

Review

Friction and Lubrication of Large Tilting-Pad Thrust Bearings

Michał Wasilczuk

Faculty of Mechanical Engineering, Gdansk University of Technology, Narutowicza 11/12, Gdańsk 80-233, Poland; E-Mail: mwasilcz@pg.gda.pl; Tel.: +48-58-347-1937; Fax: +48-58-347-2742

Academic Editor: Michel Fillon

Received: 28 November 2014 / Accepted: 11 March 2015 / Published: 3 April 2015

Abstract: Fluid film bearings have been extensively used in the industry because of their unbeatable durability and extremely low friction coefficient, despite a very low coefficient of friction dissipation of energy being noticeable, especially in large bearings. Lubricating systems of large tilting pad thrust bearings utilized in large, vertical shaft hydrogenerators are presented in this paper. A large amount of heat is generated due to viscous shearing of the lubricant large tilting pad thrust bearings, and this requires systems for forced cooling of the lubricant. In the dominant bath lubrication systems, cooling is realized by internal coolers or external cooling systems, with the latter showing some important advantages at the cost of complexity and also, potentially, lower reliability. Substantial losses in the bearings, reaching 1 MW in extreme cases, are a good motivation for the research and development aimed at reducing them. Some possible methods and their potential efficiency, along with some effects already documented, are also described in the paper.

Keywords: fluid film bearings; tilting pad bearings; energy efficiency; friction losses

1. Introduction

Fluid film bearings have been extensively used in the industry since at least the industrial revolution. The slider bearings used in ancient times were most probably operating in mixed friction regimes because of low speeds and low precision. The bearings operating in the regime of fluid friction can be characterized by unbeatable durability and extremely low friction coefficients. The invention of tilting pad thrust bearings by A. G. M. Michell in Australia [1] and, independently, by A. Kingsbury in the USA [2] was so ingenious that hardly any design modifications were necessary for decades, which can

be judged by comparing the original design of a tilting pad thrust bearing as patented by Kingsbury in 1912 [3] with a contemporary bearing manufactured by Kingsbury Inc. (Figure 1).

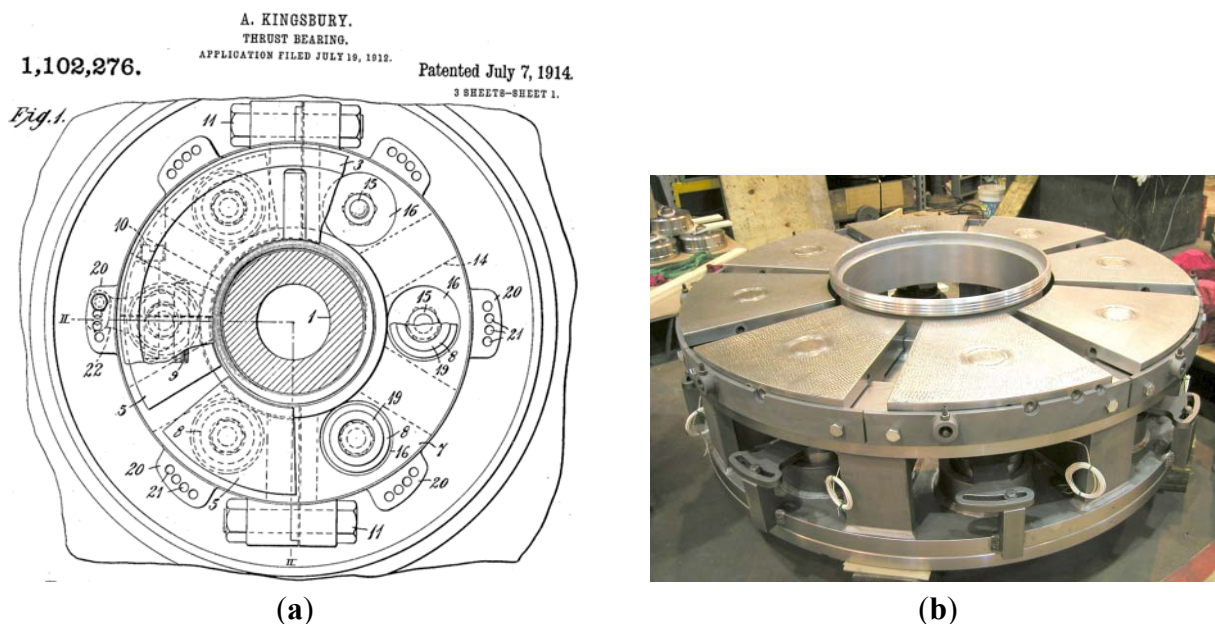


Figure 1. Tilting pad thrust bearing according to Kingsbury patent (a) and a contemporary one—courtesy of Kingsbury Inc. (b).

In the contemporary economy, however, there is always a need for improving machine performance. Two aspects of increasing efficiency can be distinguished. The first one is machine efficiency itself—the machines have to be more efficient so the losses, including friction in the bearings, have to be minimized. For example, in the case of hydroelectric generators, part of the hydraulic energy dissipated in the bearings is not converted to electric energy and decreases the overall efficiency of the machine. In this way, the attempts to decrease losses in the bearings have the same importance as the attempts to improve the geometry of the rotors of the water turbines.

The other aspect of increasing overall machine efficiency is decreasing their downtime and limiting the number and consequences of failures. In hydrodynamic bearings, there is a contradiction between energy efficiency and margin of safety, very well illustrated by the Stribeck curve. Therefore, in order to allow more efficient, and in the same time more risky, bearing operation, it is necessary to predict bearing performance using more accurate bearing models. However, since the bearing failures can be caused by a variety of reasons, it is also important to improve bearing diagnostic tools to be able to prevent their failures.

The paper is mainly based on the author and his team's experience with large hydrodynamic thrust bearings used in vertical shaft hydrogenerators. An example of a large tilting pad thrust bearing is shown in Figure 2. It is shown without the outer part of the housing and without the shaft. The bearing consists of several sector shaped pads supported so that tilting in both the circumferential and usually the radial direction is possible. The vertical shaft of the machine (not shown) goes through the inner cylindrical part of the housing shown in the photograph. A bell shaped thrust block is mounted on the shaft in such a way that during operation its thrust collar slides over the surface of the pads. Bearings are usually bath lubricated. Small rings visible on the pads are the pockets for the oil supplied from the hydrostatic

jacking systems used during starting and stopping to provide sufficient generation of the fluid film lubrication even at low speeds.



Figure 2. Tilting pad thrust bearing (courtesy of Waukesha Bearings).

