ANALYSIS GUIDE FOR
VARIABLE FREQUENCY DRIVE OPERATED CENTRIFUGAL PUMPS

by

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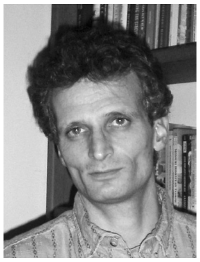

Thomas F. Kaiser has been an Engineering Analyst with Sulzer Pumps (US) Inc., in Portland, Oregon since 1996, leading the analysis team since 2001. In this position he is responsible for performing stress, finite element, rotordynamic, and vibration analyses for new and installed centrifugal pump applications, calculation tool development, and seismic qualification analyses. He began his career in 1994 with Sulzer Pumps Headquarters in Winterthur, Switzerland, as a development engineer working in the field of rotordynamics.

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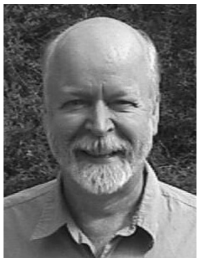

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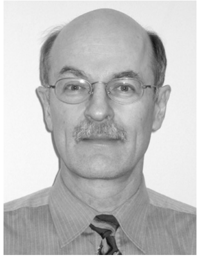

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ABSTRACT

The use of variable frequency drives (VFD) in pumping applications with variable-duty requirements provides the user with a variety of benefits, including potentially significant energy savings and improved reliability achieved by means of speed reduction and avoiding part-flow operation. Energy savings are primarily realized by running the equipment at high levels of efficiency and optimal operating speeds, matching the generated pump head to the exact system requirements without the use of energy consuming control valves. Running pumps at lower operating speeds and avoiding part-flow operation also positively influences component life and between maintenance intervals. The primary mechanical challenge of any VFD application is the wide continuous operating speed range. Excitation frequencies of fixed speed applications miss most natural frequencies of the structure, rotor, etc., and therefore potentially harmful resonance conditions often do not occur. This is no longer the case with VFD applications, where excitation frequencies become variable and the likelihood of encountering resonance conditions is greatly increased. Problems

and failures in pumps and associated systems that are not caused by resonance are generally not VFD related and are therefore not discussed.

The paper gives an overview of medium-voltage VFD technology as well as the main categories of resonance conditions of concern with regard to mechanical vibrations of pump/motor sets. The analytical and experimental identification of resonances related to lateral rotor, torsional rotor, structural, and acoustic dynamics are discussed in detail. The applicable set of analyses and, if necessary, the corresponding appropriate corrective measures, are designed to help ensure operation free of harmful resonance conditions and problems caused by excessive mechanical vibrations.

INTRODUCTION

Continuous operation under resonance conditions may result in excessive equipment vibrations, reduced in-between maintenance intervals, and premature equipment failure. Resonance conditions with centrifugal pump applications can be divided into the four categories of *lateral rotordynamics, torsional rotordynamics, structural dynamics,* and *acoustic resonance*. Each of these categories requires its own specific set of analyses and checks allowing up-front identification of resonance conditions and corresponding corrective action. Resonances may then be avoided, moved (to operating points where the resulting mechanical vibrations are acceptable), or be detuned altogether. These analyses can also help eliminate the need for expensive factory string tests aimed at investigating vibration performance.

A basic understanding of the most commonly applied and available variable frequency drive (VFD) technology and its rapid development over the last few decades is helpful in the assessment of VFD-related vibration problems and the selection of the optimal set of analyses and checks.

The case studies presented in this paper give a detailed illustration of the analysis procedures and methods that can be employed in order to successfully identify resonance conditions of concern with VFD applications. The same methods and tools can also be used to study the effect of design modifications aimed at detuning resonances. The actual analysis work should be carried out by individuals specifically trained for the task. On a broader level, the paper indicates the type of analyses and checks considered necessary as well as standard analyses that may be omitted. The information presented may therefore be used as an end-user guideline for selecting/purchasing of analysis support from the original equipment manufacturer (OEM) or engineering consultants.

The rotordynamic software tools used for the case studies are pump OEM in-house developments. Commercially available rotordynamic software may be used instead. All structural analyses were performed applying a general purpose finite element analysis software.

EXCITATION SOURCES AND AMPLIFIERS

This section describes the relevant excitation mechanisms and amplifiers of mechanical vibrations.

