CALCULATION OF THE OPERATIONAL CHARACTERISTICS OF THE IMPULSE GAS-BARRIER FACE SEAL

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Abstract:
The problem of reducing leaks along the pump or compressor shaft of pumped liquids and gases into the environment is very urgent. Serious difficulties have to be faced when sealing the shafts of machines that pump aggressive, toxic, explosive and fire-hazardous environments. According to modern occupational safety requirements, such pumps and compressors should use double seals with a barrier medium whose pressure exceeds the sealed one by 0.05–0.2 MPa. Currently, liquid-lubricated double mechanical seals are widely used in chemical production equipment, however, over the last decade of the 20th century, leading companies have developed a number of designs of double gas mechanical seals for pumps and chemical production devices, which significantly exceed liquid-lubricated seals in their performance characteristics. The vast majority of these seals use a gas-dynamic principle of operation, i.e. spiral, logarithmic, T-shaped or other micro grooves are made on the sealing faces of their rings, which, when rotated, create an additional gas dynamic force that ensures the functioning of these seals with a micron gap between the sealing pair. In this paper, the design, principle of operation and engineering methodology for calculating the main characteristics of an impulse gas barrier face seal, in which one pair of sealing rings performs the functions of a double mechanical seal, is considered. The design is simple, compact and, thanks to the more advanced principle of creating a gap between the sealing pair, is able to maintain operability in a wide range of sealing and barrier pressures. The existing experience of operating seals of this type on chemical production pumps has confirmed their high efficiency, reliability and safety.

Key words: mechanical face seal, impulse principle of operation, flow rate balance equation, barrier gas, non-contact operation

INTRODUCTION
Over the past decades, various designs of mechanical seals with a self-regulating gap have been created. The size of the operation gap can reach 3…30 microns, and in some cases 40…50 microns. These seals work reliably enough at high speeds in a wide range of operating parameters, providing small leaks and leak-proofness when the shaft of the rotary machine does not rotate. The sealing action in non-contact mechanical seals is carried out due to the large hydraulic resistance of the operation gap (thin gap between the working surfaces of the sealing rings). The size of the gap makes it possible to ensure effective cooling of the sealing rings surfaces due to leakage of the sealed product and at the same time meet the requirement of leak-proofness [1, 2]. According to leading experts, such seals have practically no restrictions by the parameter $p_v$.

Along with single seals, more complex double sealing systems are widely used in the industry. In them, instead of one seal, two are used, between which a buffer liquid is supplied. This liquid is supplied under a pressure slightly exceeding the pressure of the sealed product (usually by 0.1–0.3 MPa). From the space between the seals, a certain amount of buffer liquid enters the sealed casing of the rotary machine (through the internal seal) and into the atmosphere (through the external seal). Such a scheme not only promotes lubrication, flushing and cooling of the seal, but also allows you to completely eliminate leakage of the sealed product into the area behind the seal [3, 4]. Double sealing systems, in which both seals are mechanical, are recognized as the most effective among the
existing ones. This is due to the long service life and cost-effectiveness, as well as the ease of maintenance and operation of the mechanical seal compared to seals with a different operating principle [5, 6, 7]. Buffer liquids used in them must have satisfactory lubricating properties, high thermal conductivity, stable composition at operating temperature, minimal chemical activity to the materials of the sealing parts, good compatibility with the sealed product [8]. They should not be toxic.

Despite the great advantages of liquid-lubricated double seals compared to single seals, they are still not without certain disadvantages. Firstly, the buffer liquids used in seal have a strictly defined narrow temperature range of operation, going beyond the boundaries of which negatively affects the operation of the unit [9, 10]. Secondly, the cost of such a liquid is often comparable to the cost of the sealed product itself, which leads to an increase in the cost of operating the pump. Thirdly, the leakage of buffer fluid through the internal seal into the sealed casing of the pump reduces the purity of the sealed product. Finally, the operation of the double seal requires a complex and expensive system for pressure monitoring, cleaning, cooling and buffer liquid supply [11, 12, 13]. Thus, the use of double seals on liquid lubrication is associated with large additional costs for the operation of the entire pump.

