43rd Turbomachinery
30th Pump SYMPOSIA
GEORGE R. BROWN CONVENTION CENTER
HOUSTON, TX | SEPT. 22 - 25, 2014

EXAMPLES OF BALANCING METHODS: FOUR-RUN AND LEAST-SQUARES INFLUENCE COEFFICIENTS

TROY FEESE, P.E.
ENGINEERING DYNAMICS INC.
SAN ANTONIO, TEXAS

EDI ENGINEERING DYNAMICS INCORPORATED
TEXAS A&M UNIVERSITY
TEES TURBOMACHINERY LABORATORY
TEXAS A&M ENGINEERING EXPERIMENT STATION

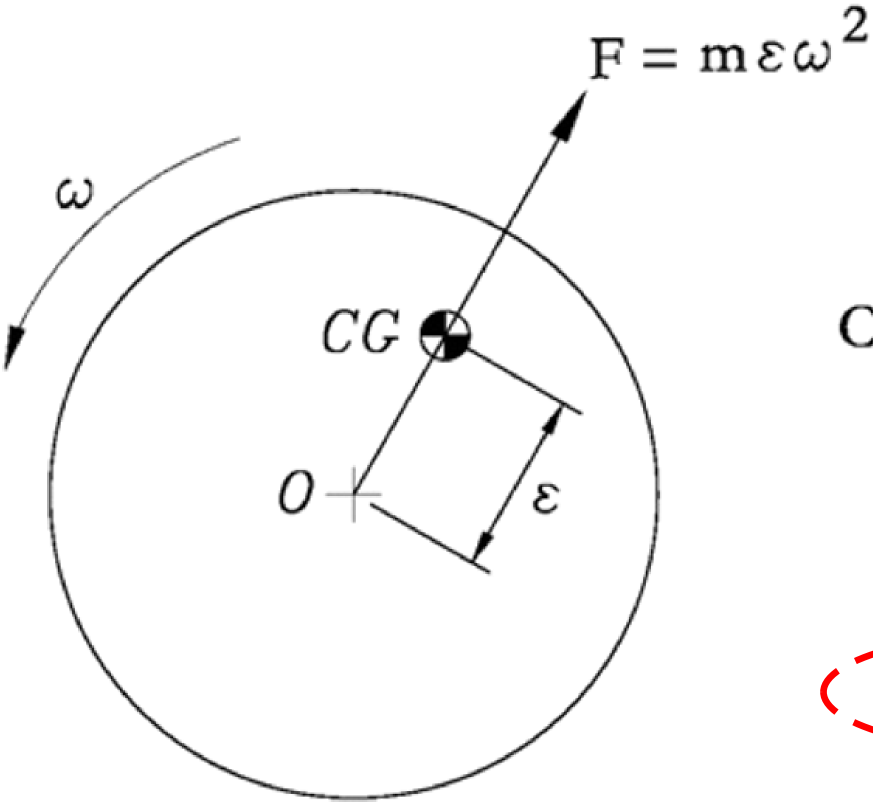
BIO – TROY FEESE

- Senior Engineer at Engineering Dynamics in San Antonio, TX
- 24 years performing torsional vibration, lateral critical speed, stability analyses, and FEA of structures / foundations
- Field studies of rotating and reciprocating machinery
- Lecturer at EDI Annual Seminar – San Antonio Riverwalk
- Published papers / articles on torsional vibration, lateral critical speeds, and balancing
- Member of ASME, Vibration Institute, Contributed to API 684, and GMRC Torsional Sub-Committee
- BSME from The University of Texas at Austin (1990)
- MSME from The University of Texas at San Antonio (1996)
- Licensed Professional Engineer in Texas (1996)

Introduction

- Balancing is often required to reduce vibration at 1× running speed.
- Balancing in-place is also referred to as **Field Balancing** or **Trim Balancing**.
- Common balancing techniques:
  - Single-Plane Vector Method
  - Four-Run Method (No Phase)
  - Least-Squares Influence Coefficients

When Center of Gravity Differs from Axis of Rotation This Causes Imbalance



O = Journal Axis

CG = Center of Gravity

F = Centrifugal Force

m = Mass of Section

ε = Eccentricity

ω = Angular Velocity

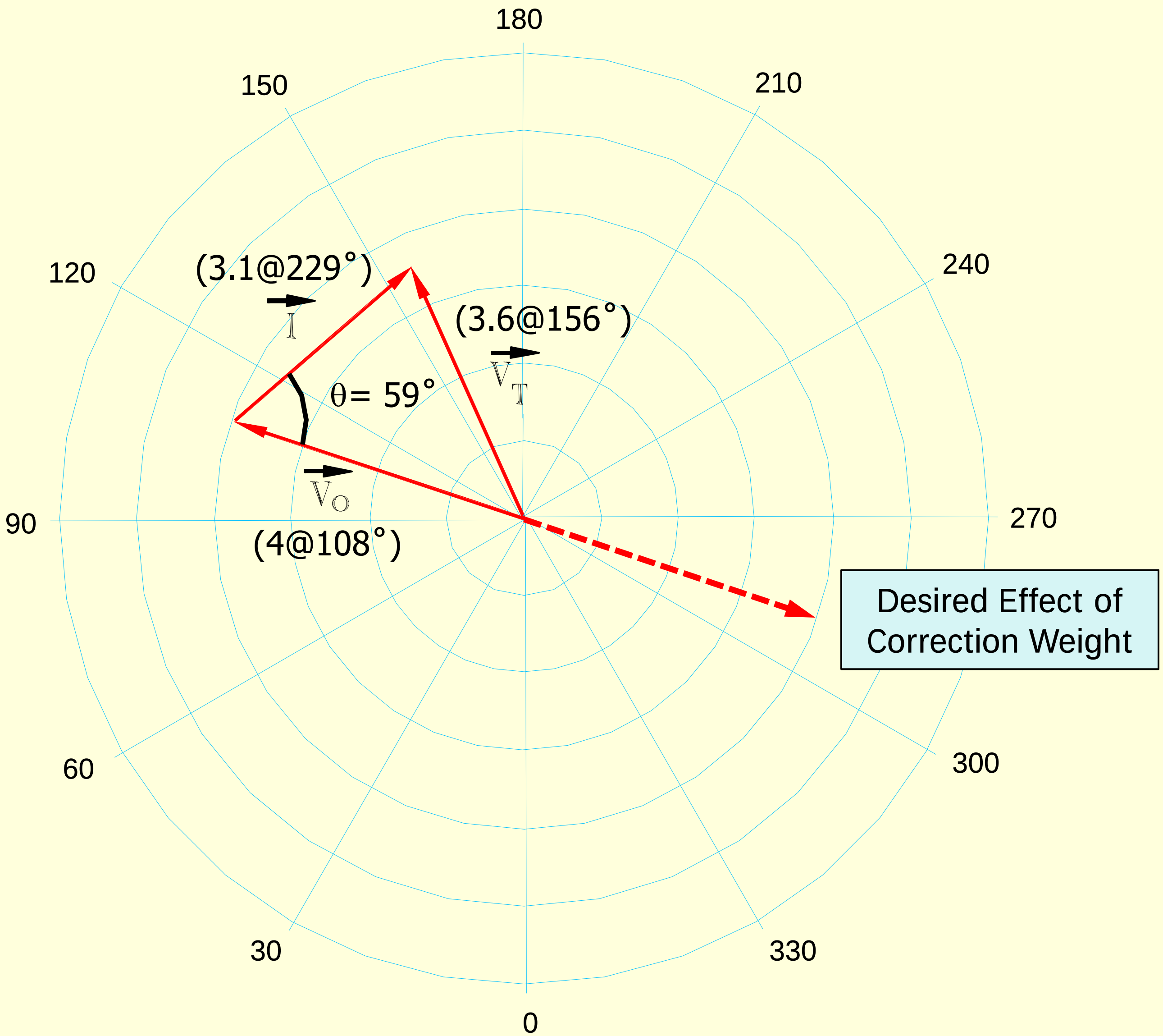
Sources of Imbalance

- Fan in dirty service.
- Variation in material density due to voids, porosity, or finish.
- Unsymmetrical parts.
- Bent shaft, erosion, wear, or other damage.
- Tolerances in fabrication, machining, or assembly.
- Shifting of parts due to shaft distortion, insufficient shrink fit, aerodynamic forces, or thermal effects.

