TURBINE REMANUFACTURE—ONE OPTION FOR RELIABILITY AND EFFICIENCY IMPROVEMENT

by

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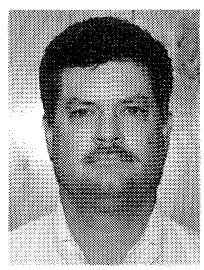
Mr. Rasmussen has a B.S. degree (Mechanical Engineering) from the University of Wisconsin and an M.B.A. from Kent State University. He and his company were recently honored with Power Magazine's 1994 Powerplant Award for their participation in turbine renovation and life extension projects.



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ABSTRACT

A case study of a remanufacture of a steam turbine is presented. The turbine has been in service for 20 years. Thermal and physical stresses have taken their toll on the turbine casing and supporting hardware.

Remanufacture was planned with innovative processes and state-of-the-art component upgrades. A unique heat treatment process was designed to reshape the casing and relieve residual stresses thermally induced into the casing over its operating life. Using heat to reshape the case minimized the amount of remachining time required.

A team of three vendors was combined to maximize technical and physical resources and accomplish this task in 24 days.

The goals of the remanufacture included:

- Recovery of lost horsepower and efficiency due to steam path deterioration
- Increase efficiency through the application of packing, tip, and nozzle seal upgrades
- Elimination of chronic maintenance problems

Major topics include:

- Casing disassembly and damage assessment
- Weld restoration of erosion damage and steam path blading
- Distortion modelling and evaluation
- Heat treatment of case
- Assessment of heat treatment results
- Machining steps required
- Design and installation of retractable packing
- Design and installation of tip seals and nozzle seals
- Successful startup and operational history

INTRODUCTION

The project described herein is believed to be unique in the industry for several reasons: the time window for repairs was small, and the efficiency enhancements were untried for this particular turbine application. Goals set for the turnaround were extremely aggressive regarding the extent of planned scope and the short duration of the schedule. The intentions of the project team were to achieve success through indepth planning during a 10 month period prior to shutdown. Plans for efficiency recovery were equally aggressive. The primary efficiency enhancement application had never been attempted on a compressor train driver. A compressor train represented a particular challenge, since the startup procedure required coming up to speed under load.

Overview of Turbine and Application

The subject steam turbine (Figure 1) is the driver for three charge gas compressors in an ethylene plant. Commissioned in the early 70s, the unit has been subjected to near continuous service except for scheduled plant maintenance outages. The unit is mounted on an open-air, unprotected deck. Operating specifications are listed in Table 1.

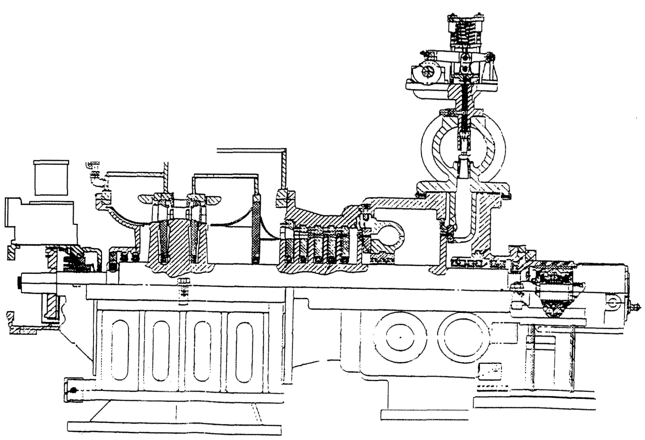

Figure 1. Longitudinal Section of Steam Turbine.

Table 1. Turbine Operating Specifications.

Parameter	Normal
Horsepower	27,644
Inlet steam pressure	1500 psi
Inlet steam temperature	850°F
Extraction/Admission pressure	650 psi
Exhaust pressure	4 in-HgA
Maximum inlet flow	500,000 lb/hr
Maximum speed	5475 rpm
Stages	10 Rateau - 9-10 double flow opposed

The turbine control system is designed to control two parameters: inlet steam header pressure and compressor suction pressure. The 1500 psi steam supply for the turbine is produced in nine furnaces within the ethylene plant. The turbine controls monitor the 1500 psi steam pressure and adjust six inlet admission valves to maintain pressure within reasonably constant parameters. Two extraction valve controls monitor compressor suction pressure. When a lower pressure is required (i.e., higher speed), the valves open and allow additional steam flow into the last nine turbine stages. Requests for higher suction pressure (lower speed) result in a closure of extraction valves, thereby forcing more inlet steam into the plant 650 psi steam header.

The turbine design provides for 1500 psi steam to enter the steam chest through one side only. The steam chest consists of an external cavity housing six governor valves. The individual nozzle chambers are designed as elongated passages (cast pipes) in a "finger" arrangement feeding a bank of bolted-on nozzle segments. Four nozzle segments/fingers are integral to the upper steam chest and two nozzle segments/fingers are mounted to the lower case. Connection to the lower fingers is via external piping attached from the steam chest housing to the lower case. The steam chest nozzle segments/fingers are connected to the upper case by a rectangular "race track" type flange. Adjustment/alignment is possible since this section is movable in all horizontal planes, within the limits of the bolt holes. Because this is a blind assembly, it is extremely difficult to verify perpendicularity and correct nozzle clearance. The "race track" seals the 650 psi steam and has been a chronic source of steam leaks.

Plant startup procedures presents a challenge to the turbine, in that 1500 psi steam is not available until the plant furnaces are operating at full firing rates. Economic and environmental considerations require that the turbine and the three compressors be capable of running with a minimum of furnaces online. This requirement is achieved by starting the turbine/compressor train on available plant 650 psi steam. Startup steam is admitted to the turbine through extraction nozzles in the second stage. The 650 psi steam is capable only of starting the unit and running the train at minimum governor speed. When 1500 psi steam becomes available, the 1500 psi trip valve is opened and the turbine begins normal operation.

Known Problems

Past inplace turbine overhauls had revealed the following problems:

- Gaps would appear in the splitline when the horizontal joint bolts were loosened.
- Overtorquing the horizontal joint bolts was required to close the horizontal joint splitline.
- Diaphragms in the turbine high pressure (HP) section could not be set low enough to match the rotor centerline.
- The exhaust end bearing had a 0.040 in vertical offset, which had to be adjusted to bring it in line with the turbine centerline during reassembly.
- The nozzle steam path and low pressure (LP) diaphragm fit areas were damaged from wet steam erosion and solid particle transit (Figure 2).
- The nozzle fingers (chambers) (Figure 3) and, therefore, the nozzle segments were misaligned to themselves, to the turbine centerline, and to the rotor control stage blades.
- Some nozzle partitions were damaged or missing.
- Past startups were very rough because of packing rubs.

These problems presented time-consuming challenges to maintenance. Turbine overhauls were an ordeal of trial and error compromises. Since some of the diaphragms could not be set low

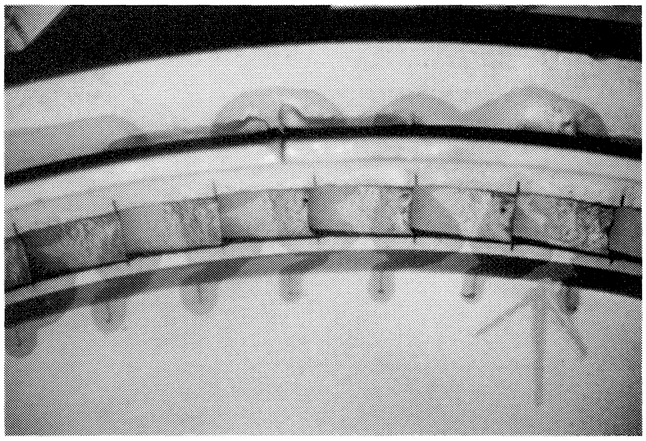

Figure 2. View of Nozzle Erosion.

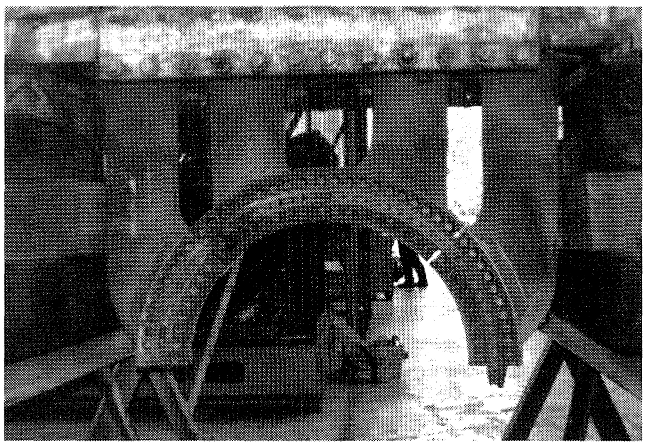

Figure 3. Incoming View of Nozzle Fingers.

enough, the difference was scraped from the internal labyrinth seals. Obtaining minimum seal clearance required several episodes of putting the cover on, torquing until the gap closed, removing the cover, measuring the seal imprints, scraping the seals, then repeating the process. Installing and removing the cover was time consuming. However, when working with a warped horizontal joint, the turbine closure time increased by a factor of five. Closing the high pressure section splitline gap required several iterations of stud heating and torquing until a 0.0015 in feeler gauge could not be inserted anywhere along the horizontal joint.

Timeframe

Due to a scheduled major upgrade of the entire plant, a long (45 day) outage was planned. The customer decided to take advantage of the extended outage interval to remove the turbine from the train, correct all outstanding maintenance problems, and return the turbine to design efficiency.

Returning the turbine to design efficiency was an important economic goal. Turbine efficiency was conservatively estimated to be 10 percent below original design due to the poor steam path conditions and case distortion. Preliminary calculations indicated that returning the turbine to design efficiency would result in a payback of less than one year. Improvements over design efficiency would reap further benefit.

Critical path estimates for removal and reinstallation of the turbine, decontamination, and reinventory of the process side, established the maximum shop time for turbine remanufacture at 24 days. A schedule exceeding 24 days would adversely affect the overall outage critical path. After 24 days, significant economic impact would begin to lengthen project payback. Lost revenue from extended downtime would quickly render the entire project uneconomical.

Several vendors were asked to evaluate time requirements to complete the desired remanufacture. Only one believed the job could be completed in the allotted time. The primary difference was in this vendor's use of new technology. Their plan involved heat treatment of the entire case to correct the casing distortion prior to machining the case. The vendor was confident that most of the casing distortion could be removed using thermal processes. The objective of the heat treatment was to eliminate, or at least minimize, horizontal joint machining. If splitline joint machining could be avoided, then the steam chest mounting face would not require machining. Also, the turbine centerline would remain in the same relationship with the horizontal splitline. In this way, time consuming boring steps could be eliminated.