An alternative to expensive double seals on liquid lubrication are the so-called “dry” seals, in which gas is used as a lubricating medium. The double “dry” seals used in pumps with a barrier gas neutral to the sealed product can operate in a wide temperature range (starting with applications in cryogenic technology and ending with the sealing of liquid polymer melts). The cost of the barrier gas (nitrogen or air is usually used) is small compared to the cost of a buffer liquid (in industrial conditions it can be $ 3-5 per year per pump). Small leaks (on the order of several normal litres per minute) and the low cost of the barrier gas make it possible to significantly simplify the system of preparation and supply of the barrier medium, while maintaining a low level of contamination of the sealed product and, most importantly, the environment [14, 15, 16, 17]. Currently, gas non-contact seals have been created for various machines (centrifugal compressors, gas blowers, steam and gas turbines, turbo expanders, turbojet engines) for a wide variety of parameters: sealed medium – hydrogen, carbon dioxide, ammonia, water vapor, etc.; the temperature of the compacted medium is -150...+300°C; sealing pressure: up to 25 MPa; sliding speed: up to 200 m/s.

LITERATURE REVIEW

The characteristics of “dry” seals listed above make us study these relatively new seals more closely. Therefore, we will consider in detail the latest achievements in this field of technology development. Usually, friction pairs using the thermo-gas-dynamic principle of operation are used as contact dry mechanical seals [18, 19, 20, 21]. An example is the double seal “Dura Seal GB-200” of “Durametallic Corporation” [22, 23, 24, 25]. In this double seal, the barrier gas is supplied between the internal standard contact and external thermo-gas-dynamic sealing rings at a pressure exceeding the sealed by 3.5 bar. The inner pair of sealing rings operates at a pressure drop of 3.5 bar, and the outer – at drops up to 13.8 bar. This mode of operation of the external seal became possible thanks to special pads and grooves made on one of the sealing rings. This seal works in the temperature range from -40°C to +120°C with rotor speeds from 0 to 3000 rpm. An example of a “dry” mechanical contact seal can also be a seal of the company “BW/IP International Inc.” [26, 27]. In the design of this double seal, the inner pair of sealing rings operates in non-contact mode, and the outer pair operates in contact mode. One of the rings of the outer pair has a special wavy surface. This ring shape allows the pair to work like a thermo-gas-dynamic one. The seal design has a wider range of operation compared to the one described earlier: operating pressure up to 41.4 bar, temperature from -40°C to +204°C, relative sliding speed of the rings from 1.5 to 30.5 m/s. The pioneer in the use of gas-lubricated seals in pumps was “John Crane Int.”. For the first time, gas-lubricated seals were installed in 1988 on a pump pumping 90% nitric acid [28, 29]. The advantages of “dry” gas dynamic seals in comparison with contact seals have led to their interest in cryogenic technology [30, 31]. The company “EG&G Sealol” produces a seal on the surface of the sealing rings of which radial microgrooves are made, having a significant elongation in the circumferential direction of the spiral shape [32, 33, 34, 35]. The company “Durametallic Corporation” produces a non-reversible seal “Dura Seal GF-200” with spiral grooves united by an annular channel [36, 37] and reversible “Gaspack” with special T-shaped grooves [26, 27]. The company “Feodor Burgmann Dichtungswerke GmbH & Co.” has advanced even further in experiments with the shape of gas dynamic grooves: grooves of a wedge-shaped “three-dimensional” shape are applied on the surfaces of the seals of this company [38].

A fundamental step in the development of gas seals for pumps was the rejection of the concept of double seals (seals containing two pairs of sealing rings). This became possible due to the combination of two main functions of the barrier gas: sealing and lubricating. An example is the non-contact gas-static seal of the Japanese company “Nippon Pillar Co. Ltd”. This seal has the ability to remotely control the size of operation gap by changing the pressure of the supplied barrier gas. “A.W. Chesterton Co.” has developed a “4400 TwinHybrid” gas seal in which one pair of sealing rings also performs the functions of a double seal [39, 40].

