ARPN Journal of Engineering and Applied Sciences

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CONCEPT OF COMBINED GAS-DYNAMIC MECHANICAL SEAL AND DISCHARGE DEVICE OF AIRCRAFT ENGINE ROTOR SUPPORT

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ABSTRACT

The thrust bearing in aircraft engines and power plants takes in the axial force, which is transmitted through the power components on the engine attachment points to an aircraft. The magnitude of the axial force depends on the engine parameters. In modern engines with high values of thrust or power, the axial force magnitude will exceed the permissible value almost always. Therefore, the problem of thrust bearing unloading is an important scientific task. The discharge methods used in modern aircraft engines are associated with the extraction of the engine air flow, which reduces its effectiveness. The paper proposes the discharge device design for the thrust bearing of an engine rotor based on the gas-dynamic seal use. In this device, the seal will perform its basic function of oil chamber sealing and the additional function consisting in the thrust bearing discharge. The article also describes the discharge device construction and the methods of its operation are described. The gas-dynamic mechanical seal with spiral grooves is considered as the core element of the developed device. The paper presents the mathematical model of such a seal. Also, the results of the theoretical and experimental studies of the designed discharge device, confirming its performance.

Keywords: aircraft, thrust bearing, discharge device, gas-dynamic seal, spiral grooves, flexible annulus.

1. INTRODUCTION

One of the main trends in the aircraft engine development is a steady increase of the cycle parameters: the total pressure and gas temperature increase at the turbine inlet. In aircraft engines, power plants and other turbomachines the engine pressure may exceed the ambient pressure by more than forty times [1]. Large differences in pressure and the axial velocity change lead to the formation of the excess axial force [2], which is accepted by thrust bearing. To ensure the required level of uptime the amount of force accepted by bearing in aircraft engines is strictly limited [3]. This value makes 3 tons for the thrust bearings of the aircraft engines. However, due to the high pressures and speeds inside the engine this force may be greater than the maximum value. In this case, the use of additional measures for the thrust bearing discharge is an imminent one.

There are two basic ways to discharge the rotor thrust bearing supports. The axial force reduction acting on the thrust bearing may be accomplished using the compressor air cavity supercharge or by the breathing of the compressor cavities. Such methods are implemented in a variety of engines. For example, the cavity behind the last compressor wheel communicates with the second loop through the hollow pillars of the main combustion chamber to unload the front support of the gas generator rotor in the aircraft engines AL-31F [4]. The discharge cavity (Figure-1) in this case is formed by two labyrinth seals installed on the disk rim of the tenth stage and on the high pressure shaft. The compressor cavity connection (of high pressure chamber) with an outer contour (low pressure cavity) reduces the axial force value to an acceptable one. Thus, the second method of the axial force discharge is realized.

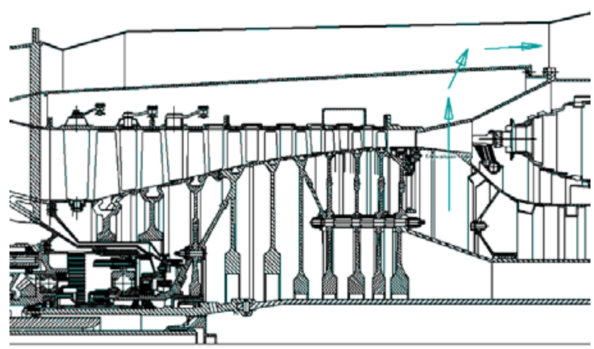

Figure-1. Discharge cavity in AL-31F engine.

Example of the first method use may be illustrated by D-36 aircraft engine design analysis [5]. In this case, the air comes through the holes in the back shaft cone into the low pressure compressor shaft cavities which passed the rear labyrinth seal of the low pressure compressor (Figure-2). After that, the air comes through the cone holes of the front shaft into the compressor cavities. Thus, by connecting the cavities with high and low pressure, the pressure in front and behind the compressor is leveled, which leads to the desired reduction of the axial force.

The disadvantage of these methods is that their implementation requires operational air extraction in turbomachines from the compressor air-gas channel. The air extraction the compression of which requires a certain amount of energy leads to the air flow reduction (or other working substance) to which the heat is supplied, and hence to the proportional decrease of the whole turbomachine efficiency.

