Pump Vibration Troubleshooting

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The Centrifugal Pump (End Suction)

Open Impeller

Shrouded Impeller

Shroud

Hub

Flow

Suction

Impeller

Shaft

Discharge

Vane

Seal

Casing

Mechanical Solutions, Inc.
Engineering Analysis, Test & Technology
Impeller rotation versus the volute tongue
Potential for Cavitation from Pressure Depression at Impeller Eye

No Cavitation

Cavitation
Scope Traces of Cavitation Event

Cavitation Event

Microphone SPL 85dB

Inboard Pressure +/-150 psi pk

Outboard Pressure +/-150 psi pk

Inboard Accel on Suction Wall +/- 300 G pk

Time (msec) =>
Axially Split Case Dbl. Suction Pump

- 180 degree “drip pocket” allows bearing hsg and stuffing box to flex downward due to pressure pulsations, e.g. at 1x or vane pass.
- Double suction impeller typically equalizes thrust side-to-side, but at off-design is prone to axial shuttling.
- Suction recirc & cav have been issues.
“Donut” Pump with Balance Disk

- 180 Less robust than barrel pump, but fewer Fn problems.
- Designs where thrust disk totally replaces oil lubricated thrust bearing have encountered rub and binding problems during process upsets.
Hero or Enemy?
The Thrust Balancing “Disk”

Can have rubs at start-up or during severe process transients, may permit “axial shutting”.

Piedmont Chapter
Pacific BFI Barrel Pump

**Barrel Pump with Balance Drum and Stiff Shaft**

- 180 degree “drip pocket” allows bearing hsg Fn near vane pass
- Pedestals weaken with age, structural modes drift into 1xRPM resonance
- Diffuser “strong-back” plates can resonate at vane pass
Barrel Pump with Balance Drum, and Thin Shaft

- Rotor critical speeds sensitive to Lomakin Effect and therefore wear ring & drum clearances
- Pedestals weaken with age, structural modes drift into 1xRPM resonance
Barrel Pump with Balance Disk

- Balance disk is theoretically prone to “control system” phasing instabilities, causing axial shuttling
- Pedestals weaken with age, structural modes drift into 1xRPM resonance
Worthington WNC

Barrel Pump with Composite Balance Drum/ Disk

- Balance disk portion may have Ax shuttling
- 180 degree “drip pocket” bearing hsgs have Fn’s near vane pass
- Pedestals weaken with age, structural modes drift into 1xRPM Fn
- Develops rotating stall easily at lower flows
- Design is particularly prone to thermal differential growth top v. bottom causing rotor binding/ rubs
Axial Split Case Pumps: e.g. Bingham MSD

Axially Split Case Issues:

- Acoustics in x-over
- Top casing half “humps, allowing interstage jet leakage
- Axial shutting at off-design conditions
- High sensitivity of 1st bend mode to center bushing clearance
Pumps for Wastewater Applications

- Excitation of structural natural frequencies
  - 1X, 2X, Vane Pass
- Packing vs. mechanical seals
- Material selection
  - ‘Wear’ parts – rings, etc.
  - ‘Non-Wear’ parts – casings and impellers
- Vertical with intermediate shafting
- Submersible
  - Wet pit
  - Dry pit
Chemical & Pharmaceutical Process Pumps

- Nozzle Loads
  - Distortion
  - Stress in non-metallic pumps
- Bearings – low precision
  - Affect on rotor behavior
- Baseplates
  - Flexibility
- Seals
  - Nozzle loads effect on mechanical seals
  - Packing – over-tightening
- Product lubricated bearings in mag drive and canned motor pumps
Slurry Pumps

- Similar issues to process pumps, distortion due to nozzle loads is less
- Material selection and specification
  - Abrasive pumpage
  - Special coatings and liners
Vertical Turbine Pump (VTP) Issues:

- Above ground “Reed Freq”
- VFD as a “shaker”
- Lineshaft is “violin string”
- Shaft enclosure tube Fn’s
- Disch nozzle loose joints
- Sump vortices or odd flows
- Column piping acoustics
Vertical Pump Shaft Thrust Effects

FREQUENCY AFFECTED BY:
1) T DIRECTLY
2) $k_2 \ll k_1, k_3$

"VIOLIN STRING" EFFECTS, INFLUENCED BY AXIAL THRUST AND BEARING ECCENTRICITIES
The “Concept” of Imbalance

UNBALANCE "EXCITATION" OR "FORCING" FREQUENCY

UNBALANCE BECAUSE OF A BROKEN BLADE

\[ \omega \text{ radians per second} = 2\pi f \]
Vibration
Problem No. 1:

1x Running Speed
What Is Resonance?

