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# TEST RESULTS FOR PTFE-FACED THRUST PADS, WITH DIRECT COMPARISON AGAINST BABBITT-FACED PADS AND CORRELATION WITH ANALYSIS

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## ABSTRACT

*The use of PTFE- faced pads in large vertical axis hydro-generators was pioneered in Russia in the 1970's, prompted by a series of failures of conventional babbitt-faced bearings. Some advantages claimed include higher specific loading, lower power loss and the omission of oil-lift facilities. There is strong interest in the Industry concerning this material, but limited data are available on actual performance. Some results from extensive testing of PTFE-faced pads are given, for two sizes of pad. These are compared directly size-for-size with results for babbitt bearings of nominally the same area. The power losses for the two types of bearing were found to be almost identical. Some of the effects observed during testing are described and discussed, including the effect of creep. The test results are compared with predictions using the GENMAT analysis software. A method of allowing for creep in numerical modeling is discussed.*

## INTRODUCTION

PTFE (polytetraflorethylene) is usually classified as a material for use in dry sliding applications. The coefficient of friction against steel can be as low as 0.05, dependent on load and surface finish, and the rate of wear can be very low. There are numerous practical applications of the material, ranging from bushes for aircraft control linkages to prosthetic joints. The wear rate drops to a steady value when a transfer film has formed on the counter-surface. Lubricants were known to interfere with the formation of a transfer film, and grease or oil lubrication were rarely used in PTFE-metal contacts.

A major departure from use as a dry-sliding material has been the development in the 1970's of PTFE-faced thrust bearings in hydro generators, in the former Soviet Union and the Peoples Republic of China (PRC). The pads run fully immersed in mineral oil, and in many cases the only change made was to remove the babbitt from pads and replace it with PTFE. Prior to this application of PTFE (according to Alexandrov [1] and Shen [2]), the usual type of bearing had disk-supported babbitt-faced pads, which were quite lightly loaded as judged by modern standards. Following demands for increased power, nation-wide programs were enacted to upgrade hydro generators. Shen [2] gives an interesting account of the problems that occurred in China, for example 'break-down occurred in almost every hydroelectric unit through the 1970's'. The upgrade called for an increase of PV from (typically) less than 50 MPa.m/s to 55 – 91. The most usual problem was high temperature in the center and center-trailing areas of the pad, apparently due to excessive crowning. This led to wiping of the central areas of the pads and subsequent cascading damage from pad to pad as smeared babbitt from one pad broke away and entered the film of the following pad. Simmons *et al.* [3] give an extended review.

At present PTFE pads with loading as high as 10.2 MPa are in use, although most PTFE bearings are run at pressures only slightly higher than the babbitt bearings that were replaced. According to numerous reports (many of which are not formally published) the replacement with PTFE-faced pads 'completely eliminated the wiping-out problem' and allowed the development of bearings of 3000 – 4000 tonnes capacity. It is reported that at present over 80% of hydro generators in the former Soviet Union have this type of bearing, and at least 400 units in the PRC. These developments have attracted strong interest in the Industry in Europe and the Americas but few data are available. Some units in the UK and Ontario have been retrofitted with PTFE-faced thrust bearings, as described by Simmons *et al.* [3], Knox [4], and Mohino *et al.* [5]. However few direct comparisons have been made between PTFE-faced and babbitt-faced bearings. This paper gives size-for-size comparisons for two bearing assemblies with pads of approximately 130 mm x 140 mm size and 306 mm x 260 mm size. The PTFE and babbitt bearings were tested in sequence in the same test machines.

## Advantages claimed

In publications such as [1,2] and others, the advantages over conventional babbitt- faced bearings are described as:

1. Reliable operation to specific pressures of 10 MPa and above
2. Reduced power loss
3. Reduced thermal crowning of the pad, so that features used to reduce thermal deflection in babbitt-faced pads need not be applied
4. Reduced oil-film temperature. (As claimed in some publications)
5. No requirement for an oil-lift (jacking) system during starting and stopping
6. No scoring damage to the runner surface in the event of a failure
7. No dwell time required before re-starting
8. More relaxed tolerances on pad thickness and flatness

## Characteristics of PTFE

The creep properties and lower mechanical strength of PTFE require special attention in the design of PTFE-faced bearings.

