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A Gas Lubricant Combined Support-sealing Node

S V Falaleev [1,*], H Nadjari[1] and A S Vinogradov[1]

[1]Aircraft engines design department, Samara national research university, Moscow highway 443086, Samara, Russia

[*] Corresponding author. Email address: sergey_falaleev@mail.ru

Abstract. The purpose of the research provided in this article is to develop a gas-dynamic device capable of performing the functions of support sealing, unloading devices for axial thrust bearings and damping of axial vibrations of the rotor. Some kinds of seals applied in supports of aircraft engines are known. A face gas-dynamic seal is one of the most effective and standard technology solution for compressors. As the basic element of the developed device, a face gas-dynamic seal with spiral grooves is considered. It also includes the fundamental mathematical calculation of such devices and the experimental research outcomes that form the basis of which such devices can be produced and adapted for use.

1. Introduction

With the development of aircraft gas turbine engines, there is a continuous increase in parameters of operating cycle. High-pressure differentials in turbo compressors and considerably high rotating speeds of rotors lead to the formation of over amplifications of forces on radial thrust bearings [1]. Traditionally labyrinth or graphite seals are used to seal the oil cavities of the rotor supports of gas turbine engines [2]. Usually unloading cavities are used to unload the radial thrust bearings, this is performed in the compressor or turbine, where air at high pressure is supplied or air it is dumped into the second contour of the engine. Both of these actions cause a decrease in the axial force applied on the engine rotor. Dry friction or hydrodynamic dampers are used to decrease the radial vibration of rotors [3], dampers are not provided for in axial vibrations of rotors. In some cases in aviation Gas turbine engines with high power parameters, it is impossible to eliminate excitation sources of axial vibration completely [4], therefore it is necessary to use axial vibrations dampers in rotor supports to decrease the vibration to an acceptable level. That is why is necessary to develop highly effective devices for unloading radial thrust bearings of aircraft gas turbine engines.

2. Principle operation of a face gas-dynamic seal

The main element of construction proposed in this paper is a face gas-dynamic seal. As mentioned above, application of such seals is influenced by its ability to self-regulate, which is to maintain the required amount of clearance when changes in external loadings occur.

A face Gas-dynamic seal consists of the rotating hard-alloy ring fixed on the shaft, and an axially-movable carbon-graphite ring placed in the body where preliminary contraction is done by springs. On the rotating ring, there is a pressure head site on which spiral grooves and the sealing ring corbel separating the high-pressure cavity from the cavity of low pressure are executed. Rubber mating rings are usually used as secondary seals. The gas-dynamic face seal works by the principle of equilibration of the gas static and gas-dynamic forces operating on the axially-movable and rotating rings. When at rest, the sealing rings are pressed together by the influence of gas static forces and forces of preliminary compression caused by springs. As the shaft gas comes to the narrowed spiral grooves and encounters resistance of sealing corbel, it compresses, forming zones of elevated pressure. Pressure in the face slot increases therefore the axially movable ring is removed from the rotating ring, forming the guaranteed clearance 0,002mm … 0,005mm. Thus, during operation, the surface seal rings do not contact with each other.

The increase in the loading force leads to a reduction of the clearance in the seal and to a simultaneous increase in the gas-dynamic pressure in the clearance and bearing capacity that leads to recovery of the initial clearance. The reverse process takes place as loading reduces.

There are exists various design options of gas-dynamic chambers [5, 6]. The "Spiral grooves" is most popular and well-studied design form of gas-dynamic chambers. For the first time, the theory of seals with spiral grooves has been studied in the works of E A Muijderman [7].

3. The proposed design of the combined "seals- unloading device damper" node

The additional axial gas-dynamic bearing which serves both for unloading of the rotor, and for damping axial vibrations(that represents axial gas-dynamic damper) will structurally differ from face gas-dynamic seal, in that the non-rotatable seal ring will not be movable in the axial direction, and will rest against the face surface of the housing.

The proposed design of the thrust bearing which will also function as a rotor support seal and damper of axial vibrations is shown on figure1.

