ASIA TURBOMACHINERY & PUMP SYMPOSIUM
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Technical Brief:
Natural Frequency Whirl Instability in an Industrial Gas Turbine

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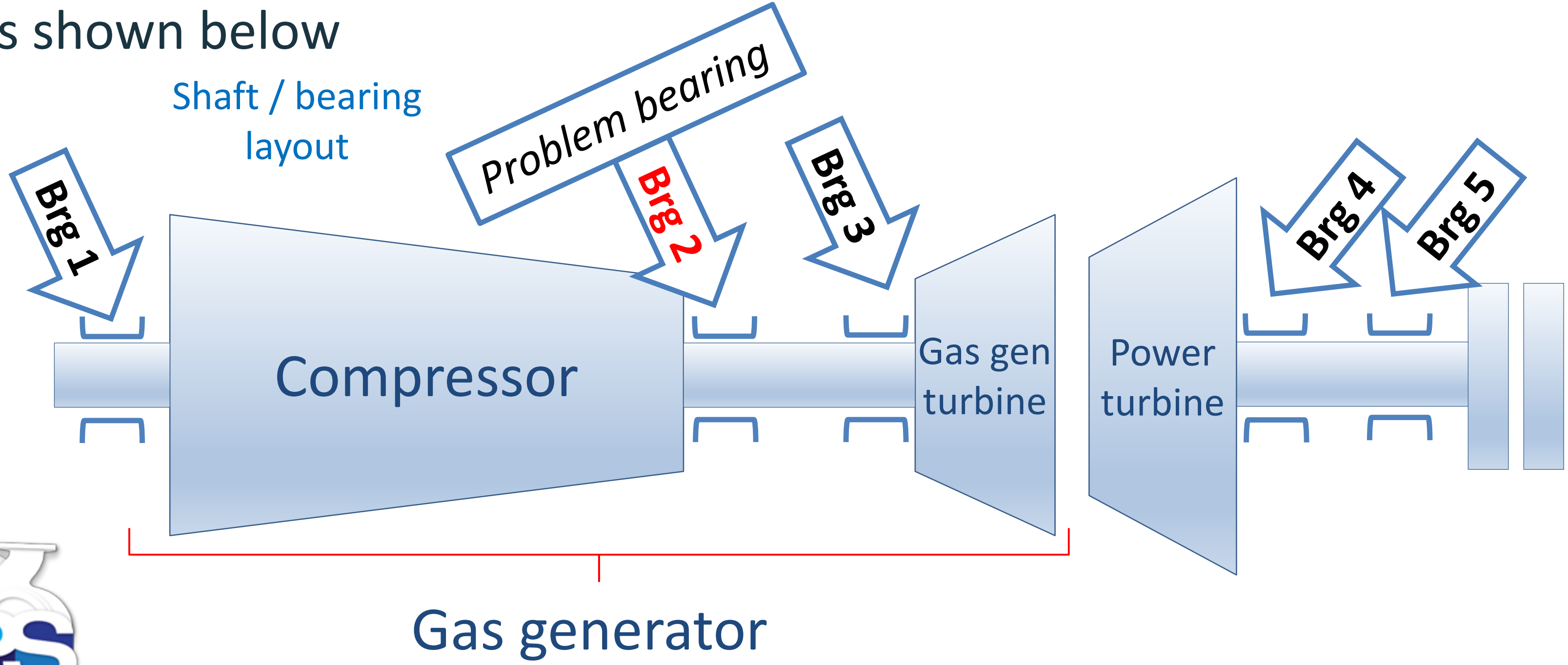
The machine

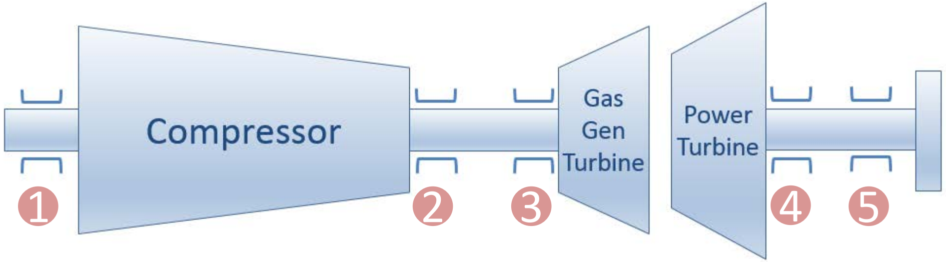
- Gas turbine driven compressor set ‘A’ (1 of 3)
- Engine is a two-shaft 70 MW industrial gas turbine, bearing layout as shown below



The issue

- Bearing 1 failed after 16,000 hours operation
- Vendor replaced Bearing 1 in situ
- Vibration returned to normal, operation continued, but:
- At 17,500 hrs, vibration at ***Bearing 2*** stepped up to 50µm / 2 mils pp, and continued to trend up
- Vendor recommended to remove engine from service for off-site overhaul, operator requested a second opinion

Shaft / bearing arrg't.

