# **Steam Turbine Rotor Crack**

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#### Abstract

GE's Machinery Diagnostic Services team was invited to perform vibration measurement on a steam turbine installed to confirm the suspicion for turbine shaft crack. Machine train consists of steam turbine (63 MW), generator (177 MW) and gas turbine (114 MW). In June 2008 there was a regular overhaul after 10 years of operation. The steam turbine had 118 starts over the last ten years. Steam turbine operated normally after the overhaul, with low shaft absolute vibration levels (less than 20 µm pp). In January 2009, after 89 starts in half a year, vibrations increased after 3 hours of operation, vibration levels reached 300 µm pp and pedestal casing expansion difference of about 1.0 mm was observed. In April 2009, the steam turbine was overhauled again. No crack was detected. The analysis of the data allowed drawing some conclusions. The steam turbine rotors showed abnormal thermal and load sensitivity with the abnormal rotor response. The rotor behavior, confirmed high probability of the shaft crack presence on the steam turbine rotor. The modified original slow roll rotor response from June 20, 2008 compared to startup and shutdown slow roll rotor response from July 29, 2008 clearly confirmed the change in the rotor bow, most probably due to shaft crack. The first bending mode resonance frequency was changed from approx. 1750-1800 rpm to 1550-1620 rpm to 1063-1412 rpm. Lower balance resonance speed indicates decrease in rotor stiffness possibly due to shaft crack. The decrease in effective damping demonstrated by higher synchronous amplification factor was likely indication of pending crack. This kind of response could be explained by opening and closing of crack on rotor temperature change. The repeatability of the abnormal rotor response behavior, regarding angle position, confirmed high probability of the shaft crack presence on the steam turbine rotor. The machines with frequent starts and stops are exposed to increase number of passing through the rotor balance resonance and suffer from extensive heating and cooling shocks. This can contributes to the possible shaft crack. The Non-Destructive Tests have been done only on accessible places not under the rotor blades. No shaft crack has been detected. The rotor eccentricity was measured using dial gauges in 16 planes every 45°, to evaluate rotor bow. The results showed increased eccentricity in the section 8 in the direction 180-225°. Finally decision was made to look more in details and the crack was found under the 1<sup>st</sup> stage bucket. The crack was symmetric, 360° circumferential. A rotor model was then built using XLTRC2 software from Texas A&M University in order to see influence of the crack on rotor responseThis case history describes an example of how 1X rotating phase angle was corroborated with other symptoms to identify a rub on a turbo-compressor. The customer had noticed an increasing vibration trend and requested GE to collect data while the machine was operating at nominal speed. Vibration data was collected using an ADRE 408 unit. GE's Machinery Diagnostic Services team was invited to perform vibration measurement on a compressor to investigate reasons of high vibrations

### **1** Introduction

The steam turbine (63 MW, 3000 rpm) is condensing type, downward flow exhaust (Fig 1). The turbine rotors are machined from a solid forging of alloy steel. The rotor is machined to form a balance piston, which is designed to balance the thrust on the blading and to reduce at low value the thrust supported by the thrust bearing at any operating condition. The reduced clearances necessary to control the steam leakages are maintained by seal strips. The rotor is supported by two tilting pad bearings (5-pads, LOP) located in the bearing housing. The front-end pedestal of HP casing contains the thrust bearing



Figure 1: Machine train instrumentation diagram

### 2 Symptoms

Historical background information on the steam turbine showed that it had 118 starts over the last ten years and 89 starts during the last six months, following a recent overhaul. Machines with frequent starts and stops are exposed to an increased number of excursions through the rotor balance resonance; they also suffer from extensive thermal cycling with associated heating and cooling shocks. As such, the numbers of stress cycles incurred following the last overhaul were noted as possible contributors to the possible shaft crack

