DESIGN MODIFICATIONS AND ACTIVE BALANCING ON AN INTEGRALLY FORGED STEAM TURBINE ROTOR TO SOLVE SERIOUS RELIABILITY PROBLEMS

by

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ABSTRACT

This paper describes the design modifications and active balancing on a 19,000 hp integrally forged steam turbine rotor to improve its dynamic characteristics and reliability. The turbine was purchased as a replacement for an identical turbine used in a syn gas train. Ever since original installation, the turbine had been plagued with high vibration problems on the coupling end, excessive bearing temperatures, and reduced coupling life. A rotordynamic analysis proved that there was sufficient separation margin between the turbine's operating speed and any critical speed. However, at operating speed, the rotor had a high vibration response at the coupling, even with a very small unbalance. A design modification was thus incorporated on the rotor to make it less sensitive to unbalance. At operating speed, this modification lowered the predicted coupling end shaft deflection to approximately a third of its original value. An active balancer was then assembled on the rotor to provide the flexibility of online trim balancing. The replacement turbine has been installed and is operating with the lowest vibration and bearing temperatures in the compressor train's history. Presented in this paper is an overview of the rotordynamic analysis, the shaft welding technique used to accomplish the rotor design changes, and the balancing with the active balancer. The techniques presented in this paper can be used to improve the mechanical reliability of other machines with similar problems.

INTRODUCTION

With modern engineering techniques, machinery repair is no longer limited to "in kind" parts replacement, rotor balancing, and assembly. Today, repairing a turbomachine often includes significant design changes to improve reliability, dynamic characteristics, and performance characteristics. In this paper, one such example is presented. The subject steam turbine had a history of high vibration on the coupling end, maximum radial bearing temperatures of approximately 225°F, and reduced coupling life. In one instance, the turbine shaft suffered a fracture just inboard of the coupling end bearing journal, resulting in a catastrophic failure. The steam turbine has a power output of 19,000 hp at an operating speed of 10,800 rpm. The inlet temperature on the turbine is 950°F and the inlet pressure is 1500 psig. Due to the operating conditions of the turbine, the rotor is of an integrally forged design with six stages, the last two of which are double flow stages. The rotor with the failed shaft end is shown in Figure 1.

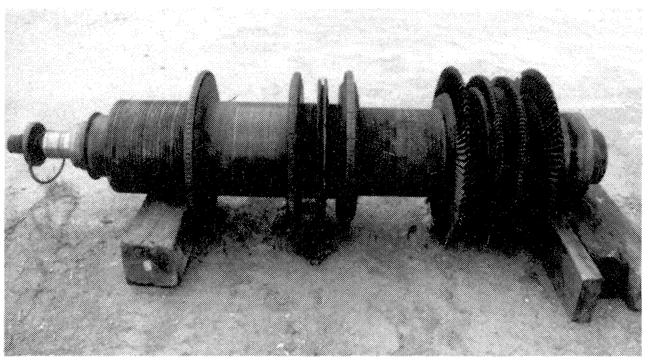

Figure 1. The Rotor with the Shaft Failure on the Coupling End. (The rotor was sectioned just inboard of the fracture for a failure analysis.)

A surplus machine was purchased as a spare, due to the problems encountered. The purchased turbine had seen a number of years of service and several years of storage. Thus, the replacement turbine required a complete overhaul before installation. At the onset of the overhaul, it was also decided to conduct a complete rotordynamic analysis to better understand the rotor's operating characteristics. The rotordynamic analysis revealed that, per API guidelines, there was sufficient separation margin between the rotor's operating speed and any critical speed. However, at its maximum continuous operating speed, the rotor was extremely sensitive to coupling unbalance. The approach taken to correct this design problem was twofold. First, the rotor shaft was redesigned to reduce deflection at running speed. Second, an active balancer was assembled on the rotor to provide the flexibility of online trim balancing. Presented in this paper is an overview of the rotordynamic analysis, the shaft welding technique used to accomplish the rotor design changes, and the performance of the active balancer.

THE ROTORDYNAMIC ANALYSIS

To reduce operating speed vibration problems on the original machine, a balance ring had been added on each end of the rotor, outboard of the bearing journals. Periodic field balancing had always been required for safe operation of the machine, but this only partially alleviated the problems. Coupling end vibration and bearing temperatures remained too high, and short coupling and bearing life persisted. The insight that would be provided by the rotordynamic analysis was thus considered invaluable.

The Rotor-Bearing Model

The rotor-bearing model was developed from actual rotor dimensions and existing bearing prints. Shown in Figure 2 is a schematic of the rotor-bearing system. The existing coupling half-weight and balance rings were modelled as concentrated masses at the appropriate locations. The existing bearings were five pad load-on-pad (LOP) with centered pivots, for which speed dependent bearing stiffness and damping coefficients were generated from 2500 rpm to 13,000 rpm.

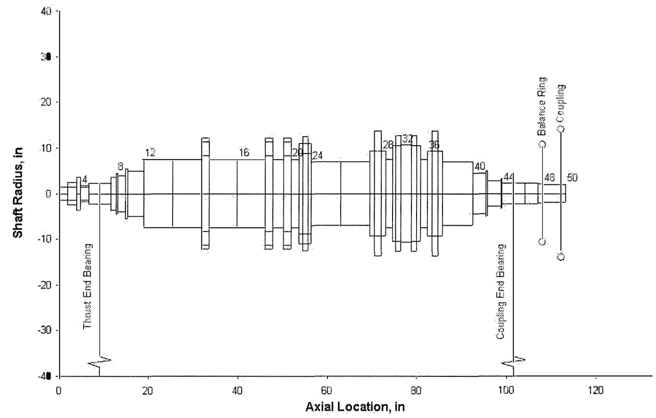

Figure 2. Rotor-Bearing Model Used for the Rotordynamic Analysis.

