Michał Wodtke[*]

HYDRODYNAMIC THRUST BEARINGS WITH POLYMER LINING

HYDRODYNAMICZNE ŁOŻYSKA WZDŁUŻNE Z POLIMEROWĄ WARSTWĄ ŚLIZGOWĄ

Key words

hydrodynamic thrust bearings, tilting-pad bearings, polymer lining

Słowa kluczowe:

hydrodynamiczne łożyska wzdłużne, łożyska z wahliwymi segmentami, pokrycie polimerowe

Abstract

Polymeric linings of sliding surfaces of the hydrodynamic bearings have been used successfully for over 50 years. Despite of their long history of operation and research, they have not become widespread in industrial applications. This fact may be surprising, considering the conclusions that have been published concerning bearing operation and design.

This paper summarizes the current state of the art of the tilting-pad thrust bearings with a polymer lining of pad sliding surfaces. Bearing design and the most commonly used polymeric materials are described. The results of the latest theoretical and experimental researches by both bearing manufacturers and at scientific centres are presented. Observed properties of the bearings with

---

[*] Gdańsk University of Technology, Mechanical Engineering Faculty, ul. Narutowicza 11/12, 80-233 Gdańsk, Poland, e-mail: mwodtke@pg.gda.pl, tel. 58 347 25 84.

polymer lining were compared to the properties of the bearings covered with Babbitt, which is the most frequently material used as lining for hydrodynamic bearings.

INTRODUCTION

The first study of the tilting-pads thrust bearings with polymer lining (pure PTFE – polytetrafluoroethylene) took place in the early 1970's in the former Soviet Union **[L. 1]** (acc. to other sources, simultaneously also in China **[L. 2]**). It was taken as a result of the negative operational experience of the large thrust bearings of water turbines with vertical shafts. Their Babbitt-faced bearings were subjected to frequent breakdowns, especially during transient states of machine operation. As a result of a comprehensive testing program in different operating regimes (steady-state and transient), polymer lined bearings showed very favourable properties, e.g., reliable operation under high specific pressure (up to 6 MPa) or a reduction in the power losses by increasing the temperature of the lubricating oil without lowering the bearing margin of safety. Another benefit was a lower value of the friction coefficient during start-ups, compared to Babbitt-faced bearings. Starts were possible without a hydrostatic jacking system, but that was unfortunately accompanied by wear of the PTFE sliding layer. Interest in polymer-lined bearings began to grow among manufacturers worldwide as a consequence of the positive experience that came from the first Soviet applications of such bearing design.

Presently, polymer lined bearings are offered by the largest bearing manufacturers (from U.S. and Japan). Operational experiences of the polymer lined bearings show that they can operate in very demanding conditions, beyond the usual range of Babbitt bearing applications, e.g., in high temperatures, at heavy loads, lubricated with low viscosity process fluids (e.g., with water **[L. 3, 4]**) and/or with contaminated fluids (in pumps, compressors or gears). In the case of water turbines undergoing upgrades in order to increase efficiency, which usually results in an increase of the axial shaft loads, polymer lined bearings are frequently used as an alternative to white metal bearings **[L. 5]**. According to the data contained in **[L. 6]**, in the years 1978–2012, bearings with a polymer sliding layer (probably only Soviet production) have been implemented in 211 hydropower plants. The average specific bearing pressure of these applications was 4.6 MPa. This is much more than the safe limit for Babbitt bearings operation (2.9 MPa).

Polymer lined bearings did not find wide industrial use despite a number of advantages. This is due to several reasons: the conservatism of users which do not want to change well known solutions, a slightly higher cost of purchasing this type of bearings (in the case of small bearing size the cost is about 2 times larger), and the lack of full knowledge of the phenomena accompanying polymer lined bearings operation (despite a long history of use, the number of available studies on this topic is still relatively small).

