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Air Cooled Bearings for Use with High Speed Electric Motors

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All hydrodynamic bearings require a continuous supply of cooled lubricant. In the case of large, high speed electric motors for industrial applications, oil is most commonly provided by an external lubrication system design to API or equivalent specifications. This paper describes the design and some of the experimental work done in the development of a new range of self contained, air cooled bearings designed to eliminate the need for such costly external lubrication systems. Experimental work was carried out using a 140 mm shaft diameter test bearing operating at a range of duties up to a maximum specific load of 2.07 MPa and a maximum speed of 4500 rev/min. Measurements made include those for oil delivery against speed utilising an innovative, dual mode oil ring described in the paper.

1. INTRODUCTION

Hydrodynamic bearings are fitted to high speed, horizontal shaft, electric motors for a very wide range of industrial applications in preference to rolling element bearings for reasons of longer life and greater reliability. In every application, satisfactory hydrodynamic bearing performance requires ensuring an adequate system for oil supply to the working faces and managing the disposal of heat generated in the bearing. Typical smaller and lower speed machines use an oil ring, which may hang loosely on the shaft or be fixed to it, to generate oil circulation within the bearing, and fins on the casing to dissipate the heat generated. The effectiveness of the oil ring and the limitation of casing surface area are obstacles to extending the simple oil ring concept to larger and faster machinery. For example, difficulties due to high ring speeds and a reduction in oil delivery rate start to appear in two pole motors at shaft diameters above 100 mm. The problem may well be exacerbated if the surface area of the casing is restricted by a centre flange mounting.

In previous designs it has been usual when these oil delivery and casing surface area limits have been encountered to employ a completely separate oil conditioning system to provide cooled lubricant to the bearing. Oil conditioning units, combining the functions of oil circulation and cooling, are generally provided in accordance with API specification 614 which implies the design of complex systems to ensure a reliable supply of oil. The purpose of this paper is to describe the design and underpinning experimental work behind the development of a new range of completely self-contained, air cooled bearings which extend the envelope of machinery size and speed which can be air cooled without the costly overhead of external oil conditioning systems.

The internal circulation system for the new bearings makes use of an innovative dual mode oil ring in which oil is collected *both* from the outside of the ring at start up and run down *and* from inside the rim of the oil ring at operating speed. Oil collected in this latter fashion is conveyed, under self generated pressure, to an external air blast cooler before being returned to the bearing, still under pressure and in quantities equivalent to those provided by a conventional lubrication system. Figure 1 shows a schematic diagram to illustrate the principle of the dual mode oil ring. Details of its embodiment in a practical bearing design are given in the next section.

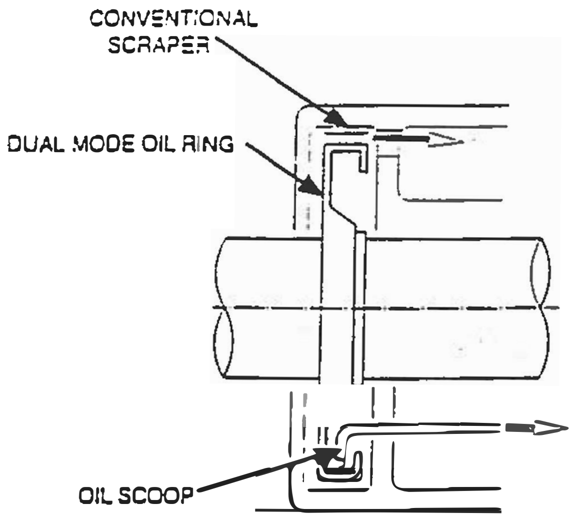

Figure 1. Schematic diagram showing dual mode oil ring

2. BEARING DESIGN

The main features of the test bearing are shown in Figure 2 with basic dimensional and operating duty information in Table 1. The overall size is comparable to that of bearings currently used for medium and large electric motors with a similar conservative loading of the white metal surfaces. The operating speed however is far beyond that of a conventional air cooled design. In the experiments described in the following sections it was demonstrated that the design load and speed combinations given in Table 1 are in fact well within the safe operating scope of the new bearing.

The primary duty for this class of bearing is to support radial load, normally the weight of a machine's rotor. This load is carried by a white metal (babbitt) lined journal bush supported in a spherical seating to allow accurate alignment between bush and shaft on setting up. An important secondary requirement in some cases is to provide a location capability in either axial direction. The units which require this feature are equipped with plain thrust faces, machined on the end of the bush, which act against corresponding collars on the shaft.

The bearing casing is split on the horizontal centre line for assembly purposes and designed with a flange mounting for attachment to an electric motor or other high speed machinery. Lubricant is kept from leaking from the bearing, even at high speeds, by non-contacting aluminium labyrinth baffles. The baffles are mounted on the ends of the casing and can be adjusted to suit shaft position.

Oil outlet and inlet ports, shown on the lower half casing, are used for pipework connections to a small external oil/air heat exchanger which may be positioned to suit the cooling system of the overall machine. In the case of an electric motor it is usual to position the heat exchanger close to the motor's cooling fan.

