VANE PASS VIBRATION—SOURCE, ASSESSMENT AND CORRECTION—A PRACTICAL GUIDE FOR CENTRIFUGAL PUMPS

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ABSTRACT

With the ever-increasing use of hand held vibration analyzers, data collectors, and predictive maintenance programs, a more complete understanding of vibrations is ever more important. This tutorial treats one of the most prevalent pump problems, vane pass pressure pulsations and vibrations. At this time, we are still not able to predict with great accuracy the amplitudes of vane pass pressure pulsations nor the mechanism that transforms them to vibration. However, with a mixture of practical experience and theory, we can assess the vibration and, when required, determine the best corrective action.

Presented are causes, methods of assessment with suggested acceptance criteria, and finally, proper corrective procedures and preventive measures. First, a good understanding of the cause, acceptance criteria, and corrective measures is imperative. Second, a complete set of vibration measurements taken in a systematic order with proper equipment, are needed to isolate the problem and assure all possibilities are covered. Finally, based on thorough diagnostics the root cause(s) of the vibration can then be identified and the best corrective action can be selected, whether it is by working on the source of excitation or the resultant problem vibrating part.

INTRODUCTION

There has been a noticeable increase of reported vane pass vibrations over the last couple of decades, and it is believed to be related mainly to five factors.

• The specified vibration criteria are becoming tighter, possibly overly tight in certain cases. Also, operating ranges as a percentage of the pumps best efficiency point (BEP), for which vibration criteria are good, is increasing. Although tight vibration criteria have its merits, they should be based on realistic criteria including orientation and frequency range. For lateral vibrations with frequencies greater than three times the pump operating speed, it is suggested that 0.4 in/s (10 mm/s) is acceptable.

• There has been a proliferation of vibration analyzers, specifically hand held instruments. Nearly every maintenance

person now has one. Vibration analyzers are one of our best means of measuring machine condition, so users need to be provided with a practical understanding of vibrations and realistic/objective acceptance criteria with which to judge the vibrations.

• There has been an increase in the use of variable speed drives (VSDs), and this trend should continue since VSDs should be more efficient. The problem with VSDs is that the possibility of tuning in a resonant condition with either a structural or an acoustic natural frequency increases with increasing operating range. Additionally, VSDs do not guarantee operation at the BEP. Thus, vibrations may be further aggravated.

• Standard pumping limits are gradually expanding toward higher speed, higher energy units. Typically, vibration will increase proportionately with energy content, which is defined here as the fluid density multiplied by the impeller peripheral velocity squared ($\rho\, u_2^2$).

• Pump designs are being optimized to reduce weight and manufacturing costs, thus the new lightweight designs are more susceptible to vibration in the range of vane pass frequencies. Over the last couple of decades, worldwide competition for market share has resulted in an increased effort to reduce pump manufacturing costs, and thus has resulted in reduced weight designs (less material costs). Reduced cost designs come at the expense of less robust (mass) designs.

Also, pumps that were originally built and tested before the "digital era" may not have been fully tested for vibrations, especially at frequencies above operating speed. Additionally, pumps that operated normally at standard nominal speeds (1500, 1800, 3000, and 3600 rpm) may have natural frequencies near these speeds, which can result in high vibrations if the pump is put into variable speed operation.

Within a pump process, fluid properties may vary over time and will effect a change in operating conditions, thus increase the possibility of an acoustic resonance and higher vibrations.

Sometimes specified requirements result in increased vane pass vibrations, for example, the requirement for continuous head rise to shutoff results in larger B_2's (width of impeller at outer diameter), and thus larger pulsations. Also suction specific speed requirements may increase the impeller inlet diameter, thus extending the flow range where suction recirculation will occur at higher percentages of BEP.

The general market tendency is moving toward higher speed, higher energy, lightweight equipment operating at variable speed, and thus is more susceptible to elevated vane pass vibrations. This, in combination with increased vibration awareness through the growing use of analyzers and more stringent acceptance criteria, warrants a closer look at the physical aspects of vane pass vibrations, diagnostics, and correction.

Most cases of elevated vane pass vibration are found on the bearing housing, and thus this tutorial will mainly address this problem. However, the same methods may be extended to other systems and types of vibration, e.g., shafts, auxiliary piping, etc.

SOURCE AND AMPLIFIERS

Vane pass pressure pulsations are inherent to diffuser and volute type pumps, and are produced by the impeller vane wake as it passes the diffuser or volute leading edge. Some of the energy produced by the pressure pulsations is dissipated as structural vibration, and these vibrations are quite often amplified by a resonant condition.

Vane Pass Pressure Pulsations

From tests, we know that normal vane pass pressure pulsation levels measured at the pump's outlet are around one percent, possibly as high as two percent, of the total developed head. A well designed set of impeller and diffuser/volute hydraulic passages may alleviate vane pass pressure pulsations, but will never completely eliminate them. This is due to several factors that

influence the magnitude of the pressure pulsations and the resulting structural vibration.

Vane pass pressure pulsations in centrifugal pumps are created by the wake flow from the impeller blade trailing edge and its interaction with the diffuser or volute cutwaters. Per Equation (1), the frequency of vane pass pressure fluctuations is equal to the pump rotor rotational speed, N, multiplied by the impeller vane number, z_2, and harmonics of n (1, 2, 3, etc.) thereof.

$$f_{vp} = n\, z_2\!\left(\frac{N}{60}\right) \qquad (1)$$

Normally the wake flow at the impeller outlet is the strongest source of pressure pulsations in a centrifugal pump. The pulsations are elevated at partial flow (and to a smaller extent, at flows above BEP) when the wake broadens due to flow separation and recirculation, and the resulting turbulence and vortices that are generated (Figure 1).
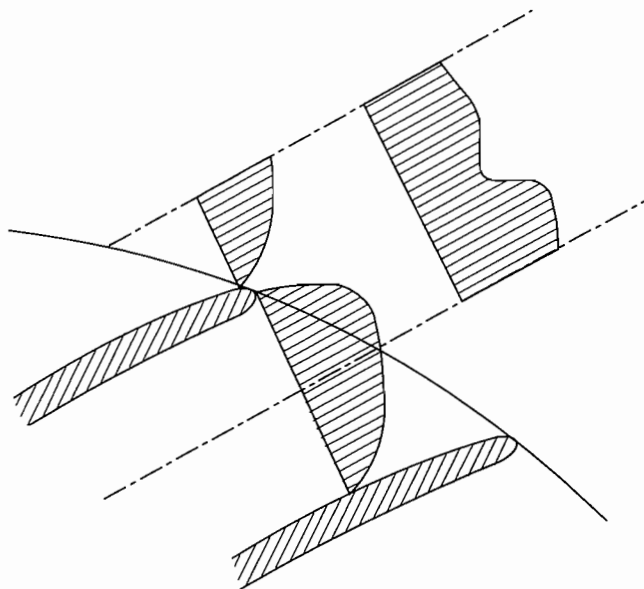

Figure 1. Velocity Profile at Impeller Outlet.

Broad band pressure pulsations and pressure pulsations at once per revolution are also produced by centrifugal pumps as a result of impeller and stator (diffuser or volute) geometry and pump operation. Large-scale turbulence within the hydraulic passages causes pressure fluctuations of a broad band nature, in particular at partial flow due to flow separation and recirculation. Pressure fluctuations at rotational frequency are generated by circumferential irregularities in the hydraulic passages of the impeller and other deviations from rotational symmetry, and result in an effect normally referred to as "hydraulic unbalance." Although these forms of pressure pulsations are not specifically treated here, their effects can be approached in much the same manner as for vane pass pressure pulsations. A typical centrifugal pump's vibration spectrum is presented in Figure 2.
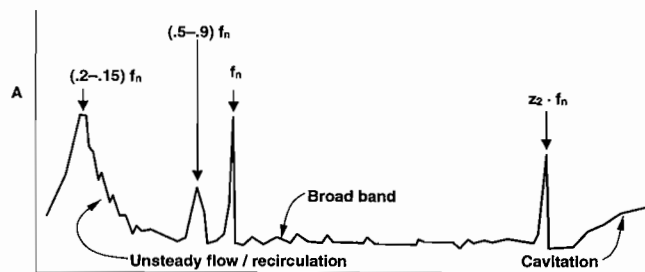

Figure 2. Standard Vibration Spectrum.

Both broad band and discrete frequency pressure pulsations are intricately related to the specific hydraulic design of the pump's inlet, impeller, and stator, and the flow distributions throughout resulting from pump operation. Because of the complex flow distribution, accurate theoretical prediction of the pressure pulsation magnitudes is not possible. Further, the system in which a pump is operated also has a strong impact on the level of pressure pulsations measured, due to the acoustical and flow characteristics of the piping. Therefore, statistical data based on numerous documented tests with different pump configurations are normally used to estimate the effects.

Generation of Vane Pass Pressure Pulsations

Influence of Speed

The fluid wake represents a deficit in the relative velocity within the rotational system. However, it represents a peak velocity in the absolute reference frame, as shown in Figure 3. Since the boundary layer velocity is near zero relative to the vane (liquid "adhering" to the surface), it nearly reaches $c_{max} = u_2 = \pi D_2 N / 60$ (where D_2 is impeller outer diameter, and N is pump speed in rpm in the absolute reference frame). The average absolute velocity at the impeller outlet is roughly $g H / (\eta_h u_2^2)$, or $0.5u_2$ for specific speed $Ns < 1800$, or in metric units $nq < 35$ (Figure 3).
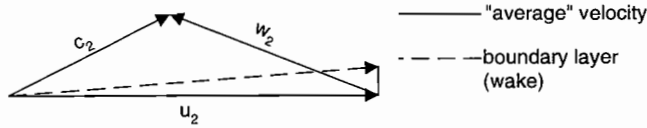

"average" velocity

– – – –boundary layer (wake)

Figure 3. Impeller Outlet Velocity Triangle.

The variation in stagnation pressure caused by the wake at the diffuser or volute is in the order of Equation (2).

$$\Delta p_d = 0.75 \, \frac{\rho}{2} u_2^2 \tag{2}$$

Δp_d is a measure for the unsteady hydrodynamic load on the diffuser vane or volute tongue (Figure 4). The static pressure downstream of the impeller, say at the discharge, also varies with time, but has a smaller amplitude in the order of Equation (3).

$$\Delta p = \rho \, \frac{u_2^2}{2} \Delta p^* \tag{3}$$

Where Δp^* is the dimensionless pressure fluctuation for the pump suction and discharge, and is based on statistical data (Figure 5). Therefore high-speed pumps have a higher potential for high vibrations, but are generally built to higher standards.

Often this scaling law is well fulfilled, but unfortunately some of the deviations discussed below can introduce errors of a factor of 10 or more, in particular cases. However, no general easily applied procedures are available to arrive at a better estimate.

Different impeller vane and stator vane combinations result in different effects. Pulsations from blade to blade may be in-phase (phase resonance) or out-of-phase or somewhere in-between, thus amplifying or compensating (reducing) amplitudes. The speed of the pump also has an impact on phase resonance, which depends on the size of the pump, the Mach number, and the combination of impeller and diffuser vane numbers. Phase resonance occurs when Equation (4) is fulfilled.

