# METHODS OF INVESTIGATION AND SOLUTION OF STRESS, VIBRATION, AND NOISE PROBLEMS IN PUMPS


by
**William D. Marscher**
**Technical Director**
**Mechanical Solutions, Incorporated**
**Parsippany, New Jersey**
and
**Jack Campbell**
**Chief Consultant**
**Rota-Tech, Incorporated**
**Denver, North Carolina**


*William D. (Bill) Marscher is Technical Director with Mechanical Solutions Incorporated, in Parsippany, New Jersey. His 28-year career involves the design, analysis, and troubleshooting of pumps and other turbomachinery. His capabilities include FEA, rotordynamics, and modal vibration testing. Mr. Marscher's innovations won the 1986 Dresser Award and the 1983 ASLE Hodson Award. He pioneered the use of FEA for solving pump casing stress and leakage problems, and piping nozzle load effects. He is the originator of a specialized vibration test procedure that led to the solution of pump problems in power plant, water, chemical, and API installations.*

*Mr. Marscher was Chairman of the 1993 ASME RoCon rotating machinery conference, the 1998/99 President of the Society of Tribologists and Lubrication Engineers, and, since 1980, has been on the ASTM standards committees for cavitation erosion and metal fatigue. He has written chapters for several handbooks and is coauthor of* Centrifugal Pumps.

*Jack S. Campbell is the President of Rota-Tech Incorporated, a rotating machinery technical services company, in Denver, North Carolina. He specializes in troubleshooting, alignment, vibration analysis, and hot alignment techniques.*

*Prior to forming his own company, Mr. Campbell was employed by Duke Power Company in Charlotte, North Carolina, for 16 years as a rotating equipment specialist. He was responsible for the direction and maintenance activities on condensate and feedwater systems pumps.*

*Mr. Campbell has been a frequent lecturer at EPRI and the International Pump Users Symposium, has been an EPRI and Pump Symposium session organizer, and is a past member of the International Pump Users Symposium Advisory Committee.*

## ABSTRACT


This tutorial discusses mechanical problems that occur in pumps and their systems, and how pump users can best detect and solve these problems. The answers to pump operational or reliability problems are often not obvious, and indepth troubleshooting must be performed to reach a successful resolution. It is shown how a concise and orderly review of all apparent problems should be developed that begins with checks of the original installation, continues through startup procedure review and data monitoring, and includes evaluation of current operational behavior.

The tutorial discusses various problem types and typical causes. For example, the discussion includes overstress fractures, cyclic fatigue, rotor rubbing and binding, and pump-to-driver misalignment. It also shows how these problems can have their root cause in nozzle loading, thermal distortion, resonance, recirculation during running off-BEP, and improper warmup cycles. More importantly, it covers the proven methods of analyzing and testing to determine the nature of such problems, and how to solve them. Also included are some basic mechanical concepts for inspection and assembly of pumps that can help improve reliability. A number of case histories show, by example, how best to apply these inspection, analysis, and test methods. Enough detail is given concerning the method of solution in each case that the attendees should be able to try the solution methods back at their plants. An interesting example is also given showing how sometimes apparently unrelated failures actually have a common cause.


## INTRODUCTION

When failures occur in pumps and their associated systems, they generally fall into one of four categories: fracture, fatigue, rubbing wear, or leakage. Fracture occurs due to excessive loading, for example, from higher than expected pressure or nozzle loading beyond recommended levels. Fatigue requires that the imposed loads be oscillating so that stresses cyclically surpass the endurance limit of the cracking material. Fatigue in pump components is most commonly caused by excess vibration, which in turn is caused by the rotor being out of balance, by the presence of too great a misalignment between the pump and driver shaft centerlines, by excessive vane pass pressure pulsations, or by large motion amplified by a natural frequency resonance.

Rubbing wear and seal leakage imply that the rotor and stator are not positioned relative to each other within design tolerances. This can happen dynamically, and in such a case, excess vibration is generally the cause. When the wear or leakage is at a single clock position in the casing, unacceptable amounts of nozzle loading and either resulting or independent pump/driver misalignment are likely causes. In high energy pumps (especially hydrocracking and boiler feedpumps), another possibility for rubbing at one location on the stator (or for an axial rub or a thrust bearing wipe) is too rapid a change in temperature, which can cause a mismatch in the length and fit of each component, since these change with temperature.

If any of this brings to mind a past or present pump problem that you have experienced, you are in good company. Over 90 percent of all problems fall into the categories listed above. Fortunately, there are certain approaches and procedures that can be followed that minimize the chance for encountering such problems, or which help to determine how to solve such problems if they occur. These approaches and procedures are the subject of this tutorial.

## CASING STRESS AND DISTORTION ANALYSIS

Today there are a wide variety of methods that can be used in determining how much pressure or nozzle load a casing can tolerate before it is likely to crack or leak. Manual calculations can be performed with a calculator, and done properly, are generally accurate within better than a factor of two. More accurate calculations are generally done using the finite element analysis method ("FEA" or "FEM") on a PC. However, either the manual or computer method is only as accurate as the assumptions and information that get fed into it, and in the end, the best accuracy is provided by some form of test. Examples of tests that determine stress are application strain gauges or brittle lacquer "stress-coat" to determine stress concentrations (careful—both are very temperature sensitive). Dynamic stresses can be determined by using analysis calibrated by test results from shaft proximity probes or seismic probes (velocity probes or accelerometers) to determine vibration levels and the location of natural frequencies, which when poorly damped become the rotor system's "critical speeds."

Manual calculations for steady stress fall mostly into three categories:

- Pressure vessel
- Hollow beam
- "Hot rod"

The pressure vessel calculations can become very complicated if they include a lot of detail. An example of this is the ASME Boiler and Pressure Vessel Code Section III or Section VIII calculations. However, simple calculations can be sufficient in some cases to determine whether excess pressure is likely to be a problem. One of these simple calculations is to assume that the typical stress in the casing walls equals the internal pressure, times the maximum radius of the casing (not including the nozzles), divided by the wall thickness. When stress is calculated in this way, it is usually on the low side of reality, because of the presence of stress concentrations, such as a fillet radius at the casing end or at the volute sidewall. Therefore, it is best to multiply this number by a safety factor, generally between 2½ and 5.

For casings with large sections of wall that are not cylindrical, or which tend to be flat, a safer estimate is to assume that the stress may be as high as half the internal pressure, times the square of the maximum casing wetted dimension, divided by the square of the wall thickness. Usually, the true maximum stress is about one-fifth this value, so by using this estimate without any additional factor, the possible effects of stress concentrations are already included.

The hollow beam calculation assumes that the peak nozzle stress can be predicted as roughly three times the moment on the nozzle, multiplied by the nozzle radius, divided by the nozzle moment of inertia. Handbooks such as Marks' Mechanical Engineering handbook (1967) or Roark & Young's *Formulas for Stress & Strain* (1975) can be used to determine the stress in such cases, although the resulting number should be multiplied by about 2½, again because of the good possibility of the presence of stress concentrations in the region near the nozzle connection to the casing or volute walls.

A "hot rod" calculation checks out the maximum change in dimension of the casing between room temperature and the operating temperature. The maximum operating temperature of the casing is generally no greater than the operating temperature of the liquid being pumped. The casing expands as it is heated, and the amount of this expansion is roughly equal to the difference in the casing temperature and room temperature, times the length of the casing, times the thermal expansion coefficient of the casing. For reference, the thermal expansion coefficient is about seven millionths of an inch per inch of length per degree Fahrenheit (about 13 millionths of a meter per meter length per degree Celsius) for most steel, and about nine millionths of an inch per degree Fahrenheit (about 16 millionths of a meter per meter per degree Celsius) for 300 series stainless. Likewise, the rotor thermal growth is roughly the liquid temperature minus room temperature, times the length of the rotor from the suction end of the shaft to the thrust bearing, times the rotor's thermal expansion coefficient. This formula can be useful in determining whether enough clearance has been left for differences in casing versus rotor thermal expansion during startup or shutdown, or if the casing is uninsulated and runs much cooler than the liquid-immersed rotor. The numbers can get larger than expected. For example, for a six foot long carbon steel shaft (roughly two meters) at 400°F (about 200°C), the amount of growth is about 160 mils (4 mm). Sudden immersion of the rotor and casing at this temperature will result in this growth of the rotor, while the greater bulk of the casing causes it to warm up and expand more slowly, possibly causing a severe axial rub.

Another type of thermal growth that can cause binding problems is the curvature that can take place over the length of the casing or of the rotor, due to the differential temperature between the lower and upper extremities. The radius of this curvature is roughly the diameter of the component, divided by the lower versus upper temperature differential, and further divided by the thermal expansion coefficient, $\alpha$, of the component. From this curvature versus the length of the component, the amount of clearance taken up can be estimated with arc length formulas. For example, the "humping," h, of the casing or rotor (whichever has the temperature differential $\Delta T$ top-to-bottom across it) of diameter D and length L, where the "humping" is relative to the bearing-to-bearing centerline, is approximately:

$$h = \rho - \tfrac{1}{2} * (4 * \rho^2 - L^2)^{0.5} \qquad (1)$$

where:

$\rho = D/(\alpha * \Delta T)$

As a rule of thumb, binding becomes possible when the upper versus lower casing temperature differential exceeds about 100°F, with rubbing beginning at about half this value. This is the reason why many users of boiler feedpumps and hydrocracking pumps put their pumps on slow roll when the pump is taken offline, why warmup cycles are often carefully specified and followed, why casings may be insulated (besides the energy cost savings), and why excess seal injection water has sometimes led to rubbing and fatigue problems in shafts.

