AN APPLICATION OF A HYDROSTATIC PRESSURE LIFT SYSTEM FOR CONTROL OF VARIABLE THRUST LOADS

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ABSTRACT

Thrust bearing failures on older equipment, especially large steam turbines, can be costly problems to remedy. A unique solution to a steam turbine thrust bearing problem that confronted Southern California Edison (SCE) is examined. The prohibitive cost of correcting the problem had forced SCE to consider mothballing the turbine. An evaluation of the situation by experts in the area of thrust bearing design produced a cost effective solution—a hydrostatic assisted thrust bearing design. The solution was implemented, and the results have proven satisfactory.

INTRODUCTION

Southern California Edison Long Beach Generating Station was facing a decision about mothballing Unit 9 steam turbine. The Long Beach Generating Station is one of two combined cycle facilities in the SCE system. This generating station produces 570 megawatts of electrical power for the surrounding communities. Seven combustion gas turbines, providing 420 megawatts, are used to power the boilers that drive Units 8 and 9 steam turbines. Over the years, continued uprates of the
original steam turbines has resulted in thrust bearing failures causing maintenance and operating difficulties.

After an in-depth evaluation, SCE suspected that a contributing cause of the thrust bearing failures was a misalignment between the turbine rotor and the generator. This misalignment, coupled with inadequate oil lubrication to parts of the thrust plate during low speed operation led to chronic thrust bearing problems. The obvious solution was to realign the generator; however, the estimated repair costs for this work exceeded the expected benefits from a major capital expenditure on a unit that would run only three to four months of the year. However, if the unit could be made operable at a reasonable cost, the result would be fuel and pollution free megawatts.

That was the situation. After further evaluation of the problem, a practical cost effective solution was determined; replace the thrust bearing with one that could withstand the misalignment and could have sufficient oil lubrication at low speeds.

The solution has been implemented and the plant has now been operating successfully for two years. Information is supplied on the history of the station along with an analysis of the problem, the unique solution, and the results of the modifications.

HISTORY

The Long Beach Facility has retired all but two of the original steam turbines installed at the site. In the 1920s the Long Beach Facility consisted of 11 steam turbines producing 550 megawatts of power. Unit 8 was replaced by unit 8R in 1943. More about that later. Units 1 through 7 were retired from service over a 10 year period from 1954 to 1964. Units 8R, 9, 10, and 11 were operated until 1967, when unit 8R and 9 were retired. In 1976, seven combustion gas turbines were installed at the site, and unit 8R and 9 were returned to service. In 1979, unit 10 and 11 were destroyed during operation when flooded by a high tide from the Pacific Ocean. Units 8R and 9, along with the seven combustion gas turbines, are still in place today.

The original Unit 8 and 9 were built as 50 MW, 50 cycle condensing turbines with two extraction points for supplying steam to local refineries. On November 24, 1942, Russia was under military attack and the need for power generating equipment was critical. The United States Government confiscated Unit 8 and placed it on a ship to Russia. On December 1942 the unit was lost somewhere in the Atlantic Ocean when the ship on which it was being transported was sunk. Unit 8R, rated at 73 MW, replaced Unit 8 and was commissioned on October 23, 1943. Unit 9 is the only one of the original 11 steam turbines that remains.

PROBLEM DESCRIPTION

In the late 1940s, the United States converted from 50- to 60-cycle power. This required increasing the operating speed of the steam turbines from 1500 to 1800 rpm. This increased rpm was the beginning of greater demands on Unit 9. In 1976, when Unit 9 was returned to service to operate on the steam produced by the exhaust heat recovery system, the two previously used extraction points were blanked off resulting in rerating the unit from 50 MW to 69 MW. However, because of other problems, Unit 9 could not be tested at the new rating. From 1977 to 1986, combustion gas turbine reliability problems prevented the plant from reaching full capacity. In 1986, all seven gas turbines were brought to reliable service enabling Unit 9 to be run for the first time at its full output capability. The several upgrades undertaken over the years finally pushed Unit 9 to its limits, and thrust bearing problems began.

