

EXPERIENCE IN THE USE OF FLEXURE PIVOT TILT PAD BEARINGS IN BOILER FEEDWATER PUMPS

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ABSTRACT

Boiler feedwater (BFW) pumps are among the most critical rotating equipment in power plants. The bearings in these pumps must be able to attenuate rotor vibrations and other hydraulic forces while still maintaining a high degree of reliability. This paper discusses the problems experienced with the original fixed geometry bearings, and the progression in bearing upgrades to spherical pivot tilt pad bearings, followed by the more recent

application of flexure pivot (FP) bearings. The first set of FP journal bearings was installed in 1994 during a major turnaround. The conventional thrust bearing was also modified to an FP style bearing to help eliminate the brinelling and wear on the pivots and the leveling links. While FP bearings have had success in relatively small high speed integrally geared compressors, this was their first application in a critical utility machine. The analysis and design considerations, along with the vibration data obtained in the field before and after the final conversion to FP bearings, will be presented and discussed.

INTRODUCTION

In the design of high energy boiler feedwater pump bearings, special consideration must be placed on the stability and unbalance response analyses. This is because the bearing load varies in both magnitude and orientation, due to the presence of hydraulic side forces. Furthermore, wear rings in high energy pumps may act like sleeve bearings and load or unload the fluid-film bearings by changing the magnitude and orientation of the net load. This condition often causes stability problems for pumps using fixed geometry sleeve type bearings. In order to overcome the limited stability margin of fixed geometry bearings, tilting pad bearings are often employed. The inherent design of the tilt pad bearing eliminates subsynchronous vibrations. However, some of its shortcomings introduce other problems. These shortcomings have been discussed in detail by Zeidan (1992) and will not be dwelled upon in this paper except where applicable.

The thrust end cross-section of the BFW pump is shown in Figure 1. The pump is rated at 28,274 hp with a design flow of 8780 gpm at 4850 psi discharge pressure. The pump operates at a speed of 5430 rpm. The shaft carries six impellers (stages) and the total rotating assembly weighs 2123 pounds.

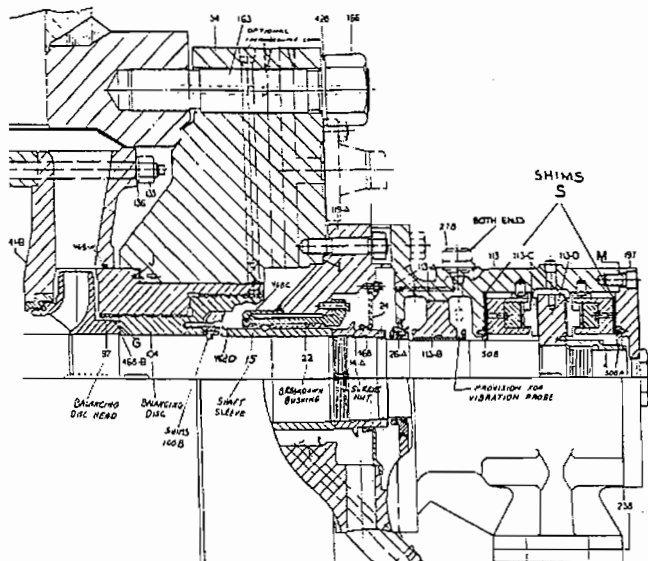


Figure 1. Cross-Section of the Pump Thrust Bearing End.

The pump was originally supplied with sleeve type bearings. These bearings had a very short life due to stability induced vibrations. The vibration signatures obtained using proximity probes and accelerometers indicated high vibration levels. Frequency spectrum analysis of the vibration signal revealed a significant subsynchronous component. Further investigation showed that the vibration was concentrated at a frequency of 66 Hz, which is approximately 45 percent of the operating speed. Such vibrations are characteristic of the oil whirl phenomenon. The oil whirl indicates the presence of a hydrodynamic instability caused by the crosscoupling inherent in the sleeve type bearings

and impeller wear rings. The frequent problems experienced with the pump finally led to a decision for modifying the bearings. A study was initiated for examining the feasibility of retrofitting the pumps with tilt pad bearings as a measure of suppressing the subsynchronous vibration.

CONVERSION TO SPHERICAL PIVOT TILT PAD JOURNAL BEARINGS

The study, which examined the limited stability characteristics of the bearings and reviewed the vibration data and operating history, resulted in an engineering recommendation for a tilt pad bearing retrofit. The plant personnel concurred with this recommendation and proceeded with implementing the proposed conversion. The tilt pad bearings offered greater stability, which was essential to solving the current subsynchronous vibration problem. Other advantages were also cited for the conversion that included embedded thermocouples for more accurate bearing health monitoring. The original sleeve bearings had a thin babbitted liner that could not be fitted with embedded thermocouples. The new tilt pad bearings are direct drop-in replacements, requiring no modifications to the bearing housing.

