RADIAL AND THRUST BEARING PRACTICES WITH CASE HISTORIES

by

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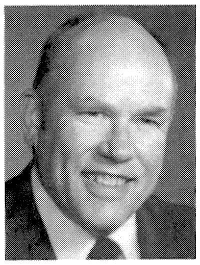

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ABSTRACT

Case histories for 13 selected bearing design conversions for specific end results are presented. They were selected out of a group of bearing designs over a 25 year span to give a varied array of non-repetitive case histories. In addition, some ideas for proper measurements are also included.

INTRODUCTION

Bearings, their proper application and fitting, rate as the most important factor in the success or failure of a turbomachine's operation. The cost of successful bearings does vary widely in the marketplace. Bearing stiffness and damping can strongly affect the rotor's resonance, or may have no effect whatever, as the rotor system may be totally dependent on the rotor shaft's flexibility. Improper fitting and confirmation of radial bearing journal-to-babbitt clearance ranks as the highest single contributor to troubleshooting operating units (Figure 1).

Setting thrust bearing rotor float on centrifugal compressors and specifically, steam turbines, will continue to remain a simple, yet little understood procedure. This confusion is usually derived from extremely poor instructions, written in the vendor's instruction manuals. Such instructions never outline the goal beforehand and generally are not understood. This same misunderstanding perpetuates into setting the nozzle-to-blading standoff(s). At best, the manuals explain what is needed, but rarely why it is necessary.

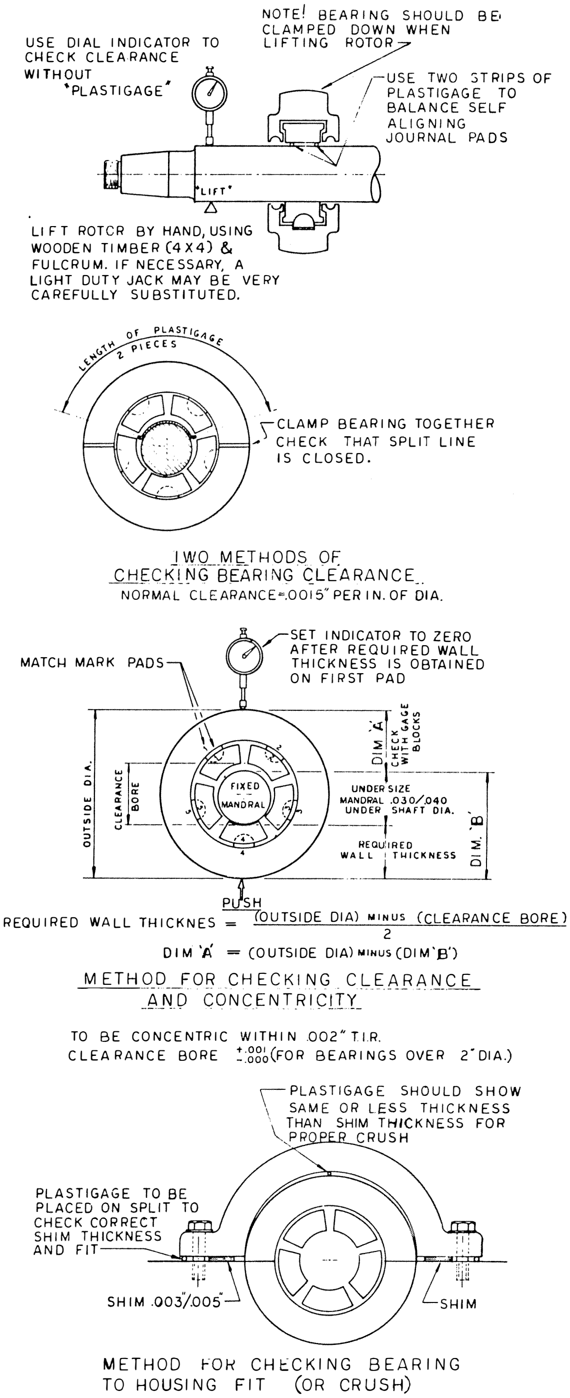


Figure 1. Checking Methods for Self-Aligning Tilting Pad Journal Bearings.

To illustrate using a steam turbine, these instructions could be improved with the addition of some basic steps (the numbers are only typical):

1. The rotor's position is very important and is referenced off the first nozzle ring. The blade standoff is controlled by a measurement from the blade shroud to the nozzle. When the rotor is in an active (downstream) position, the gap is 0.045 in. When the rotor is in a counter (inactive) (upstream) position, the gap is 0.033 in to 0.035 in.

2. The active (normal) position is controlled by Shim A. The inactive (counter) position of the rotor is controlled by Shim B. With both shims removed, the *total* rotor float can be verified from the same nozzle reference. This should be 0.110 to 0.130 in.

3. With Shim A in place and Shim B removed, the rotor is moved against the nozzle and two dial indicators are "zeroed". The rotor is then thrusted in the active (downstream) direction toward the exhaust and the gap measurement is taken. If the gap is short (for example, 0.030 in), then Shim A is reduced in thickness by 0.015 in, by grinding, to provide the 0.045 in measurement.

4. The top half on the bearing assembly and housing *must* be installed and the down float gap must be rechecked. Any discrepancy should be corrected, e.g., dowels bent, upper and lower faces contacting Shim A must be flush, and each half (upper and lower) of Shim A should be the same thickness. (It is strongly preferred that Shim A be a single thickness of steel—even if a *new* shim must be machined for thicker dimensions.)

5. With the active position set at 0.045 in, the inactive (counter) float must be set. This cold float is primarily to allow for thermal expansion between the thrust bearings.

6. With Shim B in position, the rotor is thrust towards the governor. The movement is recorded on dial indicators as before. If the movement is less than 0.010 in to 0.012 in, then Shim B is reduced in thickness as required. If the movement is greater than 0.012 in, Shim B must be increased in thickness as required.

7. Finally, the top half of the bearing assembly/housing must be reinstalled and a recheck made to confirm the clearances, as in Step 4.

Note: Whether the active or inactive thrust (standoff) is set first will generally depend on whether single or dual thrust discs are incorporated.

Instructions for setting the clearances for a compressor are very similiar, however, the total rotor float might be 0.250 in, and the starting reference may be a sleeve spacer B to set the inactive movement to one-half of the total float of 0.125 in, minus the rotor thrust bearing float of 0.015 in to 0.110 in. Shim A could be adjusted in thickness to give 0.015 in to 0.020 in float. It would be normal for a centrifugal compressor to thrust to the suction end. Further, it seems logical that a thrust bearing be located at the suction or "cool" end of the casing.

Thrust failures are so crippling to operation that "automatic" shutdowns are mandatory to protect rotors and casings, but not necessarily to save the bearings. Portions of the thrust bearings become "sacrificial lambs" against a total wreck. The limits for shutdown must be great enough to ride out acceptable upsets, such as the loss of superheat (wet steam), change of the molecular (mol) weight of the process, momentary surge, etc., yet short enough to not have rotor contact and still leave some babbitt material. These shutdown limits will always show damage on inspection and impress the machine operations people. Two charts (Figures 2 and 3) taken from American Petroleum Institute (API) 670, Revision 2, are shown for better understanding. Alarms at 15 mils (0.38 mm) and shutdowns at 25 mils (0.63 mm) have proven to be adequate since 1964.

Some case history and practices will be discussed to illustrate some principles.

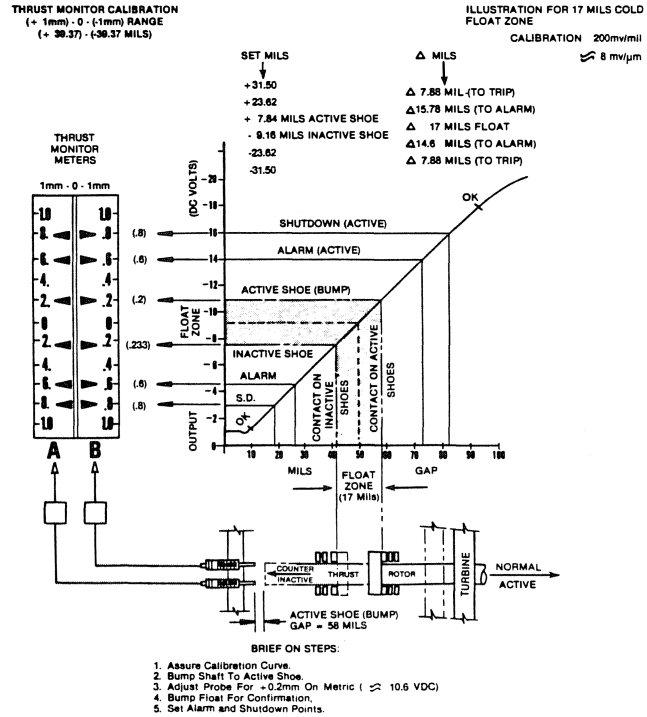


Figure 2. Typical Calibration Curve for Axial Position-Steam Turbine-Metric Based (from API 670 Standard, 2nd Revision, 1985, Vibration, Axial Position and Bearing Temperature Monitoring System).

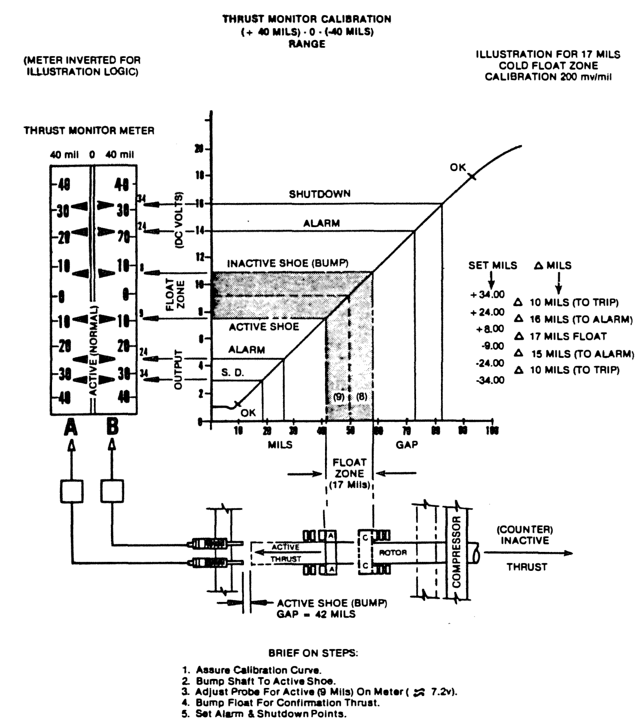


Figure 3. Typical Calibration Curve for Axial Position-Compressor-English Based (Ref: same as Figure 2).

