TILTING PAD JOURNAL BEARINGS WITH SPRAY-BAR BLOCKERS AND BY-PASS COOLING FOR HIGH SPEED, HIGH LOAD APPLICATIONS

by
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Dr. Nicholas, a member of ASME, STLE, and the Vibration Institute, has authored over 35 technical papers, concentrating his efforts on tilting pad journal bearing design and application. He received his B.S. degree from the University of Pittsburgh (Mechanical Engineering, 1968) and his Ph.D. degree from the University of Virginia (1977) in rotor and bearing dynamics. Dr. Nicholas holds several patents including one for a spray-bar blocker design for tilting pad journal bearings and another concerning by-pass cooling technology for journal and thrust bearings.

ABSTRACT

With performance considerations driving the design of rotating equipment, operating speeds and journal loads have increased greatly in recent years, often exceeding generally accepted design limits. In order for tilting pad journal bearings to operate at load and speed levels that exceed these limits, newer innovative designs are necessary. This paper concerns the development and implementation of a reduced temperature tilting pad journal bearing design that features spray-bar blockers and by-pass cooling. The sequence of design improvements began with evacuated housings with directed lubrication, then progressed to offset pivoted pads, spray-bar blockers, and then finally to behind-the-pad by-pass cooling. Many design application examples are discussed illustrating the progression of innovations leading to the final reduced temperature bearing configuration. Actual application unit loads and surface speeds start at around 350 psi at 250 ft/sec, progressing to about 495 psi at 285 ft/sec and ending up at approximately 770 psi at 350 ft/sec and 470 psi at 420 ft/sec.

INTRODUCTION

With performance considerations driving the design of rotating equipment, operating speeds and journal bearing static loads have increased greatly in recent years, often exceeding generally accepted design limits. For high performance gearbox applications including integrally geared compressors, the tilting pad pinion bearings are often subjected to both high speeds and high loads. The challenge for the bearing designer is to ensure that these high performance bearings do not operate at elevated temperature levels.

The journal surface speed limit often quoted by bearing designers is 300 ft/sec, 91 m/sec (Nicholas, 1994). For example, Herbage (1972) states that "Turbulence and hot oil carryover can cause severe problems at speeds beyond 18,000 feet per minute" (300 ft/sec). In another example from Zeidan and Paquette (1994), "The upper limiting value typical of some of the highly loaded bearing applications is usually set to 300 ft/sec."

The tilting pad journal bearing unit load is defined as the journal resultant load (gravity plus gear) divided by the product of the bearing diameter times the pad axial length, $L_u = W_r/(L*D)$. Quoted design unit load limits are between 200 psi, 1.4 MPa (Nicholas, 1994) and 300 to 350 psi, 2.1 to 2.4 MPa (Zeidan and Paquette, 1994). The lower limit is generally associated with nongeared units while the higher limit is associated with geared units. Geared units can tolerate higher unit load levels since the gear loading diminishes greatly at slow speeds during startup and shutdown when bearing babbitt surfaces wear from supporting heavy journal loads at insufficient rotor speeds.

In order for tilting pad journal bearings to operate at load and speed levels that exceed the generally accepted limits quoted above, newer innovative designs are necessary. One motivation to operate at higher surface speeds has been driven partly by the stringent rotordynamics specifications that turbomachinery manufacturers must adhere to in order to sell their rotating equipment. Large journal diameters result in less severe critical speeds. However, larger diameter journals result in high surface velocities, often exceeding 300 ft/sec. Approaching or exceeding a 300 ft/sec journal surface speed would mean operating in the turbulent regime (Edney, et al., 1996). Abramovitz (1977) states that "There is an upper limit in speed beyond which ... turbulence occurs. This need not be feared or avoided, since turbulence ... has no intrinsic harmful effects on bearing behavior. In effect, it can be treated as equivalent to operating with an increased lubricant viscosity. This then results in higher power loss, load capacity, and temperature rise ...." From this statement, one can infer that exceeding 300 ft/sec with turbulent operation should not be a problem as long as care is taken to ensure that the bearing is properly cooled.

To address bearing operation in excess of 300 ft/sec and/or 350 psi, many advanced tilting pad journal bearing designs have been developed. One of the earliest designs developed in 1980 (refer to ACKNOWLEDGEMENT) was called directed lubrication and featured spray nozzles that sprayed lubricating oil toward the leading and trailing edges between each set of pads. Other more recent designs include a leading edge groove (LEG) design (Dmochowski, et al., 1993, and Brockwell, et al., 1994), a Pocket Feed® design (Gardner, 1994), a leading edge "flow director" design called the Advantage™ (Ball and Byrne, 1998), and a spray-bar blocker with by-pass cooling design (Nicholas, 1998, 2001, and 2002). One feature that is common in all of these designs is the more efficient use of cool inlet oil.

This paper concerns the development and implementation of the spray-bar blocker with by-pass cooling reduced temperature tilting pad journal bearing design for high speed and high load applications. The sequence of design improvements began with evacuated housings and conventional spray bars for directed lubrication (Nicholas, 1994), then progressed to offset pivoted pads (Simmons and Lawrence, 1996), spray-bar blockers (Nicholas, 1998 and 2001), and then finally to behind-the-pad by-pass cooling (Nicholas, 2002). Many design application examples are discussed illustrating the progression of innovations leading to the final reduced temperature bearing configuration. Actual application unit loads and surface speeds start at around 350 psi at 250 ft/sec, progressing to about 495 psi at 285 ft/sec, and ending up at approximately 770 psi at 350 ft/sec and 470 psi at 420 ft/sec.

FLOODED VERSUS EVACUATED HOUSINGS

Figure 1 illustrates a 1980s vintage flooded, pressurized housing design with oil inlet nozzles and drain holes between each set of pads. This was the best tilting pad bearing available at the time for cool operation. As speeds and loads increased, increasing the bearing oil flow often produced only marginal decreases in bearing operating temperatures. Two major problems associated with cooling a tilting pad bearing are illustrated in Figure 2. Unfortunately, a substantial percentage of hot oil is carried over by the journal from the trailing edge of the upstream pad into the leading edge of the next downstream pad. Secondly, the cool inlet oil mixes with the discarded preceding pad's hot oil before it enters the downstream pad's leading edge (Nicholas, 1994).

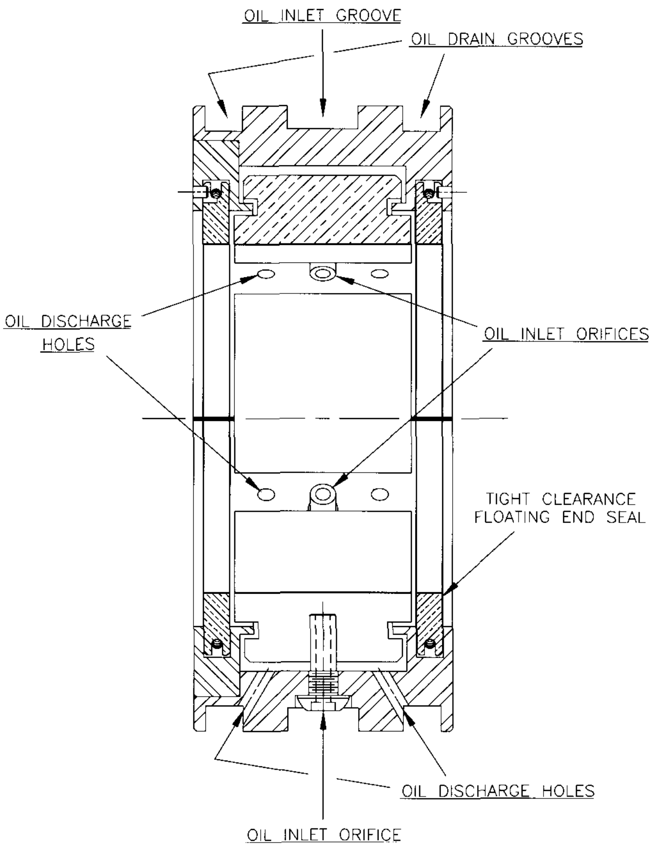

Figure 1. Flooded Pressurized Housing Design with Oil Inlet Nozzles.

