ABSTRACT

A series of failures of a radial inflow hot gas expander driving a centrifugal compressor in a nitric acid plant plagued the second author’s former company over a 24 year period. The 19,000 hp single-stage radial inflow expander inlet and discharge blade tips would fail in service, destroying the inlet nozzle vanes and expander wheel, and causing extensive damage to the rotor, bearings, seals, and shrouds. To finally resolve this severe reliability problem, an engineering team consisting of both authors’ company’s personnel was formed to pinpoint the root cause of failure and develop a corrective action plan. A parametric solid model of the expander wheel was created, and forced response stress and modal analyses were conducted on various design configurations using finite element techniques. Impact resonance testing of several expander wheels was performed to validate the results of the finite element modal analyses. Design changes were implemented based on the analyses and a new expander wheel was manufactured. The startup procedures of the machine were also modified and heat treatments on the expander material were optimized. The new expander wheel has operated successfully with the geometric and operational startup procedure changes without a failure, resulting in record service life to date.

This paper chronicles the history of these failures, expander design optimization studies and changes, operating procedure changes, and heat treatment changes for the A286 wheel material. The paper also documents the operational reliability attained in current operations, providing a validation of the engineering. Though the application of the engineering is unique for this case, the methods used in this investigation and problem resolution can be readily applied to numerous root cause failure analyses of rotating machinery.

INTRODUCTION

Solutia Inc. (formerly Monsanto Company) produces nylon fiber for the carpet and tire industries, and nylon pellets for the plastics business, at their large nylon manufacturing facility north of Pensacola, Florida. A key building block in the nylon process is nitric acid, which is manufactured in the Chemical Intermediates section of the Pensacola plant.

The most critical piece of rotating machinery in this process is the nitric acid air compressor. This is an extremely unique compressor, with only a much smaller version of it in operation at a 300 ton per day nitric plant in the United States. Conventional nitric acid compressor trains typically employ a low pressure compressor in a single housing connected to a high pressure compressor in an adjacent housing driven by a multistage steam turbine on one end and a hot gas axial flow expander on the other end through flexible couplings. This compressor train, however, comprises a single housing and one shaft incorporating one radial expander wheel, two open face compressor wheels, and a single-stage steam turbine all having curvic couplings and bolted together. A cross section of the machine is shown in Figure 1.
In the center body section is a 34.4 inch diameter open faced low pressure titanium impeller mounted back-to-back against a 27.4 inch diameter open faced high pressure titanium impeller through a 14 inch diameter curvic coupling. The first stage compressor boosts air from atmospheric to approximately 45 psig. An intercooler is used to reduce the 430°F to 470°F air off the first stage to 110°F to 180°F entering the second stage depending on production demand. The second stage impeller raises the final discharge pressure to 130 psig to 150 psig at a temperature of 410°F to 465°F. Two stub shafts fabricated from 15-5PH stainless steel are attached to the impellers by curvic couplings tied together by 10 inch diameter Inconel® 718 studs tensioned to 100,000 psi. Total air flow ranges up to 470,000 lb per hour.

The 21.9 inch diameter 17-4PH stainless steel single-stage steam turbine wheel is bolted onto the first stage stub shaft through a 14 inch diameter curvic adapter with an internal spline drive and overhangs the inlet end tilting pad journal bearing. The steam turbine produces approximately 8500 hp with 640 psig superheated steam at 720°F. Exhaust pressure is 175 psig.

The 35.1 inch diameter high pressure hot gas expander fabricated from an A286 alloy forging is bolted onto the second stage stub shaft via another 10 inch diameter curvic coupling. It is a radial inflow design, with hot gas entering radially and exhausting axially. The expander overhangs the second stage tilting pad combination journal and thrust bearing and produces up to 19,500 hp with nitrous oxide (NOx) gas at temperatures up to 1250°F and inlet pressure up to 225 psig. Exhaust gases at approximately 18 psig and 720°F are used to drive a NOx compressor module just downstream. Total hot gas flow is approximately 330,000 lbm/hr. A bank of 21 fixed position inlet nozzle vanes are mounted on an A286 nozzle plate to direct hot gas flow into the inducer blades of the expander wheel.

The entire air compressor rotor is 117.2 inches long and weighs 2700 lb. Operating speed ranges from 11,100 to 12,550 rpm, depending on production demand.

The rotor is mounted in a horizontally split barrel housing weighing a total of 46,000 lb. The air compressor module, commonly referred to as an “air plug” (shown in Figure 2), is mounted on rollers so that it can be wheeled around the shop during maintenance and assembly. The air plug is installed axially into the permanent housing in the nitric acid plant. A total of three air plugs are used. One is a “ready to go spare,” and the third is usually in some stage of rebuilding or reassembly. In addition to these complete air modules, high cycle time expander wheels are retained and stored as potential emergency spares.

HISTORY

Early Background

The enduser company originally operated four small 150 ton per day nitric acid units utilizing four two-stage centrifugal compressors powered by a three-stage backpressure steam turbine and a multi-stage axial flow hot gas expander. These compressor trains operated reliably from startup in 1953 until the early 1970s. Then Environmental Protection Agency (EPA) emissions noncompliance issues, combined with ever increasing maintenance and reliability problems, dictated the need for a newer nitric acid plant that would also be more energy efficient. A much larger 1000 ton per day plant that would be able to meet the required 250 parts per million (ppm) NOx gas emissions target at the least possible initial cost was the goal. Several different conventional design nitric acid compressor trains with multiple casings were available. However, all the alternative replacements were more expensive than the enduser wished to spend at that time. In addition, these extremely long equipment trains (nearly 100 feet long) would not easily fit in the available space. This situation set the stage for the procurement of the much more compact, single housing design.

The nitric air compressor module selected was designed and manufactured in the mid 1970s by a small southern California firm specializing in very high speed turboexpander applications. The version marketed and developed was a 10:1 scale up of much smaller models that had successfully been produced since the late 1960s.