Review of Vector Method

- V_O represents the original vibration reading (as found or baseline)
- V_T is the vibration due to the trial weight plus original vibration
- Vector I (influence) is determined by subtracting V_O (original) from V_T (trial)

Single-Plane Vector Method

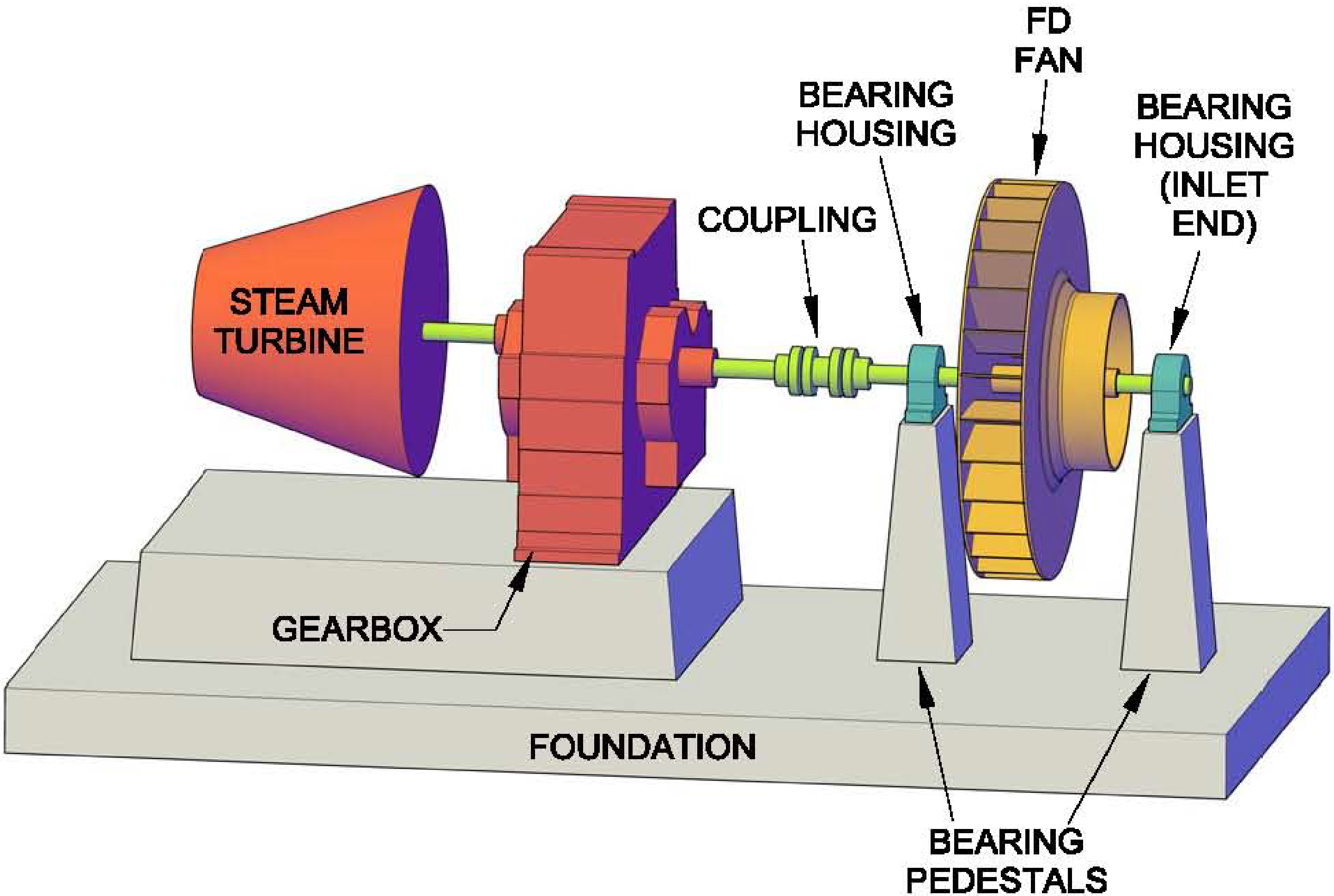


- Correction Weight =

  $$\text{Trial Weight} \cdot |V_O| / |I|$$

- Location of CW is determined by angle θ
- Should remove TW before installing CW

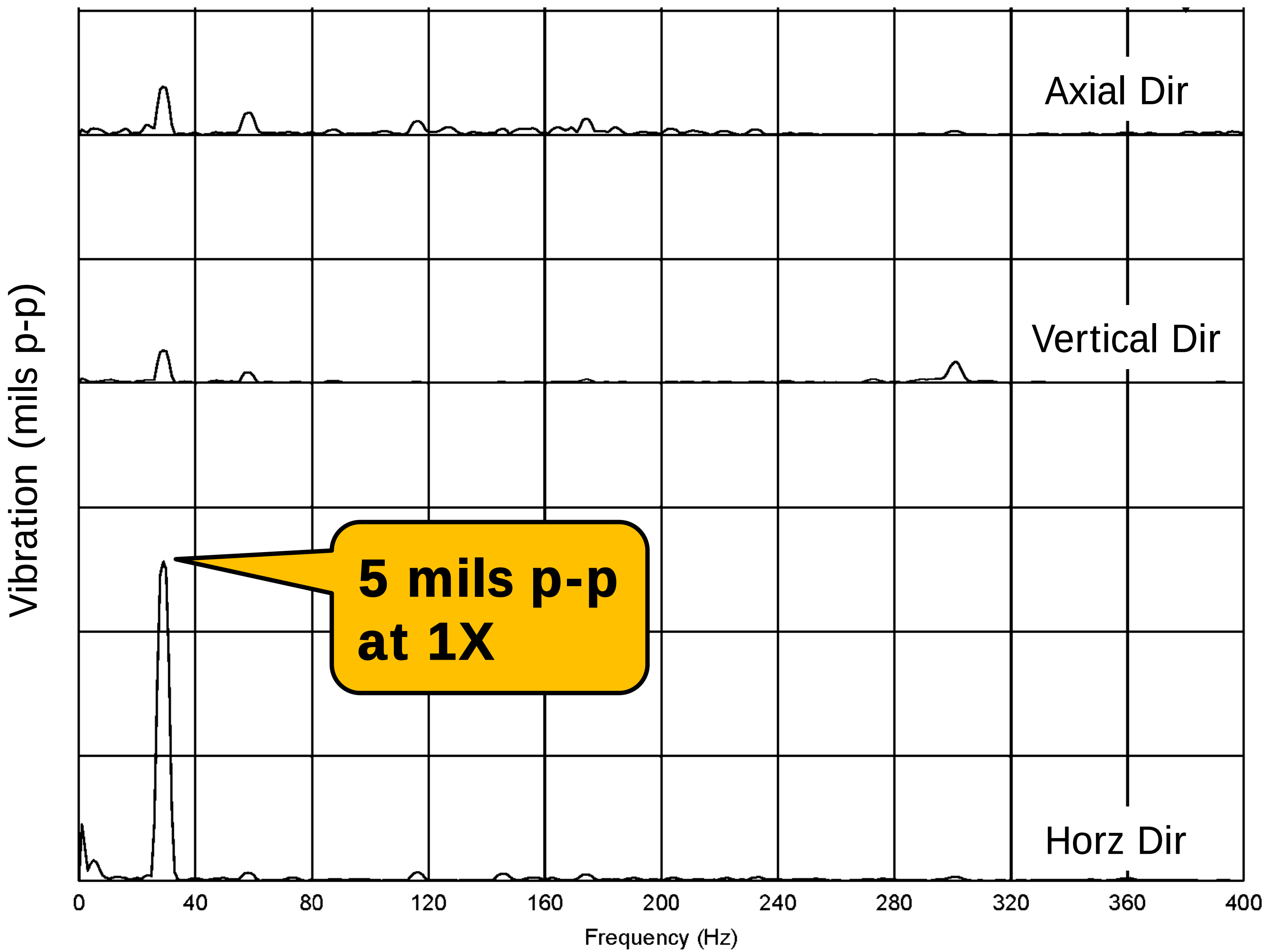
CASE 1:
Forced Draft (FD) Fan



Background Information

- Plant Personnel Reported High Vibration of FD Fan After Replacing Roller Bearings
- Second Set of Bearings Were Installed, But Vibration Remained High
- Predominant Vibration at 1× Running Speed of 1745 RPM (29 Hz)
- Reported Difficulty Balancing the Fan

