Trouble-Shooting Bearing Problems In Large Hydroturbine Generators

If hydro plant operators are aware of the signals that can alert them to potential harm to generator bearings and act promptly to investigate and evaluate any problems, failures should be averted.

By Stanley Abramovitz

Editor's Note: This article continues Hydro Review's *recent focus on hydroturbine bearings. In the July 1991 issue, the article "Shedding Light on Hydroelectric Thrust Bearing Problems" discussed bearing problems caused by changes in manufacturing processes and by previously unknown oil pressure effects. This article reviews two aspects of trouble-shooting: discovering and defining potential problems that might cause bearing failure; and pinpointing the cause of failures to prevent similar ones in the future.*—M.B.

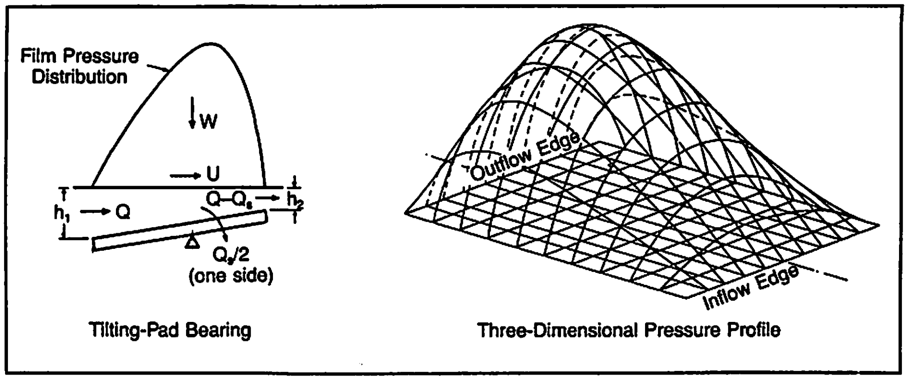

Figure 1: The pressure profile on a self-acting pad bearing shows that the lubricant film pressure is zero at the pad edges and rises to a maximum on the pad's center line toward the outflow edge. In the sketch on the left, W = load; U = runner velocity; h = film thickness; Q = oil flow; and Q_s = side oil flow.

The primary requirement for bearings in hydroturbine generators is that they be reliable and trouble-free. This consideration is particularly critical in large machines where bearing failures can be extremely costly, not only in terms of the replacement hardware and labor, but also because unscheduled turbine downtime can cause a possibly lengthy loss of power production. As soon as an operator sees the first signs of a harmful condition that may indicate only slight bearing distress, he should act immediately to shut down the unit to avoid a possible bearing failure. In doing so, the problem may be easily remedied.

The fluid-film bearings used in large hydroturbines have no inherent life limitation. The first Kingsbury thrust bearings were installed in 1912 at the Holtwood Hydro Plant in Pennsylvania on the Susquehanna River, and are still operating efficiently and reliably after almost 80 years with only routine maintenance. The reason for this longevity is that most of the large bearings installed in hydroturbines have been properly and conservatively designed by reliable, experienced manufacturers.

Although analyses of bearing designs occasionally will show that the bearing was only marginally adequate for the anticipated conditions, more often, bearing problems are caused by poor or unexpectedly severe operating conditions. These conditions include dirty oil, high inlet-oil temperatures, inadequate bearing maintenance, material corrosion and erosion, shock, excessive loading, thermal and mechanical deformations, and improper reconditioning of bearing surfaces and components.

In addition, just the mere size of a large bearing can create problems. For example, thermal and elastic bearing-surface distortions can cause surface distress that may lead to eventual catastrophic failure. In addition, the large size magnifies the inevitable distortions and misalignments in the turbine structure, bearing housing, and turbine shaft. Although the magnitude of most of these inaccuracies cannot be accurately predetermined, bearing designers should attempt to minimize and accommodate their potentially harmful effects.

Unusual bearing conditions typically require design analysis by a trouble-shooter who has theoretical knowledge

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of the bearing types used in large hydroturbines, supplemented by extensive field experience in the solution of bearing problems.