Mechanical and Hydraulic Unbalance

Mechanical unbalance occurs when the mass centerline of a rotating component does not coincide with the shaft centerline. A certain level of dissymmetry of the weight distribution is unavoidable in rotating equipment. For the case of two-plane balancing, the unbalance measured in US customary units is defined in Equation (1). The factor K is a balance constant, W is the mass per balance plane (or journal), and N is the rotor speed. The SI unit equivalent definition is shown in Equation (2) with the ISO Balance Quality Grade G, rotor-mass m, and the angular speed of rotation ω. The unbalance force rotates with rotor speed (1×) and is therefore a sinusoidal function of time when viewed in a stationary (nonrotating) coordinate system. The corresponding definition of the unbalance force F in US units and SI units is given in Equations (3) and (4), respectively.

$$U = K * \frac{W}{N} \quad \textbf{(1)}$$

$$U = G * 10^{-3} * \frac{m}{\omega / 60} = G * 10^{-3} * \frac{m}{2\pi * N / 60} \quad \textbf{(2)}$$

$$F = U * 0.000162 * \omega^2 \sin(\omega t + \varphi) \quad \textbf{(3)}$$

$$F = U \omega^2 \sin(\omega t + \varphi) \quad \textbf{(4)}$$

Geometric deviations between the individual impeller channels create a nonuniform pressure distribution at the impeller outlet, which also rotates with rotor speed (1×). The resulting radial hydraulic force has much the same effect as mechanical unbalance and is therefore referred to as hydraulic unbalance. Hydraulic unbalance increases with increasing flow-rate and usually exceeds mechanical unbalance by factors. Unbalance affects lateral rotor and structural vibrations but not torsional rotor vibrations.

Self-Excited Vibration

Self-excited vibration, also known as rotor instability, is most commonly associated with radial journal bearings, annular seals, and hydraulic impeller-casing interaction. Self-excited vibration caused by lightly loaded cylindrical journal bearings/guide bearings in vertical pump application are the most common cause of instability in centrifugal pumps. The corresponding vibration frequency typically lies between 0.40 and 0.50 times running speed (subsynchronous vibration), indicating a tangential mean fluid velocity c_u inside the tight bearing clearance per Equation (5). The parameter R denotes the rotor radius at bearing location and ω is the angular shaft speed.

$$c_u = 0.40 \ldots 0.50 * \omega * R \quad \textbf{(5)}$$

Pumps with excessively worn annular seals can show the same phenomenon with vibration frequencies in the 0.7 to 0.9 times running speed range (also above 1× running speed in case of tangential fluid entry velocities > ωR).

Instabilities are caused by the nonsymmetrical pressure distribution of the vibrating shaft, which creates a force component acting in the direction of the shaft orbit. This force feeds energy to the rotor and thus the shaft orbital movement is accelerated. Instability occurs in case the energy put into the rotor exceeds the direct damping opposing the same vibration.

Many vertical pump applications show a vibration component at or near 0.5× running speed in their amplitude spectrum (also referred to as oil whirl or bearing whirl). Instability usually only occurs in case a structural or lateral rotor natural frequency is at or near this 0.5× running speed frequency, changing the oil whirl into an oil whip condition with potentially destructive vibration levels. In case the operating speed is increased after the onset of instability, the vibration frequency will typically remain nearly constant, locked into the natural frequency of the structure or rotor as indicated in Figure 1.

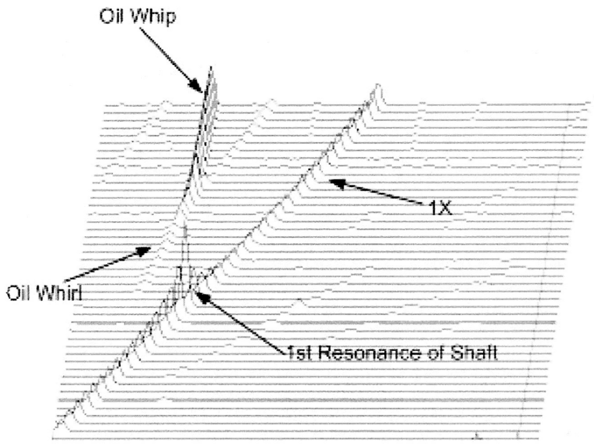

Figure 1. Waterfall Vibration Plot.