CASE HISTORY ONE (THRUST)

Two 7600 hp steam turbine trains driving two centrifugal air compressor casings, each at 6000 cpm, had failed 27 thrust bearings over a period of years all with operational runs of less than one year. Air gauges, and later eddy current thrust limit probes, were installed to prevent further major damage, but production outages were totally unacceptable.

The initial thrust load equaled 450 psi babbitt pressure, with new condition labyrinths, rotor clearances and axial spill strip tolerances. The effective thrust bearing babbitt area was 23 in^2. Babbitt metal temperature was approximately 240°F. The six-pad thrust bearing required self-levelling by a spherical seat, which basically can only self-level at warm-up speed/load conditions (Figure 4).

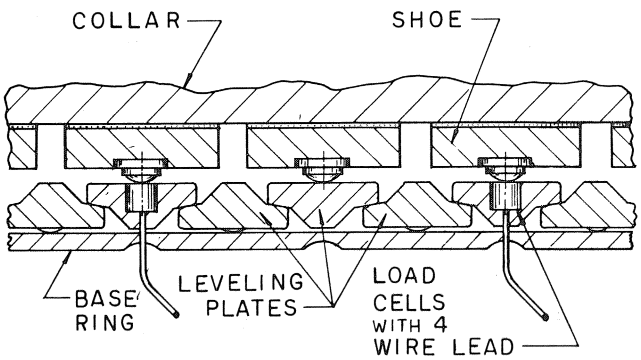


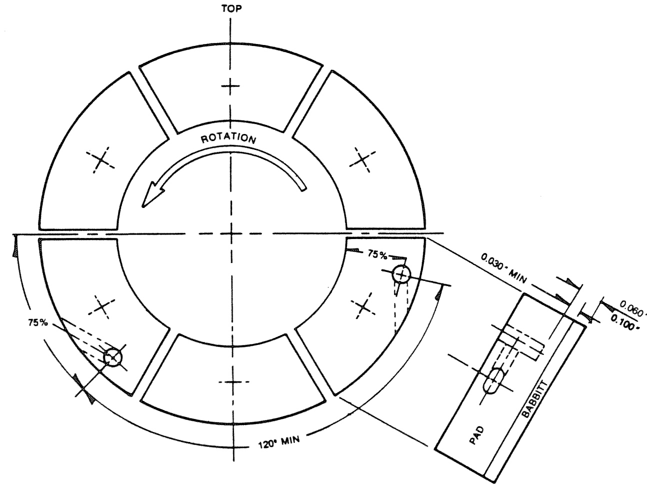
Figure 4. Thrust Bearing Tilting Pads on Self-Leveling Links with Load Cells Receiving Force from Pads.

Solution

The bearing chamber was machined in place utilizing a field erected boring bar with an air operated star feeder. A tilting pad, 8 in diameter, six-pad thrust bearing, with articulating level links, was installed with load cells on 50 percent of the pads and thermocouples on 50 percent of the pads in the 75 percent radius, 75 percent pad length position. While an area increase from 23 in^2 to 32 in^2 was affected, the ability to self-level and the tangential oil exit were felt to be the primary correction. Temperatures were lowered to 190°F. Load cell and temperature measurements confirmed one another and the temperatures were noted to be adequately sensitive (not over sensitive) to load changes. Four years of data were recorded on a direct current (DC) strip recorder. No further failure occurred on either machine for the six remaining years of this 18 year life process (Figure 5).

Note: Two other lessons were learned during this retrofit study. 1) It was quite simple to install a test rig of the existing bearings, using a 50 hp motor drive and two sets of original bearings mounted on a short rotor. One set of thrust bearings was modified to have hydraulic pistons (pins) to push the pads in order to load the test bearing in thrust. 2) Improper drilling through the steel backing pads by the manufacturer to give air relief prior to peening and babbitt puddling led to "dimpling" in the babbitt contact face. Dimples (depressions) cause turbulence, increase temperature and eventually striate grooves back to the tracking edge from the dimples. A pressure dam bearing uses this same pocket principle [1,2] for developing pressure in a corrective way to stabilize the rotor against oil whirl (Figures 6, 7 and 8).

Note 2: Thrust movement air gauges have an effective limit of about 25 mils (0.63 mm), even though 1 psi/mil sensitivity is obtained in both directions.



NOTES: 1) Temperature sensor shall be located 0.060" - 0.100" from the bearing running face and not less than 0.030" from the babbitt (white metal)/pad interface. Finish holes with a bottoming drill and break all corners.

2) Route the sensor lead from the bearing to outside the machine through a penetration fitting. Properly secure the sensor lead to prevent damage due to whip, chafing, windage, and oil with no internal connections. Sensor lead is not to refrain pivoting thrust shoes.

Figure 5. Typical Thrust Bearing Temperature Sensor Installation.

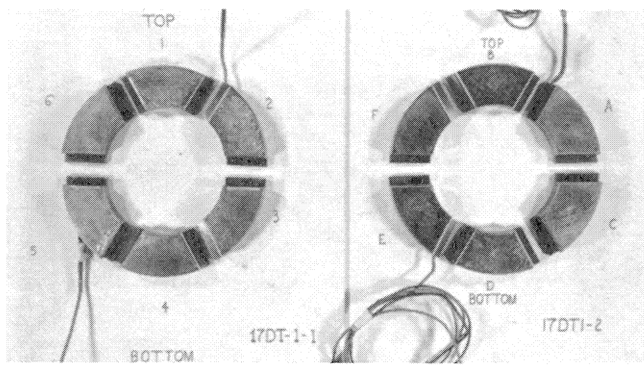


Figure 6. Dimpled Thrust Bearing Pads from Air Pilot Hole in Peening Support Pads Prior to Pretinning and Babbitting-7500 HP Steam Turbine, 1960.

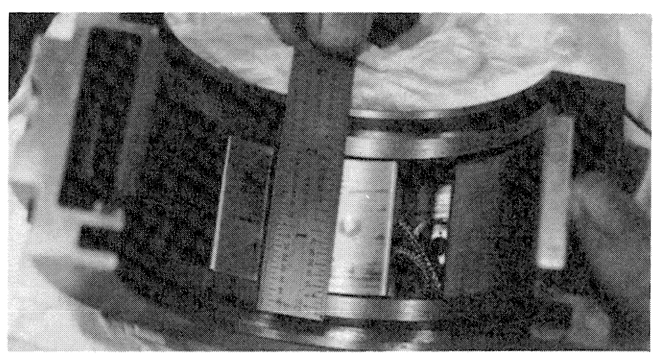


Figure 7. Dimpled Radial Tilting Pad Bearing from Centrifugal Compressor Bearing Retainer Assembly. Dimples due to air voids from pretinning and babbitt puddling over temperature sensor—1980.

CASE HISTORY TWO (RADIAL)

A steam turbine operating at 7000 hp and 10,5000 cpm was driving two centrifugal bodies and would go into oil whirl exceeding 5 mils at 43 percent of speed as the third steam inlet valve began to open. The bearings were pressure dam type.

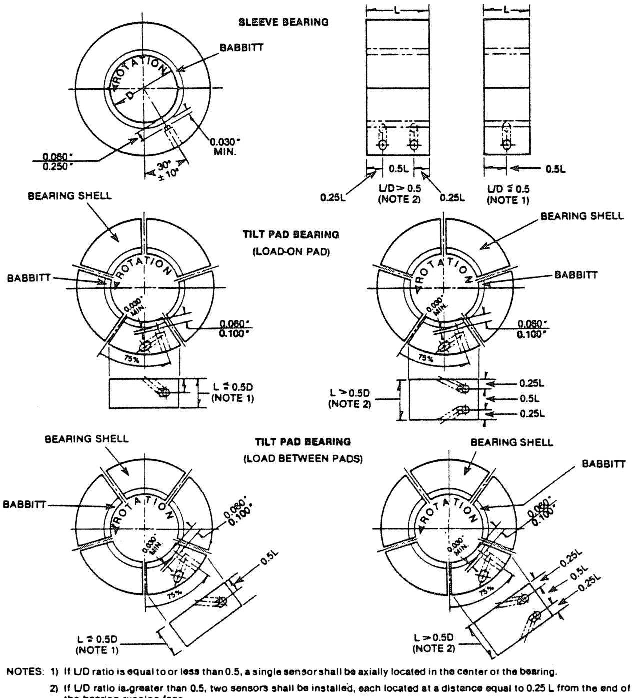


Figure 8. Typical Radial Bearing Temperature Sensor Installation (API 670, Rev. 2). Note 0.030 in minimum between babbitt interface and sensor. Also note placements of temperature sensors.

First, the steam valves were "gagged" open and the turbine was brought up to speed and loaded using the trip and throttle valve (TTV). No whirl was experienced, indicating that *partial admission* lifts to the rotor initiated the whirl (Figure 9).

Solution

The vendor had an existing design for a four lobe bearing which required two weeks to obtain. This bearing resolved the problem for a total of fourteen years without incident (to date).

Note: To allow operation during the two weeks hiatus, the pressure dam bearing was rotated to be placed on the major axis of the journal motion. This position was determined from the orbit of the shaft-to-bearing motion. The anti-rotation pin was removed and relocated to put the dam at the 9 o'clock position. The turbine was then loaded manually and the governor valves were put in service from an open position, thus allowing the turbine to operate in a limited whirl cycle of 2.5 mils to 3.0 mils at a 43 percent speed frequency (0.43X).

CASE HISTORY THREE (THRUST)

Two hot gas expanders driving axial and radial flow compressors, along with a startup helper turbine, were reduced in thrust bearing life from several years to 4000 hours, after converting to synthetic oil. The conversion had been made after a million dollar fire loss from a coupling failure, with subsequent oil spray onto the 1200°F turbine casing (Figure 10).