Except for the smaller footprint, the promises of higher efficiency, lower initial cost, and faster delivery times were not met, due to a long series of unforeseen technical and reliability issues.

Among the many initial problems experienced on the test stand in 1975 and early 1976 were expander discharge blade tip failures near the corners closest to the shroud. This problem first occurred on an expander in February 1976. This problem appeared to be resolved when a contracted consulting firm recommended a very unique fix. The discharge blade tip was slotted by electrostatic discharge machining (EDM) approximately 3.3 inches from the end. A vibration damping rivet was then fitted along the slot line in a hole approximately 1 inch from the shroud line with a radiused stop hole 2 inches further in as shown in Figure 3. This frictional damping technique was incorporated on all the hot gas expander wheels furnished to the three chemical companies.

Unfortunately, these initial blade failures were just the tip of the iceberg. During initial operation in late 1975, the first compressor train delivered to a fertilizer company, experienced back-to-back expander inlet blade tip failures on both air modules within less than 10 days of operating time. Engineering analysis at the time indicated the failures to be in the number, orientation, and tear drop shape of the fixed inlet nozzle vanes that were causing a blade resonance failure. Similar expander blade tip failures were experienced in early 1976 by the other fertilizer chemical company. Because of these multiple unexpected catastrophic expander failures, the enduser delayed shipment of their air compressor modules to Florida until...
mid-1976 after the nozzle vane induced problems had been corrected by the original equipment manufacturer. In June 1976, the enduser first attempted to start up the new nitric acid plant with the new compressor modules. Within the first week they experienced their first expander failure during a late night field balancing operation when the expander discharge was inadvertently dead headed and the inlet blades failed from overheating. Less than three months later, an overheated thrust bearing on the second air compressor module led to a catastrophic axial rub failure of the rotor, including the other hot gas expander. These back-to-back failures required the enduser to revert to using the four old 150 ton per day conventional nitric units until replacement expanders could be manufactured and the damages repaired on the air compressor rotor components.

The enduser was not alone in their misery, however. Between November 1976 and May 1977 all four of the other air compressor modules at the other two companies experienced catastrophic failures with the inability to run for more than three weeks without a serious labyrinth seal rub, rotor to shroud rub, bearing failure, high vibrations, or expander wheel failure. For these reasons, the other two nitric acid producers scrapped their air compressor modules and purchased more conventional proven nitric compressors. The enduser considered this option in 1977 and 1978, but elected to stay the course.

In March 1977, the enduser reinstalled a repaired air compressor module with a new expander wheel. In August, between a pair of steam turbine wheel failures, the replacement expander suddenly came apart with no advance warning. No clear cause for this failure could be determined. Over three years transpired before the next expander wheel failure in March 1981. Like the earlier failures, all the inlet blades of the expander were torn off approximately 3 inches to 4 inches below the outer tip (Figure 4). As was the case with all the previous failures, no gradual rise in vibration or telltale phase angle shift in the vibration probe data was evident.

History of 13-Bladed Expander

After the repeated failures of the original 12-bladed expander design, the services of an independent consulting engineer were engaged to develop an alternative design. This design effort was completed in 1982 and the first 13-bladed expander wheel was manufactured in 1983. The 13-bladed design had a notably greater blade wrap angle as well as significantly thinner blades. In addition, no EDM slots or vibration damping slots were included. This change proved to be a bad mistake, however, as the exducer section blade tips broke off at the corners of several blades just three days after the expander was put into service in July 1983. This failure was preceded by a sudden increase in radial vibration at 1× rpm, followed by a high vibration trip within 10 seconds. When the exhaust end of the compressor was opened up for inspection, a 2 inch by 2 inch triangular section was observed to have broken off three blades at the exducer, while other blades had started to crack in the same area.

History of 13-Bladed Expander

After reverting to using older 12-bladed expanders for the next two years, the 13-bladed expander was repaired and reinstalled in August 1985. This time the expander was retrofitted with the EDM slots and vibration damping rivets. In addition, all the discharge tip corners were clipped off. This module ran successfully for nine months with no apparent problems. Then in May 1986, a sudden step change in vibration on the expander end bearing was noticed. The compressor did not trip out, but the 1× rpm vibration shot up by 1.2 mils (.05 inch) with an 80 degree phase angle change. No significant further change in vibration or phase was recorded for the next week of operation. When the compressor was shut down and inspected eight days later, a triangular chunk of metal approximately ½ inch by 1 inch in size was missing out of the discharge blade right next to the rivet hole.

In June 1987, a second 13-bladed expander was installed on a spare compressor module. This expander wheel operated satisfactorily for only eight days! Then a failure almost exactly like the one in May 1986 occurred with similar symptoms. The step increase in vibration was not large enough to trip out the compressor, but it was apparent the problem had not been corrected. A field balance correction could have been made to compensate for the lost material, but no one was confident the expander would stay together. Figure 6 shows a photo of the failed expander wheel.
A third 13-bladed expander was later manufactured. Resonance testing was performed to optimize the size and location of the rivet hole. However, this expander wheel was never put into service because metallurgical inspection revealed multiple deep subsurface flaws and voids in the inlet blading that would very likely have led to another premature failure of severe proportions. This incident ended experimentation with the 13-bladed expanders, and no more have been used since.

**Attempt at Weld Repairing Worn Expander**

In between the unsatisfactory experiences with the 13-bladed expanders, an attempt was made to weld repair the badly eroded inlet blade tips of a used 12-blade expander. Solid particle erosion of the inducer blades is a common malady for users of radial inflow expanders in fluid catalytic cracking (FCC) service, and the expander in this nitric acid plant is no exception. In 1985, after a catastrophic rotor wreck, not caused by an expander failure, the services of an engineering consulting company were engaged to replace the eroded inducer blade tips on an old 12-bladed expander with over 24 months of accumulated running time. Welding of A286 is not advisable without a complete heat treatment cycle; however, in the case of the expander wheel a complete heat treatment cycle would result in deformation of the wheel geometry. The enduser was assured that it could be done successfully with properly applied local heat treating and a hot isostatic processing (HIP) treatment to restore the forging grain structure. The eroded tips were cut off approximately 3 inches below the outside diameter. After HIP treating and metallurgical inspection, new tips approximately 3 inches long by 1.90 inches wide by .45 inches thick were welded onto the end of the cut off blades (Figure 7). The blades were then finish machined to original size. Dynamic balancing and overspeed spin testing to 14,000 rpm were performed as additional tests to help ensure reliable performance.