Of the many different styles of tilt pad bearings, the spherical pivot tilt pad bearings, also commonly referred to as ball-in-socket bearings, were the preferred choice for this application. These bearings have many advantages over the conventional style line and point contact tilt pad bearings. The spherical pivot design, shown in Figure 2, has significantly lower contact stresses in the pivot, compared with tilt pad designs that utilize a point or line pivot. This design is made by generating a socket in the back of the pads that mate with slabbed-off balls, which are bolted to the shell or retainer. Conversion to a spherical pivot reduces contact stresses while allowing tilting motion in both the circumferential and axial directions.

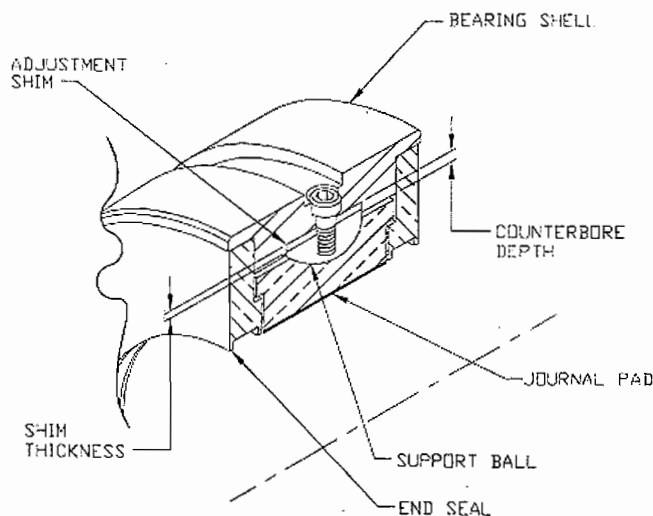


Figure 2. Cross-Section of the Ball-in-Socket Bearing Assembly.

In spherical pivot tilt pad bearings, the balls are assembled in the housing and the set bore is adjusted by grinding shims located between the ball and the housing. These shims also provide compensation for the slight difference in depth of each of the counter bores in the shell due to manufacturing tolerances. Thus, the shims may have varying thickness and, therefore, cannot be interchanged. Any mix up of the parts and their location during inspection or installation could result in a bearing with a noncircular bore, causing severe wiping of the bearing pads or bearing end seals. The plant experienced some of the symptoms pertaining to this type of mishap during a turnaround. Inadvertently switching the location of the balls and shims may cause an elliptical

bearing bore and can, in extreme cases, cause the shaft to wipe the bearing end seals. The consequences of interchanging the location of the ball pivots and shims can be severe. However, the plant personnel were aware of that and took the necessary precautions. Nevertheless, the bearing complex design makes them less robust and unable to handle BFP upset conditions.

Another problem associated with all conventional tilt pad bearings is pivot wear. Although the pivot contact stresses with the "ball-in-socket" design are lower than those with point and line contact tilt pad bearings, there is still relative motion between the ball and the socket. Shaft synchronous whirling and other vibrations caused by hydraulic forces in the pump will force the pads to oscillate, following the dynamic motion of the shaft. This will eventually cause wear of the ball pivots and the mating sockets and over time lead to an increase in bearing clearance, resulting in larger vibration levels and bearing dynamic loads. This will in turn result in further pivot wear and the cycle repeats. This propagation of pivot wear results in continuous performance degradation and steady increase in synchronous vibration levels. Figure 3 shows typical pivot wear on the balls and sockets of the two loaded pads from one of the BFW pump tilt pad bearings. These bearings had been in service for approximately two years. The severe fretting resulted in an increase of 0.002 to 0.003 inches in the bearing clearance. In order to avoid an unplanned shutdown due to excessive synchronous vibrations, the plant was forced to replace the bearings at every two year maintenance cycle.

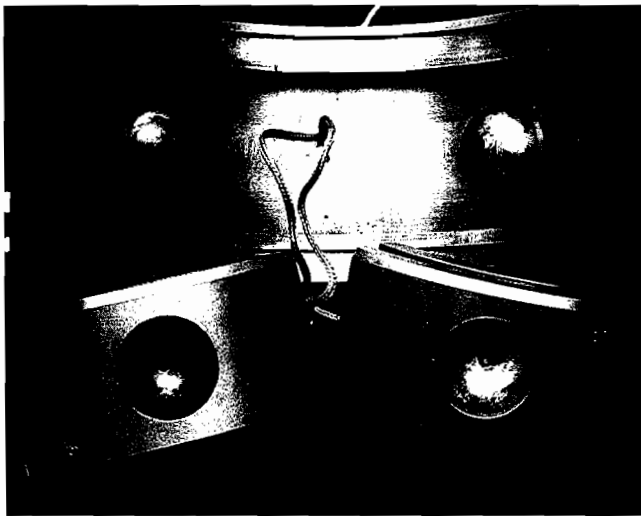


Figure 3. Pivot Wear on the Balls and the Mating Socket.