The elastic modulus for the pure unfilled material is approximately 500 MPa at 22°C and a compressive stress of 5 MPa. The modulus drops to 10% of this value at 150°C and 35 MPa. Both of the bearings described in this paper were tested to 10 MPa or greater, so consideration of the local deformation of the pad surface at these conditions is relevant. Bearing #1 had a 5 mm layer of PTFE. If plane-strain is assumed, with a Poissons ratio of 0.46, the local depression of the pad surface in the central area at 150°C and 35 MPa could apparently reach 750 micron relative to the unstressed edges.

This exceeds the likely oil film thickness by a factor of 30 – 50. Local deformation of the surface would relieve the pressure, but at an average pressure of 10.2 MPa (the highest in commercial use in a hydro application), it is likely that the maximum local pressure would exceed 25 MPa. This would still give relative deformations that are an order of magnitude greater than the general level of film thickness. Surface deformations of a much greater relative magnitude occur in EHD contacts, but nonetheless if modeling of PTFE-faced bearings is to be attempted, close attention must be paid to elastic and plastic effects in the PTFE layer.

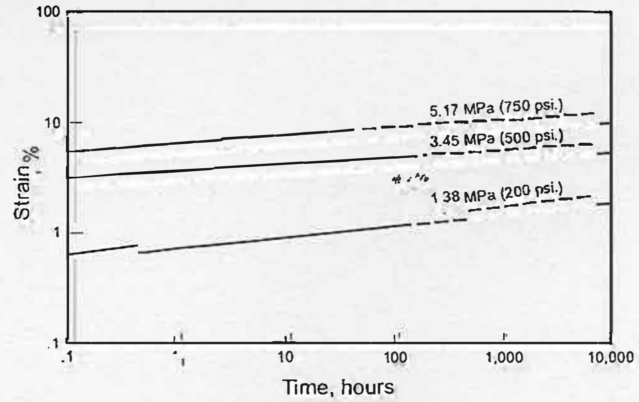


*Fig 1. Creep properties of pure PTFE in compression at 100°C.*

Figure 1 shows the creep rate of PTFE in compression at 100°C as found in tests such as ASTM D695-80. The highest compressive stress for which creep data could be found is 5.17 MPa (750 psi), which gives a creep strain of 10% after 200 hours. This implies that the film shape must adjust continuously as creep occurs. As an additional factor, the recovery of PTFE from compressive load is non-linear. At 100°C and a compressive stress of 15 MPa, the strain is 25%. The residual strain on release of the loading is 14%.

These difficulties were overcome in an ingenious production system developed in the 1960's for dry-sliding applications. PTFE powder was roll-impregnated into a 0.25 mm thick layer of tin-bronze that had been sintered onto a steel strip. The PTFE-metal bond was completed by re-sintering above the PTFE transition temperature of 327°C, which allowed the PTFE to seep into the interstices of the porous bronze, giving a strong mechanical bond. The thickness of the PTFE layer was limited to 0.12 mm to limit the extrusion of the material from the bush. Also, it was considered (by the inventors) that the bearing should be replaced if the wear exceeded this amount. The overall assembly gave the advantages of the low friction and wear of PTFE, with the mechanical strength of the steel backing. The composite strip could be used as thrust washers or swaged into journal bushes. The same general principle was used to bond the material in the test bearings.

## Bonding method used for thrust pads

The construction of the pads for bearing #1 is shown in Fig 2. A 5 mm plate of pure PTFE is clamped against a matrix of copper wire matting and the assembly is heated. PTFE is extruded into the matrix for about 1.0 – 1.5 mm giving a mechanical bond when cooled. The matrix is then soldered to the steel backing. The face of the finished assembly is ground and the tapers are added at the leading and trailing edges. Some flexibility is claimed for the copper matrix. When the pads were proof tested in simple compression at 13.7 MPa, the overall deflection was 0.100 – 0.150 mm. Using the properties of PTFE at room temperature and assuming plane-strain (with Poissons ratio = 0.46) indicates that about half of this deflection occurs in the PTFE layer. More recent techniques of production (Knox, [4]) use a thinner layer of filled PTFE, with the matrix being almost filled with solder. Assemblies using sintered tin-bronze on a steel base are also under development.