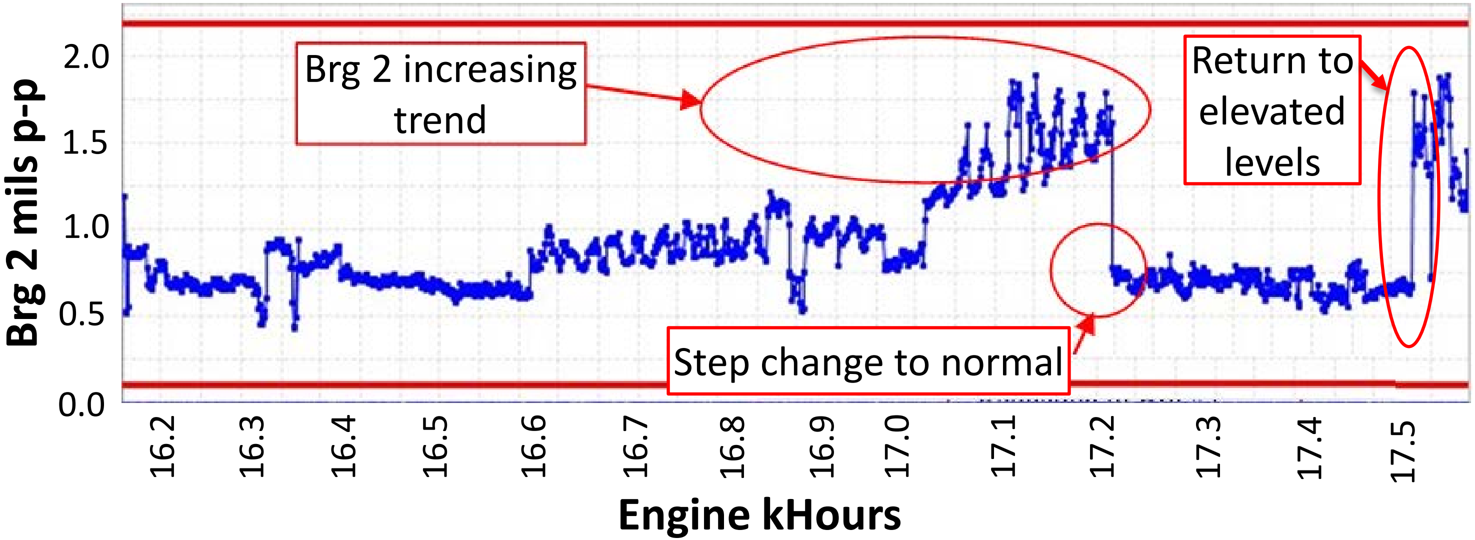


Problem narrative

Shortly after Bearing 1 replacement, ***Bearing 2*** exhibited increasing vibration amplitude trend (from 16,600 – 17,200 hours), then:

- An unexplained step change back to normal without intervention
- At around 17,500 hrs, high vibration returned at Bearing 2

Human-machine interface display – Bearing 2 vibration amplitude trend



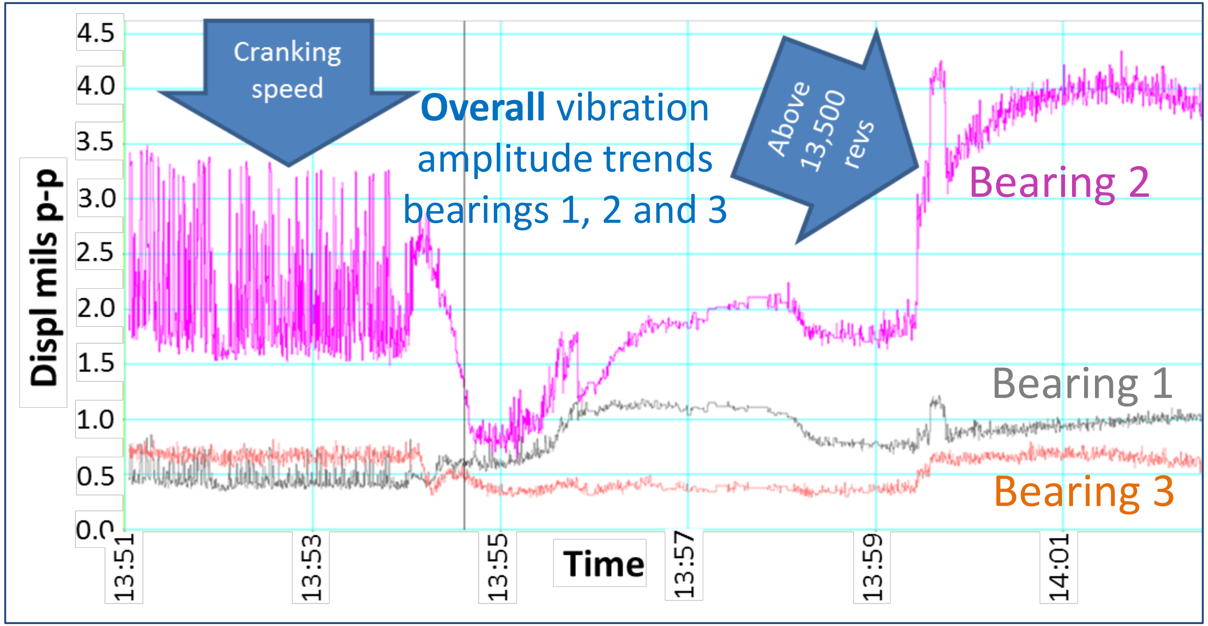
Data acquisition

- Human-machine interface only records *overall* vibration levels
- Detailed vibration data from protection panel is not archived
- Vendor attended site and acquired run-up data from gas generator shaft cranking speed of 3,300 RPM (*not* 0 RPM), using a multichannel dynamic signal analyser
- Vendor analysis report recommended withdrawal from service; data was later passed on to Wood's vibration, dynamics and noise team for further analysis

