Piedmont Chapter

Pump Vibration Troubleshooting

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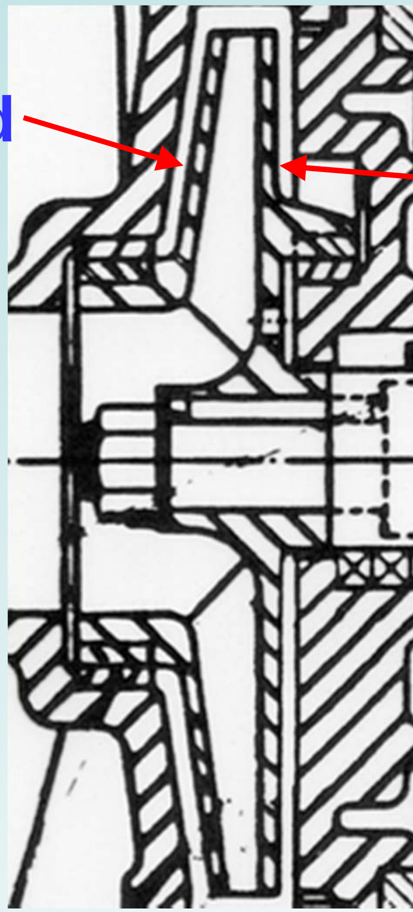
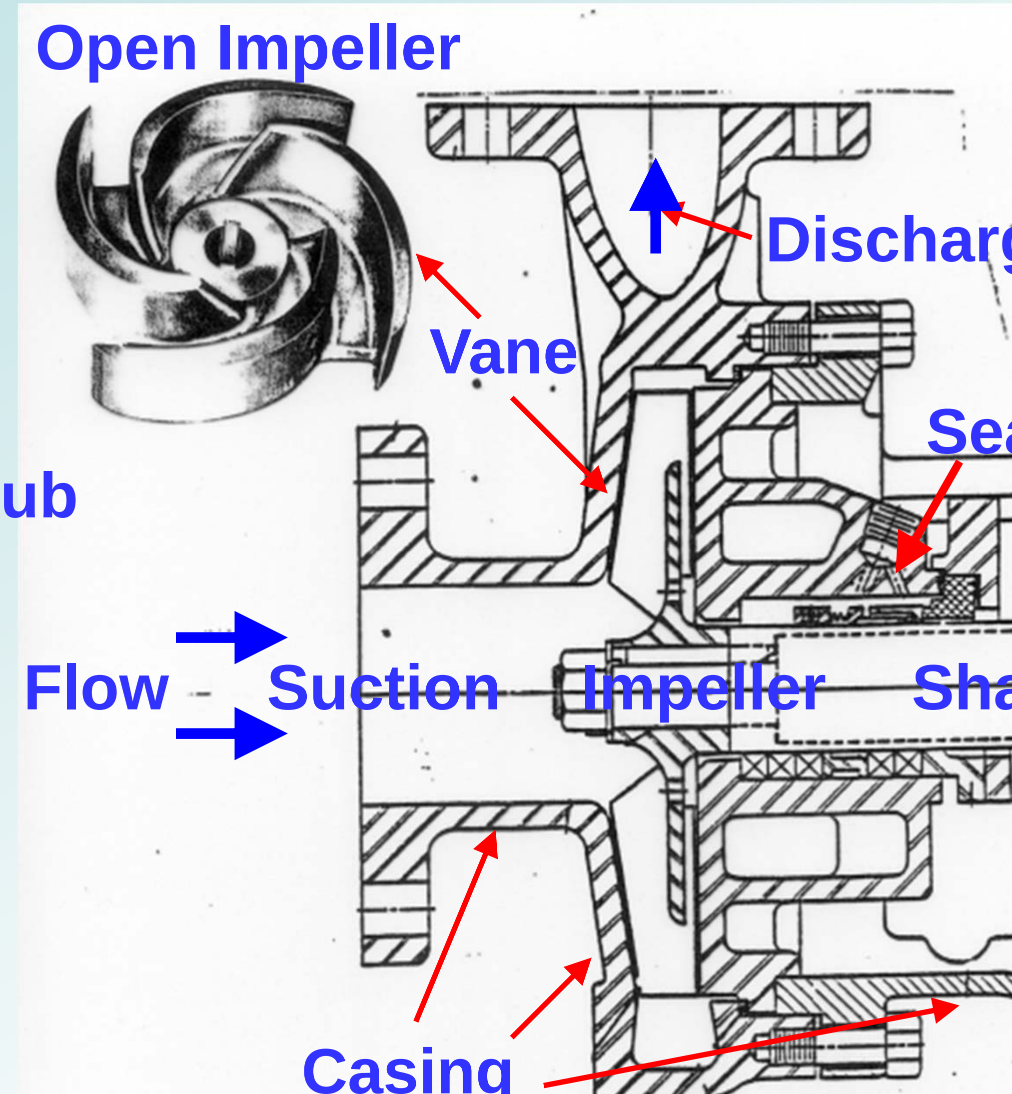
The Centrifugal Pump (End Suction)

Shrouded Impeller





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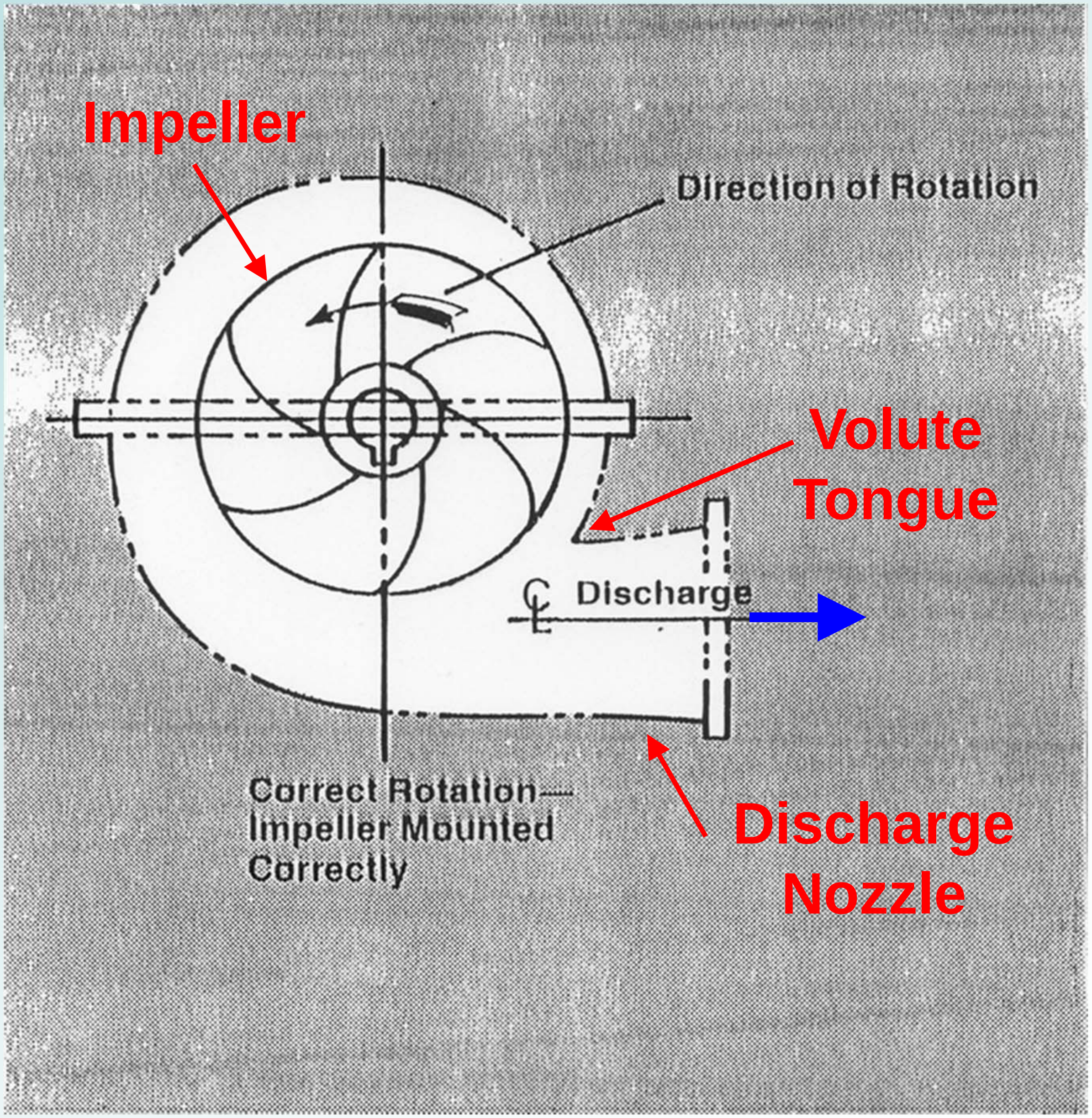
Impeller rotation versus the volute tongue



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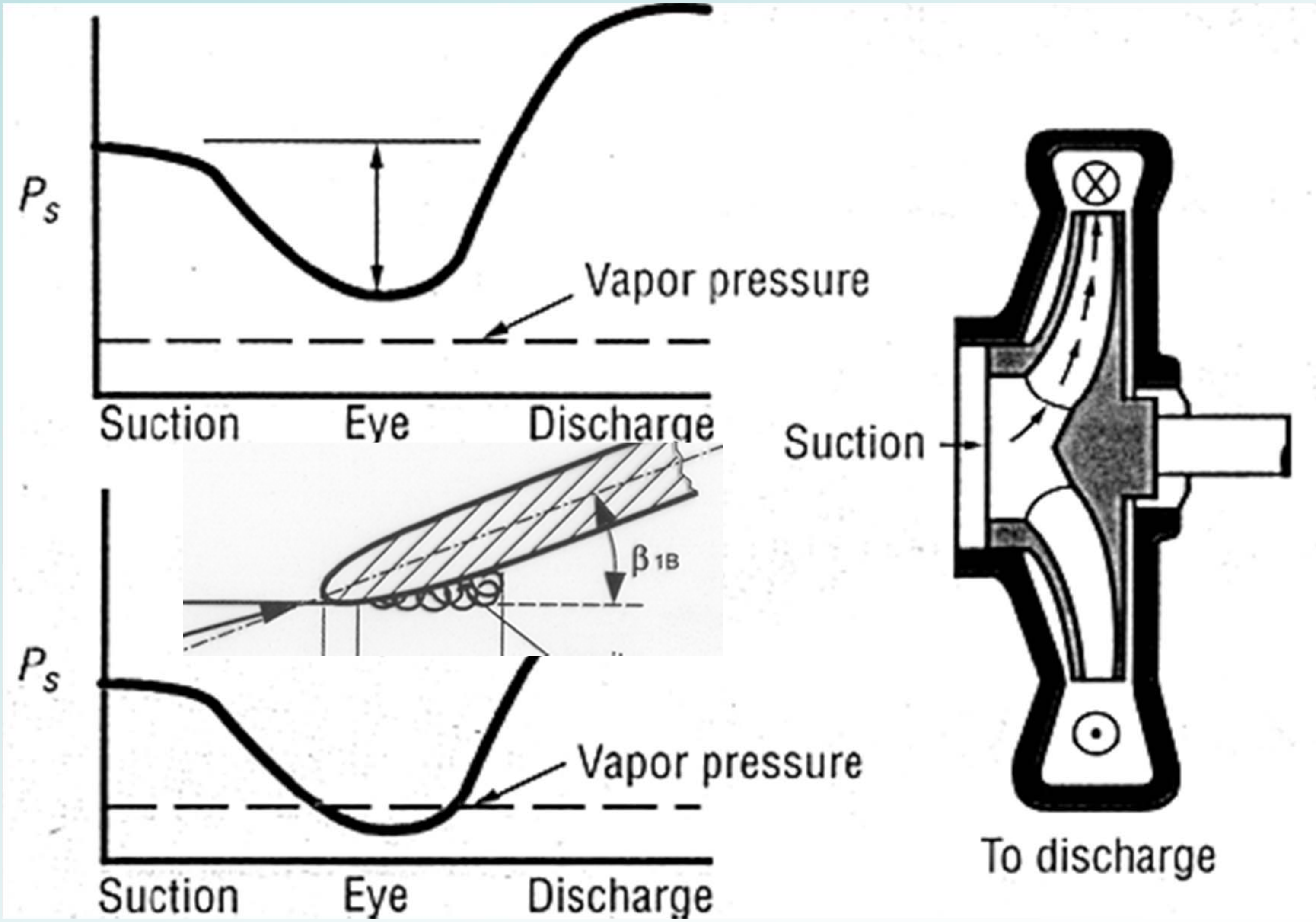
Potential for Cavitation from Pressure Depression at Impeller Eye

No Cavitation

Cavitation



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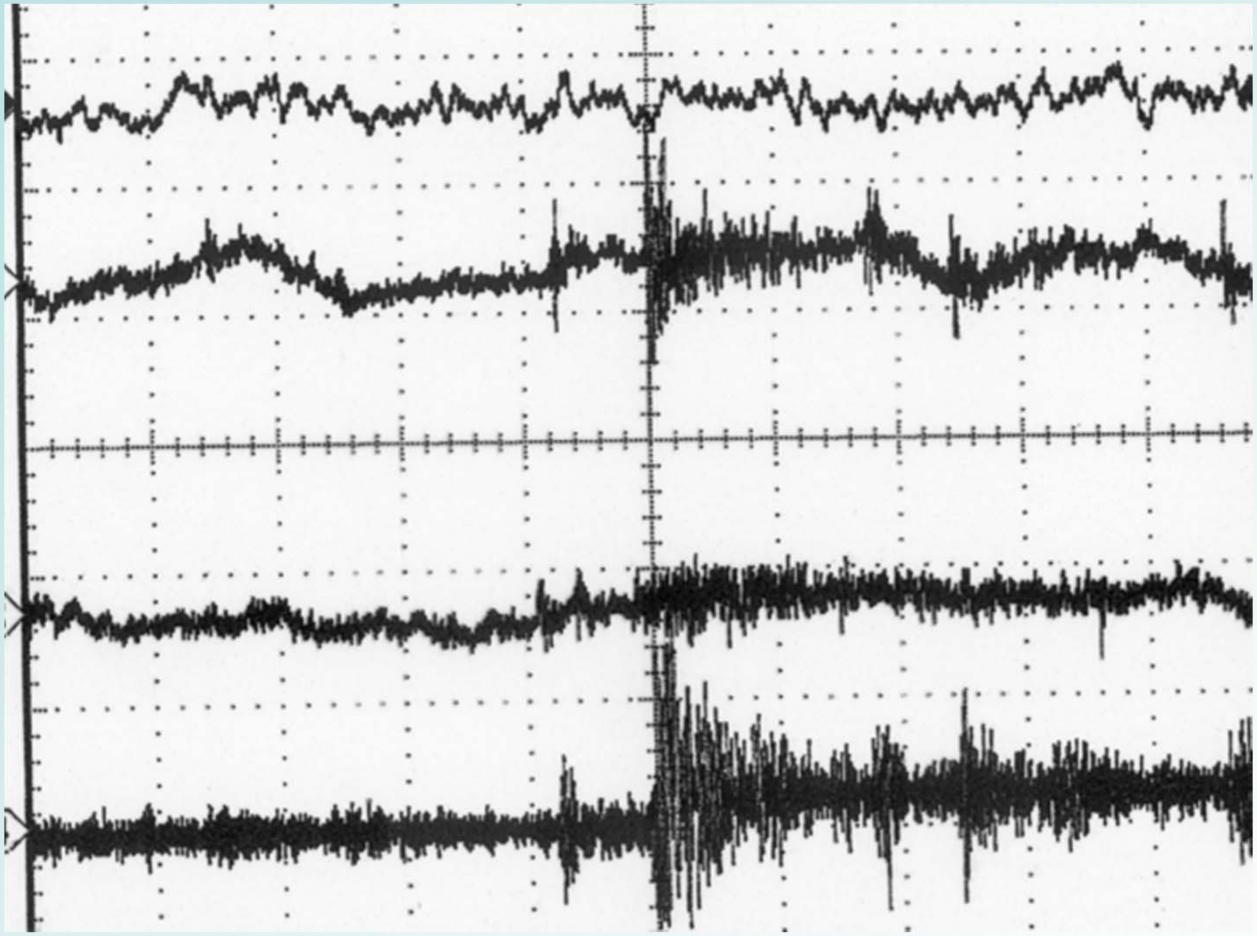
Scope Traces of Cavitation Event

Cavitation Event

v



Microphone SPL 85dB

Inboard Pressure
+/-150 psi pk

Outboard Pressure
+/-150 psi pk

Inboard Accel on
Suction Wall +/- 300 G pk

Time (msec) =>

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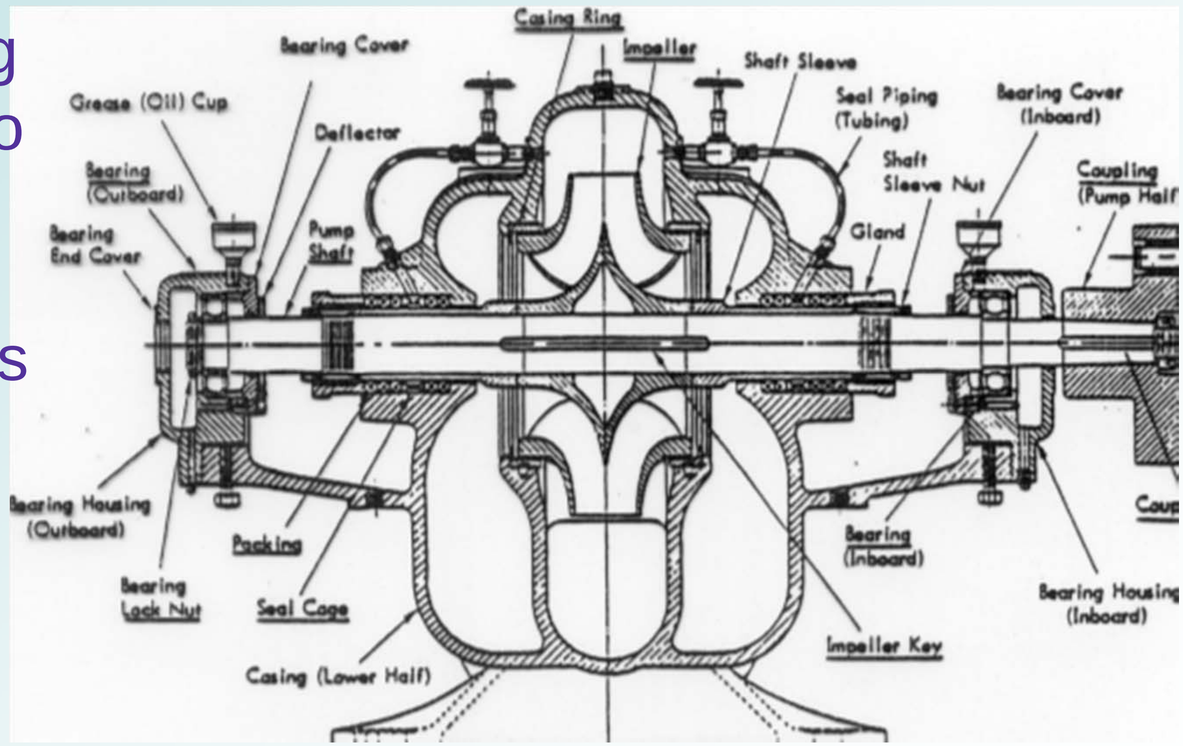
Axially Split Case Dbl. Suction Pump

- 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

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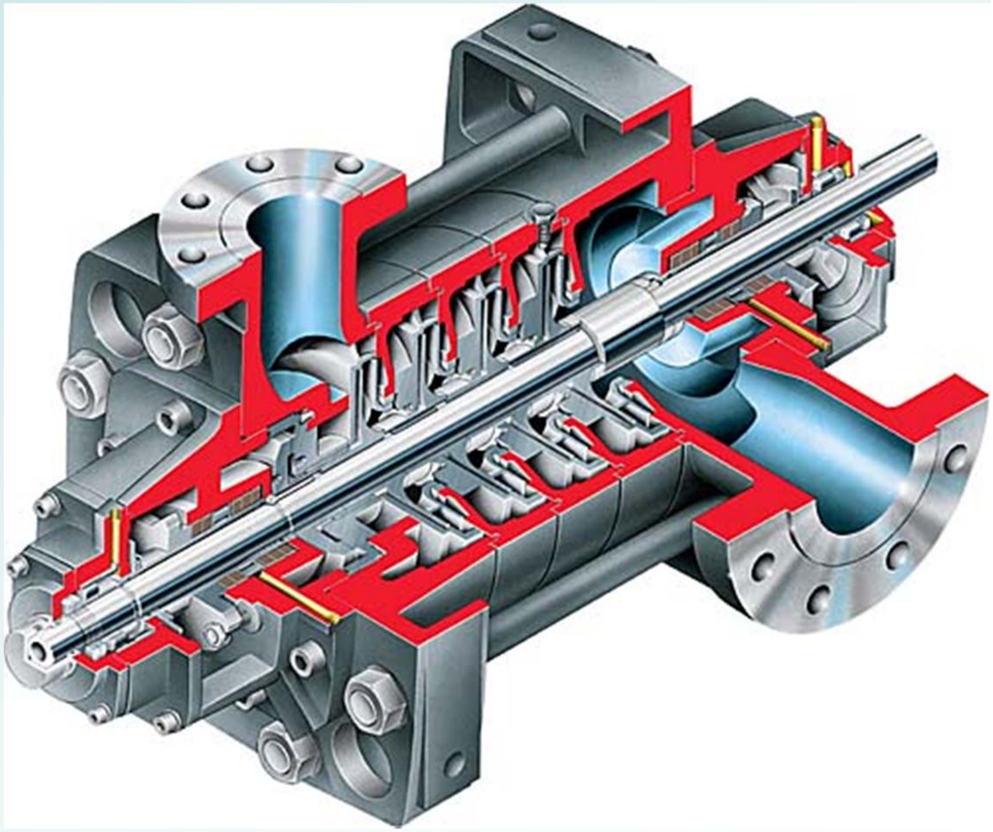
Boiler Feedwater Ring-Section Pumps

“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"

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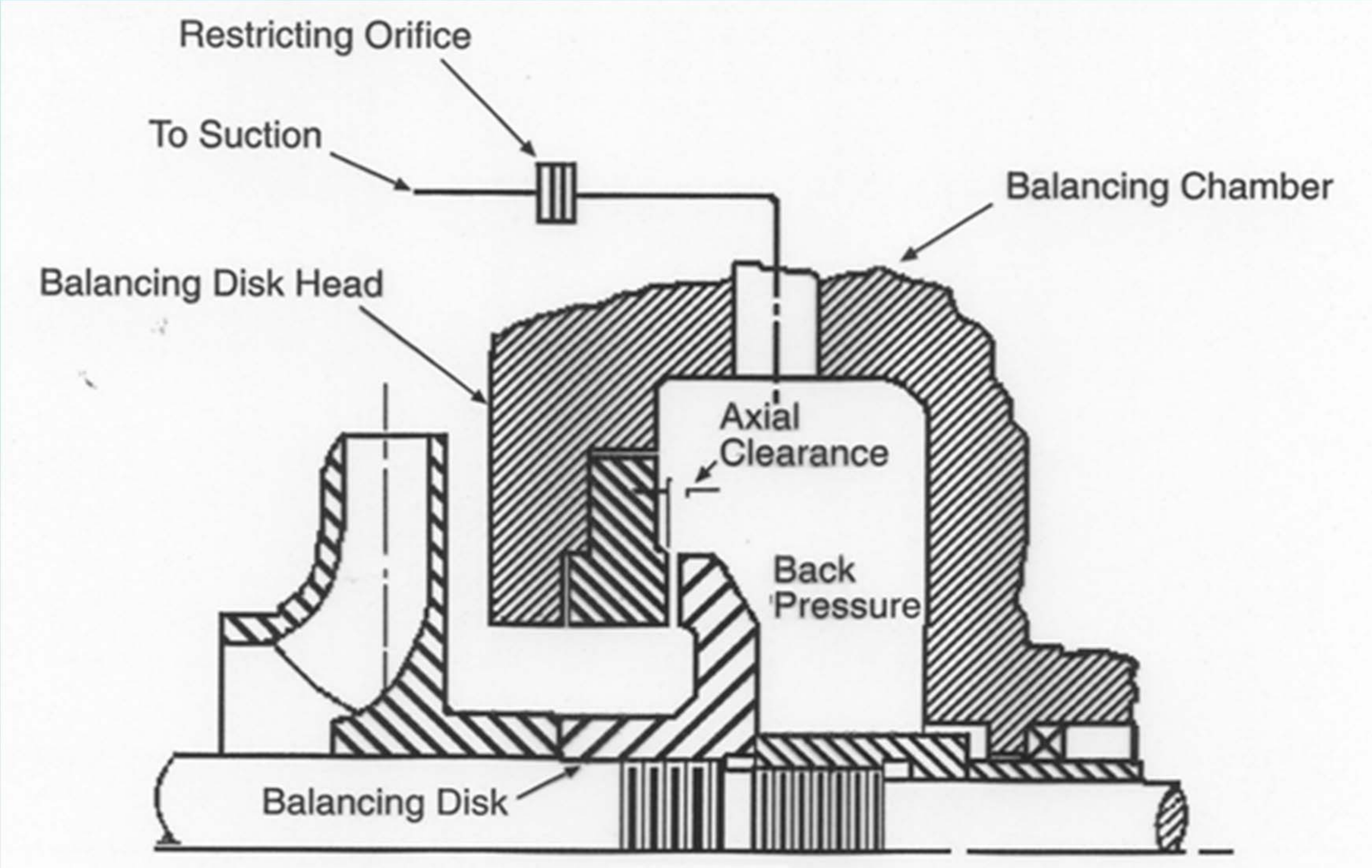

Can have rubs at start-up or during severe process transients, may permit "axial shutting".

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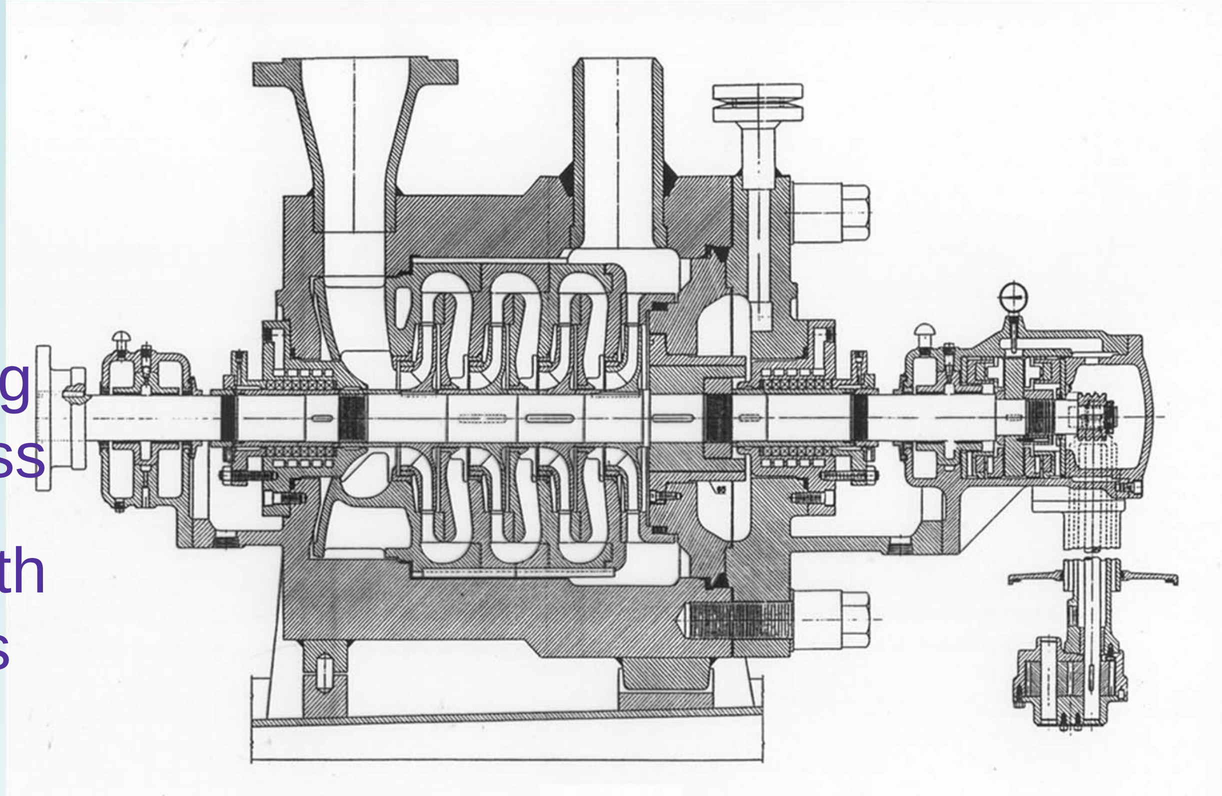
Pacific BFI
Barrel Pump

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

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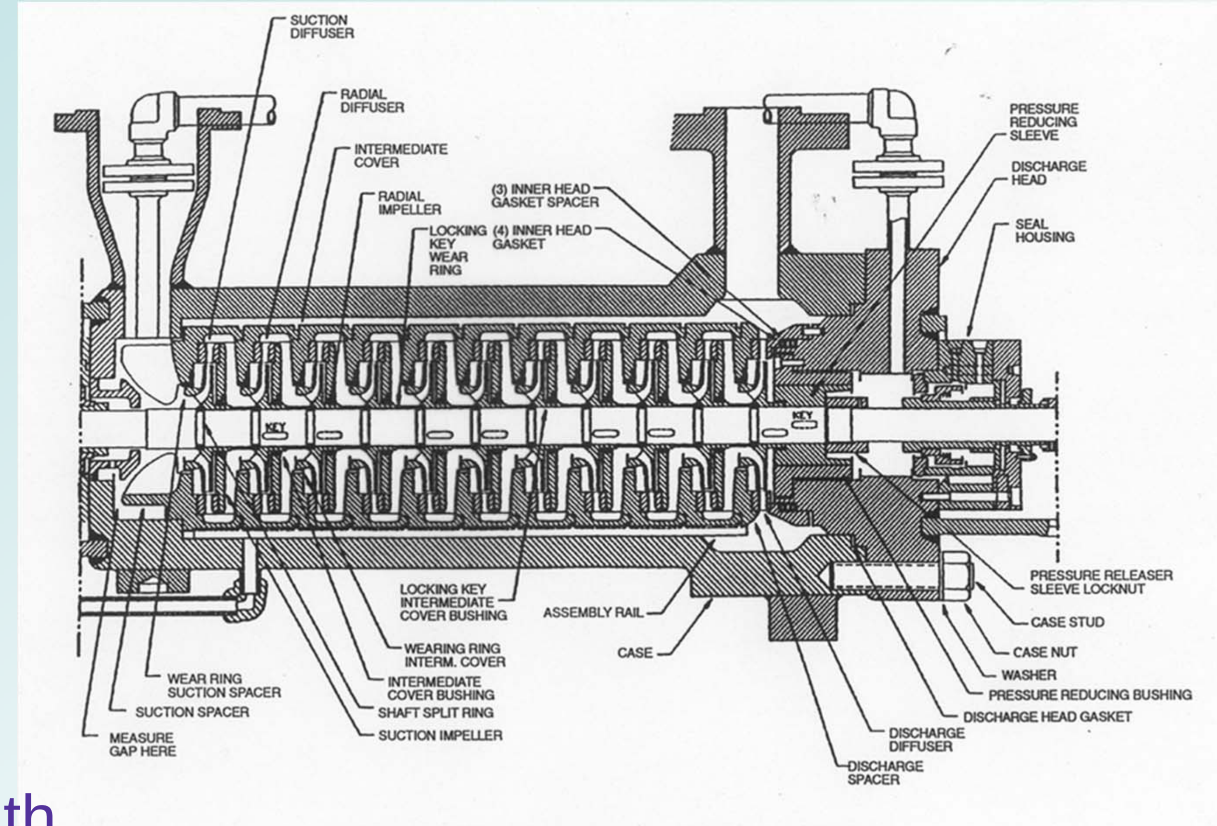
Pacific RLIJ



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

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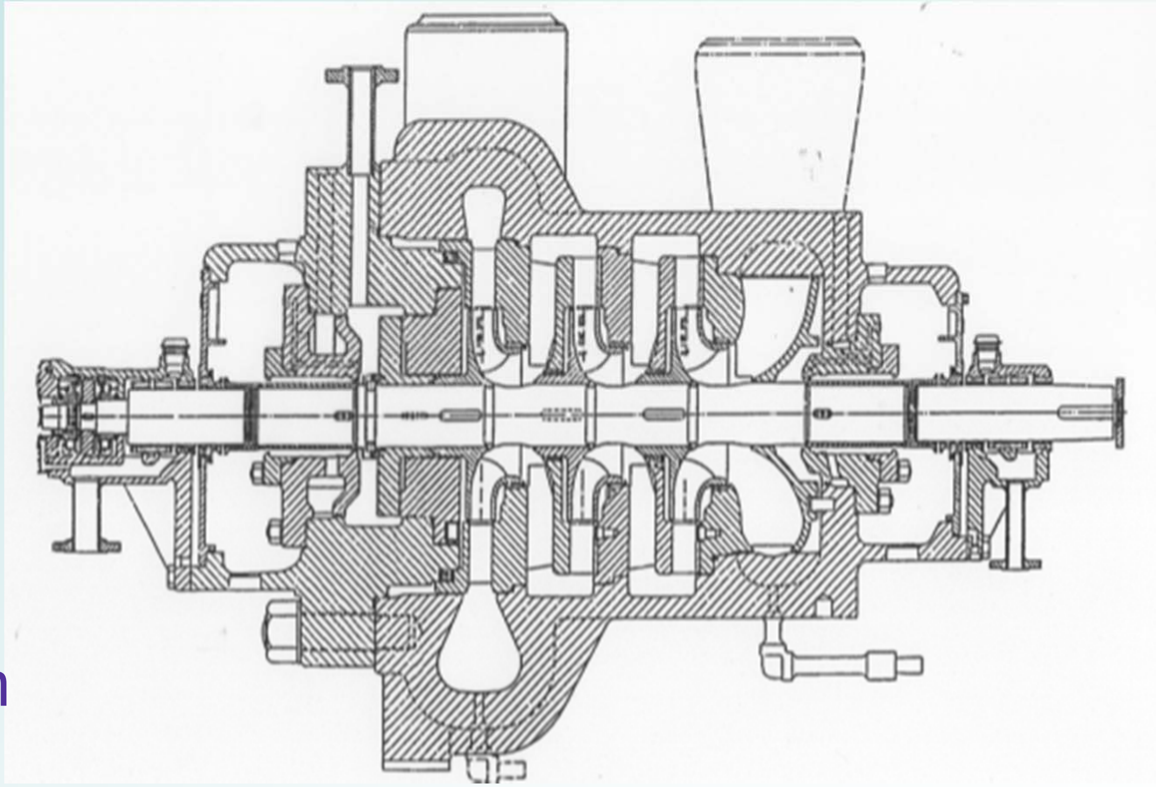
Ingersoll-Rand C

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



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Worthington WNC

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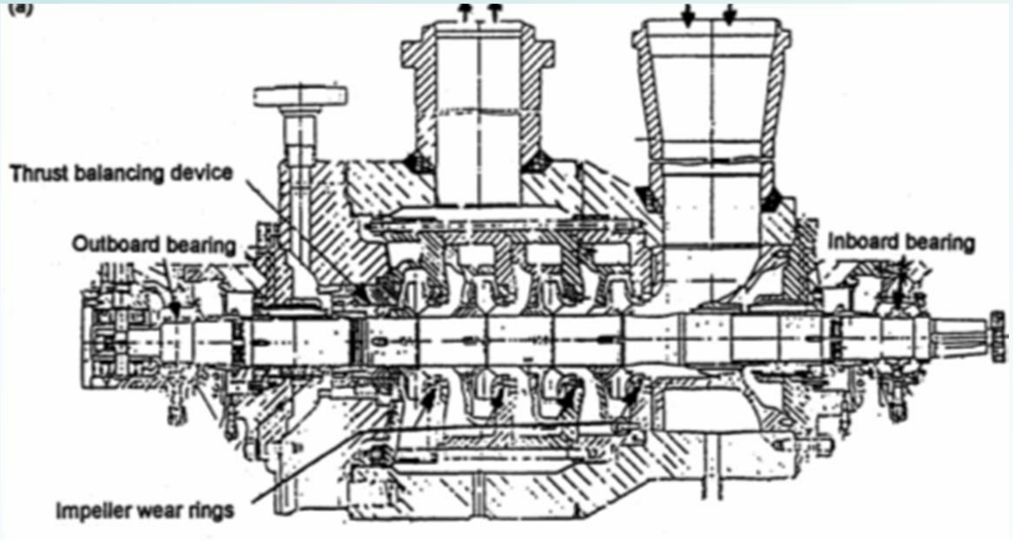
Barrel Pump with Composite Balance Drum/ Disk

- 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

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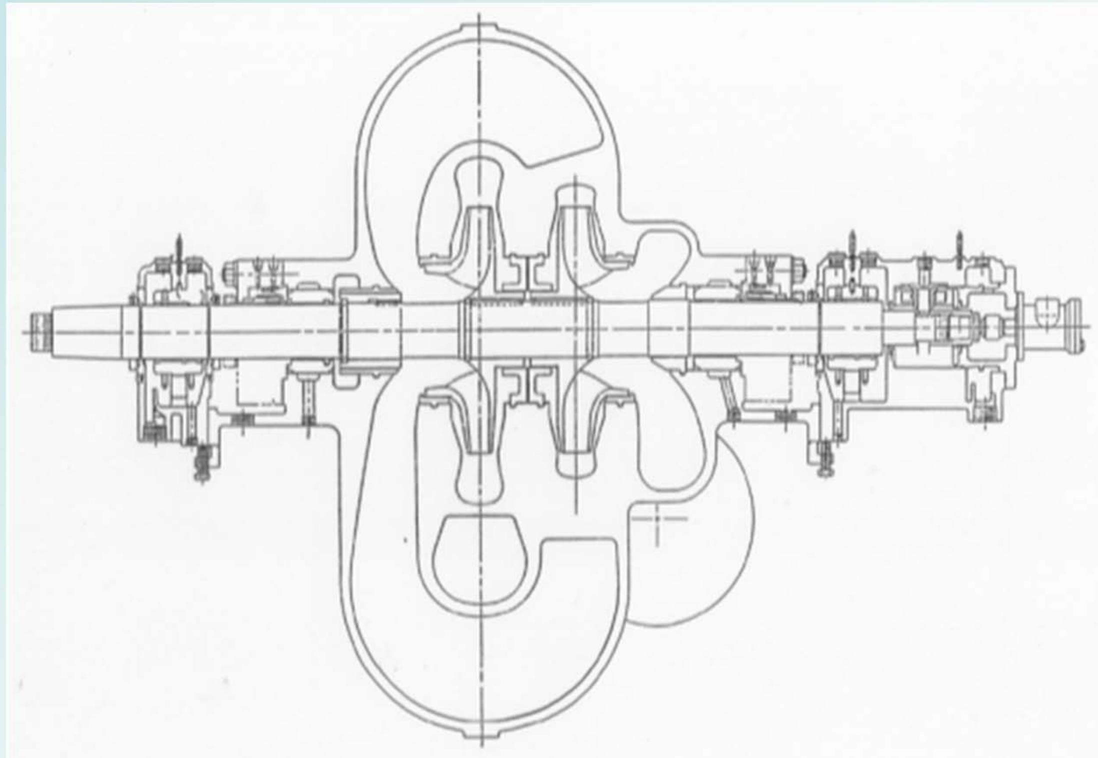
Axial Split Case Pumps: e.g. Bingham MSD



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 1^{st} bend mode to center bushing clearance

