Rub, Morton Effect, Parametric Excitation, and Shaft Crack in Rotating Machinery - Real Cases

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Presenter Bios

Dr. John Yu joined Bently Rotor Dynamics Research Corporation in 1998, followed by General Electric - Bently Nevada in 2002. He has performed not only rotor dynamic research but also machinery vibration diagnostics for customers worldwide, and is now Senior Technical Manager of Machinery Diagnostic Services at Bently Nevada. He has over 50 technical papers in refereed journals and conference proceedings. He holds a PhD in Mechanical Engineering from University of Alberta, and is an ASME Fellow.

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Abstract

This presentation discusses rub, Morton effect, parametric excitation, and shaft crack cases in rotating machines. Rub occurs more often besides unbalance and misalignment. The current rub topic covers all types of rub. The Morton effect has similar vibration pattern as the Newkirk effect, though it occurs not that often. Parametric excitation here is referred to subharmonic behavior such as ½X vibration, due to either rub or bearing looseness. Since ½X is touched here, shaft crack manifested by 2X vibration is also discussed briefly.

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Rub Diagnostics

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1. Introduction

Types of rub

Almost always the case in the field on turbomachinery

- **Newkirk effect (thermal rub behavior) manifested by 1X vibration excursion**
- **Forward precessional full annular rub with 1X vibration when passing resonance speed**
- **Reverse precessional full annular rub (dry whirl/whip)**
- **Bouncing type of rub with fractional frequency (parametric excitation such as ½X at speed of 2 times resonance)**
- **Morton effect (technically not a rub)**

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1. Introduction (Cont.)

Root-causes of 1X vibration excursion:

Newkirk effect

Thermal transient

Steam loading

Changes in alignment

Changes in unbalance

Morton effect

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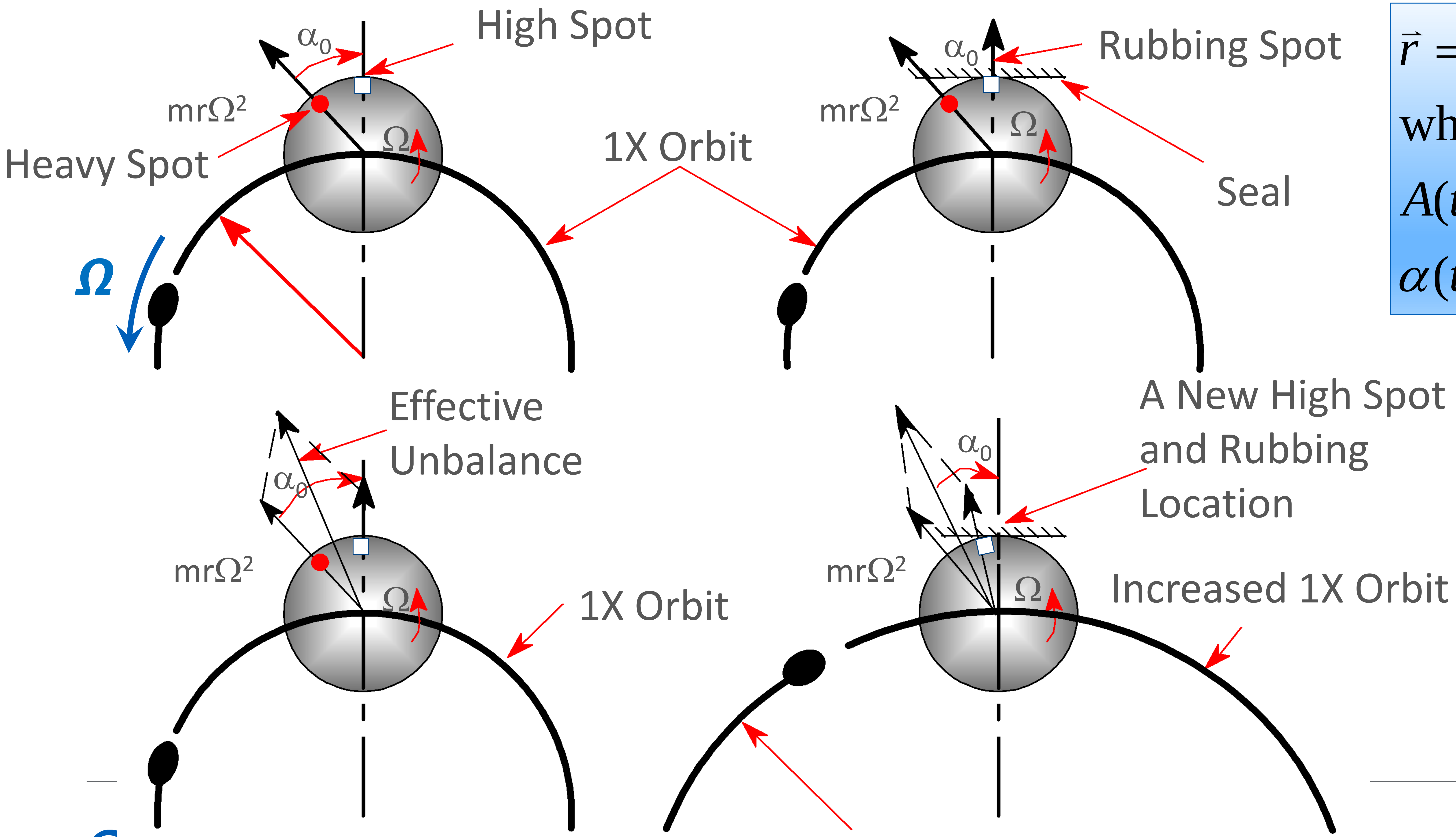
1. Introduction (Cont.)

Thermal Bow Effect of Rub – causing 1X amplitude & phase to change



$$\vec{r} = A(t)e^{j[\Omega t - \alpha(t)]}$$

where

$A(t)$: Amplitude varying with time

$\alpha(t)$: Phase lag varying with time

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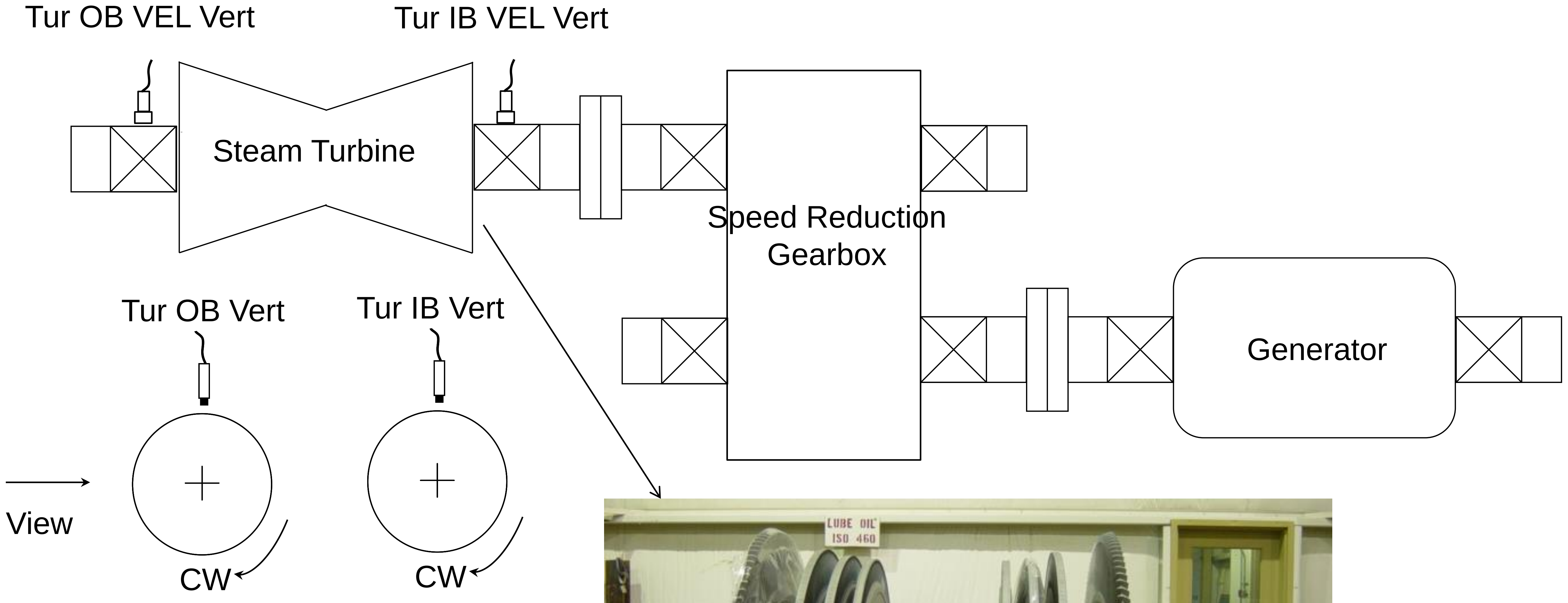
Case 1: Steady-state and transient data

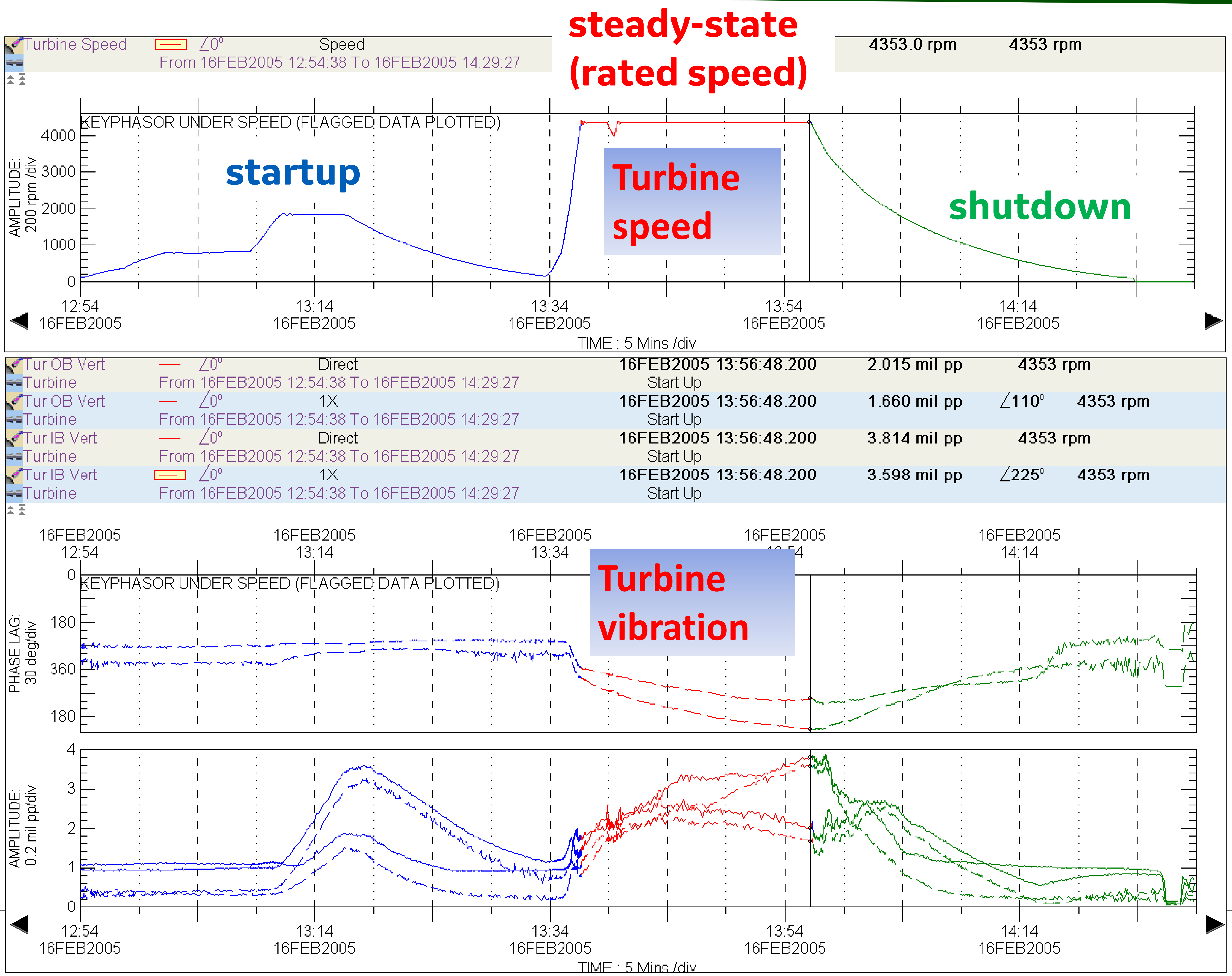
- 9-MW Steam Turbine Generator
- 4350 rpm on turbine and 1800 rpm on generator
- High turbine vibration tripped the unit



Steam Turbine Rotor

Case 1: Steady-state and transient data (Cont.)

Trend plots



Turbine inboard vibration increased from 1 mil pp to 4 mil pp 20 minutes after reaching rated speed

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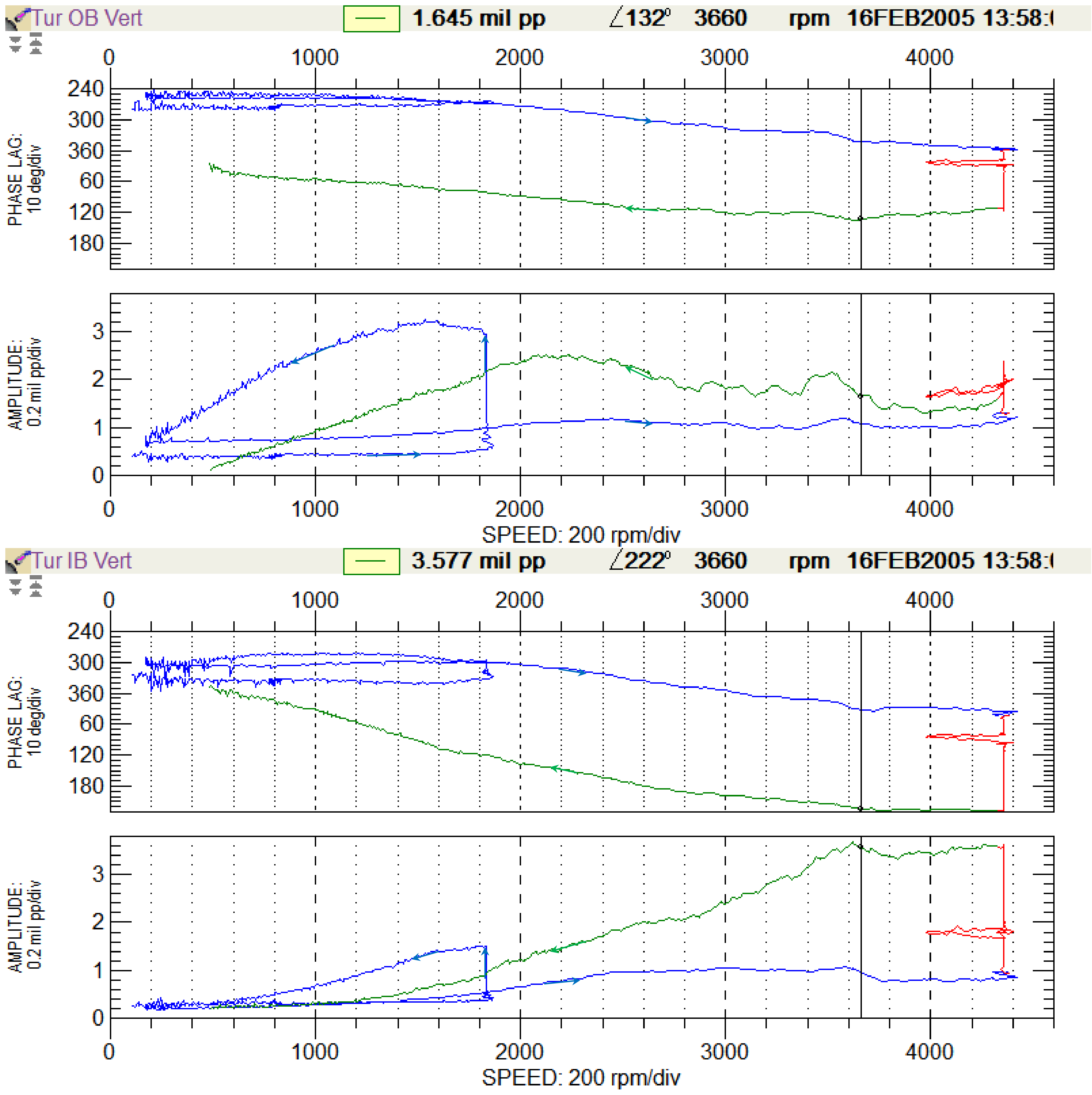
Case 1: Steady-state and transient data (Cont.)

1X Bode plots from proximity probes



- **High vibration presented at both high and low speed range.**
- **When speed is low, high vibration is mainly due to shaft bow.**

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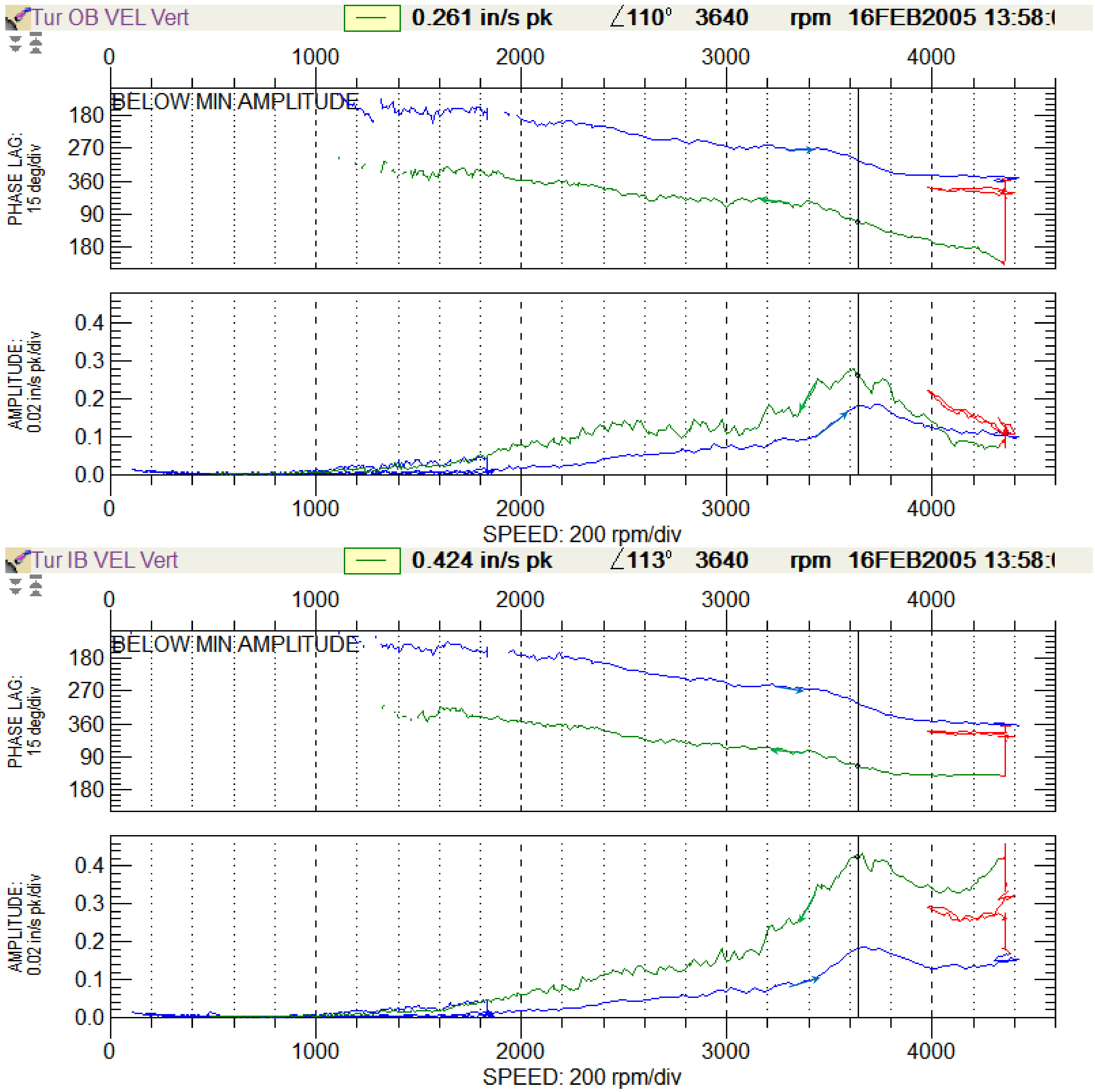
Case 1: Steady-state and transient data (Cont.)

1X Bode plots from seismic transducers



- **High vibration only presented at high speed or critical speed ranges.**
- **Thermal shaft bow at low speed cannot be verified.**

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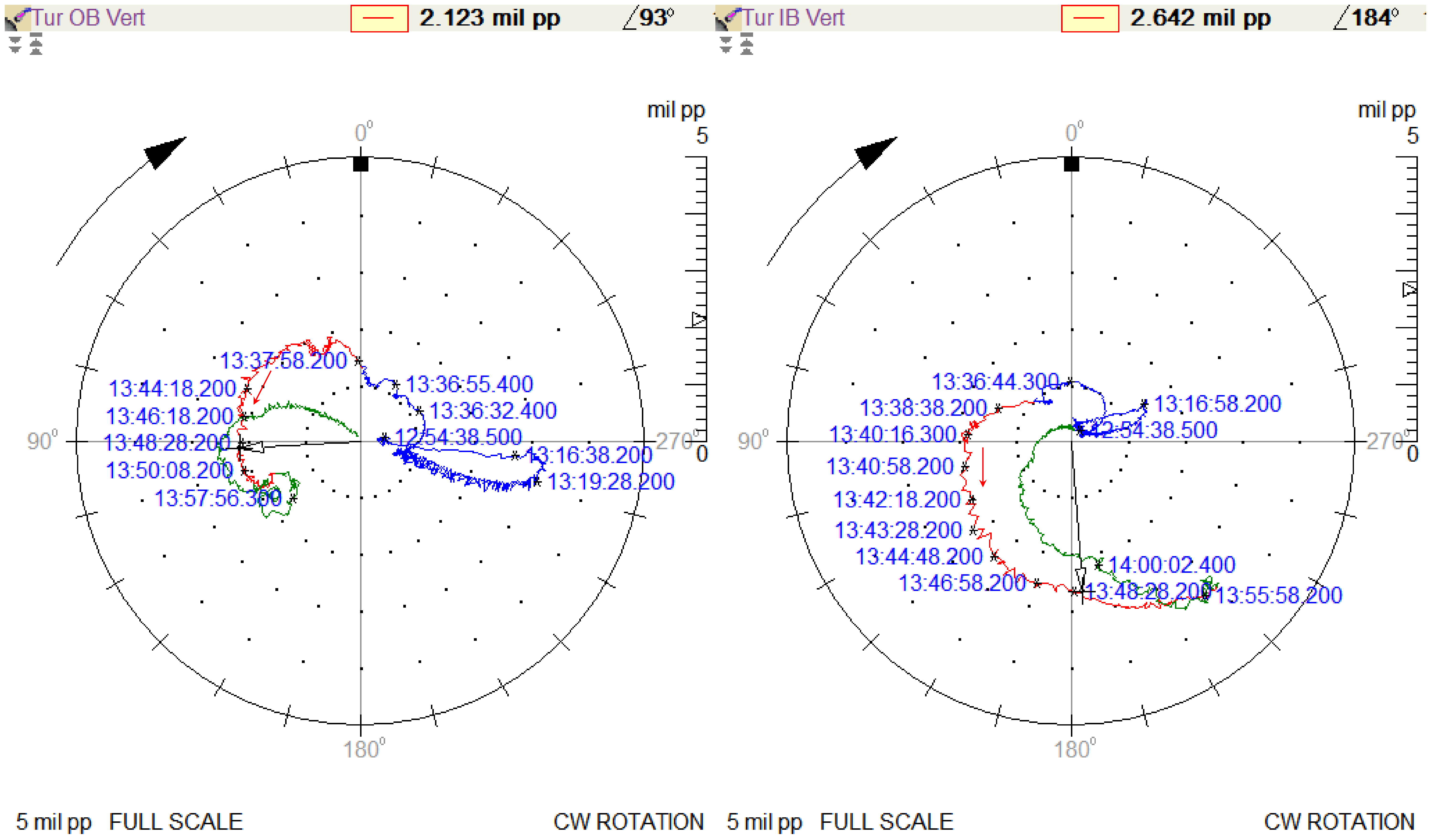
Case 1: Steady-state and transient data (Cont.)

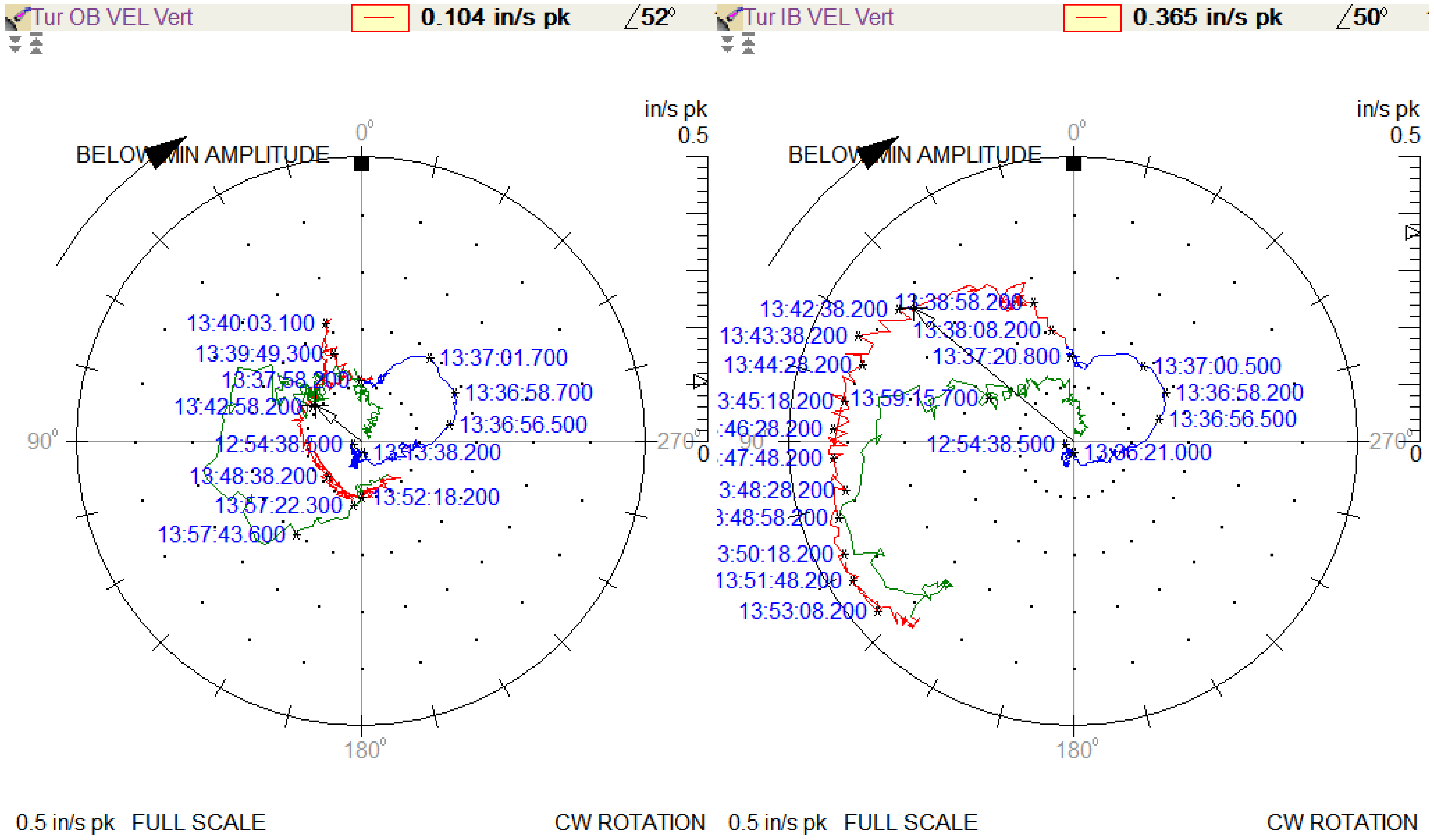
1X polar plots from proximity probes



- **1X reverse rolling vectors at rated speed or steady-state condition.**
- **Less meaningful when speed is changing or in transient condition.**

Case 1: Steady-state and transient data (Cont.)

