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Selected operational Problems of high-speed Rotors supported by Gas Foil Bearings

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In this paper, several issues related to the operation of gas foil bearings were discussed. The description of the foil bearings' operation was performed, with a special focus on the following issues: the friction processes taking place between the bearing elements together with the friction-generated heat, the appropriate selection of structural and coating materials, ensuring the bearing preload, and the vibration amplitudes in run-up and run-down of the rotor. The operational problems discussed are supplemented with practical examples. A very thorough understanding of the issues at stake makes it possible to specify more precisely the potential areas of application for foil bearings and take into account their operational properties in the contemplated implementations.

1 Introduction

With the development of high-speed fluid-flow machinery, there is a growing need for innovative bearing systems. Foil bearings are ideally suited for such applications as they allow stable and high-speed operation of the rotors at elevated temperatures (Agrawal, 1997; Bonello and Pham, 2014a). They do not require an external lubrication system to be applied for their proper functioning, even under such conditions. Excellent dynamic properties of foil bearings are achieved by using additional elastic-damping elements (usually made of thin metal foils in which a properly modified surface layer is of particular significance). Such bearings have many advantages that make them the preferred option for fluid-flow machines such as gas and vapour microturbines, compressors or expanders (Agrawal, 1997; Bruckner, 2010; DellaCorte, 1997). In their typical applications foil bearings are used to damp vibration, even at high rotational speeds, which is far more difficult to accomplish using gas bearings with high radial rigidity. This results from the fact that the vibration-damping element in such bearings is a specially shaped set of foils which, during operation, interact with each other and with the internal surface of the bush (Bonello, 2014b; Larsen and Santos, 2013; Żywica et al. 2016c). Therefore, a very important issue is the selection of the appropriate shape and thickness of the top and bump foils (Hoffmann et al., 2014; Kim et al., 2009; Larsen et al., 2014). Classical gas bearings operating at high rotational speeds support their loads solely on a thin layer of gas, the vibration-damping properties of which are rather poor. This disadvantage of classical gas bearings is not present in foil gas bearings.

Obviously, foil bearings also have some disadvantages which render them unsuitable for some types of rotating machinery. As foil bearings are still a long way from widespread use, their availability is limited. They are usually manufactured on request and have to be adapted to the actual conditions of use. As the experience gained by the authors of this article has shown, such bearings are also the origin of operational problems that have been reported in the scientific literature only rarely until today. Chief among these problems are wear of the mating surfaces, high starting torque and significant journal displacements occurring during speed and load changes. Many engineers and scientists all over the world constantly work to eliminate as many of these problems as possible, and the subsequent generations of foil bearings are characterized by better and better load capacities (Agrawal, 1997; Heshmat et al., 2005). In general, significant attention is placed on optimization of bearing design and tribological issues, including the selection of suitable constructional and functional materials. Sliding layers in foil bearings may be made of different metals including metal-ceramic composites and plastics (DellaCorte, 1997; Żywica et al., 2016b). Based on the literature review and the authors' experience, it can be concluded that in the case of bearings operating at low temperatures the best results are achieved by using soft sliding coatings for top foils which come into contact with a hard and wear-resistant journal (Kiciński and Żywica, 2014; Żywica et al., 2016b). When bearings are exposed to high temperatures (above 200°C) metal-ceramic composites are often used (DellaCorte, 1997; Jahanmir et al., 2009).

A foil bearing is a mechanical system, the modelling of which is extremely difficult. A model of such a bearing has to take into account several physical phenomena such as non-linear deformations of thin foils having a complex geometry, friction and wear processes on the contact surfaces, heat exchange, thermal deformations, flow-related phenomena taking place within the lubricating gap and fluid-structure interactions (Kozanecki et al. 2011; Żywica et al., 2016c). Therefore, in this case, computational models have limited reliability and are usually used only at the initial design stage. In practice, this means that each manufactured foil bearing goes through a series of tests before application in a target machine. This is done on specially designed test rigs, allowing simulating real operating conditions (DellaCorte, 1997; Tkacz et al., 2015). An effective implementation of foil bearings requires a lot of experience and long-lasting pre-implementation tests during which the bearings operate under extreme conditions, i.e. they are subjected to maximum loads at various rotational speeds. All the problems discussed herein were encountered by the authors of this article in the process of developing and testing new foil bearings. It is worth familiarising yourself with the issues covered here because the knowledge about them should facilitate the elaboration of new bearing systems and their implementation.