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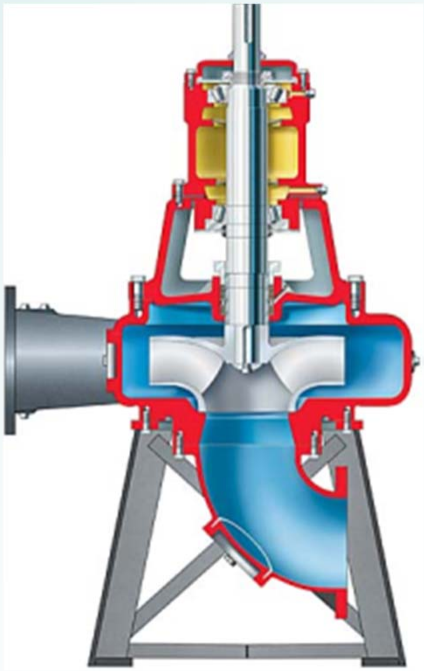
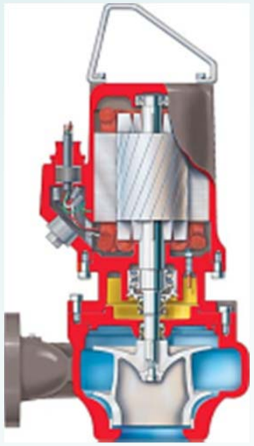
Pumps for Wastewater Applications

- 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





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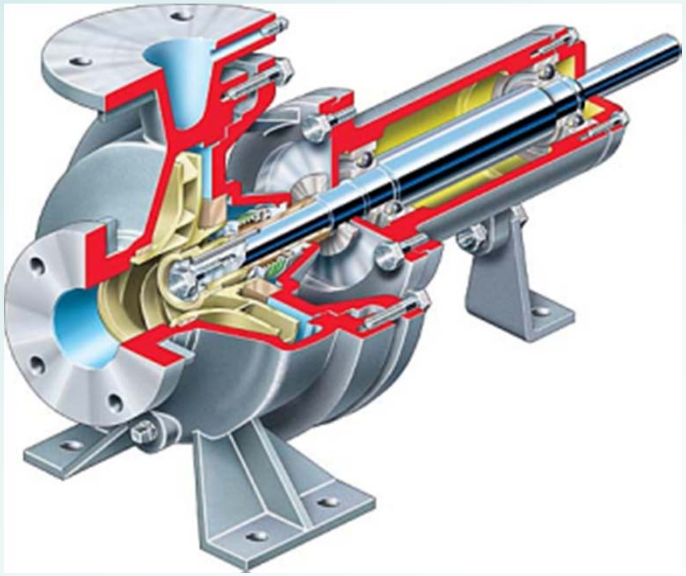
Chemical & Pharmaceutical Process Pumps

- 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



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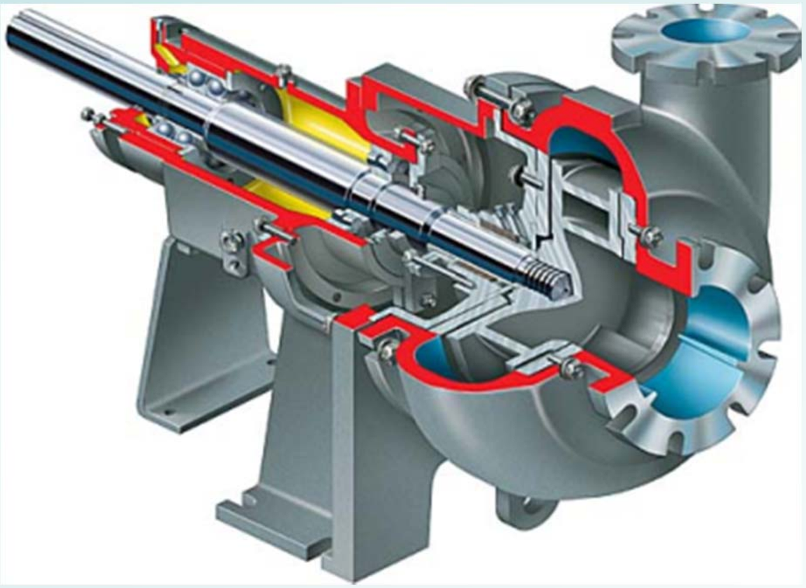
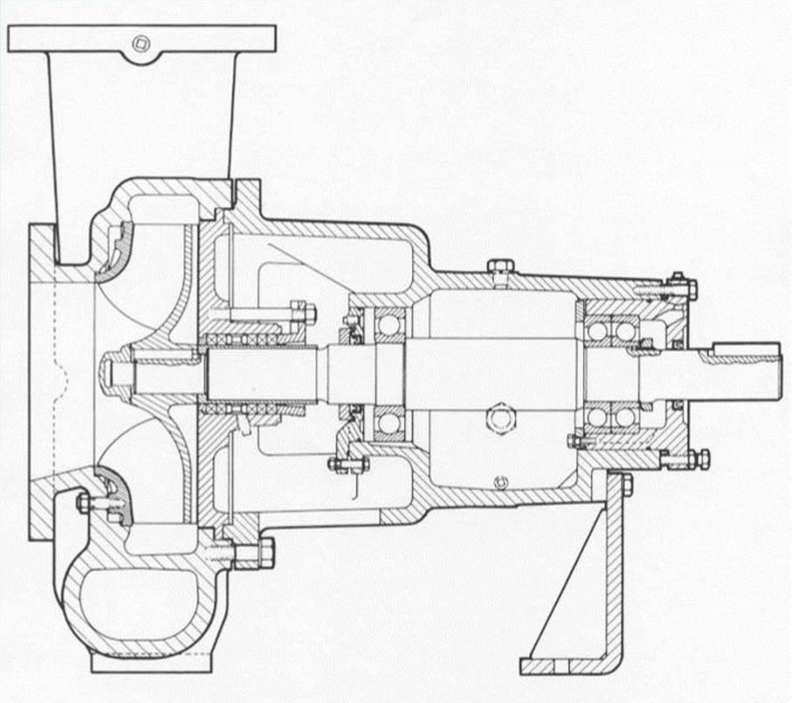
Slurry Pumps

- Similar issues to process pumps, distortion due to nozzle loads is less
- Material selection and specification
  - Abrasive pumpage
  - Special coatings and liners





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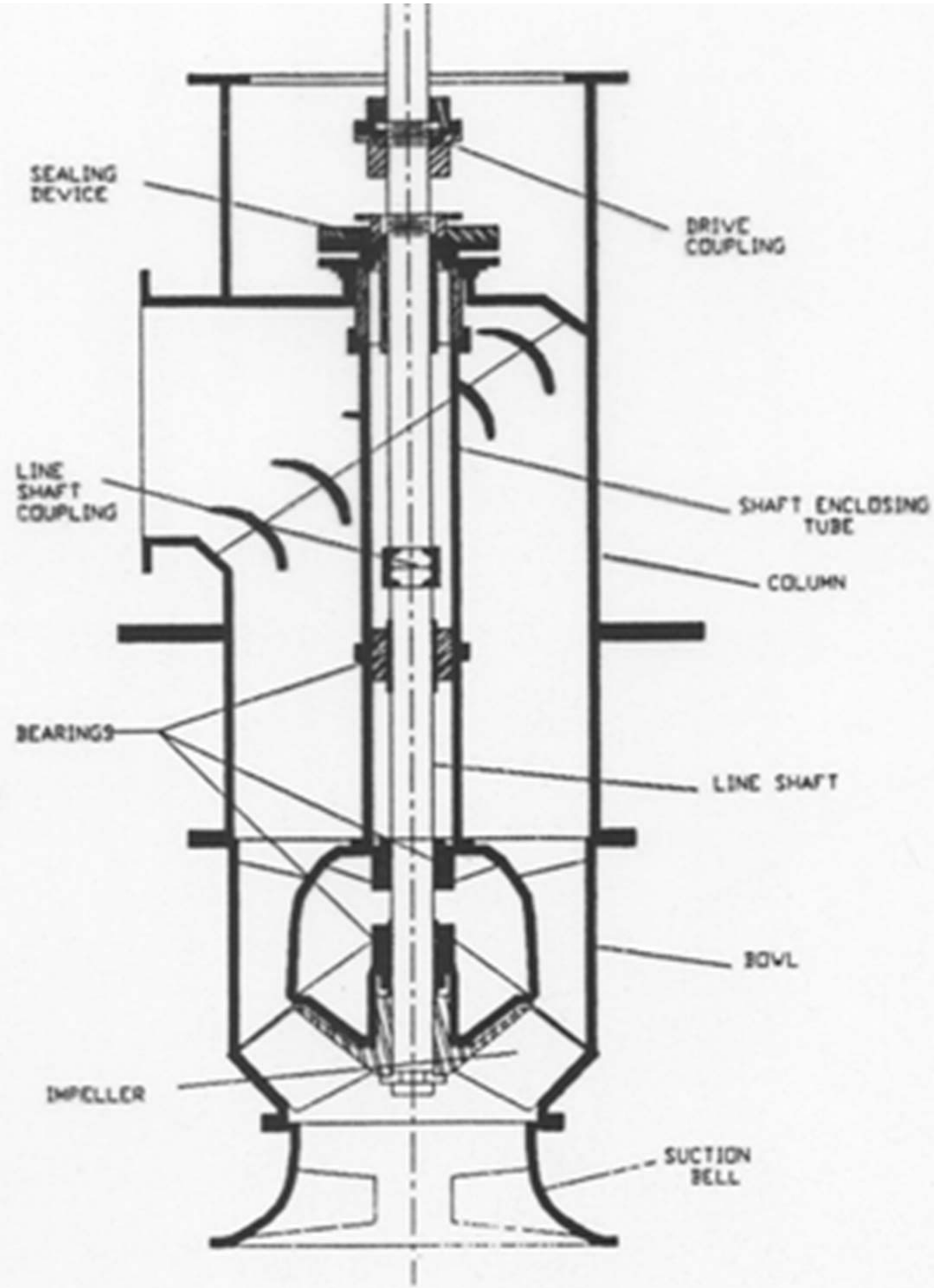
Vertical Turbine Pump (VTP) Issues:

- 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



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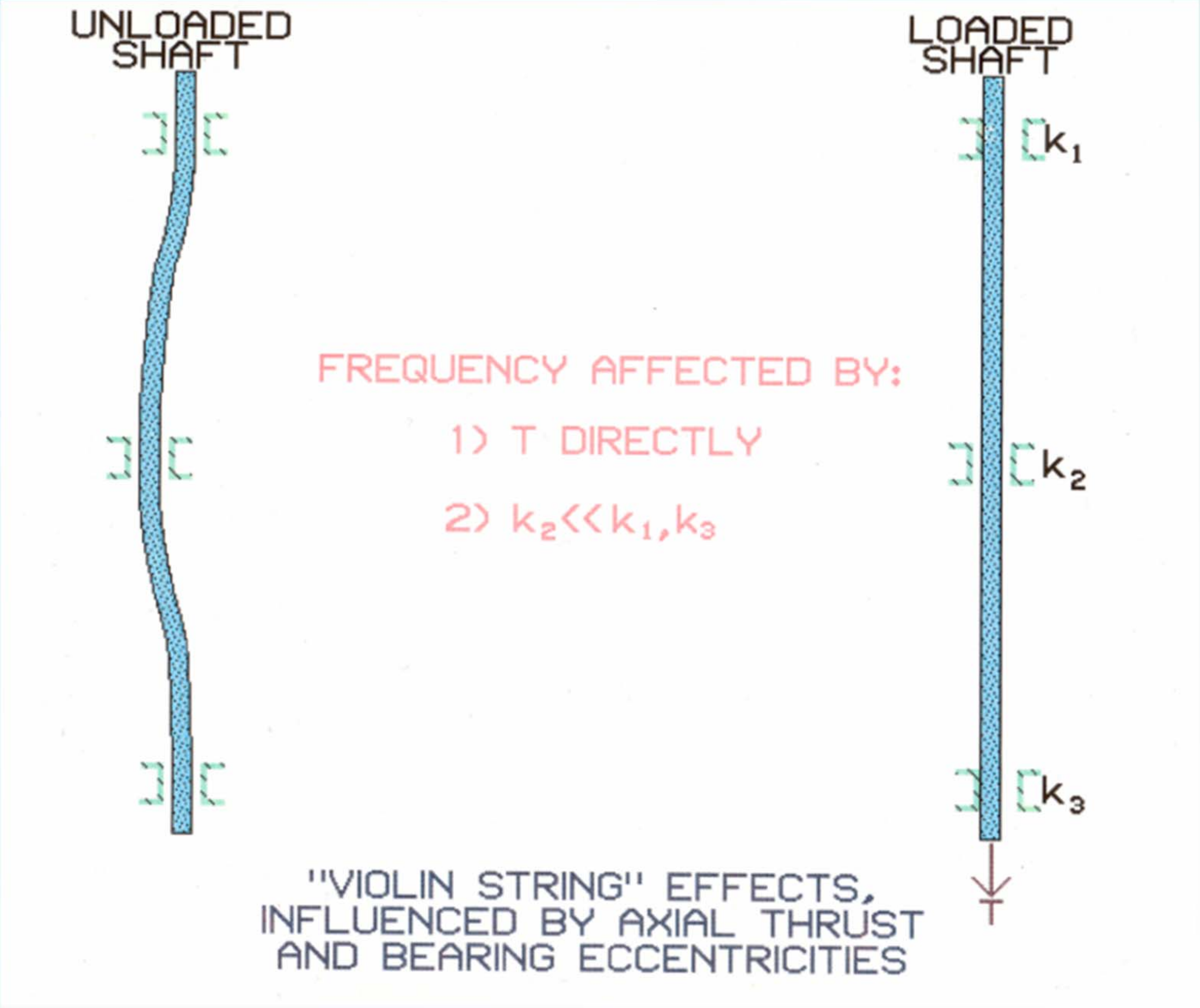
Vertical Pump Shaft Thrust Effects



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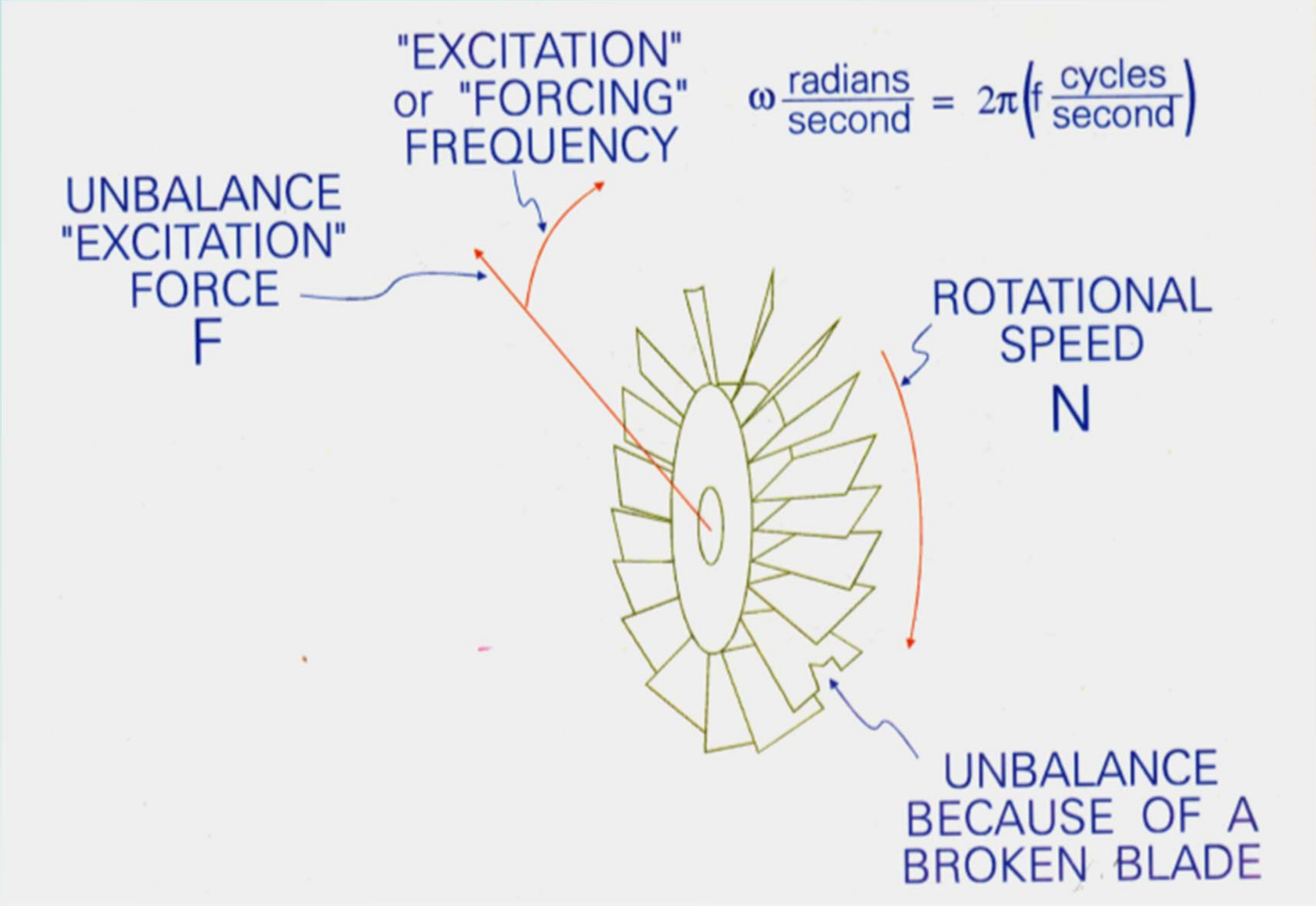
The "Concept" of Imbalance



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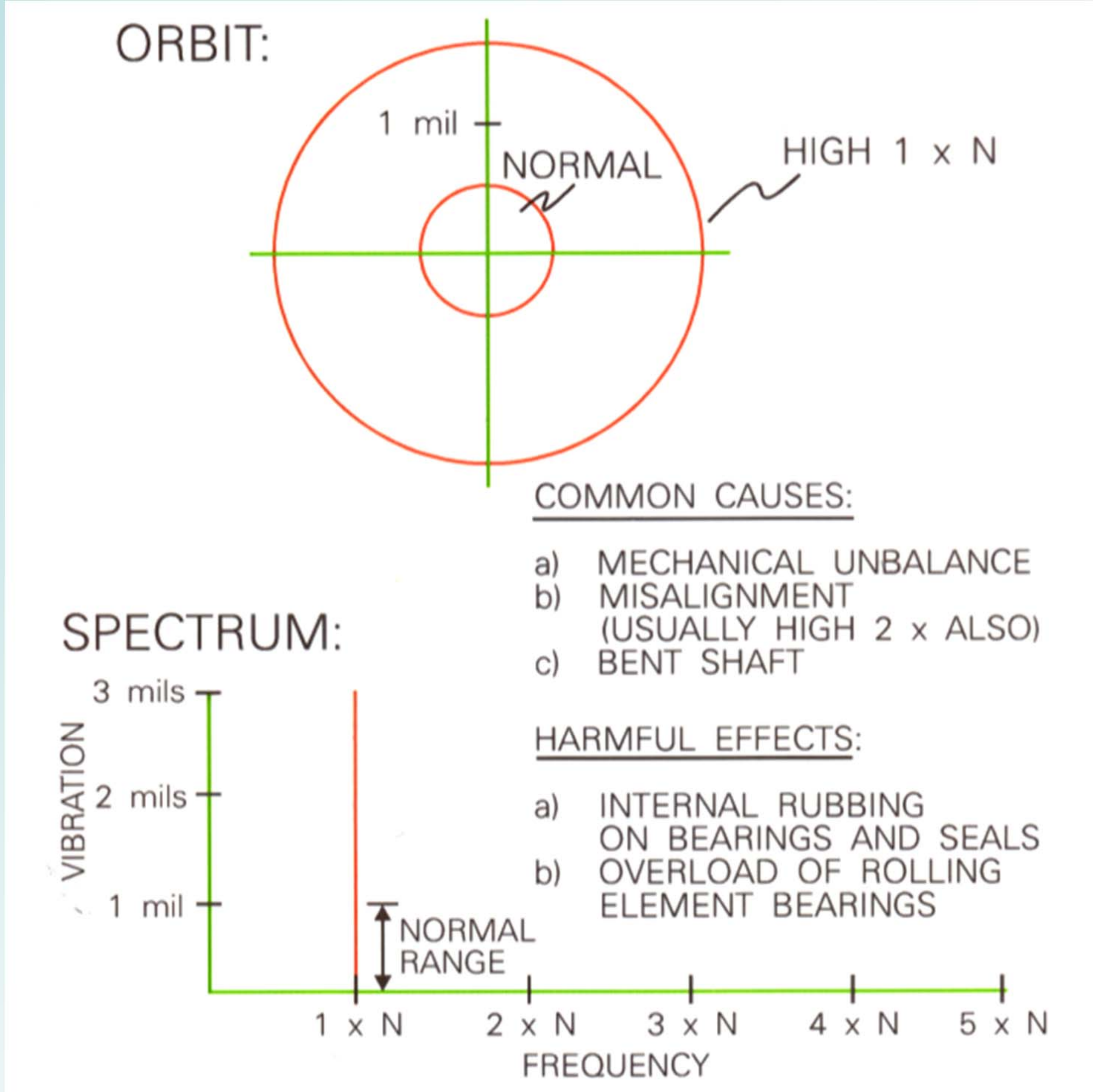
Vibration Problem No. 1:

1x Running Speed



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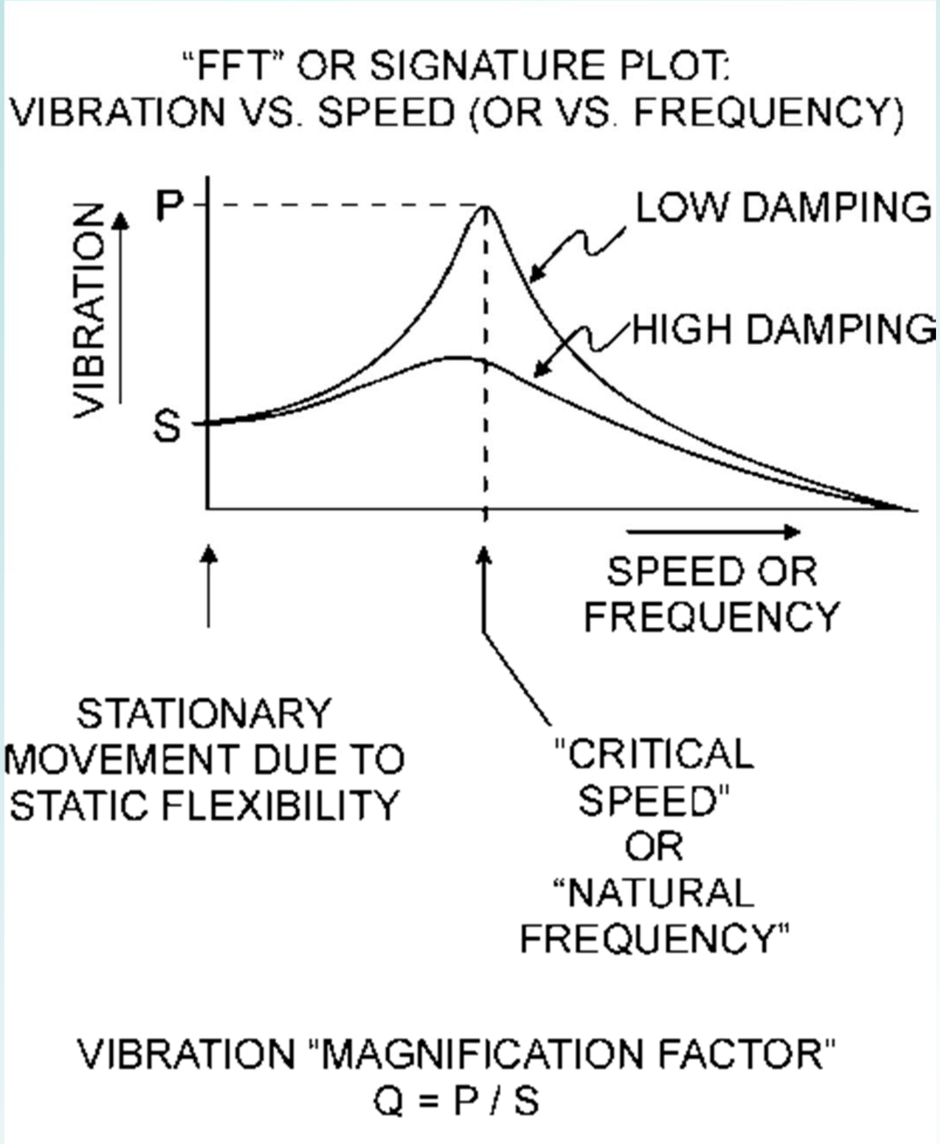
What Is Resonance?



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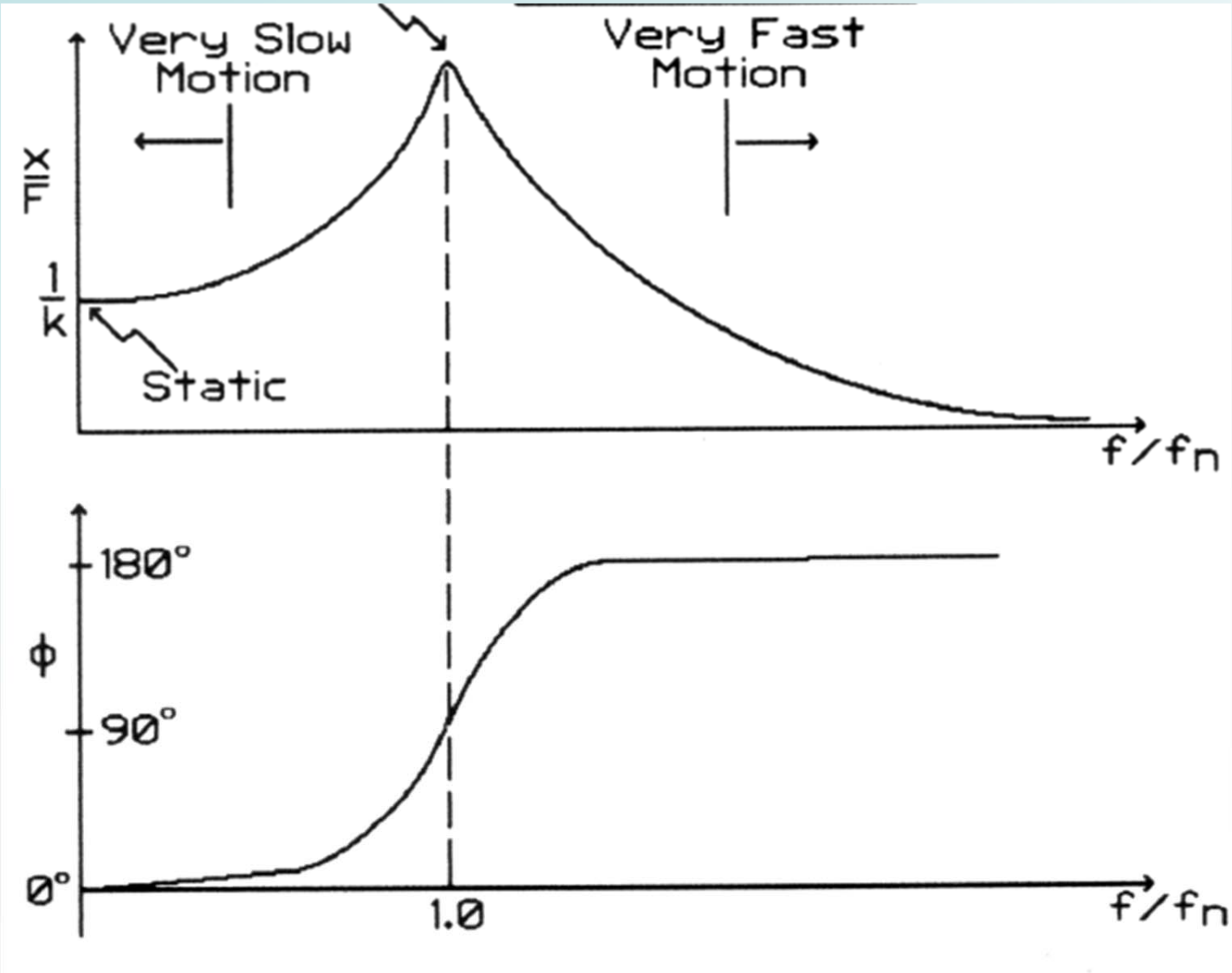
Vibration & Phase vs. Frequency

Natural Frequency



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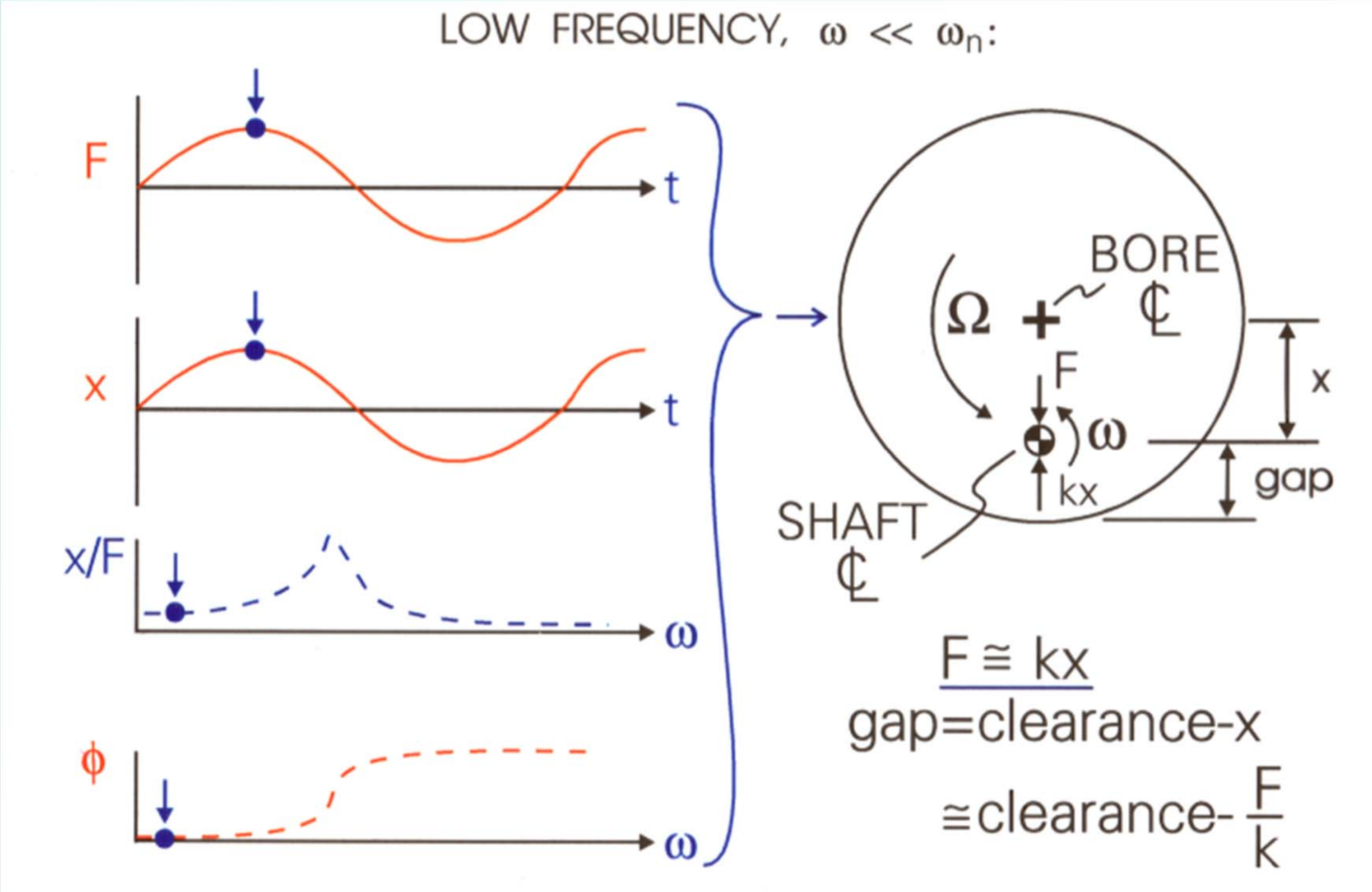
Shaft Response to Imbalance



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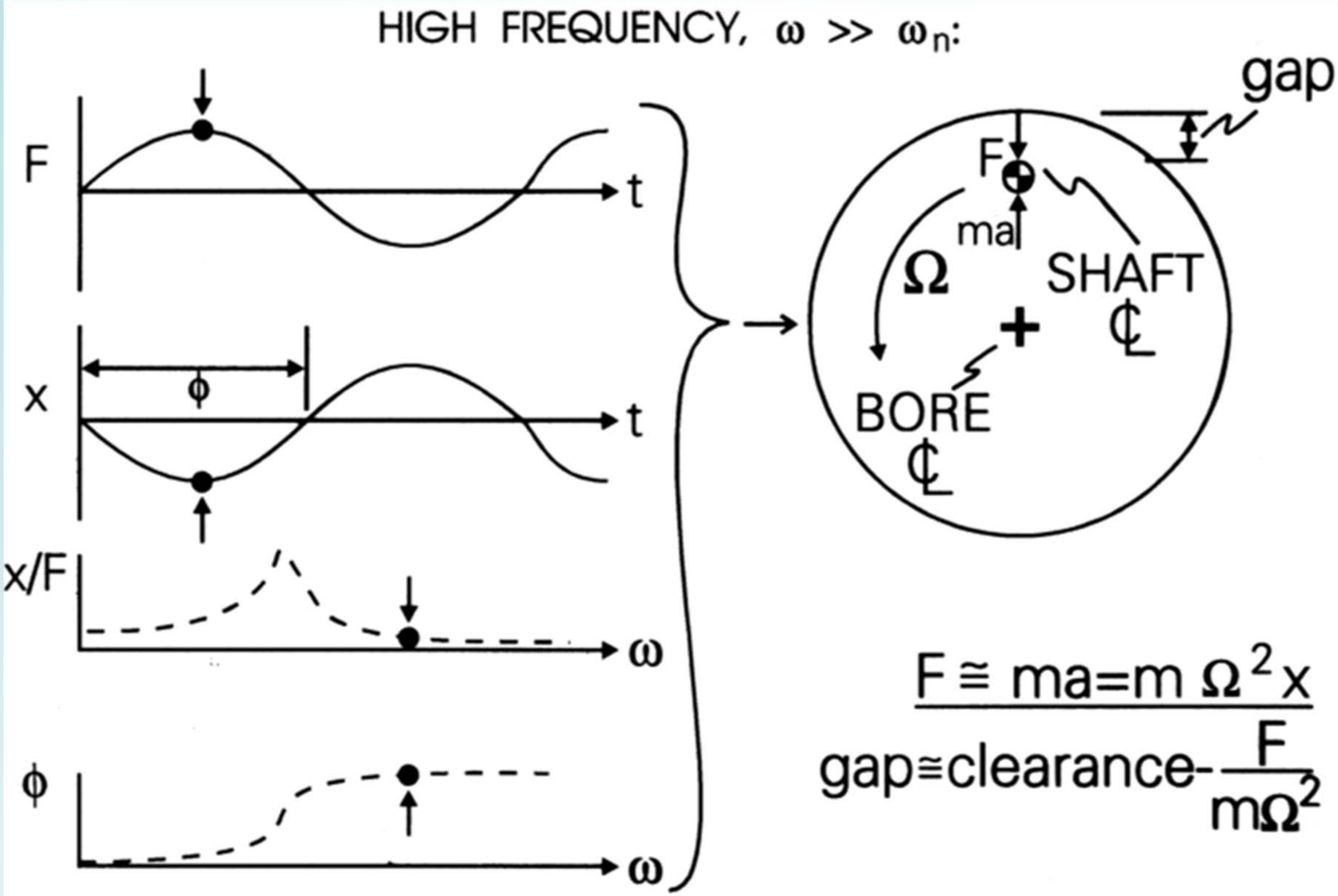
Shaft Response to Imbalance



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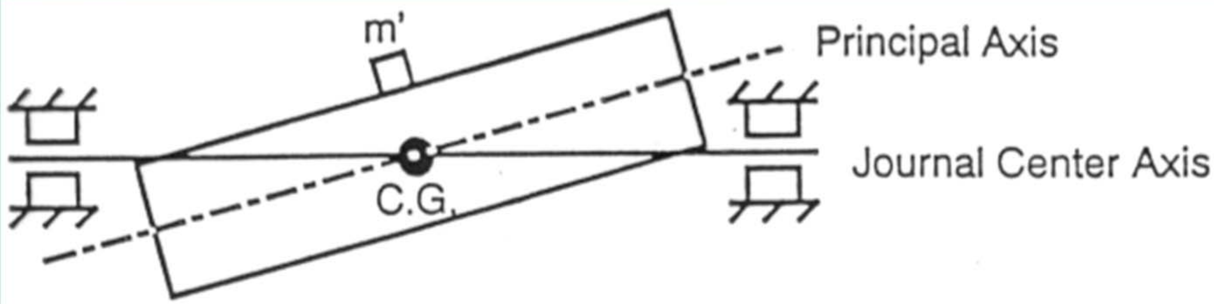
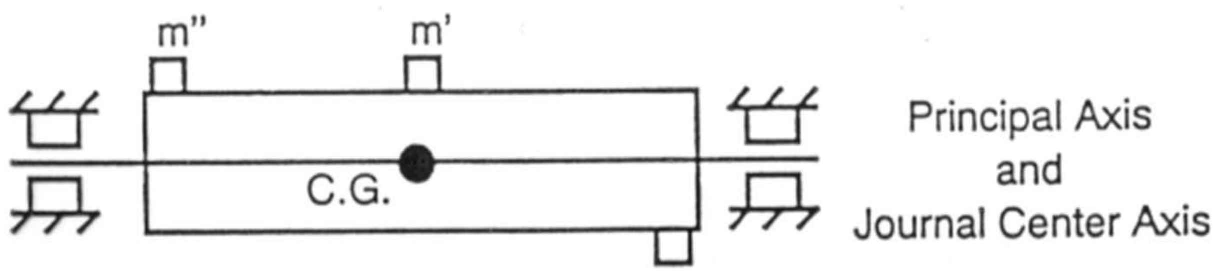
Balance: Single vs. Two Plane



Dynamic Balance: Principal Axis and Journal Center Axis Coincide

F = 0	"Static"
M = 0	"Dynamic"



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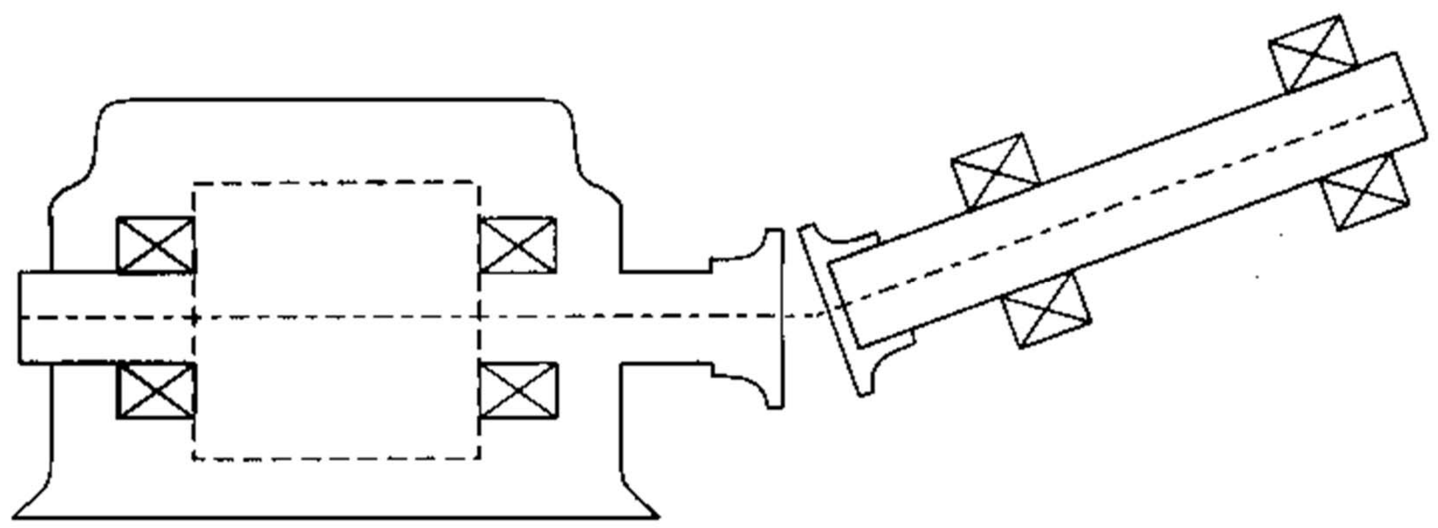
Angular Misalignment