Reviewing the Fundamentals Of Types of Bearings

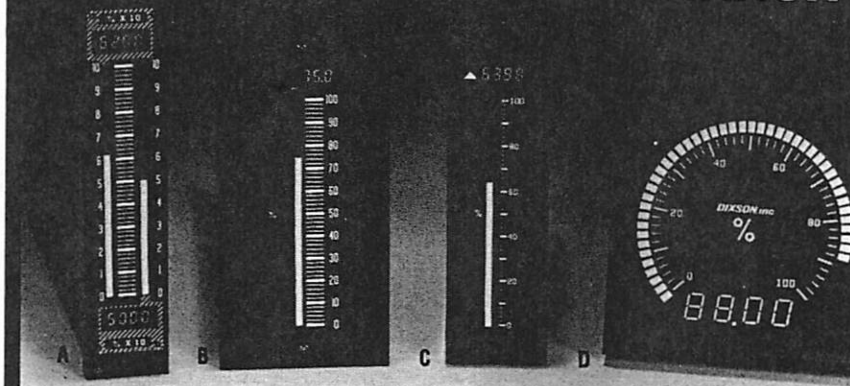
Before being able to trouble-shoot bearing problems, it's important to have a solid understanding of the various hydroturbine bearing types. Unlike the conventional-size bearings found in pumps, turbines, and compressors, hydroturbine bearings are far larger—sometimes as large as 12 feet in diameter. While small damaged bearings are usually replaced, the owner of a large bearing rarely has that luxury. When large bearings are damaged, they normally are repaired, but only after an evaluation to determine what caused the distress and what type of repair will be both safe and economical.

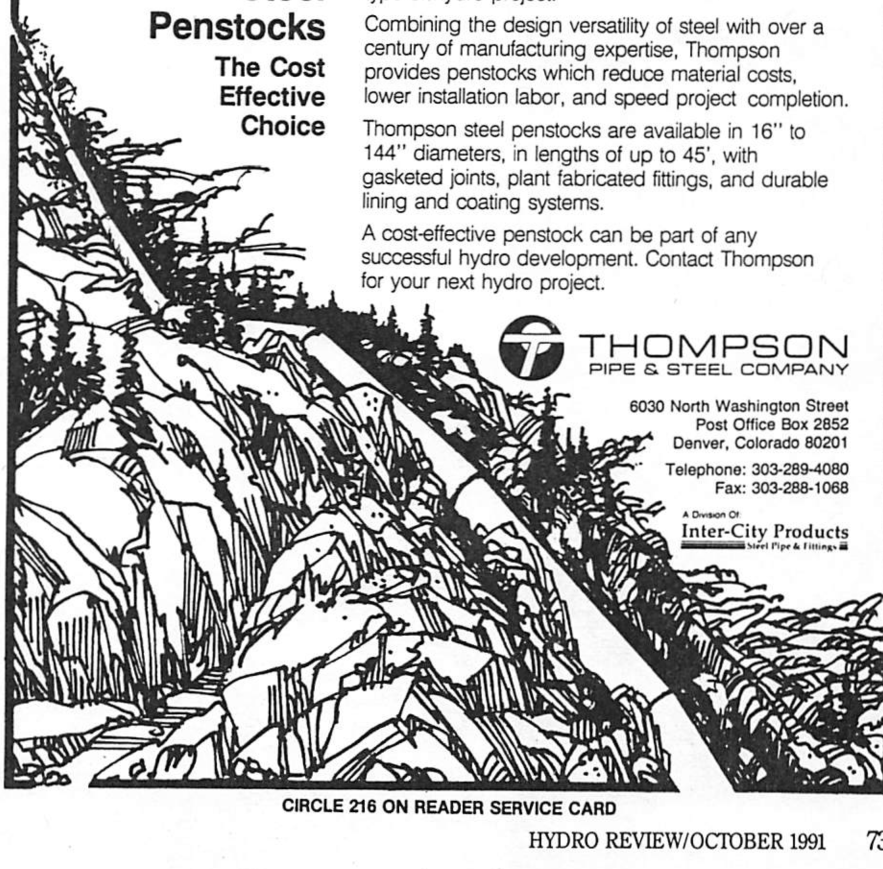
This brief overview of bearing fundamentals and types applies to large hydroturbine bearings; it does not include bearings for such turbine components as wicket gates and blade shanks.

Although small hydroturbines may use ball or roller bearings, the thrust and guide bearings used in large turbines are the sliding-surface, fluid-film type, which develop and maintain a thin, pressurized film of lubricant between their fixed and moving surfaces. The lubricant is normally oil, although some relatively small guide bearings use water as a lubricant. For oil, film thickness may range from 0.0015 to 0.0025 inches. A film of less than 0.0010 inch thick would be marginal for large bearings.

Figure 1 shows the fluid configuration and pressure profile that occurs with a tilting-pad bearing in the most common mode, the self-acting (hydrodynamic) mode. In this mode, the film pressure is self-induced by the relative motion between the two bearing surfaces. Straight-sleeve-type journal bearings also form a natural geometric film wedge when the journal is displaced in its bearing.

