

## MAJOR REVAMP AND RETROFIT OF IDENTICAL PROCESS COMPRESSOR TRAINS

by

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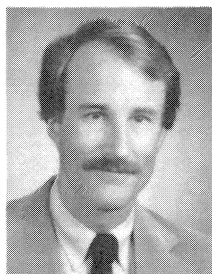
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flow increase. The first train was modified in April 1986, and the second was modified in August 1986. An additional extensive uprate of our existing steam turbine driver to almost double the design power rating will push performance even beyond the envisioned maximum rate.

This project was developed and managed by plant engineers. This paper documents the development, manufacturing, and startup phases of both the performance uprate as well as a seal system redesign for increased reliability. The mechanical redesign was primarily done by the original equipment manufacturers (OEMs). Contributions from corporate specialists, after market suppliers and independent consultants were also vital to the project's success. Measures taken to ensure the performance of the contractors and confidence in the reliability of the end products are described. The mechanical performance is up to company expectations now, but not without first overcoming a few obstacles. Actual operating rates are beyond expectations, primarily due to the effect of liquid carryover into the process compressor. Troubleshooting and performance testing techniques are described.

### INTRODUCTION

Two identical compressor trains are in a continuous full production mode at a Green Lake, Texas chemical plant. The process is an exothermic reaction of air, ammonia, and propylene, which not only creates the main product, acrylonitrile, but also generates large quantities of high pressure steam. This steam drives both compressor trains as well as a turbogenerator. Each train is composed of a steam turbine, an air compressor, a process compressor, and an expander (Figure 1). The process compressor has two sections with an intercooler. The train is mounted on concrete pedestals with downward piping connections for ease of maintenance. A common lube oil skid services the whole train. The unit has been in operation for over five years. The equipment is completely instrumented with vibration probes and bearing imbedded thermocouples. Both process and surge control are maintained by our plant process computer. The train is inline, driven with 30 in spacer, diaphragm couplings between each case. Spare rotors are stored for each machine and an entire spare process compressor is kept ready for an emergency changeout. The air compressor employs suction throttling and discharge blowoff for surge control. The process compressor has suction throttling and two independent recycle loops to prevent surge conditions. Production rates are pushed to the limit at all times. Shutdowns are infrequent and kept to the shortest duration possible. The company operating philosophy is based on squeezing as much production as possible from the existing equipment. The online operating philosophy requires the most cost effective, reliable method of

### ABSTRACT

Major uprates of process compressor train performance can be achieved utilizing the existing machinery. The following is a broad overview of a project to revamp two identical compressor trains to achieve levels well over original design flow. Significant internal modifications to an air compressor, a process compressor, and an expander have achieved most of our planned

increasing production with the existing equipment and the shortest production disruption.

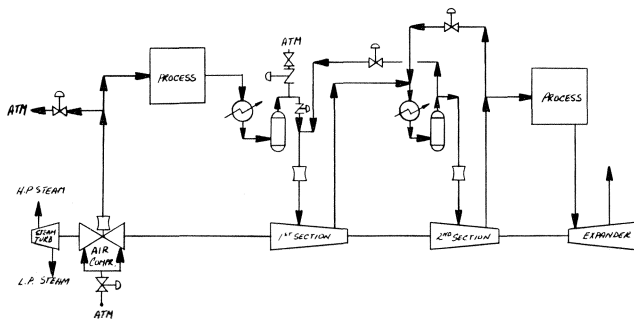


Figure 1. Compressor Train Process Flow Diagram.

## HISTORY

The compressors were designed and specified in 1978, and built in 1979 and 1980. Initial designs were concerned with both the quality and quantity of steam available for startup and the ability to operate at reduced rates without the gas expander. The steam turbine was sized to exceed the required horsepower requirement for the train at rated conditions.

Precommissioning of the compressor trains began in the summer of 1981. Startup in December 1981, went smoothly with one exception. The gas expanders all experienced blade failures on the rotating third stage, due to vibratory fatigue. A successfully redesigned blade with a thicker profile, fewer stress risers, and other design improvements was quickly provided by the OEM and installed.

The following years of operation have been without major problems. However, a number of significant problems presenting some interesting challenges to plant maintenance engineering have developed. The process compressor had a tendency to accumulate hard friable solids in low gas velocity "dead" spaces. These solids built up, eventually contacting the rotor and causing wear damage to the rotor labyrinth seal areas. Seal failure occurred periodically. This problem has been solved by a seal redesign described later. Another problem, liquid slugging, has occurred only once. The suspected cause was accumulated water in the process compressor inlet lines, slugging the compressor on startup or at low speed. The coupling between the air and process compressor failed after the coupling hubs had spun and galled to the shafts. Later a slight bow with a torsional twist was discovered in both compressor shafts.

The steam turbine experienced recurrent split-line leaks in the high pressure section of the case. Online leak repair was only partially successful. The turbines have seen an intermittent subsynchronous vibration at the natural frequency of the rotor. This has been attributed to seal rubbing. In addition, a second subsynchronous vibration component visible in the spectrum analysis and evidenced by a slowly bouncing vibration monitor needle had appeared. Increasing the crush on the spherical seat of the self-aligning, tilt-pad bearings by 0.002 in over the manufacturer's value has solved the problem.

The compressor trains have been fairly reliable overall. The extra capacity capability put into the steam turbines during the design stage allowed the plant to run at rates slightly exceeding design for the past several years. It was a desire to further increase production that germinated the eventual revamp of the machinery to safely accommodate to the new high flow of design rates.

## REVAMP

Operation of the compressor train was repeatedly pushed to the known operating "limits." Several of these "limits," when investigated with the manufacturer were well below the actual design limitations. After removing as many artificial limitations as possible, the plant engineers were faced with a 550 psi maximum allowable, turbine, first stage, pressure limitation. At this point, the expander flow limits had been exceeded. The expander recovered about 62 percent of the power provided to drive the process compressor. The high inlet pressures required to pass the required flow across the expander was nearing the pressure rating limitations of some of the upstream pressure vessels. The process system was capable of passing flow far in excess of what the expander could handle.

Once the limits of the equipment had been reached, the focusing switched to the complete system, and it was found that the horsepower in the process compressor and the expander were not effectively utilized. The plant engineering staff performed tests on the process system to derive pressure vs capacity relationships. They pointed out that a significant portion of the horsepower used to compress the process gas was unnecessary. Using this data, a project was proposed that would use less power to achieve the same production rate. The power savings translated into lower steam consumption at our maximum throughput. Excess steam could be put through an existing turbine generator. The projected increase in electrical revenues was the initial basis for project justification.

## IMPLEMENTATION

In defining the scope of modifications to the compressor train, several objectives were established: reuse of the existing cases; achievement of the highest capacity possible; using the existing cases; maintain minimum turndown; ability to meet startup conditions; dynamically stable revamped rotors as proven by computer simulations; and, torsionally stable train as proven by a computer simulation.

The first step in redesigning the compressors and expander was definition of the pressure drop and flow requirements. Tests were performed to develop the pressure requirements at various production rates. Other tests were used to derive empirical formulas for the pressure losses through the piping and equipment. From this data, an ideal expander pressure versus flow curve was developed.

Work started with the expander manufacturer to define the limits of gas flow through the existing casing with decreased inlet pressure and increased inlet temperature conditions. The OEM's design investigation revealed that the existing casing could be used to give the desired flow at the specified pressure and temperature. New internals and rotor would be required. There would be an efficiency loss with an associated degree of uncertainty due to the very high gas inlet velocities.

During this development work, plant engineers were engaged simultaneously in discussions with industry consultants, the OEMs, some independent repair shops, and other manufacturers about their ability to support this project. A need was determined to evaluate the industry alternatives, and see if there were any better ideas in order to be assured of getting a fair price. Based on this evaluation, the company established confidence in the OEMs' abilities, ideas, and the cost fairness. In the final analysis, the OEMs were considered best able to support this project. They had a primary advantage over other competitors, due to their detailed knowledge of the existing machine internals. Using their input, rough definition of the project and its scope were obtained. This included the total replacement of all internals in the air compressor, the process compressor, and the gas expander.

The OEMs were contracted to provide a complete design package. This included final internal component selection, hydraulic performance, mechanical design stability, a firm scope of supply and cost quotation. These contracts were negotiated incorporating a defined scope for each study, the specific results to be obtained, a maximum cost exposure, and a provision for crediting the cost of the studies towards any hardware purchase. Contracting the design work to the OEMs assured them that this project had the plant management's support, and, therefore, that the engineers were not asking them to perform free engineering development work.

Throughout the conception, definition, execution, and operational evaluation of this project, a number of different resources were consulted. Those resources included in-house technical specialists, the original equipment manufacturers, other equipment manufacturers, independent repair shops, and independent engineering consultants. The plant engineers felt it was important to maintain outside and independent industrial contacts, to increase the number of options available and to ensure the quality of the final product.

For each vendor, a complete and detailed scope of supply was submitted as a part of the purchase order for the engineering studies and the revamped hardware. These scopes of supply were invaluable as a communication tool between buyer and seller and between engineer and purchasing agent. Each scope was developed or reviewed in joint meetings which started with a statement of intent. The scope of supply was the final word clarifying the company's needs and the vendor's responsibility as manufacturing problems developed. The various scopes also aided project closeout as a checklist to ensure all the expected material, including the consumable spare parts, critical speed analyses, drawings, etc., was supplied as requested. Outside independent consultants were employed to review the OEM's engineering work in areas considered critical or as having high potential risk. In this project, no discrepancies were uncovered; however, it was felt that independent analysis of critical items is a necessary ingredient for ensuring high reliability of the final product.

## STEAM TURBINE

Original performance guarantee point of the steam turbine was just under 6000 hp at 4854 rpm. The steam conditions were rated at 650 psig, 625°F inlet to 55 psig exhaust into the plant low pressure steam header.

The turbine itself is a ten stage, multivalve, noncondensing steam turbine. It has an electronic governor with hydraulic actuator control. Overspeed trips are both mechanical and electronic. The bearings are all tilt pad, lubricated by a common lube oil console. Imbedded thermocouples in the pads provide temperature information. Proximity probes send vibration signals to a control room monitor.