Unfortunately, this experiment proved unsuccessful. After taking about 16 hours to field balance the retrofitted rotor, the rebuilt air compressor ran less than 24 hours in service before experiencing a catastrophic failure in February of 1987. The compressor tripped out on high vibration with no gradual ramp up in vibration or phase angle change recorded. After the operators attempted to slow roll the compressor after trip out, the vibration meters pegged out at 5 mils (.2 inch) at only 800 rpm. When the air compressor module was pulled out and inspected the following afternoon, all the weld repaired inlet blades were found to have failed just below the centerline of the new welds. Figure 8 shows a photo of one of the failed blades. Subsequent inspection by enduser metallurgists and engineering consulting company representatives deduced the failure to be caused by altered grain structure in the heat affected zone below the weld.

**Recent History with 12-Bladed Expanders**

After another discharge end blade failure of an 11 month old 12-bladed expander in July 1989, the nitric unit operated quite reliably for over 43 months. Then in February 1993, the hammer dropped again. A significant drop in compressor airflow and speed, along with a step increase in expander suction pressure, occurred without warning late one night. In addition, vibration at $\frac{1}{1000}$ rpm increased by .5 mils (.02 inch) and the phase angle shifted 60 degrees. This combination of changes had not been observed before, and had the operating and engineering personnel somewhat confused. After running for another week with no other notable changes, the nitric unit was shut down and the expander exhaust end opened up for inspection. No evidence of damage to the discharge blades could easily be seen. However, when a video probe was extended up into the inlet of the expander, severe damage could be seen on the inlet blade tips and inlet nozzle vanes. When the module was pulled, the full extent of the damage and its root cause was evident. The inlet strainer had come apart and had been ingested into the expander suction. Damage to the expander wheel and nozzles due to ingestion of the inlet strainer can be seen in Figure 9. The irony of this event was that the very device that was intended to protect the expander and nozzle vanes had destroyed them. Subsequent metallurgical analysis determined embrittlement of the strainer and inlet hot gas piping was the root cause of the failure. By early 1994 the entire 316 SS inlet piping was replaced with Inconel® 800HT material, including the strainer.

The next expander failure occurred in December 1995. This time a conventional thin bladed wheel lost its inlet blade tips after...
approximately 24 months of accumulated service time over several different removal and replacement cycles. The failure occurred very suddenly and without warning during a nitric unit startup. In November 1996, another expander with the same thinner blade geometry failed suddenly without warning during full load operation. It had been in service for 10 months. In each case the inlet nozzle vanes were destroyed, but little collateral damage suffered by the rest of the rotor and housing components. Based on these continued failures, it was evident that significant changes were needed in either the material of construction, blade thicknesses, operating speeds, or operating temperatures.

The first design change to the 12-blade expander wheel made in 1997 was to increase the expander blade thickness along the shroud line. This geometry change was implemented since the last two expander failures had much thinner blades from the inducer tips to the exducer tips. Blade width was increased approximately .015 inch at the inlet tip, to .069 inch from the point of greatest curvature to the rivet slot, to .047 inch near the discharge tip. Figure 10 is a photograph of the thick bladed expander wheel. This increased width raised the blade stiffness and natural frequency. The first author’s company’s first involvement came after 1997 with the revised material specifications for the next two A286 forgings procured to improve their internal grain structure and thereby increase their creep rupture strength.

Because of the extremely long lead time to procure these special forgings and machine them to the final finished form, the enduser did not receive them until mid 1998. In the interim, they operated two other compressor modules with the conventional thin bladed designs until January 1998. After an operator error and bypass valve problem that allowed the compressor casing to fill up with lube oil, the #3 air module was pulled out of service. When the shroud was removed to inspect the expander blades more closely, several bent inlet blades were discovered (Figure 11). The root cause was found to be a large piece of 316 SS shim material, used to center the expander blades between the nozzle plate and shroud, had broken off and impacted the expander inlet blades. None of the blades were cracked, but the expander was considered to be unfit for continued operation.

After the shim failure, the spare #1 module was reinstalled. Unfortunately, the thrust collar on this module was severely damaged during a startup just two weeks later when a slug of steam condensate destroyed the thrust bearing. Since the #2 module was not yet rebuilt, this situation required the enduser to reinstall the #3 module with the bent expander blades. The #3 module then ran satisfactorily for four more months until May 1998. The enduser had wanted to install the #2 module with the new thick bladed expander design around this time. However, machining and balancing on this expander had been extended until July because of delays at the curvic coupling machining facility.

Experience with Two Piece Expander

Because of the continuing concern over the sustained reliability of the bent bladed expander, the #2 module was built up utilizing a unique two piece expander that had been fabricated nine years earlier but never used. This two piece expander comprised a new inducer section bolted onto an existing undamaged exducer section. No rivets were installed at the blade interface. The consultant who designed these two piece expanders relied on frictional damping to minimize blade flutter and deflection. One of these two piece expanders had operated satisfactorily for six months back in 1978 for the other user.

