PTFE-Faced Journal Bearing Technology for Marine Propulsion Applications
Technologie des Coussinets à Patins Revêtus de PTFE Appliquée à la Propulsion Marine

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This paper presents the results of a test programme in which a 500 mm diameter, PTFE-faced, segmented pad journal bearing was tested at both low speed and high axial misalignment. The bearing consisted of 8 PTFE lined journal pads, each 270 mm long and was subjected to specific loads up to 2.5 MPa (LxD), sliding speeds in the range 5 to 100 rev/min (0.13 – 2.6 m/s) and axial misalignments up to 1.0 mrad. The test results show that the bearing was able to tolerate combinations of low speed and high misalignment that would not be considered suitable for a traditional whitemetal bearing.

Cette communication présente les résultats d'un programme d'essais aucours duquel un coussinet à patins de 500 mm de diamètre, garni de PTFE, a été testé à basse vitesse et avec un délignage axial simultanément. Le coussinet était composé de 8 patins recouverts de PTFE, de 270 mm de long chacun, soumis à des charges spécifiques allant jusqu'à 2,5 MPa (LxD), des vitesses de glissement dans l'intervalle de 5 à 100 rev/min (0,13 – 2,6 m/s) et des délignages axiaux allant jusqu'à 1,0 mrad. Les résultats d'essais montrent que le palier a pu supporter des combinaisons de basses vitesses et délignages élevés qui ne seraient pas considérées comme adaptées à un palier "régule" (ou métal blanc) traditionnel.

1 Introduction

The purpose of this paper is to present the results of an experimental investigation into the performance of a 500 mm diameter PTFE (polytetrafluoroethylene)-faced, segmented pad journal bearing under aligned and misaligned conditions. The experimental programme is based on duty conditions that would be commonplace in marine propulsion applications, including operation at very slow speeds and under overload conditions that, for conventional whitemetal-lined bearings, would normally require the use of high pressure oil injection.

Propulsion shaft bearings used in marine applications are required to operate across a wide speed range with running at very low speeds being routine. At the same time these bearings often have to accommodate axial misalignment of the shaft through the bearing. This misalignment is not constant but varies due to component wear and hull flexure arising from loading and sea state. A common bearing design employed to cater for misalignment is a plain journal bush with a spherical outer diameter. This spherical diameter allows the bearing surface to be aligned during installation, but its ability to accommodate changes in shaft alignment thereafter is restricted due to friction inhibiting the movement of the bush inside the bearing casing. Thus, the ability of a journal bush to accept misalignment in operation is mainly determined by the thickness of the lubricating oil film between the shaft and bearing surface. Under slow shaft speed conditions the oil film is frequently so thin, that plain bushes offer very limited ability to accept misalignment. This can mean that sometimes bearing realignments are necessary through the life time of the vessel to maintain satisfactory alignment between bush and shaft. Other measures often employed at the design stage are the use of very modest bearing specific loadings and hydrostatic high pressure oil injection to supplement the hydrodynamic oil film a low shaft speeds.

An established design alternative to the spherical bush is the use of segmented, tilting pad journal bearings which have been used for many years to support propulsion shafts in a wide variety of seagoing vessels. These bearings are comprised of separate pads each with a pivot that allows the pad to tilt relative to the rotating surface of the shaft to form a convergent, load-carrying oil film. One of the main arguments cited in favour of such bearings for marine applications is their ability to accommodate changes in misalignment during the life of the vessel without the need for bearing realignment. The suggestion is that it is the flexibility of each individual pad which produces the tolerance to shaft misalignment.

Much of the published work into the effects of journal bearing misalignment concerns either bushes and/or higher speed machinery [1]. By contrast, notwithstanding the claims made on their behalf, there has been little prior systematic investigation into the misalignment capability of segmented journal bearings at low operating speeds.

The bearing reported in this paper uses PTFE as the pad facing material. Whilst the use of PTFE as a hydrodynamic thrust bearing material for major industrial applications is now well documented, especially for hydro-power applications, its use in journal bearings, despite some recent publications [2, 3] is much less well reported. It is known, however, that segmented PTFE journal bearings have been operating successfully in the field for some time at specific loads of up to 2.9 MPa [4].

Previously, PTFE has most frequently been deployed in hydro-power installations where the potential advantages of higher thrust load, reduction in power loss and the possible elimination of high pressure oil injection at start up have been regularly cited. In marine applications, however, it is the potential reduction in overall bearing size, made possible by a higher specific load, together with a more durable material and the potential of PTFE to accommodate misalignment that makes it an attractive option compared with traditional whitemetal (Babbitt). Filled grades of PTFE have also been shown to be a better design choice than either pure PTFE or whitemetal for both wear resistance and breakaway friction at start-up [5]. This is of significant importance for marine propulsion bearings where frequent start-ups and shut-downs are a feature of normal operation.

The ability of PTFE-faced thrust pads to operate where there is some misalignment across the face of the pad has been reported previously [6] in a case where the sliding speed was 24 m/s and the oil film thickness relatively generous. In marine propulsion applications the sliding speeds involved are generally less than 5 m/s meaning the inherent capability of the oil film to accommodate any misalignment is much reduced.