Some parameters are listed below, in order to illustrate the conditions of operation of large thrust bearings [4]:

- Minimum thickness of the lubricant layer separating rotating collar from a pad surface is approximately 20–50 μm , which is less than the thickness of a human hair;
- Relative speed is equal to 40–45 m/s; this is about 150 km/h;
- The amount of heat generated in the bearing in some cases reaches 1 MW—heat flux generated in the film is equal to approximately 400 kW/m^2 ;
- Axial load can reach several meganewtons (MN), which is over one thousand tons;
- Specific load may reach 5–6 MPa, but in the large bearings is usually limited to 2–3 MPa;
- Outer diameters of the thrust bearings exceed 5 metres—Three Gorges (China), Itaipu (Brazil)—5.2–5.3 m.

Like in other hydrodynamic bearings, the operational range of large tilting pad thrust bearings is limited by three limits: film thickness, mechanical, and temperature limit [5].

The main problems of the design and operation of large tilting pad thrust bearings are [4]:

- Excessive pad deformations affecting generation of hydrodynamic pressure,
- Uneven load sharing among bearing pads,
- Excessive temperatures affecting the bearing alloy,
- Inadequate load carrying capacity at transient states.

Generating pressure distribution of favourable profile and adequate magnitude can be difficult, especially in large bearings, because of excessive thermo-elastic deformations of pads. In addition, taking the whole bearing into account is challenging because of uneven distribution of load on the bearing pads. Generation of a large amount of heat in the fluid film can cause excessive temperature harmful to oil and the bearing surface and increases thermal distortions. In this way, the main problems in large thrust bearings are also connected with the amount of energy losses. By reduction of these losses,

which is equivalent to the reduction of heat generated in the bearings, the problems caused by heat generation and transfer will also be reduced. Pressure in the hydrodynamic film is generated due to relative motion of the surfaces separated by the layer of viscous fluid. Low speed during machine start up and shutdown makes it difficult to generate adequate pressure allowing for full film lubrication (complete separation of collar and bearing by a lubricant film); hence, in many cases, hydrostatic assistance is provided at start-ups and shutdowns.

2. Lubrication Systems of Large Tilting Pad Thrust Bearings

2.1. Bath Lubrication with Internal or External Cooling Systems

Usual methods of lubrication of large thrust bearings did not change much since they were first invented—in vertical shaft machines, it is natural to arrange a bath lubrication with the pads immersed in oil. Because of large amounts of heat generated in the bearing, up to 1 MW in extreme cases, it is necessary to provide forced oil cooling. In the example shown in Figure 3, internal oil coolers are built in the housing. Cold water circulating in them cools the oil. In older designs, internal coolers built in the bearing housing were a standard design. In such a design, the size of the housing limits the size of the coolers and, while water circulation inside the coolers can be controlled, the character and intensity of oil flow outside the coolers is not easily controllable. This is not the case in the external cooling systems, where oil is pumped out of the bearing housing and cooled in standard heat exchangers, in which due to their design, optimum heat exchange parameters exist. Higher output and higher intensity of heat transfer are possible in external cooling systems because:

- The area of the heat exchanger is not limited by the size of the bearing housing.
- The heat transfer coefficient depends on the velocity of both fluids inside the cooler and when turbulent flow of both fluids is possible, which intensifies heat transfer—usually the heat transfer coefficient is 110–180 W/m²K for a built-in water oil cooler [6], and 600–680 W/m²K in an external plate heat exchanger (PHE) [7].
- The counterflow arrangement helps to fully utilize the natural differences of water and oil temperatures, which is not possible in internal cooling because of the complex flow pattern of oil inside the oil bath.

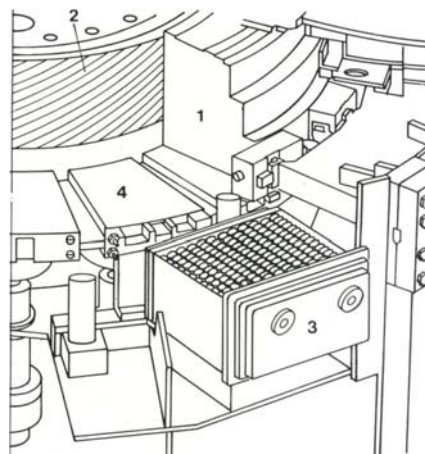


Figure 3. Radial/axial bearing of Itaipu Power Plant. Internal oil cooler marked with 3 [8].

These are probably the reasons why the external system that provided better heat transfer coefficients became more common. Apart from higher efficiency, there are also other advantages of using external cooling systems, and the following list of benefits can be specified:

- The cooler output can easily be adjusted by the changing number of plates (in PHE).
- High velocity of fluids and turbulent flow prevents sedimentation of contaminants on the plates, and the output is not affected by the contamination (in PHE).
- The use of standard elements in the cooling systems increases quality and reliability
- Due to external coolers, water is not pumped into the bearing housing, so the danger of water leakages inside the housing is decreased—special design of the seals in PHE further increases the reliability.
- Oil circulation helps to provide continuous filtration of oil by a fine filter built in the system.

An example of benefits of introducing a newly designed external cooling system is the case for Polish Porąbka Żar Pumped Storage Powerplant (PSP). The decision was taken after an analysis of operational data [9]. One of the conclusions was that thrust bearing temperature could be kept below 80 °C when the oil bath was cooler than 36 °C which, in turn, was only observed with water temperatures below 14 °C, and that was the case only in colder parts of the year. In the summertime, the cooling water temperature was increasing to 20–22 °C, then oil bath temperature reached 42–44 °C, and the bearings operated at an alarm level of 85 °C.

The cooling system designed by the Gdansk University of Technology team and installed in the power plant is shown in Figure 4. The main and reserve oil pump driven by an electric motor can be seen in the photograph, as well as a plate heat exchanger. A precise full flow oil filter is also installed in the system, but it cannot be seen in the photo.

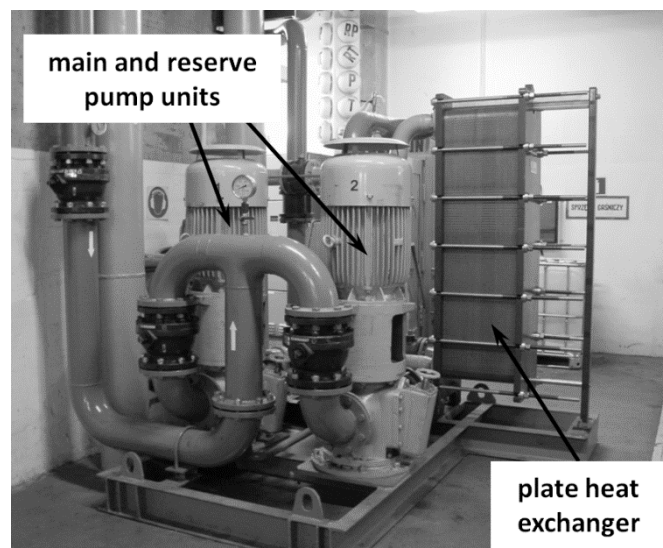


Figure 4. Photo of an external cooling system of Porąbka Żar PSPP.