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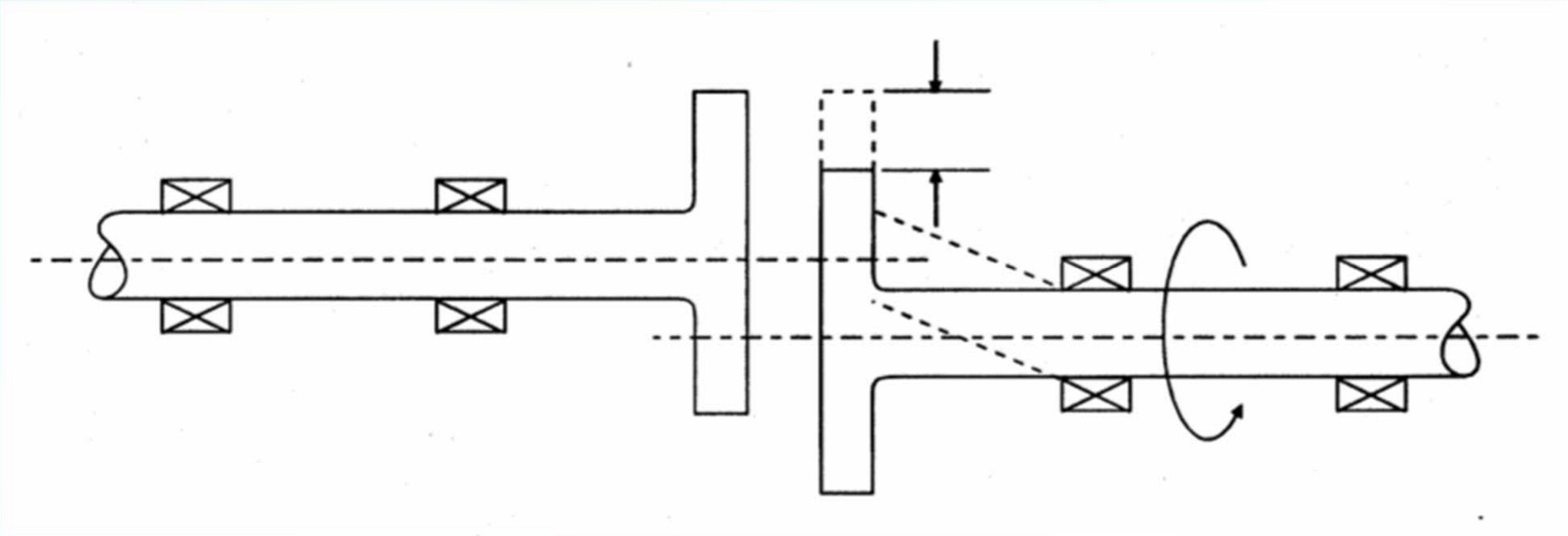
Offset Misalignment



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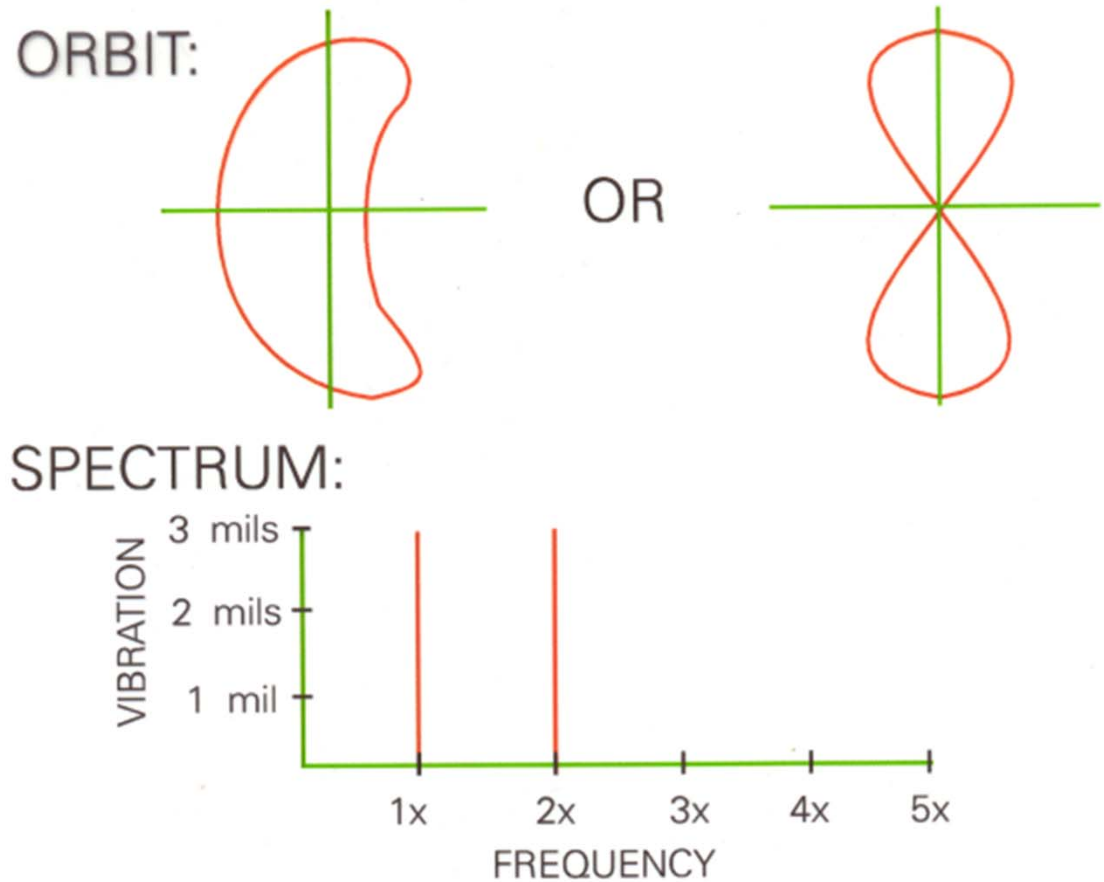
Vibration Problem No. 2:

1x & 2x .Running Speed



COMMON CAUSES:
- a) MECHANICAL MISALIGNMENT
- b) LOOSENESS IN BEARING RETENTION
- c) SEVERE SHAFT OR BEARING HOUSING CRACK

HARMFUL EFFECTS:
- a) INTERNAL RUBBING
- b) COUPLING WEAR
- c) SHAFT FATIGUE

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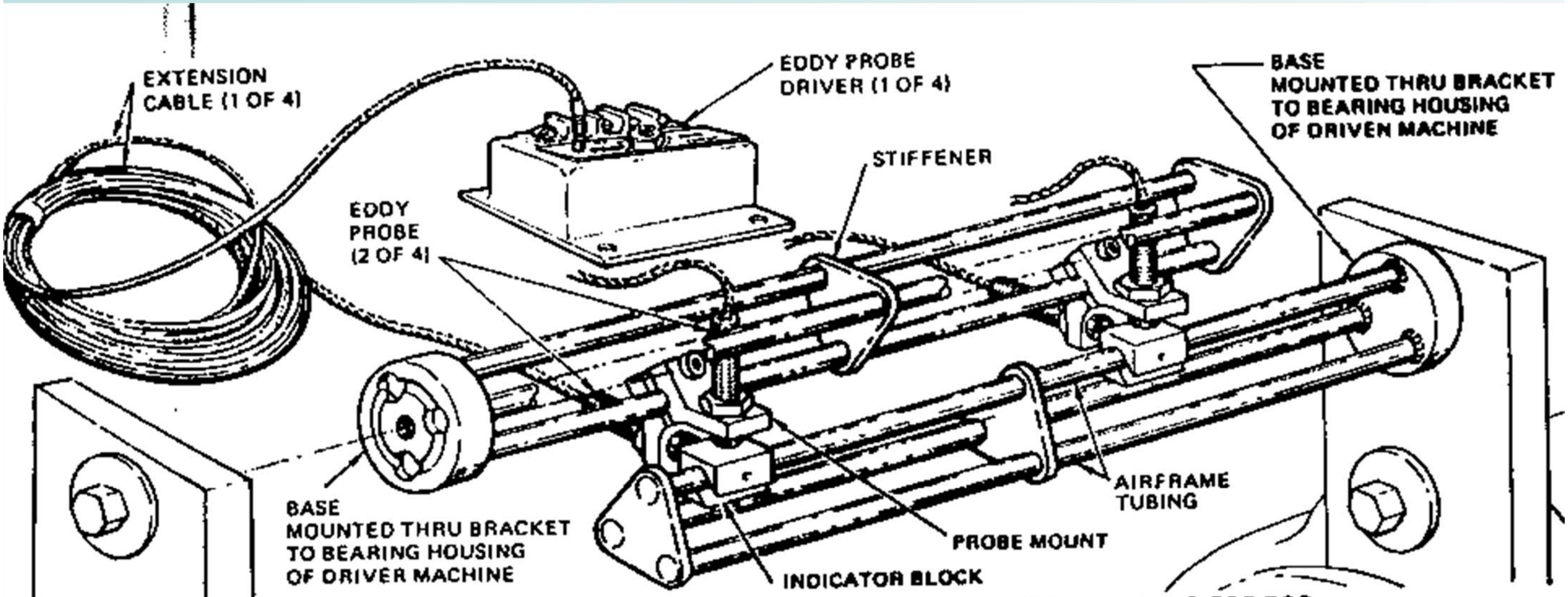
Observing Alignment Continuously with “Dodd Bars”



Produced Under License from Chevron Research Company. U.S.Patent No.3,783,522.

High Energy Density Equipment Alignment Limits

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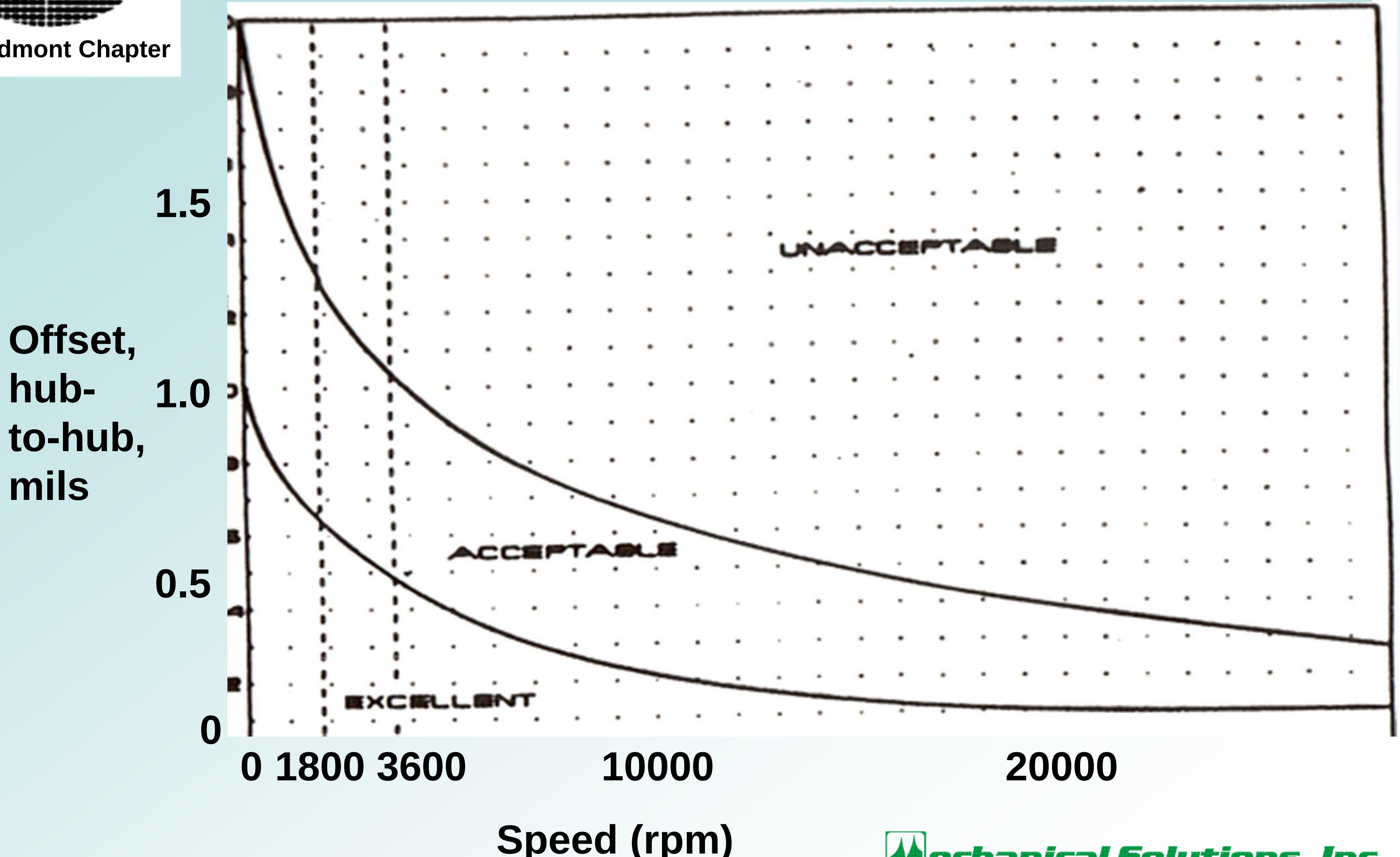

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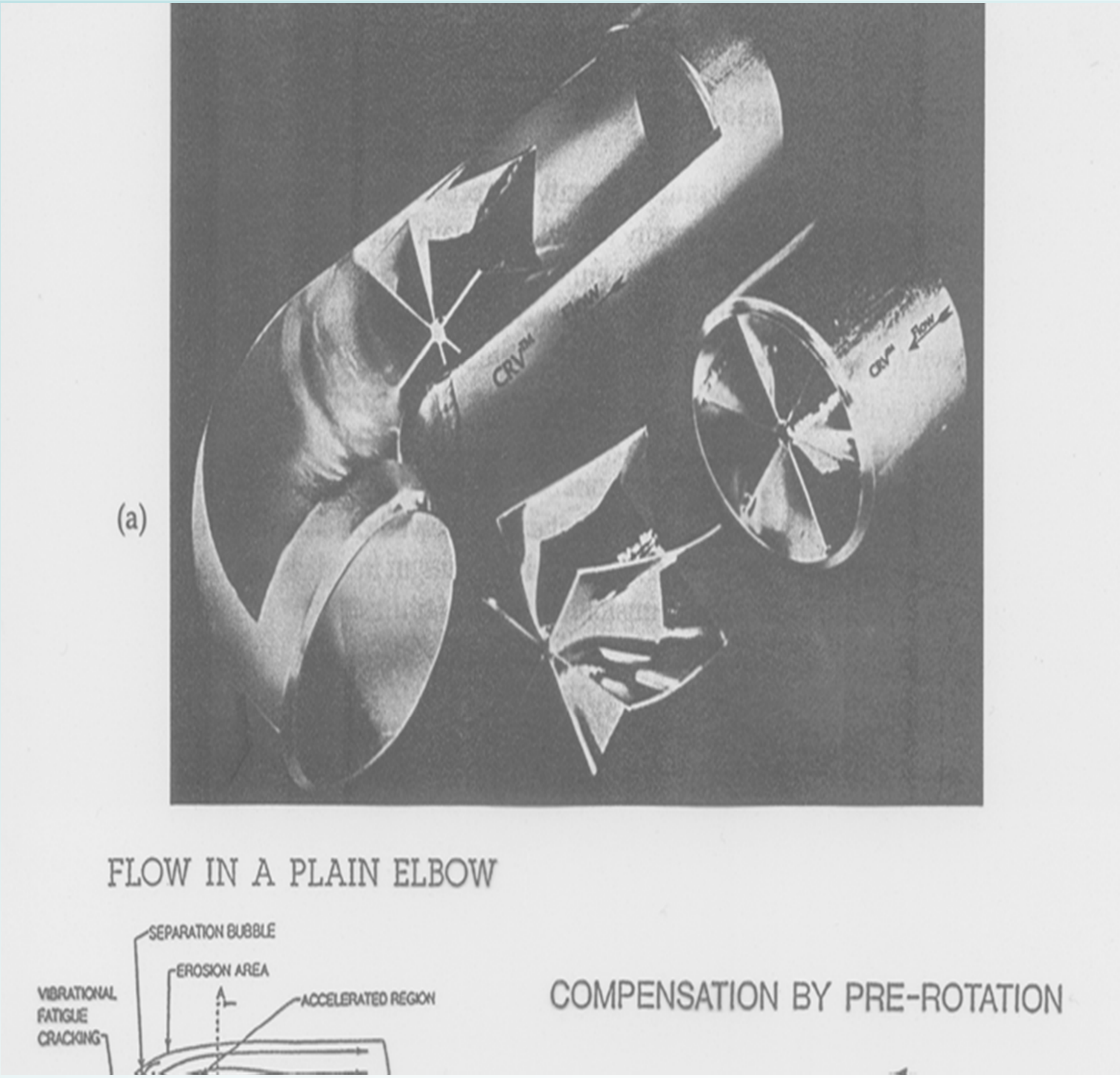
Flow Through Elbows

(Courtesy Cheng Fluid Systems, Inc.)



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Stationary Piping Load Sources

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

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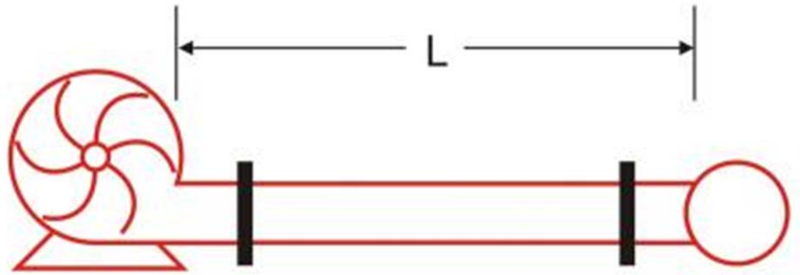
Piping Acoustic Natural Frequencies



a = Speed of Sound in Fluid

$$fn_{A_{1/4}} = \frac{a}{4L}$$

$$fn_{A_{1/2}} = \frac{a}{2L}$$

For Natural Frequency Number “i”:

$$fn_{A_i} = \frac{(2i - 1)\,a}{4L}$$ for quarter wave (closed one end)

or $$\frac{ia}{2L}$$ for half wave (both ends open or both ends closed)

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"Lomakin Effect" in Centrifugal Pumps

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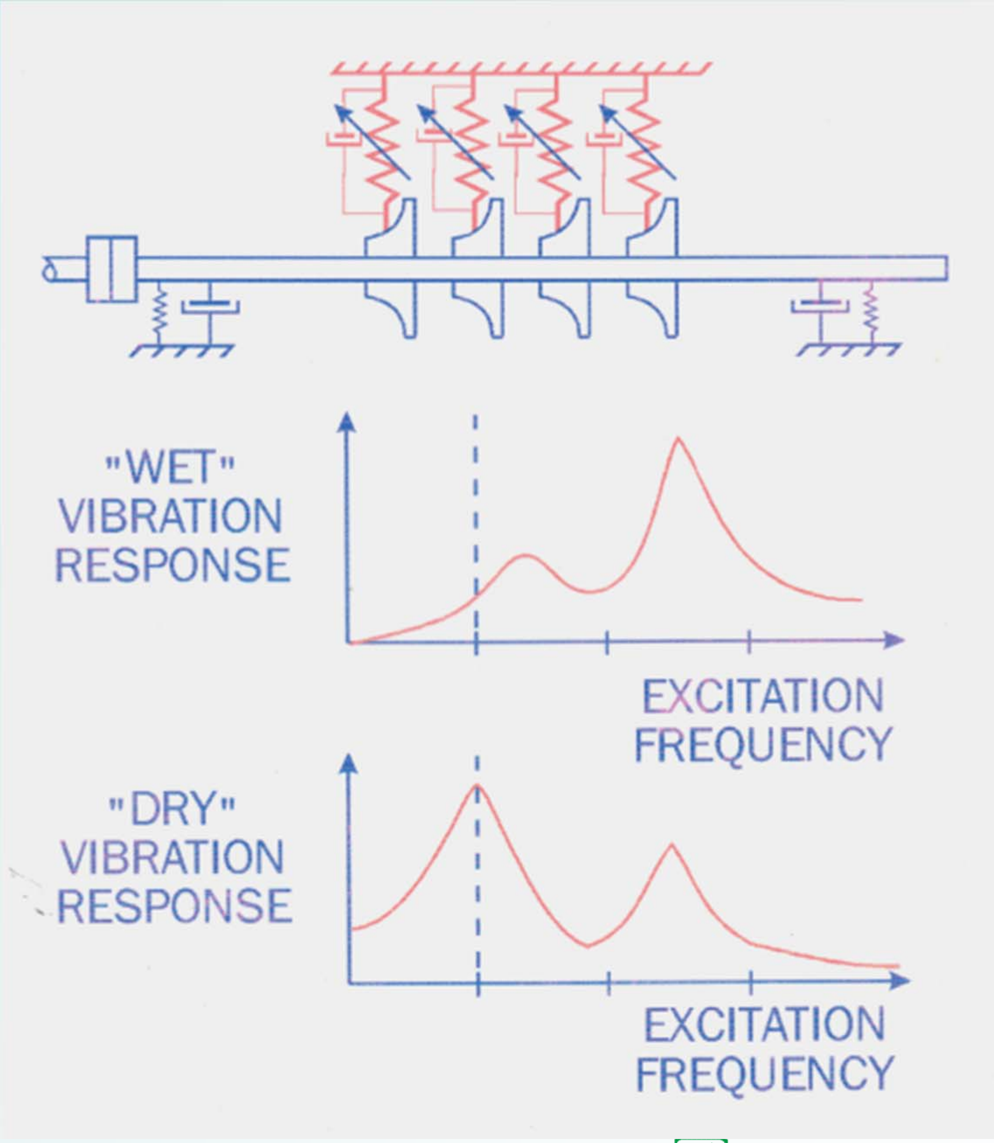
Concept of the “Wet Critical Speed



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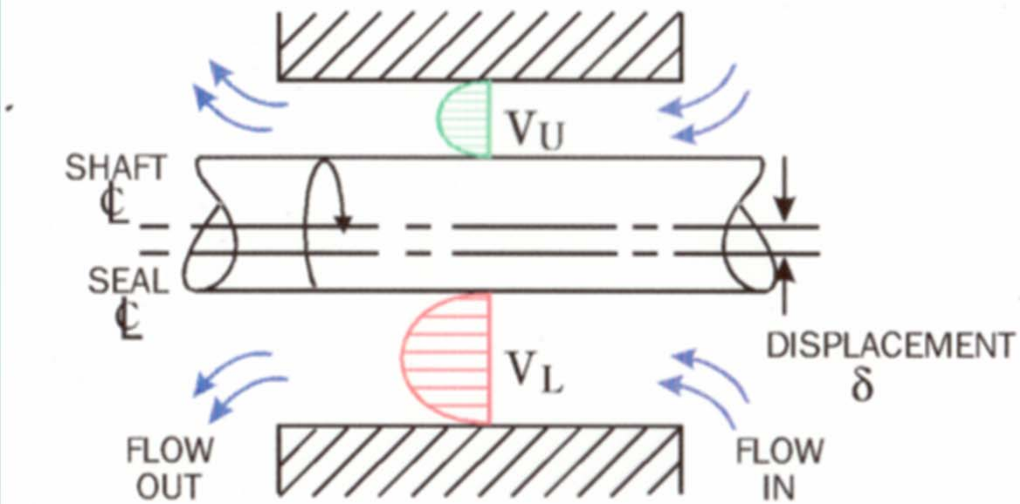
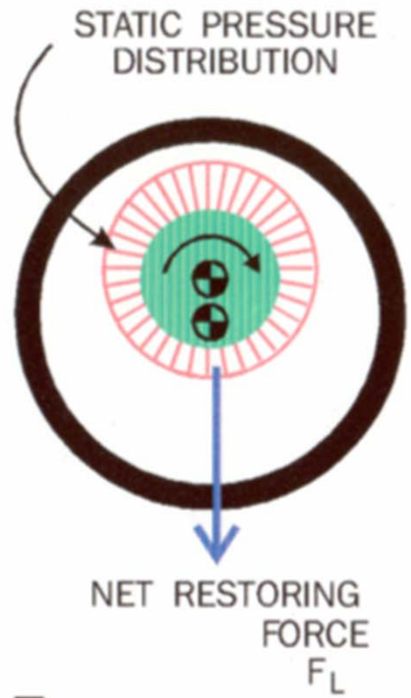
Lomakin Effect

$$P_{static} = P_{stagnation} - \frac{\rho V^2}{2g_c}$$





CRITICAL FACTORS:

- CLEARANCE
- Δ P
- GROOVING

$$K_L = \frac{\Delta F_L}{\Delta \delta}$$

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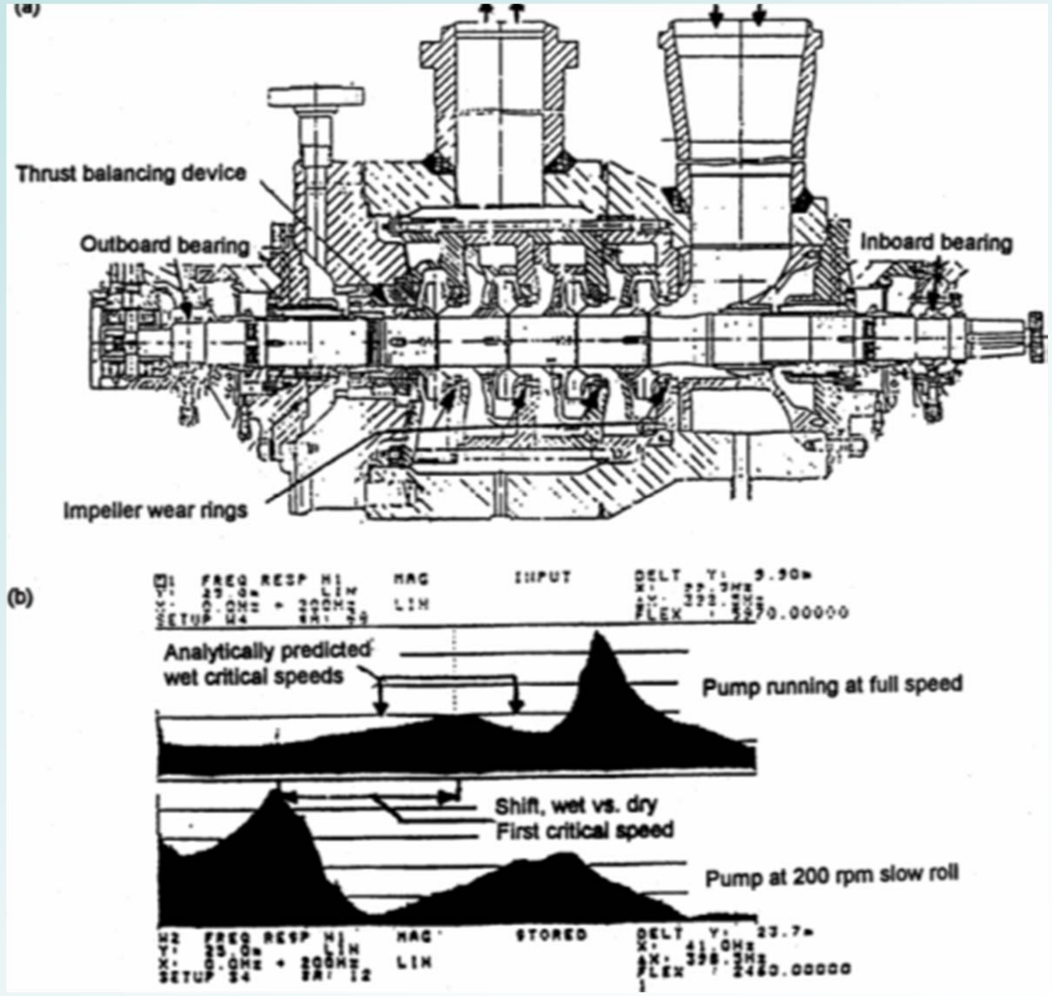
Lomakin Result:

Critical Speeds Shift Up



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Approximate Calculation of Lomakin Stiffness

R= Radius

L= Length

C= Radial clearance

ΔP=Pressure drop

λ= fric factor

$$k_{xx} \cong \frac{RL\,\Delta P}{c} K_{xx}$$

$$K_{xx} \cong \frac{\pi\sigma}{(1+2\sigma)^2} \cong 0.04 \quad (L \cong 2R)$$

$$\cong 0.40 \quad (L << 2R)$$

$$\text{where} \quad \sigma = \frac{\lambda L}{c}$$

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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

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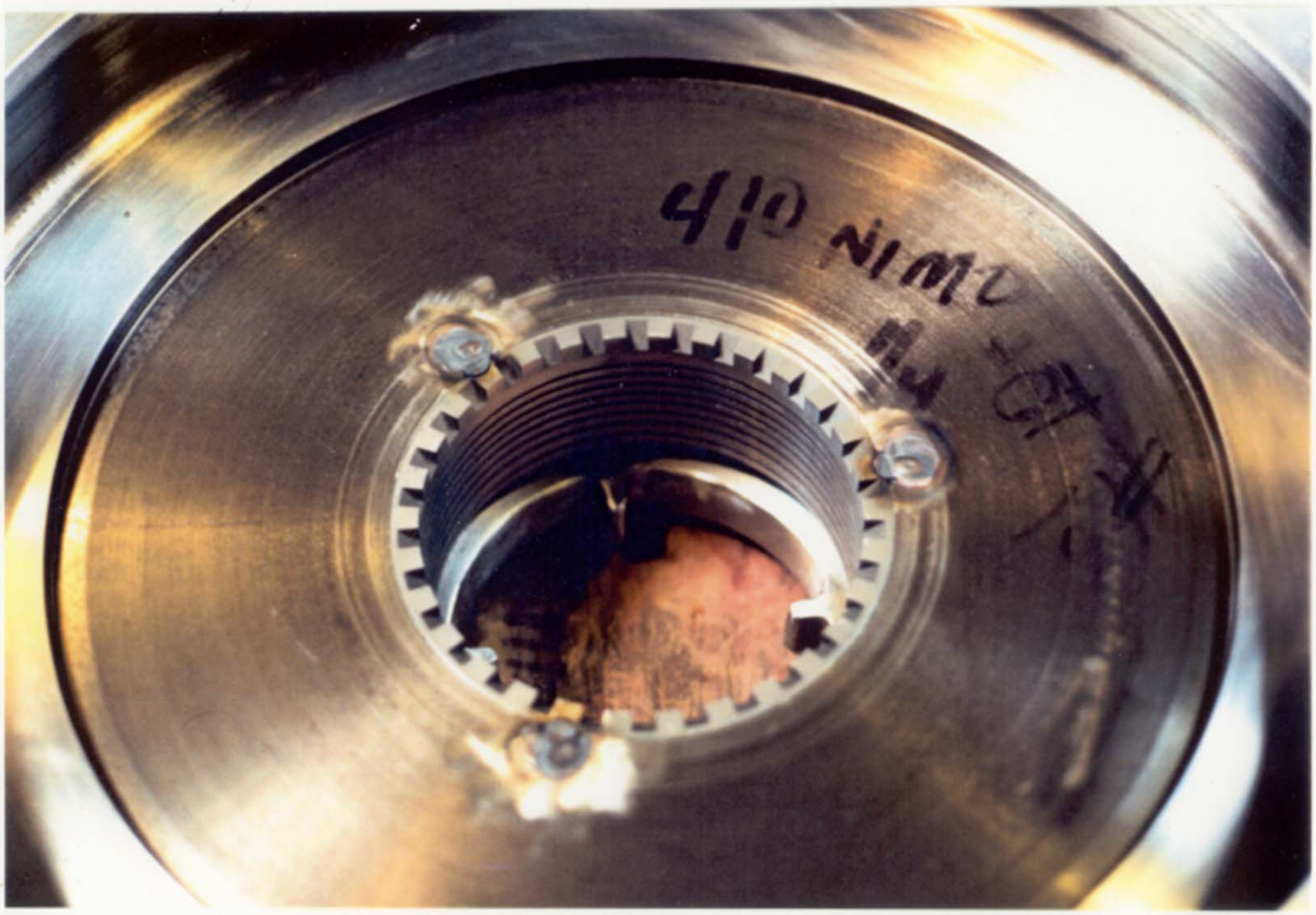
Wild Cards: Swirl & Grooving



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Impeller Radial Support Forces:

Sulzer/ EPRI Tests

Normalization:

$$k^* = \frac{1}{\pi r_2^2 B_2 \rho \omega^2} k$$

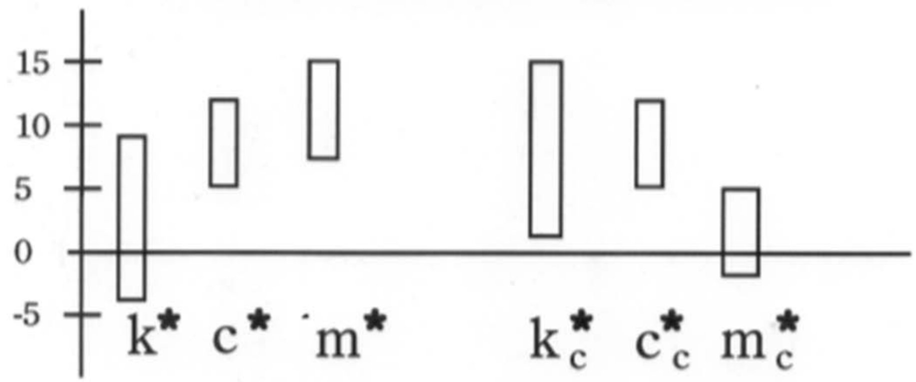
$$c^* = \frac{1}{\pi r_2^2 B_2 \rho \omega} c$$

$$m^* = \frac{1}{\pi r_2^2 B_2 \rho} m$$

r_2 = Impeller Radius, B_2 = Impeller exit width

ρ = Density, ω = rotational angular frequency

* = Normalized quantities (dimensionless)



For circular orbit of whirl frequency Ω:

Radial (in dir. of displ.) Tangential (in dir. of rot.)

$$F_r^*/x = -k^* - c_c^*\left(\frac{\Omega}{\omega}\right) + m^*\left(\frac{\Omega}{\omega}\right)^2 \qquad F_t^*/x = -k_c^* + c^*\left(\frac{\Omega}{\omega}\right) + m_c^*\left(\frac{\Omega}{\omega}\right)^2$$

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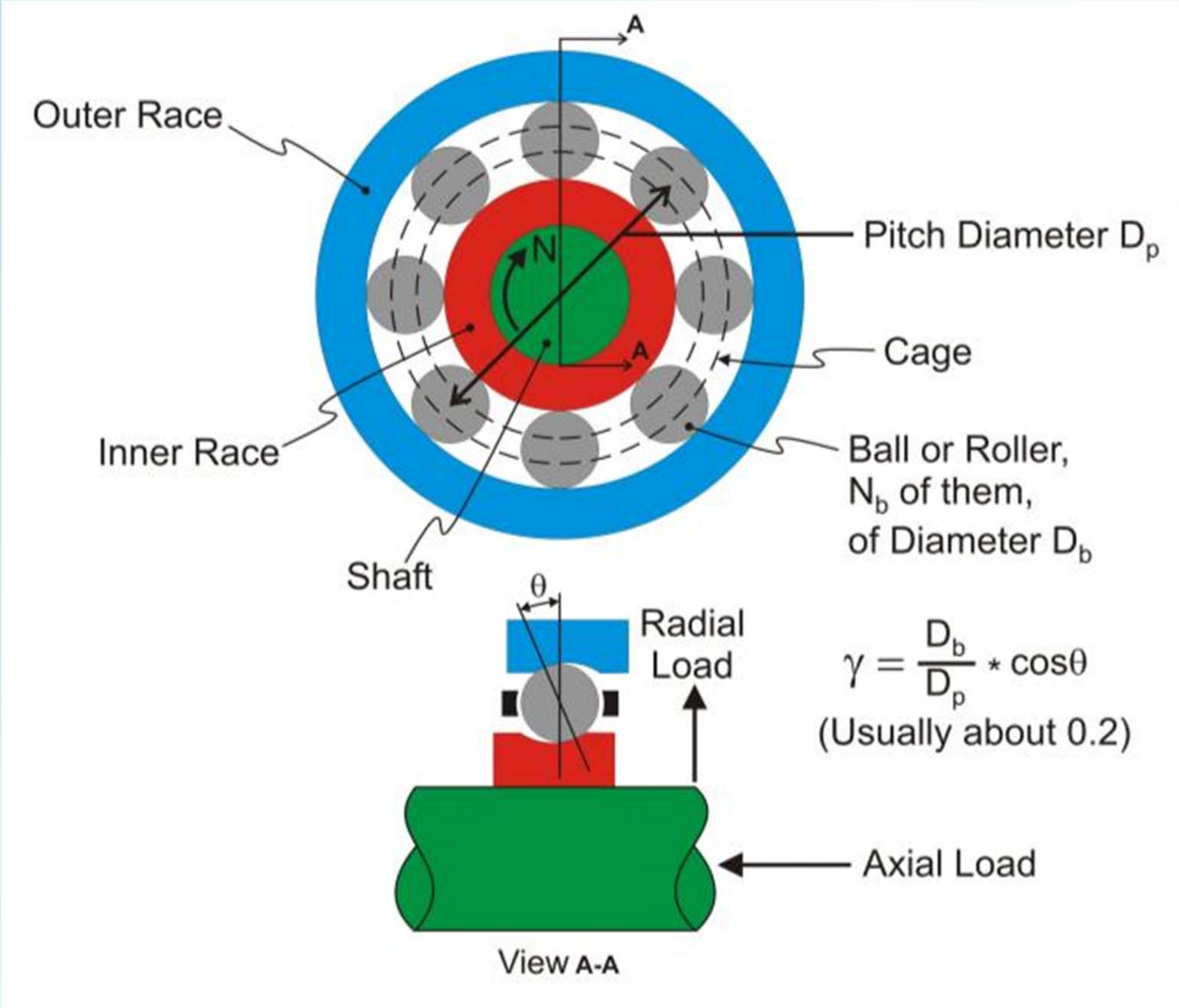
Some Typical Exciting Frequencies

FREQUENCY	SOURCE
0.05 - 0.35 x	DIFFUSER STALL
0.43 - 0.48 x	INSTABILITY
0.500 x	RUBBING
0.65 - 0.95 x	IMPELLER STALL
1 x	IMBALANCE
1 x +2 x	MISALIGNMENT
#Vanes x	VANE/VOLUTE GAP
#Blades x	BLADE/DIFFUSER GAP

Rolling Element Bearing Parameters



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Rolling Element Bearing Frequencies

$$FTF = \frac{N}{2}(1-\gamma)$$ Fundamental Train Frequency

$$BSF = \frac{D_p}{D_b} * \frac{N}{2}(1-\gamma^2)$$ Ball Spin Frequency

$$BPFO = N_b * \frac{N}{2}(1 - \gamma)$$ Ball Pass Frequency Outer Race

$$BPFI = N_b * \frac{N}{2}(1 + \gamma)$$ Ball Pass Frequency Inner Race

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Lateral Excitation Forces (Sulzer)

Normalization: $F_R^* = \frac{F_R}{\rho g H D_2 B_2}$

D_2 = Impeller diameter,
B_2 = Impeller exit width
ρ = Density, g = Gravity, H = Head
F_R^* = Normalized radial force, nondimensional

