Operational Data for a Large Vertical Thrust Bearing in a Pumped Storage Application©

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This paper presents operating temperatures for a 2756 mm (108.5") 10-shoe thrust bearing installed in a vertical turbine generator in a pumped storage application. The rated output of the generator is 350 MW at a rotational speed of 257 RPM. Temperature data will be presented for shaft speeds up to 257 RPM and loads that ranged from 8.00 to 11.12 MN (1.8 to 2½ million pounds) thrust.

The data presented demonstrates the relative effect that the hydrostatic lift has on bearing operating temperatures. Some conclusions are drawn based upon the effect that the hydrostatic lift has on bearing operating temperatures.

INTRODUCTION

Fluid film thrust bearings are at risk unless a full hydrodynamic oil film is developed and maintained. This bearing risk condition occurs at starting, stopping and reversing, or whenever the operating speed falls below a certain minimum. When this occurs, full hydrodynamic lubrication is replaced by boundary lubrication, which leads to wear of the bearing babbitt face. One solution to this problem has been the introduction of high pressure oil between the bearing surfaces to establish a hydrostatic lift (high pressure lift).

This paper describes the in-service data collection for a large pivoted shoe or pad vertical thrust bearing equipped with a hydrostatic lift. This particular thrust bearing arrangement is used in each of the six identical pump turbine generators installed at the world's largest pumped storage hydroelectric facility located in Bath County, Virginia. These units are nominally rated at 350 MW generating and 400 MW pumping. Each unit operates at 257 rpm and the dead weight on the thrust bearing is approximately 816000 Kg (1.8×10^6 lbs).

In-service data collection was instituted after several minor wipes of certain thrust bearings had occurred during the commissioning of these units. Investigations resulted in the identification of two problem areas. The first was the discovery of leaks in some of the high pressure lift systems between the check valve and thrust shoes. These leaks prevented the development of hydrodynamic pressure in these shoes and resulted in a corresponding loss of load capacity. The second area identified was a temperature imbalance between the oil bath and feed tube supply systems resulting in thrust shoe radial distortions. The bearings were supplied with an ISO VG 68 lubricant.

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DESCRIPTION OF THE THRUST BEARING

The thrust bearing is an adjustable, pivoted 10-shoe type bearing with an outside diameter of 2756 mm (108.5") and an inside diameter of 1575 mm (62.0") (see Fig. 1). Each shoe rests on a thick plate or shoe support that is instrumented with a strain gage type load cell to measure thrust load. This assembly is supported by a jackscrew which is

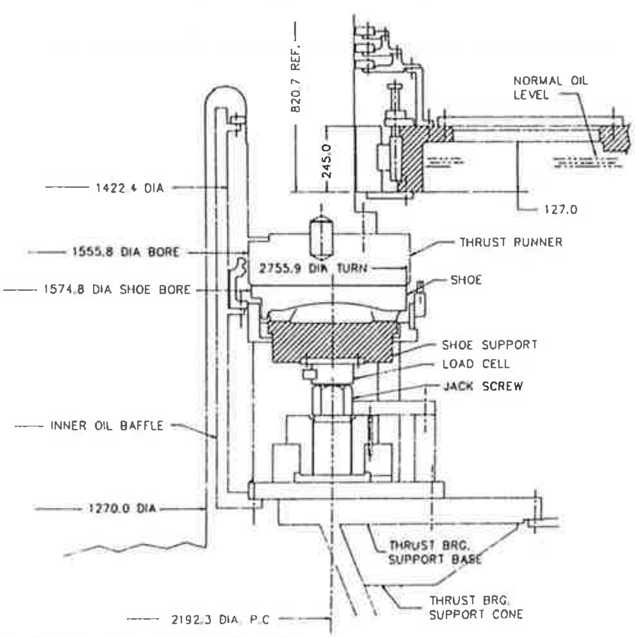

Fig. 1—Thrust bearing configuration

used to level the shoe and support, thereby equalizing the thrust load that each of the shoes must carry.