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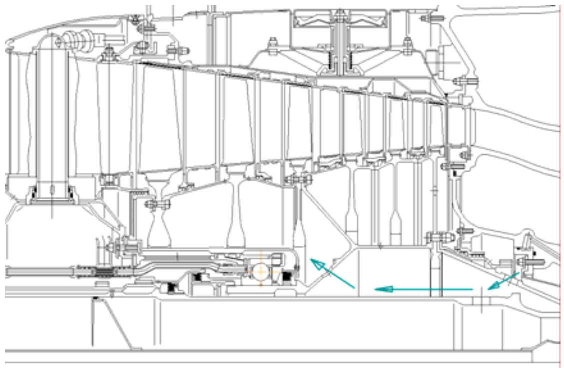

Figure-2. Discharge cavity in D-36 engine.

Such design decisions related to the air extraction from the flowing part, are also given in the patent of the largest engine design company SNECMA [6], presenting 14 constructive ways to connect the discharge cavity with the the outer contour of the turbojet. However, all these methods have the aforementioned disadvantage.

The traditional way of turbomachine rotor discharge is the use of the discharge disc, on which the axial force appears directed in the opposite direction. The example of such a solution for a centrifugal pump is disclosed in the patent [7], wherein the discharge disk and the face seal are mounted for the discharge device operation with the centrifugal pump impeller. The examples of such solutions also exist in gas turbine engines. This method has the obvious drawback: the use of additional disk increases the structure size and weight, which is especially important, if we solve the problem of the axial force reduction for the aircraft engine.

Thus, the thrust bearing discharge problem solution for an aircraft engine without the air extraction from the air-gas channel and without the significant size increase is a promising one.

2. DISCHARGE DEVICE DESIGN AND OPERATION

The developed design of the discharged thrust bearing of the turbomachine rotor (Figure-3) contains a flexible annulus, the main dimensions of which are selected according to the operating forces. This rotor design also contains the gas-dynamic apparatus which can be presented as the gas-dynamic mechanical seal. This design does not need to use additional working substance in excess of that amount, which is inevitably passes through the seal.

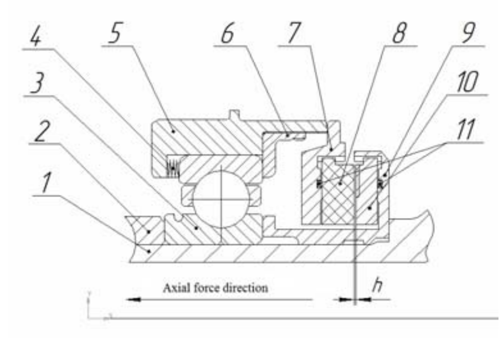

Figure-3. Developed discharge device scheme.

The flexible annulus is mounted between the end of the bearing and the housing (bearing glass). It has mutually offset projections at both ends (Figure-4), which abuts on one side into the end face of the outer bearing ring, set on a shaft with which it the axial force is passed, and, on the other side it abuts in a glass retaining the external bearing ring.

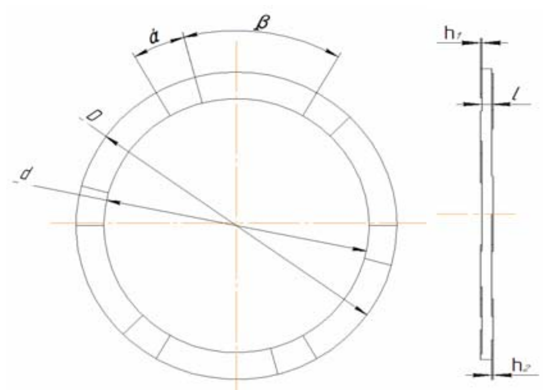

Figure-4. Thrust ring design.

The ring dimensions (D and d diameters), the ring thickness l, the dimensions h1 and h2 of projections and their number determine the ring stiffness and should be determined by calculation to provide the desired value of the elastic ring deformation under the impact of the setp surplus axial force.

The gas-dynamic seal consists of a fixed and a rotating ring. The rotating ring has gas dynamic chambers which may have different configurations. The distinctive feature of this seal is its ability for self-regulation [8]. The required value of the sealing gap is ensured by the equal forces on the non-rotating ring. Each pressure value of the sealed environment will correspond to a definite value of the sealing gap.

For the operation of the discharge device in the direction of axial force impact the following elements shall be installed sequentially: a gas dynamic seal, a thrust bearing and an elastic ring. During the ring assembly the seal with a larger gap h is set. Under the action of the axial force the flexible ring is compressed, which leads into the seal gap reduction to the calculated value. In end gas-dynamic seal the gas force is generated at the axial gap of 2... 10 mcm (with a smaller gap the sealing surface touch is possible). Upon reaching the gap rated value the seal receives the part of the axial force acting on the bearing. The mounting gap value h is determined by the elastic ring deformation under the axial force impact. In its turn, the excessive value of the axial force, perceived by the discharging device, is determined by the sealing ring area, by the magnitude of the permissible minimum gap in the discharge device and by the gas-dynamic chamber geometry.