Multichannel dynamic signal analyser

Data review – start-up amplitude trends

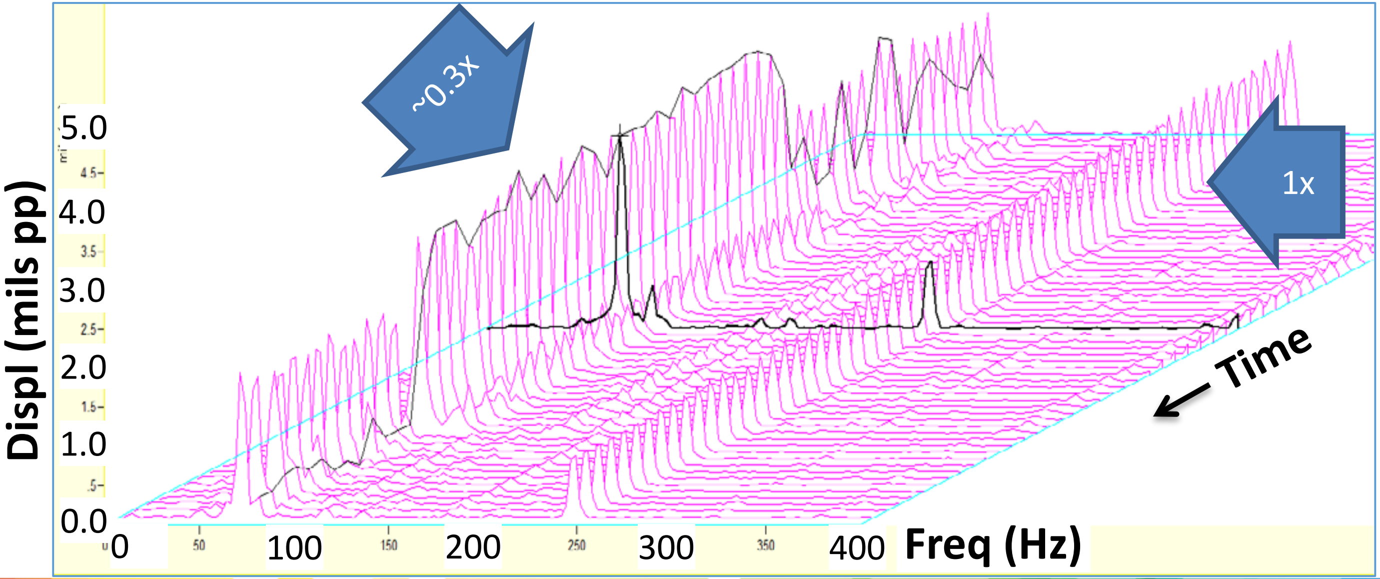
- At gas generator cranking speed of 3,300 RPM, large amplitude fluctuations observed at Bearing 2 to 90 µm / 3.5 mils pp
- Settles down as speed ramps up, however:
- At 13,500 RPM, a step increase to ~100 µm / 4 mils pp at Brg 2 only
- No issues were noted on the power turbine



Data review – waterfall amplitudes (Brg 2)

- High vibration amplitudes appear at around 13,500 RPM, reaching their worst at 14,200 RPM
- Main frequency component was at ~0.27 – 0.3x turning speed (65.0 -67.5 Hz), at over 75 µm / 3 mils p

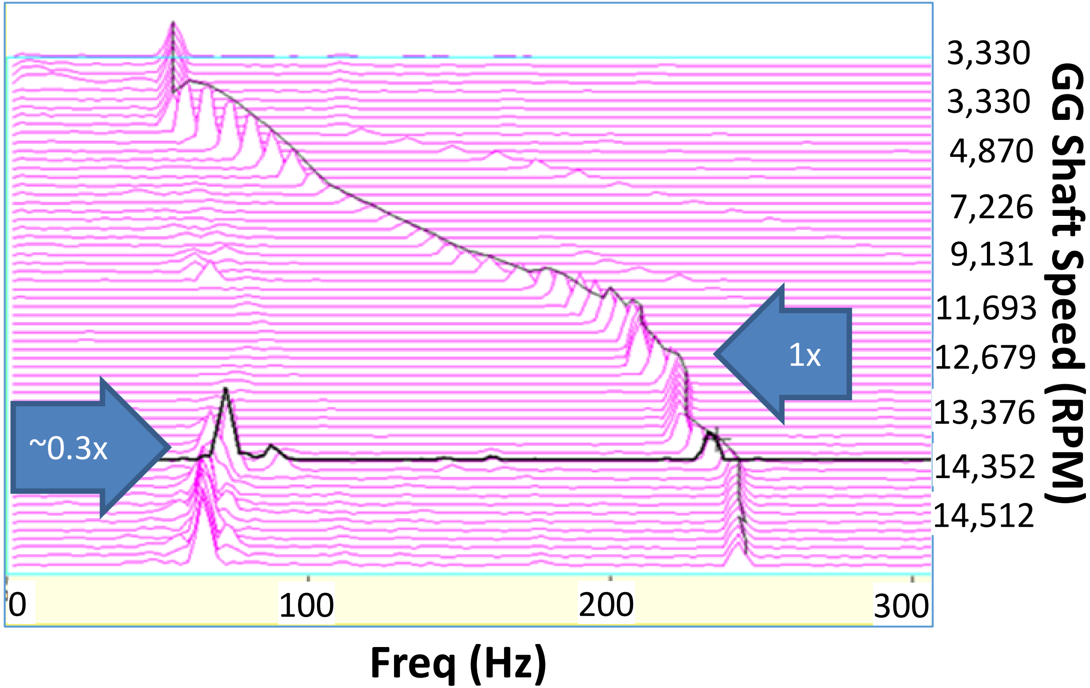
Bearing 2 start-up waterfall at onset



Data review – start-up waterfall plot (Brg 2)

- Waterfall plot (read from rear to front) shows the high amplitude whilst cranking is at 1x, also suggests critical circa 4000 RPM
- At 13,500 RPM, we see onset of a large sub-synchronous frequency component – initially, around 0.3 orders of turning speed, but drops down a little

