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Sub-Synchronous Vibration on Centrifugal Compressor with Tilt-Pad Bearings

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Author Biographies

Dr. John J. Yu joined Bently Rotor Dynamics Research Corporation in 1998, followed by General Electric - Bently Nevada. Since 2002, he has been troubleshooting machinery vibration problems in the field directly, and is now Global Technical Leader for Machinery Diagnostic Services. He holds a PhD in Mechanical Engineering from University of Alberta, and is a fellow of ASME.

Nicholas Hanson joined Flint Hills Resources as a Machinery Reliability Engineer in 2013. He has been working with rotating machinery in the Chemical and Oil & Gas industry for 10 years. He holds a degree in Mechanical Engineering from the University of Minnesota.

Abstract

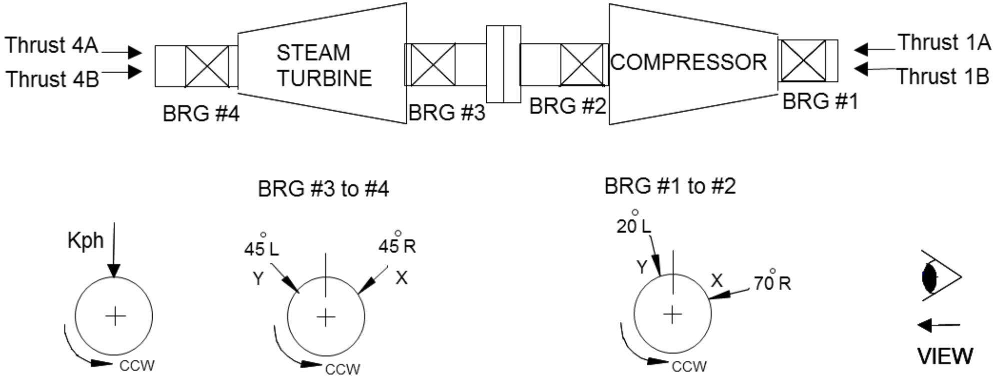
This presentation provides a success story that sub-synchronous vibration was eliminated on a centrifugal compressor. The sub-synchronous vibration originally occurred at high operational speed on this 5-stage hydrogen recycle centrifugal compressor supported by tilt-pad bearings. Vibration data is reviewed to find its root-cause. Each possible malfunction is discussed to see if it was likely the root-cause. It is shown that sub-synchronous vibration not caused by surge or stall could still happen even with tilt-pad bearings. A solution was implemented to successfully resolve the issue.

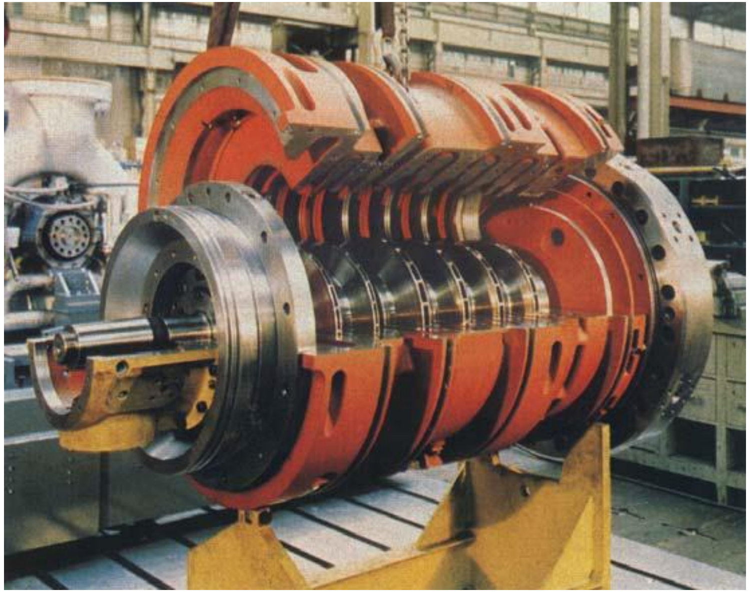
Outline

1. Introduction
2. Problem Statement
3. Vibration Data Review
4. Conclusion and Recommendation
5. Final Results
6. Lessons Learned
7. Most Recent Update