Figure 2 shows a second bearing mode, the externally pressurized (hydrostatic) bearing, which uses an external pump to induce flow and pressure. Large hydroturbines frequently use a hydrostatic lift during turbine start and stop, both for thrust bearings and for horizontal- and inclined-journal bearings.





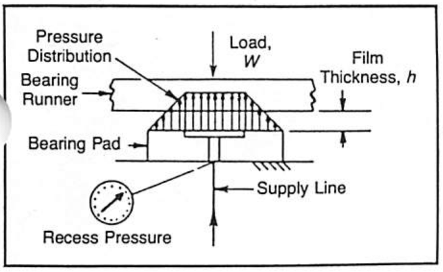


Figure 2: Most large hydroturbines use a hydrostatic lift to create a film of lubricant only during turbine start up and stop, although full-time hydrostatic thrust and journal bearing lifts are in use for special operating conditions.

Thrust Bearings

In vertical hydroturbines, the thrust bearing carries the major bearing load. This load includes a large vertical dead-weight starting load, plus the even larger hydraulic load that is added during operation. Figure 3 shows the normal tilting-pad thrust bearing, which has a number of segmented pads, or shoes, each resting on a pivot so that it is free to tilt to the optimum film-wedge angle.

Figure 4 shows alternative approaches to mounting pads. Not shown is a sensitive equalization method that uses pads supported on hydraulically connected, oil-filled bellows.

Guide Bearings

Vertical turbine guide bearings normally do not support high constant radial-shaft loads. However, there can be significant loads due to mechanical unbalance, electromagnetic pull unbalance at the generator, and hydraulic forces. Additional forces can result from temperature distortions, structural effects, and bearing hydrodynamics. Designers have difficulty in determining the importance of individual effects and the interrelationships between effects, so they tend to design guide bearings conservatively, allowing for a large but reasonable load capacity. Horizontal- and inclined-turbine journal bearings have higher and more defined radial loads.

Two types of vertical turbine guide bearings are common. One is a straight cylindrical-sleeve type with oil grooves; the other is a tilting-pad type, similar to the tilting-pad thrust bearing. The tilting-pad type has several advantages:

—it can be made to accommodate some shaft misalignment;

—clearances can be easily adjusted; and

—the segmented pad allows for more efficient cooling of the oil films.

Each tilting pad may be provided with pumping grooves. A typical tilting-pad guide bearing might have 20 pads in a 100-inch-diameter circle.

Critical design considerations for both types of journal/guide bearings are oil flow and the temperatures of the inlet oil and the oil film. Also critical are diametrical clearance for the sleeve type and pad-surface curvature and assembled clearance for the tilting-pad type. Both types may be affected in the field by differential thermal expansion that can alter the diametrical clearance and by design tolerances that may be exceeded during manufacturing.

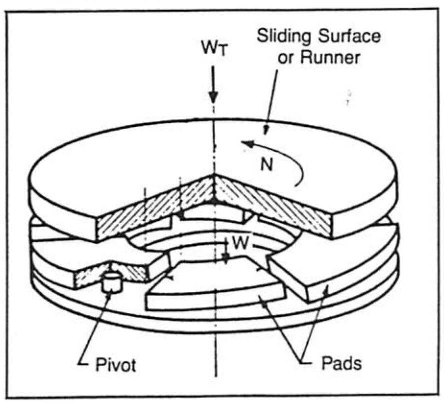


Figure 3: Tilting-pad thrust bearings incorporate segmented pads mounted on pivots. A typical bearing might include 14 pads on a 128-inch outside diameter circle. In this figure, W_T = total thrust load; N = rotational speed; and W = pad load.

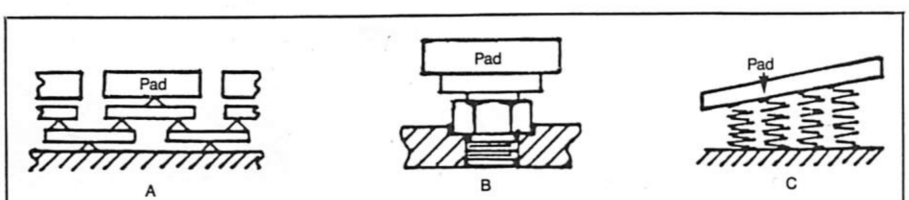


Figure 4: Alternative methods of pad mounting include (A) a self-leveling pad equalizing system used by some manufacturers. (Although more expensive, the more complex mount is easy to install and provides improved self-aligning capabilities); (B) a pad pivoted on a jack screw, a technique commonly used for very large bearings. At the site, each pad is jacked up to the thrust runner (also called "collar") surface, and fixed when all pad surfaces are in exactly the same plane. This design can help stabilize the generator rotor; and (C) each pad mounted on a spring-bed mattress, which provides freedom to tilt and equalize, and some ability to self align.

