PUMP VIBRATION ANALYSIS FOR THE AMATEUR

by

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INTRODUCTION

Maintenance is the art of prolonging or restoring the useful operating condition of a piece of equipment. Many engineers consider maintenance to be based on industrial mythology or folklore and as a result engineering principles are not always applied to maintenance. Vibration monitoring coupled with sound engineering judgement could be one of the most valuable tools of maintenance but its value is somewhat restricted at present. There are frequently two groups of personnel involved with large machinery; those who understand the machinery and those who understand the monitoring equipment. Frequently, there are differences of opinion. Resolving these differences is hard to accomplish, because the text books available on the subject of pump vibration are out of date. More effort must be made to update the information available and to improve our abilities to cope with the problems faced in trying to achieve economical pumping equipment runs. The problem of how to develop an understanding of what vibration monitoring data from large single stage or multistage pumps should mean to the average engineer, operator or maintenance person will be addressed. In short, it is a first aid manual on what to do for the "patient," the pump, until the doctor or vibration expert arrives.

CAUSES OF VIBRATION IN CENTRIFUGAL PUMPS

In every pump, dynamic forces of mechanical and hydraulic origin are present and a certain amount of vibration is therefore inevitable. To ensure the safety of the pump, the vibration must be kept within acceptable limits. If there is a high level of vibration, or if it increases markedly during the course of time, problems of a mechanical or hydraulic nature are indicated. The decision to remove a piece of pumping equipment from service for repairs should be based more on practical economic factors than on any arbitrary standards for vibration. Production losses, repair costs due to extensive damage, and personnel safety concerns must be weighed, and previous experience used to arrive at the final decision to shut down or to run the pump.

There are many potential causes of vibration. Some of the causes are shown on the vibration spectrum of Figure 1. The pump design and construction characteristics must be evaluated and the history of the pump's operational reliability considered, before positive conclusions can be drawn as to the cause of the vibration and safe operational levels of vibration. Detailed measurements and analyses of the vibration characteristics, the pump, and the pumping system are often needed for a reliable diagnosis.

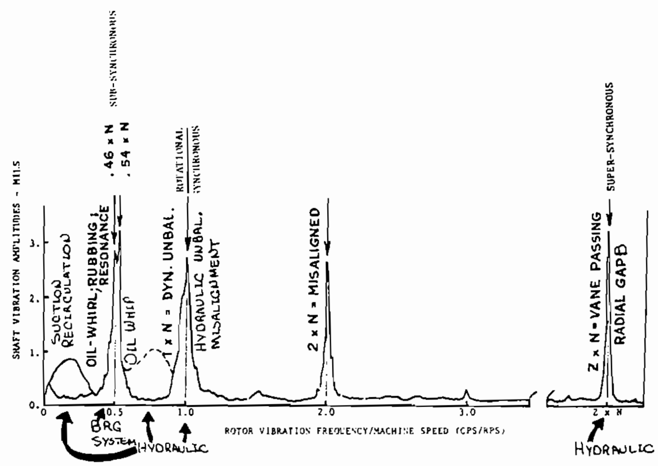

Figure 1. Frequency Spectrum of Pump Vibration Problems.

The first step in any analysis program is to draw a distinction between vibrations that are system-related and vibrations from a mechanical or hydraulic problem in the pump.

SYSTEM RELATED PROBLEMS

First, let's address vibration caused by a pumping system related problem. A typical pumping system can be divided into seven elements.

- the driver,
- mechanical transmission of power through either the coupling, and in some cases, a gear reducer, or increaser,
- the driven pump,
- the suction piping section and valves,

- the discharge piping and valves,
- the instrumentation for control of the pump flow, and
- the foundation.

Some system related vibration problems may be obvious, but many are interrelated with those vibrations emitting from the pump itself. Some typical system-related problems include the following:

- Unfavorable dynamic behavior of the foundation, supporting structures or piping (resonance excitation by forces at rotation speed).
- Excitations from the coupling area, especially due to misalignment of the driver or eccentrically bored coupling hubs.
- Excitations from the vibrations of the driver (motor, steam turbine, gearing).
- Unfavorable incoming flow conditions such as cavitation, intake vortex, or suction recirculation.
- Flow disturbances in the suction piping due to poor design and layout, especially in valves.
- High pressure pulsations due to hydraulic instability of the entire pumping system.
- Control system/pump interaction during startups or other periods of low flow.

Methodical investigation will help to identify vibration sources. The investigation into the causes of the vibration problem should include a look at all the possibilities starting at the most basic parts of the system.

PROBLEMS EXTERNAL TO THE PUMP

Foundation

Both dynamic and static forces must be considered for soil support of foundations. Well designed pump foundations and base should have the required rigidity to withstand the axial, transverse, and torsional loadings of rotating machines. Some conservative rules of thumb, as shown in Figure 2, that apply to good foundation design for API or larger pumps are:

- The mass of the concrete foundation should be five times the mass of the supported equipment.
- Imaginary lines extended downward 30 degrees to either side of a vertical line through the machinery shaft should pass through the BOTTOM of the foundation and not the sides.
- The foundation should be three inches wider than the baseplate all the way around the machinery up to 500 hp and six inches wider for larger machines [1].

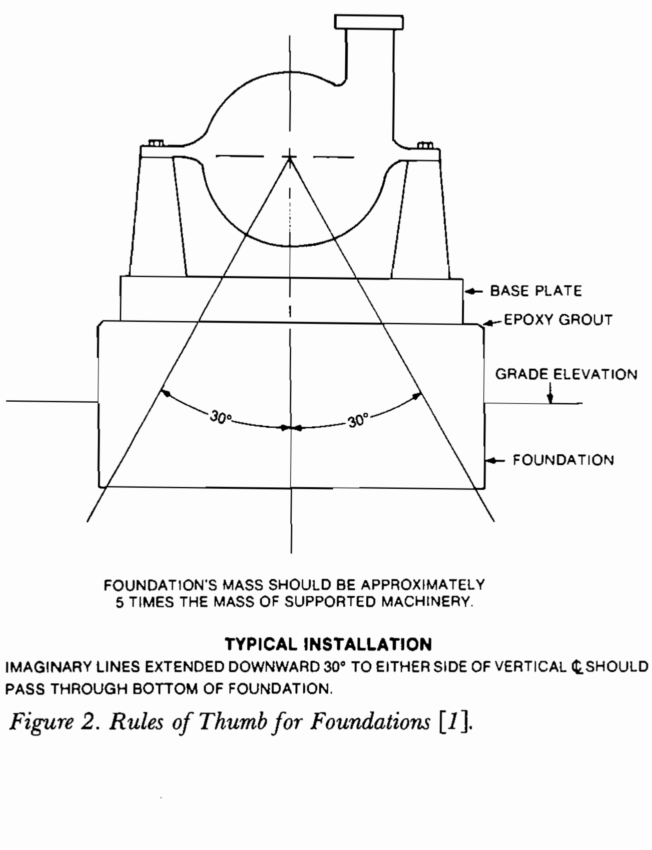

Figure 2. Rules of Thumb for Foundations [1].

