TURBINE FAILURE AND RECONSTRUCTION

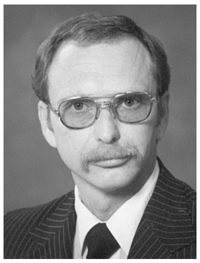
by

Charles. R. Rutan

Senior Engineering Advisor, Specialty Engineering

Lyondell Chemical Company

Alvin, Texas



Charles R. (Charlie) Rutan is Senior Engineering Advisor, Specialty Engineering, with Lyondell Chemical Company, in Alvin, Texas. His expertise is in the field of rotating equipment, hot tapping/plugging, and special problem resolution. He has three patents and has consulted on turbomachinery, hot tapping, and plugging problems all over the world in chemical, petrochemical, power generation, and polymer facilities.

Mr. Rutan received his B.S. degree (Mechanical Engineering, 1973) from Texas Tech University. He is a member of the Advisory Committee of the Turbomachinery Symposium, and has published and/or presented many articles.

ABSTRACT

This paper is an extended case history of a turbine catastrophic failure due to stress corrosion and the reconstruction of the turbine, compressor, gearbox, and the drive motor after a fire. Wheel stresses and the design upgrades to minimize the potential for failure while reducing the subsequent damage if a failure occurred of the complete system will be discussed.

INTRODUCTION

A Lyondell manufacturing facility has a recycle gas compressor that continuously recycles ethylene and ethylene oxide to the reactors. On December 28, 1996, the drive turbine of the compressor machine train catastrophically failed.

General Description

The equipment consists of a compressor driven at one end by a steam turbine and at the other end by a motor through a speed-increasing gear. The driving motor, gear, compressor, and turbine are mounted on a common bedplate. The compressor train is equipped with nonlubricated diaphragm couplings, water seal system, control and lube oil systems, and the accessories necessary for the safe and efficient operation of the unit. Design data can be found in Table 1.

The operation of the compressor train is somewhat unique for a chemical plant. The turbine is started and once it reaches 6000 rpm, the motor is started and the compressor train comes up to 8676 rpm. Figures 1 and 2 show the layout of the compressor platform.

During the summer, at peak energy consumption, the turbine is used to augment power when electrical power shedding occurs for the local electrical grid. This process lets steam down to a level required by the process and uses the steam developed in the process reactors.

A major overhaul and an uprate of the turbomachinery train had been performed in the spring of 1996. The new motor, couplings, and gearbox were installed at this time. However the turbine, compressor, and foundation did not require modifications as the uprate

Table 1. Design Data.

Turbine	
Number of stages	3
Rating, bhp (kW)	5050 (3768)
Speed, rpm	Original 7680, New 8676
Maximum continuous speed, rpm	Original 7757, New 8676
Tripping speed, rpm	9000
Normal inlet pressure, psig	760
Normal inlet steam temperature, °F	626
Normal exhaust pressure, psig	250
Wheel diameters	22.616
First rotor response speed (critical), rpm	4700
Rotor material	ASTM A470 Class 8
Rim welds for the first and second wheels	F8
Rim welds for the third wheel	F7
Bucket material	Original #1 AISI 422 SS, New TG-410AB001
Compressor	
Number of stages	1
Speed, Rpm	8676
Gas handled	Recycle mixture
Molecular weight	25.79
K	1.305
Z	0.964
Inlet pressure, psia	235
Inlet temperature, °F	85
Discharge pressure, psia	317
Discharge temperature, °F	135.5
Rating, bhp	13,390
Gear	
Type	Double helical, speed increaser
Rating, bhp	17,285
Input speed, rpm	1793
Output speed, rpm	8676
Gear ratio	1:4.8388
Service factor	1.4
Motor	
Type	Induction, premium efficiency
Rating, bhp	15,030
Speed, rpm	1793
Phase	3
Volts	12,470
Efficiency, %	97.3
Hertz, cycles per second	60
Service factor	1.25

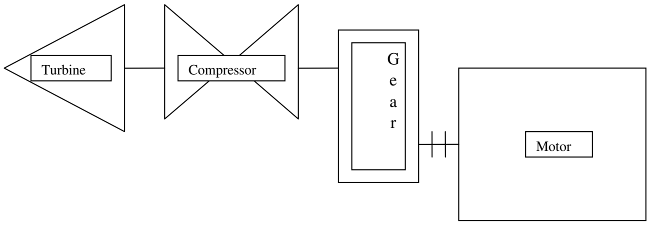


Figure 1. Compressor Platform Layout Plan View.

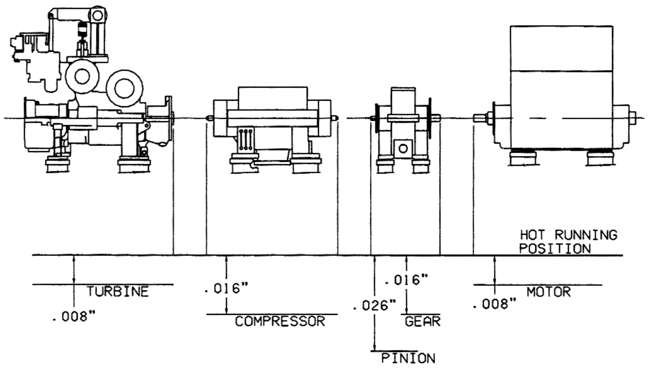


Figure 2. Compressor Platform Layout.

was a speed increased from 7757 rpm to 8676 rpm and an increase in horsepower occurred to meet the new load requirements of the compressor. The compressor manufacturer completed a thorough dynamic and torsional analysis and found no issues. The turbine and compressor rotors were removed, inspected, and refurbished locally. Inspection included:

- A thorough cleaning.
- Visual inspection.
- Dimensional verification.
- Magnetic particle
  - Shafts.
  - Integral wheels.
  - Buckets.
  - Shroud bands.
  - Tenons.
  - Impellers.
- Complete electrical runouts.
- Complete mechanical runouts.
- Incoming low speed balance.

Minor wear of the turbine and compressor was reported, which required some handwork. The rotors were low-speed balanced, then at-speed balanced, and returned to the facility where they were installed in their respective cases. The nozzles, diaphragms, packing boxes, etc., were similarly inspected with the same result. New packing and bearings were installed. The plant had most of the normal spare parts including a new spare compressor rotor. Plant management did not see the value in a spare turbine rotor, spare gears, or a spare motor. The vibration monitoring system and turbine controls (completely manual) were not upgraded. The time limitation for the installation was the time it took to dump the reactor catalyst and reload new catalyst.

Due to the process safety concerns the facility had, as per the standard operating procedure (SOP), all of the automatic shutdowns of the entire compressor train were bypassed:

- Vibration
- Thrust
- Low lube oil pressure
- Low lube oil level
- Overspeed
- Low sealing water flow
- Low sealing water pressure
- Low sealing water flow

With the exception of the motor electrical overloads located in the motor control center, all the instrumentation was locally monitored and controlled from a foul weather building located on the compressor platform.

When the turbine failed (Figures 3, 4, 5, and 6), the inboard bearing cover split into two pieces (Figure 7) and the outboard bearing cover rolled to one side of the turbine (Figure 8). Oil began spewing out all over the platform and the atomized oil ignited. The ignition source was the molten metal caused by the heat generated from the metal-to-metal contact rub in the bearing areas. The platform was engulfed in flames burning all the instrumentation, electrical wiring, and the foul weather building. The conduit melted and the electrical insulation melted away exposing the wire that came in contact with other wiring and the metal structure (Figure 9). At this point, the production operator arrived on the seen and was about to turn on the firewater monitor, located at grade level on the platform. However, when he saw all the electrical arcing he decided that it would be unsafe and elected to let the fire burn itself out, which occurred when all the lube oil had been consumed. This was not a good idea. The compressor mechanical water end seals did not fail and the trip throttle valves located at the inlet and inner stage admission did close, but the motor continued to turn at-speed until the motor current overloads tripped.

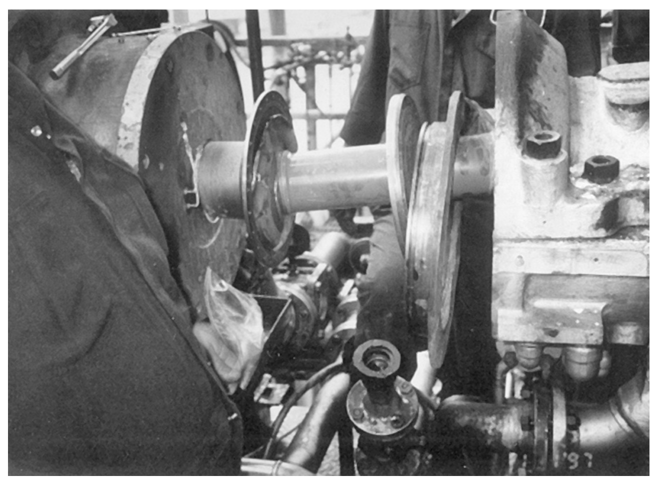


Figure 3. Inboard Compressor Turbine.

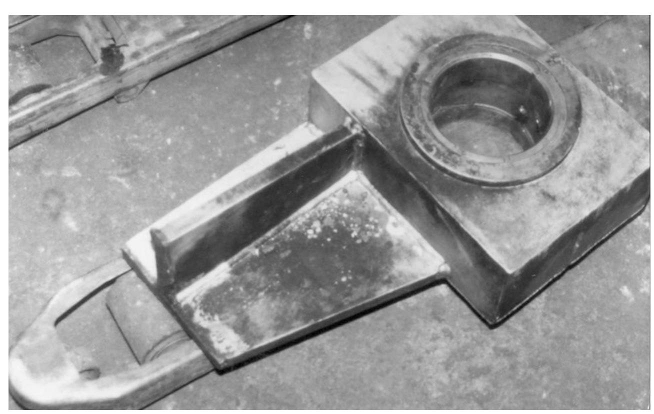


Figure 4. Valve Rack Support.

