AVOIDING FAILURES IN CENTRIFUGAL PUMPS

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ABSTRACT

Pump reliability problems are responsible for a large amount of the maintenance budget and lost-opportunity cost at chemical plants, refineries, and many electric utilities. This tutorial outlines the typical reasons for pump failures, and how they can be avoided in many cases by applying the right kinds of analysis and criteria during the pump selection process. Specifically, important issues include where the pump will operate on its curve (preferably close to best efficiency point (BEP)), how its net positive suction head required (NPSHR) compares to the worst case suction head available, the design of the piping hydraulics in the suction and discharge piping close to the pump, and the manner in which piping nozzle loads will be accommodated. Whether or not the ideal pump has been selected, installation must be performed in a manner that avoids hot misalignment, soft foot issues, and thermal bowing of the casing. Once the pump is installed as part of an overall process or system, it is shown how proper startup and steady operation procedures will avoid binding the pump rotor due to temperature differential within the rotor or casing.

Appropriate types and locations of instrumentation will be discussed, and various other condition monitoring methods and criteria. The most productive troubleshooting test methods will be discussed. The usefulness of detailed vibration testing (especially operating deflection shape plotting) and experimental modal analysis "bump" testing will be illustrated. When and why rotordynamics analysis or finite element analysis might be performed will be discussed, as well as what kinds of information these analyses can provide to an end user that could be critical in making decisions about premature shutdown or permanent modifications that should be scheduled for the next outage. Predictive maintenance test and evaluation procedures will be discussed, and it will be shown how these can be used to diagnose problems in many cases. Some specific case histories will be discussed in the context of typical or particularly problematic situations that plants have faced, and what types of solutions were effective at inexpensively providing a permanent fix.

INTRODUCTION

Centrifugal pumps are generally very reliable pieces of equipment. Pump manufacturers and component suppliers, such as bearing and seal manufacturers, have spent a great deal of research money to continuously improve the performance and mechanical reliability of their equipment. Industrial users of pumps have also contributed to this increase in pump mechanical technology, by wrestling with the problems that are inherent in certain tough applications, particularly applications that require large variations in load or temperature.

This tutorial will begin with a discussion of the major mechanical reliability issues that are faced by today's pump user, as well as the design features, installation issues, and maintenance approaches that have proven useful in minimizing repair costs and downtime. Modern monitoring and troubleshooting methods are also described. Where space constraints do not allow enough detail to be given for a reader to access or implement a described technology, references will be given that provide additional details.

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.

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 that help to determine how to solve such problems if they occur. These approaches and procedures are part of the subject of this tutorial.

AN ENCYCLOPEDIA OF KEY MECHANICAL COMPONENTS AND ISSUES

The major mechanical components of a centrifugal pump, and their functions, are listed below in alphabetical order, with references (shown in parentheses) listed at the end of this paper. A good introduction to the design and application issues associated with many of these components and for various styles and application of centrifugal pumps is provided by Sembler (1999). The primary purpose of many of the components listed below is to perform a hydraulic function, i.e., produce flow and head at an acceptable efficiency. However, these components also affect pump integrity and reliability.

Balance Drums and Thrust Disks

A thrust balancing device, in addition to the thrust bearing, is often used in a multistage pump to help absorb the accumulated axial thrust of the pump impellers. Generally, this thrust is toward the pump suction, and can be tens of thousands of pounds in a high head pump. These devices work by exposing one side of a radial area to the discharge pressure of generally the last stage, while the other side is exposed to suction pressure by connecting it with a balancing leak-off line to the pump suction piping. Thrust balance disks have a larger range of thrust they can absorb than is available in a balance drum, which is of constant diameter from one end to the other, because the disks have an axial "pinch point" near their OD that increases pressure differential across the disk in opposition to the direction of its motion. However, because of the coupling together of force and motion in thrust disks, the disks can be prone to "axial shuttling" instability under pulsating flow conditions, such as might occur during throttling or operating near runout. Also, it is difficult to precisely predict the static pressure in the side recesses in front of the impeller shroud and behind each impeller hub, because the exterior surfaces of the impeller tend to act as a pump themselves in a manner that is strongly affected by impeller discharge flow angle and wear ring or throttle bushing (Childs, 1991). Therefore, although designers try to always keep net impeller versus balance device thrust toward the suction, sometimes there are thrust reversals that can buckle the thin shafts of multistage axially split pumps or cause increased rotor vibration in all pumps (Makay and Szamody, 1980). Sometimes this situation is cured by placing a smaller orifice in the balance leak-off line, or by closing down on the "Gap A" between the impeller hub and shroud OD and the casing opposing sidewalls, which also strongly discourages any tendency for axial shuttling.

Baseplate

The pump baseplate is the interface between the casing feet and the foundation. A baseplate and the foundation have some degree of flexibility, and therefore are a contributing factor in the overall stiffness with which the mass of the pump is "grounded" mechanically to the earth. Therefore, the baseplate and foundation are often a key factor in establishing the so-called "reed" frequencies of a pump, the vibration motion that particularly vertical pumps often exhibit near running speed (Marscher, 1986b, 1990b).

This factor in the installation and qualification of new pumps is often overlooked by civil engineers and mechanical contractors when they design and construct a new or revised pump installation. Generally, in such instances the engineer or contractor is trained to be concerned about the structural strength of the floor being adequate to support the weight of the equipment, but does not realize that vertical strength is not at all necessarily the equivalent of bending stiffness. The greater the distance of the pump center of mass and/or radius of gyration (with respect to reed motion) relative to the "footprint" of the pump baseplate, the more important this effect is likely to be. This is especially true for variable speed operation, where in vertical pumps it even becomes likely that a reed critical frequency will be excited by normally acceptable values of imbalance and misalignment, acting at 1× running speed. This issue is also more likely to be a problem in buildings with large expansive floors, which have numerous columns and concrete subfloor beams to support the equipment weight, but have relatively low local bending stiffness because such bending stiffness is not needed to vertically support the pump. In addition to new plant construction, a situation where excessive bending flexibility has plagued the final installation is the upward rerating of existing water stations where existing floors are required to take on increased sizes of pumps.

Another important aspect of baseplates and their stiffness involves the ability of horizontal pumps to resist twisting deflections in either horizontal or vertical planes due to forces and moments enforced on suction and discharge nozzles by the piping, or by hydraulic forces on unrestrained flexible expansion joints. Such twisting can cause substantial misalignments between the pump shaft and the shaft of its driver due to baseplate contortion. This can be a problem even if the pump and motor are mounted on a common baseplate, if the baseplate is too flexible and not fully grouted in place.

Bearings

Bearings are the primary support that suspends the rotor system, and that carries the loads that the functioning of the impellers and the other fluid components impose on the rotor. This includes, but is not limited to, weight, imbalance, misalignment, and volute or diffuser side loading and residual axial thrust. Bearings may be either fluid film or rolling element (also called antifriction) bearings. Fluid film bearings include the inexpensive but instability-prone plain journal bearings, and contoured bore (in order to enhance stiffness and/or stability) bearings such as pressure dam or multiaxial groove bearings, as well as the very stable tilting pad bearings. Rolling element bearings include cylindrical roller bearings (carry high radial load but little axial load), ball bearings (primarily for limited radial load, but in angular contact configuration can take significant thrust load at the expense of some radial load capability). In most industrial pumps, fluid film bearings are lubricated by ISO 32 (SAE 10) oil circulated by a lubrication system (Patel and Coppins, 2001), and rolling element bearings are lubricated by water resistant grease (Marscher, 1999). Studies have shown that bearings are directly or indirectly (e.g., because of seal failure or inadequate lubrication) responsible for about half of all pump premature shutdowns.

Various alternative bearings have been suggested in recent years. The magnetic bearing supports a rotor strictly through an "active" magnetic field. "Active" means that the strength and direction of the field are purposely varied based on feedback from some shaft proximity probes, which is necessary since passive magnets are too weak by themselves to support industrial pump shafts. It was thought by many, including the author, that these bearings would become very popular in certain services, especially hermetic (e.g., canned pump and motor) services or services in which rotordynamic response or stability had been a historical problem. Unfortunately, magnetic bearings have a poor load capacity (at most about 200 psi specific load capacity, and more realistically about 100 psi), and users have not been convinced that they have demonstrated consistently reliable operation in the face of electrical faults or power outages.

Hydrostatic fluid bearings are different from classical hydrodynamic bearings in that they rely on very high pressure being supplied to each of a series of circumferentially-spaced pockets. These pockets intentionally leak around the periphery surrounding each pocket to a drain, and this leakage rate increases as the clearance increases between the bearing stationary pocket and the shaft. The increased leakage on the opening side versus the closing side of the bearing gap allows the pressure on the opening side to quickly bleed-down, so that the net pressure differential around the shaft acts to keep the shaft centered. In addition to this "passive" (i.e., needs no control) type of bearing support, hydrostatic or mixed hydrostatic/hydrodynamic (i.e., designed to incorporate some hydrostatic with some traditional "pressure wedge" fluid film bearings) have many of the same controllability advantages of magnetic bearings (or disadvantages, depending on your point of view—if you can control it, you can miscontrol it), if a shaft position feedback control or "servo" is used to modify the amount level of high pressure fed to each pocket. This enables them to potentially resist and even control rotor excursions and instabilities. Furthermore, they can have their pocket-feed pressure cranked-up to the point that the bearing load capacity is quite high. They were placed in Canadian deuterium uranium (CANDU) reactor coolant pumps many years ago, and performed well in that service from a mechanical performance and reliability point of view. However, there was a significant price in the CANDU application, in that it was reported that roughly 5 percent performance penalty had to be absorbed in order for the bearing to operate, because of the large amount of high pressure leakage that takes place out of the pocket sealing walls, even at as-new very close clearance settings (order of 1/3 to 1/2 mil per inch of shaft diameter). In any service in which hydrostatic or mixed hydrostatic/hydrodynamic bearings are placed, a careful analysis should be made of the increased lubricant flow that will be required, and the cost of flowing this much high pressure fluid versus the overall operating cost of the pumping system.