		Diffuser	180° Double Volute	Single Volute
Static	Suction impeller (suction asymmetry)	.01 - .08 (.04)	.02 - .15 (.05)	.05 - .35 (.15)
	Normal impeller (no suction asymmetry)	.01 - .06 (.03)	.01 - .10 (.04)	.03 - .25 (.06)
Dynamic	Broad Band up to 1.2 times rotational frequency	.01 - .15 (.03)	.01 - .12 (.03)	.01 - .12 (.03)
	Hydraulic unbalance, at rotational frequency	.005 - .03 (.02)	.005 - .10 (.03)	.05 - .30 (.05)

Ranges for F_R for Q=25% to 125% of Design Point
Values in brackets: typical for Design Point

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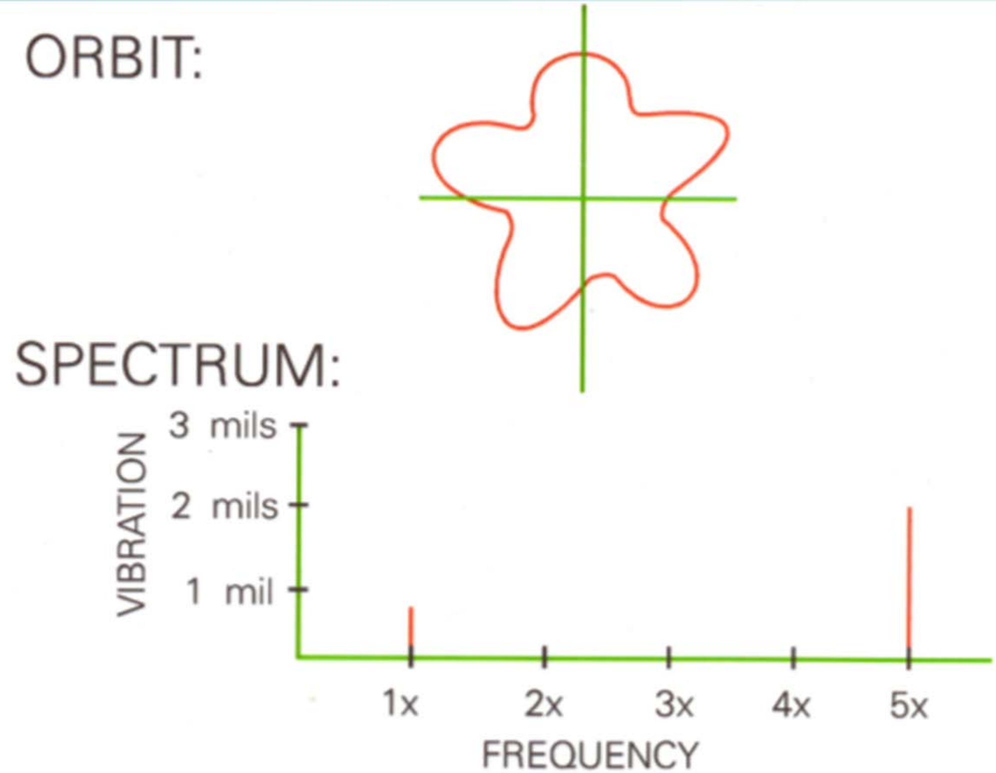
Vibration Problem No. 3: High Vane Pass



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

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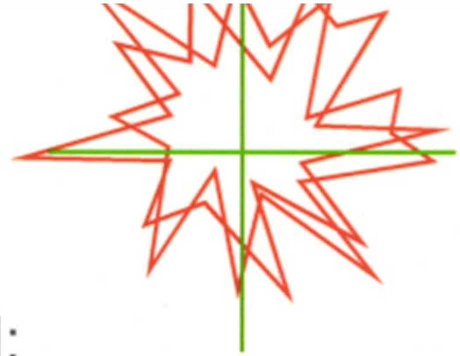
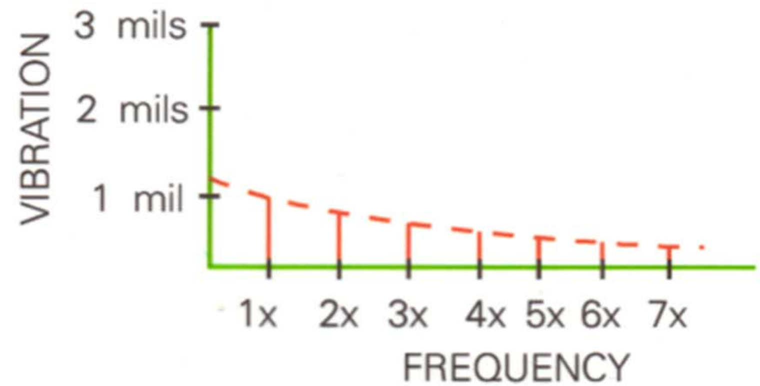
Vibration Problem No. 4: High Harmonics of Run Speed

ORBIT:



SPECTRUM:



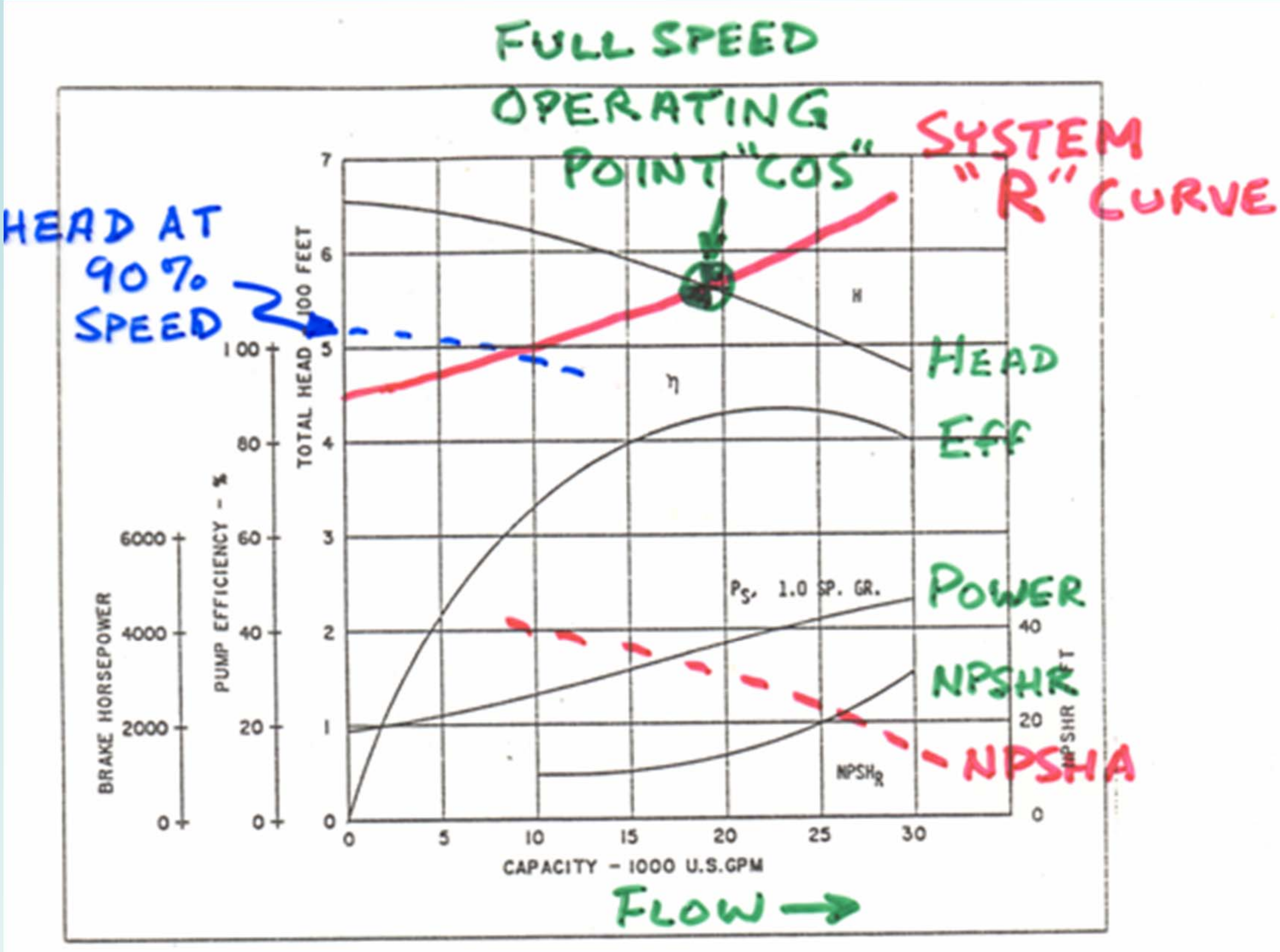
COMMON CAUSES:

a) SCRATCH IN THE SHAFT OR CHROME PLATE THICKNESS VARIATIONS AT BENTLY PROBE
b) RUBBING
c) VERY LOOSE "RATTLING" INTERNAL COMPONENT, SUCH AS IMPELLER

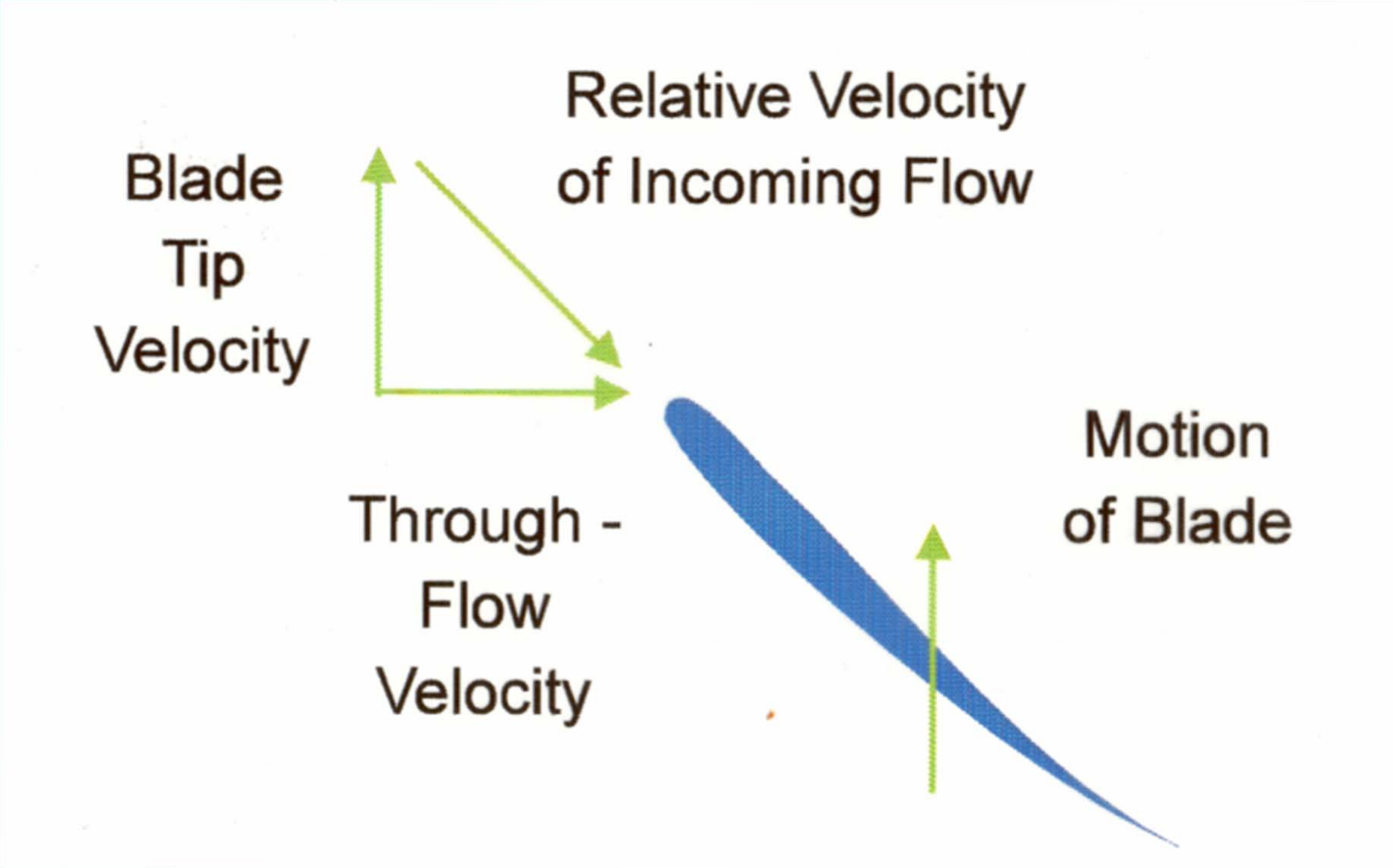
HARMFUL EFFECTS:

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"



Vane Stalling at Low Flows

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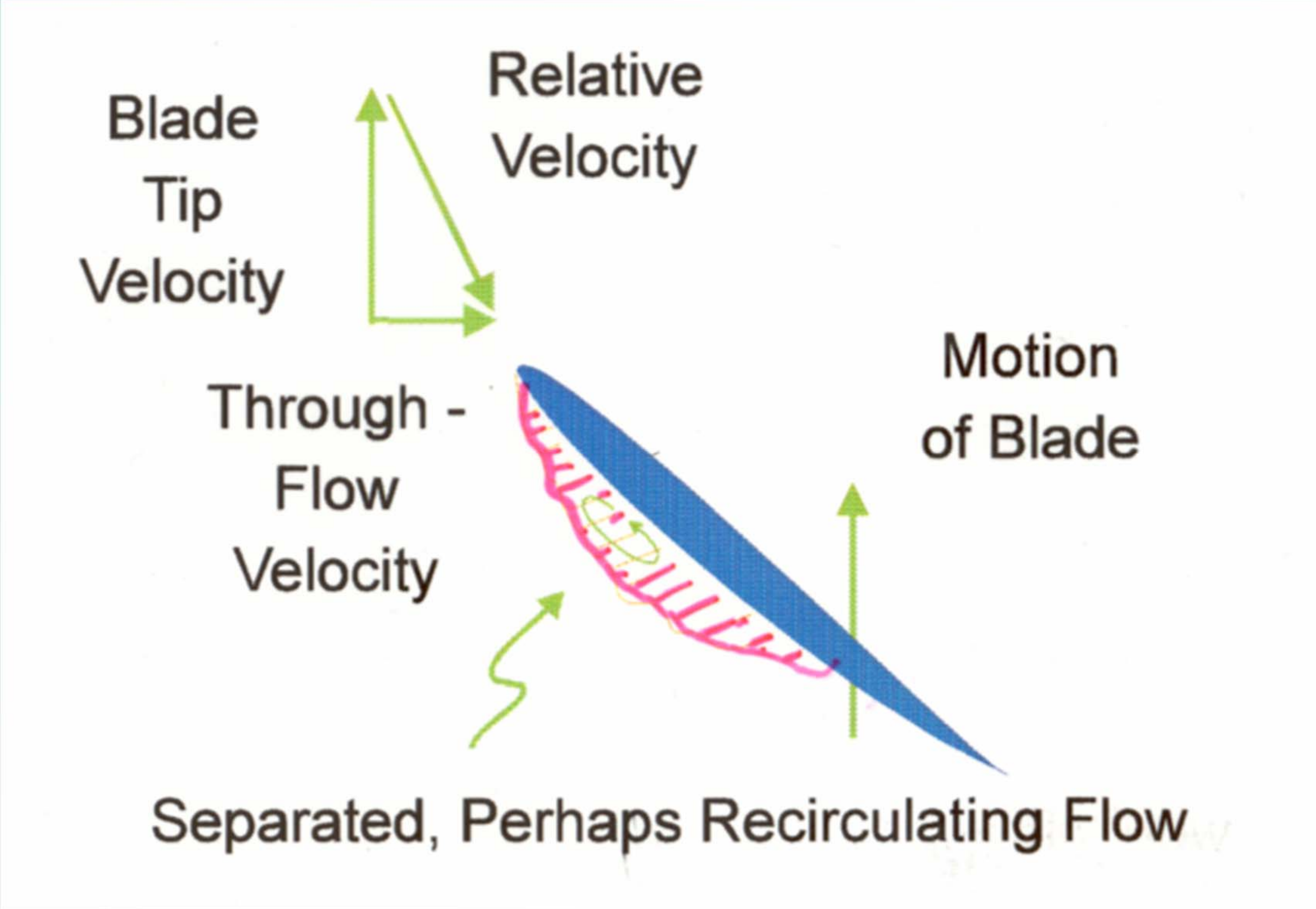

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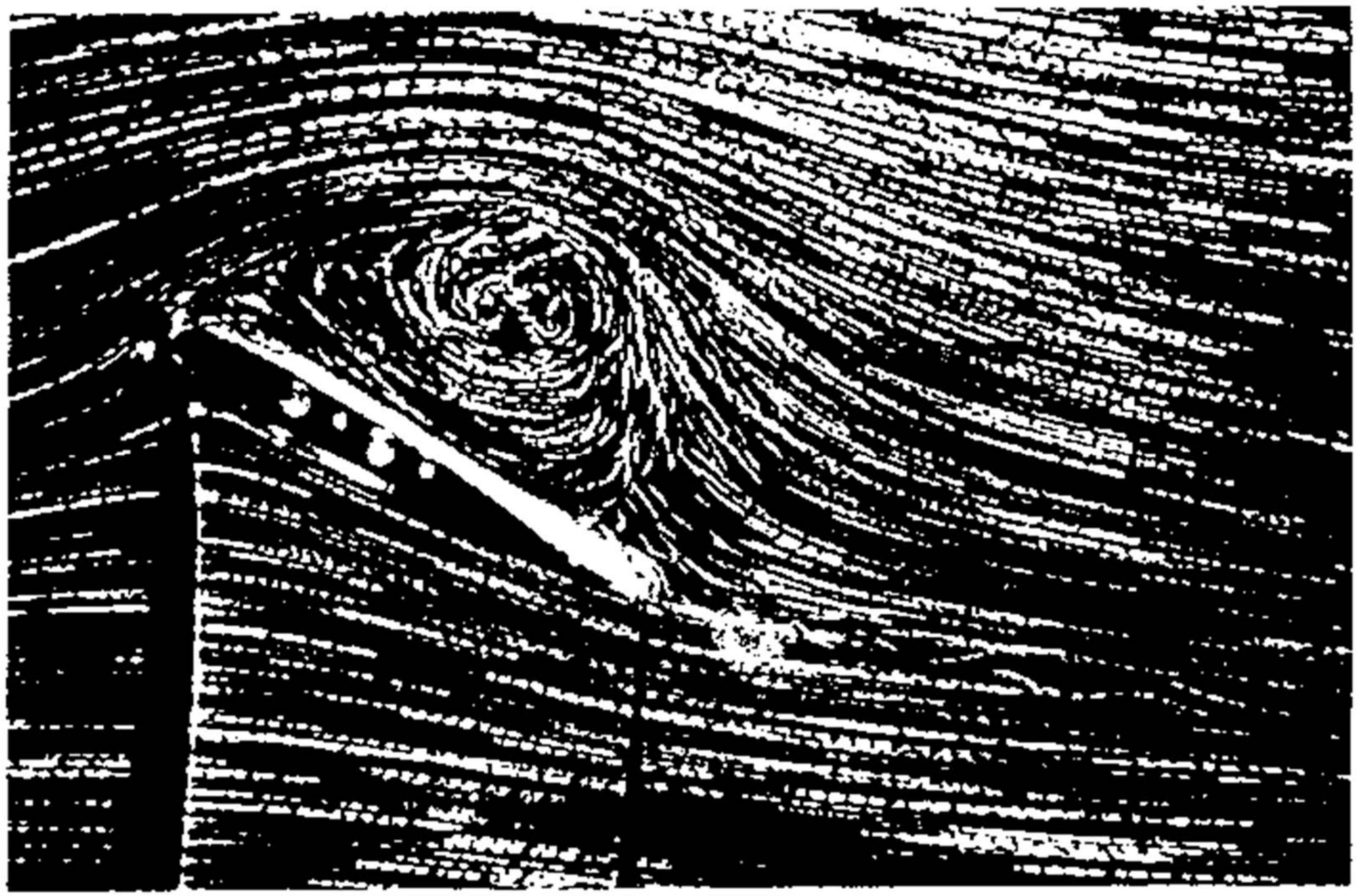
Example of Stalled Blade



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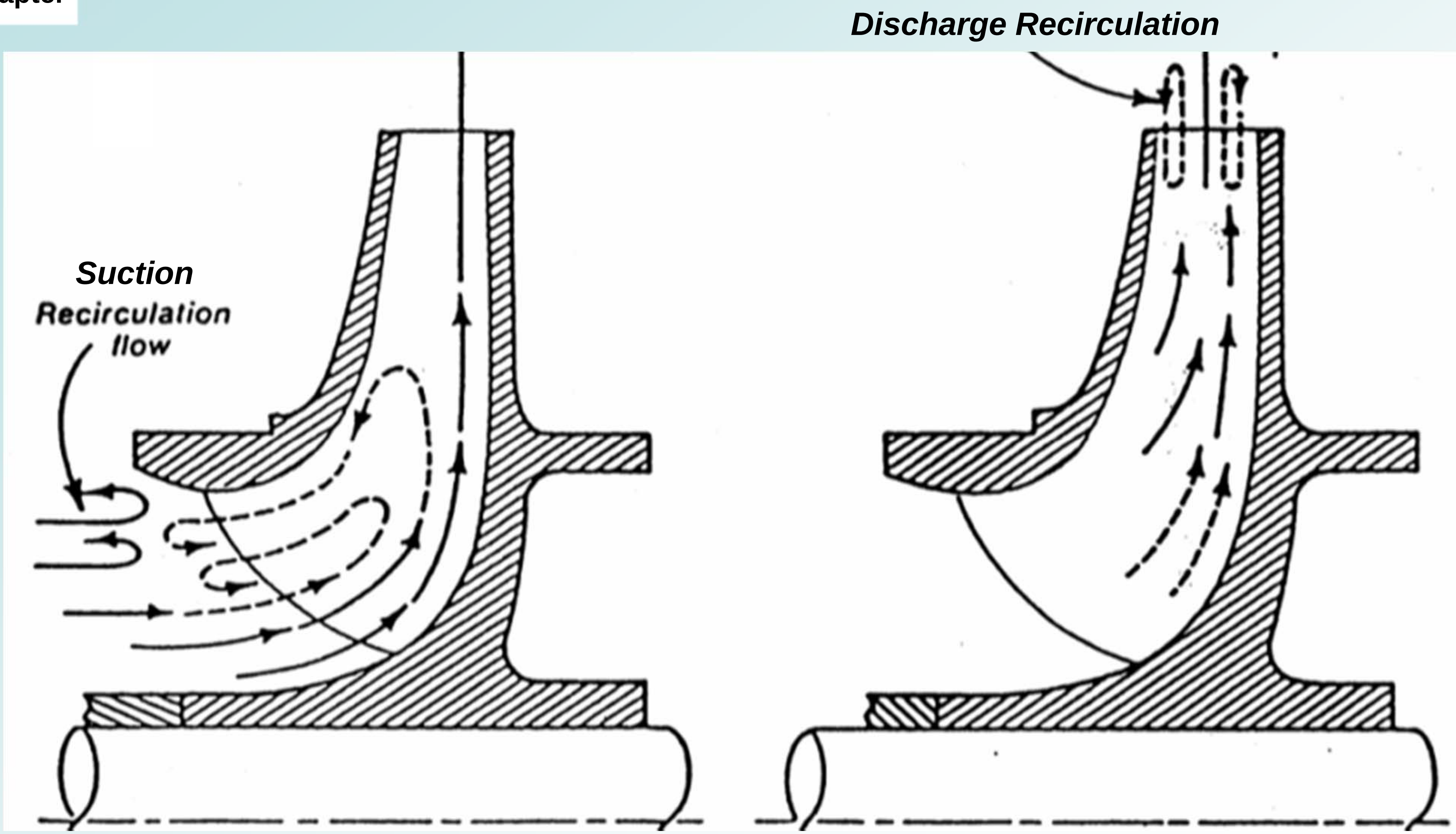
Onset of Internal Recirculation



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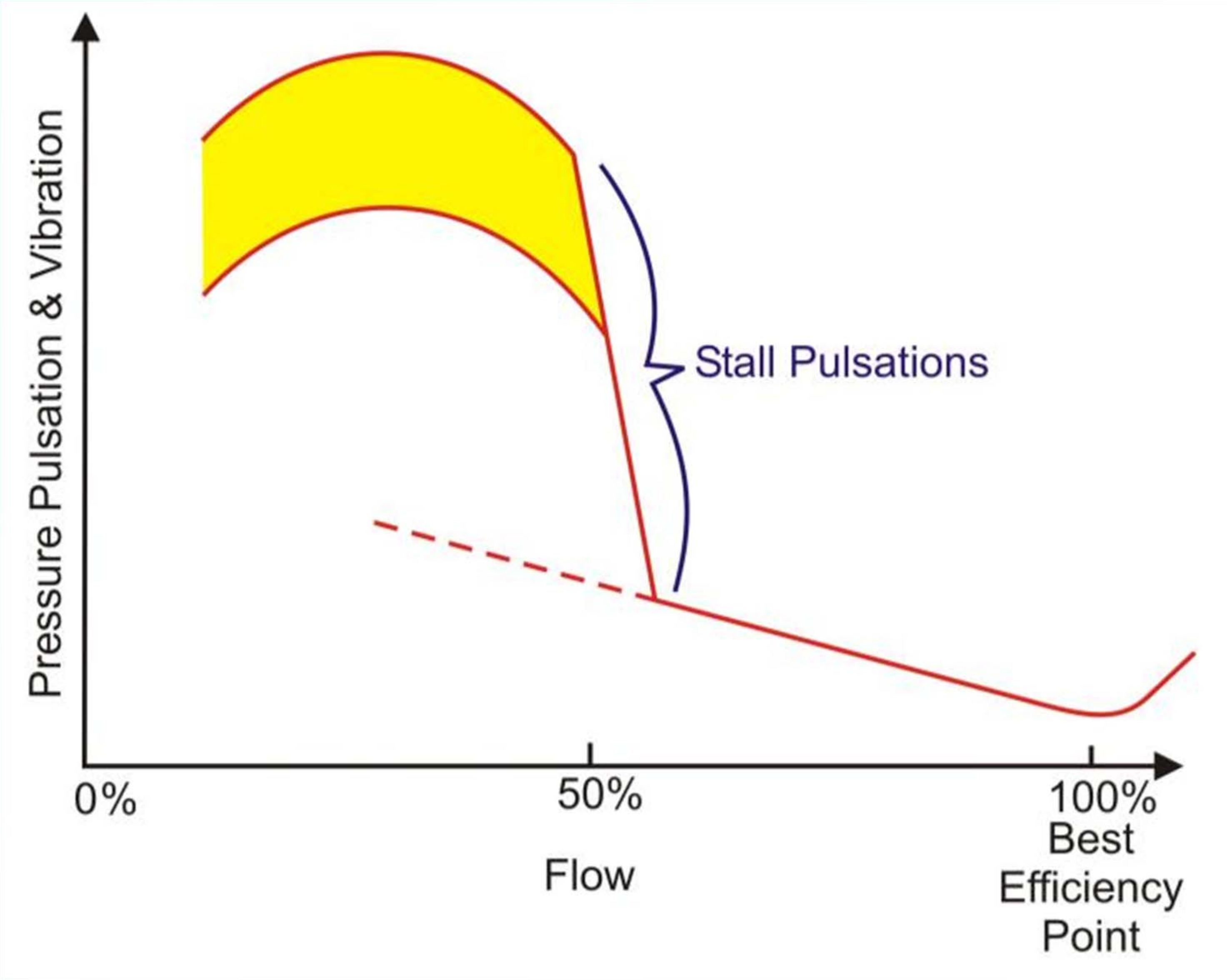
Vibration & Pulsation vs. Flowrate



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An Unexpected Hydraulic Problem:

Rotating Stall

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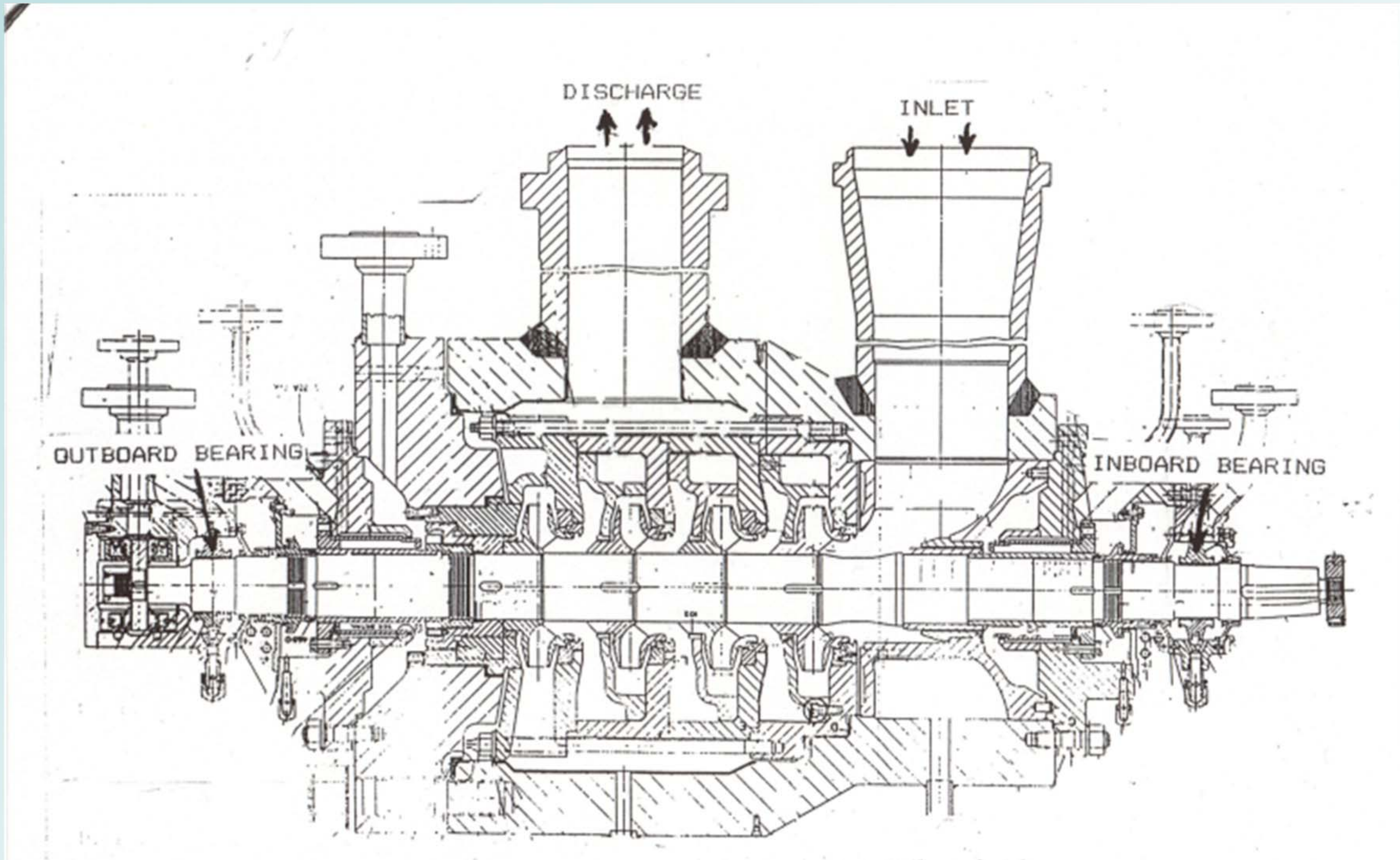
Four Stage High Speed Boiler Feed Pump



Four Stage Pump Configuration

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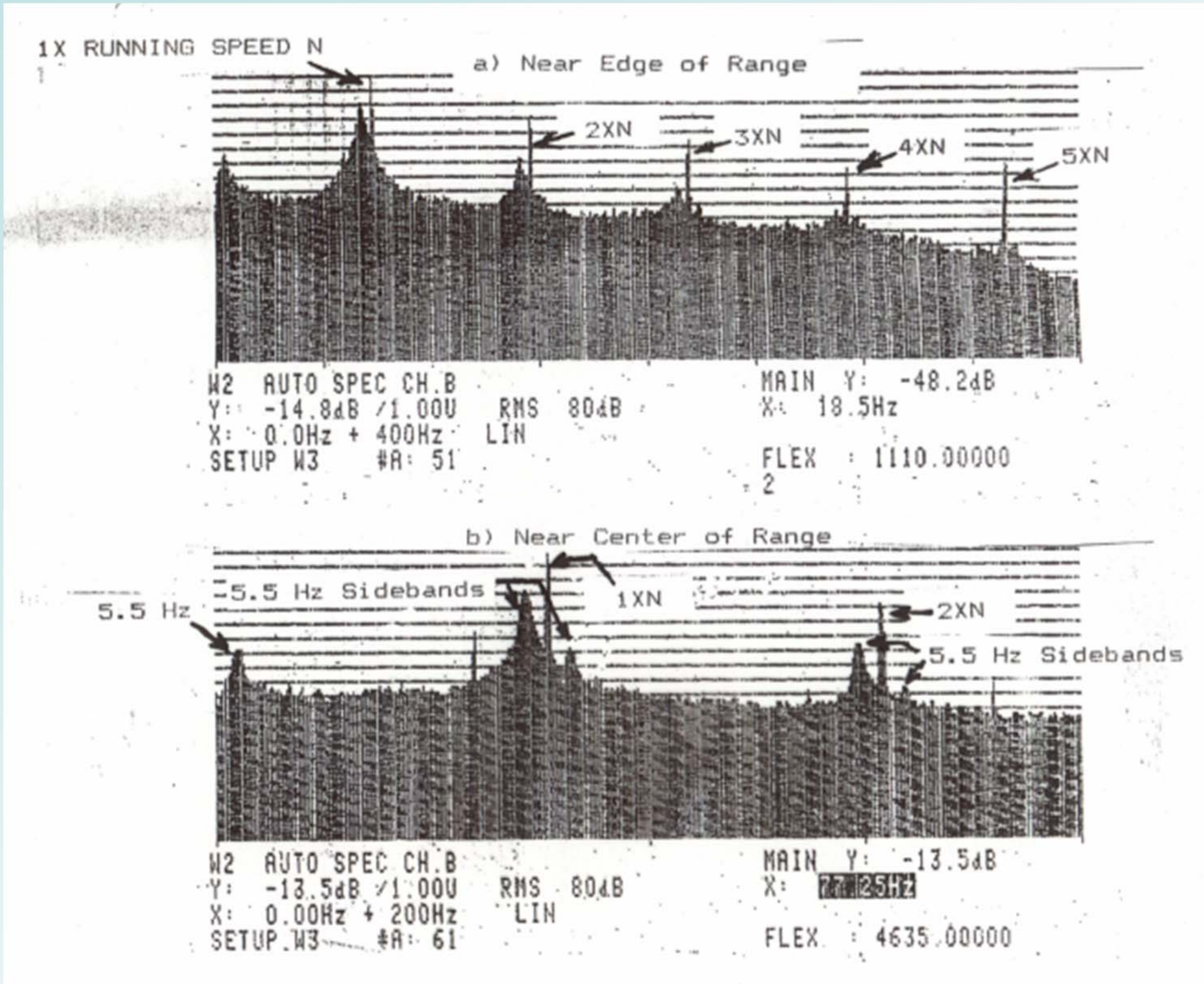
Rotating Stall Pulsation Spectrum



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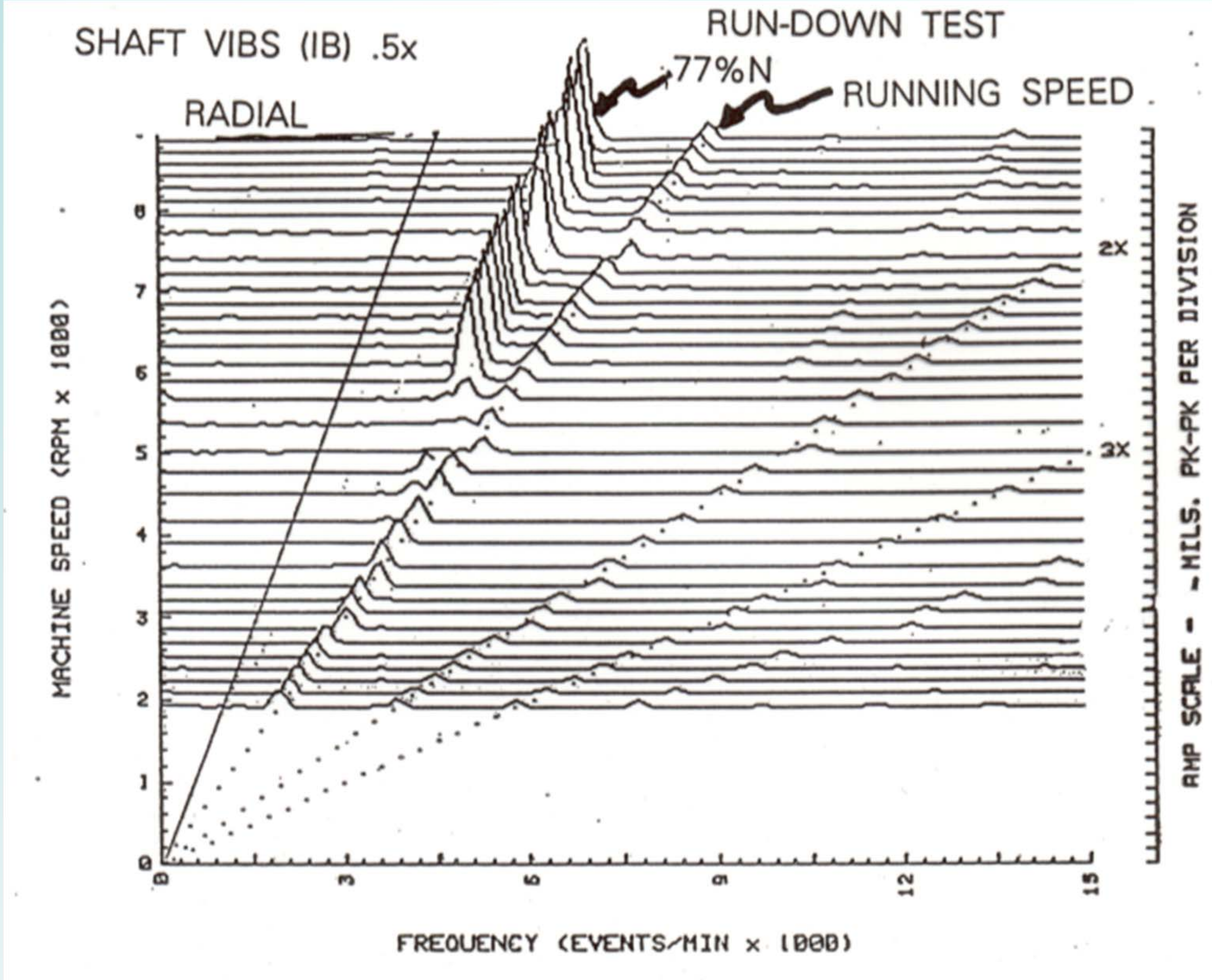
Stall Vibration vs. Speed