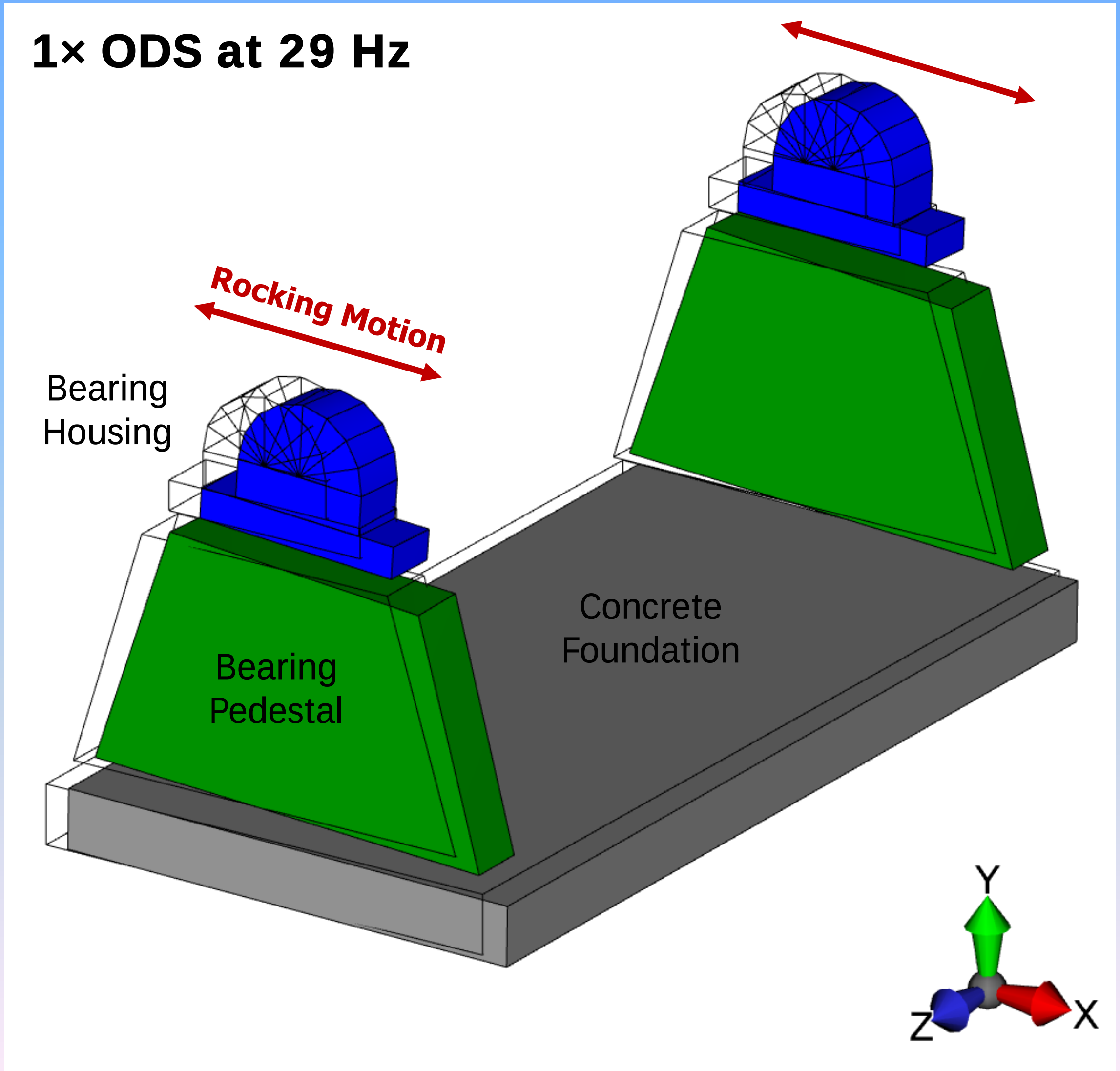
Fan Inlet Bearing



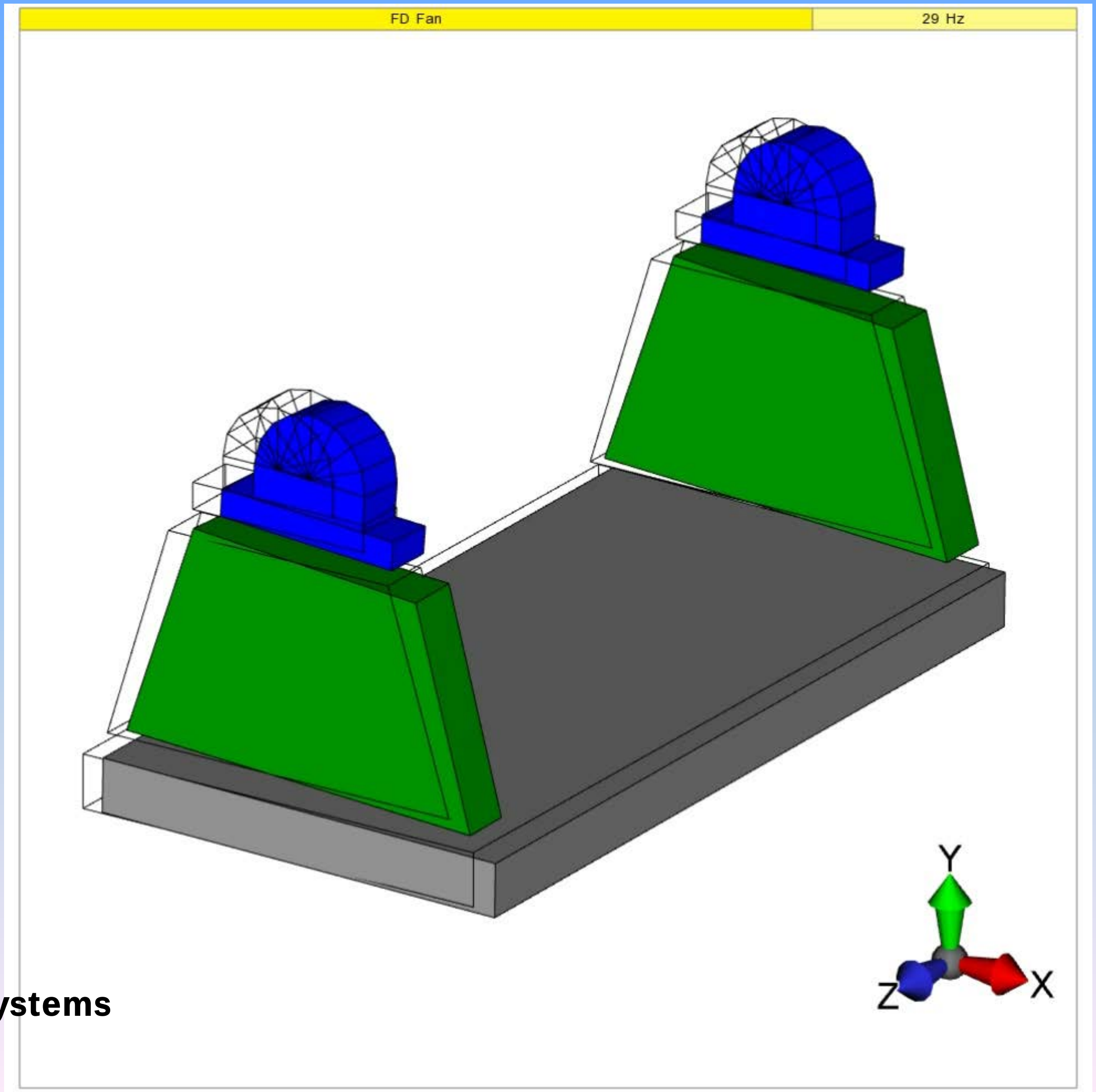
Operating Deflection Shape (ODS)

- 3D representation from basic dimensions
- Tri-axial accelerometer used to measure vibration in three directions at 18 points
- Vibration in displacement (mils p-p)
- Phase angles determined from transfer function and stationary accelerometer
- Modal software used to animate motion at 1× running speed (29 Hz)

