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

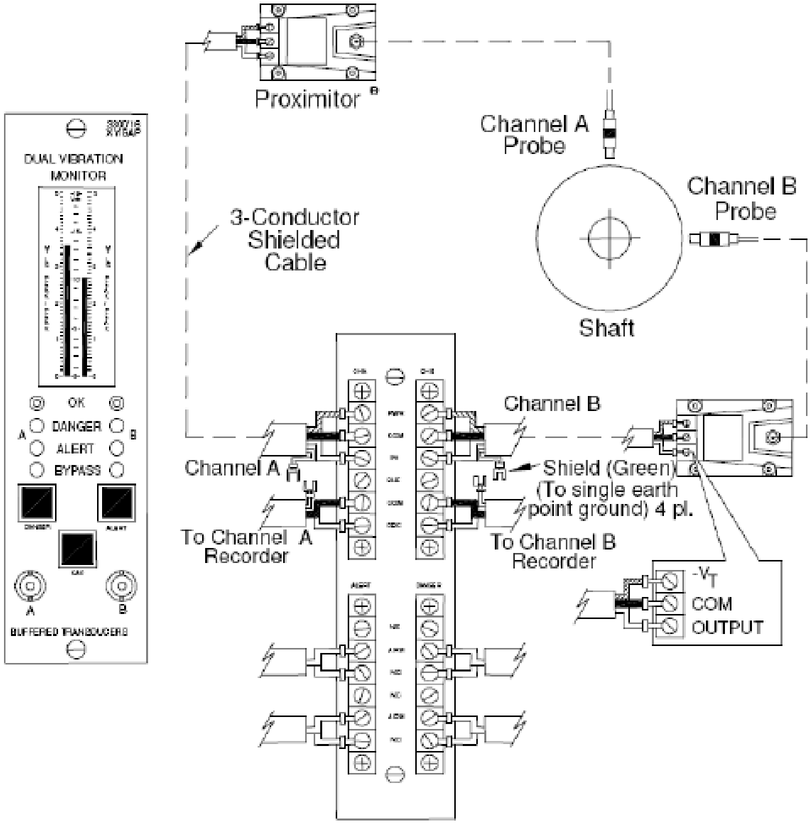
General Purpose vs Turbomachinery

- 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.

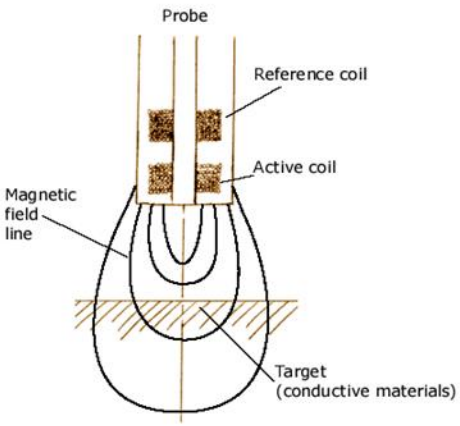
Measurement
Proximity Probes

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

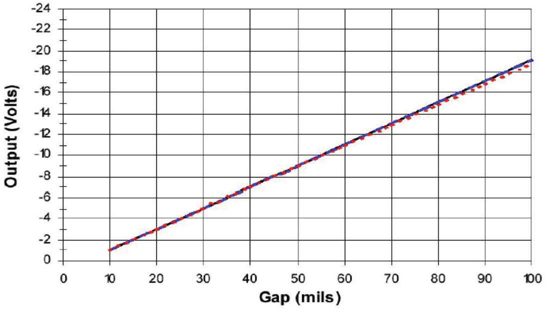


Proximity Probes

- Proximitor
  - excites the probe creating a magnetic field which is affected by the target(shaft)
  - produces a voltage proportional to the shaft displacement.



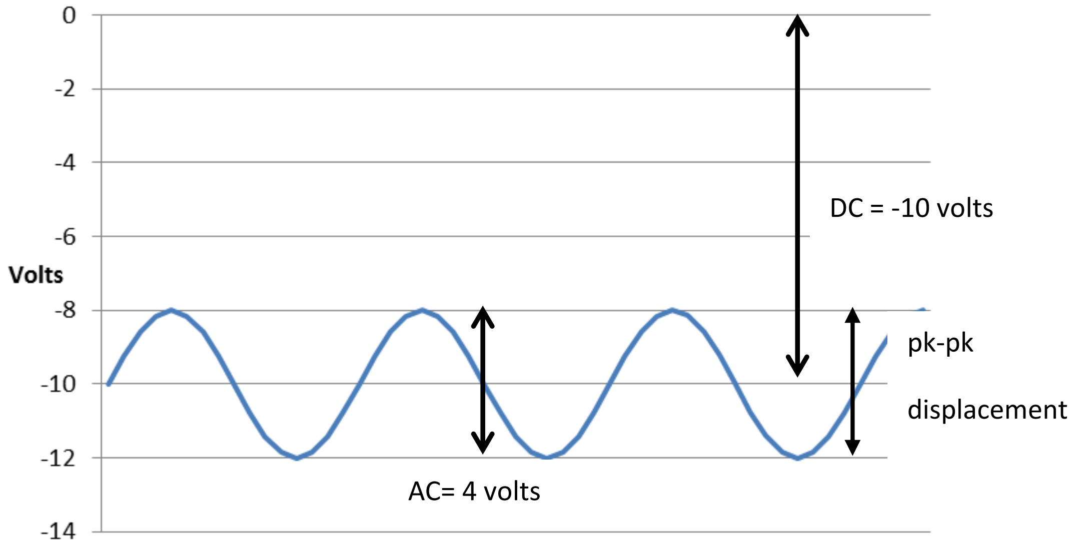
- Properly installed and maintained – very rugged and reliable



- Default sensitivity for 4140 is 200 mV/mil

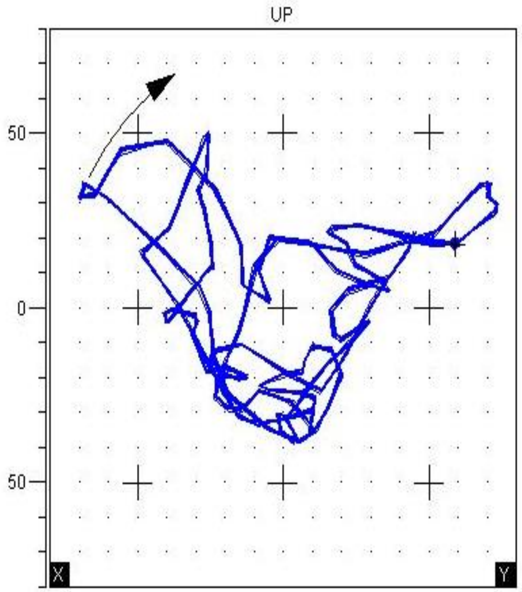
Proximity Probe Output

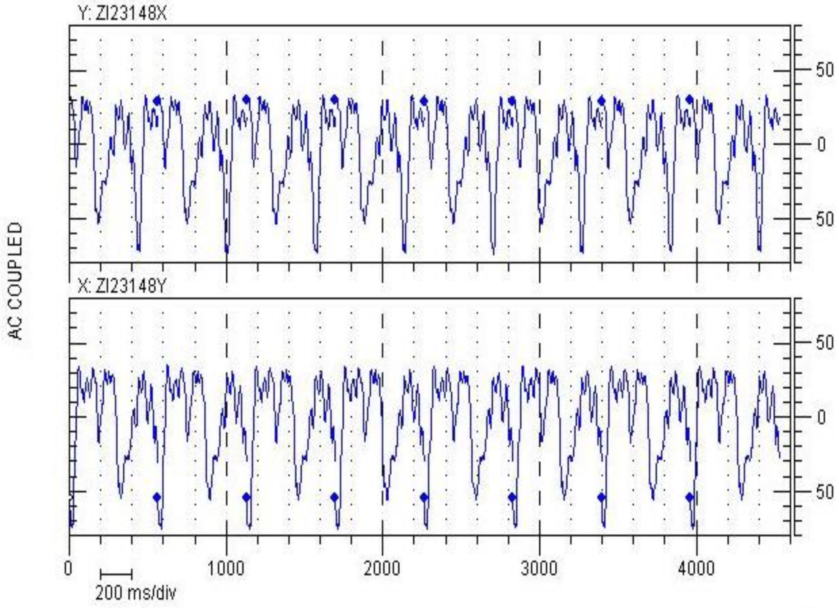
- DC component represents shaft position
- AC component represents shaft vibration



Proximity Probes - Pitfalls

- Target surface condition – scratches show up as vibration





Proximity Probes - Pitfalls

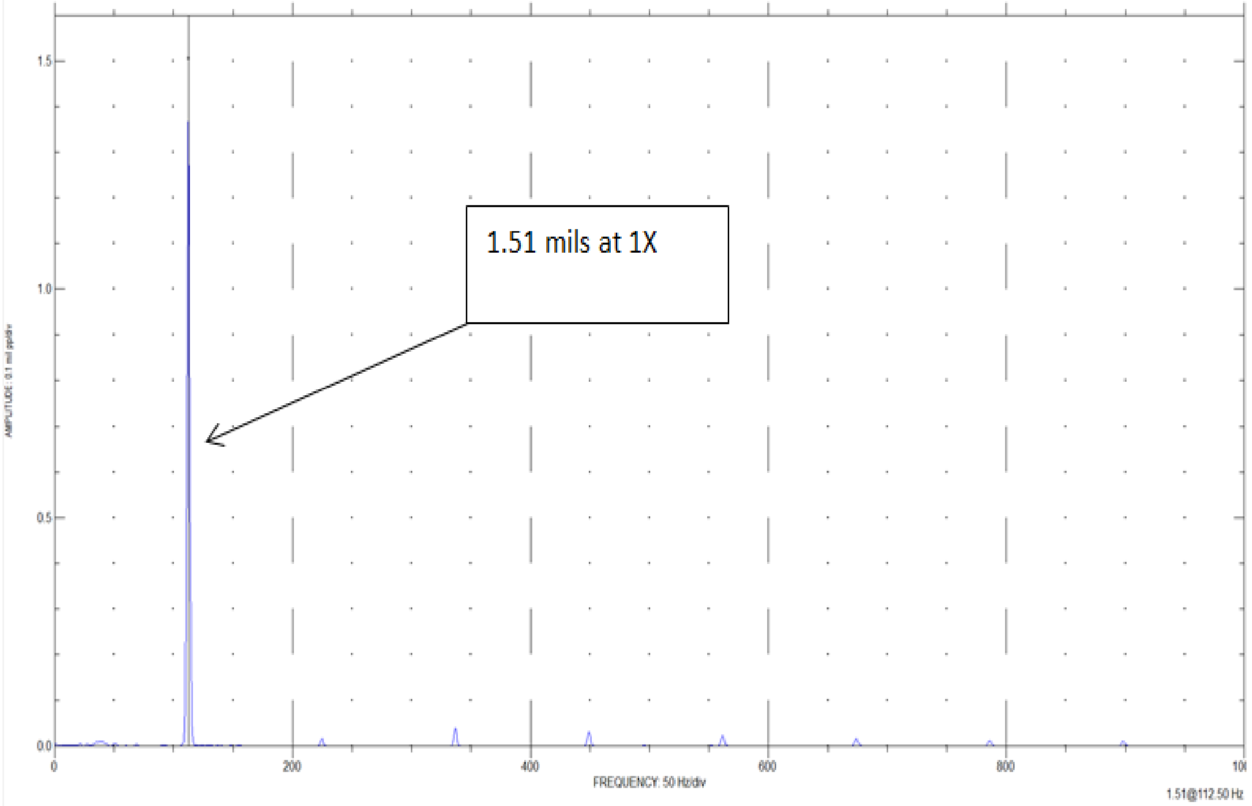
- Magnetism – if a residual magnetism exists in the shaft or target, this will affect the probes

- Incorrect target material – coatings and/or non-ferrous materials will affect probe readings

- Wrong extension cable – proximitor, extension cable, and probe are a matched set (total lengths of 1, 5, or 9 m)

