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Computational studies of stuffing box packing seal wear mechanism using the Archard model

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Abstract

The wear model of the stuffing box packing seal, particularly the surface of the protective sleeve of the shaft, is presented. Modeling was performed using the ANSYS software, using the Transient Structural module which have the built-in Archard wear model. The wear model was validated in accordance with the results of previous experimental studies considering the effect of applied pressure. It was determined that when the degree indicators in the Archard equations are m = 1.5 and n = 1.3, the proposed wear model allows obtaining sufficiently accurate wear intensity values. The values of relative wear for different materials of protective sleeves are compared. Graphs of the contact pressure distribution along the width of the friction pair at different values of the applied pressure and linear rotation shaft velocities are given. The results of the shaft surface shape change due to the loss of material were obtained. The resulting change in shape is considered idealized, since this study does not consider the presence of abrasive inclusions in the medium, uneven pressure distribution on the packing gland, temperature changes, etc. However, this study can be useful in predicting the wear of the main components of the stuffing box packing seal.

Key words: wear, Archard model, stuffing-box packing seal, contact pressure distribution, material volume loss.

Introduction

The wear of stuffing box packing seals is a long-standing problem in mechanical engineering, because of the dynamic behavior, they are subject to various tribological interactions, such as friction, wear and corrosion. This can significantly affect the service life. Thus, solving the problem of the wear of packing seals is of great importance for increasing the reliability and efficiency of centrifugal machines.

The peculiarity of the stuffing-box packing seals work is that during the service life the surface of the packing does not have time to undergo a direct process of wear. In the process of operation, it gradually loses its sealing properties: it decreases in size under the influence of the applied pressure and the packing gland, due to which the clearance between the packing and the shaft increases; the impregnation of the packing is washed out, due to which the friction coefficient increases, and as a result, this is accompanied by an increase in heat generation. Thus, the replacement of the packing ring is performed before its fibers are rubbed.

Therefore, it is the shaft or the protective sleeve of the shaft that undergoes direct wear in the stuffing-box packing seal. And therefore, the amount of wear of the surface of the shaft or sleeve, as in the case of [0], can be determined experimentally by weighing.

Literature review

Currently, research is being actively conducted on the calculation of the service life of mechanical seals, since they are widely used for centrifugal machines sealing. The main parameter in determining the durability of sealing elements is the degree of wear of the contact pair surfaces. Firstly, researchers are focused on the study of the wear of seal surfaces under the influence of various parameters for each individual case. For example, the influence of temperature deformations [0], taking into account the combined abrasive and adhesive wear [0], taking into account the uneven wear of surfaces [0], the influence of contact state that changes over time [0], or the effect of elastohydrodynamic lubrication [0-0]. The choice of materials for the friction pair is also analyzed, in particular based on carbon, which significantly affects the service life of the seal [0, 0].

Stuffing-box packing seals are one of the types of mechanical seals. They play a key role in various engineering systems, where reliable sealing of liquids is required. These seal designs are installed in centrifugal machines and prevent leakage of liquids or gases, ensuring the tightness of the system. The importance of the durability of these seals cannot be overstated, as any failure can lead to leakages, increased maintenance costs and environmental issues.

Although previous research has made significant progress in understanding wear mechanisms and its modeling in mechanical seals, there remains a notable gap that applies to stuffing box packing seals. Most likely, this is due to a significantly lower resource of packing seals compared to traditional mechanical seals. Therefore, this direction of research has prospects, since new modernized designs of radial and face packing seals are being created [0]. These designs are characterized by an increased service life. Therefore, the creation of a wear model of the stuffing box packing seal can contribute to increasing the service life of the seals.

Previous studies have demonstrated the importance of wear modeling in understanding and predicting contact pair wear. Computational tools such as finite element analysis (FEA) and computational fluid dynamics (CFD) are used to model wear behavior in mechanical seals. In general, when solving wear-contact problems, most researchers rely on the Archard wear model, including studying wear in mechanical seals. In our case, ANSYS software stands out as a reliable computational tool that offers the ability to simulate wear based on Archard's wear law and predict its effect on seal performance.