The pump benefited from the improved stability characteristics of the tilt pad bearings compared with the original sleeve bearings. Unfortunately, many of the limitations and shortcomings of conventional tilt pad bearings have become apparent. These limitations are summarized as follows:

- The number of parts and complexity of tilt pad bearings is much greater than the simpler fixed geometry bearings. Maintenance, inspection, and assembly are more difficult.
- The design and manufacture of multiple parts results in components that are not interchangeable. This often leads to problems during overhaul and inspection of the bearing components. Maintenance personnel are required to have special training to handle such complex assemblies.
- Multiple piece design requires longer installation time and is more difficult to verify critical bearing parameters.
- Dynamic loads cause relative motion between the pad and the pivot resulting in pivot wear. This reduces the bearing life and results in increased vibration levels over time.

CONVERSION TO FLEXURE PIVOT JOURNAL BEARINGS

In light of all the problems with the spherical pivot bearings and additional expenses to rectify them, the plant ultimately decided to pursue alternative bearing designs. Plant engineering decided to pursue a relatively new design, the flexure pivot (FP) tilt pad bearing, which addressed many of the problems inherent with the existing tilt pad bearings.

FP bearings are very similar in concept to conventional style tilt pad bearings except for the tilting mechanism. Instead of a two-piece mechanical pivot with relative movement, as is the case with conventional tilt pad bearings (Figure 4), FP bearings utilize a one-piece pivot design in which the pad is supported on a flexible beam or "web." The web is carefully designed to allow the pad to flex (tilt), as shown in Figure 5, and still provide adequate radial support to carry the load imposed by the shaft and the pump hydraulic forces. One of the major advantages of the flexing web element is the absence of any relative motion between parts. Thus, there is no degradation in service due to pivot wear. Furthermore, the design and new manufacturing methods utilized with the FP bearing provide additional benefits that will be discussed in the following sections.

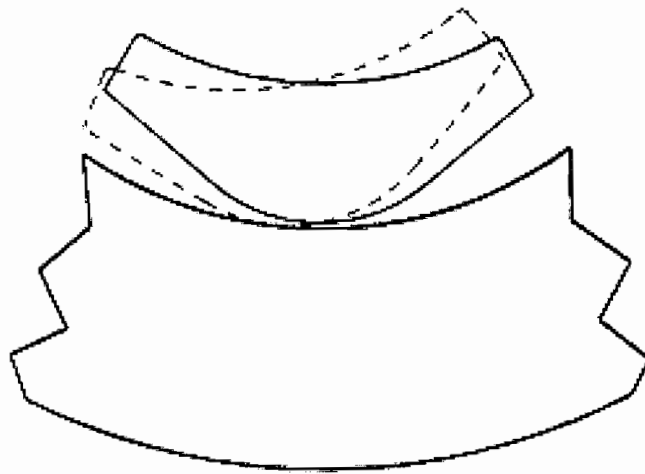


Figure 4. Schematic Showing Two-Piece Pivot Design of Conventional Tilt Pad Bearings.

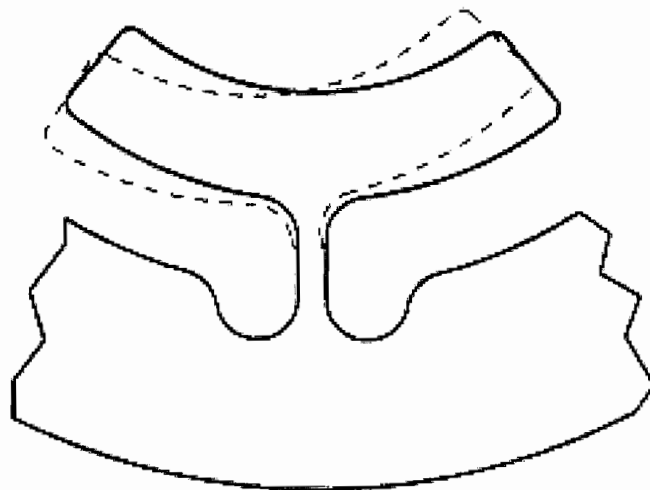


Figure 5. Schematic Showing One-Piece Pivot Design of FP Bearings.

Bearing Design and Analysis

As previously stated, the design of the flexible web is critical to proper bearing operation. The web is designed to provide an optimum value of rotational stiffness dictated by the stability requirements of the application. The rotational stiffness of the pivot determines the amount of crosscoupled stiffness coefficients (K_{xy} and K_{yx}) in the fluid film. Vance (1988), has shown that a positive K_{xy} and a negative K_{yx} will add vectorially to produce a destabilizing force tangential to the whirl orbit of the rotor as shown in Figure 6. This destabilizing force in a bearing can only be controlled by either adding damping or by reducing the crosscoupled stiffness coefficients to make its effect negligible. Therefore, bearing stability in a FP bearing is directly proportional to the difference in the crosscoupled stiffness coefficients ($K_{xy} - K_{yx}$) as shown by Zeidan (1992). Figure 7 shows a plot of ($K_{xy} - K_{yx}$) as a function of the web (pivot) rotational stiffness for this FP bearing. For high values of the rotational stiffness (right side of the curve), the bearing has high crosscoupling and behaves like a fixed geometry sleeve type bearing. At low rotational stiffness values (left side of the curve), the bearing has low crosscoupling and performs like a conventional tilt pad bearing.

