Paper Title: Performance and surface characteristics of slow oscillating journal bearing types subjected to various motion pattern

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Abstract: In hydropower application, journal bearings are used in mechanisms for turbine water flow regulation. Self-lubricated bearings are nowadays standard in Sweden and several other countries. However, hydropower is an experience based industry and requires evidence of good long term function for new technologies. This has led to usage of several bearing types which were earlier successful and but are now partly failing. This investigation shows performance of several bearing types exposed to different oscillating motion pattern under dry conditions. A specially designed test rig was employed for this purpose and 4 bearing types were subjected to two different motion patterns and in total 8 bearing types were tested. The results show surface characteristics connected to tribological behaviour in terms of friction and wear and indicate suitability for wicket gate guide vane applications.

Key words: Hydropower, self-lubricated bearings, small oscillating motion, test

1. INTRODUCTION

Hydropower units have an important role in the generating of electrical power. Besides energy production, the units' capability of controllable output changes offers an effective tool to ensure good electrical grid stability. These stability control regulations can mean one or a few daily starts and stops or just regulations leading to minute based small governing wicket gate system adjustments. Problems in the wicket gate guide vane mechanism will lead to altered machine running or in the worst case a total stop. One critical machine element in the mechanism is the guide vane bearing whose friction and wear characteristics must be accounted for in the design.

Grease lubricated bronze bearings are considered to be an old standard technology used for guide vane bearings but there has always been a desire to reduce the need for maintenance lubrication in hydropower. From early installation of self lubricated bushings, the Vattenfall experience show examples of functioning 40 years old bushings using polymer/fiber lined or lubricant plug bronze bearings. However, everything has not been problem free. Poor function has also been found in a number of newly refurbished (by retrofitting or re-designing) machine using bronze bearings with lubricant plugs. Disassembly during 2006-2009 has shown both deep scoring of surfaces and complete seizure. The first sign of problem has been the lack of regulating force (servomotor capacity) to adjust the guide vanes. Polymer lining has also been seen to have delaminated from the bronze backing.

In 1999 J. A. Jones et al [1] mentioned real application examples using self-lubricated bushings suffering from material swell, seizure or excessive wear in Northern America. Poor experience was also reported in 2001 by J. C. Jones [2] from 30 upgraded machines from 1992 and onward. Increased friction has been the cause for lack of servomotor capacity using lubricant plugged bronze bearings and delaminated lining has been seen in case of polymer lined bearings. Swelling of lubricant plugs reduced the bearing clearance and increased the friction. In some case, the grease lubrication systems were reinstalled.

In [2] it is pointed out that no effective independent tests were performed until mid 1990s from which the results were presented in 1999 [1]. The authors [1] mentioned that testing is usually not done by end users or main turbine suppliers. Bearing suppliers do their own testing in house or outsource it. The interpretation of test results to forecast performance in reality is difficult. It has been seen that the bearing suppliers can easily setup a test in which their own bearing show the best performance. The influence and importance of including small oscillation in wicket gate bearing material evaluation has been reported by Gawarkiewitcz et al [3]. Some materials can cope with small oscillations by having a material shear deformation instead of contact surface sliding. This can be an advantage for materials having lower elasticity modulii compared to those of bronzes.

Test results shows that self-lubricated bushings can match or surpass the performance of oil- or grease lubricated bronze bushings (high leaded tin-bronze, alloy UNS-93200) [1]. These are somewhat similar to those shown by J. Ukonsaari [4] in a comparative study on self lubricated bearings vs. a tin-bronze bushing lubricated by environmentally acceptable oil.

There are several possible explanations for the wicket gate bearing failures in recent years. Hydropower is a conservative industry willing to use functioning technology over and over again. In the case of using lubricant plugged bronzes, the 40 year old recipe of both lubricant and material can very well have changed over the years. One of the reasons is changes in environmental laws striving to reduce the usage of lead, which otherwise is an excellent seizure risk reducing substance for bronzes and lubricants in sliding contacts. Another important reason is the lack of adjustments in the design to suit the selected bearings. Such examples can be seen for bearings using a run-in lubricant film but the installation is not possible to do without scraping a significant amount of the film off the bearing surface. This can cause a worsened run-in process. Another factor to consider is the friction, which can be difficult to predict.