Although the bearing operates totally submerged in an oil bath (which is usually the case with these large bearings), size constraints made it impractical to put coolers in the oil bath. External coolers were therefore used. Oil was removed from the bottom of the oil bath and pumped back into the thrust bearings through a piping manifold with spray arms located between each thrust pad. Figure 2 shows the details of this oil feed tube supply system. This system allowed cool fresh oil to be delivered to the vicinity of the leading edge of each thrust shoe. Before installation, the system was flow tested at a rate of 2500 liters per minute (660 GPM) through the manifold and at 250 liters per minute (66 GPM) through each spray arm.

The thrust shoe was designed to be approximately square which is the optimum configuration to maximize unit loading and film thickness while keeping power loss to a minimum. During normal operation, the babbitt face of a pivoted shoe bearing tends to become convex around the point of support due to the load it is carrying. To minimize this tendency, the thrust shoes utilize relatively thin sections in these areas to significantly reduce the temperature difference between the face and back of the shoe (Fig. 4). Each shoe is supported on a pivot plate at four strategically computed areas located some distance from the center of the shoe (Fig. 3). The central area of the shoe is purposely weakened so that the parabolic pressure distribution in the oil film will tend to make the shoe concave instead of convex. In normal operation, the central area of the shoe becomes warmer than the outside edges, causing differential thermal expansion. Thus, the concavity due to the load and the convexity from the heat tend to cancel each other out, resulting in a more or less uniform plane for the shoe surface (Figs. 3 & 4). Although the shoe is centrally pivoted, during operation the leading and trailing edges unwrap, thus simulating an offset pivoted-shoe for increased load-carrying capabilities (*1*). This shoe design and support system is a unique approach that, through years of experience and hundreds of applications, has demonstrated an ability to minimize distortions. There are many other shoe designs and support systems that are intended to minimize shoe distortion. One such approach was detailed by Kawaike, Okano and Furukawa in 1977 Ref. (*4*) and another by Nelson, Plummer, McCulloch in 1984 Ref. (*3*).

To facilitate starting and stopping, a high pressure lift system was incorporated in the bearing. This was accomplished by a machined groove (Fig. 5) in the center of each thrust shoe that is connected to an external high pressure oil supply. Normal breakaway pressures were found to be around 13.11 MPa (1900 PSI) and shoe operating pressures ranged between 7.59 MPa and 10.35 MPa (1100 and 1500 PSI) (see Fig. 3).

DESCRIPTION OF INSTRUMENTATION

As stated earlier, each shoe rests on a strain gage type load cell intended to allow the proper adjustment of the thrust bearing. In this paper, however, the emphasis will be primarily on the temperature results obtained by an array of thermocouples and resistance temperature detectors (RTD's) strategically located throughout the shoe (Fig. 5).

