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Analysis of dynamic characteristics for face gas dynamic seal

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Abstract

Increasing of aircraft gas turbine engines and power plants efficiency is one of important design problems. One of possible ways of this problem solution is decreasing of leakage through the support seals. Some kinds of seals applied in supports of aircraft engines are known. A face gas dynamic seal is one of the most effective. The considerable part of damages in face gas dynamic seals takes place from dangerous vibration in it. A two-mass dynamic model which is the most applicable in practical cases for face gas dynamical seal is considered in the present paper. The theoretical and experimental analysis of dynamic condition of seal is fulfilled. Range of applicability for face gas dynamical seals on amplitude and frequency is defined. It is possible to use the results of present researches for development of new seal structures, and for application of seals at compressors with magnetic supports.

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Keywords: dynamic model; oscillations, speed, gap, deformations; seal; stiffness, leakage; frequency of rotation; supports of the aircraft engine

1. Introduction

To a present time the gas industry gained a large experience for gas compressors with electromagnetic bearings. Using of electromagnetic bearings in gas compressor units allows making it “dry”, without using of lubrication in rotor supports [1]. It has a significant influence on reliability of gas compressor units. Such machines, together with obvious advantages, have certain features that it is necessary to consider in its exploitation. An admissible range of amplitude of rotor vibration in electromagnetic bearings is determined by capabilities of face gas dynamic seals which are used in gas compressor units [1-3]. Main part of damage in a face gas dynamic seals takes place as a result

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of danger vibration. Therefore it is necessary to investigate the dynamics of face gas dynamic seals to identify their operational limitations [4-6]. To calculate the dynamic characteristics of face gas dynamic seals it is necessary to determine their static characteristics. Theoretical foundations of gas flow calculation in face gas dynamic seals it is possible to find in works of Muijderman [7], which considers an influence of gas compressibility, number and geometry of spiral grooves, modes of gas flow. In [8] mathematical model of face gas dynamic seals is modified for taking into account a sealing ring deformation from force and thermal factors. It provides engineering methods of calculation for real value and shape of gap, pressure distribution in the slot of seal, leaks, stiffness and damping coefficients in gas layer and face gas dynamic seal elements. Using of these methods allows adequately simulating of dynamic characteristics for face gas dynamic seals. Analysis of exploitation experience for gas compressor units with electromagnetic bearings shows that it is necessary to fulfill a research of face gas dynamic seals dynamics at rotor vibration amplitude till 250…300 μm for a frequency range 15…200 Hz.

2. Mathematical model of face gas dynamic seal

There are some dynamic models of face gas dynamic seals known in a present time. The most simple is used by Feodor Burgmann company [9]. It considers the face gas dynamic seal as a solid body (non-rotating seal ring) connected to the rotor and case by elastic elements (working layer and springs). This model considers only axial oscillations and doesn't take into account damping in the working layer.

In [8] a single-mass dynamic model of face gas dynamic seal is researched. It is considered as a composition of solid body (non-rotating seal ring), inertialess elastic-viscous suspension (working layer) and elastic element (spring). Also secondary seal is taken into account, which is represented by element with stiffness, damping and dry friction. Model with a relaxation mechanism of damping, which uses the damping force which is not acting directly between the mass of non-rotating ring and rotor part, but through elasticity, is used for description of gas layer. Three types of oscillations are possible is a model of the face seal: axial, angular and bending. Generally, axial, angular and bending displacements of the movable ring provide mutual influence due cross-linking. Therefore the angular beatings can excite an axial vibration in the seal, etc. It is shown experimentally [10] that this model is sufficient for the most of practical cases.

Analysis of existing and prospective face gas dynamic seal structures showed that the most correct is the dynamic model presented in Fig. 1. It is developed on a base of the above-described model [8] and consists of two masses. Clamp (Msp) is mounted in the body of a turbo-machine and pressed to non-rotating ring (Mr) with a set of springs with stiffness Ce.e. Mass of springs can be taken into account in the model by adding 1/3 mass of springs to the mass of the clamp.