As a result of the search for a seal design alternative to the existing gas-dynamic ones, also running on gas lubrication, but at the same time free from technologically difficult, potentially unreliable and prone to clogging gas-dynamic grooves and steps, it was decided to turn to the increasingly popular in CIS impulse seal, which uses the impulse principle of operation to maintain a stable operation gap [41]. Thanks to the impulse method of creating a lubricating film, the new gas-barrier face seal does not require high-tech and time-consuming profiling of the
operation surfaces with gas-dynamic grooves and steps. It is successfully operated [42]. The experience of the seal operation shows that its operation is characterized by the stability of the non-contact mode and leaks comparable to leaks through gas-static seals. However, despite the existing experience, it is still not known how effectively such a method of preventing leakage of the sealed product will work if the values of the operating parameters are unstable, i.e. their values will “float” relative to the expected nominal operating value, which is currently determined experimentally for each seal at the design stage.

METHODS, RESULTS AND DISCUSSION

The design of the gas-barrier seal discussed in the article is shown in Figure 1.

![Fig. 1 The impulse gas-barrier face seal](image)

The seal contains only one pair of sealing rings, one of which is non-rotating and fixed in the machine casing 3 and sealed with two rubber O-rings 4. The rotating ring 2 has the ability to move axially and is mounted on the sleeve 5, which is fixed on the shaft of the machine. Springs 6 provide the contact of both sealing rings in the absence of shaft rotation. In the stator sealing ring, supply channels 7 are made, through which the barrier medium (gas) is supplied directly into the operation gap. On the face surface of the rotary ring there are closed chambers 8, which, having the ability to connect to the supply channels 7 of the stator ring when rotating the rotary ring 2.

The seal functions as follows: in the absence of shaft rotation, the barrier gas pressure is not enough to open the operation gap and the seal works as a parking one. Due to its very low viscosity, the gas evenly fills the gap and prevents the sealed product from leaking through it. When the shaft rotates, the closed chambers 9 connect one after another in turn with the supply channels 8, through which barrier gas enters the chambers. Due to the compressibility of the gas, the pressure in the chambers increases to the value of the pressure of barrier gas, and after the chamber leaves the supply channel, the pressure in it gradually decreases due to the outflow of gas from it. The rate of pressure drop in the chamber depends on the size of the operation gap formed between the sealing rings when the shaft rotates, because gas flows out of the chambers into the sealed product and into the atmosphere, therefore, the greater the resistance of the sealing sections above and below the chamber, the smaller the amplitude of the pressure change in the chamber and vice versa. In addition, the value of the minimum gas pressure in a closed chamber $p_{\text{min}}$ depends on the speed of rotation of the rotor, that is, the time interval $T-t$ between two consecutive fillings of the same chamber. With the geometry of the sealing sections of the sealing rings selected in a certain way, the hydraulic load coefficient and the combination of the sealed $p_4$ and barrier $p_2$ pressures, the minimum gas pressure in the closed chambers $p_{\text{min}}$ remains higher than $p_4$. Taking into account that the distance between the chambers in the circumferential direction is small, the pressure on the inter-chamber space can be considered equal to the average pressure in the chambers $\bar{p}_2$. Consequently, a continuous zone is created inside the sealing gap (on the annular section occupied by the chambers), in which the gas pressure exceeds the pressure of the sealed product $p_4$. Thus, the barrier gas in the seal performs two functions – it creates a solid layer of lubricant and ensures the locking of the sealed product, i.e. prevents its leakage into the atmosphere.

Such combined barrier face seals, in which the barrier medium performs two functions at once, are not new. Hydrostatic seals have been described in [39, 40] for a long time, in which the barrier liquid is supplied through the supply channels into a continuous annular groove (or pockets separated by thin lintels) made on the face of the stator sealing ring. But such barrier seals are not widely used due to the fact that for normal operation they need a flow limiter of the barrier medium (throttle), which must be adjusted sufficiently fine to provide the required characteristics (operation gap of 0.002-0.004 mm, minimum flow rate of the barrier medium and high static stiffness of the lubricating layer).