The materials of polymeric linings and the bearing design

PTFE used initially for bearing sliding layers, despite obvious advantages (low friction coefficient, limited adhesion) also demonstrated significant disadvantages, e.g., a large amount of creep even in relatively low temperatures, and a strong dependence of mechanical properties on temperature and low wear resistance. For this reason, much research was carried out all around the world to find the best possible material for the specific requirements of hydrodynamic bearings. Composites of PTFE (polytetrafluoroethylene) and PEEK (polyetheretherketone) are presently the most commonly used. According to **[L. 7]**, a typical PTFE composite for sliding layers is filled with a carbon fibre (10–20%) and molybdenum disulphide (4–6%). In the case of PEEK composites, two compositions are in use – PEEK filled with PTFE (8–12%) and PEEK filled with the carbon fibre (27–33%) and PTFE (1-3%, all values determine the mass fraction of the component). Composite materials compared to pure polymers show very good creep and wear resistance **[L. 8]** and a simultaneously low coefficient of friction against steel. Compared to Babbitt, polymers have significantly different physical properties **(Tab. 1)**. This is probably the deciding factor of the favourable course of phenomena in the bearing.

Table 1. Selected physical properties of the bearing linings materials [L. 8]
Tabela 1. Wybrane właściwości fizyczne materiałów pokryć łożysk **[L. 8]**

Parameter		**Babbitt**	**PEEK**	**PTFE**
Elastic modulus	[GPa]	54	3.5	0.5
Melting temperature	[°C]	240	341	327
Coefficient of thermal expansion	[10^{-5}/K]	2	4.7	10
Coefficient of thermal conductivity	[W/mK]	63	0.25	0.25

The polymers have a significantly lower thermal conductivity. For this reason, the heat generated in the fluid film is not transferred to the pad as intensively, as in the case of white metal bearings. This reduces the temperature gradient across pad thickness, and consequentially decreases the influence of the negative effect of the pad thermal deformation. This is especially important in the case of large size bearing pads. Polymers also have a much higher maximum operational temperature. In the case of bearing alloys, it is assumed that they can operate safely at a maximum temperature of ~ 130°C. On the other hand, PTFE and PEEK composites can operate, for short periods, even at temperatures up to 250°C. Babbitt bearings show a significantly larger value of the elastic modulus in comparison to polymers. Increased compliance of the pad-sliding layer is advantageous, because it allows the production of noticeable lining deformations in the area of the maximum oil pressure. In this case, the geometry of the gap can be formed into a ”pocket” shape, which is close to the optimal shape. An additional benefit of the higher elasticity of the sliding layer is the possibility of equalizing the load distribution among individual pads of the bearing, especially in the case of the PTFE composite.

The design of tilting-pad thrust bearings with polymeric lining of the sliding surface differs from Babbitt bearings mainly in the method of the joining of the sliding layer to the pad. The polymers show a limited adhesion, so their direct bonding to steel is very difficult. For this reason, the bonding of polymer layer to pads requires the application of intermediate layers. In practice, two solutions are used most frequently. One with wire mesh (copper or steel) soldered to the pad and the other with a layer of bronze powder sintered on the pad **(Fig. 1)**. Intermediate layer of copper wire mesh has been used in the first Soviet applications of polymer bearings, and it is used mainly for bonding PTFE. The sintered bronze layer was developed in the 1960’s by Glacier Metal Co., and it is used, among others, in the production of PEEK lined bearings. In both solutions, bonding is achieved by the pressing of the polymer sheet against the intermediate layer at elevated temperatures. This produces a polymer flow into the pores of the intermediate layer and allows the formation of an intersection of both materials. Another known solution is to manufacture thrust bearings pads made almost entirely (except for support) of PEEK composite **[L. 3]**. In this case, there is no need to cover the pad-sliding surface with the lining, and an additional benefit is the increased ability to compensate manufacturing and assembly errors due to a lower pad stiffness **[L. 9]**.

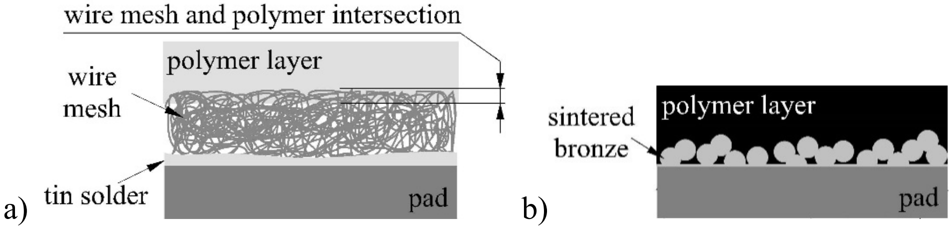

Fig. 1. Solutions of the intermediate bonding layers, a) with the wire mesh soldered to the pad, b) with the layer of sintered bronze