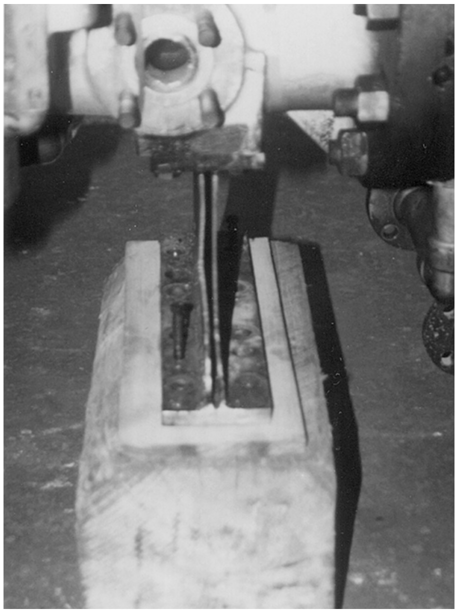


Figure 5. Turbine Outboard Wobble Foot.

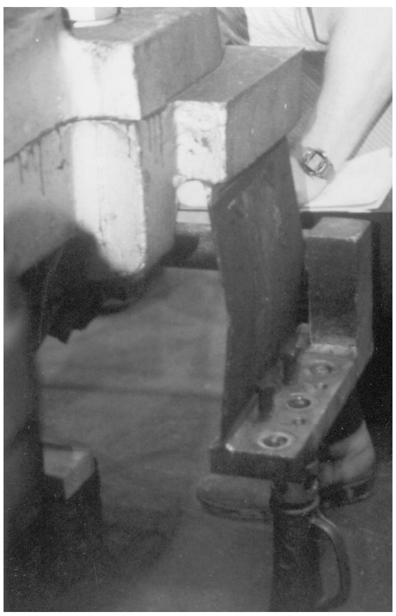


Figure 6. Exhaust End Wobble Foot.

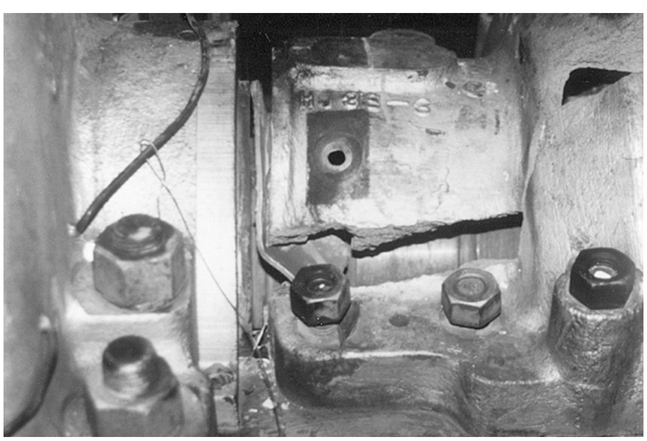


Figure 7. Turbine Inboard Bearing Cap Split.

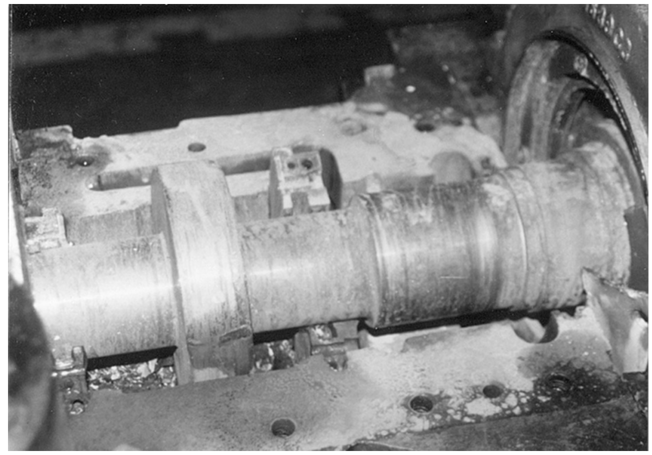


Figure 8. Turbine Outboard Bearing Cap.

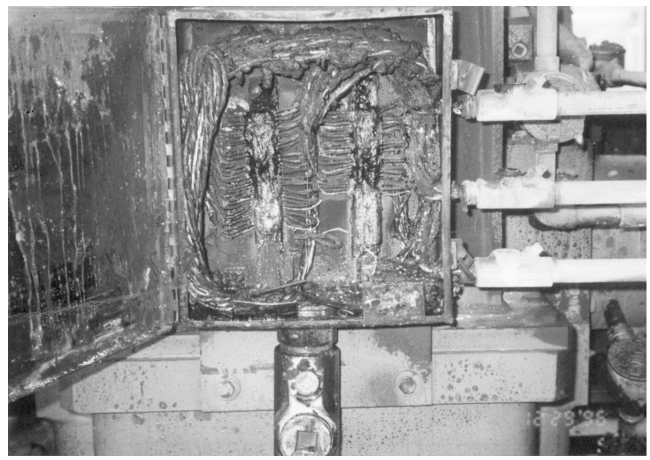


Figure 9. Typical Electrical Instrumentation Junction Box.

The turbine was supplied with steam from two sources. The inlet or head end steam was purchased from another chemical company that has a plant located near this plant. Low-pressure steam was generated in the unit reactors, and then introduced between the second and third stages. At this location inside the turbine the stationary nozzles are located in the upper half and a diaphragm in the lower half of the case. As the incident investigation progressed, a notification of a caustic excursion in the boiler feedwater makeup had occurred on December 24, 1996. The source was the company that supplied the high-pressure inlet steam to the turbine. No additional information was supplied at that time. Operations personnel took a condensate sample and found that it had a 13 pH. Supervision did not realize that a pH of 13 was a serious issue and could cause stress corrosion cracking of steels. The steam supplier had sent, by mail, a formal notification of the caustic excursion to the plant manager, but he did not receive it until December 30, 1996. This notification was too late by six days.

On the morning of January 3, 1997, the compressor platform was released to maintenance to begin the job of damage assessment and repair. Structural consultants, piping experts, instrumentation, electrical, piping, and rotating equipment engineers were brought in to see what could be salvaged and what items had to be replaced. The motor, gearbox, and compressor case (with rotor) were returned to the original manufacturers for inspection. The compressor rotor, turbine end, was severely damaged (Figure 10).

Bearing journal, labyrinth oil seals, and coupling areas would require submerge arc welding and machining to bring these areas back to the original specified tolerances. Additionally, the rotor had

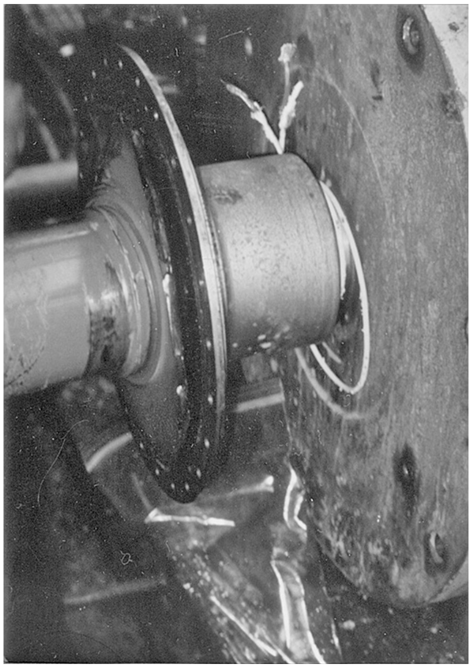


Figure 10. Compressor End.

a 5 mil kink, turbine side, starting at the compressor end seal area extending to the shaft end. A decision was made that no effort would be made to repair the compressor shaft at this time because a new spare compressor rotor was available; so all the attention was focused on the turbine repairs.

The motor sustained minimal damage to the bearing journal and oil seal areas. After the repair of the journal areas and the installation of a new bearing, the motor was tested and returned to the plant with a fresh coat of paint.

Minor damage was found upon inspection of the gear. The bearing journal areas were hand dressed. The gear teeth required regrinding and new bearings were installed. The gearbox was then tested and returned to the plant with a fresh coat of paint.

While the above was transpiring, the decision was made that repairing the turbine rotor was more expedient than waiting for a new forging. A detailed list of all the components that could be repaired, replaced with stock inventory, and fabricated was developed. An outside consultant was contacted to determine what caused the turbine failure. The first two wheels suffered severe damage. It was learned that one complete segment of the third wheel was missing (Figures 11, 12 and 13).

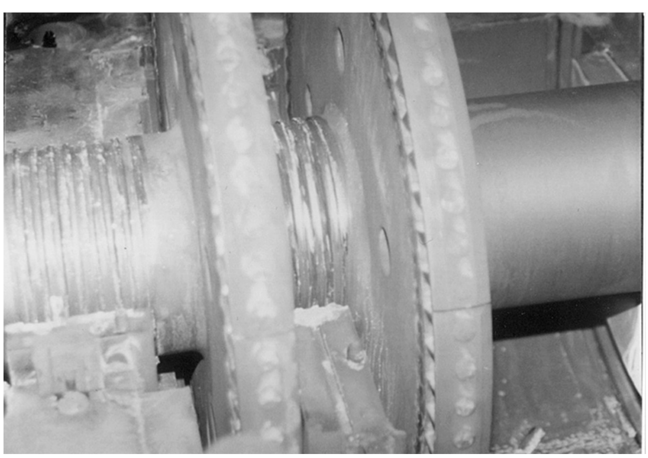


Figure 11. Turbine First and Second Stages.