The basic elements of the gas-dynamic device: The non-rotatable ring 1 (which can be made from graphite, or silicon carbide with covering) and the rotating ceramic ring 2 on which face surface spiral grooves for creation gas-dynamic pressure in face clearance are executed. The ring 2 is installed in the sleeve 3, sealing is provided by the rubber ring 4, torque transmission by the element 5. The ring 1 is mounted in the housing 6, sealing is provided with the rubber ring 7, torque transfer is provided by the milling 8. The device also provides an encapsulation of support together with the labyrinth seal 13 (that is acts as seals) and additional bearing working together with the ball bearing 9. Ball bearing is mounted in the housing sleeve through the radial damper 11 and the face elastic ring 12.

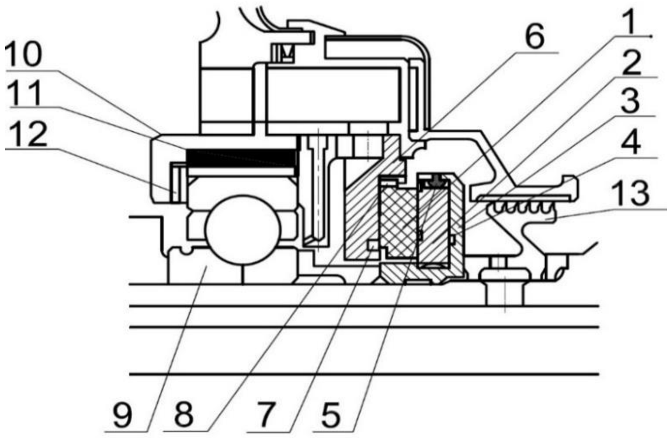

Figure 1. The design of a gas-dynamic device.

In statics, the clearance between rings 1 and 2 is established about 100 microns. In this case, the effect of spiral grooves is low and the device works as a normal slot-hole seal. As the engine begins to operate the axial force increases, the elastic ring 12 is deformed and the rotor is relatively gradually displaced to the left of the stator. The ring 12 should be designed in such a way, when an axial force of 3 tons is applied, the axial shift produced is 90…98 microns. In this case the axial clearance in the gas-dynamic device becomes equal 2 … 10 microns, spiral grooves begins to work more effectively, and the device unloads additional axial forces of rotor and provides an effective sealing for the supports, and damping of axial vibrations.

4. The fundamental theoretical design of gas-dynamic seals

The structural design of a face seal with spiral grooves is shown in figure2.

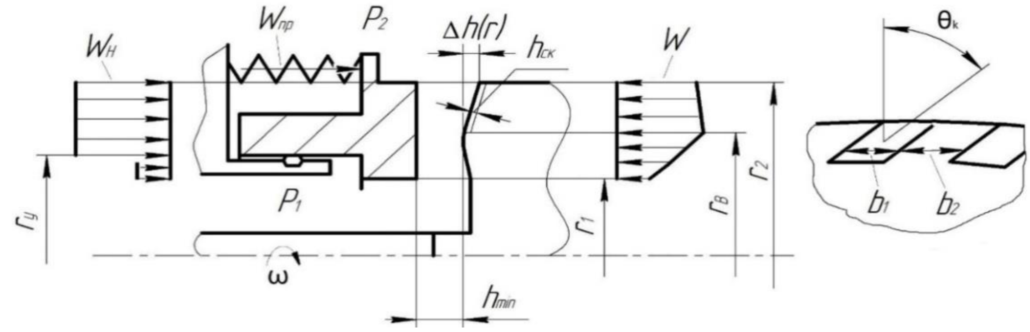

Figure2. The structural design of a face gas-dynamic seal with spiral grooves.

The load bearing capacity (applied on non-rotating ring seal) - W and the loading force - $W_и$ formed from the action of outer pressure - p_2 internal pressure - p_1 and spring forces- W_{np}. From the side of the non-rotatable ring of pressure p_1 and p_2 are also separated by the secondary seal which is on the radius r_y .The sealing clearance is formed by the external radius r_2 and internal radius r_1. The current value of the clearance $h(r)$ is the sum of minimum clearance and the clearance $h_{\min}$ which is formed as a result of deformation $\Delta h(r)\, h(r) = h_{\min} + \Delta h(r)$. Spiral grooves are carried out on the ring rotating with angular speed ω and characterized by following geometrical parameters: radius of the arrangement of spiral grooves $r_в$, depth of spiral grooves h_{ck}, discharge angle of spiral grooves θ_k, the width of spiral groove and crossing point width between grooves b_1.

