CENTRIFUGAL PUMP MECHANICAL DESIGN, ANALYSIS, AND TESTING

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ABSTRACT

This tutorial discusses hydraulic and mechanical issues that are of primary importance in centrifugal pumps and their systems, and how pump users can best address these issues. It is shown how a concise and orderly review of these issues can be developed that includes specifying installation system requirements, predicting approximate vibration and nozzle load behavior, quantitative evaluation of sealing and monitoring needs, and startup monitoring and assessment.

The tutorial examines various design types and options in their selection and analysis. The discussion includes selecting pump design options that allow the user to meet flow requirements, avoiding overstress fractures and cyclic fatigue, and minimizing rotor rubbing and wear. The relationship of these issues to improper warmup cycles, resonance, and recirculation during running off-BEP are presented. In addition, proven methods of analyzing and testing to determine the nature of such issues, and how to establish a system that will also be trouble-free when the key is first turned are reviewed. A number of case histories show by example how best to apply the analysis, evaluation, and test methods. Enough detail is given concerning each method so that the attendees should be able to try the methods back at their plants.

DESIGN CRITERIA

Design of a centrifugal pump system intends to balance the following considerations:

- Capital cost (initial)
- Performance
- Reliability
- Operational costs (life-cycle)

For many organizations only the first two items are considered in the purchase decision, with initial cost being of primary importance. Focus on initial costs and performance often leads to high life-cycle costs and decreased reliability.

The initial cost of a system can often be quite low when compared to the operational costs of the equipment over time. Factors such as the cost of power, repair costs, and lost production are less commonly considered in most purchasing processes. Making sound, informed purchasing decisions during the front end of the purchasing process can often improve performance, increase reliability, reduce life-cycle costs and occasionally reduce initial purchase costs. The first section of this tutorial explains some of the design options that are available and the pros and cons of their use.

BASIC PUMP DESIGN—COMPONENTS

A basic centrifugal pump system consists of the following elements: piping, pump, sealing system, seal auxiliaries, and a driver. This tutorial discusses some of the common component options available and how they can best be applied to improve the performance and reliability of your pump systems.

Basic pump components are reviewed individually to introduce the design options and to provide the reader with a working knowledge of the pluses and minuses of each. The intent is for the reader to gain a basic understanding of each option presented allowing them to ask the proper questions in the purchasing and repair processes to obtain a product that meets their requirements (performance, reliability, future needs) for their processes.

CASINGS

Volutes

Single

Single volutes are by far the most common type of diffusion device in centrifugal pumps (Figure 1). Their prevalence is based on their ease of manufacture and higher performance in specific speeds above 600 (US). They have a larger useable range than some other diffusion device designs that are presented, making them a good general service choice. Their main drawbacks are that they lead to larger diameter casings, and the fact that they impose higher radial loads as the pump moves away from the best efficiency point (BEP) of the pump (Figure 2). The radial load imposed on the impeller is at its lowest point, theoretically at the pump's BEP. To either side of this, the radial load increases, both steady load as well as vibrational load. The load is generally highest at the shutoff condition in the pump. These loads develop due to the uneven pressure distribution around the periphery of the impeller at off-BEP conditions. The reader is directed to Brennen (1994) for a complete description of the forces causing these steady-state radial loads.

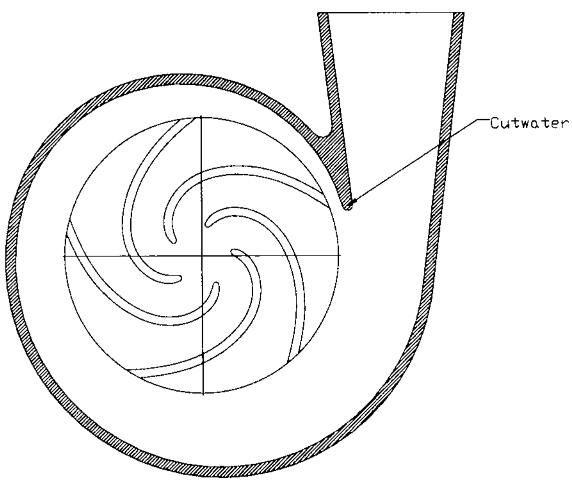

Figure 1. Single Volute Casing.

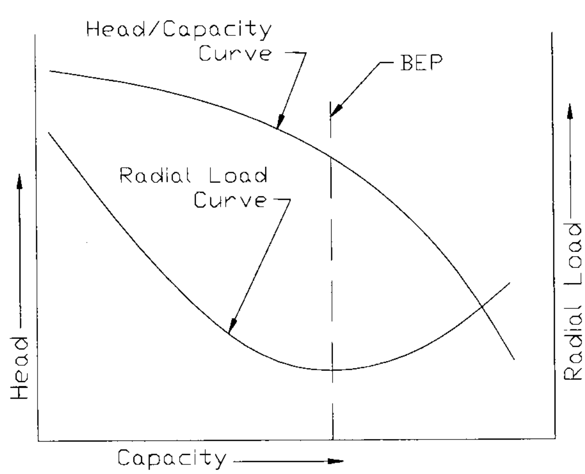

Figure 2. Single Volute Radial Load Curve.

Double

Double or "twin" volutes solve the radial load problem by splitting the volute channel into two sections (Figure 3). This provides lower radial loads than are possible with single volutes, and is a common method of manufacture in larger pumps such as horizontal split-case designs. Radial loads can be drastically reduced with this type of volute, with reductions in radial loads of 75 percent being common. The disadvantage of this type of volute is a slightly lower efficiency at the pump's BEP, and higher manufacturing costs. The increased manufacturing costs are a result of the additional casting cores required to form the second volute channel, and the difficulty in properly suspending the cores. These volutes also present problems in the cleaning of the core material from the volute passage when they are used on smaller pumps.