## Bearings tested

| *Size* | *Bearing #1* | *Bearing #2* |
|---|---|---|
| Mean pitch diameter, mm | 464 | 912 |
| Pad size: | | |
| Circum., on mean radius, mm | 133 | 306 |
| Radial, mm | 140 | 260 |
| Thickness of PTFE, mm | 5 | 2 |
| Overall thickness of pad, mm | 40 | 38.1 |
| Number of pads | 8* | 8* |
| Support system | Offset radial line | Set of 25 springs, offset |
| *also tested with 4 pads removed | | |
| *Test conditions* | | |
| Sliding speed (on mean radius) | to 41m/s | 8 pads,24 m/s; 4 pads,28 m/s |
| Specific loading | pads,10.2 MPa | 8 pads,5 MPa 4 pads,10 MPa |
| Lubricant, ISO | 32 | 32 |
| Bath temperature, °C | 40 - 70 | 70 |
| *Suppliers* | Electrosilia, Russia | Liao Yuan Scientific Institute, China |
| Principal references for test facilities | Simmons *et al.* [3] Horner *et al.* [6] | Yuan *et al.* [7] |

*Table 1 Bearing size and test conditions*

The size of bearing #1 is typical of a medium-large pump and smaller than a significant hydro application. The pads were too small to show the thermal deformation problems associated with large thrust pads, but were convenient for initial tests. The pads for bearing #1 as supplied were intended for use as a set of eight, but were also tested as a set of four to extend the range of specific load. Thermocouples were installed in the PTFE about 3 mm from the surface but, as will be shown, these gave no indication of the temperature in the oil film. Later in the program some of the thermocouple holes were drilled straight through to the surface, with the beads of the thermocouples positioned to be approximately 3 mm from the surface. These were found to give a more representative measure of film temperature.

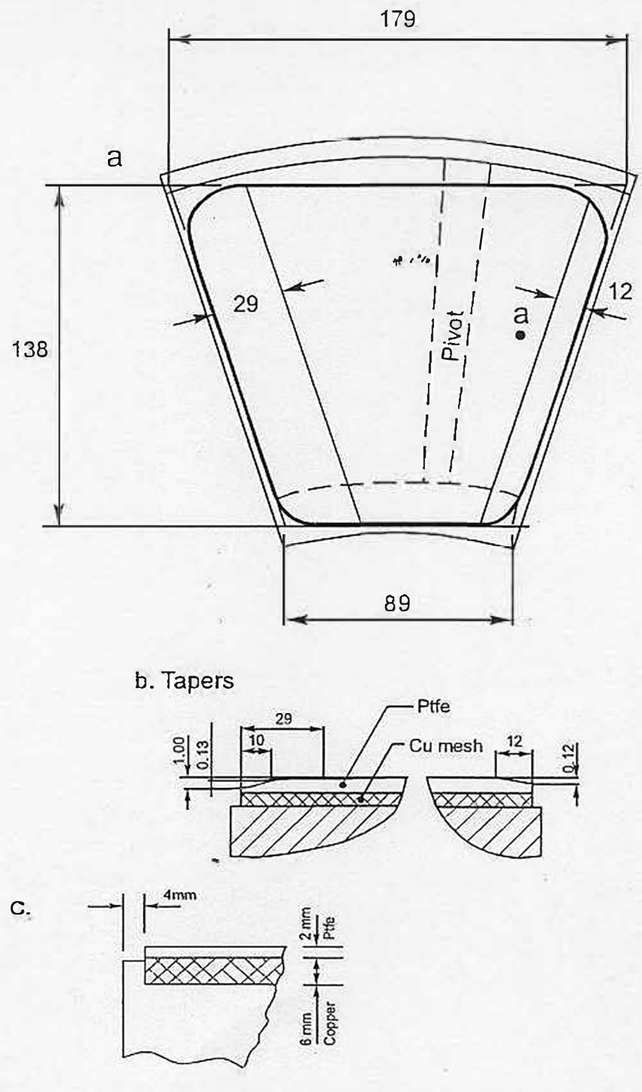


*Fig 2. a. Dimensions of pads in bearing #1*
*b. Tapers on lead and trail edges*
*c. Edge support in bearing #2*

The pads were supplied with extensive tapers on the lead and trail edges, as shown in Fig 2 a,b. These are much larger than would normally be considered necessary and are presumably to assist in the formation of a film, especially at start-up. Usually only leading and trailing radii are necessary. However the trailing edge taper appears to perform a useful function in supporting the trailing edge, as is discussed later. The area of the first taper on the leading edge and the trailing taper were not considered when calculating the specific pressure on the pad face.

