Numerical analysis of the non-contacting gas face seals

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Abstract. The non-contacting gas face seals are used in high-performance devices where the main requirements are safety and reliability. Compliance with these requirements is made possible by careful research and analysis of physical processes related to, inter alia, fluid flow through the radial gap and ring oscillations susceptible to being housed in the enclosure under the influence of rotor kinematic forces. Elaborating and developing mathematical models describing these phenomena allows for more and more accurate analysis results. The paper presents results of studies on stationary ring oscillations made of different types of materials. The presented results of the research allow to determine which of the materials used causes the greatest amplitude of the vibration of the system fluid film-working rings.

1. Introduction

The stricter standards for emissions to the environment impose on the constructors of sealing nodes extremely strict requirements for the design of non-contacting face seals. It should be emphasized that these seals are used practically in all branches of industry and work in diametrically different operating and environmental conditions. The non-contacting face seals have been widely used in rotary machines for many years, including compressors and pumps, mixers, etc., where they play an extremely important role in the separation of sealed media from the environment. Main task for the seals is to maintain maximum tightness, but maintaining this task proves extremely difficult, due to a number of external and internal factors that affect the behavior of the adopted in construction height of the radial gap.

In case of the non-contacting face seals lubricated with gas, the methods enabling to maintain a stable layer of medium separating working rings is the use of various types of geometric modifications (microstructures) on the race of the working rings. Introduction of modifications in the form of conical, radial or helical channels and texturing of the face or impulse seals results in an increase in the force generated in the fluid film. The increase in ring separating force allows to maintain a stable fluid film and eliminates the contact of the sealing rings during operation. Another important aspect of the research on non-contact gas face seals is the study of vibration analysis of the sealing rings. Depending on the construction, the con-contacting face seals can be divided into two groups: FMS - Flexibly-Mounted Stator and FMR - Flexibly-Mounted Rotor. In both of these design variants, kinematic constraints are derived from the anti-ring rigidly attached to the shaft of the rotor machine.

The analyses literature presents the results of numerical calculations of a complex mathematical model including: radial gap height function, nonlinear Reynolds equation for compressible medium, and vibration dynamic equation for ring vibration in the housing. To correct the system of equations...
describing stationary ring oscillations, it is necessary to know the pressure distribution in fluid film. Practically in all scientific publications, the two-dimensional Reynolds equation is solved using numerical methods, usually Finite Volume Method (FVM) or Finite Element Method (FEM) [1], rarely, authors use a finite difference method [2] or meshfree methods [3].

While reviewing the literature of recent years, it can be concluded that practically all types of applied race modifications have a positive influence on the non-contacting gas face seals [4–9]. This has influence on the reduced wear on the work surface (sliding) of the rings while minimizing leakage, friction temperature and thermal distortions [10–15]. Achieving these results is possible in the case of a proper design of a sealing node containing non-contacting gas face seals. However, this requires considerable financial resources to carry out a large number of experiments. An alternative way to achieve the desired effect is to develop a computational aided design apparatus for this type of non-contacting seals with respect to operating requirements.

This paper presents the results of ring vibration analyzes, being housed in a housing for a non-contacting face seal based on the author’s computer program. Simulation studies were performed for a ring with conicity for four types of materials, i.e. carbon-graphite, carborundum, stellite, tungsten carbide. The subject matter was aimed at determining the influence of the physico-chemical properties of the material used on the ring [16–18], which was flexibly attached to the housing for correct operation of gas non-contacting face seal.

2. Gas face seal physical model

The design assumption concerning the non-contacting gas seals permits the operation of such seals with minimal leakage.

This means that between the front surfaces of the cooperating rings, there is a small radial piston of with height of up to several micrometers, choking leakage. Based on the work [4] and [19, 20], Figure 1 presented a scheme of the non-contacting FMS gas face seal.

![Figure 1](image-url)

**Figure 1.** Schematic cross section drawing of a non-contacting gas face seal; 1 – stator, 2 – rotor, 3 – taper angle, 4 – spring, 5 – casing, 6 – shaft, 7 – O-ring, 8 – steady pin.

