10th International Scientific Conference
IRMES 2022
"Machine design in the context of Industry 4.0 – Intelligent products"
26 May 2022, Belgrade, Serbia

SEALING OF THE HIGH SPEED BEARING ASSEMBLIES WITH ONE ELASTIC SUPPORT

Miloš STANKOVIĆ
Nenad KOLAREVIĆ
Dimitrije MIHAJLOVIĆ
Milosav OGNJANOVIĆ
Nikola DAVIDOVIĆ
Marko MILOŠ

Abstract: *High speed bearing assemblies are widely applied in many industries, especially in the airspace and automotive. Design of these assemblies is highly challenging due to some opposed factors affecting their proper functionality. On the one side, there is a condition of rotor-dynamics, which implies the shaft to rotate as stable as possible, retaining its original shape. In regards to this, it was observed that having one elastic and one stiff support, rather than having two stiff supports, provides better dynamic behaviour. On the other side, due to high rotational speed, these assemblies have to be sealed contactlessly, by means of labyrinth, centrifugal or similar types of seals. These seals operate with very small gaps between their rotor and stator. In order to obtain the lowest possible gap in the sealing which allows displacements of the shaft supported by one elastic and one stiff support, a specific design solution is developed. Testing procedure and results, together with design solutions are presented in this paper.*

Keywords: *centrifugal seal, high speed assembly sealing, bearing stiff and elastic support*

1. INTRODUCTION

With the development of high speed assemblies the special attention was applied to sealing solutions. Due to high speeds, use of the contact seals would result in a great amount of losses, in the form of released heat. Thus, the contactless seals came up as a convenient solution, becoming a subject of many researches.

An interesting overview of the seals in high speed assemblies (aircraft gearboxes) was brought by Rahman et al. [1], while the overview of the labyrinth seals is presented by de Souza Barros et al. [2]. Guoqing Li et al. [3] examined a centrifugal seal with the oil-throwing tooth, different from the gas sealing – labyrinth seals. Subramanian et al. [4] performed a numerical calculation on the labyrinth seal, taking into account its centrifugal growth. The same authors extended their research by examining influence of both, centrifugal and thermal effect calculated by CFD software [5]. Nowadays, the centrifugal seals are improved by introducing sealing fluid, brought into the seals through special inlets [6].

In order to satisfy a rotor-dynamics criteria of the compressor shaft, many design solutions were proposed. Some examples are analysed and presented by Kolarević et al. [7]. It is observed that having one stiff and one radially elastic or both elastic supports on the shaft, provides much better rotor-dynamic behaviour (lower radial displacements along the shaft), if compared to both stiff supports. The influence of the bearing support flexibility onto the prediction of the critical speeds of the rotor was analysed back in 2008 by Nicholas et al. [8]. They proposed analytical model as well as experimental verification for 4 types of rotor. Design solution in which the one support is put into the squirrel cage, provides radial elasticity on that support. The squirrel cage elasticity is calculated and optimized for the given criteria. Zhang et al. proposed a multi-objective optimization method and confirmed it experimentally [9]. Wang et al. proposed a new mapping approach to optimize the shape of the squirrel cage slots [10].

2. DESIGN TASKS AND SOLUTION

The unit under test (UUT) is the bearing assembly for turbo shaft engine TSE-200 Phoenix of the company EDePro, which operates at 60000 RPM. In order to obtain the operation at this regime, it is necessary to design contactless seal. Any contact on such high speeds, would result in enormous friction losses and wear. Some of the possible solutions for the high speed assembly sealing would be labyrinth seal (Fig. *1*, a), which consists of a very long and relatively narrow (snake like) gap, causing significant pressure drop which prevents oil leakage. Another solution is centrifugal seal (Fig. *1*, b) [11]. The rotor of this seal is designed to have "steps and teeth" and, when rotates, it forces oil radially to the chambers on the seal stator. The oil is collected in these chambers, then forced by gravity and drainage pump into the drain out system. The second solution, a two stage (two chambers) centrifugal seal is adopted for further development.