Measures against self-excited vibration include:

- Reducing the mean tangential velocities in journal bearing and annular seal tight clearances. This may be achieved by means of applying rough stator surface finish, swirl breaks at annular seal entry, stator-side honeycomb or hole patterns, smooth rotor surface finish, etc.
- Increasing bearing radial load by means of applying additional (intentional) misalignment in vertical pumps.
- Avoiding critical speed situations between 0.5× running speed excitation and lateral or structural natural frequencies.
- Restoring design clearances in case of excessively worn annular seals.

Self-excited vibrations affect lateral rotordynamics and structural vibrations, not torsional rotordynamics.

Vane-Pass Pressure Pulsations

Vane-pass pressure pulsations are generated by the impingement of the nonuniform impeller wake flow on the volute cutwater or diffuser vane tips. Figure 2 depicts the nonuniform fluid velocity profile at the impeller outlet. These pressure pulsations travel through the system at the speed of sound of the pumpage. The frequency of the vane-pass pressure pulsations, acting on the stator, is proportional to the pump rotational speed N, the impeller vane count z_2, and multiples (n) thereof as illustrated in Equation (6).

$$f_{VANE-PASS} = n * z_2 * (N / 60) \quad \textbf{(6)}$$

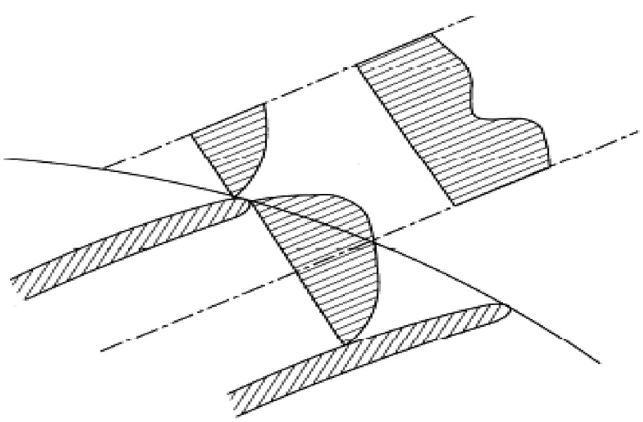

Figure 2. Wake Flow at Impeller Outlet.

The main factors influencing the pressure pulsation magnitude are the radial gap between the impeller outer diameter and the volute/diffuser cutwater (B-gap), the percent of best efficient point (BEP) operation, and the impeller outlet velocity u_2.

B-Gap

The radial distance between the impeller vane trailing edge and the volute cutwater or diffuser vane leading edge (B-gap) heavily influences the pressure pulsation amplitudes. According to investigations published in (Guelich and Bolleter, 1992), pressure pulsation amplitudes decrease on average with a power of (−0.77) on the relative radial gap as illustrated in Equation (7). For example, a 2 percent B-gap will produce pressure pulsation amplitudes three times higher than a 9 percent B-gap on an otherwise identical pump.

$$\Delta P* \approx \left(\frac{D_3}{D_2} - 1\right)^{-0.77} \quad \textbf{(7)}$$

Percent BEP Operation

In Figure 3, statistical data from 36 measurements of single and multistage pumps are plotted as dimensionless root mean squared (rms) values for the frequency range of 1.25 to 20 times running speed, which covers vane-pass frequency. Pressure pulsations are normalized according to Equations (8) and (9) for US units and SI units, respectively. ΔP_{RMS} is the rms value of the dimensional pressure pulsation measurement, ρ is the fluid density, and u_2 is the fluid velocity at impeller outlet. Flow is normalized as shown in Equation (10) with Q being the effective flow and Q_{BEP} representing the best-efficiency flow. The curves displayed in Figure 3 shows the strong dependency of pressure pulsations from the operating point with reference to percent of BEP operation (Guelich and Egger, 1992).

$$\Delta P*_{RMS} = \frac{9269 * \Delta P_{RMS}}{\rho * u_2^2} \quad \textbf{(8)}$$

$$\Delta P*_{RMS} = \frac{2 * \Delta P_{RMS}}{\rho * u_2^2} \quad \textbf{(9)}$$

$$q* = Q / Q_{BEP} \quad \textbf{(10)}$$

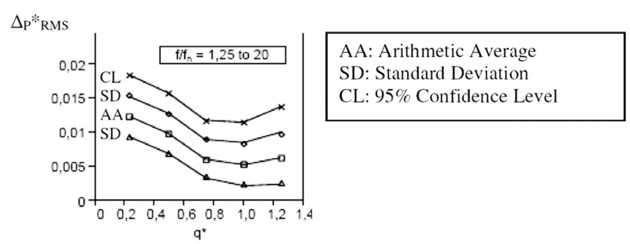

Figure 3. Pressure Pulsations Versus Percent of BEP Operation.

Impeller Outlet Velocity

Experience indicates that pressure pulsations in geometrically similar pumps roughly increase with the square of the circumferential speed as shown in Equations (11) and (12) for US units and SI units, respectively:

$$\Delta P_d = 0.75 * 0.000162 * \frac{\rho}{2} u_2^2 \quad \textbf{(11)}$$

$$\Delta P_d = 0.75 * \frac{\rho}{2} u_2^2 \quad \textbf{(12)}$$

The stagnation pressure ΔP_d is a measure of the unsteady hydrodynamic load acting on the volute or diffuser vane.

Other Influencing Parameters

Parameters affecting the nonuniformity of the impeller wake flow and geometric parameters have an influence on the generation and amplitude of pressure pulsations. Among these parameters are:

- Thickness and form of the impeller vane trailing edge.
- Number of impeller vanes and combination of impeller vanes to volute vanes/diffuser vanes (e.g., odd versus even number of impeller vanes in double volute type pumps).
- Impeller specific speed. Pressure pulsations generally increase with increasing specific speed.
- Impeller geometry, particularly vane exit angle and shape of volute cutwater (blunt versus hydraulically smooth).
- Staggering of impellers along shaft of multistage pumps and staggering of the two halves of double suction impellers.

Vane pass pressure pulsations primarily affect structural vibrations (e.g., bearing housing vibrations). A certain pressure pulsation level will always be present in centrifugal pumps and does not need to represent a problem. Excessive pressure pulsations can result in high pump and piping vibrations, particularly in combination with structural resonance or acoustic resonance. This may result in vibration levels beyond alarm or shutdown and cause fatigue failures in auxiliary piping, instrumentation, etc.

Other Excitation Sources

Other excitation sources include broadband hydraulic forces due to recirculation, cavitation, and forces due to rotating stall.

Amplifiers

Vibration levels usually become excessive when amplified by resonance. A resonance condition occurs when an excitation frequency is within a few percent of a relevant natural frequency. In that condition, the excitation force is acting again once the vibrating component has come full cycle after the last "impact" by the force. The excitation force and the vibration are synchronized, and the vibration amplitude increases until limited by nonlinear effects. With regard to mechanical vibrations in VFD operated centrifugal pumps, resonance conditions can be divided into four categories: structural resonance and torsional rotor resonance are typically lowly damped and are likely to result in high levels of mechanical vibration when properly excited. Lateral rotor resonances are in some cases highly damped and operation on or near such a condition ("critical speed") may be perfectly acceptable. Acoustic resonance conditions, amplifying mechanical vibrations via amplified pressure pulsations, are usually only lowly damped. The various resonance categories are discussed in detail in the following sections.

LATERAL ROTORDYNAMICS

General

The damped lateral rotordynamic behavior of a centrifugal pump rotor is determined by the rotor geometry, the rotor mass and inertia, and the interaction forces occurring between the rotor and journal bearings, annular seals, and casing. Impeller wear rings, close-clearance bushings, and balance pistons are typical examples of annular seals. Casing interaction occurs at impeller location, between the wear rings in case of a closed impeller design, and is generally destabilizing. These interaction forces are nonlinear but may be linearized around a particular static rotor equilibrium position. Interaction forces vary with operating speed, pumpage specific gravity and viscosity, load, state of wear, etc. Solving the linearized homogeneous Equation of Motion (13) results in a set of eigenvalues.

$$\underline{\underline{K}}\,\underline{x}+\underline{\underline{D}}\,\underline{\dot{x}}+\underline{\underline{M}}\,\underline{\ddot{x}}=\underline{0} \quad \textbf{(13)}$$

The linearized equation of motion consists of a stiffness-matrix K, damping-matrix D, mass-matrix M, and vectors of displacements (x), velocity ($\dot{x}$), and acceleration ($\ddot{x}$). The complex eigenvalue λ_X is defined in Equation (14). The imaginary component ω_X represents the angular natural frequency. The eigenvalue determines the corresponding natural frequency f (Equation [15]), modal damping value D (Equation [16]), and mode shape as shown in Figure 4. A mode with negative damping D represents an instable system. The rotor system is laterally stable when all significant modes provide positive modal damping levels.

$$\lambda_x = \alpha_x + i*\omega_x \quad \textbf{(14)}$$

$$f = 2*\pi*\omega_x \quad \textbf{(15)}$$

$$D = -\frac{\alpha_x}{\sqrt{\alpha_x^2+\omega_x^2}} \quad \textbf{(16)}$$

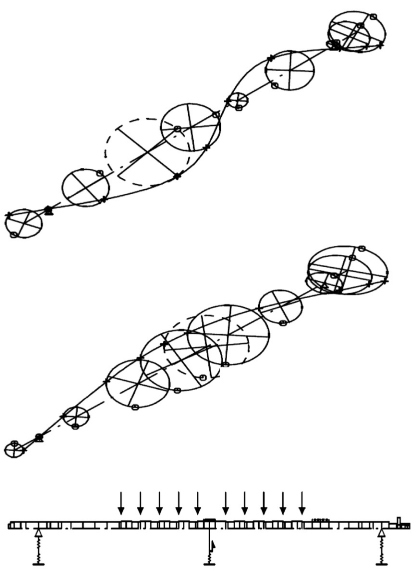

Figure 4. Lateral Mode Shapes and Mechanical Model.

The evaluation of the lateral rotordynamic behavior can either be done by solving the homogeneous equation of motion (eigenvalue calculation) or by specifying a set of excitation forces and subsequent solution of the nonhomogeneous equation of motion (forced response analysis).

The eigenvalue approach and evaluation of results applying a combined frequency-versus-damping-ratio criterion is further discussed in this paper. This approach is less ambiguous compared to a forced response analysis because it avoids the subjective process of determining and applying excitation forces (typically a combination of mechanical and hydraulic unbalance loads).

The results of a damped lateral rotordynamic analysis are best presented in the form of a Campbell diagram as illustrated in Figure 5, plotting the natural frequencies and modal damping factors versus pump operating speed. The intersection between a speed-dependent natural frequency line and the synchronous speed excitation line is called a critical speed and represents a resonance condition.

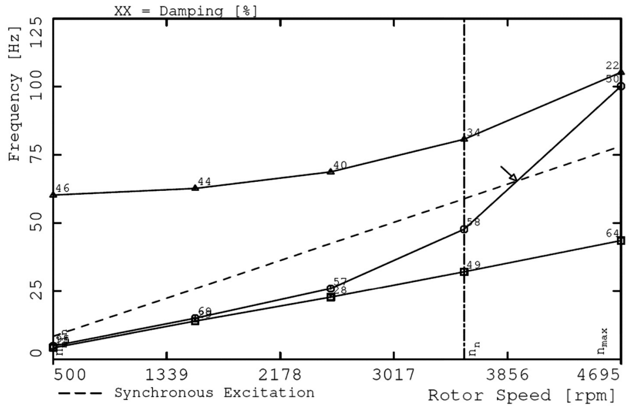

Figure 5. Campbell Diagram.

A widely used eigenvalue acceptance criterion is defined in Annex I of the API 610 standard (Eighth Edition, 1995; Ninth Edition, 2003; Tenth Edition, 2004) and the ISO 13709 standard (2003), respectively. The combined frequency-versus-damping criterion, depicted in Figure 6, is applied to each of the calculated lateral modes, limiting the evaluation to modes within a natural frequency range of zero to 2.2 times running speed. This frequency range covers the most typical and significant rotor lateral excitation forces including subsynchronous excitation, mechanical and hydraulic unbalance and misalignment:

- Subsynchronous excitation (journal bearings): 0.4… 0.5× running speed (typical)
- Subsynchronous excitation (annular seals): 0.7… 0.9× running speed (typical)
- Rubbing: multiples of 0.5×
- Mechanical unbalance: 1× running speed
- Hydraulic unbalance: 1× running speed
- Misalignment: 1× and 2× running speed

