Thrust block performance improvements using polymer lined tilting pads

J. Butler* MEng AMIMechE

** Michell Bearings Limited, UK*
* Corresponding Author. Email: john.butler@michellbearings.com

Synopsis

At the beginning of the twentieth century, the introduction of A.G.M. Michell's tilting pad bearing solution revolutionised naval thrust block design and performance. Even with today's modern designs, the fundamental concept remains unchanged, a testament to the relative simplicity and reliability of his elegant solution. The tilting pad bearing solved the inherent limitations, problems and power losses associated with the preceding multi-collar fixed geometry designs.

One aspect of thrust block design that continues universally today is the use of tin-based whitemetal (Babbitt) as the lining material of choice for bearing tilting pads. While whitemetal has proven successful since its conception in 1839, its physical characteristics introduce limitations on naval thrust block design in terms of size and performance. In recent years, interest in, and research into, polymer lined tilting pad bearings has increased as opportunities for improved performance and efficiency continue to be sought. Polymers such as PTFE (polytetrafluoroethylene) and PEEK (Polyetheretherketone) have been shown to provide much improved specific load capability and reduced coefficients of friction compared to whitemetal.

This paper reviews the potential benefits of using polymer lined tilting pads in naval thrust block designs, providing reference to previous experimental work. A typical generic operating envelope and shaft diameter are then used to develop thrust block designs as a case study covering use of different pad lining materials.

By utilising the advantages of polymer lined tilting pads a significant improvement in bearing performance, propulsion system efficiency and signature reduction is shown resulting in reduced thrust block size, weight, power loss and breakout torque; improved minimum continuous speed capability and the lack of requirement for high pressure oil injection jacking systems.

Keywords: hydrodynamic bearings, thrust bearings, thrust blocks, marine propulsion, polymers, PTFE, PEEK

1. Introduction

1.1. The naval thrust block

Marine vessels having propulsion delivered by conventional propeller shafts require a thrust bearing to absorb the propulsive thrust generated. Up until the turn of the twentieth century, shaftline main thrust bearings (thrust blocks) were predominantly of the fixed geometry design with a series of thrust collars and counter-surfaces. The multi-collar bearings were heavy and generally troublesome in operation as the plain, fixed geometry faces were designed with little knowledge of the principles of hydrodynamic lubrication. A.G.M. Michell's development of the tilting pad thrust bearing provided a significant improvement upon the *status quo* resulting in a rapid switch away from the previous multi-collar designs (Simmons & Advani, 1987). The efficiency of the tilting-pad bearing is illustrated by the comparison shown in Figure 1, which demonstrates the large savings in space, weight and auxiliary equipment.

Whilst the change of the elements carrying thrust load was rapidly accepted due to the considerable savings and reliability improvements, the materials of construction of these thrust blocks were not subject to the same revolution. Then, as now, the stationary parts of the bearing were faced with tin-based whitemetal (Babbitt) or occasionally made from varied bronzes. The rotating shafts and collars were steel or iron. The lubricants were mineral oils of various viscosity ratings. These substances remain the materials of choice for the vast majority of hydrodynamic bearings produced today.

Author's Biography
John Butler is Design Office Manager at Michell Bearings, overseeing the detail design engineering of new product. He holds a degree in Mechanical Engineering from Durham University and joined Michell Bearings as a graduate in 2016. His research interests relate to the improvement of hydrodynamic bearing designs by inclusion of novel features and materials.

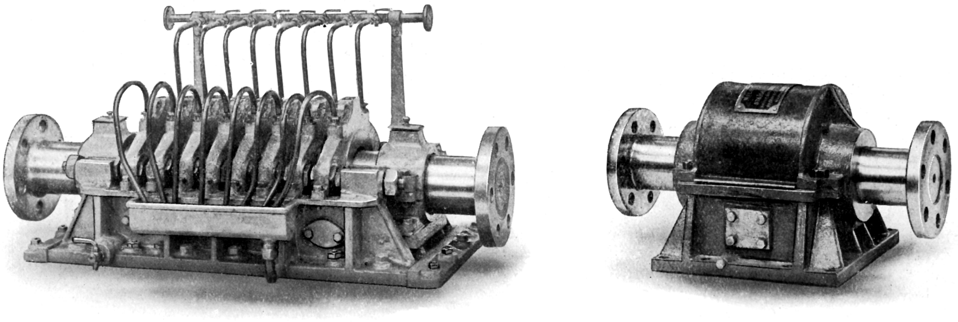

Figure 1 Comparison, to scale, of a multi-collar thrust bearing (left) and the equivalent tilting pad thrust block (right), as illustrated in *The Michell Bearing Book*

1.2. Materials and progress

Whitemetal (Babbitt) has been the predominant facing material for industrial and marine hydrodynamic bearings since the introduction of the tilting pad bearing. It offers advantages of dimensional stability, ease of repair, good compatibility with commonly used lubricants and shaft counter-surfaces, and the ability to absorb or embed particles from the lubricant which may otherwise cause damage.