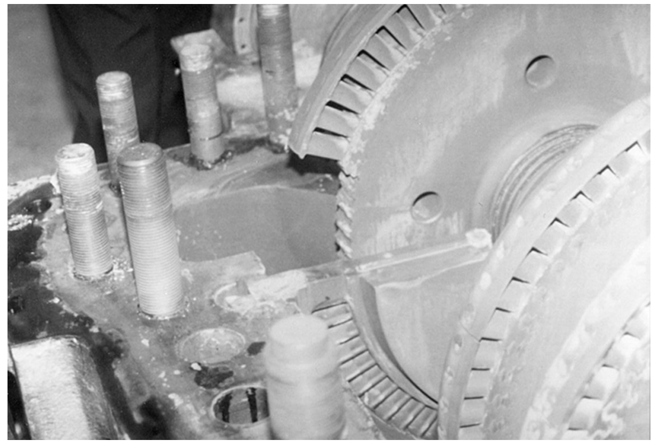


Figure 12. Turbine Third Stage.

Figure 13. Third Wheel.

When the remainder of the turbine rotor was magnetic particle inspected the Christmas trees (bucket tenons) of the third wheel looked like they had grown hair (Figure 14). The cracks were sectioned and inspected. Caustic was present and the wheel had failed due to stress corrosion cracking. A section of the wheel, identical to the section that had failed (an eight-bucket packet), was removed and weighed. A quick calculation with this weight, radius from the center of the rotor and speed, was performed. It was determined that when the section came off the rotor, the turbine experienced an unbalanced force of 160,000 lb. If any of the shutdowns had been activated, and the compressor train had automatically been shut down, then the damage would have been bad, but not catastrophic. Fortunately, the compressor mechanical water end seals did not leak and remained intact. If these seals had failed and the ethylene oxide gas had been released to the atmosphere, there was a very high probability that an additional event would have occurred.

STEAM CONTAMINATION

Stress corrosion cracking is one failure mechanism that has been well documented. Please refer to the attached list of references at the end of the paper for more information. Turbine manufacturers are all in agreement that the steam purity must be maintained at the lowest practical level of contaminants. It should not exceed 3.0 ppb Na, cation conductivity of 0.2 μmho/cm. Total suspended solid shall not exceed 0.1 ppm (100 ppb) and pH 8.0 to 10.0 (inclusive) during normal operation. The manufacturers' recommendations are intended to limit sodium compounds, such as caustic (NaOH) and sodium chloride (NaCl). One manufacturer has published the following limits during periods of abnormal operation: for short

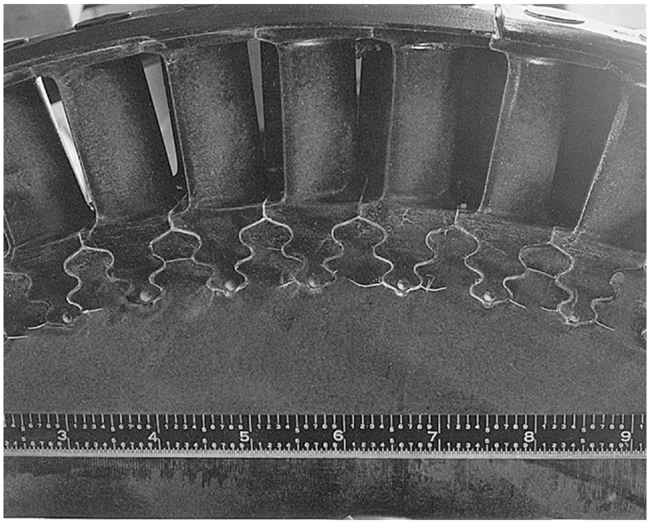


Figure 14. Third Stage Sectioned.

periods, not to exceed 100 hours per incident and accumulating 500 hours or less in a 12 month operating time, 6.0 ppb Na and 0.5 μmho/cm should not be exceeded. During emergency conditions, for periods of 24 hours or less with accumulations not exceeding 100 hours in a 12 month operating time, 10.0 ppb and cation conductivity of 1.0 μmho/cm should not be exceeded. The plant did not, at the time of the incident, have the capability of measuring anything other than the pH of the steam condensate. Just as a reminder, the plant had measured a 13 pH of the steam condensate. This situation was rectified during the period of time it took to rebuild the compressor train. Typical specifications for steam quality are found in Tables 2, 3, 4, and 5.

Table 2. ABMA Guidelines—Boiler Feedwater, Boiler Water, and Steam Specifications Applicable for Steam Drums Operating Between 0 to 300 PSIG.

Total dissolved solids[1]	700-3500 ppm
Total alkalinity[2]	140-700 ppm as $CaCO_3$
Suspended solids	15 ppm (max)
Total dissolved solids[2,3] steam	0.2-1.0 ppm (max expected value)

Notes:
1 Actual values within range reflect the total dissolved solids in feedwater.
2 Actual values within the range are directly proportional to the actual value of total dissolved solids of boiler water.
3 These values are exclusive of silica.

Water/steam related parameters are found in Tables 6, 7, and 8. The overall quality of the combined "fresh" and recoverable condensate (vacuum and/or suspect) makeup ultimately impacts boiler efficiency and operating continuity and steam purity. Poor steam purity can lead to carryover related problems such as steam turbine fouling. Lastly, continuous and intermittent blow-down control can help ensure optimum boiler water cycles of concentration, and reduce unwanted impurities.

The following information covers online sodium analysis and its relationship to total solids/total-dissolved solids.

> "A sodium analyzer is used to determine steam purity by measuring the sodium ion present in steam with the use of specific ion electrodes. Specific sodium ion electrodes (Note: Samples streams must be cooled with a sample cooler to prevent damage to various components of the sodium analyzer) have been developed which are extremely accurate (±2 percent) and reliable. As a result of this and because sodium is the most common ion found in boiler water, it is an excellent indicator of boiler water carryover problems. When using the specific sodium ion technique, the approximate total solids present in steam is calculated by multiplying the sodium ion content by a factor of three. Thus, a sodium ion reading of 0.33 ppm would represent approximately 1.0 ppm of total solids."
>
> "By running steam purity evaluations, determination can be made of the carryover TDS (total dissolved solids) in steam as well as evaluate methods for reducing TDS in the steam by making mechanical or chemical changes in boiler operation. Other applications for sodium analyzers include monitoring demineralizer effluent and condensate." (Drew, 1994)

Table 3. ASME Guidelines—Boiler Feedwater, Boiler Water, and Steam Specifications Applicable for Steam Drums Operating Between 0 to 300 PSIG.

Boiler Feedwater	
Iron	0.100ppm or 100 ppb as Fe
Copper	0.050 ppm or 50 ppb as Cu
Total hardness	0.300 ppm or 300 ppb as $CaCO_3$
pH	Report
Boiler Water	
Silica	150 ppm as SiO_2
Total alkalinity[1]	350 ppm[2] as $CaCO_3$
Specific conductance	3500 mmho
pH	Report

Notes:
1 Minimum level of hydroxide alkalinity in boiler below 1000 psi must be individually specified with regard to silica solubility and other components of internal treatment.
2 Maximum total alkalinity consistent with acceptable steam purity. If necessary, the limitation on total alkalinity should override conductance as the control parameter.

The above parameters and limits should be reviewed with the site water treatment provider.

Table 4. ABMA Guidelines—Boiler Feedwater, Boiler Water, and Steam Specifications Applicable for Steam Drums Operating Between 301 to 450 PSIG.

Total dissolved solids[1]	600-3000 ppm
Total alkalinity[2]	120-600 ppm as $CaCO_3$
Suspended solids	10 ppm
Total dissolved solids (steam)[2,3]	0.2-1.0 ppm

Notes:
1 Actual values within range reflect the total dissolved solids in feedwater.
2 Actual values within the range are directly proportional to the actual value of total dissolved solids of boiler water.
3 These values are exclusive of silica.

Table 5. ASME Guidelines—Boiler Feedwater, Boiler Water, and Steam Specifications Applicable for Steam Drums Operating Between 301 to 450 PSIG.

Boiler Feedwater	
Iron	0.050 ppm or 50 ppb as Fe
Copper	0.025 ppm or 25 ppb as Cu
Total hardness	0.300 ppm or 300 ppb as $CaCO_3$
Boiler Water	
Silica	90 ppm as SiO_2
Total alkalinity[1]	300 ppm[2] as $CaCO_3$
Specific conductance	3000 mmho

Notes:
[1] Minimum level of hydroxide alkalinity in boiler below 1000 psi must be individually specified with regard to silica solubility and other components of internal treatment.
[2] Maximum total alkalinity consistent with acceptable steam purity. If necessary, the limitation on total alkalinity should override conductance as the control parameter.

The above parameters and limits should be reviewed with the site water treatment provider.

Table 6. Water/Steam Related Parameters Specifically for 451 to 600 PSIG Steam Generators (ASME Guidelines Unless Otherwise Stated).

