Case History:
Field Balancing of a Bowed Steam Turbine Rotor
Vibration Institute Regional Training Conference
Peek ‘n Peak Resort, Clymer, NY
April 30th, 2015

My Background…

- Stan Bognatz
- 30+ years Rotating Machinery & Reliability field experience
- Prior to founding M&B in 2005, managed & staffed GE / Bently Nevada’s Machinery Diagnostic Services (MDS) field-engineering team for nearly 20 years
- Specialize in critical machinery analysis (gas/steam/hydro-turbines; high-speed drives, etc.); balancing of turbo-machinery; modal/ODS analysis; optical / laser / 3D alignment; predictive maintenance; oil analysis; customer training
- Authored technical papers on diagnostics, modal analysis, balancing, optical alignment, systems integration & oil analysis
- B.S. degree, Mechanical Engineering, Penn State University
- Masters Degree in Business Administration, Wilkes University
- Registered Professional Engineer (in PA)
- Certified Level III Vibration Analyst via the Vibration Institute
- Certified Maintenance & Reliability Professional (CMRP) via the Society for Maintenance & Reliability Professionals (SMRP)
Our Company

- We provide rotating machinery users a value-driven, OEM-independent source for reliability-based engineering services & instrumentation:
  - Advanced vibration analysis, machinery diagnostics
  - High-speed / Multi-plane rotor balancing
  - Machinery alignment / Optical alignment / 3D-Laser Tracking
  - Predictive Maintenance services & program management
    - Vibration, Thermography & Oil analysis
    - Motor current signature analysis
  - Vibration / condition monitoring Instrumentation
    - Sales & service of Bently Nevada, Metrix & other OEM equipment
    - Turnkey installation & design; sub-contractor management
    - Calibration & maintenance
  - Customer training courses
Case History Summary

- Nov-2014: A 60MW HP-IP Steam Turbine Rotor incurred a permanent bow after the turning gear motor failed to start following a hot shutdown.
- Subsequent time on turning gear & slow-rolling at 1000 rpm did not relieve the bow. The unit could not be started without vibration exceeding 10 mils at 1500 rpm, well before the 1st critical speed at 2100 rpm.
- Multiple start-up attempts were made over several days, with no significant change in the vibration characteristics with each run.
- Site / operational constraints dictated the rotor could not be removed for corrective straightening & shop balancing, and the site elected to perform in-place balancing.
- M&B was contracted to oversee the project, and we successfully field-balanced the rotor to ~4 mils, allowing a return to service and reduced lost generation revenues.

This presentation will review some key areas of this interesting project:
- Machinery layout / rotor geometry
- Event timeline & data analysis
- Balancing calculations and overall methodology
- Some pros & cons of doing a field balance solution in this situation

Machinery Layout

- 3 Bearings for the HP-IP & LP turbines
  - Note Bearing #2 is “shared”, which creates interesting dynamics when balancing on the HP-IP or LP rotors

- Vibration monitoring system & data:
  - Bently Nevada 3300XL, 8mm proximity probes at each bearing in XY configuration
  - All signals brought into a Bently 3500 rack, and ported to ‘System1’
  - All data in this presentation was prepared remotely using System1
Event Timeline & Data Analysis

- November 2014: machine operating acceptably; 1X-filtered vibration ~ 3 mils
- 12/4/14, mid-day: the unit was shut-down due to switch-yard issues
  - Unit coasted down ok; HP-IP vibration < 3 mils through 1st critical at 2100 rpm
  - Rotor then reached zero-speed, but the turning gear motor would not start
  - The rotor sat stationary for “some time” while the motor issue was reviewed
  - Site eventually began turning the shaft manually, and re-started the unit later that evening

- Bode plots below show the low vibration at Bearing #1 & #2 during shut-down:
12/4/14 – 2 Failed Start-up Attempts