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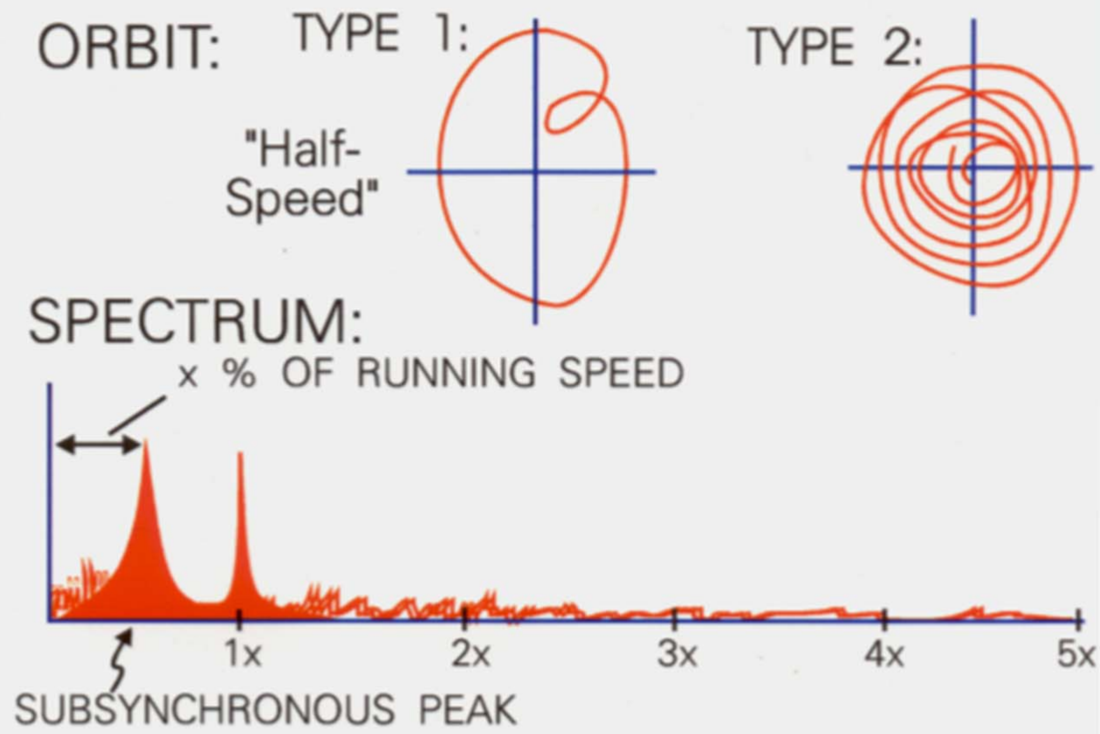
Vibration Problem No. 5: Subsynchronous (below 1x)



POSSIBLE CAUSES:

TYPE 1:

a) x = 40 TO 49% : BEARING INSTABILITY

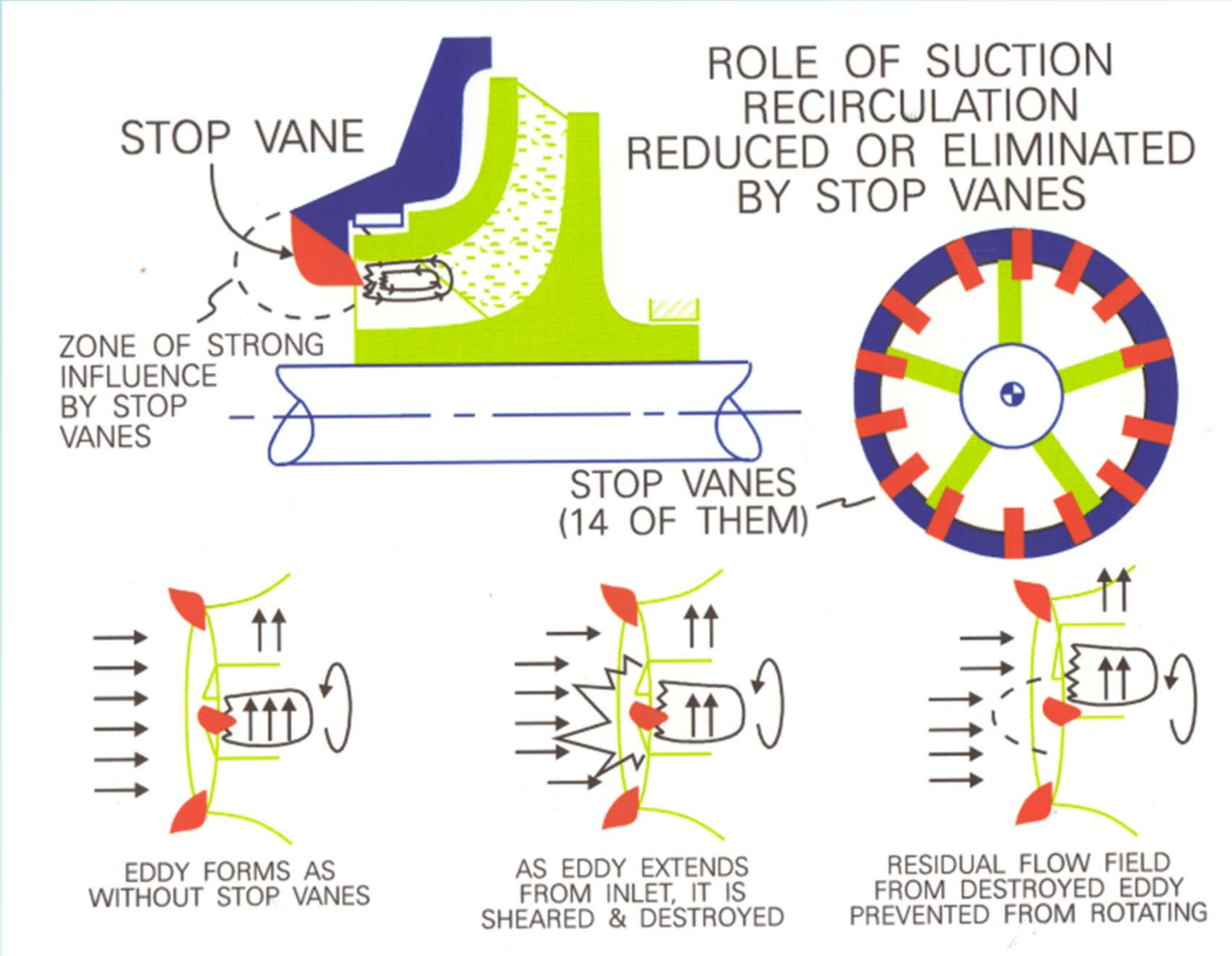
b) x = 50% EXACTLY: SEVERE RUB (OR EXACTLY 1/3 & 2/3)

c) x = 5 TO 30% : DIFFUSER STALL

TYPE 2:

d) x = 65 TO 95% : 1. IMPELLER STALL 2. SUCTION RECIRCULATION

e) GENERALLY HIGH "FLOOR" 0 - 1x : CAVITATION


ROLE OF SUCTION RECIRCULATION REDUCED OR ELIMINATED BY STOP VANES
STOP VANE
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

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Rotor Dynamics

- Critical Speeds & Mode Shapes
- Forced Response
- Stability

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Rotordynamics Is Best Evaluated by Computer

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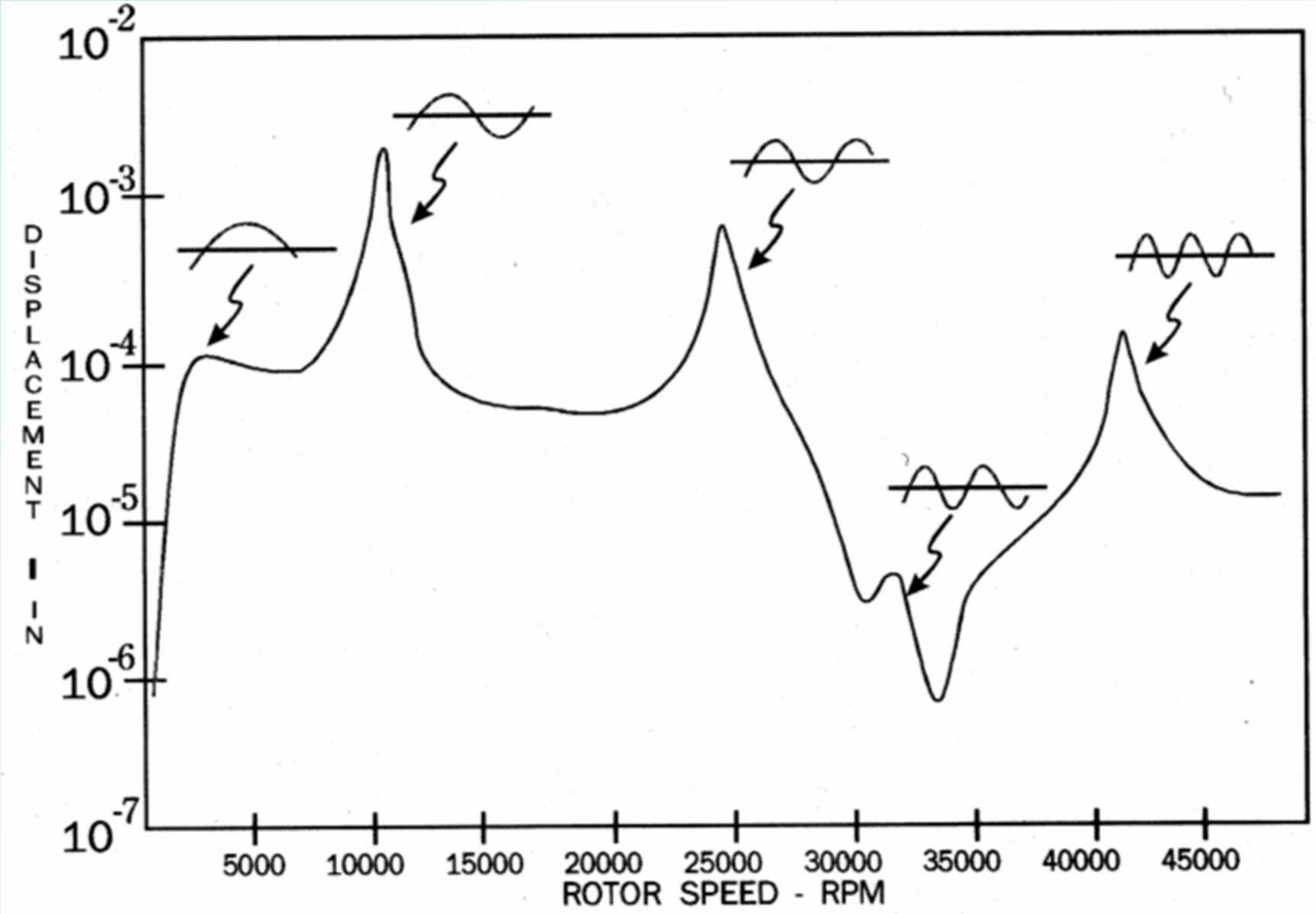
Typical Rotor Vibr. Response vs. Speed Exhibits Several Natural Frequencies



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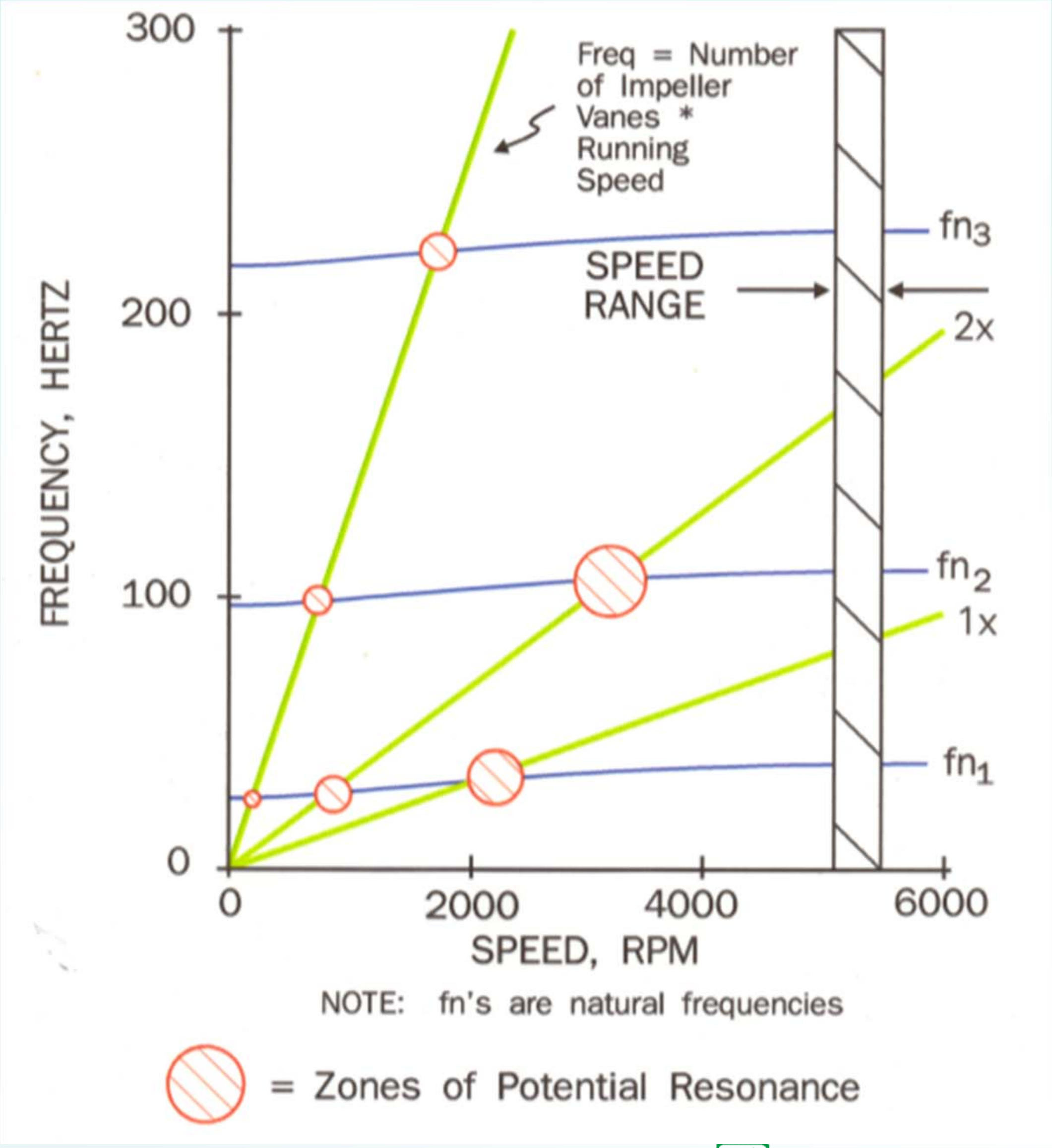
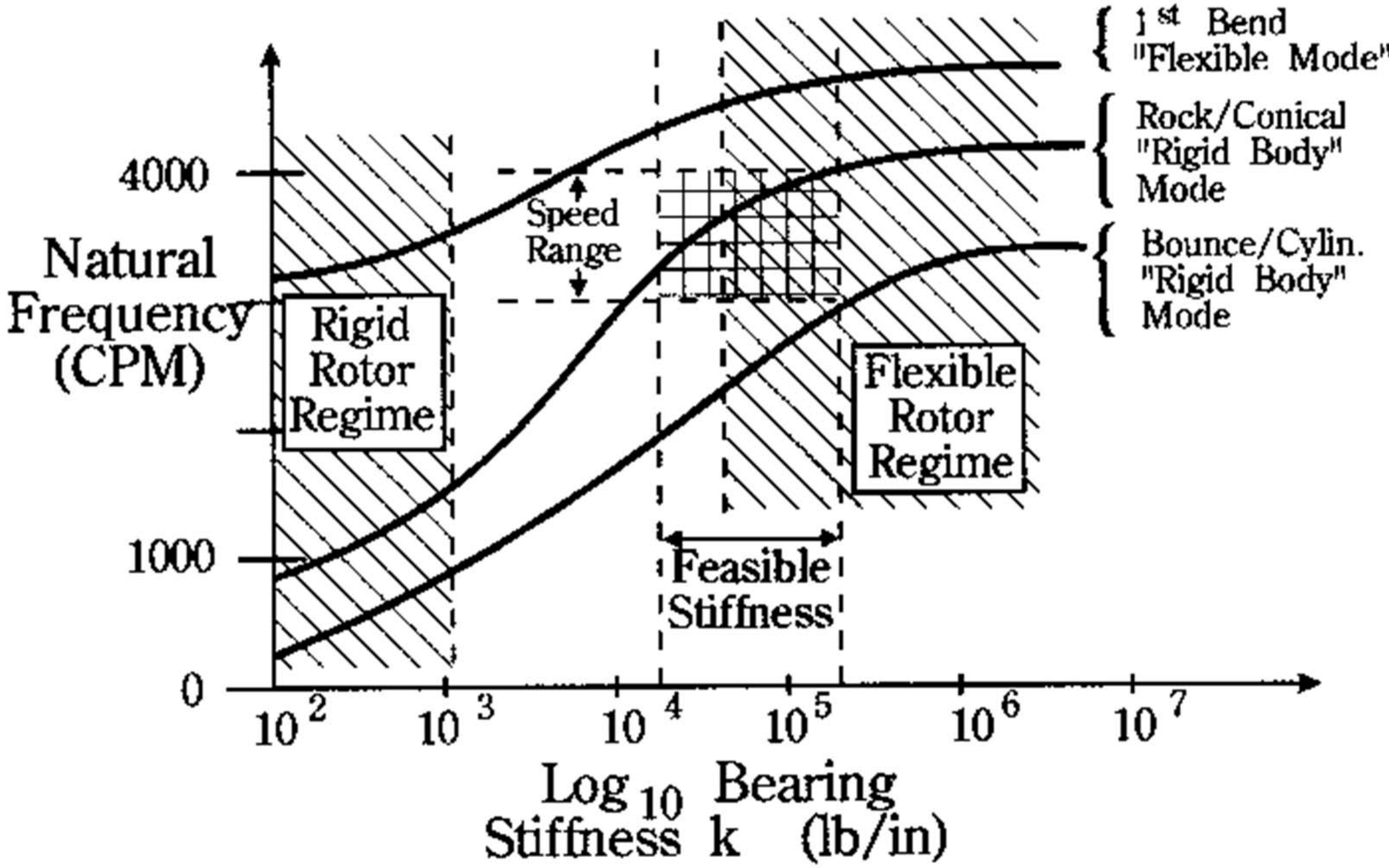
Avoiding Resonance w/ Campbell Diagram

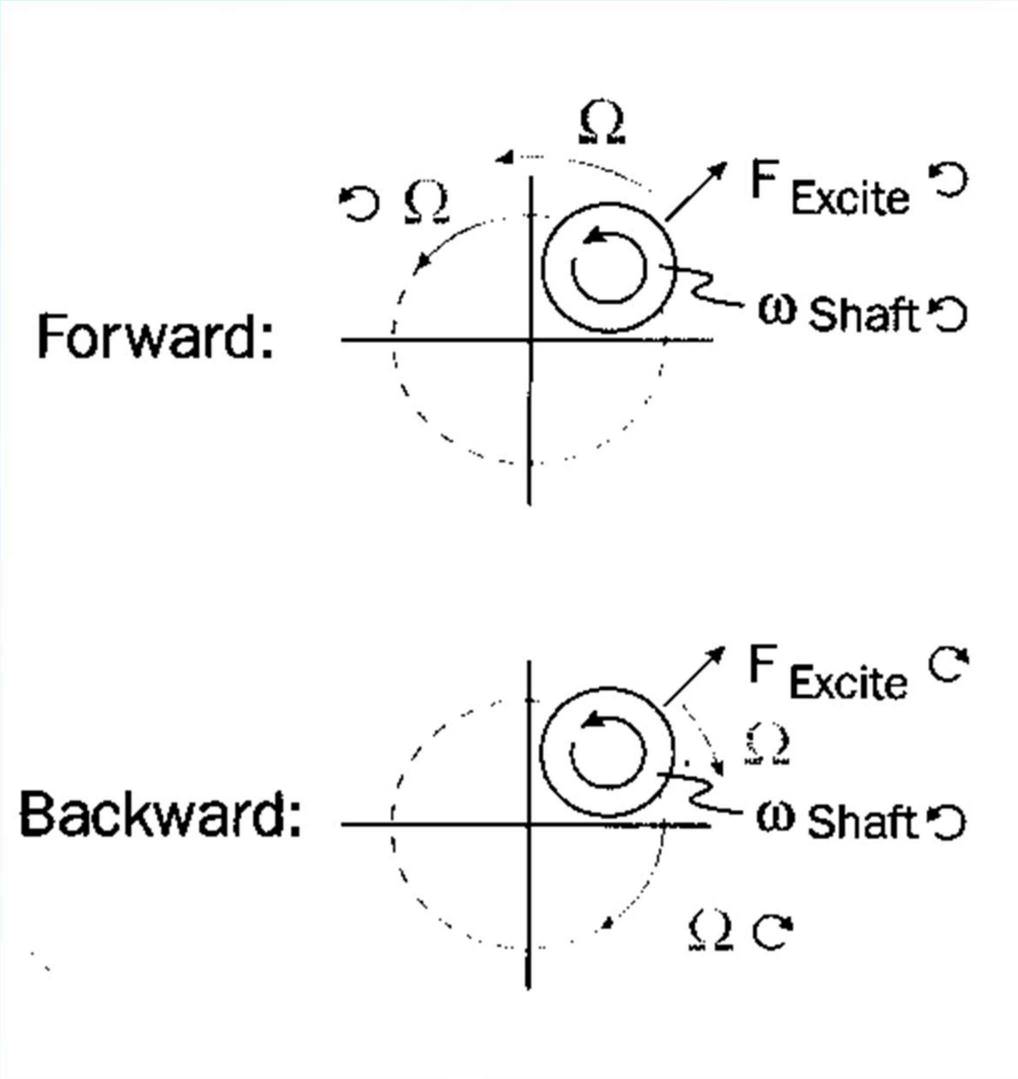


Rotordynamic Critical Speed Map



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



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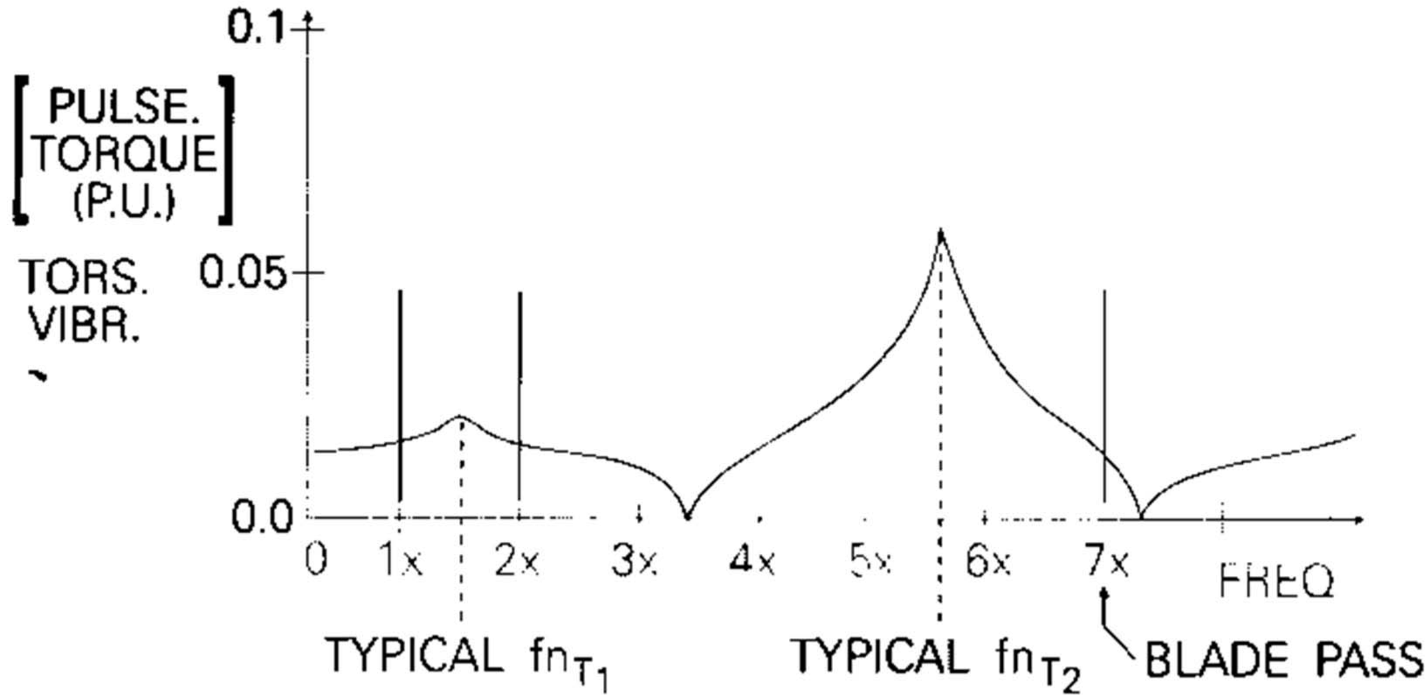
Typical Torsional Critical Speeds and Worst Case Excitations



- VALUES @ MIN. FLOW (BEP IS 2x - 5x LOWER)
- VALUES MAY VARY BY ~ +/- 0.05
- SOME EXCIT. @ VANE PASS x2, x3, . . ., xi $\left(\sim \frac{0.05}{i}\right)$
- VFD's: LINE FREQ, 2x LINE FREQ, 6x/12x/18x N_{MOTOR} x2/POLES

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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.)

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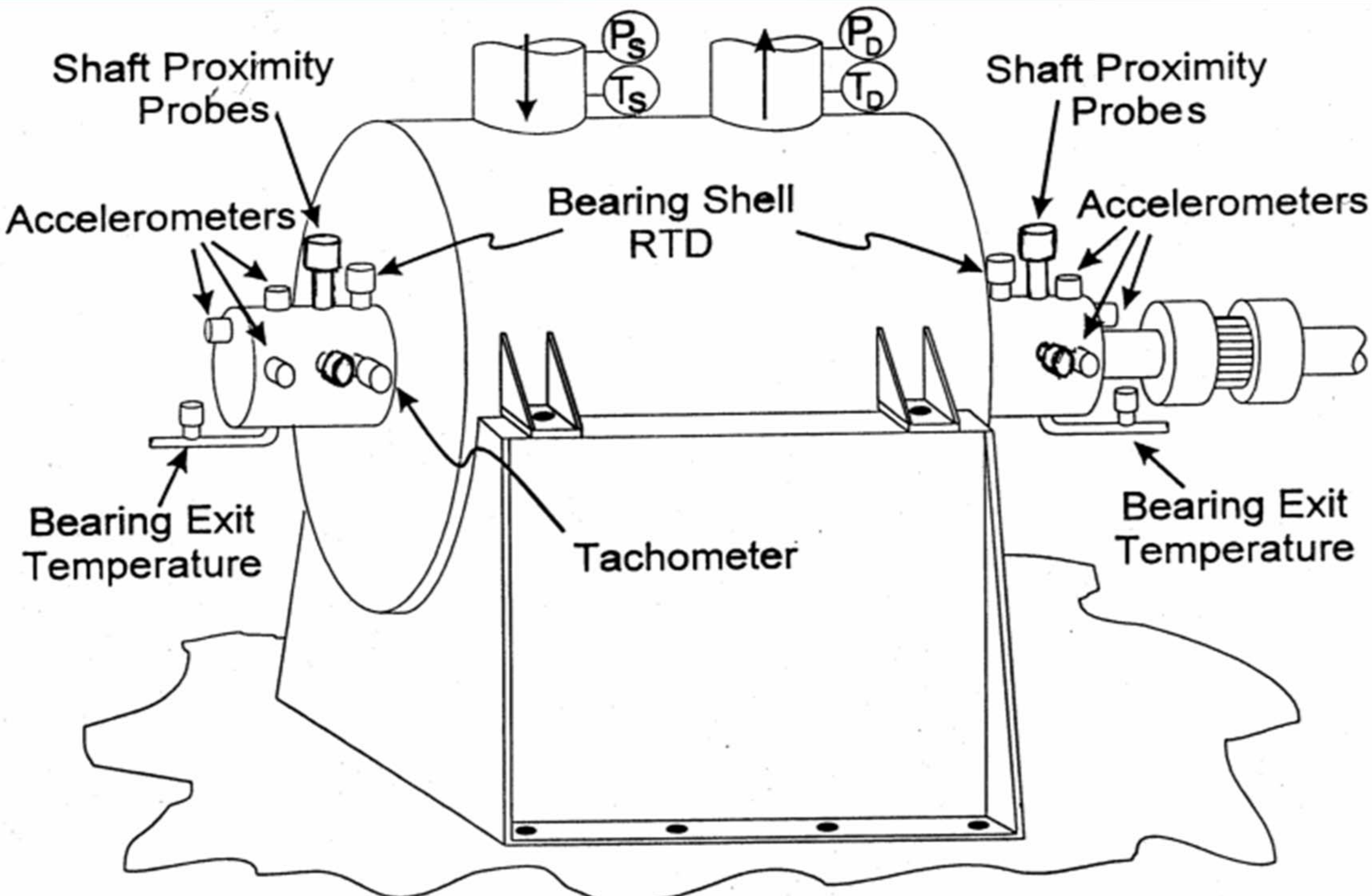
Instrumentation Options



Turbomachine Well Instrumented According to Current Practice

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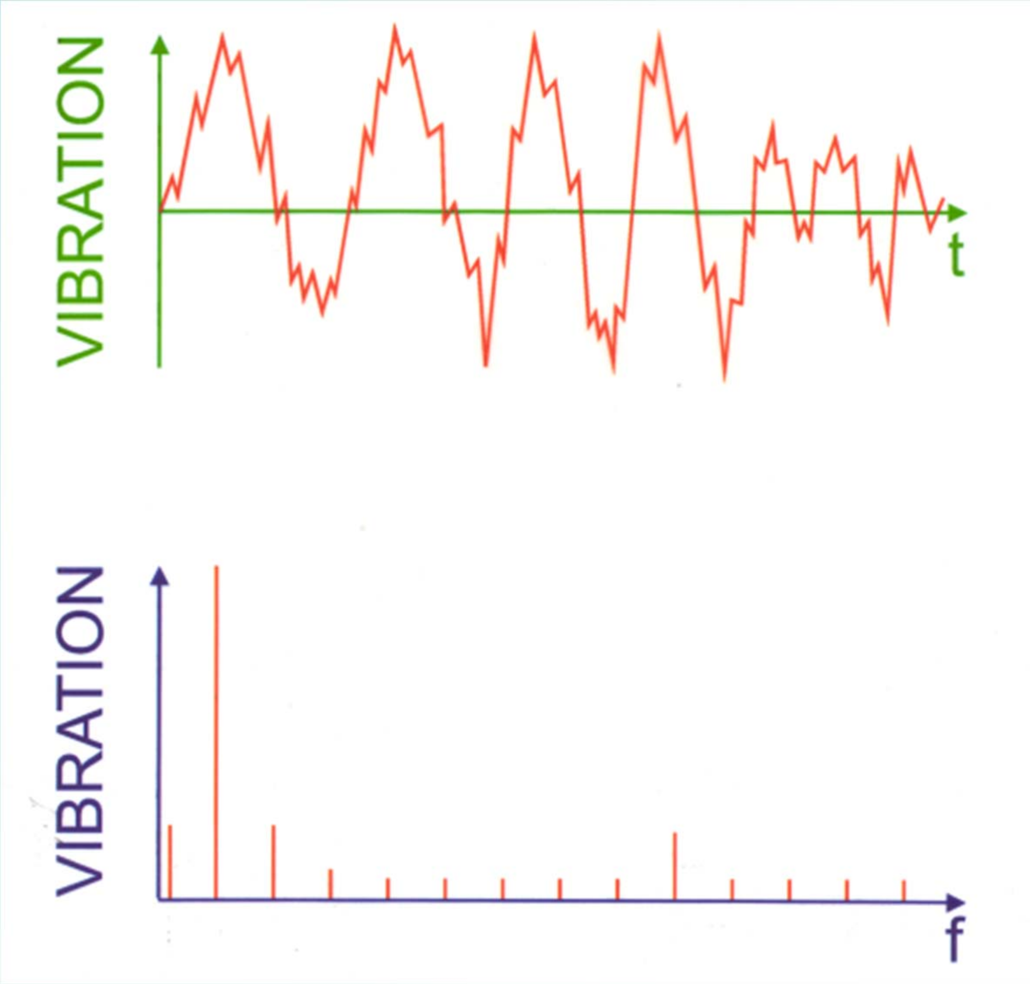
Converting Vibration vs. Time to Vibr vs. f with “FFT”: Fast Fourier Transform



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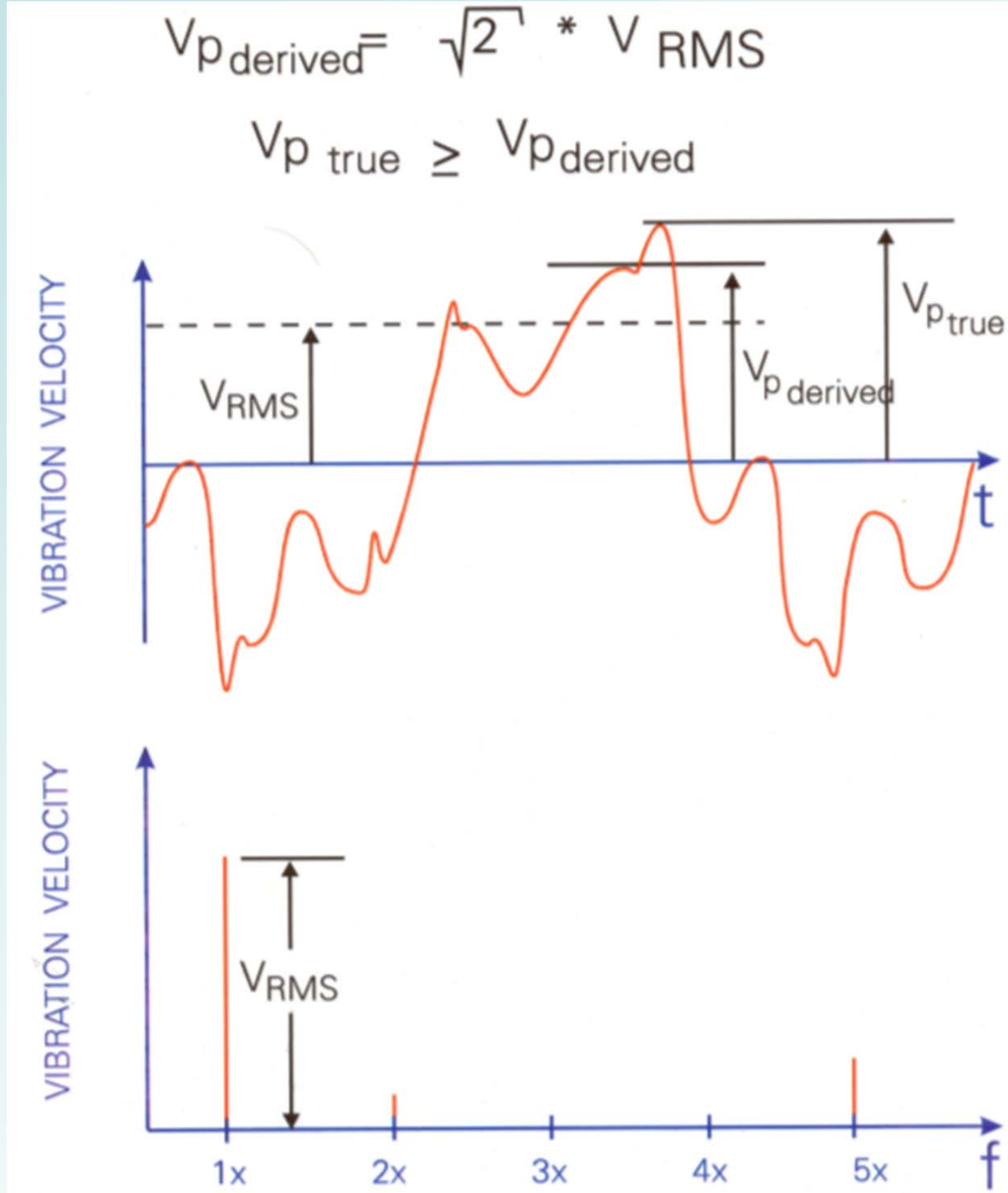
Meaning of RMS vs. Peak Vibration



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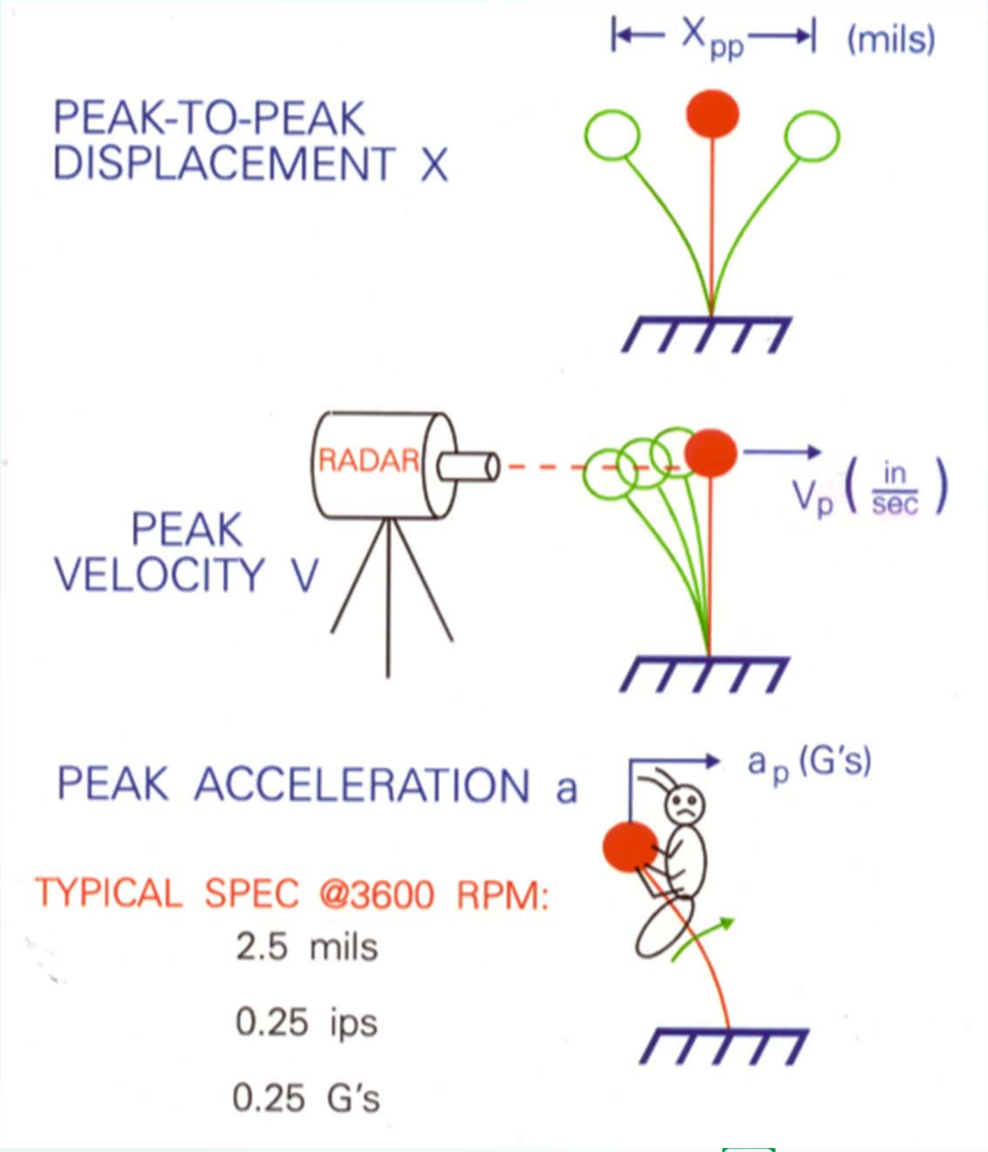
Vibration Measure Numbers