Stability Analysis

The stability analysis was conducted from 2500 rpm to 13,000 rpm. Calculations were done at nominal, minimum, and maximum bearing clearances, and no stability problems were seen. However, one of the mode shapes discovered was of significant concern. The study showed that the rotor operated above two natural frequencies, and well below the third natural frequency of approximately 15,700 cpm. Shown in Figures 3, 4 and 5 are the first three forward precessed, flexible mode shapes of the rotor-bearing system, superimposed on the rotor model. In Figure 5, the third mode is seen to have high deflection on the coupling end of the shaft. It was thus imperative to excite this rotor mode during the unbalance response analysis.

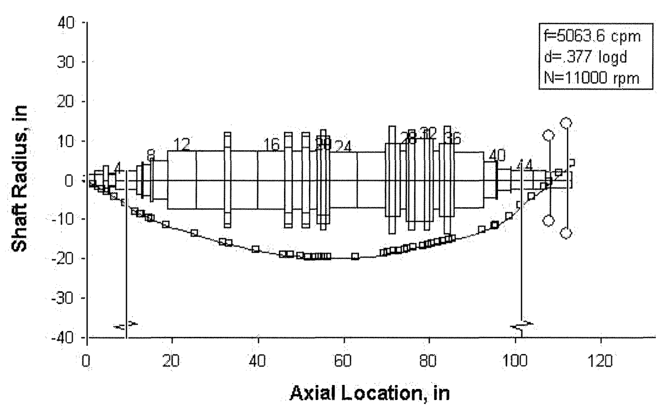

Figure 3. First Forward Precessed Flexible Mode—Original Shaft Configuration.

Unbalance Response Analysis

In order to characterize the response of the rotor to unbalance, three unbalance cases were used:

- Static unbalance at the center of the rotor (Station 23)
- Couple unbalance at quarterspan (Stations 14 and 37) of the rotor
- Unbalance at the coupling (Station 50)

Unbalance response plots for the three cases listed above were generated at the following stations:

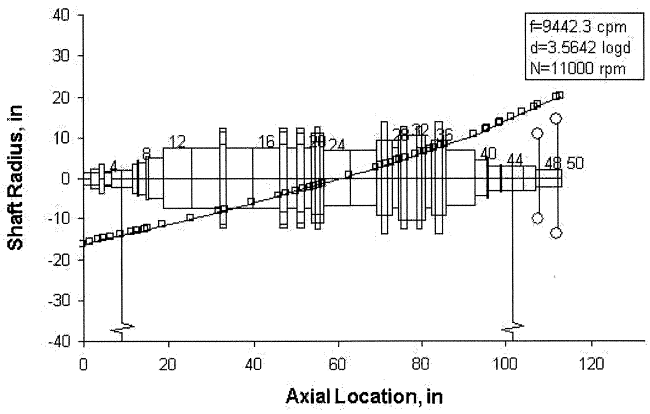

Figure 4. Second Forward Precessed Flexible Mode—Original Shaft Configuration.

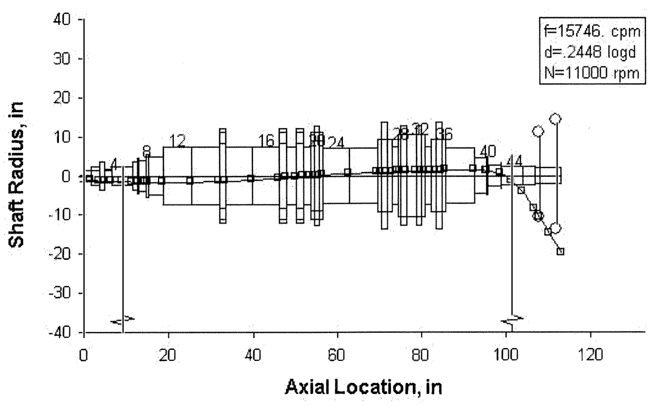

Figure 5. Third Forward Precessed Flexible Mode—Original Shaft Configuration.

- Station 6: Thrust end radial bearing
- Station 45: Coupling end radial bearing
- Station 23: Rotor midspan
- Station 50: Coupling plane

This allowed an accurate study of the rotor deflections at different locations on the rotor, throughout its speed range. Shown in Figure 6 is the predicted coupling end bearing response to an unbalance placed on the coupling (Station 50). Figure 7 shows the response at the coupling plane to the same unbalance. The actual unbalance amount placed on the coupling was two times the API low speed rotor balancing tolerance (i.e., 8 W/N). At operating speed, the response at the coupling is significantly higher than the response at the bearing, showing the excessive deflection on the outboard end of the shaft (Figure 5). The absolute magnitude of shaft deflection was also unacceptable. Due to the shape of the mode, a bearing redesign was not an effective way of controlling the shaft end deflection. The only viable option was to modify the shaft geometry.

Rotor and Bearing Design Modification

The rotor design modification was conducted by analyzing and optimizing different shaft geometries on the coupling end of the rotor. Complete rotordynamic analyses were conducted for each geometry chosen. An overlay of the old design and the final modified shaft design is shown in Figure 8. The shaft bearing journal diameter was increased from 4.5 inches to 6.0 inches, while the coupling end of the shaft was changed from a straight keyed connection to a tapered hydraulic fit. This design modification

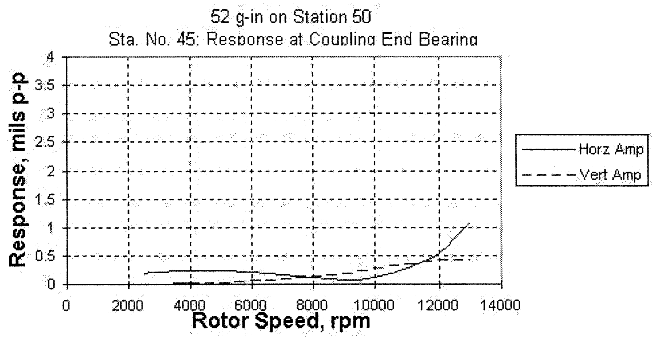

Figure 6. Coupling End Bearing Response (Station 45) to an Unbalance Placed on the Coupling (Station 50), Original Shaft Configuration.