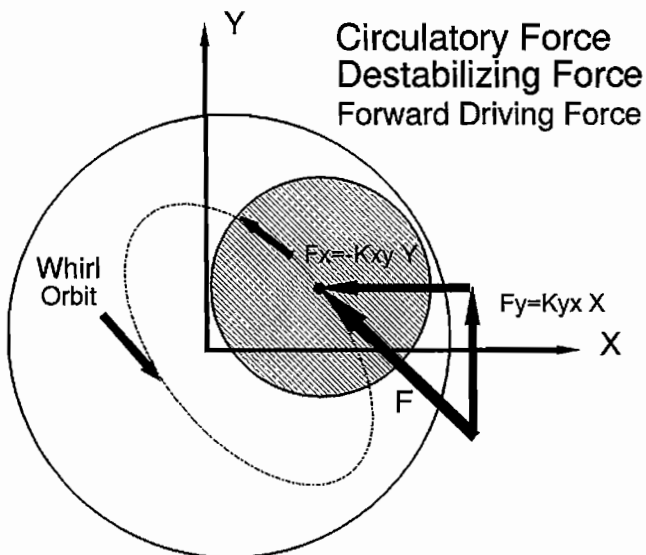


Figure 6. Schematic of the Destabilizing Cross-Coupled Force.

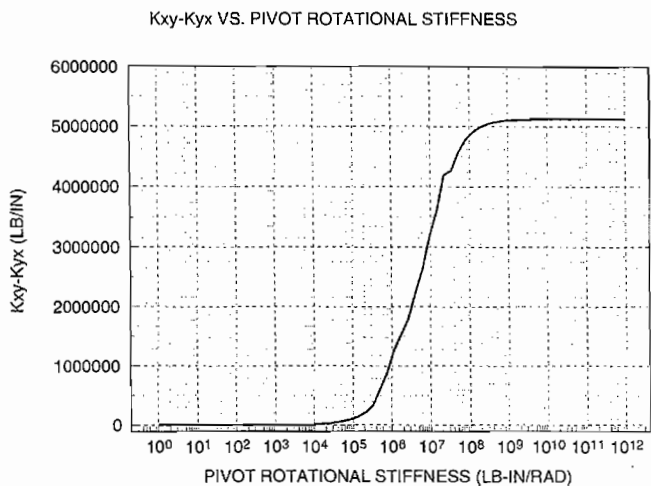


Figure 7. Variation of the Cross-Coupled Coefficients with Pad Rotational Stiffness.

In the design process of the web to achieve the desired rotational stiffness, a comprehensive stress analysis is also performed. This is accomplished by modelling a single pad using a state-of-the-art finite element program. All of the critical pad and web geometry have been categorized to allow for parametric analyses as shown by Jain (1992). An optimization routine is then executed to calculate the optimum web geometry resulting in the lowest stress for a desired pivot rotational stiffness. The finite element model of the FP bearing is shown in Figure 8. As compared with conventional tilt pad bearings, especially point and line contact pivots, the FP bearing has a much larger pad support area resulting in lower stresses. In addition to using static pad loads, which are predicted using bearing analysis programs, the dynamic bearing loads are also incorporated in the stress analysis. The Modified Goodman diagram, shown in Figure 9, can then be constructed for the web stresses to determine the factor of safety for infinite life. The predicted factor of safety for this bearing is 6.74, providing an ample safety margin and increased reliability.

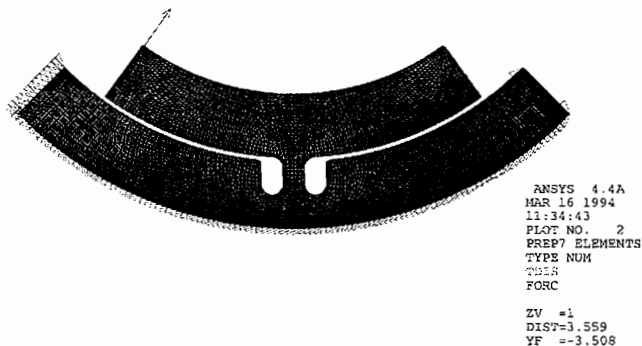


Figure 8. Finite Element Model of FP Journal Bearing.

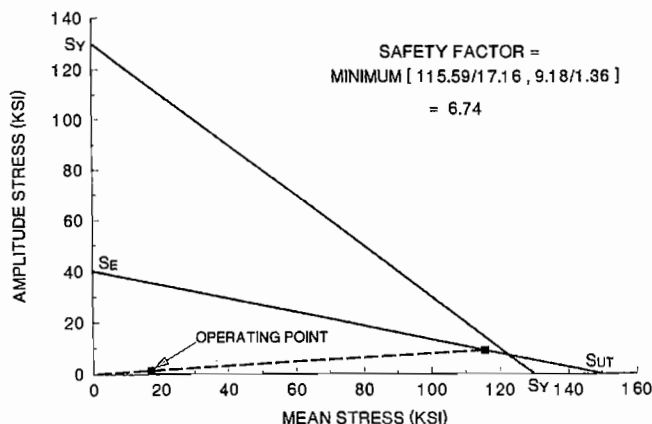


Figure 9. Modified Goodman Diagram.

