rejection, can have a significant effect on the processes in the friction pair. Therefore, in order to successfully design mechanical seals as dynamic systems, computational tools developed on the basis of studying the physical processes occurring in them should be used.

The aim is to develop a method for computing loads in the friction pair of mechanical seal, sealed liquid leakage, friction power losses in the friction pair as well as deformations of sealing rings.

Computing loads in the friction pair of mechanical seals. The condition for axial balance of the axially-movable ring of mechanical seal (Fig. 3) is written as

$$F + F_n = F_c + F_s \pm F_m, \quad (1)$$

where F — axial pressure force, which presses axially-movable ring to the seat, $F = p_1 S_1 + p_2 S_2$; F_n — force of elastic element; $F_c = p_c S_c$ — force of contact pressure; F_s — force driven by hydrostatic pressure P_s in the end clearance; F_m — friction force on secondary seal preventing displacement of axially-movable ring in one direction or another. Square signs S_1 and S_2 are considered positive if pressure forces acting on them increase contact pressure in the friction pair, and negative if the corresponding pressure forces open the end joint. Taking into account the accepted rule of signs $S_c = S_1 + S_2$, and expression for force F may be written as

$$F = \Delta p S_c (k + p_2 / \Delta p).$$

Here $\Delta p = p_1 - p_2$ — sealed surplus pressure; $k = S_1 / S_c = b_1 r_1 / b_c r_c$; $b_1 = r_e - r_1$, $b_c = r_e - r_i$; $r_p = 0.5 (r_e + r_1)$; $r_c = 0.5 (r_e + r_i)$. If $S_2 > 0$, then $k < 1$ (balanced seal), if $S_2 < 0$, so $k > 1$ (loaded seal). If the pressure downstream is not zero ($p_2 > 0$), then $k_e = k + p_2 / \Delta p$ should be taken as an effective coefficient of load.

Force of hydrostatic pressure that opens the end joint is determined by the pressure distribution in the clearance gap and can be obtained by summing the elementary pressure forces throughout the entire contact area S_c :

$$F_s = \int_{(S_c)} p_s ds = \bar{p}_s S_c,$$

Fig.3. Diagram of the forces affecting the axially-movable ring of sealing ring

where gap mean hydrostatic pressure

$$\bar{p}_s = \frac{1}{S_c} \int_{(s_c)} p_s dS.$$

Since the end clearance corresponds to contact surface roughness, therefore, it has not yet been possible to theoretically determine the pressure distribution in the clearance gap. Only with relatively large clearance gaps ($h_0 / \bar{R} > 3 - 5$), corresponding to the liquid lubrication regime and peculiar to hydrostatic seals, pressure in the end channel changes from inlet area to outlet area according to the law close to linear:

$$p_s = p_2 + \Delta p \frac{r - r_1}{r_2 - r_1}.$$

Herewith

$$\bar{p}_s = 0.5 (p_1 + p_2). \quad (2)$$

Taking into account the hydraulic force, balance equation (1) may be written as $p_c S_c = \Delta p S_c k_e - \bar{p}_s S_c + F_n F_m$, whereof

$$p_c = \Delta p k_e - \bar{p}_s + \frac{(F_n - F_m)}{S_c}. \quad (3)$$

The sign of friction force depends on whether it prevents the contact surfaces from closing with each other under load. The force of an elastic element is usually assumed to be somewhat larger than the friction force; therefore, the last summand in balance (3) is small and can be neglected. For a liquid friction mode (2), it may be written as

$$p_c = \Delta p (k - 0.5) + \frac{(F_n + F_m)}{S_c}, \quad (4)$$

that is, the contact pressure is completely determined by load factor k and the sealed surplus pressure Δp . It can be seen from (4) that at $k < 0.5$ there is a danger of opening the end joint. Therefore, take $k = 0.55 \div 0.85$ for balanced seals and for loaded seals $k = 1.1 \div 1.2$.