The seal consists of two rings, of which one stator (1) is mounted flexibly in the housing, and the other rotor (2) rotates with the shaft (6) of the rotor machine. The radial gap, which occurs between the cooperating faces, is filled with a sealed medium and forms a gas cushion. Fluid film has certain elastic-damping properties and induces the transfer of vibrations from the rotor to the stator.
Figure 2. Schematic cross section of a non-contacting mechanical face seal.

Typically, execution errors, mounting errors or shaft defects during operation cause an unbalanced alignment of the sealing rings (Figure 2) and generation of additional axial and angular inducing oscillations from the rotor. With high rigidity of the layer separating the rings or under stable operating conditions, the sealing faces may be parallel to each other, meaning that they move synchronously and the gap is practically constant near the nominal value \( h_0 \).

3. Gas face seal mathematical model

The mathematical model of a non-contacting gas face seal is a discrete-continuous system of differential equations, i.e.:

- equations describing fluid motion in a film,
- equation of stream continuity (mass conservation equation)
- ring vibration equations flexibly mounted in the stator housing.

3.1 Lubrication analysis

In the works concerning the non-contacting gas face seals, the pressure distribution in the gap is determined on the basis of the Reynolds equation. [21, 22, 19] described by the dependency:

\[
\nabla \left[ p h^3 \nabla p - 6 \mu \omega r p \hat{e}_\theta \right] = 12 \mu \frac{\partial (p h)}{\partial t}
\]

where \( p = p(r, \theta) \) – pressure, \( h = h(r, \theta) \) – film thickness function, \( \mu \) – gas viscosity, \( \omega \) – shaft angular velocity.

Along with the boundary conditions and periodicity condition:

\[
p(r, \theta) \big|_{r=a} = p_i; \quad p(r, \theta) \big|_{r=a+} = p_o
\]

\[
p(r, \theta) \big|_{\theta=0} = p(r, \theta) \big|_{\theta=2\pi}
\]

As mentioned earlier in the operation of the device, both the stator and the rotor perform axial and angular vibrations, which has influence on the function describing the height of radial gap:

\[
h(r, \theta) = h^* - h^* = h_0 + \left( z^* - z^* \right) +
+ r \left[ (\alpha_x^* - \alpha_x^*) \sin(\theta) - (\alpha_y^* - \alpha_y^*) \cos(\theta) \right] + (r - r) \beta
\]

\( \alpha_x, \alpha_y, \) – angular vibration angles
where \( z', \alpha_x', \alpha_y' \) – stator axial and angular degrees of freedom, \( z', \alpha_x', \alpha_y' \) – rotor axial and angular degrees of freedom.

In the model in question, it was assumed that the rotating ring moving along the machine shaft performs harmonic angular vibrations \( \alpha_x', \alpha_y' \) – in the form of:
\[
\alpha_x' = \alpha' \cos(\omega t), \quad \alpha_y' = \alpha' \sin(\omega t)
\]
and their derivatives over time:
\[
\dot{\alpha}_x' = -\omega \alpha' \sin(\omega t), \quad \dot{\alpha}_y' = \omega \alpha' \cos(\omega t)
\]

On the other hand, the parameters \( \alpha_x', \alpha_y' \) are the answer in the form of angular vibrations of the stator. The mutual misalignment of a pair of rings can be represented graphically as on Figure 2.

\[
\alpha_x'(0) = \alpha'_z, \quad \alpha_y'(0) = 0, \quad \dot{\alpha}_x'(0) = 0, \quad \dot{\alpha}_y'(0) = \omega \alpha'
\]

From dependencies (5) and (6) result also the initial conditions for angular and axial displacements at \( t = 0 \) (eq. (7)).