2 Temperature Distribution and Wear in Foil Bearings

The experimental research was carried out for the foil bearing consisting of a single top foil and three bump foils. The photo showing all parts of the disassembled bearing is presented in Figure 1. The journal diameter is 34 mm and the bush width 40 mm. The foils were manufactured using sheet metals with 0.1 mm thickness and are made of a nickel-chromium alloy (Inconel). The top foil is coated on one side with the coating made of a synthetic polymer (PTFE) that has supreme sliding properties. The molybdenum-coated bearing journal (made of steel) is plasma sprayed. The foil was profiled by cold forming and the journal surface was subjected to grinding in order to achieve the surface roughness average of 0.63 Ra. The bearing bush (made of bronze) has been prepared in such a way that it can be mounted on a bearing support and the temperature can be measured in 12 locations inside the bearing. Thermocouples were inserted into the bearing through gaps and holes in the bush; all of them operated at a sampling rate of 128 Hz.

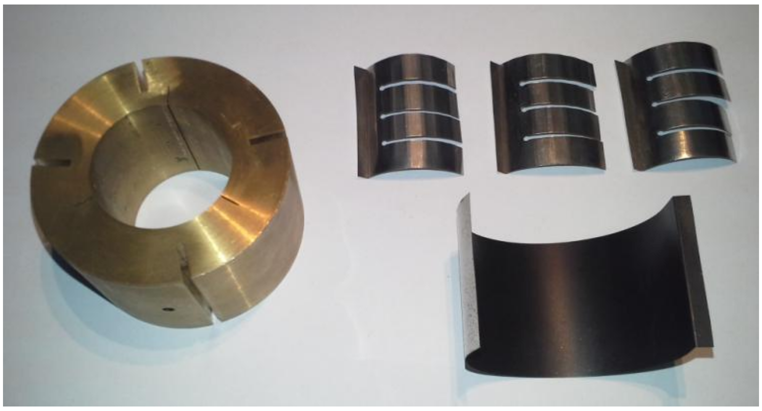

Figure 1. Parts of the foil bearing prepared for assembling

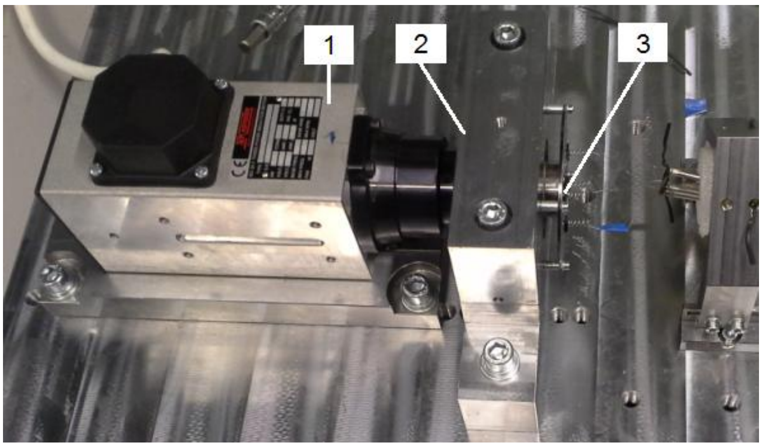

Figure 2. Foil bearing test rig (1 – electro-spindle, 2 – bearing support, 3 – journal)

The experimental studies were carried out on the test rig at the IMP PAN in Gdańsk. The test rig is adapted to testing multi-supported rotors, but only one bearing support was used for this research. The journal of the tested bearing was mounted directly on the electro-spindle shaft. A picture of the test rig is shown in Figure 2. The maximum rotational speed of the electro-spindle is 24,000 rpm. All parts of the test rig were fixed to a solid steel plate, which was equipped with anti-vibration rubber pads. The planned experimental research consisted in increasing the journal's speed up to 15,000 rpm within 55 seconds and then maintaining the same speed until the thermal equilibrium of the bearing node is reached. The operating temperature of a foil bearing is a very good diagnostic symptom since in the case of improper bearing operation its value rapidly increases (Żywica et al., 2016b).

