



45TH **TURBOMACHINERY** & 32ND **PUMP** SYMPOSIA
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GEORGE R. BROWN CONVENTION CENTER

Vibration Analysis for Turbomachinery

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Content

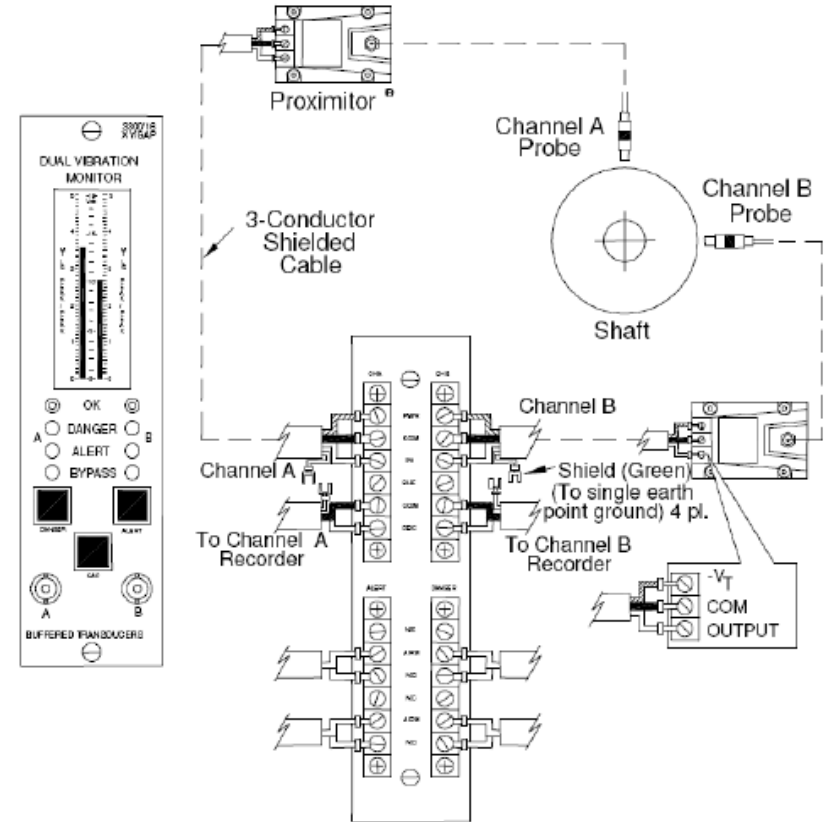
- Differences between vibration analysis of general purpose and turbomachinery
- Measurement
 - Proximity Probes
- Plots w/ case studies
 - Bode/Polar
 - Spectrums – Cascade/Waterfall
 - Orbit
 - Shaft Centerline

General Purpose vs Turbomachinery

- General purpose machinery – pumps, motors, fans, typical operates below 1st critical, most troubleshooting accomplished using spectrums
- Turbomachinery – compressors, turbines, tilt-pad bearings, etc., normally operates above 1st critical, requires analysis of many different plots to determine root cause of vibration issues.

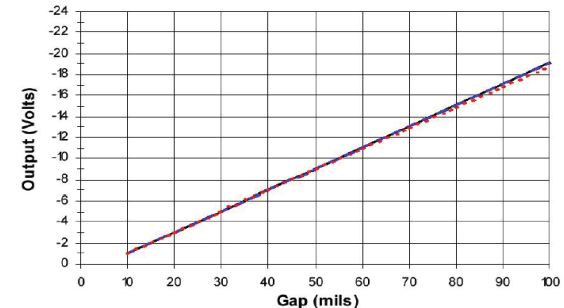
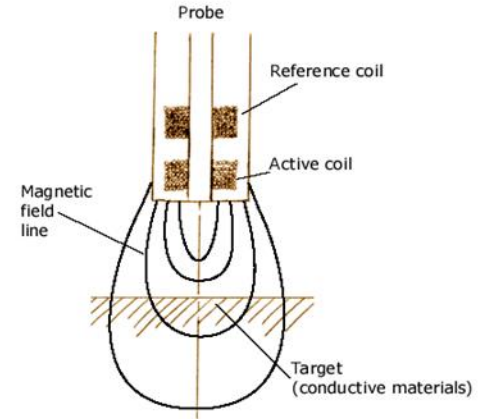
Measurement Proximity Probes

- Measures actual shaft displacement
- Includes probe, extension cable, and proximator which are a matched set
- Limited to < 1000-1200 hz



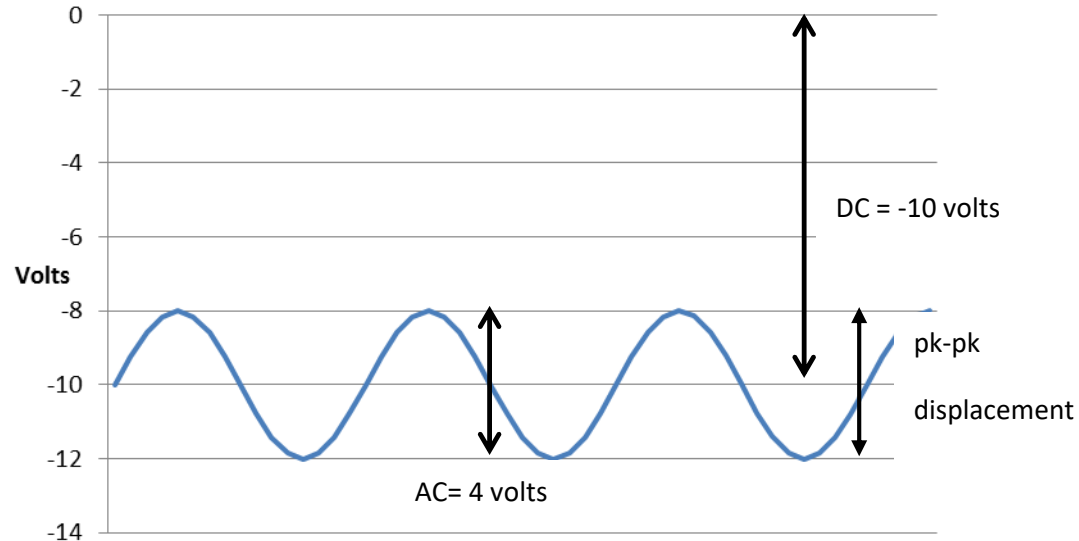
Proximity Probes

- Proximitors
 - excites the probe creating a magnetic field which is affected by the target (shaft)
 - produces a voltage proportional to the shaft displacement.
- Properly installed and maintained – very rugged and reliable
- Default sensitivity for 4140 is 200 mV/mil



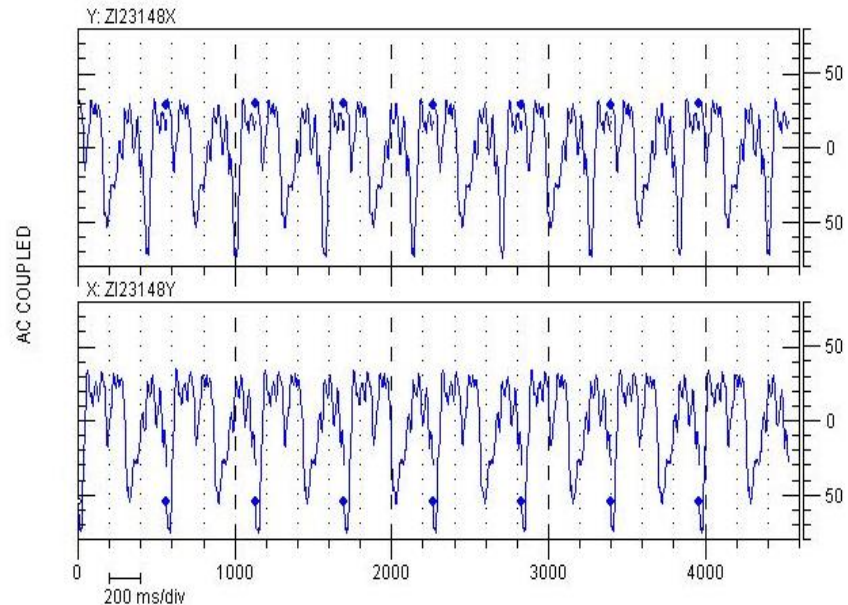
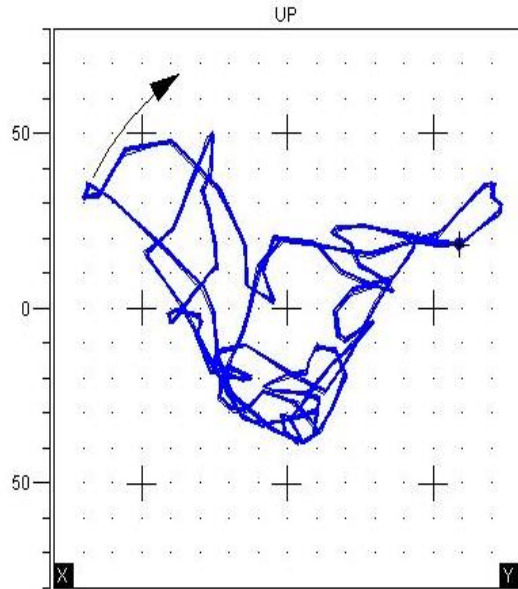
Proximity Probe Output

- DC component represents shaft position
- AC component represents shaft vibration



Proximity Probes - Pitfalls

- Target surface condition – scratches show up as vibration

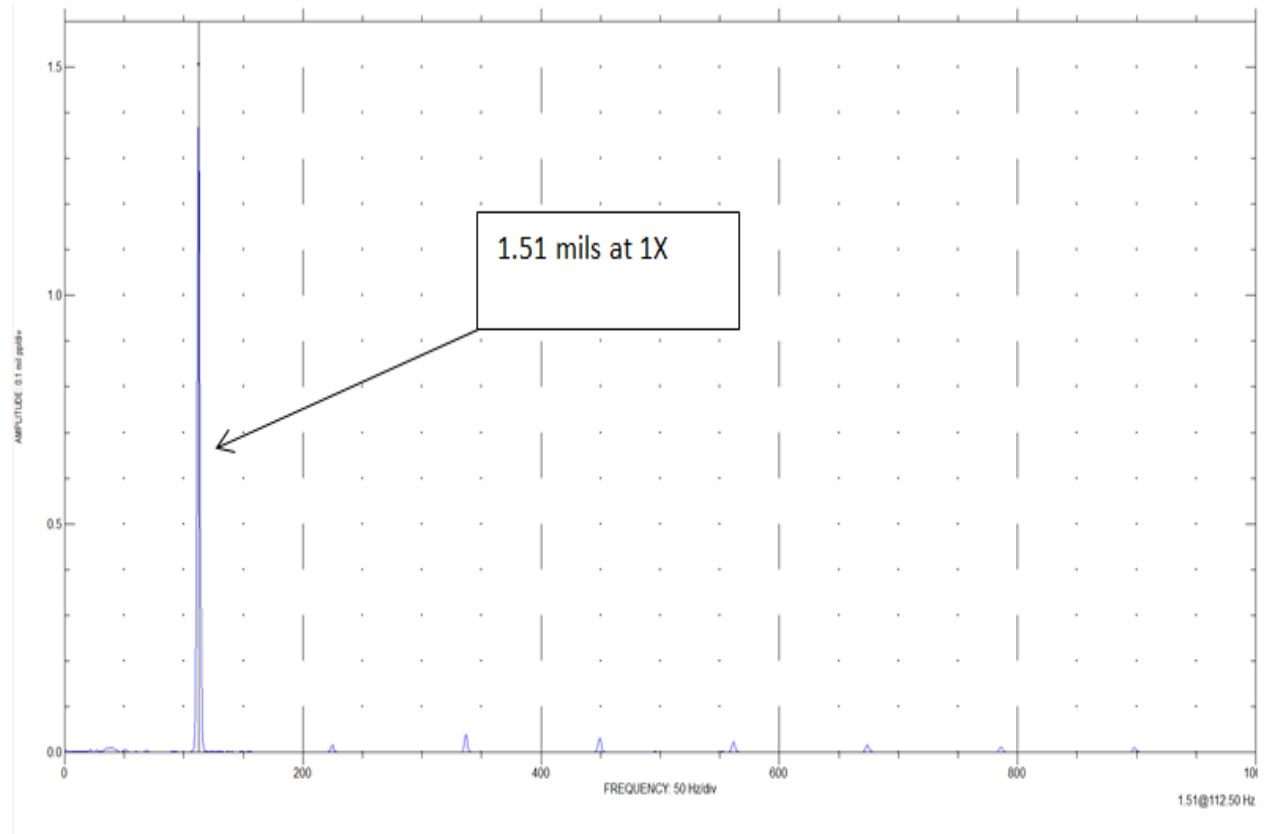


Proximity Probes - Pitfalls

- Magnetism – if a residual magnetism exists in the shaft or target, this will affect the probes
- Incorrect target material – coatings and/or non-ferrous materials will affect probe readings
- Wrong extension cable – proximator, extension cable, and probe are a matched set (total lengths of 1, 5, or 9 m)