However, it is a material with a low melting point and poor fatigue strength, both of which place limitations on the operating envelope of bearings using whitemetal as a facing material. These limitations have historically been based on limits of experience from successful operation of similar bearings. There have been many publications over the years attempting to rationalise the many 'rules of thumb', including the particularly comprehensive examples of Leopard (1976) and Martin (1969).

More recently, Ettles et al. (2005) have considered the localised stress state in the whitemetal surface along with appropriate temperature-based material properties to calculate a local bearing factor of safety index against the inherent material limitations of the common whitemetal lining.

In the later part of the twentieth century, it was recognised that the facing material of thrust bearing pads represents a limitation on the operational envelope of bearings for highly fatigue-loaded bearings (Pratt, 1969) and low speed, high load bearings (Baiborodov, et al., 1977). Regardless, the operational duties and high emphasis on reliability and conservative design has meant that development of new bearing materials has not generally been observed in actual use for marine and naval propulsion applications.

1.3. Present uptake in the market

Despite a lack of uptake in marine applications, polymer-faced hydrodynamic bearings, and thrust pads faced with polytetrafluoroethylene (PTFE) in particular, have been specified extensively in the field of vertical axis hydroelectric generators. Such hydrogenerators support very high thrust loads (the largest have loads in excess of 46000 kN) (Ferguson, 1999). At large thrust pad sizes, the effect of thermal and mechanical deformations are great (Ettles, 1980). Such deformations are significant for bearings designed to work with minimum film thicknesses of the order of 0.025 to 0.050 mm, and consequently many mechanisms of varying sophistication were developed to control the deflection of large thrust pads (Baudry, et al., 1959).

PTFE lining of tilting pad thrust bearings was developed in Russia during the 1960s and 1970s, with the intention of improving the reliability of existing and new hydrogenerators. By 1990 the vast majority of hydroelectric power stations in Russia were fitted with PTFE-faced thrust bearings. Since then, there has been a significant amount of work conducted to understand the performance parameters of PTFE thrust pads. Replacement of whitemetal with PTFE is a common upgrade for existing hydrogenerators, supported by multiple generator Original Equipment Manufacturers and plant operators performing renewal work (Mohino, et al., 2002).

One exception to the blanket statement regarding a lack of uptake of polymer bearings within the marine and naval sectors was a bearing described by Knox and Simmons (2006). This vessel was propelled by a pair of large water jet propulsors and weight savings were of very high importance. Use of PTFE faced thrust pads for the thrust faces resulted in a saving of vessel mass of 4000 kg – a reduction to approximately 4800 kg dry weight per thrust block.

2. Potential benefits of polymer bearing linings

In order to enumerate the differences between polymer bearing linings and traditional whitemetals, a series of bearing performance ‘limitations’ are listed and the polymer option is compared to the baseline, traditional whitemetal faced design.

2.1. Minimum film thickness

Hydrodynamic lubrication is defined by the presence of a fully-formed film of lubricant in the bearing gap. Conventional bearing design is based upon working to a limiting value of film thickness which, while related to the fundamental tribological factors of surface roughness, is generally a value chosen from experience. As an example, Martin (1969) encompasses factors of pad surface roughness, uneven load sharing between pads, and dishing of pad support structures to arrive at a minimum acceptable calculated oil film thickness.

Whilst the limiting film thickness value for acceptable hydrodynamic lubrication may not vary between facing materials, polymer-faced bearings, and particularly PTFE, exhibit improved performance in the boundary lubrication regime encountered at very low speeds. Typical bearing polymers, PTFE and PEEK, both exhibit lower static friction when compared to a reference steel-on-whitemetal contact.

The reduced friction coefficient offers benefits of reduced start-up torque. In addition, the lower coefficient of friction during boundary lubrication results in less wear and heat generation during these situations, resulting in a more robust bearing.

