REDESIGN OF A HIGH SPEED COMPRESSOR DRIVE STEAM TURBINE TO OVERCOME CAPACITY, EFFICIENCY, AND RELIABILITY ISSUES

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ABSTRACT

A case study of the redesign of a syngas compressor drive turbine is presented. The unit was installed as part of a relocated ammonia plant, and experienced reliability and performance problems from the start. A completely redesigned steam path was installed with the intention of correcting these issues. The unit continued to experience poorer than expected efficiency, and was forced down due to a first stage blade failure. A comprehensive engineering analysis was conducted to determine the cause of the blade failure and performance shortfall. Several inherent issues with the original turbine configuration were uncovered that contributed to the reliability and efficiency issues. A second, complete redesign of the steam path was installed to overcome these inherent problems to provide reliable, efficient operation of the turbine.

INTRODUCTION

An ammonia plant was removed from its original location and installed at a new site. Shortly after this plant started operations, problems were experienced with the synthesis gas compressor steam turbine driver as its efficiency was much lower than expected. Major reliability problems also occurred as the unit lost a blade within the first year of operation. Its poor efficiency and reliability continued to negatively impact the ammonia plant cost of operation despite various attempts to correct them. The decision was made to redesign the turbine to correct the efficiency and reliability issues and at the same time provide additional power that would allow increased plant capacity. The initial redesign effort did not correct the performance issues and a first stage blade failed shortly after the redesigned turbine was started up. A comprehensive evaluation of the turbine was conducted to determine the cause of the performance shortfall and blade failure. This evaluation revealed several inherent problems with the original turbine configuration. A second redesign of the turbine was conducted to correct these inherent problems and thereby provide efficient, reliable operation of the turbine.

BACKGROUND

Overview of Turbine and Operating Conditions

The subject steam turbine (Figure 1) is a six stage, single automatic extraction, condensing steam turbine. The high pressure section of the turbine is a single Rateau stage with six individual nozzle arcs fed by six control valves. All of the steam that exits the first stage is extracted through an opening in the lower half of the casing. A portion of the extraction steam is admitted into the low pressure section of the turbine via a single external control valve. The last two stages of the turbine are a double flow design. This was a common design practice for turbines of this vintage and was done to allow the use of shorter blades in the exhaust section of the turbine by splitting the flow between two sets of stages. Shorter height blades had fewer design challenges due to lower centrifugal forces and higher blade natural frequencies. The double flow last two stages are fed with steam via an internal crossover duct in the exhaust casing that feeds steam from the exit of the fourth stage. The fifth stage does not have a conventional nozzle diaphragm, but rather is fed by a radial flow nozzle that provides circumferential velocity of the steam in the chamber between the two rows of stage five rotating blades. The unit was originally designed as a double end drive. In its current application it drives a single centrifugal compressor from the low pressure end of the turbine shaft.

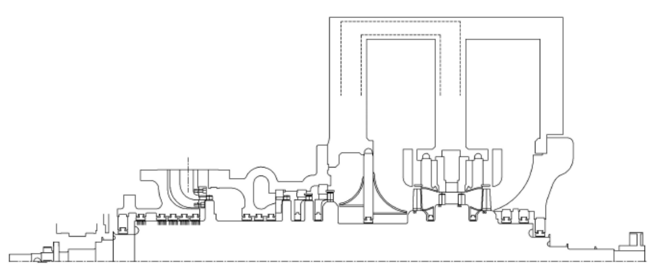

Figure 1. Longitudinal Section of Original Steam Turbine.

The relatively high inlet pressure of 1450 psig and temperature of 900°F along with the high speed of 10,850 rpm posed significant challenges in the blade design. Operating specifications for the turbine as it was originally designed are listed in Table 1.

Table 1. Original Turbine Operating Specifications.

Inlet steam pressure	1450 psig
Inlet steam temperature	900°F
Extraction pressure	500 psig
Exhaust pressure	4.0 "HgA
Maximum HP section flow	350,000 lbm/hr
Maximum LP section flow	75,000 lbm/hr
Turbine speed	10,850 rpm
Turbine rated output	19,000 HP

Turbine Performance and Reliability Issues

The turbine was not meeting expected shaft output levels based upon original equipment manufacturer (OEM) specified data. The performance map for the turbine provided by the original equipment manufacturer indicated that the turbine efficiency should be in the range of 72 percent to 75 percent over much of the operating range. Measurement data taken on the unit indicated that actual performance levels ranged from 48 percent to 59 percent. Table 2 lists performance data from the OEM map and actual measured data on the turbine.

