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A Numerical Analysis of the Grooved Surface Effects on the Thermal Behavior of a Non-Contacting Face Seal

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Abstract

The paper analyzes heat transfer in a non-contacting face seal with micro-grooves on the end face of the sealing ring. The thermohydrodynamic model of the face seal comprises the equation of the pressure distribution and the energy equation for the fluid film in the clearance gap and the equation of heat conduction in the ring. The mathematical model consisting of coupled partial differential equations was solved numerically using a specially developed computer program. A parametric analysis was conducted to establish the effect of the shape and parameters of the microgrooves on the distribution of temperature in the fluid film and the sealing ring.

Keywords: Face seals; temperature distribution; numerical analysis.

1. Introduction

The application of non-contacting face seals with microstructures on the sliding surfaces of the sealing rings is increasing. They are used in turbomachines, for example, in the chemical and aviation industries. The main reason for their popularity is low wear, low leakage rate and small power losses.

The works dealing with the hydrodynamics of spiral groove face seals, e.g. Refs. [1, 2], or thrust bearings, e.g. Ref. [3] suggest that microstructures such as grooves or dimples (laser surface texture) on the end faces of the rings, e.g. Refs. [4, 5, 6] contribute to an increase in the hydrodynamic force in the clearance gap. Microstructures with a properly selected shape and geometrical parameters guarantee

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stable and reliable operation of the seal. New solutions to the design of spiral groove face seals require in-depth theoretical and experimental analysis. The theoretical studies concerning the thermohydrodynamic problems of face seals based on more or less complex mathematical models are an important contribution to the optimal design. In Ref. [7], the authors present a thorough review of the theoretical and experimental studies concerning heat generation and transfer as well as thermal deformations in face seals with flat, smooth (texture-free) sliding surfaces (races).

In the literature on the subject, there are very few publications concerned with the influence of the microstructures on the thermal properties of face seals, i.e. the amount of heat generated in the clearance gap and the heat transfer efficiency of the sealing rings.

In Ref. [8], Tournerie et al. present a thermohydrodynamic model of a face seal and an algorithm of its numerical solution. Their detailed parametric analysis aims at determining the effect of the wavy face, and the misalignment of the ring surfaces as well as the radial grooves on the temperature distribution in the fluid film in the clearance gap and in the rings. They show that the misalignment and the waviness of the end faces does not have a considerable influence on the changes in temperature.

Zhou et al. [9] conducted an interesting theoretical analysis of a spiral groove face seal, including analytical and numerical solutions. Heat dissipation in the fluid film in the clearance gap and the distribution of temperature in the rings were calculated analytically, whereas the distribution of pressure in the clearance gap and the thermal deformations of the rings were determined from numerical solutions. Extensive multi-coefficient coupled calculations were performed to determine the optimal geometrical parameters of the spiral grooves.

This paper presents a three-dimensional thermohydrodynamic model of a face seal and the results of its numerical solution conducted by using a specially developed computer program. The main aim of the numerical calculations was to determine the influence of the shape and characteristic parameters of the microstructures, e.g. the depth of spiral grooves and the number of grooves along the ring circumference, on the temperature distributions in the elements of a non-contacting face seal.

2. Subject and methodology

The analysis was conducted for an FMS-type non-contacting face seal with microstructures on the end surfaces of the flexibly mounted stator. In the schematic diagram of the seal (Fig.1), we can also see the surfaces of the sealing rings where heat transfer occurred.

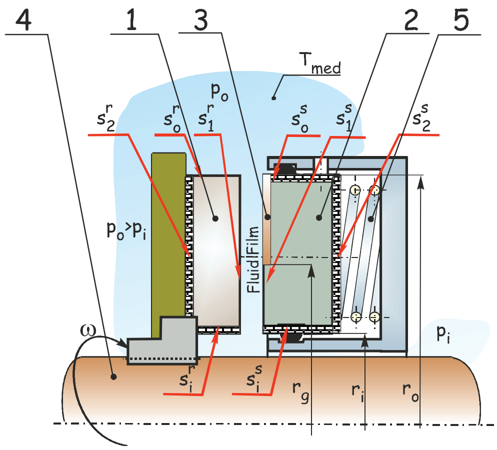

Fig. 1. Schematic diagram of a seal with microstructures on the end face of the stator 1, 2 - seal rings, 3-face grooves, 4- shaft, 5-spring

The parametric analysis was carried out for three types of microstructures, i.e. geometrical modifications of the end face of the stationary ring (Fig. 2).