These guidelines can be used to quickly evaluate the pump foundation as a source of vibration potential. Foundation vibrations are usually at rotational speed. If an inadequate foundation is identified as the source of a vibration problem, adding concrete mass may reduce the tendency of the foundation to respond to vibrations and transfer them. For example, if a concrete mass equal to the full weight of the machine is added to the base of the machine, the resulting velocity energy will be approximately half that of the original design. If twice the mass is added, the vibration energy will be cut to about 1/3 of the amount experienced on the original without the added mass. While often excellent for minimizing vibration, adding mass is a costly remedy after the foundation installation is completed [2].

On the other end of the spectrum, many ANSI style chemical process pumps in the range of 5 to 50 hp are frequently installed with no specially constructed foundations or foundations that do not meet the above guidelines.

Piping Design

Fluid flow disturbances in valves and piping create vibrational waves which travel freely in the piping, transfer to pipe hangers, and appear at the pump or at remote locations. Loose fitting valve gates or plugs, excessive throttling by use of either the discharge valve or a control valve, improperly sized piping and tortuous flow paths give rise to noise and vibration. Dynamic hydraulic forces may result from not understanding the total pumping system. As an example, in hot reboiler services of a petrochemical plant, the pump flow is frequently controlled from the tower bottoms temperature. The control valve, located between the pump and the reboiler, may cause considerable piping vibration to occur due to slug-flow pressure pulsations from the flashing in the reboiler. One possible correction is the relocation of the control valve between the column or tower and the reboiler. The same problems may arise in hot water pumping systems.

Sometimes piping fittings such as block valves, check valves, and strainers must be stacked vertically to accommodate pipe rack space limitations rather than to satisfy noise and vibration elimination concerns. Piping tees, valves, strainers and elbows may cause flow disturbances. Good practices that may be used to reduce these problems on the suction piping include:

- Valve stems and tee branches should be installed perpendicular to the pump shaft, not parallel to the shaft.
- Piping should have about five pipe diameters of straight run before the suction flange.
- To prevent resonant vibrations, long piping runs should be supported at unequal distances. The pipe wave can travel through the pipe hangers if they are at equal distances.
- Careful and generous use of pipe hangers to prevent the transmission of vibration through the structure.
- Secure anchoring and generous application of expansion joints and loops in the pipe [2]

While these are simple rules, they are frequently overlooked or ignored in piping design.

As it always does when dealing with machinery, the question arises as to how much vibration can be permitted in the piping system? One rule of thumb, that works very well, states that the permissible unfiltered velocity readings taken on the piping at the midspan of its supports can be three times the permissible readings taken on the bearing caps. Experience indicates that bearing cap readings in the range of 0.5 to 0.6 inches per second are the concern level for a pump and 1.0 in/sec is the emergency shutdown level. These pump vibration guidelines would then give 1.5 to 3.0 in/sec as the limit for piping. Maten has a very good discussion on the subject of piping vibration limits [3].

Piping Installation

Dynamic pipe reaction frequently imposes high vibration forces on pump casings. Heavy forces imposed by pipe will sufficiently distort casings and bearing housings to create vibration. Pump casings are designed for specific shaft and flange loadings. Frequently neither the pump vendor or any industry standards define very well the acceptable external forces and moments about the flanges. In the petrochemical industry, limits are outlined in API Specifications 610. These limits are somewhat vague, although the sixth edition (1981) and the soon to be released seventh edition (1987) have addressed this problem in greater detail. Even with the most accurate values of the forces and moments set by the standards, piping loads may be harmful to the pump unless proper engineering practices are followed. Provisions must be made to independently support piping and provide for thermal expansion of both the pump and its piping. This is especially important in hot service pumps. New pipe design on pumps operating above about 400°F should be carefully reviewed or audited by the owner or other independent party for correct input and desired results before construction. A good source for guidance in this area is Steiger [4].

Many mistakes have been made when "piping-up" to pumps, with some resulting in equipment destruction or a continuing reliability problems that are difficult to diagnose. The following general practices should be followed when "piping-up" pumps:

- Never connect piping to the pump prior to grouting.
- The piping system should be fabricated starting at the pump flanges, then work toward the pipe rack.
- Excessive loads to the pump flanges must be avoided during piping fabrication by using temporary braces and supports as required.
- Extreme care in the alignment of piping components must be used during initial fabrication to prevent problems later. Many times, a completed, tested, and to all appearances acceptable, piping system has required cutting, fitting, welding, and retesting, due to lack of caution during initial fabrication, with regard to alignment.
- Review the piping support design carefully. A good system design will support the piping loads and forces using spring hangers and bracing, etc., which do not require removal during normal maintenance.
- The design settings of all pipe hangers should be clearly indicated on the hanger nameplate, perferably stamped, so that time and the weather will not eradicate this important information.
- After all the fabrication and testing work is complete, loosen piping flanges. Release all lock-pins from spring hangers and remove any temporary bracing previously used. Adjust the system supports as necessary to free the pump from all piping strains.
- The piping fabrication error that can produce the largest piping strain is nonparallel flange faces. A feeler gage will help to detect this problem. Generally speaking, if a difference can be detected in the two facing planes, piping strain will exist.
- To check for piping strain, place dial indicators to monitor both vertical and horizontal movement of the pump shaft. Suction and discharge flanges should be made up separately, with indicator readings observed continuously. Should movement exceed two mils, piping strain is considered excessive and should be corrected. Loosen pipe until alignment is complete, then tighten and retest for strain.

Misalignment Between Components

A major cause of vibration is misalignment between the pump and its driver. Machines that normally operate at elevated temperatures must tolerate vibration during a temporary cold misalignment, until normal operating temperature is reached. Even though extreme caution is exercised in initially aligning shafts of coupled components, misalignment may still result from:

- loose foundation bolts,
- flexible baseplate designs,
- piping loads from dynamic, static or thermal causes,
- bearing deterioration from internal misalignment or lubrication problems,
- misconceptions about relative thermal growth between the driver and the pump,
- incorrect positioning of driver and pump such that distance between shaft ends (DBSE) exceeds the flexing limits of the coupling, and
- failure to use dowels to maintain the alignment once, it is established.

While flexible couplings will accommodate sizable misalignment without impairing the life of the coupling itself, that same amount of misalignment may cause damage to pump or driver bearings and mechanical seals.

