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STEAM TURBINES IN COMBINED CYCLE : RELIABILITY & PERFORMANCE IMPROVEMENT SOLUTIONS IAGT 2017

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Astract

Installations of the D-11 combined cycle steam turbines began in the late 1990's and early 2000's, paired with the Frame 7F gas turbine technology such as 2017FA and 2019FA. Presently there are hundreds of these installations around the globe, nearly 200 across the United States and Canada alone. Since this time, several fleet-wide issues have emerged, affecting component reliability, unit availability and steam turbine performance/output. Several of the problems noted have also been observed in other steam turbine models used in combined cycle applications. This paper will discuss key issues, causes as well as potential solutions that are available to owners and operators to aid in the planning process for future maintenance outages.

Turbine Configuration

The D-11 steam turbine design consists of a combined, opposed-flow, HP / IP section with a single-shell construction and a two-flow LP. This is a common configuration that has been used successfully for many years in both fossil and combined cycle applications. In this configuration main steam enters the turbine at the bottom of the high pressure shell via two separate stop and control valves. The flow of HP steam continues in one direction, exits via the cold reheat line where it returns to a 3-pressure HRSG and is re-heated. This re-heated intermediate pressure steam enters the center of the casing via the hot reheat piping and flows in the opposite direction of steam in the HP section. This design results in a relatively even temperature gradient from the center of the casing to the ends. The highest temperature steam in the system enters at the center of the shell and gradually reduces its temperature as it flows outward toward the end packings and bearings.

The combined HP / IP section utilizes a single shell construction, one design for the 7F gas turbine for 60Hz markets and another for the 9F 50 Hz markets. The shells have standard interstage diaphragm support locations and technology is that of impulse, or wheel and diaphragm.

A 'Structured' Steampath

In order to maximize steam turbine thermal efficiency, engineers in the design process need to know the steam conditions a turbine will operate at, such as the inlet pressure, temperature and flow rate as well as the same for the exhaust. With this information, the rotating and stationary steampath airfoils can then be optimized for the given parameters. However this creates many challenges, from the schedule that would be required for such engineering development, as well as associated costs.

In order to keep cost and cycle down, it is possible to structure a family of components for a range of steam conditions. Knowing the gas turbine technology (7FA/9FA) and HRSG configuration, along with a maximum inlet temperature of 1050F would leave only a few variables with left to structure the family of components. This includes operational speed, firing capabilities of HRSG and exhaust backpressure. In the case of HRSG firing, this is generally an end user requirement based on peaking capability. Operational speed would be dictated by the grid frequency and exhaust backpressure influenced by several factors including geographic location and cooling method (air vs. water).

Consideration of the above has led to the structured configurations for the IP and the LP, whereas the HP would have some degree of customization for performance. For configurations using 207FA – this would include 11 HP stages, 7 IP stages and a DFLP utilizing either a 762 mm (30") L-0 blade family, an 851 mm (33.5") L-0 or a 1016 mm (40") L-0. For 209FA configurations, this would include 10 HP stages, 8 IP stages and either an 851 mm (33.5") L-0 or a 1067 mm (42") L-0.

Fleet Issues

Known reliability and performance issues for combined cycle steam turbines include the following items to be discussed in more detail:

- D-11 Fleet Issues
- Known Reliability and Performance Issues
    - N2 Packing Casing
        - N2 Joint Leakage

        - Casing Ledge Cracks
    - HP casing joint leakage
    - Diaphragm Creep Distortion (diaphragm dishing)
    - Bowed rotors
    - Seal Rubs and Clearance Issues
    - L-0 Blade Cracking and Erosion
- A-10 Fleet Issues
- Alstom HP-IP Blade
    - Shim Migration
    - Root Cracking

N2 Packing Casing

Problems with the N2 packing casing (shown in Figure 1) located near the midspan of the HP / IP rotor include joint leakage and ledge cracking. Since the rotor must be removed in order to perform necessary N2 repairs, the potential exists for these repairs to cause an extended outage. For example, if the cracks are severe and cannot be machined out, but need to be weld repaired. Given the combination of the issues with the N2 box owners/operators may elect to pre-purchase a spare N2 casing to have on-hand in order to mitigate risk.

Figure 1: N2 Packing Casing

Steam leakage at the N2 packing casing horizontal joint is shown in Figure 2.