- **Startup at 7:15 pm (shown here):**
  - Slow-roll / run-out was 1-1.2 mils before start-up
  - Vibration increased rapidly above 600 rpm, reaching 11 – 12 mils at 1,500 rpm
  - 1X-filtered shaft orbits remained oval / elliptical, and were forward precessed; no signs of rotor rubbing
  - The unit was shut-down and placed on turning gear for ~ 3 hours
- **Re-start at 10:20 pm:**
  - High vibration again, same pattern
  - Slow-roll was higher (2+ mils) during start-up, nearly doubling from previous run
- At this point the site recognized that the rotor may have incurred a bow earlier in the day, so the unit was shut-down and put on turning gear for the night.

12/5/14 – 1st Failed Start-up

- **During the 1st Start-up the turbine was taken directly to 1500 rpm**
  - Slow-roll / run-out during early part of start-up was ~ 1 mil, so the rotor did not appear bowed based on that data, and site continued to increase speed
  - Vibration increased steadily, nearly identical to the 12/4/15 runs:
    - Bearing #1: 10 – 11 mils at 1550 rpm
    - Bearing #2: 9 mils
    - Bearing #3: 3 mils
  - The unit was shut-down and placed back on turning gear
12/5/14 – 1st Failed Start-up

- The 1X polar plots at Bearings #1 & #2 showed a high-magnitude, in-phase relationship, indicating a significant 1st mode unbalance response.
- Severe vibration, and the rotor was still operating well below the 1st critical speed of 2100 rpm.

Typical 1st Mode Rotor Response

- 1st Mode characteristics:
  - At low speed (~200 rpm) 1X phase angle points to the Heavy Spot.
  - At resonance (1830 rpm) the amplitude peaks and phase has shifted 90 degrees.
  - Above resonance, amplitude decreases and phase shift approaches 180 degrees.
  - The resonance peak phase angle is nearly equal at both ends of the rotor.
  - Peak amplitudes may be a different for asymmetrical rotors or unequal unbalance distributions.

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Note no significant phase angle change from slow-roll up through 1500 rpm.
The unit was re-started a few hours later and taken to 1920 rpm, with severe vibration:
- Bearing #1: 10 mils at 1580 rpm, and then climbed to 13 mils
- Bearing #2: initially peaked at 11 mils at 1700 rpm, and was decreasing until 1850 rpm, but then vibration began increasing very rapidly, reaching 22 mils
- Bearing #3: reached 10 mils near 1800 rpm, and eventually peaked at 13 mils

So, at 1920 rpm the unit was shut-down (manually?)

Vibration at Bearing #1 continued to climb as turbine speed decreased, reaching and holding ~17 mils from 1700 down to 1500 rpm
- This is a common behavior during a "heavy" rub.
- As the rotor continues to bow from friction & heating, it cannot clear the rub.
- As speed continues to drop, the centrifugal forces eventually decrease enough to reduce the dynamic bow, and the rub "lets go".

The unit was placed back on turning gear until 12/7.

It is still unknown why plant operators allowed the vibration to reach the extreme levels before shutting down. The turbine should have been shut back down when it was obvious the vibration was going to exceed 8 – 10 mils.

Classic heavy rub: vibration during shut-down was higher than start-up, and remained high for an extended period with a “flat top” on the bode plot.
12/5/14 – 2nd Failed Start-up – Extreme Vibration

**Brg. 2X – Start-up**

- Classic heavy rub:
  - Vibration increasing rapidly with only minimal change in speed above 1900 rpm
  - ‘Flat-top’ bode plot during shut-down with higher levels than during start-up

**Brg. 2X – Shut-down**

- Rotor slow-roll / run-out at low speed during shut-down did not show any severe bow. Odd behavior considering the vibration levels during the rub.

12/5/14 – 2nd Failed Start-up – Extreme Vibration

**Brg. 3X – Start-up**

- Bearing 3 was going along for the ride.

