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PHYSICAL MODEL AND CALCULATION OF FACE PACKING SEALS

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Increasing the reliability and service life of dynamic-pump shaft seals is the most important requirement for their creation. The most common type of seals is still traditional stuffing box seals, which are controlled leakage assemblies that are periodically restored during operation. A radical change in the design of standard stuffing box seals is the transition to face packing seals with a constant pressure against the packing. It is shown that face packing seals can successfully combine the advantages of mechanical face seals and the simplicity and relatively low cost of traditional stuffing box seals. Mechanical face seals, in which one of the packing rings is replaced by a stuffing box packing, have advantages that significantly expand the application of traditional stuffing box seals. A scheme and a physical model of the face packing seal operation are described. During the operation of the seal, the packing is pushed away from the mating metal surface by the pressure of the medium. In this case, a confusor gap is formed, the length of which is proportional to the ratio of the sealed pressure to the pre-compression pressure of the packing. The calculation of the distribution of the hydrostatic pressure and gap along the radius of the face joint of the seal is presented. The irregularity of the contact pressure along the radius, caused by the pressing out of the packing by the sealed inlet pressure, causes premature wear of the overloaded areas of contact surfaces. Expressions are proposed for estimating friction power losses in face packing seals. It is shown that these losses are significantly lower in comparison with the power losses in traditional stuffing box seals. Assessment of the thermal state of face packing seals has been carried out. An expression has been obtained for determining the flow rate that provides the average contact-surface temperature not exceeding the permissible value. Our studies have shown that the load factor of face packing seals, in contrast to mechanical face seals, must be close to unity. The obtained dependencies make it possible to calculate face packing seals at their design stage.

Keywords: *face packing seal, physical model, contact pressure distribution, design features.*

Introduction

The technical level of modern seals is constantly growing due to the tightening of operating requirements, which limit or exclude the external leakage of the medium being sealed. Taking into account that the quantity of operating pumping equipment is growing, leakages through seals result in enormous energy and product pumpage losses. Large costs are spent on neutralization and utilization of leakages of the media being pumped [1].

Increased requirements are imposed on the reliability and service life of seals, especially with account taken of the growth of the equipment unit power, automation of continuous technological processes, in which forced downtime due to failure of seals leads to large economic losses significantly exceeding the costs directly for seals. Replacing and repairing seals requires that significant costs be spent on payments to skilled manual workers and expensive materials. For pumping equipment, especially for the shaft seals of dynamic pumps, this is confirmed by operating experience – up to 70% of pump failures occur due to failure of seals [2].

Any stuffing box seal is an adjustable leakage assembly, that is, the excess of leakage level can be corrected by moving the pressure device without disassembling the pump. Since the increase in leakage is determined by the relatively constant wear of the friction pair over time, the possibility of sudden failures of this design is small. Replacing a stuffing box packing does not require that the pump be removed, and is generally not considered a pump failure. The resource of the assembly is determined by the achievement of the maximum permissible wear of the shaft protection sleeve whose replacement requires that the pump be removed and disassembled. Thus, any stuffing box seal is an assembly that can be adjusted and periodically restored during operation [3].

The most radical change in the design of a traditional stuffing box seal, when constant packing pressure is applied, is to move to a face packing seal. Such a design has the advantages of a mechanical face seal: automatic operation, definite specific loads in contact, good heat dissipation with no need to compensate for shaft runouts and misalignments [4]. During the period of creating first designs of face packing seals, there was no methodology for their calculation and design, which was primarily due to the lack of studies of the processes occurring in the face pair, as well as the complexity of the physical and mechanical properties of the stuffing box packing. Detailed studies of the sealing process, the results of which are outlined in works [5, 6, 7], made it possible to construct its physical model and approach to the creation of a practical method for calculating face packing seals.