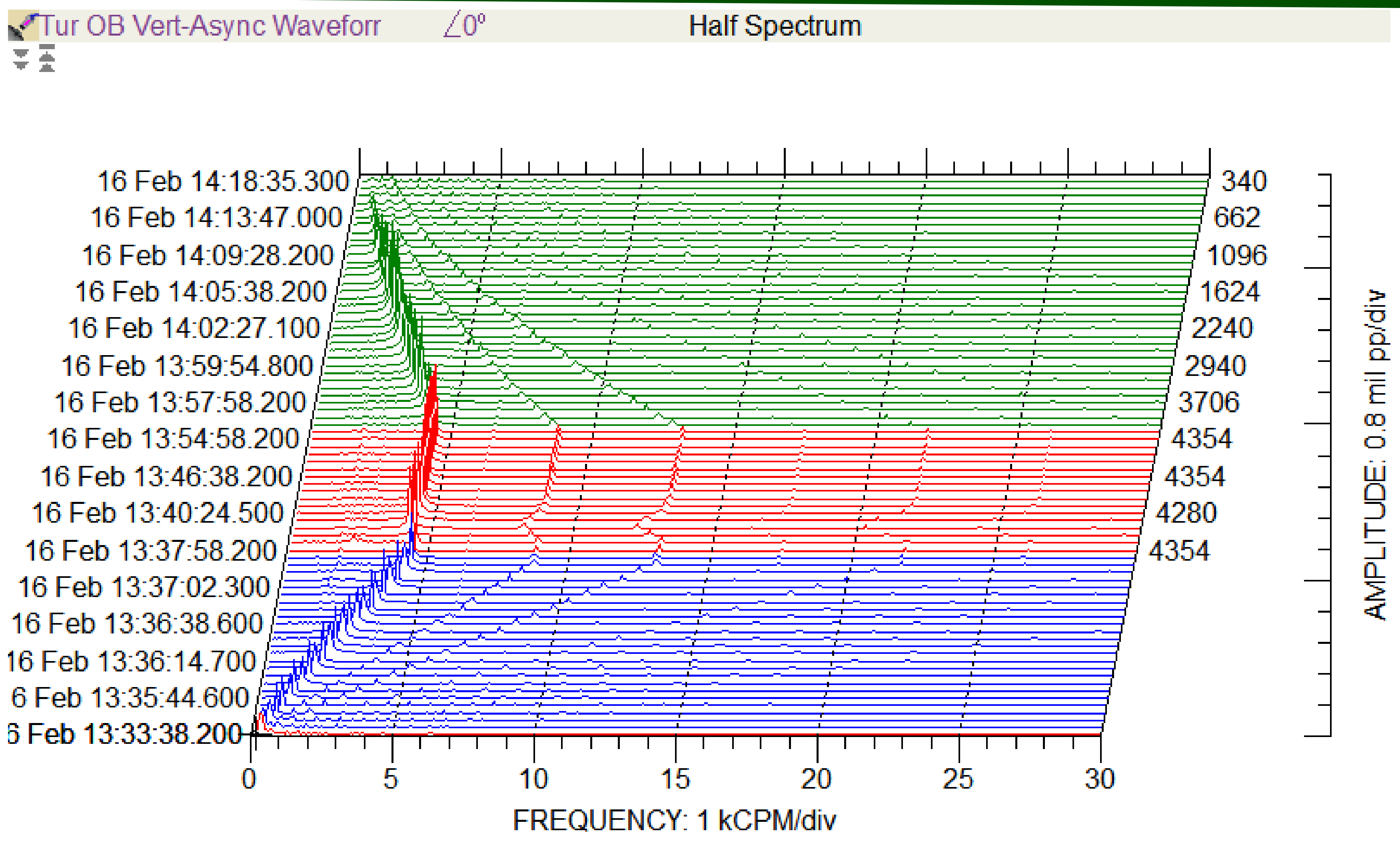
1X polar plots from seismic transducers

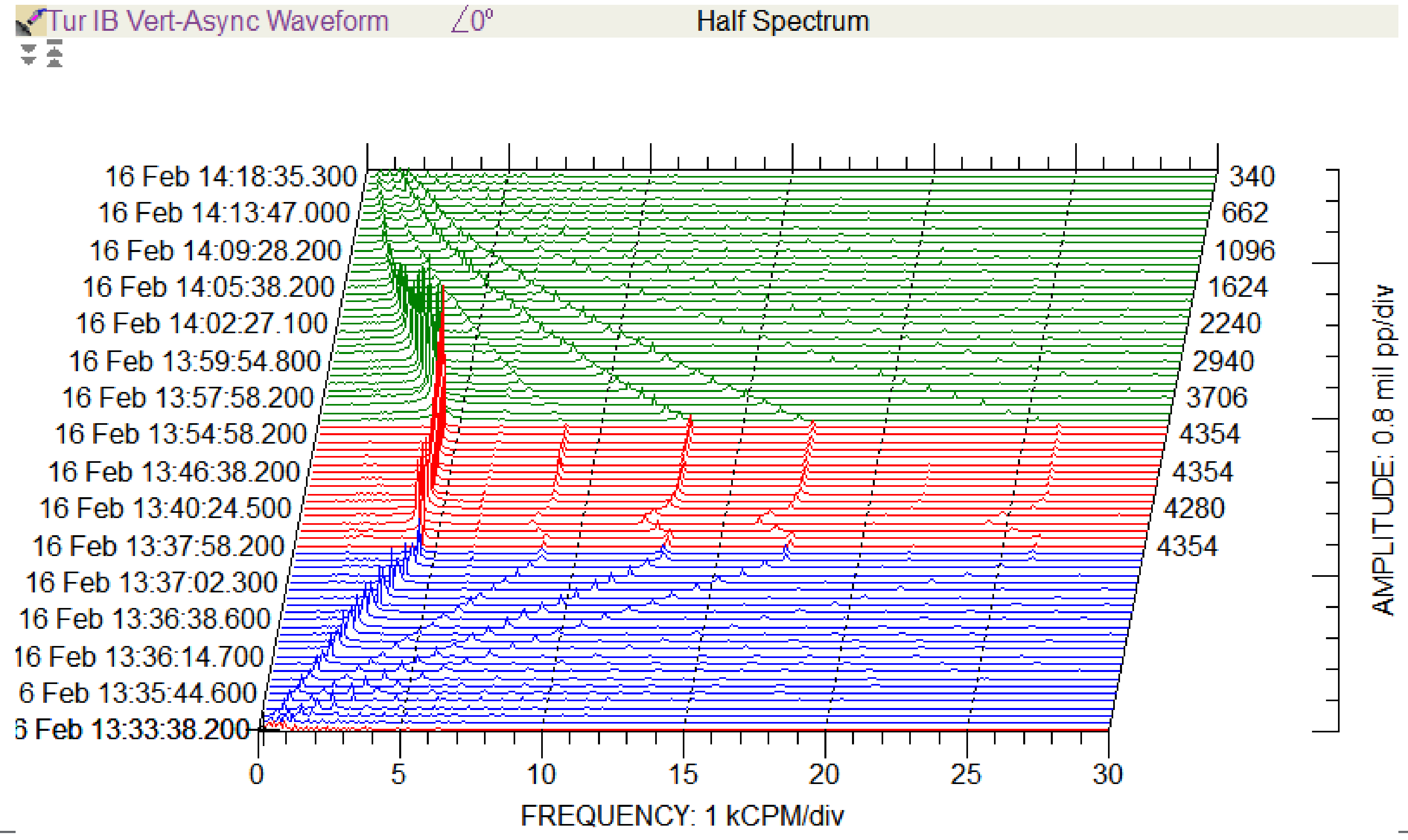


- 1X reverse rolling vectors at rated speed or steady-state condition.
- Less meaningful when speed is changing or in transient condition for rub diagnostics.

Case 1: Steady-state and transient data (Cont.)

Waterfall plots from proximity probes





- **Super-harmonic components (2X, 3X, etc.) could exist all the time due to runout on probe-viewing area whether a rub occurs or not.**
- **Rich harmonics may not be a good indicator of rub from proximity probes**

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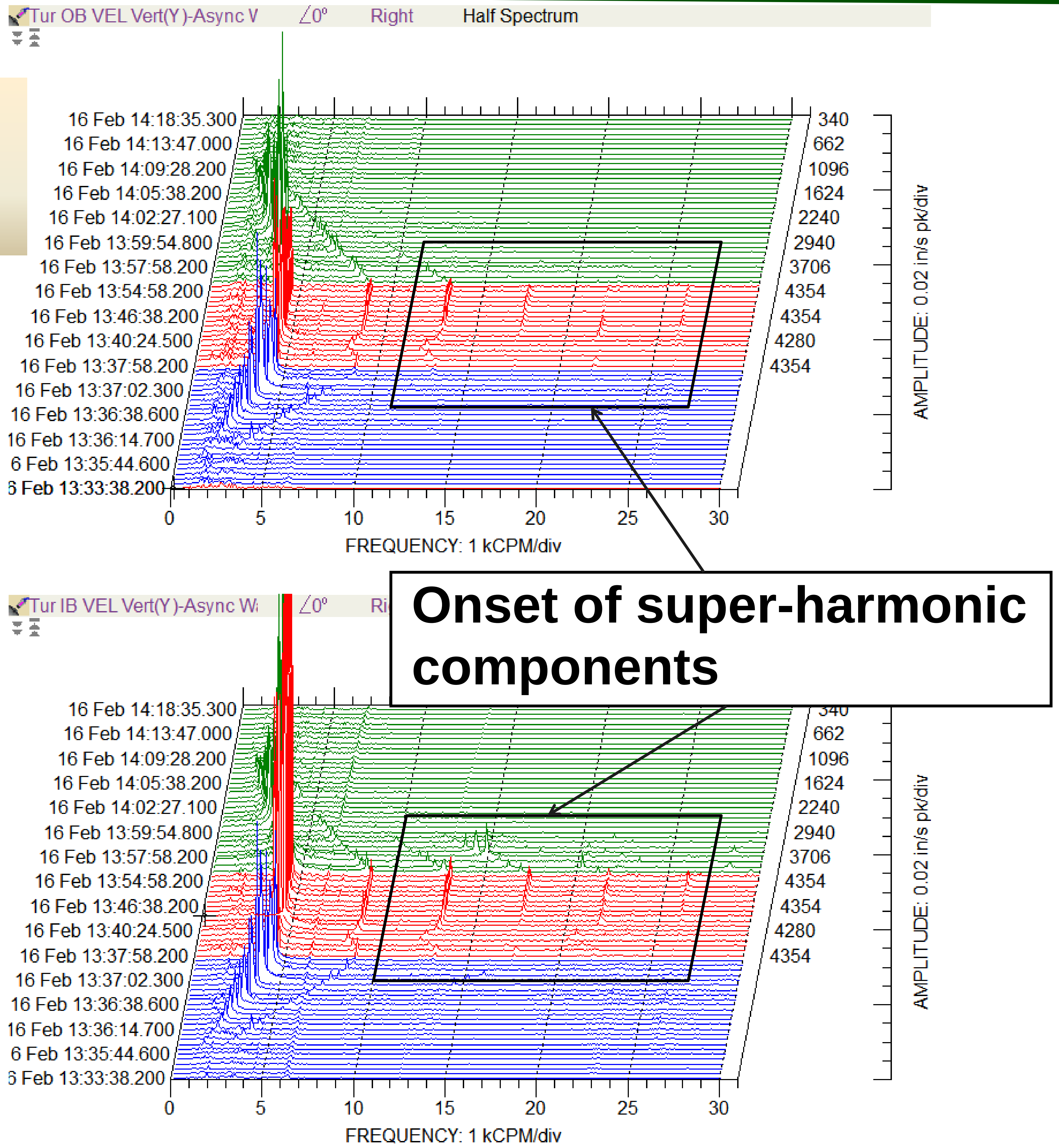
Case 1: Steady-state and transient data (Cont.)

Waterfall plots from seismic transducers



- **True readings of super-harmonics if any from seismic transducers.**
- **Rich harmonics is a good indicator of rub from seismic transducers.**

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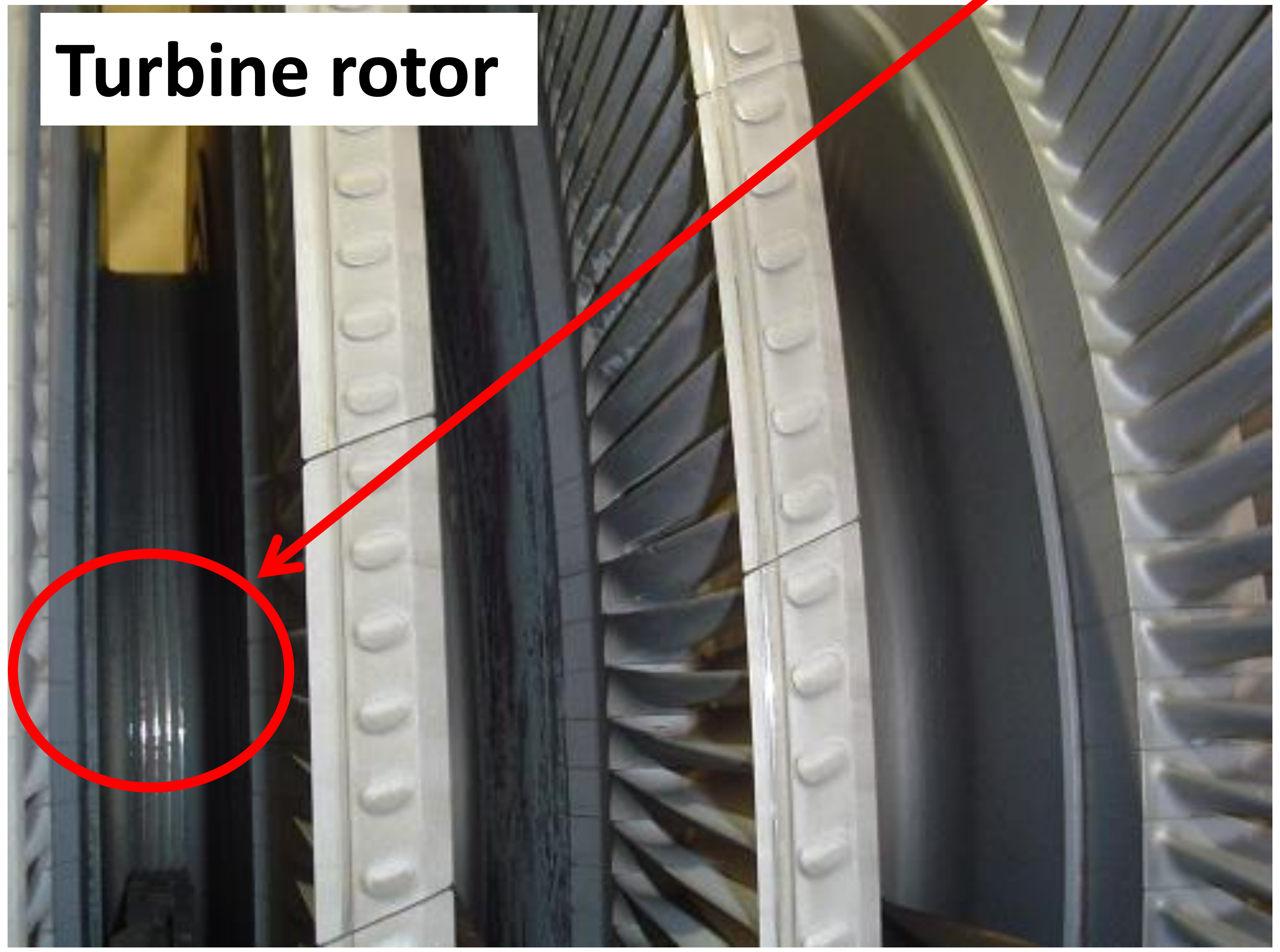
Case 1: Steady-state and transient data (Cont.)

Signs of rub as seen from turbine rotor and top case



Seal clearance setup was found to be incorrect!

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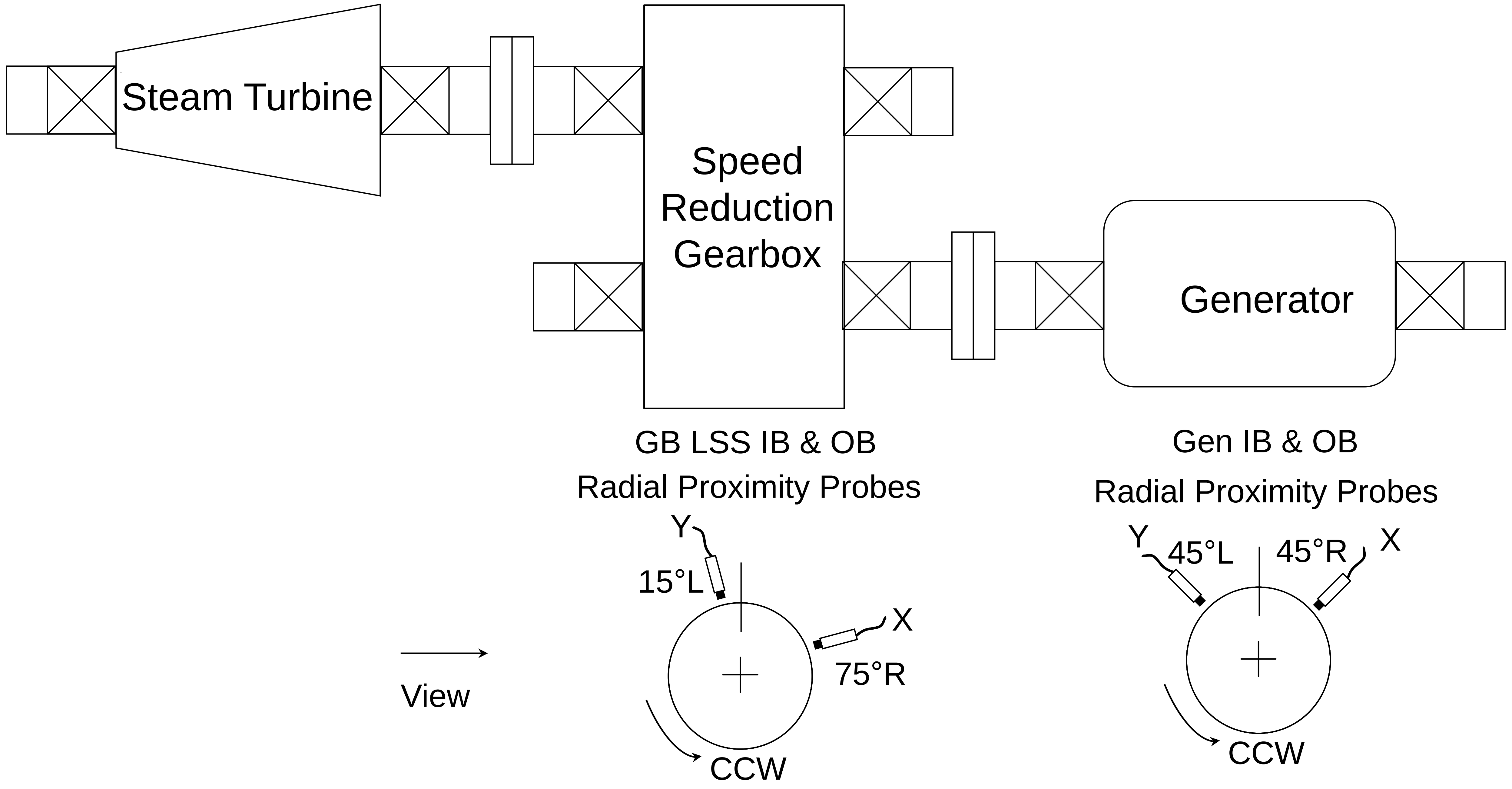
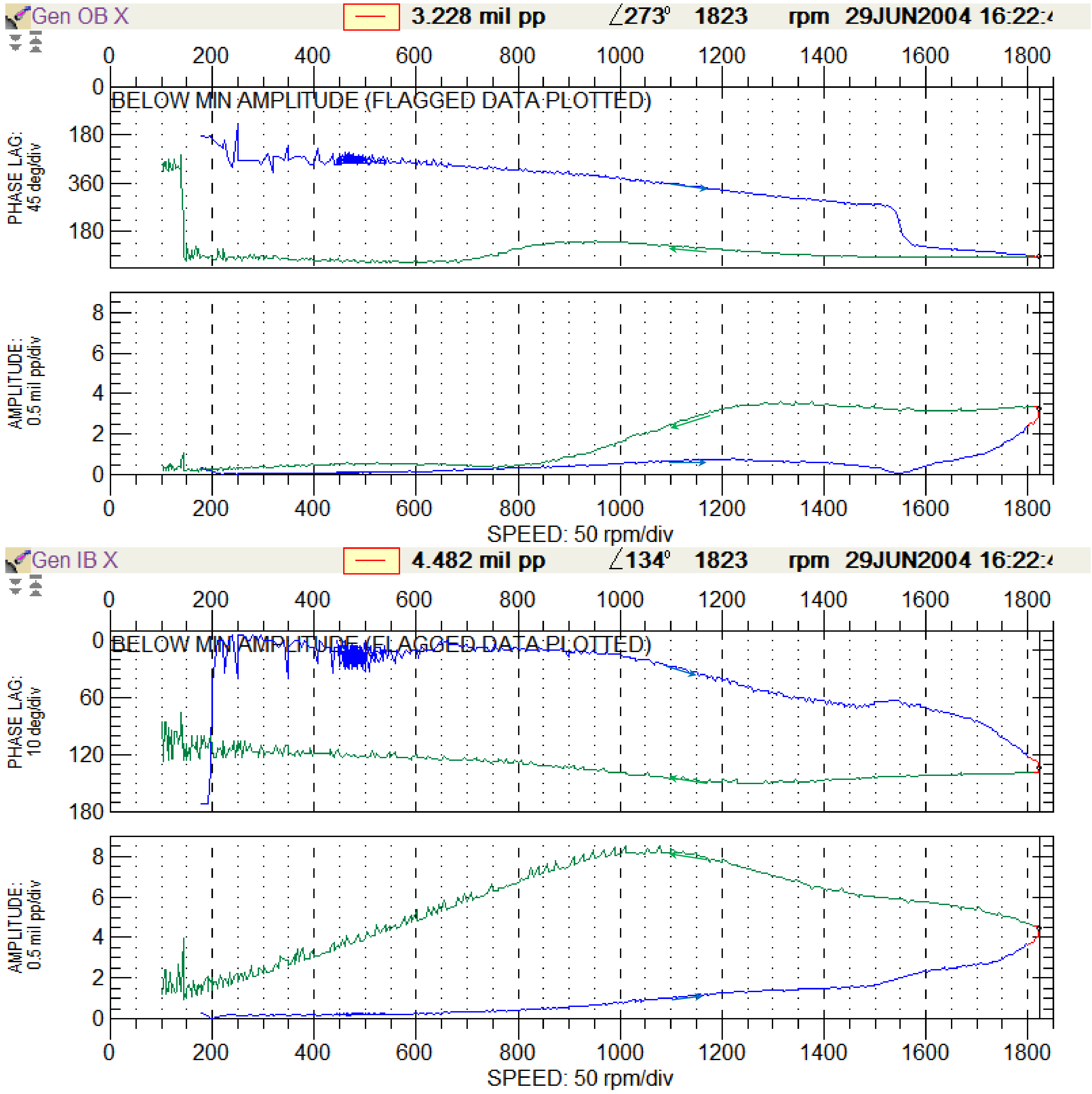
Case 2: Transient data only

- 28-MW Steam Turbine Generator
- 3000 rpm on turbine and 1800 rpm on generator
- High generator vibration tripped the unit



Case 2: Transient data only (Cont.)

1X Bode plots from proximity probes



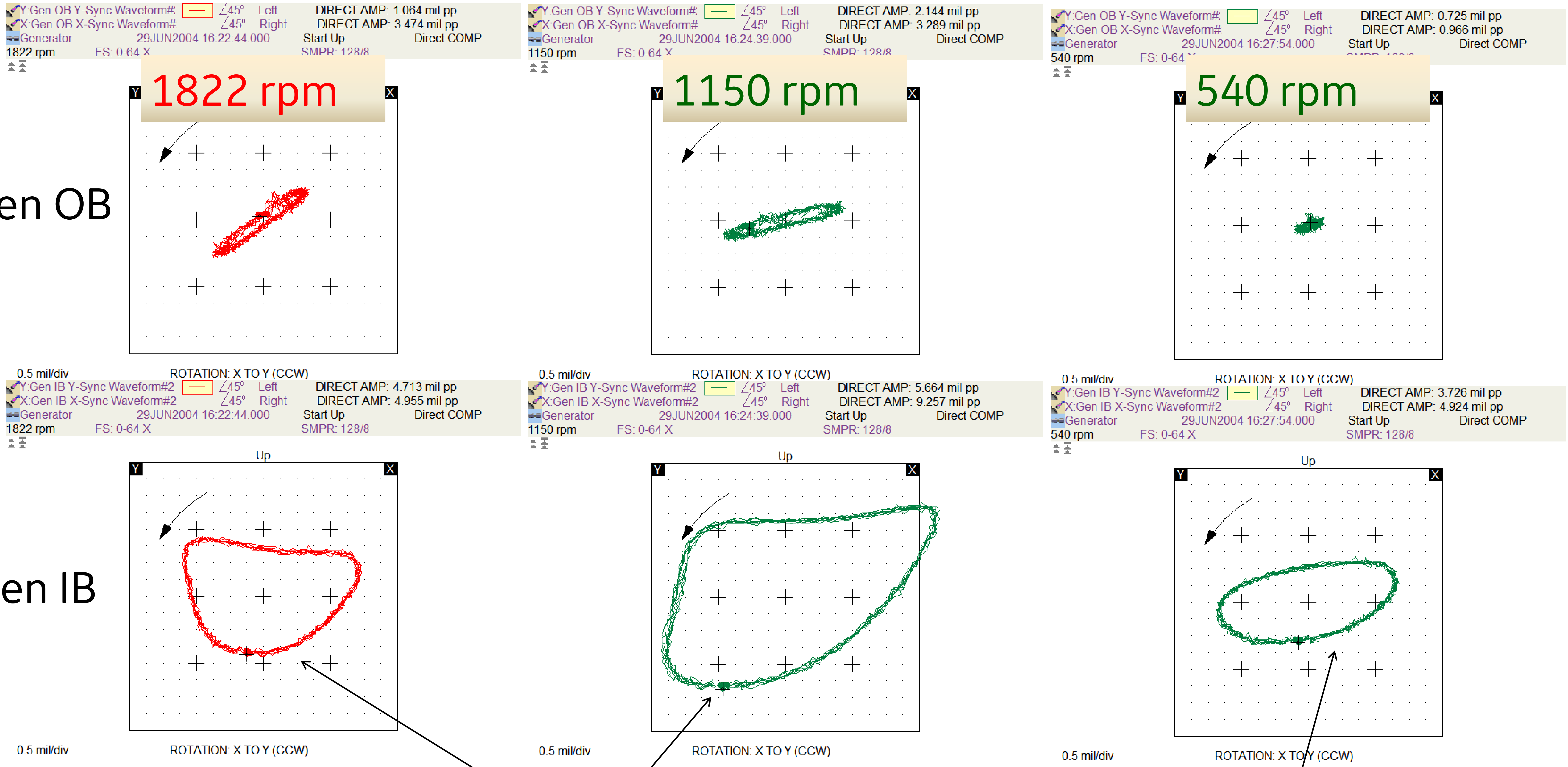
- **Significantly higher vibration during shutdown than during startup.**
- **This indicates thermal bow due to a rub.**

Case 2: Transient data only (Cont.)

Orbit plots (waveform-compensated)

Gen OB

Gen IB



Distorted orbits with abrupt changes due to rub

Rub disengaged

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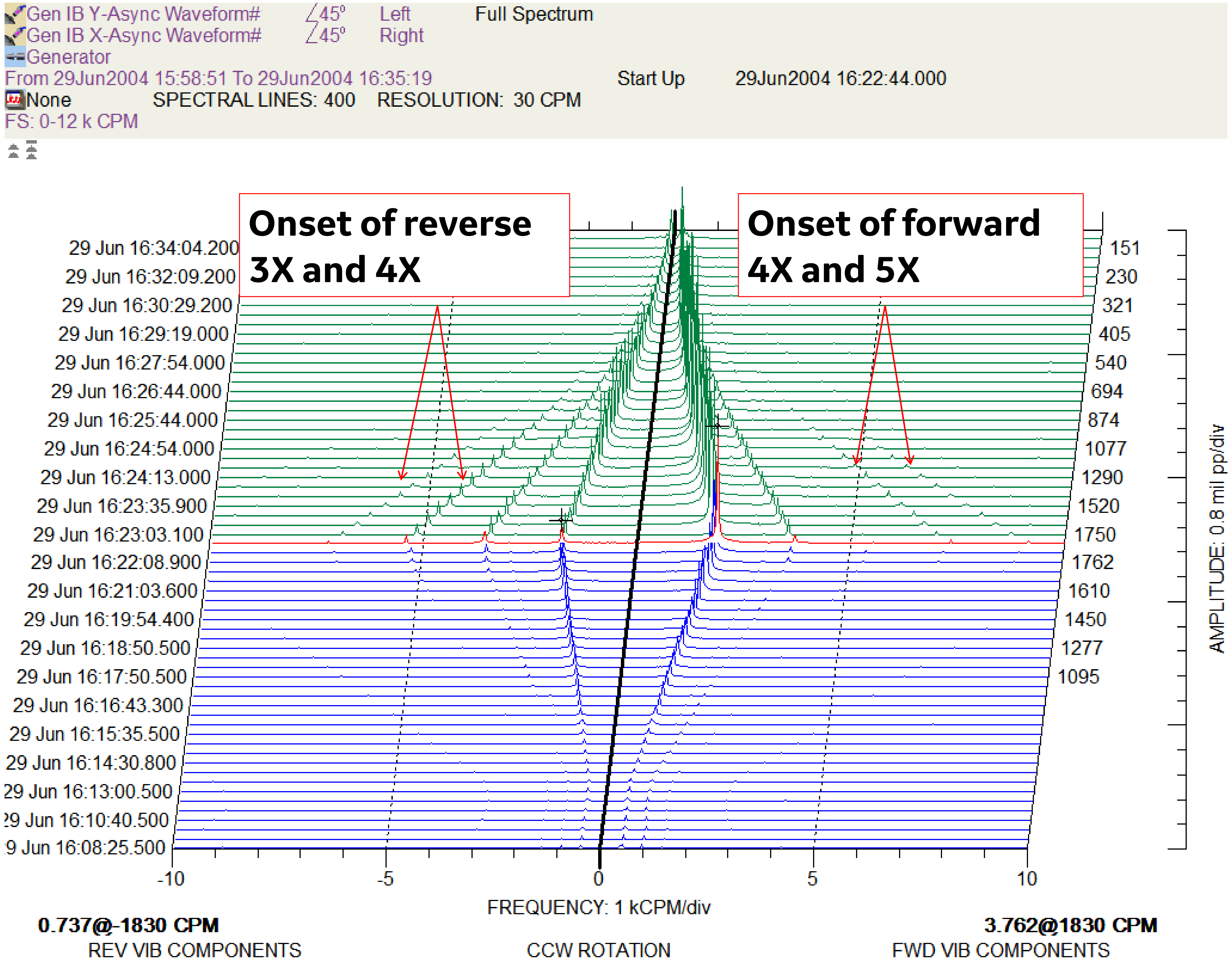
Case 2: Transient data only (Cont.)

Full-spectrum waterfall plots



- **Rich super-harmonic components due to distorted orbit shapes**

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Rub diagnostic rules in steady-state vs. transient condition

- In steady-state condition, polar plots give the most meaningful indications of rub.
- In transient conditions, bode and orbit plots may give the most meaningful indications of rub.
- In transient conditions, trend or polar plots may give less meaningful information for rub diagnostics.
- Distorted orbit may pinpoint the rub location in steady-state/transient condition. But many rubs can occur when orbits are very smooth and look normal.

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Summary and Discussion

Selection of vibration data plots for rub diagnostics

Type of plot	Speed condition	Key features to look for
Trend	Stead-state	Mainly screening to see If 1X vibration excursion occurs
Polar	Stead-state	How 1X vectors are rolling against shaft rotation
Bode	Transient	Comparison of shutdown vs. startup data
Orbit	All	Any distorted orbit (waveform-compensated)
Waterfall	All	Onset of rich harmonics

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There are more rub cases including those due to grounding brush (forward 1X rolling vector) in the following paper:

Yu, J. J., 2013, Rub Diagnostics Based on Vibration Data, ***ASME*** Paper GT2013-94203, International Gas Turbine & Aeroengine Congress & Exhibition, San Antonio, Texas.

Rub due to Oil Coking/Varnishing

- **Oil coking:**

due to heating of oil from hot steam, gas flow, etc.

- **Oil varnishing:**

due to oil quality, over loading, electro-static discharge (ESD), degradation of oil, etc.