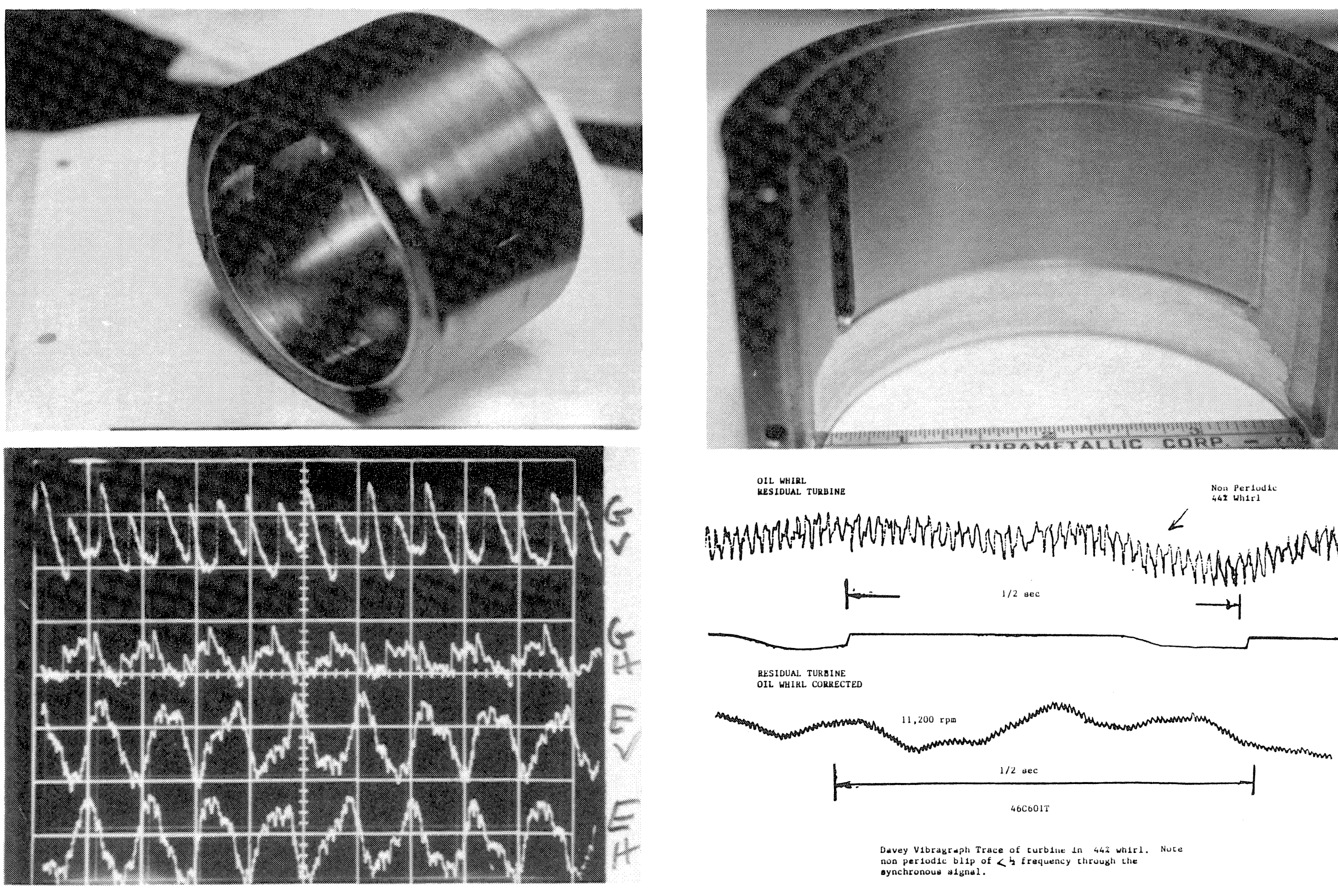


Figure 9. Pressure Dam Bearing Removed After Pure Oil Whirl at 43-44 Percent Speed. Contact of babbitt by journal has occurred, i.e., eccentricity ratio = 1.0. Davey Vibrograph tape from shaft "fish tail" also shown. Non-periodic (stationary) blip is 0.43X frequency.

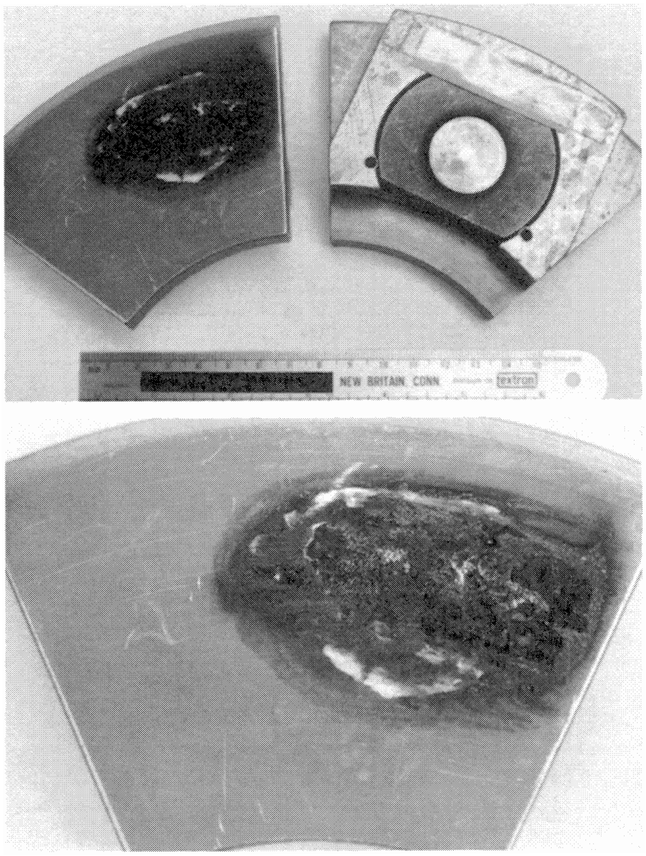


Figure 10. Thrust Pad from a Hot Gas Expander Turbine Suffering from Overheating (Baking) due to Synthetic Oil. This bearing had 4000 hours duty at removal.

Solution

After fire prevention isolation valves were installed, the oil was converted back to conventional turbine oil with no ill effects.

Note: A conversion to synthetic oil can be more easily done today than 15 years ago because of better synthetic oils. Consideration for additional cooling and oil flow may be necessary over hydrocarbon or mineral oils. In this specific case, the lubricity, specific gravity and viscosity had been matched well, but the synthetic oil had a lower specific heat, resulting in more heat for the same load, leading to a half-year service life.

CASE HISTORY FOUR (RADIAL)

Two incidents in one week occurred on two turbocompressor trains. The first incident had a compressor vibrating violently at around 1000 cpm, coming to a speed of 5600 cpm. The solution came after inspecting the bearing. (Preparatory to this startup, the compressor bearings had been inspected on a preventive maintenance external checklist.) The pressure dam bearings had been installed in reverse, i.e., journal and oil rotation did not enter the dam's feed involute and approach (converge on) the dam. The bearing position was corrected.

On a similar problem of the turbine driver of the same train, the lathe work on the babbitt to form the pressure dam was incomplete. Conventionally, the pressure dam is oriented to have a pad angle of about 125 degrees-140 degrees. Dam depths have been too deep to be effective for years (Figure 11). The oil entering this dam, which is perhaps 75 percent of the bearing width, may feed into an involuted slot and often enters at the horizontal feed slot. A relief groove may also be provided in the center of the lower half to remove babbitt area, thereby increasing bearing pressure. The machine work was somewhat sloppy and a rib of surface metal remained, which interrupted the oil flow causing the rotor to vibrate at full speed (Figure 11) [3].

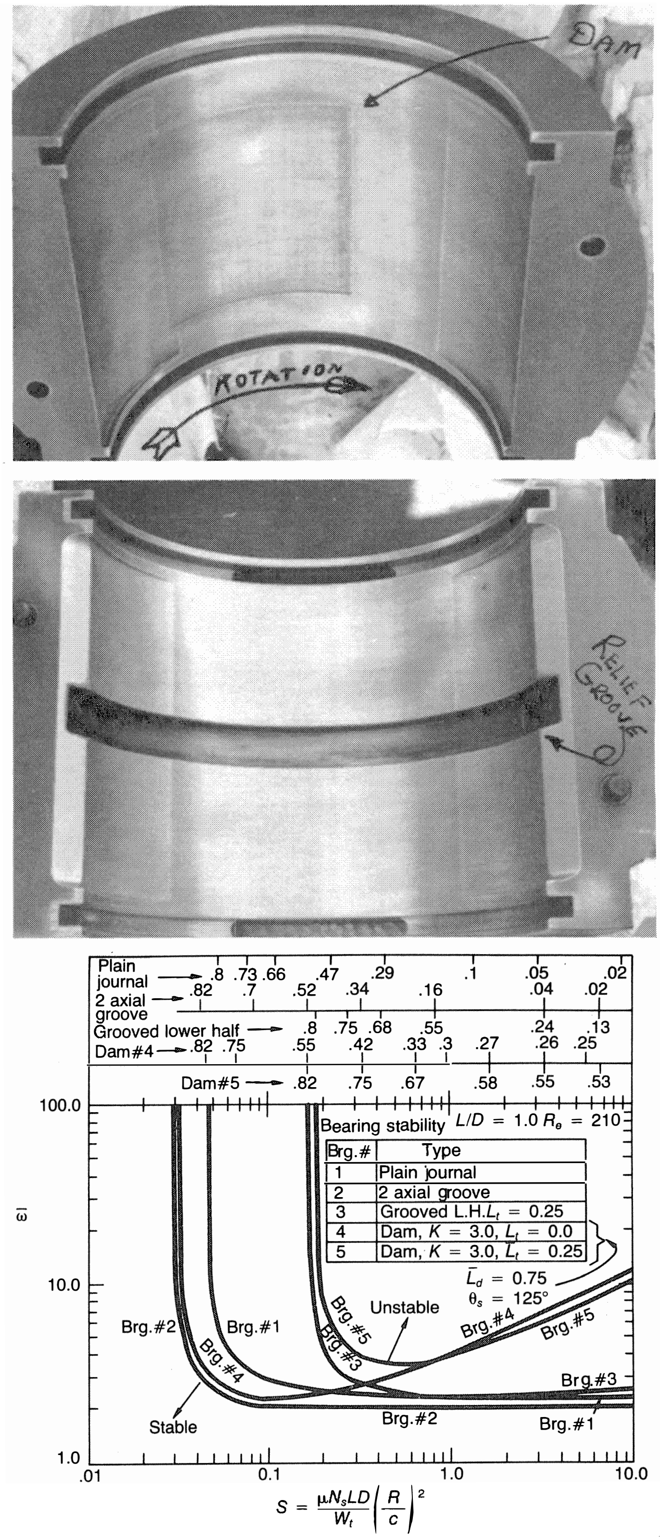


Figure 11. Pressure Dam Configuration with Relief Groove in Lower Half. Relative stability curve is also shown for comparison [3].

