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An End-User's Guide to Centrifugal Pump Rotordynamics

Mechanical Solutions, Inc.
Test ▪ Analyze ▪ Solve ▪ Design ▪ Products

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Abstract

This tutorial discusses concepts and methods involved in performing and evaluating centrifugal pump rotordynamic analysis. The presentation includes Lomakin Effect, Gyroscopic Effect, Cross-Coupling, Rotordynamic Stability, Critical Speeds and their Mode Shapes, Forced Response, common Excitation Forces (both hydraulic and mechanical), and typical plant rotordynamic problems and solutions. Case Histories are included to provide examples of successful use of rotordynamic analysis.

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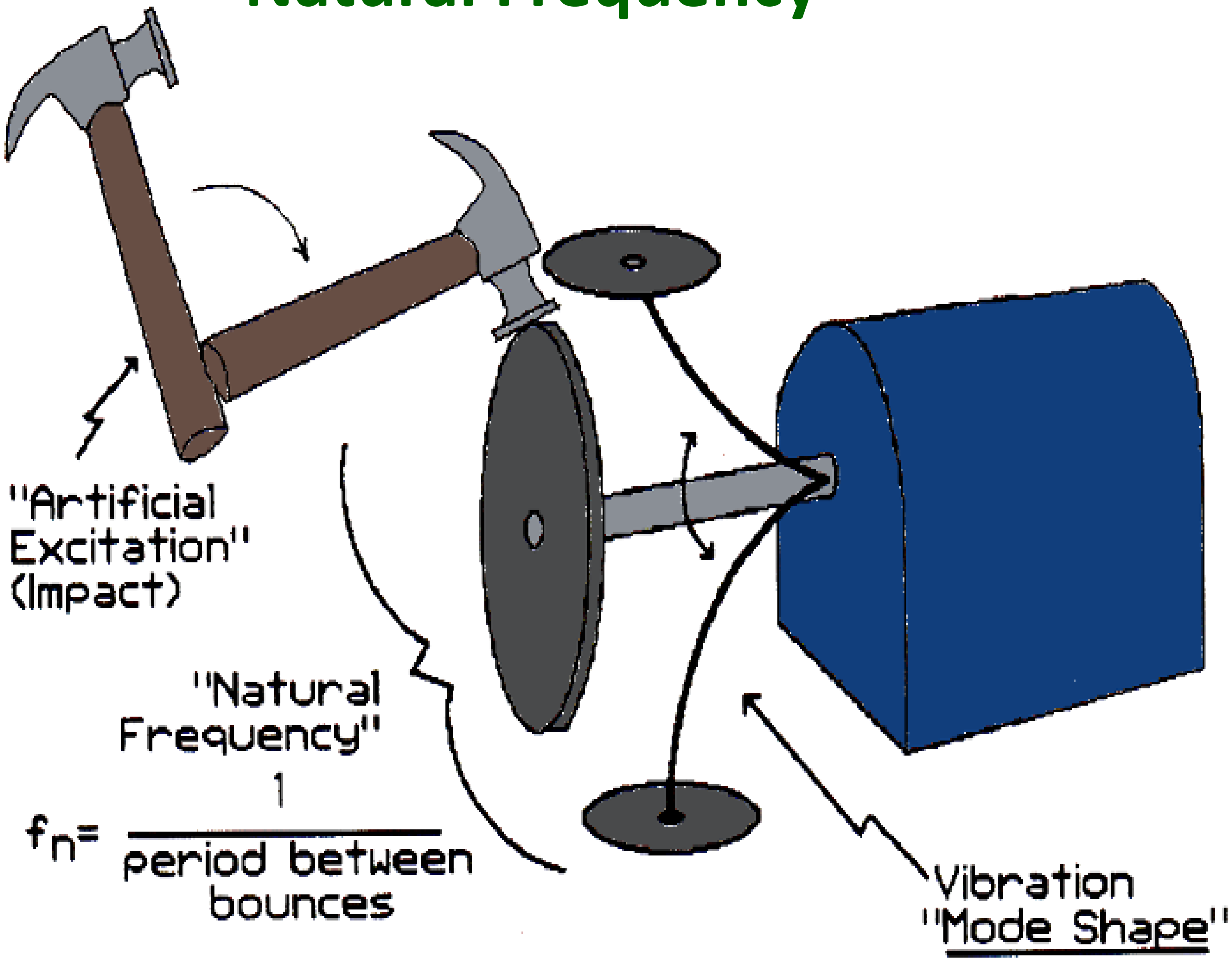
Rotor Dynamics

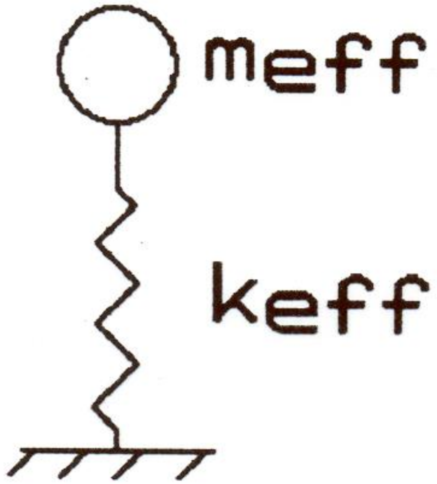
- Critical Speeds & Mode Shapes
- Forced Response
- Stability
- Specifications
  (API-610 12th & 684, HI 9.6.4, HI 9.6.8, ISO 10816-7)

Natural Frequency



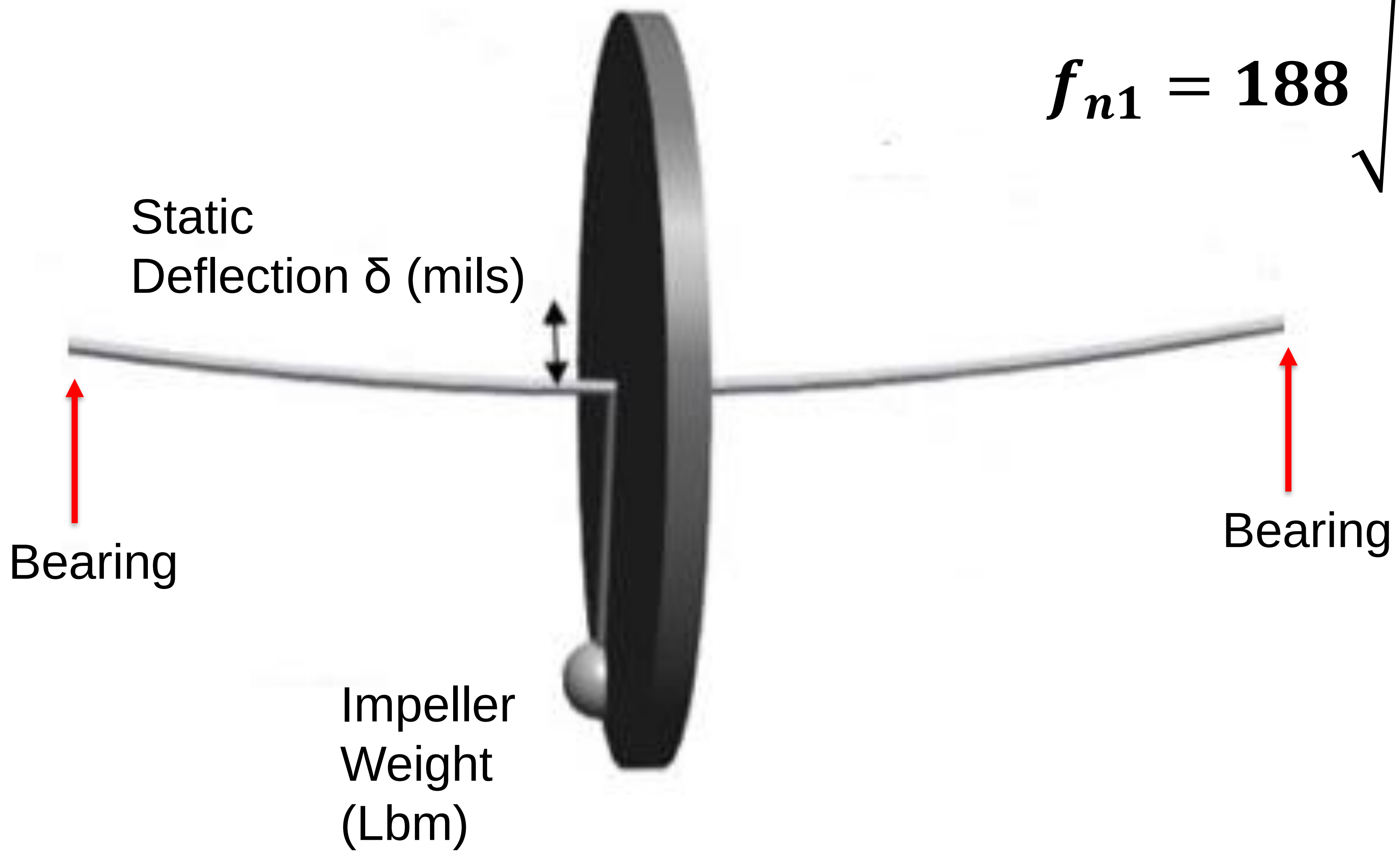
Approximating Natural Frequency

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

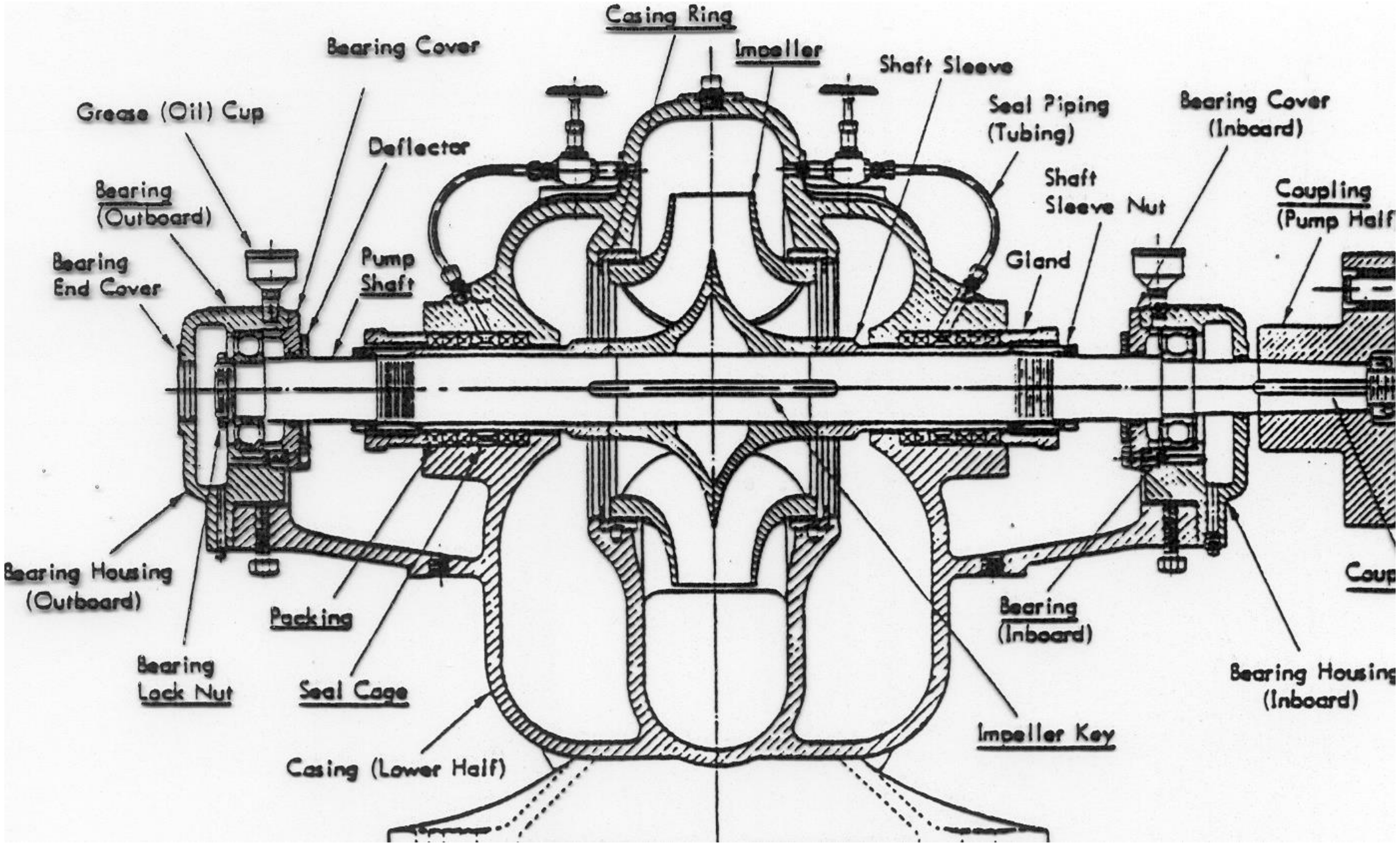


Approximate Natural Frequency Calculation

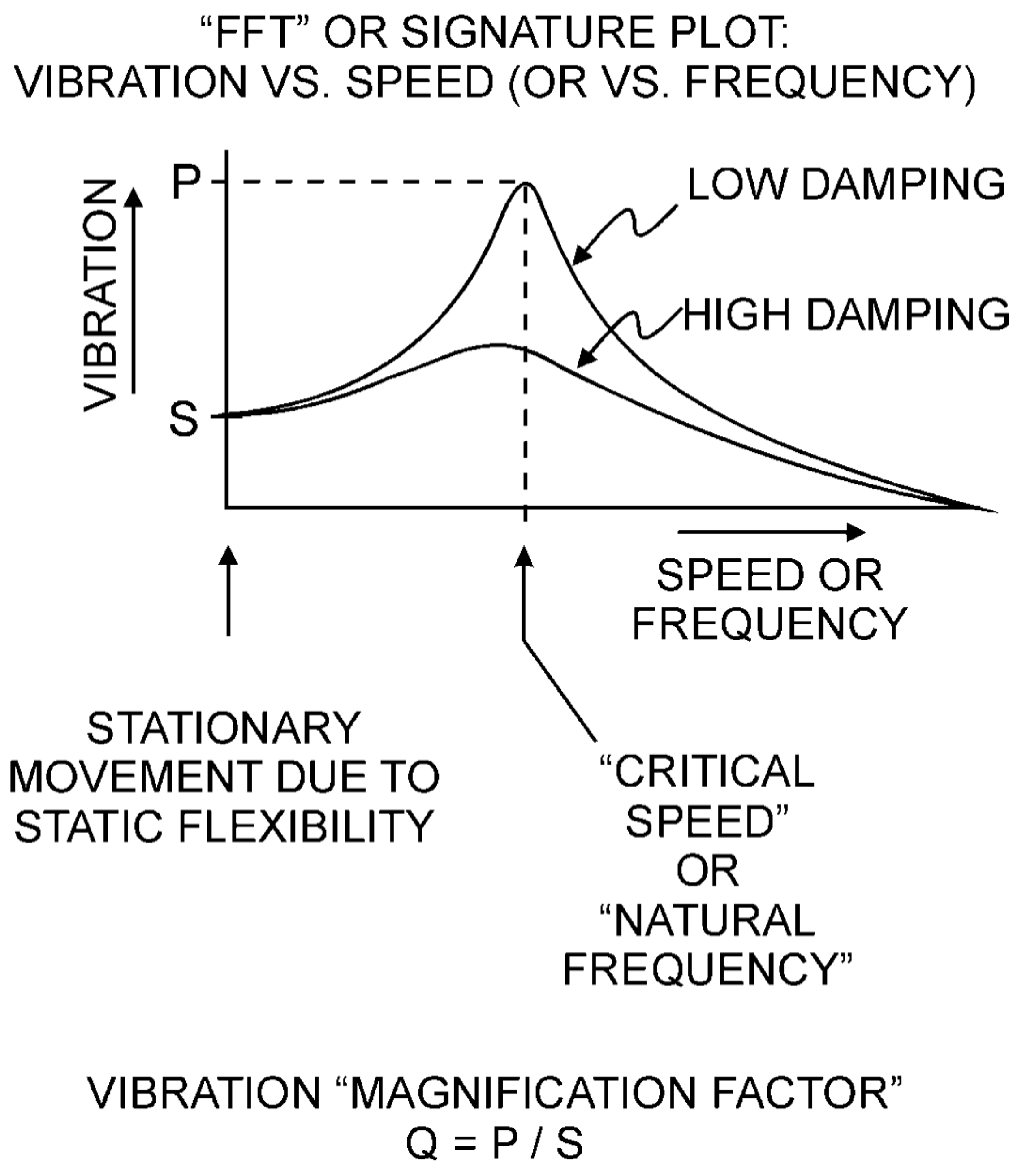
$$f_{n1} = 188\sqrt{\frac{1}{\delta_{static}}}\,(cpm)$$



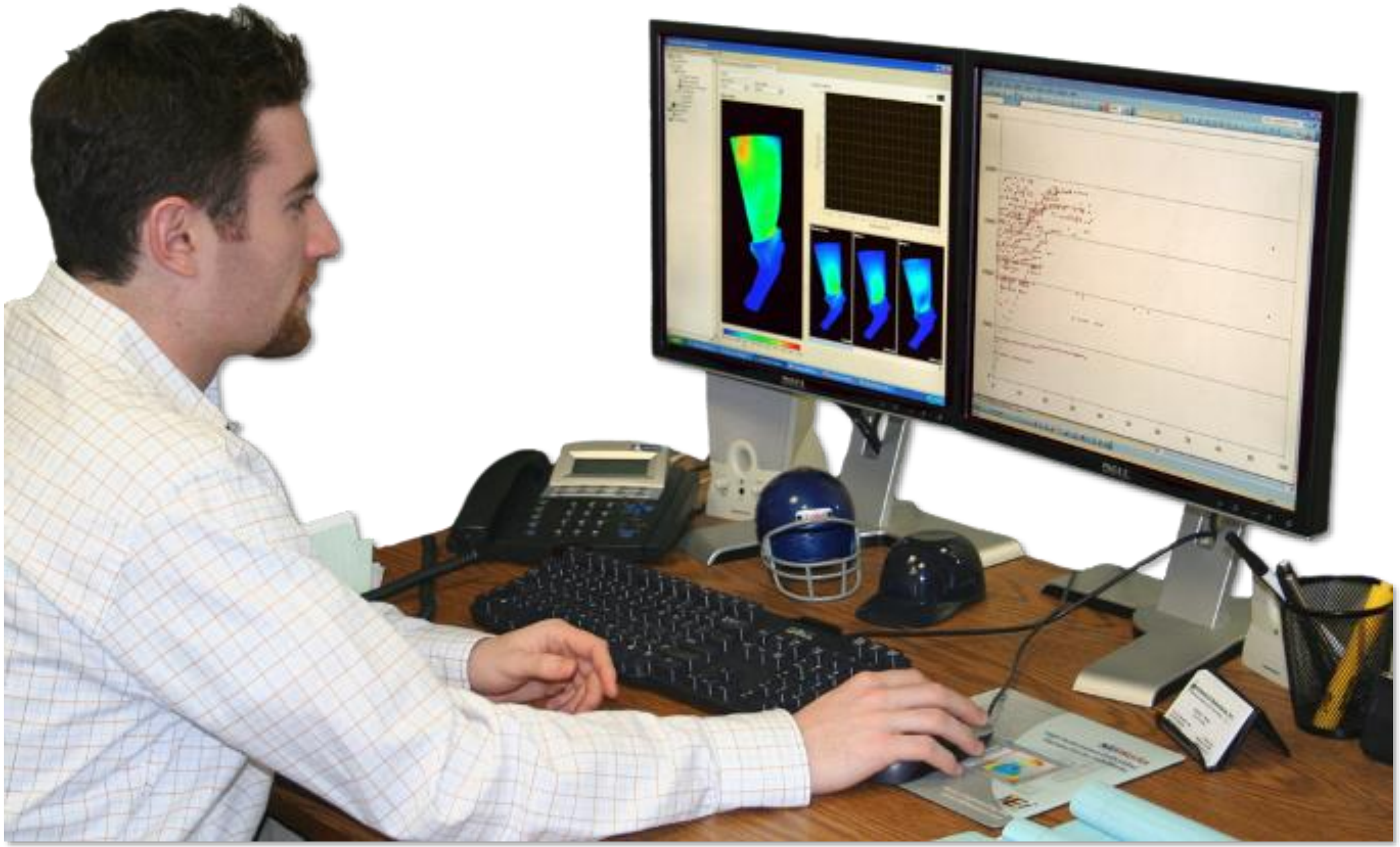
Example: Axially Split Case Double Suction Pump



Natural Frequency Resonance



Rotor Dynamics Is Best Evaluated by Computer



Types of Rotordynamic Computer Programs

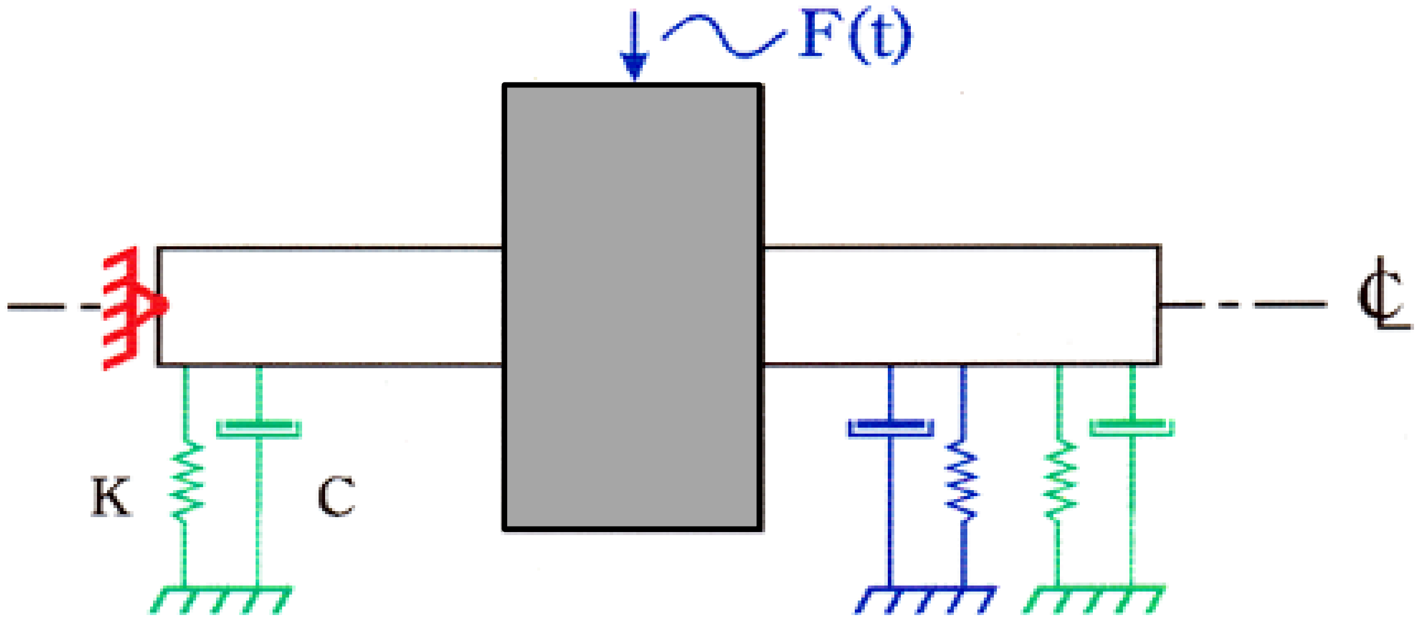
Transfer matrix (type of “Holzer Method”)

- Limited in its “stations” or “nodes”
- Very fast computation time
- Older “Legacy codes” use this method

Finite element analysis

- Can include essentially unlimited nodes
- Can easily couple in the structure effects
- Can use general purpose codes (e.g. ANSYS)

