Development of a simplified mathematical model for a foil gas bearing operation

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Abstract. A theoretical study of a gas flow process in a clearance, changing during operation between a compliant foil of a foil gas bearing and a rotor of a high-speed turbomachine, is presented. The approaches to mathematical simulation of the position of the compliant foil of the gas bearing, deforming under the influence of gas pressure on the outer surface and a system of distributed springs on the inner surface, are developed. An example of numerical simulation of the foil gas bearing in order to obtain an elastic line of the compliant foil under the influence of gas forces, acting in a wedge-shaped clearance, is considered.

1. Introduction

At the present time, worldwide interest in the development and implementation of the foil gas bearings in high-speed impeller machines for various purposes has increased [1]. This is due to the high rotational speeds of the impellers, the growing demand for their reliability, overall dimensions and weight. All this necessitates the improvement of bearing units.

Foil gas bearings have many advantages [2] in comparison with hydrodynamic and grease-lubricated bearings, for example, capability at high and low temperatures, low noise and vibration, durability and reliability, lack of contact between a stationary stator and a rotating rotor at nominal operating mode, low friction and heat generation, the absence of an extensive lubrication system and auxiliary systems for it, extremely low weight and dimensions.

Based on a review of gas bearings, it was revealed that one of the advanced designs is the foil gas bearing, developed by Dennis Weissert [3]. This bearing is a system of compliant and spring foils, installed in special grooves of the retaining cartridge, which is a stator (figure 1).
Figure 1. Compliant and spring foils in special grooves of the retaining cartridge.

This design provides minimal elastic preload using the compliant foils when the rotor is stationary. An antifriction coating on compliant foils surfaces prevents wear during the start and stop mode. When the rotor begins to turn, a gas flow starts to enter between the rotor and the compliant foil of bearing due to viscous friction, resulting in a wedge-shaped microchannel is formed. In this regard, an urgent task is to simulate the working processes that occur in the clearance and in the secondary (mechanical) elastic system of the journal foil gas bearing.

2. Structure of foil gas bearing

At the beginning, it is crucial to consider the construction and principle of operation of the foil gas bearing. The basic design of the bearing is shown in figure 2. The compliant and spring foils are pressed against the rotor when it is stationary. The quantities of this force and the prior deformation of the elastic beams of foil springs are assigned as low as practicable.

Figure 2. Construction of foil gas bearing:
1 – spring foil (spring of secondary (mechanical) stiffness); 2 – compliant foil; 3 – rotor (shaft); 4 – retaining cartridge; 5 – basic part.

At the beginning of rotation, gas starts to enter the clearance between the surfaces of the shaft and the compliant foil under the action of viscous friction, produced by the rotor of a turbomachine. The gauge pressure, forming in the clearance, tends to move away the compliant foil from the shaft. However, forces from many distributed beams of foil spring act on the compliant foil from the opposite side. The stiffness values of flat beams are comparable to the stiffness of the lubricating film, referred to the unit surface of the compliant foil. Therefore, a microchannel is formed between the outer surface of compliant foil and the shaft due to the combined deformation of the compliant foil and beams of foil spring. This microchannel has a complex «spoon-shaped form» [4], which is often represented as a wedge-shaped equivalent (figure 3).
Figure 3. Wedge-shaped microchannel:

\[ P_{\text{in}}, P_{\text{out}} - \text{gas pressure at the inlet and outlet of the microchannel; } \]
\[ h_{\text{in}}, h_{\text{out}} - \text{clearances at the inlet and outlet of the microchannel.} \]

The shape of the microchannel, in turn, determines the picture of the viscous gas flow, the shape of the pressure profile and also the value of the lifting force in the fluid film. The force arises due to the gas flow through the wedge-shaped clearance, which allows the rotor to become completely airborne. Moreover, the achievement of the steady state is characterized by equilibrium between the gas forces in the clearance and the elastic forces from the cantilevers of the spring of secondary (mechanical) stiffness. The design of foil spring is shown in figure 4.

Figure 4. Design of foil spring:

\[ l - \text{length; } h - \text{width; } 1 - \text{cantilevered beams (cantilevers).} \]

The spring, shown in figure 4, has beams with different dimensions along its width, which were invented by Robert Bosley [5]. This design provides a smooth change in the stiffness of the bearing elastic elements that increases the wear resistance of the system consisting of compliant and spring foils, installed in T-shaped retainers (figure 2). In addition, the relative sliding of a large number of contact points between cantilevered beams, smooth compliant foils and the inner surface of retaining cartridge during torsional vibrations of the rotor system provides high damping [6], which is explained by high Coulomb friction. This interaction between the elements of the bearing provides energy dissipation that allows the journal foil gas bearing to work under condition of moderate unbalance of the rotor. Above all, damping prevents unstable oscillations and limits their amplitude at resonance to a final value [7].

At the inlet and outlet of the microchannel, various pressure fields will be formed along the width of the bearing compliant foil depending on the clearance between it and the rotating shaft. However, the pressure field structure [8] will be similar to a pressure distribution, shown in figure 5.
The pressure field, shown in figure 5, has evident maximum $P_{\text{max}}$, which corresponds to the region before the minimum cross section in the wedge-shaped clearance (figure 3).