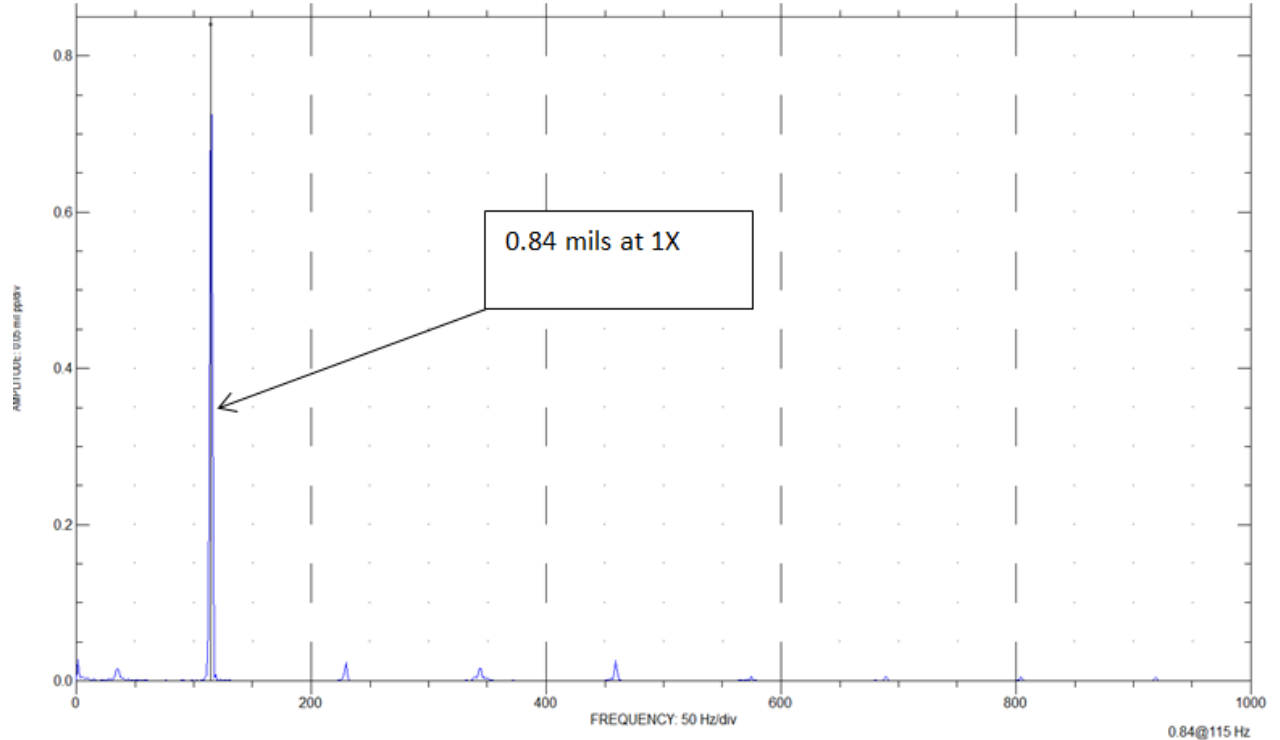
Proximity Probes - Pitfalls

- Rotor kit with correct proximator, no extension cable



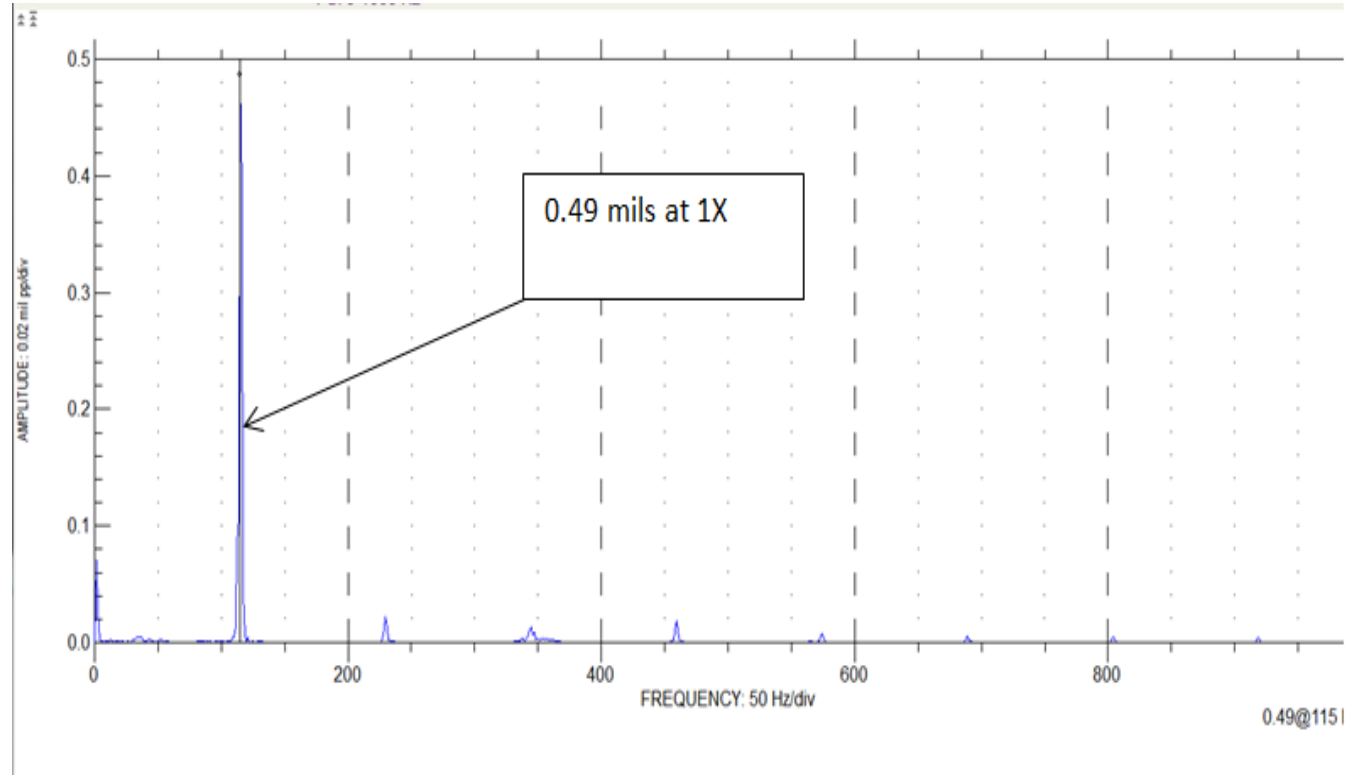
Proximity Probes - Pitfalls

- Same rotor kit, probe, proximator, added 4 m extension cable



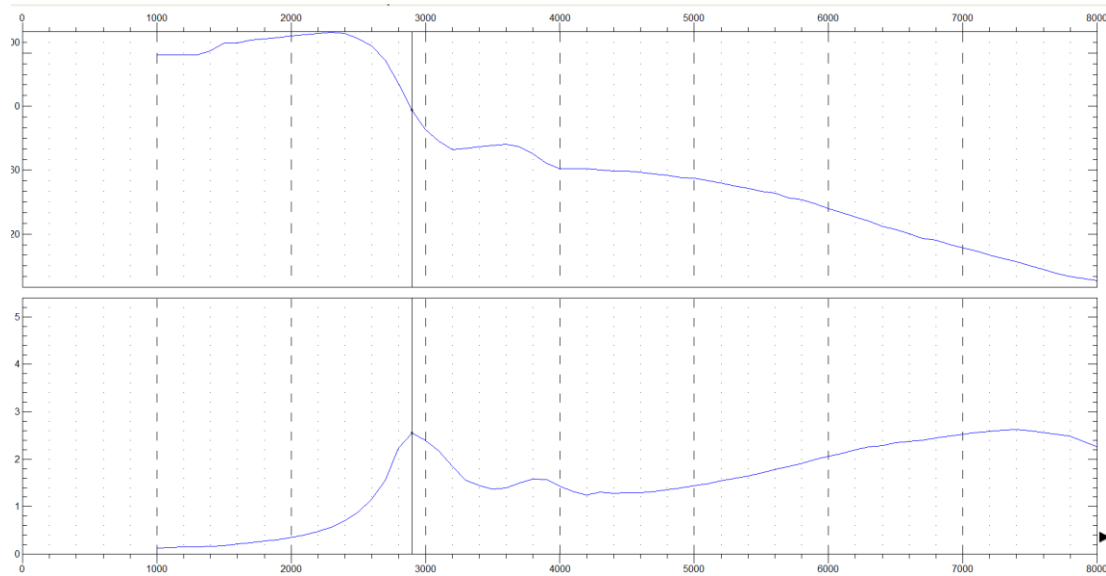
Proximity Probes - Pitfalls

- Same rotor kit, probe, proximator, added 8 m extension cable



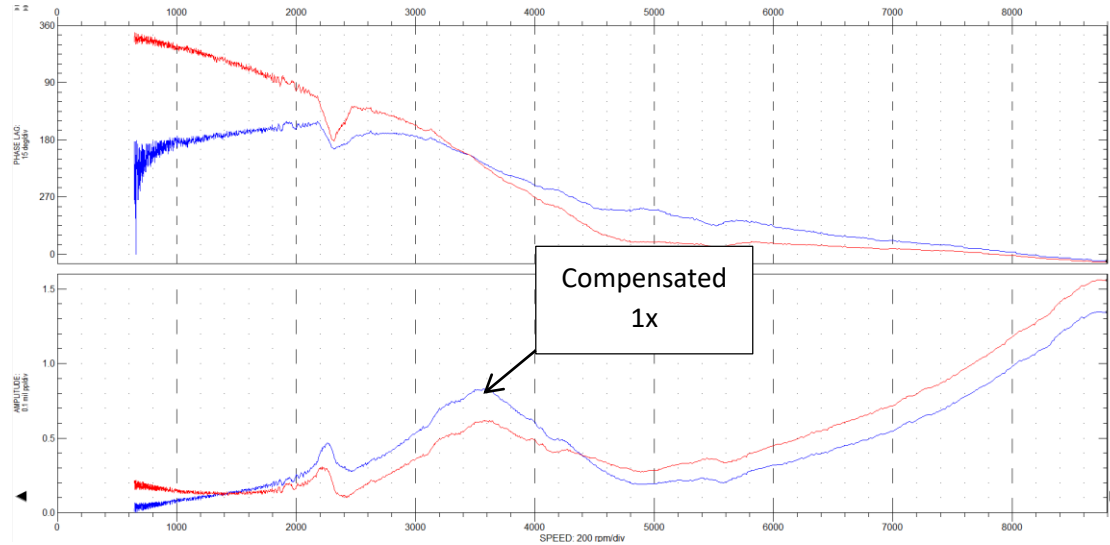
Bode Plots

- Vibration amplitude vs speed
- Critical shown by peak amplitude and phase change
- Uses
 - Determine natural frequencies (modes)
 - Indication of system damping
 - Tune rotordynamic models



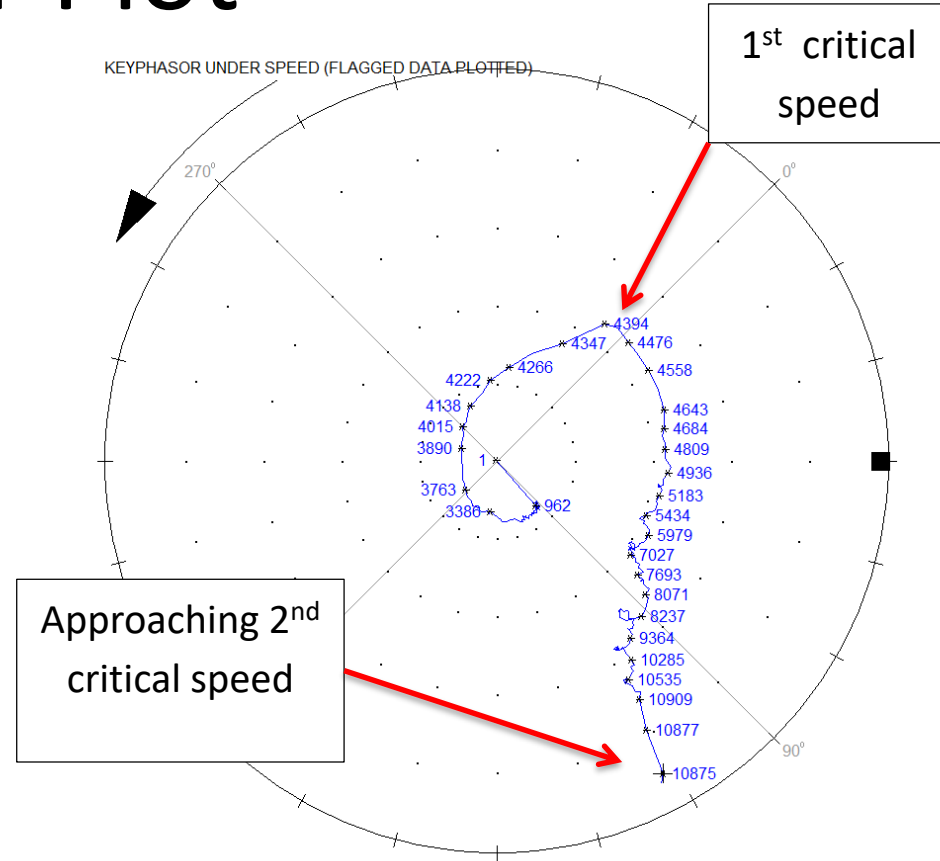
Bode Plot - Compensation

- Runout Vector – probe output at slow speed (<500 rpm), not vibration
- Compensated – run-out vector subtracted from raw reading



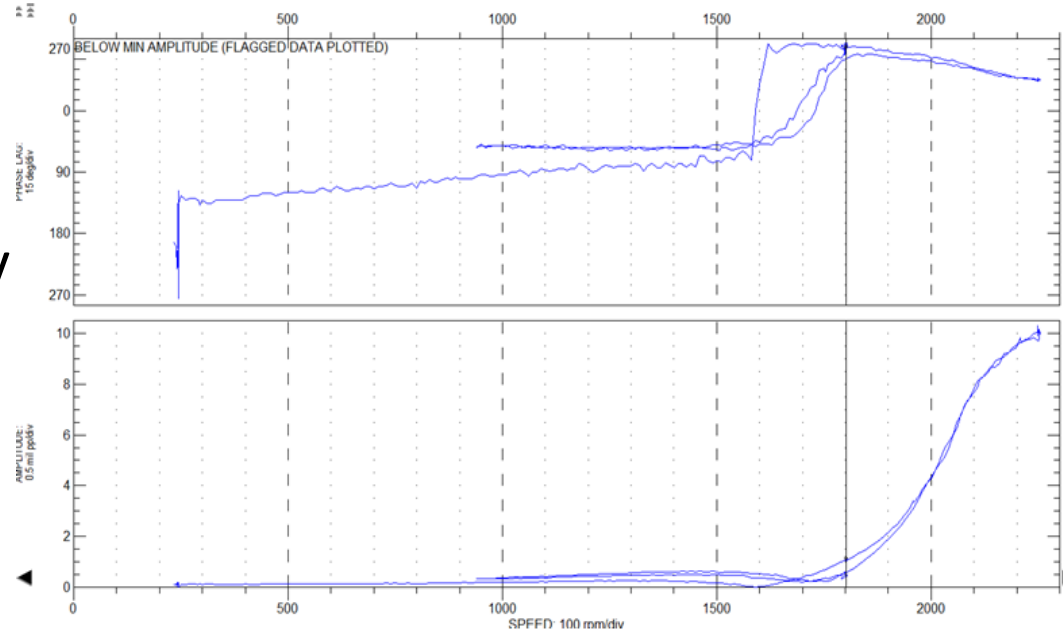
Polar Plot

- Polar Plot – same data as Bode Plot, just different format
- Critical speed defined by maximum peak and approximately 180° phase change



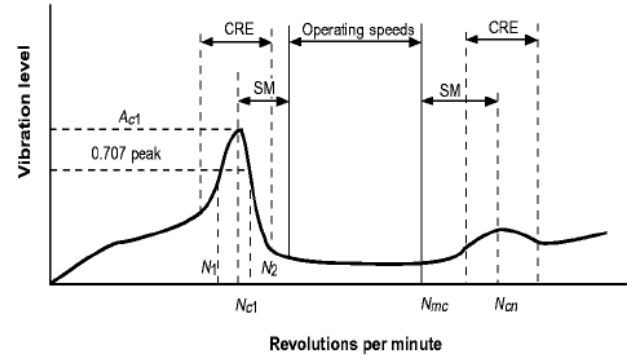
Bode Plots – Resonance Detection

- 4 pole generator FAT
- Identified 1st critical only 400 rpm above running speed during overspeed test..
- Problem could occur if natural frequency drops close to running speed.



Amplification Factor

- AF is a good ND indication of damping present in system
 - $AF \propto \frac{1}{\text{damping}}$
- ½ power method
- Change in AF can indicate change in the hydrodynamic bearings since they provide a lot of the damping in the system



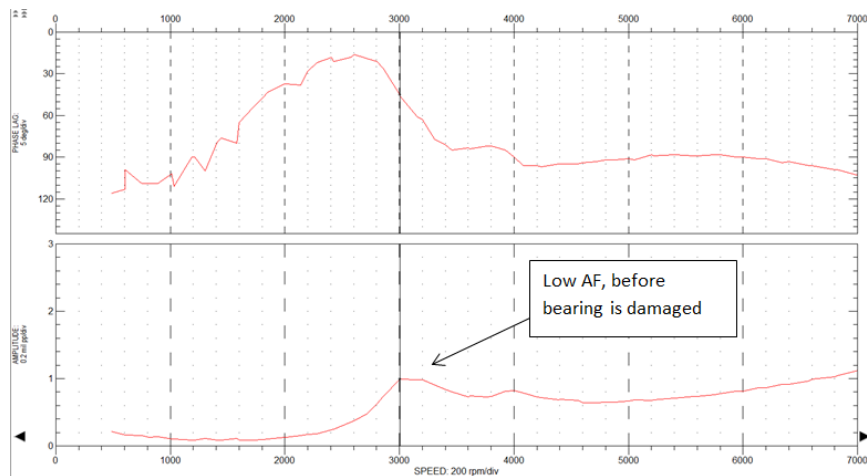
- N_{c1} = Rotor first critical, center frequency, cycles per minute.
- N_{cn} = Critical speed, $n\#$.
- N_{mc} = Maximum continuous speed, 105%.
- N_1 = Initial (lesser) speed at $0.707 \times \text{peak amplitude (critical)}$.
- N_2 = Final (greater) speed at $0.707 \times \text{peak amplitude (critical)}$.
- $N_2 - N_1$ = Peak width at the half-power point.
- AF = Amplification factor.

$$= \frac{N_{c1}}{N_2 - N_1}$$
- SM = Separation margin.
- CRE = Critical response envelope.
- A_{c1} = Amplitude at N_{c1} .
- A_{cn} = Amplitude at N_{cn} .