A secondary seal, mounted between the non-rotating ring and the clamp, is presented as an element with stiffness (C1), damping (b1) and dry friction (R1). There is an inertialess viscous-elastic suspension (working layer Cdy) between the non-rotating ring (Mr) and the rotating bushing. The face of the rotor transmits an impact on face gas dynamic seal with axial and angular components of oscillation amplitude z1 and α1. The ring can have an additional bending oscillation component θ. Bending oscillation of the ring is compensated by elastic deformation of the sec-
ondary seal, thus they are not transmitted to the clamp. The analysis of such model with considered moving is absent in present time.

Three types of oscillation are possible in the face seal: axial, angular and bending. Oscillations of non-rotating ring are described by the system of motion equations:

\[ m \ddot{z}_2 + P + W_z = 0; \quad I \ddot{\alpha}_2 + M_\alpha + L_\alpha = 0; \quad I_p \ddot{\theta}_2 + M_\theta + L_\theta = 0, \]

Here \( m, I, I_p \) are mass and inertia moments of the ring; \( z_2, \alpha_1, \theta_\alpha \) are axial, angular, bending movement of the ring seal; \( P, M_\alpha, M_\theta \) are axial force and hydrodynamic moments acting on the ring from side of the gas layer; \( W_z, L_\alpha, L_\theta \) are force and moments acting on the ring from the back side. Impact of rotor on the system is considered as three components: axial \( z_1 = z_{1,0} \sin \omega t \); angular \( \alpha_1 = \alpha_{1,0} \sin \omega t \); bending \( \theta_1 = \theta_{1,0} \sin \omega t \). Here \( z_{1,0}, \alpha_{1,0}, \theta_{1,0} \) are amplitudes of impact.

In general, for research of face gas dynamic seal dynamics it is necessary to consider simultaneously all three equations of the system. Analysis of values of cross coefficients of stiffness and damping [8] shows that in this two-mass model (Fig. 1) these types of oscillations are practically possible: axial oscillations \( z_1 \rightarrow z_2 \rightarrow z_3 \); angular oscillations \( \alpha_1 \rightarrow \alpha_2 \rightarrow \alpha_3 \); composition of axial and angular oscillations \( z_1 \rightarrow z_2, \theta_1 \rightarrow \theta_3 \).

It is shown in [5] that it is possible to consider the axial and angular oscillations separately and to sum up a movement of rings. In [8] it is shown that for variable size of gap the pressure profile and the bending moment in gap seal change too. Therefore it is necessary to consider the axial and bending oscillations together.

Let we firstly consider the first type of oscillations. In this case, all elements of the system have only axial movement. If to consider the seal as a system with lumped parameters, it is possible to obtain the equations of motion of ring and clamp as:

\[ M_r \ddot{z}_2 + c_{dyn}(z_2 - z_1) + c_1(z_2 - z_3) + b_1(\dot{z}_2 - \dot{z}_3) = 0; \]
\[ M_{sp} \ddot{z}_3 + c_1(z_3 - z_2) + b_1(\dot{z}_3 - \dot{z}_2) + c_{ce}z_3 = 0. \]

Let we consider the harmonic excite \( z_1 = z_{1,0} \sin \omega t \). Axial displacement of the ring is \( z_2 = z_{2,0} \sin (\omega t + \varphi) \). The system of linearized equations for the axial oscillation of masses in face gas dynamic seal in operator form is written as follows:

\[ M_r s^2 \ddot{z}_2 + c_{dyn}(z_2 - z_1) + c_1(z_2 - z_3) + b_1(\dot{z}_2 - \dot{z}_3) = 0; \]
\[ M_{sp} s^2 \ddot{z}_3 + c_1(z_3 - z_2) + b_1(\dot{z}_3 - \dot{z}_2) + c_{ce}z_3 = 0. \]

Here \( s = i\omega, i = \sqrt{-1} \). Dry friction \( R_1 \) is equivalent to friction damping [8] and added to the damping \( b_1 \).

