Friction and Wear Characteristics of ThorPlas Bearings and Their Application in Hydro Turbines

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ABSTRACT

ThorPlas is a proprietary engineered thermoplastic polymer bearing material which is homogeneous and self – lubricating. The material is formulated with a special combination of lubricants to offer effective performance in a wide range of bearing applications including high pressure - low speed conditions. ThorPlas is designed to operate dry or wet (in water), typically for applications such as in wicket gate and control mechanism for hydro turbines. If required, ThorPlas can also operate in oil lubricated environments such as in Kaplan Turbine blade bearings and low pressure and high speed applications such as pumps. This paper focuses on the friction and wear characteristic of ThorPlas bearings based on test results. First of all, the friction, wear, creep and edge loading test results are presented in detail. And then, special design considerations are discussed such as pressure calculation, edge loading, abrasive handling and seals.

In conclusion, based on friction and wear test results, ThorPlas bearings offer high performance suitable for wicket gates, control mechanism and other hydro turbine applications.

1.0 Friction and Wear

ThorPlas bearing material is a result of research and development work for more than 10 years. The experimental investigations can be divided into two groups: 1) the material characterization and 2) bearing simulation. The material characterization tests were to determine the physical properties of the material including mechanical properties such as tensile, shear and impact strength, water absorption; the thermal properties such as linear thermal expansion and temperature limit, and the tribological properties such as friction and adhesive and abrasive wear. The test results are published in the ThorPlas® Bearings Engineering Manual [5] which is in the public domain. It is not the focus of this paper to present this group of tests.

This paper deals with the second group of tests which was to simulate the performance of material in bearing configurations, especially, under high pressure and low speed oscillating movement. The U.S. Army, Corps of Engineers had developed a test specification to evaluate the performance of self – lubricating bushings for wicket gate application in hydro electric turbines. The specification defines an accelerated wear test to simulate the operational condition of hydro turbines. Most self-lubricating products in use today have been subjected to the requirement of this test program.

Powertech Labs Inc [1] has tested ThorPlas per the specifications defined by the U.S. Army, Corps of Engineers. This was an accelerated wear test. Before the accelerated wear test, the bearing sample was statically loaded to 23 MPa (3300 psi). The shaft displacement relative to the test block was continuously monitored. In the first 4 hours the shaft was oscillated ± 5 degrees at 0.1 Hz every 5 minutes. For the next 20 hours, the shaft was oscillated the same amount, but every 10 minutes. This period of time was used as the permanent set and creep measurement phase for both wet and dry conditions.

After the initial operation described above, the accelerated wear test began. The applied radial load of 23 MPa (3300 psi) had, in addition, superimposed a dynamic load of 7 MPa (1000 psi) at 2 Hz. The shaft rotated ± 1 degree continuously at 2 Hz. Every 15 minutes the radial dynamic load was paused and the shaft rotated ± 15 degrees at 0.1 Hz (10 seconds to complete the swing). During this period of time, only static radial load (23 MPa) was applied. The testing ran for a minimum 120 hours for both wet and dry. This test duration under the loading condition described above approximately represents 13 years of actual service life of a typical turbine [8].

The stainless steel 17- 4 PH shaft was 127/126.97 mm in diameter with a hardness of RC40. Surface finish was $R_a = 0.4\,\mu m$. The ThorPlas bushing with a 4.68 mm thick wall thickness was installed into a steel shell. The measured coefficient of friction was shown on Fig -1a & b.

Fig -1a: Coefficient of friction under dry

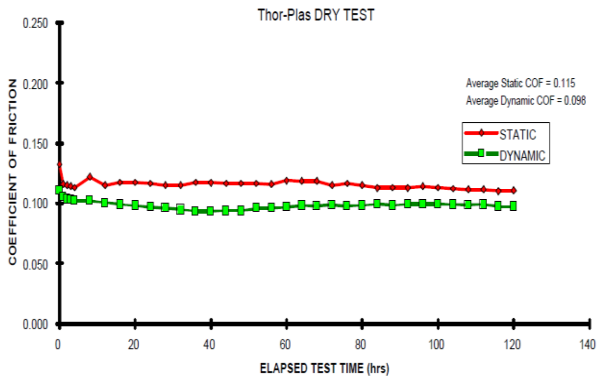
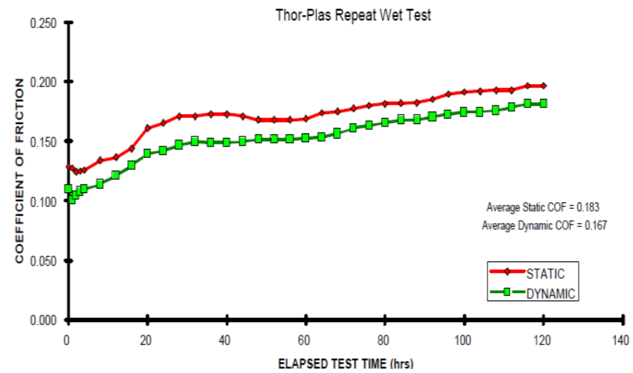