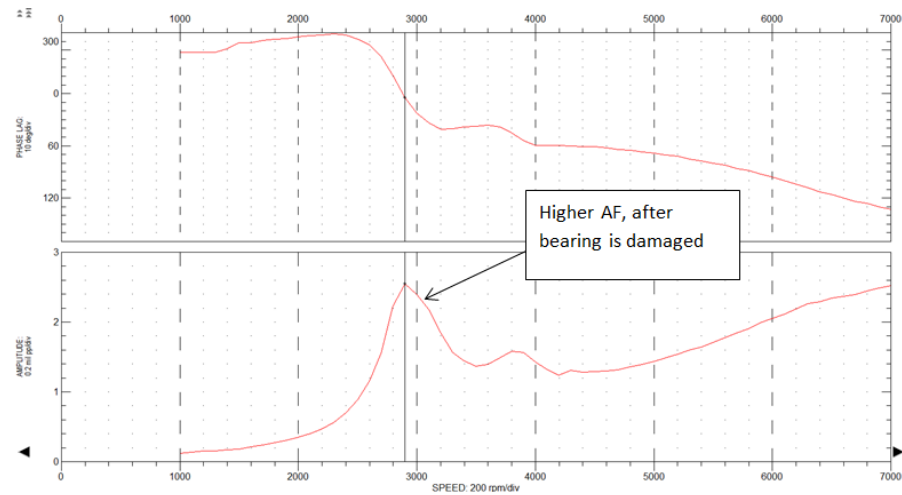
Amplification Factor

Detecting Bearing Damage in Centrifugal Compressor

Before Trip



After Trip



Amplification Factor

Detecting Bearing Damage

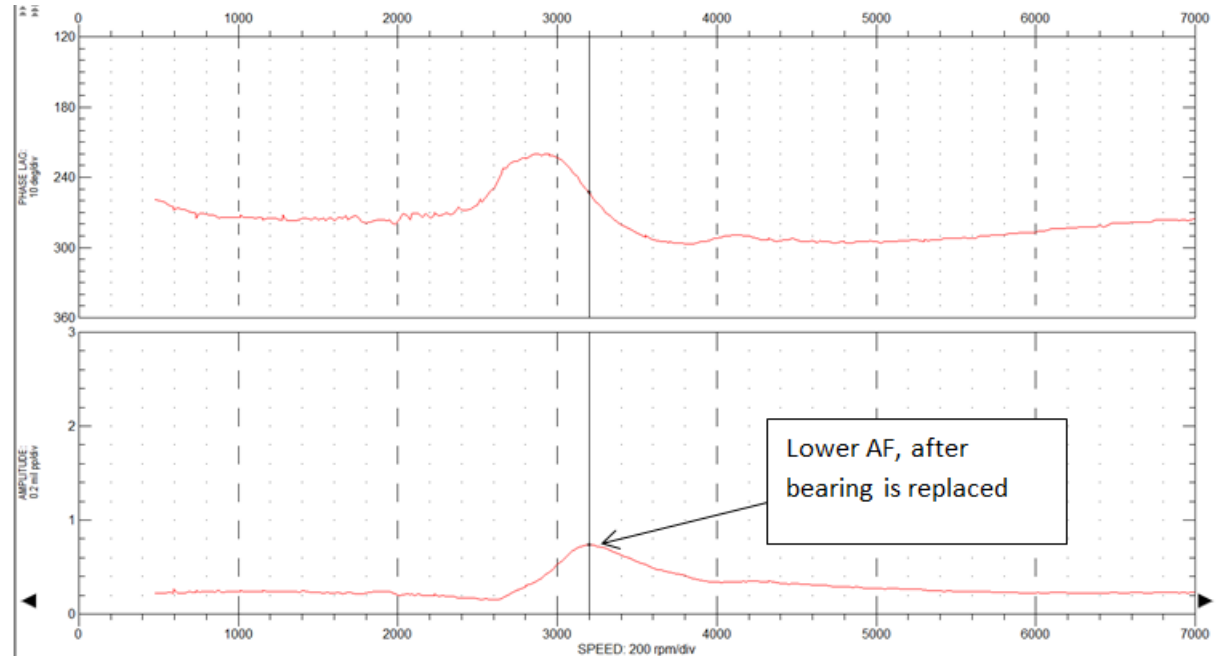
- Bearing showed evidence of loss of lube
- Clearance was 30 % above maximum
- Increase in clearance reduces damping and increases AF



Amplification Factor

Detecting Bearing Damage

- Both radial bearings replaced
- AF returns to normal level

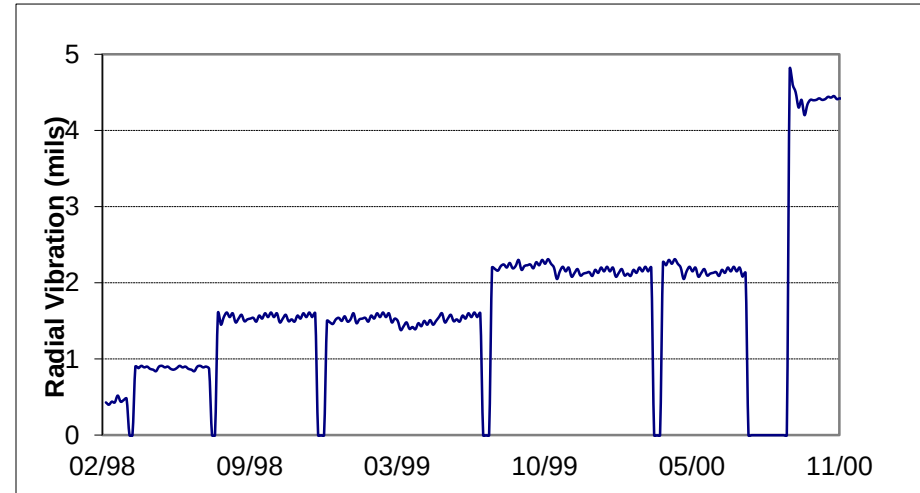


Amplification Factor

Unbalance

Barrel Compressor

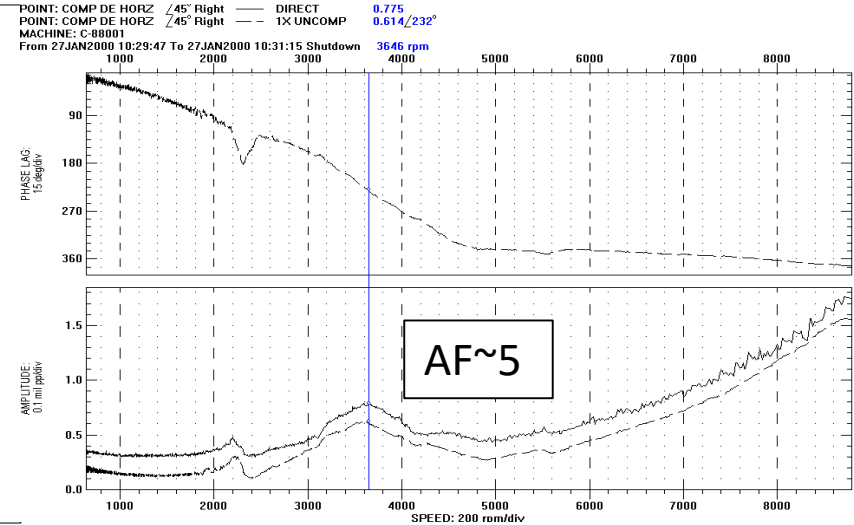
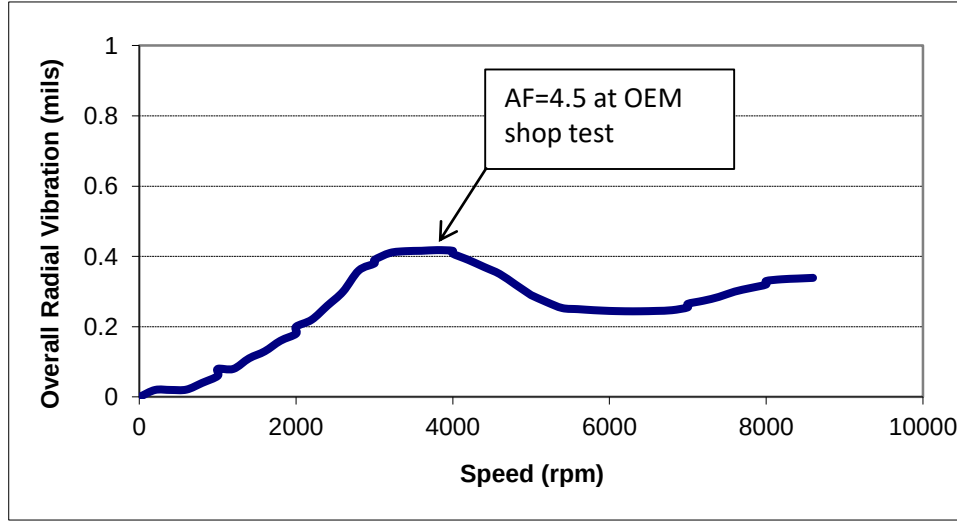
- Radial vibration increases after some trips, but not all
- No indication of bearing damage
- No loss of lubrication



Amplification Factor Unbalance

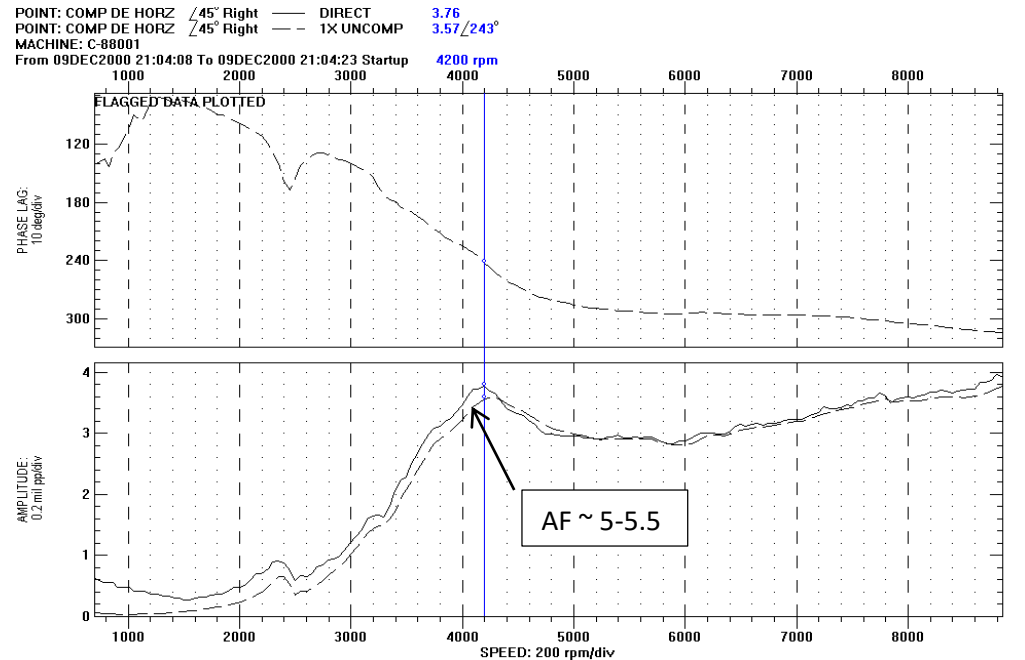
FAT Test

2 Years later



Amplification Factor Unbalance

- After extended plant outage, compressor restarts with high radial vibration
- Rotor inspection shows large amount of fouling, cleaned and returned to service

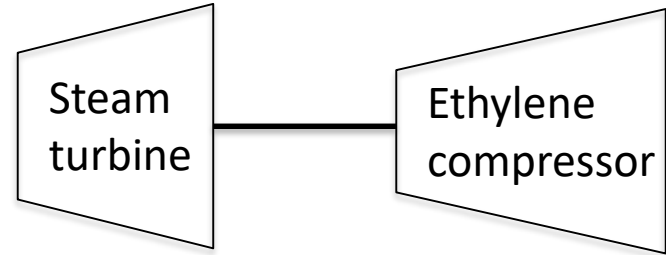


Tune Rotordynamic Models

- Measured natural modes used to “tune” rotordynamic models to better represent actual conditions

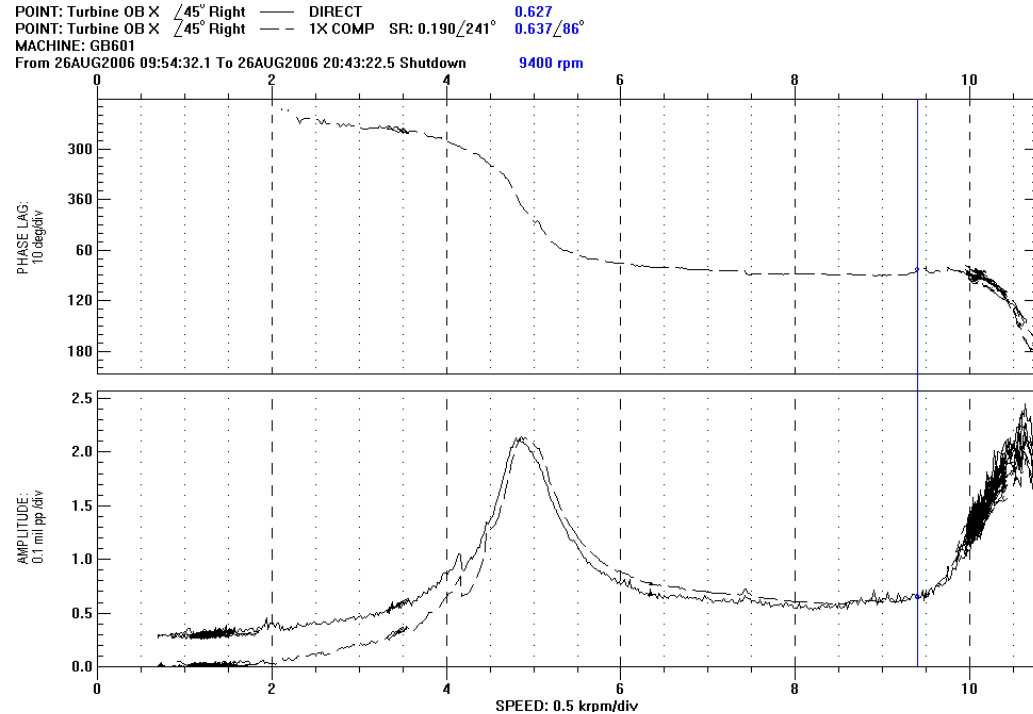
High Speed Steam Turbine

- 8,000 hp steam turbine, driving ethylene refrigeration compressor
- Turbine normally operates between 8,500-9,500 rpm
- High radial vibration on outboard end during overspeed trip testing.



