# Hydrodynamic characteristics of the face seal taking into account lubricant film breakdown, inertial forces and complex clearance form

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**Abstract.** Sealing systems of turbomachinery determine possibilities of intensification of work processes in them. Face seals are most common in high speed turbomachinery. The research objective in this article is improving calculation methods for hydrodynamic parameters of face seals. Numerical and analytical calculation methods for hydrodynamic parameters of face seal with a complex clearance form characterized by tapering and undulation have been developed in the article. The numerical method has been developed using the finite volume method and considers inertial forces. Known formulas are derived from analytical expressions of process fluid leakage through the clearance of a complex form in equating tapering and undulation parameters to zero. The developed methods are applicable for hydrodynamic face seals and for contact face seals.
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**Keywords:** face seal, clearance, tapering, undulation, distribution of pressure, bearing capacity, leakage, lubricant film breakdown

## Introduction

***Face seals (Fig. 1) are the basic type of seals of turbomachinery rotor supports [1].***

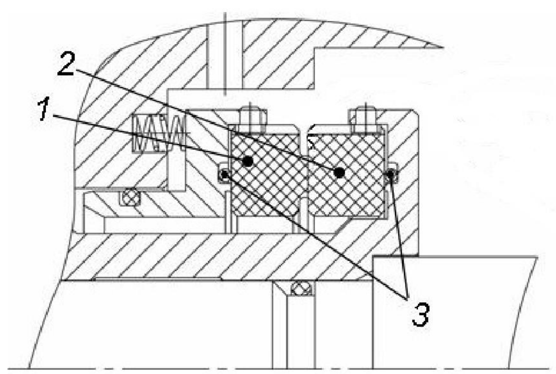


***Fig. 1. Face seal diagram:***

1 – non-rotating ring; 2 – rotating ring; 3 – secondary seals

*Set of issues arising in designing face seals that require comprehensive calculation investigation [2-6]: generation of the hydrodynamic pressure in the clearance, friction in the clearance, power and thermal deformations of the friction couple, friction couple cooling, rotation of the shaft and its axial movements. All these issues are interrelated.*

When designing turbomachinery face seals the main task is to determine hydrodynamic parameters of a lubricant film in the clearance. When analyzing the processes a number of generally accepted allowances are accepted due to the difficulty of initial differential equations that are caused by peculiarities of turbomachinery seal work: liquid is uncompressible, working fluid flow in the gap is laminar, change of the clearance value in the course of time is much less than its nominal value. The actual form of the clearance is determined as follows. Through optimization of geometry of friction couple rings 1 and 2 (Fig. 1) bending moments affecting the rings from the sealed medium pressure are balanced. Thus, it is possible to minimize the clearance tapering value. Thermal and strength tasks are solved successively. First, temperature distribution is determined and then deformations of friction couple rings are calculated using thermal problem solution.

However, despite fundamental researches in this field [1,4,6] calculation methods for such seals have not been developed by now for conditions of their application in high speed turbomachinery supports, in particular, in supports of aircraft engines. Therefore, the purpose of researches in this article is improving calculation methods for hydrodynamic parameters of face seals.

Contact face seals have a hydrodynamic bearing capacity in lubricant film breakdown in the clearance. The possibility of cavitation in hydrodynamic devices was theoretically and experimentally shown by Sergeev S. I. [7]. This theory was further developed for hydrodynamic dampers [8]. At the initial stage the vapour phase can be in the form of small bubbles. Further, vapour generation bubbles become larger and lubricant film breakdown occurs where the medium can enter. Lebeck [1] notes that the cavitation area has hydrodynamic traces, i. e. it is mixed with the fluid-filled areas. The breakdown area has a complex form and in extreme case it can occupy the clearance half-wave length. If pressure increases in the flow boiling is stopped, couples are condensed and the gases gradually dissolve. Cavitation results in pressure redistribution in the film area. As shown by Trampler [9], it can be neglected and we should take the pressure in the lubricant film equal to the pressure obtained based on the solution for the film without cavitation, and in the cavitation area we should take the pressure equal to the saturated vapor pressure. Insignificant loss of accuracy in this case is justified by simplification in task solution. This approach is used in calculation of hydrodynamic dampers of rotor supports. Lebeck [1] used the same approach when developing the analytical method of calculation of the face seal with an undulating surface; however, he significantly simplified the task. He took that the area of lubricant film breakdown occupies the clearance half-wave length and examined only the case when the ambient pressure is equal to the atmospheric one. However, in actual conditions back pressure can take place and the actual clearance has not only undulation but also tapering. Tapering of the clearance is the result of impact of power and thermal loads. Undulation of the clearance can occur due to irregularity in distribution of the pressure and temperature in the circumferential direction. The importance of obtaining the analytical expressions is also conditioned by the fact that use of wide-spread software tools for calculation of face seals based on the finite-element method is limited as hydrodynamic models included into them do not take into account specific features in lubricant thin films.