The tolerance on dimensions and flatness of the pads of both bearings appeared to be much looser than for babbitt bearings. In bearing #1 two of the pads were supplied with sets of five concentric rings pressed into the face of the pads (near the corners) with a special tool. The depth varied from 100 micron (outer ring) to 180 micron (inner ring). These were explained to be wear indicating devices, allowing pad wear to be determined by observing how much of the rings were worn away.

The babbitt faced pads used for comparison were made to identical dimensions as the PTFE-faced pads, with the same tapers on the face. As with the PTFE pads, when calculating the specific pressure, the trailing taper and the first section of the double leading taper were neglected. The power loss was calculated from the flow and temperature rise of the oil supply, and includes losses in the support and reaction bearings. About 85% of the overall losses can be assigned to the test bearings.

### Accuracy of measurement

The uncertainty of measurements for temperature was one degree. The absolute error in power loss measurement was estimated as 5%, although the relative error between bearing types was taken as 3%.

### Analysis model

The computed results were obtained using the GENMAT software described in references [8,9].
Some of the features included are:

1. An integrated treatment of heat transfer in the film, pads and rotor in three dimensions, which included the presence of the PTFE and copper matrix layers
2. Thermo elastic deflection of the pads and deformation of the surface layer. (The deformation of the surface layer was allowed for by treating the PTFE and copper mesh in plane-strain, with Poissons ratio ν = 0.46. The local influence of stress and temperature on the modulus of the PTFE layer was included. Some of details the material properties are given in an appendix to this paper)
3. Super-laminar and turbulence effects in the film
4. The inclusion of hydrostatic oil lift
5. Hard support of the pads, for example on line or disk supports, or on an arbitrary arrangement of springs (as for bearing #2)
6. "Carry-over" of hot oil between pads is allowed for by solving the turbulent boundary layer equations in the grooves between pads
7. Thermal bowing of the rotor is included

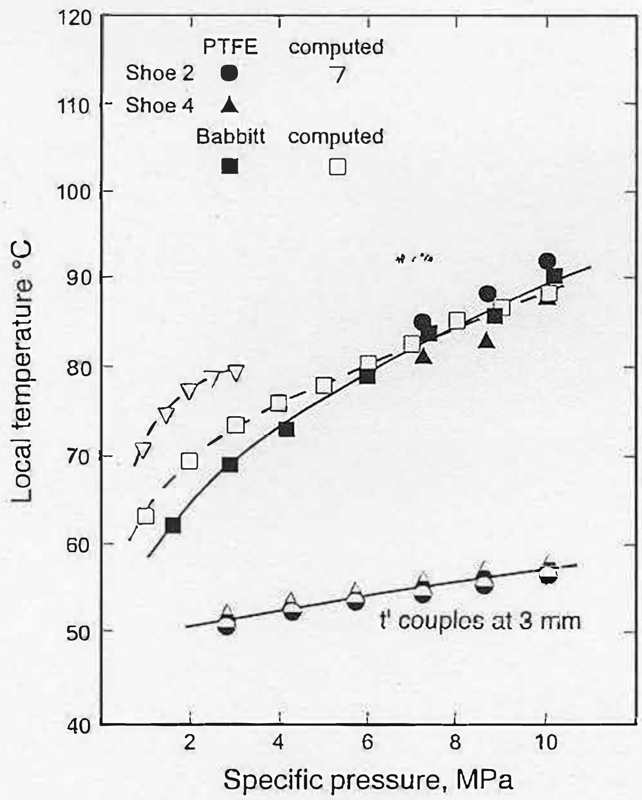


*Fig 3 a. Measured and computed temperatures in bearing #1 at location 'a' (Fig 2a) at the 57%, 84% radial/circumferential position. Speed 800 rpm, bath temperature 49 – 51° C.*
*Results for location 'a' with the thermocouple encapsulated in PTFE are shown as half-filled symbols, in the range 51 – 58° C*

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