A numerical solution of temperature distribution in the clearance and the sealing rings of the non-contact face seal

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Abstract. The study presents a numerical analysis of the temperature distribution in the clearance and the temperature distribution in the rings of a non-contact face seal. Non-contact face seals designed for turbine pumps to meet the requirements of lower leakage, longer life, and more frequent starting and stopping are included in this test. A thermo-hydro-mechanical model of the non-contact face seal clearance has been presented. Based on this model, dimensionless differential equations were derived to determine the temperature distribution in the clearance seal gap and the temperature distribution in sealing rings. The numerical solution of differential equations allowed the presentation of temperature characteristics for different parameters of the analyzed non-contact face seal. The analysis took into account hydrodynamic and thermodynamic phenomena in the clearance of the face seal depending on pressure, leakage flow rate and temperature of the sealing fluid, clearance dimensions, thermal parameters of the sealing rings, and operation parameters of the rotating machine. The temperature distribution in a non-contact face seal can also be used to determine the thermal deformation of the sealing rings. The presented numerical model is useful in the design of non-contact face seals because it allows the selection of parameters stabilizing the thermo-hydro-mechanical state of the two mating sealing rings.

1. Introduction

Non-contact face seals, lubricated with seal fluid (liquid, gas), have potentially low leakage, low friction, and low surface wear in high pressures on high-speed rotor machines. Non-contact face seals are successfully used in industrial pumps, compressors, and turbines and also are now increasingly finding their way into demanding aerospace applications [1], [2]. Higher effectiveness of a face seal is obtained by introducing some design and parametric changes that will cause the hydrodynamic effect in the clearance. In a non-contacting face seal, a radial (face) or axial (along the shaft axis) clearance is formed by two sliding elements, one of which is rigidly attached to the shaft or housing and the other flexibly mounted to the spring or controlling element. The leak tightness, durability, and operational reliability of such a seal depend on the design of the sealing joint as well as the properties and working parameters of the sealing fluid. The influence of perturbation factors related to the flow rate, change in the temperature of the sealing fluid, and vibrations of the machine shaft should also be accounted for.

This work discusses a model of a non-contact face seal in a turbomachine (fluid flow machine) used for identifying temperature distribution in the clearance in the function of seal parameters in the conditions of viscous friction and boundary Coulomb friction. [3], [4]. For the assumed calculation model of the non-contact face seal, the temperature distribution as a function of pressure, clearance high, and shaft speed will be determined. These seals when operating above a certain critical speed,
enter into a condition known as thermoelastic instability (TEI) that causes localization of pressure and excessive heat generation that manifests themselves in the form of hot spots at the sliding surface [5].

Non-contact face seals with water as the sealing medium have been designed for turbine pumps to meet the requirements of lower leakage, longer life, and more frequent starting and stopping [6]. The main task of non-contact face seals is to maintain a fluid-filled clearance separating the mating seal rings. By properly adjusting the clearance height can minimize the leakage of sealing fluid and significantly reduce power losses and heat flux in non-contact face seals. The non-contacting face seal consists of two mating sealing rings: the stator (stationary ring) mounted in the machine housing, and the rotor (rotating ring) mounted on a rotary shaft of the turbo-machines. Fig. 1 presented a cross-section of a non-contacting face seal that is being analyzed.

Figure 1. Cross-section of a non-contacting face seal: 1 – machine housing, 2 – joining mandrel, 3 – secondary static seal, 4 – stator, 5 – flexibly mounted rotor (FMR), 6 – secondary seal, 7 – clamping ring, 8 – spring, 9 – casing, 10 – clamping mandrel, 11 – sealing medium, 12 – working medium, 13 – seal clearance, 14 – rotating shaft.

Reliability and failure-free operation of non-contact face seals depend on the heat exchange conditions in the sealing clearance and thermal deformations of the mating rings [7]. In analytical studies, it is assumed that the heat flux generated in the fluid film in the sealing clearance is transferred to the sealing rings. Then the heat from the sealing rings is transferred to the sealing medium (see Fig. 1).