**Main part**

*First, we shall review the numerical calculation method for hydrodynamic parameters of the face seal.*

To calculate the parameters of the hydrodynamic face seal with microgrooves the mathematical model has been developed based on use of the final volume method [1]. The principle of the method is as follows. The whole volume of face ring clearance is divided into sectors; each sector in its turn is divided into nine control volumes. The example of such a distinguished volume is shown as a cube in Fig. 2.

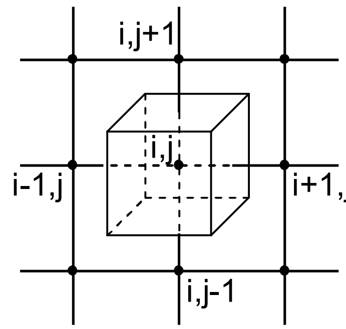


**Fig. 2. Example of section areas division**

Based on the condition of equality of flows through the control volume in the radial and circumferential direction we identify the pressure value in each point.

Taking into account the centrifugal inertial forces the equation for determining pressure $p$ in the point will be as follows [10]

$$p_{i,j} = \frac{B_{i,j}p_{i-1,j} + C_{i,j}p_{i+1,j} + D_{i,j}p_{i,j-1} + E_{i,j}p_{i,j+1} + F_{i,j} + G_{i,j}}{A_{i,j}},$$

with coefficients

$$A_{i,j} = \left(-\frac{h^3}{12\mu r}\right)_{i-1/2,j}\frac{\Delta r}{\Delta\varphi} - \left(\frac{h^3}{12\mu r}\right)_{i+1/2,j}\frac{\Delta r}{\Delta\varphi} - \left(\frac{h^3 r}{12\mu}\right)_{i,j-1/2}\frac{\Delta\varphi}{\Delta r} - \left(\frac{h^3 r}{12\mu}\right)_{i,j+1/2}\frac{\Delta\varphi}{\Delta r};$$

$$B_{i,j} = \left(\frac{h^3}{12\mu r}\right)_{i-1/2,j}\frac{\Delta r}{\Delta\varphi};$$

$$C_{i,j} = \left(\frac{h^3}{12\mu r}\right)_{i+1/2,j}\frac{\Delta r}{\Delta\varphi};$$

$$D_{i,j} = \left(\frac{h^3 r}{12\mu}\right)_{i,j-1/2}\frac{\Delta\varphi}{\Delta r};$$

$$E_{i,j} = \left(\frac{h^3 r}{12\mu}\right)_{i,j+1/2}\frac{\Delta\varphi}{\Delta r};$$

$$F_{i,j} = \frac{r_{i,j}\omega\left(h_{i-1/2,j} - h_{i+1/2,j}\right)}{2}\Delta r;$$

$$G_{i,j} = -\left(\frac{h^3 r}{12\mu}\right)_{i,j-1/2}\frac{3\rho\omega^2}{10}r_{i,j-1/2}^2 + \left(\frac{h^3 r}{12\mu}\right)_{i,j+1/2}\frac{3\rho\omega^2}{10}r_{i,j+1/2}^2.$$

Where $h$ –clearance value, $r$ – radius, $\varphi$[fi]– angle in the circumferential direction, $\rho$[rho] –

density, $\mu$[myu] – dynamic viscosity, $\omega$[omega] – rotor speed.

Initial values of pressure in the clearancee of the seal can be calculated using the iteration method.

Leakage through the seal is determined by summing up the lubricant flows through control volumes in the internal radius of the face seal. At that, we can use known formulas [1].