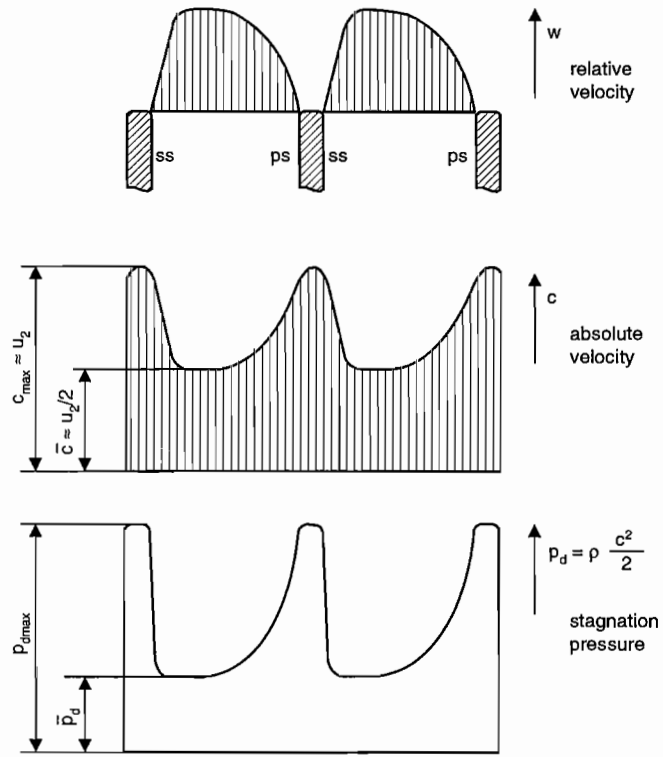$$\text{Phase resonance} \quad \frac{n \, z_2 (1 +/- M')}{z_3} = 1 +/- 0.25 \tag{4}$$

Figure 4. Pressure Profile at Impeller Outlet.

CL = 95% confidence limit
SD = standard deviation
AA = arithmetic average

$$\Delta p^*_{RMS} = \frac{2 \, \Delta p_{RMS}}{\rho \, u_2^2}$$

ρ = density
u_2 = impeller tip speed
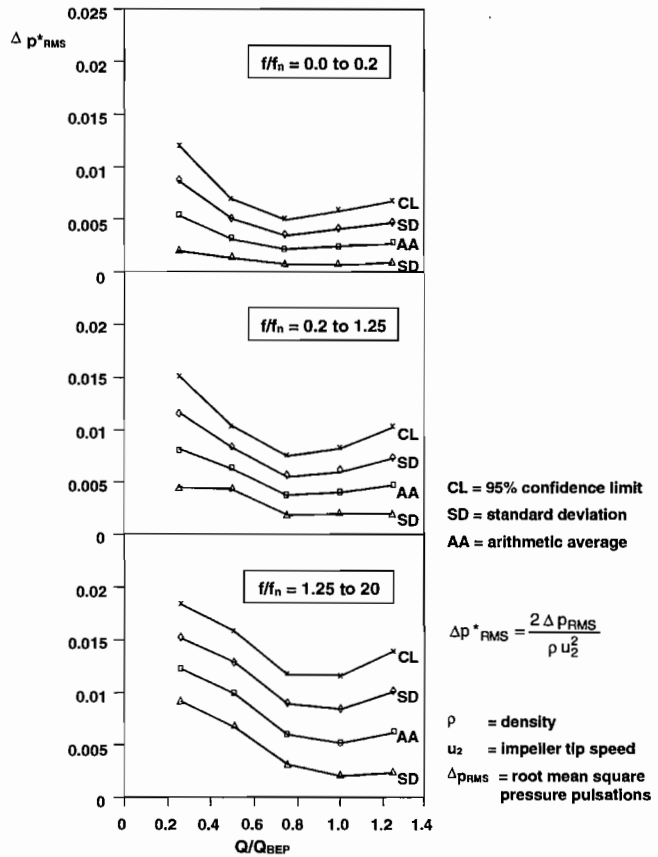Δp_{RMS} = root mean square pressure pulsations

Figure 5. Pressure Pulsation Versus Percent BEP.

Pressure pulsations are likely to increase dramatically for combinations of multiples of z_2 and z_3 that result in a difference of

zero or one, and for higher Mach numbers. Simple rules have been developed to avoid adverse combinations of blades z_2 and z_3. Equation (5) may be used to avoid radial forces and Equation (6) may be used to avoid pressure pulsations.

Radial forces $n\,z_2 - m\,z_3 \neq 1$ (5)

Pressure Pulsations $n\,z_2 - m\,z_3 \neq 0$ (6)

In Equations (5) and (6) small values of n and m result in larger forces, thus values of one should be avoided. Values above three are not of concern. Phase resonance is generally not a problem in volute type pumps with five or more impeller vanes.

Propagation of Pressure Pulsations Through Pump and System

Vane pass velocity fluctuations generate sound pressure waves that travel through the system at the speed of sound in the fluid. Despite high local velocity and pressure variation at the outlet of the impeller, little compression of the liquid takes place, and only a small amount of energy is radiated as pressure pulsations into the system. Introducing diffuser vanes or volute tongues at the outlet of the impeller create an additional source of noise due to the impingement of the wake flow from the impeller. The impingement process generates further velocity fluctuations in the near region, resulting in an even stronger source of acoustic energy (this can be inferred from the strong impact that the distance has between diffuser and impeller vanes on the pressure pulsations).

Most of the energy associated with the impeller discharge velocity fluctuation is dissipated as heat by liquid drag and vortex decay within the fluid, but a small amount of the energy is used to compress the fluid locally. Only this latter part is radiated as acoustic energy into the liquid, and it is this acoustic energy that can be measured far from its source as pressure pulsation.

Influence of Gap Between Diffuser and
Impeller Vanes on Pressure Pulsations

The radial gap between the impeller vane outer diameter and the diffuser or volute cutwaters is considered to be the most important design parameter that affects the pressure pulsations at blade passing frequency. In Figure 6, test data from various literature sources are plotted that show the influence of this gap on the normalized values by Equation (3). On average, pressure pulsation decreases with a power of -0.77 of the radial gap between impeller and diffuser/volute vanes and is formulized in Equation (7).

Gradient $\Delta p^* \approx \left[\dfrac{D_3}{D_2} - 1\right]^{-0.77}$ (7)

Data from performance and field tests show the following trends for vane pass frequency:

• The levels of pressure pulsations in the suction branch are of the same magnitude as in the discharge branch, and are in the same range for both single stage pumps and multistage pumps.

• At 25 percent of BEP flow, the pressure pulsations are roughly twice as high as at BEP.

• Virtually all pressure pulsations (dimensionless values at the 95 percent confidence limit) are below 0.015 at the BEP flow and below 0.02 at 25 percent of BEP flow (Figure 5).

Test data have also been collected relative to the dynamic hydraulic radial force (F_r), acting in all directions perpendicular to the axis of the pump rotor, resulting in standard factors (Figure 7) to be used in Equation (8).
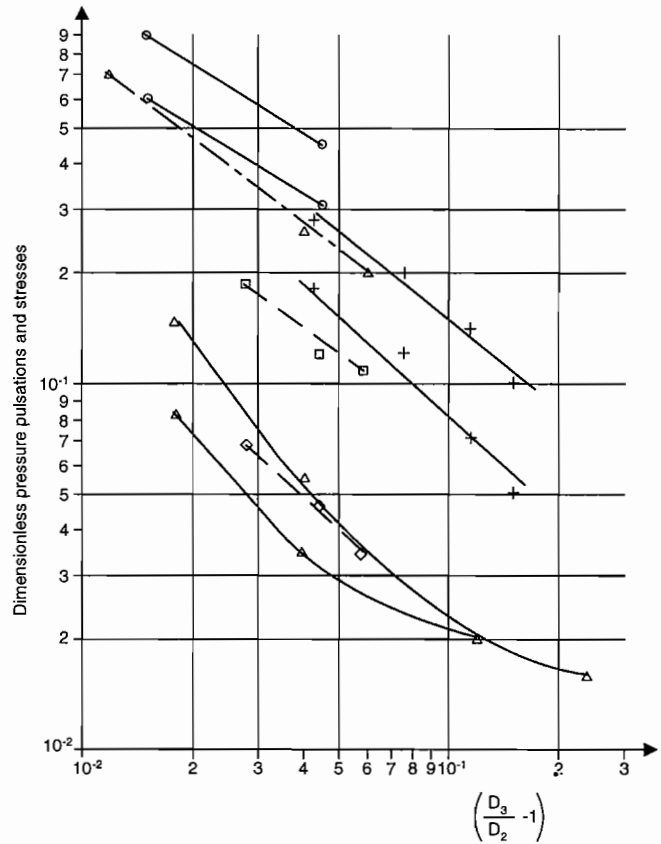
$$F_r = K_r\,\rho\,g\,H_{st}\,D_2\,B_2^*$$ (8)

Figure 6. Influence of B Gap.

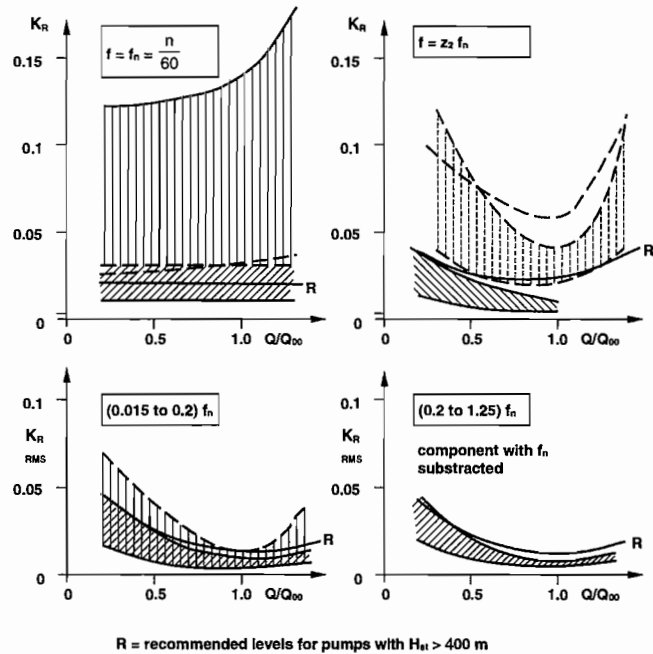R = recommended levels for pumps with $H_{st} > 400$ m

Figure 7. Radial Force Coefficient K_r.

Hydraulic radial forces (F_r) for multistage pumps with impeller vanes staggered from impeller to impeller are summed as root mean square (RMS) values per Equation (9).

$$F_{r,tot} = F_r \sqrt{Z_{st}}$$ (9)

Effects of Pressure Pulsations

Low-level pressure pulsations, and the associated noise and dynamic loading of a variety of pump components, are unavoidable and in general have no detrimental effect. Excessive pressure pulsations, however, can cause severe damage, particularly if associated with a system or structural resonance.

In the following, the consequences of excessive pressure pulsations are described (the list may not be complete).

• Dynamic loading at $(z_2 f_n)$ frequencies of diffuser vanes and volute tongues, and subsequent breakage of diffuser vanes and volute tongues at the inlet, are due to fatigue. Poor casting quality and poor finish of the vane fillet radii aggravate this risk.

• Even though the dynamic loading of the diffuser vanes is expected to increase with increasing pressure pulsations, no direct correlation between the dynamic bending stresses in the vanes and the pressure pulsations can be expected, as has been shown by Offenhaeuser (1973). The reason is that the velocity distribution (and therefore the vane loading) is not constant over the leading edge. This is particularly true at partial flow conditions.

• Dynamic loading of impeller side plates and impeller vanes at $(z_3 f_n)$ frequencies

• The dynamic loading of the impeller vanes is thought to be less than the dynamic loading of the diffuser vanes, because of the impingement of the wakes on the latter.

• Breakage of the impeller side plates due to dynamic loading may be associated with a resonance between a natural frequency of the impeller side plates and the exciting pressure pulsations. The shroud thickness and casting quality plays a very important role too.

• Dynamic loading of casing and bolts at $(z_2 f_n)$ frequencies

• Bearing housing vibrations (resonance) at $(z_2 f_n)$ frequencies, sometimes causing breakage of instrumentation or auxiliary pipes

• Baseplate vibrations may be excited from pressure pulsations. Multiplying the area of the (discharge or suction) nozzle times the measured or expected pressure pulsation amplitude can provide a rough estimate of these excitation forces. By using the spectrum of the pressure pulsations, a spectrum of excitation forces can be estimated and any resonance conditions with baseplate natural frequencies evaluated. Vane pass, 1×, and broad band pressure pulsations may be summed using the square root of the sum of the squares (SRSS) method per Equation (10).

$$\Delta p^* = \sqrt{\left(\Delta p^*_{0-0.2}\right)^2 + \left(\Delta p^*_{1x}\right)^2 + \left(\Delta p^*_{vp}\right)^2} \qquad (10)$$

• Vane pass pressure pulsations travel through the piping and can lead to a number of problems:

  • Standing waves and resonances

  • Upset of control systems

  • Excitation/breakage of instrumentation lines

• Finally, pressure pulsations are a primary cause of noise:

  • Airborne noise radiated from casing, nozzles, and bed-plates

  • Structure-borne noise radiated into piping and foundation

  • Liquid-borne noise in the piping

• Impellers with a very high suction recirculation at part load (inducers in particular) can cause low frequency pulsations in a surge-like manner when an expanding and collapsing vapor core form in the center of the suction pipe.

Other system caused pressure pulsations produced by vortex shedding at valves, elbow, etc., generally have frequencies, known as Strouhal frequencies, well below vane pass frequency. Also, cavitation (higher level broad band) and waterhammer events (normally lower frequency) within the system generally are easily recognized and different from vane pass pulsations.

Generally, the pressure pulsations generated by vane pass are not of a magnitude, which, on their own, would be considered harmful unless a resonant condition exists.

Resonance

Structural vibrations at vane pass frequency can be amplified by either a resonance with a structural natural frequency, or a resonance with a standing sound pressure wave in the piping system, or pump internal hydraulic passageway. *Resonant conditions are the leading cause of unacceptable vibrations.* If all sources of possible resonant conditions are removed, the possibility of unacceptable vibrations is greatly reduced. If no resonant conditions exist and the vibrations are still unacceptable, then the hydraulics or pump operation (off BEP for example) is producing unduly high pressure pulsations, and/or the structural system is inadequate.

Unfortunately many of the contributing factors leading to a resonant condition can be realized only after the pump is installed on its own base at site, and pumping the intended product at the specified operating speed and capacity. Particularly for pumps, bearing bracket/housing structural resonance and long crossover acoustic resonance (for multistage opposed impeller pumps) account for the larger part of the pump problems related to vane pass vibration. Also, resonance with pump support, piping, and shaft systems may occur.

Structural Resonance

Structural resonance occurs when the frequency of a periodic excitation force, or "forced vibration," exists at or near a natural frequency, a situation that can lead to large vibration amplitudes. To define resonance and establish the basis for further evaluations, the following is a discussion of the basic concepts.

Pumps are multidegree of freedom structural systems, and are generally composed of a support structure (foundation and baseplate), pump casing, external bearing housings, rotor, piping, and appurtenances. Each of these components or combination of components, like any elastic system possessing mass, has many different natural frequencies (bending, twisting, axial, etc.) These natural frequencies are simply the frequencies at which the pump structure will vibrate when placed into motion in the absence of external forces. The mass, stiffness, and damping distribution throughout the structure determine the characteristics of the natural frequency.

A single degree of freedom system, shown in Figure 8, may be used to model a structure (for example, a bearing housing, Figure 9). The simplest model to show this phenomenon is with a massless spring with stiffness K connected to a concentrated mass M at one end and to ground (fixed) at the other end (cantilevered beam). If the mass is then displaced and set free, the spring-mass system will oscillate at its natural frequency. For the simple spring-mass system, the undamped natural frequency can be expressed by Equation (11).

$$f_n = \frac{\sqrt{\dfrac{K}{M}}}{2\pi} \qquad (11)$$

This simple model only has one degree of freedom (translation, motion along one axis), thus only one natural frequency exists. For this system and for more complex structures, the lowest natural frequency is often referred to as the "reed" frequency. More complex structures (larger number of degrees of freedom) can be modelled by finite element and other methods.
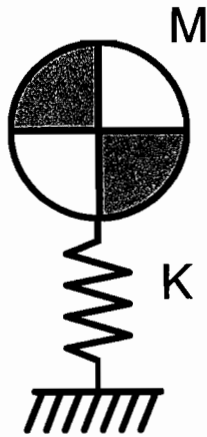
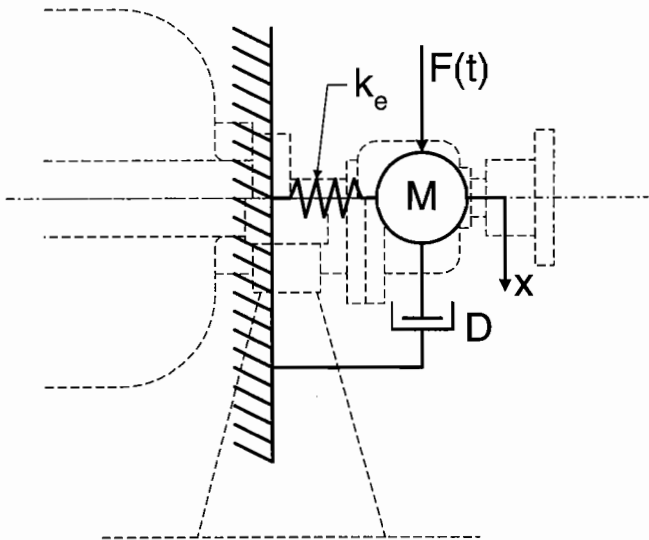*Figure 8. Single Degree of Freedom Model.*

Figure 9. Bearing Housing Model.

The vibration characteristics of the structural system will also vary, depending on how the excitation is applied. In Figure 10, there are three examples with a single degree of freedom model and pure sinusoidal excitation, one with constant excitation, one with unbalance excitation, and one with base excitation.

Phase angle, an important characteristic of a response, is the measurement of the relative time between the excitation and the reaction. At frequencies well below the natural frequency, the phase angle is near zero (reaction is in phase with the excitation force). At excitation frequencies equal to the natural frequency, there is a phase shift of 90 degrees, and at excitation frequencies above the natural frequency, the phase shift will be a full 180 degrees. This is important for diagnostics of a resonance.

There are three measurable characteristics that help in evaluating the severity of a resonant condition. These are separation margin, amplification factor (modal damping), and mode shape.

Separation margin reflects the difference between a natural frequency and the excitation frequency, and is normally expressed as a percentage of the excitation frequency. By definition, a separation margin of zero indicates an exact resonance. Because resonant vibration can become quite large, specific separation margins are required to minimize vibration.

Amplification factor Q, or often referred to as the quality factor, graphically depicted in Figure 11 and formulated in Equation (12), is the ratio of dynamic displacement to static displacement caused by the excitation force, and relates vibration amplitude, ratio of
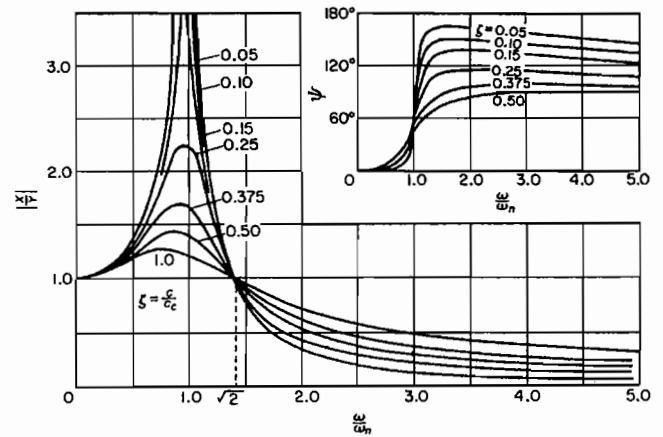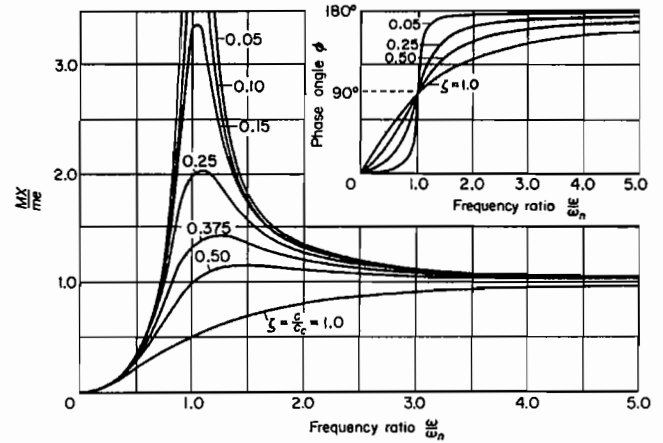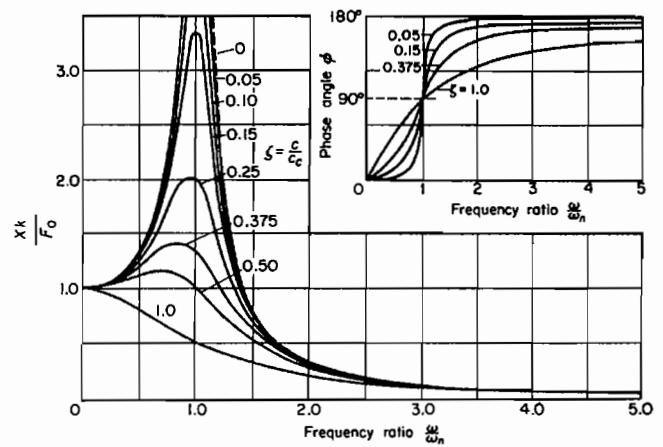

Figure 10. Typical Single Degree of Freedom Response Plots, with Constant, Unbalance, and Base Excitation, Respectively.

excitation frequency to natural frequency, and available damping. Two other commonly used measures of the amplification of a vibration are modal damping ratio D in Equation (12) (available modal damping/critical damping for a single mode) and logarithmic decrement δ per Equation (13) (measure of the decay rate of oscillation amplitude, refer to the damped response plot in Figure 12).

$$Q = \frac{N_c}{N_2 - N_1} = \frac{1}{2D} \qquad (12)$$

$$\delta = \ln\left(\frac{\hat{X}_n}{\hat{X}_{n-1}}\right) = \frac{2\pi D}{\sqrt{1-D^2}} \qquad (13)$$
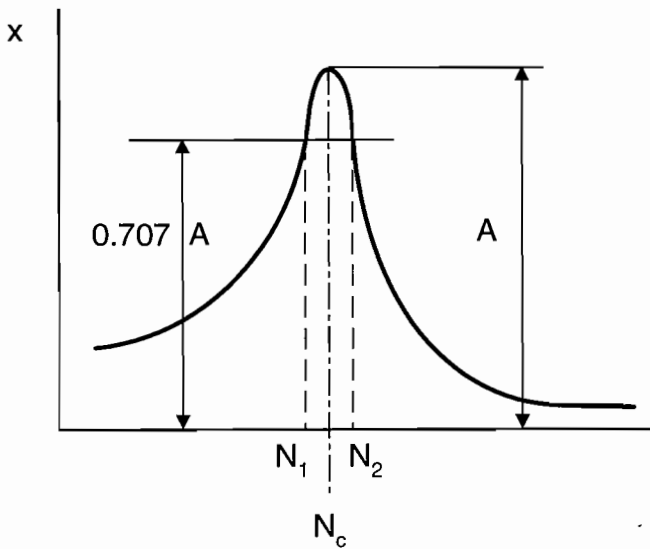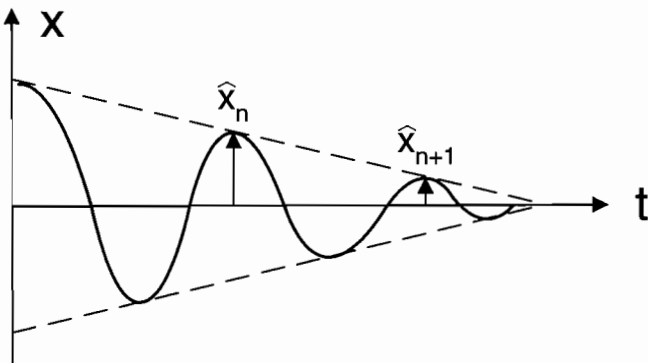
Figure 11. Amplification Factor.

Figure 12. Damped Response Plot.

Figure 13. Mode Shape Plots.

Equivalent viscous damping is used in place of Coulomb or solid damping, since they are typically difficult to determine directly, and are amplitude and frequency dependent. The undamped and damped natural frequencies for a typical steel structure are nearly the same, since structural damping is low.

Other Structural Resonances

Resonance may occur between vane pass pressure pulsations and diaphragm sections (structural membranes) or flat plate sections, for example, impeller shrouds or flat sections in the pump casing or pipe walls.

Resonance with a rotor natural frequency may also exist. In pumps, rotor lateral natural frequencies are generally speed dependent, due to the stiffening effect from the hydraulic interaction at annular seals (wear rings, bushings, etc.). This is typically referred to as the Lomakin effect. Since we are only able to measure rotor vibration at or near the bearings, it may be difficult to determine if a rotor natural frequency has been excited, and within the pump what the actual amplitudes of vibration are. Secondary effects, such as vibration, phase, waveform, etc., need to be investigated. There are methods of testing for rotor natural frequencies in operation, but these methods require special test equipment and experience.

Acoustic Resonance

Acoustic resonance occurs when a periodic excitation generates pressure waves traveling at the speed of sound, with a wave length (or quarter, half, two times the wave length) equal to the length of a fluid filled channel that has reflective conditions at the channels ends. Pressure waves are reflected, or a portion thereof, at valves, orifices, reductions, and expansions in the fluid channel. The reflected wave can result in an amplified standing wave. Amplification factors as high as 20 to 30 times the excitation energy are not uncommon.

Similar to structural resonance, basic acoustic theory, generally referred to as "organ pipe" theory, can be used to model the phenomenon adequately to verify possible resonant conditions in the various fluid passages. Fluid channels in pumps and piping are long and slender (channel width normally much less than the pressure sound wavelength related to vane pass), and can therefore be modelled as a one-dimensional system. The basic relationship

For each natural frequency, there is an associated unique mode shape (Figure 13) that depicts the vibration deflection pattern of a particular structure. Mode shapes show points of expected maximum displacement as well as minimum (nodal) points of the structure. Therefore it becomes clearer where corrective measures may be made. More energy is required to excite higher order modes to the same displacement amplitude. Likewise, stresses in higher order modes with the same displacement amplitude will be higher.

With regard to vane pass frequency, bearing housings will normally be resonant with first order modes, pump and support structures will normally be resonant with first or possibly second, and pump rotors will normally be resonant with second or higher orders.

Modal damping has an influence on natural frequency (Equation (14)) and to a greater extent on the magnitude of the vibration. The higher the damping, the lower the amplification factor will be. The spring-mass model above would vibrate forever without damping. Additionally, without damping, amplitudes of the resonant vibrations become infinite, even with the smallest of excitations. Typical steel structures have between two and five percent damping, depending on the type of construction. Bolted type constructions having higher damping than welded structures. Also, different materials have different damping characteristics, for example, cast iron has higher structural damping than steel. The damped natural frequency can be determined as follows:

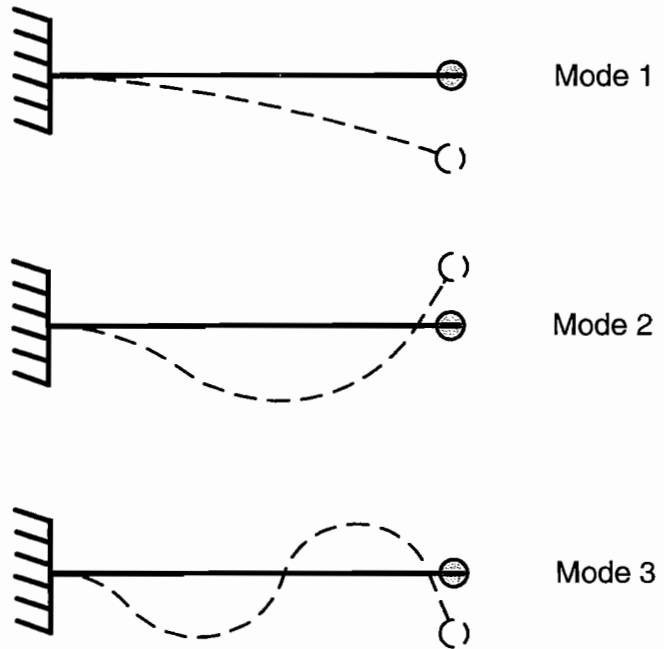$$f_n = \frac{\sqrt{\frac{K}{M} - \left(\frac{C}{2M}\right)^2}}{2\pi} \qquad (14)$$

between frequency, f, speed of sound, a, and wavelength, λ, is given in Equation (15).

$$f = \frac{a}{\lambda} \qquad (15)$$

For this tutorial, the excitation mechanism is vane pass pressure pulsations, and thus the frequencies of interest are vane pass and vane pass harmonics.

Resonances occur at multiples of half-wave and quarter-wave lengths, depending on passage length and the boundary condition at the ends (Figure 14). The natural frequencies of a particular passage of length L (for example, a multistage pump long crossover, Figure 15) for half wave resonance (open-open or closed-closed) is given by Equation (16), and for quarter wave resonance (open-closed) is given by Equation (17). Thus a resonance can be defined when $a / f = \lambda = 2 L / n$.
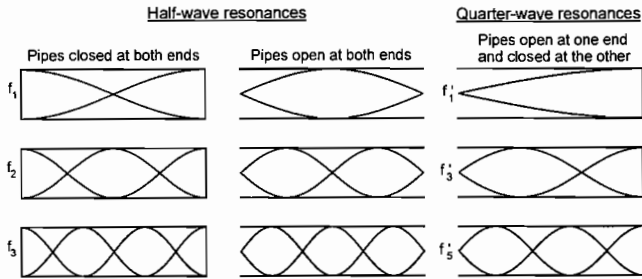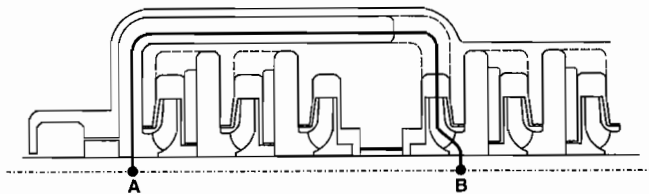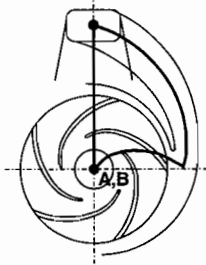

Figure 14. Organ Pipe Acoustic Modes.

Figure 15. Acoustic Resonance Plots (Pump Long Crossover).

Half wave resonance (open - open or closed - closed) $f_n = \dfrac{n\,a}{2\,L}$ (16)

Quarter wave resonance (open - closed) $\quad f_n = \dfrac{(2\,n - 1)\,a}{4\,L}$ (17)

Since damping is quite low for these types of systems, amplification peaks are typically high and narrow. Therefore, small changes in pump operating speed or the speed of sound can have a great effect on the pressure pulsation amplitudes.

Unfortunately, one variable that is sometimes difficult to evaluate is speed of sound in a particular fluid. A general formula for speed of sound is given (Equation (18)). There are several

fluids, particularly in the petrochemical industry, for which speed of sound or even its bulk modulus is not available, presumably for proprietary reasons. The main variables needed to calculate the speed of sound is density, ρ, and isentropic bulk modulus, K_s. Since density and bulk modulus for a particular fluid are temperature and pressure dependent, so is the speed of sound.

$$a = \sqrt{\frac{K_s}{\rho}} = \sqrt{\frac{(C_p/C_v)|P_2 - P_1|}{|\rho_2 - \rho_1|}} \qquad (18)$$

The flexibility of the fluid containing structure also has an influence, and may be simply defined by the structure effective diameter, d, wall thickness, t, and modulus of elasticity, E. The speed of sound in a fluid filled passage is defined by Equation (19).

$$a = \sqrt{\frac{K_s}{\rho\left(\dfrac{d\,K_s}{E\,t} + 1\right)}} \qquad (19)$$

Due to the lack of speed of sound data, a chart (Figure 16) relating speed of sound to density has been devised for estimation purposes only. The chart is based on theory and experience, but has some potential for error (±15 percent).
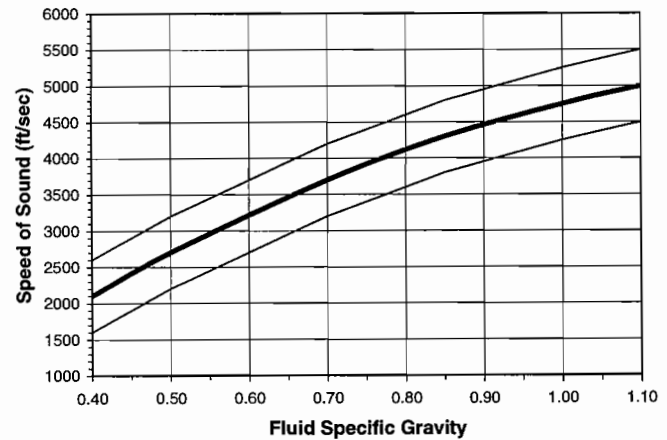

Figure 16. Speed of Sound Plot Versus Density.

Transfer Mechanism

The mechanism that transforms the pressure pulsations to bearing housing vibrations is not so obvious. Possible paths include through the pump casing or via the shaft. Yet shafts are relatively flexible compared with bearing bracket/housings, and, to a large extent are supported by the interaction forces at the annular seals. Also rotors are excited at stator $(z_3\ f_n)$ frequencies. So, a force applied to the impeller is, for the most part, transferred to the pump casing via the annular seal interactions. Pump vibration may also be driven by the reaction forces of the pressure pulsations within the piping system, and will be directed along the axis of the pump nozzle. However, the most probable mechanism of energy transfer to the bearing housing is pressure pulsations within the diffusers/volutes. These pulsations cause casing displacements, like a bellows. For a double volute pump, pulsations are alternating sides, thus more likely to excite lateral bearing housing modes.

The transfer of energy may be modelled by applying the excitation to the bearing housing, which for simplicity could be modelled as a mass-spring-damper system, per Figure 17. The transferred energy is applied either at the pump casing/bearing bracket intersection (base excitation), or as a forcing function at the shaft. Analysis with more complex models indicates that little

energy is transmitted to the bearing housing from the shaft, while the bearing housing vibration has a greater influence on the shaft vibrations. Of course resonant condition also has an effect, but in general the trend remains the same. So reducing casing displacements (vibration) reduces bearing housing, and possibly shaft, vibration.
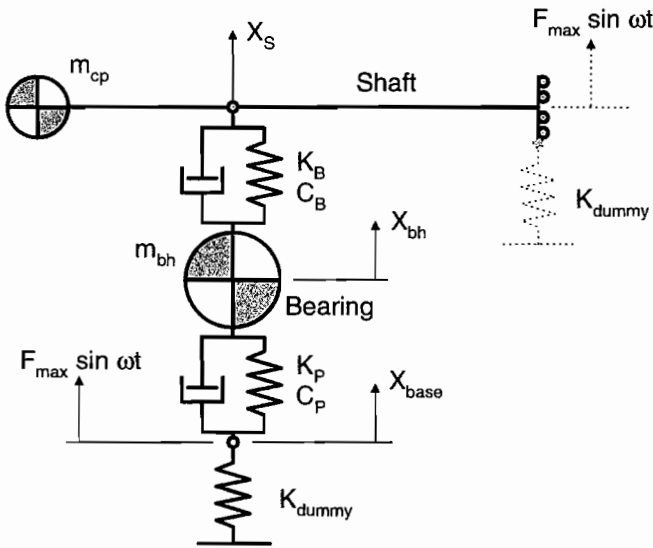

Figure 17. Simple Spring-Mass-Damper Model.

ASSESSMENT (Vibration Analysis)

Vibration analysis is performed to determine whether a vibration is excessive and, if so, why. High excitation forces, resonance (structural or acoustic), or weak structure all influence vibration levels.

Measurement and Diagnostic Technique

Since structural vibration is easily measured, vane pass excitations are generally quantified in terms of either bearing housing or shaft vibration, as these are the two most favorable methods of monitoring a pump's condition. Ordinarily, bearing housing vibration is measured in units of velocity, and shafts measured in units of displacement. Due to the higher frequency of vane pass, it is less noticeable with shaft displacement readings (Figure 18), therefore most vane pass vibration problems are found with bearing housing readings.
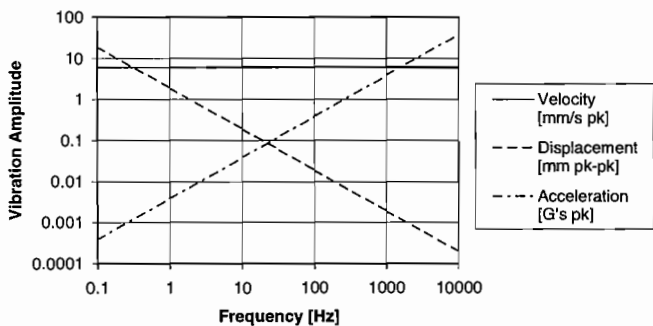

Figure 18. Relation of Vibration Units—Velocity, Displacement, and Acceleration.

When vane pass vibration is suspected, accurate diagnosis of all the contributing factors is necessary to determine the best corrective action. Sound diagnostics should be used to determine the correction instead of the old "trial and error" methods. To achieve an accurate diagnosis, full pump operating data, the proper equipment, and systematic routines are essential.

Operating data required to assess the magnitude of the excitation include flow, operating speed, suction and discharge pressure, product temperature and density, and, if possible, bulk moduli or speed of sound, impeller vane diameter to diffuser/volute lip clearance, impeller vane number, and stator cutwater number.

To measure vibration, a multichannel analyzer is preferred; however, a single channel with a tachometer (trigger) input would suffice. A standard set of readings on the bearing housing can verify with high certainty whether vane pass excitation exists. Measurements along the pump length, noting amplitude, frequency, and phase, will reveal the operating deformation shape. From this, the relative movement, it will become clear whether or not the whole pump is vibrating or just the bearing housings, etc. Impact tests should be conducted to determine if a resonant structural natural frequency exists, which would contribute to the amplitude of the vibration. Finally, pressure pulsation measurements should be taken within the suction and discharge piping, and any accessible pump internal passageway.

Noise is generated by structural vibration, incorrect impeller diffuser/volute vane number combinations, off-peak operation, or acoustic resonance. Acoustic resonance is usually the worst, and may be an indication of vane pass problems. Noise measurement may set off a warning, but, to date, without very special equipment, discerning a vane pass vibration severity with noise data is difficult, and therefore is not addressed here.

Every pump is different, having larger or smaller shafts, roller element or journal bearings, and stiffer or less stiff pump and bearing supports. Therefore shaft stiffness, bearing type, and bearing support stiffness need to be evaluated before the method of vibration measurement and acceptance level can be set.

General

The analyzer should be initially set for a range that covers from ≈ 0 Hz to two times vane pass frequency, and have a resolution that gives at least one line for each unit of frequency (Hz). If the overall vibration is much larger than any visible peak in the spectrum, then the range needs to be increased to check for possible causes, higher harmonics of vane pass or other excitations.

The signal (displacement, velocity, acceleration, pressure, or straingauge measurement as a function of time) is then treated by fast Fourier transformation to produce spectrums that give peak or RMS amplitudes as a function of frequency. Figure 19 shows an example. The spectrum shows distinct peaks at blade passing frequency ($z_2 f_n$) and its harmonics (often the second harmonic comes out strongly), as well as the rotational frequency, f_n, and its harmonics. Apart from the above, broad band pressure pulsations are present due to turbulence and/or cavitation. They increase strongly due to flow recirculation at part load, and in particular at low frequencies.

Noncontacting proximity probes are used for shaft measurements and are generally mounted in the bearing housing. Thus the measurements are relative to the stator. Accelerometers are used to measure stator vibrations, and are normally transformed to velocity readings. Velocity or seismic probes may be used to measure stator vibrations, but probe size and frequency range need to be checked for acceptability. High frequency piezoelectric pressure transducers are used for pressure pulsation measurement. Straingauges are used to measure structural strain or stress. Due to the labor intensive application of straingauges, their use is not considered practical. However if placement of pressure probes is not possible, straingauges may be an alternative to determine pulsations within a pipe section by backing out the pressure required to produce the measured strain amplitude.

If broad band vibrations are predominant, measurements may be read using a log scale ($dB's = 20 \log_{10}$ (vibration units)) rather than a linear scale (velocity, displacement, etc.). Vibration peaks are
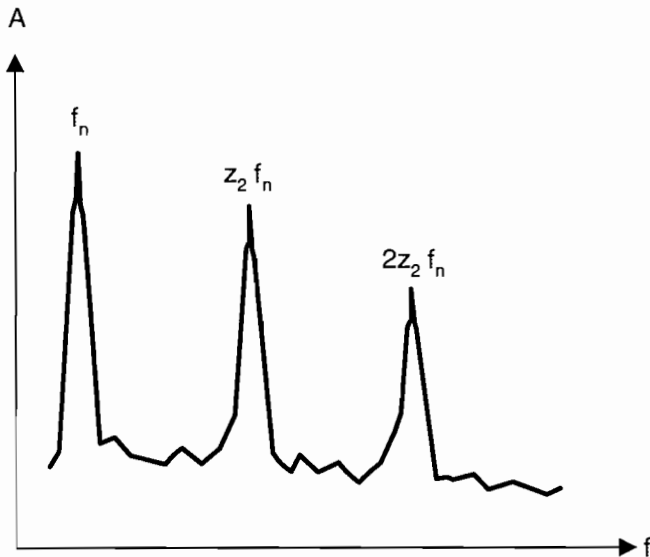
Figure 19. Example Spectrum.

Figure 20. Standard Vibration Measurement Locations on Pump.

more easily found using dB scales, and natural frequencies excited by broad band vibrations can be found there.

Check for and eliminate other possible sources of excitation in the vane pass frequency range, such as roller bearings, variable frequency driven electrical motor, belts, gears, etc. Other pulsating excitations (nonsinusoidal in form) at harmonics of operating speed may produce excitations at vane pass frequency. For example, synchronous rubs or loose parts produce multiple harmonics of operating speed, thus also providing excitation at vane pass frequency.

Vibration probes, including the mounting, need to be checked for their own natural frequencies. The mounting affects the probes' natural frequency, stud, magnet, hand held "stinger" type, clean or dirty, smooth or rough surface. Only probes with natural frequencies well above two times vane pass frequency should be used, and the surface where the measurements are to be taken should be clean and smooth. Standard field vibration monitoring tools should include degreasers and scrapers.

Standard Set of Vibration Measurements

Bearing housing and shaft measurements are taken, primarily for initial assessment of the vibration severity. Bearing housing measurements are generally taken with accelerometers at the locations PIV, PIH, POV, POH, and POA (Figure 20). Shaft measurements are taken with proximity probes at the locations PIX, PIY, POX, and POY, and are generally 45 degrees to the vertical (Figure 20). From these measurements, the standard spectrum plots are produced and are generally referred to as "signature" plots.

Also, a standard set of readings should be taken on all drive equipment (electric motors (Figure 21), turbines, gears, etc.) and other equipment in adjacent areas. These data are used to see if any other equipment is affected by the pumps vibration, or alternatively, if any other equipment is providing an external source of excitation to the pump in the frequency range of interest. This information can also be used to confirm any corrections made to the pump structure will not result in a resonance with another source of excitation.

Operating Deformation Shape (Forced Vibration)

The operating deformation shape of the problem vibration may be developed by taking multiple measurements at several locations on the pump, pump support structure, piping, and auxiliary piping (refer to Figure 22 for standard locations). These measurements
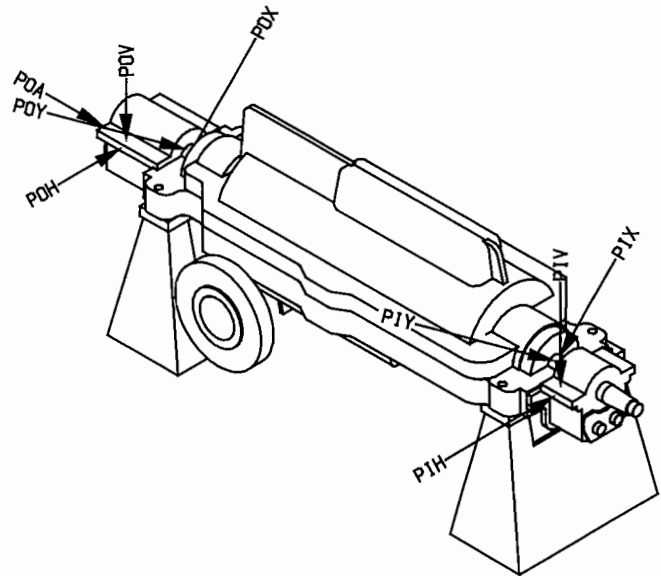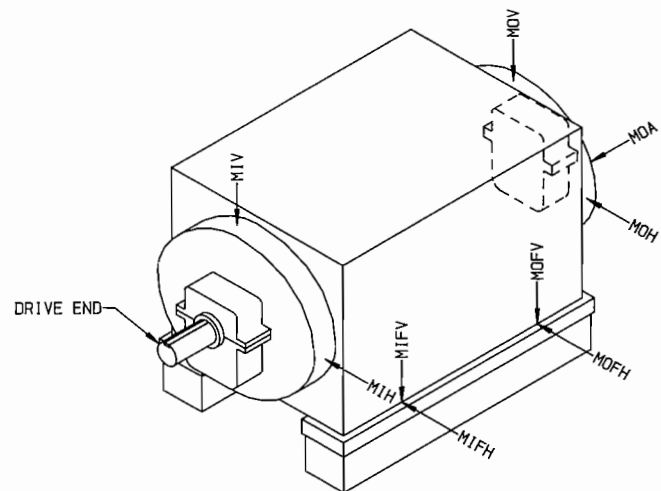

Figure 21. Standard Vibration Measurement Locations on Motor.

should be taken while the pump is operating at the condition (speed and capacity) that produces the unwanted vibration.

Results of the measurements are used to create an operating deformation shape plot. There is commercial software available for this type analysis; however, simple spreadsheets may suffice. From this it can be determined which parts are affected by the vibration and to what degree.

For this analysis, a two channel or a single channel with external trigger (set on $1\times$ source, once per revolution) should be used to enable the plotting of data with phase relationship. Single channel analyzers may be used and can produce usable results. However, without the phase relationship, the deformed shape may not be accurate, as all vibrations must be assumed to be in phase. The data should be filtered to the frequency of interest, in this case vane pass. As long as the data plotted are of consistent units, it will not matter what the units are, displacement, velocity, or acceleration, but displacements are suggested.

During this process, all locations of excessive vibration are noted. Once these areas have been isolated, more refined sets of measurements may be taken in these regions to further determine the elements that most effect the vibration amplitude. For example, on a bearing housing/bracket, measurements may be taken on both sides of a bolted joint to determine the flexibility of the joint.
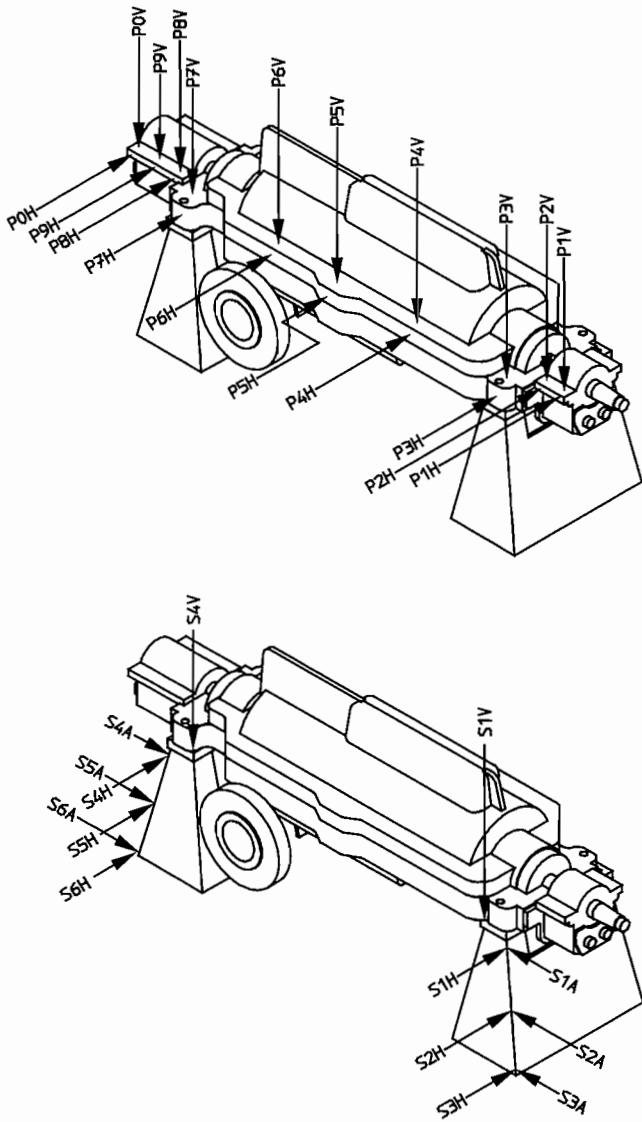
Figure 23. Example Impact Provisions.

Figure 22. Operating Deformation Measurement Locations.

Impact Testing (Natural Frequency Determination)

Impact tests are performed to determine structural natural frequencies and should be conducted on bearing housings, pump casing, piping, auxiliary piping, and any other suspect parts. Figure 23 shows standard impact test provisions. This is best accomplished with an instrumented hammer, which contains a force sensor in the head, in combination with a two channel analyzer. There are two good reasons for using an instrumented hammer. First, it provides a couple of ways to verify the existence of a natural frequency by giving phase relationship and coherence. Second, the transfer function and damping may be determined. Phase is important because a phase change of 90 to 180 degrees (180 degrees for a single mode and <180 degrees for multiple nearby modes), in combination with a response peak, verifies a natural frequency. Coherence is the ratio of coherent output power from each channel. Thus, a high ratio indicates that the two sources are related. Plots of the transfer function (vibration/excitation) and phase angle provide the best information for diagnostics. Using the transfer function eliminates the differences in the excitation force over the frequency range.

Without the instrumented hammer, it can only be assumed that every response peak represents a natural frequency. If an instrumented hammer is not available, then any large piece of
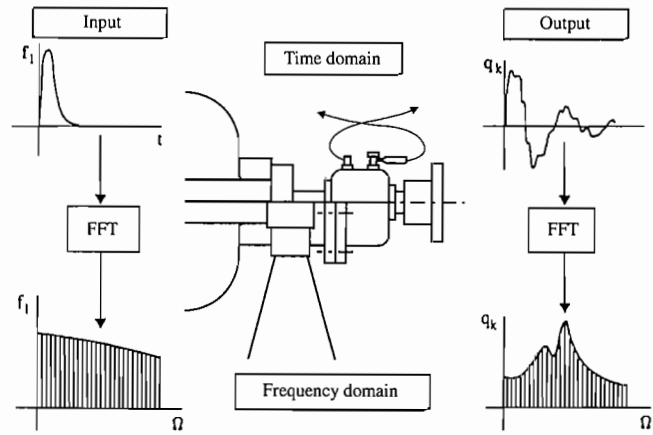
wood, or a rawhide or rubber mallet will do. The hammer weight should be selected so that sufficient excitation is introduced into the system to make measurements, but not so heavy as to cause damage to the equipment. Normally a 2 to 5 lb (1 to 2.5 kg) hammer for pumps is a good average size range. A hard metal hammer excites only higher frequencies, and may not excite the frequency range of interest completely. Likewise, too soft a hammer only excites low frequencies and will certainly not excite the vane pass frequency range.

Having the ability to do impact or modal testing while the pump is operating improves the results, especially at the bearing housings where the mass of the rotor carried by the bearings depends on the operating condition, and can be as little as a third of the mass at standstill. Also, for journal bearings at rest, the shaft is loose and resting in the bottom of the journal. When the pump is running, the hydrodynamic forces lift and support the rotor, both in the vertical and the horizontal directions. There is equipment and software available for this type of analysis, but due to the specialty, it is not further treated here.

Transient Analysis (Startup, Coastdown, or Operation with a Variable Speed Drive)

This analysis helps determine locations (frequencies) of response peaks of the operating pump. Figure 24 shows example transient plots. These response peaks may also indicate the existence of a natural frequency, which may be different from those determined while the pump is at stand still with impact tests. Although the pump structural natural frequencies will not vary greatly, shaft natural frequencies may change as much as 100 percent due to rotor casing interactions at wear parts and impellers. Bearing housing natural frequencies may also change (be increased), due to these same interactions that result in less shaft weight being carried by the bearings, thus a lower modal mass for the bearing housing.

Data are generally taken at the standard points (Figure 20), using "peak hold" or "peak averaging," to assure good resolution of the response peaks. For this type of analysis, a trigger (tachometer) is necessary so that output may be related to pump operating speed. Data need to be filtered to vane pass frequency. As for impact testing, phase is important for determination of a natural frequency.

Pressure Pulsation Measurements

Pressure pulsation measurements are used to determine the amplitude of the pulsations and if a standing pressure wave (acoustic resonance) is present. Unfortunately, most pumps and pumping systems are not fitted with tapped holes for pressure probes. Further, the proper location of the pressure probe is most important, since accidental placement near a nodal point in the standing sound pressure wave would result in very low, and what would appear to be normal, readings.
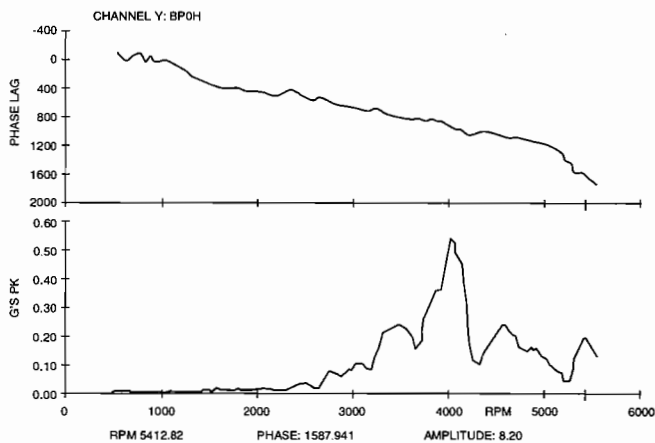
Figure 24. Example Transient Plots.

Measurements should be taken in the pump suction and discharge piping, and any accessible hydraulic passages within the pump (particularly long crossovers in multistage pumps). High frequency piezoelectric pressure transducers are used. If possible, it is best that the transducer is placed directly in the subject piping or pump hydraulic passageway. If placed in instrument piping, check that all air is removed and that the location of the probe is not affected by a possible nodal point or a standing sound wave located in the instrument piping.

A two-channel analyzer may be utilized to check whether the pressure pulsations are causing the pump vibrations, by placing an accelerometer or proximity probe on the second channel and checking the coherence between the two channels.

Pressure pulsations might be represented in different forms. However, the RMS values represent the energy content in a defined frequency band. Therefore, it is suggested to use these general values as determined with Equation (20) when evaluating the effect of pressure pulsations on the component or system.

$$\Delta p_{RMS} = \frac{\Delta p_a}{\sqrt{2}} = \frac{\Delta p_{p-p}}{2\sqrt{2}} \qquad (20)$$

It should be noted that pressure pulsations measured in the suction or discharge nozzle are only a fraction of the pulsations at the impeller outlet. Therefore, such measurements can give no reliable indication of the unsteady loading of impeller side plates, diffuser vanes, tie bolts, or other components.

Also, the system can have a great influence on the pressure pulsations measured at the inlet or outlet nozzle of a pump or anywhere else in the system. This impact is caused by wave reflections, wave interference, resonances, and standing waves. Furthermore, pressure pulsations from sources other than the pump under consideration can falsify the measurements (e.g., valves, ·orifices, other centrifugal or piston pumps, etc.).

Miscellaneous Measurements

Check all adjacent equipment and structures for vibration including harmonics of the pump operating speed, with particular attention given to drive equipment (gears, motors, etc.) A first round of measurements may be made with a single channel analyzer in monitor mode. If any suspect readings are found, they should be noted. And if a two-channel analyzer is available, then a second test to check phase and coherence may be performed by placing one probe on the pump and a second probe on the suspect equipment.

Comments Regarding Predictive Maintenance Trending

Vane pass vibrations vary with operating condition (operation at BEP, off BEP, pump speed, etc.). Thus when trending pump

vibrations, a wrong conclusion could be reached if the vibration readings are not taken at the same duty points and pumpage temperatures. Thus operating conditions, flow, head, temperature, etc., should also be observed and recorded.

Acceptance Criteria

There are several standards and other publications that propose acceptance criteria for pump vibrations such as API, ISO, HI, and others. However, because of the great variety in pump designs and the complexity of the dynamic structural systems, the criteria should be considered as general guidelines. The main purpose of acceptance criteria is to set specific levels, below which we can be reasonably certain that no failures will result, and above which the probability of mechanical failure increases significantly. Setting these guidelines for acceptable levels of vibration is difficult, but should be based on a rational/objective criterion so that perfectly well operating equipment is not rejected, resulting in wasted resources.

Pump support structures and bearing housings are structural members supporting a rotor running in tight clearances (at bearings, annular seals, and mechanical seals), and often support auxiliary piping and other appurtenances. Thus displacement and strength requirements should form the bases for vibration criteria. For example, shafts supported by journal bearings, acceptance based on bearing clearance makes sense, while for roller element type bearings, bearing housing measurements in terms of RMS velocity would be suggested due to the direct mechanical link between shaft and housing.

Higher frequency vibrations (like vane pass) result in lower level displacements and stresses, relative to a constant vibration velocity for a primary bending mode (Figure 18), and would suggest that higher amplitudes of vibration velocity may be allowed at higher frequencies. However, because there is little or no literature about failures resulting from discrete higher frequency vibrations of shafts, bearings, mechanical seals, auxiliary piping, etc., the vibration guidelines per Table 1 are suggested (refer also to Figure 25).

Table 1. Vibration Limits.

| Shaft vibration (unfiltered) at bearing for journal type bearings: compare relative shaft displacement peak-to-peak, with bearing diametrical clearance C_d (refer to Figure 25 for typical range of minimum bearing clearances)	
	Allowable displacement peak-to-peak
Good performance	$\leq 0.33\ C_d$
Improvement desirable	$> 0.33\ C_d$ and $\leq 0.66\ C_d$
Correction required	$> 0.66\ C_d$
Limits on shaft vibration filtered to vane pass frequency or harmonics thereof related to operating point	
$(0.90 - 1.10)\ Q_{BEP}$	Allowable velocity peak
$(0.75 - 1.25)\ Q_{BEP}$	0.28 in/sec (7 mm/sec)
$(0.50 - 1.40)\ Q_{BEP}$	0.35 in/sec (9 mm/sec)
	0.60 in/sec (15 mm/sec)
Bearing housing vibration unfiltered (for frequencies less than 10 Hz, the equivalent displacement at 10 Hz becomes the limit)	
	Allowable velocity RMS
Good performance	≤ 0.28 in/sec (7.1 mm/sec)
Improvement desirable	> 0.28 in/sec and ≤ 0.43 in/sec
Correction required	> 0.43 in/sec (11.0 mm/sec)
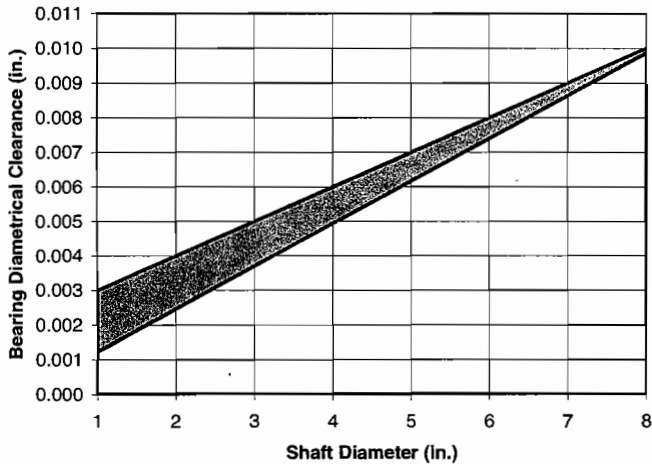
Figure 25. Typical Range of Minimum Journal Bearing Diametrical Clearances.

