Michell Bearings

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ADVANCES IN BEARING DESIGN FOR REMOTE HYDRO POWER LOCATIONS

J. E. L. Simmons, Consultant
Vickers plc, Michell Bearings
Newcastle Upon Tyne, England

ABSTRACT

This paper describes the design and experimental investigation of a new, high speed thrust and journal bearing for hydro power applications. The experimental work covered tests in which the bearing was operated in a self-contained mode using water cooling as well as in a conventional external circulating oil arrangement. A range of test conditions were applied substantially in advance of those available previously for a bearing of this sort and in particular very high machinery overspeeds were simulated sucessfully. This work opens up the possibility of providing substantial hydro installations free from external lubrication systems. The attendant cost reduction and maintenance implications of this are considerable and relevant especially in those more distant locations where human and mechanical services are not readily available.

1. INTRODUCTION

There have been numerous experimental studies published over many years concerning the behaviour of hydrodynamic bearings. In the main these reports have considered radial loading (journal bearings) or axial loading (thrust bearings). In practice, however, the majority of bearing designs are configured to accept a combination of radial and axial loads. It is perhaps unfortunate therefore that this class of combined bearings should be poorly represented in the literature.

This paper covers part of a recent extensive experimental programme to investigate the performance of a new, large, high speed thrust and journal bearing which incorporates some innovative features and is designed for a range of demanding industrial applications such as hydro-generators and water turbines. Because of the very considerable horizontal and vertical loads involved and the need to apply these independently a completely new test rig was devised which is described in the paper.

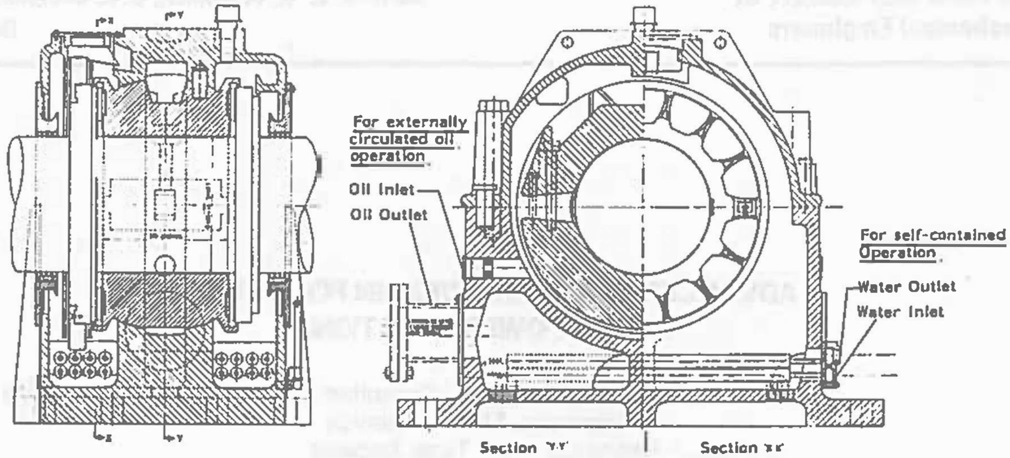
2. TEST BEARING

Basic dimensional information about the test bearing is given in Table 1 and the main features shown in Figure 1. The 400 mm shaft has twin collars to accommodate axial loading which may be applied continuously in either direction. The journal bush is situated between these collars. Tilting thrust pads are mounted in the ends of the bush abutting the loading surfaces of the two collars. The bush itself is located within the bearing casing in a spherical seating which allows accurate alignment with the shaft on setting up. Figure 2 shows the journal bush and thrust pads when removed from the bearing and Figure 3 is a view of the assembled bearing with the casing top removed showing the bush in position between the two thrust collars.

The bearing casing, horizontally split on the shaft centre line, is designed for strength to absorb with safety very large horizontal and vertical working loads. In this experimental programme the maximum values applied were 478 kN and 293 kN respectively. Oil is kept from leaking from the bearing even at high speeds by specially designed, multi-barrier baffles. These baffles are mounted on the casing and can be adjusted to suit the shaft.