It is possible to represent a dynamic response of the gas layer as:

\[ c_{dyn} = c_{el} + i\omega b_1, \]

Here \( c_{el} \) and \( b_1 \) are elastic and damping components of the dynamic response of the layer.

The natural frequency of oscillation of the ring is \( \omega_0 = \frac{C_{el}}{\sqrt{M_r}} \).

The transfer function is \( K(s) = \frac{Z_2}{Z_1} \).

Let we introduce dimensionless parameters:

\[ \tilde{\omega} = \frac{\omega}{\omega_0}; \quad \tilde{D} = \frac{b_1}{M_r \omega_0^2}; \quad k_1 = \frac{M_{sp} \omega_0^2}{c_1 + c_{ce}}; \quad k_2 = \frac{b_1 \omega_0}{c_1 + c_{ce}}. \]

The transfer function is obtained with these parameters as:

\[ k(s) = \frac{(1 - k_1 \tilde{\omega}^2) - k_2 \tilde{D} \tilde{\omega}^2 + i \tilde{\omega}(k_2 + \tilde{D}(1 - k_1 \tilde{\omega}^2))}{(1 - k_1 \tilde{\omega}^2) - \tilde{\omega}^2(1 - k_1 \tilde{\omega}^2) - k_2 \tilde{D}} + i \tilde{\omega}(k_2 - k_2 \tilde{\omega} + \tilde{D}(1 - k_1 \tilde{\omega}^2)). \]
By separation of the real and imaginary components of the \( K(s) = U(\omega) + iV(\omega) \), it is possible to define amplitude-frequency (AFC) \( \mu_h(\omega) = (U^2 + V^2)^{0.5} = z_{2.0}^2/z_{1.0} \) and phase-frequency (PFC) \( \psi_0(\omega) = \arctan(V/U) = \varphi \) characteristics. Dependency of size of the gap on time is:
\[
\Delta h = z_{2.0} \sin(\omega t + \varphi) - z_{1.0} \sin \omega t,
\]
here \( z_{2.0}, z_{1.0} \) are amplitudes of movement of the ring and the end face of the rotor; \( \varphi \) is phase shift.

Let we consider the second oscillation type. In this case, all elements of the system have only angular displacement. The equations of motion of the ring and clamp have the form:
\[
\begin{align*}
I_r \ddot{\alpha}_2 + c_{\text{dyn},a}(\alpha_2 - \alpha_1) + c_1(\alpha_2 - \alpha_3) + b_{1a}(\dot{\alpha}_2 - \dot{\alpha}_3) &= 0; \\
I_{sp} \ddot{\alpha}_3 + c_{1a}(\alpha_3 - \alpha_2) + b_{1a}(\dot{\alpha}_3 - \dot{\alpha}_2) + c_{e,oa}\alpha_3 &= 0.
\end{align*}
\]

It is possible to calculate the dynamic coefficients for the angular oscillations with sufficient accuracy by the equation \([8] c_a = cR^2/2, b_a = bR^2/2\). Here \( c \) and \( b \) are the stiffness and damping for axial oscillations.

The transfer function \( K(s) = \alpha_2/\alpha_1 \) is similar to the previous case. It is more convenient to obtain a solution in a complex form.

Let we consider a third type of oscillation. In this case, rotor and clamp have only axial displacements and seal ring has only axial and bending displacements. The motion equations of ring and clamp have the form:
\[
\begin{align*}
I_r \ddot{z}_2 + c_{\text{dyn},a}(z_2 - z_1) + c_1(z_2 - z_3)r_0 + EJ\theta_2 &= 0; \\
M_r \ddot{z}_3 + c_{\text{dyn},z}(z_2 - z_1) + c_{\text{dyn},a}\theta_2 + c_1(z_2 - z_3) + b_1(\dot{z}_2 - \dot{z}_3) &= 0; \\
M_{sp} \ddot{z}_3 + c_1(z_3 - z_2) + b_1(\dot{z}_3 - \dot{z}_2) + c_{e,z}z_3 &= 0.
\end{align*}
\]

Here \( r_0 \) is radius of the location of elastic element; \( E \) is modulus of elasticity of ring; \( J \) is moment of ring inertia at a bend; \( c_{\text{dyn},a}, c_{\text{dyn},z} \) and \( c_{\text{dyn},z0} \) are dynamic response coefficients (the first letter of index refers to form of disturbing influence of rotor face, second one to form of ring displacement).