Fig -1b: Coefficient of friction under wet



The result indicates that the coefficient of friction under wet condition was slightly higher than dry. The reason for this is not conclusive. A Scanning Electron Microscopy (SEM) coupled with an Energy Dispersive X-ray Analysis (EDX) was performed by University of Toronto [6] to analyze the molecule concentration of lubricant on the surface of the samples under new and worn conditions after testing. It showed that the amount of one of the solid lubricants was reduced. However, the overall concentration of this lubricant was still higher than required. This might partially explain the higher wet friction, but more investigation is required to confirm this. In contrary to the friction, the wear level observed under wet condition was much lower than dry. The measured adhesive wear (unadjusted) is demonstrated in Fig-2a and 2b below:

Fig – 2a: Wear measurement under dry

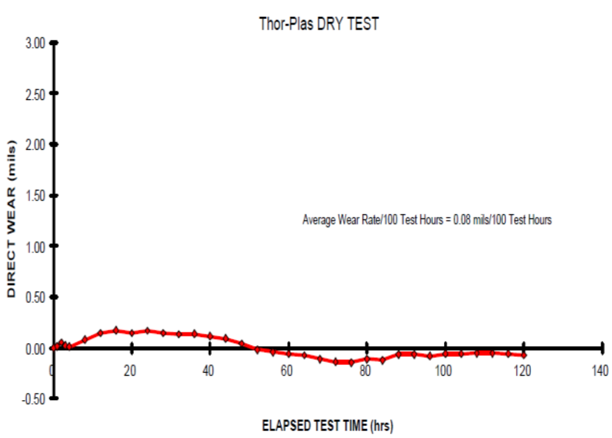
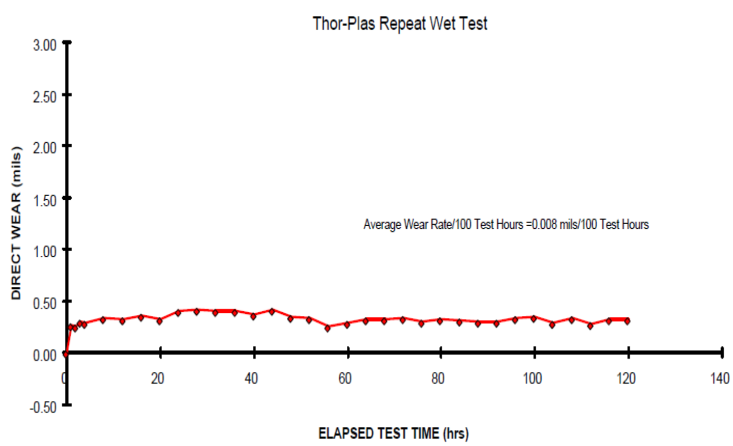