Considering all test results and increasing real application experience, it might seem strange to hear about new failures from recently installed machines. Choosing a bearing suitable for the actual conditions and design for assuring the expected conditions should be adequate. However, bearing selection can be difficult without relevant information. Experiences have shown that a bearing type working very well in one machine, can fail in the next machine. This necessitates the need of dissemination and publications of relevant test results so as to enhance the number of successful self-lubricated bushing installations - for potential increase in machine availability and life. This investigation highlights tribological performance and sliding surface characteristic differences using two different motion patterns using some selected bearing types. The results have been discussed in terms of suitability for wicket gate guide vane applications.

2. TEST METHOD AND SETUP

2.1. Test-rig

The tests were carried out in a specially designed oscillating journal bearing friction and wear tester. The test rig has two bearing halves pressed against an oscillating shaft (Figure 1). The test rig allows control and monitoring of speed, load, friction, decreasing displacement between journal bearings (linear wear), and temperature.

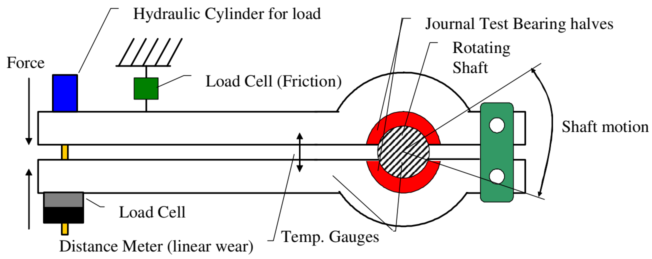

Figure 1. Journal bearing test-rig, viewed from above.

2.2. Guide vane bearing conditions

It has earlier been established that small oscillations occur in guide vane bearings [1]. One recording of a 45 MW Vattenfall Kaplan turbine in Sweden showed that 99 % of 5798 (1 motion every 7.5 minute) guide vane servo motor motions led to bearing motions below 2.5 mm sliding distance. These motions were governing system controlled for compensating the electrical grid frequency offset from an ideal 50 Hz. Water flow induced vibrations on guide vanes can lead to much higher frequency motions as those tested in [1]. ∅

Sliding speed in real applications is normally very low (without water flow induced vibrations), normally below 10 mm/s. When there is low motion speed and long pauses between each motion the bearing temperature will be close to the ambient temperature. In hydropower wicket gate applications, an assumption of ambient temperature of 5-35°C could be representative. The variation depends on machine layout and water temperature.

Bearing pressure in guide vane bearings is typically about 20-30 MPa. Edge pressure from deformation and misalignment can give maximum local contact pressure of 100 MPa or more.

Intentional actions for guide vane bearing run-in are rarely seen in hydropower. During the commissioning and initial adjustment, the machine is tested with a function test program which includes function test with dry wicket gate and about a minimum of 5 real starts and stops.

2.3. Test materials, profiles and conditions

The shaft material was EN 1.4462 (SS 2377): A ferrite-austenitic (duplex) stainless acid resistant steel. The steel contains chrome and molybdenum. The shaft diameter was ∅40 mm and they were ground to a surface roughness R_a 0.3-0.5 μm.

The tested bearings were 30 or 32 mm in length. The bearing suppliers or sales organizations were responsible for manufacturing of test bearing halves to get the desired clearance. Information about the test-rig including dimensions and clearances were provided to them and thoroughly discussed when required. The descriptions of test bearings evaluated are given below.

Oiles JM3: Oiles 500 JM3 SL4: Bronze (88 % Cu, 12 % Tin), lubricant plugs with graphite and lead.
Oiles JM7: Oiles 500 JM7 SL4: Bronze (80 % Cu, 10 % Al, 5 % Fe, 5 % Ni), lubricant plugs with graphite and lead.
Thordon: Thordon ThorPlas, a homogeneous thermoplastic, self-lubricating polymer bearing.
Orkot: Orkot TXM Marine is a reinforced medium weave polymer material including PTFE (Teflon).
DEVA BM: DEVA BM9P is a thin walled sintered bronze material on a steel backing, the sintering includes PTFE.
DEVA TEX: deva.tex is a glass fibre reinforced carrying layer with a fibre strengthened epoxy resin including PTFE.
PAN SOBz: PAN SoBz10 GF is a bronze (12 % Sn, 2 % Ni, rest Cu) with lubricant plugs and run in powder sprayed on.