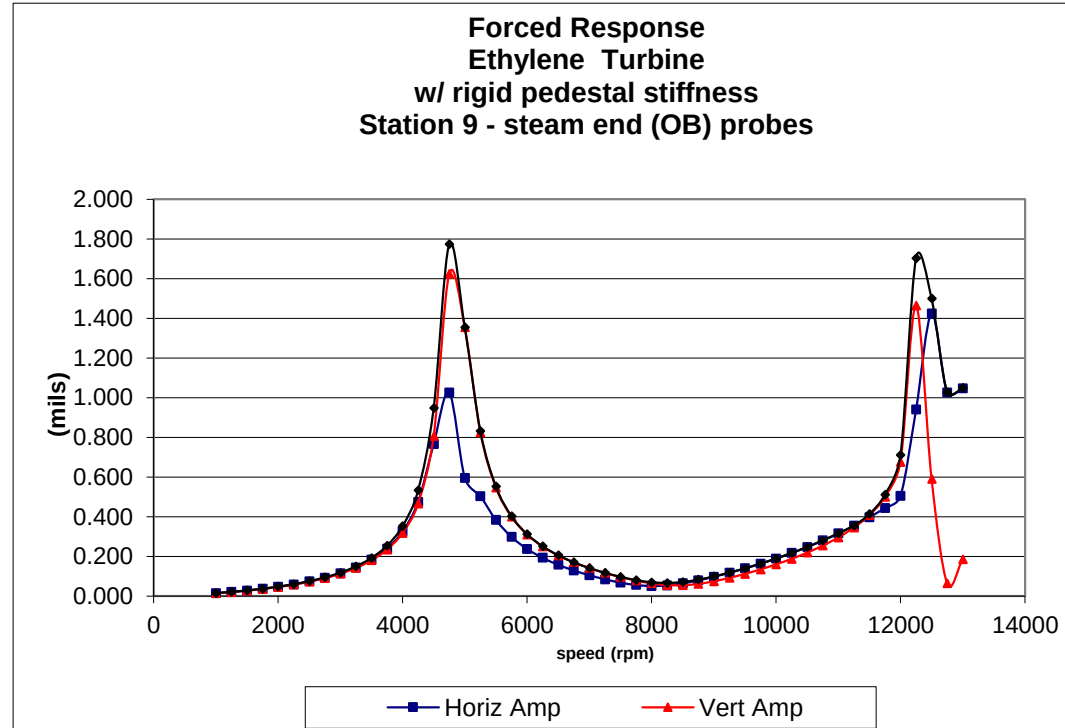
Tune Rotor Models

- High radial vibration on steam turbine outboard end during overspeed trip testing, 10,700 rpm
- Turbine normally operated below 10,000 rpm with vibration < 1 mil



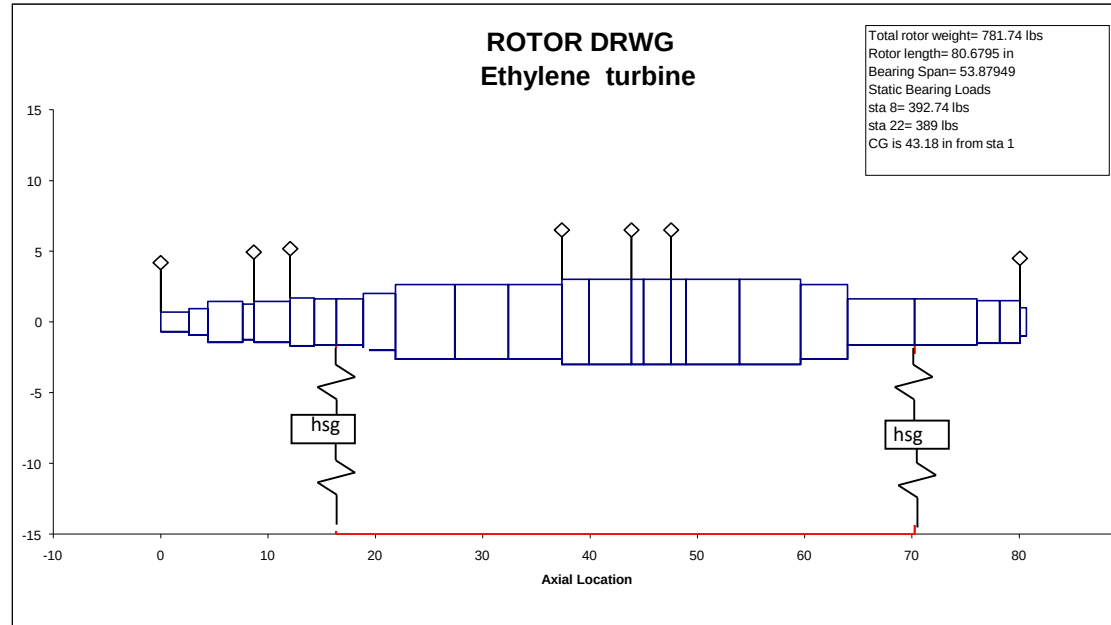
Tune Rotor Models

- Rotor model built to evaluate cause of vibration
- Model predicts 2nd critical much higher than vibration in field.
- Rotor model assumes rigid supports



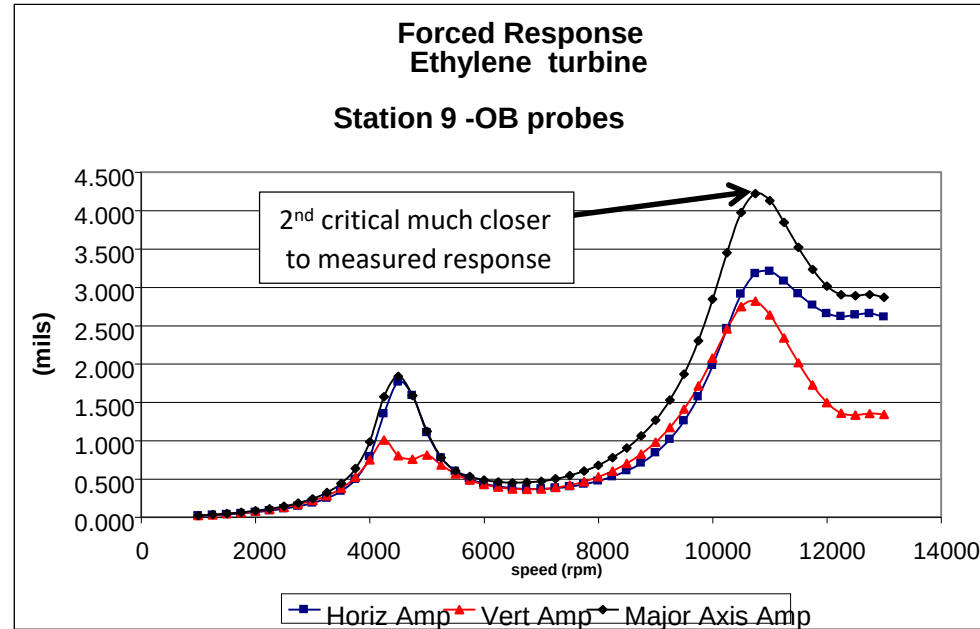
Tune Rotor Models

- Turbine outboard bearing has a “wobble foot” design to allow for thermal expansion
- Bearing housing support stiffness added to rotor model



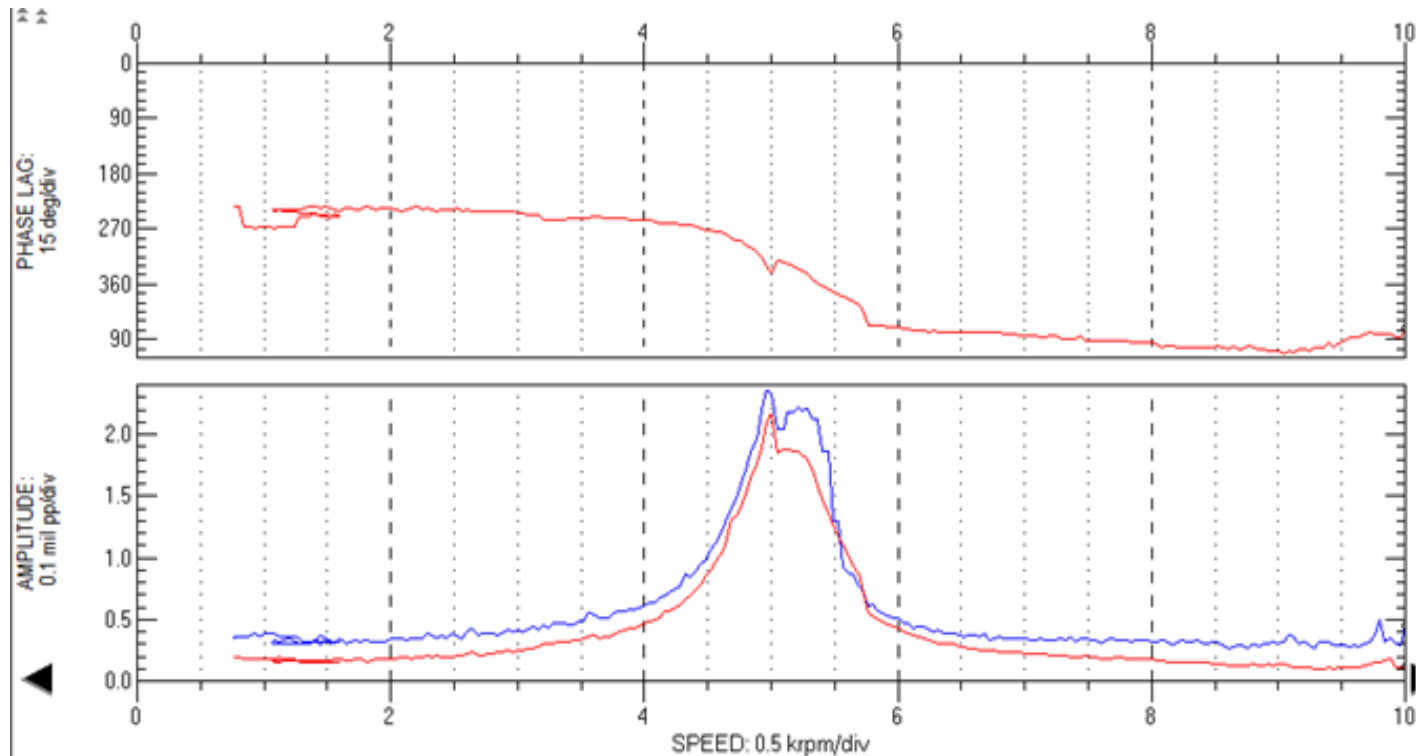
Tune Rotor Models

- Adding bearing support stiffness to model lowers predicted 2nd critical
- Turbine OB bearing clearance was shimmed to 0.0005 in below minimum design



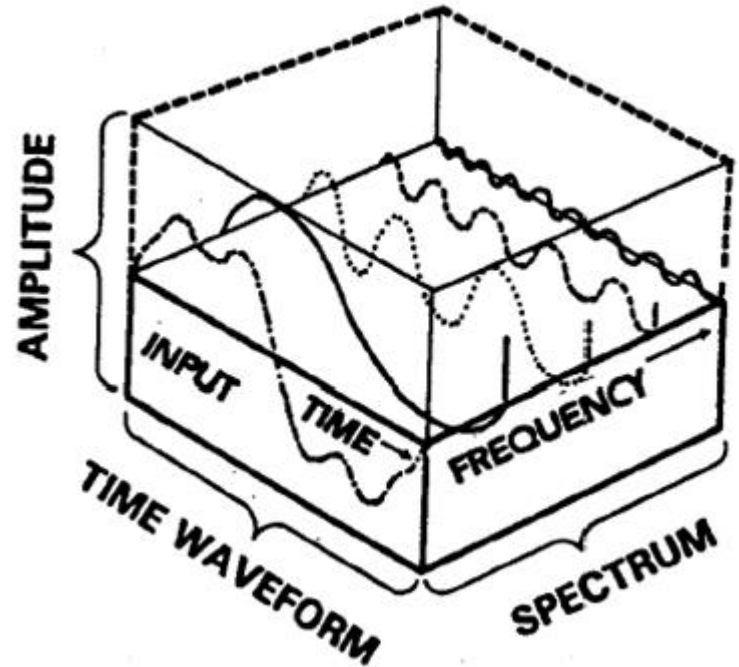
High Speed Steam Turbine

After bearing clearance change



Spectrum Plot

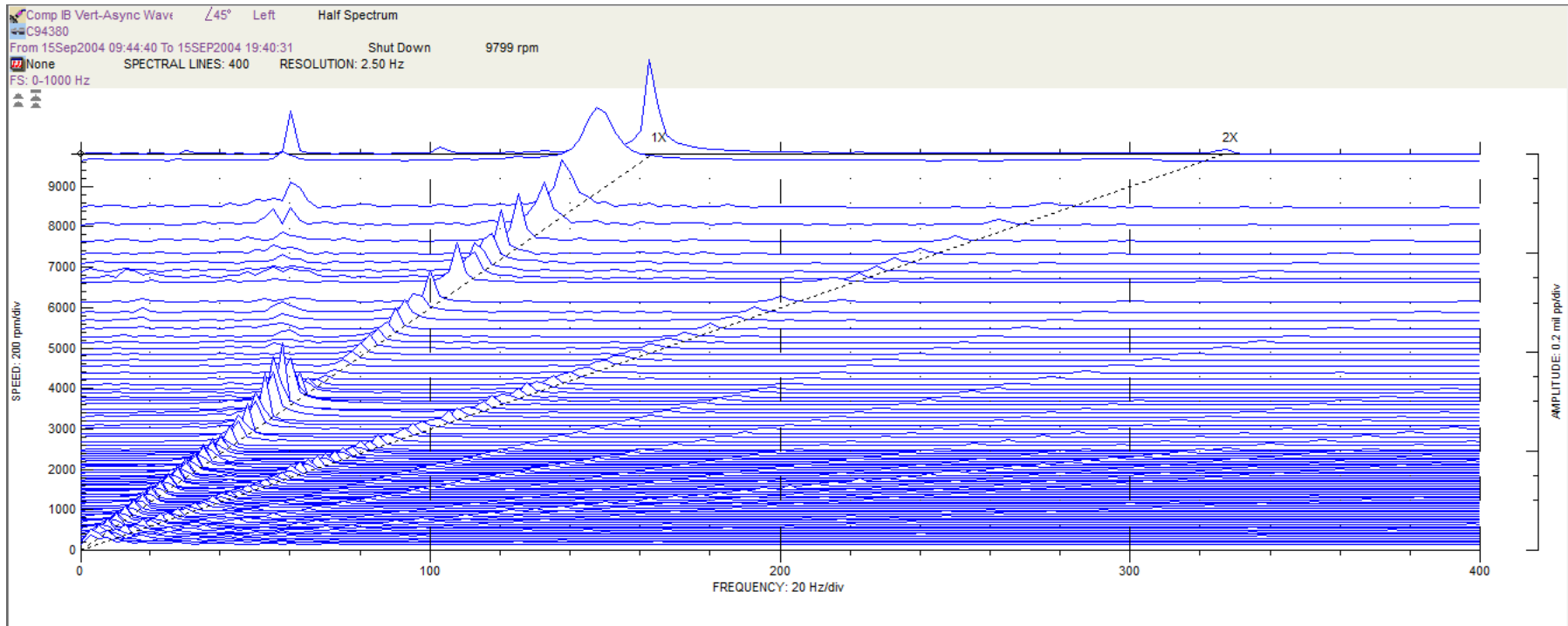
- FFT (Fast Fourier Transform) used to convert from time domain to frequency domain
- Spectrum excellent tool for determining frequency of different components
- Data acquisition time may limit use during transient events