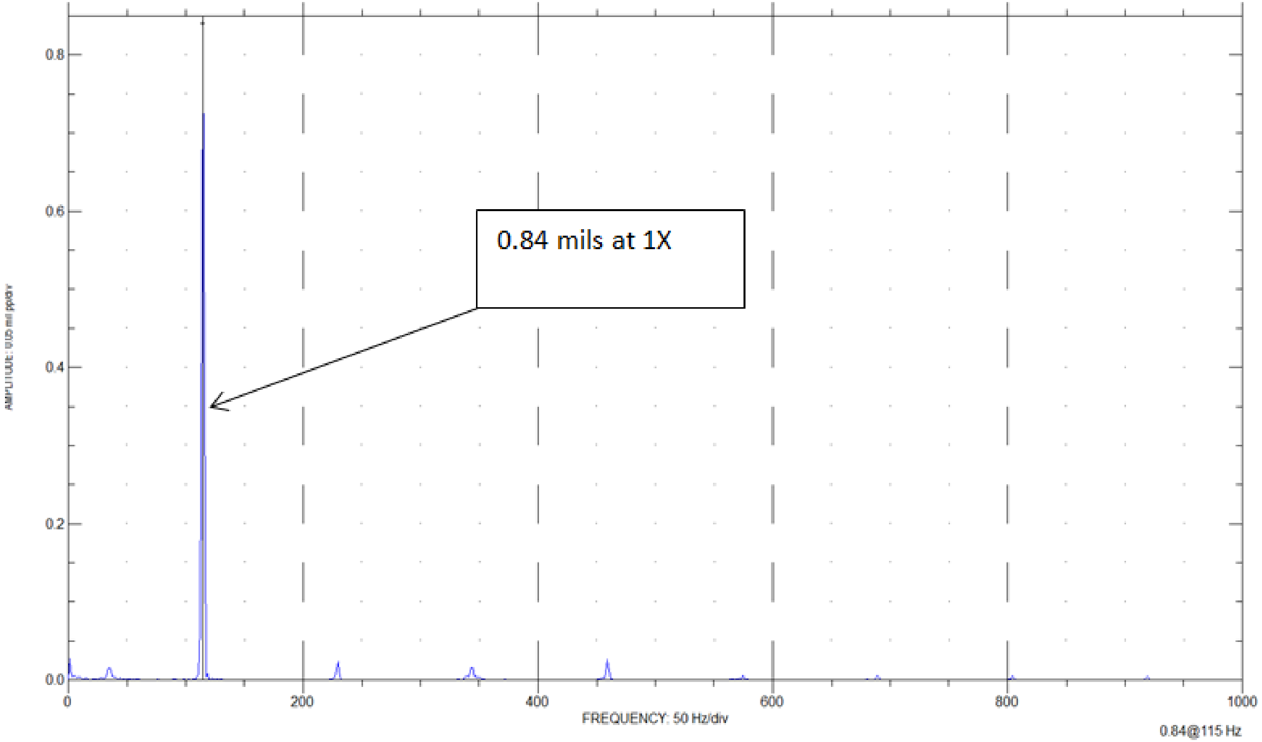
Proximity Probes - Pitfalls

- Rotor kit with correct proximitor, no extension cable



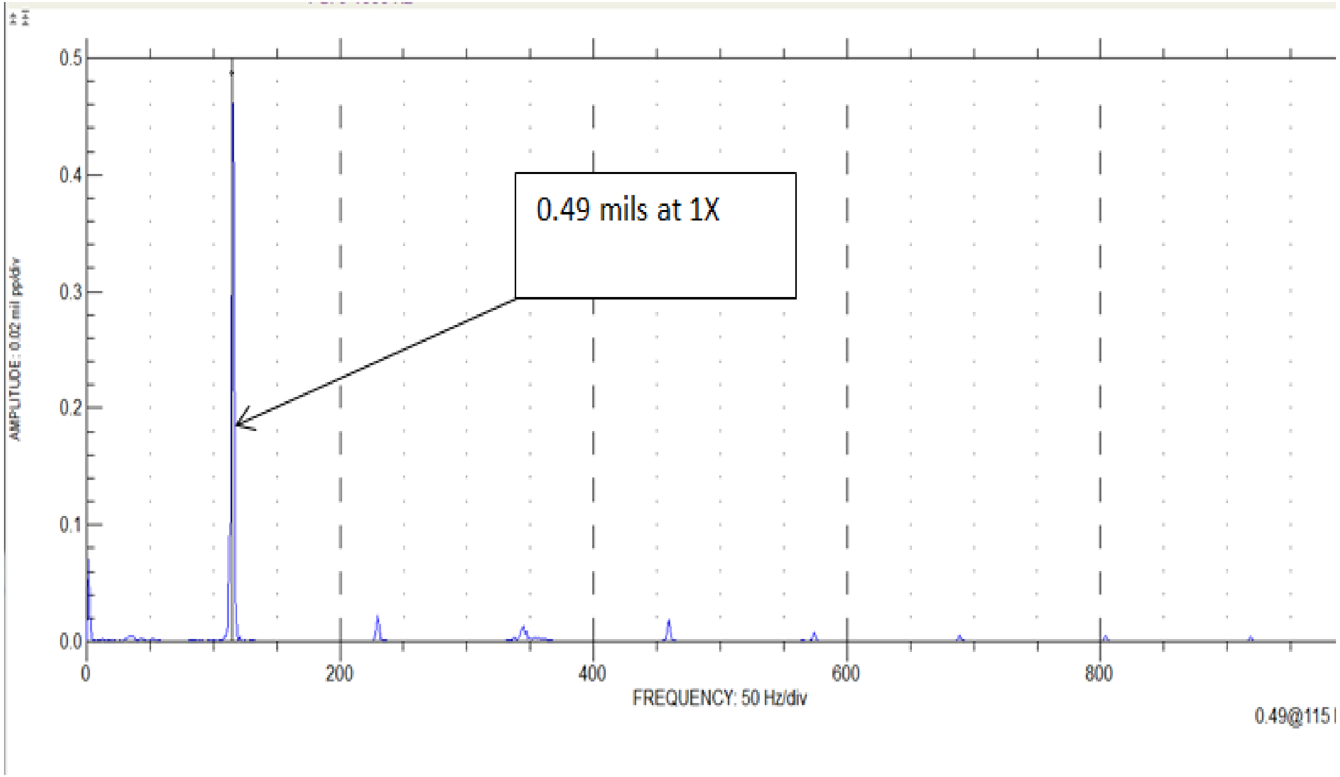
Proximity Probes - Pitfalls

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



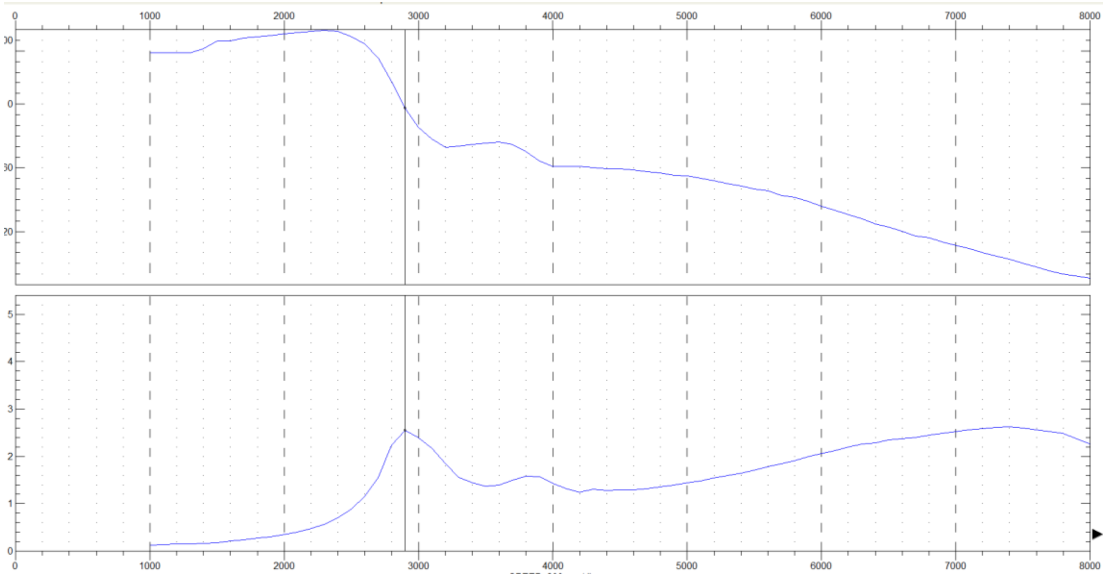
Proximity Probes - Pitfalls

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



Bode Plots

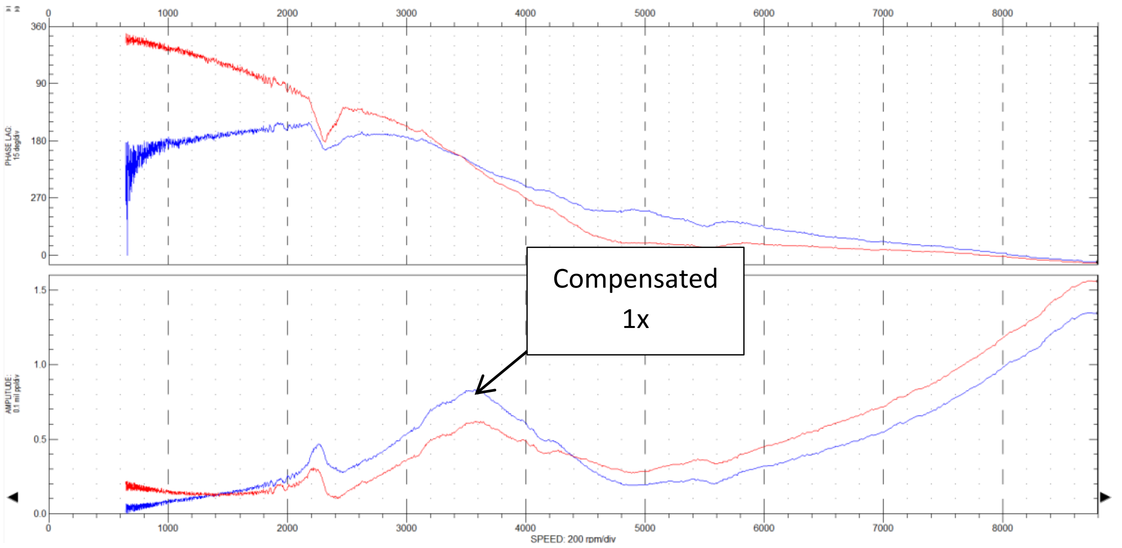
- Vibration amplitude vs speed
- Critical shown by peak amplitude and phase change
- Uses
  - Determine natural frequencies (modes)
  - Indication of system damping
  - Tune rotordynamic models



Bode Plot - Compensation

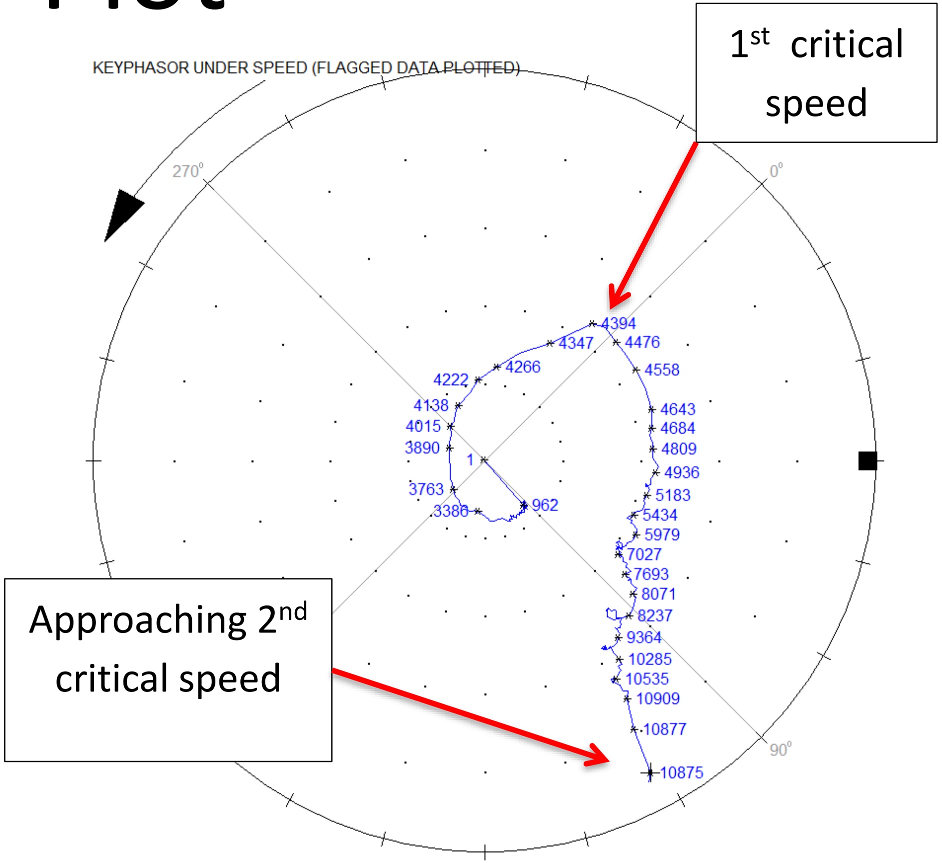
- Runout Vector – probe output at slow speed (<500 rpm), not vibration

- Compensated – run-out vector subtracted from raw reading



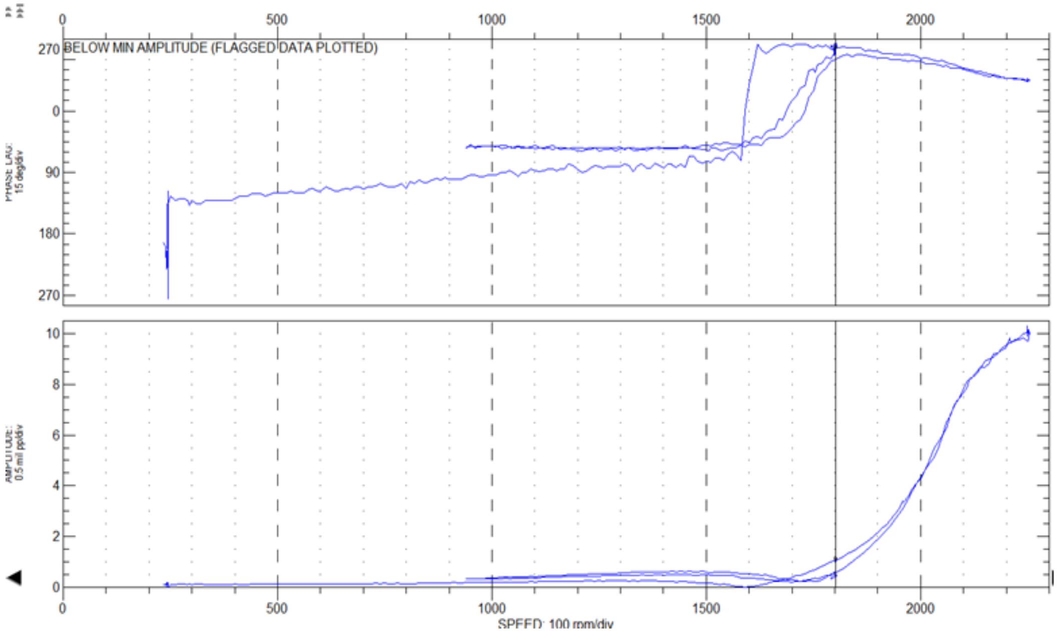
Polar Plot

- 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

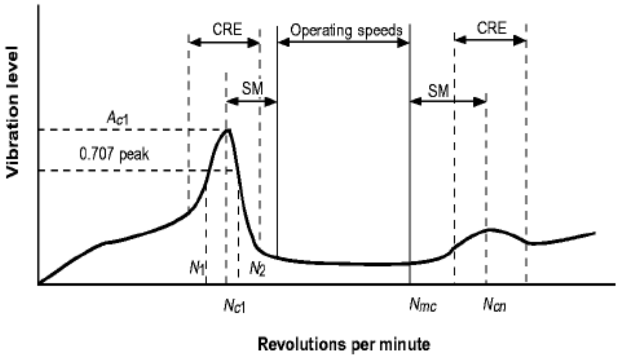
- 4 pole generator FAT
- Identified 1st critical only 400 rpm above running speed during overspeed test..



- Problem could occur if natural frequency drops close to running speed.

Amplification Factor

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

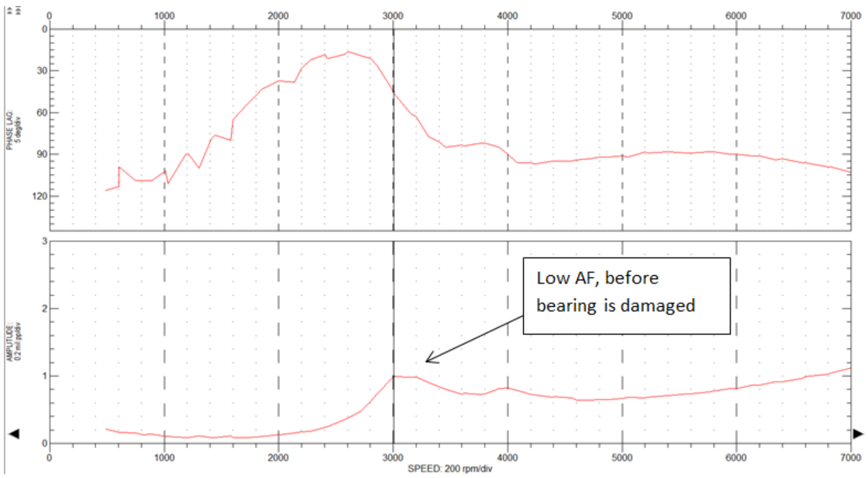


N_{c1} = Rotor first critical, center frequency, cycles per minute.
N_{cn} = Critical speed, nth.
N_{mc} = Maximum continuous speed, 105%.
N_1 = Initial (lesser) speed at 0.707 × peak amplitude (critical).
N_2 = Final (greater) speed at 0.707 × peak amplitude (critical).
$N_2 - N_1$ = Peak width at the 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}.

