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Selected dynamic problems of the face packing seal

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Abstract

This paper analyzes the statics and dynamics of face packing seals, in which one of the sealing rings is made of elastic sealant. The study involved analyzing the axial vibration of the flexible ring for two configurations of its housing in the seal and determining the frequency characteristics and the condition of maintaining face-to-face contact between the sealing rings with considerably different elastic properties subjected to two types of external excitations. The derived analytical relationships were verified using a calculation example for real operating conditions of face packing seal in the pump.

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Keywords: face packing seal, design, dynamics

1. Introduction

Face packing seals were first designed about 30 years ago [1, 2] and since then they have been regularly modified [3, 4]. As a type of face seals, they are simple to design and relatively easy to fabricate. Face packing seals combine the benefits of longitudinal packing seals and mechanical face seals.

The theoretical analyses of face packing seals have included static calculations, which focus on determining the permissible loads of sealing rings and the leakage rates [5, 6]. In the technical literature, there are not many works dealing with the dynamics of face contacting seals, let alone face packing seals. The works [7, 8] describe the conditions under which there is a loss of leak-tightness for a face-to-face contact of the rings.

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This paper analyzes the axial vibration for two configurations of a face packing seal. The study involved determining the frequency characteristics and the condition for maintaining contact between two sealing rings with completely different elastic properties

2. Seal design and properties

The design of the face packing seal (Fig. 1) is similar to those of a mechanical face seal. The principal difference is that one of the sealing rings in a face packing seal is made of a soft packing characterized by elastic and plastic properties. The sealing is achieved on the face of the sealant ring 1 mounted in the axially flexible housing 2, and pressed against by the spring 5 to the face of the metal lid 3 screwed to the chamber of the stuffing box 6. The spring 5 ensures initial compression of the ring 1. During operation, however, an additional load causes by the pressure *p* of the sealed fluid. Thus, in this design of the face packing seal it is possible to optimize the load of the sealing ring.

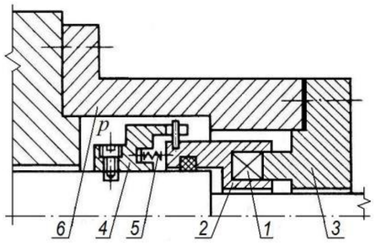

Fig. 1. Diagram of the face packing seal [8]: 1- sealing ring with the packing (sealant); 2- flexibly mounted housing of the ring, 3-closing cover, 4- positive drive device, 5- springs, 6- stuffing box.

Since one of the surfaces in contact is the surface of the soft packing, it is not essential that the surface of the metal ring in the sliding contact be precision machined as is the case with mechanical face seals ($Ra \le 0.1$ μm) [9]. It is necessary, however, that the smoothness of the surface be relatively good. In face packing seals, the contact surface area is smaller than in longitudinal seals, hence lower friction resistance, lower power loss and better heat removal. Generally, it can be assumed that face packing seals make use of many good, practically verified design solutions typical of mechanical face seals [10].

3. Static analysis

Static calculations are used to determine the initial axial displacement of the closing cover, which is determined on the basis of the required contact pressure p_c between the sealant ring and the seat seal ring to guarantee leak tightness. In the first approximation, the soft sealant can be treated as linearly elastic material [6]. Fig. 2 shows the calculation schemes and dynamic models of seals with axially flexible rotary (Fig. 2a) and non-rotary housing (Fig. 2b) with an elastic ring made of a sealant.

The symbols in the models above (Fig. 2) stand for: m – mass of the axially flexible housing with the ring; k, k_1 – equivalent (reduced) coefficient of elasticity of the springs and the coefficient of elasticity of the soft sealant, c – coefficient of damping, z_e – kinematic excitation, p_e – pressure of the sealed fluid, s_0 – axial displacement of the seat seal ring (closing cover).