Planned Remanufacture Scope

Over a 10-month period preceding the outage, the project team met on a number of occasions to plan the turbine remanufacture. The project team included the customer, vendor engineering, technical and shop personnel, and a consulting firm. The first aspect covered was a review of historical data derived from the previous two turbine overhauls. Data from a previous mapping of the casing splitline bottom half led team members to believe that the case possessed a longitudinal "hump." It was determined that an opening appraisal of casing ovality and the relative concentricity of the diaphragm fit areas must be performed at the earliest opportunity. Mapping of the horizontal joint flange faces would be performed on each casing half in parallel with indicating the turbine bores. Examination of the steam path was planned to follow case mapping. Another planning concern was the past problems that overhaul teams had when setting the HP diaphragm vertical position. Measurements to determine the relative concentricity of the diaphragm fit areas to the theoretical turbine centerline were also planned. An evaluation of the accumulated data would then be performed prior to entering the heat treatment process. Goals of the planned scope were:

- Correct splitline distortion.
- Return all axial and diametrical fits to proper size, position, and relation to turbine centerline.
- Repair damaged/missing nozzles and steam path components.
- Correct nozzle alignment.
- Repair/buildup steam chest sealing surface to maintain seal and vertical position.
- Machine diaphragm fits to provide radial clearance at operating temperature.
- Modify diaphragm supports to allow thermal growth.
- Return all sealing surfaces to as new condition.
- Bore bearing housings concentric with turbine centerline.
- Replace all vertical and horizontal splitline bolts.
- Renew all seals. Add tip seals and nozzle seals, if possible.
- Install retractable interstage packing.
- Install redesigned inlet and extraction valves.
- Total shop time must be less than 24 days.

A preinspected, fully repaired, and operating speed-balanced rotor would be installed in the turbine case when repairs were completed.

Desired Results of Stress Relieving

The critical timing of the remanufacture required hard decisions or gambles on applying new technology. Stress relieving was scheduled to consume three to five days of the schedule. During this time, no other work could occur on the casing. The conventional paradigm is to make chips from start to finish. Taking a break in midstream for a sauna bath just did not feel right. Two factors influenced the decision to try the stress relieving.

Since the case had 25 years of residual stresses, any machining would act as a stress relieving operation. Major cuts could release stresses that would require yet another cut to obtain dimensional stability. Depending on the level of stresses, up to three cuts could be required to produce an acceptable product. If these residual stresses could be eliminated or minimized through total case stress relieving, then if machining were required, dimensional stability could be obtained in one cut.

Machining the horizontal splitline would solve the problem of leakage and warpage, but it would introduce a whole set of other problems. Decking the splitline, in effect, lowers the turbine centerline. However, the lower nozzles are fixed in the case, and this establishes the rotor centerline. Thus, the rotor must remain on the nozzle centerline. Cutting the horizontal splitline forces the following additional work:

- Adjust bearing housings true to nozzle centerline.
- Cut, weld, and recut seal hook fits in casing (not on casing CL, but nozzle CL).
- Bore casing ID to allow proper diaphragm positioning.
- Weld buildup and remachine steam chest sealing flange.

Past experience of the stress relieving consultant indicated a high probability that the splitline could be returned to a near flat condition without machining. Comparing the proposed schedule of stress relieving and light touchup machining to the straight heavy machining, the stress relieving option consumed less time. However, due to numerous unknowns, this option was not without risk. If it did not work, at least one week of precious machining time would be lost.

Even with all the unknowns, the stress relieving option was selected. This was the only option that allowed completion of the remanufacture in the allotted timeframe.

TEARDOWN, INITIAL INSPECTIONS, AND EVALUATIONS

Immediately after offloading the assembled turbine from the transport, a team was set to work to remove the steam chest and valve rack assembly from turbine casing. Due to space restrictions, special low profile hydraulic type wrenches had to be used to loosen a number of the bolts. The project team, aware that all disassembly activities would impact critical path, had prepared several tooling methods for nut and stud removal. Following the steam chest removal, the crew promptly began the process of loosening the HP heated and LP casing stud bolts.

The rotor axial position, total thrust, and a bump check (without thrust shims) were taken and recorded. The bump check was performed after removal of the upper nozzle/valve rack assembly and again with the upper shell removed. The rotor was then removed and set aside, since it would be replaced with the fully refurbished spare rotor.

Casing NDE

The bearing housings, diaphragm, and extraction stage were removed, grit blasted, and magnetic particle (NDE) inspected. The upper and lower turbine case were laser mapped then sandblasted and NDE inspected. NDE revealed several cracks in the LP casing and struts. After NDE, visual and dimensional inspections were performed.

Mapping the Horizontal Joints

As stated earlier, the project team fully expected to encounter a turbine shell with an axially oriented "hump." In two previous overhauls, assembly teams experienced difficulty closing the horizontal joint gap and had to resort to overtorquing the bolts to obtain a metal-to-metal fit. The last overhaul, performed in 1992, had proved sufficiently difficult to warrant an attempt to understand the nature of the distortion. The horizontal joint flange on the bottom half of the case had been mapped using an optical device. This last overhaul, however, had been performed onsite, on the exposed (uncovered) turbine deck. The bottom half of the case was not removed from its foundation and it was difficult to differentiate elevation and/or keyway problems from casing distortion. Similarly, for the top half of the casing, no convenient way existed to properly set up and level the component for optical mapping. Readings taken on the bottom half, however, clearly indicated that a problem existed (regardless of root cause), and that the problem exceeded 0.060 inches in longitudinal distortion.

The primary advantage enjoyed by this project team was having the entire turbine in a service facility. Each casing half was positioned with the splitline flange in the up position. Each half was then mapped with a laser (Figure 4) to obtain the entire splitline topography. The laser mapping revealed that significant problems did indeed exist in the horizontal joint. The nature of the problem was a complete surprise. Both halves of the case horizontal joint had a vertical offset of 0.028 to 0.030 in at the vertical joint intersection. The vertical joint apparently had been assembled originally at the factory, with some type of gasket material. The vertical splitline gap varied 0.030 in from the top of the case to the splitlines on either side. The horizontal joint had a 0.030 in offset at the intersection of the vertical joint. The race track was also mapped and found to be flat within 0.002 in.

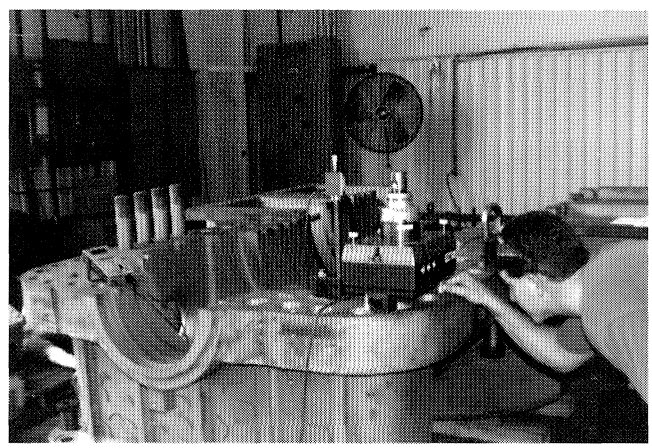

Figure 4. Laser Mapping Setup.

The first laser mapping operation determined the extent of existing casing distortion in the horizontal splitline. It was now clear that some distortion did exist, and the total elevation readings confirmed the readings obtained with the optical equipment onsite in 1992. However, the nature of the distortion was of considerable concern to the project team. The "longitudinal hump," expected by the project team, was instead a vertical offset between high pressure and the low pressure casings. The vertical offset was caused by the nature of the vertical joint. One could almost speculate that the four quarters of the case had been manufactured in separate facilities and then brought together for final assembly. When the casing quarters were assembled, the findings are summarized below:

- The vertical and horizontal splitlines were not square; they did not form 90 degree angles.
- The vertical joints, top and bottom, had a horizontal offset created by a twist between the HP and LP sections.
- The horizontal joints had an offset at the vertical joint between the HP and LP casings.

• The horizontal joint had low areas in the HP case near the vertical joint.

• The LP case lower half exhibited a "droop" on one side near the LP bearing housing vertical joint.

The project team decided to move on to mapping the casing bores in order to complete their understanding of the true shape of the casing before making any decisions as to how to proceed.

Indicating the Bores and Diaphragm Axial Fits

In order to ensure that the casing fit, bore, face and splitline dimensions, conditions, and relationships were thoroughly understood, the project team used redundancy in measuring techniques. Bores in each casing half were first measured with inside micrometers, with vertical heights taken from a straight edge across the horizontal splitline. Next, the bores were indicated using digital indicators and an aluminum mechanical tubing inspection bar. Finally, the tight wire method was utilized to verify readings obtained using the other two methods. The rationale for the redundancy was simplistic: "Measure twice (or three times); cut once." The project team knew that there would be no room in the schedule for mistakes or rework.

The measurement program resulted in some startling findings. The diaphragm fit grooves were offset (upper from lower) and not square with each other. The upper case was longer than the lower case by almost 0.050 in. The HP section of the case lower half possessed a bow or hump in the HP diaphragm fits that reached a maximum of approximately 0.060 in. It was now clear to the project team why assemblers in previous overhauls had been unable to set the HP diaphragms low enough to match rotor centerline. The LP diaphragm fits had the same problem, to a lesser degree.

Re-Evaluation

The project team now had a clear understanding of the current casing condition. Instead of longitudinal "hump," the casing had a four-way "step." The vertical joint located between the HP and LP was misaligned. This created a mismatched four-way joint at the horizontal and vertical joint intersection. Although some longitudinal distortion did exist, most of the problem had been caused by the initial machining of the joint faces during manufacture. The vertical joint gaskets had, undoubtedly, been the result of an attempt to correct a four-way joint that was not square. Similarly, case ovality, though existing, was minor in nature.

The "hump" in the HP section diaphragm fit areas was also of concern. These bores were simply not in alignment with the rest of the turbine centerline. The casing exhaust end vertical joint that supports the bearing and packing area was out of square and off center.

Throughout its life, the turbine four-way joint had been forced square through the application of wedge-shaped gaskets in the vertical joints and through the efforts of maintenance crews by overtorquing the horizontal joint bolts. This turbine suffered from both thermally and mechanically induced stresses. The project team realized that even if all residual stresses were relaxed, the casing problems would still exist to a significant degree.