Table 2. Turbine Efficiency.

Operating Point	Inlet Pressure (psig)	Inlet Temp (F)	Extraction Pressure (psig)	Exhaust Pressure (" HgA)	Throttle Flow (#/hr)	Extraction Flow (#/hr)	Exhaust Flow (#/hr)	Shaft Output (HP)	Overall Efficiency (%)
OEM Map - 19,000 HP	1450	900	490	4.00	350000	296200	53800	19,000	72.6
OEM Map - Max Power Zero Extraction	1450	900	490	4.00	75000	0	75000	11,500	74.2
Measured Data	1485	900	500	5.92	293160	216820	76340	12,512	47.7
Measured Data	1485	900	500	5.92	267200	195970	71230	14,317	59.3
Measured Data	1485	900	500	5.92	273700	178100	95600	15,435	55.0

Turbine overall efficiency shown in Table 2 is the turbine shaft output divided by the maximum possible power that could theoretically be generated from the available energy in the steam that passes through the unit. Figure 2 shows the steam conditions for the "OEM Map - 19,000 HP" operating point.

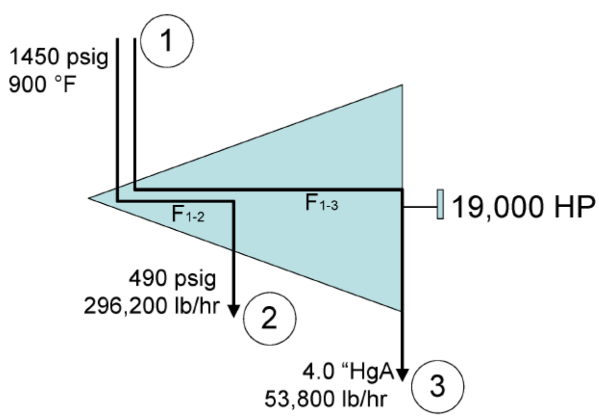

Figure 2. OEM Specified Operating Conditions.

In an extraction turbine there are two steam flows that must be considered to determine the maximum theoretical power the turbine can generate, one going from inlet to extraction (point 1 to point 2) and one going from inlet to exhaust (point 1 to point 3). The maximum power that could be generated from the steam going from inlet to extraction (P_{1-2}) is the steam flow (F_{1-2}) times the available energy in the steam being expanded from point 1 to point 2 (AE_{1-2}). The maximum power that could be generated from the steam going from inlet to exhaust (P_{1-3}) is the steam flow (F_{1-3}) times the available energy in the steam being expanded from point 1 to point 2 (AE_{1-3}). Efficiency is defined as the actual shaft power divided by the sum of P_{1-2} and P_{1-3}. The following is a sample calculation of turbine overall efficiency for the conditions shown in Figure 2:

- AE_{1-2} = 129.53 Btu/lbm
- F_{1-2} = 296,200 lbm/hr
- P_{1-2} = 296,200 lbm/hr * 129.53 Btu/lbm * (1 hp/2546.699 Btu/hr) = 15,065 hp
- AE_{1-3} = 526.03 Btu/lbm
- F_{1-3} = 53,800 lbm/hr
- P_{1-3} = 53,800 lbm/hr * 526.03 Btu/lbm * (1 hp/2546.699 Btu/hr) = 11,113 hp
- Efficiency = 100 * (19,000 hp/(15,065 hp +11,113 hp) = 72.6 percent

The turbine experienced a blade failure on the last stage within the first year of operation. Blade failures occurred repeatedly forcing the unit down and negatively impacting the plant's cost of operation. The last stage blade was an unshrouded (freestanding) design with a "T-root" style dovetail. The blade natural frequencies on this stage were in the range of low multiples of rotational speed of the turbine. Manufacturing variations on the stationary diaphragm can create stimulus on the blade at low multiples of running speed, potentially resulting in stimulus frequencies resonant with the blade natural frequencies. The unshrouded design has no coupling at the blade tip to suppress the fundamental tangential mode of vibration, and therefore the blade is very responsive to resonance with this mode. Since this is a variable speed turbine, it is very likely that a resonant condition will occur at some point across the speed range, and the resulting vibratory stresses would be sufficient to cause the blade to fail. The persistent efficiency and reliability issues forced the decision to undertake a redesign of the turbine steam path in an attempt to correct these problems.