2.2. Maximum surface temperature

As mentioned in section 1.2, the existing temperature limitations for whitemetal bearings are to an extent empirical and derived from experience. Temperature limits of anywhere between 70 and 130 °C may be specified by machine designers or design standards, based on the risk appetite of the end user and historic experience in the field. Ettles’ proposal (2005) for deriving operational limitations from fundamental material properties does generally validate the upper limits of these established rules.

Because of low thermal conductivity compared to metallic materials, the polymer facings employed on thrust and journal pads result in steep temperature gradients across the thickness of the polymer layer. Direct measurement of the polymer temperature is difficult even in laboratory conditions due to the scales involved (Zhou, 2016). Where extensive tests have been carried out on PTFE-faced bearings, the operating limitations for oil film temperature (and hence polymer surface temperature) have found to be similar to those of whitemetal bearings with an upper limit to the tested operating envelope up to 7 MPa of approximately 130 °C reported by Dixon et. al (2016).

Limiting temperatures for PTFE and PEEK can be related to local conditions on the pad face approaching and exceeding the glass transition temperature of the polymer material, and due to increased susceptibility to creep under long-term compressive stress.

2.3. Peak lubricant film pressure

The mechanical strength of the bearing facing material places limitations on the operation of tilting pad bearings. Tin-based whitemetals exhibit considerable reduction in yield and fatigue strengths as temperature increases, even far below the theoretical ‘melting’ temperature.

The computational evaluation of bearing performance for metallic facings is well-understood, and the methods used across the industry generally treat the pad facing and backing materials as linear elastic. By contrast, the behaviour of PTFE under a load is non-linear, and also the elastic modulus of the material varies considerably across the typical working temperature range of industrial bearings. Calculation methods for PTFE-faced bearings have been created which make use of both temperature-independent linear (Glavatskih & Fillon, 2006) or temperature-dependent properties (Wodtke & Wasilczuk, 2013), or more advanced analyses incorporating material non-linearity as well as temperature dependence (Ettles, et al., 2003).

When bearings are analysed using methods which take into account the compliance of the PTFE face, it is found that the ‘peak’ of the oil film pressure profile is flatter and more rounded, making more efficient use of the bearing surface area for a given mean bearing pressure. Wodtke (2013) showed a reduction in peak film pressure with PTFE having a peak of 82% of the baseline whitemetal-faced pad. Ettles (2003) reported a ratio of approximately 75% between computed peak pressure for a pair of thrust pads differing in their facing material.

Reduction in the peak oil film pressure means that a bearing will be working further away from the mechanical limits of the facing material, and leads to the opportunity for savings in material and power losses by reducing bearing surface areas.

2.4. Size limitations – bearing deflection

Hydrodynamic bearings, particularly thrust bearings, are subject to size effects, principally arising from thermal deformations and which start to become significant at pad radial widths of approximately 254 mm and above (Ettles, 1980). These effects can be controlled to a certain extent by using a larger number of smaller bearing pads to carry the same load, and by use of complex support systems designed to compensate thermal- and pressure-induced distortions.

The optimal profile for the running face of thrust bearings is close to flat, or with a convex profile in the same order of magnitude as the minimum lubricant film thickness (Raimondi, 1960). The load carrying capacity of bearings is significantly reduced in the presence of crowning in excess of this, and therefore techniques to reduce any crowning can be especially beneficial.

Polymer facing materials have much lower thermal conductivity than metals. Because the hot oil film is insulated from the bearing pad backing material, the magnitude of the temperature differential through the thickness of the bearing pad is significantly reduced, and consequently the magnitude of thermal bending is much smaller. This allows for simpler support systems to be used, or for the retrofit of bearings with deficiencies in support design.

2.5. Misalignment and geometry error

Polymer hydrodynamic bearing linings, as discussed in section 2.3, are inherently more compliant than metallic surfaces. This results in a much improved ability to compensate for relative misalignment or errors in manufacture of bearing components.

One interesting example which demonstrates this capability is the application of PTFE-faced thrust pads to the Thissavros pumped-storage hydro plant described by Knox (2006). The support system for the bearing runner face was inadequate and resulted in a calculated difference in surface height of 0.226 mm radially across the thrust face of the runner. A generally accepted rule of thumb is that whitemetal bearing performance becomes seriously degraded at levels of misalignment approaching a unity ratio compared to the calculated minimum film thickness of the bearing. In this case the predicted minimum film thickness for the whitemetal pad would have been in the region of 0.040–0.050 mm. Whilst such a film thickness would usually be considered comfortable, it is clearly inadequate when compared to the predicted runner face deflection and failure of the whitemetal pads resulted.