When the bearing is self-contained, the oil inlet and outlet shown in Figure 1 are not used. Oil is circulated in the bearing by means of a pick up ring mounted on one of the collars and partially immersed in the sump. Certain features are provided on the inner surfaces of the upper and lower half bearing casings in the region of the oil pick up ring to counteract the tendency for oil to be thrown from the circumference at high speeds. In this way ample amounts of oil were delivered at ring surface speeds much greater than is usually quoted for fixed ring lubrication of this sort.

Oil adhering to the pick up ring as it rotates is diverted at the top of the bearing by a deflector to oil ways leading to the journal bush in the conventional manner. It should be noted however that the deflector is not in contact with the oil ring but separated from it



400 mm SHAFT DIAMETER EXPERIMENTAL THRUST AND JOURNAL BEARING

FIGURE 1

by a small gap. There is therefore nothing to cause wear in either component. Provision is made so that some of the oil delivered from the pick up ring is available to the bush surfaces immediately on start up and there is thus no risk of an oil starvation failure in the first few revolutions of the shaft before the full internal flow has become established. The bush surface is machined to allow some of the oil to pass directly to the ends and provide a satisfactory supply for the thrust faces. Oil squeezed from the ends of the journal combines with that in the regions of the thrust bearings and from there falls back into the reservoir in the lower half casing.

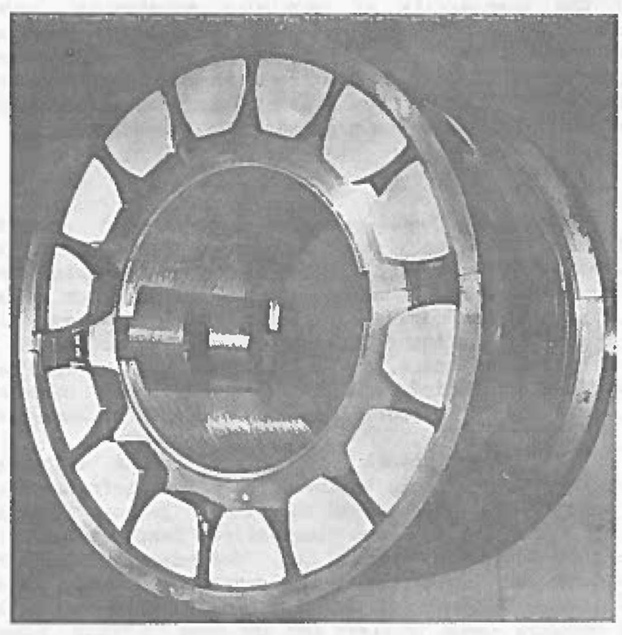
Heat generated in the bearing is absorbed by two multi-tube water coolers immersed in the oil of the bearing sump. The coolers are made from copper alloy tube with external wire windings to give a good rate of heat transfer from the oil. In the experimental work described later in this paper, water was provided to these self-contained bearing coolers at a temperature of about 15°C. There is however considerable design latitude in this figure and substantially warmer cooling water temperatures are acceptable providing sufficient quantities are available.

TABLE 1

Journal diameter	400	mm
length	370	mm
diametric clearance	0.50	mm
Thrust Pads outside diameter	685	mm
inside diameter	491	mm
surface area per thrust face	138,600	mm²
Oil pick up ring outside diameter	860	mm
Casing maximum dimensions		
height	1,150	mm
axial length	875	mm
width	1,440	mm
Oil grade	ISO VG 46	

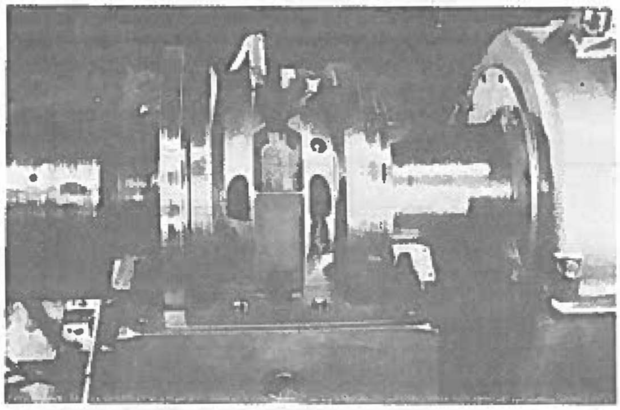
TEST BEARING DETAILS

For operation with an external supply, constant temperature oil is supplied via the inlet (Figure 1) to an annular channel immediately behind the bush and thence to the journal and thrust faces in the same way as for self-contained working. A suitable, constant oil level is maintained in the lower half casing by means of a weir incorporated in the large bore oil outlet. The oil ring remains in place to provide alternative circulation and protection for the bearing both during start up and run down in the event of a supply failure. for example in the rundown period following a supply failure. This is an important feature in ensuring the safety of the bearing under all circumstances.