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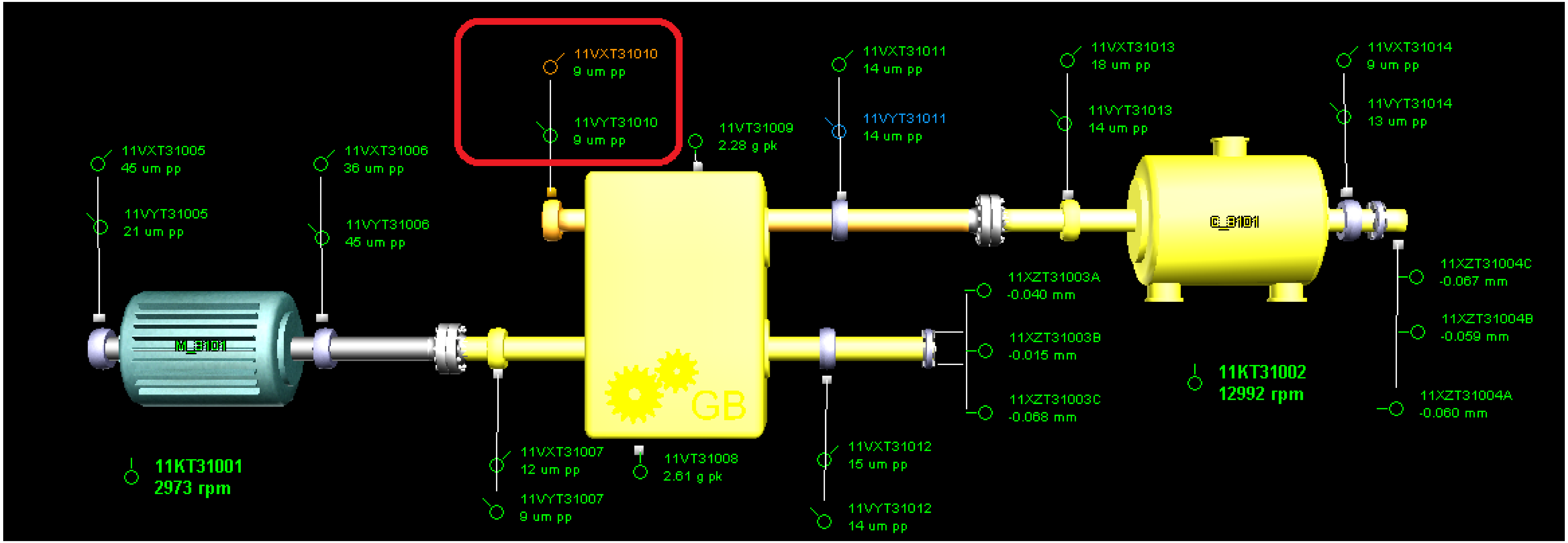
Case: Rub due to oil coking/varnishing



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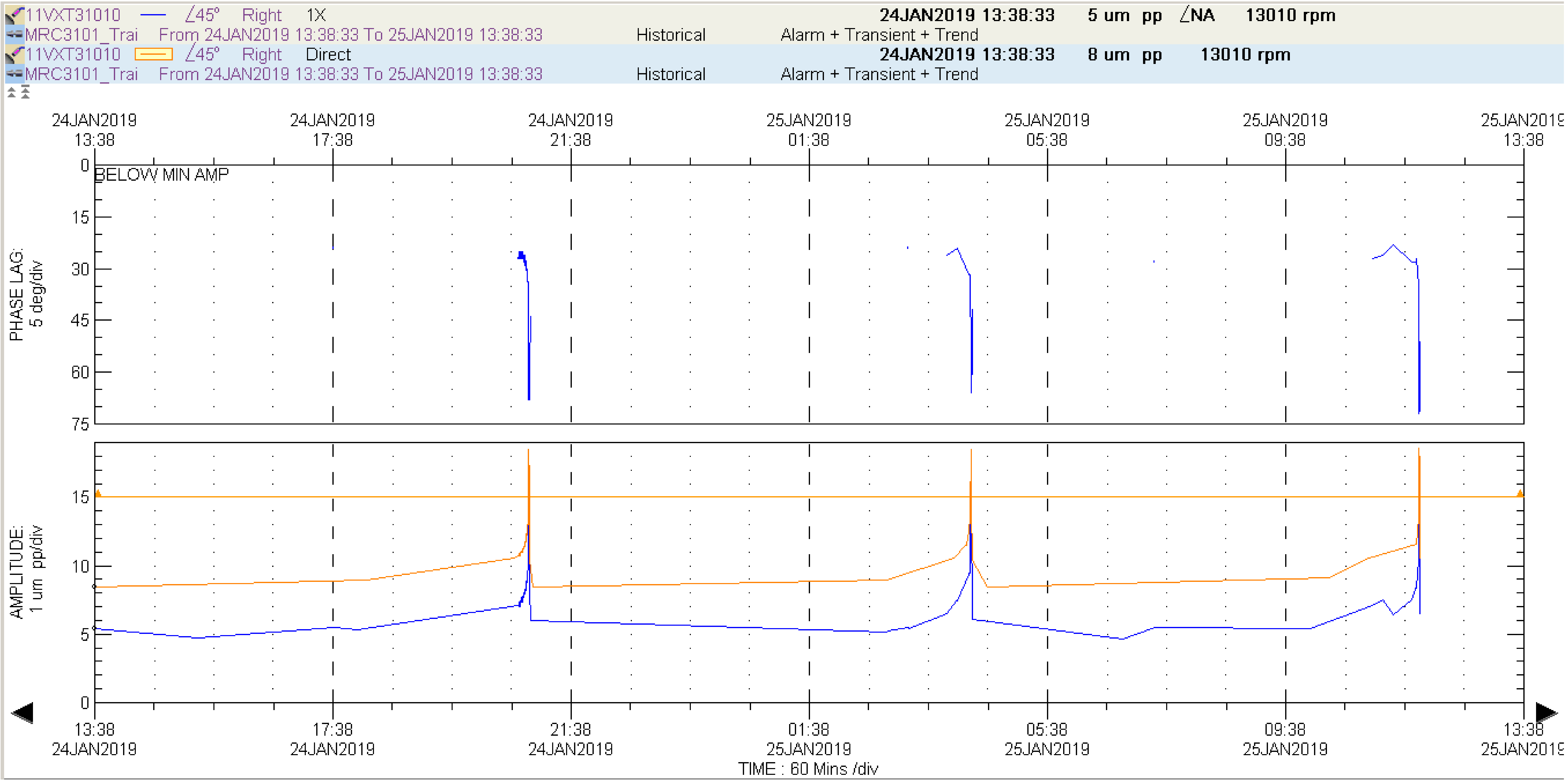
Case: Rub due to oil coking/varnishing



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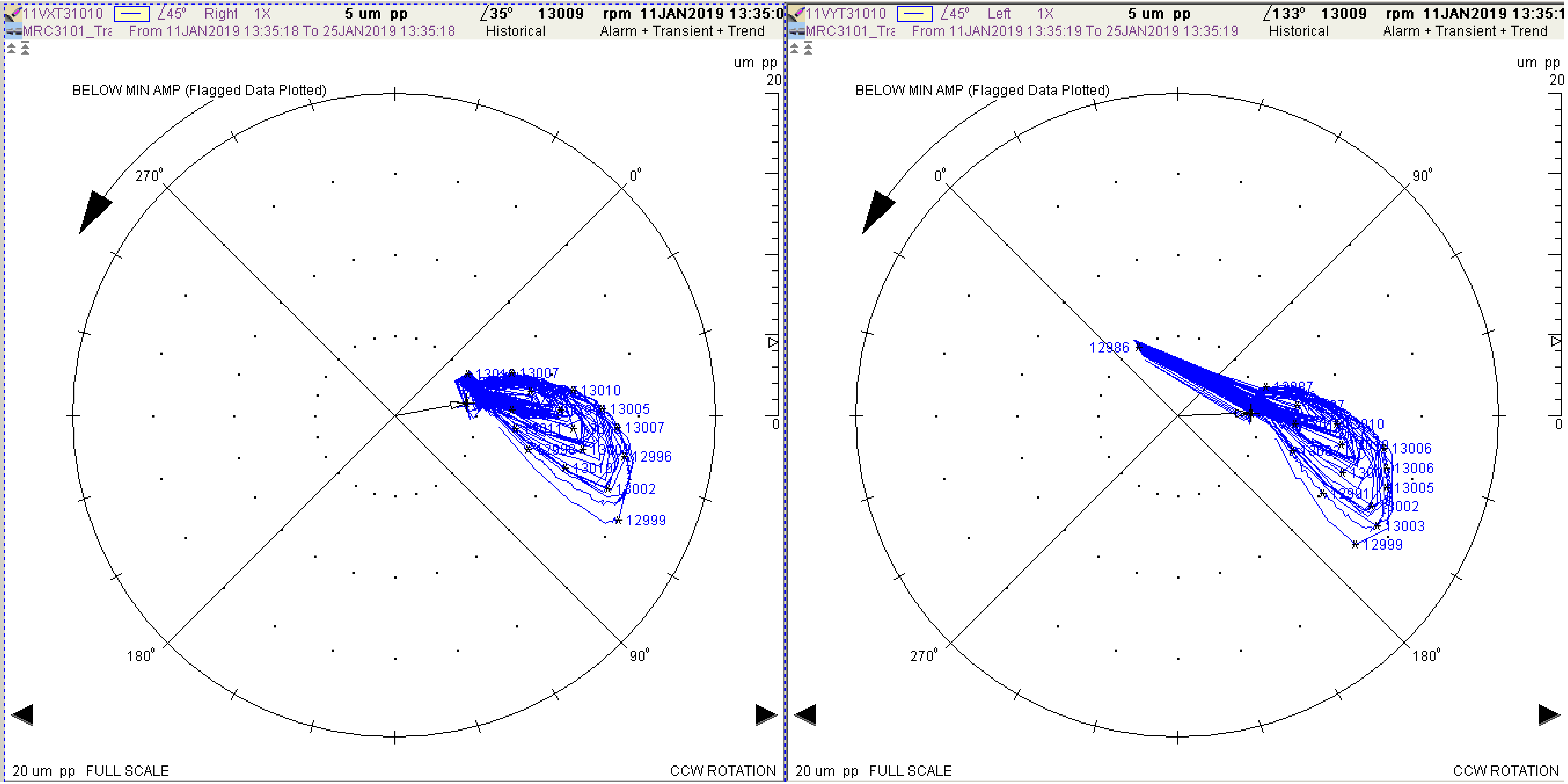
Case: Rub due to oil coking/varnishing



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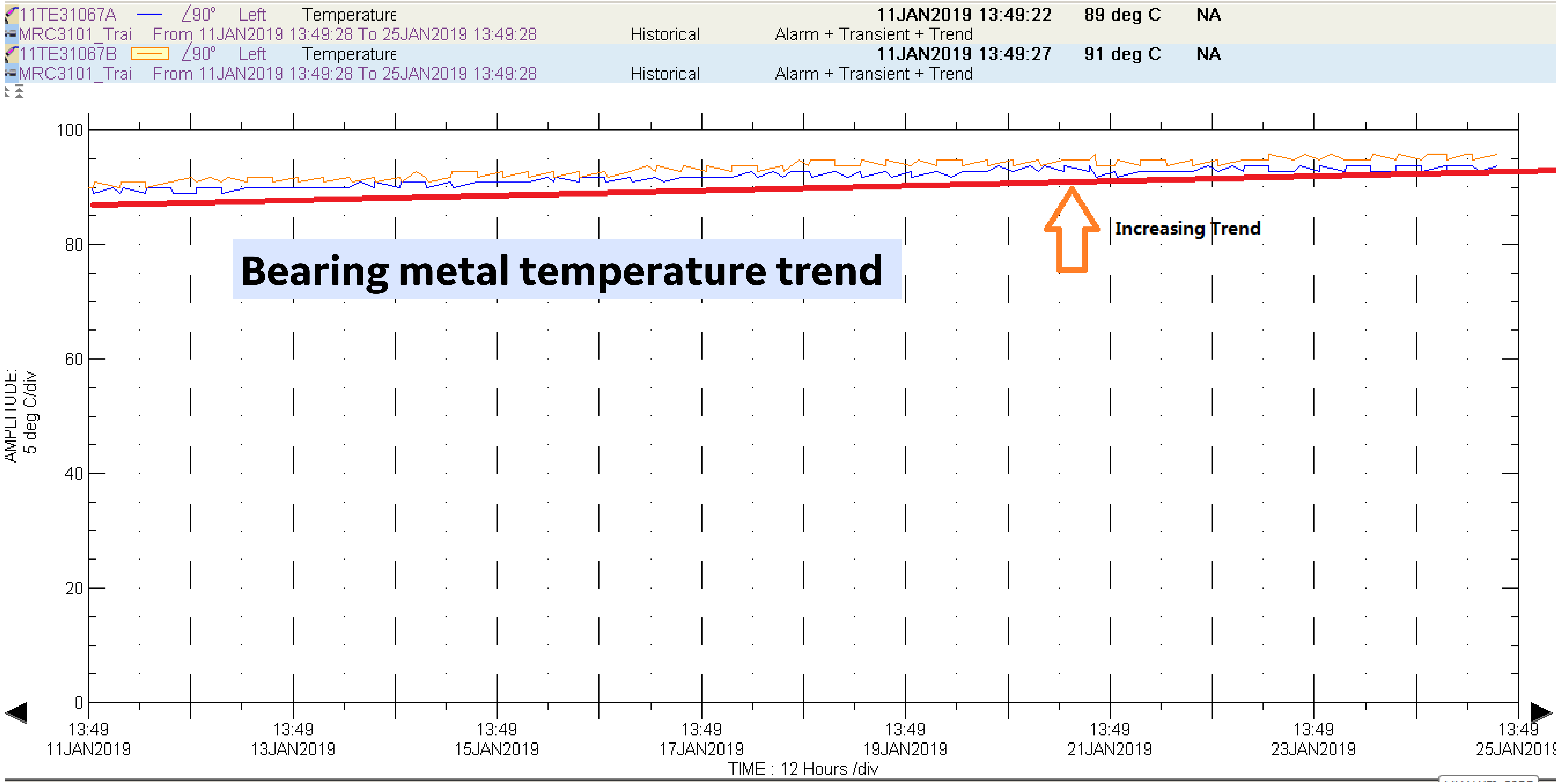
Case: Rub due to oil coking/varnishing



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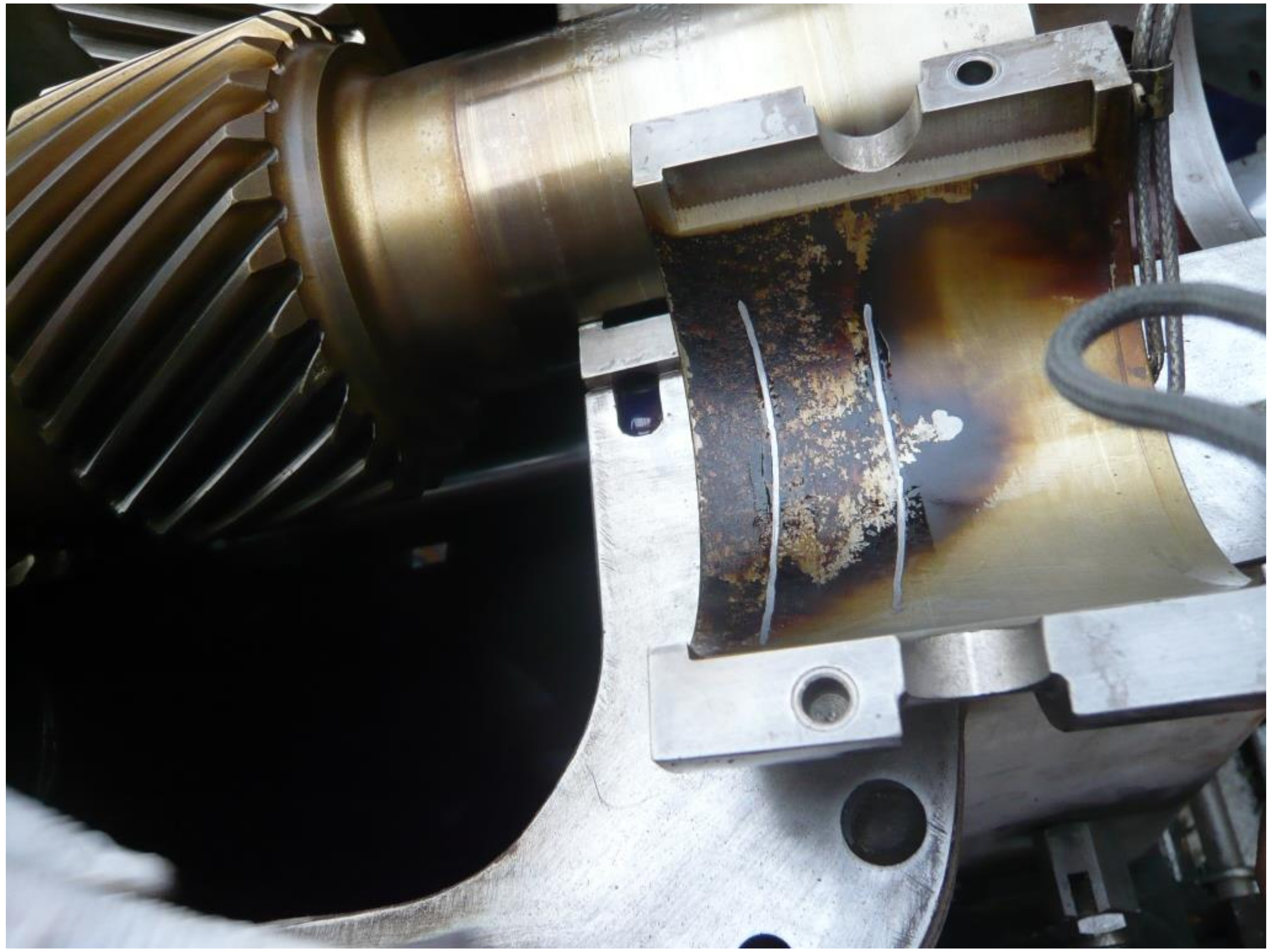
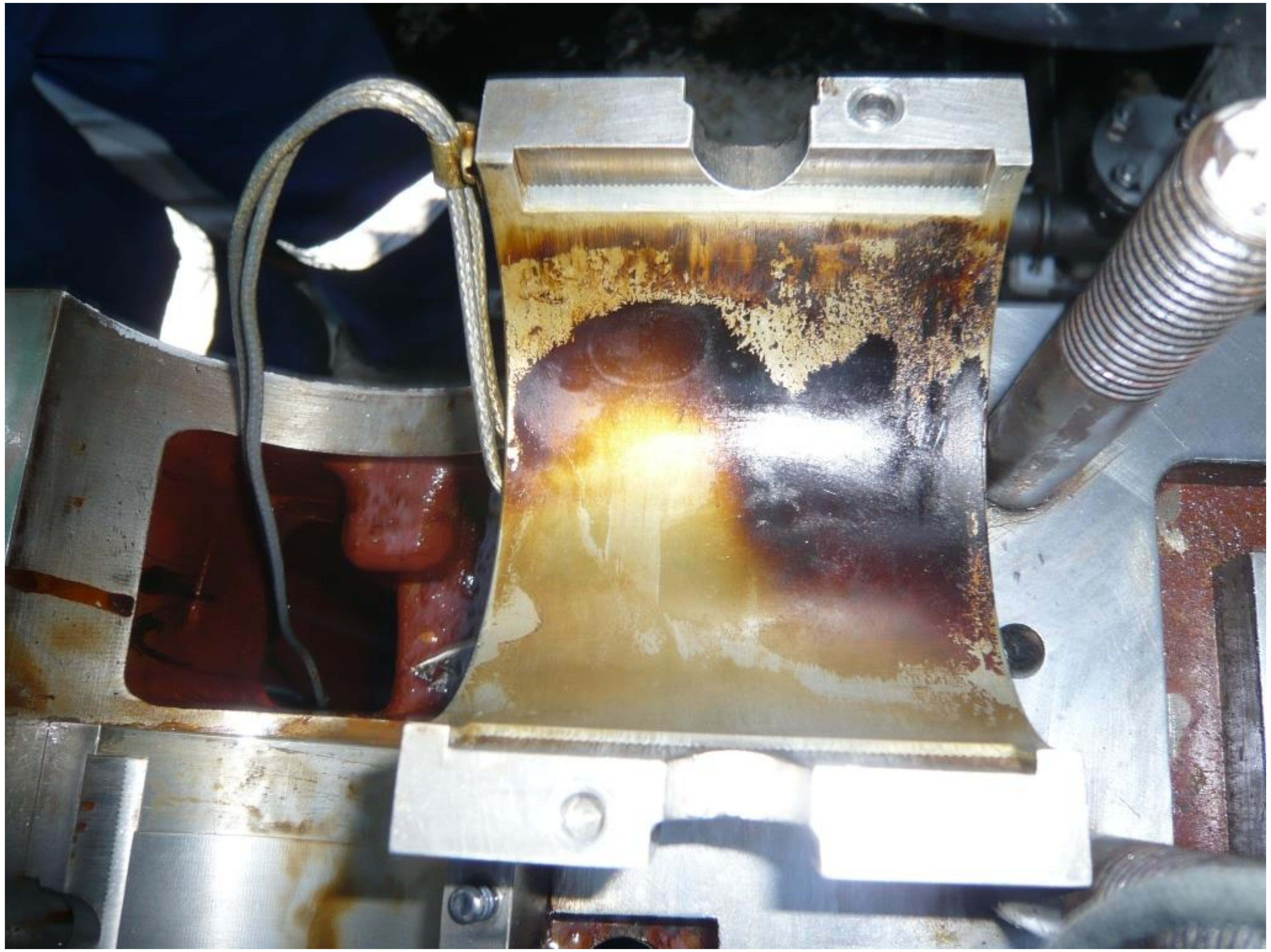
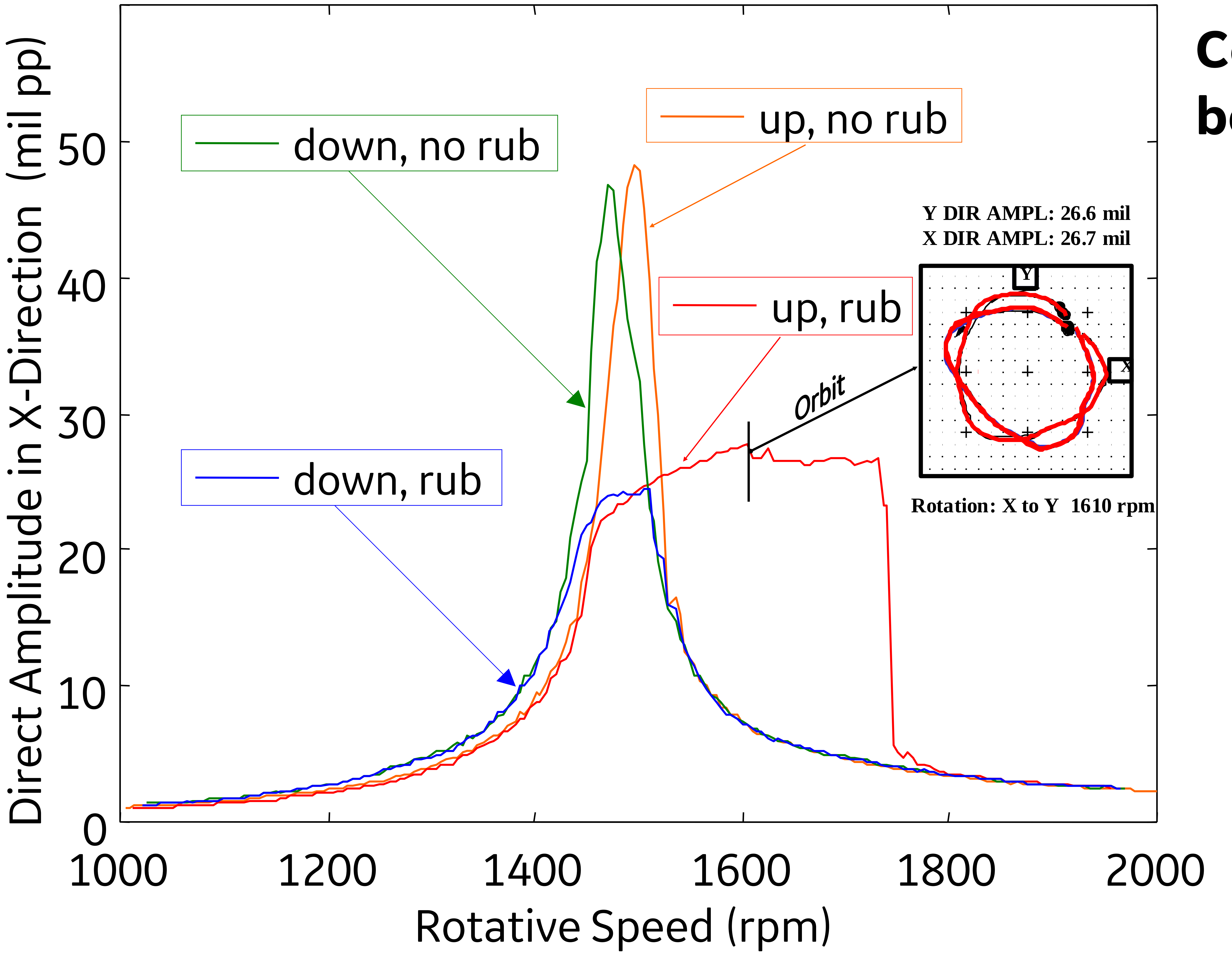
Case: Rub due to oil coking/varnishing





Forward full annular rub case on a rotor-kit



Comparison of direct responses between rub and no-rub cases

Note:

- **Resonance speed region with rub is greatly increased during runup.**
- **The rotor may bounce inside the seal with almost continuous contact.**

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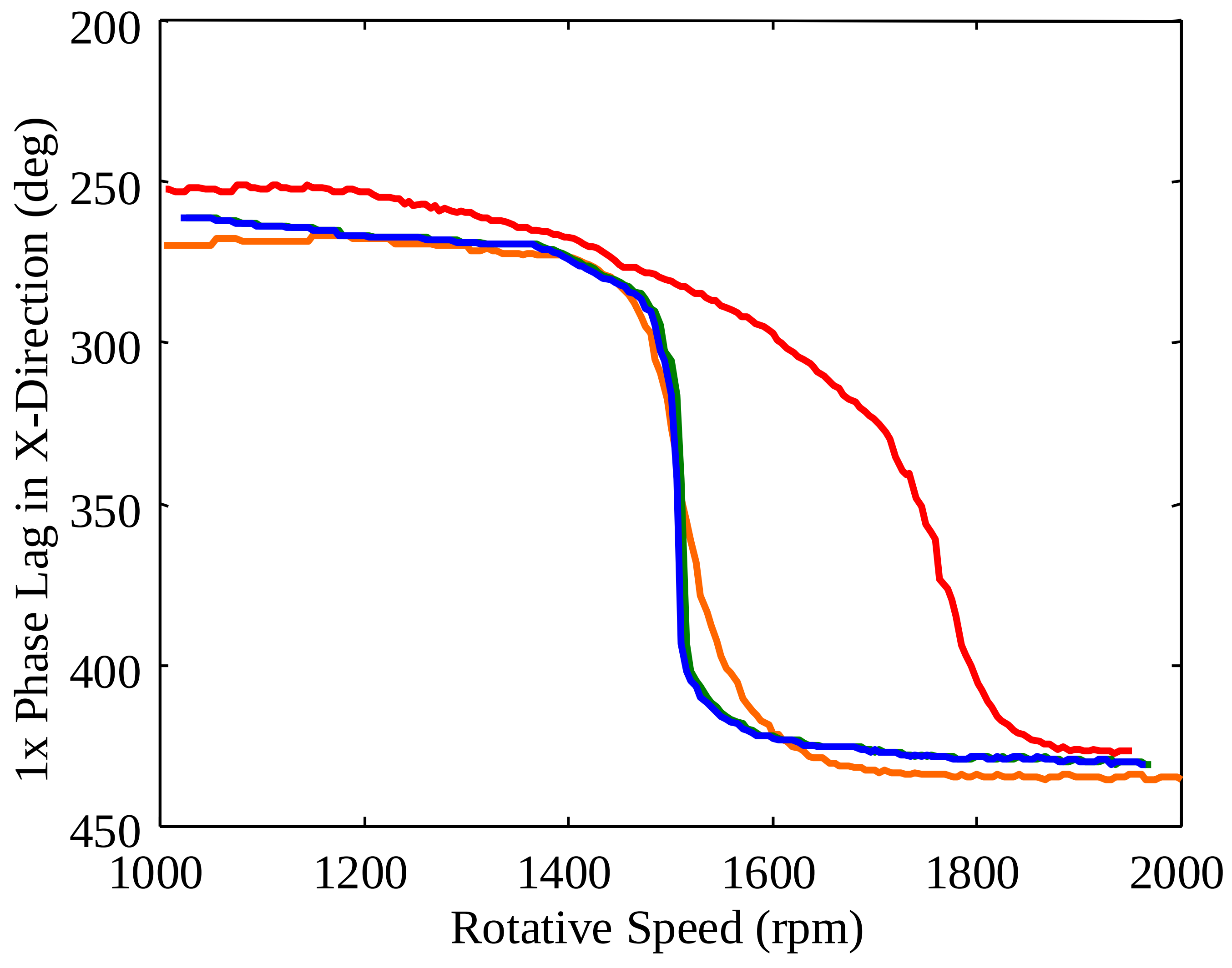
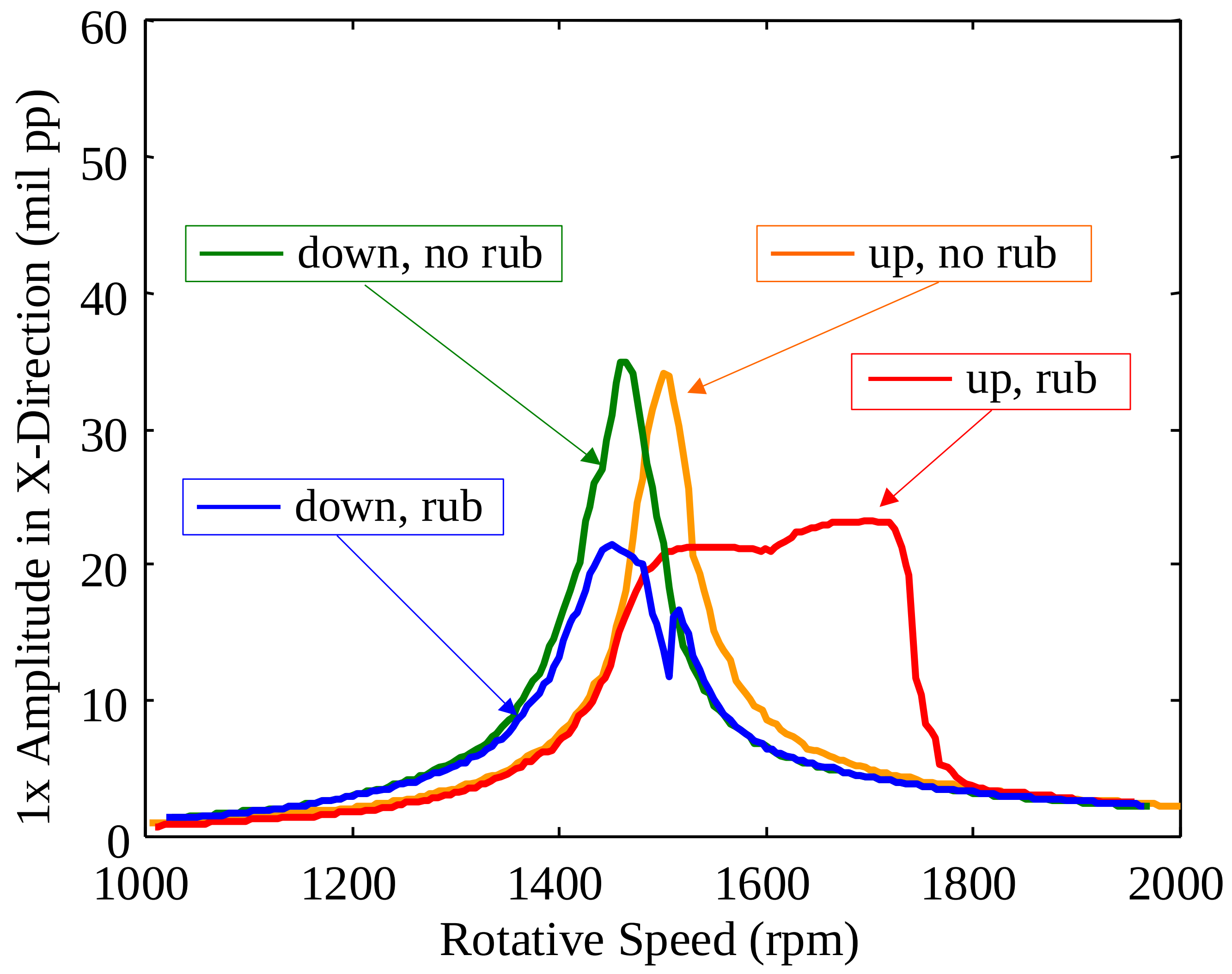
Forward full annular rub case on a rotor-kit