During this first year of operation at the new rating level, a foreign object became lodged in the nozzle block of Unit 9 and bent the first row of blades over, resulting in steam path blockage. The thrust bearing failed when the thrust collar moved from thrust overload. The thrust collar retaining nut stripped the shaft threads and allowed the thrust collar to slide down the shaft. The rotor moved ⅛ in downstream causing major damage to the rotating and stationary turbine components.

SCE selected a manufacturer other than the OEM to supply a new rotor and refurbish the stationary components. The original shrink-on wheel construction was replaced with a new solid rotor design to allow shorter startup times. The shrink-on wheel design could result in loose wheels from differential thermal growth if started too quickly, and the wheels could move into contact with stationary parts. The manufacturer also supplied a standard integral thrust collar, eliminating the possibility of the previous thrust collar failure. The thrust bearing was further modified from a 360 degree cage to a split cage for ease of installation during maintenance. Temperature monitoring of the thrust bearing was also improved. The number of thermocouples was increased from 4.0 to 8.0. The turbine casing was also modified to accommodate the new rotor. The unit was returned to operation October 23, 1988.

Immediately after returning Unit 9 to service, the power company began monitoring the unit thrust bearing temperatures more closely. By January 1989, performance monitoring data revealed a normally high thrust bearing temperature of 180°F which was normally 170°F. These temperatures continued unabated until June 1989, when Unit 9 was opened for a thrust bearing inspection. During inspection, the lands of the tapered land thrust bearing were found to be severely worn. Approximately 60 percent of the thrust shoe taper had been worn away. Also, the thrust bearing plate was found in a cocked position. The program mapped out at that time to correct the misalignment condition included removal of the thrust bearing and cage, machining of the journal bearing housing face and thrust cage, resurfacing and reinstallation of the tapered land thrust bearing.

One year later, January 14, 1990, Unit 9 was taken out of service for an inspection of the new rotor. Thrust bearing temperatures and differentials had been at high levels up to the time of the inspection. During this inspection, major damage was found in the thrust bearing. Approximately 40 percent of the surface area had been wiped. The thrust cage and the tapered land bearing were remachined and recontoured back to factory specifications. The unit was returned to service on April 30, 1990.

On December 19, 1990, the thrust bearing on Unit 9 experienced a sudden and catastrophic failure. The rotor moved toward the generator end 0.180 in. All seven gas turbines and both steam turbines were in service at the time, and Unit 9 was at 54 MW. Inspection revealed the active thrust bearing babbit material had melted and been wiped off the plate by the centrifugal force imparted by the thrust collar. In addition, the thrust collar had deep scratches. A spare tapered land thrust bearing was recontoured and the thrust collar was machined smooth and flat in place. The cage was machined to bring the rotor to its original thrust position specifications. Unit 9 was returned to service on January 22, 1991.

The site continued to closely monitor thrust bearing temperatures from January 22, 1991 to March 22, 1991, when Unit 9 was again removed from service for a thrust bearing inspection. This inspection revealed that the active thrust bearing was wiped 1.0 to 3.0 mils. At this point, the turbine shells were removed to inspect for internal damage, but no damage was found.

In the summer of 1991, SCE made the first attempt to find the root cause of the Unit 9 thrust bearing problems. The OEM calculations of theoretical thrust loading showed no obvious excessive thrust load problems. The rotor blades had been previously coated to retard corrosion and erosion. It was consid-
ERED THAT THE COATING MAY HAVE REDUCED THE BLADE OPENING ENOUGH TO INCREASE THRUST LOADS. THE COATING WAS BLASTED OFF TO INCREASE THESE CLEARANCES AND LESSEN THRUST. WHILE THE ROTOR WAS OUT, A LASER TRANSIT SURVEY WAS DONE TO DETERMINE TURBINE ELEVATIONS END TO END. DURING THE SURVEY, IT WAS FOUND THAT THE GENERATOR END OF THE UNIT WAS 1 INCH LOWER THAN THE TURBINE END. THIS DISCOVERY RAISED CONCERNS ABOUT THE EFFECTS ON THE THRUST BEARING FROM POSSIBLE MISALIGNMENT FORCE WHILE ON TURNING GEAR, WHEN ONLY BOUNDARY LUBRICATION IS PRESENT. A THREE-DAY TEST ON TURNING GEAR WAS UNDERTAKEN WITH BOTH THE AUXILIARY OIL PUMP AND TURNING GEAR OIL PUMP IN OPERATION. AFTER THREE DAYS, THE THRUST BEARING WAS REMOVED AND FOUND TO BE SEVERELY BUT EVENLY WORN. IT WAS CONCLUDED THAT MISALIGNMENT AND LACK OF LUBRICATION RESULTED IN THE WEARING OF THE THRUST PLATE. A NEW THRUST BEARING WAS INSTALLED. AT THIS TIME, SCE CONSIDERED CORRECTING THE FRONT TO BACK TILT BUT REJECTED THE MEASURES NEEDED AS TOO COSTLY, I.E., IN EXCESS OF ONE MILLION DOLLARS.