400 mm JOURNAL BUSH AND THRUST PAD ASSEMBLY REMOVED FROM BEARING CASING

FIGURE 2



ASSEMBLED BEARING WITH CASING TOP REMOVED

FIGURE 3

The mode of operation with an external oil supply described here is known conventionally as circulated oil lubrication and is distinct from a third possible mode, forced lubrication, not covered in the present series of experiments. In forced lubrication oil is also supplied to the bearing from an external source but is delivered under pressure directly to the clearance between bush and shaft.

3. EXPERIMENTAL APPARATUS

The usual procedure for testing a journal bearing is to mount the bearing between two fixed support bearings and apply an external load, for example by hydraulic jacks between the base of the test bearing casing and the bed plate. In the case of thrust bearings, load is commonly applied by means of hydraulic pistons acting behind the thrust ring of a second bearing opposing that under test. The reactions of the two bearings are balanced by bolting both casings to a common bed-plate. For a combined bearing where radial and axial loads have to be applied the situation is more complicated because the need to apply the radial load to the casing prevents it being fixed to the bedplate. It is therefore necessary to provide different means for absorbing the reaction to the thrust load. In the past smaller horizontal thrust and journal bearings have been tested at Michell Bearings' experimental facility in Newcastle upon Tyne using a system of temporary axial stays positioned between the test bearing and that being used to apply the thrust load. In the present case the subject bearing was much bigger than any similar bearing examined previously and the loads to be applied were very large as Table 2 indicates. Consequently an alternative solution was sought and this resulted in the experimental apparatus shown in Figures 4 and 5.

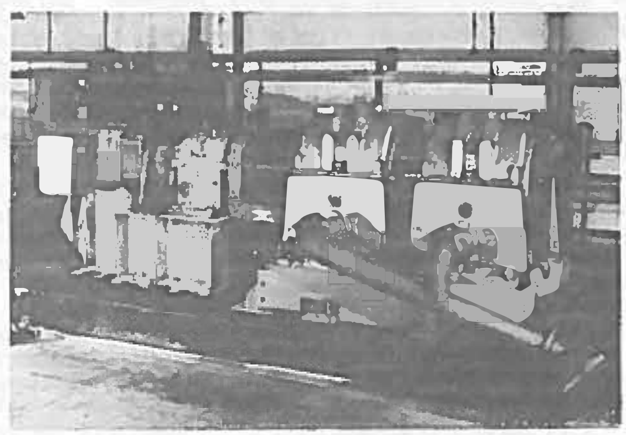
In the new arrangement the experimental bearing is mounted between two support bearings as usual. One of the support bearings, a 280 mm diameter, self-aligning journal, and a variable speed DC drive motor are mounted on a fabricated structure which is itself rigidly bolted to a massive cast bed-plate. The experimental bearing and a second thrust and journal bearing of the same size are mounted together on a second fabrication which is secured to the bedplate by a hinged connection. The second 400 mm diameter bearing is positioned directly above the pivot as Figure 5 shows. The purpose of this latter bearing is to act as the second shaft support bearing and also to be the means by which axial load is mposed on the test bearing in the conventional way. adial load is applied by two hydraulic jacks positioned etween the hinged fabrication and the bed-plate immediately below the experimental bearing. The test rig is designed so that thrust and journal loads can be applied independently or at the same time.

Motive power for the shaft is provided by a 300 KW DC motor with a thyristor drive giving continuously variable speed control up to 1500 rpm. Restrictions on available torque meant that it was not possible to start the system up under load. Thus experimental practise was to drive the shaft to the speed of a particular test and then to apply the loads as required. It can be noted, however, that the experimental bearing is designed with internal lubrication arrangements, described earlier, which operate very quickly when motion of the shaft commences. This provides the bearing, like other similar bearings, with a considerable potential capacity for starting up under load.

