Proactive Shop Strategy to ensure a smooth post-outage startup without field balancing

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Primary Goal

- Planned outages should result in a smooth startup the first time, without the need for field balancing.

- Four categories of focus:
  - Outage planning
  - Runout measurement and assessment
  - Balancing
  - Field alignment

Two Key Causes of Post-Outage Vibration

1. Unmeasured, unobserved, and uncorrected **non-perpendicular rotor couplings**

2. Improperly balanced (or unidentified) residual **distributed mass eccentricities**

   - Both are "static" causes integral to the rotor, which can be proactively identified and resolved in the shop
   - By resolving these two areas, a smooth restart can be ensured

The Risk of Assumptions

- Applying OEM methods and assumptions about new rotors to used service rotors in the shop, ***without proper and thorough verification***

  - Assuming rotors are concentric
  - Assuming couplings are perpendicular

  And especially problematic…

  - Assuming that **any** found defects can be "balanced"

- Outage scope must incorporate the complete and thorough verification of the above points, with the correct and necessary procedures defined and quantified

Outage Planning

- Review and amend outage scope ahead of time to incorporate points of assessment to better assure smooth turbine-generator dynamic operation
- Must review shop procedures, and service provider contractual Terms and Conditions (T&Cs) for ability to make amendment(s)
- Synchronize plant outage schedule with shop work activities based on amended outage scope

Key Outage Steps

1. Condition assessment of rotordynamic behavior (and alignment) prior to & during shutdown by collecting vibration data

2. Thorough physical runout measurement and mathematical 1x and 2x evaluation (full body, couplings, faces, rims)

3. Machining (if determined necessary)

4. Balancing by Quasi-High Speed Balancing method in 2N+1-planes (minimum three planes) on balancing machines

5. Verification of 16-point coupling rim/gap measurements during reinstallation and (re)alignment based on improved rotor train condition

Outage Planning

- Guarantees identification and resolution of all eccentricities, whether induced from misalignment or intrinsic to the rotor or couplings
- These eccentricities are the basis of unwanted vibration and damaging forces when rotor is returned to operation
- Resolution of found problems is based on specific unit data and facts alone
- Takes into account true rotor-bearing/support behavior, and eliminates assumptions, leaving no “surprises”

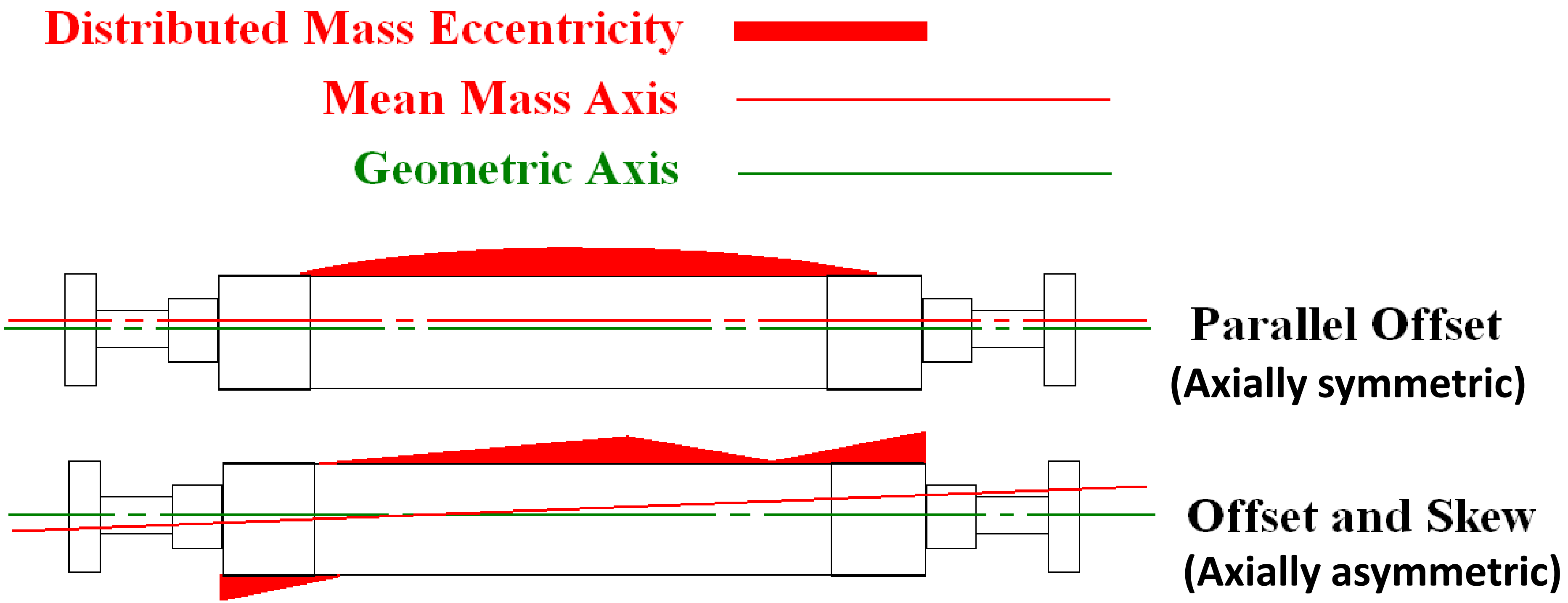
Current Rotor Service Procedures

Specifically, regarding balancing methods, and field alignment methods and tolerances…

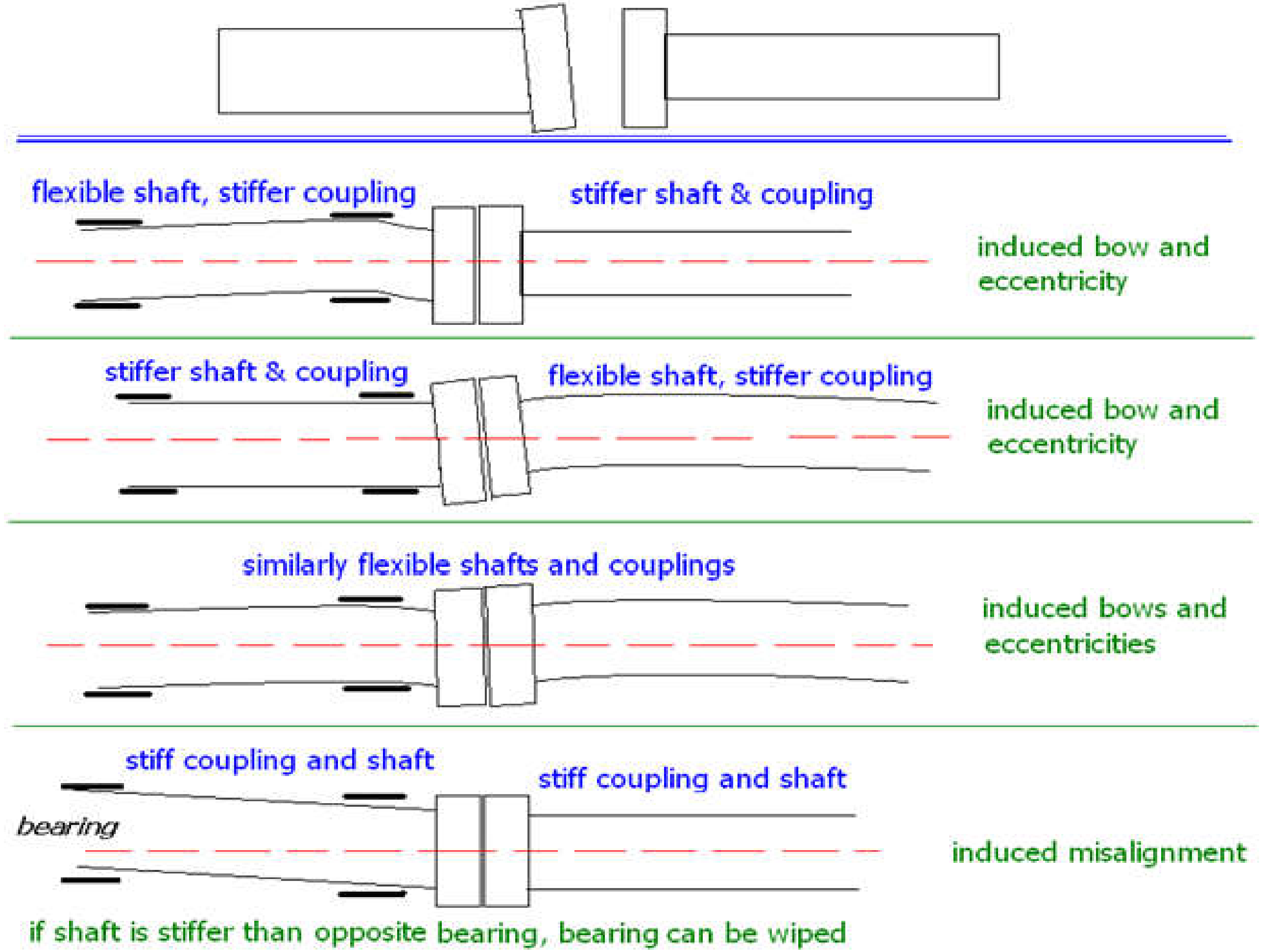
- Developed for and work well for **NEW** installations, with all rotor tolerances to OEM design and factory specs
    - Procedures contain **assumptions** on rotor condition
    - It is **required** that rotors meet factory dimensional specs for the standard methods to be reliably successful

Rotordynamic Effects of Eccentricity

- Definition of eccentricity: (differs from concentrated "unbalance")
  - Any distributed mass that notably alters or shifts the overall mean mass centroidal axis of the rotor itself (> 2 mils)



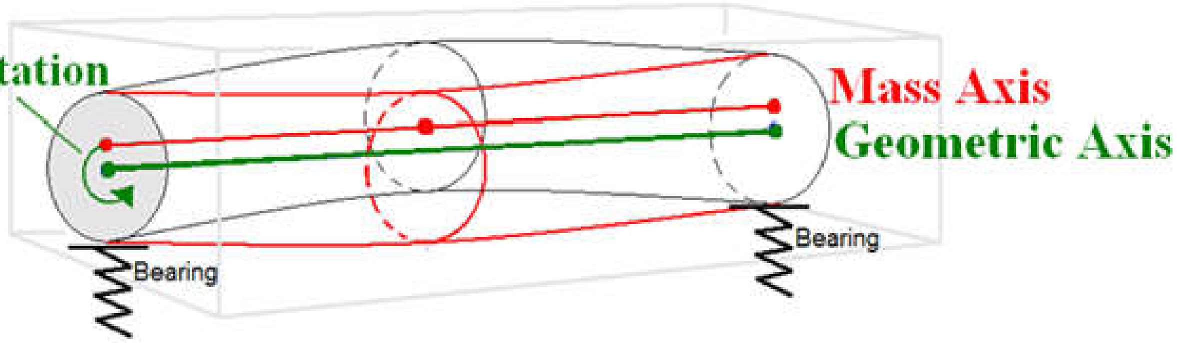
Induced Eccentricity from Off-Square Couplings



Bowed/Eccentric Rotor: Mass Axis not Coincident to Geometric Axis

We want the rotor to spin balanced about its geometric axis at all speeds...

Like this:



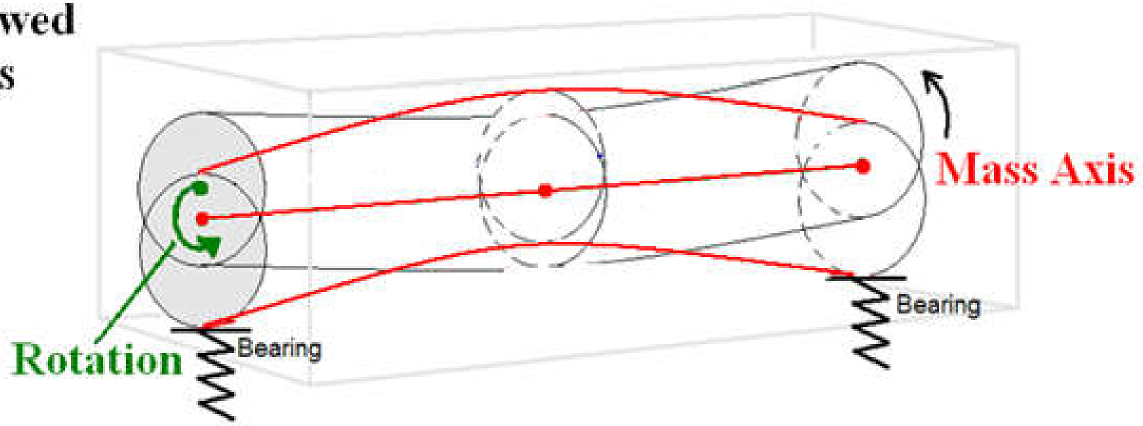
However,

Any rotor's natural tendency is to rotate about its actual center of mass axis, which is offset due to eccentricity

When coupled, this natural tendency is constrained. This produces forces and vibration.

Natural tendency of a bowed rotor is to rotate about its center of mass axis.

Like this:



(produces very high bearing forces if bearing clearances are insufficient to allow the increased displacement)

Resolving Eccentricity

- Our goal is to bring the mass axis coincident to the rotor's journal axis
  … by "mirroring" it with balancing weights, not by "unbending" the rotor
- This ensures the rotor's natural state of rotation is about its journal axis, in line with its couplings
- **All eccentricity can be found and resolved in the service shop before installation and startup**

Pre-Outage Condition Assessment

Get prior to and during shutdown:

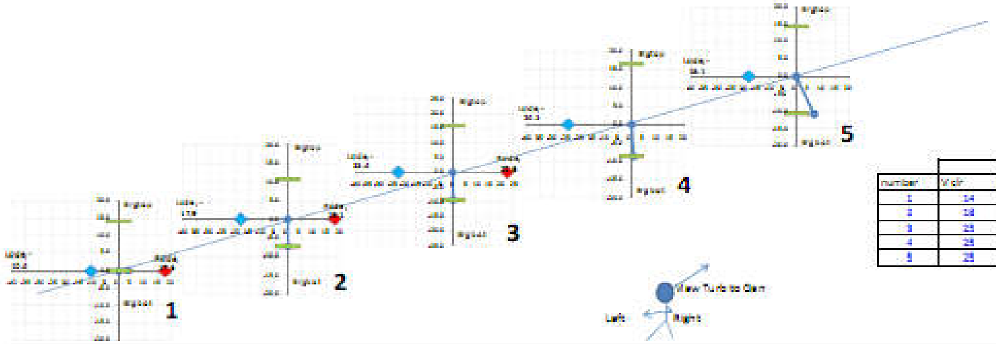
- DC shaft centerline position **from standstill** (off gear) through 1st critical speed range and to full speed/load
- Vibration amplitudes/phase through all speeds, with two probes per axial location if at all possible
- Shaft orbits through all speeds
- Bearing and pedestal seismic readings
- Bode, Polar, and Full Frequency Spectrum plots

Pre-Outage Condition Assessment

Purpose:

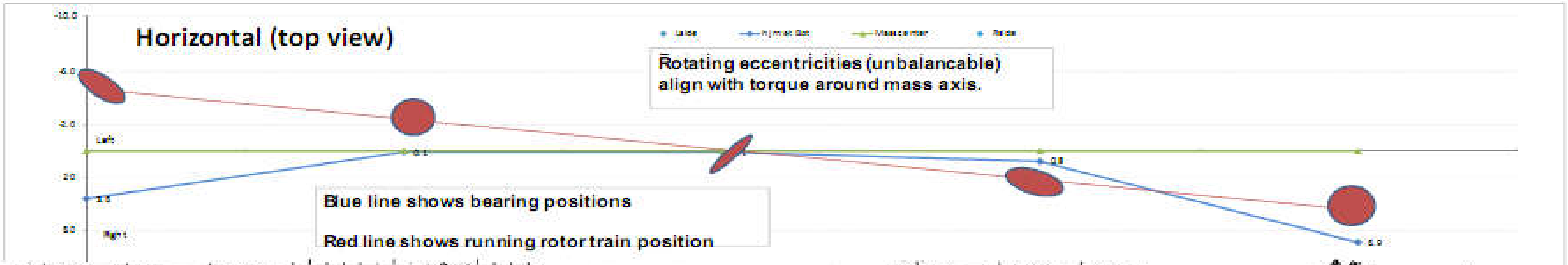
- Verify dynamic condition, resonances, evidence of eccentricities or misalignment, static stability of journals (SCL path) or other problems
- Can point to root cause of vibration issues, and identify possible solutions, and help with scheduling machine shop work if needed
- Determine operating deflection shape (ODS)
- Determine alignment condition and bearing positions