Boiler Feedwater Limits	
Organic	Nondetectable (TC/TOC min. detection ~ 0.5 ppm)
Iron	0.030 ppm or 30 ppb as Fe
Copper	0.020 ppm or 20 ppb as Cu
Total hardness	0.200 ppm or 200 ppb as $CaCO_3$
Dissolved oxygen*	10 ppb as O_2 (w/o oxygen scavenger)
Boiler Water	
Silica**	35 ppm as SiO_2 (ABMA max.)
Total alkalinity	250 ppm as $CaCO_3$
Specific conductance	2500 mmho/cm
Total solids	2500 ppm (ABMA max.)
Suspended solids	100 ppm (ABMA max.)
Steam Purity Limit	
Total dissolved solids	0.2-1.0 ppm (max. expected value)
Silica	0.02-0.03 ppm, or 20-30 ppb as SiO_2 (ABMA limit)

Notes:
* Well-designed and operated deaerators can reduce oxygen to as low as 7 ppb.
** Silica limit based on limiting silica in steam.

ASME guidelines unless otherwise stated.
The above parameters and limits should be reviewed with the site water treatment provider.

Comment: It is assumed that the author considers total solids and total dissolved solids as being synonymous. Therefore, if the TDS in steam concentration (ppm) is determined analytically, or taken from ASME/ABMA steam purity guidelines, the theoretical sodium concentration can be determined by multiplying the TDS concentration by 0.33.

Table 7. Water/Steam Related Parameters Specifically for 901 to 1000 PSIG Steam Generators (ASME Guidelines Unless Otherwise Stated).

Boiler Feedwater Limits	
Organic	Nondetectable (TC/TOC min. detection ~ 0.5 ppm)
Iron	0.020 ppm or 20 ppb as Fe
Copper	0.015 ppm or 15 ppb as Cu
Total hardness	0.050 ppm or 50 ppb as $CaCO_3$
Dissolved oxygen*	10 ppb as O_2 (w/o oxygen scavenger)
Boiler Water	
Silica**	8 ppm as SiO_2
Total alkalinity	100 ppm as $CaCO_3$
Specific conductance	1000 mmho/cm
Total solids	1250 ppm (ABMA max.)
Suspended solids	40 ppm (ABMA max.)
Steam Purity Limit	
Total dissolved solids	0.1-0.5 ppm (max. expected value)
Silica	0.02-0.03 ppm, or 20-30 ppb as SiO_2 (ABMA limit)

Notes:
* Well-designed and operated deaerators can reduce oxygen to as low as 7 ppb.
** Silica limit based on limiting silica in steam.

ASME guidelines unless otherwise stated.
The above parameters and limits should be reviewed with the site water treatment provider.

Table 8. Water/Steam Related Parameters Specifically for 1001 to 1500 PSIG Steam Generators (ASME Guidelines Unless Otherwise Stated).

Boiler Feedwater Limits	
Organic	Nondetectable (TC/TOC min. detection ~ 0.5 ppm)
Iron	0.010 ppm or 10 ppb as Fe
Copper	0.010 ppm or 10 ppb as Cu
Total hardness	None detectable
Dissolved oxygen*	10 ppb as O_2 (w/o oxygen scavenger)
Boiler Water	
Silica**	2 ppm as SiO_2
Total alkalinity	Not specified, dictated by boiler water treatment program
Specific conductance	150 mmho/cm
Total solids	1000 ppm (ABMA max.)
Suspended solids	20 ppm (ABMA max.)
Steam Purity Limit	
Total dissolved solids	0.1 ppm or 100 ppb (max. expected value)
Silica	0.02-0.03 ppm or 20-30 ppb as SiO_2 (ABMA limit)

Notes:
* Well-designed and operated deaerators can reduce oxygen to as low as 7 ppb.
** Silica limit based on limiting silica in steam.

ASME guidelines unless otherwise stated.
The above parameters and limits should be reviewed with the site water treatment provider.

Cleaning

Cleaning rotors after being exposed to steam contaminants involves the removal of water-soluble and water-insoluble deposits. The original equipment manufacturer (OEM) should be contacted for recommendations concerning water washing and grit blast specifications and procedures. The author prefers to grit blast prior to water washing. For the general cleaning of steam turbine rotors, the recommended material is aluminum oxide with a particle size of 220 mesh maximum. All the sensitive areas must be protected by masking, such as the journal bearing areas, thrust disk, threads, rotor shaft ends, vibration probe, etc. It is vital to have an experienced individual performing this task. If the blast gun is held in one place too long, metal will be removed in addition to the contaminants. The only reason for water washing is to remove water-soluble deposits in the dovetail regions of the bucket and wheel assemblies and the tenon areas of the bucket and shroud band areas. In general water washing consists of three steps:

1. A general rinse and high-pressure water spray to remove the majority of the water-soluble deposits on the surface.
2. Completely submerging the turbine rotor assembly in a water bath. The bath must consist of demineralized water with a conductivity of less than 1 micro siemens (μS) per centimeter and have a pH range of 5.0 to 7.
3. A high-pressure water wash to remove seepage products following the soaking is the final step. The seepage must be tested and the pH must be below 7 to 10. If the seepage is above a pH of 10, then the rotor must be resubmerged in the cleaning bath.

Step 3 must be repeated as many times as required to lower or raise the seepage pH into the recommended range.

Turbine rotors can be water washed at slow roll speeds. At-speed water washing is *not* recommended by any of the turbine manufacturers. This will remove water-soluble deposits such as salts (NaCl) and turbine blading if not executed very carefully. The deposits in the turbine that are not water-soluble, such as silica, can only be removed by abrasive blasting of the rotor with the critical areas, e.g., bearing journals and vibration probe, masked.

BACK TO THE STORY

The turbine case was sent to welding and after a thorough cleaning the rotor was stress relieved then put into a lathe. The rotor was turned (machined) true. The wheel areas were actually undercut in the rotor shaft. The next step in the process was to begin submerged arc welding. Weld metal was deposited about 3 inches on the radius, then the rotor was turned to a rough dimension and the rotor was magnetic particle inspected. If the rotor passed the magnetic particle inspection the weld material was ultrasonically tested to find any inclusions in the welds. When an inclusion was found, greater than 1/32 inch, the inclusion was ground out and repaired via welding (Figures 15 through 19). This procedure was repeated many times until the rotor was sent to final machining and inspection (Figures 20 through 22).

During this period discussions were conducted to determine if there was any way to reduce the stresses on the third wheel. An agreement was reached that the steel bucket and shroud bands would be replaced with titanium Z lock buckets that would reduce the forces on the third wheel. A full engineering study was completed and no issues were found with the change of the bucket material.

As the rotor work was proceeding, the turbine case was being weld repaired. As can be seen this was not an easy matter (Figures 23 through 26). First, the split lines of the upper and lower halves were weld repaired and rough machined. Then, the register fit areas were weld repaired and rough machined. The turbine case halves were then sent to stress relieving. Once this operation was completed, the case was returned to the machine shop and final machining was completed and the case was dimensionally checked.

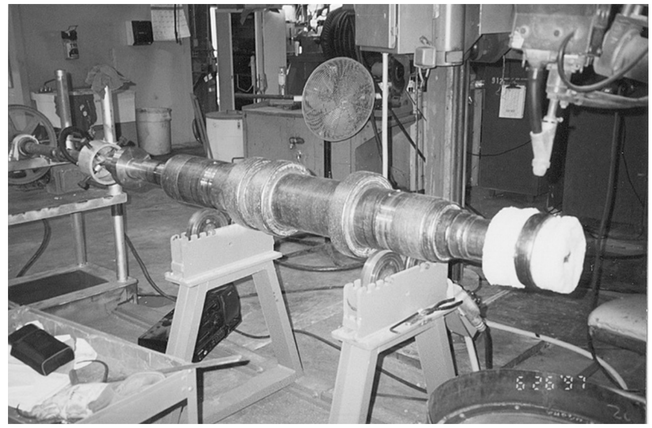


Figure 15. Initial Weld Pass of Turbine Rotor.

Figure 16. Submerged Arc Welding.

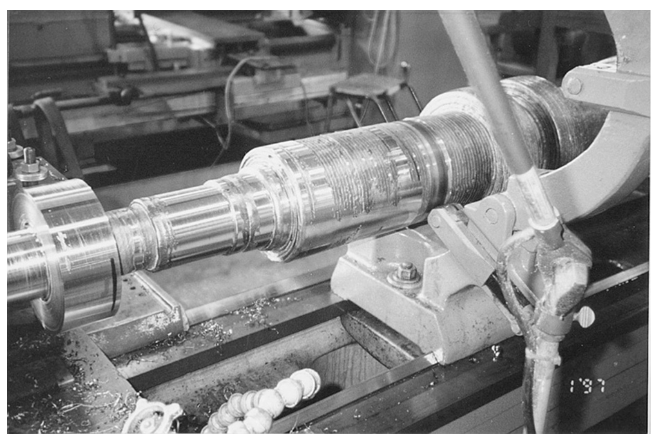


Figure 17. Machining after Initial Weld Pass.

The original cast bearing bracket was fabricated while other work was performed. The valve lift beam had damaged supports and had to be fabricated along with other “miscellaneous” components. The rotor was at-speed balanced to 0.5 mm/sec/bearing (the author’s specification), then installed in the repaired case and returned to the plant (Figure 27). All the above work detailed took six weeks.