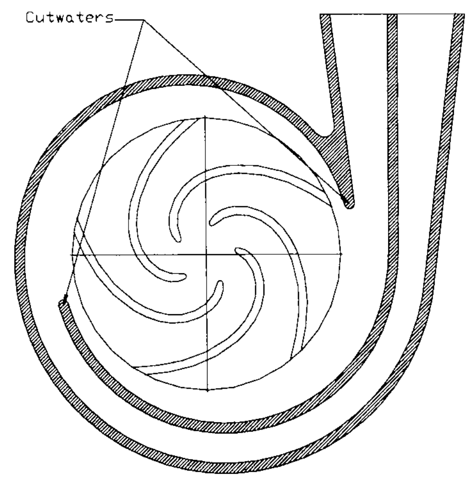

Figure 3. Typical Double Volute Casing.

Multiple

Multiple volutes are simply an extension of the double volute concept. If two is good, more must be better. They are less common, but are used to further reduce the radial loads in the pump casing and for strength and manufacturing reasons. In the past some nuclear coolant pumps used quad volute designs to reduce the amount of material involved and therefore reduce the overall cost of the component. This was done as a cost saving measure and really offered no mechanical or hydraulic benefit. Most research on the topic indicates that there is generally no mechanical advantage to using more than the double volute design.

Concentric

Concentric volutes are very common in lower cost pump designs. As can be seen in the diagram (Figure 4), the casing is circular and maintains a set distance from the OD of the impeller. Because of this simple geometry they eliminate some of the machining and foundry issues that are common in producing true volutes. This also makes them very popular in inexpensive stamped stainless designs and pumps machined from billets. A positive benefit to this type of pump is that the highest radial loads are at the runout condition and they diminish as the flow is reduced (Figure 5), which can be particularly useful in wastewater service.

Diffusers

Vaned Diffusers

Vaned diffusers are very common in high energy engineered pumps and compressors (Figure 6). They consist of a series of diffusion passages at the exit of the impeller. Each of these diffusion passages works as a small volute gradually diffusing the flow. In general the efficiency of the vaned diffuser is better than

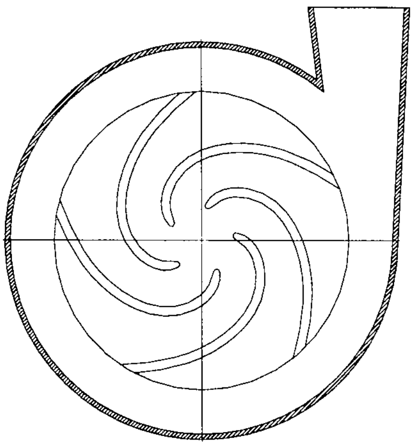

Figure 4. Typical Circular Volute.

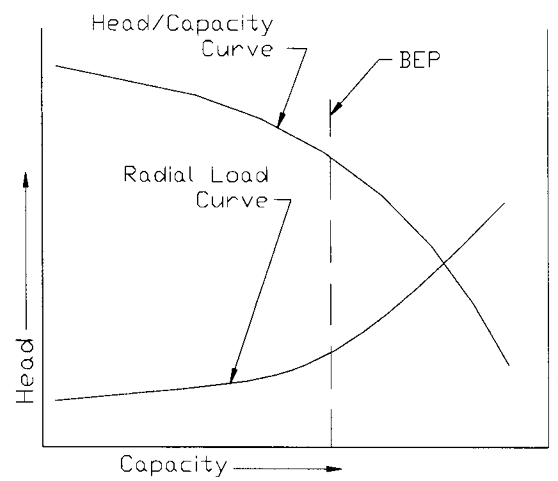

Figure 5. Radial Load Curve for a Circular Volute.

volute designs near the best efficiency point. Due to the separation of the flow into several diffusion passages located symmetrically around the impeller, the steady-state radial loads are almost nonexistent in diffuser designs. The disadvantage to this type of design is that the operating range of the vaned diffuser is generally much smaller than other types of diffusion devices. This is caused by the fact that the angle of incidence between the diffuser vane and the angle of the incoming flow must be relatively small. This occurs only when the inlet flowrate is near the design flowrate for the diffuser. As the flowrate departs from the design flow, the angle of incidence increases causing fluid separations and diminishing the effectiveness of the diffuser. Another disadvantage is the cost of producing quality diffusers. This generally adds another component to the pump and thus increases its cost.

Vaneless

Vaneless diffusers are very common in radial compressor designs and can be found in more advanced pump designs. The vaneless diffuser consists of a vaneless space at the discharge of the impeller. The vaneless space allows the fluid to diffuse (slow down) as it moves outward radially. As can be seen in Figure 7, there are several configurations of vaneless diffusers. The lack of a vane causes very little radial load in this design. The efficiencies of the vaneless diffuser can be very good if ample radial space is provided to gradually diffuse the flow. The disadvantage of this type of design is that it generally increases the overall size of the machine because vaneless diffusers typically require a larger diameter casing, with the exception of a continuous crossover diffuser that provides this extra diffusion length tangentially and axially.

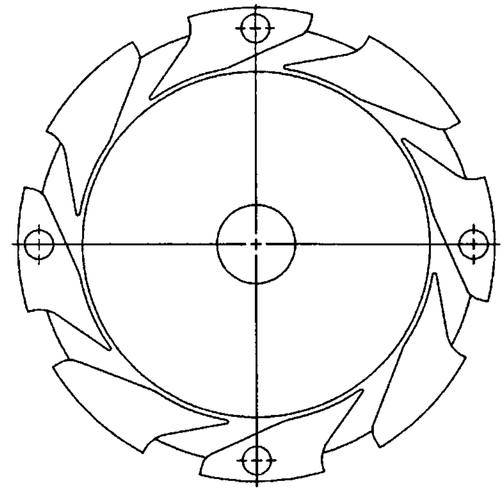

Figure 6. Integral Vaned Diffuser and Return Channel.