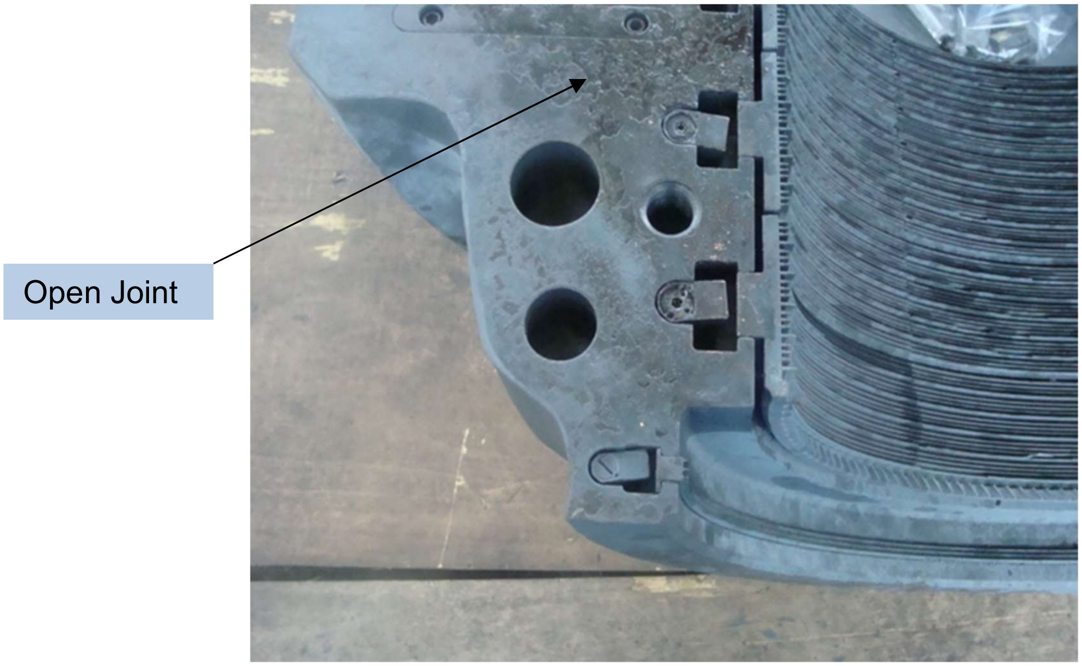

Figure 2: N2 Packing Casing Horizontal Joint Leakage

Loose nuts are observed upon disassembly, and on occasion the studs difficult to remove resulting in the need for machining out the studs. Once the N2 shell is split, evidence of steam leakage is observed across the horizontal joint area as shown in Figure 2. This leakage decreases efficiency of the HP section and causes steam cutting of the studs at the joint area. Leakage also impacts the ability of retractable seals to function properly as the pressure drop across the retractable seal is altered, decreasing segment closure forces – effectively leaving retractable seals in their open configuration during full speed full load operation.

Analysis performed on this area indicates high temperature creep damage of the stud/casing threads occurs, combined with stud preload relaxation. The majority of the creep damage of the OEM 422 bolting material occurs within the first year of operation. Over time when operating at high temperatures, preload relaxation prevents the ability of the joint to remain sealed against further leakage. Furthermore, for distorted joints, a large portion of bolt preload is needed to close the joint at reassembly. Little to no force remains to compress the joint and seal it in operation resulting in further leakage of high temperature steam into the bolt holes, further exacerbating the issue of casing thread creep damage. This has resulted in tertiary creep damage of the casing threads, or thread failure.

OEM recommendations and solution involve changing the bolt material, however, without addressing the lower half casing thread issue.

Another solution option is to change not only the stud material, but also to utilize a double-nut configuration as shown in Figure 3. This solution eliminates the need to use the casing material for threading altogether. This approach is especially beneficial for units that were once designated as base-load but have since increased in cycling duty as the thermal cycling resulting from starts/stops.

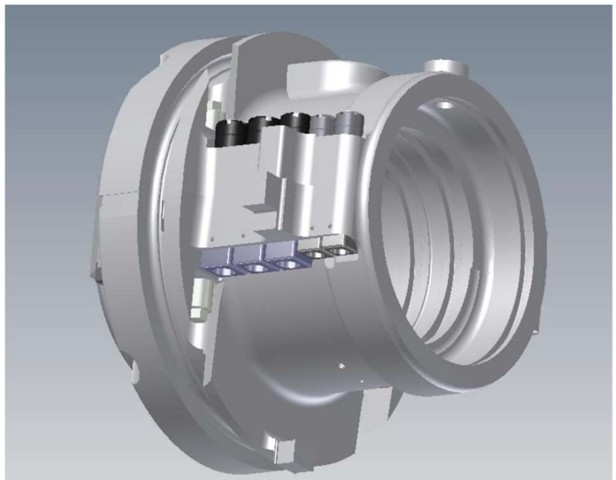

Figure 3: Redesigned N2 Packing Box with Double-Nut Configuration

Ledge cracking as shown in Figure 4 is observed at the main fit to the turbine HP shell.

Figure 4: N2 Packing Casing Ledge Cracks

These cracks occur in the main fit fillet radius on the active face (HP end) both upper and lower halves. Cracks initiate in this area due to low cycle fatigue and creep rupture damage due to the high stress concentration (small radii) combined with high temperature operation of approximately 566 C (1050F).

Solutions available for the N2 ledge cracking issue may be a repair by machining of a larger radii at the shell fit area which removes the crack in the process (shown in Figure 5), or replacement of the entire N2 packing casing altogether (shown in Figure 6).

Figure 5: Machining of N2 Main Fit Radii

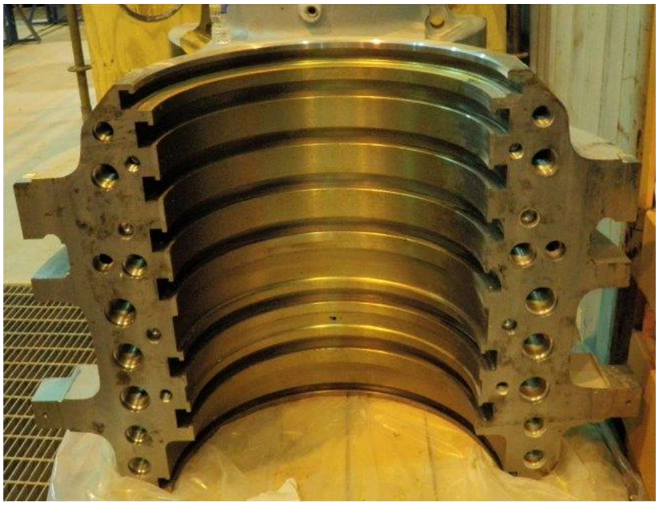

Figure 6: New N2 Packing Casing

HP Shell Leakage Packing Casing

Similar in nature to the N2 packing casing leakage is the HP shell (shown in Figure 7) leakage. As in the case of the N2 box, the HP shell also exhibits leakage area as shown in Figure 8.