It was soon discovered that better results were achieved by applying the cool inlet oil more effectively. One method of reducing the carryover and hot-cool oil mixing is to introduce cool inlet oil directly into the pad's leading edge (Dmochowski, et al., 1993; Brockwell, et al., 1994; Gardner, 1994; Ball and Byrne, 1998; and Nicholas, 1994, 1998, and 2001). This effectively prevents mixing and blocks some of the hot oil carryover. Tanaka (1991) published results showing the cooling effect of removing the end seals of a bearing design similar to the one shown in Figure 1. Tanaka's test bearing is illustrated in Figure 3 with sample results shown in Figure 4.

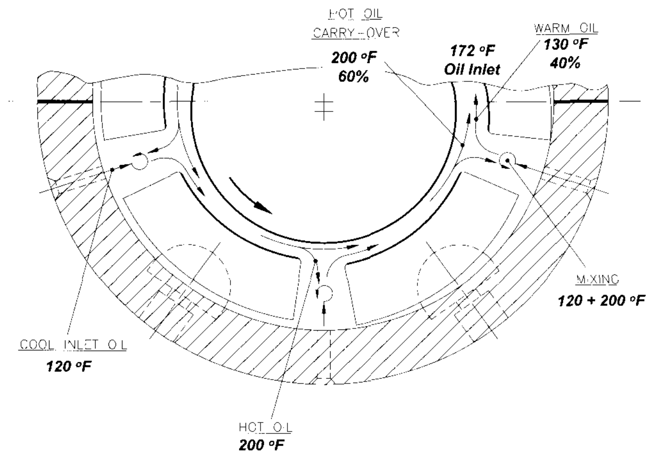

Figure 2. Pad-to-Pad Hot Oil Carryover.

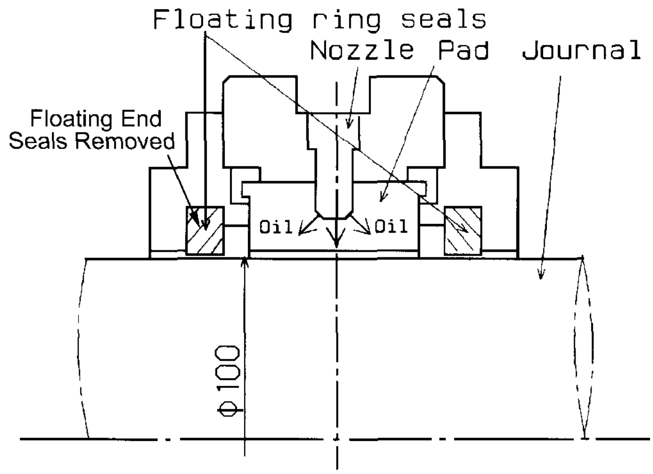

Figure 3. Cooling Effect of Removing End Seals. (Courtesy, Tanaka, 1991)

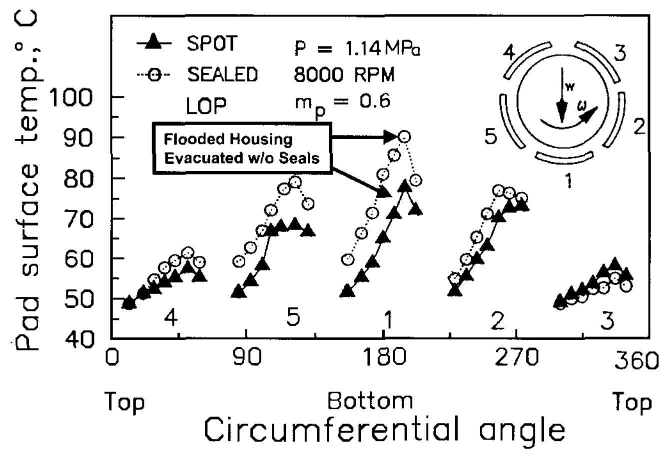

Figure 4. Flooded Versus Evacuated Housing. (Courtesy, Tanaka, 1991)

Tanaka's test data led to the design shown in Figure 5 that features conventional spray bars, open end seals, and open housing drains (i.e., an evacuated housing with directed lubrication, Nicholas, 1994). From Figure 6, the effectiveness of the evacuated housing, directed lubrication design of Figure 5 is compared to the flooded, pressurized housing design of Figure 1 for a 4.0 inch four pad tilting pad journal bearing. At 15,000 rpm, 262 ft/sec, the pressurized housing embedded temperature sensor reading was 235°F compared to the evacuated housing embedded temperature sensor reading of 198°F (Nicholas, 1994).

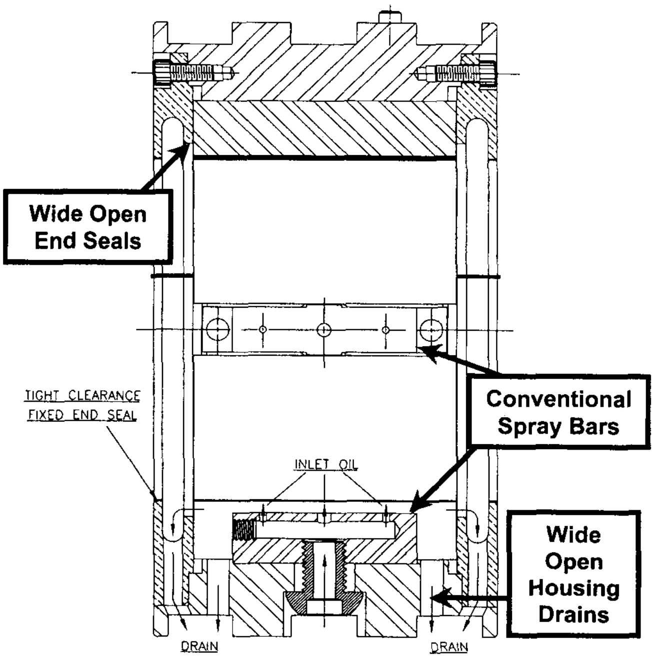

Figure 5. Evacuated Housing, Directed Lubrication Spray Bar Design.

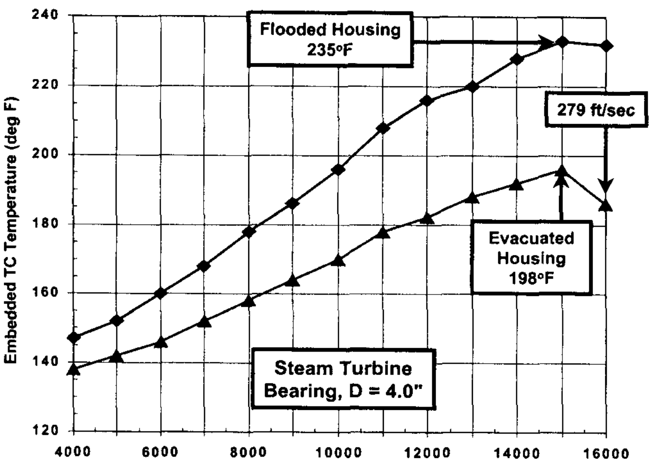

Figure 6. Flooded Versus Evacuated Housing with Directed Lubrication.

Other authors have also made similar comparisons. Harangozo, et al. (1991), compared flooded lubrication, directed lubrication (i.e., evacuated housing), and leading edge lubrication for a four pad tilting pad journal bearing. They concluded that "... over the speed range and load range of the tests, directed lubrication temperatures of pads 1 and 2 are lower than the corresponding flooded lubrication temperatures." Pads 1 and 2 are the loaded pads. However, their test bearing's pad preload range was negative 0.62 to negative 0.75. Large negative preloads are not normally encountered in bearings for industrial turbomachinery.

More results may also be found in DeCamillo and Brockwell (2001) comparing a conventional flooded five pad tilting pad journal bearing design to an evacuated LEG design. The test bearing was 6.0 inches (152 mm) in diameter operating at 320 psi and 279 ft/sec.