There are two cases where stresses are directly proportional to the velocity. First, for a vibration wave traveling through a piece longitudinally, defined as $\sigma = v\,(E\,\rho)^{0.5}$. Where v, E, and ρ are the wave velocity, modulus of elasticity, and density of the structure, respectively. These stresses are well below the endurance limit for standard materials, even at higher velocities and stress concentrations. Second, stresses are proportional to maximum lateral vibrational velocities at specific resonant frequencies, each with a specific mode shape. Generally these conditions do not exist in practice.

To further demonstrate that vibrations at a discrete frequency in the vane pass range are much less likely to result in failure than vibration at lower frequencies, the following points are made:

• When performing vibration analyses, mode shape is important. For all vibrations that occur below the first resonant mode, stresses are proportional to displacement. In general, most pump casing vibrations at vane pass fall within the first mode.

• Because vibration infers cyclic structural displacements (bending modes), and structural displacements are directly related to stress, one of the main concerns is fatigue. Therefore, number of stress cycles and material endurance limits need to be evaluated. Also, cumulative fatigue cycles need to be assessed, including vibration at other discrete frequencies and broad band, especially at lower frequencies, startup, shutdown, and other cycles. The endurance limit is simply the stress level below which a component will not fail while exposed to an infinite number of cycles. For example, fatigue life at 10^7 cycles will be reached in less than 19 hours of operation for a vane pass frequency of 150 Hz (1800 rpm (5 vanes) / 60), or less than two hours to reach 10^6 cycles (typical fatigue diagram in Figure 26). If there are no other cyclic loads (startup/shutdown cycles, thermal cycles, minimum flow operation, etc.) that produce higher stresses, then safe operation may be assumed after a few weeks of operation.

• When considering mechanical seals, there is no literature that indicates that lateral vibration at vane pass frequencies has resulted in seal failure. When considering velocity at a seal face, lateral vibrations are quite low. For example, a three-inch seal face diameter running at 1800 rpm, $V_{face} = N\,\pi\,D\,/\,60 = 1800\,\pi\,3\,/\,60 = 282$ in/sec $\gg V_{vib}$ of, say, 0.5 in/sec. Additionally, if we consider a five vane impeller, the displacement is equal to $60\,V_{vib}\,/\,(\pi\,f_{vp}) = 60\,(0.5)\,/\,(5\,\pi\,1800) = 0.001$ inches in the range of allowable runout. However from experience, axial vibration is much more damaging than radial vibrations for mechanical seals, thus these seal vibrations require more scrutiny.
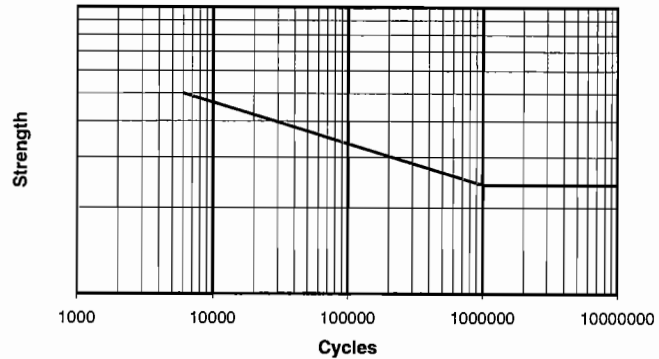

Figure 26. Typical Fatigue Diagram.

Acceptance Criteria for Pressure Pulsations

In the absence of other standards, it is suggested to use Figure 5 and Equation (3) as a level of pressure pulsations that are allowable with respect to continuous safe pump operation.

There are essentially two options to assess the adequacy of a specific pump with respect to pressure pulsations. They are:

• Check of geometric design features

• Measurements during operation

It is recommended to base an initial assessment on a check of the geometric design features rather than on acceptance testing for a number of reasons:

• The pressure pulsations measured depend strongly on the characteristics of the system. There is no standard procedure for such tests that would assure comparable results. In order to avoid the dependence of the measured pressure pulsations on the location of the transducer (i.e., in order to avoid the problem of standing waves), various laboratory methods are available to determine the acoustic energy radiated into a pipe by intensity measurements. The true acoustic energy emitted from the pump could, however, be determined only if nonreflecting pipe terminations are installed (which is extremely difficult in a liquid carrying system).

• The allowable pressure pulsations depend on the design of the pump, baseplate, and piping.

If pressure pulsations are measured in the plant or on the test bed, there are two means to reduce (but not eliminate) the uncertainties introduced by standing waves and resonances:

• Vary the test speed and calculate the average of normalized pulsations measured at different speeds (normalize according to Equation (5)).

• Average the pressure pulsations measured at different locations in the discharge or suction pipe.

CORRECTION

Vane pass vibration can be reduced by working either on the source of excitation or the resultant problem vibrating part. Based on the diagnosis and knowledge of corrective methods, the most effective and economical corrections can be made.

Vane Pass Pressure Pulsations

Correction to impeller vane outlets and diffuser/volute inlets may be made; however, the resulting change in performance needs to be assessed by a hydraulics engineer. Impeller vane outlet tips may be trimmed, under filed, over filed, cut at an angle, or otherwise profiled to reduce the wake. Diffuser/volute inlet tips may be cut back, cut back at an angle, or profiled to reduce the impingement forces from the wake produced by the impeller.

Pump energy level as well as specific speed may be used to judge the potential for a vane pass vibration problem. Certainly

higher energy levels result in a higher potential for forced vibration, possibly from vane pass. Higher specific speed pumps have larger hydraulic passageways with wider impeller outlets, and therefore more exposed vane tip creating wakes. Also, impeller and diffuser/volute geometry play a big part in vane pass intensity.

Recommendations to Minimize Pressure Pulsation

Table 2 lists those design features that should always be considered when the risk of excessive pressure pulsations is assessed. These design characteristics primarily determine the pressure pulsations generated by the pump. Table 3 lists additional parameters that might be corrected to reduce pressure pulsations. Figures 27 and 28 depict impeller corrections that reduce vane pass excitations.

Table 2. Recommended Design Features to Obtain Low Pressure Pulsations.

Parameter	Recommendation	Comment
Distance between impeller and stator vanes	$D_3/D_2 > 1.04$ for $H_{st} > 500m$ $D_3/D_2 > 1.06$ $D_3/D_2 > 1.06$ $D_3/D_2 > 1.10$	Mandatory diffuser Suggested diffuser Mandatory volute Suggested volute
Rotor/Stator vane combination	$n\, z_2 \neq z_3$ $n = 1, 2, 3$	Mandatory—For double volute pumps, 5 and 7 vane impellers are suggested
Staggering of impeller vanes	Stagger impeller vanes for double-entry pumps and for multistage pumps from stage to stage	
Impeller vane trailing edge	Apply appropriate profile to reduce width of the wake (Figure 27)	Trailing edge thickness must not be made too small for reasons of mechanical strength
Impeller vane number	Select number of impeller vanes to avoid structural or acoustical resonances	
High energy pumps	Use multistage pump with staggered impeller vanes to lower overall vibrations	
High energy-high specific speed pumps	Use two pumps in parallel	
Operation at off BEP	Select pump that will be operated at BEP	Use variable speed drive, if necessary

Part load recirculation at the impeller outlet leads to an increase of the pressure pulsations at distinct frequencies (such as blade passing), as well as broad band pulsations. Partial flow suction recirculation is thought to increase mainly the low frequency pressure pulsations. Hence a design check should include those geometric features that determine part load flow recirculation.

Of the corrections to the hydraulics, only a change to vane number has the potential to completely eliminate the vibration at a particular frequency. However when changing vane count, it is necessary to make sure that a new resonant condition is not created, particularly where VSD and pump operation with various fluids is expected.

A hydraulics engineer should oversee all corrections to the hydraulics, the heart of the pump. When making changes to the hydraulics, the actual performance requirements of the pump need to be considered. Often pumps are operated at different points than initially selected, thus the pump may be operating well off BEP. Therefore hydraulic corrections may also be made to better fit the actual required operating points.

It is nearly impossible to totally eliminate vane pass pressure pulsations; however, good hydraulic design should minimize the effects. If the vibrations are too strong, correction to the hydraulics alone will most likely not reduce vibration to an acceptable level. Therefore system or structural corrections will be required.

Table 3. Means to Reduce Pressure Pulsations.

|	Parameter	Comment
1	Oblique cut impeller vane trailing edge (Figure 28)	Effective means since D_3 / D_2 increases
2	Twisted impeller vane at outlet	Not always effective
3	Reduce impeller vane loading (near outlet)	Reduce angle
4	Proper profiling of diffuser vanes and volute tongue at inlet	Mechanical strength requirements must be observed
5	Oblique cut volute tongues	
6	Avoid/reduce flow perturbations upstream of pump (bends, especially multiple bends in different plane valves, booster pump with strong pulsations swirl break in suction pipe intake or in pump pits)	
7	Proper design of inlet bend to avoid vortices (including Karman vortices from ribs)	
8	Reduce thickness of impeller shroud at the impeller outer diameter (volute type pump only)	Mechanical strength requirements must be observed
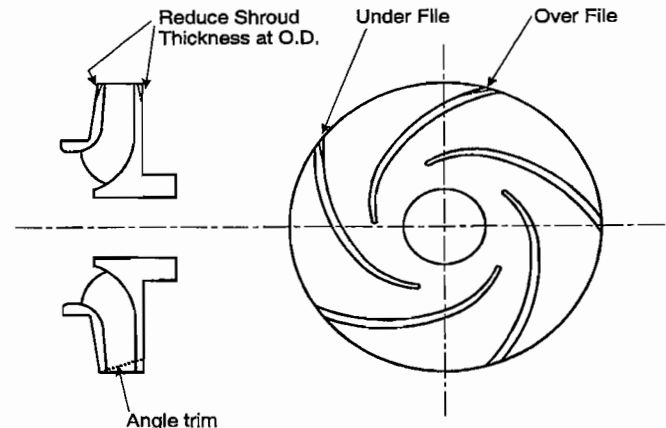
Figure 27. Correction at Impeller Vane Tips.

Weak Structure

If there are no resonant conditions and the excitation forces are reasonably low or cannot be reduced, the structure will need to be stiffened. For a weak structure, added mass, damping, or a vibration absorber will not help, unless there is a resonance within approximately 30 percent of operating speed.

To assist in locating the areas that are weakest or contribute most to the displacements, the operating displacement shape is most useful. Once the weak members have been identified, the method of application and the amount of additional stiffness need to be determined. Additionally, the frequencies of any structural natural modes need to be known so that the proper correction can be made without creating a resonant condition.