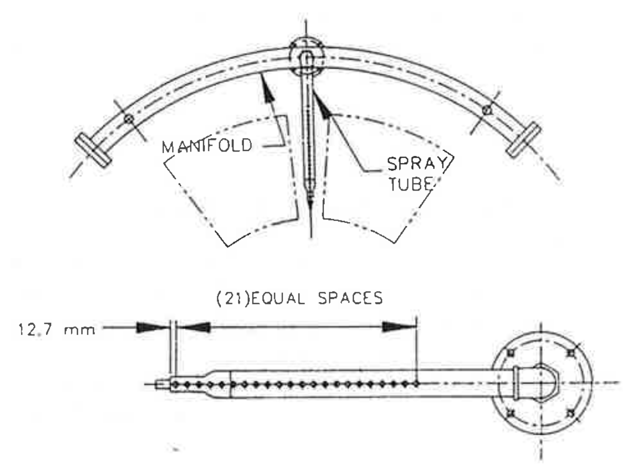

Fig. 2—Oil supply system arrangement and detail of spray tubes

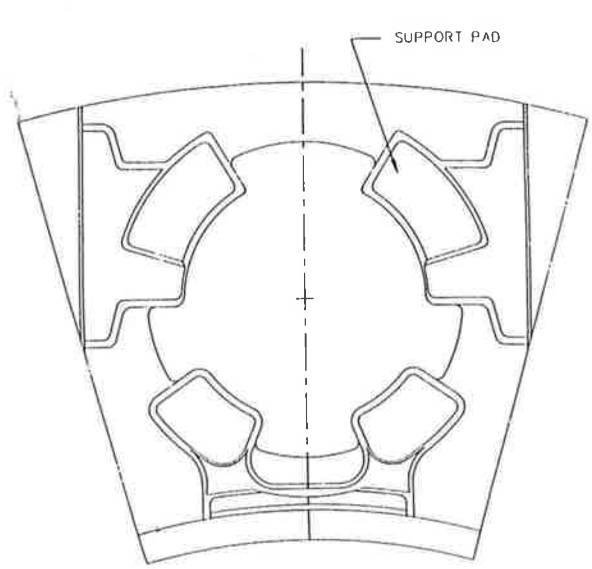

Fig. 3—View of Back of thrust shoe showing support pads

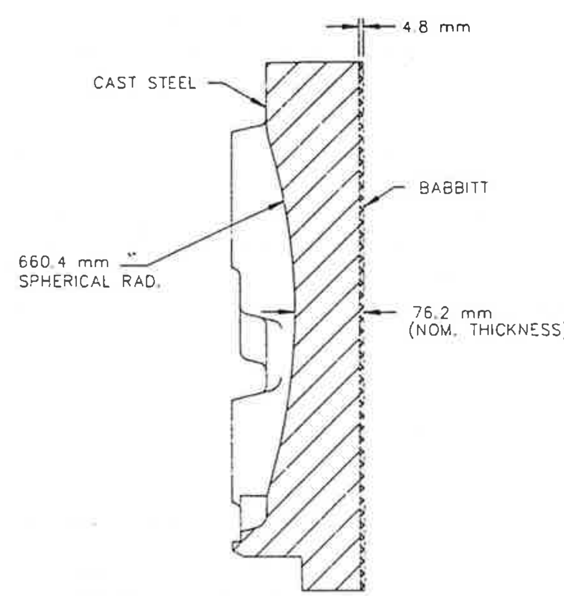

Fig. 4—Cross section through center of thrust shoe

Seventeen thermocouples were placed 9.5 mm (3/8") from the babbitt face. The babbitt thickness of the thrust shoes was 4.8 mm (3/16"). This resulted in the thermocouple tips being located 4.8 mm (3/16") into the steel. There was also an array of 13 thermocouples strategically located 9.6 mm (3/8") deep in the steel from the back of the shoe. Two shoes were also instrumented with six additional thermocouples located at the 20-26, 1-12, and 8-2 positions shown in Figs. 5 & 6. Resistance temperature detectors (RTD's) were also located at the leading and trailing edges of every other shoe of the thrust bearing to monitor bi-directional operation (see Fig. 7).

DESCRIPTION OF TESTS

As mentioned before, minor wipes were not only caused by the problems with the high pressure lift system, but also by an excessive temperature imbalance between the spray system and the oil bath. Bearing wipes occurred shortly after startup when the oil bath temperature was approximately 40 to 45°C (104–113°F) and the ambient temperature of reservoir water was low. This resulted in the oil from the spray bar system being introduced to the thrust bearing 10–20°C (18–36°F) cooler than the oil bath temperature. This caused the thrust shoes to become convex as a result of the cool oil being sprayed in the vicinity of the leading and trailing edges of the shoes. Therefore, all the testing was conducted with the spray system oil warmed to a minimum of 35°C (95°F) before initial roll. Testing to simulate summertime operating conditions was also conducted, where the bath temperature and cooling water would be warmer. These tests were run with identical oil supply bath and feed tube system temperatures of 49°C (120°F).