Two of these special expanders had been manufactured in 1988 and 1989 after the disastrous welding experiment failure in 1987 noted earlier. They had both been dynamically balanced and spin tested years earlier, but never installed. As fate would have it, they never should have been. Earlier in 1998 the two piece expander was spin tested again in a spin pit to 14,000 rpm for two minutes. The complete rotor was also high speed vacuum bunker balanced at 12,300 rpm with satisfactory results.
Only one week after the two piece expander was installed it failed catastrophically. Evidence of increasing vibration had been evident up to two days prior to failure; however, inventory was very low and the shut down of the unit was delayed. When the exhaust end of the module was opened up, the cause of the shutdown was very apparent. Most of the discharge end blades were bent and twisted, while three had lost chunks of metal of the end tips (Figure 12).

The #3 module with the bent expander was then reinstalled. As it turned out, this module was operated for seven more months on three separate runs because of an unfortunate series of thrust bearing damage incidents to the other compressor modules. When the bent expander was finally retired from service in June 1998 after 17 months accumulated run time, it was still running satisfactorily. To this day all our engineering, operating, and maintenance personnel are amazed by this.

The last expander failure occurred in December 1999. This was another premature failure with only a three week run after initial startup. It was the last expander with the historic thin bladed geometry. Ironically, the #3 module it was installed on was the one that the bent expander had been removed from service. This failure occurred shortly after the air compressor module operated for over 10 minutes in the 8300 to 8600 rpm speed range. This action was taken to try to alleviate a high vibration problem over 3 mils (.12 inch) at the normal prelight off-speed range of 8700 to 9000 rpm. Analitical studies by the first author’s company found a severe expander inlet blade resonance at 8519 cpm, which should have been avoided. Operating circumstances at the time did not enable this speed range to be avoided in the extremely cold weather that night that limited the prelight off-speed to 8700 rpm with this module.

ANALYSIS/CORRECTIVE ACTIONS

Because of all these failures that the enduser had been unable to prevent, in 1998 the services of the first author’s company’s Engineering Department were enlisted to identify the cause of failure and provide a solution.

The average life of an expander wheel to that date had been approximately 11 months, at which time it typically suffered a catastrophic inducer tip failure. Typical consequences from the failure included significant damage to the entire expander end of the machine requiring extensive repair.

Several studies had been conducted over the past 15 years or so to determine the actual cause of failure of the expander wheels, so that the correct approach could be taken to improve the design and extend expander life. Causes listed in the failure reports varied from stress rupture, to tensile overload, to forging defects, to statements such as “A cause for the overall failure of the high pressure expander wheel was not identified from this analysis.”

A threefold approach to pinpoint the cause of failure was prescribed. Three separate analyses were conducted as listed below:

- **Metallurgical evaluation**—The metallurgical evaluation conducted was in two distinct phases. The first was to analyze the type of failure by evaluating the actual fracture face. The second was a series of stress rupture tests and metallographic studies to evaluate the condition of the expander blade material in the close vicinity of the blade failures.

- **Impact resonance analysis**—Detailed measurements were taken on several expander wheels to determine the natural frequencies of the expander blades at the inducer and exducer sections. Extensive ring testing was conducted to enable plotting of the actual mode shapes. The results of this analysis were also used as a baseline for the finite element analysis.

- **Finite element analysis**—The expander wheel was scanned on a coordinate measuring machine for exact profiles. These data were then combined with profile data supplied by the end user, and a detailed solid model was developed. The solid model was then imported into a finite element analysis code and a detailed stress analysis was performed. In addition to the stress analysis, a complete analytical modal analysis was conducted using finite element techniques to determine the natural frequencies and mode shapes of the expander wheel blades. The results were then compared to the actual measured results to check for accuracy.

The results of these three separate investigations are listed below with a summary of the conclusions of each independent study.

**Metallurgical Evaluation**

To confirm the root cause of the failures and determine the overall material condition, mechanical testing, metallography, and fractography were performed on two separate expanders designated expanders #1 and #2. Shown in Figure 13 are the approximate locations where the test specimens were taken. For comparison, an in situ metallographic examination of a third expander, “Bubba,” which saw over 25,000 hours of operation without failure, was also performed. Also, a section of the fractured damping slot from expander #1 was examined metallographically.

**Figure 12. Wrecked Two Piece Expander Wheel with Shroud Still in Place.**

**Figure 13. Approximate Specimen Locations.**
5737 that specifies a solution annealing heat treatment of 1650°F for two hours, followed by aging cycles of 1400°F for 16 hours and 1350°F for 16 hours. The expander wheels were in operation for approximately 7000 hours prior to failure with an average inlet temperature of about 1250°F at speeds between 11,700 and 12,400 rpm.

Two tensile tests per expander were conducted at room temperature. The average results are shown in Table 1. Hardness testing was conducted in several areas of the blades, and no significant deviation among the various locations were found.

Table 1. Tensile Properties, Hardness, and Grain Size.

<table>
<thead>
<tr>
<th></th>
<th>T_u, ksi</th>
<th>T_y, ksi</th>
<th>%El</th>
<th>Hardness</th>
<th>Grain, ASTM *</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actual, #1</td>
<td>148</td>
<td>101</td>
<td>24</td>
<td>322 HB</td>
<td>4-8.5</td>
</tr>
<tr>
<td>Actual, #2</td>
<td>149</td>
<td>100</td>
<td>23</td>
<td>315 HB</td>
<td>4-8.5</td>
</tr>
<tr>
<td>AMS 5737</td>
<td>&gt;140</td>
<td>&gt;95</td>
<td>&gt;12</td>
<td>277-363 HB</td>
<td>&gt;1</td>
</tr>
</tbody>
</table>

* AMS 5737 does not specify grain size. Specification noted is per AMS 5734

Stress rupture testing of a couple of 0.160 inch reduced diameter specimens was conducted at 1500°F and 13 ksi. These conditions would produce an average rupture life in new material of about 28 hr. The specimen from the #2 expander fell slightly short of the expected, but rupture ductility in both specimens was excellent. The typical minimum criteria for elongation and reduction of area are 5 percent. These data were plotted below against the Larson-Miller curve (Figure 14) for A286 in both the 1650°F and the 1800°F annealed conditions. The test results of the two test samples are shown in Table 2.

