The influence of nonuniform heat transfer coefficient distribution on the value of thermal deformations in mechanical gas dynamic seal rings

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Abstract. Currently, many different aircraft engine types exist. They have a wide range of operating conditions: the flight speed can vary from zero to hypersonic values, the flight altitude from near-earth conditions to space flights. During the flight, loads are not kept constant. Engines have significant dynamic overloads, temperature and pressure differences, can operate in conditions of high humidity or dust, as well as in various aggressive environments. The development of aviation and rocket engine building is associated with a continuous increase in the working thermodynamical cycle parameters.

Further improvement of gas turbine engines (GTE) can be carried out in two directions. The first is using new advanced engine designs based on a modernized cycle, with a different fuel, original design, and power scheme. This direction also includes the application of new high-performance elements in the engine: bearings, seals, coatings, etc. The second direction is related to the improvement of existing structures, increasing their efficiency by more detailed working conditions forecasting based on the use of modern computing software systems. This article presents a study of the potential mechanical face gas dynamic seal (FGDS) in the support in an aircraft engine compressor. Aircraft engines are characterized by multi-mode operation and high temperature levels in the internal cavities. For a FGDS operating with a working gap of several microns the exact determination of the deformation value is a very important result of the design process. This article considers the influence of the convective heat transfer coefficients distribution on the value of sealing rings thermal deformations. To achieve this goal, two compaction models were created: the gas-dynamic model for determining the convective heat transfer coefficients in the ANSYS CFX software program; and the structural model for calculating the resulting deformations. Interaction between the models was performed based on a specially created methodology. The obtained results can be used in the design of a FGDS for the supports in the compressors of aircraft engines and power plants.

1. Introduction

Traditional types of seals for aircraft engines (for engines of the first four generations) consist of two large groups with contact and non-contact seals. In aforementioned group the labyrinth or gap seals were mostly included. Contact and contactless seals have "mirror" advantages and disadvantages. If
labyrinth seals have a low tightness and unlimited resource, then contact seals have a strictly limited resource and high tightness. In this regard mechanical contactless seals occupy a certain intermediate position. There are face gas static and face gas dynamic seals, or FGDS.

In a FGDS the sealing rings are separated by a guaranteed layer of gas. High pressure in the sealing gap is created by using special gas dynamic grooves. In addition, the gas layer in the gap must have the necessary stiffness to prevent ring contact in case of accidental perturbations or minor axial vibrations. Figure 1 shows the working scheme of a FGTS in an engine support with gas-dynamic special grooves. These grooves were produced in the form of spirals. The seal consists of a rotating ring 2 with spiral grooves 1 that were made on it. The rotating ring is mounted on the shaft 4. The axially movable ring 3 is installed in the housing 5. Between the housing 5 and the ring 3 the secondary seal 7 is used and the elastic element 6. A guaranteed gap is formed only under condition of two components equality. There are loading force-$W_0$ acting from the outside of the ring 3 and the bearing capacity-$W$, formed from the pressure distribution in the sealing gap.

![Figure 1. Face gas dynamic seal scheme with spiral grooves](image)

![Figure 2. FGDS working principle](image)

The working principle of a FGDS is shown in figure 2. The advantage of this seal type is its ability to self-regulate, i.e. to maintain the gap with minor changes in pressure or displacements of rings 2 and 3 (figure 1).

FGDS are mainly used in ground and industrial power plants. But there are examples of the FGDS modification application (radial face seal with gas dynamic grooves and carbon ring) in aircraft engines. A FGDS in the support of a transport aircraft engine is shown in figure 3.

![Figure 3. A radial face gas dynamic seal in transport aircraft engine support](image)
Constantly increasing requirements for reliability, tightness and weight of seals have led to increased interest in developing mechanical seals with gas and liquid lubrication or mechanical contactless seals [1]. They are characterized by a guaranteed gap between the sealing surfaces. Therefore, the seals work with constant leakage, but almost without wear. The gap in a FGDS is equal to several microns and consequently a high level of tightness is achieved. The leakage level is close to the leakage through the contact seals. A FDGU theoretically does not have circumferential speed limits, and the compacted pressure level is limited only by permissible leaks and force deformations. The power of friction at the sealing rings in a FGDS is approximately ten times less than the power of friction in tradition contact seals. However, the use of such a seal must be based on careful design calculations. A slight change in deformations can lead to a change in the characteristics of the seal and even to its destruction. Changes in deformations can occur as a result of various static and dynamic factors. This paper focuses on analyzing the gap changes resulting from changes in the convective heat transfer coefficients.