INITIAL TURBINE REDESIGN

An investigation was conducted to determine what modifications could be done to the turbine to address the reliability and efficiency issues. Additionally there was an opportunity to improve plant operation by increasing the capacity of the high pressure section of the turbine to pass more 1485 psig steam, and extracting more 500 psig steam. Table 3 shows the new operating parameters for the turbine.

Table 3. Redesign Operating Conditions.

Parameter	Original Design	Redesign
Inlet steam pressure	1450 psig	1485 psig
Inlet steam temperature	900°F	900°F
Extraction pressure	500 psig	500 psig
Exhaust pressure	4.0 "HgA	5.9 "HgA
Maximum HP section flow	350,000 lbm/hr	400,000 lbm/hr
Maximum LP section flow	75,000 lbm/hr	50,000 lbm/hr

Steampath Redesign

The turbine was completely redesigned including a new bladed rotor assembly, all new stationary nozzle plates and nozzle diaphragms, all new seals, new bearings, new inlet control valves, and significant modifications to the casing. Flow area on the first stage was increased to provide the capacity to pass 400,000 lbm/hr of steam through the high pressure (HP) section and generate more power on the stage. In order to keep stresses within design limits with the increased loading on the blade resulting from the increased blade height and torque, a significantly wider airfoil was used. A three-hook, axial entry dovetail was used to accommodate the high centrifugal loading at 11,000 rpm along with an integral cover with two tenons to carry an outer cover. Figure 3 shows the new stage one blade.

At the reduced maximum low pressure (LP) section flow it was possible to design a single flow last stage blade that could fit within the constraints of the existing casing. The redesign utilized a single 5 inch tall blade to replace the two rows of 3.7 inch blades on the original design. The conversion to a single flow exhaust section eliminated the pressure loss through the crossover along with the inefficient radial flow nozzle on stage five. The new last stage blade was designed with an optimized 3-D airfoil for improved efficiency, along with a three hook axial entry dovetail and a z-lock integral cover. The z-lock cover provides continuous coupling of the blade tips to suppress the fundamental tangential mode of blade vibration (which is a primary concern for tall blade reliability) along with increased damping. The suppression of vibration and the damping significantly reduce potential vibratory stress in the blade compared to the original unshrouded blade that failed. Figure 4 shows the new last stage blade.

Figure 3. Redesigned Stage One Blade.

Figure 4. Redesigned Stage Six Blade.

Rotating and stationary airfoils on all stages of the turbine were redesigned for optimum efficiency at the new operating conditions along with tip seals to reduce leakage losses. Figure 5 shows a comparison of the old technology airfoils typical in the original design and the modern, high efficiency airfoils utilized in the new steam path. The old stationary nozzle designs accelerate the steam through the converging inlet channel and then turn the flow at the exit resulting in high cross channel pressure gradients and therefore high secondary losses. The new design nozzle passage turns the flow and then accelerates it resulting in low secondary and profile losses. The improved inlet wedge angle on the new nozzle efficiently accepts a wide range of inlet steam flow angles lower potential for flow separation. The old rotating blade design was characterized by flat and circular arc construction with sharp surface curvatures conducive to flow separation. The new blade design provides uniform flow acceleration virtually eliminating flow separation and minimizing profile losses. Thinner trailing edges on the new design minimize wake losses and blade inlet angles are optimized to minimize incidence loss.