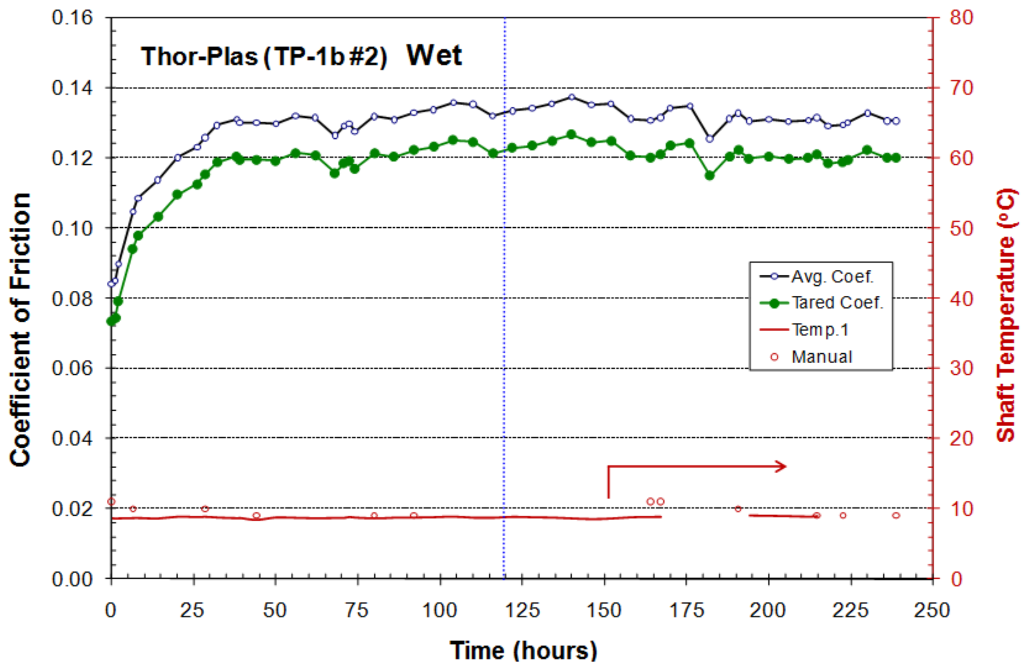
Fig -2b: Wear Measurement under wet



In Fig -2b, after making the initial displacement zero, the calculated wear rate for wet test was actually lower than dry condition. The average adhesive wear during 100 hours was 0.08 mils (1 mil = 0.001 inch) for dry while it was 0.008 mils for wet. According to report [8], the 100 hours testing is approximately equal to 3.98 miles travel which is 6405 meters. Based on this traveling distance and bearing static pressure, the wear factor (as defined later section of this paper) is 1.378×10^{-11} $(\mathrm{mm}^2/\mathrm{N})$ for dry operation.

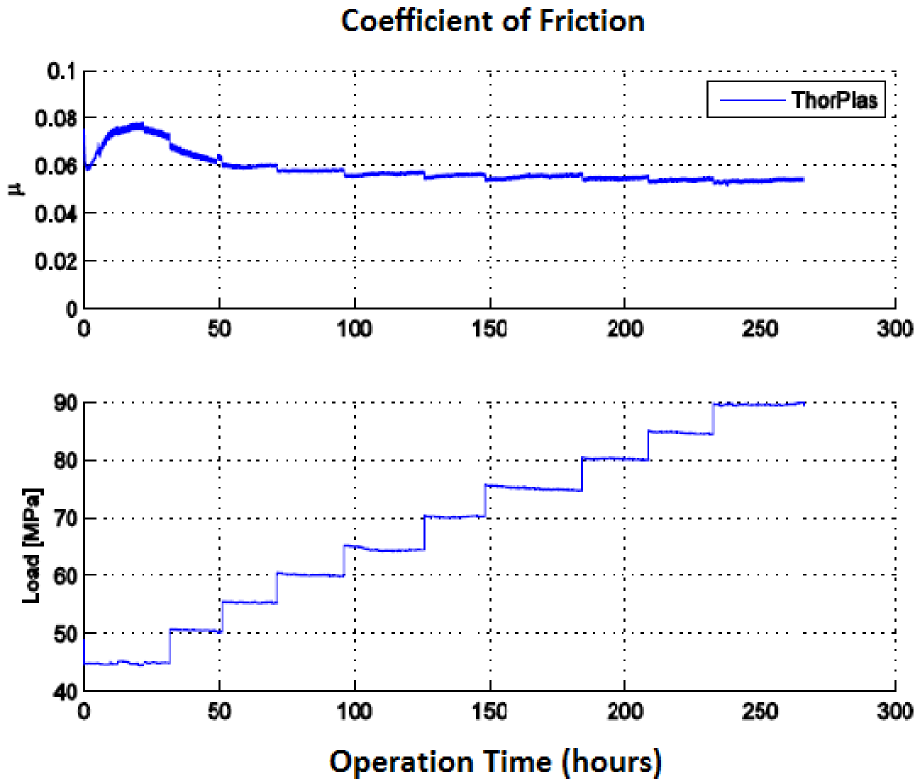
From Fig -1b, it was unclear whether the increasing trend of friction would continue or level off after certain period of time in wet condition. Evidences during ThorPlas wet testing at PowerTech seemed to suggest that although cooling fluid was introduced into the shaft to reduce heat during the much accelerated test, cooling appeared to be less efficient in wet test, possibly due to the effect from water present between the shaft and bearing. To remove the concern about the possible continuing increase in friction with time in water, Thordon performed its own testing using the High Pressure Oscillating Motion (HPOM) test rig [7], under nearly identical loading and oscillating conditions, but extended the testing time to 240 hrs (simulating over 25 years of service life based on typical cycles for wicket-gate bearings). As shown in Fig-3 and from repeating tests, it was observed that friction increased only at early times before reaching its equilibrium and then became stabilized around 0.13 (or 0.12 after tare correction) throughout the duration of test.