Comparison of 1X responses between rub and no-rub cases

Note:

- **Synchronous (1x) precession is dominant.**
- **1x phase angle changes smoothly during runup, quite differently compared with no-rub cases.**

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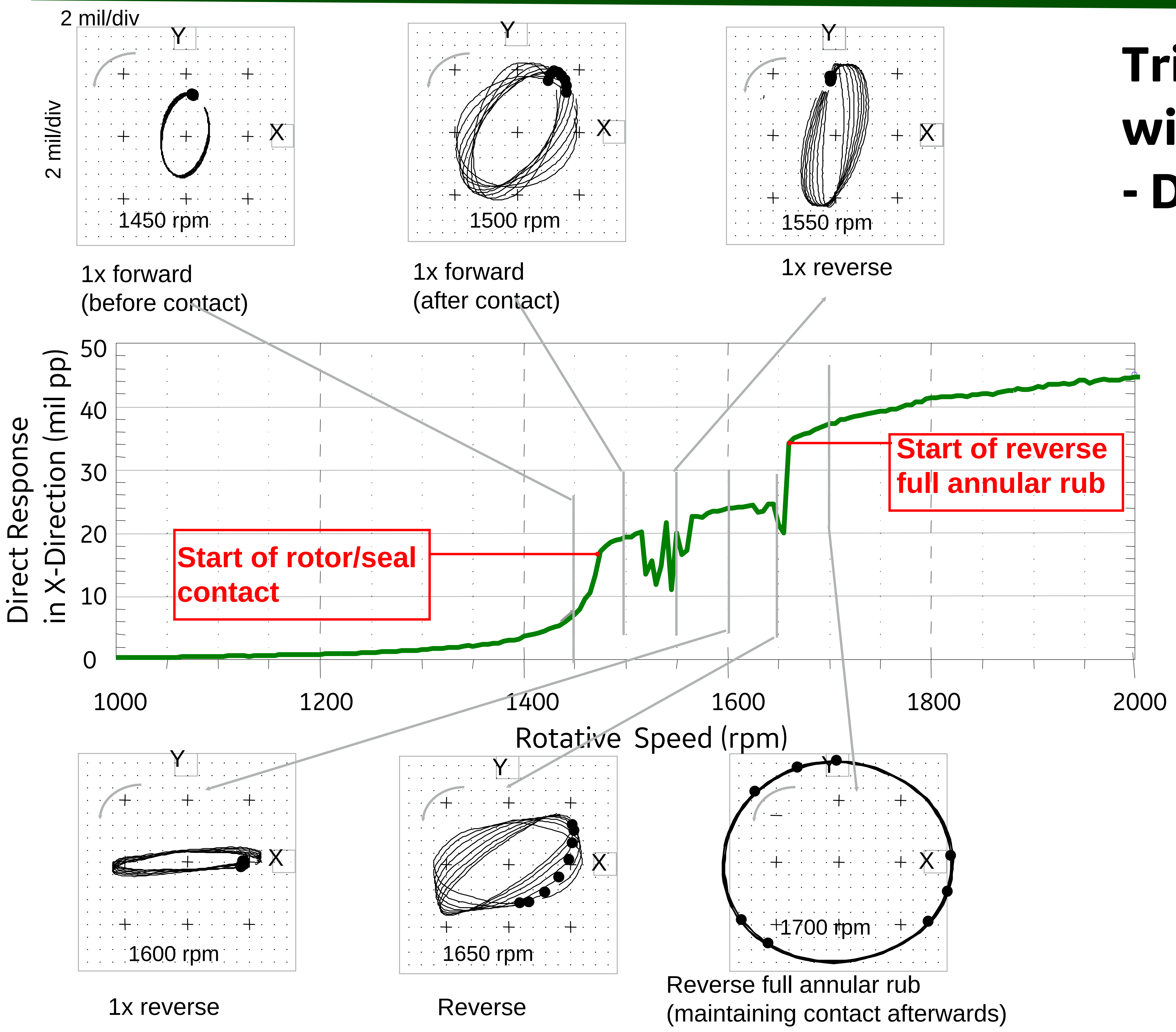
Reverse full annular rub case on a rotor-kit



Triggered during startup without outside disturbance
- Direct/orbit response

Note:

- **Three regions are distinguished during transition from contact initiation to reverse full annular rub:**

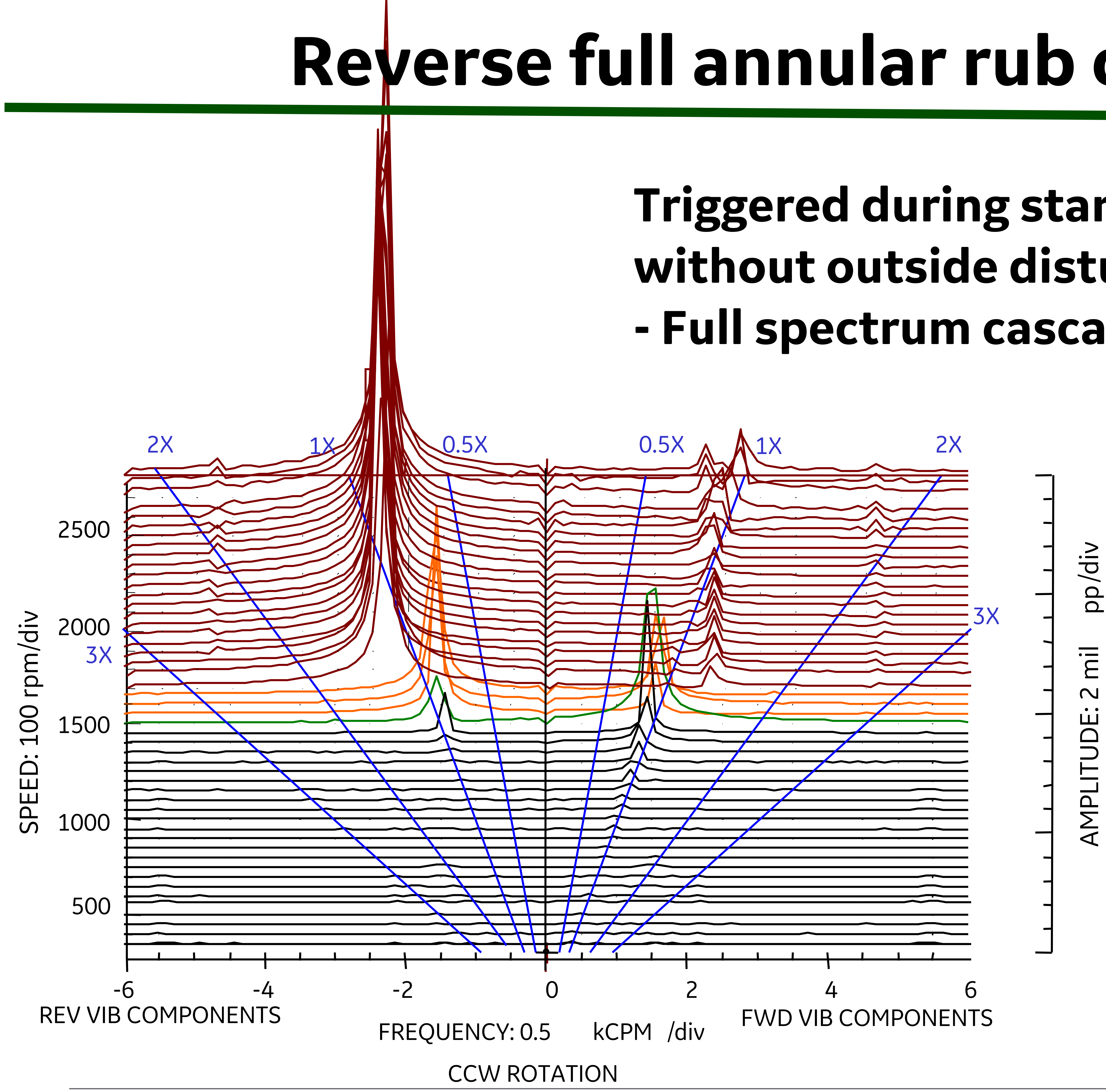
  (1) Dominant 1x forward

  (2) Dominant 1x reverse and then transitional speed-independent reverse

  (3) Reverse full annular rub with a fixed frequency

Reverse full annular rub case on a rotor-kit

Triggered during startup without outside disturbance - Full spectrum cascade



Note:

- **Synchronous 1x response component almost disappears during the full annular rub.**
- **Three speed regions during contact :**
  (1) Dominant 1x forward
  (2) Dominant 1x reverse and then speed-independent reverse
  (3) Reverse full annular rub with a fixed frequency (2300 cpm in this case)

Full Annular Rub, including both forward and reverse dry whirl/whip, are typically not seen in real machines.

The following papers can be viewed in *ASME* Journal of Engineering for Gas Turbines and Power for detailed information

Yu, J. J., Goldman, P., Bently, D.E., and Muszynska A., 2002, Rotor/Seal Experimental and Analytical Study on Full Annular Rub, ***Transactions of the American Society of Mechanical Engineers, Journal of Engineering for Gas Turbines and Power***, Vol.124, pp.340-350.

Yu, J. J., 2012, On Occurrence of Reverse Full Annular Rub, ***Transactions of the American Society of Mechanical Engineers, Journal of Engineering for Gas Turbines and Power***, Vol. 134, 012505.

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Morton Effect

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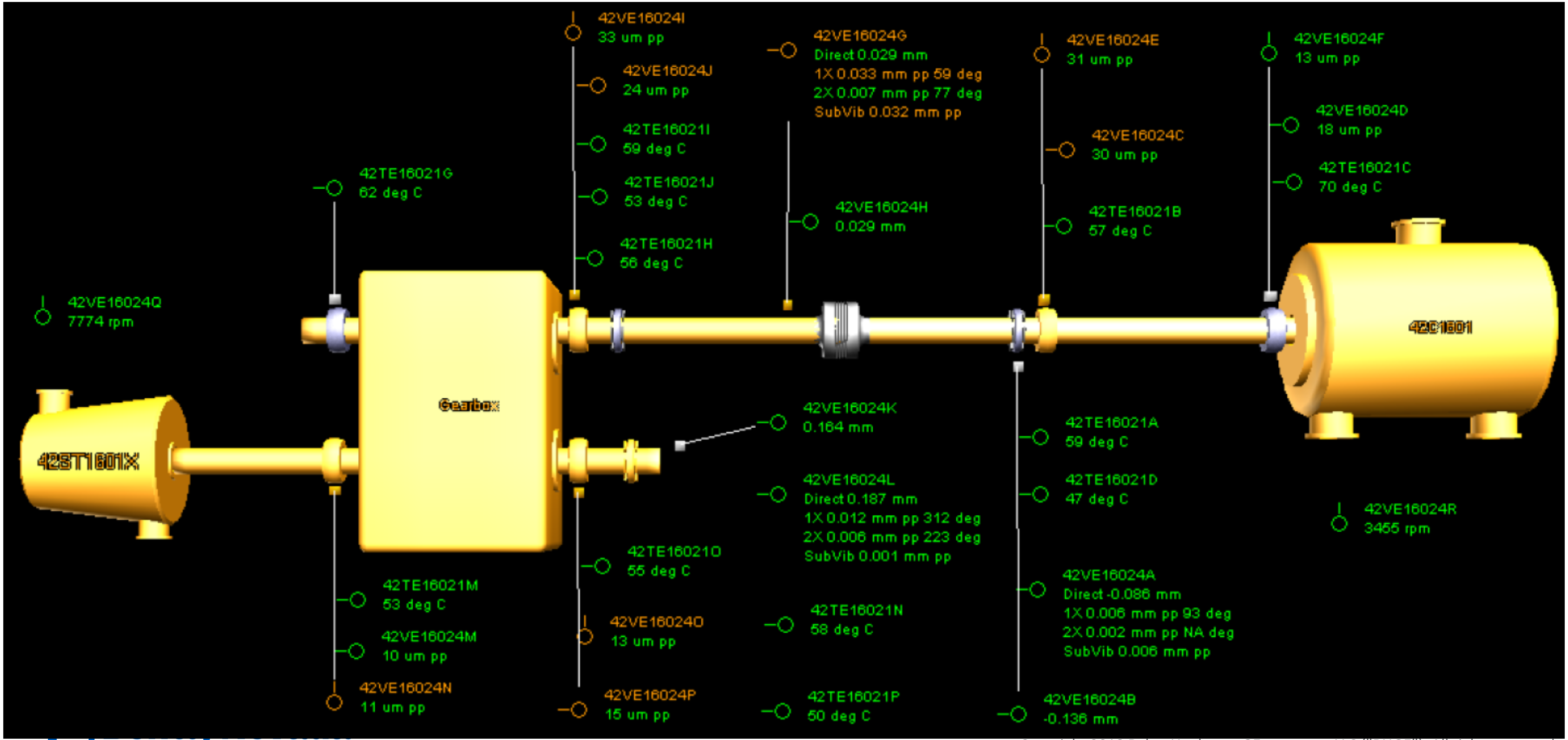
Morton Effect

- “Rubbing” on oil film.
- Looks like seal rub, but no rub sign on stationary parts.
- Often very smooth 1X rolling vector on polar plot. Looks like smooth orbit on polar plot.
- Typically on overhung compressor bearing

Morton Effect Case



Morton Effect Case

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Morton Effect Case

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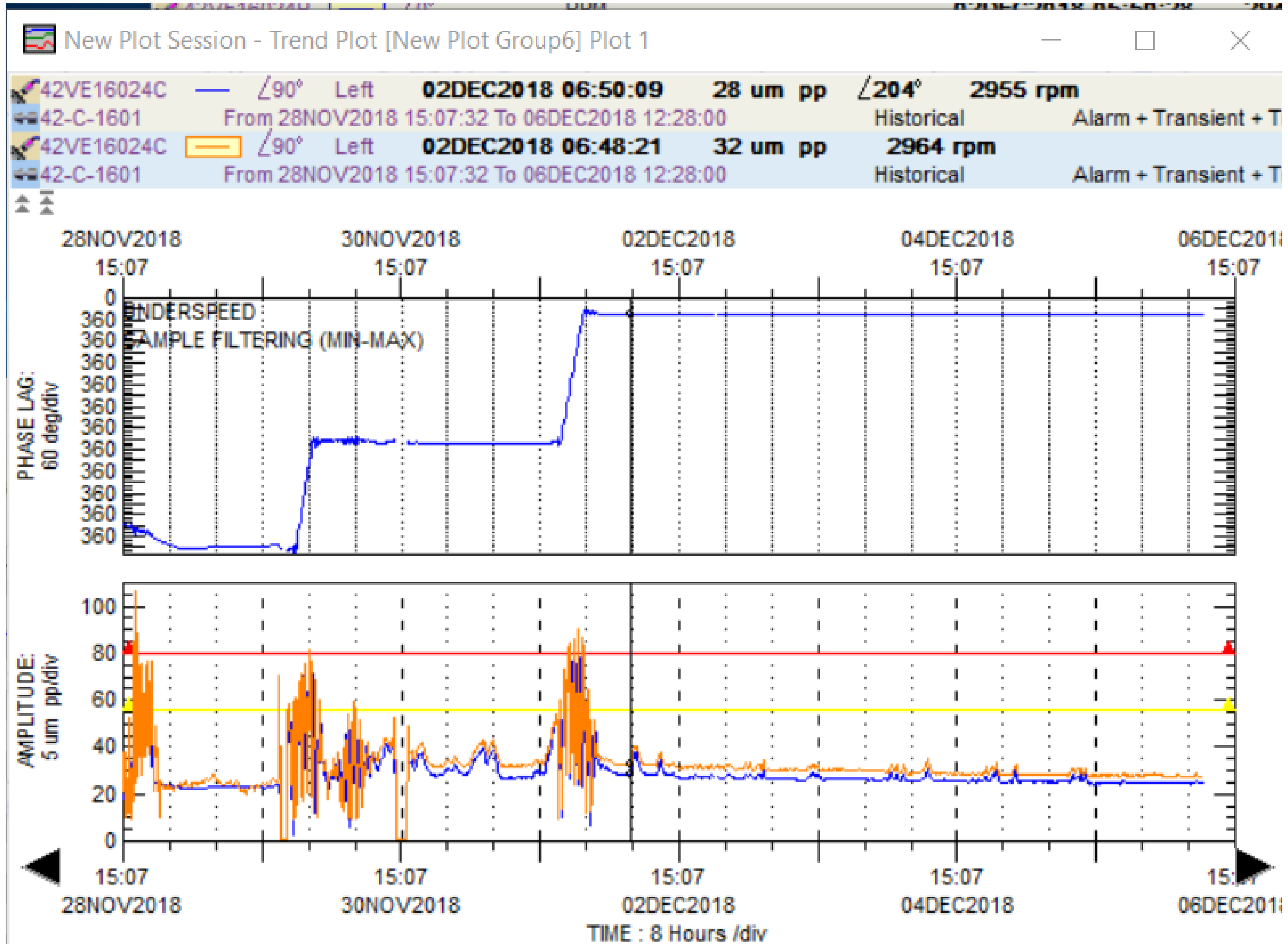
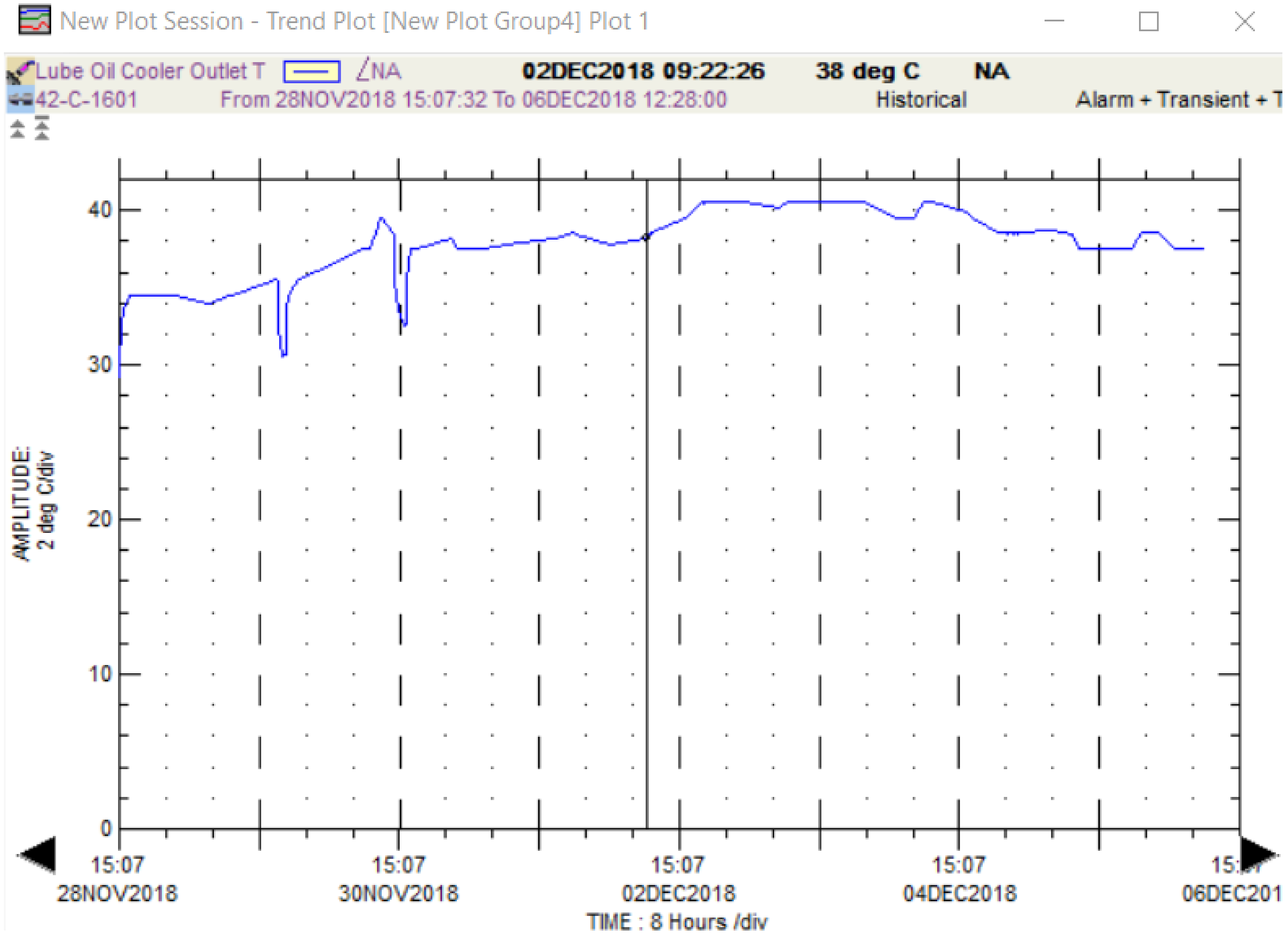
Morton Effect Case





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Parametric Excitation

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Introduction

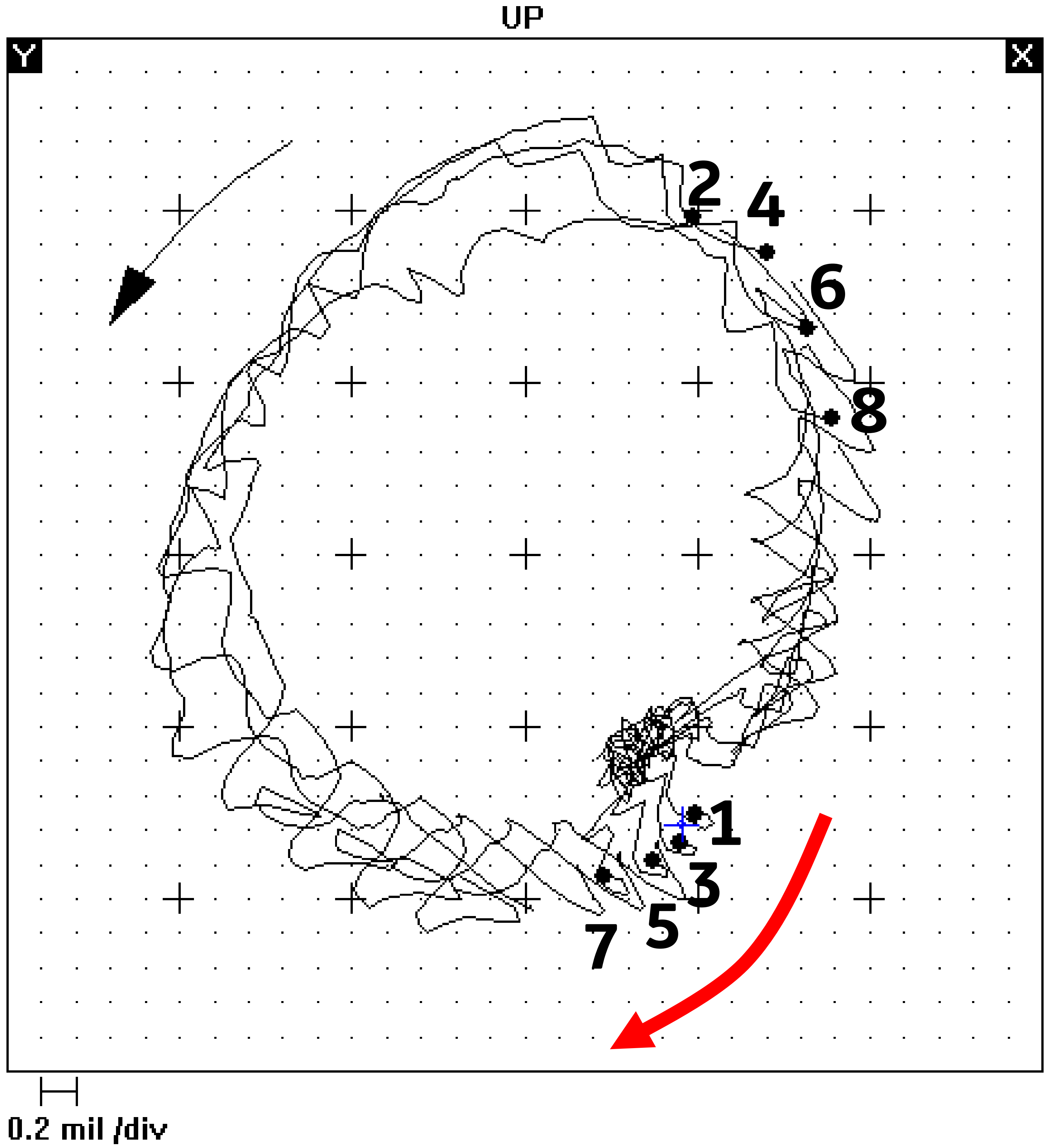
Subsynchronous vibration

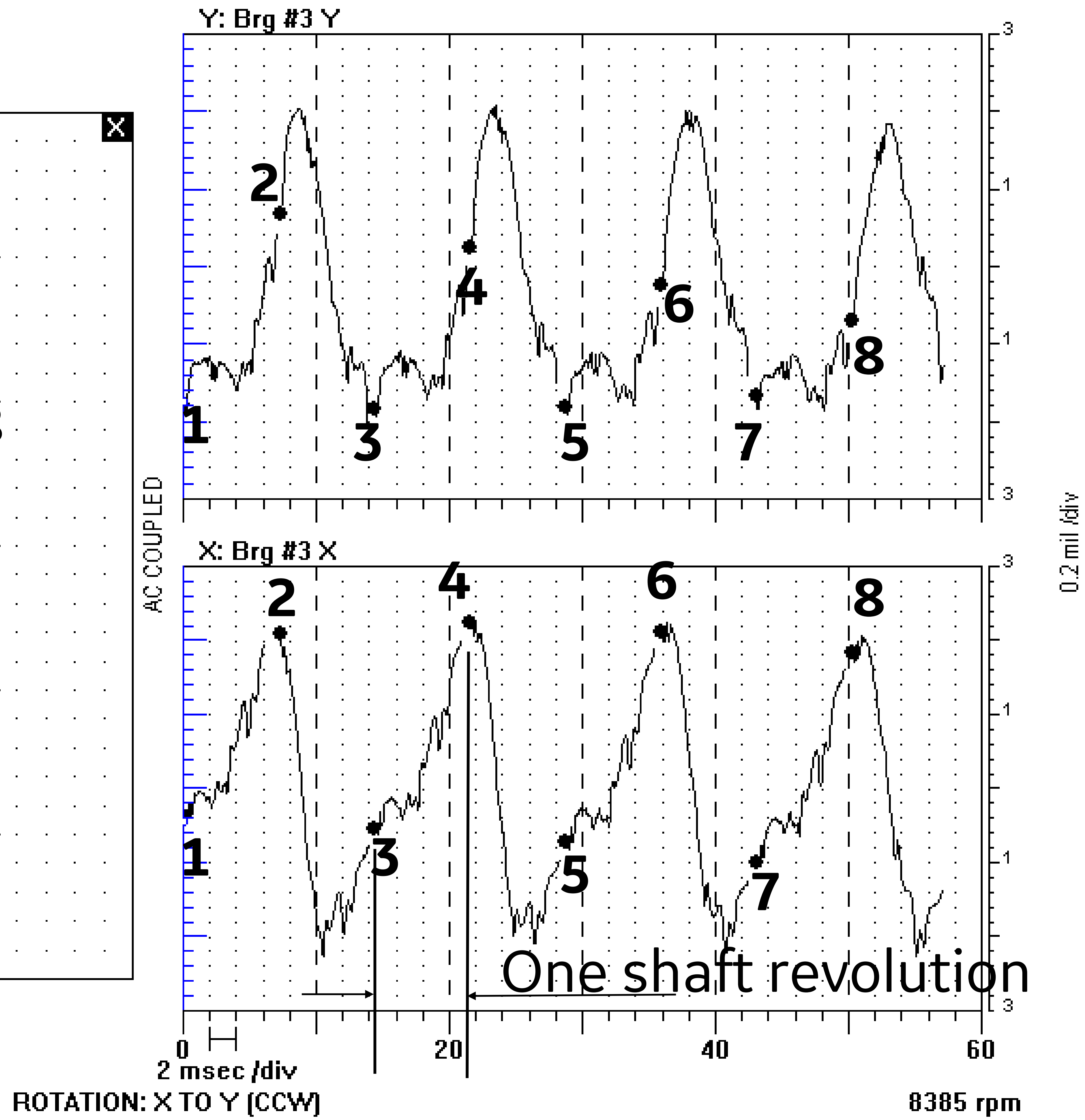
- Non-fractional (such as oil whirl, approx. 0.35X to 0.47 X)
- Fractional or subharmonic (such as ½X, ⅓X, ¼X, etc.)