Figure 14. Larson-Miller Plot of Stress Rupture Results from Both Expanders.

Table 2. Stress Rupture Test Results.

<table>
<thead>
<tr>
<th></th>
<th>% Elongation</th>
<th>% Red. Area</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>57</td>
<td>94</td>
</tr>
<tr>
<td>#2</td>
<td>45</td>
<td>94</td>
</tr>
</tbody>
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The fracture surface from the #2 expander (Figure 15) was damaged by apparent abrasive blasting making fractography difficult, but the surface from #1 was in excellent condition. Figure 16 shows the fracture origin at the trailing edge corner of the vane. The fracture propagated toward the vane center and final failure occurred by tensile overload.

Figure 15. Fracture Surfaces Following Sectioning, #1 on Left and #2 on Right.

Figure 16. Light Optical Micrograph of #1 Fracture Origin.

Closer study using scanning electron microscopy (Figure 17) revealed high cycle fatigue striations across the entire primary fracture surface. Unfortunately, the origination site was obscured by secondary impact damage and the exact point of initiation was not discernible.

Metallographic specimens were taken from the areas adjacent to the fractures. The microstructures of both expanders were similar (Figure 18). Most areas of the specimens had a duplex grain structure with about 70 percent ASTM 8.5, and 30 percent ASTM 4 grain size (refer to Table 1). Light to moderate intergranular precipitation and some eta phase had formed, probably during service. No signs of oxidation or corrosion attack were observed. Damping slot hardness and grain size (Figure 19) were also similar to the other examined areas. The microstructure of the “Bubba” expander (Figure 20) was considerably different with no indications of eta phase formation and a grain size of ASTM 2.5. However, it should be noted that this in situ metallographic technique provides information on the near-surface structure that may not be representative of the overall condition.
On evaluating the heat treatment procedure for the existing A286 wheel it was seen that the heat treatment used could be improved to obtain superior high temperature material properties. Shown below, in Figure 21, is a plot of the average stress rupture life for A286 in the machine’s operating temperature range. The stress level in the plot shown is 25 ksi, which is the approximate static stress in the wheel at the fracture location.

From the plot it can be clearly seen that the stress rupture life of A286 with the 1800°F heat treatment is significantly better than with the 1650°F heat treatment. The aging process could also be improved by machining the expander blade profiles to within 0.25 inch of their final profile prior to the heat treatment. This would ensure better material properties throughout the vane section.

The following key factors were determined from the metallurgical evaluation:

- The failures were undoubtedly the result of high cycle fatigue. Striation spacing indicated extremely high frequencies propagating the crack until ultimate overload.
- At the expander inlet inducer tips operated close to the material’s temperature property limitations
- The overall condition of the expander material was good to excellent. Tensile properties, hardness, and grain sizes all met the original specifications. There were signs of service induced aging, but stress rupture properties were found to be acceptable.
- A change in heat treatment would result in improved mechanical properties for the service conditions.

**Impact Resonance Analysis**

The resonance analysis included measured natural frequencies and mode shapes from impact tests and an interference study based on the operating speed range and the number of rotating blades and stationary vanes. The primary expander wheel in this study was new and designated as Bubba II. Other expander wheels discussed include:

- (a) two-piece,
- (b) Bubba, and
- (c) expander wheel #1.

Expander wheel #1 is distinguished by a minor failure at the damper rivet slit.
The natural frequencies of the expander wheel were measured by impacting the wheel with a modal hammer and simultaneously recording the impact force signal and the vibration of a wax (or glue) mounted miniature accelerometer, via a fast Fourier transform (FFT) spectrum analyzer. The expander wheel was isolated from background vibrations by hanging it in a free-free position from an overhead crane. Figure 22 shows the measurement setup along with the measurement instrumentation. Impact measurements were recorded at numerous locations along the expander wheel blade. The measurement locations along a complete blade for the two-piece wheel can be seen in Figure 23. Similar measurement locations were also used for the Bubba II expander wheel.

Figure 22. Picture of Measurement Setup.

Figure 23. Picture of Measurement Locations on Two-piece Expander Wheel.

The results of the impact resonant testing revealed multiple inducer and exducer natural frequencies throughout the operating speed range. The natural frequency measurements verified that there were possible resonant problems. Mode shapes of natural frequencies in resonance provided insight to the physical location of the problem.

The mode shape for a natural frequency was determined from the geometric location of the impact, the magnitude of the transfer function, and the sign of the imaginary part of the transfer function. In this study only the magnitude of the vibration normal to the surface was measured and this had to be taken into consideration when comparing the mode shapes with those predicted in the finite element analysis.

The mode shapes presented here also served the purpose of qualifying the predictions from the finite element analysis.

- The investigation showed that the expander wheels had a significant number of natural frequencies within the prelight off-operating speed range, all the way through the operating speed ranges.

- Time domain signal traces showed that the damper rivet was effective at damping the exducer section vibrations. Furthermore, the slit for the damper rivet effectively isolated the exducer section from the inducer section. Hence it was believed that the inlet vane passing frequencies would not excite the exducer modes.

Finite Element Analysis

A 3-D computer model of the expander wheel was created using scanned points taken with a coordinate measuring machine (CMM). The computer model of a one piece expander wheel was created from scans of a used one piece expander wheel. The model was sectioned to reduce the computational intensity. After sectioning, the model was meshed using 10 node tetrahedrons and mesh controls to produce adequate sized elements in the regions of concern. The elemental model of the expander wheel is shown in Figure 24. The model was then exported and stress and modal analyses were performed.

Figure 24. Meshed Expander Wheel Model.

After the model was created and sectioned, and imported into a computer software simulation program, symmetrical boundary conditions were created at the sectioned surfaces. Material properties for A286 at an average operating temperature of 1000°F were assigned to the model. Additionally, an angular velocity was applied to the model, which corresponded to a running speed of 12,400 rpm.

