Hydrodynamic Bearings - Robust Design Ensures Success
Paliers Hydrodynamiques – Une Conception Robuste est la Garantie du Succès

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When Michell and Kingsbury successfully applied their solution to Reynold's equation, the tilting pad bearing came into existence. The design was simple and elegant and offered significant performance advantages to users. Whitemetal (Babbitt), a material conceived in the 19th Century, was utilised as a sacrificial lining.

Today, more than 100 years on, the same fundamental concept appears to be going strong. Over the years, robust designs have prevailed and designers have enhanced their products and improved performance. Materials have also improved and alternative linings to whitemetal have emerged. Fluid film bearings are well established across a huge variety of applications, many with demanding operating and environmental conditions.

The authors look at some of the successful designs that have become commonplace, how designs have progressed and how lining materials have evolved. Finally, some recent bearing developments are reported and conclusions are drawn about the requirements of plain bearings in the future.

Quand Michell et Kingsbury ont réussi à appliquer avec succès leur solution à l'équation de Reynold, le palier à patins pivotants a vu le jour. La conception en était simple et élégante et offrait aux utilisateurs des avantages significatifs en termes de performances. Le métal blanc (ou Babbitt), un matériau conçu au XIXe siècle était utilisé comme revêtement sacrificiel.

De nos jours, plus de 100 ans après, le même concept fondamental n'a rien perdu de sa valeur. Au fil des années, les conceptions robustes ont prévalu et les concepteurs ont apporté des améliorations à leurs produits et à leurs performances. Les matériaux ont également été améliorés et d'autres revêtements que le métal blanc ont fait leur apparition. Les paliers à film fluide sont aujourd'hui bien établis dans une très grande variété d'applications, dont beaucoup dans des environnements et des conditions de fonctionnement particulièrement contraignants.

Les auteurs examinent quelques unes des conceptions les plus performantes devenues usuelles, comment les conceptions ont progressé et les matériaux de revêtement évolué. Enfin certains développements récents de paliers sont évoqués et des conclusions sont tirées quant aux exigences futures des paliers lisses.

1 Introduction

Albert Kingsbury and Anthony George Malden Michell are each credited independently with the invention of the tilting pad bearing. Kingsbury's US patent was granted in 1910 [1] whilst Michell's thrust bearing patent [2] was granted in 1905 and his journal bearing patent [3] in 1912. The Michell patents describe the idea of a series of segmented bearing 'blocks' each pivoted on the back so that "the film of oil between each block and the surface on which it bears may be more compressed or constricted at the rear end than at the front or leading end of the block, a condition which favours the entry of the oil between the surfaces at the leading end." Figure 1 shows an illustration from Michell's 1905 patent.

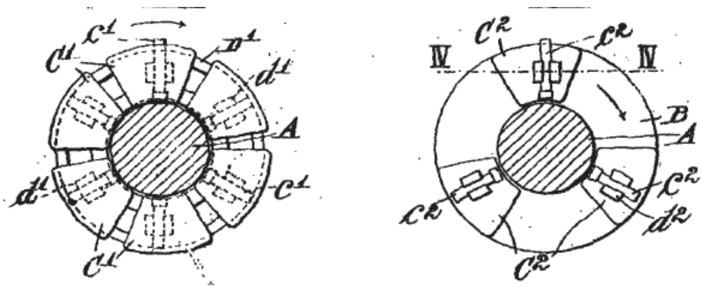
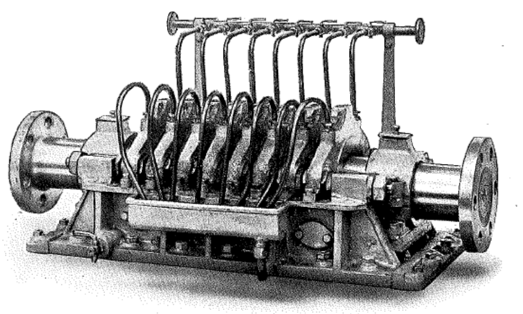


