Influence of feeders on operating characteristics of the impulse seals

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Abstract. The paper presents an refined calculation of impulse face seals with a self-regulated gap. Unlike the existing methods of calculation, wherein, for the sake of simplicity, influence of the feeder conductance is neglected, this article describes conductance effect on the operating characteristics of the seal. Static calculation method of a face impulse seal of the high-speed pump is presented.

1. Design and working principle of the seal
The simplest design of a single-stage impulse seal (Figure 1) differs from the mechanical face seal only by the fact that closed chambers 2 are located on the end face of the axially movable ring 1, and several radial feeding channels 5 are made on the rotating support ring 6, and these channels are opened towards the sealing hole [1]. Sealing medium under sealing pressure $p_1$ is injected through these channels into the chambers during the short time intervals $t_\beta = \beta_c / \omega$ while the rotating channels 5 pass by the chambers 2. At these moments, pressure $p_2$ in the chambers increases in an abrupt manner to a value $p_{2\text{max}} = p_1$.

Figure 1. Scheme of the impulse face seal.
Nature of the pressure change in the chamber over a period of \( T = 2\pi/\omega n_i \) \((n_i\text{-number of feeders})\) between two next injections depends on the hydraulic resistance or conductance \( g_i \) of the feeders and conductance of the internal (on the side of the condensed pressure) \( g_1(z) \) and \( g_3(z) \). Figure 2 (a) [2] shows approximate graphs of pressure variation in a separate chamber: the larger is the gap, the less is \( p_{2\min} \) and the averaged pressure \( \bar{p}_2 \) over a period \( T = 2\pi/\omega n_i \).

A similar picture takes place during the expansion process, when a feeder is outside the sector \( \beta_c \) on a time interval \( T - t_c \). After a feeder leaves the sector \( \beta_c \) occupied by the chamber, pressure \( p_2 \) in the chamber begins to decrease due to the outflow of the compressed medium through the outer flat gap 4; \( A_1, A_2 \) - areas of flat annular gaps walls (Figure 2, (b)). Pressure drop continues until the next injection (Figure 2 (a)), and depth of reduction, i.e. the value \( p_{2\min} \), depends on the face gap: the larger is the gap, the less is \( p_{2\min} \). A combination of parameters wherein pressure reaches the minimum value \( p_3 \) within the time \( t < T = 2\pi/\omega n_i \) is possible. If a gap value decreases, amplitude of pressure changes in the chamber decreases, and the average pressure increases.

**Figure 2.** Pressure changes in a separate chamber during the period \( T \) (a) and steady pressure diagrams on the end faces of the axially movable ring (b) depending on the face gap: \( z' > z \).
The shorter is the time between injections, the lower is the depth of pressure drop $p_{2\text{min}}$ in the chambers, the higher is the averaged pressure $p_2$ in the chambers the greater is the force $F_\omega$ opening the end joint.

Pressure force $F_\omega(z)$ opening the face joint depends on the pressure $\bar{p}_2(z)$ and, consequently, on the gap size. As the gap size is reduced, pressure increases, and its balance with the external force $F_e$ (Figure 3), not depending on the gap size, is upset. Under the influence of positive difference $F_\omega - F_e > 0$ on the axially movable ring, gap size decreases ($\Delta z$) so that the balance $F_\omega = F_e$ is recovered. The sharper the dependence $F_\omega(z)$ drops, the less the new equilibrium position $M'$ deviates from the original position of $M$.

![Figure 3](image_url)

**Figure 3.** Effect of the slope of a curve $F_\omega(z)$ on static deflection under the influence of an external load $F_e$.

Thus, there is a negative feedback between the face gap $z$ (adjustable value) and the force $F_\omega$ (regulating effect), which provides self-regulation of the face gap (Figure 4).

![Figure 4](image_url)

**Figure 4.** Structural scheme of the impulse seal as an automatic control system.

Seal performance is based on the developing of high-frequency pressure impulses in the discharge chambers, so it is called the impulse seal. Impulse frequency of the total axial force of pressure in the
gap, acting on the axially movable ring – \( n_c, n_e \) \( (n_c = 2\pi/\beta_0, \) number of chambers \) is large in comparison with rotation speed. Because of large damping in the micron face gap and inertia of the axially movable ring, the ring hardly reacts to such high-frequency impulses. For these reasons, high-frequency forced oscillations of the ring in the static calculation are not taken into account.