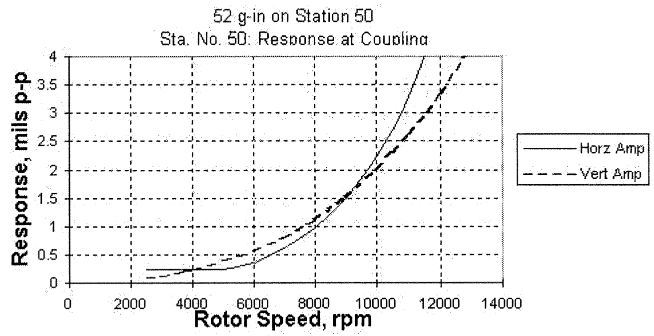

Figure 7. Coupling Plane Response (Station 50) to an Unbalance Placed on the Coupling (Station 50), Original Shaft Configuration.

increased the shaft stiffness, thereby reducing shaft deflection due to unbalance. However, increased shaft diameter means increased surface speed and increased bearing temperatures. Three changes were made to the bearing design to lower the peak temperature. First, the bearing oil flow was increased by approximately 20 percent. Second, the pad surface pressure was decreased by switching from a five pad load-on-pad configuration to a five pad load-between-pad configuration. Third, the bearing pads were changed from a central pivot to a 0.6 offset pivot, enabling better oil flow to each pad. The predicted bearing temperature thus dropped from 225°F to 175°F.

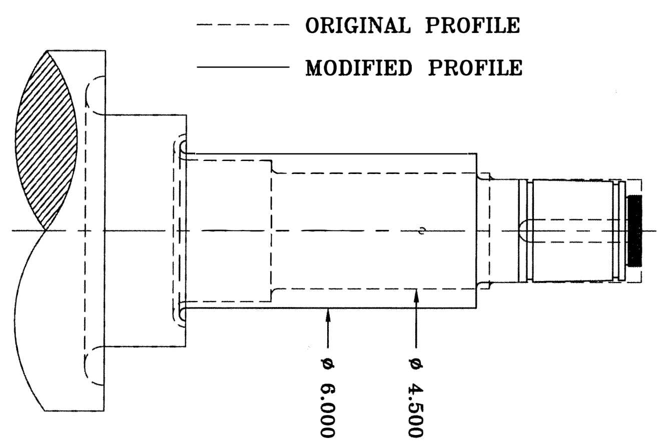

Figure 8. The Original and Final Modified Shaft Profiles.

Shown in Figure 9 is the third mode, after the shaft and bearing modification. The mode is still similar in shape to the original mode shape shown in Figure 5. However, the difference is that the mode now has a natural frequency of approximately 22,500 cpm, compared to 15,746 cpm before shaft modification. This represents an increase of approximately 43 percent in the natural frequency, resulting in further separation of the undesirable mode from the operating speed of the rotor. Shown in Figures 10 and 11 are the responses at the coupling end radial bearing and at the coupling, respectively, again with an 8 W/N unbalance on the coupling. A comparison of Figures 7 and 11 reveals that the predicted original response was approximately three times the response predicted for the modified shaft. The shaft modification had no impact on the first two modes.

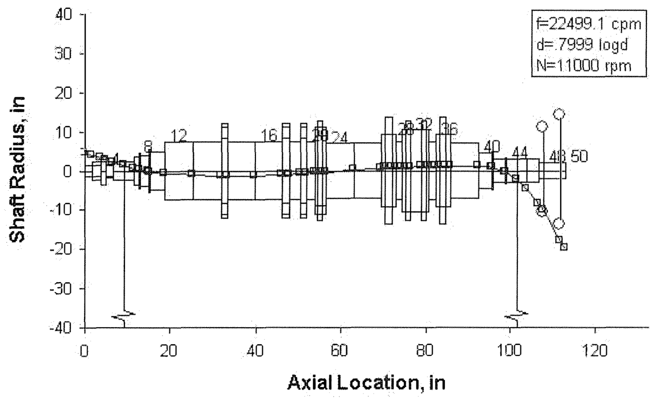

Figure 9. Third Forward Precessed Flexible Mode—Final Modified Shaft Configuration.

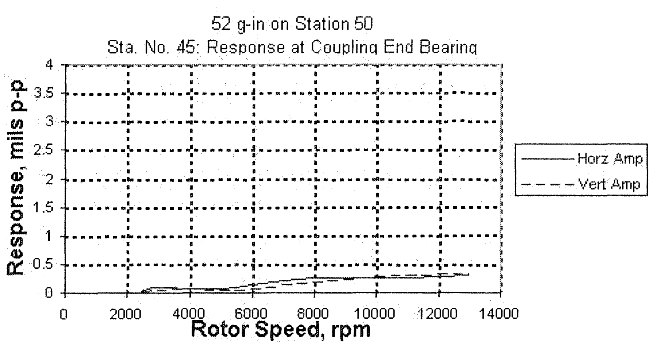

Figure 10. Coupling End Bearing Response (Station 45) to an Unbalance Placed on the Coupling (Station 50), Final Modified Shaft Configuration.

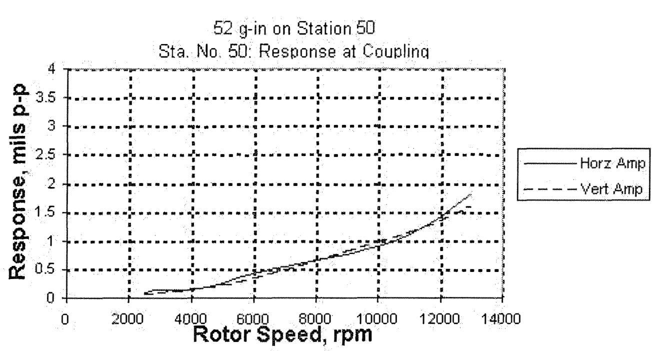

Figure 11. Coupling Plane Response (Station 50) to an Unbalance Placed on the Coupling (Station 50), Final Modified Shaft Configuration.