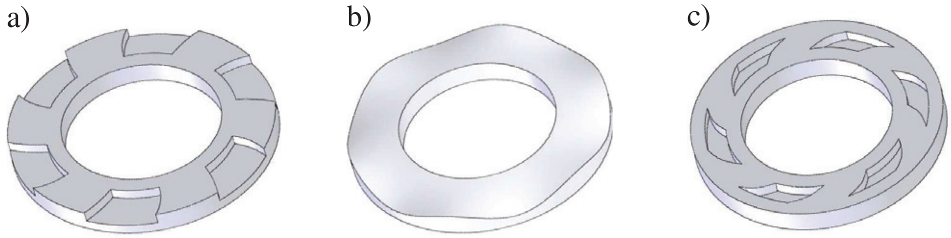

Fig. 2. Geometrical modifications of surfaces: (a) Pw1, (b) Pw2 and (c) Pw3

The first micropattern denoted by Pw1 (Fig.2a) is open spiral grooves described with a known formula [3, 9]:

$$r = r_g e^{\theta \tan \alpha} \,, \tag{1}$$

where: r_g – end radius of the spiral grooves; - α spiral angle.

The second microstructure designated by Pw2 is the circumferential waviness of the end face (Fig.2b), while the third microstructure, denoted by Pw3, is closed spiral grooves (Fig.2c).

To conduct the simulation tests it was necessary to develop a model describing the flow of the fluid through the clearance gap and the generation and transfer of heat.

3. Governing equations of the mathematical model

The mathematical model describing the behavior of a non-contacting face seal was formulated by determining the function of the height of the clearance gap between the end faces of the sealing rings, i.e. the stator and the rotor. The function of the clearance gap height in a cylindrical system can be written as:

$$h = h(r,\theta) = h_o + h^g(r,\theta), \tag{2}$$

where: $h^g(r,\theta)$ – parametric function describing the depth of the spiral groove.

The components of the flow rate and their derivatives along the corresponding coordinates were written in the form [10]:

$$v_r = \frac{1}{2\mu}\frac{\partial p}{\partial r}\left(z^2 - z(H^r + H^s) + H^r H^s\right) \,, \tag{3}$$

$$v_\theta = \frac{1}{2r\mu}\frac{\partial p}{\partial \theta}\left(z^2 - z\left(H^r + H^s\right) + H^r H^s\right) + \frac{V_\theta^{\,r} - V_\theta^{\,s}}{\left(H^r - H^s\right)}\left(z - H^s\right) + V_\theta^{\,s} \,, \tag{4}$$

$$\frac{\partial v_r}{\partial z} = \frac{1}{\mu}\frac{\partial p}{\partial r}\left(z - \frac{1}{2}\frac{(H^{r2} - H^{s2})}{(H^r - H^s)}\right) \,, \tag{5}$$

$$\frac{\partial v_\theta}{\partial z} = \frac{1}{r\mu}\frac{\partial p}{\partial \theta}\left(z - \frac{1}{2}\frac{\left(H^{r2} - H^{s2}\right)}{\left(H^r - H^s\right)}\right) + \frac{V_\theta^{\,r} - V_\theta^{\,s}}{\left(H^r - H^s\right)} \quad (6)$$

The distribution of pressure in the fluid film is described by the Reynolds equation, which, for a laminar flow of an incompressible fluid, neglecting the inertial forces, can be written as [2, 8, 11]:

$$\frac{\partial}{\partial r}\left(rh^3\frac{\partial p}{\partial r}\right) + \frac{\partial}{\partial \theta}\left(\frac{h^3}{r}\frac{\partial p}{\partial \theta} - 6\mu\omega rh\right) = 0 \quad (7)$$

where: $\mu(r,\theta,z)$ – dynamic viscosity of the fluid; h – clearance gap height; p – pressure in the fluid film; r,θ – directions of the axes of the coordinate system.