The results of the stress analysis showed relatively low stress levels, 20 to 30 ksi, in the inducer tips where the majority of the failures had occurred.

Two variations of the modal analysis were performed. The first modal analysis was a free-free model performed using room temperature material properties in order to calibrate the computer model with the impact resonance results. The results of the room temperature modal analysis correlated well with the physical measurements taken. The second modal analysis was performed using the material properties of the expander wheel at an average operating temperature of 1000°F and using the prestressed static analysis condition to incorporate the angular velocity associated with normal operation. Table 3 contains the material properties of A286 at room and operating temperature used for this analysis.
Vane tip failure. The fatigue nature of the inducer vane tip failure throughout its startup range that were a likely cause of the inducer revealed that the expander impeller had several natural frequencies lographic evaluation, impact testing, and stress and modal analysis be the most probable cause of inducer vane failures. Several modes were specifically identified that were determined to presented a serious problem with the design of this expander wheel.

The modal analysis revealed a significant number of natural frequencies found in and around the operating speed and prelight off-speed represented a serious problem with the design of this expander wheel. Several modes were specifically identified that were determined to be the most probable cause of bending inducer vane failures.

The results of this preliminary investigation including the metallographic evaluation, impact testing, and stress and modal analysis revealed that the expander impeller had several natural frequencies throughout its startup range that were a likely cause of the inducer vane tip failure. The fatigue nature of the inducer vane tip failure also validated this theory. However, it was also seen that the heat treatment of the material could be significantly improved. An optimization of the heat treatment procedure for improved high temperature toughness, as well as a blade redesign to optimize the blade inducer tip profile, were subsequently pursued.

Since the preliminary analyses revealed numerous inducer vane tip modes, the next step of the analysis was to identify the modes that the failure could be attributed to. The original model was developed from an expander wheel that had been in service (#1) and had some fairly significant tip degradation. For the analysis a new solid model was developed from prints for the “Bubba II” wheel.

In order to identify the resonant mode of failure, startup and operational data of the expander were evaluated:

- 800 rpm – 1000 rpm Slow roll
- 1000 rpm – 8500 rpm Acceleration through critical speed (1800 rpm/sec)
- 8500 rpm – 9200 rpm Prelight off-dwell zone (several hours)
- 9000 rpm – 12,500 rpm Unit startup (2½ hours)
- 11,500 rpm – 12,500 rpm Operational speed range
- 1202°F (650°C) – 1256°F (680°C) Operational temperature range
- 20 – 35 Number of starts and stops per year
- 3 – 5 runs, 2 – 4 times per year Balance corrections

After carefully studying the operational data the two speed ranges that need to be free from any resonances are:

- 8500 rpm – 9200 rpm Prelight off-dwell zone (several hours)
- 11,500 rpm – 12,500 rpm Operational speed range

The machine has 21 expander inlet guide vanes. This establishes the “resonance free” vane pass frequency ranges required as:

- 2975 Hz – 3220 Hz Prelight off-dwell zone (several hours)
- 4025 Hz – 4375 Hz Operational speed range

The stress distributions of all the modes were thus plotted to identify the modes that resulted in high stresses in the location of failure. Furthermore once “dangerous” frequencies were identified, forced response analyses were conducted with a variable frequency fixed load applied at the blade tip. This forced response analysis was performed on the original design impeller as well as on each design change iteration.

The critical natural frequencies were identified by comparing the stress plot contours to the actual failure mode. A review of the stress plots revealed that the failure of the inducer vane tip was most likely attributed to the bending modes of the inducer vanes. This is because the maximum stress location of the bending modes is on the backplate side of the inducer vanes, which corresponds to the actual fracture initiation site. The torsional modes cause a high stress at the center of the inducer vanes, which does not coincide with the fracture initiation site. Figure 25, Figure 26, and Figure 27 show the von Mises stresses of three modes that the original design inducer vane tip failure could be attributed to. These modes are at 1179 Hz, 2972 Hz, and 3830 Hz, respectively.

The Campbell diagram with the three modes that could cause high inducer vane tip stresses at the fracture location is shown in Figure 28. In order to identify which mode was the most likely cause of the failures it was necessary to identify which modes were most likely to be encountered during normal operation. In this case both the 2972 Hz mode at the lower speeds of the prelight off-speed, and the 3830 Hz at the lower operating speed seemed likely candidates. In a high cycle fatigue failure the combined static and

<table>
<thead>
<tr>
<th>Property</th>
<th>Values at 70 °F</th>
<th>Values at 1000 °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>0.286</td>
<td>0.286</td>
</tr>
<tr>
<td>Modulus of Elasticity</td>
<td>29.1x10^6</td>
<td>23.5x10^6</td>
</tr>
<tr>
<td>Shear Modulus</td>
<td>10.4x10^6</td>
<td>8.4x10^6</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.306</td>
<td>0.328</td>
</tr>
</tbody>
</table>

Table 4. Expander Wheel Natural Frequencies at Room and Operating Temperatures.

<table>
<thead>
<tr>
<th>Mode #</th>
<th>Frequency @ Room Temperature Condition</th>
<th>Frequency @ Operating Temperature Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>647</td>
<td>585</td>
</tr>
<tr>
<td>2</td>
<td>972</td>
<td>875</td>
</tr>
<tr>
<td>3</td>
<td>1444</td>
<td>1301</td>
</tr>
<tr>
<td>4</td>
<td>2132</td>
<td>1924</td>
</tr>
<tr>
<td>5</td>
<td>2247</td>
<td>2029</td>
</tr>
<tr>
<td>6</td>
<td>2322</td>
<td>2093</td>
</tr>
<tr>
<td>7</td>
<td>2367</td>
<td>2136</td>
</tr>
<tr>
<td>8</td>
<td>2464</td>
<td>2222</td>
</tr>
<tr>
<td>9</td>
<td>2625</td>
<td>2366</td>
</tr>
<tr>
<td>10</td>
<td>2879</td>
<td>2606</td>
</tr>
<tr>
<td>11</td>
<td>3488</td>
<td>3140</td>
</tr>
<tr>
<td>12</td>
<td>3640</td>
<td>3272</td>
</tr>
<tr>
<td>13</td>
<td>3835</td>
<td>3454</td>
</tr>
<tr>
<td>14</td>
<td>4052</td>
<td>3649</td>
</tr>
<tr>
<td>15</td>
<td>4266</td>
<td>3864</td>
</tr>
<tr>
<td>16</td>
<td>4493</td>
<td>4044</td>
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<tr>
<td>17</td>
<td>4767</td>
<td>4307</td>
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<td>18</td>
<td>5081</td>
<td>4575</td>
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<tr>
<td>19</td>
<td>5210</td>
<td>4687</td>
</tr>
<tr>
<td>20</td>
<td>5319</td>
<td>4773</td>
</tr>
</tbody>
</table>