The cooling system proved successful and the analysis of the operational data showed that, with a new system, it was possible to keep the thrust bearing temperature around 77 °C, even at the highest cooling water temperature of 22 °C, while before modernization it was about 83 °C (Figure 5).

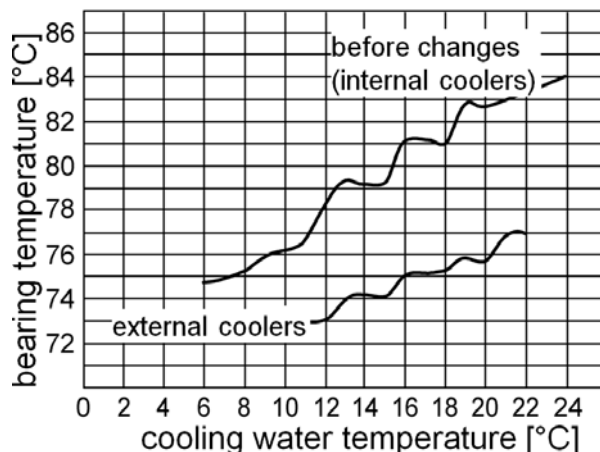


Figure 5. Temperature of the thrust bearing as a function of cooling water temperature with external and internal cooling systems in Porąbka Żar Pumped Storage Powerplant (PSP).

Comparison of bearing temperatures in three units in summer months is presented in Figure 6. One can observe that the bearing temperature in two units with traditional internal cooling systems usually exceeds 80 °C and often approaches an alarm limit of 85 °C, while in the machine with a modernized external cooling system, the temperature does not exceed 80 °C. External cooling systems also present some drawbacks—especially additional energy consumption by the pumps and complexity of the system. Because of small differences in temperature between oil and water, the required oil flow is large—in this case, in the cooling system of approximately 300 kW capacity, the required oil flow was about 1200 L/min, and the power of the electric motor was 37 kW. One of the pumps was a reserve unit.

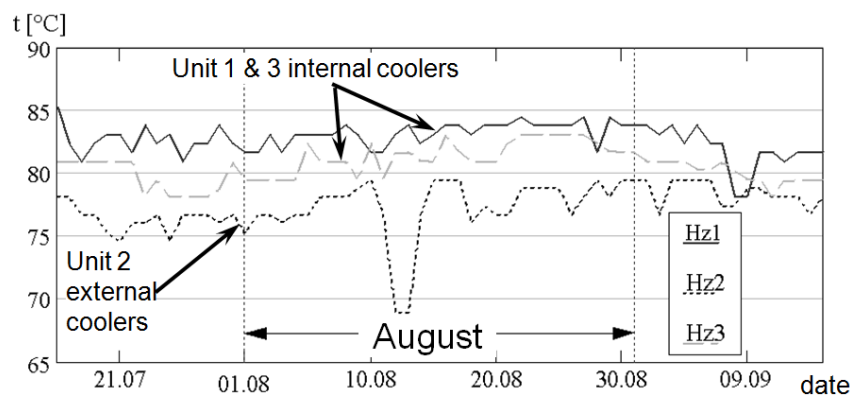


Figure 6. Temperature of the thrust bearing as a function of cooling water temperature with external (Hz-2) and internal cooling systems (Hz-1 and Hz-3) during summer months in Porąbka Żar PSP.

2.2. Directed Oil Supply

In external cooling systems, the oil is pumped into the housing, and special systems of cooled oil supply can be arranged—schematic view of such a system is presented in Figure 7 [10].

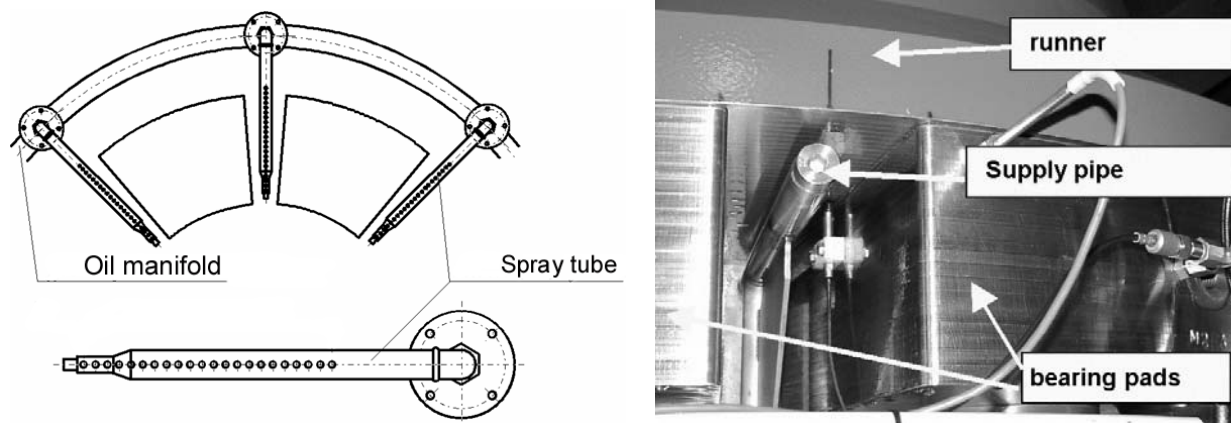


Figure 7. A schematic view and a photograph of the system of cold oil direct supply [10].

Although various special installations of fresh oil supply to the zone between bearing pads exist, there is no systematic evidence of their efficiency. On the contrary, calculations carried out with the use of Computational Fluid Dynamics (CFD) showed that such direct oil supply does not considerably decrease bearing temperature or inlet temperature considerably. Calculations were performed for a relatively large (OD = 1.8 m) thrust bearing of a low speed (180 rpm) hydrogenerator. In the results presented in Table 1 [11], one can see that only after a threefold increase of oil flow, from nominal 330 L/min to 990 L/min, the decrease of inlet temperature becomes noticeable, and still maximum bearing temperature hardly exceeds 1 °C.

Table 1. Bearing maximum temperature and inlet temperature for bath lubrication and direct supply with different oil flow.