CASE HISTORY FIVE (THRUST)

A synthesis (syn) gas high pressure (HP) compressor for a 1000 tons per day (T/D) methanol plant operated for several months before loading the thrust bearing to a deflection of 22-25 mils, which activated the axial position shutdown monitor and shut the unit down. Typically, no radial vibration increase was noted, but there had been an increase in the static balance chamber pressure behind the balance piston of about 10 psi (350 psig to 360 psig). Also, the operator had noted the balance line had been increasing in temperature.

Solution

On disassembly, the balance piston labyrinth clearance was 45 mils versus a design clearance of 15 mils. The balance piston may have been installed with excessive clearance, possibly burned in before mechanical shop testing. This was confirmed on bundle reinstallation, as the lower tilting pad carrier did not lift the rotor off its labyrinths, requiring a relocation (5 mils lift, 4 mils left) (recentering) of the bearing housing in the casing head. No repeat of this problem occurred in the next 15 years (Figure 12).

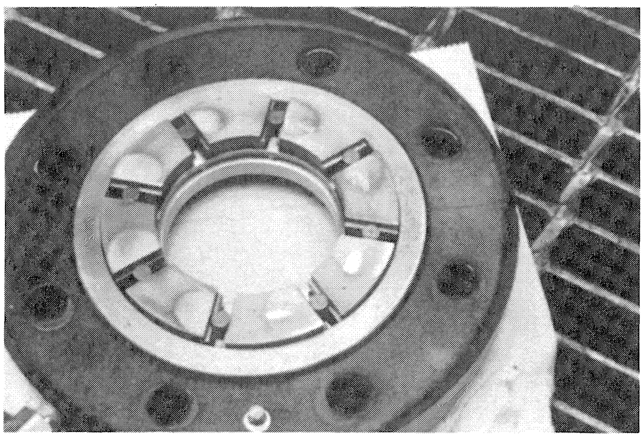


Figure 12. Thrust Pads on Overload with Rotor Deflection at 0.025 in. Taken from HP Barrel Compressor Operating at 10,500 rpm.

CASE HISTORY SIX (THRUST)

A similar series of thrust bearing failures occurred on a syn gas train in which the thrust bearings failed six to eight times, due to a "cavitation" type failure, as opposed to thrust load. Further, it always occurred during surge, due to a drop in molecular (mol) weight of the gas (loss of CO_2 feed in the gas). This rotor will oscillate in the axial direction during surge because it is a high pressure case, 350 psig to 700 psig, with five of eight stages active, due to a three stage blank (bridge over) (Figure 13).

Solution

After a discussion with a tribologist, an outside bearing company was directed to design a new active copper backed pad thrust assembly with *directed* lube provision along the *articulating* level links, rather than beam detectors (a poor leveler at best). This installation has proven to be successful for more than five years, while suffering surges and loss of mol weight severe enough to pop relief valves, yet without failure. The logic was to defeat cavitation, *negative* pressure, via spray nozzles; i.e., *positive* pressure at the pad entrance.

Note: This cavitation damage has a similar appearance to black scab or "wire wooling". Wire wooling was experienced in the 1970s on 15-5 ph thrust collars at 500 ft/sec surface speeds.

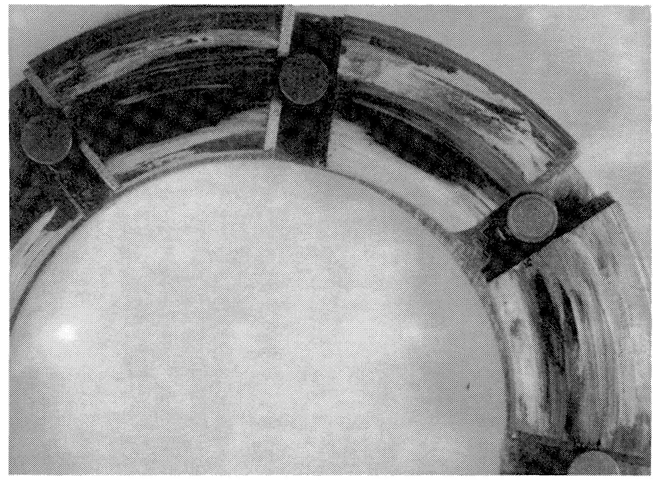


Figure 13. Similar Thrust Pads to Figure 12, but with "Black Scab" Appearing Failures Due to Cavitation. Corrected successfully with directed lube designs.

One is susceptible to wire wooling with chrome or manganese contents over one percent and surface speeds over 66 ft/sec (20 m/sec).

CASE HISTORY SEVEN (RADIAL)

A barrel compressor using five pads, load-on-pad (LOP) bearings, which operated at 11,000 cpm with relative vibration 3 mils peak-to-peak (P/P) (76 μm) first indicated a bad balance. This run-up followed assembly on a repair, following eight years of successful and smooth service. Case vibrations were typically very low (0.04 inches per second peak (ipsp)) (2 μm/sec) (Figure 14).

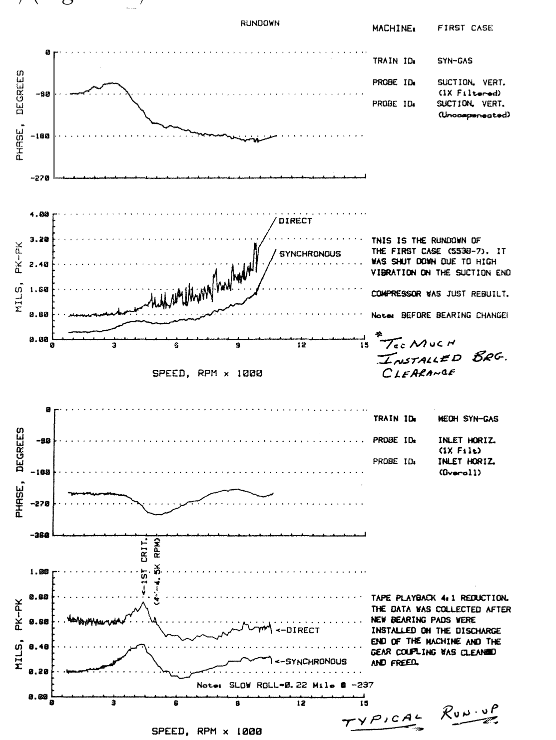


Figure 14. Bodé Plot of a Barrel Compressor Which Had Been Successfully Repaired and Balanced but Installed with Excessive Clearance (11 mils versus 7 mils) from Pad Retainer Brinnelling. The retainer was machined out, chrome plated and rebored to obtain the proper pivot circle (clearance).

Solution

Since a lift check had not been performed on reassembly (normally a common practice), the train was shut down for four hours for a lift check. *Eleven* mils of clearance was determined and *seven* mils was required. A spare retainer with pads was assembled, confirmed by plug gage check and installed. The vibration was low through the critical speeds, operating in the 1 mil P/P (25 μm) range, and less at the full operating speed.

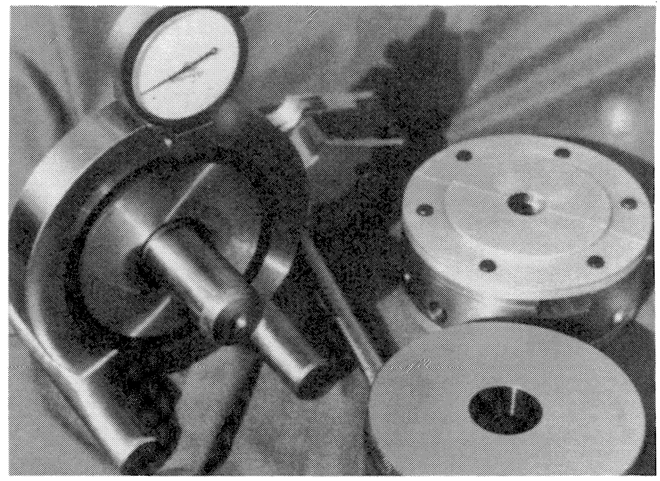
Note: A three-step gauge is shown (Figure 15) for tilting pad bearing checks in the retainer. Also, a mandrel checker is shown (Figures 16 and 17) which allows individual checking of the pads.

Since pads are generally made the same thickness (must be checked with a pin or knife micrometer), it is necessary to note brinelling indentions in the retainer bore. Indentions as much as five mils deep have been noted and corrected. The correction could entail machining out the indentions, chrome plating and remachining the proper bore, thereby reestablishing the proper pivot circle and clearance. Some designs like the one shown (in Figure 18) allow the spherical balls which receive the pad spherical seats to be readjusted. These can also be electronically gauged as shown in Figures 16 and 17.

CASE HISTORY EIGHT (RADIAL)

A 22 MW (30,000 hp) steam turbine driving a gear and alternator went into 43 percent pure oil whirl after 6 MW of load was applied. Whirl continued up through higher loads, 13 MW, 16 MW, etc. The bearings were plain axial groove bearings with 60° relief on each side at 7 in and 8 in, a length to diameter ratio (L/D) of 0.5 and a diametrical clearance of 11 mils and 12 mils, respectively. There was a load relief groove in the top half for no apparent reason. The operating speed was 5511 cpm. The vibration went above six mils at the whirl frequency and was not whirling at no load.

The turbine manufacturer converted to a lemon bore (elliptical) bearing with 2.5:1 convergence and oriented this bearing's butt joint 30 degrees against the rotation direction. This put the turbine in a limit cycle of up to 1 mil at 43 percent



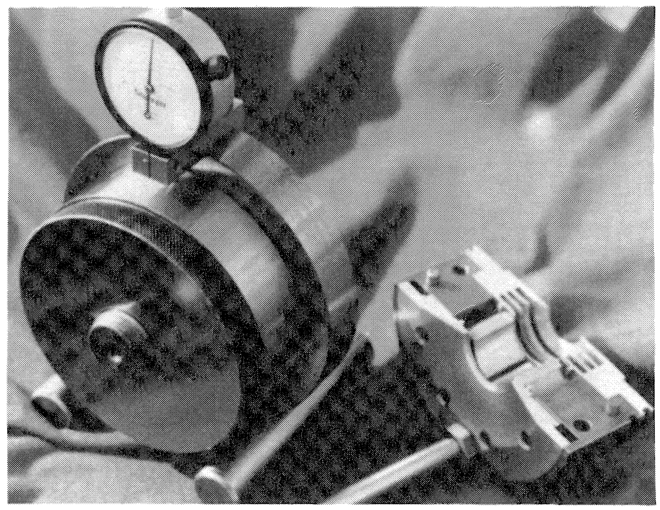
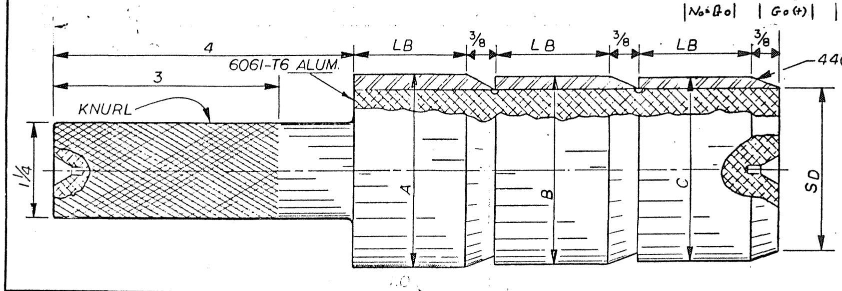


Figure 16. Small Mandrel Checker.