The Archard wear model, proposed by Archard in 1953, is the fundamental basis for predicting wear in tribological systems. This model quantifies wear as the proportional loss of material volume due to sliding or relative movement between surfaces. The Archard equation (1), which includes the wear coefficient (K), the applied load (P), and the sliding distance (S), is widely used to predict the wear volume loss. The simplicity and versatility of this model make it a valuable tool for wear analysis, especially in seals.

$$W = K \times S \times P \tag{1}$$

The current study is aimed at providing information on the wear rate of the contacting surfaces of stuffing-box packing seals, focusing on the numerical modeling of this phenomenon in a centrifugal pump. However, it does not include taking into account the features of manufacturing and metalworking of individual sealing components or comprehensive experimental verification, which remain the object of further research.

Purpose

The purpose of this work is to create a computer model of the surface wear of the stuffing box packing seal, to compare the obtained results with the experiment, and to use this model to solve the wear-contact problem considering the Fluid-Structure Interaction (FSI) problem of the stuffing box packing. This model will make it possible to determine the contact pressure distribution more accurately when changing such parameters as the applied pressure and the shaft rotation speed. In addition, it is planned to investigate the change in the contact pressure distribution during the wear of the contacting surfaces, to determine the amount of wear and, as a result, the change in the shape of the shaft contact surface. The obtained results in general will make it possible to predict the resource of the main components of stuffing box packing seals more accurately. They will also become the basis for further design modernization of stuffing box packing seals.

1. Materials and methods

1.1. Validation of the radial stuffing-box packing seal wear model

In the ANSYS software, the modified Archard equation (2) is used, where $\dot{W}$ – wear rate, K – dimensionless wear coefficient, H – hardness of the material being worn, v – sliding speed, P – applied pressure.

$$\dot{W} = \frac{K}{H} \times v \times P \tag{2}$$

The most interesting is the wear coefficient *K*, which can be determined only as a result of conducting an experiment. So, at work [0] given values of wear intensity (mg/h), applied pressure (MPa), sliding speed – 3 m/s. The material of the protective sleeve is stainless steel, the hardness of which can be considered 1/3 of the yield strength (σ_y) [0]. For stainless steel, the value of the yield point is 207 MPa. Therefore, the value of hardness is equal to (3)

$$H = \frac{\sigma_y}{3} = \frac{207}{3} = 69 \text{ (MPa)} \tag{3}$$

Then, from equation (2), we can express the coefficient K (4), the values of which are given in table 1:

$$K = \frac{\dot{W}H}{vP} \tag{4}$$

Also, an important point in solving the wear problem is finding the correspondence between the experimental wear intensity (J) and the Archard wear rate (W). The correspondence is given in formula (5):

$$\dot{W} = J\frac{10^{-6}}{\rho \cdot 3600}, \tag{5}$$

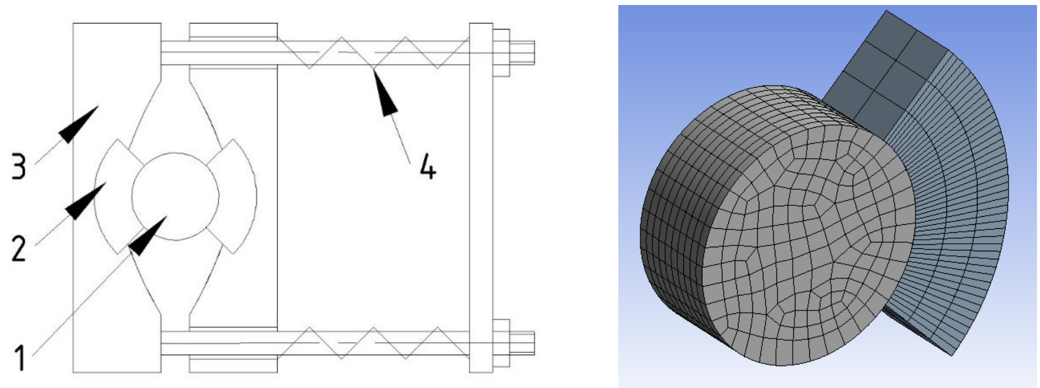
where ρ – the density of the sleeve material.