Q_{lubr} (L/min)	Lubrication System	T_{max} (°C)	T_{inlet} (°C)
330	bath	68.0	51.9
	direct supply	68.0	51.5
660	bath	67.8	50.3
	direct supply	67.2	48.9
990	bath	67.7	49.7
	direct supply	66.6	47.2
1320	bath	67.6	48.9
	direct supply	66.2	45.7

3. Attempts to Reduce Bearing Losses

3.1. No Bath Lubrication

The friction coefficient, results when shearing of the fluid film is very low, but total losses in the bearings are also the result of mixing of oil within the bearing housings. The latter component may even exceed the component of shearing in the film, especially at high rotational speeds. Shearing of the film is a loss component which cannot be avoided in fluid film bearings, while the other components are not necessary to generate film lubrication. Components of the losses and the results of calculations performed for high and low speed bearings are shown in Figure 8 [12]. One can see that while at about



40 m/s, 30% of the total loss is generated outside the film; however, at a low speed with a water turbine bearing ($v = 14.5$ m/s), it is less than 10%, with more than 90% of the total loss generated in the film. The results show that, although no large decrease of losses can be expected by elimination of churning losses in classical medium sized turbines of low rotational speed, the benefits can be larger in reversible machines, which usually are built for higher rotational speeds, or in larger bearings in which sliding speed is higher.

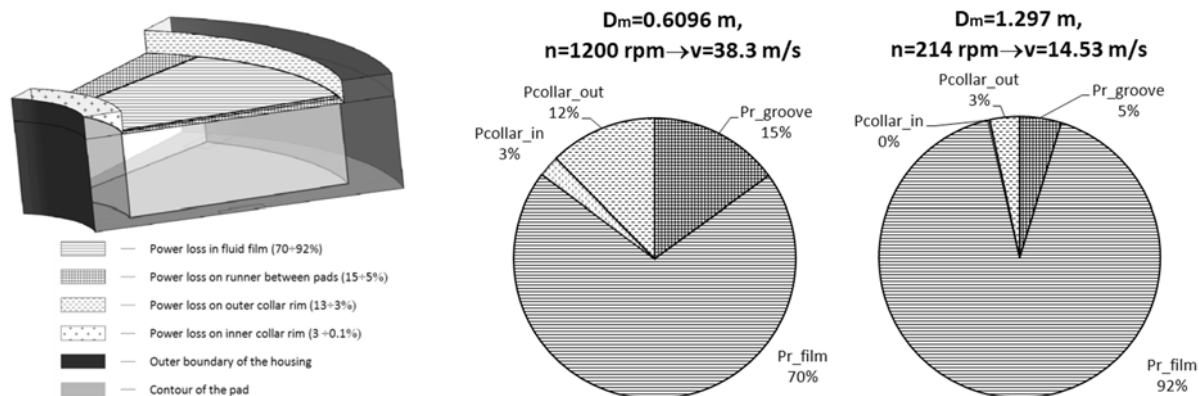


Figure 8. Components of power loss in a bearing with flooded lubrication, and comparison of power loss in two regimes of bearing operation depending on size and rotational speed [12].

Because of such proportions between film and out-of-film losses, various methods of supplying oil directly to the film were developed, such as spray lubrication systems or oil supply grooves installed at the inlet zone of the pad. Methods of direct supply are utilized especially in high speed bearings, contributing to a decrease of losses reaching 45% [13–15]. Two examples are shown in Figure 9.

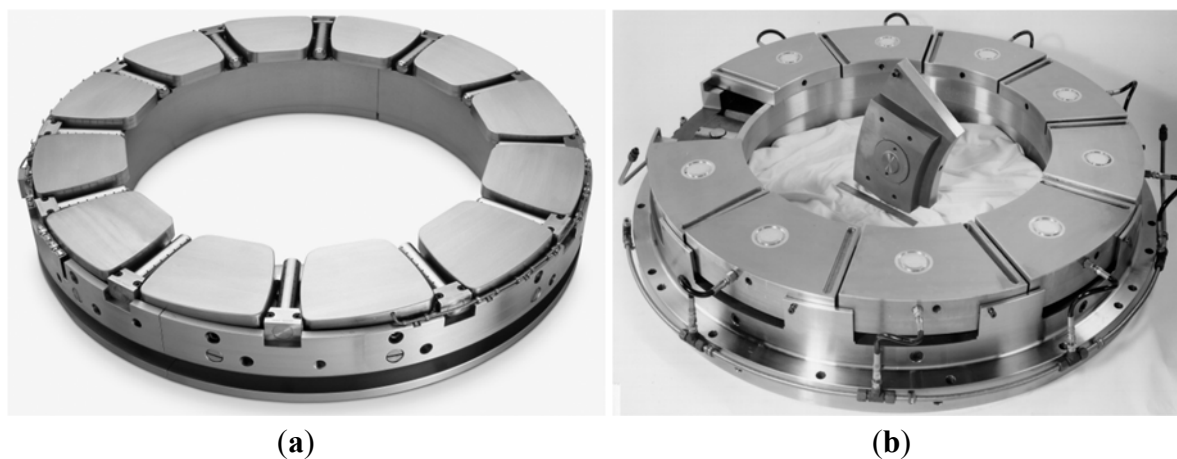


Figure 9. Examples of direct oil supply (a) WBC™ “Directed Lubrication” (courtesy of Waukesha Bearings; (b) Leading Edge Groove® (courtesy Kingsbury Inc.).

Interestingly enough, such systems have not been incorporated in large thrust bearings of hydrogenerators, according to the author’s knowledge, and in these machines, it is still a common practise to use bath lubrication. From the point of view of machine reliability, bath lubrication systems are beneficial due to fail-safe properties. Even if the machine is shut down for some emergency reason,

the bearing is still lubricated and enables safe coasting. With the no-bath system, any stop in the oil supply may cause severe damage to the bearing unless a redundant supply system is provided. Lubricating systems will be described in more detail in the next paragraph.

3.2. Decrease of Viscosity—Standard Oils, High Viscosity Index (VI) Oils

Ferguson, Yuan and Medley [16] were testing the influence of oil grade and temperature in a test rig with a large thrust bearing using three oil grades—ISO VG 68, VG 46, and VG 32 and increasing oil bath temperature. They managed to achieve a threefold decrease of the power loss, but at the cost of a decrease in film thickness. In the case of ISO VG 32 oil at 33 °C of bath temperature, friction loss was equal to 73 kW, and after the increase of bath temperature to 90 °C, it dropped to approximately 27 kW, but, simultaneously, the film thickness decreased from 22 μm to 14 μm . The authors stated that the film thickness of 14 μm is adequate, but, nevertheless, the safety margin is considerably lower. If one recalculates the results of Ferguson *et al.*, keeping the same oil film thickness, then the change of oil grade or its temperature does not affect the bearing power loss.