Replacement PTFE-faced thrust pads were fitted to a machine without removing the underlying problem. When inspected after 570 hours of operation, there was no visible wear to the PTFE face in spite of the adverse duty conditions.

2.6. Electrical currents

Hydrodynamic bearings are frequently used as part of or in the vicinity of electrical machinery which can generate differences in potential between shafts and foundations. Bearing failures have frequently been noted which arise from this phenomenon. The mechanism of failure is from deterioration of the bearing working surfaces by pitting caused by electrical arcing across the thickness of the lubricant film (von Kaehne, 1964). Remedies to this phenomenon are well known, either providing an alternative current flow path to ground which bypasses the bearings; or providing electrical insulation to the bearings themselves. Both options represent extra expense, and if a bearing assembly must be insulated at its foundations then additionally all instrumentation and services must be designed to maintain the integrity of the insulation.

Polymeric bearing linings are generally electrically insulating so long as the formulations are not ‘filled’ with conductive components such as graphite. If desired, insulating lining materials can be used for all bearing working surfaces to provide full protection from electrical discharge across the lubricant film.

As electrical propulsion systems in surface ships and submarines become more commonplace and increasingly assume the role of main prime mover, systems design will have to ensure that all shaftline bearings are protected from stray currents.

2.7. Supporting equipment

Typical shaftline bearings may require several items of supporting equipment or services. These include seawater or freshwater systems to provide cooling to the bearings, and external lubricating oil circulation systems for bearings that do not have ‘self-contained’ designs. Additionally, high pressure oil injection (or ‘jacking’) systems may be employed to ensure operation of the bearings at low shaft speeds where full hydrodynamic films are not generated.

Polymer-faced bearings present opportunities to either reduce the capacity of this supporting equipment, or indeed do away with it entirely.

The increased load capacity which is possible with polymer bearing facings (through the mechanisms described in sections 2.1 to 2.3) allows for smaller bearings to be used, with commensurate reductions in power losses. This reduces the demand for cooling water and lubricating oil flow, and allows extended operation under emergency conditions where water or oil supply is disrupted.

The improved low speed and start-up performance described in section 2.1 can mean that high pressure jacking systems are often made redundant. Removal of such systems can greatly simplify overall vessel system design and operating philosophy and procedures, with effective savings in excess of the capital cost of said equipment.

3. A case study in thrust block design

3.1. Duty conditions

Whilst fundamentally performing a single task in supporting propeller (or propulsor) thrust forces, thrust blocks are subject to a number of distinct duty conditions depending on the type of vessel, with some known to be particularly challenging for traditional whitemetal-faced bearing elements.

When considering bearings fitted to the main propulsion shaft line, there is a clear distinction to be made between surface ships and submersible vessels. For surface ships, the thrust generated by the propulsor is dependent upon the shaft speed, and the thrust load with a stationary shaft is zero. This presents an ideal application for a hydrodynamic bearing (Simmons & Henderson, 1989) as the load carrying capacity of the bearing is matched with the duty to which it is subjected.

Conversely, and importantly in the field of naval propulsion, the above considerations do not apply to thrust blocks for submarines. In these applications, there is an additional component of the thrust load which is derived from hydrostatic pressure applied to the sections of the shaft external to the pressure hull, meaning that the most challenging duty condition for the thrust block becomes starting shaft rotation at full submergence depth, along with maintaining continuous low speed operation at depth. The additional hydrostatic thrust loading also means that total thrust for a given propulsive power will be higher.

Many shaftline system designs make use of a journal bearing integrated within the thrust block to support a proportion of the total shaft weight, removing the need for a separate lineshaft bearing. Both plain cylindrical bore journal bushes and tilting pad journal arrangements are commonly used.

3.2. Thrust block components and arrangement

A typical arrangement of a water cooled thrust block as supplied for conventionally powered submarines is illustrated in Figure 2. This shows a cross-section through a bearing with a 'centre flange' or saddle-type mounting arrangement where the interface with the ship structure is close to the horizontal midplane of the bearing, as opposed to the bearing forming a pedestal arrangement.