The leak tightness of the face-to-face contact of the sealing rings is guaranteed firstly by the springs, which press the soft sealant ring 1 against the seat seal ring 3 (Fig. 1), and secondly, the force exerted by the pressure of the sealed fluid.

$$F_o = A_e p_e = A_0 K p_e \tag{1}$$

where $A_e = \pi\left(r_2^2 - r_e^2\right)$ – hydraulically loaded area, $A_0 = \pi\left(r_2^2 - r_1^2\right)$ – surface area of the face contact of the rings, $K = A_e / A_0$ – hydraulic load factor.

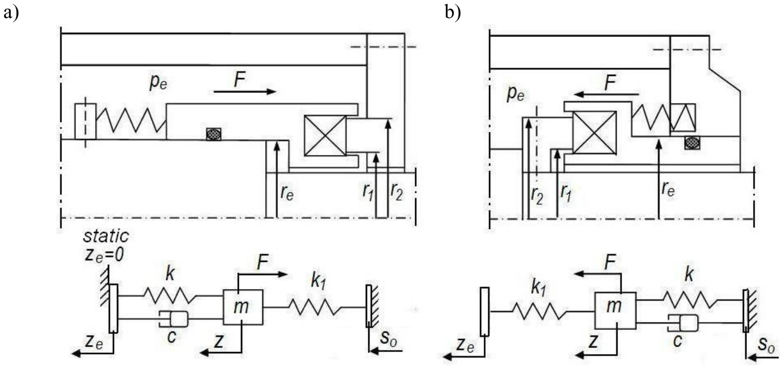

Fig. 2. Calculation schemes and the dynamic models of two seal configurations of a) the axially flexible rotary housing and b) the axially flexible non-rotary housing of an elastic ring.

The hydraulic load factor K is commonly used both in theory and design of mechanical face seals. It characterises the conditions of friction between the sealing rings and determines the service life of the whole seal. The conditions of the static equilibrium of the housing with the sealant ring (m) for both configurations of the housing (Fig. 2a,b) can be written as follows:

- diagram (A):

$$\left(k+k_1\right)z_0 = k_1 s_0 - F_0 \quad \Rightarrow \quad z_0^A = \frac{k_1}{k_1+k}s_0 - \frac{F_o}{k_1+k} = \kappa_1 s_0 - \frac{KA_o p_{eo}}{k_1+k} \tag{2}$$

- diagram (B):

$$\left(k+k_1\right)z_0 = k s_0 + F_0 \quad \Rightarrow \quad z_0^B = \frac{k}{k_1+k}s_0 + \frac{F_o}{k_1+k} = \kappa s_0 + \frac{KA_o p_{eo}}{k_1+k} \tag{3}$$

$$\kappa = \frac{k}{k+k_1}; \quad \kappa_1 = \frac{k_1}{k+k_1} \tag{4}$$

where κ, κ_1 – relative coefficients of elasticity; z_o – initial (in the static equilibrium state) displacements of the housing about unloaded position ($F_o = s_o = 0$); F_o – predetermined force to maintain contact between the rings exerted by the pressure of the sealed fluid p_{eo}.

In special cases, when the seal design does not include springs $\left(k \approx 0\right)$ or there is a rigid ring instead of a soft sealant $\left(k_1 >> k\right)$, the relative coefficients of elasticity or stiffness are $\kappa \approx 0,\ \kappa_1 \approx 1$, respectively. These values of the coefficients correspond to those typical of the design of mechanical face seals.

4. Forced vibration of the axially flexible housing

The axially flexible housing with the sealant ring is subjected to axial vibration about the static equilibrium position (Fig. 2). The external excitations are assumed to include inevitable small axial vibration of the shaft (as a kinematic excitation) and possible pressure pulsations of the sealed fluid.