Anyway, the tendency to decrease oil viscosity grades is observed in modern machinery—engine oils of SAE viscosity class 0W/20 [17] or 5W/30 are recommended for contemporary cars, while several years ago, 20W/40 oils were most common.

Research on the application of synthetic oils in bearings is being carried out at Lulea University. Testing at the laboratory was first conducted to analyze the influence of various types of oils, including standard mineral oils, synthetic ester-based oil, and rapeseed oil on film thickness and friction losses [18]. The results showed that a decrease of power loss occurs at a cost of decreasing minimum film thickness (Figures 10 and 11).

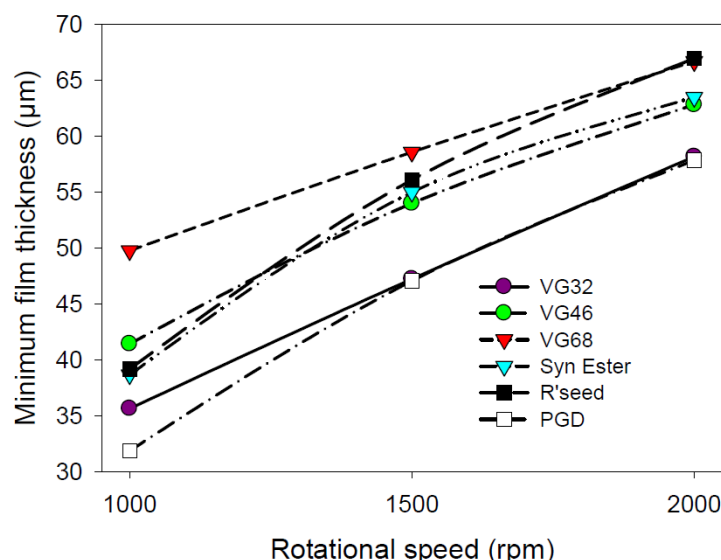


Figure 10. Minimum film thickness as a function of rotational speed of a journal bearing lubricated with different oils [18].

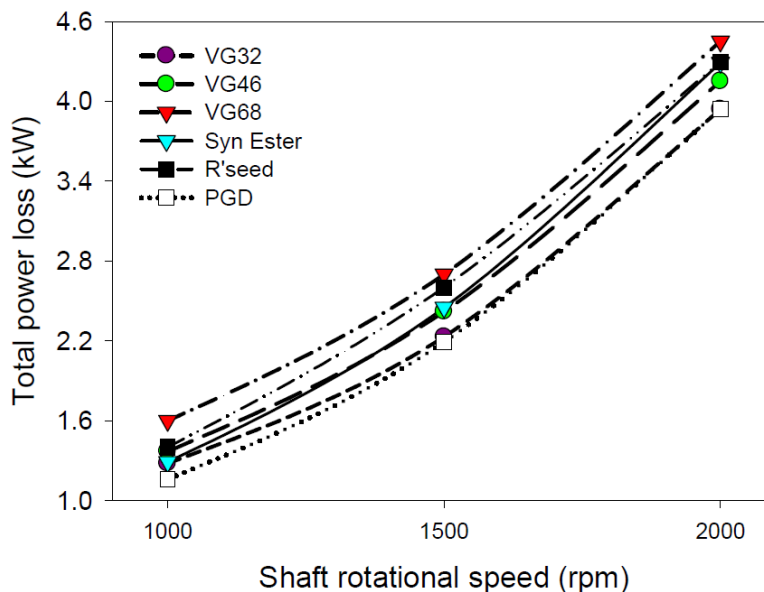


Figure 11. Total power loss as a function of rotational speed of a journal bearing lubricated with different oils [18].

Apart from laboratory measurements at the test rigs, the research was comprised of field tests. In a real 10 MW water turbine installed in Porjus power plant [19], ISO VG 68 mineral oil was replaced with a synthetic oil of VG 46 (*i.e.*, kinematic viscosity of 46 mm²/s at 40 °C) but with a viscosity index equal to 150 (as compared to approximately 100 for mineral oil). The viscosity of the tested oils at 40 °C and 100 °C is presented in Table 2, and one can observe that, while both the oils have similar viscosity at 100 °C, the synthetic oil is much less viscous at 40 °C.

Table 2. Specifications of mineral oil and high VI oil based on synthetic esters.

Parameter	TurbWay SE	ISO VG 68
Viscosity at 40 °C (mm ² /s)	46	69
Viscosity at 100 °C (mm ² /s)	8.1	8.8
Viscosity index	149	98

According to the results presented in [19], in winter months, the oil bath temperature was equal to 24 °C, and the bearing temperature to 84 °C, while in the summer, it was respectively 35 °C and 88 °C. Oil with a different VI made it possible to decrease the maximum temperature of the bearings by 7 °C (with a thicker film) but, on the other hand, oil bath temperature was 5 °C higher. As a result, the total friction loss in the journal bearing was decreased by 18.5%, and the gain in power was assessed as 30 kW, which increases the machine efficiency by 0.3%.

3.3. Contribution of Polymer Lined Bearings

Polytetrafluoroethylene (PTFE) lined bearings used in water power plants in Russia in the 1970s were extensively tested in field tests for the first time in the world [20–23]. Among the results of these applications was the possibility to use bearings with much higher specific loads—even up to 10 MPa, which in the case of special designs, enables the designers to decrease bearing size while also



considerably decreasing friction losses and torque at the same time. An example of a thrust test bearing operating for 20 years with no severe wear and tear at a specific load exceeding 11 MPa was given by Glavatskih [24]. Due to the possibility of a substantial increase with a specific load, pad width was decreased from 525 to 350 mm and mean radius was decreased from 980 mm to 938.5 mm, which contributed to the overall decrease of friction losses. While it seems to have been quite common in the former USSR to build special polymer lined bearings, in the western world, the introduction of PTFE or polyether ether ketone (PEEK) lining was done without changes in the geometry of the bearings [25] and [26], so the effect on power losses was only possible if the bearing operated at higher temperatures. This possibility for reducing losses was tested in a real power plant. In the tests, an increase of oil bath temperature of up to 60 °C allowed for a decrease in losses of 20%–30% [21]. In a recent paper by Pajączkowski *et al.* [27], a modernization of large thrust bearings with the use of Polypad™ lining at Itaipu Power plant is proposed (Figure 12). In this proposition, the outer bearing diameter is decreased from 5.2 m to 4.25 m and, according to the calculations, the total power loss should decrease by 44% from 890 kW to 502 kW.