The steam turbine was not modified during the April revamp. After the startup of the revamped first train, it was determined that more driving horsepower was desired. Future uprate of the steam turbine to well over double its original rating had been discussed with the manufacturer. An uprate of this magnitude would require a complete new set of internals. Modifications may include reducing the number of stages from ten to four or five. In addition, blades would be lengthened to fill the maximum internal diameter of 25 in from the current 20 in diameter. The shaft may be salvageable; but that is doubtful, since the wider wheels require longer keys. A larger trip and throttle valve may also have been required. The major design constraints are the first stage pressure limitation and the internal diameter of the case. There would be some efficiency loss. These modifications are major and very expensive. In addition, process engineers felt that other areas of the plant needed

debottlenecking prior to achieving maximum compressor train output. The major turbine uprate was postponed.

As a temporary measure, the manufacturer suggested removing the second stage blading and diaphragms. Although this measure sacrificed an estimated 1.5 percent efficiency, it allowed considerably higher flow, and therefore, higher horsepower within the first stage case pressure limit. This step was in line with desired horsepower supply requirements and certainly within the budget. Temporary horsepower ratings were at 9000 hp at 5200 rpm. The plant is currently exceeding that quoted rating by an additional ten percent. First stage case pressure has dropped to about 530 psig.

The manufacturing step of the uprate was simple and quick. Airfoils were machined off and the rotor was rebalanced. Blade roots were left in the disk to allow reblading of the stage, if necessary.

Valve cams on the first steam turbine were reset to new positions per the manufacturer's instructions. Valves on the second steam turbine were not reset. At startup, it was possible to pass more steam through the second turbine. Upon inspection, valves on the first train did not open fully on full governor travel. The cams were reset again to the original position to allow full steam admission.

One particularly troublesome problem was finally solved. Previously, steam leaks on the casing horizontal splitline at the high pressure end had been very troublesome. Leak repair was done while operating with a sealant injection into the leaking area by a contractor. The frequency of repair was unacceptable, as was the possibility of a shutdown in the event of a large unstopable leak. Four months prior to the August shutdown, a complete strategy had been developed for eliminating the leaks. First, the splitline was inspected and honed flat to eliminate all burrs, scratches, and raised surfaces. A "wire drawn" erosion line was carefully welded and honed flat. A lead wire check was then performed. Shim stock, 0.005 in thick, and one amp lead fuse wire, 0.015 in thick, were located along the splitline. The top half of the case was then installed with one third of the bolts tightened to approximately one half of the required torque load. The lid was then removed and the lead wire thickness was checked with a micrometer. Forty points were checked with 90 percent being within one thousandth of the shim stock thickness. Maximum deviation was only 0.003 in. The manufacturer's allowable deviation was 0.005 in. Satisfied that the flatness was not the problem, the plant engineers used asbestos tape and triple boiled linseed oil to seal the splitline and replaced the lid.

All new B-16 grade bolts were then installed in both cases. The rounded ends of the bolts were machined flat, top and bottom. The case was then reassembled. A contract bolting service company was hired to hydraulically tighten the bolts. An ultrasonic transducer was used to check for proper bolt tension. The manufacturer's recommended tension of 60,000 psi was used for all bolts. Two teams of service representatives started on the low pressure end, simultaneously bolting each side until finished. Previously, millwrights with hammer wrenches just beat the bolts as tight as they could get them. Low profile hydraulic wrenches provided even torque when ultrasonically monitored. To date, only one minor steam leak has developed. The even application of the specified torque to the new bolts has apparently solved the problem.

## AIR COMPRESSOR

The original performance criteria delivered several thousand pounds per minute of atmospheric air to our process system at about 30 psia discharge pressure. The rated condition was 105 percent mass flow, although the plant often exceeded 125 per-

cent mass flow. The new performance targets for the compressor train were based on maximizing the mass flow of air to the reactor.

The air compressor is of the double flow design having one 36 in discharge and two 36 in suction lines (Figure 2). The case is rated for a maximum air flow of 100,000 icfm. It is supported on four integral feet, two fixed and two sliding, which allow for axial expansion. A total of six closed welded impellers, three to a side, had diameters of 36, 34 and 36 in, respectively, and were made of 4330 alloy steel. Impellers are keyed and have a light shrink fit. The shaft is made of 4340 steel, has a 101 in bearing span and an 8 7/8 in diameter. Shaft sleeves are 12 percent to 14 percent chrome steel and the seals are of the aluminum labyrinth type. All of the bearings are tilt pad type and pressure lubricated. The case is cast iron, ASTM A247 Class 40 and the diaphragms are cast iron, ASTM-A278 Class 30 material. The coupling is a dry diaphragm type with a 30 in spacer. It has a 2.0 service factor. The hubs are taper fit and hydraulically mounted. Although the coupling guard is non-lubricated, it is totally enclosed and open to oil spray from the bearing housings. These coupling guards have been a source of nuisance oil leaks since the original startup. The assembled compressor weighs in at 50 tons.

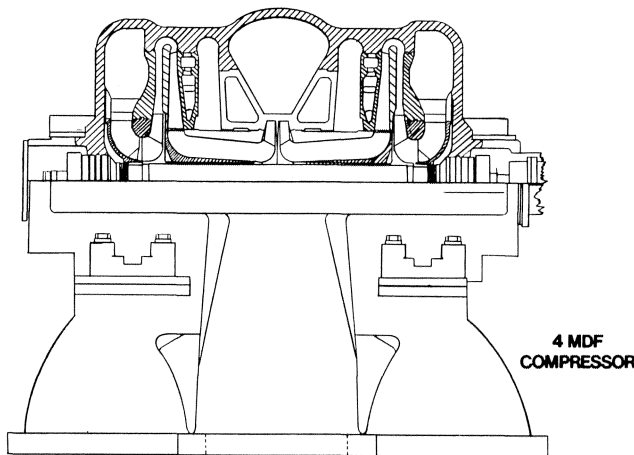


Figure 2. Air Compressor Cross Section.

Up-rated performance of the air compressor is the primary factor in boosting the production rate. The modification raised air rates significantly (Figure 3). Normal discharge pressures increased slightly, primarily due to additional friction loss at higher flow rates.

Physical changes to the compressor were limited to elements in the flow path. Original impeller diameters of 36, 34 and 36 in were enlarged to 38, 36 and 32 in diameters. Impeller eye diameters were increased at each stage to allow more flow. Six of the original impellers were adaptable to the new service. They were trimmed to a 32 in diameter. Two operating rotors and the common spare were modified. Twelve new impellers were ordered along with three sets of spacers, two sets of shaft seals and two sets of inlet guide vanes.

The company supply policy had required the use of four 410 stainless steel impeller forgings that had been stored in reserve at the manufacturer's warehouse. After the new forgings arrived, the manufacturer checked the stored 410 SS forgings, only to find that they were only 37 in in diameter, and could not be used to make 38 in impellers. The stainless steel forgings stayed in the storehouse while two new 38 in 4330 forgings were ordered. Negotiations concerning the manufacturer's responsibility to pay for the mistake ended in a stalemate. Two sets of diaphragms from each compressor were salvaged and

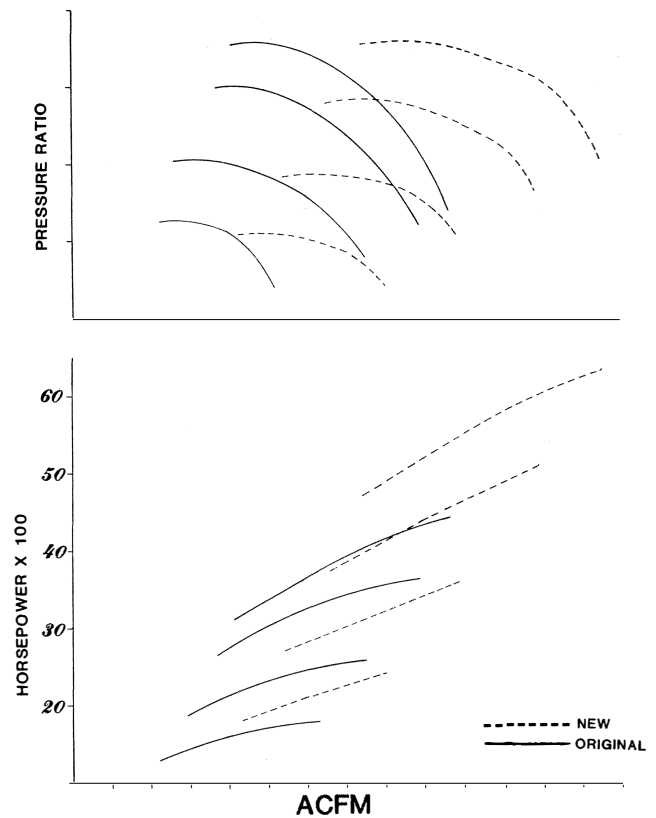


Figure 3. Air Compressor Performance Curves.

modified for reuse in the new up-rated design. A complete set of new diaphragms was ordered for the first train, to shorten the turnaround time for the revamp. The diaphragms from the first train were modified and installed in the second train. The second train diaphragms were modified, preserved, and put in storage as spares.

All forgings were required to have at least four test coupons adequately identified for future use by the plant in weld repair procedures. Documentation of the new impellers included material certification, heat treatment data, mechanical test results (which included yield strength, tensile strength, elongation and reduction of area data, and several Brinnell hardness readings), dimensional checks of critical dimensions both before and after the overspeed test, magnetic particle inspection after overspeed, dye penetrant inspection after overspeed, and final inspections.

A number of questions came up concerning discrepancies between the company's specification for material hardness and the manufacturer's specifications. These were resolved as being due to differences in the as quenched hardness for AISI 4330 and 410 stainless steel tempered to identical mechanical strength levels. Both metals were quenched to produce 100 percent martensitic structures and then tempered to meet the desired mechanical properties. The hardness of the martensite is a direct function of the carbon content. This explained the manufacturer's deviation (270 BHN) from the specification, which has a minimum yield strength of 80,000 psi and a Brinnell hardness number (BHN) of 190 to 235 for the second stage impellers. The first stage impellers were all AISI 4340 with a specified minimum yield of 90,000 psi and a range of 235 to 285 BHN, which matched the manufacturer's hardness criteria. However, in reviewing the documentation, values of 228 BHN, below the specified minimum value 235 BHN had been approved. The manufacturer's explanation was that the mechan-

ical properties are the primary concern. Hardness tests are simply a process control device to assure against manufacturing or heat treating deviations. The company specialist agreed that the deviation from the minimum was not felt to be significant.