**The force opening the film sealing joint (film bearing capacity),**

$$W = \int_0^{2\pi} \int_{r_1}^{r_2} prdrd\varphi.$$

Hardness of the lubricant film

$$C = -\frac{dW}{dh}.$$

Loss of capacity for friction in the face clearance

$$N = \mu\omega^2 \int_0^{2\pi} \int_{r_1}^{r_2} \frac{r^3}{h(r,\varphi)} d\varphi dr.$$

Minimum clearance in the seal gap $h_{min}$ is calculated using the balance of closing and opening forces

$$W = F,$$

where force $F$ takes into account the impact of pressure on the rear side of the seal ring and the spring pressing force.

Based on the above equations the software has been developed that allows to obtain basic seal characteristics taking into account pressure redistribution in the area of lubricant film breakdown, centrifugal inertial force and clearance complex form. In the area of lubricant breakdown the pressure equal to the saturated vapor pressure is taken (Fig. 3). Initially the fluid film parameters (viscosity, density) are set at a certain temperature of contacting surfaces of seal rings derived from thermal problem solution. As the lubricant film is thin (~ 1 mym) it is heated and has a temperature of contacting surfaces of seal rings. Then, as the task is solved using the iteration method, we insert the viscosity and density values in each mesh point in relevant pressure and temperature.

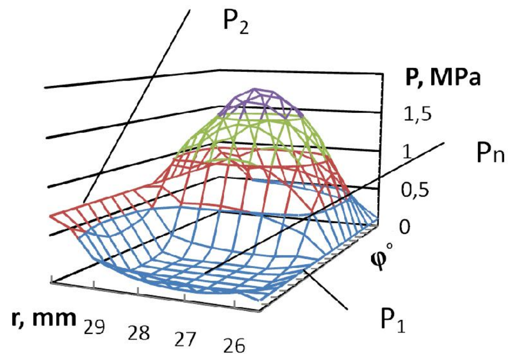


**Fig. 3. Rated distribution of pressure in the face seal gap in the sector with one wave**

This calculation method was successfully applied in calculation of seals of turbomachinery and aviation engine supports [10,11].

*Further, let us consider the analytical calculation method for hydrodynamic parameters of the face seal. We shall consider the face seal the clearance of which is characterized by tapering and undulation (Fig. 4):*

$$h = h(r,\varphi) = h_{\min} + \frac{r - r_1}{r_2 - r_1}\Delta h + e(1 + \cos k\varphi)$$

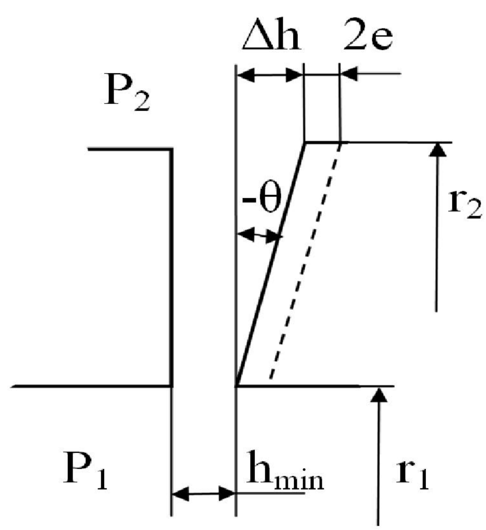


**Fig. 4. Diagram of the face gap characterized by tapering and undulation**

Where $e$ – undulation amplitude, $\theta$ [teta] - tapering angle of the sealing surface. Or having selected the elements depending on $r$ and $\varphi$[fi] the clearance can be expressed as

$$h = h_0 + r tg\theta; \quad h_0 = h_c + e\cos k\varphi;$$
$$h_c = h_{\min} - r_1 tg\theta + e; \quad dh/d\varphi = -ek\sin k\varphi.$$

The Reynolds equation for the case under consideration is as follows [8]

$$\frac{d}{dr}\left(rh^3\frac{dp}{dr}\right) = 6\mu\omega r\frac{dh}{d\varphi}. \qquad (1)$$