Rys. 1. Rozwiązania warstw pośrednich łączących polimer z segmentem, a) z plecionką z drutu lutowaną do segmentu, b) z warstwą spiekanego brązu

For other pad parameters (e.g., shape or size) polymer bearings do not differ significantly from Babbitt bearings. However, due to the possibility of carrying much larger loads, the design of the pad support requires special attention, because it operates in conditions of higher contact pressure **[L. 10, 11]**. According to **[L. 5]**, PTFE bearings operate at the best performance when the position of the pad support is shifted circumferentially towards the trailing edge (PTEE ~ 53% compared to Babbitt ~ 60%). However, there is no confirmation of this observation in the results available in the literature.

A separate problem, which seems not well studied, is the influence of the lining thickness on the bearing performance. In technical applications, the thickness of the PTFE lining bonded to the pad through the wire mesh is not more than 10 mm. In the case of PEEK bonded to steel with the use of a bronze layer, the thickness of the lining is up to 4 mm. The author knows two works from the literature that contain a theoretical analysis of the influence of the lining thickness on the PTFE bearing performance **[L. 12, 13]**. Obtained results indicate that the increase in the thickness of the pad lining increases the maximum film temperature, and it decreases the inlet and minimum gap thickness and reduces the maximum oil pressure (acc. to **[L. 12]**, after reaching a minimum at certain lining thickness, the maximum oil film pressure starts to the increase with the increase of the lining thickness).

An example of the results of an analysis of the effect of the lining thickness on the circumferential profiles of the gap thickness was shown in **Fig. 2**. It is worth noticing that increasing the lining thickness to 2 mm (larger layer compliance) results in an increase in the ”concavity” of the sliding surface near the film outlet.

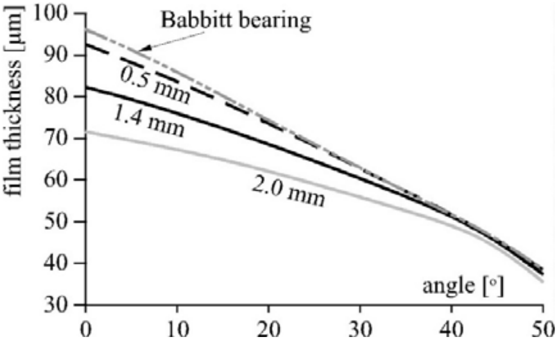

Fig. 2. Circumferential film profiles as a function of PTFE layer thickness [L. 13]
Rys. 2. Obwodowe profile grubości filmu w zależności od grubości warstwy PTFE **[L. 13]**

Theoretical studies

Theoretical studies of polymer lined bearings were carried out using the same techniques as were developed to analyse the Babbitt-faced bearings, e.g. **[L. 13, 14]**. TEHD (thermo-elasto-hydrodynamic) bearing models were used. They are basing on the solution of the generalized Reynolds equation, which allows one to take into account fluid flow and thermal phenomena in the oil film, heat transfer, thermo-elastic pad deformations, and pad tilting. TEHD models, to obtain a solution, require a definition of the boundary conditions, especially for phenomena that are included in the analysis in a simplified manner. This applies mainly to heat transfer from the pads' free-walls and the film inlet temperature. The mentioned parameters were the subject of separate studies, but only for Babbitt-faced bearings. Since the values of the boundary conditions parameters (e.g., heat transfer coefficient **[L. 15]**) influence the results of the analysis, there is the possibility that boundary conditions assumed for analysis of the polymer-lined bearings can cause considerable uncertainty in the calculations. In the author's opinion, this is a significant limitation to analysis.

In theoretical studies of the Babbitt bearings, the presence of the sliding layer, which has different physical properties compared to the pad, is usually not taken into account. This is reasonable, because the mechanical and thermal properties of the Babbitt and steel are comparable. In the case of polymer-lined bearings, such a simplification it is not acceptable, due to the significant difference in the physical properties of the two bonded materials. In theoretical studies dealing with polymer-lined bearings, the sliding layer is always treated as an isotropic material. A separate problem of the theoretical research is the value of the elastic modulus of the sliding layer required in analysis. It has a direct influence on its compliance, which influences the determination of deformations and the shape of the film. Except for **[L. 16]**, the value of the elastic modulus of the sliding layer (polymer + intermediate layer) has not been the subject of a detailed research. The values used in the calculations come from catalogue data or were assumed arbitrarily, without providing the sources or detailed justification.

