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Turbine Generator Balancing – Identifying Modally Effective Solutions

29-Jan-2009

By:
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Rotating Machinery Services for… "Maintenance That Matters"
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30 Minutes is Not Enough…..

We probably won't be done with this discussion in 30 minutes, so:

- You can download the entire presentation at: www.mbesi.com/downloads/downloads.htm

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Hurry Up! Put a Weight In…..

Did you ever:

- Need several runs to get good results?
- Balance a 'simple' fan, thinking it should take 1 shot '180 out', only to need several runs to finish the job?
- Have OK running speed vibration but high vibration in resonance?
- Wonder why operators will start up a unit fast but can't control it on the way down, and how does this affect vibration?
- Wonder how to balance a 1^{st} or 2^{nd} critical, while also achieving good vibration levels at running speed?

How Much Does Ineffective Balancing Cost?

- What are some of the 'costs' of ineffective balancing?
  - More runs = more time with high vibration = more stress = less safety
  - More runs = extra Operations time = more labor $
  - More runs = more Lock-Out / Tag-Out = more time
  - More runs = extra analyst & millwright time = more labor $, typically OT$
  - More runs = less 'visible' success for you personally

- These all are important, but the highest cost is lost generation

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How Much Does Ineffective Balancing Cost?

- Using a 100 MW plant and $100 / MW-hr as an example, gross revenues lost per day of downtime are:
  - Power produced = 100 MW x 24 hrs = 2,400 MW-hrs
  - Revenue = (2,400 MW-hr) x ($100/MW-hr) = $240,000 per day
- So **1** extra startup when balancing a main turbine causes about 1 shift of generation to be lost; cost = $80,000 per 100 MW
  - This could be **much higher** during peak generation periods
  - But who's counting…..
- Similar production losses can be incurred on critical, non-spared boiler fans or feed pumps if they cause downtime or reduced generation
- So does that first balance shot count?

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What Makes Balancing More Difficult?

- Not having the right vibration transducers installed
- Not having the right diagnostic hardware
- Insufficient training and / or experience
- Not enough time to stay proficient with skills & tools
- Lack of good historical balance data
- Not understanding the machinery dynamics

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As Consultants, What Do We See?

- From our perspective many items contribute to the problem:
  - Permanent monitoring systems mis-configured or malfunctioning
  - Insufficient diagnostic instrumentation
  - Not enough manpower and experience with an array of machines
  - Not knowing what else causes 1X vibration in turbo-machinery
  - Not having thermal growth data & an accurate hot alignment
  - Not understanding how misalignment hampers balancing
  - A lack of historical balance response data
  - Can't calculate shots from historical data for reliable, 1st-shot performance in critical situations
  - Insufficient understanding of transient data

Essential Rotor Dynamics

- We will focus this discussion on several aspects of rotor dynamics that are important when balancing turbo-machinery:
  - Rotor Type - Rigid vs. Flexible Rotors
  - Rotor Resonance (Critical) Identification
  - Rotor Mode Shape Identification
  - Balancing Strategy

- Mode Shape identification, and modally correct balancing, becomes very important when working with the solid-flanged couplings found on T-G units.

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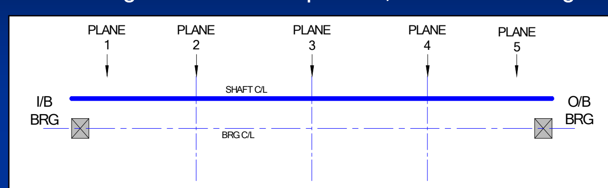
Rigid vs. Flexible Rotors

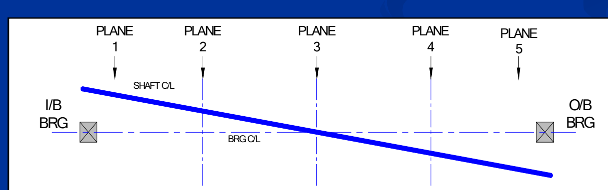
- Rigid Rotors:
  - Operating speed < 70% of 1st balance resonance (critical)
  - Or: Unbalance can be corrected in any *two* arbitrarily selected planes. After correction residual unbalance does not change at any speed up to maximum operating speed
  - Examples include most gearbox shafts, electric motors, pumps and fans equipped with rolling element bearings, and other similar machines
  - Balance weights can be placed "180° out"
- Flexible Rotors
  - Just about everything else!
    - Most High & Low Pressure steam turbines; all 1,800 & 3,600 rpm generators
    - Most boiler feed pumps and large boiler fans
  - Phase changes > ~30° indicate approaching resonance / flexible rotor
  - Balance shot angle must take this into account – can't just go 180-out

Rigid Rotor Mode Shapes

- 1st Rigid Mode – aka bouncing or translatory mode.
  - Shaft and bearing centerlines are parallel; no shaft bending



- 2nd Rigid Mode – aka conical rocking mode
  - Shaft & bearing centerlines parallel; nodal point at center; no shaft bending



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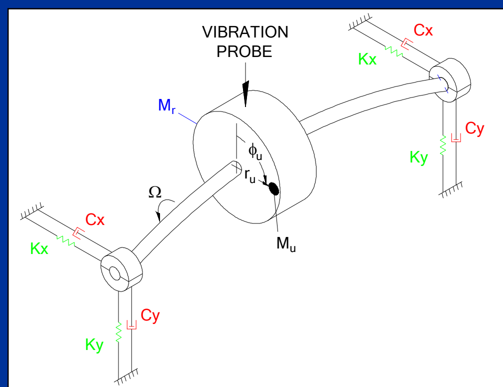
Flexible Rotors – The Jeffcott Model

- A simple model used to help explain rotor dynamics
  - One or more rigid disks mounted on a flexible shaft
    - Number of disks dictates number of resonances likely seen
  - 2 degree of freedom system subjected to rotating unbalance
  - Restraint of vibration provided by shaft, bearings and supports



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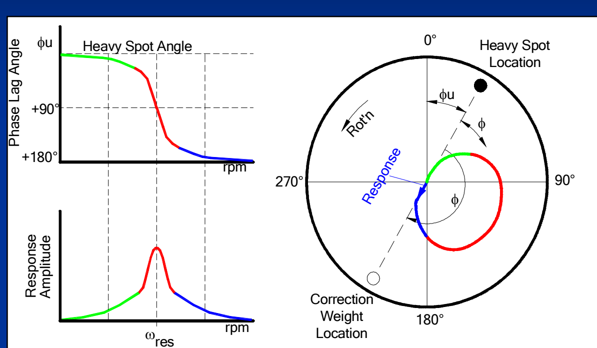
Understanding Resonances

- As RPM increases the rotor will go through one or more resonances. The 1X-filtered bode and polar plots look as follows:



- Heavy Spot is 90° with-rotation from resonance peak
- It is also the phase angle at low speed (data must be slow-roll compensated)
- Correction Weight is placed 180° from Heavy Spot

- Resonance is defined by 2 essential characteristics:
  - Phase change: 90° at resonance; ~180° above resonance
  - Amplitude peak at resonance; subsides above resonance

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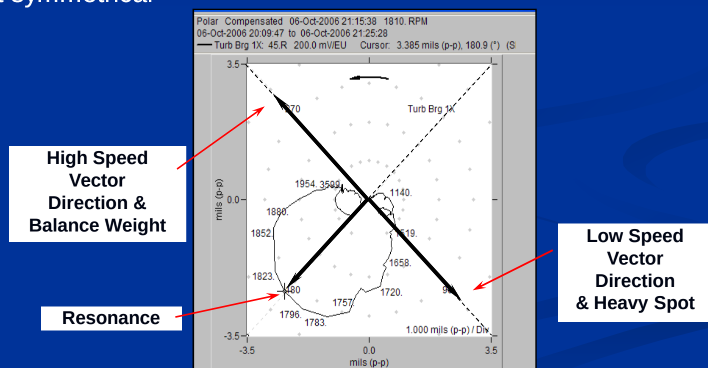
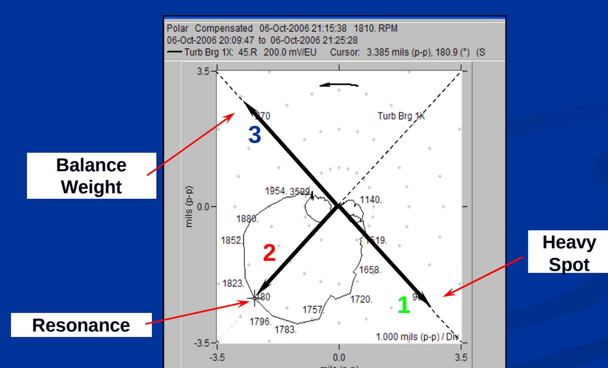
The 'Balancing-T'

- The Balancing-T uses three pieces of data to determine the most likely Heavy Spot using rotor response theory
  - Developed by Stan Bognatz
- It is an effective tool for evaluating a resonance loop when that loop is influenced by the modes of adjacent rotors or structural modes, and is not symmetrical



Drawing the 'Balancing-T'

- Leg 1 aligns with the low speed vectors & indicates Heavy Spot
- Leg 2 aligns with resonance; must be 90° APART from Leg 1
- Leg 3 aligns with high speed vectors, 90° APART from Leg 2; indicates Balance Weight location
- Rotate the T to get the best overall fit to the resonance loop



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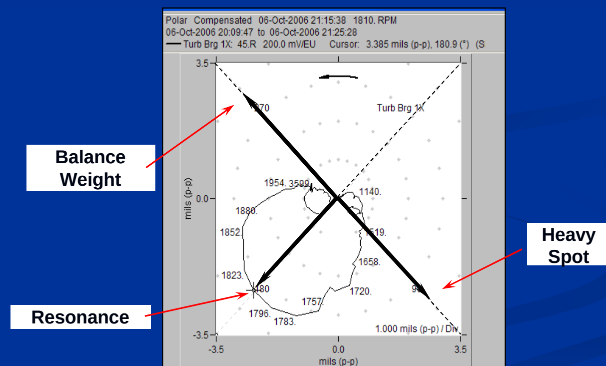
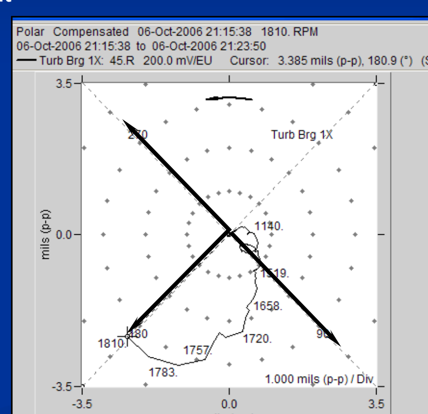
The 'Balancing-T'

- By using the Balancing-T we can be confident we have properly identified the Heavy Spot as well as possible for the given data
- Place a balance weight 180° from the Heavy Spot to begin balancing
- You should examine the X & Y Probes for both bearings to get the best indication for the rotor system; average the data as needed



Using the 'Balancing-T'

- The Balancing-T is very useful when operating speed is at or near resonance
  - A typical polar plot for resonance speed range operation is shown
  - Note there is only about ½ of a 'full' polar loop
- Orient Legs 1 & 2; look for best fit
- Balance Wt. in line with Leg 1



In general, balancing near resonance is a sensitive process.

Minor changes in the balance weight angle or amount cause non-linear response in the 1X vectors.

If a resonance occurs at the normal running speed, proceed with caution.

It is recommended that single-point calculations be performed using both running speed data and points well below resonance.

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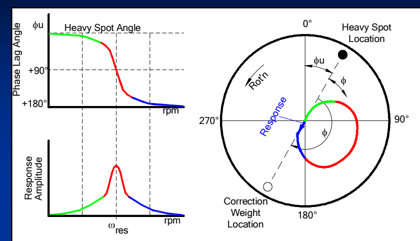
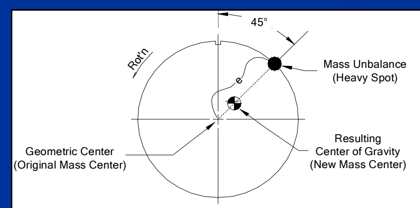
Self-Balancing Effect

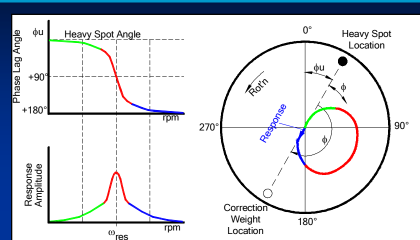
- Why does vibration peak then decrease after resonance?
- At Low speed:
  - Shaft rotates around geometric center
  - Bearing stiffness (K) dominates the total dynamic stiffness, & thus the response
- Centrifugal unbalance force increases as a function of speed squared (c.f. = m r ω^2)
  - Causes center of rotation to migrate outward toward mass center
  - Vibration increases as rotation center moves outward (due to c.f.)

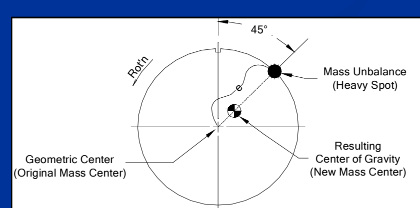




Self-Balancing Effect

- At resonance: rotation center is mid-way between geometric center and mass center
  - Dynamic stiffness at resonance is controlled by damping ($C_{XY} \propto V$)
- Above resonance:
  - Mass stiffness dominates the synchronous dynamic stiffness ($F \propto a$)
  - Mass stiffness rate of change overtakes the centrifugal force rate of change, driving vibration down
  - Eventually the rotation center coincides with the mass center
    At this point the vibration is minimized and the rotor appears to have "self-balanced"





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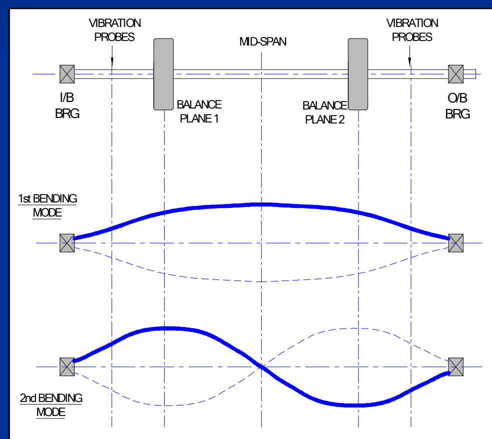
Modal Shapes

- Rotor made shapes for the 1st and 2nd modes shown below
  - Note phase responses on inboard and outboard probes for each mode.



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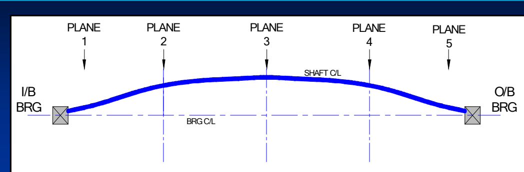
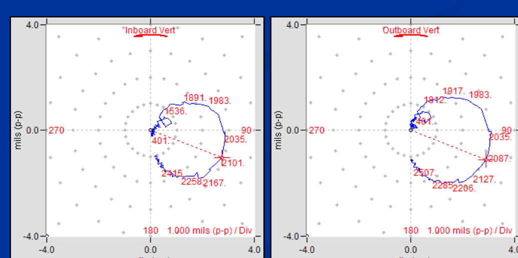
Flexible Rotor – 1st Bending Mode

- 1st Bending Mode
  - 1X phase angles are nearly the same all along the rotor
  - Deflection (vibration) is maximum at the rotor mid-span, decreasing near the bearings



  - 1X vector data from the inboard and outboard bearings shown here
  - Resonance at 2,100 rpm
  - Both bearings show identical in-phase responses, indicating 1st bending mode



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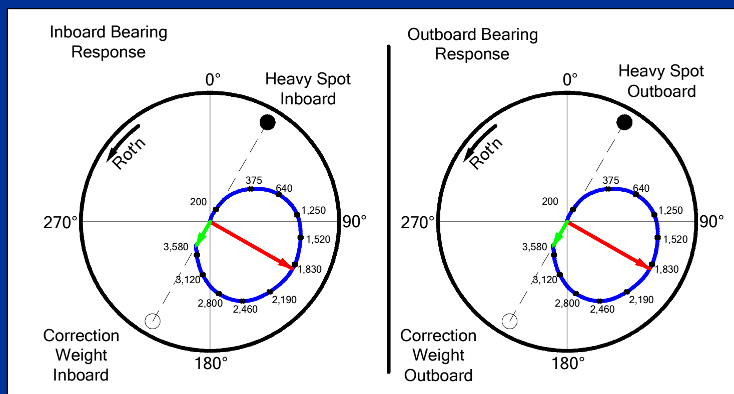
Balancing the 1ST Mode Response

- 1st mode response shows in-phase relationship of polar loop peaks
- Heavy Spots at both ends of the rotor are in-phase
- Given similar vibration, both disks should be balanced similarly



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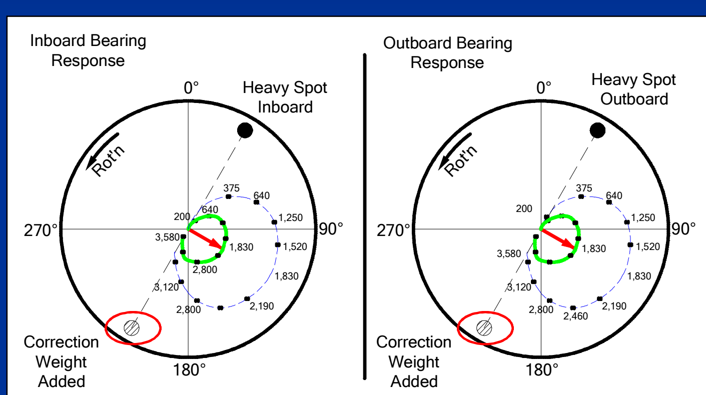
Balancing the 1ST Mode Response

- To properly balance the 1st mode of a two-plane or multi-plane rotor, we must address the unbalance at both ends of the rotor
- We must place equal balance weights at the same angular locations in each end of the rotor – a "static shot"



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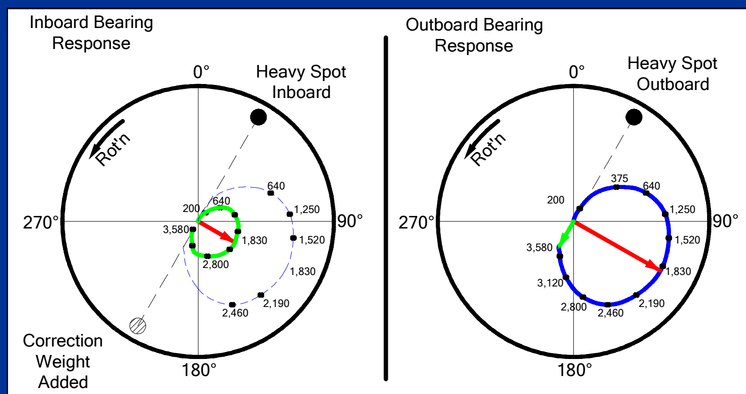
Balancing the 1ST Mode Response

- Could the rotor be balanced by placing 1 weight in one disk?
  - One end would respond favorably
  - Other end should be lower too, but difficult to predict
  - Would definitely affect the 2nd mode negatively



Balancing the 1ST Mode Response

- Using equal weights maintains a 1st mode specific response.
- When rotor enters 2nd mode the weights will be 180° out-of-phase and will cancel, thus not affecting the 2nd mode response.
- Unequal weights introduce an effect on the 2nd mode, which we are specifically trying to avoid when using this technique.

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Balancing the 1ST Mode Response

- After a "static shot" observe the polar plot changes to assess the effect of the weight pair
- Draw scaled Balancing-T's on the polar plots to visualize the C vectors at resonance and running speed
- Adjust the static shot weight pair to minimize resonance vibration
- By driving resonance vibration as low as possible, vibration above resonance (before any higher modes) will be very low !!

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Balancing the 1ST Mode Response

- Standard balancing theory calls for equal amounts of weight placed in both ends of the rotor, which is usually sound advice
- If the rotor has an accessible center balance plane, you should consider placing a single balance weight there first. It will have the best effect due to the mode shape having its highest deflection at the rotor's center.