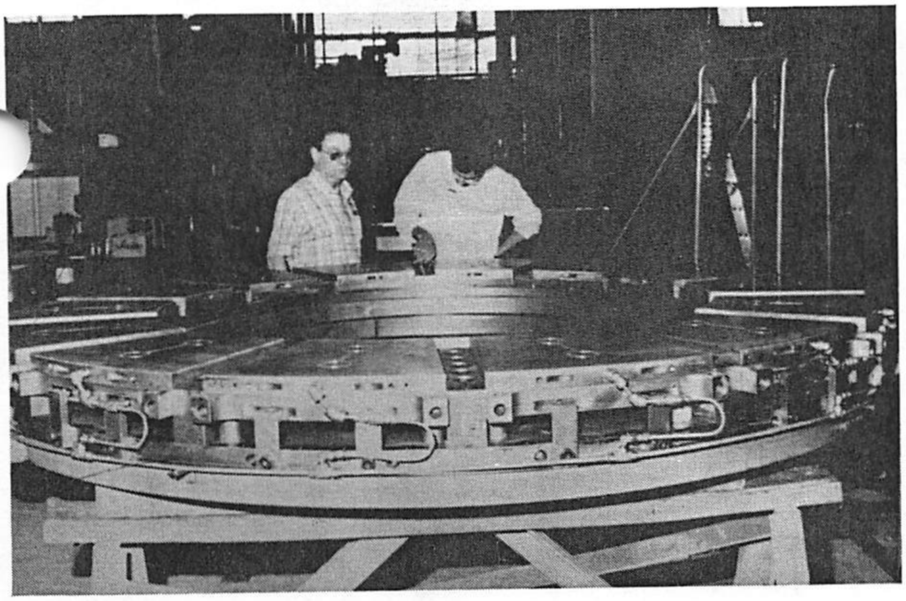
Marginal Operations Can Lead To Major Failures

The best procedure for preventing bearing failure, in addition to operating an effective routine maintenance program, is to train operators to recognize a marginal bearing condition. Bearings operating under marginal conditions are sensitive: they might run forever, or, on the other hand, even a minor adverse change in operating conditions could cause failure. Experienced trouble-shooters can detect if a bearing is operating marginally and might be destined to fail.

The performance of a fluid-film bearing is a function of two criteria: the thickness of the lubricating oil film separating the bearing surfaces, and the temperature of the oil film. Because either may be the controlling factor in evaluating the bearing, the first step in trouble-shooting a bearing problem is to analyze and evaluate the bearing design by calculating the theoretical film thickness and temperature.

Large hydroturbine bearings are normally well instrumented so that any sign of bearing distress, such as high bearing temperature, activates an alarm, and—if necessary—a shutdown. If the alarm system functions as it should, the only form of surface distress will be burnishing. Unfortunately, severe damage cannot always be avoided, particularly when harmful conditions occur quickly. Major adverse conditions, such as starvation due to loss of lubricant, very large shock loading, and dirty oil, can produce frictional temperatures high enough to result in catastrophic wear (galling, seizing, and melting of the babbitted bearing surface).

When the bearing is operating marginally, the lubricant film is extremely thin and/or hot, and the bearing-surface temperature will remain at a constant but unusually high level. (If the bearing-surface temperature is rising steadily, it indicates imminent failure.) Since peak film temperatures can be significantly higher than thermocouple readings taken below the bearing surface, only an experienced analyst can estimate a realistic peak value from the thermocouple readings.



A thrust bearing in a hydroturbine-generator is used to overcome the friction created when the shaft of the unit exerts force in a direction parallel to its axis of rotation. The top half of a bearing is a flat ring called a runner, which rests on flat shoes shaped like wedges of a pie, such as the bearing shown here. This 3-meter-diameter thrust bearing, being trial assembled in the factory, will be installed in a vertical hydroelectric generator. (Courtesy GE Canada)

When analyzing the bearing-temperature history, the investigator should be careful to account for any changes in bulk or inlet oil temperature.