Misalignment of the driver and pump usually results in a high axial vibration reading. The axial vibration may be as much as 1.5 times the vertical or horizontal readings. Vibration caused by misalignment generally occurs at the running speed of the pump, although it may also occur at multiples of the running speed. Vibration caused by misalignment can be distinguished from a resonance disturbance by monitoring the pump during a "coast down" period. Misalignment vibration will shift in frequency directly with the rotational speed. Shaft critical speeds or resonances will not shift.

Unbalanced or eccentrically bored couplings frequently cause vibration. Each coupling bolt assembly (bolt, nut, and washer) must have the same weight; and each coupling bolt hole must have a complete bolt assembly for balanced operation. Even missing set screws and improperly sized shaft keys can cause mechanical unbalance and consequent vibration. The coupling should not create axial thrust during operation even under conditions of shaft misalignment. A gear type coupling will transmit axial thrust imposed upon it while operating under a significant driving load due to frictional conditions between the two gear sets. The coefficient of friction increases as both torque and speed increase and the axial force actually transmitted through the coupling can be up to 1/3 lb/hp transmitted. The thrust load developed by this "slip-stick" situation can exceed the capabilities of the pump or its driver's thrust bearing. This high friction condition, called coupling "lockup," can cause vibration at either one or more likely two times running frequency depending on the extent of damage to the coupling. The greatest vibration is likely to transfer to the outboard end of the pump as the rotor is whipped about by the coupling using the inboard bearing as a fulcrum.

With a membrane or disc coupling, there is little likelihood of the vibration, caused by coupling misalignment or damage, being two times running frequency due to the axial "softness" of the coupling.

Gear Boxes

Gear problems can generate lots of vibration data but the number of sources of the problem make interpreting the data difficult. There are several frequencies that show up. One is gear mesh which is always present and is equal to the number of teeth multiplied by the rotational speed. Since the amplitude of the gear mesh changes sharply with load, amplitude alone is not a good indicator of the source of any problems. Manufacturing errors like excessive backlash on the gear teeth, eccentricities called "apex wobble," internal misalignment of the gears or a damage defect will create sidebands of vibration. These sidebands can be used to identify which gear is the damaged one. The shock impulse from a major gear defect can excite the natural frequencies of the gears in a set to further confuse the interpretation of the data. In general, the only way to analyze a gear box problem is to take abundant data on it and to trend any problems. Snap analyzes are very difficult to make.

Hydrodynamic problems can sometimes afflict high speed gears. The problem seems to be a combination of circumstances which makes the lube oil drain stop functioning with the result that foam or even solid oil touches the spinning gear wheels. If there is not ample room in the top part of the box, so that the oil has a chance to flow to the side walls and down relatively unrestricted, the top of the box floods. This causes excessive power loss, high temperatures, and vibration. Drain temperature increases as high as 100°F in less than a minute have been observed. This condition is called "choking." The vibration frequency of "choking" is wildly random and of little value in identification.

What triggers "choking" is not really known. There are seven gear design conditions that seem to be conducive to "choking."

- Low ratios
- Downward mesh
- Coarse pitch-large teeth profiles
- High pitch line velocities
- Gears fill a large portion of the box volume
- Small sized and/or poorly located drain lines
- Relatively high floor in the box

Since gear casing configuration and basic arrangement are not readily changed, remedies for "choking" usually take the form of modifying and augmenting the drains and vents. The bottom half of the box should be equipped with a baffle or pan so that the oil level and the drain is shielded from the windage directly hitting it.

VIBRATIONS RELATED TO THE PUMP

The preceding discussion has discussed how measured vibrations at the pump can actually be caused by system related vibration problems. Vibrations can also be produced by mechanical and fluid related problems within the pump itself. Typical vibration problems of the pump are:

- Unfavorable dynamic behavior of the rotor due to excessive wear ring, bushing, or seal clearances.
- Poor support of the rotor because of loose fits on the shaft or housing in the case of ball bearings. Excessive bore clearance or lack of "clamp" on the shell OD of sleeve bearings causes the same effects.
- Mechanical imbalance of the rotating parts due to poor balancing or careless assembly. Operational influences including cavitation erosion, deposits, corrosion, damaged impellers, galled parts, and abrasion can also cause imbalance.
- Increased radial hydraulic forces when the pump is operated outside of the design flow range. Some increase in vibration is normal when departing from the best efficiency flowrate.
- Mechanically defective bearings.
- Pump manufacturer casting and/or machining defects.

In a malfunctioning pump, there are lots of opportunities to collect massive amounts of data that can point to almost any defect that can be imagined. Monitoring systems have not received the level of acceptance and recognition they deserve because of a lack of operator support and enthusiasm, problems of detection—false alarms or no alarms, and the amount of data generated. The data compiled exceeds the operator's (and-most everyone else's) ability to cope with it. The end result is a general reluctance to accept monitoring as the total factor in maintenance of pumps. In spite of this negative atmosphere, a change in ideas is overdue, because of rapidly increasing maintenance costs. Logic and monitoring data must be tied together to answer questions if these costs are to be reduced.

Converting Vibration Data into Logical Thoughts

Associating the collected vibration data with the cause of a system and/or pump related problem is the point where most of us become confused and ineffective. Where do we start the analysis process? An analysis method used by one of the better known machinery troubleshooting experts, Sohre [5], is to relate the possible vibration causes and problems according to the ratio of the occuring frequency to the rotational speed, as shown in Figure 1.

SUBSYNCHRONOUS FREQUENCIES

Five Percent to Thirty-Five Percent Frequency Spectrum

Flow Disturbances

There is a low frequency axial vibration that occurs due to eddy current flows around the impeller from excessive impeller shroud to casing clearances, called Gap "A", and suction recirculation, both depicted in Figure 3. Flow disturbances related to suction recirculation and cavitation are always present in both diffuser and volute type pumps. These eddy currents are shown in Figure 3. As best efficiency point (BEP) operating conditions are approached, the frequency band width of the recirculation may narrow some what and sometimes the frequency will increase. Assuming that there is a sufficient margin between NPSH Required and NPSH Available, the noise or vibration associated with recirculation can be identified by stepping the operating flow toward the BEP flow. If recirculation is a cause, the vibration levels will decrease.