Solving it in boundary conditions $r = r_1, p = p_1; r = r_2, p = p_2$, we deduce

$$p = p_1 + \frac{(p_2 - p_1)I_2(r_1, r)}{I_2(r_1, r_2)} + 3\mu\omega r\frac{dh}{d\varphi}\left[I_3(r_1, r) - \frac{I_3(r_1, r_2)I_2(r_1, r)}{I_2(r_1, r_2)}\right], \qquad (2)$$

where

$$I_2(r_i, r_j) = \int_{r_i}^{r_j}\frac{dr}{r(h_0 + rtg\theta)^3} = \frac{1}{(h_0 + r_1 tg\theta)^3}\ln\frac{(h_0 + tg\theta(r_1 - r_i))r_j}{(h_0 + tg\theta(r_1 - r_j))r_i} +$$
$$+ \frac{tg\theta(r_j - r_i)(2h_0 + tg\theta(r_1 - r_i - rj))}{2(h_0 + r_1 tg\theta)(h_0 + tg\theta(r_1 - r_j))^2(h_0 + tg\theta(r_1 - r_i))^2} +$$
$$+ \frac{tg\theta(r_j - r_i)}{(h_0 + r_1 tg\theta)^2(h_0 + tg\theta(r_1 - r_j))\,(h_0 + tg\theta(r_1 - r_i))},$$

$$I_3(r_i, r_j) = \int_{r_i}^{r_j}\frac{rdr}{(h_0 + rtg\theta)^3} = \frac{(r_j - r_i)[h_0(r_i + r_j) + 2tg\theta\, r_i r_j]}{2(h_0 + tg\theta\cdot r_i)^2(h_0 + tg\theta r_j)^2}.$$

Bearing capacity of the fluid film

$$W = k\int_0^{2\pi/k}\int_{r_1}^{r_2} prdrd\varphi. \qquad (3)$$

Only numerical solution of the equation is possible (3). At that, substituting (2) in (3) it is necessary to consider that if the pressure in any point is lower than the saturated vapor pressure $p(r,\varphi)<p_n$, we take $p(r,\varphi)=p_n$.

Analytical solution of this task is possible only in neglecting the seal curving. We shall show the example when such inertial forces are considered.

The initial equation is as follows

$$\frac{d}{dr}\left(\frac{r_m h^3}{12\mu}\frac{dp}{dr} - \frac{3}{10}\frac{\omega^2 h^3}{12\mu}\rho\cdot r_m^2\right) = \frac{r_m\omega}{2}\frac{dh}{d\varphi}.$$

where $r_m$ - seal mean radius: $r_m = (r_1 + r_2)/2$.

Distribution of pressure is as follows

$$p = p_1 + \frac{(p_2 - p_1 - B(r_2 - r_1))I_4(r_1, r)}{I_4(r_1, r_2)} + B(r - r_1) +$$
$$+ E\left[I_3(r_1, r) - \frac{I_3(r_1, r_2)I_4(r_1, r)}{I_4(r_1, r_2)}\right], \qquad (4)$$

where

$$B = \frac{3}{10}\omega^3\rho r_m; \quad E = 6\mu\omega\frac{dh}{d\varphi};$$

$$I_4(r_i, r_j) = \int_{r_i}^{r_j}\frac{dr}{(h_0 + rtg\theta)^3} = \frac{(r_j - r_i)[2h_0 + tg\theta(r_i + r_j)]}{2(h_0 + tg\theta\cdot r_i)^2(h_0 + tg\theta\cdot r_j)^2}.$$

Bearing capacity of the film $W = W_1 + W_2$,

where

$$W_1 = k\int_0^{2\pi/2k}\int_{r_1}^{r_2} pr_m drd\varphi; \qquad W_2 = k\int_{2\pi/2k}^{2\pi/k}\int_{r_1}^{r_2} pr_m drd\varphi. \qquad (5)$$

Substituting (4) in (5) and integrating it we deduce the bearing capacity value of the confusor part of the film by summing two components – static and hydrodynamic:

$$W_{1stat} = \frac{\pi(p_1 + p_2)(r_2^2 - r_1^2)}{4} + \frac{(p_2 - p_1 - B(r_2 - r_1))\pi(r_2 - r_1)^2 r_m tg\theta}{4\left[(h_c + tg\theta\, r_m)^2 - e^2\right]^{1/2}}; \qquad (6)$$

$$W_{1dyn} = \frac{6\mu\omega\mathrm{kr_m}}{\mathrm{tg}\theta^3}\ln\frac{H_{2\max}^{H_{2\max}}H_{1\min}^{H_{1\min}}(H_{1\min} + H_{1\max})^{\Delta h}}{H_{1\max}^{H_{1\max}}H_{2\min}^{H_{2\min}}(H_{2\min} + H_{2\max})^{\Delta h}}, \qquad (7)$$

where

$$H_{1\min} = h_{\min}; \quad H_{2\min} = h_{\min} + 2e; \quad H_{1\max} = h_{\min} + \Delta h;$$
$$H_{2\max} = h_{\min} + \Delta h + 2e.$$