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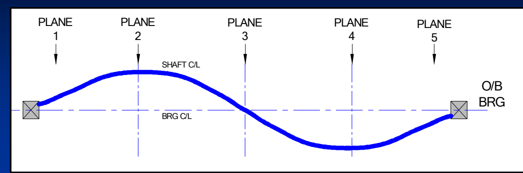
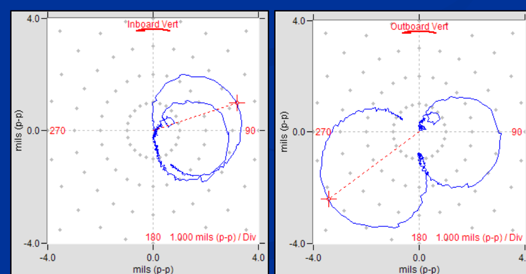
Flexible Rotor – 2nd Bending Mode

- 2nd Bending Mode
  - 1X vector phase angles are out-of-phase at the ends of the rotor
  - Amplitude maximums at rotor ¼ points
  - Nodal point at mid-span



  - The rotor ends will be 180° out-of-phase, while the 1st bending mode remains in-phase
  - The 1X polar plots for one end will exhibit a "double loop" appearance
  - The other end will have a "figure eight" appearance



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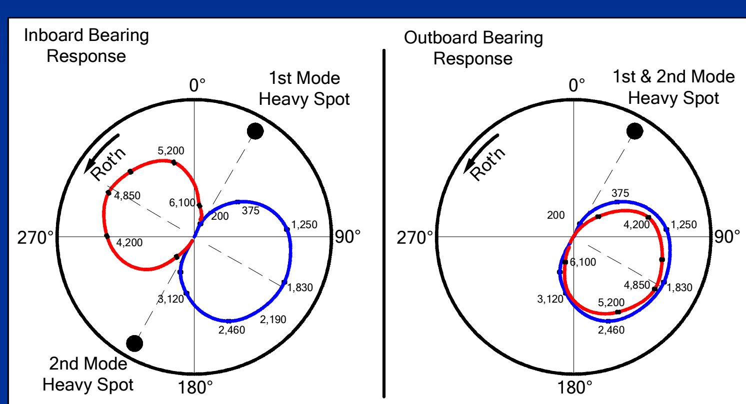
Balancing the 2ND Mode Response

- Typical 1st and 2nd mode responses shown below
- 1st mode resonance peak in phase at 135°, at 1,830 rpm
- 2nd mode resonance peak occurs at 315° inboard bearing, at 4,850 rpm
- 2nd mode resonance peak 135° on outboard = 1st resonance peak



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Balancing the 2ND Mode Response

- Since the two modes are out of phase on the inboard bearing, the Heavy Spot locations will be out of phase as well
- This means that if a single balance weight were to be used on the inboard bearing balance plane, it would be nearly impossible to reduce vibration at one speed, or in one mode, without adversely affecting the other mode
- On the outboard bearing both modes and thus both Heavy Spots are in-phase, and a single balance weight would likely reduce both resonances simultaneously
- A single balance weight placed in the outboard bearing plane would help reduce the 1st mode vibration on the inboard bearing, but it would cause the 2nd mode vibration to increase

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Balancing the 2ND Mode Response

- The ideal procedure to achieve a good balance quality for both the 1st and 2nd modes is:
  - First: use a “static shot” to minimize the 1st mode response. This will not affect the 2nd mode.
  - Second: use a “couple shot” to correct the 2nd mode. This will not affect the 1st mode.
  - The second mode will prove far more sensitive to balance weight changes due to the much higher speed.
  - Use the Balancing-T to scale and adjust the static and couple shots to achieve acceptable vibration; look at X & Y data at both bearings
  - On large turbine-generator units, and industrial gas turbines, try to keep 1X shaft vibration during resonance below 5 mils-pp. This will help ensure running speed vibration will be low.

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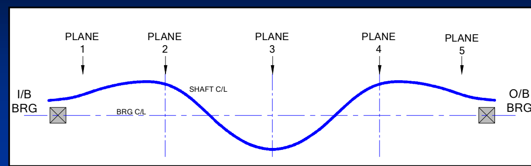
Flexible Rotor – 3rd Bending Mode

- 3rd Bending Mode
  - Double-hump shape
  - 2 nodal points
  - 3 points of high amplitude
  - Generally only on the largest 3600 rpm generators, aero-derivative gas-turbine engines, and high-speed compressors