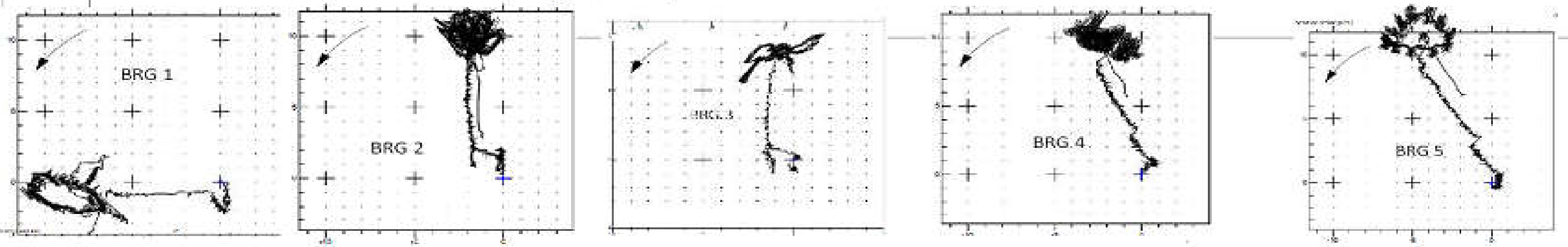
Pre-Outage Condition Assessment



Alignment Verification

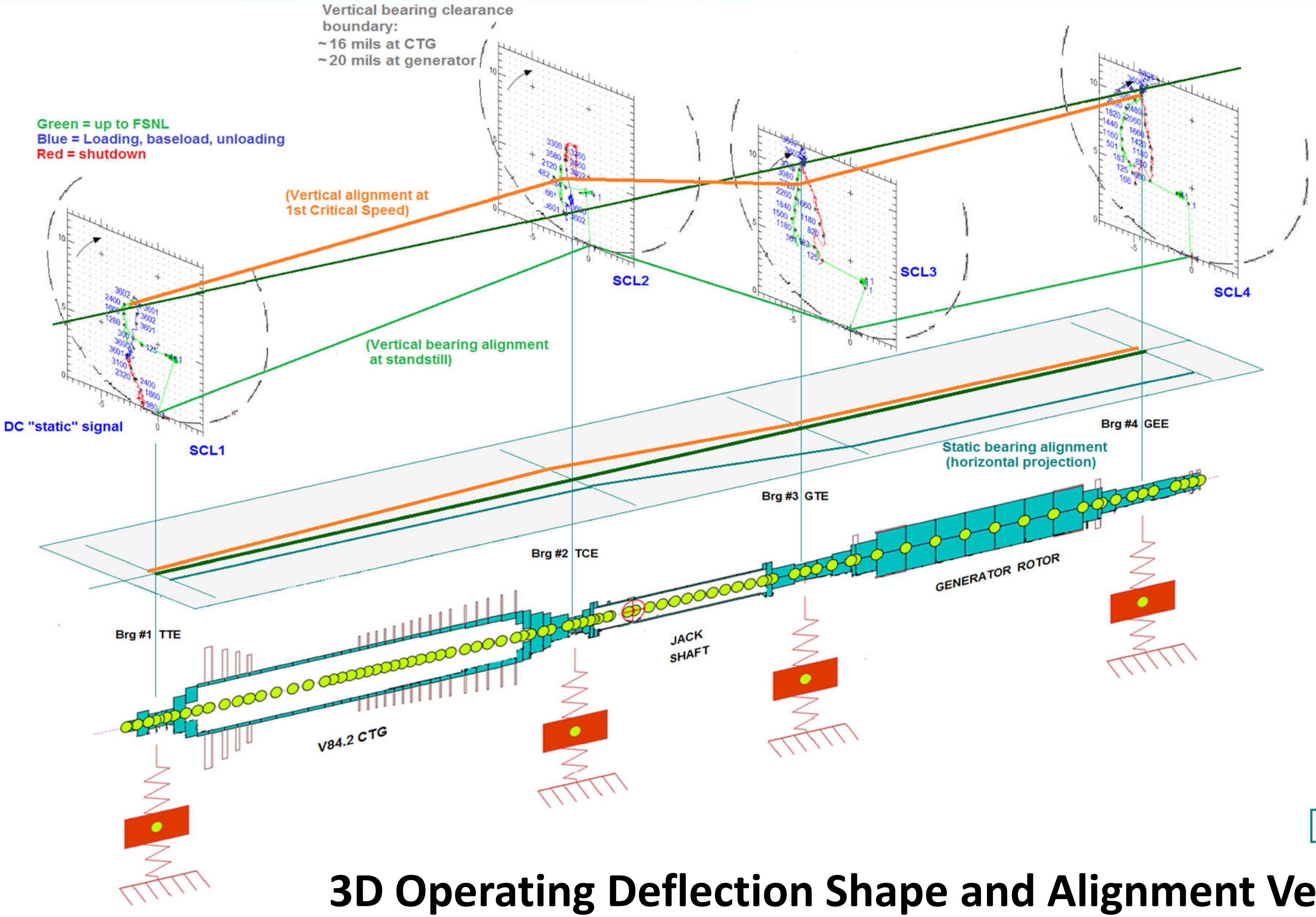
			distance		Cold Position			Bearing Geometry						
number	V clr	H clr	between brg		Vert	Horz	lgth	Lside	Rside	Brg top	Brg bot		h jrn at Bot	
1	14	28	10		-0.2	-3.6	0	-10.4	17.6	14.2	0.2	0	3.6	0
2	18	35	10		7.3	-0.1	10	-17.9	18.1	10.7	-7.3	0	0.1	0
3	25	45	10		9.3	-0.1	20	-22.4	22.6	15.7	-9.3	0	0.1	0
4	25	50	10		8.6	-0.8	30	-24.2	25.8	16.4	-8.6	0	0.8	0
5	25	50	10		10.8	-6.9	40	-18.1	31.6	14.2	-10.8	0	6.9	0





This is a case study we'll look at later...

Pre-Outage Condition Assessment



The straight dark green line represents a linearized catenary line (set at load)

The orange line represents the 3D view of the rotor train journal operating alignment at the 1st critical speed.

The lighter green line shows the 3D view of shaft and bearing alignment deviation at standstill, assuming the shaft is resting on the bottom of the bearing, and assuming a torque-induced straight operating line upon reaching base load.

It appears that the TCE bearing is high.

The horizontal projection here shows the bearing standstill position relative to a reference of the straight-line mass axis of the rotor train under maximum drive torque and inertia.

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3D Operating Deflection Shape and Alignment Verification

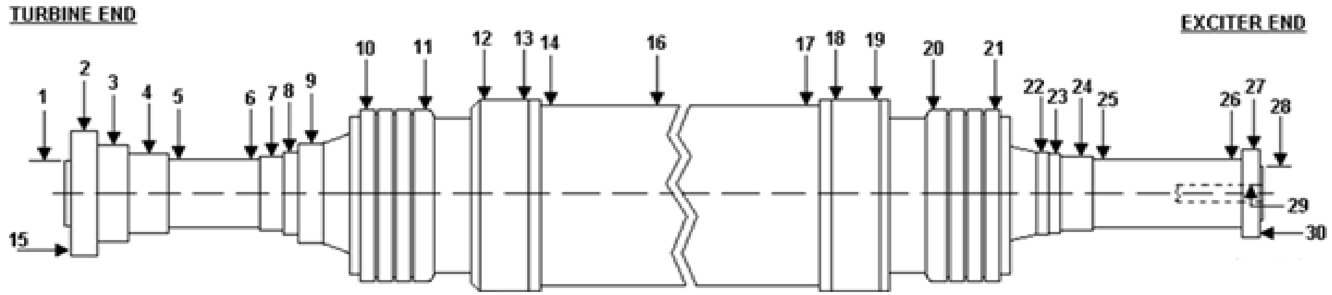
Service Shop Procedure: Runout Evaluation “As Received”

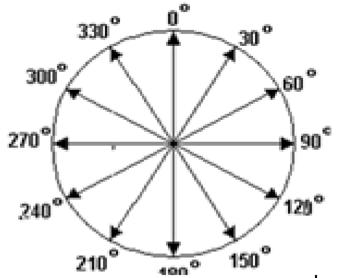
- TIR (total indicator runout) measurements and evaluation of 1x eccentricities are a critically important step
- Provides a clear map for scheduling required work and procedures to resolve all eccentricities
- No room for assumptions or skipped measurements (especially coupling faces)

- **We can identify FIVE essential conditions that must be met in the shop regarding TIR evaluation…**

Service Shop Procedure: Runout Evaluation

- **Requirement #1: Record sufficient data points**
- Record data points every 45^{o} radially (better, 30^{o}), including coupling rim and face
  - At least 8 – 12 points per measurement plane
- Record data at each axial point of diametral change of the rotor

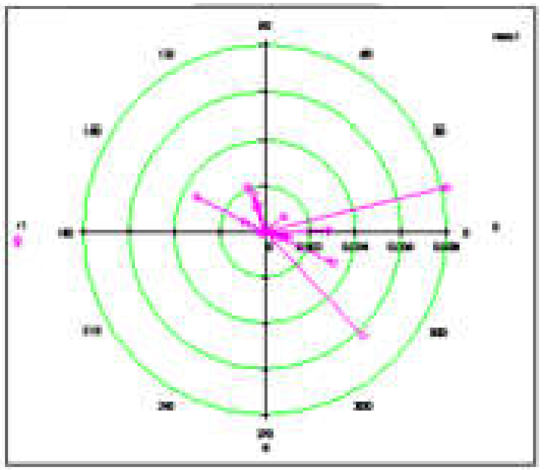


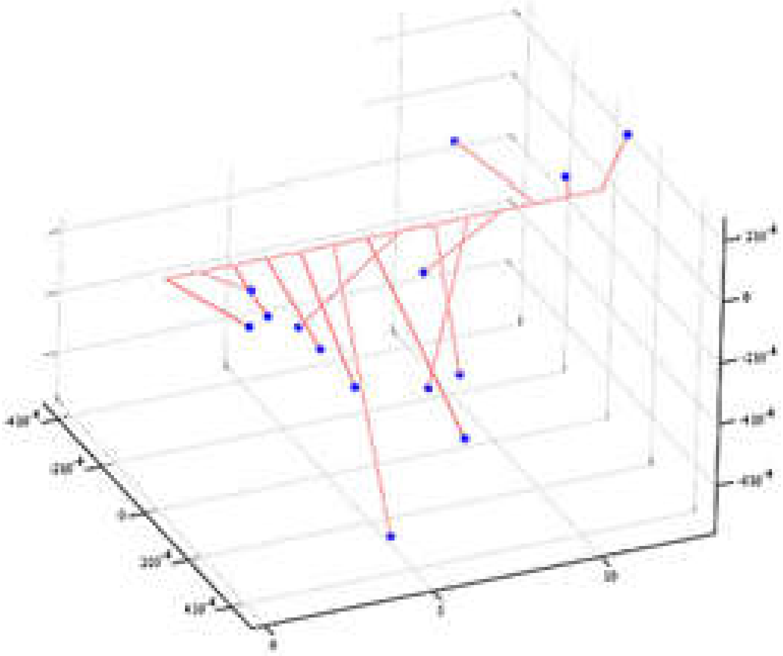


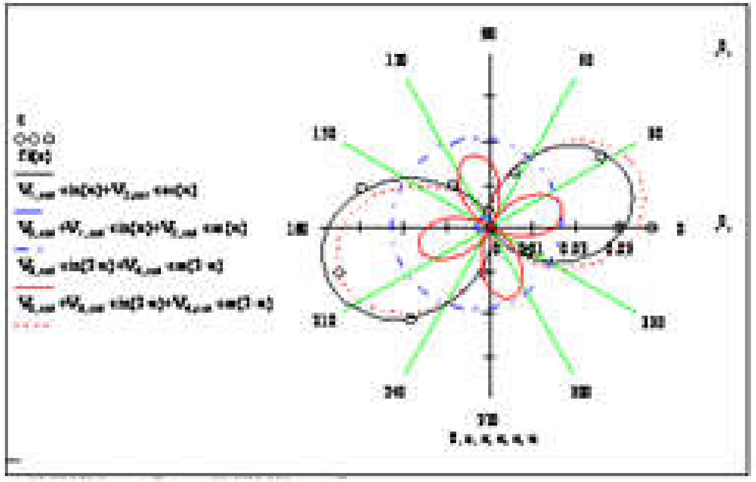
	1	2	3	D	5	6	7	10	11	12
0	0.0025	0.0011	0.0007	0.0004	0.0004	0.0008	0.0009	0.0001	0.0005	0.0371
30	0.0025	0.0015	0.0010	0.0006	0.0004	0.0003	0.0003	0.0003	0.0000	0.0302
60	0.0022	0.0012	0.0005	0.0003	0.0003	0.0000	0.0000	0.0000	0.0002	0.0139
90	0.0016	0.0004	0.0000	0.0000	0.0003	0.0005	0.0006	0.0012	0.0020	0.0039
120	0.0000	0.0000	0.0000	0.0003	0.0006	0.0014	0.0016	0.0014	0.0028	0.0129
150	0.0020	0.0006	0.0007	0.0009	0.0012	0.0021	0.0022	0.0030	0.0033	0.0311
180	0.0044	0.0008	0.0006	0.0006	0.0007	0.0011	0.0010	0.0005	0.0010	0.0363
210	0.0056	0.0015	0.0011	0.0005	0.0003	0.0005	0.0003	0.0004	0.0000	0.0282
240	0.0059	0.0014	0.0009	0.0004	0.0001	0.0002	0.0002	0.0009	0.0017	0.0105
270	0.0048	0.0008	0.0004	0.0001	0.0000	0.0006	0.0007	0.0011	0.0007	0.0000
300	0.0027	0.0003	0.0001	0.0002	0.0002	0.0012	0.0016	0.0012	0.0005	0.0105
330	0.0017	0.0004	0.0002	0.0002	0.0003	0.0013	0.0017	0.0011	0.0013	0.0296
Max	**0.0059**	**0.0015**	**0.0011**	**0.0009**	**0.0012**	**0.0021**	**0.0022**	**0.0030**	**0.0033**	**0.0371**
	Evaluated Eccentricity (one per rev)									
1X Amp	0.0021	0.0002	0.0002	0.0002	0.0003	0.0003	0.0002	0.0005	0.0009	0.0033
Angle	262.1	291.4	250.2	183.8	148.2	173	162.7	174.1	152.1	168.1
	Evaluated Eccentricity (two per rev)									
2X Amp	0.0012	0.0006	0.0004	0.0002	0.0003	0.0008	0.0009	0.0008	0.0009	0.0171
Angle:	58.7	50.5	40.6	16.2	175.8	158.8	155.7	142.5	138.3	15.4
Angle:	238.7	230.5	220.6	196.2	355.8	338.8	335.7	322.5	318.3	195.4

Service Shop Procedure: Runout Evaluation

- **Requirement #2: Mathematical evaluation for 1x (offset) and 2x (ovality) eccentricity**
- Evaluate all eccentricities relative to a common reference line (connecting the journal centers)
- Must identify amplitude and phase angle of net eccentricity at each measurement plane







Service Shop Procedure: Runout Evaluation

- **Requirement #3: Measure and evaluate runout on all coupling faces, rims, and fits**

- Properly square/concentric coupling faces are absolutely essential
- Assure bolt holes a reamed square to coupling faces
- Assure bolt heads and nut seats are square to bore
- **Perpendicular and concentric couplings are critical to achieving proper field alignment**

Service Shop Procedure: Runout Evaluation

- **Requirement #4: Journal TIR evaluation**
- Each journal should be measured in at least 3 planes
- Each journal should be evaluated independently as well for concentricity, taper, ovality, finish roughness, and any diametral deviation

Service Shop Procedure: Runout Evaluation

- **Requirement #5: Collect all TIR data on a single setup on the lathe**
- The only way to ensure that all data is evaluated to a common reference line
- Only way to achieve meaningful runout data for evaluation
- Rotor must remain free, constrained only by gravity at the journals - No coupling can be held/constrained in a chuck on the lathe during measurement

Service Shop Procedure: Runout Evaluation

- **Eccentricity tolerances for couplings and journals:**
  (following ISO 1940-1, or major OEM guidelines)
  - All journal eccentricity must be **< 0.5 mils**
  - Coupling rims and fits **< 0.5 mils**
  - Coupling faces must be perpendicular to **< 1 mil**
- **Coupling and journal eccentricity MUST be brought to tolerances by machining**
  - This will guarantee successful field alignment (by standard method of using 16-point gap/rim readings)

Service Shop Procedure: Rotor Balancing

- Balancing cannot be relied upon as a cure-all
- Eccentricities on journals & couplings cannot be resolved by balancing
- However, any eccentricity on the rotor body between the journals **CAN** be balanced by proper rigid-mode balancing in three planes
- Rotor body 1x eccentricity **over ~2 mils** requires a special balancing procedure to ensure successful operation in the field after assembly

Balancing Significant Rotor Body Eccentricity

- Key goal: **The rotor must be balanced about its geometric axis for all speeds**

- Note: An eccentric/bowed rotor will naturally rotate about its mass axis above its 1st critical speed
- This means a rotor balanced on balancing machines by standard methods of static-couple or influence coefficients will inadvertently be balanced around its mass axis
- BUT, in the field, it will be constrained to its geometric axis
  - The rotor will not be balanced for operation

- This is what often creates vibration problems, when bowed or eccentric rotors are balanced on balancing machines by traditional methods following "industry standards"

Balancing Significant Rotor Body Eccentricity

- Key goal: Restore radial rotor internal mass symmetry relative to the journal axis **FIRST**, at lower speeds, before balancing critical speed responses

- “Rigid mode balancing”
- Full process performed at lower speeds, up to just above the first critical speed
- Because this removes excitation sources at higher speeds above the 1^{st} critical speed, often this procedure alone completes the balancing job
- Saves time and cost, fewer runs, better results in operation

Balancing Significant Rotor Body Eccentricity

- Key Goal: **Must not bend or distort the rotor during "rigid mode" balancing**
  - Must distribute weights across **THREE or more** balancing planes
  - If only 2 planes (endplanes) exist, a third (midplane) must be added
- If not possible to add a central third plane, the eccentricity must be resolved mechanically:
  - Machining the full rotor to throw the centers
  - Thermal straightening

Balancing Significant Rotor Body Eccentricity

Quasi-High Speed Balancing Method

(using 2N+1 balancing planes, where N is the rotor's highest mode in its operating speed range)

- Based on theory from Finite Element Analysis
  - The rotor is conceptually divided into "Rigid Modal Elements"
  - "Rigid" means the largest modal element in the FE model that doesn't bend at any critical speed or within the full operating speed range