Still-Frame Representation



ODS Animation*



*Courtesy of
Clear Motion Systems
San Antonio, TX

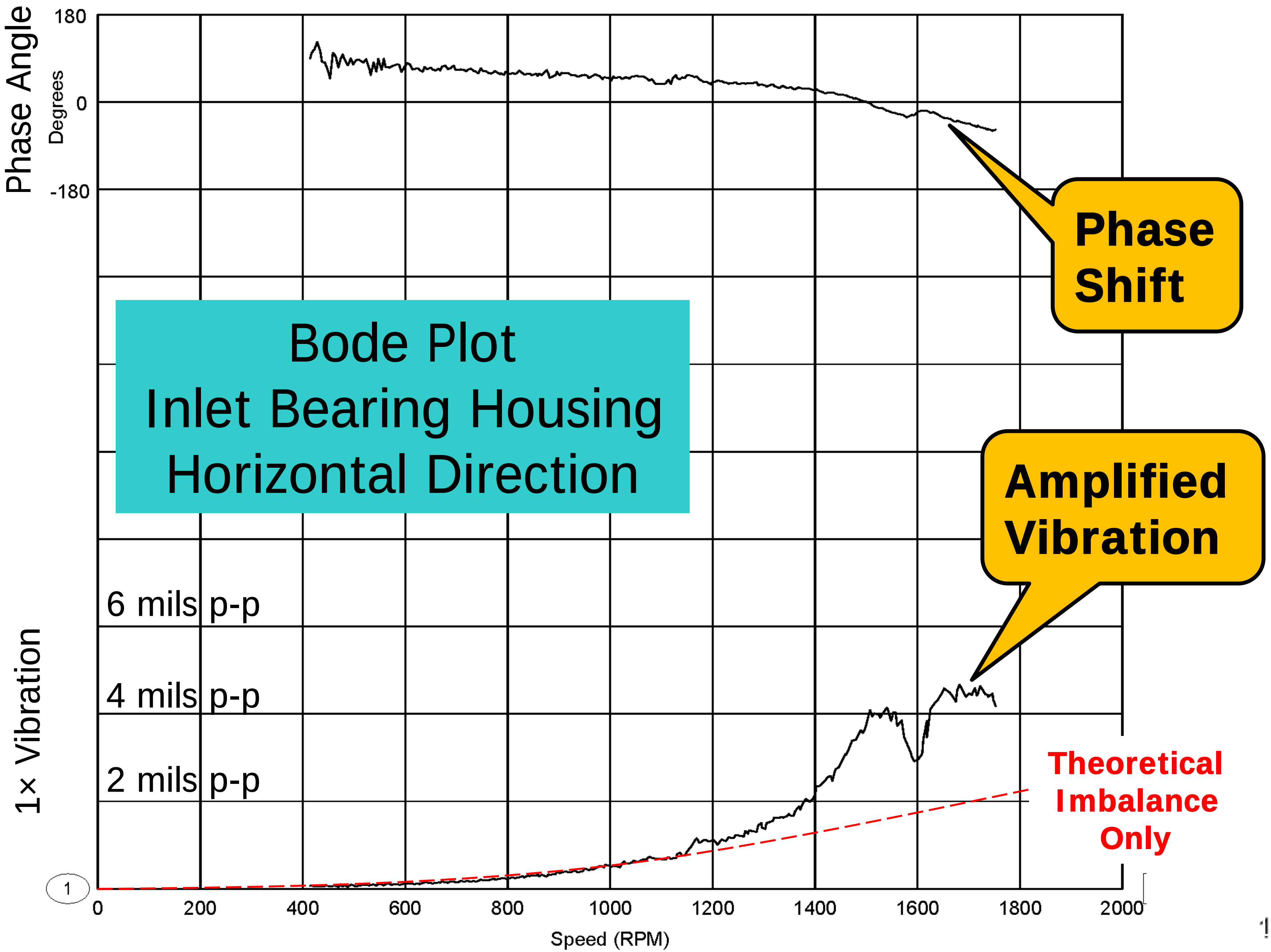
FD Fan / 29 Hz

Characteristics of Bearing Housing Vibration

- Occurred Primarily at 1× Running Speed
- Highest at Top of Bearing in Horizontal Direction (5 to 6 mils p-p)
- Inlet and Coupling Ends Move In-Phase
- No Looseness Found Between Bearings, Pedestals, and Concrete Foundation
- High Vibration Measured on Concrete Foundation (3 to 4 mils p-p)

Insensitive to Load

- Closed Louvers for Test –
  Flow Reduced from 70,000 lb/hr
  to Essentially Zero
- No Significant Change in 1× Vibration Readings on Bearing Housings
- Similar Vibration Amplitudes and Phase Angles at Each Bearing
- Indicates Static Imbalance of Fan Impeller, Not Flow Induced Vibration


Phase Angle
Degrees
180
0
-180
Phase Shift
Bode Plot
Inlet Bearing Housing
Horizontal Direction
Amplified Vibration
6 mils p-p
4 mils p-p
2 mils p-p
1× Vibration
Theoretical Imbalance Only
1
0
200
400
600
800
1000
1200
1400
1600
1800
2000
Speed (RPM)

Fan Inspection Results

- No obvious mechanical damage.
- Fan impeller is dirty, which could affect balance condition.
- Five balance weights of various sizes already welded to fan impeller.

Four-Run Balance Method

- Requires vibration data at 1× running speed. Can be mils or in/sec as long as consistent units are used.
- Phase angle data **not** required.
- Simple, graphical method.
- Computer software **not** needed.

Four-Run Balance Method

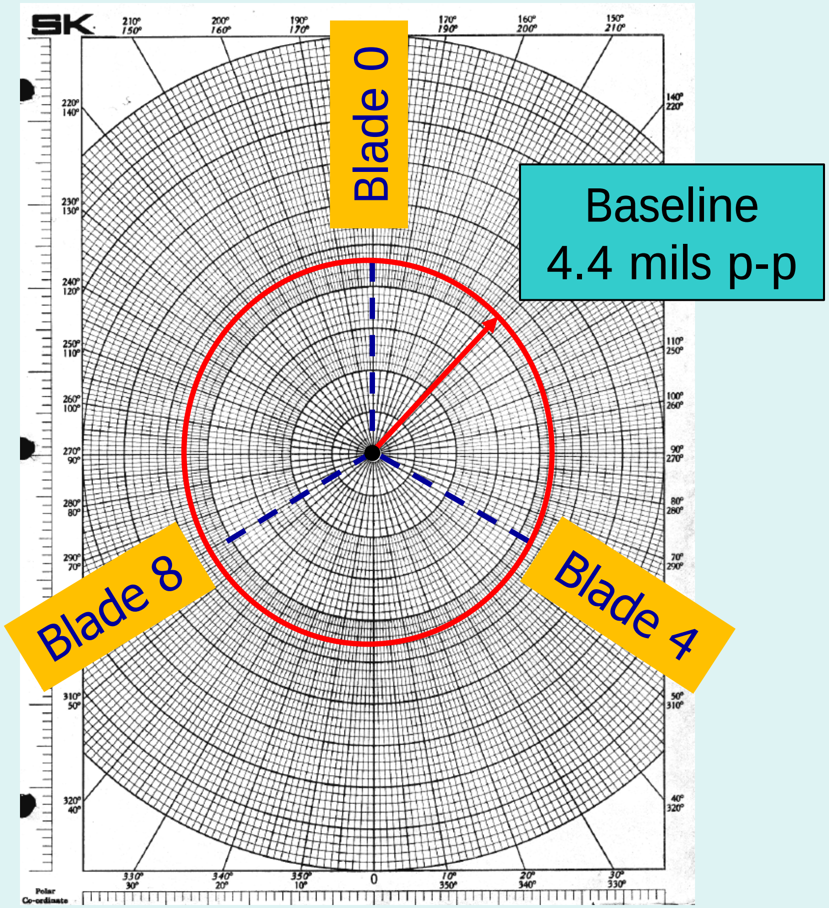
Good method for balancing near resonance since it does not rely on phase angles. Results can easily be derived using polar plot paper and a compass. Assume static imbalance of fan impeller (single plane).

Steps:

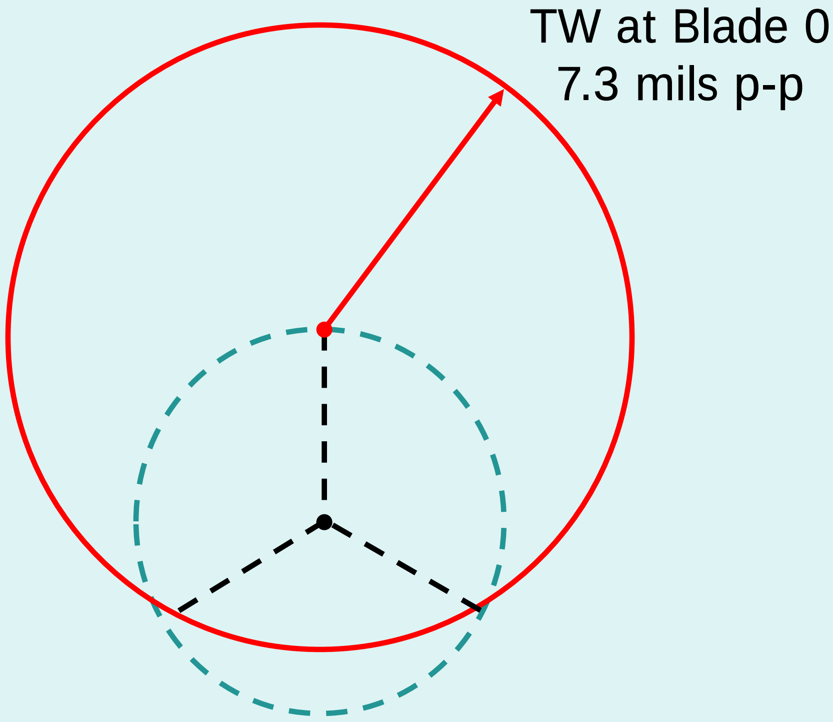
1. Number fan blades from 0 to 11, opposite rotation
2. Readings taken with the fan running at 1745 RPM
3. Speed verified with optical tach and strobe light