Mean clearance gap between the rough surfaces in the presence of contact markings ($p_c > 0.15$ MPa) can be evaluated by formula [6]

$$h = 0.5 \left(\frac{R_{a1}}{K_1^2} + \frac{R_{a2}}{K_2^2} \right), \quad (5)$$

where $K_{1,2}$ — coefficients of surface smoothness of axially-movable and base rings respectively, $K_{1,2} = \frac{R_{a1,2}}{R_{\max 1,2}}$ for mechanical seals $1 \geq K \geq 0.67$. At a high

contact pressure, when unevenness interpenetration is observed, accurate values for clearance gaps are determined from the formula

$$h = \frac{1}{20} \left(\frac{R_{a1}}{K_1} + \frac{R_{a2}}{K_2} \right). \quad (6)$$

If the difference $F_n - F_m$ is neglected, for boundary friction mode at $p_2 = 0$ it follows from the formula (3) that the coefficient of load is equal to the ratio of average contact pressure to sealed pressure: $k = p_c / p_1$.

Calculating leakage and friction power losses. Leakage via a smooth gap h between two flat ring plates by a laminar flow is determined by expression [7] $Q = \frac{\pi r_c h^3 \Delta p}{6 \mu b_c}$, if $b_c < r_c$. Clearance gap between rough surfaces is calculated using the formula (5) or (6). Friction force at the surface of ends $F_T = f p_c S_c$, and friction power losses

$$N_c = 2 \pi f p_c \omega r_c^2 b_c = S_c f r_c \omega p_c. \quad (7)$$

For the mode of liquid film lubrication $N_c = \frac{2 \pi \mu \omega^2 r_c^3 b}{h_0}$.

In the boundary lubrication mode, friction coefficient can be determined only experimentally for each pair of materials and operating conditions. According to the data of paper [5], obtained as a result of testing several hundred friction pairs at the contact pressure of 0.15–20 MPa and sliding velocity 0.01 to 50 m/s, the friction coefficient sharply decreases with increase in the contact pressure from 0.15 to 0.65 MPa. At $p_c > 0.65$ MPa, the friction coefficient of a given pair of materials becomes constant and does not depend on the pe-

ripheral velocity, pressure, width of the contact surfaces, provided that the deformations of sealing rings are small ($\varphi < 1,2 \cdot 10^{-4} \times r_e / b_c$) and temperature in contact does not exceed vaporization temperature of the boundary film, i.e., conditions of boundary lubrication are preserved. Herewith, for different pairs of materials and properties of the sealed liquid, the steady-state value of the coefficient of friction is within 0.03–0.15.

Total power supplied on the seal performance increases due to losses associated with liquid leakage $N_p = Q\Delta p$ and disk friction losses. If leakages abstract some heat from the contact surfaces, then disc losses increase liquid temperature in the seal chamber and thereby increase total thermal stress of the assembly. Liquid friction power of rotating end and cylindrical surfaces is expressed through coefficients of disk friction [8]

$$N_1 = 0,25c_1\rho\omega^3(R_2^5 - R_1^5), \quad N_2 = c_2\pi\rho\omega^3 R_1^4 l, \quad (8)$$

where R_1 and l — cylinder radius and length; R_2 and R_1 — outside and inside disc radii; ρ — liquid density; c_1 and c_2 — coefficients of disk friction [4]. It should be noted that friction power losses (8) are proportional to the cube of rotor speed and at low frequencies (up to 3000 rpm) are measured in fractions of kilowatt, therefore, taking these losses into account is advisable for high-speed machines.

Friction power in the sealing clearance is converted into heat, which spreads through the contact end surfaces along the sealing rings, creating an uneven temperature field. Contact surface heating significantly affects the friction mode: firstly, mechanical and physical properties of the materials of the rubbing bodies and the separating layer of the sealed liquid change, and secondly, the rings are subject to thermal deformations, which violate the uniformity of contact in the friction pair. Excessive heating leads to evaporation of the liquid layer. Herewith, the friction coefficient increases sharply, temperature and wear rise, i.e., there are thermal cracking of the rings and loss of tightness.

