Direction of improvement of the radial-face seals of rotor supports of the aircraft engines

Petr Bondarchuk, Sergei Falaleev
Samara state aerospace university (SSAU), Russia, Samara

Abstract: Today the radial-face contacts seals are the most wide-spread type of sealers of the aircraft engine rotor supports. In the paper the main shortcomings of the radial-face contact seals are specified the removal of which will result in increase in the operating range in terms of the pressure and temperature of the sealing air, reduction of leakages and extension of life-time. On the basis of the literature, patents and catalogues of the manufacturing companies the modern trends of improvement of the sealing structure are considered. The innovative technical solution for the radial-face contact seal with oil lubrication has been developed allowing increasing its efficiency. In order to increase the sealing reliability the hydrodynamic grooves of unique form made with the use of a laser are used. High sealing efficiency is ensured due to the simultaneous application of principles of hydrostatic and hydrodynamic lubrication. The method of calculation of seal properties has been suggested. The results of testing the new type of sealing for the engine rotor support as part of a moving-base simulator and aircraft gas-turbine engine have been presented.

Keywords: Radial-face contact sealing, hydrodynamic grooves, leakages, non-contact operation

1. INTRODUCTION

By designing the new engines improvement of characteristics is performed due to enhancement of the basic operating conditions: compression ratio, temperature of gas in front of the turbine and rotor speed [1, 2]. These factors cause increase in the power and temperature loads on the engine components including sealing.

The contacts seals including the radial-face contact seals are still being the main method of sealing the cavities of the compressor supports of gas-turbine engines (Fig. 1).

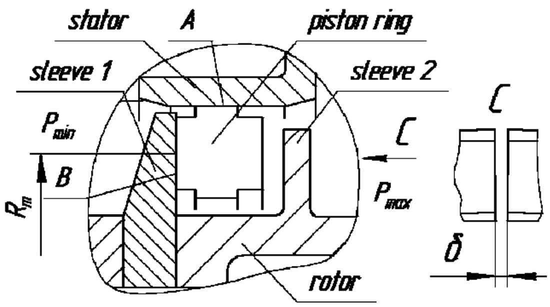

Fig. (1). Sketch of the radial-face contact sealing

The main component of the sealing structure is a graphite piston ring. It is mounted fit inside of the stator centering according to the exterior radial surface A and is retained against it due to the elastic force. On the shaft side the ring is placed between the two sleeves 1 and 2. During the engine operation the pressure on the side of the sealing cavity P_{max} forces the graphite ring against the sleeve face 1 (surface B) and against the stator (surface A). The cross-section of the piston ring is designed in such a manner that the force of friction against the surface A is larger than against the surface B. If this condition is met the piston ring does not rotate and wear takes place only upon contacting the surface B. If the forces of friction against these surfaces are similar the rings begins to rotate. The rotation rate is determined by the relation of these friction forces. The surfaces A and B shall be perpendicular to each other. It should be mentioned that in order to secure the pressing of the ring on the surface B an additional spring element may be added to the structure.

Upon breakdown of perpendicularity of the contact surfaces during operation the graphite ring maintains the flat contact with the surface A. At the same time the contact with the surface B becomes not flat but linear (Fig. 2).

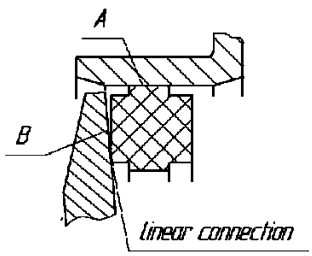

Fig. (2). Operation of the radial-face contact seal in case of skewing

In this case the pressure of contact increases along with the temperature. The wearing of both the sleeve surface and graphite ring takes place accompanied by the increase in the friction coefficient. There comes a point of time when the force of friction against the surface B begins to prevail and the seal ring begins to rotate together with the shaft. This process is of chaotic nature and by switching between the modes both the surface misalignment and ring rotational speed change. As a rule, the cause of such misalignment is the temperature strains of the sleeves and their structural envelope. Most of the leakages are predetermined by the presence of a slit in a ring. By wearing of the radial surface the increase in the gap δ (Fig.1) takes place as well as the proportional increase in the number of leakages. The design excellence of the contact seals is evaluated according to the load parameter «pressure-speed P·V» summing the factors of the surface heating, average speed and wear intensity [3, 4].

As of today in respect of the ordinary constructions there has been achieved the value of 50 MPa·m/s at the maximum sliding speed 100 m/s and temperature 700 K [5]. Upon increase in the rotation speed and rotor diameters (taken into account the necessity to maintain the rotor rigidity) this achieved value is not sufficient. For the Trent 1000 engine the high-pressure rotor speed makes 13500 rpm. At the average seal radius (Fig.1) R_m=100 mm the linear sliding speed will make 141 m/s. So, it may be concluded that design of a next-generation engine is not possible without high-performance seals of oil cavities that are efficient at high rotational speeds and increased pressure. It is also worth mentioning that for the engines and power units in operation the issue of increasing the overhaul life limited by the seals in particular remains always topical. For this reason the increase in reliability of the seals used and extension of their operational range represents the task the solution of which will allow reducing the costs due to the increased overhaul life and reduction in the number of the gas-turbine engine failures.

2. SKETCH DESIGN AND DEVELOPMENT TREND

The air leaks are reduced due to implementation of various design measures. The Burgmann company performs cutting of the complex form of the graphite ring allowing to reduce the leaks (Fig. 3).

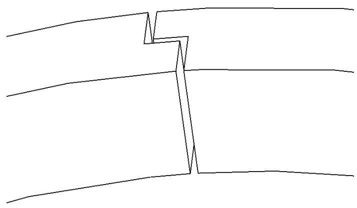

Fig. (3). Cut of the piston ring by the Burgmann company

There are more complex locks used by the companies manufacturing ordinary piston rings. The Grover Corporation company slashes the ring in the way indicated in the Fig. 4.

Fig. (4). Cut of the piston ring by the Grover Corporation

There are also known the attempts to improve the lock due to creation of the hydraulic friction inside of the passage made by cutting (Fig. 5).

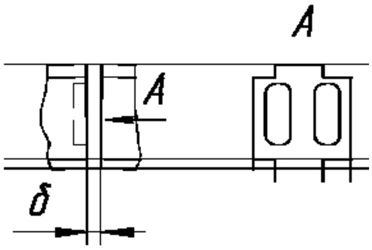

Fig. (5). Creation of hydraulic friction in the cut

The Permaseal™ seal by the Grover Corporation (Fig. 6) creates both the locking bridge and hydraulic friction.

Fig. (6). Permaseal™ seal by Grover Corporation

There have also been made attempts to block the passage of air [6] by arranging at the face surfaces of the cut a special slot in which an insert preventing leaks is put (Fig. 7).

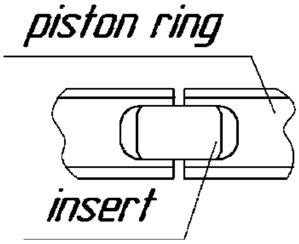

Fig. (7). Seal with an insert preventing the air leakage

In the patent [7] an element made from the porous elastically-damping material "metal rubber analogue" is used as an insert allowing additional adjustment of the force of pressing the exterior surface of the graphite ring on the support body.

The extension of the seal operating range is performed due to introduction of the aerodynamic and hydrodynamic lubrication [8] enabling the non-contact operation of the seal. The essence of the structural improvements implemented is that micro- (fine) grooves are made at the sleeve faces. By the medium injection into the sealing slit the pressure inside of it increases whereby the axial-moving ring is displaced in the friction couple a secured gap of 1...5 μm appears. Such approach allows to significantly reduce the forces of friction against the sealing face and reduce the temperature and, therefore, the misalignment value and ensure the absence of the graphite ring rotation and wear.

The largest seal manufacturers patent the new solutions in the area of use of the hydrodynamic and gas-dynamic effect in the seals. As an example, the patents of Rexnord Corporation -Aerospace housing and shaft assembly [9], Rexnord Industries, LLC - Hydrodynamic seal with circumferentially varying lift force [10], Eaton Corporation - Hydrodynamic magnetic seal [11], Stein Seal Company - Air riding seal [12], United Technologies Corporation - Non-contacting seals for geared gas turbine engine bearing compartments [13]. The sealing components with aerodynamic and hydrodynamic lubrication are patented also by engine-manufacturing companies, in particular, there is a patent of Pratt & Whitney Canada Corp. - Method and device for minimizing oil consumption in a gas turbine engine [14].

These solutions have been implemented in real production and they can be found in the public catalogues of the manufacturing companies (Fig. 8-9).

Fig. (8). Surface of the Rexnord® Mating Rings

Fig. (9). Surface of Centurion® 700 Series mechanical seals фирмы Eatone

3. STATEMENT OF THE TASK

In the Samara State Aerospace University the development and implementation of the long-life and efficient face seals with a gas and fluid lubrication in the natural gas blowers and aircraft engines is performed [8,15,16,17]. Today together with the specialists of the “KUZNETSOV” company the works on modernization of the oil-lubrication seal are performed in order to extend its life-time. In the Fig. 10 the engine unit with the fitted face contact seal is shown.

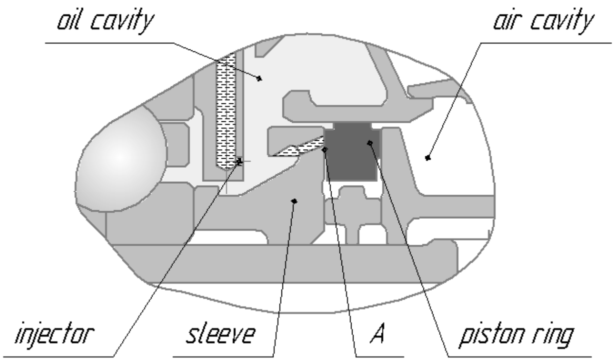

Fig. (10). Axial section of the engine support unit with a face contact seal

The seal assembly separates the compressor air cavity and the support oil cavity. The seal contact on the surface A is lubricated. A few spray nozzles force oil under the sleeve shield where due to the centrifugal forces an oil bath is formed and oil under the pressure enters the gap. The seal piston ring is made from pyrographite, the contact surface of the steel sleeve is covered with the CrMo-coating.

To enable the non-contact operation we proposed to apply the micro-grooves to the sleeve surface A (Fig. 11).

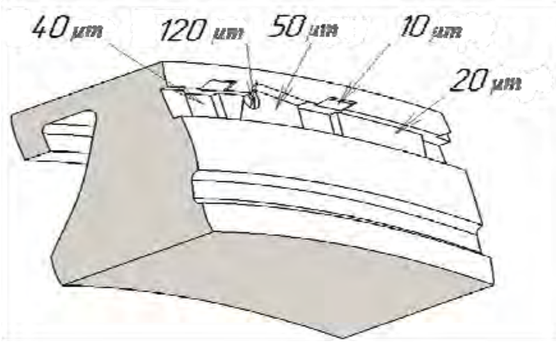

Fig. (11). Micro-grooves structure

The rest of the sleeve structural parameters remain unchanged by improvement of the standard seal assembly. To ensure the uniform oil supply a bevel around the hole is made – the material is cut to the depth of 120 μm and a specially shaped structure for the oil supply to the annular groove with the depth of 50 μm is made. The annular groove of the variable depth (20 and 40 μm) serves for the uniform distribution of oil between the hydrodynamic grooves 10 μm deep. It is needed to perform the computational investigation of the seal, select the optimal parameters, develop the technology of the hydrodynamic groove shaping, produce an experimental model and test it.