4. Baseline was 4.4 mils p-p (no weights)

5. Locate blades where TW will be applied

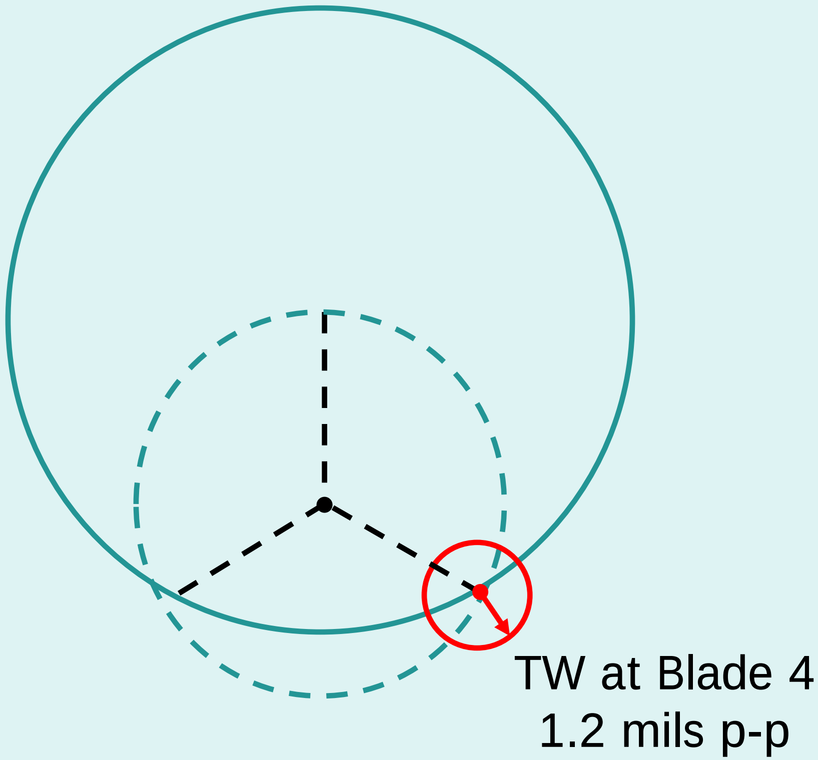


6. Trial weight selected, washer weighed 3.2 oz

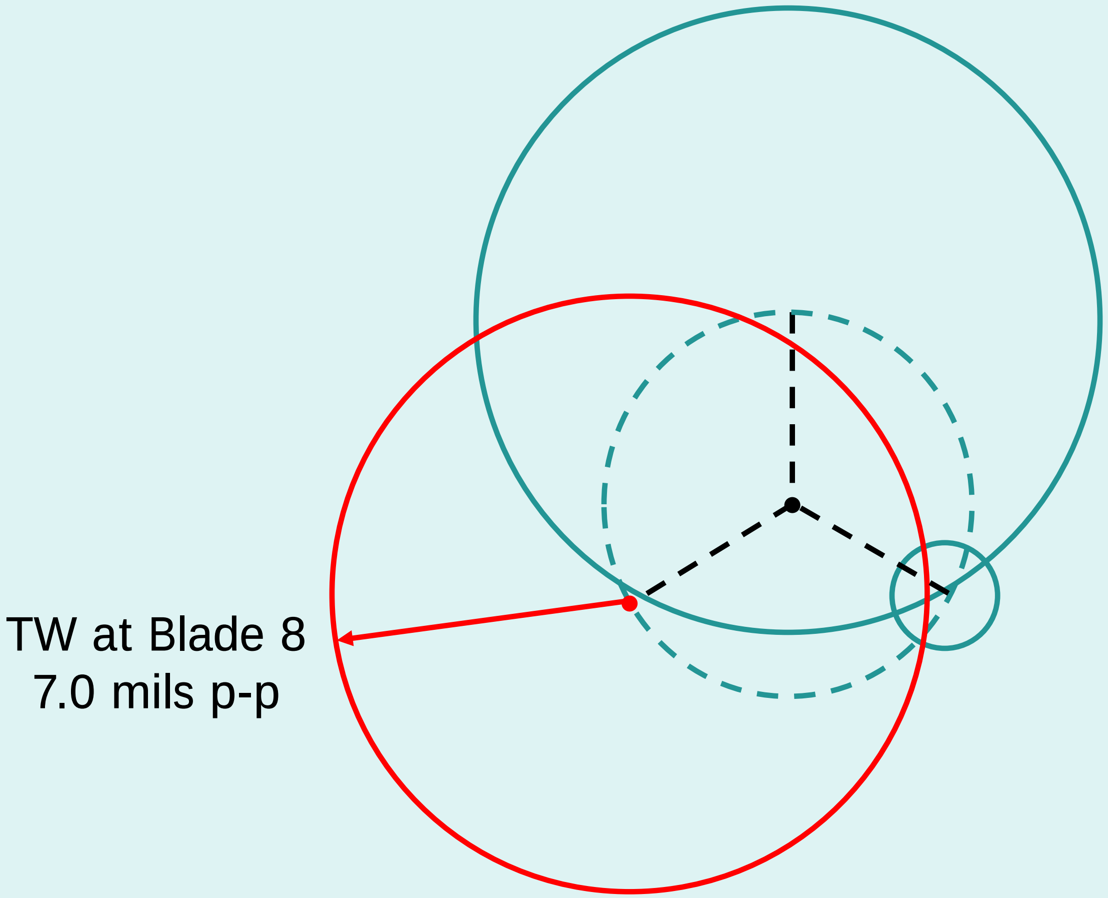


7. Washer welded to fan impeller near blade 0. Resulting vibration was 7.3 mils p-p.

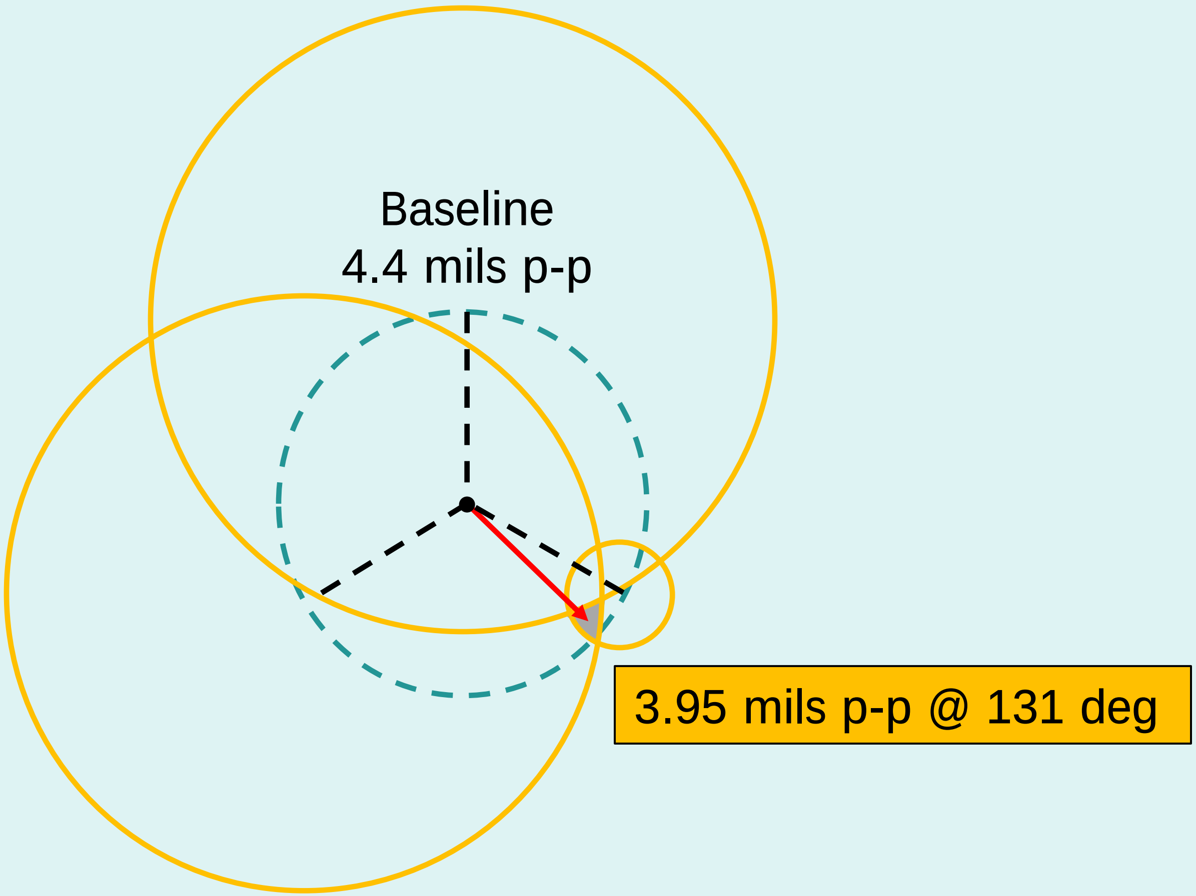
8. Washer moved to fan blade 4 (120 deg)



