

# LEAKAGE ACCOUNT FOR RADIAL FACE CONTACT SEAL IN AIRCRAFT ENGINE SUPPORT

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**Abstract.** The article is dedicated to the development of a methodology for the radial face contact seal design taking into consideration the supporting elements deformations in different aircraft engine operating modes. Radial face contact seals are popular in the aircraft engines bearing support. However, there are no published leakage calculation methodologies of these seals. Radial face contact seal leakage is determined by the gap clearance in the carbon seal ring split. In turn, the size gap clearance depends on the deformation of the seal assembly parts and from the engine operation. The article shows the leakage detection sequence in the intershaft radial face contact seal of the compressor support for take-off and cruising modes. Evaluated calculated leakage values (2.4 g/s at takeoff and 0.75 g/s at cruising) go with experience in designing seals.

## 1. Introduction

Support seals of the aircraft engine determine the reliable bearing operation and its thermal state [1, 2, 3].

Oil leakage in the air cavity is unacceptable because it leads to irretrievable oil loss and air contamination for the cockpit and cabin pressurization. Apart from that the oil from the support air cavity can get into the engine air path and burn together with fuel in the combustion chambers. This leads to the deterioration of the combustion conditions and the carbon deposition on the chambers walls. Radial face contact seal (further - RFCS) consists of a cover for separating oil and air cavity. Besides, RFCS provides the sealing between the rotor and stator. This seal is insensitive to large axial displacements, so it can be applicable on the turbine supports, where large axial displacements have taken place. RFCS provides an opportunity to have a high temperature in the seal area and it simpler face contact seal in constructional terms [1, 4, 5]. Leakage in RFCS is much less than in the labyrinth seal [5, 6, 7].

## 2. Finite-element model creation of the support

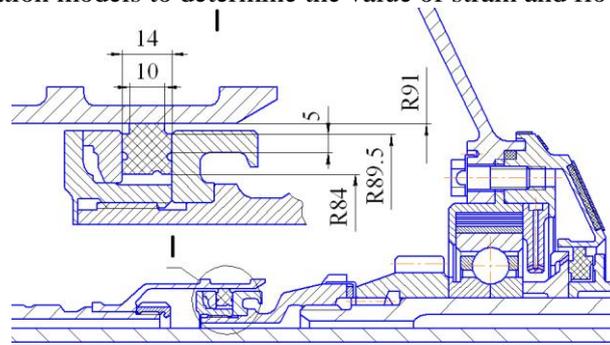
In this work intershaft RFCS of the engine compressor NK-56 was investigated. The support design with the seal is shown in figure 1. The support front wall is washed by the air from the compressor eighth stage. The same air is used for front support RFCS and intershaft RFCS pressurization. Back RFCS is pressurized with air from the tenth stage.

RFCS is installed on the nut on the medium pressure compressor shaft, and this seal consists of a split carbon ring. It is pressed to the hub by the cylindrical band with air entering from the support air



cavity. The hub is held down on the high pressure compressor shaft. Carbon ring also clamped by the end face to the other hub. This hub is pressed onto the nut. Hub and seal ring are fixed against axial displacement by a nut. The hub on the high pressure shaft is also fixed by a nut. Both nuts are locked by special elements. They are folded in the joggle at the shafts faces [5, 8].

Intershaft RFCS geometrical model is presented in figure 1. The values shown in figure 1 were used to create the calculation models to determine the value of strain and flow through the seal.



**Figure 1.** Intershaft RFCS geometrical model.

For that purpose heat, gas-dynamic and hydraulic engine calculations at the maximum (H=0, M=0) and cruising (H=11, M=0.8) mode also were done.

Axial and radial forces, heat generation in the bearing and radial face contact seal have been identified [9, 10]. On the ground of geometrical model flat, axisymmetric and finite-element model was created for subsequent thermal and structural calculations (use model was divided into 16657 elements).

**3. Heat calculation**

For the displacement and deformation evaluations of the intershaft seal parts the following calculations were performed with the imposition previously identified loads.

Thermal analysis (initial data are the boundary air temperature, the convection heat transfer coefficient and the heat flow released from the intershaft RFCS surfaces due to friction in the bearings) [3, 9].

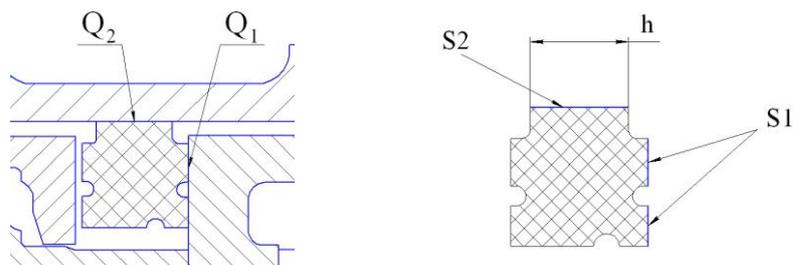
Structural analysis (initial data are the boundary pressure, axial force, with the imposition of the temperature analysis results, heat generated in contact places). The heat amount generated from friction in the RFCS can have a decisive influence on the seal assembly deformation. This deformation is determined by the formula (1) [3]:

$$Q = q \cdot S \tag{1}$$

$q$ , heat flow rate;

$S$ , contact area.