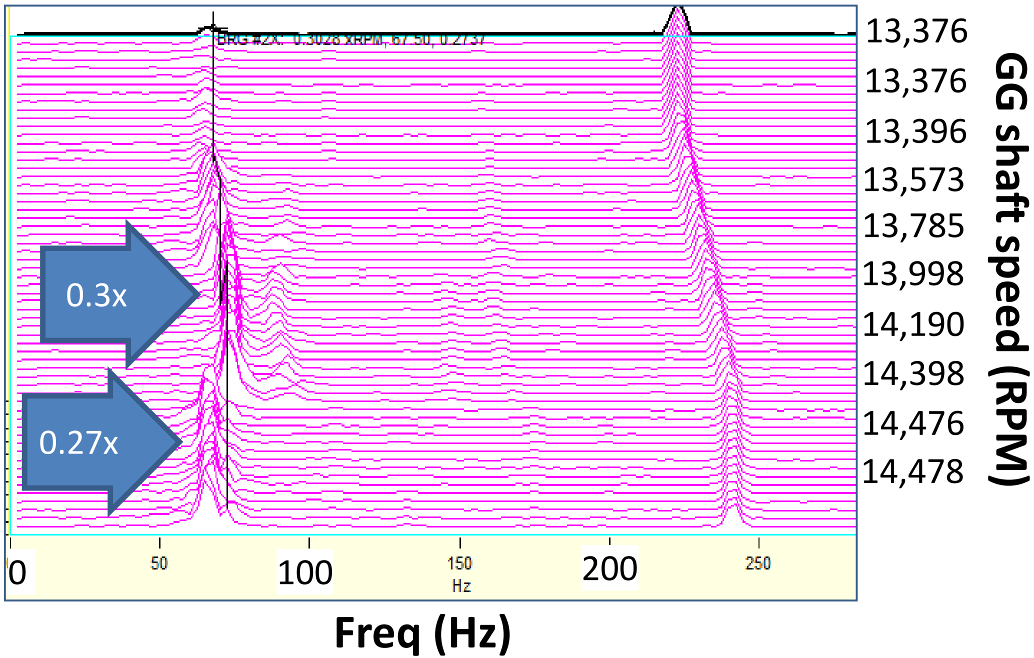
Bearing 2 start-up waterfall



Data review – start-up waterfall plot (Brg 2)

- Closer examination of the onset of this sub-synchronous frequency component indicates it tracks with running speed during the final speed increase (~0.3x)

- But then ‘locks on’ to a frequency of 65Hz, around 0.27 orders of turning speed

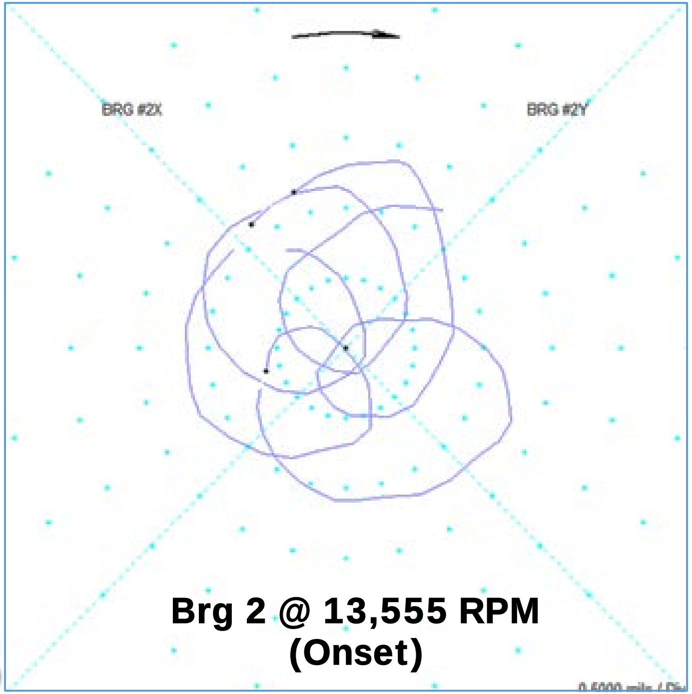


Zoom-in on bearing 2 start-up waterfall

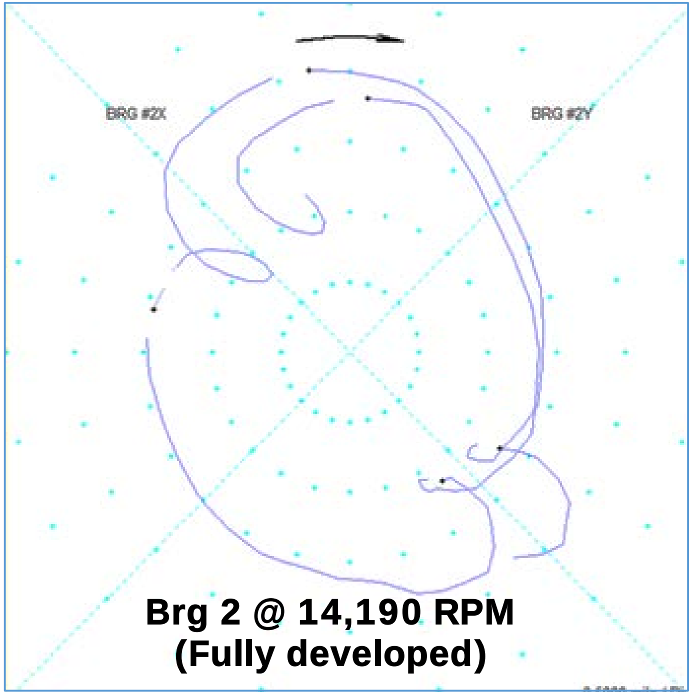
Data review - Brg 2 orbits at normal op speeds

- Two inner loops (forward precession), 3 phase markers per cycle
- Not a closed cycle because not an integer fraction

Bearing 2 orbit plot at instability onset



Brg 2 @ 13,555 RPM
(Onset)