The FP pad webs are also designed to allow motion in the axial direction as shown in Figure 10. This feature accommodates potential misalignment due to manufacturing tolerances, shaft bow, or thermal distortion.

Bearing Manufacture and Installation

Wire electric discharge machining (EDM) is employed in the manufacture of the one piece FP tilt pad bearings. Wire EDM machines, shown in Figure 11, have an accuracy of 0.000004 in. This high precision, which is only available in some of the latest EDM technology, is used to machine the bearing pad bore and set bore in one operation. This manufacturing method and the one piece construction allows the bearing set bore to be held to a total assembled tolerance of 0.0005 in or less. This is a significant improvement when compared with the assembled tolerance of

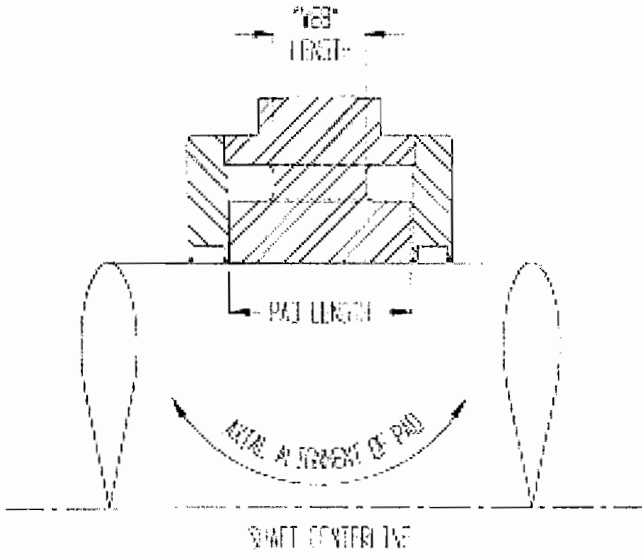


Figure 10. Cross-Section Showing Machined Scallops on Either Side of the Pivot to Provide Some Rotation Flexibility in the Axial Direction.

0.002 in obtained in a typical conventional tilt pad bearing. The differences in the tolerance stackup between a FP bearing and conventional tilt pad bearing are shown in Figure 12.

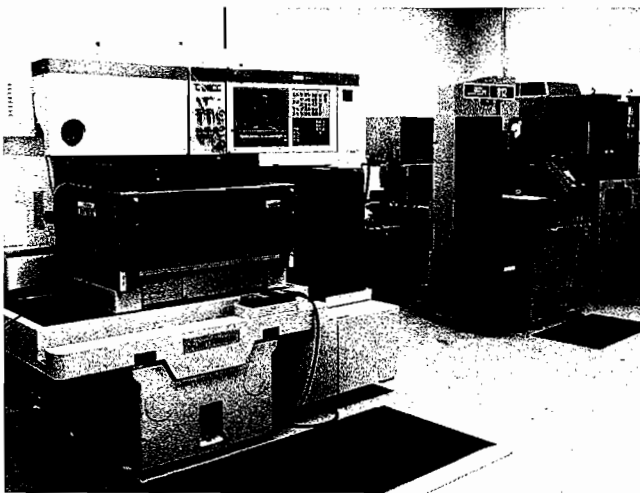


Figure 11. Wire EDM Machines Utilized in the Manufacture of FP Journal Bearings.

The one piece construction, shown in Figure 13, helps eliminate many of the shortcomings experienced with conventional tilt pad bearings. The elimination of multiple components precludes the possibility of inadvertently mixing up parts. In addition, the one-piece design simplifies handling and reduces the installation and inspection time.

Another advantage of the FP bearings became apparent after a recent rub and damage of the wear rings on two pumps still equipped with the ball-in-socket tilt pad bearings. In these pumps, the shaft sinks between the pads, since the tilt pad bearing is oriented with a load between pads configuration. The stackup of tolerances, in addition to this shaft sink between pads, coupled with operation at low speed (thin oil film), resulted in the shaft rubbing the wear rings and severely damaging the two pumps. The FP pads, unlike the pads in a tilt pad bearing, have a rotational stiffness that reduces the amount of shaft sink between pads. This, in addition to better control of concentricity inherent in the

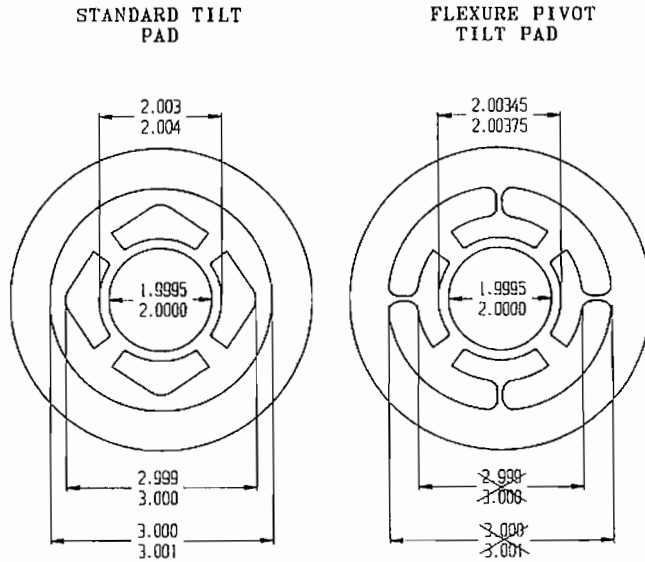


Figure 12. Comparison of Manufacturing Tolerances for Conventional and FP Bearings.