The use of gas-dynamic seal in the discharge device is equivalent to the use of an additional thrust bearing. This design may be considered as quite promising, if you take into account one of the gas turbine equipment development, based on the magnetic bearing use. In the article [9] C. DellaCorte and R.J. Bruckner offer the turbomachine rotor variant shown in Figure-5. In this case there is the combination of magnetic bearings accepting the axial and radial load from the compressor and the turbine. If we consider such a structure the bearing and seal functions may be combined in a single node [10].

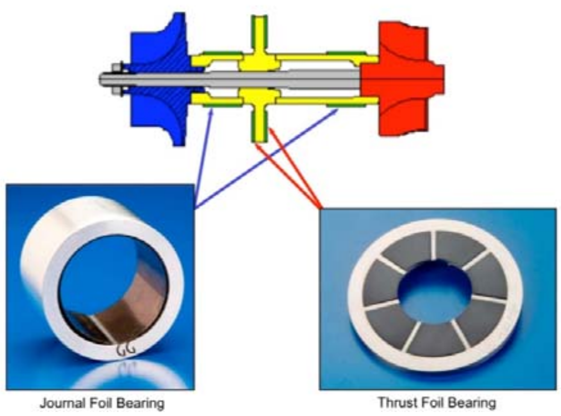

Figure-5. Turbomachine promising rotor scheme with magnite bases.

3. THEORETIC BASIS OF GAS DYNAMIC SEAL DESIGN

The main element of the proposed design is a gas-dynamic mechanical seal. As noted above, the use of such seals is conditioned by its ability to self-regulation, that is required to maintain the the necessary gap value at a certain change of the external loads [8, 11].

The principle of such a seal operation is shown by Figure-6. The loading force increase first of all leads to a seal gap reduction and then to the simultaneous gap pressure increase, which in its turn leads to the pressure restoration up to its initial value. The similar process will take place when the load is decreased.

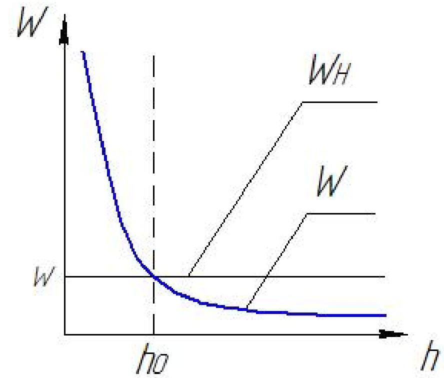

Figure-6. End gas dynamic seal operation principle.

There are many versions of gas-dynamic cameras implementation [8]. They may be executed as with the rotating ring so as with two rings which make the friction pair. The most well-studied and common form is the form of gas-dynamic cameras, named as the "spiral grooves." The seal theory with spiral grooves was presented first of all by Muijderman E.A. [12]. There are also fundamental researches of such seals in the Russian scientific literature [8, 13-16].

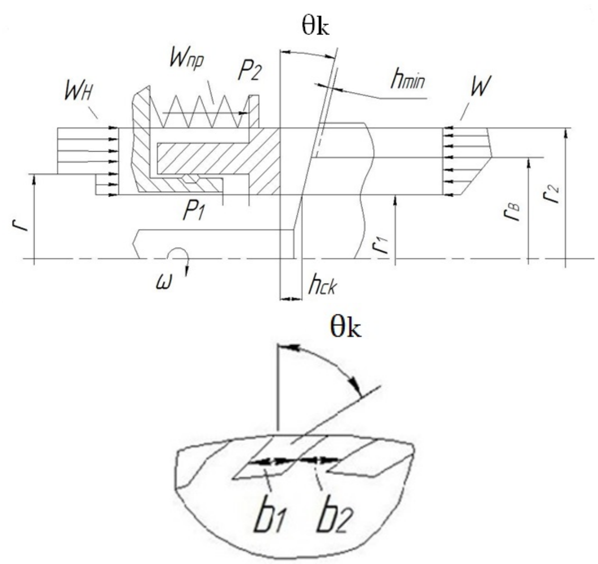

Figure-7. Calculation scheme of the end gas dynamic seal with spiral grooves.