Rotor Idealization

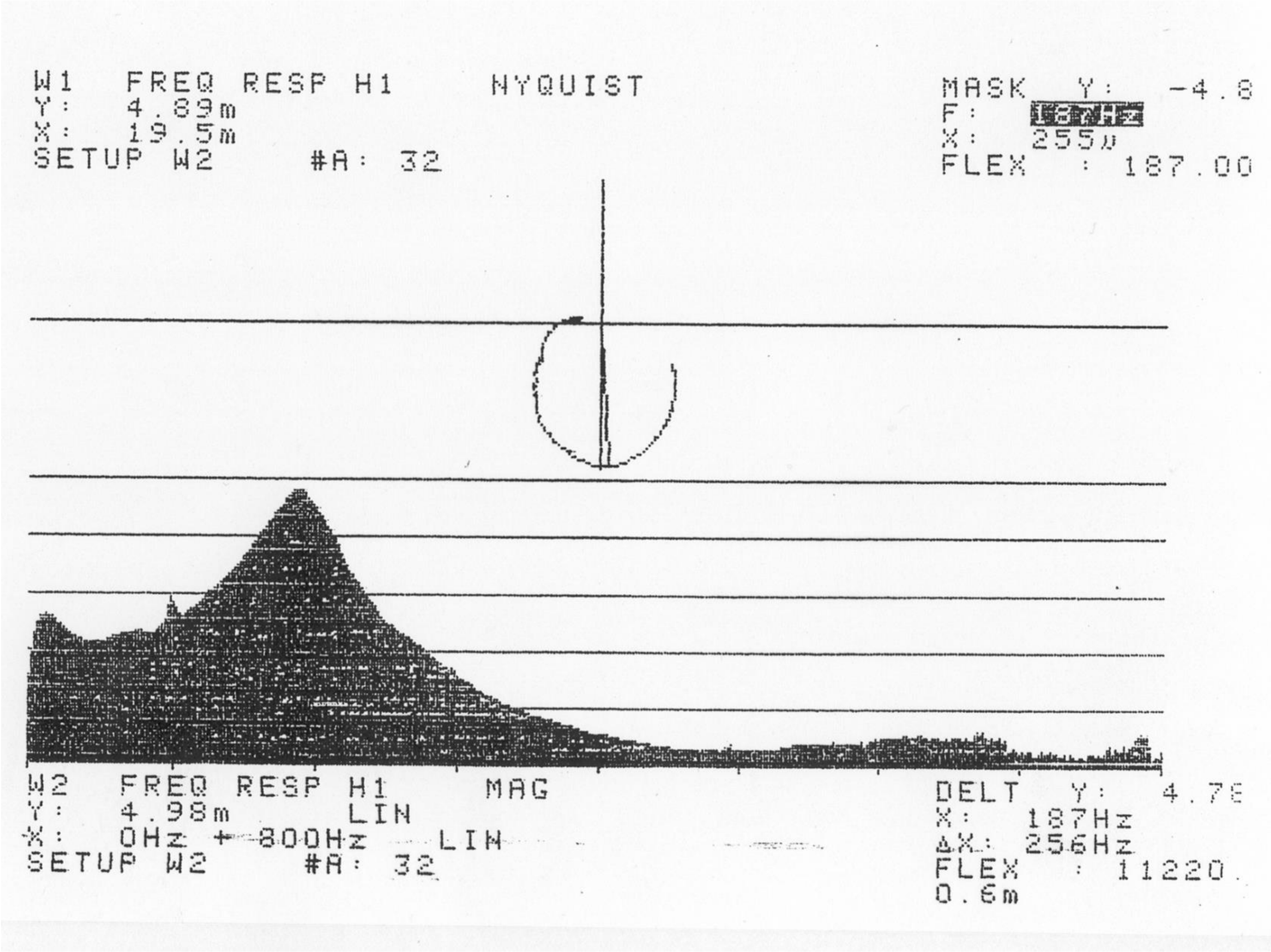


$$\{F\} = [M]\{\ddot{\delta}\} + [C]\{\dot{\delta}\} + [K]\{\delta\}$$

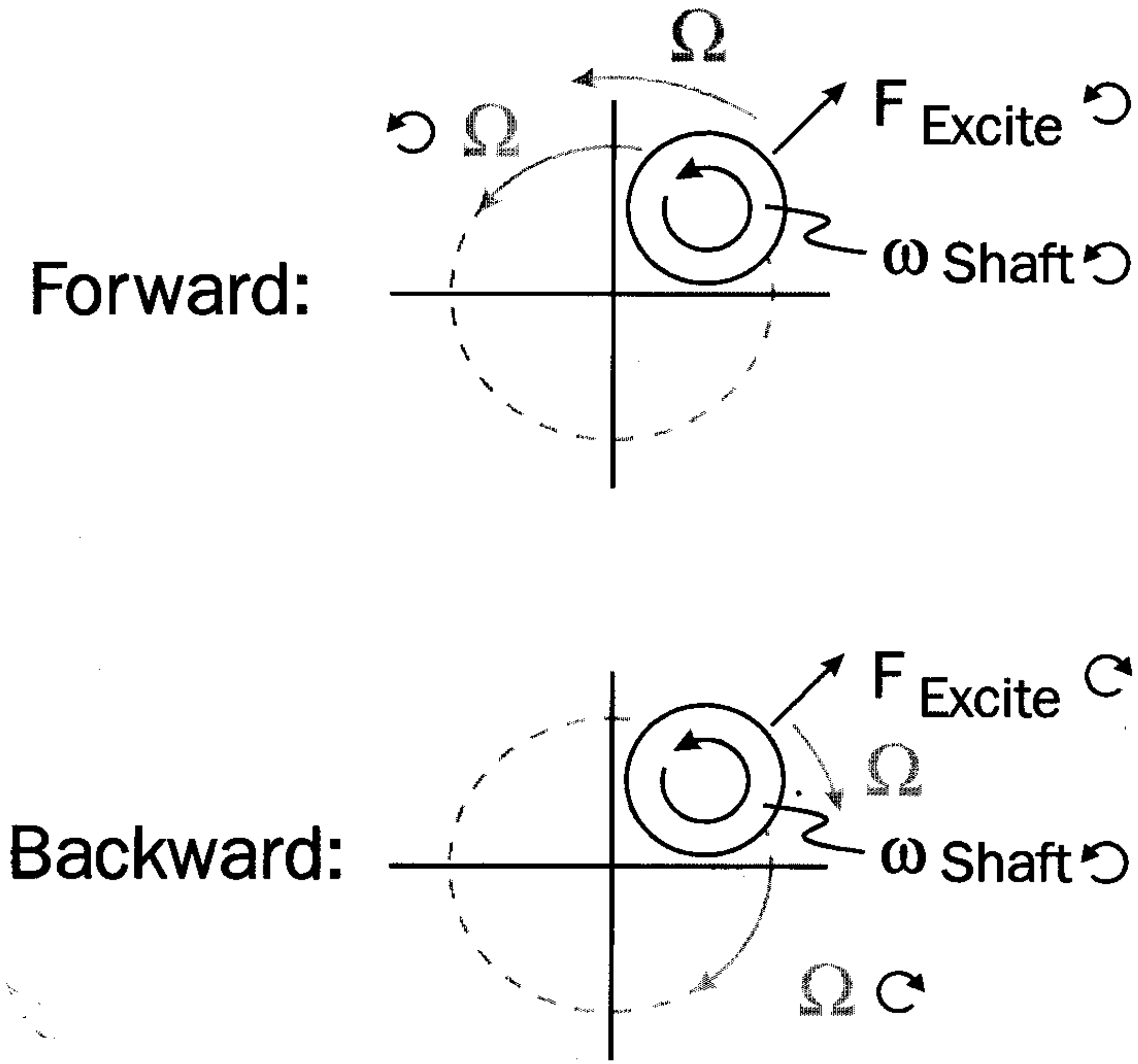
where $\delta = \begin{Bmatrix} x_1 \\ y_1 \\ x_2 \\ y_2 \\ \cdot \\ \cdot \\ \cdot \end{Bmatrix}$, $K_{station} = \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix}$

NOTE: F, C, AND K ARE A FUNTION OF: ω, ε, μ, $M_{leakage}$, SWIRL, TEMPERATURE, AND ANGULAR POSITION.

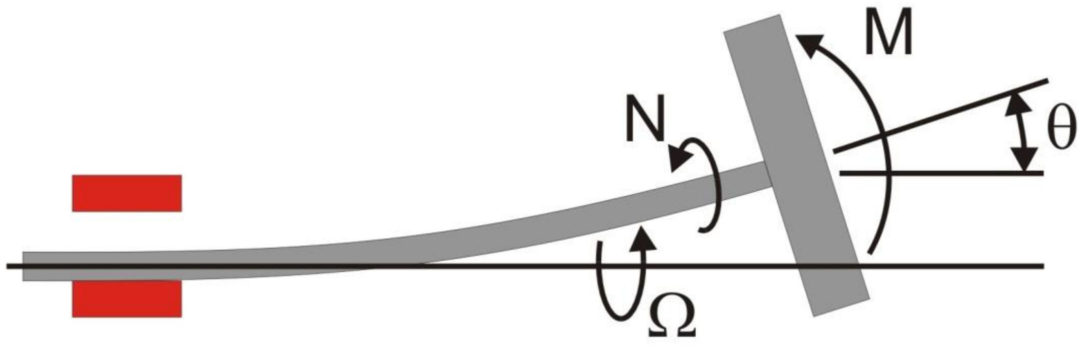
First Critical Speed Operating Bump Test
Double Suction SS Pump

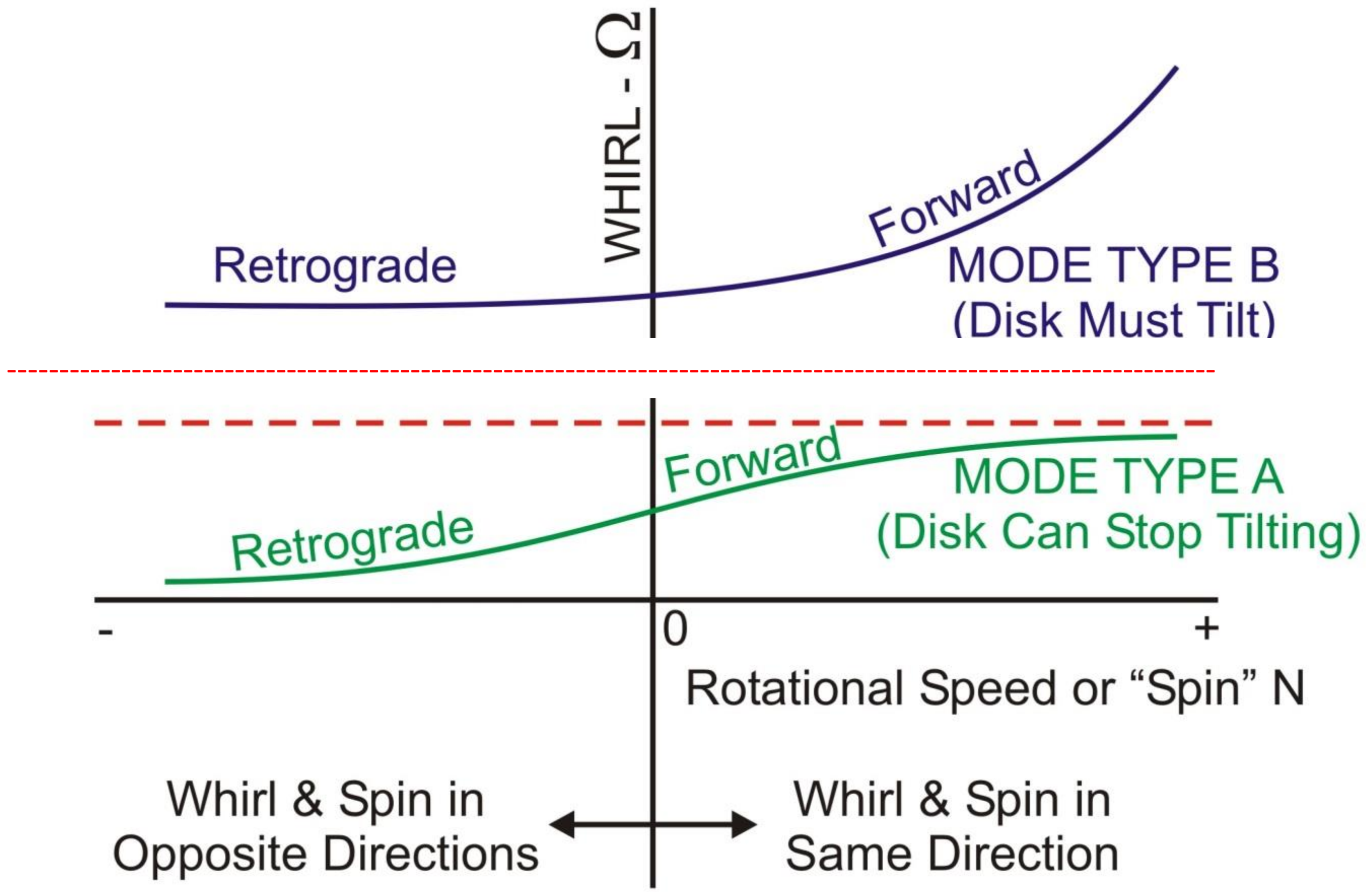


Backward vs. Forward Precession

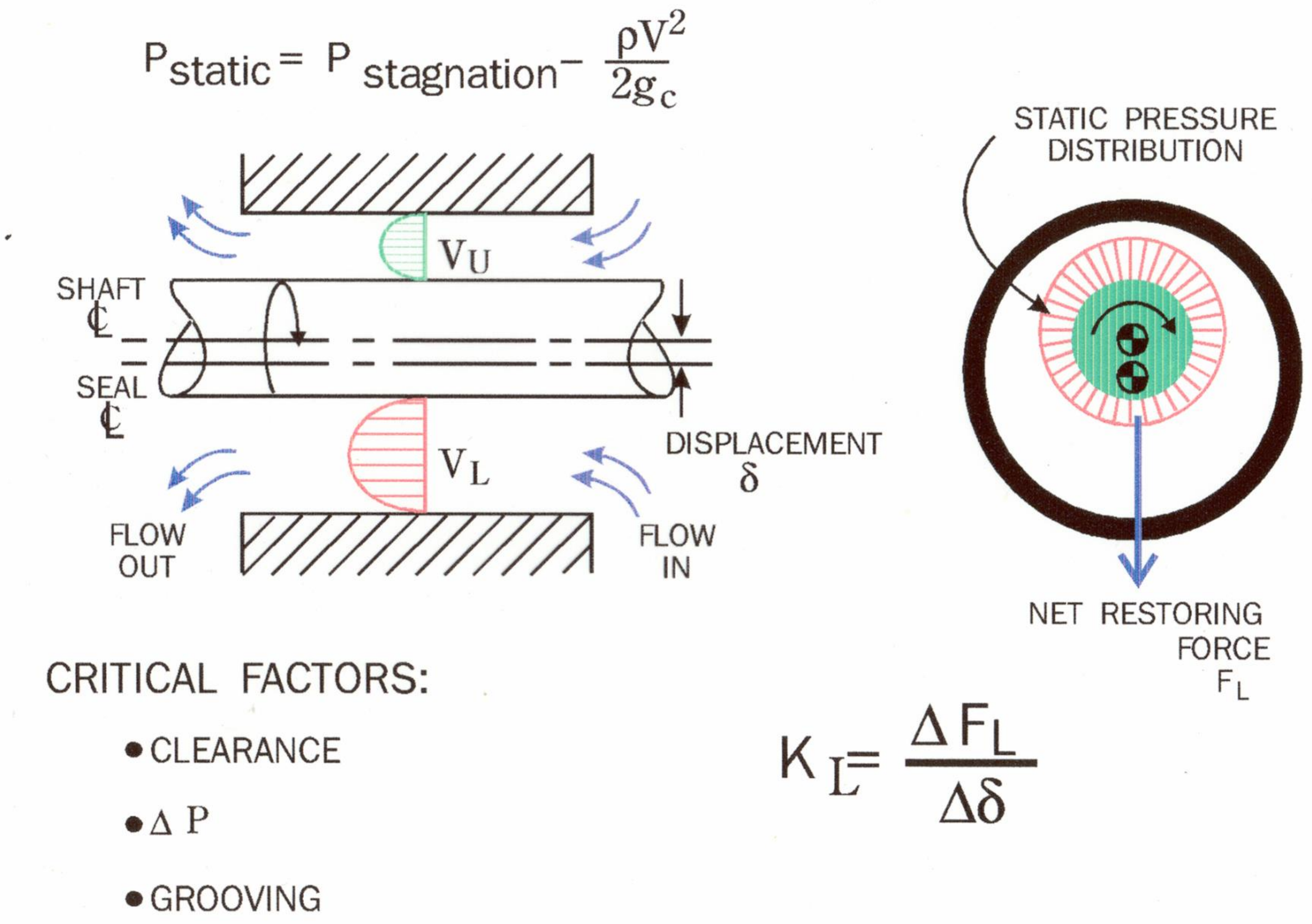


Gyroscopic Dynamics

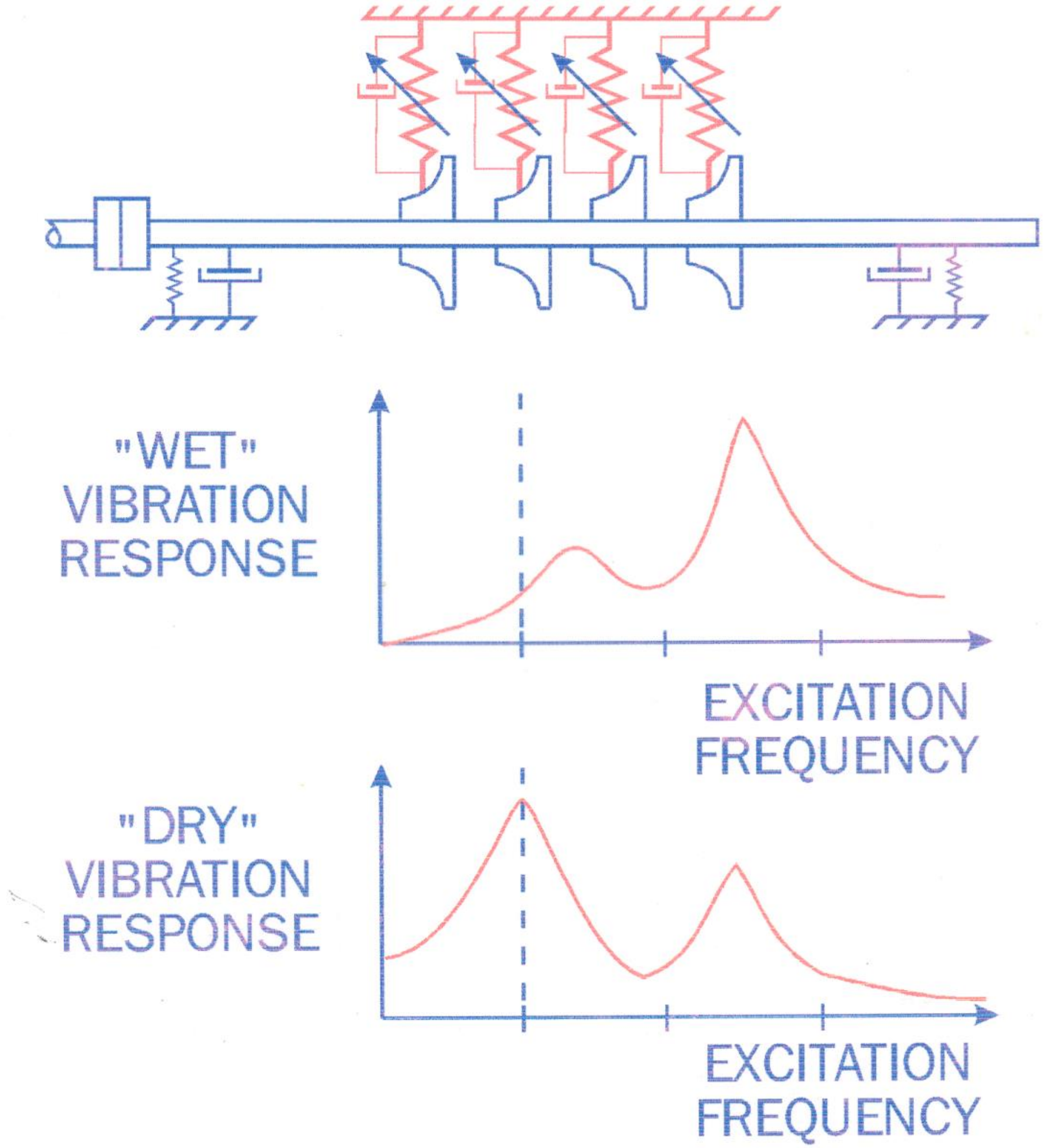




Lomakin Effect in Centrifugal Pumps

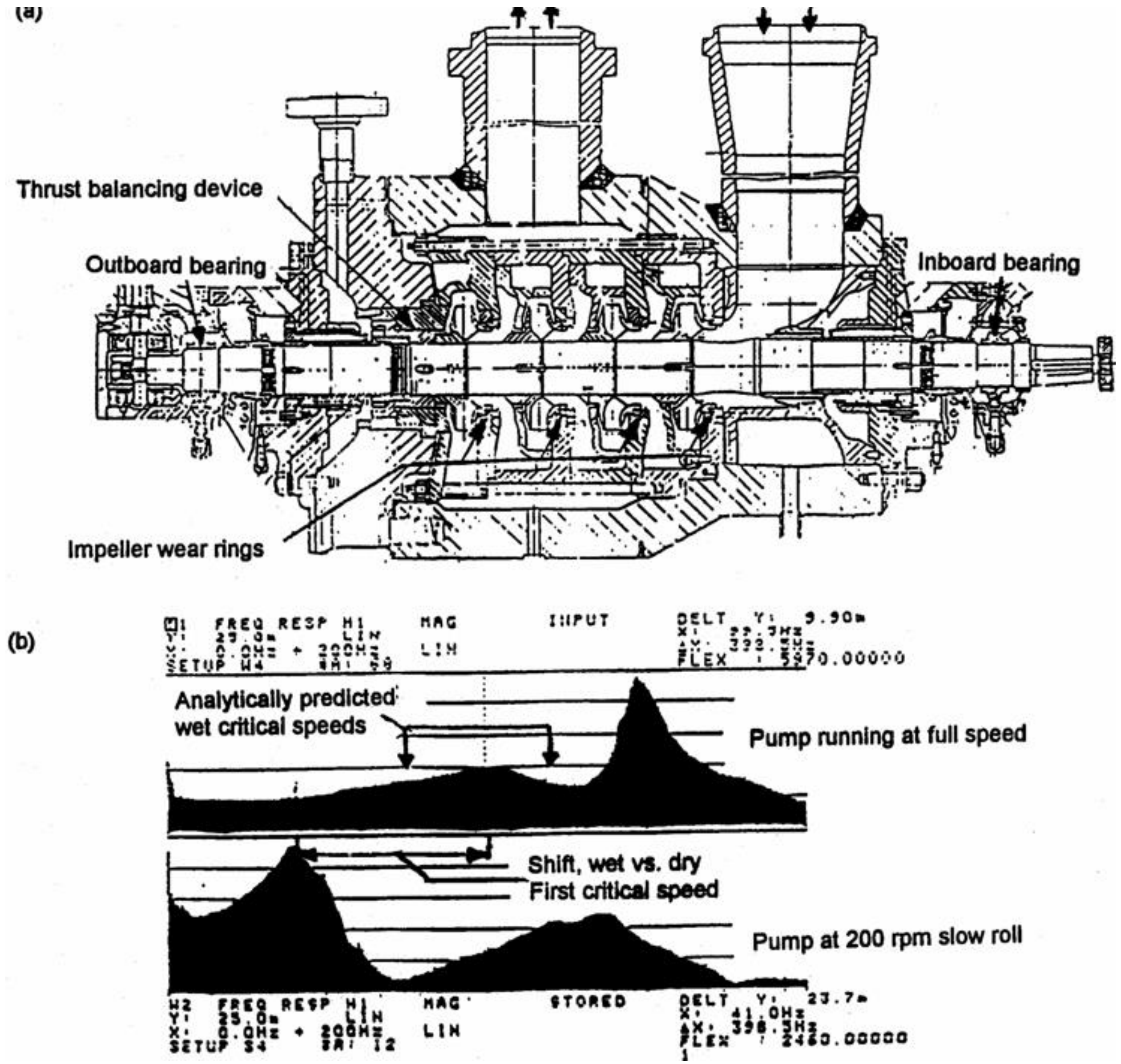


Concept of the "Wet Critical Speed"



Lomakin Result:

Critical Speeds Shift Up



Approximate Calculation of Lomakin Stiffness

R= Radius
L= Length
C= Radial clearance
ΔP=Pressure drop
λ= fric factor

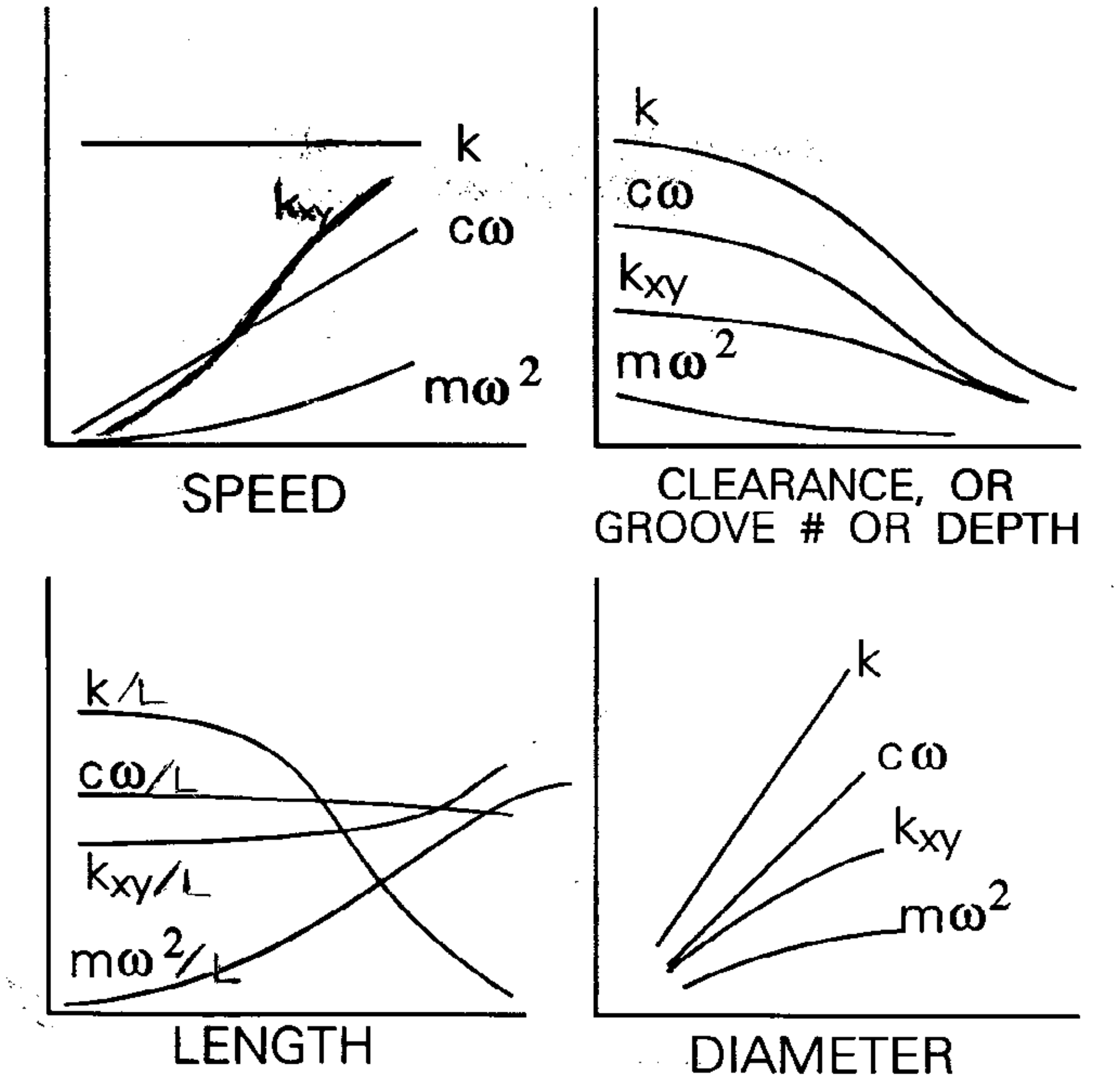
$$k_{xx} \cong \frac{RL\,\Delta P}{c} K_{xx}$$

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

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

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

Lomakin Effect: Dependence on Various Parameters



ALSO: WATCH TURBULENT -> LAMINAR TRANSITION

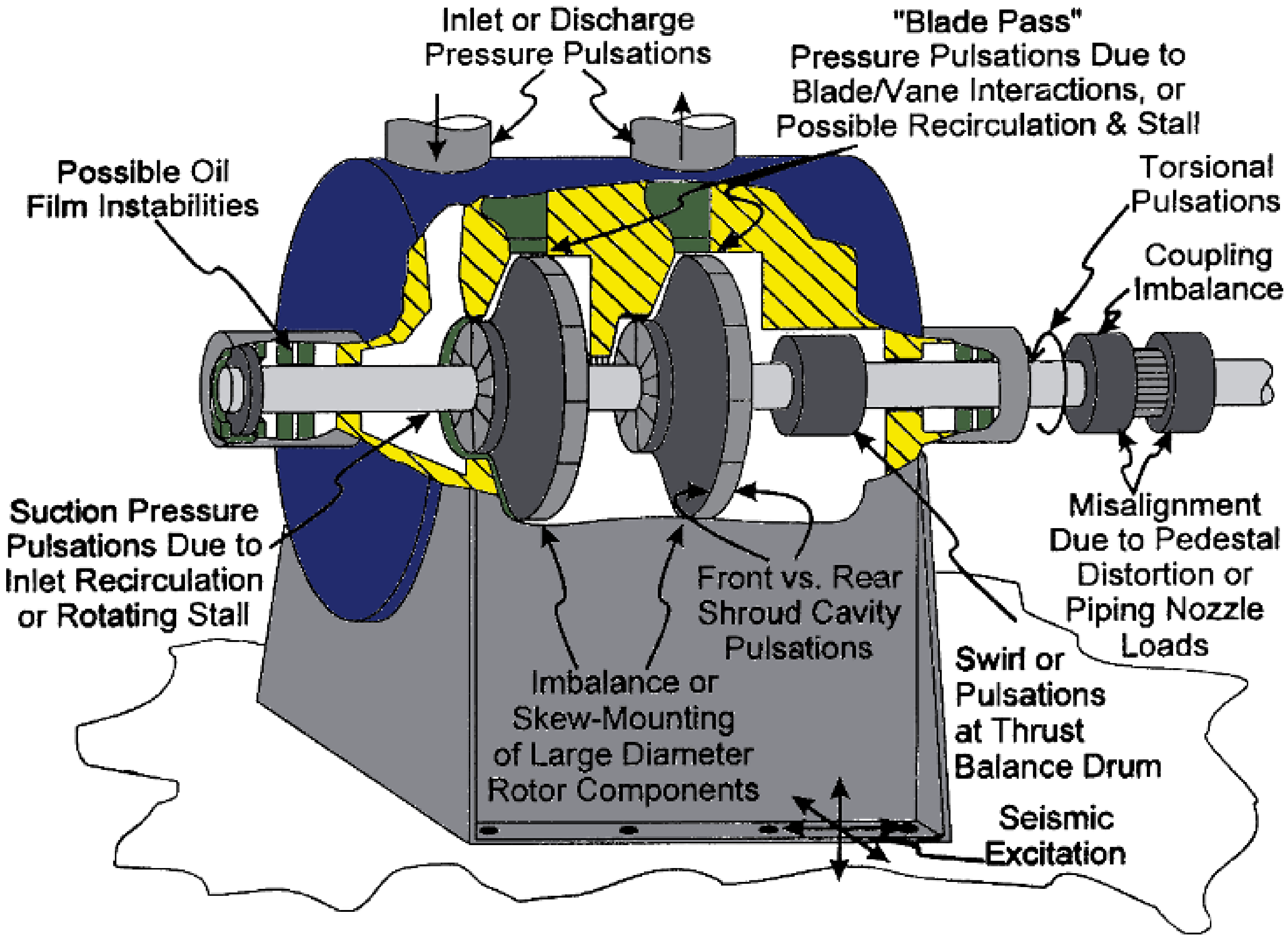
Some Key Issues in Lomakin Effect Strength:

- Grooving
- Surface Roughness
- Inlet Conditions (Swirl, Corners, Deposits, Cavitation)
- Available Total Pressure at Inlet
- Alignment, Eccentricity
- Frequency Content, Orbit Shape
- Wear or Erosion

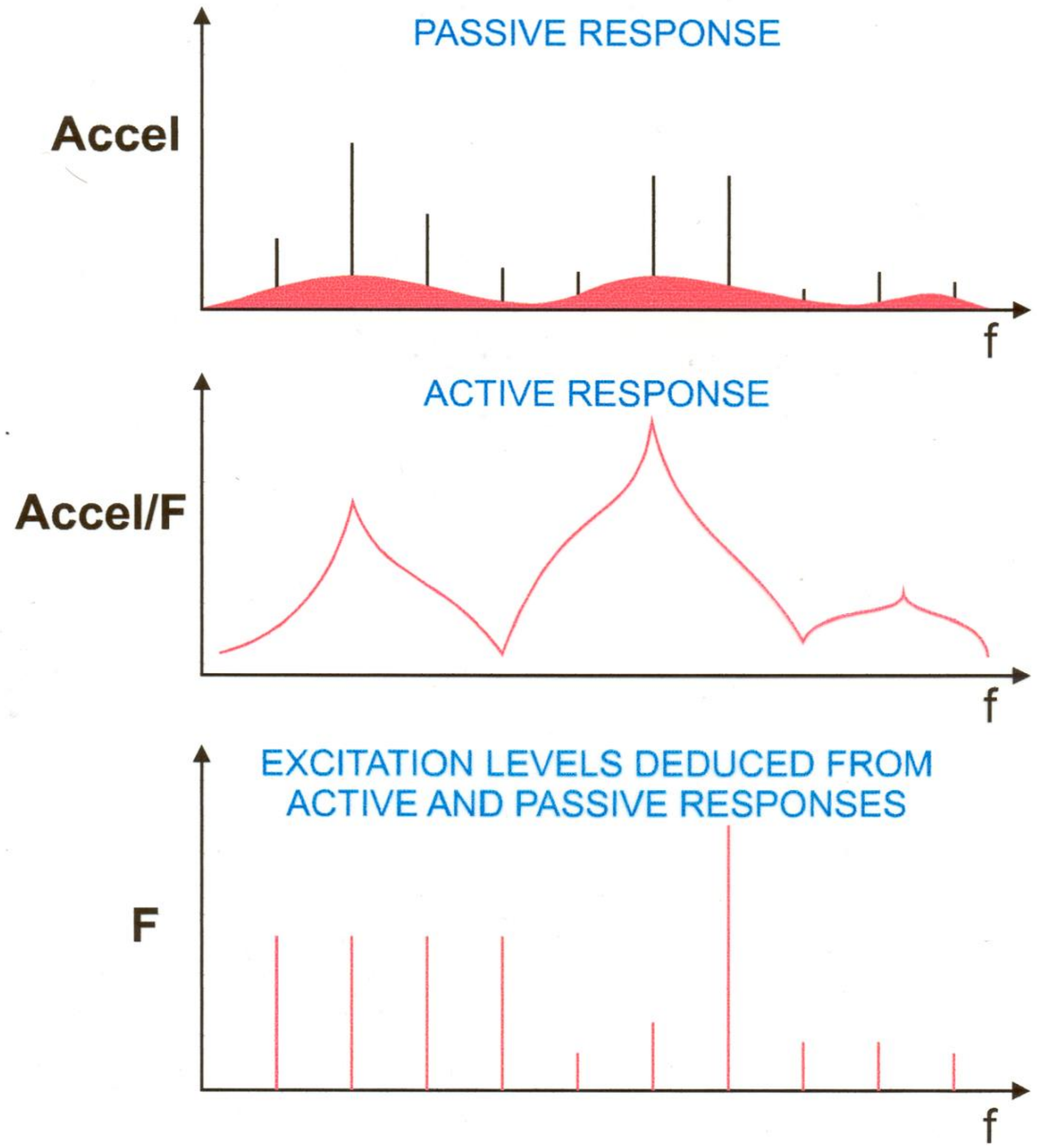
Liquid “Added Mass” Effect

- Fluid inside the impeller flow passages
- “Swept volume” of impeller exterior shape times the density of water (in *addition* to impeller mass itself)
- Added mass in close clearance annular seals (“Stokes Effect”)
- Torsional added mass is usually small, hard to predict, may be neglected

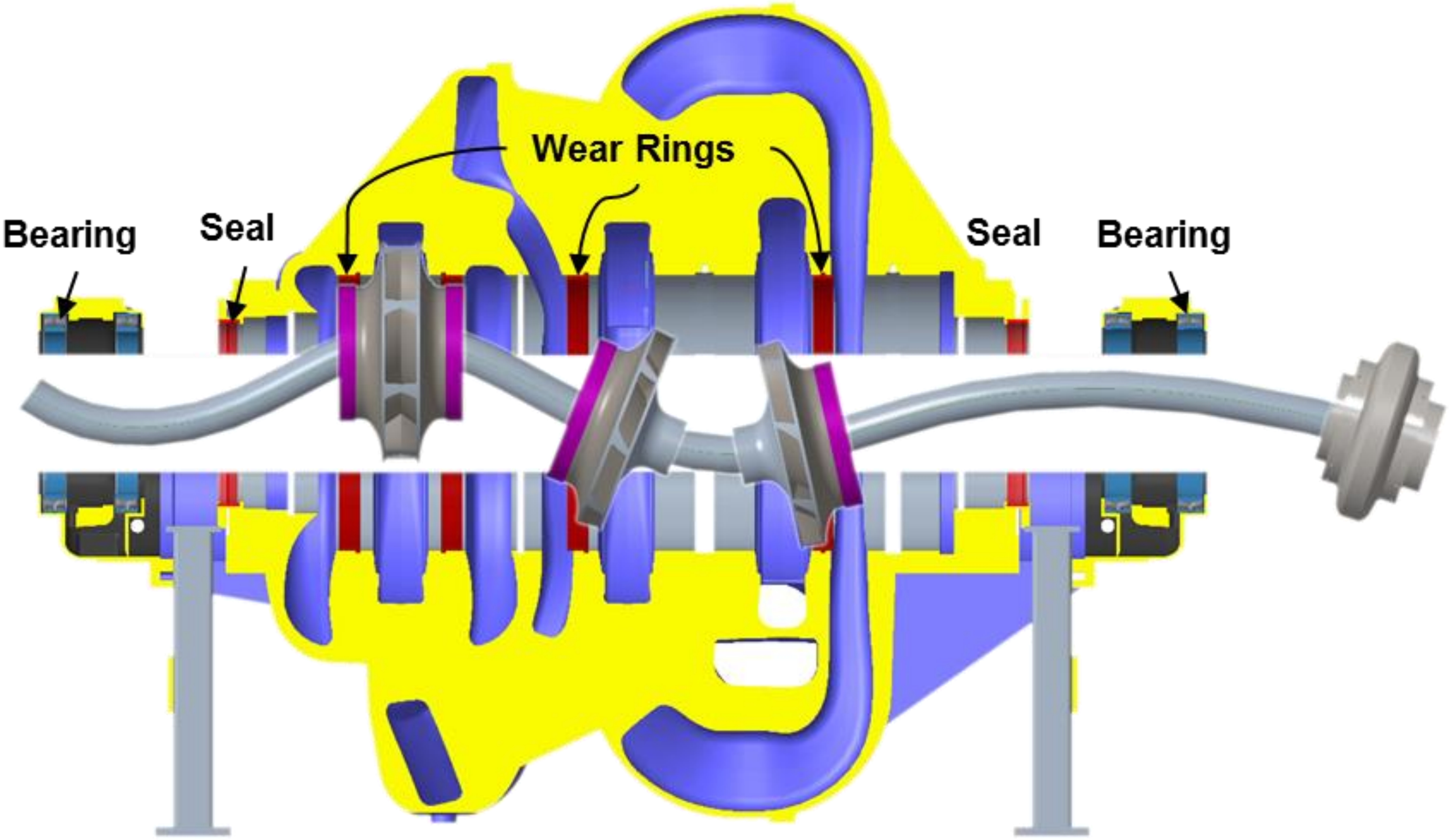
Forced Response:
Sources of Damaging Forces



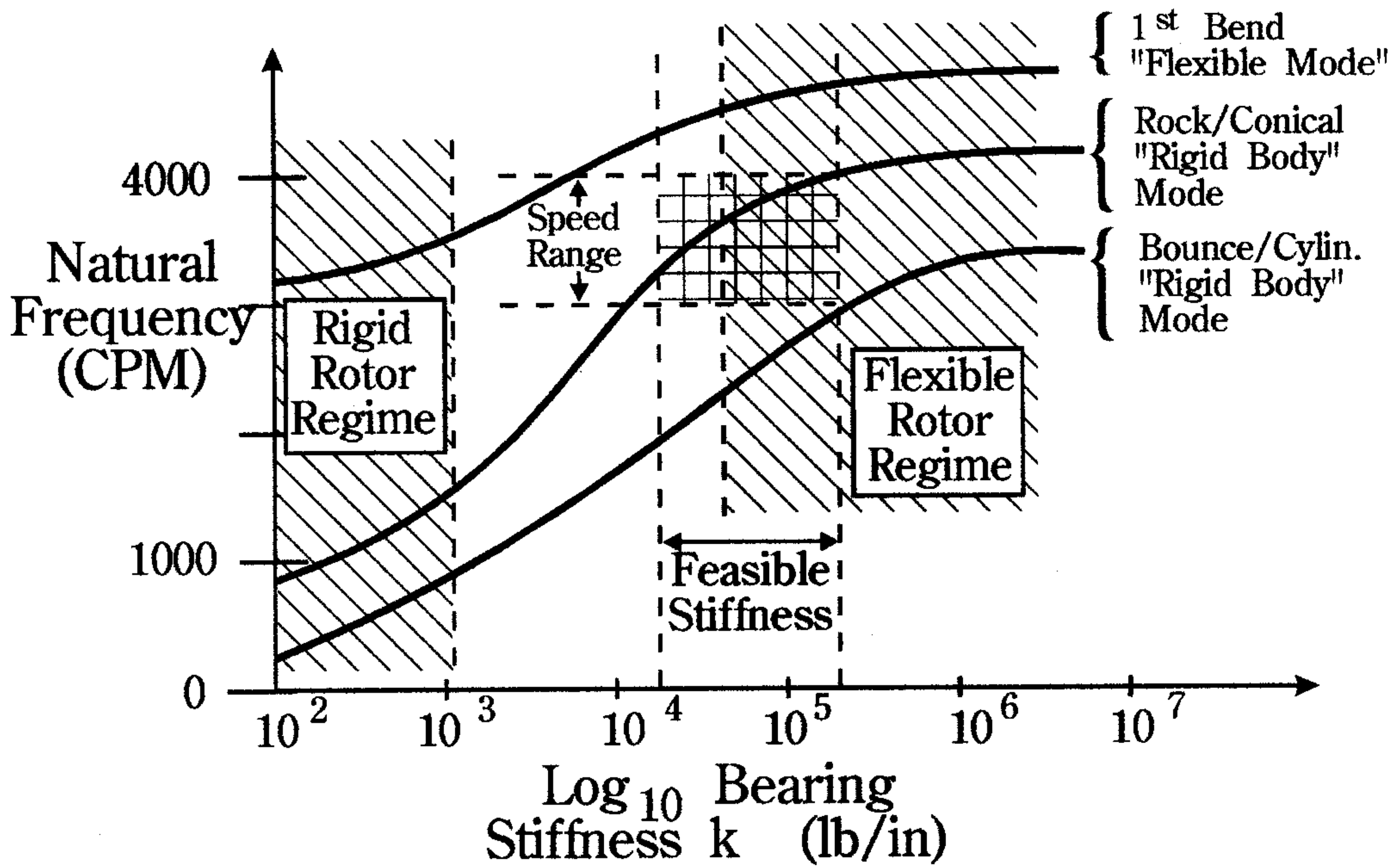
How Natural Frequencies Affect Vibration



Typical Rotor Exaggerated “Mode Shape”

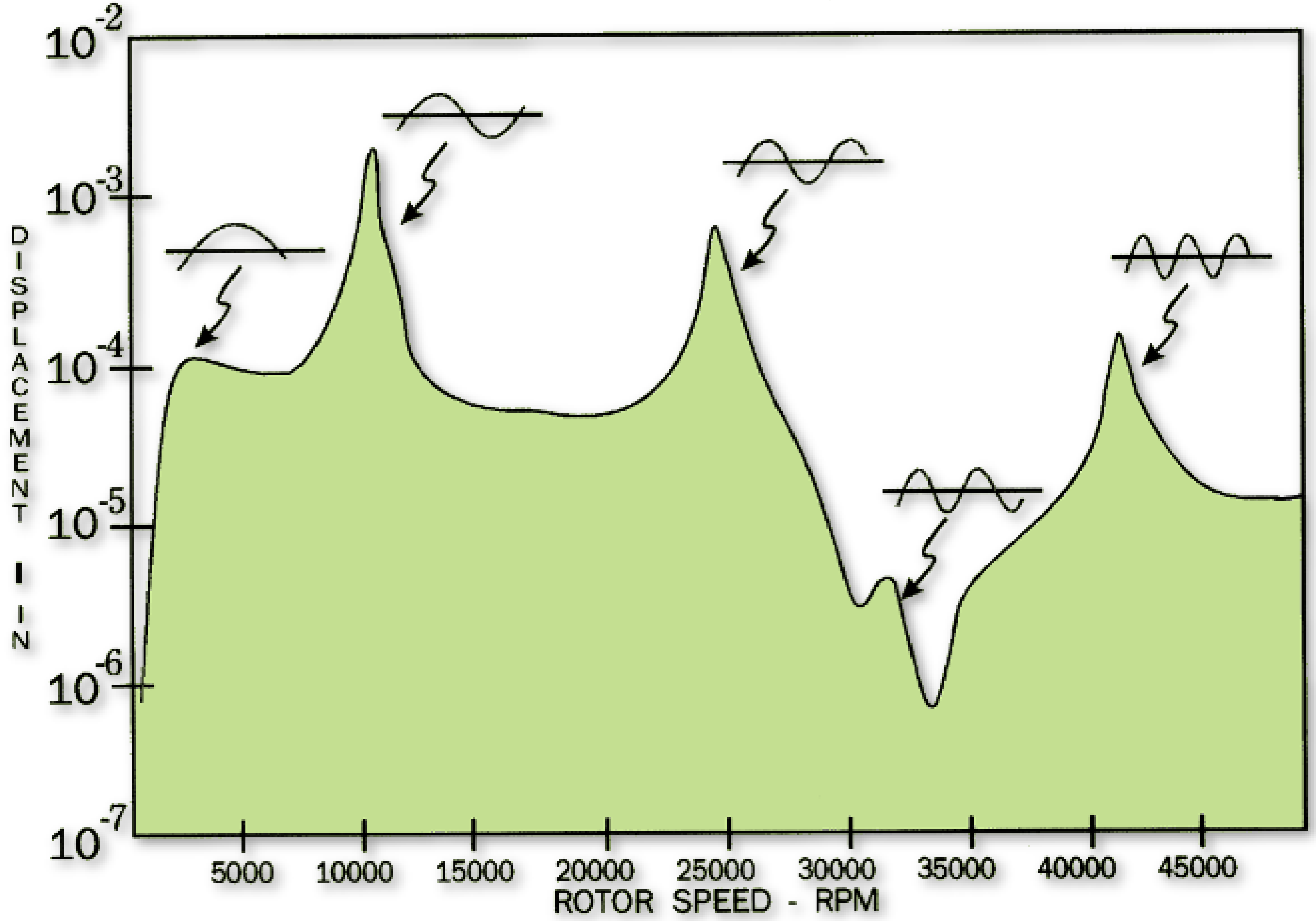


Rotordynamic Critical Speed Map

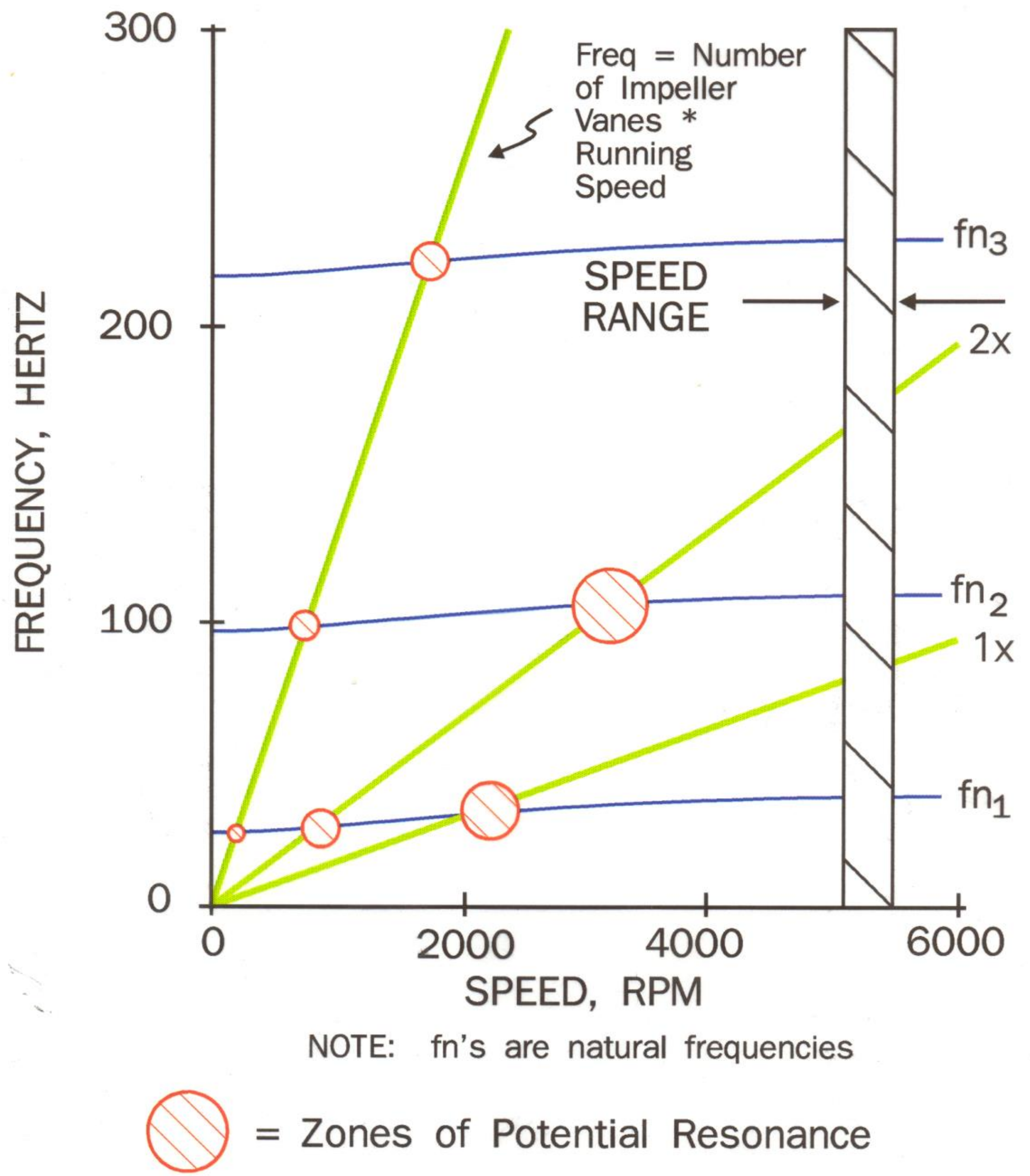


NOTE: This map is initially drawn from calculations where $k_{xx} = k_{yy}$ and $k_{xy} = k_{yx} = c_{xx} = c_{yy} = c_{xy} = c_{yx} = 0$