The temperature values measured in the central part of the bearing during its start-up are presented in Figure 3, taking into consideration the circumferential positions of the measurement points. The highest temperature occurred at the lower part of the bearing (200°) and was 59 °C. The lowest temperature increase was observed at the upper part of the bearing (20°), where the temperature was 25 °C. The temperature values measured by the thermocouples positioned at 90° and at 290° were 52 °C and 31 °C, respectively. The temperature increases at different parts of the top foil are between 5 °C and 39 °C, during time duration as small as 55 seconds.

The highest temperature rise in the lower part of the bearing resulted from the movement of the journal in the direction of the lower part of the bearing support, as in the case of a bearing subjected to heavy loads. The differences between the temperatures measured at three different locations situated on the same angular position of the bush were small and did not exceed 2 °C. For this reason, only the measurement results relating to the centrally located thermocouple are presented.

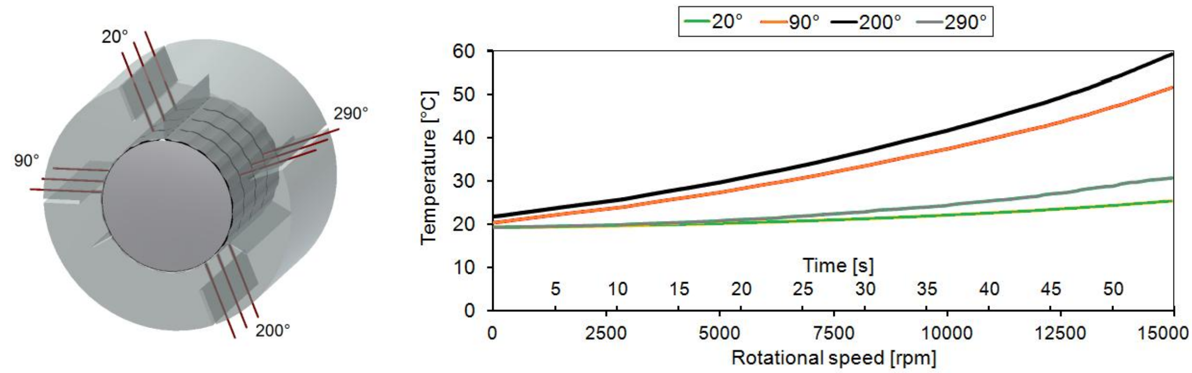

Figure 3. Thermocouples' location (left) and measured temperature of the central part of the foil bearing during acceleration of the rotor (right)

Operating the rotor at 15,000 rpm resulted in further rises in temperature, especially in the lower part of the bearing. During operation of the foil bearing, the temperature differences between various measuring points distributed circumferentially in the bush continued to increase. Despite the expected stabilization of the bearing temperature, its temperature was still rising and – after ca. 300 seconds of operation – reached very high values. The top foil had the temperature of 130 °C and 60 °C at its lower and upper part, respectively.