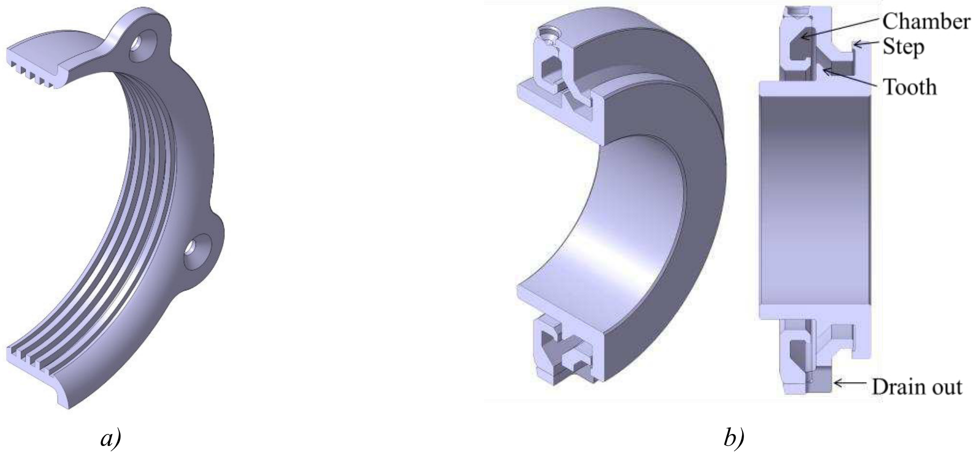


Fig. 1. a) Labyrinth seal, b) Centrifugal seal

Another specific condition of this bearing assembly is imposed by rotor-dynamic. As noted before in this paper, the behaviour of the high-speed shafts is much more favourable if they are supported by one elastic and one stiff support (Fig. *2*). Although it improves the rotor-dynamic behaviour of the rotor, the presence of the elastic support causes the gaps in the centrifugal seal to vary, with respect do radial displacements allowed by that elastic support.

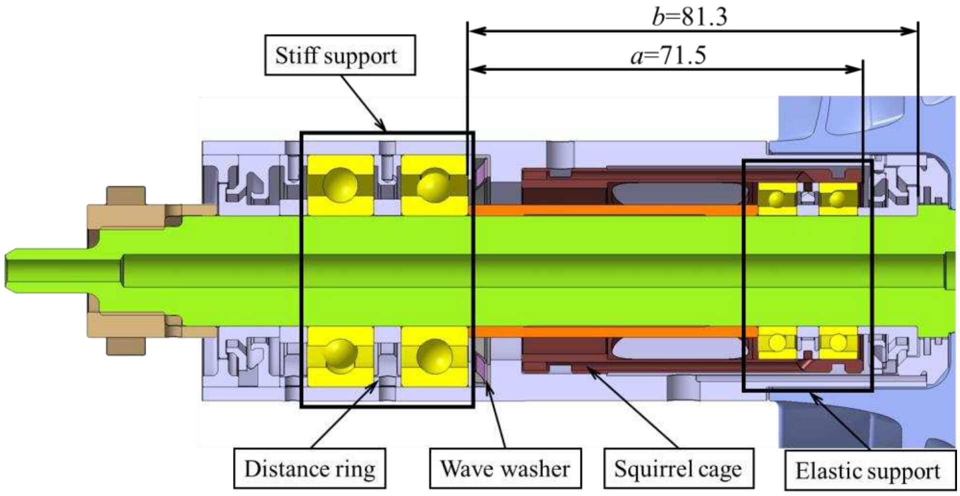


Fig. 2. Bearing assembly of jet engine with one stiff and one elastic support

The more radial freedom in the elastic support, the higher variations of the seal gap between rotor and stator. I.e. when the squirrel cage is not deformed, which is in the stationary mode, the gap in the seal has a uniform circumferential value. On the other hand, extreme gaps occur when the shaft is on the full speed rotation, and the squirrel cage deformations are maximal. By extreme gaps, it is assumed a minimal gap in one point of the shaft circumference, while at the same time the gap is maximal at the opposite point of the shaft circumference.

To secure the proper run of the assembly, a contact between the rotor and the stator should be avoided. Thus, the gap between the rotor and stator of the seal has to be greater than the gap between elastic cage and the bearing housing (Figure 3), which in this specific case equals 0.2 mm.

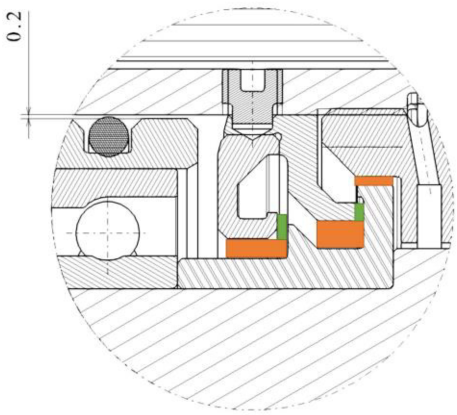


Figure 3. Axial and radial gaps in the seal, with respect to radial gap between elastic cage and bearing housing

Taking into account that the elastic cage possible displacement (a radial gap between the cage and the bearing housing) is designed to be g=0.2 mm, and regarding the proportion of the dimensions in *Fig. 2*, a minimal radial gap between the seal rotor and stator could be calculated:

$$g_x = \frac{b}{a} \cdot g = \frac{81.3}{71.5} \cdot 0.2 = 0.227 \text{ mm} \tag{1}$$

According to this equation, and for the sake of safety in preventing the contact between the seal rotor and stator, the minimal gap is slightly increased and set to 0.3 mm.