A three-dimensional model of the experimental rig was created for conducting investigations (Fig. 1) [0]. The rig consists of two pads 3, into which pre-pressed samples of stuffing box packing 2 are inserted. The samples are inserted in such a way that their position does not change during the rotation of the shaft with a protective sleeve 1. The shaft is driven into rotational motion by a motor that can change the rotation speed from 640 to 3200 rpm. The force of the applied compression is regulated by the compression of the springs (4) in the range of 10-150 N.

Table 1

Values of experimental wear intensity [0], wear rate and wear coefficient depending on the applied pressure

The intensity of wear (experimental) (mg/h)	Applied pressure (MPa)	Wear rate (experimental) ($m^3/s*10^{-14}$)	Wear coefficient (10^{-8})
0.21	0.08	0.752	0.706
0.67	0.12	2.401	1.503
0.93	0.17	3.333	1.472
2.44	0.25	8.746	2.627
3.68	0.34	13.190	2.910
7.00	0.46	25.090	4.096



a b

Fig. 1. Scheme (a) and three-dimensional model of the rig (b)

The model has two main components: a shaft with a diameter of 30 mm and a thickness of 13 mm, and a stuffing-box packing sample with a square cross section of 13x13 mm. Shaft material – stainless steel 30Ch13 (Young's modulus – 2.1 GPa, Poisson's ratio – 0.31), stuffing-box packing material – asbestos fluoroplastic AFT (Young's modulus– 50 MPa, Poisson's ratio – 0.45).

A frictional contact model was used to simulate wear. The contact surface is the surface of the sleeve, the surface of the stuffing box packing is the target surface. The coefficient of friction between the surfaces is 0.04. The contact problem was solved based on the Augment Lagrange formulation.

When creating a finite element mesh, it is important to determine the number of elements in both bodies. It is necessary to achieve a balance between accuracy and calculation time. The view of the finite element mesh is presented in fig. 1, b. The number of elements was equal to 4290, nodes to 22158.

The following boundary conditions are presented: external pressure applied to the surface of the stuffing box packing and the shaft rotation linear velocity - 3 m/s (Fig. 2). At the same time, both solid bodies are constrained to move along the y axis, and the shaft surface has the ability to rotate around its own axis of rotation.

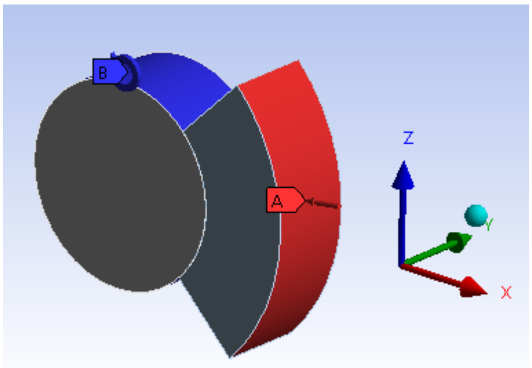


Fig. 2. Boundary conditions

To determine the wear of the shaft, the built-in command (6) was used, which uses the modified Archard formula (7). Where the coefficients *m* and *n* are degree indicators at pressure and velocity, respectively.

TB,WEAR,1,,,ARCD
TBDATA,1,K,H,m,n. (6)

$$\dot{W} = \frac{K}{H} \times P^m \times v^n. \quad (7)$$

In this study, it is assumed that the wear process occurs in the isothermal regime. This assumption assumes that the working medium flowing through the seal completely removes excess heat.

1.2. Wear of the shaft surface, considering the Fluid-Structure Interaction problem of the stuffing box packing

Determining the amount of shaft surface wear is very important for radial stuffing box packing seal. As is known, the highest level of wear is observed in the place of increased contact pressure in the friction pair. The рисунокwordcontact pressure in a face packing seal is considered. Here, in addition to the direct wear-contact problem, the FSI problem is additionally solved. A similar picture is characteristic of the radial stuffing box packing seal. In fig. 3 shows the effect of hydraulic pressure on the stuffing box packing. At the same time, part of the stuffing box packing from the applied pressure side is pressed against the surface of the shaft, and the other part at the exit from the seal is in contact with it. Therefore, in addition to contact forces, it is critically important to consider the action of hydroelastic forces to solve the wear-contact problem of the stuffing box packing seal. The algorithm flowchart for solving following problem is presented in fig. 4.