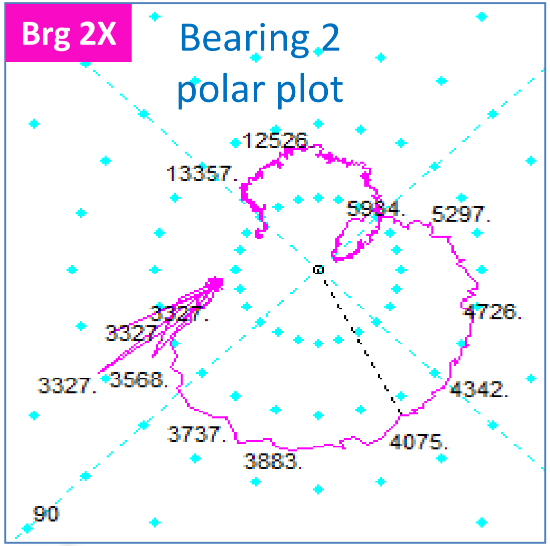


Brg 2 @ 14,190 RPM
(Fully developed)

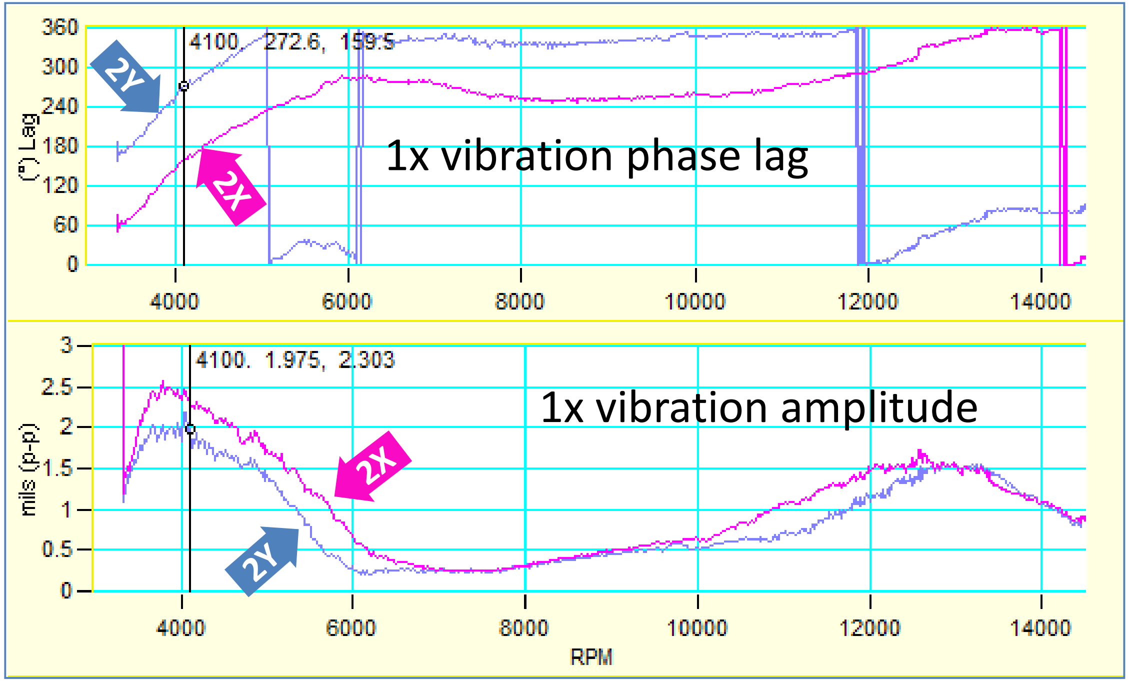
Bearing 2 orbit plot, instability fully developed

Data review – polar and bode plots (1x amp and phase)

Indicate critical at around 4,100 RPM / = 68 Hz, good match for the 0.3 orders vibration which appears at 13,500 RPM