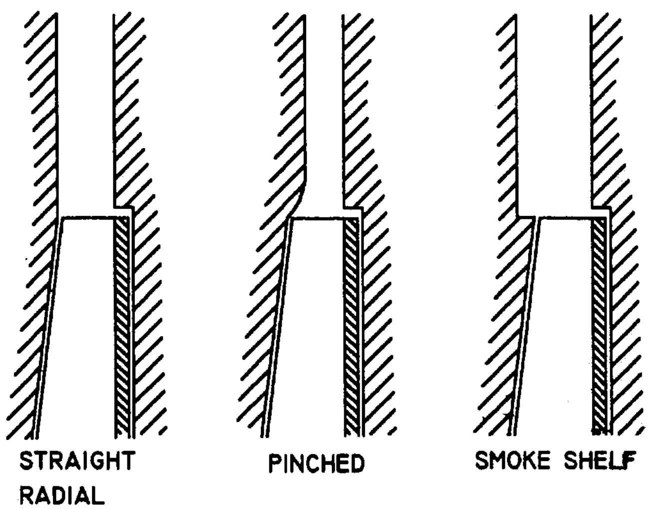

Figure 7. Common Vaneless Diffuser Styles.

IMPELLERS

Open Impellers

Open impellers are intended to be used to pump fluids containing particulate. The impellers do not have shrouds on the front or the rear of the impeller allowing them to wear particulate that might clog other impellers along the stationary walls of the front casing and rear cover. As the impeller rotates, the particulate is dragged along the stationary walls causing it to wear down so it can pass through the impeller.

An added benefit to the open impellers is the lack of a surface area on the impeller shrouds to allow the axial loads to build on. This drastically reduces the axial loads on the pump, improving bearing life.

Some of the disadvantages to this style of impeller are the added leakage areas around the front and rear of the impeller vanes. This often causes efficiencies of these impellers to be lower than impellers of similar specific speed but of different styles. Another drawback is the thickness of the vanes. The vanes must generally be thicker due to the lack of a shroud to help support them. Because of the thicker vanes the impeller must necessarily have fewer vanes and thus have less control over the fluid.

Closed Impellers

Closed impellers (Figure 8) have shrouds on both sides of the impeller providing less leakage and better control of the fluid flow.

These impellers can be prone to clogging from particulate in the pumpage. Generally the impeller efficiency is slightly better than open and semi-open impellers due to the lower leakage rates around the impeller vanes. They are also generally thought to have better control of the fluid direction because the added shrouds rotate with the vanes, preventing the increased drag on the fluid imposed by the stationary walls. While the support provided by the two shrouds on this style of impeller mechanically allows for thinner vanes, the vane number and thickness are somewhat restricted in cast impellers by the need for wall sections of consistent thickness and the need to be able to remove the impeller cores.

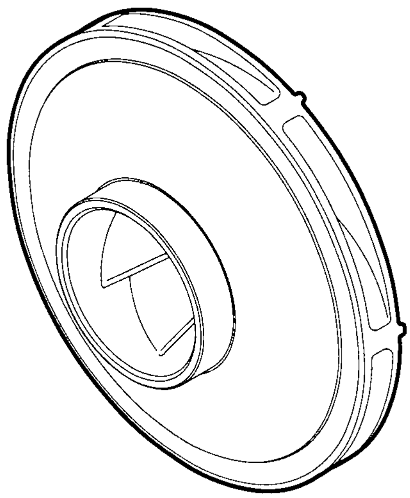

Figure 8. Closed Impeller. (Courtesy of Price Pump Co.)

Semi-Open Impellers

Standard Semi-Open Designs

This design, and variants of it are probably the most common type of pump impeller in the world (Figure 9). As the name implies, the impeller is a hybrid of the open and closed designs. The design has one shroud commonly on the back of the impeller vanes. This shroud provides support for the impeller vanes as well as preventing leakage around one side of the vane. They share many of the advantages of both the open and closed impellers, such as self-cleaning impeller passages, excellent control of the fluid in the vane passages, and the ability to use fairly thin vanes. The absence of one of the shrouds often allows for more advanced vane profiles to be used including the addition of splitter vanes that would be difficult to use on closed impeller designs.

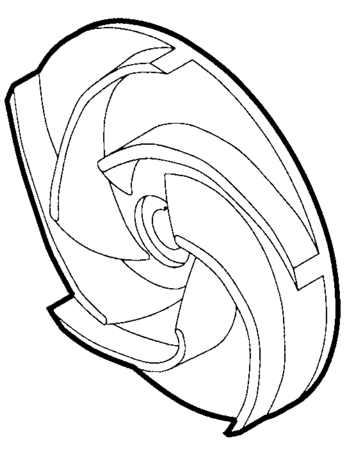

Figure 9. Semi-Open Impeller. (Courtesy of Price Pump Co.)

SEAL CHAMBERS

Many very informative papers have been presented at this symposium over the years about the various types of seal chambers and the effect on seal performance (Figure 10). The reader is directed to examples such as Adams, et al. (1993), for a detailed comparison of seal chamber designs.

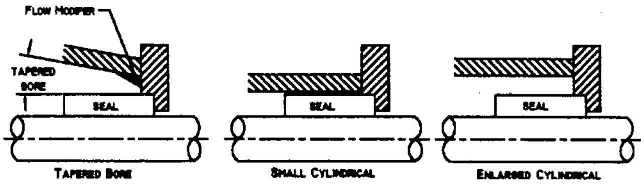

Figure 10. Various Common Seal Chamber Designs.