**Brg. 3X – Shut-down**

- Considering the LP Turbine rotor's symmetrical layout, and with nearly double the vibration at Bearing #2, the data was pointing to the HP-IP rotor as the rub source.
12/7/15 – Failed Start-up Attempt

- 12/7/15: unit re-start at 7 am
  - Vibration reached 10 mils at 1500 rpm
  - Unit was shut-down (no sense taking any chances, right??)

![Graph: Vibration vs. Speed]

- Moving along….

- 12/8/14:
  - We were eventually contacted by the site to discuss situation
  - Vibration data from the various runs was gathered & reviewed

- Various items were discussed and checked as follows:
  - Run-outs measured with dial indicators adjacent to the bearings. These showed less than 2 mils, so no significant bow was noticeable (but, these readings were very close to the bearings).
  - The HP turbine rotor was bore-scoped on the last row of blades (the only accessible area); there was no visible damage that would have accounted for the large 1X vibration (i.e., damaged / lost blading).
  - Since there was no ‘smoking gun’, a decision was made to remove the turbine shell for an internal inspection.

- 12/13/14:
  - Shell removal was completed
  - Some rubbing damage was found on the blade shrouds and inter-stage diaphragms near rotor-center, and at/near the N2 glands (labyrinth seals)
  - Dial indicator run-out readings showed a significant rotor bow across the rotor
Rotor Run-out Readings

- 12/15/15 Run-out vectors:
  - Bearing #1: 5.0 mils @ 180°
  - Mid-Rotor: 9.0 mils @ 176°
  - Bearing #2: 5.0 mils @ 202°

The rotor was clearly bowed at/near the center, with similar phase angles at all locations.

What’s are some possible solutions?

- Without considering the lost generation costs, the ideal engineering solution with the highest probability of success would involve:
  - Remove the rotor from the machine for straightening
  - Reduce the mechanical bow as far as possible to reduce centrifugal forces
  - OEM experts indicated generally good results straightening smaller rotors (< 6’), but had lower success rates with large turbine rotors
  - Perform shop-balancing using rotor center balance planes, reduce vibration as low as possible
  - Reinstall the rotor and run at full speed / load; trim balance as needed using rotor end-planes

- Alternative: attempt to balance-out the bow without removing the rotor.
  - Technically feasible if enough balance force can be created
  - Potential risks:
    - Because of the large balance weight that would likely be needed, it would be necessary to have the turbine open so that the inner balance planes could be used
    - If the 1st balance shot was inaccurate, it would be necessary to remove the turbine shell, again
    - Rotor bow eventually relieves itself during operation, possibly negating the balance work (monitoring 1X vectors should provide sufficient warning)

- 12/17/14:
  - Site decides to reassemble the unit & run at 1000 rpm to attempt to alleviate the rotor bow
  - Last-ditch attempt, low probability given the number of runs already performed
  - Sometimes, as a consultant, resistance is futile…
Santa Claus, where are you?

- 12/24/14: assembly completed & unit started:
  - Vibration was 3.8 mils at 1000 rpm, and 6.0 mils at 1220 rpm, same as before
  - No gifts for Christmas!
  - Decision was made to shut-down, remove the shell again, and proceed with balancing

- 12/30 – 12/31/14 – HP-IP turbine shell removal completed:
  - Mapping of existing balance weights in each HP-IP balance plane was done
  - Run out data showed a 0.015" bow at 157 degrees, with maximum deflection at rotor center

Happy New Year!

- 1/1/15:
  - After receiving the new rotor run-out data and HP-IP balance weight maps, the total unbalance force present in the rotor could be calculated.
  - Knowing that, we could theoretically calculate a balance weight adjustment to correct the vibration & help us ring in the New Year.
  - This would only work if we could actually install and/or relocate enough balance weights to produce a sufficiently large force to counter-act the rotor bow.