In the works of Glavatskih and Fillon **[L. 13]** and Ettles et al. **[L. 14]**, the results of theoretical analyses for PTFE-faced tilting-pad thrust bearing were presented. For comparison, calculations were also carried out for Babbitt-faced bearings of the same size. In both works, polymer sliding layer deformations were calculated in a simplified manner, e.g., in **[L. 13]** Winkler's model was applied, while in **[L. 14]**, they were calculated with the use of plane stress assumption. In addition, in the last cited work, it was mentioned (without providing details), that the influences of the temperature and the local stress on the elastic modulus of the lining were taken into account. In both papers, the authors encountered difficulties in obtaining converged solutions for higher values of the bearing load. Such problems were not noticed in the case of the Babbitt bearing calculations.

Comparison of the theoretical analysis results obtained for PTFE and Babbitt lined bearings are summarized in **Tab. 2**.

Table 2. Comparison of the results obtained for PTFE and Babbitt bearings
Tabela 2. Porównanie wyników dla łożysk z pokryciem PTFE i białym metalem

Authors	max. surface temperature T_{max}	max. film pressure p_{max}	friction losses P	film thickness	
				minimum h_{min}	inlet h_{max}
Glavatksih, Fillon **[L. 13]**	↗	↘	-	↘	↘
Ettles et al. **[L. 14]**	↗	-	↘ or ≈	-	-

↗ - evaluated values for PTFE bearing greater than for Babbitt bearing,
↘ - evaluated values for PTFE bearing smaller than for Babbitt bearing,
≈ - evaluated parameter values ~ equal for both bearings.

Generally it can be noticed that the maximum calculated temperatures of the pad's sliding surface were larger for polymer lined bearings (by up to 10°C). This is due to the thermal insulation of the polymer. The polymer layer, due to low thermal conductivity, is a barrier for heat flow to the pad interior. For this reason, the evaluated temperatures of PTFE lined pads were smaller when compared to Babbitt-faced pads and very uniform. The maximum calculated oil film pressure was smaller for PTFE lined bearings at the inlet and minimum film thickness. According to the results of work **[L. 13]**, the minimum film thickness differs between the two bearing types only slightly, while the film thickness at the inlet was lower by about 10 µm. In the case of power losses, it was noticed that they were slightly lower for PTFE lined bearings. However, it also depends on shaft speed, and for low velocities calculated power losses were comparable for both bearing types.

Analyses of the PEEK lined bearings were presented in the later works **[L. 17, 18]**. For this purpose, Fluid-Solid Interaction technique (FSI) available in the commercial software was used. It allows one to carry out coupled analysis of the fluid flow (with the use of Computational Fluid Dynamics, CFD) and mechanics (with the use of Computational Solid Mechanics, CSM). In consequence, fluid flow conservation equations were solved in the fluid part of the problem (usually using finite volume method), while thermos-elastic equations in solid part (typically using a Finite Element Method – FEM). The advantage of the FSI is the automatic exchange of the load (forces, displacements, heat fluxes, and temperatures) between the fluid and solid parts of the model. It takes place on the interaction surfaces, which are common for both parts of the model.

Ricci et al. **[L. 17]**, besides the analysis of the bearing with PEEK lining, also accomplished calculations for bearings with PTFE linings. However, due to problems with calculations, the results were obtained for different loads carried by both bearings. This makes it impossible to compare the bearing properties directly. A comparison of the calculation results for three bearing pad-lining materials (Babbitt, PTFE and PEEK) was presented in work **[L. 18]**. Calculated temperature profiles for analysed bearing lining materials are shown in **Fig. 3**.