During the pump operation, the predetermined excitations, i.e. the kinematic excitation $z_e = \delta z_e$ and the force excitation $p_e = p_{e0} + \delta p_e$, cause axial vibration of the flexibly mounted sealing ring and its housing around the position of static equilibrium $z = z_0 + \delta z$. The equations of motion for both configurations of the housing have the following form:

$$\text{(A)} \quad m\delta\ddot{z} + c\left(\delta\dot{z} - \delta\dot{z}_e\right) + k\left(z_0 + \delta z - \delta z_e\right) + k_1\left(z_0 + \delta z - s_o\right) = KA_0\left(p_{eo} + \delta p_e\right) \tag{5}$$

$$\text{(B)} \quad m\delta\ddot{z} + c\delta\dot{z} + k_1\left(z_0 + \delta z - \delta z_e\right) + k\left(z_0 + \delta z - s_o\right) = KA_0\left(p_{eo} + \delta p_e\right) \tag{6}$$

After considering the conditions of equilibrium (2), (3) and omitting the sign of the variance, we obtain the following equations of forced vibration:

$$\text{(A)} \quad \ddot{z} + 2n\dot{z} + \Omega_0^2 z = 2n\dot{z}_e + \kappa\Omega_0^2 z_e + f \tag{7}$$

$$\text{(B)} \quad \ddot{z} + 2n\dot{z} + \Omega_0^2 z = \kappa_1\Omega_0^2 z_e + f \tag{8}$$

$$\Omega_0 = \sqrt{\left(k + k_1\right)/m}; \quad n = c/2m; \quad f = \frac{A_o K p_e}{m} \tag{9}$$

where Ω_o, n – natural frequency of the system and the coefficient of damping, respectively.

The right sides of the equations of motion for both configurations are identical. Now a classic analysis of the derived equations is required, for example [11]. As mentioned above, *forced vibration* is caused by a kinematic excitation and a force excitation. The frequency of the pressure pulsations and the frequency of the axial vibration are generally equal to the angular velocity (or rotational frequency) of the shaft:

$$\delta z_e = He^{i\omega t}; \quad p_e = p_{e0} + \delta p_e; \quad \delta p_e = p_{ea}e^{i\omega t} \tag{10}$$

where H, p_{ea} – amplitudes of the harmonic excitations of the shaft vibration and the pressure of the sealed fluid.

From the linear vibration theory it is evident that the response of the linear system to two harmonic excitations is equal to the sum of the responses to the axial vibration of the shaft and the pressure pulsations:

$$z = z_1 + z_2 \quad \Rightarrow \quad z_1 = Z_1 e^{i(\omega t + \psi_1)}; \quad z_2 = Z_2 e^{i(\omega t + \psi_2)} \tag{11}$$

The next step of the analysis involves deriving the characteristics of the forced vibration separately for each excitation (10). After using the expressions for z_1 (11) and z_e (10) in equation (7), we obtain the following form of the equation ($p_e = 0$):

$$\left(\Omega_0^2 - \omega^2 + i2n\omega\right) Z_{1a} e^{i(\omega t + \psi_{1a})} = \left(\kappa\Omega_0^2 + i2n\omega\right) He^{i\omega t} \tag{12}$$

From the equation we determine the frequency transfer function in the complex form:

$$W_{1a} = \frac{Z_{1a}}{H} e^{i\psi_{1a}} = \frac{\kappa\Omega_0^2 + i2n\omega}{\Omega_0^2 - \omega^2 + i2n\omega} = A_{1a} e^{i\psi_{1a}} \tag{13}$$

After separating the real and imaginary parts in the obtained complex function (13), we find the amplitude-frequency and phase-frequency characteristics.

$$A_{1a}\left(\nu\right) = \frac{Z_{1a}}{H} = \sqrt{\frac{\kappa^2 + 4\xi^2\nu^2}{\left(1-\nu^2\right)^2 + 4\xi^2\nu^2}}; \quad \psi_{1a}\left(\nu\right) = arctg\, 2\xi\nu \frac{\kappa_1 - \nu^2}{\kappa\left(1-\nu^2\right) + 4\xi^2\nu^2} \tag{14}$$

where $\nu = \omega/\Omega_0$, $\xi = n/\Omega_0$ – ratios of the frequency and the damping, respectively.