The instrumentation was connected to a data logger and computer where 40 channels were scanned and recorded every 20 seconds. Tests were conducted both with the continuous use of the high pressure lift system, and also under normal operating conditions where the high pressure lift system was turned off after initial roll at approximately 75 percent speed (30 seconds after initial roll) and turned back on during normal stopping at approximately 75 percent speed. The test data clearly shows that there is a difference between standard hydrodynamic and a hybrid hydrodynamic-hydrostatic bearing operation. Although it was expected that continuous operation of the high pressure lift system would cause erosion, as of this writing, (1500 hours of continuous operation), there has been no evidence of babbitt erosion or cavitation damage.

During testing, hydraulic and dead weight loads totaled between 998000 Kg and 1134000 Kg (2.2 and 2.5 million lbs) on the thrust bearing. It should be mentioned that the lower MW loadings produced higher thrust loads on the bearing, i.e., at a 310 to 360 MW output, readings were 1134000 Kg (2.5 × 10^6 lbs) on the bearing; at a 400 to 410 MW output, readings were only 998000 Kg (2.2 × 10^6 lbs.) on the thrust bearing. This was confirmed not only by the shoe temperature readings, but also by the thrust shoe load cells.

To minimize the amount of data presented, the test results reported will be for the generating mode only, which happens to be the highest load case. Although in many applications the turbine pump direction normally has the higher loads, it was not the case in this application. Test results will be shown with various curves of individual thermocouple locations on the thrust shoe and also isothermal plots of the entire thrust base.

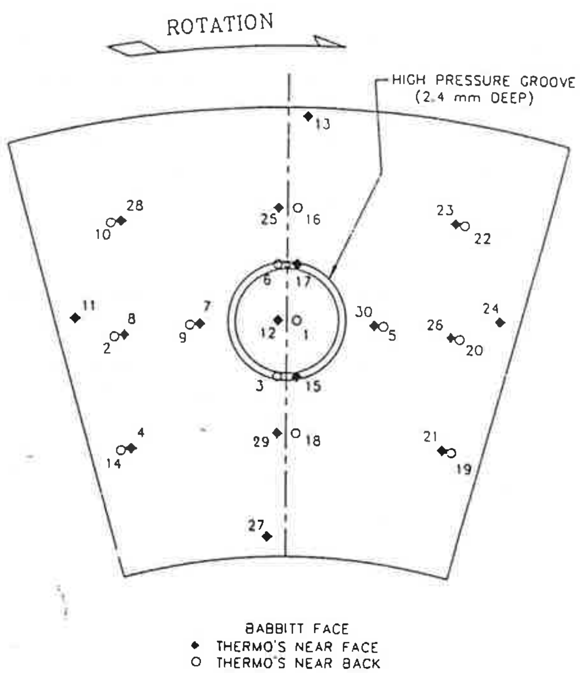

Fig. 5—View of thrust shoe face showing location of thermocouples and high pressure lift groove.

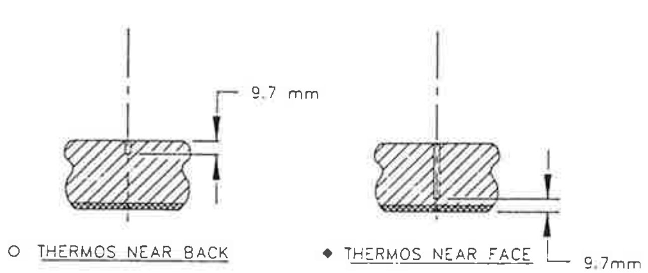

Fig. 6—Shoe cross section showing thermocouple holes. All thermocouples were grounded in bottom of hole and epoxied in place.