Bearing Housings

Bearing housings provide structural support for the bearings, and their stiffness must be accounted for in series with the bearing stiffness in determining the amount of effective support stiffness that the bearing provides to the rotor. The most common vibration problems with horizontal pumps (other than imbalance and alignment problems, which account for 90 percent of all vibration spec violations) involve either rotordynamic motion, or the cantilevered (i.e., like a reed bending) flexing of the bearing housing by itself or in combination with swaying of the pump casing (Marscher, 1990a). An effective fix for a bearing housing that is too flexible is to add stiffening gussets between the exterior housing sidewalls and the flange to the casing. In the case of bearing housings with “drip pockets” that bypass the stuffing box area, as is typical, if the drip pockets are open across the top half then the stiffness of the bearing housing attachment is greatly reduced, especially in the vertical direction. In addition, this design allows the bearing housing to “roll” under horizontal side loading from the shaft. A valid fix for such a situation is to design a stiffly attached cover or bridge that forms a “360 degree” bearing housing attachment, and such a design upgrade is available from some manufacturers. The author has used finite element analysis to accurately and inexpensively model proposed stiffening arrangements, to predict their effectiveness in advance, and to explore alternate solutions. In this manner, outage time is effectively used, and the fix is ensured to work without any “back to the drawing board” issues.

Bowl Assembly

Vertical turbine pumps such as deepwell pumps generally consist of several component parts that are not typical of other styles of centrifugal pumps (Marscher, 1990b). One of these is the bowl assembly, which is sometimes called the “liquid end” of the pump. The bowl assembly is the impellers, diffuser vanes, and their casing and bearings. Typically, problems in the bowl assembly have to do with bowl or lineshaft bearing wear, first stage wear ring wear, or cracking of the transition piece that connects the bowl to the column piping or the column piping to the discharge head bottom. Bearing problems and wear ring wear can be due to a rotordynamic problem, especially if the pump is run past its best efficiency point (BEP), near the high flow low head “runout” point, when lack of discharge pressure down-thrust on the impellers allows the shafting to whip at amplitudes up to 10× more than normal. However, more often the root cause is excess turbulence at the suction, severe abrasives in the pumpage, or vibration of the bowl assembly itself. Turbulence at the suction can be caused by vortices in the sump, which can often be cured by sump baffling or redesign (refer to the Hydraulic Institute Standards (HI/ANSI, 2000)), or in mild cases by straightening vanes at the pump inlet, combined with a “nose bearing” in front of the first stage if it does not already have one. Severe abrasives are best handled by rubber bearings (e.g., the cutlass style, as discussed by Marscher, 1986b), although these will quickly fail if the pump ever needs to run dry, for even a brief time. Dry running is best handled by carbon composite bearings (e.g., Graphalloy®), although they are not very abrasive resistant. The author has had success with Graphalloy® sandwiched between two shorter bronze bearings for cases that need combined abrasion and shock resistance along with limited dry running capacity. Wear due to vibration of the bowl casing is generally an issue if one of the combined bowl/column/baseplate modes has the same frequency as either running speed (this is typically an S or W shape, and will wear out some bearings consistently but not others) or a fluid whirling frequency below running speed (in the range of 1/2 running speed for whirl inside the bowl and column, and about 1/4 running speed for whirl outside and around the bowl and column). In such instances, it is best to either add mass (usually a large amount is needed) to the bowl or to use thicker column piping in order to detune the problem natural frequency from the whirl, using finite element analysis ahead of time to ensure that another natural frequency does not get tuned into a problem frequency range in the process.

Breakdown or Throttle Bushings

These are generally grooved annular clearance zones, between a shaft sleeve and a hardened ring. They can be placed in various locations in pumps, depending on their style, such that they provide a partial seal between the discharge pressure of one or more stages and the suction pressure or outside atmosphere. As these wear, they can change the axial thrust versus flow characteristics of the pump, which can cause a sudden change in pump vibration if thrust reversals begin to occur (Makay and Szamody, 1980). This situation requires bushing replacement.

Casing

The pump casing is known as the frame or housing on certain other types of machinery. Its purpose is to provide the ultimate support, relative to ground, of the rotor, and to provide a well-sealed flowpath for the liquid being pumped. Casings can be made of various materials, including cast-iron, bronze, carbon steel, and stainless steel. Especially in larger pumps, casings are nearly always cast. The particular material selected depends upon its strength, cost, and corrosion resistance, and for abrasive services on its erosion resistance. Mechanical issues important to casings are their resistance to distortion and breakage under pressurization and piping nozzle loading, the tendency for multistage casings to take on a “bow” on the order of the internal clearances if warmed up too fast or if one end of the pump is not free to expand (refer to the analysis of this later in this tutorial under *“Casing Thermal Bowing”*), the occurrence of leaks at the gaskets or split flange bolts, and the difficulty of certain types of casings of avoiding loss of interior gasket preload, and associated internal leakage and wire-drawing damage. Relative to the latter two problems, the switch for environmental reasons some years back from asbestos to vegetable fiber and other less resilient gaskets led to flange sealing leaks and diaphragm seal blowouts where there were none before. Carbon fiber gaskets have been found to have a “spring-back” resiliency after initial compression as good as asbestos, and are recommended in these situations.

Casing Nozzle Loading Effects

Common causes of casing nozzle loads are piping length changes or bend curvature changes due to thermal growth, pressurization strain, Bourdon-tube action, unrestrained discharge lines, installation misalignment (mating up of flanges with the help of a come-along), driver/pump misalignment, and soft-foot problems. In turn, excessive pump nozzle loads sourced in the piping can induce excessive nozzle neck stress, casing internal misalignment and rubbing, and most importantly driver/pump misalignment, with possible high vibration and bearing or seal wear. Manufacturers typically will quote the maximum nozzle loads that they allow at each pump flange. The loads are in the form of direct forces (vertical, axial, or horizontal) at the flange face, and moments (torques) with the twisting axis about one of those three directions. API 610 (1995) provides a widely used table of allowable loads for various flange sizes.

Casing Pressurization Effects

Casing pressurization can lead to reliability problems including casing overstress, but more likely problems consist of unexpected split-flange leakage, or internal leakage across an axially-split case diaphragm.

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 (FEA) method on a personal computer (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-coats" to determine stress concentrations (careful—both can be very temperature sensitive and require very good application technique). Dynamic stresses at cracking locations can be determined by strain gauges, or by using analysis calibrated by test results from shaft proximity probes or seismic probes (velocity probes or accelerometers) to determine vibration levels and associated motion of a portion of the casing versus others.

Manual calculations for steady stress fall mostly into three categories:

- Pressure vessel
- Hollow beam
- Heated 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 (2001) 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 about three.

For casings with large sections of wall that are not cylindrical, or that 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 probable effects of stress concentrations are already included.

The hollow beam calculation assumes that the peak nozzle stress (including stress concentration) 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' Standard Handbook for Mechanical Engineers* (1996) or *Roark's Formulas for Stress and Strain* (Young and Budynas, 2001) can be used to determine the stress in cases where loads other than pure bending moments are applied to the nozzle flange, although their resulting number should be multiplied by about three, 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.

Casing Thermal Bowing

Casing problems can occur from other than pressurization effects. One of the most important of these other issues is thermal growth and/or bowing.

A heated 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 inch of length per degree Fahrenheit (about 16 millionths of a meter per meter length 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 6 ft long carbon steel shaft (roughly 2 m) 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 ρ is roughly the diameter of the component, divided by the lower versus upper temperature differential, and further divided by the thermal expansion coefficient α of the component. From this curvature versus the length of the component, the amount of clearance taken up can be estimated either graphically or with arc length formulas. For example, the "humping" h of the casing or rotor (whichever has the temperature differential ΔT top-to-bottom across it) of diameter D and length L, where the humping is relative to the bearing-to-bearing centerline is approximately:

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

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 feed pumps 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.

Cavitation—See *"Impeller"*

Column Piping

Vertical turbine pumps such as deepwell pumps generally consist of several component parts that are not typical of other styles of centrifugal pumps. One of these is the column piping, which connects the bowl assembly (usually below ground) to the discharge head (usually above ground). This piping typically provides the pathway for the discharged liquid to proceed to the discharge head, and provides structural support for the lineshaft and its bearings along the sometimes considerable distance from the bowl to the discharge head. Typical issues with column piping are discussed in the *"Bowl Assembly"* section. As discussed in that section, cracking of the transition piece between the column piping at the bowl, or between the column piping and the lower discharge head, can be due to a structural natural frequency resonance. Another reason can be waterhammer pressure spikes that propagate upstream through the pump, starting at a check valve that suddenly opens or closes. In such a case, place some form of damper on the check valve, or purchase a valve with a slower opening characteristic.