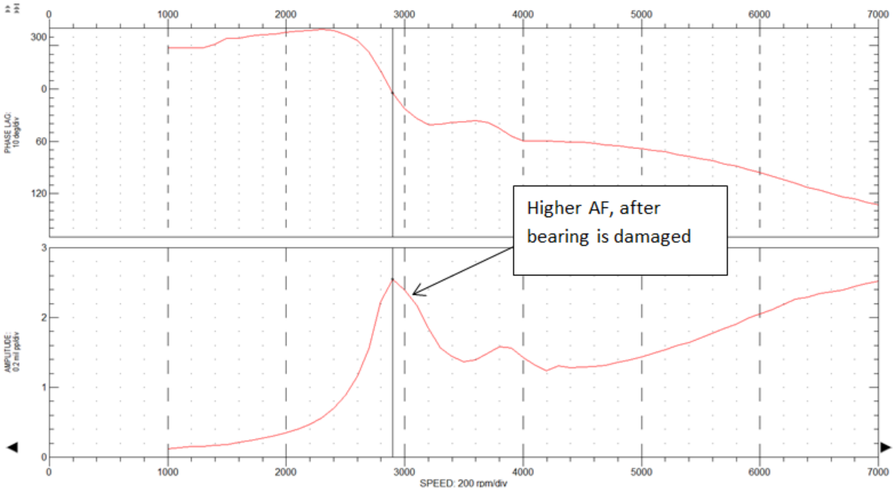
Amplification Factor

Detecting Bearing Damage in Centrifugal Compressor

Before Trip



After Trip



Amplification Factor
Detecting Bearing Damage

- Bearing showed evidence of loss of lube

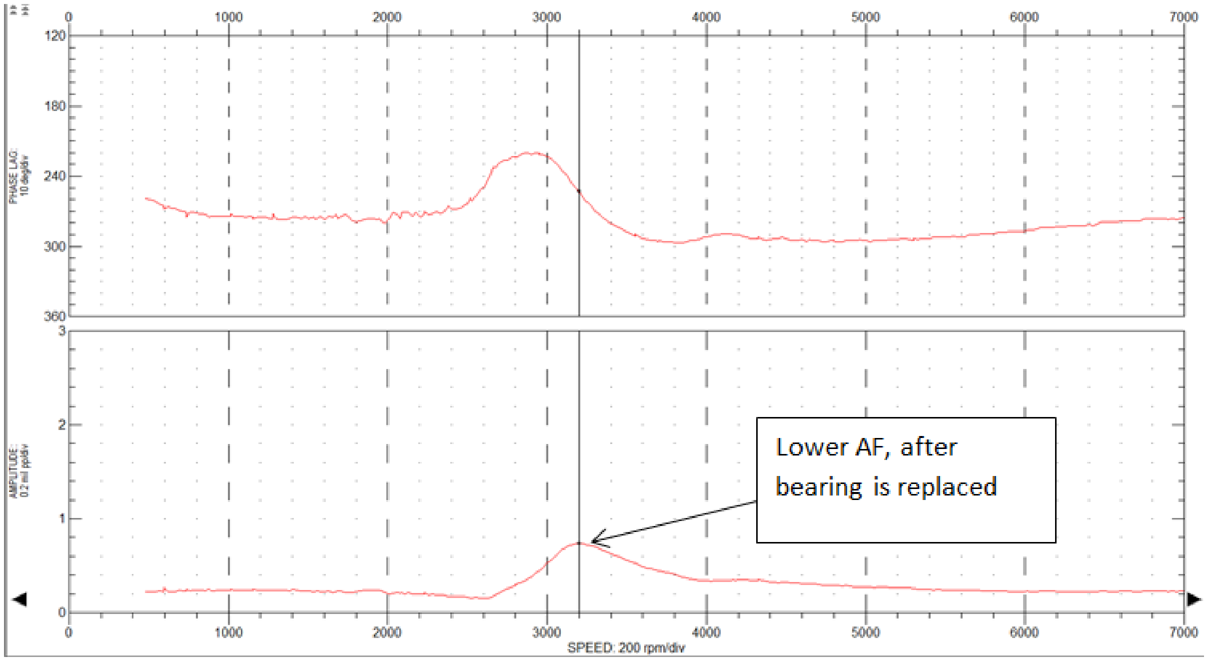
- Clearance was 30 % above maximum

- Increase in clearance reduces damping and increases AF

Amplification Factor

Detecting Bearing Damage

- Both radial bearings replaced
- AF returns to normal level



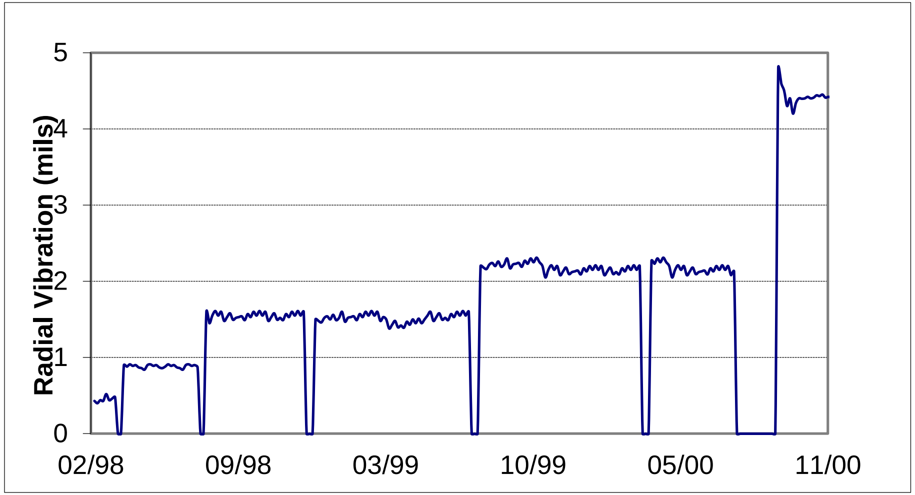
Amplification Factor
Unbalance

Barrel Compressor

- Radial vibration increases after some trips, but not all

- No indication of bearing damage

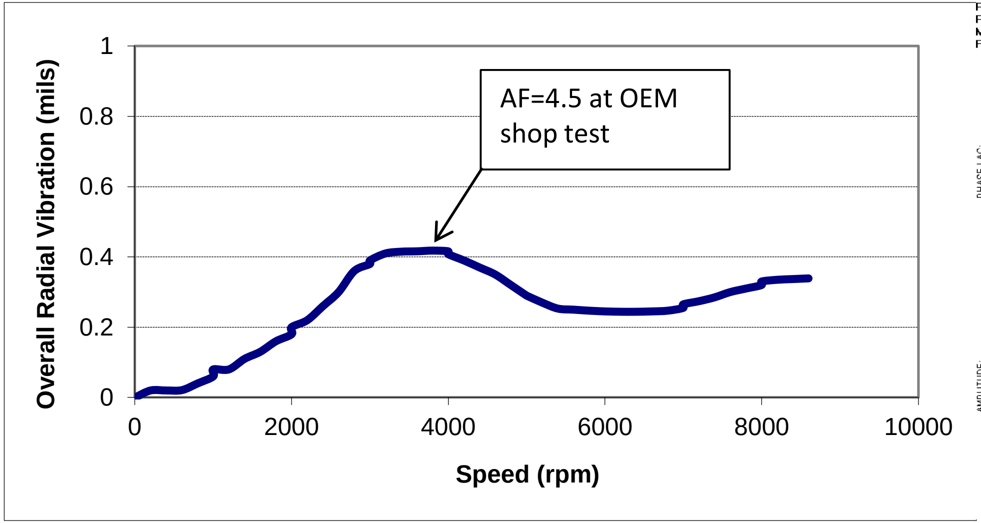
- No loss of lubrication



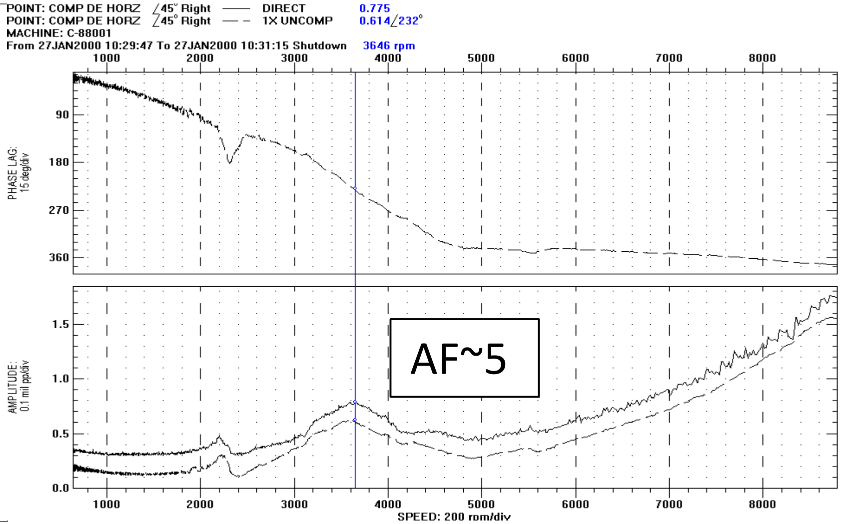
Amplification Factor

Unbalance

FAT Test



2 Years later

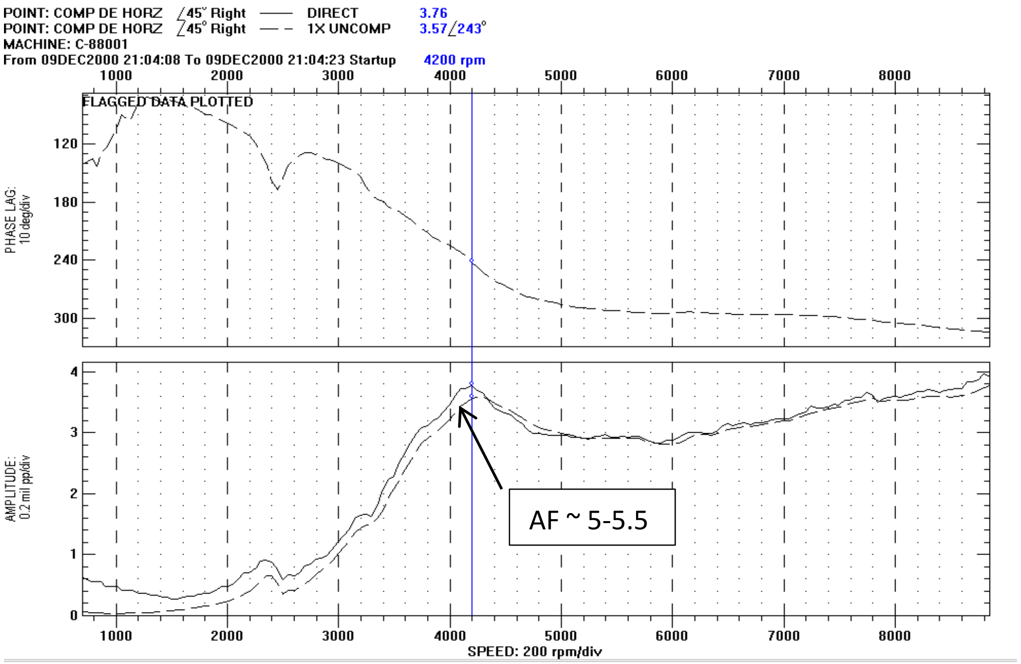


Amplification Factor

Unbalance

- After extended plant outage, compressor restarts with high radial vibration

- Rotor inspection shows large amount of fouling, cleaned returned to service

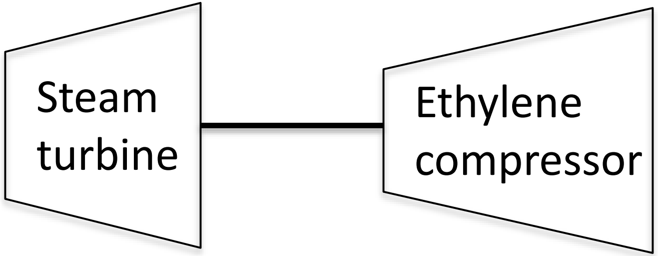


Tune Rotordynamic Models

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

High Speed Steam Turbine

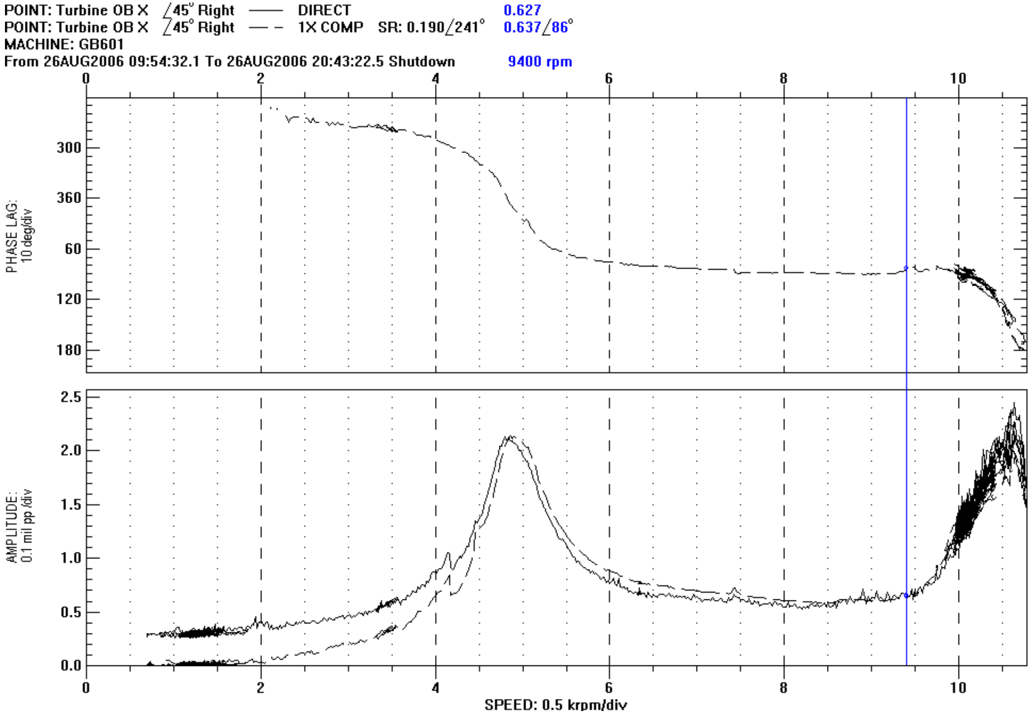
- 8,000 hp steam turbine, driving ethylene refrigeration compressor
- Turbine normally operates between 8,500-9,500 rpm
- High radial vibration on outboard end during overspeed trip testing.