The process of converting mechanical work (frictional resistance) to thermal energy occurring in the radial clearance gap of the face seal can be described by means of the following energy equation.

$$\rho C_p\left\{v_r\frac{\partial T}{\partial r} + \frac{v_\theta}{r}\frac{\partial T}{\partial \theta} + v_z\frac{\partial T}{\partial z}\right\} = \mu\left\{\left(\frac{\partial v_r}{\partial z}\right)^2 + \left(\frac{\partial v_\theta}{\partial z}\right)^2\right\} + \lambda\frac{\partial^2 T}{\partial z^2} \quad (8)$$

The equation presents a full diathermal model in which temperature is a function dependent on three coordinates $T = f(r,\theta,z)$. In the model, the heat is conducted along the fluid film thickness (along the Z-axis), which requires considering the variation in temperature in this direction not only in the fluid film but also in the sealing rings.

The energy equation (8) and the pressure equation (7) take account of the change in the dynamic viscosity of the fluid resulting from the change in the temperature according to the relationship used for both non-contacting face seals and sliding bearings [7, 8, 9]:

$$\mu = \mu_f e^{\left(-b\left(T_m - T^f\right)\right)} \quad (9)$$

where: μ_f – dynamic viscosity at the reference temperature T^f; b – thermoviscosity coefficient $\left[1/{}^oC\right]$, (for water b=0.0175).

The local average temperature in the fluid film can be determined from the following relationship:

$$T_m = \frac{1}{\left(H^r - H^s\right)}\int_{H^s}^{H^r} T\,dz. \quad (10)$$

Like in Ref. [7], we assume that the stationary ring is insulated (Fig.3) and the heat generated in the fluid in the clearance gap is transferred by conduction to the rotor and then by natural convection to the surrounding fluid.

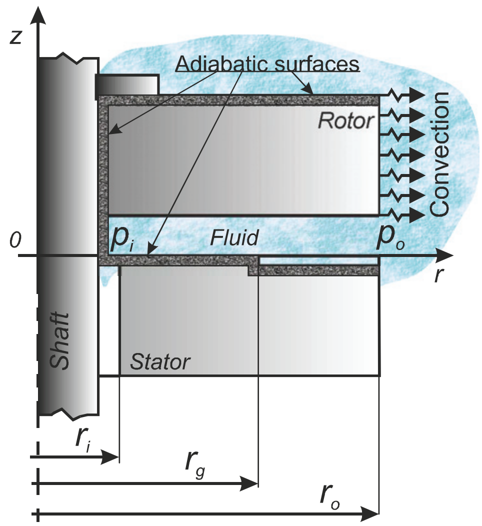

Fig. 3. Schematic diagram of the heat transfer in the fluid film-sealing ring system

For steady-state conditions, the three-dimensional temperature distribution in the ring (Fig. 3) with a constant heat transfer coefficient is described by the Laplace's differential equation.

$$\frac{1}{r}\frac{\partial T}{\partial r}+\frac{\partial^2 T}{\partial r^2}+\frac{1}{r^2}\frac{\partial^2 T}{\partial \theta^2}+\frac{\partial^2 T}{\partial z^2}=0 \qquad (12)$$

To solve this equation we need appropriate boundary conditions. As shown in Fig. 3, heat transfer by conduction occurs between the end face of the ring and the fluid in the clearance gap, which can be written as an equality of heat fluxes:

$$k_f \left.\frac{\partial T^f}{\partial z}\right|_{z=h^f} = k_r \left.\frac{\partial T^r}{\partial z}\right|_{z=0} \quad and \quad T^f = T^r . \qquad (13)$$

The heat transfer occurring on the surface of the ring in contact with the surrounding fluid is in the form of natural convection:

$$-k\left.\frac{\partial T}{\partial r}\right|_{r=r_o} = H_w\left(T\big|_{r=r_o} - T_{med}\right) . \qquad (14)$$

The mathematical model formulated above was solved numerically. The Reynolds equation (7) was solved using the Finite Difference Method (FDM), while the energy equation (8) and the heat conduction equations (12) were solved by means of the Finite Volume Method (FVM). The complete numerical algorithm was used to write a computer program in C++ [12] for conducting extensive parametric analysis of heat transfer, taking account of the geometry and material of the sealing rings and the operating conditions.

4. Results of numerical simulations

The results of the calculations of the temperature distributions in the fluid film and the sealing ring for the geometrical and operational parameters of the analyzed face seal are shown in Table 1.