Interestingly, the evacuated housing, directed lubrication design concept has been successfully applied to tilting pad thrust bearings to lower operating temperatures well before 1991. Discussions concerning this may be found in Herbage (1977) and, more recently, Whalen (1996).

OFFSET PIVOTS

A well-known design tool for reducing operating temperature and increasing load capacity in tilting pad thrust bearings is to offset the pivots. Test results showing operating temperature reductions for offset pivoted thrust pads compared to centrally pivoted pads may be found in Gardner (1975 and 1988). Gardner's 1988 results "... indicate that pad temperatures decrease significantly as the pivot is advanced from the central (50 percent) position to at least the 70 percent position ...." Also, theoretical predictions correlated with test data may be found in Whalen (1996) comparing centrally pivoted thrust pads to 60 percent offset pivoted pads.

Offset pivots may also be applied to tilting pad journal bearings. Simmons and Dixon (1994) present temperature data for a 200 mm (7.87 inch) five pad tilting pad bearing for unit loads up to 600 psi and surface velocities up to 345 ft/sec for centrally pivoted pads. Offset pivoted pads are considered in Simmons and Lawrence (1996) where temperature comparisons are made showing a significant pad operating temperature reduction for offset pivots. Some sample results are illustrated in Figure 7. For 500 psi, maximum temperatures are reduced from 245°F for centrally pivoted pads to 215°F for 60 percent offset pivoted pads.

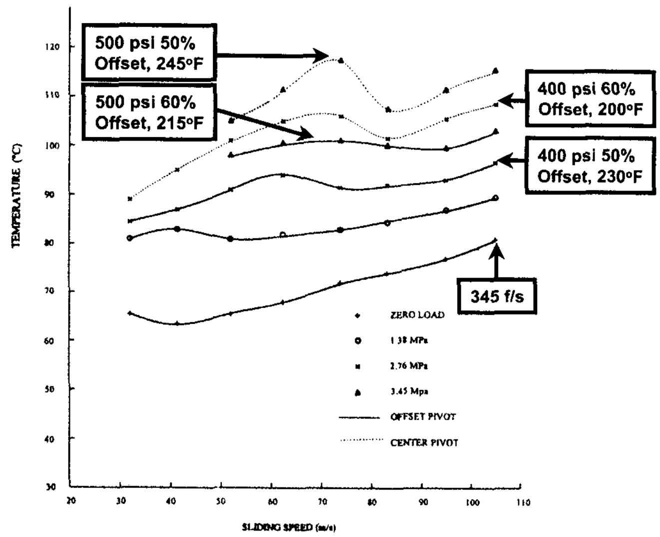

Figure 7. 50 Versus 60 Percent Offset Pivoted Pads. (Courtesy, Simmons and Lawrence, 1996)

Bouchoule, et al. (1995), tested a 160 mm (6.3 inch) five pad journal bearing operating at 100 m/sec (328 ft/sec) with loads up to 3.5 MPa (508 psi). They conclude that "Shifting the pivot from the central position to at least the 55 percent position leads to a decrease of the maximum temperature of about 15°C ...."

Brockwell, et al. (2001), presents pad operating temperature data for a 6.0 inch five pad tilting pad bearing, comparing offset pivoted to centrally pivoted pads for unit loads up to 320 psi and surface velocities up to 279 ft/sec. The authors concluded that "In all test conditions, the offset pivot bearing ran cooler than the center pivot bearing. In some cases the difference was as much as 20°C." Similar data may also be found in DeCamillo and Brockwell (2001).

From these results, it is clear that offsetting the pad pivots is an effective design tool in reducing tilting pad journal bearing operating temperatures compared to centrally pivoted pads.

16,000 HP Pinion Bearing

Armed with the knowledge that evacuated housings, conventional spray bars for directed lubrication, and offset pivots all contribute to cooling a tilting pad journal bearing, these design features are utilized for a 16,000 hp high speed pinion bearing application. The retrofit design is illustrated in Figure 8 and features triple orifice conventional spray bars, open end seals with an inner seal tip clearance of 50 mils, and a 65 percent pivot offset. The journal diameter is 5.75 inches with a 11,635 lb resultant load, 352 psi, operating at 9869 rpm, 248 ft/sec. A photo of the actual bearing is shown in Figure 9. After retrofitting seven gearboxes with this design, the bearings have been running since 1998 free of journal bearing problems. Sample embedded temperature sensor readings for three of the boxes are 171/180°F, 184/176°F, and 183/176°F for the nondrive-end/drive-end high speed pinion bearings at full load.

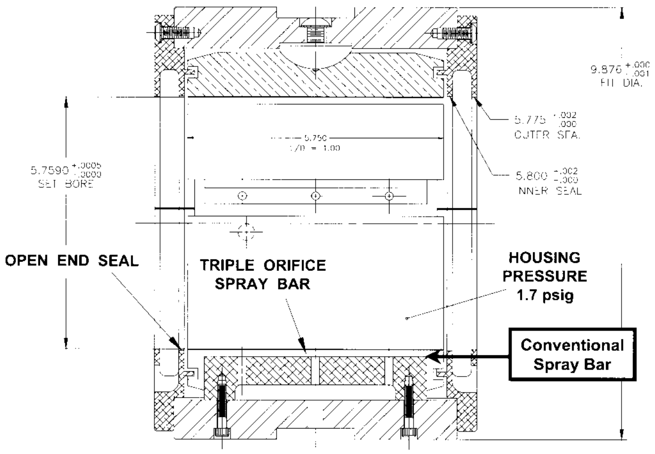

Figure 8. 16,000 HP Pinion Bearing, Evacuated Housing with Directed Lubrication.

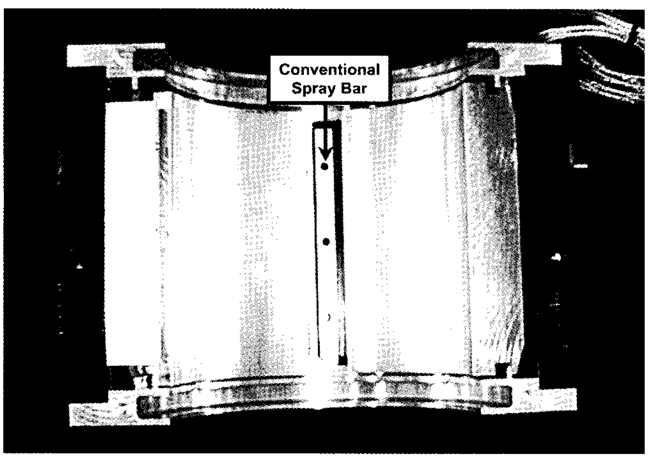

Figure 9. 16,000 HP Pinion Bearing, Lower Half with Conventional Spray Bars.

SPRAY-BAR BLOCKERS

In order to handle increasingly severe applications, more efficient use of the cool inlet oil is necessary. A design innovation that was developed to address this issue is the spray-bar blocker (Nicholas, 1998 and 2001). Illustrated in Figure 10, the design features a spray-bar blocker between each tilting pad that effectively blocks much of the pad-to-pad hot oil carryover and directs that hot oil into the housing drain holes located between each set of pads. It also sprays cool inlet oil directly into the leading edge of the downstream pad. Besides allowing the cool inlet oil to directly enter the pad leading edge, the design prevents the trailing edge hot oil and the cool inlet oil from mixing. It also minimizes hot oil carryover and, with the wide open housing drains and open end seals, the evacuated housing eliminates churning losses. A photo showing a spray-bar blocker and the open housing drains is shown in Figure 11.

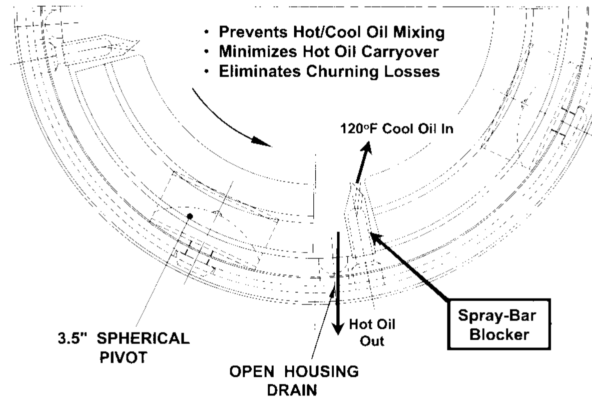

Figure 10. Spray-Bar Blocker Design to Minimize Hot-Oil-Carryover.