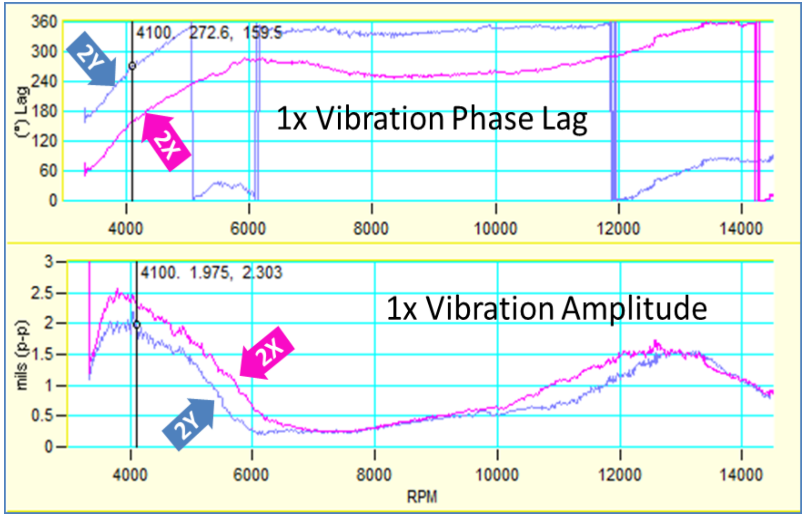


Bearing 2
bode plots

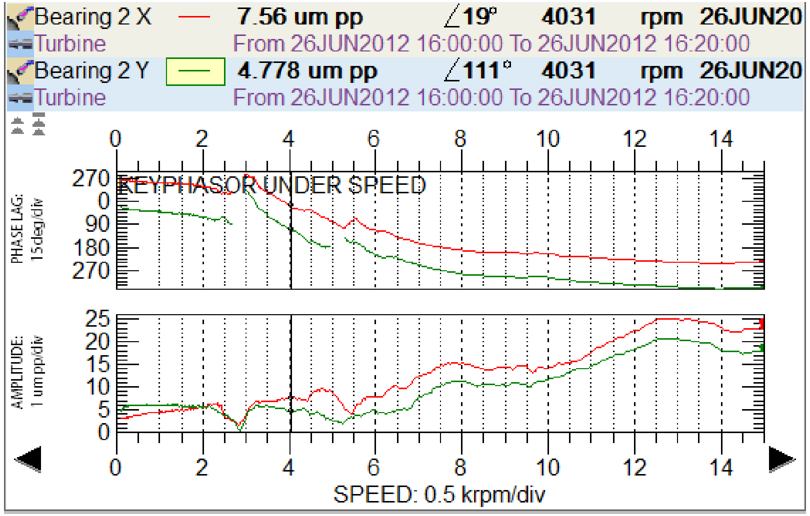


Data review – bode plot comparison

Problem machine Bearing 2 bode plots



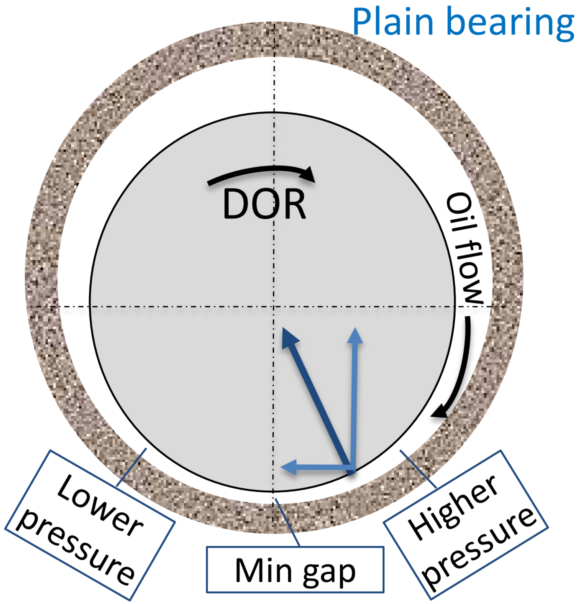
Healthy machine Bearing 2 bode plots

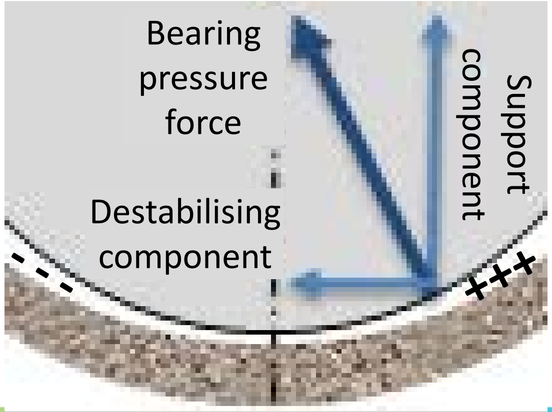


When compared with a healthy example of the same model, 1st critical (as determined by phase change) similar but amplification factor appears much higher

Discussion

- Symptoms bear similarities to oil whirl / oil whip, as presented in many vibration texts
- Oil whirl arises in plain journal bearings[1] because fluid pressures are asymmetric either side of the minimum gap – higher on the upstream side
- Resultant bearing force has tangential, destabilising component, normal to the bearing support force
- Usually, shaft just runs a little to one side of the shaft centreline in the DOR, but …
- It is this cross-coupling force which, if it overcomes loading and damping forces, results in oil whirl

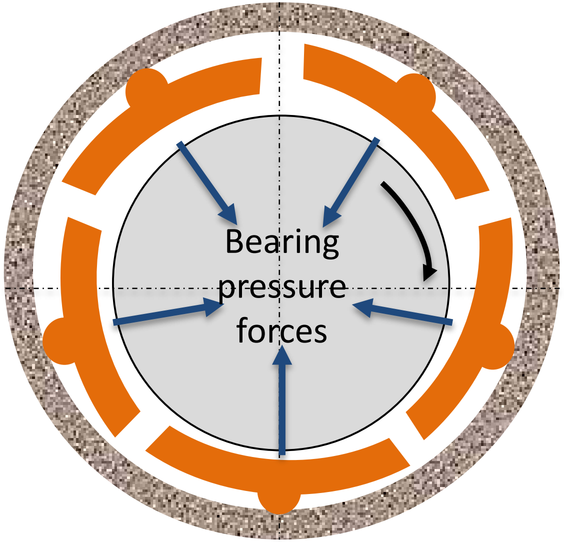




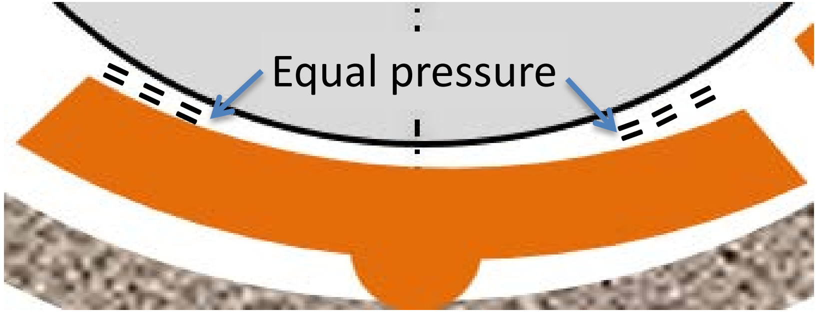
[1]and other fixed geometry bearing designs

Discussion

- But the machine in question uses tilt pad journal bearings (TPJBs), with 5 pads, in a load-on-pad (LOP) configuration
- As each pad can pivot, it constantly adjusts angle to maintain equilibrium pressure across its face; resultant force always points at shaft centre
- Thus each pad tilts such that the leading edge is a little further from the journal than the trailing edge – as we typically see in the wear pattern
  - This bearing type ***cannot*** produce cross coupling forces – but there *are* other contenders…