The bearing surface, which should be inspected during a scheduled maintenance shutdown, can also indicate a marginal situation. If the oil film is too thin and hot, the babbitt tends to creep, which can be detected as ripples on the surface where creep flow occurred. Tin babbitts will creep at temperatures ranging from 375°F for unit loads below 200 psi to 270°F for steady loads of 1,000 psi.

A dark baked film on the bearing surface is another indication of marginal operating conditions. This film is the result of high temperatures deteriorating the oil lubricant. Failure may result from either the buildup of the baked material or the debris that flakes from the buildup.

Conditions that can create a marginal situation include: an inadequate bearing design; turbine operating conditions that are more severe than anticipated in the design; the use of an improper grade of oil; excessively high inlet-oil temperatures; deformations and misalignments exceeding those provided for in the design; an inadequate bearing "fix;" and the selection of the wrong type of bearing.

Thrust Bearing Problems Relate to Curvature

In addition to problems caused by the selection of an inappropriate bearing type or by inadequate design, another common thrust bearing problem is associated with surface deformation of the pads. As Figure 5A shows, during turbine operation, the thermal gradient across the thickness of the pad causes the pad surface to become spherically convex (crowned). When large, heavily loaded bearing pads are exposed to hot oil films, but the surrounding oil is relatively cool, the pad surface can become so spherically crowned as to substantially reduce the developed film pressure, and therefore reduce the load capacity and film thickness. Temperature ratcheting can occur, eventually leading to film temperatures high enough to cause wiping and then catastrophic failure.

Pad surface curvature is well understood, and manufacturers have used many methods to control it. Figure 5B shows one approach: mounting the pad on a platform that supports the pad closer to its perimeter, leaving it less rigid at its center. The oil film pressure on the weaker center mechanically flattens the pad surface in a way that compensates for the thermal crowning.

Other designers have used cooling galleries behind the babbitt surface, with circulating cold water to reduce the film temperature and therefore the differential temperature across the pad thickness. Another method is to insulate the sides and back of the pad to keep its temperature constant throughout.

Figure 4C shows a pad mounted on a spring-bed support. This type of mount has an inherent flattening capability; however, the spring-bed and pad thickness must be carefully designed for the operating conditions. Marginal conditions and failures have occurred when excessive mechanical flexibility in the pad eliminated the needed amount of crown, and the pad failed because it was too flat. Conversely, because the springs are preloaded, it is possible to have a failure caused by excessive crown if the load on the pad is less than the spring pre-load. This failure can occur when the thrust load is reduced in an effort to "help" the bearing.

Figure 6 shows a pattern of surface distress common in thrust-bearing failures, although the same type of failure also may cause a circular wear pattern. The burnished or wiped surface indicates a pad that, because of excessive crown, is unable to fully support the thrust load. When this occurs, the pad thermocouple shows an upward temperature trend, giving the operator time to shut down the turbine before catastrophic wear results. The proportions of the wear pattern depend on the amount of crown and how promptly the turbine is shut down. Aside from a poor pad design or the excessive temperature differential discussed earlier, excessive crowning can be caused by partial oil starvation or by high frictional heat due to oil contamination.

For thrust runners that rotate in only one direction, the pivot point can be offset to help the pad achieve the required hydrodynamic film. For pump-turbines that are capable of rotation in both directions, central pivots are needed. Also, some designers normally choose central pivots to avoid the possibility that the pad may be installed for the wrong direction of rotation. In some installations, marginal bearing operating conditions have been corrected by installing pads with properly designed offset pivots.

Curvature Also Affects Journal and Guide Bearings

The clearance, oil flow, and temperature problems that affect sleeve

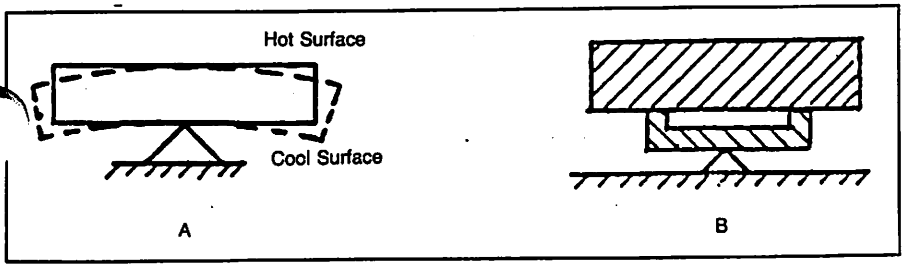


Figure 5: Sketch A shows how a thrust pad that is flat when it is at rest (solid outline) will assume a crowned shape (dashed outline) when it is exposed to a hot oil film on its bearing surface. Sketch B shows one technique for compensating for this curvature: the pad support is designed to weaken the center of the pad, so that the pressure of the oil film against it will act to flatten the surface.

and tilting-pad guide bearings normally are well understood. A more subtle source of guide-bearing distress is related to the curvature of the tilting pad. In order for the tilting-pad bearing to operate properly, the designer must consider two radial clearances: the "machined-in" or "pad" clearance, and the "assembled" or "bearing" clearance.