All of the final rotor assembly was done at the manufacturer's repair shop. No modifications to the case, seal system, or bearings were required. Completed parts were shipped to the plant where they were sorted and organized to reduce confusion and downtime during the plant shutdown.

### PROCESS GAS COMPRESSOR

The process gas compressor was extensively revamped, both mechanically and aerodynamically. The compressor is a two section machine with an intercooler. The process gas is primarily nitrogen with water vapor and various hydrocarbons. Solids tend to build up in low velocity areas on the impellers and in the compressor seals. Original design flow was several thousand pounds per minute with some liquid knockout at the interstage cooler. Rated speed was just under 5000 rpm. Inlet temperatures were rated at 105°F due to the hot and humid Texas Gulf Coast summers. Discharge pressure approached 50 psig. Polytropic efficiency was about 70 percent for section one and 77 percent for section two.

The compressor has a horizontally split casing, with five stages in two sections (Figure 4). The casing has three integral

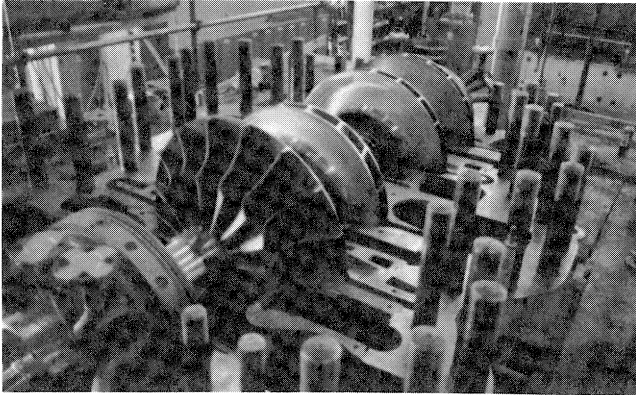


Figure 4. Process Compressor.

feet, two fixed and one wobble foot to allow axial thermal growth. The suction and discharge flanges are 36 in and 30 in for section one and 30 in and 20 in for section two. The maximum flow capacity for the case was 60,000 acfm. The case itself is made of cast steel while the diaphragms are cast iron with stainless steel inserts at the impeller discharge volute. There were five stages, two in section one and three in section two. The first section, stage one, had an open 38 in cast impeller. The other four stages were enclosed welded impellers. All of the impellers were made of 410 stainless steel. The shaft has stainless steel sleeves and impeller spacers. It has a diameter of 8 7/8 in and a between-bearing span of 110 in. The bearings are all of the tilt pad type. Thrust bearing rated loading was 264 psi with a maximum allowable of 500 psi. The compressor weighs in at over 50 tons, dangerously more than the manufacturer listed 36 tons. Initial buffer gas requirements to the shaft labyrinth seal system were stated at 4.2 lb/min at one psia for each outer seal and 10.1 lb/min at five psia for the discharge end seal.

Performance changes were significant for the uprate (Figure 5). Flow was increased significantly through the compressor while discharge pressure was reduced. The maximum continuous operating speed was raised to 5300 rpm.

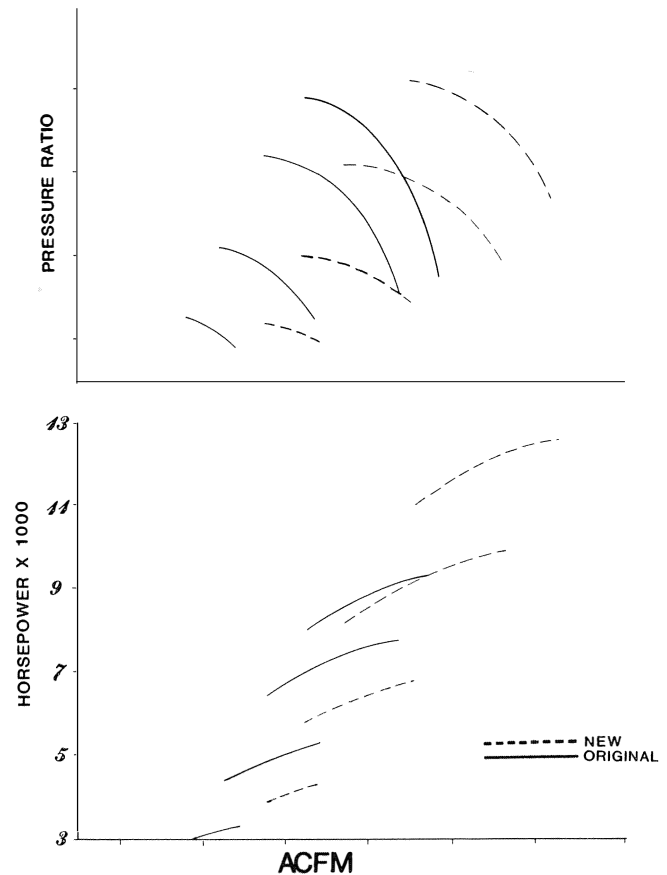


Figure 5. Process Compressor Performance Curves.

Mechanically, the changes have been sweeping. The redesign of the sealing system and addition of the balance piston are covered in a separate section. The aerodynamic section was extensively revamped. Bearings and the casing remained the same. Two relatively minor changes were made to the casing bore. The shafts were reused and several of the diaphragms were modified.

The manufacturer's scope of supply included all new impellers, sleeves, spacers, balance pistons, impeller eye and interstage Fluorosint 500® seal strips, and inlet guide vanes, modifications of reuseable diaphragms, and new replacement diaphragms. A complete set of new diaphragms was ordered to shorten the first turnaround schedule. Each set of parts was listed and described in the scope of supply to eliminate any confusion. All of the radial and thrust bearings were reuseable. All spare parts of a new design were also included in this scope of supply. The company deemed it important to have spare parts on hand during startup.

The new first stage impeller was 44 in in diameter and had open construction. The material was 410 stainless steel and was specified to have a maximum strength between 70,000 psi and 90,000 psi and a 22 Rockwell C hardness factor. The second stage impeller was 38 in in diameter and had a closed construction. The specified material was 17-4 PH with a minimum yield strength of 100,000 psi. Maximum continuous speed was 5300 rpm. This second stage impeller was the limiting factor in setting the maximum continuous speed for the train. Hardness was to be 33 Rockwell C and a 255 to 311 Brinell hardness number. Rotating labyrinth teeth were manufactured on the impeller eyes. The third and fourth stage impellers were similar to each other in physical characteristics, with 40 inch diameters and closed construction. The material was 17-4 PH with a

minimum yield strength of 90,000 psi, a Rockwell C of 33, and a 255 to 311 Brinell hardness number. All 17-4 PH impellers were to meet NACE MR-01-75 specifications. A fourth "back-up" forging was ordered for the second stage, due to expected difficulty in meeting the NACE specification and a 100,000 psi minimum yield strength.

The manufacturer supplied impeller spacers and shaft sleeves were made of 410 stainless steel per company design drawings. The sleeves and spacers have rotating labyrinth teeth. The sleeves were to be Kanigen® coated for extra wear resistance. The new balance piston was also made of 410 stainless steel and was designed by the manufacturer.

Meetings were held between plant engineers and the manufacturer after the award of the purchase order, to assure agreement on all points to the written scope of work, and to review design engineering. Particular attention was paid to the surge and stonewall characteristics of the compressors and the surge control system. The surge control system was sized and specified by the compressor manufacturer with design input from plant engineers. The schedule was also a primary concern of the discussion. Based on a promised delivery date of 2/24/86 for all materials to the manufacturer's Houston repair shop for assembly, a react or shutdown was scheduled 4/1/86 for modification of the first train. All manufactured materials for all of the equipment were to be shipped by 3/10/86. This deadline was missed. Several parts for the April 1986 shutdown had to be pushed through on an emergency basis to meet the shutdown schedule. Parts for the second compressor were delayed several months, and barely arrived in time for the August 1986 shutdown and revamp. Parts for the spare process compressor were not received until February 1987. At this writing, spare compressor parts ordered in the original scope of supply still have not been received. Fortunately, the needed delivery of the items deviated significantly from the the company's original expectation.

The manufacture of the components proved to be very difficult. Seal sleeves that were to be Kanigen® coated were reportedly done improperly several times by the manufacturer's sub-vendor. Finally, they were acceptably coated by a local vendor in Houston. Major manufacturing problems developed with the impellers. The process gas inherently contains various quantities of hydrogen cyanide, carbon dioxide, and water vapor. Alone, these chemicals cause no problem for 17-4 PH material. The presence of traces of chlorides or sulfides in the process gas, however, would cause stress corrosion cracking of the impellers. The 17-4 PH material was chosen due to its higher strength which allowed for higher maximum continuous speed for the compressor. Since 17-4 PH impellers were a new experience for the company, a survey was taken of current users in process gas applications. No particular problems were uncovered. The manufacturer stated that it would be difficult to meet the NACE specification and requested that the impellers be heat treated for a minimum yield strength of 105,000 psi and a hardness range of 277 to 352 Brinell. Their justification was that the NACE specification normally applied to gases containing hydrogen sulfide. Their request was not accepted due to the probability of sulfide contaminants at up to 10 ppm in the feedstocks. In addition, the corporate metallurgist felt that there was no good reason why a quality heat treatment shop could not meet both requirements and that if the material did not meet the specifications, it could be reheated.

The first problem appeared with the chemical analysis of the 17-4 forgings. Chromium content was only 14.73 percent which was below the 15 percent to 17.5 percent specified for the 17-4 PH material. The chromium content was in the range of 15-5 PH stainless steel. These forgings were accepted on the basis that mechanical properties, erosion resistance and weldability

of both materials were equivalent and that replacement forgings would have lengthened the delivery schedule.

This was just the start of material problems for the manufacturer. Four of the thirteen impellers that were ordered had to be scrapped. Although involvement of plant engineers was specifically requested in all major repairs, they were not aware of any of the problems with the impellers until after the schedule had unexpectedly slipped several weeks. Obtaining detailed information from the vendor regarding the problems that they were having was difficult. Witnessed inspection of the components by plant personnel was not performed prior to shipment, due to the distance to the manufacturer's shop and the fact that the impellers were inspected and shipped several days, even weeks, apart. Quality assurance documentation often arrived well after the rotors had already been stacked at the manufacturer's repair shop. For instance, documentation for the second stage impeller did not arrive until a week before its scheduled installation in the first train. The impeller documentation showed that Brinell hardness readings were 321 BHN, above the stated maximum of 311 BHN. A Brinell hardness tester was brought into the plant. Twenty readings were taken on the disk and cover. Four of the readings on the disk ranged from 320 to 341 BHN. The manufacturer was notified that the company was rejecting that impeller. Unfortunately, one of the three impellers had already been declared scrap due to 0.008 in oval discontinuity in the bore. The choices were to accept the impeller, attempt a repair of the oval bore, or wait several weeks until the third impeller was finished. The impeller was conditionally accepted, with assurances from the manufacturer that stress corrosion cracking would not occur. Close inspection of the impeller for any signs of stress corrosion cracking was scheduled for the next available opportunity.