Tilt pad journal bearing



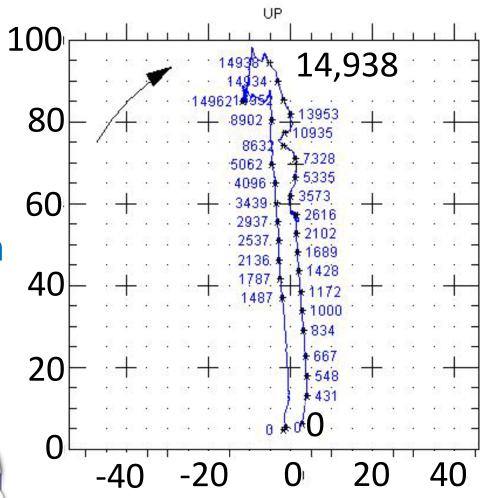
Diagnosis

- Forward acting cross coupled forces are also produced by aerodynamic effects, and by fluid rotation at labyrinth seals. Normally, stiffness, loading and damping is sufficient to suppress these and maintain stability.
- Whilst it cannot be discounted that one or more such forces had increased here, experience and recent maintenance history suggest a simpler explanation
- Vendor have advised that such symptoms are commonly seen when bearing clearances have increased significantly due to wiping or other damage (e.g., fulcrum wear)
- This results in increased lube end leakage from the bearing, significantly reducing damping

Discussion

- Unfortunately, start-up data provided was not captured from 0 rpm, so we were unable to obtain meaningful Shaft Centre Line plots.
- Such plots enable us to determine shaft running position within the bearing, and compare with clearance to determine eccentricity ε:

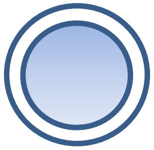
Example centre line plot (not from this case)