The project team decided to break the vertical joint in the upper half and evaluate the vertical joint. The joint face revealed only minor damage. The rabbet fit, however, was determined to have been machined off center. It was then decided to loosen the vertical joint bolts on the bottom half and evaluate the case from a true "crosshair" intersection perspective. This improved the turbine centerline difference between the HP and LP from 0.030 into 0.011 in. However, repositioning the vertical joint created a gap at the horizontal joint on either side of the vertical joint flange. This was caused by an upward "bow" in the HP casing.

The project team was faced with a formidable dilemma. The casing clearly suffered from thermal distortion. But, no amount of thermal manipulation could correct problems such as the top half of the case being longer than the bottom half or the joints being machined out of square. The project schedule now appeared to be in jeopardy. It was clear that the case was in need of both heat treatment and considerable corrective machining.

On the advice of the consultant, the team elected to attempt a complex compromise. The strategy was to seek the best all-around averages and then raise the entire casing to annealing temperature, relieve residual stresses, and allow some areas of the case to yield. This would be a thermal attempt to obtain an acceptable horizontal joint flatness and sealing capability. If this could be achieved, at least a significant portion of the boring could be eliminated.

The "step" in the vertical joints would be "slipped." The vertical joint bolts were loosened, top and bottom, and the HP and LP casings were adjusted for the best average to turbine centerline and horizontal joint gap. The vertical and horizontal bolts were then torqued to maximum and the casing was prepared for heat treatment.

Nozzle and Diaphragm Inspections

The alignment holes for the extraction housing in both upper and lower case halves were worn to an elongated condition and offset, possibly from previous repairs. The lower nozzle had some previously ground out partitions and some damage. NDE of the extraction nozzle partitions revealed a total of 15 cracked trailing edges. The upper and lower antirotation pins were both eroded and worn, and the hole on the upper half had to be reamed to clean up.

Visual and NDE inspection of the diaphragms revealed moderate foreign object damage to the partitions, light to moderate cracking of trailing edges, and medium to heavy wash out at most weld interfaces.

The diaphragm splitline locking devices were not as shown on the OEM drawing. Instead, they are a tab and screw arrangement that will allow the diaphragm to move vertically during transient thermal conditions. Because of this, the planned splitline constraint modification was determined not to be necessary. The addition of shim packs to the top and bottom of the diaphragms would not be required, since the turbine case has provisions for shimming the top and bottom.

Diaphragm numbers three through six had radial shroud bands, tack welded on, but no radial seals. Number seven diaphragm had a bolted-on shroud band, with two integral radial seals that showed some wear and tear. The radial tip seals in the double-flow stage were completely missing.

Bearing/Bearing Housing Inspections

One of the known problems was that the exhaust end bearing retainer required machining 0.040 in off-center to allow proper alignment to the seal housing. Since this bearing had previously been through a wreck, the bearing bore was assumed to be offset. This combination bearing and seal housing was removed and set up in a vertical mill. Measurements taken with dial indicators showed the bearing housing true to the seal housing. However, measurements also demonstrated the vertical face to be out of square by 0.045 in. Measurements on the casing matching flange also showed it out of square by a greater amount.

Valve Rack/Steam Chest Inspections

The control valves, gears, linkages, and cams were removed from the steam chest for separate cleaning and inspection. The steam chest was sandblasted and an NDE was performed. Only minor cracking was found at six previously seal-welded pressure taps on the lower upstream side of the large cylinder. The race track was in good condition, with no pitting or steam cutting evident. All six control valves and both extraction valves were completely disassembled, and the valve bodies were inspected. No indications were found. The valve seats in the steam chest had minor damage.

The steam chest was set up on a horizontal mill with the nozzle fingers up, and the race track mating area was indicated in as close to zero as possible. Then the nozzle faces and nozzle block bores were swept with an indicator to determine twist, relative position to each other, and perpendicularity to the race track. The outer faces were within 0.009 in TIR, and the inner faces within 0.008 in TIR, with only the last three to four inches at the horizontal splitline showing a significant increase, to −0.015 in right side, and −0.021 in left side. Most of this excess can be attributed to grooves caused by heavy rubbing at some time in the past; otherwise, the faces would be within the TIR noted previously.

Casing Heat Treatment—Theory

Steam turbines are subjected to large thermal differentials during transient operating modes such as during startup and shutdown. During startup, steam heats the inner surface of casings causing this material to want to expand, whereas the outer surface of the case remains cooler, remaining in compression. The thermal stresses induced into the turbine during normal operation cause the inner diameter of the casing to go into high tension, while the outer surface remains in lower tension.

Through years of operation, turbine casings are repeatedly subjected to these high thermally induced stresses, which often exceed the yield point of the material. Over time, some of these stresses become locked into the material due to local yielding of adjacent materials. This is particularly prevalent in adjacent areas of casings with significantly different wall thickness. These phenomena result in both casing distortion and residual casing stresses. The residual stresses, if left untreated, continue to increase in both magnitude and quantity.

As the stress levels continue to rise, yielding increases causing the casing to distort out-of-round. In some cases the horizontal joints begin to warp, either radially or longitudinally, and gaps open in the splitline. The joint gaps allow steam leakage, thereby reducing turbine output. Eventually, the increasing stress levels are manifest in either cracks and/or global casing distortion.

The thermal straightening process is simple in concept. New or additional thermal stresses are induced into the outer casing surfaces. The new stresses are designed to oppose the existing stresses that have built up over time. Process complexity lies in determining the precise location and quantity of stress that must be induced to achieve the desired result. Success depends on acquiring sufficient information to determine where to apply heat; how much heat to apply, and how long to apply the heat. The countering induced stresses must be sufficiently high so as to cause adjacent material to yield, moving the case material back into the desired positions and planes. In the instance of a distorted case, the goal would be to reround an out-of-round case, or reshape a case with longitudinal distortion.

Stress relief of the casing would use the following procedure. By heating the entire casing to the stress relief temperature, at which it will exhibit a lower yield strength, approximately three to four ksi, the residual stresses are allowed to relax through local yielding. This local yielding takes place as grains in the metal reorganize to a lower or stressfree orientation. Throughout this process, all stresses above the current temperature dependent yield strength are removed by relaxation. The post repair residual stresses that remain are below this value and will remain low until influenced by any high operating stress.

The important parameters to consider when performing an annealing or stress relief heat treatment are discussed later.

In many instances (such as the one being discussed), this may involve more than one set of properties, due to the case being constructed of different materials for high temperature, high pressure sections, and lower temperature, lower pressure sections. It is also important to understand material properties and heat treatment of associated attached components. The following information is required to determine heat treatment parameters:

- Thermal transients to achieve maximum temperature
- Allowable thermal gradient (axial and radial directions)
- Desired annealing temperature
- Yield strength at annealing temperature
- Hold time at annealing temperature
- Allowable thermal transients during cool down

It is necessary to consider the historical/operational rapid heating on the ID and its probable effects over unit life to date. These considerations must be combined with an understanding of the case configuration (thick- *vs* thin-walled sections) and case section operating temperatures.

Analyses will then lead to an understanding of residual stress levels and its locations within the case. Only when this understanding is reached can parameters for thermal manipulation and/or mechanical loading and motion constraints be set up.

Physical Setup and Operation

Casing heat treatment was performed in a 12 ft × 12 ft × 16 ft portable furnace (Figure 5). The furnace is capable of being fired electrically, or with natural or LP gas. For this application, LP gas was utilized as the firing medium. A primary air blower provided additional oxygen for combustion. Additional blowers provided circulation of the heated air. The furnace is fabricated from angle iron frames and expanded metal with detachable "blankets" or "pillows" manufactured from high temperature glass wool sewn into welding cloth. The hearth was constructed of fabricated piers and firebrick.

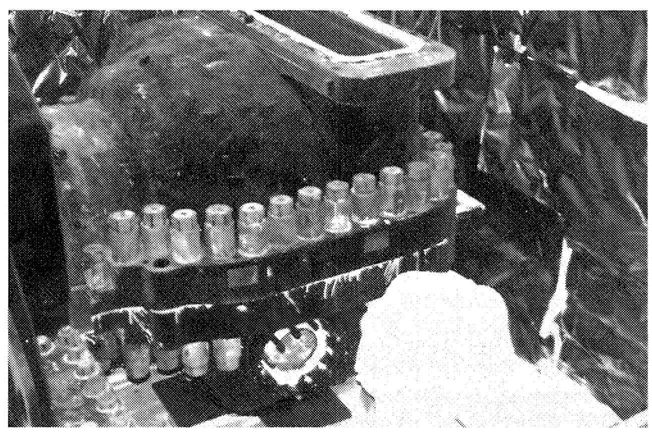

Figure 5. Casing Being Installed in Furnace.

Thermocouples were welded to the case in specific locations to monitor the actual metal temperature. Additional thermocouples were located in the furnace to monitor furnace air temperature. All locations were fitted with redundant thermocouples for security reasons. The heat treating process was controlled and monitored by a remote station (Figure 6).

Removal and reinstallation of the bolted on nozzle sections would have impacted overall project critical path. During the post teardown evaluation phase, the project team elected to perform weld repairs of the nozzle vanes using a GTAW process with Inco-82 filler material. Using this weld material eliminated the necessity of removing and replacing the upper and lower nozzle segments for post weld heat treatment. Performance of the nozzle repair *in-situ* also eliminated the necessity of replacing the nozzle segment bolts. Such bolts are routinely destroyed during the removal process.

Performance of the weld repair of the nozzle vanes using a 400 series stainless steel would have required post weld heat treatment of the nozzle section at a higher temperature than that planned for the case. In addition, the nozzle segment bolts are in close proximity to the nozzle vanes. These bolts were manufactured from

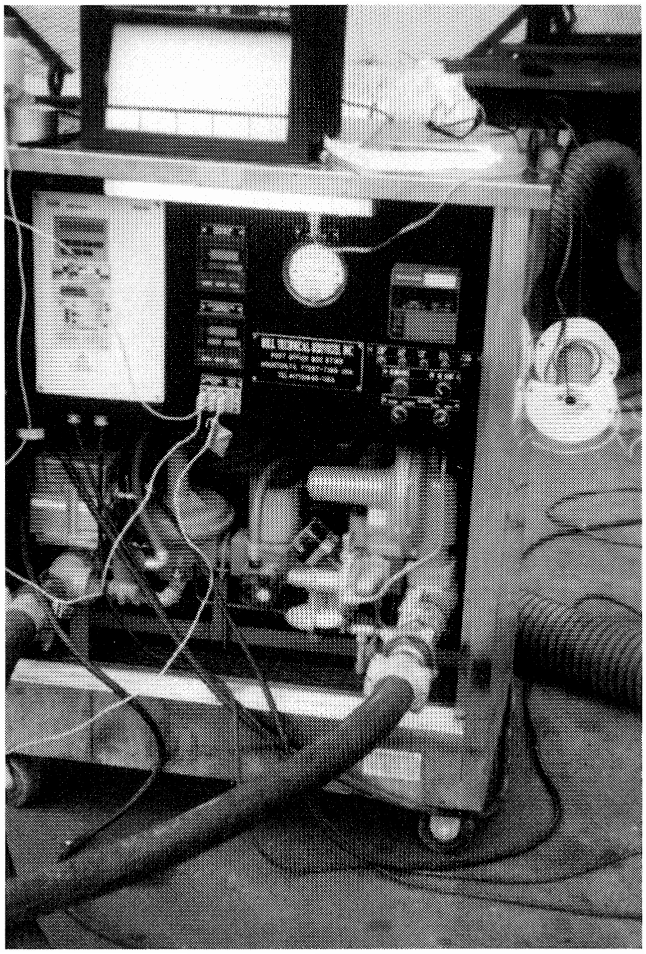

Figure 6. Furnace Control System.

a material that, if subjected to the planned casing heat treatment temperatures, could, potentially, be reduced in strength. Locally raising the temperature within the furnace to obtain proper post weld heat treatment temperature for the nozzle vanes would further exacerbate the risk to the nozzle segment bolts. Since the Inco-82 material did not require post weld heat treatment, it was decided to cool the nozzles and bolts during stress relief and keep them below their normal maximum operating temperature of 873°F. This was accomplished with insulation and cooling air (Figure 7). Thermocouples were attached to each nozzle and monitored throughout the cycle to ensure adequate cooling was achieved.