1. Introduction

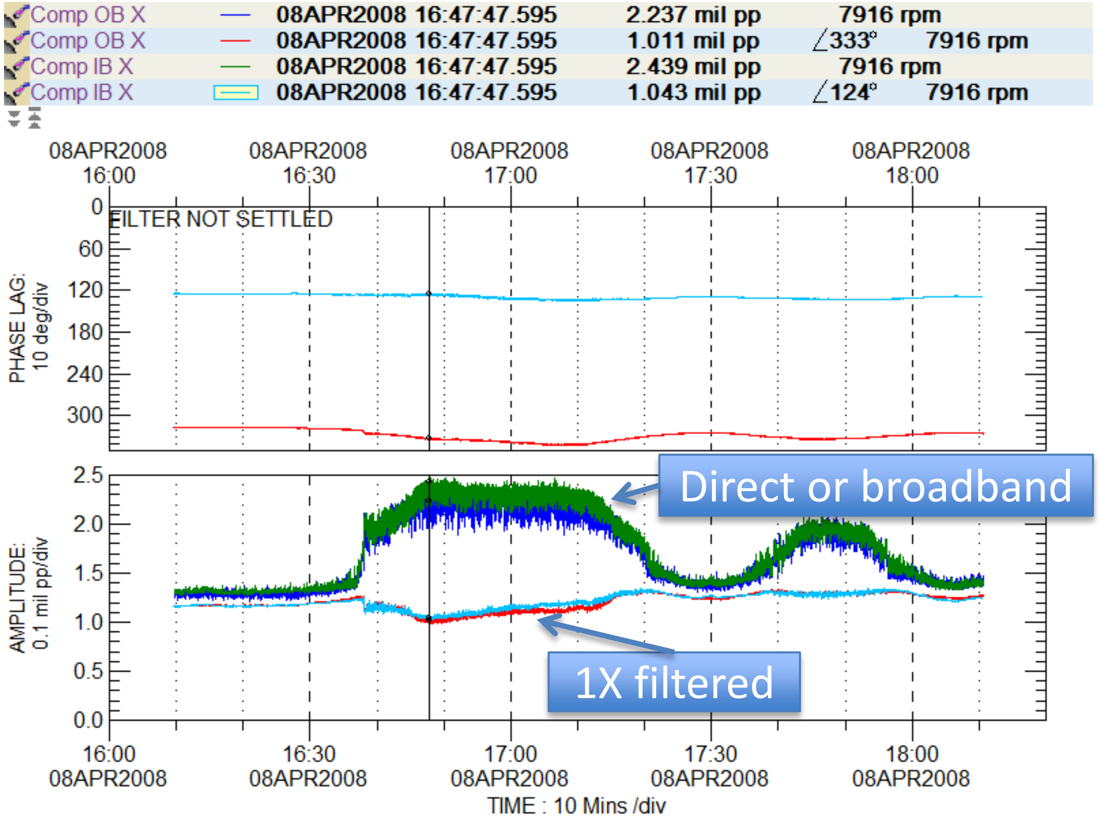
- 5-stage hydrogen recycle centrifugal compressor
- Supported by 5-pad tilt bearings with LBP arrangement
- Maximum speed ~8600 rpm
- Vibration monitored by X-Y pairs of proximity probes at all bearings, with site convention of looking from outboard to inboard of the compressor.





2. Problem Statement

- The compressor experienced high vibration of ~ 2.5 mil pp with **Sub-Synchronous Vibration (SSV)** amplitude of ~ 1.5 mil pp at frequency of ~ 3000 cpm, at both inboard and outboard bearings.
- Very trivial level of **SSV** from steam turbine bearings.

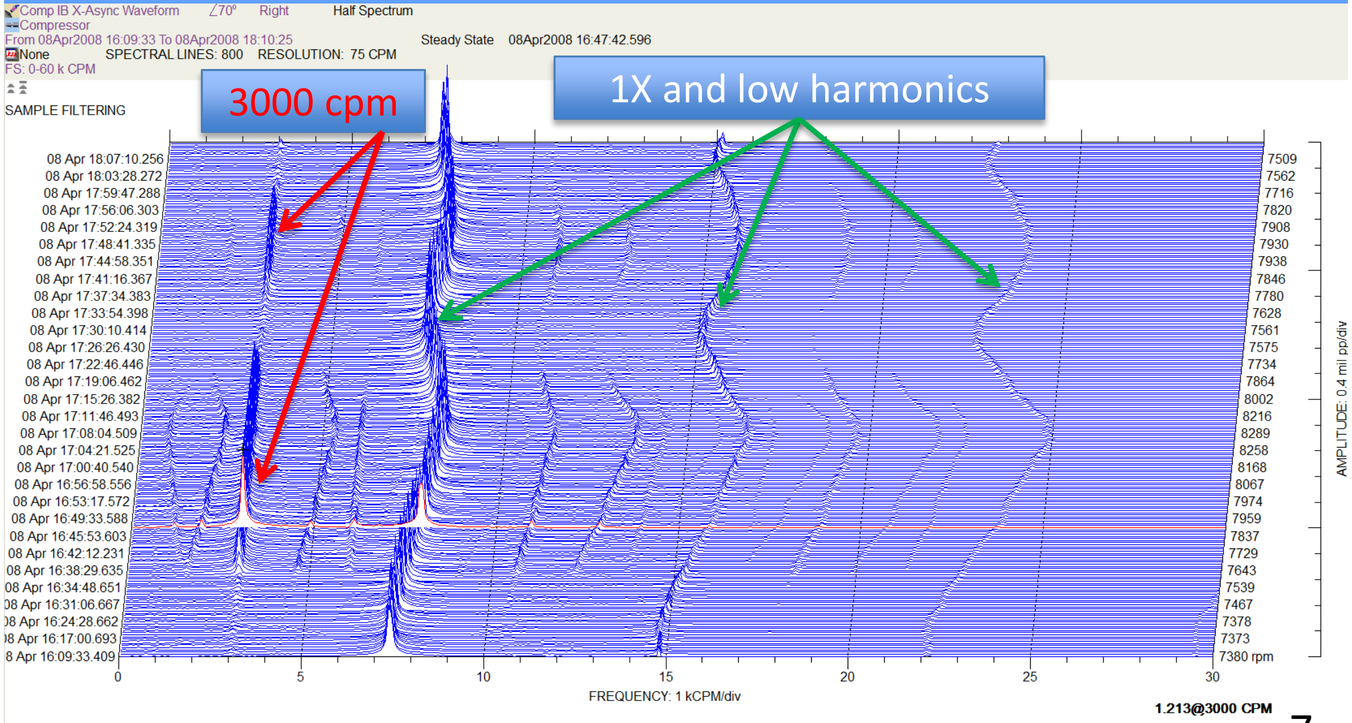


3.1 Vibration Data Review – Waterfall plot

Vibration components:

- 1X and harmonics – normal
- SSV at 3000 cpm – abnormal

from Compressor Inboard Bearing X-probe

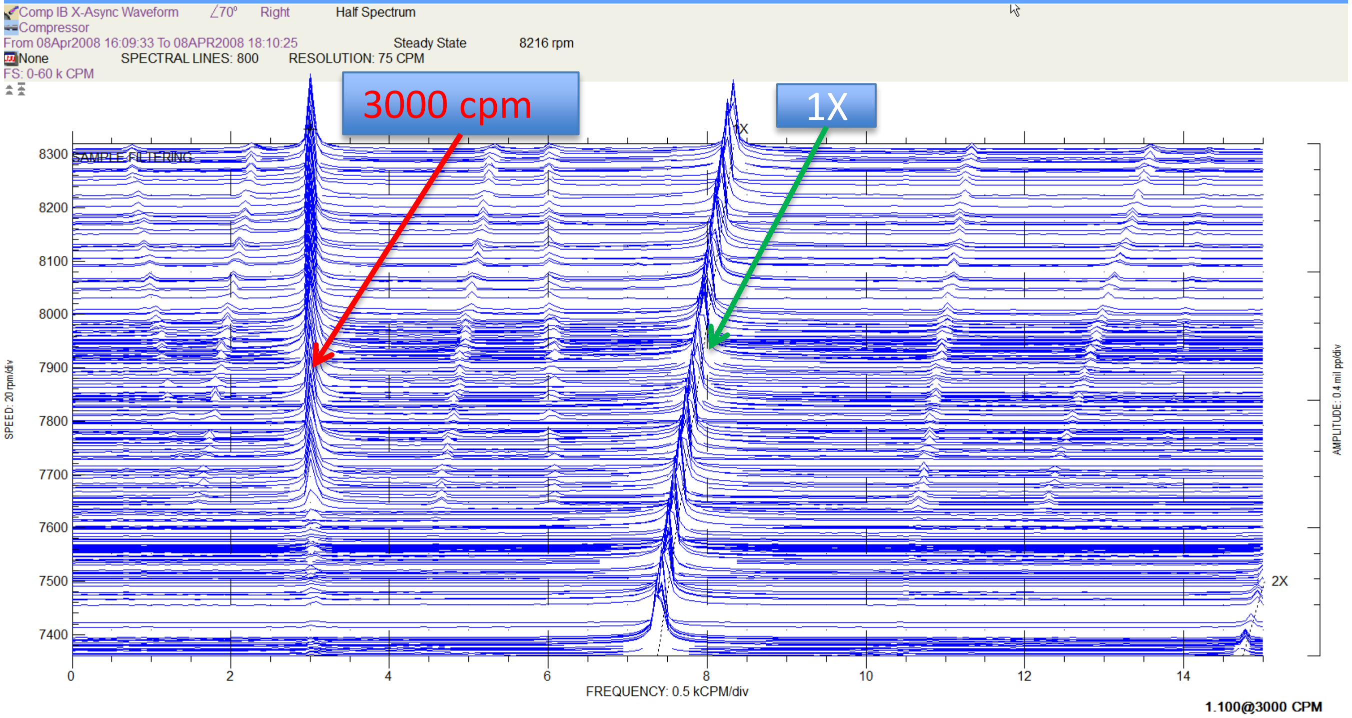


3.2 Vibration Data Review – Cascade plot

Characteristics of SSV:

- Not tracking with speed.
- Almost constant frequency of ~ 3000 cpm, likely the lowest natural frequency.