Tapered Bores

Smooth

Taper bore seal chambers have become very prevalent in most types of pumps. Tapered bores have the advantage of being self-purging. Gases that would build in the seal chambers are allowed to escape into the pumpage and purged out the discharge of the pump. Their main disadvantage is the increased susceptibility to erosion caused by higher circumferential velocities due to the larger seal chamber opening being exposed to the rotating impeller. This effect is exacerbated when back pumpout vanes are present on the rear shroud of the impeller. In fluids with particulate in pumpage, the erosion damage to the seal gland can be extensive in a short period of time.

With Flow Modifiers

To break up the damaging circumferential flow patterns, flow modifiers (speed bumps) are added to the seal chamber. These are generally cast into the seal chamber and consist of small ribs protruding into the seal chamber from the walls. As the flow passes over the walls, the ribs direct the flow toward the shaft centerline and out of the seal chamber. This promotes circulation in the seal chamber and expels the particulate from the seal chamber.

Small Cylindrical

Small cylindrical bore seal chambers are close-fitting chambers that have evolved from the packing gland. These chambers are simply the packing gland bored through to allow the fluid to pass freely into the seal chamber. This type of chamber is very common but can increase a variety of problems in operation. Some of these problems are the entrapment of air and gases. Due to the close confines of the seal chamber, flow patterns can sometimes be inadequate to purge the cavity of gas that gets entrapped within it. Other issues include the lack of lubrication and cooling to the seal faces because of the tight confines of the chamber to the seal. Often the seal head is close to the same diameter as the seal chamber bore. This makes it very hard for natural circulation to provide enough flow to the seal faces to properly cool and lubricate them. The addition of seal flushes (suction or discharge) is generally recommended for this type of seal chamber.

Enlarged Bore

The enlarged bore seal chamber is an improvement of the small cylindrical design. The larger bore diameter allows for better circulation of the fluid in the seal chamber, which leads to better cooling and lubrication of the seal faces.

HYDRAULIC AXIAL BALANCE

Hydraulic axial balance refers to the balancing of the axial loads generated by the hydraulic forces on the impeller. As the impeller generates head, the pressure on the shrouds of the impeller is increased. Due to geometric features and sealing surfaces this often causes an uneven distribution of the pressure on the shroud surfaces. This uneven pressure leads to axial loads on the impeller. These axial loads can lead to bearing and seal problems. The reader is directed to Lobanoff and Ross (1992), Brennen (1994), Japikse and Baines (1994), and Japikse, et al. (1997), for a detailed explanation of how the radial forces are developed and some rudimentary techniques for calculating their magnitude. The reader is cautioned to use these calculations as a best guess for the load's imposed. Physical testing is the only reliable method that can accurately predict the axial loads in a piece of equipment.

Double Suction Impellers

One of the most effective techniques used to balance hydrodynamic axial loads is the use of double suction impellers (Figure 11). By using virtually the same impeller profile on both sides of the impeller, the developed forces are nearly equal. These impellers are inherently balanced and develop very low axial loads. One side of the impeller is generally designed to develop a small axial force to load the thrust bearing slightly and prevent the impeller from hunting back and forth during operation.

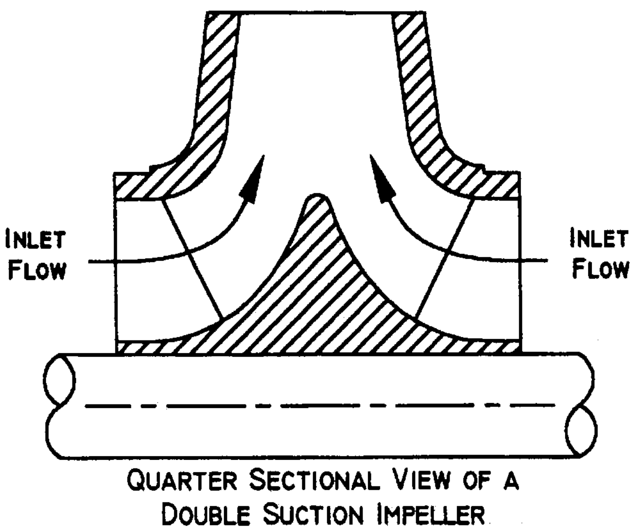

Figure 11. Double Suction Impeller.

A variant of the double impeller concept, used in multistage designs, is the mounting of an even number of impellers on a single shaft so that they oppose each other. The axial loads are then equalized on the rotor as a whole. The challenge and expense in this type of design are the intricate casing design and coring of the internal passages that allow the liquid to pass to multiple suctions and discharges. This type of design is common on large multistage axially-split-case pumps.

Balance Holes

Balance holes are simply small holes drilled in the shroud(s) of the impeller (Figure 12). Generally these holes are drilled as close to the pump centerline as can be mechanically accommodated to maximize the pressure reducing effect. Their purpose is to allow the fluid pressures on each side of the shroud to equalize. Due to the fact that balance holes are drilled at a fixed radius from the shaft centerline and have a fixed diameter, they are most effective at only one point on the performance curve of the pump. As you move away from this point, large axial forces can build on the impeller. Another problem that can be encountered from the use of balance holes is the reduction of the pressure in the seal chamber below the boiling point of the liquid. If the holes are sized to almost completely equalize the pressures on the impeller shrouds, the pressure in the pump suction can often drop to very low pressures at high flowrates. The low pressures combined with the added heat from the pumping action and mechanical seal can cause the fluid to boil and thus cause the mechanical seal to run dry. Another detrimental effect of the balance holes is the disturbance of the inlet flow pattern in the main impeller passages. As the fluid passes through the balance hole, it is accelerated causing a small jet that shoots out into the flow stream of the main impeller flow. This can cause disturbances to the flow pattern and potentially increase the NPSHR for the pump.