Tune Rotor Models

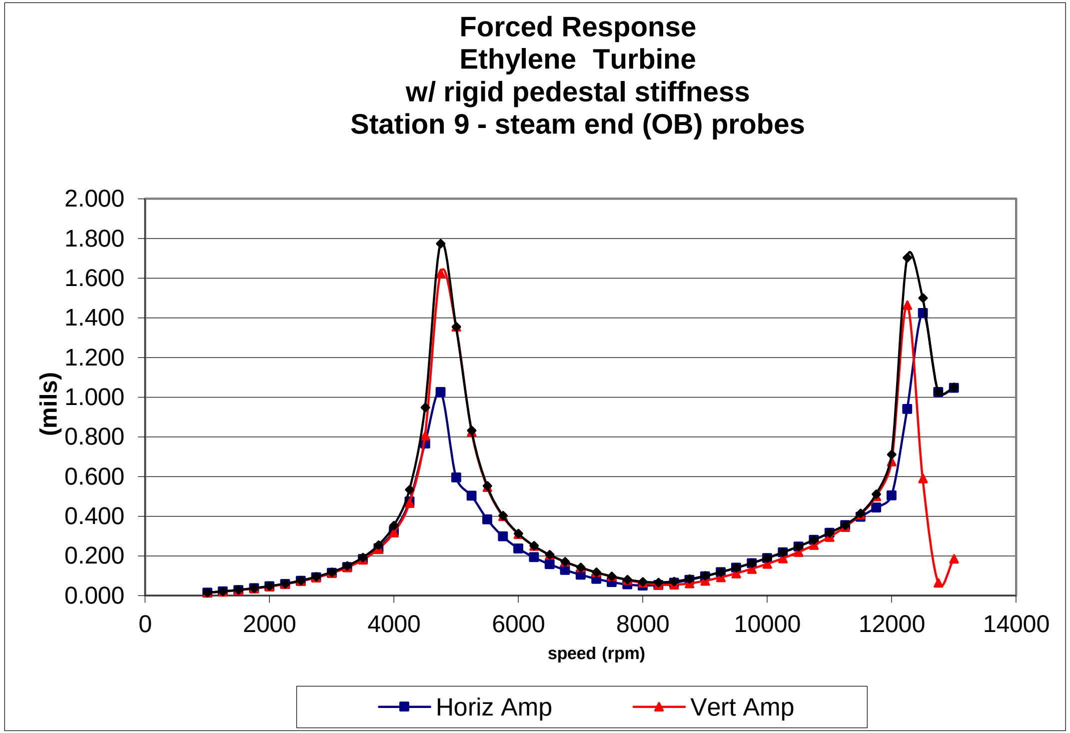
- High radial vibration on steam turbine outboard end during overspeed trip testing, 10,700 rpm

- Turbine normally operated below 10,000 rpm with vibration < 1 mil



Tune Rotor Models

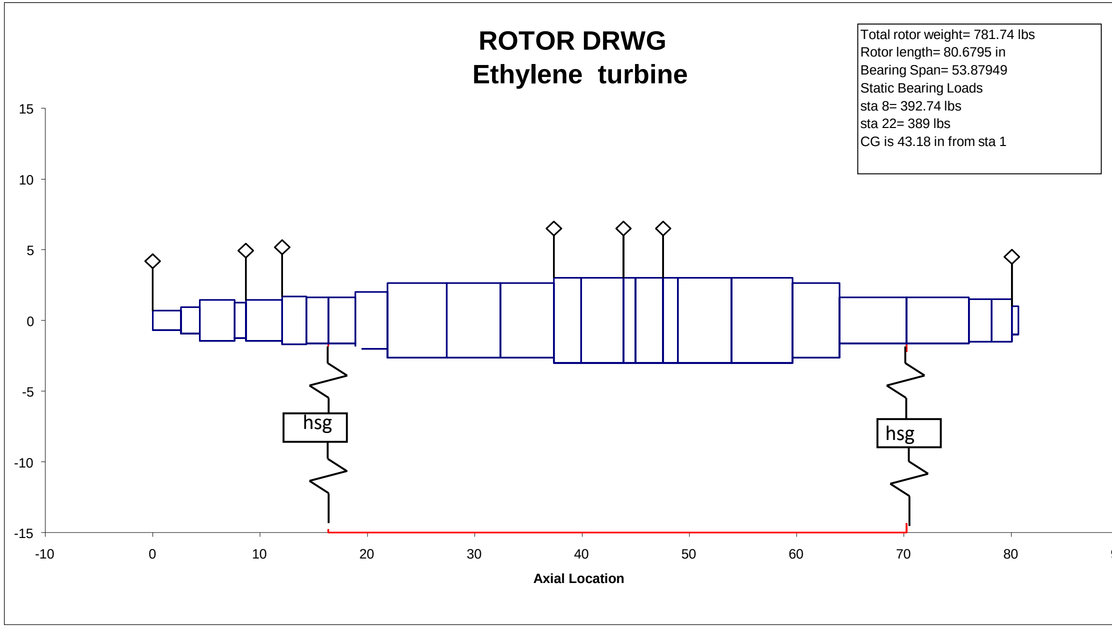
- Rotor model built to evaluate cause of vibration
- Model predicts 2nd critical much higher than vibration in field.
- Rotor model assumes rigid supports



Tune Rotor Models

- Turbine outboard bearing has a "wobble foot" design to allow for thermal expansion

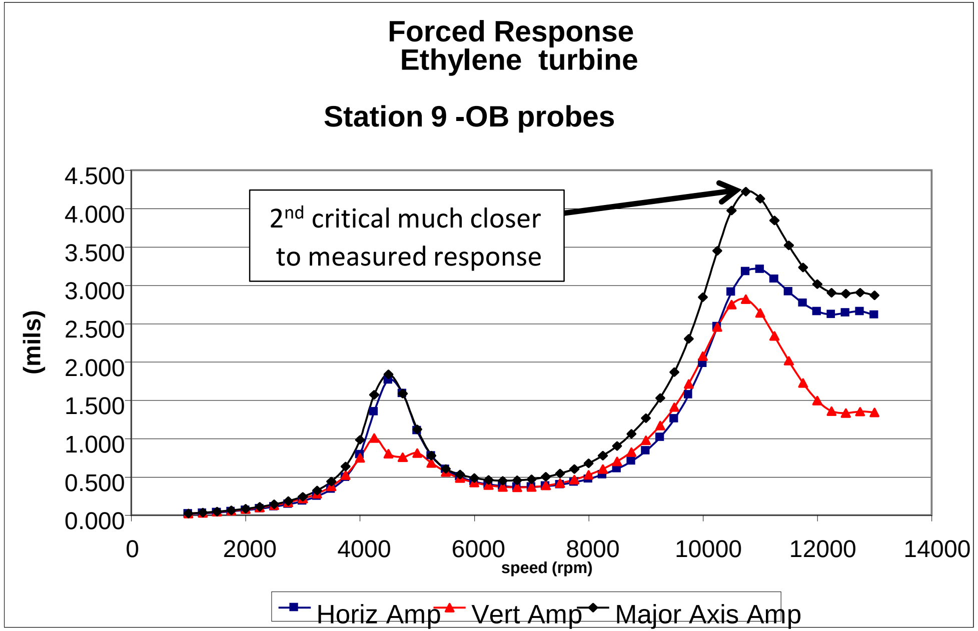
- Bearing housing support stiffness added to rotor model



Tune Rotor Models

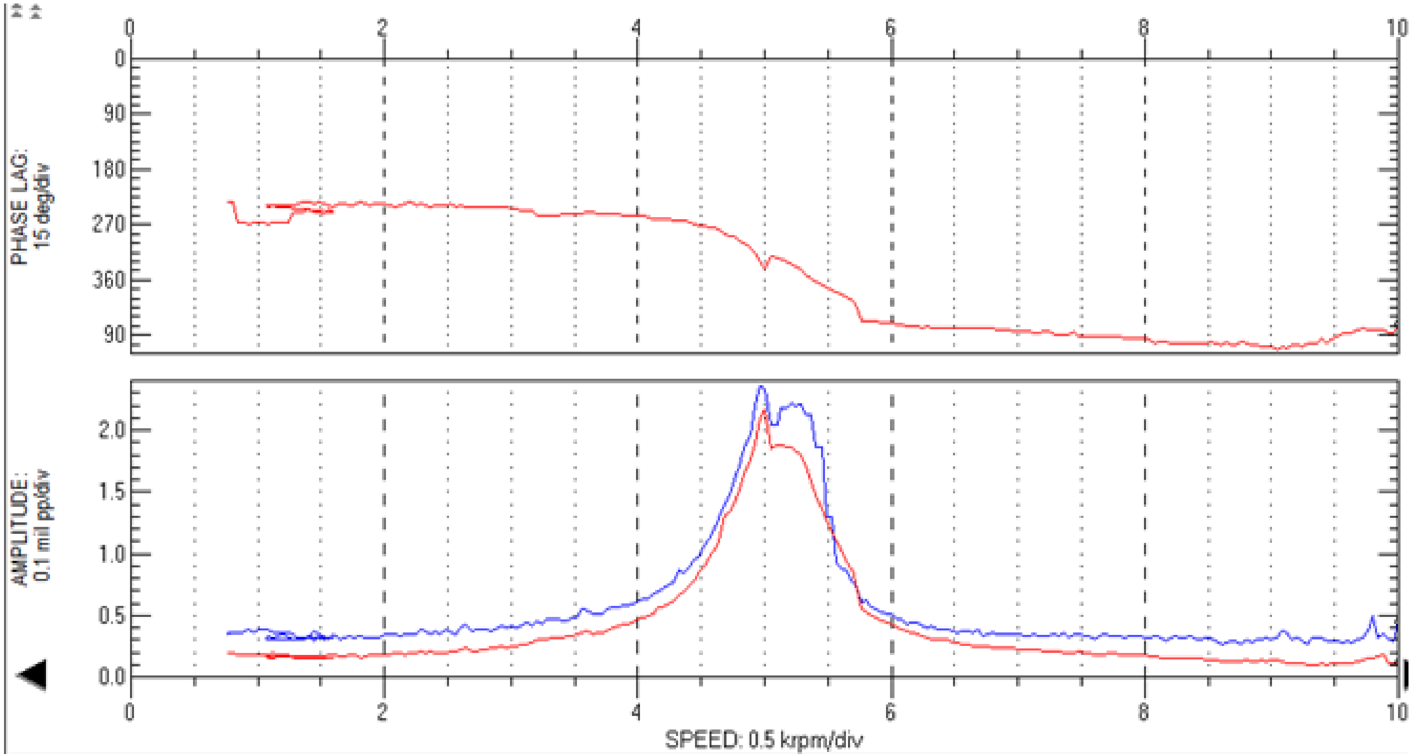
- Adding bearing support stiffness to model lowers predicted 2nd critical

- Turbine OB bearing clearance was shimmed to 0.0005 in below minimum design



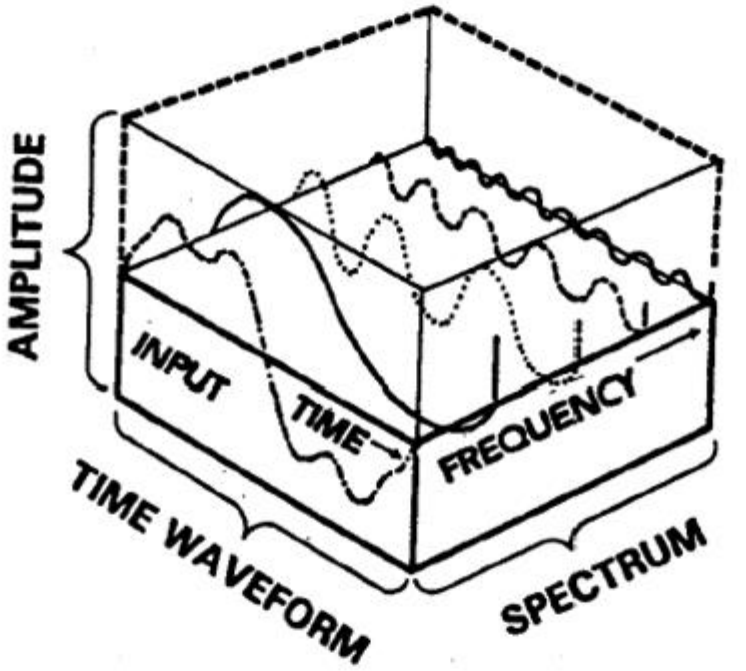
High Speed Steam Turbine

After bearing clearance change



Spectrum Plot

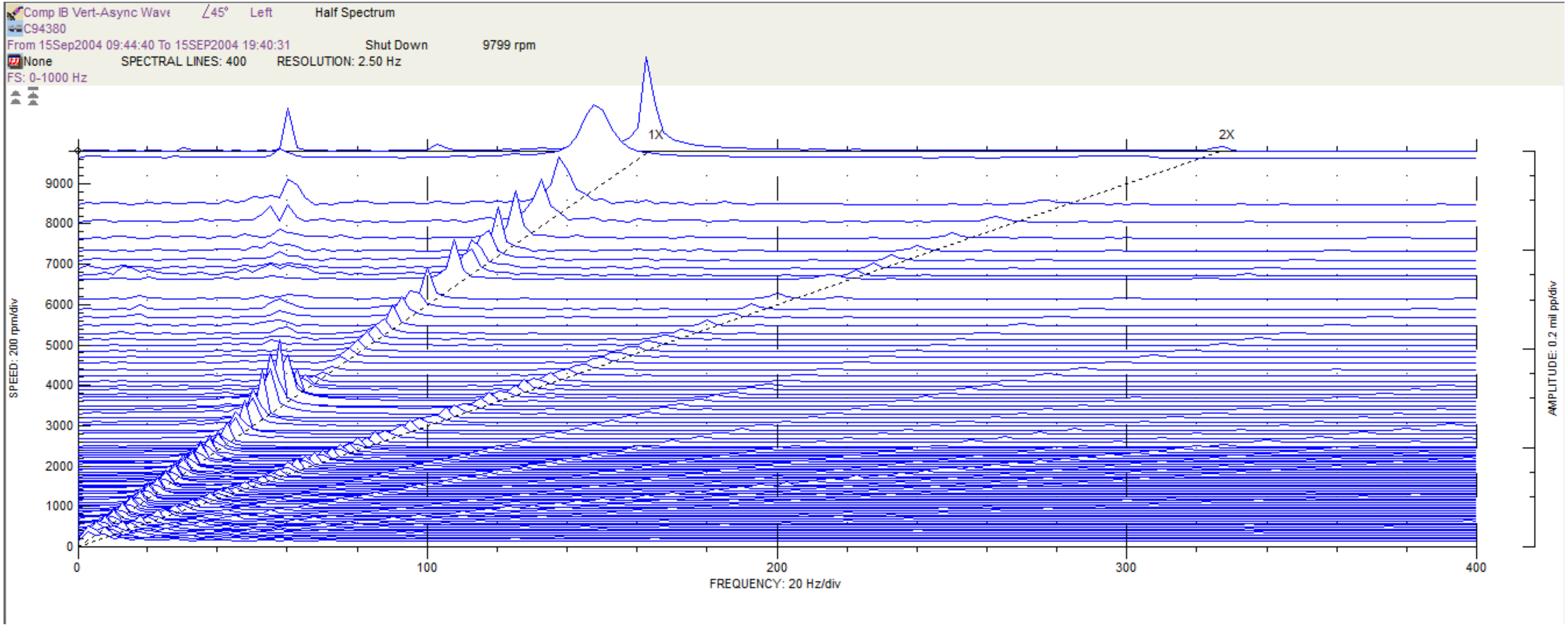
- FFT (Fast Fourier Transform) used to convert from time domain to frequency domain
- Spectrum excellent tool for determining frequency of different components
- Data acquisition time may limit use during transient events