		BEARING BORE DIMENSIONS CLARK BRG.				MACHINING TOLERANCES A	B	C	LB	SD*	MONSANTO IDENTIFICATION
PENSACOLA AIR A & B	AGR6-8 CT*21164A1	PT.*609-239-001	BRG. BORE	5.5070	To 5.5095	5.5102/5.5100	5.5095/5.5093	5.5072/5.5070	2½	4½	
44C-101	1M.7	UNIT# 1-7-2179	BRG. BORE	2.5025	TO 2.505	2.5057/2.5055	2.5050/2.5048	2.5027/2.5025	1½	2	
46C-602-A	2M9-7	UNIT# 2-7-2181	BRG. BORE	3.1285	TO 3.130	3.1307/3,1305	3.1300/3.1298	3.1287/3.1285	1½	2	
46C-602-B	1M9-8	UNIT# 1-8-2180	BRG.BORE	2.5025	TO 2.505	2.5057/2.5055	2.5050/2.5048	2.5027/2.5025	1½	2	
46C-601-A	2M8	UNIT# 2-8-2192	BRG. BORE	3.1285	TO 3.130	3.1307/3.1305	3.1300/3.1298	3.1287/3.1285	1¾	2	
46C-601-B	2BC-10	UNIT# 2-10-2193	BRG. BORE	4.130	TO 4.132	4.1327/4.1325	4.1320/4.1318	4.1302/4.1300	1¾	3½	
45C-201-A	553 B/7	UNIT# 553-7-2177	BRG. BORE	4.5045	TO 4.5065	4.5072/4.5070	4.5065/4.5063	4.5047/4.5045	2	3½	
45C-201-B	2BC-8-5	UNIT# 2-5-2178	BRG. BORE	3.0045	TO 3.0070	3.0077/3.0075	3.0070/3.0068	3.0047/3.0045	2	2	
45C-202	24X24 BOOSTER	UNIT# 5-1-2176	BRG. BORE	4.8810	TO 4.8830	4,8837/4,8835	4.8830/4,8828	4,8812/4,8810	2	3½	
						No Go	Go (+)	Go (-)			

* — NOMINAL DIMS. SEE PROC. NOTE #4



PROCEDURE

1. ROUGH TURN 440C ANNEALED DISCS INTO SHRINK RINGS.
2. HEAT TREAT SHRINK RINGS:
   a. HEAT TO 1500°F FOR ½ HOUR
   b. INCREASE TO 1850-1900°F FOR ½ HOUR
   c. AIR QUENCH.
   d. TEMPER AT 1000°F FOR 2 HOURS.
3. GRIND FOR FIT WITH OVER STOCK ON O.D. DIMENSIONS A, B and C.
4. SHRINK 440C RINGS ON TO ALUM. HOLDER. MACHINE ALUM. O.D. FOR .002" INTERFERENCE FIT.
5. FINISH GRIND DIMENSIONS A, B and C, LEAVE TRUE CENTERS.

Figure 15. Three-step Go—No Plug Gage for Tilting Pad Radial Bearings.

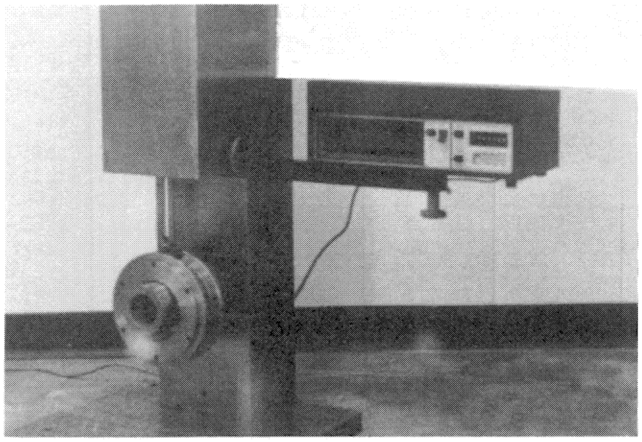


Figure 17. Electronic Gage Being Used on Tilting Pad Radial Bearing with Adjustable Spherical Ball and Seat Pads.

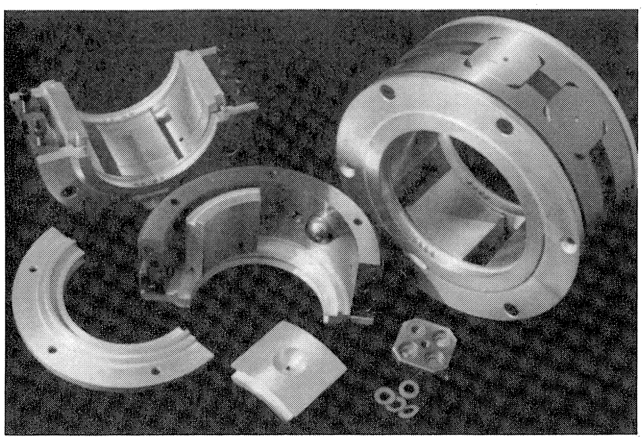


Figure 18. Radial Bearing with Spherical Seat Supports to the Retainer.

speed and was "shakey," unsteady and unacceptable. Babbitt temperatures were 235°F (Figures 19, 20 and 21).

Solution

The company installed an offset half bearing [4] which was installed with its butt joint 45 degrees off the bearing housing split line (against the rotation direction). This bearing was stable at all loads, entirely removed the sub-rotation 43 percent frequency and has operated six years without fault. Babbitt temperatures dropped to 170°F. Only the exhaust end 8 in × 4 in bearing was converted to the offset half bearing style.

CASE HISTORY NINE (THRUST/RADIAL)

A low pressure (LP) condensing steam turbine at 7500 hp and 11,000 cpm drives through a 17,000 hp topping turbine (HP) in a syn gas train driving two barrel compressors. The thrust bearing monitor for axial movement of the rotor progressed over a period of time to 28 mils and the unit shut down. Two mils of babbitt were remaining on the trailing edge of the pads and "frosting" was noted in a tapered pattern from the trailing edge progressing with more babbitt towards the leading edge, which still had 30 mils (0.76 mm) of original babbitt remaining.

New pads were installed and the same mechanism repeated, but over one to two months less time, actually four months.

A degaussing team was brought in to measure the gauss magnetic field levels about the train. Levels as high as 100

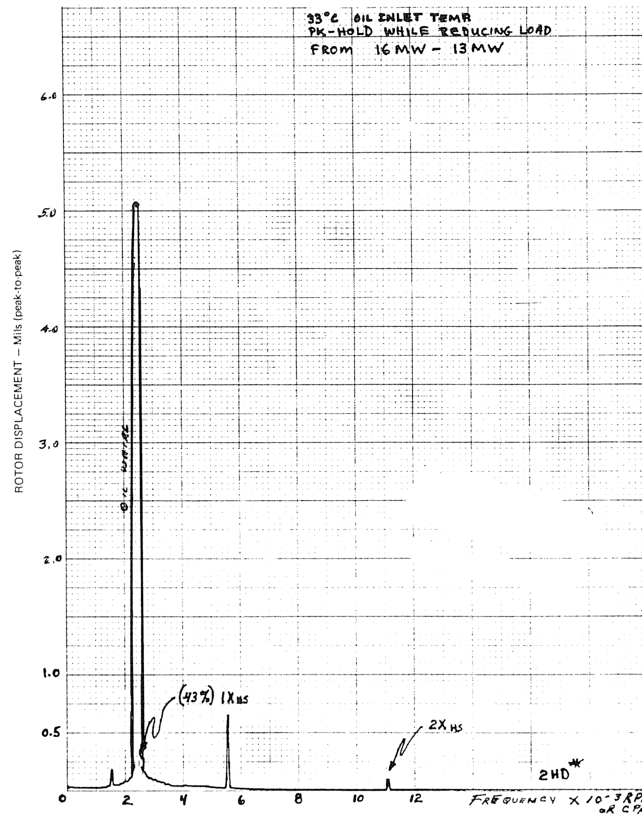


Figure 19. Spectrum of 30,000 HP Turbine Shaft in Pure Oil Whirl with Axial Groove Bearing.

gauss could be detected, with hot spots at the condensing turbine and LP barrel compressor.

Problem

Improper welding, i.e., the welding ground attachment location, was not in extremely close proximity to the weld work, had been performed. A steam line thermowell leak had been repaired in the steam line near the LP turbine, and a weld repair had been made in the LP barrel compressor's thrust balancing line (Figure 22).

Solution

The removed LP turbine rotor was degaussed to one gauss and the balance of rotor cases (rotor inside) was degaussed to a maximum level of six gauss. Two grounding brushes were installed. An axial type fine wire brush was installed on the blind end of the HP compressor shaft. A radial fine wire brush [5] was installed on the oil wetted coupling spacer tube at the LP turbine exhaust. These direct current (DC) bleeds were monitored for more than a year until the next scheduled turnaround. Operations was overjoyed after experiencing three runs of less than one year. All four rotors and four closed casings were degaussed separately at that time and the radial brush was removed. Two years later this sytem was rechecked for electromagnetic buildup and found clean.

The thrust bearing was upgraded to seven in from six in with copper backed pads and flooded type directed lubrication using spherical pad seats resting on level links. This was a 50 percent load capability uprate and babbitt temperatures on the clean (unsalted) turbine startup reduced from 225°F to 160°F.