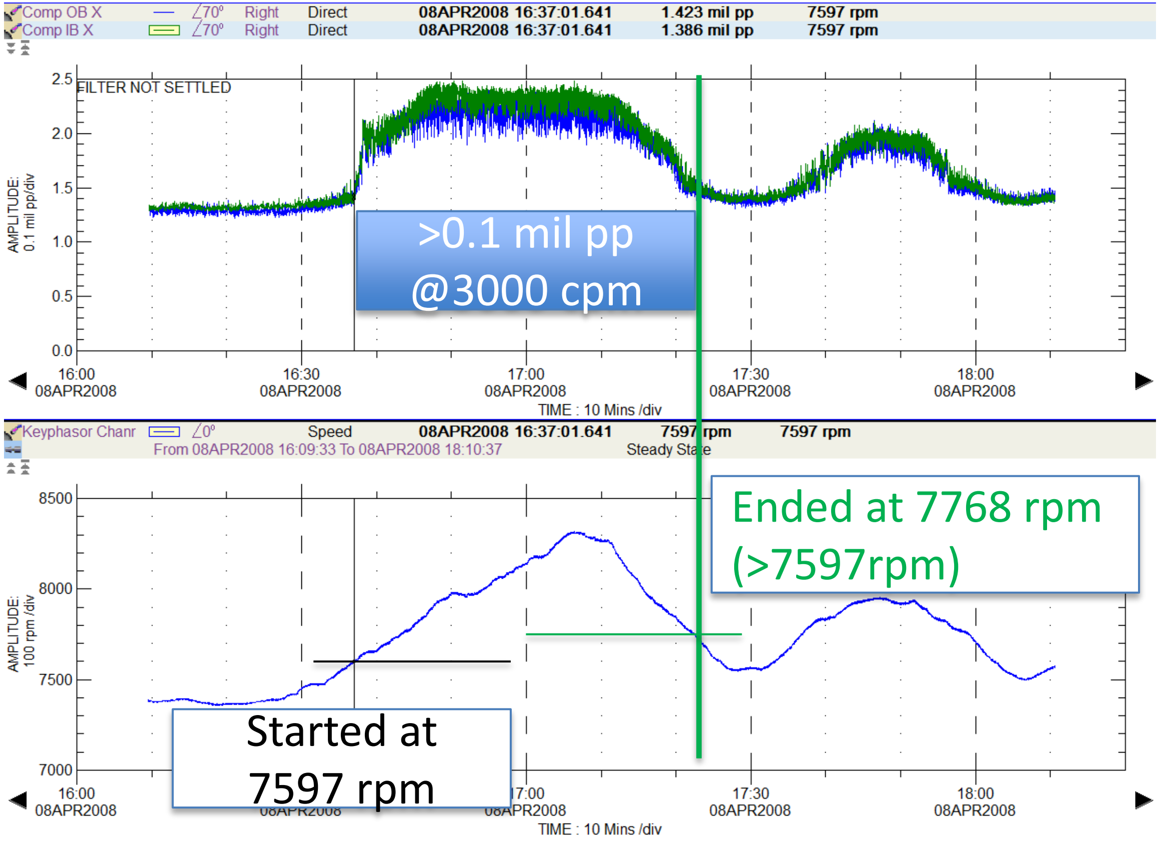
from Compressor Inboard Bearing X-probe



3.3 Vibration Data Review – Operation condition

When did SSV occur?

- At high speed that is also related to high horsepower.
- SSV was able to disappear at speed even higher than the onset speed.
- Molecular weight decreased after SSV lasted for a while but before reducing speed.

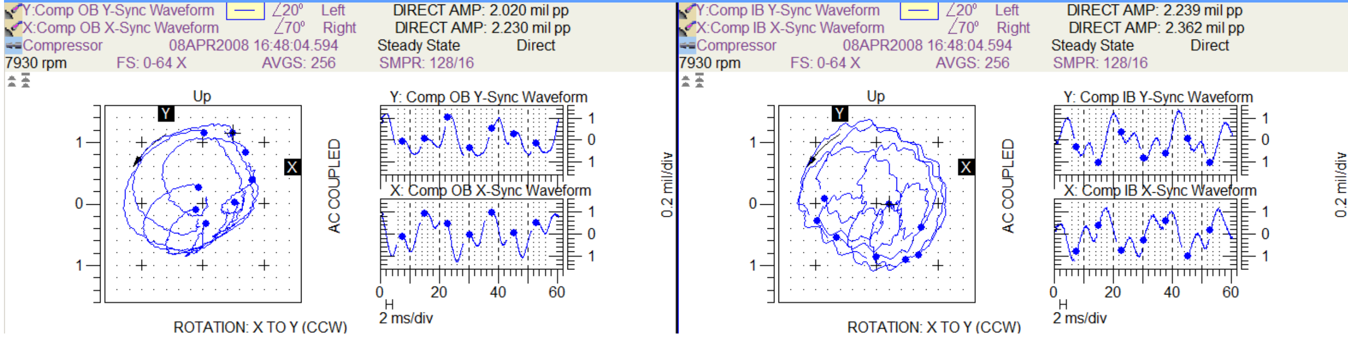


3.4 Vibration Data Review – Orbit and spectrum plots

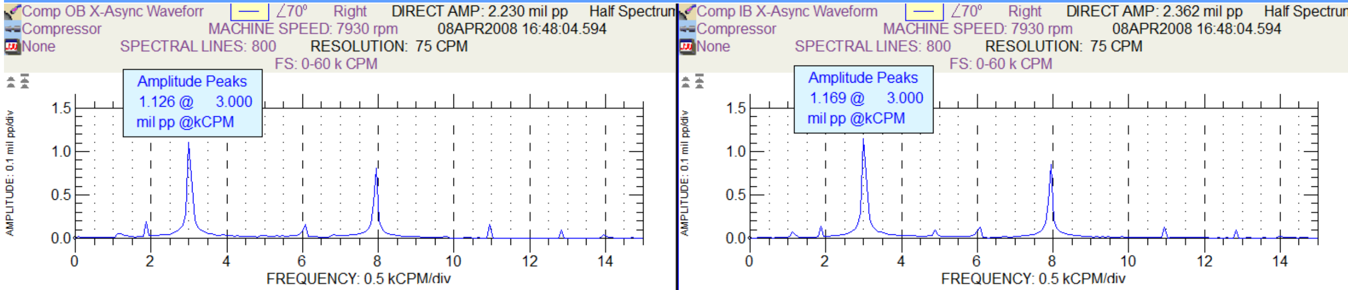
Shaft vibration:

- Varying orbit shapes due to combined SSV and 1X, etc.
- SSV level even higher than 1X as shown in spectrum plots