Typical Rotor Vibration Response vs. Speed Exhibits Several Natural Frequencies



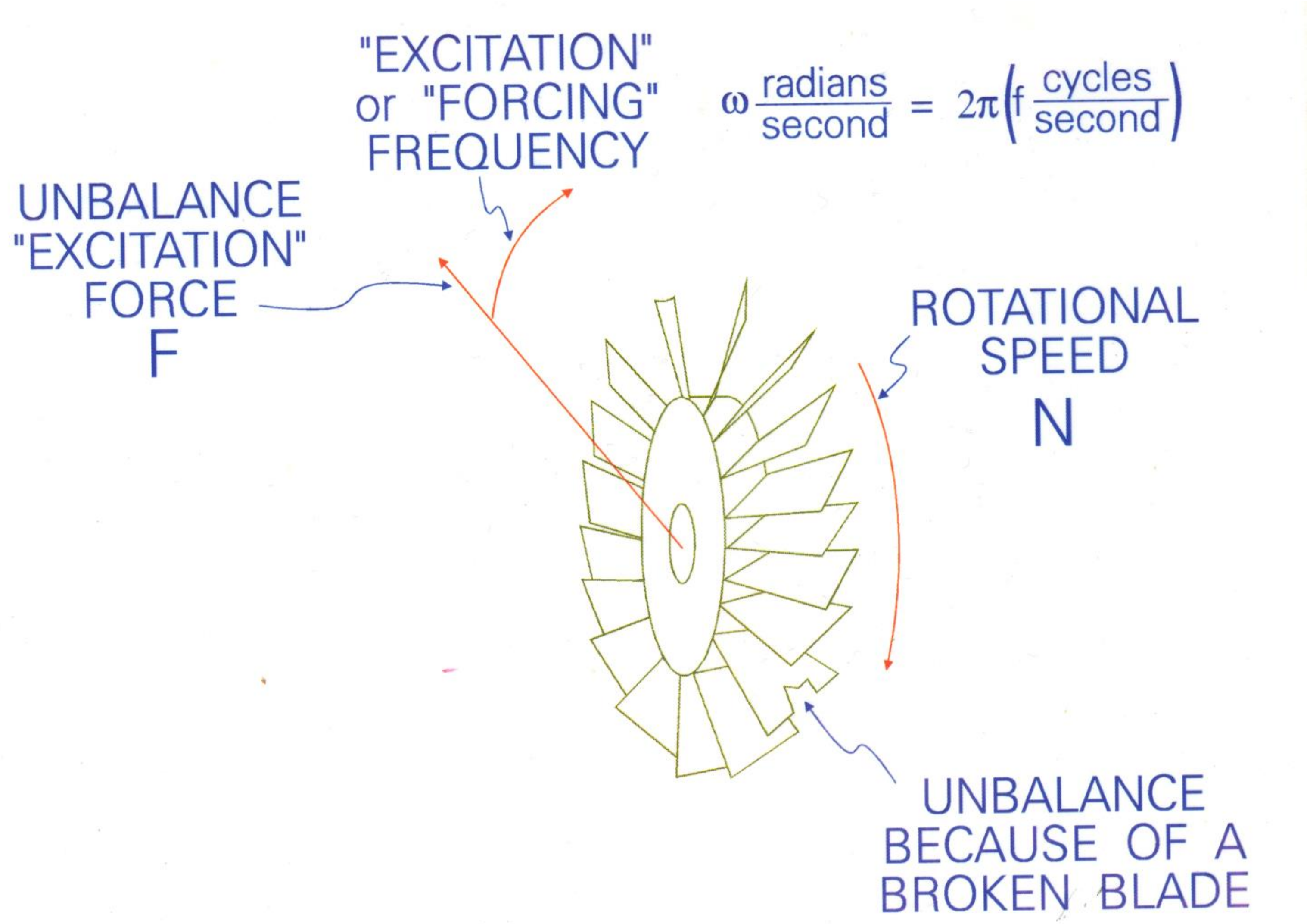
Plotting Resonance w/ Campbell Diagram



Rotordynamic Exciting Frequencies

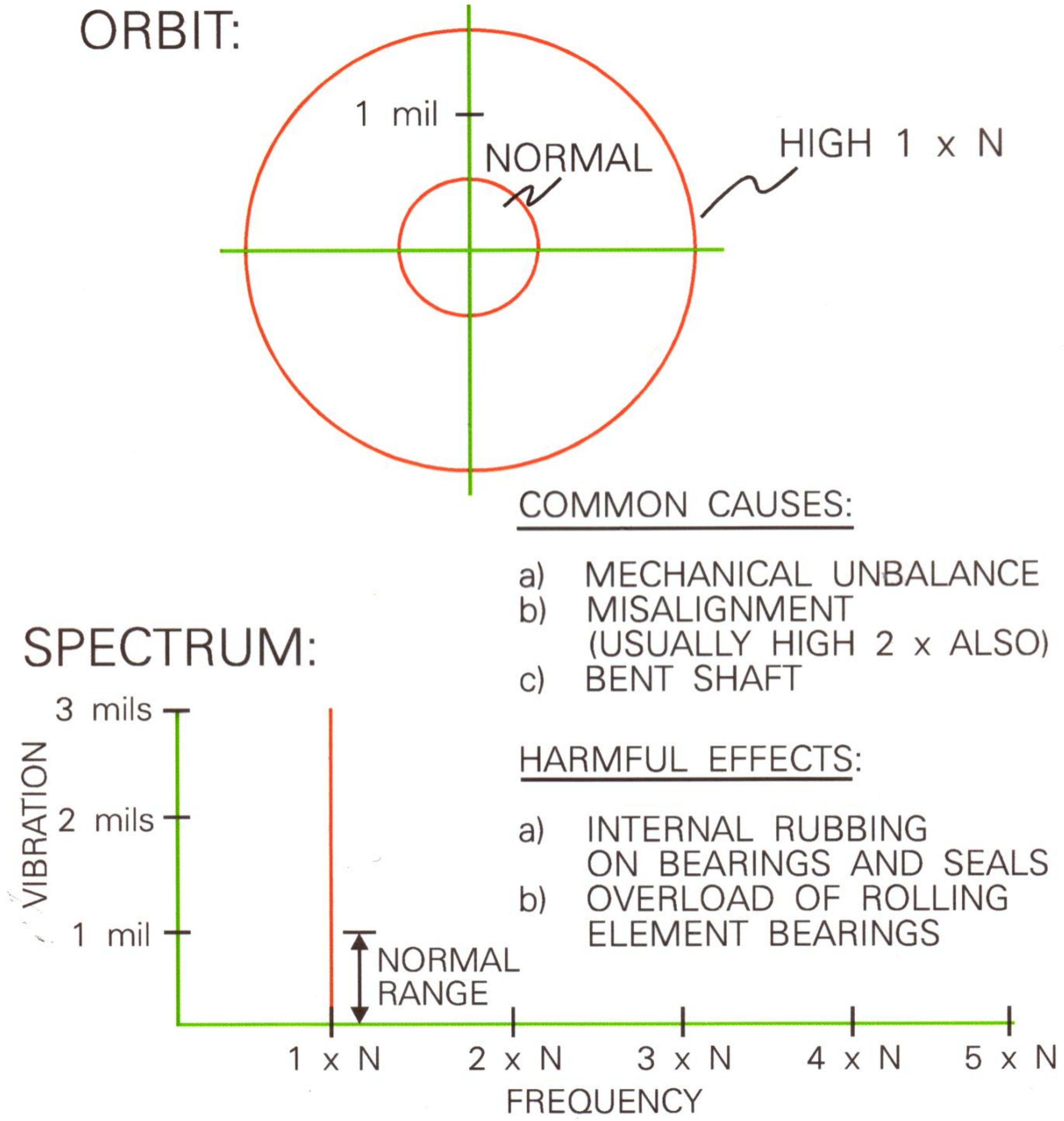
FREQUENCY	SOURCE
0.05 - 0.35 x	DIFFUSER STALL
0.43 - 0.48 x	INSTABILITY
0.500 x	RUBBING
0.65 - 0.95 x	IMPELLER STALL
1 x	IMBALANCE
1 x +2 x	MISALIGNMENT
#Vanes x	VANE/VOLUTE GAP
#Blades x	BLADE/DIFFUSER GAP

Vibration Cause by Oscillating Force
Example: Imbalance



Vibration Problem No. 1:

1x Running Speed



API-610 Allowable Imbalance

API-610 gives a table:

- Between 4W/N and 24W/N
- Depends on rotor bore fit
- Depends on other details
- Weight is lbm per bearing, N is RPM
- Imbalance is oz-inches

Note:

4W/N = ISO G 0.66 (i.e. < ISO G 1.0)

24W/N = ISO G 4.0

40 W/N = ISO 6.6 (i.e. about ISO G 6.3)

<u>Vibration Problem No. 2:</u>

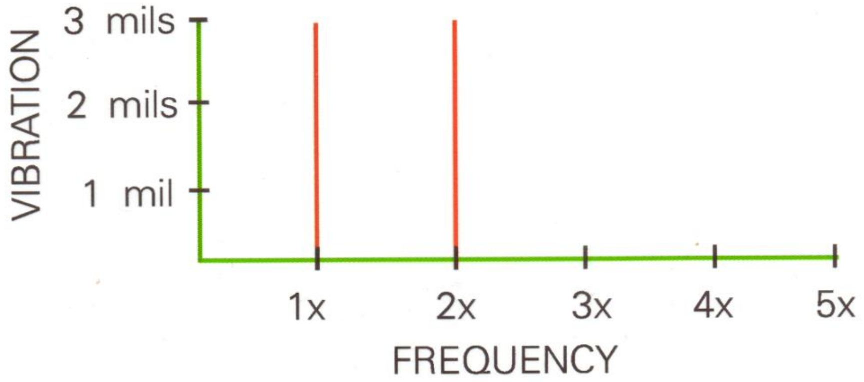
1x & 2x Running Speed

IMPORTANT VIBRATION PROBLEMS IN TURBOMACHINERY

2. HIGH 1x AND 2x

ORBIT:

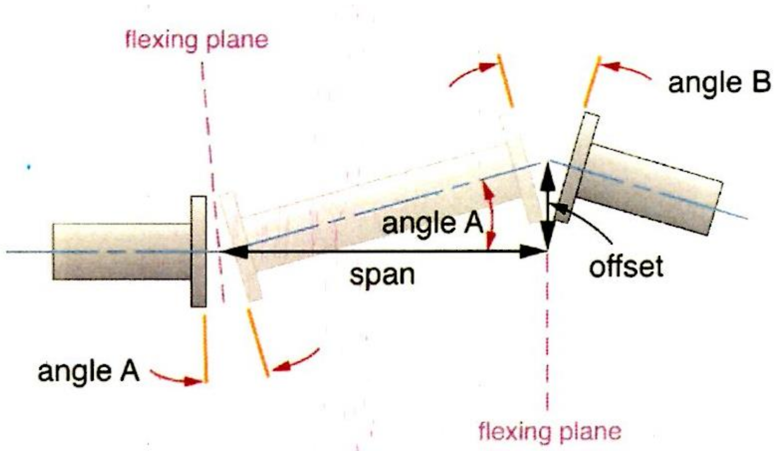
SPECTRUM:



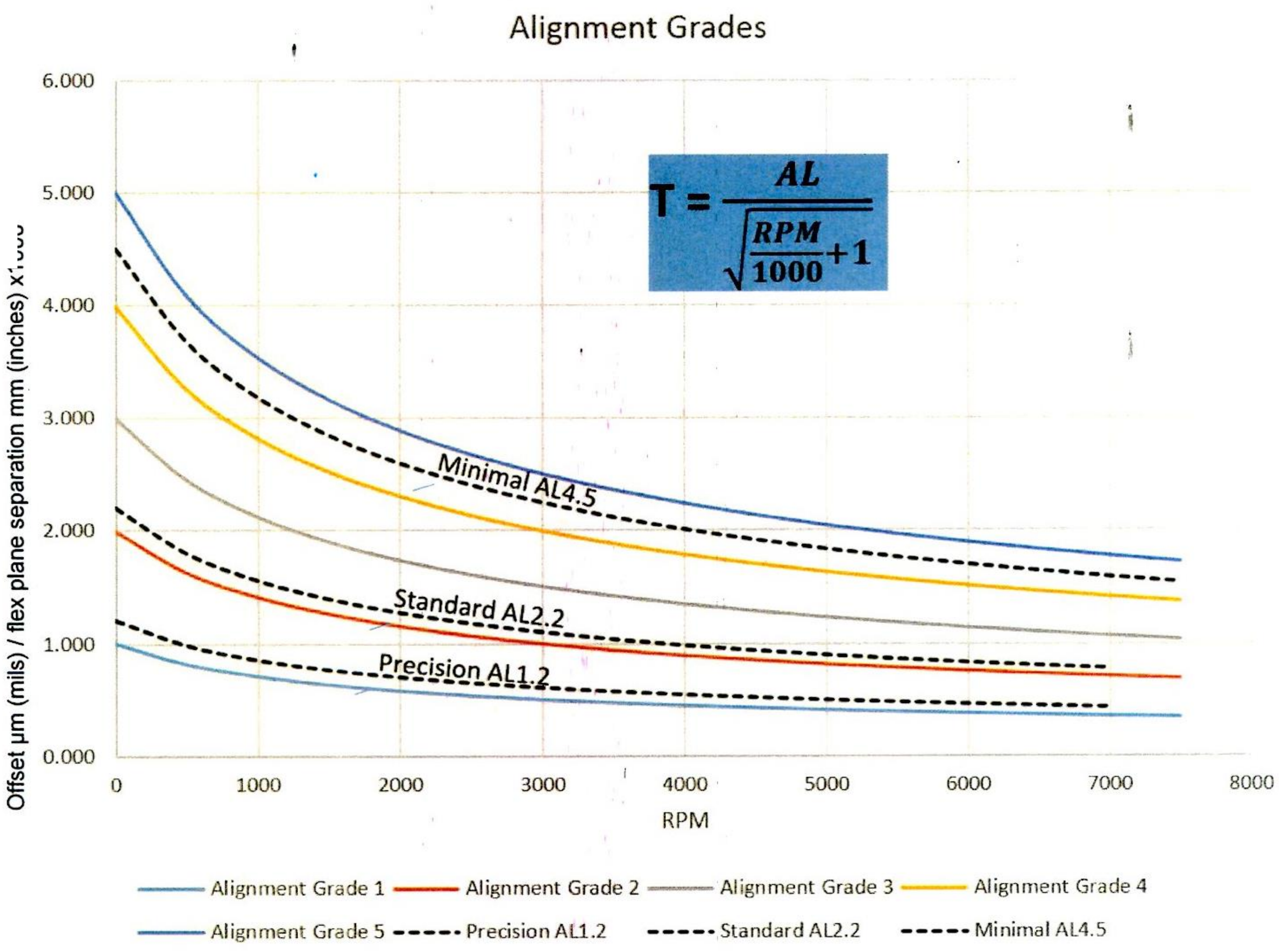
COMMON CAUSES:
a) MECHANICAL MISALIGNMENT
b) LOOSENESS IN BEARING RETENTION
c) SEVERE SHAFT OR BEARING HOUSING CRACK

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

ASA/ANSI Alignment Limits



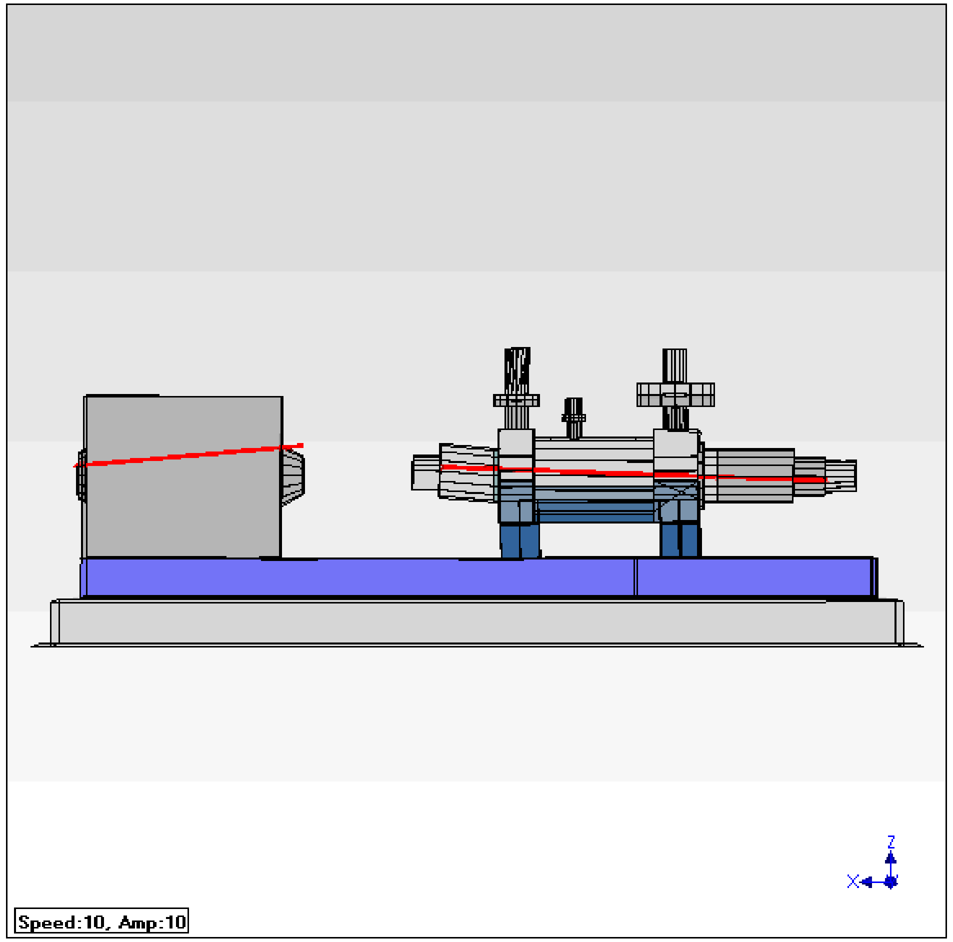
Misalignment represented as angles at the flex planes: $angle\ A(or B) = \frac{offset}{span}\left(\frac{mils}{inch}\right)$



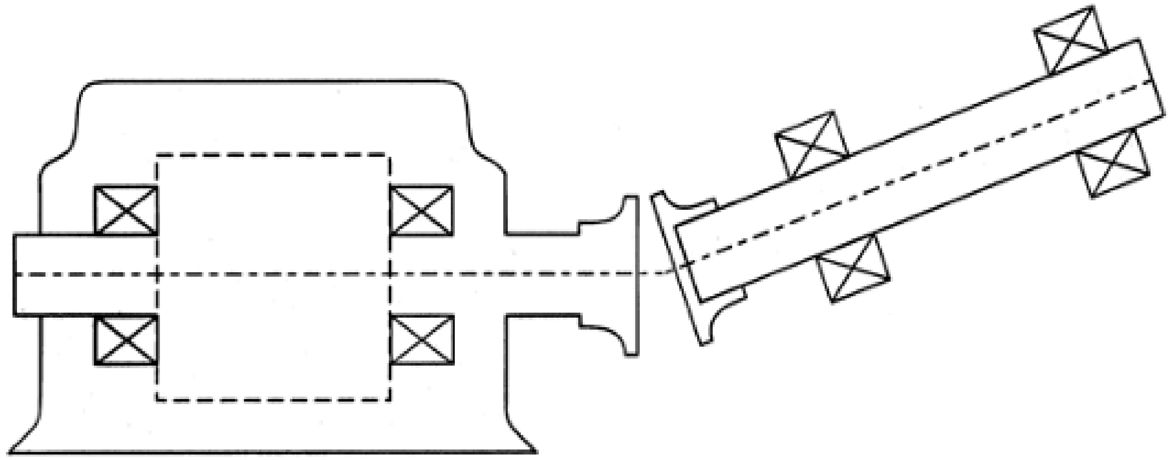
For Offset Only, hub- to-hub, mils, divide vertical values by two.

Source:
ANSI/ASA S2.75-2017/Part 1

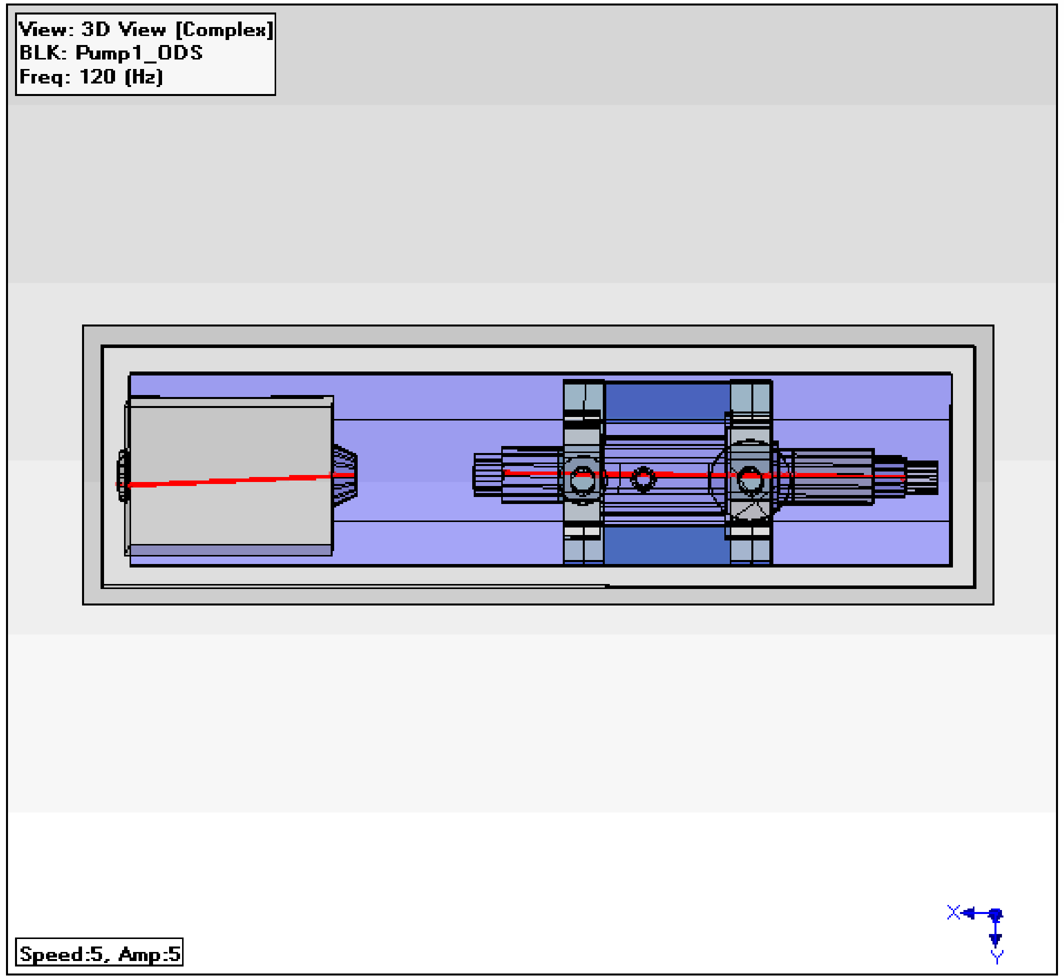
Angular Misalignment

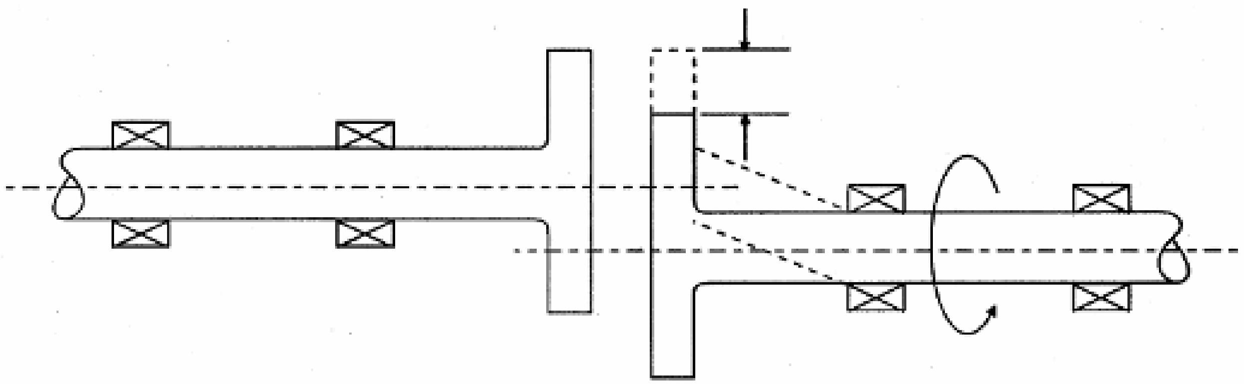


An ODS is created by collecting vibration data and using specialized processing. It shows exaggerated motion (but to scale) at specific frequencies. Left - animated Operating Deflection Shape (ODS) of typical angular misalignment.