Pressure on the border between zones of pressure and forcing - $\bar{p}_в$:

$$\bar{p}_в = \left[\frac{\bar{p}_1^2 \int\limits_{\bar{r}_в}^{\bar{r}_1} \frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3} + \int\limits_{\bar{r}_1}^{\bar{r}_в} \frac{d\bar{r}}{\bar{r}H^3} + \frac{2\lambda}{H^2} \int\limits_{\bar{r}_1}^{\bar{r}_в} \frac{d\bar{r}}{\bar{r}H^3} \cdot \int\limits_{\bar{r}_в}^{\bar{r}_2} \frac{B_0 \bar{r} d\bar{r}}{A_0\left(1-B(\bar{r}-1)\right)^2}}{\int\limits_{\bar{r}_в}^{\bar{r}_2} \frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3} \int\limits_{\bar{r}_1}^{\bar{r}_в} \frac{d\bar{r}}{\bar{r}H^3}} \right]^{\frac{1}{2}} \tag{1}$$

Pressure distribution in the leakage zones and pressurization is determined by formulas:

$$\bar{p}_l = \left[\bar{p}_1^2 + (\bar{p}_в^2 - \bar{p}_1^2) \frac{\int\limits_{\bar{r}1}^{\bar{r}} \frac{d\bar{r}}{\bar{r}H^3}}{\int\limits_{\bar{r}_1}^{\bar{r}_в} \frac{d\bar{r}}{\bar{r}H^3}} \right]^{\frac{1}{2}} \tag{2}$$

$$\bar{p}_н = \left[1 - \left(1 - \bar{p}_в^2\right) \frac{\int\limits_{\bar{r}_в}^{\bar{r}} \frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}}{\int\limits_{\bar{r}_в}^{\bar{r}_2} \frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}} + \frac{2\lambda}{H} \left(\int\limits_{\bar{r}_в}^{\bar{r}} \frac{B_0 \bar{r} d\bar{r}}{A_0\left(1-B(\bar{r}-1)\right)^2} - \frac{\int\limits_{\bar{r}_в}^{\bar{r}_2} \frac{B_0 \bar{r} d\bar{r}}{A_0\left(1-B(\bar{r}-1)\right)^2}}{\int\limits_{\bar{r}_в}^{\bar{r}_2} \frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}} \cdot \int\limits_{\bar{r}_в}^{\bar{r}} \frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3} \right) \right]^{\frac{1}{2}} \tag{3}$$

Here $\bar{r}$ - dimensionless radius $\bar{r} = r / r_1$; $\bar{p}$ - dimensionless pressure $\bar{p} = p / p_2$;

H - Dimensionless clearance $H = h_{\min} / h_{ск}$; λ - compressibility parameter $\lambda = \dfrac{6\mu\omega r^2}{P_2 h_{ск}^2}$;

A_0, B_0 - Constants of the spiral groove; B - conicity parameter $B = \dfrac{\Delta h(r)}{r_2 - r_1} \cdot \dfrac{r_{\min}}{h_{\min}}$.

In pressurization zone, between radiuses r_2 and r_1 of expression for A_0, B_0 are presented in equation [2]:

$$A_0 = \frac{H_1^3 + \bar{b}_k\left(1-\bar{b}_k\right)\left(H_1^3 - 1\right)^2 \cos^2\theta_k}{\bar{b}_k + H_1^3\left(1-\bar{b}_k\right)}, B_0 = \frac{\bar{b}_k\left(1-\bar{b}_k\right)\left(H_1^3 - 1\right)\left(H_1 - 1\right)\sin\theta_k \cos\theta_k}{\bar{b}_k + H_1^3\left(1-\bar{b}_k\right)} \quad (4)$$

Where H_1 - the parameter considering the amount of clearance on the current radius r $H_1 = 1 + \dfrac{h_{ck}}{h_{\min} + \Delta h(r)}$;

$\bar{b}_k$ - The dimensionless width of the spiral groove $\bar{b}_k = \dfrac{\bar{b}_1}{b_1 + b_2}$.