Condition Monitoring Probes and Systems

There are a variety of methods available today, each one being most appropriate for a certain range of problems and not very reliable in the detection of others. Therefore, typically two or more methods should be combined, with the choice of methods depending on the particular pump and its service. Methods to choose from include (in order of popularity) visual inspection, performance tracking, vibration monitoring, oil monitoring analysis, electric current spectrum monitoring, thermography, and acoustic emission analysis. The first two are inexpensive and essential for nearly all pumps, in the author's opinion. The other four methods require someone properly trained in data acquisition and data interpretation, and therefore involve the extra expense of an inhouse expert or outside consulting expertise. Today, automated permanently installed probes and devices compete with probes and readout screens that are integrated into a compact handheld portable unit. In the case of permanently installed proximity probes, make sure that their installation holders, which are often long extensions from the bearing housing, are stiff enough and do not have a natural frequency in the running speed or vane pass frequency range, or you will get false highs in your vibration readings. In the case of portable devices, make sure that the "stinger" or magnet mount is firmly attached to the surface, and has no chance to rattle. Condition monitoring methods are discussed in detail in the second part of this tutorial. More details are provided in the *STLE Tribology & Lubrication Handbook* (Marscher, 1997).

Coupling

The coupling attaches the driver (e.g., motor) shaft to the pump shaft. Couplings may be rigid or flexible. Rigid couplings require very good tolerances and assembly procedures, or else driven-end bearings may be overloaded due to misalignment. Flexible couplings allow a certain amount of misalignment, although vibration increases for imperfect alignment. The vibration increase may be acceptable, and this can be judged by reference to various public vibration acceptance specifications, such as the vibration sections of the Hydraulic Institute/ANSI standards (HI/ANSI, 2000) or API 610 (1995). A general rule-of-thumb is that any rigid coupling or flexible coupling with a single flexible location should be aligned preferably within ± one mil (25 microns), and to no worse than double this amount. For flexible couplings with more than one flexible point (or "location-of-engagement"), the amount of allowed misalignment depends on speed as well as the spacer length (i.e., the distance between the locations of engagement (Crease, 1977)). Beware of some charts produced by well-meaning coupling manufacturers in this regard, however (as pointed out by Blake, 1960; Dodd, 1974; and Kirk, 1983). These charts are based on how much misalignment the coupling can tolerate without damage to itself, which is normally much more than the pump or its driver can tolerate. Once again, refer to public specifications for guidance, both in terms of their specified maximum misalignment, as well as the allowable vibration level that the misalignment subsequently produces. However, a general rule-of-thumb is that no more than 1/4 to 1/2 mil (6 to 12 microns) of misalignment per inch of spacer length should be tolerated.

Magnetic drive couplings are a special category of noncontact coupling that have become very popular in hermetically sealed chemical services. They are discussed under the *"Drivers, Magnetic Drives"* section.

Discharge Head

Vertical turbine pumps such as deepwell pumps generally consist of several component parts that are not typical of other styles of centrifugal pumps. One of these is the discharge head, which serves several duties. The first is that it connects (generally through a right angle turn) the flow coming up the column piping to the discharge piping of the flow system. Second, it mechanically connects the pump to the baseplate, and therefore indirectly to ground. Flexibility of the discharge head must be modeled in detail in any analytical model (such as a finite element model (Marscher, 1990b)) of a vertical turbine pump structure meant to accurately predict natural frequencies and mode shapes.

Discharge Header

The discharge header is a large diameter pipe that accepts the discharge of one or more pumps in a common fashion. Care must be taken that acoustic resonances do not exist because of accidentally tuned lengths of either the header or the feeder lines from the individual pumps (Marscher, 1997).

Discharge Nozzle—See *"Piping Nozzles: Discharge"*

Driver

The driver is generally either an electric motor (usually an induction motor with some limited slip as a function of load) or a steam turbine. The driver is connected to the pump through a drive coupling, and supplies all the energy to cause the pump to produce head and flow.

Electric motors are the most common type of pump driver. These are usually configured to run constant speed, but in situations where variable flow and/or head are required, the most efficient method of accomplishing this is by use of a variable frequency drive, or VFD, also known as an adjustable speed drive, or ASD.

In terms of vibration and particularly torsional response, care should be exercised with systems involving VFDs (Marscher, 1990b). This is especially true for vertical pumps that 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.

Drivers, Magnetic Drives

Magnetic drives (Sembler, 1999), or commonly "mag drives," consist of rare earth magnets mounted on the pump rotor, driven magnetically by a set of electromagnetic stator coils surrounding that portion of the rotor, where there is some form of thin "can" between the rotor and stator. They are limited in the amount of torque they can transmit, but that amount of torque is relatively high for chemical services, and is continually being improved. Reliability issues with these or with the similar technology of a canned motor pump (in that can the motor stator is inside the can, and no metal intervenes between the electrical rotor and stator), are loss of liquid to their process-lubricated bearings due to a starved suction or overheating, and lack of feedback concerning rotor vibration (unless proximity probes are installed, which is rare). There is a performance penalty due to eddy current losses in the rotor, and windage losses between the large diameter rotor and its close-clearance can. In addition, negative stiffness and very large side loads can occur if unexpected swirl occurs in the can area.

Drivers, Steam Turbines

Steam turbine details are beyond the scope of this tutorial. For the purposes here, the main factor to consider is that steam turbine casings and supports typically get much hotter than the pump that they drive, leading to unequal thermal growth, and offset misalignment at the coupling. By offset misalignment, what is meant is that the coupled shafts stay parallel to each other, but are radially displaced. Also, often the two ends of the turbine do not grow the same amount, causing angular misalignment at the coupling, which as the name implies involves the shafts not staying parallel with each other, but developing crossed axes. This can cause faster than normal coupling wear, and can lead to excess vibration and/or overheating of the inboard bearings of the turbine or, more likely, the pump. In such a case, the cold alignment may check out correct, but during fully-heated-up operation the unit could be in severe misalignment. Therefore, hot alignment should be checked, either by immediate inspection as soon as the warmed up unit is turned off (proper warmup can take several days for the foundation of large units to warmup and play its role correctly), or better with the aid of lasers or Dodd bars (Dodd, 1974) installed and zeroed out when the unit was cold-aligned. If the unit needs to run over a range of hot or pressure conditions (remember that the steam turbine alignment can also be affected by exhaust vacuum as well as inlet pressure), then a compromise alignment may be required between the cold and worst-case hot running condition.

Driveshaft

The driveshaft connects the driver, such as an electric motor or steam turbine, to the pump shaft. Not all installations have a separate driveshaft, but when they do it is typically hollow, supported at least at some points by pillow block bearings, and driven through Cardan joints (Hooke's couplings or "U-joints"). The driveshaft typically has its own lateral critical speeds, independent of the driver of the pump, or even whether they are connected, for example during a test. However, the torsional critical speeds of the driveshaft by themselves have no direct technical meaning. Hollow drive shaft lateral natural frequencies can be detuned or damped by injecting Styrofoam® or grease to completely fill the hollow cavity. Torsional natural frequencies typically are best solved by adding a properly tuned flexible or elastomeric coupling in the rotor system, or by adding a flywheel at a high-torsion antinode.

Electric Motors—See *"Driver"*

Expansion Joints—See *"Piping Flexible Couplings and Expansion Joints"*

Feet and Foundation

Pump "feet" are the bosses that provide for attachment, generally by bolting, of the pump to components (such as the baseplate), which eventually lead to the pump assembly being connected mechanically to earth through the floor or "foundation." The feet and their bolts are often more important than the pump casing itself in resisting casing motion due to forces and moments enforced on the pump nozzles by the piping system (Reichert, et al., 1970). Rotational slip of the feet/baseplate interface, for example, has been shown by tests to increase pump/motor coupling misalignment by up to a factor of two in many ANSI and ISO end suction pumps, relative to what would occur if the slip were prevented, e.g., by doweling. The issue of a "soft foot" involves improper shimming of the pump feet to ensure that all four are in firm contact with the pedestal or foundation (or an effectively similar problem can be caused by a grout void or a cracked foundation), and is discussed in the *"Shims"* section.

Filters, Suction Strainers, and Centrifugal Separators or Cyclones

Filters and other methods of straining foreign particles from the flow are generally needed more for some reason in the process than for protection of the pump. The primary pump issue in this regard is being certain the suction pressure does not decrease below the net positive suction head required (NPSHR) due to filter clogging, to avoid head loss and erosion damage due to cavitation. A good practice is to have a maintained gauge at the pump suction flange that continuously displays pump suction pressure. Methods of calculating net positive suction head available (NPSHA) from the suction pressure reading are given in the HI/ANSI Standards (HI/ANSI, 2000).

Flexible Couplings—See "Piping Flexible Couplings" or "Coupling"

Header—See "Discharge Header" or "Suction Header"

Impeller

The impeller is the component that imparts energy to the pumped flow. In a centrifugal pump, the impeller takes axial flow at the pump inlet, and changes its direction into a more or less radial direction, at the same time increasing the head of the flow due to the centrifugal force field that it must pass through. Axial flow and mixed flow pumps create less head per stage, but are generally able to handle much larger flows for a given casing diameter, and impart all or most of the energy to the flow due to the lift effect within the pump vanes, which act as airfoils as in an airplane propeller. The same types of materials are typically used for impellers as for casings, and impellers are generally (but not always) cast because of the low cost of production in large quantities.

It is critically important in pump selection and installation to make an assessment of the impeller head and flow versus what the impeller was optimally designed for. Select a pump that will typically operate close to the best efficiency point 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 (Fraser, 1985). 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 (refer to 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.