Offset Misalignment



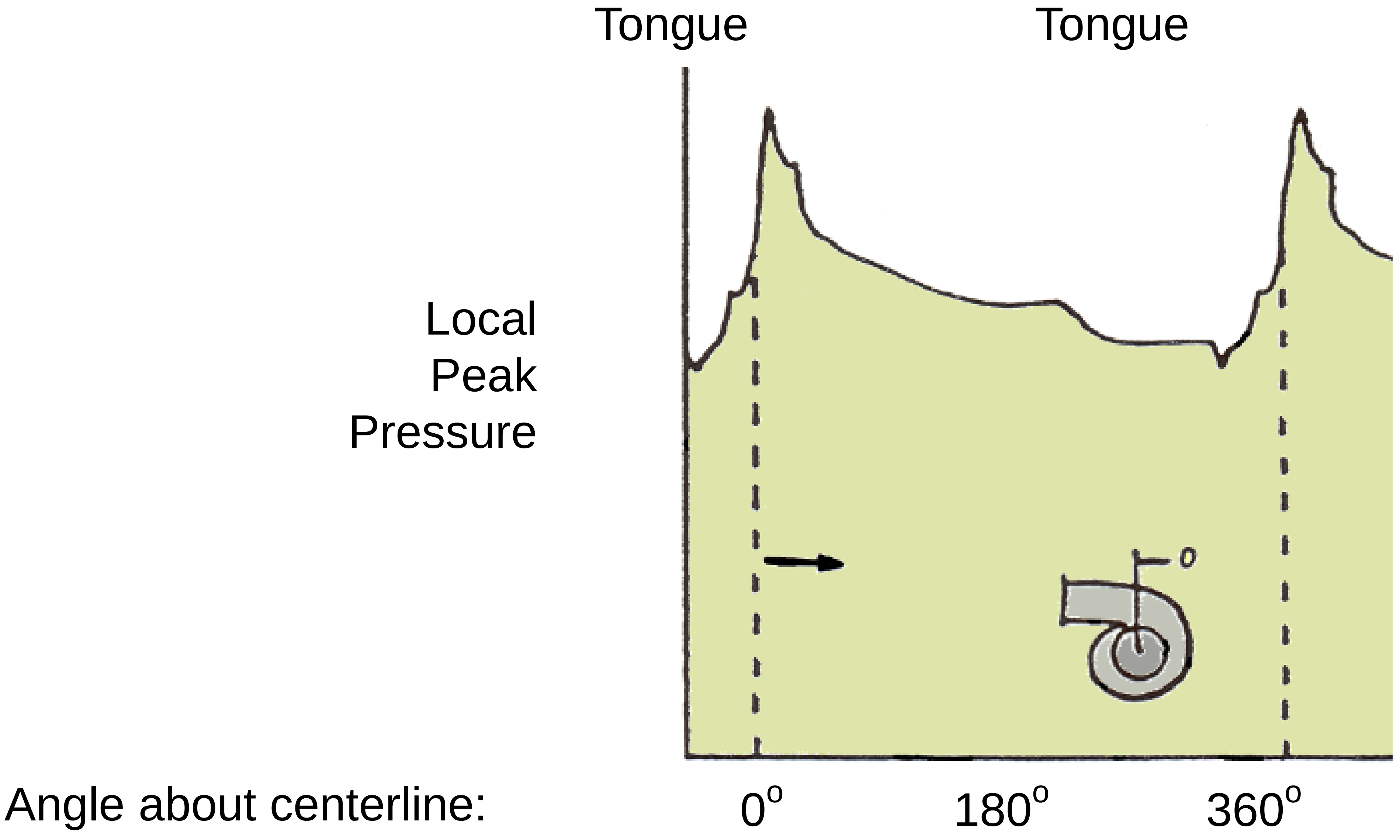


An ODS is created by collecting vibration data and using specialized processing. It shows exaggerated motion (but to scale) at specific frequencies. Left - animated Operating Deflection Shape (ODS) of typical parallel misalignment.

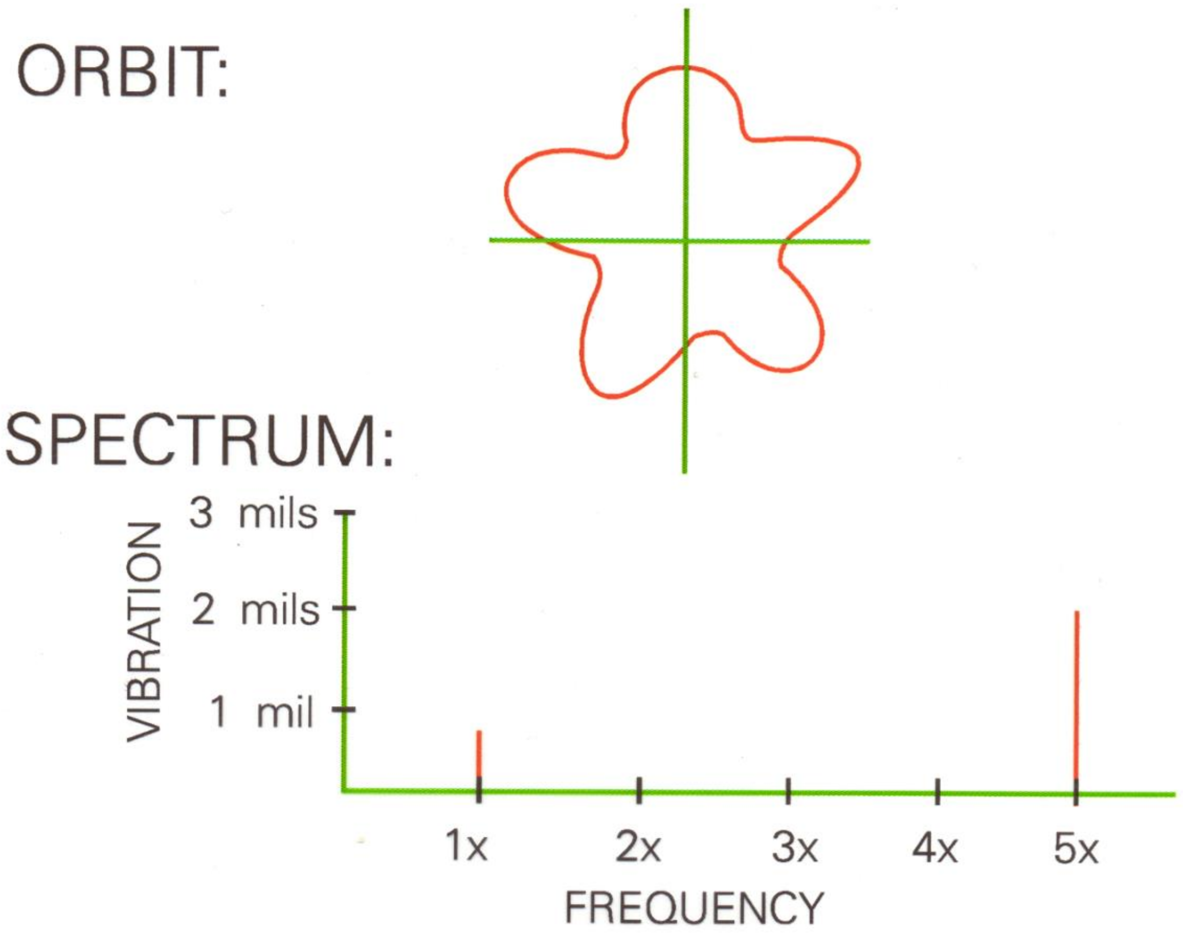
Method of Accounting Misalignment Force

- Determine net offset between pump & driver coupling hubs
- Assign 1/2 of this as effective imbalance eccentricity to coupling hub, and 1/4 of this to spacer spool piece, acting at pump coupling hub
- Run unbalance response analysis including this effective extra "imbalance"

“Vane Pass” Vibration Source



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

Method of Including Vane Pass

- Put in as an effective imbalance load
- Good approximation since vane pass load depends on square of speed, just like imbalance
- Level of effective imbalance discussed later. Can be as high as discharge pressure times discharge “projected area” (D x blade height) times 0.36. Usually about 10x lower than this.

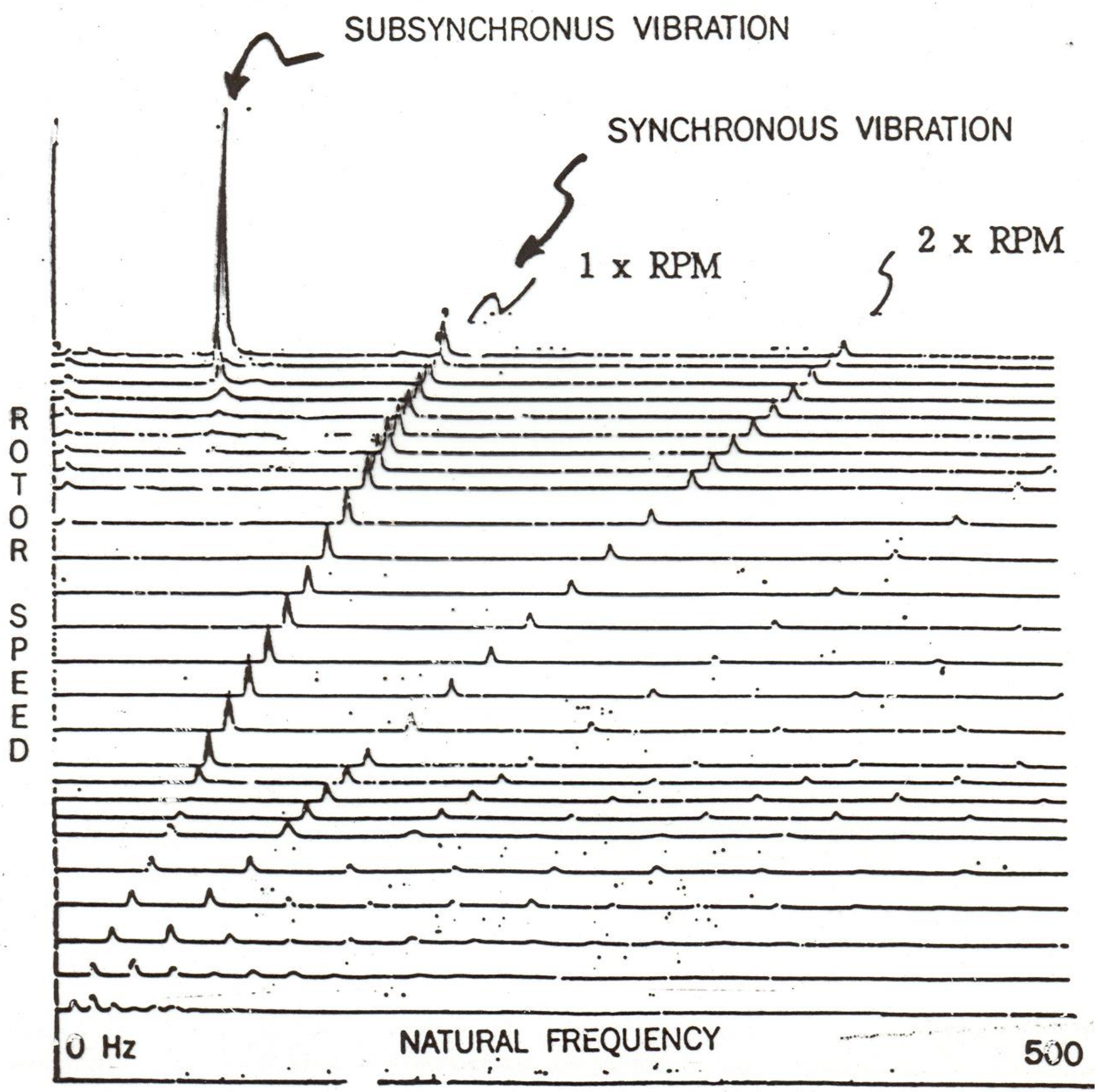
The Most Frequent "Subsynchronous" Vibration Problem:

Rotordynamic Instability

Symptoms:

- "Half speed" (48%N) whirl
- Other frequencies occur!!
- Loop-de-loop orbit
- Cross-coupling > Damping

Unstable Rotor Whirl at "Half" Running Speed



Cross-Coupled Stiffness

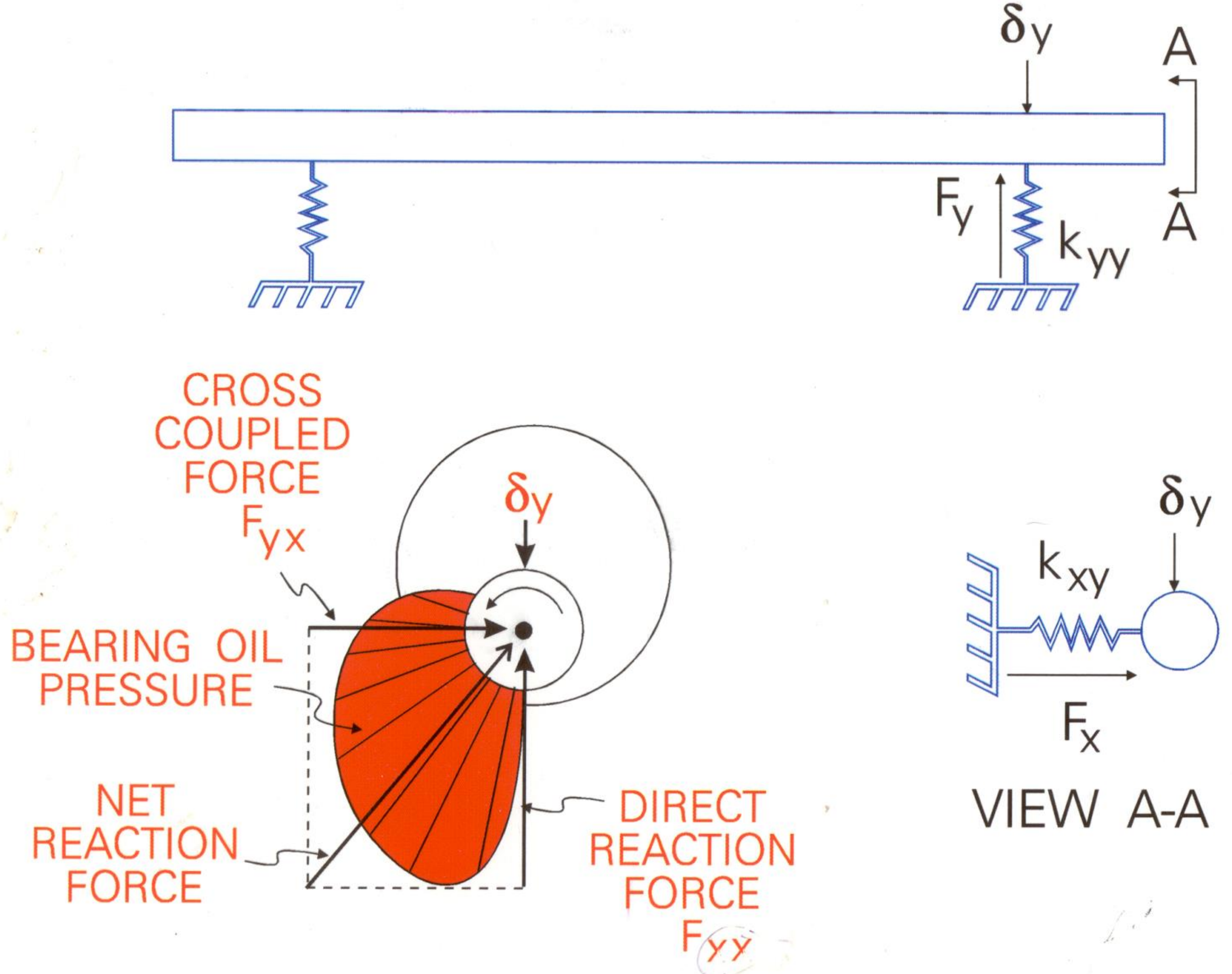
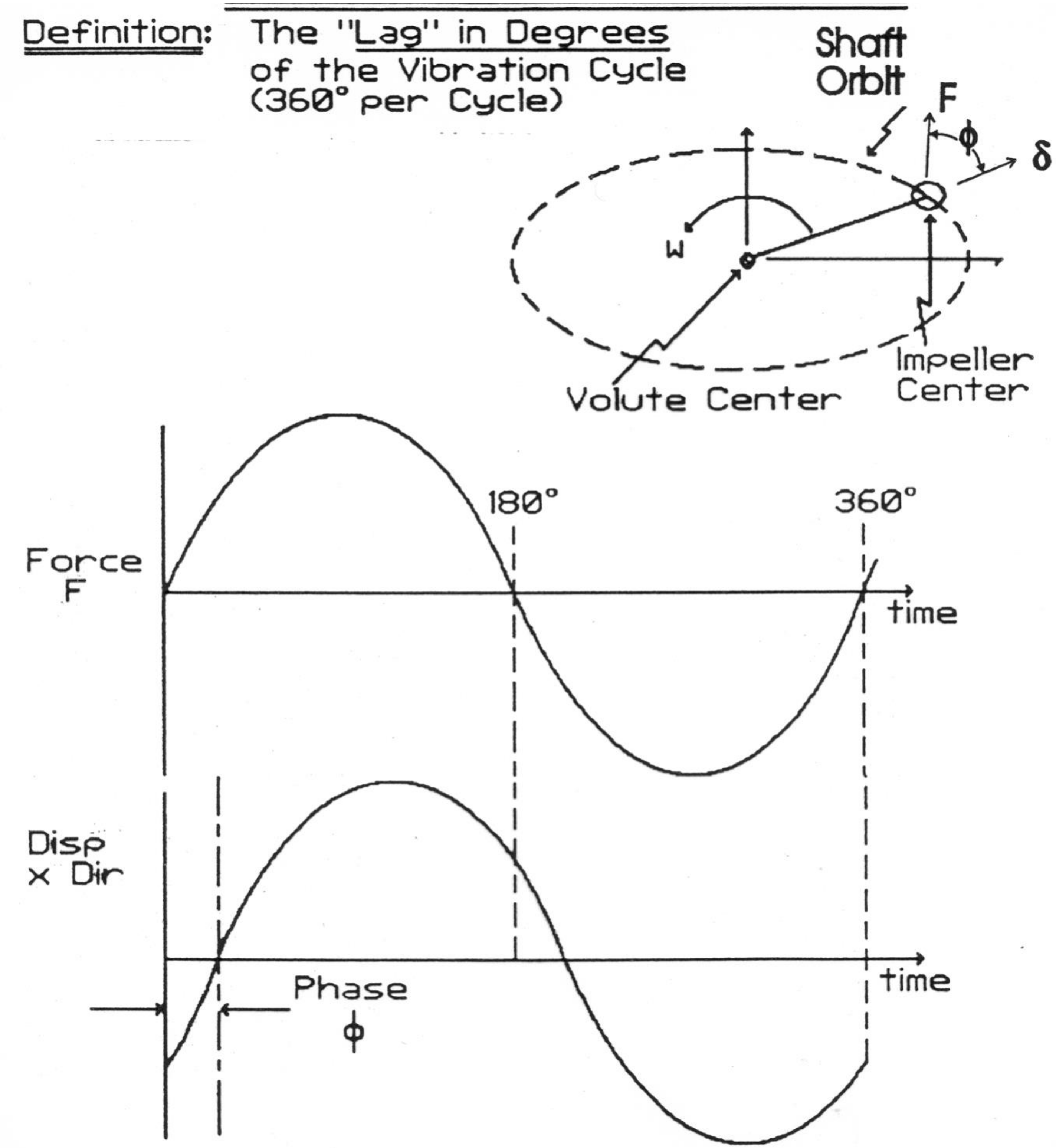
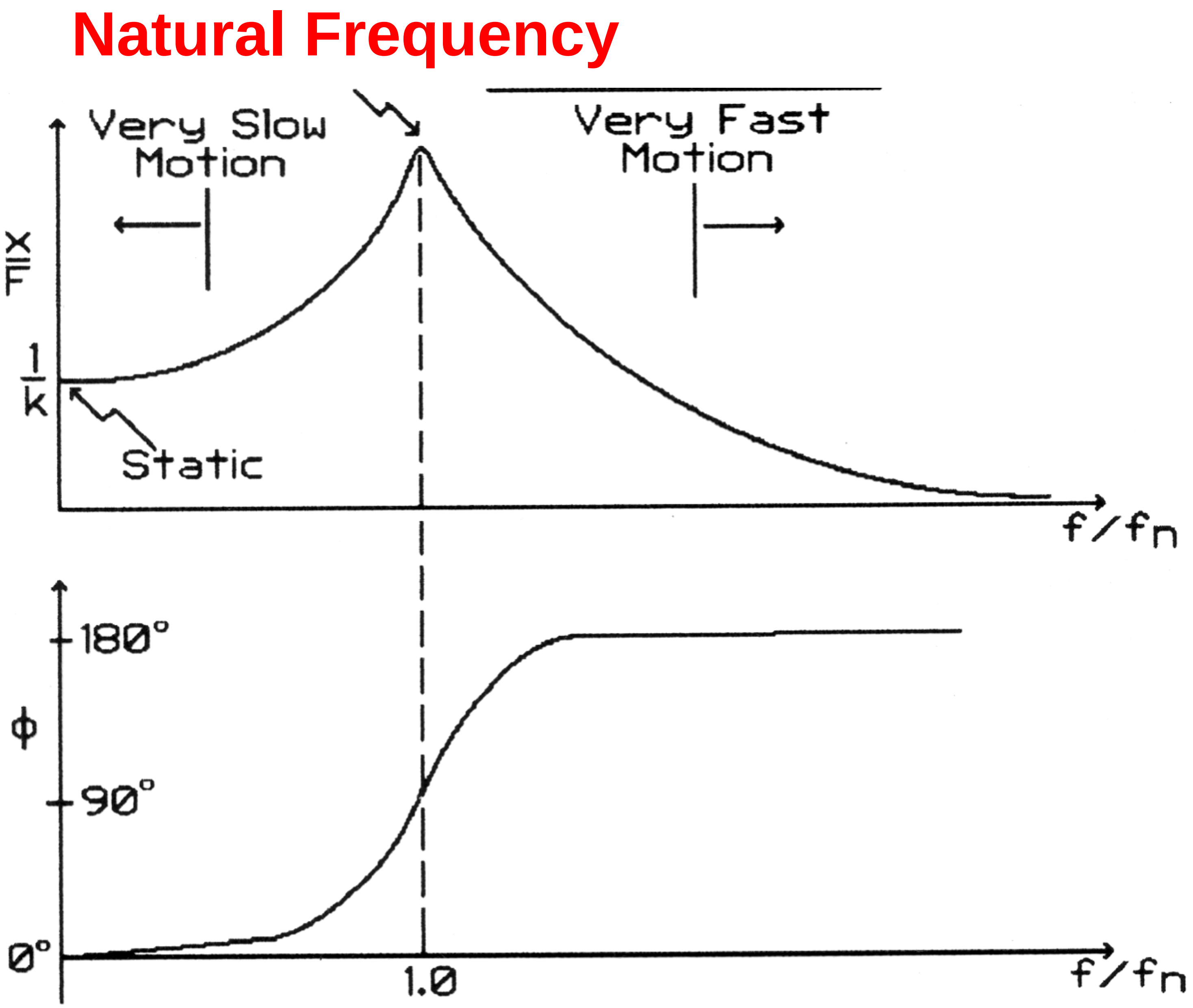


Illustration of Phase Angle



Vibration & Phase vs. Frequency



How Cross-Coupling Leads to Instability

1) CROSS-COUPLING:

CAUSE (DISPLACEMENT)
& EFFECT (SHEAR FORCE)
ARE 90° OUT OF PHASE AT
LOW ω/ω_N

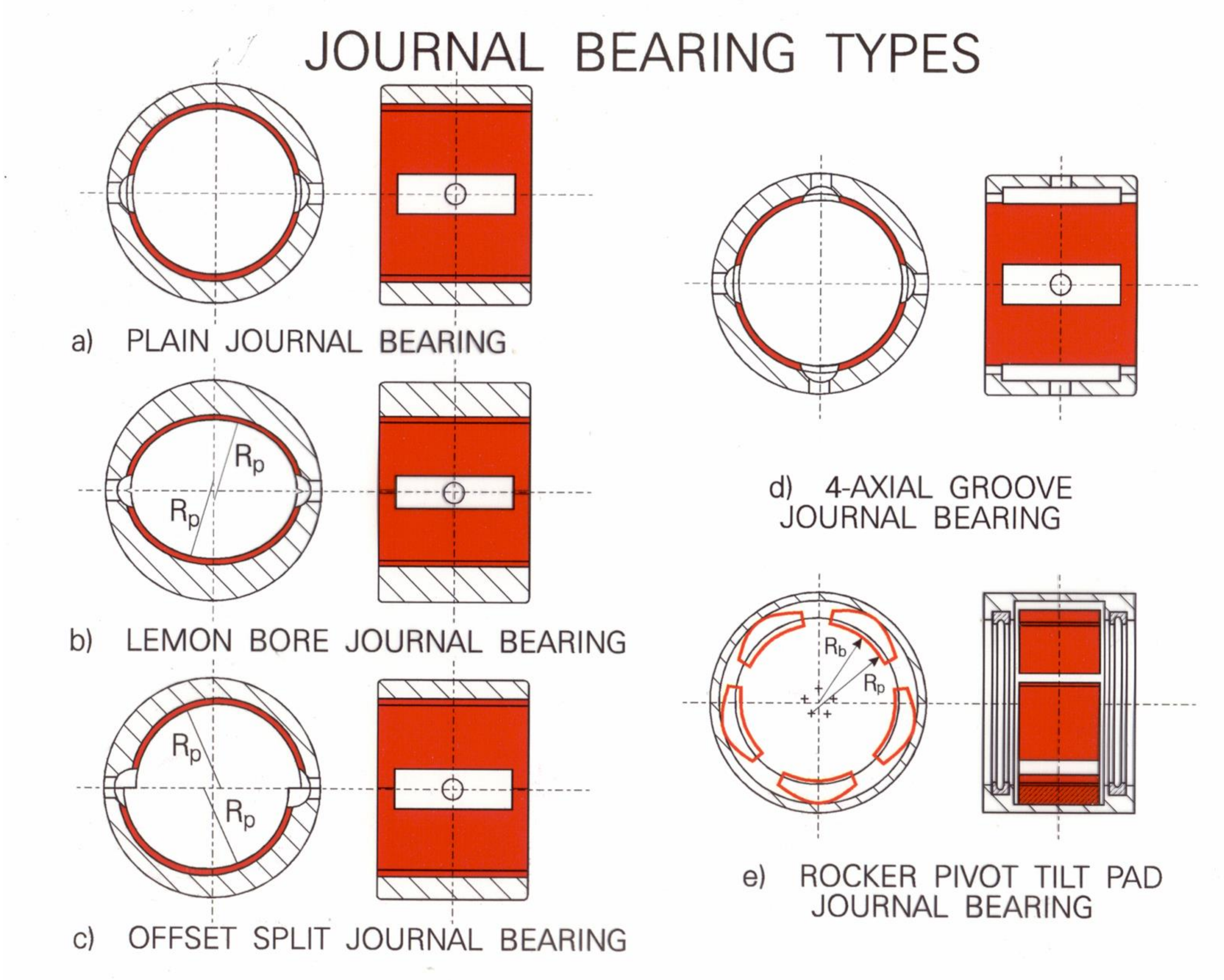
2) CAUSES SLOW PRECESSION OF
ROTOR, WITH PRECESSION SPEED
OF ~1/2 ROTOR SPEED

3) WHEN $\omega_{PRECESSION} = \omega_{1st\ CRIT\ SPEED}$,
ADDITIONAL 90° PHASE SHIFT
OCCURS

⇒ Response displacement ends up in the same direction as the cross-coupling force, in the same direction as the minimum clearance.