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

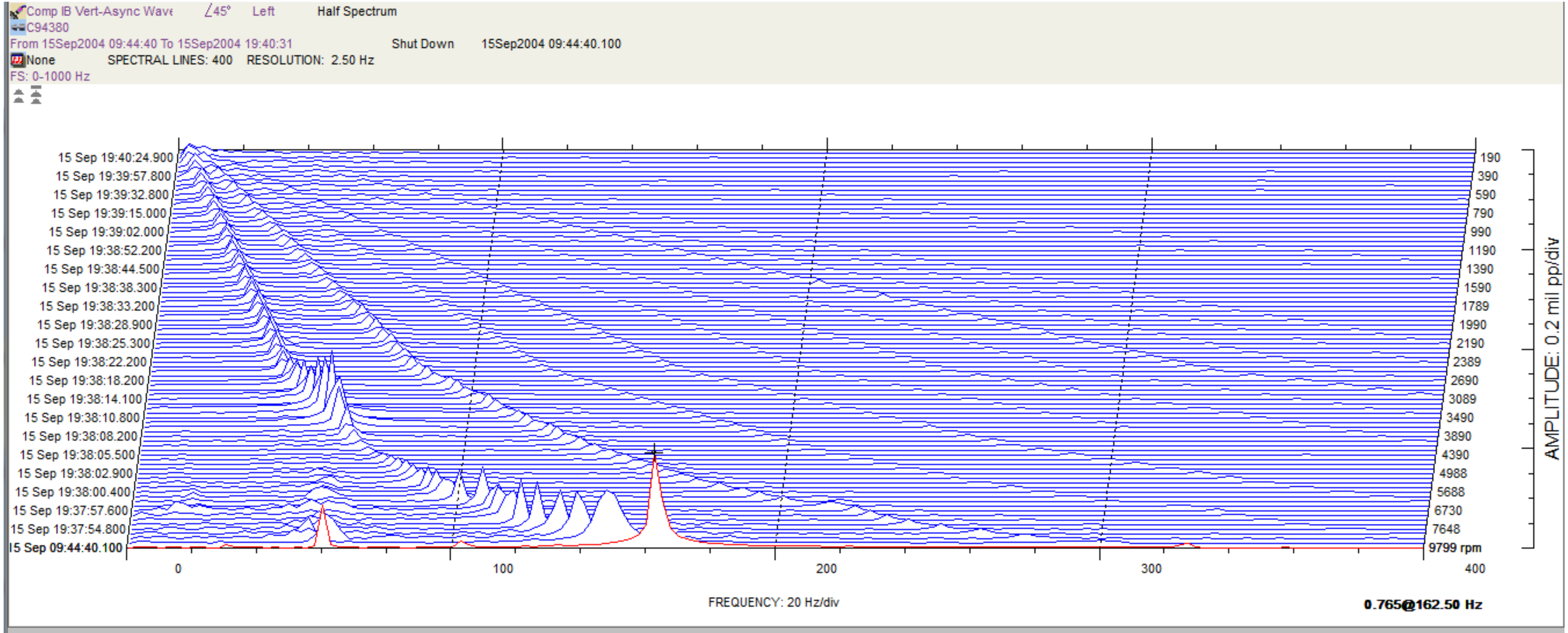
Cascade

- Multiple spectrums plotted versus speed



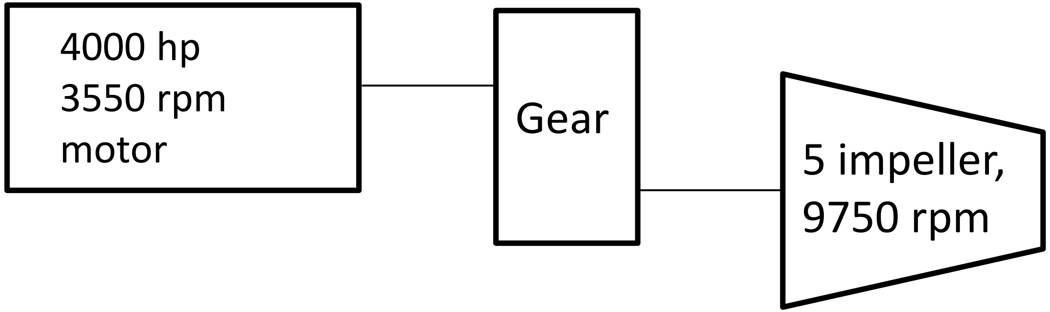
Waterfall Plot

- Multiple spectrums plotted versus time



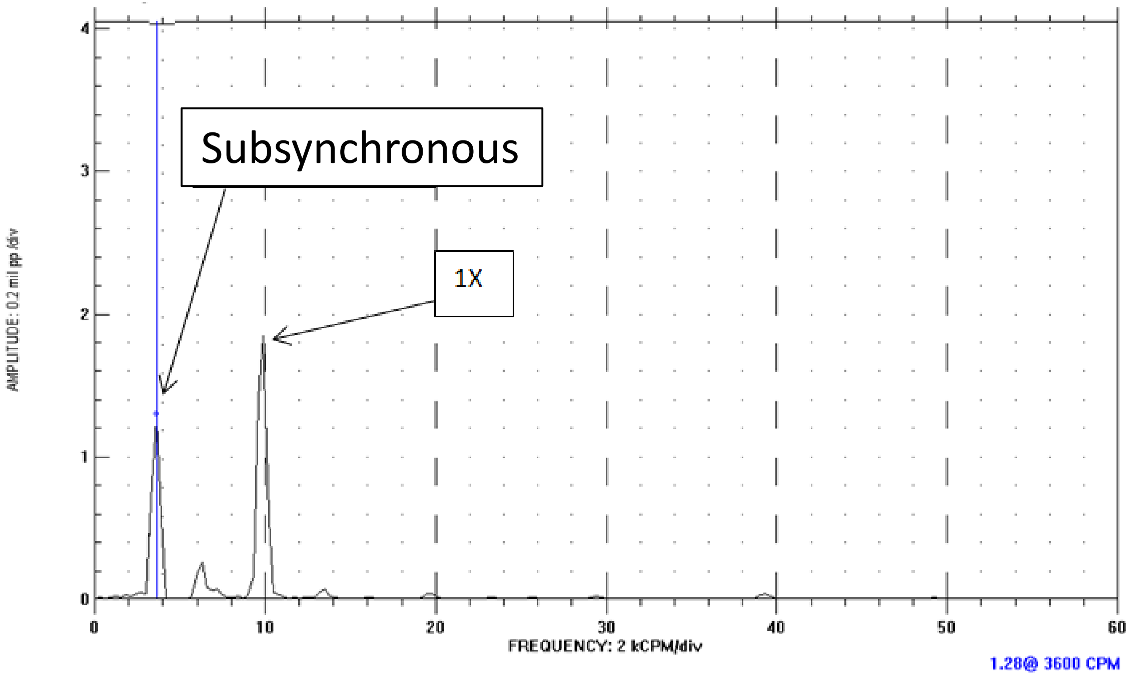
Centrifugal Compressor
w/ sub-synchronous instability

- 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

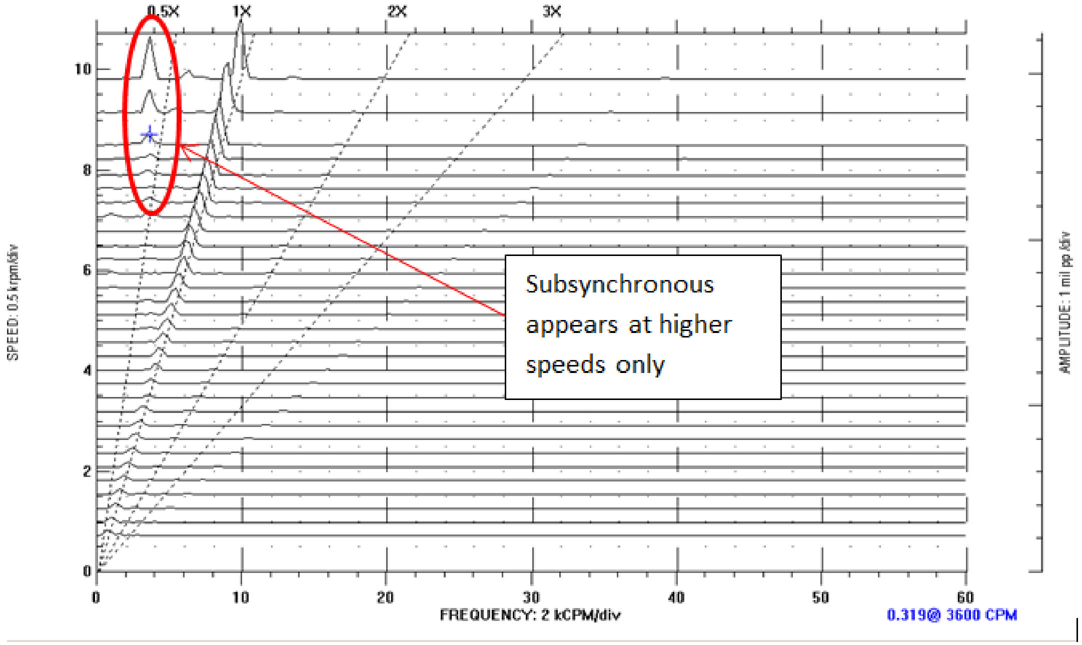


Centrifugal Compressor Instability

Sub-synchronous vibration

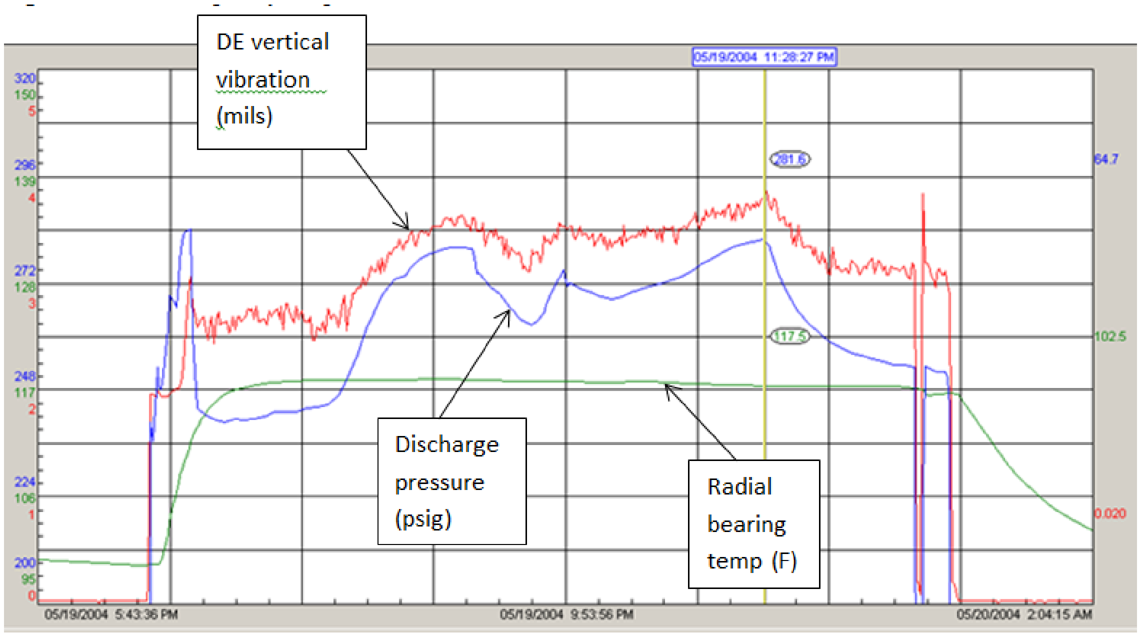
- Cascade plot shows subsynchronous peak appears only at higher speed

- Indicates that sub-synchronous could be exciting 1st critical



Centrifugal Compressor Instability
Sub-synchronous vibration

- Vibration tracked closely with discharge pressure



Centrifugal Compressor Instability

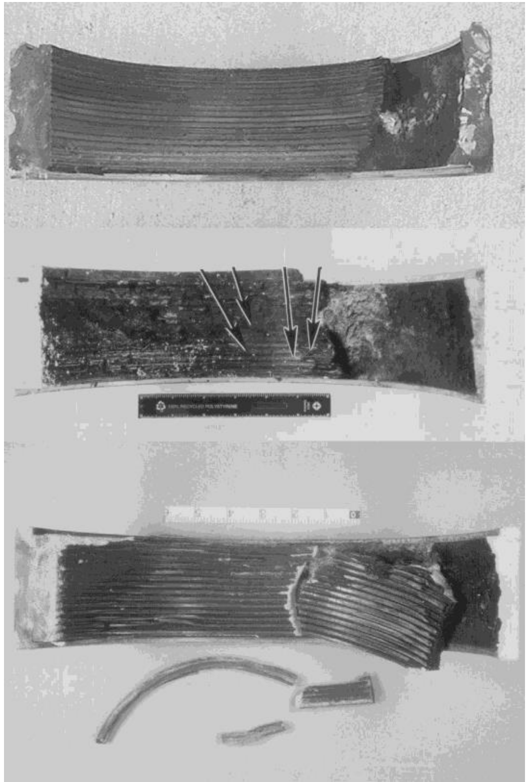
Balance Drum Seal History

- 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

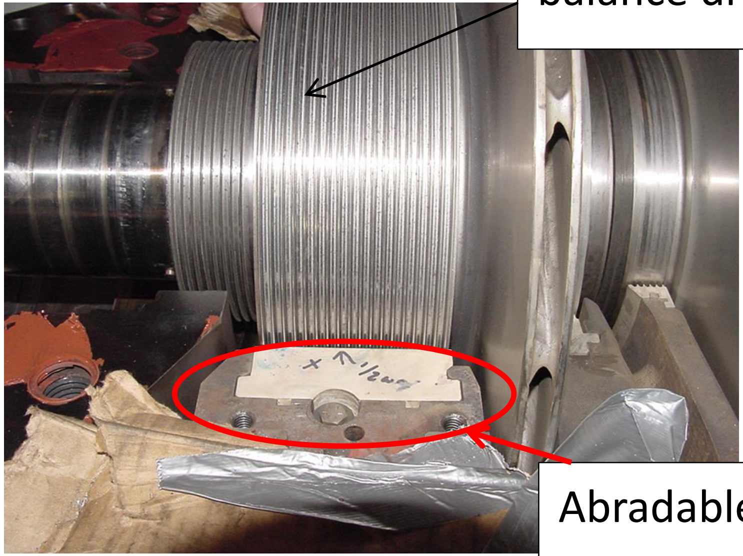


Centrifugal Compressor Instability

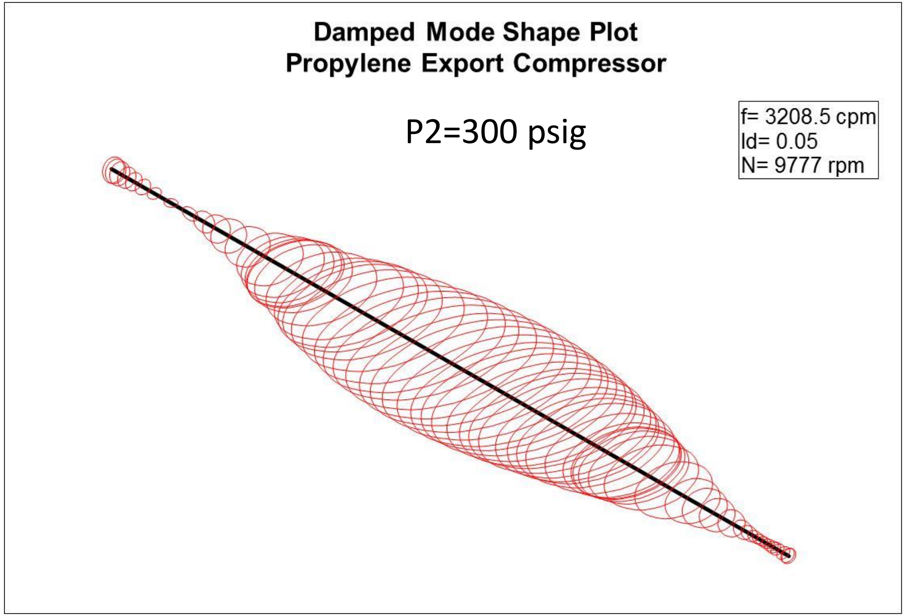
Balance Drum Seal

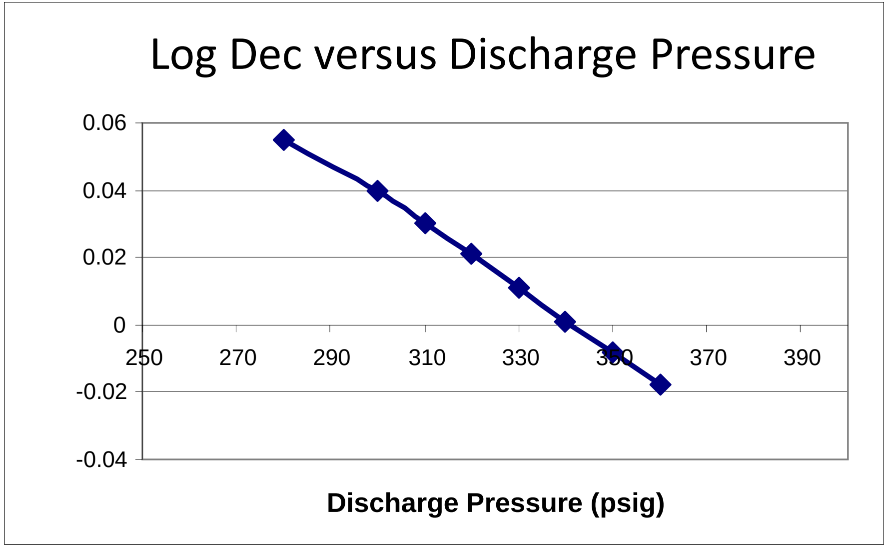
- End gap clearance increased further during latest overhaul (2004) due to multiple failures in past