The design of the pump suction has significant mechanical ramifications. Both the mechanical connection of the suction flange, as well as the hydraulic design upstream of the pump impeller are of key importance in this regard.

The main issue of hydraulic concern is that sufficient suction static pressure be present to avoid cavitation. Today, it is understood that this means more than merely having sufficient NPSHA to satisfy the 3 percent head drop NPSHR published by the manufacturer. At NPSHA, as much as 3× the NPSHR incipient cavitation (usually inaudible) can cause serious erosion of the suction side of impeller vanes or wear ring exits. Even if NPSHA is high enough to avoid cavitation under normal circumstances, it can still be caused in local sectors by skewed or swirling flow in the inlet pipe, as can be the result of an elbow too close to the pump suction flange, by too severe a reducer near the suction flange, or by vortices in the inlet sump. If the pump is operated too far away from its BEP, the angle of attack of incoming flow on the rotating impeller vane can be different from anticipated by the pump designer at that pump speed, and vane stalling can occur at either the suction or discharge, leading to suction or discharge recirculation, respectively. Such internal flow recirculation can cause cavitation on the pressure-side of vanes, and can cause tornado-like eddies that rotate with the impeller, but at a somewhat slower speed, exciting rotor critical speeds at unexpected frequencies.

If stalling occurs due to operation of the pump at flows too far below the BEP, then it can become coherent and can rotate backward through the impeller at a consistent and unexpected frequency, which can strongly excite a rotor critical speed; this is known as rotating stall, and is sometimes misinterpreted as rotordynamic instability.

Cavitation resistant alloys have been developed in recent years, which typically are also corrosion resistant as well. Various types of duplex stainless steels, including Ferrallium® and Hydralloy®, are in this category. The material microstructure of these materials is analogous to plywood, in that it is made up of layers or "lamellae." This discourages corrosion species or surface flaws due to erosion or cavitation from turning into a crack that propagates deep into the metal, causing gross material loss or even through-cracks.

Keys

As stated for retaining rings, the best method of ensuring expected rotordynamic behavior and of minimizing imbalance is to use heavy press or shrink fits of components such as impellers on the shaft. However, this makes disassembly for maintenance difficult. Therefore, light press fits or slightly loose "close running fits" may be used. In such cases, keys are used to transfer torque and to prevent random rotation (and associated fretting wear) of slip-fit components. When keys are used in pumps running with high surface speeds at the key location, they should be trimmed back to the "half key" configuration flush with the coupling or impeller hub, to avoid introducing an unexpected imbalance into the system.

Lineshafting

Vertical turbine pumps such as deepwell pumps generally consist of several component parts that are not typical of other styles of centrifugal pumps (Marscher, 1986b). One of these is the lineshafting, which connects the impeller(s) in the bowl assembly to the motor coupling, often through tens or even hundreds of feet of vertical distance. Problems with lineshafting generally occur in either the bearings or couplings. In spite of its inherent flexibility, lineshafting with more than three bearings seldom has rotor critical speed or stability problems because of the statistical way in which the nonlinear stiffness of these bearings is constantly tuning to different values at different bearings, making the rotor natural frequencies a "moving target" that cannot get resonant with typical forces such as running speed imbalance.

Lineshaft Bearings

Vertical turbine pumps such as deepwell pumps generally consist of several component parts that are not typical of other styles of centrifugal pumps (Marscher, 1986b). One of these is the lineshaft bearing, which supports the lineshaft laterally relative to the column piping. This bearing typically has a very long L/D ratio and relatively large clearance-to-diameter ratio, and much of the time behaves more like a "bumper" than a classical bearing. Typically the bearing walls are grooved rubber (e.g., cutlass style bearing, which is particulate erosion resistant, but cannot run dry), bronze (okay versus erosion and shock, okay for brief dry running), or carbon composite (runs dry, but is erosion-challenged and not good against shock loads).

Lineshaft Couplings

Vertical turbine pumps such as deepwell pumps generally consist of several component parts that are not typical of other styles of centrifugal pumps (Marscher, 1986b). One of these is the lineshaft coupling, which connects one segment of lineshaft (typically about 10 ft long) with the adjacent segment. Many lineshaft couplings are some kind of slip fit collar, dropped over a key from the top and held in place by gravity. These are relatively flexible, acting like a pinned joint. Some lineshaft couplings are of a screw/union type, and these are much more rigid, able to transmit a bending moment as well as the barrel of the shafting. In the case of the former, shaft vibration measurements along one length may have very little effect or representation of vibration in the adjacent length. In the case of the latter, be sure that the screw threads do not allow the coupling to unscrew itself, which is sometimes an oversight in services where the pump might need to go through reverse rotation, for example during a back-pressure blow-down condition.

Lubrication Systems

Lube systems may be as simple as grease packing, or may provide a continuous supply of cool inlet oil as part of an oil-ring splash or forced lubrication system. Larger, high-power density pumps (e.g., multistage pumps) require the latter. Generally, for forced oil the viscosity is set near ISO 32 or SAE 10 oil specs, and inlet oil temperature is within 20° of 140°F. It is critical to keep water out of the oil, either due to seal failures or due to condensation on the sump walls in humid environments. Water can actually increase the viscosity of an oil, which aids hydrodynamic film development in journal bearings, but will corrode many bearing internal components, and causes oil to lose its tendency to adhere to lubricated surfaces (called "lubricity" in some quarters, a term that has fallen out of favor). Proper filtration is essential (at least to 30 microns, and research has shown substantial reduction in wear if filters as small as 5 microns are used). A filter bypass should be provided with a check valve that opens in case of excessive lube pump pressure buildup due to filter clogging. A prominent research institute's statistics show that most unexpected bearing failures occur either due to oil badly contaminated by water (especially due to seal failures) or foreign particles, or due to a starved lubrication system. Regularly scheduled oil monitoring analysis can often detect faulty situations long before a failure, and should be implemented in critical service pumps on at least a three month (and preferably more frequent) basis. A thorough and up-to-date discussion of pump lube systems and requirements is provided by Patel and Coppins (2001).

Magnetic Drives—See "Drivers, Magnetic Drives"

Motors—See "Drivers"

Pedestal

The pedestal is a generally concrete platform that provides the final custom interface between the foundation (or floor) and the pump baseplate and/or soleplate. It may be used to adjust the height of the pump relative to the floor, or to extend the "footprint" or "purchase" of the pump of a particularly thin floor or flexible foundation. Therefore, a pedestal generally should be included in any structural vibration analysis of a pump (Reichert, et al., 1970), particularly in the case of vertical pumps. Cracking (internal or external) of the pedestal can lead to casing natural frequency changes, such that critical speeds drift into problem frequency ranges (e.g., 1× running speed), unexpectedly increasing vibration, and causing bearing and seal reliability problems. Therefore, inspect and weld-repair pedestals in older units when such vibration occurs.

Piping

Obviously, piping is used to convey the pump fluid to (suction piping) and from (discharge piping) the pump. Suction piping needs to have a relatively large diameter, to minimize the pressure drop to the pump suction from the fluid's original source, and to keep the liquid velocity reasonably low at the pump inlet. High velocity at the inlet reduces static pressure there, making the pump more prone to cavitation. In addition, excessive swirl and/or nonaxisymetric static pressure or flow velocity distributions at the first stage impeller suction (i.e., inlet) must be minimized. Such phenomena lead to local pockets of cavitation and/or varying angles-of-attack of the impeller vanes relative to the incoming flow. The latter can cause vane airfoil stalling with associated "recirculation" cavitation (this can occur on the vane pressure, rather than suction side) and often significantly increased vibration. Piping elbows not part of the pump design that are too close to the inlet, nonsymmetrical or too aggressive pipe reducers, suction isolation or check valves too close to the inlet, and prerotation of fluid at the pump intake in the sump are common causes of these problems. The best policy is to keep disturbances in the suction piping such as reducers at least 10 diameters away from the pump's suction flange (HI/ANSI, 2000).

In suction piping it is also very important to avoid flow interruptions that could cause a lower NPSH than expected in the system design. Make certain that there are no mating flanges that are offset relative to each other or that contain misaligned gaskets.

The discharge piping is generally not as critical as the suction piping. However, valves, piping expanders or reducers, or elbows should be kept five diameters or more away from the pump's discharge nozzle. Even if this rule is followed for the suction and discharge, however, it is still possible to have difficulties with one additional issue: acoustic resonance. Quarter wave or (less frequently) half wave resonances can be established as sound waves reflect back to the pump from sudden openings or obstacles. If the frequency matches one times running speed or vane pass frequency, pressure oscillations can build up like the dribbling of a basketball.

Piping Flexible Couplings and Expansion Joints

Care must be taken concerning how the piping is attached to the pump nozzles. Thermal expansion, Bourdon tube pressure effects, gravity loading of the liquid in the filled pipe, and assembly issues (e.g., using a come-along to strain a pipe nozzle into junction with the pump prior to bolting up the flange) can lead to forces and moments on the pump nozzles that cause excess stress and distortion in the pump casing and its foundation.

A less commonly known source of such problems is unrestrained hydraulic force of the nozzles, especially the pump and/or piping discharge nozzle. 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. Just because the rocket is free to accelerate because it is not tied down while the casing is not free to move, does not mean that the pump casing 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.

Piping Nozzles: Discharge

The discharge nozzle of the pump is the exit for the flow that has been pressurized by the pump. The main reliability issues in this area are the potential mechanical or hydraulic loads that might be imposed on the nozzle by the piping or by an unrestrained expansion joint, respectively, as addressed in the allowable nozzle load table in API 610 (1995).