In leakage zone, between radiuses and the same equations will take the form:

$A_0 = 1 - B(\bar{r} - 1)$, $B_0 = 0$

Knowing pressure distribution in clearance, it is possible to define seals-hermiticity, and also axial thrust perceived by a friction pair.

The gas-dynamic force that opens the clearance is determined by the formula:

$$W = \bar{w} p_2 \pi\left(r_2^2 - r_1^2\right), \bar{w} = \frac{2}{\left(\bar{r}_2^2 - \bar{r}_1^2\right)}\left[\int_{\bar{r}_1}^{\bar{r}_в} \bar{p}_l \bar{r} d\bar{r} + \int_{\bar{r}_в}^{\bar{r}_2} \bar{p}_н \bar{r} d\bar{r}\right] \quad (5)$$

The research on the applicability of face gas-dynamic seals for unloading of the rotor from the excess axial force is based on the theory of 'face gas-dynamic seal'. The geometrical parameters of the face seals are shown in (Table 1).

Table 1. Some geometrical parameters of the friction pair of face gas-dynamic seal.

Parameter	r_1	r_2	r_y	r_b	Quantity of spiral grooves	$h_{ск}$
Value, мм	90	113	193	100	12	0,006…0,008

Figure3, shows the relationship between the extra bearing capacity change of the gas layer in the bearing clearance on the size of the clearance. Experimental values which were taken at the stand are indicated on the graph by x and triangular symbols [8, 9]. Equal to the depth of the spiral groove which a gas bearing can have. The extra load-bearing capacity - is the difference of current bearing capacity and minimum bearing capacity at the size of the clearance. The value of the load-bearing capacity was obtained through calculations taking into account the geometry of the sealing rings, the forces of the springs and the operating pressure, while the experimental clearance was calculated from the leakage of air measured through the end clearance during the experiment. Air differential pressure – 0.4 MPa, rotor rotational speed - 5500 and 10000 RPM.

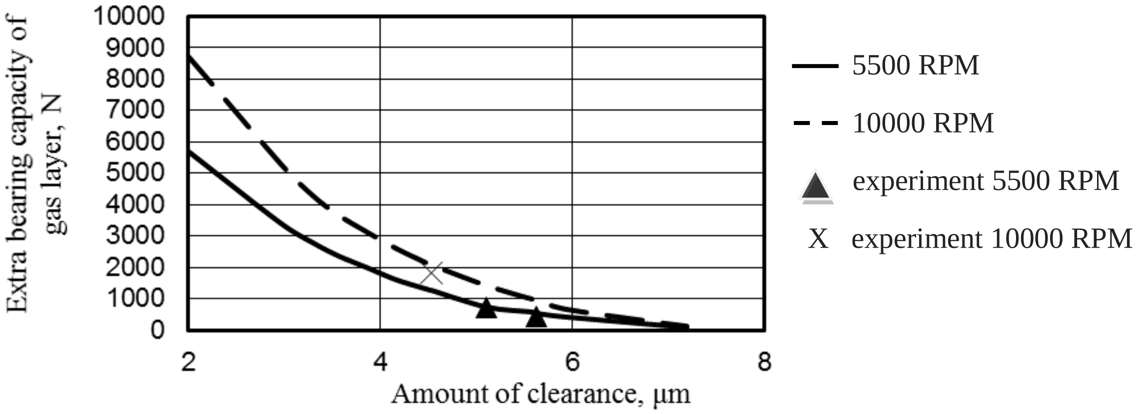

Figure 3. Design and experimental values of bearing capacity of the axial gas-dynamic bearing.