Figure 7 shows how the radii of the two clearances are different, providing the crown essential for this type of bearing. The assembled clearance can be set using jack screws or other means to fix and maintain the adjustment. Although the pad surface will tend to increase its radius of curvature due to the thermal gradient across the pad thickness, this increase in "crown" may be insufficient—or occur too late—to prevent a failure if the initial pad clearance is inadequate.

There have been a number of cases where tilting-pad guide bearings, 6 to 8 feet in diameter, were running hot or had failed, and the condition was corrected merely by improving the pad-surface curvature.

Very large pads can be inspected by fitting them against the turbine shaft and measuring the amount of curvature with a feeler gage. It is not uncommon to find pad surfaces that are "flat" or incorrectly curved. They may have been supplied that way initially, or they could have been deformed over time or been incorrectly reconditioned.

Hydrostatic Lifts Provide Insurance, Diagnosis

Figure 8 shows a typical hydrostatic lift system. These systems, which use an external pump to develop an oil film separation between bearing surfaces, are used to prevent surface damage during the period when a thrust bearing is starting up, and again when it

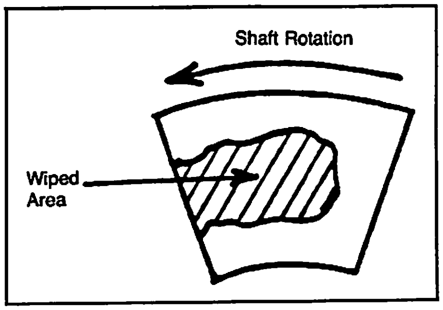


Figure 6: A burnished or wiped surface pattern indicates the bearing's pad surface is excessively crowned.

comes to rest during stopping. Depending on the dead-weight load, these are the times when the turbine speed normally is insufficient to develop a reliable self-acting fluid film. The lift is usually turned off or on, depending upon whether the turbine is starting or stopping, when the shaft reaches about 75 percent of operating speed.

Each pad has a check valve to prevent back flow and bleeding of the hydrodynamic oil film pressure when the lift system is shut off, and the system includes a pressure regulator. Flow-control valves at each pad provide equal flow to the pads, and are

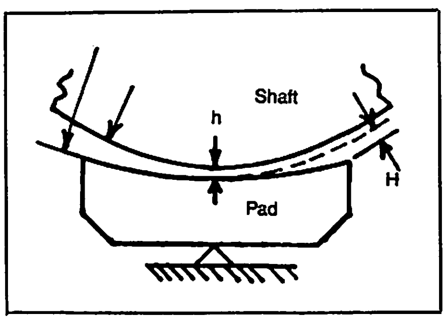


Figure 7: Guide-bearing pad clearances, with the "machined-in" clearance greater than the "assembled" clearance. This provides the pad surface with a "crown," or greater radius of curvature than the shaft. Crowning of the pad surface is essential for proper bearing operation. In the figure, h = minimum oil film thickness and H = "crown" height.

matched with the pump flow to generate the desired pad-film stiffness. The valves also prevent the loss of the entire flow through one partially loaded pad. Because the oil is pumped directly into the center of each bearing pad, filtration is critical. Dirty oil or a blown or ineffective filter in the lift circuit can cause an immediate bearing failure.

Hydrostatic lifts are not just for thrust bearings; they also are installed on inclined and horizontal-shaft journal bearings. In rare instances, they are used on a full-time basis to provide an additional factor of safety for a bearing that is operating marginally.

Problems arise when a lift system has not been well designed to develop a thick enough or stiff enough film. Some designers may consider film stiffness unimportant because the system is intended to be used only for the short interval during starting or stopping. However, some types of hydroturbines generate acceleration thrust loads in addition to the dead-weight load, and a "soft" lift system may be unable to prevent surface damage. Occasionally, turbines may be restarted quickly after shutdown, while pads are still significantly crowned from heat effects. In those situations, the hydrostatic-lift effect is reduced. For relatively little added cost or effort, a hydroturbine owner can secure a well designed lift system that provides a degree of insurance.