To get more accurate results of solutions, it is necessary to set the thermal boundary conditions that are close to the real ones. In order to determine them, all heat sources that exist in the sealing unit should be considered. The heat sources in the compressor supports are shown in figure 4.

Figure 4. Heat sources in a FGDS unit for an aircraft engine compressor support

There are numerous highly important international publications dedicated to examining FGDS. They include, for example, the works of L. P. Ludwig [2], A. O. Lebeck [3], I. Zuk [4], R. Salant [5], H. K. Mueller [6], I. Etsion and I. Green [7, 8]. The author of the classical theory for gas dynamic lubrication in an FGDS is E. A. Mujdermann [9]. I. Green's investigations about the continuous improvement of the dynamic model for a FGDS is still ongoing [10]. Most articles of the rest researchers are devoting to the study the performance of various FGDS structures and grooves different operating conditions. In the countries of the former USSR, the mathematical modeling of a FGDS was performed in the works of S.V. Pinegin, A.V. Emelyanov, Yu.B. Tabachnikov [11], Yu. V. Peshti [12], Yu.Ya. Boldyrev, B.S. Grigoriev, G.A. Luchin [13], A.I. Golubev [14], G.N. Den [15], V.A. Marcinkovsky [16]. Extensive applied investigations of seal operation and calculation features are contained in the works of B.M. Gromyko, A.E. Chernov [17], V.A. Melnik [18], E.A. Novikov [19], E.P. Krevsun [20]. Improved mathematical models together with the solution of applied problems for the maintenance and further development of their own designs were performed by V.A. Maximov [21], V.A. Marcinkovsky [16], S.V. Falaleev [1]. The analysis of ring thermal deformations was conducted in the works of other researchers without taking into account the nonuniform distribution of convective heat transfer coefficients and its changes under different aircraft engine modes [1, 14, 17, 18, 19, 21 etc.].
2. Calculation seal models and examination method

An analysis of existing publications confirms that a study on the thermal state of FGDS in the compressor support of an aircraft engine with the above-listed features is relevant. The proposed design of such a seal in the aircraft engine compressor support designed by the company PSC “Kuznetsov” is shown in figure 5.

![Figure 5. FGDS installed in compressor support](image)

It has already been noted that the existing studies of thermal deformations in mechanical face seals were performed under the assumption of a constant (or uniform) convective heat transfer coefficients distribution between the boundaries of the considered surface sealing ring area [1, 14, 17, 18, 19, 21]. In this case, the influence of uniform distribution for coefficients along the sealing ring perimeter on the shape and size of the sealing gap was studied. To implement this research project a calculation method was developed. A scheme of this method is shown in figure 6. The seal model with bordering cavities was created in the ANSYS CFX software program. The calculation method includes the following steps: creating the model and boundary conditions, checking the meshing quality by $y^+$ parameter, performing the first hydraulic calculation with adiabatic walls and determining the total temperature near the walls $-T^\text{adiabatic}$, setting the wall temperature $-T^*$, re-conducting the hydraulic calculation and determining the value of heat flows-Q for each surface under study, calculating the convective heat transfer coefficients and temperature distribution. The coefficient value is calculated using the formula:

$$\alpha_i = \frac{Q}{T^* - T^\text{adiabatic}}. \tag{1}$$

The developed method (figure 6) was tested for a FGDS with spiral grooves installed in the support of an aircraft engine compressor. The choice of such a seal for the study was made for the following reasons. Firstly, mechanical gas dynamic seals can potentially be considered for use in the compressor support. Secondly, the existing mathematical models and improved calculation methods allow us to accurately determine the pressure distribution in the sealing gap. This is required for the accurate prediction of the deformations level and value. Simulation of the compaction pressure distribution was performed based on a mathematical model developed by E.A. Mujdermann [9]. For traditional mechanical contact seals (face seals, radial face seals, radial moving seals) pressure distribution accurate mathematical models do not exist.
The hydraulic model of the sealing unit cavities is shown in figure 7. High-pressure air from the cavity in front of the compressor passes through the labyrinth seal into the cavity in front of the gas dynamic seal (figure 7a). The cavity between the seals is connected to the cavity behind the tenth stage of the high-pressure compressor to provide a calculated differential ratio on the mechanical seal. Research was conducted for four flight modes. Figure 7b shows the boundary conditions corresponding to the parameters of the first mode. Data for other flight modes is shown in table 1.