If to linearize the equations in a similar manner as in the first example, it is possible to obtain a system of three equations with three unknowns:
\[
\begin{align*}
A_1\theta_2 + B_1z_2 + C_1z_3 &= D_1z_1; \\
A_2\theta_2 + B_2z_2 + C_2z_3 &= D_2z_1; \\
B_3z_2 + C_3z_3 &= 0.
\end{align*}
\]

Here \( A_i, B_i, C_i, D_i \) are coefficients (complex numbers).

By solution of this system of equations it is possible to find the transfer functions in the form \( z_2/z_1 \) and \( \theta_2/z_1 \), where \( r_0 \) is a radius of pressure center of the gap seal. Equations for AFC and PFC are not shown because they are too large. In addition to it, the solution in computer program is found numerically by using of complex numbers, therefore relevance of obtaining of analytical expressions is significantly reduced.

3. Theoretical studies of face gas dynamic seal

Let we take for the analysis a face gas dynamic seal with spiral grooves for natural gas compressor NC-16M [10]. We will analyze both the main and backup stages, since they operate at different pressure drops with different values of gap. Figures 2-5 show the results of calculation of gap value changes during period of oscillations by the theory developed in this article and based on the bending deformation of seal graphite ring. The pressure drop at the first stage is 5.2 MPa, in the second stage it is 0.5 MPa. The oscillation frequency is 100 ... 200 Hz. The amplitudes of axial vibration of rotor face is 100 ... 300 \( \mu \)m, the amplitude of the angular oscillations is up to 1 mrad.

Analysis of Fig. 2 shows that the presence of axial rotor impact with amplitude 100 ... 300 mm and frequency 100 Hz has no strong influence on the work of face gas dynamic seal. However, when the oscillation frequency is 200 Hz (Fig. 3) at amplitude is 200 \( \mu \)m, size of the gap becomes close to 1 \( \mu \)m, for amplitude 300 \( \mu \)m it is 0.3 \( \mu \)m. Experience shows that in such gaps there is a possibility of touching of the sealing surfaces. Leakage increases 2 ... 3 times, it can lead them out allowed range.
Fig. 2. Changing of the size of the gap in the main stage of face gas dynamic seal (oscillation frequency is 100Hz):

- - - 0.1 = 0.1 mm;            0.2 mm;                      0.3 mm

Presence of both axial and angular rotor impact (Fig. 4) makes a situation significantly more complicate. At frequency 200 Hz and amplitude of angular rotor face oscillation 1 mrad a collision of surfaces will occur when oscillation amplitude of sealing surfaces of rotor face is more than 200 μm. The calculations show that at a frequency 100 Hz the size of the gap reduced to a value about 0.5 ... 1 μm.

A feature of the second stage of face gas dynamic seal is low pressure drop. In this case, there is no bending of graphite ring from the pressure drop and gas dynamic force in sealing gap prevails over force which closes a sealing seam. It leads to the work of stage with increased gap. Analysis of oscillations of face gas dynamic seal’s backup stage (Fig. 5) shows that, in spite of a lower level of stiffness of gas layer which depends on the operating pressure of gas, presence of increased gap allows to respond satisfactory on oscillations of rotor face with high amplitude and frequency.

Fig. 3. Changing of the size of the gap in the main stage of face gas dynamic seal (oscillation frequency is 200Hz):

- - - z0.1 = 0.1 mm; 0.2 mm; 0.3 mm
Thus, our theoretical studies of face gas dynamic seal’s dynamics show that seal is properly working under axial vibration amplitude of the rotor to 250 ... 300 µm with a frequency up to 100 Hz. Appearance of vibration frequency 200 Hz makes the face gas dynamic seal inoperative or requires a limiting of the amplitude up to 100 µm. Therefore, manufacturers of magnetic bearings must exclude the possibility of high frequency oscillations with increased amplitude of rotor vibration.