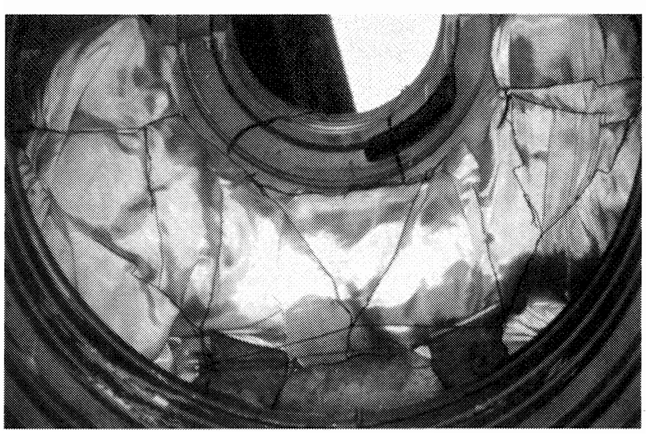

Figure 7. Cooling Passage.

Temperature ramps were controlled by two parameters: temperature differential per hour, and maximum and minimum casing temperatures recorded by the thermocouples.

Planned Results

Ideally, the desired result was that the casing would return to its original manufactured (neutral or zero stress) state after stress relief. This implies no casing distortion would remain and all residual stresses were removed from the casing. As a result, the horizontal joints would be in perfect contact with each other and the casing inner diameters would be perfectly round (assuming that the OEM manufactured it this way). Furthermore, all casing bolting, etc. (not removed or being replaced), would not be reduced below the acceptable material strength range for the component. All components would be able to be placed back into service.

Evaluation of data obtained during teardown and casing mapping modified some of the specific results desired. However, the overall desired results remained intact.

REVISIONS RESULTING FROM INSPECTIONS

Prior to stress relief, the shells were bolted together and tightened up with the old bolts. The unit was placed in the gas-fired furnace and brought up to annealing temperature where it was held for six hours. After the six hour period, the furnace temperature was lowered with a controlled cool down to 400°F. Total elapsed time was 30 hours (Figure 8).

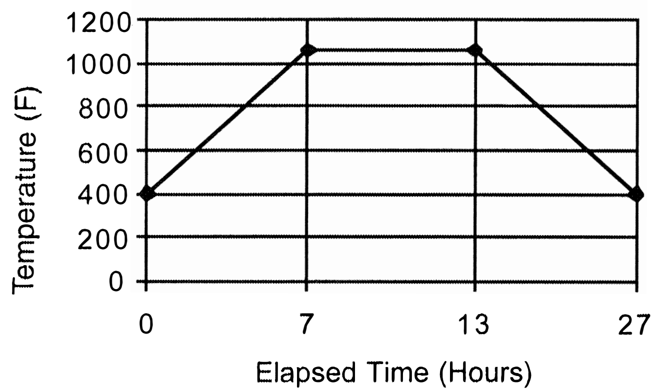

Figure 8. Stress Relief Temperature Chart.

• Objective 1—Take the vertical and horizontal splitline "offsets" into account and reduce the distortion in the upper and lower halves HP and LP casings as much as possible by heat treating. If possible, eliminate the necessity for horizontal and/or vertical joint machining. Eliminate thermal rerounding, as this operation was deemed unnecessary through a review of case mapping findings. Relax casing residual stresses to reduce the tendency for future distortion and to improve maintainability.

• Objective 2—Provide horizontal and vertical joint sealing without requiring over stretching the bolts throughout the length and diameters of the HP and LP casings.

• Objective 3—Position nozzle blocks in one plane that is perpendicular to the upper flange on the steam chest.

• Objective 4—During heat treatment, the temperature of the nozzle segment bolts in the HP lower half was not to exceed the maximum operating temperature of the turbine, when in service 873°F.

Actual Results Achieved

Following cool down of the case to ambient (Figure 9), the horizontal joint was unbolted and checked with feeler gauges. It was determined that the horizontal joint had been corrected to an acceptable state where approximately 25 percent of the OEM specified bolting torque would be required to obtain a metal-to-metal seal. Overall, the HP and LP gaps after heat treatment were reduced 0.005 in and 0.010 in, respectively. This was determined to be acceptable. Machining of the entire horizontal joint was not required. While unbolted, the horizontal joint gap was reduced by 0.005 in and 0.010 in, for the HP and LP, respectively, when compared to the prestress relieved condition. While bolted, no gap existed except in two small areas on the HP case near the vertical joint. The out-of-roundness of the casing was improved by 0.005 in and 0.002 in radially on the HP and LP, respectively. The LP experienced minor change, because initially it was only 0.009 in out-of-round, compared with the HPs 0.017 in radial distortion, on average. Also, the residual stresses were relaxed out of the casing, which should provide years of distortion-free operation, assuming normal operating procedures are followed.

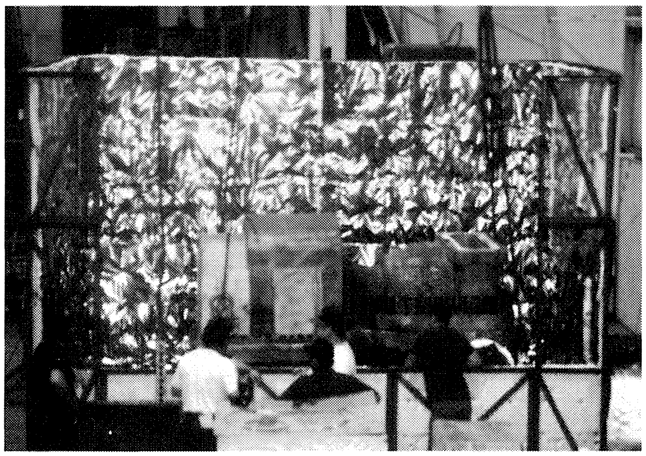

Figure 9. Turbine Case in Furnace After Stress Relief.

When the upper and lower cylinder halves were bolted together, the steam chest was positioned in the upper half. The nozzle block face and bore were indicated with a boring bar. The face was determined to be out-of-round radially by 0.009 in maximum; the bore was 0.005 in. Given these indications, thermal repositioning of the nozzle fingers was not needed due to the minimal twist and axial movement. Minor welding with 309L material was performed on the block face. This was subsequently machined to ensure a proper seal with the nozzle seal ring.

Feeler gauge readings were taken during the repair process. These checks were taken at various times with various amounts of bolt force. An illustration of the readings is shown in Figure 10, taken before and after repairs, on one side of one half of the HP case, in the bolted and unbolted state. The readings are typical of the decrease in horizontal joint gaps in the turbine section, as a result of the heat treatment process.

Before and after heat treatment, hardness tests were performed at various locations on the turbine casing and bolting hardware. This was to ensure that no unacceptable material property changes occurred throughout the repair process. The hardness readings were converted to an approximate, localized, ultimate tensile strength. Comparison of the before and after readings demonstrated that no significant changes in material properties resulted from casing heat treatment.

Casing Re-Evaluation

The casing halves were grit blasted again to remove the scaling resulting from the heat treatment process. All bolt holes were tapped to clean the threads. The splitlines were laser mapped again, and the results indicated that machining of the entire splitline

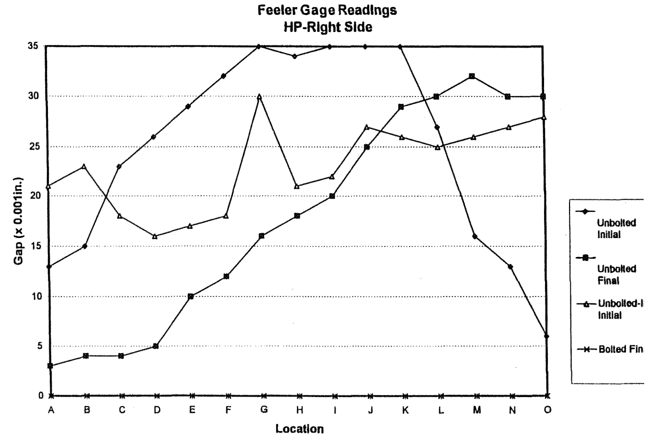

Figure 10. Casing Feeler Gauge Readings.

would not be necessary. It was decided to unbolt the lower case a the vertical joint. The gasket material was removed and the join was cleaned and inspected. A 0.025 in soft metal shim wa installed in the upper and lower vertical joints to compensat axially for the gasket that was removed. The new gasket was seale and all new B-16 studs were installed and torqued. The joints wer doweled in place.

There were, however, low areas on the LP casing at the HP/LI vertical joint intersection. A blue check during the top-on sea check confirmed the low areas. Later in the project, afte machining, these low areas were welded up and machined on th lower casing to eliminate the low spots as much as possible.

Although the splitline was not perfectly flat, it was easil brought to zero gap with very little bolt torque. However, due t the mismatch in the diaphragm grooves and the weld repair required due to erosion, every axial fit inside the turbine require machining.

Considerable debate surrounded the reassembly of the vertica joint. Replacement of the gasket was not an issue, but whether t leave the offset or make the horizontal joint flat was. After a review of all the data, it was determined the casing must be assembled an bored with an offset at the four-way joint. Complete removal of th offset would mean having to machine all bores within the case. I addition, the rabbet fit would require repositioning (throug machining) to allow for the required movement of the vertica joint. Since all of the vertical face fits already had to be machined it became clear to the project team that reproducing a perfec horizontal joint would not be possible without negatively affecting overall project schedule.