Diagram of a Face Packing Seal and its Features

A face packing seal is a mechanical face seal in which one of the sealing rings is replaced by a stuffing box packing (Fig. 1). Sealing is achieved due to the face pre-compression of the packing ring (*3*) located in the axially movable sleeve (*2*) to the support part (*4*). As in mechanical face seals, the pre-compression is carried out by the elastic element (*1*), and during operation, the pressure of the sealed medium is used for this, which allows optimum contact pressure at design conditions by selecting an appropriate load factor. When developing face packing seals, one can use all the best design solutions accumulated by the practice of mechanical face seals. Thus, face packing seals can successfully combine the advantages of mechanical face seals and stuffing box seals with their simplicity and relatively low cost [5].

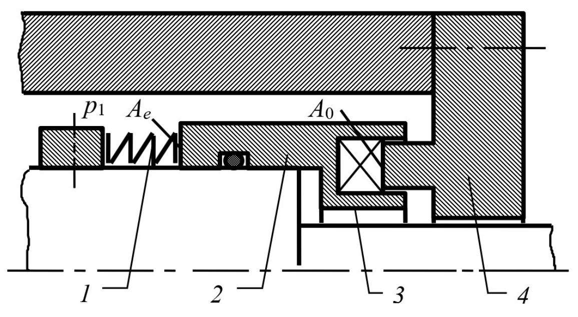

Fig. 1. Diagram of a face packing seal

The friction pair, as in traditional stuffing box seals, is formed by solid metal and soft elastic-plastic surfaces. In traditional stuffing box seals, the frictional surfaces are cylindrical, with the solid heat-conducting surface (shaft or protective sleeve) rotating, and the stuffing box packing (which has poor thermal conductivity) being immovable. In face packing seals, frictional surfaces are flat (annular belts), and, as in mechanical face seals, either the solid or soft surface can rotate. Reduced friction area and improved heat dissipation allow face packing seals to operate reliably at higher load indicators (sealed pressure and rubbing speed p_1v), with small, close to drip, leakages and increased resource compared to traditional stuffing box seals. Face packing seals require tens of times lower packing consumption, since instead of a package set one ring is used, and its resource increases. The service life of traditional stuffing box seals is largely determined by shaft wear, which cannot be effectively compensated by lateral packing deformation. In a face packing seal, the wear of the support ring does not affect the tightness of the assembly, and the magnitude of wear is practically not limited.

Since one of the contact surfaces is a soft packing, the precision machining of friction pairs is not required. It is required for mechanical face seals where the permissible non-flatness of contact surfaces is no more than 0.9 μm. The performance of mechanical face seals is disturbed by the force and temperature deformations of the friction pair, even if their value is within 1–3 μm, and face packing seals are not sensitive to the elastic deformations of structural elements. As in mechanical face seals, the support ring and / or the packed bushing have freedom of axial and angular movements capable of compensating technological and operational misalignments.

Replacing a damaged mechanical face seal requires that the pump be disconnected from the drive, and this disrupts the alignment of the unit. In a face packing seal, the wearing sealing ring is rolled up from a piece of the packing and is either removed or inserted into the annular chamber of the stuffing box sleeve without disassembling the pump.

The practical implementation of the listed features and advantages allows one to significantly expand the scope of using stuffing box seals while maintaining high reliability and tightness.

Physical Model of a Face Packing Seal

The analysis of the results of the experiments described in [6, 7] made it possible to lay the foundations for the construction of a physical model of the sealing mechanism of a face packing seal. During the operation of the seal, the packing is pushed away from the mating metal surface by the pressure of the medium. In this case, a confusor gap is formed, the length of which is proportional to the ratio of the sealed pressure to the pre-compression pressure of the packing. Outside the gap, in the area where the packing contacts directly with the mating surface, the contact pressure increases, and this area plays the main sealing role. Leakages are mainly due to the filtration flow through friction pair microlabyrinths. Taking into account the variety of packings, liquids being sealed, and operating conditions, we will seek an approximate solution to the problem of pressure distribution in the contact pair. When constructing a computational model, the choice of simplifying assumptions should be limited by the requirement that they do not distort the qualitative pressure distribution picture.