Since all the controls and vibration/temperature monitoring equipment were destroyed in the fire, this opportunity was taken to upgrade these systems. New state-of-the-art turbine controls and monitoring systems were installed that allowed the startup of the train to go smoothly. The startup procedure was:

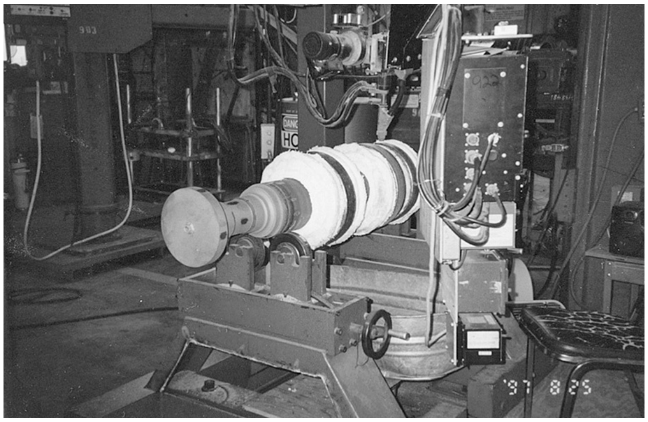


Figure 18. Turbine Rotor Stress Relieving.

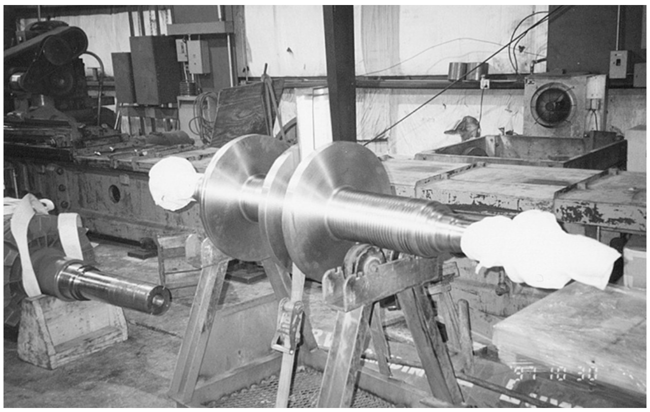


Figure 19. Turbine Rotor Welding Completed.

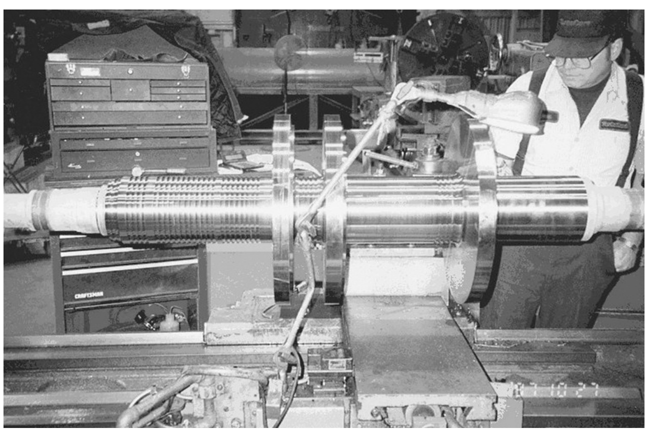


Figure 20. Turbine Rotor Final Machining.

1. Line up and start the water to the compressor water seals.

2. Line up and start the lube/control oil system.

3. Test the shutdown turbine overspeed shutdown devices.

4. Perform three shutdowns:
   a. Low oil trip
   b. Thrust trip
   c. Overspeed trip

5. Couple the turbine to the compressor train.

6. Slow roll the turbine at 500 rpm for 1 hour to allow the turbine case, turbine rotor, and oil systems to warm up close to their operating temperature.

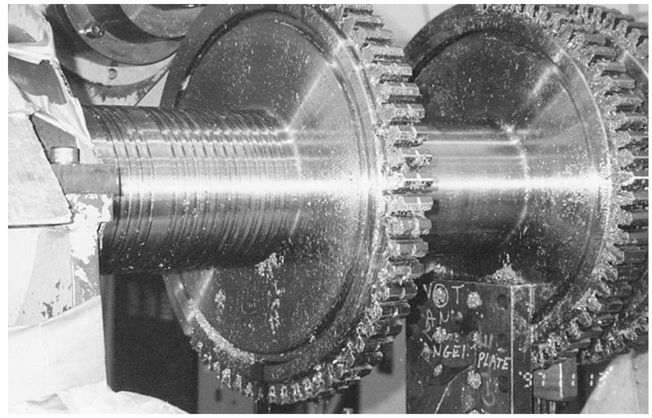


Figure 21. Cutting Bucket Dove Tails in Turbine Wheels.

Figure 22. Loading Buckets into Turbine Rotor Wheels.

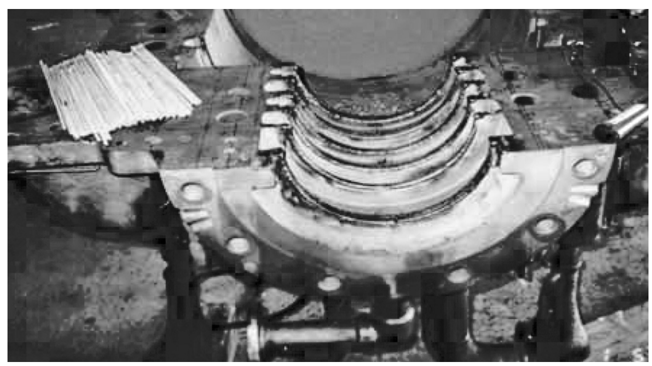


Figure 23. Upper Half of Case.

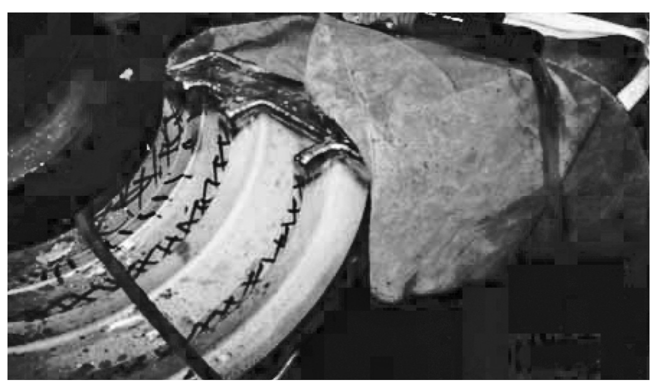


Figure 24. Case Split Line.

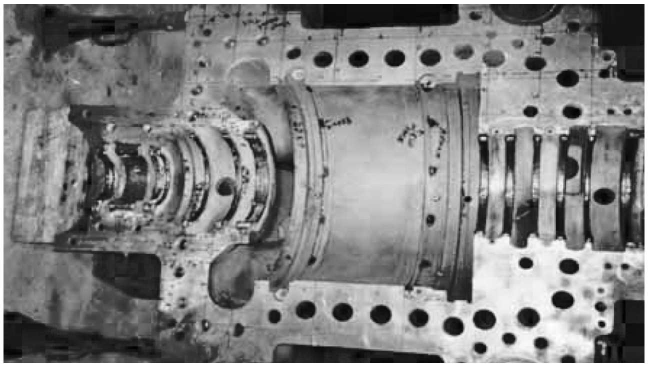


Figure 25. Bottom Half of Case.

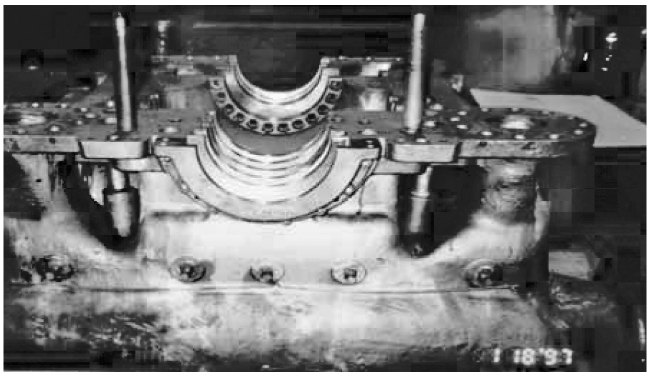


Figure 26. Upper Half of Case after Welding Stress Relieving and Machining.

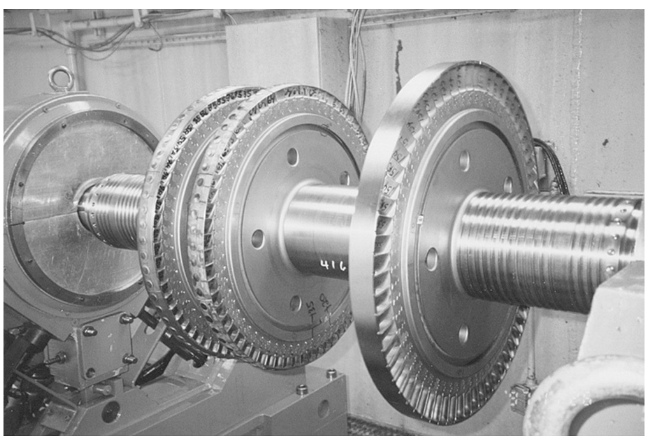


Figure 27. Finished Turbine Rotor in the At-Speed Balance Bunker.

7. Step the turbine speed up at 500 rpm per step until the speed set point, which is 500 rpm below the critical speed.

8. Press fast ramp on the electronic governor, the turbine ramps up at 250 rpm/second until 500 rpm above the first critical speed.

9. Set at this speed for 30 minutes, allowing all the turbine component temperatures to become stable.

10. Once the minimum operating speed set point is reached, continue to jog up at 500 rpm steps until the high-speed stop (maximum continuous operating speed) is reached.

11. Bypass the high-speed stop and run the turbine up to the overspeed trip set point.

Once the turbine was soloed it was shut down and coupled to the compressor/gearbox/motor and restarted. The startup of the compressor train steps is identical until step 10 (listed above), when 6000 rpm is reached, the motor is started, and the equipment is brought up to the normal operating speed of 8676 rpm. This step takes about four seconds, then the machine is turned over to operations. The vibration levels were less than 1 mil in any plane. Bearing temperatures, thrust positions, compressor process temperatures, etc., were at their normal operating conditions and life was good!