Fig - 3: Friction testing for ThorPlas (wet) with HPOM test rig



The friction and wear characteristic of ThorPlas bearings was re-confirmed by another independent test. This was performed by Luleå University of Technology in Sweden [3]. The test was implemented on a test rig to simulate the wicket gate bearing operation in hydro turbines. The bearing was loaded from 45 MPa to 90 MPa. The maximum sliding speed was 5.4 mm/s. Test condition was dry with air cooling to shaft and bearing. The shaft material was stainless steel 2377 with a surface finish $R_a \approx 0.4\ \mu\mathrm{m}$. The coefficient of friction was measured from 0.05 to 0.08 (see Fig - 4). The coefficient of friction showed a general trend of reduction as pressure increased. Under pressure of 45 MPa, the coefficient of friction rose slightly with time after commencement of test and then went down thereafter while it showed a clear reduction over time for higher pressure.

Fig – 4: Coefficient of Friction reduces with time and increased pressure



The bearing wear and operational cycles were measured for each pressure level. The result is summarized in Table -1. Based on adhesive wear measurement of the same test, a wear factor can be calculated. The wear factor is defined based on the wear law in common practice:

$$\Delta = k \cdot p \cdot L \qquad (1)$$

Where: $k =$ Wear factor (mm^2 / N)
$\Delta =$ Measured wear (mm)
$p =$ Bearing Pressure (N / mm^2)
$L =$ Traveling distance (mm)

With using measured wear data and reversing equation (1), the wear factor is calculated:

$$k = \frac{\Delta}{p \times L} \qquad (2)$$

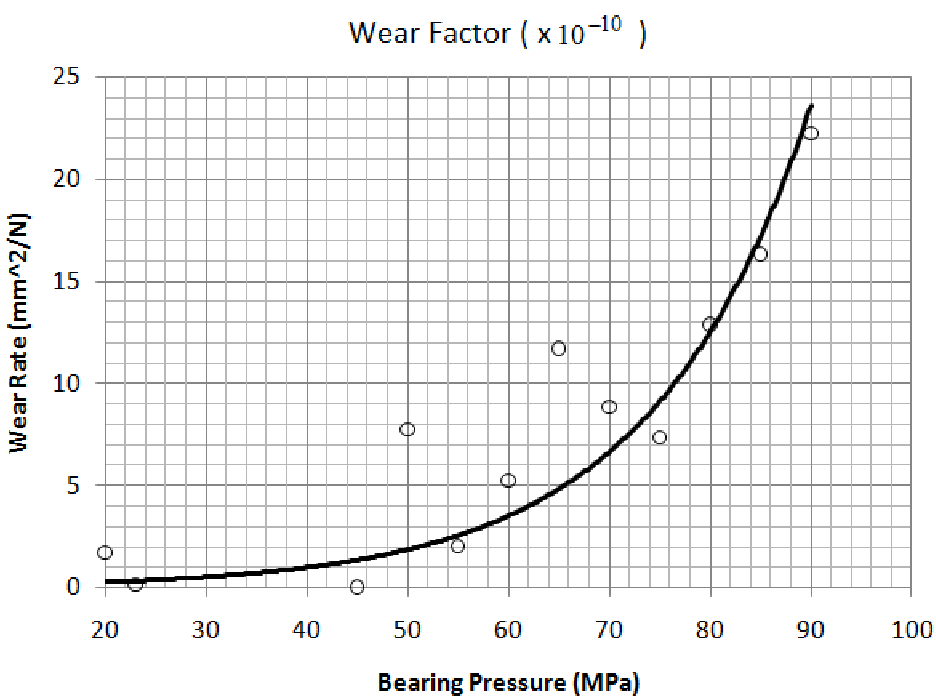
Based on the variables listed above, the dimension of wear factor is (mm^2 / N). The calculated wear factor is listed in Table -1 and also represented in Fig - 5. It is clear that the wear factor increases with increasing bearing pressure and follows approximately an exponential function of bearing pressure. The calculated wear factor by equation (2) is very important for engineering because it comes directly from measurement of the bearing. With help of these data, design engineers are able to estimate the potential wear by knowing bearing pressure and traveling distance.