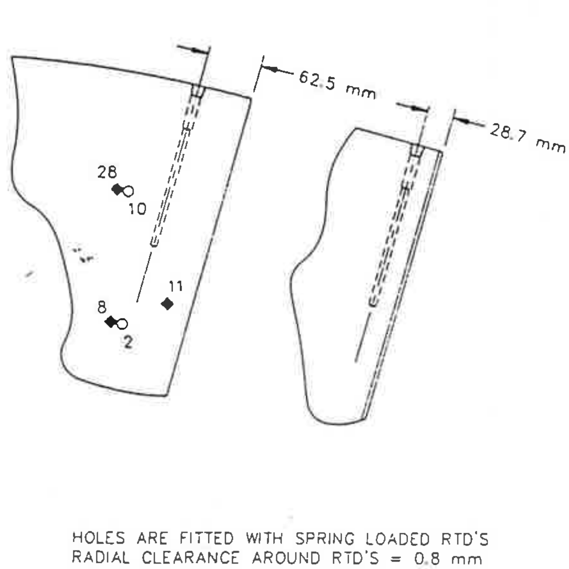

Fig. 7—Location of RTD's in thrust shoes

OPERATING RESULTS

It was discovered early in the testing that the RTD's provided a poor indication of operating temperatures. The maximum temperatures measured by the RTD's were 80–90°C (176–194°F), which was 10–20°C (18–36°F) cooler than the face thermocouples located at the mid-diameter from the center of the shoe to the trailing edge. Also, the response time for the RTD's compared to the thermocouples was very poor Ref. (2). In fact, several wipes had occurred and there was no indication registered by the RTD's, and wipes that had occurred with the thermocouples in place were immediately recognized by temperature shifts of 30–40°C (54–72°F). Therefore, only thermocouple temperature data will be used in the curves and isotherms. To reduce the vast temperature data to a manageable amount, only the thermocouples located along the middle radius of the shoe will be utilized in the discussion.

Figure 8 shows the temperature responses recorded by thermocouples 5 (back) and 30 (face) for both high-pressure lift on-and-off operation with the turbine in the generating mode (9.8 to 11.1 MN/2.2 to 2.5 × 10 lbs) for the first 30 minutes of rotation. When operating with the high pressure lift system on (solid curves), the thermocouple located just below the babbitt (T/C 30) is reporting, as would be expected, higher temperatures than the thermocouple located at the back of the shoe (T/C 5). The response of T/C 30 to rotation is almost immediate, while there is a lag of about 1 minute before T/C 5 starts to respond. The slope of the temperature response curve for T/C 30 is also greater than that of the T/C 5 curve, but after about 25 minutes, the two curves become nearly parallel with a 10°C (18°F) difference between the top and bottom of the shoe. The maximum temperatures reported for each is 92°C (197.6°F) for T/C 30 and 82°C (179.6°F) for T/C 5 and occurs at the 30 minute mark.

The effect that the use of the high pressure lift system has on bearing operating temperatures can be seen by contrasting the dotted and solid curves for both T/C 5 and T/C 30. Use of the high pressure lift system can be seen to have a significant impact on bearing temperatures at these locations. Operating without the high pressure system has increased the overall temperature of the shoe. Not only have the maximum temperatures increased to 98°C (208.4°F) for T/C 30 and 86°C (186.8°F) for T/C 5, but so has the slope of the temperature curves. Temperature data for thermocouples located on the middle radius of the shoe is shown on Figure 9. Reviewing the temperature data in Figs. 8 and 9, three observations can be made.

The difference in temperature between operating with and without the high pressure lift system is not very large (Fig. 8). The reason for this can be attributed to the proximity of thermocouples 5 and 30 to the high pressure lift groove. When the high pressure lift is used, cool, non-aerated oil from this groove washes over the shoe at this location, and the thermocouple reports these lower temperatures.