$$\text{DAT} = \frac{\text{Number of spectrum lines}}{F_{\max}}$$

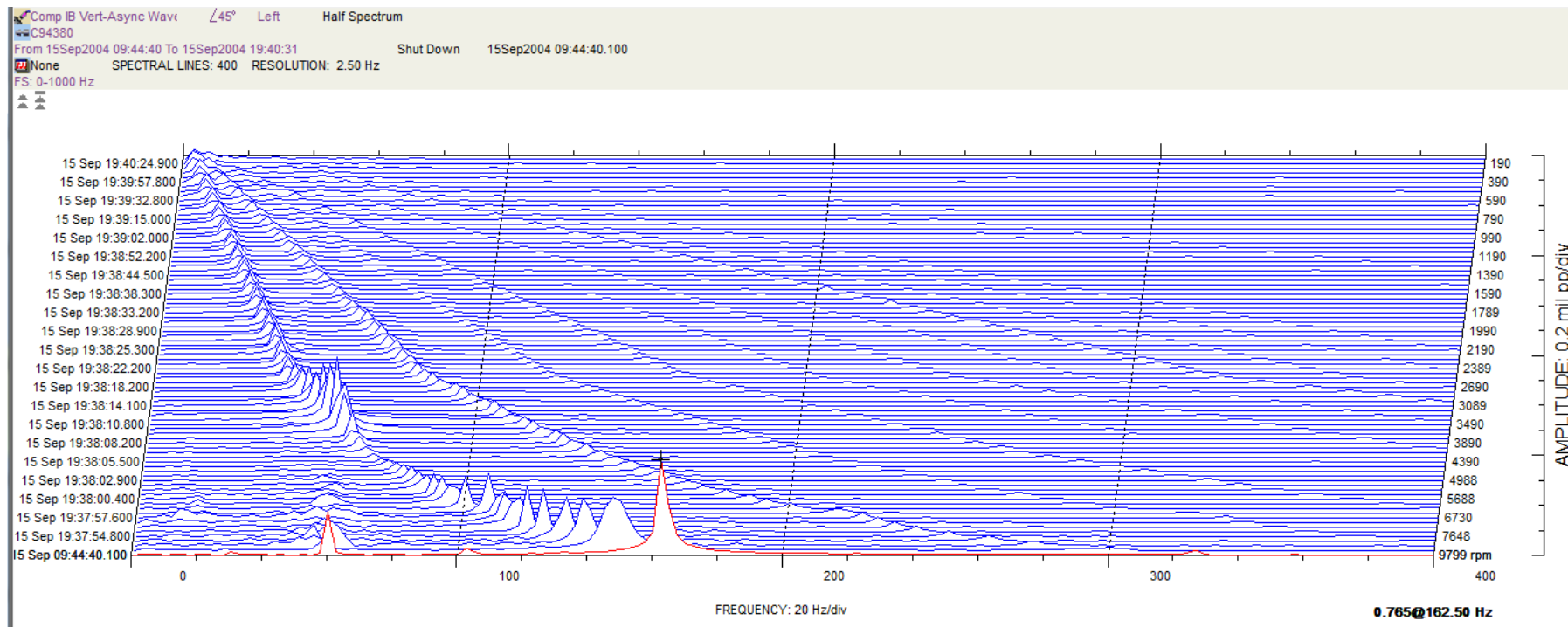
Cascade

- Multiple spectrums plotted versus speed



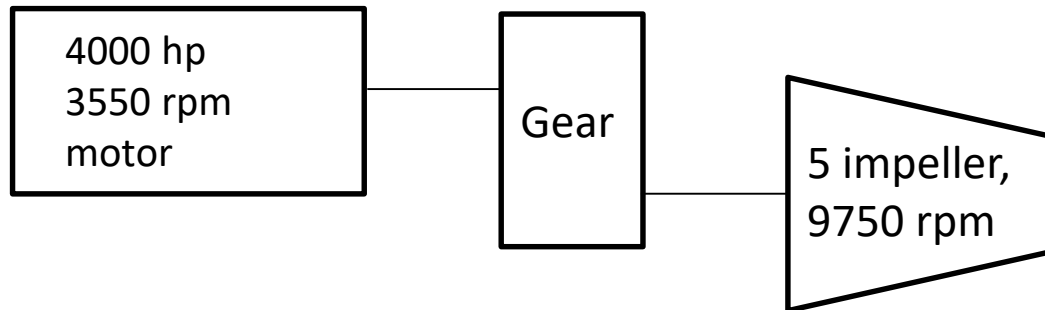
Waterfall Plot

- Multiple spectrums plotted versus time



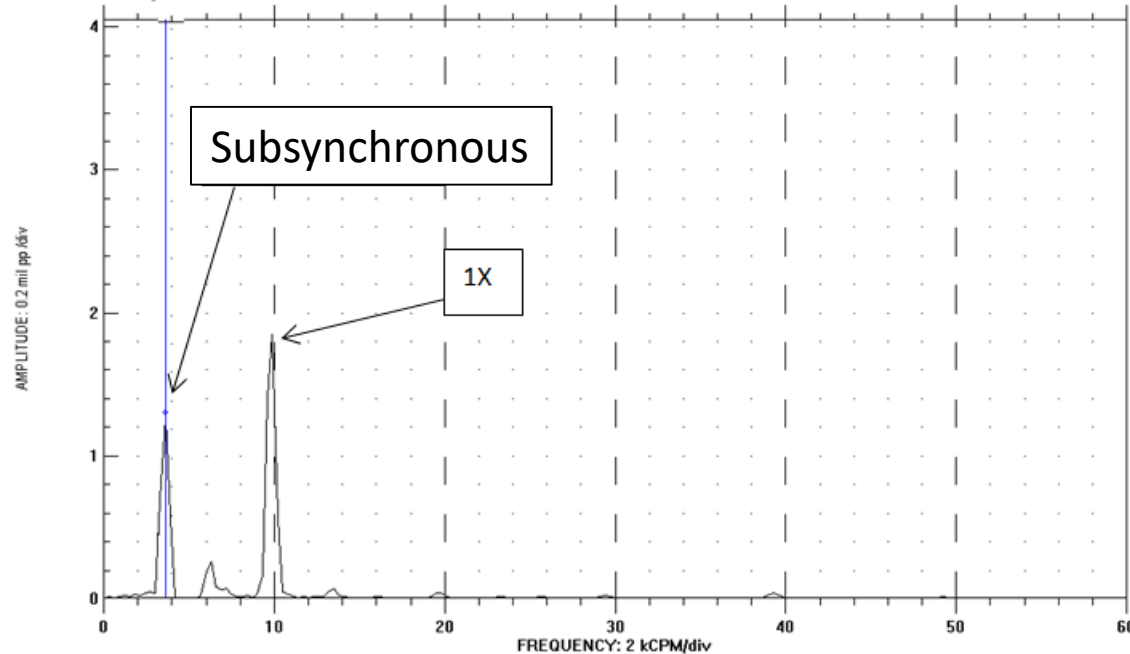
Centrifugal Compressor w/ sub-synchronous instability

- Single casing, straight through
- Propylene export
- $P_1=30$, $P_2=300$ psig
- 5 pad, LBP bearings
- Dry gas seals
- Balance drum has rotating labyrinth and abraidable stationary



Centrifugal Compressor w/ sub-synchronous instability

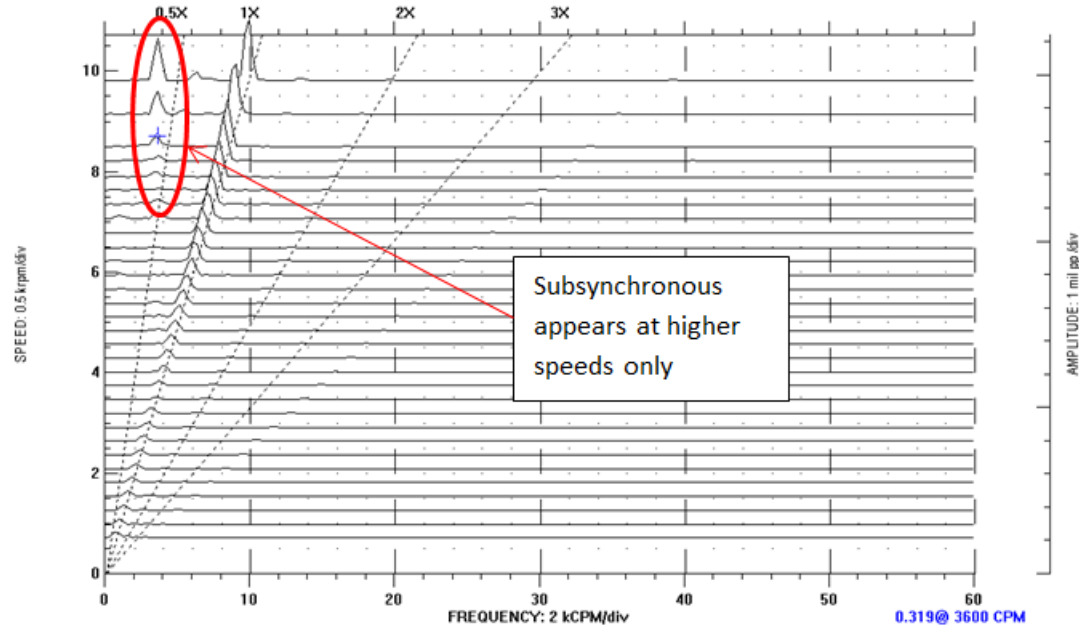
- Sub-synchronous vibration appears on start-up after overhaul in 2004
- 3600 cpm which is close to rotor's 1st mode



Centrifugal Compressor Instability

Sub-synchronous vibration

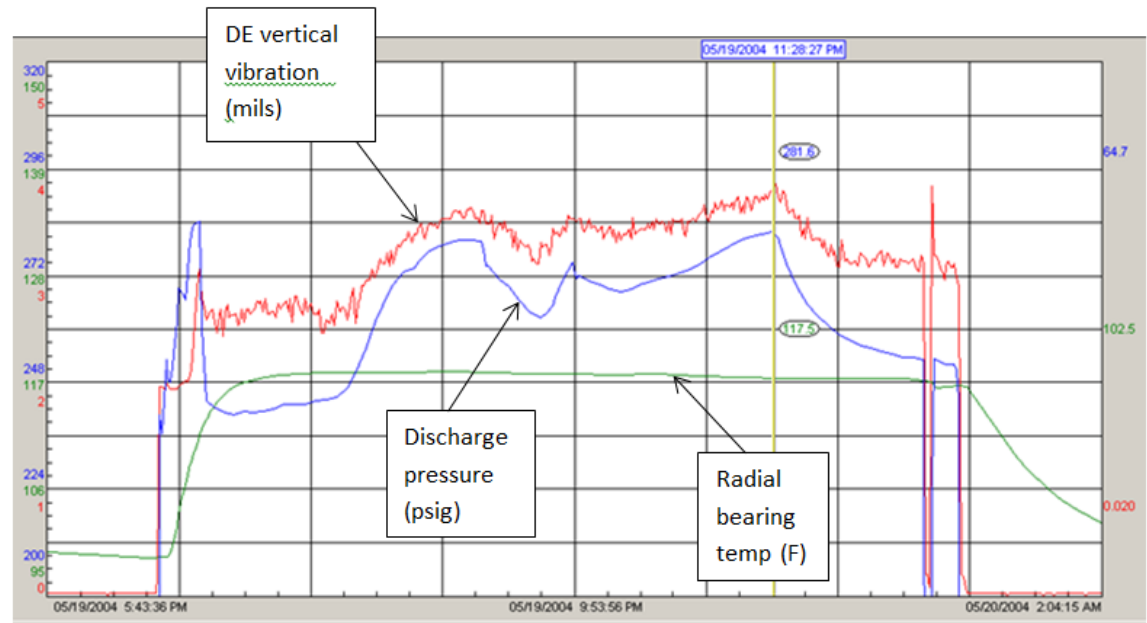
- Cascade plot shows subsynchronous peak appears only at higher speed
- Indicates that sub-synchronous could be exciting 1st critical



Centrifugal Compressor Instability

Sub-synchronous vibration

- Vibration tracked closely with discharge pressure

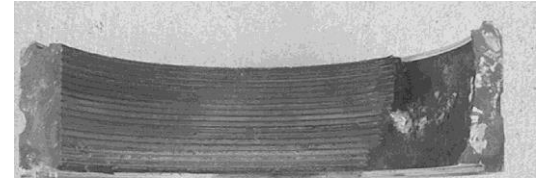


Centrifugal Compressor Instability

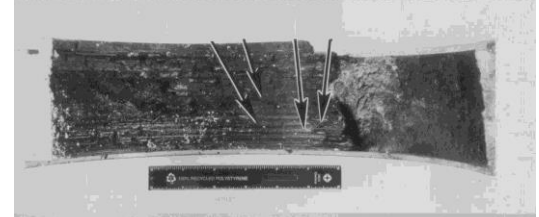
Balance Drum Seal History

- Abradable balance drum seal had failed every 3-4 years since installation in 1991
- End gap between seal was increased after 2001 failure
- Latest (2004) overhaul to inspect seal to see if failure imminent

2001



1997



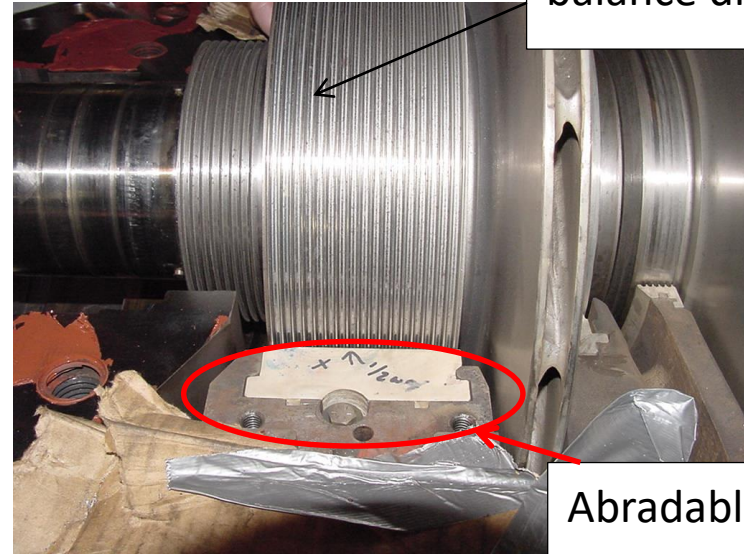
1994



Centrifugal Compressor Instability

Balance Drum Seal

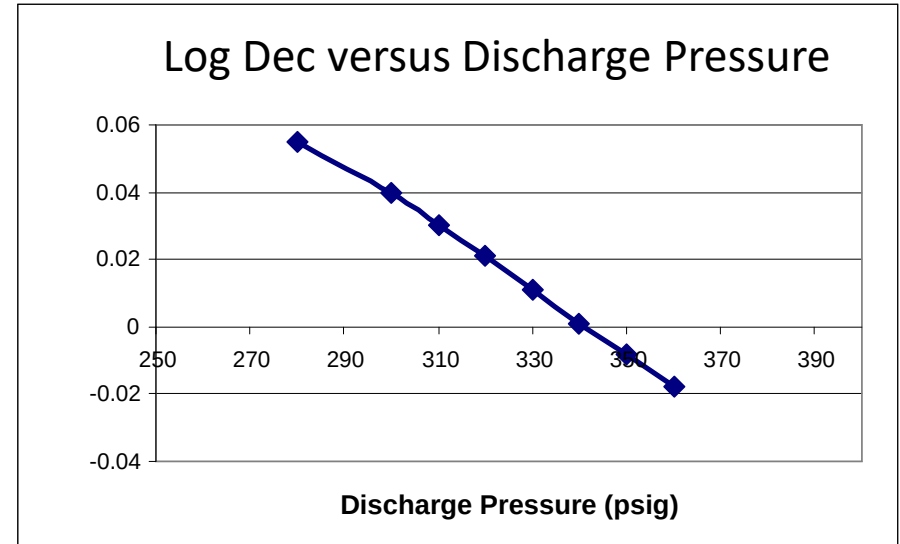
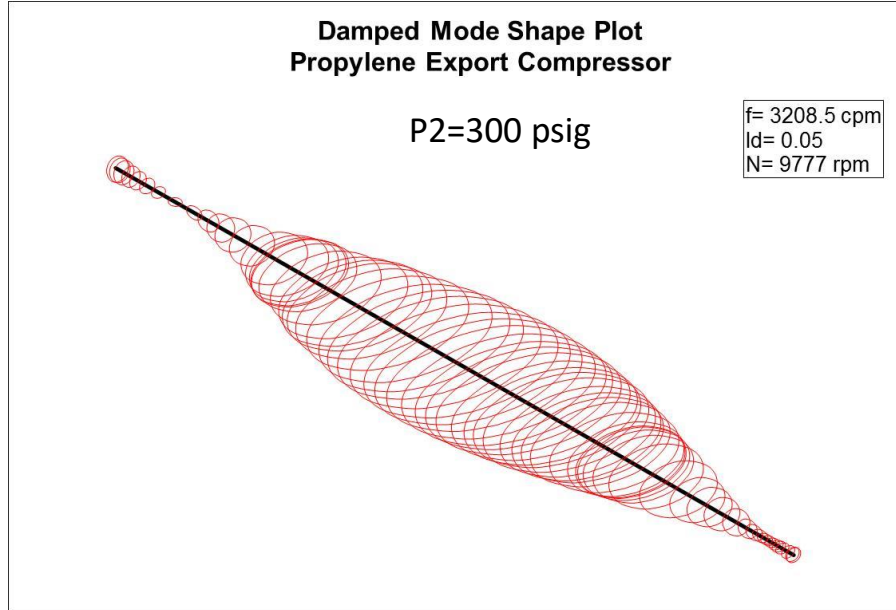
- End gap clearance increased further during latest overhaul (2004) due to multiple failures in past
- Rotor model built to evaluate problem