Remedies For Recirculation

Suction recirculation flows are made considerably worse by operation away from the best efficiency point or BEP. The

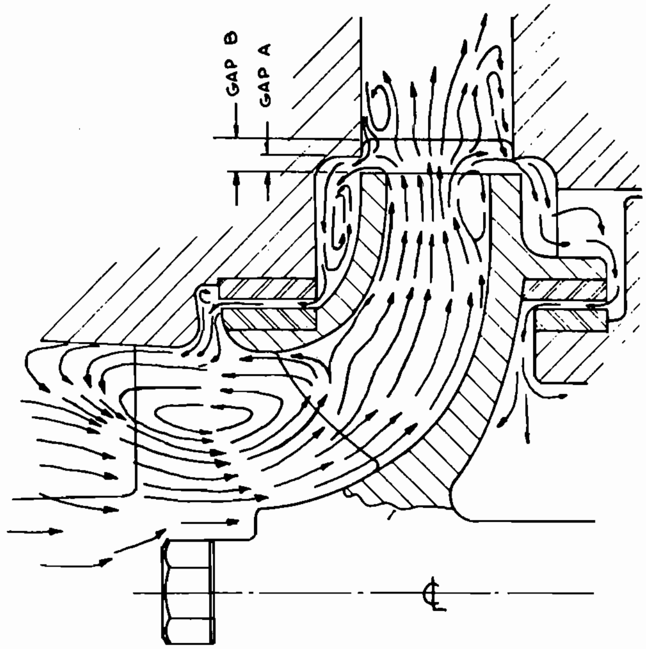

Figure 3. Eddy Flow Pattern at Suction Eye and Impeller Vane Tip at Off Design Flow Operations.

only real remedy is to operate the pump as far out on the curve near BEP as possible. There are four *Rules of Thumb about the recirculation phenomenon that should be remembered.*

• *As the energy level (measured by the head developed and the horsepower) increases, the recirculation driving force increases for operations away from the BEP.*

• *If the impeller diameter has been reduced by trimming, the flow distribution across the exit width of the impeller becomes more unstable. The tendency for the high pressure liquid to return to the low pressure side of the vane and create tip recirculation is greatly increased.*

• *The onset of recirculation can occur at almost any flow rate from near 100 percent down to about 40 percent of BEP depending on the pump design and other factors. Flows below about 50 percent of BEP should be avoided entirely.*

• *Suction recirculation, when it occurs, will cause strong AXIAL vibrations, especially with single stage and double suction horizontal pumps.*

Correction Of Gap "A"

The head developed by a centrifugal pump is frequently adjusted by trimming the impeller diameter. How the impeller is trimmed can influence the vibration levels experienced on the pump on account of problems associated with excessive Gap "A." No hard and fast guidelines for the mechanical aspects of impeller trimming exist, but there several pump construction and hydraulic design factors to consider while making the decision.

For most pumps, cutting the entire impeller, vanes and shrouds, as shown in Figure 4(a) will often increase the axial vibration and other problems associated with Gap "A" (shroud to case clearance), due to the uneven flow distribution at the impeller exit area. Trimming the vanes only tends to even out the exit flow pattern and reduce the recirculation tendencies at the exit area. Gap "A" should be about 50 mils (radial) for minimum axial vibration because of tip recirculation [10].

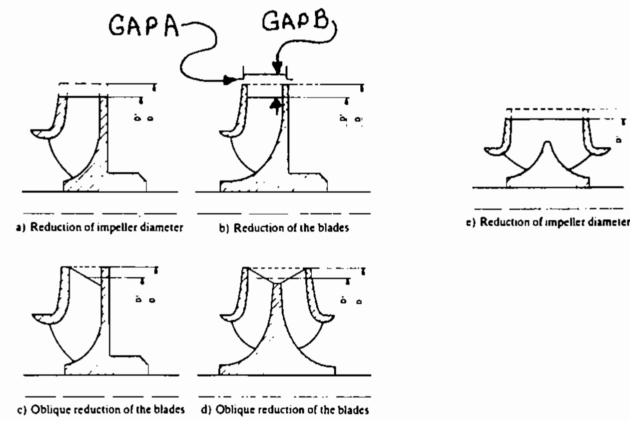

Figure 4. Impeller Diameter Reduction

In most cases, it is best to trim only the vanes as shown in Figure 4(b). The oblique cut in Figure 4(c) yields a more stable head curve because the tendency for tip recirculation and the possibility of suction recirculation being established is greatly reduced due to a more uniform flow distribution at the exit area. Structural strength of the remaining shrouds is a factor in the decision as to how to trim the impeller. There may be too much unsupported shroud left after a major reduction in diameter. It is often best to cut the impeller vanes obliquely, as shown in Figure 4(d), leaving the shrouds unchanged and solving the structural strength problem along with improving the exit flow pattern.

The double suction type pump is especially sensitive to problems caused by increased Gap "A," so trimming the entire impeller is not a good choice for this style pump.

Forty Percent to Fifty-five Percent Spectrum Frequency

Oil Whirl

A condition known as "oil whirl" may occur in lightly loaded sleeve bearings (under 90 to 100 psi). It is a self-sustaining type of rotary motion and can be explained by considering a wedge of oil traveling around the bearing at its average velocity or half the surface speed of the shaft. The flowing wedge of oil lifts and drives the shaft ahead of it in a forward circular motion within the bearing clearance. This motion is oil whirl. This oil film flow is due to fluid friction and has an average speed of less than one half of journal surface speed (42 percent to 47 percent), which is the characteristic frequency of this instability. This vibration may be caused by excessive bearing clearances or a low lube oil temperature. Several bearing conditions can produce oil whirl:

• light unit loading,

• excessive clearances,

• high surface speeds,

• oil properties (primarily high viscosity due to cold temperatures), and

• initial deflection of the rotor which generates a nonsymmetrical oil wedge along the axial length of the bearing.

Resonant or Critical Speed

All rotors have certain rational speeds at which the shaft vibrates at its natural frequencies. The natural frequencies are those frequencies at which the rotor would vibrate if struck by a heavy blow while stationary.

Many large multistage pumps operate above the first resonant "dry critical" speed and below the second. Low speed pumps typically operate below the first "dry" critical speed. Just barely enough is known about criticals to enable the building of machinery which will run. Design numbers out of a computer and what happens in the real life machine in the field are often something quite different. Some selected quotations from one of the best brief discussions on this subject from Gopalakrishnan [6], may be helpful in understanding this subject.

> In a rotating machine handling gas, the critical speed is basically determined by the mass and stiffness of the rotor and its bearings. In a centrifugal pump, the presence of liquid brings in several additional effects. For any centrifugal pump to function, it must have some close clearance areas. The number, location, and dimensions of these are dictated by the design philosophy. These areas include wear rings (sometimes called seal rings), throttle bushings, etc., each of which is, in effect, a running fit between a rotating sleeve within a stationary bushing. The effect of such fits is evidenced in several ways. First, the liquid in the bushing resists the radial displacement of the sleeve. The ratio of this resisting force to the displacement is called its stiffness. Secondly, the liquid also produces a restoring force proportional to the transverse velocity of the shaft movement. The ratio between this force and velocity is the damping coefficient of the bushings. Finally, the liquid also produces a restoring force proportional to the transverse acceleration of the sleeve. The ratio of this force to the acceleration is frequently termed the virtual or hydrodynamic mass of the bushing.