In addition, the first stage wheel, an open impeller made of 410 stainless steel, was distorted beyond the possibility of salvage during heat treatment. More problems affected the fourth stage wheels. Reportedly, one impeller came apart on the overspeed test. A second impeller was found to have cracks along the blade weld seam after overspeed. The manufacturer's inspection revealed that the cracking was due to either slag or some other defect in the first pass weld. Structural design capability for the impeller was calculated to be 7000 rpm. The manufacturer cut out the cracked areas and rewelded them. They were then checked with magnetic particle and dye penetrant inspection. Unfortunately, the impeller had already been finished machined. The reheat treatment caused a wavy distortion in the impeller and it had to be scrapped. The third impeller was rushed through manufacturing to meet the schedule while two additional impeller forgings were ordered. With the new weld procedure, the problems were eliminated and subsequent impellers passed the overspeed test without any signs of cracking.

Problems were also encountered in the retrofit of the reusable diaphragms. New impeller outer diameters required machining of diaphragm surfaces and the installation of new wider diameter stainless steel inserts. In a few instances, the casting thickness was non-uniform and machining the grooves for the inserts broke through the diaphragm wall. These holes were carefully evaluated. Since they did not effect either structural or aerodynamic integrity, they were not repaired.

During stacking, one of the effluent compressor shafts was found to be bowed from a suspected liquid slug. Shaft bow was 0.0015 in. A vertical stress relief procedure failed to significantly reduce the shaft bow. Other methods of straightening seemed more likely to increase the bow than straighten the shaft. The rotor was restacked without modifying the shaft. The air compressor end of the shaft had been damaged. The coupling hub had spun on the shaft prior to the failure of the coupling diaphragm. The hub was galled to the shaft and had to be cut off.

The manufacturer's official recommendation was to scrap the shaft. The company, after consulting with the Houston repair shop, elected to chrome plate the affected area. After restacking the shaft and the sleeves, the chrome plated journals and hub areas were ground on an offset axis, to reduce the effect of the bow, while maintaining a fair degree of concentricity. The stacked rotor was then balanced and shipped. The manufacturer recalculated stresses on the hub for lower coefficient of friction between the chrome coated shaft and the hub. It was found to still be within an acceptable safety factor for the uprated conditions. Typically, the diaphragm is designed to break first. Spinning of the hub on the shaft prior to diaphragm breakage was attributed to a possible mistake in the method of checking the hydraulic hub for proper location and fit. O-rings should be left out for the first hub position and contact checks.

When all of the materials were received onsite, the spare process compressor was rebuilt in the plant maintenance shop. A number of minor problems surfaced. The first concerned modification of the case. A four by one inch notch and a shoulder bevel had to be cut in the compressor bore. The manufacturer sent a field crew with a portable boring mill to do the job. The work took over two weeks of ten and twelve hour days to finish. It also appeared to be an unsafe working condition for the machinists. Upon inspection, the cut was of poor quality, and had to be ground out with a hand grinder in order to properly fit the diaphragm. All subsequent case work was done by a local machine shop in their large horizontal boring mill. The local shop only took two days, had much better quality, and, in the final evaluation, cost less.

Diaphragm fit problems surfaced with the second case, particularly in fitting the discharge wall. Amazingly, some of the bores and axial dimensions in the top half of the case did not match the bottom half casing. Some diaphragms and the discharge wall had to be specially machined to fit properly. This was not a particularly difficult problem to solve, although it puts the cases' interchangeability with spare diaphragms in question.

The field installation of the new compressors was relatively straightforward. The entire case was swapped out at once so that only piping boltup and shaft alignment needed to be done. There were also some minor piping additions that were done for the new buffer gas system.

## EXPANDER

Original performance design of the expander was over 3000 horsepower (hp) at 4854 rpm. Process gas normal conditions were rated at over 50 psia and 650°F inlet to an exhaust pressure of 15.5 psia. Flow at the rated pressures and horsepower was near the design limitations.

The original design was a four stage Rateau blading, integral rotor in a cast steel casing. The casing consisted of two different steam turbine exhaust end casings bolted together. One end was fixed while the other end had a "wobble foot" to allow for axial thermal expansion. The inlet and exhaust flanges were 20 in and 30 in, respectively. The shaft was sealed by several sets of springloaded, labyrinth packing with a 4.5 lb/hr. nitrogen buffer gas injection. The first critical speed was 3200 rpm. The rotor was made of forged alloy steel with stainless steel buckets and shrouds. The rotor has a 64 in bearing span, and weighed 1400 lbs. There were three interstage case drains between the diaphragms. The expander was insulated with foil backed, teflon faced, stainless steel mesh enclosed, custom-fitted blankets.

The uprated design reduced inlet pressure to under 50 psia and increased potential inlet temperature an extra 100°F. This, coupled with the fact that the uprate significantly increased mass flow, required a significant increase in flow capacity. The

higher mass flow ratio was primarily responsible for an increase in the horsepower rating to over 4000 hp at 5545 rpm. raising the inlet temperature from 650°F to 750°F while keeping the mass flow constant increases horsepower output by an estimated 12 percent. Exhaust pressure increased about one to two percent due to increased flow rates.

Mechanically, only the rotors and diaphragms were changed (see Figure 6). The casing, bearings, and labyrinth seals remained the same. Three new integral three stage rotors were manufactured, two for the process trains and one common spare. Two new sets of diaphragms were also manufactured. Although the rotor and diaphragms fit into the same relative envelope, the changes were quite significant. The blades for the new rotors had a cross section roughly eight times larger, were longer, and considerably stronger than the originals. Blade material is 403 stainless steel. There were less blades on the new rotor, approximately 60 per stage. The blades were riveted into a shroud, five or six to a packet. The number of stationary blades also decreased while the flow area increased several times. Only one casing modification had to be made. One internal circumferential shoulder had to be manually ground off at a 45 degree angle to accommodate the middle diaphragm (Figure 7).

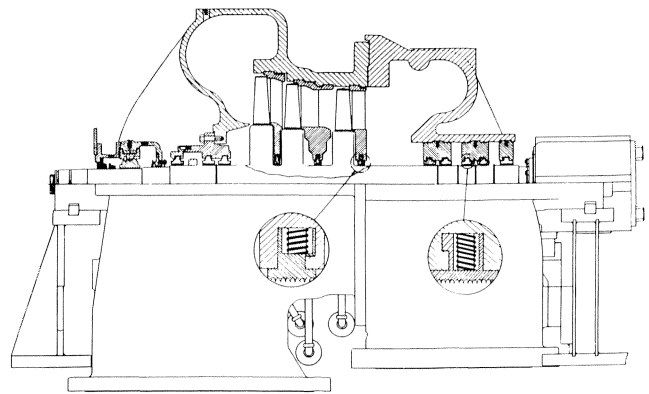


Figure 6. Expander Cross Section.

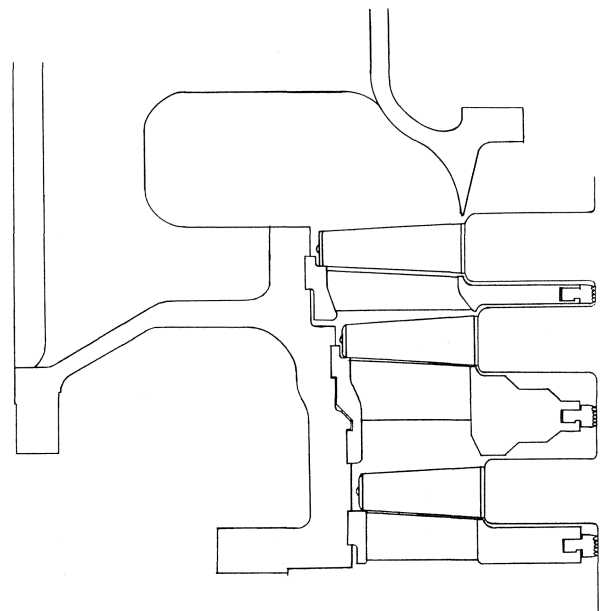


Figure 7. Expander Diaphragm Cross Section. (Note that the second stage diaphragm bridges two diaphragm slots).

The expander shaft outboard end was extended four inches, to permit the future addition of a turning gear in the event that one would be required. The outboard bearing housing coverplate was modified to accommodate the longer shaft.

A 20 in butterfly valve in the suction line was set to trip on an electronic overspeed or train trip signal. In the event of a coupling break, the valve would start to close as the rotor exceeded the emergency overspeed trip point setting of 5824 rpm. The valve had a piston actuator using compression spring force to close and pneumatic pressure force to open. The trip signal opened a solenoid dump valve which bled off pneumatic pressure allowing the compression spring to close the butterfly valve. Air pressure opened the valve to 60 percent open, where a mechanical stop prevented further travel. The valve was set at 60 percent open, in order to ensure closure within the one second time period before the rotor reached destructive speed in the event of a breakaway.

At 60 percent open, it created a major flow restriction with a  $C_v$  of 4200 and a pressure drop of eight psia. At 70 percent open, it had a  $C_v$  of 7000 and a pressure drop of only three psia; however, the valve closing time increased from 1.0 to 1.16 seconds. The rise in speed in the event of a breakaway is estimated at 3000 revolutions per second by the manufacturer. Opening the valve fully would increase the closure time and result in the catastrophic failure of the rotor. To decrease the valve closure time, two additional solenoid dump valves were installed on the pneumatic side of the actuator piston. Closure time was reduced to under one second from the full open position. The pressure drop to the expander inlet was minimized, increasing net horsepower output.