Various tests were done in the machine shop to confirm this possibility prior to the MDS personnel arrived at site. Nondestructive testing (NDT) and ultrasonic testing (UT) was also performed along the rotor, but only on accessible location, not under the rotor blades. No shaft crack was detected. Rotor eccentricity was measured using dial gauges in 16 radial planes every 45 degrees to evaluate rotor bow. The results showed increased eccentricity (0.09.-0.10mm) in the section near the first blade row, oriented 180-225degrees from the Keyphasor<sup>®</sup> reference position. Because the results were not conclusive, the customer planned to use vibration measurements to verify the possible presence of shaft crack(s) during the machine operation

### 3 Data analysis

The Synchronous Amplification Factor in general, is a measure of the Quadrature Dynamic Stiffness (effective damping) of the rotor system. In general, higher SAF indicates, that the rotor system has lower damping and is more sensitive to unbalance at the resonance.

The Half-Power Bandwidth method is the ratio of the rotor speed at the resonance and the difference of the speeds at the 70% of the amplitude peak at the resonance. The 70% points correspond to the –3dB points (or "Half-Power" points) on the curve

When comparing data from 2008 and 2009 the 1X compensated Bode plots on Fig 3 show only slightly higher calculated Synchronous Amplification Factors using Half-Power Bandwidth Method. The SAF change is from 6.8 to 7.3 measured in the bearing 1 and from 6.1 to 6.7 measured in the bearing 2. Higher Synchronous Amplification Factors possibly indicates decreased rotor system effective damping. Decrease in rotor spring stiffness demonstrated by shifted balance resonances and decrease in effective damping demonstrated by higher SAF are symptoms of pending crack



Figure 2: 1X Bode plot for the startup of June 2008 (Blue/Red) and July 2009(Green/Orange)



Figure 3: Synchronous amplification factor for June 2008 (Bottom) and July 2009 (Top)

Fig 2 shows the startup of June 2008 and July 2009. Slow roll vectors from 2009 startup are significantly (3 times) higher than in 2008, despite 20 hours of rotor turning gear rotation. Increased 1X slow roll vectors indicates increased rotor bow, with the assumption that rotor runout was not changedFor the compared startups the 1<sup>st</sup> balance resonance frequency changed from approximately 1750-1800 rpm to 1550-1620rm. A Lower balance resonance speed indicates a decrease in rotor spring stiffness. The spring stiffness decrease and the increase of the rotor bow demonstrated by changes in slow roll vectors are expected symptoms in case of rotor crack. In general, the symptoms of the propagating transverse crack can increase rotor stiffness asymmetry, which is producing the 2X frequency component, if the rotor is subjected to steady unidirectional radial load. The 2X component can be greatly amplified, as the rotor operates at half of any balance resonance speed, because of exciting the resonance mode by 2X vibration. Fig 4 shows that the

above mentioned symptoms **are not visible in our case**. It is noteworthy that lack of 2X vibration does not exclude the possibility of a rotor crack. Significant 2X vibration can be absent due to either the shape of the crack, which may not cause significant asymmetrical stiffness (e.g., a circumferential crack), or due to insufficient radial preload. Fig 5 shows the difference between startup and shutdown in July 2009. At beginning of Start Up (200-800rpm), a significant change of amplitude without phase change was noticed. At constant speed (800rpm) a significant change of amplitude without phase change is noticed. Further there is visible significant difference in the slow roll vectors amplitudes comparing the same day startup and shutdown on July 29, 2009. The amplitude rise from 36umpp to 73umpp represents double increase. The phase of the slow roll vectors is the same. The phase during the shutdown 2 from 3000 rpm is not changing, neither through the 1<sup>st</sup> balance resonance region of the steam turbine bearing1. The rotor 1<sup>st</sup> balance resonance was shifted lower from startup 1551-1621rpm to 1063-1421rpm at shutdown.

A lower balance resonance speed indicates decrease in rotor/support stiffness, which can be caused by many factors; however the presence of crack, by reduction of the rotor stiffness can cause this effect. This significant difference in rotor starts up and shut down behavior, indicates serious thermal sensitivity of the rotor



Fig 4 – 2X Bode plot from June 2008(Red) and July 2009(Blue).