- **These two types of vibration can be confused.**
- **Oil whirl, sometimes called *half-frequency whirl*, is not at fractional ½X frequency. It is typically below 0.5X.**
- **Though high resolution spectrum helps to determine the frequency (unable to zoom to “0” Hz), orbits with Keyphasor® dots, provide the best tool to distinguish between these two types of vibration.**

Introduction

0.47X (Oil Whirl)





Keyphasor dots are moving on the orbits

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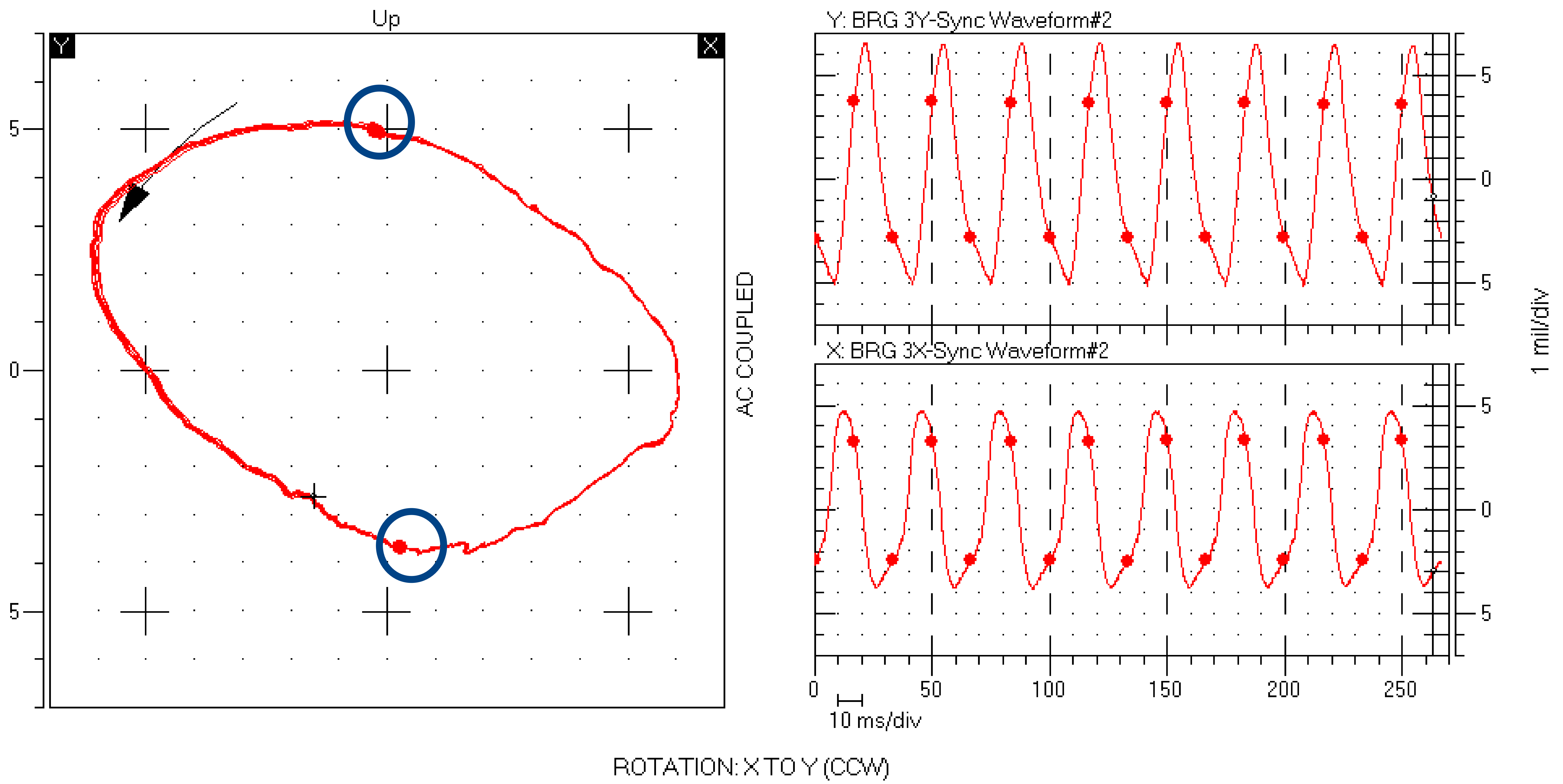
Introduction

½X Vibration



Keyphasor dots are locked on the orbits

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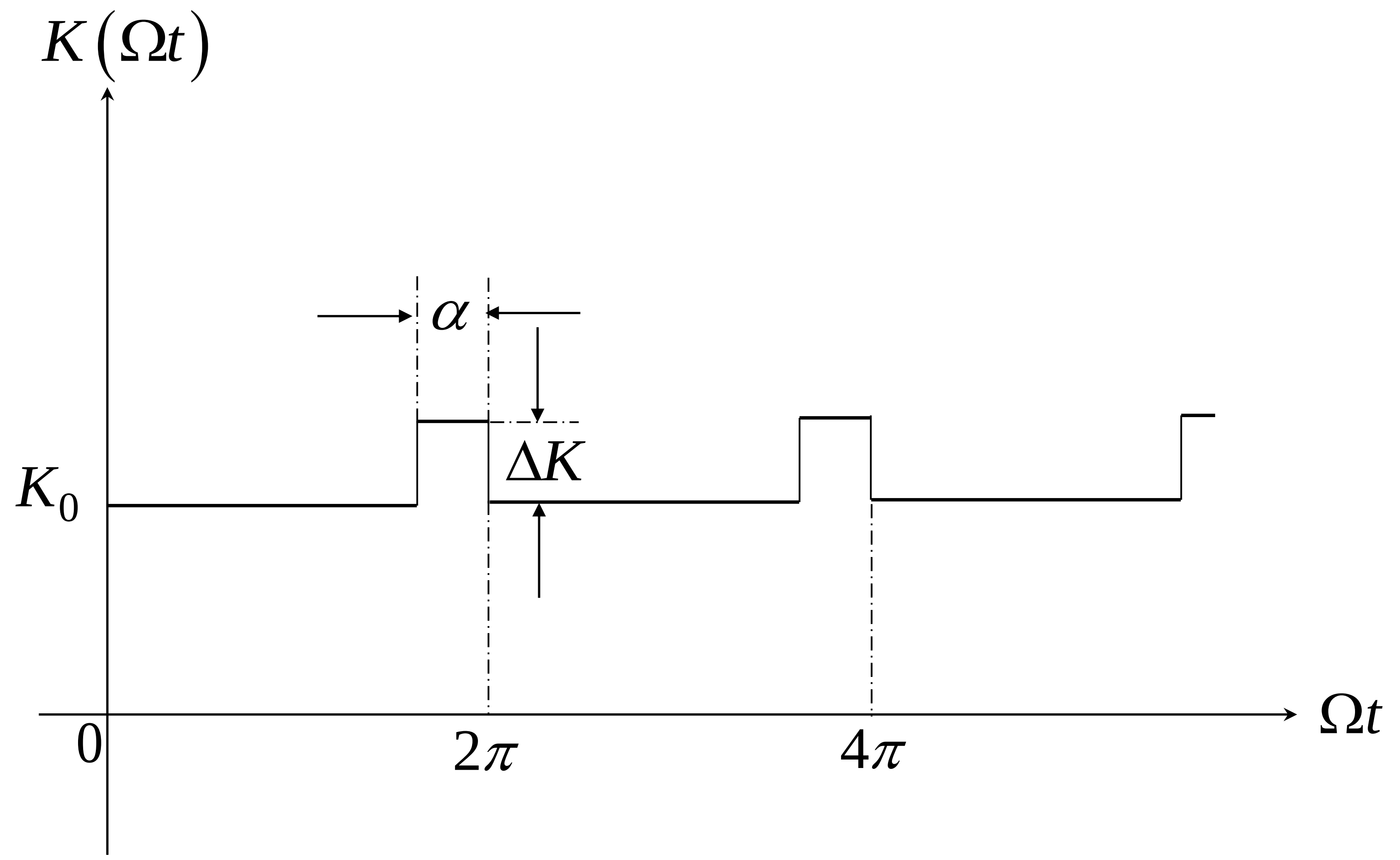
Theory

Equation of motion of the Jeffcort model:

$$M\frac{d^2x}{dt^2}+D\frac{dx}{dt}+K(\Omega t)x=mr\Omega^2\cos(\Omega t-\varphi)$$

If we have nonlinear step-changing stiffness $K(\Omega t)$ within each synchronous 1X vibration cycle



$$K(\Omega t)=K_0+\frac{\alpha\Delta K}{2\pi}+\frac{2\Delta K}{\pi}\sum_{n=1}^{\infty}\frac{\sin\frac{n\alpha}{2}}{n}\cos\left(n\Omega t+\frac{n\alpha}{2}\right)$$

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Theory

For the case n=1 (neglecting damping), the homogenous solution can be simplified into the Mathieu Equation

$$\frac{d^2x}{d\tau^2}+\left(\delta+2\varepsilon\cos 2\tau\right)x=0$$

where $\boxed{\tau=\frac{1}{2}\Omega t+\frac{1}{4}\alpha}$, $\delta=\frac{1}{\left(\frac{1}{2}\Omega\right)^2}\frac{K_0+\frac{\alpha\Delta K}{2\pi}}{M}$, and $\varepsilon=\frac{1}{\left(\frac{1}{2}\Omega\right)^2}\frac{\Delta K\sin\frac{\alpha}{2}}{\pi M}$

Thus, the principal instability region is approx. determined by

$$\left|\delta-1\right|<\left|\varepsilon\right|$$

and the unstable solution is dominantly composed of $\cos\tau$ and $\sin\tau$ terms.
This is exactly the ½X vibration.

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Theory

Assume α (range where ΔK occurs) is small. $\longrightarrow$ $\sin\frac{\alpha}{2} \approx \frac{\alpha}{2}$

Thus the unstable speed region due to step-changing stiffness:

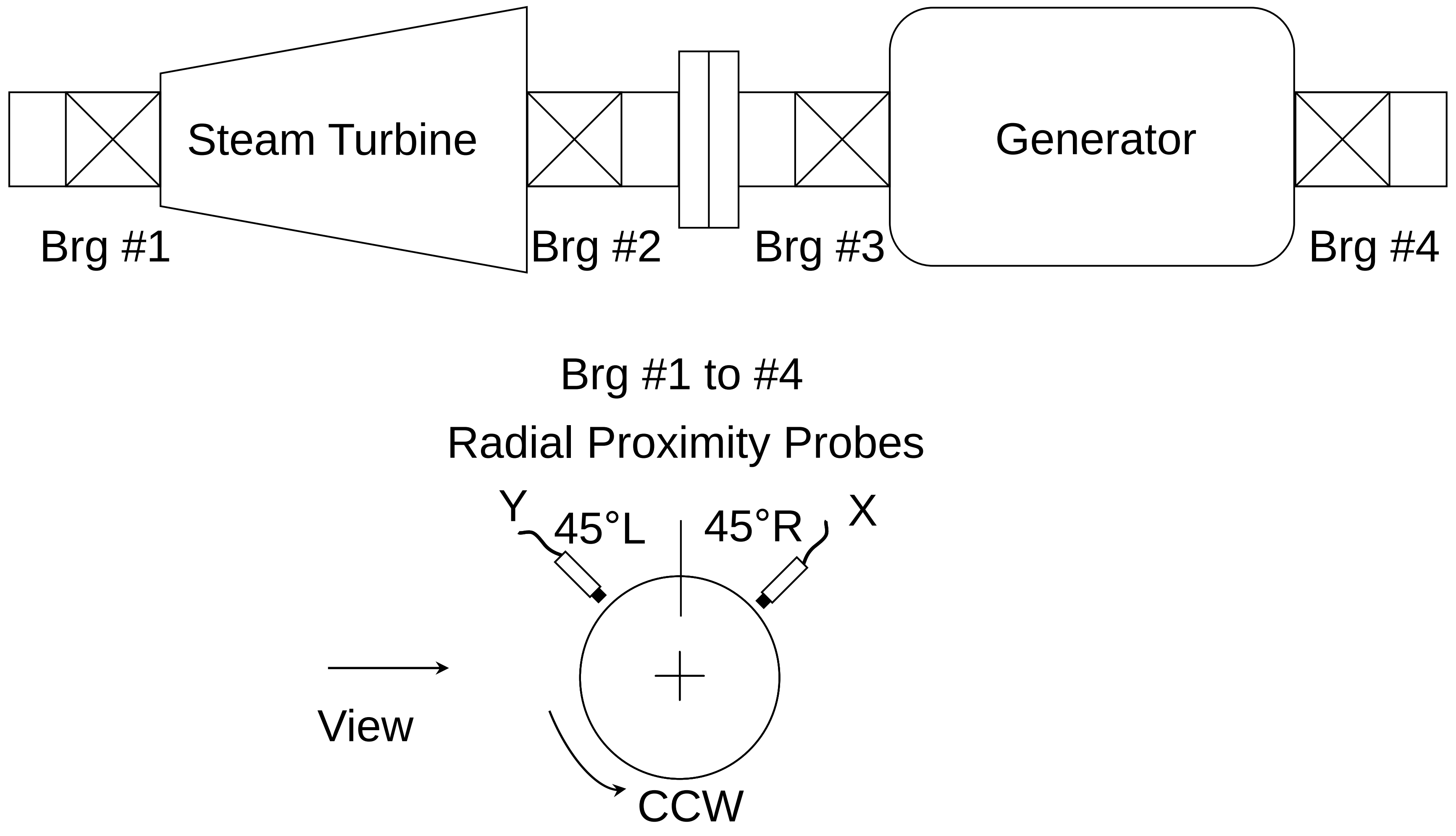
$$2\omega_n < \Omega < 2\omega_n\sqrt{1+\frac{\alpha\Delta K}{\pi K_0}}, \quad \text{for } \Delta K > 0$$ Normal-tight

and

$$2\omega_n\sqrt{1+\frac{\alpha\Delta K}{\pi K_0}} < \Omega < 2\omega_n, \quad \text{for } \Delta K < 0$$ Normal-loose

where $\omega_n = \sqrt{\frac{K_0}{M}}$

Case One



- 3600-rpm, 25 MW steam turbine generator.
- Journal bearings.
- Vibration excursion from approx. 2 mil pp (~50 µm pp) to 12 mil pp (~300 µm pp) at Bearing #3 at 2 MW, due to onset of ½X vibration.

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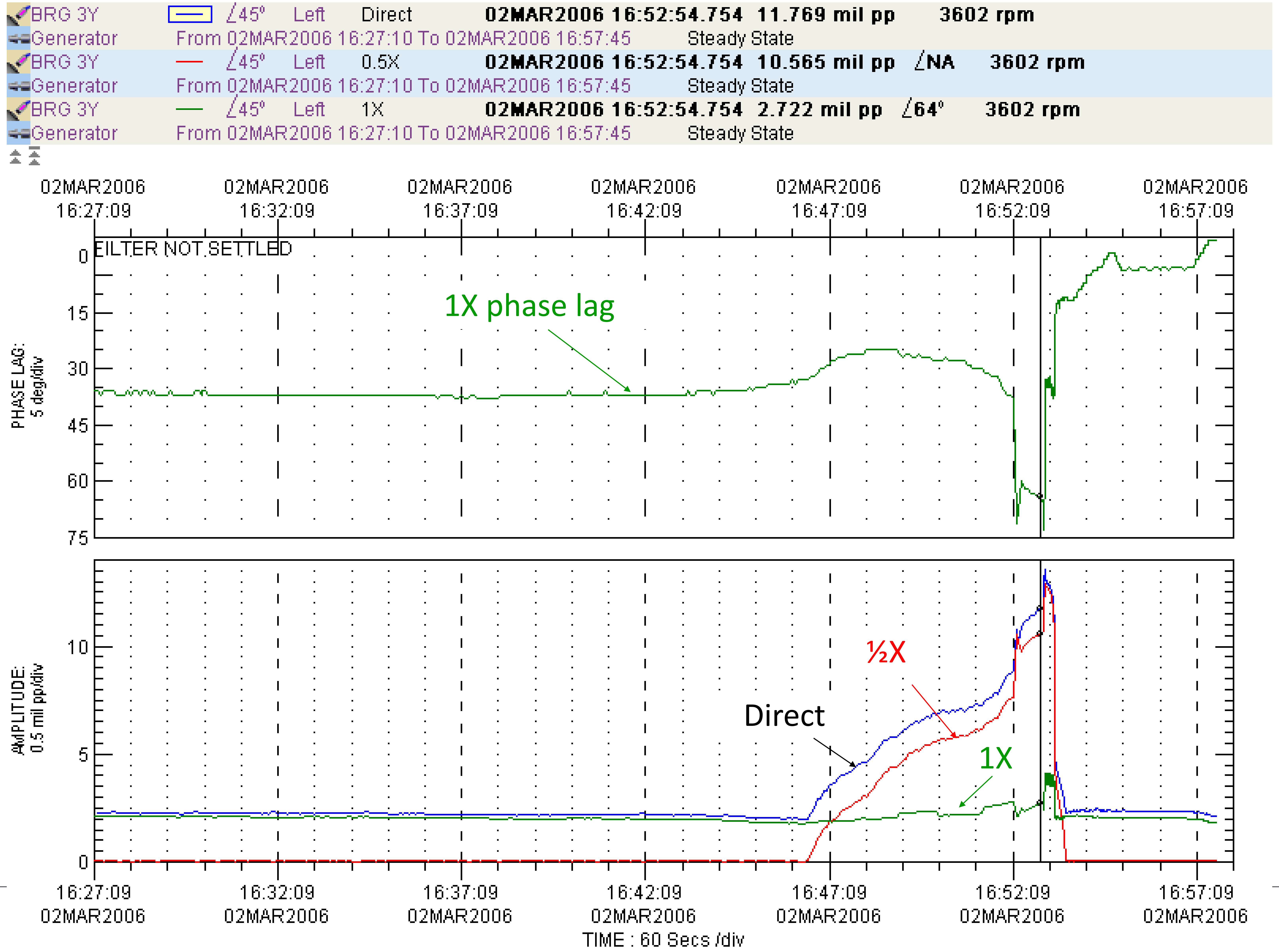
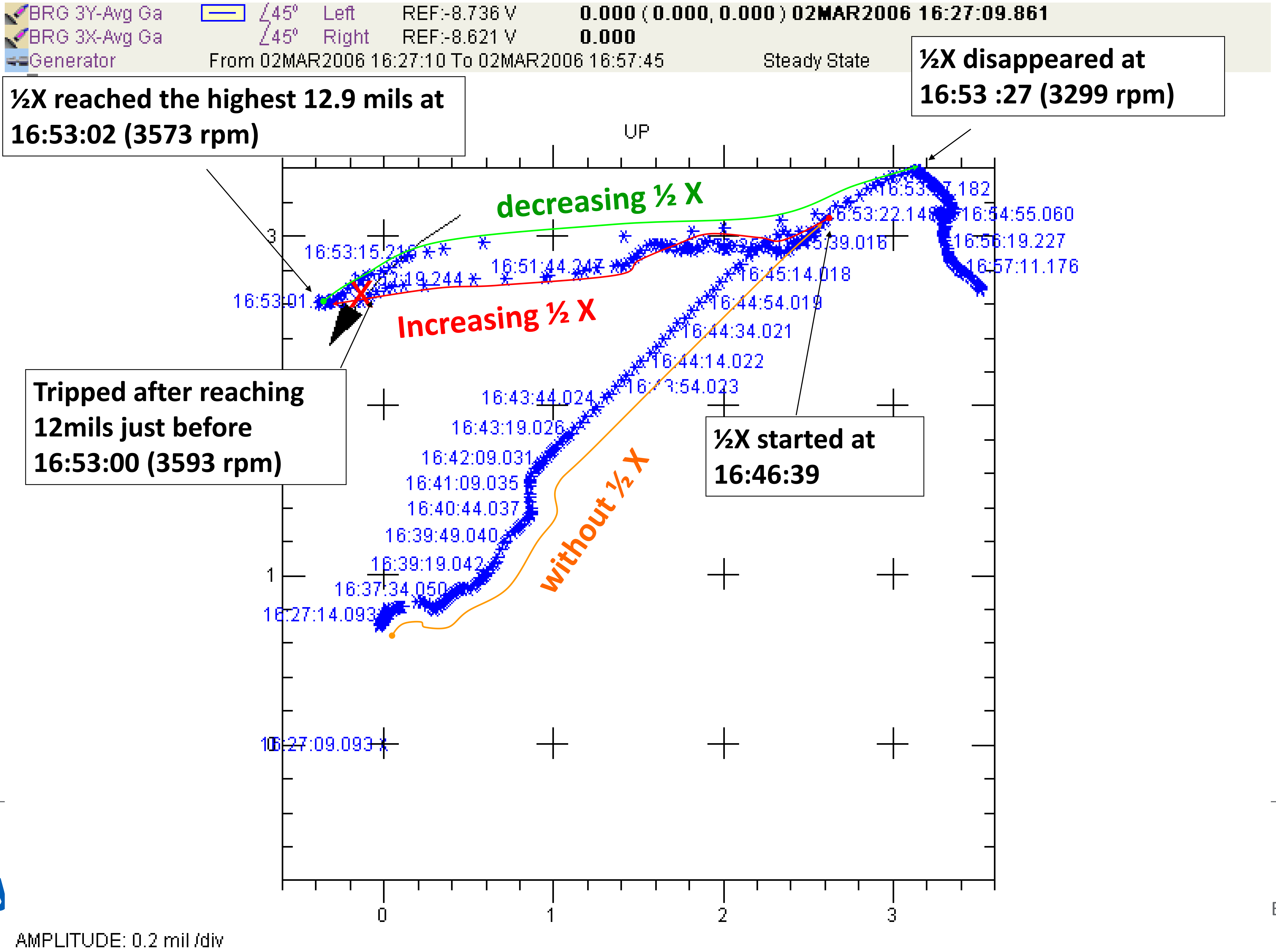
Case One