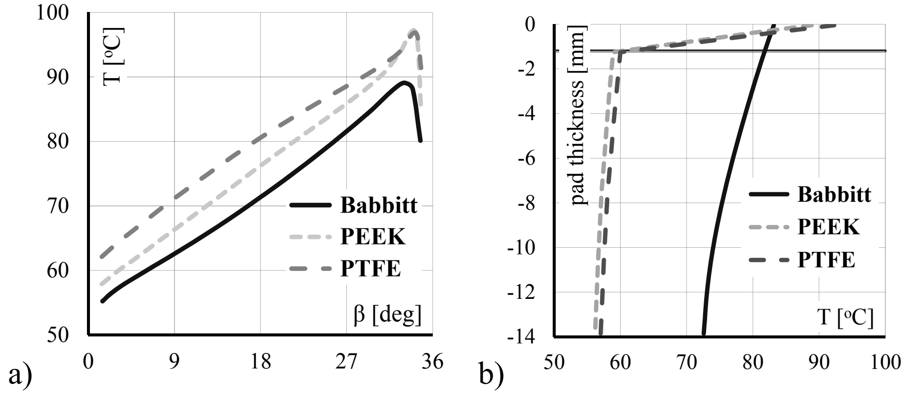

Fig. 3. Temperatures – results of FSI [L. 18]: a) pad-sliding surface – circumferential profile at mean bearing diameter, b) across pad thickness (75%/75% location)
Rys. 3. Temperatury obliczone FSI **[L. 18]**: a) powierzchni ślizgowej segmentu – profil obwodowy na średniej średnicy łożyska, b) na grubości segmentu (punkt 75%/75%)

The highest temperatures of the sliding surface were calculated for bearings with PTFE linings, while the lowest was for Babbitt bearings. The shape of the temperature drop profile across pad thickness differs significantly when comparing the results for Babbitt and polymer lined bearings. In the case of polymer-lined bearings, there is large temperature drop clearly visible across lining thickness (horizontal line in **Fig. 3b** is a boundary between pad and lining). The temperature gradient across the polymer lined steel pad backing is low (3–4°C), whereas, in the pad covered with Babbitt, it reaches 10°C. Changes in the other analysed parameters (T_{max}, p_{max}, h_{min}), as compared to results for white metal bearings, are larger in the case of PTFE than PEEK bearing. This is mainly due to the greater value of the elastic modulus for PEEK than PTFE (see **Table 1**). It seems interesting that the influence of lining thickness and lining material properties on the film profiles is very similar. The higher value of the lining elastic modulus (it is equivalent to smaller lining thickness and lower compliance) results in a film profile shape closer to the Babbitt bearing **(Fig. 2)**.

A very interesting example of PEEK lined bearing calculations is also presented in **[L. 19]**. The analysis was carried out for the high value of the bearing specific pressure (up to 12 MPa). Presented results (temperature and film profiles) proved that, under such high load, the PEEK lined bearing can still operate in hydrodynamic regime.

Experimental studies

Experimental studies of the polymer lined hydrodynamic bearings require special techniques for measuring the temperature which is the most frequently used diagnostic parameter, due to the thermal insulation of the sliding layer. Temperature measurements in a backing of the polymer-lined pad are not representative. With the use of them, it is difficult to interpret the temperature of the oil film. For example, according to tests carried out by Glavatskih **[L. 20]**, a change in the bearing load from 0.5 to 2 MPa (at 3000 rpm) results in an increase in the PTFE lined pad backing temperature by only ~5^{o}C, while in the case of Babbitt bearing, it is even up to ~20^{o}C. For this reason, the thermocouples are most often placed in the holes drilled across the thickness of the pad lining **[L. 14, 22, 23]**. In this way, they have contact with the oil in the oil film. Other methods of temperature monitoring were also developed and tested. For example, drain holes **[L. 13, 21]** or special cylindrical inserts made of copper with a thermocouple placed in the hole drilled in the pad's sliding surface were used **[L. 19]**.

The experimental results (temperatures, pressures, power losses and the film thickness) for polymer-lined thrust bearings in steady state conditions can be found in **[L. 13, 14, 20]** (for PTFE) and in **[L. 11, 22, 23, 24]** (for PEEK). They confirm the results of theoretical works mentioned above.