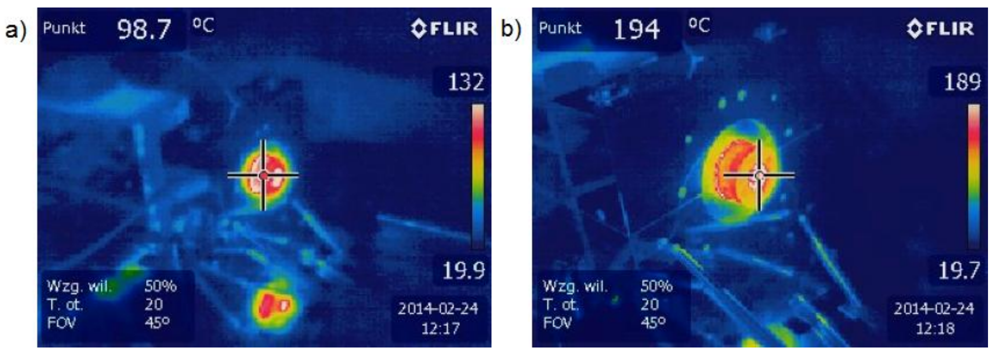

Figure 4. Foil bearing temperature measured using a thermal imaging camera: a – temperature on the external part of the journal, b – temperature in the journal opening

The temperatures of other parts of the bearing were measured using an infrared camera. The results of these measurements are presented in Figure 4. The shaft temperature measured on its cylindrical surface (near the bearing) was around 100 °C. However, the highest temperature value (about 200 °C) was observed in the shaft's hole. Given that there was a serious risk of rapid damage to the bearing, the electro-spindle was stopped. The thermal resistance of the coating covering the top foil was defined by the manufacturer at approx. 200 °C. Its damage may have led to a direct contact between the two raw surfaces not adjusted to such operation. That situation could be dangerous for the bearing and test rig, including electro-spindle.

The bearing was then disassembled in order to evaluate the technical condition of its components. A quick visual inspection of the top foil revealed that it sustained permanent damage at several locations (Figure 5). In those locations in which the top foil was supported by the bump foil, the slide coating was ground down, which caused the friction between the journal and the top foil's construction material. This is why so high operating temperatures had been recorded. The technical condition of the bearing made its further operation impossible. Since the slide coating was ground down only at the lower part of the bearing, it can be said that the reason for this was the journal and bush eccentricities. The deliberate movement of the journal towards the bush caused that the bearing became overloaded. Some skewing of the journal and bush was also observed. Under such conditions, a lubricating film did not form itself in the lower part of the bearing and the journal and top foil surfaces were not properly separated. Only the material covering the surface of the top foil was ground down because it has lower thermal resistance and hardness than the molybdenum coating of the journal.

Figure 5. Damage of the sliding surface in the top foil

The analysis of the results obtained from the research showed that the bearing was not operating properly in the test run configuration applied. Accordingly, the bearing load had to be reduced to increase its lifespan. In order to reduce the bearing load in this system, the precise alignment of the electro-spindle shaft and the bush should be carried out. A similar effect can be achieved by using the so-called floating bush bearing, in which the bush position is adjusted to actual operating conditions. The implementation of this type of solutions is rare and one of the preconditions for this to happen is a small bearing load. When it comes to foil bearings testing, it may be observed, however, that sometimes there is a rapid wear of the sliding layer covering the foils, despite the relatively high compliance of the set of foils. Therefore, one cannot exclude the need to precisely align a rotor supported by foil bearings before running any tests.

Another important issue is the appropriate selection of both structural and sliding materials. In the case discussed above, the sliding layer of the top foil was made of a soft polymer and it was in contact with the journal covered with molybdenum layer. Although this pair of materials successfully underwent testing with lower loads, the test discussed herein resulted in a significant damage to the sliding material. This was caused by too high level of bearing load and the problem of forming a gaseous lubricating film. The bearing operation under such conditions led to a sharp rise in temperature to a level at which the sliding layer no longer possessed its sliding properties. This was obviously followed by an almost immediate wear of this layer.

An interesting result of this work is the observation of a substantial increase in the journal temperature, while there was a relatively low increase in the temperatures of the bush and the bearing support. This was indicative of the problem with a poor heat transfer from the sliding surface of the bearing towards the bush. The thorough analysis of this issue by using a numerical model has demonstrated that the problem is mainly linked to the bearing structure, and more precisely to the small cross-section of the bump foil (Żywica et al., 2016a). The thin bump foil separating the top foil from the bush forms a barrier for heat flow towards the bush. This causes fast heating of both the top foil and the journal. It should be borne in mind that a local temperature rise in a foil bearing can lead to a rapid wear of the sliding layer and a bearing itself. Furthermore, it can happen as the result of even temporary overloads.