Orbits at OB & IB Bearings

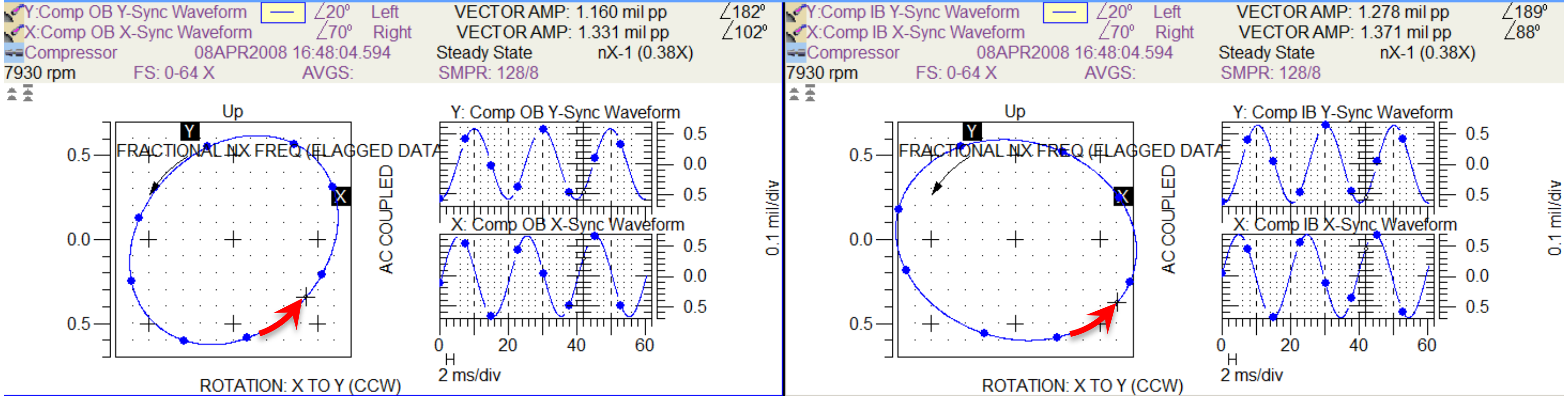


Spectrum plots from X-probes at OB & IB Bearings



3.5 Vibration Data Review – SSV orbits

SSV (~ 0.38X filtered at 7930 rpm) orbits at OB & IB Bearings



SSV properties:

- Fairly circular orbits with forward precession.
- In-phase between IB and OB, indicative of the self-excited vibration with the 1st mode.

4.1 Conclusion and Recommendation

Root-causes:

- Oil whip? – unlikely
  - ➢ Tilt-pad bearings with narrow oil rings of 0.017” clearance.
  - ➢ No “hysteresis” effect was observed on oil bearings as SSV ending speed was even higher than its beginning speed.
- Surge ? – definitely not now
  - ➢ Surge occurred a few months ago that might affect aerodynamic stability, but no sign of surge right now.
- Misalignment ? – less likely
  - ➢ Though shaft centerline plots are not available due to lack of startup/shutdown data, orbits look normal.

4.2 Conclusion and Recommendation

Root-causes (cont.):

- Rub? – not now
  - Rub at natural frequency of 3000 cpm would be reverse processional (dry whip), but the current one is forward.
  - Slight changes in 1X vibration were due to changes in speed.
  - Bouncing type of rub would be at fractional frequency tracking with speed. Several loops in orbit were due to combined different frequencies, not contact.
- Aerodynamic instability from seals – possible
  - Current abradable seal clearances of 0.026-0.030”, doubling the previous spec 0.013-0.015” diametral clearance.

4.3 Conclusion and Recommendation

Root-causes (cont.):

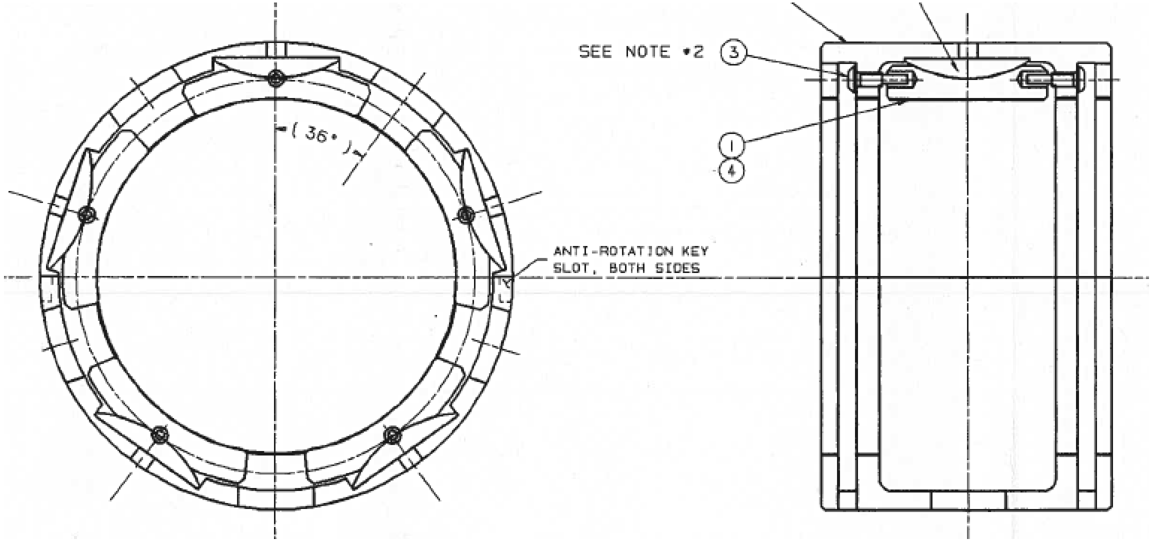
- Aerodynamic instability from compressor impellers – possible
  - Destabilizing impeller forces exist, similar to turbines and pumps (Thomas-Alford forces), though less.
  - It was reported that no SSA occurred at 8000 rpm with molecular weight of 2.
- Higher bearing clearance – possible
  - Previous bearings (0.004” diametral clearance) replaced with the current ones (L/D=1.6”/3.5”, 0.0055”clearance) with self-aligning spherical contact surface, probably reducing their damping to cope with destabilizing forces from seals and impellers.