Approximately an hour after startup, the compressor inboard and outboard bearing temperatures begin to rise, and the vibration in both planes started to rise about 0.2 of a mil per hour. All the other operating parameters remained unchanged. The maximum radial shaft vibration level, in any plane, was considered to be 4.5 mils. The orbit of the compressor shaft in the bearings was perfectly round and at synchronous speed. A spectrum analysis of the vibration also indicated that the vibration was predominately at synchronous speed. It appeared that an unbalance existed and was slowly moving away from the compressor shaft centerline. The compressor was shut down and allowed to come to ambient temperature and then restarted. At speeds below the first critical speed, the vibration levels of the compressor rotor were slightly above those seen initially on the first startup. Once the critical speed of the compressor was gone through, the vibration levels rapidly increased to 4.5 mils and continued to climb. Analysis of the vibration indicated that at this point a rub was being picked up and thus the vibration climbed at a faster rate. Two additional attempts to start up were made before the conclusion to split the compressor case and determine the cause of the vibration.

The compressor case was split and some minor rubbing was apparent, but it was not significant enough to cause the vibration levels to be seen until a dial indicator was put on the compressor lower half and the runout was checked at the center of the rotor. A 45 mil bow was discovered. The rotor was removed from the case and returned to the manufacturer's machine shop. Dimensional checks were repeated. Runouts were repeated. Except for the instance when the runout at the center of the rotor was performed, 5 mils of runout were measured, which was acceptable for this rotor. But this was not close to the 45 mils seen in the field. Since all readings were back to within tolerances, the rotor was returned to the plant and reinstalled in the compressor. The thought was that something must have shifted on the compressor rotor during the transport by truck to the machine shop.

After the reinstallation of the rotor, the compressor was closed up and recoupled and the machine train started up. At this point, the repair was started on the compressor rotor that was originally installed in the compressor during the initial wreck. The compressor train was restarted, as per procedure, and the exact same sequence of events occurred as detailed above.

The compressor case was shut down and split, once again, and the condition of the rotor was as it had been left. Management was not happy and life was not very good. The rotor was again checked. There was a 45 mil bow in the center of the rotor, so the rotor was returned for a thorough inspection. The same results were seen in the shop as before, rotor runout at the midspan was 4 mils. The rotor was unstacked and reassembled and everything was to specifications. A decision was made to return the compressor rotor to the facility. Once installed in the lower half of the compressor case, a runout check at midspan would be performed, and it was. The runout at midspan was exactly as measured in the shop, 4 mils. Armed with this information a decision was made to complete the installation and startup.

As expected by some, the same sequence of events occurred. Life at this time was terrible! When the rotor was removed from the case this time, it was transported with the thrust disk and the end seals were left on the rotor. The rotor was set into v-blocks, seals supported, and runouts were retaken. The 45 mil runout at midspan was found this time. As the rotor was disassembled, a dial indicator was placed at the midspan so any changes could be seen.

In the shop, with at least a dozen spectators, the thrust disk nut was loosened and the midspan went from 45 mils to four mills. The OEM representative and the author were watching the dial indicator when it changed. We would have called each other a liar if we had not both seen the change. Fortunately, the compressor rotor from the initial wreck was repaired and the rotor was ready for installation. It was installed into the compressor and the compressor train was restarted, as per the operating procedure. The plan was to get the unit up then go back to the new rotor and perform a very detailed inspection and determine what was causing the 41 mil change in the rotor assembly. The machine train was restarted and vibration levels were less than 1 mil peak-to-peak at the operating speed of 8676 rpm. Life was good again.

The problem compressor rotor was stripped (impellers removed) and every type of test to the 17-4ph shaft was performed. Nothing could be found that would explain why the rotor would bow. The final test was to hang the shaft vertically and stress relieve it. Once the stress relieving was completed it was allowed to cool, then it was placed in a low speed balance stand and shaft runouts were taken. The problem was immediately determined, a 45 mil bow was found in the shaft at its midspan. The shaft had not been stress relieved properly. The OEM bought us a new shaft and reassembled the rotor. Several years later during a major compressor overhaul the new rotor was installed and it ran perfectly.

One would think that everyone lived happily ever after but they would be wrong. The compressor train ran about three months, then the author received a call that the vibration levels at the bearing journal had pegged the vibration monitor at 5 mils and the temperature of the turbine bearings was starting to climb. Operations was advised to shut down immediately and life had turned again. The unit was shut down. The turbine bearings were inspected and nothing was found. Digital data recordings of the vibration were reviewed and it appeared that a mass unbalance suddenly occurred. The turbine case was split and four titanium buckets had broken off at the blade root and were lying in the bottom of the case. Now a way had to be found to start up the compressor train without the turbine. A modal and tensional analysis was completed. The only way the compressor train could be brought up was to leave the turbine side coupling that was mounted on the compressor rotor on the end of the rotor; that overhung moment was needed to have a stable machine train. Startup procedures were rewritten to prevent the motor from tripping due to the extended rampup time that was expected for motor overloads. The time out was set from 8 seconds to 15 seconds and it actually took 13 seconds to reach 8676 rpm with the compressor completely unloaded. About three weeks later the motor tripped due to overloads and this was a surprise to operation personnel because they were not aware this shutdown set point was not increased to give them any warning. This did prove that they could trip without warning and the plant would not exceed a safety critical variable. All the safety systems functioned as designed; ethylene oxide gas was directed to the flare.

Now the reason the titanium buckets failed could be investigated. The buckets were removed from the third wheel and the rotor was inspected thoroughly and the wheel was in as-new condition. It was finally determined that the steam at the inlet of the third stage was causing very high alternate loading of the buckets because the purchased steam was still super heated and the “reactor” steam was just at the saturation temperature at this point in the turbine. Options were defined and it was decided to add thickness, about fi inch to the third stage wheel and install thicker buckets and these buckets would have shroud bands. The modifications were made to the turbine rotor and the rotor was installed in the case on the compressor platform awaiting an opportunity to couple up to the compressor. We did not have to wait long—two months before an electrical outage brought the unit down. During the outage the turbine was coupled to the compressor and started up, as per the operating procedures. The turbine rotor forging that had been ordered was then received and machined to the new specification. It was then put into a climate controlled storage hanging vertically as a spare.

ANOTHER TURBINE FAILURE

In mid-1999, after several years of smooth operation the turbine rotor vibration began to rise. At 2.5 mils the first vibration alarm sounded in the control room alerting the operation personnel that the turbine had a problem. The other equipment vibration levels remained normal. Fortunately a field operator was in the immediate area and reported a “strange noise” emitting from the turbine. The resident mechanical engineer was called and he told operations that if the vibration reached 4.5 mils to shut down the compressor train. Within a minute, operations had to shut down the compressor train.

Upon reviewing the vibration data, all indications were that there was a mass unbalance and it was getting worse. The decision was to uncouple the turbine and run the compressor with the motor alone. Maintenance forces were mobilized and the turbine was dismantled after a quick inspection of the bearings showed no signs of damage. The case was lifted; the problem was obvious. The shroud band of one of the bucket packets was pealing off like an 18-wheeler retread.

Metal pieces were recovered from the recycle gas drive compressor turbine after its failure in May. They were believed to be part of the shroud assembly that covered the turbine buckets. The metal pieces, all smaller than 1 inch in length and fi inch in width, were heavily banged and deformed. However, one round nipple-shaped piece, which is believed to be one of the tenons, appeared to have an undamaged fracture surface. The turbine components had been in service for about two years.

Analysis

SEM Fractography

The metal pieces were first ultrasonic cleaned in a concentrated chemical cleaner to remove grease and loose surface films, and then the “nipple” and one of the larger pieces were examined under a scanning electron microscope (SEM).

- *Tenon (or nipple) piece*—It was quite fortunate that the fracture surface was mostly undamaged except in areas near the circumference of the tenon (Figure 28). However, the fracture surface is severely oxidized, which masked out the fine details on the surface. Still, it could be determined to be a fatigue failure (Figure 29) with the typical feature of fatigue striations (Figure 30).

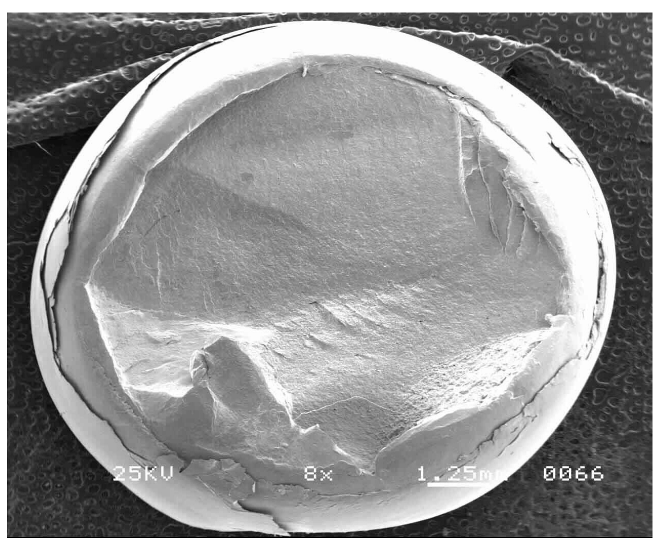


Figure 28. Bucket Tenon.