Rotating
labyrinth on
balance drum

Abradable
stationary seal

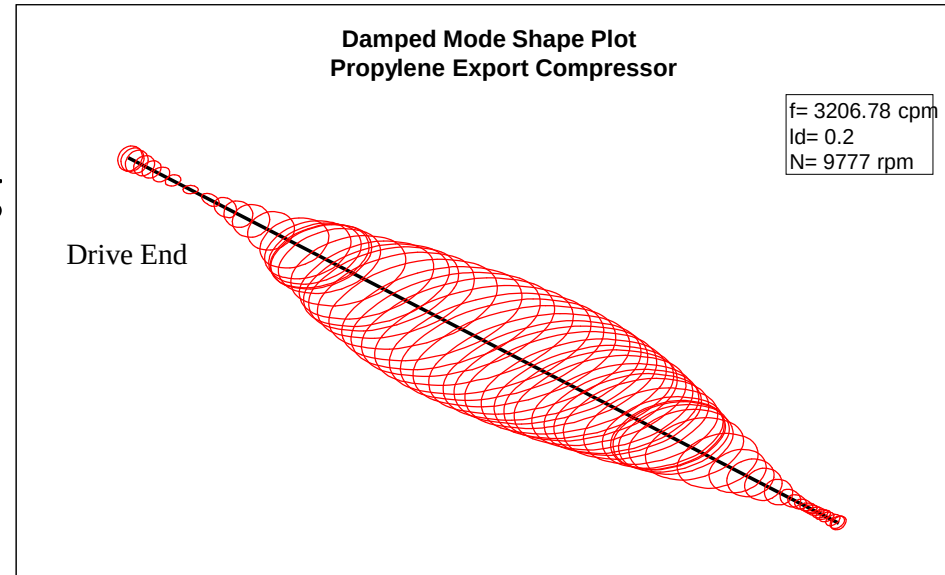
Stability Analysis



- Compressor has low margin of stability

Centrifugal Compressor Instability

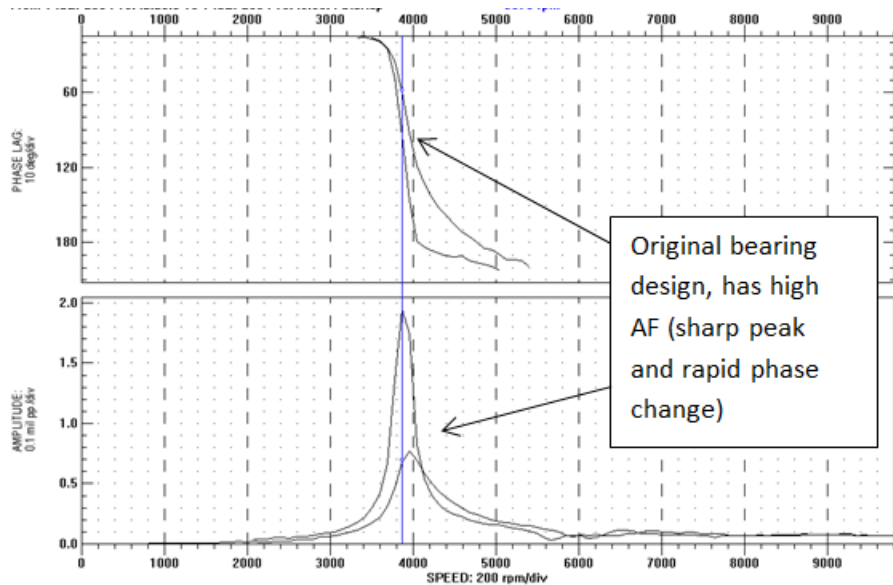
- Balance drum seal was root cause but difficult to replace
- Bearing optimized by increasing L/D, changed configuration to LOP, lowered preload
- Calculated 1st mode log dec increased from 0.05 to 0.2



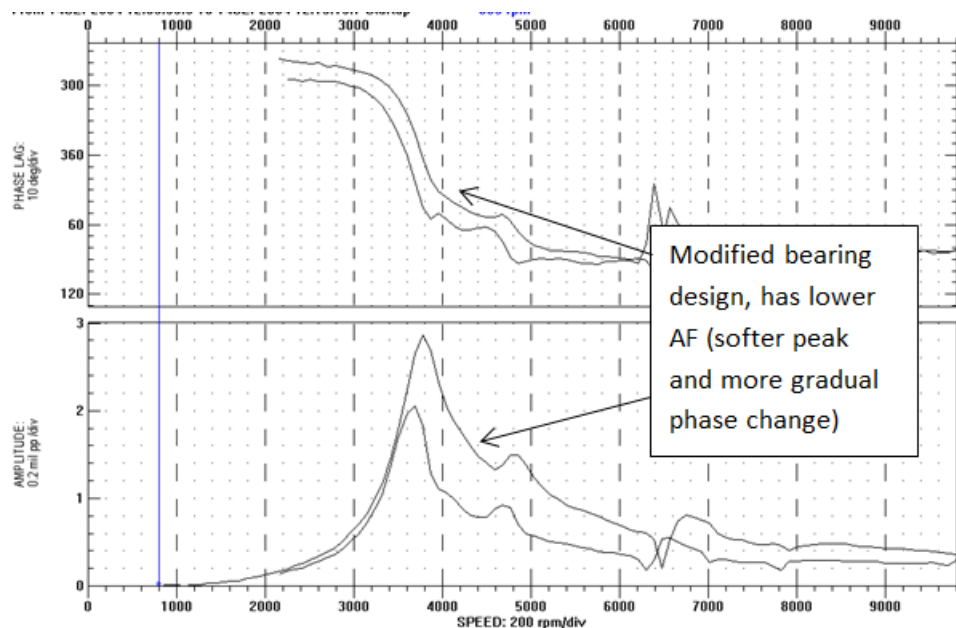
Centrifugal Compressor Instability

At-speed testing

Original LBP bearing

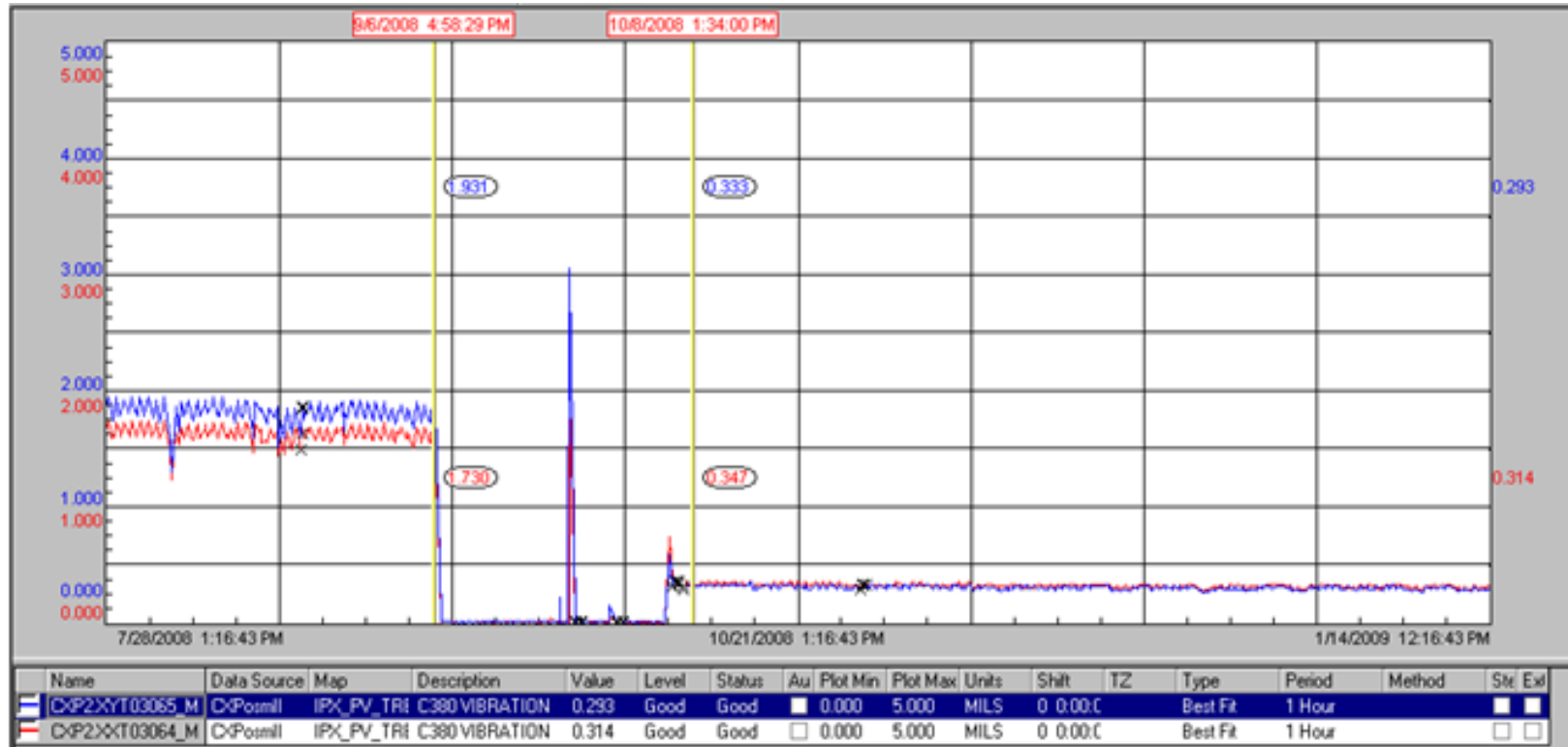


Optimized LOP bearing



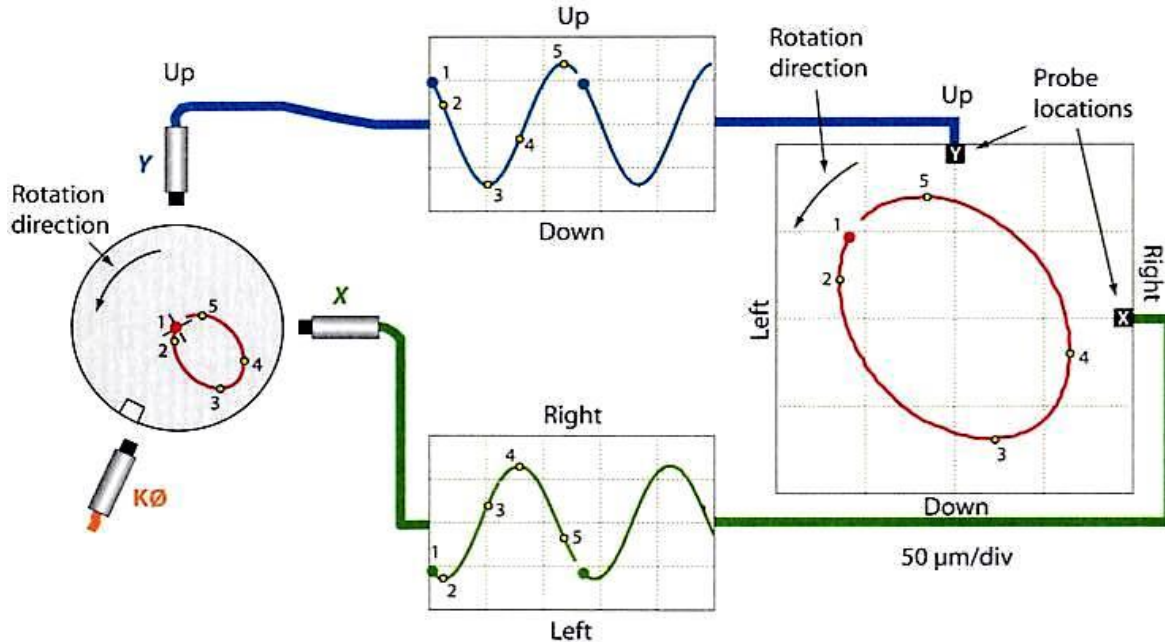
Centrifugal Compressor Instability

Bearing Change Results



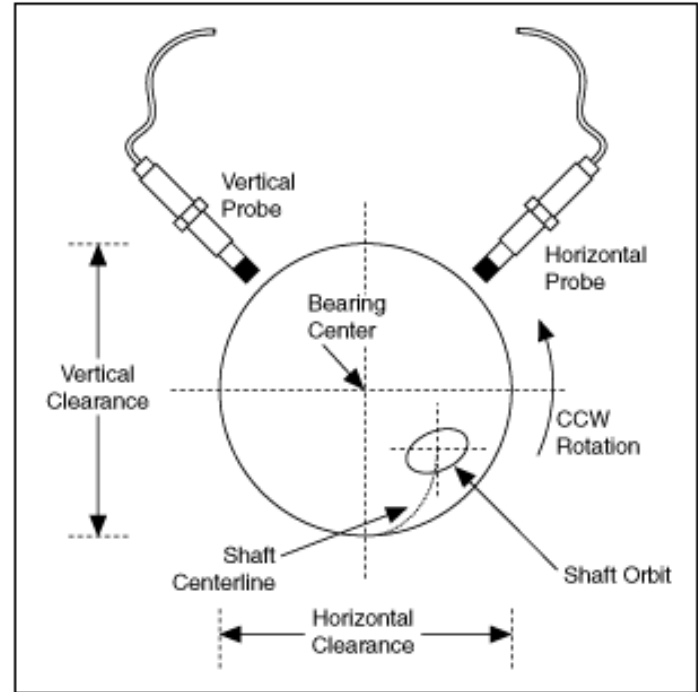
Shaft Orbit

- Orbit shows rotor procession in bearing
- Created by plotting Y vs X time waveform
- Blank-bright sequence shows rotation



Shaft Centerline

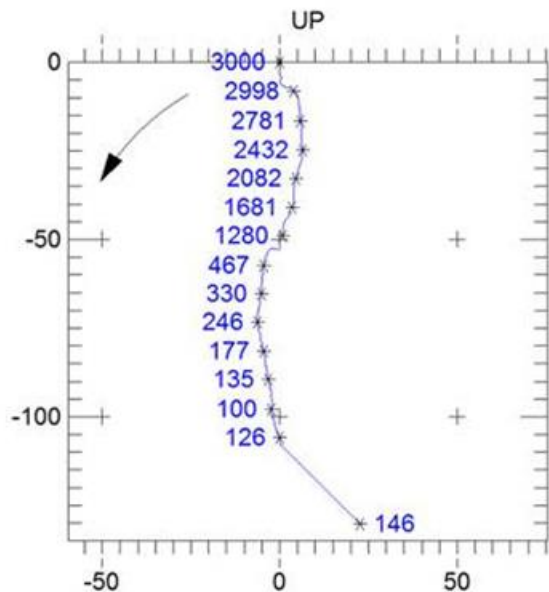
- Similar to orbit, but DC portion used to show center of shaft orbit
- Excellent for showing rubs, excessive clearance, misalignment
- Must be careful with transient thermal effects