Available experience of exploitation of gas compressor units with magnetic bearings and face gas dynamic seal [10] showed that provision in exploitation of amplitudes of axial and radial rotor oscillations not more than 50 µm
with the permission of it short-terms increasing up to 100 ... 130 μm for transient operation modes does not lead to a violation of face gas dynamic seal work. It confirms qualitatively the accuracy of theoretical results obtained in this article.

It should also be noted that developed dynamic model of face seal was successfully used for design of high-efficiency seals for aircraft engine supports and high-speed turbo-machinery [11, 12].

4. Experimental studies of face gas dynamic seal

There is a series of experiments on dynamic test rig [13] needed to estimate a working ability of face gas dynamic seal at increased rotor oscillation amplitudes. A rotor sleeve on which the face gas dynamic seal is mounted is attached as cantilever to the shaft of the test equipment which is installed on two bearings in a case (Fig. 6). Radial displacement of rotor sleeve and its bias is technologically achievable. Axial and radial beat of rotor sleeve were selected as the determining parameters. On non-regular operation mode of the magnetic suspension the radial rotor oscillation takes place relative to stator with an amplitude from 0.1 mm to 0.3 mm. To simulate this process on test rig a necessary beat (0.1; 0.2; 0.3 mm) of the rotor sleeve relatively to the axis of rotation was used. Combination of radial and axial beat was obtained by skew of the rotor sleeve. Radial beat of shaft in process of assembling is controlled near the second stage of seal, the axial beat is measured on face surface of the rotor sleeve seal. There is an increasing of shaft oscillation amplitude when rotor is rotating due to action of dynamic force.

Scheme of measurement system is shown on Fig. 6. Measurement of shaft’s dynamic radial beat under the second stage of seal in horizontal and vertical directions was done by eddy-current sensors; it can be converted into a dynamic radial and face beat of the first stage of face gas dynamic seal due to the presence of tough kinematic connection. Measurement results were recorded and processed by the block PXI of company «National Instruments» [14].

![Fig. 6. Scheme of the measurement system for dynamic tests:](image)

1 – Analog converter PXI, 2 – driver,
3 and 4 – meters of gas leaks, 5 – vibration analyzer AM-012, 6 – speed sensor, 7 – monitor

In experimental researches a following results were obtained. Radial displacement of shaft with eccentricity e = 0.1, 0.2 and 0.3 mm (without shaft skewing) practically doesn’t have influence on face gas dynamic seal at a frequency 90 Hz. Air leakage in face gas dynamic seal is increasing in the presence of shaft skewing. In series of
experiments a limiting amplitudes of shaft oscillations were identified. After working of face gas dynamic seal with shaft skewing (shaft radial displacement was 0.3 mm, and face beat of shaft sleeve was 0.25 mm) on regular operating mode (shaft rotation frequency is 90 Hz) fragments of graphite wear on outer diameter of sealing ring were found. At restart of this operation mode there was an emergency stop from appearance of significant angular oscillations of face gas dynamic seal details. At the beginning air leakage already reached the limit value, then there were jamming of graphite ring from falling of rubber seal (secondary seal) into gap between the graphite ring and the sealing block and this rubber ring could not follow to oscillations of carbide ring.

5. Conclusions

Face gas dynamic seals are the main type of seal for rotor supports of compressor machines. The developed theory allows studying of dynamic characteristics of such seals. It’s very usable for designing and development of face hydrodynamic seals. Due to this a time and material resources for development of seals and accordingly compressors sharply reduces.

Studies of face gas dynamic seal dynamics show that it works satisfactory with axial or radial rotor vibration amplitude to 250 ... 300 mm with a frequency up to 100 Hz. Presence of vibration with a frequency 200 Hz, especially in presence both of axial and angular vibrations makes the face gas dynamic seal inoperable or requires the oscillation amplitude limits.

Results of the present research will be useful for both of researchers and engineers involved in designing of face gas dynamic seals.

References