The natural frequencies at operating conditions shifted approximately 10 percent lower due to the significant change in the modulus of elasticity. Table 4 lists all the nonrepeating natural frequencies found via the computer model.

Table 3. Properties of A286 at Room and Average Operating Temperature.
cyclic stress must be large enough to cause failure in a finite number of cycles ($<10^6$). In the case of a vane pass excitation at operating speed, $10^6$ cycles would be reached in approximately 4 minutes of operation, making a natural frequency in the operating speed range an unlikely cause for failure. A more thorough review of the prelight off-operation found that the speeds varied significantly and tended to be lower during winter startups due to the colder air. Additionally, multiple runs were occasionally required in order to balance the rotor. The nature of operation would mean an intermittent build up of cycles that could lead to failure of the expander wheel.

Ten different design configurations were analyzed. The designs analyzed (Designs D1 to D10) can be seen in Figure 29 and Figure 30. Three completely different design approaches were used. The approach chosen for the first four design iterations was to stiffen the blade, and consequently shift the natural frequencies of consequence out of the prelight off-dwell zone. The problem encountered with this approach was that with reasonable geometric changes, only a slight upward shift was seen in the bending mode of concern. This was not sufficient to ensure resonance-free operation. When a significant geometric change was made, it was found that the increased mass caused a significant increase in the wheel centrifugal stresses, resulting in unsatisfactorily high stresses at the base of the blade. The approach was thus abandoned.
Another important observation that came to light from this analysis was that increasing the taper of the inducer vane tip cross-section caused a greater stress differential between the back plate side and the shroud side of the blade. The reason for this was that making the blade section thicker on the back plate side increases the outer fiber's distance from the neutral axis in a bending mode. This causes a higher stress at the surface when the blade is in resonance. To obtain a better comparison between the original design and the first four designs, response analyses were also conducted on the new designs with the same variable frequency fixed load case. The peak stresses in designs D1 to D4 were seen to be higher than in the original wheel.

The second approach was to reduce the diameter of the existing wheel, thereby raising the natural frequency of the wheel. This did not produce good results either, since a large diameter change was required to obtain the natural frequency shifts required. Since a smaller wheel diameter would decrease the wheel power, this approach was also abandoned.

After reviewing the results from the first two design approaches it was clear that shifting the natural frequency was not an option. Thus, the next approach was to make a design modification that would not shift the natural frequency, but would lower the peak stress in the blade when the wheel traversed the bending modes. With this approach in mind designs D6 to D10 were generated to obtain a more even stress distribution in the vane from the back plate side to the shroud side as the blade traversed its bending modes. The final design D10 provided the most satisfactory results. Comparing the results of the D10 design with the original design, a decrease in the peak stress levels for all three modes was achieved. This was achieved while maintaining an almost identical cross-sectional area and steady-state centrifugal stresses compared with the original blade design. The most dramatic reduction of stress is in the second bending mode, which is the mode that interferes with the lower end of the prelight off-dwell zone. It is important to understand that the absolute stress numbers were not considered since the exact load and damping conditions were not known. What was important however, was the relative reduction of the peak stress levels. Table 5 compares the peak stress levels and the peak stress reduction for the three inducer bending modes from the original designed expander to the D10 design.

Table 5. Relative Peak Stress Levels in the Inducer Bending Modes.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Natural Frequency (Hz)</th>
<th>Speed (RPM)</th>
<th>Peak Stress (ksi) - Relative Numbers</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Original Design</td>
<td>Final Design</td>
<td>Original Design</td>
</tr>
<tr>
<td>1</td>
<td>1179</td>
<td>1371</td>
<td>3546</td>
</tr>
<tr>
<td>2</td>
<td>2972</td>
<td>2985</td>
<td>8529</td>
</tr>
<tr>
<td>3</td>
<td>1024</td>
<td>1052</td>
<td>1100</td>
</tr>
</tbody>
</table>

Note: Frequencies and speeds based on inlet nozzle count.

In addition to reducing the peak stress levels of each mode, a great deal of effort was also invested toward not moving the natural frequencies. This was done for the simple reason that the second bending mode is at the bottom speed range of the prelight off-dwell zone. Any small increase in stiffness pushes this frequency higher, landing it within the prelight off-dwell zone. This can be very dangerous and cause premature failure.

CONCLUSIONS

The analyses revealed that the cause of the failures was most likely due to the resonance of the inducer vane tip bending modes. The most dangerous of these modes is the second bending mode that lies at the lower end of the prelight off-dwell zone. This mode had a natural frequency of 2985 Hz in the final design (design D10), which meant that it was excited by the 21 inlet guide vanes at a rotor speed of 8529 rpm.

Ten different design configurations were analyzed to obtain an optimized geometry for the expander inducer vanes. The final design chosen (design D10) shows a negligible change of natural frequency but a significant peak stress reduction in the inducer blades at resonance. This directly translated to many more successful startups and shutdowns, thus extending the life of the impeller. However, it was clearly understood that if the vane natural frequency is excited for a prolonged period, failure was still imminent.