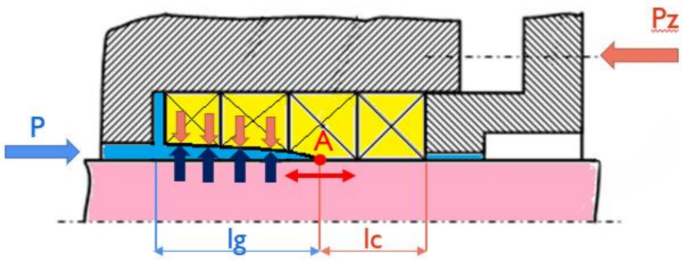


Fig. 3. FSI consideration in stuffing-box packing seal. P – applied pressure; Pz – pressure acting on the sleeve; lg – zone with a significant clearance; lc – contact zone; A – transition point between the zones, which changes its position along the package depending on the operating conditions

The simulation was carried out by changing the pressure of the working medium and the shaft rotation speed. The values of operating parameters are presented in Table 2.

Table 2

Operating parameters

Parameter	Value				
Applied pressure (MPa)	0.2	0.4	0.6	0.8	1.0
Linear velocity of shaft rotation (m/s)	1	2	3	4	5

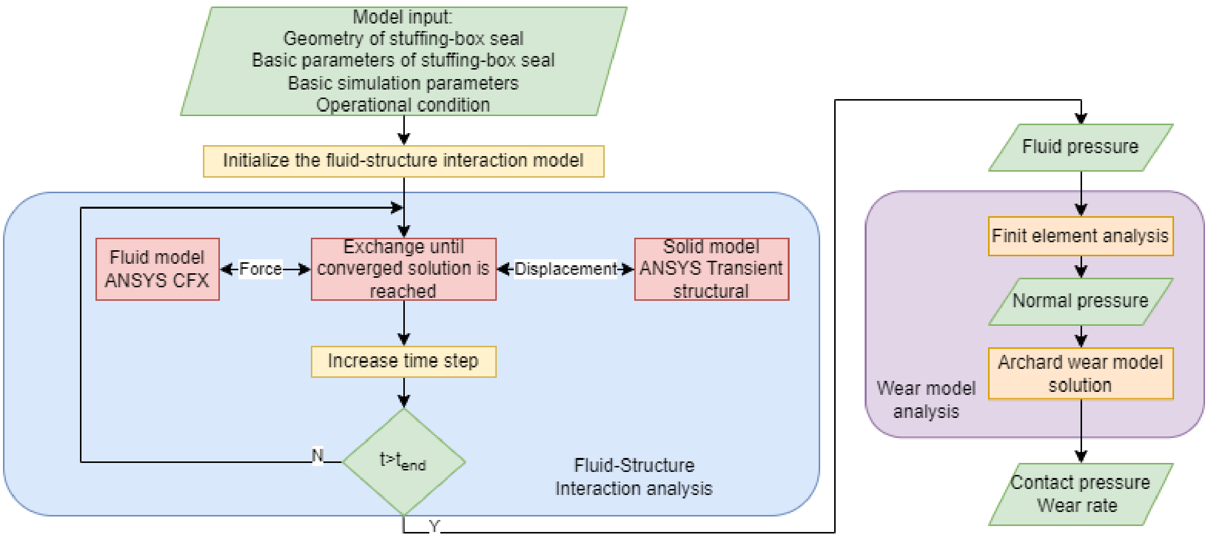


Fig. 4. Algorithm flowchart for solving the wear-contact problem of stuffing box packing seal

Results

2. Results of wear model validation

For the validation of the stuffing-box packing seal wear model, numerical studies were performed with different combinations of *m* and *n* degree indicators. In the result, it was determined that with the values of the degree indicators m=1.5 and n=1.3, the proposed wear model coincides with the experiment. The results of the research are presented in fig. 5, which shows the graph of the dependence of the volume loss rate on the applied load. The graph shows that the values of the wear volume at the corresponding loads, obtained during modeling, generally coincide with the experimental data. The deviation does not exceed 10%.