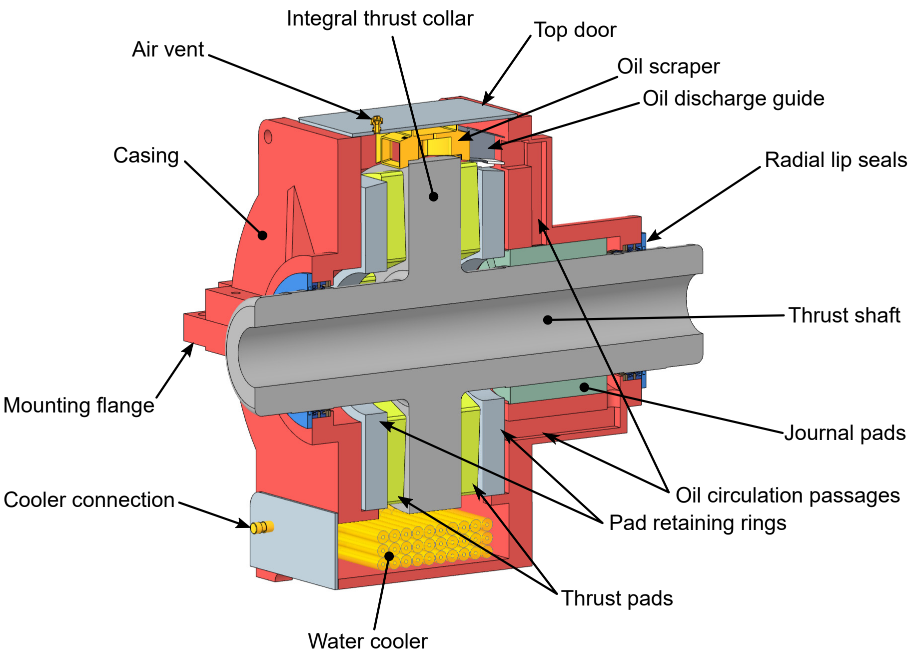

Figure 2 Annotated drawing showing components of a typical water cooled self-contained thrust block

Thrust pads are arranged ahead and astern of the thrust collar, and a series of journal pads surround the shaft on the forward side of the collar.

A cooler is provided in the bottom half of the bearing to remove heat from the oil. Oil circulation is achieved by the motion of the thrust collar and an oil scraper at the top of the bearing to divert oil to the working elements.

Owing to the requirement to start up under load, the oil level inside of the bearing is chosen to ensure that an adequate proportion of thrust pads are submerged in all possible conditions of pitch and roll.

3.3. Design study objective

To provide a quantitative example which demonstrates the possible gains listed in section 2, a representative thrust block design has been developed for the duty conditions listed in Table 1 below. The duties have been selected to represent a typical submarine with non-nuclear propulsion. In common designs such submarines do not contain main reduction gearboxes and such boats forego a main lube oil circulation system, and therefore a self-contained, water cooled bearing as described in section 3.2 is the favoured choice.

Table 1 Design study bearing duty parameters

Parameter	Value	Unit
Shaft speed (continuous)	175	rpm
Shaft speed (max. transient)	200	rpm
Shaft speed (target min. continuous)	20	rpm
Thrust load (max. continuous)	1100	kN
Thrust load (max. transient)	1253	kN
Hydrostatic thrust load	600	kN
Journal load	70	kN
Shaft diameter	355	mm
Lubrication	Bath lubricated	
Oil type	Mineral oil, ISO VG 68	
Cooling	Water cooled	
Water temperature (max.)	35	°C

The thrust loads in Table 1 are based upon an invented nominal load arising from hydrostatic effects on the shaft of 600 kN at the vessel maximum depth, which is to be considered for starting up at depth and also during continuous slow-speed running. Additionally, a 500 kN propulsive thrust load was chosen at the maximum continuous rating of the propulsion system. A power law with exponent 2 was used to relate hydrodynamic thrust load to speed and to derive a thrust load for the over-speed condition.

3.4. Design process

To carry out the design study, three potential designs were developed, as described in the following sections.

3.4.1. Baseline, whitemetal design – option 1

A design was created for the Table 1 duty conditions based upon typical conservative design practices with regard to bearing specific load values. Calculations of bearing performance were used to refine the design in terms of the required water cooling capacity and performance of the thrust block in off-design conditions such as operation at minimum speeds following sustained operation at high speeds and oil bath temperatures.

The output of the design was a thrust block with the following parameters:

Table 2 Baseline whitemetal thrust block sizes and properties

Parameter	Value	Unit
Thrust surface	0.504	m²
Journal length	355	mm
Cooling coil relative length	100	%
Cooling water flow rate	3000	L/hr
Thrust pressure (normal/max/hydrostatic)	2.18 / 2.49 / 1.19	MPa
Journal pressure	0.49	MPa
High pressure jacking	Not required	

3.4.2. High-load whitemetal design – option 2

An additional comparison design was developed which continues to make use of whitemetal-faced thrust and journal pads, however, the size of the bearing elements was reduced where possible with the intention of making the most efficient use of the traditional facing material. A specific load of 4.2 MPa was targeted at the maximum thrust load which is in line with common industrial practice (Simmons & Henderson, 1989).