In a similar way, we find a response to an excitation for configuration two (Fig. 2b)

$$A_{1b}(\nu)=\frac{Z_{1b}}{H}=\frac{\kappa_1}{\sqrt{\left(1-\nu^2\right)^2+4\xi^2\nu^2}};\qquad \psi_{1b}(\nu)=-arctg\frac{2\xi\nu}{1-\nu^2} \tag{15}$$

The response of the pressure pulsations is the same for the both design configurations. In this case, the frequency characteristics are as follows:

$$A_2(\nu)=\frac{Z_2}{p_{ea}}\cdot\frac{k+k_1}{KA_0}=\frac{1}{\sqrt{\left(1-\nu^2\right)^2+4\xi^2\nu^2}};\qquad \psi_2(\nu)=-arctg\frac{2\xi\nu}{1-\nu^2} \tag{16}$$

The introduced characteristics (14 ÷ 16) enable us to determine the amplitude of the axial vibration for the system (flexibly mounted ring) for an arbitrary angular velocity of the shaft at the predetermined amplitudes of the harmonic excitations H and p_{ea}:

$$Z_{1a}=A_{1a}H;\quad Z_{1b}=A_{1b}H;\quad Z_2=A_2\frac{KA_0p_{ea}}{k+k_1} \tag{17}$$

Condition of maintaining contact. During normal operation of the seal, the contact between the ring faces the required leak-tightness. The loss of contact during machine (pump) operation is unacceptable.

Let us consider the conditions at which for both seal configurations (Fig. 2) there is a loss of contact. As mentioned above, for model (A), the value of the force responsible for the squeeze of the sealant at the static equilibrium position is $\Delta z^A = s_o\text{-}z_o^A$, for model (B) $\Delta z^B = z_o^B$. If the overall amplitude of the forced vibration of the flexible housing of the sealant exceeds the value $\Delta z^{A,B}$, the face contact will be periodically interrupted. Under the most unfavourable conditions for synchronous vibrations with coinciding phases $(\psi_1=\psi_2=0)$, the condition of maintaining contact is met by the following inequality: $Z_1^{A,B}+Z_2<\Delta z^{A,B}$.

$$A_{1a,b}(\nu)H+A_2(\nu)\frac{KA_0p_{ea}}{k+k_1}<\kappa s_o+\frac{KA_0p_{eo}}{k+k_1} \tag{18}$$

The results of the analysis of the axial vibration of the elastic ring (made of a soft sealant) in contact with the rigid ring can be used for the design of new face packing seals.

5. Numerical example

The dynamics of the real face packing seal was quantitatively described using the calculated frequency characteristics for the double-sided pump.

Processing pump: pumping the water at the pressure $p_e=0.65$ MPa, nominal angular velocity of the shaft $\omega_n=150$ s^{-1}, diameter of the shaft (shaft sleeve) at the point of assembly of the seal 120 mm, $r_1=70$ mm, $r_2=90$ mm (Fig. 2), cross-sectional area of the cord sealant 20×20 mm, $A_0=10.1\cdot10^{-3}$ m^2, hydraulic load factor $K=1.2$, coefficients of elasticity of the springs and the sealant $k=10^5$ N.m^{-1}, $k_1=5\cdot10^5$ N.m^{-1}, mass of the housing with the sealant ring $m=6$ kg. According to relationships (4) and (9), $\kappa=0.17$, $\kappa_1=0.83$; $\Omega_0=316$ s^{-1}, $\nu_n=\omega_n/\Omega_0=0.47$. The Fig. 3 shows the calculated amplitude-frequency and phase-frequency characteristics for three values of the damping ratio: ξ= 1- 0.05; 2- 0.1; 3- 0.3.