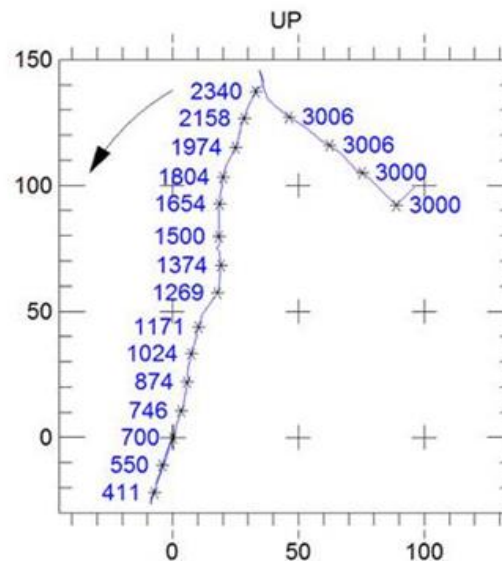
Industrial Gas Turbine

Internal rub

Before

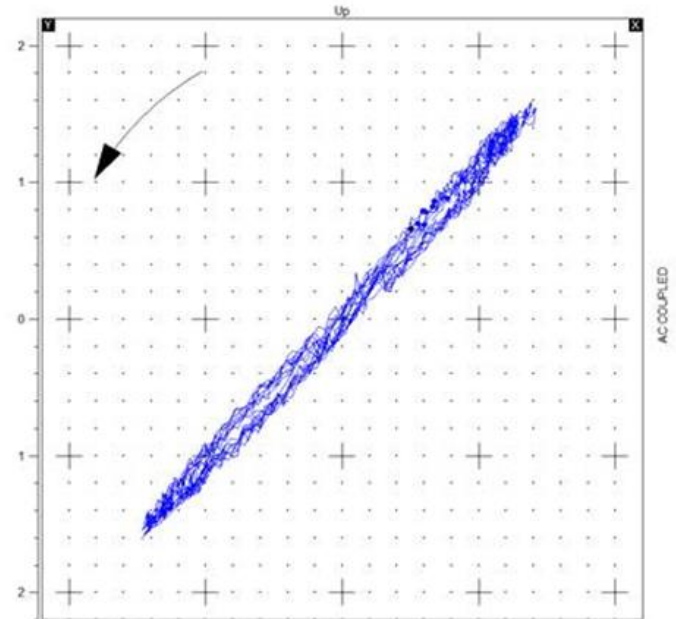
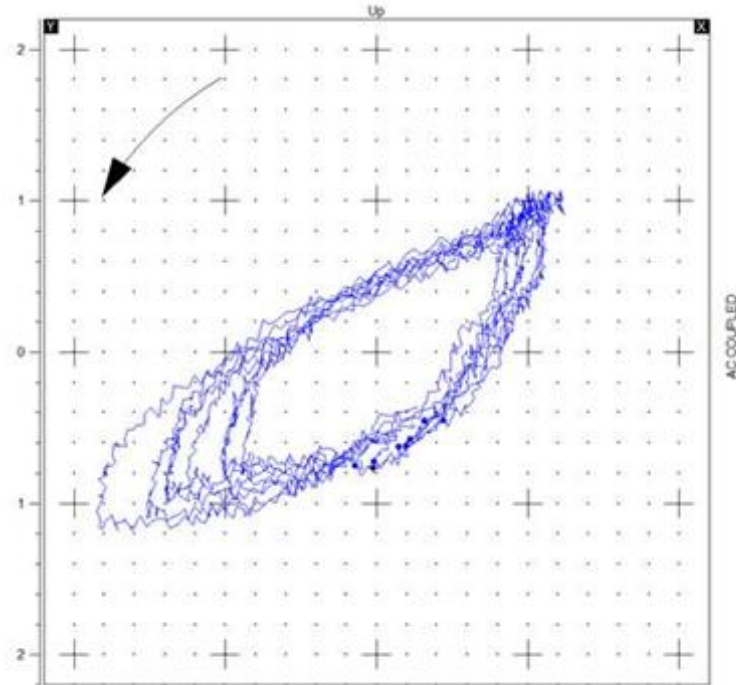


After



Industrial Gas Turbine

Internal rub – progression



Industrial Gas Turbine

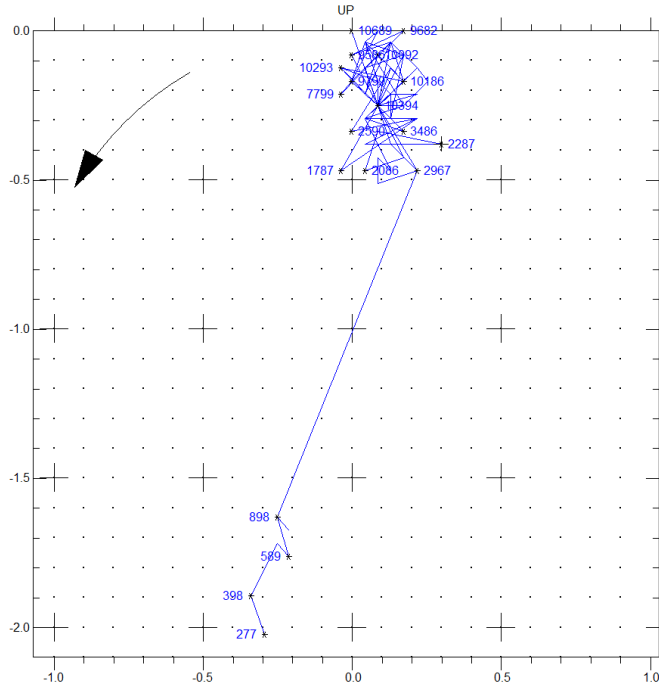
Internal rub

- Inspection revealed damaged #3 bearing
- Alignment change made and bearing replaced

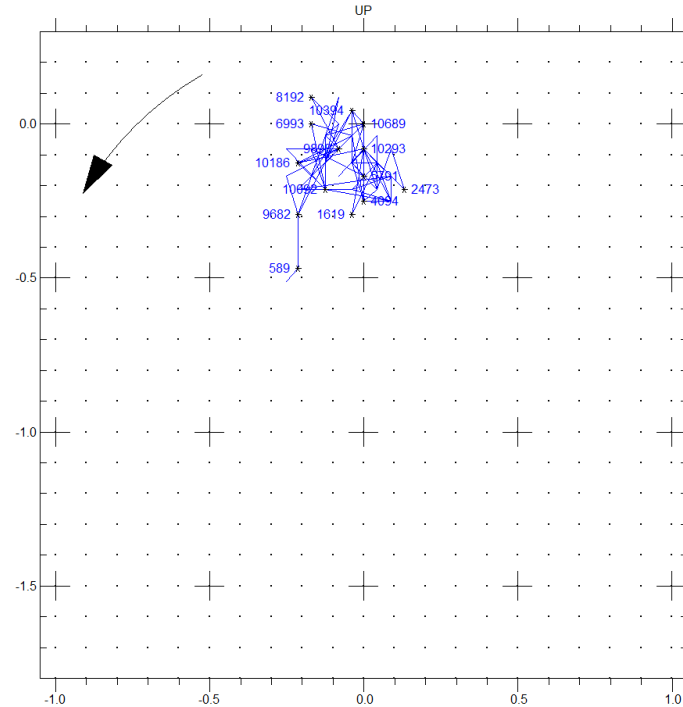


Insufficient bearing clearance

Compressor OB

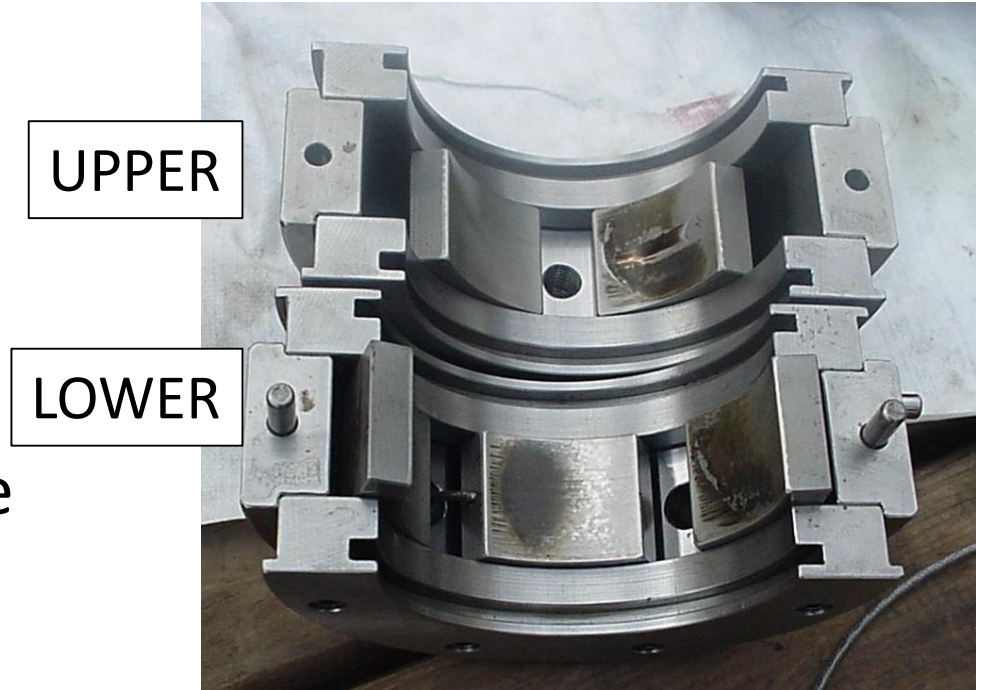


Compressor IB



Shaft Centerline Plot

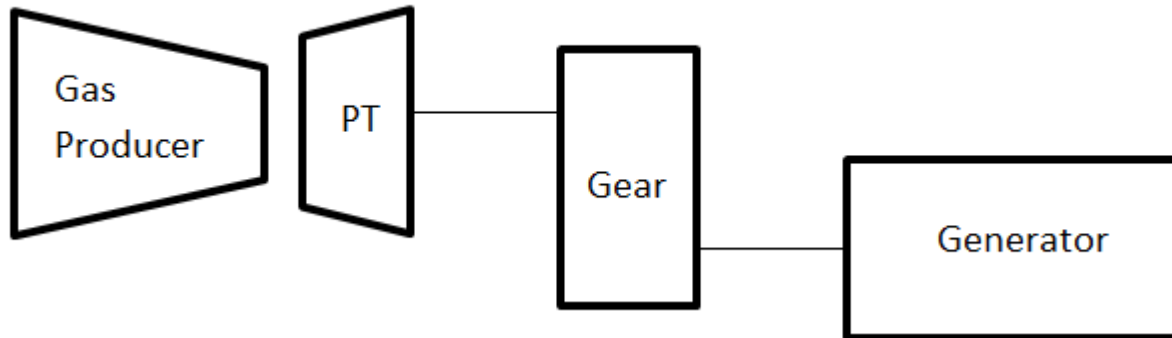
- IB Bearing clearance was too low
- Pads should never be scored in top half of bearing on a beam style compressor



Gas Turbine Generator

High Gearbox Vibration

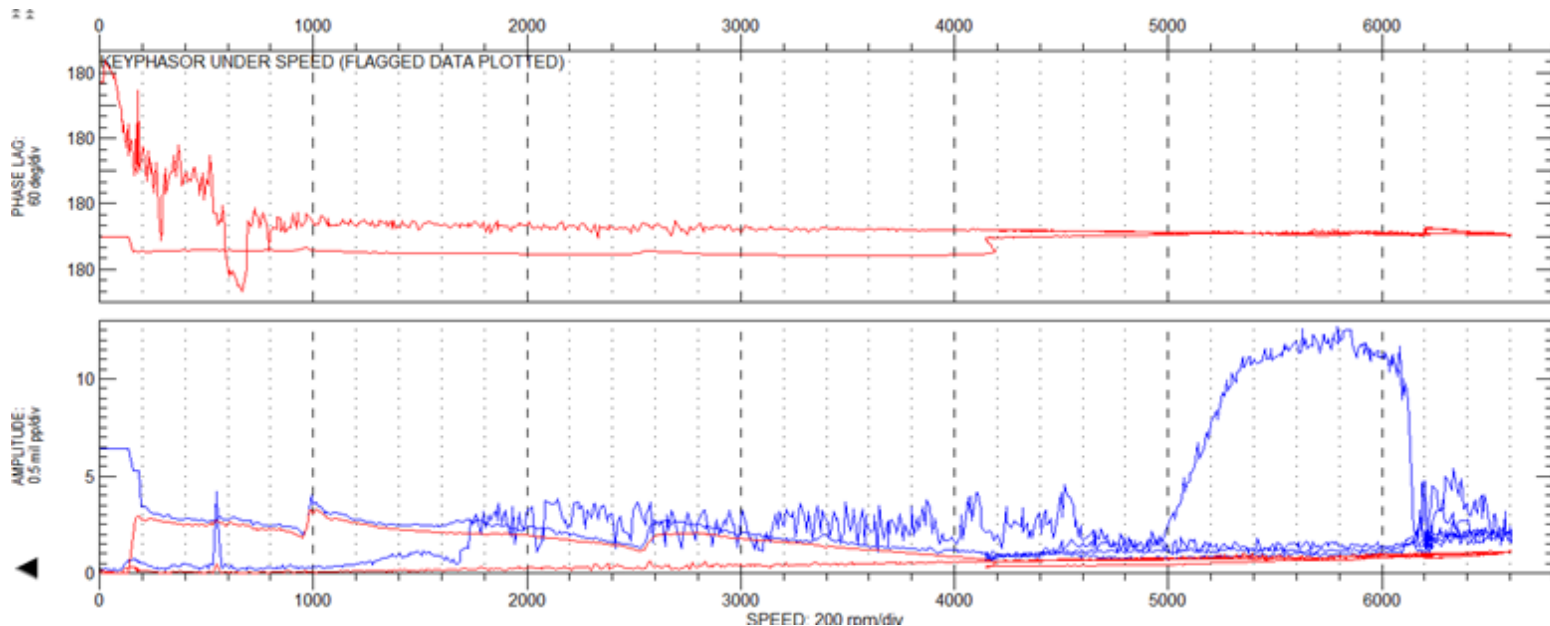
- 30 MW, 2-shaft aero-derivative turbine generator
- PT-6200 rpm, generator-1800 rpm, double helical parallel reduction gear set



Gas Turbine Generator

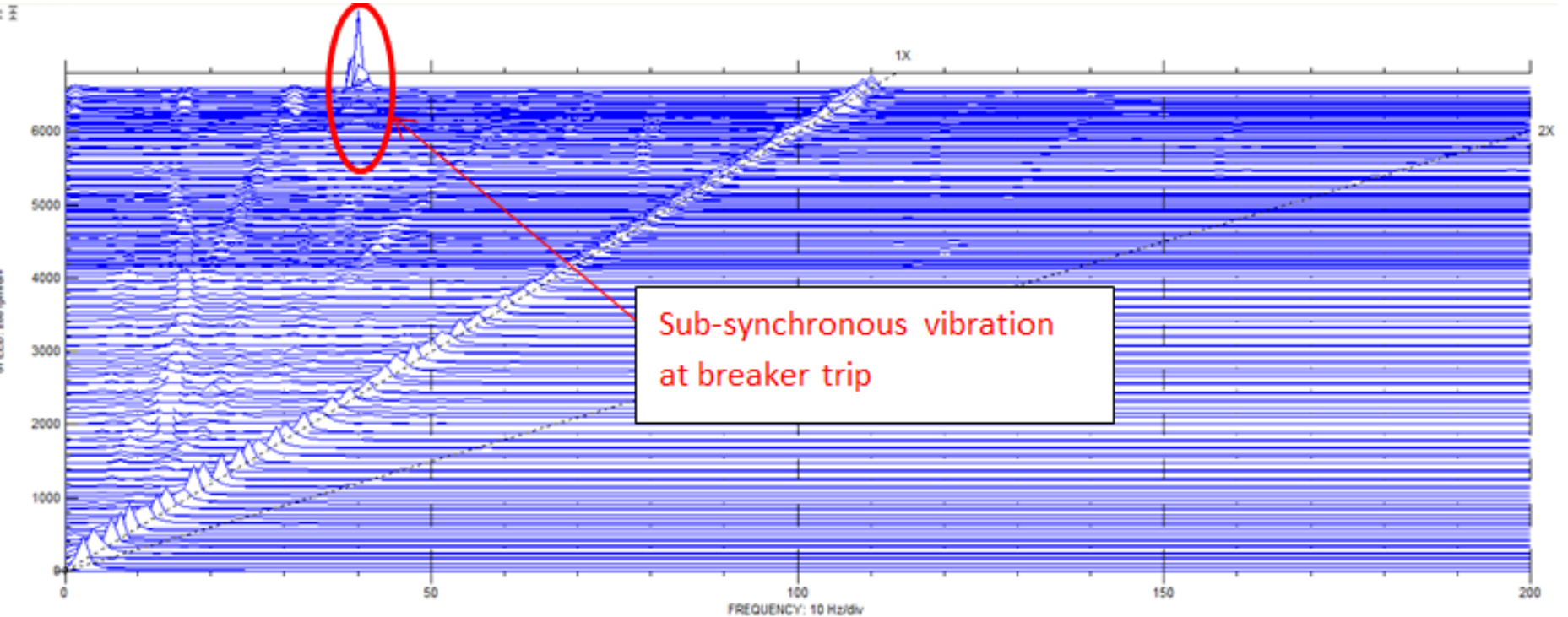
High Gearbox Vibration

- High vibration occurs on pinion during open breaker test (FAT), when load is rejected