Fig. 1 – An Illustration from Michell's Patent No 875 Granted in 1905

It is not until one compares the tilting pad bearing to the technology of the time that one appreciates the significance of the invention. Prior to the tilting pad invention, bearings were characterised by unreliability and unpredictable lifespans. A very clear example of this is thrust blocks which absorb the thrust from the propeller of ocean going vessels. In the 19th century marine thrust block designs utilised multi-collar technology and, for at least one merchant liner, 22 collars were required to transmit the thrust onto the flat bronze counter-faces [4]. Balancing the load so that each thrust element carried equal load was particularly problematic and continual adjustment was required. In addition, high power losses were inherent with the multi-collar design. Figure 2 shows a photograph of a typical design and illustrates the complexity of the arrangement. In contrast, Figure 3 depicts the equivalent, circa 1916, Michell tilting pad thrust block design for the same duty. The design utilises a single collar thus reducing the complexity and lowering power losses whilst, simultaneously, allowing significantly larger loads to be transmitted. The immediate impression is that the Michell design is much simpler resulting in installation benefits and much lower maintenance requirements. It was estimated that the Royal Navy saved £500,000 GBP worth of coal in 1918 alone as a result of fitting Michell's tilting pad bearings [5].



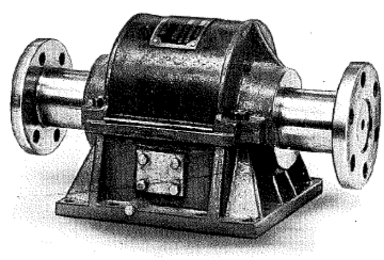


Fig 2 – Multi-Collar Thrust Bearing ***Fig 3 – Equivalent Michell Thrust Block***
source : The Michell Bearing Book, 3rd Edition, first published 1916

In a similar fashion, Kingsbury's bearing fitted to Unit 5 of the Holtwood Hydroelectric Station in 1912 was said to carry 100 times the load taken by the previously fitted roller bearings [6]. During overhaul in 1950 the bearing was reinstalled having been described as in 'perfect condition'. Clearly the tilting pad design resulted in a step change in bearing performance and reliability.

The authors' argument is that robust designs that enjoy longevity are inherently reliable and often have a deceptive simplicity about them. Both Kingsbury and Michell had been able to recognise the significance of the work undertaken by Tower [7] [8] and Reynolds [9] in the 1880's, the stimulus for which was the poor reliability of railway axle bearings. Michell was the first to extend Reynolds' classical work and obtain solutions which took account of side leakage. Tilting pad designs were developed that positioned a pivot point over the centre of fluid pressure thus creating a component that always positioned itself in the optimum position. This concept is still as relevant today as it was back then and, as has been observed [10], bearing designs today still bear great similarity to those of the past. This should not be seen as a negative as surely it is a case of admiration that, despite advances in technology, sound, well thought out designs that are inherently reliable can continue to flourish and satisfy their requirements for a great number of years.

It is a very similar story for the plain bearing lining of choice. Whitemetal (Babbitt) has been in use since its conception in 1839. In the early 19th century dissimilar metals were being used to run against each other. Bronzes were often used as a bearing surface which were followed by the use of tin and lead [11]. However, whilst both tin and lead were excellent at embedding dirt particles and operating in boundary lubrication conditions, neither of these materials had the strength to carry the desired loads. Isaac Babbitt patented an alloy in 1839 whereby copper and antimony were added to tin in order to improve strength and hardness. Thus whitemetal came into existence. Despite the growth in the use of alternative materials in many other industries, whitemetal is still the principal material that is associated with plain bearings today.

This said, plain bearing designs have not stood still. More demanding requirements have pushed designers into finding ways to extend both the operating envelopes and the life of products. Poor designs have fallen by the wayside whilst robust designs have endured. In the next section, the authors look at some successful bearing solutions and bearing features which have stood the test of time.

2 Robust Designs

The passage of time has seen plain bearings becoming progressively more reliable. Whilst there is more than one way to solve a given problem, well designed bearings are sympathetic to the requirements of the application, regardless of their style and should operate successfully without incident, remaining inconspicuous.

In terms of thrust pad designs and thrust pad supports, three types of thrust pad are very commonplace. They are the conventional line pivoted pad (Figure 4), the button/spring plate supported pad (Figure 5) and the spring mattress supported pad (Figure 6).