The gas dynamic seals are used in aircraft engines and power plants [17]. The designing of such seals shall take into account not only the processes occurring in the sealing gap, but also the seal operation as the part of the engine systems or its components [18-22]. The seal design scheme with spiral grooves is shown by Figure-7. The Reynolds equation for such a seal may be written as follows [9]:

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$$\frac{d}{d\bar{r}}\left(\bar{r}\cdot A_0\frac{d\bar{p}^2}{d\bar{r}}\right)=2\frac{\lambda}{H^2}\bar{r}B_0, \quad (1)$$

where $\bar{r}$ = non-dimensional radius $\bar{r}=r/r_1$;
$\bar{p}$ = non-dimensional pressure $\bar{p}=p/p_2$;
H = non-dimensional gap $H=h_{\min}/h_{ск}$;
λ = compaction parameter $\lambda=\frac{6\mu\omega r^2}{P_2h_{ск}^2}$;
A_0, B_0 = Whipple spiral groove constants [7, 9].

The constant grooves equations in the pumping area between r_2 and $r_в$ radii will be the following [10, 12]:

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

$$B_0=\frac{\bar{b}_k\left(1-\bar{b}_k\right)\left(H_1^3-1\right)\left(H_1-1\right)\sin\theta_k\cos\theta_k}{\bar{b}_k+H_1^3\left(1-\bar{b}_k\right)}.$$

The same facilities will take the following equation in the leakage area betweenr$_1$ and $r_в$, radii:

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

$$B_0=0,$$

where B is the taper parameter:

$$B=tg\theta\frac{r_{\min}}{h_{\min}}.$$

The equation (1) may be solved by taking into account the extreme values $\bar{p}=1$ at $\bar{r}=r_2$, $\bar{p}=p_1$ at $\bar{r}=\bar{r}_1$. When crossing the boundary between the discharge and leakage zones the pressure and gas flow shall be continuous ones. As a result, we obtain the mechanical seal charecteristics with spiral grooves in a dimensionless form.

The gas-dynamic force, revealing the gap is determined by the following formula:

$$W=\bar{w}p_2\pi\left(r_2^2-r_1^2\right),$$

where $\bar{w}$ is dimensionless gas dynamic form, determined according to the following formula:

$$\bar{w}=\frac{2}{\left(\bar{r}_2^2-\bar{r}_1^2\right)}\left[\int_{\bar{r}_1}^{\bar{r}_в}\bar{p}_{ут}\bar{r}d\bar{r}+\int_{\bar{r}_в}^{\bar{r}_2}\bar{p}_н\bar{r}d\bar{r}\right]. \quad (2)$$

In formula (2) $\bar{p}_{ут}$ is the leakage zone pressure, and $\bar{p}_н$ is the pump zone pressure.

The border pressure between the pressure and pump zones:

$$\bar{p}_в=\left[\frac{\bar{p}_1\int_{\bar{r}_в}^{\bar{r}_1}\frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}+\int_{\bar{r}_1}^{\bar{r}_в}\frac{d\bar{r}}{\bar{r}H^3}}{\int_{\bar{r}_в}^{\bar{r}_2}\frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}\int_{\bar{r}_1}^{\bar{r}_в}\frac{d\bar{r}}{\bar{r}H^3}}+\right.$$

$$\left.+\frac{\frac{2\lambda}{H^2}\int_{\bar{r}_1}^{\bar{r}_в}\frac{d\bar{r}}{\bar{r}H^3}\cdot\int_{\bar{r}_в}^{\bar{r}_2}\frac{B_0\bar{r}d\bar{r}}{A_0\left(1-B(\bar{r}-1)\right)^2}}{\int_{\bar{r}_в}^{\bar{r}_2}\frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}\int_{\bar{r}_1}^{\bar{r}_в}\frac{d\bar{r}}{\bar{r}H^3}}\right]^{\frac{1}{2}}. \quad (3)$$