4. COMPUTATIONAL INVESTIGATION

In order to perform the computational investigation it is needed to explain the mechanism of operation of the suggested seal structure. Oil supplied under the shield of the rotating sleeve has an increased pressure due to the centrifugal force effect. Under this pressure it is pressed through the orifice holes in the sleeve to the ring chamber at the face surface. And further, in particular under the centrifugal force, it flows through the gap to the external cavity, i. e., back to the support oil cavity. This ensures the hydrostatic principle of the lubrication generation within the seal gap [18]. The value of the oil pressure in the annular groove is determined by the relation of the hydraulic friction of the orifice holes and the seal gap. By means of calculation such seal parameters are selected that will ensure that the oil pressure in the annular chamber does not exceed the pressure of the sealing air. This excludes the oil ingress into the cavity with the sealing air. In order to increase the hydraulic force in the annual chamber slit in the outward radial direction the hydrodynamic grooves are made. The making thereof enables generation of the hydrodynamic force in the gap; i. e. the seal is of a hybrid nature. In this case the rigidity of the lubricating film is increased which ensures the satisfactory dynamic properties of the seal [19].

The dimensions of the designed micro-structure have been calculated with the use of the finite-volume method based on the proprietary know-hows [15, 16, 20]. The standard distribution of pressure over the sleeve segment including a single groove is shown in the Fig. 12.

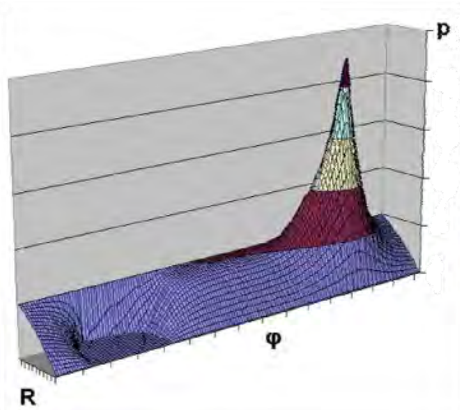

Fig. (12). Pressure distribution over the sleeve segment including a single groove

The dependence of the calculated value of the hydrodynamic force in the slit for the fixed gap size according to the rotor speed is shown in the Fig. 13.

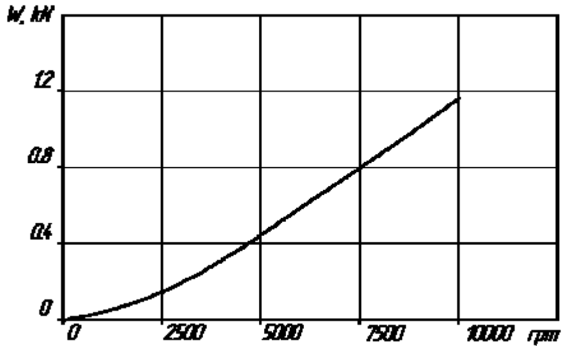

Fig. (13). Dependence of the load capacity on the rotor speed

Performance of such computational investigation allows determining the optimal parameters of the radial-face seal components ensuring its efficient operation as part of the engine support.

5. TECHNICAL IMPLEMENTATION

The grooves were made at the Samara State Aerospace University with the use of the EV15DS laser by Telesys (Fig. 14).

Fig. (14). Laser machine for groove making

A sleeve piece with the groove made is shown in the Fig. 15.

Fig. (15). A sleeve piece with hydrodynamic grooves

After making of the hydrodynamic grooves the finishing of the sleeve surface shall be performed. The designed hydrodynamic groove technology allows ensuring the required depth with the accuracy 0,5 µm. The roughness of the micro-groove bottom makes Ra=0,4 µm.

6. EXPERIMENTAL RESULTS

The tests have been performed on the basis of a moving-base simulator of the “KUZNETSOV” company designed for implementation of conditions of the support seal operation as part of the engine. The first testing stage is checking the seal under advanced operating conditions. The schedule of the seal operation is indicated in the Fig. 16. The slot size δ was controlled before testing and made 0,12 mm and the thickness of the graphite ring equaled 8,93 mm.

