A Numerical Analysis of the Temperature Distributions in Face Sealing Rings

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Abstract

Simulation tests were conducted to establish the influence of the operational parameters on the distributions of temperature in the sealing rings and the fluid film. A mathematical model was developed for a non-contacting face seal. It included a system of coupled differential equations with partial derivatives describing three-dimensional temperature distribution. The solution of the formulated model required developing algorithms and applying numerical methods in the form of a computer program in C++. The equations with partial derivatives were solved using the Finite Volume Method (FVM).

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1. Introduction

The durability and reliability, and therefore correct operation, of non-contacting face seals are largely dependent on the thermal conditions. The heat flux generated in the clearance gap is responsible for the non-uniform distribution of temperature both in the fluid film and the sealing rings. An excessive increase in the temperature causes changes in the properties of the fluid and sometimes even its vaporization. In the last 30 years, many theoretical and experimental studies have been conducted to analyze the thermal phenomena in face seals, which confirms how significant the problem is.

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This paper presents a short review of the latest theoretical approaches to the temperature distribution in seal elements. The first works dealing with heat flow in face seals provide us with one-dimensional mathematical models of the distributions of pressure and temperature in the clearance gap as well as heat conduction in the rings. Such simplified mathematical models can be used to obtain full analytical solutions. In the well-known monographs by Mayer and Golubiev [1, 2], the average temperature in the friction zone for a pair of sealing rings is calculated by using the Fourier’s law. Another simple one-dimensional model of heat conduction in a sealing ring was proposed by Buck in Ref. [3]. He assumes that, for heat-removing fins with the length much greater than the thickness, the temperature distribution is uniform in an arbitrary cross-section. Two-dimensional models of the temperature distribution in sealing rings solved analytically can be found, for example, in Refs. [4, 5]. Dumbrava and Morariu [4] describe a thermohydrodynamic model of a double non-contacting seal. They obtain an analytical solution of the temperature distribution in the fluid film and the sealing rings by assuming that the temperature along the clearance gap height is constant. Pascovici and Etsion [5] also present a thermohydrodynamic analysis for a face-to-face double seal. By applying an analytical solution, they calculate the radial distribution of temperature in the fluid film and on the surfaces of the rings. They assume that the stator surface is insulated and that the whole heat emitted in the film is conducted through the rotor. They use the assumption presented in Ref. [4] that simplifies the heat flow in the rotor into the form of parallel straight lines.

A large number of theoretical studies concerned with the thermal analysis of face seals present analytical and numerical solutions. The heat dissipation in the fluid film is most often calculated analytically, whereas the heat conduction equations and the equations of thermoelasticity for the sealing rings are solved numerically. One of the first publications providing a numerical solution of a two-dimensional temperature field in the sealing rings is the work by Li [6]. The main boundary condition to solve the heat conduction equation was the assumption of a constant level of the friction heat produced in the clearance gap. Other well-known publications on the subject include works by Zeus [7] and Zhu [8], who calculated the heat dissipation in the fluid film analytically and the temperature distributions in the rings numerically using the finite element method. In Ref. [9], which is an extensive monograph, Lebeck analyzes a mechanical face seal as a thermal system. He proposes analytical solutions to calculate the temperature in the clearance gap and the ring deformations caused by an external mechanical load, which contribute to radial taper and waviness of the surface.

Another group of publications provide only numerical solutions to the coupled equations of fluid flow through the clearance gap, the energy equations for the fluid film and the heat conduction equations in the rings for real boundary conditions. Knoll et al. [10] present a three-dimensional thermohydrodynamic model of a face seal. They provide an algorithm for a numerical solution with the finite element method and the calculation results for the basic thermal and mechanical characteristics of the seal, i.e. the temperature distributions in the fluid film and the rings, leakage rate, power losses and pressure distribution. Advanced thermohydrodynamic (THD) and thermoelastohydrodynamic (TEHD) models of non-contacting face seals are presented in numerous works by researchers from Universite de Poitiers. For instance, Tournerie et al. [11] describe the fundamental equations of thermohydrodynamic (THD) lubrication and the equations of heat conduction in the rings. They solve the mathematical model numerically using the finite difference method. They also conduct a parametric analysis to determine the effect of the clearance gap geometry and the properties of the ring materials on the temperature distribution in the fluid film and the rings. In Refs. [12 and 13], Brunetiére et al. review the theoretical and experimental findings on the thermal phenomena in non-contacting face seals. They present a general TEHD model of a face seal including steady-state kinematic and dynamic models, simplified Navier-Stokes equations for turbulent flow, a general form of the Reynolds equation and the energy equation with appropriate boundary conditions. In the first part [12], they present a numerical calculation procedure for the formulated mathematical model and the determined distributions of temperature fields both in the rings and the fluid film. In the second part [13], they perform an extensive parametric analysis
for different parameters of the seal and heat transfer conditions. The authors emphasize high conformity of the simulation results with the experimental data [12].