Figure 7: D-11 HP Shell

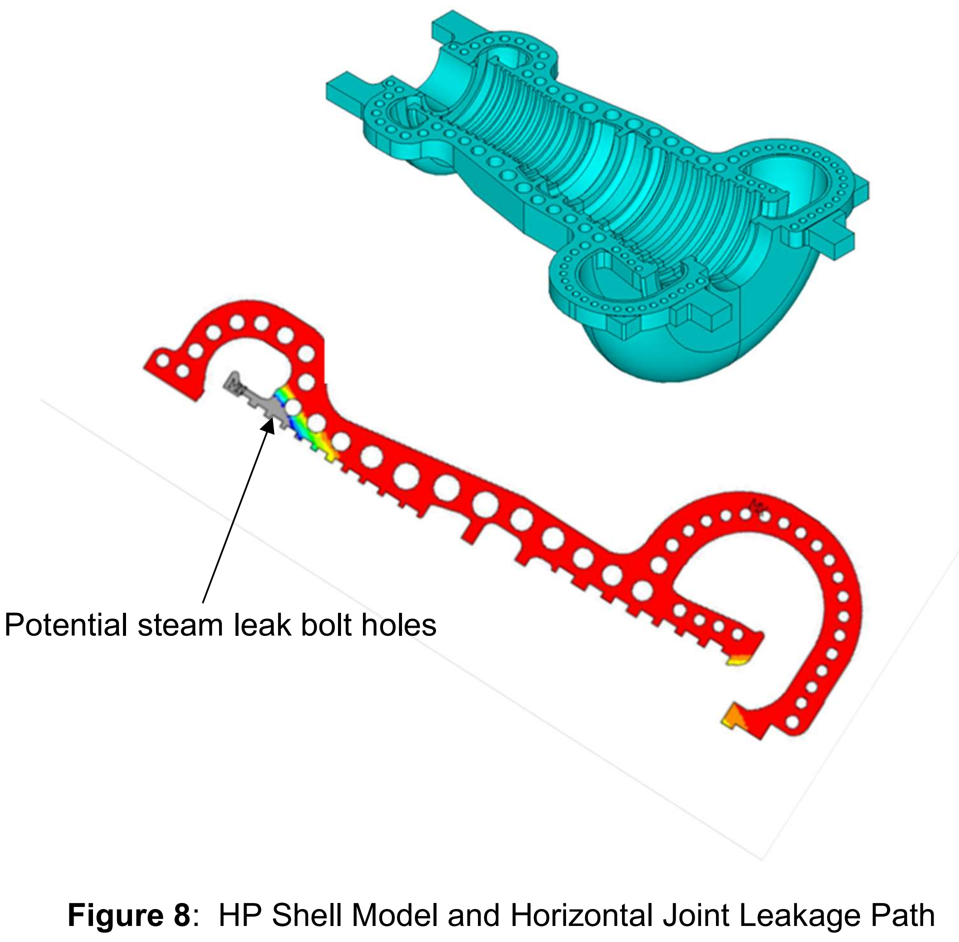

Figure 8: HP Shell Model and Horizontal Joint Leakage Path

Diaphragm Problems

Distortion of diaphragms in the way of creep deformation occurs in the hot stages of the HP and IP sections. This is typically observed upon opening inspection as diaphragm drop checks are performed. In addition to the drop checks, axial rubs with the rotor are observed as are fractured trail edges of the units. Developed solutions for this creep distortion include the installation of stiffeners at the stationary nozzle lead edge sections as shown in Figure 9.

Figure 9: Preparation For Diaphragm Stiffener Welding

Bowed Rotors

Bowing of the HP/IP rotor (shown in Figure 10) is seen across the fleet, due in part to rubs during startup / shutdown, distortion of the casing, water induction events, FOD, improper techniques relating to slow roll, rotor thermal stability / growth problems. Residual stresses and creep/plastic deformation can result, causing a bow to remain in the rotor.

Traditional methods for straightening a bowed rotor include:

- Hung rotor / vertical stress relief
    - Usually only half of original TIR removed by stress relieving
- Heat lathe
    - Similar results as hung rotor method as creep and plastic deformation is not addressed
- Localized heating / hot spotting
    - Residual stresses introduced may relax at different rates than existing stresses

    - Danger of overshooting or local tempering from poor heat input control
- “Throwing” of journals
    - Requires deviated bearing diameter to maintain acceptable clearance
    - Bow may continue to increase over time
    - Interchangeability of spare bearings affected

Figure 10 shows an alternative option available to owners/operators is a proprietary horizontal straightening method.

Figure 10: HP / IP Rotor Preparing for Straitening Process

This method was developed many years ago and has been used successfully as a long term solution. This process involves first understanding some of the underlying material issues for the specific rotor through extensive initial NDE techniques. This inspection process yields a greater understanding of potential residual stresses in the rotor, their location and relative magnitude. Following the NDE, an FEA is performed for each rotor to support a custom straightening process. Figure 11 shows typical results achieved during this process.