Static characteristic is found from the equation of axial equilibrium of an axially movable ring. In the equilibrium position, pressure force \( F_s \), opening the face gap, balances external pressure force \( F_e \) and force of elastic elements \( F_k = F_e + F_k \). Averaged pressure \( p_2 \) in the chambers over a period between successive injections determines the force \( F_s \). For simplicity, a linear pressure change along the face gap radius is assumed (Figure 5).

![Figure 5. Pressure distribution on the end faces of the axially movable ring.](image)

2. Calculation of the averaged pressure in the chambers

To estimate pressure in the chambers, it is necessary to consider a radial flow of viscous compressible fluid in a flat channel, which looks like a sector with a central angle \( \beta_e \) and a radial size \( r_3 - r_1 \), formed by sealing surface elements and separated by a flow chamber (Figure 1, (b)).

![Figure 6. Segment of a flat channel with a flow chamber.](image)

Right wall of the gap wherein feeders are located, rotates; the left one has a freedom of axial displacement and moves inside the micron-sized face gap. Circumferential component of the flow is not taken into consideration. The flow in the channels is nonstationary. A feeder (shown in dashed
lines), passing over time \( t_c = \beta_c/\omega \) through a liquid-filled chamber, step-like supplies pressure \( p_1 \) to it. As a result, pressure rises to the maximum value \( p_1 \), compressing liquid in the chamber. After a feeder leaves the sector \( \beta_c \), liquid volume compressed in the chamber flows out and pressure decreases to the initial minimum value. The expansion process takes place over a period of time \( T - t_c \). After that, compression takes place again, and the process repeats (Figure 2, (a)).

During compression, difference in the liquid volume inflowing into the feeder and inner gap throttle \( (Q_i + Q_f)(dt) \), and outflowing \( Q_3(dt) \), from the outer one, is compensated by the liquid, which fills the volume \( -dV \) released in the chamber because of previous liquid compression:

\[
(Q_i + Q_f - Q_3)(dt)_c = -dV
\]

(1)

In the process of expansion, volume of the outflowing liquid \( Q_3(dt)_p \) is greater than the volume of the inflowing one \( Q_i(dt)_p \) by the amount \( dV \) with the opposite sign:

\[
(Q_3 - Q_i)(dt)_p = dV
\]

(2)

Equations (1) and (2) of the volume balance differ only by flow rate \( Q_i \), through the feeder and the initial condition: compression begins with the minimum pressure, and expansion with the maximum one.

As defined by [5], the modulus of volume elasticity of a liquid is defined

\[
-E = -V_0 \frac{dp}{dV}
\]

from which

\[
dV = -V_0 \frac{dp}{E}
\]

(3)

Here \( dV = V - V_0 \) - difference between the final \( V \) and initial \( V_0 \) volumes of the liquid, \( V_0 \) - unchanged volume of the chamber, \( E \) - isothermal modulus of liquid volume elasticity; \( Q_r \) - flow rate through the feeder, \( Q_i, Q_3 \) - flow rates outflowing from the internal and external face throttles of the sector \( \beta_c \).

Modulus of water elasticity is \( E \approx 2200 \) MPa. To seal the pump with discharge pressure \( p_1 = 18 Mlta \), \( p_2 \approx 0.5p_1 \) the relative volume of compression is \( \Delta V/V_0 \approx 10^{-2} \), i.e. it is approximately equal to 1% of the chamber volume. Therefore, there is no need to expect a large effect from compressibility on the seal performance. The change of volume \( \Delta V \) during compression and expansion differs only in sign.

Adding to term the equations (1) and (2), we obtain an equation for the entire period \( T \)

\[
Q_i(dt)_c + (Q_i - Q_3)(dt)_c + (dt)_p = \frac{V_0}{E} [dp)_c + (dp)_p]
\]

The sums of incremental components of time and pressure cover the entire period \( T \) between the next injections, therefore, one can indicate: \((dt)_c + (dt)_p = dt, \quad (dp)_c + (dp)_p = dp\):

\[
Q_i(dt)_c + (Q_i - Q_3)dt = \frac{V_0}{E} dp
\]

(4)

Differentials of independent variables are constant quantities:

\[
(dt)_c = t_c - 0 = \beta_c/\omega, \quad dt = \Delta T = T - 0 = 2\pi/n_1\omega
\]