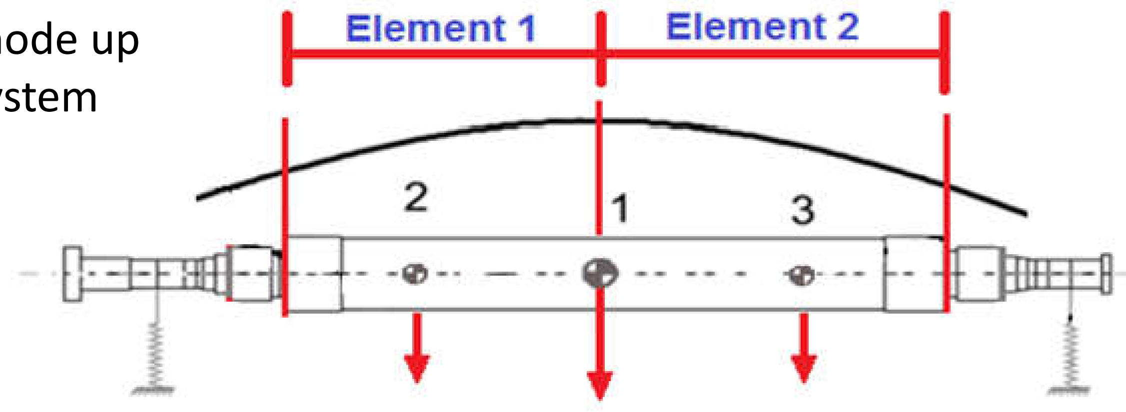
Also based on the principle:

  - *A truly rigid rotor (beam element) can be balanced in any **2** arbitrarily-selected planes*

Balancing Significant Rotor Body Eccentricity

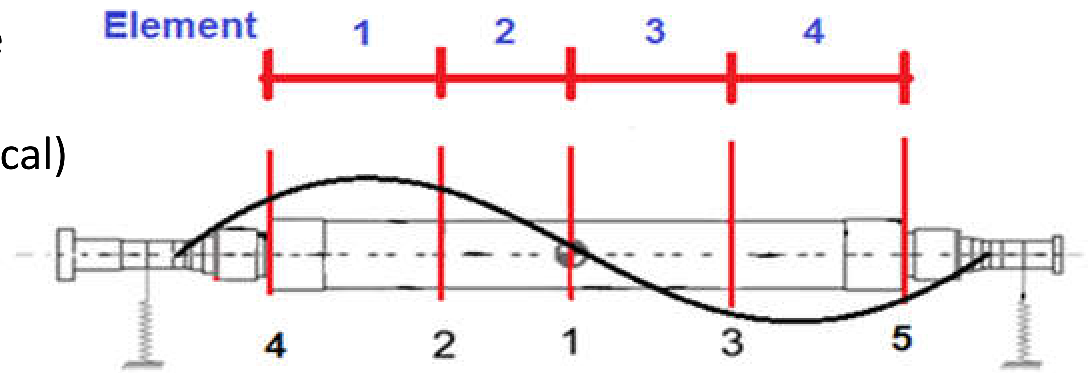
Quasi-High Speed Balancing Method

(rigid mode up to 1st system critical)



"Rigid Element" divisions (use as balancing planes)

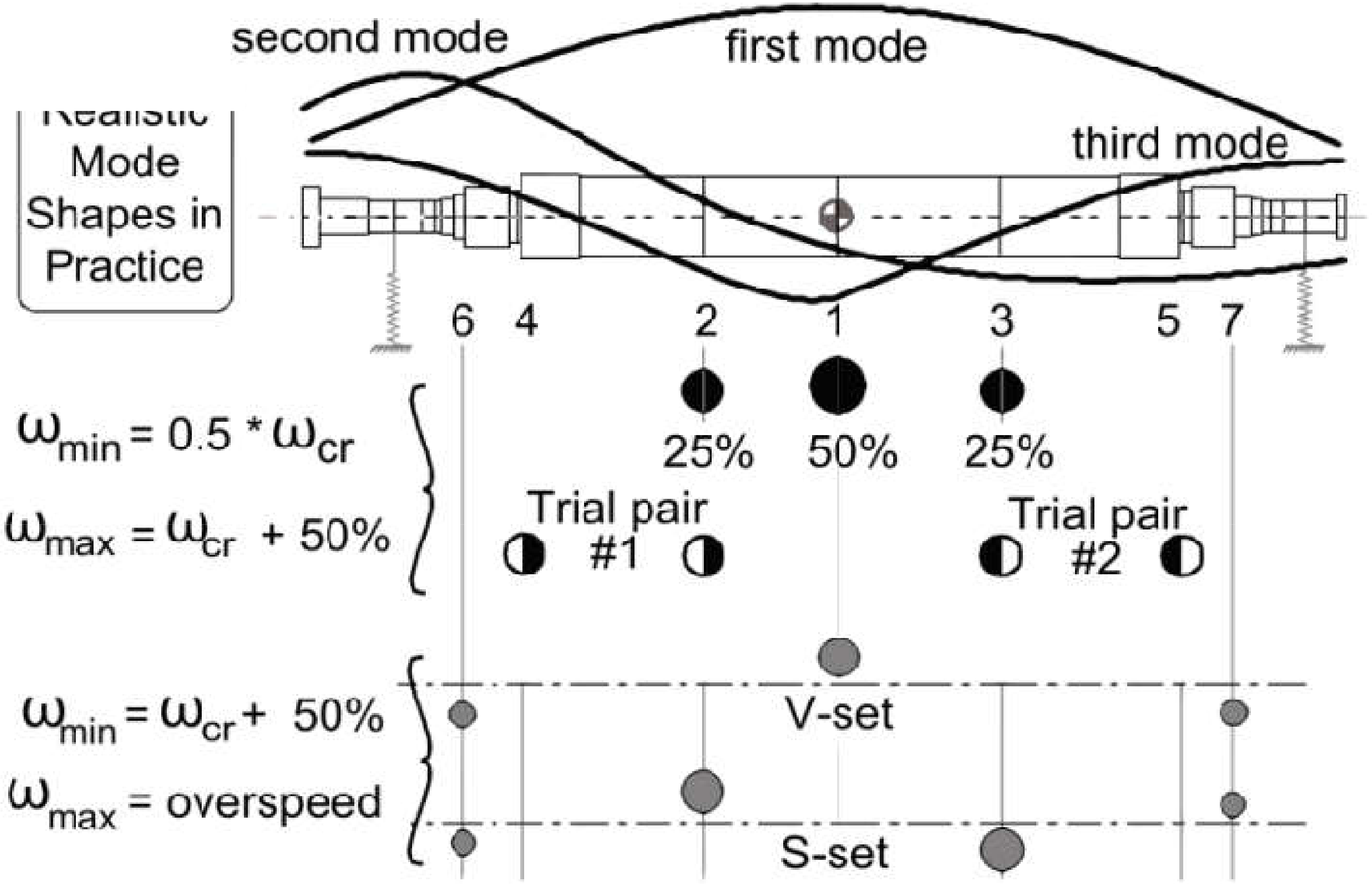
(rigid mode above 1st system critical)



- Axial weight distribution prevents all bending/distortion
- The rotor runs "Dynamically straight"
- The rotor behaves as if it were concentric
- Remains balanced about its geometric axis at all speeds

Balancing Significant Rotor Body Eccentricity

Balancing higher modes:

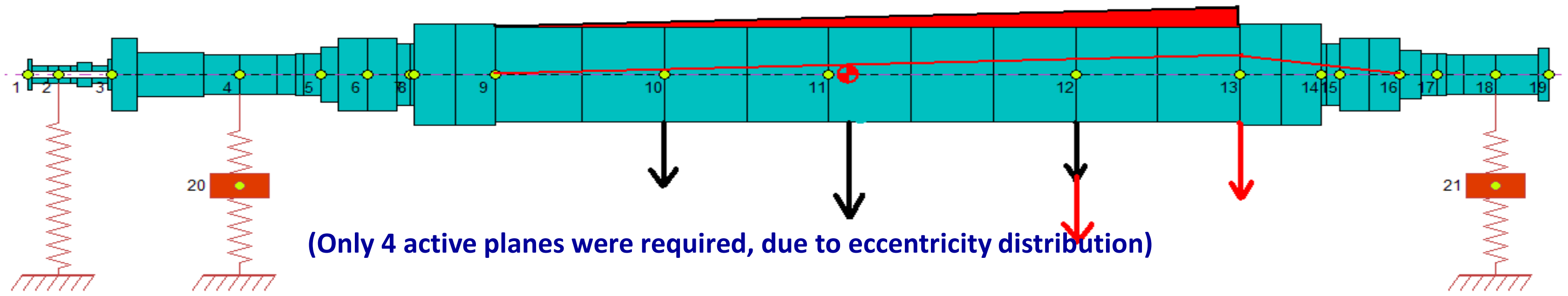


Balancing Significant Rotor Body Eccentricity

- Example Results of 2N+1 Balancing Method

N=3 , 2N+1 = 7 planes

Rotor with body eccentricity

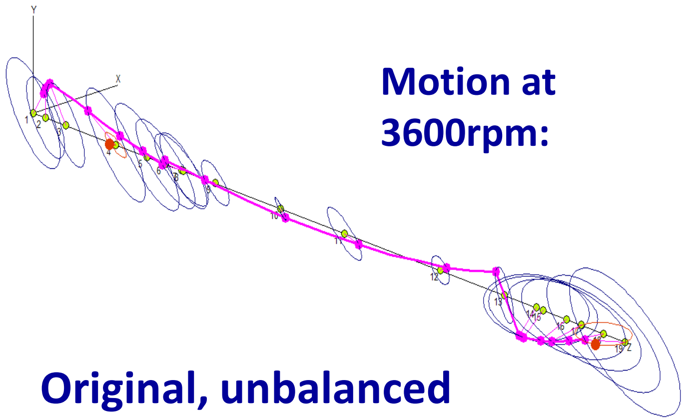


(Only 4 active planes were required, due to eccentricity distribution)

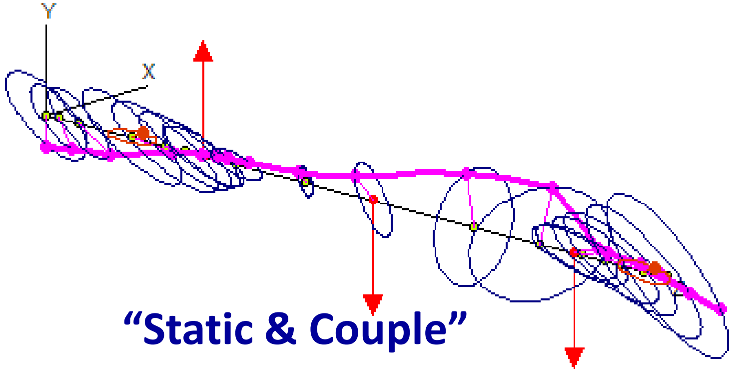
(Arrows represent balancing weight placement)

- Comparison of results to standard balancing method:
  - 2N+1 Method: negligible motion at journals, undistorted, low forces

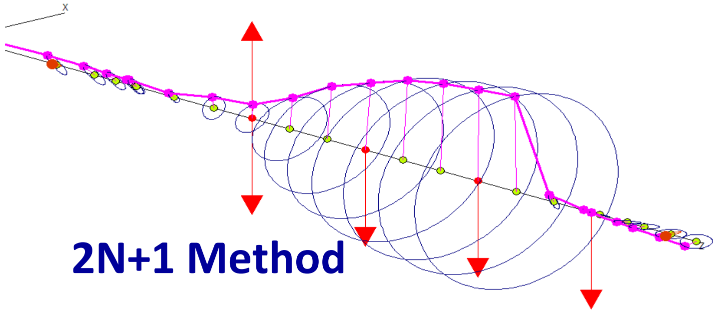
Motion at 3600rpm:



Original, unbalanced



"Static & Couple"



2N+1 Method

Key Takeaways in Balancing Eccentric Rotors

- Mandatory to correct the 1st critical speed response with correction weights placed in three planes simultaneously
- Use 2N+1 balancing planes if TIR is larger than 2 mils or 1x evaluated body eccentricity is > 1 mil
- Resolve rigid mode forces **first**, before any balancing at higher speeds
- Weights should not bend or distort the rotor throughout its full speed range
- Restore symmetry to the rotor about its geometric axis

Field Coupling Alignment Verification

- Evaluation of standard 16-point rim and gap field alignment data during installation
- Bearing and rotor alignment by these measurements is assured ONLY IF the couplings are first verified to be concentric and square to journals
- These readings can be analyzed to distinguish the contribution caused by misaligned bearings versus that from off-square coupling(s)
- Horizontal side to side gap difference must be kept at < 0.002” maximum
- Bearing horizontal moves must always follow both **gap** and **rim** measurement
- Vertical rim offset for purposes of bearing loading for “increased stability” is not a recommended practice

Note: Industry standard forms with data evaluation by averaging the 16-point readings can allow excessive variation in bearing alignment

Field Coupling Alignment Verification

- Evaluation of standard 16-point rim and gap field alignment data during installation

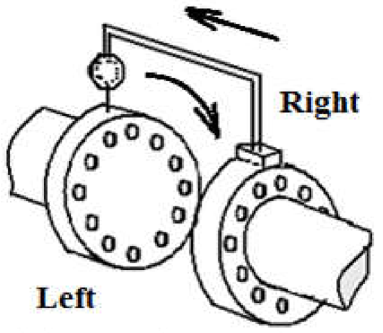
Template by: **Z-R Consulting**

Coupling Face Alignment Evaluation

Plant and Unit

Date:

Indicator reading on: GEN



Indicator mounted on: JS

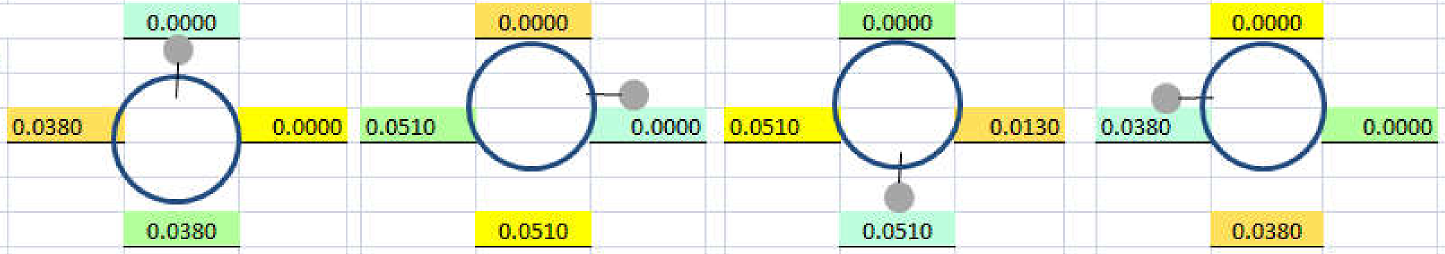
Enter standard 16-point coupling face data in the box below :

Dial Indicator Position:		FACE Top	FACE Left	FACE Bottom	FACE Right
	0	1.4860	1.5240	1.5240	1.4860
	90	1.4730	1.5240	1.5240	1.4730
	180	1.4730	1.5240	1.5240	1.4860
	270	1.4860	1.5240	1.5240	1.4860

minimum per row:
1.4860
1.4730
1.4730
1.4860

Dial Indicator on: *TOP* *RIGHT* *BOTTOM* *LEFT*

GAPS -in mm

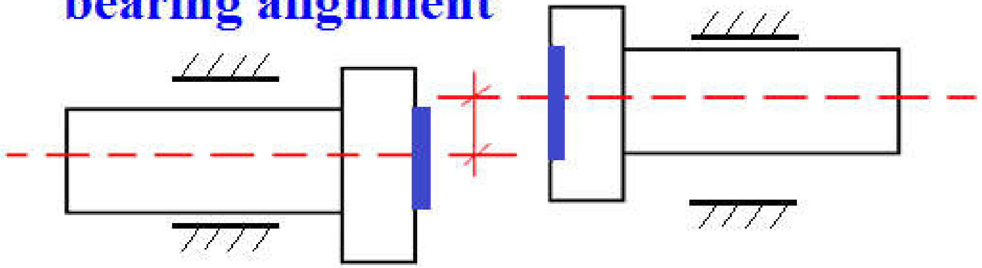


Dial Indicator on	Top gap	Left gap	Right gap	Bottom gap
TOP	0.0000	0.0380	0.0000	0.0380
RIGHT	0.0000	0.0510	0.0000	0.0510
BOTTOM	0.0000	0.0510	0.0130	0.0510
LEFT	0.0000	0.0380	0.0000	0.0380

Face readings: If gap remains on the bottom across all cases, couplings are likely ok, and bearings are not exactly aligned (some preload)
If gap rotates around, then the couplings are off square

Effect of Coupling Eccentricity

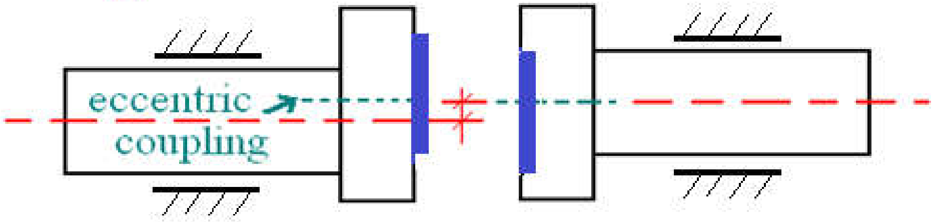
Coupling alignment as horizontal bearing alignment



Large tolerance ok when closing couplings, assuming couplings are good

Only, journals will be mislocated in bearings after closing the couplings

Eccentric coupling will create bad alignment



Coupling eccentricity MUST be limited to <0.5 mil, per ISO 1940. Anything larger will create a "crank" in the rotor train

- Coupling defects create compromised alignment
- ISO 1940 tolerances for coupling/bearing alignment are ~10x higher than eccentricity tolerances
- Many bad rotors get reinstalled because rotor eccentricities can be hidden by liberal alignment tolerances

Summary

For a successful post-outage first restart without the need for field balancing:

- **Two main causes of vibration**:

  1. Misalignment during installation, usually from using off-square couplings that were never evaluated or corrected
  2. Insufficient balancing approach for > 2 mils of distributed mass eccentricity or rotor bow