Table 1. Basic geometrical and operational parameters of the FMS-type seal

Parameter	Symbol	Value	Unit
Type of sealed fluid		water	
Inner radius	r_i	0.035	$[m]$
Outer radius	r_o	0.040	$[m]$
Spiral angle	α	20	$[\,^o]$
Dynamic viscosity at 20°C	μ	$1 \cdot 10^{-3}$	$[Pa \cdot s]$
Clearance gap height in steady state	h_o	$6 \cdot 10^{-6}$	$[m]$
Pressure along the inner radius	p_i	$0 \cdot 10^5$	$[Pa]$
Pressure along the outer radius	p_o	$5 \cdot 10^5$	$[Pa]$
Angular velocity	ω	1200	$[rad / s]$
Depth of spiral groove	h_g	$(6,12,24) \times 10^{-6}$	$[m]$
Number of modifications	N_g	6,10,12	—

Figure 4 shows distributions of the average temperature in the fluid film in the radial clearance gap. Three different microstructures were analyzed (Fig. 2).

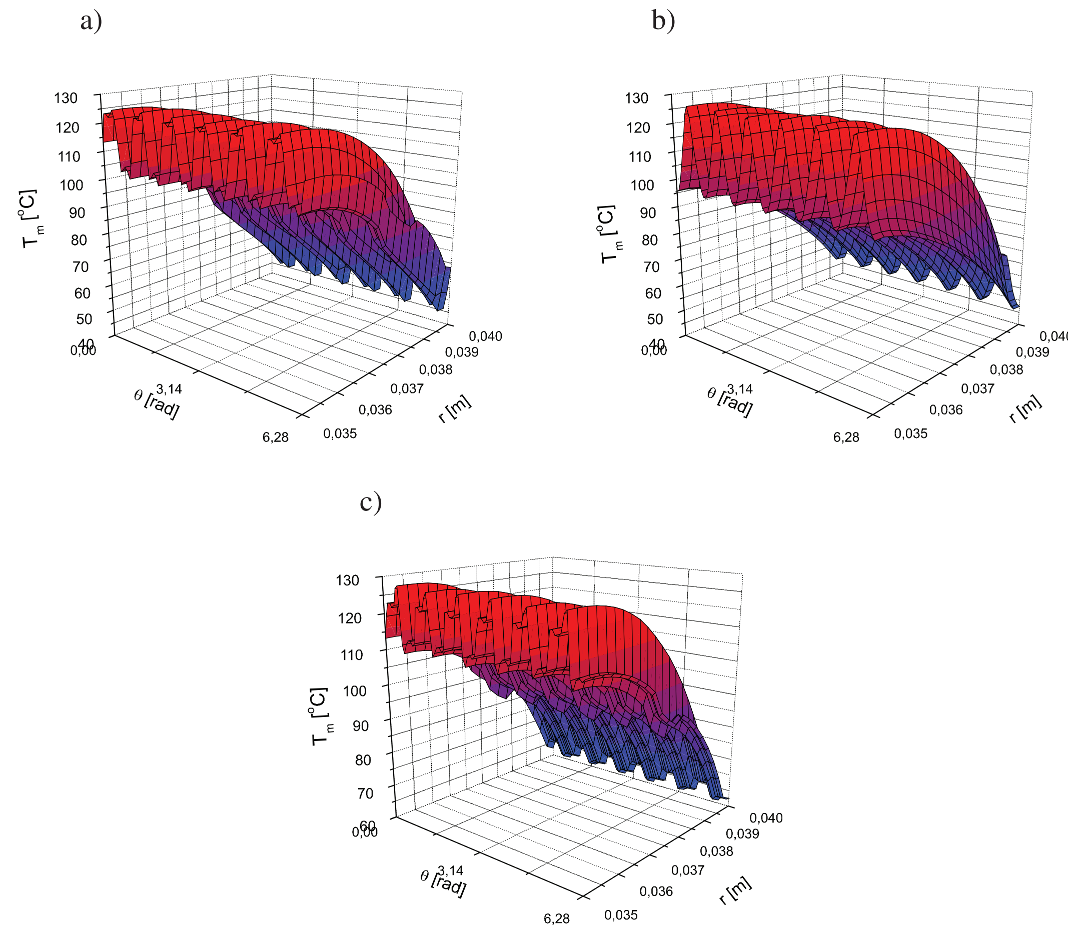

Fig. 4. Distribution of the average temperature of the fluid in the radial clearance gap for three surfaces: (a) Pw1, (b) Pw2, (c) Pw3

From the results presented in Fig.4 it is clear that the shape of the surface modification of one of the sealing rings has a considerable effect on the temperature distribution in the fluid film. For surfaces Pw1 and Pw2, the differences in temperature are greater than for surface Pw3. This is due to the smoother flow of the cold fluid through the microgrooves to the clearance gap. For surface Pw2, with closed microstructures (cavities), the temperature is the lowest along the outer radius r_o where the ring is in direct contact with the cold sealed fluid.