In addition, it should be noted that in work [0], the amount of relative wear of a sleeve made of different materials and with different hardening methods was also experimentally investigated. Since in this work there is no information about specific values of the hardness of the shaft, but only the difference in the intensity of wear is shown. Therefore, the hardness values were selected from reference literature [0]. A comparison of the results of simulation and experiment in the form of the relative wear for different materials of the shaft is shown in fig. 6. The unit was taken as the value of the relative wear indicator for the material of the shaft made of cast iron 18. After analyzing the obtained dependence, it can be concluded that although the values of relative wear for the calculation model are underestimated in comparison with experimental data, the tendency to decrease the amount of wear for different types of materials in the sequence shown in Figure 6.

Thus, it can be stated that Archard's model can be applied in studies of shaft wear in stuffing-box packing seals.

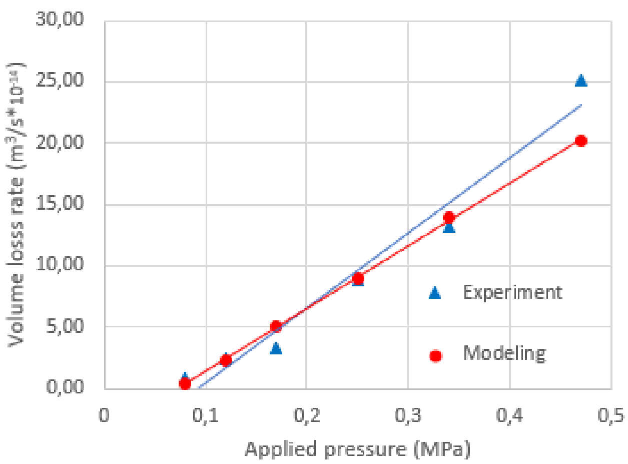


Fig. 5. Dependence of volume wear on the applied pressure

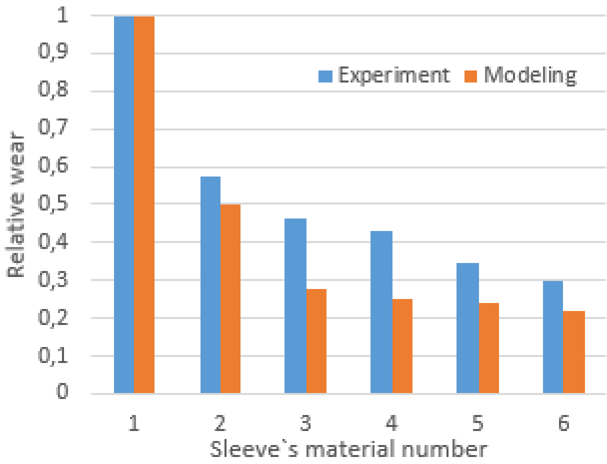


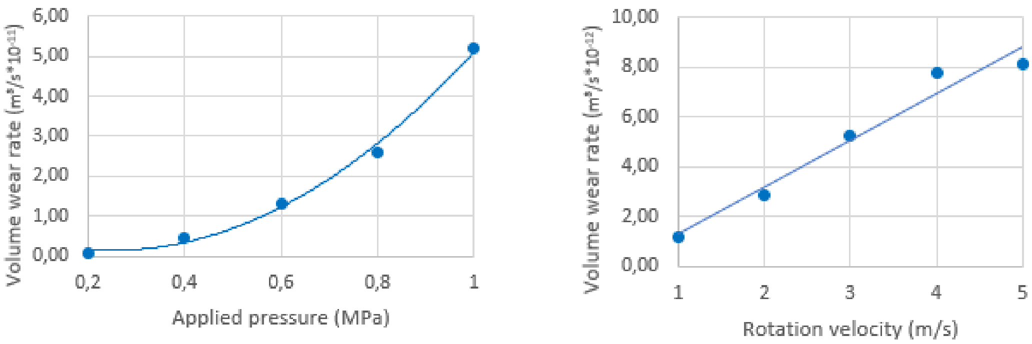
Fig. 6. Relative wear of sleeves made of different materials: 1 – cast iron 18; 2 – cast iron 18 with induction hardening; 3 – steel 40Ch with isothermal hardening; 4 – cast iron 18 with induction hardening and carburizing; 5 – steel 20 with nitriding; 6 – steel 45Ch

3. The results of solving the wear-contact problem considering FSI problem

The wear intensity and the contact pressure distribution along the width of the friction pair were chosen as the results of the study.