A pump rotor's "dry" critical speed will not remain constant for life as it may be influenced by such operating factors as the mass, stiffness, and damping characteristics of the rotor and its supporting structure. A small amount of damping will cause the critical to exhibit a sharp vibration peak over a narrow range of speed. A large amount of damping such as the leakage at balancing drum or wear rings can cause the critical to exhibit a broad peak over a wide range of speed.

The "wet" critical can be at almost any frequency above or below the running speed since it also changes with wear at the leakage points. Normally the critical is not seen at all due to suppression by the damping effect of the leakage points.

Some quotes from the literature may further clarify the problems [7].

> The forces occurring in the wear rings and any balancing drum have a considerable influence on the vibrational behavior of the rotor. The balancing drum has a great influence because of its large area and high pressure differential. The restoring force acting in opposition to rotor deflection has the effect of increasing the natural frequency, while the forces acting in the tangential direction can lead to instabilities in the form of self-excited vibrations.
>
> At rest, the rotor has a slightly lower natural frequency in liquid because the liquid has the effect of an additional mass. With increasing rotational speed, the pressure differential across the labyrinths and balancing piston increases, as a result of which the above mentioned restoring forces are built up and the natural frequency, which then depends on the speed, is raised.
>
> Vibrations excited by unbalance forces occur at the rotation frequency (synchronous vibrations) or at a higher harmonic frequencies, without the occurrence of subsynchronous vibrations. Subsynchronous vibrations only become noticeable in the case of self-excited vibrations. These can occur, for example, in boiler feed pumps with heavily worn wear rings, since with increasing wear ring clearance, not only the natural frequencies but also the rotor damping rates are reduced.

Oil Whip

If the cpm of an oil whirl condition is near a rotor resonance or critical that is below rotational speed, oil whirl excites and locks on to this resonance, producing oil whip. Oil whip is a forward circular orbit of the shaft at the resonance speed. Since it is a shaft vibration occurring at the natural frequency of the shaft when the speed of the shaft is two or more times the natural frequency, the speed can be less than or greater than 50 percent of running speed. This might be considered the stabilized version of the half-frequency whirl, the constant frequency being due to the persistence of the first natural critical of the rotor.

Severity of Oil Whirl or Whip

The shaft vibration resulting from oil whirl or whip can vary from completely acceptable levels to highly destructive levels. Since the whirl is in the oil film itself, the basic tendency is very destructive, because the capacity of the bearing is reduced to near zero. As a rule of thumb, oil whirl movement should not exceed 50 percent of the bearing clearance. At higher vibration levels, the machine must be shut down immediately to avoid serious damage.

Correction of Oil Whirl or Whip

There are several methods that can be used either individually or in combination to eliminate oil whirl or whip:

- Install new bearings with reduced clearances.
- Preload the bearing by a pressure dam or reducing the net area of the bearing, raising the unit loading above 150 psi.
- Raising the oil temperature. In rare cases, lowering the temperature can be effective in breaking up the oil whirl.
- Install tilting pad journal bearings which cannot whirl.

Three of these suggestions involve bearing redesigns. Numerous bearing designs have been developed to reduce the tendency for a shaft to whirl or whip in a bearing. These designs can be listed in the following order, from the worst to the best, of ability to prevent whirl or whip:

Design Type	*Anti-whip Characteristic*
Cylindrical	None
Elliptical	Slight
Pressure Pad	Fair
Multiple Groove	Fair
Tilting Pad	Excellent

The tilting pad bearing "fix" is the best one because it is not wear sensitive and it is more reliable. The bearing surface of this design is divided into three or more segments, each of which is then pivoted at the center. Each pad tilts in order to form a wedge shaped oil film, which tends to force the shaft toward the center of the bearing. This is the most effective anti-whip design.

Rubbing

Some partial internal rubs will cause "half frequency" vibrations because they excite a resonance frequency. Other rubs may produce synchronous or random multiples above running speed. Viewing the orbit of the vibrations on a scope may aid in determining if a rub is taking place.

Sixty Percent to Eighty Percent Frequency Spectrum

Hydraulic In Nature

This random frequency vibration is particularly found in multistage pumps. Its causes are hydraulic unbalance, the effects of leakage points such as wear rings, balancing drums or center stage bushings and impeller-volute interaction, when operating away from the design point. At flows below about 50 percent of BEP, these dynamic forces may be very strong with both diffuser and volute types of pumps. Remedies such as anti-swirl devices can reduce the effects of leakage points such as at the balancing drum bushing of multistage straight through flow pumps, but these are trial and error efforts. Total correction is difficult, since very little is known about proper volute design.

Mechanical In Nature

Subsynchronous vibration components between 0.5 and 0.75 times rotational frequency can occur in high speed multistage pumps because of excitation of a natural frequency by loose bearing housing fits and/or excessive bearing clearance. When these frequencies appear, the situation has to be taken very seriously; the rotor can be destroyed in a matter of minutes or sometimes in seconds. Both the bearing housing and bearing clearances should always be checked during the assembly of any pump.

One Hundred Percent Frequency Spectrum—Synchronous Speed

Apparent Unbalance

A once per revolution (synchronous) vibration is frequently cited as a sure sign of simple mechanical unbalance. The analysis is given that a short session on a balancing machine will correct all problems associated with the pump rotor. This premise is more often incorrect than correct. The once per revolution can also be a symptom of several other rotordynamics problems other than mechanical unbalance. These other problems resemble mechanical unbalance in nature but have entirely different origins. The resulting vibrational forces cause stresses in the rotor and its support structure. The vibrations from these forces can result in:

- excessive wear of bearings, bushings, seals, couplings, and gears.
- fatigue failure of rotating components such as shafts, impellers, wear rings coming loose and spinning.
- fatigue failure of stationary structures such as bearing supports, foundation bolts, and piping.
- transmission of vibrations to adjacent machinery and causing of additional problems.
- generation of noise.

The origins of the problems that produce synchronous vibrations are many and varied. Some are built into the pump while others are generated by poor operational or maintenance practices. The following discussion covers some of the more commonly encountered causes.

Mechanical Unbalances

The impellers of many centrifugal pumps are sand castings. Investment cast impellers are greatly preferred to sand castings. A sand cast impeller will have some degree of internal physical variations, of dimensions or of mass, which will appear as either (1) mechanical unbalance or (2) hydraulic unbalance. A good dynamic balancing can correct some of the problem but not all of it.