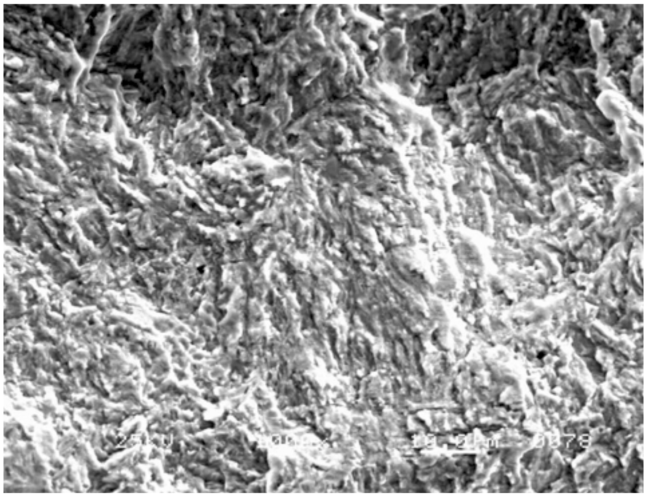


Figure 29. Fatigue Failure.

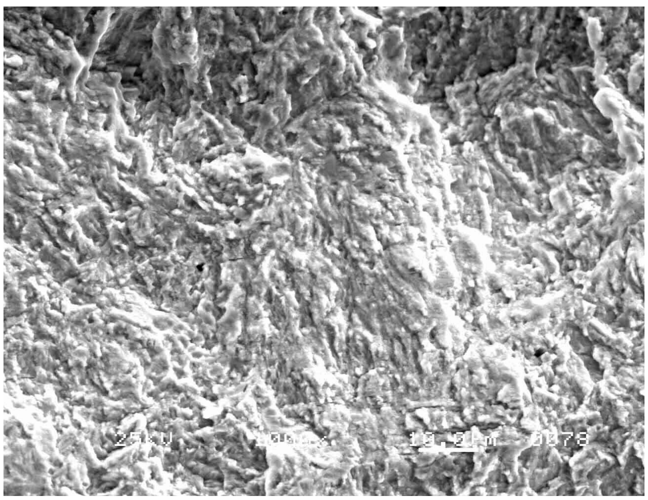


Figure 30. Fatigue Striations.

On one end of the fracture, there was clear sign of final ductile overload fracture (Figure 31). Also known as microvoid coalescence (MVC). Another final fracture area is found about 90 degrees to the first one. Here the microvoids appeared to be distorted or elongated (Figure 32), which indicates the fracture is at a different orientation. Efforts to locate the crack initiation site by tracing back from the final fracture areas to the opposite end of the fracture surface were not successful, as the area was too damaged to be recognizable (Figure 33).

A feathery area (Figure 34) across from the second final fracture area was determined to be a fatigue area (Figure 35). When viewed at higher magnification, all in all, the fatigue area was estimated to cover over 70 percent of the fracture surface.

• *Shroud piece*—There was not much to look at, because the "fracture surface" had been damaged (rubbed) and also heavily oxidized (Figure 36). In a cracked area, near one end of the surface, it was also damaged, but it had some scale attached to it (Figure 37). Even the surface inside the crack was found to be severely oxidized (Figure 38). Heavy scaling was also found on the side surface of the shroud piece (Figure 39).

Chemical Analysis (by X-Ray Fluorescence)

• *Based metal*—The chemical compositions of the tenon and shroud pieces were found to be close to that of 422 stainless steels. They are listed below in Table 9.

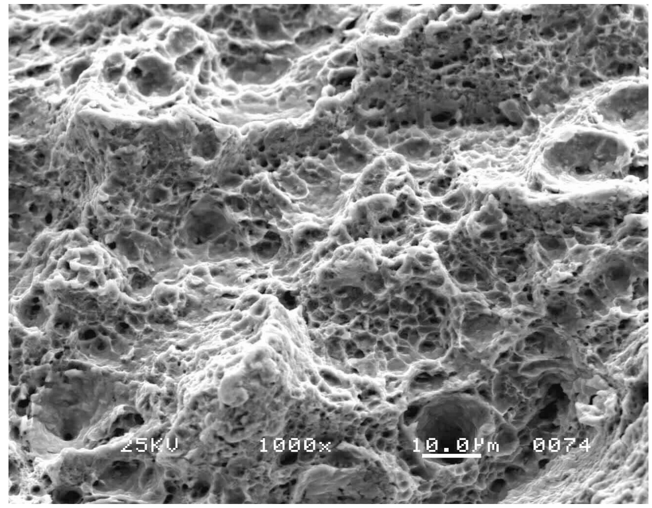


Figure 31. Ductile Overload Failure.

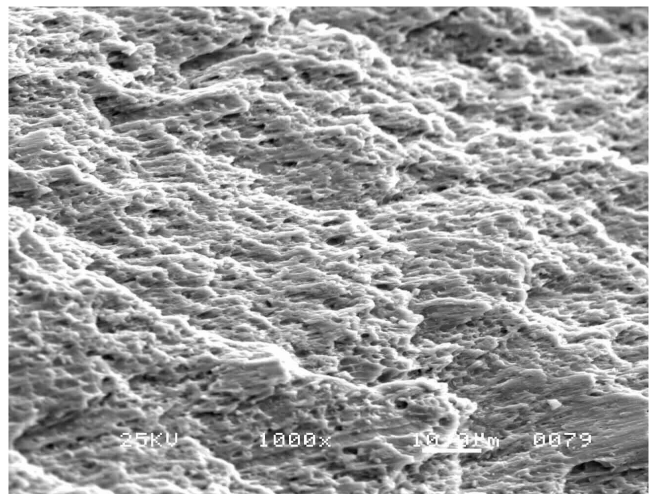


Figure 32. Elongation of Microvoids.

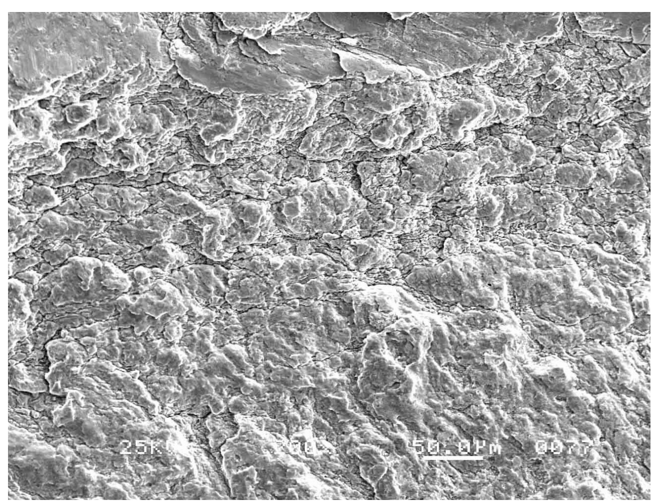


Figure 33. Badly Damaged Area.

• *Oxidized surfaces*—The oxygen content ranged from 8 to 14 wt. percent or 24 to 35 atomic percent. In some areas the oxide films were so thick that only 2 to 3 percent of chromium was detected. Other than the basic elements like iron, nickel, and manganese,

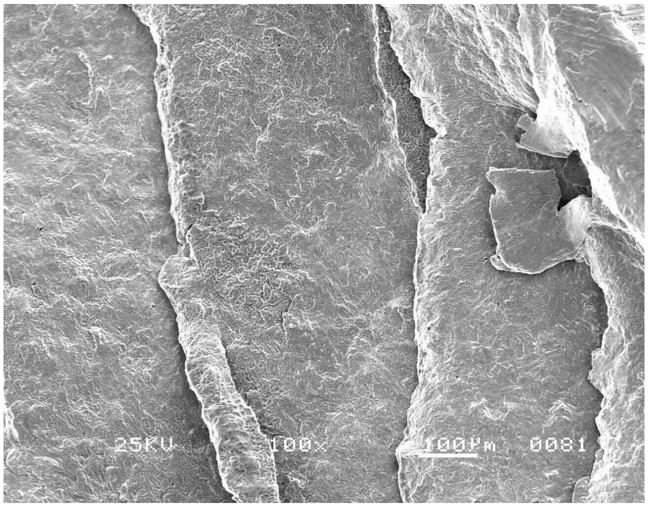


Figure 34. Feathery Area.

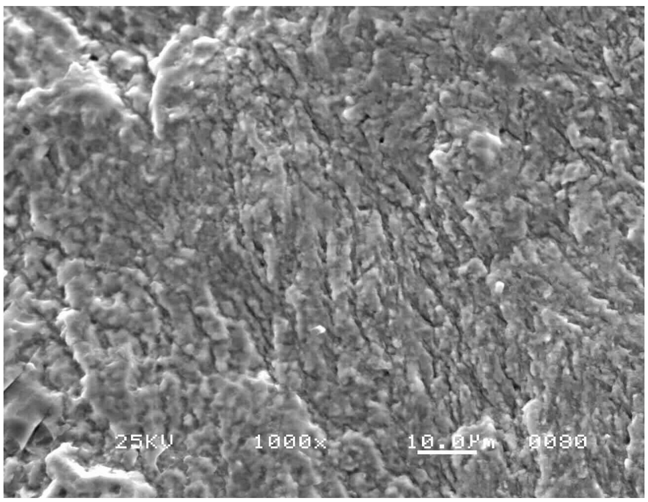


Figure 35. Final Fracture Area.