## MECHANICAL GUIDELINES FOR PUMP SELECTION AND OPERATION

- Avoid operation far from the best efficiency point ("BEP") of pumps. Contrary to intuition, centrifugal pumps do not undergo less nozzle loading and vibration as they are throttled back, unless the throttling is accomplished by variable speed operation. Operation well below the BEP at any given speed, just like operation well above that point, causes a mismatch in flow incidence angles in the impeller vanes and the diffuser vanes or volute tongues of the various stages. This loads up the vanes, and may even lead to "airfoil stalling," with associated formation of strong vortices (miniature tornadoes) that can severely shake the entire rotor system, and can even lead to fatigue of impeller shrouds or diffuser plates or "strong-backs." The rotor impeller steady side-loads and shaking that occurs at flows below the onset of suction or discharge recirculation (Fraser, 1985) leads to the strong possibility of rubs, and excessive rotor loads that can damage bearings. Many plants buy equipment that has more capacity than is needed, to allow for future production expansion, but in doing so ensure years of unreliable performance of potentially reliable machinery. Never run a pump for extended periods at flows below the "minimum continuous flow" provided by the manufacturer. Also, if this flow was specified prior to about 1985, it may be based only on avoidance of flashing and not on recirculation onset, and should be rechecked with the manufacturer.

- Avoid unrestrained expansion joints (piping "flexible joints") at large nozzles. The pressure across the cross-section of such nozzles, times that area, becomes a large thrust, similar to the thrust at the exit of a rocket nozzle. The rocket is free to accelerate because it is not tied down. Just because the pump casing is not free to move does not mean that it is not distorted by the thrust at the unrestrained nozzle; in fact, it makes the casing more likely to distort. This thrust can even overstress the nozzle, or indirectly cause excessive distortion in the casing or baseplate, leading to severe operating driver/pump alignment problems and possible rubs.

- During system commissioning, violation of vibration specifications is a common problem, particularly in variable speed systems where the likelihood of an excitation force's frequency equaling a natural frequency is enhanced. In vibration troubleshooting, investigate first imbalance, then misalignment, and then natural frequency resonance, in that order, as likely causes. In the case of a resonance, modal impact testing is a very effective and proven method of quickly finding the reason for the resonance, so that it can be fixed permanently. Typical fixes include selective bracing, or alternately adding mass to areas of maximum vibrational movement. Modal testing is best done while the machine is operating, so that the bearings and seals are "charged" and supporting the rotor in a manner typical of the pump's operating condition. Try to ensure that your manufacturer or any third party consultant that you hire has the capability for performing these "bump" tests while the pump is operating.

## VIBRATION ANALYSIS

One of the most common problems in new pump installations is vibration. This is particularly true if the pump is installed in the vertical position, if the pump is run at variable speed, or if the pump is to be steadily run at flows well below the design point. The problem vibrations most commonly discussed in the literature are lateral shaft vibrations, i.e., rotordynamic motion perpendicular to the pump axis. However, problem vibrations can also occur in the pump stationary structure, especially in vertical pumps. In addition to lateral vibration, vibration can occur in the axial (thrust) direction, or can involve a torsional (twisting) motion.

Vibration and other unsteady mechanical considerations should include analysis of:

- Rotordynamic behavior, including critical speeds, forced response, and stability.
- Torsional critical speeds and oscillating stress, including startup/shutdown transients.
- Piping and nozzle load-induced unsteady stress and misalignment-causing distortion.
- Fatigue of high stress components due to oscillating torque, thrust, and radial load.
- Bearing and seal steady and dynamic behavior.
- Lubrication system operation during normal operation and trip coastdowns.

Unsteady fluid dynamic considerations include assessment of:

- Levels of oscillating pressure under part load operation, to minimum continuous flow.
- System operational control capabilities, including failsafe protection, to prevent, for example, running near shutoff with resulting recirculation and flashing.
- Acoustic (e.g., like a trumpet) resonances in combined pumps and systems.

The mechanical issues can be judged with the aid of API, ANSI, ASME, ISO, DIN, HI, and other standards. The fluid dynamic considerations generally require the aid of a specialist, either from the plant's engineering group, from a large manufacturer, or from a consulting company.

An important concept is the "natural frequency," the number of cycles per minute that the rotor or structure will vibrate at if it is "rapped," like a tuning fork. Pump rotors and casings have many natural frequencies, some of which are generally in or close to the operating speed range. The vibrating patterns that result when a natural frequency is close to the running speed, or some other strong force's frequency is known as a "mode shape." Each natural frequency has a different mode shape associated with it, and where this shape moves the most is generally the best place to try a "fix," such as a brace or an added mass.

If the excitation force frequency and the natural frequency are within a few percent of each other, this causes "resonance." In resonance, the vibration energy from the last "hit" of the force has come full cycle, and is restored up when the next hit takes place. The vibration in the next cycle will then include movement due to both hits, and will be higher than it would be for one hit alone. The vibration motion keeps being amplified in this way, until its large motion uses up more energy than the amount of energy that is being supplied by each hit. Unfortunately, the motion at this point is generally quite large, and damaging.

Resonance is illustrated by someone playing basketball—his dribbling (the exciting force) synchronizes with the ball bounces (the ball's natural frequency), but if he is uncoordinated, the ball will not bounce very high. The imbalance force in the pump, which oscillates high and low in a given direction once per revolution, should not start "dribbling" the rotor. In other words, the natural frequencies of the rotor and bearing housings should be well separated from the frequencies at which "dribbling" type forces will occur, which tend to be 1× running speed (typical of imbalance), 2× running speed (typical of misalignment), or at the number of impeller vanes time running speed (so-called "vane pass" vibrations from discharge pressure pulses as the impeller vanes move past the volute or diffuser vane "cutwater").

In practice, the vibration amplification Q due to resonance is usually between a factor of two and twenty higher than it would be if the vibration force was steady instead of oscillating. Q depends on the amount of energy absorption, called "damping," that takes place between hits. In an automobile body, this damping is provided by the shock absorbers. In a pump, it is provided mostly by the bearings and the liquid trapper between the rotor and stator in the "annular seals," like the balance piston. The amount of vibration amplification that is typical when a force that acts at a certain frequency (like running speed) passes through a natural frequency is illustrated in Figure 1.

Two curves are given in Figure 1: one for high damping (not much amplification), and one for low damping. One way to live with a resonance (okay in a pinch, but not recommended) is to increase the damping by closing down annular seal clearances, or switching to a bearing that, by its nature has more energy absorption (e.g., a journal bearing rather than an antifriction bearing).

Another important concept is the "phase angle," which measures the time lag between the application of a force and the vibrating motion that occurs in response to it. This is illustrated in Figure 2.

A phase angle of zero degrees means that the force and the vibration due to it act in the same direction, moving in step with one another. This occurs at very low frequencies, well below the

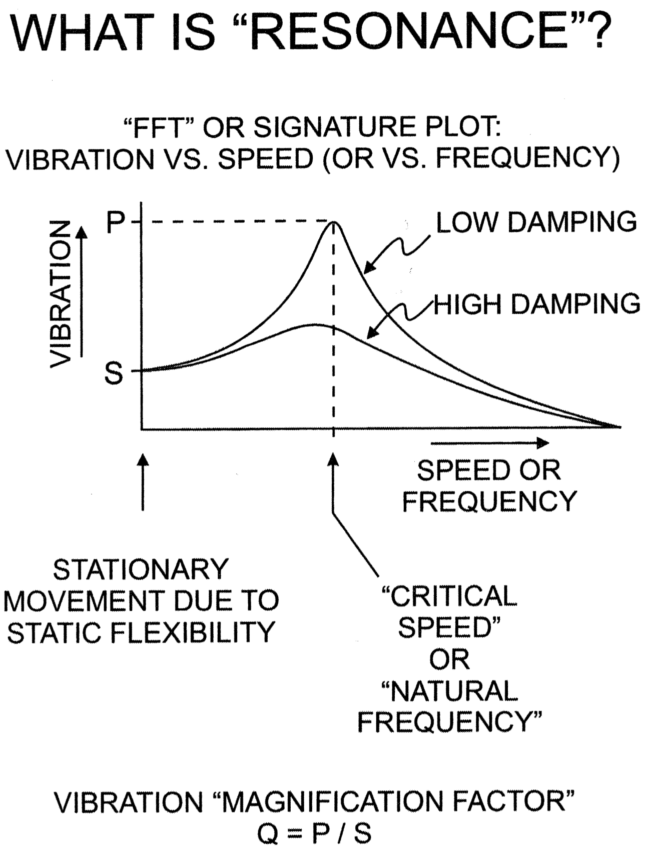


*Figure 1. Vibration Amplification Versus Frequency Plot.*

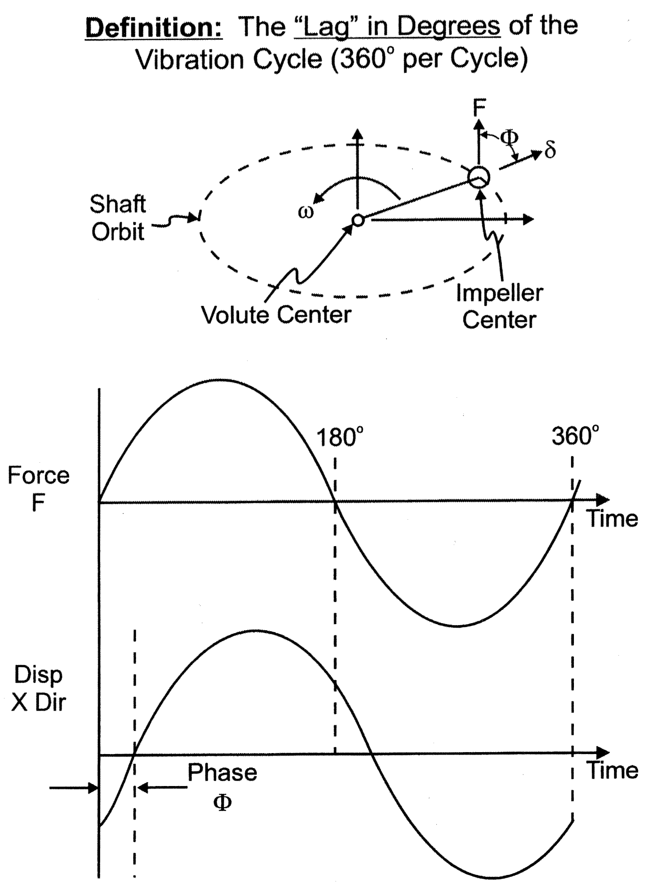


*Figure 2. Phase Angle.*

natural frequency. An example of this is a force being slowly applied to a spring. Alternately, a phase angle of 180 degrees means that the force and the vibration due to it act in exactly opposite directions, so that they are perfectly out of step with each other. This occurs at very high frequencies, above the natural frequency.