BY DECEMBER 1991, HIGH THRUST BEARING TEMPERATURES PROMPTED ANOTHER INSPECTION. THE THRUST BEARING WAS FOUND TO BE WORN EVENLY AND UNEVENLY THIS TIME. THIS PROMPTED AN INVESTIGATION OF THE PEDESTAL (STANDARD) SUPPORTING THE THRUST BEARING. THE THRUST BEARING IS A COMBINATION THRUST/JOURNAL BEARING ASSEMBLY. THE WHOLE ASSEMBLY IS SUPPORTED BY A SPHERICAL BEARING WITH 0.0 TO 0.002 IN CLEARANCE TO ALLOW THE BEARING TO ADJUST IN SERVICE FOR ALIGNMENT. THE SUPPORT IS DESIGNED TO MOVE WITH THERMAL GROWTH OF THE TURBINE CASING AND, IF OBSTRUCTED, CAN LEAD TO UNDESIRABLE MISALIGNMENT. THE FRONT STANDARD BASEPLATE CENTERLINE KEY WAS FOUND TO BE SEVERELY WORN, WHICH COULD INDICATE THE STANDARD WAS MOVING UNEVENLY DURING STARTUPS. IT WAS DECIDED TO OVERHAUL THE STANDARD. THE THRUST BEARING BALL HOUSING FIT WAS RENWED TO ASSURE FREEDOM OF MOVEMENT.

ON JANUARY 16, 1992, A PROGRAM WAS BEGUN TO MAKE SURE THE GOVERNOR STANDARD WAS FREE TO SLIDE PROPERLY DURING OPERATION. A SLIDE TEST WAS CONDUCTED TO DETERMINE THE FRICTION BETWEEN THE FRONT STANDARD AND THE BASEPLATE. THE HORN KEYS AND THE ELEVATION/AXIAL KEYS WERE REMOVED AND THE TURBINE CASING WAS SUPPORTED ON ROLLERS BETWEEN IT AND THE STANDARD TO ELIMINATE ANY BINDING BEFORE ATTEMPTING THE PUSH/PULL TEST. A HYDRAULIC JACKET WAS USED TO MOVE THE STANDARD AND THE HYDRAULIC FLUID PRESSURE WAS MEASURED TO DETERMINE THE Jacking FORCE. IT WAS FOUND THAT THE FRICTION VALUES WERE WITHIN THE 0.3 COEFFICIENT OF FRICTION SPECIFICATION.

FROM THE SUMMER OF 1992 TO JANUARY 1993, ADDITIONAL EFFORTS WERE BEGUN TO FIND THE ROOT CAUSE OF THE THRUST BEARING TEMPERATURE PROBLEM. A THIRD PARTY SUPPLIER WAS CONTACTED TO PROVIDE OBSERVATIONS AND RECOMMENDATIONS. IN DECEMBER 1992, DURING A COLD PLANT OUTAGE (A COMPLETE PLANT SHUTDOWN), MEASUREMENTS OF THE JOURNAL THRUST BEARING CAVITIES WERE MADE FOR POSSIBLE BEARING MODIFICATIONS. FROM JANUARY 1993 TO MAY 1993, PARAMETERS FOR A SPECIFICATION FOR PROBLEM SOLUTION WERE DEVELOPED. IT WAS CONCLUDED BY THE POWER COMPANY THAT A THRUST BEARING WAS NEEDED THAT WOULD MEET THE FOLLOWING REQUIREMENTS:

- WITHSTAND POSSIBLE MISALIGNMENT FORCE WHILE ON TURNING GEAR
- REDUCE TEMPERATURE DIFFERENTIALS ON THE ACTIVE SIDE PADS
- OPERATE WITH AN IMPROVED LUBRICATION SYSTEM
- BE SUPPLIED AT A REASONABLE COST

DESCRIPTION OF THE NEW COMBINATION JOURNAL/THRUST BEARING

AS MENTIONED EARLIER, THERE WAS CONCERN ABOUT POSSIBLE MISALIGNMENT FORCES ON THE ACTIVE THRUST BEARING. COMBINING THIS WITH EXTENDED PERIODS ON TURNING GEAR WITH ONLY BOUNDARY LUBRICATION RESULTED IN DEGRADATION OF THE BEARING SO THAT FAILURE OCCURRED WHEN THE UNIT WAS PUT ON LINE. AFTER EVALUATION OF THE PROBLEM, THE POWER COMPANY DECIDED TO CHANGE THE OLD TYPE BEARINGS (ELLIPSE BORE JOURNAL BEARINGS AND TAPER THRUST PLATE) TO A DIFFERENT DESIGN. THE NEW DESIGN BEARING (FIGURE 1) IS A COMBINATION JOURNAL AND THRUST BEARING. IT REPLACES THE OEM BEARING AND IS INSTALLED IN THE TURBINE WITH A 0.0 TO 0.002 IN INTERFERENCE ON A CYLINDRICAL FIT WITH THE TURBINE CASE (FIGURE 2). IN ADDITION TO THE REDESIGNED THRUST BEARING, A LIFT OIL SYSTEM WAS ADDED.

Figure 1. Thrust Bearing Location.

BEARING DESCRIPTION

THE JOURNAL BEARING IS A BALL AND SOCKET TYPE. IT IS DESIGNED FOR MAXIMUM ALIGNMENT CAPABILITY WITH FIVE STEEL PADS, LOAD BETWEEN PADS. BOTH THE ACTIVE AND INACTIVE THRUST BEARINGS ARE A SELF-EQUALIZED DESIGN WITH DIRECTED LUBRICATION (EVACUATED CAVITY TYPE). THE SELF-EQUALIZING FEATURE IS ACCOMPLISHED BY A SERIES OF INTERMESHED ROCKERS THAT DISTRIBUTE THE LOAD EVENLY. THE OIL FROM THE TUBE OIL SYSTEM FEEDS THROUGH NOZZLES AT THE LEADING EDGE OF EACH PAD (FIGURE 3), ensuring positive lubrication. Oil flow through the thrust bearing is enabled by drain holes sized to maintain a slight amount of oil in the bottom of the thrust cavity. This oil will be picked up by the collar for lubricating the thrust surface while operating on the turning gear. When the turbine is operating at normal speed, the natural effect of the thrust collar spin will push the oil out through the drain.

The thrust pads are mounted with ball-and-socket pad supports. Thrust pads are of chromium-copper alloy for better heat conduction than steel. Each of the 16 pads (eight on the active and eight on the inactive bearing) has an individual hydrostatic pressure lift supply, which feeds a hydraulic pocket through a high pressure check valve. A separate lift oil manifold is provided for both the active bearing and the inactive bearing. From the manifold, the hydrostatic lift oil flows through a ⅜ in flex hose to the individual pad.

HYDROSTATIC PRESSURE LIFT SYSTEM

The hydrostatic pressure lift (HPL) system is a complement to the redesigned thrust bearing. It provides increased lubrication
during startup and shutdown and during low speed operation, such as when the turbine is on the turning gear. In addition, the hydrostatic pressure lift system provides additional thrust bearing cooling during high load operation. In the HPL system, oil is supplied at required pressure to both active and inactive thrust bearings, and the flow is controlled by separate flow control valves for the inactive side and for the active side. Since the flow rates are fixed, the pressure in the system, and therefore hydrostatic performance, is a function of the bearing load only.

**Components**

The hydrostatic pressure lift (HPL) system is independent of the thrust and journal bearings lubrication oil system. The HPL oil is delivered by an electric-driven pump to both thrust bearings. Each thrust bearing pad has an individual flexible line to one of the two lift oil manifolds, and each line is fitted with a calibrated orifice and a check valve. Refer to Figure 4 for a diagram of the HPL system.