Hydraulic Unbalance

Hydraulic unbalance is the result of uneven flow patterns and volumes between vanes or liquid passages due to their uneven spacing, caused during the sand casting process by pattern wear, core shift, poor metal flow, and shrinkage. Good quality control at the foundry or the use of investment castings is the only correction for this condition. It is impossible to eliminate or even to reduce hydraulic unbalance by mechanical balancing of the impeller, since the cause is hydrodynamic and its action is unpredictable. Some pumps can exhibit synchronous response and some subsynchronous response because of hydraulic unbalance. The only remedy for the problem is to change the impeller to one of better quality with equal and uniform liquid channels.

Bowed Shafts

Bowed shafts are a common cause of unbalance. Every effort should be made to avoid conditions that help create shaft bows because they are difficult, sometimes impossible, to straighten and/or balance out. Here are a few causes of shaft bow:

- *Gravitational sag.* Large pump shafts with enlarged clearances which are allowed to sit in one position for a long period may develop bows. Upon startup, the shaft may vibrate violently.
- *Thermal distortion.* During the startup of hot oil pumps, a bow may develop in the rotor shaft because of heat distribution. The rotor must be slowly brought up to speed while uniformly heating it to avoid creating a large thermal bow. The casing may also be bowed due to heat distribution.
- *Thermal Expansion.* Thermal growth can cause vertical and horizontal misalignment that must be accommodated by the coupling. Internal misalignment can also result. The relative growth of a typical multistage hot oil pump of the barrel or double casing design is shown in Figure 5. The casing grows away from the support point while the rotor grows toward the support. Since the thrust bearing is attached to the casing, part of the rotor growth is canceled out.

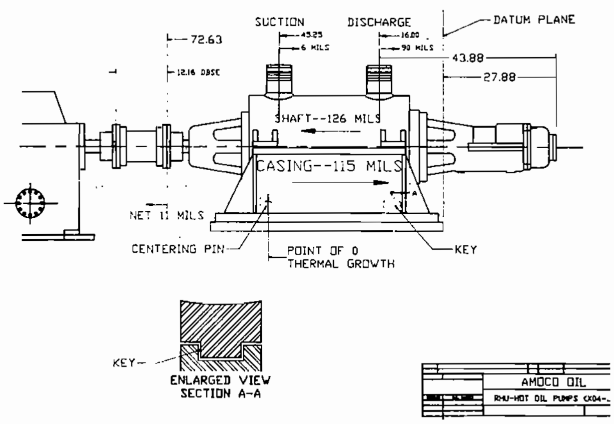

Figure 5. Thermal Expansion of Hot Pump.

Control of this growth is very important to both internal and external alignment. The heat expansion of pump parts can affect the mechanical performance of the pump in several ways.

- Usually the parts of the rotating element are of different materials than the pump case, and the running clearances may be reduced below a safe minimum due to their different rates of thermal expansion.
- Shaft sleeves, impeller rings, and case wearing rings, which are tightly fitted, may become loose when hot, because of a difference in expansion, especially during quick start-ups.

The rotating element may get out of alignment when the case is hot due to heat distortion of the case. Internal leakage further aggravates the thermal distortion, since uneven heating occurs along the leakage path. If the pump is put on stream cold and is brought up to temperature gradually over several hours, then there is sufficient time for all parts to reach the same temperature with little case distortion. However, if hot oil is charged to the pump suddenly, the various pump parts do not reach the same temperature at the same time. Usually, the upper part of the outer case reaches its highest temperature an hour or more before the bottom because the flow tends to be in the top of the pump near the inlet and discharge nozzles [8].

• In barrel or double wall type pumps, shut down leaves liquid trapped in two areas of the casing. One area is the inner liquid channels of the bundle and another area is the annulus between the bundle and the casing (barrel) that is sealed at the discharge end. Thermosyphon action occurs in each area. Field experience indicates a temperature of up to 175°F can develop. This temperature differential causes greater expansion of the casing top half, which pushes the bearings downward and the pump center upward, in relation to the center line, so that a half turn of the rotor can cause binding at leakage points such as wear rings and balancing drums. The amount of deformation depends of the location of piping nozzles including flush, cooling and balance line connections. Several hours must elapse before the pump is uniformly cool [9].

Putting the rotor on turning gear is a common practice to solve the problem of thermal sag of the rotor. Recommended turning gear speed is usually 35-60 cpm, which is enough to establish an oil film wedge in the bearings. Such low speeds are not enough to create circulation in the liquid channels of the pump and thereby aggravating the thermosyphon effects and causing more distortion [9].

• In dealing with pumps that will operate at temperatures greater than 500°F, extreme care must be taken to ensure that all auxiliary piping be welded to the case and the case rough machined prior to final heat treatment. This will prevent distortion due to stress relieving by temperature cycles in service.

Shaft Rubbing

Rubs or contacts of stationary parts like oil baffles, glands, seals, or bushings on the rotor will cause hot spots which can bow the shaft. Thermal bows in shafts can cause vibration at running speed or sometimes twice running speed. The bow tends to act like an unbalance condition giving the one times running speed characteristic. The tendency is toward two times running speed in very severe rubs. For every rotation of the shaft, a vertical impulse on the outboard and inboard end of each bearing will occur, thus there are two impulses.

Improperly Fitting Parts

Loose components may cause vibrations that shorten equipment life. One example is the unbalance imposed on a rotating assembly by the shift in mass center of loose rotor components. The design fit for a pump impeller cannot be a heavy fit as used in turbines or compressors because of assembly problems. Loosely fitting nonrotating and rotating parts generate vibrational waves that travel freely through a machine and its structure. To avoid these problems the following guidelines for rotor assembly are important:

• The impeller to shaft fit must be accurately controlled and be a slight interference fit. Many impeller castings are not properly stressed relieved, so bore dimensions may change while in hot service, or when they are heated for removal. Careless or excessive application of heat during installation or removal can cause distortion and loss of fit.

• The impellers must be precision balanced on a solid mandrel.

• The shaft, thrust collar, balancing drum and all other rotating parts must be precision balanced.

• All fits and runouts; impeller to shafts, keyways, etc., must be carefully checked.

• Stack the rotor (and diffusers if final assembly) in a vertical position.

• Heat the impellers as necessary.

• Quick cool the toe or eye area of the impeller with the flow of shop air from a ring about two inches larger than the shaft.

• Dial indicate the rotor on the shaft, wear ring and bushing areas for runout due to stacking deviations. Runout should be below two mils. If runout exceeds this level, look for skewed impellers or a heat bowed shaft.

• Check balance the entire rotor.

Loss of Balance

A progressively worsening mechanical unbalance may be caused by uneven wear or corrosion. Pump rotors are particularly susceptible to corrosion and erosion, major causes of mechanical unbalance. Water may corrode boiler feed pump impellers. Catalyst or ash fines may erode slurry pump impellers. Cavitation or suction recirculation flows may pit and erode any centrifugal pump impeller where marginal suction conditions exist.

Sleeve Bearings—Excessive Clearance

Excessive clearance or damage in sleeve bearings is normally one times running frequency. The traditional cylindrical bore bearing has some basic problems for use in the larger and higher speed pumps. To keep bearing temperatures at desired levels, larger clearances must be used. This runs counter to the need for keeping rotor vibration to acceptable values. Excessive clearance or damage can normally be determined by comparing the vibration level reading on the bearing housing, as close to the shaft as possible, with the vibration reading taken on the shaft itself. The pickup must be in the same plane and direction when taking both readings. Before bearings are replaced, a careful dimensional check should be made of the bearing bore and the shaft journal surface to spot any bore size change, out of roundness, or shaft wear.

Sleeve Bearings—Mechanical Looseness

Looseness of bearing caps or brackets usually results in one times running speed vibration, a large number of harmonics and sometimes partial frequencies in the vibration spectrum. Vibration levels are generally higher in the vertical plane than in the horizontal plane. Bearing housing bores and a bearing shell or liner diameters are sized to impose an exact "crush" or holding force. Check the bearing shell outside diameter to the housing bore to be sure there is sufficient "crush" to hold the bearing. This clearance should be checked using "plastigage" strips between the top of the shell and the bearing housing while five mil shims are placed on both sides of the split line of the housing. The squeeze of the "plastigage" should indicate the clearance. The "crush" should be at least metal to metal, with one or two mils of interference preferred.

Ball Bearings—Looseness

Unfortunately, many pump builders do not indicate the proper bearing fits for shaft and housings to guide shop repairs. The original dimensions of both the housing and the shaft will change from time to time from oxidation, fretting, damage from locked bearings and other causes. Every bearing handbook has tables to aid in selecting fits. The vibrational effect of looseness on the bearing fits is different for the housing and the shaft.

Housing Fits

Ball bearing fits in the bearing housing are of a necessity slightly loose for assembly. If this looseness becomes excessive, vibration at rotational speed and multiple frequencies will result. Obtaining a loose fit of the antifriction bearing outer race when mounted in the bearing housing is done by measuring the bearing OD to verify compliance with required dimensions. Do not install bearings with ODs outside of the given tolerance band since this might result in either excessive or inadequate outer race looseness.

Rules of Thumb:

- Bearing OD to housing clearance—about 3/4 mil loose with 1-1/2 mils max.
- Bearing housing out of round tolerance is one mil max.
- Bearing housing shoulder tolerance for a thrust bearing is 0 to 1/2 mils per in of diameter off square up to a maximum of two mils.

Shaft Fit

A loose fit of the shaft to the bearing bore will give the effect of an eccentric shaft, at a one times running frequency vibration pattern. The objective of the shaft fit is to obtain a tight fit of the antifriction bearing inner race, when mounted on the shaft. The bearing bore should be measured to verify inner race bore dimensions. Do not install bearings with an ID outside of the given tolerance band since this might result in either excessive or inadequate shaft tightness.

Rules of Thumb:

- Fit of bearing inner race bore to shaft is 1/2 mil tight for small sizes—3/4 mil tight for large sizes.
- Shaft shoulder tolerance for a thrust bearing is 0 to 1/2 mil per inch of diameter off square up to a maximum of 1 mil.

Soft Foot

An improperly shimmed foot on a pump or its driver can cause vibration. As any residual unbalance of the rotor passes over the "soft foot," the support structure deflects, giving a characteristic of one-time running frequency.

Pipe Strain

Piping strain has already been discussed as part of the pumping system. It generally has the effect of creating misalignment.

SUPER-SYNCHRONOUS

Vane Passing Frequency

Vane passing frequency vibration is a hydraulically induced vibration at a frequency determined by the number of impeller vanes, the number of stationary vanes and the pump cpm. Detection of this problem sounds easy but can be confusing. Vane passing frequency is the number of impeller vanes times cpm for a double volute pump, IF the number of impeller vanes is an even number. IF the number of impeller vanes is an odd number, it is the number of impeller vanes times cpm times two for the same pump. IF it is a diffuser pump, the vane passing frequency is the number of impeller vanes times the number of diffuser vanes times cpm. A quick reference for this information may be viewed in Table 1.

Table 1. Vane Passing Frequencies

No. Impeller Vanes	Volute Style (Double)	Diffuser Style
Even	No. vanes × CPM	No. impeller vanes × diffuser vanes × CPM
Odd	No. vanes × CPM × 2	No. impeller vanes × diffuser vanes × CPM

Remedies for Vane Passing Vibrations

Correction of Gap "B"

Gap "B," as shown in Figure 2, is a major factor in the magnitude of the vibration level from vane passing frequency. Careful machining of the volute or diffuser tips to increase gap "B" while maintaining gap "A" has been used for over fifty years to reduce the vibration levels of vane passing frequency. The pulsating hydraulic forces acting on the impeller can be reduced 80 percent to 85 percent, by increasing the radial gap from about one percent to six percent. Definitions and recommended dimensions for the radial gaps are presented in Table 2. Locations of Gap "A" and Gap "B" are shown in Figure 3. Many pump handbooks state that shortening the diffuser or volute channels will reduce efficiency. Experience has shown that there is little or no loss of overall pump efficiency when the diffuser or volute inlet tips are recessed. The maintenance of efficiency in actual pumps probably results from the reduction of various energy-consuming phenomena: the high noise level, shock, and vibration caused by vane passing frequency, and the stall generated at the diffuser or volute inlet [10].

Random Positioning of Impellers

In the diffuser style pump, a complete set of hydraulics can be created specifically for each pump application because the diffusers are cast separately from the case and can be readily changed. In the volute design, the volutes themselves can be relocated only by a very expensive pattern change. In multistage volute pumps the degree of positioning of the volutes is severely limited by case design, so it is necessary to randomly cut the keyways in the impellers to

Table 2. Recommended Radial Gaps for Pumps.

Type	Gap "A"	Gap "B"* Percentage of Impeller Radius		
		Minimum	Preferred	Maximum
Diffuser	50 mils	4%	6%	12%
Volute	50 mils	6%	10%	12%

$*B = \underline{100}\ \frac{(R3-R2)}{R2}$ R3 = Radius of diffuser or volute inlet, R2 = Radius of impeller

NOTE: If the number of impeller vanes and the number of diffuser/volute vanes are both even, the radial gap must be larger by about 4%.

assure that vanes on adjacent impellers are not aligned. Frequently, this random selection of keyway positioning is not done when manufacturing the impellers and the vanes on the impellers line up, producing a high vane passing frequency vibration. The alignment of impellers and volutes in each stage should be carefully observed during witness testing or reassembly on a new pump, or when replacing the impellers during maintenance.