Phase angle is important because it can be used, together with peaks in vibration frequency field data, to positively identify natural frequencies, as opposed to excessive excitation forces. This is necessary in order to determine what steps should be taken to solve a large number of vibration problems. Phase angle is also important in recognizing and solving rotordynamic instability problems that typically require different solutions than resonance or excessive oscillating force problems.

## MANUAL VIBRATION ANALYSIS

For certain pumps, particularly single stage pumps, rotordynamic analysis can be simplified without significant loss of accuracy. This allows manual methods, such as mass-on-spring or beam formulas, to be used. For example, for single stage double suction pumps, simply supported beam calculations can be used to determine natural frequencies and mode shapes. Other useful simplified models are a cantilevered beam with a mass at the end to represent a single stage end-suction pump, and a simply supported beam on an elastic foundation to represent a flexible shaft multistage pump with stiffness (as explained below) at each wearing ring, interstage bushing, and the thrust balance device. A good reference for these and other models is the handbook by Blevins (1984).

An example of how to apply these formulas will now be given for the case of a single stage double suction pump. If the impeller mass is M, the mass of the shaft is $M_s$, the shaft length and moment of inertia ($= \pi D^4/64$) are L and I, respectively, and E is Young's Modulus of Elasticity, then the lowest natural frequency (the "reed" mode) in cycles per minute is:

$$f_{nl} = (120/\pi)[(3EI) / \{L^3(M+0.49M_s)\}]^{1/2} \tag{2}$$

If the eccentricity of the impeller relative to the bearing rotational centerline is e, then the unbalance force is simply:

$$F_{ub} = Me\omega^2 / g_c \tag{3}$$

and the amount of vibration displacement expected at the impeller wearing rings is:

$$\delta = (F_{ub}^3 * L) / (48EI) \tag{4}$$

For hydraulic radial forces, $F_{ub}$ may be replaced by the hydraulic radial force $F_r$ (estimated by the manufacturer, or estimated worst case as 0.36 times the axial width of the water passage at the OD, times the outside diameter, times the difference between the discharge and suction pressures) to determine δ.

However, the degree to which hydraulic forces occur is a complicated and design-specific matter. Besides issues of impeller vane design, Makay (1980) introduced the concept of vibration due to axial pressure pulsations on the surfaces of the impeller shrouds, due to large clearance at "Gap A" (the minimum clearance between the rotating shrouds and stationary casing walls), and (sometimes) dramatic increases in impeller vibrations due to vane pass pulsations when there is an excessively small clearance at "Gap B" (impeller vane versus diffuser or volute vane gap of less than four percent to six percent of the impeller diameter).

The most accurate means to determine rotor natural frequencies, and the only reliable way to assess overall rotor stability, is with a complete rotordynamics analysis. This is because there are complications in analyzing rotors versus performing similar analysis on stationary structures. For example, regardless of the bearing type used in a particular pump, the reaction forces that occur in the bearings in response to vibration and even static loads are not straightforward. Besides the direct restraining force of each bearing that acts exactly opposite to rotor motion, there are also other important forces that act perpendicular to this motion, namely damping and "cross-coupling" (bearing reaction "spring" force that acts perpendicular to the shaft motion). These can be as large as or larger than the direct force, allowing them to dominate the vibration. Unlike other types of manual or computer (e.g., most FEA) vibration analysis, a rotordynamic analysis includes the effects of these forces perpendicular to the motion, allows the dependency of reaction forces on speed to be modelled, and includes impeller, balance disk, and coupling gyroscopic effects.

In addition to exciting fluid forces due to the action of the impellers, and reactive fluid forces occurring in the bearing, strong fluid forces can occur in the pump "annular seals," i.e., the wear rings, interstage bushings, and balancing device clearance gaps. The most important aspect of these forces in industrial pumps is generally called the "Lomakin effect." In this effect, each annular seal acts to some extent as a bearing, usually tending to stiffen the rotor support and raise the natural frequencies to higher values, at least until the clearances wear. However, as pointed out by the work of Childs (1982), the expected "stiffening" can actually become "destiffening" if enough fluid swirl is present at an annular seal inlet, and other effects such as annular seal "effective mass" and cross-coupling should be accounted for as well. The Lomakin effect is particularly strong in multistage pumps, because multistage rotors are relatively long and flexible. Since the annular seals are primarily in the central portion of the rotor, where they exercise considerable leverage on the first bending mode of the rotor, the contribution of seal stiffness to the rotor support can be comparable to the rotor stiffness itself.

## TORSIONAL ANALYSIS

Lateral rotordynamics can often be analyzed without including other pumping system components such as the driver, pump casing, pedestal, foundation, or piping. However, torsional vibration of the pump shaft and all types of vibration of the pump stationary structure are system-dependent, because the vibration natural frequencies and mode shapes will change significantly depending on the mass, stiffness, and damping of components other than those included within the pump itself.

Although torsional vibration problems are not common in pumps, complex pump/driver trains do experience torsional vibration problems. This can be checked by calculation of the first several torsional critical speeds and of the forced vibration response of the system, due to excitations during startup transients, steady running, trip, and motor control transients. The forced response should be in terms of the sum of the stationary plus oscillating shear stress in the most highly stressed element of the drivetrain, usually the minimum shaft diameter.

Generally, calculation of the first two torsional modes is sufficient to cover the range of potentially significant resonances. To estimate these, the pump/driver rotor system must be modelled in terms of at least three bodies: the pump shaft assembly, the coupling hubs and spacer, and the driver rotor. If a gearbox is involved, each gear must be separately accounted for in terms of both inertia and gear ratio. If a flexible coupling is used, the coupling stiffness will generally be similar to the shaft stiffnesses, and must be included in the analysis. Estimates of coupling torsional stiffness are listed in coupling catalogs. Usually, a range of stiffness is available for a given coupling size, so that troublesome torsional resonances can be detuned without changing the rest of the system.

Care should be exercised with systems involving variable frequency drives (VFDs), especially for vertical pumps, which are structurally flexible and tend to have more natural frequencies close to or even below the peak operating speed. Besides sweeping the excitation frequencies through a large range and increasing the chance of matching running speed to one of these natural frequencies, VFD controllers provide new excitations at various "control pulse" multiples of the motor running speed, commonly at 6× and 12×, and often at whole-fraction submultiples as well. The control's manufacturer can predict these frequencies and their associated torque oscillation strengths.

## VIBRATION TROUBLESHOOTING PROCEDURES

The most common types of vibration tests fall into two categories:

- *Natural-excitation signature analysis tests*—Running the pump at a steady operating condition of interest, and collecting data from pairs of transducers at important locations to determine vibration amplitude versus frequency plot spectrum "signatures" and component "orbits" (position versus time traces in a plane perpendicular to the shaft axis), due to forces occurring naturally within the pumping system.
- *Shutdown and startup transients, using "peak average" plotting*—In these tests, if possible, run the pump up and down in speed slowly, while documenting frequency spectrum signature changes due to forced response and instabilities occurring in the pumping system throughout the transient. This is similar to cascade plotting, but is accomplished with a single spectrum, with the aid of a technique available on most analyzers called "peak averaging." Peak averaging retains the maximum vibration amplitude value attained at any given frequency during the period over which the "averaging" is done.

In addition to these common tests, experimental modal analysis "EMA" has been found to provide information that is key to understanding and eliminating vibration problems, particularly if these problems are a result of resonance.

## EXPERIMENTAL MODAL ANALYSIS OR "RAP" TESTING

Experimental modal analysis (EMA) is a method of vibration testing in which a known force (constant at all frequencies within the test range) is put into a pump, and the pump's vibration response, due exclusively to this force, is observed and analyzed. EMA can determine the natural frequencies of combined casing, piping, and supporting structure, and if special data collection procedures are used, EMA can also determine the rotor natural frequencies at the pump operating conditions (Marscher, 1986). Separately, the frequencies of strong excitation forces within the pump can be determined by comparing the vibration versus frequency spectrum of the pump's EMA artificial force response with the signature analysis spectrum of the pump's response to the naturally occurring forces from within the pump and from its attached system and environment.

The main tools required to do EMA are a two channel FFT spectrum analyzer, a microcomputer with special software, a set of vibration response probes such as accelerometers or proximity probes, and an impact hammer designed to spread its force over a frequency range that covers the test range, as if the results of a number of shaker tests were combined.

The impact hammer has an accelerometer in its head that is calibrated to indicate the force being applied. During an EMA test, the signal from the hammer input force accelerometer is sent to one channel of the spectrum analyzer, and the signal from the vibration response probe is sent to the second channel. Dividing, at each frequency, the second channel by the first channel gives the "frequency response function" (FRF) of the pump and its attached system. The peaks of the FRF are the natural frequencies, and the width and height of the peaks indicate the damping of each natural frequency, as discussed by Ewins (1984).

As discussed by Marscher (1986), cumulative time averaging may be used in this technique to statistically reduce the amount of vibration response signal due to undocumented residual unbalance, misalignment, and hydraulic forces, relative to that due to the known artificial excitation force produced by the instrumented impact hammer. Previously, determination of natural frequencies in the presence of running vibrations has been a problem for modal analysis, limiting its practical use to stationary, nonoperating machines in quiet environments. The new method may be applied to machines at any operating speed and load.

There are several advantages to using impact modal analysis methods, rather than the more traditional vibration test methods, such as "shaker testing," at one frequency at a time. A typical EMA test to determine natural frequency locations throughout the frequency range of interest takes about two minutes, compared with about two hours for a comparable shaker test. One hundred or more such tests are necessary to solve many difficult types of field vibration problems. Therefore, it is practical for EMA to sort through a complicated modal test database consisting of FRF plots of response vibrations at many locations, due to hitting at a chosen location representative of where a significant exciting force might operate. The result of this sorting is accurate prediction of the frequency and damping of each natural frequency within the range of the test, and the ability to create moving "cartoons" of the vibration "mode shape." In some EMA computer programs, this information can also be used to automatically predict the best locations for added masses, dampers, or stiffeners to solve the vibration problem associated with a given mode.

In performing vibration troubleshooting, generalized charts matching symptoms to possible causes can be useful for many typical or simple problems. However, do not rely too heavily on such lists, especially if their initial application does not lead to immediate resolution of the problem. Persistent pump vibration problems are usually due to an unexpected combination of factors, some of which are specific to the particular pumping system, like mechanical or acoustical piping resonances, or hot running misalignment of the pump/driver due to thermal distortions of the piping or baseplate.

## SPECIFIC GUIDELINES FOR VIBRATION MEASUREMENTS

The following measurements are suggested as a minimum for predictive maintenance or vibration troubleshooting of any style pump:

- What the vibration level is on both bearing housings on the pump, and on the pump-side (i.e., "inboard") bearing housing on the driver in the vertical, horizontal, and axial directions.

- How hydraulic performance compares to design. In other words, for a given speed and capacity (i.e., flowrate), how close is the temperature-compensated head of the pump to the curve supplied by the manufacturer, especially near the design or best efficiency point ("BEP")? Is the head and capacity steady when the operator tries to hold the pump at a constant speed? Is the motor or steam turbine driver required to provide more power than expected?

- What the bearing shell or lubricant exit or sump temperatures are, at least approximately.

- Whether the suction pressure is steady at a given operating point, and well above NPSH requirements.

- Whether unusual noises are present at certain operating conditions, and if so, what their main frequencies are, as picked up by a microphone and fed into the vibration analyzer.

For multistage pumps, particularly, the following measurements are also recommended:

- Vibration and steady position of the shaft relative to the housing near each bearing, using proximity probes permanently installed in each bearing housing to monitor vertical, horizontal, and axial displacement.

- Axial steady or "DC" position of the shaft relative to the housing near the thrust bearing (the axial vibration proximity probe can be used for this).

- What the monitored leakage rates and exit temperatures are in the thrust balancing device (if any) leak-off line and seal coolant feed or lubrication lines (if any).

- Whether any wear particles or pumpage contamination are visible in samples taken of the lubricant on a regular basis.

- What the pump shaft and casing natural frequencies are, and what the vibration response to a unit load near the bearings is at these frequencies, as determined by experimental modal analysis, if possible.

## ALIGNMENT AND PRELOADS

Strictly speaking, "loads" in terms of force and "preloads" from a design and assembly standpoint are not one in the same. However, some engineers prefer the use of "preloads" to identify unidirectional loads that are applied by the stationary structure to a rotating shaft, due to the relative positioning of the two. Two categories of preloads defined in this manner are internal and external. Internal, e.g., bearing preloads deal with forces generated from within the machine and go far beyond the scope of this tutorial. There are various types of external "preloads" that impact the structure or casing of the machine. These include piping loads (forces and moments), and the possibility of a "soft foot." The only type of external preload that is likely to exist on shaft is misalignment. The immediate effect of a preload due to misalignment is to force the shaft into one sector of a bearing.

A good indication of preloads, and their magnitude and direction, can be determined with the use of pairs of proximity probes (90 degrees opposed to each other) close to the bearing, and the use of bearing metal thermocouples. For example, consider a machine such as a turbine generator train, or a high energy pump such as a feedwater pump. In this case, the vibration may be low on one bearing with the temperature high, while the adjacent bearing may have higher vibration and lower temperature. This would be a typical indication of misalignment preload.

The amount of preload may be related directly to the amount of misalignment or coupling type. For example, spring type couplings, such as a diaphragm coupling, exhibit the least amount of preload on a bearing and its supporting structures, while a rigid type coupling will impose the most preload.

Figure 3 depicts several shaft orbits acquired from eddy current probes on a sleeve bearing machine. A circle or ellipse, as shown in the first two orbits, is the norm you expect to see when no unidirectional loading or preloads are present. As you move from left to right in the figure, greater preloads are encountered. The last orbit is where the shaft is located in the bottom of the bearing, due to a large amount of misalignment, and the results may show up as a classic 2× shaft speed (notice that a proximity probe in the vertical direction would detect the occurrence of two highs and two lows per revolution). An elevation in bearing temperatures may also accompany such a scenario. Remember, however, that there are other things that may cause 2× vibration. However, because misalignment occurs perpendicular to the shaft orbit maximum and forces the orbit to flatten, the sensors perceive this as 2× running speed. A steady state preload will cause the shaft to move eccentrically to an eccentric position within the bearing. These type orbits are seen most often in machines with gear type couplings.

Misalignment preloads may also unload a bearing and result in less temperature on one bearing, and such an unloaded bearing may even create an opportunity for an unstable shaft, which may result in large orbit oil whirl, and resultant serious (and deteriorating) rubbing.

External preloads may be due to accidental cold misalignment, or may be indirectly caused by piping loads. In addition, the

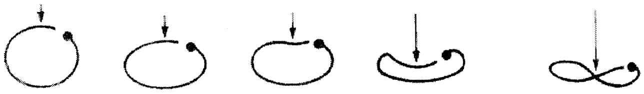

*Figure 3. Effects of Increasing Misalignment Preload.*

presence of a soft foot (whether classic soft foot or induced by excessive piping loads) can also contribute to the problems caused by preloads.

Large pumps with alignment keys pose another problem with piping strain and/or casing deformation in the context of misalignment. There is a choice of where these forces enter and react with the pump. They can enter through the piping and make the pump move relative to the driver, or they can transfer through the keys and the supporting base/pedestal of the pump casing, causing some distortion in the casing (minimal for a properly designed casing, if nozzle loads meet API 610 criteria), but basically maintaining alignment. If reacted at the keys, these forces eventually will be transmitted into the superstructure supporting the pump pedestal, such as the baseplate and grouting of the machine.

Smaller machines with antifriction bearings pose special problems with regard to preloads, because the stiffness of the bearings tends to result in preload, causing excessive bearing internal forces that cannot easily be measured, as opposed to large displacements of the shaft, which can be measured with proximity probes or with dial indicators in terms of cold offset eccentricity. Detection techniques may need to include infrared thermography along with seismic vibration analysis, in order to detect preloads that do not have much effect on shaft eccentricity or orbit, but have a significant effect on alignment and reliability of the machines in terms of shaft fatigue and bearing life.

Machines that utilize a gearbox for speed increasing or decreasing will have preloads associated internally to the machine, which act upon alignment in complex ways while the machine is in operation. Vibration testing, including torsional testing, is generally required to sort out problems in this area. One of the most reliable torsional test techniques uses some kind of strain sensor on the shaft, which transmits its results to ground through slip rings or radio telemetry. However, an expert in gearbox analysis can "demodulate" the vibration readings from the gearbox shaft and housing to determine torsional versus lateral vibration effects.

Often piping strain as it relates to misalignment is overlooked. This is particularly true in the horizontal direction. Minor piping growth in the piping designers eyes can be a major amount in the eyes of rotating machinery personnel. The piping growth due to heat can have a severe impact on misalignment of machines. Some of this misalignment can be accounted for with transient alignment monitoring and corrections. In terms of the resulting residual misalignment, keep in mind that in the process of piping strain impacting alignment, it also impacts wear of parts.

Cold piping strain in the horizontal direction must be accounted for and remedied. Radial and axial pump keys under the casing will not eliminate the adverse effects of these loads, since the resulting forces against the keys may constrain the pump, but add the loads to the casing in a manner that excessively distorts it. Therefore, in the case of misalignment symptoms, it may be found that piping may need to be cut and rewelded from its original installed position.

## SOFT FOOT PROBLEMS

Induced soft foot may never be checked on machines where the driven machine is not the machine to be moved. While machine reliability may be improved with the correction of soft foot problems on the MTBM, additional improvement can usually be realized with checks made on the stationary machine. This is particularly true when pipe strain is involved, such as with a centrifugal pump. Where pumps are concerned, regardless of size, the induced type soft foot will typically cause problems with the stationary-to-rotating component interface, including mechanical seals, bearings, casing rings, etc.

Intermediate to large size machines, such as double suction type pumps, used in the service of condensate booster or feedwater pumps may pose a problem with the flexibility of the feet of the pump. Due to the large sizes of piping connected to these pumps and the generally small size of the casing, the problem with casing flexibility (casing deformation) becomes an issue. "Proper alignment," by the indication that good readings and data may be present, can give one the sense that everything is okay within the components of the driver and driven machine, when, in fact, this may not be the case. Soft foot, in the general sense, has to do with the planar position of all four feet of a machine. In general, soft feet are corrected with shims, eliminating the problem at the feet.

All too often, however, this is checked only on the motor or driver of the machine train. When, and if, the driven machine is checked for a soft foot, this is checked and remedied (if possible) on the driven machine, such as a centrifugal pump. The user may then "open a can of worms." The user in such cases is brought to the realization that less obvious but major problems may have existed all along, and were the most significant root cause of the machine's lack of reliability. The time consumed to remedy such problems may be great, and can create last minute scheduling conflicts during an outage. Plant maintenance should be prepared for this possibility when beginning the task of solving problems with a piping-induced soft foot.

If spring supports (cans) are utilized as piping supports or hangers, then these must be analyzed to determine the correct hot and cold set positions, which can be considerably different from each other. These types of supports should generally be set to accommodate the total load of the piping, which (it must not be forgotten!) includes the weight of the liquid in the piping.

The most opportune time to make soft foot checks is when the unit is coming offline for scheduled outage work. Operations must maintain the piping filled with liquid during this timeframe. Alignment equipment, such as a laser, can be mounted rather quickly as soon as the machine is at rest, and tagged for all safety precautions. An "as found" alignment should be performed at this time. Immediately following should be the soft foot check.

Any keys or foundation pins should be removed and all four feet loosened on the pump, and alignment again checked. Dial indicators mounted in locations around the feet should be monitored.

Using quality alignment equipment and aligning to tight tolerances is allowing mechanics to become much more proficient in alignment tasks. However, it should be remembered that this is only part of a sometimes complex task needed to be performed to achieve complete machine reliability.

## CASE HISTORIES

### *Elimination of Misalignment-Related Reliability Problems*

A steam generator feedwater pump was continually exhibiting problems with wear and rubbing of the throttle bushings and sleeves, which required replacement at unacceptable intervals. The pump rotor exhibited rubbing when rotated by hand during the performance of shaft alignment. Inspection over a period of time revealed rubs in the pump, and higher than normal vibrations according to installed proximity probes. Due to the rubs internal to the pump in the sleeve and bushing area, and since it was not known if a failure could possibly occur during a run between refueling outages, the bushings and sleeves were changed at the periodic interval of every refueling outage.

Beginning in 1989, the pump-to-turbine alignment was monitored during shutdown and cooldown of the machines. The last monitoring task was performed in the middle of 1994. Improvements were gradually seen in the wear of the bushings and sleeves from past "evolutionary" changes in alignment.

Incremental changes to alignment were made over a period of five years to ensure excessive moves were not made that could influence the reliability of the pump and turbine. In part, this incremental approach was due to a new alignment technology being used, and the need for convincing management that these moves would not be detrimental to the machine.

Data taken from the 1994 outage were as follows:

- "As found" alignment, May 4, 1994:
  - Vertical offset = 14 mils
  - Horizontal offset = +2 mils
  - Vertical angularity = +3 mils
  - Horizontal angularity = −3.5 mils

A computer-based laser system was used to capture and record all data. The dimensions were entered into the computer and were saved under the heading "initial data." The initial data file contained information that the computer needed to calculate all movements of the pump relative to the turbine. Under the distances heading, the dimensions were entered as follows:

- LF (left foot) and RF (right foot) specified the feet of the stationary machine
- C1 and C2 were distances to the two coupling flex planes
- BF and FF referred to the front foot and back foot of the MTBM

Figure 4 shows the graphics displayed on the computer at the beginning of data collection. Figures 5 and 6 are the graphics on the computer at the end of data collection. The "as built" alignment was as follows:

- Vertical offset = 10.5 mils
- Horizontal offset = 12.5 mils
- Vertical angularity = 3 mils
- Horizontal angularity = 2.5 mils

The above alignment numbers were plotted on a graph with the MTBM. These data were also plotted against the as-found data and the cold data recorded on the computer.

Comment = 1B FWP COOLDOWN 4-94

*** Distances**
LF_RF (inch) = 90
RF_C1 (inch) = 38
C1_C2 (inch) = 17
C2_FF (inch) = 10
FF_BB (inch) = 42
*** MONITOR SYSTEM DATA (horiz.)**
H_Device_No. (0 bis F) = 0
H_=_Value (mils) = - 34.6
H_| |_ Value (mils) = 16.9
HM_C1 (inch) = 11
HM_P (inch) = 28.75
*** MONITOR SYSTEM DATA (VERT.)**
V_Device_No. (0 bis F) = F
V_=_Value (mils) = 31.5
V_| |_ Value (mils) = - 24.0
VM_C1 (inch) = 4.5
VM_P (inch) = 22

******ALIGN** Coupling Data
Coupl1_Dia (inch) = 10
| |_ Vert. Value (mils) = 0
| |_ Horiz. Value (mils) = 0
==_Vert. Value (mils) = 0
==_ Horiz. Value (mils) = 0
*** Alarm Data**
Par. offset tolerance (mils) = 8
per distance (inch) = 1
*** Save / request parameter**
Cycle time (min) = 1
Saving duration (h) = 1
Data filename (no ext.) = 1B4944
*** Lens constants**
LC_V (inch) = 13.27
LC_H (inch) = 13.19

*Figure 4. Initial Data.*

The machines were monitored via computer data points relative to other possibly relevant parameters, including turbine vacuum, pump inlet and discharge pressures, turbine interstage pressures, stop valve temperature, and suction header temperature. These parameters were plotted against all alignment changes. Figures 7, 8, 9, 10, and 11 depict the surprisingly large changes in alignment with the changes in some of the conditions. For example, Figures 7 and 8 plot changes in vertical and horizontal offset versus turbine vacuum, Figures 9 and 10 plot alignment angularity versus stop valve temperature, and Figure 11 plots angularity versus suction header temperature.

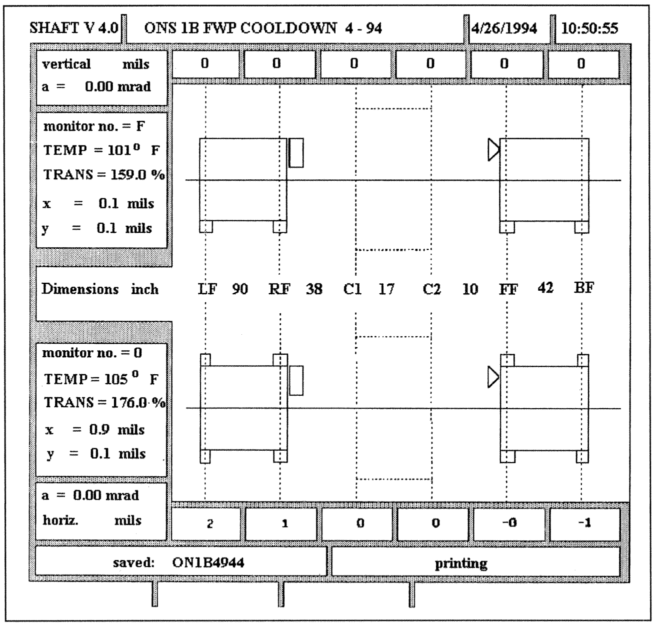


*Figure 5. Screen 1 at End of Data Collection.*

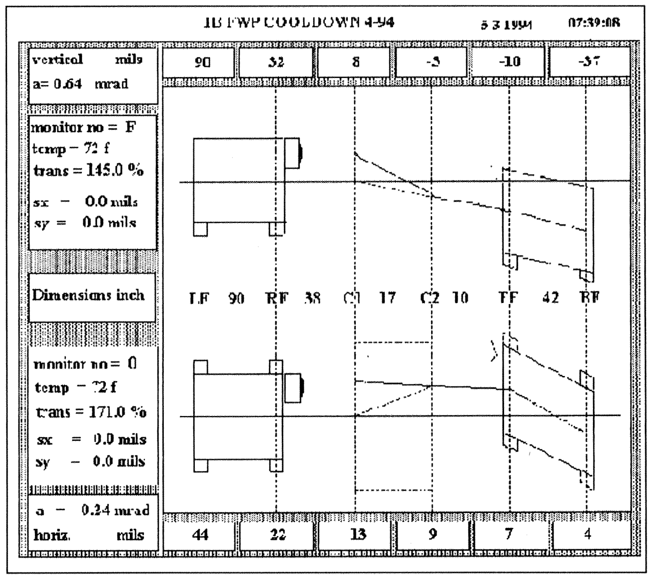


*Figure 6. Screen 2 at End of Data Collection.*

It is obvious from the numbers plotted in Figures 7, 8, 9, 10, and 11 that substantial changes were required in the alignment of this machine from the time of its installation up to the mid-1994 outage. Such changes can occur due to foundation settling, base or pedestal sag due to gradual relaxation of residual stresses, or adjustment of the pumping system (e.g., location of pipe hangers, or changes in configuration of the piping, even at long distances away from the pump).

*Soft Foot Correction*

Piping-induced soft foot coupled with the effects of excessive piping strain, if unremedied, can cause continual problems when a plant is trying to align a machine, let alone achieve satisfactory machinery operation.

When soft foot checks were made on the machines in this example using a laser system, erratic and nonrepeatable readings