"FFT" OR SIGNATURE PLOT:
VIBRATION VS. SPEED (OR VS. FREQUENCY)

VIBRATION "MAGNIFICATION FACTOR"
\[ Q = \frac{P}{S} \]

STATIONARY MOVEMENT DUE TO STATIC FLEXIBILITY

"CRITICAL SPEED" OR "NATURAL FREQUENCY"
Vibration & Phase vs. Frequency

Natural Frequency

![Graph showing vibration and phase vs. frequency](image-url)
Shaft Response to Imbalance

LOW FREQUENCY, $\omega \ll \omega_n$:

- $F(t)$
- $x(t)$
- $x/F(t)$
- $\phi(\omega)$

$F = kx$

$\text{gap} = \text{clearance} - x$

$\approx \text{clearance} - \frac{F}{k}$
Shaft Response to Imbalance

\[ F \equiv ma = m\Omega^2 x \]

\[ \text{gap} = \text{clearance} - \frac{F}{m\Omega^2} \]
Balance: Single vs. Two Plane

Dynamic Balance: Principal Axis and Journal Center Axis Coincide

\[ F = 0 \quad \text{"Static"} \]
\[ M = 0 \quad \text{"Dynamic"} \]
Angular Misalignment
Offset Misalignment
Vibration
Problem No. 2:
1x & 2x Running Speed
Observing Alignment Continuously with “Dodd Bars”
High Energy Density Equipment Alignment Limits

Offset, hub-to-hub, mils

Speed (rpm)

Excellent

Acceptable

Unacceptable

0 1800 3600 10000 20000
Flow Through Elbows

(Courtesy Cheng Fluid Systems, Inc.)
Stationary Piping Load Sources

- UNRESTRAINED EXPANSION JOINT (LIKE ROCKET NOZZLE, F=P*A)
- "BOURDON TUBE" STRAIGHTENING
- THERMAL GROWTH / MISMATCH
Piping Acoustic Natural Frequencies

\[ a = \text{Speed of Sound in Fluid} \]

\[ f_{nA_{1/4}} = \frac{a}{4L} \]

\[ f_{nA_{1/2}} = \frac{a}{2L} \]

For Natural Frequency Number “i”:

\[ f_{nA_i} = \frac{(2i - 1) a}{4L} \] for quarter wave (closed one end)

or \[ \frac{ia}{2L} \] for half wave (both ends open or both ends closed)
“Lomakin Effect” in Centrifugal Pumps
Concept of the “Wet Critical Speed”
Lomakin Effect

\[ P_{\text{static}} = P_{\text{stagnation}} - \frac{\rho V^2}{2g_c} \]

**Critical Factors:**
- CLEARANCE
- \( \Delta P \)
- GROOVING

\[ K_L = \frac{\Delta F_L}{\Delta \delta} \]
Lomakin Result:
Critical Speeds Shift Up
Approximate Calculation of Lomakin Stiffness

\[ k_{xx} \approx \frac{RL \Delta P}{c} \]

\[ K_{xx} \approx \frac{\pi \sigma}{(1+2\sigma)^2} \approx 0.04 \quad (L \approx 2R) \]

\[ K_{xx} \approx 0.40 \quad (L \ll 2R) \]

where \( \sigma = \frac{\lambda L}{c} \)

\( R = \) Radius
\( L = \) Length
\( C = \) Radial clearance
\( \Delta P = \) Pressure drop
\( \lambda = \) fric factor
Some Key Issues in Lomakin Effect Strength:

- Grooving, Surface Roughness
- Inlet Conditions
  (Swirl, Corners, Deposits, Cavitation)
- Available Energy
  (i.e. Total Pressure at Inlet vs. Outlet)
- Alignment, Eccentricity
- Frequency Content/Orbit Shape
- Wear & Erosion
Wild Cards: Swirl & Grooving
Impeller Radial Support Forces: Sulzer/ EPRI Tests

Normalization:

\[ k^* = \frac{1}{\pi r_2^2 B_2 \rho \omega^2} k \]
\[ c^* = \frac{1}{\pi r_2^2 B_2 \rho \omega} c \]
\[ m^* = \frac{1}{\pi r_2^2 B_2 \rho} m \]

\[ r_2 = \text{Impeller Radius}, \quad B_2 = \text{Impeller exit width} \]
\[ \rho = \text{Density}, \quad \omega = \text{rotational angular frequency} \]
\[ * = \text{Normalized quantities (dimensionless)} \]