PAN 12: PAN 12 GF is a bronze (10 % Sn, 5 % Pb, rest Cu + Ni) with lubricant plugs and run in powder sprayed on.

Two different test profiles were used: profile 1 aimed to simulate a machine running with starts and stops and profile 2 to simulate a machine under electrical frequency stabilization regulating mode:

Profile 1 (P-1): Totally 1466 m sliding distance including 10,000 cycles where each cycle was

- 60° forth, 6 s pause, 60° back, 5 s pause.
- 50° forth, 5 s pause, 50° back, 4 s pause.
- 40° forth, 4 s pause, 40° back, 3 s pause.
- 30° forth, 3 s pause, 30° back, 2 s pause.
- 20° forth, 2 s pause, 20° back, 1 s pause.
- 10° forth, 1 s pause, 10° back, 2 s pause.

Profile 2 (P-2): Totally 2433 m sliding distance including 350,000 cycles of 10 degrees forth, 1 s or 2 s pause, 10 degrees back, 1 s or 2 s pause (the pause length was selected for each individual bearing type to keep the shaft temperature under 30°C).

The optional run-in prior to test profile was 50 cycles of 60° forth, 1 s pause, 60° back and 1 s pause.

Sliding speed was set to a maximum 5.6 mm/s and the speed acceleration was linear. Bearing housing was cooled and fan cooling of the shaft was employed for non-copper based bearing types.
The loading was set to achieve a mean bearing pressure of 20 MPa. Each performed test showed a spread of mean bearing pressure between 18.3 to 20.9 MPa, see Table 1.

2.4. Test procedure

The bearings and shaft were mounted in the test-rig prior to cooling to reach stable temperatures. When stable temperature prevailed the load was applied and the run-in and/or the test were started. Material creep over time was not considered.

Friction values were sampled at full sliding speed 0.2 s after every motion start. This corresponds to a dynamic friction value.

Several tests were conducted using the same shaft by shifting the contact position. New shaft surface was used at every test. All copper based bearings had a run-in lubricant film applied on the bearing surface. The mounting of the bearings in the test-rig was done carefully to protect the run-in film except in one case, for Oiles JM3.3, to investigate the influence of preserving the film or without preserving it since it can be difficult to preserve the run-in film in reality.

Table 1. Different bearing and test number, test profile, optional run-in and applied mean bearing pressure for each test. * Run-in film was intentionally partly scraped off using counter shaft surface. ** Sliding distance was increased.

Bearing-test number	**P-1**	**P- 2**	**Run-in**	**No Run-in**	**Bearing pressure (MPa)**
Oiles JM3.1	X			X	20.0
Oiles JM3.2	X			X	18.3
Oiles JM3.3*		X		X	20.2
Oiles JM3.4		X	X		20.0
Oiles JM7.1	X			X	20.6
Oiles JM7.2	X			X	19.9
Oiles JM7.3	-	-	-	-	-
Oiles JM7.4		X	X		19.8
Thordon.1	X			X	20.9
Thordon.2	X			X	20.2
Thordon.3		X		X	20.3
Orkot.1	X			X	20.2
Orkot .2	X			X	18.5
Orkot.3		X		X	19.9
DEVA BM.1	X			X	19.4
DEVA BM.2**	X			X	18.4
DEVA TEX.1	X			X	19.3
DEVA TEX.2**	X			X	20.0
PAN SOBz.1	X			X	19.6
PAN SOBz.2	X			X	19.5
PAN 12.1	X			X	20.1
PAN 12.2	X			X	19.4

All run test surfaces were photographed prior to and after the tests. Selected surfaces were further analysed in SEM-EDS (Scanning Electron Microscope – Energy Dispersive Spectroscopy).

3. RESULTS

For tests with copper based bearings, the shaft temperatures were about 15°C and for non-copper based bearings the shaft temperatures were 20-25°C.