4.4 Conclusion and Recommendation

Therefore, the source of the instability might have come from the seals and impellers, not from the tilt-pad bearings. However, the current higher bearing clearance might have reduced stability margin of the whole compressor-bearing system. A recommendation was made to reduce the bearing clearance.

4.5 Conclusion and Recommendation

Changes made: Bearings were replaced with tighter clearance (reduced from 0.0055” to 0.0035” diametral, bearing preload increased from 0.27 to 0.53).

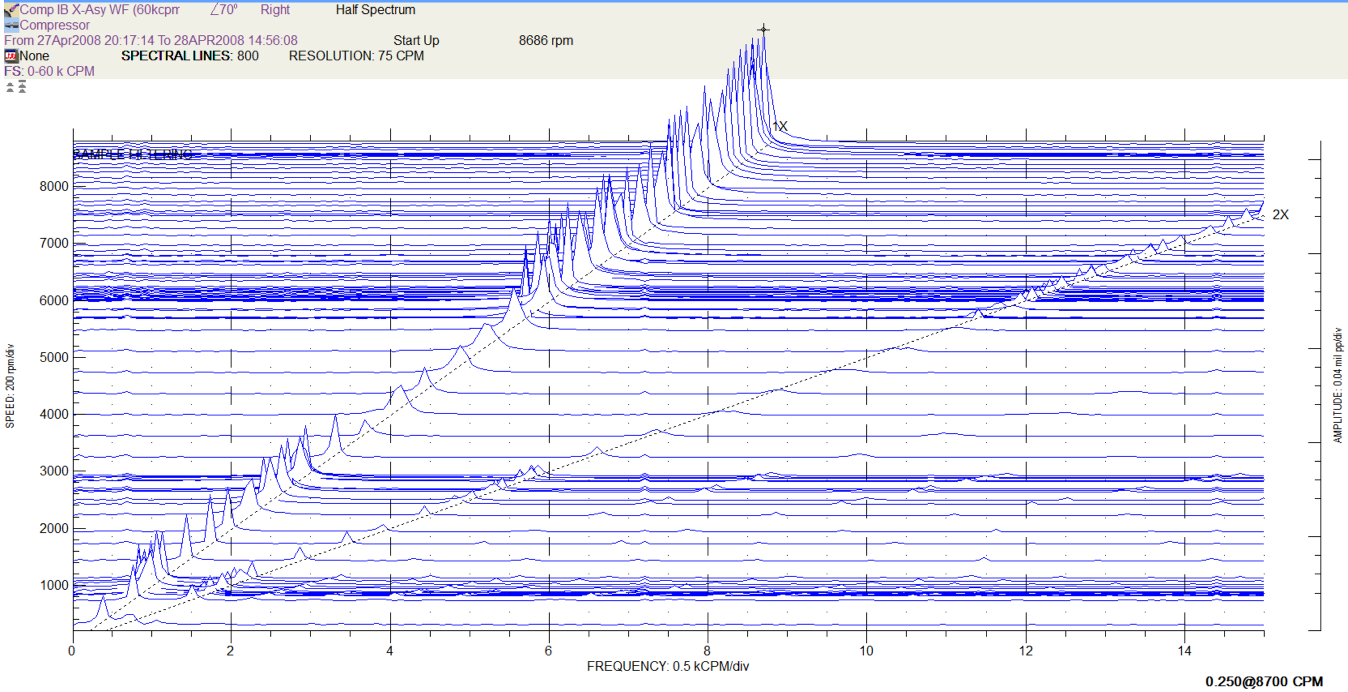