For circular orbit of whirl frequency \( \Omega \):

- Radial (in dir. of displ.)
  \[ F_r^* = -k^* c_c^* \left( \frac{\Omega}{\omega} \right) + m^* \left( \frac{\Omega}{\omega} \right)^2 \]
- Tangential (in dir. of rot.)
  \[ F_t^* = -k^* c^* \left( \frac{\Omega}{\omega} \right) + m^* \left( \frac{\Omega}{\omega} \right)^2 \]
### Some Typical Exciting Frequencies

<table>
<thead>
<tr>
<th>FREQUENCY</th>
<th>SOURCE</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05 - 0.35 x</td>
<td>DIFFUSER STALL</td>
</tr>
<tr>
<td>0.43 - 0.48 x</td>
<td>INSTABILITY</td>
</tr>
<tr>
<td>0.500 x</td>
<td>RUBBING</td>
</tr>
<tr>
<td>0.65 - 0.95 x</td>
<td>IMPELLER STALL</td>
</tr>
<tr>
<td>1 x</td>
<td>IMBALANCE</td>
</tr>
<tr>
<td>1 x +2 x</td>
<td>MISALIGNMENT</td>
</tr>
<tr>
<td>#Vanes x</td>
<td>VANE/VOLUTE GAP</td>
</tr>
<tr>
<td>#Blades x</td>
<td>BLADE/DIFFUSER GAP</td>
</tr>
</tbody>
</table>
Rolling Element Bearing Parameters

- Outer Race
- Inner Race
- Shaft
- Cage
- Ball or Roller, $N_b$ of them, of Diameter $D_b$
- Pitch Diameter $D_p$
- Radial Load
- Axial Load

\[
\gamma = \frac{D_b}{D_p} \times \cos \theta
\]

(Usually about 0.2)
Rolling Element Bearing Frequencies

\[
\begin{align*}
FTF &= \frac{N}{2} (1 - \gamma) \quad \text{Fundamental Train Frequency} \\
BSF &= \frac{D_p}{D_b} \times \frac{N}{2} (1 - \gamma^2) \quad \text{Ball Spin Frequency} \\
BPFO &= N_b \times \frac{N}{2} (1 - \gamma) \quad \text{Ball Pass Frequency Outer Race} \\
BPFI &= N_b \times \frac{N}{2} (1 + \gamma) \quad \text{Ball Pass Frequency Inner Race}
\end{align*}
\]
Lateral Excitation Forces (Sulzer)

Normalization: \[ F_R^* = \frac{F_R}{\rho g D_2^2 B_2} \]

- \( D_2 \) = Impeller diameter,
- \( B_2 \) = Impeller exit width
- \( \rho \) = Density, \( g \) = Gravity, \( H \) = Head
- \( F_R^* \) = Normalized radial force, nondimensional

<table>
<thead>
<tr>
<th>Static</th>
<th>180° Double Volute</th>
<th>Single Volute</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction impeller (suction asymmetry)</td>
<td>Diffuser</td>
<td></td>
</tr>
<tr>
<td></td>
<td>.01 - .08 (.04)</td>
<td>.02 - .15 (.05)</td>
</tr>
<tr>
<td>Normal impeller (no suction asymmetry)</td>
<td>Diffuser</td>
<td></td>
</tr>
<tr>
<td></td>
<td>.01 - .06 (.03)</td>
<td>.01 - .10 (.04)</td>
</tr>
<tr>
<td>Broad Band up to 1.2 times rotational frequency</td>
<td>Diffuser</td>
<td></td>
</tr>
<tr>
<td></td>
<td>.01 - .15 (.03)</td>
<td>.01 - .12 (.03)</td>
</tr>
<tr>
<td>Hydraulic unbalance, at rotational frequency</td>
<td>Diffuser</td>
<td></td>
</tr>
<tr>
<td></td>
<td>.005 - .03 (.02)</td>
<td>.005 - .10 (.03)</td>
</tr>
</tbody>
</table>

Ranges for \( F_R \) for \( Q=25\% \) to 125\% of Design Point
Values in brackets: typical for Design Point
Vibration
Problem No. 3:
High Vane Pass

Common Causes:
- a) "GAP B" TOO TIGHT
- b) DISCHARGE RECIRCULATION
- c) FLAT OR DAMAGED VOLUTE TONGUES
- d) INTERNAL RESONANCE OF DIFFUSER WALLS OR VANES