Trend plot at Bearing #3 Y-Probe



Case One

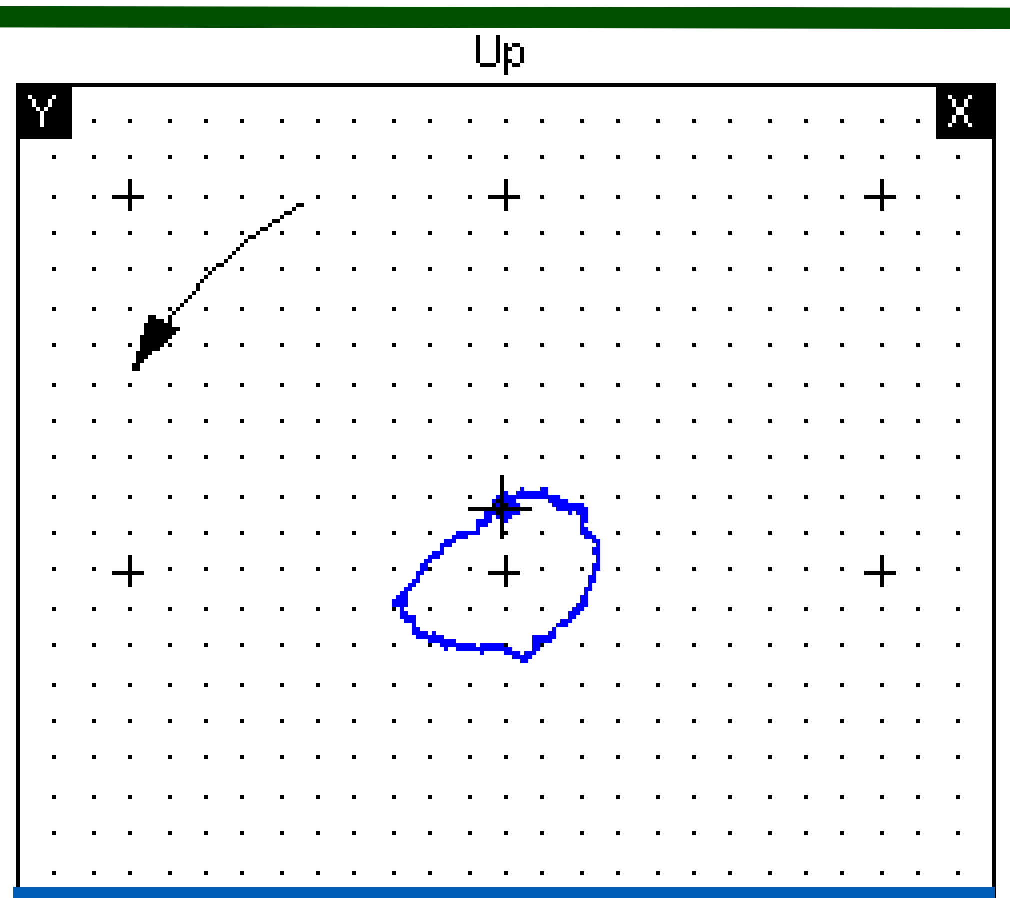
Shaft centerline plot at Bearing #3



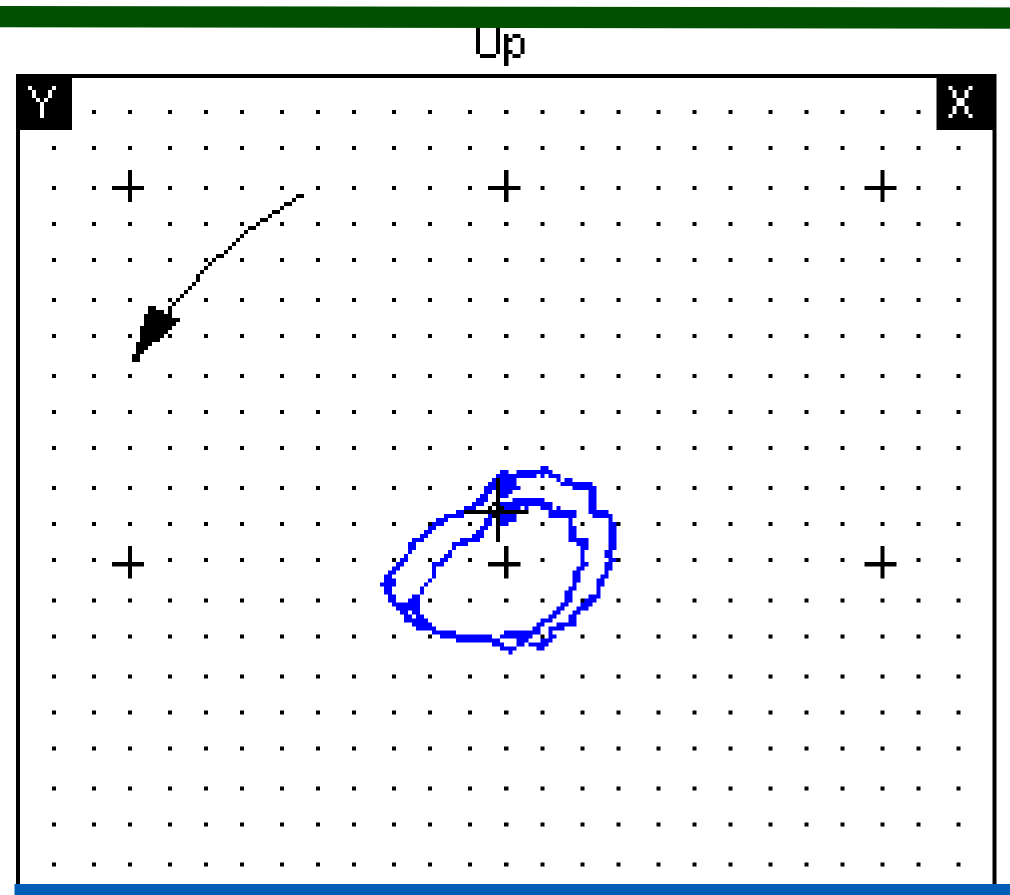
Case One

Orbit plots

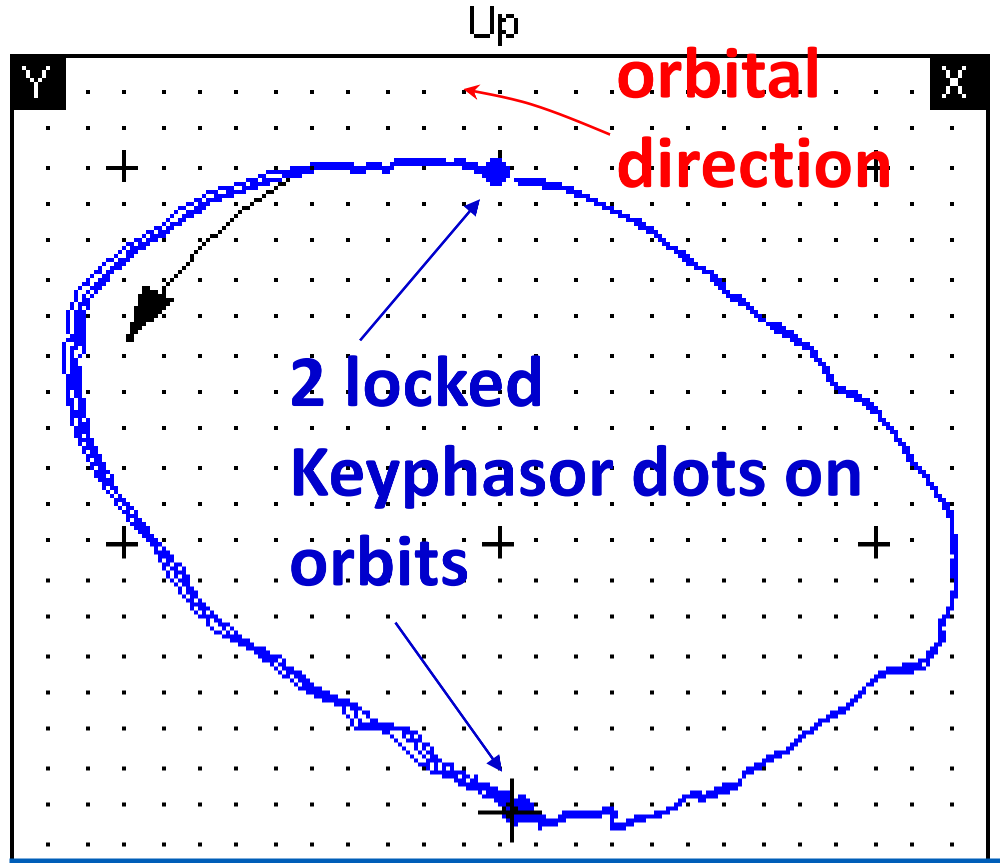
at Bearing #3



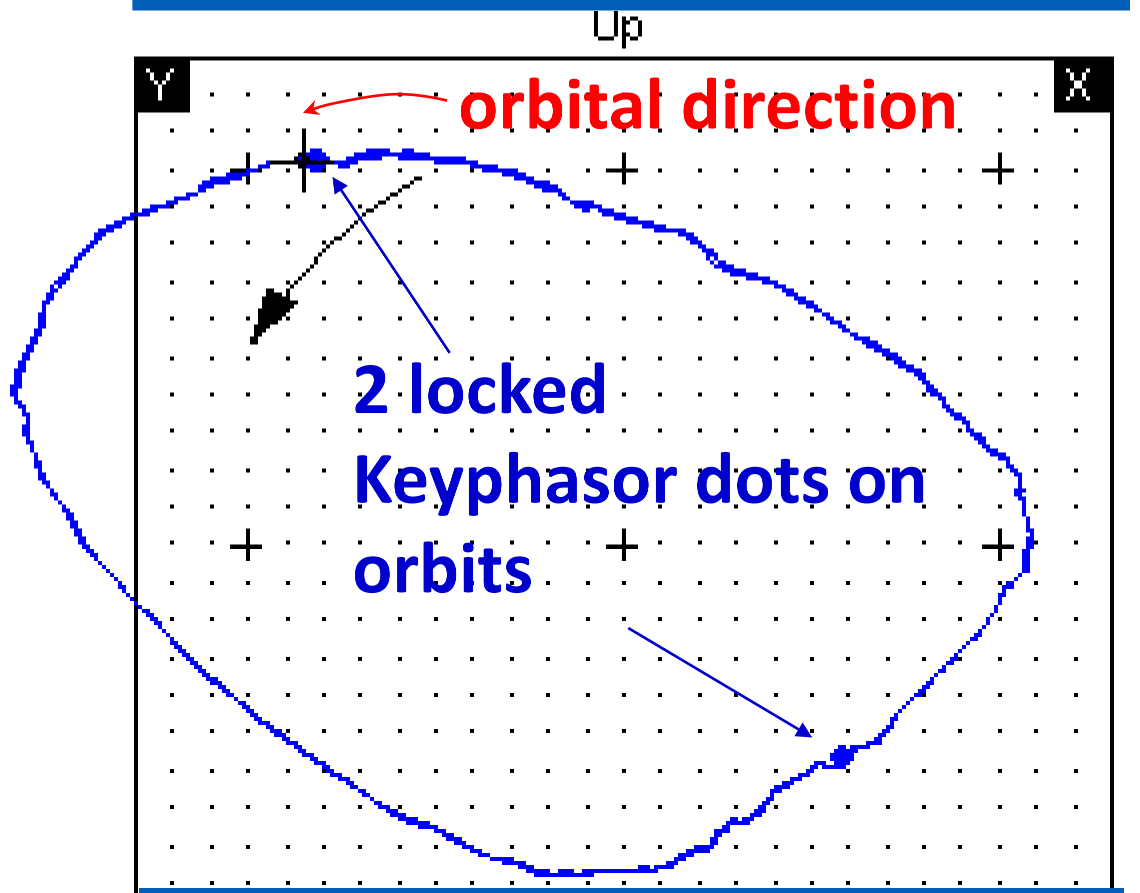
(a) Without ½X at 16:46:34



(b) Starting ½X at 16:46:39



(c) Full-blown ½X at 16:52:59



(d) Maintaining ½X at 3466 rpm at 16:53:14

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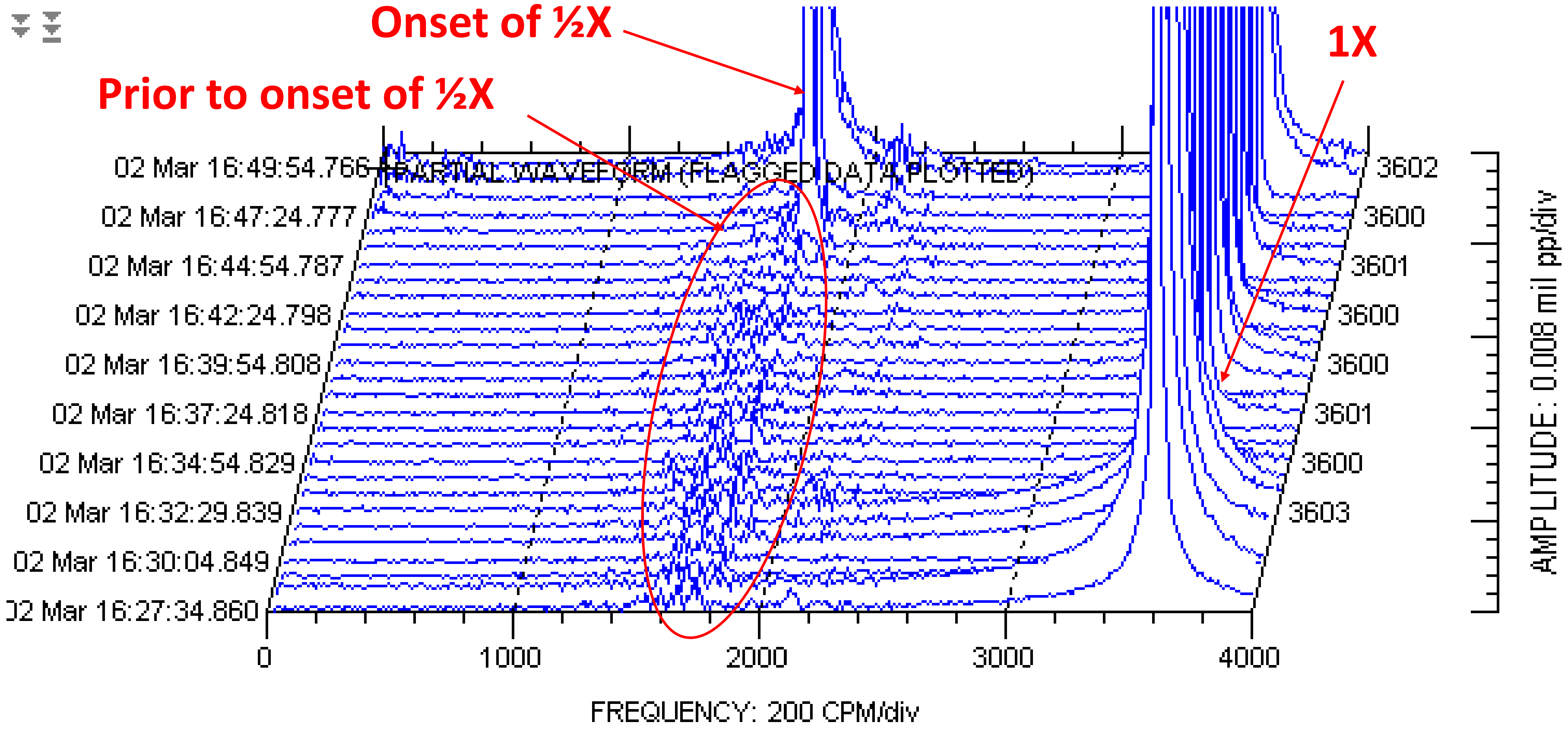
Case One

Waterfall plot from Bearing #3 Y-Probe



Indicative of natural frequency at near 1800 cpm or half of the running speed (most likely due to suppressed oil whip), prior to onset of ½X.

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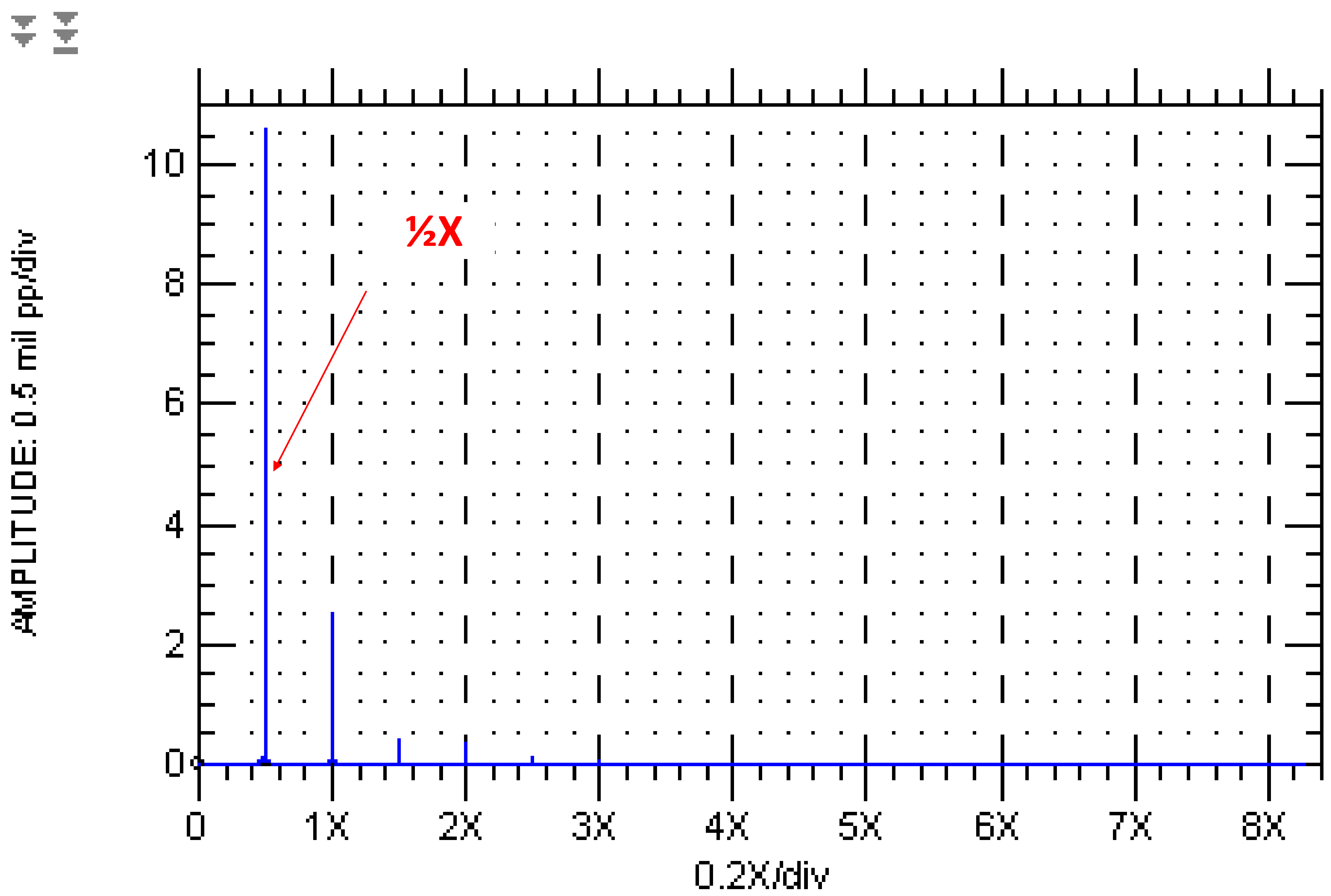
Case One

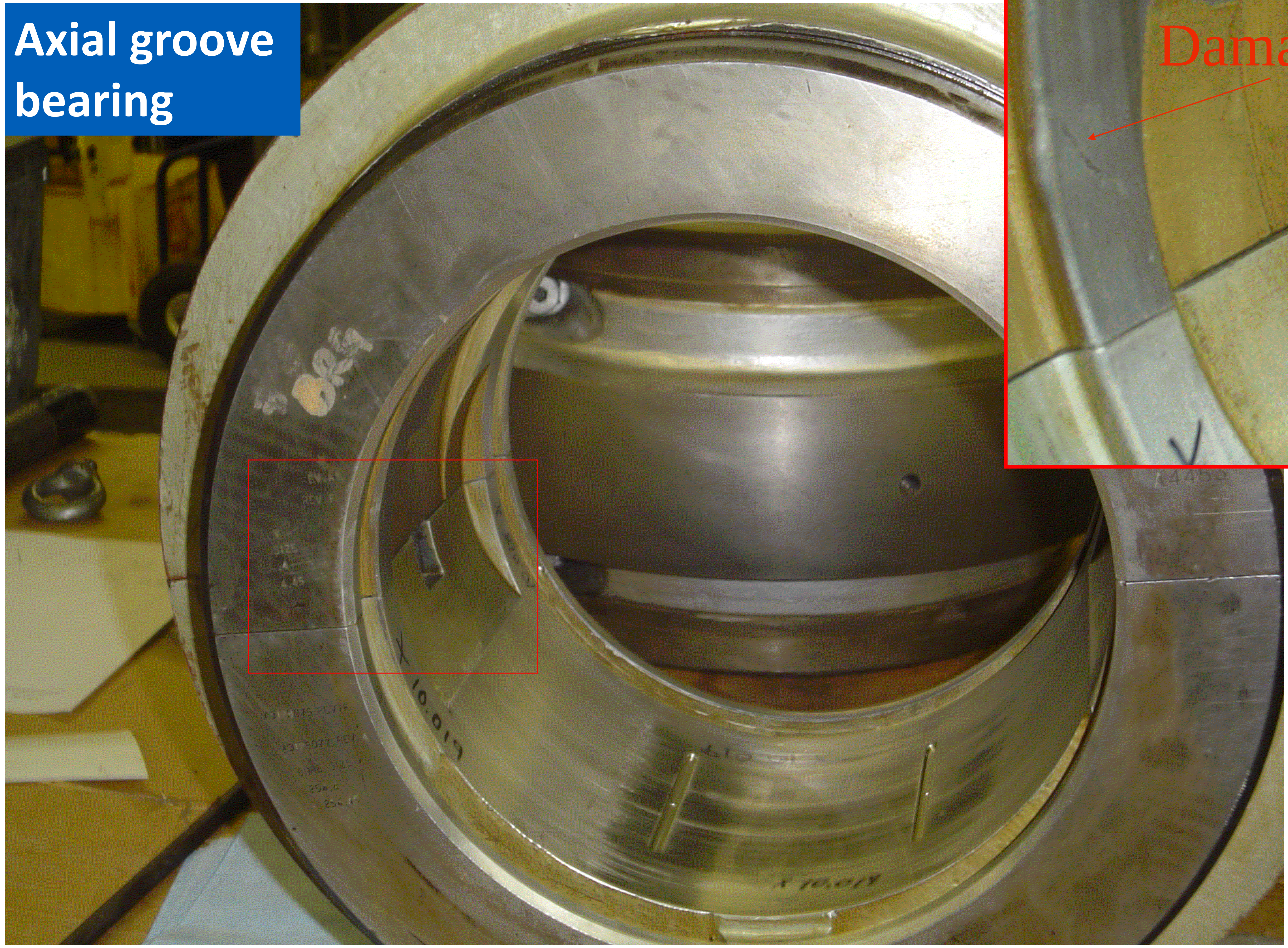
Spectrum plot from Bearing #3 Y-Probe



Primarily ½X component without significant harmonics

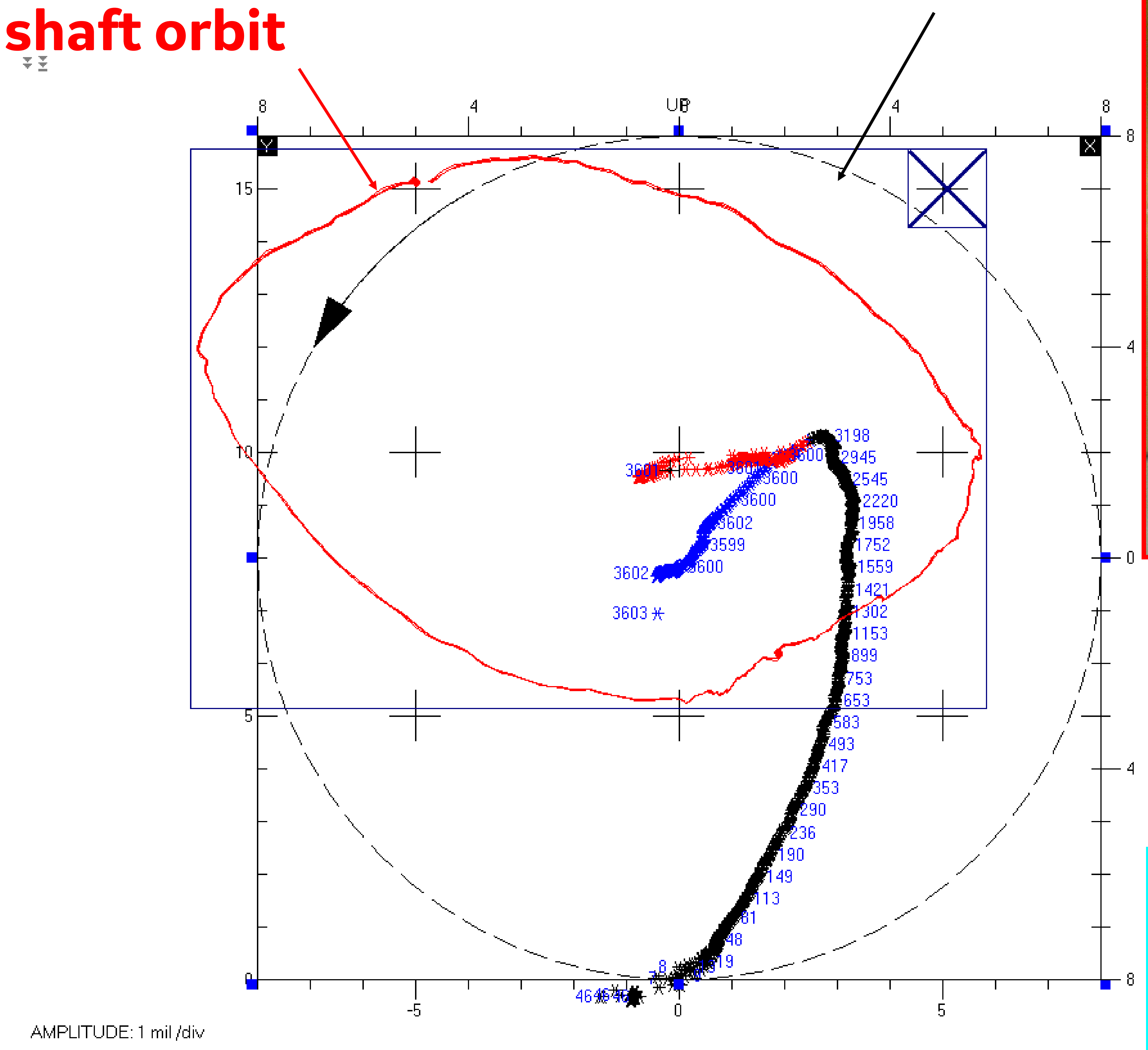
Case One

Inspection on Bearing #3



- Oversized bearing. Diametral clearance 0.005 inch (127 μm) higher than its spec
- Damage on the bearing surface.

Case One



High vibration damaged the bearing wall.

After replacing this oversized and damaged bearing with a new one, no ½X vibration has occurred since then.

There are more ½ X vibration cases in the following paper:

Yu, J. J., 2010, Onset of 1/2X Vibration and Its Prevention, ***Transactions of the American Society of Mechanical Engineers, Journal of Engineering for Gas Turbines and Power***, Vol. 132, 022502.

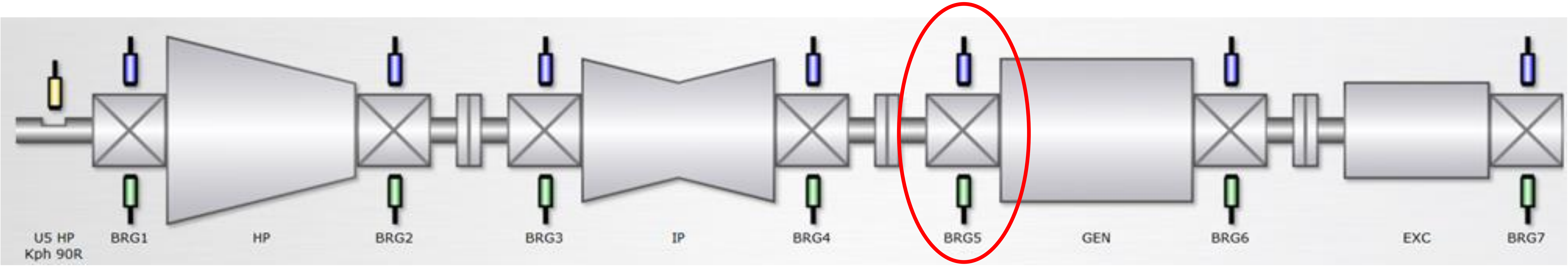
Case Two

- Cross-compound steam turbine generator unit with HP speed 3600 rpm, LP speed 1800 rpm, and rated output 750 MW.
- Vibration monitored by proximity probes at each bearing, plus seismic sensors. Data collection performed to support its startup after outage.

Case Two

Problem:

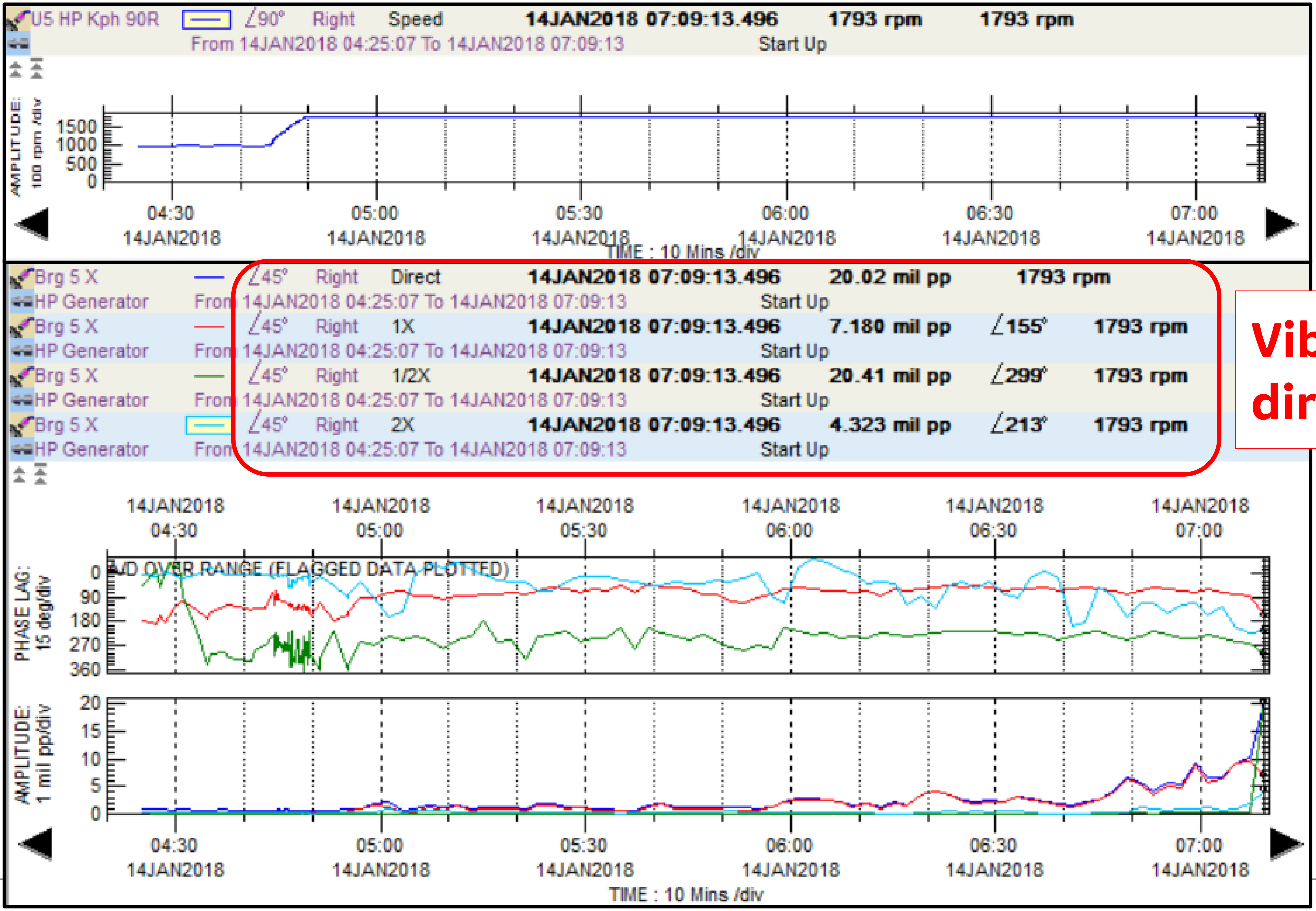
- During startup when the unit was staying at warmup speed of 1800 rpm, vibration reached 20 mil pp at HP generator drive end bearing (Brg#5) and therefore tripped the unit.
- Its root-cause needed to be determined as soon as possible to avoid any delay in unit startup.



Bently Nevada

Case Two

Trend plots



Vibration excursion in direct, 1X, ½X, and 2X

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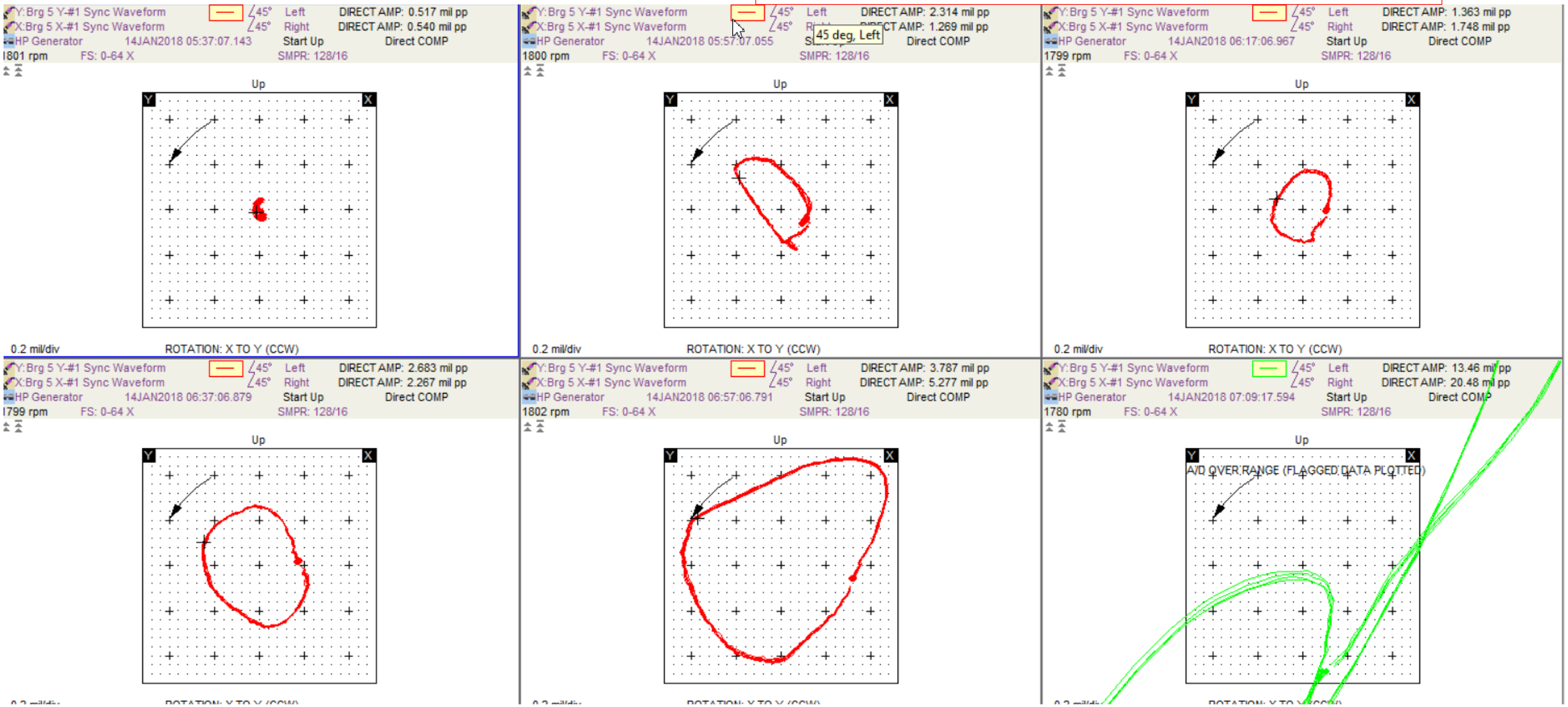
Case Two

Waveform-compensated orbits:

Changing with time at 1800 rpm



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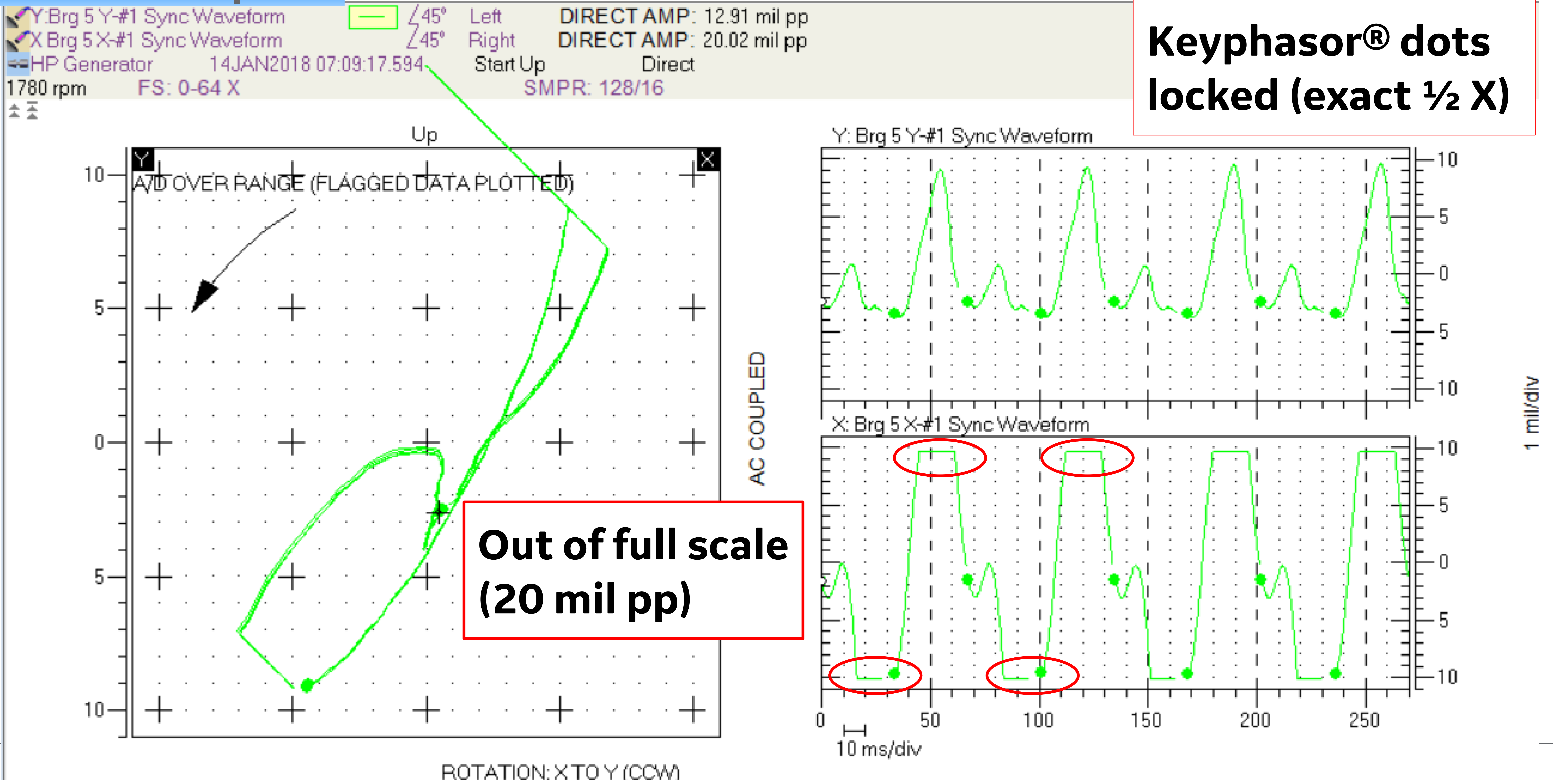
Case Two