9. Washer moved to fan blade 8 (240 deg)



10. Find Approximate Intersection Point of Circles Representing the Three Trial Runs



11. Correction Weight =

$$3.2 \text{ oz} \cdot 4.4 \text{ mils} / 3.95 \text{ mils} = 3.6 \text{ oz}$$

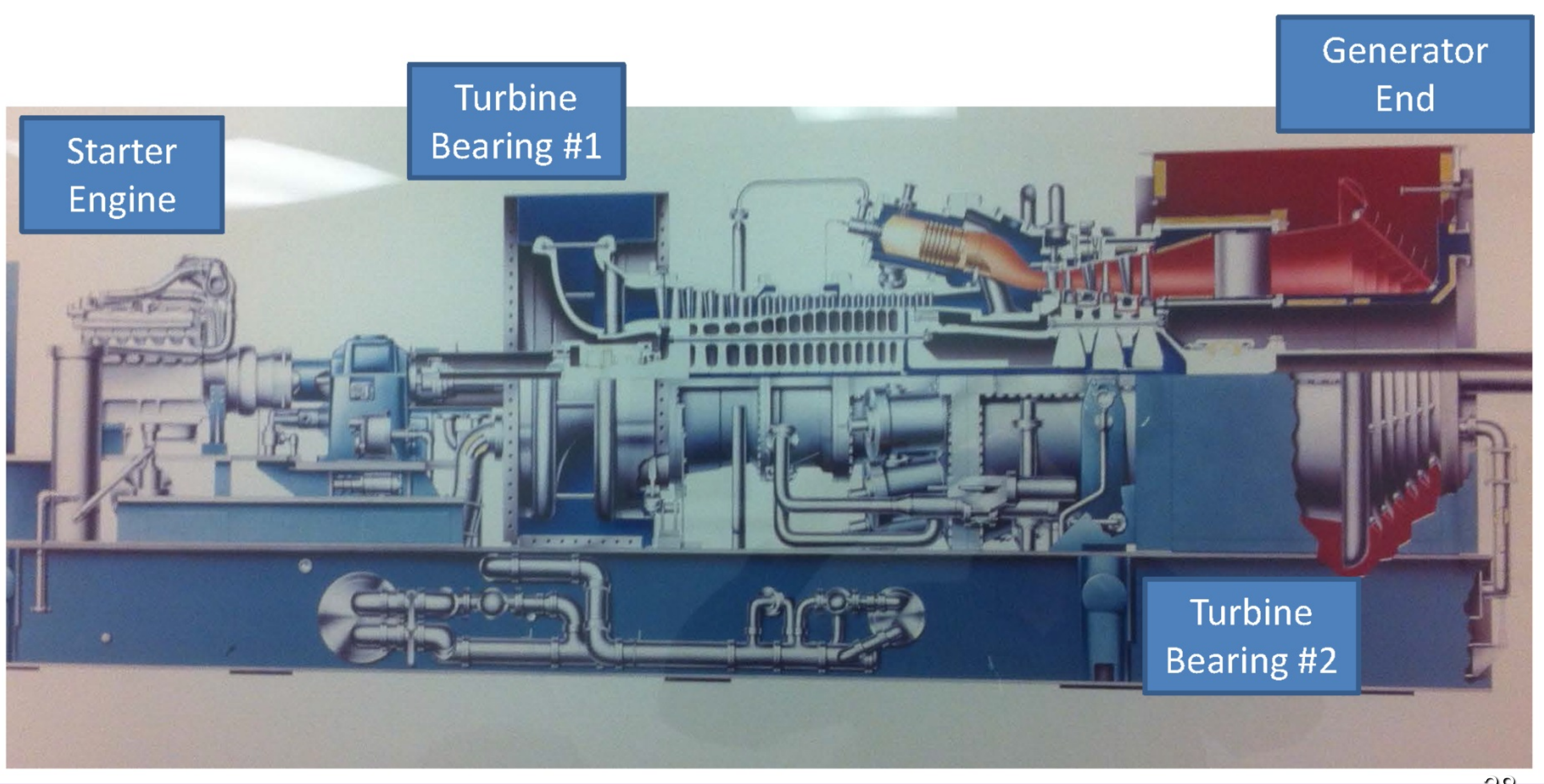
Balance Summary

- Fan blade 4 was optimum location
- Correction weight was 3.6 oz, slightly more than trial weight of 3.2 oz
- Bearing vibration was reduced to 1.2 mil p-p (0.1 ips peak at 29 Hz)
- Final balanced condition was considered acceptable for operation

Case 1 – Conclusions

- When balancing near a resonant condition, phase angles may vary. Using simple four-run method was good option for the FD fan.
- Natural frequencies of the fan rotor, impeller, and foundation should have a separation margin of at least 10% from the operating speed range to avoid high sensitivity to small amounts of imbalance, fouling, etc.
- The FD fan moves air at ambient temperature so thermal effects are not prevalent like an induced draft (ID) fan or turbine would be.

Case 2:
Balancing Gas Turbine

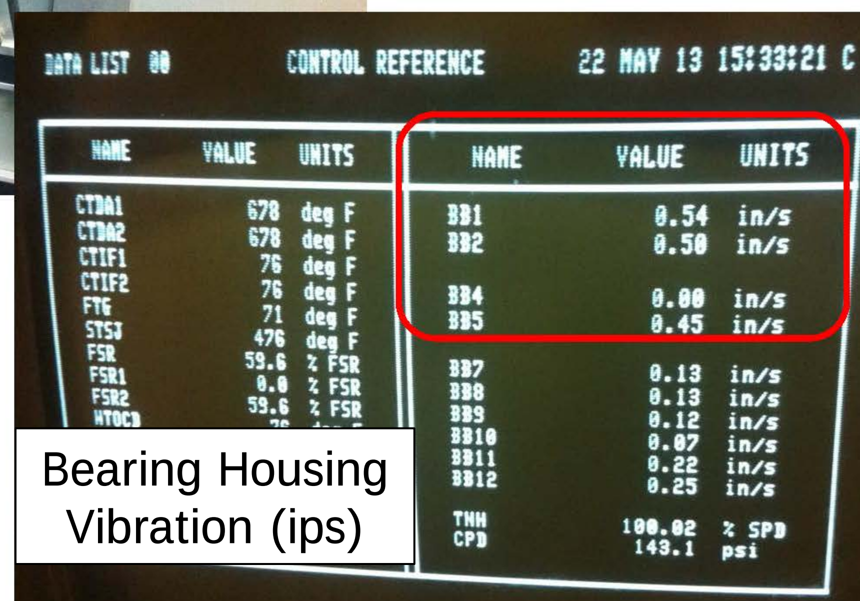


Background

- Turbine has history of high vibration since commissioning 20 years ago.
- Previous balance attempts were largely unsuccessful.
- The keyphasor (KP) was unreliable making it difficult to reuse influence coefficients.
- Several other problems found with couplings, bearing pitting, and magnetism.

Vibration Measurements

Bently Panel for Shaft Proximity Probes (mils)

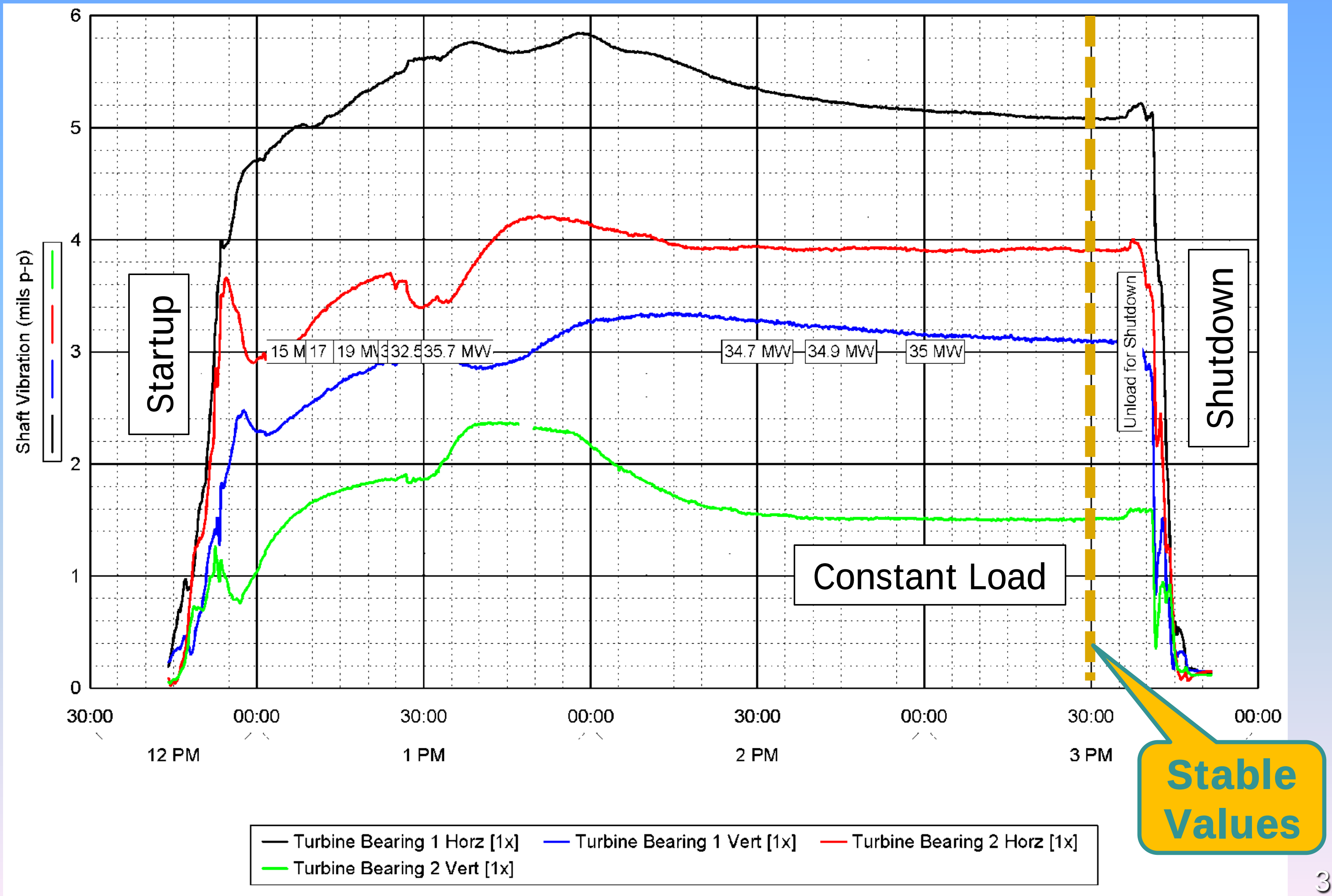


Bearing Housing Vibration (ips)

Observations

- A temporary optical KP was installed.
- Vibration amplitudes and phase angles were trended over several hours.
- It was determined that 3 hours were required to stabilize the turbine vibration while generating 35 MW of power.
- Previous balance attempts did not allow sufficient time for heat soaking of rotor.

Vibration Trend (3-Hr Period)



Influence Coefficients

- Determined from trial weights and vibration measurements.
- Goodman (1964) applied least squares.
- Assumed linear behavior.
- Can be used with multiple balance planes and operating speeds.
- Predicted residuals indicate if rotor can theoretically be balanced.

Influence Coefficients (cont.)

- Commonly used for dynamic balancing.
- Matrix operations may require calculator or computer program.
- Use vibration amplitude and phase at 1× running speed in multiple directions.
- Must subtract runout vectors from proximity probe readings.

Steps:

- Obtain baseline vibration (amplitude and phase) after machine is heat soaked and readings stabilize.
- Install trial weight and retake vibration readings.
- The angular location of trial weight is typically referenced to the key phasor, opposite shaft rotation.
- Repeat for each balance plane.
- The influence coefficients are calculated by subtracting the baseline from the trial data and dividing by the trial weight.
- Solve for the correction weight(s) needed to minimize residual vibration.

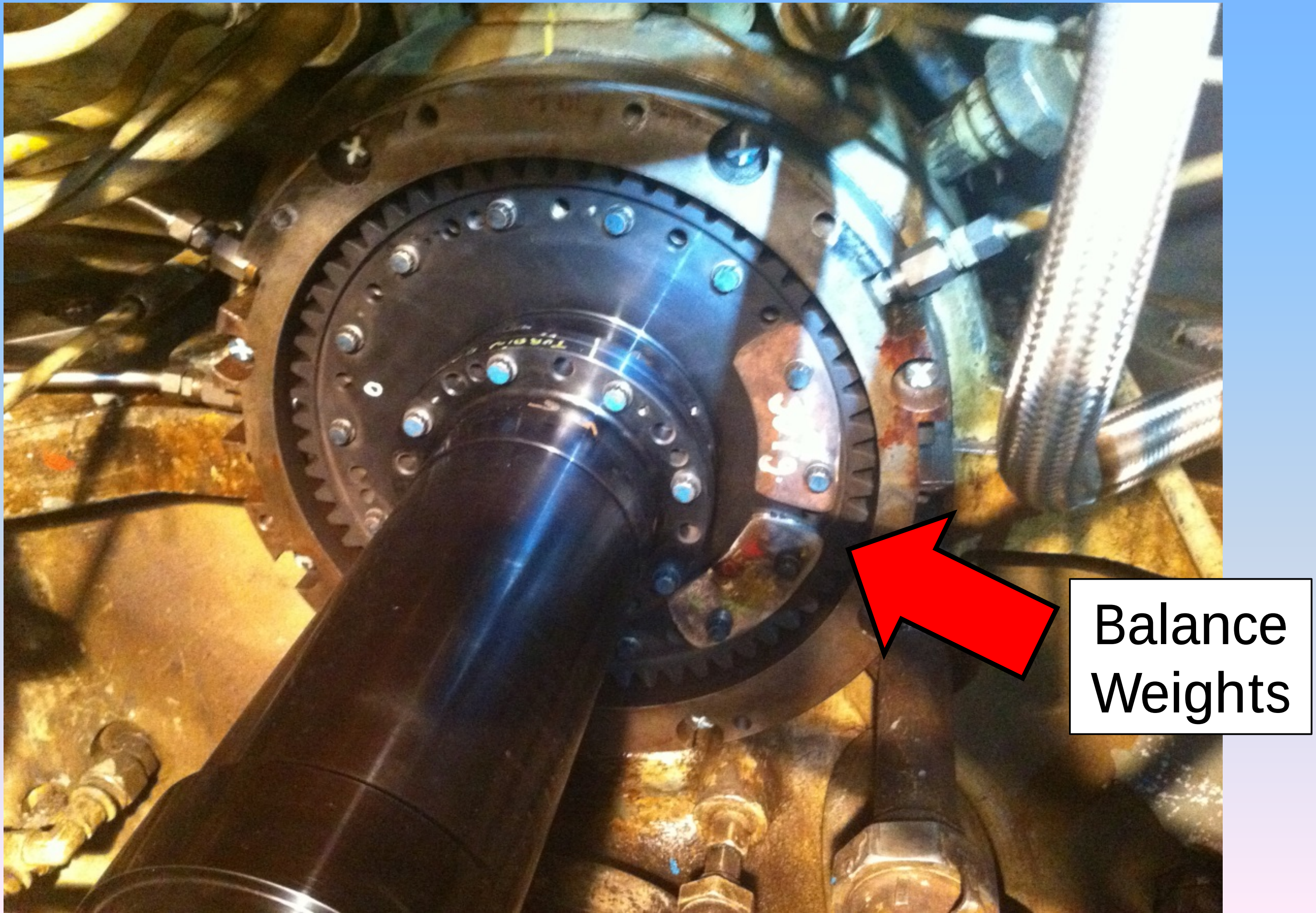
10% Force Method

“Rule of thumb” for sizing initial trial weight:

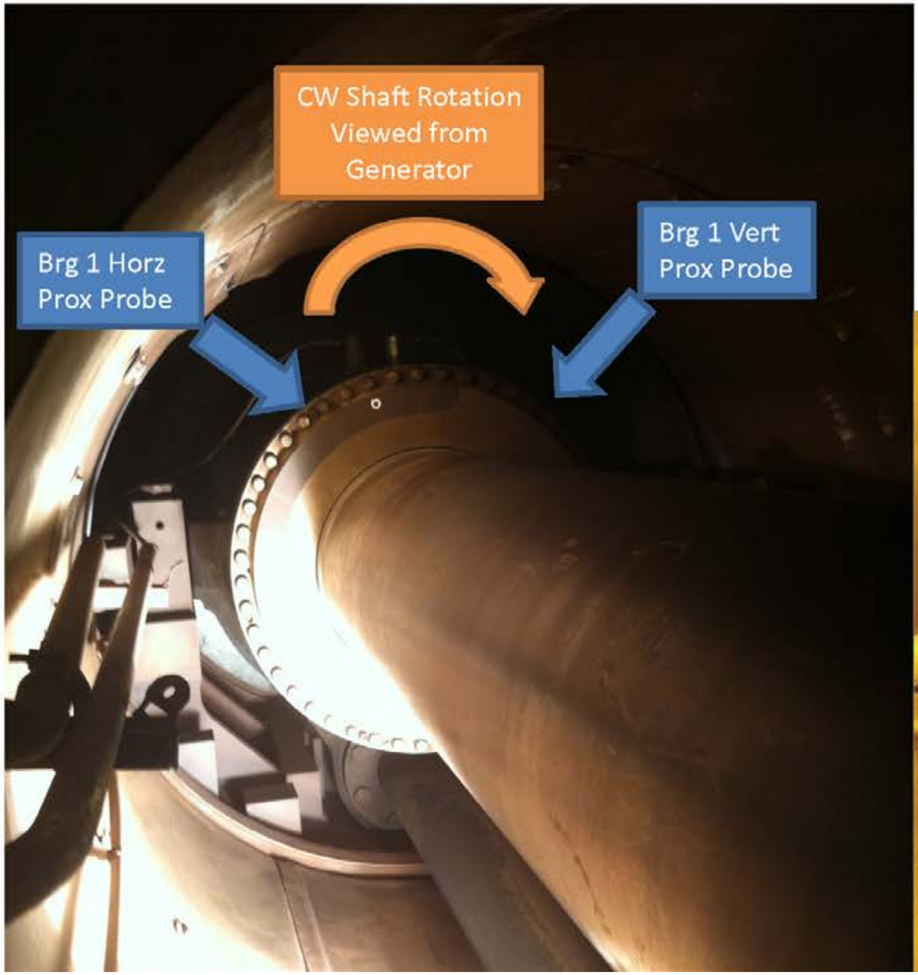
$$U = \frac{56347 \cdot W}{N^2}$$

Where: U = Residual Imbalance (oz-in)
W = Journal Weight (lbs)
N = Speed (RPM)