Fig 5 – 1X Bode plot for startup and shutdown in July 2009. Blue: direct, Red: 1X uncompensated, Dark green:2X, Light green: beginning of SU, Black: constant speed.

Fig 6 and Fig 7 show the trend plots of direct and 1X for steam turbine bearings during startup and loading, recorded on July 27, 2009. The amplitudes of shaft relative vibration on bearing 1 and 2, are reaching the minimum approximately in 1 hour after the synchronization. The shaft relative vibration phase is changing 180 degrees in this region (pink) at constant load and is changing again 180 degrees in the second region (orange) as the load is increased. In addition, the shaft relative vibration phase is changing 180 degrees in the speed region 2300 rpm-3000 rpm (Blue-Direct, Red-1X), where no balance resonance is expected



The DCS process trends (Fig 8) are showing the correlation between the eccentricities, steam turbine shaft relative vibrations and casing temperatures. When the steam turbine casing temperature reached the maximum (about 387°C), the vibrations and eccentricity measurement dropped to minimum and then started to increase. The machine was manually tripped after 2 hours at 120MW due to turbine bearings and clutch vibration growth over the alarm limits. After machine trip, vibrations, mainly on bearing 1 and eccentricity continued to increase. The eccentricity pk-pk reading increased to 200umpp at 14rpm then started to decrease



Figure 8: DCS trend showing eccentricity, shaft relative vibrations and casing temperatures

A second start up was performed after the previous trip. The steam turbine was synchronized to minimum load 9MW and vibration behavior was tracked on constant speed and load. The steam turbine vibration started on low vibration levels (60umpp), following the amplitude slow decrease with rotor heating. Approximately in 3 hours from the start, when the steam turbine casing temperature reached the maximum (about 372°C), the vibrations and eccentricity measurement drooped to minimum and started to increase with no phase change. This erratic rotor response was the reason to manually trip the machine, with suspicion for rotor crack. After the machine trip vibration on bearing 1 and eccentricity continued to increase. The eccentricity pk-pk reading increased to 96umpp at 14rpm and then started to decrease



Fig 9 – Direct and 1X trend plot for bearing N°1.

Fig 10 – Direct and 1X trend showing plot for bearing N°2.

Fig 9 and Fig 10 show visible amplitude and phase change during steady state at constant load. The amplitude of shaft relative vibration on bearing 1, is reaching the minimum in 1 hour after synchronization. The phase is not changing significantly. The amplitude of shaft relative vibration on bearing 2 is oscillating and is reaching maximum at this moment. The shaft relative vibration phase is changing 180 degrees in this region (orange) and is changing again 180 degrees in the second region (green).



Fig 11 – 1X polar plot for bearing N°1 at steady state



Fig 12 – 1X polar plot for bearing N°2 at steady state

Fig 11 and Fig 12 show abnormal unidirectional rotor response change on temperature change which is another symptom supporting the possibility of presence turbine rotor crack. The change of the shaft relative vibration and eccentricity (Pk-Pk) measurement, together with phase step changes (in 1-2min), always in the same direction, suggest sudden changes in the rotor bow possibly caused by closing and opening shaft crack

Polar plots (Fig 11 and Fig 12) for bearings 1 and 2 show the unidirectional change of the shaft relative vibration during the rotor heating (orange). As the rotor thermally equalize, in our case in 1 hour after the synchronization (in 3 hours from start up), the direction of the vibration suddenly change to the opposite direction (green). The possible explanation of thermal sensitivity for the axially heated rotor is that the transverse crack acts as the insulation. The heat transfer is faster on the opposite side of the crack and the rotor starts to bow in this direction, causing the crack closing until temperature equalizes. There are also some models described in the literature for radial heat transfer, concluding that for transverse crack the heating of the rotor should cause the temporary closing the crack (surface of the shaft expand more and faster than the center of the shaft) until temperature equalizes