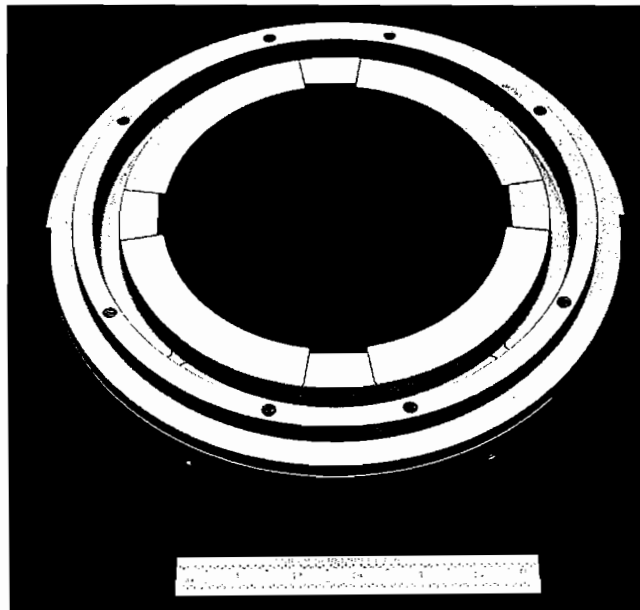


Figure 13. Photograph of FP Bearings Utilized in this Application.

manufacture of these bearings, has prevented the occurrence of any rub problems with the FB bearings.

FIELD VIBRATION MEASUREMENTS

The main goal of the FP bearing conversion was to eliminate pivot wear in the existing tilt pad bearings while maintaining the improved stability characteristics present with the tilt pad bearings. In addition to achieving these goals, the synchronous vibrations were also significantly reduced in comparison with the vibration levels measured with the "ball-in-socket" bearings. The first FP bearings were fitted in November 1994 and have been running since that time. Figure 14 shows the synchronous whirl orbit with the FP bearing. These data were taken after seven months in operation and the orbit size has not changed since installation. Conversely, Figure 15 shows the synchronous whirl orbit of the "ball-in-socket" bearings after 19 months of operation. The vibration amplitudes have continued to increase to the current level of approximately 0.005 in, peak-to-peak.

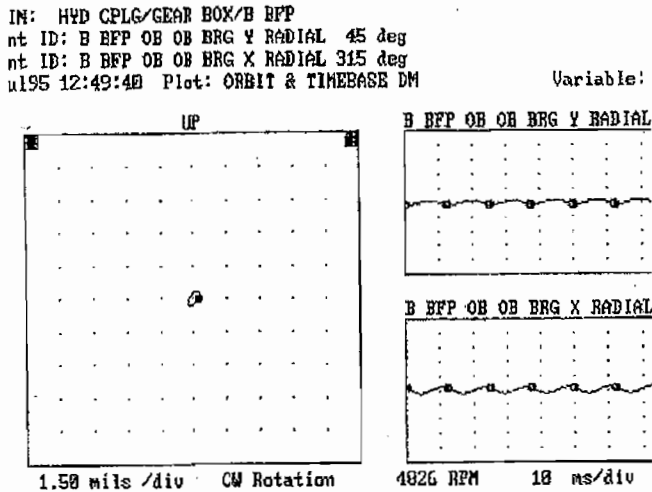


Figure 14. Vibration Measurements with FP Bearings.

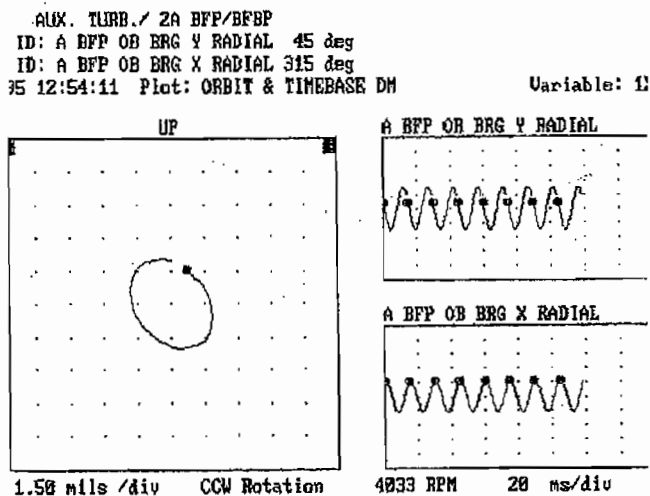


Figure 15. Vibration Measurements with the Conventional Tilt Pad Bearings.