Table 3 Adventurous whitemetal thrust block sizes and properties

Parameter	Value	Unit
Thrust surface	0.300	m²
Journal length	236	mm
Cooling coil relative length	69	%
Cooling water flow rate	2500	L/hr
Thrust pressure (normal/max/hydrostatic)	3.67 / 4.18 / 2.00	MPa
Journal pressure	0.84	MPa
High pressure jacking	17 L/min @ 77 bar	

However, the reduced thrust pad size on this design resulted in an acceptable minimum continuous speed of 35 rpm compared to the 20 rpm required by the case study specification. The axial length of the journal bearing was also reduced as placing the threshold of hydrodynamic operation at 35 rpm allows for a higher journal specific load.

To overcome this limitation on minimum continuous running, and to provide surety of start-up at depth, the bearing design requires high pressure jacking facilities on the ahead thrust face and journal bearing. Depending on overall propulsion system design, the auxiliary hydraulic jacking unit may require a redundant configuration.

3.4.3. PTFE-based design – option 3

Thirdly, a thrust block design was developed to make use of polymer bearing facing technology. A carbon/graphite filled grade PTFE material was chosen, as there is a background of research in the literature covering use of PTFE for duties appropriate for the application. The size of the bearing elements was reduced to make use of the increased load capacity of PTFE facings both at normal running speed and under low speed and start-up conditions. At high speed, the thrust pad loading is within experience for PTFE faced bearings operational on surface ships of 5.5 MPa (Knox & Simmons, 2006).

PTFE facing is also proposed for the journal component of the thrust block. Previous testing of oil-lubricated, PTFE-faced journal pads for marine applications by the author’s company to evaluate continuous low speed operation has validated a performance envelope of 0.13–3.9 m/s at 2.0 MPa (Dixon, et al., 2010). The proposed bearing is comfortably within this region. The reduced journal length from 3.4.2 is retained, but due to the advantages of PTFE no oil jacking is required.

Table 4 PTFE thrust block sizes and properties

Parameter	Value	Unit
Thrust surface	0.228	m²
Journal length	236	mm
Cooling coil relative length	57	%
Cooling water flow rate	2150	L/hr
Thrust pressure (normal/max/hydrostatic)	4.82 / 5.49 / 2.63	MPa
Journal pressure	0.84	MPa
High pressure jacking	Not required	

3.5. Design results and comparison

In addition to the sizes of the bearing elements and cooler, described in Tables 2 to 4, mechanical design models were generated for options 1 and 3. A size comparison illustrating the reduction in size is shown in Figure 3. The reduction in size of the block between the two options is quite apparent, even though the shaft diameter has remained constant.

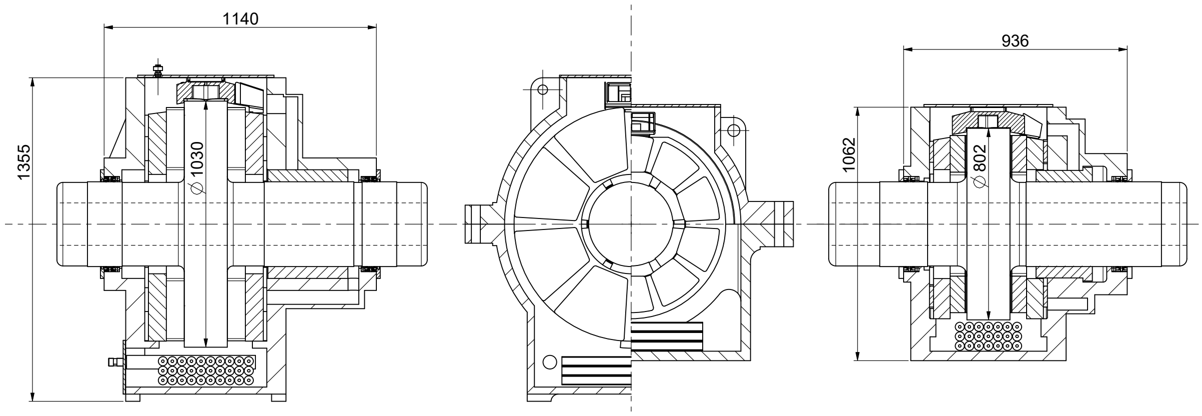

Figure 3 Comparison in size between baseline whitemetal thrust block design (left) and proposed PTFE thrust block (right)