Fig 13 – 1X polar plot for bearing N°1 at 3000 rpm Fig 14 – 1X polar plot for bearing N°2at 3000 rpm

Fig 13 and Fig 14 show the original rotor response (orange) **at 3000rpm** at Full Speed No Load (FSNL) on June 20, 2008 (Orange) and the modified rotor response 1 (Blue) at FSNL at the beginning of steady state and the response 2 (green) at FSNL at the end of steady state on July29, 2009. The polar plots display the same direction for the rotor response 1 and response 2

Further, it is visible that original unbalance response vector is acting against the static bow unbalance, due to crack. The crack creates static bow, which is dominant at low speeds. This static bow unbalance is interacting with the dynamic unbalance, due to unbalance forces dominating on higher speed. The resultant 1X rotor response depends on the relative phase of the static and dynamic unbalance. The basic rule in shaft crack detection says that the cracked rotor will very likely experience the bow and the bow is likely to change over the time, with development of the crack. The change in rotor bow will change the slow roll response as well as the effective location and magnitude of the heavy spot, which will change the 1X rotor response



Fig 15 – 1X effect vector for bearing N°1 due to change unbalance



Fig 16 – 1X effect vector for bearing N°2due to change unbalance

On Fig 15 and Fig 16 the effect vectors show the rotor unbalance change, caused by the rotor-modified bow, due to shaft crack. The effect vectors for bearing 1 and bearing 2 are in the same direction 190deg-220deg from YE-201 reference transducer (0° reference), what confirms the same rotor bow effect on both bearings in the same direction

The amplitude ratio of the effect vectors is similar for each bearing (effect vector 2 / effect vector 1=1,3), what confirms the similar effect of the rotor bow, due to Shaft Crack on each bearing

#### 4 Conclusions

From the analysis of the test performed the following conclusion can be drawn:

- The modified original slow roll rotor response clearly confirms the change in the rotor bow, most probably due to shaft crack.
- The first bending mode resonance frequency was changed from approx. 1750-1800rpm (SU June20, 2008) to 1550-1620rm (SU July29, 2009) to 1063-1412rpm (SD July29, 2009). Lower balance resonance speed indicates decrease in rotor spring stiffness, possibly due to shaft crack.
- The decrease in effective damping demonstrated by higher Synchronous Amplification Factor (SAF) is likely indication of pending crack.
- The Steam Turbine shaft shows abnormal thermal and load sensitivity, with the abnormal rotor response. This kind of response can be explained by opening and closing of crack on rotor temperature change.
- The repeatability of the abnormal rotor response behavior, regarding angle position, confirms high probability of the shaft crack presence on the steam turbine rotor. The estimated direction of the shaft crack is 180-200deg from YE201 reference transducer, measured against the rotation. It represents, approximately opposite direction to the keyphasor notch reference. The excepted shaft crack location is probably closer to the steam turbine bearing1

The recommendation was to remove the rotor from the machine and confirm the presence of a crack using the Nondestructive Testing (NDT) techniques on the entire rotor, with special attention to regions that were not tested before (i.e., areas that including the blade grooves). Because of the costs to de-blade the rotor, these tests were not previously performed; however, the vibration test results described in this article were considered sufficient justification to incur the additional costs of de-blading and NDT re-testing. The rotor was removed from the machine, an impact hammer test was performed, and a decreased first balance resonance at 19Hz (1140rpm) was confirmed, as predicted in the report. The buckets from stages 1 through 28 were removed to investigate the bottom of the dovetail. A rotor crack was found under the first-stage bucket. The crack was symmetric and 360° circumferential, explaining the absence of an observed increase in the 2X frequency component



Fig 17 – Picture showing the crack found under the 1<sup>st</sup> stage bucket

## 5 Rotor Modeling

Original Rotor Model was created in CALC19 –old modeling software, therefore new model (Fig 18) had to be created in XLTRC2 software from Texas A&M University. Objective was to simulate the crack and estimate the depth.