(5)

A pressure differential, accurate to a second-order quantity, can be presented as following:
\[ dp = (dp)_c + (dp)_p \approx \Delta p = \Delta p_c + \Delta p_p = p_{2\ max} - p_{2\ min} + p_{2\ min} - p_{2\ max} = 0 \]

Taking into account the obtained relations, an equation (4) after dividing by \( dt \) represents an approximate, time-independent equation of flow rate balance:

\[ Q_i \frac{d}{dt} + Q_1 - Q_3 = 0 \]  

(6)

For a laminar flow, the flow rate is linearly dependent on pressure drops:

\[ Q_i = g_i(p_1 - p_2), \quad Q_1 = g_1(p_1 - p_2), \quad Q_3 = g_3(p_2 - p_3) \]  

(7)

where

\( p_2 - \) averaged value of pressure in the chambers.

Conductance values of the face throttles for laminar flow are proportional to the cube of the gap \( z \) [3,7] and are expressed by the equations:

\[ g_i = g_{i0}u^3, \quad g_3 = g_3u^3, \quad u = z/\ z_n, \]

\[ g_{i0} = \frac{\beta_c z_n^3}{12\ \mu \ln(r_3/r_2)}, \quad g_3 = \frac{\beta_c z_n^3}{12\ \mu \ln(r_21/r_1)} \]  

\[ \beta_c \approx \beta_{c,1} \approx \beta_{c,2} = \beta_{c,3} \approx \beta_{c,4} \approx \beta_{c,5} \approx \beta_{c,6} \]

(8)

\( z_n = (10...20) \ \mu m \) is a nominal face gap accepted for this device design, \( u = z/\ z_n \) - dimensionless current gap, \( \mu \) - dynamic viscosity of the fluid to be sealed.

Feeder conductance \( g_i \) depends on the shape and dimensions of the cross-section and it is determined experimentally, as a rule. Conductance estimate of an open radial channel can be obtained using the Hagen-Poiseuille equation for tubes of the circular cross section [6]:

\[ g_i = \pi d_i^4 / 128 \mu \]

\[ \mu = \mu_i \]  

(9)

where

\( d_i, l_i \) - diameter and length of an equivalent tubular feeder.

Having substituted a equation for flow rate in (6) and using (5), it is possible to obtain the general equation of balance for volumetric flow

\[ \left( g_i \frac{d}{dt} + g_1 \right)(p_1 - p) - g_3(p - p_3) = 0 \]

from which the averaged pressure in the chambers can be found, depending on the dimensionless face gap:

\[ p = \left( g_i \frac{d}{dt} + g_1 \right)\frac{p_1 + g_3 p_3}{g_i \frac{d}{dt} + g_1 + g_3} \]

Presents the resulting expression to the dimensionless form by introducing the following description:
\[ \psi_2 = \frac{p_2}{p_n}, \quad \psi_1 = \frac{p_1}{p_n}, \quad \psi_3 = \frac{p_3}{p_n}; \quad \alpha_{n1} = \frac{g_{1n}}{G_1}, \quad \alpha_{31} = \frac{g_{3n}}{G_1}, \quad G_1 = g_1 \frac{t_c}{T} = g_1 \frac{B_c}{2\pi} \]

\( p_n \) - nominal value of sealing pressure. As a result, one gets

\[ \psi_2 = \psi_1 + (\alpha_{n1} \psi_1 + \alpha_{31} \psi_3) u^3 \]

\[ \psi_2 = \frac{\psi_1 + (\alpha_{n1} \psi_1 + \alpha_{31} \psi_3) u^3}{1 + (\alpha_{n1} + \alpha_{31}) u^3} \] (11)

3. Regulating effect and hydrostatic stiffness

Regulating effect is a total axial force of pressure influencing on the contact surface of the axially movable ring, which depends on the face gap. Using a linear pressure diagrams shown in Figure 5, we calculate the pressure force \( F_3 \) on the contact surface, opening the face gap (regulatory impact), pressure force \( F_e \) which presses a ring to the support disk (external load) and the force \( F_k \) of the elastic elements (control input):

\[ F_3 = 0.5(p_1 + p_2)A_i + p_2A_2 + 0.5(p_2 + p_3)A_3 = F_{i0} + A_2p_2(u) \] (12)

\[ F_{i0} = 0.5(A_1p_1 + A_3p_3) \]

\[ A_i = \pi(r_2^2 - r_{22}^2) \quad A_2 = \pi(r_2^2 - r_{21}^2) \quad A_3 = \pi(r_2^2 - r_1^2) \]

- effective area of contact relative to pressure \( p_2 \):

\[ F_e = B_1 p_1 + B_3 p_3, \quad F_k = k(\Delta + z) \]

\[ F_e = B_1 p_1 + B_3 p_3, \quad F_k = \pi(r_2^2 - r_{21}^2), \quad B_2 = k(\Delta + z) \]

As it can be seen from expressions (12) and (14), only the component \( A_2p_2 \) of the force \( F_3 \) depends on the face gap value, therefore, this component causes regulatory impact.

The equation of axial equilibrium \( F_e = F_k + F_e \), taking into account the fact that the face gap value is negligibly small in comparison with the preliminary deformation of the elastic elements \( z << \Delta \), after substitution of the forces (12) and (14) takes the form:

\[ A_2p_2 = -0.5(A_1p_1 + A_3p_3) + B_1 p_1 + B_3 p_3 + k\Delta \]

where \( k \) is a reduced coefficient of axial stiffness of the elastic elements. It is necessary to divide this equation term-by-term by \( A_2p_2 \) and pass on to dimensionless forces, introducing the following notations:

\[ \chi = F_k / A_2p_2 = k(\Delta + z) / A_2p_2 \approx k\Delta / A_2p_2 \quad \phi_e = F_e / A_2p_2 = \frac{1}{A} (B_1 \psi_1 + B_3 \psi_3) \]

\[ \phi_2 = \frac{F_2}{A_2p_2} = \phi_{i0} + \psi_2(u) \quad \phi_{i0} = \frac{F_{i0}}{A_2p_2} = \frac{1}{2A} (A_1 \psi_1 + A_3 \psi_3) \]

As a result, we obtain a dimensionless regulatory impact effect:

\[ \psi_2(u) = \frac{B_1 - 0.5A_1}{A} \psi_1 + \frac{B_3 - 0.5A_3}{A} \psi_3 + \chi \] (16)

For further transformations, one can use the obvious equality \( B_1 + B_3 = A_1 + A_2 + A_3 \) (Figure 5), introduce the notation of dimensionless areas \( K, \sigma \), and open the difference \( \phi_e - \phi_{i0} \):
\[ K = \left( B_f - 0.5 A_f \right) / A, \quad \sigma = \left( B_3 - 0.5 A_3 \right) / A = I - K; \quad \varphi_e - \varphi_{st0} = K \psi_1 + (I - K) \psi_3 \]  

(17)

In this case, axial equilibrium equation (16) of an axially movable ring is:

\[ \psi_2(u) = K \psi_1 + (I - K) \psi_3 + \chi \]  

(18)

where \( \psi_2(u) \) is also determined by the equation (11), which is the consequence of a balance equation of the flow rate. Joint solution of a balance equation of the flow rate and a equation of axial equilibrium of an axially movable ring, makes it possible to express a dependence of the gap value on external disturbances \( \psi_1, \psi_3 \) and the driving force \( \chi \), i.e. determine the static characteristics.

The steepness of curve \( F_s(z) \) (Figure 3) is determined by the slope of the tangent, i.e. by means of a derivative \( k_s = \partial F_s / \partial z \), which is called a coefficient of hydrostatic stiffness. In the dimensionless form, stiffness coefficient is expressed by the equation \( \kappa_s = \partial \varphi_s / \partial u \). Since in the expression \( \varphi_s \) (15) only the second additive \( \psi_2(u) \) depends on the gap value, using the equation (11), one obtains

\[ \kappa_s = \frac{\partial \varphi_s}{\partial u} = \frac{\partial \psi_2}{\partial u} = \frac{\alpha_{3i} - \alpha_{3c}}{1 + (\alpha_{3i} + \alpha_{3c}) u^3} \cdot 3u^2 \Delta \psi \]  

(19)

The greater is the steepness \( F_s(z) \), i.e module \( \kappa_s \), the smaller is the gap position deviation from the initial equilibrium state when the external load \( F_s \) changes (Figure 3), the minor is the static error. Thus, in order to improve static characteristics, it is necessary to increase modulus of the hydrostatic stiffness coefficient, i.e. increase conductance \( \kappa_{3n} \) and reduce \( G_i \). Attempts to reduce \( \alpha_{3i} \) are rarely successful, since conductance of the face gap is determined by the micron-sized gap, and conductance of feeders - by the diameter of a channel - 0.4 ... 1.0 mm. Many different designs of throttles have been developed for bearing supports with gas lubrication [8,9]. Such throttles can be also used for impulse seals. Coefficient of hydrostatic stiffness (19) is less than zero, and it provides stability of the equilibrium position. Hydrostatic stiffness makes it possible to estimate minimum value \( \omega_0 \) of the own frequency of axially movable ring vibrations and, if necessary, to escape possible resonance. In the equilibrium position:

\[ \omega_0 = \left( \frac{k + k_s}{m} \right)^{1/2} = \left[ \frac{k}{m} \left( 1 + \frac{A p_n \kappa_s}{k z_n} \right) \right]^{1/2} \]  

(20)

4. Static and discharge characteristics

Static characteristic is a dependence of the steady-state gap value on external disturbances level. It is determined from the joint solution of equations (18) and (11):

\[ K \Delta \psi + \psi_3 + \chi = \frac{\psi_1 + (\alpha_{3i} \psi_1 + \alpha_{3c} \psi_3) u^3}{1 + (\alpha_{3i} + \alpha_{3c}) u^3} \]  

From this equality one can find:

\[ u = \left\{ \frac{\alpha_{3i} [(1 - K) \Delta \psi - \chi]}{[K (1 + \alpha_{3i}) - 1] \Delta \psi + (1 + \alpha_{3i}) \chi} \right\}^{1/3}, \quad \alpha_{3i} = \frac{G_{3n}}{S_{1n}} \]  

(21)

Equation (21) enables to make some important practical conclusions and recommendations.

1. Working range of the seal is limited by positive values of the face gap \( u > 0 \) in the entire variation range of the external influences \( \Delta \psi \) and \( \chi \). In addition, these equations are reduced to inequations:
\[
\frac{1}{1 + \alpha_3} < K < 1, \quad \Delta \psi > \frac{\chi}{1 - K} \tag{22}
\]

The minimum value of the permissible sealed differential pressure, upper value in the indicated range \( K \) (22) is not limited. Operating range does not depend on the conductance of the feeders \( g_i \).

2. In the nominal mode \( \Delta \psi \approx 1 \), parameters included in (21) must provide face gap value close to the optimal one: \( z \approx z_n, u \approx 1 \) It is possible if numerator and denominator of the equation (21) are equal. From this equality, we find a load factor providing nominal gap parameters at \( \Delta \psi \approx 1 \):

\[
K_n = \frac{I + \alpha_{11}}{I + \alpha_{11} + \alpha_{31}} - \chi \tag{23}
\]

The load factor determined by this equation complies with the condition (22) \( K_n < 1 \).

Geometric parameters and preliminary compression force of elastic elements included in (23) are selected from constructive and technological considerations. Required value of the load factor \( K_n \) is provided by the appropriate choice of the internal radius \( r_i \) (Figure 5) of the loading area \( B_1 \) (14). The last one is related to the load factor by the equation (18), whereof it follows:

\[
B_1 = K_n A + 0.5A_i = \pi \left( r_3^2 - r_4^2 \right), \quad r_i = \left\{ \frac{v^2}{2} - \frac{1}{\pi} \left( K_n A + 0.5A_i \right) \right\}^{0.5} \tag{24}
\]

Equation (21) shows that the load factor \( K_n \) and the force of elastic elements \( \chi \) reduce the gap value, and pressure drop increases it. If \( \chi = 0 \), a gap (21) does not depend on the condensed pressure difference and maintains a constant value. Limiting value of the gap when the condensed differential pressure increases is expressed by the equation:

\[
\lim_{\Delta \psi \to \infty} u_{\Delta \psi} = \left\{ \frac{\alpha_3(1 - K)}{K(1 - K)\alpha_3} \right\}^{\frac{1}{3}} \tag{25}
\]

To calculate the flow rate, an expression for the total flow rate through the outer (output) flat annular throttle can be used. The flow rate \( Q_3 = g_3(p_3 - p_1) \) and conductance (4) were determined for a sector with a central angle \( \beta_c \) (Figure 6). For the entire gap, the angle \( \beta_c \) in equations (4) is to be replaced by \( 2\pi \):

\[
Q = \frac{2\pi}{\beta_c} Q_3 = \frac{2\pi}{\beta_c} g_3n p_n u^3 (\psi_2 - \psi_3) \tag{26}
\]

It is necessary to assign it to the basic flow rate equation: \( Q_n = \frac{2\pi}{\beta_c} g_3n p_n \) and the required flow rate in a dimensionless form is obtained as following:

\[
\overline{Q} = \frac{Q}{Q_n} = u^3 (\psi_2 - \psi_3) \tag{27}
\]

We substitute the value \( \psi_2 (u) \) (11) in the last equality:
If the expression for the gap (21) is used, then from equation (27) it is possible to obtain an analytical dependence of the flow rate on the compressed pressure and on the force of the spring preliminary compression. However, this dependence turns out to be too lengthy, so to calculate the flow rate it is more convenient to use a numerical calculation of the face gap by equation (21). A simple estimate is presented by the equation (27) for the nominal gap \( u = 1 \):

\[
\overline{Q} = \Delta \psi \left[ \frac{1 + \alpha_i}{1 + \alpha_i + \alpha_j} \right] \quad (28)
\]

It is necessary to consider an important case of the compressible pressure dependence on the rotor speed. In centrifugal machines with an adjustable drive, pressure to be compressed is proportional to the square of the rotor speed \( p_i = C \omega^2 \) [3]. Here \( C \) is a generalized parameter determined by geometry of the machine flow section and preserving an approximately constant value at all rotational frequencies. In the nominal mode \( p_{in} = C \omega^2 \), therefore

\[
C = \frac{p_{in}}{\omega^2}, \quad p_1 = C \omega^2 = \frac{\omega^2}{\omega_n} = p_{in} \Omega^2, \quad \psi_1 = \frac{p_1}{p_{in}} = \Omega^2;
\]

\[
\Delta \psi = \frac{p_1 - p_3}{p_n} = \left( \Omega^2 - \psi_3 \right), \quad p_n = p_{in}
\]

Thus, in static (21) and discharge (27) characteristics, a replacement \( \Delta \psi = \Omega^2 - \psi_3 \) is to be done:

\[
u = \frac{\alpha_i \left[ 1 - K \left( \Omega^2 - \psi_3 \right) - \chi \right]}{\left[ K (1 + \alpha_i) - 1 \right] \left( \Omega^2 - \psi_3 \right) + (1 + \alpha_i) \chi} \quad (30)
\]

\[
\overline{Q} = \nu^3 \Omega^2 - \psi_3 \left[ \frac{1 + \alpha_i \nu^3}{1 + (\alpha_i + \alpha_j) \nu^3} \right] \quad (31)
\]

In this case, independent external influence is not the condensed pressure drop, but the rotor speed.

5. Calculation procedure and numerical example

Static calculation procedure will be considered using engineering calculation example of an impulse mechanical seal for a high-pressure feed pump of a smaller size and higher rotational speed.

1. Initial data: \( p_3 = 0.9 \text{MPa} \), \( p_n = p_{in} = 18 \text{MPa} \), \( \omega = \omega_n = 6000 \text{r/c} \)

Sealing medium – water, \( \mu = 0.653 \cdot 10^{-3} \text{Pa s} \), \( E=2,2 \cdot 10^4 \text{MPa} \).

2. For design reasons, we choose dimensions of the contact surface: \( r_1 = 0.05 \text{m}, \ r_3 = 0.07 \text{m}, \ r_2 = 0.5(r_1 + r_3) = 0.06 \text{m}, \ r_{21} = 0.057 \text{m}, \ r_{22} = 0.063 \text{m}, \ l_1 = l_3 = 0.007 \text{m} \), and also sizes and number of chambers and feeders: \( n_i = 2, n_c = 18 \), \( V_0 \approx 3 \cdot 10^{-7} \text{m}^3 \), \( d_i = 0.4 \cdot 10^{-3} \text{m} \), \( l_i = 8 \cdot 10^{-3} \text{m} \), \( \beta_s = 2 \pi / n_c = 0.349 \), \( \beta_c = b_c / r_2 = 0.25 \text{ rad} \).