Harmful Effects:
- a) FATIGUE IN INSTRUMENTATION WIRE CONNECTIONS OR DRAIN PIPE CONNECTIONS
- b) IF INTERNAL RESONANCE IS THE CAUSE, FATIGUE CRACKING OF THE RESONATING PART
Vibration
Problem No. 4: High Harmonics of Run Speed

**COMMON CAUSES:**

- a) Scratch in the shaft or chrome plate thickness variations at Bently probe
- b) Rubbing
- c) Very loose "rattling" internal component, such as impeller

**HARMFUL EFFECTS:**

- a) From cause (a), none (except stomach problems!)
- b) Premature wear from (b)
- c) Progressively looser fits and worsening situation from (c)
Marked-up Copy of Pump Curves
Flow Rate and the “Angle-of-Attack”

- Blade Tip Velocity
- Relative Velocity of Incoming Flow
- Through-Flow Velocity
- Motion of Blade
Vane Stalling at Low Flows
Example of Stalled Blade
Onset of Internal Recirculation

Discharge Recirculation

Suction Recirculation Flow
Vibration & Pulsation vs. Flowrate
An Unexpected Hydraulic Problem:

Rotating Stall
Four Stage High Speed Boiler Feed Pump

Four Stage Pump Configuration
Rotating Stall Pulsation Spectrum
Stall Vibration vs. Speed
Vibration
Problem No. 5:
Subsynchronous (below 1x)
ROLE OF SUCTION RECIRCULATION REDUCED OR ELIMINATED BY STOP VANES

ZONE OF STRONG INFLUENCE BY STOP VANES

STOP VANES (14 OF THEM)

EDDY FORMS AS WITHOUT STOP VANES

AS EDDY EXTENDS FROM INLET, IT IS SHEARED & DESTROYED

RESIDUAL FLOW FIELD FROM DESTROYED EDDY PREVENTED FROM ROTATING
Rotor Dynamics

- Critical Speeds & Mode Shapes
- Forced Response
- Stability
Rotordynamics Is Best Evaluated by Computer
Typical Rotor Vibr. Response vs. Speed
Exhibits Several Natural Frequencies
Avoiding Resonance w/ Campbell Diagram

![Campbell Diagram](image_url)

**FREQ = Number of Impeller Vanes * Running Speed**

**SPEED RANGE**

- fn₁
- fn₂
- fn₃

**NOTE:** fn’s are natural frequencies

= Zones of Potential Resonance
Rotordynamic Critical Speed Map

NOTE: This map is initially drawn from calculations where $k_{xx} = k_{yy}$ and $k_{xy} = k_{yx} = c_{xx} = c_{yy} = c_{xy} = c_{yx} = 0$
Forward vs. Backward Precession

Forward:

Backward:
**Typical Torsional Critical Speeds and Worst Case Excitations**

- **VALUES @ MIN. FLOW (BEP IS 2x - 5x LOWER)**
- **VALUES MAY VARY BY ~ +/- 0.05**
- **SOME EXCIT. @ VANE PASS x2, x3, ..., xi (~ \( \frac{0.05}{1} \))**
- **VFD’s: LINE FREQ, 2x LINE FREQ, 6x/12x/18x \( N_{MOTOR} x^2/POLES \)**
Typical Vibration Problems

- Imbalance at 1xN (40% chance)
- Misalignment at 2xN and 1xN (40% chance)
- Natural Frequency Resonance (10% chance)
- Everything Else (10% chance)
  (Motor electrical problems, pump or system hydraulic problems, foundation problems, etc.)
Instrumentation Options

Turbomachine Well Instrumented According to Current Practice
Converting Vibration vs. Time to Vibr vs. f with “FFT”: Fast Fourier Transform
Meaning of RMS vs. Peak Vibration

\[ V_{p\text{ derived}} = \sqrt{2} \times V \text{ RMS} \]

\[ V_{p\text{ true}} \geq V_{p\text{ derived}} \]
Vibration Measure Numbers

- Peak-to-Peak Displacement $X$
- Peak Velocity $V$
- Peak Acceleration $a$

Typical Spec @3600 RPM:
- 2.5 mils
- 0.25 ips
- 0.25 G's
Vibration Measurement Format

1) “UNFILTERED” or “TOTAL”:

\[ X_{\text{TOTAL}} = \Delta x_{pp} \approx \frac{2}{0.707} \sqrt{X_n^2 + X_{2N}^2 + \ldots + X_{5N}^2} \]

2) “FILTERED”:

\[ X_{\text{RMS}} \]

3) “Spectrum” or “Signature”:
All Spectral Information,
Unfiltered Total and
Filtered Values & f’s.
Housing vs. Shaft Vibration: Which Should Be Measured?
Natural Frequency