The graph in Fig. 5 shows changes in the temperature in the circumferential direction along the mean radius r_m for h_o=6 10^{-6}[m].

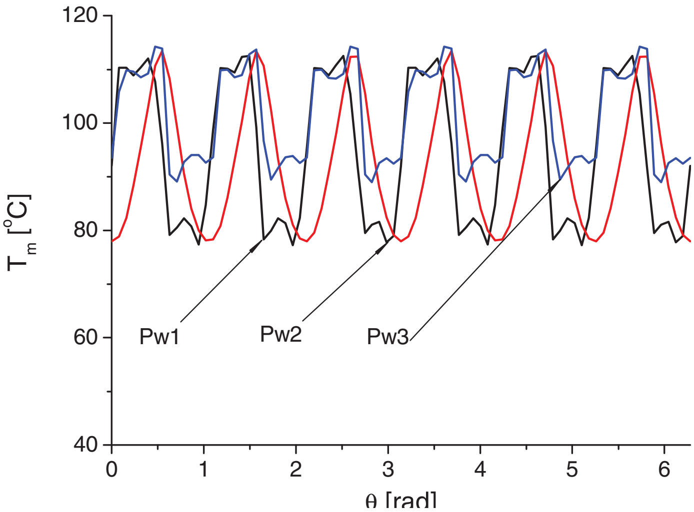

Fig. 5. Distribution of the average temperature along the radius r_m

From the above distributions it is clear that the temperature around the circumference falls by about 30°C. This will affect the dynamic properties of the fluid film separating the sealing rings. The variation in the average temperature in the clearance gap results directly from the modifications of the stator surface. It can be seen that the average temperature of the fluid is the highest in the area where the clearance gap height ($H \approx\sim h_o$) is equal to the nominal height of the radial clearance gap.

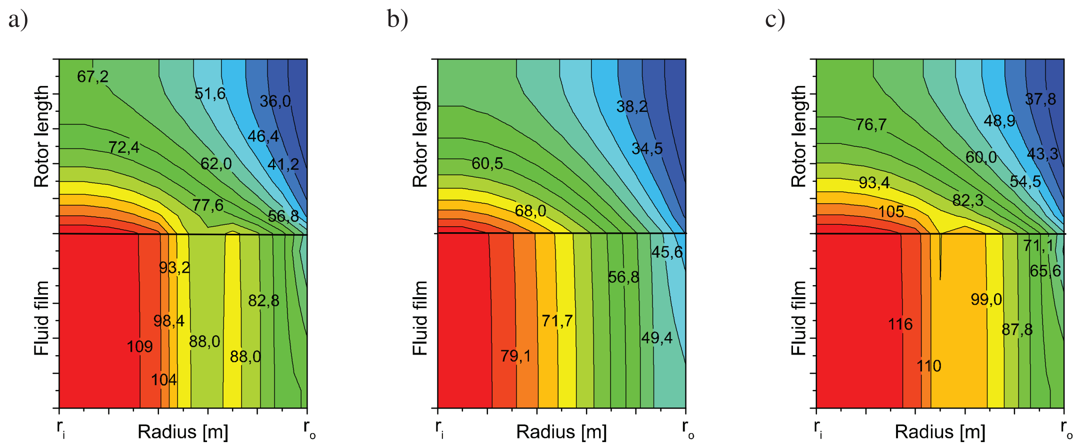

Fig. 6. Distribution of the temperature fields in the fluid film and the rotor for $\theta = \pi$

In Fig. 6 we can see distributions of the temperature fields in the fluid film and the rotor for three rings with the depth of spiral grooves h_g=$12\ 10^{-6}[m]$ for surfaces Pw1 and Pw3 and the angle $\beta = 0.0012$ for surface Pw2.

The numerical calculations indicate that, along the outer radius where the process takes place, i.e. where the working fluid enters the grooves (microstructures) at a predetermined pressure p_o, the temperature in the clearance gap is slightly higher than the temperature of the sealed fluid. This the cold fluid mixes with the hot fluid at the inlet and that heat is transferred through the sealing rings.

The graphs in Fig. 7 illustrate changes in the average temperature in the clearance gap according to the depth of spiral grooves.