![Figure 3. Thrust Pad.](image)

Figure 4. Hydrostatic Pressure Lift System.

The HPL system is made up of the following components:

- Booster pump
- Electric motor
- Filters
- Temperature gauges
- Pressure gauges
- Adjustable range relief valve
- Adjustable range flow control valve
- Adjustable pressure switch

**Design Calculations**

The hydrostatic performance of the thrust bearing is a function of the total load and the flow rate. The flow rate to each bearing is fixed by the flow control valves (FCV) at 10 gpm to the inactive side and 12.5 gpm to the active side. Since the flow rates are fixed, the pressure in the system, as mentioned, is a function of bearing load only. Pressure upstream from the FCV is maintained by the pressure relief valve. From Equation (1), it can be estimated that for a 300 psi total thrust load with an
approximately 25 in² lift surface, the system should supply approximately 2300 psig.

Art. Equation 1 MUST go here. GUIDES...

ADVANTAGES OF THE REDESIGNED SYSTEM

The journal bearing ball-to-socket improves bearing load distribution. The ball-to-socket contact allows each pad to tilt both axially and circumferentially. This ensures that any misalignment will be evenly distributed across the bearing load area. The use of this type pad support system eliminates the problems associated with possible lock-up of the spherical seated bearings previously used. Also, the ball-and-socket design reduces stresses at the pad to housing contact by allowing the load to be distributed over a larger pivot area. This maintains a constant bearing-to-shaft clearance, which changes in other designs because of wear.

Direct lubrication of the bearing pads improves efficiency. By supplying oil through fixed orifices at the edge of each journal pad, this system supplies only the oil needed and avoids extra oil in the bearing cavity, which would just increase the temperature of the bearing through churning. This situation is similar in the thrust bearing. The oil is injected through the nozzles to the leading edge of each pad assuring positive lubrication and continuous cool oil supply.

The self-equalizing feature of the thrust bearings compensates for angular misalignment between the thrust collar and thrust bearing support and distributes thrust loading circumferentially to give equal loading on all pads. Ball-and-socket thrust pad supports reduce contact stresses over point contact eliminating brinelling of the thrust pad pivots, and, therefore, maintaining axial tolerances to extend run times.

The HPL system has proven successful in three important stages of turbine operation: at startup or shutdown, while on turning gear (when slow speeds occur), and at high thrust bearing temperatures (high speed and high thrust load). While on the turning gear, the system provides additional lubrication for the bearings. At high thrust bearing temperatures, this system provides additional cool oil to the bearing.

The major change in design of the active thrust bearing to a self-equalizing type with the HPL system provide significant advantages over the old style thrust plate. Specifically, these advantages are the following: compensation is made for misalignment, pad-to-pivot contact stress is reduced, bearing load is more evenly distributed among the pads, thrust bearing lubrication is improved at low speed, thrust bearing cooling is improved at high turbine loads, and parasitic oil churning losses are reduced.

CONCLUSION

The redesigned thrust bearing system has allowed SCE to continue using a vital low-pollution component of their power generation system. The redesigned thrust bearing with HPL allows continued use of the steam turbine without the expensive capital outlay for thrust bearing repairs. In addition, the redesigned system allows increased output from Unit 9.

SCE is presently using the HPL system in the following manner: With the HPL supply off, the power output can be increased to 46MW, where the temperature on the active thrust bearing rises to approximately 220°F. At this point, the HPL system is placed in operation. With the HPL system in operation, the thrust bearing temperature decreases to 170°F. This improved cooling allows SCE to continue to raise power output to 68MW.

From September 1993, to March 1995, the following steps were taken:

- Installing the redesigned thrust bearing
- Added a hydrostatic pressure lift system to the tilt-pad thrust bearing
- Testing the new thrust bearing system

Among the advantages accrued from the redesigned thrust bearing system are the following:

- Eliminates premature bearing failure at startup
- Eliminates premature bearing failure while on turning gear
- Eliminates progressive thrust bearing wear
- Decreases thrust bearing operating temperature
- Decreases temperature differential between individual thrust bearing pads
- Allows operating on turning gear with existing axial misalignment

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