### 3.2 Kinematic of face seal

The ring, which is flexibly attached to the housing, performs a complex movement that can be described by the system of differential equations of ordinary form:

\[
\begin{align*}
\ddot{z'} + c \dot{z'} + k z' + F_{seq} &= F_z \\
J \ddot{\alpha}_x' + c_k \dot{\alpha}_x' + k_k \alpha_x' - M_{si} - M_{seq} &= M_x \\
J \ddot{\alpha}_y' + c_k \dot{\alpha}_y' + k_k \alpha_y' - M_{seq} &= M_y
\end{align*}
\]

where \( c \) – axial damping coefficient, \( c_k \) – angular damping coefficient, \( F \) – force, \( J \) – transverse moment of inertia, \( k \) – axial stiffness coefficient, \( k_k \) – angular stiffness coefficient, \( m \) – stator mass, \( M \) – moment, \( M_{si} \) – moment due to stator initial misalignment, \( M_{seq} \) – equilibrium moment.

The equations of the movement flexibly attached in the stator housing are stored in an inertial system \( OXYZ \) (Figure 3) similarly as in the paper [1].

The constants \( c, k \) mean the damping factor and the axial elasticity, respectively \( c_k, k_k \) – coefficient of damping and angle elasticity. The coefficients depend on the properties of the elastic-damping elements of the stator mounting (Figure 4) and are linked together according to the following relations [19]:
\[
c_k = 0.5 c r_{s1}^2, \quad k_k = 0.5 k r_{s2}^2
\]

\( r_{s1} \) and \( r_{s2} \) respectively the radius on which the spring is mounted and the secondary seal – o-ring, \( r_{s1} = (r_o + r_b) / 2, \quad r_{s2} = r_b \). Where \( r_o \) - stator outer radius, \( r_b \) - stator balance radius.
Figure 3. Kinematic model of the non-contacting face seal: 1 – rotor, 2 – stator.

In the system of equations (8), it was assumed that: $m$ – ring weight, was determined according to dependency [20]:

$$m = (\frac{1}{3} + 2B/3) \frac{\pi rl(r'^2 - r^2)}{2}$$  \hspace{1cm} (10)

The transverse moment of inertia is described by the equation: $J_x = J_y = J = \int r^2 \, dm$.

Figure 4. Geometry of the stator.
The values of force $-F_z$ and moments $-M_x$ and $M_y$, occurring on the right side of the system of equations (8) depend directly on the pressure distribution in the gap. The pressure, however, depends on the height of the gap, the assumed flow pattern, and the operating conditions of the seal.

$$
F_z = - \int_{\Omega} p(r, \theta) \, d\Omega, \quad F_{eqz} = - \int_{\Omega} p_{eq}(r, \theta) \, d\Omega
$$

$$
M_x = - \int_{\Omega} p(r, \theta) r \sin(\theta) \, d\Omega, \quad M_{eqx} = - \int_{\Omega} p_{eq}(r, \theta) r \sin(\theta) \, d\Omega
$$

$$
M_y = \int_{\Omega} p(r, \theta) r \cos(\theta) \, d\Omega, \quad M_{eqy} = \int_{\Omega} p_{eq}(r, \theta) r \cos(\theta) \, d\Omega
$$

(11)

Static and dynamic forces and moments are calculated by numerical integration of the gas film pressure $p$ and $p_{eq}$ (in a state of static equilibrium) in the area $\Omega$ of the ring. Based on such formulated mathematical model, a computational algorithm and a C++ computer application has been developed allowing for simulation studies.

4. Results and discussion

Parameters used to perform numerical calculations for non-contacting gas face seal, were compared in Table 1. One adopted, as a working medium, the air with dynamic viscosity $\mu = 1.8 \cdot 10^{-5} \text{ (Pa} \cdot \text{s)}$.

One considered the sealing structure which working rings have an outer radius of 60 mm and an internal radius of 48 mm. It has also been assumed that the rotor performs rotational movement at an angular speed of 2094 rad/s, just as in operation. One assumed the stator material density respectively for Carbon-graphite $- \rho = 1.8 \cdot 10^3 \text{ (kg} \cdot \text{m}^{-3})$, Carborundum $- \rho = 3.1 \cdot 10^3 \text{ (kg} \cdot \text{m}^{-3})$, Stellite $- \rho = 8.4 \cdot 10^3 \text{ (kg} \cdot \text{m}^{-3})$, Tungsten carbide $- \rho = 1.5 \cdot 10^4 \text{ (kg} \cdot \text{m}^{-3})$. The calculations were made similarly as in the paper [5] the height of radial gap at level $h_o = 6 \text{ \mu m}$.

| Table 1. Characteristic operation parameters of the gas face seal. |
|---------------------------------------------------------------|
| Parameter                        | Value | Parameter                        | Value             |
|----------------------------------|-------|----------------------------------|-------------------|
| Stator inner radius $r_i$ (mm)   | 60    | Stator density $\rho$ (kg $\cdot$ m$^{-3}$) | var               |
| Stator outer radius $r_o$ (mm)   | 48    | Angular excitation frequency $\omega$ (rad/s) | 2094             |
| Pressure at inner radius $p_i$ (MPa) | 0.101 | Support axial stiffness $k$ (N $\cdot$ m$^{-1}$) | 5 $\cdot$ 10$^6$ |
| Pressure at outer radius $p_o$ (MPa) | 0.2  | Support axial damping $c$ (N $\cdot$ m$^{-1}$) | 300              |
| Dynamic viscosity $\mu$ (Pa $\cdot$ s) | 1.8 $\cdot$ 10$^{-5}$ | Nominal clearance height $h_o$ (m) | 6 $\cdot$ 10$^6$ |
| Balance ratio $\beta = (r_o^2 - r_i^2)/(r_o^2 - r_i^2)$ | 0.6   | Angular excitation amplitude $\alpha^*$ (rad) | 150 $\cdot$ 10$^6$ |
| Stator length $l$ (mm)           | 50    | Taper angle $\beta$ (rad)       | $0.5 \cdot h_o$  |
|                                  |       |                                  | $(r_o - r_i)$     |
Figure 5 (a) shows the changes in the minimum radial gap height depending on the number of revolutions for four different stator densities $\rho$. By analyzing graphically presented results, it can be seen that for the carbon graphite and carborundum, the period of transient vibration is elongated and the stabilization of the vibration amplitude of the ring flexibly attached to the housing occurs only after rotation of the shaft of the rotor machine.

The increase in inertia force occurs together with increase of the density $\rho$, and greater inertia forces the stator to respond more rapidly to the sudden motion caused by the rotor displacement. This is perfectly visible to the ring made of stellite. The transition period is significantly shorter (up to 3 revolutions), but the amplitude of the angular vibration increases, which has influence on a significant reduction in the minimum radial gap height.

![Graph showing the changes in the minimum radial gap height](image)

(a)

![Graph showing the transient dynamic response](image)

(b)

**Figure 5.** (a) change of minimum radial gap height, (b) transient dynamic response.

The use of stator material with even higher density $\rho$, destabilizes the operation of the working rings system. For the parameters collected in Table 1 for the non-contacting gas seal, the amplitude of
the angular vibration increases drastically, leading to the discontinuity of the gaseous film and the contact of the working ring races.

5. Conclusions

The presented paper presents simulations of the non-contacting face seal, lubricated with gas, with flexibly mounted stator. Numerical calculations consisted in solving a mathematical model involving the nonlinear Reynolds equation and the equation of motion for the stationary ring. The demonstrated example of numerical calculations proves that the type of material used on working rings in non-contacting face seal can influence its proper non-contact operation. Sealing operation destabilization can be caused by too large mass of the ring flexibly attached to the housing, and may also lead to excessive wear of the cooperating working rings. Another undesirable occurrence is the occurrence of dry friction and excessive temperature increase of the sealing ring leading to the occurrence of thermoelastic deformation. Increasing the amplitude of the angular vibration also causes an increase in leakage. In order to avoid costly repairs and increase the life of the non-contacting seals used, computer programs are developed that can be useful tools to assist in selecting seals for specified operating conditions.

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