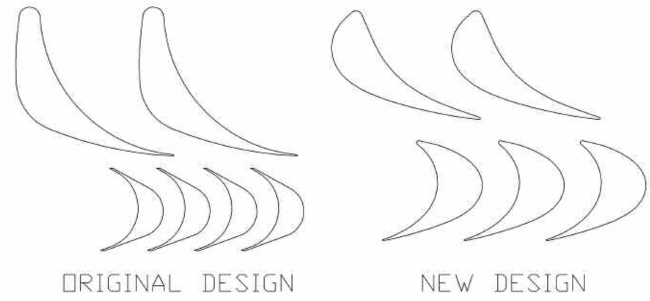

Figure 5. Comparison of Airfoil Profiles.

In order to convert the turbine to a single flow exhaust, significant casing modifications were required. Blanks were welded into the exhaust casing to block off the crossover duct. The casing was stress relieved in the area of the welds and all of the casing fits were remachined to accommodate the new steam path. A new inner casing/flow guide was installed to carry the new stage five and six diaphragms. Figure 6 is a cross-section drawing of the redesigned turbine showing the new steam path and single flow exhaust arrangement. Figure 7 is a photograph of the new rotor being installed in the turbine.

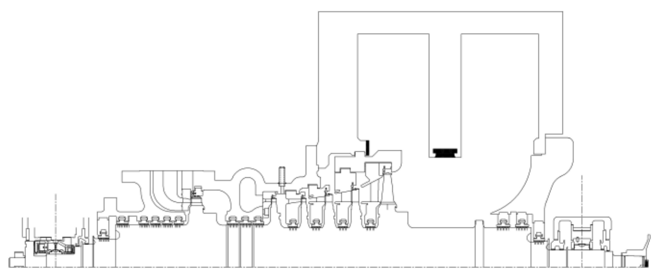

Figure 6. Cross-Section of Initial Redesign.

Figure 7. Installation of New Steam Path.

Startup Issues

The redesigned turbine was installed and run at part load for an extended period of time. The high pressure journal bearing temperature was very high and continued to climb as steam flow to the turbine was increased. The elevated bearing temperature was determined to be the result of steam forces from the partial arc of nozzles on the first stage pushing the HP end of the rotor into the side of the bearing resulting in elevated reaction and undesirable loading angle on the HP journal bearing. Figure 8 shows the configuration of the first stage nozzle arcs and valve opening sequence. With the first three valves open, the steam force exerted on the stage one wheel pushes the turbine rotor down and to the left. This steam force coupled with the rotor gravity load results in a very high load on the HP journal bearing, which caused the elevated bearing temperature. A new V1 camshaft assembly was installed with revised opening sequence as shown in Figure 9. With this opening sequence, the maximum bearing load was reduced and the unit was able to increase load without exceeding bearing temperature limits.

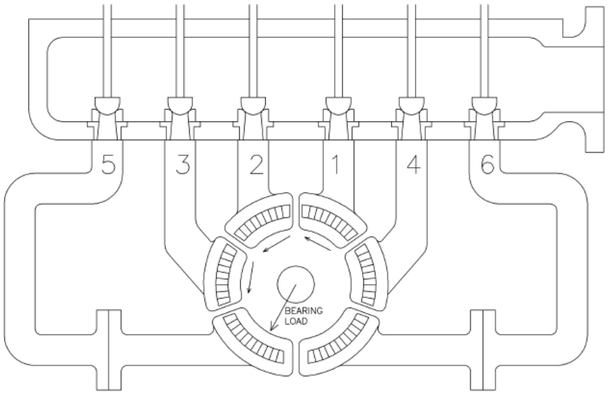

Figure 8. Original Valve Opening Sequence.

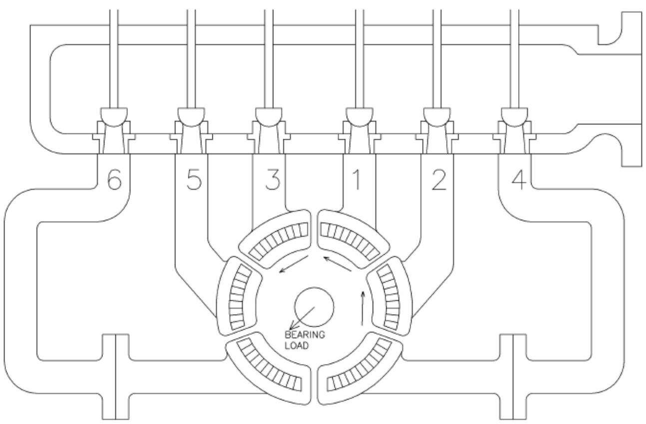

Figure 9. Revised Valve Opening Sequence.