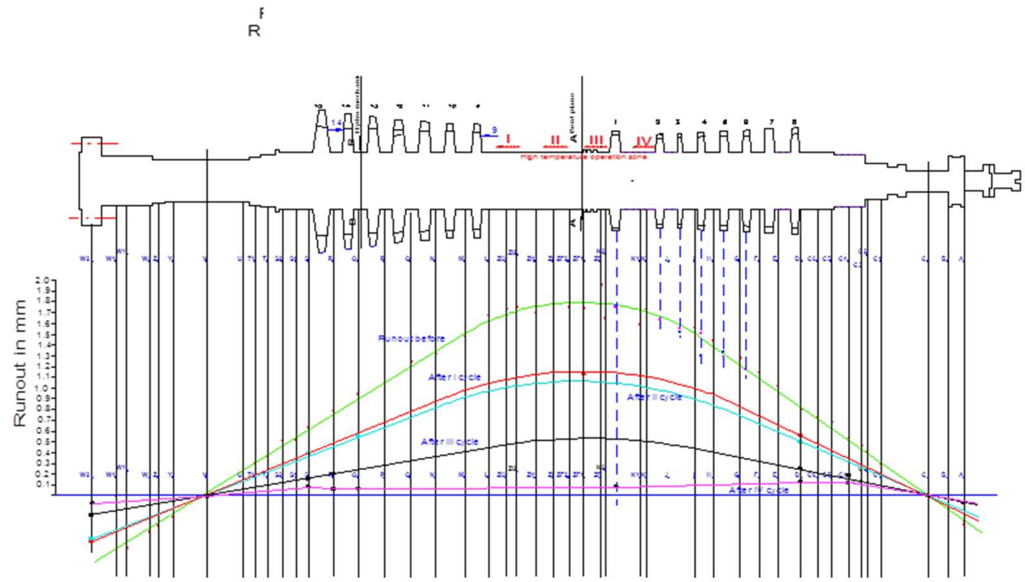

Figure 11: Before and After TIR

Seal Rubs and Clearance Issues

Results from a steampath audit are shown in Figure 12 for the HP casing horizontal clearances and Figure 13 for axial clearances.

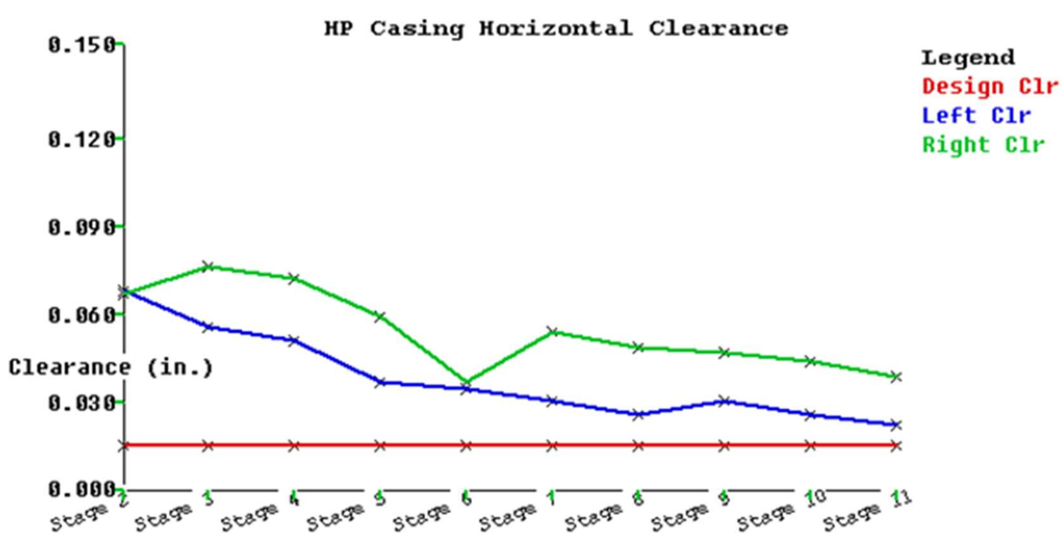

Figure 12: HP Casing Horizontal Clearance

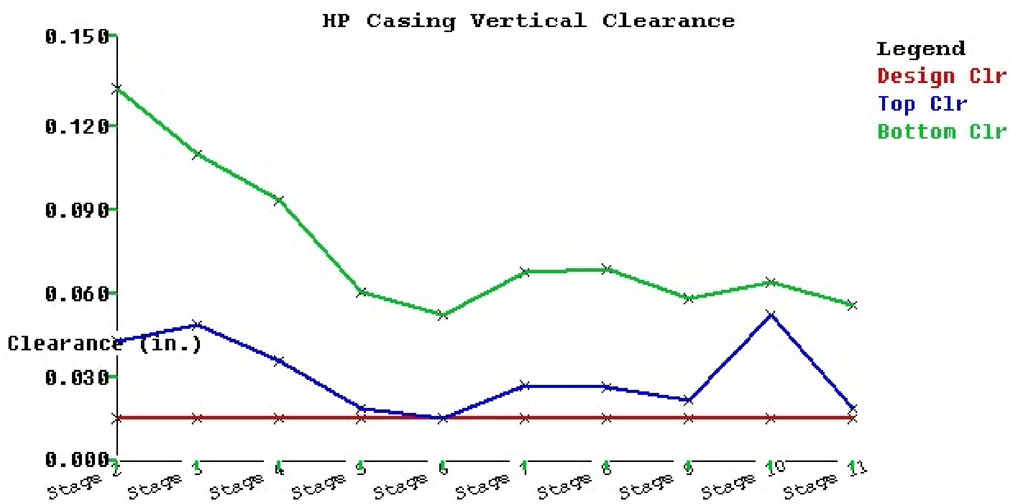

Figure 13: HP Casing Vertical Clearance

For the HP Conventional Packing:

- Avg. horizontal wear = 35 mils
- Avg. top wear = 15 mils
- Avg. bottom wear = 60 mils

IP Conventional Packing:

- Avg. horizontal wear = 15 mils
- Avg. top wear = 20 mils
- Avg. bottom wear = 65 mils

Wear on the bottom segments is significantly greater than top and sides. Similar wear pattern is evident also on rotating on tip seals.

Also noted are heavy rubs of the lower half horizontal joint. Although clearances are generally larger on the bottom, the wear is substantial at all locations, see Figure 14.

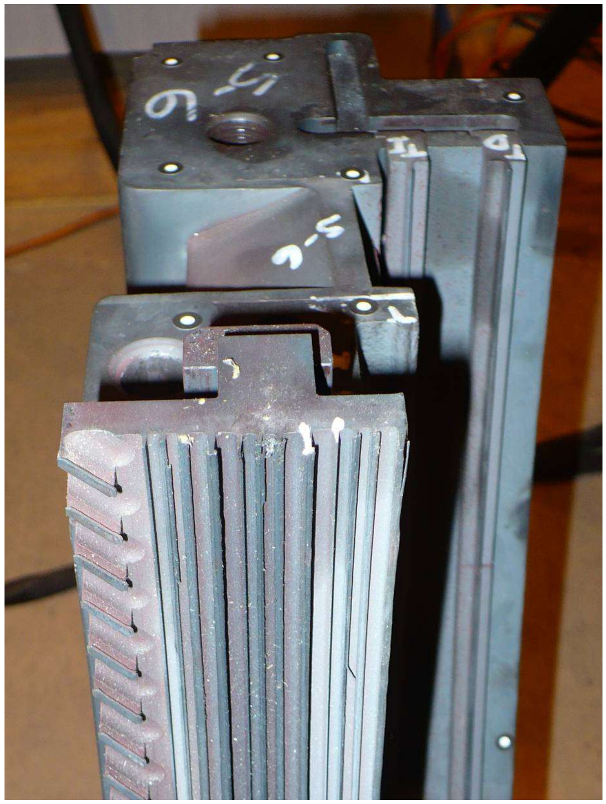

Figure 14: Heavy Seal Rub

Resulting from these seal rubs and a combination of other factors is rotor vibration. The HP/IP rotor in the D-11 is very flexible and is sensitive to midspan rubs. Further analysis indicates an amplification factor of over 24 as shown in Figure 15. Rotor trip on high vibration is 7.5 mils peak to peak, with a max displacement at seal location of .156 mils peak to peak as shown in Figure 16. Therefore, with a radial displacement of 78 mils, subtracting the operating clearance any casing distortion reduces the effective seal clearance.