Table - 1: Measured Adhesive Wear and Wear Factor (Dry)

Pressure (MPa)	Wear (mm)	Cycles	Travel Distance (mm)	Wear Factor (mm^2/N)
20*	0.005	10000	1466000	1.7053×10^{-10}
23**	0.00203	N/A	6405000	0.1378×10^{-10}
45	0.00004	1350	197910	0.0450×10^{-10}
50	0.004658	820	120212	7.7490×10^{-10}
55	0.00092	560	82096	2.0376×10^{-10}
60	0.00384	830	121678	5.2597×10^{-10}
65	0.01095	980	143668	11.7258×10^{-10}
70	0.0069	760	111416	8.8471×10^{-10}
75	0.0104	1280	187648	7.3897×10^{-10}
80	0.0121	800	117280	12.8965×10^{-10}
85	0.0163	800	117280	16.3510×10^{-10}
90	0.0367	1250	183250	22.2525×10^{-10}

* Based on a separate test program with exactly same test rig configuration
** Based on PowerTech test [1]

Fig – 5: Wear factor as function of bearing pressure



2.0 Creep Test in Bearing Configuration

Creep is a condition whereby bearings continuously deform over a long period of time under a constant load. Engineering practice brings out the question if ThorPlas bearings would deform excessively over a long period of time under high pressure condition. The best method to answer this question is to test the actual bearing under simulated load conditions. Powertech Inc has performed a test by directly loading eight bearing halves in tandem [2].

Fig – 6: Schematic of Creep Test Rig

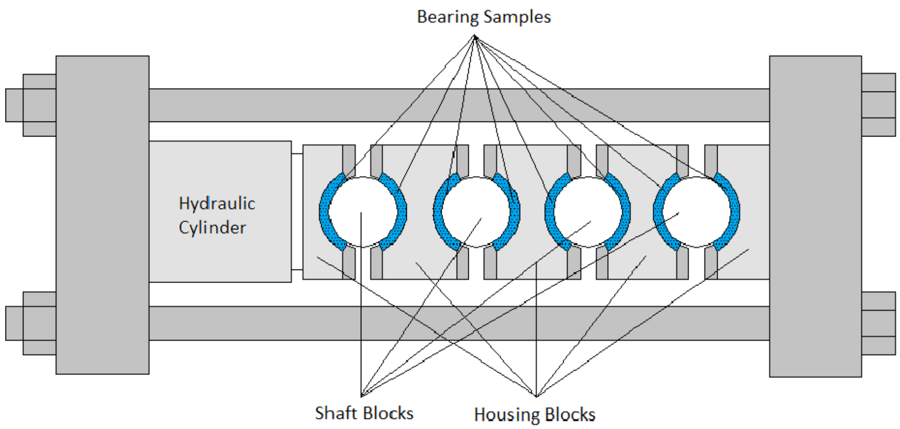
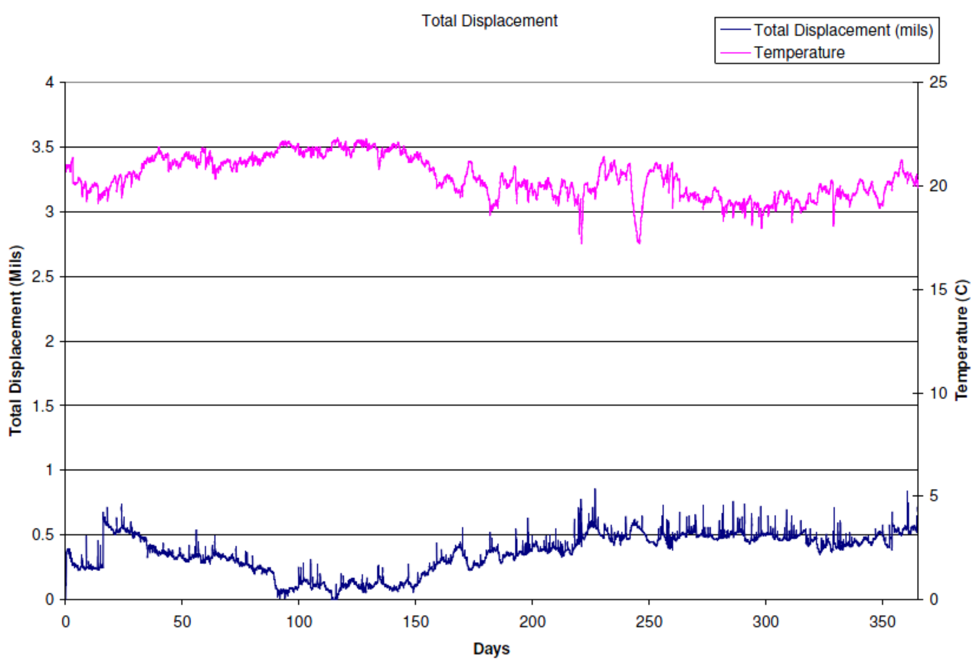