As a result of the calculation performed using the developed method, the average values of the convective heat transfer coefficients and their distribution in the open areas of the friction pair rings were obtained. Figure 8 shows the results of calculating the distribution of convective heat transfer coefficients corresponding to the first mode. Analysis of this figure allows us to conclude that averaging the values of \( \alpha_i \) leads to significant errors. For some walls, this is especially pronounced. For example, for wall No.2, the average AI value is 287 W/m\(^2\)K, while the maximum and minimum values are 609 W/m\(^2\)K and 201 W/m\(^2\)K, respectively. In the first case, the approximation error is 112%, in the second-30%. In addition, the distribution of \( \alpha_i \) along wall No.2 is nonlinear.
Table 1. Boundary conditions for the examined modes

| Flight modes | Inlet 1 | Outlet 1 | Inlet 2 | Outlet 2 | Rotation |
|--------------|---------|----------|---------|----------|----------|
|              | $P_{\text{in1}}, \text{kPa}$ | $P_{\text{out1}}, \text{kPa}$ | $G, \text{g/s}$ | $P_{\text{out2}}, \text{kPa}$ | $n, \text{rpm.}$ |
| No. 1        | 225,6   | 150      | 40,5    | 147,2    | 7055     |
| No. 2        | 285,3   | 150      | 43,5    | 196,1    | 8505     |
| No. 3        | 390,5   | 150      | 51,5    | 274,6    | 9280     |
| No. 4        | 495,3   | 150      | 86,5    | 294,2    | 9612     |

Figure 9 shows the change in the average values of $\alpha_i$ for engine modes (from the speed of rotation). It is obvious that the values of the convective heat transfer coefficients will increase with increasing speed. It is important to emphasize the extent to which these values change. When the speed increases by 36% from 7055 rpm to 9612 rpm, the average $\alpha_i$ value for wall No.2 increases from 287 W/m²K to 804 W/m²K. In relative terms, the increase is 180% or 2.8 times. The obtained results indicate that it is necessary to take into account the changes in the coefficients $\alpha_i$ for modes and their distribution along the walls.

![Figure 8](image1.png)

Figure 8. Average and variable values of the convective heat transfer coefficient $\alpha_i$ along the perimeter of the sealing rings

![Figure 9](image2.png)

Figure 9. Changing the coefficient of convective heat transfer $\alpha_i$ for flight modes

When designing a seal, it is particularly important to determine its effect on the deformation of the sealing rings and the associated tightness. The resulting deformation was studied using a finite element
model created in the ANSYS Mechanical APDL software package. For the flight mode No.1, two thermal calculations were made: with a uniform ($\alpha=$const) and uneven ($\alpha=$var) distribution of the convective heat transfer coefficient, which resulted in a field of temperature distribution in the sealing rings. Then, using the results of thermal calculations, structural calculations were performed. Full schemes of boundary conditions for thermal and structural calculations for both cases are shown in figure 10. Fixing the non-rotating ring in both cases is performed in the same node, so as not to interfere with the rotation of the section. The heat flow in the gap is set using special elements that form a thermal bridge between the rings.

3. The results of calculations of FGDS deformations in the support of a compressor and their analysis

As a result of calculations based on the models shown in figure 10, the temperature distribution for the four studied modes was obtained. The temperature distribution for the first mode for the cases is given in figure 11. Comparison of the results shows that both the value of the average temperature of the seal ring and the nature of its distribution change simultaneously. The temperature difference does not actually change. For a graphite ring, the temperature changes from 549.0 to 550.1 K in the case, while in the case the temperature changes from 555.1 to 555.8 K. In both cases, the difference is about 1K.

At the same time, in the first case, the average temperature of the ring is 549.6 K, and in the second 554.4 K. The difference in the average temperature for the absolute value is almost 6K. In the case of $\alpha=$const, the most heated area is located in the middle of the sealing face. With a variable distribution of $\alpha=$var, the heated section is shifted to the outlet of the sealing gap. These factors lead to changes in the size of the deformations of the sealing rings.