Simplifying the physical phenomena occurring in a non-contact face seal, one can write a mathematical model useful for analytical solutions of hydrodynamic and thermodynamic problems. The following simplifying assumptions are usually made [8]: the sealing gap is axially symmetrical, the liquid is Newtonian and incompressible, mass forces are neglected. The values of basic parameters sealing medium (water) and steel rotor rings of the non-contact face seal used in the analytical model solutions are presented in Table 1.

Estimating of the heat generation rate is important in designing non-contact face seals. For low viscosity fluids such as water, heat is mostly generated mainly as a result of sliding friction at the ring seals. The frictional heat is a function of mechanical load (spring force) and fluid load (fluid pressure), friction coefficient, rotational speed of the seal's ring, and sliding contact area. The heat generated in the clearance and the sealing rings must be dissipated effectively to avoid thermal degradation of the face seal. Both conductivity and convective heat transfer are important for the sealing efficiency in a non-contact face seal. For this reason, the thermal conductivity of the stator and rotor is important. Also, the heat generated in the clearance on the surface of the sealing rings is dissipated in the fluid by
convective heat transfer. Most non-contact seals fail long before their wear, which is why high temperature is identified as one of the main causes of their failure [9], [10].

Table 1. Basic parameters and it values.

| Parameters                      | Values                                      |
|---------------------------------|---------------------------------------------|
| \( r_i \) – inner radius of the rings | \( r_i = 0.04 \) m                          |
| \( r_o \) – outer radius of the rings | \( r_o = 0.045 \) m                        |
| \( h \) – the nominal height of clearance (film thickness) | \( h = 1 \times 10^{-6} \) m            |
| \( \omega \) – angular speed       | \( \omega = 500–1500 \) rd/s               |
| \( T_m \) – absolute temperature of sealing medium | \( T_m = 20^\circ \) C                  |
| \( \rho \) – medium density       | \( \rho_0 = 998 \) kg/m\(^3\)                 |
| \( c_p \) – fluid specific heat   | \( c_p = 4189.9 \) J kg\(^{-1}\) K\(^{-1}\) |
| \( \mu \) – fluid dynamic viscosity | \( \mu = 0.001 \) Pas                        |
| \( l_1 \) – thickness of a rotor  | \( l_1 = 0.005 \) m                         |
| \( l_2 \) – thickness of stator   | \( l_2 = 0.005 \) m                         |
| \( p_i \) – gauge pressure along the inner radius | \( p_i = 0 \) Pa                         |
| \( p_o \) – gauge pressure along the outer radius | \( p_o = 1 \times 10^6 \) Pa          |
| \( \lambda_1 \) – thermal conductivity in the rotor | \( \lambda_1 = 42 \) W m\(^{-1}\) K\(^{-1}\) |
| \( \lambda_2 \) – thermal conductivity in the stator | \( \lambda_2 = 120 \) W m\(^{-1}\) K\(^{-1}\) |
| \( \alpha_a \) – average heat transfer at the face of the ring | \( \alpha_a = 545 \) W m\(^2\) K\(^{-1}\) |
| \( \alpha \) – forced convection heat transfer for water | \( \alpha = 1800 \) W m\(^2\) K\(^{-1}\) |
| \( \alpha_c \) – natural convection heat transfer for water | \( \alpha_c = 300 \) W m\(^2\) K\(^{-1}\) |

The thermo-hydro-mechanical schematic diagram model of the non-contact face seal clearance is shown in Fig. 2. The sealing medium of the main pump seal is water; which is regarded as an incompressible fluid. In the numerical tests, the working fluid as air under atmospheric pressure was assumed.

Figure 2. Schematic diagram of the thermal-hydro-mechanical model of a non-contact face seal clearance: 1 – rotor (rotating ring), 2 – stator (stationary ring).

The following simplifying assumptions were made in the non-contact face seal model: the continuous laminar flow of sealing medium; the seal rings are rigid; the effect of fluid inertia force is ignored; the velocity gradient in the film thickness direction is ignored; Newton fluid flow is isothermal in a steady state. When the non-contact sealing surface is in the parallel face state, the clearance \( h \) is a constant value, and then the radius \( r \) is the only integral variable. The basic characteristics of the flow through the parallel clearance of non-contact face seals are determined by the Reynolds equation [11], but the article considers the axisymmetric problem.
2. Determination of temperature distribution in the clearance of the non-contact face seals

The isothermal heat flux in the clearance of non-contact face seals is determined by equation [12]

\[ \dot{Q}_2 + \dot{Q}_1 - P_h = \dot{Q}_r - \left( \frac{\partial \dot{Q}_r}{\partial r} dr \right) = -\frac{\partial \dot{Q}_r}{\partial r} dr \quad (1) \]

Equation (1) has been simplified to the following form

\[ \frac{\partial \dot{Q}_r}{\partial r} dr + \dot{Q}_{12} = P_h \quad (2) \]

After introducing a dimensionless radius \( \bar{r} = \frac{r - r_i}{r_o - r_i} \), was obtained

\[ \frac{\partial \dot{Q}_r}{\partial \bar{r}} d\bar{r} + \dot{Q}_{12} = P_h \quad (3) \]

where \( \dot{Q}_r \) is the heat transfer in the sealing gap,

\[ \dot{Q}_r = c_p \dot{m}_r \left( T(\bar{r}) - T_m \right) = c_p \rho q_L \left( T(\bar{r}) - T_m \right) = c_p \rho q_L \Theta \quad (4) \]

where \( \dot{m}_r \) is the mass flow rate, \( \Theta \) is the excess temperature in the clearance, \( \Theta(\bar{r}) = T(\bar{r}) - T_m \), \( q_L \) is the volume leaks (volume flow rate) through the clearance between the seal rings, \( \rho \) is the density of the working medium,

\[ q_L = \frac{\pi h^3}{6 \mu \ln \left( \frac{r_o}{r_i} \right)} \left[ \frac{3 \rho}{20} \left( r_o^2 - r_i^2 \right) \omega^2 + \left( p_o - p_i \right) \right] \approx K_o \omega^2 + K_p \left( p_o - p_i \right) \quad (5) \]

where:

\[ K_o = \frac{\rho \pi h^3}{40 \mu \ln \left( \frac{r_o}{r_i} \right)} \left( r_o^2 - r_i^2 \right), \quad K_p = \frac{\pi h^3}{6 \mu \ln \left( \frac{r_o}{r_i} \right)} \]

After taking the flow rate (4) into account, the heat transfer in the sealing gap is

\[ \dot{Q}_r = c_p \rho \left( K_o \omega^2 + K_p \left( p_o - p_i \right) \right) \Theta \quad (6) \]

Dissipated power was determined based on internal friction in the clearance of non-contact face seal [13]:

\[ P_d = \frac{2 \mu \pi r_m^3}{h} \omega^2 D(\bar{r}) d\bar{r} = K_d \omega^2 D(\bar{r}) d\bar{r} \quad (7) \]

where \( A \) is the surface of the sealing rings, \( A = 2 \pi r_m \, dr \), \( r_m \) is the mid radius of the seal's ring surfaces, \( r_m = \sqrt{r_o r_i} \), \( \omega \) is the shaft angular speed, \( v = \omega r_m \), \( v \) is the linear velocity, \( K_d \) is the factor of dissipated power,

\[ K_d = \frac{2 \mu \pi r_m^3}{h} \left( r_o - r_i \right) \quad (8) \]
$D(\bar{r})$ is the dissipation function along the clearance length assumed according to [14]

$$D(\bar{r}) = \sin^3(\pi \bar{r}) \quad (9)$$

$\dot{Q}_{12}$ – heat transfer at the face of the seal rings,

$$\dot{Q}_{12} = \dot{Q}_1 + \dot{Q}_2 = \lambda_1 a_1 \tanh(a_1 l_1) 2 \pi r_m (r_o - r_i) \Theta d\bar{r} +$$

$$+ \lambda_2 a_1 \tanh(a_2 l_2) 2 \pi r_m (r_o - r_i) \Theta d\bar{r} = k S \Theta d\bar{r} \quad (10)$$

where: $k = \lambda_1 a_1 \tanh(a_1 l_1) + \lambda_2 a_2 \tanh(a_2 l_2)$, $a_1 = \sqrt{\frac{2 \alpha_o}{\lambda_1} \frac{r_o}{r_o^2 - r_i^2}}$, $a_2 = \sqrt{\frac{2 \alpha_o}{\lambda_2} \frac{r_o}{r_o^2 - r_i^2}}$,

$S = 2 \pi r_m (r_o - r_i)$

The dimensionless differential equation is given in the form

$$\frac{\partial \dot{\Theta}}{\partial \bar{r}} + K_i(\omega) \Theta = K_\phi(\omega) \sin^3(\pi \bar{r}) \quad (11)$$

where:

$$K_i(\omega) = \frac{k S}{c_p \rho (K_\omega \omega^2 + K_p p_o)} \quad (12)$$

$$K_\phi(\omega) = \frac{K_d \omega^2}{c_p \rho (K_\omega \omega^2 + K_p p_o)} \quad (13)$$

Based on the numerical solution of the differential equation (11), the distribution of excess temperature in the sealing clearance for the adopted hydrodynamic and geometric parameters of the non-contact face seal was obtained (see Fig. 3).

**Figure 3.** Distribution of excess temperature $\Theta(\bar{r})$ along in the sealing clearance for dissipation function $\sin^3(\pi \bar{r})$.

And the temperature distribution $T(\bar{r})$ along in the sealing clearance is written as follows

$$T(\bar{r}) = T_m + \Theta(\bar{r}) \quad (14)$$
The temperature distribution $T(\vec{r})$ along the sealing clearance depends on the medium temperature $T_m$ and the excess temperature in the clearance $\Theta(\vec{r})$. Optimal hydrodynamic and geometric parameters of the non-contact face seal can be selected based on the temperature distribution $T(\vec{r})$ along with the sealing clearance.

### 3. Determination of temperature distribution in the sealing ring

Based on the direction of the heat flux and mechanical energy in the sealing clearance and the sliding rings of the non-contact face seal shown in Fig. 2, the temperature distribution in the axial direction of the sealing ring was determined.

The one-dimensional heat flux in the axial direction in the sealing ring (rotor) was determined [15]

$$\dot{Q}_z - \left( \dot{Q}_z + \frac{\partial \dot{Q}_z}{\partial z} \right) - \dot{Q}_0 = 0$$

Equation (15) has been simplified to the following form

$$\frac{\partial \dot{Q}_z}{\partial z} dz + \dot{Q}_0 = 0$$

where $\dot{Q}_z$ is the heat flux in the axial direction, $\dot{Q}_0$ is the heat transferred per unit time for natural convection between the ring surface and the sealing fluid,

$$\dot{Q}_0 = \alpha_o A_o \left[ T(z) - T_m \right]$$

where $T(z)$ is the temperature difference between the surface and the fluid, $\alpha_o$ is the free convective heat transfer coefficient, $A_o$ is the heat transfer surface of the ring,

$$A_0 = 2 \pi r_o \, dz$$

Finally given

$$\dot{Q}_0 = \alpha_o \, 2 \pi r_o \, [T(z) - T_m] \, dz$$

The second term of the equation (16) describes the thermal conductivity in the sealing ring according to Fourier's law,

$$\frac{\partial \dot{Q}_z}{\partial z} = -\lambda \, A \frac{d^2 T}{dz^2}$$

where $A$ is the cross-sectional area of the sliding ring,

$$A = \pi \left( r^2 - r_o^2 \right)$$

By transforming the differential equation (16) of the temperature distribution in the axial direction of the sealing ring, the following is obtained

$$\frac{d^2 T}{dz^2} - a^2 \left[ T(z) - T_m \right] = 0$$

where $a$ is a constant coefficient,

$$a = \sqrt{\frac{2 \alpha_o \, r_o}{\lambda_1 \left( r^2 - r_o^2 \right)}}$$
After introducing the excess temperature $\Theta(z) = T(z) - T_m$ into the equation (22), then we obtained

$$\frac{d^2 \Theta}{dz^2} - a^2 \Theta = 0$$  \hspace{1cm} (24)

The following initial conditions to the numerical solution of the differential equation (24) were introduced,

for $z = 0$: $\Theta = \Theta_a$, and for $z = l_1$: $\frac{d \Theta}{dz} = 0$.

where $\Theta_a$ is the average temperature on the side surface of the face seal as the arithmetic average of temperatures,

$$\Theta_a = \frac{T_m + T_c}{2}$$  \hspace{1cm} (25)

The analytical solution of the differential equation (24) for boundary conditions is as follows

$$\Theta(z) = \frac{e^{a(l_1 - z)} + e^{-a(l_1 - z)}}{e^{a l_1} + e^{-a l_1}} \Theta_a$$  \hspace{1cm} (26)

Figure 4 compares the distribution of the excess temperature $\Theta(\bar{z})$ in the axial direction $\bar{z}$ of the sealing ring for the numerical and analytical solutions. The dimensionless coordinate $\bar{z}$ for the thickness $l_1$ of the sealing ring ($\bar{z} = z/l_1$) has also been introduced. The curve in red results from the numerical solution of the second-order differential equation (24), and the curve in blue results from the analytical equation (26). As shown in Fig. 4, the numerical and the analytical solution of the excess temperature distribution $\Theta(\bar{z})$ in the axial direction $\bar{z}$ of the sealing ring overlap. It follows that when the excess temperature is distributed in the axial direction of the sealing ring, the analytical solutions give accurate results.

**Figure 4.** Comparison of the distribution of excess temperature $\Theta(\bar{z})$ in the axial direction $\bar{z}$ of the sealing ring: 1 - analytical solution (blue line), 2 – numerical solution (red line)
4. Conclusions
In the numerical solution of the temperature distribution in the clearance and sealing rings of the non-contact face, a seal rings model is adopted in which the thermodynamic and hydrodynamic conditions are stabilized. As part of the numerical test, non-contact face seals designed for turbine pumps were included. For turbine pumps, the following requirements must be met: low leakage and longer service life with more frequent starting and stopping. Since the disturbance of heat flow in these pumps can cause changes in thermal and hydrodynamic conditions in the non-contact face seal, therefore, a detailed numerical analysis of the temperature distribution in the clearance and sliding sealing rings was necessary. A thermo-hydro-mechanical model of the non-contact face seal clearance has been presented. Based on this model, dimensionless differential equations were derived to determine the temperature distribution in the clearance seal gap and the temperature distribution in sealing rings. The numerical solution of differential equations allowed the presentation of temperature characteristics for different parameters of the analyzed non-contact face seal. The analysis took into account hydrodynamic and thermodynamic phenomena in the clearance of the face seal depending on pressure, leakage flow rate and temperature of the sealing fluid, clearance dimensions, thermal parameters of the sealing rings, and operation parameters of the rotating machine. The temperature distribution in a non-contact face seal can also be used to determine the thermal deformation of the sealing rings. The presented numerical model is useful in the design of non-contact face seals because it allows the selection of parameters stabilizing the thermo-hydro-mechanical state of the two mating sealing rings.

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