Fig – 7: Measured total displacement and temperature



A special test rig was constructed for the test holding eight test samples in the form of half bearing in tandem between housing and shaft blocks. A hydraulic cylinder was used to apply load to the blocks. The total movement of eight samples was monitored with two SKF CMSS65 eddy current displacement transducers. Fig – 6 schematically shows the work principle of the rig. The bearing pressure was controlled constantly at 55 MPa (8000 psi). The test lasted 360 days (1 year). Fig – 7 shows the total displacement recorded over the year. The measurement was the total amount of creep for all eight bearing halves. It is reported that the total displacement was 0.58 mils (0.0147 mm) [2]. If divided by 8 bearing halves, the amount of creep for each bearing half is 0.073 mils (0.00184 mm). The wall thickness of each sample was 9.53 mm. The creep to wall thickness ratio is 0.00184/9.53 $= 1.93 \times 10^{-4}$ mm/mm/year. From this test, it concludes that at given level of bearing pressure, the creep of ThorPlas is essentially negligible. Therefore, for most engineering applications, creep is not a concern.

3.0 Edge Loading Test

Fig – 8: Schematic of edge loading test principle

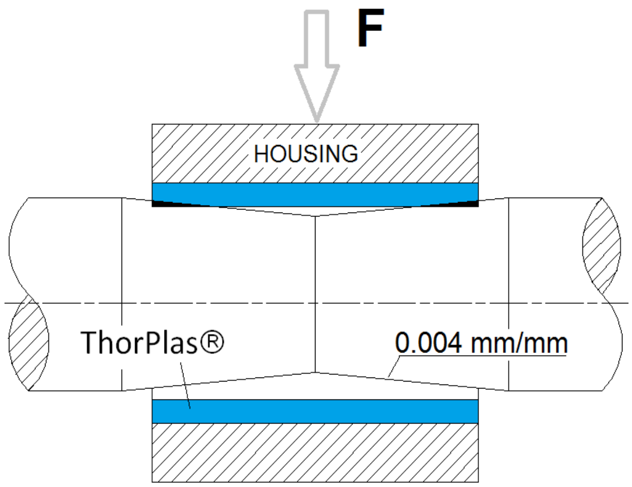
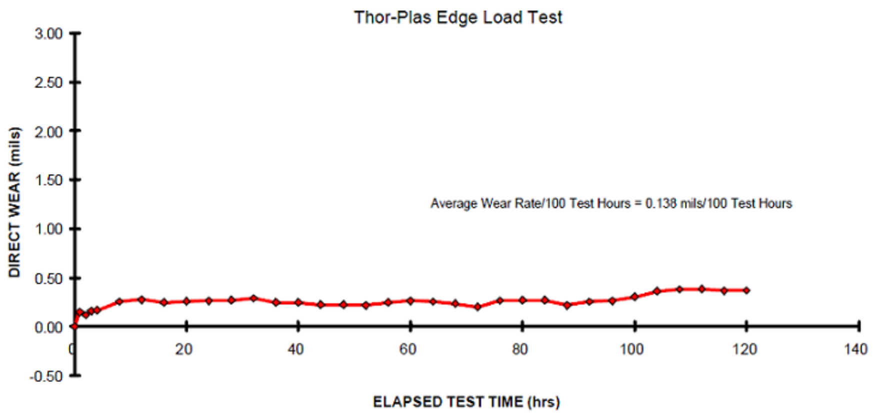
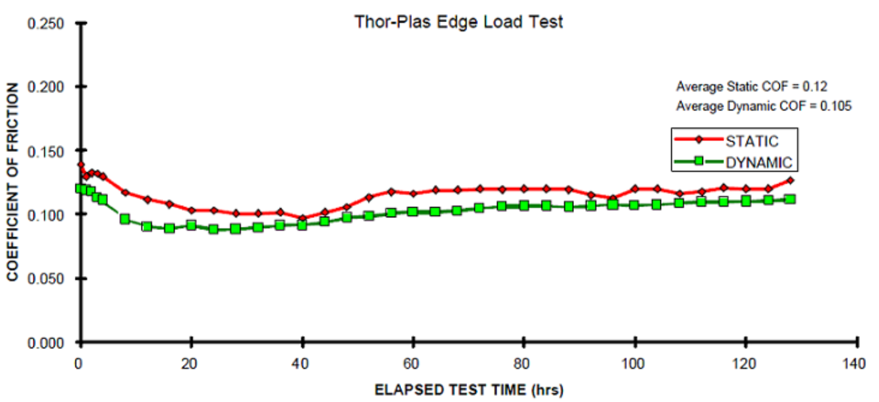