Piping Nozzles: Suction

The suction nozzle of the pump is where the piping conveying the incoming flow attaches to the casing. It is important that the flow from the system be well-distributed coming into the suction nozzle, and that mechanical loads are not excessive, as applied to the nozzle by hydraulic pressure (for unrestrained nozzle connections across expansion joints) and by piping end forces and moments (where expansion joints are not used). In particular, avoid the use of flow components such as certain reducers, elbows, or strainers in the suction that might decrease suction pressure excessively compared to the vapor pressure at which the liquid will cavitate, or that will cause the flow velocity to be maldistributed or unexpectedly swirling as it enters the nozzle. The amount of suction pressure available in terms of feet of head, absolute, is known as net positive suction head available (NPSHA), and must be kept above a minimum level set by the manufacturer based upon factory tests of suction lift (vacuum) tolerance.

Recirculation—See *"Impeller"*

Recirculation Lines and Valves, Bypass Lines

In many pumping systems, low flow operation is necessary for part of the service, and discharge throttling is necessary. In this case, if the pump is run too far below its operation BEP, suction or discharge recirculation may occur, with resulting excessive vibration or cavitation damage. If the flow is near deadhead conditions (e.g., as can happen in decoking operations), the water around the pump impeller recirculates without leaving the casing, leading to rapid heat-up and possible flashing of the water in the pump. In such cases, a recirculation or "bypass" line can be installed between the pump discharge and the suction line. Make certain to install such lines at least 10 pipe diameters in from the suction flange. Such lines can be left open, but whatever flow occurs in the line at higher system flow conditions adds directly to the pump operating expense. Therefore, a valve is often placed in this line, and is opened only when needed. This is often a modulating valve, which can be opened different amounts depending on the flow required to keep the flow in the pump above the manufacturer-specified minimum continuous flow.

Retaining Rings

As stated for keys, the best method of ensuring expected rotordynamic behavior and of minimizing imbalance is to use heavy press or shrink fits of components such as impellers on the shaft. However, this makes disassembly for maintenance difficult. Therefore, light press fits or slightly loose "close running fits" may be used. In the axial direction, retaining rings are generally used. These are especially important for impeller retention, since impellers generally have their suction side facing the direction from which they were slid onto the shaft, and typical thrust loading of individual impellers tends to push them back in this direction.

Rings—See *"Wear Rings"*

Rotor Systems and Rotordynamics—See *"Shaft or Rotor"*

Seals

Seals prevent the high pressure fluid in a pump from leaking back to the suction or to the external environment. Common sealing mechanisms are mechanical seals of various types (a static and a rotating carbon or ceramic face in near-contact with each other across a typically axial face), packing rope wound between a stator "stuffing box" chamber at the rotating shaft or replaceable sleeve, and serrated "labyrinth" noncontact seals. Seals are considered the Achilles heel of most pumps, and a common source of reliability problems. As discussed by Wilcox (2001), this is not because seal manufacturers do not know their trade, but because sealing between a stationary casing and a rapidly rotating shaft is a very difficult thing to accomplish. Also, many end users accidentally misapply a given seal design, or unknowingly do not take steps necessary to ensure that a proper flow of fresh sufficiently cool liquid, known as "flush," is provided to the seal cavity. In addition, either misalignment or excessive vibration tends to degrade primarily close clearance areas, of which seals are chief.

The surfaces are installed to form a tight sealing gap, such that a hydrodynamic film builds up between the surfaces that generally allows a small and controlled amount of residual leakage. By using spiral grooved surfaces and/or a buffer fluid (such as buffer gas in a so-called dry gas seal) leakage can be reduced to zero, at an increased cost of the seal assembly. Environmental seals are critical to pump operation not only to reduce undesirable emissions, but also to keep nonlubricating fluid away from the bearings. Although bearings are often protected by some other form of seal, such as a lip seal and/or a "flinger" or "slinger" disk, if sprayed pumpage is allowed in the vicinity of the bearing then some of it will generally penetrate into the lubricant. This is a common root cause of bearing failures.

Seal problems have been reported by several studies to be responsible for roughly half all pump reliability complaints. When chronic seal difficulties persist, it is usually not because of an inherently bad seal design or manufacture, but rather because of excessive vibration, misapplication of the particular seal type, or running of the seal in a dry or "flashed" (fluid in the cavity boils) condition. Steps should be taken to ensure that the seal is supplied with cool enough injection fluid at all times that the rotor is in motion, and that this fluid is under sufficient pressure that it can provide adequate flushing flow rate through the stuffing box. Also, although the fluid should be cool, it should not be so cool that it will cause thermal kinking of the shaft or thermal shock of the sealing faces.

Besides packing and mechanical seals, the term "seals" often refers as well to the close clearance area in wear rings, breakdown bushings, and balance devices. These are discussed elsewhere in this tutorial, under subsections named for these components.

Seal Stuffing Box

The stuffing box is at the end of the shaft, and is where the packing or seal is placed that keeps the pumped fluid from leaking out into the atmosphere (Wilcox, 2001). This is the so-called breakdown or environmental seal. If seal failures chronically occur and seal flush fluid is of proper temperature and pressure, then alignment (offset or angular) and/or vibration of the shaft relative to the casing should be carefully studied in the stuffing box area.

Sealing with Mechanical Seal Versus Packing

The environmental seal may be either a mechanical seal or packing. A mechanical seal consists of at least two parallel axial disk surfaces or radial cylindrical surfaces, one rotating and mounted to the rotor and one stationary and mounted to the casing, while packing is conceptually similar to the waxed rope that is wound around a valve stem.

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 feed pumps), 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.

In the case of packing, adjustments are required from time-to-time to limit leakage, as the packing wears. Make sure in such circumstances that the packing is not overtightened, because otherwise rapid wear of the shaft surface may take place, as well as "bluing" and possible permanent loss of shaft local mechanical properties due to frictional heat. Some degree of leakage is necessary in packing to ensure that it stays lubricated.

Some applications that have used packing need to have less leakage due to stricter Environmental Protection Agency (EPA) requirements, for example, or want to avoid the continuous maintenance that is associated with packing (its squeezing "lantern rings" must be retightened periodically). In such cases, replacement with a mechanical seal is typically possible, but care must be taken that the vibration or driver misalignment inherent in the unit is not more than the seal can tolerate. In addition, the extra damping provided to a shaft by packing can suppress critical speeds of the rotor, which suddenly surface when the seal "upgrade" is performed. Impact modal testing of the operating rotor should be performed to investigate this possibility before such upgrades are implemented in critical services (Marscher, 1986a).

Dry gas seals are mechanical seals with grooved faces that act like impellers to prevent the flow of liquid through the sealing interface (Wilcox, 2001). The seals are installed with a clean buffer gas between the seal and the zones external to the pump flowpath. It is this gas that fills the sealing cavity between the seal faces, keeping them separated and nearly friction and wear-free, but without gradual leakage of the sealed liquid. These seals have allowed sealed pump configurations (as opposed to "sealless" pumps as discussed below) to meet zero emission standards, and allow mechanical seals to operate at higher speeds and to deal effectively with more difficult fluids, such as those that tend to flash easily in the seal faces.

Shaft or Rotor

The shaft is the backbone of the pump rotating element, and all rotating components are attached to it. Impellers and other components such as seal sleeves and thrust balance devices may be attached to the shaft by a key in combination with either a close running slip fit or with various degrees of press fit. The advantage to the former is that it makes impeller removal for maintenance purposes easier, but at the expense of uncertain shaft rigidity, and the possibility of impeller/shaft stick-slip. The uncertain rigidity can lead to variation in critical speed values from pump-to-pump of the same type. The stick-slip under the heavy side-loading during operation can cause hysteretic damping at the shaft interface, which if strong enough can cause rotor instability, as is discussed by Allaire and Flack (1978).

Shaft vibration is typically a complicated issue, and is known as "rotordynamics." The most accurate means to determine rotor natural frequencies, and the only reliable way to assess overall rotor stability or the "forced response" due to a given level of imbalance, is with a complete rotordynamics analysis performed with specialty computer programs. This is because there are complications in analyzing rotors versus performing similar analysis on stationary structures (Yamamoto, 1954). 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), and these can be as large 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 modeled, 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 (Black, 1969, 1979; Allaire and Flack, 1978; Childs, 1982, 1988, 1991; Marscher, 1989). The most important aspect of these forces in industrial pumps is generally called the "Lomakin effect" (Lomakin and Bedger, 1957; France, 1987). 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.

Shaft Sleeves

Hardened cylinders are sometimes keyed and/or shrink-fit onto the shaft in areas where metal-to-metal rubbing contact or contact against seal packing might occur. This provides wear resistance, and makes eventual shaft refurbishment easier and less expensive. It is important that sleeves are wear-compatible with the part that opposes them, as identified in galling charts such as the one in Marscher's chapter in the ASM Handbook (Marscher, 1993). In addition, there should be at least a 50 BHN (Brinnell) hardness difference between the parts across the rotating and stationary clearance.

Shims

These are thin strips of metal placed between the surfaces of pump feet and the baseplate or other important load-bearing interfaces. These are used to provide final adjustment of the location of one component (such as the driver) relative to another component (such as the pump casing). What is sometimes lost in this process is the stiffness that was intended across the shimmed interface, because too small a portion of the interface is shimmed or because some small foreign particles get trapped between the shim and one of the shimmed components. This creates a situation known as a "soft foot," which can result in significant unexpected misalignment when the pump is under load (for example because of pump torque-up or nozzle loading), but that goes away when the pump operation ceases. Such an issue is generally detectable by mounting a dial indicator on a mounting rigid to the floor or a neighboring baseplate-to-foundation bolt, and watching for movement as the pump foot bolt is tightened and loosened slightly. In cases of a soft foot, the best solution is always to properly reshim to eliminate its root cause, and to subsequently realign the driver and the pump. Compromise hot operational alignment must be performed if for some reason the soft foot cannot be totally removed (e.g., due to voids in the grouting under the foot rather than improper shimming). Note that in general it is bad practice to stack shims five or more deep, since each stacking interface provides an opportunity for lost stiffness.

Soleplate

The soleplate is an additional metal interface sometimes placed between the baseplate and ground to aid in either providing an effectively wider footprint on a soft foundation, or in facilitating removal of the combined pump and baseplate while still allowing a layer of permanent grout between the pump and foundation. It can affect the pump's dynamic performance if it is too thin to properly transfer bending moment at significantly greater moment arm to the foundation or to discourage cracking of the grout.

Suction Header

The suction header is a large diameter pipe that feeds one or more pump inlet lines in a common fashion. Care must be taken that acoustic resonances do not exist because of accidentally tuned lengths of either the header or the feeder lines to the individual pumps.

Suction Nozzle—See *"Piping Nozzles: Suction"*

Suction Recirculation—See *"Impeller"*

Suction Strainers—See *"Filters"*

Sump

The sump is a reservoir of liquid that is designed as the inlet region of a vertical deepwell or a submersible pump. It is meant to collect the liquid prior to its being pumped, in a manner that discourages air entrainment or vortex formation, and that encourages the uniform flow of relatively low velocity liquid into the pump inlet. Sump design is not intuitive, and must be performed by an expert in order to avoid potentially serious flow maldistribution that could cause severe vibration or cavitation erosion, and air entrainment problems that could lead to severe vibration or accelerated corrosion of pump and piping internals. The Hydraulic Institute recently produced an excellent guide on the design and evaluation of sumps (HI/ANSI, 2000).

Throttle Bushings—See *"Breakdown or Throttle Bushings" or "Balance Drums and Thrust Disks"*

Thrust Collar

The thrust collar is a circular plate that is attached to the shaft by, for example, a nut loading the other side of the ring against a shaft shoulder. It forms the rotating portion of a fluid film thrust bearing. The faces of this ring then interact through a lubricating film with stationary rings on each side of the thrust collar, such that residual axial thrust is transferred to the casing, and eventually to the foundation. Typically, the stationary rings have pivoting pads, which allow the thrust load to be more uniformly distributed, minimizing thrust bearing wear and maximizing load capacity. So-called leading edge grooves have been found beneficial in maximizing the lubricating capability of the oil film on such pads. Rather than using the thrust collar/fluid film bearing concept, some pumps (particularly small ones) use angular contact ball bearings that take axial thrust as well as radial loading. In certain circumstances (e.g., relatively large diameter thrust collars), the "moment coefficient" (how much restoring torque is creating transverse about an axis to the shaft as the shaft is tilted at the thrust collar) associated with the thrust collar has been found to have significant effects on a pump's rotordynamic behavior.

Thrust Disks—See "Balance Drums and Thrust Disks"

Wear Rings

Wear rings, sometimes called impeller neck rings, are the annular cylindrical surfaces that form a tight clearance gap between a "covered" impeller's rotating shroud and the surrounding stationary casing piece. Their main purpose is to provide a noncontacting partial seal that prevents discharge fluid in the vicinity of the impeller exit from leaking back into the lower pressure suction area. A secondary function of wear rings is to provide hydrostatic rotor support through a phenomenon known as "Lomakin effect," as discussed above, which allows the wear rings to act not only as seals, but also as weak but strategically placed bearings. Childs has performed substantial research for a variety of pump applications of pump wear rings and balance drums concerning the rotordynamic coefficients associated with Lomakin effect (e.g., Childs, 1982, 1988, 1991). Lomakin effect can result in roughly a doubling of the first critical speed in multistage high head pumps. Higher critical speeds are also affected, although generally less. A more thorough discussion of Lomakin effect is given in the vibration condition monitoring section below. Wear rings can be either plain (i.e., smooth surface) or grooved. Grooved wear rings are less prone to allowing particles in the pumpage to cause rotor seizure, and also provide a better sealing function. However, grooved rings have less Lomakin effect. If the grooves are deep or wide enough (order of greater than 0.150 inch), then they may virtually eliminate Lomakin effect. Also, as wear rings increase their radial clearance due to rubbing wear or erosion, Lomakin effect is lost in a ratio roughly proportional to the clearance increase. This can cause rotor critical speeds to shift down as elements wear out, possibly causing unexpectedly high resonant vibration at some point before the pump would normally be refurbished due to decreased hydraulic performance.

CONDITION MONITORING

There are a variety of methods available today, each one being most appropriate for a certain range of problems, and not very reliable in the detection of others. Therefore, typically two methods should be combined, with the choice of methods depending on the particular pump and its service. Methods to choose from include visual inspection, performance tracking, vibration monitoring, oil monitoring analysis, electric current spectrum monitoring, thermography, and acoustic emission analysis. The first two are inexpensive and essential for nearly all pumps, in the author's opinion. The other four methods require someone properly trained in data acquisition and data interpretation, and therefore involve the extra expense of an inhouse expert or outside consulting expertise. A more thorough discussion on this subject is provided by Marscher (1997).

Vibration

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 also can occur in the pump stationary structure, especially in vertical pumps. In addition to lateral vibration, vibration can occur in the axial direction or can involve a torsional 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 running near shutoff with resulting recirculation and flashing, for example.
- 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 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's bounces (the ball's natural frequency), but if he is uncoordinated the ball will not bounce very high. You do not want the imbalance force in your pump, which oscillates high and low in a given direction once per revolution, to start "dribbling" your rotor. In other words, you want the natural frequencies of your rotor and bearing housings to be well separated from the frequencies that "dribbling" type forces will occur at, 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 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 20 higher than it would be if the vibration force was steady instead of oscillating. Q depends on the amount of energy absorption, called "damping," which 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 trapped 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 this figure: 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).

Typical exciting forces that cause vibrations in pumps fall into the following categories, with the associated frequencies (cycles of vibration motion per second) shown:

- Balance: 1× running speed N
- Alignment: 2× (sometimes 1× and/or 3×) N
- Impeller vane passing pulsation, especially due to tight "Gap B": number of vanes × N. This is typically stronger in single volute versus twin volute or diffuser pumps.
- Cavitation: A broad frequency range, especially low (below 1× N and very high (20 to 70 kHz)

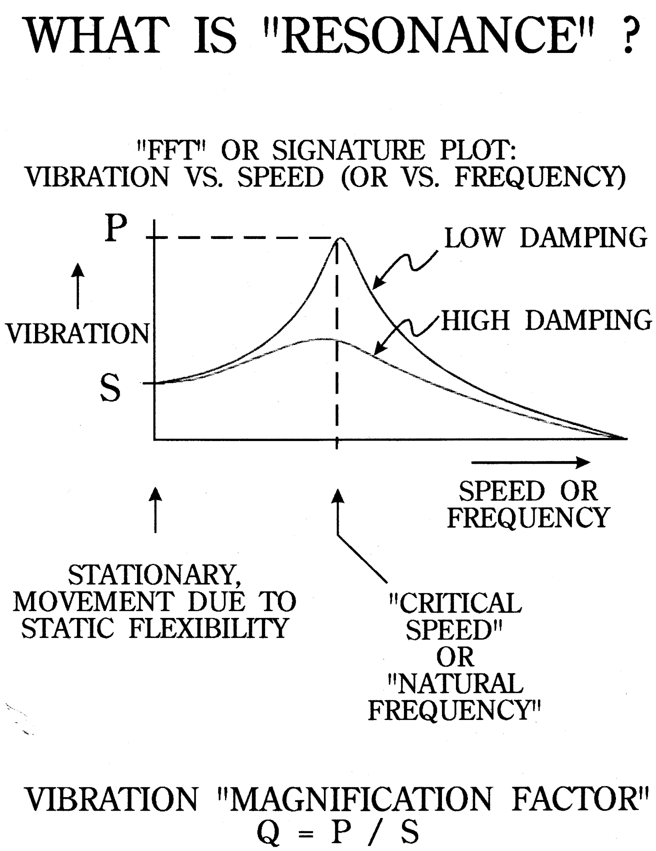

Figure 1. Illustration of Natural Frequency Resonance, and Effects of Damping.

- Rubbing and/or component looseness: Multiples of running speed (1×, 2×, 3×, etc., and often ½× and 1½× running speed)
- Running at flow too far below BEP (typically 50 percent or less): "Rumbling" noise and unexpected frequency peaks around 1× N, i.e., at about ± 25 percent of 1× N
- Resonance of the pump structure natural frequency with 1× N, especially in vertical pumps, and especially with VFD operation: 1× running speed, on top of a "skirt"

Simplified Methods to Calculate Vibration

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 the effects of imbalance on a single stage end pump with the impeller cantilevered relative to the bearings. If the impeller mass is M, the mass of the shaft is M_s, the shaft length and moment of inertia (=π 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:

$$f_{n1} = (60/2\pi)\left[(3EI)/\left\{L^3\left(M + 0.49M_S\right)\right\}\right]^{1/2} \qquad (2)$$

If the eccentricity of the impeller relative to the bearing rotational centerline is e, and the rotational speed is ω rad/s, then the unbalance force is simply:

$$F_{ub} = Me\omega^2 / g_c \qquad (3)$$

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

$$\delta = \left(F_{ub} * L^3\right) / (3EI) \qquad (4)$$

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 and Szamody (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 4 percent to 6 percent of the impeller diameter).

Torsional Vibration

Lateral rotordynamics can often be analyzed without including other pumping system components such as the driver, pump casing, pedestal, foundation, or piping (Gunter and Li, 1978). 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. The pump/driver rotor system must be modeled in terms of at least three bodies: the pump shaft assembly, the coupling hubs and spacer, and the driver rotor. If a gear box 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.

Vibration Monitoring Methods

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 "Bump" Testing

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 exclusively due 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, as well (Marscher, 1986a, 1990a). 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 to 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, as shown by example in Figure 2.

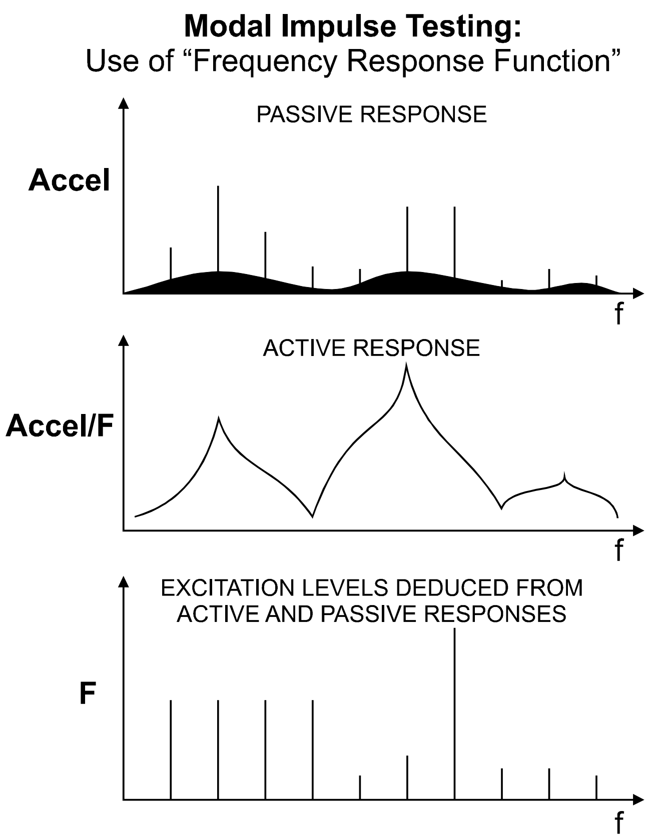

Figure 2. Vibration Frequency Spectrum Dependence of Natural Frequencies and Forces F.

The main tools required to do EMA are a two channel fast Fourier transform (FFT) frequency 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 (1986a), cumulative time averaging is 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 to 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 data base 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 be used also 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 (Marscher, 1997) 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.

Typical instrumentation used for vibration testing are the accelerometer, the velocity probe, and the proximity probe. Accelerometers are by far the most common today, since the modern ones are rugged, and are accurate from about three cycles per second (i.e., 3 Hz) to 10 kHz or more. Velocity probes directly detect vibration velocity (i.e., how fast a bearing housing vibrates instead of how much it vibrates), which is what most modern specifications relate to, but have a lower frequency capability than accelerometers, and accelerometer outputs can be electronically integrated to change them into velocity readings. Proximity probes are mounted to a bearing housing typically, and measure the dynamic shaft displacement (i.e., direct movement) relative to the housing. They are obviously the most sensitive measurement to detect potential rubs in the bearing, but are more difficult to install (elaborate housings and wiring must be installed, whereas accelerometers can be attached with a screw to a tapped hole or even temporarily attached with a securely placed magnet). Keep in mind that all three vibration measurement forms—acceleration, velocity, or displacement—are just three ways of describing the same motion.

Whatever sensor is used, there are decisions to be made about how its output is going to be measured and interpreted. Years ago, the only way to look at the electrical voltage signals (which are proportional to the vibration) from these probes was to watch a wave of how they changed in time, on an oscilloscope. Sometimes these vibrations were (and still are) shown in the vertical and horizontal directions together in the same screen, so the oscilloscope dot shows the instantaneous position of the shaft center at the axial location of the two perpendicular proximity probes being plotted, for example. This shows how the shaft center is whirling in an "orbit," and the size of the orbit tells you how much clearance is being taken up within the bearing. Today, the oscilloscope type of information has not lost its use (especially to see such things as flawed bearing ball pass "spikes" for example), but also the vibrations are translated into the various levels at a range (spectrum) of frequencies, through use of an FFT analyzer. When the vibration has been sorted out in this way in terms of how much is occurring at each frequency, then an interpretation can be made of why an excessive amount of vibration would occur at a given frequency—for example high vibration that was almost all at 1× running speed would make the troubleshooter think of imbalance (which has to rotate at exactly the running speed) as being the cause.

However, the other issue is one of deciding whether or not the vibration is too high. Even after years of study, there is still a lot of controversy on this subject (Rathbone, 1939; Blake, 1964; Hancock, 1974; Makay and Szamody, 1980; Marscher, 1987), and many different specifications exist. The most widely accepted specification in the chemical, power, and petrochemical industries is the API 610 (1995) specification for horizontal pumps, and the ANSI/Hydraulic Institute Standards (HI/ANSI, 2000) for vertical pumps. In horizontal pumps, vibration is measured in the vertical, horizontal, and axial direction at both of the bearing housings on both the pump and the driver. In vertical pumps, vibration is typically measured at the motor top in the horizontal direction perpendicular and parallel to the pump's horizontally-facing discharge nozzle. Be careful in specifications that you properly interpret your readings in a manner consistent with the specification's intent. For example, sometimes peak vibration is used (i.e., the level from zero to the maximum or minimum of the vibration "wave"), and sometimes root mean square (RMS) of the wave is used, which is typically about 0.707 times the peak. Depending how the different frequencies are phased relative to each other (i.e., when one's peak occurs versus another's), this 0.707 can change, so that the "true" peak as seen on an oscilloscope would often be different—typically higher—than a "derived" peak that was based on the RMS value divided by 0.707. API and the other spec agencies like ISO (e.g., ISO 1940 (1986, 1997) and ISO 2372 (1974)) have not determined what to do about this yet, since most measuring equipment is set up to measure RMS and to report peaks based on derived peaks, but the true peak is what best represents the probability of rub.

Another confusing aspect of some specifications is when it speaks of "filtered" versus "unfiltered" readings. Unfiltered simply means that the whole signal is being considered at once, while "filtered" means that only the amount of vibration at one given frequency at a time (e.g., only at 1× running speed, or only at 2× running speed) is to be compared to the specification limit.

Vibration Troubleshooting Procedures

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.

Oil Monitoring Analysis

A new technology has reached maturity over the last decade known as oil monitoring analysis (*STLE Tribology & Lubrication Handbook*, 1997). This involves monitoring the condition of the bearing lubricant, both to determine whether the lubricant needs replacing, as well as to use the lubricant as a "telltale" of whether there are problems in the pump or its driver. Typical methods include chemical analysis and the observance of the amount of particulates in the flow. Chemical analysis can consist of determining the oil's total acid number (TAN) or total base number (TBN) (determines if the antioxidation capability of the oil has degraded, implying excessive temperature in the bearing), running a spectroscopic analysis in which the atomic spectra of the small particles and other constituents in the oil are determined (determines if bearing metal is flaking off, or if dirt or contaminants are entering the oil), or running a water-detection test such as a Karl Fischer test. Particulate analysis can take the form of observing the degree to which light is blocked by a lubricant sample (implies how many particles are in the oil), or can consist of "ferrography" in which metal particles are collected from the flowing oil by a magnetic strip, and examined under a microscope, for shape, size, and color. Other specialty tests are possible, from the intuitive (smell and color) to the very sophisticated (electron microscope examination of wear particles).

Oil analysis is better than vibration for picking up the early stages of problems in rolling element bearings or gears, or for detecting severe misalignment where each rotor is "pinned" against its bearing housing by the misalignment forces.

Thermography

Thermography involves mapping out the surface temperatures on a machine or system component, looking for lack of symmetry where symmetry should exist, e.g., around the periphery of a bearing housing (Marscher, 1997). It can be a very sensitive method of discovering particular problems. In the case of the bearing housing, for example, if a heavy misalignment was present, a hot spot would be observed on one side of the pump bearing housing, and a hot spot would be also observed on the opposite side of the driver bearing housing. Thermography is most easily implemented using thermally-sensitive video equipment, but infrared cameras are still very expensive, particularly if temperatures are shown in color rather than as shades of gray. Although it takes a little more time, exactly the same information is available for a much smaller budget by plotting on paper the readings from a surface thermometer, surface thermocouple, or an inexpensive infrared gun. In the case of either a $40,000 infrared camera or a $200 infrared gun, be careful that what is being detected is emitted radiation, which depends not only on the true temperature, but also on the emissivity of the surface, which can change up to about a factor of two depending upon the amount of dirt, roughness of the surface, and whether the surface has been freshly painted. If readings are taken off a piece of material (e.g., fresh duct tape) firmly pasted on the surface rather than from the surface itself, problems with this issue can be avoided.