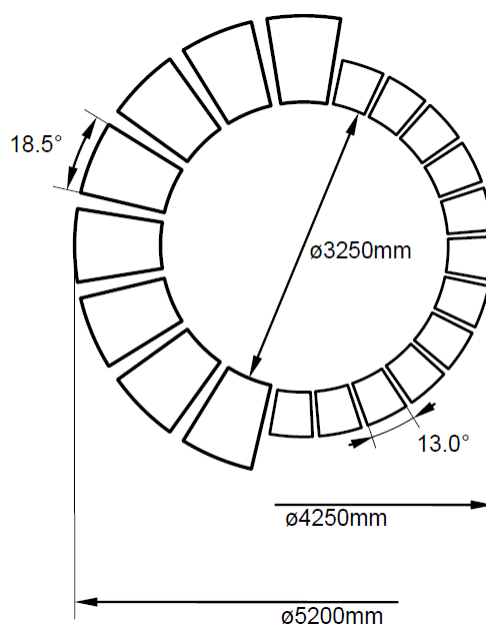


Figure 12. Comparison of the dimensions of an original thrust bearing of Itaipu power plant hydrogenerators with pads lined with Babbitt (**Left**) and the proposed modified bearing with the pads covered with Polypad™ lining (**Right**) [27].

3.4. Hydrostatic Bearings

Even though the losses are low in comparison to machine power, they should be considered in the energy balance of the whole machine because, due to machine and bearing size, the absolute value of losses is considerable. In the technical specifications of the offers, hydrogenerator manufacturers have to specify friction losses in the bearings, and they are verified during machine commissioning. An example of one of the Polish power plants shows that 0.3% of machine loss of efficiency is caused by bearing losses. The measured value of losses in the upper thrust/radial bearing setup was equal to 250–300 kW depending on oil bath temperature. Additionally, in the hydrogenerator there are two radial

bearings in which energy loss was calculated to be equal to 50–60 kW per bearing, so that the total energy dissipated in the bearings is $250\text{--}300\text{ kW} + 2 \times 50\text{--}60\text{ kW} = 350\text{--}420\text{ kW}$. The losses of 350–420 kW compared to 125 MW of the machine output decrease machine efficiency by approximately 0.3% [28].

Thrust bearings of vertical shaft hydrogenerators are normally built as hydrodynamic bearings with hydrostatic jacking systems activated only in the periods of start-ups and shutdowns. Therefore, the hydrostatic pockets installed in the faces of the pads are small and shallow, so as not to affect the hydrodynamic pressure generation potential. A hydrostatic bearing can be considered as an alternative to the common design, and sometimes as a means of solving bearing problems, hydrostatic systems are also used in normal bearing operations [29]. On the other hand, a hydrodynamic bearing used as a hydrostatic (hybrid) bearing is far from optimum. A hydrostatic bearing optimal from the point of view of energy consumption (Figure 13) should have a large pocket. The size of the pocket (b), according to the theory [30], should be equal to 60% of the pad width (B). Such a bearing is best in terms of energy consumption for pumping. A relatively small area outside the pocket is a benefit from the point of view of friction losses, because the shearing of a thin fluid film occurs only in this area, while the shearing gradient in the thick film area of the pocket is much smaller.

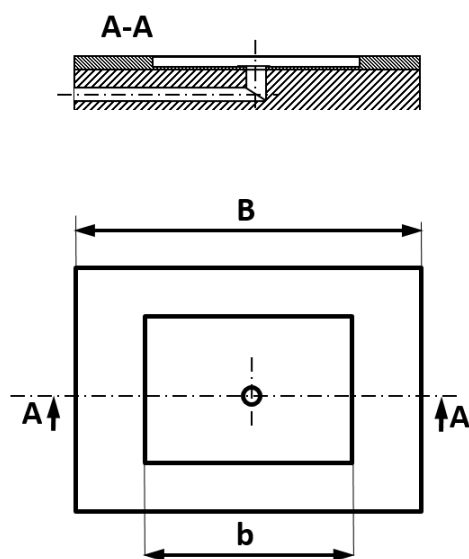


Figure 13. Rectangular section of a hydrostatic bearing with an optimum size pocket.

Calculations were carried out for such a hydrostatic (hybrid) bearing applied in the machines of Porabka Żar power plant [28] with the following assumptions: OD = 1.3 m, ID = 0.85 m, rotational speed 600 rpm, bearing axial load 2.25 MN, lubricating oil ISO VG-46 at constant temperature of 60 °C, film thickness of 40 μm (at present, according to measurements [31], the thickness in hydrodynamic regime hardly exceeds 20 μm). The calculations were performed assuming that the bearing will consist of 6, 8, 10 or 12 pads. The results in Table 3 show outstanding performance of such a bearing. Depending on the number of pads, the friction loss of the bearing can be even four times smaller than that of the hydrodynamic bearing, for which operational data are given as a reference (Figure 14). In the most feasible version with 10 pads, the pressure in the pocket is relatively small, the necessary oil flow is 42.7 L/min, the power consumption for pumping is only 7.3 kW, and the total loss

is about half of the present losses in the hydrodynamic bearing. These results illustrate the fact that a substantial decrease of friction loss, due to a thicker film (in which shearing gradient is decreased) and a smaller thin film area can be obtained at a relatively small cost of energy used for pumping.

Table 3. Simulation of the performance of hydrostatic thrust bearings for Porąbka-Żar power plant.

Case	Number of Pads	Oil Viscosity (Ns/m ²)	Pump Output (L/min)	Film Thickness (µm)	Pocket Pressure (MPa)	Pumping Power (kW)	Film Friction Loss (kW)
HS 6	6	0.018	42.7	40	22.2	12.1	70.9
HS 8	8				16.6	9.0	95.5
HS 10	10				13.3	7.3	118.1
HS 12	12				11.1	6.1	141.7
HD	16				Hydrodynamic bearing with measured power loss of 250 kW, given for reference		

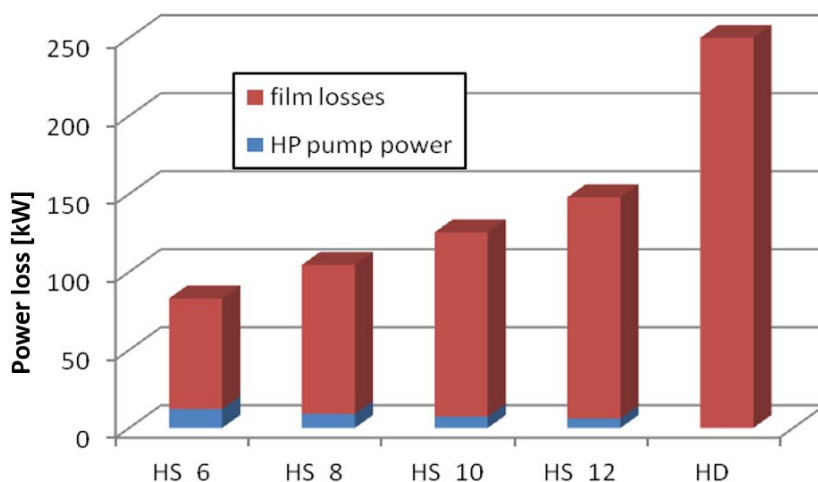


Figure 14. Comparison of power losses in hydrostatic bearings of various pad numbers and in the real hydrodynamic bearing with 16 pads shown for reference.