If there is a structural natural frequency below the excitation frequency, increasing stiffness will move the natural frequency into a higher range. This could result in a resonant condition, which will most likely produce higher vibration levels. Therefore, adding
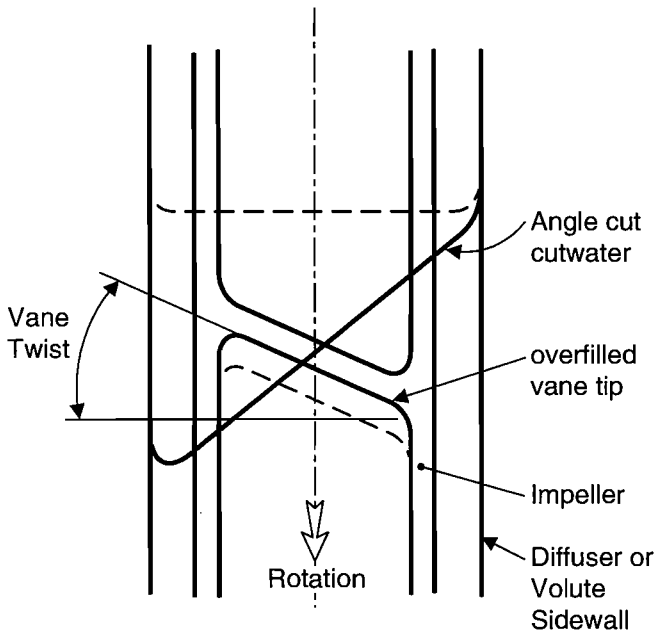
Figure 28. Angle Cut Volute Cutwater.

mass in combination with stiffness to maintain a separation margin between natural frequency and excitation frequency may be necessary. The alternative would be to increase the stiffness enough to push the natural frequency well above the excitation frequency. However in practice, it is difficult to increase the stiffness of an existing piece of equipment by the amount required to move a natural frequency from below to above an excitation frequency, and maintain an acceptable separation margin.

If the structural natural frequency is above the excitation frequency, then stiffness may be added, but care must still be taken not to push the natural frequency into resonance with other possible excitations or harmonics of vane pass frequency.

Additionally, any structural changes made to stiffen a structure or otherwise move a structural natural frequency in one direction will most likely influence the natural frequencies in other directions, depending on the type of change.

Resonant Structure

If a resonant structure is found to be the root cause, the same general considerations for correcting a weak structure should be followed. However, additional stiffness may be removed, mass added or removed, or a dynamic vibration absorber used, in order to displace the natural frequency, or "detune" the system.

Adding or Removing Structural Mass and/or Stiffness

The single degree of freedom system described above may be used to model most pump parts (for example, a bearing housing) to determine the amount of stiffness and/or mass to be removed or added to move a resonant natural frequency away from the excitation frequency (Equation (11)).

First the mass is approximated, then the stiffness can be calculated for a given natural frequency. A new stiffness and/or mass may then be determined that will move the natural frequency to the required separation margin. This generally works well for changes in mass, but to determine the required change in stiffness is more difficult. Most structural systems consist of several contributing parts and quite often with bolted joints in-between, thus it becomes difficult to sort out how much contribution each section makes to the total structural stiffness.

By adding mass on a fixed speed pump, the ratio of excitation to natural frequency point is moved to the right on the forced

response plot (Figure 10). For a VSD pump with added mass, the natural frequency can be lowered to an operating point of lower power content.

The safest way to prevent a resonant condition, especially for designs intended for varied operation, is to provide a rigid structure (structural natural frequencies are much greater than excitation frequencies). Unfortunately, we have found that greatly increased structural sections are required in order to provide a rigid structure. Pump casing-bearing bracket-bearing housing and bearing bracket/housing bolting all need to be greatly increased in strength.

Hand calculations are good for simple structures and approximations, but for more complex structures, finite element analysis can help determine or verify corrective action. However, depending on the size and type of model, and boundary conditions, results can be off by as much as 50 percent. One of the most difficult areas to model for modal analysis with finite elements is a bolted joint. Because bolted joints are typically prestressed and behave in a nonlinear way, the model needs to be quite comprehensive to reduce error. However once a resonance has been determined in an existing structure, it is much easier to adjust the analytical model.

Care should be taken to assure that pump integrity is not affected when corrections to the structure are made (for example, alignment).

Dynamic Vibration Absorber (Undamped)

Because most pumps and pump components (not including multistage rotors) can be modelled as single degree of freedom systems, a vibration absorber may be the solution if more permanent corrections (changing structural stiffness or mass) are not feasible. A vibration absorber is simply an additional spring and mass with a natural frequency equal (or nearly equal) to the original resonant frequency, and which is fixed to the original resonant vibrating part. The addition of this second spring K_2 and mass M_2 creates a second degree of freedom, thereby changing the vibration characteristics of the structural system. The new system now has two degrees of freedom. Thus two new and different natural frequencies that straddle the original resonant frequency are created (Figure 29).
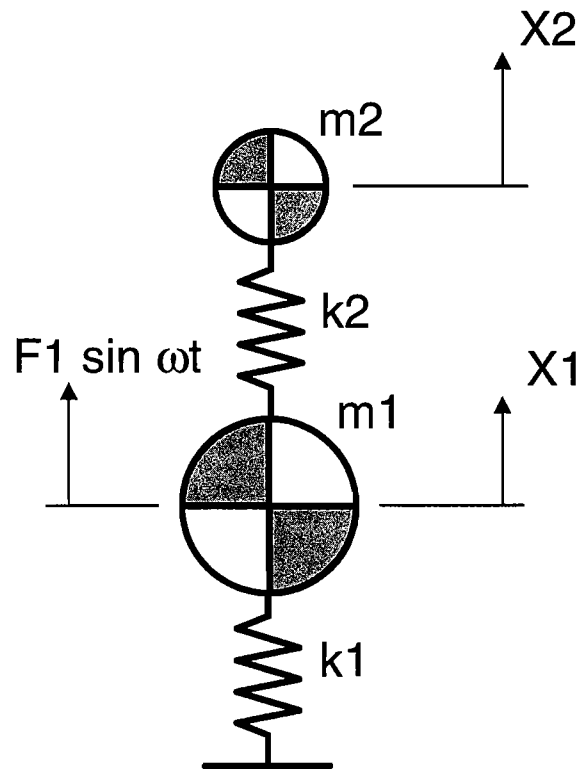

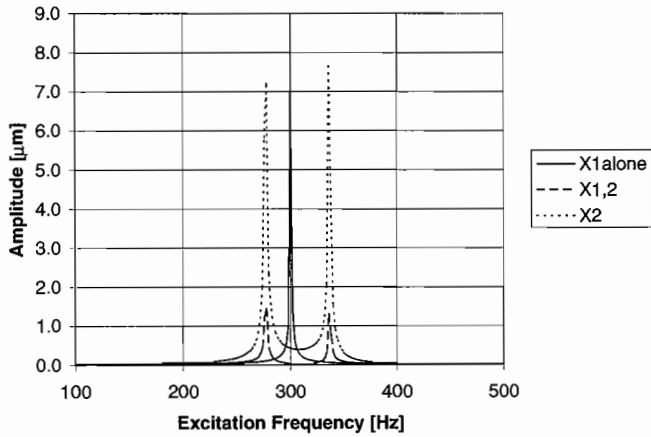*Figure 29. Two Degree of Freedom Model.*

Figure 30. Response Plots for Two Degrees of Freedom.

The separation margin of the new natural frequencies depends on the mass of the absorber M_2 and the ratio of the natural frequencies of the original part to the absorber $(K_1/M_1)/(K_2/M_2)$ (Figure 30). The larger the mass of the vibration absorber, the larger the separation margin. Also, for example, if the natural frequency of the absorber is less than the original piece, then the new natural frequencies will be skewed to a lower value, but still straddling the original natural frequency. Graphical aids depicting these dependencies may be easily made by plotting the ratio of natural frequencies versus mass ratios.

Several factors need to be checked before applying a vibration absorber. Vibration absorbers are generally only good for fixed speed units, depending on obtainable separation margins. Since separation margins are dependent on the mass of the absorber, and the amount of mass that can be safely applied is dependent on the existing structure and interference with other natural frequencies, the separation margin may be limited. The additional mass of the vibration absorber also needs to be accounted for, with regard to other natural frequencies. Additionally, if the subject part is asymmetrical (natural frequency in one direction is different from those in the other directions), the effects of the vibration absorber needs to be checked in all directions. The vibration absorber will normally have higher amplitude of vibration if a resonance occurs with its natural frequency, than the original piece. If this condition is expected, it should be checked for stresses. There are commercially available vibration absorbers with built-in damping.

Acoustic Resonance

When a standing wave is found to be a problem and its location known, the fluid filled passages may be lengthened, shortened, have a restriction, expansion chamber, or other specialized attenuator added in order to change the acoustic characteristics. Standing pressure waves may also be eliminated or reduced by improving the pump's hydraulics, as seen above.

In some cases, corrections to the hydraulic passages are not economically feasible. Consideration of pump performance effects must also be made. For example, long crossovers of multistage pumps, generally being a cast-in part, are expensive to modify and changes will affect the pump performance. In these cases, it may be less costly and easier to remove or reduce the excitation source (change impeller vane count, increase "B" gap, etc.).

Simple orifices, which do not overly affect the system performance (reduce pressure), generally will not change the acoustic characteristics enough to reduce pulsations to acceptable limits, and likewise for venturi tubes.

Injection of a gas into the system will change (lower) the speed of sound of the fluid, and thus change the acoustic characteristics of the system. However, depending on the process, injected gas may not be an option, and, further, the injection of a gas requires an additional system and the associated expenses.

Both performance and structural integrity need be considered when modifications are made.

CONCLUSION

Vane pass vibration concerns the effects of fluid-structure interaction on the vibration behavior of a flexible system (pump, pump components, and pump supports). It is our basic belief that we know what the excitation source is and what is necessary to minimize it. In general, we know the structural system and how to modify or detune it to avoid response to the source of excitation.

Providing optimized hydraulic designs (discussed above) should minimize vane pass excitation, but cannot completely eliminate them. Thus, system characteristics (resonances) need to be evaluated.

With regard to the pump structure, a stiffer and more massive structure is better, at the same time taking care not to be in a resonant condition. Although more and better tools (finite element analysis, etc.) are available to the present day engineer, they still are not 100 percent reliable. This is due to modelling difficulties (nonlinear effects, bolted joints, etc.) and unknown system related aspects (foundation stiffness and piping arrangements). On the other hand, once a resonance has been determined in an existing structure, it is much easier to adjust the analytical models and determine the proper corrective action.

In some cases, where variable speed drives are involved, the only way to avoid a resonant condition is by locking out certain specified speeds from the operating range.

With good diagnostic technique and a practical knowledge of corrective methods, the most effective, yet economical, corrections can be made.

NOMENCLATURE

A	=	Amplitude
a	=	Speed of sound
B_2^*	=	Total impeller width at outer diameter
C	=	Damping
c	=	Absolute velocity
C_d	=	Diametrical clearance
C_p/C_v	=	Ratio of specific heats
D	=	Damping ratio (percent of critical)
D_2	=	Impeller outer diameter
D_3	=	Diffuser or volute inlet diameter
D_a	=	Diameter of measuring location
d	=	Pipe diameter
E	=	Modulus of elasticity
e	=	Displacement
F_r	=	Radial force
f	=	Frequency
g	=	Gravity constant (386.1 in/sec or 9.81 m/sec)
K	=	Stiffness
K_r	=	Radial force coefficient
K_s	=	Bulk modulus
L	=	Length
M	=	Mass
M'	=	Mach number = $\pi D_a N / (60\, a)$
m, n	=	Integers 1, 2, etc.
N	=	Speed of pump shaft in rpm
P	=	Pressure
Δp	=	Pressure pulsation
Δp_d	=	Stagnation pressure
Q	=	Capacity or amplification (quality) factor
t	=	Pipe wall thickness or time
u	=	Circumferential speed
w	=	Relative velocity
X	=	Displacement
z_2	=	Number of impeller vanes
z_3	=	Number of diffuser or volute cutwaters
z_{st}	=	Number of pump stages
δ	=	Logarithmic decrement

η_h = Hydraulic efficiency $\approx (\eta_{pump})^{0.5}$
λ = Wave length
ρ = Density of fluid
$'\Omega$ = Frequency

Subscripts

2 = Impeller exit
3 = Diffuser inlet
a = Amplitude
c = Critical
max = Maximum
n = Harmonics, 1, 2, 3, etc.
p-p = Peak-to-peak
vp = Vane pass

Superscripts

$*,'$ = Dimensionless quantity

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