Fig -9: Friction and wear under edge loading





Edge loading is one of the major concerns for turbine designers when selecting bearings for wicket gate applications. Shaft misalignment is used to describe a situation where several bearings on one shaft line are not concentric or the axes of shaft and bearings are not parallel. In terms of wicket gates, there are two major causes of shaft misalignment leading to edge loading: 1) the imperfect installation and 2) insufficient shaft stiffness. There are no statistics that can be used to predict what the maximum possible shaft slope to bearings should be. A case study [4] demonstrates that a shaft slope at bearing of 0.0023 mm/mm can cause the pressure at the edge to be 4 times the design pressure for a ThordPlas bearing and 15 times over designed for a common bronze bearing.

PowerTech Inc designed a test to simulate the edge loading condition of wicket gate bearings by means of machining a double taper with a slope 0.004 mm/mm on the shaft [8]. This amount of shaft slope is considered to be very high and covers the most severe edge loading working conditions. Fig -8 presents the principle of test rig construction.

All test parameters were set the same as those for the dry accelerated wear test as described in section 1.0 except that the shaft was ground to a taper as defined in [8]. The result [1] concluded that there was no evidence of damage on ThorPlas bearing after 130 hours of testing. The coefficient of friction and wear rate (Fig -9) is slightly higher than observed with the perfectly aligned shaft.

4.0 Bearing Design for Hydro Application

4.1 Wicket gate bearings

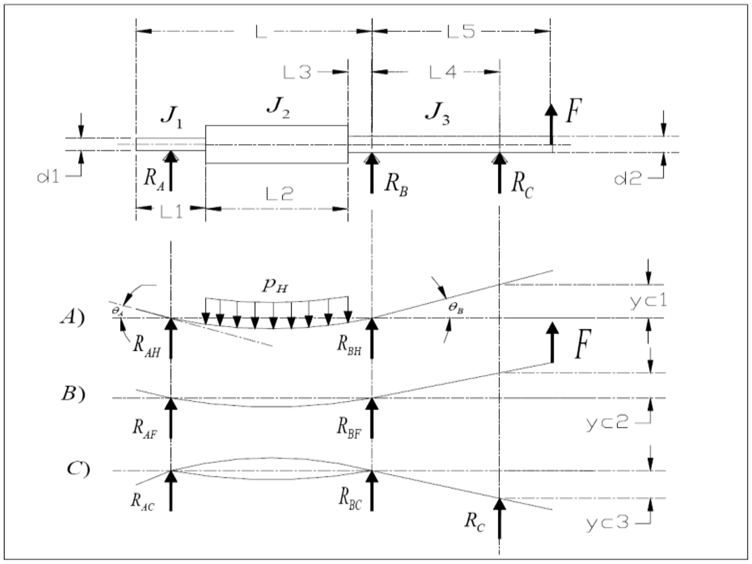
The maximum design pressure over projected area is 31MPa for low speed continuous operation of ThorPlas bearings. The best way to determine the reaction forces of wicket gate bearing is to use Finite Element Analysis (FEA). If an FEA program is not available, the bearing reactions can be calculated using a partial analytical method proposed by N.N. Kovalev [9].