Gas Turbine Generator

High Gearbox Vibration



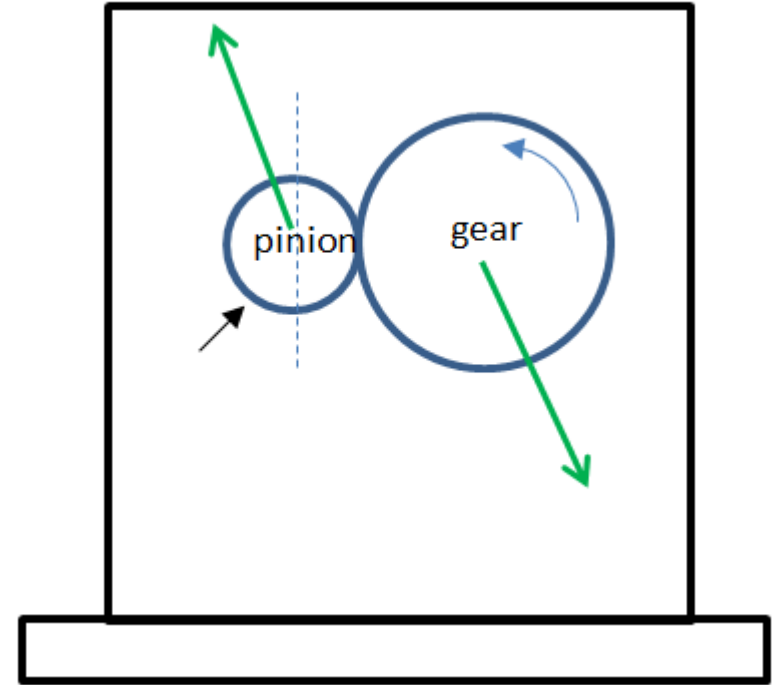
Gas Turbine Generator

High Gearbox Vibration

- Initial hypothesis was a torsional resonance excited by the open breaker sequence.
- Torsional vibration was found to be very low, no torsional critical at 40 hz.

Parallel Gearbox (speed reduction)

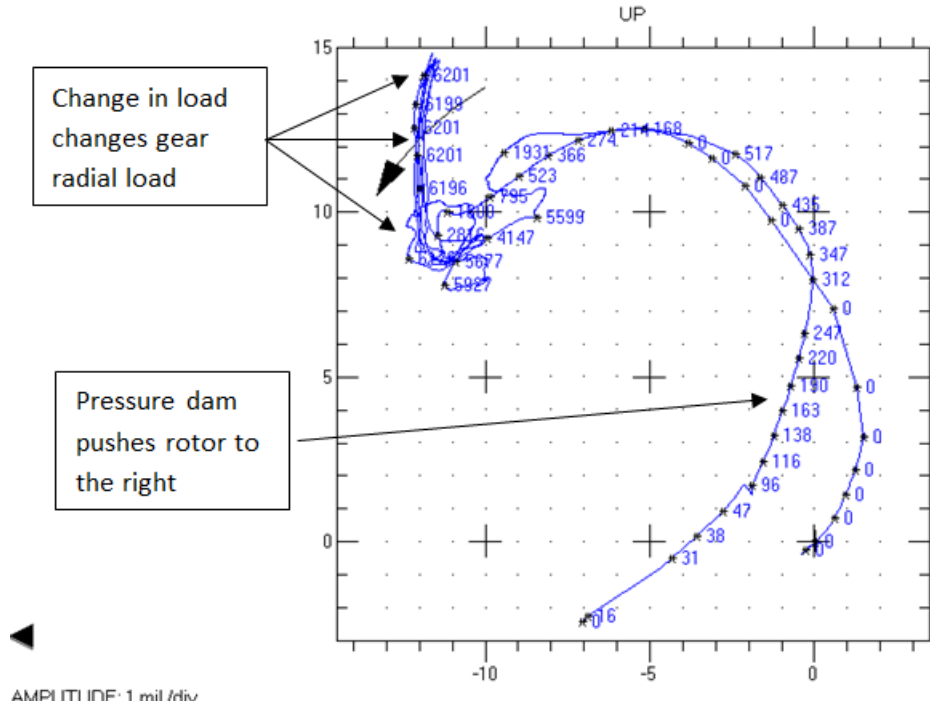
- Gear radial load pushes down on bull-gear and up on pinion
- Gear load more than enough to lift weight of pinion and keep it stable (no whirl)
- Pressure dam bearing in lower half of pinion used to load pinion up at low power conditions



Gas Turbine Generator

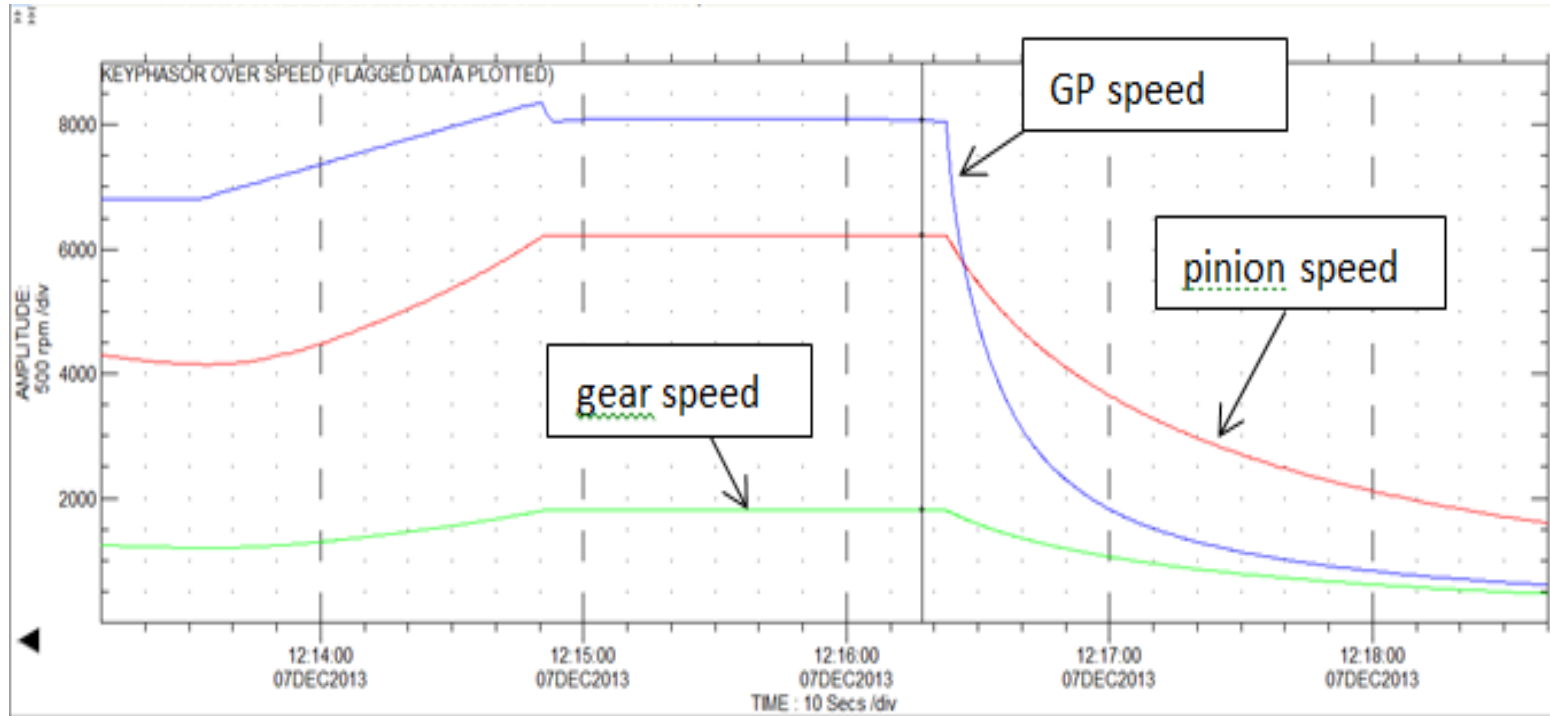
High Gearbox Vibration

- Shaft centerline plot shows effects of:
 - changes in load
 - pressure dam bearings



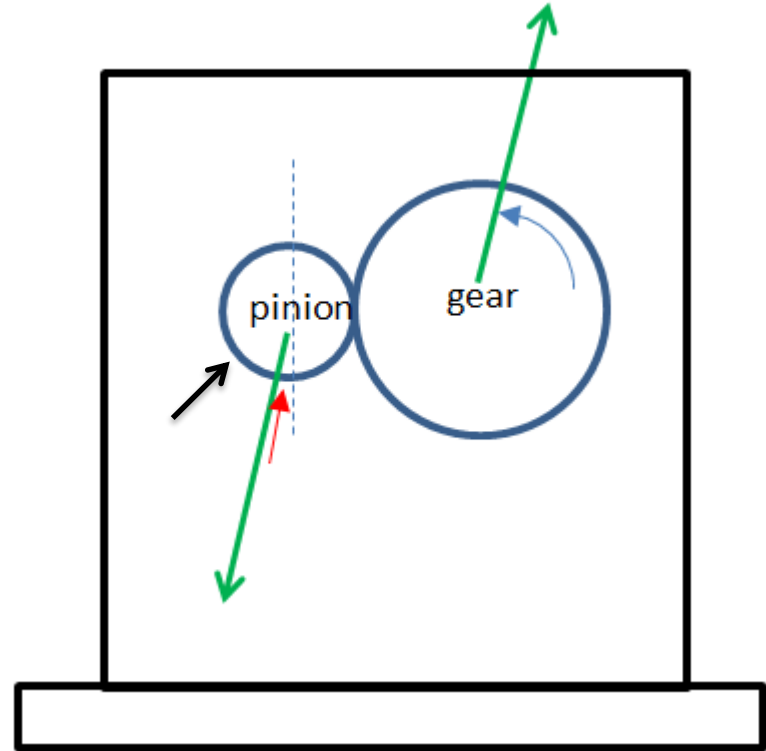
High Gearbox Vibration

- Speed trends show that open breaker test causes inertia of generator to act as a brake



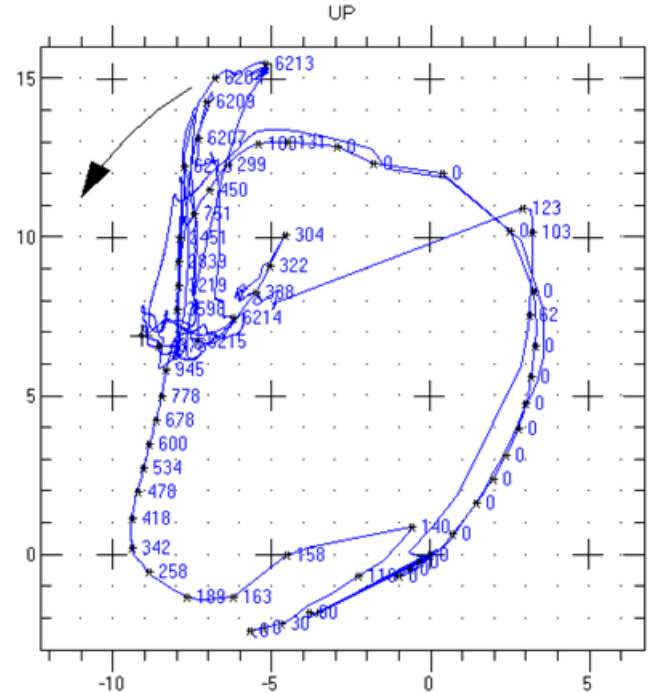
Pinion Bearing Modifications

- Open breaker reverses gear load and pushes down on pinion
- Pressure dam in lower half of pinion bearing rotated to 20° so that it is opposite gear load during reversal



High Gearbox Vibration

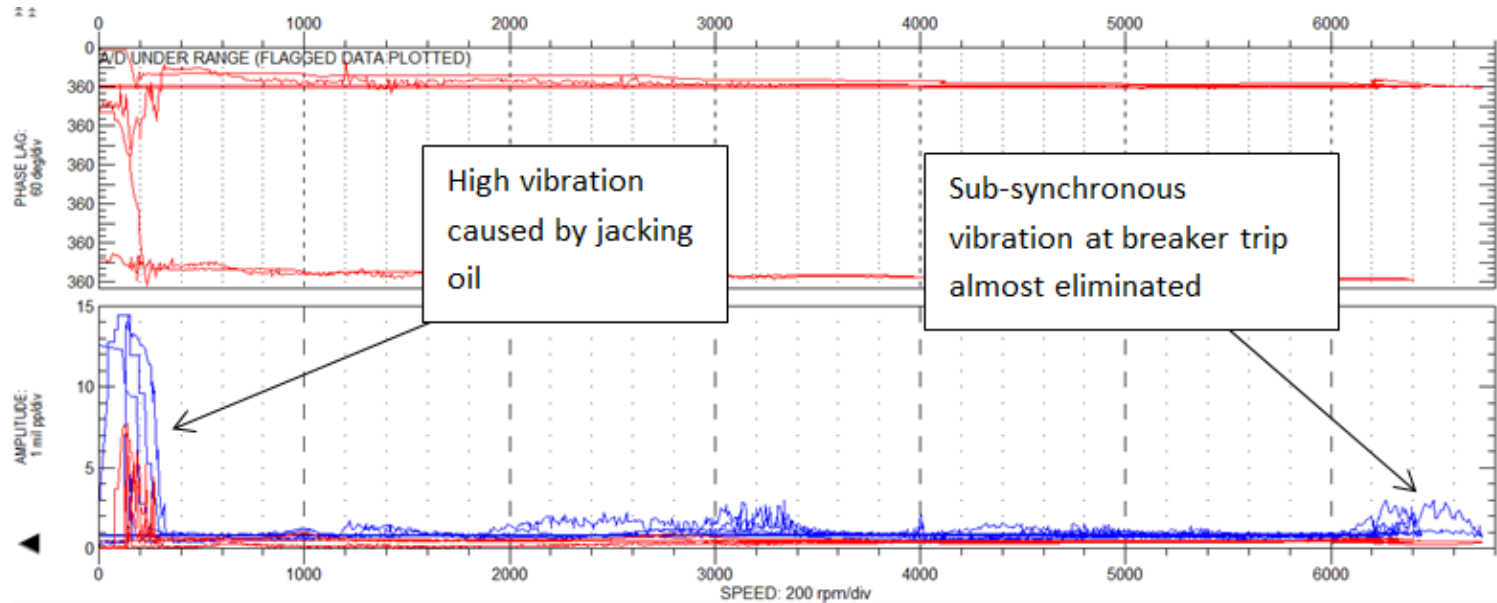
- Note that pinion comes almost straight down after pressure dam re-located.
- Load changes still present in shaft centerline



Gas Turbine Generator

High Gearbox Vibration

- Bearing modifications minimize sub-synchronous during open breaker test



Conclusions

- Complexity of turbomachinery normally requires the use of multiple plots to properly diagnosis vibration problems.
- Understanding of rotordynamics and the ability to build rotor models key to designing correct solutions.

Questions?