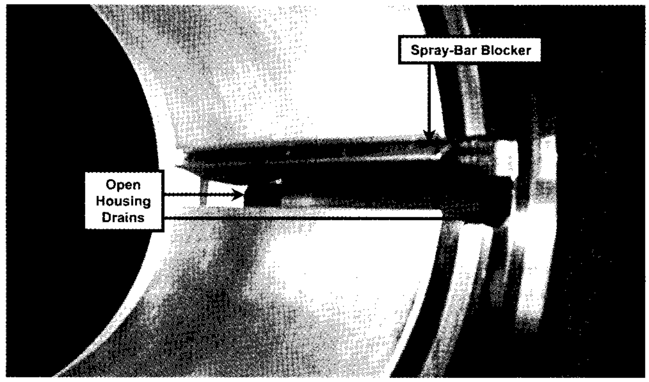

Figure 11. Spray-Bar Blocker with Open Housing Drains.

54,000 HP Pinion Bearing

With the spray-bar blocker, evacuated housing, and offset pivot design features, a more severe application can now be addressed: a high speed pinion bearing in a new 54,000 hp gearbox (Evans, et al., 2000). This design is illustrated in Figure 12 and features five orifice spray-bar blockers, open end seals with an inner seal tip clearance of 0.5 inches, wide open housing drains, and a 65 percent pivot offset. The journal diameter is 10.0 inches with a 34,108 lb resultant load, 341 psi, operating at 5546 rpm, 242 ft/sec. Photos of the actual bearing are shown installed in the gearbox in Figures 13 and 14 with the upper, loaded half shown in Figure 15. At the time, this was the highest horsepower gearbox ever manufactured by that particular gear vendor. The box was commissioned in 2000 and has been operating ever since except for a brief downtime to change gear sets. The bearings have been operating at full load with embedded temperature sensor readings of under 85°C (185°F).

BY-PASS COOLING

Chen, et al. (1999 and 2001), presented an isothermal journal bearing that contained methanol in heat transfer chambers located circumferentially behind the bearing's outside diameter as shown in Figure 16. As the liquid methanol heats up at the bottom, loaded half of the bearing, the methanol vaporizes and rises to the top, carrying away the heat and transferring it to the top half of the bearing. Figure 17 shows typical results comparing the operating temperatures of a conventional bearing to the much lower temperatures of the isothermal bearing.

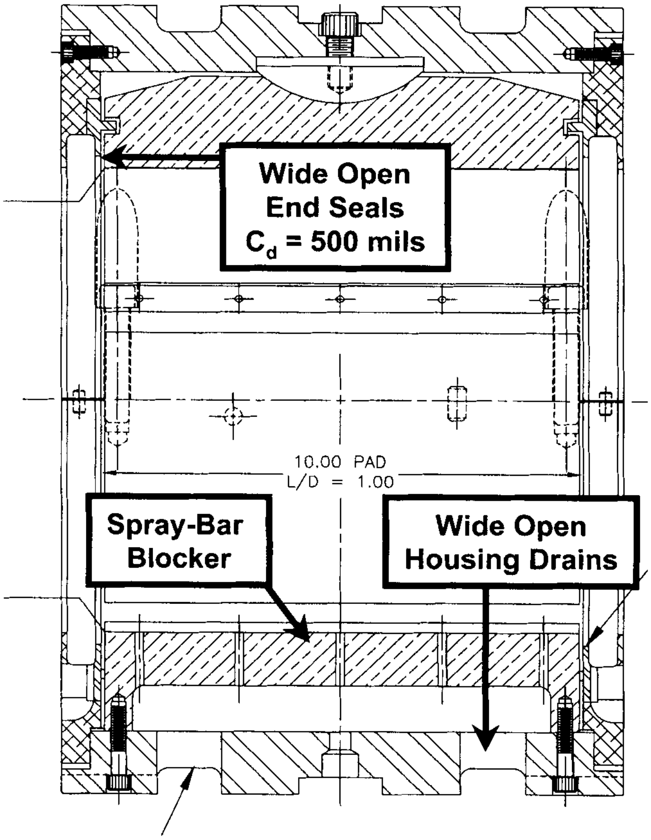

Figure 12. 54,000 HP Pinion Bearing, Evacuated Housing with Spray-Bar Blocker.

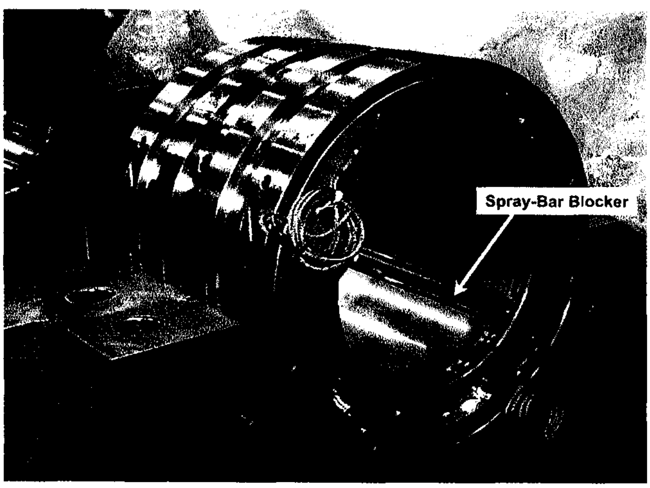

Figure 13. 54,000 HP Pinion Bearing with Spray-Bar Blocker in Gear Case.

While the isothermal journal bearing technology may not be practical in industrial application, the concept has merit and can be applied to a tilting pad journal bearing. The innovation is called behind-the-pad by-pass cooling and it utilizes cool inlet oil instead of methanol to transfer heat away from the babbitt surface (Nicholas, 2002). As shown in Figure 18, using a babbitted chrome copper pad with circumferential heat transfer chambers, cool inlet oil is directed onto the back of the pad and into the heat transfer grooves. Figure 19 illustrates how the spray-bar blocker directs the by-pass cooling oil directly into the housing drains. The by-pass cooling oil does not participate in lubricating the bearing but serves only to transfer heat. This design takes advantage of the high thermal conductivity of chrome copper, which is 187 Btu/(hr*ft*°F) compared to 29 for steel and 32 for bronze.

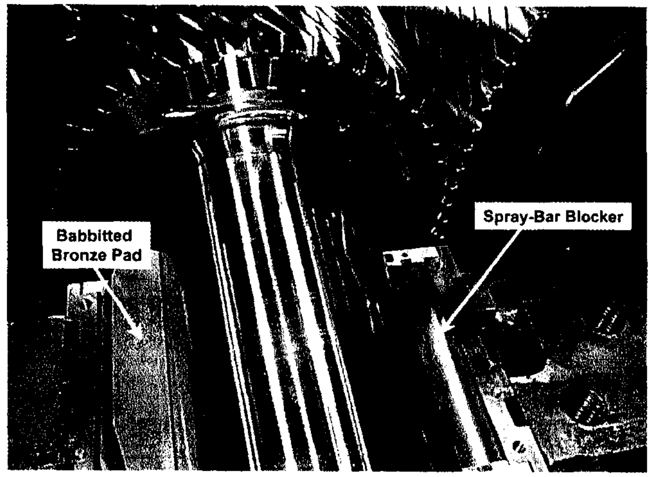

Figure 14. 54,000 HP Pinion Bearing, Lower Half in Gear Case.

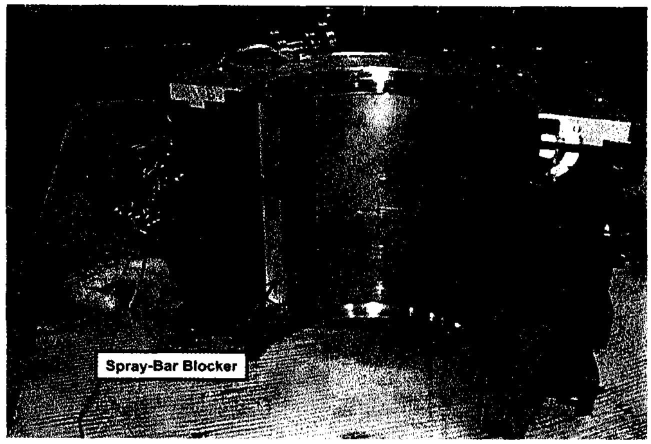

Figure 15. 54,000 HP Pinion Bearing, Upper Loaded Half.

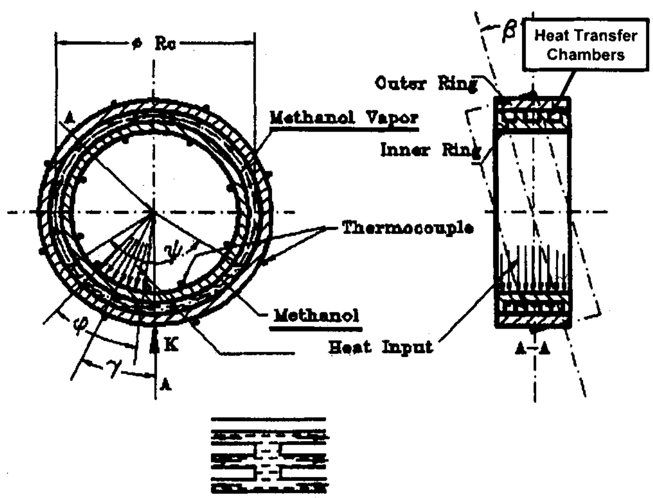

Figure 16. Isothermal Journal Bearing. (Courtesy, Chen, et al., 1999 and 2001)

Babbitted copper alloy journal pads have been known to reduce tilting pad journal bearing operating temperatures. Bouchoule, et al. (1995), concludes that "The bearing design … for which the

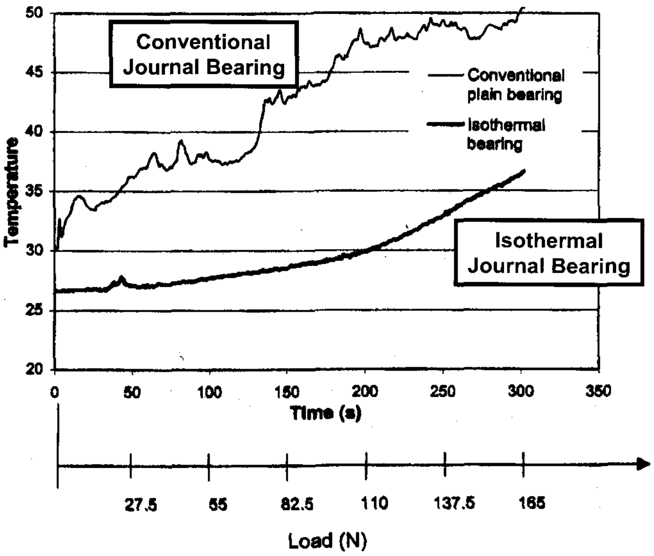

Figure 17. Isothermal Journal Bearing Test Results. (Courtesy, Chen, et al., 2001)

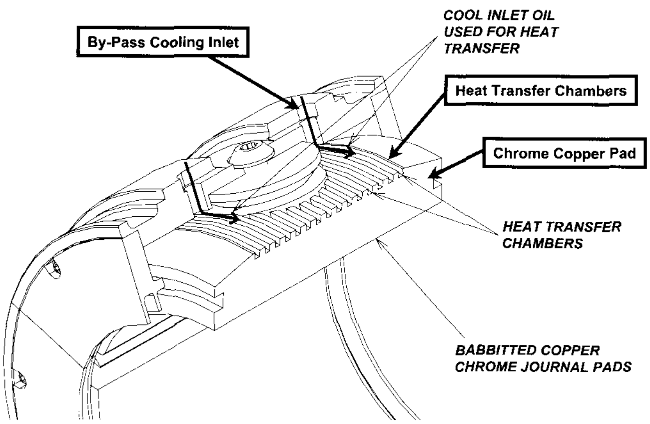

Figure 18. Behind-the-Pad By-Pass Cooling, Bearing Schematic.

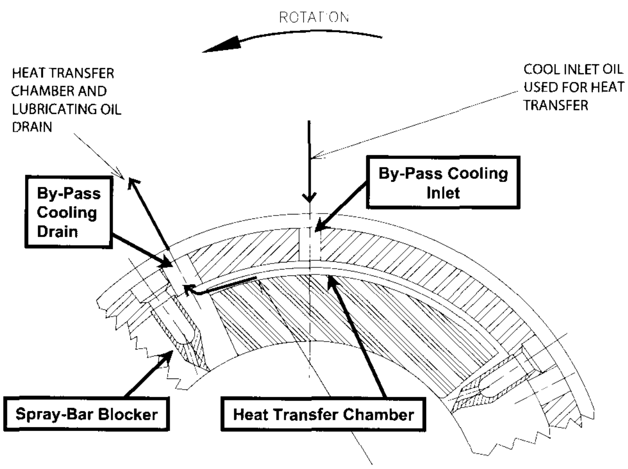

Figure 19. Behind-the-Pad By-Pass Cooling, Pad Schematic.

pads are made in copper-bronze alloy, with spherical offset pivots, contribute to improve its performance, notably temperature (21°C at nominal conditions) and power loss."

As before with evacuated housings, directed lubrication, and pad pivot offset, utilizing copper alloy pads in thrust bearings has been a well-known design tool to reduce tilting pad bearing operating temperatures. For example, Gardner (1975) presents test data for a 6.0 inch tilting pad thrust bearing showing significantly lower operating temperatures with babbitted copper alloy pads compared to babbitted steel pads.

34,500 HP Pinion Bearing

Now with the by-pass cooling, babbitted chrome copper pad, spray-bar blocker, evacuated housing, and offset pivot design features, a much more severe application can be addressed: a high speed pinion bearing in a new 34,500 hp gearbox. This design is illustrated in Figure 20 and features chrome copper pads, behind-the-pad by-pass cooling, five orifice spray-bar blockers, wide open end seals, wide open housing drains, and 65 percent offset pivots. The journal diameter is 6.0 inches with a 17,783 lb resultant load, 494 psi, operating at 10,896 rpm, 285 ft/sec. Photos of the chrome copper pads are shown in Figure 21. Note the 65 percent offset spherical pivot and the heat transfer chambers.

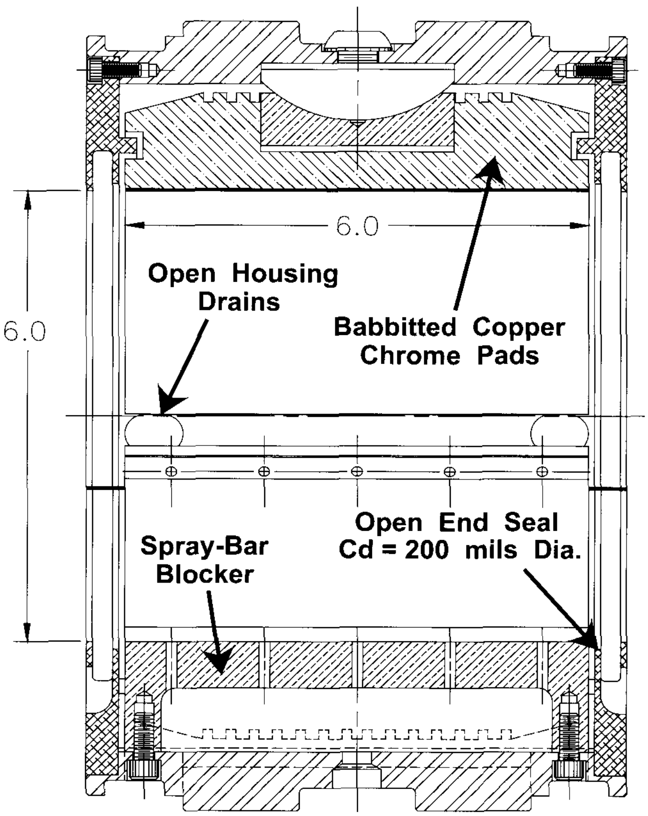

Figure 20. 34,500 HP Pinion Bearing with Chrome Copper Pads and By-Pass Cooling.

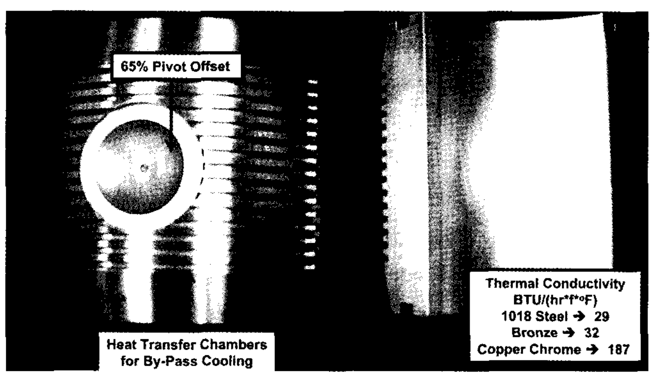

Figure 21. 34,500 HP Pinion Bearing, 65 Percent Offset Chrome Copper Pads with Heat Transfer Chambers.

During no-load testing with an oil inlet temperature of 119°F, the full speed embedded temperature sensor readings were 187, 192, 192, and 194°F without the by-pass cooling. Rerunning the test with by-pass cooling, the full speed embedded temperature sensor readings were 172, 172, 173, and 172°F. Total oil flow to the gearbox was 208 gpm without by-pass cooling and 214 gpm with by-pass cooling, a difference of 6.0 gpm or 3.0 gpm per bearing. During commissioning, these gearboxes ran at full load in 2002 with pinion bearing embedded temperature sensor readings under 190°F.

16,500 HP Pinion Bearing

Consider a more recent example: a high speed pinion bearing in a new 16,500 hp gearbox. This design is similar to Figures 20 and 21 featuring chrome copper pads, behind-the-pad by-pass cooling, spray-bar blockers, wide open end seals, wide open housing drains, and 65 percent offset pivots. The journal diameter is 5.25 inches with a 9562 lb resultant load, 463 psi, operating at 14,050 rpm, 322 ft/sec. After commissioning in early 2003, this gearbox operated at 60 percent load with pinion bearing embedded temperature sensor readings of 156, 164, 165, and 156°F.

CENTRIFUGAL COMPRESSOR APPLICATIONS

Gearboxes are not the only application for this technology. High speed centrifugal compressor bearings also require special features to keep the bearings operating at a reasonable temperature. Compressor bearings often operate at higher surface speeds in order to satisfy the stringent rotordynamics specifications. Large journal diameters result in less severe critical speeds, pushing surface velocities near 300 ft/sec.

A typical centrifugal compressor bearing with an evacuated housing, conventional spray bars for directed lubrication, and bronze pads is shown in Figure 22. Many have been used on new compressor applications where the loading is relatively light (under 100 psi) but with surface velocities ranging from 192 to 229 ft/sec. All applications produce operating temperatures in the 170 to 185°F range.

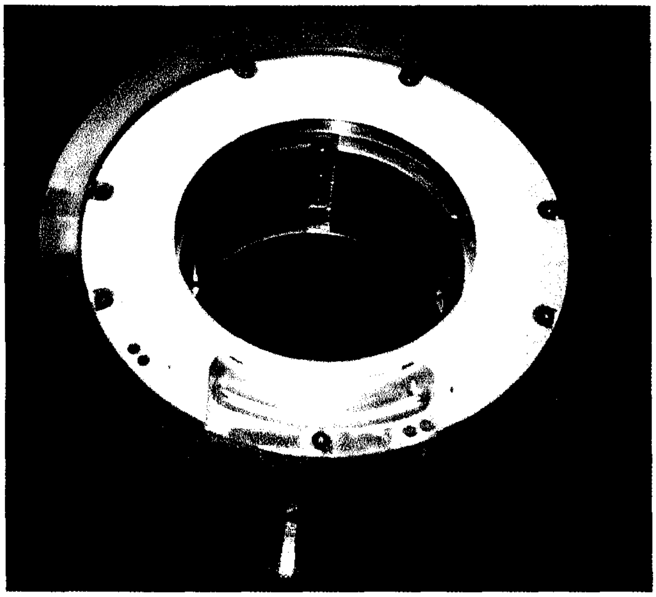

Figure 22. Evacuated Housing, Directed Lubrication Design for Centrifugal Compressor Applications.

A slightly more severe application is illustrated in Figure 23 where this 4.0 inch bearing operates at 245 ft/sec. This centrifugal compressor bearing design features chrome copper pads, by-pass cooling, spray-bar blockers, an evacuated housing, and offset pivots. Field operation at full speed resulted in embedded temperature sensor readings of 166, 165, 157 and 156°F.

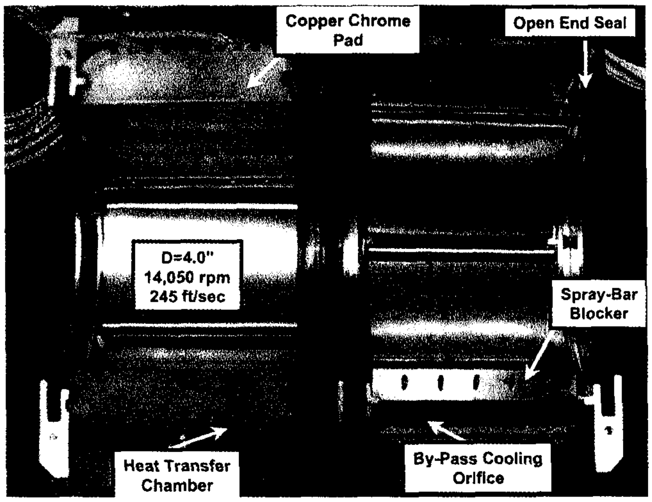

Figure 23. Centrifugal Compressor Application with Chrome Copper Pads and By-Pass Cooling.

Other successful centrifugal compressor applications include a 8.5 inch tilting pad bearing with conventional spray bars for directed lubrication, bronze pads, and an evacuated housing operating at 206 ft/sec with test stand temperature readings between 162 and 182°F. This compressor is part of the train that contains the 54,000 hp gearbox discussed in the spray-bar blockers section. The compressor has been in operation since 2000 without bearing problems.

Two other severe applications include a 6.0 inch tilting pad bearing running at 247 ft/sec and a 7.0 inch tilting pad bearing running at 289 ft/sec, both with conventional spray bars for directed lubrication, bronze pads, and evacuated housings. With an oil inlet temperature of 134°F, full speed test stand operation resulted in embedded temperature sensor readings of 187°F for the 6.0 inch bearing and 182°F for the 7.0 inch bearing.

MISAPPLICATIONS AND DISADVANTAGES

Since Tanaka (1991) published his test results, many designers have tried to emulate the open end seal, evacuated housing design. Since this design features wide open end seals and often wide open housing drains, the housing cavity pressure is very near atmospheric, often below 1.0 psig. Conversely, conventional flooded housing designs have typical housing pressures that range from 5 to 15 psig for a 20 psig oil inlet. The low housing pressures created by evacuated housings can lead to oil starvation and subsynchronous vibration if improperly applied. Figure 24 illustrates one such improper application. This gearbox pinion bearing design with open end seals produced a subsynchronous vibration due to a dry friction rub from oil starvation. With long L/D = 1.0 pads, the single oil inlet orifice hole in the housing between each set of pads was insufficient to distribute the inlet oil along the entire pad leading edge axial length. The problem was solved with a conventional spray bar retrofit as in Figure 9.

Another misapplication of an evacuated housing design may be seen in Figure 25. This gearbox pinion bearing, with long L/D = 1.0 pads, contains no end seals. The mushroom oil nozzle fails to provide lubricant to the pad axial ends resulting in oil starvation, a dry friction rub, and subsynchronous vibration. Again, the problem was solved with a conventional spray bar retrofit similar to Figure 9. The resulting babbitt damage from oil starvation and a dry friction rub may be seen in Figure 26.

An evacuated housing design may not work in a high speed balance vacuum. Even though the inlet oil is introduced to the housing through a spray bar at around 20 psig, the housing cannot maintain a positive pressure and the oil immediately atomizes, resulting in oil starvation. This again may manifest itself as a susynchronous

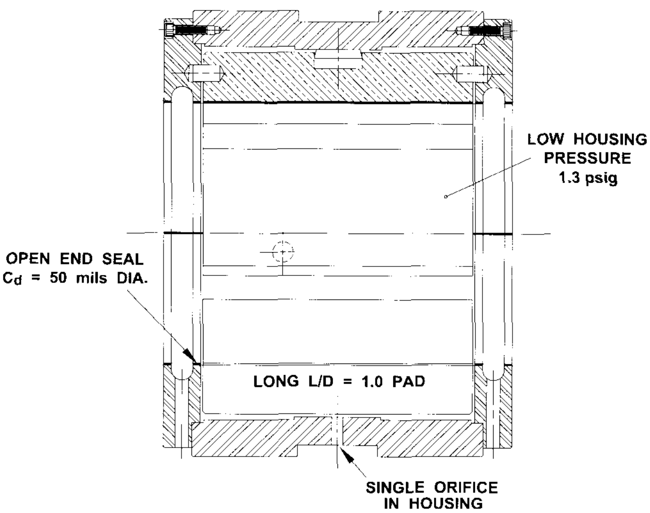

Figure 24. Evacuated Housing Misapplication #1—Oil Starvation, Dry Friction Rub, Subsynchronous Vibration.

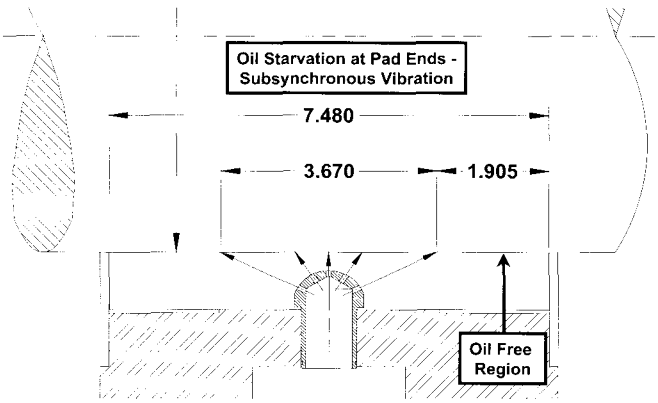

Figure 25. Evacuated Housing Misapplication #2—Oil Starvation, Dry Friction Rub, Subsynchronous Vibration.

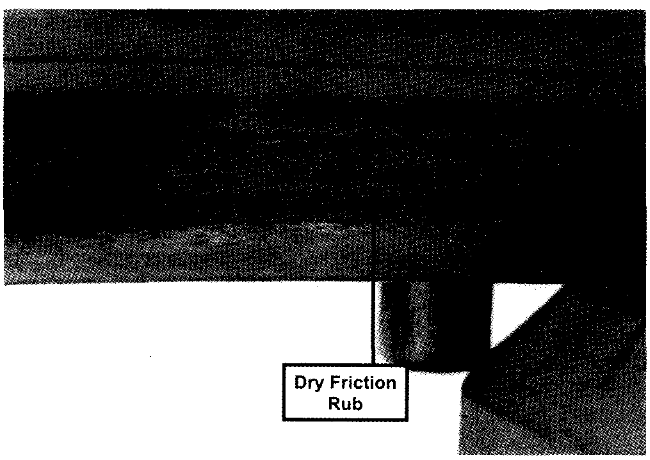

Figure 26. Evacuated Housing Misapplication #3—Dry Friction Rub Babbitt Damage, Subsynchronous Vibration.

rotor vibration. The solution is to use dummy end seals with a reasonable clearance and, if they exist, to temporally block the open housing drains. Dummy end seal clearances of around twice the bearing clearance will produce a reasonable housing pressure of around 10 psig. In this case, the inlet oil exiting the spray bars will not atomize even in a high speed balance vacuum bunker.

Another disadvantage of the evacuated housing design for tilting pad bearings is the lack of lubricant contained in the housing. Of course, that is one of the major reasons that makes the design so effective in lowering operating temperatures. However, the lubricant encased in a flooded housing acts as a reserve during coastdown in the event of a trip caused by a lubricant supply failure. If this is a concern, a gravity fed rundown tank may be utilized.

A disadvantage of offset pivoted pads is that they are not bi-rotational. That is, their performance deteriorates if the rotor reverse rotates (DeCamillo, 2003). For example, a 220 mm (8.661 inch) ethylene compressor tilting pad journal bearing with four 60 percent offset pivoted pads operates at 4900 rpm with a unit load of 153 psi. The deterioration in bearing performance for reverse rotation may be seen in Figures 27 and 28. At 4900 rpm, there is a 19 percent decrease in minimum film thickness (Figure 27) and a 24 percent increase in pad temperature (Figure 28) for 40 percent offset pads (reverse rotation) compared to 60 percent offset pads. While this does represent a degradation in bearing performance, reverse rotation normally occurs for a short time duration and at low speeds. Reverse rotation would be even less of a problem for gearbox journal bearings since the gear load, which can be up to 95 percent of the resultant load, is essentially zero during reverse rotation.

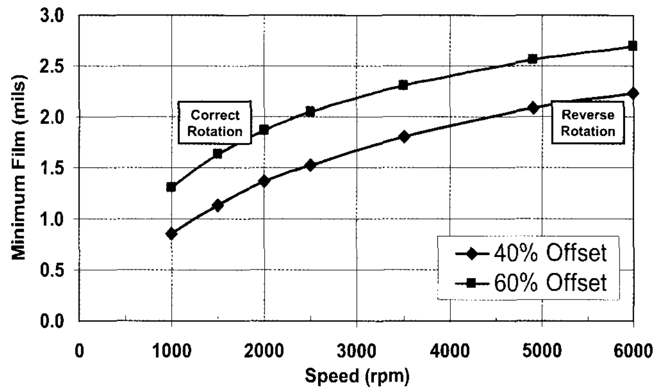

Figure 27. Effect of Reverse Rotation on Minimum Film Thickness, 60 Percent Offset Pivots.

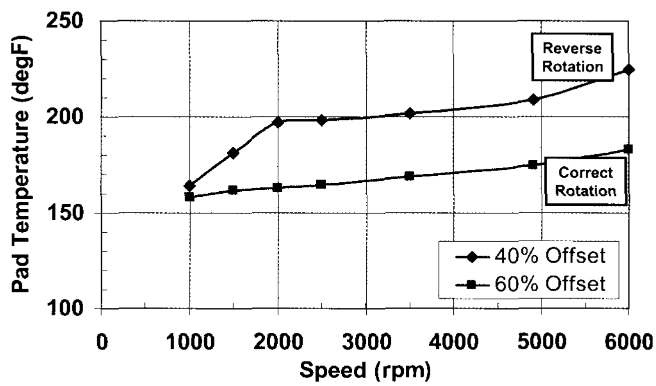

Figure 28. Effect of Reverse Rotation on Maximum Pad Temperature, 60 Percent Offset Pivots.

The disadvantages of by-pass cooling with chrome copper pads are summarized below:

- About a 10 percent increase in oil flow is necessary for the by-pass cooling compared to a conventional bearing.
- Since the by-pass cooling removes a considerable amount of heat, the lubricant drain temperature may be higher than anticipated. This is more a function of the application since both high speeds and high loads contribute to high heat production in the oil

film. In any case, care must be taken to properly size the oil coolers.

- The chrome copper pads that are subjected to by-pass cooling may thermally distort more than a steel pad or a bronze pad. Further investigation is necessary.
- Chrome copper is a more expensive material, and it is more difficult to machine compared to steel or bronze. This adds to the cost of the bearing.
- Chrome copper is susceptible to hydrogen sulfide contamination. A degasser is necessary when used in conjunction with oil seals.
- Chrome copper is susceptible to embrittlement of the babbitt bond causing separation of the babbitt lining. This tends to occur during operation at elevated temperature levels over an extended time period. Additional care must be taken during babbitting to prevent this from occurring.
- Steel backed babbitted pads are required by specifications set fourth by the American Petroleum Institute, API, and an exception must be made to the API specifications in order to use chrome copper material. The specification (API 617, 2002) states that "The bearings shall be precision bored with steel-backed babbitted replaceable liners, pads, or shells." Often for severe applications, users will specify an alternate pad material such as bronze or chrome copper.
- As bearing loads increase, the unloaded pads may have a tendency to flutter. While pad flutter is not directly caused by the by-pass cooling design, it may indirectly cause this problem by allowing tilting pad journal bearings to operate at much higher loads. Pad flutter may be easily eliminated by chamfering or profiling the leading edge of the unloaded pads (Edney, et al., 1996).

ULTRA SEVERE APPLICATION TEST DATA

An ultra severe high speed pinion bearing application requires a 6.0 inch tilting pad journal bearing to run at 10,723 rpm, 281 ft/sec, with a load of 716 psi and at 16,000 rpm, 419 ft/sec, with a load of 471 psi. Another pinion application requires a 5.0 inch tilting pad journal bearing to run at 16,000 rpm, 349 ft/sec, with a load of 769 psi and at 20,000 rpm, 436 ft/sec, with a load of 388 psi. The bearings were designed with by-pass cooling, chrome copper pads, spray-bar blockers, evacuated housings, and 65 percent offset pivots. The 5.0 inch bearing was tested by the gear vendor (refer to ACKNOWLEDGEMENT) prior to testing the gearbox. The test setup is shown in Figure 29. The bottom half of the bearing and the test rotor are shown in Figure 30. The chrome copper pad with heat transfer chambers can clearly be seen along with one of the spray-bar blockers.

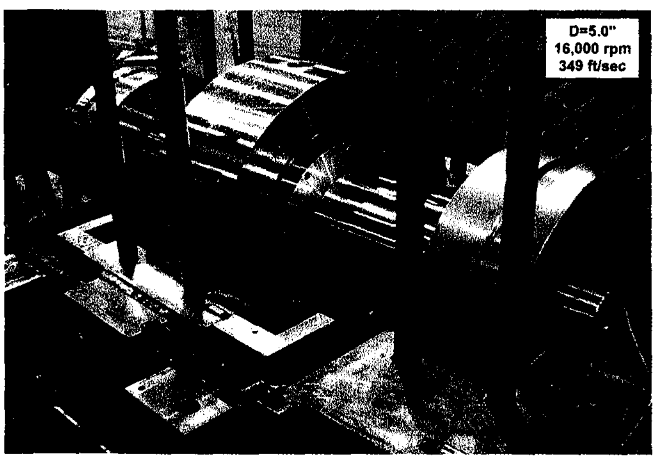

Figure 29. High Speed, High Load Bearing Test Rig. (Courtesy Lufkin Industries)

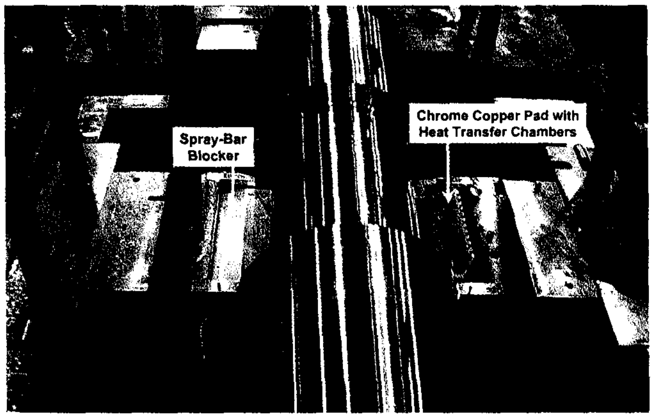

Figure 30. Ultra Severe Application with By-Pass Cooling—Test Setup. (Courtesy Lufkin Industries)

The results of the bearing's embedded temperature sensor readings as a function of applied load at a constant speed of 16,000 rpm, 349 ft/sec, is shown in Figure 31. The two lines on the plot represent the two embedded temperature sensors located upstream of the pad pivot and on either side and equidistant off of the pad's axial centerline. The load was increased with a hydraulic ram up to 862 psi where the temperature sensors read below 220°F. A photo of the lower, loaded half of the bearing is shown in Figure 32. The babbitt surface looks as-new with no evidence of wiping.

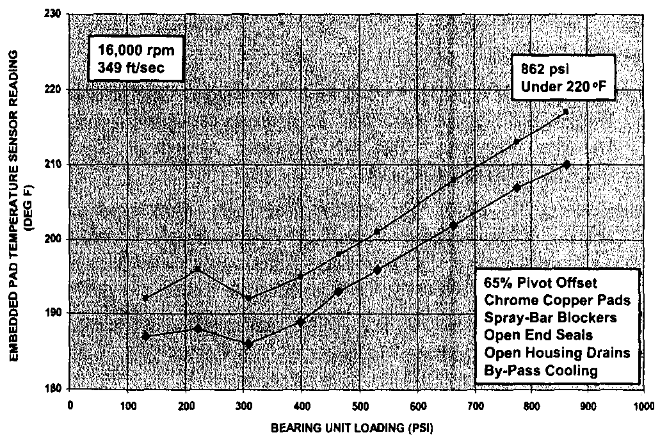

Figure 31. Ultra Severe Application with By-Pass Cooling—Test Results.

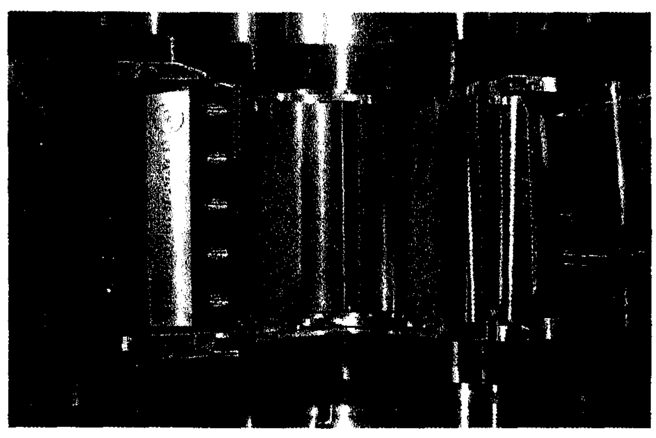

Figure 32. Ultra Severe Application—Loaded Pads after Test with No Babbitt Damage.

CONCLUSIONS

The following design features help to reduce tilting pad bearing operating temperatures:

- Evacuated housings with spray bars for directed lubrication
  - Wide open end seals
  - Wide open housing drains
- Offset pivots up to 65 percent
- Spray-bar blockers
- Babbitted chrome copper pads
- Behind-the-pad by-pass cooling

These features were successfully applied to several high speed, high load gearbox pinion tilting pad journal bearing applications ranging in severity from moderate to ultra. Actual application unit loads and surface speeds ranged from around 350 psi at 250 ft/sec, progressing to about 495 psi at 285 ft/sec, and ending up at approximately 770 psi at 350 ft/sec and 470 psi at 420 ft/sec. Several high speed centrifugal compressor tilting pad journal bearing applications also utilized these features for successful operation ranging from 192 to 245 ft/sec.

NOMENCLATURE

C_d = Bearing diametral clearance (mils)
D = Journal diameter (in)
L = Tilting pad axial length (in)
L_u = Bearing unit load, (psi)
W_r = Bearing resultant load, gravity + gear (lb)

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