5.1 Final Vibration Results

After replacing bearings

SSV virtually suppressed to zero at all operational conditions

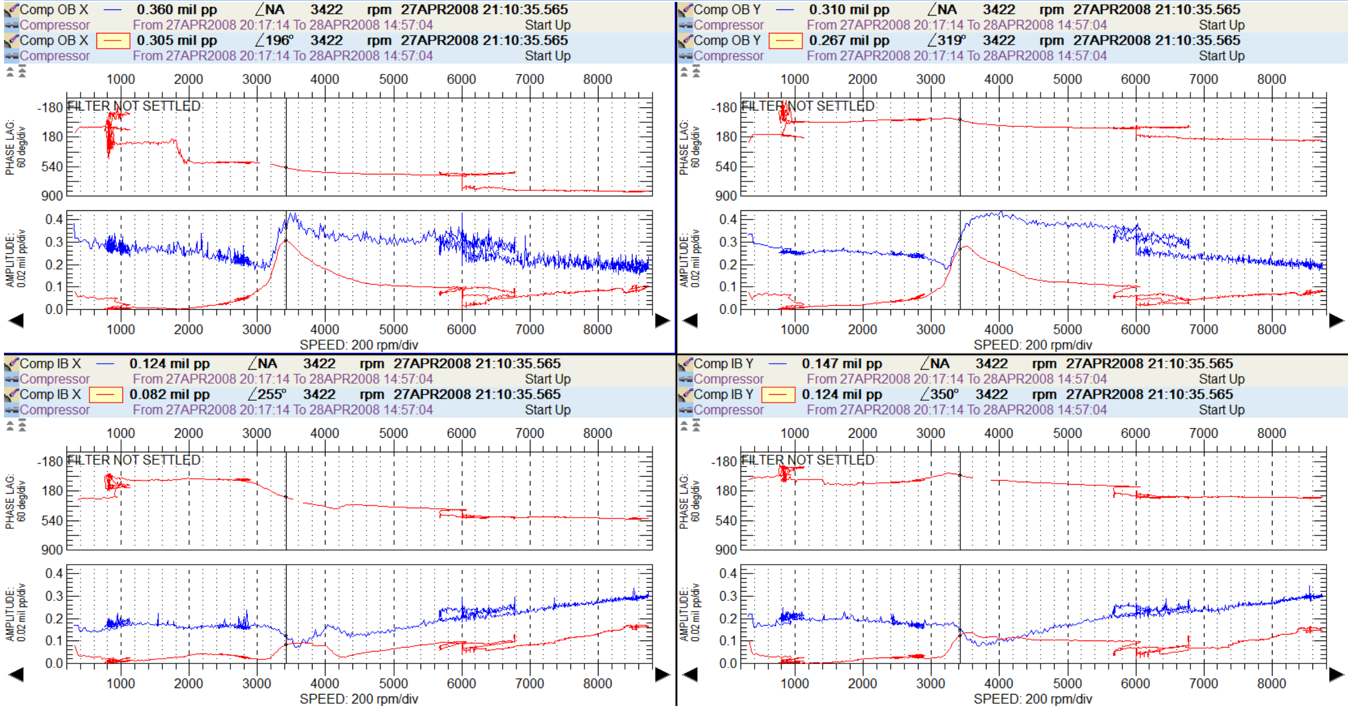
Cascade plot from Compressor IB X-probe



5.2 Final Vibration Results

The 1st resonance increased from ~ 3000 rpm to ~ 3400 rpm

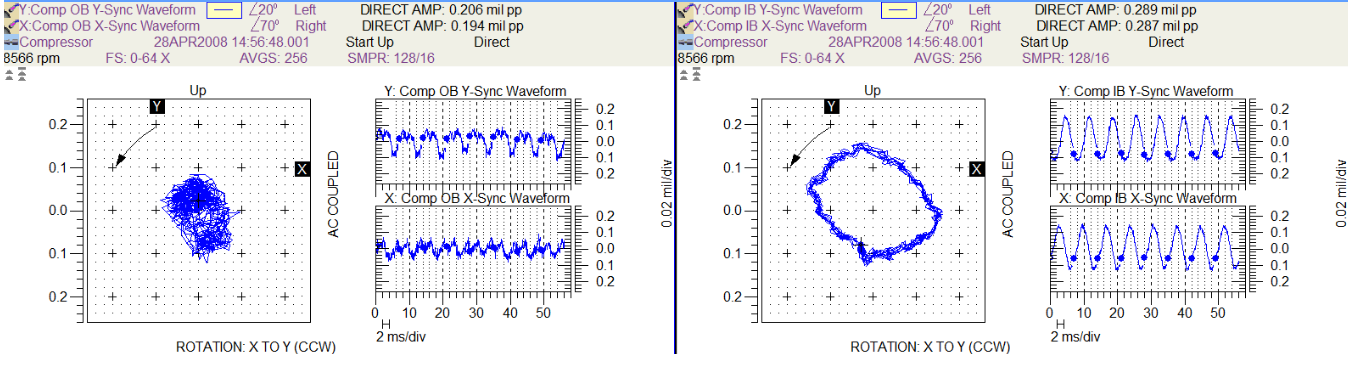
Bode plots from compressor bearings



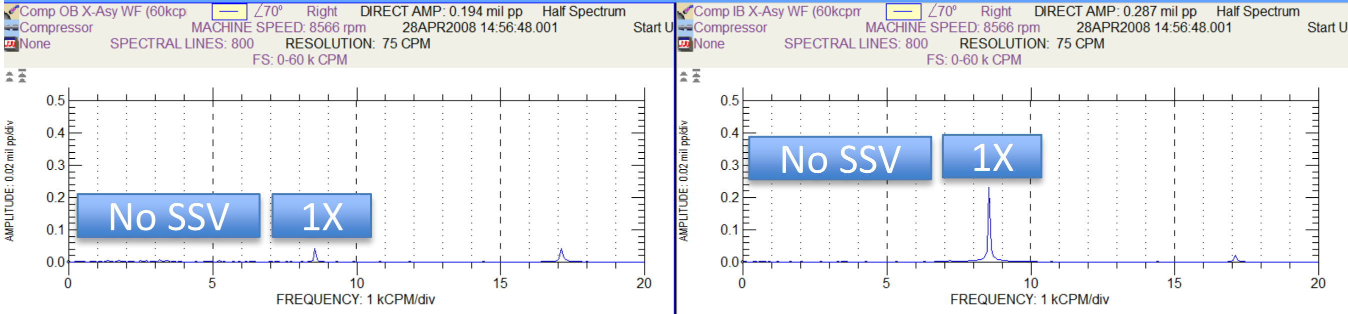
5.3 Final Vibration Results

At 8566 rpm, overall vibration amplitude became < 0.3 mil pp at both IB and OB bearings, as the unit was operating near the anti-resonance point.