3.1. Friction and linear wear

The Oiles 500.1-4 showed different behaviours for friction and linear wear, see Figure 2. The repeated JM3.1 and JM3.2 tests showed clearly different results i.e., higher friction and higher wear rate. The small motion test (Oiles JM3.3-4) showed significant step variation for the friction values, but no significant differences were seen when the run-in film was partly scraped off. Frictional variation behaviour was generally not coinciding with steps seen in the linear wear measurement. Small motion pattern showed lower friction and wear compared to larger motion pattern (Oiles JM3.1-2).

The Oiles 500 JM7.1-2 and 4 showed a spread of the friction curves and also variations in the linear wear, see Figure 3. Small motion pattern (Oiles JM7.4) showed more step variations. Negative wear can be realised by material transfer onto the shaft counter surface.

Thordon.1-3 showed the most consistent behaviour with small variations regarding friction and wear, see Figure 4. Negative wear can be partly due to material transfer but in this case it can also be influenced by initial run and shaft heating. The bearing material has low conductivity and owing to this the shaft runs at a higher temperature compared to that with copper alloy bearings.

For one of the Orkot.1-3 tests the initial friction and wear were higher compared to the other two tests and the friction decreased towards the end of the test, see Figure 5. Orkot showed the lowest friction values overall and no influence from applied motion pattern was seen.

DEVA BM.1 test showed low linear wear but increasing friction towards the end, see Figure 6. In the second test on DEVA BM.2, the running distance was increased and this showed that the friction varied slightly throughout the test. The linear wear had a high wear rate during the first 5 m sliding distance and a few step variations along the way. The wear rate was low. The DEVA TEX.1 showed a similar end of test friction increase which in the end of the second test DEVA TEX.2 shows change in curve slope. Wear rate was very low.

PAN SoBz.1-2 showed high but rather stable friction level and linear wear, see Figure 7. PAN 12.1-2 showed more differences between the two tests where one test showed increasing friction.

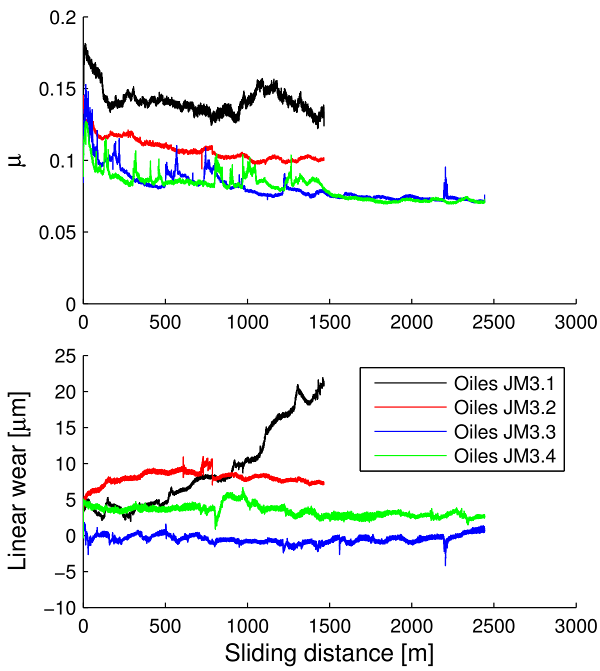

Figure 2. Friction and linear wear for Oiles 500 JM3 SL4, see Table 1.

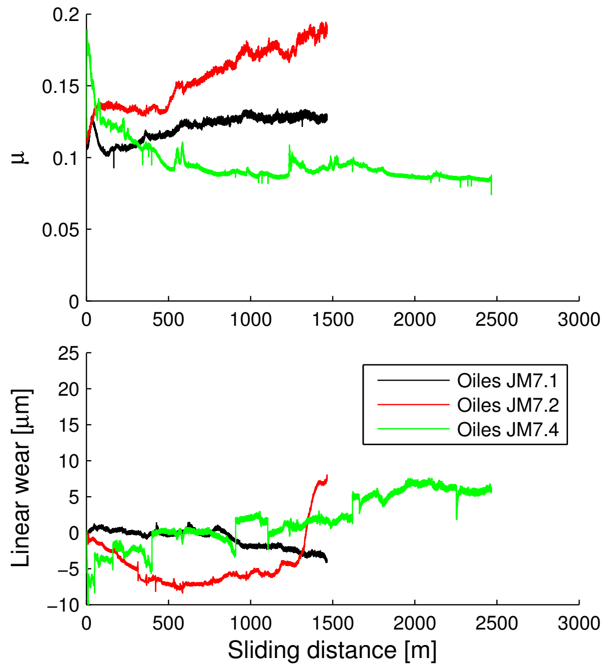

Figure 3. Friction and linear wear for Oiles 500 JM7 SL4, see Table 1.

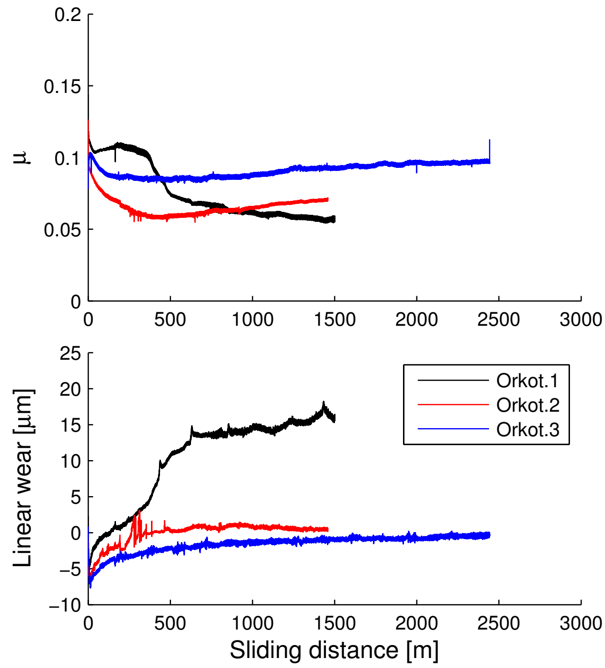

Figure 5. Friction and linear wear for Orkot.1-3, see Table 1.

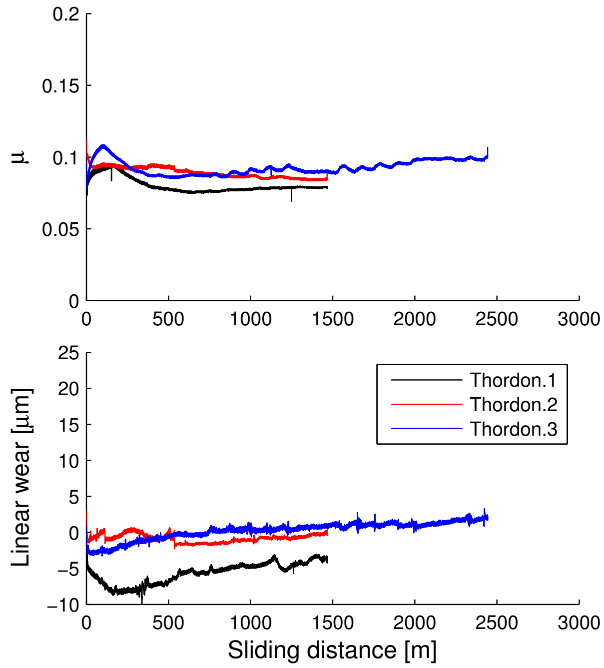

Figure 4. Friction and linear wear for Thordon.1-3, see Table 1.

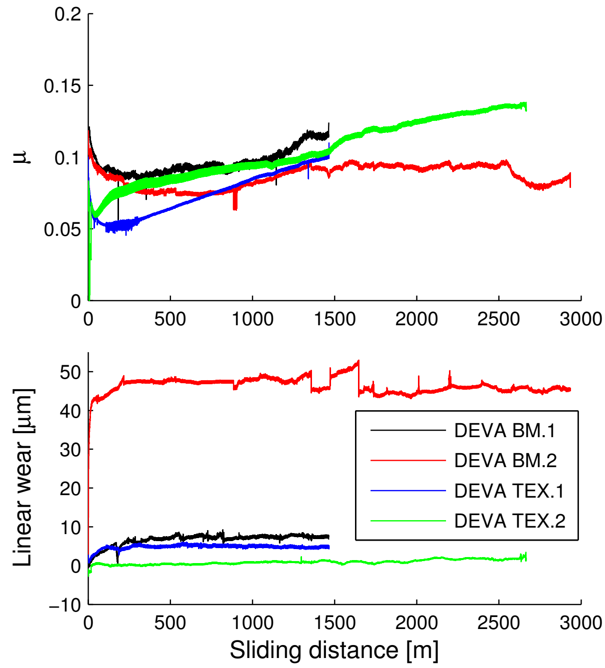

Figure 6. Friction and linear wear for DEVA BM.1-2 and DEVA TEX.1-2, see Table 1.

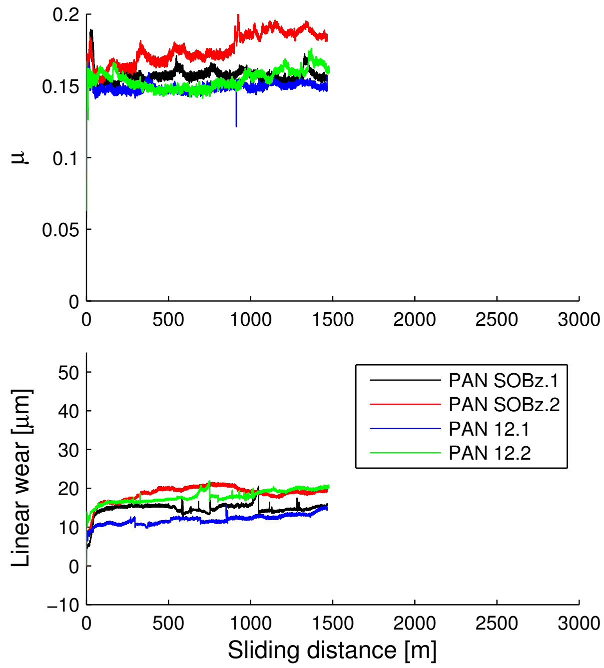

Figure 7. Friction and linear wear for PAN SoBz.1-2 and PAN 12.1-2, see Table 1.

3.2. Surface characteristics

Visual inspection of tested bearing specimens showed a different appearance for the Oiles 500 JM3 (and JM7) when running with different motion patterns. The bearings tested under small stroke oscillating motion pattern showed clearly different areas with variations of lubricant- and material transfer to the shaft compared to the more homogenous appearance of various motions test, see Figure 8 e)-f) vs. b)-d). The distinguished fields formed between and on lubricant plug position on shaft counter surface position becomes even clearer when studying Oiles 500 JM7, see Figure 9. Surface analysis using Optical microscopy showed that some particles attached to the shaft surface were more than 50 μm in height. Analysis using SEM-EDS showed that the largest particles were pure lead (Pb). The different fields seen in Figure 9 a) were found to contain lead (Pb), fluorine (F), copper (Cu), aluminium (Al), carbon (C) and iron (Fe). In some regions, no iron was found. The visual inspection from the Thordon and Orkot tests showed very similar appearance from both test cycles, see Figure 10. Shaft analysis showed that the contacting surface was similar to the original for the Orkot tests. The SEM-EDS showed that some areas contained flourine and other did not. The carbon content was always higher compared to the original shaft surface indicating material transfer. The shaft surface from a Thordon test showed small visual plastic transfer to the shaft. The surface analyse showed that the fields of plastic transfer showed different surface roughness values and characteristics compared to the original surface. The SEM-EDS indicated that the surface consisted of two fields with more or less plastic material (carbon) transferred to the shaft. The carbon content was significantly higher in the contact zone.