Electric Current Spectrum Monitoring

Oak Ridge National Laboratory (ORNL) and the U.S. Navy pioneered this method. It involves basically looking with an oscilloscope or FFT analyzer at the output from an inductive "clamp-on" electric current probe, which is clamped (like an alligator clip) around a motor phase lead (Marscher, 1997). The interpretation is much like interpreting a vibration spectrum, such that whatever frequencies show up strongly in the current are physically interpreted relative to whether they are too high in current, and if so what would cause this. The most common problem frequencies are induction motor 2× slip frequencies (including 1× speed ± 2× slip "sidebands"), and of course the line frequency, twice the line frequency, the rotor bar passing frequency, and the stator slot passing frequency. Broken rotor bars, for example, lead to increased slip sidebands and stronger levels of slotpass. Theoretically, faults in the pump that affect the motor torque and therefore current should also be evident in the spectrum, but generally vibration is more sensitive for pump problem detection.

An exception is shown in Figure 3 for a 1250 hp 4160 V vertical turbine pump, in which the electric current spectrum (when plotted logarithmically to emphasize the low level "floor" of the spectrum) clearly shows a hump due to the pump/motor first torsional natural frequency at 261 Hz (where the vertical dotted line is in the lower figure), as identified separately by torsional impact test as well as by finite element modeling. Note that the torsional does not show up in the vibration trace from the top of the vertical motor, and in fact it would not be expected to (as opposed to a lateral natural frequency, which would show up in vibration, but not in the current).

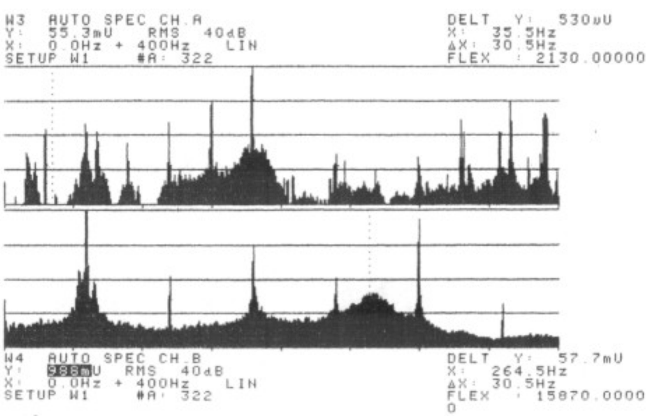

Figure 3. Top Trace—Vibration Versus Frequency; Bottom Trace—Same Pump Current Versus Frequency.

Specific Guidelines for Pump Parameter Measurement and Interpretation

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 pump, and on pump-side (i.e., "inboard") bearing housing on 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., flow rate), how close is the temperature-compensated head of the pump to the curve supplied by the manufacturer, especially near the design or 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
- Whether the suction pressure is steady at a given operating point, and well above NPSH requirement
- 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 a FFT analyzer

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

- Vibration 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

CASE HISTORIES

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

A major U.S. petroleum refinery had a serious gear box 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 through a right angle 1:1 gear box 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 in one case finite element analysis) to understand and cure the problem over the several years since installation. On two separate occasions, replacement of the gear boxes 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 the author found that all rotor system torsional natural frequencies were close to their predicted values, and were not near the unit's constant operating speed.

Impact modal testing was performed on all exposed stationary as well as rotating components, using the cumulative time averaging method referenced in the discussion above. This impact testing was performed successfully while the unit continued to operate. 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, as shown in Figure 4, 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 (through a small grease fitting placed at one end near the U-joint, with an air relief hole drilled at the opposite end near the other U-joint) to damp out this unusual vibration. The gear box noise immediately fell a factor of 10, and all gear box problems ceased.

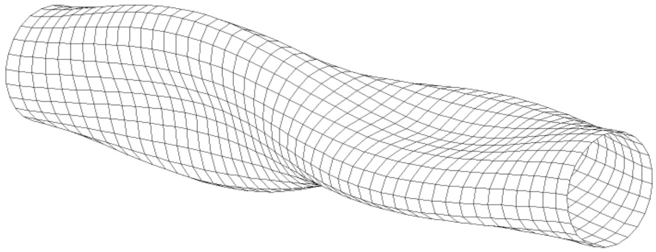

Figure 4. Hollow Drive Shaft Second Bell Mode.

Although the grease-filling procedure was meant only to be a proof-of-principle concerning the nature of the problem and to serve as a temporary fix, the procedure worked so well that the plant decided to keep the shaft in this configuration permanently.

Reliability Problem Resolution by Careful Combination of Rotordynamic Analysis with Test

A northeastern power plant had experienced chronic boiler feed pump 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, the author was called in for a "third party" opinion.

Impact vibration testing using the cumulative time averaging procedure referenced above 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, as illustrated by the critical speed map in Figure 5, 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. The particular clearance was the 135 degree long pressure dam pocket, which was supposed to be 10 mils depth, but was mistakenly 40 mils depth. 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 4 ft (over 1 m) 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 the following:

- There were 35 mils of misalignment at the coupling due to the hydraulic loads on the pump discharge flange, far in excess of API 610 (1995) 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 (1995).

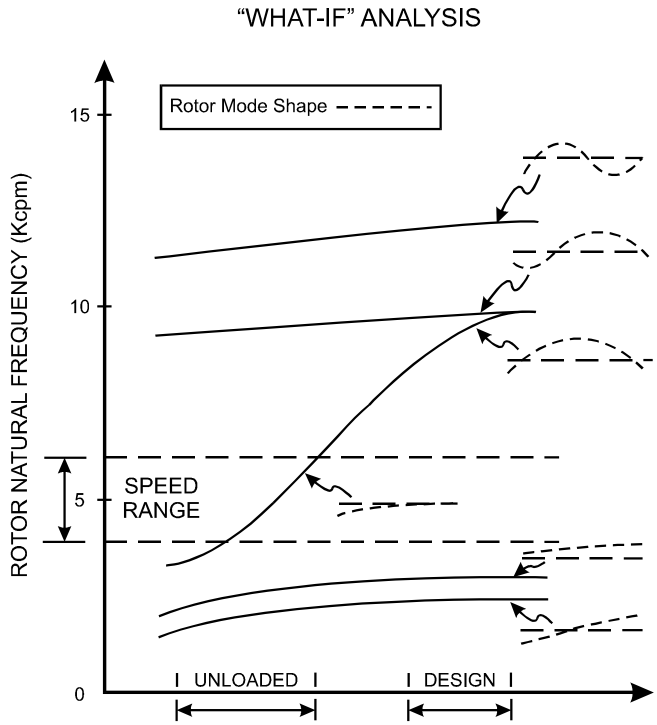

Figure 5. Variation of Rotor Critical Speeds with IB Bearing Stiffness.

CONCLUSIONS

The purpose of this tutorial has been to explain pump mechanical concepts, terminology, recent component developments, and proven methods of finding and diagnosing problems, so that those involved in the selection, operation, and maintenance of centrifugal pumps are able to maximize their effectiveness on behalf of the plants that employ them.

Although the heart of a centrifugal pump design consists of its hydraulic passage shapes, a wide variety of mechanical issues need to be addressed to ensure effective and reliable operation. This is particularly true for pumps of high energy density, such as boiler feed pumps and certain refinery pumps. Therefore, it is important for those who write specifications or make purchase decisions to understand the benefits of various mechanical options and technologies available to them with regard to centrifugal pumps; plant downtime costs as well as maintenance costs often will be dramatically affected depending upon the factors weighed in the pump selection process. Similarly, operations personnel need to understand the mechanical versus production costs of operating the pump at various off-design capacities, and the level of risk that they take if they continue to operate when the pump shows symptoms of various types of problems. Finally, maintenance groups should be aware of the often subtle signs of pump degradation, how most problems and their symptoms are component-specific, and the modern tools and components available for the solution of chronic problems.

Machinery issues such as the effects of nozzle loads and procedures for checking acceptability of vibration can seem deceptively simple. In reality, it takes education and experience to reach the correct conclusion relative to the many interrelated issues associated with choosing and operating a centrifugal pump. The purpose of this tutorial is to provide a "jump-start" to the attendees in this process, and to perhaps add a different perspective and a few new tricks of the trade for more experienced pump users and troubleshooters. Below are some parting comments that encourage implementation of the material presented in this tutorial:

- 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 much more likely to result in the correct conclusions than more traditional "manual" techniques.

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)
FRF = Frequency response function
f = Frequency (cycles per minute, cpm, or cycles per second, Hz)
f_n = Natural frequency (cycles per minute, cpm, or cycles per second, 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)
α = Thermal expansion coefficient
δ = Vibration displacement amplitude, peak-to-peak, or shaft deflection (mils or mm)
Δ = Shaft bending displacement (mils or mm)
ρ = Density (lbm/in or kg/mm)
ω = Vibrational frequency (radians/s)

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A more detailed discussion of the case histories presented in this tutorial, as well as a variety of additional pump case histories, is presented on the webpage: www.mechsol.com.

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ACKNOWLEDGEMENTS

The author would like to thank Mechanical Solutions, Inc. (MSI), for its support of the author in the construction of this tutorial. In particular, Leo Andronis, Bill Kelly, Michael Marscher, and Jim McGinley contributed much of the test data, analysis results, tables, and figures of this tutorial, and thanks to Karen Rizzi for her support in assembling the figures and putting the test in presentable format. Many thanks also to the manufacturers and users of pumps around the world who have employed MSI in the pursuit of the many interesting design developments, analyses, and troubleshooting tests that provided valuable background information and experiences. In addition, the author would like to express his gratitude to the many key individuals in the pump industry with whom he has had the pleasure to work with over the years on various projects.