Even though the analysis revealed that the response of the modified shaft was one third the original shaft, the rotor was still fairly sensitive to coupling plane unbalance. It was thus decided to high speed balance the rotor and add an active balancer just inboard of the coupling, to trim balance the rotor onsite if necessary.

ROTOR SHAFT MODIFICATION

As mentioned earlier, the rotor was of an integrally forged design, so only two options were available for the modification of the shaft. The first option was to manufacture a complete new rotor with the modified design. This was prohibitive from a cost and time standpoint. The second option was to build up the shaft surface with weld, and machine it down to the modified dimensions. This repair technique, when well engineered, produces excellent results (Pardivala and McLaughlin, 1996), and gives the design engineer a great flexibility at the fraction of the cost of a new rotor. Rotor welding techniques have thus started to gain wide acceptance in the repair industry. The rotor material was ASTM A470 Class 8 steel, which is readily weldable. Some of the results of the weld qualification tests conducted on this material can be seen in Table 1. There was no significant variation in hardness through the base metal, the heat affected zone, and the weld overlay. All of the other mechanical properties of the weld overlay are also comparable to the mechanical properties of the base metal.

Table 1. Weld Qualification Tensile Test Results after Stress Relieving.

Weld Qualification - Tensile Test Results					
Test	Location of Test	Yield Strength	Ultimate	% Elongation	% Reduction in
1	Weld metal	78.9	105.0	26.5	64.7
2	Weld metal	81.5	106.5	25.2	65.8
3	Weld metal	78.8	105.5	26.3	64.6
4	Base metal	90.0	118.8	20.0	50.8
5	Base metal	91.1	119.7	18.9	53.2
6	Base metal	90.8	118.2	19.3	52.9

Figure 12 is a view of the shaft, after it was machined down in preparation for weld buildup. A semiautomatic submerged arc welding process was used to build up the shaft, and weld parameters were controlled throughout the process to closely duplicate those used during the qualification tests. After the weld buildup was completed, the shaft was rough machined and subjected to thorough magnetic particle and ultrasonic inspection. Stringent acceptance criteria were used to ascertain the quality of the weld. The shaft was then wrapped and vertically stress relieved. Surface hardness values were recorded and were within the acceptable limits. The welded area was then subjected to another magnetic particle and ultrasonic inspection. After passing the inspections, the shaft was final machined to size, and a final magnetic particle inspection was conducted. Shown in Figure 13 is the completed rotor. A closeup of the welded shaft end is shown in Figure 14.

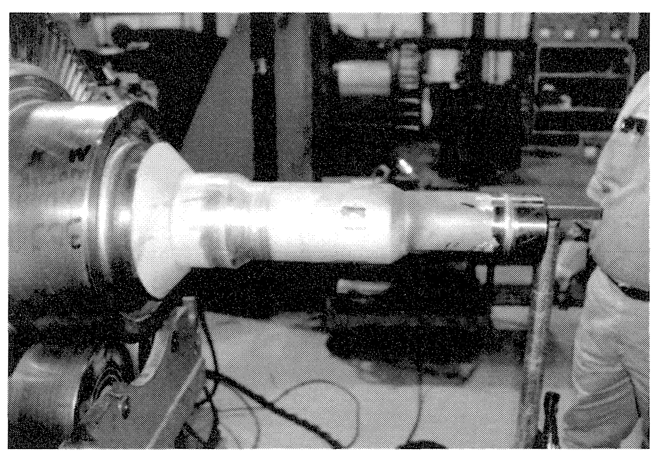

Figure 12. The Shaft End, Machined Down for the Weld Overlay.

APPLICATION OF THE ACTIVE BALANCER

As the turbine response was predicted to remain fairly sensitive to coupling unbalance, a serious consideration was given to active

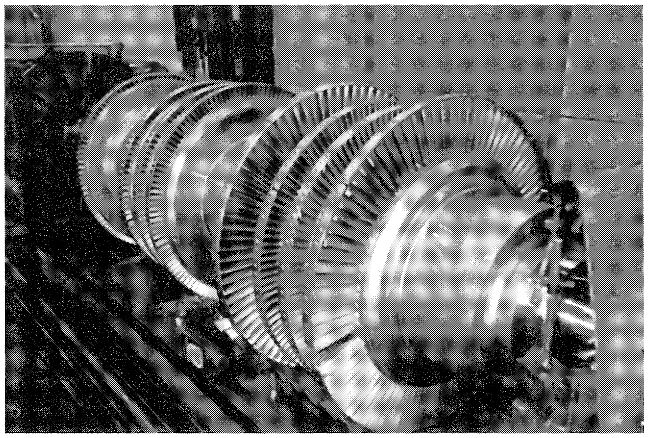

Figure 13. The Completed Rotor after Welding.

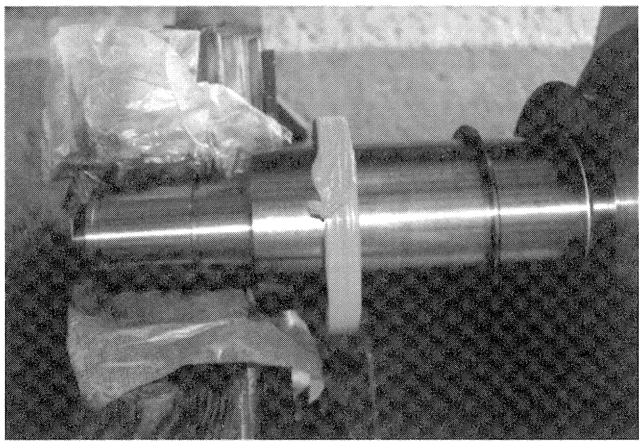

Figure 14. Closeup of the Welded Shaft End.

balancing technology to enhance field reliability. Active balancing is a method of controlling synchronous vibration by actively changing the balance of the rotor as it continues to operate.