In this paper, the three-dimensional thermohydrodynamic model of a face seal was solved numerically using a specially developed computer program. The equations with partial derivatives were solved by applying the Finite Volume Method (FVM), which was different from the approach discussed in Refs. [6, 7, 11, 12], where the finite element and finite difference methods were used. Moreover, the energy equations were solved by introducing a little-known boundary condition [14] for a predetermined temperature distribution at the interface between the fluid film in the clearance gap and the surrounding fluid. The major aim of the numerical calculations was to determine the effect of the operational parameters on the distributions of temperature in the elements of a non-contacting face seal.

2. Governing equations and boundary conditions

The thermohydrodynamic (THD) model of a non-contacting face seal comprises a complex system of interdependent differential equations with a number of boundary conditions that are necessary to find a correct solution. The theoretical considerations concerning non-contacting seals, e.g. Refs. [4, 5, 9, 11, 15] assume that the flow through the radial clearance gap is steady-state, the fluid is Newtonian and incompressible and the mass forces for a small Reynolds number (Re<800) are negligible. The pressure distribution in the fluid film can be described by the Reynolds equation:

$$\nabla \left[ h^3 \nabla p - 6 \mu \omega r h \theta \right] = 0 ,$$

where:

- $\mu$ – dynamic viscosity coefficient;
- $\omega$ – angular velocity;
- $h$ – function describing the height of the clearance gap dependent on the coordinates $(r, \theta)$;
- $p$ – distribution of pressure in the fluid film and $r$ - radius.

Except for the pressure equation, it is necessary to consider the energy equation (the Fourier-Kirchhoff equation), which, after some simplifications required for non-contacting face seals [12], can be written as:

$$\rho_c \nu \left\{ \nu_r \frac{\partial T^f}{\partial r} + \frac{\nu_\theta}{r} \frac{\partial T^f}{\partial \theta} + \nu_z \frac{\partial T^f}{\partial z} \right\} = \mu \left\{ \left( \frac{\partial \nu_r}{\partial z} \right)^2 + \left( \frac{\partial \nu_\theta}{\partial z} \right)^2 \right\} + k \frac{\partial^2 T^f}{\partial z^2},$$

where: $\nu_r, \nu_\theta$ – distributions of velocity along the radial and circumferential coordinates $\theta$, respectively; $T^f$ - temperature distribution in the fluid film.

Equation (2) describes changes in the temperature in the fluid film filling the radial clearance gap. The left-hand side of the above equation represents the heat transfer by convection, while the right-hand side terms are: the dissipation of the kinetic energy of the fluid and the thermal energy conduction according to the Fourier’s law along the height of the radial clearance gap, respectively.
The rings are usually solid structures conducting heat in each of the three directions of the coordinate system. Heat generated in the clearance gap is removed to the surroundings mainly through the rings and then further via natural convection to the surrounding fluid. For steady-state conditions, the three-dimensional distribution of temperature in the sealing rings (Fig. 1) at constant heat conduction coefficients is described with the Laplace differential equation:

$$\frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} = 0. \quad (3)$$

Like in Ref. [11], the temperature distribution was determined for an FMS-type non-contacting face seal (Fig. 1), in which the surfaces of the flexibly mounted stator $S_0^f, S_1^f, S_2^f, S_3^f$ and the surfaces of the rotor $S_4^f, S_5^f$ are completely separated from the surroundings and no heat transfer occurs.
The fourth kind boundary condition is satisfied on the end faces of the rings - the rotor $\overline{S}_r^T$ and the stator - $\overline{S}_s^T$ being in contact with the fluid in the radial clearance gap. This is the case of heat transfer by conduction, where the heat flux at the interface between the rings and fluid film has the same value as that flowing into the seal elements.

$$
\frac{k_f}{\partial z}\left|_{z=h^f}\right. = k_r\frac{\partial T'}{\partial z}\left|_{z=0}\right. \quad \text{and} \quad T' = T',
$$