Rotordynamic model of the motor was created and typical set of calculations was presented. The model is not validated because this requires additional experimental data, not available at the moment. However the method of validation was described in case the project will be continued. Therefore the included calculations are for demonstration purposes only. Design decisions should not be made based solely on conclusions from the model until the model validated. A basic validation of the model geometry was performed but an impact test with comparison of results with free-free mode calculation should be performed to validate the rotor model.

Lateral analysis is performed using number of tools. A typical rotordynamic sequence is made of the following steps:

- Build shaft model & validate
- Expand to rotor model & validate
- Run undamped critical speed map
- Include bearings in the model
- Run damped response
- Run forced response

The linear analysis is concerned with the solution of the differential equation below:

#### $M\ddot{x} + C\dot{x} + Kx = F(\omega)$

For an unforced solution the forcing function on the right hand side  $F(\omega)$  is set to zero and the homogenous differential equation is solved to find the eigenvalues and eigenvectors. The eigenvalues are complex numbers representing the frequency and damping of each mode. The eigenvectors give the relative magnitudes of vibrations of the nodes for each mode. To find the undamped natural frequencies and mode shapes the equation is solved with the damping matrix ignored. The damping affects the response most at resonance. For excitation frequencies away from resonance, one can get reasonable and conservative estimates of response if the damping is not included. The free -free analysis is performed for the rotor without including the effect of supports.

The most common forced solution is the unbalance response. The unbalance force is at the same frequency as the running speed and its magnitude is related to the square of running speed. While the unforced mode shapes give only the relative magnitude, the forced unbalance response predicts the estimated level of

vibration for assumed unbalance. This harmonic response analysis is the most common type of analysis. Fig 18 shows the model



Fig 18 – Rotor model with fine tuning

A free-free analysis is then performed to calculate frequencies for comparison to frequencies measured in an impact hammer test. If a machine runs near or above any rotor modes with a lot of shaft flexing and little damping it could be worth making sure the shaft model is accurate as possible. Comparison with measured frequencies can be used to validate the model, or fine tune it. The free-free analysis doesn't include any information from the support.

An undamped critical speed analysis is then performed to provide a quick look at the vibration of critical speeds with bearing stiffness. The undamped, unforced system is solved at each stiffness value. Note that the direct stiffness of the actual bearing used in the model is overlaid on the map. A critical speed will arise at the points of intersection. One had to be concerned with the critical speeds that occur near to the running speed range.

Note that the first bending mode at zero stiffness is the first free-free mode calculated in the free-free analysis. A way to validate the model is to have a test done with the shaft hung in ropes. The differences between the calculated free-free mode and the impact test results are used to ensure correctness of the model. A damped analysis should then be performed to provide the mode's frequency and amount of damping.

The unbalance response is the most common lateral analysis. The value of the unbalance is determined by the amount, phase angle and axial location. For the unbalance response analysis the load distributions are chosen such that they affect the relevant modes most adversely

Once the model was tuned using different parameters for the preload and the bearing clearances the crack was simulated by a change in rotor diameter at the location of the crack. In the case of strong variation in diameter, it is suggested to take into consideration in the model the part of the material which does not participate to the stiffness by an intermediate element, as in Fig 19.



Fig 19 - Rotor model with Outer Diameter of 0.2m at the location of the crack

Fig 20 shows the results from the model when using an Outer Diameter of 0.2m at the location of the crack. Critical speed and SAF were reduced significantly because of the crack. This model was created only for demonstration purpose and shouldn't be used for design



Fig 20 - Cracked Rotor model showing change in SAF and critical speed



Fig 21 – Picture showing the crack surface

## References

[1] Bently D.E, Hatch C.H, Fundamental of Rotating Machinery Diagnostics, Bently Pressurized Bearing Company, 2002