Active balancing differs from conventional active vibration control in that the controlling forces are generated in the rotating reference frame. Approaches to vibration control using magnetic bearings or piezoelectric elements are well documented (Palazzolo, et al., 1991; Lum, et al., 1995). These types of force actuators generate forces in the stationary frame and must have a control bandwidth at least as fast as the rotational speed. These "stationary" frame control methods can be used to control either rotor deflections or transmitted bearing forces, but typically not both simultaneously. Active balancing, by compensating for the unbalance excitation source, has the potential to simultaneously control rotor deflections and transmitted forces.

Active balancing has been utilized on precision grinding machines for decades, to reduce vibration and improve machine life and manufacturing precision (Koepfer, 1996). Several methods of moving balance weights on a rotating shaft have been used in industry. Some of these balance weight actuators are DC servomotors, water jets, dense fluid chambers, pneumatic actuators, thermal elements, etc. (Caruso and Hoffman, 1964; Chen, et al., 1994; Koepfer, 1996; Gosiewski, 1985; Zumbach, et al., 1992). A key challenge affecting reliability of such devices is the method used to transfer operating power from the stationary reference frame onto the rotating shaft. Slip rings and other mechanical contacting methods wear out and must be replaced often. One alternate approach is a rotary transformer that passes power and data inductively across an air gap (Hackett and McCleer, 1993).

Initially, a dense fluid transfer balancing device was considered for application to the steam turbine. After further analysis, the high rotational speed (10,800 rpm) and relatively high temperature of the environment ruled out this technology because of stress and electronic reliability issues. However, another innovative balancing device had recently been developed for use in aircraft gas turbine engines (U.S. Air Force, 1994) and high speed machining spindles. This new technology was chosen subsequently for application to the syn gas compressor train.

The patented (Dyer, 1997) electromagnetic device incorporates two counterweighted rotors, as shown in Figure 15. The balance rotors can be moved to any desired position relative to a rotating shaft, in less than a second, using permanent magnet actuators. The rotating portion of the device is fixed to the rotating shaft and contains the moveable balance counterweight rotors. These dual counterweight rotors attach to the balancer housing with thin section ball bearings. The rotating balancer is surrounded by a stationary coil assembly, and separated from the rotating assembly by a radial air gap of about 0.030 inch. The stationary electrical coils pass actuation power across the air gap, in the form of a magnetic field. When no change in balance correction is required, the device remains passive. Permanent magnets allow the counterweight rotors to be locked in place with no power. Hall effect sensors mounted to the stationary coil housings, detect the passing of magnetic targets fixed to rotating components, to facilitate sensing of the shaft speed and phase and balance rotor positions. The control system receives vibration signal inputs from shaft proximity probes. This "noncontacting" device had been tested on spindles operating at up to 15,000 rpm, and showed much promise for application to all types and speeds of high speed rotating machinery.

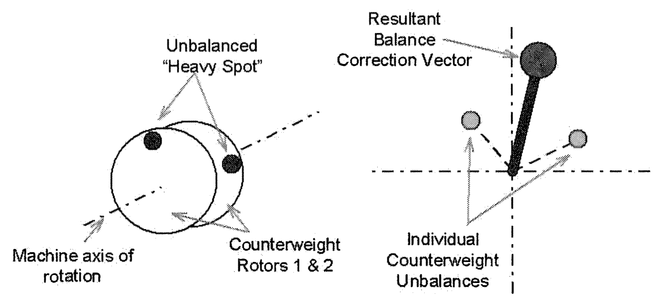

Figure 15. Schematic of Balancer with Two Counterweighted Rotors.

Locating a balancing device at the coupling plane would theoretically allow for the cancellation of the coupling assembly unbalance excitation to which the turbine rotor was fairly sensitive. Furthermore, previous authors (Bishop and Parkinson, 1972) have presented the argument that a single active balancing plane would suffice for many rotors. This argument was based on two assumptions. First, the critical speeds must be sufficiently separated to ensure that only a single mode dominates response near any given critical speed, and second, the active balance plane must not lie at a node for any of the dominant mode shapes. Vibration at speeds between critical speeds was also assumed to be acceptably low. This single plane approach was later verified for relatively small-scale rotors in laboratory conditions (Lee and Kim, 1987; Lee, et al., 1990). However, the presence of structural damping, anisotropic support stiffnesses, etc., could potentially lead to multiple modes, contributing significantly to dynamic response at any given speed. Such conditions could potentially lessen the effectiveness of single plane active balancing, especially if the residual unbalance was not located in the same plane as the active balancer.

BALANCER TESTING IN AN OPERATING SPEED BALANCE BUNKER

As this would be the first known active balancing installation on industrial turbomachinery, initial tests of the active balancer were conducted in an operating speed balancing facility. Balancing integrally forged rotors after extensive repairs always poses a challenge, because the exact locations and magnitudes of unbalance are never known. Multiplane high speed balancing of the turbine was therefore necessary for satisfactory performance, and to ensure that the balancer was only called on to trim balance variations at the coupling in the actual installation. Thus, the turbine was permanently balanced for all speeds up to operating speed prior to installing the active balancer with the new bearings. Figure 16 shows the bearing pedestal velocities after the high speed balancing to be well within specification. The rotor deflection was also minimized all the way to operating speed. This also provided a good baseline for testing the active balancer. As expected, the frequencies of the modes in the balance bunker were lower than predicted, and lower than that seen in the field. This was due to the lower pedestal stiffness used during high speed balancing. Figure 17 shows the final bearing pedestal velocities with pedestal stiffeners turned on, which better simulates the operation of the rotor within the casing.