3. Simplified mathematical model for a foil gas bearing
Gas constantly circulates between the rotating shaft and the compliant foil at the nominal operating mode of the foil gas bearing. The gas flow regime in this bearing is laminar that is confirmed by the mathematical model of the gas flow, described in [2]. Gas influences on the compliant foil and spring foil through the pressure, deforming them. Thus, each beam of the foil spring is aligned under the influence of deformation on the inner surface of retaining cartridge in a certain way (figure 6).

Deformed foil spring forms a system of distributed beams which react to the gas load through the compliant foil. Since the pressure distribution is non-uniform, the beams must have different stiffness. Therefore, the task of modeling the position of the compliant foils of a gas bearing is reduced to creation of an equivalent system, where each cantilever is replaced by spring with the similar stiffness (figure 7).
The midline of the system, consisting of compliant and spring foils, is considered for simplifying the problem, as the maximum load arises there. In addition, the constant stiffness for the beams along the length of foil spring is taken. Ultimately, the task of determining the deformations of the compliant foil is reduced to calculation of a beam deflection with width \( h \), fixed in two supports, which is affected by the distributed load from the gas force \( q(x) \) (figure 8).

![Figure 8. Foil gas bearing schematic for mathematical modelling:](image)

1 – compliant foil; 2 – equivalent spring (1,2…n); \( q(x) \) – distributed load; \( R_A, R_B \) – reaction of support; \( l, h \) – length and width of compliant foil; \( P_{\text{max}}, P_{\text{min}} \) – regions with minimum and maximum pressure.

The compliant foil has a certain deflection before installation in T-shaped retainers (figure 2) which differs slightly from the deflection in the fixed position. The foil spring is flat in the initial state. However, the stiffness of its matrix is a negligible quantity in comparison with its cantilevers, considered undeformed before the start of shaft rotation. Therefore, it is possible to make an assumption at mathematical modelling that the distributed load \( q(x) \) acts on the undeformed system (figure 8). As a result of the analysis of this system, simultaneous equation (1) was obtained, in which the pressure from the gas is balanced by the elastic forces of the compliant and spring foils.

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\begin{align*}
\sum_{i=0}^{n} F_y &= 0: R_A + R_B + Q + \sum_{i=1}^{n} F_i = 0, \\
\sum_{i=0}^{n} M_A &= 0: R_A \cdot h + Q \cdot d + \sum_{i=1}^{n} F_i \cdot x_i = 0, \\
\sum_{i=0}^{n} M_B &= 0: R_B \cdot h + Q \cdot (h-d) + \sum_{i=1}^{n} F_i \cdot (h-x_i) = 0, \\
\phi(x) &= \int_0^h \frac{M_s(x)}{E \cdot I_z} \cdot dx + C_1, \\
f(x) &= \int C_1 \cdot x + D_1,
\end{align*}
\]

where \( Q = \int_A^B q(x) \cdot dx \) is the concentrated force, N; \( F_i = k_i \cdot f_i \) is the elastic force of equivalent spring, N; \( f_i \) is the deflection of equivalent spring, mm; \( k_i \) is the stiffness of equivalent spring, N/m; \( \phi(x) \) is
the slope of beam; \( f(x) \) is the deflection of compliant foil; \( M_z(x) \) is the moment of forces acting on compliant foil from distributed load and elastic force of equivalent springs, N\( \cdot \)m; \( E \) is the Young modulus of compliant foil, Pa; \( I_z \) is the moment of inertia with respect to axis \( Z \), kg\( \cdot \)m\(^2\); \( C, D \) are the integration constants; \( d \) is the arm of concentrated force \( Q \), mm.

4. Simulation of fluid film hydrodynamics
When the geometry of the bearing, its elements and, therefore, wedge-shaped microchannel are known, the schematic, shown in figure 9, can be used.

![Figure 9](image)

**Figure 9.** Schematic for numerical simulation of foil gas bearing:
1 – gas domain; 2 – compliant foil; 3 – equivalent springs; 4 – wedge-shaped clearance (thickness of minimal cross section is specified in the range of 0.15 to 0.05 mm);
\( d_{in} \) – inner diameter of gas domain is equal to diameter of rotor 31.7 mm.

To obtain the distributed load \( q(x) \) all along the compliant foil, it is necessary to solve the fluid film task by setting the boundary conditions for the rotation speed of shaft and eccentricity between the rotor and the retaining cartridge. Numerical methods make it possible, using successive approximations, to obtain the shape of a microchannel in which the gas load is balanced by reactions from the elastic elements of the foil spring (figure 10).

![Figure 10](image)

**Figure 10.** Compliant foil deflection along axis \( Y \) due to distributed load \( q(x) \) at equivalent spring stiffness in the range from 10 to 15 N/mm.
The values of the gas forces acting on the compliant foil, the moments in the supports, and stiffness of equivalent springs can be obtained through gasdynamic and static calculations to get the desired wedge-shaped clearance and elastic line of the compliant foil by transferring the results of the hydrodynamic simulation to the system of equations (1).

**Conclusions**

The design of the foil gas bearing and its principle of operation are considered. The simplified mathematical model for the foil gas bearing operation has been developed. An example of numerical simulation of gas bearing with known geometry of the compliant foil, spring foil and wedge-shaped microchannel is given. The next step of research is planned to compare the results of numerical simulation with experimental data in the nominal operating mode of the foil gas bearing in order to verify this mathematical model.

**References**

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