CONVERSION TO FLEXURE PIVOT THRUST BEARINGS

The encouraging results with the FP journal bearings prompted the investigation into using FP bearings to replace the active thrust bearings. These bearings are of the self-equalizing ball-in-socket style. The leveling links, shown in Figure 16, exhibited signs of wear and brinelling. The sockets, shown in Figure 17, also showed significant wear. The standard style FP thrust bearings do not have leveling links to provide additional misalignment capability as is the case with the Kingsbury style bearings. A FP thrust bearing used on the Dresser Rand DR990 gas turbine (Jain, et al., 1994) implemented a spherical fit on the back face of the FP thrust retainer. This sphere mated with a female socket in the housing, thus providing ample alignment capability. In the gas turbine design, the retainer is a nonsplit one piece assembly. This design was complicated with the fact that the FP retainer and the mating female socket had to be horizontally split per API for ease of installation.

Bearing Design and Analysis

The existing thrust bearings had babbitted copper backed pads with steel ball pivots. One of the pads was equipped with a load cell for direct load measurement and monitoring. In designing the FP thrust pads, a finite element code was utilized to model the pad

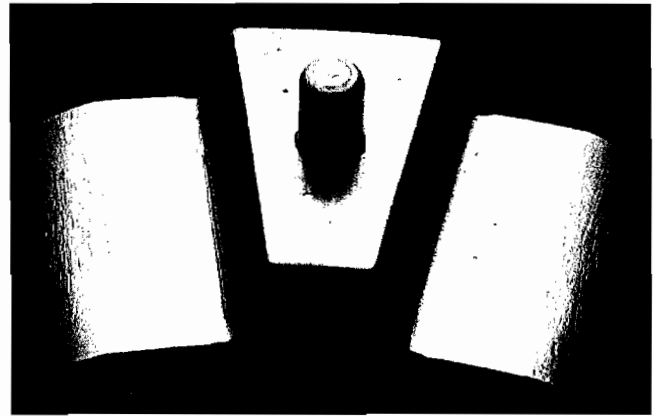


Figure 16. Brinelling on the Thrust Bearing Leveling Links.

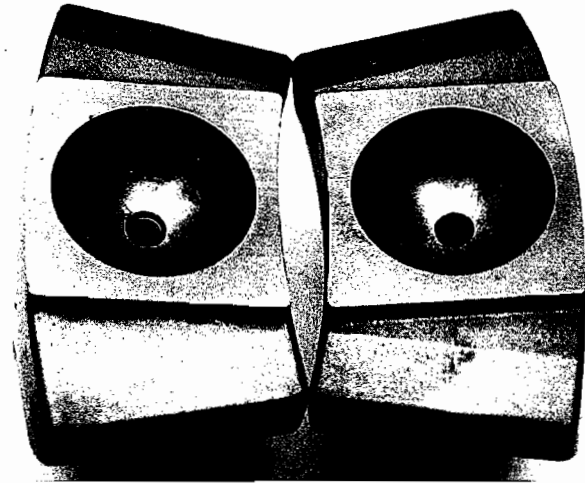


Figure 17. Pivot Wear on the Socket of the Upper Leveling Link.

structure. The pad structure consisted of a steel post, to provide the desired flexure necessary for the pad tilt, and a copper pad that was mechanically fastened to the steel post. Figure 18 is a cross-section of the finite element pad structure showing the two different materials making up the thrust pad. This bearing, like the existing spherical pivot bearing, has an offset pivot design to allow greater load capacity and lower operating temperatures. The copper pads are capable of dissipating approximately three times the heat dissipated by the steel pads, thus providing a greater safety margin.

The fluid film pressure contours on the pad thrust surface at maximum operating conditions are shown in Figure 19. Figure 20 is a finite element plot showing an exaggerated deflection (tilt) of the pad and the maximum stresses on the flexible post supporting the pad.

The FP thrust bearing assembly is presented in Figure 21. As discussed above, one of the pads was equipped with a load cell for direct load measurement and monitoring. The load cell used is 0.35 in thick and one inch in diameter. The load cell was assembled in the pad and tested to verify its linearity and proper operation before installation. Figure 22 shows a detailed cross-section of the thrust pad containing the load cell. The FB thrust bearing was installed in October of 1996 and has been running satisfactorily since startup.