The calculated values of the deformation of the sealing rings for the first studied mode at constant and variable values of the convective heat transfer coefficients are shown in figure 12. At $\alpha=$const, the taper value was 1.17 $\mu$m, at $\alpha=$var-2.38 $\mu$m, increasing by 100%. As was shown in the second Chapter, this change in taper can significantly change the characteristics of the mechanical gas dynamic seal. A twofold increase in thermal deformation will affect the wear processes in end contact seals of traditional types.
Changing the taper by mode (depending on the speed of rotation) is shown in figure 13. At constant values of AI, the taper will change from 1.17 μm to 2.15 μm (by 84%) in the studied speed range. As already noted, for α=var, the absolute values of the strain at each point will be greater. When switching from the minimum to the maximum mode, the taper changed by about the same amount (by 79%) from 2.38 μm to 4.25 μm.

![Figure 11. Temperature distributions in friction pair rings for flight mode No.1 conditions with constant and variable values of convective heat transfer coefficients](image)

![Figure 12 Calculated values of deformation of the sealing rings for the conditions of flight mode No.1 at constant and variable values of convective heat transfer coefficients](image)

![Figure 13. The dependence of the taper of the graphite ring to the rotational speed of the engine](image)
4. Conclusions
As a result of the performed computational study, it was found that taking into account the distribution of coefficients of convective heat transfer over the surface of the sealing rings allows adjusting the calculated value of thermal deformation several times. I.e., when designing a sealing unit in the support of an aircraft engine, it is necessary to achieve the required distribution of these coefficients. This study allowed us to formulate the following conclusions.
1. The distribution of the coefficient \( \alpha \) for the model under study has a pronounced uneven character, and when averaging it, too large an approximation error will occur. Thus, for one of the studied surfaces, the average value is \( \alpha_{\text{ave}}=609 \ \text{W/m}^2\text{K} \), while the maximum and minimum values are \( \alpha_{\text{max}}=609 \ \text{W/m}^2\text{K} \) and \( \alpha_{\text{min}}=201 \ \text{W/m}^2\text{K} \). Also, the coefficient of convective heat transfer \( \alpha \) during the transition from cruising mode varies greatly when switching from one mode to another. For the same surface, it increases from the value \( \alpha_1=287 \ \text{W/m}^2\text{K} \) to the value \( \alpha_2=804 \ \text{W/m}^2\text{K} \), which corresponds to 180%.
2. In the thermal structural calculation of the thermal state of the FGDS rings, it was found that the results obtained taking into account the uneven distribution of the convective heat transfer coefficient \( (\alpha=\text{var}) \) differ from the results obtained with a uniform distribution of the convective heat transfer coefficient \( (\alpha=\text{const}) \). The temperature range at \( \alpha=\text{const} \) is \( \Delta T=\text{const}=549.0 - 550.1 \ \text{K} \), and at \( \alpha=\text{var} \) the range is \( \Delta T=\text{var}=554.7 - 555.8 \ \text{K} \). From this, we can conclude that when taking into account the uneven distribution of the convective heat transfer coefficient for this model, heat is withdrawn by 1% less than without it. The account of the uneven distribution of heat transfer coefficients leads to different temperature distribution in the FGDS rings. When calculating with \( \alpha=\text{const} \), the area in the center of the sealing gap is the most heated; when \( \alpha=\text{var} \), it moves to the lower boundary of the sealing gap. This can be explained by the fact that the distribution of \( \alpha \) has a pronounced non-constant character, and when averaging it, there is too large an approximation error.
3. Different temperature distribution leads to changes in the deformations in the seal. In the structural calculation, taking into account the uneven distribution of the convective heat transfer coefficient \( \alpha \), the taper value of the graphite ring was \( \Delta h_{\text{Gr}}=2.38 \ \mu\text{m} \), and in the calculation with a uniform distribution - \( \Delta h_{\text{Gr}}=1.17 \ \mu\text{m} \). This allows us to conclude that when heat removal is reduced by 1%, the taper value of the graphite ring increases by 2 times. The value of the taper of the graphite ring \( h_{\text{Gr}} \) for calculation with \( \alpha=\text{var} \) when switching from cruising mode No. 1 to cruising mode No. 4 increases from the value \( h_{\text{Gr}}=2.38 \ \mu\text{m} \) to the value \( h_{\text{Gr}}=4 \ \mu\text{m} \), which corresponds to 80%.

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