The main tasks of thermal calculation are to determine maximum temperatures in the friction pair and to assess temperature deformations of the rings in order to ensure such conditions in the structure under which temperatures and deformations do not exceed the allowable values. Calculation of the thermal state is based on heat balance equations. Average liquid temperature in the seal chamber is determined from the equality of total heat flow released during the seal performance and convective removal from its case (Fig. 4). Heat balance equation is written as

$$N_c + N_1 + N_2 = N_k N_p, \quad (9)$$

where heat removal from the case $N_k = S_k \alpha_k (t_1 - t_2)$, and heat removal via the seals as a result of leakage $Q N_p = Qc\Delta t$. Where S_k — reduced area of the

Fig. 4. Diagram of heat flows in mechanical seal

seal case, from which heat is released to the external environment; α_k — heat transfer coefficient; c — specific heat capacity of the sealed liquid; Δt — increase in its temperature, taking into account the removed heat; t_1 and t_2 — temperature of the liquid inside the seal case and temperature of the external environment. If the seal is provided with forced cooling, then the corresponding heat flow rate must be added in the right side of equation (9). During normal seal performance, when leakages are small (approx. $10 \text{ cm}^3/\text{h}$), their influence on the thermal state can, as a rule, be neglected ($N \approx 0$).

Temperature in the friction pair is determined on the basis of heat balance equation for the sealing rings from surrounding them sealed leakage and external environment (see Fig. 4) $N_c = N_a + N_b$, where N_a and N_b — heat flows released respectively from the rotating ring A and base B rings.

Temperature field of sealing rings is generally described by a second-order nonlinear differential equation for a parabolic derivative. If weak dependence of the heat conduction coefficient on temperature is not taken into account in a stable thermal process, then temperature field can be described by Laplace's equation with complex boundary conditions. But at the same time, numerical solution of the problem for rings of complex shape is connected with significant difficulties. In this case, the methods of electronic modelling of thermal state of the seal are effective.

For a rough estimate of temperature on the contact end surfaces, let us simplify the problem by taking hollow cylinders (Fig. 5, a) with a heat flow uniformly distributed on the contact surface $q = N_c / S_c$ and constant in length heat transfer coefficients of cylindrical surfaces washed by the sealed liquid or the external environment as a design model of the rings,

$$\frac{d^2\theta}{dx^2} - m^2\theta = 0, \quad (10)$$

under boundary conditions

$$x = 0 : \theta = \theta_0, \quad x = l : \frac{d\theta}{dx} = -\frac{\alpha}{\lambda} \theta_1,$$

where $\theta = t(x) - t_2$; $m = (\alpha L / \lambda S)^{0.5}$; t_2 — ambient temperature; λ — thermal conductivity of the ring material; α — heat transfer coefficient from the ring surface to the external environment; $L = 2\pi r_2$ — perimeter of the ring surface on which heat is transferred; $S = \pi(r_2^2 - r_1^2)$ — area of the ring cross section.

Equation is (10) solved according to the law of temperature change along the ring:

$$\theta(x) = \theta_0 \frac{\text{ch}[m(l-x)]}{\text{ch}ml}. \quad (11)$$

Meanwhile, decrease in temperature along the length is caused by heat transfer to the external environment. Inserting (11) into Fourier equation $N = \lambda \frac{d\theta}{dx} S$, we obtain the formula for the heat removed per unit of time:

$$N = \lambda S m \theta_0 \text{th} ml. \quad (12)$$

In real seals, in different areas of the ring, its surfaces can be in contact with different media or can be made of materials with various thermal conduc-

tivity coefficients. Therefore, the calculation model can be represented by cylinders (Fig. 5, *b*) with different heat transfer coefficients and different ambient temperature in certain areas of the surface. Herewith, total heat transfer of each ring consists of the sum of heat transfer from individual sections:

$$N_a = N_{a1} + N'_{a1} + N_{a2}, \quad N_b = N_{b1} + N_{b2} + N'_{b1}. \quad (13)$$

Here the first index is the ring number, and the second characterizes external environment of the given section: 1 — the sealed liquid with temperature t_1 ; 2 — external environment (air) with temperature t_2 . Each term is calculated by formula (12) taking into account the corresponding parameter values:

$$S_a = \pi (r_{a2}^2 - r_{a1}^2), \quad S_b = \pi (r_{b2}^2 - r_{b1}^2),$$

$$r_{a1} = r_{b1} - r_1,$$

$$l_{a1} = l_a, \quad l'_{a1} = c_a, \quad l_{a2} = l_a - c_a,$$

$$l_{b2} = l_b, \quad l'_{b2} = l_b - c_b, \quad l_{b1} = c_b,$$

$$L_{a1} 2\pi r_{a2}, \quad L'_{a1} = L_{a2} = L_{b2} = 2\pi r_1,$$

$$L_{b1} = L'_{b2} = 2\pi r_{b2}.$$

For the section of rings contacting with the sealed liquid $\theta_{01} = t_0 - t_1$, for areas contacting with the external environment, $\theta_{02} = t_0 - t_2$; t_0 — temperature at the contact of the rings. Inserting (13) into the heat balance equation, we obtain

$$N_c = (A_{a1} + A'_{a1} + A_{b1}) \theta_{01} + (A_{a2} + A_{b2} + A'_{b2}) \theta_{02},$$

whereof

$$t_0 = \left(\frac{N_c}{B_1} + t_1 + \frac{B_2}{B_1} t_2 \right) / \left(1 + \frac{B_2}{B_1} \right), \quad (14)$$

where B_1 and B_2 — heat transfer coefficients into the sealed liquid and external environment, $B_1 = A_{a1} + A'_{a1} + A_{b1}$, $B_2 = A_{a2} + A_{b2} + A'$. Taking into account that heat transfer from the surfaces of the rings, which are offset towards to the external environment ($\alpha'_{a1}, \alpha_{a2}, \alpha_{b1}, \alpha'_{b2}$), is small in comparison with heat transfer of the rotating surfaces (α_{a1}, α_{b2}), it can be written as $B_1 \approx A_{a1}$, $B_2 \approx A_{b2}$, or

$$B_1 = S_a \lambda_a m_1 \operatorname{th} (m_1 L_1), \quad m_1 = [\alpha_1 L_{a1} / \lambda_a S_a]^{0.5},$$

$$B_2 = S_b \lambda_b m_2 \operatorname{th} (m_2 L_2), \quad m_2 = [\alpha_2 L_{b2} / \lambda_b S_b]^{0.5}.$$

Consequently, change in the cross-section average temperature along the ring can be determined by formula (11), and average temperature on the contact surfaces of mechanical seal can be determined by formula (14). More accurate calculation, taking into account the change in temperature along the ring radius, was carried out in papers [4, 9].

Based on the conditions of heat resistance of contact surfaces and preservation of the liquid film in the clearance gap at a given pressure, temperature in the friction pair is limited t_* value. Herewith, allowable friction power can be determined for the given conditions of heat removal using formula (14) $N_* \leq (B_1 + B_2) t_* - B_1 t_1 - B_2 t_2$, and taking into consideration (7) and (14), allowable value of operational load:

$$(\omega r_c \Delta p f)_* \leq \frac{1}{k_e S_c} [(B_1 + B_2) t_* - B_1 t_1 - B_2 t_2].$$

It can be seen from (14) that in order to reduce temperature in the friction pair, it is necessary to reduce friction power losses and increase heat removal, intensifying heat transfer (increasing Nusselt number), increasing heat transfer coefficients and cross sections of the rings. The ring length is included only in the argument of the hyperbolic tangent, whereof limit value is one. Since for $ml = 1.6$ th $(ml) = 0,9217$ (limit value), it is ineffective to increase this argument further by increasing the ring length. Relevance of various methods for reducing temperature in the contact area should be determined based on the specific operating conditions.

Deformation of the sealing rings. 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 as a rotary movement of the ring cross-section 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 [10]:

$$\varphi = \frac{M_t + M_p}{El_y} y_c,$$

where y_c — radius of centroid of section; I_y — second area moment towards to the axis Oy, passing through the centroid and perpendicular to the ring axis; E — elasticity modulus of the ring material; M_t , M_p — force moments towards to 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 [11] $M_t = \int_{(s)} E\beta\theta x dx$, i.e., a decrease in the tempera-

ture 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. Herewith, the components of temperature moment decrease, which correspond to the intersection areas farthest from Oy axis, and are subject to the action of large temperature gradients $(\theta x)_{\max}$ (Fig. 5, c).

For the ring shown schematically in Fig. 6, 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

$$\varphi_t = \frac{6\beta (r_1 + r_2)}{l_3} \int_0^l \theta(x) x dx, \quad (15)$$

and taking into account (11) to distribute temperature over the ring length

$$\varphi_t = \frac{12\beta\theta_0 (r_1 + r_2)}{m^2 l^3} \frac{\text{sh}^2(ml/2)}{\text{ch}(ml)}. \quad (16)$$

Formula (16) may be used for rough estimation of temperature deformation. [6] gives the formula obtained from (15) on the assumption that the temperature over the ring length changes linearly. In this case, the deformations turn to be significantly underestimated.

Moment M_r about an axis Oy of radial pressure forces acting on a cylindrical surface with radius r_2 , and length l_1 , will be obtained if we sum up the moments of projections of elementary pressure forces (see Fig. 6) onto the plane xOz : $dM_r = -p_1 l_1 r_2 x_1 \sin \alpha d\alpha$. Total moment over two cylindrical surfaces is written as

$$M_r = \int_0^{\pi/2} dM_r = -p_1 (l_1 r_2 x_1 + l_2 r_e x_2).$$

If, when calculating the moments of radial forces, the projections of elementary pressure forces change, then when calculating the moment of axial forces, arm of the elementary force is variable: $dM_a = p_1 b_1 r_p^2 \sin \alpha d\alpha - p_c b_c r_c^2 \sin \alpha d\alpha$. Taking into consideration that $p_c = kp_1$, $k = S/S_c = b_1 r_p / b_c r_c$, it should be written

$$M_a = p_1 (b_1 r_p^2 - k b_c r_c^2) = p_1 b_1 r_p^2 (1 - r_c / r_p). \quad (17)$$

Fig. 6. Diagram of sealing ring deformation

It follows from the formula (17) that the moment of axial forces depends on load coefficient. For balanced seals $k < 1$, $r_p = r_c$, $M_a > 0$, for loaded — $k < 0$, $r_p < r_c$, $M_a < 0$. If $k = 1$, then $r_p = r_c$, $M_a = 0$. Consequently, the module of the moment of axial forces is greater, the more load coefficient differs from one. 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 [12].

Total slope of the elastic curve (14) is determined by algebraic expression of moments or algebraic expression of the corresponding components of slope of the elastic curve ($\varphi = \varphi_t + \varphi_p$). 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: $\delta = \varphi b_c$. Based on experience of many years in the development and operation of mechanical seals in various conditions [6], 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. During the design of mechanical seals, it is necessary to calculate the expected deformations of sealing surfaces of the rings to determine their shape in the end joint.

Conclusion

At the present time mechanical seals are widely used in pumps, compressors and devices of chemical industry. Their design and production require profound knowledge of complex interrelated processes of lubrication, force and temperature deformations and proficiency in the modern computational methods that allow maximum number of the above-mentioned factors to be taken into account. The developed method for calculation of the mechanical seal parameters allows operational factors to be taken into account during the design phase. The papers aimed at improvement of computational methods for mechanical seals will be relevant and in demand for a long time.

REFERENCES

1. Gaft, J. and Marcinkowski, M. (2004), "A choice of the seal for the shaft of the pump", *the Proceeding of Pump users International Forum*, Science and Engineering, September 29-30, Karlsruhe, 2004, pp. 45-52.
2. Martsynkivskyi, V.A. and Shevchenko, S.S. (2018), *Nasosy atomnykh elektrostantsiy: raschet, konstruirovaniye, ekspluatatsiya* [Pumps of nuclear power plants: calculation, design, operation], CHF izd-vo «Universitetskaya kniga», Sumy, Ukraine.
3. Gaft, J. (2005), "Comparative economic indicators of the main types of pump rotor seals", *Trudy 11-y mezhdunarodnoy nauchno-tekhnicheskoy konferentsii «Germetichnost', vibronadezhnost' i ekologicheskaya bezopasnost' nasosnogo i kompressornogo oborudovaniya»* [Proceedings of the 11th International Scientific and Technical Conference "Tightness, Vibration Reliability and Environmental Safety of Pumping and Compressor Equipment"], Vol. 1, 2005, pp. 38-45.
4. Melnyk, V.A. (2008), *Tortsevyeye uplotneniya valov. Spravochnik* [Mechanical shaft seals. Directory], Mashinostroyeniye.
5. Mayer, E. (1978), *Tortsevyeye uplotneniya* [Mechanical seals], Mashinostroyeniye, Moscow, USSR.
6. Golubiev, A.I. (1974), *Tortsovyeye uplotneniya vrashchayushchikhsya valov* [Mechanical seals of rotary shafts], Mashinostroyeniye, Moscow, USSR.
7. Kondakov, L.A. (1972), *Uplotneniya gidravlicheskikh sistem* [Seals of hydraulic systems], Mashinostroyeniye, Moscow, USSR.
8. Krevsun, E.P. (1998), *Tortsovyeye germetizatory vrashchayushchikhsya valov* [Mechanical seals of rotary shafts], Art-Feks, Minsk, Belarus.
9. Mueller, H.K. and Nau, B.S. (1998), *Fluid sealing technology*, Marcel Dekker, New York, USA.
10. Błasiak, S. and Kundera, C. (2012), "A numerical analysis of the grooved surface effects on the thermal behavior of a non-contacting face seal", *Procedia Engineering*, Vol. 39, pp. 315-326.
11. Falaleev, S.V. and Vinogradov, A.S. (1996), "Creation of workable in conditions deformations of mechanical seals", *Trudy VIII Mezhdunarodnoy nauchnotekhnicheskoy konferentsii «NASOSY-96»* [Proceedings of the VIII International Scientific and Technical Conference "PUMPS-96"], Vol. 2, IPP «Mriya» LTD, Sumy, pp. 53-60.
12. Adamczak, S., Kundera, C. and Swiderski, J. (2017), "Assessment of the state of the geometrical surface texture of seal rings by various measuring methods", *IOP Conference Series: Materials Science and Engineering*, Vol. 233, no. 1, pp. 12-31.

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МЕТОДИКА РОЗРАХУНКУ ТОРЦОВИХ
МЕХАНІЧНИХ УЩІЛЬНЕНЬ ЯК ДИНАМІЧНОЇ СИСТЕМИ

Розроблено структурну схему торцового ущільнення як динамічної системи, яка дає уявлення про взаємодію основних її елементів. Запропоновано методи розрахунку навантажень в парі тертя торцового ущільнення, протікання ущільнюваної рідини, втрат потужності на тертя в торцовій парі, а також деформацій кілець ущільнень. Це дозволяє враховувати експлуатаційні фактори, що впливають на роботу ущільнення, на етапі їх проектування.

К л ю ч о в і с л о в а: торцове ущільнення, силові і температурні деформації, потужність тертя, протікання.

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