To minimize this work, a decision was made to reassemble th cases with only a 50 percent reduction in offset: to 0.014 in on th right and 0.017 in on the left.

Time Savings Achieved through Heat Treatment

If the casing heat treatment had not yielded acceptable results the horizontal joint would have required a full machining operation. Since the nozzle chambers "fingers" were integral wit the steam chest, machining material off the casing horizontal join would have changed the vertical position of the upper half of th nozzle in relation to the rotor centerline. To compensate for th change, the steam chest "race track" sealing surface would have required a weld buildup to raise the nozzle back into its original vertical position in relation to the rotor. A subsequent machining operation, utilizing a complex setup on a CNC horizontal milling machine programmed circle interpolation milling, would have been required.

In addition, milling the horizontal joint would have necessitated remachining virtually all the casing bores. Although the project team had prepared contingency plans for such a development, scheduling forecast was for an additional six days. Four of the extra days would not be critical path but the additional scope would have negatively affected overall project delivery by at least two days.

REPAIR ACTIVITIES

Casing Weld Repair

Machining a weld prep on all the casing groove axial fits was accomplished via a six inch portable boring bar. After preliminary machining, the cases were separated, and the fits that had been machined were repair welded. Weld repair in the high pressure end was accomplished using a GTAW method and ER 70s filler material. Filler material for the low pressure end was 309L. The 309L material was selected because of its better erosion resistance. Weld time was reduced by breaking the vertical joint. This allowed concurrent weld repair of all four pieces.

The LP diaphragm fits were welded (Figure 11) because little parent material was left on the support side of the fit. Crush pads were added to the LP bearing housing bores to compensate for the boring.

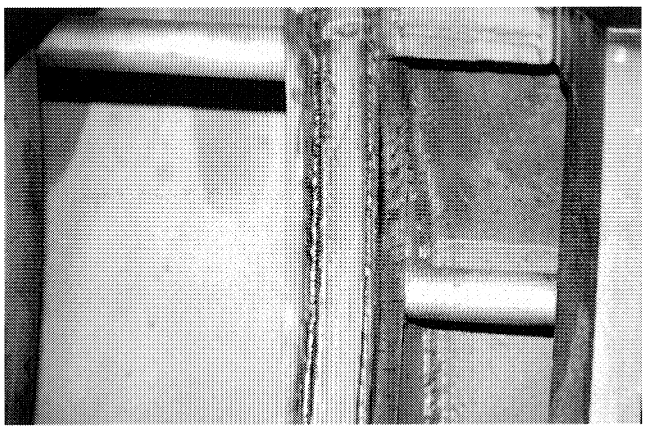

Figure 11. Diaphragm Fits Being Welded.

Case Machining

The casing was leveled and the boring bar installed (Figure 12). The top half was installed and adjusted to minimize material removed. Dowels were reamed and installed to hold this position. The steam chest and upper nozzle fingers were installed. These were moved as necessary to align with the centerline of the machine. A light skim cut on the nozzle face produced a surface perfectly perpendicular to the machine centerline. Since this one stage is responsible for one-third of the total horsepower, every effort was being made to produce a near-perfect fit.

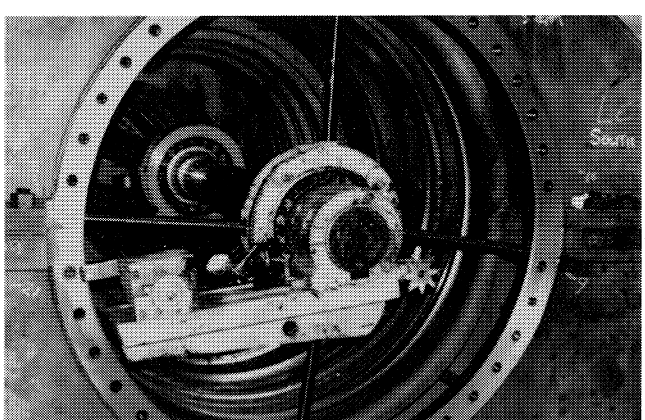

Figure 12. Boring Bar Installed in Casing.

The remaining cuts were straight boring bar work. The IDs were cut to relieve the residual center hump and the diaphragm groove faces were cut to achieve 100 percent cleanup (Figure 13). After each groove was cut, the matching diaphragm was cut to achieve the proper axial position.

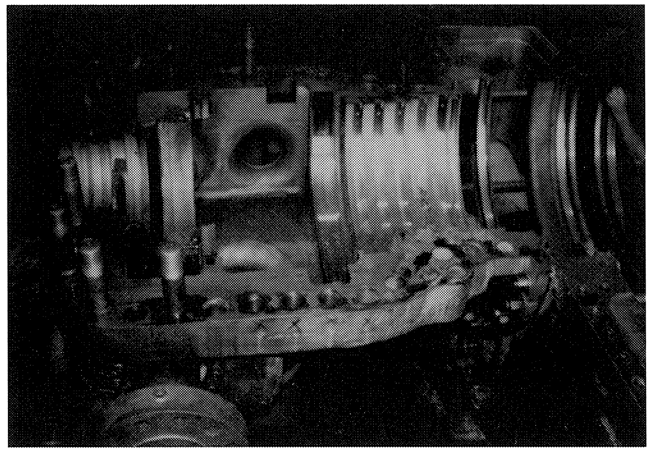

Figure 13. Upper Half of Case After Final Boring.

The bores were measured and swept with an indicator using a six inch boring bar supported on spiders in three places and optically setup to eliminate sag. The faces of the diaphragm fits, including the extraction stage, were machined square up with the rotor axis. Some of these bores required up to 0.050 in of material removal to true the faces.

Nozzle Repair

The nozzle segments were determined to be repairable in place, therefore, the nozzles were not removed (Figure 14). To ensure there were no unseen problems, the nozzles were checked with a fiber optic borescope, on the high pressure side. The results were recorded on video tape for future inspection and reference.

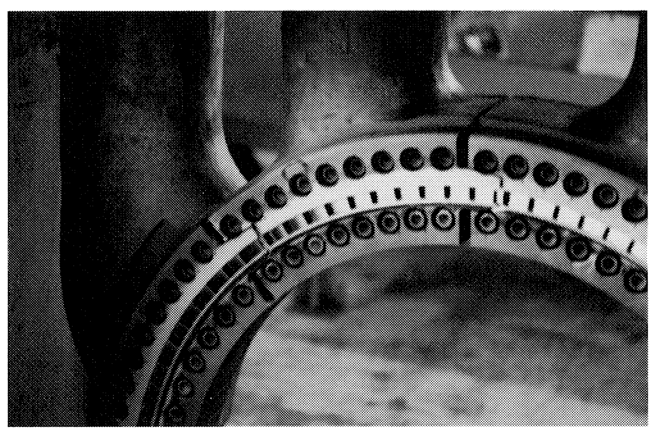

Figure 14. Repaired Nozzle Fingers.

The extraction nozzle was weld repaired to return its area to design (Figure 15). A 410 stainless steel GTAW process was used. The extraction nozzle ring was post weld heat treated along with the diaphragms. The antirotation/alignment pin hole for the extraction stage upper half was found elongated and had to be reamed to correct. A new stepped alignment pin was made from B-16 material and installed at final assembly.

Diaphragm Repairs

There are many, sometimes opposing, theories and procedures existing relative to repairing diaphragms and seals. The main

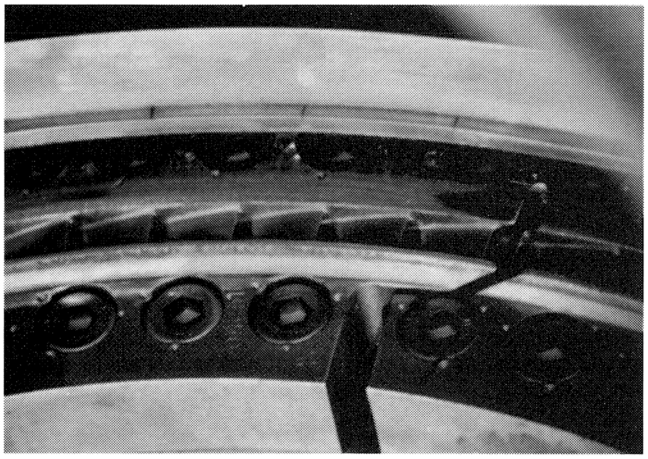

Figure 15. Closeup of Nozzle Vanes.

dissension revolves around the optimum exit edge thickness of diaphragm partitions. This difference of opinion concentrates on improved efficiency with thin edges (0.015 to 0.020 in) *vs* improved longevity in a solid particle erosive atmosphere with thick edges (0.040 in). The same argument is relative to the shape of seal teeth: the sharper the teeth, the more advantageous the flow coefficient; the thicker the tooth, the more resistant to solid particle erosion and foreign object damage. The project team reached its decisions by reviewing the current condition of steam path components and the past history of the turbine. Based on the location and nature of erosion found in the turbine, it was determined that the unit suffered more from wet steam erosion, downstream from the moisture line, than it did from solid particle erosion. For this reason, slightly thicker trailing edges (0.025 in) were opted for and a low stress design tooth was employed for the nozzle seals. Tip seals, which were integrally machined into bolted on seal rings were built rugged, capable of withstanding minor foreign object transit.

It was apparent from the opening appraisal that significant steam leakage had historically resulted from pathways created by the mismatch of the turbine internals. For this reason, each potential location for energy savings was investigated and corrected. Some of these aspects were:

- Diaphragm bore and hook fit tolerances
- Horizontal joint gaps
- Sealing face setback
- Steam seal face runout and erosion
- Crush pin fit
- Partition surface finish and contour
- Sidewall repairs
- Total and individual throat area
- Control throat openings, pressure side emphasis
- Distortion resolution
- Dowel pin corrections
- Seal edge corrections
- Keyway resolutions
- Root seal repairs
- Resolution of horizontal blade joint

All of the above items were taken into consideration and addressed in order to obtain the maximum improvement in steam path efficiency. The diaphragm partitions and set back faces were repaired, and the washed out welds were ground out and rewelded (Figure 16). Several required weld repairs and machining of the horizontal splitlines at the outer diameter. All the diaphragms were cut back and welded on the sealing faces, and all received crush pads on the opposite faces. This was done to save time at assembly, because all the diaphragm fits were welded and machined and brought to a uniform width.

Figure 16. Diaphragm Partitions Being Welded.

Bearing Housing Correction

From measurements of the bearing housing (Figure 17) and the casing, it was determined that the cause of the bearing misalignment was a severely warped vertical mounting face. Correction of this problem involved machining both vertical faces true and installing a soft metal shim to keep axial spacing the same (Figure 18).

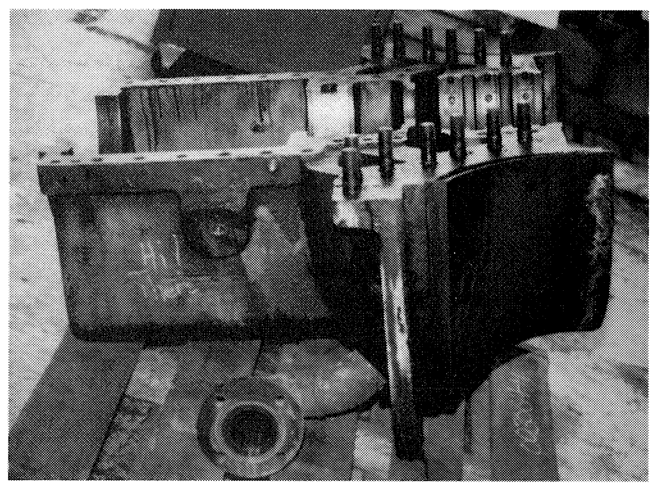

Figure 17. Incoming View of Bearing Housing.

Even though the bearing fit was true to the seal fits, it averaged 0.010 in out-of-round with several areas greater than 0.010 in. A total of 0.021 in on diameter was removed to achieve 100 percent cleanup.

The transverse casing-to-bearing housing keys were slightly undersize and were replaced with new keys, made to size after the key slots were cleaned and trued up. New tube-type spacers and clearance washers were made as well.

Valve Rack Rebuild

Past problems with governor valve stability and reliability led the customer to seek help from the OEM. The original valves (Figure 19) were a single seated Venturi type with a mushroom

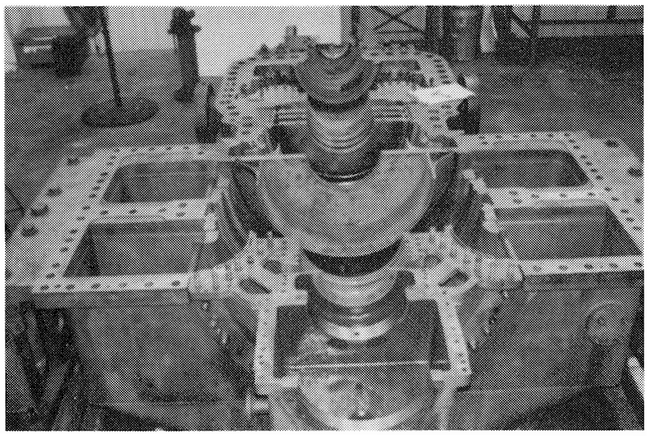

Figure 18. View of Repaired Bearing Housing.

shaped head. At certain lifts, or steam flows, the steam forces would become unbalanced and cause considerable vibration throughout the valve train. This led to premature wear on the valve gear components, instabilities in speed control, and frequent online maintenance. In addition, the inlet valves were attached to their lifting mechanism via threads on the valve stem. Rotation of the valve was prevented by a radial set screw that dogged into the threaded area of the stem. This antirotation device produced a stress riser and had become the initiating point of several past valve failures.

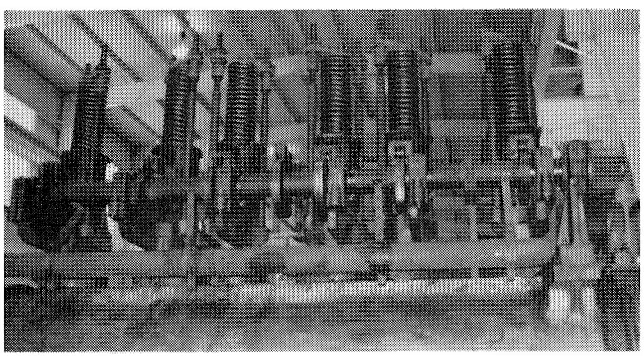

Figure 19. View of Valve Rack. Incoming.

The OEM offered a redesigned valve package (Figure 20) that addressed the problems that had been encountered. The revised valves were manufactured such that attachment to the stem is accomplished by a split clamp instead of threads. This eliminated the stress risers in the threads. Modifications to the valves included making the head straight with very little taper on the seat. Four small diameter holes were drilled through the valve body to aid in equalizing forces across the valve.

EFFICIENCY UPGRADES—RETRACTABLE PACKING—THEORY OF OPERATION

One area where the turbine was provided with upgraded components was the shaft packing, a critical area of potential leakage and efficiency loss. Historically, turbine designers have attempted to reduce shaft leakage through the installation of seals that reduce the flow coefficient by maintaining a close tolerance between the shaft and the turbine casing. Maintaining these close tolerances without incurring damage to the seals has been a continuous problem; a problem that seemed to have no practical solution until recently.

Reported results of upgrading to retractable packing vary and are affected by a number of significant conditions. These include, but are not limited to the following items:

Figure 20. View of Valve Rack. Final.

- Alignment of the turbine internals
- Repaired edge thickness, areas, finishes, and contours of diaphragm partitions
- Distortion of internals
- Condition of blade or bucket steam path
- Location of efficiency measuring instrumentation
- Interpretation of results
- The degree to which the turbine packing had been rubbed prior to installation of retractable packing

It is recognized that other improvements in the steam path to correct losses associated with erosion, mechanical damage, and deposits contribute to increased efficiency and/or output. However, in general, the typical improvement from the use of the advanced packing alone is calculated to be about one to two percent in heat rate and two to three percent in output.

Most turbine manufacturers design packing rings with springs designed to force the packing ring segments toward the shaft to a close clearance position. A typical conventional packing ring design (Figure 21) consists of six segments, each of which is held in place by a flat or, in some cases coil design, spring located behind the packing. The spring holds the segment in its minimum clearance position. There is, on the average, 0.025 in of clearance between the rotating shaft of the turbine and the stationary shaft packing.

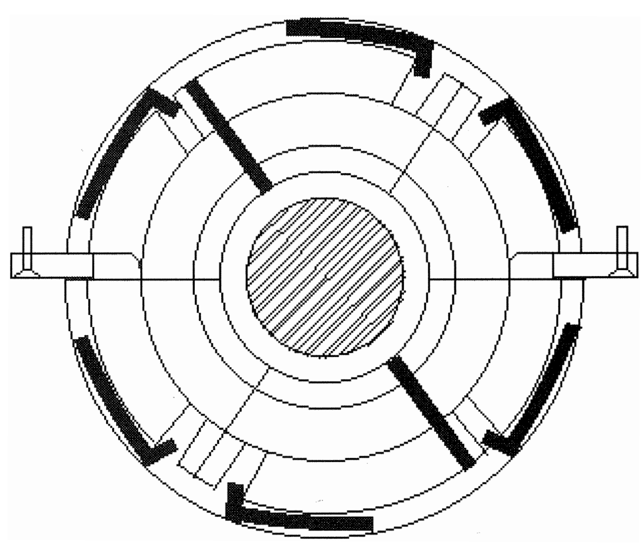

Figure 21. Conventional Packing.

Packing is designed so that, as steam flows past each tooth, there is a pressure drop. The pressure drop across the packing ring teeth is relatively linear in nature. Upstream steam pressure, and the spring force, act on the back side of the packing ring. Downstream pressure, past the steam joint face, also acts on the back side of the packing ring. By design, the forces acting in the closing direction will always be greater than those in the opening direction. This is true for either impulse or reaction design turbines.

A typical steam turbine diaphragm consists of the shaft packing, nozzles, and spill strips or tip seals. Historical data indicate that, in most units inspected, the radial spill strips or tip seals were rubbed approximately 50 percent more than the shaft packing. This is because spill strip design holds the seal rigidly in place, whereas the shaft packings are spring-backed. At low loads and steam pressure, the forces acting in a closing direction on the packing are lower. When rubbing occurs, the springs allow the packing ring segments to move away from the shaft to some degree. The conclusion may be drawn that, when rubbing on the seals takes place, it does so in a relatively short period of time. If the rubbing occurred over a long period, the teeth on the shaft packing would rub out the same amount as the tip seals.

As steam flow and pressure increase, the ability of the spring to allow the packing segment to move away from the shaft is significantly reduced. High flows eliminate radial movement of the packing segment, due to the steam force and the coefficient of friction that exists between the packing and the holder at the steam seal face. Under this condition, the shaft packing becomes, essentially, rigid and should rub just as much as the tip seals. However, since tip seals rub 50 percent more than shaft packings, another conclusion can be drawn: rubbing must occur at relatively low steam flows where flows acting to close the ring are small enough to allow some movement away from the shaft in the event of a rub. The two conclusions are: rubbing occurs in a relatively short period of time, and rubbing occurs at relatively low steam flows.

Thermal gradients are a major cause of packing rubs. Because turbines are made of rings split at the centerline, unequal heating during certain conditions causes distortions in these rings. As the inside of the ring is heated it expands faster than the outside of the ring. This causes the half ring to try to spread at the splitline. As the splitline area moves outward, the bottom moves upward toward the shaft. This reduces the packing clearance and is one of the causes of a rub.

Thermal gradients are not the only reason packing rubs take place. As the rotor is brought up in speed during startup, it approaches the first critical, the rotor's natural frequency. The resulting high vibration, and a naturally occurring V-shaped rotor bow, occur simultaneously with diaphragm distortion. This is the most likely time for a rub to occur. The casing and diaphragms are distorted and a reduced clearance condition exists on the bottom. The rotor is in high vibration and bowed. When a rub occurs at this point, the rotor high spot will rub the packing on the bottom creating a localized heating and increasing the rotor bow. When this occurs, the tip seals rub out along with the packing rings.

Logically, if packing clearances could be opened during the startup and shutdown period, then closed during normal load operation, most severe packing rubs could be avoided, and consequently, leakage and damage could be controlled. This situation has been achieved by the introduction of advanced retractable packing.

The advanced packing operates in a reverse fashion compared to standard packing. In standard packing, the spring pressure holds the packing toward the turbine shaft, and as the turbine load increases, the steam pressure adds force behind the packing. In retractable packing, the springs hold the packing away from the shaft, and as turbine load increases, the steam pressure behind the packing forces the packing toward the turbine shaft.

The basic concept used in the improved packing is that the pressure on the back of the ring is greater than the pressure on the toothed side of the ring. This pressure differential increases with throttle flow and can be used to overcome the spring and friction forces acting on the individual packing segments. By proper design of the coil springs between the packing segments, the pressure forces acting on the packing can be utilized to cause the packing to move from a large clearance to a small clearance at a predetermined flow condition.

The advanced packing design is shown in Figures 22 and 23. The spring, normally located behind the packing ring, has been removed and replaced with a coil spring located in a hole machined in the butt of the ring segment. The coil springs cause the packing ring segments to move radially away from the shaft, and take up the amount of space between the back side of the packing and the packing ring holder. This distance is normally 0.100 to 0.125 in. This dimension does not include design radial clearance (normally an additional 0.025 in).

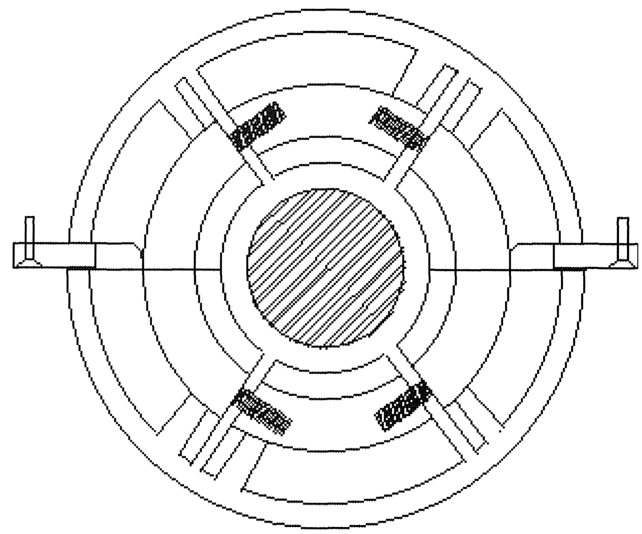

Figure 22. Retractable Packing.

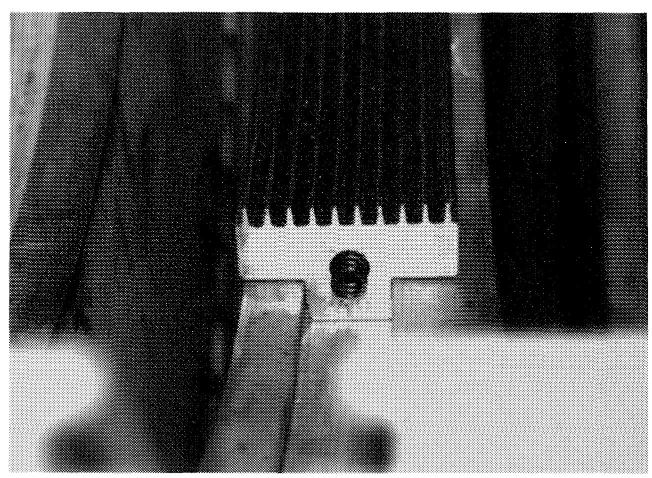

Figure 23. Closeup of Brandon Packing Segment.

In addition to the spring modification, a slot has been introduced on the back of the packing ring, at the center of each segment. The slot is designed to allow additional upstream pressure behind the ring segments. As steam flow increases through the turbine, steam pressure behind the packing ring will overcome the spring and friction forces, thus causing the packing ring to close. In each stage, the packing is designed to close well beyond the point where thermal gradients occur and well above the critical speed range.

No major rework to the turbine is required to fit the improved packing. This allows the installation of the improved packing to be treated as a standard part replacement rather than a modification. The size of the spring and the depth of the holes drilled in the packing rings are predetermined based on the expected pressures around the packing.

Why this Turbine

An increasingly competitive ethylene market led to a desire, on the part of the user, to reduce steam turbine operating costs. These costs were reflected in the high cost maintenance and also in fuel due to the decreased efficiency of the aging turbine. Over time, distortion and damage to turbine components had increased steam leakage and were believed to be the major factors resulting in lost efficiency. Industry evidence indicates that steam leakage alone can account for as much as 80 percent of the efficiency losses in a turbine. Past history also indicated high vibration when passing through the critical speeds. It was felt that rubbing packing contributed to this high vibration.

An innovative program was conceived to restore steam turbine efficiency without major modifications to components. New turbine seal technology coupled with improved repair methods were applied to enhance the efficiency gains.

The following discussion will identify the particular areas of the turbine steam path that were considered critical to performance and describe the changes made in these areas to improve efficiency.

Prior to the outage, no accurate mechanism existed to measure turbine performance. What operators were sure of, however, was that the process was turbine limited. The planned retrofit with new compressor internals would further exacerbate this problem. Accurate horsepower measurement equipment was installed during the turnaround, allowing proper verification and interpretation of test results. In line with the philosophy of continuing improvement programs, a practical approach to unit testing was developed. The goal of the program is to document and evaluate turbine performance trends. Some aspects of the program are:

- Review plant operations and procedures to confirm measurement technique.
- Review condition of existing instrumentation.
- Recommend added or upgraded instrumentation.
- Install instrumentation.
- Evaluate post outage performance data.

As part of the initial evaluation and because of the effect of cyclic thermal variations, the project team elected to measure observed distortion of internal parts. The temporary distortion causing thermal variations, most prevalent during transient operating conditions, eventually "take a set," becoming permanent. It was believed that the resulting distortion (out-of-round condition or longitudinal hump) was a major cause of excessive internal leakage and an accompanying loss of efficiency. Recorded measurements consisted of the following items:

- Critical rotor diameters
- Packing casing bores
- Diaphragm bores
- Potential tip seal bores

The historical data review resulted in the customer providing the following data package to vendor engineering.

- Turbine heat balance(s)
- Turbine cross-section drawing
- Past inspection reports, as found clearance data
- Past maintenance records and past overhaul reports
- Past corrective action maintenance records

Review of the turbine maintenance history was one of the most important aspects of project planning. The corrective maintenance records provided useful information that pointed out trends of component degradation. Problems encountered during past overhauls could be expected again. Realizing that past problems tend to be repeated, if not recognized, the team studied each problem until understood, and then devised means for their future prevention. The study was also helpful in identifying where the potential disassembly delays might occur. The review allowed the customer team members to make practical suggestions for improving the disassembly/reassembly process.

Opening Appraisal

The appraisal began prior to the removal of the rotor. Important horizontal joint gap measurements were taken as the bolts were loosened. Radial packing and tip seal clearance measurements were taken and recorded. These readings must be taken before the rotor is removed. The appraisal value was greatly enhanced by the full participation of the customer's machinery group who benefitted from having been involved in previous overhauls. Awareness of sources of loss, analyses of observed conditions, investigation into potential improvements, and the probability of success provided incentives to obtain maximum information in minimal time. The educational value of such participation was an aspect of opportunity for all project team members, but more important was the continued presence of persons knowledgeable of the basis for suggested improvements. Still a further reason for such participation was the often observed benefit that two or three engineers, of divergent experience, discussing any obscure phenomena, can often solve the puzzle, whereas a single observer might overlook important data.

A thorough examination of the critical areas in the steam path was essential to make informed judgments about the efficacy of planned steam path repairs and upgrades. Data derived during teardown were used in making subsequent determinations about the need to change direction or methods. All concerned knew that exceptional effort had been made at this point to perform a complete and detailed observation of every critical area and component. Each variation or out-of-character detail had to be noted. It was felt that thoroughness at this point would save time later during the remanufacturing period. The steam path examination included the following activities:

- Examine the quality of critical aerodynamic components.
- Identify unusual causes of performance loss.
- Identify and photograph expected causes of loss.
    - Identify losses caused by mechanical damage.
    - Identify losses caused by steam path deposits.
    - Identify losses caused by erosion.
    - Identify losses caused by excessive leakage.
- Determine extent of distortion problems.

Once the casing examination was complete, a critical evaluation of the data obtained during the examination was performed. This evaluation addressed all potential causes for the phenomena observed during the examination. The steam path appraisal was used to review and confirm planned methods for improving efficiency. While all critical areas of the turbine were subject to evaluation, some of the areas under review and consideration were the following:

- Tighter packing seal clearances
- Design and installation of tip seals
- Upgrade of components in the seal areas

Historical data indicated that severe rubs had occurred during startups. The project team believed that there were two causes of seal damage and that most of the damage was occurring during turbine startup and shutdown. The causes were believed to be vibration and distortion. It was known that, during startup and shutdown the turbine was susceptible to vibration as the rotor was brought through its critical speeds. The team surmised that even though the end bearing vibration may not be excessive, the more critical center span was subject to large deflection. In addition to the vibration, diaphragms and packing boxes, which hold the stationary packing rings, were believed to be subjected to large temperature differentials during startup. It was believed that this condition was causing the normally round packing ring holders becoming egg shaped. The assumption was that the distortion, when combined with rotor deflection, lead to reduced clearances, which resulted in packing rubs. It was possible that the resulting packing rubs were creating heat and further bowing the shaft, exacerbating the packing rub.

More important than the temporary distortion, most of the stationary turbine parts and the turbine case were believed to have become permanently distorted.

A total of 10 rows of retractable packing were installed in the HP, N2, and LP sections of the turbine (Figure 24). It was decided to install retractable packing throughout the turbine, including the LP section. The outer rings (N1 and N3) were the exception. If retractable packing had been installed in the outboard ring, on either end of the turbine, dirt and oil could be drawn into the seal system during startup and shutdown when the packing rings were operating at maximum clearance. The rings inboard of the outer rings were also left in the conventional design to maintain proper steam sealing and steam header pressure. Retractable packing was installed in the inboard ring of the HP packing gland.

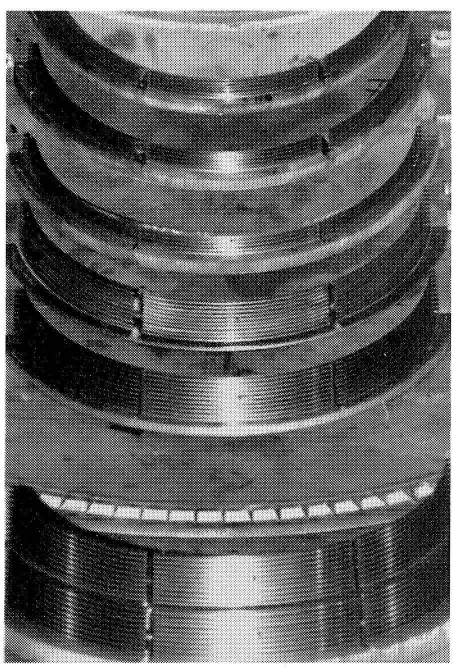

Figure 24. Brandon Packing Rings Installed in Diaphragms.

Nozzle Seal Modifications

The original turbine design provides a ring mounted in the case that forms a shroud or cover over the first stage rotor shroud, and is designed to have little or no axial clearance to the nozzle outer face. To this ring, a radial seal was added.

The case bore that holds this ring was off square, and would not allow the desired clearance to be obtained. To correct this condition, the ring itself was modified by machining, welding, and hand dressing to fit the nozzle with 0.008 to 0.010 in axial clearance at ambient temperature.

A radial seal was added to the extraction nozzle as well. The project team felt that estimated shell pressure in this second stage area more than justified the addition of a radial seal. A seal groove was machined in the extraction nozzle ring during the vane repair process. The nozzle seal was then manufactured, installed, and machined to the proper clearance.

Tip Seal Upgrades

The project team decided that a decrease in nozzle seal clearance and the addition of tip seals would be possible, due to the expected avoidance of a bowed rotor (normally caused by packing rubs). The improvement from the addition of tip seals may be expected to exceed the benefits of improved diaphragm packing clearances.

Originally, the seventh stage diaphragm had a bolted-on ring with radial seals (Figure 25). This ring was replaced with a new ring of the same design, but tighter clearance (Figure 26). The third through sixth stages were modified to accept bolted-on radial seal rings. The application of radial seal rings reduced blade tip sealing clearance to less than half that of original design in every stage. In several stages, the resulting clearances were less than 25 percent of design.

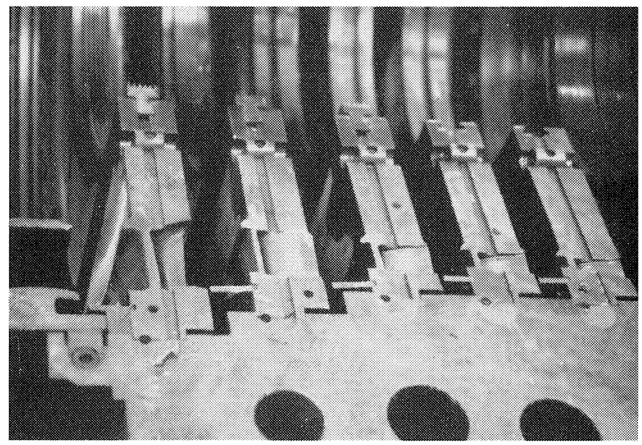

Figure 25. Old Diaphragm Seal Rings.

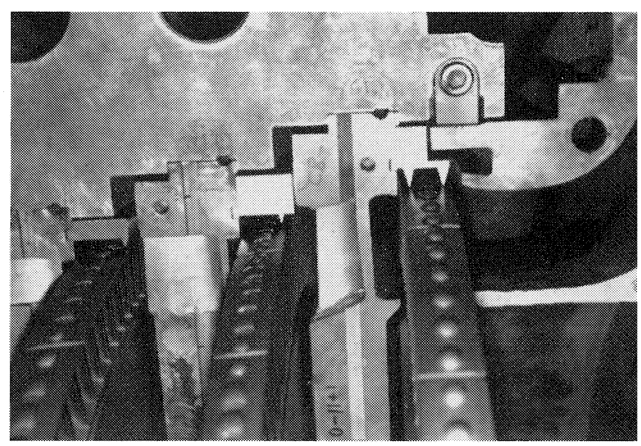

Figure 26. New Seal Rings.

REASSEMBLY

The luxury of assembling a turbine inside a machine shop was a treat for the customer members of the project team. From the perspective of the other team members, the opportunity to rebuild an entire turbine in just over three weeks was an exciting and satisfying experience. Each component was custom fitted to ensure perfect alignment. The result was a high pressure nozzle ring perfectly square and concentric to the first stage rotating blades

All axial spacing was per design and, for the first time, the seal bores were all concentric with the rotor. This also allowed all seal clearances to be set on the low side of the tolerance band.

Because the diaphragm fits were bored to a 100 percent radial clearance, the diaphragms could now be set up without wedging them into the shell. This correction allowed the diaphragms to grow vertically, during the transition from ambient to operating temperature, while the packing rings were retracted and seal clearances were large.

Each diaphragm was indicated in to center, then shimmed at the bottom, to limit total vertical movement, and at the sides to center it on the shim. Horizontal clearance was held to 0.003 to 0.005 in total. The upper halves were centered, shimmed, set to mate with the lower half joint and form a proper seal.

The HP packing housing was relocated axially, to sit 0.045 in closer to the thrust end from the original location. This was performed to center the seals in the shaft grooves. Original design packings were installed in the outer HP packing grooves and all three packing grooves on the LP end.

All horizontal and vertical splitline bolting was replaced with new B-16 studs (Figures 27 and 28), nuts, and washers before final assembly.

Figure 27. New Bolting.

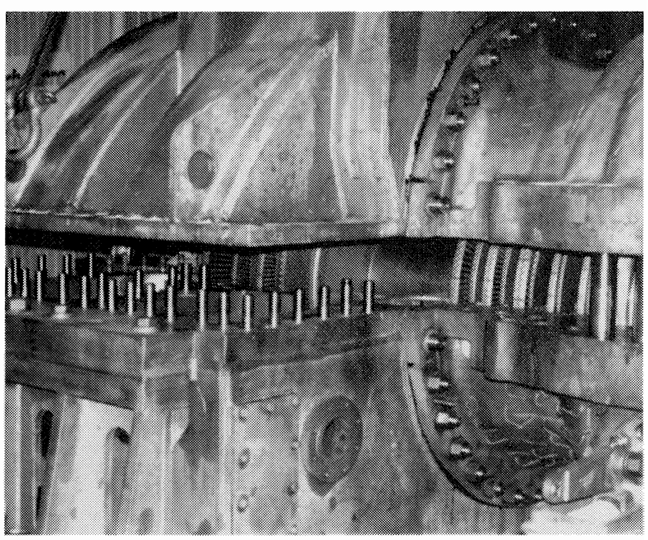

Figure 28. Turbine Case Bolting.

After assembly, the steam chest was doweled. The dowels are angled and pass through a void. It was not clear at that stage how effective they would be. Because of this, small steel blocks were installed via welding. During subsequent overhauls, these can be measured against one another, on all four corners of the steam chest.

The bearing caps were not sealed and the RTDs were not set, because the shipping shims would still require removal. The HP wind back seal and shield were shipped separately. All external openings were blanked, and the case was primed and painted with high temperature gray paint (Figure 29). The complete remanufactured turbine was then transported to the customer for installation (Figure 30).

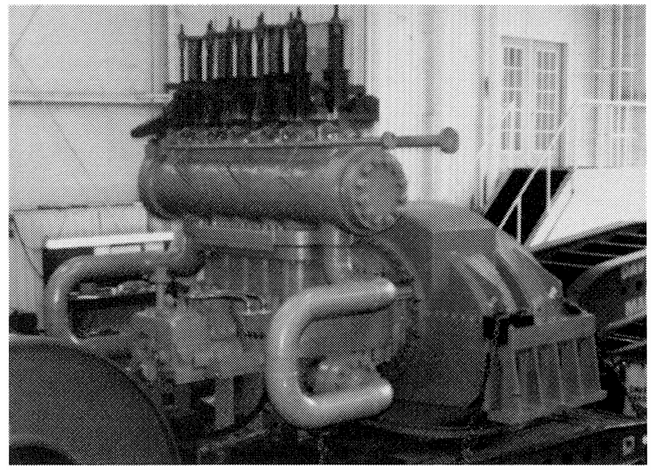

Figure 29. Turbine on Truck.

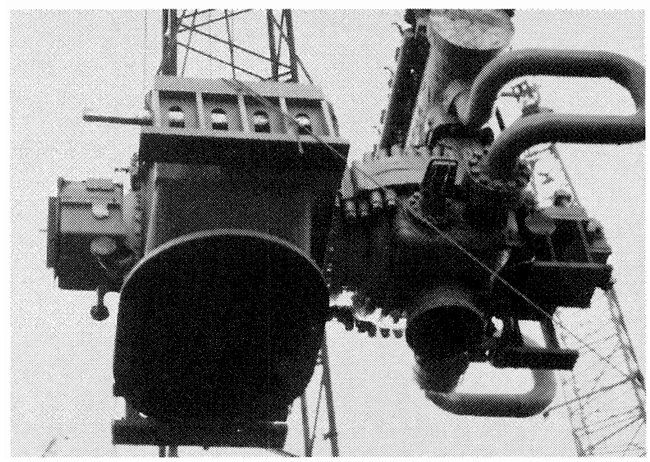

Figure 30. Turbine Installation.

STARTUP AND RESULTS

Startup was not without incident. During the reassembly, the high pressure packing was found mismatched to the corresponding rotor lands. The packing appeared to be too far downstream. This was corrected and proved to be a mistake. It is believed that starting the turbine, with 650 psi steam admitted into the extraction nozzle area, was the cause of the problem. The extraction nozzle is behind the first stage, almost in the midspan of the rotor. It is probable that the startup steam caused the rotor to rise in temperature much faster than the thick walled high pressure casing. The high differential in thermal expansion caused a “rotor long” scenario, which, in turn, caused an axial high pressure packing rub. While the high vibration caused some anxious moments, everything cleared up after the casing was allowed an extended heat soak. With each successive outage, more is learned about the quirks of a particular turbine installation.

CONCLUSIONS

Performance since startup has exceeded expectations. No external steam leak is the first obvious sign of success. The HP splitline and the steam chest to casing joint had been chronic leakers. Due to compressor problems, unit horsepower requirements are much higher than design. The recently installed torque monitor coupling between the turbine and first compressor shows an excess of 35,000 hp when the machine is under full load. Considering the turbine nameplate is 27,600 hp, the performance is appreciated. The turbine produces this power while running at maximum continuous speed with rotor vibration less than 0.5 mil. The new governor valves have shown no sign of instability throughout their operating range. Previously, at certain flows, the valves would shake so hard as to destroy the linkage.

In light of current compressor performance problems, all the turbine performance enhancements have proven worth the effort. Distortion and erosion correction, along with the retractable packing, steam path repairs, blade tip seals, and new governor valves have joined forces to make a marginal turbine into a real power house. The turbine should easily tolerate another 20 years of use and abuse.

Stress relieving the case proved to be the single most important factor in meeting the required schedule. Instead of a gamble, as originally envisioned, the stress relieving operation was an application of practical engineering. Careful evaluation of all parameters, customizing a detailed procedure, and strict adherence to this procedure saved at least a week of machining and aggravation.

This project required a cooperative effort of all parties involved. Early agreement of project goals and the atmosphere that everyone's input added value produced a win-win situation. The customer got a remanufactured turbine in better than new condition with no impact on the turnaround schedule. The vendor reaped financial rewards for his efforts, plus valuable experience for future remanufacture jobs.

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