Load was increased on the unit and initial measurements of turbine performance indicated lower than predicted efficiency and shaft power. As additional measurement data were being taken to confirm actual performance levels the unit tripped on high vibration. The unit was opened and a failed first stage blade was found. The redesigned turbine was removed, and a spare (original design) turbine was installed while an investigation of the blade failure and performance shortfall was conducted.

INVESTIGATION OF BLADE FAILURE AND PERFORMANCE SHORTFALL

A comprehensive evaluation was conducted to determine the cause of the blade failure and performance shortfall.

Stage One Blade Failure

A piece of the failed blade was sent to a metallurgical lab for failure analysis. Although the fracture surface was somewhat compromised due to the broken blade getting bounced around within the steam path, the mode of failure was determined to be fatigue. Figure 10 shows the failed stage one blade and damage to adjacent blades.

Figure 10. Failed Stage One Blade.

Finite element analysis of the stage one blade and wheel was completed in an attempt to uncover the cause of the blade failure. All steady stresses were evaluated and determined to be within acceptable limits. Potential vibratory stresses due to resonance with nozzle passing stimulus were evaluated, and again were determined to be within acceptable limits. Another potential source of harmonic stimulus on the blades is the partial arc of steam being fed by the nozzle plate. On most industrial steam turbines, the nozzle plate is designed so that as the valves are sequentially opened, there is always only one continuous arc of nozzles feeding steam to the stage one blades. The stimulus from the partial arc of nozzles and resulting vibratory stress was calculated assuming a single nozzle arc and found to be within acceptable limits. However, the stage one nozzle plate configuration on this unit created multiple separate arcs of nozzles as shown in Figure 11.

Figure 11. Original Stage One Nozzle Configuration.

The nozzle plate is made up of six segments, each fed by a control valve through a separate duct. The four inboard valves feed four upper half nozzle segments, and two outboard valves feed lower half nozzle segments via jumper pipes external to the turbine. This arrangement creates gaps between the nozzle segments creating a stimulus on the blades as they pass in and out of the active nozzle arcs. This unusual arrangement could potentially result in a harmonic stimulus resonant with natural frequencies of the stage one blade and wheel system that may have contributed to the blade failure.

The results of the failure analysis did not point to any one cause that would be expected to cause the blade to fail in such a short period of operation. However, the existing turbine configuration and operating conditions made this a very challenging blade design. The single stage HP section results in very high stage one blade loading. A modern turbine would typically be designed with a two or three stage HP section for these operating conditions, which would dramatically reduce the load on each row of blades. The existing nozzle plate configuration could create a harmonic stimulus on the blades that could potentially result in elevated vibratory stress levels. If these issues could be addressed reliability of the stage one blades could be significantly improved.

Performance Shortfall

Evaluation of measured performance data indicated that the LP section of the turbine was operating at expected efficiency levels. However, the HP section efficiency was measured to be approximately 55 percent versus the predicted efficiency of approximately 70 percent. Measured performance data on the spare turbine with the original blade design showed HP section efficiency to be approximately 52 percent. Investigation of the shortfall revealed issues inherent with the original turbine configuration that resulted in poorer than predicted efficiency.

Nozzle Inlet Duct Pressure Losses

The nozzle inlet duct arrangement shown in Figure 12 creates numerous restrictions for the steam passing from the control valve to the nozzles. Figure 13 shows the geometry of the inside of the duct. This configuration requires the steam to make a sharp turn at the bottom of the port to feed out circumferentially and axially to feed the nozzle arc. There is also an obstruction in the pipe that the steam must bypass before reaching the nozzle port.

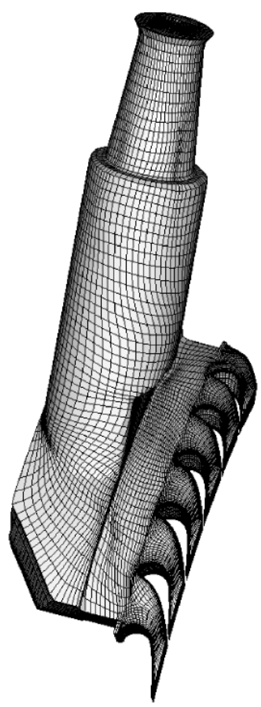

Figure 12. Model of Nozzle Duct.