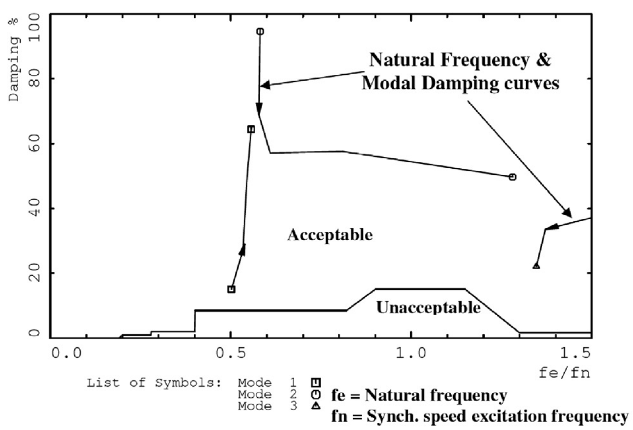

Figure 6. API 610 Lateral Rotordynamic Acceptance Criterion.

The API 610 (2004) lateral evaluation criterion also requires consideration of operating speeds outside of the defined pump continuous operating speed range. A speed range of 25 percent of minimum continuous speed to 125 percent of maximum continuous operating speed needs to be investigated.

While, for cases in violation, the cited API standards allow proof of acceptability by means performing additional unbalance forced response analysis, this approach is not recommended for obvious reasons. An unbalance forced response analysis applies excitation sources at synchronous speed (1×), which cannot excite subsynchronous modes. It is therefore recommended to consider the modal-damping-versus-frequency-separation acceptance criterion illustrated in Figure 6 as binding. With a few exceptions not discussed in this paper, rotor designs in violation should be modified to meet this acceptance criterion. Design modifications aimed at improving lateral rotordynamic stability can be divided in two categories. A first category aims at increasing the frequency separation margin between lateral modes and synchronous excitation speed by means of increasing the rotor stiffness (K_{xx}) or reducing the rotor mass (M). A second category of modifications intends to increase modal damping.

Design modifications aimed at increasing lateral rotor natural frequencies:

- Decreasing coupling overhung length and/or coupling weight in case of overhung dominated modes (K_{xx}, M)
- Changing of impeller material from steel to aluminum (M)
- Increasing shaft size (K_{xx})
- Decreasing between-bearing span (K_{xx})
- Tightening or restoring annular seal clearances (K_{xx})
- Eliminating stator-side serrations applied to reduce leakage (K_{xx})
- Applying stator-side circumferential grooves in balance pistons and center and throttle bushings. This reduces the adverse effect of piston tilting onto the direct radial annular seal stiffness (K_{xx}).
- Changing from inline to back-to-back configuration, which reduces bearing spans and also adds damping at the center of the pump (K_{xx})

Design modifications aimed at increasing modal damping:

- Applying stator-side swirl breaks at the entrance of impeller eye wear rings. This reduces the circumferential inlet swirl, which in turn reduces destabilizing annular seal cross-coupled stiffness.
- Optimizing the journal bearing design. Journal bearings with length-over-diameter ratios above one should be avoided. The destabilizing effect of cross-coupled journal bearing stiffness can be reduced or eliminated by switching from cylindrical bearings to multilobe or tilting-pad designs.
- Loading of vertical pump line-shaft bearings by means of applying intentional misalignment between bearings and rotor
- Applying rough annular seal stator surface finish, stator-side honeycomb or hole patterns, smooth rotor surface finish

In case a fixed speed centrifugal pump is converted to VFD operation, the *continuous VFD operating-speed range may already be sufficiently analyzed and covered by the original fixed-speed lateral rotordynamic analysis*. Fixed-speed lateral analyses should be carefully reviewed on a case-by-case basis before deciding whether a new lateral analysis for the VFD operated application is necessary or not.

Case Study—Standard Lateral Rotordynamic Analysis Procedure

The dynamic lateral stiffness and damping levels provided by the journal bearing fluid film depend on the stiffness of the bearing housing/support structure itself. With most horizontal pumps, the lowest bearing housing natural frequency is well above the first few lateral rotor bending mode natural frequencies. In these cases, the corresponding *journal bearing support stiffness can be considered as near-rigid and constant over the entire speed range* of concern. The situation is entirely different with most vertical pump applications. Vertical pumps are typically structurally flexible and significant structural modes may appear at, near, or below operating speed. This requires calculation of the dynamic bearing support stiffness by means of performing harmonic response

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