3 Vibrations of the Rotor with Foil Bearings

Sets of foils, which are the structural components of foil bearings, improve dynamical properties of the rotors supported by such bearings. In classical gas bearings consisting of a journal and a bush (which are both rigid components), the gaseous lubricating film is the only vibration-damping element, which, at high relative speeds between the sliding surfaces, forms a thin and rigid layer separating a journal and a bush from each other. Such a construction has very limited vibration-damping abilities. The addition of an elastic-damping element between a journal and a bush makes it possible to obtain more desirable dynamic properties. The change of the properties of a set of foils can be carried out by the careful selection of foils geometry and their constructional materials, so as to achieve a stable operation of the rotor–bearings system at high rotational speeds and at different loads.

Experience with foil bearings so far showed that the rotors supported by such bearings are characterized by a reliable operation, even at very high rotational speeds (Bonello, 2014; Bruckner, 2010). In general, this matched what our expectations were. Foil gas bearings also have some drawbacks compared to classical gas bearings, among which the most important are the following: high vibration levels at some speeds and shift of the critical speed towards lower values. These drawbacks mean that serious operational problems might arise in certain foil bearing configurations. This is of particular relevance for microturbines' constructions aiming at decreasing the gaps above vanes in such a way as to improve total efficiency by reducing energy losses. Similar problems we can encounter when designing bearing systems for high-speed electric generators in which there are very small gaps between the rotor and the stator. In such systems, the application of foil bearings may not be able to make up for losses resulting from lower efficiency of the machine and/or its shorter life span, because when the machine operates under extreme conditions, the damage to its rotating elements, and as a consequence of that, damage to the machine itself is likely to happen.

In this part of the article were presented the results of research aimed at the determination of vibration amplitudes at different rotational speeds of the rotor. Additionally, we conducted the analysis of the impact of ambient temperature on the operation of the rotor supported by two foil bearings. The rotor is propelled by the electro-spindle and rotation of the spindle shaft is conveyed by the coupling (Figure 6). The electro-spindle and bearing supports rest on a steel plate. The smooth rotor (shaft without any disc) is made of stainless steel. The rotor diameter is 34 mm, its length is 435 mm and the distance between the bearings is 245 mm. In order to protect the shaft against wear, the bearing journals were coated with chromium oxides and then grinded. The characteristic of the foil bearings was discussed in the previous part of this article. The elevated temperature was obtained by the use of the so-called heat gun (which acted as a hot air blower) and the infrared illuminators shown in Figure 6. In order to check the system’s vibrations at different temperatures and for the whole range of rotational speeds, the measurements were conducted during the run-up of the rotor, increasing its speed up to 24,000 rpm at a constant acceleration.

The measurement results obtained for both room and elevated operation temperatures are presented in Figure 7 and Figure 8, respectively. In these graphs, the resonance area can be clearly identified, which occurred around a rotational speed of 8,000 rpm. The analysis of the results shows that the change in ambient temperature of the system resulted in a change of the resonant speed. When the bearing operated at room temperature (approx. 25°C), the highest vibration amplitudes were observed at a rotational speed of around 8,400 rpm. After the temperature was increased to 100°C, the highest vibration amplitudes were recorded at approx. 7,800 rpm. The decrease of the resonant speed can be explained by a higher compliance of the bearing’s foils when operating at an elevated temperature. In this context, attention must also be given to the increase in the maximum vibration amplitude levels in both bearings. Looking at Figure 7 and Figure 8, it can be observed that the maximum peak-to-peak vibration amplitude, relating to a horizontal direction, changes from 0.18 to 0.28 mm for bearing number 1 and from 0.09 to 0.13 mm for bearing number 2. Similar differences can be observed as regards the vibration amplitudes in the vertical direction recorded in bearing number 1. However, the maximum vibration level in bearing number 2, measured in the vertical direction, was less dependent on the ambient temperature level.

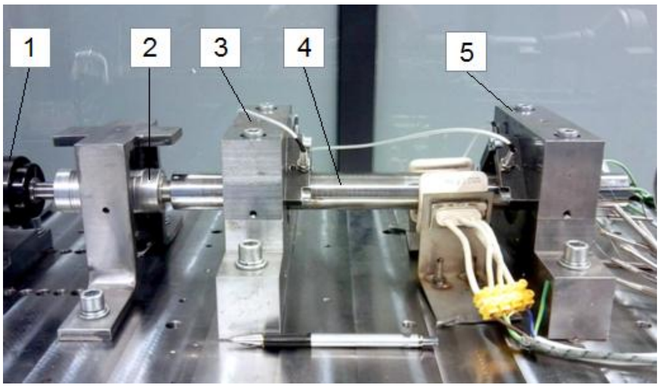

Figure 6. Configuration of the test rig for the rotor with two foil bearings (1 – electro-spindle, 2 – coupling, 3 – bearing support no. 1, 4 – shaft, 5 – bearing support no. 2)

At lower and higher rotational speeds (outside the resonance range), the rotor supported by foil bearings was characterized by a stable operation and the maximum vibration level did not exceed a few dozen micrometres. There was no increase in the vibration level or any signs of unstable operation, even at maximum speed (24,000 rpm). Thus, it can be concluded that independently of ambient temperatures, a very stable operation of the rotor–bearings system has been achieved, under the condition that the current rotational speed remains outside the resonance area of the rotor. In the case of a real machine having a similar rotor–bearings system, it is apparent that only the speeds above the resonant speed (i.e. from approximately 12,000 rpm) could be taken into consideration for determining the operating speed range. A stable gaseous lubricating film did not form itself in any of the bearings operating below the rotational speed of 8,000 rpm. A continuous and reliable operation in this speed range would, therefore, be impossible.

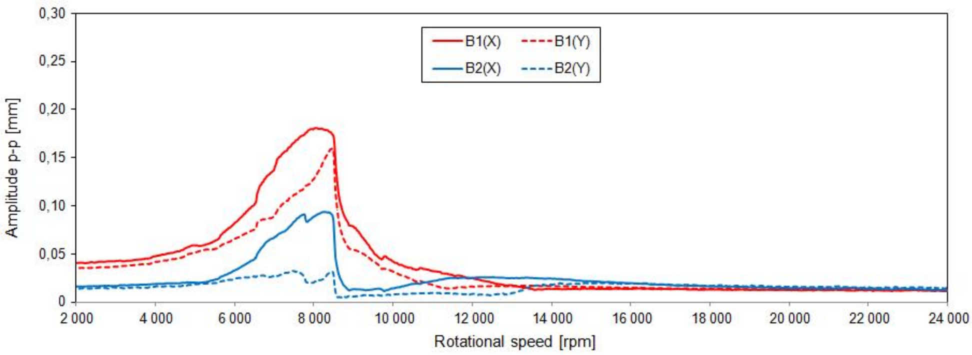

Figure 7. Peak-to-peak vibration amplitude of the shaft vs. rotational speed at 25°C (B1 – bearing no. 1, B2 – bearing no. 2, X – horizontal direction, Y – vertical direction)