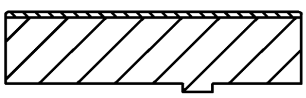


Fig. 4 – Conventional Pivot

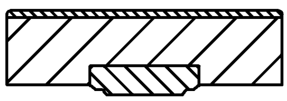


Fig. 5 – Button Support

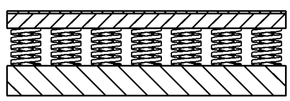


Fig. 6 – Spring Mattress

The type of thrust pad support used is generally dependent upon size, whereby as size increases the pad support mechanism becomes more sophisticated. The vast majority of thrust pad applications can be satisfied using conventional pivots and button supports, however, as the size of the thrust pad increases, the challenge for the designer is to control the amount of thermal and mechanical distortion otherwise failures can occur both during transient conditions [12] and steady-state performance. For large bearing designs, such as those employed in hydroelectric applications the radial width of thrust pads can exceed 900 mm. The spring mattress design, introduced in the 1940's [13], is particularly versatile and is capable of operating successfully in this size range. In 1997 it was reported [14] that General Electric had over 800 bearings of this type in operation in hydroelectric applications. Other choices of thrust pad support systems are available and, over the years, machine manufacturers, rather than bearing specialists, have developed their own in-house techniques of providing bearing solutions. In addition to the spring mattress designs, elastomer support designs, membrane support designs and double layer designs (Figure 7) are also used successfully.

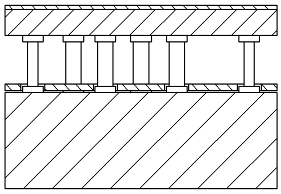


Fig. 7 – Double Layer Thrust Pad Design

It is also worthy of note that, whilst sector shaped thrust pads are the most prevalent and versatile design type, circular shaped thrust pads have been successfully operated in a number of applications since their conception in 1968 [15]. This type of thrust pad must therefore also be considered to be a robust design.

When it comes to radial bearings, plain cylindrical bushes are undoubtedly the most utilised. Despite being the oldest and simplest design, the plain cylindrical bush is remarkably versatile in terms of its operating envelope, its size range and its ability to satisfy requirements across a wide range of applications. Even though the operation of the plain bush was not understood until the 1880's, they had been employed in their whitemetal lined form since 1839 (when Babbitt invented the alloy). When the tilting journal pad design emerged in 1912 the plain bush was not superseded but rather tilting pads designs found their place in environments whereby they could satisfy more onerous operational requirements.

Whatever the application, whether horizontal or vertical, tilting pad or journal bush, all bearings must be furnished with lubricant in sufficient quantity that satisfies the bearing duty requirements. Oil circulation can be achieved by means of external pumps and, in cases where shear losses are high this may be the only option, but the simplest concept is to use the motion of the rotating shaft to circulate the lubricant. This method is tried and tested and has proven extremely robust in a number of guises. Figures 8 and 9 show two methods employed in horizontal bearings using pick-up rings to circulate the oil to the working surfaces. In each case the working bearing surfaces are above the standing oil level at start up and so the rings must deliver oil with haste within the first few revolutions of the shaft. Both methods are in very common use today but have been used for over a hundred years. In fact it is reported that loose ring bearings were exhibited in London as early as 1848 [16]. Whist both designs have the different core strengths their robustness cannot be denied.

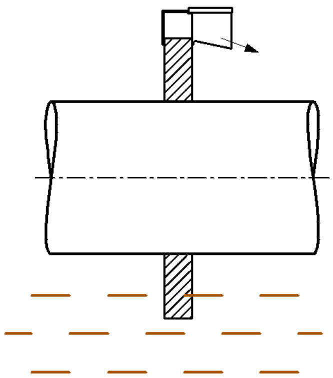


Fig. 8 – Fixed Ring Lubrication

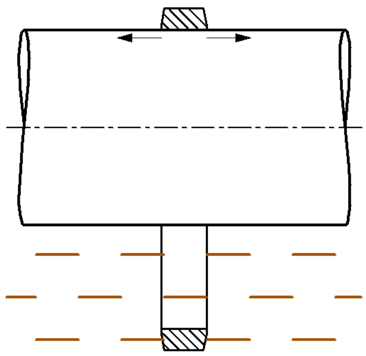


Fig. 9 – Loose Ring Lubrication

Similarly in vertical shaft bearings, the rotating collar can be used to pump the oil around the bearing. Figure 10 shows the oil flow path in a Michell vertical bearing. This design results in a very compact bearing with very effective cooling. The arrangement was conceived in the 1970's and has proven very popular. For larger vertical bearings with higher operational power losses, the pumping action of the collar can be utilised to create sufficient pressure to circulate the oil through an external cooler. Figure 11 shows a cross-section of a bearing utilising this type of oil distribution. Again this design has been proven to be robust over many years of use [17].

