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

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Introduction

• Balancing is often required to reduce vibration at $1 \times$ 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

\[ F = m \varepsilon \omega^2 \]

O = Journal Axis
CG = Center of Gravity
F = Centrifugal Force
m = Mass of Section
\varepsilon = Eccentricity
\omega = 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

\[ \theta = 59^\circ \]

Desired Effect of Correction Weight

\( (4@108^\circ) \)

\( (3.1@229^\circ) \)

\( (3.6@156^\circ) \)
• Correction Weight = Trial Weight \cdot \frac{|V_o|}{|I|}
• Location of CW is determined by angle $\theta$
• Should remove TW before installing CW
CASE 1: Forced Draft (FD) Fan
• 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

5 mils p-p at 1X
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

1× ODS at 29 Hz

- Bearing Housing
- Bearing Pedestal
- Concrete Foundation

Rocking Motion
ODS Animation*

*Courtesy of Clear Motion Systems
San Antonio, TX
Characteristics of Bearing Housing Vibration

- Occurred Primarily at $1 \times$ 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 $1 \times$ Vibration Readings on Bearing Housings
- Similar Vibration Amplitudes and Phase Angles at Each Bearing
- Indicates Static Imbalance of Fan Impeller, Not Flow Induced Vibration
Bode Plot
Inlet Bearing Housing
Horizontal Direction

Phase Angle

Degrees

0

-180

17

1x Vibration

6 mils p-p

4 mils p-p

2 mils p-p

Speed (RPM)

0

200

400

600

800

1000

1200

1400

1600

1800

2000

Theoretical Imbalance Only

Phase Shift

Amplified Vibration

2 mils p-p

4 mils p-p

6 mils p-p

×

Vibration

Imbalance

Only
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)

TW at Blade 4
1.2 mils p-p
9. Washer moved to fan blade 8 (240 deg)

TW at Blade 8
7.0 mils p-p
10. Find Approximate Intersection Point of Circles Representing the Three Trial Runs

11. Correction Weight = 
   \[ \frac{3.2 \text{ oz} \cdot 4.4 \text{ mils}}{3.95 \text{ mils}} = 3.6 \text{ oz} \]
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)

- **Startup**
- **Constant Load**
- **Shutdown**

- **Stable Values**
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

149.6 grams
### Summary at Base Load

#### 1X Vibration (mils p-p @ deg)

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

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

- Left graph: Orbit plot for Turbine Bearing 1 Horz / Turbine Bearing 1 Vert, labeled as "As Found".
- Right graph: Orbit plot for Turbine Bearing 2 Horz / Turbine Bearing 2 Vert, labeled as "Balanced".

[Detailed description of the graph plots is not provided in the text fragment.]
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


References (cont.)