Unbalance calculation notes:

- Both the TIR Run-out data and the 1X vibration at a particular speed can be used to calculate the existing unbalance force in a rotor.
- TIR data is purely the “mechanical” state of the rotor.
- Vibration data reflects the dynamic condition of the rotor, and accurately takes into account the rotor system’s synchronous dynamic stiffness (mass, fluidic, and inertial stiffness terms).
- Comparison of these two CF values will provide a range of values to help guide in balance weight selection.

- So, if we are going to balance the rotor, we need a weight amount & angular location.
  - Vibration-based CF calculations will often provide a more accurate weight amount.
  - However, for a bowed rotor, the TIR data will provide the best phase angle value.
  - The balance weight would be installed opposite the TIR ‘high spot’.
As-Found Balance Weight Distribution

The majority of the existing weights were, ironically, at the rotor center, and nearly in-phase with the rotor bow. Should be an easy exercise to just relocate them, right?? Well, a couple days later, no problem…
Proposed Solution, 1/2/15

<table>
<thead>
<tr>
<th>Stage 1</th>
<th>Vector Change</th>
<th>Total Vector</th>
</tr>
</thead>
<tbody>
<tr>
<td>Remone</td>
<td>Install</td>
<td>Rad.</td>
</tr>
<tr>
<td>8.25</td>
<td>8.25</td>
<td>16.50</td>
</tr>
</tbody>
</table>

- Removing an existing weight and reinstalling it 180 degrees away essentially doubles the weight's effect. It's a vector thing…

Balance Weight Maps

Removing an existing weight and reinstalling it 180 degrees away essentially doubles the weight's effect. It's a vector thing…
**Results**

- **1/6/15**: Unit startup was successful!
  - Start-up: 6 mils through the 1st critical; acceptable starting point
  - On-line: Vibration started 6 mils at 3600 rpm / full-speed/no-load (marginal)
  - Unit was kept on-line to warm-up & thermally stabilize the turbine casing & piping
  - Vibration began trending down quickly, stabilizing at 4 mils at 75% load
  - Unit kept on-line for 1 week, no significant 1X vector amplitude or phase angle changes

- **1/13/15**: Shut-down & re-start (not vibration related)
  - 4 mils while on-line prior to shut-down, and again after start-up
  - HP-IP 1st critical vibration remained at 6 mils during start-up & shut-down
  - Rotor appeared to be stable

- **1/14/15 – 3/6/15**: 1X vibration was stable….
  - Bearing #1: 2.8 – 4.0 mils
  - Bearing #2: 3.6 – 4.5 mils
  - Bearing #3: 3.5 – 4.0 mils

- **3/17/15**: LP turbine trim balance – 220g couple-shot to reduce Bearing #2 vibration
  - Bearing #1: 2.0 – 2.4 mils
  - Bearing #2: 3.0 – 3.1 mils
  - Bearing #3: 0.7 – 1.5 mils
**Conclusion**

- **Primary risk at this time** – We continue to monitor the HP-IP turbine 1X vectors for any changes in amplitude or phase that might indicate the bow was changing.

- **Some lessons learned by the client:**
  - Operators need to understand the dynamics of rotor rub behavior in terms of what is happening with the overall vibration, and how they manage the situation to avoid damage.
  - Monitoring & understanding how eccentricity & slow-roll readings provide primary rotor bow indications, and that run-out small changes can produce high 1X vibration during start-up.
  - When a rub does occur, understand that vibration can change rapidly, and that without automatic high-vibration trips that operators may not be able to respond quickly enough to avoid machine damage.

- **Some key points as the consulting engineer on this project:**
  - A dedicated on-line vibration monitoring & data tracking system (in this case System1) captured all acquired data over several months of testing and allowed us to solve this problem remotely, and most of it while I was on vacation over the holidays, providing the client with project continuity through a difficult period.
  - Detailed inspections & record keeping, and a solid understanding of basic balancing principles allowed us to solve a challenging problem.
  - Good communication is critical – especially when working remotely - to ensure the correct data is gathered and all action items are fully understood.
Thank you!!
Any questions?

A copy of the presentation can be downloaded from: