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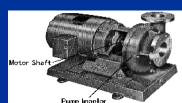
Transient Speed Vibration Analysis

Insights into Machinery Behavior

07-Dec-2007



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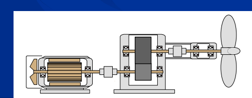

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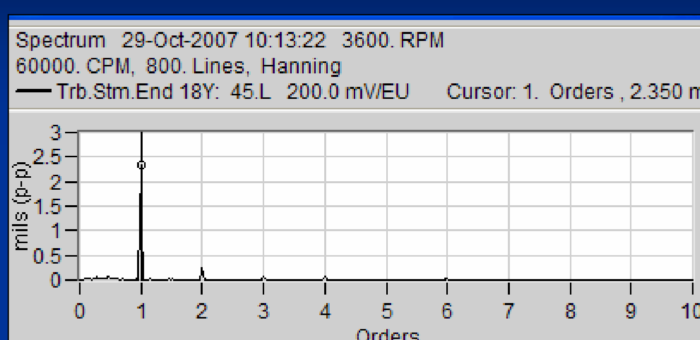
Ok, analysts….

- What is wrong with this turbine?



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2

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Possibilities

- What causes a predominant 1X Vibration?
  - Unbalance
  - Misalignment
  - Rotor Resonance
  - Structural Resonance
  - Rub
  - Coupling Lock Up
  - Oversize bearing
  - Bowed Shaft
  - High 1X Slow Roll / Runout
  - Cracked Shaft

Transient Vibration Analysis

- Transient <u>Speed</u> Vibration Analysis
- Acquisition & analysis of data taken during startup and shut down
- Provides significant insight into the rotor and structural dynamics that cannot be had with only steady state analysis
- This information includes:
  - Unbalance “Heavy Spot” Locations
  - Rotor Mode Shapes
  - Shaft Centerline Movement / Alignment
  - Bearing Wear
  - Shaft Runout
  - Critical Speeds / Resonances
  - Rotor Stability
  - Bearing Wear
  - Foundation Deterioration, and others

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Transient Data Sampling

- This is not 'PdM' data acquisition
- Multiple channels (8 – 30+)
- All channels sampled simultaneously & synchronously
- All data referenced to a once-per-revolution speed / tach signal

Instrumentation

- What are some instrumentation requirements for transient data?
- Here's a "short" list of desired abilities for transient data acquisition:
  - Minimum channel count of 8, with 16 or more channels preferred
  - Synchronous sampling of all channels
  - 2 or more tach channels
  - Accurately sample data at low rotor speeds (< 100 rpm)
  - Measures DC Gap Voltages up to -24 Vdc
  - Produce DC-coupled data plots (for shaft centerline & thrust data)
  - Provide IEPE / accelerometer power
  - Electronically remove low speed shaft runout from at-speed data
  - Display bearing clearances; plot shaft movement with available clearance
  - Specify RPM ranges for sampling, and RPM sampling interval
  - Produce bode, polar, shaft centerline, and cascade plots for data analysis
  - Tracking filter provides 1X and other programmable vector variables

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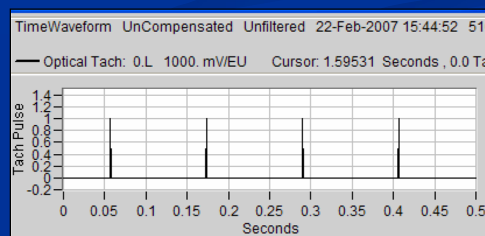
The Need for (Rotor) Speed

- A key (the key) component to successful transient analysis is a reliable once-per-revolution tachometer signal
- This signal provides a triggering pulse for the instrument tracking filter
- It lets us establish a rotor phase angle reference system
- For machines without a permanent vibration monitoring system, a portable laser tachometer can be used to provide a TTL pulse
  - We have had excellent results with Monarch Instrument's PLT-200
  - Observes optically reflective tape attached to the shaft
  - Senses optical tape 25' away feet at angles of ±70°!
  - Clean TTL pulse output
  - Very reliable trigger
  - Use with ZonicBook/618E dedicated Tach inputs



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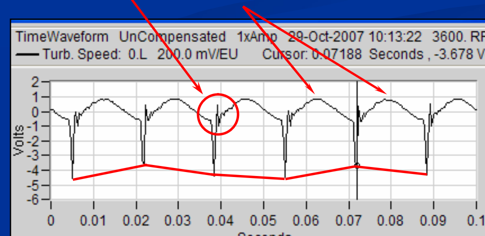
7

The Need for (Rotor) Speed

- Machines with permanent vibration monitoring systems often use a proximity probe to observe a notch or keyway in the shaft
  - This provides a DC voltage pulse output
  - Signal must used as an analog tach input on the ZonicBook/618
- Some signals create triggering problems due to signal quality:
  - Overshoot / ripple – causes multiple triggers per revolution
  - Overall signal contains an AC vibration signal – causes multiple triggers
  - The bottom of each pulse is not at the same voltage level – causes misses samples



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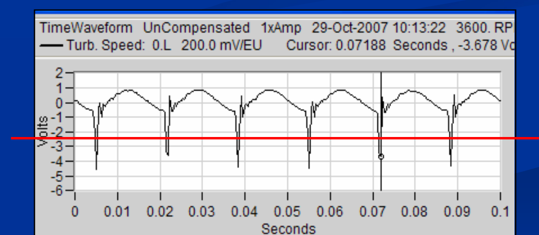
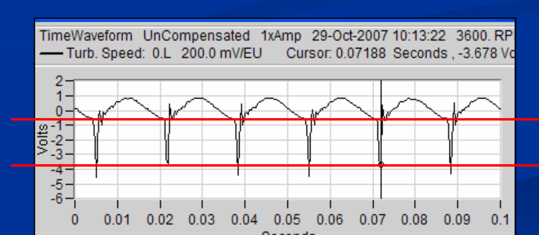
The Need for (Rotor) Speed

- If your instrumentation does not properly trigger using Auto tach:
  - Try manually adjusting the trigger voltage level to a voltage that only allows the instrument to that voltage level and corresponding slope (+ or -) once per revolution
  - A trigger setpoint of -2.0 to -3.0 Vdc would work nicely here



The Need for (Rotor) Speed

- If reliable triggering cannot be established, a signal conditioner such as Bently Nevada's TK-15 Keyphasor Conditioner can be used to modify the signal
- It can simultaneously 'clip' the top and bottom portions by applying bias voltages, thus removing any ripple / overshoot from the pulse, and producing a more TTL-like pulse
- The signal can also be amplified



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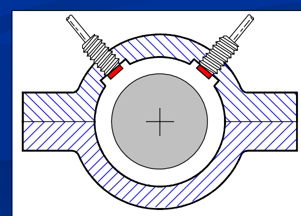
Transducer Selection vs. Machine Design

- Journal & Tilt Pad Bearings:
  - Machines with heavy /rigid casings, and 'light' rotors
    - Most steam turbines, barrel compressors, gearboxes, large pumps, etc.
    - Proximity Probes in X-Y Configuration at each bearing
    - Provide direct measurement of shaft-relative vibration
    - Seismic probes (accels) yield attenuated signals & phase lag
  - Machines with lower case-to-rotor mass ratios or flexible supports
    - Gas turbines, LP turbine pedestals, air machines, fans, pumps, motors
    - Use Proximity & Seismic when possible
    - Flexible supports provide comparable shaft & casing vibration – both are important
- Rolling-Element Bearings:
  - Accelerometers
    - True Vertical & Horizontal Planes
    - Aligns probes close to major & minor stiffness axes



Configuration & Sampling Guidelines

- ΔRPM & ΔTime Sampling Intervals
  - Generally sample at ΔRPM of 5 to 10 rpm for most machinery
    - Produces high quality data plots
    - Keeps database sizes reasonable
  - Need to consider the total speed range over which data must be sampled
  - Will speed will oscillate during the startup?
    - Turbine startups; VFD drives
  - ΔTime sampling during startup provides data during heat soak / idle periods
    - 20 to 30 seconds between samples, unless process conditions are changing rapidly
- Try to estimate total database size required and ensure system will not truncate database during sampling

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Configuration & Sampling Guidelines

- Fast Ramp Rates – AC Motors
  - Induction motor startup will be very fast, accelerating quickly and smoothly from zero to full speed
  - Startup lasts only 10 – 40 seconds after the breaker is closed
  - Can data acquisition keep pace with the ramp rate?
- 3,600 rpm motor; 40 second startup time
  - 3600 / 40 = 90 rpm per second
  - At ΔRPM = 5, we would be trying to capture = 16 samples per second
- What can we expect from our data?
  - Examine data acquisition settings:
    - Fmax
    - Lines of resolution
  - ZonicBook/618E: 2,000 Hz Fmax; 1600 LOR
    - 1 sample = 0.8 seconds = 72 rpm change between start and end of sample
    - Data smearing
  - Fmax = 1,000 Hz; 200 LOR
    - 1 sample = 0.2 seconds = 18 rpm
    - Set ΔRPM at 20 - 30

Transient Data Plot Types

- Bode
- Polar
- Shaft Centerline
- Waterfall / Cascade

- Before discussing these data plots, we need to review the importance of slow roll compensating our 1X-filtered shaft vibration data to remove the effects of runout

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Slow Roll Compensation

- Slow-roll: mechanical and electrical shaft runout in the target area of a proximity probe
  - Defects that create a non-dynamic 'false' vibration signal
  - Adds vectorally to the true dynamic vibration at any speed
  - Prox probe cannot distinguish between runout and true vibration
- We need to electronically remove slow roll for accurate results
- For most turbo-machinery:
  - Sample vibration at low speeds, typically below 300 rpm
  - Reasonably sure there will be little dynamic shaft motion
  - The measured signal will contain the runout of the probe target area
- Most data acquisition systems allow runout signal to be store and then digitally subtract it from any at-speed vibration
- The differences can be dramatic….

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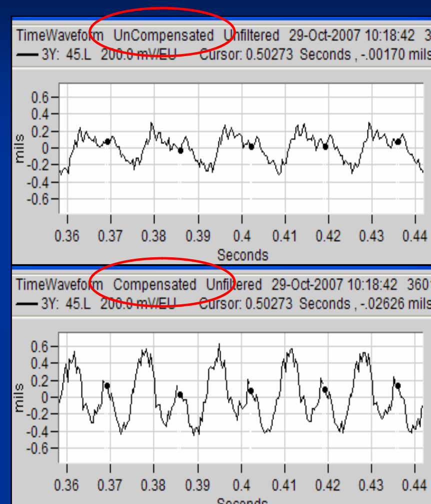
15

Slow Roll Compensation

- Time-waveform plot
  - Uncompensated
    - Dominant 1X signal, some 2X
  - Compensated
    - Significant 2X
- Different conclusions?



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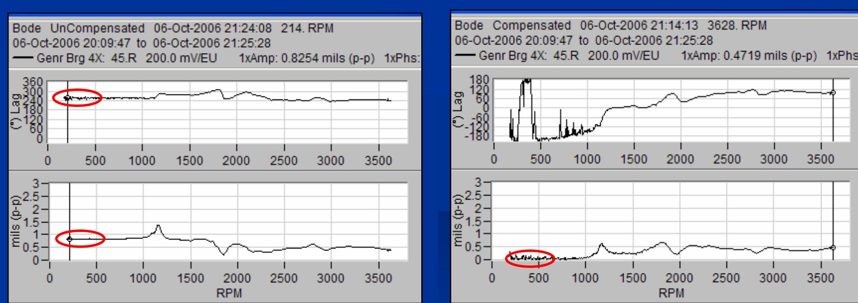
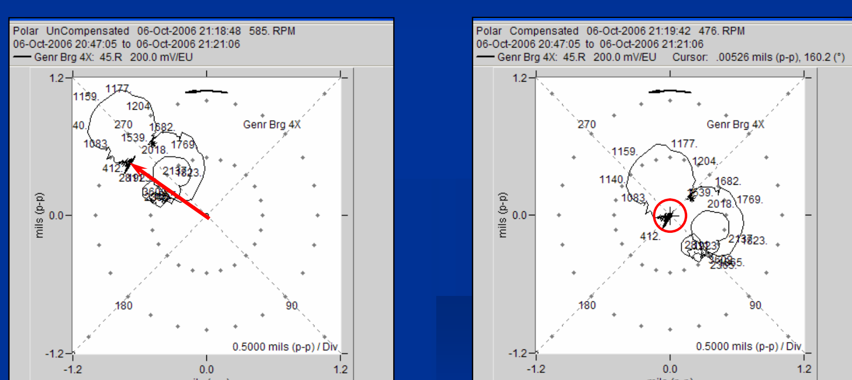
Slow Roll Compensation

- Selecting the Correct Slow Roll Speed Range on a Bode Plot
  - Look for low speed region with constant amplitude and phase
  - This ensures we are not including any 'true' dynamic activity
- Effect on Bode plot
  - Cannot predict how the bode plot will look like after compensation
  - Amplitude should be near zero at the speed the slow roll was sampled



Slow Roll Compensation

- Polar Plot Compensation
  - Slow roll vector merely moves the entire plot
  - "Shape" of the data (polar loops) not changed
  - Can be done visually
  - Properly compensated plot will begin at zero amplitude



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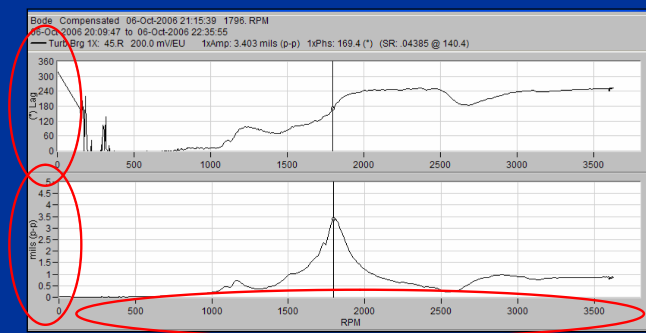
Transient Data – Bode Plots

- Bode plots typically show the 1X vector response in X-Y format
  - Filtered Amplitude
  - Phase Lag Angle
  - Rotor Speed



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19

Bode Plots

- They help provide the following information:
  - Slow roll speed range & slow roll vector values
  - The location of the “High Spot”, i.e., the rotor’s vibration response
  - The location of the “Heavy Spot”, i.e., the physical location of a residual unbalance on the rotor
  - Amplitude, phase & frequency of rotor and structural resonances
  - The presence of ‘split’ resonances
  - Amplification Factor, Damping Ratio & Separation Margin for a resonance

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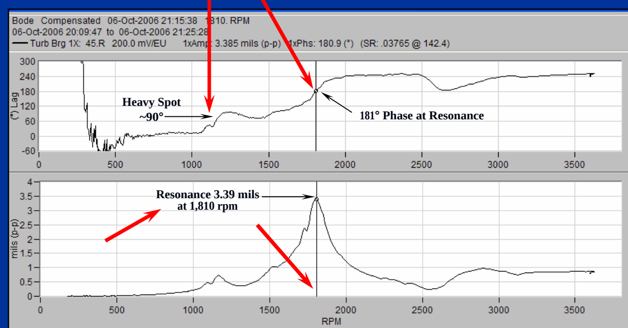
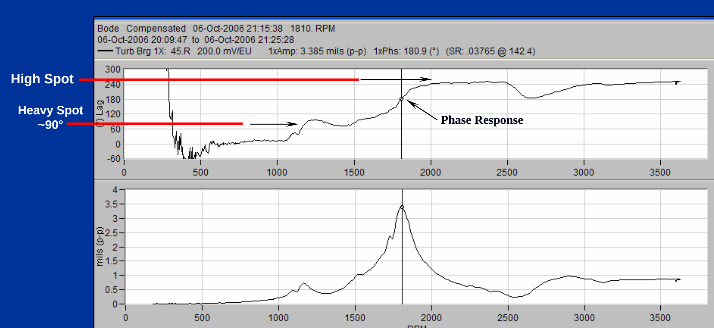
Bode Plots

- Resonance information
  - Amplitude – peaks at resonance
  - Phase – must show a 90 degree shift from low speed
    - Starting point can be difficult to determine if other modes are present
  - Frequency (rpm)



Bode Plots

- The "High Spot" = the measured 1X vibration response
  - Changes as a function of speed
- The "Heavy Spot" = physical location residual rotor unbalance
  - Hopefully doesn't change ☺
  - Relationship to High Spot depends on whether the rotor operates below, near, or above its resonance



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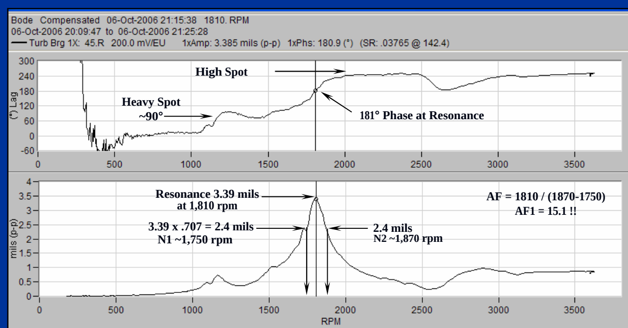
Bode Plots

- Amplification Factor (AF) a measure of damping for any mode
- API Half-Power Bandwidth Method:
  - Multiply (peak amplitude x 0.707)
  - Determine corresponding speeds N1 & N2 == half-power bandwidth
  - AF = resonance frequency divided by half-power bandwidth



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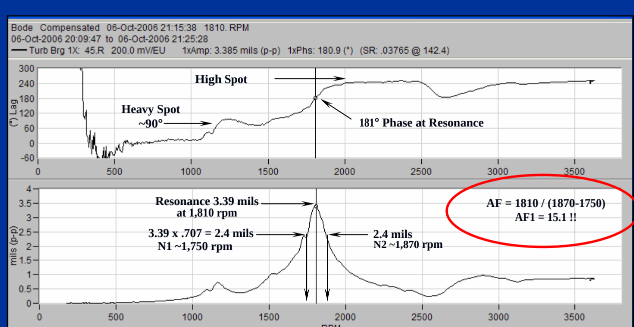
23

Bode Plots

- Amplification factor API guidelines:
  - < 2.5 = Critically damped; essentially no resonance seen
  - < 5.0 = very good
  - < 8.0 = still acceptable
  - 8 – 10 = marginal
  - > 10 = poorly damped



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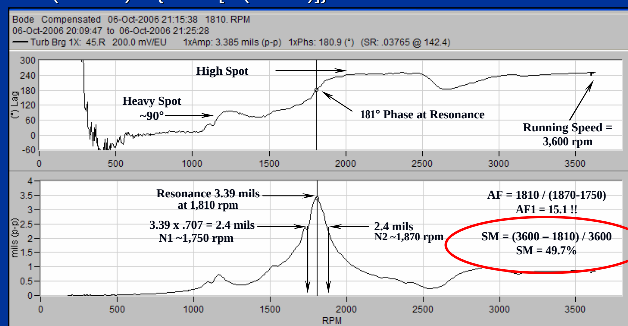
Bode Plots

- Separation Margin - API guidelines:
  - AF < 2.5: critically damped; no SM required.
  - AF = 2.5 - 3.55: SM of 15% above MCS & 5% below MOS
  - AF > 3.55 & resonance peak < MOS:
    - SM (%MOS) = 100 - {84 + [6/(AF-3)]}
  - AF > 3.55 & resonance peak > trip speed:
    - SM (%MCS) = {126 - [6/(AF - 3)]} - 100



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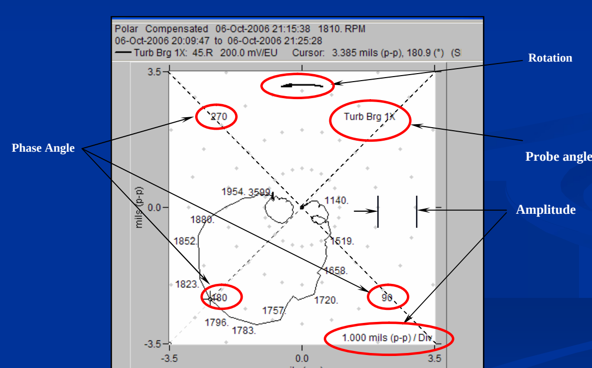
Polar Plots

- Polar plots (Nyquist diagrams) present the same information as bode plots, graphing amplitude, phase, and frequency
- Data is plotted in polar (circular) coordinates
- Direction of rotation



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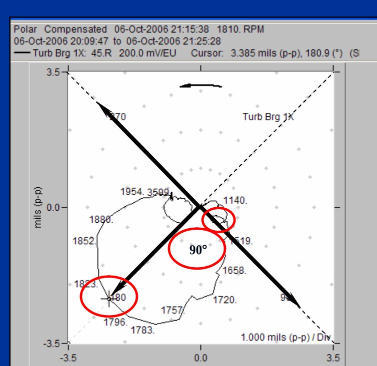
Polar Plots

- Key advantages over bode plots:
  - Easier data interpretation - resonances appears as loops
  - Plot is oriented to the vibration probe & referenced to machine casing
  - Slow roll compensation is easily performed, even visually
  - Incorrect compensation is easily identified
  - The High Spot and Heavy Spot have immediate physical meaning, being directly transferable from the plot to the machine.
  - We can easily identify the 1st and 2nd rotor modes and determine the ideal locations / planes for balance weights
  - Structural resonances are easy to identify
  - Speed normally increases opposite the direction of rotation, providing precessional information

Polar Plots

- Resonances (rotor & structural) appear as loops
  - Same amplitude peak and 90 degree phase change as bode plot
  - Loop is easier to identify than bode plot activity
  - Structural resonances appear as small inner loops



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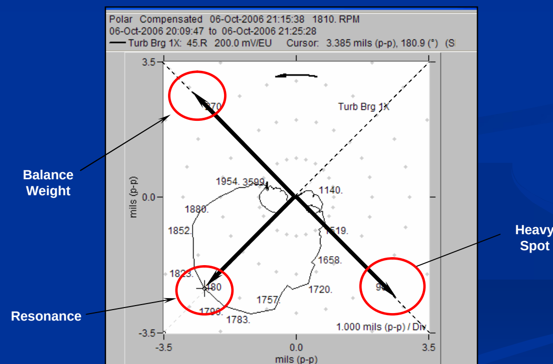
Polar Plots

- Heavy Spot, High Spot, and the "Balancing-T"
  - Heavy Spot for any resonance located in the direction the polar loop starts
  - Heavy Spot should be 90 degrees with rotation from the resonance peak
  - Phase angle well above resonance should approach 180 from Heavy Spot, and be 90 degrees against rotation from the resonance peak



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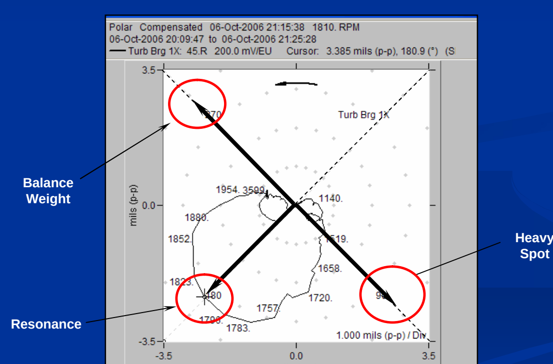
Polar Plots

- Heavy Spot, High Spot, and the "Balancing-T"
  - The Low Speed, Resonance, and High Speed data should all lead to essentially the same balance weight location.



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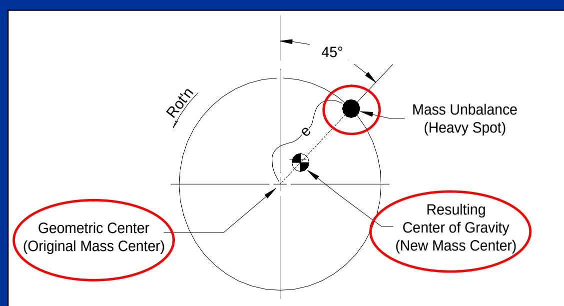
Why Do Resonances Occur?

- Unbalance creates a CG that is not coincident with the geometric center
- At low speeds, shaft rotation occurs around the geometric center
- As speed increases, the rotor seeks to "self balance", with the center of rotation migrating from the geometric center toward the mass center
- The 'self-balancing' process is what occurs as we pass through resonance
- At the resonance peak, rotation is centered half-way between the geometric and mass centers, resulting in maximum vibration



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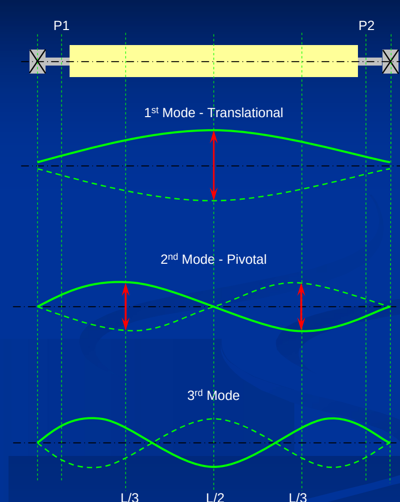
31

Resonances & Mode Shapes

- Rotor may have more than one resonance depending on operating speed and mass distribution
- Each resonance has an associated mode shape. For symmetric rotors:
  - 1st Mode is in-phase from end to end
  - Maximum deflection at rotor center
  - 2nd mode out-of-phase end-to-end
  - Maximum deflection at 1/3 points
- 'Single disks' such as 1-stage fans and pumps will have only 1 rotor mode (if they operate fast enough)
- Multi-disk (compressors, turbines, some pump) or distributed mass rotors (motors, generators) may have more than 2 modes
- Do not confuse rotor mode shapes with structural mode shapes



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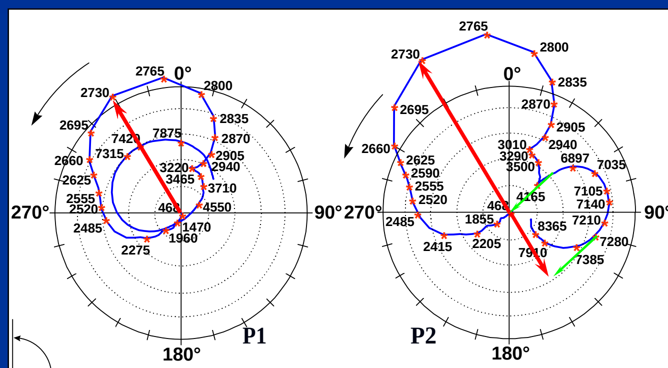
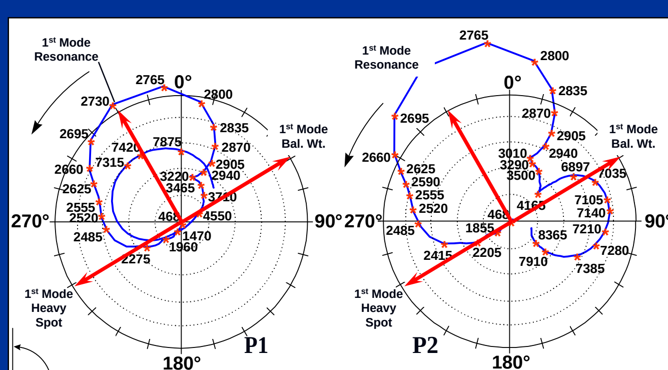
Polar Plots

- Typical 1st and 2nd mode responses
  - 1st mode at 2,730 rpm – in-phase across the rotor
  - 2nd mode at 7,420 rpm – out-of-phase across the rotor
    - 2nd mode must be compensated to remove residual 1st mode effects if present



Polar Plots

- 1st Mode balance solution – the classic “Static” Balance
  - Balance weights on both ends of rotor in same angular locations
  - We can balance the 1st mode without affecting the 2nd mode
  - The static balance weight pair will cancel each other out during the 2nd mode, having no effect on 2nd mode vibration



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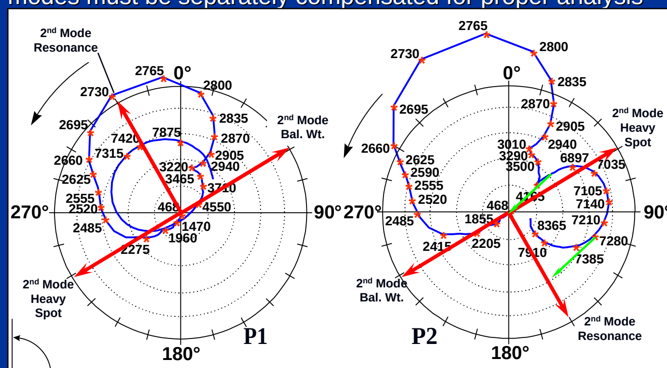
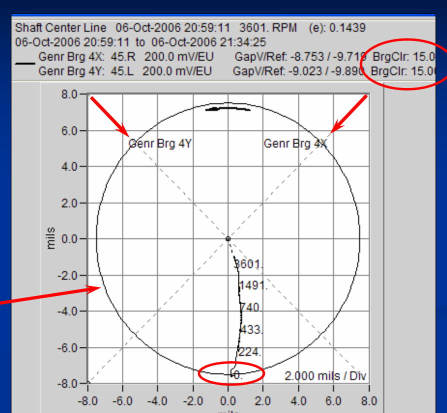
Polar Plots

- 2nd Mode balance solution – the "Couple" Balance
  - Balance weights on both ends of rotor in opposite angular locations
  - We can balance the 2nd mode without affecting the 1st mode
  - The couple balance weight pair at out of phase during the 1st mode, canceling each other out
  - 2nd modes must be separately compensated for proper analysis



Shaft Centerline Position

- Shaft centerline plot shows the average movement of a shaft within the bearing
  - Plots DC gap voltage changes from X-Y proximity probes
  - Only applies to proximity probes
- Plots should be made in relation to the available diametral bearing clearance
- We must also assume a reference position (typically the bottom of the bearing for horizontal machinery)



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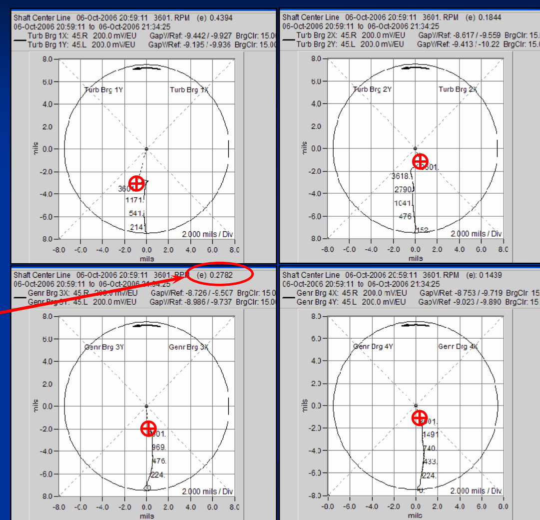
Shaft Centerline Analysis

- Used Qualitatively
  - Look at motion paths & final positions
  - Shows misalignment and preloads across the machine
- Used Quantitatively
  - Need accurate bearing clearance data!
  - Measure Bearing Wear
  - Eccentricity Ratio
  - Shaft Attitude Angle



Eccentricity Ratio

- Any forcing function placing a lateral (radial) preload on the rotor can result in a change is shaft centerline position.
- Preloads can result from:
  - Misalignment Thermal growth
  - Casing Deflection
  - Pipe Strain
  - Rotor rubbing
  - Pumping
  - Gravity
- Misalignment from Thermal Growth / Deflection is the main cause of excessive lateral preloading !
- We use Eccentricity Ratio as the key measure of the shaft's response to lateral preloads

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Eccentricity Ratio

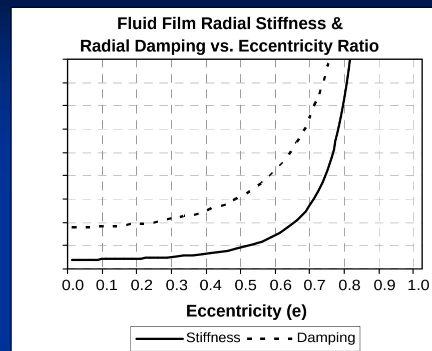
- Eccentricity is the ratio between the distance from the shaft center to the bearing center, divided by the <u>radial</u> bearing clearance
- Usually abbreviated as ‘e’ or ‘ER’
- Example 1:
  - Given: 15 mils <u>radial</u> bearing clearance
  - Shaft centerline data shows the shaft operating 3 mils from bearing center
  - ER = 3 / 15 = 0.2 → running ‘high’ in the bearing
- Example 2:
  - Shaft is operating 9 mils from the bearing center
  - ER = 9 / 15 = 0.6 → typical
- If the shaft is centered in the bearing, what is ‘e’?
  - ER = 0 / 15 = 0
- If it is against the bearing wall?
  - ER = 15 / 15 = 1

Average vs. Dynamic Eccentricity Ratio

- The previous slides showed ‘Average’ ER. This is the shaft position obtained when we only consider the DC Gap Voltage data
- If we also consider the shaft vibration at any given average position, we have the Dynamic Eccentricity Ratio
- For example, given 15 mils radial bearing clearance:
  - If shaft centerline data shows the shaft 9 mils from bearing center, Average ER = 9 / 15 = 0.6
  - Given 6 mils pk-pk of 1X shaft vibration using 200 mV/mil probes, the dynamic shaft motion varies +/- 3 mils from the average position
  - Dynamic ER = Avg ER +/- [(peak-to-peak vibration / 2) / radial clearance]
- For our data:
  - Dynamic ER = 0.6 +/- [(6 / 2) / 15] = 0.6 +/- 0.2
- So the Dynamic ER varies from 0.4 to 0.8 with each shaft rotation
- At 0.8 we see the shaft is in close proximity to the bearing wall. We might want to reduce the vibration and/or modify the alignment to reduce babbitt stress