  - 1st and 2nd modes balanced as noted above
  - 3rd mode may require center-plane shot (often only available in shop)
  - 3rd mode will look like a 1st mode at the bearings; watch for precense of 2nd mode in the transient speed data
- Nodal points
  - Note that a rotor will only have nodal points directly at the bearings for each mode when the bearing stiffness is sufficiently high.
  - As bearing stiffness becomes relatively lower, the nodal points will have a tendency to move outside of the bearings

Resolving Running Speed Vibration

- Is vibration at operating speed & load acceptable?
  - Be sure to monitor full range of generator loads to document 1X load vectors
  - Slow roll compensate all data to get true dynamic vibration levels before deciding
  - Amplitude on permanent vibration monitors can be misleading - significant slow roll is often present on journals - often 0.5 to 1.5 mils on older machines!

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Approaches to Multi-Plane, At-Speed Balancing

- #1: 2-Plane Influence Vector Solution
  - "Fixed Point" numerical solution – only 1 speed / load data point
  - 2 planes = 2 vibration probes for input
  - Software / calculator required; check data by graphing each plane
  - Works great when historical influence vector data is available (oz/mil, mils/oz, lag)
  - Can create a solution without historical data – need 2 calibration runs + solution run
- #2: Least-Squares Balance Solution
  - Numerical solution; software / calculator required; powerful & flexible
  - Addresses mix of speeds, loads, vibration probes and balance planes
  - Data points can be weighted for greater influence on solution
  - Does not address modal shapes (resonances) if data is not properly selected
- #3: Static-Couple Solution
  - Fixed Point Solution – usually done graphically; 2 planes = 2 probes
  - Historically used by OEMs (GE, Westinghouse, etc.)
  - Addresses static & couple components separately
  - Works well; be sure to use slow roll compensated data

- Use more than 1 method to cross-check results; solutions must agree / compliment highest observed mode or resonance vibration will increase

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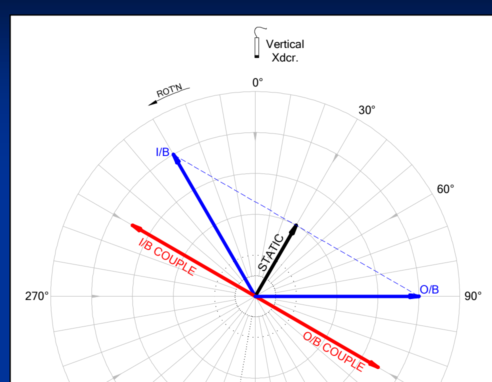
Static / Couple Derivation

- Draw I/B and O/B vectors
- Connect the vectors heads
- Bisect the connecting line – this is the static component
- Transfer the connecting line to the origin – this is the couple component
- Balance sensitivity (weight & lag) for each mode should be determined from historical data



  - Example: for 2 mils of static unbalance, with a sensitivity of 8 ounces per mil, a static shot of (8x2) = 16 oz would be required.
  - If the rotor was well above the 1st resonance (as shown by the dominant couple mode) the weight would have very little lag and be placed at the angle shown

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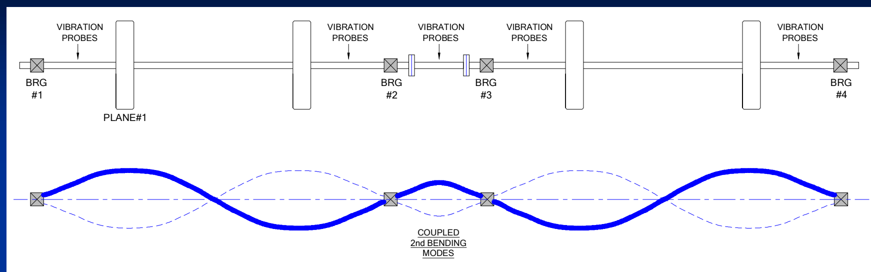
Multiple Rotors



- There are many combinations of mode shapes in the field
- First, review rotors individually to determine their solutions
- Then, see if individual solutions complement adjacent rotors
- It is not uncommon to successfully balance a rotor and drive vibration up on adjacent rotors; may need to do multiple planes & rotors
- Do not neglect long jack-shafts – they can often be used effectively for trimming running speed vibration

Thank You – Any Questions?