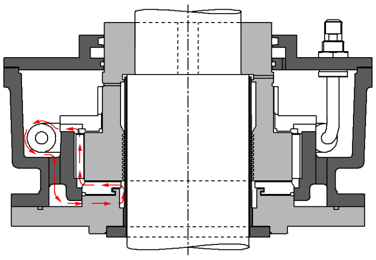


Fig. 10 – Michell Vertical Bearing

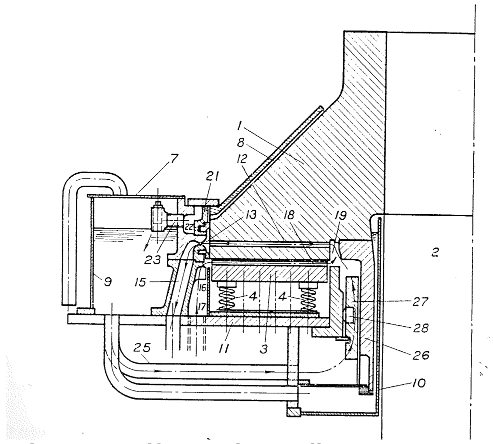


Fig. 11 – Self Pumping Collar Arrangement (source: Michell AGM, 'Lubrication: Its Principles and Practice')

3 Progressive Improvements

For products to remain successful they must keep pace with customer requirements. Whilst the basic concept is largely unchanged, evolutionary progress has certainly been made. Consequently, many aspects of the robust designs outlined in the previous section have been developed and improved over the years.

In 1907 the first tilting pad bearing in service was installed for a centrifugal pump at Cohuna, Victoria Australia [18]. This Michell bearing had a specific load of 1.5 MPa running at 200 rpm. In 1910 another Michell bearing for a steam turbine application was in operation with a specific load of 3.5 MPa and running at 1750 rpm [18]. In terms of specific load the latter loading is one that can be readily associated with bearings operating today. Whilst there will always be some special designs operating at surprisingly high loads, most main-stream applications have bearings operating at specific loads in the order of 4.2 to 5.5 MPa suggesting that, over the last 100 years, only very modest increases in loadings of the order of 20% have been made. The reasons for this are varied but, in addition to bearing design limitations, factors such as longevity of existing operating plant, the adequacy of existing designs and overall industry conservatisim are all influential.

Clearly, over the last century, knowledge of bearing operation has improved and advances in a number of areas have resulted in improved and more reliable bearing operation. The demands of machine designers to make faster or larger machines has forced bearing designs to improve.

In 1950, tilting pad thrust bearing speed limitations were quoted as being 70 m/s at the mean radius [19] for steam turbine applications. Today both tilting pad thrust bearings and journal bearings operate in excess of 120 m/s. Engineers have developed methods of extending the speed range using techniques of delivering the oil to the working surface that reduce or virtually eliminate churning losses and reduce operating temperatures [20] [21] [22]. These same principles have been adopted for tilting pad journal bearings with similar results. Further enhancements have been made to bearings for high speed machinery by using high conductivity backing materials [23] such as copper chromium to reduce bearing temperatures and extend the upper speed boundary. Significant effort has been applied to determining the effects of varying geometry on rotordynamic behaviour [24] and designs specifically targeting rotordynamic problems have been developed [25], [26].

The best examples that illustrate how size has increased over time are to be found in large hydroelectric schemes. Such schemes as the Three Gorges project in China and Itaipu in Brazil have thrust bearing runner diameters in excess of 5 metres. Both spring mattress and double layer designs are employed at Three Gorges. Figure 12 shows one of the Andritz designed spring mattress thrust bearings employed at Three Gorges (Sanxia). The outside diameter of the runner is in excess of 5.4 metres and the thrust load supported is more than 34 MN. The double layer design has an outer diameter of 5.2 metres and supports 47.6 MN [27]. To put this into context, Kingsbury's 1912 thrust bearing design at Holtwood supported 2.2 MN with a runner diameter of 1.2 metres [6] and in the 1950's runner diameters were quoted as reaching 10 feet (3 metres) [28].

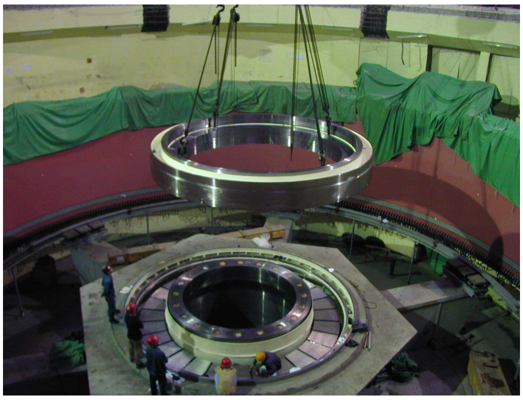


Fig. 12 – Sanxia Generator Thrust Bearing
(courtesy of Andritz Hydro Canada Inc.)