Table 1
Experimental Bearing: Dimensional Details and Design Duty

Journal diameter		140 mm
	length	85 mm
	diametric clearance	0.24 mm
Axial location:		
	outside diameter	185 mm
	inside diameter	145 mm
	surface area	9335 mm^2
Oil ring outside diameter:		400 mm
Casing dimensions:		
	flange diameter	620 mm
	axial length	290mm
Design duty		
	shaft speed	3600 rev/min
	journal load	16.4 kN
Lubricant		ISO VG 32 turbine oil
Cooling air		40°C at 10 m/s
Heat Exchanger		Serck BE11 25

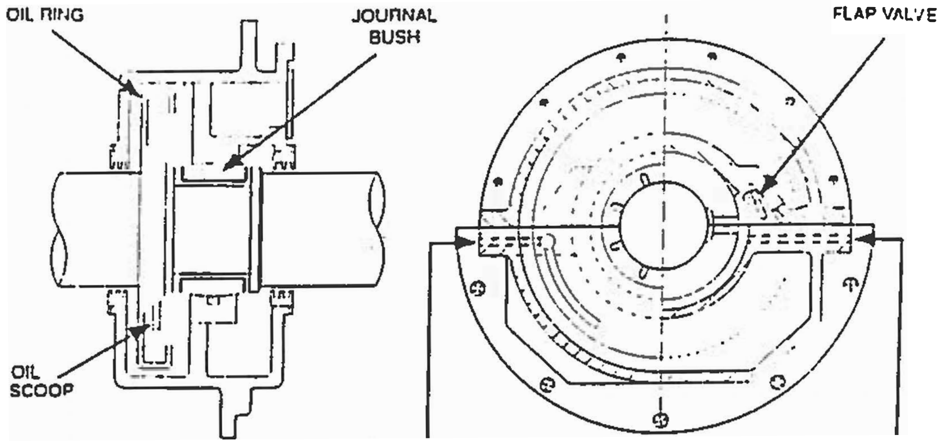

Figure 2. Test Bearing

The dual mode oil ring is mounted on the shaft adjacent to one of the location collars. Operation of the oil ring in supplying lubricant to the working faces and driving oil around the cooling circuit of the bearing is as follows. In the standing condition, prior to shaft rotation, the oil reservoir in the lower half of the casing, the heat exchanger and connecting pipe work are filled with oil. At start up oil is picked up on the external periphery of the oil ring and immediately conveyed to the top of the bearing where it is deflected by a scraper built into the interior of the upper half casing. The lubricant flows through ports in the upper casing to supply the annulus on the exterior surface of the bush and thence to oil ways leading to the working surface in a conventional fashion. Simultaneously, some oil from the reservoir adheres to the inner periphery of the oil ring. As speed increases a proportion of this oil on the inner surface is collected by a specially designed scoop positioned in the lower half casing. Conversion from dynamic head to pressure head at the scoop provides the impetus for driving lubricant around the cooling circuit without the need for any additional moving parts. Oil emerging from the outlet port passes via pipework connections to the heat exchanger and back to the inlet port shown on the illustration. From this point the oil flows directly to the annulus behind the bush from which the working surfaces are supplied. The effect is not only to provide the bearing with a continuous source of cooled oil but also to close the important flap valves fitted to the oil ports in the upper half casing thus pressurising the feed to the bush. A second result of closing the flap valves is the retention of a volume of oil in the upper half casing which is available to flow through the bush during shut down as the shaft comes to rest.

In normal, steady state operation oil which has passed through the working surface of the bush drains back to the lower half casing where it is gathered on the inner surface of the oil ring and collected by the oil scoop to complete the cycle and provide a continuous supply for the pressurised lubrication circuit.

The effect of the oil collection and circulation system described here is to cause the reservoir of standing oil the lower half casing to empty quickly as the bearing picks up speed so that in normal operation the external periphery of the ring is rotating in air rather than in a bath of oil. In this way the penalty of excessive energy consumption which is characteristic of high speed fixed oil ring systems is avoided.

3. EXPERIMENTAL APPARATUS

The experimental set up, shown in outline form by Figure 3, involved three bearings with their casings all bolted to a rigid frame and linked by a common shaft. A centrally positioned loading bearing, whose working element consisted of a half bush sitting on top of the shaft, was positioned between two support bearings. External load was applied by means of an hydraulic piston built in behind the half bush of the loading bearing causing it to bear down on the shaft. The shaft in its turn reacted against the two supporting bearings, one of which was the test bearing as described in the previous section. Figure 4 shows an end view of the test bearing mounted on the apparatus described here.

Motive power for the apparatus is provided b
variable speed DC motor also bolted to the comn
frame and connected to the shaft via a timing
drive to achieve the required speed range.

The oil inlet and outlet ports of the test bear
were connected by rigid pipe work to a Se
compact air blast heat exchanger. In order
simulate air conditions in an electrical mach
adjacent to its cooling fan, the heat exchanger w
located in a short length of ducting about two met
away from the bearing. For the majority of t
experimental work air at 40°C was blown throu
the duct at a speed of 10 m/s.

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