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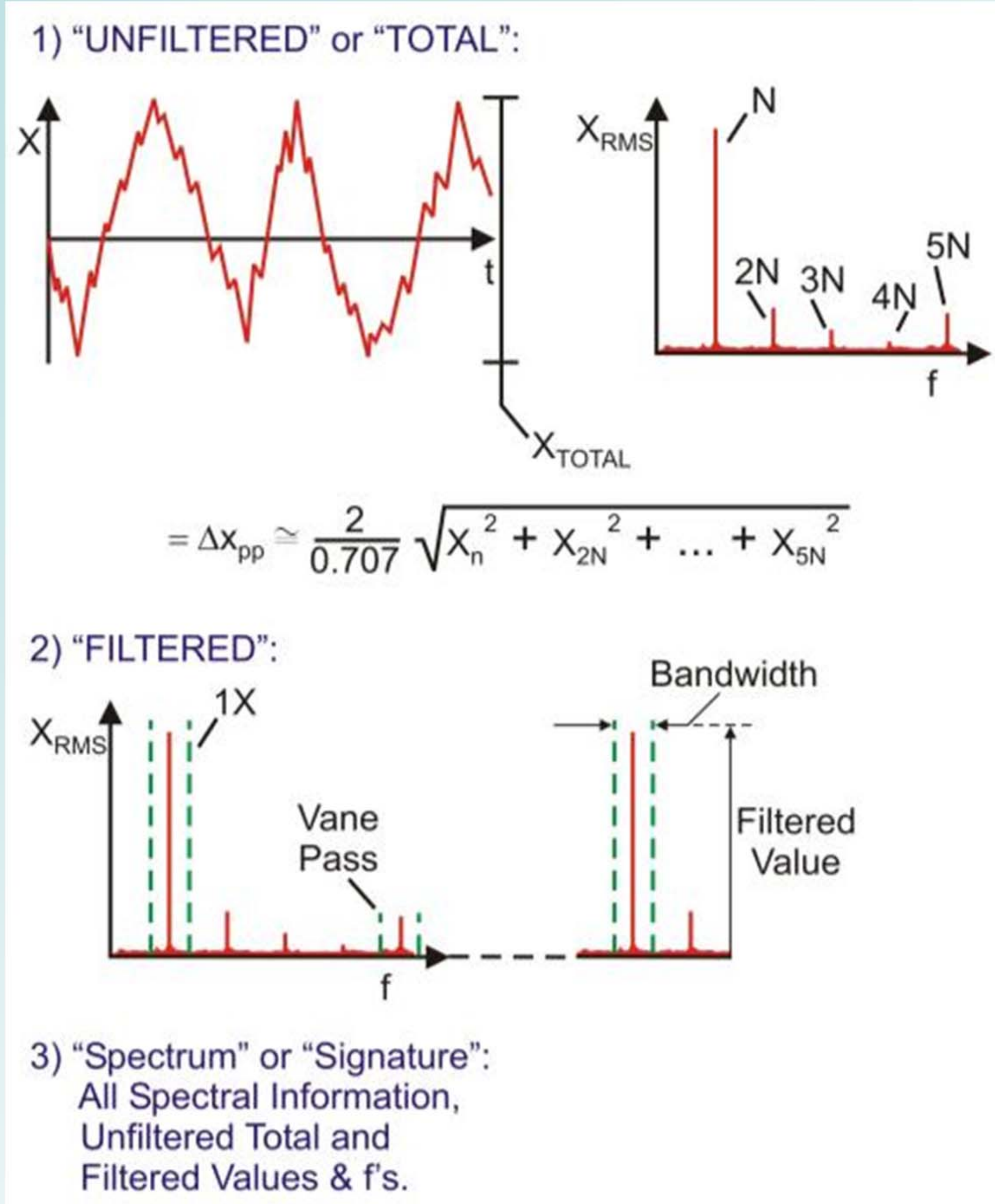
Vibration Measurement Format



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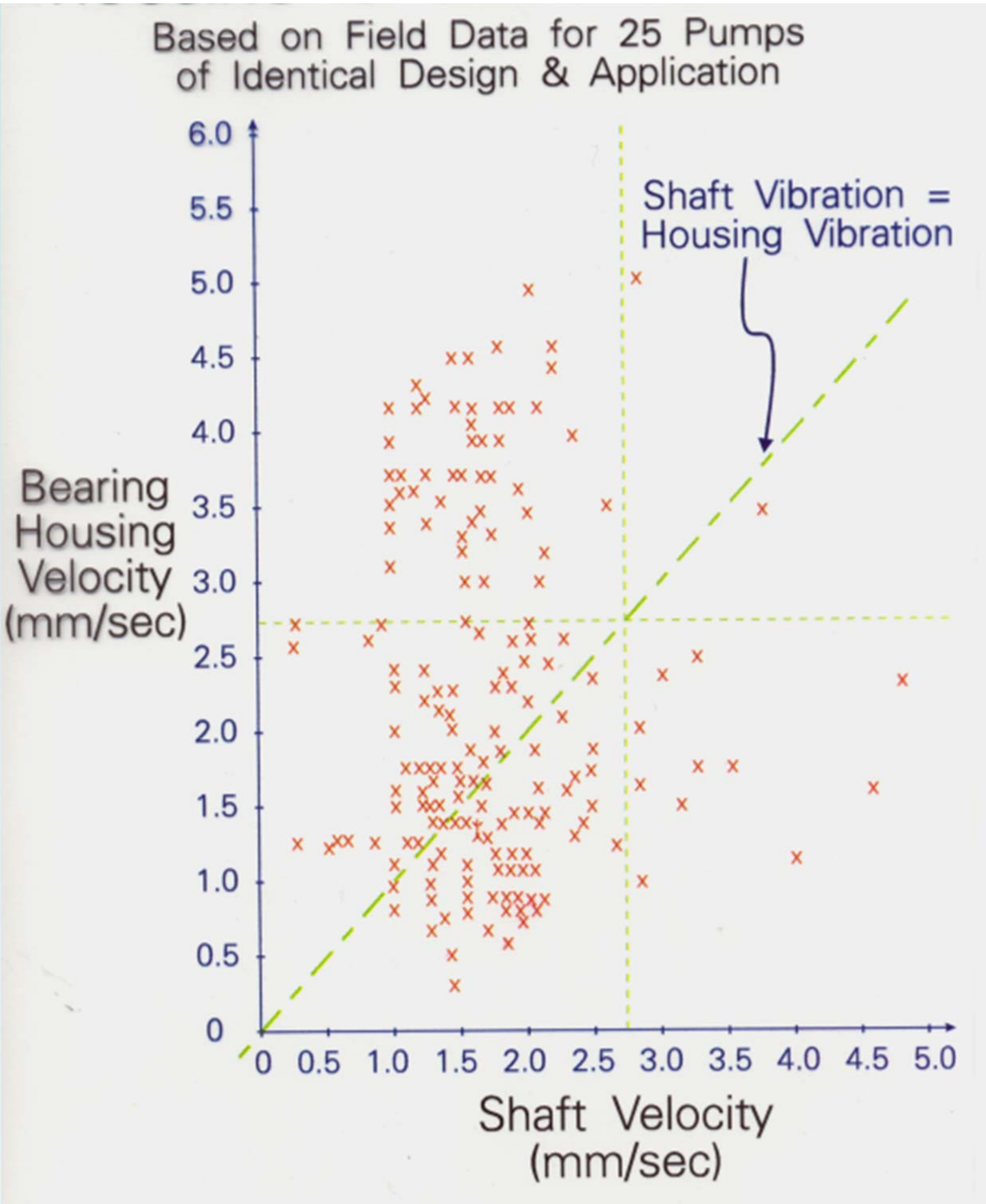
Housing vs. Shaft Vibration: Which Should Be Measured?



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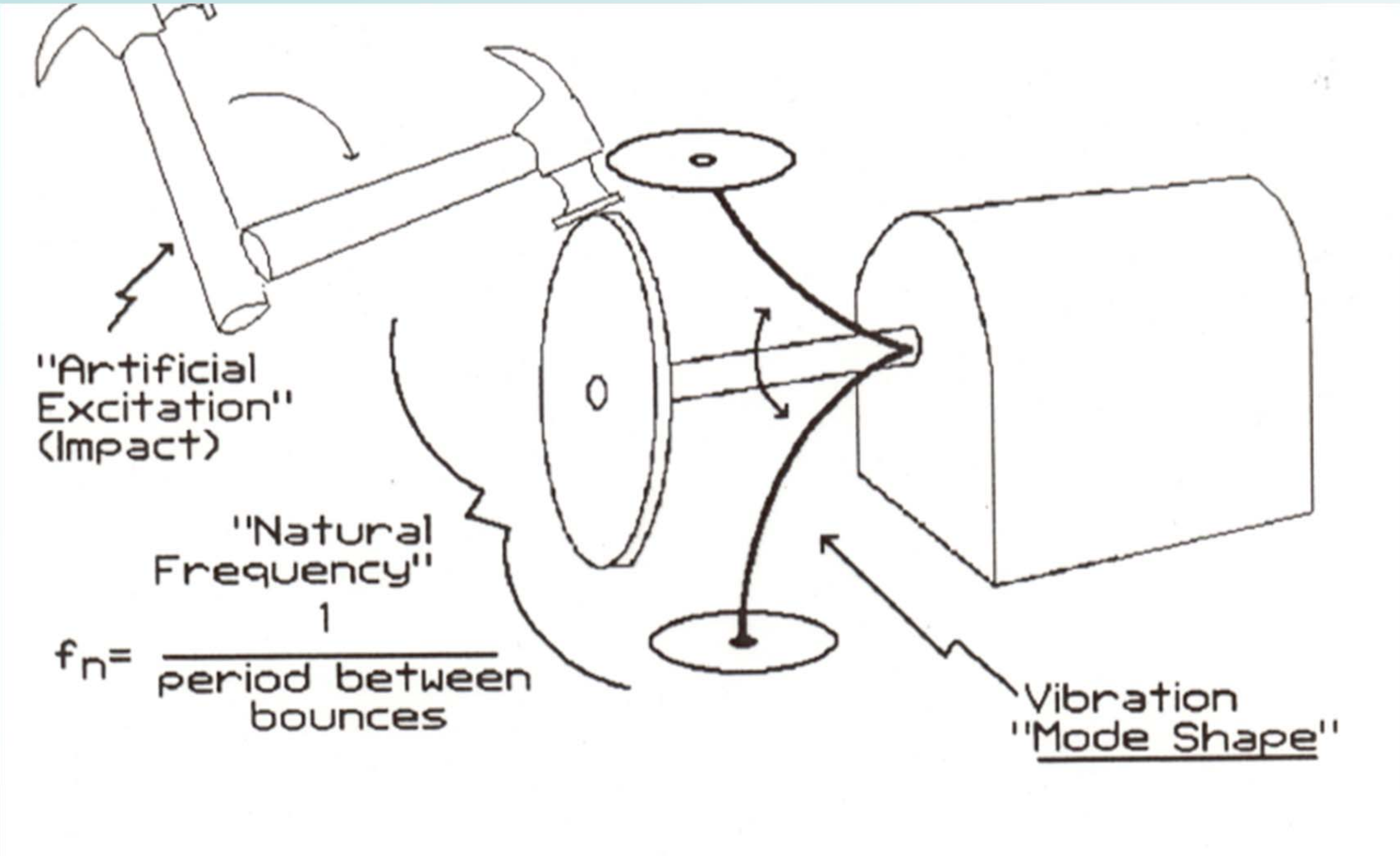
Natural Frequency



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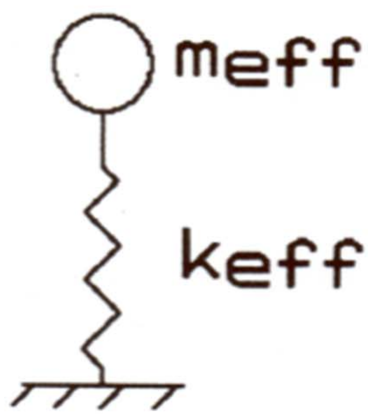
Approximating Natural Frequency

$$f_{n_1} = \frac{60}{2\pi}\sqrt{\frac{k_{effective}}{m_{effective}}}$$



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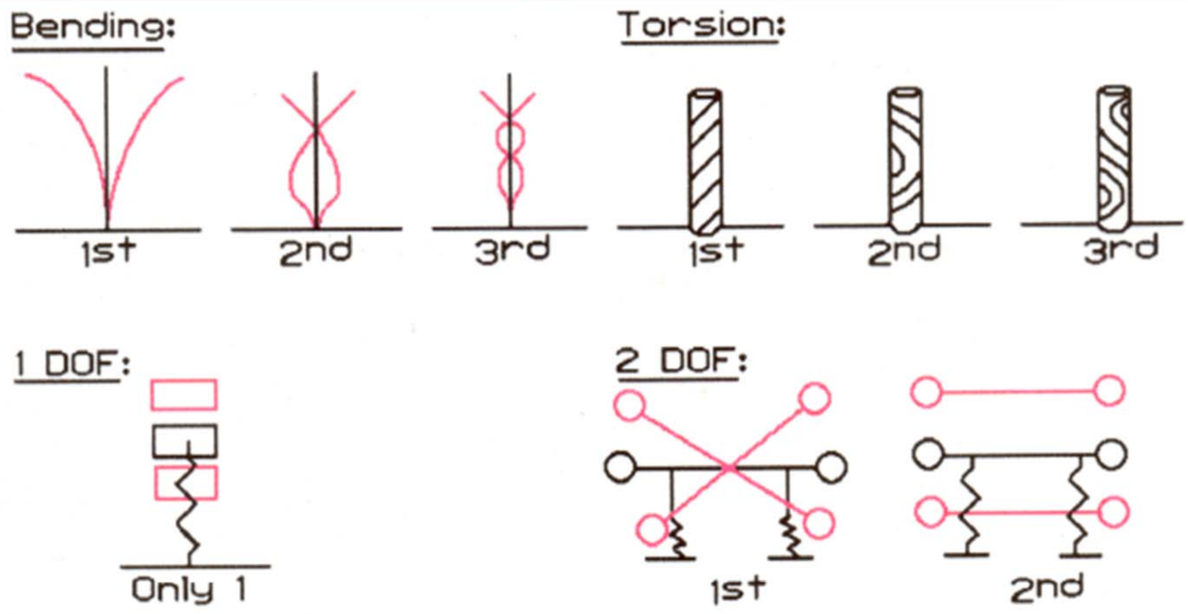
Vibration "Mode Shapes"



Orders of Excitation

of Excitation Pulses per Rev.

Resonance

When Order of Excit * Rotor Speed = Mode Freq.

$$\Longrightarrow \frac{\omega}{\omega_{nd}} = 1.0$$

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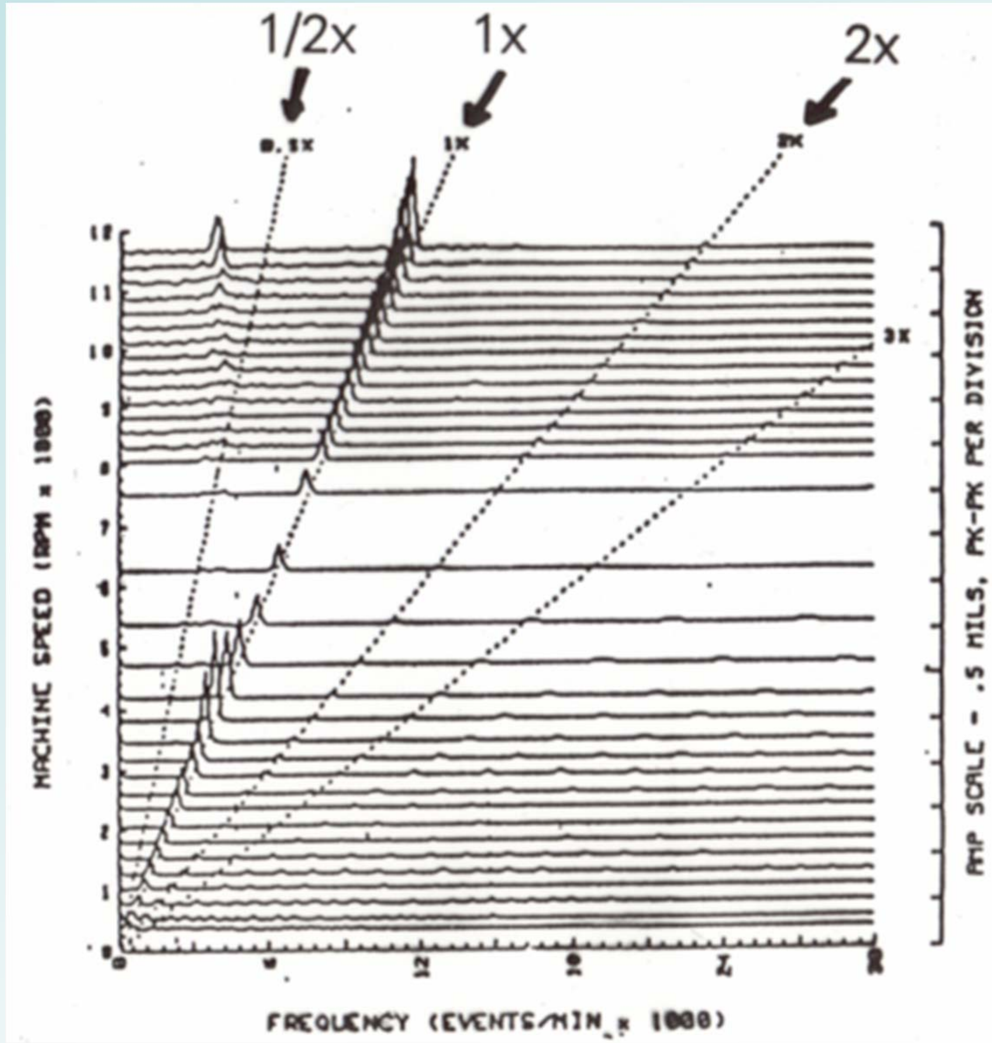
Finding Nat. Freq. Resonances with a "Waterfall" Plot of Stacked FFT's from Test



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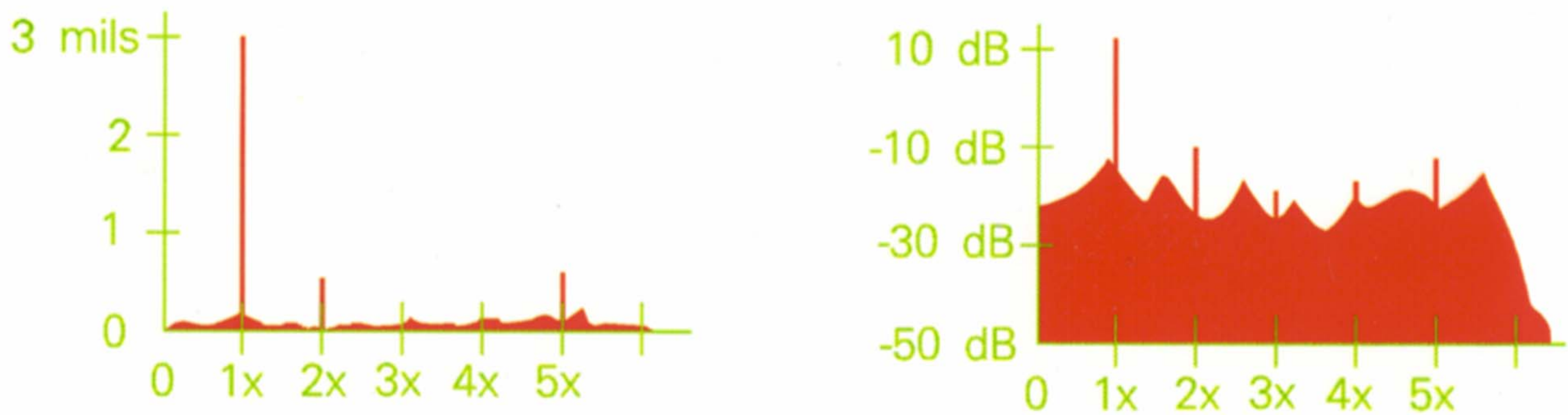
Approximate Identification of Natural Frequencies



→ PLOT VIBRATION AS dB

→ PLOT FREQUENCY AS LINEAR

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A Modal Field Test In-Progress on an Operating Pump

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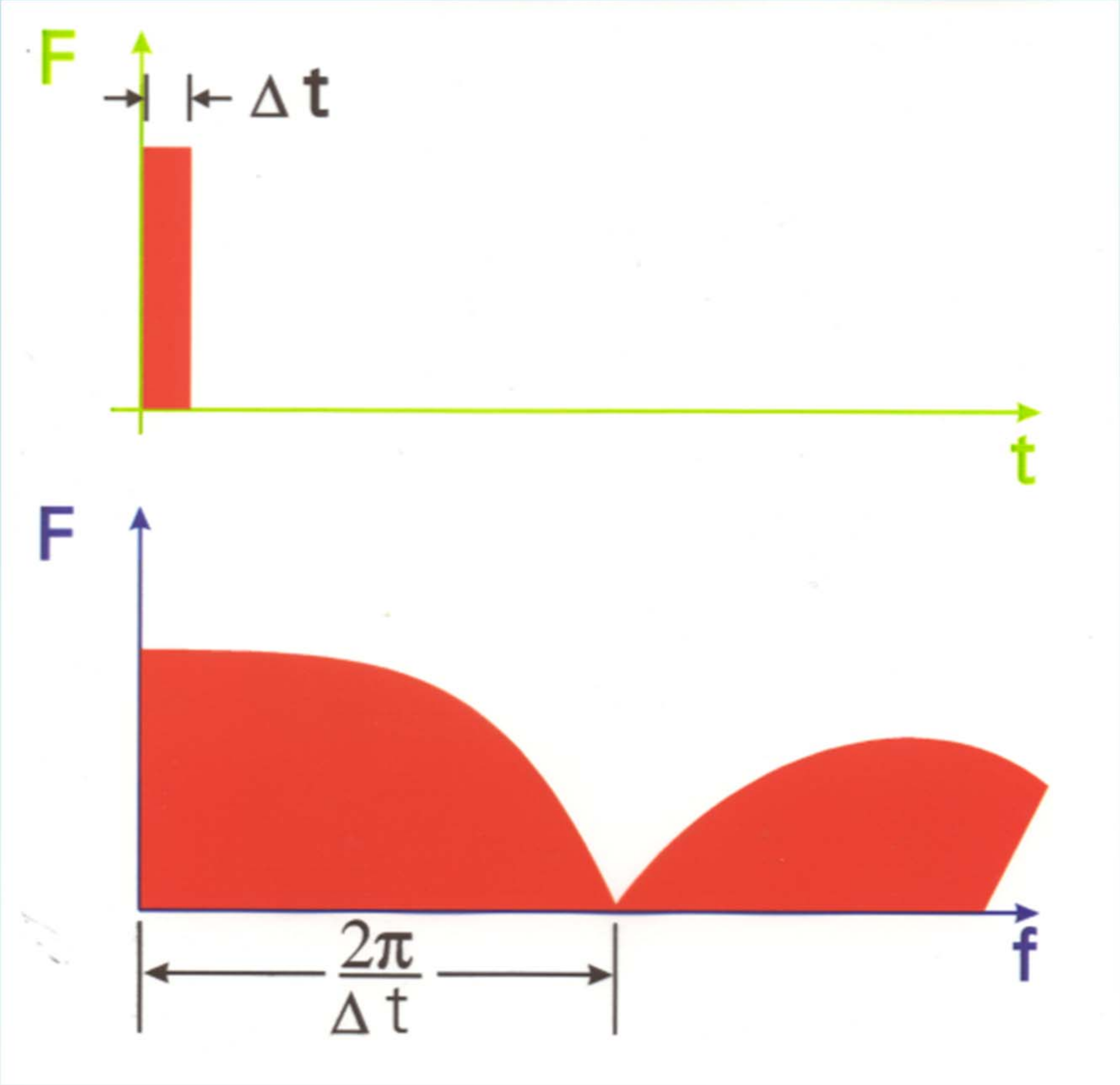
Short Force Application Excites Wide Frequency Band



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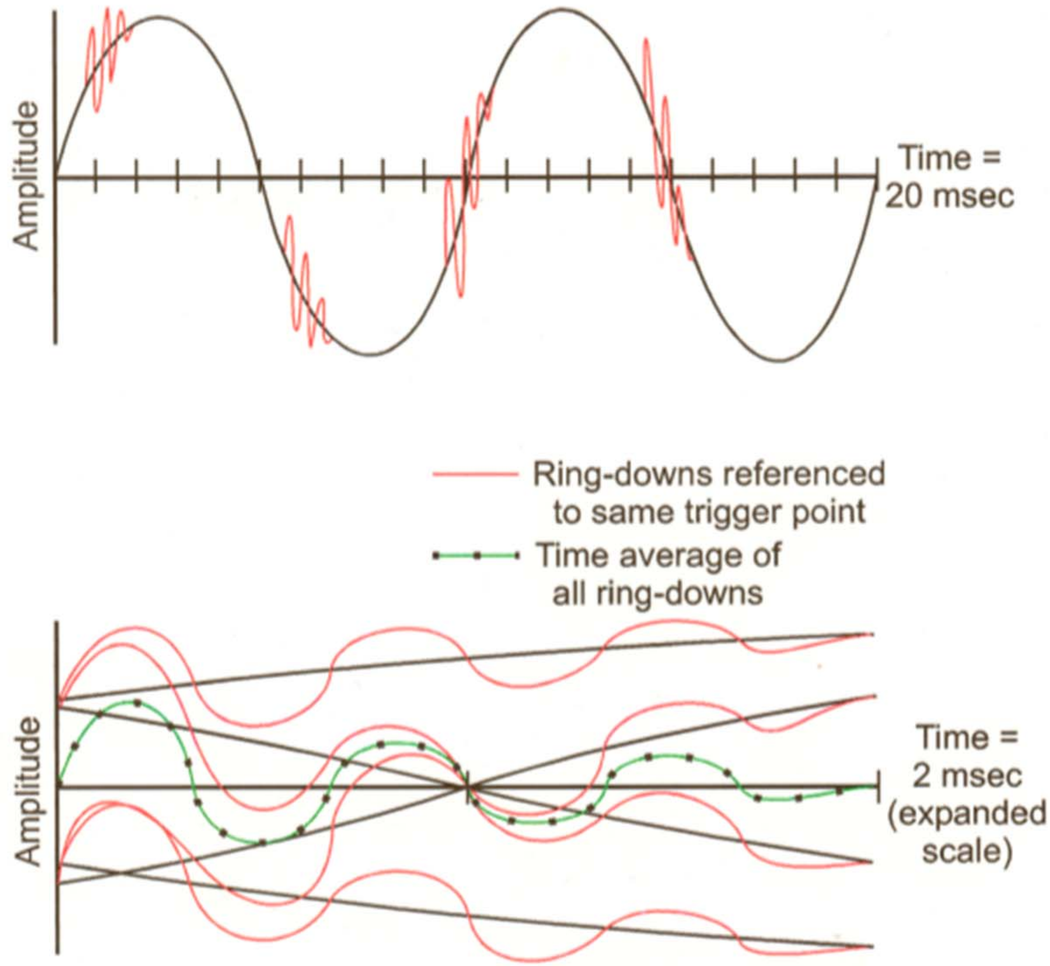
Principle of Time Averaging



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Rotor Impact Testing
Case History

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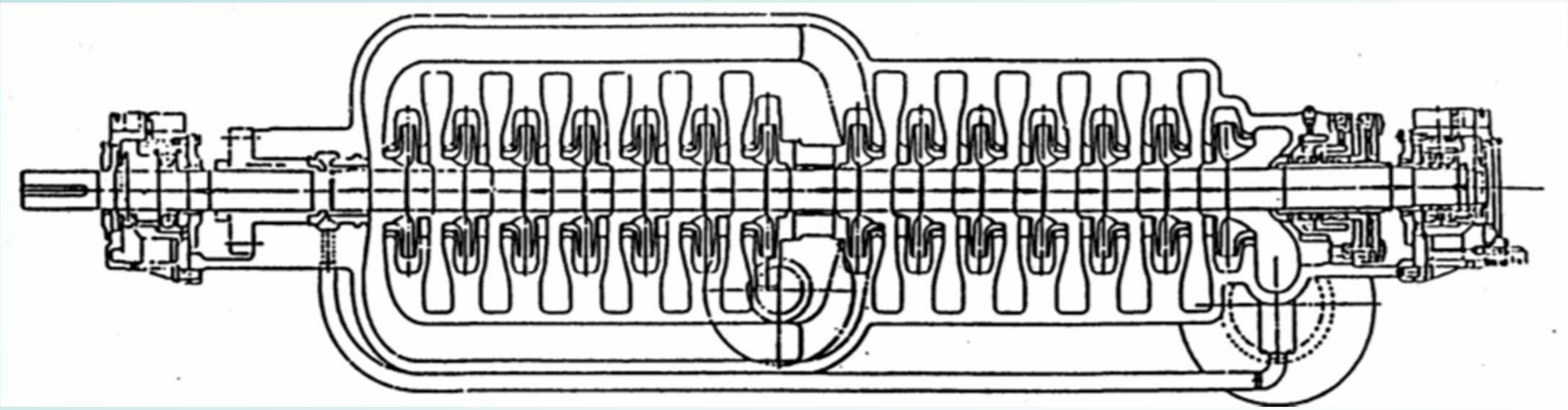
14 Stage
Axially Split Case Pump



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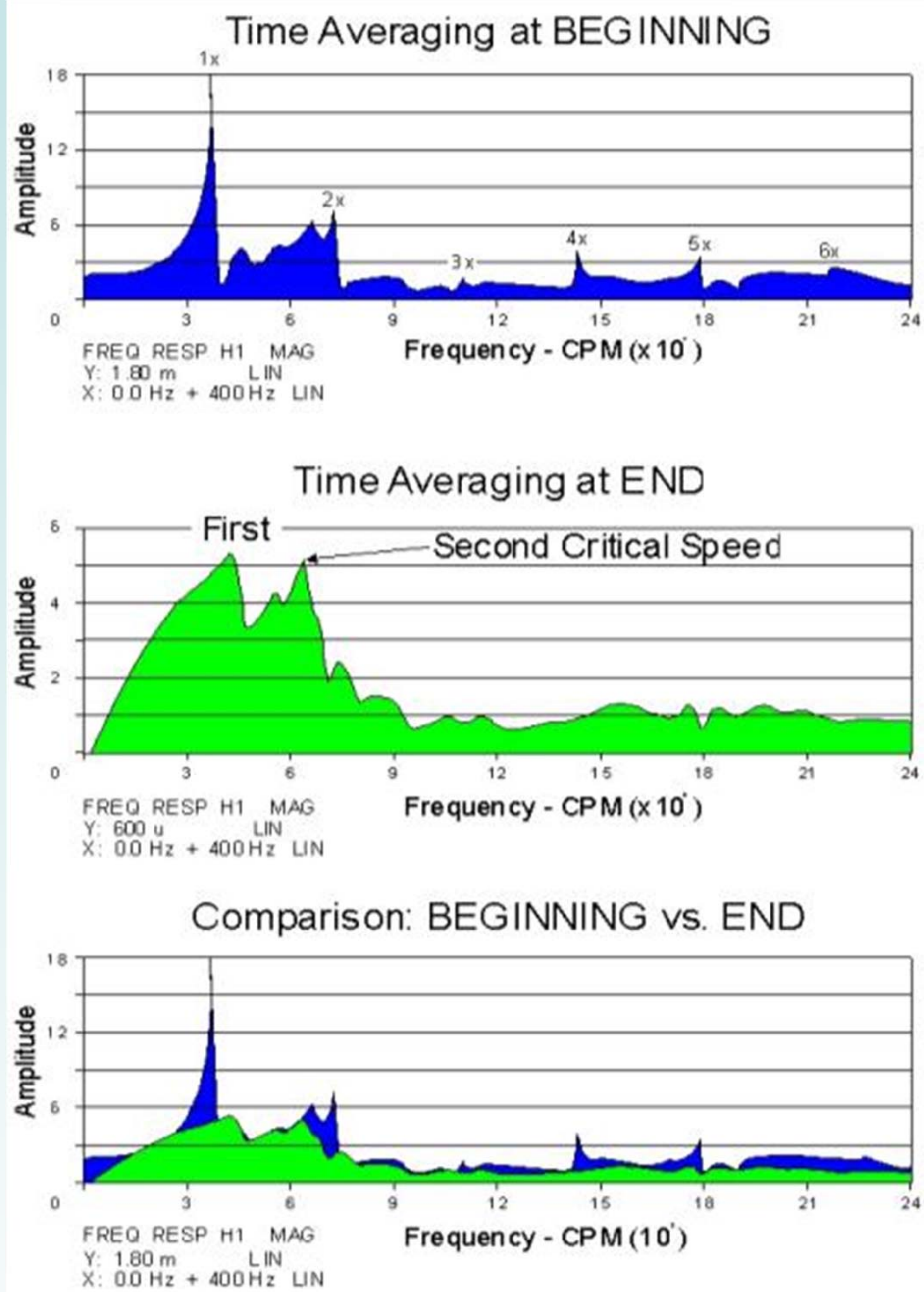
<u>Time Averaging Field Example:</u> Watch the Harmonics Disappear

In the End, Natural Frequencies Are Clear



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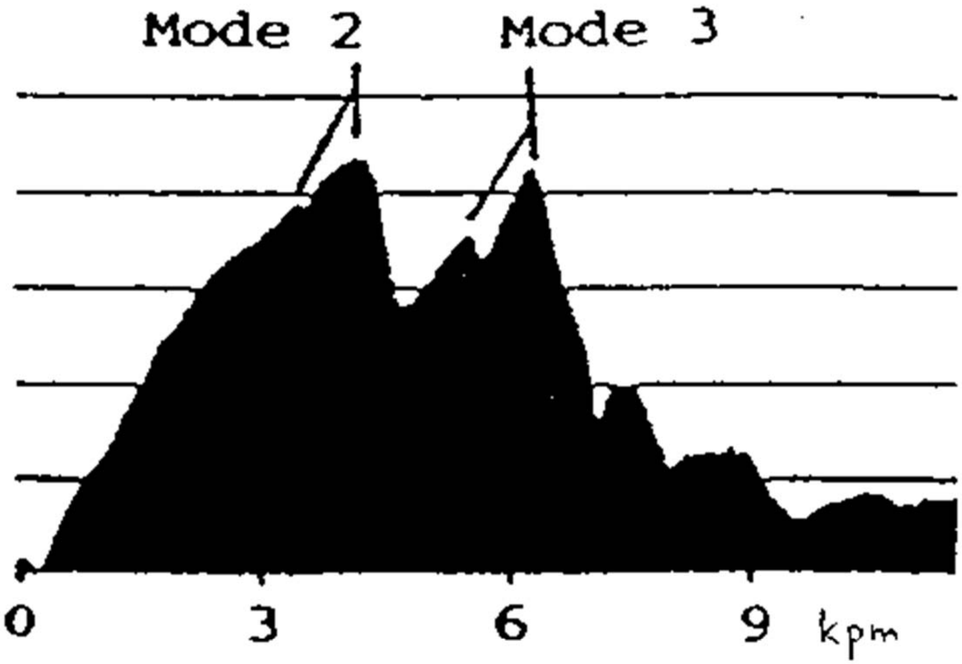
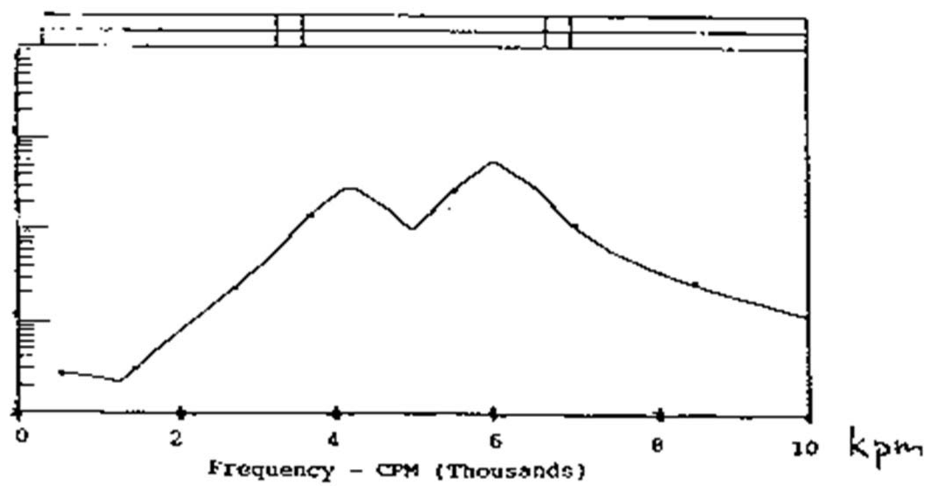
Impact Test

Critical Speeds of 14 Stage Pump



Modal Test Results for Shaft of Pump
Figure 10 after 200 Impacts.