Consider the loading diagram of the packing ring of a face packing seal (Fig. 2). When analyzing the stress-strain state, the packing is considered as an isotropic elastic material with physical and mechanical characteristics (modulus of elasticity and Poisson's ratio), independent of the loading value.

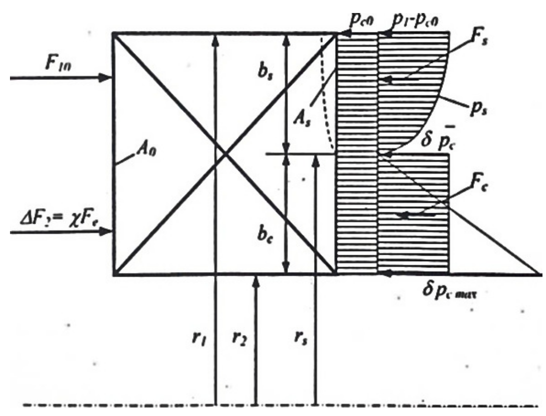

Fig. 2. Loading diagram

The external loading of the packing is carried out in two stages: the pre-compression by the force F_{10} of elastic elements during the installation of the seal into the pump and the final load by the force of the pressure of the liquid being sealed. The force F_{10} creates the pre-contact pressure $p_{c0} = F_{10} / A_0$.

From the condition of compatibility of the axial deformations of elastic elements and the packing, there has been determined that part ΔF_2 of the pressure force $F_e = p_1 A_e$ of the liquid being sealed which is transmitted to the packing and is balanced by the hydrostatic pressure force F_s in the gap and the additional contact pressure force F_c:

$$\Delta F_2 = \chi F_e = F_s + F_c \,, \tag{1}$$

where $\chi = k_2/(k_1 + k_2)$ is the transfer coefficient, or the basic loading coefficient that shows how much of the external force F_e is transferred to the packing; k_1, k_2 are the coefficients of stiffness of the elastic elements and packing, respectively.

From equality (1), after calculating the force F_s, we find the contact pressure force F_c that arises at the area of direct contact.

The hydrostatic pressure in the gap is determined by simultaneously solving the axial strain equation for the packing and the radial pressure flow equation. When calculating the deformations of the packing, it should be borne in mind that the packing is pre-compressed by the force F_{10}, and is a pre-loaded element. The deformation of the packing, or the size of the gap is determined by the formula

$$h(r) = b(p_s - p_{c0})/E \,, \tag{2}$$

where $b = r_1 - r_2 = b_s + b_c$ is the height of the packing ring.

Thus, apart from the presence of microchannels, the formation of the gap is possible only when the pressure p_1 at the inlet of the seal exceeds the contact pressure p_{c0} due to the force of the pre-compression of the packing. As the hydrostatic pressure p_s decreases, the gap narrows, and when it becomes equal to the contact pressure p_{c0}, the packing contacts with the mating surface (section b_c).

Calculation of Face Packing Seals

The flow rate of the radial pressure flow through the flat annular channel with a gap h and length dr, at which the pressure dp_s is throttled, can be represented by the Hagen-Poiseuille formula

$$Q = \frac{\pi h^3 r}{6\mu dr} dp_s \,,$$

and, taking into account (2), we arrive at the differential equation

$$Q\frac{dr}{r} = \frac{\pi E b^3}{6\mu}\left(\frac{p_s - p_{c0}}{E}\right)^3 \frac{dp_s}{E} \tag{3}$$

with boundary conditions $r=r_1$: $p_s=p_1$; $r=r_s$: $p_s=p_{c0}$ (the liquid being sealed is supplied from the side of the outer radius).