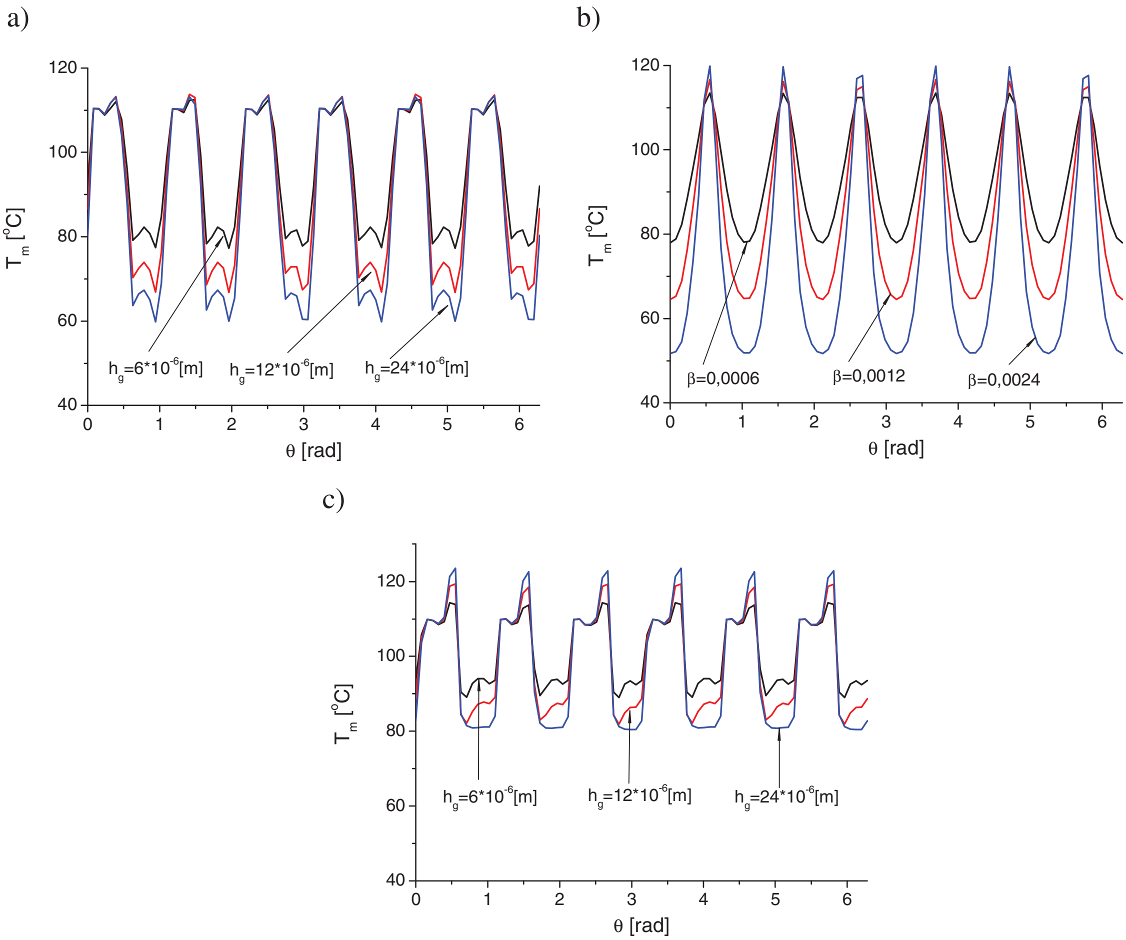

Fig. 7. Distribution of the average temperature along the radius r_m for (a) surface Pw1, (b) surface Pw2, (c) surface Pw3

From the graphs in Fig.7 it is clear that the depth of the surface modifications has significant influence on the temperature distribution. A fourfold increase in the groove depth – from 6x10^{-6} to 24x10^{-6} [m] - causes that the temperature along the mean radius falls by more than 20^{o} C. This is particularly visible for the spiral groove surface (Pw1) and the wavy face (Pw2). In the case of surface Pw3 with closed microstructures, a decrease in the temperature in the cavities is small, which results from the fact that there is no flow of the cold fluid into the clearance gap. We can see that, for this surface, the average temperature is the highest along the mean radius.