Pressure distribution in the leakage and pumping area is determined according to the following formulae:

$$\bar{p}_{ут}=\left[\bar{p}_1^2+\left(\bar{p}_в^2-\bar{p}_1^2\right)\frac{\int_{\bar{r}_1}^{\bar{r}}\frac{d\bar{r}}{\bar{r}H^3}}{\int_{\bar{r}_1}^{\bar{r}_в}\frac{d\bar{r}}{\bar{r}H^3}}\right]^{\frac{1}{2}}. \quad (4)$$

$$\bar{p}_н=\left[1-\left(1-\bar{p}_в^2\right)\frac{\int_{\bar{r}_в}^{\bar{r}}\frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}}{\int_{\bar{r}_в}^{\bar{r}_2}\frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}}+\frac{2\lambda}{H}\times\right.$$

$$\times\left(\int_{\bar{r}_в}^{\bar{r}}\frac{B_0\bar{r}d\bar{r}}{A_0\left(1-B(\bar{r}-1)\right)^2}-\frac{\int_{\bar{r}_в}^{\bar{r}_2}\frac{B_0\bar{r}d\bar{r}}{A_0\left(1-B(\bar{r}-1)\right)^2}}{\int_{\bar{r}_в}^{\bar{r}_2}\frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}}\times\right.$$

$$\left.\left.\times\int_{\bar{r}_в}^{\bar{r}}\frac{d\bar{r}}{\bar{r}A_0\left(1-B(\bar{r}-1)\right)^3}\right)\right]^{\frac{1}{2}}. \quad (5)$$

Knowing the pressure distribution in the gap, which is determined according to the relations (3, 4, 5) one may determine the seal tightness and the axial force accepted by a friction pair [23]. The seals with spiral grooves may be used as a sealing medium for gas, but may be operable at the sealing of liquids or biphase media [24, 25]. If the value of the axial force acting on the thrust bearing is known, it is possible to design the discharge apparatus for a turbomachine rotor. This design should

VOL. 9, NO. 10, OCTOBER 2014 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

include two main stages: the selection of the elastic ring geometric parameters and the friction pair rings.

4. ESTIMATED AND EXPERIMENTAL RESULTS OF EXPLORED DISCHARGE DEVICE

The main parameters of the elastic ring (Figure-4), which shall be selected during its design are the thickness l and the number of protrusions. The remaining parameters of the ring are determined the bearing assembly dimensions.

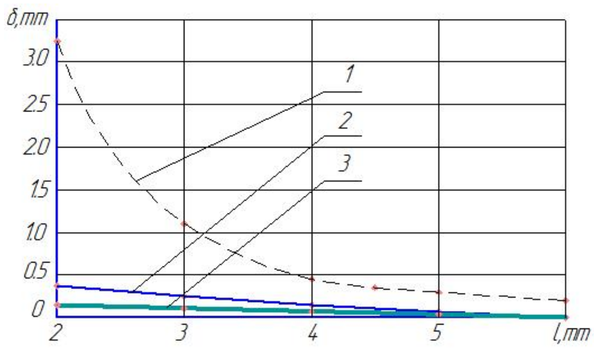

Figure-8. The dependence of elastic ring deformation on its thickness at different numbers of projections: 1 - 3 projections, 2 - 6 projections, 3 - 9 projections.

According to the design of the discharge device, the elastic ring shall be deformed by the valueof an enlarged sealing gap in the seal. For the considered support design of the aircraft engine and the operating conditions of loading the required value of the elastic ring axial deformation shall be equal to 0.1. mm.

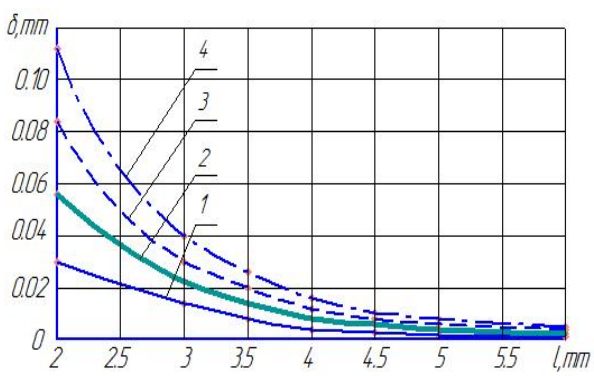

Figure-9. Elastic ring deformation dependence on its thickness at different axial load: 1 - 1 t, 2 - 2 t, 3 - 3 t, 4 - 4 t.

This deformation value shall correspond to the axial force, which was equal to 3 t. The analysis of Figure-8, Figure-9, Figure-10 allows to select the ring thickness and the number of protrusions.