Integrating (3), we get

$$Q\frac{24\mu}{\pi E b^3}\ln\frac{r_1}{r}=\psi_1^4-\psi^4, \quad Q\frac{24\mu}{\pi E b^3}\ln\frac{r_1}{r_s}=\psi_1^4 \tag{4}$$

where $\psi=(p_s-p_{co})/E, \quad \psi_1=(p_1-p_{c0})/E.$

From the second expression (4), we can find the flow rate if we know the length of the gap (radius)

$$Q=\frac{\pi E b^3}{24\mu\ln\frac{r_1}{r_s}}\psi_1^4 \tag{5}$$

or, using the measured flow rate, we can determine the gap length

$$r_s=r_1\left(1+\frac{\pi E b^3}{24\mu Q}\psi_1^4\right)^{-1}.$$

If we exclude the flow rate from formulas (4) by dividing the first expression into the second, then we obtain the law of pressure distribution in the gap

$$\psi=\psi_1\left(1-\ln\frac{r}{r_1}\bigg/\ln\frac{r_s}{r_1}\right)^{1/4}.$$

Since the ratio of the radii under the signs of the logarithms is close to unity, then in the expansion of the logarithms in a series we retain only the linear terms $\ln r/r_1 \cong r/r_1-1$. Then the formula for the pressure distribution also will take a more convenient form for integration

$$\delta p_s=p_s-p_{c0}=E\psi_1\left(\frac{r-r_s}{b_s}\right)^{1/4}, \tag{6}$$

where b_s=r_1–r_s is the width of the annular strip on which a gap is formed between the packing and support ring.

Substituting the obtained pressure into formula (2), we find the law of variation of the gap along the radius

$$h(r)=b\frac{p_1-p_{c0}}{E}\left(\frac{r-r_s}{b_s}\right)^{1/4}. \tag{7}$$

Having integrated the pressure (6) over the gap, we obtain the hydrostatic force F_s, partially balancing the external load $\Delta F_2=\chi F_e$

$$F_s=1{,}6\pi r_s b_s(p_1-p_{c0})\left(1+\frac{5}{9}\frac{b_s}{r_s}+\frac{5b_s}{4r_s}\frac{p_{c0}}{p_1-p_{c0}}\right).$$

Now, from the equilibrium condition (1), we can to find the additional contact pressure force F_c acting on the annular belt b_c=r_s–r_2 where the packing contacts with the support ring

$$F_c=\chi F_e-F_s,$$

as well as the additional medium contact pressure

$$\delta\overline{p}_c=F_c\ /\pi\left(r_s^2-r_2^2\right).$$

After some transformations, we reduce the last expression to the form

$$\delta\overline{p}_c=\frac{p_1}{1-\alpha}\left[k\chi-0{,}8\alpha\left(1-\frac{p_{c0}}{p_1}\right)\right], \tag{8}$$

where $\alpha=A_s/A_0, \quad A_s=2\pi r_s b_s, \quad k=A_e/A_0$.

The total contact pressure in the packing at the area of direct contact

$$\overline{p}_c=p_{c0}+\delta\overline{p}_c.$$

An approximate estimate of the maximum additional contact pressure can be obtained if we assume that the pressure $\delta\overline{p}_c$ at the area of direct contact varies linearly from zero to the maximum value $\delta p_{c\max}$ (Fig. 2).

The used simplified approach based on the analysis of uniaxial deformations does not allow constructing the contact pressure distribution along the radius. For this, it is necessary to consider the static problem of hydroelasticity, taking into account the volumetric stress-strain state of the packing. This problem is solved by numerical methods [8].

Formula (8) makes it possible to estimate the value of additional contact pressure and its irregularity, and also gives a qualitative idea of the influence of the main parameters of the seal on the working conditions of the packing.