In the investigated seal  $S_1$  and  $Q_1$  - contact area and heat flow rate released from the friction with the high pressure shaft; values  $S_2$  and  $Q_2$  refer to the average pressure shaft. These parameters are shown in figures 2 and 3.



**Figure 2.** Heat flow rates locations.

**Figure 3.** Contact areas.

Heat flow rate can be defined as in equation (2):

$$q = \Delta p \cdot f \cdot \omega \tag{2}$$

$\Delta p$ , differential pressure;

$f$ , sliding friction coefficient;

$\omega$ , angular ring surface velocity relative to the shaft surface.

Vast amount of troubles caused by the ring rotation speed determination. The angular RFCS ring rotation velocity can be gauged through equation (3). This formula was proposed by the author of [11, 12]:

$$\omega = \left[ F_A (m_k \cdot R_S)^{-1} \cdot \left( \frac{f_T}{f_R} \cdot \frac{R_A + f_R \cdot R_O}{R_O + f_T \cdot R_A} - \varepsilon^{-1} \right) \right]^{1/2} \tag{3}$$

$F_A, F_R$ , unbalanced axial and radial forces from the differential pressure on the sealing ring;

$f_T, f_R$ , friction coefficients in the axial and radial friction pairs (their values can be determined on the basis of the following recommendations [6, 13, 14]);

$m_k$ , carbon seal ring mass;

$R_A$ , resultant action force (from the thrust washer on the RFCS ring) radial location;

$R_S$ , radius of the seal ring mass center;

$R_O$ , RFCS outer radius;

$\varepsilon = F_A \cdot F_R^{-1}$ , coefficient.

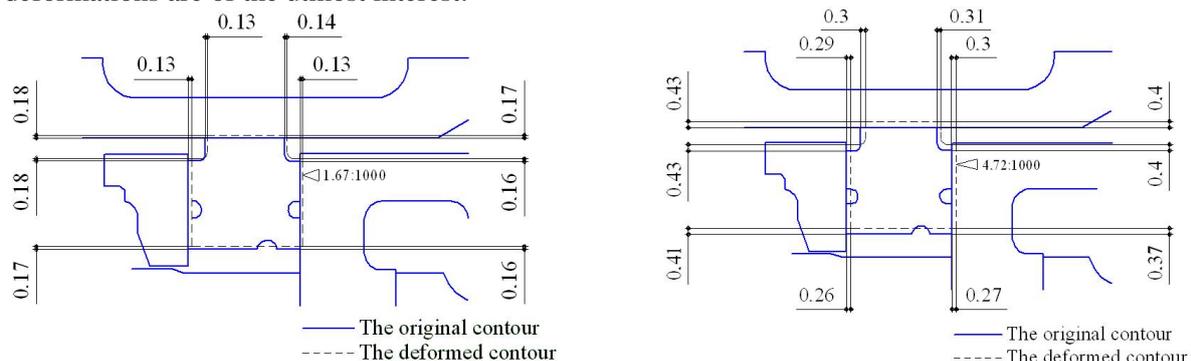
Based on the foregoing calculation and heat distribution analysis in the model were done.

Previously mentioned findings were used for the structural calculation. Then the counted results of the support and seal assembly deformations were analyzed.

The calculations found the temperature distributions when the engine is running in two modes: cruising and takeoff. As a result, maximum temperature has the following values: 248 °C for cruise mode and 599 °C for takeoff. At cruising carbon ring is heated more evenly. The difference between the maximum and minimum ring temperature is equal to 34 degrees. At takeoff the temperature difference between the ring faces is more sudden and it is equal to 77 degrees. This is two times more compared to cruising mode.

**4. Structural calculation**

After receiving the calculated temperature distribution and results analysis structural calculations for the researched modes was performed. The calculation results are shown in figures 4 and 5. The maximum deformation in the cruise mode is observed in the upper seal cover. Carbon ring deformations are of the utmost interest.



**Figure 4.** The ring displacement relative to the rest position at cruising.

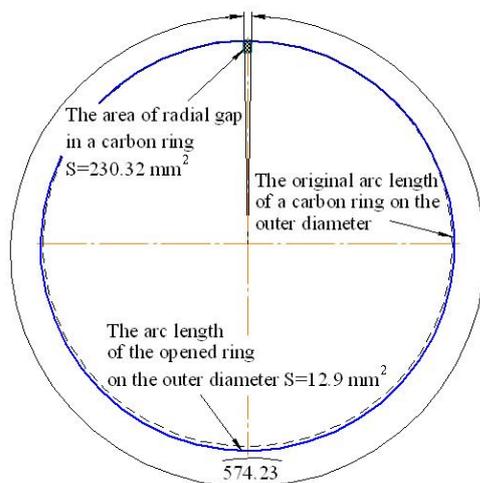
Linear and angular strain corresponding to the cruising regime is shown in figure 4. The figure analysis shows: the ring moves to the right by 0.13 mm and 0.17 mm up from its original position. The angular strain amounted to the following values: about 6' to the face surface, and the same value for the radial. In general, the angular deformation is negligible and both sealing gaps remain plane parallel form. Taper is equal to 1.67  $\mu\text{m}$ .

The maximum deformation for the entire support at takeoff is equal to 2 mm. It corresponds to the same part as on cruise mode and attributed to the same reasons. In the seal assembly (figure 5) maximum strain corresponds also to the upper seal cover and has value of 0.85 mm.

So, the maximum deformation in the seal parts is increased two times more (approximately 110%). The ring is deformed as at cruising: it is shifted to the right by 0.3 mm, and up to 0.17 mm. This conclusion was made after the figure 5 analysis.

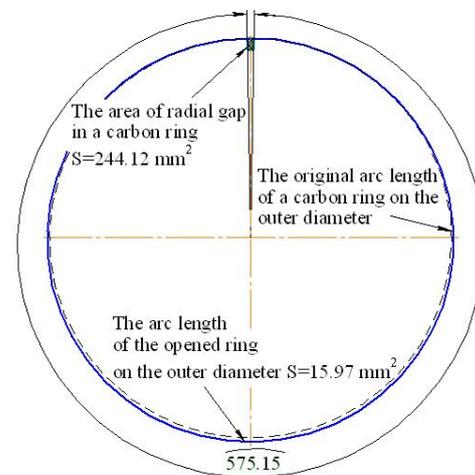
The axial deformation increased in 2.3 times, while the radial remained at the same level. The cross section rotation angle has increased significantly. It has grown in 2.7 times (from 6' to 16'). The excessive taper at takeoff (4.72  $\mu\text{m}$ ) can have an impact on work of the gas-dynamic grooves. The research have shown the small effect of pressure from the boundary air flow on the deformation, compared to thermal extensions. This value is  $\sim 1...3\%$  and it is within the calculation error. The heat released from the friction between the contacting surfaces has a decisive influence on the deformation magnitude. It leads to a radial deformation and completely determines its value.

The existence of the seal ring radial deformation causes a gap change in the RFCS and impairs its leakage. The upper seal cover being leads to the same problems. A tangential and slotted gap changing is displayed in figures 6 and 7.



**Figure 6.** Changing of slotted and tangential gaps at cruising.

**Figure 5.** The ring displacement relative to the rest position at takeoff.



**Figure 7.** Changing of slotted and tangential gaps at takeoff.

Slotted gap arises in the carbon seal ring split. This can occur because of different seal ring and the upper seal cover thermal expansions.

Obviously, after the transition from the takeoff mode to the cruising, the slotted gap area decreased from 244.12  $\text{mm}^2$  to 230.32  $\text{mm}^2$  (about 6%) and the tangential gap area is also reduced from 15.97  $\text{mm}^2$  to 12.20  $\text{mm}^2$  (about 31%). As one can see, the gaps do not change in proportion; this is due to the predominant influence of the upper seal cover radial deformation. Analysis of changes in the gaps areas clearly shows: the slotted gap has a decisive influence on the seal leakage. For take-off mode conditions it is 94% of the total area, and 95% for cruising. It would appear reasonable, leakage will vary in proportion to the gap size and leaking through the tangential gap will be at the 5% level of

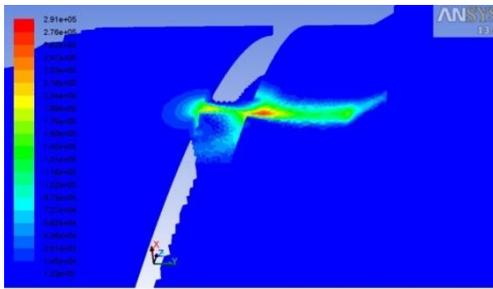
their total volume. The research have shown that if it is necessary to analyze the seal leakage the flow rate changing through the slotted gap is sufficient to investigate.