In addition to the visual demonstration of the reduction in size and weight, quantitative data for bearing performance are shown in Table 5. These data are tabulated for both the maximum continuous rated duty and also the overspeed duty condition. A significant power loss saving can be seen, with the high load whitemetal design a reduction of 33% and the PTFE design an even larger 44% reduction in power loss at the overspeed condition. The cooling water flow rate for the PTFE bearing is 28% reduced compared to the baseline. The discrepancy is due to the more compact bearing size requiring more cooler tubes in parallel to achieve a given total cooling surface area and rating.

Table 5 Performance parameters of proposed designs

Parameter	Unit	Whitemetal design, baseline		Whitemetal design, high load		PTFE design	
		175 rpm	200 rpm	175 rpm	200 rpm	175 rpm	200 rpm
Power loss	kW	10.7	13.1	7.4	8.8	6.1	7.3
Bath temperature	°C	49.3	52.5	49.2	52	49.3	52.1
Max. thrust pad temp.	°C	64.7	69	71.2	77.6	77.1	83.8
Max. journal pad temp.	°C	55.3	58.4	58.2	61	58.3	61.1
Cooling water temperature rise	K	3.1	3.7	2.6	3.1	2.4	2.9

Table 6 presents a second set of quantitative data, this time dealing with the inherent properties of the thrust block designs not related to specific operating conditions. The minimum continuous speed allowable for the PTFE bearing design is improved even compared to the conservative whitemetal thrust block which has a total thrust surface area more than 2× larger.

The breakout torque has been calculated using a coefficient of friction reported by Dixon and Humble (2015). The values for both PTFE and whitemetal are based on a dwell time of 24 hours under load and are therefore considered to be conservative estimates with respect the possible improvement granted by a low-friction polymer facing. Such a reduction may allow for savings for emergency propulsion motors or turning devices.

The sum of the total bearing mass and mass saved from the thrust shaft between the baseline and PTFE options comes to 2896 kg versus 4110 kg, a saving of 29.5% in total.

Table 6 Additional specifications of proposed designs

Parameter	Unit	Whitemetal design, baseline	Whitemetal design, high load	PTFE design
Minimum continuous speed	rpm	18	35	11
Breakout torque (* = jacking operational)	kNm	57.6	3.0* / 50.0	23.4
Mass of bearing (dry), excluding shaft	kg	4110	-	3360
Shaft mass change	kg	0	-	–464

4. Conclusions

Use of polymer-faced tilting pads in a thrust block design typical of a conventionally propelled submarine application has been calculated to offer significant improvements. In the specific case of PTFE-faced pads, bearing performance has been improved by reducing the minimum speed capability beyond the baseline design. The overall power loss is significantly reduced, meaning that there are lower requirements upon supporting cooling plant and improving the overall efficiency of the vessel. Breakout torque is improved by incorporation of PTFE-faced bearing elements, with a reduction of 59% versus the baseline whitemetal design. Whilst the whitemetal design can be altered and supplemented by high pressure jacking to achieve intermediate weight and size benefits, the addition of such a system has potential signature implications and is not required for a polymer faced option. The bearing and shaft of the case study application are reduced by over 1200 kg in total, a considerable proportion of the baseline.

Overall, it appears that a compelling case can be made for specifying PTFE facings in naval thrust block applications. The performance of the technology is well understood and widely accepted in industrial applications, particularly hydro power. The author expects similar positives, with the exception of considerations of startup thrust load, would be seen in a study for a surface vessel.

Acknowledgements

Figure 1 is reproduced from (Dixon & Humble, 2015) with permission from Michell Bearings Limited.

References

Baiborodov, Y. I. et al., 1977. Performance of elastic steel-plastics segments in step bearing of hydrogenerator no. 9 of the Volga-Lenin power station. *Hydrotechnical Construction,* 11(10), pp. 1013-1018.

Baudry, R. A., Kuhn, E. C. & Cooper, G. D., 1959. Performance of Large-Waterwheel-Generator Pivoted-Pad Thrust Bearing Determined by Tests under Normal Operating Conditions. *Trans AIEE. Part III: Power Apparatus and Systems,* 78(4), pp. 1300-1309.

Dixon, S. J., Bruce, P. & Waggott, J. D., 2016. Developments in PTFE faced thrust bearings for usein vertical pump applications. In: *13th IMechE European Fluid Machinery Congress, 3-4 October 2016, The Hague, Netherlands.* London: IMechE.