Let us carry out an estimative calculation of the distribution of hydrostatic pressure and gap along the radius of the face joint of a face packing seal. For example, consider a face packing seal with typical parameters: r_1=0.06 m, r_s=0.05 m, $\mu=5\cdot10^4$ N·c/m^2, E=300 MPa, p_1=1.6 MPa, α=0.5, k=0.9, χ=0.8. The results of calculations, using formulas (6) and (7), are shown in Fig. 3. The flow rate calculated by formula (5) is $3\cdot10^{-7}$ m^3/s, which is close to the experimentally obtained values $Q=2\cdot10^{-7}$–$3\cdot10^{-7}$ m^3/s [6].

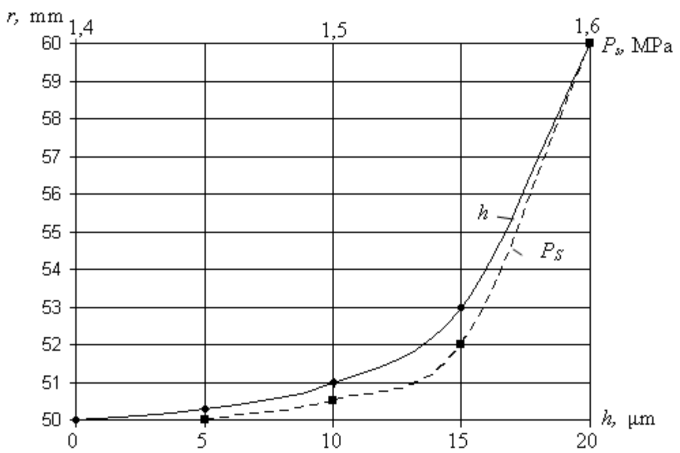

Fig. 3. Distribution of hydrostatic pressure and gap along the face joint radius

The irregularity of contact pressure along the radius, caused by the pressing of the packing by the sealed pressure at the inlet section, causes premature wear of overloaded areas, which is why, for packing end seals, the problem of equalizing the contact pressure along the width of the sealing belt remains relevant [7].

The results obtained make it possible to calculate the friction power loss in the contact pair in the area of direct contact

$$N_c = N_{c0}\left[1+\frac{p_1}{p_{c0}}(k\chi-0.8\alpha)-0.2\alpha\right], \quad N_{c0}=0.5\pi f p_{c0}\omega b(r_1+r_2)(r_s+r_2). \tag{9}$$

The calculations show that the power of fluid friction is an order of magnitude less than the losses in the contact area [9].

In the example considered above, at the rotor speed ω=300 s^{-1} and the friction coefficient f=0.02, the calculation using formula (9) gives N_c=0.478 kW. For comparison, note that in a traditional stuffing box seal with three packing rings, for the same parameters, the calculated friction power losses are 2.15 kW, i.e. four and a half times the loss in a face packing seal. Substantial energy savings are another important benefit of face packing seals.

It can be seen from formula (9) that, in addition to the friction coefficient and peripheral speed, the friction power is significantly influenced by the pre-compression pressure of the packing and the pressure of the liquid being sealed, as well as the load coefficient k and the main load coefficient χ. The calculated power can be used to evaluate the thermal state of a face packing seal, just as it is done for mechanical face seals.

An approximate estimate of the temperature in the friction zone can be obtained if we neglect the heat removal through the surfaces of the supporting disk and the stuffing box packing, i.e., if we assume that heat removal is carried out only by leakages through the face packing seal

$$N_Q = \rho c Q \Delta t, \tag{10}$$

where ρ, c are the density and specific heat capacity of the liquid being sealed, Q is leakages, Δt is the increment of the average temperature of contact surfaces with respect to the temperature of the fluid at the friction zone inlet area. Comparing (9) and (10), we get

$$\Delta t = \frac{N_c}{\rho c Q}. \tag{11}$$

Unfortunately, the flow rate through a face packing seal, which is included in (11), cannot be estimated theoretically, which is why in design calculations, it has to be taken on the basis of the experience of operating such seals in similar conditions. More often, the inverse problem is solved: from equality (11), the flow rate Q_* is determined, which is necessary for the average temperature on the contact surface not to exceed the permissible value Δt_* to prevent the packing from burning.