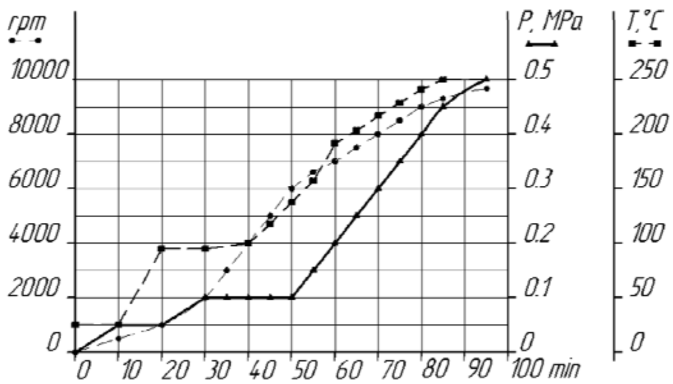

Fig. (16). Schedule of the seal operation during the first testing stage

The test bench on the basis of which testing was performed is a processing one that is why no continuous leakage recording was performed. The maximum leakage value made 3,5 g/s while the permissible range is up to 4 g/s. After disassembly it was found out that the gap δ and the ring thickness remained the same. The design documentation provides for the change of these two parameters, it was the first time that no changes have been observed. The long-term tests were performed at the test bench under n=8000 rpm, T_{air}=210...230°C, P=0.3 MPa with recording the air leakage through the seal. (Fig. 17).

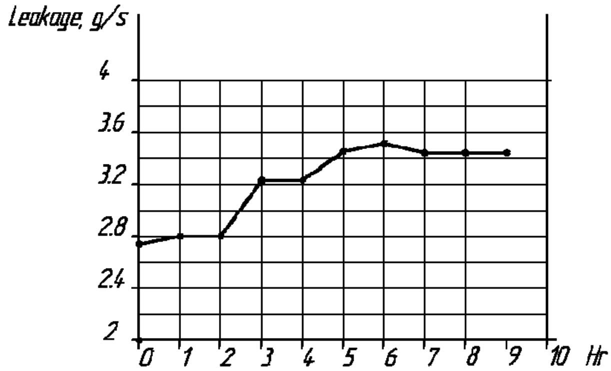

Fig. (17). Leakage during the long-term testing

Also, no changes of the geometric parameters under control have been recorded during testing. There has also been performed testing of the seal as an engine part with the similar results. The appearance of the sleeve surface after all tests is shown in the Fig. 18.

Fig. (18). A sleeve piece with hydrodynamic grooves after testing

7. CONCLUSION

The analysis of patents and developments of different companies has shown that there is a continuous search for the technical solutions enabling to ensure the high tightness and lifetime of the seals of engine rotor supports. The suggested concept of the new radial-face seal is in line with the modern trends of the sealing development [4, 21, 22]. The original technical solution on introduction of hydrodynamic lubrication in the seal assembly of an aircraft engine will allow increasing the service life rates of the radial-face contact seal. However, a number of issues remain unsolved. The first issue is the purity of the oil supplied to the sealing gap [23]. The efficient compact dynamic filters shall be designed. The second significant issue is the possible oil caking in the micro-grooves under the heavy conditions of the engine operation when the seal parts are heated to a high temperature [24, 25]. The third issue is production and application of the wear-resistant coatings for the contact surfaces that will prevent the oil from sticking to these surfaces [24]. The recent achievements in nanotechnology allow expecting a positive result.

CONFLICT OF INTEREST

The author confirms that this article content has no conflict of interest.

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ABOUT THE AUTHORS

Petr Bondarchuk, Assistant of department of construction and design of aircraft engines. Samara State Aerospace University. The author's major is face contactless seals.

Sergei Falaleev, Dr., Prof., Chief of department of construction and design of aircraft engines. Samara State Aerospace University. The author's major is face contactless seals.