3. To calculate areas of face bands we use the following equations:

\[
A_1 = \pi (r_3^2 - r_2^2) = 2.92 \cdot 10^{-3} \text{m}^2, \quad A_2 = \pi (r_{22}^2 - r_{21}^2) = 2.62 \cdot 10^{-3} \text{m}^2, \quad A_c = 64 \cdot 10^{-6} \text{m}^2.
\]

\[
A_3 = \pi (r_{21}^2 - l_1^2) = 2.35 \cdot 10^{-3} \text{m}^2, \quad A = 0.5(A_1 + 2A_2 + A_3) = 4.9 \cdot 10^{-3} \text{m}^2.
\]

4. We determine conductance of gap throttles (8) and their dimensionless values (10).

Calculation results are given in Tables 1-3 for 3 nominal gap values. It makes possible to evaluate the effect of \( z_n \) on static characteristics.
Table 1. Conductance values of gap throttles

| \( z_n, \mu m \) | \( G_i \)   | \( g_{1n} \)   | \( g_{3n} \)   | \( \alpha_{11} \) | \( \alpha_{13} \) |
|-----------------|----------|----------------|----------------|----------------|----------------|
| 10              | 3.19 \times 10^{-13} | 2.6 \times 10^{-14} | 30             | 36.8          |
| 15              | 9.57 \times 10^{-12} | 1.077 \times 10^{-12} | 8.768 \times 10^{-13} | 8.89          | 10.91         |
| 20              | 2.552 \times 10^{-12} | 2.078 \times 10^{-12} | 3.75          | 4.6           |

5. We calculate coefficients of hydrostatic stiffness (20).

Table 2. Coefficients of hydrostatic stiffness

| \( z_n, \mu m \) | 10 | 15 | 20 |
|-----------------|----|----|----|
| \( \kappa \)    | -0.069 | -0.18 | -0.281 |

\( \kappa_s < 0 \), i.e. the stability condition holds for all 3 values of \( z_n \).

6. We calculate load factor \( K_\alpha \) (23), which provides nominal gap value at the nominal mode, loading area B1 and inner radius r4 (24). To estimate effect of the pre-compression force of elastic elements, we calculate three of its values: \( x = 0.004, 0.01, 0.03 \).

Table 3.

| \( z_n, \mu m \) | \( K_\alpha \) | \( B_1 \) | \( \mu m^2 \) | \( r_\alpha \) | \( \mu m \) |
|-----------------|--------------|----------|--------------|--------------|----------|
| 10              | 0.97         | 0.964    | 0.944        | 6.21 \times 10^{-3} | 6.19 \times 10^{-3} | 6.09 \times 10^{-3} | 0.054 | 0.054 | 0.054 |
| 15              | 0.92         | 0.914    | 0.894        | 5.97 \times 10^{-3} | 5.94 \times 10^{-3} | 5.84 \times 10^{-3} | 0.055 | 0.055 | 0.055 |
| 20              | 0.85         | 0.844    | 0.824        | 5.62 \times 10^{-3} | 5.61 \times 10^{-3} | 5.51 \times 10^{-3} | 0.056 | 0.056 | 0.056 |

7. According to the equations (21) and (27) static and consumption characteristics are constructed (Figures 7, 8).

**Figure 7.** Discharge characteristics for various \( z_n \).
6. Conclusion

The paper deals with a face impulse seal of a high-speed centrifugal pump, which has several advantages over traditional mechanical face seals. The extended analysis described above made it possible to estimate influence of feeder conductance on the static characteristics of the impulse seal. In particular, the steady-state value of a dimensionless face gap under equal external forces is proportional to the cubic root of the relative conductance $\chi_{i\alpha}$:

$$u \sim \sqrt[3]{G_i/g_{in}} \sim \sqrt[3]{g_i n_i \beta_i / g_{in} 2\pi}.$$  

From the derived characteristics it can be seen, that in the set pressure range of the sealed liquid, face gap value slightly differs from the base value, and so optimal conditions of the seal operation are guaranteed.

The main problem of impulse seals is the development of structures and calculation of feeding channels with high hydraulic resistance (with low conductance). Impulse seal performance is significantly affected by the force of preliminary compression of elastic elements. Increasing of this force limits working area and increases static error, so it is necessary to monitor the load value and uniformity of its distribution along the circumference.

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