⇒ Enough damping trumps cross-coupling

Anti-Whirl Bearing Fixes



Issues in Rotordynamics
Not Well Enough Understood

- Hydrodynamic “active” forces
- Impeller reaction coefficients
- Swirl incoming to annular seals
- Effect of spiral grooving in seals
- Unique seal groove geometries

Impeller Excitation Forces: Sulzer/ EPRI Tests

Alternate Rule-of-Thumb:

F_R^* =0.1 to 1.0 K

Where K= Steppanoff Radial Thrust Factor

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

D_2 = Impeller diameter, B_2 = Impeller exit width

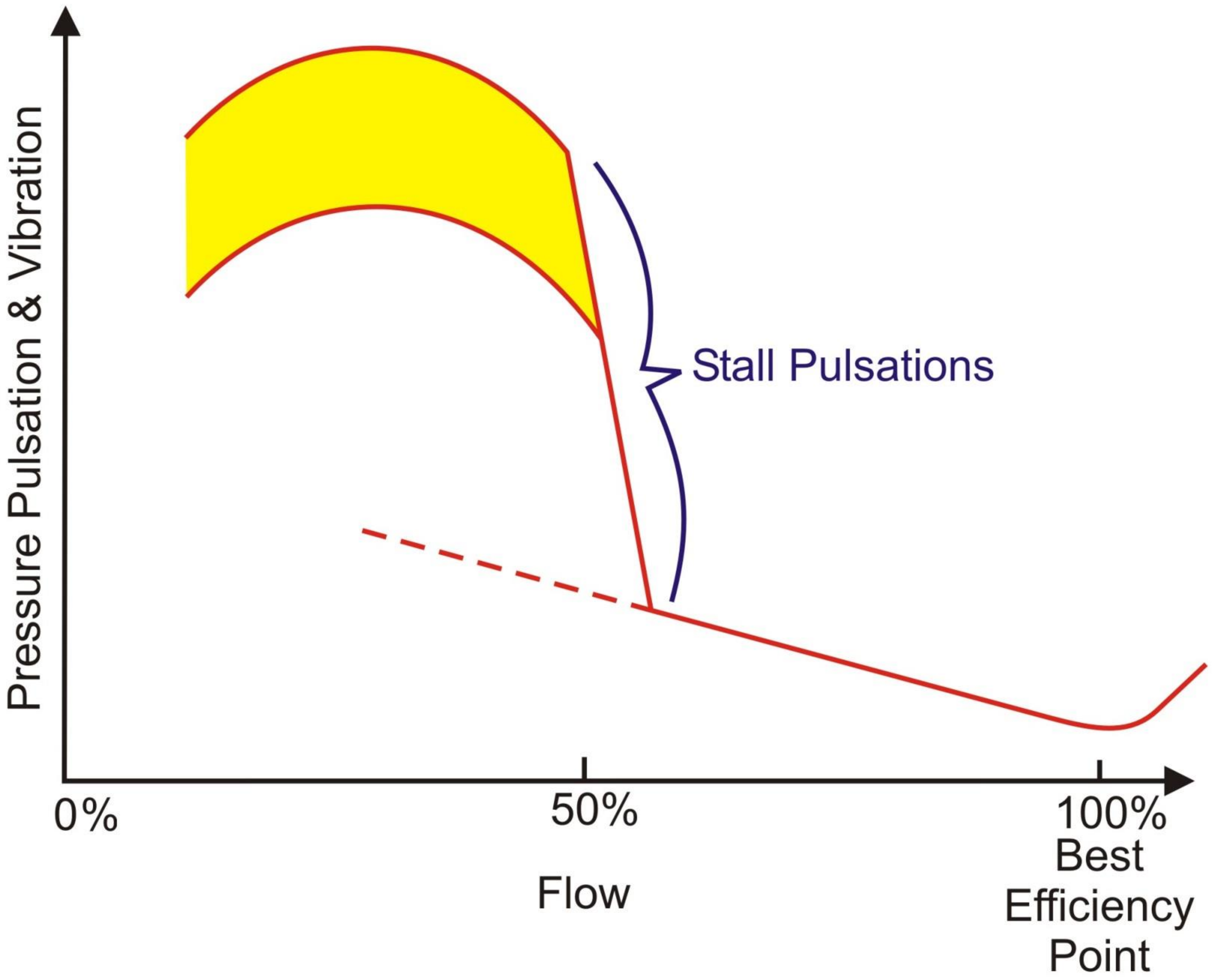
ρ = Density, g = Gravity, H = Head

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)

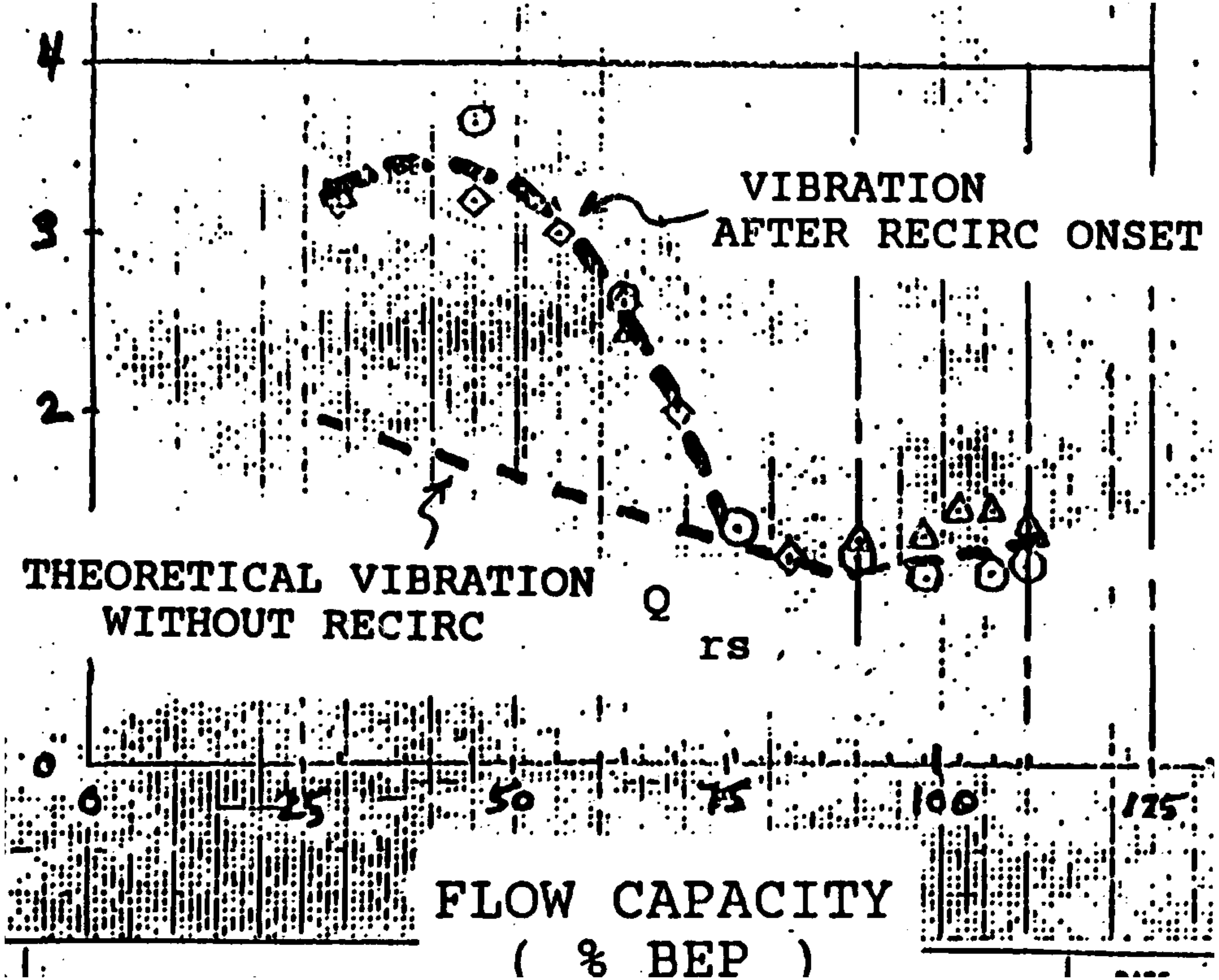
Ranges for F_R for Q=25% to 125% of Design Point
Values in brackets: typical for Design Point

Vibration Pulsation vs. Flow



Actual Field Data Vibration vs. Flow
Axially Split Case Double Suction Pump

Vibration
(mils p-p)



API-610 Accounting of Flow Effects

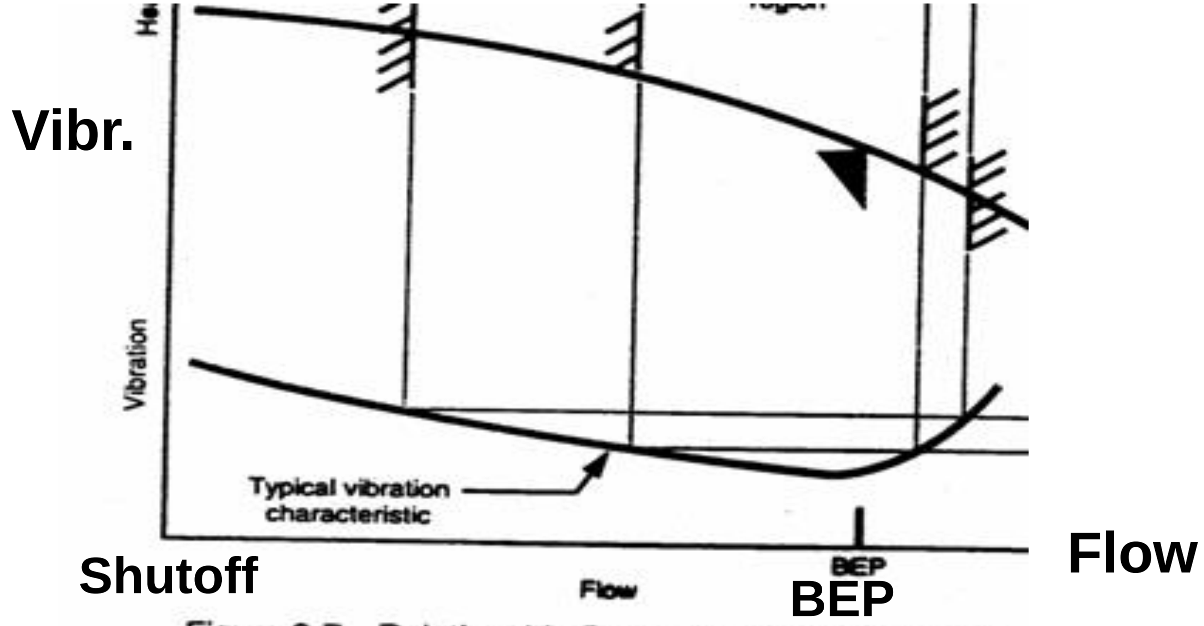
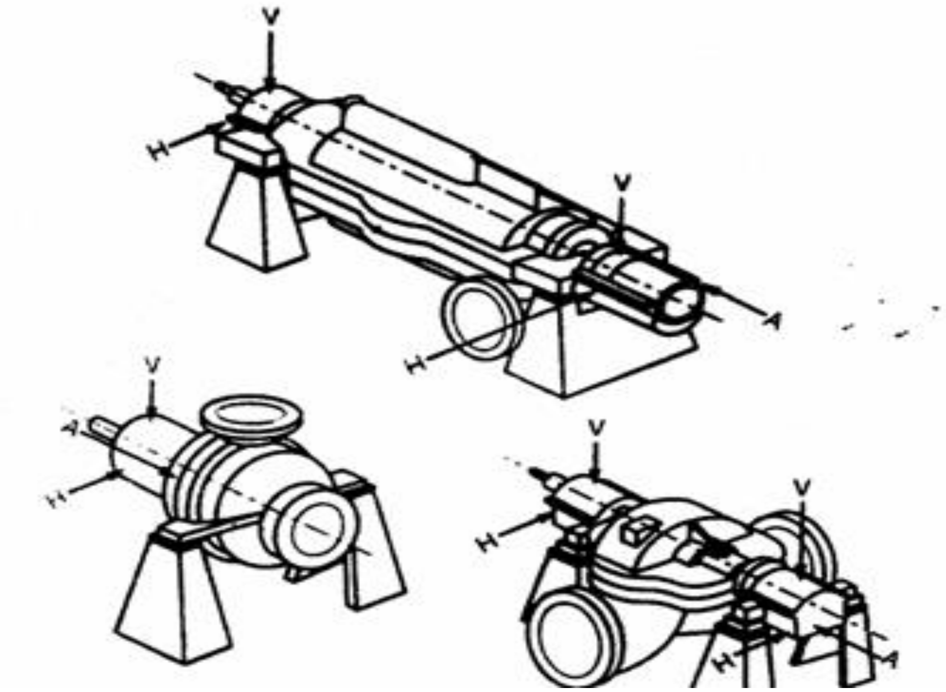
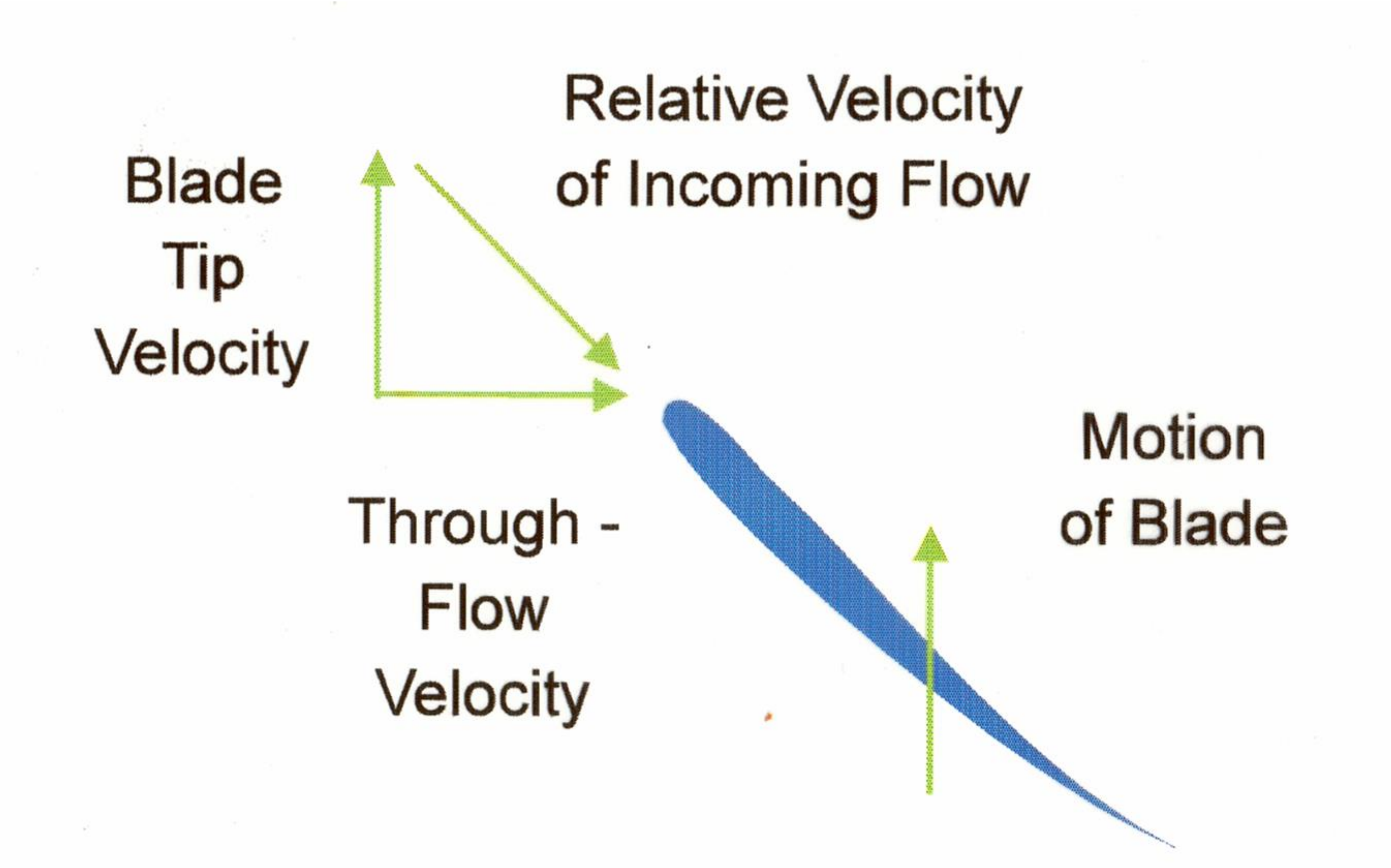


Figure 2-7—Relationship Between Flow and Vibration

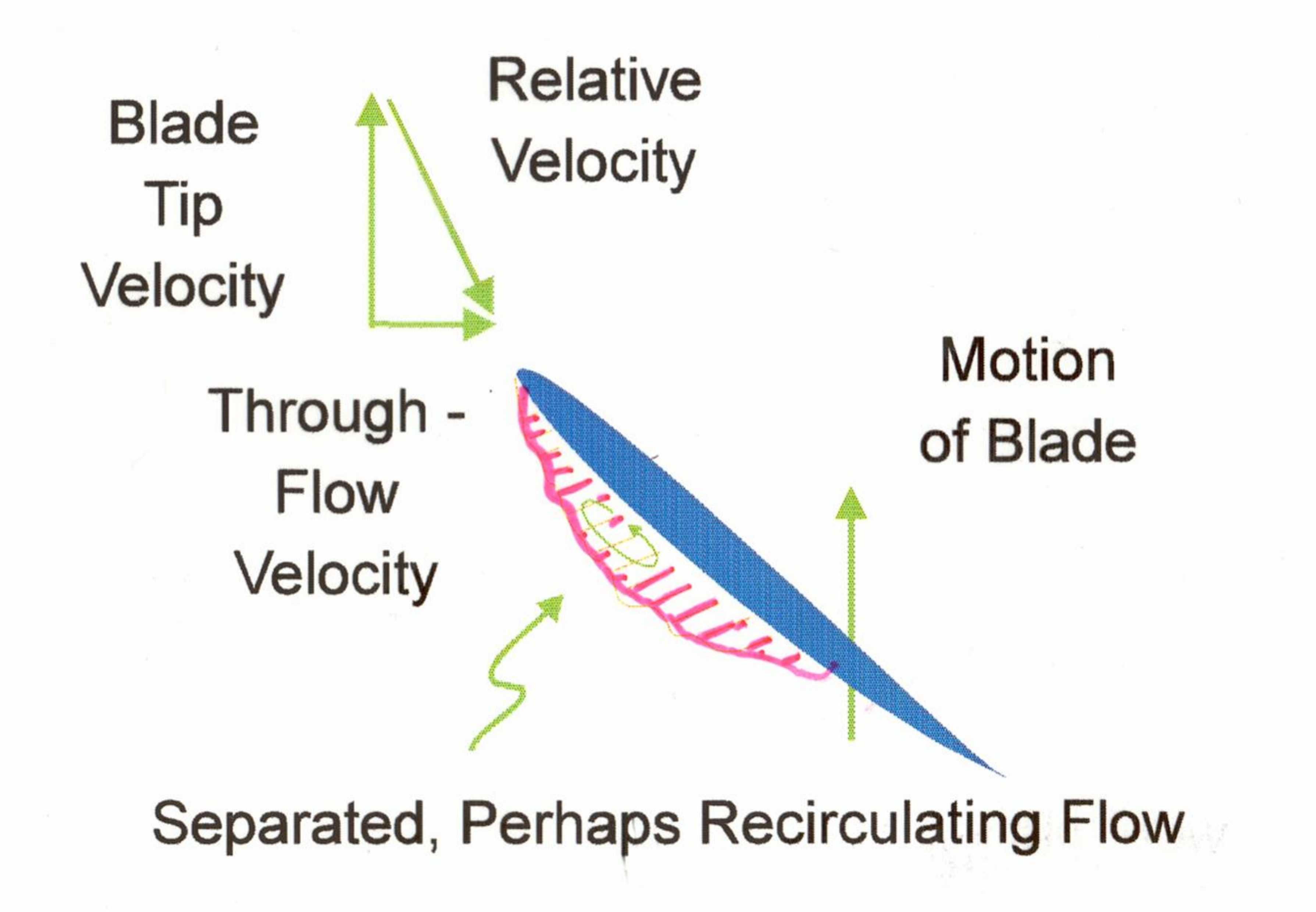


Recommended vibration measurement locations

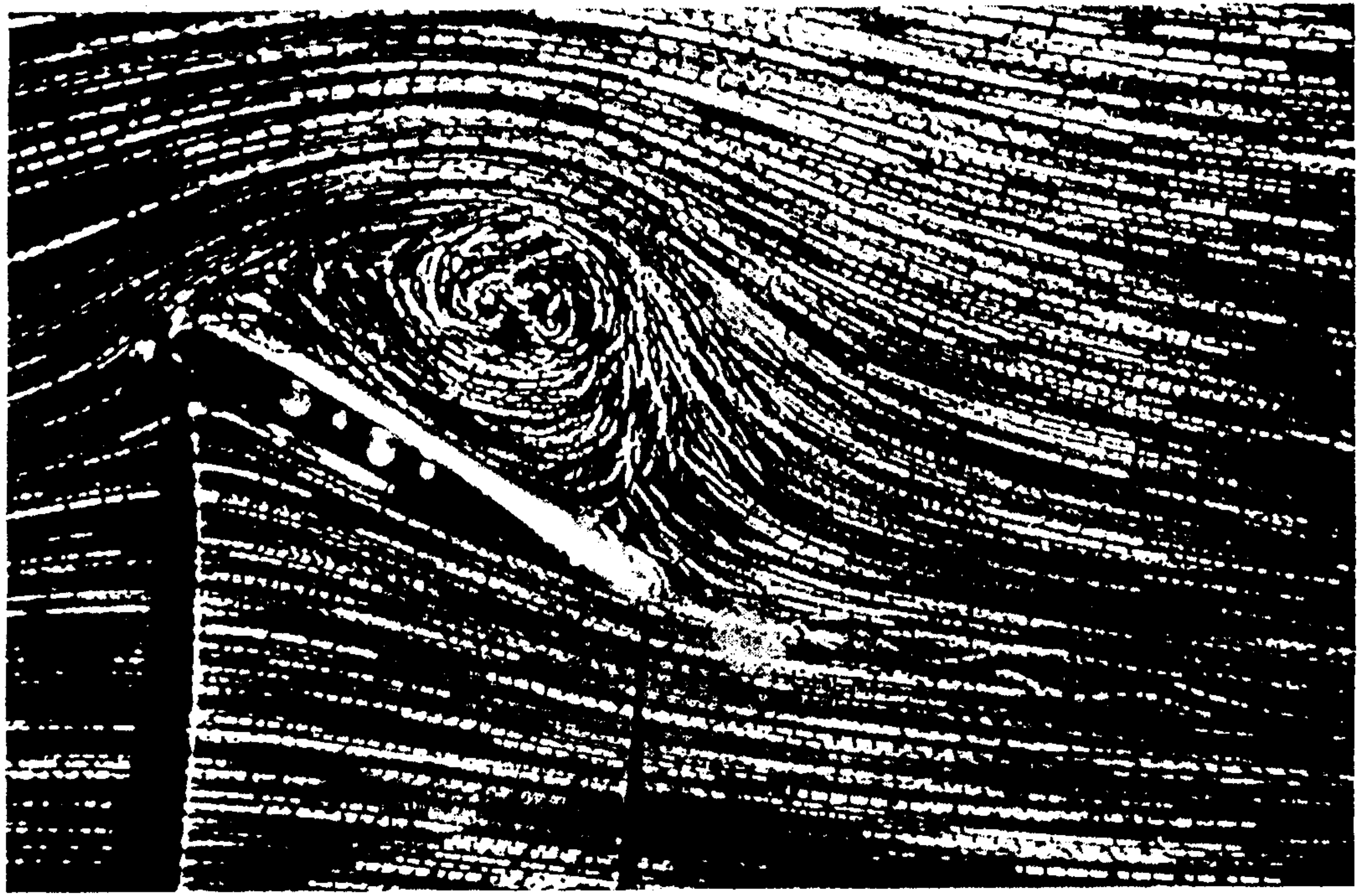
Flow Rate and the "Angle-of-Attack"



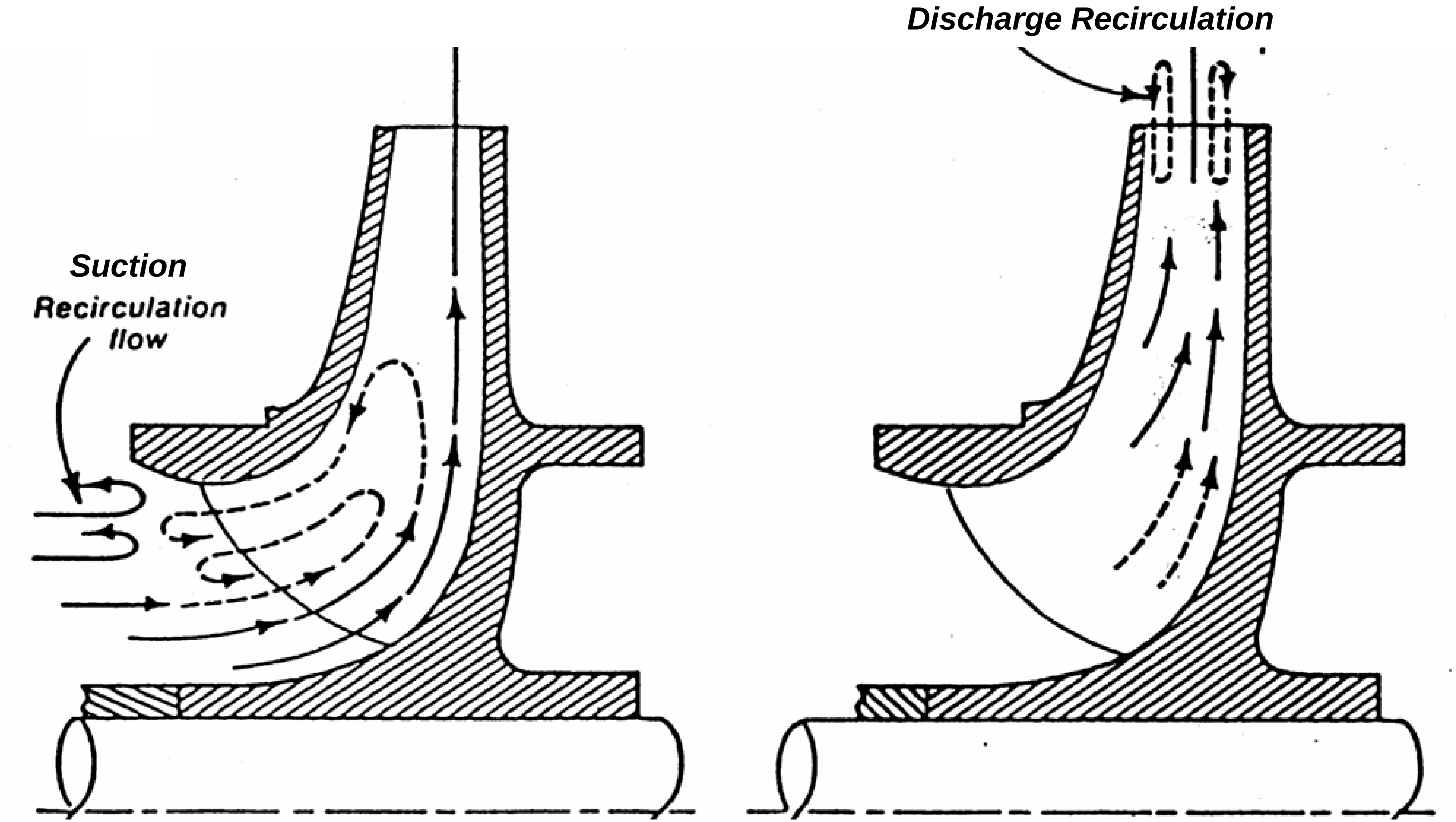
Vane Stalling at Low Flows



Example of Stalled Blade



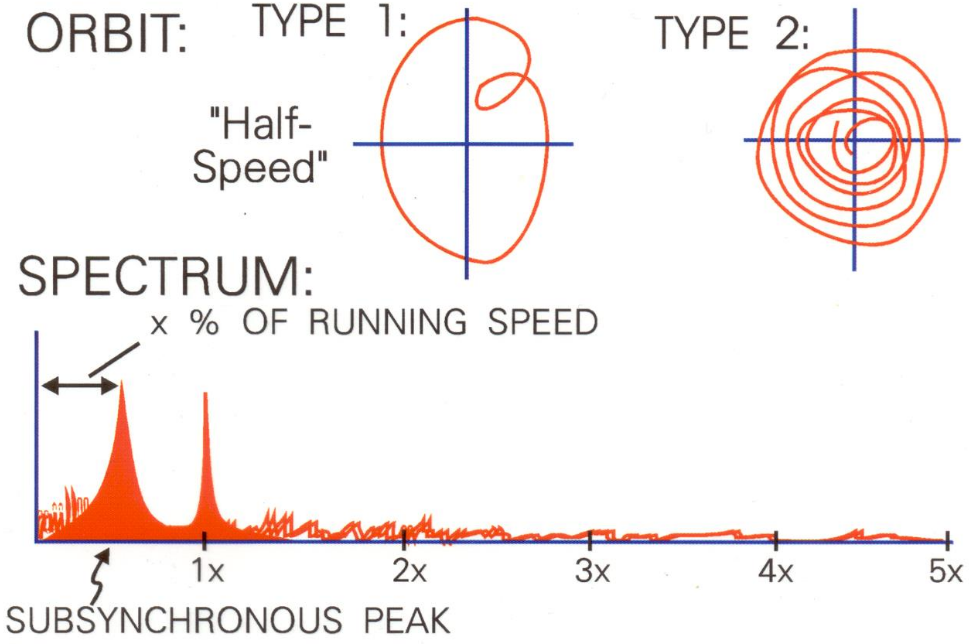
Onset of Internal Recirculation



Vibration Problem No. 4: Subsynchronous (below 1x)

IMPORTANT VIBRATION PROBLEMS IN TURBOMACHINERY

5. "BROADBAND" SUBSYNCHRONOUS VIBRATION

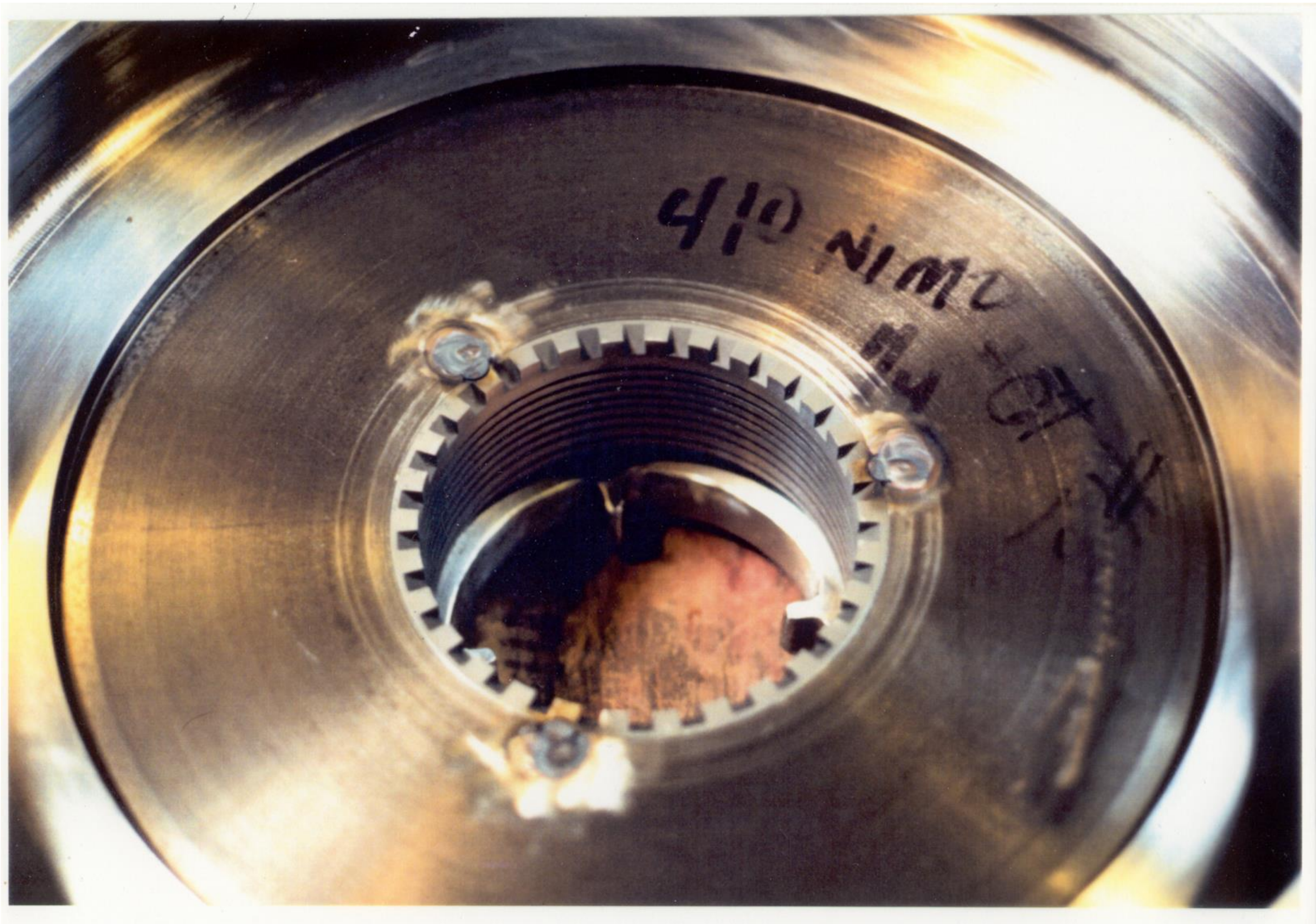


POSSIBLE CAUSES:

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

Suppressing Incoming Swirl



Impeller Radial Reaction Forces: Sulzer/ EPRI Tests

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

D_2 = Impeller diameter,

g = Gravity, H = Head

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

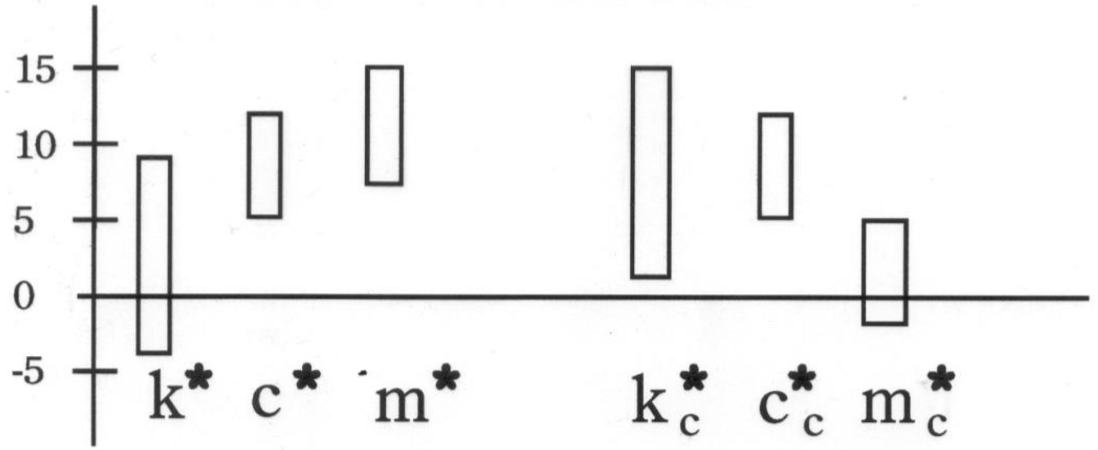
$$c^* = \frac{1}{\pi r_2^2 B_2 \rho \omega} c$$

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

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.)

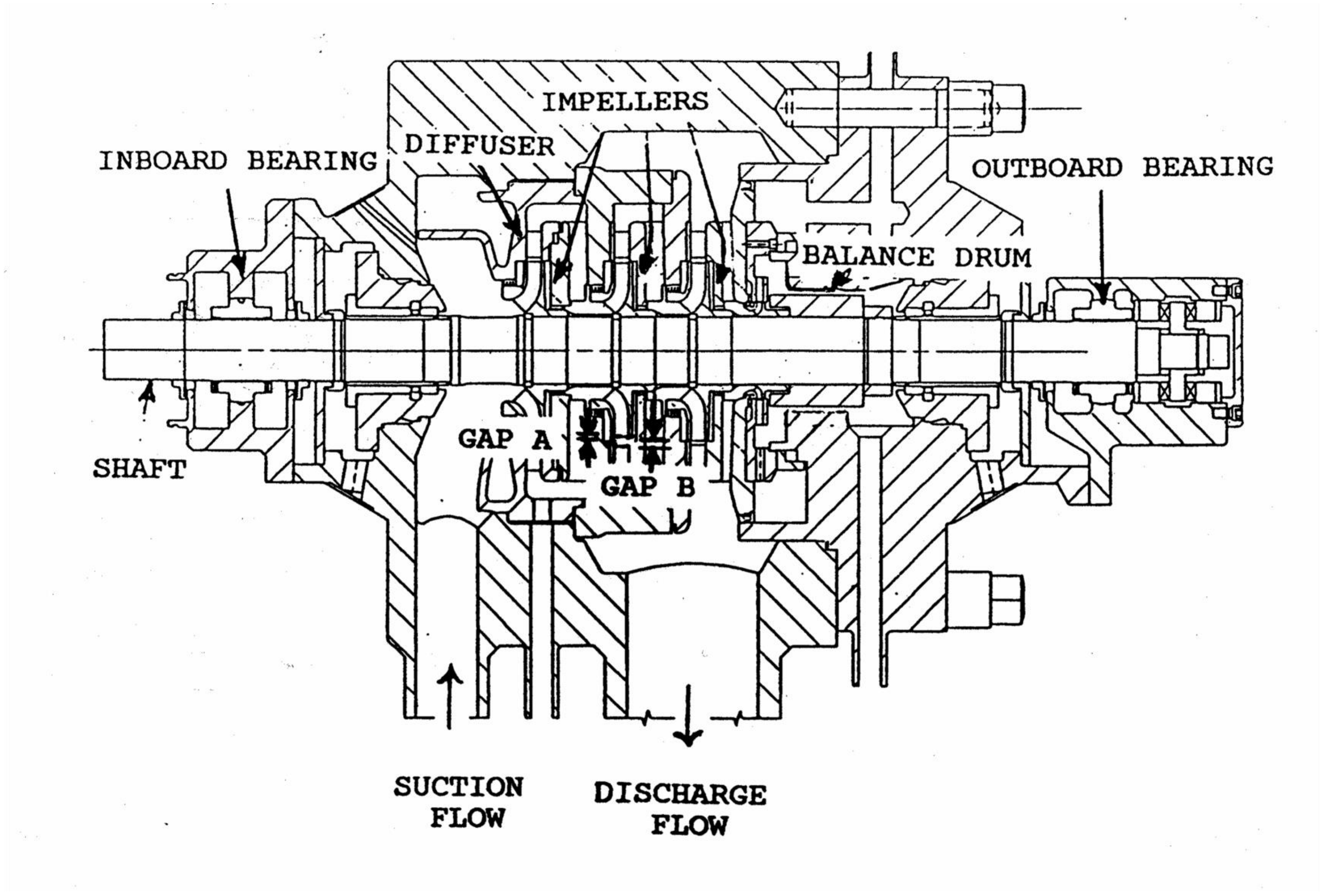
$$F_r^* = -k^* - c_c^*\left(\frac{\Omega}{\omega}\right) + m^*\left(\frac{\Omega}{\omega}\right)^2$$

Tangential (in dir. of rot.)

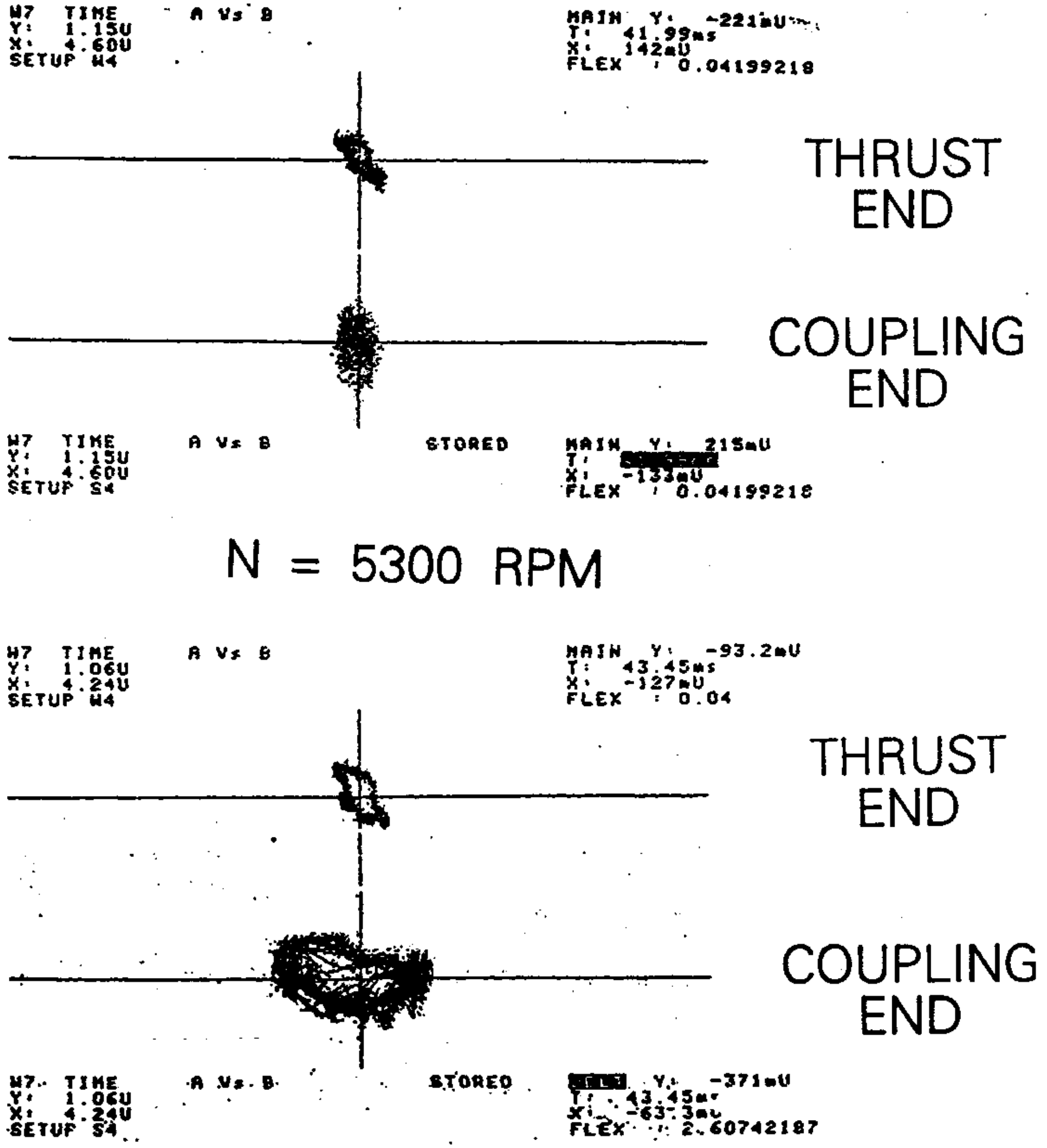
$$F_t^* = -k_c^* + c^*\left(\frac{\Omega}{\omega}\right) + m_c^*\left(\frac{\Omega}{\omega}\right)^2$$