Manufacturing and measuring techniques have clearly improved over the years allowing such large components to be produced with improved accuracy, however it is the pre-existing robust designs that have been built upon. Even with improved manufacturing methods, the predominant factor that has allowed engineers to increase size and speed by such margins is the ability to simulate the performance of bearings using computer models. With advances in computer technology, more elaborate theoretical models, for both thrust and journal bearings could be employed more readily and with increasing sophistication. The 1970's saw designers increasingly concentrating on the accurate prediction of performance with temperature becoming the focal point. More robust guidance on the safe operation of bearings began to emerge [29] and the impact of journal bearing geometry on rotordynamic behaviour was investigated [30]. The focus on temperature continued into the 1980's with optimum positions for temperature measurement being established [31].

With the passage of time, knowledge of plain bearing operation has steadily improved and today customers expect that bearing performance in terms of temperature, film thickness and power loss can be predicted with a high degree of certainty. It is also expected that accurate oil film coefficients [32] can be provided allowing sound rotordynamic predictions to be generated. Robust bearing designs can only be produced using credible predictive software that has been extensively tested against known bearing performance and refined over a number of years. The use of condition monitoring equipment is now virtually standard on every bearing so a good correlation between predicted and actual performance provides the customer with the necessary confidence that the bearing will be reliable. Latterly there has been the natural progression towards finite element solutions [33], [34] [35] however current finite element packages find the fluid film layer difficult to handle and, depending upon the sophistication of the model, results can be variable with significant, sometimes prohibitive, computing resources being required [36].

4 Bearing Linings

On first inspection, the basic whitemetal alloy seems very simple; an alloy of tin (or lead), copper and antimony. However, it is only when the list of requirements of a bearing lining are considered that one appreciates how well the whitemetal alloy is suited to the purpose: sacrificial, seizure resistant, good embeddability, fatigue resistant, corrosion resistant, conformable, ductile, readily castable, and the list goes on. The material has even proven to be highly robust when subjected to explosive shock in naval warfare situations. Today it is still the most common lining for plain bearings and must be considered to be a significant success.

Despite its longevity of use, whitemetal has many limitations which metallurgists have continually sought to overcome to improve its robustness. In 1976 it was reported that 'not many years ago' there were over 450 different whitemetal alloys in existence [37]. It is highly doubtful whether many of these alloys offered any significant advantages over one another but additives such as arsenic and cadmium were recognised as improving grain refinement and adding strength. Such a vast number of alloys was not sustainable and the standardisation of whitemetal alloys is now largely complete. Whitemetal bearing failure mechanisms are also well documented [38]. Alloying elements such as lead, arsenic and cadmium are no longer used due to their toxicity and other coating materials such as copper-lead have also been rendered obsolete for the same reasons. This has led to further development of whitemetal that has focused on increasing the content of alloying elements together with the addition of silver and zinc to improve creep resistance and strength [39]. Tests have shown that the increased alloying content does improve strength but concerns have been expressed over the potential for brittle bond formation [40].

A good alloy is only one element of providing a satisfactory lining. A robust process is essential and there are many elements to control. The molten whitemetal must be poured at the correct temperature, the backing material must also be at the correct temperature, the surface of the backing material must be prepared correctly and the cooling cycle is paramount to ensure a refined grain structure. Bonding techniques have obviously improved over time and mechanical anchoring (dovetail grooves) to the backing material is no longer required. The practice of ultrasonic testing (UT) of the whitemetal bond has been in existence for many years and the process has now been standardised [41]. Using this technique the quality of the bond can be readily verified. Despite these advances, there are still bearings in operation containing dovetail grooves. These grooves can actually detract from the bond integrity as well as making the UT inspection process very problematic but the fact that they are still in existence is an indicator of the slow moving nature of some industry sectors.

In addition to the whitemetal alloy and bond integrity, the condition of the working surface itself has also been subject to investigation. Surface shape and finish have both been studied with respect to enhancing bearing performance, both during transient and normal operating conditions. In some cases large thrust pads have been hand scraped to incorporate a micro-pattern. The belief is that this micro-pattern retains oil to reduce friction at start-up [28]. It has been reported that this process serves no useful purpose [42] however other authors have reported that the use of micro-patterned thrust pads results in reduced friction by up to 10% compared to the equivalent un-patterned pads as well as producing slightly lower operating temperatures [43]. Recently focus has been given to the effect of different 'dimple' geometries, attempting to understand the role that the quantity and shape of the dimple has on bearing performance [44] [45] [46].