Orbits at OB & IB Bearings

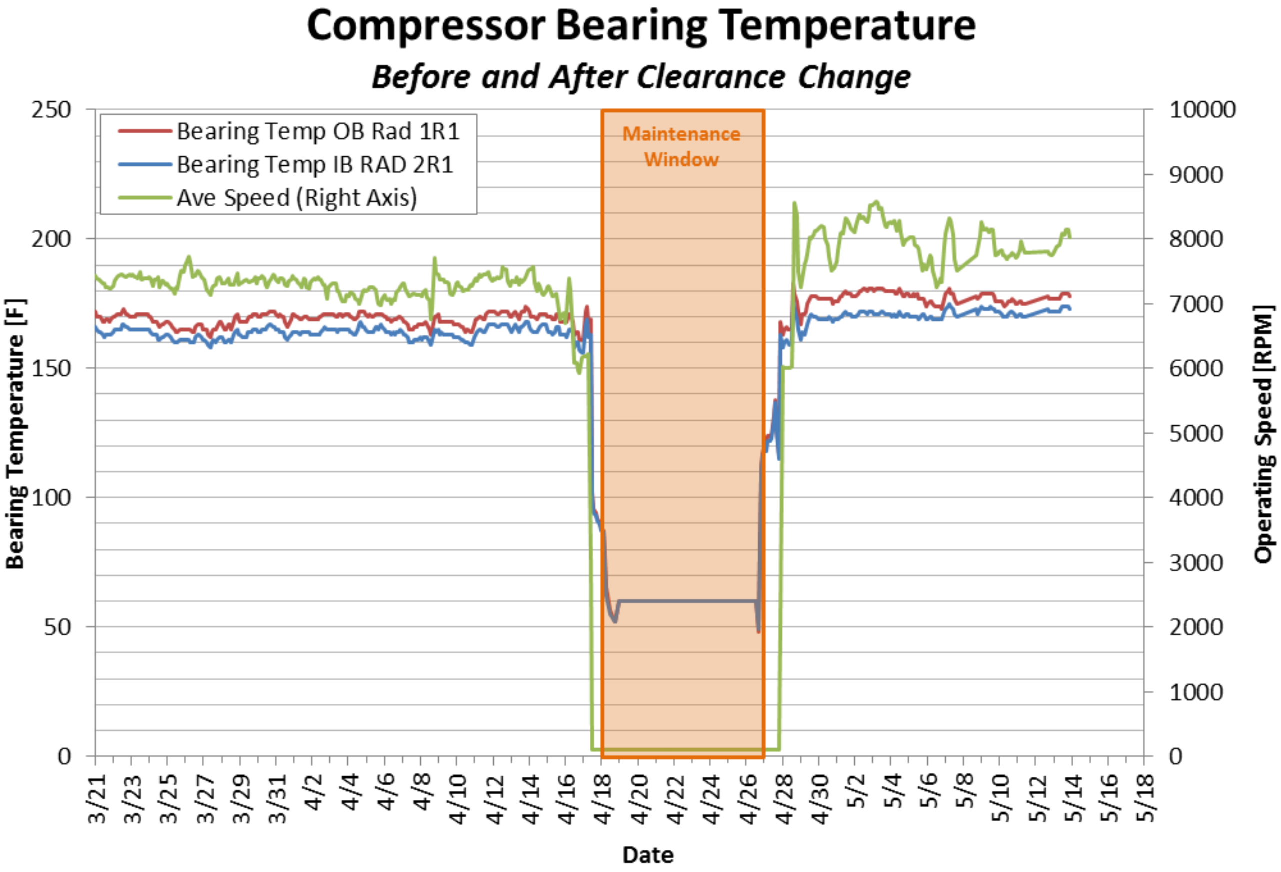


Spectrum plots from X-probes at OB & IB Bearings



5.4 Final Bearing Metal Temperatures

Bearing metal temperatures increased slightly (<10°F) due to reduction in bearing clearance, but not exceeding 180°F (alert limit being 240°F).



6. Lessons Learned

- Sub-synchronous vibration not caused by surge or stall could still happen even with tilt-pad bearings, because centrifugal compressor destabilizing forces always exist, such as those from seals and impellers to possibly make the unit rotordynamically unstable.
- Sometimes it is impossible to eliminate the sources of instability. However, since the stability margin is determined by overall rotor-bearing system properties, modification of other parts in the field may resolve the issue.
- In this case, it seems that the overall stability margin could be enhanced by decreasing bearing clearances. The effective damping might still be increased by raising both stiffness and damping of the bearings.

7. Most Recent Update

- No sub-synchronous vibration has occurred since the changes in bearings in 2008.
- In 2012, shaft was stiffened, which raised the 1st critical speed to about 3700 rpm.