The mechanical design was thoroughly evaluated prior to the start of manufacturing. Rotordynamic diagrams and mechanical design were reviewed by plant maintenance engineers and an outside structural/vibration consultant. Campbell diagrams, Goodman-Soderberg diagrams, SAFE interference diagrams, ANSYS generated mode shape and frequency diagrams and calculations, a lateral vibration analysis, and a torsional analysis were all part of the review. The review brought out a number of legitimate user concerns and resulted in some changes in the design. Due to a previous blade failure, blading design was subjected to intense scrutiny. The proposed blades were rejected on the basis that the trailing edge of the airfoil extended off the platform, creating a potential failure point. The manufacturer agreed to supply a new blade design with the airfoil totally over the platform.

Concerns were also expressed about interferences shown on the Campbell diagrams for each stage. The first stage had a second tangential resonance within our operating range. Lowering minimum governor speed to 3600 rpm created an interference at the last stage rocking mode. The manufacturer stated that, although the interferences were present in the operating range, the mode shapes were not conducive to the development of enough energy at resonance to cause any problem.

The new critical speeds were at 3000 rpm for the first and at 6900 rpm for the second critical. The coupling still had above a 2.0 service factor. The thrust bearing service factor had a new load of 192 psi, and was rated for a 412 psi maximum loading. New thrust bearing temperatures were well below the 200°F limit set by the manufacturer.

Goodman-Soderberg diagrams were also reviewed with close attention to the philosophy of the determination of amplification and safety factors. The manufacturer stated that they were very conservative in their design. Their minimum allowable safety factor was one and a half. The overall stress levels were limited to no more than two thirds of the variable yield strength, even at amplification factors of twelve. The evaluation revealed no problems. Across the operating range, the first stage wheel has

a safety factor between two and one half and three for amplification factors of seven to nine. The second stage had a lower safety factor—two to two and one half, due to identical amplification factors and higher horsepower. The third stage had a safety factor just under two. This was caused by an amplification factor of ten to twelve, due to a harmonic at five times running speed. A second analysis was done to determine the effects of raising the inlet temperature an additional 100°F. For the expected stresses, the rise in temperature significantly lowered the safety factor. The first stage safety factors were reduced from 2.45 to 1.88 for the blades and 2.48 to 1.91 for the root. Stage two safety factors were reduced from 2.08 to 1.93 for the blades and 2.18 to 2.02 for the root. The third stage was below 550°F and safety factors were not significantly affected by the increase in temperature.

Detailed ANSYS generated mode shapes for blades and blade packets. Each packet consisted of five blades riveted into a shroud. Vibration response was calculated using ANSYS finite element modelling. Each packet was modelled and the harmonics checked. Blades were also modelled. Deflections of the blades were fairly large due to resonance and a high excitation force generated by the axial velocity of the gas.

Axial deflection of the diaphragms was also scrutinized very closely. Calculations were done for a 31 psi drop across the expander. The first stage had an axial deflection of 0.037 in with the expected pressure differential of 12 psi across the diaphragm. The second stage had 0.025 in deflection for a 9.5 psi differential. The third stage, with a much thinner diaphragm, had 0.072 in deflection for a 9.3 psi differential. The deflection is calculated at the edge of the diaphragm (shaft seal). Although the diaphragm had 0.250 in clearance at the center edge, it was only 0.050 in at the root of the stationary blade. Deflection along the length of the diaphragm cross section was checked and found to be 0.045 in at the blade root, leaving only 0.005 in clearance. The manufacturer modified the design to provide the nominal, as installed, no flow clearance of 0.100 in. This left a safe 0.055 in clearance at the maximum flow, maximum deflection condition and reduced the possibility of a rub.

Performance details were also examined. The manufacturer expected a significant velocity loss at the inlet flange face. Presently the expander recovers about 60 BTU per pound of flow. Using energy (BTU/lb mass) lost =  $(V_e/223.7)^2 \times 3$ ; a loss of 4.0 BTU/pound was present at maximum flow conditions. This loss did not warrant redesign of the case.

The following quality assurance measures were specified to provide maximum confidence in the rotor and diaphragm construction. Material certification, magnetic particle test, ultrasonic test, complete dimensional check including coupling hub contact, concentricity check, probe area electrical and mechanical runout check, residual magnetism, and a twelve-point residual balance test, per API Standards, were part of the testing and documentation of the rotor. The manufacturer did not have overspeed test capabilities, so an overspeed test was not performed. All of the documentation for the material certifications, magnetic particle test, and the ultrasonic test was reviewed by the corporate metallurgist. Everything else was witness checked and tested in a final inspection before release for shipment. Only three minor problems were discovered. The turning gear rotor extension had high residual magnetism, and the circumferential gap between shroud bands was non uniform and smaller than manufacturer's specification allowed. The extension was quickly degaused. The gap specification was a manufacturing tolerance only. The gap was within operating tolerance limits. Contact checks of the shaft's coupling hub taper initially showed poor results on all three rotors. All three rotors had "slight raised lip and small burrs" along the



O-ring shoulders that prevented the ring gauge from sliding onto the shaft properly. Light sanding with emery cloth corrected the problem. Ring gauge contact was good.

One major problem was encountered during manufacturing. In grinding the taper on the shaft for the coupling hub, the manufacturer mistakenly used a lesser taper than the specified 1/2 in/ft taper. Correcting the shaft to the correct taper effectively lengthened the coupling gap by about 0.200 in. After reviewing current coupling gap and shim location data taken during the last shutdown, along with the engineers' ability to attain a small degree of adjustment in setting the new diaphragms, it was decided that the error could be accommodated. In order to assure interchangeability, all three rotors were ground to the same dimensions. New coupling nuts were designed to make up for the fact that the hub was an additional 0.200 in up the shaft.

Field work was done under the close supervision of the manufacturer's service engineer and a service representative. The service engineer was primarily responsible for determining the final positioning cuts to be made in the diaphragms after a trial fit with the new three-stage rotor. He also supervised the machining at a local shop. Each expander case was stripped and cleaned. All new bearings and seals were installed in addition to the uprate design diaphragms and rotors. The job went smoothly and without any significant problems. Diaphragm hold down and alignment screw taps drilled and tapped by the manufacturer did not match those in the case. They had to be redrilled and tapped in the field in a different position. The manufacturer should have left the drilling and tapping for the plant machinists.

## ANTISURGE SYSTEM

As part of the engineering design package, the compressor vendor was required to perform an analysis of the existing antisurge system and recommend modifications required for the compressor's new performance parameters. The operating system is inherently resistant to operating conditions that could induce surge. It is a once through process with few potential restrictions. The antisurge system was required to respond to these potential restrictions.

The highest potential for surge is during startup, when the air compressor discharge must be blowing off to atmosphere and the process compressor must be recirculating. There is a potential to generate heat from zero flow windage in the expander. During this time, there is also a potential for the process compressor to pull excessive vacuum on the intermediate components. Besides surge, another major concern was having enough horsepower, since the expander contributes no horsepower during startup. The revamp increased each compressor's capacity while lowering discharge head. Since the steam turbine rated output was very close to the horsepower required, the system was analyzed very carefully to assure startup ability.

The original surge control system had a blowoff valve to atmosphere on the air compressor discharge. This existed exclusively for startup purposes. The process compressor had a recirculation line from the second section discharge back to the first section suction that controlled flow through the entire machine. The recirculation control valve was controlled by an analog surge control device that sensed case differential pressure vs inlet flow corrected for inlet pressure and temperature. There was also a second recirculation line from the first section discharge to first section suction that was controlled by a speed switch. Below a certain speed, it was possible that the large flow capacity of the first section could stonewall or choke the second section. Opening of the valve reduced flow into the second section. Suction line coolers removed the heat of compression during recirculation. The new compressive heat load

was calculated, based on anticipated worst case startup conditions, and was determined to be well within the capability of the coolers. Finally, there was a vacuum breaker valve that allowed makeup mass flow into the process compressor to reduce the potential for drawing excessive vacuum.

Currently, surge control is manipulated by the plant's integrated control system. The existing control loop was modified to control only the first section of the compressor using a ten percent margin from the surge line (Figure 1). A second analog control loop was added to control the second section using a 15 percent margin. The two sections were given different margins, because the second section will surge first as volume flow is restricted. The surge control valves do not open during normal operation. Operating points are typically much closer to maximum flow conditions than the surge line.

Dual ball type control valves were used in the recirculation loops in lieu of one large control valve. Concerns over noise and reliability were the deciding factors requiring dual valves.

The second significant change was the addition of a suction "damper" on the inlet to the first section of the process compressor. This inlet damper is a large butterfly type valve. The damper reduces mass flow to the process compressor, thereby reducing horsepower during the startup phase. The damper also decreases the possibility of pulling excessive vacuum on upstream components and eliminated the need to enlarge the existing vacuum breaker. The inlet damper has a mechanical stop to prevent its total closure.

## PROCESS COMPRESSOR SEAL SYSTEM

The original shaft seal design incorporated two different buffer gas injections into labyrinth seals. Inert purge gas, which is nitrogen with minimal organic vapor, is used on the inner shaft seals while pure nitrogen is used on the outer shaft seals. The suction end used a low pressure inert purge gas injected between process and a vent port to the process flare. Flow was controlled by maintaining a five psi differential pressure over suction pressure. The flare port was similarly sealed from atmosphere using a nitrogen injection. A 1.5 psi differential pressure was maintained between the nitrogen injection port and the flare port.

The discharge end shaft seal was similar except it used high pressure inert purge gas as a buffer between the process, at second stage discharge pressure, and the flare port. It had a shorter labyrinth seal between the injection port and process, due to space constraints. The labyrinths had stationary teeth riding over a smooth stainless steel shaft sleeve. The labyrinth seals between the vent port and the process gas are made of Fluorosint 500®. This material was favored to be in contact with the process gas, due to its chemical resistance properties. The seal cartridges were also made from this material. The outermost labyrinth seals, containing the nitrogen injection ports, were made from aluminum.

A number of problems were experienced with this seal system: erosion of the Fluorosint 500® labyrinth teeth; high nitrogen costs; higher than expected buffer gas flows; and, high maintenance replacement cost. In one case, loss of a seal resulted in the contamination of the lube oil system with solids from the process gas. Another problem was the buildup of solids between the stationary labyrinth teeth. Solids would harden and actually cut into the stainless steel shaft sleeve.

To improve the shaft sealing system, the plant engineers set out with the following objectives: reduce nitrogen consumption; improve mechanical reliability; and, reduce the maintenance replacement costs.

These objectives were met by a number of progressive and simultaneous steps (Figure 8). First, 410 SS seal cartridges were designed and fabricated by an after market supplier, with

the inner bore lined with smooth, replaceable Fluorosint 500® inserts. Second, rotating labyrinth teeth were added to the shaft sleeve design. The rotating steel labyrinth teeth were more durable than the stationary Fluorosint 500® teeth. This design is less susceptible to erosion damage and accumulation of solids buildup. Third, all diametral clearances were reduced to the minimum required for assembly purposes. Since the Fluorosint 500® is an abrasible material, diametral clearances were reduced from 0.012 in to 0.004 in. The clearance under the aluminum labyrinth was also reduced from 0.016 in to 0.008 in (and was later opened back to 0.015 in. These new clearances were discussed with the manufacturer prior to manufacture and installation. Rubbing of the rotating teeth against the Fluorosint 500® inserts was evident on initial startup during slow roll, but quickly dissipated as the inserts abraded.

Fourth, a balance piston was added to reduce the sealing pressure on the discharge end (Figures 9 and 10). The balance piston has two sets of rotating teeth sealing against two Fluorosint 500® strips. A buffer gas flow of about 25 acfm is injected between the stationary abrasible strips through about twenty equally spaced 5/16 in diameter holes. The radial clearance is a nominal 0.004 in.

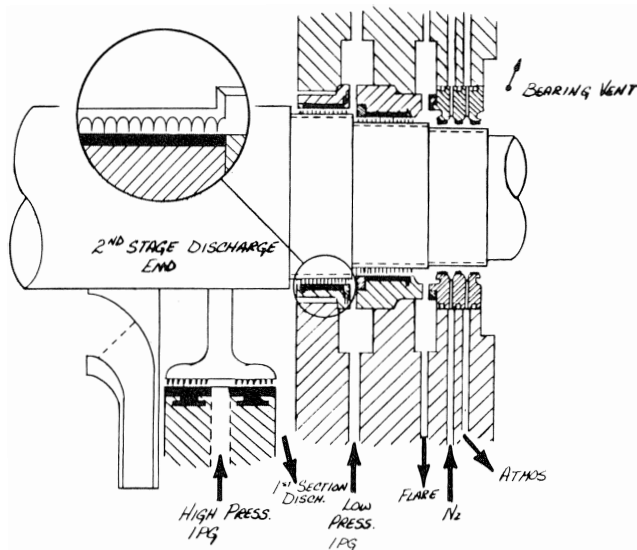


Figure 8. Process Compressor Discharge End Seal System Modification.

High pressure buffer gas injection was installed on the balance piston to reduce the chances of fouling in the labyrinth area. The low pressure side of the balance piston vents to the first section discharge line. The inner labyrinth shaft has a low pressure inert purge injection replacing high pressure injection. By reducing the sealing pressure, it was reasoned that the pressure at each injection port would be reduced, thereby reducing the required nitrogen injection pressure and, hence, nitrogen flow.

Fifth, the number and diameter of the injection holes in the outer aluminum labyrinth shaft seals were increased. This improved the reliability of the seal by ensuring the injection pressure measured outside of the compressor would more closely match the actual sealing pressure at the labyrinth seal. A new outer seal is currently being designed to further reduce clearances, hence nitrogen injection costs, to an absolute minimum.

#### INITIAL STARTUP OF THE FIRST TRAIN

Getting the train up to minimum governor control speed for the first time after the revamp was fairly difficult. The compres-

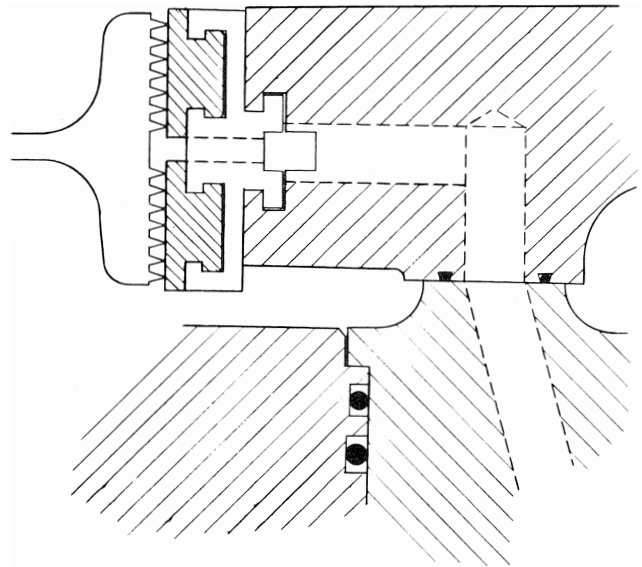


Figure 9. Balance Piston Diagrammatic Cross Section.

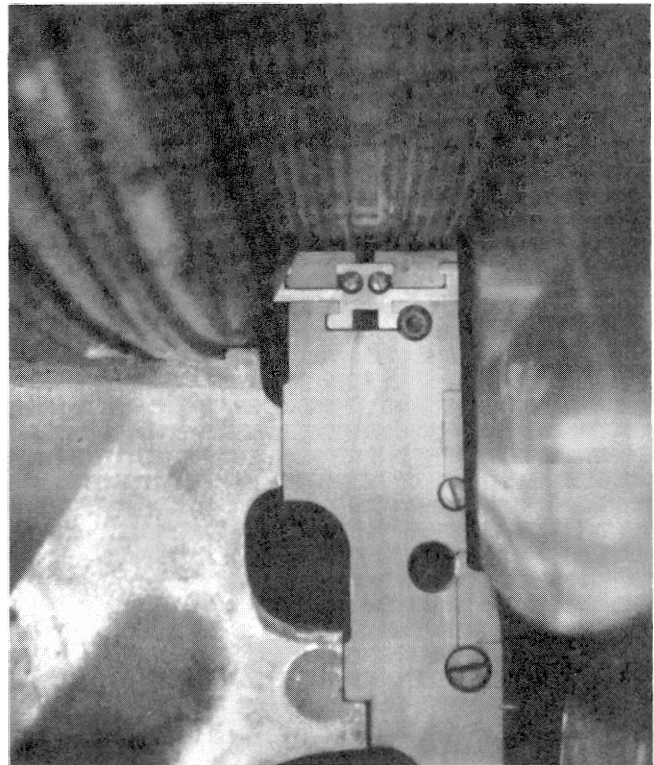


Figure 10. Balance Piston with Fluorosint® Seals.

sors were all running with new temperature and pressure conditions. A panel monitor and a portable FFT analyzer were used to monitor vibration. Bearing temperatures and process conditions were monitored on the plant computer.

The compressor train was brought up to a slow roll of 500 rpm. Air was being circulated throughout the system and the process had not started. The air compressor exhibited some brief vibration due to seals wearing in, as evidenced by vibration dropping gradually from 3.0 mils to 0.6 mils. After an hour of smooth running, the speed was increased to 1500 rpm. The process compressor ramped up to 5.0 mils vibration fairly rapidly, tripping the train. Restart attempts tripped in the same

manner. The bearings and the outboard aluminum seals were pulled. A rub was apparent on the aluminum labyrinth teeth. The seal clearance was checked with a feeler gauge and found to have an 0.008 in clearance, half of the manufacturer's recommended 0.016 in clearance. These clearances were purposefully tightened. The aluminum labyrinth seal clearances were opened up to 0.015 in and the compressor was reassembled. The compressor train was restarted and vibration levels were quite low, 0.4 mils.

While still slow rolling, expander vibration began to increase. The increase was almost logarithmic with time, slowly increasing from 0.21 mils to over 5.0 mils in a few seconds to trip the train. Several attempts at startup were made, all with the same ramping vibration pattern. The FFT analyzer showed virtually all of the energy was concentrated in a long thin pulse at running speed (Figure 11). The speed of the compressor was changed, but the amplitude of vibration appeared independent of rotating speed. Audible noise and noise heard through an ultrasonic listening device indicated a hard rub. Inlet and outlet piping supports, spring hangers, and slide plates were checked for freedom of motion and found to be free. As a final measure, 50 lb cooling steam was injected into the inlet. This lowered the inlet temperature gradually to 500°F from 670°F. As the temperature of the inlet gas dropped, so did the vibration. With the expander running smoothly at 0.6 mils, speed was brought up to minimum governor, 4000 rpm. Inlet temperature was then slowly raised by about twenty degrees an hour until 750°F, the maximum allowable inlet temperature, was reached. Vibration stayed at below 0.6 mils throughout the temperature changes, finally settling out at 0.3 mils. Subsequent startups revealed that the expander vibration was sensitive to the rate of temperature increase. A teardown inspection of the internals in August revealed very hard shaft rubs along the bottom inlet end seals. A new operating procedure was drafted to bring inlet temperature up slowly.

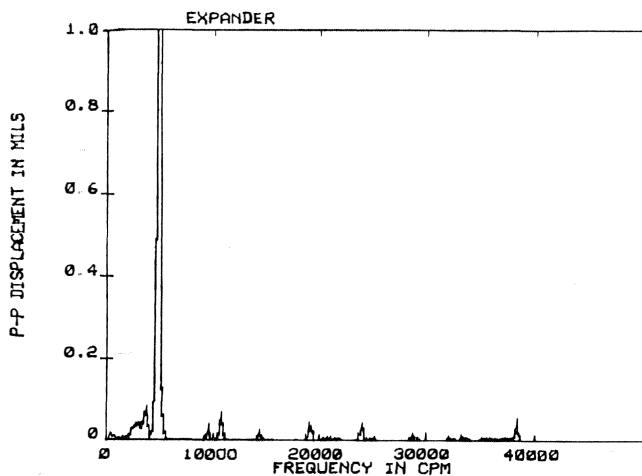


Figure 11. Expander Vibration Spectrum.

The steam turbine also showed a relatively high vibration level of 1.5 mils at slow roll. Spectrum analysis from both probes revealed a sharp amplitude at synchronous speed. The time wave form indicated a sharp jump. The cause was determined to be a scratch in the shaft. Vibration rose to 1.9 mils at operating speed. Alarm and shutdown levels in the monitor were adjusted slightly to protect against an accidental trip.

## PERFORMANCE TEST

A performance test was necessary after the first week of operation. Overall performance of the train was measured by

plant management in terms of air flow delivered to the process. The air rate of the uprated train was poor, at or below that of the original compressor train. Naturally, top management was upset. A performance test was set up with the help of a service engineer from the compressor manufacturer.