Other than the traditional bearing alloys discussed above, there are a number of non-metallic materials, such as ceramics that have been developed with specialist applications in mind. It is not possible to cover all lining materials in this paper but it is acknowledged that specialist linings do exist and have found their niche. However, certain polymeric materials, that could be considered specialist lining materials, are worthy of mention because they have found, or are finding, favour in more mainstream plain bearing applications. Both PEEK (polyetheretherketone) and PTFE (polytetrafluoroethylene) have been used to replace whitemetal in both thrust and journal applications. PTFE is now well established in the hydro industry where it has been in use since the 1970's [47] but PEEK has recently been fitted into a small number of hydro applications with reported success [48]. Both materials make claims to be capable of operating at higher loads than whitemetal and like for like comparisons suggest that this is the case [49] [50]. Other advantages of PTFE in thrust pad applications are; higher starting load capability, reduced thermal crowning during operation and relaxation of manufacturing tolerances. The compliant nature of the material also allows for a greater acceptance of runner misalignment and structural deflections [51].

Despite the higher load ratings, most hydro applications using polymer lined thrust pads are operating at similar, or modestly higher, specific loads to that of whitemetal. This is due either to the conservative nature of the industry or due to the fact that a pre-existing whitemetal bearing has been subsequently retrofitted with a polymer pad. A good example of the latter is a PTFE lined thrust bearing which was installed by Michell Bearings into Unit 4 at Ffestiniog Power Station in 1996. With the exception of the elimination of the hydrostatic jacking feature, the virgin PTFE lined thrust pad replaced the previous whitemetal pad on a like for like basis. The bearing is shown in Figure 13 and the parameters are shown in Table 1. The same set of thrust pads have been in operation since they were installed 19 years ago and, as of May 2015, they have completed 71,667 hours of operation without the pads having had any refurbishment at any time. The operational profile means the thrust bearing is subject to as many as 13 start/stops per day. Although the bearing is modestly loaded, the longevity of operation without intervention significantly exceeds the life of the equivalent whitemetal lined bearing. This demonstrates the reliability and durability of the PTFE material, providing a robust solution for the customer.

Fig. 13 Ffestiniog Unit 4 Thrust Bearing

Number of pads	12
Pad outside diameter	1640 mm
Pad inside diameter	875 mm
Offset line pivots	0.6
Oil type (ISO VG)	68
Sliding speed at MPD	29 m/s
Specific load	2.8 MPa

Tab. 1 – Data for Ffestiniog Thrust Bearing

5 Some Recent Developments

Polymeric materials are of obvious interest if there is the prospect of adding value for customers with increased durability, reduced bearing dimensions (due to higher unit loadings) and reduced complexity. The current performance and reliability of PTFE technology is such that many large machines are now designed and built with PTFE faced pads as the first choice.

Figure 14 shows a 15% carbon, 2% graphite filled PTFE lined thrust bearing supplied by Michell Bearings to Andritz Hydro Canada Inc. for Caojie power station on the Jialing River in China. The bearings were of spring mattress support construction and the use of the PTFE material was specified. Four machine sets were supplied with the first machine set being commissioned in 2010. The bearing parameters are given in Table 2. The specific loading is at the upper end of the load range that is usually associated with similar large sized bearings lined with whitemetal but it was the added benefits associated with PTFE's ability to limit thermal distortion [13] that was the primary interest in this case.

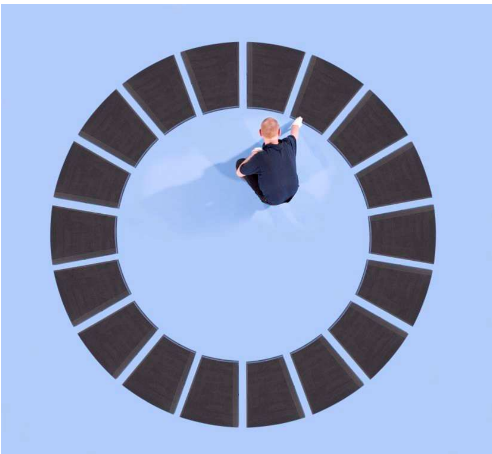


Fig. 14 – Caojie Generator Thrust Bearing

Number of pads	18
Pad outside diameter	3850 mm
Pad inside diameter	2520 mm
Spring mattress support	Offset
Sliding speed at MPD	12-33 m/s
Thrust load	26 MN
Specific load	4.5 MPa