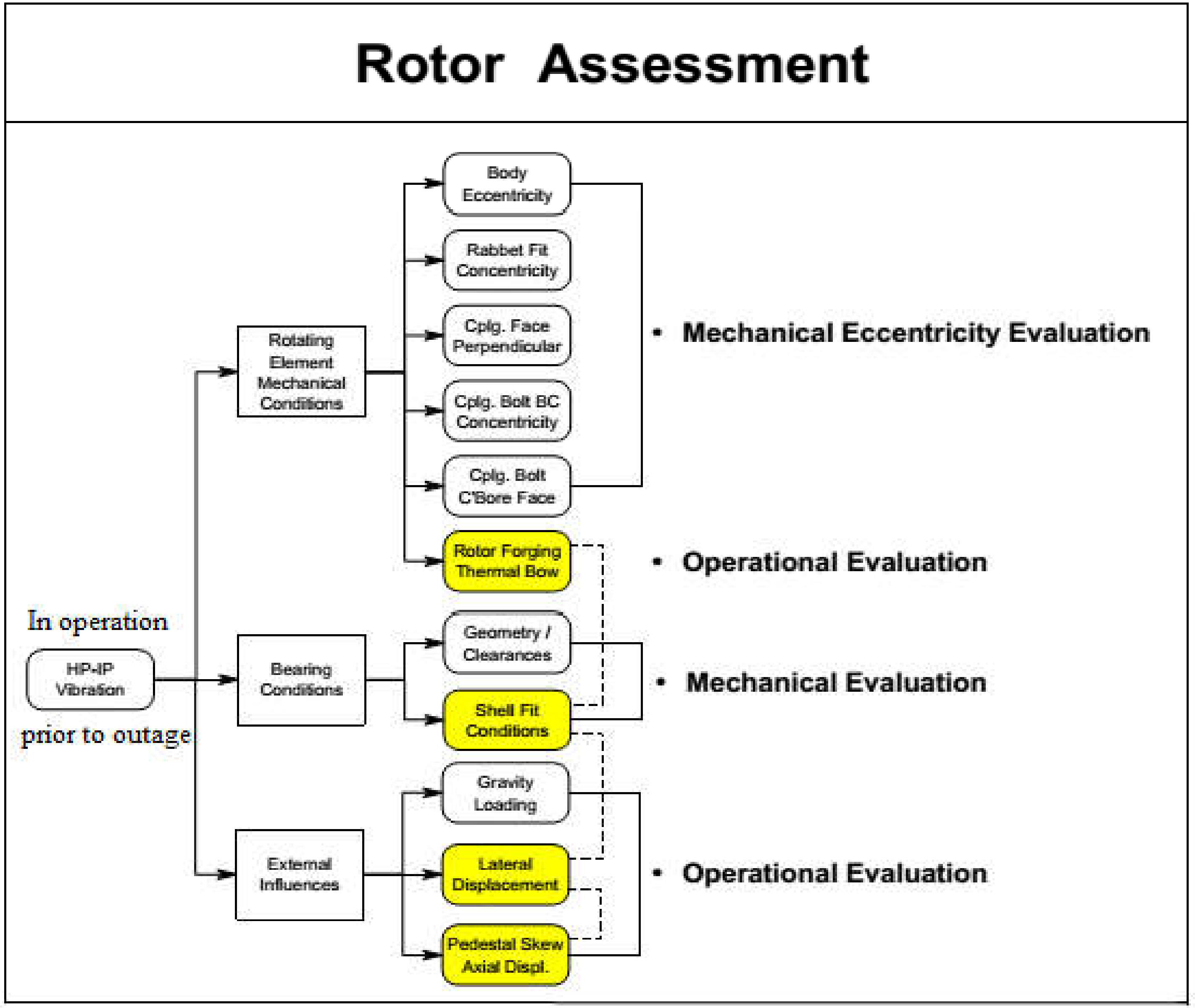
Summary

For a successful post-outage first restart without the need for field balancing:

- **Must incorporate into the outage process:**
  - Leave no unchecked assumptions on rotors "as received" and after any machining and "as left" prior to balancing
  - Measure and evaluate full rotor TIR, including couplings using sound shop practices
  - Bring any coupling/journal to OEM specs by machining
  - Balance rotors with > 2 mils eccentricity using 2N+1 balancing planes (1st critical solution in 3 planes)
  - Assess field coupling alignment data during assembly

- *When all rotor eccentricities are identified and resolved in the service shop, a smooth startup can be guaranteed*

Summary



Rotor Assessment
In operation
HP-IP Vibration
prior to outage
Rotating Element Mechanical Conditions
Bearing Conditions
External Influences
Body Eccentricity
Rabbet Fit Concentricity
Cplg. Face Perpendicular
Cplg. Bolt BC Concentricity
Cplg. Bolt C'Bore Face
Rotor Forging Thermal Bow
Geometry / Clearances
Shell Fit Conditions
Gravity Loading
Lateral Displacement
Pedestal Skew Axial Displ.
• Mechanical Eccentricity Evaluation
• Operational Evaluation
• Mechanical Evaluation
• Operational Evaluation

Case Studies

1. Effects of Misalignment
   - 185 MW Steam turbine-generator
2. Field coupling gap tolerances
   - 240 MW Steam turbine-generator
3. Shop balancing of an “unusable” generator rotor
   - 600 MW generator rotor with “thermal sensitivity”
4. Effects of a bowed IP rotor
   - 800 MW steam turbine-generator
5. A “simple” shop balancing correction
   - 60MW CTG generator rotor

Case Study #1: Effects of Misalignment

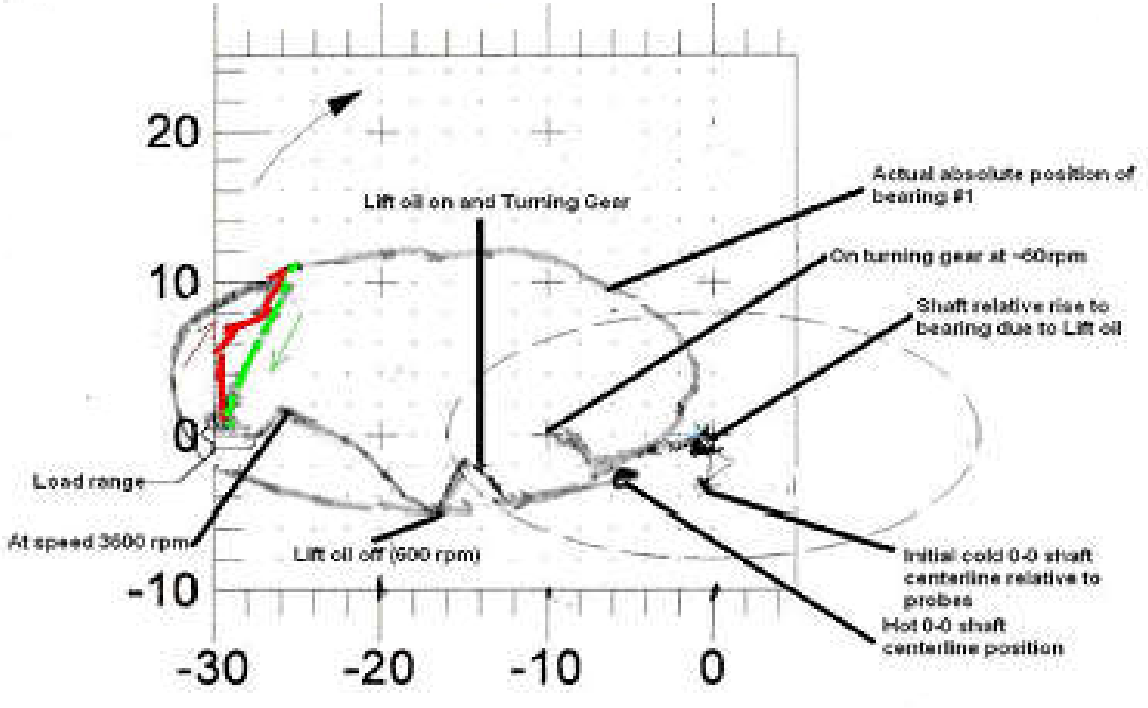
185 MW Steam turbine-generator, 1 year old

- Following major damage from LP turbine blade loss, turbines were overhauled and reinstalled, generator was not touched
- Angular misalignment was found between LP to Gen coupling, which would require a 0.100” shift of Gen EE bearing to bring couplings to tolerance
- Instead, compromise was made by distributing misalignment across all couplings
- Upon restart, HP front bearing wiped at initial loading, impure oil blamed
- Second restart, HP front bearing wiped again

- The shaft centerline plot told the story...

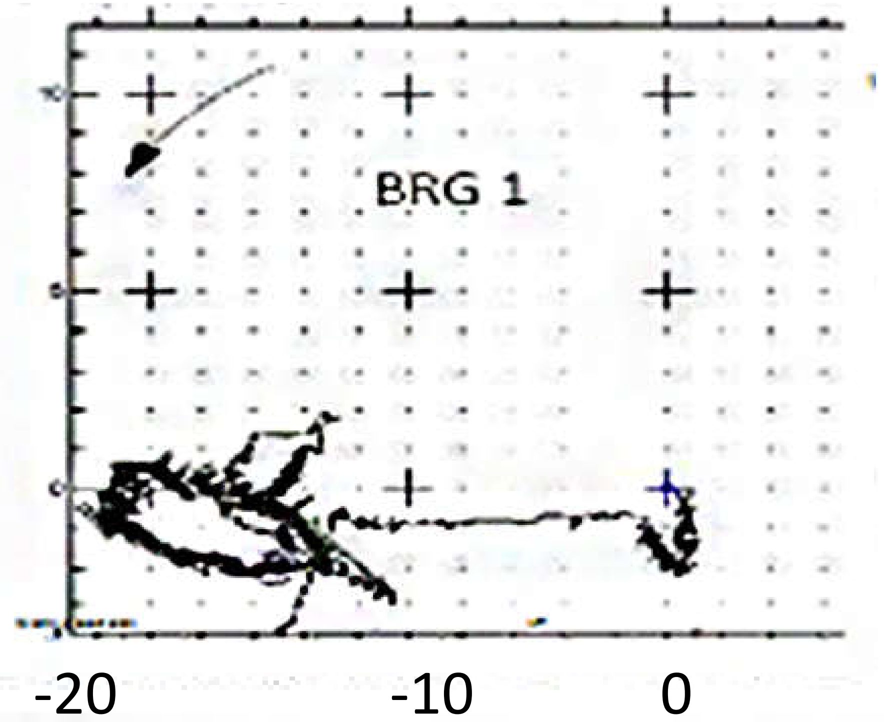
Case Study #1: Effects of Misalignment

- Shaft centerline motion of HP front journal showed 15 mil horizontal move from standstill to 600rpm, plus 15 mils more going to load



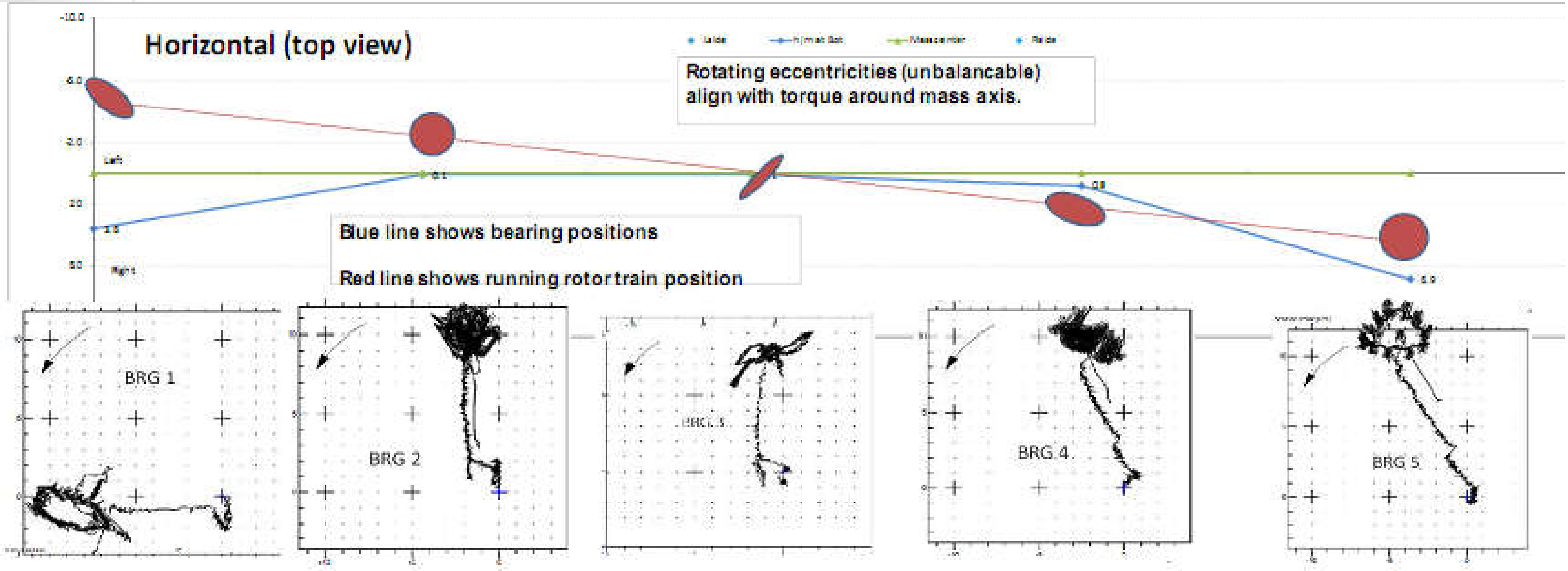
Bearing 1 shaft centerline motion

Superimposed Brg #1 orbit: 2 mils/division



Case Study #1: Effects of Misalignment

- The inertia driven self-straightening of the heavier Generator + LP rotors pushed the lighter HP rotor horizontally until hitting its constraint point at bearing #1



(Shaft Orbits superimposed onto shaft centerline plots)

Case Study #1: Effects of Misalignment

- FE modeling determined the side forces from misalignment were 30,000 lbs on bearing #1 from the HP rotor "spring", plus expected gravity load
- Resulting bearing load exceeded the compressive strength of the babbitt
- Additionally, the bearings had used replacement cheaper babbitt with less load capacity than OEM specs
- The solution:
  - Repair the bearing with stronger, OEM babbitt material
  - Move the HP front bearing 20 mils to the left (the maximum attainable), with recommendation for LP-generator alignment within a year's time
- The unit was operated for 5 years in this state, until the generator developed a ground fault, and full realignment was completed, with no problems since.

Case Study #2: Field Coupling Gap Tolerances

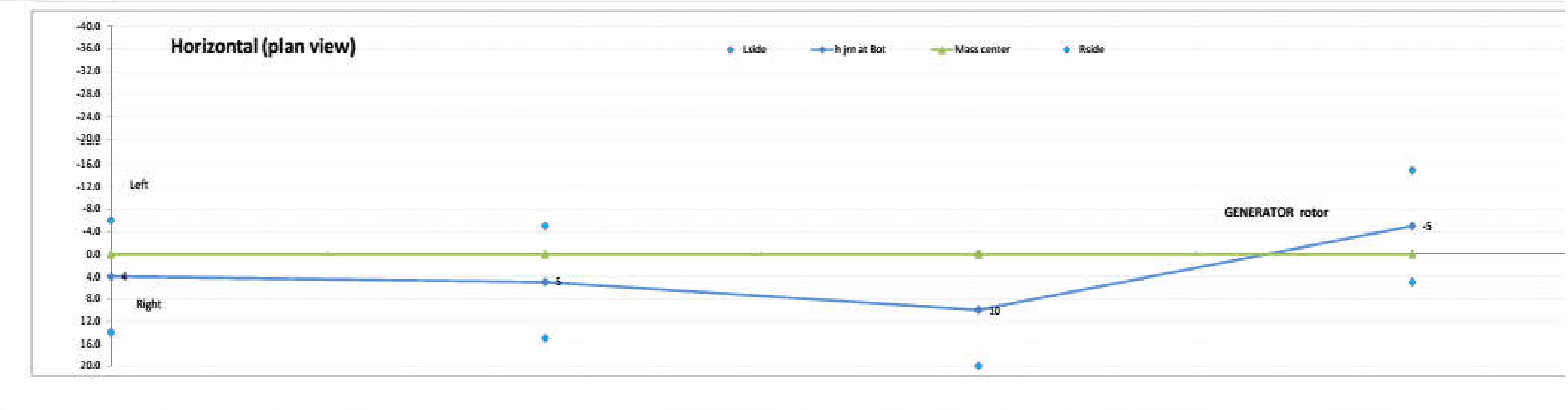
240 MW Steam turbine-generator in combined cycle, FIVE sister units

- Three are installed on high tuned concrete foundations, no vibration problems.
- Two are on steel foundations. Steel platform is supported by series of coil springs mounted over steel columns. These both had vibration issues.