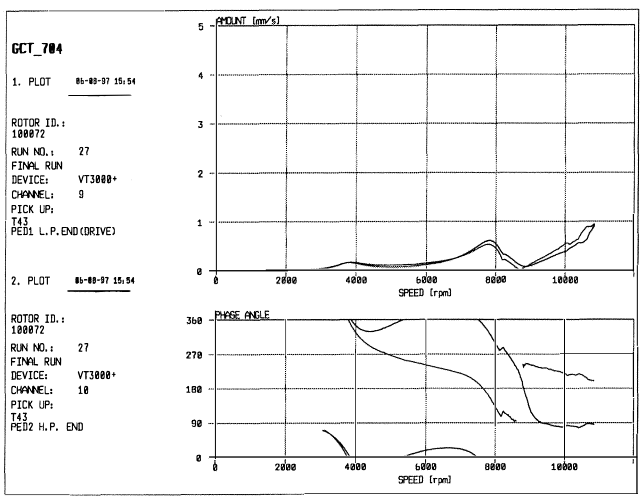

Figure 16. Bearing Pedestal Velocities after High Speed Balancing.

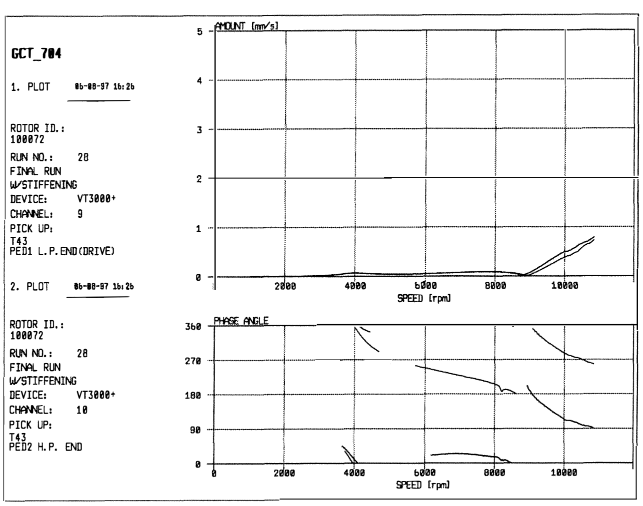

Figure 17. Bearing Pedestal Velocities after High Speed Balancing—With the Pedestal Stiffeners Turned On.

The balancer was then mounted on the drive coupling hub of the redesigned steam turbine. A combination of shaft proximity probes and bearing pedestal velocity sensors were used to monitor the performance of the active balancing system in various unbalance tests. A schematic of the test setup with sensor and unbalance locations is shown in Figure 18.

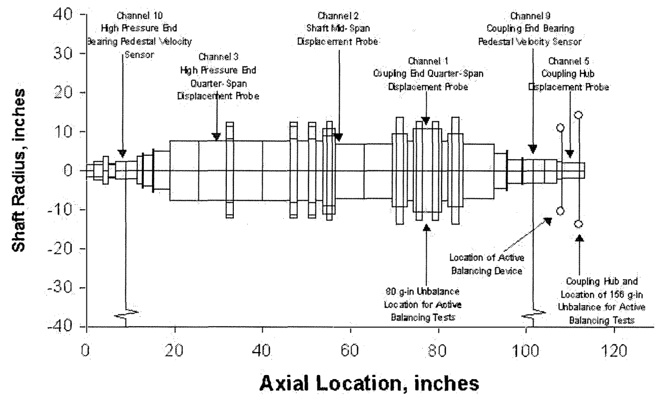

Figure 18. Schematic of the Test Setup.

The turbine rotor was then run up to its maximum continuous operating speed with the balancer, and vibration at all sensors recorded. These data were used to verify the effect of added overhung mass and the residual unbalance contained in the active balancer. The balancer was then exercised at various turbine speeds up to the operating speed of the turbine, and the resulting changes in vibration were recorded to calculate the influence coefficients of the balancer. This test was also used to confirm proper operation of the balancer and control system at all turbine operating speeds. An unbalance of 156 gram-inches was then placed on the coupling hub, as shown in Figure 18, to represent a potential coupling assembly unbalance. The magnitude of the unbalance, however, was much greater than the turbine would ever experience with an API 671 (1990) coupling. The turbine was then run at various speeds up to its operating speed, and the vibration was measured before and after active balancing at these speeds. The capacity of the balancer used in this test was 255 gram-inches. This value was chosen based on balance actuator limitations, desired balance resolution, and the sensitivity of the rotor to unbalance in the coupling plane. The balancer was seen to be able to easily correct for the coupling unbalance added. The balancer was also given full automatic control at rotor speeds of 3500 rpm to 7000 rpm. Figure 19 is a two-dimensional projection of the measured deflected shape of the rotor at two of the active balancing speeds. The results shown indicate that at these speeds, active balancing was able to reduce rotor deflection significantly.

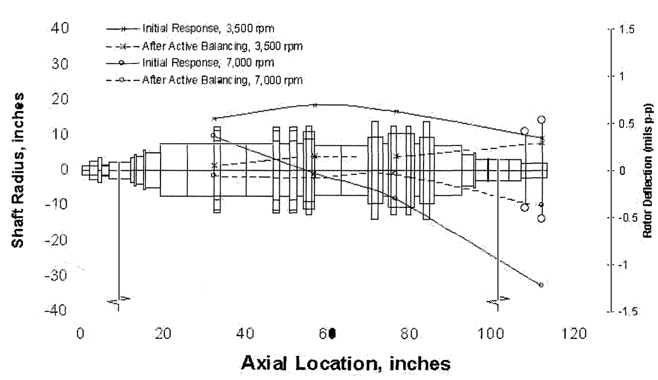

Figure 19. Two-Dimensional Projection of the Rotor Deflected Shapes Before and After Active Balancing.