- Rotor model built to evaluate problem



Stability Analysis

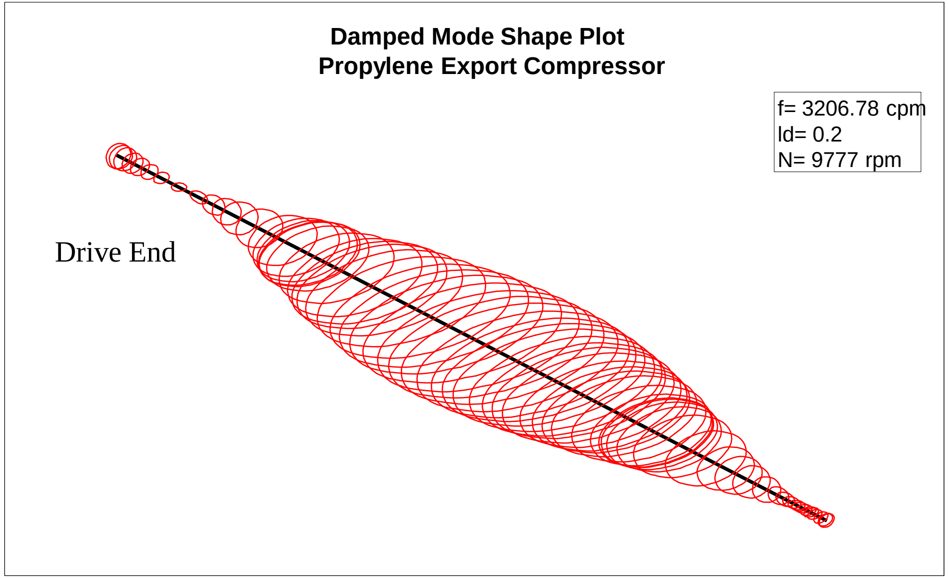




- Compressor has low margin of stability

Centrifugal Compressor Instability

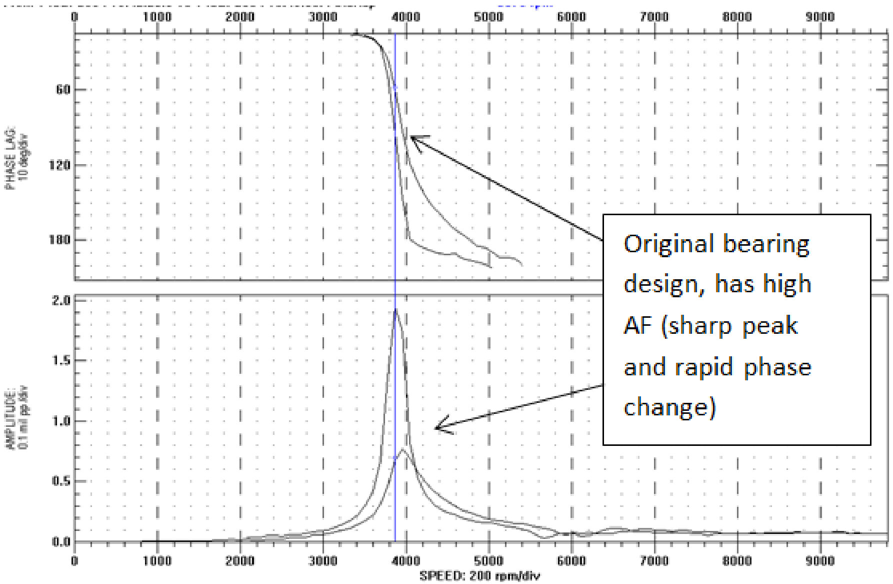
- Balance drum seal was root cause but difficult to replace
- Bearing optimized by increasing L/D, changed configuration to LOP, lowered preload
- Calculated 1st mode log dec increased from 0.05 to 0.2



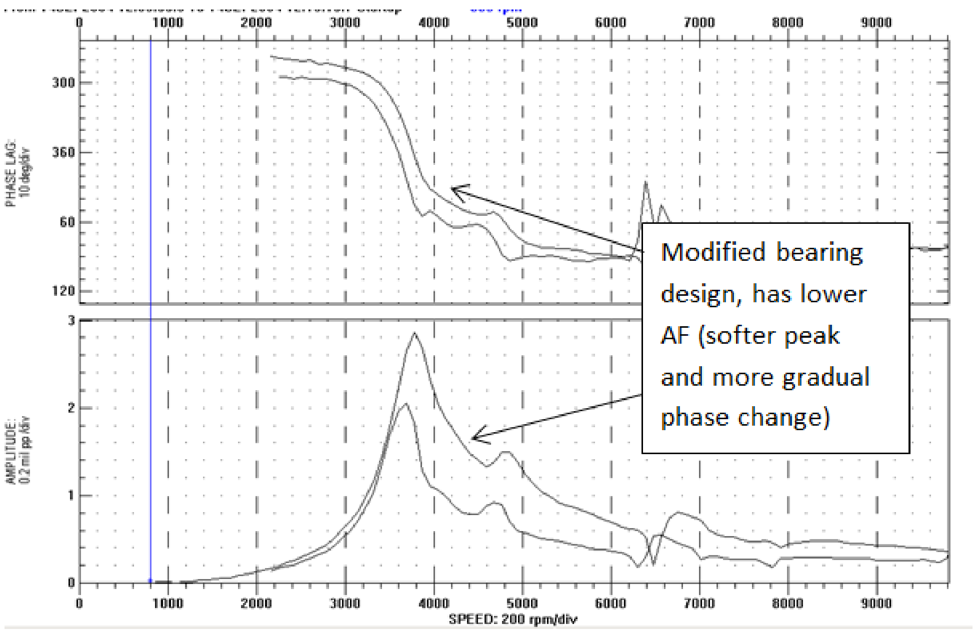
Centrifugal Compressor Instability

At-speed testing

Original LBP bearing

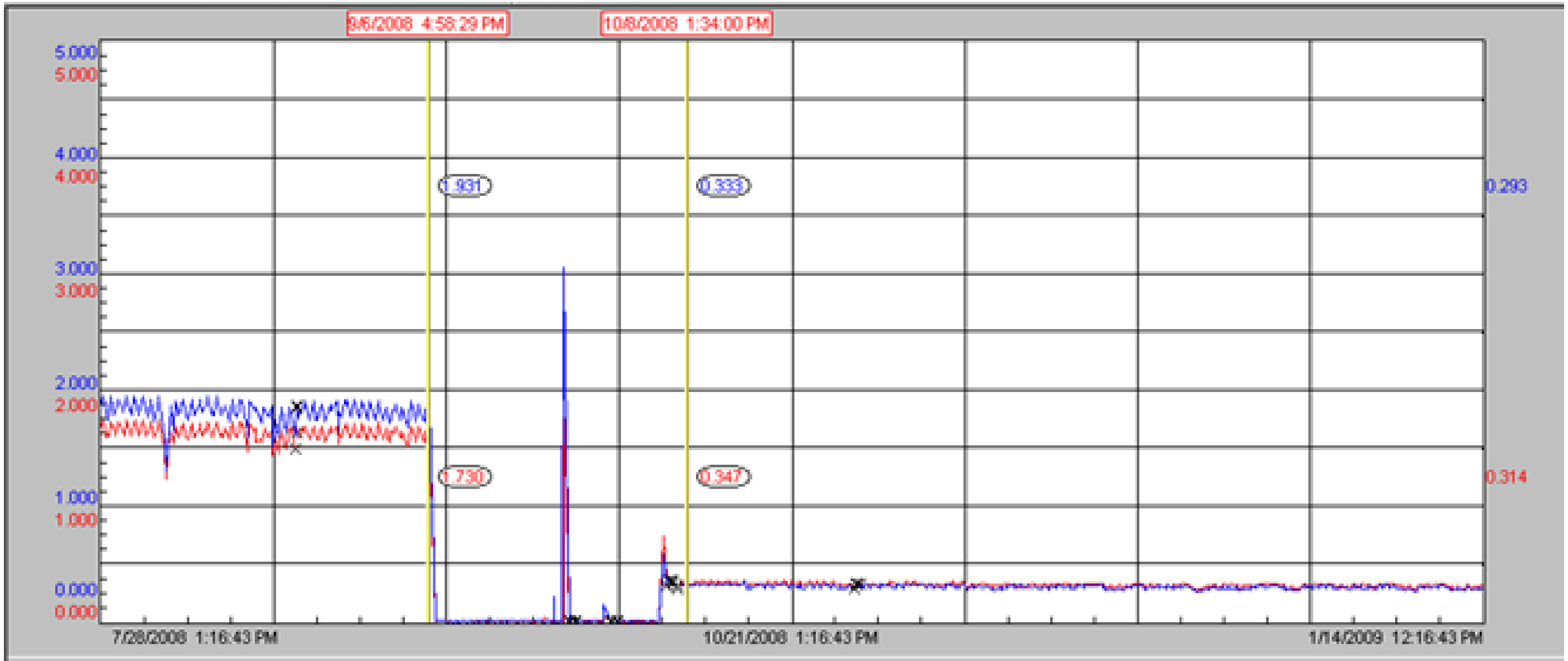


Optimized LOP bearing



Centrifugal Compressor Instability

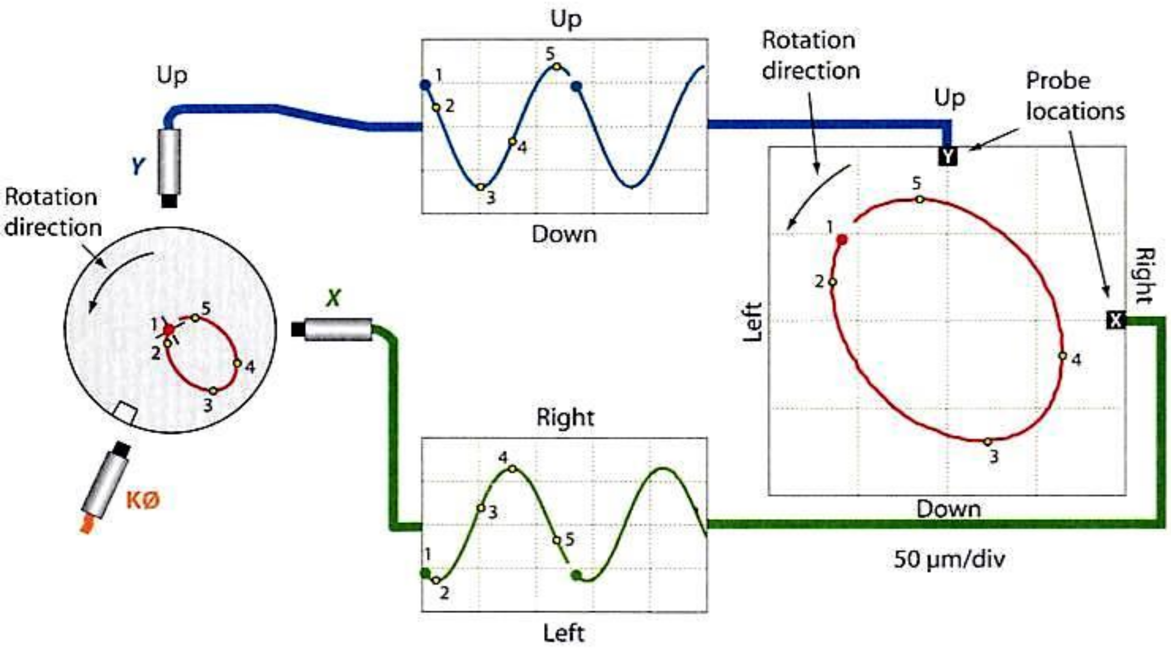
Bearing Change Results



	Name	Data Source	Map	Description	Value	Level	Status	Au	Plot Min	Plot Max	Units	Shift	TZ	Type	Period	Method	Sta	Ext
	CXP2XYT03065_M	CXPosmII	IPX_PV_TRI	C380 VIBRATION	0.293	Good	Good		0.000	5.000	MILS	0 0:00:(		Best Fit	1 Hour			
	CXP2XXT03064_M	CXPosmII	IPX_PV_TRI	C380 VIBRATION	0.314	Good	Good		0.000	5.000	MILS	0 0:00:(		Best Fit	1 Hour			

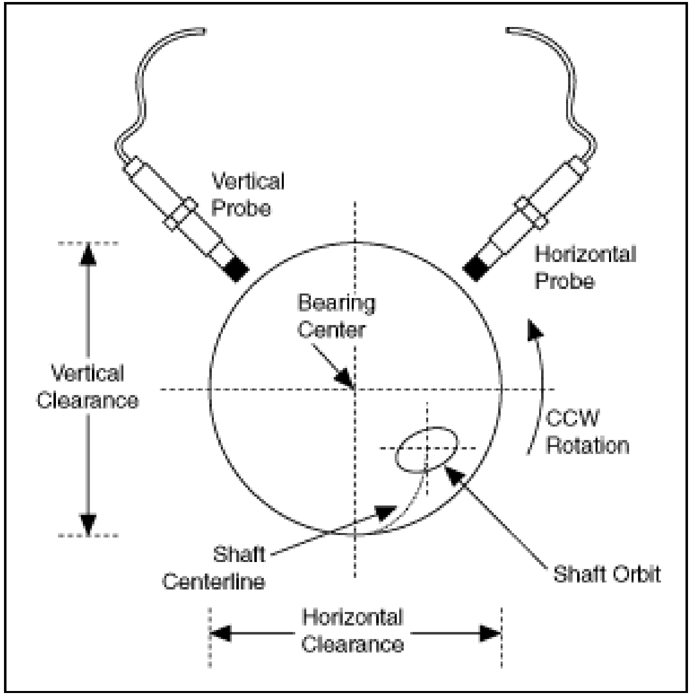
Shaft Orbit

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



Shaft Centerline

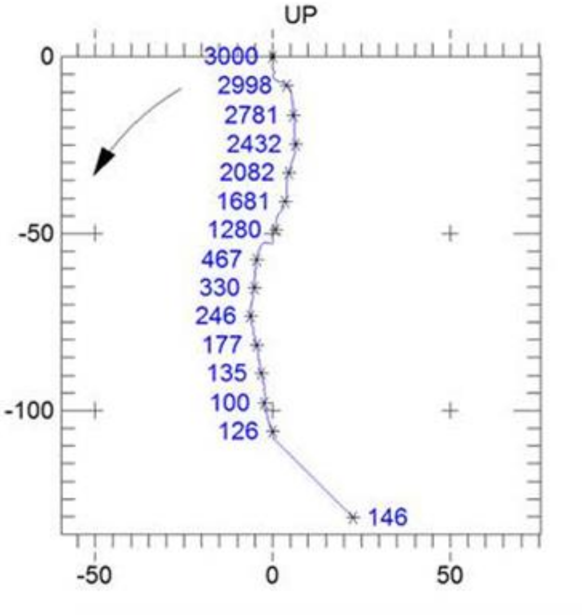
- Similar to orbit, but DC portion used to show center of shaft orbit
- Excellent for showing rubs, excessive clearance, mis-alignment
- Must be careful with transient thermal effects