Tab. 2 Data for Caojie Thrust Bearing

PTFE technology was originally developed to overcome problems associated with the performance of large thrust pads. For this reason its use as a lining material on small thrust pads has received much less attention. It has been reported that PTFE lined thrust pads with a radial width of 138 mm can sustain loads up to 20MPa [52]. PEEK has also been shown to be capable of sustaining loads of 16.2 MPa on a 62.5 mm thrust pad but at sliding speeds of 67.7 m/s where oil film thicknesses are higher [50]. In many applications however, such as pump applications, sliding speeds are much lower and any load increases must be carried on comparatively smaller oil films.

Michell Bearings has recently undertaken tests to determine if PTFE technology can yield advantages for small thrust pads operating at sliding speeds in the range 2.2 to 16.5 m/s (measured at the thrust pad mean pressure diameter) which encompasses the majority of pump applications. In addition, the thrust pads in question were also subjected to high load breakaway cycles before any hydrodynamic testing took place.

5.1 Small PTFE Thrust Pad Breakaway Tests

A set of 12 small offset pivoted thrust pads lined with a 15% carbon, 2% graphite filled grade PTFE were subject to a total of 1300 starts/stops under specific loads ranging from 3 MPa to 7 MPa. Load was increased in 0.5 MPa increments and a minimum of 100 starts/stops were completed at each load. In total 400 starts/stops were completed at 7 MPa. The thickness and surface finish of all thrust pads were measured before the start of the tests and at intermediate points during the test programme as well as at the end.

The test rig, shown in Figure 15, comprised of housing containing the set of 12 pads which had been split into two groups of 6 thrust pads placed either side of a central collar with a surface roughness of 0.4 microns Ra. A hydraulic piston behind one group of 6 pads was used to apply the load. The housing was filled with oil so that all the thrust pads were fully submerged in oil. The shaft could be turned by means of a calibrated torque wrench allowing the measurement of breakaway torque and hence the coefficient of friction. The thrust pad details are given in Table 3.

The daily test routine comprised of 20 start/stop cycles with a one minute dwell period between starts. The load was applied for a minimum of 24 hours prior to the commencement of the daily routine. All tests were carried out at an ambient temperature of circa 20 ^{o}C.

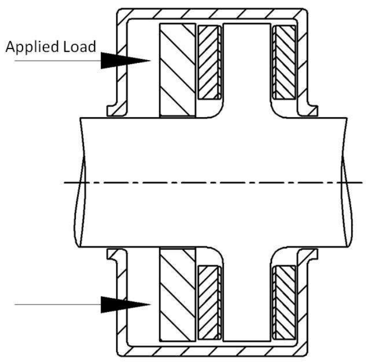


Fig. 15– Breakaway Test Rig

Number of pads	12
Pad outside diameter	251 mm
Pad inside diameter	161 mm
PTFE roughness	1.0 microns Ra
Offset line pivot	0.6
Specific loading	3 to 7 MPa

Tab 3 Data for Small PTFE Thrust Pads

It was found that over the course of the test programme the PTFE surface took on a more polished appearance and the surface finish of the polished area improved to a final value of between 0.4 and 0.7 microns Ra. Although the extent of the polished area increased over the course of the tests, the original machining marks were still evident over the full extent of the face. In all cases the breakaway was very smooth with no juddering. Figure 16 shows a pad which is representative of the set.

Fig 16 – Thrust Pad – Post Breakaway Testing

The coefficient of friction was found to be in the order of 0.06 to 0.08 for the cycles with a 1 minute dwell period and 0.08 to 0.12 for the 24 hour dwell cycle. The measured values were very similar to those measured by other authors [53] [54]. The coefficients did reduce through the course of the tests from the higher end to the lower end of the quoted ranges. There was no discernible difference in thrust pad thickness between the start and the end of the test. Equivalent tests using whitemetal thrust pads loaded up to 4 MPa yielded coefficients in the range 0.18 to 0.31.

5.2 Small PTFE Thrust Pad Hydrodynamic Tests

The same set of thrust pads (Table 3), which had previously underwent 1300 breakaway cycles was then subjected to hydrodynamic testing at specific loads up to 7 MPa. The test bearing which housed the PTFE thrust pads was mounted onto a vertical test rig fitted with a loading module to either lift or pull the shaft against the thrust pads under test.