The impulse seal presented in the article, despite its apparent similarity to hydrostatic, has fundamental differences in the design and method of creating an operation gap, and most importantly, it has a completely different, much more flow rate and static characteristic. In other words, in a sufficiently wide, at least 0.2 MPa, range of changes in the pressure drop between the barrier gas and the sealed product $\Delta p = p_1 - p_2$, the flow rate and static characteristics change little. This was made possible thanks to the impulse sealing principle, where the barrier gas flow limiter (throttle) is the closed chambers themselves and the flat surfaces (the gap between the sealing sections of the rings). A single chamber cannot absorb more barrier gas than its volume, gas compressibility and pressure difference will allow:

$$\Delta p = p_1 - \bar{p}_2 \text{min}.$$  

With an increase in the barrier pressure $p_1$, the average pressure in the chambers increases slightly, because the time $t = (2a_1 + 2a_2)/\omega$ during which the chamber is connected to the supply channel is limited and at the same time much less than the time $T-t$ during which the barrier gas flows out of the chamber through the sealing gap into the sealed product and atmosphere.

Unlike gas-lubricated double seals, the gas-barrier impulse face seal presented in the article has a unique feature associated with the ability to fine-tune the size of the operation gap by changing the pressure of the barrier gas. Due to this, the reliability and durability of the sealing unit is significantly increased, since in case of failure of the non-contact operation mode of sealing rings, which can
be fixed by a sharp increase in the flow rate of the barrier gas, it is enough to increase the barrier pressure with the help of a regulator and thereby create an additional opening force in the operation gap sufficient to detach from the dangerous mode.

Despite the apparent simplicity, face seals operating with a micron operation gap are a complex dynamic system in which the gap must be automatically maintained in the range of optimal values (0.002-0.004 mm) under changing external factors. The main task of calculating impulse face seals is to determine the sizes of sealing rings that provide the operability of the seal in a given range of rotational speeds, sealing and barrier pressures with minimal possible losses of the barrier gas and exclude the possibility of a breakthrough of the sealed product through the operation gap, that is, a continuous circumferential area in which the pressure of the barrier gas is higher than the pressure of the sealed product. To do this, it is necessary to derive formulas linking the size of the operation gap with the pressures of the barrier gas and sealed product, as well as the speed of rotation, after which it is possible to determine the flow rate of the barrier gas through the seal and the value of the averaged pressure of the barrier gas on the section where the closed chambers are located.

For engineering calculations of face seals with a self-regulating gap, it is customary to use the following assumptions:

- the gas flow in the operation gap is steady, isothermal, laminar, subsonic;
- the properties of the barrier gas correspond to the properties of an ideal gas;
- the sealing surfaces of the rings form a flat operation gap;
- pressure in closed chambers increases and decreases in time according to a linear rule;
- the flow of the barrier gas in the operation gap is pressure, radial and axisymmetric, i.e. the pressure does not change in the circumferential direction;
- small high-frequency (with a frequency \( \omega \) where \( k \) is the number of chambers) axial vibrations of the axially movable ring are not taken into account;
- the inertia forces of the gas in the operation gap and the friction forces in the secondary seal are negligible and are not taken into account in the calculation.

To determine the barrier gas pressure in closed chambers, consider the balance of mass gas flow rate through the annular section of the operation gap with a central angle \( \alpha \) equal to the angular length of the closed chamber (Fig. 2). During the operation of the seal, the barrier gas enters the chamber only when it connected with the supply channel, therefore, the balance of gas flow rate from the chamber into the sealed area and into the atmosphere will be determined by the expression:

\[
Q_{m3} = Q_{m1}(T - t) + Q_{m2}(T - t),
\]

where:

- \( Q_{m1} \) is the mass flow rate of the barrier gas into the sealed area through an external face throttle with a central angle \( \alpha \) and limited radii \( r_s \) and \( r_s \), during the time \( T - t \) (the period between the chamber’s fillings of the barrier gas through the supply channels),

\[
T = 2\pi/(\omega z),
\]

- \( z \) is the number of supply channels;

- \( Q_{m2} \) – the mass flow rate of the barrier gas into the environment through the internal (under the chamber) face throttle with an angle \( \alpha _ s \) and radii \( r_s \) and \( r_s \), during \( T - t \);

\[
Q_{m3} = \frac{V_k}{RT^0} (p_{2\text{max}} - p_{2\text{min}}),
\]

- \( R \) – gas constant;

\[
T^0 – \text{the absolute temperature of the barrier gas in the operation gap.}
\]

**Fig. 2 Geometrical dimensions of the rotating sealing ring with chambers**

According to the method of calculating the impulse seal on a liquid lubricant [20, 21], obtained the equation of the balance of flow rates (1) in terms of conductivity:

\[
g_{m3}(p_1 - \bar{p}_2) = g_{m1}(\bar{p}_2 - p_3^2) + g_{m2}(\bar{p}_2 - p_3^2),
\]

where:

- \( p_1 \) is the barrier gas pressure at the inlet to the seal;
- \( \bar{p}_2 \) – the barrier gas pressure averaged over the time between two fillings in a closed chamber;
- \( p_3^2 \) is the pressure behind the seal (usually atmospheric).

The formulas for calculating the conductivities have the form:

\[
g_{m1} = \frac{a_k x^2}{24 \mu R T^0 \ln (\frac{r_s}{r_s})} \left(1 - \frac{\alpha + 2\dot{\alpha}}{2\pi}\right),
\]

\[
g_{m2} = \frac{a_k x^2}{24 \mu R T^0 \ln (\frac{r_s}{r_s})} \left(1 - \frac{\alpha + 2\dot{\alpha}}{2\pi}\right),
\]

\[
g_{m3} = \frac{2V_k}{RT^0}.
\]

Taking the size of the operation gap at the nominal operation mode as the base \( x_0 \), then, expressing the conductivities in terms of the base values, obtained:

\[
g_{m1} = g_{m1b} x_0^3,
\]

\[
g_{m2} = g_{m2b} x_0^3,
\]

\[
g_{m3} = g_{m3b} \Omega = \frac{2V_k}{RT^0 T_s} x_0 \Omega.
\]
Then determine the gas pressure \( \bar{p}_i \) in the chambers averaged over the time \( T-t \) between the chamber fillings. To do this, obtained the flow rate balance equation (2) in dimensionless form, dividing the right and left parts into basic values \( x_0, p_0, \omega_0 \) and basic conductivity \( g \). The size of the operation gap \( x \), the pressure of the barrier gas \( p_b \), the angular velocity \( \omega \) at the nominal operating mode of the seal are taken as the basic ones.

The dimensionless flow rate balance equation has the form:

\[
\Omega (\psi_1 - \psi_2) = \frac{\#_{\text{basis}}}{\#_{\text{basis}}} \left( \psi_2^2 - \psi_1^2 \right) + \frac{\#_{\text{basis}}}{\#_{\text{basis}}} \left( \psi_2^2 - \psi_1^2 \right),
\]

where:
\[ u = x/x_0 \text{ – is the dimensionless size of the operation gap; } \]
\[ \Omega = \omega/\omega_0 \text{ – dimensionless angular velocity of rotation; } \]
\[ \psi = p_1/p_0, \psi_2 = \bar{p}_2/p_0, \psi = p_3/p_0, \psi_4 = p_4/p_0 \text{ – dimensionless pressures of the barrier gas in the chambers, behind the seal and in the sealed machine casing, respectively.} \]