The controller utilized an adaptive control law (Dyer, 1997) based on the well-known influence coefficient method (Lund and Tonnesen, 1972). In general, the controller used an initial guess of the influence coefficient to command the balance rotors toward the optimal positions, based on the measured vibration. With each subsequent movement of the correction weights, the vibration was again measured and the influence coefficients reestimated, using a weighted averaging scheme. The adaptive control system performed successfully in the tests, regardless of the accuracy of the initial influence coefficient guess. Each balance cycle required anywhere from one to three seconds, depending on initial conditions. At rotational speeds above 7000 rpm, with higher unbalance sensitivity, a more conservative approach was taken during the active balancing tests, and the balancer was controlled manually, using influence coefficient estimates and trial and error fine-tuning.

The coupling hub unbalance was then removed and an unbalance of 80 gram-inches was placed at the quarterspan of the turbine. The turbine was again rotated at various speeds, and vibration measured before and after active balancing. Figure 20 shows the results of the active balancing at the various speeds. The results indicated that the active balancer was able to reduce vibration on both ends at the certain speeds, but at operating speed when the thrust end vibration decreased, the coupling end vibration increased. This was the result of attempting to compensate for an unbalance located some distance away from the correction plane. After testing, the turbine was assembled and shipped to a major ammonia production facility. Figure 21 shows the assembled turbine prior to shipment, and Figure 22 shows the coupling end of the turbine with the coupling hub and balancer installed.

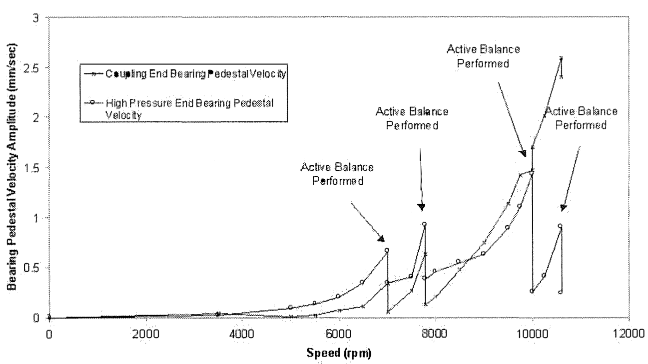

Figure 20. Results of the Active Balancing with the Quarterspan Unbalance.

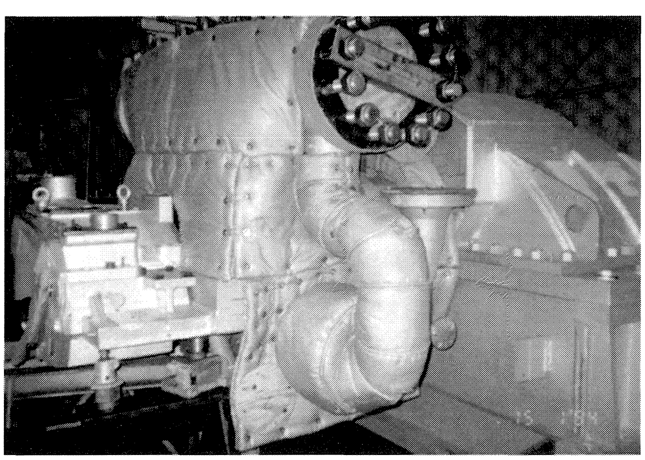

Figure 21. Assembled Turbine Prior to Shipment.

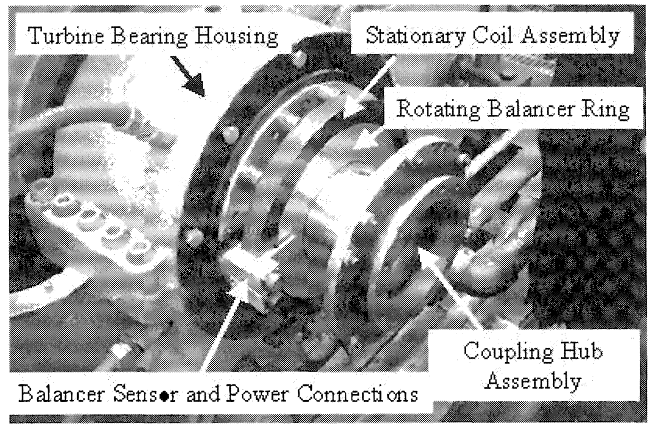

Figure 22. Turbine Coupling End, with the Coupling Hub and Balancer Installed.

FIELD INSTALLATION

The steam turbine drives two compressors in a syn gas train. Even though a flexible diaphragm coupling was used between the turbine and the low pressure compressor, a second active balancer was installed on the compressor coupling hub, to mitigate any cross-influence of the turbine balancer through the coupling. The compressor active balancer had a maximum balance correction capacity of 113 gram-inches. A host controller was developed to coordinate the two balancing devices and accomplish true two-plane balancing for the coupling. Figure 23 is a system schematic for the installation. Power amplifiers for the balancing devices were mounted on the compressor deck adjacent to the machines, to minimize the power cable run. All control instrumentation was placed in the process control room in a single rack. Each balancer was linked to its own controller and both controllers communicated with host control software running on a personal computer. The mechanical layout of the balancing devices on each machine is shown in Figure 24, with the balancer assemblies contained inside the coupling guard. Balancer power and sensor cables were routed through the bottom half of the coupling guard through sealed flanges. To minimize the potential for oil leakage, the flanges were located near the upper edge of the guard. Existing shaft proximity probes located just outboard of each bearing were utilized to provide vibration signal feedback for the active balancing control system.