The results obtained give a qualitative idea of what is happening on the friction surface of the packing, and allow a more reasonable approach to the development of reliable and cost-effective face packing seals.

The most important parameter that determines the design and its performance is the load factor (unlike for mechanical face seals), its value, as a rule, should be close to unity. Two areas of sealed pressures can be distinguished

– low pressure area <0.5 MPa; in this area, reliable sealing is ensured by load factors k=0.9–1.1;

– high pressure area> 0.5 MPa; in this area, the load factor must be k>1.

Conclusions

The scheme and physical model of a face packing seal, which made it possible to explain its main features, are described. During the operation of a face packing seal, the packing is pushed away from the mating metal surface by the pressure of the medium. As a result, a confuser gap is formed. Its length is proportional to the ratio of the sealed pressure to the pre-compression pressure of the packing. Outside the gap, in the area of direct contact of the packing with the mating surface, the contact pressure increases, and this area plays the main role in sealing. Leakages are mainly due to the filtration flow through irregular micro-labyrinths in the friction pair.

An analysis of the distribution of hydrostatic pressure and gap along the radius of the face joint of a face packing seal is carried out. Expressions for determining the friction power losses in face packing seals are obtained and compared with the power losses in the traditional design. Dependencies are proposed to conduct the thermal calculation of face packing seals.

The results of theoretical studies can serve as a basis for creating methods of calculation of face packing seals.

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Фізична модель і розрахунок торцового сальникового ущільнення

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Підвищення надійності і ресурсу ущільнень валів динамічних насосів є найважливішою вимогою під час їх створення. Найбільш поширеним типом ущільнень залишаються традиційні сальникові ущільнення, які являють собою вузли з регульованим витоком і періодично відновлюються в процесі експлуатації. Радикальною зміною конструкції традиційного сальникового ущільнення є перехід до торцового сальникового ущільнення з постійним тиском на набивання. Показано, що торцові сальникові ущільнення можуть успішно поєднувати в собі переваги механічних торцових ущільнень із простотою і порівняно низькою вартістю традиційних сальникових. Механічне торцове ущільнення, в якому одне з ущільнюючих кілець замінено сальниковою набивкою, має переваги, які суттєво розширюють сферу застосування традиційних сальникових ущільнень. Описано схему і фізичну модель роботи торцового сальникового ущільнення. В процесі роботи ущільнення набивка відтісняється від відповідної металевої поверхні тиском середовища. При цьому утворюється конфузорний зазор, протяжність якого пропорційна відношенню тиску, що ущільнюється до тиску попереднього стиснення набивки. Наведено розрахунок розподілу гідростатичного тиску і зазору по радіусу торцового стику ущільнення. Нерівномірність контактного тиску по радіусу, що обумовлена віджимом набивки ущільнюваним тиском на вхідній ділянці, викликає передчасний знос перевантажених областей контактних поверхонь. Запропоновано вирази для оцінки втрат потужності на тертя в торцовому сальниковому ущільненні. Показано, що ці втрати істотно нижче порівняно з втратами потужності тертя в традиційному сальниковому ущільненні. Проведено оцінку теплового стану торцового сальникового ущільнення. Отримано вираз для визначення витоку, який забезпечує середню температуру на контактній поверхні, що не перевищує допустимого значення. Дослідження показали, що коефіцієнт навантаження торцових сальникових ущільнень, на відміну від механічних торцових ущільнень, повинен бути близький до одиниці. Отримані залежності дозволяють виконувати розрахунок торцових сальникових ущільнень на етапі їх проектування.

Ключові слова: *торцове сальникове ущільнення, фізична модель, розподіл контактного тиску, особливості проектування.*

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