Dixon, S. J. & Humble, G., 2015. Hydrodynamic Bearings - Robust Design Ensures Success. In: *14th EDF/Pprime Workshop: Influence of design and materials on journal and thrust bearing performance, 8-9 October 2015, Futuroscope, France.* Poitiers: University of Poitiers.

Dixon, S. J., Waggott, J. D. & Simmons, J. E. L., 2010. PTFE-Faced Journal Bearing Technology for Marine Propulsion Applications. In: *9th EDF/Pprime (LMS) Poitiers workshop: Improvement of bearing performance and evaluation of adverse conditions, 7-8 February 2010, Futuroscope, France.* Poitiers: University of Poitiers.

Ettles, C. M., Knox, R. T., Ferguson, J. H. & Horner, D., 2003. Test Results for PTFE-Faced Thrust Pads, With Direct Comparison Against Babbitt-Faced Pads and Correlation With Analysis. *Journal of Tribology,* 125(4), pp. 814-823.

Ettles, C. M. M., 1980. Size effects in tilting pad thrust bearings. *Wear,* 59(1), pp. 231-245.

Ettles, C. M., Seyler, J. & Bottenschein, M., 2005. Calculation of a Safety Margin for Hydrogenerator Thrust Bearings. *Tribology Transactions,* 48(4), pp. 450-456.

Ferguson, J., 1999. Upgrading thrust bearings for hydro machines. In: *Meeting of Canadian Electrical Association, March 1999, Vancouver.* s.l.:s.n.

Glavatskih, S. B. & Fillon, M., 2006. TEHD analysis of thrust bearings with PTFE-faced pads. *Journal of Tribology,* 128(1), pp. 49-48.

Knox, R. T., 2006. PTFE faced bearings; Thissavros - a case study. In: *Hydro 2006: maximising the benefits of hydropower, 25-27 September 2006, Port Carras, Greece.* Sutton: Aqua-Media International.

Knox, R. T. & Simmons, J. E. L., 2006. PTFE Faced Bearings for Marine Applications. In: *SNAME 11th Propeller and Shafting Symposium, 12-13 September 2006, Williamsburg, VA.* Jersey City, NJ: SNAME.

Leopard, A. J., 1976. Tilting pad bearings – limits of operation. *Lubrication Engineering,* 32(12), pp. 637-644.

Martin, F. A., 1969. Tilting pad thrust bearings: rapid design aids. *Proc. IMechE. Part 3C: Conference Proceedings,* 184(12), pp. 120-138.

Mohino, A. A., Loshkarev, V. P. & Goldenberg, S. I., 2002. Using PTFE-lined thrust bearing pads in hydro units. *Hydro Review*, June, pp. 44-51.

Pratt, G. C., 1969. New developments in bearing materials. In: *International Automotive Engineering Congress, 13-17 January 1969, Detroit, MI.* New York: SAE, p. 690112.

Raimondi, A. A., 1960. The Influence of Longitudinal and Transverse Profile on the Load Capacity of Pivoted Pad Bearings. *ASLE Transactions,* 3(2), pp. 265-276.

Simmons, J. E. L. & Advani, S. D., 1987. Michell and the development of tilting pad bearings. In: D. Dowson, C. M. Taylor, M. Godet & D. Berthe, eds. *Fluid Film Lubrication - Osborne Reynolds Centenary.* Amsterdam: Elsevier, pp. 49-56.

Simmons, J. E. L. & Henderson, N., 1989. Naval thrust bearings. In: *IMarE centenary year conference proceedings: Past present and future engineering in the Royal Navy, 6-8 September 1989, Plymouth, UK.* London: IMarE.

von Kaehne, P., 1964. *Shaft voltages and bearing currents: a survey of published work.* Leatherhead, Surrey: Electrical Research Association.

Wodtke, M. & Wasilczuk, M., 2013. Effect of Coating Material Properties on Tilting-pad Thrust Bearing Performance. In: *12th EDF/Pprime Workshop: Solutions for performance improvement and friction reduction of journal and thrust bearings, 17-18 September 2013, Futuroscope, France.* Poitiers: University of Poitiers.

Zhou, J., 2016. Temperature Monitoring of PEEK Bearings. In: *STLE Annual Meeting and Exhibition, 15-19 May 2016, Las Vegas, NV.* Park Ridge, IL: STLE.

Word count total 4382
Less tables 64 41 44 41 74 47 (311)
= 4071