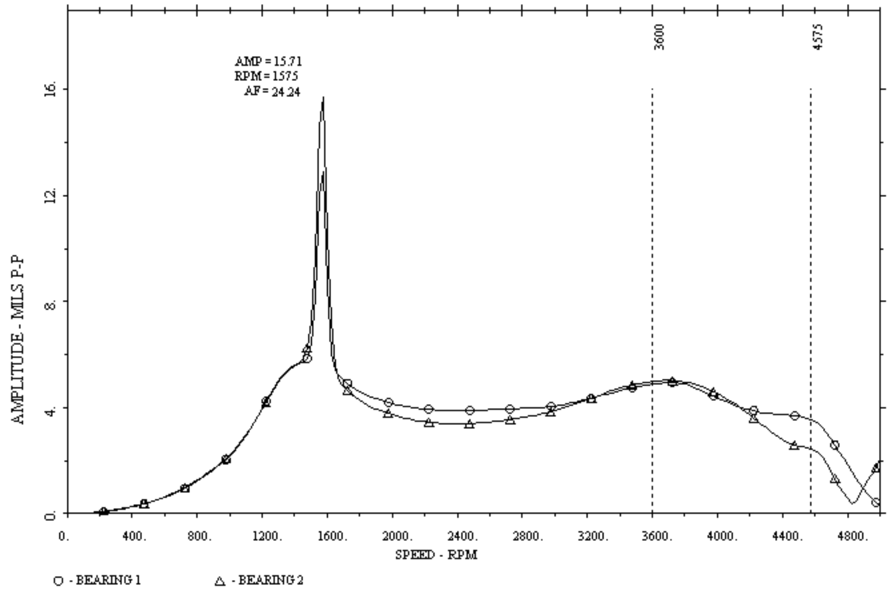

Figure 15: HP/IP Rotordynamic Evaluation

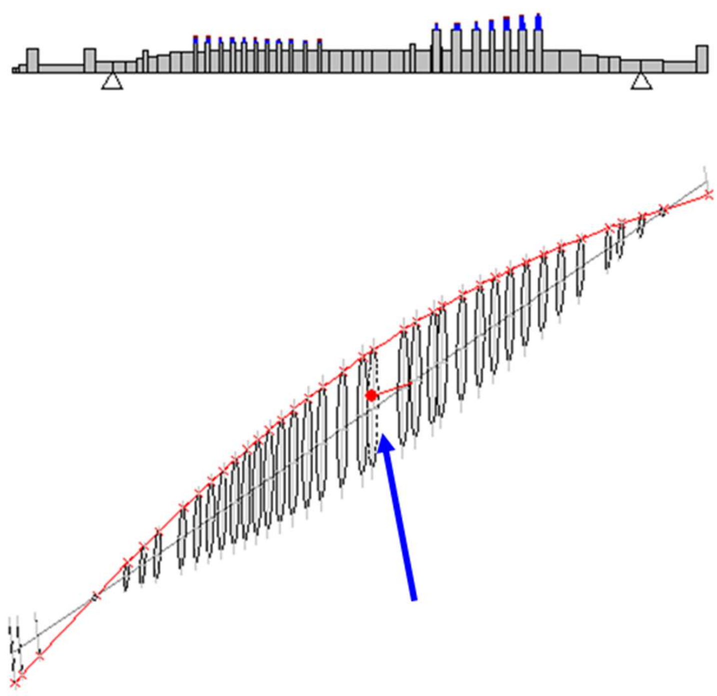

Figure 16: HP/IP Peak Rotor Displacement Orbit

In order to avoid a rub and subsequent wear during startup, while the rotor passes through first critical speed and peak displacement, a seal development is provided as an option to owners/operators. These retractable seals have additional clearance during startup and shutdown as shown in Figure 17.

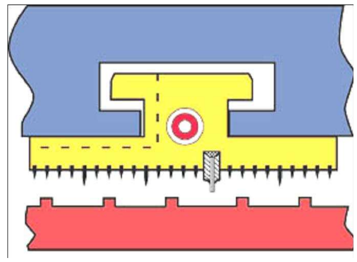

Figure 17: Seal in Retracted State

Once full speed is reached and load is applied, the seal retracts as shown in Figure 18.

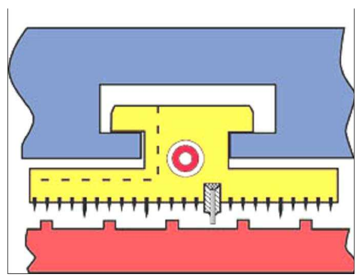

Figure 18: Seal in Operational State

Figure 19 shows a typical spring segment which provides the force necessary to maintain the seal open until sufficient pressure drop across the seal forces it closed. The design of this spring ensures closure takes place at load. Figure 20 shows a similar spring segment which also incorporates a compliant brush seal for added efficiency.

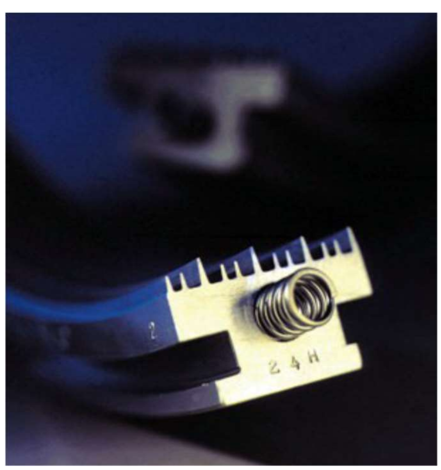

Figure 19: Retractable Seal Segment

Figure 20: Retractable Brush Seal

Similar in nature to the interstage and gland seals, are the tip seals used to seal the areas over the rotating blade shrouds as shown in Figures 21 to 23. Usage of these seals minimizes tip clearance and reduces leakage over previous designs.

Figure 21: Tip Seal Segments

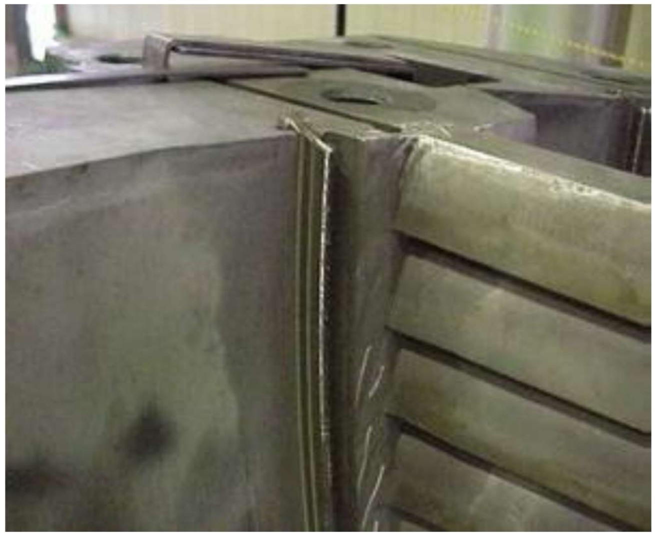

Figure 22: Tip Seal Segments Installed in Stationary

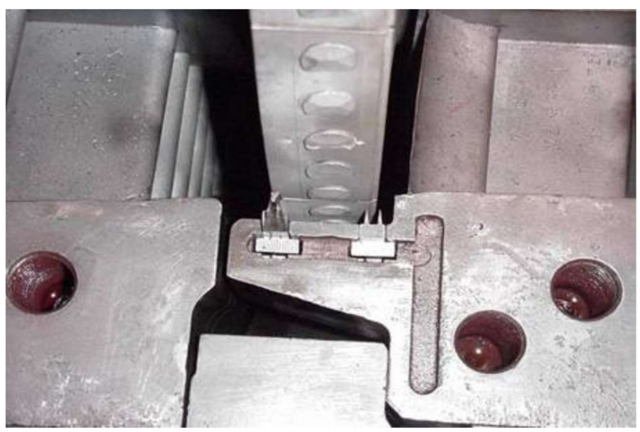

Figure 23: Tip Seal Segments Installed in Rotating

Seal Rubs and Clearance Issues

Tall LP blades are subjected to high centrifugal loads and water droplet erosion during operation. Specific to this fleet is cracking of the pin finger roots. The cracks occur at the corners or transition points on the fingers as shown in Figure 24.