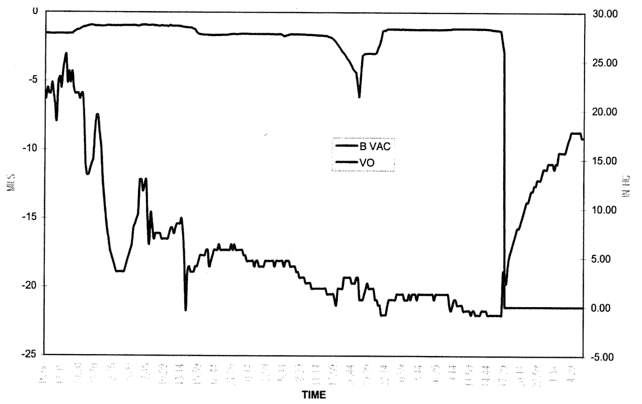


*Figure 7. Vertical Deflection Versus Turbine Vacuum.*

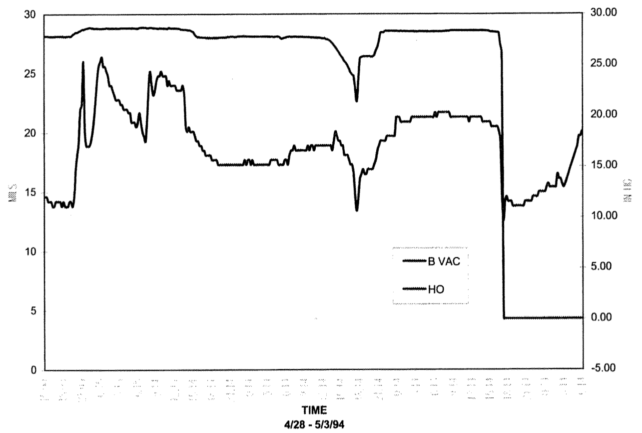


*Figure 8. Horizontal Deflection Versus Turbine Vacuum.*

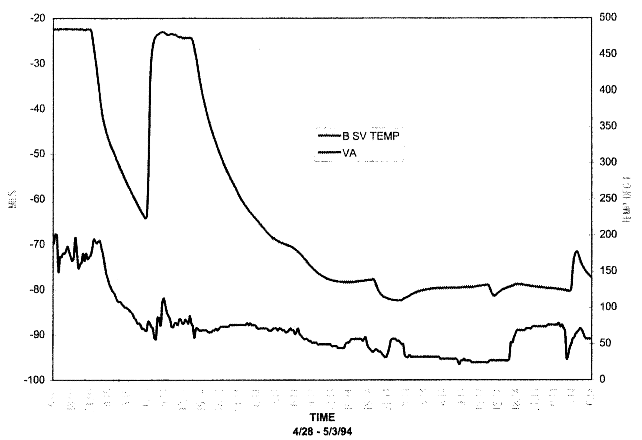


*Figure 9. Vertical Alignment Angularity Versus Stop Valve Temperature.*

were being acquired. Based on these readings and the changes transpiring during alignment, along with the inability of the plant personnel to move the pump easily, the following checks were made. It had been felt for some time that the amount of sleeve and bushing wear and high bearing temperatures in the pumps in question were stemming from excessive external preloads, as discussed previously.

Figure 12 is a graphic of the subject pump showing the hold down feet labeling and the jacking bolt labeling. The jacking bolts were brought into contact with the pump feet. Then the pump, when loosened, lifted from the sole plates by the following amounts:

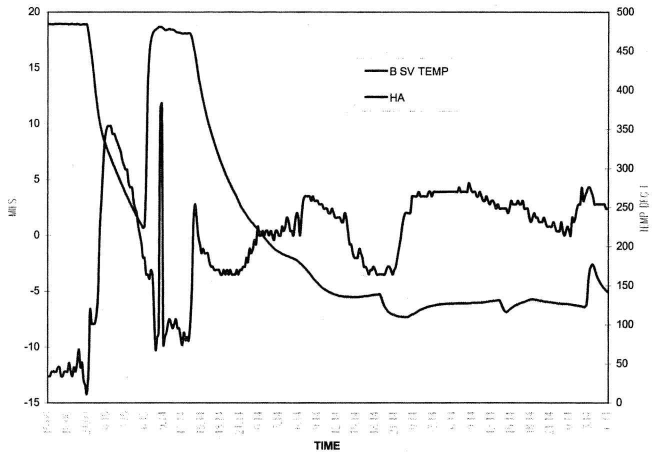


*Figure 10. Horizontal Alignment Angularity Versus Stop Valve Temperature.*

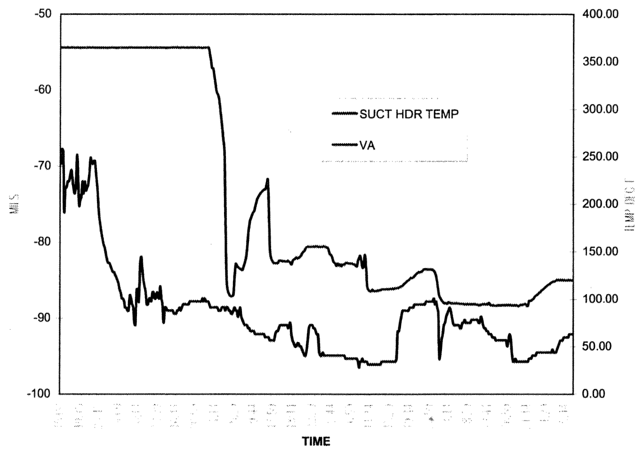


*Figure 11. Angularity Versus Suction Header Temperature.*

- Foot 1, .080 in
- Foot 2, .038 in
- Foot 3, .068 in
- Foot 4, .005 in

As can be seen from the above numbers, *all* feet were soft, due to the severity of the piping induced loads pulling them all up from secure bolted-down contact with the base.

The distance between shafts was measured prior to unbolting the pump, and the dimensions recorded before and after unbolting the pump. The values were:

- Before shaft spacing = 12.021 in
- After shaft spacing = 11.390 in
- Before/After difference = 0.631 in

This table, and the accompanying illustration in Figure 13, reveals the movement of the pump after the holddown bolts and the jacking bolts were loosened.

In this particular case, preparations were made for these checks. Prior to the system being drained, stops were installed in the spring hangers. In some cases, the appropriate stops are attached to the hanger for use when the need arises. In such cases, the stops may not be of the appropriate length to stop the spring. If this is the case, new stops must be cut from stock prior to draining the system. This stop should closely match the filled condition of the piping. In the case of this example, for instance, when the above numbers were found, the system hanger supports required adjustment.

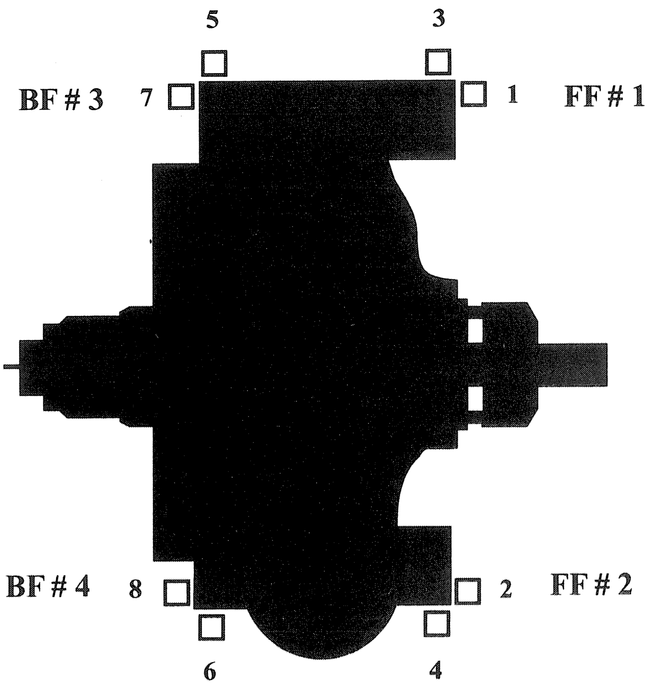


*Figure 12. Pump Foot/Jacking Bolt Labeling for Soft Foot Test.*

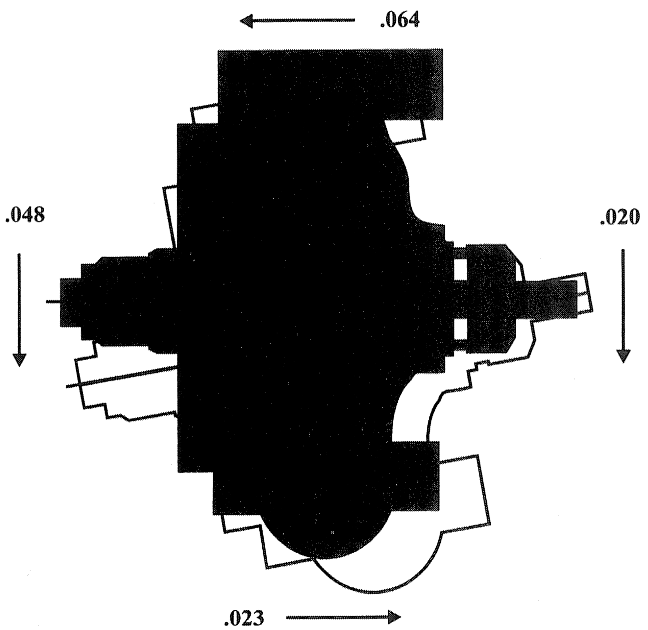


*Figure 13. "Spring" Movement of Pump Due to Soft Foot Condition.*

*Benefit of Up-Front Analysis Prior to Installation*

A series of centrifugal pumps were to be purchased and installed in a petrochemical plant in Taiwan. Prior to purchase, the user contracted to perform a complete mechanical design evaluation, including rotordynamic, torsional critical speed, and rotor forced response analysis. In addition, the pressure-carrying capability of all casings at full and part load were to be assessed. Some of the pumps were driven by motors through variable frequency drives (which have unique and strong torsional excitation harmonics), and some of them were driven by steam turbines. Analysis showed that all aspects of the design were acceptable with sufficient factor of safety, except for the second torsional natural frequency, which was excited by 2× line frequency of the motor (this is usually the strongest torsional excitation frequency in electrical motors), and by the partial admission steam turbine's 2× running speed, which could be easily excited by misalignment. A low-speed-shaft coupling modification was evaluated through "what-if" torsional analysis, modelled with a well known general purpose FEA program. It was found that by changing to a stiffer model of the same coupling (still an off-the-shelf catalog standard), the torsionals were moved up and away from the 2× excitations, without creating any new problems. Subsequent witnessed shop testing, after the equipment had been assembled, verified that the torsional frequencies were within several percent of where the analysis predicted they would be, and that no torsional or rotor bending natural frequencies were causing any critical speeds to occur in the operating speed range. The pumps were installed, and have now operated for nearly 10 years without any mechanical difficulties.

*Identification and Solution of a Complex System Vibration Problem Using Modal Testing*

A major U.S. petroleum refinery had a serious gearbox failure problem, coupled with a severe high-pitched noise in violation of OSHA standards, in some service water pumps. These pumps were driven at variable speed by a steam turbine though a right angle 1:1 gearbox and hollow drive shafting. Many experts from the pump, turbine, and gear manufacturers, and from independent consulting firms, had tried unsuccessfully to use vibration signature testing (and sometimes FEA analysis) to understand and cure the problem over the several years since installation. Replacement of the gearboxes with some carefully built to more stringent tolerances had no effect. It was suspected that the problem involved a torsional critical speed, excited by gear-meshing frequency. However, torsional testing performed by one of the authors found that all rotor system torsional natural frequencies were close to their predicted values, and were not near the unit's single operating speed.