It is important to note that the analyses were conducted at a metal temperature of 400°F because that was believed to be the prelight off-metal temperature. At shutdown, when the impeller tip metal temperature is higher, a significant downward shift would be seen in the natural frequencies due to the lowered modulus of elasticity of the material. As an example, a metal temperature of 800°F will result in approximately a 6 percent drop in the frequencies of the calculated bending modes.

RECOMMENDATIONS

Design Changes

Below were specific recommendations crucial to impeller inducer vane life extension:

- It was recommended that the prelight off-dwell zone be carefully controlled to range from 8900 rpm to 9200 rpm. This prevents the excitation of the second bending mode and thus prolongs life.
- It was recommended that the geometry of the impeller vane be modified to the design configuration D10 to reduce the inducer vane stress at resonance. This would result in the impeller being able to traverse more startups and shutdowns, thereby increasing its operating life.
- It was recommended that the heat treatment be changed from the 1650°F to the 1800°F+ precipitation hardening heat treatment. This would give the material increased rupture strength. It was also recommended that aging cycles be conducted after machining the impeller. This would ensure better material properties throughout the vane section.

Corrective Process and Diagnostic Actions

Among the process changes made, the machine operating instructions were revised to caution operators about the potential hazards of running the air compressor at speeds below 8700 rpm for an extended period of time. The speed range of 8300 to 8700 rpm was to be avoided during preparation for nitric unit light-off. This is particularly a problem in the cold winter weather when the denser air causes the compressor to produce more flow but at a lower speed than in summer or during a restart when the unit is still hot. The last three inlet blade failures on conventional expanders all occurred in the winter. The following quotes are taken from “Nitric Acid Air Compressor Startup Procedure” E2-467 (Price, et al., 2000) Item #200:

SPECIAL NOTES

a. Critical speed range for the air compressor is from 3600-6000 rpm. Very high vibrations are experienced in this range. Serious compressor damage can be done if the air compressor is not rapidly accelerated through this speed range.

b. High vibrations on the steam turbine may sometimes be observed right after going through critical speed. They will usually subside before the unit is lit off. If not, contact Engineering for diagnostics assistance to confirm whether it is safe to proceed with nitric unit startup.

c. After going through critical speed the air compressor needs to be running at least 8700 rpm (8900 rpm or more is preferred). Sustained operation at speeds in the 8300 to 8700 rpm range excites a natural frequency in the
HIGH PRESSURE expander inlet blades that can result in a catastrophic blade failure. This problem with low speed is most likely to occur in very cold weather since the denser air consumes more horsepower from the turbine.

d. Do not operate the air compressor in the 8300–8700 rpm range for more than five minutes. If sufficient rpm cannot be attained, trip the compressor out and contact Engineering for technical assistance.

In December 2001 the original steam governor valve was replaced with a new 12 inch governor valve. The replacement valve has a 7 percent greater Cv flow coefficient that enables it to pass more steam flow at less pressure drop. During three cold weather startups in the December 2001 to January 2002 period, the new governor valve enabled a 40 to 80 rpm increase in prelight off-speed, depending on the ambient temperature. This was not as much of a speed increase as the enduser had hoped for, but it has proven sufficient to get past the dangerous 8519 rpm identified in the engineering study.

Diagnostic procedures now include greater use of acceptance circle monitoring on each bearing to better detect erratic shifts in phase and radial vibration. Software alarms for synchronous vibration are set at 1.25 mils (.05 inch) with a 120 degree phase and radial vibration. Software alarms for synchronous circle monitoring on each bearing to better detect erratic shifts in performance until June 2001. The #1 module was then removed and stored in the compressor maintenance shop where it served as a proven primary spare. The only reason it was taken out of service was to certify the performance and reliability of the #2 module that had ingested foreign object debris (FOD) into the first stage compressor impeller and had to be repaired.

When the shroud on the expander was removed for inspection later in September, no evidence of any incipient cracking could be detected through visual means or dye penetrant testing. Only moderate rounding of the inlet blade tips, caused by solid particle erosion, was observed. As noted earlier, all radial inflow expanders in FCC and nitric acid service suffer wear damage from solid particulates exfoliated from the heat exchangers and hot gas piping upstream.

In June 2002 the #1 module with the new design expander wheel was reinstalled as part of the planned rotation cycle between the three air plugs. For 18 months the new expander has been in service through March 2003; operating speeds ranged from 11,080 rpm up to a maximum of 12,540 rpm. Expander inlet temperatures ranged from 1150 to 1250°F. Expander inlet pressures ranged from 140 psig at minimum production rates to 225 psig at maximum production rates in the winter of 2000/2001. No significant vibration or performance problems were encountered during this period.

With the combination of a sound top quality A286 forging, the new design that reduces stress levels in the critical resonant fatigue zone, revised operating practices that limit inlet temperature to 1240°F, closer operator attention to maintaining prelight off-speeds above 8700 rpm, high velocity oxygen fuel (HVOF) coated inlet guide vanes to reduce the wear effects of solid particle erosion, comprehensive metallurgical inspection after each nine to 12 month run, and vigilant vibration monitoring by enduser engineers, normal operating life is expected to increase from the historic average of 11 or 36 months or more in the current decade.

Results Achieved in 2000 to 2003 Operations

After the December 1999 expander failure, the #2 module with the thick bladed profile was reinstalled. It operated successfully through the summer of 2000 until the rebuilt #1 module with the new designed expander wheel was ready. In September 2000 the new module was installed. It ran satisfactorily without any mechanical reliability problems and excellent aerothermal performance until June 2001. The #1 module was then removed and stored in the compressor maintenance shop where it served as a proven primary spare. The only reason it was taken out of service was to certify the performance and reliability of the #2 module that had ingested foreign object debris (FOD) into the first stage compressor impeller and had to be repaired.

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REFERENCE


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Figure 31. Rotor in At-Speed Balance Machine.


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