3.5. Water Lubrication

Bearing in mind very low viscosity of water and the environmental benefits of its use, instead of lubricating oils, one may also study the potential of water lubricated bearings. They are commonly used in the marine industry for propeller shafts [32,33] and research and applications for journal bearings of horizontal and vertical shaft water turbines are also known [34], but this refers mainly to smaller machines. Most probably, due to high loads and the consequences of failures, the application of water lubrication to large thrust bearings of vertical shaft hydrogenerators has not been realized in practice, according to the author's knowledge. Theoretical calculations, however, show that the energy savings could be even larger than in the case of oil lubricated hydrostatic bearings. A similar calculation as above, carried out for a water lubricated HS bearing (assuming OD = 1.3 m, ID = 0.85 m, rotational speed 600 rpm, bearing axial load 2.25 MN, film thickness of 20 µm and water lubrication at the temperature of 40 °C) shows that the total power loss would drop to 34 kW (seven times lower than in the existing HD bearing)—power loss will be a sum of film losses of 14 kW and pumping power of



20 kW. Such a bearing would require a water pump capable of delivering approximately 150 L/min at a pressure of 10 MPa. The reliability of such a pump is of vital importance to the reliability of the system.

4. Discussion and Conclusions

In Table 4, various methods of decreasing losses in the bearings presented in this paper are gathered and the benefits resulting from their introduction are commented on, according to the research results of various researchers.

Table 4. Methods of decreasing power loss in hydrodynamic thrust bearings.

Method of Friction Loss Reduction	Effects	Remarks
Decrease of viscosity—standard oils	Confirmed experimentally in large bearings, even a three-fold decrease of power loss, but at the cost of decreased film thickness	Decrease of the margin of safety
No-bath lubrication	Substantial decrease of power loss confirmed experimentally in tests in small bearings Smaller effects in large bearings (approx. 10%) checked only by calculations	New ideas required to provide safety in case of failures
Decrease of viscosity—high VI oils	In large journal bearings, decrease of power loss by almost 20%. A reduction by 13% in thrust bearings checked by calculations only	Oil price is the only barrier to extend the application, no other changes in bearings necessary
Polymer lining	20%–30% savings possible just by increase of oil bath temperature, 40%–50% savings possible in modified bearings of higher specific loads and decreased dimensions	Increase of oil temperature may affect its durability
Special hydrostatic bearings	In field tests, considerable decrease of temperature and increase in film thickness observed. Benefits of up to 50% were checked by calculations but require special design hydrostatic bearings	Not often used. New bearing designs and hydraulic systems necessary. Methods of providing safety in case of failures
Water lubrication	6–7 times reduction of losses is theoretically possible.	hydrodynamic or hydrostatic bearings—substantial development required, but the benefits may be remarkable

After pointing out some of the methods, some conclusions can be drawn. First, it seems that the use of high VI lubricants offers moderate but immediate benefits and does not require any changes in the bearing design. Larger benefits can be expected if specially designed hydrostatic bearings were introduced, but this requires further research on the methods of preventing the consequences of lubricating systems failures—fail safe properties must be found. Similar problems, *i.e.*, the need for bearing re-design and further research, is required if no-bath lubrication is to be introduced. Water lubrication offers higher potential benefits, but the research problems are also larger and include development of appropriate pumps. Polymer lined bearings can contribute much to energy savings,

provided that retrofits are designed to carry higher specific loads, which enables a decrease in diameter.

Acknowledgements

The author would like to thank all his Colleagues participating in various researches described in this review paper, especially doctors Leszek Dąbrowski, Michał Wodtke, Piotr Pajęczkowski and Grzegorz Rotta.

Conflicts of Interest

The author declares no conflict of interest.

References

1. Simmons, J.E.L.; Advani, S.D. Michell and the development of tilting pad bearings. In *Fluid Film Lubrication—Osborne Reynolds Centenary*; Elsevier: Amsterdam, The Netherlands, 1987.
2. The Kingsbury bearing at Holtwood. In *ASME Identification Number HH 0587*; ASME: New York, NY, USA, 1987.
3. Kingsbury, A. Thrust Bearing. U.S. Patent 1102276, 7 July 1914.
4. Wasilczuk, M.; Wodtke, M.; Dąbrowski, L. Large Hydrodynamic Thrust Bearings and Their Application in Hydrogenerators. In *Encyclopedia of Tribology*; Jane Wang, Q.J., Chung, Y.W., Eds.; Springer Science+Business Media: New York, NY, USA, 2013; pp. 1912–1926.
5. Leopard, A.J. Tilting Pad Bearings—Limits of Operation. *Lubr. Eng.* **1975**, *32*, 637–644.
6. Stryczek, S. *Napęd Hydrostatyczny (Hydrostatic Drive Systems)*; WNT: Warsaw, Poland, 1984; p. 454.
7. Dąbrowski, L.; Wasilczuk, M.; Lewandowski, S. Układy chłodzenia oleju smarującego łożyska wzdłużne turbin wodnych (Cooling systems of hydrogenerator thrust bearings—in Polish). *Tribologia* **2002**, *33*, 201–213.
8. *Itaipu hydroelectric project. Generation equipment*. ALSTOM Hydro: Baden, Switzerland; pp. 69–74.
9. Dąbrowski, L.; Wasilczuk, M.; Lewandowski, S. Badania eksploatacyjne łożyska wzdłużnego pompoturbiny—analiza wpływu czasu pracy pompy hydrostatycznego smarowania na warunki pracy łożysk (Field tests of a thrust bearing of a pump-turbine—Analysis of the time of hydrostatic pump operation on bearing properties—in Polish). *Tribologia* **2002**, *181*, 185–200.
10. Rotta, G.; Wasilczuk, M. Modeling lubricant flow between thrust-bearing pads. *Tribol. Int.* **2008**, *41*, 908–913.
11. Rotta, G.; Wasilczuk, M. Obliczeniowa analiza wybranych systemów dostarczania smaru w hydrodynamicznych łożyskach wzdłużnych (Calculation analysis of systems of oil supply in hydrodynamic thrust bearings). *Tribologia Teoria i Praktyka*. **2009**, 161–169.
12. Wasilczuk, M.; Rotta, G. On the Possibilities of Decreasing Power Loss in Large Tilting Pad Thrust Bearings. *ISRN Tribol.* **2013**, doi:10.5402/2013/732790.