As in formula (7) the denominator has $tg\theta$, if there is no tapering ($tg\theta=0$) we can use the formula for $W$ which is deduced from solving equation (1) without taking into account tapering but taking into account curving of the sealing surface:

$$W_1 = \frac{\pi p_1(r_2^2 - r_1^2)}{2} + \pi(p_2 - p_1)\frac{\left[r_2^2\ln\frac{r_2}{r_1} - \frac{r_2^2 - r_1^2}{2}\right]}{2\ln\frac{r_2}{r_1}} + \frac{3}{16}\mu\omega k\left\{\frac{1}{h_{\min}^2} - \frac{1}{(h_{\min} + 2e)^2}\right\}\times$$
$$\times\left[(\mathrm{r}_1^2 + \mathrm{r}_2^2)\ln\frac{r_2}{r_1} - (r_2^2 - r_1^2)\right]\frac{(r_2^2 - r_1^2)}{2\ln\frac{r_2}{r_1}}.$$

Analysis of formulas (6) and (7) shows that consideration of inertial forces does not affect the hydrodynamic component of the bearing capacity and results in decreased static component of the bearing capacity.

In the confusor part of the clearance

$$W_1 = W_{1stat} + W_{1dyn}.$$

In the diffuser part of the clearance

$$W_2 = W_{2stat} - W_{2dyn}.$$

At that

$$W_{1stat} = W_{2\text{stat}}; \qquad W_{1\text{dyn}} = W_{2\text{dyn}}.$$

Thus, if there is no lubricant film breakdown in the clearance, $W = 2W_{1stat}$, that is pressure increase in the confusor part of the clearance due to hydrodynamics is compensated through pressure decrease in the diffuser part of the clearance. This result shows that if there is no cavitation clearance undulation does not create the hydrodynamic force.

If there is lubricant film breakdown in the cavitation area the pressure $p=p_n$ and the "negative" part of the hydrodynamic bearing capacity in its absolute value becomes less than the "positive" one. In this case the seal has the hydrodynamic bearing capacity.

For convenience of calculations we shall introduce simplification changing the form of the cavitation area (Fig. 3) into the rectangular one (Fig. 5) which is located throughout the length of the diffuser part of the clearance and its radial boundaries $r_{н1}$ and $r_{н2}$ coincide with the boundaries of the cavitation area in its central part.

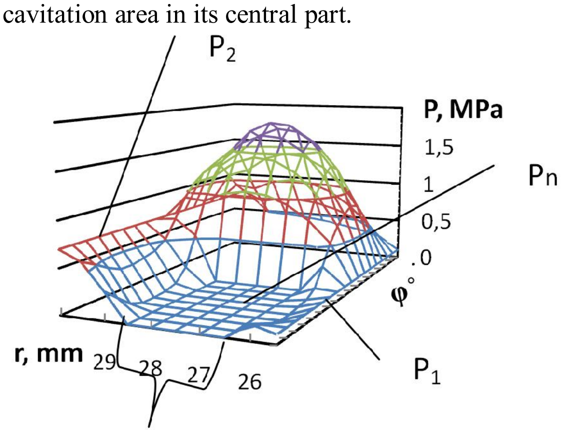


**Fig. 5. Accepted distribution of pressure in the face seal gap in the sector with one wave**

We shall find the boundary of the cavitation area at $\varphi = 1{,}5\pi / k.$ Thus, $E = 6\mu\omega e k, h_0 = h_c$. We shall consider the formula determining the pressure (4) without taking into account inertial forces. With $r = r_{н1}$ and $r = r_{н2}$ the pressure is $p=p_n$ . Substituting this condition in (4) we deduce

$$\frac{p_1 - p_n}{E} - \bar{k} I_4(r_1, r_н) + I_3(r_1, r_н) = 0, \qquad (8)$$

where

$$\bar{k} = \frac{E I_3(r_1, r_2) - (p_2 - p_1)}{E I_4(r_1, r_2)}$$

- constant value for specific geometry and seal working conditions.