- Both units on steel foundations have a similar problem with appearance of a ~15Hz subsynchronous frequency component at the generator EE bearing.
- The subsynchronous vibration increased with load, increasing to the trip point.
- Steel platform was also vibrating horizontally at ~15HZ
- One unit, besides the subsynchronous vibration, also had a problem with generator EE side high bearing temperature, reaching ~250 F at high load
- This unit was forced to operate at reduced load

Case Study #2: Field Coupling Gap Tolerances

- OEM focus had been on subsynchronous vibration component, and tried several generator bearing modifications without success
- Our analysis, tracking the DC shaft centerline position from standstill gaps, to gear, through the speed range and load range, found horizontal misalignment between the LP and generator rotors

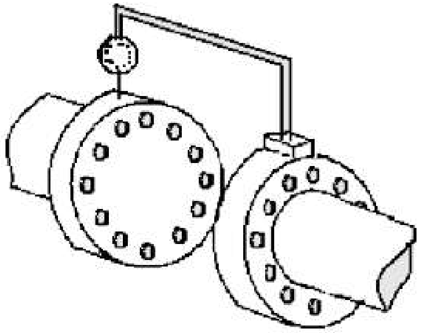


Case Study #2: Field Coupling Gap Tolerances

16-point field alignment data further confirmed misalignment and unresolved off-square coupling faces, despite gaps being within “specs” when averaging the measurements

Indicator reading on: **GEN**

Rim Data (Mils)	
top	0.000"
right	0.000"
bottom	0.000"
left	0.000"



Indicator mounted on: **LP**

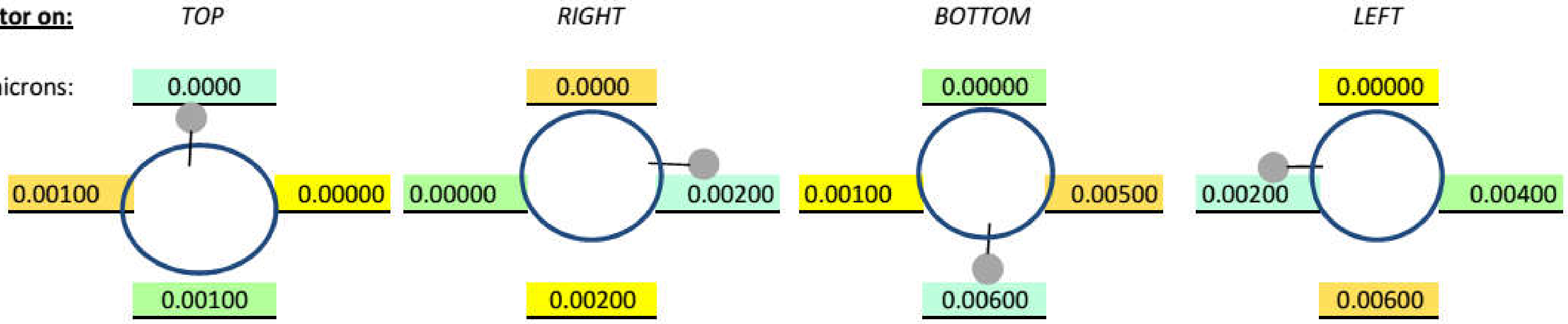
Enter standard 16-point coupling face data in the box below :

Dial Indicator Position:		FACE Top	FACE Right	FACE Bottom	FACE Left
	TOP	0.0280	0.0280	0.0290	0.0290
	RIGHT	0.0280	0.0300	0.0300	0.0280
	BOTTOM	0.0260	0.0310	0.0320	0.0270
	LEFT	0.0270	0.0310	0.0330	0.0290

minimum per row:
0.02800
0.02800
0.02600
0.02700

Dial Indicator on:

Gap in microns:



Face readings: If gap remains on the bottom across all cases, couplings are likely ok, and bearings are not exactly aligned (some preload)
If gap rotates around, then the couplings are off square

Case Study #2: Field Coupling Gap Tolerances

The problem:

- Coupling faces were not evaluated in the shop, assumed ok, and liberal OEM field alignment tolerances allowed horizontal gaps up to 4 mils, leading to misaligned rotors
- With increasing torque (load), the inertial self-centering forces from misalignment became a driving force to excite the rotor's fundamental resonant response at its 1^{st} critical speed of ~15Hz (900rpm)
- This response was possible because the vertical steel springs provided "zero" horizontal dynamic stiffness, so the forces were transferred in a single degree of freedom into horizontal motion through the generator pedestals
- The misalignment also "pushed" and loaded the generator EE journal horizontally into the bearing

The Solution:

- Recommended to correct the generator to LP misalignment to eliminate the driving force
- However, the plant did not want to correct misalignment, and instead continued operating at reduced load, until a short time thereafter, the generator rotor developed a ground fault and had to be replaced, and then was finally realigned

Case Study #3: Shop Balancing of an "unusable" generator rotor

600 MW generator with "thermal sensitivity"

- The rotor was taken out of service as "thermally sensitive", with vibration displacement increasing proportionally to MW load
- The rotor was rewound and shop balanced by the OEM, with no improvement when placed back in service
- Rotor was removed again to check for electrical faults, but none were found
- OEM recommended to discard and replace the rotor

- The plant and a non-OEM service requested another opinion and investigation to diagnose the root cause

Case Study #3: Shop Balancing of an "unusable" generator rotor

- The first step of analysis was to mathematically evaluate the most recent shop TIR data
- The rotor body forging showed 1x eccentricity of ~0.004". It was also revealed that the generator TE side overhang was bowed, and coupling rim was eccentric by ~ 0.004".
- We suspected the rotor's "sensitivity" was actually mechanical in nature, proportional to torque/load, due to driving the bowed rotor and bowed overhang

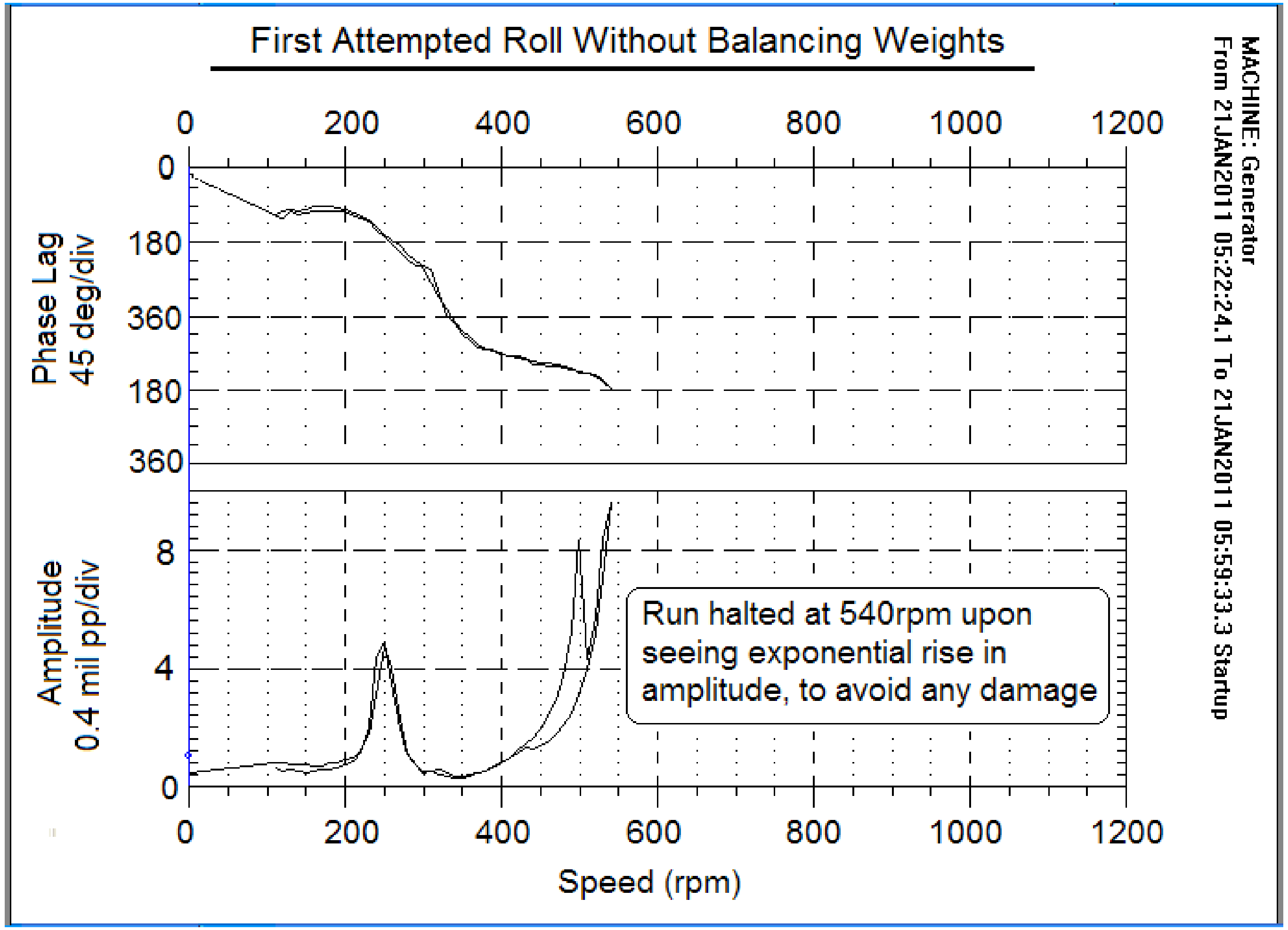
- We recommended:
  - Machine coupling face to less than 0.001" perpendicular to TE side journal
  - Machine a reference band on coupling rim to less than 0.001" TIR to journal
  - Balance the rotor at 1st critical speed using the Quasi-High Speed Balancing method in three simultaneous balancing planes

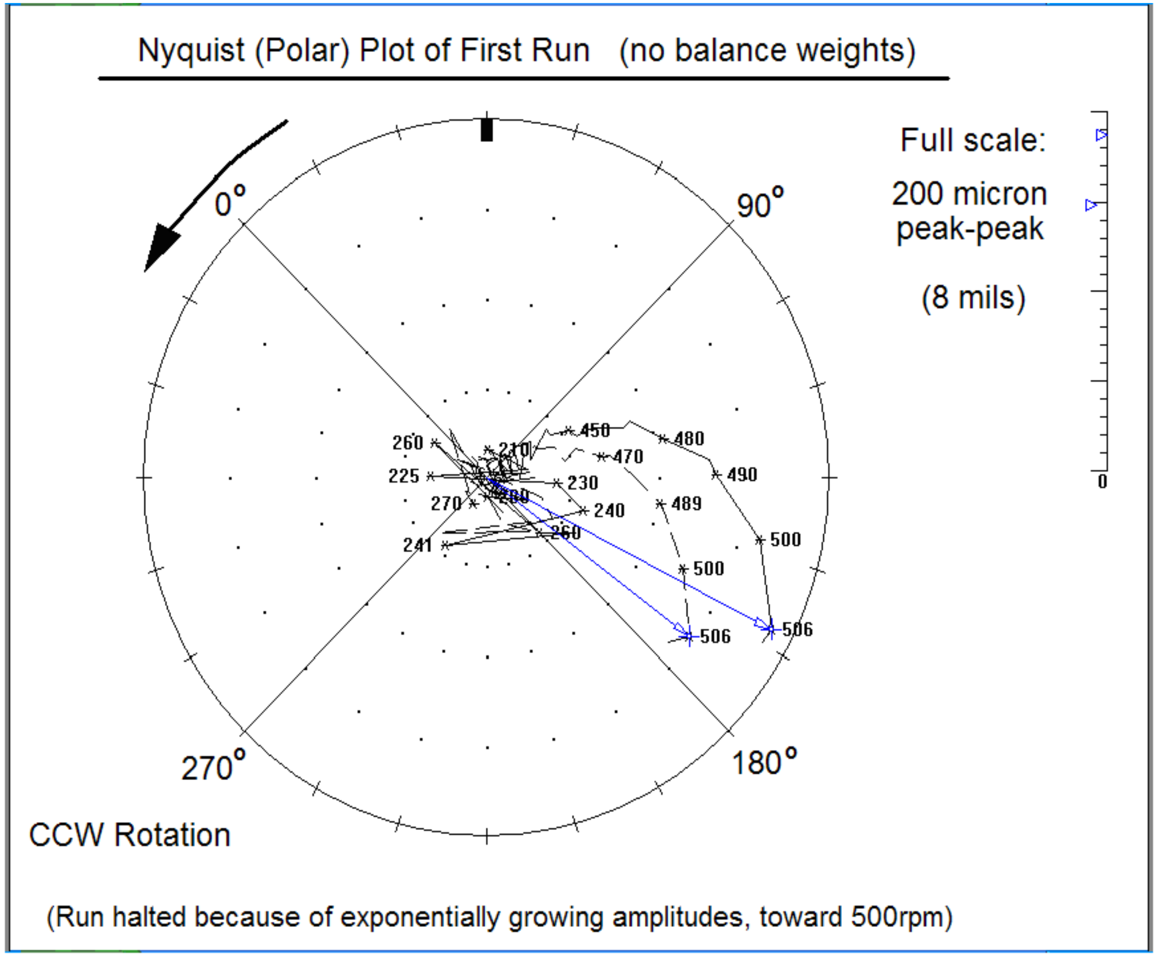
Case Study #3: Shop Balancing of an “unusable” generator rotor

- After machining was completed, initial balancing was first tried by a shop balancing engineer using “industry standard” modal balancing.
- The balancer spent over forty runs without a solution, struggling with compromise between “static” and “couple” balancing
- Either first critical response was high, or running speed vibration was high

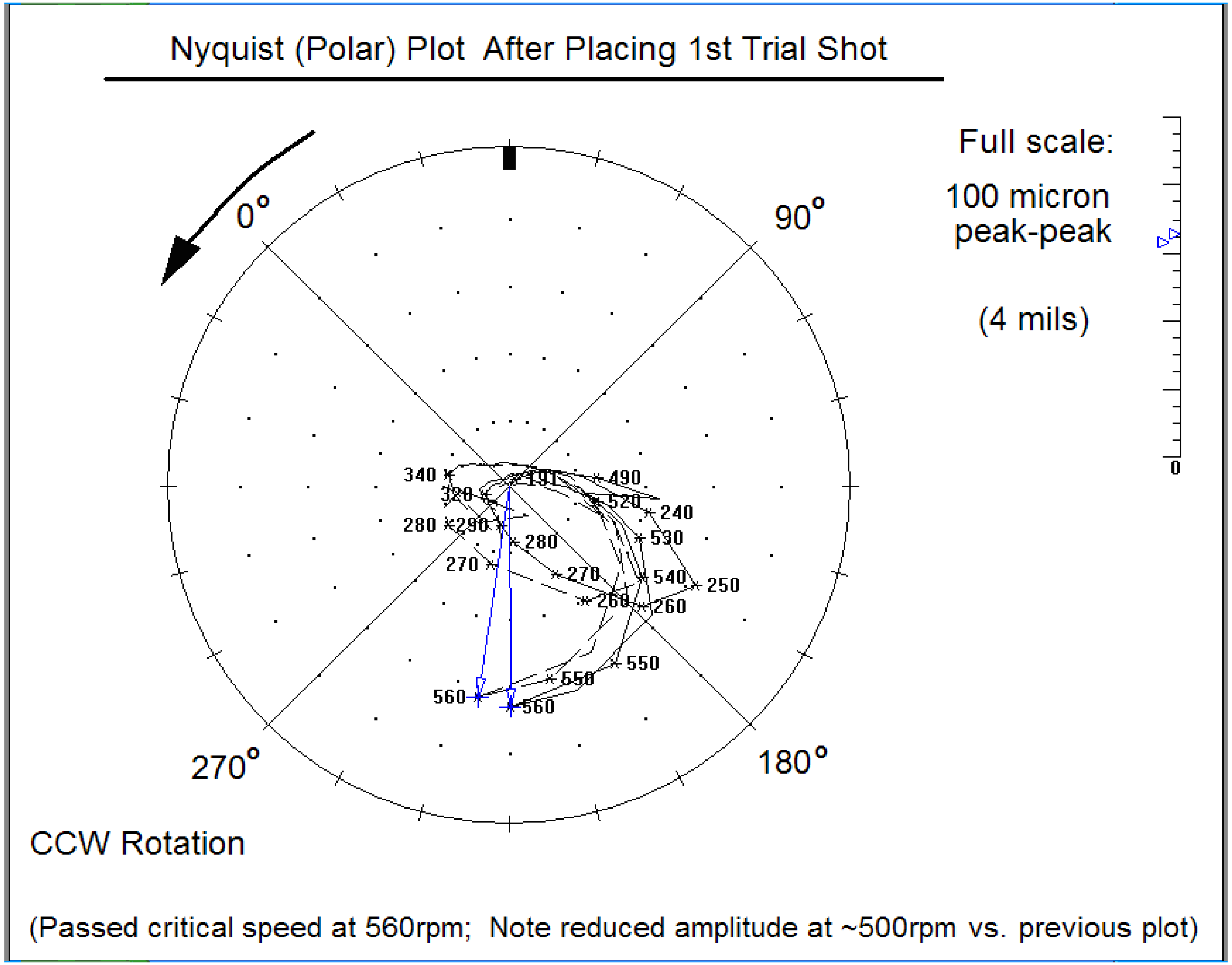
- Balancer requested assistance, and the QHSB method was used
- Solution at 1st critical speed was found by distributing the initial amount of the balance correction weights in three planes; 50% in the mid-plane, and 25% in each ¼ planes, to better axially mirror the eccentricity distribution.
- Rotor balancing at the 1st critical speed, at second critical speed, at operating speed and overspeed, and electrical “heat run” at rated excitation current was completed in **nine runs**.