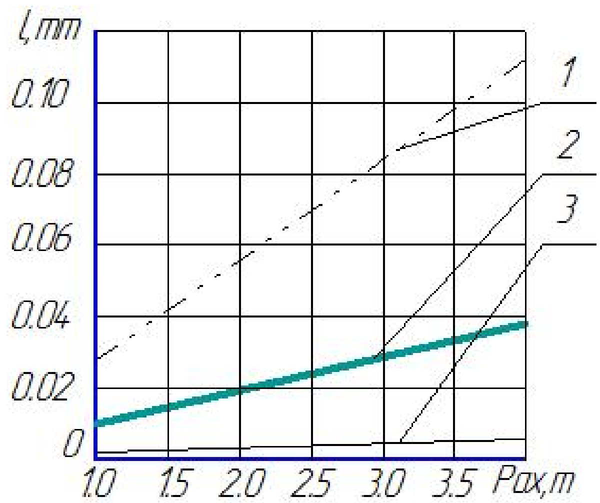

Figure-10. Elastic ring deformation dependence on axial load at different thickness: 1 - 2 mm, 2 - 3 mm, 3 - 6 mm.

For these conditions, these values made 4.5 mm and 6, respectively. The deformation value was determined by the calculation within ANSYS software with finite elements.

Another important issue is the choice of the gas-dynamic compaction dimensions. As compared with the conventional bearing seal, which solves only the problems of tightness the discharge device seal shall accept the part of the axial load. The calculations show that the end face area shall be increased to 3 ... 4 times for the given conditions.

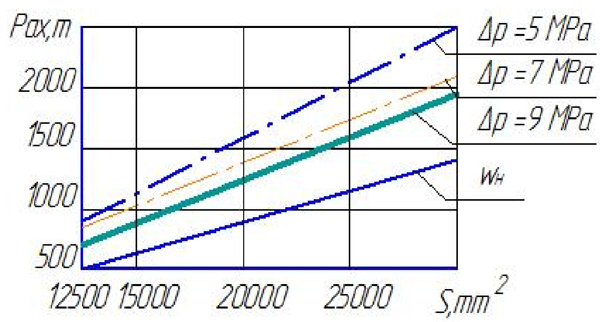

Figure-11. The dependence of the end face sealing area on the axial force acting at various differential pressures.

The obtained calculated dependence of the end face sealing area on the perceived axial force for the given geometry is shown by Figure-11. The choice of seal dimensions are especially important for the gas turbine rotor support unit conditions due to the serious shortage of space. However, despite the enlarged seal size, the use of the proposed discharge device gives great advantages compared with the known methods.

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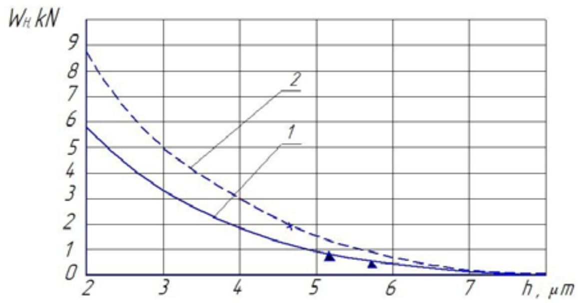

Figure-12. Estimated and experimental values of the axial gas dynamic bearing carrying capacity: 1 - 5500 rev/min, 2 - 10000 rev/min.

The existing theory of seals with spiral grooves and the software developed on its basis allowed to perform the comparison of the calculated values for the bearing capacity with the experimental data. Figure-12 shows that a good agreement was achieved. The error of the carrying capacity determination does not exceed 5%. This suggests to state that the proposed design of the discharge device may be designed for other conditions with the required level of accuracy.

5. CONCLUSIONS

The existing methods for the thrust bearing discharge from the axial force in turbomachines, and, particularly in gas turbine engines are associated either with its efficiency decrease or with the significant complication of the structure. Therefore, a simple and effective discharge device development may be considered as the actual technical problem.

The proposed design of the discharge device consists of an elastic ring and a gas-dynamic device. The gas-dynamic seal is reasonable to use for this purpose. Since the seal is an obligatory element of the bearing assembly, the entire structure becomes more complicated due to the introduction of only one additional detail: an elastic ring. One may use the seal with any form of gas-dynamic cameras as a gas-dynamic seal. For example, the most common seal is the seal with spiral grooves.

The design of this discharge device is reduced to the choice of the elastic ring geometrical parameters (which is largely similar to the spring selection process) and the size of the friction pair for the gas-dynamic seal. The existing design experience allows predicting the effort perceived by a gasdynamic device accurately. The discrepancy between the experimental data and the design calculation did not exceed 5%. At that the bearing assembly dimensions will be increased slightly.

ACKNOWLEDGEMENTS

This work was supported by the Ministry of education and science of the Russian Federation in the framework of the implementation of the Program of increasing the competitiveness of SSAU among the world's leading scientific and educational centers for 2013-2020 years.

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