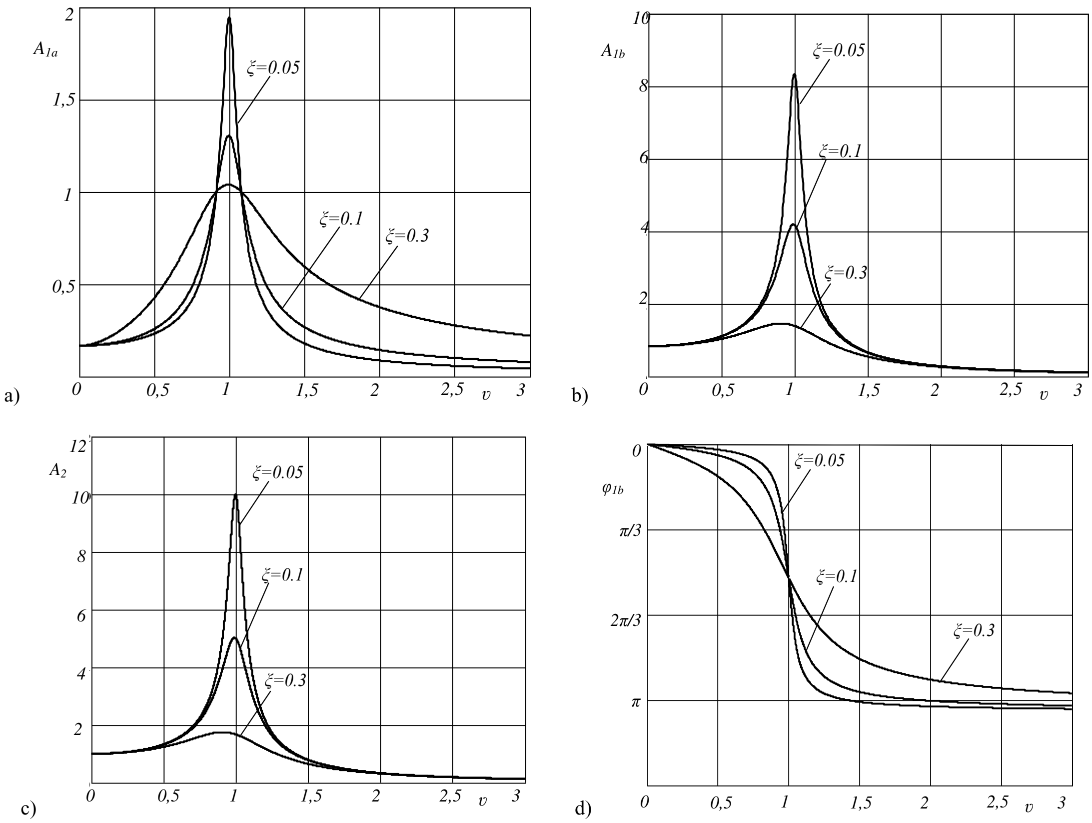

Fig. 3. Amplitude-frequency characteristics in response to: a), b) axial vibration of the shaft, c) pressure pulsations, d) phase-frequency characteristics.

According to relationship (18) the initial deformation of the springs and the sealant, Δz, for which the face-to-face contact of the rings is maintained also during resonance vibration, is not less than 0.4 mm. In the example considered here, it can be recommended with an appropriate safety factor that Δz = 4 mm.

6. Conclusion

From the results obtained, the following general conclusions can be formulated. The axial vibration of the housing with the flexible ring made of a soft sealant is mainly dependent on the amplitude of the kinematic excitation in the form of axial vibration of the shaft.

Since the natural frequencies of the axial vibration of the housing with the flexible ring are much higher than the nominal frequencies of the rotating shaft, resonance conditions rarely occur. As the stiffness of the sealant is high, the natural frequency of the axial vibrations of the flexible housing with the sealant ring considerably exceeds the nominal (working) angular velocity of the shaft, which suggests that resonance conditions are hardly probable.

For the seal configuration where the housing with the flexible ring performs a rotary motion and the kinematic excitation acts directly on the springs, the amplitude of resonance vibration is an order of magnitude lower than that for a non-rotary housing.

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