All of the required instrument taps were already in place. Although the setup does not meet ASME PTC-10 requirements, a high degree of accuracy was expected. Inlet and outlet temperature and pressure gauge connections existed on all of the machines. Rotational speed was measured by two digital tachometers. Flow measurement was taken across orifice plates for the steam turbine and expander and with flow tubes on the air compressor and effluent compressor discharges. A manometer was hooked to the upstream and downstream taps on the venturi flow tube. One minor problem was that static pressure readings were taken downstream of the venturi rather than at the 1/4 in static pressure tap in the flow tube. This was calculated to create less than one percent effective error for flow. The flow tube itself is accurate within 0.5 percent. Readings were adjusted for deviations in relative humidity, barometric pressure, ambient temperature, and specific gravity of the process gas relative to specified conditions. Readings were taken with one calibrated, hand-held thermocouple, one hand-held pneumatic indicator, and one water manometer. By using the same instruments, the possibility of error between two separate instruments was eliminated. Process conditions were held stable by operations. A chemical analysis was done on process streams on both sections of the process compressor and the expander to determine molecular weight and specific gravity of the gases.

Six tests were performed overall at different speeds and operating conditions over three days. Three tests were done at maximum output. The others were done at low speeds, one of which throttled the inlet flow to the expander. Finally, one was done at high speed with the air compressor inlet valve throttled. Four people took data. Three were in the field. One monitored the train performance in the control room to assure consistency of the data.

For actual comparison purposes, the full power test runs were compared to the specified 125 percent case from the operating conditions on API data sheets filled in by the manufacturer. Major deviations from these parameters were listed and horsepower gains and losses were tallied. Major performance problems were uncovered in each machine, with the exception of the steam turbine which had not been modified. The air compressor was showing a low efficiency at about 74.4 percent, well below the quoted 80 percent. This low efficiency was costing a few hundred horsepower. Inlet temperatures into each stage of the effluent compressor were 114°F and 118°F, several degrees over the maximum rated inlet temperatures of 105°F. This cost over 100 BHP for each section. In the effluent compressor section, one efficiency was much higher than expected—86.3 percent vs a predicted 74 percent. This was a net savings of a few hundred horsepower based on calculations. The efficiency increase was theorized to be the result of a liquid fog carryover out of the suction separators. As the liquid aerosol passed through the compressor it vaporized, creating a continuous intercooling effect. This phenomena is covered in depth in a later section.

The expander was not performing as expected, even with higher than normal inlet temperature boosting horsepower by about 250 HP. The expander performance data could not be directly compared with manufacturer supplied performance curves. New performance curves were extrapolated from the manufacturer's curves to represent speeds and temperatures at the operating points. These curves showed that pressure drop across the expander was lower than expected for the mea-

sured flows. The horsepower balance for the train also showed that less horsepower was being generated by the expander than expected for the actual flow rates. Calculations indicated that 12 percent less horsepower was being generated.

Based on the findings of the performance tests, an action plan was developed and implemented for correction of the known problems. Test data was sent to the various manufacturer's engineers for their own evaluation. Meetings were then set up with the manufacturers, to review results of their evaluation and to determine further testing needs and the anticipated scope of repair. The air compressor and process compressor were made by the same manufacturer. The performance engineer assigned to the project stated that, typically, the air compressor design produces slightly higher head and efficiency than predicted. The data, however, showed the efficiency to be five percent lower and the pressure coefficient to be 2.5 percent lower than expected. This suggested an internal problem—possibly an excessive amount of recirculation or incorrect hardware positioning. A list of potential errors was generated:

- Discharge scrolls reversed with respect to rotation.
- Diaphragms reversed with respect to rotation.
- Impeller and shaft labyrinths are out of clearance tolerance or badly damaged.
- The third stage inlet guides, which are angled to provide pre-swirl counter to rotation, are reversed.
- Excess buffer gas seal leakage.
- Inlet wall and diaphragm split line gaps.

Due to the relatively smooth operation of the compressor since its uprate and the large volume of recycle required, seal leakage and split line gaps were ruled out. Reversal of internal parts was perceived as the most likely cause of the performance shortfall. Plans were made to open and inspect the first train during a total plant shutdown scheduled for August 1986.

The process gas compressor performance was also in question due to the actual data's relation to the expected performance on the pressure coefficient and efficiency curves (Figures 12 and 13). The most obvious problem was that inlet temperature was much higher than expected, boosting the actual cubic feet per minute ACFM significantly along with the corresponding horsepower required. Chemical cleaning of the suction coolers at the first opportunity solved that problem. The performance engineer also stated that the open impeller appeared to have a ten percent larger capacity than expected. This tended to lift actual data above the pressure coefficient curve. In addition, the higher calculated efficiency was due to liquid entrainment. Liquid entrainment has a positive effect in that evaporation of

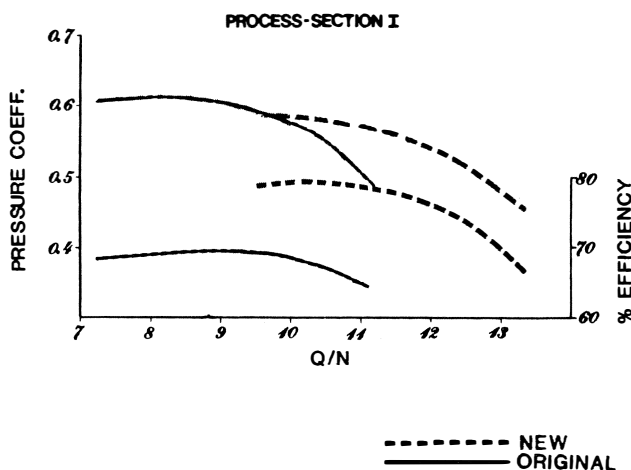


Figure 12. Process Compressor Expected Performance Curves.

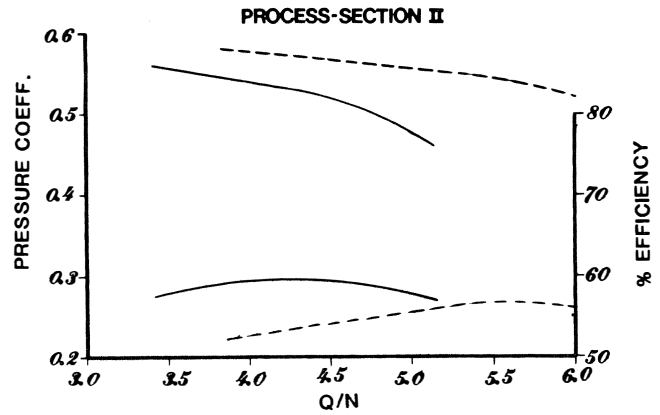


Figure 13. Process Compressor Expected Performance Curves.

liquid particles in the stream acts like an intercooler increasing calculated efficiency. However, film evaporation of liquid on the thermowell can distort discharge temperature readings to well below thermodynamically expected values. The performance engineer stated that although there was a net increase in performance, as supported by the net horsepower balance, the real efficiency and pressure increases were much lower than the calculations indicated. He also stated that he had no way of quantifying actual compressor performance for our two phase flow. The final conclusion was that the compressor was performing better than expected. Exactly how much better is not known. A strain gauge torque meter would give the best definition of actual horsepower, but was viewed as an unjustifiable cost.

Expander performance was well off the predicted performance curves. Inlet pressure versus flow diagrams revealed that the expander was not generating the specified resistance per pound of gas flow. Initially, the manufacturer maintained that the plant's flow readings were in error. Flow instruments were double checked for both calibration and calculation of the flow. The expander inlet flow meters involved were orifice plates with pipe flange pressure taps. Measured flow readings were confirmed by gas analysis and stoichiometric calculations. The manufacturer's lead engineer made a site inspection, but still maintained the problem was in the plant's flow measurement. The manufacturer's calculations were supported by an independent consultant. Nevertheless, the company requested a reduction in nozzle area to increase inlet pressure. Nozzle area was reduced on the plant's second set of diaphragms (which had not yet been installed) at the manufacturer's factory and double checked in the field. Trailing edges were cut at both ends, bent, and re-welded to reduce the cross sectional flow area. The area reduction was based on field data points and expected pressure drop between stages. Pluggage of the case drains prevented the acquisition of accurate interstage pressure data. All three stages were modified proportionally.

In conjunction with the fixes scheduled for the first train and the modifications to the second train, the steam turbines were evaluated for potential stepwise horsepower increases. Removal of the second stage would allow an additional 25 percent more steam flow, and this would significantly increase output. The spare turbine rotor was modified for the second shutdown.

## PLANT SHUTDOWN

A total plant shutdown took place in August 1986. Revamp of the entire second train as well as modifications to the first train were the critical path items. Completion of the turnaround

on the first train was to be done as quickly as possible. Eight days were allowed for turnaround of the first train, while thirteen days were allowed for the second train. The critical path item for the first train was the removal and modification of the expander diaphragms. After the cases were opened and measurements were taken, the first train diaphragms were removed and shipped to the manufacturer's closest qualified repair shop for modification of the cross sectional flow area. The manufacturer's service engineer accompanied the diaphragms to ensure quality workmanship and the quickest possible turnaround. The manufacturer's facility would have been preferred, but it was in the midst of a labor strike. Turnaround on the diaphragms lasted about six days from shipment to return. In the meantime, the second set of diaphragms, already modified, was machined to fit the first train case. Measurements taken on both cases confirmed that this exchange would not cause a sloppy diaphragm fit in either case.

The first train air compressor was opened and the locations of all internals were checked. Two inlet guide vane halves to the third section were interchanged. The vane orientation had choked the inlet eye area and had distorted the flow preswirl into both third stage impeller eyes.

The first train steam turbine rotor was replaced with the spare rotor. The second stage diaphragms were removed and stored for possible future use. Inspection of the turbine internals revealed hard seal rubs on one area of the the shaft labyrinth seals. A sweep of the internals with a mandrel revealed misalignment of the case, bearings, and diaphragms. Due to pressure to startup and a relatively trouble-free history, the spare rotor was installed and the unit reassembled without realignment of the internals. A hydraulic bolting service with ultrasonic tension checking equipment was used to bolt up the turbine. Two teams of two men each simultaneously bolted up each side of the turbine within ten hours.