Note: This same phenomenon repeated in three lesser, but similar, incidents in one plant in 1980 and 1984. Another near miss was headed off at another plant during an ethylene plant

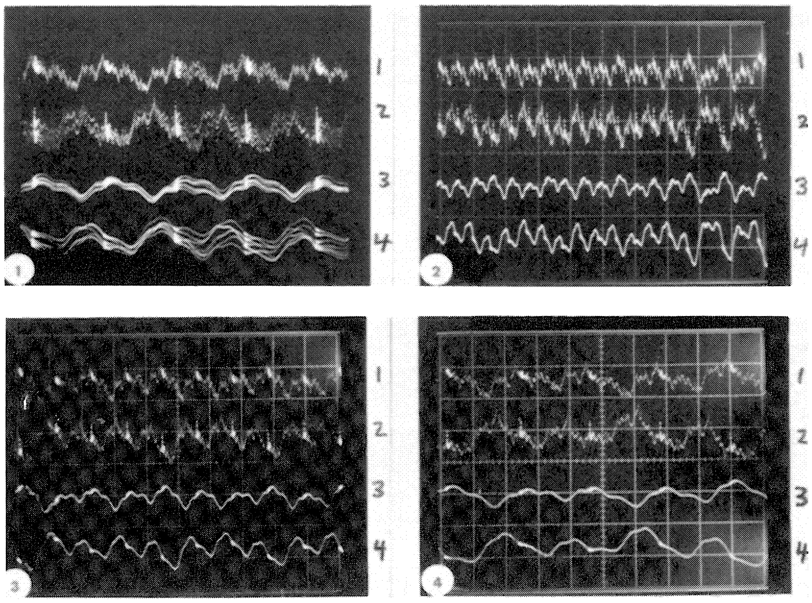


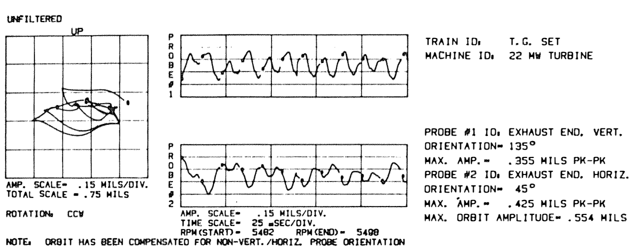
Photo 1 @ ½ Sec Exposure
Photo 2 @ 20 m Sec/Division Sweep

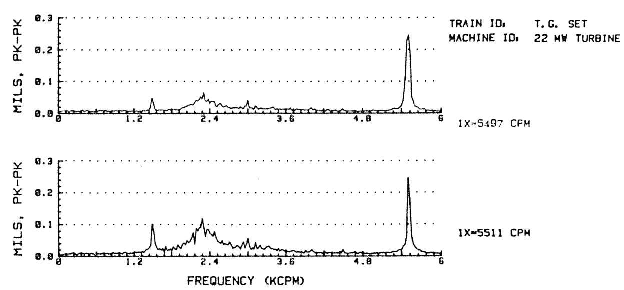
Photo 3 @ 10 m Sec/Division Sweep
Photo 4 @ 5 m Sec/Division Sweep

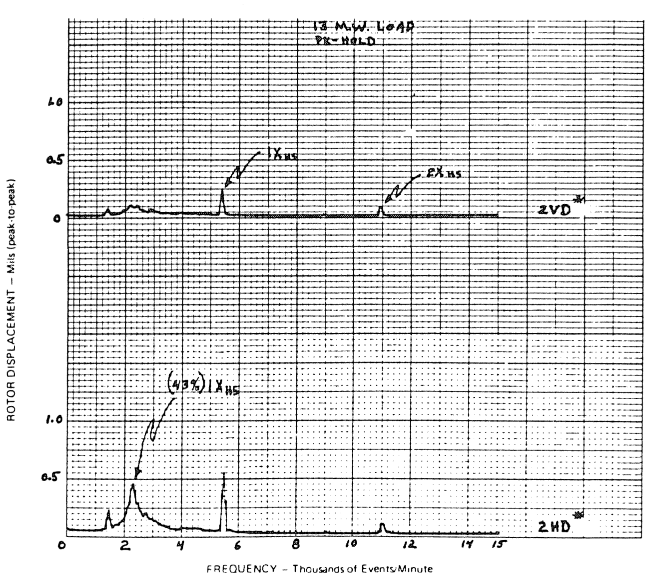
Trace 1 = Turbine Exhaust Verticle Raw
Trace 2 = Turbine Exhaust Horizontal Raw
Trace 3 = Turbine Exhaust Vertical Low Pass
Trace 4 = Turbine Exhaust Horizontal Low Pass

22MW T/G SET

Load: 13MW Speed = 5511 RPM Scale =
1/2 Thousand Peak-To-Peak/Division







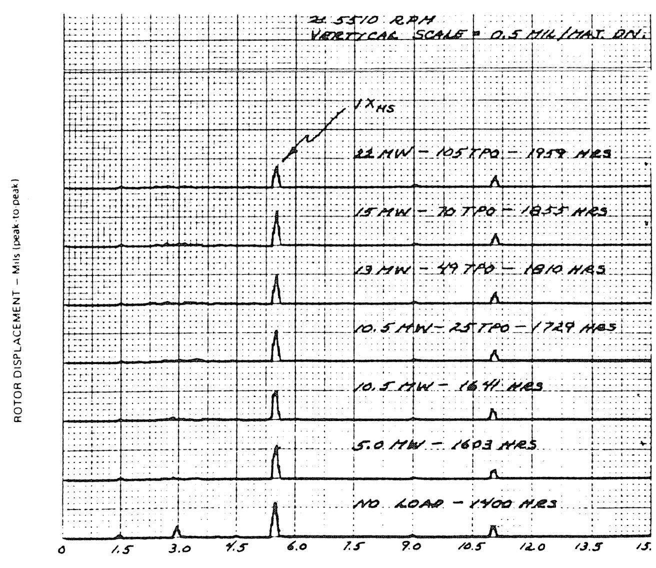


Figure 20. Scope Traces of Restrained (Limit Cycle) Oil Whirl on Lemon Bore Bearing.

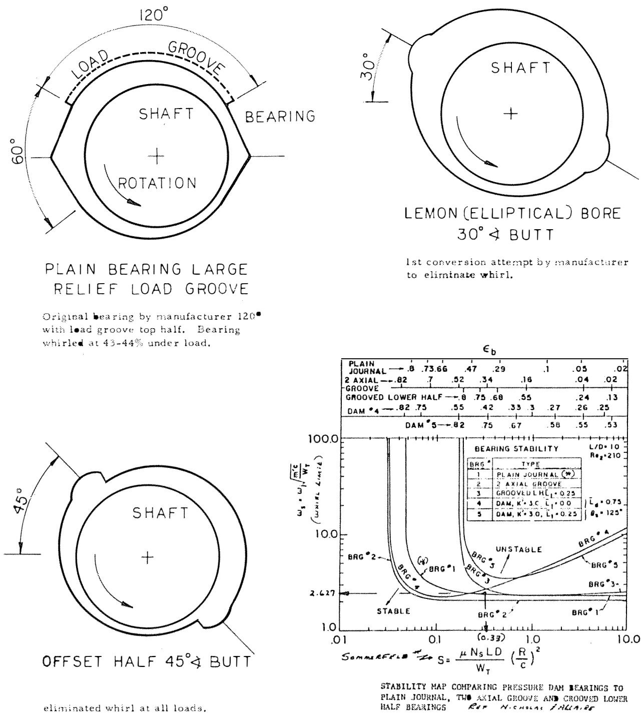


Figure 21. Summary of Three Bearings Employed in Correction. Original axial groove by turbine OEM, lemon bore by OEM, offset half bearing by user.

construction in 1980. In three projects, the couplings were not installed until alignment checks were completed in order to remove a possible conductor should improper welding occur unnoticed.

CASE HISTORY TEN (THRUST)

An 11,000 hp turbine drives two barrel compressors and a gear increaser. It operates on 575 psig steam exhausting to a 4 in Hg abs condenser. A similar turbine at 15,000 hp drives two refrigeration compressors and a gear increaser under the same steam conditions.

Each turbine has a display of the first stage pressure, which is a pressure tap immediately behind the first row of nozzles from the steam chest. This pressure is quite indicative of blade fouling from salt, silica or other foulants. It was difficult to make a year's run without the first stage pressure increasing by 100 psig or more.

The air compressor has routinely been the "Achilles' heel" and, with a first stage pressure rise from 290 psig up to 400 psig, the thrust bearing axial displacement monitors would shut this turbine down.

Solution One

The active thrust bearing was redesigned to have copper backed pads with center pivot, self-leveling and flooded directed lubrication nozzles at each pad. The size was changed from a 6 in to a 7 in bearing with a new collar and a modified bearing housing. This gave an 18 in^2 to 24 in^2 effective area increase, with a double rate of heat removal via the copper backed babbitt, for a minimum of a 50 percent total increase in the total load carrying ability.

The inactive bearing was converted from flat fluted babbitt faces to tapered land fluted faces. The flutes are for oil supply across the flats.

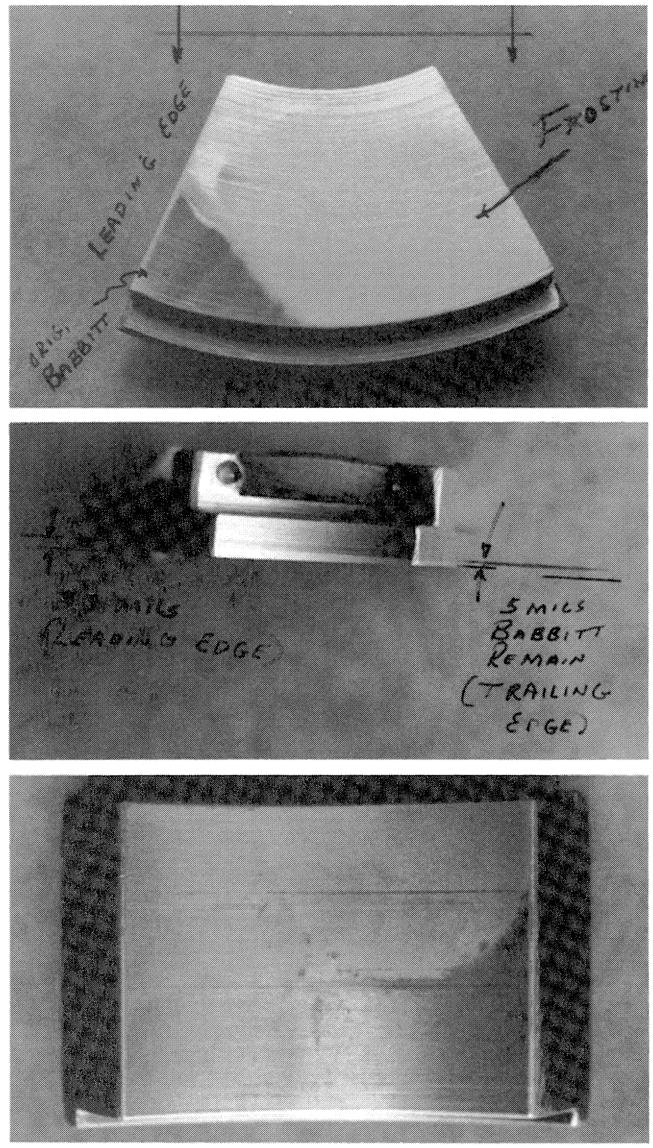


Figure 22. Frosting on Both Thrust and Radial Pads.