Substituting expressions for $I_3$ and $I_4$ in (8). The boundaries of the cavitation area are solutions of the equation

$$a r_н^2 + b r_н + c = 0,$$

where

$$a = h_{\min} + tg\,\theta r_1 - \bar{k} tg\,\theta + 2h_{\min}^2 \frac{p_1 - p_n}{E} tg^2\theta;$$

$$b = 4h_{\min}^2 \frac{p_1 - p_n}{E} h_0 tg\,\theta - 2tg\,\theta r_1^2 - 2\bar{k} h_0;$$

$$c = \bar{k} r_1 (h_{\min} + h_0) - h_0 r_1^2 + 2h_{\min}^2 \frac{p_1 - p_n}{E} h_0^2.$$

Thus,

$$r_{н1} = -\frac{b}{2a} - \sqrt{\left(\frac{b}{2a}\right)^2 - \frac{c}{a}}; r_{н2} = -\frac{b}{2a} + \sqrt{\left(\frac{b}{2a}\right)^2 - \frac{c}{a}}.$$

If the discriminant is

$$\left(\frac{b}{2a}\right)^2 - \frac{c}{a} \le 0$$

there is no cavitation.

To calculate $W_2$ we can use the following formula:

$$W_2 = W_{1cmam}\Bigg|_{\substack{p_1 = p_1 \\ p_2 = p_н \\ r_1 = r_1 \\ r_2 = r_{н1}}} + W_{1cmam}\Bigg|_{\substack{p_2 = p_2 \\ p_1 = p_н \\ r_2 = r_2 \\ r_1 = r_{н2}}} + \frac{\pi p_n (r_{н2}^2 - r_{н1}^2)}{2}. \qquad (9)$$

If there is no tapering

$$r_{н1,2} = r_m - \frac{(p_2 - p_1)(h_{\min} + e)^3}{6\mu\omega ek(r_2 - r_1)} \pm$$
$$\pm \sqrt{\left[\frac{(p_2 - p_1)(h_{\min} + e)^3}{6\mu\omega ek(r_2 - r_1)}\right]^2 - \frac{(h_{\min} + e)^3 (p_2 + p_1 - 2p_n)}{6\mu\omega ek} + \frac{(r_2 - r_1)^2}{4}}.$$

Comparing the data calculated using formula (9) with the numerical solution of formula (3) showed that discrepancy does not exceed 5…7% which is acceptable.

Leakage through the seal can be deduced as follows:

$$Q = k\int_0^{2\pi/k} c_1 d\varphi = k\int_0^{2\pi/2k} c_1 d\varphi + k\int_{2\pi/2k}^{2\pi/k} c_1 d\varphi,$$

where

$$c_1 = \frac{[(p_2 - p_1) - EI_3(r_1, r_2) - B(r_2 - r_1)]r_m}{12\mu I_4(r_1, r_2)}.$$

Total leakage throughout the seal gap is deduced using the following formula:

$$Q = \frac{\pi r_m}{6\mu(r_2 - r_1)}\left(p_2 - p_1 - \frac{3}{20}\rho\omega^2(r_2^2 - r_1^2)\right) \times$$
$$\times \left\{ \frac{tg^4\theta(r_2 - r_1)^4}{16\sqrt{\left[h_{\min} + e + tg\,\theta\frac{(r_2 - r_1)}{2}\right]^2 - e^2}} + \left[h_{\min} + e + tg\,\theta\frac{(r_2 - r_1)}{2}\right]^3 + \right. \qquad (10)$$
$$\left. + \frac{\left[h_{\min} + e + tg\,\theta\frac{(r_2 - r_1)}{2}\right]}{2}(3e^2 - tg^2\theta(r_2 - r_1)^2)\right\}.$$

Using formula (10) we can obtain the leakage equation for simpler cases (plane-parallel clearance, wavelike clearance, tapered clearance) in equating $e$ or $\theta$[teta] to zero. The equations obtained are similar to the known ones [1,6].

When calculating contact face seals the concept of the equivalent clearance is used the value of which is determined by roughness parameters [1]. Such clearance as a rule is characterized by undulation and tapering. Application of the formulas obtained allows to evaluate the hydraulic force in the gap of the contact face seal and to determine an actual contact load.

**Conclusions**

Face seals are the basic type of seals of turbomachinery rotor supports. The developed theory of face seals allows to conduct the comprehensive research of such seals and to optimize their features. Using it we can design and bring both hydrodynamic face seals and contact face seals by the calculation or calculation and experimental method. Due to this time and resource expenditures for processing the seals and, thus, turbomachinery dramatically decrease. The formulas obtained will be useful for both research workers and engineers involved in face seals designing.

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