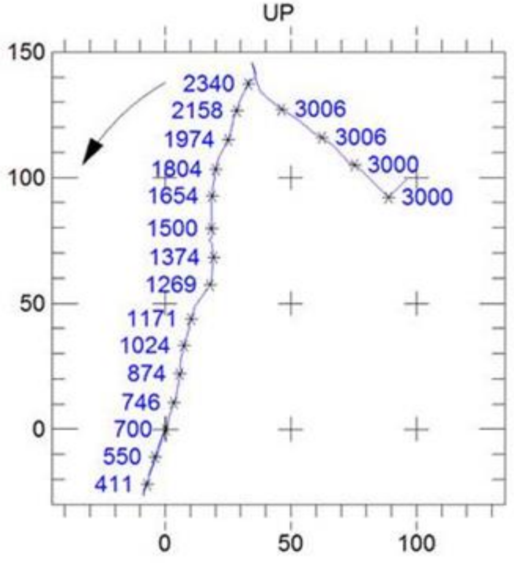
Industrial Gas Turbine

Internal rub

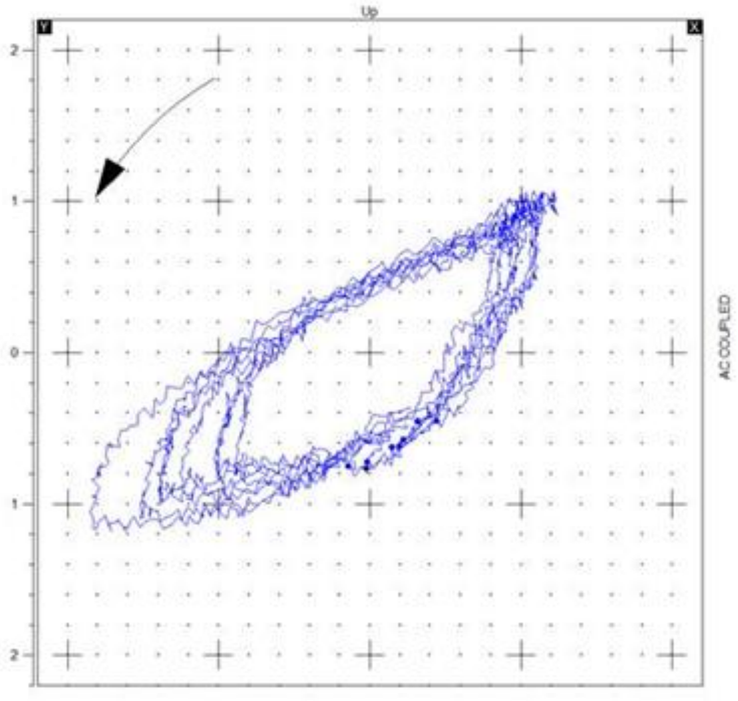
Before

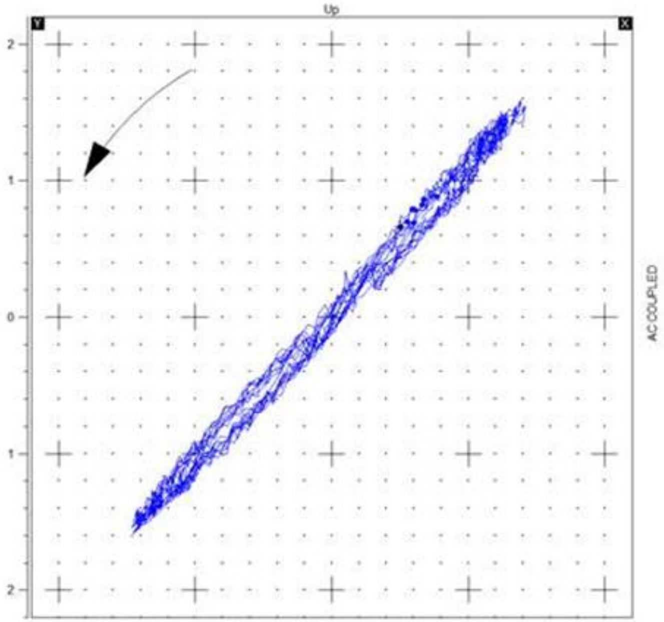


After



Industrial Gas Turbine
Internal rub – progression





Industrial Gas Turbine Internal rub

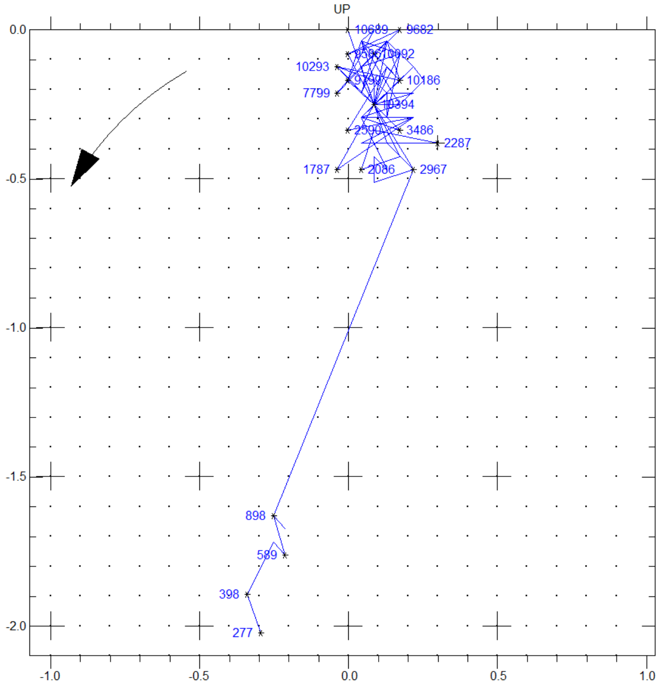
- Inspection revealed damaged #3 bearing

- Alignment change made and bearing replaced

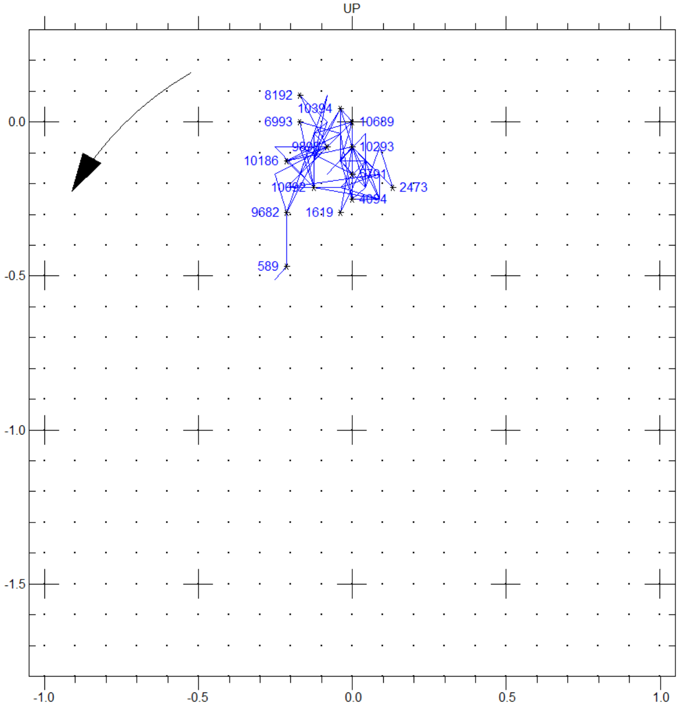
Shaft Centerline

Insufficient bearing clearance

Compressor OB



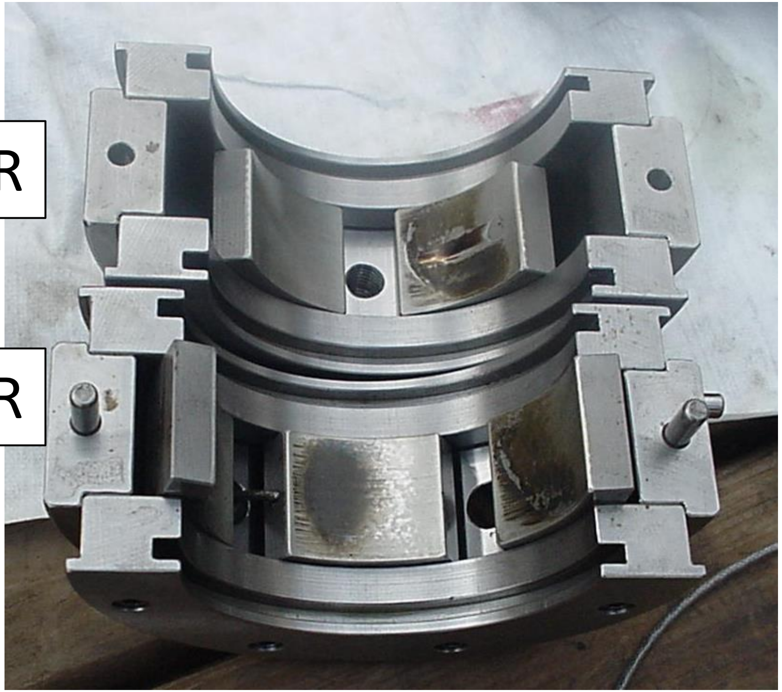
Compressor IB



Shaft Centerline Plot

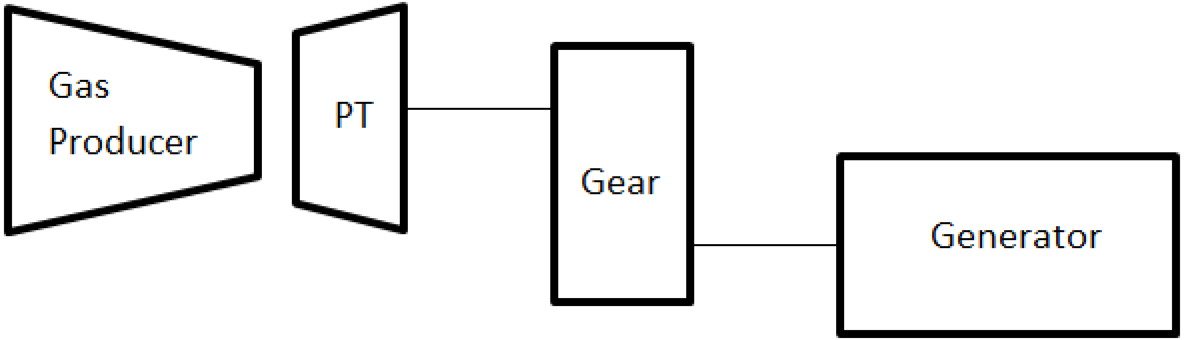
- IB Bearing clearance was too low

- Pads should never be scored in top half of bearing on a beam style compressor



Gas Turbine Generator
High Gearbox Vibration

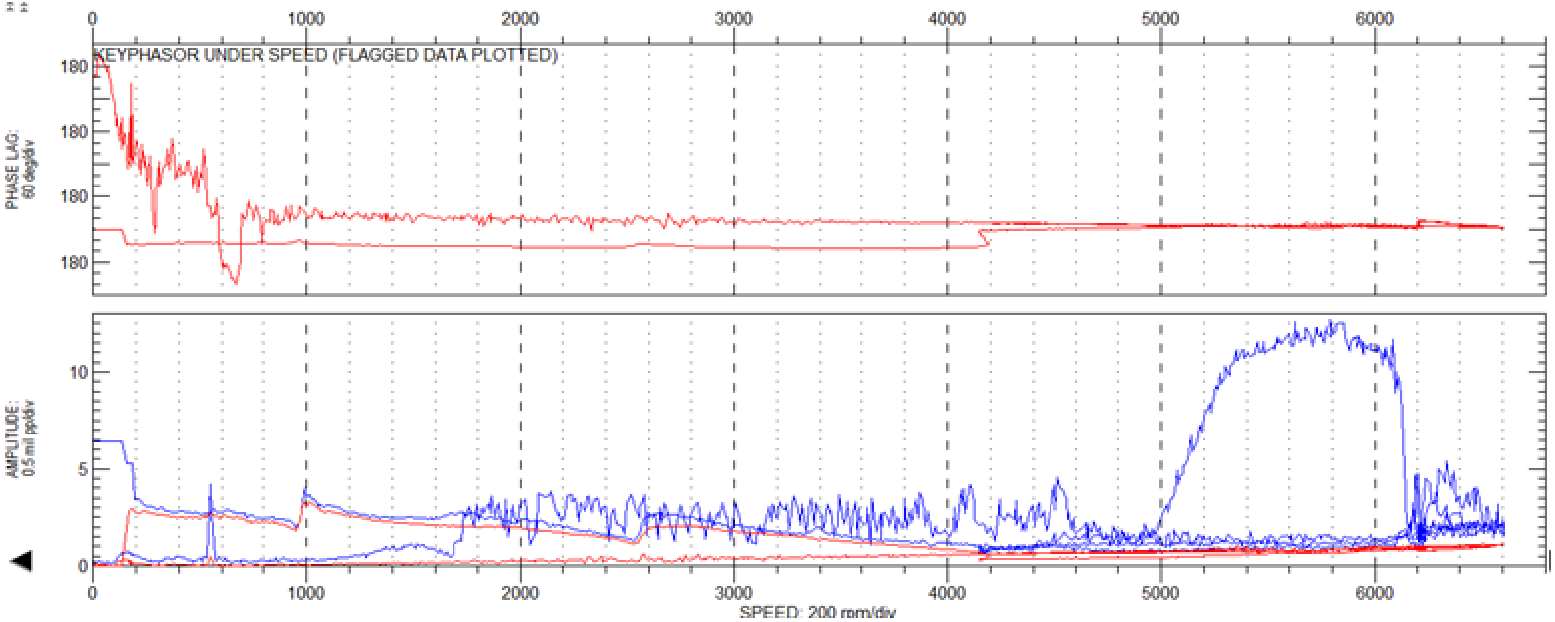
- 30 MW, 2-shaft aero-derivative turbine generator
- PT-6200 rpm, generator-1800 rpm, double helical parallel reduction gear set