Impact modal testing was performed on all exposed stationary and rotating components, using the cumulative time averaging method referenced in the discussion previously. None of the results indicated the presence of any natural frequencies close to the excited gear meshing frequency, until the four foot long hollow drive shaft was impact tested while it was operating. The surprising test results showed that the hollow shaft, when under torque, had a "bell-mode" almost exactly at the gear meshing frequency. The mode shape of the excited natural frequency was such that the hollow shaft ovalized with very little damping, causing the shaft length to oscillate as the cross-section cyclically ovalized. Subsequent analysis showed that the unexpected axial movement was through the "Poisson effect," which states that as you strain a component in one direction, it automatically deflects at the same time in the perpendicular direction. The driving force was shown by further testing to be the combined torsional and axial load from the bull/pinion gear meshing. The drive shaft was filled with grease to damp out this unusual vibration. The gearbox noise immediately fell a factor of 10, and all gearbox problems ceased.

*Reliability Problem Resolution by Careful Combination of Rotordynamic Analysis with Test*

A Northeastern power plant had experienced chronic boiler feedpump failures for eight years, since the unit involved had been switched from base load to modulated load. The longest that the turbine-driven pump had been able to last between major rotor element overhauls was five months. The worst wear was seen to occur on the inboard side of the pump. The turbine was not being damaged. The pump OEM had decided on the basis of detailed vibration signature testing and subsequent hydraulic analysis that the internals of the pump were not well enough matched to part-load operation, and proposed replacement of the rotor element with a new custom-engineered design, at a very substantial cost.

Although the problem showed some characteristics of a critical speed, both the OEM and the plant were sure that this could not be the problem, because a standard rotordynamics analysis showed that the factor of safety between running speed and the predicted rotor critical speeds was over a factor of two. However, the financial risk associated with having "blind faith" in the hydraulics and rotordynamic analyses was considerable. In terms of OEM compensation for the design, and the plant maintenance personnel and operational costs associated with new design installation, the combined financial exposure of the OEM and the plant was about $350,000. Because of this exposure, one of the authors was called in for a "third party" opinion.

Impact vibration testing using the cumulative time averaging procedure referenced previously quickly determined that one of the rotor critical speeds was far from where it was predicted to be, and, in fact, had dropped into the running speed range. Further testing indicated that this critical speed appeared to be the sole cause of the pump's reliability problems. "What-if" iterations using the OEMs rotordynamic computer model showed that the particular rotor natural frequency value and rotor mode deflection shape could best be explained by improper operation of the driven-end bearing. The bearing was removed and thoroughly inspected, and was found to have a critical clearance far from the intended value because of a drafting mistake on the bearing's drawing, which was carried over each time the bearing was repaired or replaced. Installation of the correctly constructed bearing resulted in the problem rotor critical speed shifting to close to its expected value, well out of the operating speed range. The pump has since run for years without need for overhaul.

*Misalignment Caused by Nozzle Loading*

A large double suction single stage pump, with an impeller diameter of four feet (over one meter) and a running speed of 600 rpm, was designed with close impeller vane/volute tongue clearance to reach an aggressive efficiency level in a facility where energy was at a premium. During installation, it was found that vibration levels got as high as the operating clearances in the wearing rings (25 mils, or 0.6 mm, diametral), with the primary component at running speed. There was no possibility of a resonance in this pump, since both the shaft and the bearing housing natural frequencies were above the 1× and 2× excitations, and the 3× excitation due to suction flow asymmetry, which is common in this style pump. The vane pass frequency of 4200 cpm was far removed from the shaft first and second noncritically damped natural frequencies of 2850 and 19,000 cpm, respectively.

The reason for the high vibration was found to be 35 mils of misalignment at the coupling, due to the hydraulic loads on the pump discharge flange being far in excess of API 610 levels. The 48 inch (1.2 m) discharge had a piping expansion joint at the flange, with no tie-bars in place across the flange to carry the resulting thrust. After removal of the piping forces through a grounded bulkhead bolted to the discharge flange, the pump's large 1× and 2× vibration levels were reduced to acceptable values per API 610.

## CONCLUSIONS

Machinery issues such as the effects of nozzle loads and procedures for checking misalignment or vibration can seem deceptively simple. In reality, the issues often become interrelated forming complex patterns of information that are difficult to decipher. The messages of this tutorial are to:

- Analyze machinery "up front," before installation, and preferably before purchase. If you do not have an inhouse group to do this, hire a third party consultant, or make it part of the bidding process that the manufacturer must perform such analysis for you in a credible manner. However, there are many "ballpark" checks and simple analyses that you, as a nonspecialist, can do for yourself.
- Be very careful about the size of the pump you buy versus what you truly need for your process and its pumping system. Do not buy significantly oversized pumps that then must spend much of the time operating at part load.
- Be very careful in assessing and controlling piping loads. Expansion joints may relieve some thermal expansion, only to result in a huge hydraulic thrust, making the situation worse rather than better.
- In the case of rotordynamics, alignment monitoring, and natural frequency resonance testing, the use of computerized tools are actually generally faster and, therefore, less expensive than older and simpler techniques when you must analyze or troubleshoot multistage pumps. Proper use of the proper tools will give the right answer, first try. Those of you who have spent years trying to track down the cause and eliminate pesky alignment and vibration problems on machinery using previous techniques will appreciate the value of this.

## NOMENCLATURE

| | | |
|---|---|---|
| BEP | = | Best efficiency operating point of the pump |
| C | = | Radial clearance in the sealing gaps (in or mm) |
| c | = | Damping constant (lbf-s/in or N-s/mm) |
| D | = | Shaft diameter (in or mm) |
| E | = | Elastic modulus or Young's modulus (psi or N/mm) |
| EMA | = | Experimental modal analysis |
| F | = | Force (lbf or N) |
| FEM | = | Finite element method |
| FRF | = | Frequency response function |
| f | = | Frequency (cycles per min, cpm, or cycles per sec, Hz) |
| $f_n$ | = | Natural frequency (cycles per min, cpm, or cycles per sec, Hz) |
| $g_c$ | = | Gravitational unit (386 in/s or 9800 mm/s) |
| I | = | Area moment of inertia (in or mm) |
| k | = | Spring constant (lbf/in or N/mm) |
| L | = | Shaft length (in or mm) |
| M | = | Bending moment (in-lbf or N-mm) |
| m | = | Mass (lbm or kg) |
| N | = | Shaft rotational speed (revolutions per min, rpm) |
| t | = | Time (s) |
| V | = | Vibration velocity amplitude, peak (in/s or mm/s) |
| X | = | Vibration displacement amplitude, peak (mils or mm) |
| x | = | Instantaneous vibration displacement from equilibrium (mils or mm) |
| v | = | Instantaneous velocity of vibration (in/sec or mm/s) |
| a | = | Instantaneous acceleration of vibration (in/s or mm/s) |
| $\alpha$ | = | Thermal expansion coefficient |
| $\delta$ | = | Vibration displacement amplitude, peak-to-peak, or shaft deflection (mils or mm) |
| $\Delta$ | = | Shaft bending displacement (mils or mm) |
| $\sigma$ | = | Shaft bending stress (psi or N/mm) |
| $\rho$ | = | Density (lbm/in$^3$ or kg/mm$^3$) |
| $\omega$ | = | Vibrational frequency (radians/s) |
| $\omega_n$ | = | Shaft first bending natural frequency (radians/s) |

## REFERENCES


Blevins, R. D., 1984, *Formulas for Natural Frequency and Mode Shape*, Malabar, Florida: Robert Krieger Publishing Company.

Childs, D., October 1982, "Finite Length Solutions for Rotordynamic Coefficients of Turbulent Annular Seals," ASME/ASLE Lubrication Conference, ASME Paper 82-LUB-42.

Ewins, D. J., 1984, *Modal Testing: Theory and Practice*, Wiley, New York: Research Studies Press.

Fraser, W. H., 1985, "Centrifugal Pump Hydraulic Performance and Diagnostics," *Pump Handbook*, New York, New York: McGraw-Hill.

Makay, E. and Szamody, O., September 1980, "Recommended Design Guidelines for Feedwater Pumps in Large Power Generating Units," EPRI Report CS-1512.

Marks, L. S., Ed., 1916-1951, Baumeister, T., Ed., Seventh Edition, 1967, *Standard Handbook for Mechanical Engineers*, New York, New York: McGraw-Hill Book Company.

Marscher, W. D., December 1986, "Determination of Pump Rotor Critical Speeds During Operation through Use of Modal Analysis," Proceedings ASME 1986 WAM Symposium on Troubleshooting Methods and Technology, Anaheim, California.

Roark, R. J. and Young, W. C., 1975, *Formulas for Stress and Strain*, Fifth Edition, Rainbow-Bridge Book Company.

## BIBLIOGRAPHY

Agostinelli, A., Nobles, D., and Mockridge, C. R., 1960, "An Experimental Investigation of Radial Thrust in Centrifugal Pumps," Journal of Engineering for Power, Transactions of the ASME, *82*.

Allaire, P. E., et al., 1978, "Dynamics of Short Eccentric Plain Seals with High Axial Reynolds Number," Journal of Spacecraft AIAA, *15*, (6).

Anwar, I. and Colsher, R., 1979, "Computerized Time Transient Torsional Analysis of Power Trains," ASME 79-DET-74.

API 610 for Centrifugal Pumps, Eighth Edition, 1995, American Petroleum Institute, Washington D.C.

Arndt, N., August 1988, "Turbomachinery Rotor Forces—Final Report," California Institute of Technology, Division of Engineering and Applied Science, Report No. 249.9.