Computational fluid dynamics (CFD) modeling was completed on the inlet duct to determine if the restrictions resulted in excessive pressure losses that could negatively impact turbine efficiency. Figures 13 and 14 are CFD results showing the steam trajectories through the duct.

Figures 13 and 14 show that the duct geometry does not allow for smooth, unrestricted flow between the control valve and the nozzle vanes. Numerous sharp turns in the steam trajectory results in excessive pressure loss as shown in Figures 15 and 16.

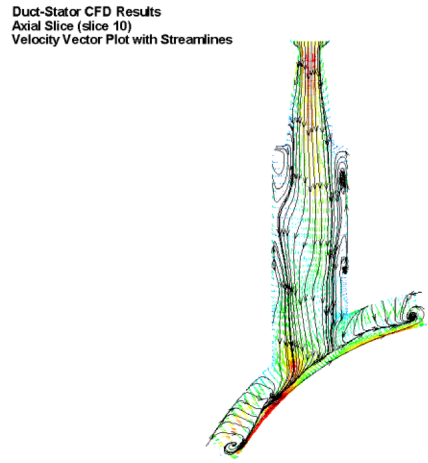

Figure 13. Steam Trajectories in Duct (A).

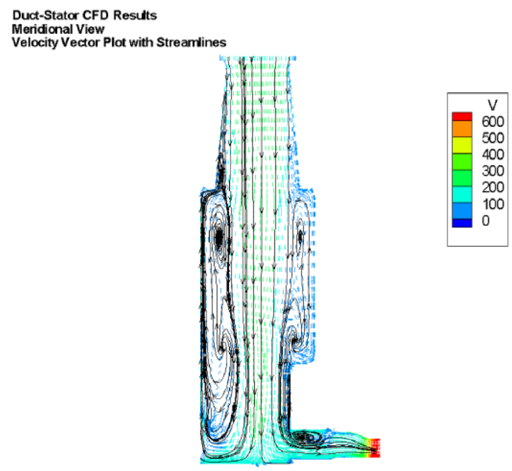

Figure 14. Steam Trajectories in Duct (B).

A significant pressure loss occurs before the steam reaches the inlet to the nozzle vanes. This pressure loss results in reduced available energy across the first stage and a reduction in HP section efficiency. Additionally, Figure 16 shows that there is a variation in pressure upstream of the arc of nozzle vanes. This variation results in a force variation across the arc that could increase the potential harmonic stimulus on the stage one blades mentioned earlier.

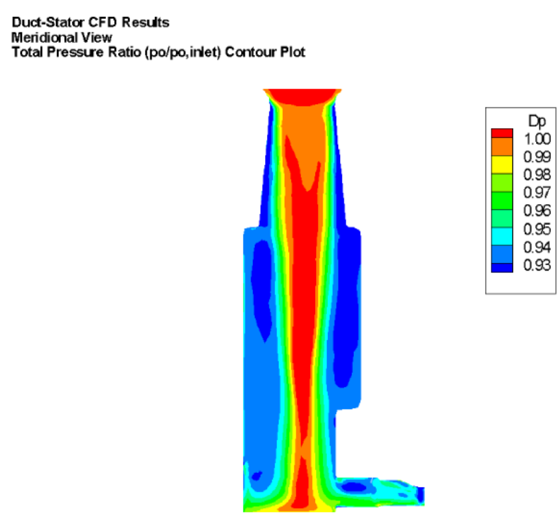

Figure 15. Pressure Loss in Duct (A).

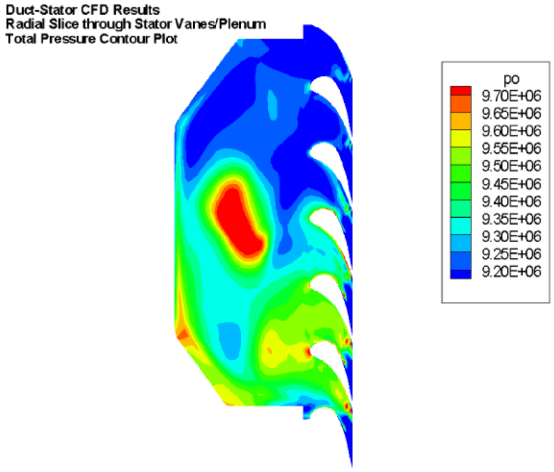

Figure 16. Pressure Loss in Duct (B).