The analysis of 'figure3', shows that at a configuration of the friction pair, where the clearance is more than 2 microns (at smaller values of the clearance, contact of working surfaces and their wear is possible) it is possible "perception" to have the axial force operating on rotor reaching limits of up to 5800 … 8800N. In order for the bearing to carry the considerably more axial load, it is necessary to increase the area of the friction pair. Therefore, it's suggested to have the width of the friction pair increased from 22.5 mm to 40 mm. The analysis on the amount of clearance in the bearing from the size of the axial load was made at a differential pressure of air of 0.4 MPa and the rotational speed of rotor 10000 RPM. The results of the analysis have shown that at rotational speeds of 10000 RPM, and with an increase in width of the friction pair of up to 40 mm, the bearing is capable of withstanding an axial force equal to 3 tons. Thus, results from experimental analysis of axial gas-dynamic bearing have shown its prospective applicability.

5. Aviation gas turbine engine gas-dynamic axial vibrations rotor damper

Dynamic characteristics analysis of the layer of gas lubricant is used for assessment of the damping force. At the same time, it is possible to use theoretical researches on the face seals [10]. The dynamic reaction of the thin coat lubricant C_d (S), can simply be presented in the form of sums of rigidity and damping, which for incompressible environments are defined by viscosity and geometrical parameters of the node, and for the compressed environments depends also on vibratory frequency.

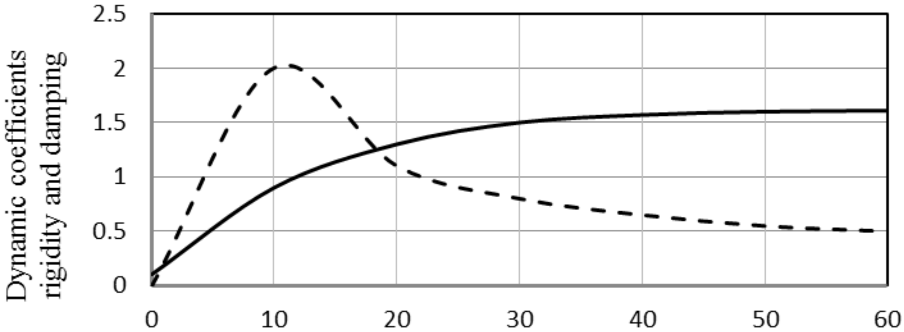

Figure 4. Dependence of dynamic coefficients rigidity $\overline{C}_y$ and damping $\overline{D}$ on compression parameter σ .

At the same time, axial movements damping is defined as.

For plane-parallel clearance: $$b_{zz} = \frac{3\pi\mu}{2h^3}\left[r_2^2 - r_1^2 - \frac{(r_2^2 - r_1^2)^2}{\ln(r_2/r_1)}\right] \tag{6}$$

This expression is applicable for incompressible environments and can be used as a first approximation for compression. More exact formulas used for the definition of damping gas layer need to be derived by use of special functions [10]. The dynamic reaction of the gas layer on the rotor is defined as the sums of dynamic coefficients of rigidity and damping $\bar{c}_d = c_d h_{\min} / (p_2 \pi (r_2^2 - r_1^2)) = \bar{c}_y + i\bar{D}$. Dependence of dynamic coefficients of rigidity $\bar{c}_y$ (represented by a solid line) and damping $\bar{D}$ (represented by the broken line) against the compression parameter σ for the specified node, $\sigma = 12\mu\omega(r_1 / h_{\min})^2 / p_2$ is presented on 'figure 4'. These results allow us to analyze dynamics of rotor systems.

6. Conclusions

The existing ways of unloading thrust bearings from axial forces in a gas turbine engines are associated with a reduction of their efficiency, or with the considerable complication of design. This article is a proposition for a new, simpler and more effective technical solution, that entails the use of a face gas-dynamic seal as an unloading device. Because seals of this type have the ability to self-regulate, the force-load on the unloading device will change depending on loading conditions, attaining an optimum value for each cycle of operation. The knowledge and experience in design provide for precise prediction of the force-load on the gas-dynamic device. The Discrepancy of experimental data to the theoretical design analysis does not exceed 5%.

A method of analysis of the unloading device characteristics is hereby proposed. These results allow us to conduct research on dynamics of rotor systems.

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