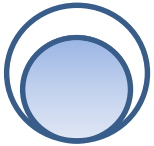
$$\varepsilon = d / c$$

d = shaft centre to bearing centre

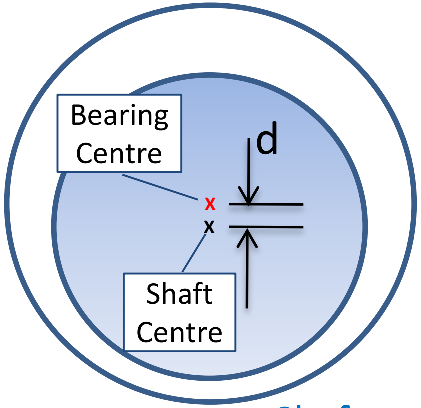
c = bearing radial clearance



ε →0



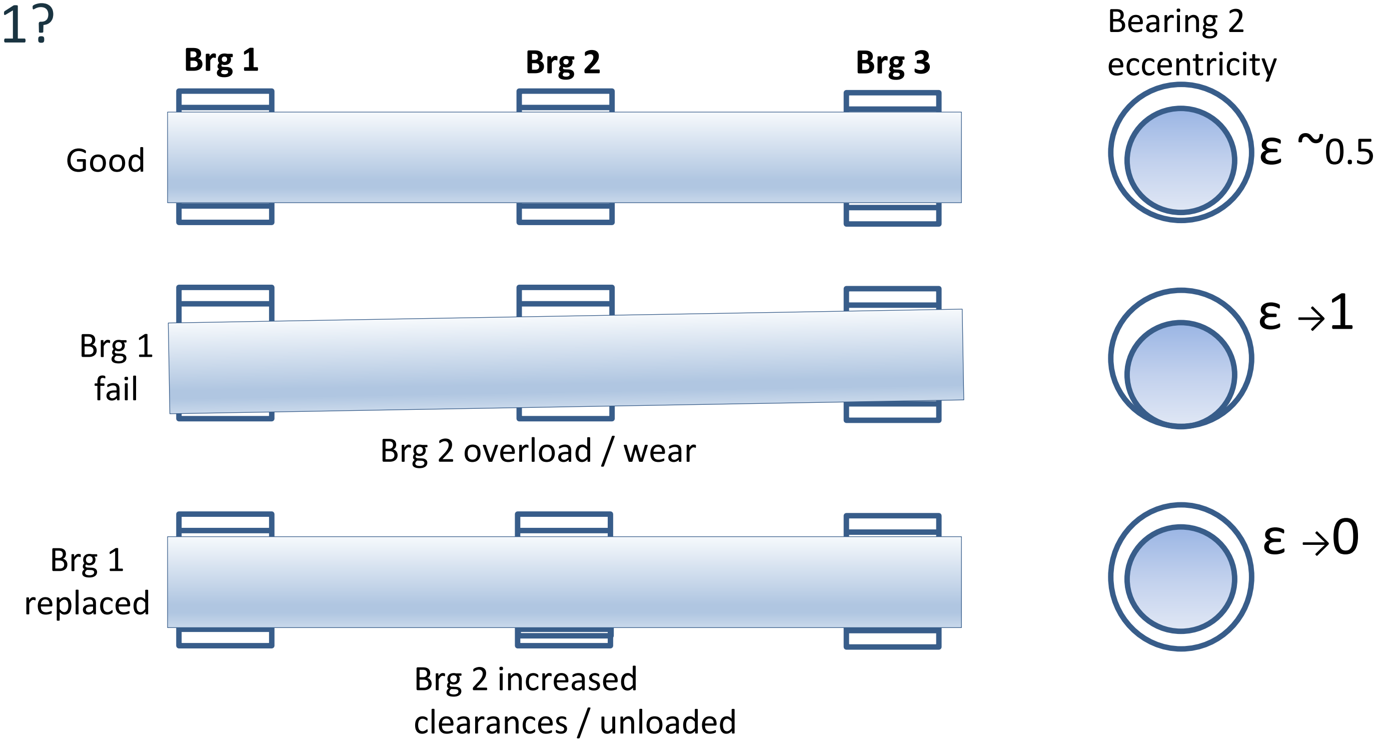
ε →1



Shaft eccentricity

Hypothesis

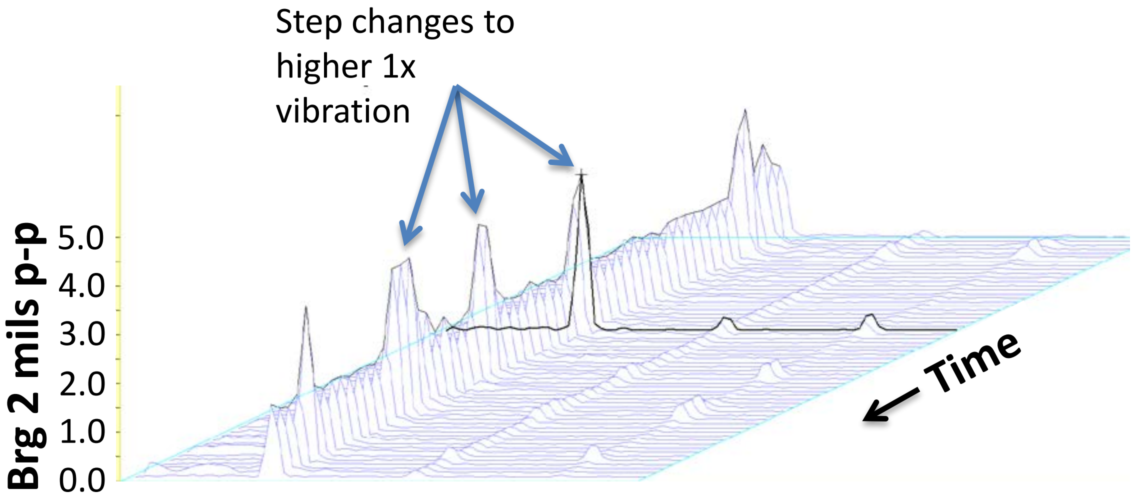
- Was Bearing 2 overloaded / damaged whilst running with damage to Bearing 1?



Data review – cranking speed (3300 rpm) waterfall

At cranking speed of 3,300 RPM, turning speed vibration exhibits step changes between two clearly distinct ‘states’ at random intervals (bistable system)

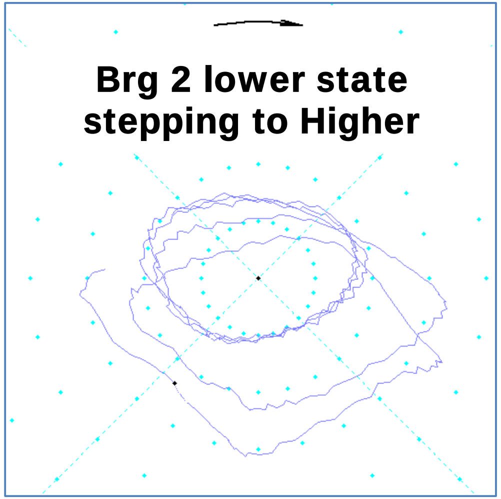
Bearing 2Y cranking speed waterfall

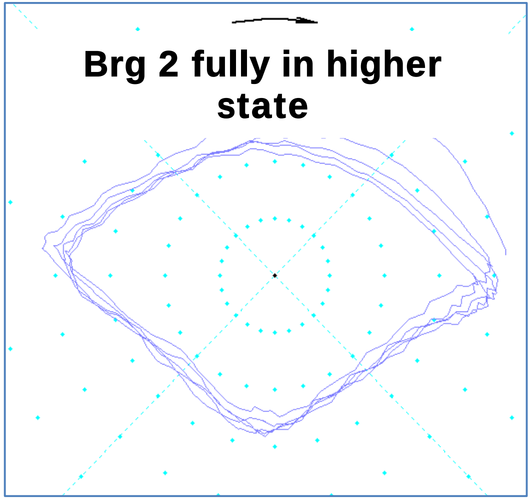


Data review – cranking speed (3300 rpm) orbits

- Left orbit shows several rotations at lower stable state, followed by step to higher state
- Right orbit shows fully in higher vibration state, shape (abrupt trajectory changes) suggests low friction rub events

Bearing 2 orbit plot during state change





Bearing 2 orbit plot in higher vibration state

Bistable vibration at cranking speed

- We have not found references to such behaviour in the literature
- We do not have an explanation, other than to note that:
  - Fluid film bearings are a ‘hardening non-linear system,’ i.e., due to clearance, stiffness is relatively soft up to a limit of deflection, then increases sharply at the ‘soft deflection limit’[3].
  - If clearances increase, this soft deflection limit will increase significantly, and,
  - Ehrich notes that non–linear systems may have multiple solutions.

More pragmatically, is further damage occurring at Bearing 2 during cranking?

[3] *Ehrich Handbook of rotor dynamics*

Current status

- Case is not complete – unit not currently required to maintain the current level of production, still in service as a poor spare
- Most probable diagnosis - likely bearing damage / increased clearances at Bearing 2, possibly a consequence of earlier Bearing 1 failure
- If required to be run prior to overhaul, recommended to *minimise cranking time* and to *reduce lube oil temperature* to lower end of tolerance – increasing viscosity in this manner increases both effective bearing stiffness and damping

Lessons learned

- It is essential when acquiring run-up / coast-down data to ensure data is acquired from standstill, i.e., 0 RPM. This allows determination of rotor running position from centreline plots.
- When an initial bearing failure occurs, this may result in overloading of other bearings on that shaft
- If continued operation with bearing damage is essential, temporary improvement of vibration amplitudes may be achieved by lowering lube oil temperatures

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Thank you

Questions?

wood.

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