Taking into account that \( \alpha_{m23}=\alpha_{m13}=\alpha_{m20}/\alpha_{m30} \) and \( G_m=\alpha_{m13}+\alpha_{m23}, \) obtained from equation (5) the characteristic of the seal, i.e. the dependence of the operation gap on the pressure of the barrier gas, the pressure of the sealed product and the speed of rotation:

\[
u = \left[ \frac{\Omega (\psi_1 - \psi_2)}{\#_{\text{basis}}} \right]^2,
\]

where:
\[ \alpha_{m1} = \alpha_{m13} \psi_1^2 + \alpha_{m23} \psi_3^2. \]

The dependence of the dimensionless averaged pressure of the barrier gas in a closed chamber on the gap has the form:

\[
\psi_2 = \left( \psi_1^2 + 4 \#_{\text{basis}} \#_{\text{basis}} \psi_1^2 (\psi_1^2 + \#_{\text{basis}} \#_{\text{basis}}) \right)^{1/2} - \Omega,
\]

where:
\[ F_c = F_0, \]
\[ F_0 = \#_{\text{basis}} \cdot \#_{\text{basis}} + p_{\text{st}} + p_{\text{st}} + F_{\text{pr}} \text{ – the force acting on the back surface of the axially movable ring (closing the operation gap); } \]
\[ F_0 = \text{the force acting on the sealing face of the axially movable ring; } \]
\[ F_{\text{pr}} \text{ – the force of pre-compression of the springs.} \]

The areas are determined by the formulas: \( S_3 = \pi (r_1^2 - r_2^2), S_4 = \pi (r_2^2 - r_3^2). \)

Unlike a liquid pressure, the gas pressure plot in a flat gap has a convex shape. The rule of gas pressure distribution along the radius of the sealing face is determined by integrating the one-dimensional Reynolds equation for gas lubrication:

\[
\frac{\partial}{\partial r} \left( r x_0 \frac{\partial p}{\partial r} \right) = 0.
\]

For isothermal \( p/p = \text{const} \) pressure flow of gas:

\[
p(r) = \frac{\#_{\text{basis}}}{\#_{\text{basis}}} \left( \psi_1^2 - \psi_2^2 \right) \left( \frac{r_{\text{out}}}{r_{\text{in}}} \right),
\]

where:
\[ \rho_0, p_{\text{in}} \text{ – is the pressure at the inlet and outlet of the operation gap, respectively; } \]
\[ r_{\text{in}}, r_{\text{out}} \text{ – radius at the entrance and exit of the operation gap.} \]

The forces acting on the outer and inner ring sections limited by radii \( r_n, r_{\text{in}}, r_{\text{out}}, r_1 \) are determined by integrating the expression (10) under the appropriate boundary conditions:

\[
F_1 = \frac{2}{3} \left( \psi_1^2 - \psi_2^2 \right) S_1, F_0 = \psi_2 S_0, F_2 = \frac{2}{3} \left( \psi_1^2 - \psi_2^2 \right) S_2,
\]

where:
\[ S_1 = \pi (r_1^2 - r_2^2), S_0 = \pi (r_2^2 - r_3^2), S_2 = \pi (r_3^2 - r_4^2). \]

Taking into account (11), obtained the expression for the force in the operation gap in a dimensionless form:

\[
\varphi_x = \frac{F_1}{\pi b S_0} = \frac{2}{3} \psi_1^2 - \psi_2^2 + \psi_2 \psi_3 + \psi_3 \psi_4 + \psi_4 \psi_5 - \psi_2 \psi_3^2 + \psi_3 \psi_4^2 + \psi_4 \psi_5^2.
\]

Equation of equilibrium of an axially movable ring in dimensionless form:

\[
\psi_3 S_3 + \psi_3 S_0 + \chi = \frac{2}{3} \psi_1^2 - \psi_2^2 + \psi_2 \psi_3 + \psi_3 \psi_4 + \psi_4 \psi_5 - \psi_2 \psi_3^2 + \psi_3 \psi_4^2 + \psi_4 \psi_5^2.
\]

where:
\[ S_0 = 0.5 S_1 + S_0 + 0.5 S_2 \text{ is the base square.} \]
\[ \chi = F_{\text{pr}}/(\pi b S_0) \text{ is the dimensionless force of the pre-compression of the springs.} \]

Taking into account the complexity of expression (13), it is solved numerically with respect to the averaged pressure in the chambers \( \psi_3 \), after which, substituting it into equation (6), find a static characteristic of the gas seal. The barrier gas enters the seal only through supply channels, so the mass flow rate of gas through the seal in one revolution of the shaft is determined by the formula:

\[
Q_{\text{m}} = \pi \kappa (Q_{\text{mb}} t_m + Q_{\text{mt}}),
\]

where:
\[ Q_{\text{mb}} = \frac{a_p (s_x u)^3}{24 \mu R t_m \ln r_{\text{out}}/r_{\text{in}}}, \]
\[ Q_{\text{mt}} = \frac{a_p (s_x u)^3}{24 \mu R t_m \ln r_{\text{out}}/r_{\text{in}}}, \]

– is the gas flow through one supply channel into the sealed machine casing and into the atmosphere at the inner-chamber gap;
\[ r_{\text{mb}} = \alpha_{\text{mb}}/\omega \text{ – the time during which the supply channel passes the inter-chamber interval } \alpha_{\text{mb}} = (2\pi - \alpha_{\text{mb}})/\omega. \]

The volumetric flow rate of the gate gas through the seal per minute (in Nl/min):

\[
Q_{\text{g}} = n Q_{\text{m}} / \rho \cdot 10^3, (15)
\]

where:
\[ n \text{ – is the speed of rotation of the shaft (rpm).} \]

The rigidity of the gas film in the operation gap is the most important characteristic of the seal, which determines its reliability, i.e. the stability of the size of the operation gap under changing operating conditions or unpredictable external influences on the seal (the movement of the rotor in the axial direction during bearing wear, etc.). The dimensionless coefficient of static stiffness of the gas seal is found by differentiating expression (12) by the dimensionless gap:

\[
-\chi = \frac{\partial \varphi_x}{\partial \chi}.
\]

Since the expression for the dimensionless force opening the operation gap has a complex form, it limit ourselves to its numerical differentiation.
The condition for static stability of the seal is the presence of a negative feedback between the adjustable value (operation gap) and external influences (pressure of the sealed medium, spring compression force), therefore, the area of static stability of the seal is determined from the condition:

\[ x_s < 0. \]  \hspace{1cm} (17)

CONCLUSIONS
With the help of the presented methodology, gas-barrier face impulse seals for pumps and compressors of chemical industries were designed and manufactured. Before installing the seals on the machines, they were run-in on an experimental stand in the laboratory of vibration reliability and tightness of centrifugal machines of the Department of General Mechanics and Dynamics of Machines of Sumy State University. A comparison of the calculated and experimental flow characteristics of gas seals has shown that in most cases a good qualitative and quantitative coincidence is found.

At the end of the article, it should be noted that the presented method for calculating the characteristics of gas-sealing seals is based on a well-known approach to the calculation of impulse-type seals, considered in [41, 42]. They assume that the pressure in a closed chamber and the pressure on the section between the chambers differ little from each other, therefore, to determine the pressure on the annular section occupied by the chambers, it is sufficient to consider the balance of flow rate through a single chamber and assume that the same average pressure as in the chamber takes place at any point of this section. From the point of view of the authors, the mentioned simplification is legitimate if the distance between the chambers is small and does not exceed several millimeters, therefore, unfortunately, questions remain open related to the influence of the number of chambers on the nature of the pressure distribution of the barrier medium on the section occupied by the chambers in the circumferential direction.

Recent experimental studies of gas-barrier seals have shown that by varying the number of chambers on the face, it is possible to achieve a significant change in flow rate and static characteristics. Moreover, not only their quantitative, but also qualitative indicators change, including static rigidity. Therefore, the main task of the upcoming research should be considered the creation of a mathematical model of impulse seals, which allows to obtain a more reliable picture of the pressure distribution in the operation gap, taking into account the number and location of closed chambers on the operation gap of an axially movable sealing ring.

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