With a relatively small modification, hydrostatic lift systems have been used as an effective diagnostic tool. A pressure tap is inserted at each pad between the entrance to its annular groove and its check valve. A pressure gage or transducer connected to each tap can measure both the film pressure during hydrostatic lift and the running hydrodynamic pressure at or near peak value.

With the lift on and no shaft rotation, a comparison of film pressures will show how good the load distribution is from pad to pad. During turbine operation, with the lift off, the pressure variations at each pad and between the various pads can provide diagnostic information. They can uncover vibrations, indicate shaft wobble, show unusual thrust forces, and indicate rough or smooth turbine operation at different megawatt loads. Although bearing taps can be a very useful tool in uncovering an elusive bearing problem, they must be in-

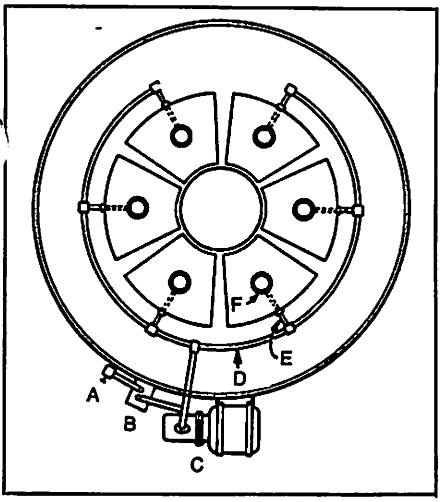


Figure 8: In a typical hydrostatic lift system, a constant-displacement pump (C) moves oil from a reservoir (A) through a filter (B) into a manifold (D). From the manifold, the oil moves through flexible hoses or piping (E) into an annular groove (F) in each pad surface.

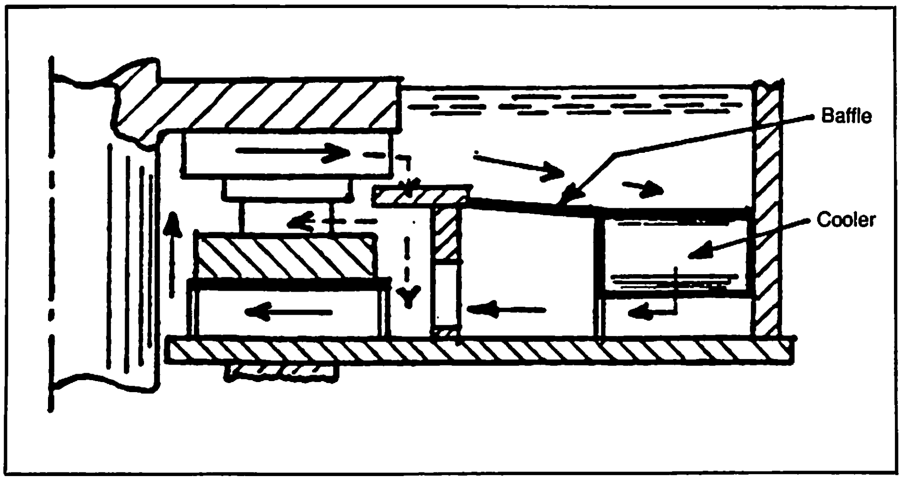


Figure 9: The oil in a bearing reservoir flows naturally because of the velocity difference, and therefore the pressure drop, between the inside and outside diameter edges of the pads.

stalled with extreme care to prevent leaks or fitting failures.

High Oil Temperature: A Red Flag

If the oil film between the pad and runner (or shaft) surfaces is too hot, the bearing can fail. If the oil temperature in the reservoir (often called the pot or bath) is excessive, it will increase the bearing-film temperature, possibly making a well designed bearing operate marginally. The temperature of the bath oil is a controllable variable that usually can be reduced to approximately 120°F.

Many large thrust bearings operate submerged in an oil bath that is maintained at the desired temperature by appropriately sized and located coolers. As Figure 9 shows, the viscous pumping action of the thrust-bearing runner provides a natural oil circulation between pads and around the bearing. Problems have occurred when the natural flow short-circuits the coolers, so that the hot oil leaving the space between pads does not contact the cooling-tube surfaces.

Excessive bath-oil temperatures often can be corrected by adding or relocating the coolers and/or the baffles that control oil flow, or by increasing the size or number of coolers. External circulating systems also can be used to direct the oil through coolers and filters, although the added hardware of the pumping system represents an additional source of problems.