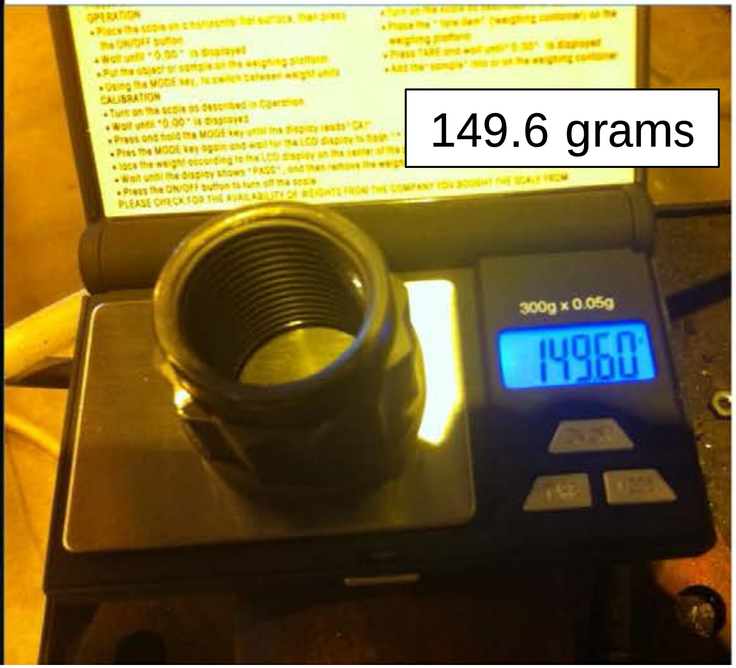
Balance Plane 1: Accessary Coupling (Gearbox End)



Balance Plane 2: Load Coupling (Generator End)



Trial Weight at Plane 2
One Nut Installed on Backside of Coupling



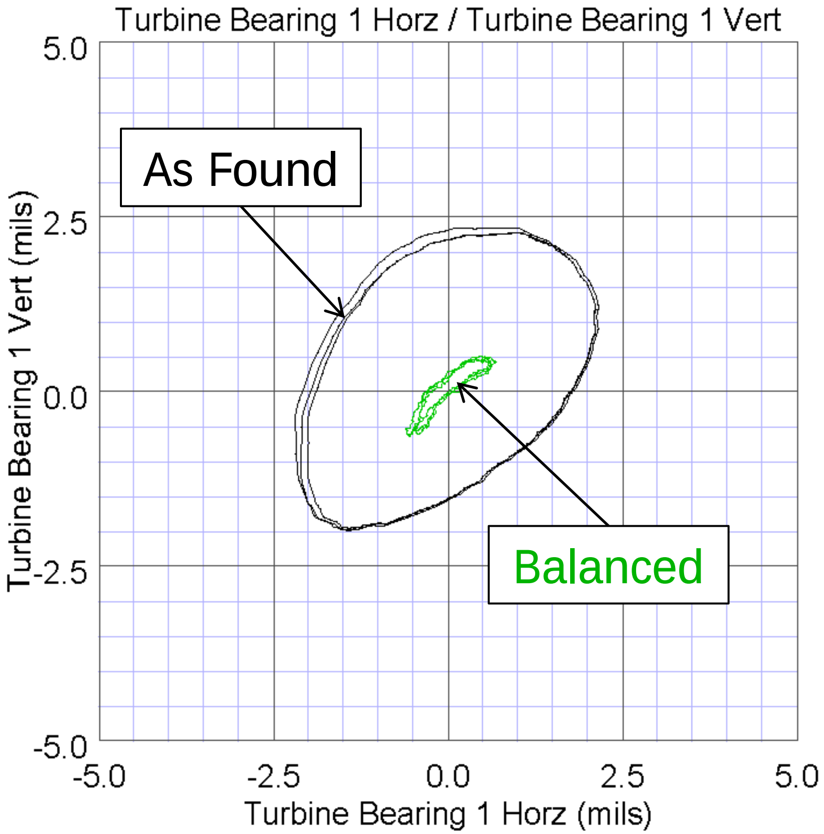
Summary at Base Load

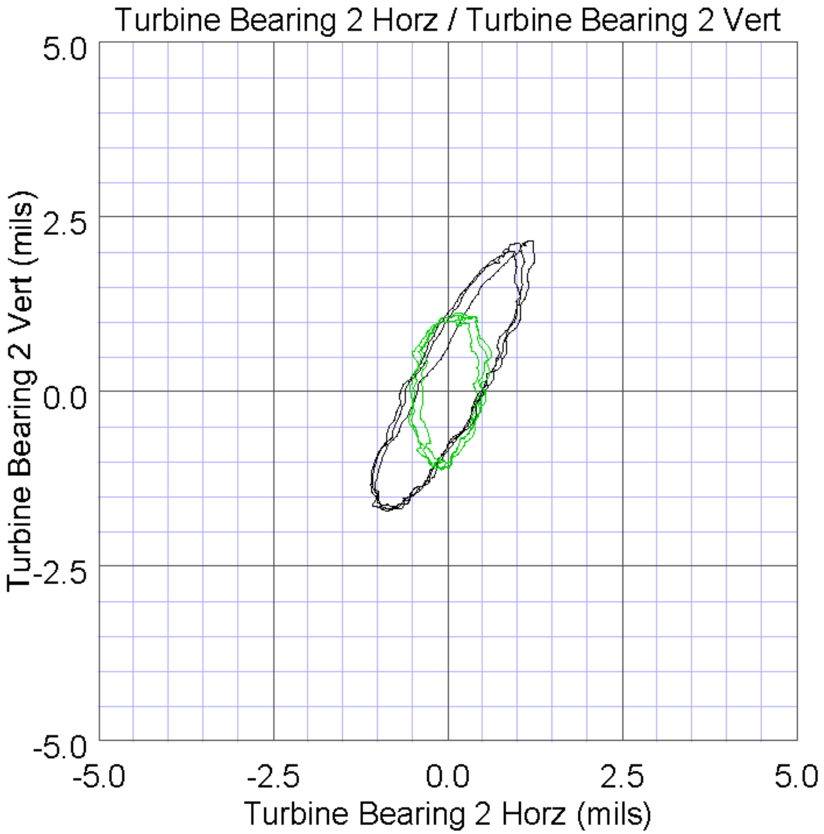
1X Vibration (mils p-p @ deg)

	Brg 1 Horz Dir	Brg 1 Vert Dir	Brg 2 Horz Dir	Brg 2 Vert Dir	Plane 1 Mass (g)	Plane 2 Mass (g)
Baseline	5.1 @ 97	3.1 @ -173	3.9 @ -53	1.5 @ -13	-	-
TW PL1	4.85 @ 77	5.2 @ -176	7.3 @ -27	1.1 @ 69	307 @ 51	-
TW PL2	5.0 @ 71	3.9 @ 169	5.1 @ -37	0.74 @ 46	-	150 @ 45
Prediction	1.5 @ 66	1.4 @ 352	0.84 @ 342	1.3 @ 13	667 @ 174	505 @ 343
Correction	1.5 @ 107	0.2 @ 6	1.75 @ -41	1.5 @ 7	555 @ 180	440 @ 345
% Change	-70%	-93%	-55%	0%		

By balancing on the turbine couplings, the shaft vibration was reduced from 5.1 to 1.75 mils p-p.

Orbit Plots:
As Found vs Balanced





Case 2 – Conclusions

- Turbine vibration readings are often sensitive to heat and load. During the testing, approx. 3 hours were required for the vibration readings to stabilize. Even after steady readings, vibration could still vary with load.
- Final correction weights were installed “out-of-phase” on both ends of the turbine. This indicates sensitivity to the conical whirl mode and not the rotor midspan mode, which would have “in-phase” vibration at both bearings.
- Large weights were required to balance the turbine, which indicates available balance planes are at ineffective locations compared with where the actual imbalance occurs in the rotor.

References

- Feese, T., "Least Squares Balance Program for the Hewlett-Packard 48GX Calculator," Vibrations Magazine, March 1998.
- Feese, T., "High Shaft Vibration of Synchronous Generator," Vibrations Magazine, December 1998.
- Feese, T. D. and Grazier, P. E., "Balance This! Case Histories from Difficult Balancing Jobs," 33rd Texas A&M Turbomachinery Symposium, September 2004.
- Fox, Randall, "Dynamic Balancing," 9th Texas A&M Turbomachinery Symposium, 1980.
- Goodman, Thomas P., "A Least-Squares Method for Computing Balance Corrections," J. Engrg. Indus., Trans. ASME, August 1964.

References (cont.)

- Jackson, Charles, "Single Plane Balancing," Texas A&M Turbomachinery Symposium, February 1991.
- MacPherson, H., Taylor, T., Feese, T., "An 11000 RPM Steam Turbine Case History," Canadian Machinery Vibration Association, 2003.
- Shreve, Dennis, "Balancing Without Phase," UPTIME Magazine, January 2011.
- Smith, D. R. and Wachel, J. C., "Controlling Fan Vibration – Case Histories," EPRI Symposium on Power Plant Fans: The State of the Art, Indianapolis, Indiana, Oct. 1981.
- Wowk, Victor, Machinery Vibration: Balancing, McGraw-Hill, 1995.