Dramatically enhanced air cooling for vertical pump bearings

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ABSTRACT

Technological advances in engineering are frequently driven by the need to solve a particular problem. Successful solutions then form the basis of a new generation of products designed for wider application. This move, from particular to general, is well illustrated by the development of a forced air-cooled, whitemetal bearing able to operate in markedly more extreme conditions than usual. This paper describes the evolution of a bearing design with greatly increased heat dissipation capability when compared with other bearings of its size and type. The resulting operating envelope therefore extends well into the area traditionally occupied by water cooled bearings.

1 INTRODUCTION

Self-contained, fluid film, tilting pad bearings are standard supply for a wide range of vertical machinery throughout the world. Figure 1 shows a cut away illustration of a typical design of such a vertical bearing with thrust (axial load) and journal (radial load) capability that has been described previously (1, 2, 3). To be self-contained all the lubricant used by the bearing is contained within the casing and it is independent of any external oil supply system. This means that the supply of cooled lubricant to the working surfaces of the bearing is not reliant on an external pump, but rather the rotation of the shaft and its attached collar provide the means of circulation within the casing. The circulation path is shown on the left hand side of Figure 1. Rotation of the lower face of the collar against the non-rotating set of thrust pads acts as a simple pump and oil is forced across the face of the bearing from the inside to the outside diameter. Constrained within the casing, the oil flows upwards, between the journal pads and back into the oil bath. It is then drawn through channels, not shown in the illustration, situated in the baseplate beneath the thrust pads and back to the inner diameter of the collar. An important feature of the bearing system is the anti-vortex shroud shown on Figure 1 which is fixed to the casing and closely around the upper part of the rotating collar. This prevents the motion of the collar generating a standing vortex in the oil bath.

Of course, considerable heat is generated in the bearing by the viscous shear of the lubricant which takes place in the load carrying fluid film. It is necessary to absorb this heat if the bearing is not to become too hot and most commonly this is done by means of a water cooler immersed in the oil bath around the circumference of the bearing. The right hand side of figure 1 shows part of such a heat exchanger together with one of its connections to an external water supply. Water coolers are typically made from copper nickel alloy tube with externally wound copper wire and situated, as shown, below the level of oil in the casing reservoir.

In installations where an external cooling water supply is unavailable, it is possible for a self-contained bearing to be cooled by heat transfer directly to the surrounding atmosphere. This is achieved by firstly providing the bearing casing with external fins to increase the area from which heat is transferred and, secondly, by arranging for a fan, mounted on the main shaft, to drive air constrained by a cowl at high velocity across this surface. The left hand side of Figure 1 shows a typical arrangement. The enhanced heat transfer coefficients achieved by this method are then usually sufficient for a range of applications with less arduous duty requirements

The particular case to be described in this paper, however, demanded an air-cooled bearing with a series of duty requirements normally associated with those of a water-cooled bearing. From the outset it was realised that this would be a challenging heat transfer problem, particularly as the range of ambient temperatures fell between -35ºC and +30ºC. The principal requirement was for a vertical shaft bearing to support a normal downward axial load of 390 kN at an operating speed of 990 rev/min. With a specific load of 3.5 MPa under normal working conditions, the heat generated within the oil bath was calculated to be 9.5 kW. Bearings generating this amount of heat are kept cool either by a continuous supply of cool oil to the bearing from an external source, or by means of a heat exchanger immersed in the oil reservoir as described above. In this case, however, neither external oil nor cool water were available and the client, therefore, requested all heat generated in the bearing be dissipated by heat transfer to the surrounding atmosphere. This was against a background for previous air cooled bearings of similar size where the heat generated was of the order of 3~4 KW. The problem facing designers, therefore, was how to increase the heat dissipation capacity of the bearing to new and unprecedented levels.