The second train was also successfully revamped within the allotted time of thirteen days. Work went fairly smoothly with few problems. The second train process compressor was replaced with the company's spare process compressor, previously rebuilt in the plant's maintenance shop. The air compressor was disassembled. New diaphragms and a revamped rotor were installed without any problems. The second steam turbine revamp, however, provided some interesting challenges. A mandrell sweep and the condition of the labyrinth packing indicated that the alignment of the case, diaphragms, and bearings was poor. All diaphragms were cleaned, reinstalled and aligned. The case bore and bearing housings were also brought into alignment. The biggest problem arose in the disassembly of the multivalves from both steam turbines. The threaded ends of the stems of four out of ten valves were badly damaged during disassembly. Fortunately, the manufacturer was able to quickly respond to the company's need. The valves were manufactured and nitride treated on an emergency basis. The unit was started up on schedule with the new valves.

The manpower requirements for this job should be mentioned. The plant utilizes a non-union maintenance contractor. Standard millwright manning levels for the entire plant are only three journey men, three apprentices, and one foreman. For the August shutdown, besides work on the two compressor trains, there was a turbine generator uprate from 13.8 to 17.0 MW and numerous miscellaneous rotating equipment jobs that could only be performed during a total plant shutdown. Work was scheduled for two ten hour shifts, seven in the morning until five at night, and six at night till four in the morning. An hour overlap between shifts was necessary for turnover and coordination between the foremen, supervisor, and manufacturer's representatives. Overall, as much work was completed in a twenty hour day as in a twenty four hour day.

Approximately sixty "traveling" millwrights were brought in by the maintenance contractor. In addition, an outside turbomachinery maintenance contractor provided twelve skilled mechanics. The outside contractor worked on the first train modifications only. As could be expected, there was some friction between the regular and the outside maintenance contractor, particularly over the use of common equipment, such as mandrils, air hammers, etc. Five service representatives from the manufacturers were brought in for varying lengths of time to oversee the quality of the work performed. Our regular millwrights temporarily became foremen to direct and facilitate the work of the "traveling" millwrights. A plant engineer coordinated and directed the millwright work and service representative support. Due to our lean supervision, a critical aspect to the success of the revamp was the ability and willingness of some of the service representatives to direct some of the work, rather than simply inspecting as things came apart and went back together.

In terms of the workforce, the "traveling" millwright force had a random distribution of skill level and work ethic. Disassembly of the second train equipment gave some opportunity to evaluate and separate the workers. Those who were motivated or skilled continued work on the compressors while the others were assigned to simple tasks or terminated. On the average, productivity of the traveling work force was fair to low. A second disadvantage of the travelers was theft. The agreement with the maintenance contractor specifies that the company would supply all tools. Virtually all of the supplied tools under 16 in long were stolen. Having had a similar bad experience during the April shutdown, several extra sets of small tools were stocked in the storehouse and made available as the field supply disappeared. Although the tool cost was a few times higher than it should have been, work did not stop due to a shortage of five inch wrenches or dial indicators. In comparison, the outside contractor, who was motivated by a chance to get in the plant, provided skilled and motivated mechanics who worked efficiently and brought their own tools. In an overall evaluation, the "travelers," in general, were not as skilled, as trustworthy, or as cost effective as the outside turbomachinery contractor. Politically, although the presence of the outside contractor caused some minor attitude problems, it has also provided some negotiating leverage with our regular maintenance contractor. The outside contractor will continue to be utilized as a supplement to our contract work force.

## CURRENT PERFORMANCE

Performance has been tested several times since the August shutdown. Historically, the first train steam turbine has had a lower maximum steam flow than the second train by about 10,000 pounds per hour steam. In addition, due to the steam system configuration, the first train has inlet steam that is 10° to 20°F cooler than the second train. After the uprate modification these differences remain, even though the calculated efficiency of both turbines is identical. Detailed measurement of internals and inspection for damage or pluggage may provide some clue as to why there is a difference in maximum flow.

The air compressors are now operating with similar efficiency although, due to lower horsepower input, the air flow rate is lower in the first train.

The process compressor is experiencing considerably more liquid carryover due to much higher flow rates. An analytical method of determining mixed flow performance is being researched in preference to installing strain gauges on the coupling. The case will be opened during the next major shutdown to determine the extent of any erosion damage due to the additional liquid flow, and to check the second stage impeller for stress corrosion cracking.

Finally, the expander on the first train produces less power than the second train expander for a number of reasons. The main factor is lower mass flow. The process gas is essentially inert nitrogen from the air compressor. Due to lower steam turbine output, there is a lower flow rate, hence a lower pressure ratio across the expander, reducing efficiency and horsepower. The first train expander has a higher inlet temperature than the second train. This actually adds horsepower to the first expander, but not enough to significantly offset the effects of lower flow.

### LIQUID CARRYOVER

This phenomena and its effect on both sections of the process compressor are not well understood at this time. The makeup of the two phase flow is not defined, as yet, in terms of chemical composition and mole percent of the liquid versus the gas phases. Size, or even weight percent of liquid in each stream, is not known. What is known is that the efficiency of the inlet separators drops radically as flow increases (Figure 14) and that liquid exists in both compressor discharge lines.

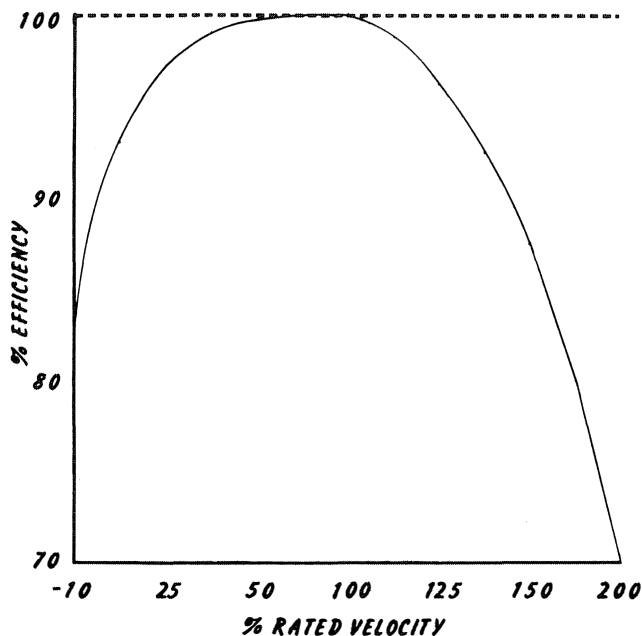


Figure 14. Typical Separation Efficiency vs Superficial Velocity through Vane Type Separation.

For the given rotating speed, a higher pressure ratio and flow rate than predicted exists. Currently, the measured flow rate, in acfm, through the process compressor at 4900 rpm, should not be achievable, according to the manufacturer's data, until the speed exceeds 5200 rpm. Both flow and pressure ratio at 4900 rpm exceed expected values quoted for 5200 rpm, although calculated horsepower is similar. The expected final production rate is constrained by a maximum continuous speed of 5300 rpm, due to the speed limitations of the process compressor second stage wheel. This flow enhancement effect points to the possibility of significantly greater production rates than originally envisioned in the uprate. Naturally, the side effects of higher flow will have to be investigated; however, the prospects for higher rates look good.

An ongoing literature search has produced few good references on wet compression in centrifugal compressors thus far. They report that wet compression typically increases the pres-

sure ratio and hence flow for a given speed. Horsepower per pound of total flow was also shown as decreasing with liquid addition. Tests were done on air compressors using water for liquid injection. A water to air weight ratio of about four percent was reported as optimum. Excess water tends to reduce efficiency, negating the positive effects of the higher pressure ratio. Disadvantages of water injection were also considered. Erosion damage and entrained solids deposition from the liquid injection are obvious effects. A second deleterious effect is movement of the surge line to the right. This is not a problem in this case, with the current high flow, low resistance operation. The authors also warn of possible mismatching of components in compressors that are not specifically designed for continuous liquid injection.

Currently, the plans are to gather more literature on the various aspects of wet compression. Since the liquid carryover and its effects have caught the company by surprise, the existing instrumentation is inadequate to provide adequate data on two phase flow. New sampling probes and shielded thermocouples are being investigated for installation during the next available shutdown. A practical methodology for the calculation of the actual performance still needs to be identified and fully developed. The mixture of different components making up the liquid carryover complicates the calculation. Current carryover is grossly estimated to be over ten percent of the dry gas weight flow, well in excess of the four percent recommended for optimum results. New, low pressure drop, suction separators have been ordered to eliminate carryover. At the same time, a liquid injection system controlling flow, composition, droplet size, etc., needs to be developed. Hopefully, enough information and possibly expertise can be found to facilitate the development of a workable system capable of maximizing the benefits of wet compression in the process compressor.

### CONCLUSION

The many aspects of a large equipment revamp, as seen from the plant engineering level, have been presented. A great deal of credit has to go to the original equipment manufacturers. Their expertise and aggressive attitude toward the project completion and troubleshooting were critical to its success. The contribution of inhouse specialists, after market suppliers, and independent consultants in both design and quality assurance roles is also recognized. Finally, the effective definition of the goals and coordination of the available resources was critical to the achievement of the desired results.

A general overview of the project reveals many phases. First, there was an incubation phase wherein the ideas were generated and plant tests were performed. Outside and after market suppliers, as well as OEMs, were contacted to determine their capabilities. Second, there was a development phase where the manufacturers were given a specific scope and contracted to develop it. Future implementation of the rerate was envisioned. Problems and constraints were planned for in advance. Measures for quality control and assurance were agreed upon. The manufacturers' designs, from system performance to blade packet resonances, were reviewed and approved prior to any release for manufacturing. Each vendor's scope of supply was detailed and finalized. Third, there was the manufacturing stage. Schedule and quality of the components were monitored at each specified step. Problems were confronted and resolved as expeditiously as possible. Detailed planning for the installation and startup took place. Arrangements were made for labor, service support, materials, tools, cranes, etc. Fourth, was the implementation phase. The equipment was revamped. Problems were resolved to ensure adherence to the shutdown schedule as well as high quality. This is where good planning and the manufacturer's emergency response capability paid off.

The train was then started up. Accurate and expedient troubleshooting and experience in using the right tools for analysis could avoid wasted time and energy. Finally, there was an evaluation phase, performance testing and evaluation of data. Careful acquisition of accurate data, especially with built-in inaccuracies due to normal process plant operation, was extremely important for producing meaningful results. Formulas for the calculational analysis of performance followed the ASME PTC-10 Compressors and Exhausters Power Test codes as closely as possible. This final evaluation phase is continuing on, even after the specified performance goals have been met to become the incubation phase for the next generation of performance and reliability improvements.

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