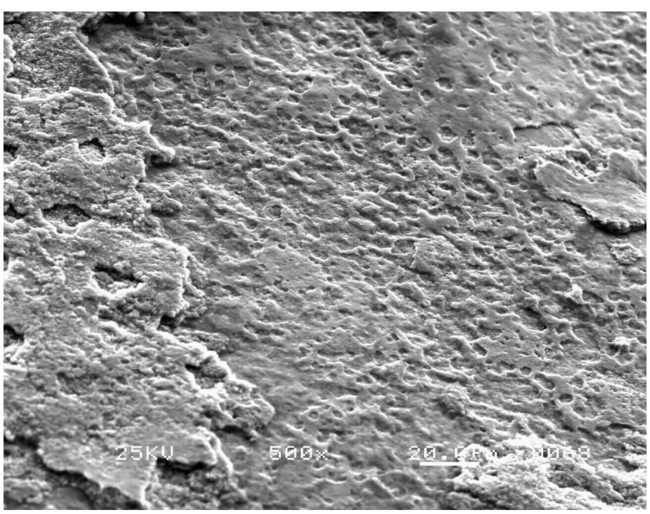


Figure 36. Fracture Surface.

there were also silicon (0.4 to 1.5 percent), vanadium (0.8 to 1.9 percent), aluminum (0.3 to 0.5 percent), zinc (up to 1 percent), copper, (up to 2.4 percent), and sulfur (0.3 percent).

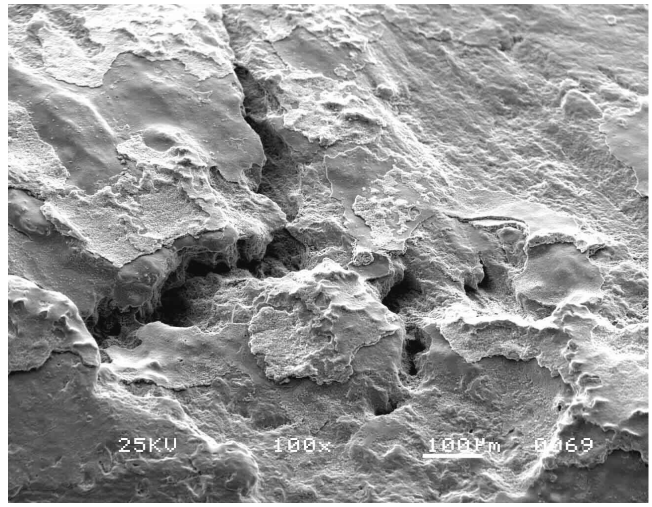


Figure 37. Scale at One End.

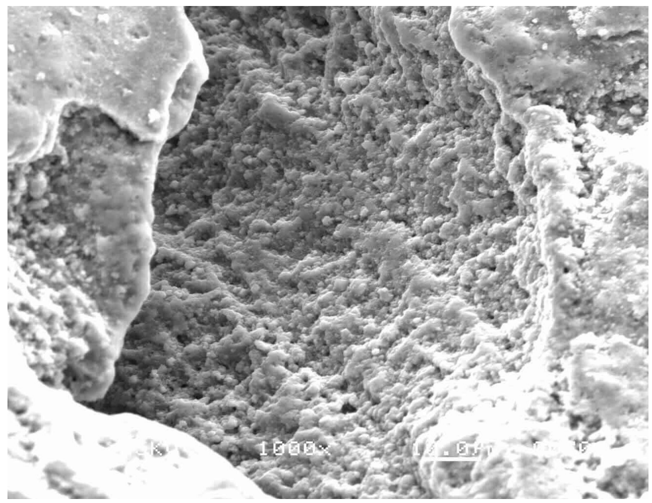


Figure 38. Severe Oxidation.

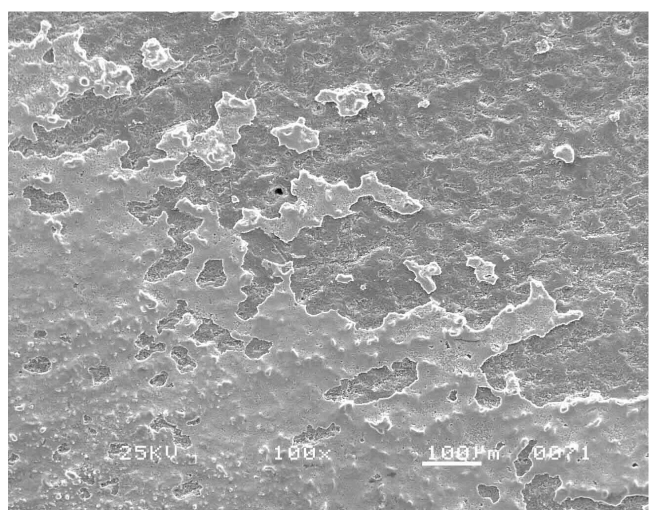


Figure 39. Shroud Piece.

• *Scale*—The surfaces of the shroud and tenon pieces, including the fracture surfaces, were covered with scale. The scale was found to be rich in calcium (3 to 4 percent), silicon (0.7 to 3.3 percent),

Table 9. Chemical Compositions.

	422 SS	Tenon	Shroud
Chromium	11.5 13.5	12.26	12.46
Molybdenum	0.75 1.25	1.03	0.97
Nickel	0.5 1.0	0.79	0.86
Manganese	1.0 max.	0.78	0.75
Silicon	0.75 max	0.62	0.46
Vanadium	0.15 0.3	< 0.1	< 0.1
Phosphorus	0.04 max.	< 0.1	< 0.1
Iron	Balance	84.5	84.5

phosphorus (1 to 3.2 percent), vanadium, (0.7 to 1.7 percent), copper (0.5 to 1 percent), aluminum (0.15 to 1.42 percent), and zinc (0.4 to 1.1 percent).

The failure of the tenon has been confirmed by SEM fractographs as fatigue, or more precisely corrosion assisted fatigue. It is also likely a high cycle (or low stress) fatigue failure because:

- The ratio of fatigue area to final fracture area is over 2 to 1. This means the fatigue crack propagated through a large portion of the cross-sectional area of the tenon before final fast fracture occurred.
- The fatigue area was heavily oxidized. In order to have such a thick oxide film on the surface, the oxidation is believed to have occurred over a long period of time. The final fracture areas, on the other hand, were not heavily oxidized because the fracture tenon was removed from the turbine shortly after the failure.

The shroud and tenon were confirmed to be 422 stainless steel. With the small amount of alloying element molybdenum and nickel, 422 SS offers better corrosion resistance and higher hardenability than the lower alloy 400s martensitic SS like 410 or 416. The excess deformation and rubbing damages on the pieces, the lack of secondary cracking, and the MVC appearance of the final fracture all indicate the material has good ductility and toughness.

The severe oxidation and scaling on the failed pieces are not believed to be beneficial to the service life of the turbine components. It was believed that the steam is of poor quality and is corrosive to 422 SS. The steam also contains a high amount of impurities. The elements found in the scales like calcium, phosphorous, silicon, and vanadium are believed to be water treatment chemicals, while the zinc, copper, and aluminum are likely to be corrosion products that were carried over by the steam. Since steam is water vapor, it is not supposed to carry many impurities. The steam used in the turbine is thus believed to be very wet and contained liquid entrainment. Besides oxidation, there was no other sign of environmental degradation like localized corrosion, stress corrosion cracking, or caustic embrittlement on the failed pieces.

The failure of the tenon is believed to be high cycle fatigue. The actual fatigue duration is not known, but is estimated to be months. The quality of steam is questionable; it is believed to have contributed to the scaling and heavy oxidation on the failed components. However, there is not enough evidence to indicate the low quality steam, or the scaling and oxidation on the parts, are the root causes of the failure. The materials of construction of the tenon and shroud are 422 SS, which has good corrosion resistance and mechanical properties. So the failure is not believed to be material related.

With the above information engineers revised the shroud design by reducing the overall width of the band by 1/2 inch, thus reducing the stress on the tenons.

LESSONS LEARNED

- Stress corrosion crack can occur very rapidly when all the right conditions are present.
- Recommended steam quality
- Using today's technology almost anything can be repaired.
- If the equipment is critical to the operation of the unit or to the facility, spare rotors are essential.
- New rotors can have unseen problems.
- 15-5ph material is far better than 17-4ph, relative to stability.
- High thrust shutdowns should not be bypassed.
- The perception of danger is not always accurate or understood.

REFERENCE

Drew, 1994, "Principles of Industrial Water Treatment," Eleventh Edition, Published by Drew Chemical Corporation, Boonton, New Jersey, p. 310.

BIBLIOGRAPHY

"The ASME Handbook on Water Technology for Thermal Power Systems," Paul Cohen, Editor-in Chief, Sponsored by the ASME Research and Technology Committee on Water and Steam in Thermal Power Systems, EPRI Research Project No. RP 1958-1.

ASTM A-380, 1999, "Standard Practice for Cleaning, Descaling, and Passivation of Stainless Steel Parts, Equipment, and Systems," American Society for Testing and Materials, West Conshohocken, Pennsylvania.Elliott Specifications for Steam Quality.

General Electric Instructions GEK-63430, "Turbine Steam Purity."

General Electric Instructions GEK-72281, "Steam Purity—Stress Corrosion Cracking."

General Electric Power Generation Global Product & Technology Support, TIL 1231-3, "Maintenance Recommendations for Cleaning Deposits and Chemical Contamination from Steam Turbine Rotors," August 15, 1997.

General Electric Power Systems GEK 98965, "Steam Turbines, Steam Purity for Industrial Turbine."

Lindinger, R. J. and Curran, R. M., 1981, "Experience with Stress Corrosion Cracking in Large Steam Turbines," The National Association of Corrosion Engineers, The International Corrosion Forum, Corrosion/81, Paper Number 7, Toronto, Ontario, Canada.

McIntyre, D. R. and Dillon, C. P., 1985, "Guidelines for Preventing Stress Corrosion Cracking in the Chemical Process Industries," MTI Publication No. 15.