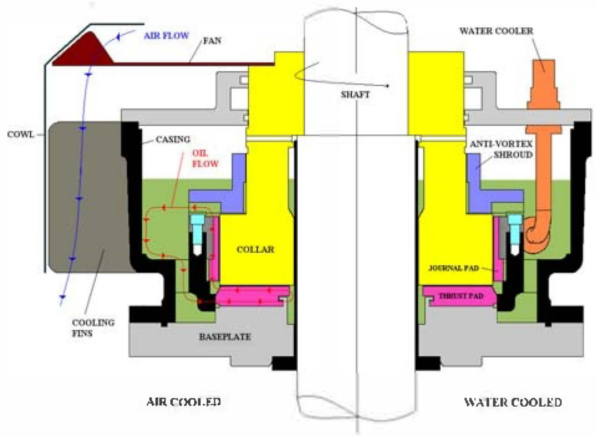

Figure 1. Conventional vertical thrust and journal bearing arrangements

2 BEARING OPERATION

The principal duty requirements for the case study application are given in Table 1 and Figure 2 shows the bearing system originally proposed by the designers. The bearing consists of a conventional main lower tilting pad thrust face, a tilting pad journal bearing to accommodate radial load and an upper surge thrust face above the collar. The purpose of this latter feature is to accommodate any axial load that might in some circumstances apply in the upward direction. The added complication of needing to allow for loads in both axial directions leads to a modification of the oil circulation into two paths as indicated. The bearing elements are enclosed by a finned, cast iron casing which forms an oil-tight enclosure and the means for dissipating the heat generated in the working section. An axial flow fan is mounted on top of the thrust collar to provide the cooling airflow. A cowl is supplied to enclose the fan and guide the cooling air over the exterior of the casing.

The bearing performance was monitored using a total of 10 thermocouples. Two thermocouples were situated in the oil bath, one thermocouple was placed at the bottom of the oil bath to sense the temperature of the oil flowing into the working section and one was placed about mid-way up the oil bath. Thermocouples were placed in the hottest areas of three thrust pads in the main load bearing thrust face and the remaining three were placed in the lower edge of one of the cooling fins, at the fin root, mid section and fin tip.

As it was unknown for such a large and relatively fast bearing as this to be cooled solely by direct heat transfer to the atmosphere, a large casing diameter was employed in order to provide sufficient surface area to dissipate the heat generated in the working section. A consequence of the large oil bath is that it was necessary to provide an oil guide immersed in the bath to ensure oil exiting from the bearing was conveyed out to the wall of the casing where heat transfer can take place. The extension of current knowledge that this design represented made it imperative that expected performance was verified by experimental testing of the prototype.

Shaft Diameter	194 mm
Speed	990 rev/min
Thrust Load – Normal	390 kN
Thrust Load – Maximum	470 kN
Thrust Surface (8 Sector Shaped Pads)	0.11 m^2
Thrust Pressure - Normal Load	3530 kN/m^2
Thrust Pressure - Maximum Load	4260 kN/m^2
Lubricant	ISO VG 32
Ambient Temperature	-35ºC to +30ºC

Table 1 Bearing characteristics

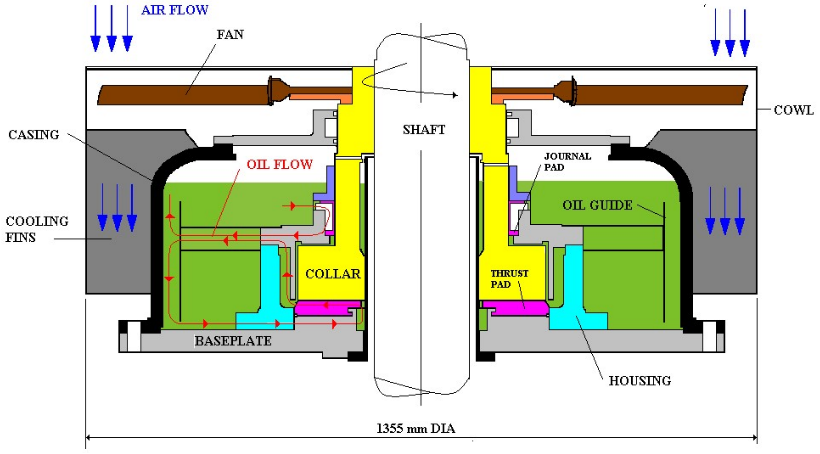

Figure 2. Original air cooled bearing proposal

3 THE EXPERIMENTAL ARRANGEMENT

The prototype bearing was mounted on one of Michell Bearings stands for experiments with vertical bearings. The stand features a variable speed DC motor to turn the test shaft, which includes a digital tachometer to record the shaft speed. The lower end of the test shaft passes through a loading module situated below the test bearing that is capable of forcing the shaft downwards and hence loading the lower, main thrust face of the test bearing. Hydraulic cylinders, within the loading module, generate the thrust loads giving a simple yet accurate method of determining the applied load.

The whole of the test stand is enclosed in an acoustic chamber. Consequently, varying the ventilation to the chamber provides a convenient means of controlling the ambient air temperature. Two thermocouples were used to measure the air temperature, one mounted above the test bearing in the air stream being drawn into the bearing and the other mounted at a suitable location adjacent to the bearing.

All experiments were conducted at the design shaft speed of 990 rpm in an ambient air temperature of 30°C. Temperature readings were normally taken after the bearing had been allowed to settle to steady state conditions. An exception to the normal procedure was for the ‘closed valve load’, a temporary overload applied for 4 minutes. In this case the temperature was recorded at the end of the 4 minute period.

Figure 3 showing the prototype bearing, with the external cowl removed, gives an idea of the scale of the project.