Baxter, R. L. and Bernhard, D. L., 1967, "Vibration Tolerances for Industry," ASME 67-PET-14.

Black, H. F., 1979, "Effects of Fluid-Filled Clearance Spaces on Centrifugal Pump and Submerged Motor Vibrations," *Proceedings of the Eighth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 29-34.

Blake, M., January 1964, "New Vibration Standards for Maintenance," *Hydrocarbon Processing and Petroleum Refining*.

Bolleter, U., September 1988, "Blade Passage Tones of Centrifugal Pumps," *Vibrations*, *4*, (3), pp. 8-13.

Bowman, D. G., Marscher, W. D., and Reid, S. R., 1990, "Pump Rotor Critical Speeds: Diagnosis and Solutions," *Proceedings of the Seventh International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 73-80.

Chen, Y. N., January 1990, "Rotating Stall in Radial Compressors and Pumps as Rossby Waves Associated with Transient Vortices," Sulzer Technical Review, pp. 51-56.

Childs, D. and Moyer, D., January 1985, "Vibration Characteristics of the HPO Turbopump of the SSME," ASME Journal of Engineering for Gas Turbines and Power, *107*, pp. 152-159.

CRC Lubrication Handbook, 1985, Park Ridge, Illinois: STLE.

Dodd, V. R., 1974, *Total Alignment*, Tulsa, Oklahoma: The Petroleum Publishing Company.

Downham, E., November 1957, "Theory of Shaft Whirling, No.5—The Influence of Plain Bearings on Shaft Whirling," *The Engineer*.

Engeda, A. and Rautenberg, M., April 1989, "On the Flow in a Centrifugal Pump Near Shut-Off Head with Positive and Negative Flows," Eleventh International Conference of the British Pump Manufacturers' Association New Challenges-Where Next? Churchill College, Cambridge.

Franz, R., Acosta, A. J., Brennen, C. E., and Caughey, T. K., September 1990, "The Rotordynamic Forces on a Centrifugal Pump Impeller in the Presence of Cavitation," Transactions of the ASME Journal of Fluids Engineering, *112*, pp. 264-271.

France, D., March 1987, "Rotordynamics Considerations in the Design of High Speed Centrifugal Pumps," Proceedings of Indian Pump Manufacturers Association.

Gopalakrishnan, S., et al., 1982, "Critical Speed in Centrifugal Pumps," ASME 82-GT-277.

Gülich, J. F. and Bolleter, U., April 1992, "Pressure Pulsations in Centrifugal Pumps," Transactions of the ASME Journal of Vibration and Acoustics, *114*, pp. 272-279.

Hancock, W. P., March 1974, "How to Control Pump Vibration," *Hydrocarbon Processing*.

Hergt, P. and Krieger, P., 1970, "Radial Forces in Centrifugal Pumps with Guide Vanes," Proceedings of IMechE, *184*, (3N).

Jaberg, H. and Hergt, P., Fall 1991, "Flow Patterns at Exit of Radial Impellers at Part Load and Their Relation to Head Curve Stability," KSB AG, Frankenthal, Germany, *172*, pp. 213-225.

Jen, C. W. and Marscher, W., January 1990, "Experimental Modal Analysis of Turbomachinery Rotors Using Time-Averaging," Proceedings of the International Modal Analysis Conference, Orlando, Florida.

Karassik, I. J., Krutzsch, W., Fraser, W., and Messina, J., Ed., 1984, *Pump Handbook*, New York, New York: McGraw-Hill, pp. 288-301.

Kikuyama, K., Minemura, K., Hasegawa, Y., Asakura, E., Murakami, M., 1987, "Unsteady Pressure Distributions on the Impeller Blades of a Centrifugal Pump-Impeller Operating Off-Design," 87-GT-144.

Kovats, A., April 1962, "Vibration of Vertical Pumps," ASME Journal of Engineering for Power.

Lees, A. W. and Haines, K. A., October 1978, "Torsional Vibrations of a Boiler Feed Pump," ASME Journal of Mechanical Design, *100*.

Marenco, G., et al., 1982, "Elastic Solution for Vibration Trouble on Vertical Pumps," Energ. Ellettr., Italy, *59*.

Marscher, W. D., February 1986, "The Effect of Fluid Forces at Various Operation Conditions on the Vibrations of Vertical Turbine Pumps," Proceedings of IMechE, Radial Loads and Axial Thrusts on Centrifugal Pumps.

Marscher, W., May 1987, "The Relationship Between Pump Rotor System Tribology and Appropriate Vibration Specifications for Centrifugal Pumps," Proceedings IMechE Third European Congress on Fluid Machinery for the Oil and Petrochemical Industries, The Hague, The Netherlands.

Marscher, W., September 1988, "Subsynchronous Vibration in Boiler Feed Pumps Due to Stable Response to Hydraulic Forces at Part Load," Proceedings of IMechE Conference on Part Load Pumping Control and Behavior, Edinburgh, Scotland.

Marscher, W., October 1989, "Analysis and Test of Multistage Pump Wet Critical Speeds," STLE/ASME Joint Tribology Conference, Ft. Lauderdale, Florida.

Marscher, W., April 1990, "The Effect of Variable Frequency Drives on Vibration Problems in Vertical Pumps," Proceedings of Water and Wastewater Conference, Barcelona, Spain.

Marscher, W., December 1990, "Vibration Test and Analysis of a Barrel Boiler Feed Pump Exhibiting Non-Synchronous Vibration," Proceedings of IMechE Seminar on Vibrations in Centrifugal Pumps, London, England.

Marscher, W., June 1991, "How to Use Impact Testing to Solve Pump Vibration Problems," Proceedings of EPRI Power Plant Pumps Symposium, Tampa, Florida.

Marscher, W. D., November 1993, "Accurate Predictive Maintenance for Rotating Machinery," Proceedings of the ASME Rotating Machinery Conference and Exposition (RO-CON '93), Somerset, New Jersey.

Marscher, W. D., 1993, "Vibration Testing and Predictive Maintenance," Chapter in CRC Lubrication and Tribology Handbook, *3*.

Marscher, W. D., 1993, "Wear and Erosion of Pumps," Chapter in ASM Metals Handbook, *18*.

Massey, I. C., 1985, "Subsynchronous Vibration Problems in High-Speed, Multistage Centrifugal Pumps," *Proceedings of the Fourteenth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 11-16.

Maxwell, A. S., September 1980, "Experience with Use of Vibration Standards," Sixth Machinery Dynamics Seminar, Toronto, NRC Canada.

Meyer, R. J., 1977, "Solve Vertical Pump Vibration Problems," *Hydrocarbon Processing*, *56*.

Miyagawa, K., Mutaguchi, K., Kanki, H., Iwasaki, Y., Sakamoto, A., Fujiki, S. Terasaki, A., and Furuya, S., April 1992, "An Experimental Investigation of Fluid Exciting Force on a High Head Pump-Turbine Runner," ISROMAC-4, Honolulu, Hawaii.

Ohashi, H., Sakurai, A., and Nishihama, J., 1988, "Influence of Impeller and Diffuser Geometries on the Lateral Fluid Forces of Whirling Centrifugal Impeller," Texas A&M University Instability Conference, College Station, Texas, N89-22907, pp. 285-305.

Pace, S. E., Florjancic, S. S., and Bolleter, U., 1986, "Rotordynamic Developments for High Speed Multistage Pumps," *Proceedings of the Third International Pump Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 45-54.

Rathbone, T., November 1939., "Vibration Tolerance," *Power Plant Engineering*, *43*.

Reichert, Hall, and Woods, 1970, *Vibrations of Soils and Foundations*, New York, New York: Prentice- Hall.

Rosemann, W., 1965, "Experimental Investigations of Hydrodynamically Induced Shaft Forces with a Three Bladed Inducer," and discussion by Grindell, A. G., Proceedings of Symposium on Cavitation in Fluid Machinery, ASME, pp. 41-50 (page 47+).

Sakamoto, A., Iwasaki, Y., Kanki, H., Baba, K., Ochi, M., Fujiki, S., Terrasaki, A., Furuya, S., and Taguchi, T., April 1992, "A Study on Vibration Characteristics of a High Head Pump Turbine Runner," ISROMAC-4, Honolulu, Hawaii.

Sanks, R. L., 1989, *Pumping Station Design*, Butterworth-Heinemann.

Shadley, J. R., Wilson, B. L., and Dorney, M. S., April 1992, "Unstable Self-Excitation of Torsional Vibration in AC Induction Motor Driven Rotational Systems," Transactions of the ASME Journal of Vibration and Acoustics, *114*, pp. 226-231.

Smith, J. D., 1983, *Gears and Their Vibration*, Marcel Dekker Publishing.

Spettel, T., April 1985, "The Dynamics of a Vertical Pump," *Orbit Magazine*, Bentley Nevada Corporation.

Soundranayagam, S., "Unsteady Operation of Axial Impellers at Part Load Conditions," Indian Institute of Science.

Tsujimoto, Y. and Acosta, A., September 1987, "Theoretical Study of Impeller and/or Vaneless Diffuser Attributed Rotating Stalls and Their Effects on Whirling Instability of a Centrifugal Impeller," Work Group on the Behaviour of Hydraulic Machinery under Steady Oscillatory Conditions, Lille, France.

Tsujimoto, Y., Acosta, A., and Yoshida, Y., 1988, "A Theoretical Study of Fluid Forces on a Centrifugal Impeller Rotating and Whirling in a Vaned Diffuser," N89-22908, Texas A&M University Instability Conference, College Station, Texas, pp. 307-322.

Walter, T. J., March 1987, "Detection of Incipient Failure in Vertical Pumps," Proceedings of EPRI Pump Symposium.

Yamamoto, T., October 1954, "On the Critical Speeds of a Shaft," Memoirs of the Faculty of Engineering, Nagoya University, Japan, *6*.