Both forward and reverse senses of rotation were tested meaning the offset pivoted thrust pads were operated with an effective pivot position of 0.6 and 0.33 depending upon the direction of rotation. The bearing was supplied with 1200 litres/hour of ISO VG 32 oil from an external source at a nominal inlet temperature of 42^{o}C for the duration of the test programme.

Typically, the daily cycle was one of establishing a steady state temperature at the required operating speed and then stepping through increments of load. In general, 2 hours of steady state running conditions were maintained at any given duty point except for the highest load of 7 MPa where steady state conditions were maintained for 8 hours at each duty point. The duty conditions are summarised in Table 4.

Test Stage	Loads (MPa)	Speeds (m/s)	Direction of Rotation
1	2.0 – 7.0	2.2 – 16.5	Forward
2	2.0 – 7.0	2.2 – 16.5	Reverse

Tab 4 – Hydrodynamic Testing – Duty Conditions

Figures 17 and18 show the same representative thrust pad after the test stages indicated in Table 4.

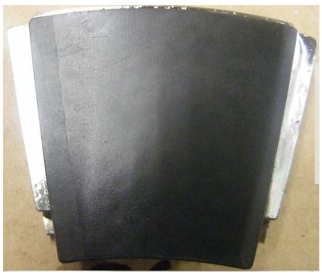


Fig. 17 – Post Test Stage 1

Fig. 18 – Post Test Stage 2

All the thrust pads were in good condition at the end of the test programme with only minor marking caused by foreign bodies present in the oil. The results show that small offset pivoted thrust pads lined with PTFE are able to sustain high specific loads in a low to moderate speed range in both forward and reverse directions of rotation. The conclusion drawn is that the compliant nature of the PTFE allowed the thrust pads to operate at duties that would result in oil films unacceptably low for conventional whitemetal pads of this size. Moreover, the pads demonstrated durability, having not only been tolerant to rotation reversals, but having been previously subjected to numerous high load starts.

6 The Future for Plain Bearings ?

The plain bearing industry and associated markets are now very mature. Whilst there are still applications that require specialist knowledge and continue to drive product development and innovation there are other sectors of the market that desire other attributes from bearings; reliability, longevity, lower costs, and environmental friendliness. Whilst such demands prevail, development will progress in an incremental 'staircase' manner much like it did in the 1970's [37]. Undoubtedly solutions for more demanding applications will continue to feed through as improvements for the wider markets.

At the present moment there does not appear to be a disruptive technology that has the potential to render current plain bearing designs obsolete per se. The inherent cost effectiveness, longevity and reliability make them attractive solutions for numerous applications to this day. There have been certain industry sectors whereby new technology has encroached or displaced incumbent designs that utilise whitemetal bearings. An example of this is in the cruise industry whereby the advantages of using podded propulsion have started to displace conventional propulsion arrangements. Initially, the most convenient arrangement seemed to be to use anti-friction bearings within the pod, but it would appear that, for the thrust bearings at least, there are still advantages in using plain thrust bearing technology [55]. Another example involves the use of magnetic bearing technology in compressor applications involved with oil pipelines. Here the remote tuning and monitoring capability of magnetic bearings yields advantages in terms of machine availability, reliability and energy security therefore justifying the additional expense of this technology.

Following the invention of the tilting pad, sliding speeds have doubled, sizes have, in some cases, quadrupled but unit loading has not seen the same degree of progress. Unit loading would seem to be an area on which to focus if more efficient solutions are to be offered. One means of doing so involves the use of alternative materials to whitemetal, such as PTFE, which also has the potential to reduce product costs. The success of such materials in more widespread markets will be dependent upon industry being receptive to such changes and recognising that many long standing, highly conservative operational norms will need to be updated.

7 Conclusions

The tilting pad concept is still going strong 110 years after its invention and seems set to carry on for some time to come. This alone should be a cause for celebration as, although designs have progressed, understanding has improved and performance of the product is enhanced, the fundamental concept remains unchanged which is a significant credit to the inventors.

A number of key product features and design elements are common place across the industry. They have proven to be robust over many years and appear deceptively simple in their conception. Hydrodynamic bearings that are well engineered have a simple elegance, are reliable and enjoy longevity whilst adding value for customers and especially end users.

There are still further improvements that can be made to plain bearings and currently polymeric linings, amongst others, still have potential to offer worthwhile performance benefits. However, industry is inherently conservative and a combination of the familiarity with and confidence in whitemetal lined products, whilst being highly positive attributes of traditional plain bearing technology, could impact the rate at which new technological developments are embedded in the future.

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