### 5. CFD modeling of the gap

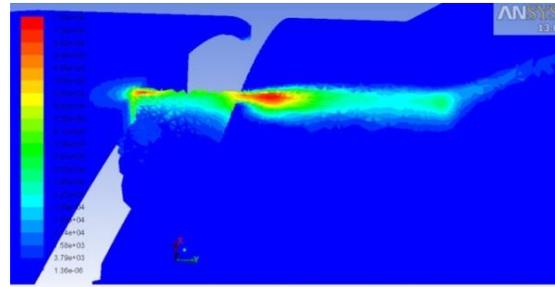
The finite elements tetrahedral mesh was used in this calculation (~490,000 elements in each model). Figure 8 shows the dynamic pressure distribution (ram air pressure) in the air flow path at cruising. The maximum pressure is located in the gap center and it is equal to 291 kPa.

On the calculation results the actuation fluid leakage amount through the gap in the carbon ring on the cruise mode was defined. It has a value of 2.4 g/s. Figure 9 shows the distribution dynamic pressure (ram air pressure) in the air flow path at takeoff. The maximum pressure is located in the gap center and takes the value 75.8 kPa.

On the results of calculation the actuation fluid leakage amount through the gap in the carbon ring on the cruise mode was defined. It is equal to 0.75 g/s. Thus, leakage at takeoff is 3.2 times larger than on cruising (2.4 g/s and 0.75 g/s, respectively).

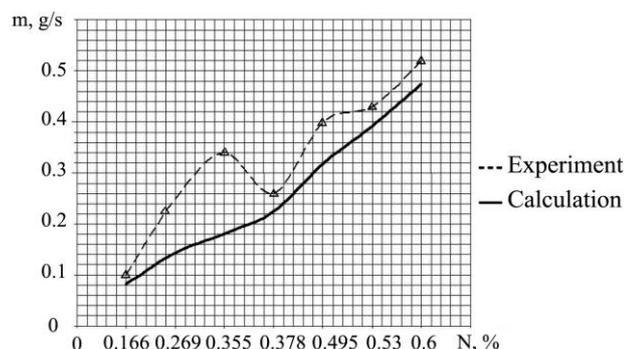


**Figure 8.** Dynamic pressure field distribution at cruising.



**Figure 9.** Dynamic pressure field distribution at takeoff.

The correctness of the generated calculation models and designing seals methodologies as a support part were verified by comparison with experimental data (figure 10) [1, 15].



**Figure 10.** Comparison of the experimental and calculated flow seal characteristics.

The difference between calculation and experimental data is in the range of 8 to 15%. It can be considered as a good result. The used calculation seals models and the seals design method as a support element were correct. It can be approved on the level of the results disarrangement basis.

### 6. Conclusion about the performed research

The performed research served as a ground for the following conclusions:

Firstly, carbon seal ring deformation behaviour does not change in the transition from the takeoff to cruising mode. Ring deflection in the radial direction also remains constant. It is amounted to 0.17 mm. In axial direction the ring is displaced by 0.3 mm at takeoff and 0.13 mm at cruising (the displacement is reduced by 2.3 times). The cross section rotation angle also varies considerably (from 16' to 6').

Secondly, the heat generated in the contact zone has the greatest influence on the size and shape of the seal gap. The heat generated in the face gap is two times more than in the radial gap. The

maximum heat flow value at the face gap is equal to 5147 W at takeoff. Axial force has an effect only on the axial seal ring displacement. It varies between 0.135 mm and 0.029 mm (reduced in 4.7 times) on cruise mode. At takeoff it ranges from 0.3 mm to 0.125 (reduced in 2.4 times). If the pressure is taken into account it leads to the deformation change in the range of 1-3%.

Thirdly, seal assembly parts radial deformation has a decisive influence on the gap magnitude change in the carbon seal ring split. After the transition from the takeoff mode to the cruising, slotted gap area decreases from 244.12 mm<sup>2</sup> to 230.32 mm<sup>2</sup> (about 6%), the tangential gap area is also reduced from 15.97 mm<sup>2</sup> to 12.20 mm<sup>2</sup> (about 31%). Analysis of changes in the gap areas clearly shows: the slotted gap has a decisive influence on the seal leakage. For take-off mode conditions it is 94% of the total area, and 95% for cruising.

Fourthly, leakage calculation through the slotted gap was implemented in FLUENT CFD complex. Calculated leakage values at takeoff is 3.2 times higher than the leakage at cruising, it has a value of 2.4 g/s and 0.75 m/s respectively. These values go with experience in the RFCS designing.

Fifthly, the significant dynamic pressure presence at the seal gap exit (up to 291 kPa at takeoff and up to 75.8 kPa at cruise) should be considered in the RFCS gas-dynamic unloading design process.

On the investigation basis the RFCS leakage calculation methodology was devised. This computing combines: deformation, leakage and thermal analyses. With the aid of the method it is possible to evaluate the leakage on the various engine operation modes.

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