Another problem for the operation of the polymer lined bearings, which was the object of experimental studies, was the breakaway friction coefficient **[L. 22, 23, 25, 26]**. Tests were also carried out for the Babbitt bearings for comparative purposes **(Fig. 4)**. According to **Fig. 4**, lower values of the breakaway friction coefficient were measured for polymer-lined bearings than for Babbitt bearings. PEEK lined bearings showed only a slightly lower value of breakaway friction coefficient, whereas, for PTFE lined bearing, it was

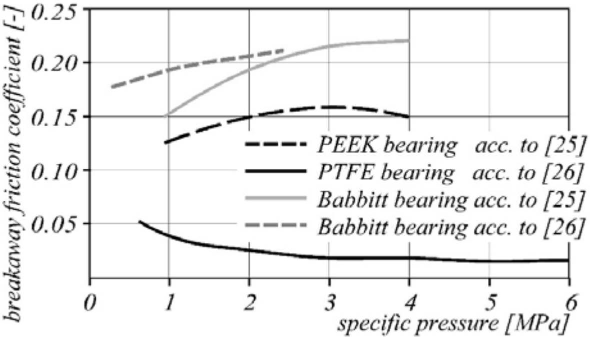

Fig. 4. Breakaway friction coefficient for PTFE and PEEK-faced bearings compared to Babbitt-faced bearing [L. 25, 26]

Rys. 4. Współczynnik tarcia oderwania łożysk PTFE i PEEK w porównaniu z łożyskiem z białym metalem **[L. 25, 26]** measured as low as one sixth of the Babbitt bearing. Additionally, acc. to **[L. 26]**, during starts of the Babbitt-faced bearing, slip-stick was observed, but it was not noticed in tests of PTFE lined bearings.

PEEK lined bearings were also the object of destructive tests in conditions of constant speed and gradually increased load (flooded lubrication) **[L. 24, 25]**. In **[L. 25]**, the tests were carried out on bearing with a mean diameter of 204.5 mm operated at 1000 rpm. Bearing failure was noticed at a specific pressure equal to 26 MPa. Before the failure, a significant temperature rise in the oil film was recorded. Inspection of the failed bearing revealed the wiping of the PEEK composite material and no signs of the polymer adhesion to the steel collar. In work **[L. 24]**, destructive tests were carried out for the bearing of similar size (209.5 mm mean diameter); however, it was at much higher shaft speeds – 6000 and 11000 rpm. Bearing failure was observed for a specific pressure equal to 16.2 MPa (at 6000 rpm) and 14.5 MPa (at 11000 rpm). At 11000 rpm, the Babbitt bearing of the same size as the polymer lined bearing was also tested. It failed under 9.6 MPa (maximum catalogue load for that bearing is 3.8 MPa). The results proved that PEEK lined bearings can operate at much higher loads (up to 50%) than the Babbitt bearings.

PEEK lined bearings were also tested with the directed lubrication system (with the spray bars) with the intention of reducing friction losses and using them in steam turbines (OD 727 mm, 3600 rpm) **[L. 11]**. It was assumed for tests that the limitation of the operational use of the bearings is the maximum temperature of the sliding surface, which is equal to 150°C for PEEK and 130°C for Babbitt. Taking this into account, a Babbitt bearing can operate under the load of 6 MPa, whereas a PEEK lined bearing can operate under a load up to 12 MPa. The increase in turbine efficiency, as a result of the application of smaller PEEK lined bearings (smaller size operating at higher loads) was estimated at 0.05 to 0.1%.

Operational experiences

The possibility of the application of polymer-lined bearings under higher axial loads than on Babbitt bearings has been used in the early development of the PTFE lined bearings in water turbines. Glavatskih **[L. 27]** described an example of such an application in a machine at Bratskaya water power plant (250 MW power output), with the bearing's nominal specific pressure equal to 5.5 MPa. A thrust bearing from one of the machines was equipped with PTFE lined pads (with copper wire mesh as intermediate layer), which had a smaller size. In that way, the specific bearing pressure was increased to slightly more than 11 MPa. During 20 years of routine operation (including 700 start-ups), the bearing showed trouble-free operation, even without a hydrostatic jacking system. Inspection of the bearing revealed the wear of the PTFE lining (~0.5 mm), but it did not has influence its safe operation. Bearing power losses were reduced even up to 50%. The results of theoretical work **[L. 28]** indicate similar benefits in the case of the application of bearings with PEEK linings.