Multistage Pump
Inboard Bearing Chronic Failure
Case History

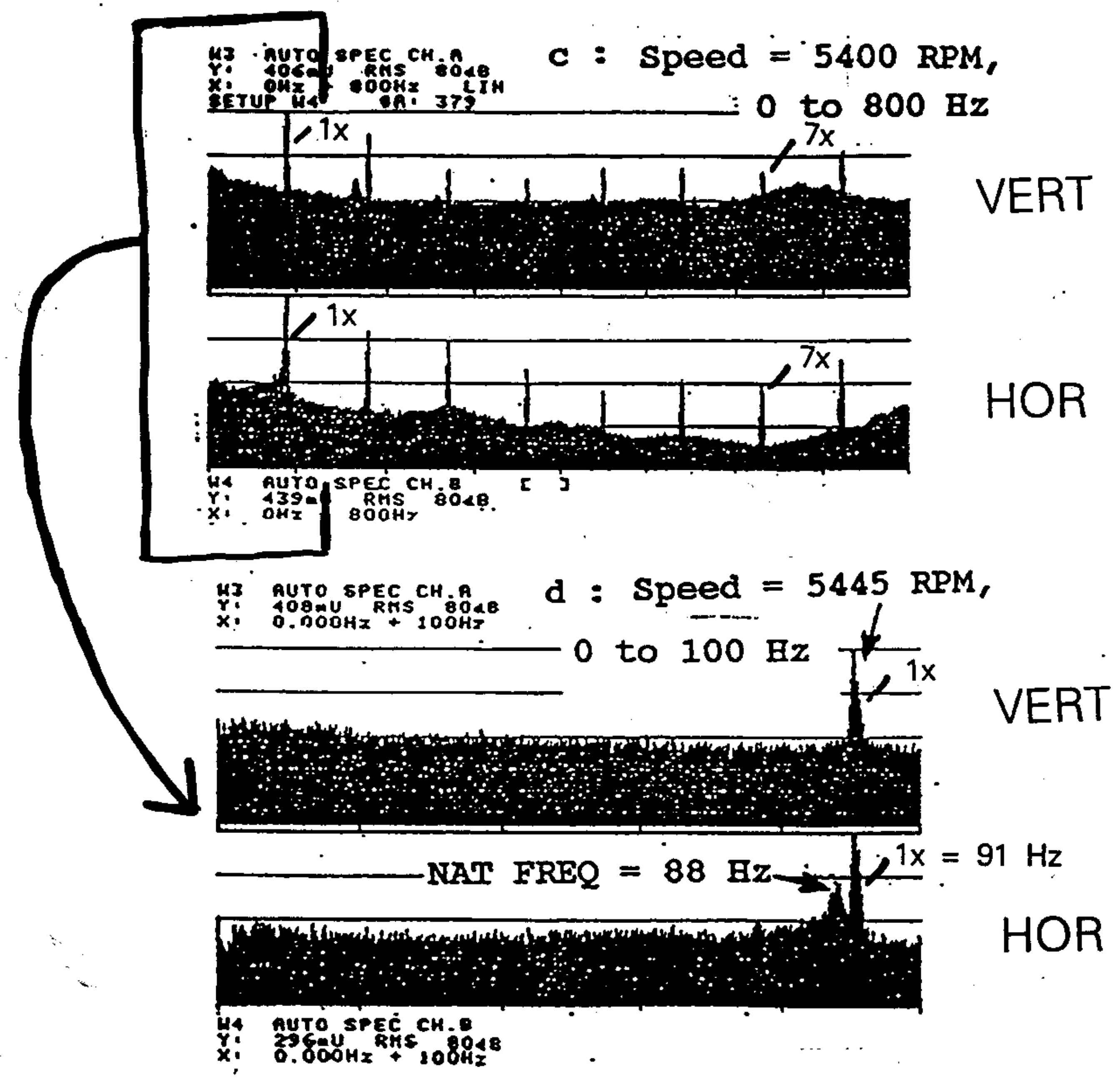
Pump with Inboard Bearing Failures



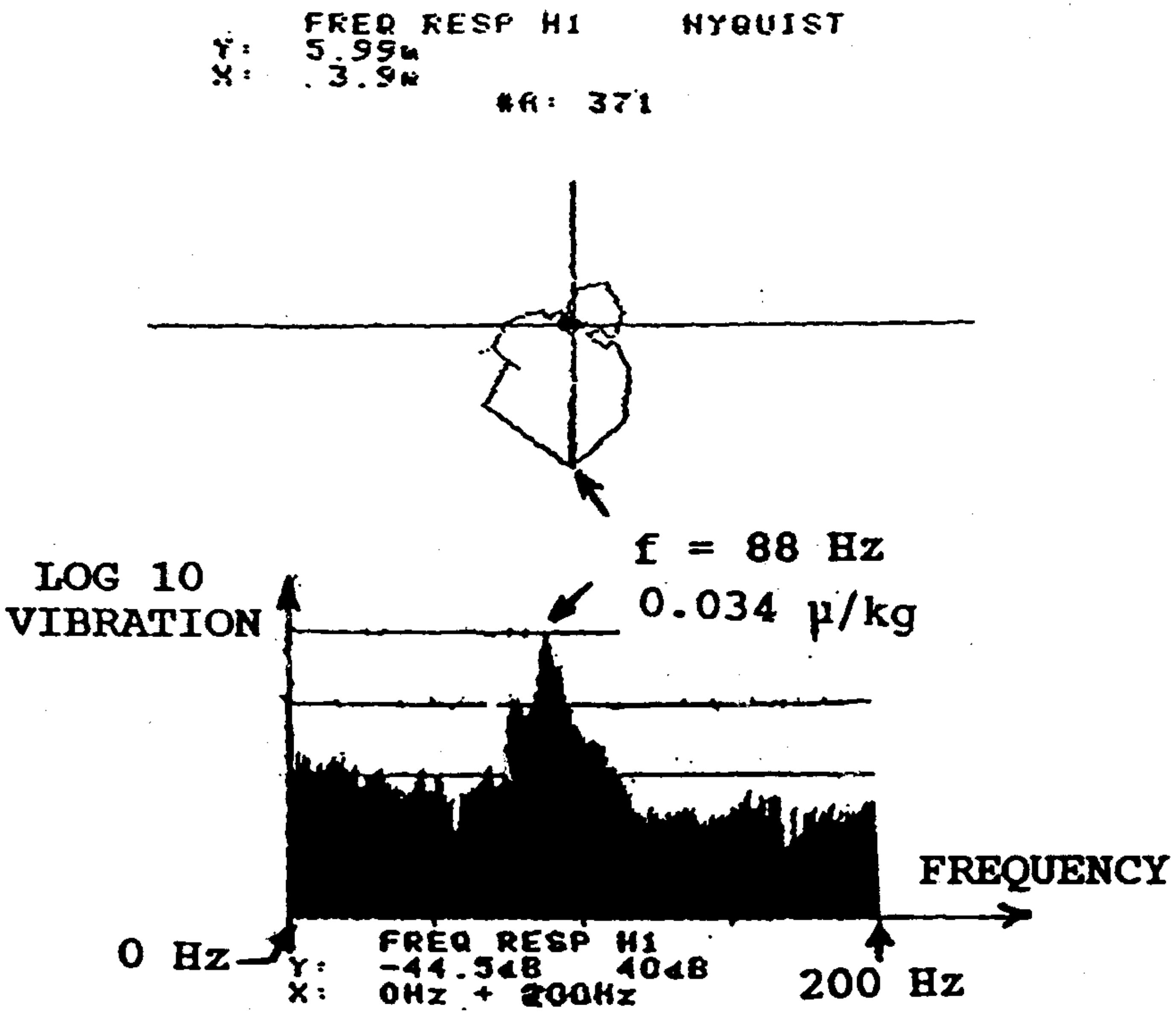
Shaft Orbits of Problem Pump



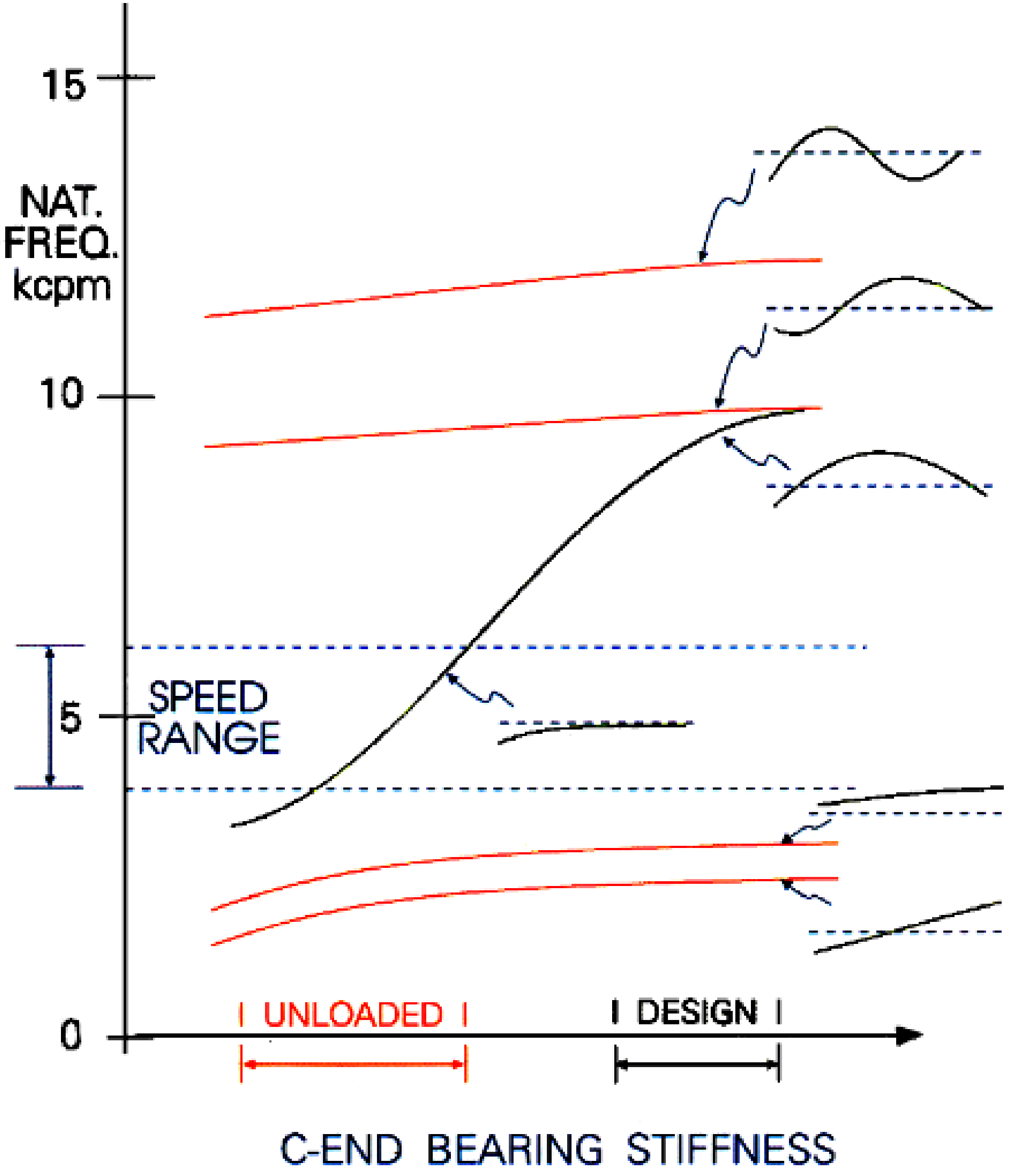
Vibration Spectra for Problem Pump



Impact Test Results Showing Shaft Critical Speed at 5280 rpm

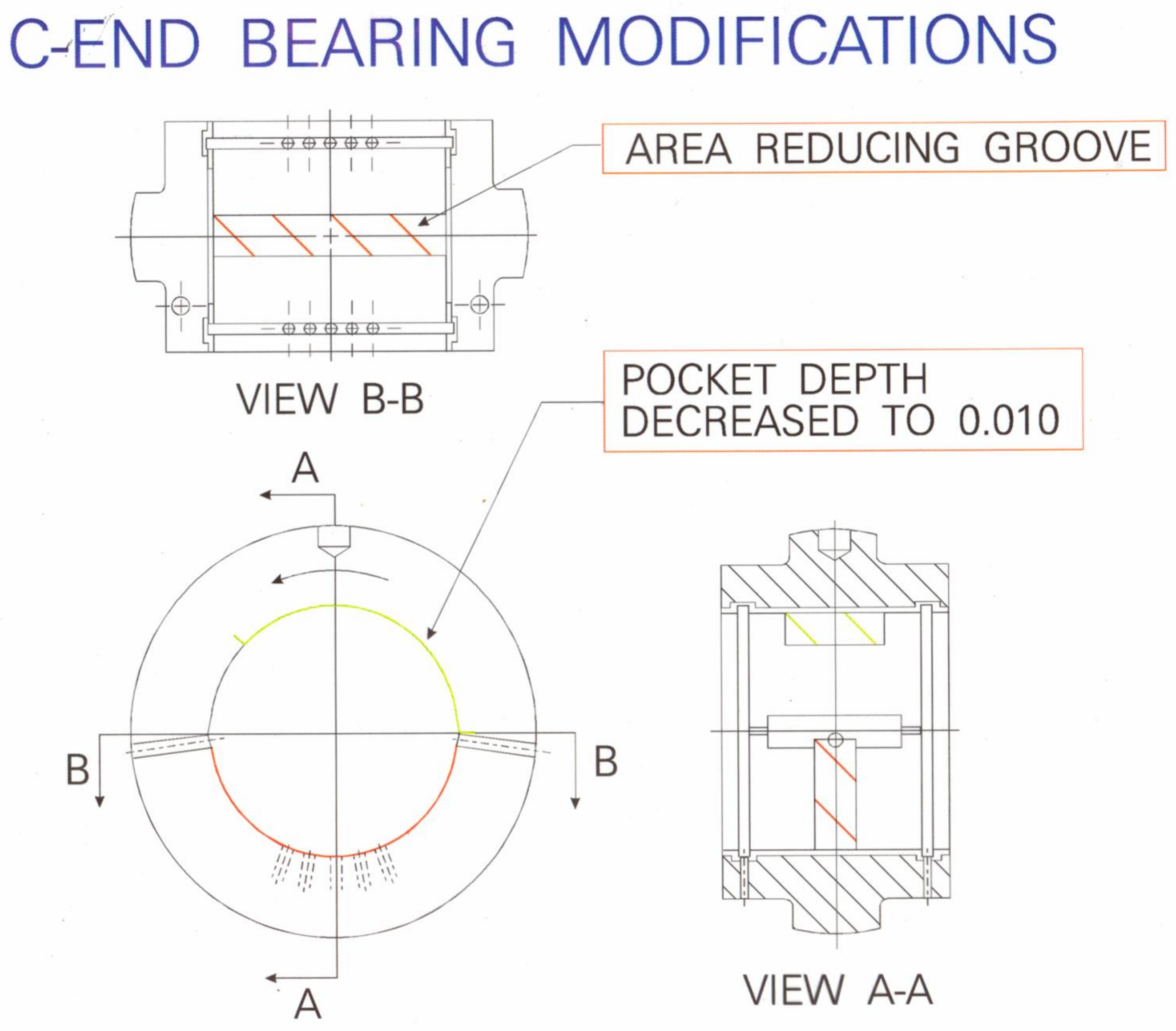


“What If” Analysis for Problem Pump

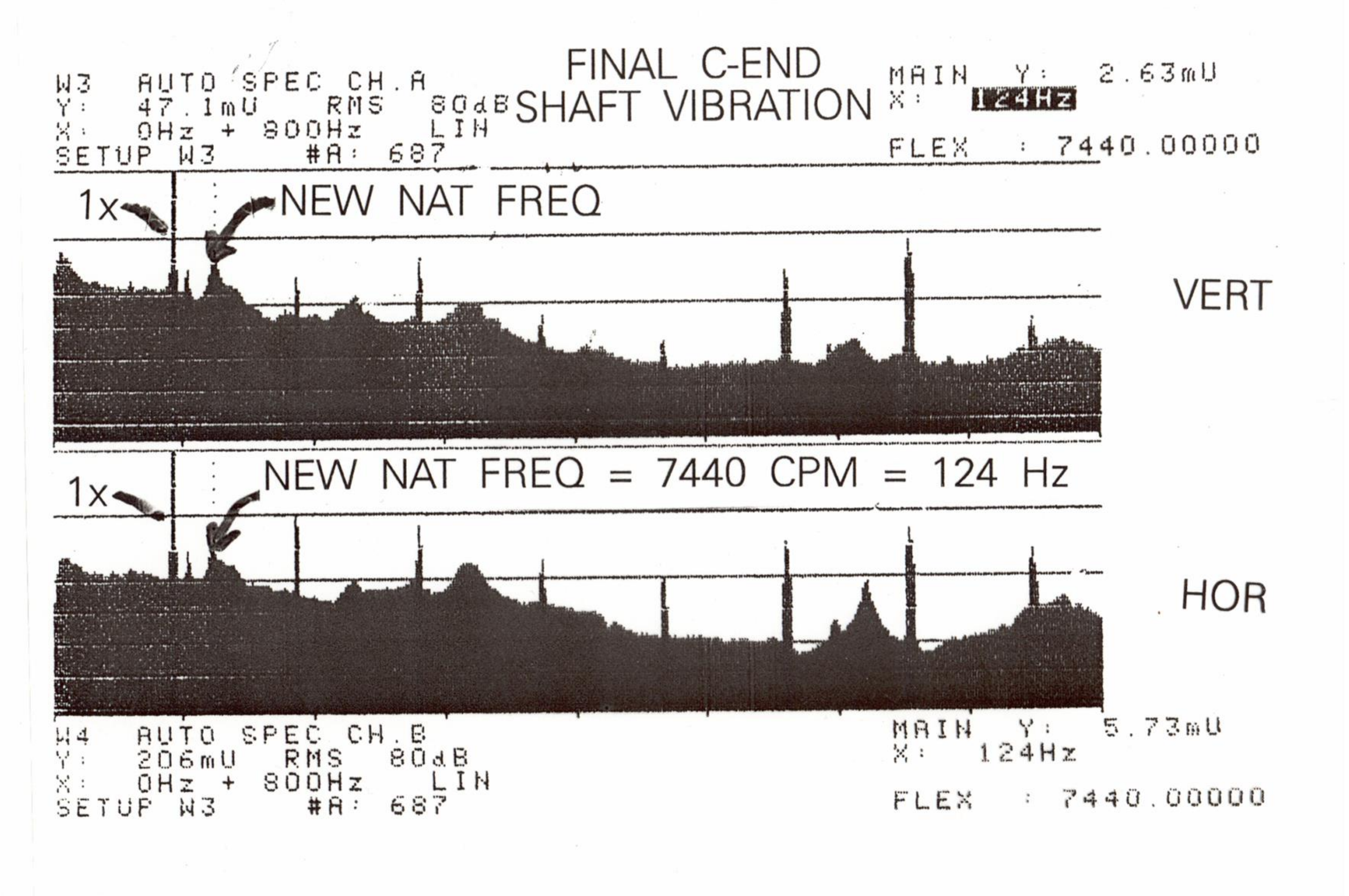


Bearing Groove Change from 0.040 in. Deep to 0.010 in. Deep

Vibration decreased a factor of ten!

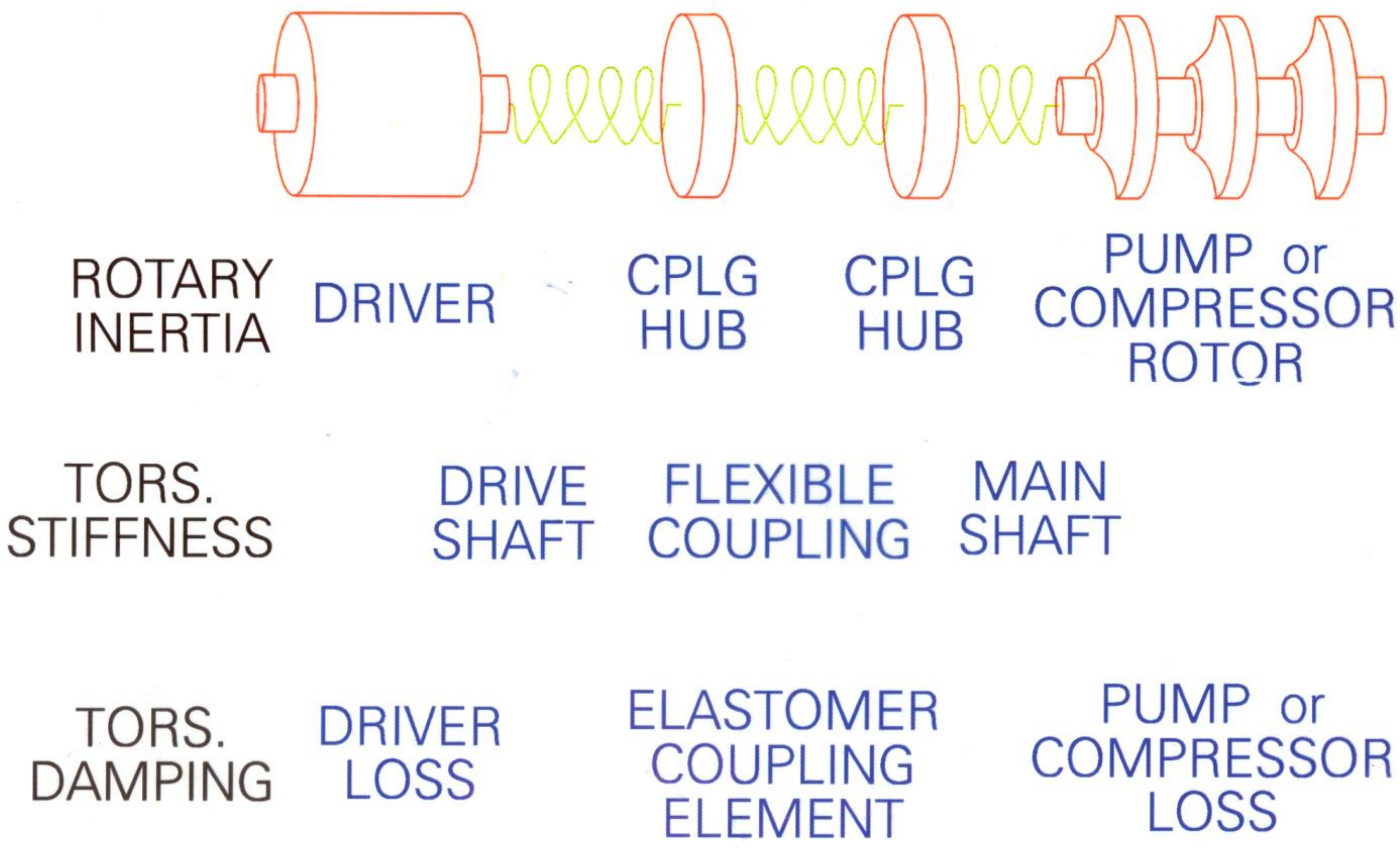


Vibration Spectrum After Bearing Fix

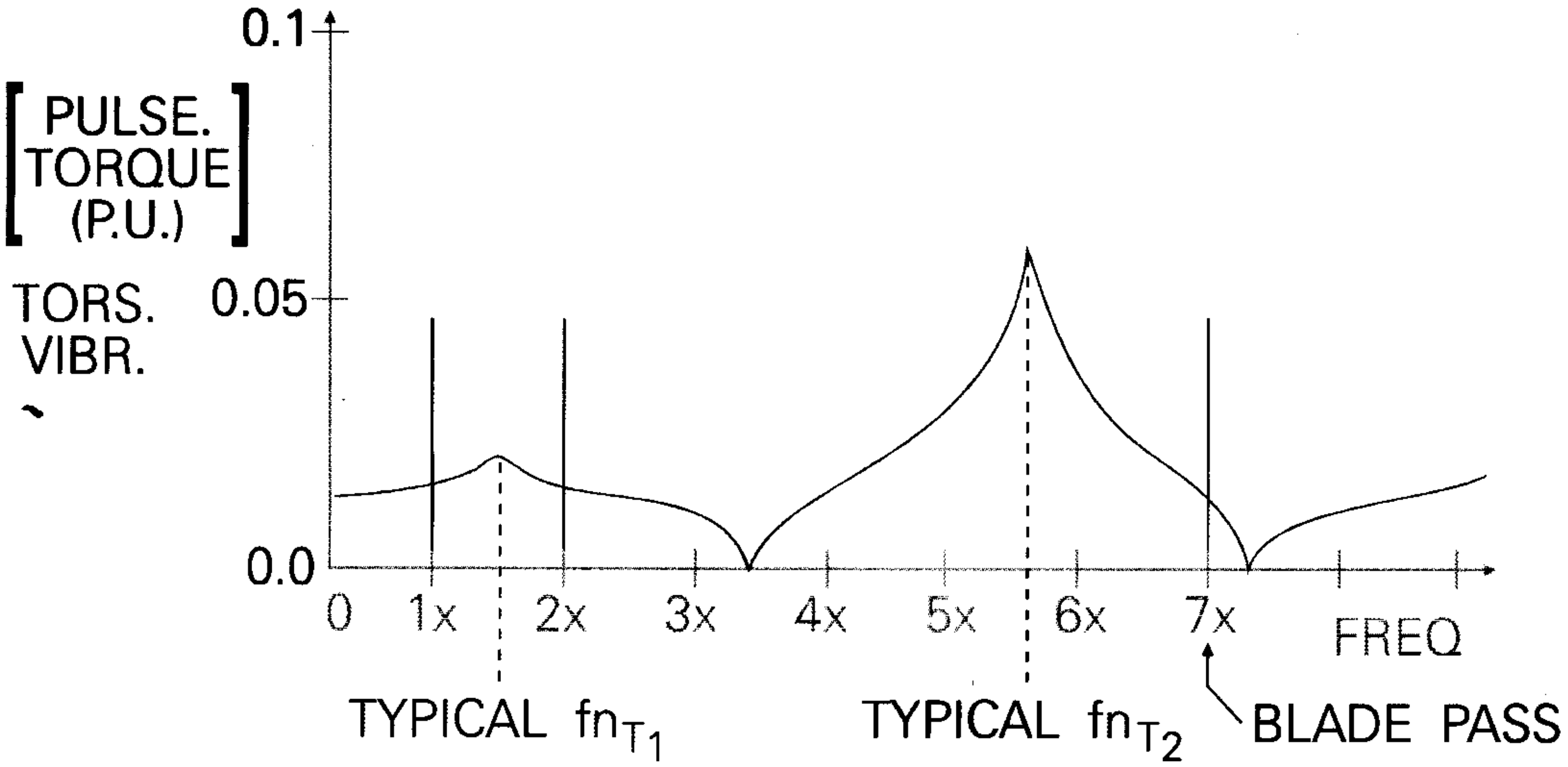


Torsional Vibration Model

Note: Fluid "Added Mass" Is nearly zero for practical pump impellers



Typical Torsional Critical Speeds and p-p Torque Pulsation Levels



- 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}

Summary of API 610
Rotordynamic Requirements

Rotor must be analyzed if:

- A similar pump is not already operating successfully
- The rotor is not "classically stiff", i.e. its 1st critical speed is not 20%+ above 1x N

API-610 & 684 Analysis Guidelines

- Do not include stiffness of either loose fit or press fit impellers, sleeves, and hubs
- Do include the full mass of all impellers, sleeves, and hubs, located at their center-of-mass
- Include fluid added mass
- Include Lomakin Effect
- Base Lomakin Effect on 1x and 2x API clearances: Meet separation margins for both extremes
- Account for bearing support stiffness & mass
- Driver not included for lateral critical speeds, but MUST be for torsionals.

API Required Rotordynamic Results

- First three lateral critical speeds
- All critical speeds up to at least 2.2x running speed
- Amplitude v. frequency & phase angle “Bode” plots (forced response plots)
- Campbell diagrams of critical speeds v. N
- No quantitative stability assessment for pumps

Separation Margin Guidelines per API-610 11th Edition

- Forcing frequency v. natural frequency “separation margin” needed depends on degree of damping.
- For damping ratio of 0.15 or higher no separation margin is required
- Fig. I.1 shows separation (up to 26%) required for lesser amounts of damping

Conclusions

- Rotordynamics is complex
- Keys to success:
    - ✓ Knowledge
    - ✓ Experience
    - ✓ The Right Tools
- API-610 and API-684, as well as HI and ISO Specs, provide a good evaluation guide
- Check out the new HI 9.6.8 pump dynamics evaluation guide!