(4)

$$
\frac{k_f}{\partial z}\left|_{z=h^s}\right. = k_s\frac{\partial T'}{\partial z}\left|_{z=0}\right. \quad \text{and} \quad T' = T^s.
$$

The heat transfer between the surfaces of the rings $\overline{S}_r^T$ and $\overline{S}_s^T$ (Fig. 1) and the surrounding fluid occurs through natural convection (Fig. 2), both for the stator and the rotor and can generally be written as:

$$
-k\frac{\partial T}{\partial r}\bigg|_{r=r_s} = H_w \left(T_{\text{in}} - T_w\right),
$$

(5)

where: $k_s, k_f, k_r$ - heat conduction coefficients for the stator, the fluid film and the rotor, respectively, $H_w$ - convective heat transfer coefficient, $T_\infty$ - surrounding fluid temperature, $T'$ - rotor temperature.

For the energy equation, the boundary condition was the nonlinear temperature distribution at the inlet to the radial clearance gap approximated with a parabolic function [14, 16].

$$
T'(r = r_s, \theta, z) = \left(3T_h^f + 3T_0^f - 6T_m\right)\left(\frac{z}{h}\right)^2 + 
$$

$$
+ \left(6T_m - 2T_h^f - 4T_0^f\right)\left(\frac{z}{h}\right) + T_0^f
$$

(6)

where: $T_h^f = T^f (z = h), T_0^f = T^f (z = 0), T_m = \frac{1}{h} \int_0^h T' dz$ - average temperature in the clearance gap.

The variability of the temperature field in the fluid film separating the rings causes a change in the dynamic viscosity of the fluid. The relationship between the dynamic viscosity and the temperature of the fluid is discussed in Refs. [5, 11, 13]. This relationship is described by an exponential law:

$$
\mu = \mu_\infty e^{(b(T - T_\infty))},
$$

(7)

where: $\mu_\infty$ - dynamic viscosity of the fluid at temperature $T_\infty$; $b$ - thermoviscosity coefficient, for water $b = 0.0175 [1/K]$.

3. Calculation procedures

The mathematical model formulated above was solved numerically. The Reynolds equation (1) was solved using the finite volume method, whereas the energy equation (2) and the heat conduction equations (3) were solved with the finite element method. Generally, the algorithm to solve these
equations is as follows. The initial distribution of temperature in the radial clearance gap and the sealing rings is assumed to be known, and this temperature is the same as that of the surrounding fluid $T'$. The next step involves determining the height of the radial clearance gap, the pressure and dynamic viscosity of the fluid in the clearance gap and the components of the velocity of flow along each axis of the coordinate system. Thus, the temperature distribution of the fluid is calculated from the energy equations (2); the heat conduction equations (3) are solved using appropriate boundary conditions. This step is repeated until the temperature distribution is determined with the required accuracy. This procedure is repeated for the subsequent time step. The proposed algorithm of numerical solution is actually a fragment of a complex computer algorithm to solve the fundamental equations of dynamics and determine the displacements of the flexibly mounted ring depending on the predetermined geometrical and operating conditions. The complete algorithm was used to write a computer program [17] in C++, which can be applied to conduct extensive parametric heat transfer analysis with a view to selecting the most suitable geometry and material of the sealing rings and operating conditions.

4. Calculation results

The numerical calculations were performed in order to determine the effect of selected parameters characterizing the seal operation, i.e. the rotational speed, the pressure of the sealed fluid and the nominal height of the clearance gap, on the distributions of temperature in the rings and the fluid film in the clearance gap.

Table 1 shows the analyzed geometrical and operational parameters of the non-contacting face seal.

Table 1. Design and operational parameters

| Design parameters                  | Operational parameters          |
|-----------------------------------|---------------------------------|
| Inner radius                       | 0.035 [m]                       |
| Outer radius                       | 0.040 [m]                       |
| Stator length $e_s$               | 0.005 [m]                       |
| Rotor length $e_r$                | 0.005 [m]                       |
| Nominal height of the clearance gap | 1, 5, 10 x10^{-6}[m]            |
| Angular velocity $\omega$         | 900, 1200, 1500 [rad/s]         |
| Pressure along the inner radius    | 0 [Pa]                          |
| Pressure along the outer radius    | 10 x10^{5} [Pa]                 |
| Fluid reference temperature $T_w$  | 20 [°C]                         |
| Convective heat transfer coefficient $H_w$ | 3000 [W/m²K] |

The physical properties of two most common materials used for sealing rings are presented in Table 2.