Gas Turbine Generator

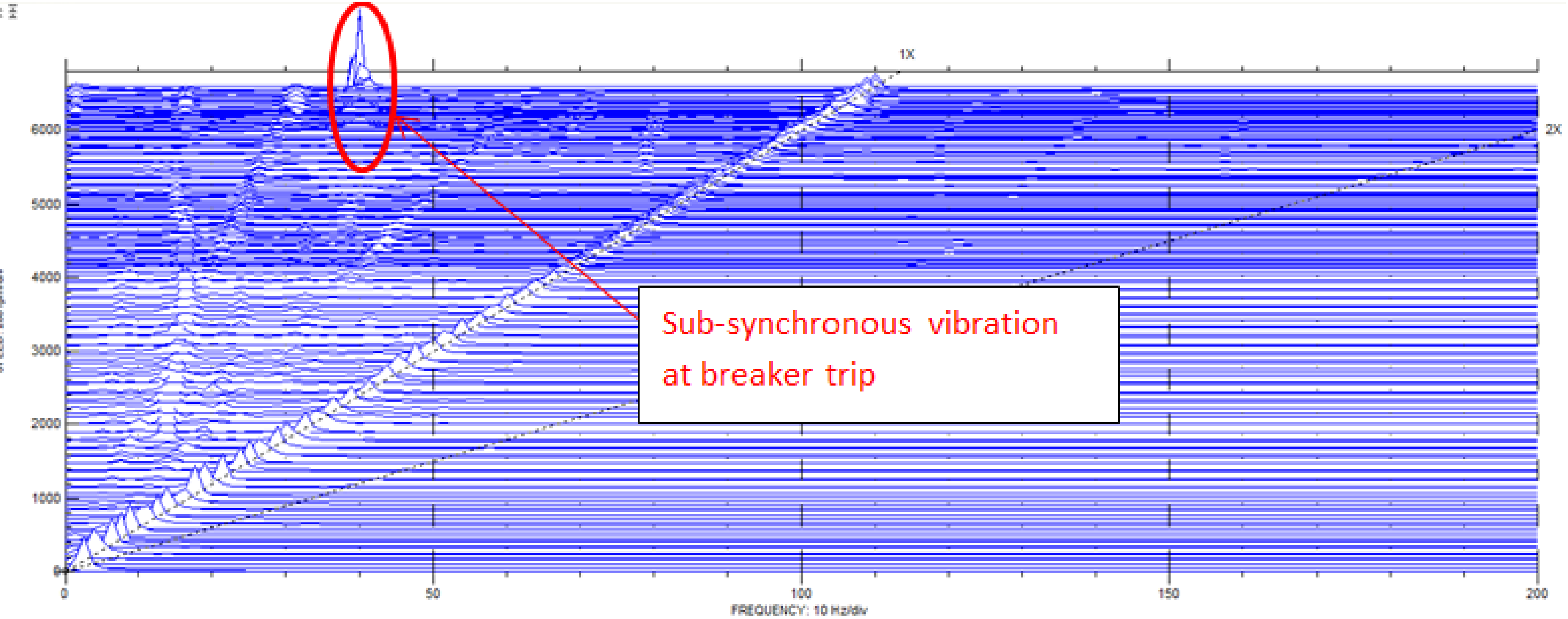
High Gearbox Vibration

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



Gas Turbine Generator

High Gearbox Vibration



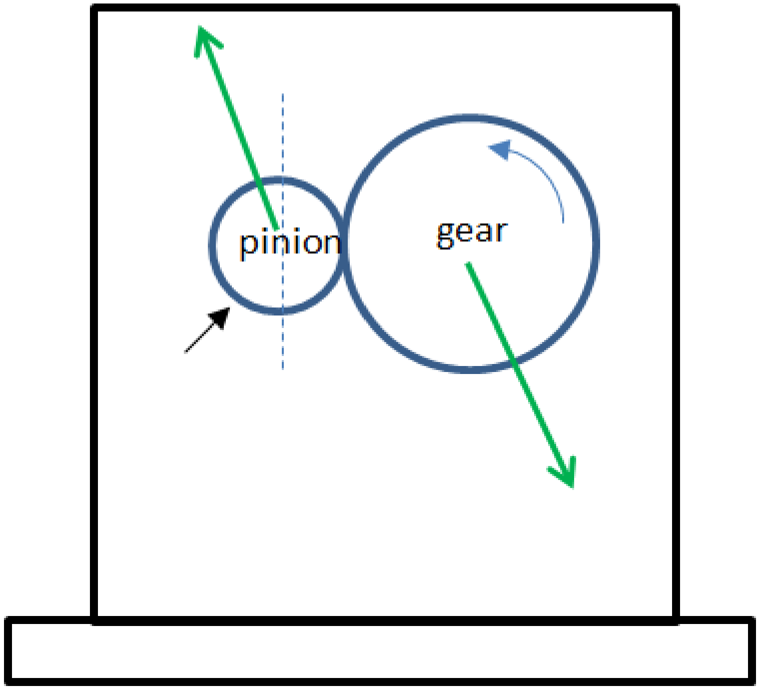
Gas Turbine Generator
High Gearbox Vibration

- 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



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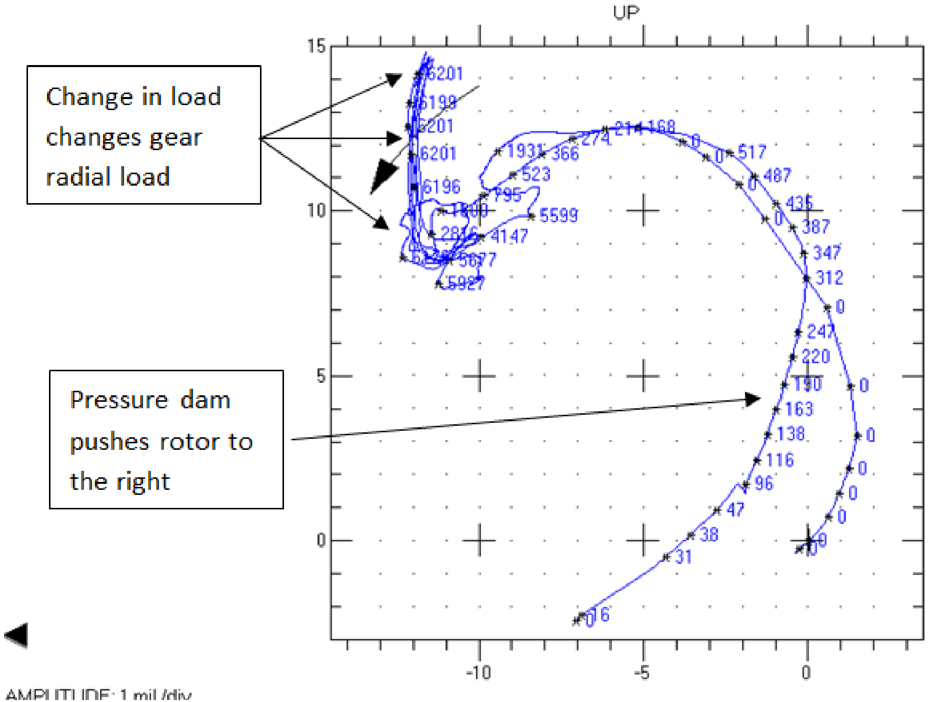
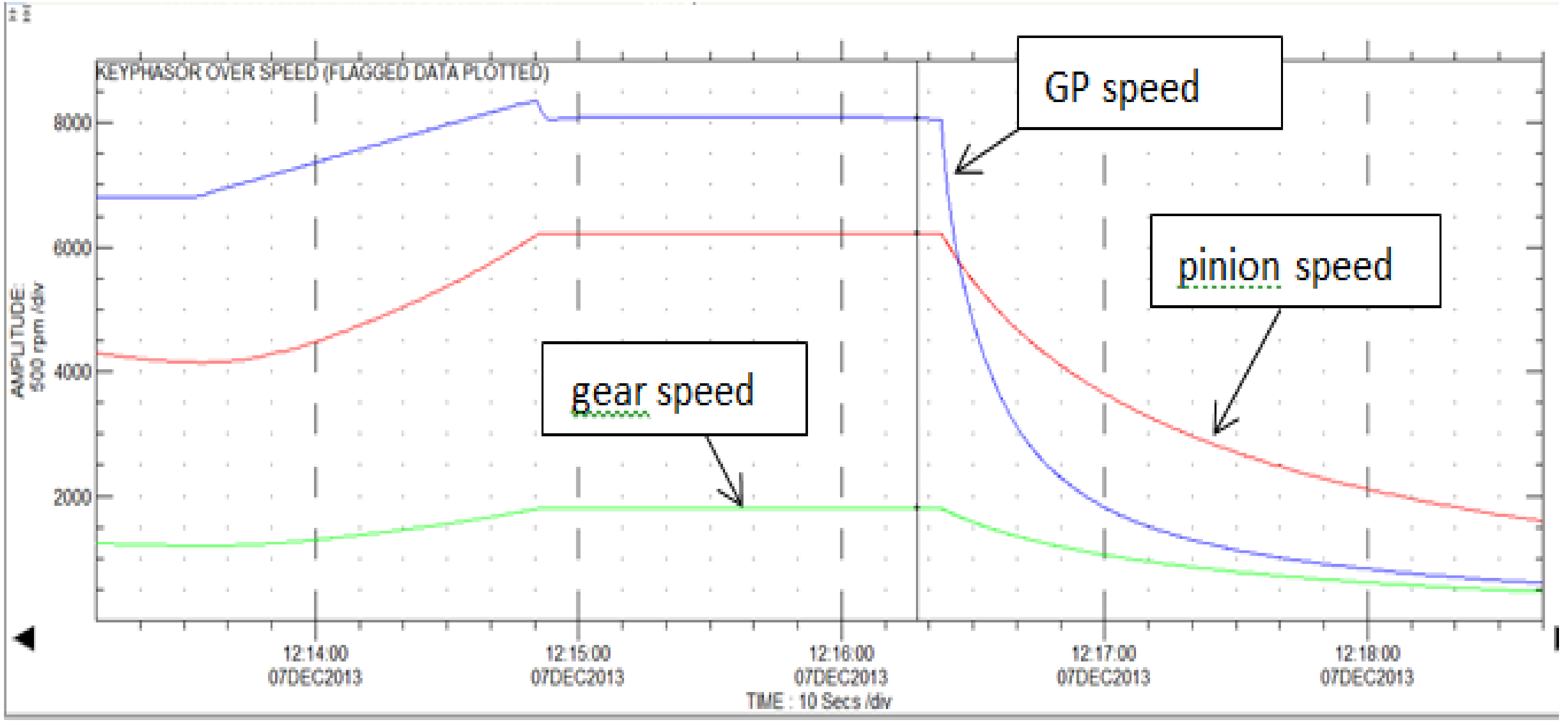
High Gearbox Vibration

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



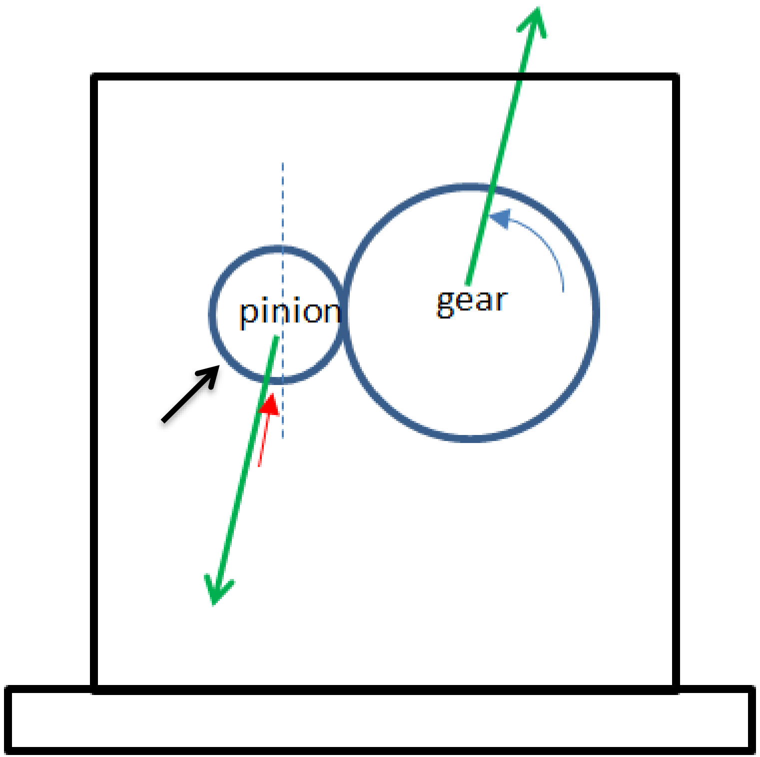
High Gearbox Vibration

- 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



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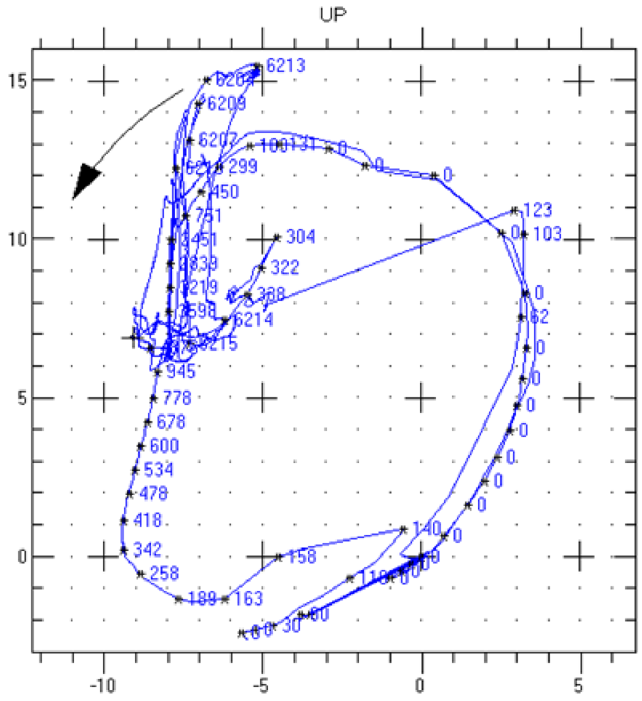
High Gearbox Vibration

- Note that pinion comes almost straight down after pressure dam re-located.

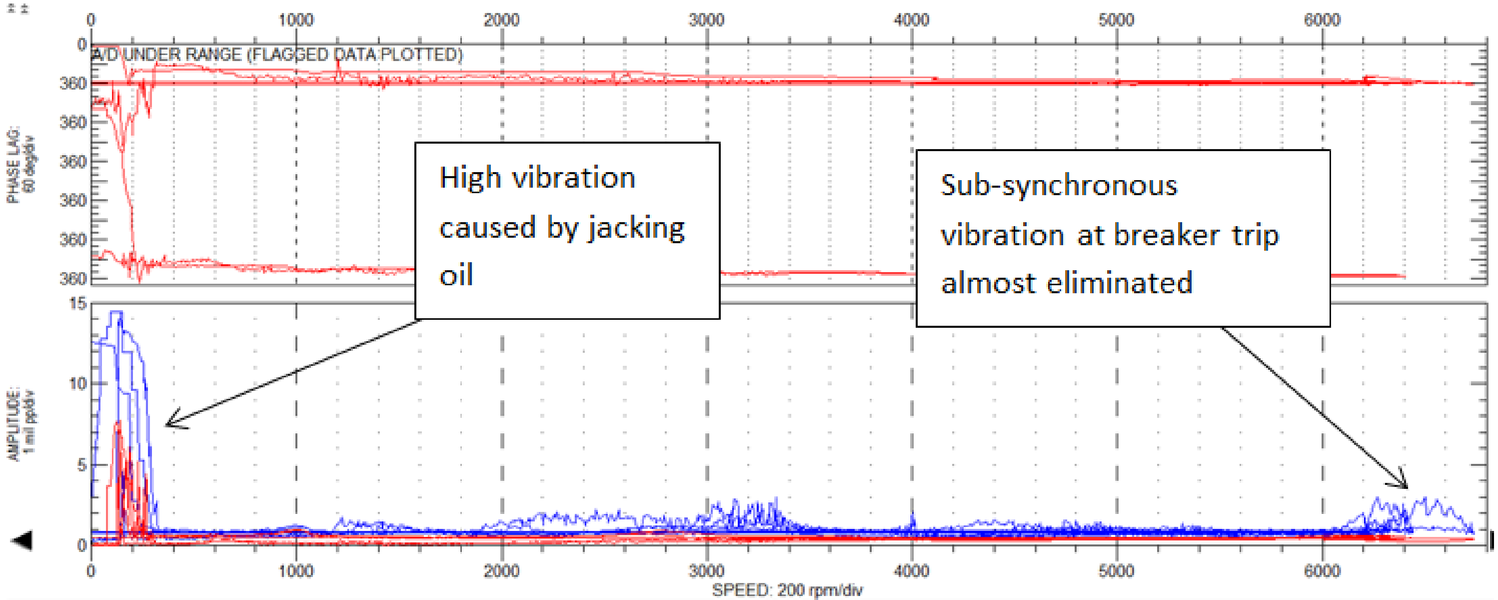
- Load changes still present in shaft centerline



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?