The increased temperature difference between the babbitt face (T/C 26) and the thermocouple located at the back of the shoe T/C 20 (40°C–72°F) can be attributed to the spray arms which deliver cool oil between the shoes (Fig. 9). The back of the shoe is more sensitive to this cool oil because of circulation caused by the collar rotation, and there is minimal affect to the face of the shoe as a result of hot oil carryover.

The higher babbitt face temperatures reported by T/C 26 (105°C–221°F maximum) result because the thermocouple is located closer to the trailing edge. The thinner oil films in this area translate into higher temperatures. This position and temperature relationship can be seen in Fig. 10 where the temperatures increase from leading edge (T/C 8) to trailing edge (T/C 26).

Figures 11 and 12 are plots of shoe isotherms showing the overall temperature effect of running with and without the high pressure lift system. Comparing these figures, it is easy to see the influence that the cool oil from the high pressure groove has on the temperature gradient across the face of the shoe. Using the high pressure lift, in this case,

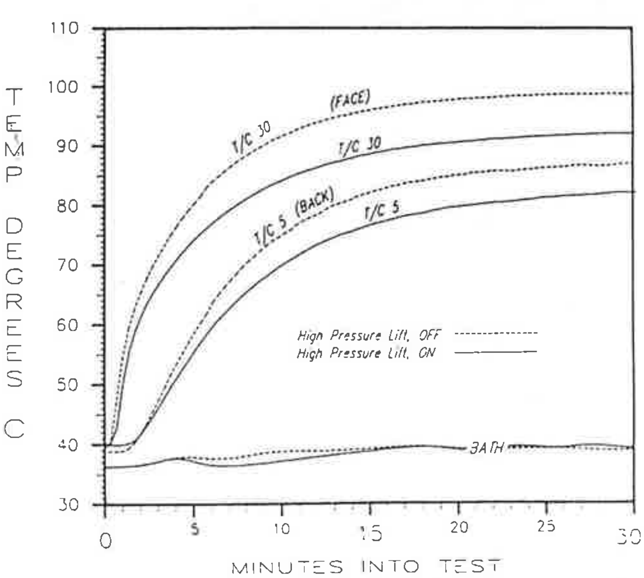

Fig. 8—Plot of shoe temperatures vs time for T/C's 30 and 5, both with and without the high pressure lift system.

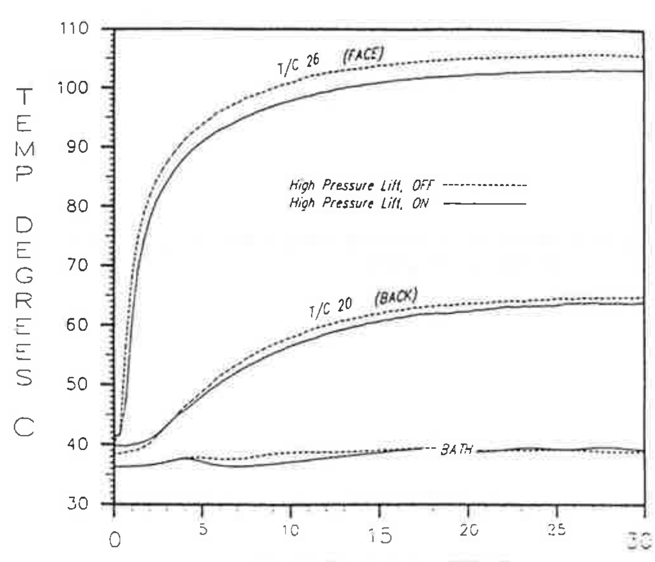

Fig. 9—Plot of shoe temperatures vs time for T/C's 26 and 20, both with and without the high pressure lift system.