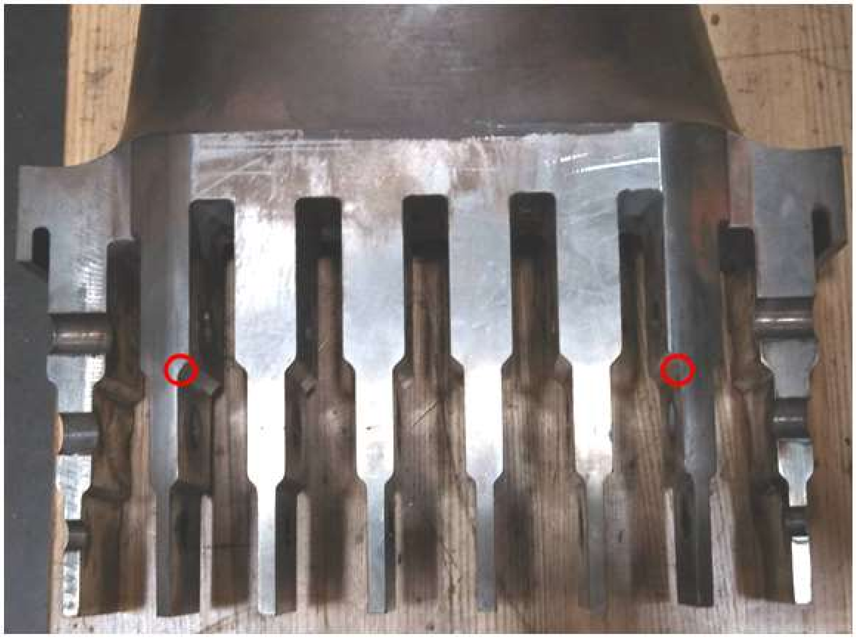

Figure 24: L-0 Pin Finger Root Cracking Locations

This cracking is caused by low cycle fatigue, related to the number of unit starts and stops. This issue is common among the 1219 mm and 1067 mm tall L-0 blades even though the 1219 mm blade is made from titanium and the 1067 mm blade is made from M152. Titanium is highly notch sensitive and the stress concentration factor is therefore higher for a blade of the same geometry but different material. Both blades require the same improvements to the root.

In order to resolve the issue and provide owners with an alternate solution to the OEM's recommended replacement a development project was launched. The result is an option to modify in-service blades by means of a geometry modification as well as proprietary surface treatment which reduces cyclic stresses in these areas. The result of which is approximately a four times multiple of life for the 'recycled' blade option over OEM design. Figure 25 shows the end result of a recycled blade.

Figure 25: L-0 Pin Finger Root Modification

Also addressed is the erosion of the in-service blade with a repair that restores the blade to a like new condition as shown in Figures 26 and 27.

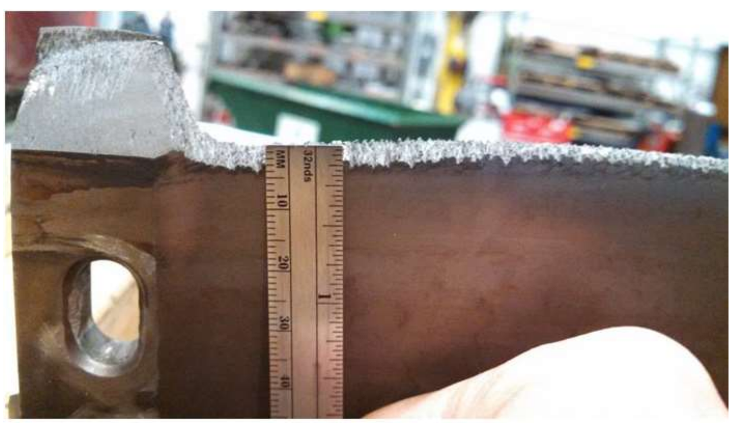

Figure 26: L-0 As found In-Service Blade

Figure 27: L-0 As Repaired In-Service Blade

A-10 Frame Issues

Issues pertaining to this fleet are similar in nature to the D-11 steam turbine. The A-10 is a single flow version of the D-11. Some of the observations and issues for this fleet are:

- Axial Seal Rubs as shown in Figure 28
  - N2 & N3 location occurrence – long rotor condition
  - Excessive leakage at higher loads limits turbine output
- Long Flexible HP rotor
  - Seal Wear
  - Rotor vibration from rubs
- Same LP blades as D-11 (34.5” M152 and 40” Titanium L-0)
- Difficult to set radial clearances and align
  - Tops-on alignment used by some customers