Orbit/timebase plot



Keyphasor® dots locked (exact ½ X)

Out of full scale (20 mil pp)

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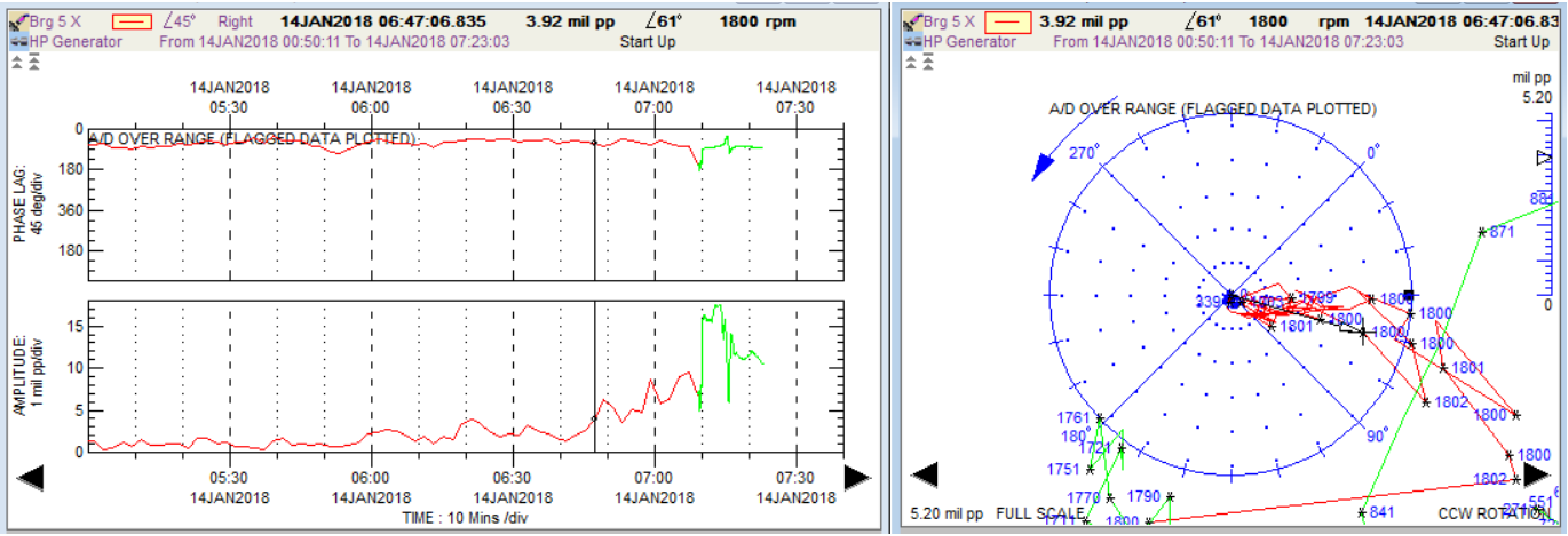
Case Two

Polar plot

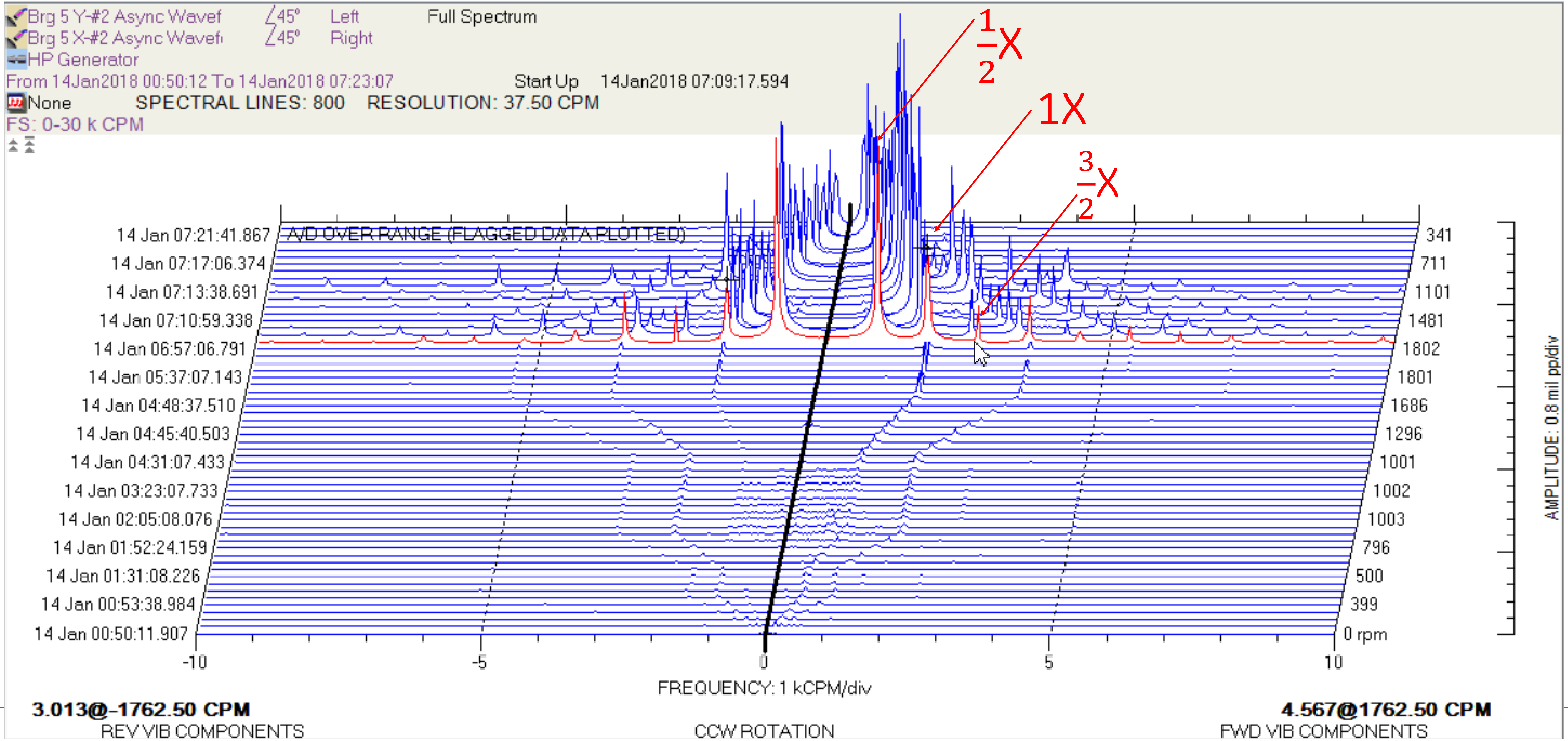
Changes in 1X vibration vector. Rub?



Case Two

Waterfall plot

Abnormal ½ X and its multiples during shutdown



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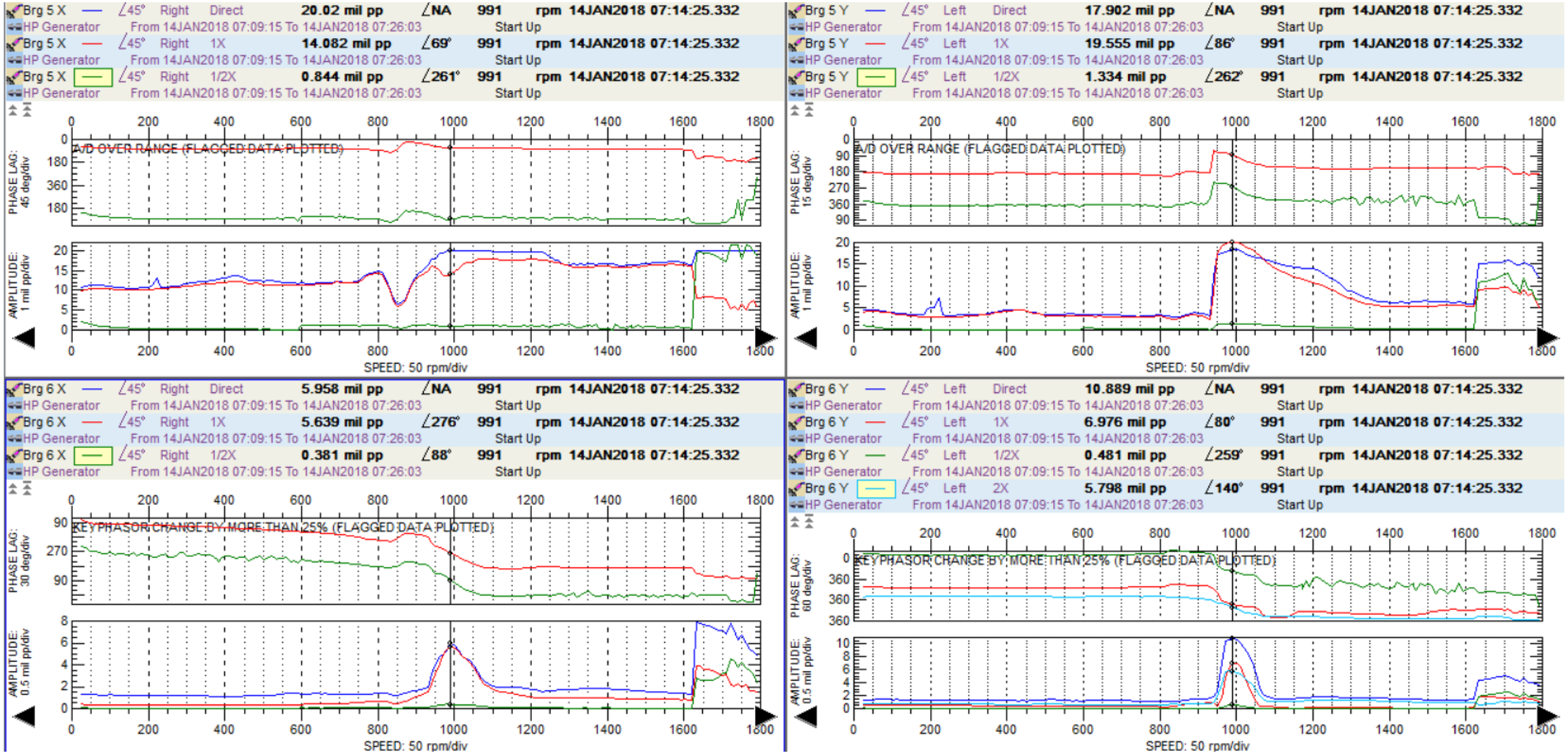
Case Two

Direct, 1X, and ½X bode plots plots

½X maintained until 1631 rpm during coast-down



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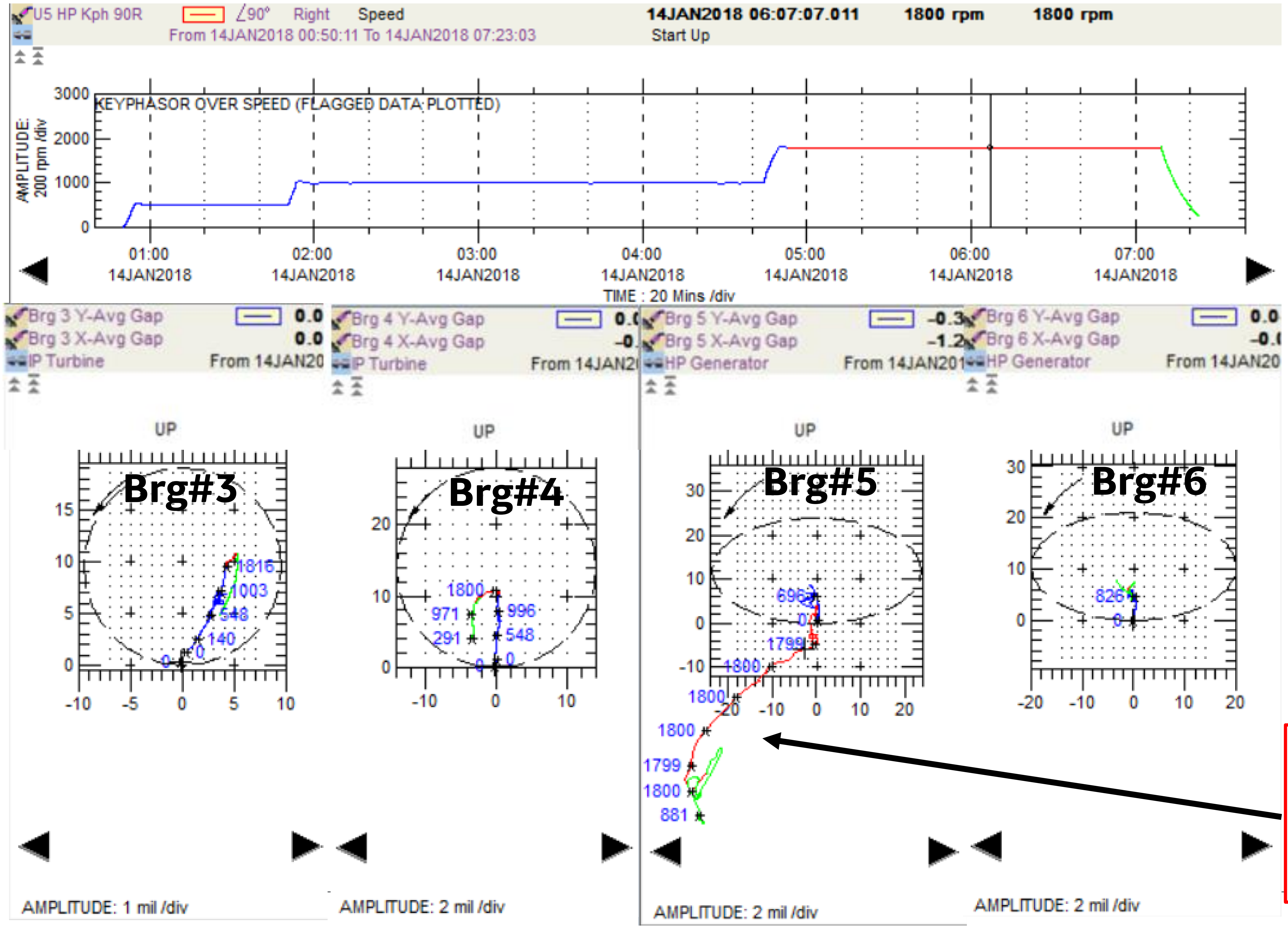
991 rpm as resonance or critical speed during coast-down

Case Two



Shaft centerline plots

Abnormal shaft centerline plot at Brg#5

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Root-cause of much higher vibration:

- **1X and ½X components ? – Yes**
  - 1X, followed by dominant ½X (20 mil pp),thus tripping the unit. ½X maintained above 1631 rpm during shutdown
- **Shaft bow ? – Yes**
  - High runout at low speed during shutdown (10 mil pp)
- **Change in bearing condition ? – Likely at Brg#5**
  - Normal shaft centerline plots except at Brg#5

Analysis

Root-cause of much higher vibration (Cont.):

- **Rub? – Yes**
  - 1X vectors changed at constant speed
  - Much higher runout during shutdown, indicative of shaft bow
  - Distorted orbits at Brg#5
  - Likely due to changes in shaft centerline position
- **What caused ½X vibration ? – oversized bearing**
  - ½ X occurred at 1631 to 1800 rpm during shutdown. The first critical speed is 991 rpm. Thus, the ½ X frequency was at 815-900 cpm < 991 cpm. Normal-tight or normal loose?

Bently Nevada

Root-cause of much higher vibration (Cont.):

- **Bearing damage? – Most likely**
  - At constant speed of 1800 rpm and during coast-down, shaft centerline position was far beyond the bearing clearance boundary at the bottom at Brg#5.
  - Unfortunately, bearing metal temperature reading was invalid due to the sensor issue.

Case Two

Conclusions

- Brg#5 appeared to have been damaged based on shaft centerline position.
- Rub seemed to occur, causing varying 1X vibration and distorted orbits, plus significant shaft bow during shutdown.
- ½X vibration occurred due to oversized bearing (normal-loose).

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Case Two

Recommendations:

- Open Brg#5 for inspection
- Check rub locations

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Case Two

Inspection & Findings :

- The findings indeed indicated Brg #5 bearing wiped, mostly at the left bottom with additional 42 mils clearance due to wear beyond as-left clearance of 24 mils in vertical direction, matching the diagnosis. Obviously the wear was due to journal rubbing against the babbitt surface.

Babbitt material transferred to

e to
generator

Babbitt wear at the left bottom

Fracture due to heat

Bently Nevada

Case Two

Inspection & Findings :

- It was found that a fine strainer was mistakenly left, causing oil reduction and starvation, and finally wiping the bearing. Bearing was shipped offsite to be re-spun.
- Rubs occurred on inner and outer oil deflectors at Brg #5 as well as that at adjacent Brg #4 generator side. All three were shipped offsite for teeth replacement.
- Brg#5 H_2 seal casing oil deflector rubbed and replaced with new one.
- Adjacent Brg #4 turbine side oil deflector clearance and alignment acceptable.

Case Two

Inspection & Findings :

- No damage was observed on Brg#5 journal surface, and its dimensions were verified to be acceptable.

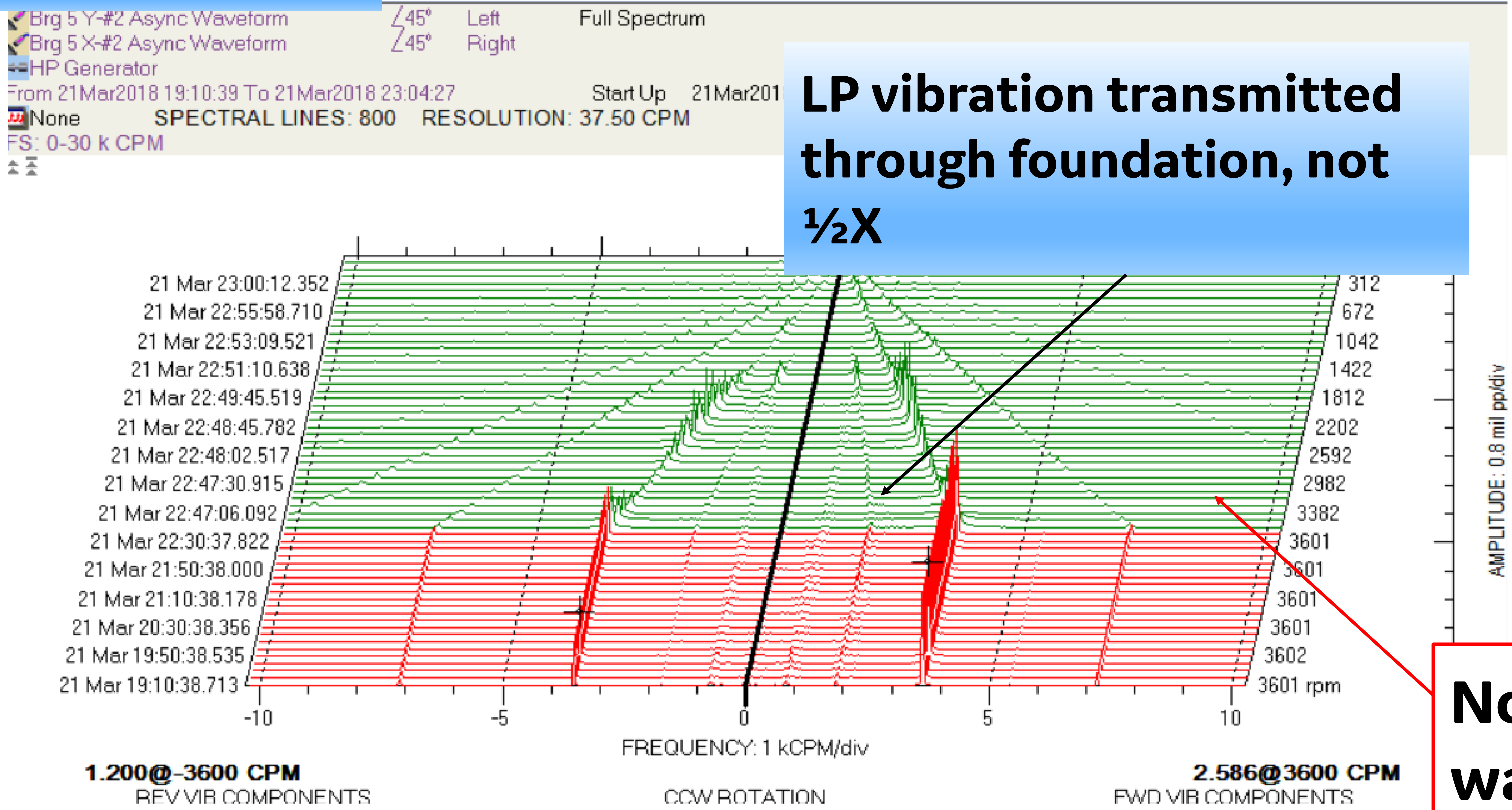
Case Two

Resolution:

- The bearing was repaired and re-installed correctly.
- Damaged oil deflectors were replaced with new ones.

Case Two

Final vibrations:



LP vibration transmitted through foundation, not ½X

Normal full-spectrum waterfall plot at Brg#5

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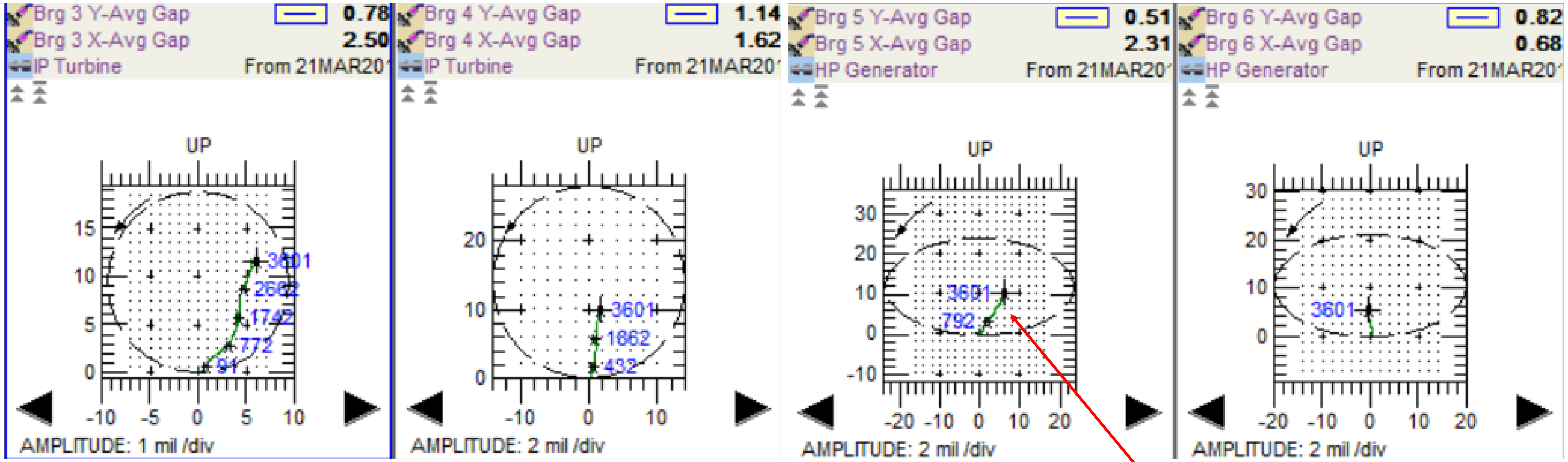
Case Two

Final shaft centreline plots:



Normal shaft centerline plot at Brg#5

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Case Two

Lessons Learned:

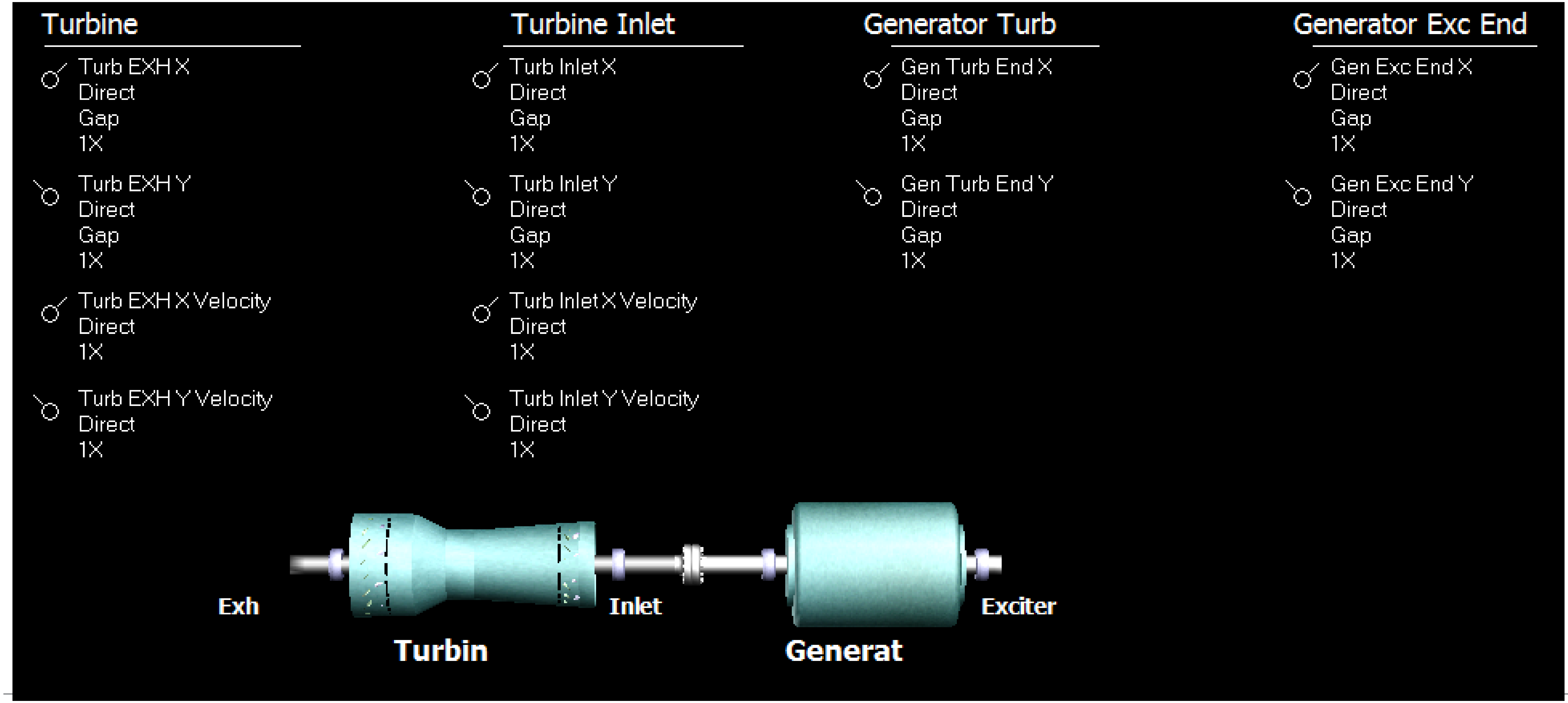
- Correct assembly after outage should be ensured to avoid costly repairs. All important sensors should be ensured working properly.
- The ½X vibration up to 20 mil pp was due to the increase in bearing clearance, not simultaneously due to a rub against the bearing wall.
- However, rubs did occur, which first damaged bearing babbitt surface and resulted in a change in shaft centerline position. This change caused the shaft to rub against oil deflectors. All these rubs were indicated by 1X vibration excursions.
- Shaft centerline is one of the most important plots in machinery diagnostics. It was used to diagnose bearing damage correctly when bearing metal temperature reading was invalid.
- Had the vibration issue be simply treated as rub, followed by re-start without in-depth data analyses and actions, further catastrophic damages would have occurred.