TABLE 2

Combined Load Condition	Absolute Load, N Thrust	Absolute Load, N Journal	Specific Load, MPa Thrust	Specific Load, MPa Journal
1	478,000	292,500	3.45	1.98
2 (3/4 Condition 1)	358,500	219,400	2.59	1.48
3 (1/2 Condition 1)	239,000	146,300	1.72	0.99

LOADING DETAILS

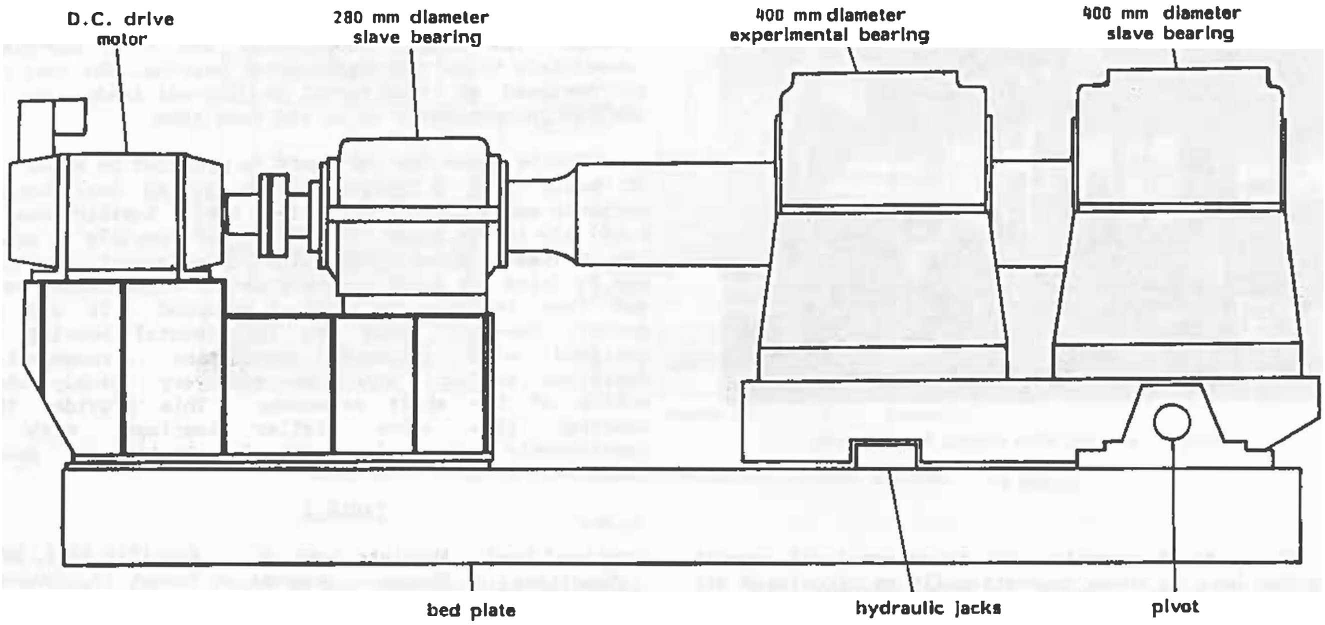


EXPERIMENTAL ARRANGEMENT

FIGURE 4

4. INSTRUMENTATION

Temperatures were measured in the bearing using stainless steel clad thermocouples. Six out of the total of 14 pads on the loaded face of the test bearing had thermocouples embedded in the trailing quadrant, "hot spot" position as defined by Horner, Simmons and Advani (1). The thermocouples were positioned 3 mm from the whitemetal surface. Three thermocouples were embedded a similar distance beneath the surface of the journal bush halfway along its length at the positions shown by Figure 6. Oil temperatures in the bearing were measured in the sump just beneath the oil pick up ring and in the oilway leading to the journal.



SCHEMATIC DIAGRAM OF THE EXPERIMENTAL ARRANGEMENT

FIGURE 5