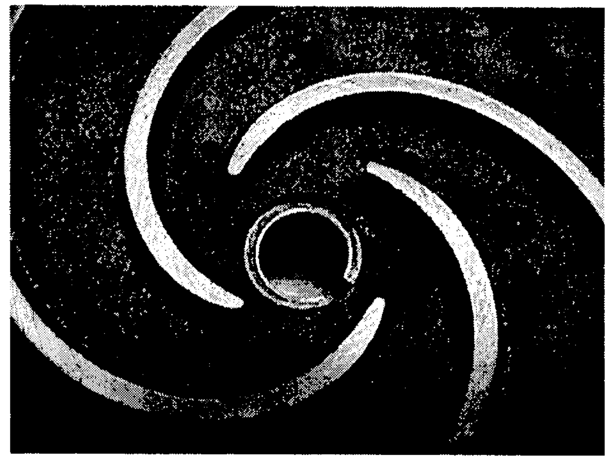

Figure 12. Semi-Open Impeller with Balance Holes. (Courtesy of MTH Tool)

Back Pumpout Vanes

Back pumpout vanes are generally small protrusions from the rear shroud of the impeller (Figure 13). They may be of a straight radial design or may follow the contour of the main impeller blade. Just like the main blade of the impeller, the function of the back pumpout vane is to pump out the fluid on the back of the impeller and thus equalize the pressure across the impeller shroud. They can be very effective in reducing the axial load on the impeller across a wide operating range of the pump. They are most effective when the top of the vane is within 0.030 inch of the rear casing. This prevents leakage around the vane and allows for efficient pumping action.

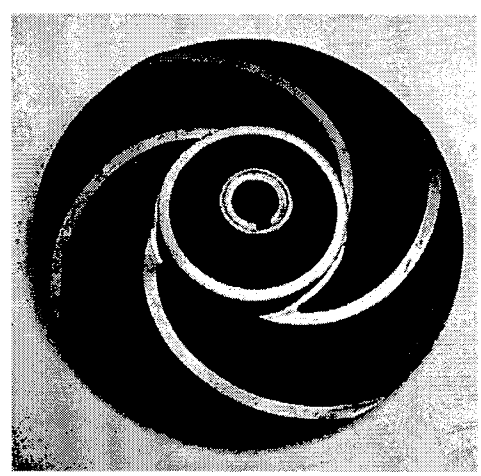

Figure 13. Impeller with Back Pumpout Vanes. (Courtesy of MTH Tool)

A problem that can occur when the vanes are extended all the way to the impeller hub is that the rotating motion of the flow can add to erosion problems in the seal cavity.

Another issue is the reduction of the pressure in the seal cavity. As with the balance holes, the back pumpout vanes can reduce the pressure on the rear surface of the shroud to levels below the vapor pressure of the product. This can again lead to boiling of the product in the seal chamber and cause the seal to run dry.

Both these problems can be overcome with the use of a throttle bushing placed in the entrance of the seal chamber and an ANSI Plan 11 seal flush. This provides a controlled pressure flush into the seal chamber and allows the pump to equalize the pressure across the impeller shrouds.

Balancing Drums and Disks

The use of balance drums is common on many types of large rotating equipment. In truth the use of balance holes is often a compact version of a balancing drum. A true balancing drum uses a drum or cylinder connected to the rotating shaft with a small radial gap along the axis between the casing and the drum (Figure 14). The area on the front of the drum (area B) is exposed to the discharge pressure of the pump. The rear section of the drum is vented back to the suction of the pump. The pressure on the rear section of the drum (area C) is much less than the pressure on the front section. The axial load generated across the impellers can be balanced by an equal (or nearly equal) and opposite force produced by the balancing drum. This method like the balance holes is only completely effective at one operating point. To overcome this flaw the addition of a variable orifice or balancing disk is often included with the balancing drum to improve its performance across the entire operating range of the pump (Figure 15). The balancing disk allows the pressure to build against area A+B, which causes the radial orifice to open. As it opens, it releases the pressure that is acting against area B. As this pressure is reduced, the pressure acting against area C overcomes the pressure on area A+B closing the radial orifice. This condition continues to self-correct through the entire operating range of the pump. However, there is the potential for unstable hydraulic oscillation in this design.

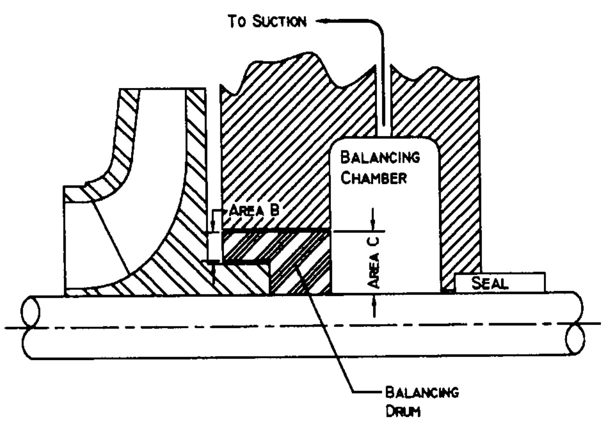

Figure 14. Balancing Drum.

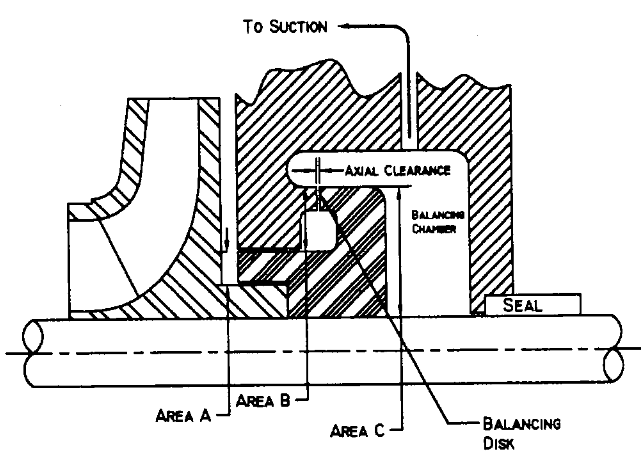

Figure 15. Combined Balancing Disk and Drum.

SEALLESS PUMP OPTIONS

The sealless pumps that are discussed here are of the magnetic drive and canned motor types. These pumps offer the advantage that the rotating shaft does not pass through the casing, and thus no dynamic seal is needed to prevent leakage. Sealing of the components is accomplished through the use of standard O-rings or compression gaskets. Common designs have the shaft rotating in fixed journal bearings or use a stationary shaft with the bearings mounted in the rotating magnet capsule/impeller assembly. Most designs use the pumpage as the lubricating and cooling fluid for the internal bearings. There are several material combinations available for the shaft, bearing(s), thrust surface(s), and casing components. The following is a brief description of some of the common materials and their benefits and shortcomings. As always, the consumer should consult the manufacturer regarding their experience with the chemicals being pumped. All known data about the chemical and process conditions should be forwarded to the manufacturer or their representative for review and proper selection of materials.

Fluids Being Pumped

Commonly the pumpage is used as the lubrication and cooling medium for the internal bearings. This requires that the fluid is able to do two things:

- Build a fluid film between the shaft and the bearing to prevent contact of the surfaces
- Provide a method to remove the heat generated by the bearings and eddy currents that are generated in the containment shell of the rotor

It must provide lubrication properties to the bearing surfaces, and be capable of forming a fluid film that will support the rotating assembly over the entire range of the pump's operation. Failure to form this hydrodynamic wedge causes the bearing surfaces to come into contact with each other. Depending on the material combinations, and the extent of the bearing contact, this can cause a slow wear of the bearings in a mild case, to sudden catastrophic failure in extreme conditions.

The pumpage must therefore be delivered to the bearings cool enough that it will not vaporize as it picks up the heat added by these components. Reputable sealless pump manufacturers can provide estimates of the temperature rise through the drive section of their pump based on your fluid properties and the operating conditions of your service.

Cleanliness of the pumpage can also be a major factor in extending the service life of a sealless pump. Abrasive particulate (harder than the bearing surfaces) can cause extreme wear in a short period of time. The use of filter and strainers is generally encouraged as long as the NPSH conditions are sufficient even when the filter is full and ready to be changed.

Shaft and Bearing Options

Shaft and bearing material selection is of paramount importance in the proper application of sealless pumps. Most manufacturers have a variety of options available. Common options are somewhat soft materials such as carbon and filled composites. On the other end of the spectrum are very hard materials such as silicon carbide (SiC) and some ceramics.

Soft Bearing Materials

The softer materials generally have some inherent lubricating ability that allows them to support the shaft during periods when the product may vaporize in the bearings. These materials can provide a safety cushion during pump upset conditions such as dry running or deadheading of the pump. Commonly they are used as the sacrificial piece of the system. If left unchecked after these periods of upset the bearings often wear to a point where the bearing clearances can no longer build the fluid wedge necessary to support the shaft in normal operating conditions. As the shaft continues to wear against the bearings, the clearances continue to widen until a major failure occurs.

Hard Bearing Materials

Hard bearing materials such as silicon carbide, tungsten carbide, and some ceramics provide excellent wear characteristics, promising virtually infinite life if the proper operating characteristics are maintained. To achieve these operating results the manufacturer must allow the bearing to properly align itself in the pump as well as accommodate the difference in thermal expansion of the "hard" materials and their surrounding support structures. This is often accomplished through the use of tolerance rings on the exterior of the bearings. When properly mounted the bearing is free to align itself with the shaft to reduce edge loading of the bearings, which can cause cracking of the bearing and/or scoring of the shaft.

These bearings are resistant to abrasion and erosion caused by particulate in the pumpage. Because they are so hard most of the particulate is unable to scratch the surface, but softer materials can smear along the surface possibly causing pitting from the adhesion of the particulate. Their Achilles' heel is their low survivability in upset and dry running conditions. Many ingenious devices have been incorporated into bearing designs using these materials, and many notable papers have been presented at this symposium over the years outlining research being done to improve the dry running capabilities of the materials. Diamond-like coatings have been used over the past couple of years and results have been favorable in the authors' experience.

Shaft Materials

These range from a metallic shaft made from the same material as the casing to shafts of solid SiC. In small mag-drive units it is common for the shaft to be solid ceramic. This provides a hard smooth bearing surface at a reasonable cost. In larger units the shafts are generally sleeved or coated with the base shaft material being metallic. The sleeves are often made from the same material as the bearing being used. Diamond-nickel and tungsten carbide coatings have also been used very effectively on metallic shafts running against hard bearings. Hard chrome coatings are a popular lower cost option that can be used against the softer bearing materials.

Nonmetallic and lined units have become very popular over the last several years. These pumps generally use ceramic or SiC shafts and perform very well in their intended conditions of service. Nonmetallic pumps often use completely nonmetallic containment shells. The lack of metal prevents the formation of eddy currents and thus reduces the heat generation in the drive section.