13. Mikula, A.M. Further Test Results of the Leading-Edge-Groove (LEG) Tilting Pad Thrust Bearing. *Trans. ASME J. Tribol.* **1988**, *110*, 174–180.
14. Bielec, M.K.; Leopard, A.J. Tilting Pad Thrust Bearings: Factors Affecting Performance and Improvements with Directed Lubrication (Paper 13-AT 341/70). In Proceedings of Tribology Convention, Brighton, Great Britain, 27–29 May 1970.
15. Mikula, A.M. The Leading-Edge-Groove Tilting-Pad Thrust Bearing: Recent Developments. *Trans. ASME J. Tribol.* **1985**, *107*, 423–430.
16. Ferguson, J.H.; Yuan, J.H.; Medley, J.B. Spring-supported thrust bearings for hydroelectric generators: Influence of oil viscosity on power loss. In *Leeds-Lyon Symposium on Tribology*; Dowson, D., Ed.; Tribology Series; Elsevier: London, UK, 1997; pp. 187–194.
17. Tanaka, H. The Effect of 0W-20 Low Viscosity Oil on Fuel Economy. *Tech. Paper* **1999**, doi:10.4271/1999-01-3468.
18. McCarthy, D.M.C.; Glavatskih, S.B.; Byheden, A. Influence of oil type on the performance characteristics of a two axial groove journal bearing. *Lubr. Sci.* **2009**, *21*, 366–377.
19. Simmons, G.F.; Kuznetsov, E.A.; Glavatskih, S.B. Powerplant lubricant selection for improved efficiency and environmental impact reduction. In Proceedings of the ASME 2010 International Mechanical Engineering Congress & Exposition IMECE 2010, Vancouver, BC, Canada, 12–18 November 2010.
20. Aleksandrow, A.E.; Płatonow, N.G. Naturnyje issledowanija raboty podpiatnikow s EMP segmentami pri powyszennoj tiemperaturie masła (Experimental testing of thrust bearings with composite lined pads at elevated oil bath temperature—In Russian). *Gidrotechniczeskoje stroitelstwo*, **1990**, *11*, 20–24.
21. Aleksandrow, A.E. Primienienije w podpiatnikach gidrogeneratorow elasticznych metaloplastmasowych segmentow s ftoroplastowym pokryciem powierzchni trenja (Substitution of the sliding layer of a hydrogenerator thrust bearing pad with PTFE composite material—In Russian). *Gidroelektrostrojenije* **1981**, *9*, 12–14.
22. Baibrodow, J.I.; Romanov, A.A. Opyt ekspluatacji elasticznych metaloplastmasowych segmentow w podpiatnikie gidrogeneratora No 9 Wolzhskoj GES. (Experience of exploitation of elastic metal-polymer lined pads in a thrust bearing of the unit no. 9 at V. I. Lenin Volzhskaya hydro-electric station—In Russian). *Gidrotechniczeskoje stroitelstwo* **1977**, *10*, 28–31.
23. Baibrodow, J.I.; Tiereshtshenko, A.W.; Aleksandrow, A.E. Rezultaty naturnych ispytanij podpiatnika gidroagregata Bratskoj GES s elasticznymi metaloplastmasowymi segmentami (Results of field testing of a thrust bearing of Bratska HEPP with elastic metal-polymer lined pads—In Russian). *Gidrotechniczeskoje stroitelstwo* **1982**, *6*, 42–45.
24. Glavatskih, S.B. Extending performance limits of tilt pad thrust bearings: A full scale study. In Proceedings of the 7th EDF/LMS Poitiers Workshop. Operational limits of Bearings: Improving of Performance through Modelling and Experimentation, Futuroscope, France, 2 October 2008.

25. Schubert, A.; Brescianini, T. Application of a PEEK coated thrust bearing on the occasion of a refurbishment of a large hydro power plant with concurrent load increase. In Proceedings of the 10th EDF/Pprime Poitiers Workshop: Condition Monitoring, Performance Improvement and Safe Operation of Bearings, Futuroscope, France, 6–7 October 2011.
26. Knox, R.T.; Moss, W.O. The development of a PTFE faced thrust bearing for Dinorwig pumped storage power station. In Proceedings of the Hydraulic Machinery and Systems 20th IAHR Symposium, Hydro Technology, and Environment for the New Century, Charlotte, NC, USA, 2000.
27. Pajaczkowski, P.; Spiridon, M.; Schubert, A.; Brito, G.C.; Marra, J.M. Itaipu Binacional hydro power plant thrust bearing design optimization for higher efficiency. In Proceedings of 2nd International Brazilian Conference on Tribology, Foz da Igacu, Parana, Brazil, 3–5 October 2014.
28. Wasilczuk, M. *Wielkogabarytowe hydrodynamiczne łożyska wzdużne—(Large hydrodynamic thrust bearings—In Polish)*; Wydawnictwo ITEE: Radom, Poland, 2012; p. 151.
29. Abramovitz, S. Using Hydrostatic Bearings as “Lifts” in Hydroturbines. *Hydro Rev.* **2000**, *19*, 38–47.
30. Rippel, H. *Projektirowanije gidrostaticzeskich podszipnikow (Design of hydrostatic bearings—In Russian translated from English)*; Izdatielstwo Maszinstrojenie: Moscow, Soviet Union, 1967; p. 16.
31. Dąbrowski, L.; Wasilczuk, M. Influence of hydrostatic pump operation period on performance of a thrust bearing of a 125 MW pump-turbine. *Mecanique Ind.* **2004**, *5*, 3–9.
32. Litwin, W.; Olszewski, A. Properties Comparison of Two Water Lubricated Rubber Bearings Based on Experimental Test. In Proceedings of the ASME/STLE 2012 International Joint Tribology Conference, Denver, Colorado, USA, 7–10 October, 2012.
33. Litwin, W.; Olszewski, A. Water-Lubricated Sintered Bronze Journal Bearings—Theoretical and Experimental Research. *Tribol. Trans.* **2013**, *57*, 114–122, doi:10.1080/10402004.2013.856980.
34. Litwin, W.; Wodtke, M.; Olszewski, A. Shaft Misalignment Influence On Water Lubricated Turbine Sliding Bearing With Various Bush Module Of Elasticity. *Key Eng. Mater.* **2012**, *490*, 128–134, doi:10.4028/www.scientific.net/KEM.490.128.

© 2015 by the authors; licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution license (<http://creativecommons.org/licenses/by/4.0/>).