Rotordynamic Result



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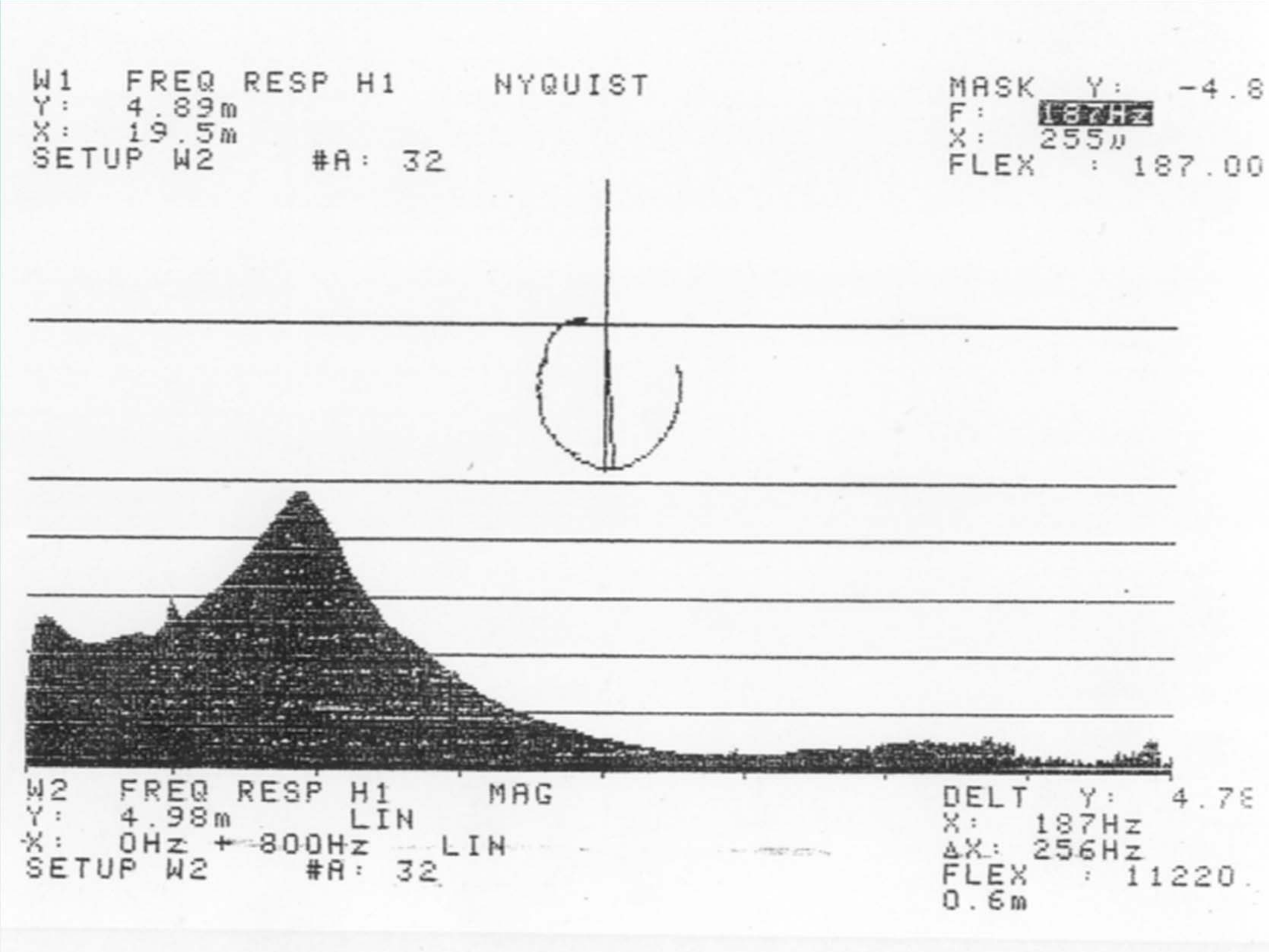
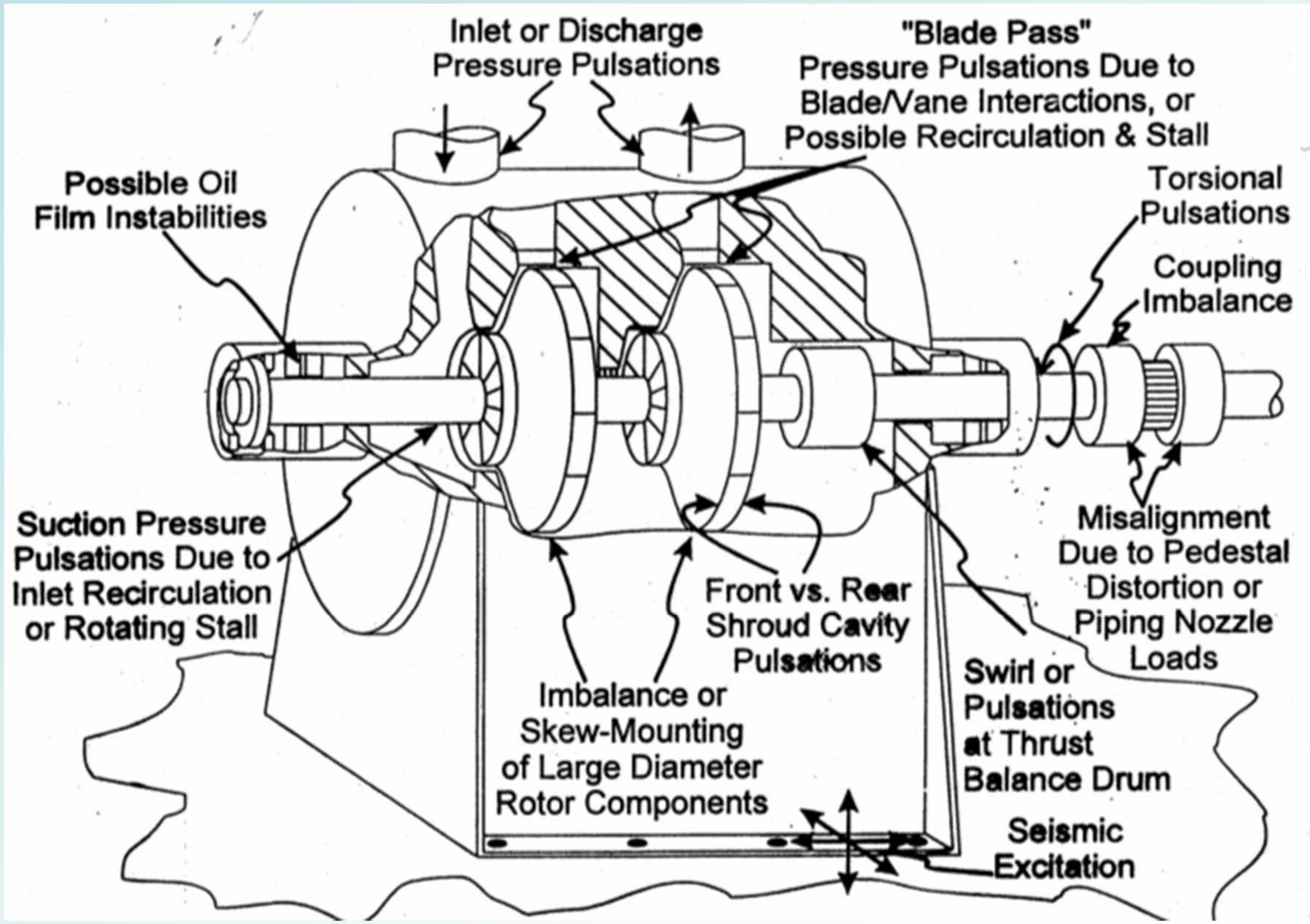
First Critical Speed, Dbl. Suction SS Pump



Sources of Damaging Forces in Centrifugal Pumps



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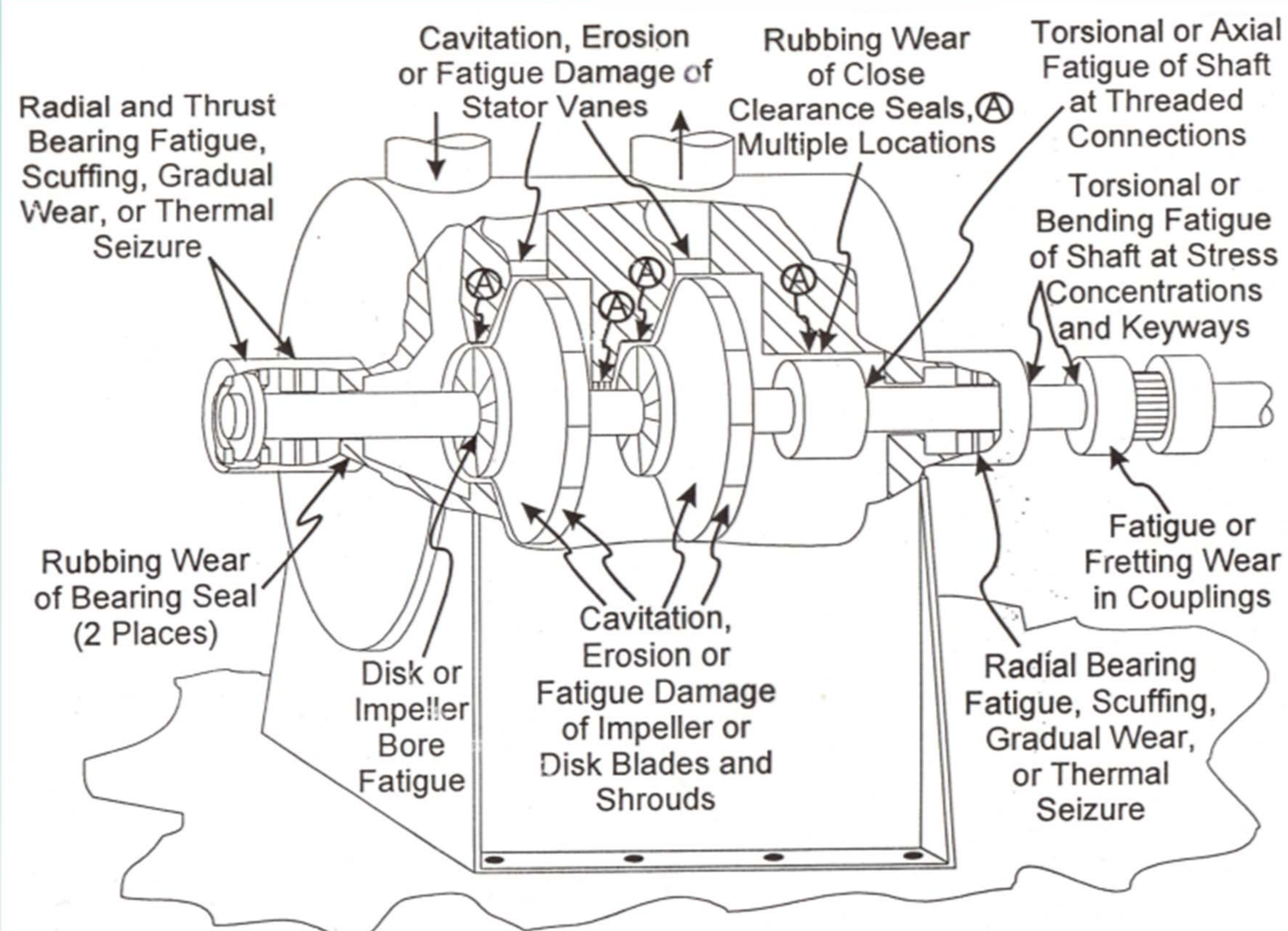
Typical Failures in Centrifugal Pumps



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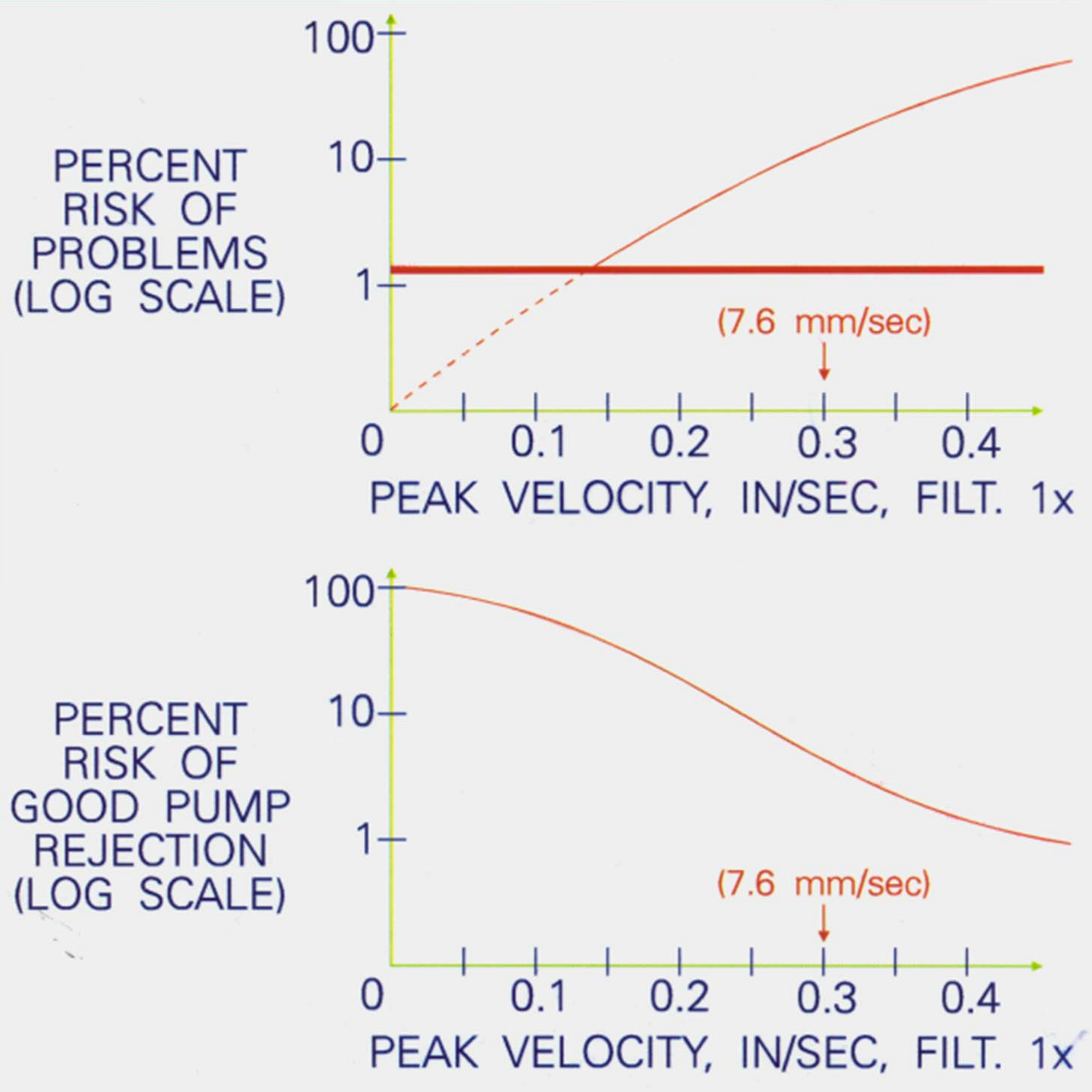
Risk of Accepting Bad vs. Risk of Rejecting Good



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Stuffing Box & Bearing Skew Case Histories

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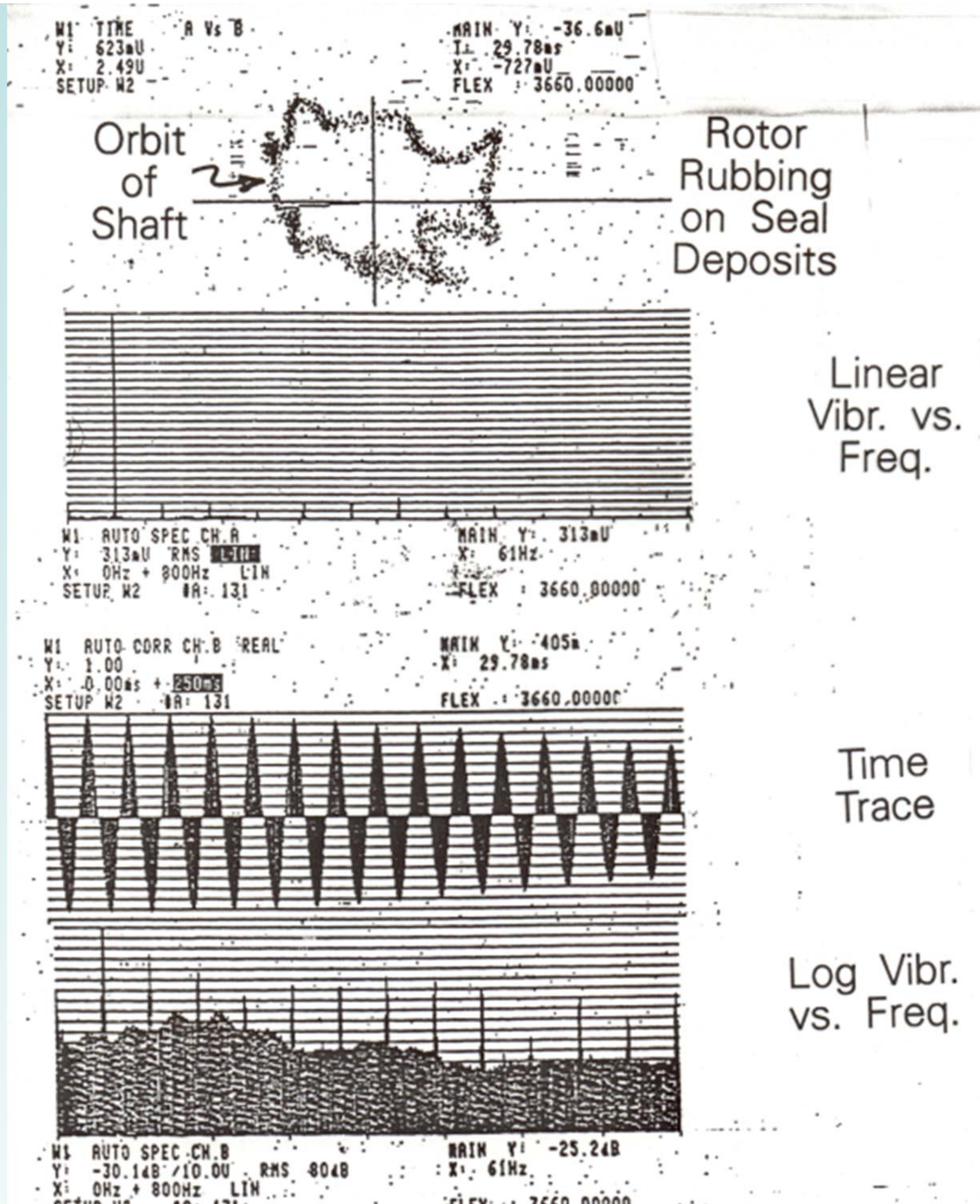
Cartridge Pump Seal Rub Example



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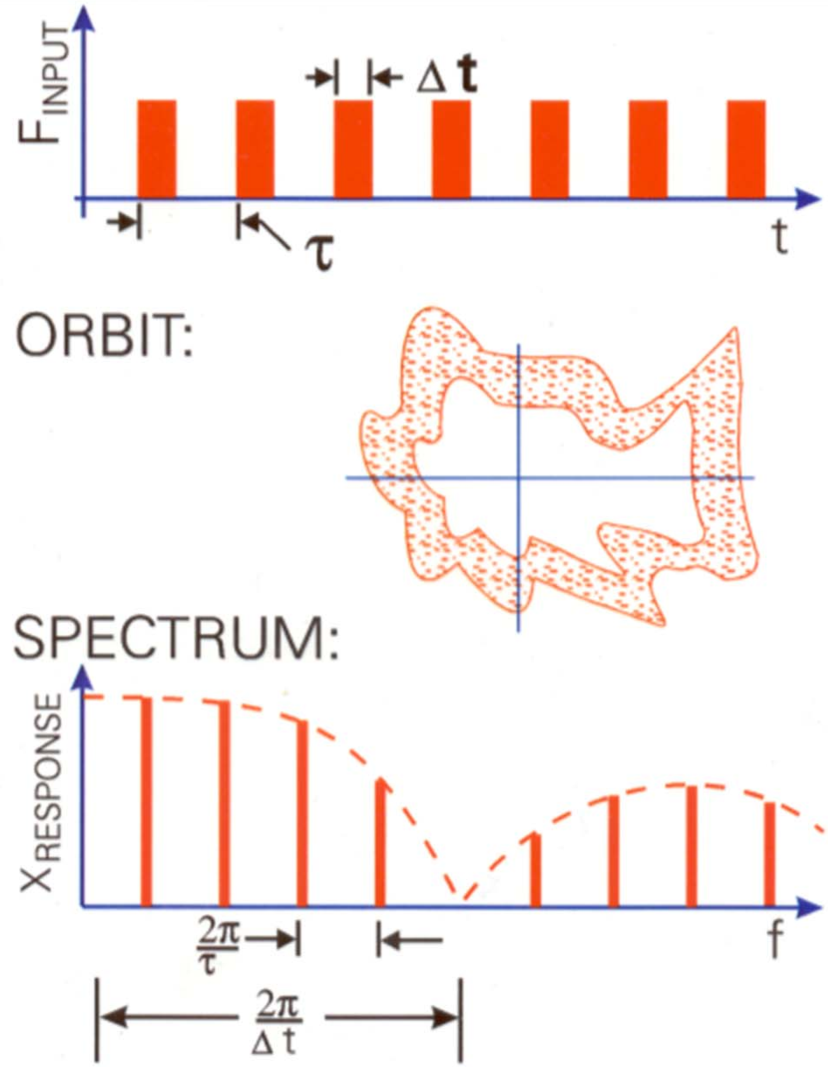
Vibration Problem #6: Repeating Impacts



POSSIBLE CAUSES:

a) RUB CONSISTENTLY OVER PART OF ORBIT
b) PINCHED OR MISALIGNED SEAL
c) JOURNAL BEARING WORN TO OVAL SHAPE

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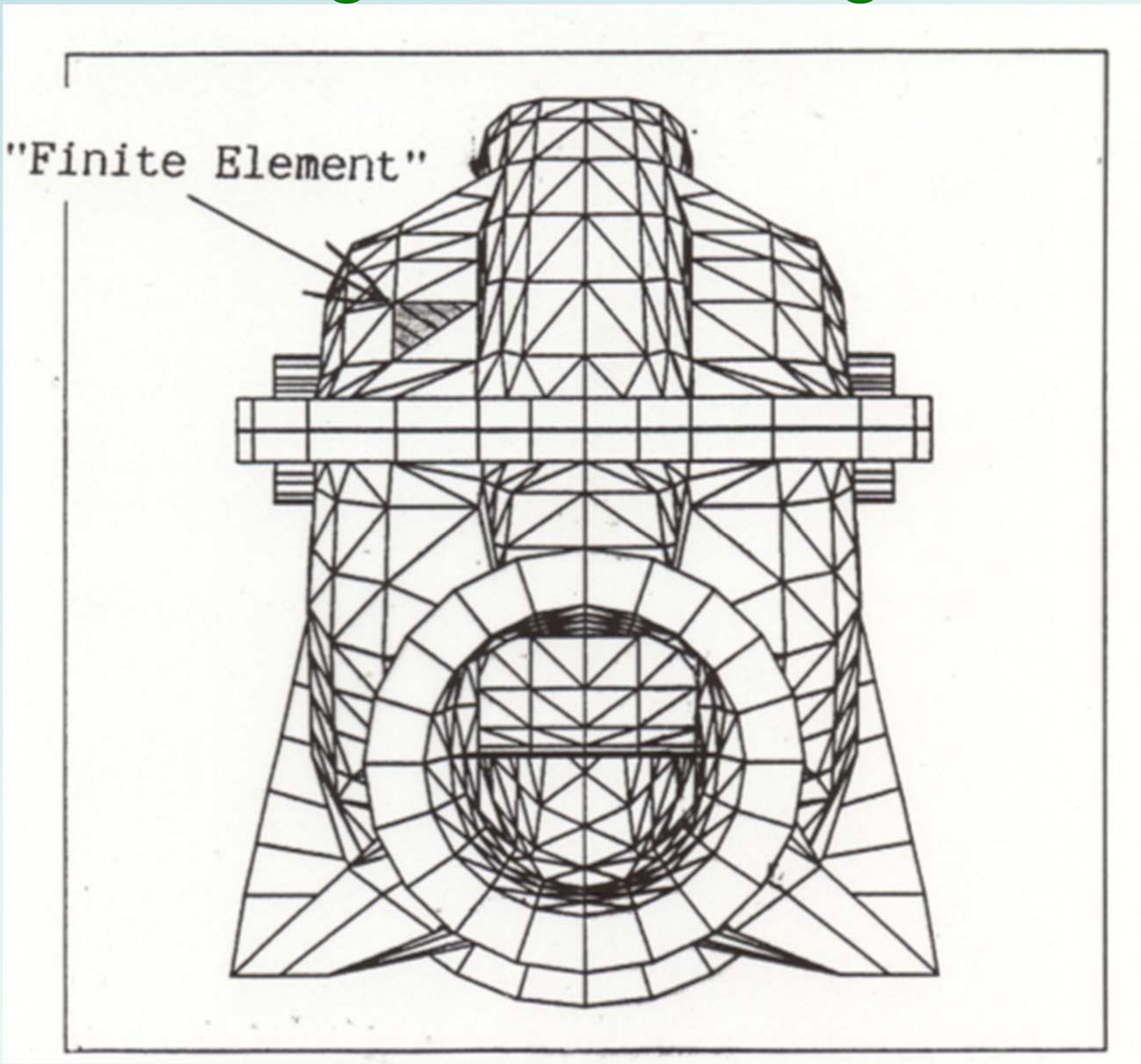
Double Suction Pump "Breathing" Tilts Stuffing Boxes



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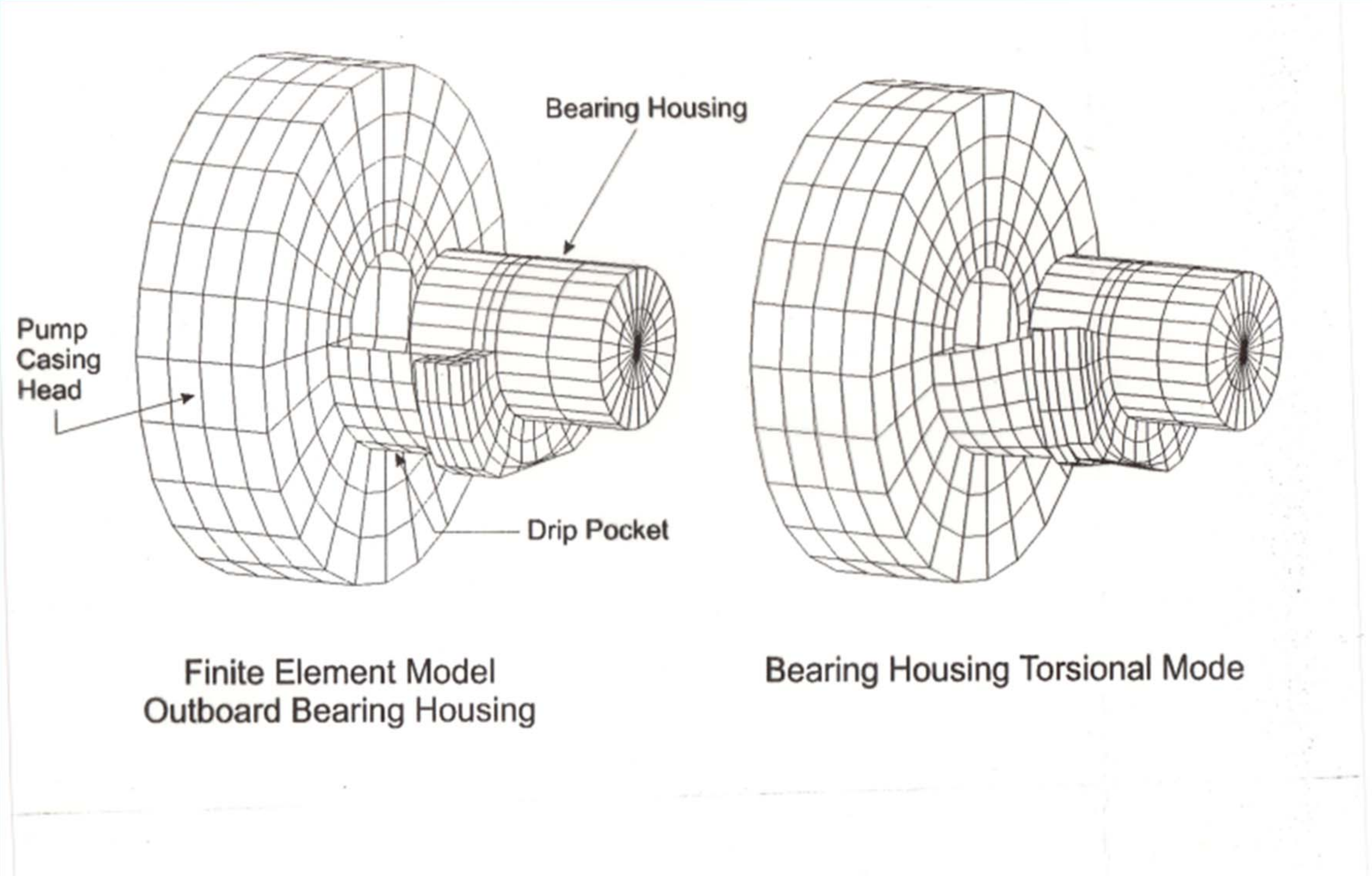
FEA Model of Twisting Pump Bearing Housing



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Vertical Pump Failure Case Histories

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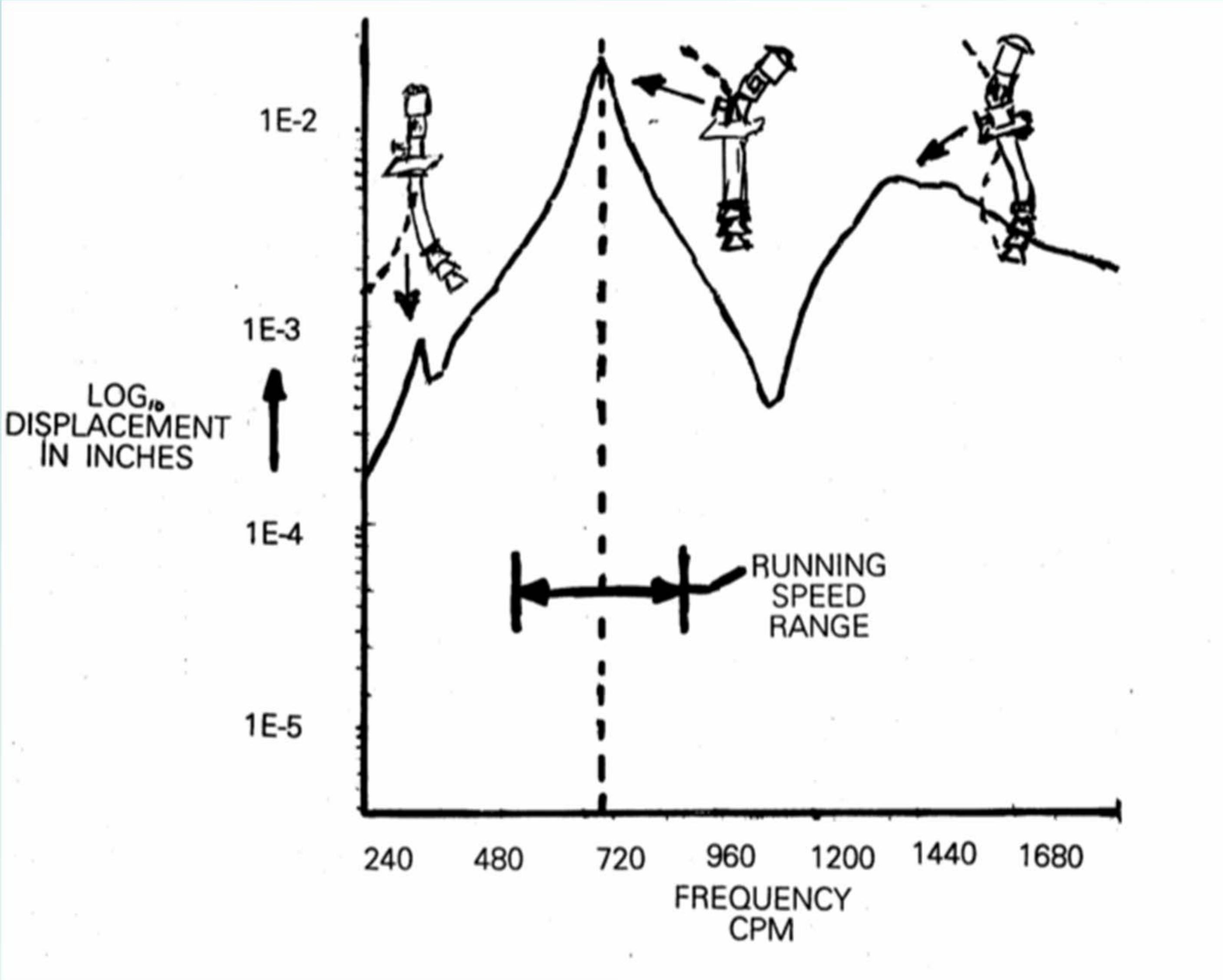
Typical Structural Vibrations of VTP's



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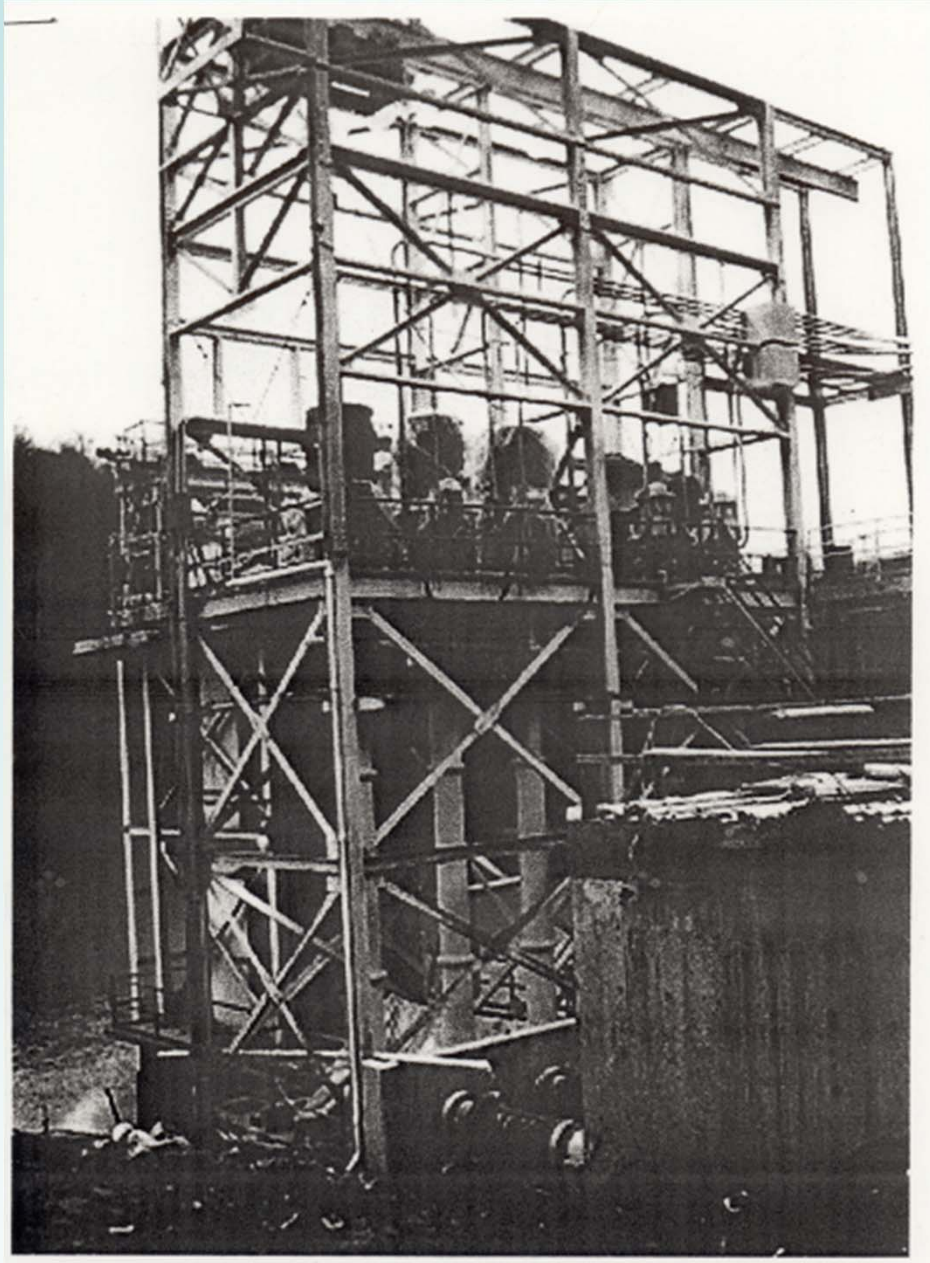
Six Vertical Pumps on Platform, with Column Exposed to Air



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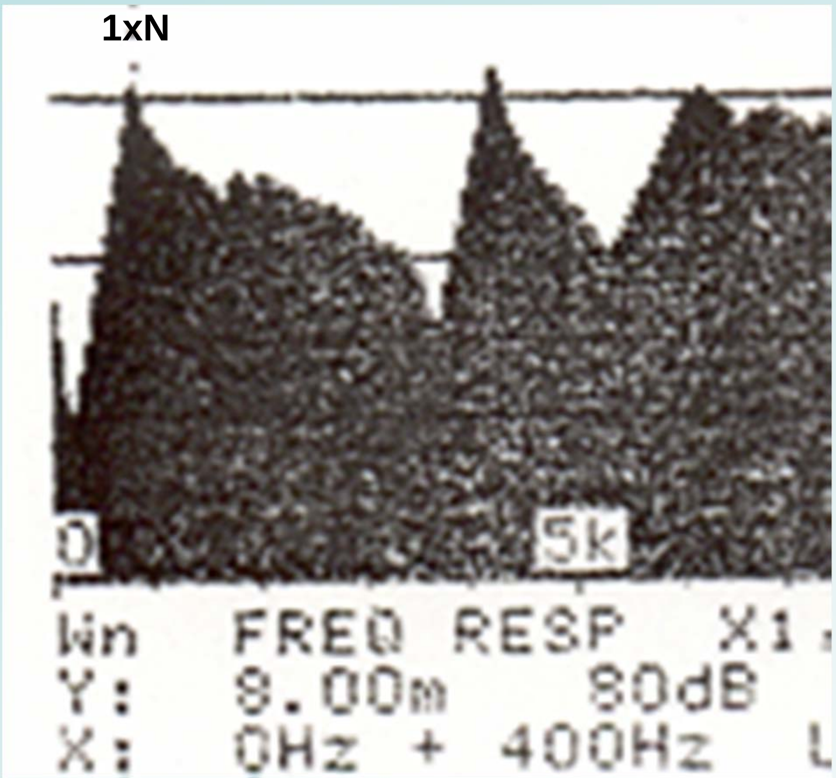
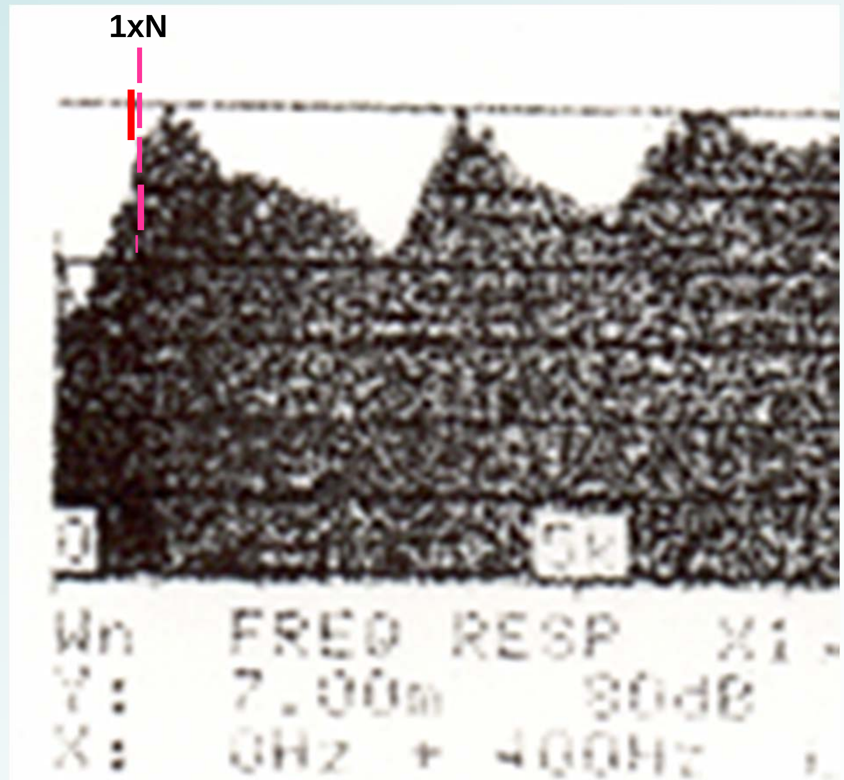
Vertical Pump Column Vibration with & without Water in Column



With Water



Without Water

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Multistage Pump Inboard Bearing Chronic Failure Case History

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Pump with Inboard Bearing Failures

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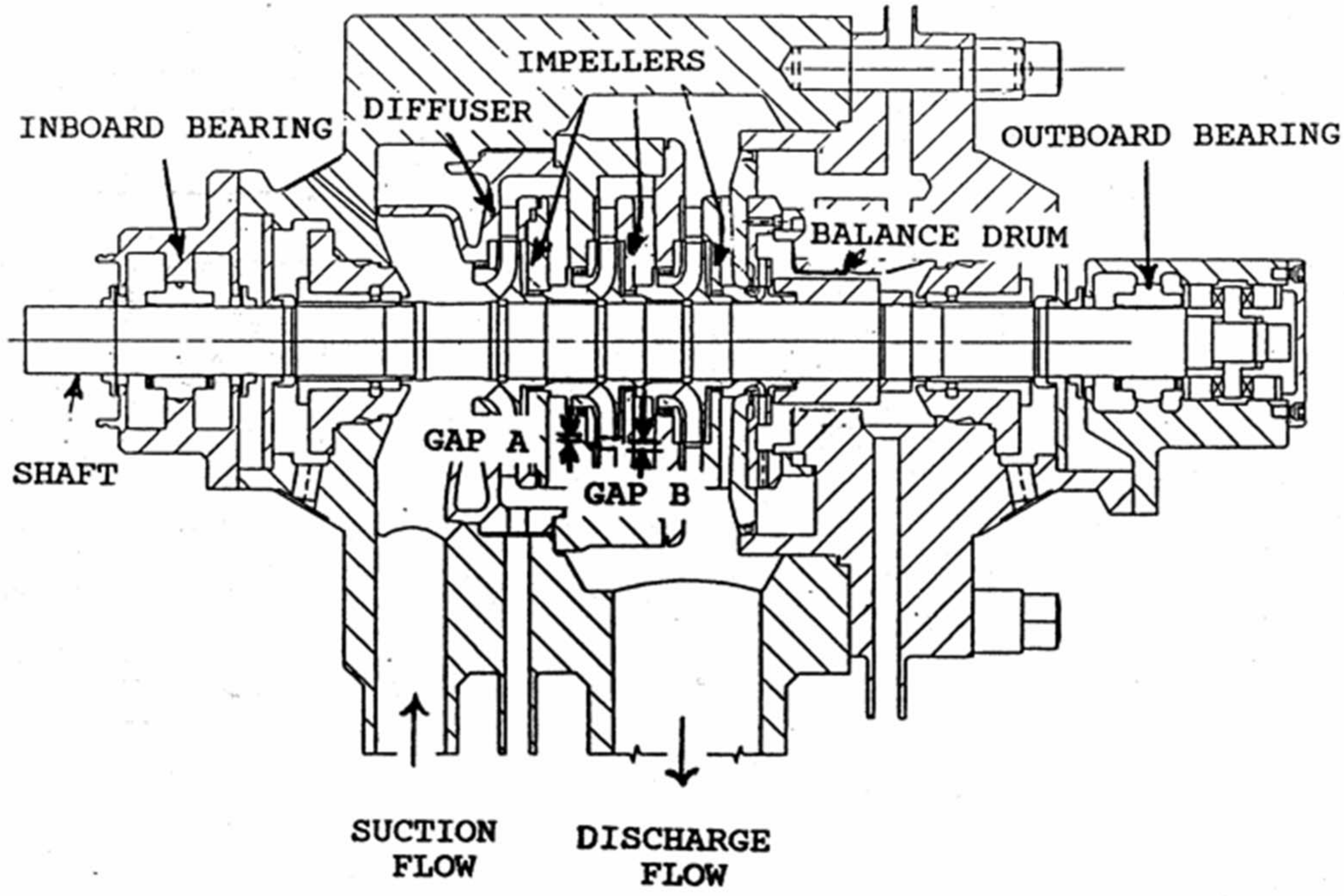