High Pressure Ratio Single Stage Design

In the high pressure section of the turbine, steam is expanded from 1485 psig to 500 psig across a single stage. This very high pressure ratio results in very high (sonic) steam velocities and associated losses. Thermodynamic and aerodynamic analysis of this stage confirmed the measured efficiency of 55 percent and revealed that the originally predicted 70 percent efficiency level would not be possible with a single stage HP section.

SECOND TURBINE REDESIGN

A second redesign effort was undertaken to address the performance shortfall and blade reliability issues. This required changes to the turbine to correct the inherent problems with the original configuration. In order to reduce the high pressure ratio across first stage to improve efficiency and reduce blade loading, the HP section was converted to a two stage design. This required significant modification to the turbine rotor. The rotor was built up with weld in the area downstream of the existing first stage wheel. Two new wheels were machined with axial entry dovetails at reduced diameters. New blades were installed on the two new HP section rotor wheels.

A new stage one nozzle box was designed to eliminate the issues with the original nozzle duct arrangement, and also allowing a reduction in the stage one wheel diameter. This design eliminated the gaps between the individual nozzle arcs and also eliminated the restrictions in the passages between the control valves and nozzles. Figure 17 is a photograph of the new nozzle box.

Figure 17. Nozzle Box.

The two outboard valves that fed the jumper pipes to the lower half were blanked off, and the valve rack was converted from a six to a four valve design with larger diameter valves.

Figure 18 is a cross-section of the new steam path. Figure 19 is a photograph of the new rotor and nozzle box installed in the turbine casing.

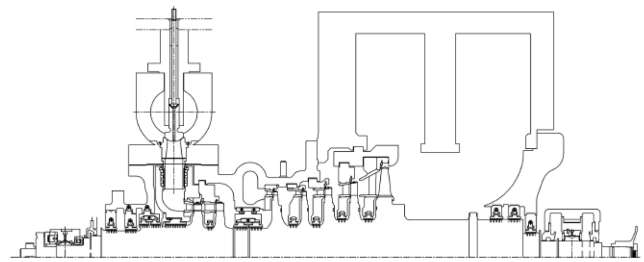

Figure 18. Final Redesign Cross-Section.

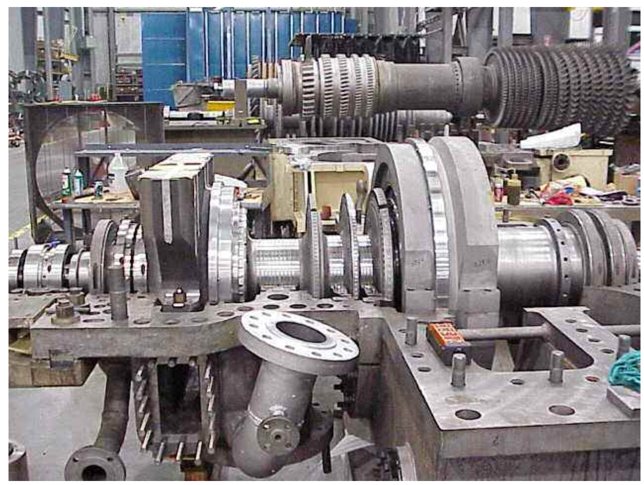

Figure 19. Final Redesign Installed in Casing.

RESULTS AND CONCLUSIONS

The redesigned turbine was installed and started up successfully. Since the startup, the turbine has performed reliably and has met the efficiency targets. The plant was debottlenecked and has been operating at the new capacity provided with the redesigned steam path. The modifications performed to the turbine turned an inefficient, unreliable turbine utilizing obsolete technology into an efficient, reliable turbine comparable to a new turbine that would be provided today utilizing the most modern advancements in high speed steam turbine technology.