Case Study #3: Shop Balancing of an "unusable" generator rotor

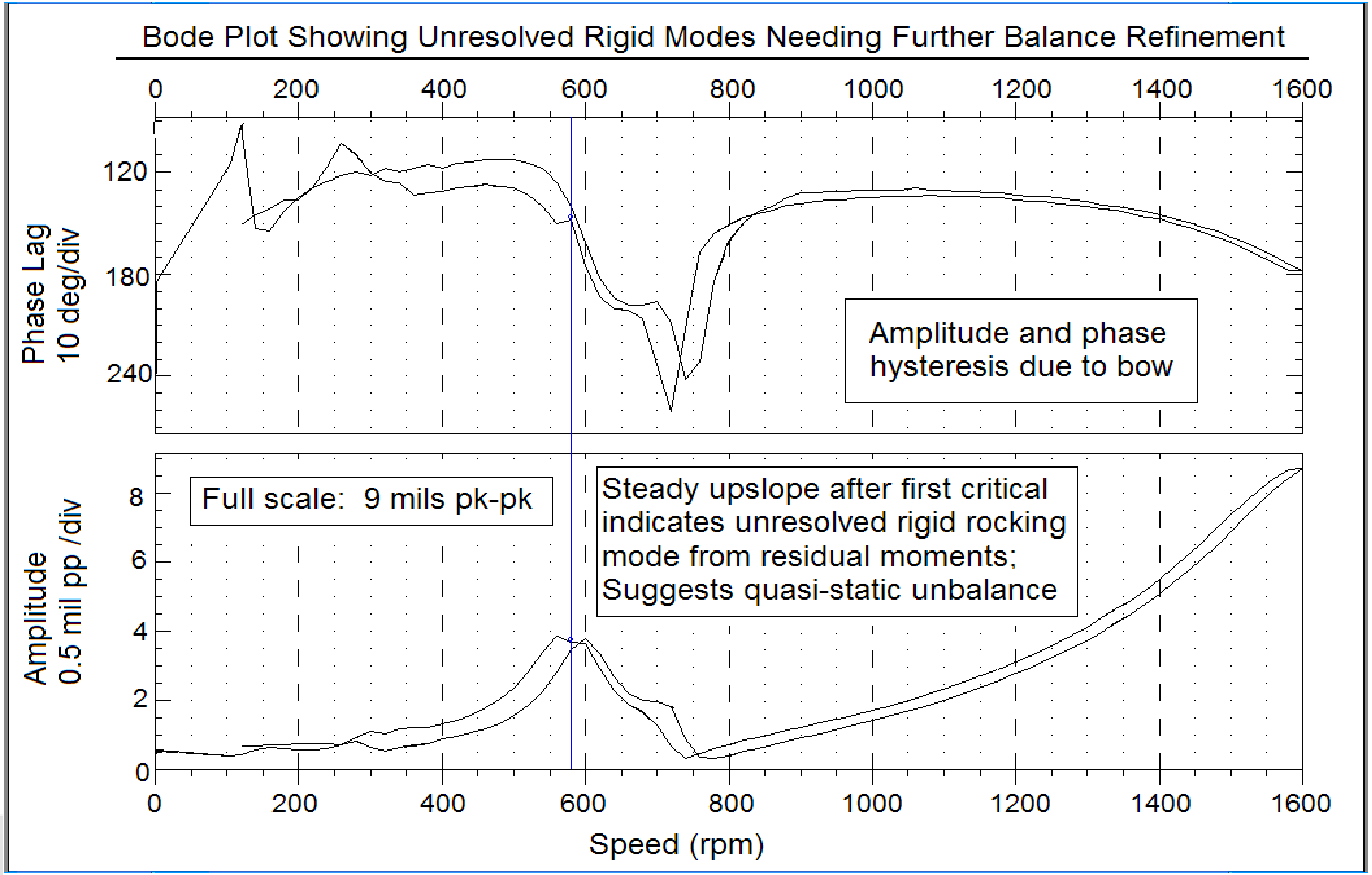




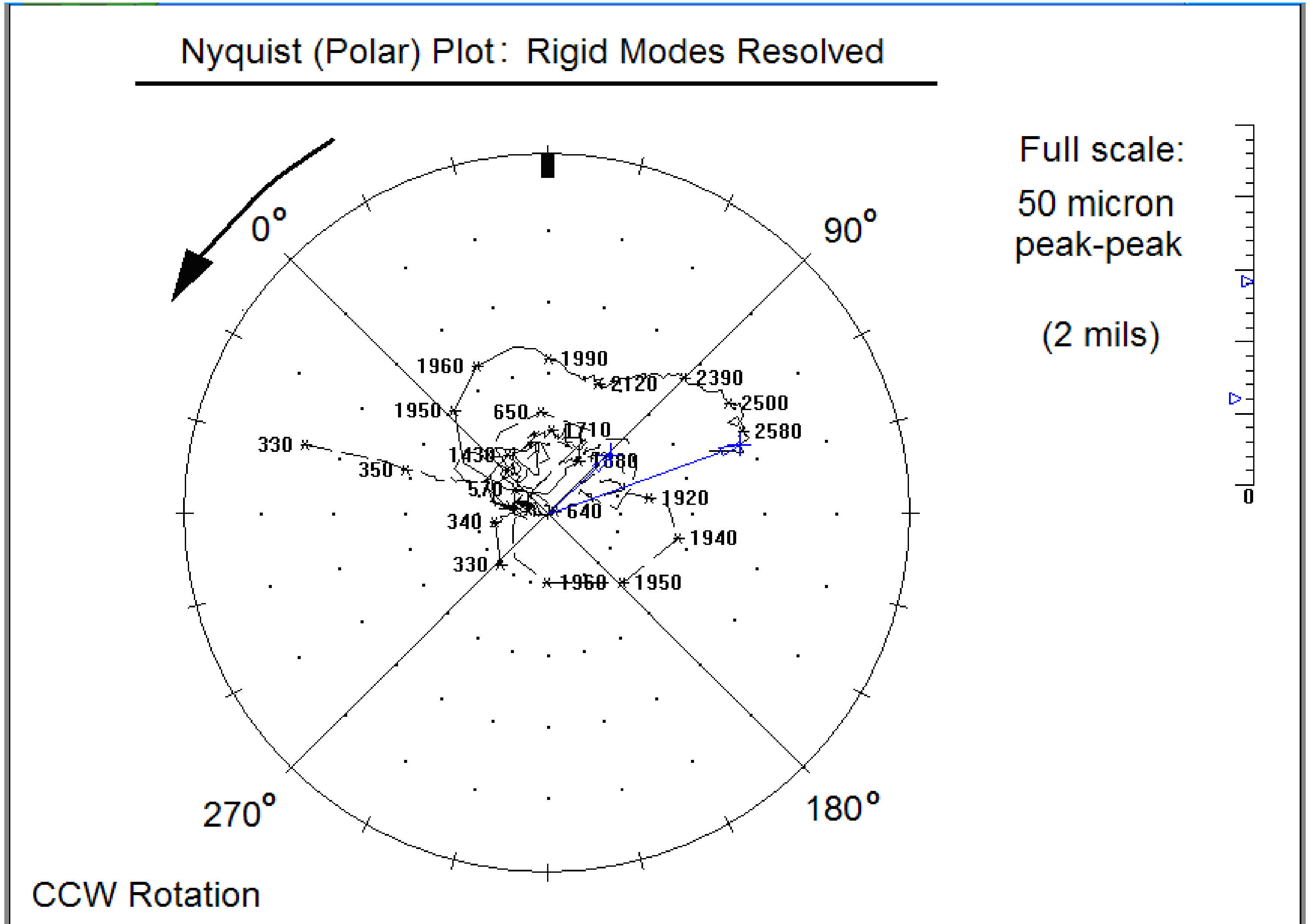
Case Study #3: Shop Balancing of an "unusable" generator rotor



Case Study #3: Shop Balancing of an "unusable" generator rotor

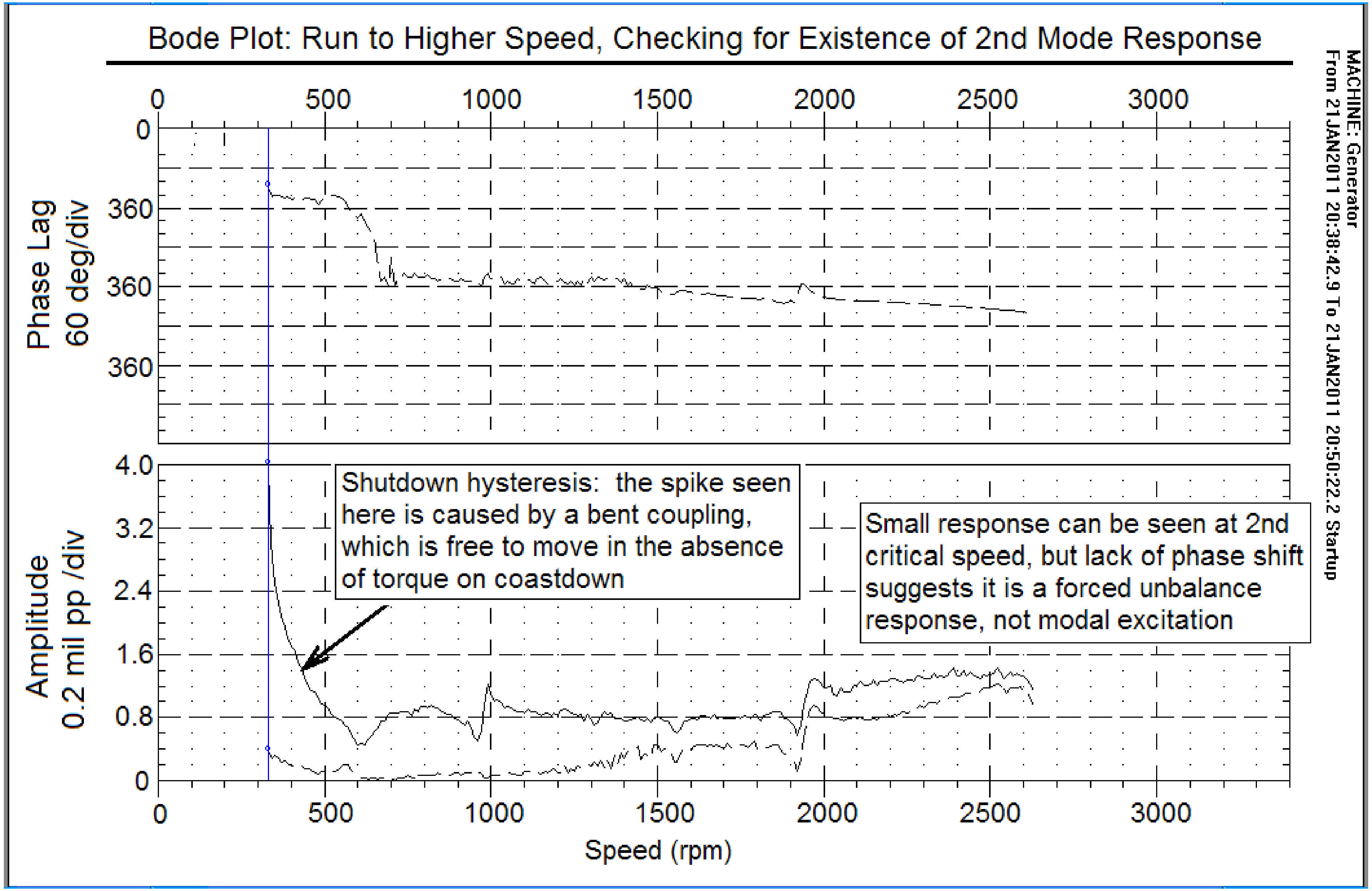


Case Study #3: Shop Balancing of an “unusable” generator rotor

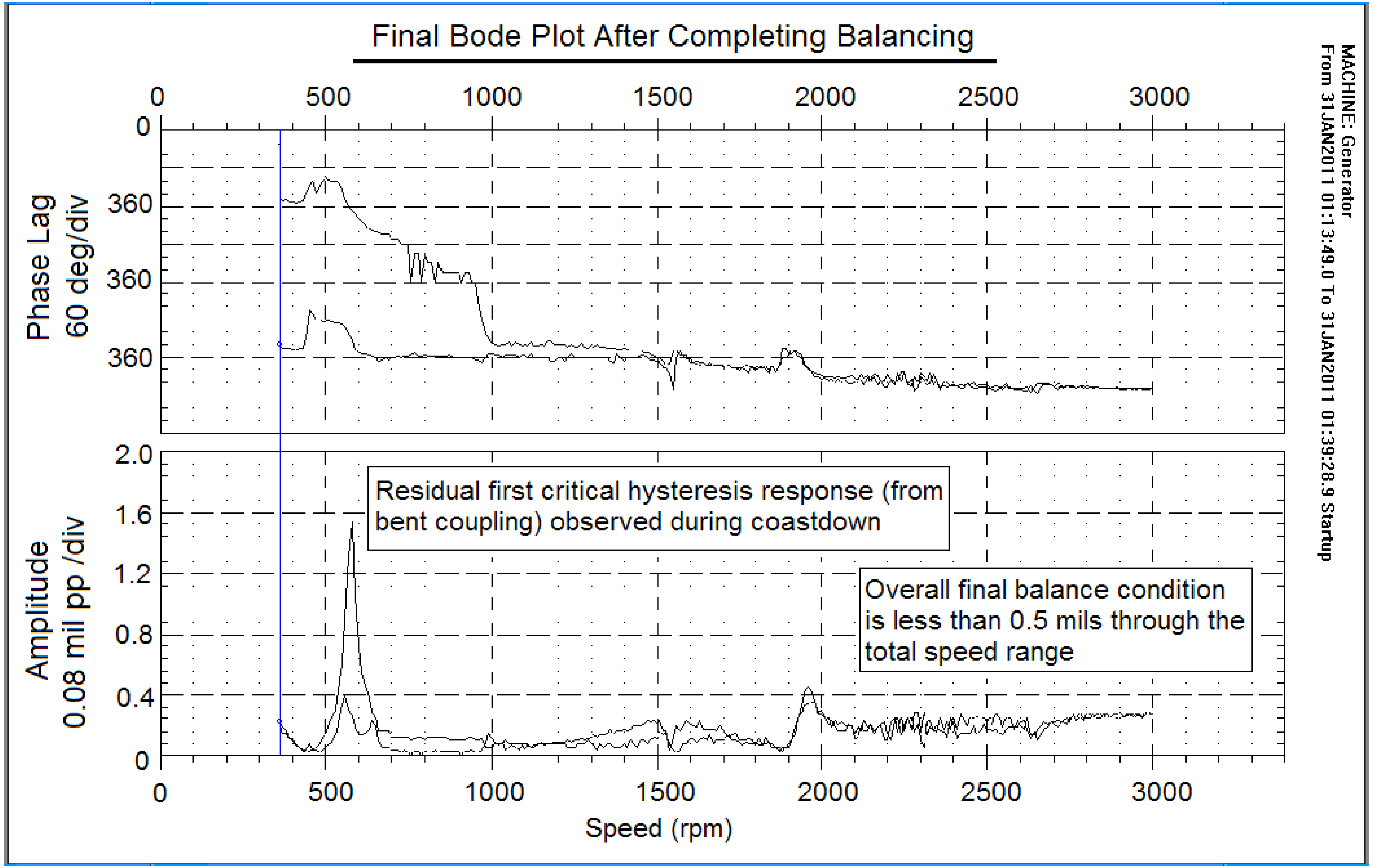


(First critical nonexistent; slight excitation remains around 2nd critical, but fully acceptable)

Case Study #3: Shop Balancing of an "unusable" generator rotor



Case Study #3: Shop Balancing of an "unusable" generator rotor



Case Study #3: Shop Balancing of an “unusable” generator rotor

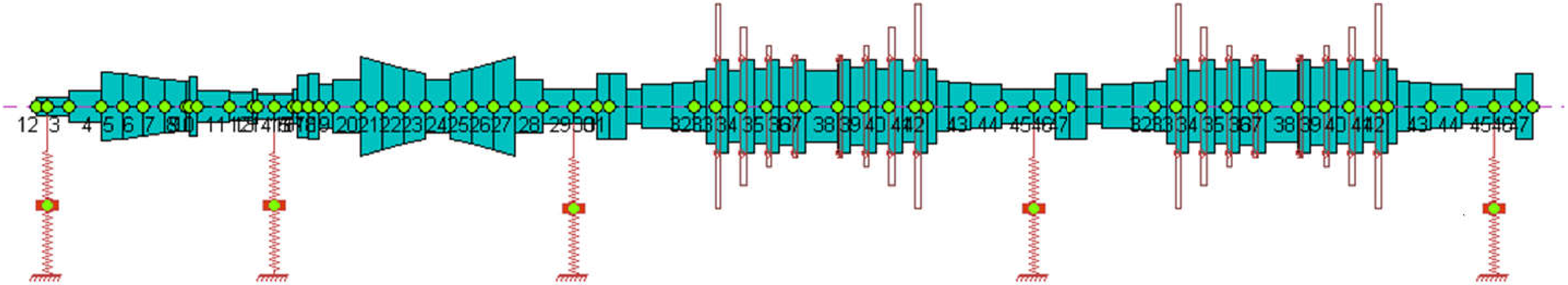
- Client requested also to perform heat run in increments of 400 amperes to rated excitation current of 2000 amperes. Rotor vibration displacement increased at each increment of the excitation current.
- Since jumps were almost instantaneous with change in current, not proportional to heating rate, it was concluded that change in vibration was of mechanical nature, proportional to the angular momentum change from the increased drive torque driving the bowed and unsupported coupling overhang rotating unconstrained on the balancing machine.

- The rotor was accepted and reinstalled.
- Alignment between LP and generator rotors was done utilizing reverse dial indicator method necessary to compensate for two pole rotor inherent second harmonic and residual bow of unsupported coupling overhang
- Turbine-generator was restarted and tested to full load without showing any previously observed “thermal sensitivity”.

Case Study #4: Effects of a Bowed IP Rotor

800 MW Steam turbine-generator (HP, IP and 2 LP turbines)

- After turbine-generator up-rating by nearly 100MW, vibrations at HP #1 bearing journal were increasing up to 0.012”(p-p) in operation proportional to load.
- Client attempted to reduce vibration by balancing the HP rotor, and had tried several contractors, but without any visible success.



Case Study #4: Effects of a Bowed IP Rotor

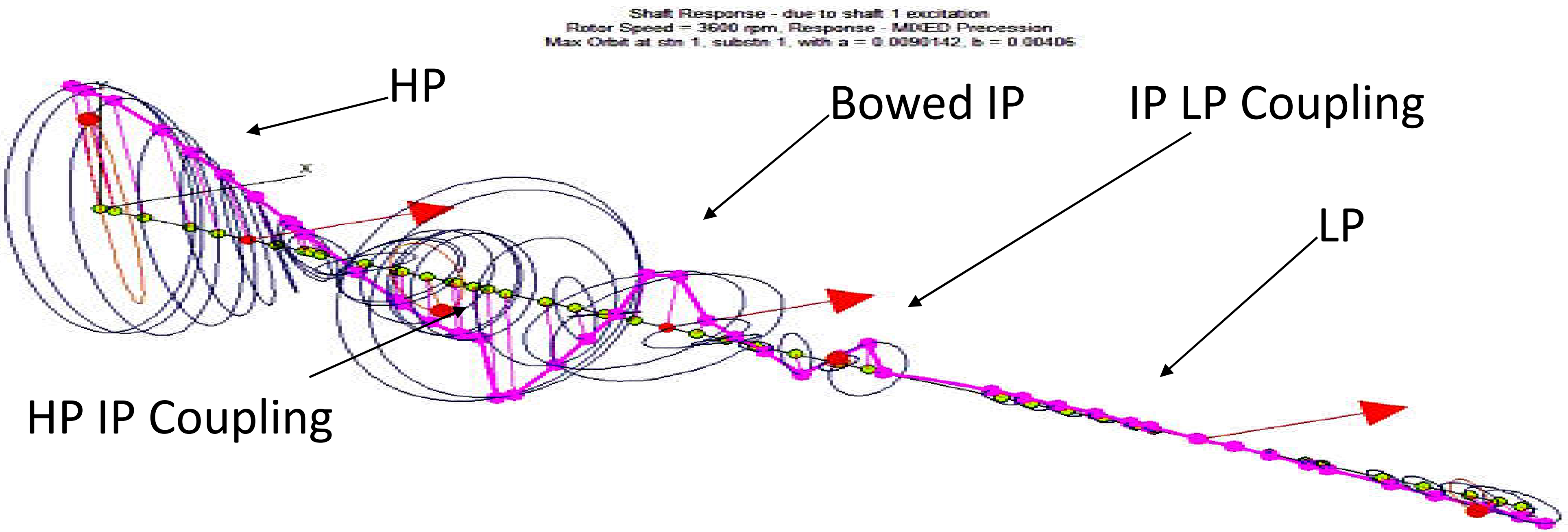
800MW Steam turbine-generator (HP, IP and 2 LP turbines)

- Z-R Consulting was called to assist in finding the root cause of vibration. During a start up for testing, DC and AC vibration data from proximity probes was acquired from rotors at standstill, at slow roll and to full speed and load.