In this method, the shaft is considered as an ideal elastic beam, point–supported at bearing centers by three bearings. The static over-determined system can be separated into 3 individual conditions: (A) -loaded by hydraulic pressure on guide vane only, (B) -loaded by gate ring pulling force only and (C) -loaded by reaction force of upper bearing R_C (see Fig -10). Each individual loading case is statically determined. After calculating each case and adding together the results, one can determine the reactions and shaft slope of the entire system. In loading case (C), there are 3 unknowns including bearing reaction R_C. The 2-dimensional system however can only solve for two unknowns and thus one more equation is required to solve all variables. Shaft deflection at the upper bearing is chosen as such variable. This requires all reactions for lower, middle and upper bearings to be expressed as a function of deflection at the upper bearing. It is designated with “y_C”. According to Fig -10, there is:

$$y_C = y_{C1} + y_{C2} + y_{C3} \tag{3}$$

The final value of “y_C” shall be equal to the running clearance of the upper bearing which is a known design parameter. This provides sufficient information to solve the reactions force and shaft slopes at bearings. In the case study [4], it was confirmed that the calculated result with this method is very close to the result obtained by FEA analysis in terms of bearing reactions and shaft slopes.

Fig - 10: Loading diagram of a typical wicket gate shaft [4]



After obtaining bearing reaction and shaft slope at the bearing, following the procedure proposed by [4], one can obtain the maximum pressure due to edge loading. The maximum edge pressure under edge loading conditions is much higher than the nominal bearing pressure defined by load over projected bearing area. Therefore, in cases of severe edge loading, the nominal bearing pressure should be designed as low as possible so that the maximum edge pressure does not exceed the material strength limit. As a rule of thumb, the maximum edge pressure shall not in any case exceed the material yield strength published in the ThorPlas Engineering Manual [5].

An abrasive environment (water) is one of the most difficult problems of wicket gate bearings. The most effective way to prevent abrasive particles from entering the bearing is to install seals into bearing wall or bearing housing, depending on the design. If installing seals into bearing wall, sufficient thickness of bearing wall is required to accommodate the seal. In this case, the bearing wall is not only determined by the requirement for interference fit but also by the minimum size of seal. Experience has proven that the machined ThorSeals (supplied by Thordon) are very effective to prevent abrasives from getting into bearings. A typical ThorSeal is designed with 1.5 to 2.0 mm diametrical interference against shaft. This means that the inside diameter of the seal at lip is generally smaller than the shaft diameter by 1.5 to 2.0 mm. Therefore, in designing the shaft, a large chamfer is required on the shoulder of the shaft section with seal so that there is no damage to the seal when installing the shaft into bearings.

4.2 Other applications

The ThorPlas bearings are self – lubricated and offer one of the cleanest installations in hydro turbine applications. Wherever the load capacities are met, they can be applied anywhere required. The most popular applications are the linkage bushings for the control mechanism and the Kaplan turbine hub bearings.

5.0 Conclusions

This paper summarized the most important test results for ThorPlas bearings in the past 10 years. From a practice application point of view, the following result is most important:

The coefficient of friction is very low. It varies from 0.05 to 0.1 for dry conditions and from 0.1 to 0.17 for wet. The adhesive wear rate is amongst the lowest of engineered polymers. Tests also showed that the wear rate under wet conditions was much lower than that under dry conditions. The wear factor increases with increasing bearing pressure and follows approximately an exponential function. It is evident that the creep of ThorPlas bearings is essentially negligible for the test pressure 55 MPa. The bearing is able to handle a very high level of edge loading. Practical experience shows a shaft slope 0.002 mm/mm is considered high. ThorPlas bearing was tested with shaft slope 0.004 mm/mm with no evidence of damage. For edge loading concern, a shaft bending and edge loading analysis is strongly recommended to ensure a satisfactory operation.

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