BEARING FRAME OPTIONS

As we continue our quest for longer mean time between failure (MTBF) and increased pump availability, the improvements being made in the bearing frames of long coupled pumps cannot be overlooked. Manufacturers have long realized that the improvement of the lubrication systems and the availability of oversized or extreme duty bearings can go a long way toward increasing MTBF. Some of the options and/or improvements listed below have been incorporated into standard bearing frame designs and others have been made available as individual options or as a package in extreme duty bearing frames.

- *Deeper oil sumps*—The deeper oils sumps have been incorporated into the bearing frames of several manufacturers. They provide a larger quantity of oil so the oil pumped from the bearings has a longer dwell time in the sump allowing it to cool prior to its next trip through the bearings. The deeper sump is also thought to reduce agitation and allow particulate and water to collect on the bottom of the sump, thus keeping the debris and moisture from passing through the bearings.
- *Improved bearing seals*—The use of labyrinth seals to prevent contamination of the oil sump has been a major contributor to the extension of bearing life. These seals prevent moisture from primary pump seal failures and wash down cleaning processes from entering the oil sump. They are also effective in preventing dirt and other particulate from entering the sump and damaging bearings.
- *Improved lubricants*—Both oil and greased bearings have benefitted from some of the advancements in lubrication technology. The new lubricants, synthetic and hybrids, allow for higher bearing temperatures, wider range of operating temperatures, and longer lubricant life between change out. This often translates into longer bearing life and increased equipment reliability.
- *Extreme duty bearings*—These bearings offer higher dynamic load ratings in the same bearing size. This allows the user to operate the pump at higher loads for a longer period of time. This is definitely a band-aid. The true solution is to select the correct pump for the application. When the correct pump has been selected, these bearings also offer a better safety margin. These bearings will run cooler and under a lower percentage of the allowable load thus extending their useful life and the MTBF for that component.

MECHANICAL ISSUES

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

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

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

Casing Stress and Distortion Analysis

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

Manual calculations for steady stress fall mostly into three categories:

- Pressure vessel
- Hollow beam
- 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 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 which tend to be flat, a safer estimate is to assume that the stress may be as high as half the internal pressure times the square of the maximum casing wetted dimension, divided by the square of the wall thickness. Usually, the true maximum stress is about one-fifth this value, so by using this estimate without any additional factor, the possible effects of stress concentrations are already included.

The hollow beam calculation assumes that the peak nozzle stress can be predicted as roughly three times the moment on the nozzle, multiplied by the nozzle radius, divided by the nozzle moment of inertia. Handbooks such as *Marks' Standard Handbook for Mechanical Engineers* (1996) or *Roark's Formulas for Stress and Strain* (Young and Roark, 1989) can be used to determine the stress in such cases, although the 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.

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 °F (about 13 millionths of a meter per meter length per °C) for most steel, and about nine millionths of an inch per °F (about 16 millionths of a meter per °C) 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 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 graphically for α 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:

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

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

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

Pump Suction Design

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.

Relative to the mechanical connection, 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.

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 net positive suction head available (NPSHA) to satisfy the 3 percent head drop NPSHR (NPSH required) 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 best efficiency point, the angle of attack of incoming flow on the rotating impeller vane can be different than 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.

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

Resonance is illustrated by someone playing basketball—his dribbling (the exciting force) synchronizes with the ball bounces (the ball's natural frequency), but if he is uncoordinated the ball will not bounce very high. 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 from discharge pressure pulses as the impeller vanes move past the volute or diffuser vane "cutwater").

In practice, pumps do not have only one natural frequency in the range of running speed, but have at least several of them, each one with a different contortion or "mode shape." At any of these natural frequencies, the vibration amplification 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. The total amplification 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 trapper between the rotor and stator in the "annular seals," like the balance piston. The amount of vibration amplification that is typical when a force that acts at a certain frequency (like running speed) passes through a group of natural frequencies is illustrated in Figure 16.

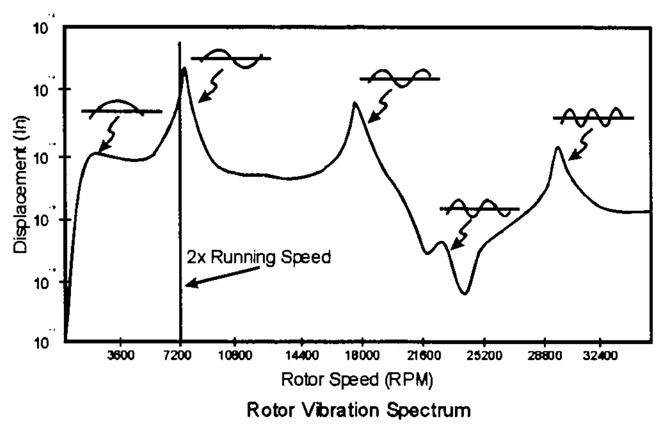

Figure 16. Vibration Amplification Versus Frequency Plot.

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).

Another important concept is the "phase angle," which measures the time lag between the application of a force and the vibrating motion that occurs in response to it. A phase angle of zero degrees means that the force and the vibration due to it act in the same direction, moving in step with one another. This occurs at very low frequencies, well below the natural frequency. An example of this is a force being slowly applied to a spring. Alternately, a phase angle of 180 degrees means that the force and the vibration due to it act in exactly opposite directions, so that they are perfectly out of step with each other. This occurs at very high frequencies, above the natural frequency.

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

Manual Vibration Analysis

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

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

$$f_{n1} = (120/\pi)[(3EI)/\{L^3(M + 0.49M_S)\}]^{1/2} \qquad (2)$$

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

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

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

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

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 outside diameter (OD), times the OD, 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).