CONCLUSIONS

The FP bearings have proved to be a "maintenance friendly" alternative to solving rotor stability problems in boiler feedwater pumps without the complications and risk associated with conventional style tilt pad bearings. These bearings have the

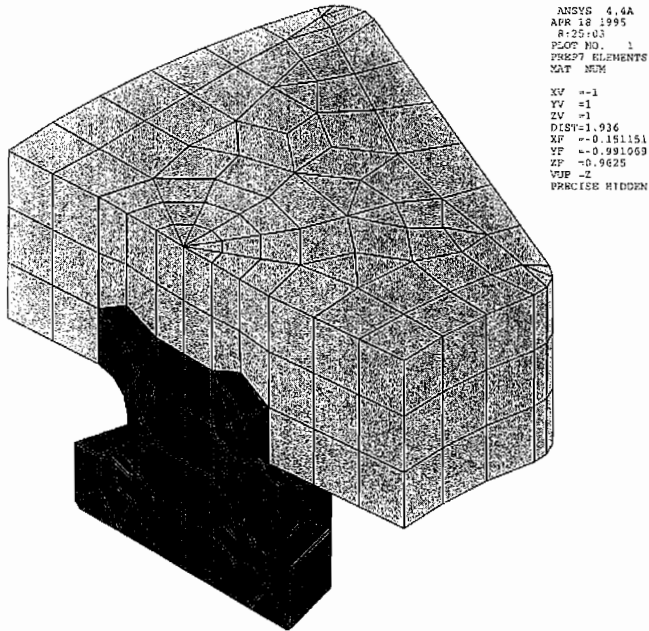


Figure 18. ANSYS Cross-Section of GP Thrust Pad.

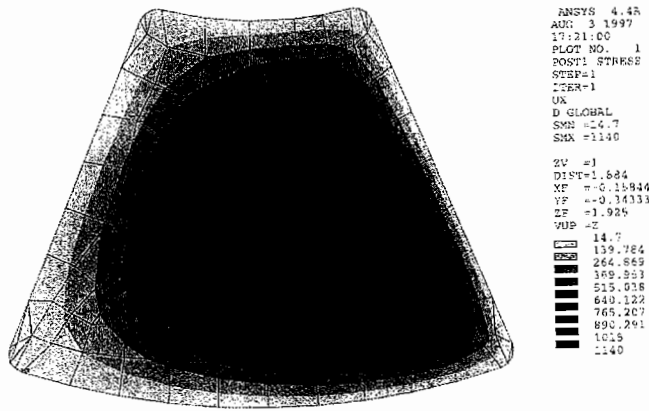


Figure 19. Oil Film Pressure Contours on the Pad Surface.

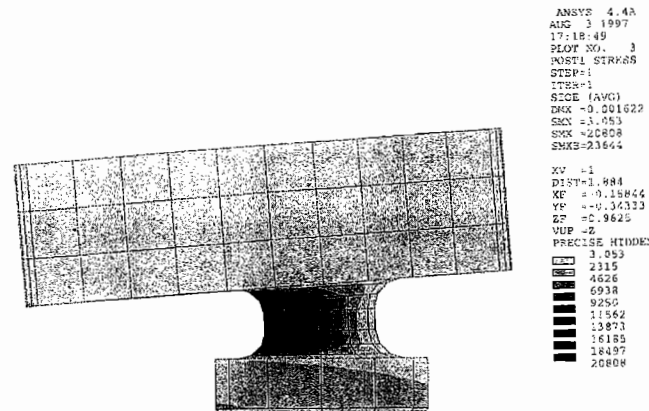


Figure 20. Exaggerated Deflection of the FP Pad and Stress Contours.

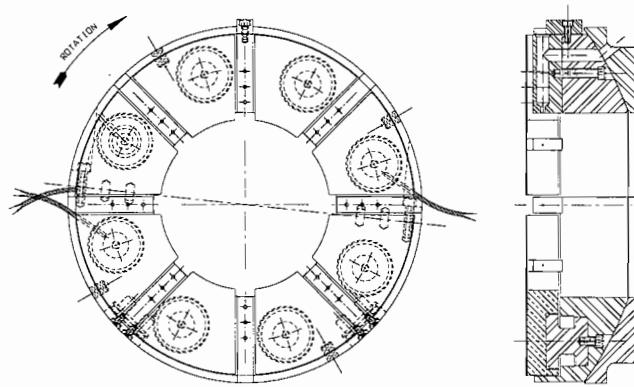


Figure 21. Thrust Bearing Assembly for FP Bearings.

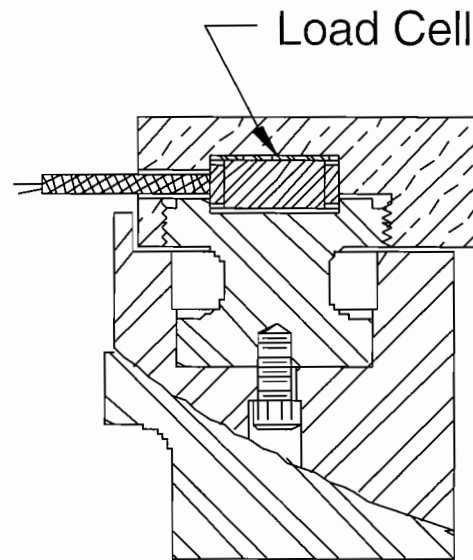


Figure 22. Cross-Section of the Pad Showing the Load Cell Configuration.

simplicity of sleeve bearings coupled with the stability characteristics of tilting pad bearings. At the time of this writing, the subject bearings have been in operation for approximately three years with excellent results. It is anticipated that the bearing will continue to operate for two to three maintenance cycles (four to six years) before needing repair or rebabbiting. The more recent FP thrust bearings have been in successful operation for about one year.

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