Other benefits of using polymer-lined bearings confirmed by use are excellent dielectric properties, reduced thermal deformations of the pad, the possibility of hot start-ups, or lesser requirements for the manufacturing accuracy of the pad-sliding surface.

SUMMARY

The paper presents an attempt at a synthetic summary of the newest state of the art in the field of hydrodynamic tilting-pad bearings with polymer-lined pads. Increased interest in polymer-lined bearings is visible among manufacturers, although they are not so commonly applied and there are a limited number of available studies in the literature. This is due to their excellent properties and the fact that the Babbitt bearings probably reached the limit of their development.

REFERENCES

1. Aleksandrov A.E.: Primienienije w podpiatnikach gidrogieneratorow elasticznych metałłopłastmasowych segmientow s ftoropłastowym pokrytiem powierchnosti trenja. Gridoteknicheskoe Stroitielstwo, 9, 1981, pp. 12–14, (in Russian).
2. Simmons J.E.L, Knox R.T., Moss W.O.: The development of PTFE (polytetrafluoroethylene) – faced hydrodynamic thrust bearing for hydrogenerator application in the United Kingdom. Proc IMechE Part J: J. of Engineering Tribology, vol. 212, 1998, pp. 345–352.
3. Henssler D., Schneider L., Gassmann S., Felix T.: Qualification and optimization of solid polymer tilting pad bearings for subsea pump application. Proc. of 44th Turbomachinery & 31st Pump Symposia, 14–17 September 2015, Huston, Texas, USA.
4. Inoue K., Deguchi K., Okude K., Fujimoto R.: Development of the water-lubricated thrust bearing of the hydraulic turbine generator. Proc. 26th IAHR Symposium on Hydraulic Machinery and Systems, IOP Conf. Series: Earth and Environmental Science, 15, 2012.
5. Dupuis M., Maricic T.: Thrust bearing PTFE re-design for pump storage generator case study. Proc. of Hydro Vision 2015, 14–17 July 2015, Portland, USA.
6. North American PTFE Bearing, list of PTFE bearings implementations in the world, website: www.ptfehydro.com/ptfe/World-REFERENCES – access 02.2016.
7. ISO 14287:2012(E). Plain bearings – Pad materials for tilting pad bearings.
8. Tanaka T.: Approaches to the safer operation of thrust and journal bearings used in turbomachinery. Proc. of 10th EDF/Pprime Workshop, A: pp. 1–12, 6&7 October 2011, Futuroscope, France.
9. Pethybridge G., New N.: Polymer bearings for sever operating conditions. Proc. of the EDF-LMS Workshop, D: pp. 1–6, Futuroscope, 7th October 2004.