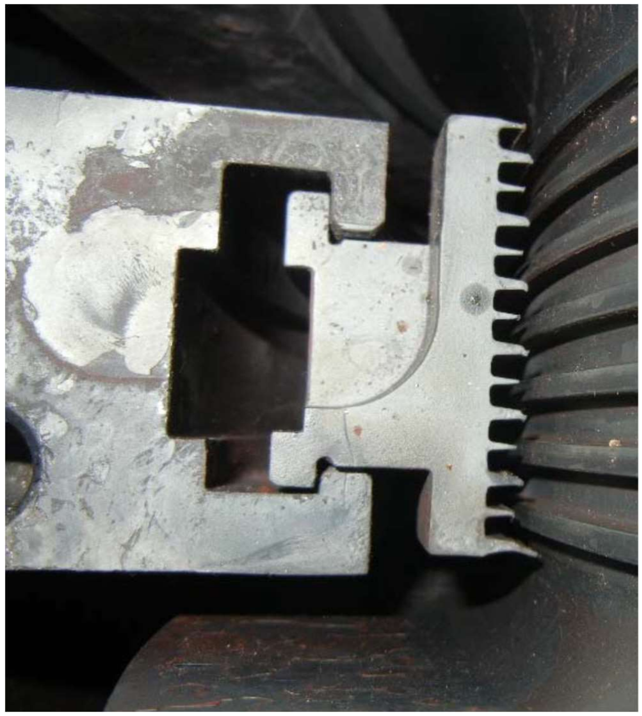

Figure 28: A-10 N2 Axial Seal Rub

Alstom HP/IP Blade Design Issues

One of the issues with this fleet of combined cycle steam turbines is related to the blade-shim design for the HP and IP first stage rotating blades. Failures have occurred by way of shim migration/failure as shown in Figure 29, as well as cracks in the highly stressed area of the blade root fillet near the load carrying hook as shown in Figure 30.

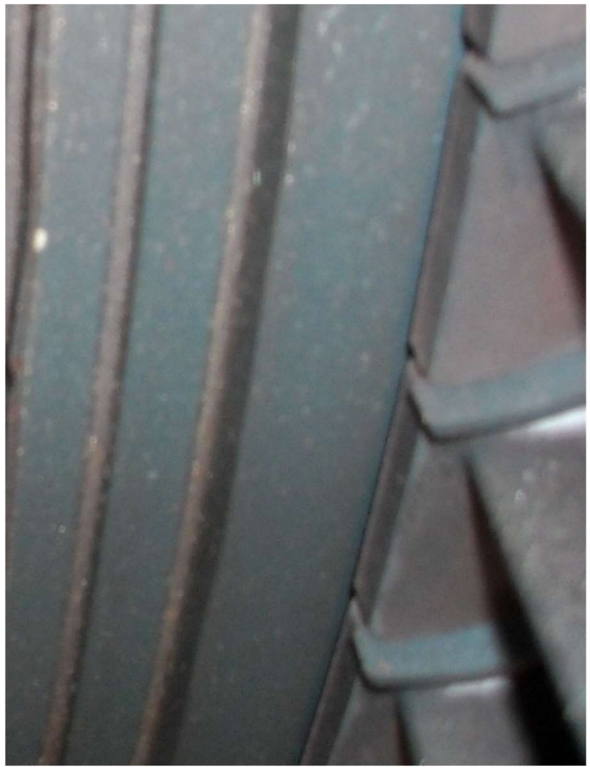

Figure 29: HP/IP First Stage Blade Root Crack

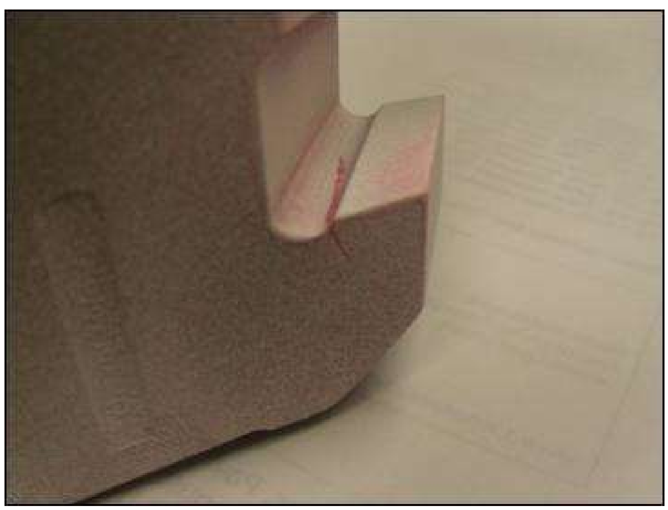

Figure 30: HP/IP First Stage Blade Root Crack

A combination of underlying factors contribute to the cause of these failures such as thermal expansion coefficient differences between the blade and shim material, the installation of blades with a clearance at the blade root, thermal ratcheting as well as not having a locking blade/mechanism or entry slot.

As an alternative to the OEM solution, a development program led to a redesign which has been installed in multiple units without issue. This new design addresses each of the underlying issues mentioned from a geometry change, material change, shim elimination and addition of an entrance slot. This solution is shown in Figure 30.

Figure 30: HP/IP First Stage Blade Root Crack

Summary

Combined cycle steam turbines are complex machines which operate in harsh environments. There is clearly a host of issues that affect reliability and/or performance of these units for which both OEM's as well as ISP's provide solutions for. It is important to note that there are options, which is best for any given power plant will ultimately depend on what the critical evaluation factors are, such as cost, cycle or scope of the project. What is best for one end user may not be the same solution for another, which is why the understanding of the issues at hand along with the options available will provide for the most appropriate evaluation.