3.1. Wear intensity

As a result of the conducted study, the wear intensity of shaft was determined depending on the change in applied pressure (Fig. 7a) and the shaft rotation linear velocity (Fig. 7b). Thus, with a change in the shaft rotation linear velocity, there is a linear increase in the wear intensity. At the same time, with an increase in the applied pressure, the wear intensity increases according to a polynomial dependence. Such a discrepancy in dependencies is explained by the fact that when calculating the wear intensity when the applied pressure changes, the wear degree indicator also changes with the change in pressure.



a **b**

Fig.7. Dependence of the wear intensity on the applied pressure (a) and the shaft rotation linear velocity (b)

3.2. Contact pressure distribution

In fig. 8 shows the contact pressure distribution over the width of the friction pair at different values of the applied pressure. The point 0 mm along the width of the friction pair is the entrance to the seal, and the point 13 mm is the exit. It can be seen from the graph that as the applied pressure increases, the contact pressure also increases. The contact pressure increases when approaching the exit from the seal. This trend is explained by the fact that the hydraulic pressure decreases as it approaches the exit from the seal (Fig. 3).

In fig. 9 a-e show the contact pressure distributions before and after wear of the shaft sleeve. It can be seen from the graphs that after wear, the contact pressure decreases along the entire width of the friction pair. Moreover, when the applied pressure increases, the contact pressure drop also increases. Change (decrease) in contact pressure in the process of modeling is one of the indirect signs of wear of contacting surfaces.

The study of the effect of changing the rotation speed of the shaft on the distribution of contact pressure was carried out at an applied pressure of 0.4 MPa (Fig. 10). Analyzing the effect of changing the rotation speed of the shaft, it can be noted that there is no clear relationship between the increase or decrease of the contact pressure relative to the increase in the linear velocity of shaft rotation, which is quite well confirmed by the results of the experiment [0].

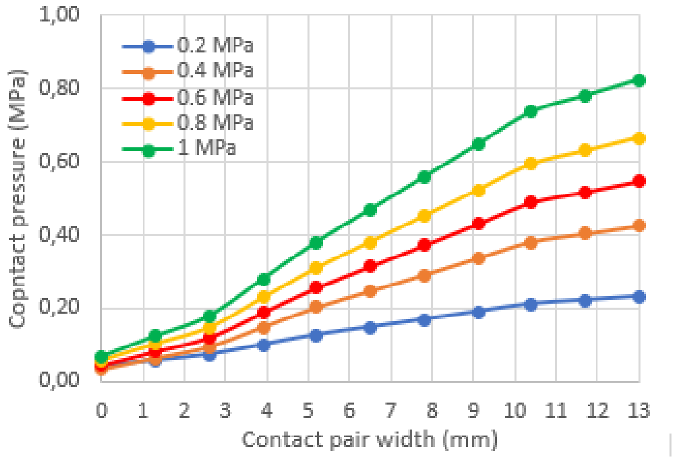
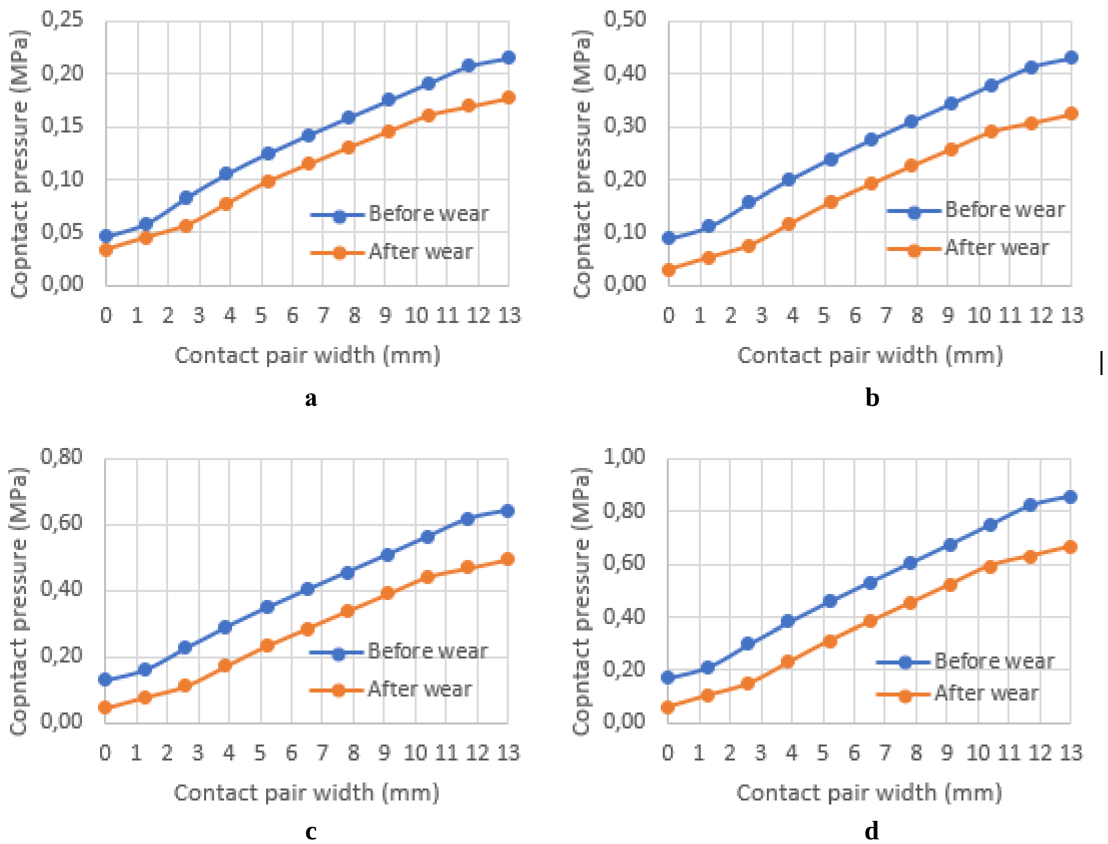
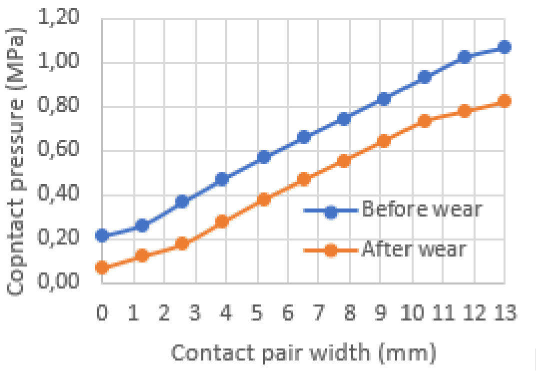