- The analysis of SCL data plots suggested that the IP rotor is bowed ~ 0.004”.
- That affected coupling faces to be non-perpendicular to respective journals.
- That caused angular misalignment between the HP and IP mass axis, which induced eccentricity in the HP rotor relative to the overall rotor train mass axis.

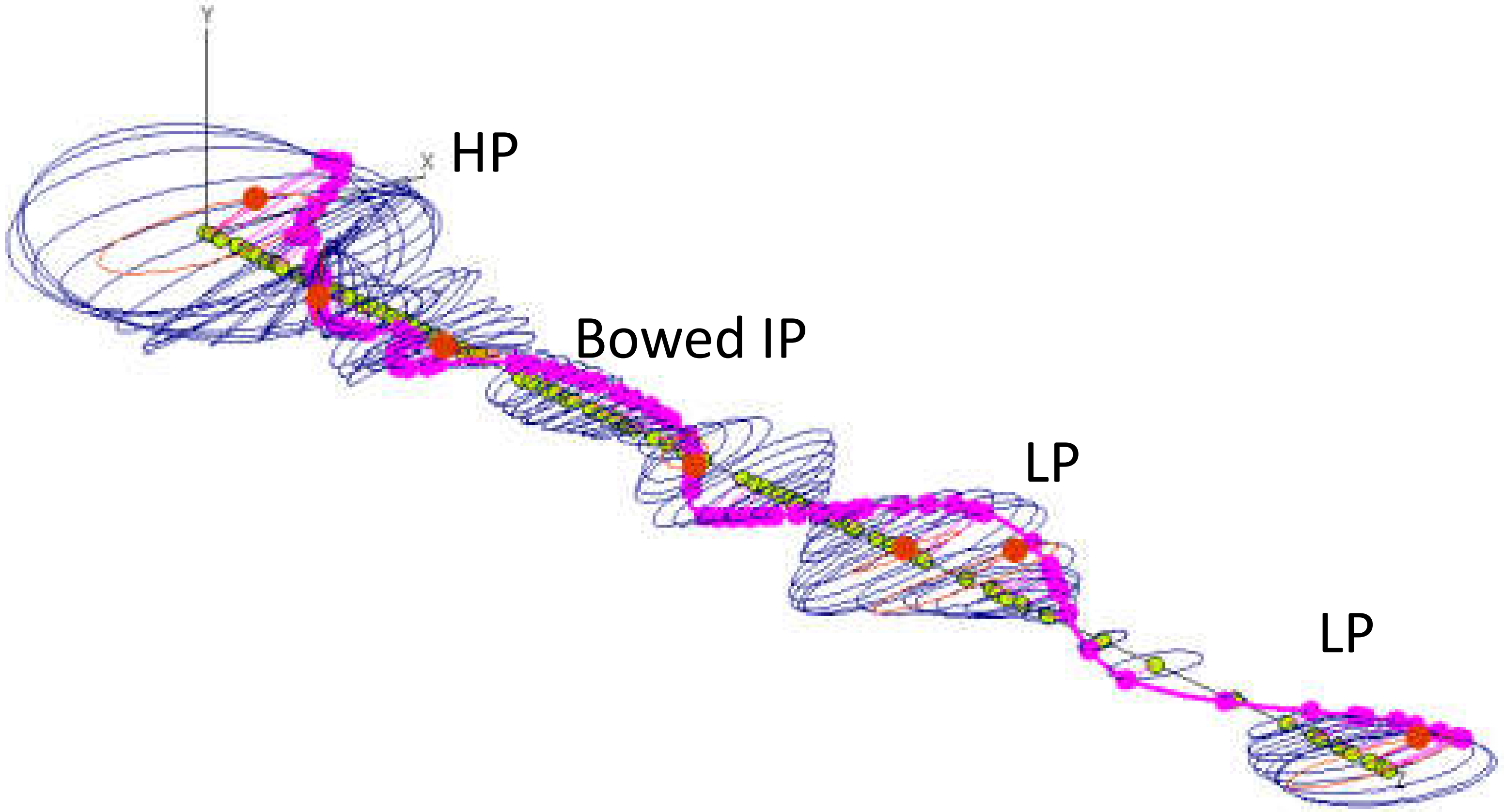
Case Study #4: Effects of a Bowed IP Rotor

- The mass axis of the rotor with the largest inertia self-centers, and all other rotor mass axes tend to self-align to this common centroidal axis. The lighter rotor (HP turbine) with eccentric masses relative to the common centroidal axis then whirls synchronously within bearing clearances.
- The bowed, shop-balanced IP produces high motion on the adjacent, perfectly balanced HP rotor due to coupling eccentricity and out of perpendicularity.



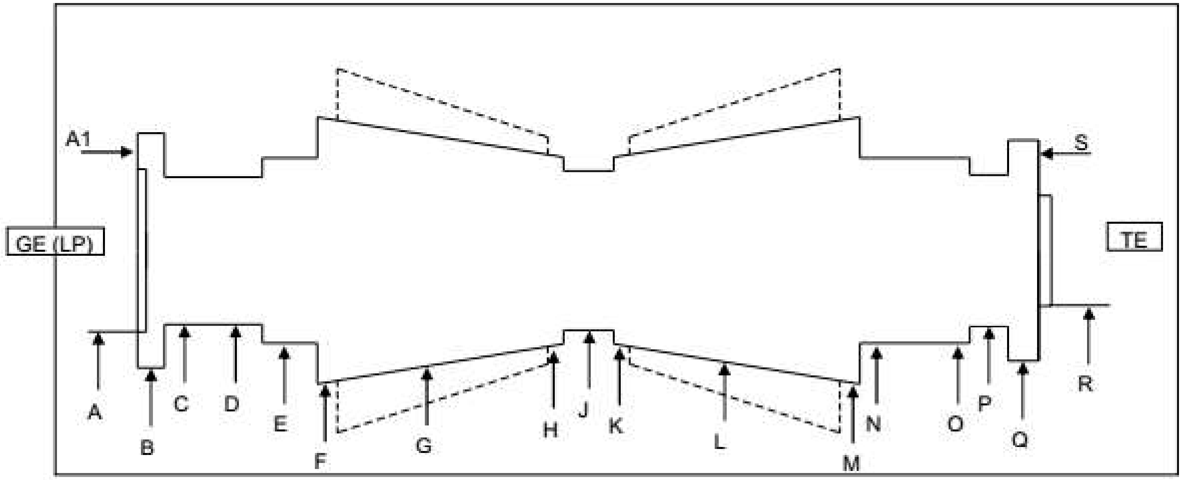
Case Study #4: Effects of a Bowed IP Rotor

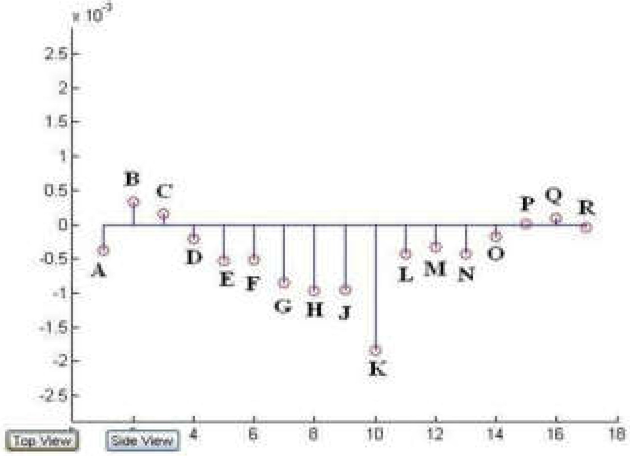
- As long as whirling is not constrained, sensors will indicate large displacement from kinetic energy, but relatively low seismic vibrations.
- The unit was allowed to continue operating in this condition for over a year, until a planned outage scheduled for removal of the IP rotor for machining correction and rebalancing.



Case Study #4: Effects of a Bowed IP Rotor

- TIR evaluation of the IP rotor showed up to ~0.002” 1x eccentricity (~4 mil TIR) on the rotor body, plus ~3 mils on the HP coupling face – this skewed the HP rotor in operation





	LP Coupling																HP Coupling		
	Face	Fit	Rim	Journal						Inlet				Seals		Dummy Journal	Rim	Fit	Face
	A1	A	B	C	D	E	F	G	H	J	K	L	M	N	0	P	Q	R	S2
0	0.0009	0.0011	0.0001	0.0000	0.0004	0.0018	0.0020	0.0010	0.0020	0.0020	0.0026	0.0030	0.0020	0.0010	0.0011	0.0003	0.0001	0.0008	0.0022
45	0.0010	0.0009	0.0002	0.0001	0.0003	0.0010	0.0015	0.0005	0.0010	0.0007	0.0005	0.0020	0.0018	0.0004	0.0006	0.0000	0.0000	0.0007	0.0046
90	0.0000	0.0000	0.0009	0.0004	0.0001	0.0003	0.0005	0.0000	0.0000	0.0000	0.0001	0.0010	0.0012	0.0003	0.0002	0.0001	0.0004	0.0012	0.0051
135	0.0002	0.0002	0.0010	0.0005	0.0000	0.0000	0.0000	0.0000	0.0000	0.0009	0.0000	0.0000	0.0006	0.0000	0.0000	0.0004	0.0004	0.0000	0.0040
180	0.0009	0.0004	0.0007	0.0004	0.0002	0.0003	0.0000	0.0005	0.0015	0.0015	0.0011	0.0010	0.0000	0.0004	0.0004	0.0003	0.0005	0.0015	0.0030
225	0.0009	0.0007	0.0005	0.0003	0.0005	0.0010	0.0010	0.0010	0.0017	0.0015	0.0030	0.0010	0.0010	0.0007	0.0005	0.0002	0.0003	0.0006	0.0008
270	0.0014	0.0010	0.0001	0.0000	0.0005	0.0013	0.0015	0.0020	0.0021	0.0023	0.0035	0.0020	0.0020	0.0010	0.0007	0.0000	0.0000	0.0008	0.0000
315	0.0012	0.0011	0.0000	0.0000	0.0004	0.0016	0.0020	0.0015	0.0018	0.0022	0.0031	0.0020	0.0021	0.0011	0.0004	0.0003	0.0001	0.0009	0.0005
0	0.0009	0.0011	0.0001	0.0000	0.0004	0.0018	0.0020	0.0010	0.0020	0.0020	0.0026	0.0030	0.0020	0.0010	0.0011	0.0003	0.0001	0.0008	0.0022
Max	**0.0014**	**0.0011**	**0.0010**	**0.0005**	**0.0005**	**0.0018**	**0.0020**	**0.0020**	**0.0021**	**0.0023**	**0.0035**	**0.0030**	**0.0021**	**0.0011**	**0.0011**	**0.0004**	**0.0005**	**0.0015**	**0.0051**
Evaluated Eccentricity																			
1X	**0.0005**	**0.0005**	**0.0005**	**0.0003**	**0.0002**	**0.0009**	**0.0011**	**0.0009**	**0.0010**	**0.0010**	**0.0019**	**0.0011**	**0.0010**	**0.0005**	**0.0003**	**0.0000**	**0.0002**	**0.0000**	**0.0026**
Phase	288	314	138	145	293	322	331	290	289	284	286	338	341	306	329	171	154	254	94

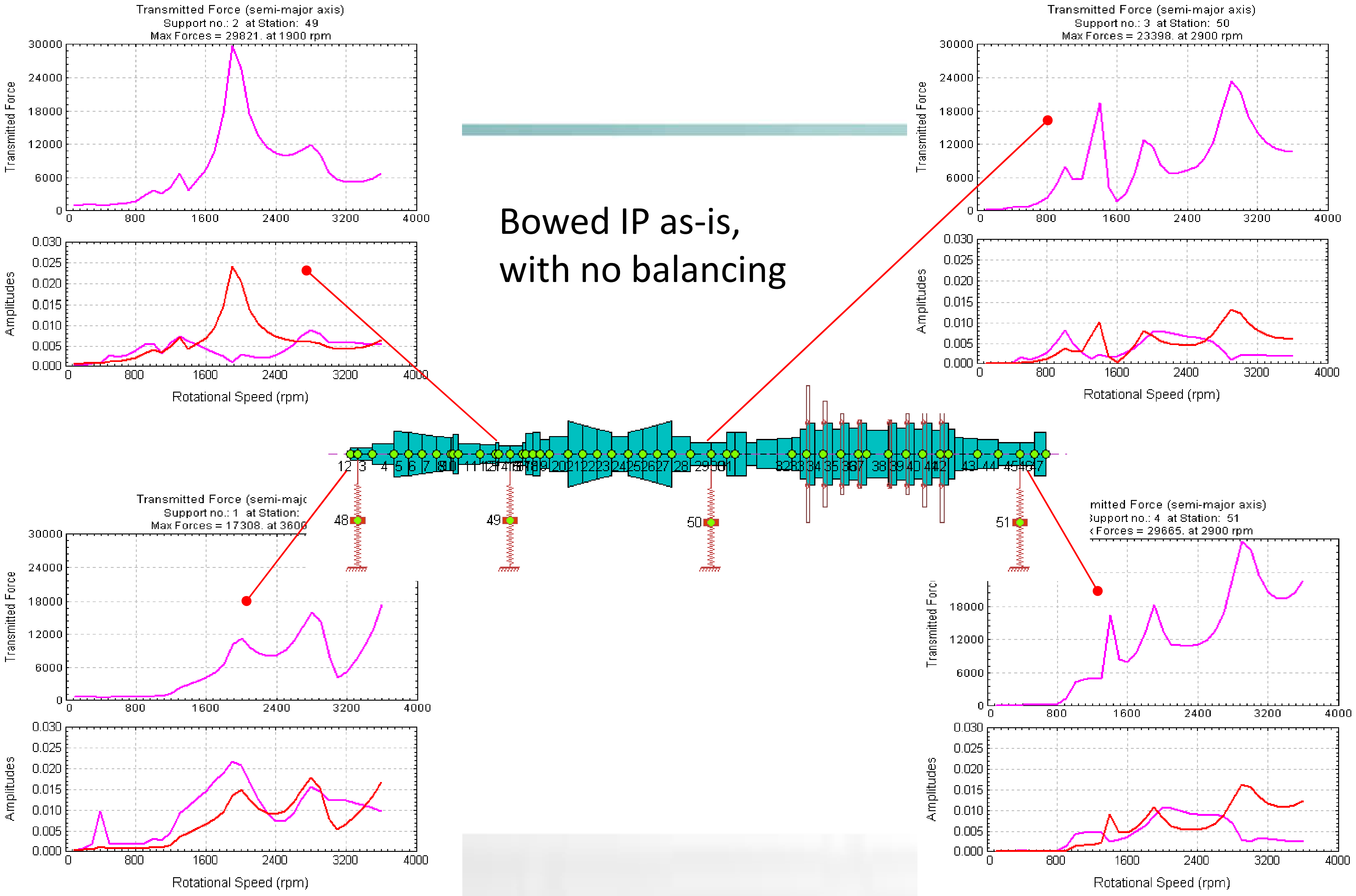
Case Study #4: Effects of a Bowed IP Rotor

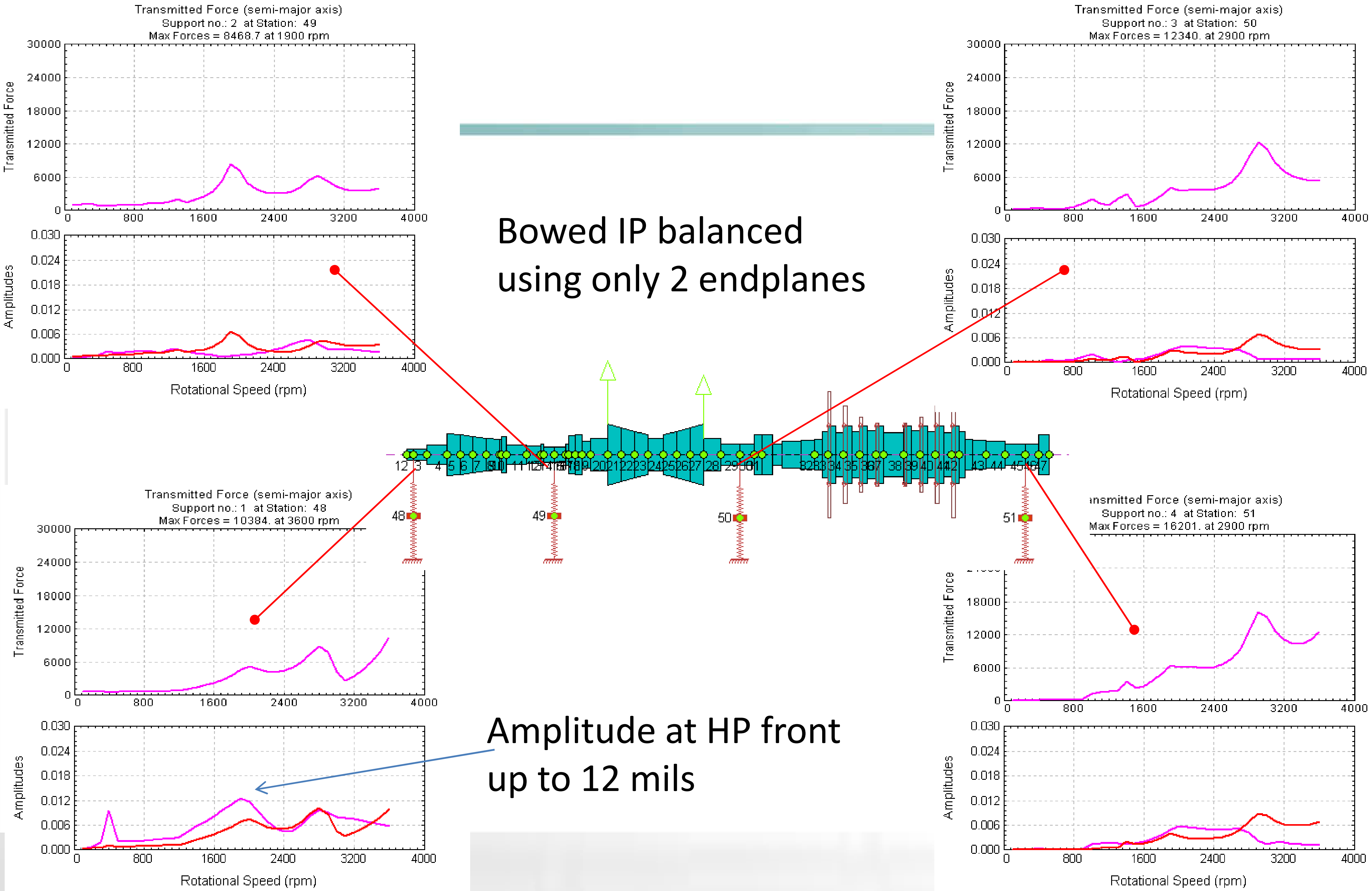
- The IP rotor had only two balancing planes, and proper balancing by the QHSB method would require a third balancing plane at the axial midpoint

- Adding a third balancing plane was not an option because of high operating temperature at the required location at the rotor midpoint
- Balancing alone in two planes would not resolve the problem of a bowed rotor as was attempted by another service provider.