Correction of Vane Shape

Impellers manufactured with blunt vane tips can also cause trouble by generating hydraulic disturbances in the volute and impeller exit area even when the impeller is the correct distance, "Gap B," from the cutwater. This disturbance may be partly or entirely eliminated by tapering the vanes by "overfiling," removal of metal on the LEADING face of the vane as shown in Figure 6 and Table 3. The same improvement can be achieved by sharpening the impeller blades on the underside or TRAILING edge of the vane as shown in Figure 7. This method enlarges the outlet area of the liquid channel and can result in up to five percent more head near the best efficiency point, depending on the outlet vane angle. At least 1/8 in of tip thickness must be left as shown in Figure 7.

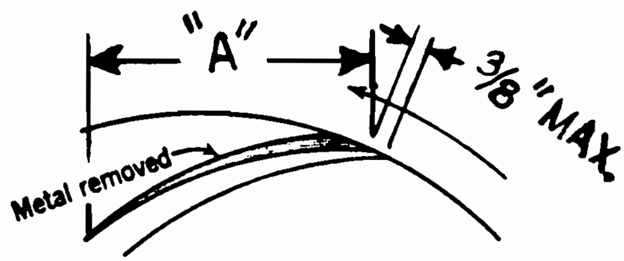

Figure 6. Impeller Vane Overfiling.

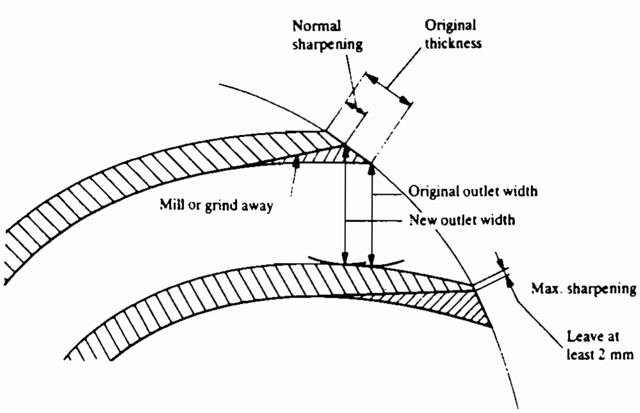

Figure 7. Sharpening of Impeller Vanes [7].

Antifriction Bearing Defects

Antifriction bearing defects are difficult to detect in the early stages of a failure, because the resulting vibration is very low and the frequency is very high. If monitoring is performed with simple instrumentation, these low levels will not be detected and unexpected failures will occur. The vibration frequencies transmit well to the bearing housing, because the bearings are stiff. They are best measured with accelerometers or shock pulse meters. Spencer and Hanson [11] have a good discussion of the reliability of four different monitoring systems for rolling element bearings.

Table 3. Length of Blend For Over Filing.

Impeller Diameter	"A" Distance of Blend
10" & Below	1-½"
10-1/16 through 15"	2-½"
15-1/16 through 20"	3-½"
20-1/16 through 30"	5"
30" & Larger	6"

There are some guidelines that can be established to evaluate bearing deterioration. For example, a ball passes over defects on the inner race more often than those on the outer, because the linear distance around the diameter is shorter. A very good discussion on early detection of the bearing distress frequencies is contained in the literature [12]. One word of caution on the use of the formulas in the reference is that parameters for the same size bearing will change with the manufacturer, so conflicting data can occur. Also, considerable dimensional data on the bearings is required to utilize the formulas.

Without going into the formulas [12], there are four dimensions of a ball bearing that can be used to establish some feel for the condition of that bearing:

- A defect on outer race (ball pass frequency outer) occurs at about 40 percent of the number of balls times running speed.
- A defect on inner race (ball pass frequency inner) causes a frequency of about 60 percent of the number of balls multiplied by running speed.
- Ball defects (ball spin frequency) are variable with lubrication, temperature and other factors.
- Fundamental train frequency (cage defect) occurs at lower than running speed values.

A simple check for verification of poor bearing condition is made by shutting off the pump and observing that the high bearing frequency remains as the pump speed reduces. This high frequency signal will normally remain until the pump stops. The frequency indication is normally from five to 50 times the running speed of the machine.

CONCLUSIONS

Vibration analysis that limits itself to collecting and "trending" data is not economical. The data must be interpreted as to a base cause and more importantly, corrections developed. If the corrections are not developed, a major portion of the economic incentives of monitoring systems are lost.

Vibration in pumps generally will come from the sources discussed briefly herein. Most of these sources are best corrected before they occur. Good sound engineering practices applied to the sizing and selection of equipment is vital. Proper planning, design and construction installation is a major factor in successful operational startup. Proper maintenance will keep the pump operating successfully.

If corrections are necessary during operation, a systematic approach with vibration analysis instrumentation and a good grasp of the construction peculiarities of the pump can identify vibration disturbances. After identification, the causes can be either minimized or corrected so that trouble free operation can be achieved.

REFERENCES

1. Dodd, V. R, and East, J. R., ASME Pump Engineering Seminar, South Texas Section, Houston, Texas (1979).

2. Meyerson, N. L., "Vibration: Its Causes, Identification and Cures," Power and Fluids, Northington Corporation (Date unavailable).
3. Maten, S., "Field Criteria for Pipe Vibration," Hydrocarbon Processsing (July 1984).
4. Steiger, J. E., "API 610 Baseplate And Nozzle Loading Criteria," *Proceedings of the Third International Pump Symposium*, Turbomachinery Laboratories, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1986).
5. Sohre, J. S., "Turbomachinery Problems and Their Correction," *Sawyer's Turbomachinery Handbook, II*, Ch. 7 (1980).
6. Gopalakrishnan, S., and Husmann, J., "Some Observations on Feed Pump Vibrations," EPRI Conference, Cherry Hill, New Jersey (1982).
7. *Sulzer Centrifugal Pump Handbook*, Sulzer Brothers Limited, 1986.
8. Karassik, I. J., Centrifugal Pump Clinic, Marcel Decker, Inc. (1981).
9. O'Keefe, W., "How to Avoid Transient-Caused Problems In Boiler Feed Pumps," Power, (June 1986).
10. Makay, E., and Diaz-Tous, I. A., "Feed Pump Reliability and Efficiency Improvements Resulting from Hardware Modifications," EPRI Conference, Cherry Hill, New Jersey (1982).
11. Spencer, D. B., Hansen, J. S., "A Better Way To Monitor Bearings," Hydrocarbon Processing (January 1985).
12. "Effective Machinery Maintenance Using Vibration Analysis," Dynamic Signal Analyzer Applications, Apllication Note 243-1, Hewlett Packard (1983).

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