Shaft Crack Diagnostics
-1X & 2X vibration monitoring

Bently Nevada

Shaft crack case

Machine train diagram



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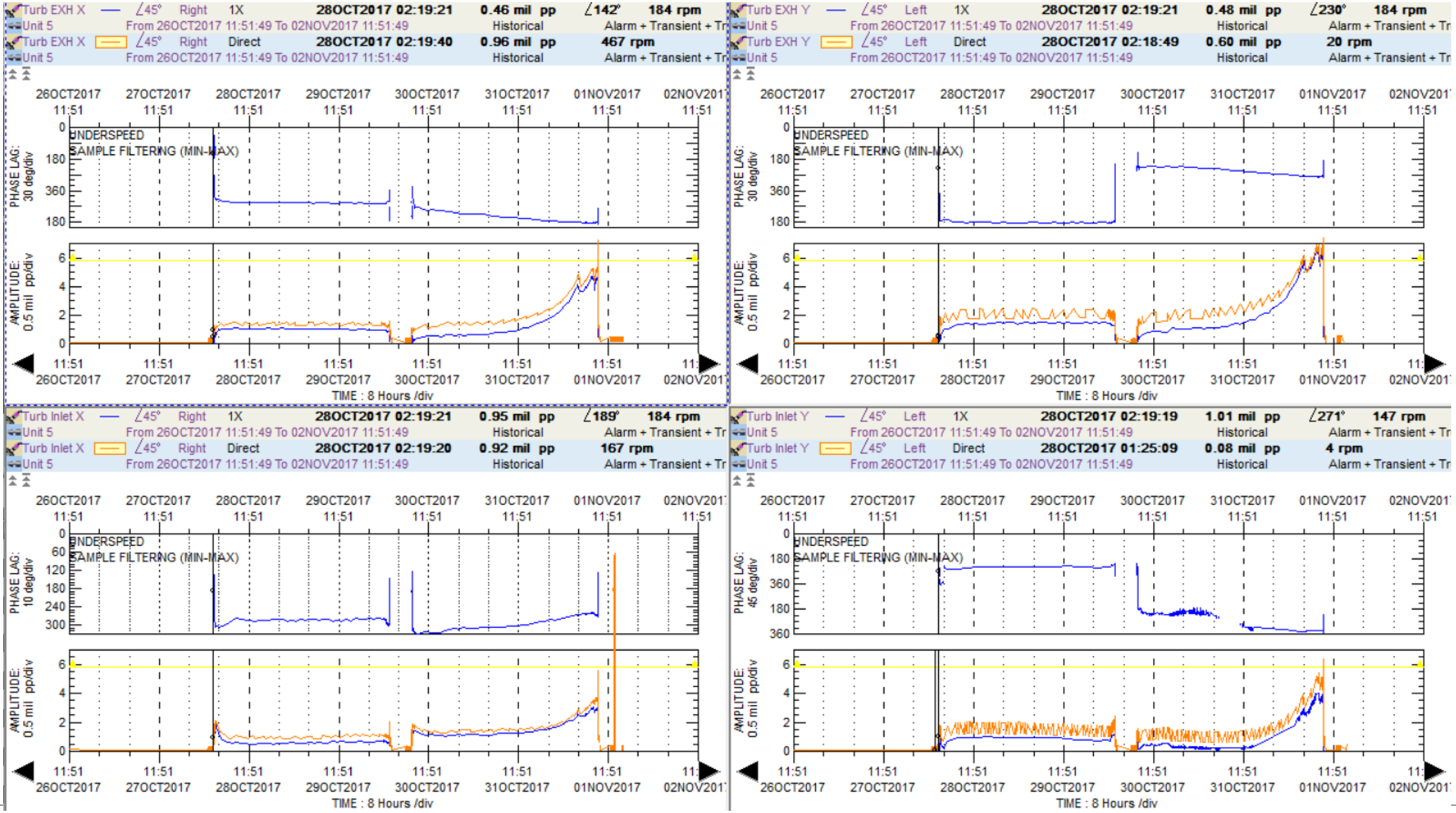
Shaft crack case



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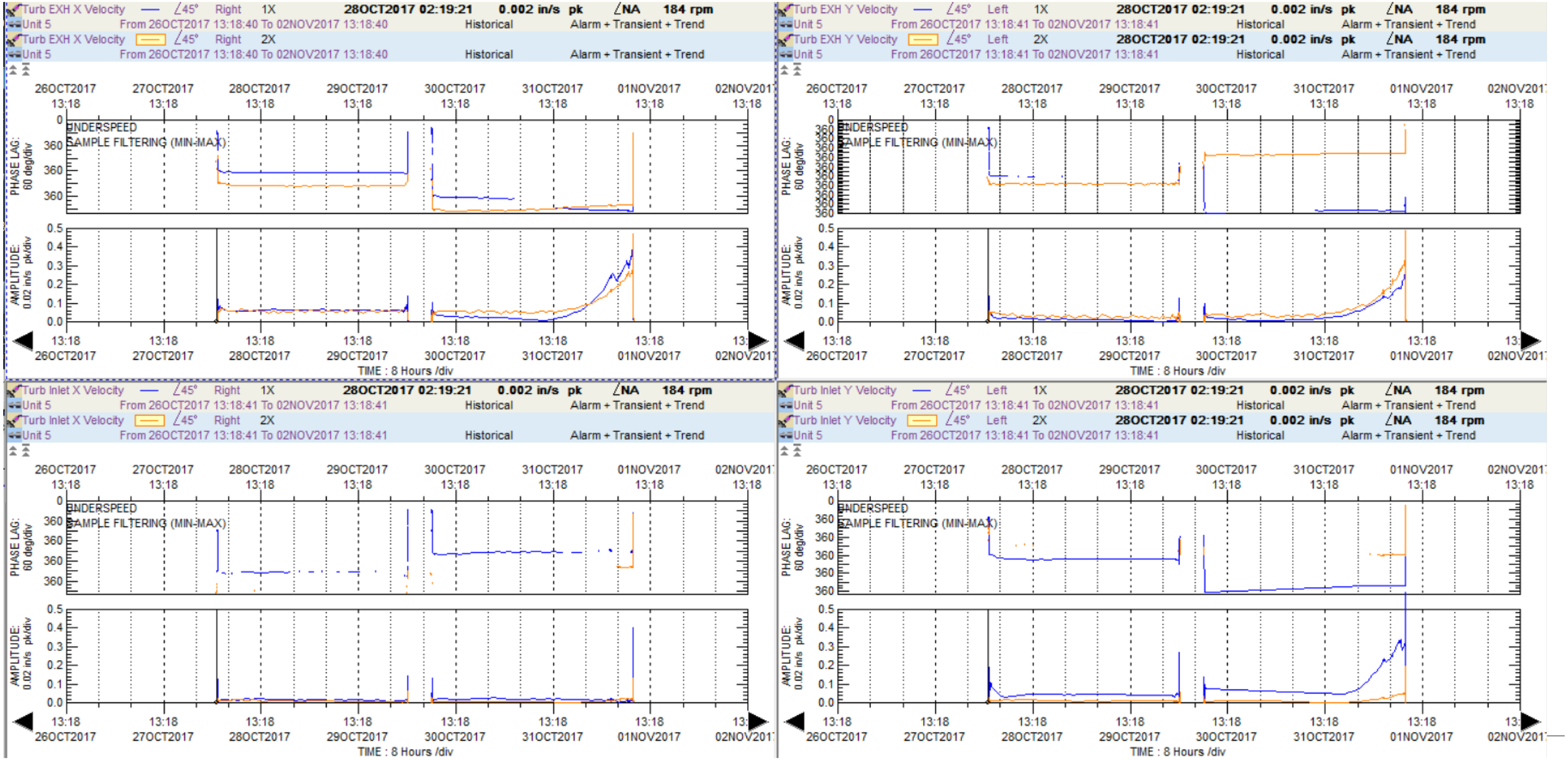
Shaft crack case



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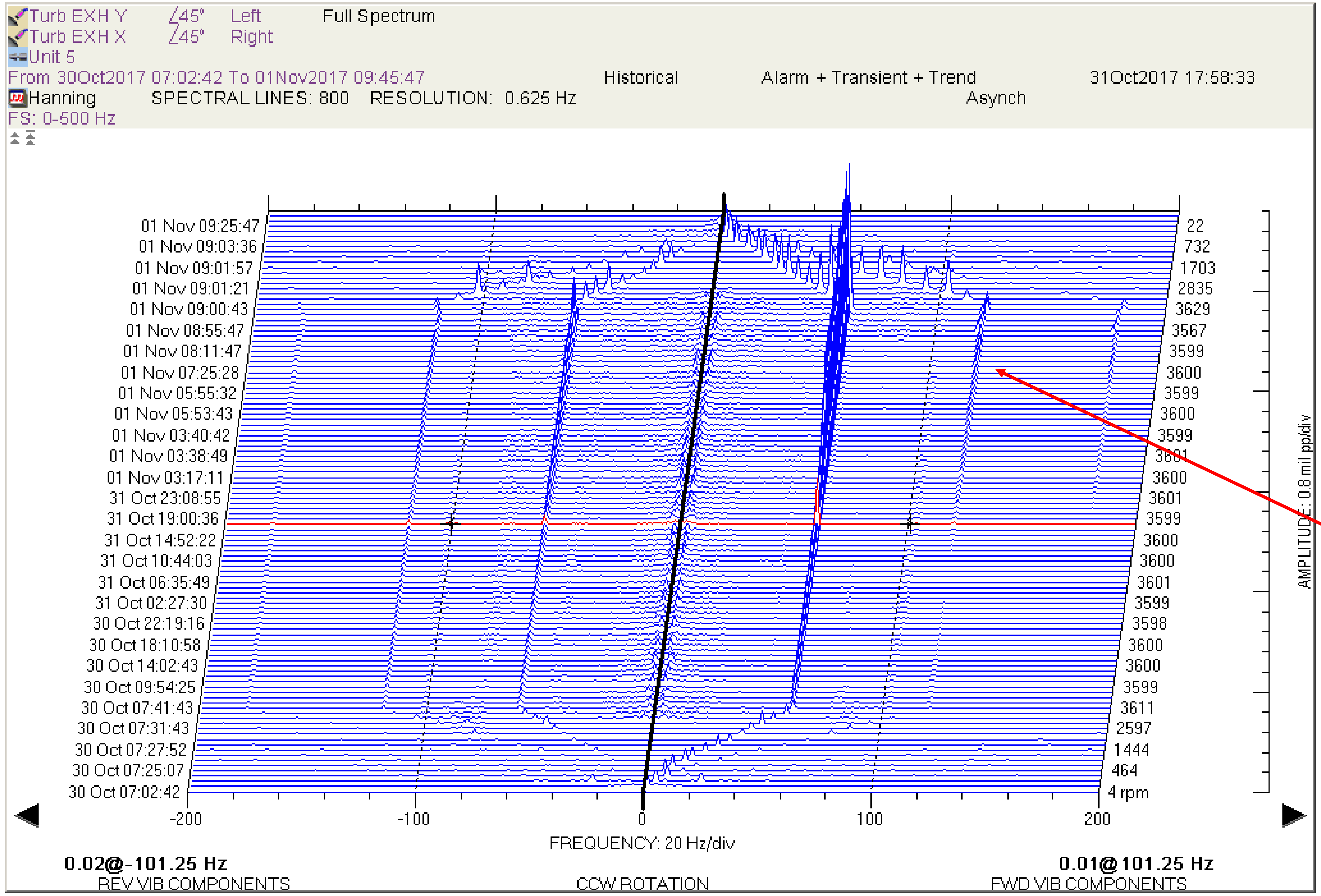
Shaft crack case



2X Vibration

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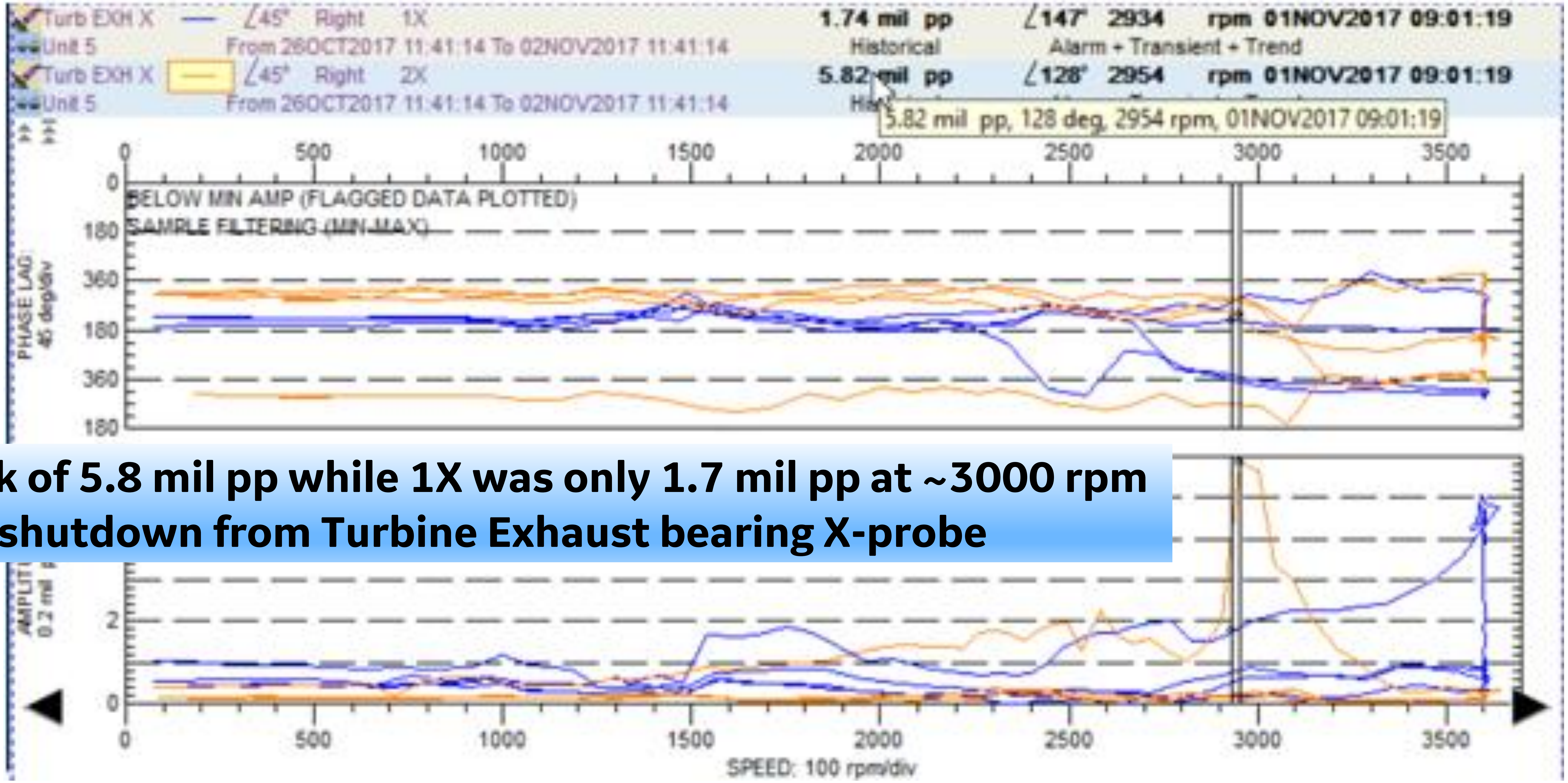
Shaft crack case

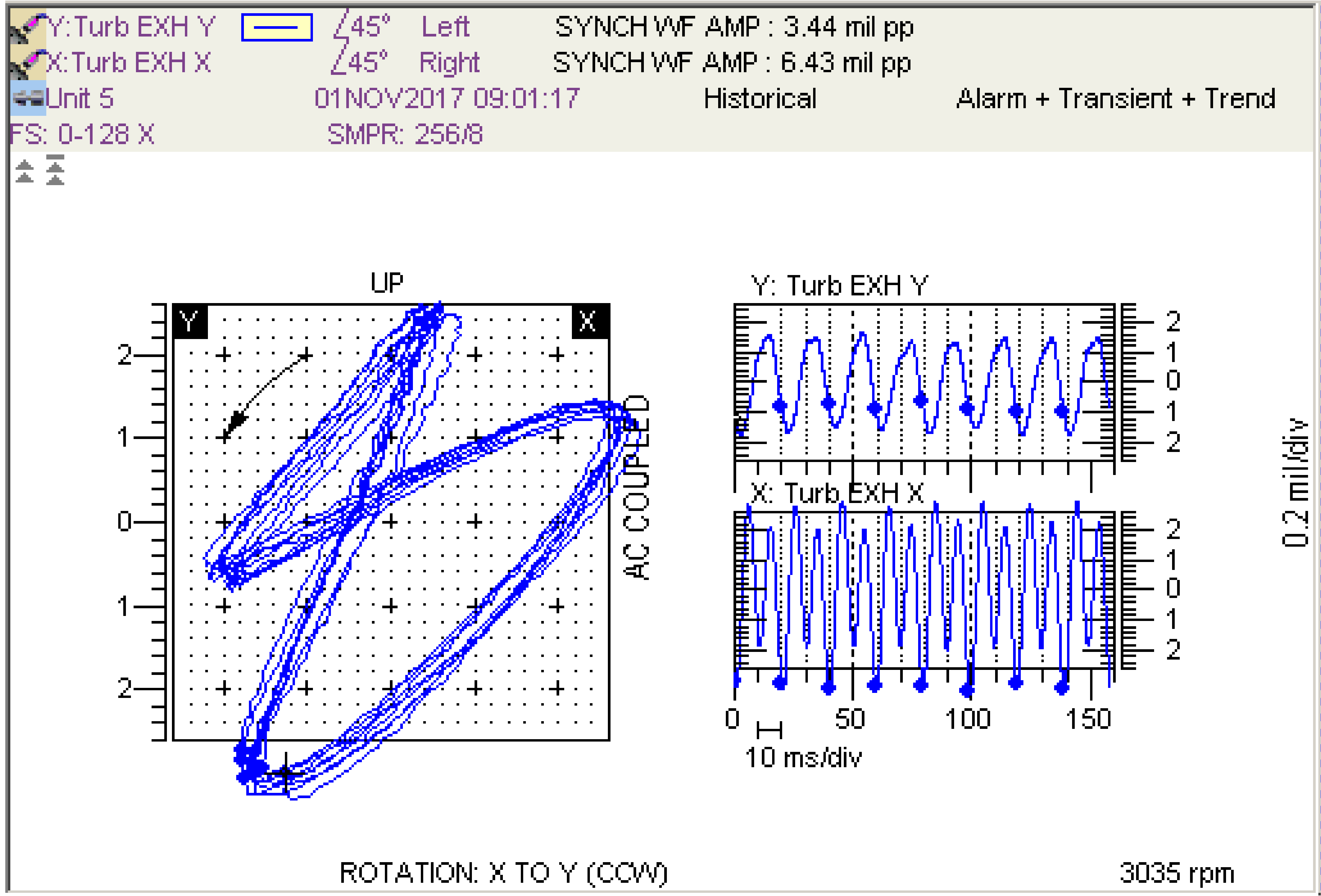


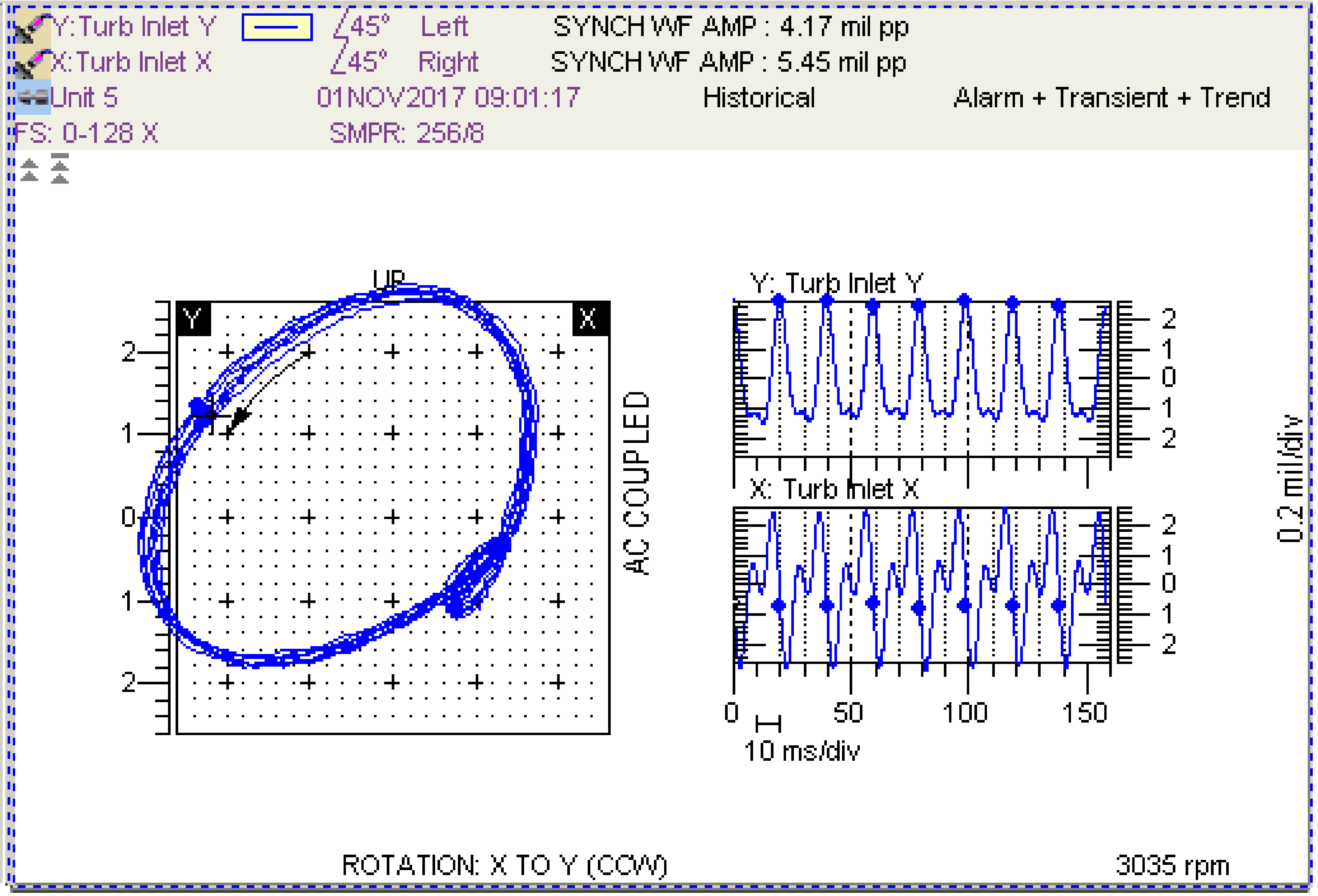
2X peak of 5.8 mil pp while 1X was only 1.7 mil pp at ~3000 rpm during shutdown from Turbine Exhaust bearing X-probe

Bently Nevada

Shaft crack case

Direct compensated orbit of the TURB EXH bearing at 3035 RPM during shutdown





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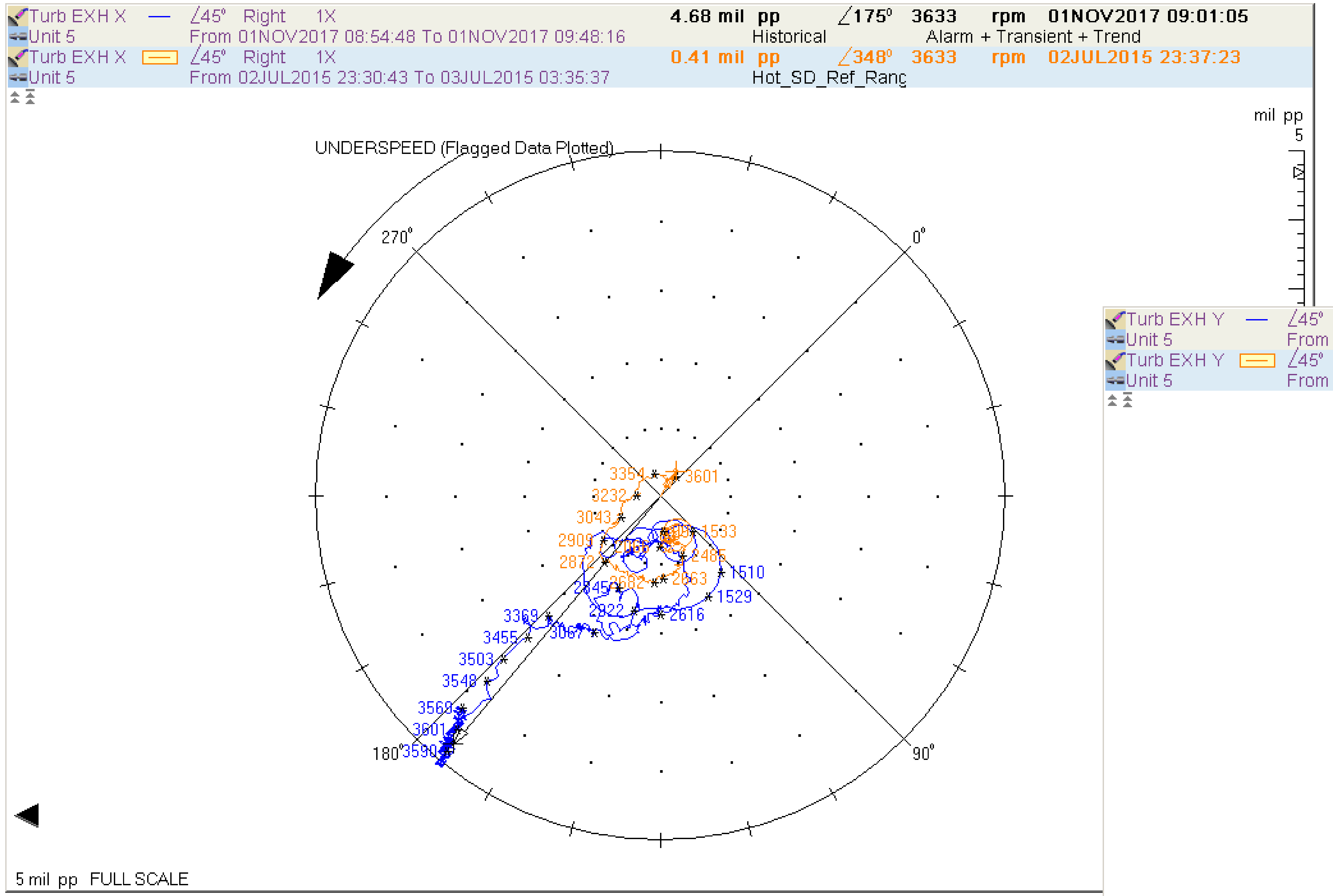
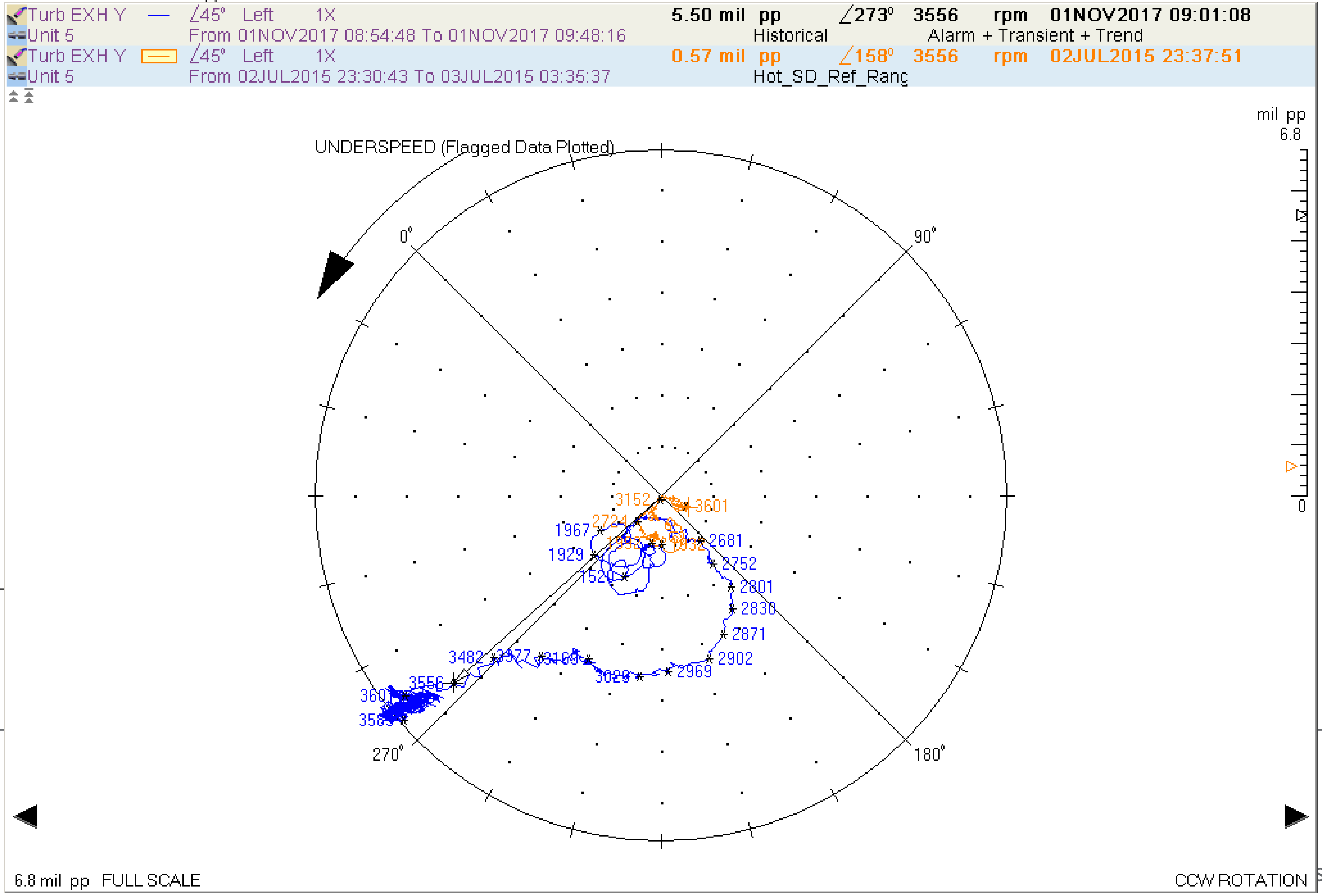
Shaft crack case



Abnormal polar plots during shutdown compared with the previous plots



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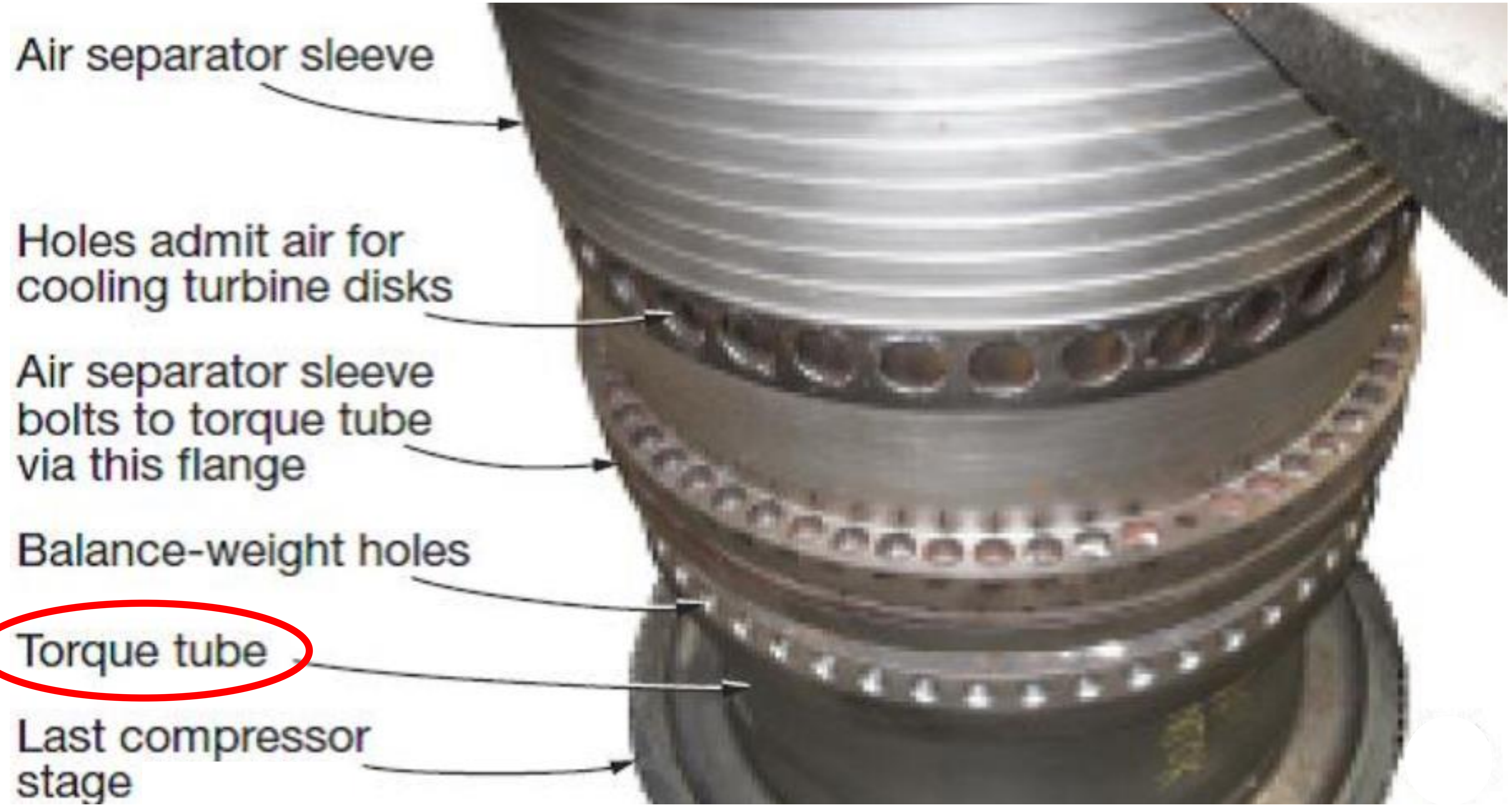
Shaft crack case

Inspection and Finding

Cracks on torque tube that joins the compressor and turbine sections of the rotor



Bently Nevada

Bently Nevada