Problem Two

With an uprate of the air turbine, the refrigeration turbine failed in 1984 due to overload (Figure 23).

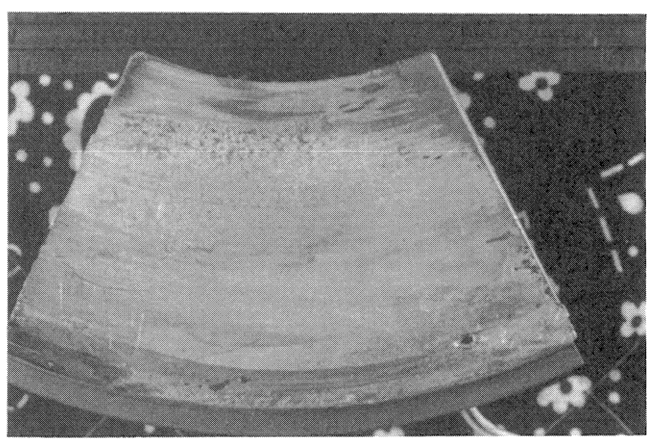


Figure 23. Pad Failures Due to Overload on a 15,000 HP Steam Turbine.

The adjacent air turbine had increased its first stage pressure from 290 psig to 432 psig which accompanied a pad temperature rise of 170°F to 220°F.

Solution Two

The 15,000 hp refrigeration drive turbine has an integral thrust collar. The active thrust bearing way, however, was converted for the following upgrades:

- Copper backed pads over steel backed pads.
- Seven pads over six pads (shorter pads).
- Offset ratio changed from 0.5 to 0.55. (Pivot is 55 percent from leading edge to assist oil into the pads. Failed pads had front (leading) edge damage).
- Flooded directed lubrication supply.
- Spherical seated level plates and pads (Figure 24).

Note: The air turbine drives the LP compressor which drives an increaser gear to the HP compressor. The LP compressor has been noted to be in its float zone on several occasions, i.e., between the active and inactive thrust bearings. The oil lubricated couplings will be converted to reduced moment dry couplings in late 1985 to decouple the axial movements of this train.

A similar move was successfully performed to decouple the condensing turbine from the topping turbine on the 22,000 hp syn gas train (four case train) more than a year ago, with the balance of that train on "hold" for conversion as more experience and feedback is gathered. The rotordynamics are not affected adversely as the reduced moment has been equal or *even less on these conversions.*

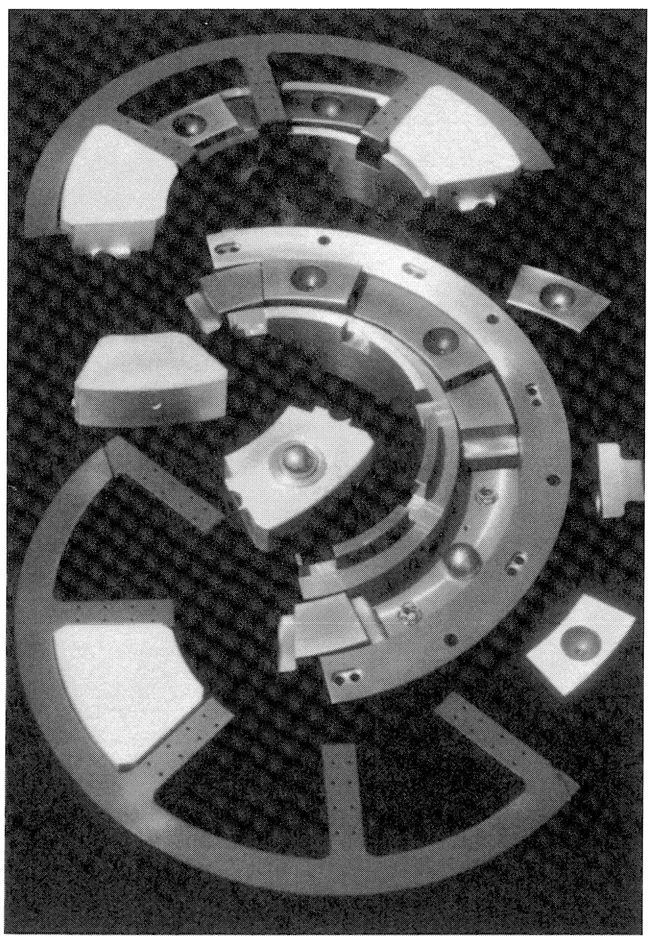


Figure 24. Details of the Redesigned Thrust Bearing.

CASE HISTORY ELEVEN (RADIAL)

Two compressor train gear boxes with plain sleeves have been recorded to show oil whirl during startup and shutdown, but not at full load (Figure 25). This has been more of an aggravation and noise sensitive phenomenon rather than damage. Only one set of bearings has been replaced in eight years and they were pounded, i.e., a "road map" pattern in babbitt.

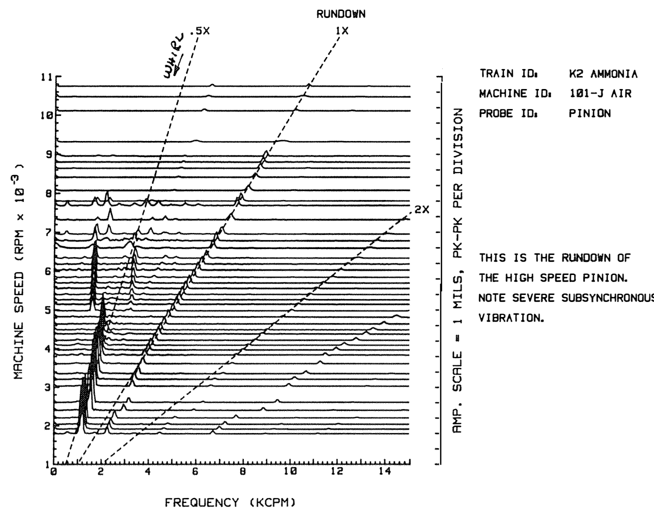


Figure 25. Rundown Data Showing the Gear Pinion Going Into Oil Whirl on Unloaded Rundown. Converse is true on startup.

Solution

These gear boxes had sleeve bearings fitted into a rather large casing. Further, the load angle on the pinion varied over 30°+ degrees during loadout. For these reasons, the solution became a thin radial profile tilting pad bearing designed using seven pads, which always keep three pads, minimum, loaded at any angle of gear tooth radial load. The assembly inserts with a radial split carrier and still allows for the top (12 o'clock) mounted proximity probe without any special modifications.

CASE HISTORY TWELVE (THRUST AND RADIAL)

An 18,000 hp steam turbine drives two ethylene centrifugal compressors at about 4500 cpm. This train was not on automatic thrust shutdown. The thrust and radial bearing failure, illustrated in Figure 26, was one of a sequence starting in the mid 1970s that ended up putting thrust axial movements on automatic shutdown in the plant. Warning (alert) alarms are at 15 mils from the commissioning clear thrust active position, with automatic shutdown (danger) at 25 mils from the same reference.

Generally, all active thrust positions are set-up within an operating unit at one-half of the average thrust float position, i.e., turbines could have 12 mils and compressors could have 17 mils. Since the float range varies, a uniform reference, e.g., +9 mils active, could be set as a position for all active thrust-to-rotor contacts. The inactive float would vary on individual machines. In this manner, all active position monitors should read about the same position, which should *highlight* any deviations. The API 670 Standard, Revision 2, for Vibration, Axial Position, and Bearing Temperature Monitors will be issued in late 1985, to better illustrate and extend coverage on many machinery protection features.