Bearing Surface Irregularities Often Symptoms, Not Problems

The material normally used for the stationary surface of a bearing is high-strength, tin-based babbitt (white metal) bonded to a steel backing. The moving thrust runner and shaft are steel, fully annealed to relieve any residual stresses. Although bearing failures occasionally occur because of poor babbitt bonding and porosity, most bearing manufacturers and service companies have excellent quality control in their babbitting facilities, and material and assembly problems are rare. Still, the hydroturbine owner can avoid this type of failure by requiring 100 percent ultrasonic and die-penetrant tests, performed to established standards.

Babbitt surface cracks and harmful

surface deformations can sometimes be related to the design of the steel bearing shell or pad backing. The source of the problem is the thermal expansion of babbitt having a non-uniform thickness, combined with the dissimilar backing material. Surface cracks and deformations have occurred in the grooving areas in large, sleeve-type guide bearings. Babbitt on bearing pads is normally metallurgically bonded to an ungrooved steel surface. Some old dovetail designs may create surface problems.

Since a bearing pad normally deforms to a convex surface profile, any insert in the steel backing can cause cracks in the babbitt surface, and failure due to fatigue. One failure of that type was caused by a lift orifice plug that was not properly designed and secured in the pad.

Miscellaneous Sources Cause Various Problems

Because hydroturbine fluid-film bearings are designed with a relatively thick lubricant film and excellent bearing materials, they are very tolerant of problems that might damage less "forgiving" bearings. Still, failures due solely to the lubricant do occur. A harmful degree of contamination and/or deterioration can result if the oil is not purified before being put into the lube system, if the piping and bearing housing are not clean, or if the oil is not periodically analyzed and replaced as required.

Instrumentation that monitors bearing performance should be calibrated periodically. Precisely calibrated instruments not only will ensure that dangerous conditions will be detected, they also will not generate false alarms and unnecessary downtime and operator concern.

Because of their relatively tight diametrical clearances, horizontal and inclined-turbine journal bearings should have a reliable and effective self-aligning capability. These bearings normally are highly loaded, and will fail if they cannot compensate for shaft deflections and deformations.

The need for a weld repair in a large bearing support structure must be evaluated very carefully. If completely effective stress relieving is not possible, relaxation distortions after on-site machining could affect pad equalization and/or bearing alignment. With bearing film thicknesses only on the order of 0.0015 inch, any repair could be more dangerous than the structural defect.

Bearing Replacement Is Not A Casual Decision

When small bearings have problems, the decision to replace them is almost automatic. With large bearings, overwhelming justification is needed. The installation and downtime costs of replacing a large and expensive bearing make it difficult to decide to replace it, even with a new and better design. In general, the only justification occurs when excessive maintenance demands result in correspondingly excessive costs and labor requirements.

As an example, a large, non-tilting-pad thrust bearing, designed and installed in the 1920s, had become sensitive to start up, possibly due to a small change in loading. It could be made to start only after several tries, each with much surface conditioning and contouring. The labor required was costly and time-consuming.

Analyses showed that satisfactory

operation of this type of bearing depended on thermal crown deformations, and the bearing required a unique set of conditions to start and operate. A new, optimized tilting-pad bearing was designed to fit into the existing space envelope. The bearing was purchased and installed, and is currently operating without any starting or running problems.

Since the space envelope around an existing bearing usually cannot be increased, a replacement bearing should include all of the best design features currently available to optimize performance. Even if these features are not normally provided on a standard model that fits into the available space, the owner should demand the most conservative, reliable, "special" design possible for the replacement bearing.

Fortunately, few hydroturbine owners are likely to be faced with a replacement decision. Hydroturbines are designed to provide long, trouble-free service and well instrumented to disclose problems. If operators are aware of the signals that alert them to potential harm, and if they act promptly to investigate and evaluate any problems, failures should be averted.

Why Trouble-Shooting Is Worthwhile

By providing routine maintenance and observing the precautions listed in this article, owners of large hydroturbine bearings can keep them operating reliably for the life of the installation.

Several years ago, the owner of a large hydroturbine with a 84-inch-diameter tilting-pad thrust bearing was concerned about excessive thrust-pad temperatures. I was asked to review the design and analyze the bearing performance. My analysis indicated that the bearing was operating under marginal film thickness conditions, in which any adverse change might result in failure. Analysis also showed the bearing was well designed and would operate reliably if the bath oil temperature, which had been high, was kept to about 120°F. Further investigation disclosed that the oil coolers were adequate in size, but, since they were improperly located, they were less effective and unable to maintain a low bath oil temperature. Additional coolers, properly located, solved the problem, and were the most economical solution for this case. As this case illustrates, analysis of a bearing problem and corrective action is far less expensive than the downtime to make repairs to a damaged bearing. □

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