The graphs in Fig. 8 show how the number of microstructures (modifications) affect the radial distribution of temperature in the sealing ring (rotor). It is clear that the influence of the number of surface modifications on the changes in the rotor temperature is negligible for the wavy face (surface Pw2). However, greater changes in the temperature are observed for the open spiral groove surface (Pw1) and the surface with closed microstructures (Pw3). In Fig. 8a, we can see that, for surface Pw1, the temperature is the lowest when the number of grooves is six. For a surface with ten or fourteen microgrooves arranged around the circumference, the temperature distribution is comparable. Similar results were obtained for surface Pw3. For a predetermined geometry of the ring and operating conditions, there exists a certain number of microstructures for which the temperature reaches a maximum.

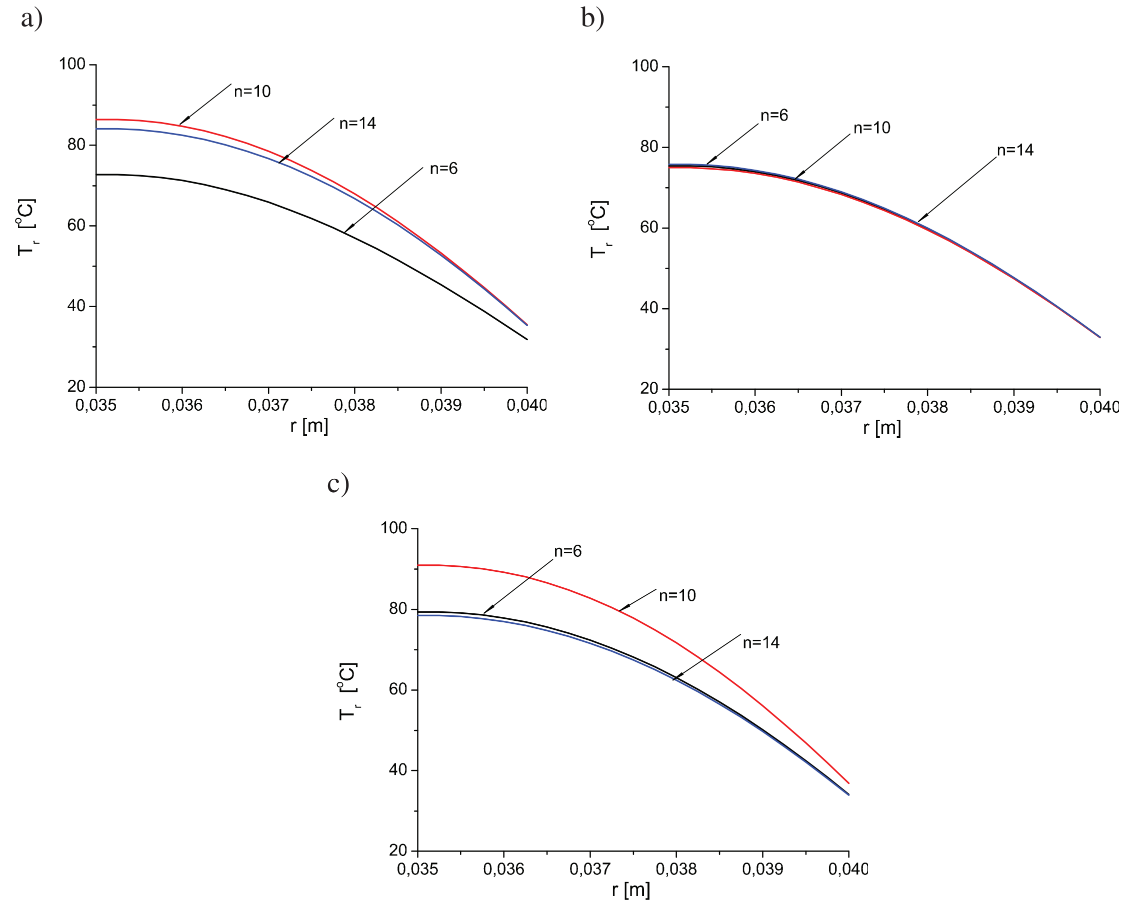

Fig. 8. Temperature distribution in the rotor (a) surface Pw1, (b) surface Pw2, (c) surface Pw3

Using the relationship (9), it was possible to calculate the changes in the dynamic viscosity μ depending on the distribution of temperature in the radial clearance gap (Fig. 9).