"Artificial Excitation" (Impact)

"Natural Frequency"

\[ f_n = \frac{1}{\text{period between bounces}} \]

Vibration "Mode Shape"
Approximating Natural Frequency

\[ f_{n_1} = \frac{60}{2\pi} \sqrt{\frac{k_{\text{effective}}}{m_{\text{effective}}}} \]
Vibration “Mode Shapes”

**Orders of Excitation**

# of Excitation Pulses per Rev.

**Resonance**

When Order of Excit * Rotor Speed = Mode Freq.

\[ \frac{\omega}{\omega_{nd}} = 1.0 \]
Finding Nat. Freq. Resonances with a “Waterfall” Plot of Stacked FFT’s from Test
Approximate Identification of Natural Frequencies

→ Plot vibration as dB
→ Plot frequency as linear
A Modal Field Test In-Progress on an Operating Pump
Short Force Application
Excites Wide Frequency Band
Principle of Time Averaging
Rotor Impact Testing
Case History
14 Stage Axially Split Case Pump
Time Averaging Field Example: Watch the Harmonics Disappear

In the End, Natural Frequencies Are Clear
Critical Speeds of 14 Stage Pump

Rotordynamic Result
First Critical Speed, Dbl. Suction SS Pump
Sources of Damaging Forces in Centrifugal Pumps

- Inlet or Discharge Pressure Pulsations
- "Blade Pass" Pressure Pulsations Due to Blade/Vane Interactions, or Possible Recirculation & Stall
- Possible Oil Film Instabilities
- Torsional Pulsations
- Coupling Imbalance
- Suction Pressure Pulsations Due to Inlet Recirculation or Rotating Stall
- Front vs. Rear Shroud Cavity Pulsations
- Imbalance or Skew-Mounting of Large Diameter Rotor Components
- Misalignment Due to Pedestal Distortion or Piping Nozzle Loads
- Swirl or Pulsations at Thrust Balance Drum
- Seismic Excitation

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Typical Failures in Centrifugal Pumps

- Cavitation, Erosion or Fatigue Damage of Stator Vanes
- Rubbing Wear of Close Clearance Seals, Multiple Locations
- Torsional or Axial Fatigue of Shaft at Threaded Connections
- Torsional or Bending Fatigue of Shaft at Stress Concentrations and Keyways
- Fatigue or Fretting Wear in Couplings
- Radial Bearing Fatigue, Scuffing, Gradual Wear, or Thermal Seizure
- Disk or Impeller Bore Fatigue
- Rubbing Wear of Bearing Seal (2 Places)
Risk of Accepting Bad vs. Risk of Rejecting Good
Stuffing Box & Bearing Skew Case Histories
Cartridge Pump
Seal Rub
Example
Vibration Problem #6: Repeating Impacts

POSSIBLE CAUSES:

a) RUB CONSISTENTLY OVER PART OF ORBIT
b) PINCHED OR MISALIGNED SEAL
c) JOURNAL BEARING WORN TO OVAL SHAPE
Double Suction Pump
“Breathing” Tilts Stuffing Boxes
FEA Model of Twisting Pump Bearing Housing
Vertical Pump Failure
Case Histories
Typical Structural Vibrations of VTP’s
Six Vertical Pumps on Platform, with Column Exposed to Air
Vertical Pump Column Vibration with & without Water in Column

With Water

Without Water
Multistage Pump
Inboard Bearing Chronic Failure
Case History
Pump with Inboard Bearing Failures
Shaft Orbits of Problem Pump

N = 5300 RPM
Vibration Spectra for Problem Pump
Impact Test Results Showing Shaft Critical Speed at 5280 rpm
“What If” Analysis for Problem Pump
Bearing Groove Change from 0.040 in. Deep to 0.010 in. Deep

Vibration decreased a factor of ten
Vibration Spectrum After Bearing Fix
Use of Dynamic Absorbers in Constant Speed Equipment to Decrease Vibration Motion
Dynamic Absorber Design - Vertical Pump/Motor

1. Weld or firmly bolt to rigid part of motor frame.
2. Discharge head.
3. 50 kg plate.
4. 25mm dia rod with threaded section about 150mm long.
5. Locknut.
Effect of Dynamic Absorber on Vibration
Gear Box Noise & Wear Problem
Conclusions

1. There’s More to Pump and System Vibrations than You Might Expect

2. Keys to Success: Knowledge, Experience, the Right Tools

3. Good Condition-Based Methods & Instrumentation Are Getting Better