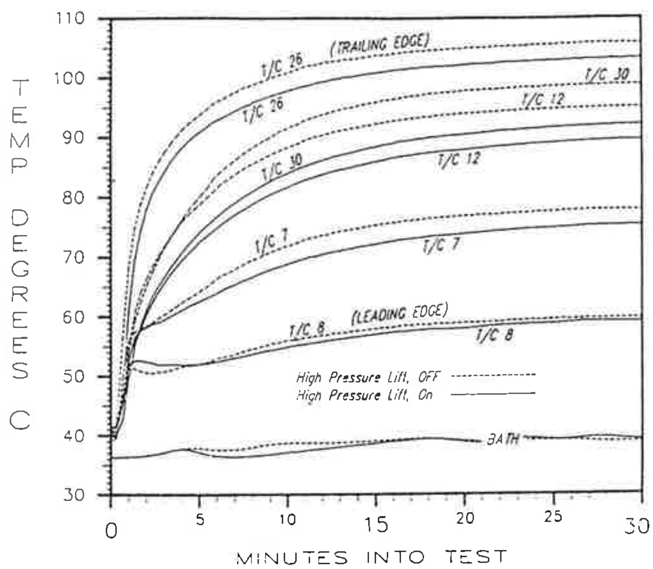

Fig. 10—Plot of shoe temperatures vs time for middle radius face T/C's, both with and without the high pressure lift system.

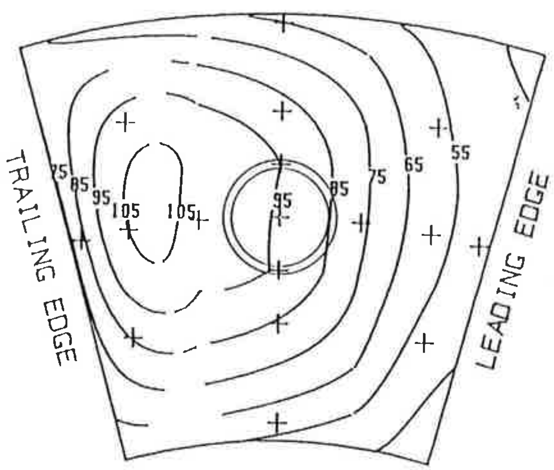

Fig. 11—Shoe isotherms at bath temperature 39°C H.P. lift off, 257 RPM and 360 MW output.

results in a 10°C (18°F) temperature reduction on the trailing edge half of the shoe. The operating temperatures for this particular case suggest that the use of the high pressure lift system for other than startup and coast-down is unnecessary.

CONCLUSIONS

1. High pressure lift systems affected bearing metal temperatures, but the maximum pad temperature is only a few degrees cooler with the high pressure lift system in operation.

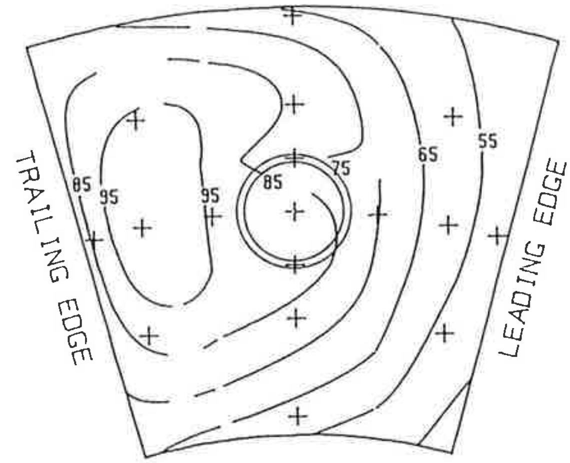

Fig. 12—Shoe isotherms at bath temperature of 39°C H.P. lift on, 257 RPM and 350 MW output.

2. Oil supply method and temperature can influence the thermal gradient through the shoe, leading to excessive thermal deflections. In the case described in this paper, babbitt wiping was caused by excessively cool oil being pumped into the bearing.
3. Location and installation techniques used on thermocouple or RTD is critical to reliably measure bearing performance.
4. Resistance temperature detectors are not reliable enough to protect against damage in bearings of this type and should not be used.

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