Stiffness & Damping vs. Eccentricity

- The radial stiffness and damping of the lubricating fluid within a bearing are functions of eccentricity ratio
- Increased eccentricity results in non-linear increases of stiffness and damping
- What effect does this have on our resonance? AF?
  - Observed frequency will increase
  - Observed AF will decrease
- Practically speaking: eccentricity ratio and shaft position should be an integral part of the transient vibration analysis process



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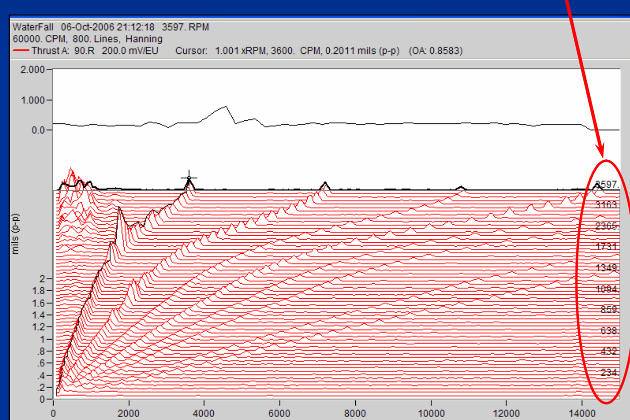
41

Waterfall & Cascade Plots

- Waterfall and cascade plots are three-dimensional graphs of spectra at various machine speeds and times. They allow us to see the entire frequency content from a location as a function of speed.



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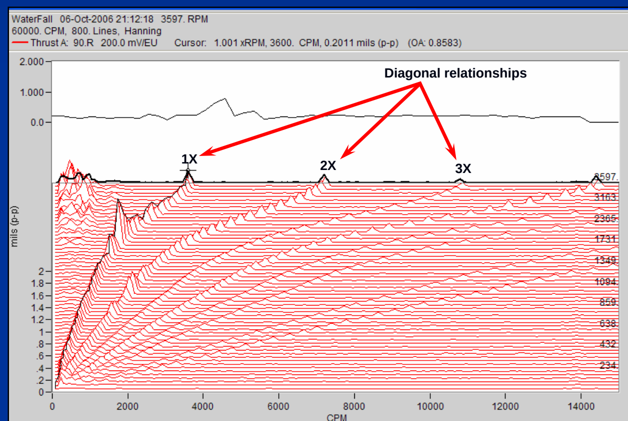
Waterfall & Cascade Plots

- Orders of running speed (1X, 2X, 3X, etc.) form near-diagonal lines in the plot



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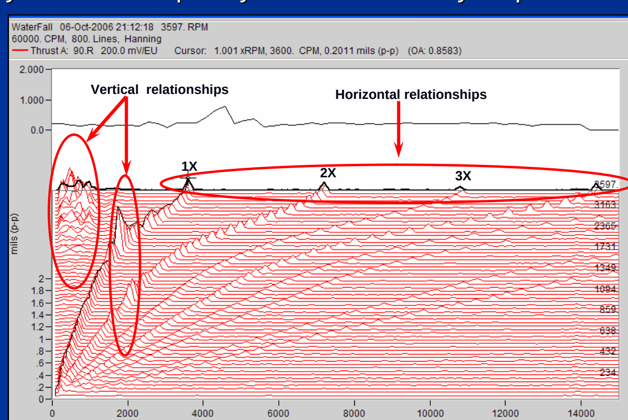
43

Waterfall & Cascade Plots

- Horizontal relationships can be analyzed for resonance, looseness, rubs, instability, etc.
- Vertical relationships onset and progress of resonant related activity, and any constant-frequency vibration that may be present.



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44

Waterfall & Cascade Plots

- Vertical relationships may be due to:
  - Adjacent machinery – check their speeds against your data
  - Ground faults or electrical noise in your instrumentation – the peaks will line up vertically at 60 Hz (3600 cpm)
  - Structural resonances will get easily excited by orders of running speed as machine speed increases or decreases
  - Oil Whirl: look for subsynchronous vibration at 0.4X – 0.48X that tracks running speed
  - Oil Whip: look for subsynchronous vibration in the at a frequency equal to the rotor's 1st lateral balance resonance
  - Anisotropic shaft stiffness due to cracking or by design – look for excitation of the 2X order line at ½ the balance resonance

Conclusion

This presentation was compiled from the author's paper, "Transient Vibration Analysis, Insights into Machinery Behavior," presented 07-Dec-2007 at the Vibration Institute's Piedmont Chapter meeting in Halifax, NC.

For a copy of that paper, please contact the author:

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