Fig. 8. Contact pressure distribution along the width of the friction pair when the pressure of the working medium changes

In fig. 11 shows the change in the shape of the contact surface. It can be seen from the graphs that the section closer to the exit from the seal experiences the greatest loss of material. That is, as mentioned above, in places of higher values of contact pressure. It is also worth noting the fact that as the shaft rotation linear velocity increases, the amount of material loss increases in direct proportion.

Separately, it should be noted that the obtained modeling results may have a slightly idealized picture of the wear of the contacting surfaces, as the effect of abrasive inclusions that may be present in the medium of the stuffing-box packing seal is not taken into account; misalignment of the shaft and seal and uneven loading of the packing gland, which holds the stuffing box packing inside the chamber, are not taken into account; the degree of washing out of the stuffing box packing impregnation is not taken into account; the effect of temperature is not taken into account. The impact of these factors can be reflected in further study.





e

Fig. 9. Contact pressure distribution before and after wear at applied pressure: a – 0.2 MPa, b - 0.4 MPa, c – 0.6 MPa, d – 0.8 MPa, e – 1 MPa

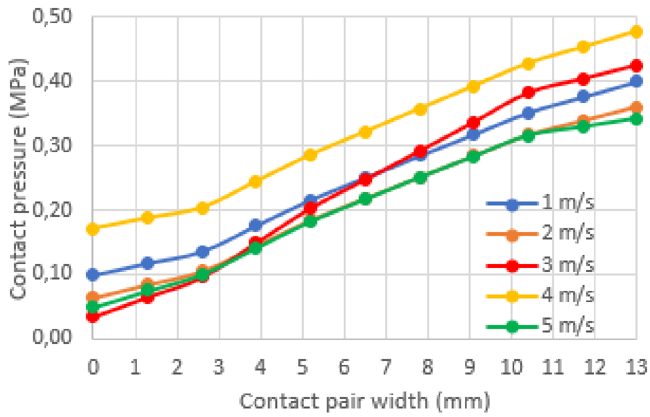


Fig.10. Contact pressure distribution along the width of the friction pair when the shaft rotation velocity changes