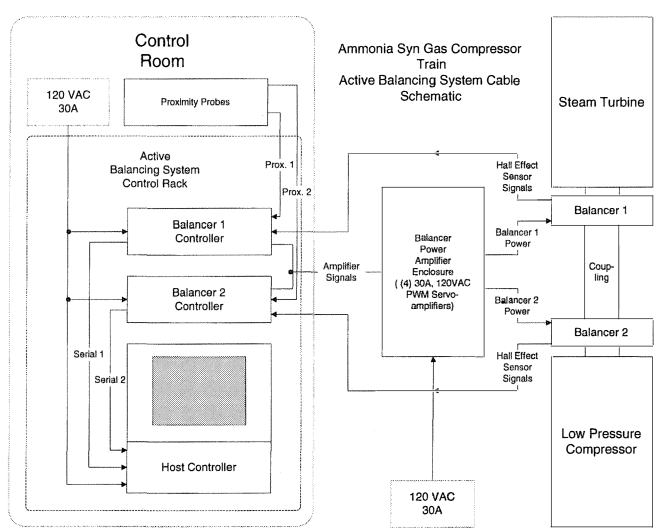

Figure 23. System Schematic for the Balancer Installation.

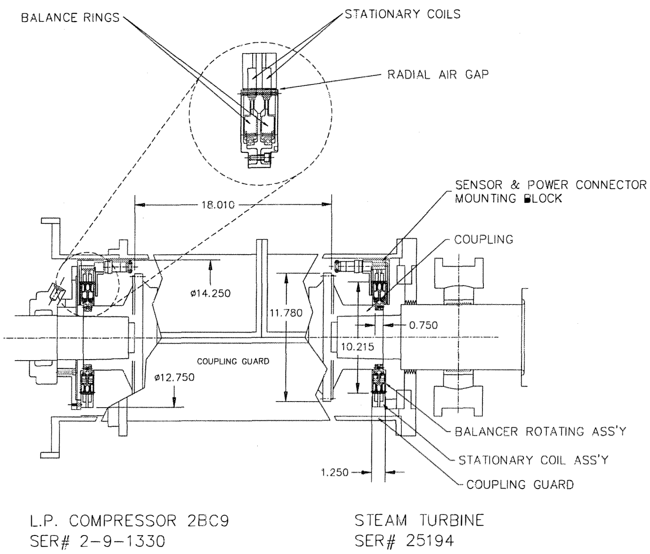

Figure 24. Mechanical Layout of the Balancing Devices on Each Machine.

FIELD RESULTS

Slow roll shaft runout values were acquired at 500 rpm and stored in the balancer controller. Upon startup, vibrations on the turbine thrust end were slightly high. This was probably due to the residual unbalance in the coupling assembly and the balancer. After reaching operating speed, the active balancing system was exercised, and the influence coefficients obtained, using each balancer and adjacent vibration probes. The host controller software allowed these influence coefficients to be computed automatically under operator control. The measured influence values indicated that the turbine balancer could effect approximately 0.67 mils peak-to-peak and 2.10 mils peak-to-peak of vibration at the coupling end and thrust end probes, respectively. This cross effect qualitatively matched the response observed during turbine testing in the high speed balance bunker.

After first acquiring influence coefficient estimates as discussed previously, the host control software then displayed estimates of the residual unbalance, corresponding to each active balance plane's influence. These estimates were then used to manually command the balance weights to compensating positions. Though the controller possessed the capability to perform automatic two-plane balancing on the coupling, the more conservative manual approach was taken to minimize any risk of potential hazards in this first installation. After moving the balance weights to the indicated compensating positions, the vibration responses at the probes were further minimized through fine-tuning. The results of the active balancing are shown in Figure 25. As can be seen, significant improvement was possible, especially on the thrust end.

RESULTS AND CONCLUSIONS

Since the installation and commissioning of the syn gas train, the turbine has operated successfully with the lowest levels of vibration in its history. Radial bearing temperatures, with the redesigned bearings, dropped from 225°F to 175°F, improving reliability and extending bearing life. From the high speed balance runs and the field data, it can be clearly seen that the shaft modifications were very successful. The unbalance response at the coupling was reduced by as much as 60 percent with the same coupling unbalance, thereby significantly improving the reliability of the rotor. The results also highlight the advantages of advanced shaft welding techniques, where a rotor design modification can be made at a fraction of the cost and time of manufacturing a new rotor.

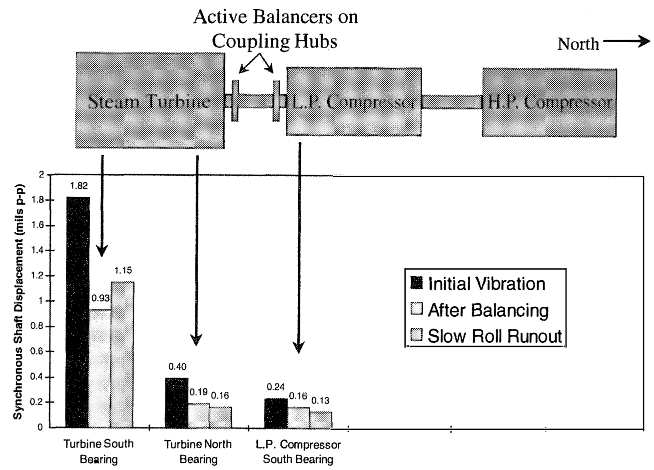

Figure 25. Field Results of the Active Balancing.

The performance of the active balancer has shown that such a system, when correctly applied, offers exceptional flexibility in balancing rotors online, providing significant savings at startup when troublesome machines need to be manually trim balanced. The system was highly effective in controlling synchronous vibration due to coupling unbalance, and to some extent quarterspan unbalance. The balancer has also proven capable of operating at high speeds in demanding industrial environments, up to 10,800 rpm and 350°F, indicating the viability of this technology.

The techniques presented in this paper have a wide range of application in modern turbomachinery repair, and can be successfully used to improve the reliability of other turbomachines with similar problems.

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