- The only permanent solution was to throw the journal centers and re-machine couplings and journals to restore symmetry between the journal axis and rotor mass axis to a tolerance of less than 0.001”

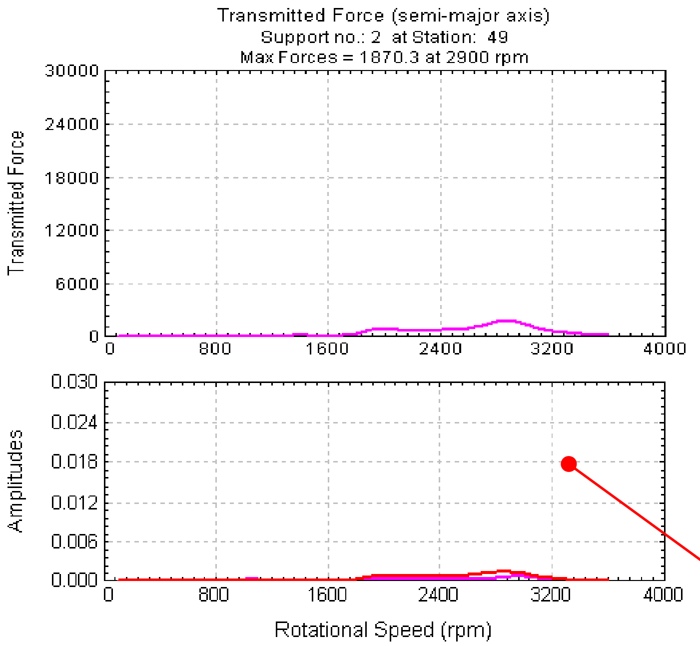
Bowed IP as-is, with no balancing



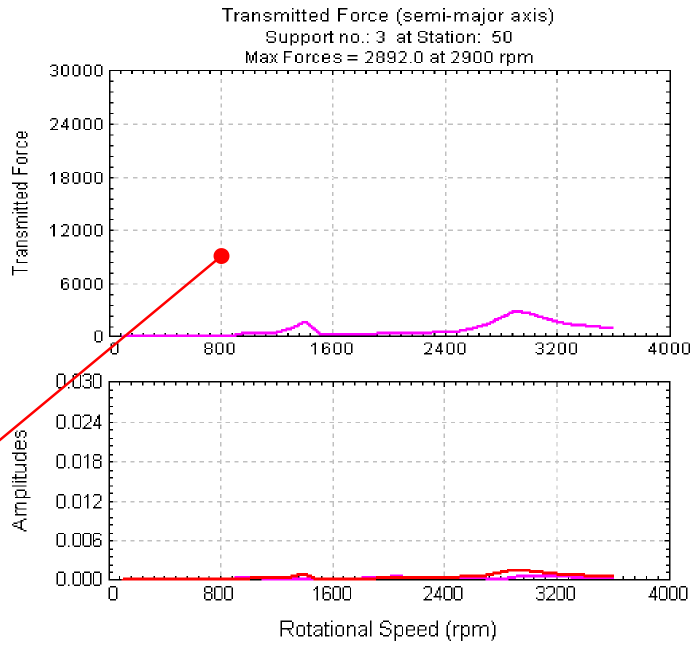


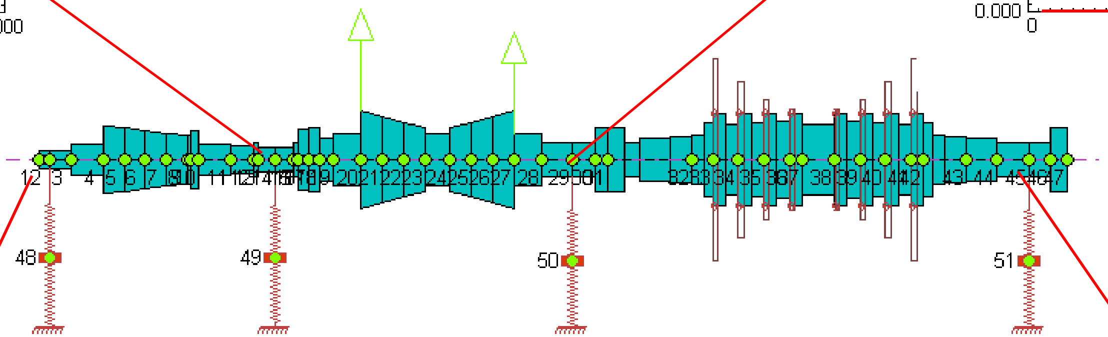
Bowed IP balanced using only 2 endplanes

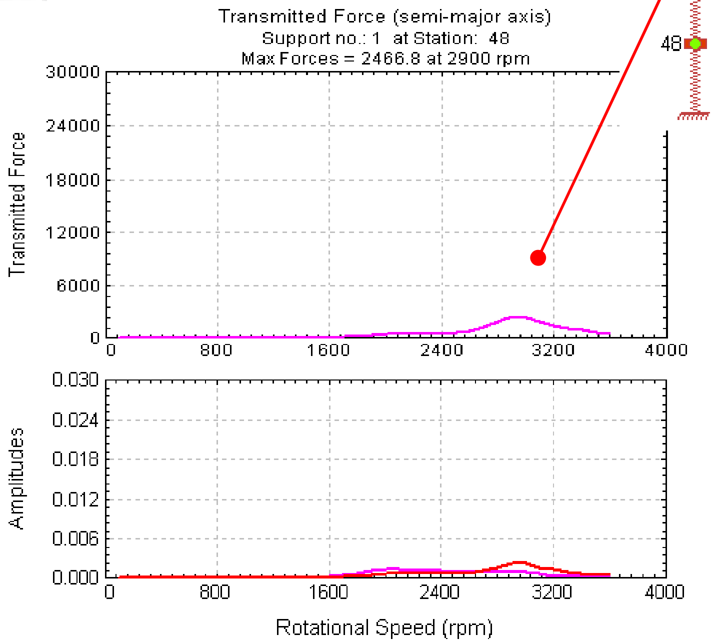
Amplitude at HP front up to 12 mils

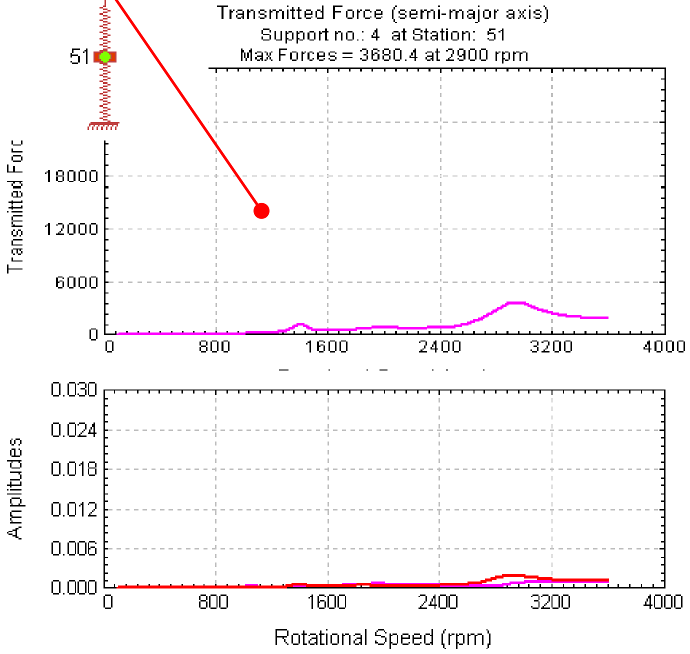


Journal centerline correction and balanced IP + coupling evaluation/repair and proper alignment





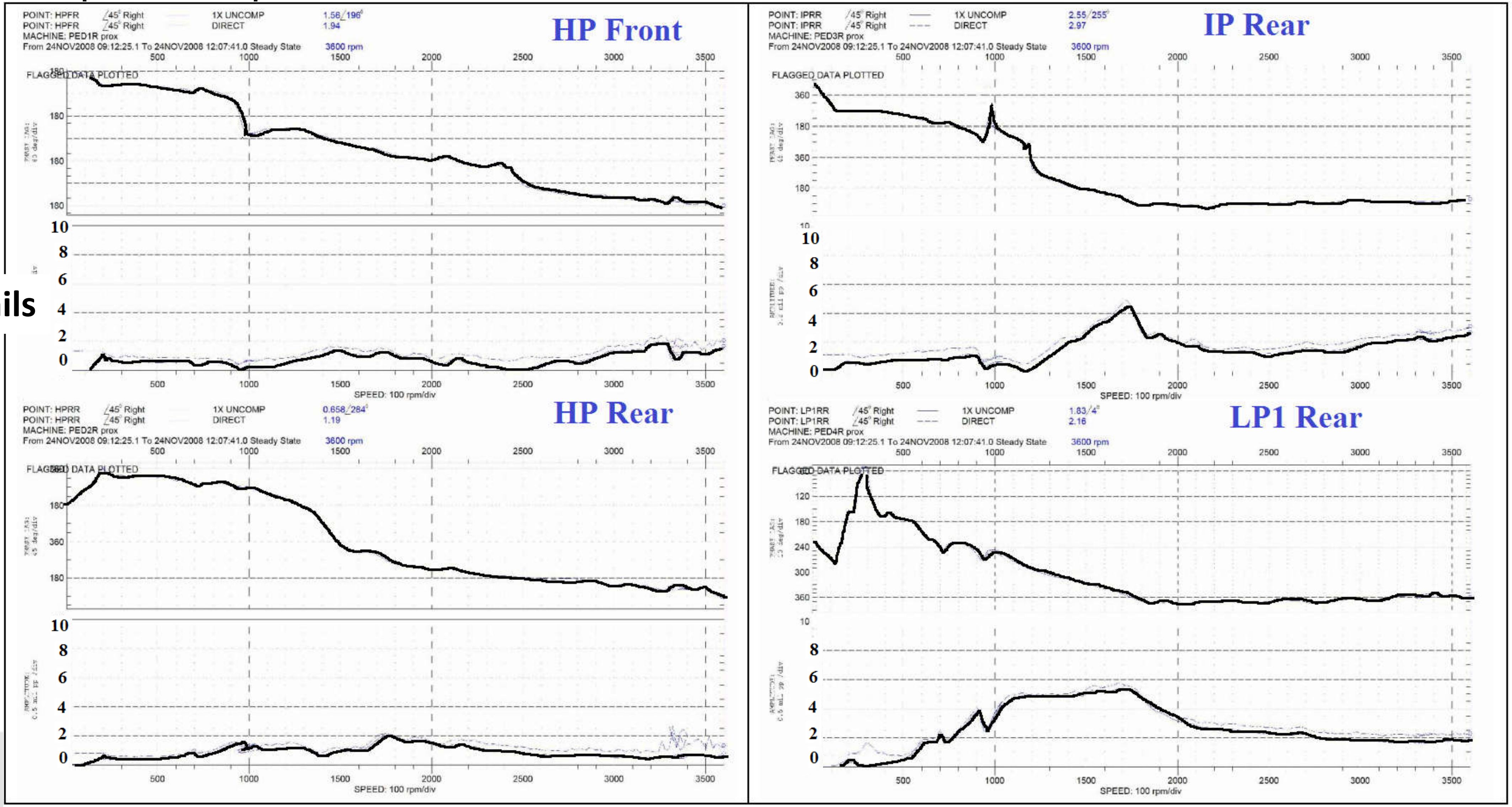




Case Study #4: Effects of a Bowed IP Rotor

Start up after repair:

mils



Case Study #5: A "Simple" Shop Balancing Correction

60 MW generator rotor from GE Frame 7 CTG

- After a rewind in the service shop, the rotor was set up for balancing on a high speed balancing machine by the shop's balancer (in air, no vacuum)

- TIR measurements had been taken and mathematically evaluated with Z-R Consulting's FFT program for 1x and 2x eccentricity.

- After ~6 hours of balancing by the shop's engineer, no compromise solution could be achieved. Either the first critical response was high or second critical response was high.

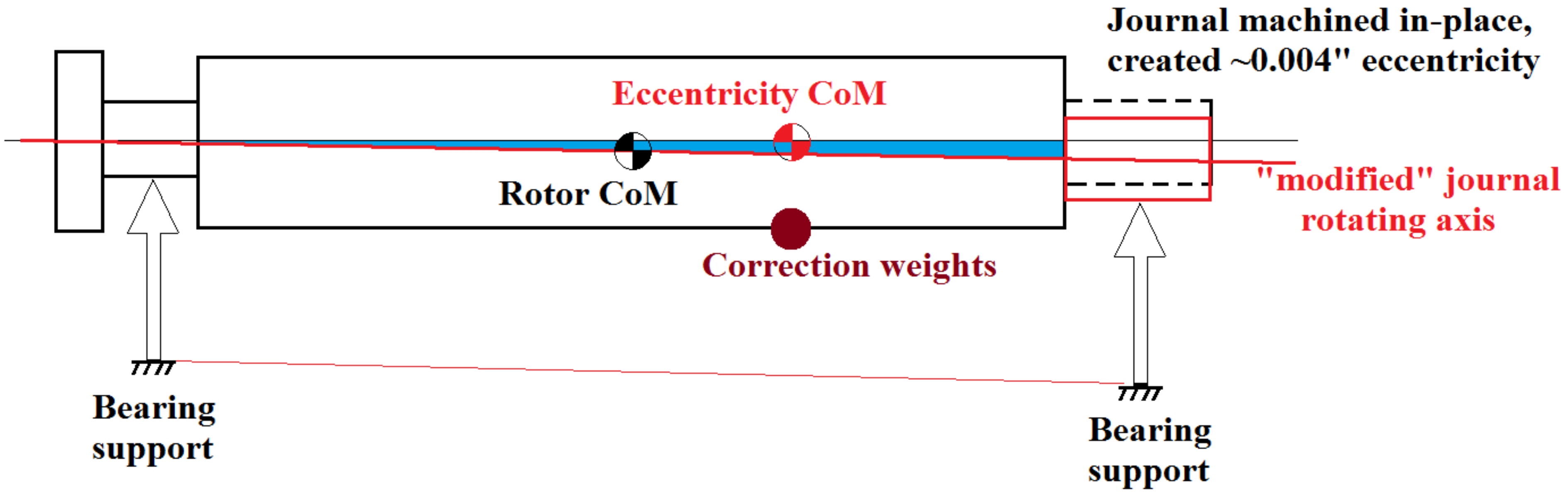
- The service shop engineer called us for immediate assistance.

Case Study #5: A “Simple” Shop Balancing Correction

- After arriving at the shop, we reviewed the TIR and evaluated 1x eccentricities.
- From the TIR review, it was suspected that the journal on the non-drive end had been in-place machined, as the journal center was radially offset by ~4 mils.
- This resulted in the rotor body acting as distributed eccentricity, now being radially offset and skewed from the journal centerline axis, toward the direction of the machined journal
- During balancing in only a single midplane, an axial moment had been created between the midplane balance weights and the center of mass of the eccentric rotor body, driving displacement amplitude

Case Study #5: A “Simple” Shop Balancing Correction

- The same amount of weight used to resolve displacement for the first critical was then shifted axially by ~30 inches, from the center of mass of the total rotor to the suspected center of mass of the eccentricity, based on the TIR evaluation.
- In the next run, the rotor was accelerated through the first critical to overspeed with fully acceptable vibration displacement.
- Since this occurred on December 24th at 11:55pm, the rotor was hence known as the “Christmas rotor”



More details and our published papers can be found at

Z-RConsulting.com