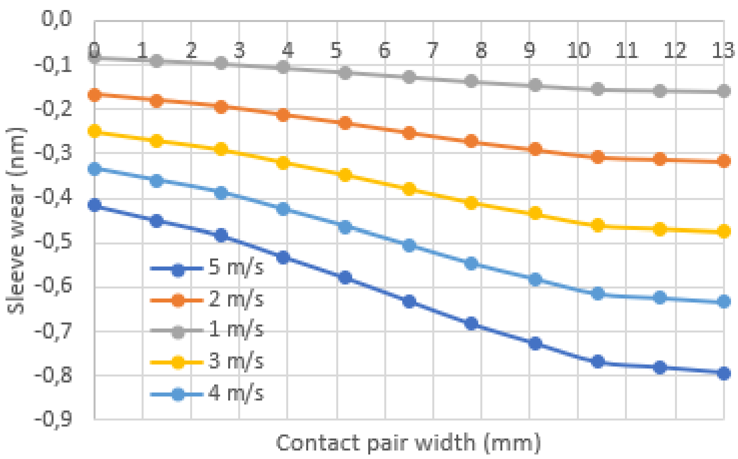


Fig. 11. Changing the shape of the contact surface during time 10^7 s

Nevertheless, despite all the mentioned limitations of the proposed model, it can be used to determine the wear of contacting surfaces. As for stuffing-box packing seals, this model can be used to predict the wear of protective sleeve surfaces and, as a result, specify the time intervals for its service.

Conclusions

The obtained investigation results indicate that the proposed stuffing-box packing seal wear model, built based on the Archard model, sufficiently accurately corresponds to the experimental data at the degree indicators m = 1.5 and n=1.3. Graphic representation of the dependence of the wear volume on the applied load confirms the correspondence of the calculation results to the experimental data with an accuracy of up to 10 %.

The given calculation results indicate that the model considers the influence of applied pressure and shaft rotation speed on the wear intensity. The difference in the influence of these parameters on the wear intensity is explained by the change in the wear degree indicator with the increase in applied pressure.

The obtained results have both theoretical and practical significance. The wear model can be used to predict the wear of the protective sleeve surfaces, which will help clarify the service intervals of the stuffing box packing seals and, therefore, increase their service life. As a result of the solution of the wear-contact problem, the understanding of wear processes in stuffing-box packing seals has been deepened. Addressing the model's shortcomings, such as lack of consideration of abrasive inclusions, no uniform loading, or temperature variation, in further studies may improve its accuracy and applicability.

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Я.І. Сапожников, А.В. Загорулько. Розрахункові дослідження механізму зношення сальникового ущільнення з використанням моделі Арчарда.

Представлено модель зношення сальникового ущільнення, зокрема, поверхні захисної втулки вала. Моделювання виконувалося за допомогою програмного комплексу ANSYS, застосовувався модуль Transient Structural із використанням вбудованої моделі зношення Арчарда. Виконано валідацію моделі зношення у відповідності до результатів попередніх експериментальних досліджень з урахуванням дії робочого тиску. Вперше визначено, що при показниках ступеня в рівняннях Арчарда m=1.5 та n=1.3 запропонована модель зношення дозволяє отримати достатньо точні значення інтенсивності зносу. Порівняно величини відносного зношення для різних матеріалів захисних втулок. Наведено графіки розподілу контактного тиску по ширині пари тертя при різних значеннях робочого тиску та лінійних швидкостей обертання вала. Отримано результати змінення форми поверхні втулки, обумовленої втратою матеріалу. Отримане змінення форми вважається ідеалізованим, так як в цьому дослідженні не враховуються наявність абразивних включень в робочому середовищі, нерівномірний розподіл тиску на натискній втулці, змінення температури та ін. Проте, це дослідження може бути корисним при прогнозуванні зношення основних компонентів сальникового ущільнення.

Ключові слова: зношування, модель Арчарда, сальникове ущільнення, розподіл контактного тиску, об’ємна втрата матеріалу