2 Test Equipment

2.1 Test Bearing

The test bearing is shown in figure 1: dimensional and other details are given in table 1. The bearing consists of a set of eight PTFE-faced and steel-backed journal pads supported in a cast iron casing. A 15% carbon and 2% graphite filled PTFE layer forms the working surface of each pad and is part of a PTFE/copper wire composite that is soldered to the steel backing of the journal pad as previously reported [7]. The journal pad PTFE face design incorporates chamfers at the leading and trailing edges. The transition from chamfer to main face is clearly visible as a straight line along the length of the journal pad surface. This line becomes less distinct after operation, particularly after running at high specific loads and with low oil film thicknesses. The bearing components were arranged circumferentially so that the load, acting vertically, was equally shared between the bottom two pads (figure 1). The journal pads were lubricated by oil fed from a circulating system into a central annulus in the casing and thence to the PTFE working faces via the gaps between the pads. The oil flows out from the ends of the journal pads and goes to drain at the bottom of the casing.

The bottom two journal pads were fitted with thermocouples as shown in figure 2 to monitor the temperature of the oil film. The thermocouples were embedded in brass inserts which were fixed into the journal pads so that the tip of each brass insert was set 1 mm back from the working surface. Nine thermocouples were positioned along the axial length of each instrumented pad, in line with the trailing edge of the pivot. Two thermocouples were placed on the axial centre line of the pads, 25% and 75% respectively from the leading edge. The thermocouple at the 75% position in the trailing half of the pad on the upward rotation side of the shaft used a design of brass insert previously used for monitoring the operating temperature of thrust pads. Oil inlet temperature, oil outlet temperature and oil temperature between the bottom journal pads were also measured.

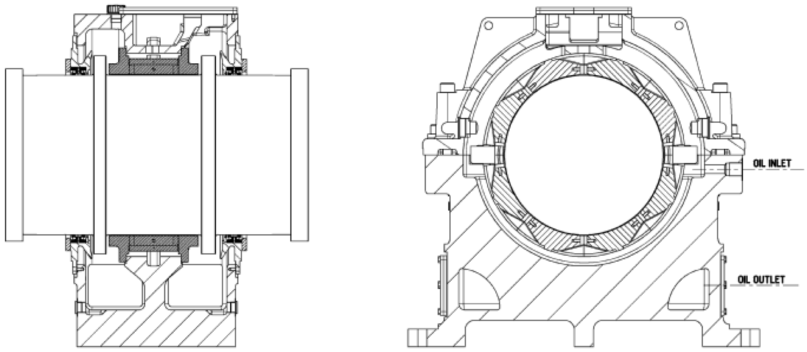

Fig 1 – Cross Section through Test Bearing

Bearing Diameter	500 mm	Oil Viscosity	ISO VG 68
PTFE surface axial length	270 mm	Oil Flow	600 l/hour
Pivot length	148 mm	Oil inlet temperature	45 +/- 2 ^{o}C
PTFE subtended angle	35^{o}	PTFE Surface Finish (Ra)	0.8 μm
Nominal pad thickness	68.5 mm	Shaft Surface Finish (Ra)	0.5 μm
Diametric Clearance	0.67 mm	Cylindricity of PTFE Bore	0.015 mm

Table 1 – Dimensional Details of the Test Bearing

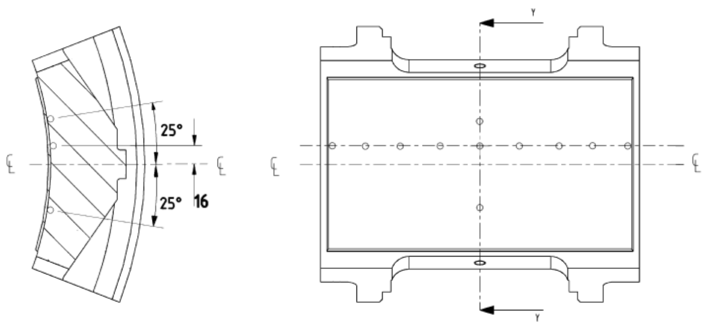

Fig 2 – Thermocouple Positions

2.2 Experimental Arrangement

The test rig arrangement is shown in photograph 1 and in diagrammatic form in figure 3. Motive power was provided by a DC electric motor capable of driving the shaft at speeds of up to 200 rev/min and as low as 1 rev/min with the addition of a suitable gearbox. The test bearing (orange in photograph 1) was situated on a floating frame sitting between two 300 mm diameter support bearings which were mounted onto the main base plate. The journal load was applied directly on the axial centreline of the test bearing using a pair of hydraulic pistons situated one each side of the shaft. The pistons applied the load through the floating platform, forcing the test bearing upwards against the shaft which in turn reacted against the upper journal pads in the support bearings. The uppermost two journal pads in each support bearing were fitted with high pressure oil injection to facilitate slow speed running.

The required axial misalignment in the vertical plane was applied by using a third hydraulic piston to tilt the floating frame upon which the test bearing was mounted. Misalignment was monitored and checked using proximity probes mounted on the test bearing casing and also using dial indicators. The action of the applied misalignment is to move the centre of hydrodynamic pressure towards one end of the loaded pads, coupled with a reduced operating film thickness at that end. Hence the misalignment produces a more heavily loaded pad end (at the non drive end) with thinner film thicknesses in comparison to the other more lightly loaded end of the pad.