10. Nakano T., Waki Y., Yamashita K., Kaikogi T., Uesato M., Yamada Y.: Development of thrust and journal bearings with high specific load for next generation steam turbine. Proc. of the International Conference on Power Engineering 2007, pp. 350–355, October 23–27, 2007, Hangzhou, China.
11. Sumi Y., Sano T., Shinohara T., Tochitani N., Otani Y., Yamashita K., Nakano T.: Development of thrust bearings with high specific load. Proc. of ASME Turbo Expo: Turbine Technical Conference and Exposition, GT2014-26798, June 16–20, 2014, Dűsseldorf, Germany.
12. Wang S., Tan Q., Kou Z.: Thermal elastohydrodynamic lubrication analysis of large scale composite thrust bearing with sector pad faced by PTFE. Industrial Lubrication and Tribology, vol. 68(1), 2016, pp. 67–75.
13. Glavatskih S. B., Fillon M.: TEHD analysis of thrust bearings with PTFE – faced pads. Trans. ASME, J. of Tribology, vol. 128(1), 2006, pp. 49–58.
14. Ettles C.M., Knox R.T., Ferguson J.H., Horner D.: Test result for PTFE-faced thrust pads, with direct comparison against babbitt-faced pads and correlation with analysis. Trans. ASME, J. of Tribology, vol. 125(3), 2003, pp. 814–823.
15. Wodtke M., Fillon M., Schubert A., Wasilczuk M.: Study of the influence of heat convection coefficient on predicted performance of a large tilting-pad thrust bearing. Trans. ASME, J. of Tribology, vol. 135(2), 2013, 021702.
16. Wodtke M., Wasilczuk M.: Evaluation of apparent Young’s modulus of the composite polymer layers used as sliding surfaces in hydrodynamic thrust bearings. Tribology International, vol. 97, 2016, pp. 244–252.
17. Ricci R., Chatterton S., Pennacchi P., Vania A.: Multiphysics modeling of a thrust bearing with polymeric layered pads. Proc. of 10th EDF/Pprime, G: pp. 1–10, 6&7 October 2011, Futuroscope, France.
18. Wodtke M., Wasilczuk M.: Effect of coating material properties on tilting-pad thrust bearing performance. Proc. of 12th EDF/Pprime Workshop, pp. 1–12, 17&18 September 2013, Futuroscope, France.
19. Pajączkowski P., Spiridon M., Schubert A.: Oil film temperature measurements for highly loaded PEEK coated bearings. Proc. of Hydro 2015 – Advancing Policy and Practice, 26–28 October 2015, Bordeaux, France.
20. Glavatskih S. B.: Evaluating thermal performance of a PTFE-faced tilting pad thrust bearing. Trans. ASME, J. of Tribology, vol. 125(2), 2003, pp. 319–324.
21. Glavatskih S.B.: A method of temperature monitoring in fluid film bearings. Tribology International, vol. 37, 2004, pp. 143–148.
22. Bouyer J., Hanahashi M., Fillon M., Fujita M.: Experimental investigation of the influence of materials on the behavior of a hydrodynamic tilting-pad thrust bearing. Proc. of 15th Nordic Symposium on Tribology, doc. 153, pp. 1–5, 12–15 June 2012, Throndheim, Norway.
23. Bouyer J., Nakano Y., Nagata M., Fillon M.: Experimental study on a hydrodynamic centered pivot tilting-pad thrust bearing. Proc. World Tribology Congress 2013, pp. 1–4, September 8–13, Torino, Italy.
24. Zhou J., Blair B., Argires J., Pitsch D.: Experimental performance study of a high speed oil lubricated polymer thrust bearing. Lubricants, 3, 2015, pp. 3–13.

25. Yamada Y., Uesato M., Tanaka M.: The tribological performance of PEEK lining bearing. Proc. of the EDF-LMS Workshop, B: pp. 1–7, Futuroscope, 7th October 2004.
26. Knox R.T., Simmons J.E.L.: PTFE faced bearings for marine propulsion applications. Proc. of Society of Naval Architects and Marine Engineers Propellers/Shafting Symposium 2006, 19: pp. 1–6, Wiliamsburg, VA, USA, 12–13 September.
27. Glavatskih S.B.: Extending performance limits of tilt pad thrust bearings: a full scale study. Proc. of the EDF&LMS Workshop, G: 1-7, Futuroscope, 2nd October 2008.
28. Pajaczkowski P., Spiridon M., Schubert A., Brito G.C., Marra J.M.: Itaipu binacional hydro power plant thrust bearing design optimization for higher efficiency. J. of Mechanics Engineering and Automation, vol. 5, 2015, pp. 95–106.

Streszczenie

Pokrycia polimerowe powierzchni ślizgowych łożysk hydrodynamicznych są stosowane z sukcesem od ponad 50 lat. Mimo tak długiej historii eksploatacji oraz badań nie znalazły one, jak dotąd, szerokiego zastosowania przemysłowego. Fakt ten może dziwić, biorąc pod uwagę wnioski płynące z eksploatacji oraz dostępnych w literaturze badań łożysk tej odmiany konstrukcyjnej.

W niniejszej pracy podsumowano stan aktualnej wiedzy dotyczącej hydrodynamicznych łożysk wzdłużnych z wahliwymi segmentami z pokryciem powierzchni ślizgowej warstwą polimeru. Opisano konstrukcję łożyska oraz scharakteryzowano najczęściej stosowane materiały polimerowe na pokrycia łożysk. Omówiono wyniki najnowszych badań teoretycznych oraz doświadczalnych prowadzonych zarówno przez producentów, jak i w ośrodkach naukowych. Wskazano również na zaobserwowane różnice we właściwościach łożysk tej odmiany konstrukcyjnej w porównaniu z łożyskami z najczęściej stosowanym pokryciem warstwy ślizgowej stopem łożyskowym.