To obtain the constant preload of the bearings, on the right side of the stiff support there is placed a special type of spring called wave washer, which is compressed for about 0.5 mm in the mounting position (*Fig. 2*). Between the bearings there are distant rings, through which is the oil brought to the bearings. Separation of bearings with distant rings, allows run on higher RPM for bearings, then it would be the case without the distant rings.

3. EXPERIMENTAL VERIFICATION

In order to verify the efficiency of the centrifugal seal, two tests were conducted – stationary and dynamic.

In the first test, the shaft was not rotating (stationary tests), and the sealing is provided only by under-pressure in seal chambers created by means of the draining pump, which forces the lubricant into the draining pipes. During the second test, the shaft rotates at the maximum speed (dynamic tests) and, in addition to draining pump, the centrifugal force also helps for directing lubricant into the drainage pipes. Both tests are supposed to give the answer on how much oil could be brought into the bearing assembly, before it comes to the leakage through the seals. Measured parameters during the tests were flow rates of oil on both, stiff and elastic support separately. Also the temperatures of the bearings were measured, in order to verify the proper run of the assembly.

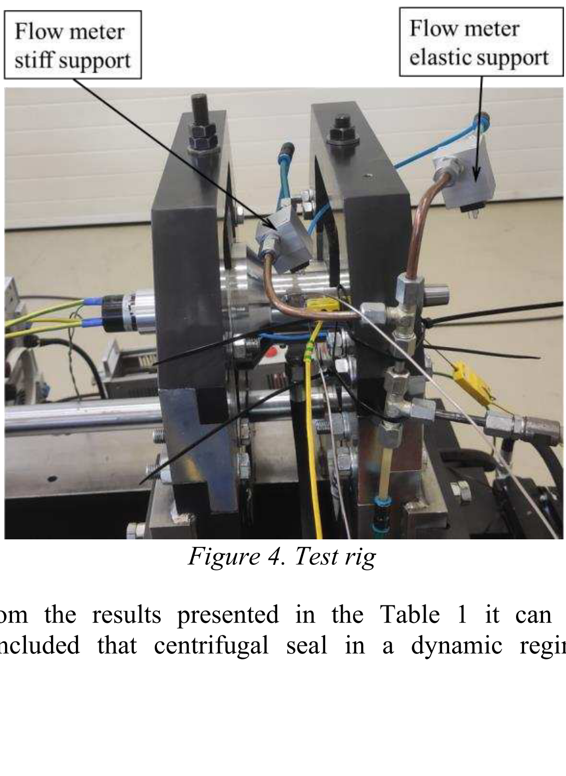


Figure 4. Test rig

From the results presented in the Table 1 it can be concluded that centrifugal seal in a dynamic regime allows about 27% higher oil flow than in a stationary regime.

Table 1. Flow rates of oil on which occurred the leakage on the seal, for the stationary and the dynamic test

	Flow rate on stiff support [g/s]	Flow rate on elastic support [g/s]	Total flow [g/s]
Stationary test	11	4,5	15.5
Dynamic test	13	6,7	19.7

These numbers are taken to define the regime of the engines' start-up. While accelerating from zero to 30000 RPM, the oil flow rates on stiff and elastic support are slightly lower than maximums in the stationary regime tests, and are respectively 10 g/s and 4 g/s. Then the flows are increased to 12 g/s and 5 g/s, and kept on these values while on full working regime.

It is important to note that bearing temperatures did not exceed 85 °C, while maximum allowed temperature for this type of bearings is 200 °C. This indicates that amount of oil brought to the bearings is enough to provide proper lubrication.

4. CONCLUSION

The task of the research was to develop and test a contactless seal for high speed assembly which can operate efficiently, placed next to the elastic support (bearing that has certain degree of freedom in radial direction). For a suitable solution it is chosen a centrifugal, two stage contactless seal.

In order to avoid clash inside the seal, its radial gap is calculated with respect to the maximal radial displacement of the elastic support (the difference between the inner radius of the bearings' housing and outer radius of the squirrel cage).

The sealing abilities of the centrifugal seals are tested experimentally. When the shaft rotates on maximum, the leakage on the seal occurs if the total oil flow exceeds 19.7 g/s. When the shaft is not rotating, the leakage occurs when total oil flow exceeds 15.5 g/s. If comparing these values, it could be observed seal performance improvement of 27% in dynamic regime. Also, it should be noted that the values obtained by tests are satisfactory for the amount of oil necessary for the lubrication of the UUT.