The most accurate means to determine rotor natural frequencies, and the only reliable way to assess overall rotor stability, is with a complete rotordynamic's analysis. This is because there are complications in analyzing rotors versus performing similar analysis on stationary structures. For example, regardless of the bearing type used in a particular pump, the reaction forces that occur in the bearings in response to vibration and even static loads are not straightforward. Besides the direct restraining force of each bearing that acts exactly opposite to rotor motion, there are also other important forces that act perpendicular to this motion, namely damping and "cross-coupling" (bearing reaction "spring" force that acts perpendicular to the shaft motion), 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. The most important aspect of these forces in industrial pumps is generally called the "Lomakin Effect." In this effect, each annular seal acts to some extent as a bearing, usually tending to stiffen the rotor support and raise the natural frequencies to higher values, at least until the clearances wear. However, as pointed out by the work of Childs (1982), the expected "stiffening" can actually become "destiffening" if enough fluid swirl is present at an annular seal inlet, and other effects such as annular seal "effective mass" and cross-coupling should be accounted for as well. The Lomakin Effect is particularly strong in multistage pumps, because multistage rotors are relatively long and flexible. Since the annular seals are primarily in the central portion of the rotor, where they exercise considerable leverage on the first bending mode of the rotor, the contribution of seal stiffness to the rotor support can be comparable to the rotor stiffness itself.

Torsional Analysis

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

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

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

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 vibrations problems, particularly if these problems are a result of resonance.

Experimental Modal Analysis, or "Bump" Testing

Experimental modal analysis 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 can be obtained, 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, 1986b). 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.

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, which 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 (1993), 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 database consisting of FRF plots of response vibrations at many locations due to hitting at a chosen location representative of where a significant exciting force might operate. The result of this sorting is accurate prediction of the frequency and damping of each natural frequency within the range of the test, and the ability to create moving "cartoons" of the vibration "mode shape." In some EMA computer programs, this information can also be used to automatically predict the best locations for added masses, dampers, or stiffeners to solve the vibration problem associated with a given mode.

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

GUIDELINES

Pump Sizing

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. This loads up the vanes, and may even lead to "airfoil stalling," with associated formation of strong vortices (miniature tornadoes) that can severely shake the entire rotor system, and can even lead to fatigue of impeller shrouds or diffuser plates or "strong-backs." The rotor impeller steady side-loads and shaking that occurs at flows below the onset of suction or discharge recirculation (Fraser, 1985) leads to the strong possibility of rubs and excessive rotor loads that can damage bearings. Many plants buy equipment that has more capacity than is needed, to allow for future production expansion, but in doing so ensure years of unreliable performance of potentially reliable machinery. Never run a pump for extended periods at flows below the "minimum continuous flow" provided by the manufacturer. Also, if this flow was specified prior to about 1985, it may be based only on avoidance of flashing and not on recirculation onset, and should be rechecked with the manufacturer.

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., flowrate), 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, at least approximately
- Whether the suction pressure is steady at a given operating point and well above NPSH requirements
- Whether unusual noises are present at certain operating conditions, and if so, what their main frequencies are, as picked up by a microphone and fed into the vibration analyzer

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

- Vibration 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

VFDs—Good for Energy, a Challenge to Reliability

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.

In terms of torsional response, discussed above, care should be exercised with systems involving variable frequency drives (VFDs). 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 controls' manufacturer can predict these frequencies and their associated torque oscillation strengths.

CASE HISTORIES

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

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

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

Reliability Problem Resolution by Careful Combination of Rotordynamic Analysis with Test

A Northeastern power plant had experienced chronic boiler feedpump failures for eight years, since the unit involved had been switched from base load to modulated load. The longest that the turbine-driven pump had been able to last between major rotor element overhauls was five months. The worst wear was seen to occur on the inboard side of the pump. The turbine was not being damaged. The pump OEM had decided on the basis of detailed vibration signature testing and subsequent hydraulic analysis that the internals of the pump were not well enough matched to part-load operation, and proposed replacement of the rotor element with a new custom-engineered design, at a very substantial cost. Although the problem showed some characteristics of a critical speed, both the OEM and the plant were sure that this could not be the problem, because a standard rotordynamic's analysis showed that the factor of safety between running speed and the predicted rotor critical speeds was over a factor of two. However, the financial risk associated with having "blind faith" in the hydraulics and rotordynamic analyses was considerable. In terms of OEM compensation for the design, and the plant maintenance personnel and operational costs associated with new design installation, the combined financial exposure of the OEM and the plant was about $350,000. Because of this exposure, one of the authors was called in for a "third party" opinion.

Impact vibration testing using the cumulative time averaging procedure referenced 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 showed that the particular rotor natural frequency value and rotor mode deflection shape could best be explained by improper operation of the driven-end bearing. The bearing was removed and thoroughly inspected, and was found to have a critical clearance far from the intended value, because of a drafting mistake on the bearing's drawing, which was carried over each time the bearing was repaired or replaced. Installation of the correctly constructed bearing resulted in the problem rotor critical speed shifting to close to its expected value, well out of the operating speed range. The pump has since run for years without need for overhaul.

Misalignment Caused by Nozzle Loading

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

The reason for the high vibration was found to be 35 mils of misalignment at the coupling due to the hydraulic loads on the pump discharge flange being far in excess of API 610 (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).

Additional Case Histories

Additional case histories, along with further details concerning the case histories listed above, are available on the Internet at www.mechsol.com.

CONCLUSIONS

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.

- 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 is 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)
FEM = Finite element method
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)
σ = Shaft bending stress (psi or N/mm)
ρ = Density (lbm/in or kg/mm)
ω = Vibrational frequency (radians/s)
ω_n = Shaft first bending natural frequency (radians/s)

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