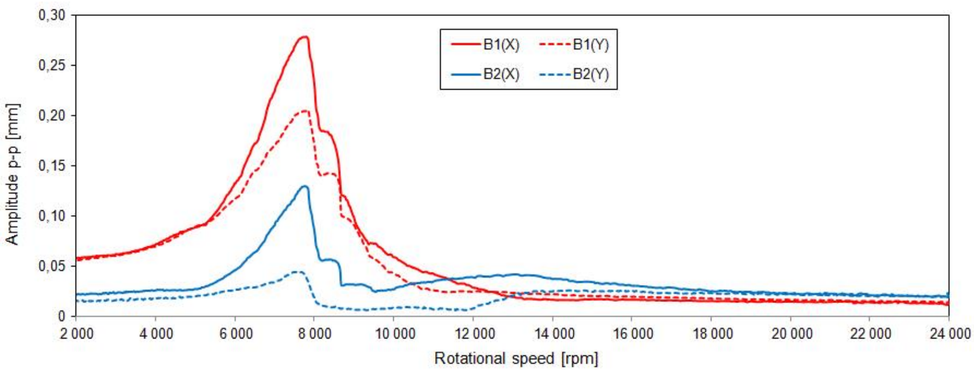

Figure 8. Peak-to-peak vibration amplitude of the shaft vs. rotational speed at 100°C (B1 – bearing no. 1, B2 – bearing no. 2, X – horizontal direction, Y – vertical direction)

The obtained results show that the amplitudes of peak-to-peak vibrations were at a level of 0.3 mm under the least favourable operating conditions. It means that radial displacement of the bearing reached a level of 0.15 mm. For microturbines used in power engineering systems, such a high level of radial vibration is not acceptable on the ground that there are very small radial clearances near the seals and in a blade system. When both the shaft and the disks have small diameters the radial clearance is usually below 0.1 mm (Kiciński and Żywica, 2014; Tkacz et al., 2015), and it depends on the flow system of a turbine (axial-flow, radial-flow or diagonal-flow turbine). In such a machine, the use of foil bearings would be likely to lead to a number of negative consequences (e.g. damage to the rotor or the seal during the first start-up). In the case in which axial foil bearings would be used, these side effects are even more likely to occur, since, in such bearings, the axial displacements caused by a load change can have even higher values than that of the lateral foil bearing types.

In order to avoid the above-mentioned problems, we would like to present a few approaches we use in our research. One such approach is the design of bearings equipped with a set of foils that is very compact and very rigid. However, this often comes at the cost of worsening vibration-damping properties. It is also possible to design flow systems in such a way, that they are more clearance resistant (but only with respect to the preferred directions). Such an effect may be obtained, for example, in axial-flow microturbines in which the blade tips are connected by a ring, which rotates inside a casing. This creates higher clearance at the ring's location, but it does not cause considerable losses since the ring is protected from the working medium flow. It is clear that the application of foil bearings in such machines requires a lot of interference in the construction of a machine embodying our idea.

4 Conclusion

The article discusses the selected issues related to the application of gas foil bearings in modern fluid-flow machinery. Compared to conventional gas bearings, foil gas bearings stand out through their ability to operate at very high rotational speeds and the lack of external lubrication system. However, their application must be preceded by a detailed analysis of a particular machine, i.e. the machine itself must be adapted to the operation with such a bearing system. The main problems existing in foil gas bearings that must be taken into account during the selection process are the following: low load capacity, poor overload resistance, high vibration levels at some speeds, large impact of ambient temperature on the bearing's operational characteristics, a very high dimensional accuracy and assembly accuracy must be maintained. The issues presented on the pages of this article relate to many problems connected with the operation of foil gas bearings. However, they are still spoken of as a bearing system of the future whose best days are still to come.

The first case study concerns the wear of a foil bearing that happened very fast and was caused by the overload which occurred as the result of a wrong position of the bush in relation to the journal. In such operating conditions, there occurred a rapid rise in temperature and the damage to the sliding layer on the top foil. The foil bearing could not be operated any longer because the top foil was so damaged that it was clear that it required replacement. The second case study is focused on the assessment of the impact of temperature and rotational speed on the vibration amplitudes of the rotor supported by foil bearings. Its results show that high vibration levels observed at some rotational speeds can cause the rubbing between the rotating and stationary bearing elements, and as a consequence of that, the damage to the machine itself is very likely to happen.

The authors of the article want to point out that the examples provided do not cover all operational problems related to foil bearing applications. It is also worth mentioning that when designing a foil bearing, manufacturing technology for foils should be developed. It must also be remembered that a proper operation of the rotor at high speeds requires the use of an initial clamp, which unfortunately impedes the run-up. The presented case studies aim to provide an accurate picture of the scale and type of the problems, which may be encountered by all those who want to design and construct fluid-flow machines equipped with foil bearings.

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