Further development of these seals could go in direction of introducing compressed fluid into the seal chambers, in order to improve the sealing performances. This compressed fluid could be air brought from the compressor of jet engine, or some external compressing device through the separate installation.

ACKNOWLEDGMENT

The authors wish to acknowledge and thank the company EDePro –Engine Design and Production in Belgrade for support and for encouragement in the project realization, as well as the support from the Serbian Ministry of Education, Science and Technological Development, contract no. 451–03-9/2021–14/200105.

REFERENCES

[1] Rahman A. H., Basha J.: Dynamic Seals For High Speed Aircraft Gearbox, Journal on Design and Manufacturing Technologies, Vol.1, No.1, pp. 24-29, 2007

[2] de Souza Barros G. P., Martinez C. B., de Faria Viana, E. M, Diniz H. A. G., Duarte W. M.: Labyrinth Seals - a Literature Review, 17th Brazilian Congress of Thermal Sciences and Engineering, Águas de Lindóia, SP, Brazil, 2018

[3] Li G., Zhang Q., Lei Z., Huang E., Wu H., Xu G.: Leakage performance of labyrinth seal for oil sealing of aero-engine, Propulsion and Power Research, Vol. 8, No. 1, pp. 13-22, 2019

[4] Subramanian S., Sekhar A.S., Prasad B.V.S.S.S.: Rotordynamic characteristics of rotating labyrinth gas turbine seal, with centrifugal growth, Tribology International, Vol. 97, pp. 349-359, 2016

[5] Subramanian S., Sekhar A.S., Prasad B.V.S.S.S.: Rotordynamic characterization of rotating labyrinth gas turbine seals with radial growth: Combined centrifugal and thermal effects, International Journal of Mechanical Sciences, Vol. 123, pp. 1-19, 2017

[6] Kunyi C., Shuangxi L.: Experimental Study on the Performance of the Gas Centrifugal Seal with Continuous Injection and Discharge of Sealing Fluid, IOP Conference Series: Materials Science and Engineering, Vol. 1081, 2021

[7] Kolarević N., Micković D., Crnojević S., Stanković M., Ognjanović M., Miloš M.: Dynamic stability of high speed turboshaft engine with reducer, IOP Conf. Series: Materials Science and Engineering Vol. 393, The 10th International Symposium Machine and Industrial Design in Mechanical Engineering (KOD 2018), 6-8 June 2018, Novi Sad, Serbia

[8] Nicholas J. C., Barrett L. E.: The Effect of Bearing Support Flexibility on Critical Speed Prediction, ASLE Transactions, Vol. 29, No. 3, pp. 329-338, 1986, DOI: 10.1080/05698198608981693

[9] Zhang W., Han B., Li X., Sun J., Ding Q.: Multiple-objective design optimization of squirrel cage for squeeze film damper by using cell mapping method and experimental validation, Mechanism and Machine Theory, Vol. 132, pp. 66-79, 2019

[10] Wang D., Zhang W., Wang Z.P., Zhu J.H.: Shape optimization of 3D curved slots and its application to the squirrel-cage elastic support design, Science China Physics, Mechanics And Astronomy, Vol. 53, No. 10, pp. 1895–1900, 2010

[11] SKF, PUB BU/P1 13383/2 EN • March 2016, page 96

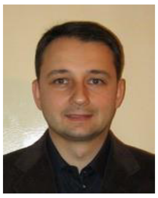
CORRESPONDENCE

Miloš STANKOVIĆ, PhD,
Scientific associate
Innovation Center
Faculty of Mechanical Engineering
Kraljice Marije 16
11120 Belgrade, Serbia
mstankovic@mas.bg.ac.rs

Nenad KOLAREVIĆ, PhD,
Assistant professor
University of Belgrade
Faculty of Mechanical Engineering
Kraljice Marije 16
11120 Belgrade, Serbia
nkolarevic@mas.bg.ac.rs

Dimitrije MIHAJLOVIĆ, PhD student
University of Belgrade
Faculty of Mechanical Engineering
Kraljice Marije 16
11120 Belgrade, Serbia
d.mihajlovic@icloud.com

Milosav OGNJANOVIĆ, PhD,
Professor Emeritus
University of Belgrade
Faculty of Mechanical Engineering
Kraljice Marije 16
11120 Belgrade, Serbia
mognjanovic@mas.bg.ac.rs



Nikola DAVIDOVIĆ, PhD, Assistant professor
University of Belgrade
Faculty of Mechanical Engineering
Kraljice Marije 16
11120 Belgrade, Serbia
ndavidovic@mas.bg.ac.rs

Marko MILOŠ, PhD, Full professor
University of Belgrade
Faculty of Mechanical Engineering
Kraljice Marije 16
11120 Belgrade, Serbia
mmilos@mas.bg.ac.rs