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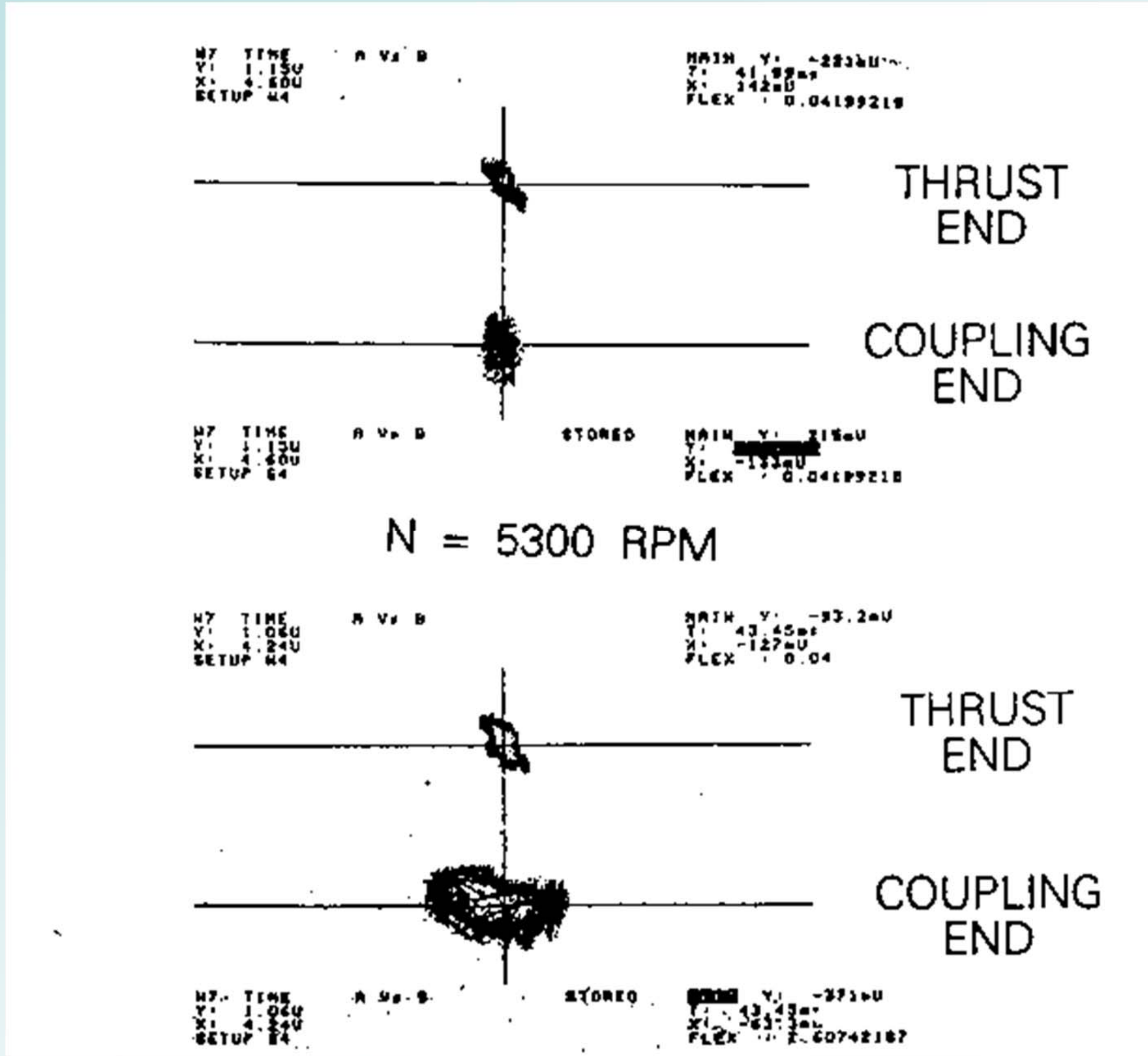
Shaft Orbits of Problem Pump



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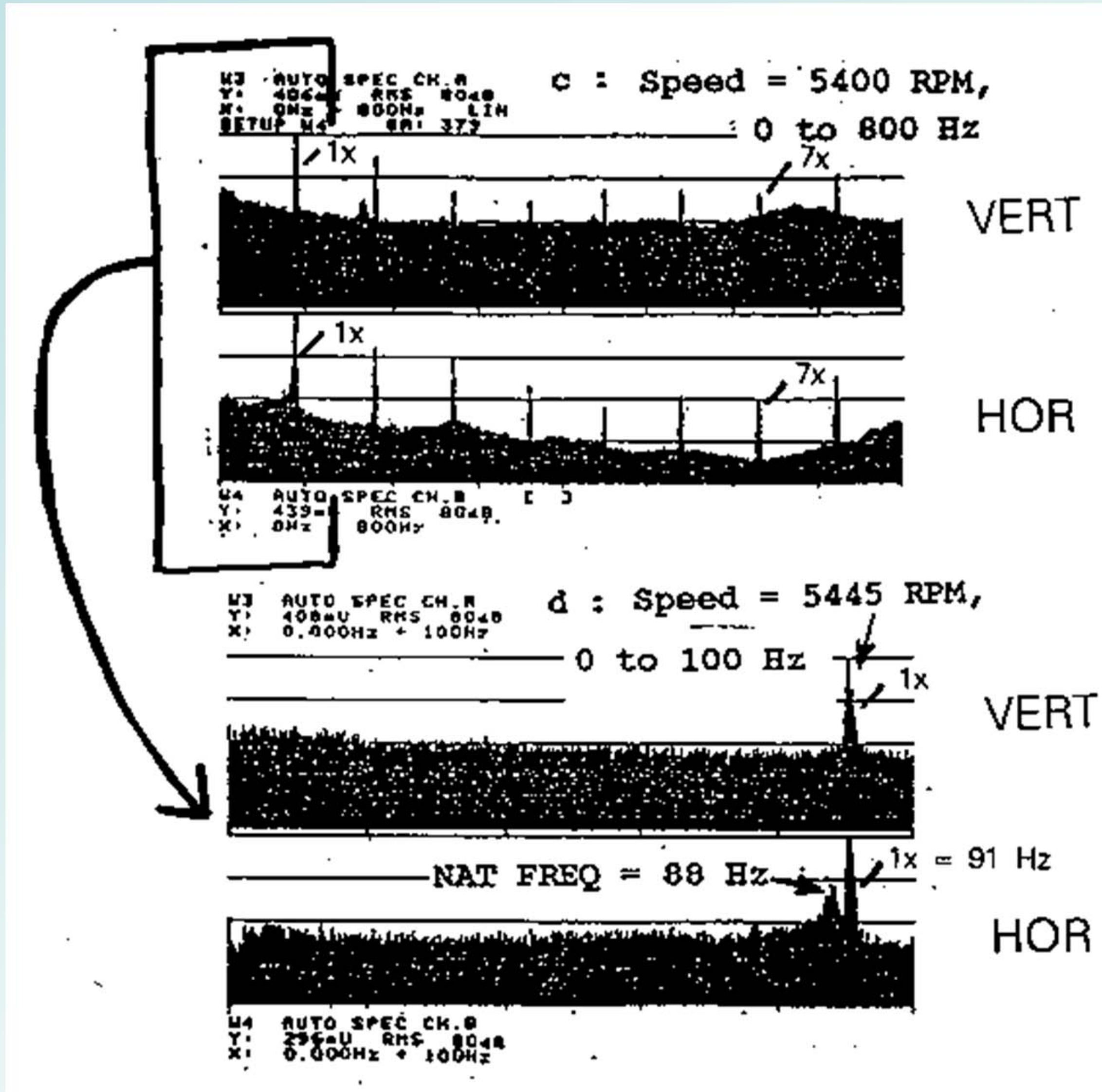
Vibration Spectra for Problem Pump



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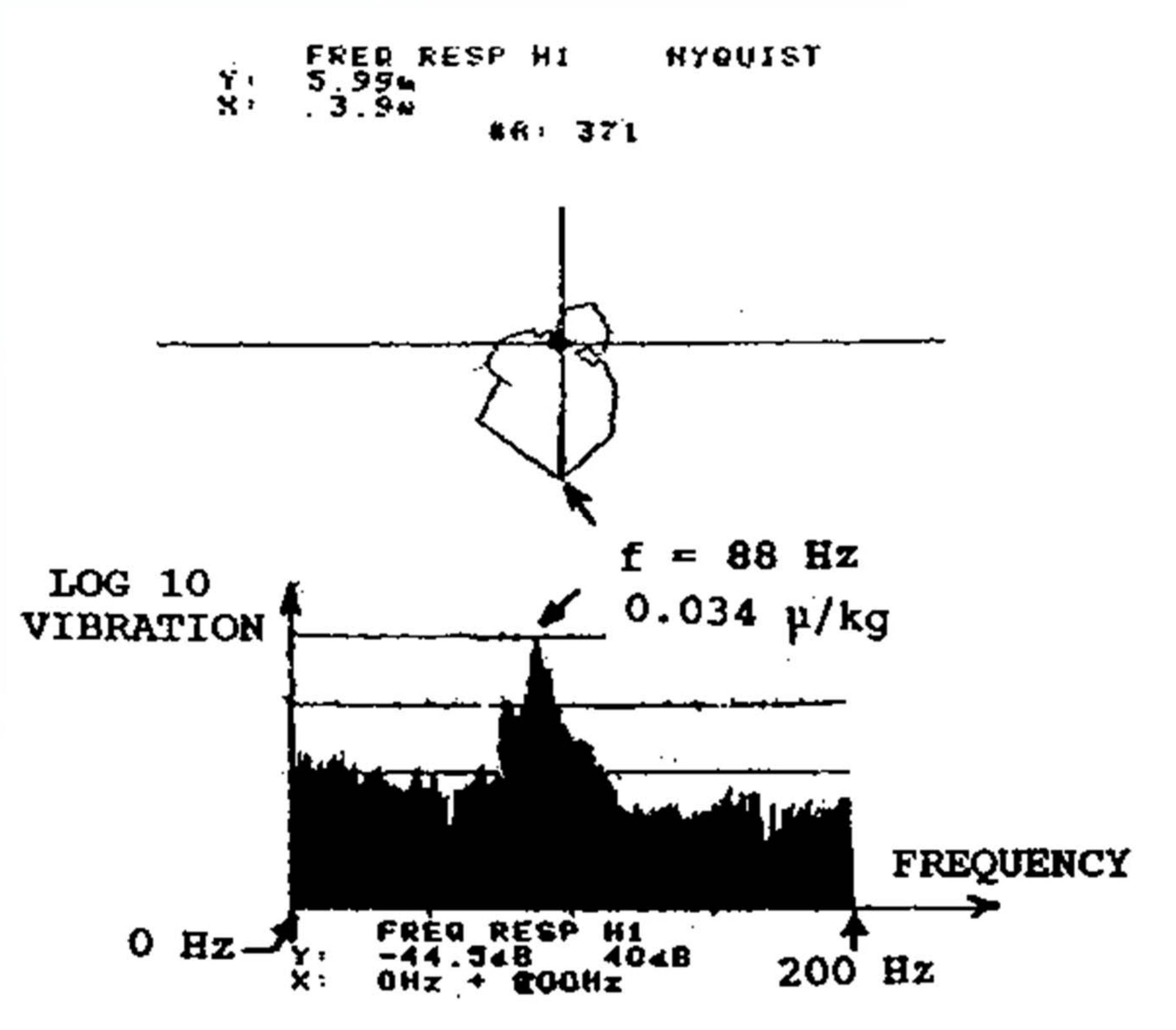
Impact Test Results Showing Shaft Critical Speed at 5280 rpm



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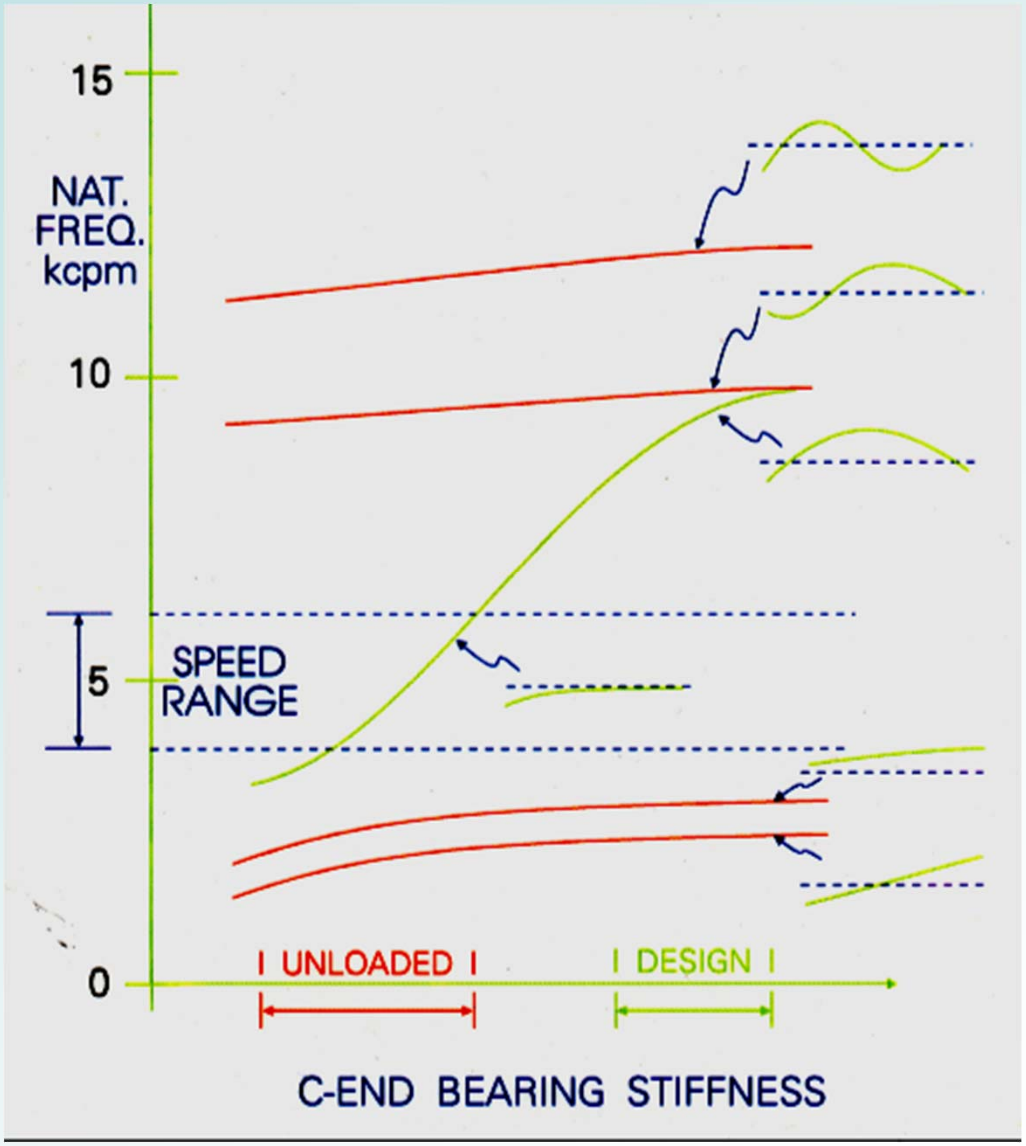
"What If" Analysis for Problem Pump



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Bearing Groove Change from 0.040 in. Deep to 0.010 in. Deep

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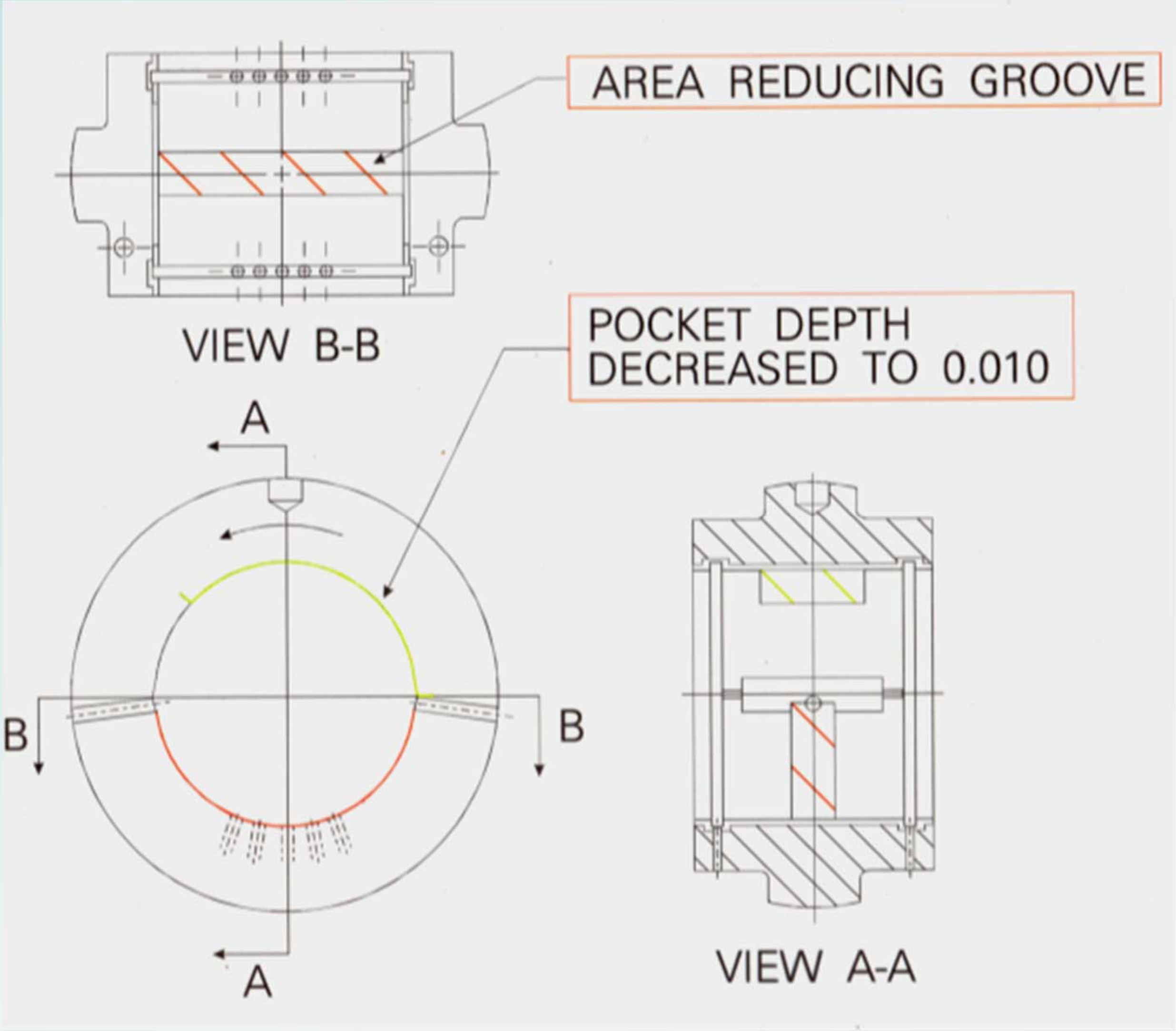
Vibration decreased a factor of ten



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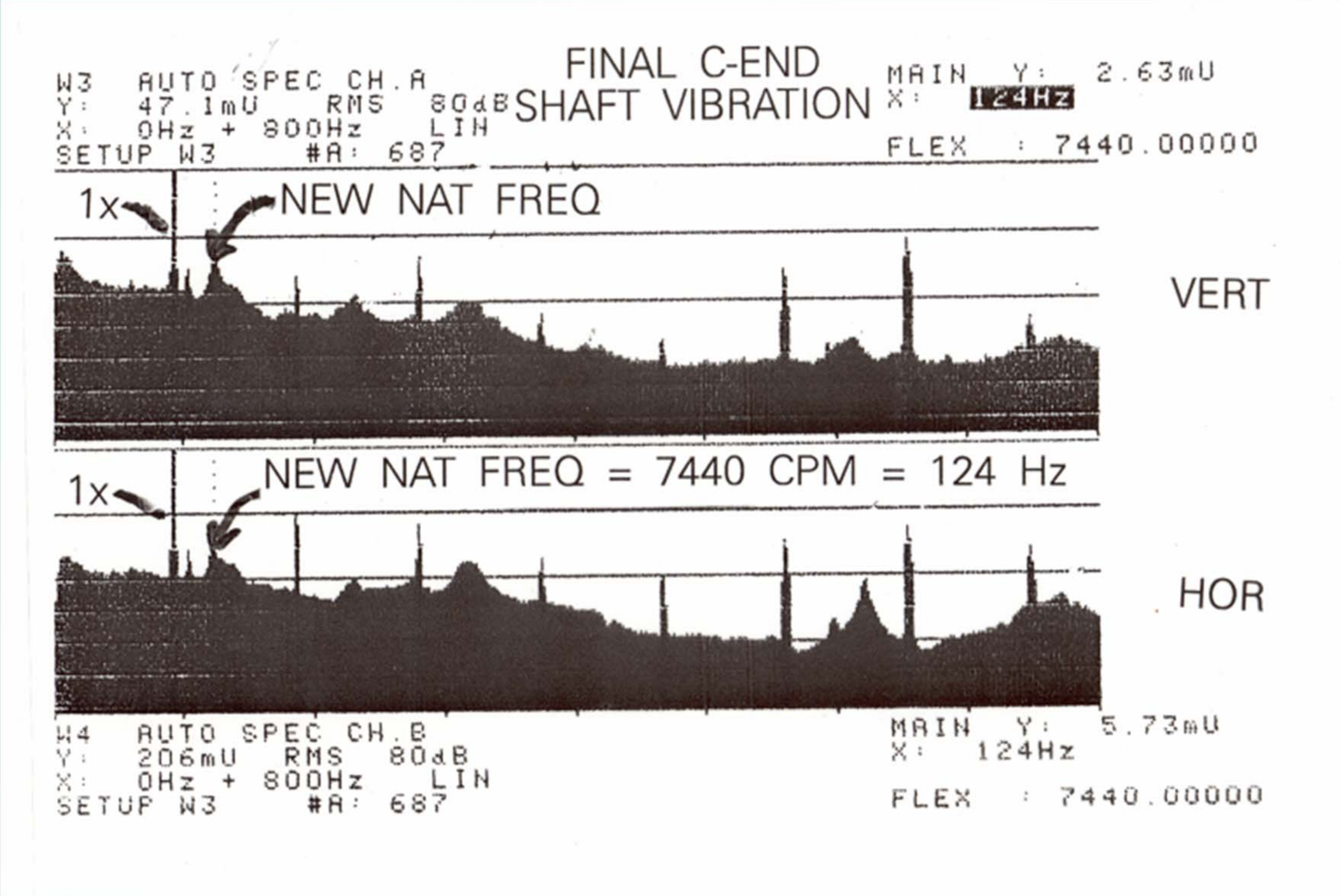
Vibration Spectrum After Bearing Fix



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Use of Dynamic Absorbers in Constant Speed Equipment to Decrease Vibration Motion

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Dynamic Absorber Design- Vertical Pump/Motor

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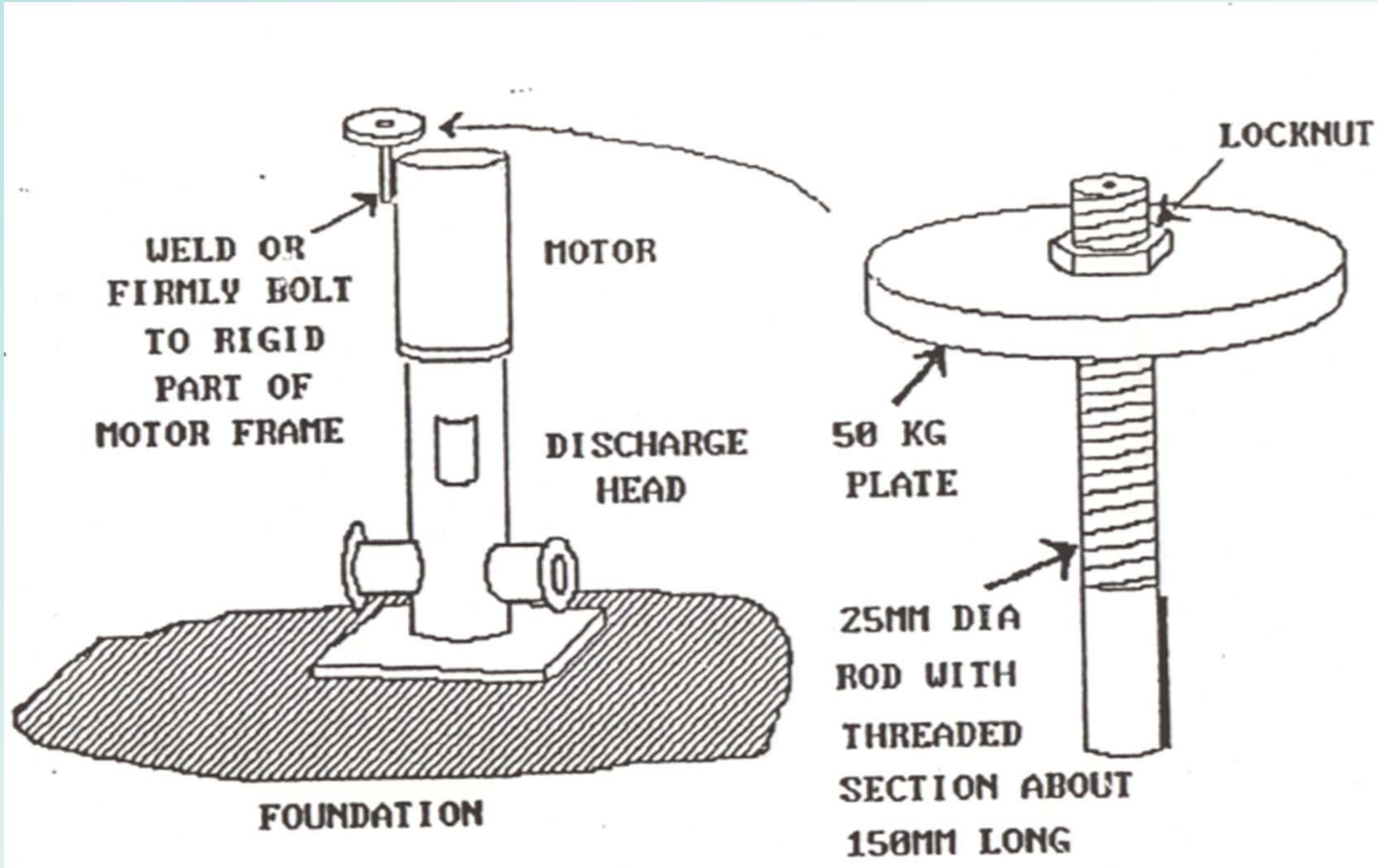

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Effect of Dynamic Absorber on Vibration

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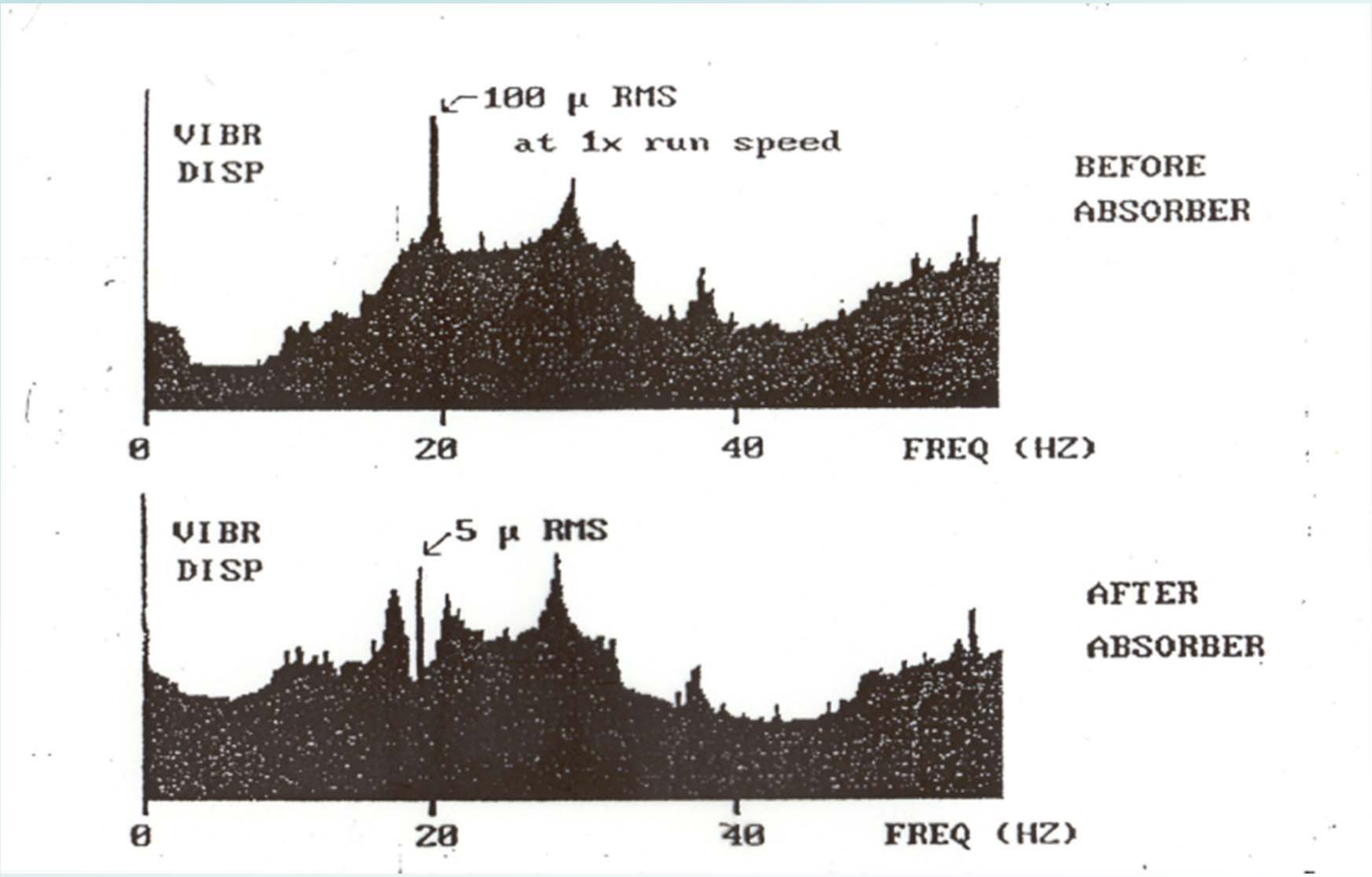

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Gear Box Noise & Wear Problem

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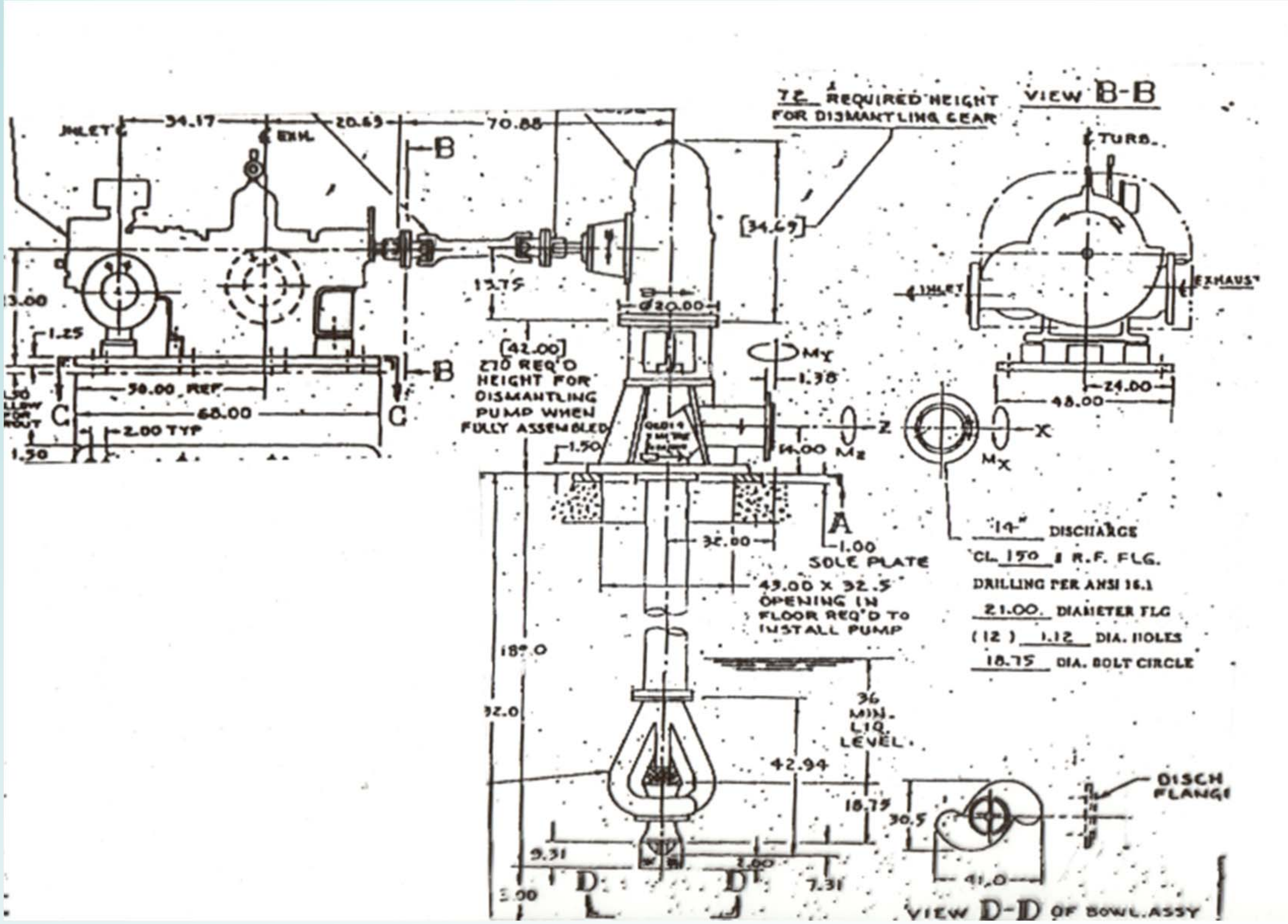

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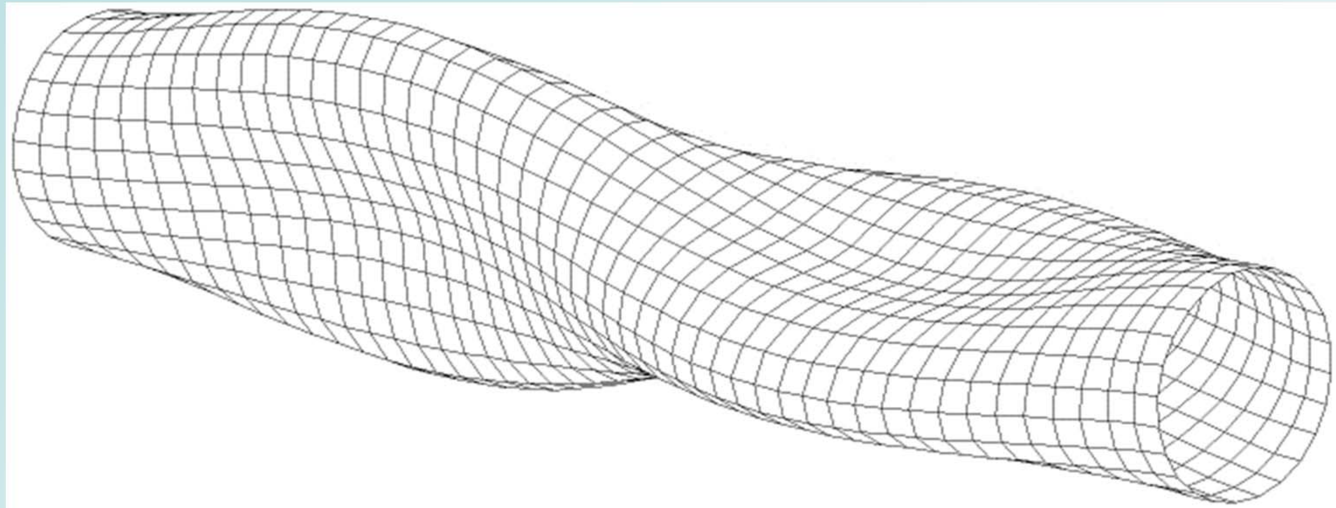

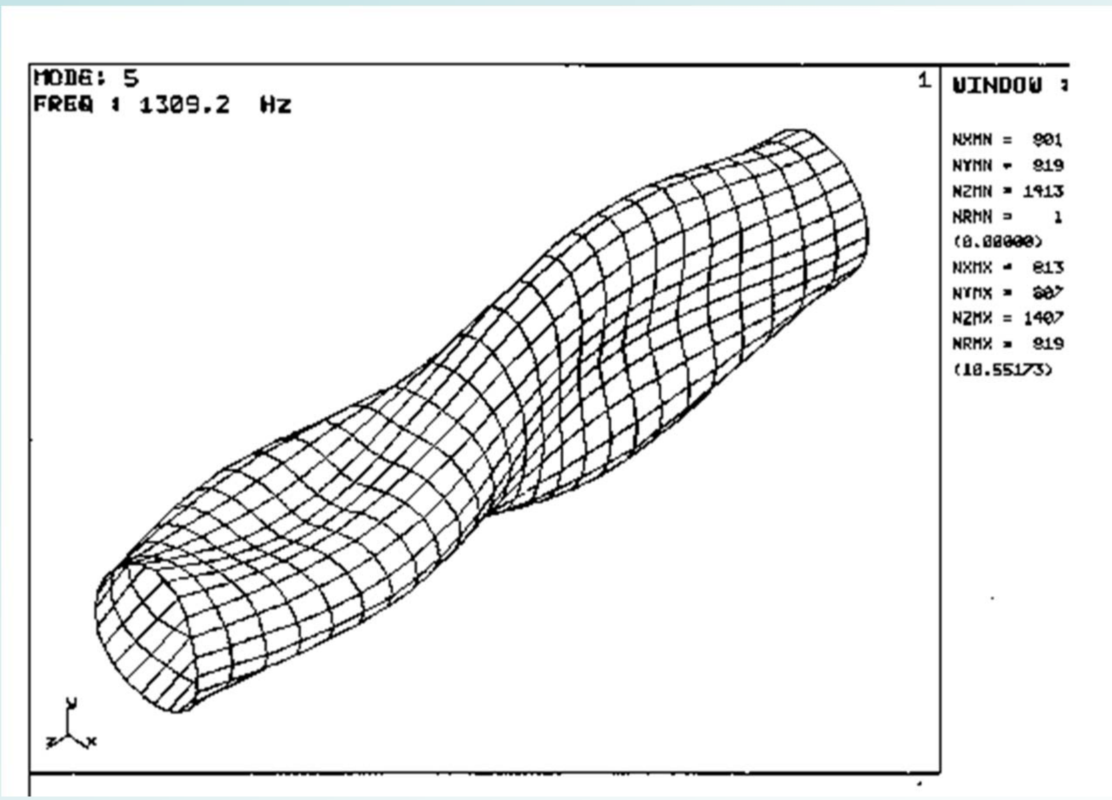

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Conclusions

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

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