Table 2. Physical properties of the materials used for the rings

| SiC - silicon carbide | WC - tungsten carbide |
|-----------------------|-----------------------|
| Young’s modulus $E$   | 410 [GPa]             |
| Poisson’s ratio $\nu$ | 0.14                  |
| Thermal conductivity $k_r$ | 120 [W/mK]         |
| Young’s modulus $E$   | 668 [GPa]             |
| Poisson’s ratio $\nu$ | 0.24                  |
| Thermal conductivity $k_r$ | 42 [W/mK]          |
The calculation results were provided in the graphical form. Figure 3 shows distributions of temperature fields in the fluid film and the sealing rings made of two different materials for three values of rotational speed of the rotor: 900, 1200 and 1500 \[ \text{rad/s} \]. Figure 5 compares changes in the temperature on the end face of the rotor along the radius.

Fig. 3. Temperature distribution in the fluid film and the sealing rings for different values of the angular velocity $\omega$.
From the diagrams in Figs. 3 and 4 it is clear that the angular velocity has a considerable influence on the temperature distribution. A 75% increase in this parameter results in more than a double increase in the maximum temperature of the fluid in the clearance gap both for silicon carbide (SiC) and tungsten carbide (WC). This is due to a rise in the frictional resistance (tangential stresses in the fluid) and, accordingly, an increase in the heat flux. Moreover, the temperatures for the rings made of tungsten carbide are almost twice as high as those for silicon carbide.

In Figs. 5 and 6, we can see the distributions of temperature fields in the fluid film and the sealing rings for different values of the nominal height of the clearance gap, $h_0$: $1 \times 10^{-6}$; $3 \times 10^{-6}$; $5 \times 10^{-6}$ [m].
Fig. 5. Temperature distribution in the fluid film and the sealing rings for different values of the clearance gap height $h_c$. 
The analysis of the calculation results shows that the temperature in the system decreases with an increase in the height of the radial clearance gap. If the height of the clearance gap rises by 4 micrometers, the temperature along the radius $r_i$ drops by nearly $120^\circ C$ for silicon carbide and by $180^\circ C$ for tungsten carbide. In the case of rings made of tungsten carbide (Fig. 7), the temperature is about $70^\circ C$ higher than in the case of silicon carbide rings at $h_o=1 \times 10^{-6} [m]$. If the clearance gap height is $h_o=5 \times 10^{-6} [m]$, the difference is smaller.

Another objective was to determine the distributions of temperature fields for various pressures of the sealed fluid $p_o$ along the outer radius $r_o$. 

![Fig. 6. Changes in the rotor temperature for different values of the clearance gap height $h_o$.](image)
Fig. 7. Distribution of temperature in the fluid film and the sealing rings for different values of pressure $p_o$
The calculation results confirm that the changes in the pressure of the sealed fluid do not have a considerable effect on the distribution of temperature in the fluid film and the sealing rings provided that the height of the clearance gap is constant. In their parametric calculations, Brunetiere et al. [13] obtained similar results. It should be noted that the mechanical deformations of the rings caused by the working pressure were not considered here, although it is obvious that the deformations change the shape and height of the clearance gap.

Further studies are required to analyze the thermal phenomena in a face seal including ring deformations.

5. Conclusion

The aim of this paper was to study the influence of the operational parameters on the temperature field in the fluid film and the sealing rings of non-contacting liquid face seals. The considerations included the relationship between selected design and operational parameters and the distribution of temperature in the fluid film and the rings.

The results of numerical calculations confirmed the findings presented in Refs. [11, 13] that an increase in the rotational velocity of the ring and a reduction in the clearance gap height caused a rise in the temperature of the fluid.

It was shown that the different distributions of temperature fields in the sealing rings were highly dependent on the operating conditions.

Due to the variability of the parameters, it might be necessary to remove large heat fluxes from the fluid film to the surroundings through the sealing rings. The large thermal deformations of the rings may contribute to considerable changes in the height of the clearance gap, and in consequence, to changes in the dynamic properties of the whole seal. There exists a close relationship between the geometry of the radial clearance gap, leakage and power losses in the seal. Estimating the temperature and thermal deformations at the design stage may prevent excessive fluid leakage and wear of the sliding surfaces of the sealing rings during operation.

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