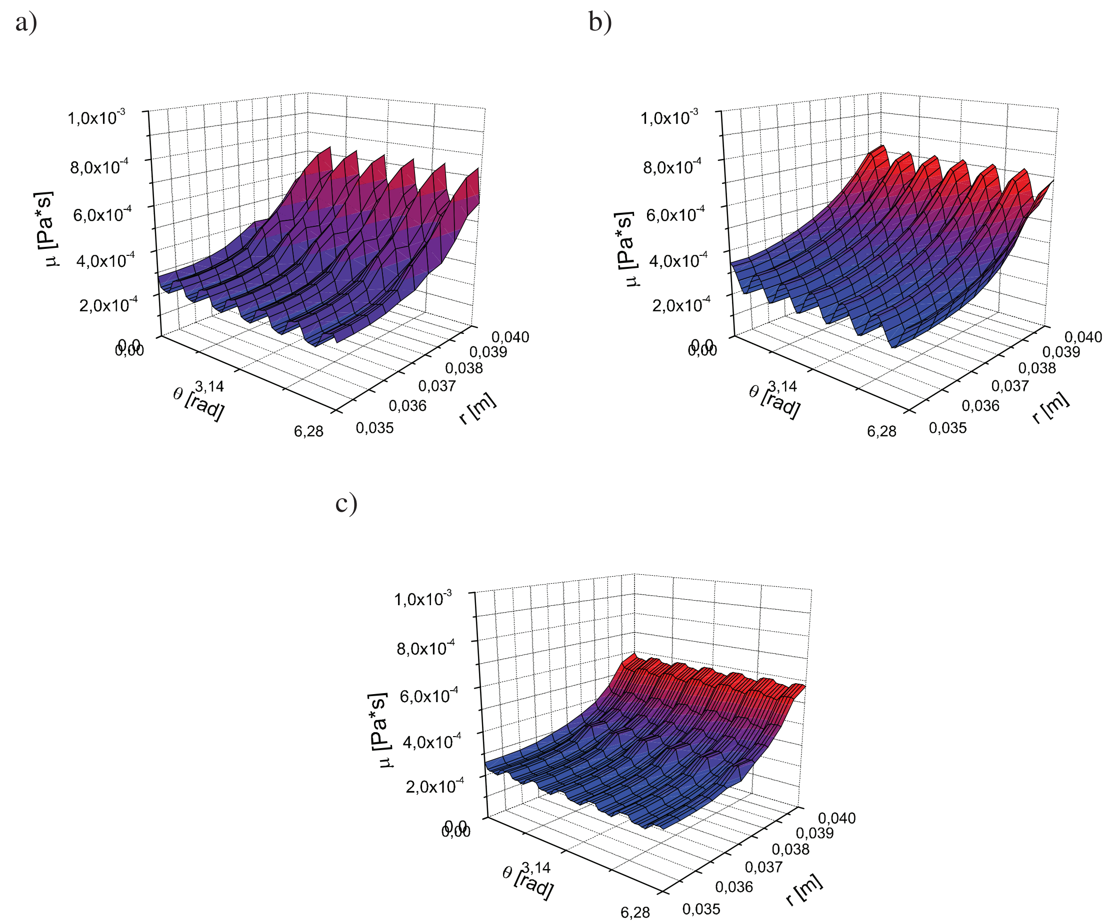

Fig. 9. Distribution of the dynamic viscosity in the radial clearance gap for surfaces Pw1, Pw2 and Pw3

The different three-dimensional temperature distributions in the fluid film have considerable influence on the fluid dynamic viscosity, which leads to changes in the dynamic properties of the seal.

5. Conclusion

The results of the numerical analysis show that the shape and geometrical parameters of the patterned surface microstructures on the end face of the sealing ring have a considerable effect on the temperature distributions in the structural elements of the seal. Three types of surface modifications were considered. The most favorable heat transfer conditions for friction heat generated in the clearance gap were reported for the open spiral grooves in contact with the surrounding fluid. The parametric analysis suggests that there exists an optimal number of grooves (microstructures) arranged around the ring circumference, for which we obtain, for example, the predetermined average temperature or the desired temperature distribution in the fluid film. The changes in the temperature in the clearance gap lead to significant changes in the dynamic viscosity coefficient and, accordingly, changes in the dynamic properties of the seal. The thermodynamic model can be used to determine the thermal deformations of the sealing rings. As a result, it will be possible to conduct a thorough analysis of the dynamics of the seal-turbomachine shaft system.

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