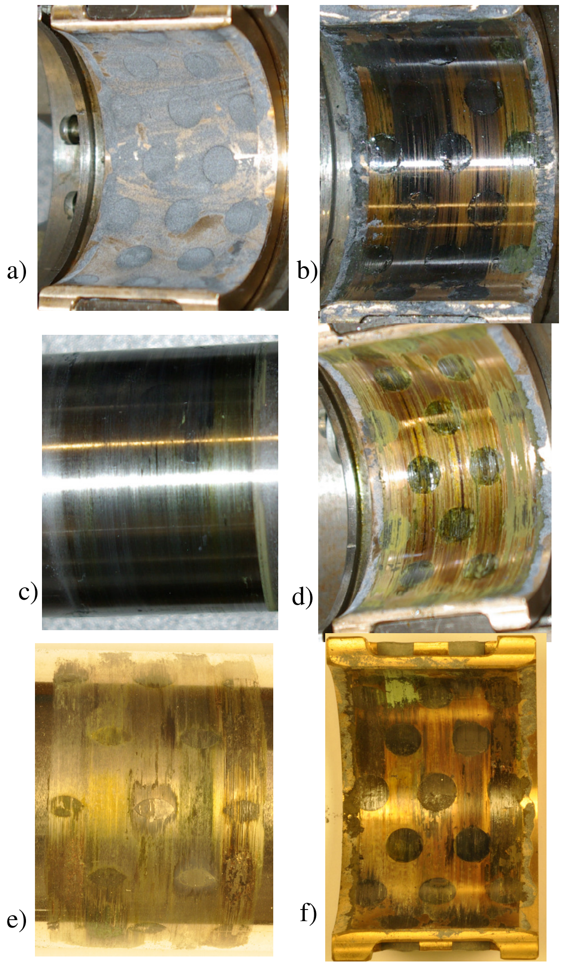

Figure 8. Photographs of Oiles 500 JM3 SL4; a) new test bearing with run-in lubricant, b) tested bearing (Oiles JM3.2), c) tested shaft (Oiles JM3.2), d) tested bearing (Oiles JM3.1), e) tested shaft (Oiles JM3.4) and f) tested bearing (Oiles JM3.4).

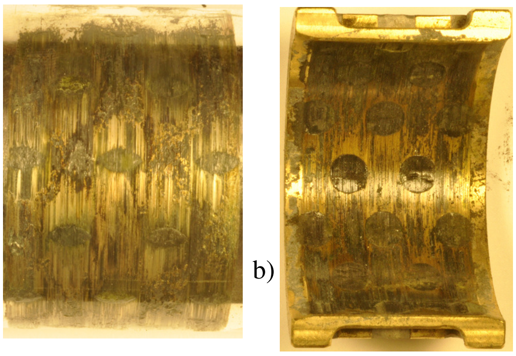

Figure 9. Photographs of tested Oiles JM7.4 a) shaft, and b) bearing.

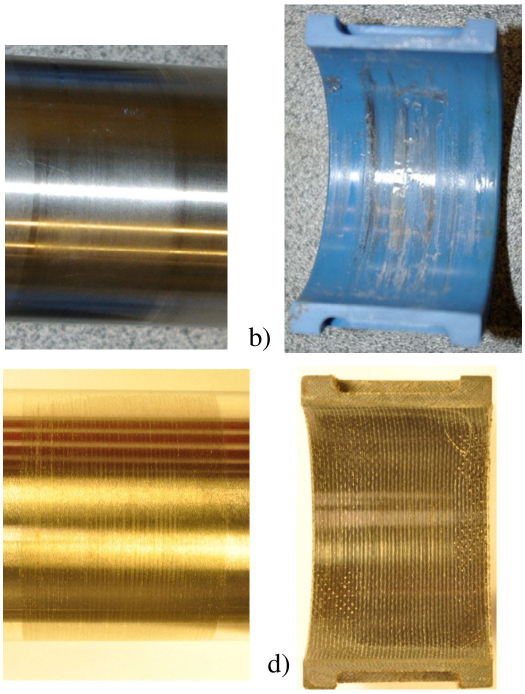

Figure 10. Shaft a) and bearing b) from Thordon test, and shaft c) and bearing d) from Orkot test.

4. CONCLUSIONS

In this work, the tribological performance and sliding surface characteristic differences using two different motion patterns using some selected bearing types have been investigated in detail. The results have shown that some bearing types show different performance when subjected to two different motion patterns. The most significant difference was seen for the variation of friction values. It was also clear that the bearing types showing different friction step variations also showed a visual appearance difference at the worn surfaces. Different fields were formed with different surface characteristics, with partly large lead particles and partly fields of different material composition.

Considering the stated conditions of small oscillating motion in hydropower wicket gate guide vane applications these tests contributes to the selection of suitable bearing types. The most suitable types are not sensitive to different motion patterns. In this study only 4 types were fully tested of which two types are considered to be more suitable. The suitability from motion pattern evaluation point of view of the other types is not answered in this study.

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