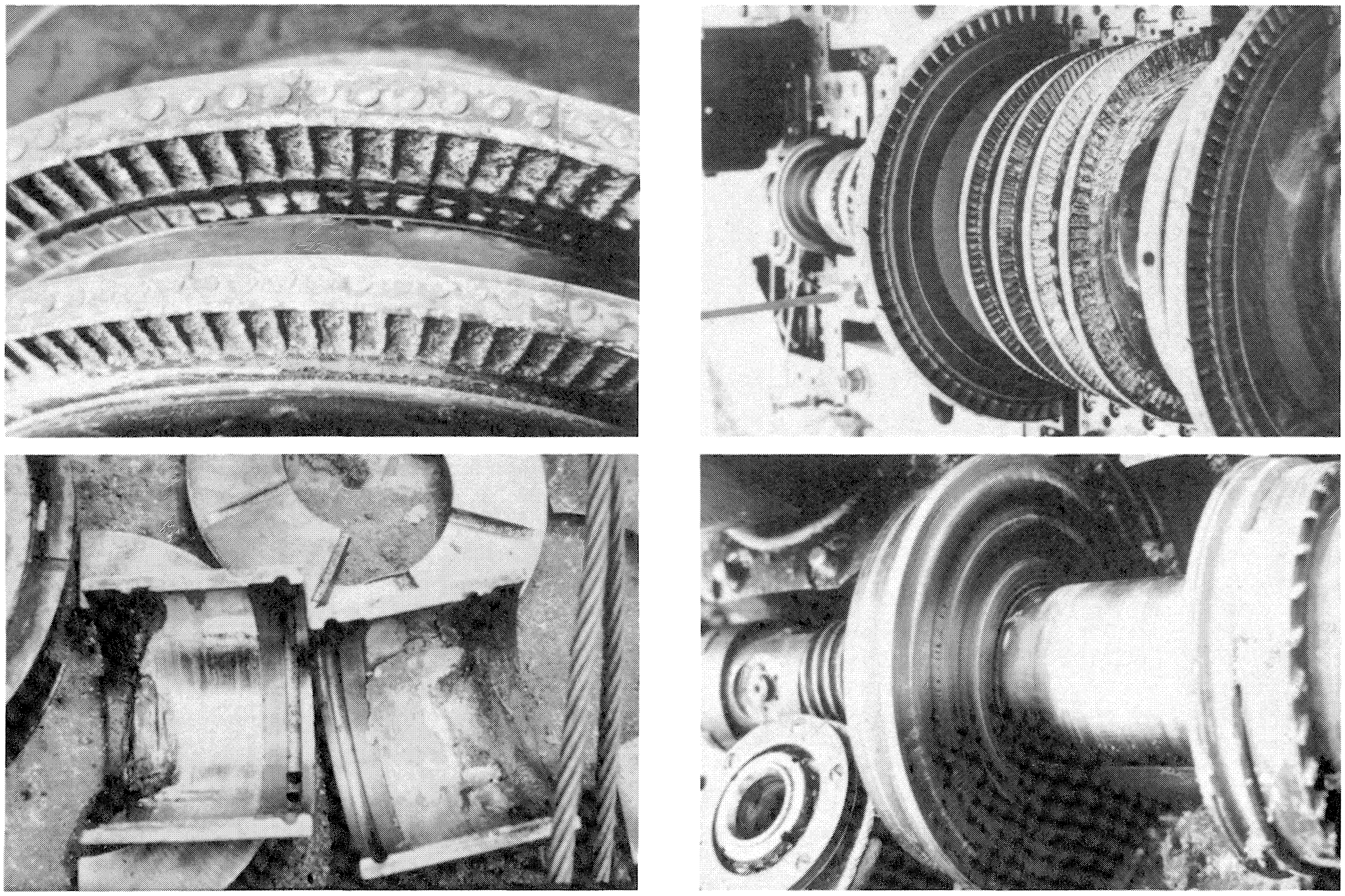


Figure 26. Thrust and Radial Bearing Damage with Hot Tears in the Thrust Collars and Salted Blading on Fourth and Fifth Stages on an 18,000 hp, 10 Stage Turbine.

Problem

The problem on this ten stage turbine was salting, due to ingestion of brine and treated water into the boiler feedwater, due to leaks in the internal exchangers of a flash evaporator. This salt, primarily NaCl, settled out selectively on the fifth and sixth stage blading (dew point of 500-525°F), obstructing the flow to a 40 percent or less flow passage area. This overloaded the tapered land thrust bearing, which melted and caused hot tears in the thrust disc. The radial bearing overheated and melted the babbitt (high tin babbitt melts at 415°F). The melted babbitt flowed down the drain oil pipe and resolidified, blocking the oil flow. The rotor journal turned blue. All turbine blading contacted the diaphragms and the shrouds cut into the diaphragms in places. Even though one million dollars was lost, the turbine was returned to service within one week, using blading tongs for blade straightening. The time-before-manual shutdown was four to five minutes. Thrust bearings will fail in less than 30 seconds.

Both thrust alarms activated and, after rotor contact was made, the radial vibration alarms activated (Figure 26).

Solution

The rotor was cleaned. The contamination to the boiler feedwater (BFW) was corrected through exchanger repairs. Steam conductivity meters and alarms were installed to monitor the steam conditions. The thrust bearings were replaced with self-leveling copper backed pads, but this was probably not needed. Self-leveling thrust pads are always a good idea to make allowances for any field imposed misalignments for whatever the reason.

CASE HISTORY THIRTEEN (RADIAL)

Two 5500 hp axial compressor trains were to be driven at 5614 cpm by 1200 cpm synchronous motors through gear increasers. The pinion (high speed) was designed to be "up" mesh as the gear (low speed) would be lifted at part load with a "down" mesh design. The pinion bearing pressure dam was placed in the lower half at 288 degrees (Figure 27). The pinion (clockwise rotation) was to operate in the −120 degrees to −135 degrees attitude position at full load, i.e., upper right hand quadrant.

Problem

The bearing analysis (DAMBRG) results utilizing finite elements can be seen in Figures 27 and 28 for the shop test, 377 hp dynamometer, and the final design load of 5500 hp at an attitude of −57.4 degrees and −73.2 degrees, respectively. The actual position of the pinion under load is directly over the oil feed slot (−89 degrees), i.e., the pinion shaft, while at an eccentricity ratio of 0.627, is *not in an area where babbitt exists*.

Measurements

The actual position of the pinion during shop testing is shown in Figure 26b. The actual position of the pinion shaft under full load in the field is shown in the eccentricity plot, Figure 27b. The pinion was utilizing only 63 percent of the bearing clearance, but did not seem encouraged to go into the upper half of the bearing.

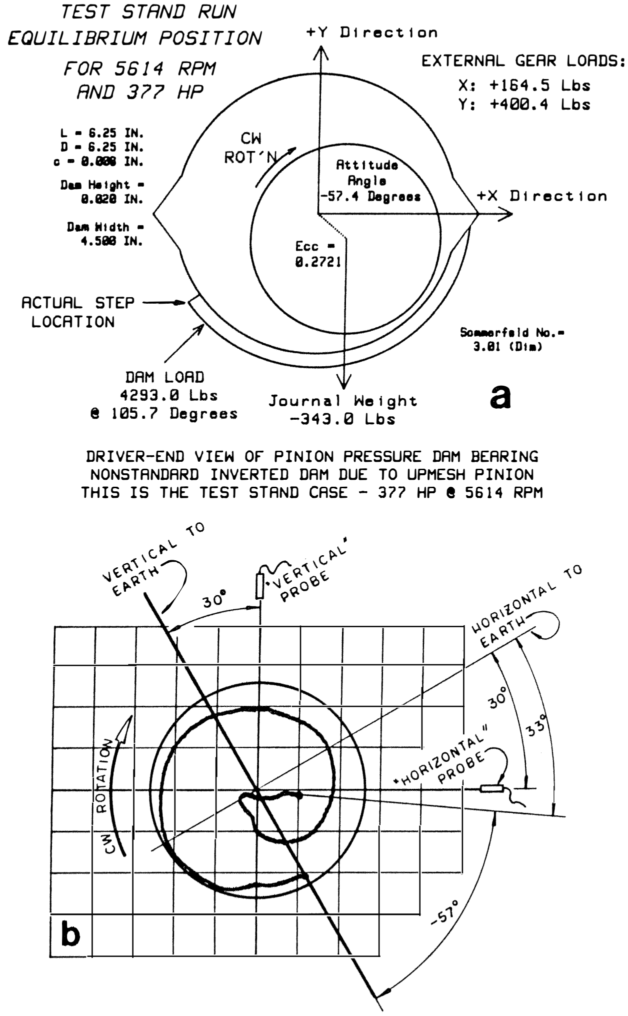


Figure 27. (a) Finite Element Analysis of the Pinion Bearing for the Shop Test at Full Speed with a 377 HP Dynamometer. (b) Actual Pinion Position as Recorded from the Two Proximity Probes at 60° and −30° Using a Two-Channel Storage Oscilloscope. The actual bearing clearance was laid out on the scope.

Solution

While the pinion has operated almost two years without problems, the bearing design has been rearranged to reposition the trailing edge of the upper half babbitt region 30 degrees further in the rotation direction, i.e., the butt joint was rolled 30 degrees clockwise. The pressure dam remains at 288 degrees. Further, the spring loaded, dual tip probe type thermocouple was relocated to 30 degrees to be more sensitive.

CONCLUSION

These 13 cases are presented to show an array of different design conversions for a specific purpose in mind. Much can be learned from careful examination of bearings, as well as proper data. A setup for data needs to be arranged to provide information to confirm or define information to be used for redesign.

Once the problem is defined it seems only logical that a plan of correction is the next step. Many people are resistant to make a change unless the original manufacturer agrees to or supports the change.

In many cases the original equipment manufacturer (OEM) will instigate and/or approve certain changes to rid a problem. However, should this not happen, why should one live with a problem rather than correct it? Apparently the OEM did not correct the problem or it would not be there.

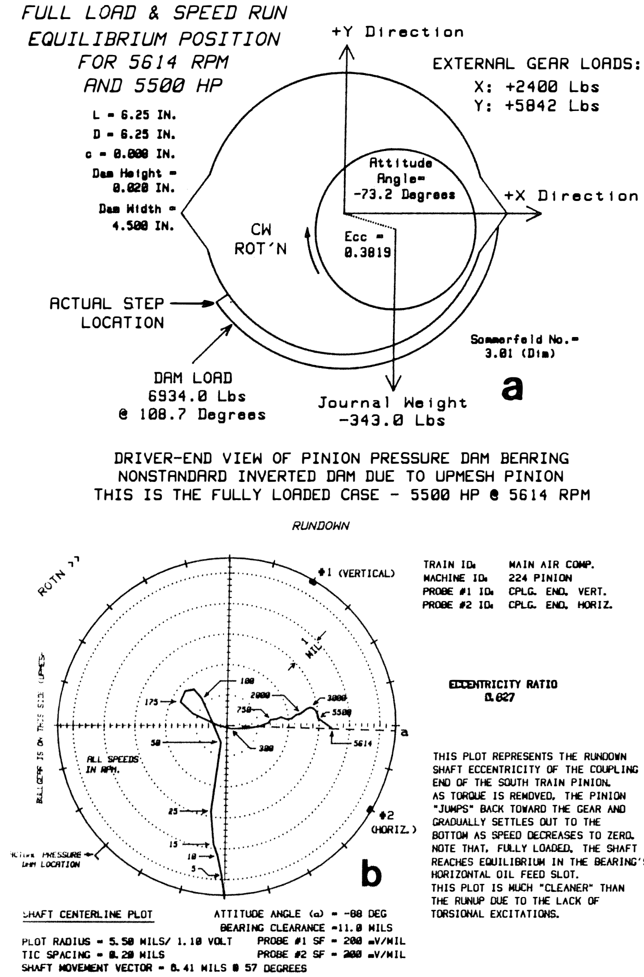


Figure 28. (a) Finite Element Analysis of the Pinion as in Figure 27 (a), but at Full Load. (b) Eccentricity Plot.

One should approach a problem solution carefully, applying good engineering practice. Do not rush out of one problem and head into another. "Only fools rush in where angels fear to tread."

There are many bearing application engineers willing to help. Generally, they provide the solutions for equipment builders in the first place. Take a bearing engineer to lunch. Do not take a bearing parts pirate to lunch—you may be the main course.

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